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STRUCTURAL OPTIMIZATION FOR CONVERSION OF ALUMINUM CAR FERRY TO SUPPORT MILITARY VEHICLE PAYLOAD

SHIP STRUCTURE COMMITTEE

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Ship
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Committee

Address Correspondence to:

Executive Director
Ship Structure Committee
U.S. Coast Guard (G-MSE/SSC)
2100 Second Street, SW
Washington, D.C. 20593-0001
Web site: http://www.shipstructure.org

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STRUCTURAL OPTIMIZATION FOR CONVERSION OF ALUMINUM CAR FERRY TO SUPPORT MILITARY VEHICLE PAYLOAD

The objective of this project was to assess the conversion of an existing large aluminum car ferry to transport military vehicles in unrestricted, open-ocean service. The report analyzes the structural modifications necessary for the application of the ABS High Speed Naval Craft criteria and the DNV High Speed, Light Craft and Naval Surface Craft criteria to a passenger ferry design. The British Columbia PacifiCat ferry was chosen as the conversion vessel for this project.

In order to satisfy the conversion requirements, it was necessary that the converted vessel be able to operate in the open-ocean, unrestricted environment. The original design of the PacifiCat ferries was developed in accordance with DNV Inshore (R4) service area restrictions, which are significantly different from the open-ocean criteria needed for military purposes.

The results of the study indicate that neither ABS nor DNV would require any structural modifications to accommodate global hull girder loads or vehicle deck loads resulting from the assumed payload. The results also show that structural modifications would be necessary to accommodate slam loads (2 metric tonnes for ABS and 30 metric tonnes for DNV). Cost estimates are provided for the structural modifications resulting from application of the rules.

T. H. Gilmour
T. H. GILMOUR
Rear Admiral, U.S. Coast Guard
Chairman, Ship Structure Committee

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16 Abstract

This project went through the typical procedures associated with developing the structural requirements/impacts for a Sealift conversion vessel: 1) Identify the payload requirements and a representative military vehicle payload for the Sealift vessel, 2) Determine the candidate vessels available for the conversion and make a final choice for the conversion study, 3) Develop a preliminary arrangement of the vehicle loadout, i.e., arrange the payload to ensure the selected ship can accommodate the vehicles, 4) Determine the structural loads, global, local and vehicle related, to accommodate the conversion, 5) Determine the structural modifications required to accommodate the new loads and 6) Develop the cost estimates to accommodate the structural modifications, including the addition of vehicle tie downs. The essence of this project was comparison of the structural requirements and resulting modifications from application of the ABS High Speed Naval Craft criteria and the DNV High Speed, Light Craft and Naval Surface Craft criteria. Comprehensive comparison of the global loads, secondary slam loads and vehicle deck loads is developed in the report along with the structural requirements to resist these loads. The report also presents a summary of the finite element analysis work that was developed to investigate the impact of the military payload on the vehicle deck structure. This was of particular interest because the nominal tire footprints for many vehicles are significantly greater than the stiffener spacing, which violates typical assumptions for the design of plate structure subjected to wheel loads.

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CONVERSION FACTORS
(Approximate conversions to metric measures)

To convert from	to	Function	Value
LENGTH			
inches	meters	divide	39.3701
inches	millimeters	multiply by	25.4000
feet	meters	divide by	3.2808
VOLUME			
cubic feet	cubic meters	divide by	35.3149
cubic inches	cubic meters	divide by	61,024
SECTION MODULUS			
inches^2 feet	centimeters^2 meters	multiply by	1.9665
inches^2 feet	centimeters^3	multiply by	196.6448
inches^3	centimeters^3	multiply by	16.3871
MOMENT OF INERTIA			
inches^2 feet^2	centimeters^2 meters	divide by	1.6684
inches^2 feet^2	centimeters^4	multiply by	5993.73
inches^4	centimeters^4	multiply by	41.623
FORCE OR MASS			
long tons	tonne	multiply by	1.0160
long tons	kilograms	multiply by	1016.047
pounds	tonnes	divide by	2204.62
pounds	kilograms	divide by	2.2046
pounds	Newtons	multiply by	4.4482
PRESSURE OR STRESS			
pounds/inch^2	Newtons/meter^2 (Pascals)	multiply by	6894.757
kilo pounds/inch^2	mega Newtons/meter^2 (mega Pascals)	multiply by	6.8947
BENDING OR TORQUE			
foot tons	meter tonnes	divide by	3.2291
foot pounds	kilogram meters	divide by	7.23285
foot pounds	Newton meters	multiply by	1.35582
ENERGY			
foot pounds	Joules	multiply by	1.355826
STRESS INTENSITY			
kilo pound/inch^2 in$\sqrt{}$in)	mega Newton MNm$^{3/2}$	multiply by	1.0998
J-INTEGRAL			
kilo pound/inch	Joules/mm^2	multiply by	0.1753
kilo pound/inch	kilo Joules/m^2	multiply by	175.3

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LIST OF ABBREVIATIONS AND SYMBOLS

This document presents two List Of Symbols pages, one for ABS and one for DNV.

LIST OF SYMBOLS – ABS High Speed Naval Craft Rules

B	Beam of Vessel, meters
B_{cl}	Distance between hull centerlines in a catamaran, meters
B_w	Maximum waterline beam, meters
B_{wl}	Waterline breadth, meters
C_b	Block coefficient
C_1, C_2 & C_3	ABS Design coefficients
F_D	ABS design area factor
F_I	ABS Wet Deck pressure distribution factor
f_p	ABS Design Stress
h_a	Vertical distance from lightest draft waterline to underside of Wet Deck, meters
$h_{1/3}$	Significant wave height, meters
L	Length of Vessel, meters
l_s	ABS length of slam load
K_V	ABS Vertical Acceleration distribution factor
K_1, K_2	ABS Design coefficients
k_1 & k_2	ABS Design coefficients
M_{sl}	ABS Slam Induced Bending Moment
M_{swh}	ABS Still Water Hogging Moment
M_{sws}	ABS Still Water Sagging Moment
M_{th}	ABS Transverse Bending Moment
M_{tt}	ABS Torsional Moment
M_{wh}	ABS Wave Induced Hogging Moment
M_{ws}	ABS Wave Induced Sagging Moment
N_h	Number of hulls in a multi-hulled vessel
N_1, N_2, N_3	ABS design coefficients
n_{cg}	Vertical Acceleration at the center of gravity of the vessel
n_{xx}	Vertical acceleration at section x-x of the vessel
p_{bxx}	Bottom design slamming pressure at any section clear of LCG

p_{sxx}	Side and transom slamming pressure
V	Speed of Vessel, knots
V_I	Relative impact velocity
W	ABS static wheel load
β	Coefficient based on aspect ratio of deck plating panel
β_{bx}	Deadrise angle at any section clear of the LCG, degrees
β_{cg}	Deadrise at Longitudinal Center of Gravity, degrees
Δ	Displacement of Vessel, metric tones, kilograms
σ_a	ABS allowable stress for design of deck plate subjected to wheel loads
σ_y	Yield stress of material
τ	Running trim at V, degrees

LIST OF SYMBOLS – DNV High Speed, Light Craft & Naval Vessel Rules

A	Design area for element to be designed for slamming calculations
A_R	Reference area for slamming calculations, crest/hollow landing calculations
a	Tire footprint dimension parallel to deck stiffening
a_{cg}	DNV Vertical design acceleration at craft's center of gravity, m/s^2
a_v	Vertical design acceleration for wheel load calculations
a_0	DNV Acceleration parameter
B	Beam of Vessel, meters
B_{MAX}	Maximum width of submerged portion of hulls – sum of both hulls for twin hull craft
B_{tn}	Breadth of cross structure, meters
B_{WL}	Maximum width of waterline – sum of both hulls for twin hull craft
b	Transverse distance between the centerline of the two hulls in twin hull craft. Also, Tire footprint dimension perpendicular to deck stiffening
b_s	Breadth of slamming reference area
C_B	Block coefficient

C_H	Correction factor for height above waterline, slam pressure calculation
C_L	Correction factor for length of craft, slam pressure calculation
C_W	Wave coefficient
c	Coefficient used in design of deck structure subjected to wheel loads
d	Coefficient used in design of deck stiffening subjected to wheel loads
e_r	Mean distance from the center of $A_R/2$ end areas to the vessels LCG
e_w	One Half of the distance form the LCG of the fore half body to the LCG of the aft half body = 0.25L if not known (0.2L for hollow landing)
F_y	Horizontal split force on immersed hull, transverse bending calculation
f_g	DNV Acceleration factor
f_l	Allowable stress reduction factor
g_0	Acceleration due to gravity, 9.81 m/s^2
H_C	Minimum vertical distance from waterline to load point in questions, wet deck slam pressure calculation
H_L	Necessary vertical clearance from waterline to load point to avoid slamming, wet deck slam pressure calculation
H_S	Significant wave height
$H_{S\,MAX}$	Maximum significant wave height in which the vessel is allowed to operate
H_1	Design coefficient for vertical bending moment calculation
k	Design coefficient for crest/hollow landing calculations
k_a, k_b	Coefficients for bottom slamming pressure calculation
k_a	Coefficient is also used to determine vehicle deck plate thickness requirement
k_c	Hull type clearance factor, wet deck slamming pressure calculation
k_l	Longitudinal distribution factor for slam pressure calculations
k_t	Longitudinal pressure distribution factor, wet deck slamming pressure calculation
k_w	Coefficient for vehicle deck plate thickness
k_1, k_2, k_3	DNV design coefficients

k_z	Coefficient used for deck stiffening design subjected to wheel loads
L	Length of Vessel, meters
L_{BMAX}	Length where $B_{MAX}/B_{WL} > 1$
l	Length of stiffener subjected to wheel load
l_s	Longitudinal extent of slamming reference area
M_B	DNV Crest and Hollow Landing hull girder bending moments
M_p	DNV Hull girder Pitch Connecting moment
M_s	DNV Transverse hull girder bending moment
M_{so}	DNV Transverse still water bending moment
M_{sw}	DNV Still water bending moment
M_t	DNV Torsional hull girder moment
$M_{tot\ hog}$	DNV Total longitudinal hogging moment acting on the hull girder
$M_{tot\ sag}$	DNV Total longitudinal sagging moment acting on the hull girder
m	Coefficient used for deck stiffening design subjected to wheel loads
n	Number of hulls in slamming calculation
n_0	Number of tires on an axle for wheel load calculations
p	Design pressure in way of tire footprint loads
p_{sl}	DNV variable used to describe various slamming pressures; bottom, Wet Deck, forebody side and bow
Q	Maximum axle load
RX	DNV Service Restriction Notation where X can vary from 0 to 6
r	Rigidity factor for design of deck stiffening subjected to wheel loads
s	Stiffener spacing
T	Draft of vessel
T_L	Draft at lowest service speed
T_0	Draft at L/2 at normal operating condition and speed
V	Speed of Vessel, knots
x	Distance from AP to position considered
Z	Section Modulus required for deck stiffening in way of vehicle footprint loads
z	Height from baseline to wet deck

α	Flare angle
β	Coefficient based on aspect ratio of deck plating panel
β_x	Deadrise angle at section under consideration
β_{cg}	Deadrise angle at LCG
Δ	Displacement of Vessel, metric tonnes, kilograms
γ	Angle between the waterline and a longitudinal line at point considered, slam calculation
ρ	Liquid density
σ	Reduced allowable stress after consideration of function of structural element being designed
σ_0	Allowable stress before application of reduction factor
τ	Running trim at V, degrees

EXECUTIVE SUMMARY

This report provides a comparison of the structural modifications resulting from the application of the ABS High Speed Naval Craft criteria and the DNV High Speed, Light Craft and Naval Surface Craft criteria to a ferry conversion. The British Columbia PacifiCat ferry was chosen as the conversion vessel to accommodate the military vehicle payload selected for the project, a portion of the USMC Marine Expeditionary Unit, MEU. The US Army Stryker Brigade Combat Team was also considered as a candidate loadout for the conversion.

In order to satisfy the conversion requirements it was necessary that the converted vessel be able to operate in the open-ocean, unrestricted environment. The original design of the PacifiCat ferries was developed in accordance with the criteria of the DNV **R4** service area restrictions. This is defined by DNV as an *Inshore* condition, which requires that a ship be no more than 5 nautical miles from safe harbor during *Winter* operations, 10 nautical miles during *Summer* operations and 20 nautical miles in a *Tropical* environment, significantly different from the open-ocean criteria required for the converted vessel.

The results of the study indicate that neither ABS nor DNV would require any structural modifications to accommodate global hull girder loads or vehicle deck loads resulting from the MEU. Application of the ABS rules would result in just over 2 metric tonnes of structural modifications to accommodate secondary slam loads whereas the DNV rules require an additional 30 metric tonnes of structure to accommodate the slam loads.

The study also includes the required modifications to the vehicle deck to accommodate vehicle tiedowns, which are considered separate from the loading criteria mentioned above. There are no differences between ABS and DNV accommodating the tiedowns but their procurement and installation costs are included as part of the estimate for both ABS and DNV.

Cost estimates provided at the end of the report compare the structural modifications for ABS and DNV. The cost estimates include the vehicle tie downs and port facility costs as well as structural modifications resulting from application of the rules.

1.0 INTRODUCTION

This project was performed under Solicitation Number DTMA1R03004 for the Department of Transportation/Maritime Administration.

1.1 OBJECTIVES

As stated in the Solicitation, there were three Objectives to be served by this project:

1. Develop the detail design of an existing large aluminum car ferry to increase its structure to handle military vehicles for unrestricted, open-ocean service.
2. Demonstrate the required changes and impacts to ship structure to accommodate the military payload.
3. Allow commercial designers to consider this aspect of a functional design in their studies to minimize conversion requirements and allow possible USN service to augment high-speed Sealift capability.

This report directly addresses the first two objectives and provides valuable insight for commercial designers to consider the inclusion of military vehicle payloads and unrestricted operation in their designs, thereby addressing the third objective.

1.2 BASIS FOR THE CONVERSIONS

As required by the Solicitation, the conversion designs were done in accordance with two sets of rules. These are:

1. ABS Rules for Building and Classing High Speed Naval Craft, 2003 [1]
2. DNV Rules for Classification of High Speed, Light Craft and Naval Surface Craft, July 2000 [2].

This report presents full discussion on the primary and secondary structural loads required for the conversion to satisfy each of these classification society rules for unrestricted, open-ocean operation. The primary loads include all typical hull girder loadings associated with a catamaran, although ABS and DNV each include primary hull girder loads unique to their own set of rules. The secondary loads include wave slamming and impacts to the vehicle deck structure due to the new military vehicle loadout and the vessel accelerations in the unrestricted environment.

A brief paragraph is provided below that outlines the sections of this report. It is organized in the same order as the Interim Deliverables that were required for the development of the project.

1.3 SEALIFT MISSION REQUIREMENTS SELECTED FOR THE CONVERSION VESSEL (SECTION 2)

Section 2 defines the Sealift mission requirements to be used as the criteria for selecting the vessel to be converted. The mission requirements include a realistic military vehicle loadout and definitions for the range, speed and endurance of the vessel to be converted.

The military vehicle loadout considered both the US Army and US Marine Corps who each have significant efforts associated with their force deployment strategies and the logistics required to support these forces. The U.S. Army's Stryker Brigade Combat Team (SBCT) and the U.S. Marine Corps Marine Air Ground Task Force (MAGTF) or Marine Expeditionary Unit (MEU) were both considered as payload candidates for this project. The MEU was the final choice and the actual payload was selected from the MEU loadout defined for the LHA(R) program.

1.4 VESSEL SELECTED FOR THE CONVERSION STUDY (SECTION 3)

Section 3 identifies the likely candidates for conversion and selects the final vessel to be used for the conversion study.

One of the surprises for this project is reflected in the final choice for the vessel for the conversion study, the British Columbia Ferry Corporation's *PacifiCat*. Although these are sound craft, they are classed in accordance with DNV with an **R4** service restriction. This denotes a severe service restriction for a vessel intended for consideration of unrestricted, open-ocean operation. When the project was originally started it was anticipated that a ship with service restriction **R0** or **R1** would be available for the study. Unfortunately, none of the **R0/R1** owner/operators was willing to provide enough information considering the structural designs of their vessel to complete the work required for this project. Given the new developments regarding US Navy interest in High Speed Vessels, HSV, and the US Army interest in Theatre Support Vessels, TSV, there was too much potential for commercial opportunity to allow distribution of structural design information through an SSC report and potentially jeopardize a competitive edge in this market share. This led to selection of the **R4** *PacifiCat's* as the final vessel for the conversion study.

1.5 PRELIMINARY VEHICLE ARRANGEMENT ON VEHICLE DECK (SECTION 4)

Section 4 presents the preliminary arrangement of the military vehicle payload on the Main Vehicle Deck of the PacifiCat, which has Upper and Main Vehicle Decks. It was not necessary to use the Upper Vehicle Deck to accommodate the military payload defined for this project. A revised arrangement is presented later in the report that more efficiently stores the vehicles to minimize the structural conversion work required on the vehicle deck. The preliminary vehicle arrangement assumed 12 inches between bumpers of stowed vehicles. Research confirmed that the US Marines typically assume 10 inches clearance between vehicles, "Marine Lifting and Lashing Handbook, Second Edition, October 1996, MTMCTEA REF 97-55-22 [3].

1.6 STRUCTURAL LOADS REQUIRED FORTHE CONVERSION (SECTION 5)

Section 5 uses both the ABS and DNV rules cited above to calculate the applicable primary hull girder and secondary loads acting on the vessel for operation in unrestricted, open-ocean environments. Tables compare and summarize the hull girder and slamming loads. This section also provides some of the original loads that were used for the design of the PacifiCat's.

The vehicle tire loads are also defined in this section. There was significant problem getting accurate determination of all the tire footprint loads. Enough representative data was available to develop reliable structural calculations for this project and determine structural modifications that would be required to accommodate the new payload. During this portion of the project it was concluded that the tire loads fit into one of three basic categories:

1. Tires with a width less than the deck longitudinal stiffener spacing,
2. Tires with a width nominally greater than the deck longitudinal stiffener spacing and,
3. Tires with a width that approaches twice the deck longitudinal stiffener spacing.

Since most deck plate calculations assume the first condition in their sizing requirements for deck plate subjected to tire loads, i.e., simple support boundary conditions for the plate panel, it was considered necessary to perform some detailed calculations for the other two scenarios to accurately determine the deck structural requirements in way of these tire loads. These other cases were analyzed using finite element analysis techniques in Section 6.

1.7 STRUCTURAL MODIFICATINS REQUIRED FOR THE CONVERSION (SECTION 6)

Section 6 provides the calculations detailing the structural modifications necessary to satisfy the loads developed in Section 5. Calculations include investigation of the modifications required for the primary hull girder loads, secondary slam loads and vehicle loads on the Vehicle Deck.

Section 6 also provides the finite element analyses, FEA, that were developed to investigate the tire size/deck longitudinal spacing issue discussed above. This FEA work presents insight to some of the structural optimization that can be realized using these tools compared to the algorithms presented by the rules.

1.8 STRUCTURAL MODIFICATION COST ESTIMATES (SECTION 7)

Section 7 provides cost estimates to accommodate the structural modifications required for these conversions in accordance with the requirements resulting from both ABS and DNV. The cost estimates also include typical costs for port facilities, material painting and procurement and installation of the vehicle tie downs required for the conversion.

1.9 CONCLUSIONS

Conclusions are presented at the end of the report that suggest the limited increases that would be required for future designs to accommodate the structure that would allow for use as an open-ocean, unrestricted craft. Owners may give consideration to this possibility and may also approach the Government to help absorb the up-front costs of fabrication and operation of the heavier ship offering their vessels for service in time of need for the military or national emergency.

2.0 SEALIFT MISSION REQUIREMENTS OF CONVERSION VESSEL

2.1 INTRODUCTION

This section defines the Sealift mission requirements for converting a commercial high-speed ferry for use in transporting military vehicle cargos. A comparison is made between the load requirements of the U.S. Army's Stryker Brigade Combat Team (SBCT) and the U.S. Marine Corps Marine Air Ground Task Force (MAGTF) or Marine Expeditionary Unit (MEU). Both of these loads represent actual or planned sealift loads for either rapid deployment (SBCT) or a lightweight, sea based combat force (MEU). After consideration of both loads, and the data available to support this project, a notional load based on the MEU is defined for this project.

The preliminary survey for the commercial high-speed craft available for conversion suggest that the converted vessel will have less capability than a purpose built dedicated military design. Based upon the assumed loads, the requirements for the commercial conversion craft have been set at approximately 1/3 less than the payload target of a dedicated military craft.

This section of the report is broken into several sections:

- Background Assumptions: Assumptions that focused the direction of the requirements development effort.
- Existing Military Vessel Characteristics Summary: Review of existing high-speed transport vessels designed to serve military missions.
- Load Planning Assumptions: Assumptions that have refined the requirements based upon particular considerations.
- Deck Height Profiles & Notes: Analysis of the SBCT and MEU to identify deck height and cargo weight requirements based upon threshold and objective requirements for numbers of sorties to move the complete unit.
- Load Planning Notes: Requirements details to support the designers in the evaluation of candidate high-speed ships for conversion. Includes critical vehicles that may impact the selection of candidate high-speed ships and the conversion requirements for each.
- Notional Load list: A representative listing of military vehicles based on a self-deployable slice of the MEU.

The notional vehicle load for the conversion vessel is provided in Table 2-1. It is based on the LHA(R) load, which is developed from the USMC MEU load. Except for tanks and tank recovery vehicles, the conversion load includes all of the vehicles contained in the LHA(R) load, albeit in smaller quantities. Further discussion on the exclusion of the tanks is provided below. For clarity, Table 2-1 shows the total number of vehicles required by the LHA(R) load and the total number of vehicles to be transported by the conversion vessel. The areas and weights shown in Table 2-1 are for the conversion vessel.

Table 2-1. Notional Vehicle Load for Conversion Vessel

NOMENCLATURE	LHA(R) QTY	Conv QTY	LN	WD	HT	WT	Area (SQ FT)	TOTAL WT (LBS)
AN/MLQ-36	1	1	255	99	126	28,000	175	28,000
AN/MRC-138B	8	2	180	85	85	6,200	213	12,400
AN/MRC-145	8	2	185	85	83	6,200	218	12,400
ARMORED COMBAT EXCA	2	1	243	110	96	36,000	186	36,000
TRK, FORKLIFT	1	1	315	102	101	25,600	223	25,600
TRK, FORKLIFT, 4K	1	1	196	78	79	11,080	106	11,080
TRAM	1	1	308	105	132	35,465	225	35,465
CONTAINER, QUADRUPLE	70	20	57	96	82	5,000	760	100,000
CHASSIS TRLR GEN PUR, M353	1	1	187	96	48	2,720	125	2,720
CHASSIS, TRAILER 3/4 T	2	2	147	85	35	1,840	174	3,680
POWER UNIT, FRT (LVS) MK-48	2	1	239	96	102	25,300	159	25,300
TRLR CARGO 3/4 T (M101A1)	4	2	145	74	50	1,850	149	3,700
TRAILER CARGO M105	6	2	185	98	72	6,500	252	13,000
TRLR, MK-14	2	2	239	96	146	16,000	319	32,000
TRLR TANK WATER 400 GL	2	1	161	90	77	2,530	101	2,530
TRK AMB 2 LITTER	1	1	180	85	73	6,000	106	6,000
TRK 7-T MTVR	24	8	316	98	116	36,000	1,720	288,000
TRK 7-T M927 EXTENDED BED	1	1	404	98	116	37,000	275	37,000
TRK 7-T DUMP	1	1	315	98	116	31,888	214	31,888
TRK TOW CARRIER HMMWV	8	4	180	85	69	7,200	425	28,800
TRK, MULTI-PURPOSE M998	45	15	180	85	69	6,500	1,594	97,500
TRK,AVENGER/CLAWS	3	1	186	108	72	7,200	140	7,200
TRK ARMT CARR	10	2	186	108	72	7,000	279	14,000
TRK LIGHT STRIKE VEHICLE	6	2	64	132	74	4,500	117	9,000
155MM HOWITZER	6	2	465	99	115	9,000	639	18,000
LAV ANTI TANK (AT)	2	2	251	99	123	24,850	345	49,700
LAV C2	1	1	254	99	105	26,180	175	26,180
LAV ASSAULT 25MM	4	2	252	99	106	24,040	347	48,080
LAV LOGISTICS (L)	3	1	255	98	109	28,200	174	28,200
LAV, MORTAR CAR	2	1	255	99	95	23,300	175	23,300
LAV, MAINT RECOV	1	1	291	99	112	28,400	200	28,400
TANK COMBAT M1A1	4	-	387	144	114	135,000	-	-
MAINT VAN	4	2	240	96	96	10,000	320	20,000
RECOVERY VEH, M88	1	-	339	144	117	139,600	-	-
Totals:	**238**	**87**					**10,629**	**1,105,123**
Total STons								553

Other notional requirements for the conversion vessel are given as:

- Speed (Fully Loaded) 36 knots
- Range 4000 NM
- Vehicle stowage capacity 500 short tons (Threshold)
  750 short tons (Objective)

2.1.1 Speed

The definition of 36 knots as the speed for the fully loaded vessel represents a consensus based on the speeds of the commercial high-speed ferries that have been identified as part of the preliminary search for candidates for the conversion. In particular, the larger commercial ferries, which represent more likely candidates for the conversion, tend to have speed ratings between 35 knots and 41 knots. As further definition is developed for this study better definition will be available for speed. However, it must also be recognized that the structural weight of the vessel will almost certainly increase as a result of the conversion. This will have the dual effect of reducing the speed and/or cargo capacity of the converted vessel.

2.1.2 Range

The range is aggressive and will certainly not be satisfied by any of the conversion vessels. As seen in Table 2-2, the Actual Ships have ranges on the order of 600NM whereas the Notional Ships will be designed to have significantly greater ranges. At this time, it has not been possible to gather the data regarding the actual ranges of the commercial vessels that have been collected for the preliminary identification of conversion candidates.

2.1.3 Vehicle Stowage Capacity

The Threshold and Objective values defined for this project are based on the cargo requirements associated with the US Army Stryker Brigade Combat Team, SBCT, even though the actual load out is based on the MEU. As shown in Appendix A, the full complement of one SBCT has a weight of 14,403 short tons. The criteria defined by the Operational Requirements Document, ORD, for the US Army Theatre Support Vessel, TSV, show that it must be able to transport the full complement of 14,403 short tons in 20 sorties and therefore defines a cargo capacity of 750 short tons per sortie. With most of the larger commercial ferries advertising cargo capacities in the range of 500 short tons it was decided to define this as the Threshold cargo capacity for the conversion vessel with an Objective value of 750 short tons. The notional load defined in Table 2-1 results in a cargo weight of 553 short tons and is consistent with the cargo capacity goals defined for the conversion vessel.

2.2 DATA ON US ARMY & US MARINE CORPS NOTIONAL LOADS INVESTIGATED FOR THIS PROJECT

Before selecting the notional load defined by Table 2-1, a thorough review of the US Army SBCT and the USMC MEU was undertaken. This review included defining the deck area and deck height requirements associated with each of the vehicle loads. It also included defining the vehicle weights associated with the various loads and insuring that the notional load defined for this project represented a realistic load and was acceptable from an overall weight standpoint.

Finally, this review also provided some definition on vehicles to be excluded from the mix transported by the conversion vessel. This included M1A1 tanks and their support vehicles as well as aviation assets. Further discussion on all of this review is presented below.

2.2.1 Background Assumptions

The following background assumptions concerning load and design requirements have been made.

- The Stryker Brigade Combat Team (SBCT) represents the US Army load for studies of future rapid deployment. See Appendix A for comprehensive load list.

The Army has established aggressive goals for achieving a lighter, more mobile and more rapidly deployed force as part of their ongoing transformation initiative. This force is to come into being over the next several decades as an interim and then final force. The SBCT is the interim force and represents the mid-term transformation to the lighter Army. As such, the SBCT is being used for Army mid term (circa 2015) mobility planning.

- The Marine Expeditionary Unit (MEU) represents the USMC load for rapid deployment. See Appendix A for comprehensive load list.

  The doctrine, strategy, and tactics of the MEU are well established, and regularly reviewed to support amphibious ship availability and construction schedules. The data in Appendix A represents the results of the Department of the Navy (DoN) Lift II study which provides a benchmark of future concepts of operations, force structure, and estimated lift requirements of a potential 2015 MEU. The actual notional load recommended for this project is a down sized version of the notional 2015 load for the LHA(R), which was based on the MEU.

- The U.S. Army Theater Support Vessel (TSV) Block I Operational Requirements Document (ORD) represents the planning requirements for size, speed, and payload for an intra-theater sealift vessel.

  The TSV program ORD requirements have been established around the SBCT. As such, this notional vessel represents the Department of Defense (DoD) developed "military" solution for transportation of the Army's future combat force.

- It is assumed that the converted high-speed commercial ships will have less capability than a purpose built dedicated military design. For the purposes of this study, the military threshold requirements for the TSV have been established as the objective requirements for the commercial conversion class and the threshold requirements for the commercial class have been set approximately 1/3 below those numbers based on expected capabilities of commercial ferries available for conversion.

  The rationale for this assumption is that in times of war, missions going into harms way, or critical deployments, a dedicated military asset such as the Army TSV will be used. These conversion ships will be used for low threat follow-on missions or deliberately planned lifts.

2.2.2 Summary of Existing High-Speed Military Vessel Characteristics

Table 2-2 provides an overview of high-speed vessels recently designed and/or used for military missions. Also included are several notional vessels including the three "blocks" of TSV's presently being developed by the U.S. Army.

Threshold and Objective are the terms defining the minimum (Threshold or T) and optimal (Objective or O) solution sets for a design. In some cases, the requirements development process may determine that these values are the same (T=O or T/O)

Table 2-2. Summary of Existing Sealift Mission High-Speed Vessels

		Length	Beam	Draft	Displacement	Speed (Loaded)	Range	Cargo Weight	Cargo Sq Ft
Proposed SR 1437 Study Requirements									
	Notional Requirements	121m (397 ft) (T)		18 ft (T), 15ft (O)		36 kts (T/O)	4000 NM (T/O)	500 STons (T), 750 STons (O)	11,700 (T), 17,300 (O)
Notional Ships:									
	Army TSV Block I	121m (397 ft) (T)		18 ft (T) 15ft (O)		36 kts (T/O)	2400 NM @36 kts (T), 4726 NM @24 kts (T), 4726 NM @ 36 kts (O)	754 STons (T), 1050 STons (O)	20,000 (T), 25,000 (O)
	Army TSV Block II	121m (397 ft) (T)		18 ft (T) 15ft (O)		40 kts (T), 45 kts (O)	4726 NM (T/O)	1050 STons (T), 1250 STons (O)	25,000 (T), 27,500 (O)
	Army TSV Block III	121m (397 ft) (T)		18 ft (T) 15ft (O)		45 kts (T), 50 kts (O)	4726 NM (T/O)	1254 STons (T), 1250 STons (O)	27,500 (T), 29,500 (O)
	Shallow Draft High Speed Sealift (SDHSS)	300 m	37 m	9.4 m	27,135 Tonnes	55 kts	8,700 NM		120,770
Actual Ships:									
	HSV X1 (Joint Venture)	96 m (314 ft)	26 m (87 ft)	4.04 m (15 ft)	1740 LTons (full load)	35 kts	600 NM	545 STons (35 STons Vehicles)	23,000
	HSV X2	98 m (319 ft)	26.61 m	3.43 m		40 kts	4000 NM *	750 LTons (deadweight)	
	TSV 1X (Spearhead)	98 m	26.61 m	3.43 m		40 kts	4700 NM *	750 LTons (deadweight)	
	USMC WestPac Express	101.0 m	26.62 m	4.2 m		33 kts	1240 NM	550 MT	70000 cu ft
	Skjold Class-MCMV Class	44 m	13.5 m	1 m		45 kts	800 NM	270 Tonnes (deadweight)	
	Jervis Bay	86.14 m (282 ft)	26 m (87 ft)	3.63 m		44 kts		415 LTons (deadweight)	

Note: * Extended range listed is based on the use of cargo deadweight for additional fuel and/or reductions in speed

2.2.3 Load Planning Assumptions

The TSV ORD requires 750 short tons cargo capacity as a threshold requirement. Because the subject study is the conversion of commercial assets, this was set as the objective requirement for conversions and the threshold was set 1/3 lower at 500 short tons cargo capacity which is also consistent with the cargo capacities of the largest commercial ferries so far identified for the conversion candidates.

The TSV ORD assumes 20 sorties to transport an entire SBCT (750 tons/sortie). As a result of previous assumption, 30 sorties will be used for commercial mission (500 short tons per sortie). This may equate to a longer deployment time depending on the number of ships available and the vessel speed.

Table 2-3 summarizes the information regarding number of sorties and threshold and objective weights associated with the SBCT and the MEU.

Table 2-3. Summary of Number of Sorties and Weight Requirements Based on TSV ORD

	Threshold		Objective	
	SBCT	MEU	SBCT	MEU
Weight	500	500	750	750
Sorties	30	7	20	5

2.2.4 Elimination of M1A1 Tank and Aviation Assets from the Notional Load

In order to develop a notional vehicle load that is consistent with the capabilities of a conversion vessel it became clear that it would be necessary to eliminate the M1A1 tank and aviation assets (helicopters) from the notional load.

2.2.4.1 Elimination of M1A1 Tank Assets

There were two issues regarding the tanks. First, transporting the M1A1 implies the necessity for transporting the M88 tank recovery vehicles. Each of these vehicles weighs less than 140,000 pounds and transporting one of each would consume almost 140 short tons or almost 28% of the total cargo weight available for the threshold capacity. It was felt that this would severely limit the number of additional vehicles that could be transported on a given sortie and was viewed as an unattractive limitation, not desirable for the conversion vessel.

Regardless of the weights, is the other big issue of the ramps that are required to load and off-load these vehicles. The RO/RO ramps required to load and off-load these vehicles are heavy structures whose design is governed by these vehicles. Their weight would not only decrease the cargo capacity remaining for military vehicles but they also require heavy cranes to position them for operation. Each of these considerations would impose further limitations by having to accommodate the M1A1 and the M88's.

2.2.4.2 Elimination of Aviation Assets

Due to special handling needs, and outsized loads, the conversion high-speed ships will not be required to accommodate aviation assets (helicopters) for the purposes of this study. As a follow-up study to this project, the final design can be re-evaluated for its ability to accommodate selected aviation assets.

Missions requiring a significant air wing are assumed to be higher priority military missions, which will receive priority consideration for military high-speed sealift assets.

Additional concerns were also raised regarding the NAVAIR requirements that might accompany the transport of aviation assets. This could have increased effects on the arrangeable areas of the vehicle decks as well as secondary systems such as fire fighting. These concerns are beyond the scope of this project. An actual conversion design would have to address this along with other system impacts such as fire fighting, HVAC and other mechanical systems.

As a result of these assumptions, the aviation element was eliminated in its entirety from the load out for the conversion efforts associated with this project.

2.2.5 Deck Height Profiles & Notes

The following deck height profiles for square footage and cargo weight have been developed by evaluating the SBCT and MEU loads as sorted by height of cargo. The requirements (per height category) were divided by the assumed number of sorties (threshold and objective) in order to obtain the actual required deck height profile per ship. For the purposes of this evaluation, and with the exceptions noted below, it has been assumed that an equal fraction of each deck height requirement is carried on each sortie. Table 2-4 summarizes these requirements.

Table 2-4. Summary of Deck Height/Area Requirements for SBCT &MEU

Square Footage	**Threshold (Sorties)**		**Objective (Sorties)**	
Deck Height	**SBCT (30)**	**MEU (7)**	**SBCT (20)**	**MEU (5)**
Below 48 inches (4 FT)	282	487	423	681
Between 49 and 96 inches (8 FT)	3,009	3,937	4,514	5,512
Between 97 and 114 inches (9.5 FT)	4,471	895	6,706	1,253
Between 115 and 120 inches (10 FT)	382	1,962	382	2,747
Between 121 and 144 inches (12 FT)	930	1,032	1,395	1,445
Between 145 and 156 inches (13 FT)	N/A	246	N/A	246
Other Cargo	279	N/A	419	N/A
Totals	9,353	8,559	13,839	11,884

Normal loadout planning includes a stow factor of 0.85 for pre-positioned loads and 0.75 for surge loads. Experience shows that factors closer to the preposition factor can be achieved even for surge loads. Based on this experience, a stow factor of 0.8 has been assumed for this study. Using the SBCT load as a worst case (larger footprint than the MEU) and rounding up to the next 100 square feet equals an expected required square footage of 9353/0.8 = 11,700 (threshold) and 13,839/0.8 = 17,300 (objective). While a stowage factor of 0.8 is acceptable for this project, it is also noted that experience with surge loads has shown stowage factors as low as 0.7.

Table 2-5 provides a summary of weight requirements of the load as a function of deck height for the threshold and objective SBCT and MEU.

Table 2-5. Summary of Deck Height/Load Requirements for SBCT & MEU

Load (lbs)	**Threshold (Sorties)**		**Objective (Sorties)**	
Deck Height	**SBCT (30)**	**MEU (7)**	**SBCT (20)**	**MEU (5)**
Below 48 inches (4 FT)	9,181	9,749	13,772	13,648
Between 49 and 96 inches (8 FT)	181,747	336,033	272,621	470,446
Between 97 and 114 inches (9.5 FT)	616,826	125,674	925,239	175,944
Between 115 and 120 inches (10 FT)	15,750	236,202	15,750	330,683
Between 121 and 144 inches (12 FT)	98,676	193,646	148,014	271,105
Between 145 and 156 inches (13 FT)	N/A	40,960	N/A	40,960
Other Cargo	47,620	N/A	71430	N/A
Totals (lbs)	969,800	942,264	1,446,826	1,302,786
Totals (STons)	485	471	723	651

The SBCT deck height between 115 and 120 inches (10 FT) is driven by the 12 155mm howitzers. Since it is not possible to carry 1/20th or 1/30th of 12 howitzers, these requirements are based on 1/12th of the force or one howitzer and are the same for threshold and objective.

The MEU deck height between 145 and 156 inches (13 FT) is driven by the M1085C Hydraulic Wheeled Excavator. Since it is not possible to carry a fraction of this vehicle, these requirements are based on the actual requirements of the excavator and are the same for threshold and objective.

The actual clear height from the Main Vehicle Deck to the underside of the structure on the Upper Vehicle Deck is approximately 4.2 meters, 165", which allows plenty of overhead for any of the assets contained in either the MEU or the SBCT.

2.2.6 Load Planning Notes

Pounds per square foot (PSF) loading requirements have been evaluated on a per deck height basis both as an overall average for the cargo to be stowed and for the heaviest individual vehicle in the height range. Table 2-6 and Table 2-7 summarize this data. In several cases, the significant value is 2-3 times the average value. This consideration may in turn drive additional load planning decisions (i.e. where a particular piece of cargo may be stowed) or basic load planning assumptions (i.e. can this piece of cargo be accommodated on the conversion class.

Table 2-6. Average Deck Loads/Height for SBCT & MEU

Overall Average of Vehicles in Deck Height Range (PSF)		
Deck Height	**SBCT**	**MEU**
Below 48 inches (4 FT)	33	20
Between 49 and 96 inches (8 FT)	61	85
Between 97 and 114 inches (9.5 FT)	138	140
Between 115 and 120 inches (10 FT)	42	120
Between 121 and 144 inches (12 FT)	107	188
Between 145 and 156 inches (13 FT)	N/A	167
Other Cargo	171	N/A

Table 2-7. Heaviest Individual Average Vehicle Load for SBCT & MEU

Heaviest Vehicle in Deck Height Range (PSF)		
Deck Height	**SBCT**	**MEU**
Below 48 inches (4 FT)	134	65
Between 49 and 96 inches (8 FT)	187	291
Between 97 and 114 inches (9.5 FT)	209	163
Between 115 and 120 inches (10 FT)	42	148
Between 121 and 144 inches (12 FT)	132	201
Between 145 and 156 inches (13 FT)	N/A	167
Other Cargo	171	N/A

2.2.7 Key Design Vehicle Lists

The key design vehicle lists below in Table 2-8 and Table 2-9 were derived from Appendix A. The master lists were sorted for the five largest vehicles in each measurement (Length, Width, Height, Weight, and Square Footage) and then the result simplified and summarized into these tables. The intent of these tables is to provide the structural and arrangements trades with some indication of particularly challenging equipment requiring access within the ship.

While information on turning radius will be available to ensure notional vehicle loading on the conversion vessel is practicable it is not expected to be a major concern. The conversion vessels are ferries with straight on/off loading access, either fore/aft or amidships and, unlike typical Sealift vessels do not require movement of vehicles along fixed ramps between decks of the ship.

Table 2-8. Key Design Vehicle List for US Army (SBCT)

LIN	Nomenclature	Model	LN	WD	HT	WT	SQ. FT.
K57821 01	HOWITZER TOWED 155M	M198	**496**	111	**117**	15750	**382**
L28351 47	KITCHEN FIELD TLR M	MKT-95	201	**152**	**132**	5260	212
R41282 01	RECON SYS NBC	M93A1 FOX	288	**118**	105	38500	236
T63093 33	TRUCK WRECKER 8X8	M984A1 WWN	**402**	102	112	**51300**	**285**
T87243 11	TRK TANK 2500 GAL	M978 WOWN	**401**	96	112	38165	**267**
T96496 05	TRUCK CARGO TAC W/L	XM1120 WOW	**401**	96	**129**	35300	**267**
YA0005 01	WATER PURIFICATION	2-1/2-TON	281	100	**128**	19480	195
Z43601 02	INFANTRY CARRIER VE	STRYKER	288	**113**	106	**42000**	226

Table 2-9. Key Design Vehicle List for USMC (MEU)

TAMCN	NOMENCLATURE	LN	WD	HT	WT	SQ. FT.
B0589	AMORED EARTHMOVER, ACE M-9	243	**110**	96	**54,000**	186
B0591	EXCAVATOR HYD WHL 1085C	**365**	97	**154**	**40,960**	246
B2482	TRACTOR, ALL WHL DRV W/ ATACH	277	94	**141**	13,000	181
B2567	TRAM	308	105	**132**	35,465	225
C4154	INFLATABLE BOAT RIGID HULL	**468**	98	80	6,000	**319**
D1061	MTVR 7 TON EXT BED	**404**	98	116	31,500	**275**
D1212	TRUCK WRECKER 5 TON 6X6	346	98	114	**38,466**	235
E0665	155MM HOWITZER	**465**	99	115	15,400	**320**
E0846	AAAV PERSONNEL (P-7)	**360**	**144**	**126**	**72,500**	**360**

2.2.8 Notional Load List

Table 2-10 provides a summary of area and weight requirements of the notional load based on deck height requirements. This is based on Table 2-1, which is the notional load list based on the 2015 projected USMC load for the LHA(R). This load has been reviewed with USMC planners and been determined to be a representative "slice" of the MEU, which could be expected to deploy as an independent unit.

Note that this notional load is sized (both by weight and footprint) to be between the threshold and objective requirements discussed above.

Table 2-10. Summary of Area and Weight Requirements Based on Deck Height for the Notional Load

Notional Load Profile	**Square Footage**	**Load (lbs)**
Deck Height		
Below 48 inches (4 FT)	298	6,400
Between 49 and 96 inches (8 FT)	5,140	396,910
Between 97 and 114 inches (9.5 FT)	1,277	181,760
Between 115 and 120 inches (10 FT)	2,849	374,888
Between 121 and 144 inches (12 FT)	745	113,165
Between 145 and 156 inches (13 FT)	319	32,000
Other Cargo	N/A	N/A
Totals	10,629	1,105,123
Short Tons		553

2.3 CONCLUSION ON SEALIFT MISSION REQUIREMENTS

The review of the US Army SBCT and the USMC MEU developed for this project provided good insight to the philosophy of these two organizations and their requirements for high-speed vessel capability. It is obvious that the ability to access high-speed transport is an important component for both force structures. The conversion vessels will help to satisfy these requirements with certain limitations that are consistent for the capabilities of a conversion vessel.

As this project proceeds, the notional load presented in this deliverable may be re-defined as a function of the commercial high-speed ferries available for this mission and its actual speed and payload capabilities. The structural modifications required for these vessels to accommodate the notional load may in fact consume a significant portion of the available displacement and limit the cargo capacity of the conversion vessels.

One of the lessons of this project will come from the weight of structural modifications that are required. If they are not significant it may become desirable for owners to include such capacity in their original designs, minimizing time and expense for conversion.

3.0 CONVERSION VESSEL CANDIDATES & THE FINAL SELECTION

3.1 SELECTION PROCEDURES AND REQUIREMENTS

This section of the report presents the procedure that was used to select a candidate vessel for the Structural Conversion study. In brief, the process undertaken was as follows:

- Determine vessel requirements based on the notional loadout defined in Section 2.
- Assemble a "menu" of candidate vessels for the conversion.
- Assess candidates versus requirements.
- Develop a set of downselect criteria and,
- Downselect to recommended conversion vessel for this project.

The information presented in this report parallels the outline given above. Thus, paragraph 3.1 of this report presents the determination of the vessel requirements. Paragraph 3.2 summarizes the list of candidate vessels, with back-up data presented in Appendix B. Paragraph 3.3 presents and discusses the alignment of the candidate vessels against the requirements. Paragraph 3.4 presents and discusses the recommendation of a single vessel for use as the conversion candidate. Paragraph 3.5 presents the characteristics of the final vessel selected and Paragraph 3.6 presents the Conclusions for this section of the report.

3.1.1 Required Vessel Characteristics

This section of the report presents the recommendation of what vessel to use for this Structural Conversion Study. The recommendation presented herein is the result of collecting data on commercial ferries and comparing this data with the needs of the project. This section of the report presents a discussion of what those needs are.

The required vessel characteristics are broken into three categories:

- Payload Characteristics: The vessel must be able to carry the payload identified earlier, (Section 2).
- Vessel Engineering Characteristics: The vessel must have been engineered to provide a target level of speed, range, sea state, and similar performance.
- Project Data Availability: There must be appropriate technical data available concerning the vessel, so that this structural conversion study project can move forward. It would be no good to find an otherwise-ideal vessel for which there is no available engineering data.

Each of these sets of requirements is discussed in detail in the following paragraphs.

3.1.2 Payload Characteristics

The derivation of the required military payload is discussed in Section 2. The payload investigation results in a notional vehicle load for the converted vessel. The notional vehicle load is presented in Table 2-1.

Note also that the notional load (Table 2-1) was set arbitrarily at 1/3 below the objective levels. This assumption was based on results from the preliminary vessel surveys that had taken place during the development of the work for this section of the study. The survey indicated that the maximum cargo displacement that could be expected for the largest ferries available for conversion is in the range of 500 short tons. Albeit somewhat arbitrary, this was used to define and validate the threshold cargo requirements. This is not a strict limitation, and as will be seen in following sections, the recommended candidate deviates from this capacity.

The notional vehicle load for the conversion vessel, Table 2-1, is based on the LHA(R) load, which is developed from the USMC MEU load. Except for tanks, tank recovery vehicles and aviation assets, the conversion load includes all of the vehicles contained in the LHA(R) load, albeit in smaller quantities.

The notional vehicle load in Table 2-1 is "translated" into a set of functional requirements in Table 3-1. The emphasis of this translation has been to express the notional vehicle load in the form of weight and vehicle lane-meter requirements, because these are the parameters that are most readily available for the candidate vessels. Once a coarse screening has been accomplished using these gross parameters, it will then be possible to attempt an actual load out arrangement on a given selected candidate vessel's vehicle deck.

In the fast ferry industry it is normal to present the vessel's vehicle deck capacity in terms of the total available length of all vehicle lanes combined. This yields a figure expressed in "lane meters."

Also note that the far right hand column in Table 3-1 presents the total vehicle weight in metric tonnes (2204 pounds per metric tonne). Table 2-1 presents the total weight in short tons (2000 pounds per short ton).

By taking the vehicle lengths given in Table 2-1 and adding a 12-inch gap between vehicles, [3], a total requirement of 438 lane meters of vehicle deck is determined in Table 3-1.

This calculation assumes that all vehicles will fit within a lane, but this obviously depends on the actual designed lane width. Commercial ferry lane widths range from a low of 6.56 feet (2 meters) up to 10.82 feet (3.3 meters). If the width of the average lane is assumed to be 9 or 10 feet than review of Table 3-1 shows that only one vehicle, the 11 foot wide Trk Light Strike Vehicle is perhaps too wide. Since there are only two of these vehicles in the loadout it is assumed that careful planning would allow for their accommodation. Therefore, this was not considered a driving factor for the initial scrutiny of the candidate vessels. Instead, attention is focused on finding candidate vessels with about 500 metric tonnes of vehicle payload capacity and 438 lane-meters of vehicle deck. Note that it may become necessary to revisit the lane width assumption for any given vessel.

Table 3-1. Conversion of Notional Vehicle Load into Lane-Meters and Metric Tonnes

NOMENCLATURE	Conv QTY	LN (in)	WD (in)	HT (in)	Wt Each Unit (lbs)	Total Wt (lbs)	Total Length (m)	Total Wt (m-tonnes)
AN/MLQ-36	1	255	99	126	28,000	28,000	7	13
AN/MRC-138B	2	180	85	85	6,200	12,400	10	6
AN/MRC-145	2	185	85	83	6,200	12,400	10	6
ARMORED COMBAT EXCA	1	243	110	96	36,000	36,000	6	16
TRK, FORKLIFT	1	315	102	101	25,600	25,600	8	12
TRK, FORKLIFT, 4K	1	196	78	79	11,080	11,080	5	5
TRAM	1	308	105	132	35,465	35,465	8	16
CONTAINER, QUADRUPLE	20	57	96	82	5,000	100,000	35	45
CHASSIS TRLR GEN PUR, M353	1	187	96	48	2,720	2,720	5	1
CHASSIS, TRAILER 3/4 T	2	147	85	35	1,840	3,680	8	2
POWER UNIT, FRT (LVS) MK-48	1	239	96	102	25,300	25,300	6	11
TRLR CARGO 3/4 T (M101A1)	2	145	74	50	1,850	3,700	8	2
TRAILER CARGO M105	2	185	98	72	6,500	13,000	10	6
TRLR, MK-14	2	239	96	146	16,000	32,000	13	15
TRLR TANK WATER 400 GL	1	161	90	77	2,530	2,530	4	1
TRK AMB 2 LITTER	1	180	85	73	6,000	6,000	5	3
TRK 7-T MTVR	8	316	98	116	36,000	288,000	67	131
TRK 7-T M927 EXTENDED BED	1	404	98	116	37,000	37,000	11	17
TRK 7-T DUMP	1	315	98	116	31,888	31,888	8	14
TRK TOW CARRIER HMMWV	4	180	85	69	7,200	28,800	20	13
TRK, MULTI-PURPOSE M998	15	180	85	69	6,500	97,500	73	44
TRK,AVENGER/CLAWS	1	186	108	72	7,200	7,200	5	3
TRK ARMT CARR	2	186	108	72	7,000	14,000	10	6
TRK LIGHT STRIKE VEHICLE	2	64	132	74	4,500	9,000	4	4
155MM HOWITZER	2	465	99	115	9,000	18,000	24	8
LAV ANTI TANK (AT)	2	251	99	123	24,850	49,700	13	23
LAV C2	1	254	99	105	26,180	26,180	7	12
LAV ASSAULT 25MM	2	252	99	106	24,040	48,080	13	22
LAV LOGISTICS (L)	1	255	98	109	28,200	28,200	7	13
LAV, MORTAR CAR	1	255	99	95	23,300	23,300	7	11
LAV, MAINT RECOV	1	291	99	112	28,400	28,400	8	13
MAINT VAN	2	240	96	96	10,000	20,000	13	9
Totals:	**87**					1,105,123	**438**	**501**

Once the initial coarse-filter is complete it will be desirable to consider a detailed vehicle arrangement on the car deck, including any obstructions that would complicate the load/unload operation. It is also necessary to ensure the vertical clearance of the Vehicle Deck will accommodate all vehicle heights identified in the military vehicle payload, and that vehicle access door locations are consistent with military vehicle maneuvering characteristics.

3.1.3 Vessel Engineering Characteristics

In addition to the vehicle payload, additional requirements for the conversion vessel are given in Section 1 as:

• Speed (Fully Loaded)	36 knots
• Range	4000 NM
• Vehicle stowage capacity	500 short tons (Threshold) 750 short tons (Objective)

The definition of 36 knots as the speed for the fully loaded vessel is described in Section 2 as a consensus based on the speeds of the commercial high-speed ferries that have been identified as part of the preliminary search for candidates for the conversion.

As seen below, the speed of the conversion candidates exhibits a very linear dependence upon the deadweight (load) imposed upon the vessel. In other words, the ships are quoted as being capable of, say, 40 knots on a deadweight of "X" or 35 knots on a deadweight of "Y". This establishes a relationship between speed and deadweight, for any given vessel.

The range requirement in Section 2 is aggressive and will certainly not be the design range of any of the candidate conversion vessels. As seen below, the candidate ships have ranges on the order of a few hundred nautical miles. This is so that they can devote their lift capacity to the carriage of cargo, and not fuel. Fuelings are relatively frequent.

The ships are, however, capable of carrying greater amounts of fuel. Indeed, the INCAT vessels come fitted with extra tank capacity to make possible vessel repositioning without the use of portable tanks. Thus, longer ranges are possible, but they will come at the expense of payload capacity.

The result is that speed, range, and payload are traded off, one against the other. Most of the large fast car ferries have extra tank capacity and can easily be configured to carry enough fuel for the given range. The problem is that since total deadweight (payload + fuel) is fixed for a given speed the weight of fuel will reduce – one-for-one – the payload lift capacity.

Further, it must also be recognized that the structural weight of the vessel will almost certainly increase as a result of the conversion. This will have the dual effect of reducing the speed and/or cargo capacity of the converted vessel. These trade-offs will be included in future reports as the work is developed.

In addition to the primary requirements of payload, speed, and range, this project also requires that the structural design of the converted vessel allow it to support open ocean, unrestricted operation. The Det Norske Veritas High Speed and Light Craft Rules, DNV HSLC, is commonly used for the design of these vessels. These rules do not support open ocean, unrestricted operation and it is necessary to investigate other DNV rulebook options for developing the structure for this notation. A slight variant on these rules is the more recent DNV High Speed, Light Craft and Naval Surface Craft, DNV HSLC&NSC. These rules include vessel types "Patrol" and "Naval", for which "Naval" only can receive an unrestricted notation.

In the DNV HSLC rules, Part 1, Chapter 1 Section 2/Table B1, repeated below as Figure 5-5, defines Service Area Restriction notations. Note 1 to this table states:

> "*Unrestricted service notation is not applicable to craft falling within the scope of the HSC Code, i.e., service and type notations* ***Passenger****,* ***Car Ferry*** *or* ***Cargo****.*"

Service Area Restriction "**R0**" ("R Zero") is the most unrestricted and it requires that such a craft can be no more than 300 nautical miles from safe harbor, obviously not open ocean. **R1** is the next level of service restriction. Higher **R** notations correspond to increasing degrees of restriction, the highest one found in the fast ferry world being the **R4** notation (coastal protected waters) found on the Canadian PacifiCat. The **R0** classification is not an open-ocean unrestricted notation but corresponds to vessel-specific limiting wave heights, generally in the range of 20 feet (6 meters).

For this conversion study the **R** notation should be as unrestrictive as possible. The more restrictive the notation the greater the impact to the load and scantling requirements necessary to satisfy the conversion.

Also please note that, the discussions presented above are in primary consideration of structural loads and scantlings, which are the specific focus of this project. Other elements of the **R** notation, and elements of the IMO regulations, would also have impacts on the vessel engineering, distributive systems, etc. For the current project, the study is limited to those items that have a direct influence on the structural design of the vessel.

Finally, regarding the vessel itself, there are structural parameters that can be considered at this stage. These include assessing the arrangement and structural arrangement of the craft, and identifying any potential problems or benefits that would complicate or simplify the conversion.

Also, one should consider the ease or difficulty of accessing the Vehicle Deck structure to accomplish the anticipated modifications that would be required for the conversion. Vessels wherein this structure is rendered inaccessible due to the presence of thin double bottoms or similar impediments may be disqualified on the basis of "practicality."

3.1.4 Project Data Availability Characteristics

A final class of parameters has been considered in performing the vessel candidate downselect. These parameters concern the availability of project data for the vessels. Several considerations on this point exist:

- Does the necessary engineering data exist? Some of the attractive candidates are older vessels whose builders have since gone out of business. In such case it may be impractical to access the engineering data required for this project.
- Is the vessel available for conversion? Some of the vessels on the list of candidates are designs only and have not yet been built. There is always the likelihood that they will not be built and it is considered inconsistent with the original assumptions for this project to use a vessel that does not actually exist.
- Is the owner of the design data likely to be willing to grant access to that data? Many ferry designers consider their structural details to be trade secrets and are very unwilling to share. The older the vessel the less likely this is to be a problem. This is also a big problem due to the highly competitive marketplace that has evolved to

support high speed vessels for the US Navy, HSV programs, and US Army, Theater Support Vessel, TSV, programs.

- Is the vessel broadly representative of the current and near future state of the art? It may be possible to get data on old vessels, and these vessels may, because of their age, actually be available for purchase and conversion, but is this the most meaningful candidate for the study? Would it not be more valuable to perform the study upon vessels that are representative of the best thinking in the industry, and not on ones that are arguably obsolete.

These parameters, as may be seen, are subjective and interrelated. Nevertheless they have been used in the downselect process as will be described below.

3.2 MENU OF CANDIDATE VESSELS

This section presents the compilation of a "menu" of candidate vessels for the conversion. In producing this list publicly available data on fast car ferries was surveyed and supplemented with in-house knowledge or direct contacts with builders.

A significant element of this effort has been to compile information in a manner that is comparable to the statement of requirements, given in paragraph 3.1. This has relied on some extrapolations and calculations, as will be discussed herein. For example, where length of lane-meters is not quoted by the builder estimates were developed by assuming a 4.5m average car length, times the quoted car capacity. The 4.5m length is a standard figure used to approximate lane-meter capacities in the international ferry trade.[1]

The calculations of available area are not perfectly accurate, because they do not necessarily take into account the vertical clearances required – in other words the stated amount of lane length may not be of a usable height for the military mission. However, this is not a fatal flaw in the analysis because the vessels are limited by their weight capacity and not by their deck area. In general, the vessels, because they are configured for carriage of low-density passenger cars and parcel trucks, have more than the required lane length, even if they are marginal on payload weight capacity. Therefore, extra effort was not justified to make the deck area calculations more precise. This is illustrated graphically in Figure 3-1, which plots the lane-meters available on the candidate ferries as a function of their deadweight.

[1] In the case of the Alaska and Canada vessels a longer vehicle length was used due to unique owner's requirements for these vessels.

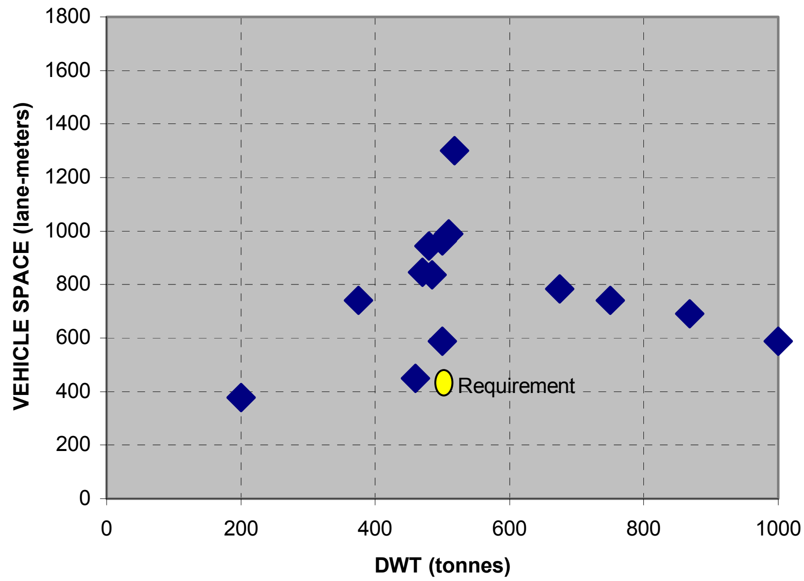


Figure 3-1. DWT Tonnage and Vehicle Deck Space for Candidate Vessels

Car deck axle loads are generally about 800 kg/axle. Special truck lanes are strengthened to 10 or 12 t/ axle.

Any of the ships can be overloaded to some extent, and thus a heavier payload carried at the expense of speed. In fact, most builders quote several different combinations of weight and speed, frequently not corresponding one to the other.

The complete list of identified candidate vessels is presented in Table 3-2. In this table the values given in black color are taken directly from builder's literature. Values calculated for this project are printed in blue color.

Table 3-2. List of Candidate Vessels and Key Characteristics.
(Values in blue are estimated – Values in black are provided by builder)

Designation	Builder	LOA	DWT	Speed	Power	Fuel Capacity	DNV Service Area Restriction	Displ.	Vehicle Capacity	Vehicle Capacity	Axle load	
		m	tonnes	knots	kW	cu m		tonnes	Lane-meters	Cars	t/axle	t/axle
				at quoted DWT	at quoted speed						Truck lanes	Car lanes
AMHS	Derecktor	71.75	unk	35.5	14400	unk	3	unk	210	35		
INCAT 74m	INCAT Tasmania	73.6	200	36	28320	32.8	1	unk	378	84	2	0.8
Auto Express 86	Austal	86	485	41	30600	140	2	unk	837	186	12	1
STENA HSS 900	Westamarin	88	480	40	44000	unk	unk	unk	945	210	unk	unk
INCAT 91m	INCAT Tasmania	91.3	510	42	28320	396	1	unk	990	220	9	2
Auto Express 92	Austal	92	470	40.5	25920	160	unk	unk	846	188	12	1
INCAT 96m	INCAT Tasmania	95.47	868	38	28320	568	1	unk	690	153	10	0.8
Evolution 10	INCAT Tasmania	96	675	38	28320	660	1	1650	785	174	10	0.8
Evolution 10B	INCAT Tasmania	97.22	750	36	28320	660	1	unk	740	164	10	0.8
Evolution 10B	INCAT Tasmania	97.22	375	40	28320	660	1	unk	740	164	10	0.8
Auto Express 101	Austal	101	500	37	25920	160	unk	unk	963	251	15	3
AFAI 110m	A Fai Ships	110	460	60	84000	400	unk	unk	450	100		
Evolution 112	INCAT Tasmania	112	500	45	36000	unk	1	unk	589	130	12	0.8
Evolution 112	INCAT Tasmania	112	1000	40	36000	unk	1	unk	589	130	12	0.8
PACIFICAT	Catamaran Ferries International	122	518	32	26000	94	4	1885	1300	250	7	1.4
STENA HSS 1500	Finnyards	126.6	1500	40	68000	235	1	unk	1688	375	unk	unk

3.3 ALIGNMENT OF CANDIDATE VESSELS AGAINST REQUIREMENTS

The next step in analysis of the candidate vessels is to sort them according to their ability to meet the payload, speed, and range requirements. The initial "filter" is to simply eliminate all vessels that carry less than about 500 tonnes of DWT. This effect is dramatic, cutting the list to the eight vessels shown below (sorted by length). The Evolution 10B and the Evolution 112 are represented twice because their builder has presented two different loading conditions for them.

- INCAT 91m
- INCAT 96m
- INCAT Evolution 10
- INCAT Evolution 10B (2 loading conditions)
- AUSTAL AutoExpress 101
- INCAT Evolution 112 (2 loading conditions)
- CFI PacifiCat
- STENA HSS 1500

These are the only vessels from Table 3-2 that can carry 500t of payload. The next discriminator is range. Some of the short-listed vessels carry only 500t of deadweight. As such, since deadweight includes both payload and fuel, the vessels with only 500t of deadweight would have no capacity left for fuel, and thus would not meet the range requirement.

In order to tackle this issue estimates for the range of the vessel, based on the stated deadweight capacity and power level were developed. The range estimate is very simple: It is assumed that Fuel = (Deadweight minus Payload.) The builder's stated power and speed were used with an assumed average fuel consumption of 200 gallons / kW-h to calculate range.

This calculation is a simplification that ignores the non-fuel elements of deadweight, such as fresh water and passengers (which introduces error in one direction), and it ignores the electric load contributions to fuel consumption (which will introduce error in the opposite direction), and it does not use an actual fuel consumption rate for the specified engines. For the scope of this project, this is a useful metric, and is probably not too far off from the actual vessel performance.

The resulting predicted performance for the listed vessels is presented in Table 3-3.

Table 3-3. Estimated Fuel Weight and Range for short listed vessels.
(Values in blue color are estimated – Values in black color are provided by builder)

Designation	Builder	LOA	DWT	Speed	Power at quoted speed	Fuel weight with 500 t of payload	Calculated Range at listed Speed
		m	tonnes	knots	kW	tonnes	n mi
91m	INCAT Tasmania	91.3	510	42	28320	10	74
96m	INCAT Tasmania	95.47	868	38	28320	368	2469
Evolution 10	INCAT Tasmania	96	675	38	28320	175	1174
Evolution 10B*	INCAT Tasmania	97.22	375	40	28320	-125	-883
Evolution 10B*	INCAT Tasmania	97.22	750	36	28320	250	1589
Auto Express 101	Austal	101	500	37	25920	0	0
Evolution 112*	INCAT Tasmania	112	500	45	36000	0	0
Evolution 112*	INCAT Tasmania	112	1000	40	36000	500	2778
PACIFICAT	Catamaran Ferries International	122	518	32	26000	18	111
HSS 1500	Finnyards	126.6	1500	40	68000	1000	2941

* Note that for two of the vessels, the Evolution 10B and Evolution 112, there is data at two different deadweights. This data shows the effect of loading the vessel. Consider the Evolution 112 vessel: The data shows that at a deadweight of 500 tonnes it is capable of 45 knots. But if it is "overloaded" to a deadweight of 1000 tonnes then the speed – on the same power – will drop to 40 knots. Similarly the Evolution 10B loses 4 knots (from 40 down to 36) when "overloaded" to 750 tonnes of deadweight.

These two examples allow investigation of the feasibility of meeting the speed and range requirements discussed above. The behavior of the Evolution 10B and Evolution 112 have been plotted in Figure 3-2, and, since only two points are given, an assumed linear trend is depicted[2]. Single data points are shown for the STENA HSS 1500, the Austal AE 101, and the CFI PacifiCat, as is a point at the target values of 4000 n mi @ 36 knots.

[2] The linear trend assumption is a simplification. It is possible to "reverse engineer" the vessels and produce a more accurate trend line. For this reason the reader should be cautious especially if extending the performance trend outside the values provided by the shipbuilder

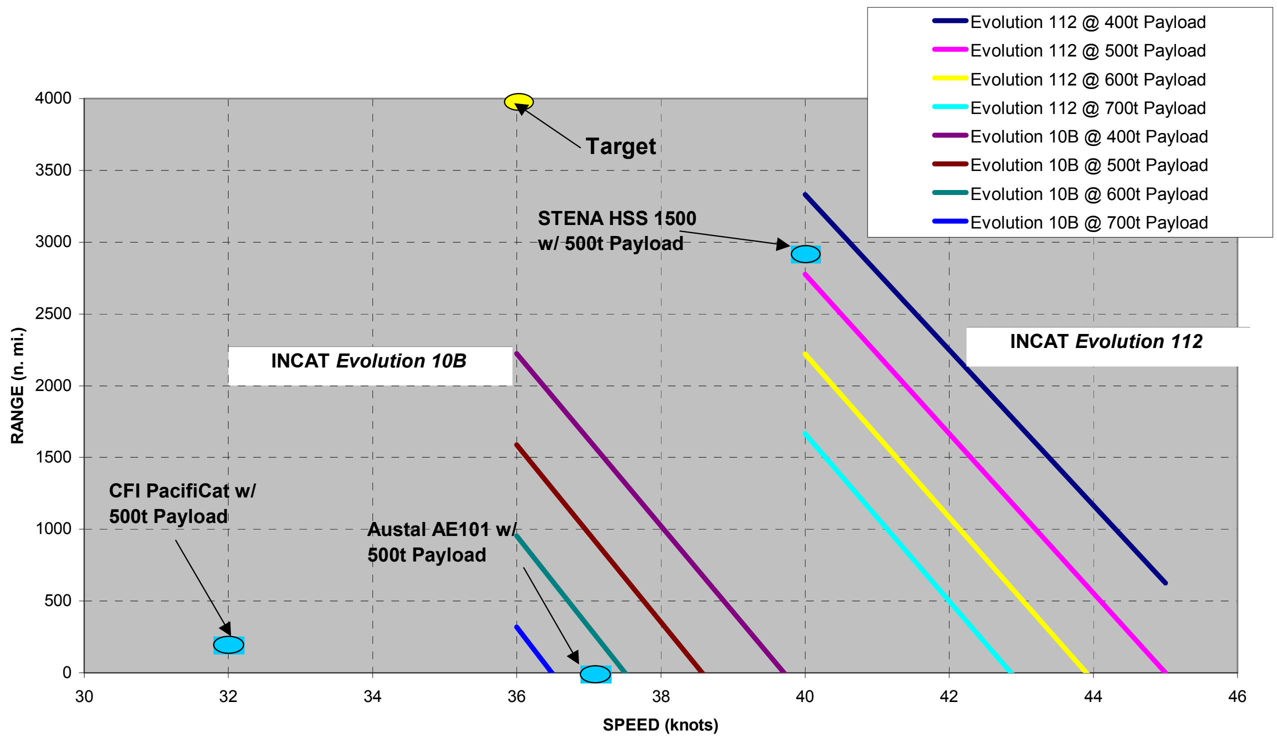


Figure 3-2. Speed, Range and Payload Performance for Candidate Vessels

This figure shows that the desired payload, speed, and range performance can be met with the Evolution 112 and STENA HSS 1500 craft. The Evolution 10B hull (96m) and AUSTAL AutoExpress 101 (101m) may be considered "second best", as it appears that it might be possible to extrapolate from them a vessel capable of over 3000 miles and over 30 knots, while the Canadian PacifiCat falls into the third tier of candidates which does not appear capable of a speed/range combination greater than 30/3000.

The remaining parameter concerns access to data. Incat, Austal, and Finnyards (builder of the STENA HSS) were contacted to inquire about the availability of data to support this project. Incat replied "We have recently taken the view with the wide technical community that we should limit information to that which is in the public domain." Austal replied that their current competition for the US Navy Littoral Combat Ship, LCS, suggested that they should not share data on the AE 101, as they feared it might weaken their competitive position. This was further exacerbated because the Principal Investigator and his employer, JJMA, were supporting a competitor to Austal on the LCS program. Finnyards replied they did have the data, but it was archived and they were not willing to expend the effort to dig it out on an uncompensated basis.

Table 3-4 summarizes the data availability for the vessels listed in Table 3-3.

Table 3-4. Availability of Structural Drawings For Listed Vessels

Designation	Builder	Does the engineering data exist?	Is the vessel available for conversion?	Is the owner likely to grant access to design data?	Is the vessel state of the art?
91m	INCAT Tasmania	**Yes**	**Yes**	No	**Yes**
96m	INCAT Tasmania	**Yes**	**Yes**	No	**Yes**
Evolution 10	INCAT Tasmania	**Yes**	**Yes**	No	**Yes**
Evolution 10B	INCAT Tasmania	**Yes**	**Yes**	No	**Yes**
Evolution 10B	INCAT Tasmania	**Yes**	**Yes**	No	**Yes**
Auto Express 101	Austal	**Yes**	**Yes**	No	**Yes**
Evolution 112	INCAT Tasmania	No - Not Built yet	No - Not Built yet	No	**Yes**
Evolution 112	INCAT Tasmania	No - Not Built yet	No - Not Built yet	No	**Yes**
PACIFICAT	Catamaran Ferries International	**Yes**	**Yes**	**Yes**	Marginal – Many route-specific features
HSS 1500	Finnyards	**Yes**	**Yes**	No	Marginal – Older vessel

3.4 VESSEL DOWNSELECTION

Paragraph 3.3 leads the reader to the conclusion that only vessel with LOA greater than about 100m are technically appropriate for this project, due to range & payload constraints. Eight vessels are identified that would be appropriate candidates for this project. Unfortunately, only one of them, the British Columbia PacifiCat, has technical data available for the projects use.

The conclusion thus seems obvious – use the available data – but it may be informative to discuss the limitations that this selection will impose upon the rest of the project.

The PacifiCat ferries were designed by International Catamaran Designs, Sydney NSW, for the British Columbia Ferry Corporation. The vessels were designed particularly for the waters, routes, and facilities of British Columbia. This tailoring to the specific needs of their owners results in a few features of the vessels that are worthy of note.

Double Ended Loading – The PacifiCats were inserted into a ferry system that currently uses double-ended monohulls. The system assumes a double ended vessel design, where vehicles will drive on over one end of the ship, and drive off over the other end. There is no provision for on-board turn-around of vehicles. This is true of the PacifiCats, and the PacifiCats embody unusual bow and stern features as a consequence. While the vessels don't sail as double enders, their car deck is very much configured like one, with the bow and stern nearly identical in plan form and terminal interface.

Double Deck Loading – BC Ferry's terminals are set up to load the upper and lower vehicle decks simultaneously, through the use of two-level shore-based ramps. This means that there are no onboard ramps permitting vehicles to drive from the upper deck to the lower, or vice versa. This is a difficult arrangement to work with regarding conversion. Earlier studies developed by JJMA for PriceWaterHouseCoopers investigated the impacts of installing on-board ramps to accommodate single point loading operations. These studies are included in Appendix B.

Three different ramp modifications were included in the studies. The first allowed for continued double ended loading operation. The other two addressed single ended operations with stern-to and bow-to loading/unloading only. The studies presented in Appendix B were not exhaustive but do represent the issues involved with each type of modification. Maintaining the double ended operation reduces the vessel capacity by the least amount but is the most expensive because it requires a forward and aft ramp as opposed to the other two systems, which require one ramp each.

Particular Terminal Interfaces – The shore-based boarding ramps at the BC Ferry's terminals are designed to lower and land upon the car deck of the ship. That is to say, that the weight of the ramp is carried upon the ship. In the case of the PacifiCats there is also a shipboard ramp that lowers first, and upon which the shore-based ramp lands. The result of this arrangement is that the shipboard ramp, and it's hydraulic machinery, have been sized not only for the weight of the ramp, but for the weight of the ramp, the shore ramp upon it, and the loaded vehicles that may be crossing. Contrast this situation to the case where the shipboard ramp is lowered onto a concrete seawall. In that latter case the shipboard machinery would never have to deal with anything heavier than the unload ramp itself, whereas the PacifiCat machinery has to deal with the ramp, the shore ramp, and the vehicles, all at once.

R4 Service Restriction – Partially due to the freeboards imposed by the shore interfaces described above, and partially because the PacifiCats were specifically designed for the climate of British Columbia, they were certificated to only a DNV **R4** Service Restriction. This service restriction, along with a 2.5m design limiting wave height, is much lower than the open-ocean, unrestricted operation sought for a vessel under this project.

The following comparison investigates these peculiarities and cautions against the earlier-established criteria for this Project:

The following down select criteria were developed for this task.

- Identify the existing service restriction notation for the craft and assess the magnitude of load increases necessary to satisfy the unrestricted notation required for the conversion. The more restrictive the existing notation, the greater the impact to the load and scantling requirements necessary to satisfy the conversion.

  The PacifiCat's unfortunately have a restrictive Service Notation (**R4**), which may require substantial structural design modifications to bring to the Unrestricted level.

- Assess the structural arrangement of the craft and any potential problems or benefits that would complicate or simplify the conversion.

The PacifiCat's are reflective of best current thinking regarding aluminum fast car ferry construction. JJMA holds a complete set of engineering drawings of the vessels, including analyses of conversion feasibility associated with the above-mentioned **R4** studies. JJMA has also received permission to use these drawings for the current project.

- Assess the ease or difficulty of accessing the Vehicle Deck structure to accomplish the anticipated modifications that would be required for the conversion.

  Physical access to the structure is excellent. Further, the PacifiCats are mothballed in North America and thus are available for shipcheck, if needed, at minimum cost to the project.

  Determine the grade of aluminum used for the existing Vehicle Deck and a rough assessment of its existing strength

- Arrangeable area of the Vehicle Deck and any obstructions that would complicate the load/unload operation. Insure the vertical clearance of the Vehicle Deck will accommodate all vehicle heights identified in the military vehicle payload.

The PacifiCat clear deck height is 14 feet in the truck lanes. JJMA participated in studies of the feasibility of increasing the truck capacity of the vessels and has data on all of the vehicle deck obstructions that might limit loading and unloading.

3.5 DETAILS OF SELECTED VESSEL

BC Ferry Corporation provided the following specifications for the PacifiCat vessels:

PACIFICAT CLASS - VESSEL SPECIFICATION

"PacifiCat Explorer", "PacifiCat Discovery", "HSF 003"
122 METER HIGH-SPEED PASSENGER / VEHICLE FERRIES
(minor differences in outfit occur between ships)

3.5.1 General Specifications

Builder	Catamaran Ferries International Inc.
Year Completed	"Explorer "– June 1999, "Discovery" – December 1999, "Voyager" – completed and mothballed August 2000
Owner	British Columbia Ferry Corporation.
Designer	INCAT Designs Sydney, Australia in collaboration with Robert Allan Ltd. of Vancouver, Canada
Length Overall	122.50 m
Length Waterline	96.00m
Beam	25.80m
Beam of Hulls	6.00m

Draft	3.76m (approx.) in salt water
Certification	Transport Canada Marine Safety, in accordance with 1994 IMO International Code of Safety for High Speed Craft (HSC), July 1993 (MSC 63/23 Addendum 2, HSC) for voyages in Canadian Home Trade III waters DNV +1A1 HSLCR4 (enclosed waters 20,20) (Can) Car ferry A EO HSC Category B Craft International Tonnage Certificate Compass installation and adjustment International Load Line Certificate (ILLC) Health and Welfare Canada; Potable Water Standard Ship Station Radio Certificate Can/CSA-B44-94. Safety Code for Elevators

Propulsion Power	26,000 kW
Electrical Power	4 x 190 kW
Passengers & Crew	1000
Total Vehicles	250 cars (1.5 tonne each) or 4 buses (22.4 tonnes each) and 200 cars
Gross Tonnage	9,022 tonnes
Displacement Tonnage	1885 tonnes
Dead Weight Tonnage	518 tonnes
Fuel Oil	66 tonnes
Water	5800 litres
Lube Oil Storage Tanks	1800 litres
Speed	34 knots at 518 tonnes dead-weight at 100% MCR

3.5.2 Structural

Design	Two slender aluminum hulls connected by a strong bridging structure consisting primarily of major transverse web frames and 2 longitudinal CVK 's and a series of minor girders.
Fabrication	Welded aluminum construction using 5083 H116 and 5383 H116 plates and 6061-T6 sections. Longitudinal stiffeners supported by transverse web frames and bulkheads.
Subdivision	Each hull is divided into 8 vented, water-tight compartments divided by transverse bulkheads and decks. The bridging structure between the hulls is fully welded to form a separate compartment. Water-tight upper and lower voids are incorporated into the 5 void spaces ahead of the engine room in each hull between frames 20 and 73.
Vehicle Decks	Main vehicle deck and upper vehicle decks are constructed with straight-line camber with the external aft and side decks flat.

Superstructure	Welded or bonded aluminum construction with longitudinal and transverse framing. Passenger accommodations and wheelhouse are supported above the T3 strength deck on anti-vibration mounts.
Axle Loads	Main vehicle deck 7.1 tonnes centre 2 lanes, 3.16 tonnes outboard lanes. Upper vehicle deck 1.375 tonnes.
Vibration	Within DNV guidelines for structural vibration limits for High Speed Light Craft with the vessel fully loaded at service speed.

3.5.3 Accommodation

Interior Outfit	Passenger spaces are finished to a first class commercial standard using lightweight materials complying with all Canadian, DNV and IMO-HSC regulations.
Interior Decks	Floor coverings of heavy-duty carpeting and Amtico (simulated hardwood) vinyl in the passenger lounges, Light-weight Colorflake in the washrooms, Altro 35 vinyl in the food preparation areas, Pirelli rubber flooring in crew spaces, Wooster stairtreads and Bolar gratings on stairway landings. The aluminum vehicle decks are profiled to provide a non-slip surface.
Insulation	The accommodation areas fully insulated with R8 insulation contoured to the ship's structure to avoid condensation traps.
Seats	Café-style light-weight tables and chairs arranged in 4 passenger lounges. Tabletops have a variety of colored vinyl laminates. Chairs are finished with a high quality wear-resistant fabric in a number of different designs.
Windows	Frameless bonded windows, consisting of clear tempered safety glass, are used in both the accommodation areas and the wheelhouse. The glass in the 4 skylights is tinted.
Wall Coverings	Hexlite 110 aluminum core honeycomb panels with decorative vinyl laminates.
Ceilings	The Hydro-Aluminum ceiling system consists of a combination of linear and open grid-systems with white and mirror finishes. Special features include skylights and a sail-cloth ceiling above the observation deck snack bar.
HVAC	Reverse cycle HVAC system capable of maintaining 22°C at 25% RH with an outside temperature of 32°C at 50% RH and maintaining 20°C with an outside temperature of minus 10°C. Passenger space – make-up fans and fans for 'purging' the passenger cabin provide 8 air changes per hour. Vehicle deck ventilation is capable of 10 air changes per hour in navigational mode and 20 air changes per hour in loading mode. Toilet and service space extraction fans provide 30 air changes per hour. Main engine combustion air is drawn directly from outside the hull through 3 stage water separators. The engine room ventilation system is designed to maintain the engine room temperature below

	45°C when the outside temperature is 32°C. Bow Thruster compartment and Voids #5 and #6 have natural intake and mechanical exhaust.
Sewage System	EVAC vacuum collection system complete with salt water flushing, 1,350 litre collection tank and chlorination / dechlorination treatment unit.
Women's Washrooms	The forward washroom includes 6 WCs, 3 Surell hand-basins with large wall mirror and electric hand dryers. The midship washroom includes 5 WC, 4 Surell hand-basins with large wall mirror and electric hand dryers. All vanities have Surell counter tops.
Men's Washrooms	The forward washroom includes 3 WCs, 5 urinals, 3 Surell hand-basins with large wall mirror, and electric hand dryers. The midship washroom includes 3 WCs, 4 urinals, 3 Surell hand-basins with large wall mirror and electric hand dryers. All vanities have Surell counter tops.
Handicapped Toilets	The common-use handicapped toilet is installed at midship on the starboard side of the passenger deck, contains 1 WC with hand rails, emergency call switch, automatic door-opener, wall-mounted hand-basin with lever handle centre set, and paper towel dispenser.
Crew Facilities	The crews' mess is located aft of the wheelhouse on the port side and includes lockers for 21 crew, table with 8 chairs, sideboard with sink, microwave, refrigerator, and coffee machine. The officers' mess is located behind the wheelhouse on the starboard side, and includes: 1 table with 8 chairs, sideboard with sink, coffee machine and fire-locker complete with 2 fire suits.
Ship's Office	The ship's office is located amidships on the starboard side and includes: 3 workstations complete with desks, shelving, filing cabinets, video display control rack, safe and video monitoring of gift shop and arcade.
Shop/Kiosk	A gift shop is located at the centre of the forward end of the main passenger deck. The shop has Amtico simulated hardwood flooring, mirrored ceilings, first-class display shelving, security cameras, exit scanners, and 4 external display cases. Both entrances have vertical security grilles.
First-Aid Room	The first-aid room is located on the port side of the passenger deck behind the wheelhouse and includes a bed, WC and hand-basin.
Other Areas	Children's playroom, video arcade, business/study carrels, engineers' work shop, bicycle racks and pet facilities.
Noise Levels	Noise levels do not exceed 75 dBA in passenger areas, and 65 dBA in the wheelhouse.
Vehicle Access	Vehicle access is via ship-based hydraulically actuated bow and stern ramps working in conjunction with shore-based ramps. Clear width of bow and stern ramps is 5.2m between curbs. Vehicle lane width is a minimum of 2.6m. Main vehicle deck clear height

	underneath structural fire protection is 4.0m. The bus lane length is 60m, and is located port of centre line of vessel. The remaining main vehicle deck lane length is 595m. Upper vehicle deck clear height underneath insulation is 2.1m, and total lane length is 655m.
Passenger Access	There are 3 separate passenger deck accesses from the main vehicle deck; stairwells port, starboard aft and centre forward. A total of 4 shore accesses for walk-on passengers are located on the port and starboard sides of the passenger deck.
Elevator	An elevator is fitted at the centre of the main vehicle deck providing access to each level of the vessel. Wheelchair access is fitted to the elevator platform at the main vehicle deck level.
Paint	Hulls above waterline, topsides and superstructure are primed and top-coated white. Hulls below the waterline are coated with primer and silicone-based non-toxic (fouling release) paint. Exposed interior structure, all concealed areas, void spaces, tank linings, jet rooms and foredeck are not painted.
Signage	Signage to Classification Society's specifications.

3.5.4 Ship Control Systems

Steering/Reverse	The electronic steering, speed and reversing control system is integrated into the main engine and water jet controls and is programmed to automatically control engine speed when the reversing buckets are deployed.
Control	Water jet control is from the wheelhouse centre, port and starboard wing consoles.
Bow Thruster	A diesel engine-driven bow-thruster is fitted at the forward end of the starboard hull to assist manoeuvring during docking.

3.5.5 Stabilisation Systems

Ride Control	A 'Maritime Dynamics' active ride-control system controls the trim-tabs to maximize passenger comfort and propulsion efficiency.
Safety	An emergency trim-tab lifting device is provided at each transom in the event of electrical / hydraulic failure. The trim-tab control system automatically raises the trim-tabs to the full-up position whenever the reversing jet buckets are deployed, in order to prevent deflection of the reverse thrust.

3.5.6 Anchoring, Towing and Berthing

Anchor	1 cast Super High Holding Power (SHHP) balanced anchor complete with crown shackle, weight 1359 KGs, anchor chain 46mm diameter stud link grade NV K3, 22 meters long with minimum breaking strength 1290 kN is installed complete with anchor cable 196m long by 45mm diameter Herzog P-7 with

	HMPE core, minimum breaking strength 1290 kN, specific gravity 1.12. The anchor is stored between the hulls under the aft port side of the main vehicle deck. The cable and anchor chain are stored on a hydraulic anchor windlass and secured in place by a pelican hook.
Towing	Panama-style fairleads and bollards are installed on the aft mooring platforms port and starboard, main vehicle deck forward port and starboard and upper vehicle deck forward port and starboard. The vessel is capable of being towed by both the forward and aft bollards on the main vehicle deck. A towing bridle is fitted.
Berthing	Variable speed hydraulic mooring capstans are installed on the port aft mooring platform and on the port forward upper vehicle deck. Each capstan has a 2.5 tonne capacity at a line speed of 20m/min for berthing. The forward capstan has a 3m long pendant control. 500mm aluminum sponsons run the full length of the hulls to protect the hulls and jets during berthing and turning in port. Two-way radio communication is provided at each mooring and anchoring station.

3.5.7 Fire Safety

Fire Detection	An addressable fire detection system covers all high & moderate risk spaces. An alarm panel is located in the wheelhouse. Toilets, stairway enclosures, and corridors are equipped with automatic smoke detectors and have manually operable call points. Engine rooms have smoke detectors and heat detectors. Water jet spaces have smoke detectors. Vehicle spaces have smoke and heat detectors. The fire detection system is supplemented by a Closed-Circuit Television (CCTV) with cameras located throughout the ship. A split screen monitor and switching arrangements is mounted in the wheelhouse to enable constant, shipwide monitoring.
Structural Fire Protection	A Classification Society approved structural fire protection system is used to protect the aluminum structure in areas of high fire risk.
Closing Devices	Ventilation fire-rated closing devices (fire dampers) are controlled from the wheelhouse, locally and automatically, in the event of fire.
Shut-downs	Emergency shut-down push-buttons, located at the engine room entrances and operable from the wheelhouse, are installed to stop ventilation fans, fuel and lube-oil pumps located in the engine rooms. A similar means of shutting down the accommodation fans and food preparation equipment is provided.
Fire Suppression System	A Hi-Fog sprinkler system provides fire extinguishing, on the passenger and vehicle decks, in the engine rooms and the bow-thruster compartment (with AFF foam) from pumping modules in

	the port and starboard #5 Void spaces below the main vehicle deck. Each automatic pumping module is sized for whole-ship operation and is monitored/controlled from the wheelhouse.
Hydrants	Fire hydrants, distributed throughout the ship, are supplied from the Hi-Fog pumping modules.
General Equipment	Portable fire extinguishers, fire suits and equipment, water-fog applicators, and fire control plans are provided.

3.5.8 Life-saving Appliances and Arrangements

Appliances	SOLAS approved lifejackets complete with light and whistle are provided for 1200 passengers (120% of the compliment) of which 100 are children's lifejackets. Life jackets for all crewmembers are supplied. Lifebuoys, complete with light and smoke signals, flares and pyrotechnics, line-throwing apparatus, and immersion suits are supplied in accordance with SOLAS and IMO-HSC.
Liferafts	8 Life Saving Appliance (LSA) 150-person life rafts, 4 deployable Marine Evacuation System chutes (MES), and 2 rescue boats, complete with launching and retrieval davits, provide for evacuation of 1200 persons. Transport Canada approved.
Arrangements	Safety cards, fire-fighting and escape plans are posted in the control stations and throughout the passenger lounges.
Rescue Boats	2 Zodiac H-472, each with 1 - 40 hp Mercury outboard motor, are provided, one on each side of passenger deck.
Rescue Boat Davits	2 aluminum alloy, SOLAS type 42, 1,000 kg SWL MOB cranes complete with electric hoist with manual back-up are installed.
Communication	A 5-station, all-master, telephone type intercom is installed with points at each MES station, the wheelhouse and elsewhere in the passenger and observation decks.

3.5.9 Machinery

Main Engines	4 MTU 20V1163 TB3 resiliently-mounted marine diesel engines, rated at 6500 kW each.
Water Jets	4 KaMeWa 112 SII steerable, reversible water jets complete with internal thrust bearings.
Transmission	4 Renk ASL 53 gearboxes.
Shafting	Engine output shaftlines: (1 - long shaftline, outboard and, 1 - short shaftline, inboard in each hull,) Geislinger 4 filament-wound carbon-epoxy CFRP shaftlines complete with membrane couplings, shaft-support bearings and bulkhead seals. Gearbox output shaftlines: 4 steel shaftlines complete with flexibox couplings, Hi-Lock fittings muff style, shaft-support bearings and stern-tube seals.

3.5.10 Auxiliary Systems and Ship Services

Cooling System	Engines, generators, reduction gears and hydraulics are cooled by raw water through heat exchangers.
Starting Systems	The main engines are air started 40 bar. Each engine room is equipped with an air compressor and receiver. There is normally a closed crossover between the systems. All generators are electrically started and provided with 'dead start' capability.
Fuel System	2 integral aluminum fuel-oil tanks, one per hull, of 37,135 litres capacity each are located in hull Void #6 port and starboard. Fuel filling stations are located at both ends of the vessel Filling is via a valved filling main sized to allow a flow rate of 2000 ltr/min at 450 Kpa fitted in a save-all with 'Camlock' type connections. Fuel tank gauging software forms part of the control station monitoring system. A remote level indication displayed adjacent to each bunker station with tank bunkering valve controls.
Lube Oil System	2 - 908 litre storage tanks installed at the jet space entrances port and starboard. Each main engine has an independent lube-oil system complete with 350 litre service tank located in the jet spaces.
Exhaust Systems	Main engine and generator exhaust pipes and silencers are resiliently mounted using stainless steel mesh blocks and resilient hangers. Exhaust pipes are thermally insulated. Main engine exhaust pipes are fitted with pyrometers and connections for checking back pressure immediately after the turbocharger outlet.
Fresh Water	1 - 5800 litre HDPE tank is fitted in the starboard Void #5 upper and includes high/low level alarm and level indication in the wheelhouse. There are 2 filling stations, one located forward and the other aft. There are 2 fresh water pump sets, one primary and one standby and 9 on-demand hot water heaters. Water to hand-basins is thermostatically controlled to 40°C. Hot and cold water pipes are polyethylene lined aluminum and fittings are gunmetal bronze. Heat tracing is fitted in areas exposed to potential freezing.
Sewage Treatment	The sewage system consists of 1 - 1,350 litre vacuum collection tank and treatment unit for onboard treatment and discharge overboard. The treatment unit is fitted with a macerator pump, salt water flushing system, chlorination and dechlorination systems.
Waste Oil/Oily Water System	The system consists of 1 - 400 litre aluminum tank, located in each engine room aft complete with an air-operated transfer pump for transfer to the main vehicle deck aft discharge station. Oily water separator not fitted on HSF 003.
Bilge System	Every lower void space is fitted with an electric, 3-phase 240vac submersible bilge pump. The presence of bilge water is indicated in the wheelhouse by means of float-switches in each bilge area.

	Control of the bilge pumps is from the wheelhouse. In bilge areas where oil contamination can occur, pump control is also located at the compartment access. Engine rooms are fitted with two bilge pumps. One spare portable electric bilge pump is provided with flexible hoses and is stowed in the car deck locker.
Anti-Fouling System	Hydrosonic Hull Tender systems on all through-hull raw water systems.
Hydraulic System	A separate hydraulic system for each water jet is installed. Hydraulic power is taken from the PTO pumps and a stand-by electric driven pump normally used for lubrication. One trim-tab power pack is located in each jet space, aft ramp hydraulic system, capstan and anchor windlass take hydraulic power from the trim-tab power packs in the starboard jet room. The forward ramp hydraulic system and forward capstan are powered by an independent power pack located in the bow-thruster compartment. The power packs are single tank systems. An electrically operated, cart-mounted emergency hydraulic power-pack complete with quick disconnects is stowed on the main vehicle deck. The emergency power pack is intended to provide hydraulic power to the ramps, capstans and windlass in the event of a hydraulic failure.

3.5.11 Remote Control, Alarm and Safety Systems

IMACS	An Integrated Machinery Alarm and Monitoring System (IMACS) enables all of the functions of the propulsion, auxiliary and electrical systems to be monitored and controlled from the engineers' console in the wheelhouse.
Closed-Circuit TV	A CCTV allows selected areas of the ship to be visually monitored from the wheelhouse.
Automatic Telephone System	An automatic telephone system links all the control positions on the ship, i.e. the wheelhouse, engine rooms, steward's office, boarding stations, etc.
In-Dock Security System	An in-dock security system is installed that, with a pushbutton/flashing light/siren arrangement, will enable communication between the wheelhouse and the various vehicle and passenger loading stations.
Emergency Sound-Powered Telephone System	An emergency sound-powered telephone system with handsets in the wheelhouse, engine rooms and waterjets spaces ensures.
Video Information	A Video Information System (VIS) is installed, consisting of flat-screen monitors located throughout the passenger areas on which advertising messages, route information and safety announcements can be displayed. The system includes DVD, VCR, and CD, AM/FM tuner.

Intercom	A 5-station, all-master, telephone type intercom with points at each MES station, wheelhouse and elsewhere in the passenger and observation decks is fitted.
Public Address	A ship-wide public address loudspeaker system is installed. Automatic interruption of the entertainment system permits important announcements to be made from any of the automatic telephones on the ship.
Alarm	General alarms clearly audible to all passengers and crew are enunciated through the PA loudspeakers by means of an alarm tone signal activated by a wheelhouse push-button.
Anti-Theft Security System	An anti-theft shop security system is fitted at the entrances to the gift shop

3.5.12 Electrical Installations

Generators	4 - 190 kW (nominal) marine, continuous-rated, self-excited, brushless diesel engine-driven alternator complete with class "F" windings are installed in the engine rooms. The generators have 110% one-hour overload capacity.
Power Distribution	600V, 60 Hz., 3 phase, 3 wire, no neutral.
Switchboards	The 2 - main switchboards, located in each engine room are equipped with 600Vac, 240Vac and 120Vac distribution systems with a power management system. Switchboards can be paralleled or operated as stand alone.
Power Distribution	Power distribution panels located throughout the ship are fed from the main switchboard at 600V, 240V and 120V via engine room transformers.
Essential Power Distribution	Essential services are supplied from distribution panels that are, in turn, fed from emergency and essential load centers. The 240V and 120V load centers receive, via automatic transfer switches, power supply from each switchboard ensuring that even with the loss of one switchboard, power to the essential systems is maintained.
Shore Power	2 - 300 amp, 600V, 60Hz, 3 phase, connections paralleled onto the ship's electric power system.
UPS Power	Battery-maintained Uninterruptible Power Supplies located in the bridge superstructure deliver power at 120VAC 24V/12VDC for the essential equipment IMACS, navigation, communication, engine and waterjet control.
Lighting 120VAC	Lighting in non-passenger areas is fluorescent. In passenger areas the lighting consists of fluorescent fixtures, recessed pot-lights, neon, and special decorative fixtures. Fixtures in engine rooms and vehicle decks are vapor proof to IP56 standards. External areas are illuminated with quartz floodlights controlled from the wheelhouse.
Emergency	30% of the installed lighting fixtures are fed from the emergency distribution system. 10% of the emergency lighting fixtures are

	located at stairways, doors and passageways and have a built-in battery back up which ensures a minimal level of lighting output for a minimum of 4 hours.
Navigation Lights	Navigational lighting fixtures receive normal and emergency power from monitored circuits with status and control from the wheelhouse.
Searchlights	Three searchlights are fitted, two forward-facing on housetop for docking purposes, manual aft facing on aft bulwarks. The forward searchlights are controlled from the wheelhouse consoles.
Cathodic Protection	A microprocessor controlled Impressed Current Cathodic Protection system (ICCP) is installed to protect the water jet tunnels against corrosion. A Sacrificial Anode Monitoring System (SAMS) monitors the condition of sacrificial anodes in the bow thruster tunnel and on the transom by means of a selector switch and Digital Volt Metre (DVM.) All three systems are located in the starboard jet room.

3.5.13 Navigational Equipment

GPS	2 x Northstar 941X, GPS/DGPS
Wind Spd/Dir Speed Log	Walker combined wind speed and direction and speed log monitoring and display
Whistle	Airchime motor-driven piston whistle with console-mounted “at will” and coded-signal control unit.
Radar	2 - navigational radar’s, Raytheon ST Mk 2 ARPA 3425/7XD, interswitch, colour displays, high-performance monitor, and 7 ft high-speed antenna. 2 - docking radar’s, 1 Raytheon Pathfinder ST Mk 2 TM2525/7XU, and 1 /7xD interswitch, colour TM display (medium resolution), 7 ft high speed antennas
ECDIS System	Raytheon Pathfinder ST ECDIS bridge station, 1 - 28 inch colour monitor,
Management System	2 - high-resolution 20 inch monitors complete with a set of electronic charts.
Navigational Sounder	Seachart 3, IMO compliant, 50kHz. echo sounder, digital display, 2 transducers, transducer COS.
Auto Pilot	Anschutz Nautopilot 2010 digital adaptive autopilot, radius and rate of turn control.
Gyrocompass	Anschutz gyrocompass standard 20GM and 3-console mounted bearing repeaters.
Magnetic Compass	Anschutz standard magnetic compass Reflecta Fiberline, binnacle and sonde.
Navtex Receiver	JRC NCR 300A.
Nauto conning system	Anschutz JBS Display and monitoring system – 4 to 20 inch monitors

Night Vision Equipment	Current Corporation Light Enhancement Night Vision Complete with Pan/Tilt camera IR Search Lite 9” Monitor 5” Flat Screen Monitor 2 x H/H Remote Controls

3.5.14 Radio Communications

Radiotelephones	4 - Sailor RT 2048 VHF console mounted radiotelephones.
Receiver	Kenwood R-5000 communications receiver.
Modem, DSC	Sailor RM 2042 DSC modem, console mounted.
EPIRB	Alden, category 1, Satfind 406 Class 1, Emergency Position Indicating Radio Beacon.
SART	2 x Alden Search and Rescue Transponder.
Shipboard Radio	1 Radio Repeater; 6 Antennas; 8 VHF H/H Radios; 1 Base Station VHF in
Repeater	Engineers console.

3.5.15 Wheelhouse Arrangement

Access	Access to the wheelhouse is through the crew or officers’ lounges or by means of doors from the external wings.
Operation	There are two forward-facing seats in front of the centre navigation console located at the forward end of the wheelhouse. The centre console contains all of the required navigation, main engine control and internal and external communication equipment. There are secondary consoles at the extremities of the bridge wings where the necessary controls, indications and communications to navigate the vessel are repeated. The engineers’ console, with seating for two engineers, is located behind the central control console. All controls, indications and communications necessary to monitor all of the ships mechanical systems and the facility to manually override the related automatic systems are located in this console. Also included is a computer workstation with bridge gear diagnostics.
Visibility	Selected wheelhouse windows are equipped with wipers and heaters. There is a built-in window washing system.
Communication	The onboard communication system is operable from the wheelhouse, enabling communication to all machinery, mooring, boarding and passenger spaces. A dedicated in-dock security system providing communication between the wheelhouse and the various loading stations to ensure safe embarkation and debarkation of passengers and vehicles.

3.5.16 Services

Food Prep	The food prep area is outfitted with a dishwasher, dish table, washroom, reach-in refrigerator/freezer, soup kettle, work surfaces, and an area for storage of mobile transport modules.
Servery	A self-service food counter, complete with hot-wells, soup wells, refrigerated sandwich display, warming lamps and heated shelves is located in the servery. A food counter, 2 combo ovens, pizza oven, conveyor toaster, microwave oven, refrigerated prep table, complete with built-in fire suppression systems, are installed.
Coffee Bar	A coffee bar is located at the forward passenger lounge complete with espresso machine, other beverage dispensers and dry displays.
Snack Bar	Another coffee bar is located on the observation deck complete with espresso machines, refrigerator, sandwich display, service counter, beverage dispensers and cash counter.
Open Deck Area	Bench seats are fitted on the open deck aft of the observation deck.

3.5.17 Manuals/Drawings

Manuals/Drawings	ISM & HSC Drawings and manuals are provided in both hard copies and electronic format.

3.6 CONCLUSION

An analysis was performed of the mission and payload requirements for the candidate vessels. From this analysis it was determined that only a vessel over 100m would be a suitable candidate for the conversion. From open literature a list was compiled of eight candidate vessels which could arguably satisfy the requirements. The vessels that best fit the requirements do not yet exist – none have been built. Of the vessels which have been built, for most of them, technical data is not available for use in this project.

Technical data is available and the vessel payload capacity is satisfactory for the British Columbia PacifiCat class ferries. This vessels range falls short of the target for this project. The vessels DNV service area restriction is also more restricting than initially hoped at the outset of this project for the selected conversion vessel. This will increase the work associated with the structural modifications for the craft

In consideration of this balance of facts, the BC Ferry PacifiCat is selected as the conversion vessel for this project.

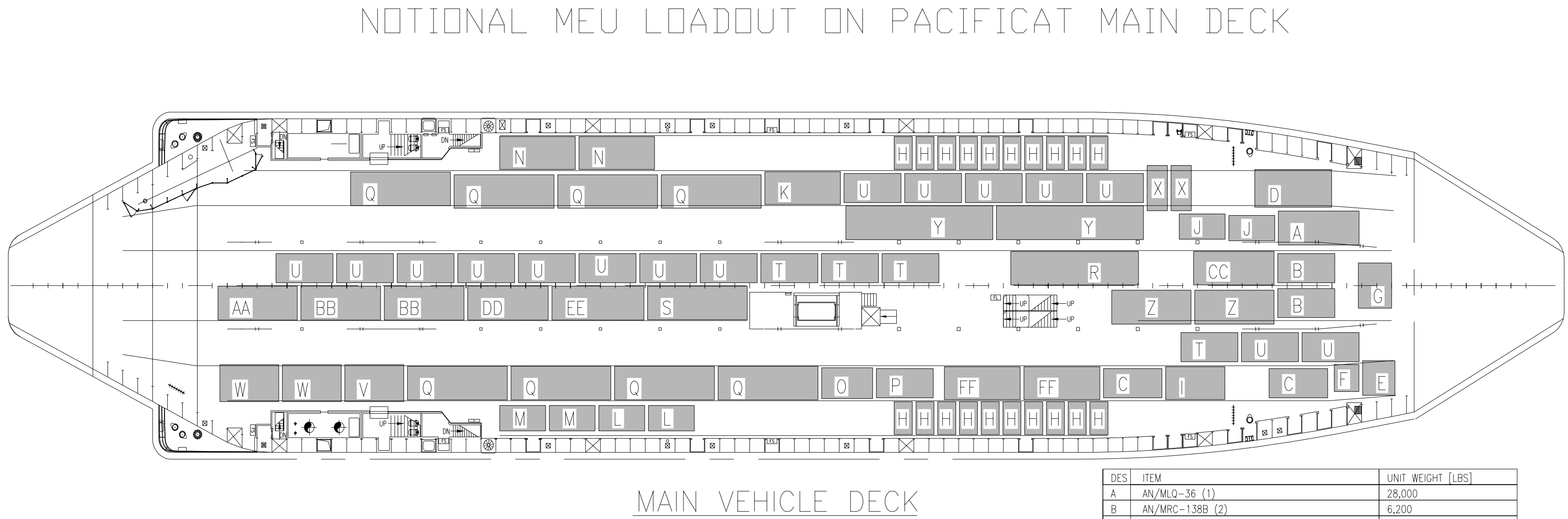
4.0 PRELIMINARY VEHICLE ARRANGEMENT OF MEU LOADOUT

A drawing is provided below in Figure 4-1 that presents the preliminary arrangement drawing of the Military Vehicle Payload identified for this conversion study. The arrangement is shown on the Main Vehicle Deck of the PacifiCat ferries used for the conversion. The PacifiCat ferries include two vehicle decks although there is no onboard ramp to get from one deck to the other. In its commercial application, all vehicle loading of both decks is accomplished at pierside loading facilities that have been specially developed to load both the upper and lower vehicle decks, i.e., there are upper and lower vehicle ramps to access the two vehicle decks. The preliminary vehicle arrangement locates the entire vehicle payload on the Main Vehicle Deck, which is the Main Deck of the vessel.

The preliminary vehicle arrangement does not consider specific structural impacts to the arrangement shown on the drawing. The requirements for the deck/tie down structure are developed in Section 6 of this report.

The notional vehicle load for the conversion vessel (Table 2-1) is provided again as Table 4-1 for easy reference following Figure 4-1, shown below. It is based on the LHA(R) load, which is developed from the USMC MEU load.

The purpose of this arrangement drawing is to show that the vehicle load out detailed in Table 2-1 will fit into the Main Vehicle Deck of the candidate vessel. The Upper Vehicle Deck is not required to accommodate the load out for this project. The vehicle arrangement includes a 12-inch (30 cm) gap between all vehicles. This was later confirmed as an acceptable gap using reference [3], which recommends that a 10-inch gap be assumed for all vehicle arrangements.



DES	ITEM	UNIT WEIGHT [LBS]
A	AN/MLQ-36 (1)	28,000
B	AN/MRC-138B (2)	6,200
C	AN/MRC-145 (2)	6,200
D	ARMORED COMBAT EXCAVATOR (1)	36,000
E	TRK, FORKLIFT (1)	25,600
F	TRK, FORKLIFT 4K (1)	11,080
G	TRAM (1)	35,465
H	CONTAINER, QUADRUPLE (20)	5,000
I	CHASSIS TRAILER, GENERAL PURPOSE M353 (1)	2,720
J	CHASSIS TRAILER, 3/4T (2)	1,840
K	POWER UNIT, FRT (LVS) MK-48 (1)	25,300
L	TRLR CARGO 3/4T (M101A1) (2)	1,850
M	TRAILER CARGO M105 (2)	6,500
N	TRLR, MK 14 (2)	16,000
O	TRLR TANK WATER 400 GL (1)	2,530
P	TRK AMB 2 LITTER (1)	6,000
Q	TRK 7-T MTVR (8)	36,000
R	TRK 7-T M927 EXTENDED BED (1)	37,000
S	TRK 7-T DUMP (1)	31,888
T	TRK TOW CARRIER HMMWV (4)	7,200
U	TRK, MULTI-PURPOSE M998 (15)	6,500
V	TRK, AVENGER/CLAWS (1)	7,200
W	TRK ARMT CARR (2)	7,000
X	TRK LIGHT STRIKE VEHICLE (2)	4,500
Y	155MM HOWITZER (2)	9,000
Z	LAV ANTI TANK (AT) (2)	24,850
AA	LAV C2 (1)	26,180
BB	LAV ASSAULT 25MM (2)	24,040
CC	LAV LOGISTICS (1)	28,200
DD	LAV, MORTAR CAR (1)	23,300
EE	LAV, MAINT RECOV (1)	28,400
FF	MAINT VAN (2)	10,000

Figure 4-1. Preliminary Notional Loadout for MEU on Main Vehicle Deck of PacifiCat

Table 4-1. Notional Vehicle Load for Conversion Vessel

NOMENCLATURE	LHA(R) QTY	Conv QTY	LN	WD	HT	WT	Area (SQ FT)	TOTAL WT (LBS)
AN/MLQ-36	1	1	255	99	126	28,000	175	28,000
AN/MRC-138B	8	2	180	85	85	6,200	213	12,400
AN/MRC-145	8	2	185	85	83	6,200	218	12,400
ARMORED COMBAT EXCA	2	1	243	110	96	36,000	186	36,000
TRK, FORKLIFT	1	1	315	102	101	25,600	223	25,600
TRK, FORKLIFT, 4K	1	1	196	78	79	11,080	106	11,080
TRAM	1	1	308	105	132	35,465	225	35,465
CONTAINER, QUADRUPLE	70	20	57	96	82	5,000	760	100,000
CHASSIS TRLR GEN PUR, M353	1	1	187	96	48	2,720	125	2,720
CHASSIS, TRAILER 3/4 T	2	2	147	85	35	1,840	174	3,680
POWER UNIT, FRT (LVS) MK-48	2	1	239	96	102	25,300	159	25,300
TRLR CARGO 3/4 T (M101A1)	4	2	145	74	50	1,850	149	3,700
TRAILER CARGO M105	6	2	185	98	72	6,500	252	13,000
TRLR, MK-14	2	2	239	96	146	16,000	319	32,000
TRLR TANK WATER 400 GL	2	1	161	90	77	2,530	101	2,530
TRK AMB 2 LITTER	1	1	180	85	73	6,000	106	6,000
TRK 7-T MTVR	24	8	316	98	116	36,000	1,720	288,000
TRK 7-T M927 EXTENDED BED	1	1	404	98	116	37,000	275	37,000
TRK 7-T DUMP	1	1	315	98	116	31,888	214	31,888
TRK TOW CARRIER HMMWV	8	4	180	85	69	7,200	425	28,800
TRK, MULTI-PURPOSE M998	45	15	180	85	69	6,500	1,594	97,500
TRK,AVENGER/CLAWS	3	1	186	108	72	7,200	140	7,200
TRK ARMT CARR	10	2	186	108	72	7,000	279	14,000
TRK LIGHT STRIKE VEHICLE	6	2	64	132	74	4,500	117	9,000
155MM HOWITZER	6	2	465	99	115	9,000	639	18,000
LAV ANTI TANK (AT)	2	2	251	99	123	24,850	345	49,700
LAV C2	1	1	254	99	105	26,180	175	26,180
LAV ASSAULT 25MM	4	2	252	99	106	24,040	347	48,080
LAV LOGISTICS (L)	3	1	255	98	109	28,200	174	28,200
LAV, MORTAR CAR	2	1	255	99	95	23,300	175	23,300
LAV, MAINT RECOV	1	1	291	99	112	28,400	200	28,400
TANK COMBAT M1A1	4	-	387	144	114	135,000	-	-
MAINT VAN	4	2	240	96	96	10,000	320	20,000
RECOVERY VEH, M88	1	-	339	144	117	139,600	-	-
Totals:	**238**	**87**					**10,629**	**1,105,123**
Total Stons								553

5.0 STRUCTURAL LOADS REQUIRED FOR THE CONVERSION

5.1 BACKGROUND

One of the primary objectives for this project is the definition of the loads that need to be resisted for the converted vessel in order for it to operate successfully in open-ocean, unrestricted service with the military vehicle payload identified above. The structural loads presented in this report include the following three categories:

1. Primary hull girder loads,
2. Secondary slam loads acting along the shell of the ship and,
3. Tire footprint loads of the military vehicles on the Vehicle Deck structure of the ship.

The primary hull girder and secondary slam loads change from the original design loads for the vessel due to the requirement that the converted vessel be capable of unrestricted, open-ocean operation. As designed and built, the PacifiCat vessels studied for this project have an **R4** service restriction in accordance with the DNV rules to which they were designed.

The loads for the first two categories have been developed using the ABS Rules For Building and Classing High Speed Naval Craft, 2003 [1] and the DNV Rules for Classification of High Speed, Light Craft and Naval Surface Craft [2]. The global and local slam loads were calculated for DNV using both **R1** service restrictions and unrestricted notations. Since ABS does not have notations that strictly parallel these definitions, the ABS loads were calculated for conditions corresponding to Naval Craft (Unrestricted notation) and Coastal Naval Craft (taken to approximate **R1** service restriction).

The third set of loads is independent of the rules used for the conversion although the accelerations predicted by the different rule sets does factor into the final design of the ship structure and tie-downs required to accommodate and secure the military vehicles.

Also critical for the conversion are the materials used for the fabrication of the vessel. There are two principal aluminum alloys used for this vessel:

1. 5083-H321 (or H116) for the hull girder plating of the vessel
2. 6082-T6 (or 6061-T6) for all extrusions

There are two vehicles decks on the PacifiCat. All of the Upper and Main Vehicle Deck structure is fabricated from extrusions and the properties for these materials will be taken from the appropriate ABS and DNV rules.

This report also presents preliminary estimates for the effect to the Vehicle Deck plate and longitudinals resulting from the vehicle loads. The procedures used in this report indicate the baseline for future calculations, which will also be supplemented with detailed analysis for specific areas of design and loading geometry. No analysis for the Vehicle Deck transverse structure is included in this report. Its presence is reflected as boundary conditions in FEA work developed to analyze the vehicle deck plate and stiffening later in the project.

5.2 ABS & DNV RULES FOR HIGH-SPEED VESSEL DEFINITION

The first objective for this section of the report is to confirm that the vessel being converted satisfies the definition for high-speed. Both ABS and DNV have their own definitions for what constitutes a high-speed vessel. Discussion on that and the ABS/DNV Rules required for this project are presented below.

There are three sets of rules that were referenced for this project. ABS has two sets of rules, one each for High Speed Craft [1] and High Speed Naval Craft [2]. All DNV criteria are contained within a single set of rules with different sections addressing naval and commercial vessels. These three rule sets are:

1. ABS Rules for Building and Classing High Speed Naval Craft, 2003 [1]
2. DNV Rules for Classification of High Speed, Light Craft and Naval Surface Craft [2].
3. ABS Guide for Building and Classing High Speed Craft, February 1997 [4]

Since it is the objective of this project to determine the structural modifications to accommodate a military vehicle payload the assumption was made that the conversions should be developed in accordance with the naval vessel criteria of each respective society, i.e., references [1] and [2].

Reference [4] is for information only. It preceded the development of [1] and forms the basis for many of the load predictions and scantling calculations presented in [1]. All global and local loads and associated scantlings for this conversion study were determined from [1] and [2].

In accordance with Objective 1.1 and Task 3.2.3 of the Statement of Work for this project, it is necessary to determine the structural load and scantling requirements for conversion of the selected ship for “unrestricted, open-ocean operation.” This terminology reflects DNV practice, which refers to “service restrictions” for all high speed craft. Discussion is presented below regarding the different philosophies behind the ABS and DNV approaches to the design of high speed craft.

5.2.1 Definitions – ABS & DNV High Speed Craft

ABS and DNV each have their own definition that determines whether a craft is high speed. The definition for ABS is contained in their Rules for High Speed Naval Craft [1] and their High Speed Craft Rules [4]. The DNV high speed criteria are taken from the DNV Rules for Classification of High Speed, Light Craft and Naval Surface Craft [2].

5.2.2 ABS High Speed Naval Craft

In accordance with ABS HSNC Part 1, Section 1, paragraph 4/1 [1], a vessel will be considered high speed when:

$$\frac{V}{\sqrt{L}} \geq 2.36$$

Where:

V = speed in knots = 37 knots for PacifiCat
L = length in meters = 96 meters for PacifiCat.

Using these values results in a ratio of 3.78 which is greater than 2.36, resulting in a craft that is high speed in accordance with the ABS criteria.

5.2.3 DNV High Speed, Light Craft

In accordance with DNV Part 1, Chapter 1, Section 2/105 [2] a high speed craft is defined as a craft capable of a maximum speed, in knots, not less than:

$$V = 7.16\Delta^{0.1667}$$

Where:

Δ = Displacement of the vessel, tonnes

Using Δ = 1885 tonnes results in a speed of V = 25.2 knots. The PacifiCat ferries have a maximum speed of 37 knots.

Therefore, based on the definitions provided in the DNV rules, the PacifiCat ferries qualify as high speed and, if certification of the conversion were desired, the notation for the vessel would include HS, assuming all design and fabrication requirements were satisfied.

When considering ABS and DNV the Light Craft notation is unique to DNV. The ability to be classed as a Light Craft is not a requirement for this conversion study. The calculation is included in this report as a matter of completeness.

In accordance with DNV Part 1, Chapter 1, Section 2/103 [2] a light craft is defined as a craft with a full load displacement that does not exceed:

$$\Delta = (0.13LB)^{1.5}$$

Where:

Δ = Displacement of the vessel, tones
L = Length of the vessel, meters = 96 meters for PacifiCat
B = Beam of the vessel, meters = 25.80 meters for PacifiCat

For the PacifiCat vessels selected for conversion, using the values indicated above results in a displacement that cannot exceed 5777.6 tonnes, which greatly exceeds the full load displacement of 1885 tonnes used for the original design of the vessel. The converted vessel will have a full load displacement similar to the original value, well within the limitation to define the vessel as Light Craft.

5.3 ABS STRUCTURAL LOADS

The following section will determine the primary hull girder and secondary slam loads that need to be considered for the conversion of the vessel.

In their Rules for High Speed Naval Craft, ABS defines three different types of naval craft:

- Naval Craft
- Coastal Naval Craft
- Riverine Naval Craft

Of these, only the Naval Craft can be operated in an unrestricted, open-ocean environment. The operating scenario defined for each of these craft is defined and shown in Figure 5-1, taken from ABS Part 1, Chapter 1, Section 3, Table B [1]:

TABLE B
Classification Type

TYPE	*DESCRIPTION*	*REFERENCE*
HSC	Indicates that the vessel complies with this Guide and the limits established in Section 1-1-4	HSNC
Naval Craft	This notation is to be assigned to a naval vessel that is intended to operate in the littoral environment, but is capable of open ocean voyages. Naval Craft are limited to a maximum voyage of 300 miles from a safe harbor when operating in the Winter Seasonal Zones as indicated in Annex II of the International Conference on Load Lines, 1996. When operating on an open ocean voyage, craft are to avoid tropical cyclones and other severe weather events.	HSNC
Coastal Naval Craft	This notation is to be assigned to a naval vessel that is intended to operate on a coastal voyage with a maximum distance from safe harbor of 300 miles and a maximum voyage of 150 miles from a safe harbor when operating in the Winter Seasonal Zones as indicated in Annex II of the International Conference of Load Lines, 1996. Coastal Naval Craft are not permitted to perform transoceanic movements.	HSNC
Riverine Naval Craft	This notation is to be assigned to a naval vessel that is intended to operate in rivers, harbors, and coast lines with a maximum distance from safe harbor of 50 miles. Riverine Naval Craft are not permitted to perform transoceanic movements.	HSNC
HSNC:	Guide for Building and Classing High Speed Naval Craft	

Figure 5-1 ABS Classification Type from ABS 1-1-3/Table B

Part 1, Chapter 1, Section 4/3 of the Rules [1] also requires that the structural design for all Naval Craft include a Direct Analysis, which is far more extensive then intended for this SSC conversion project. The Direct Analysis would be required for all new construction vessels and it is assumed that the objectives of this project can be satisfied using the rule-book approach for the structural design available through Part 3 of the Rules.

5.3.1 Use of the ABS Coastal Naval Craft Notation for R1 Equivalency

The original intent for this project was to use a candidate vessel for the conversion that had DNV notation of **R1** or greater. As a result of only being able to have satisfactory access to an **R4** vessel it was decided to slightly refine the task. Instead of taxing the converted **R4** vessel with the full structural weight required to convert from an **R4** notation to an unrestricted notation it was decided to stagger the weight impact from **R4** to **R1** and from **R1** to unrestricted. The reduction to the payload capacity resulting from structural modifications would only reflect the increase from **R1** to unrestricted.

Since the **R** notation is used by DNV and not ABS, it was decided to equate DNV **R1** to the ABS Coastal Naval Craft and equate the DNV **R4** to **R1** conversion to the ABS existing vessel to Coastal Naval Craft criteria. While this is not an exact match, it is pretty close and considered acceptable for use in this project. In accordance with DNV, an **R1** craft can operate up to 300 nautical miles from safe harbor in Summer and Tropical conditions and not more than 100 nautical miles in Winter conditions. As noted in Figure 5-1, the ABS Coastal Naval Craft can operate up to 300 miles from safe harbor on a coastal voyage and no more than 150 miles from safe harbor when operating in a Winter Seasonal Zone. Note that DNV cites "nautical miles" and ABS "miles" as the unit of measure from safe harbor. Again, it was felt that these definitions were close enough for the purposes of this project to allow for the equating of **R1** and Coastal Naval Craft.

Therefore, all intermediary loads for ABS are based on their Coastal Naval Craft requirements.

5.3.2 ABS Global Loads

Similar to other rules published by ABS, the rules for High-Speed Naval Craft includes a series of design algorithms to develop the design moments to be resisted by the hull girder. The HSNC rules also include a design algorithm to predict the slamming induced bending moment. The still water and wave induced hogging and sagging moments do not include any variables that account for the high-speed nature of the craft, i.e., they do not include any variables that are a function of speed or vertical acceleration. The slamming moment expression does include the variable n_{cg}, the vertical acceleration of the craft.

The ABS HSNC Rules presents the following equations for the calculations of the global loads acting on a twin hull vessel:

- *Wave Induced Bending Moment Amidships:* (ABS 3-2-1/3.1)

  $M_{ws} = -k_1 C_1 L^2 B(C_b+0.7)x10^{-3}$ *Sagging Moment*

  $M_{wh} = +k_2 C_1 L^2 B C_b x10^{-3}$ *Hogging Moment*

- *Still Water Bending Moment:* (ABS 3-2-1/3.1)

  $M_{sws} = 0$ *Sagging Moment*

  $M_{swh} = 0.375 f_p C_1 C_2 L^2 B(C_b+0.7)$ *Hogging Moment*

- *Slamming Induced Bending Moment:* (ABS 3-2-1/1.1.2(d))

  $M_{sl} = C_3 \Delta(1+n_{cg})(L-l_s)$

- *Design Transverse Bending Moment:* (ABS 3-2-1/3.3)

  $M_{tb} = K_1 \Delta B_{cl}(1+n_{cg})$

- *Design Torsional Moment:* (ABS 3-2-1/3.3)

  $M_{tt} = K_2 \Delta L(1+n_{cg})$

After defining all of these components for the global loading on the vessel, ABS also includes definitions for the combined, total moments acting on the hull girder that need to be considered for the structural design of the vessel. These combined moments are:

$M_t =$	$M_{swh} + M_{wh}$	*Total Hogging Moment*	(ABS 3-2-1/1.1.2(e))
	$-M_{sws} - M_{ws}$	*Total Sagging Moment*	
	M_{sl}	*Slam Induced Bending Moment*	

As noted by the equations presented above, neither the still water nor the wave induced vertical moments include any terms for the speed or vertical acceleration of the craft, unlike all of the other moment expressions included by ABS. This implies that the values of these global moment components is independent of craft type and suggests that the vertical global bending moments will be the same for a given vessel regardless of whether it is to be defined as Naval Craft or Coastal Naval Craft. This is reflected in the moment calculations summarized below. All of the other moments include the consideration for the vertical acceleration of the craft and produce different values for slam induced, transverse bending and torsional moments acting on the vessel.

Table 5-1 and Table 5-2 summarize the global loads acting on the hull girder of the PacifiCat using the ABS High Speed Naval Craft Rules [1]. These two tables present the same results in Metric and English Units.

In no case do the survival loads exceed the operational loads. While this may seem counterintuitive it is easy to explain when the variables affected by these definitions are reviewed. As shown below in Figure 5-2, the two variables defined for the Operational and Survival conditions are Design Significant Wave Height, $h_{1/3}$, and Speed, V. The change in these values and their effect on the structural design loads is reflected in the value for the design vertical acceleration, n_{cg}, which then has a direct bearing on the global moment values calculated above. For the current situation involving the notations of Naval Craft and Coastal Naval Craft with a design speed of 37 knots, the following impacts to the design equations are noted. Also, as shown below, the value for n_{cg} varies linearly with $h_{1/3}$ while varying with the square of V. Therefore, for Naval Craft changing from operational to survival would see an increase of n_{cg} as a function of $h_{1/3}$ equal to (6/4) = 1.5 while for Coastal Naval Craft the effect of $h_{1/3}$ is also seen to increase n_{cg} by the ratio of (4/2.5) = 1.6. However, both of these increases are strongly overshadowed by the reducing effect resulting from the decrease in speed from 37 knots to 10 knots. The speed reduction would serve to reduce n_{cg} for both Naval Craft and Coastal Naval Craft by the ratio of $(10/37)^2 = 0.073$.

Table 5-1. ABS Global Hull Girder Loads for Naval & Coastal Naval Notations – Metric Units

	Max Vertical Moment* kN - m	SW Hogging kN - m	WI Hogging kN - m	SW Sagging kN - m	WI Sagging kN - m	Slam Induced kN - m	Transverse kN - m	Torsional kN - m
Naval, Operational	161,123	68,966	92,157	0	-115,601	101,539	127,636	273,143
Naval, Survival	161,123	68,966	92,157	0	-115,601	83,524	104,991	224,682
Coastal, Operational	161,123	68,966	92,157	0	-115,601	95,807	120,430	257,723
Coastal, Survival	161,123	68,966	92,157	0	-115,601	83,524	104,991	224,682

Note: * The Maximum Vertical Moment is the worst-case absolute value moment comparing |SW Hogging + WI Hogging| and |SW Sagging + WI Sagging| where SW = Still Water and WI = Wave Induced.

Table 5-2. ABS Global Hull Girder Loads for Naval & Coastal Naval Notations – English Units

	Max Vertical Moment* Lton-ft	SW Hogging Lton-ft	WI Hogging Lton-ft	SW Sagging Lton-ft	WI Sagging Lton-ft	Slam Induced Lton-ft	Transverse Lton-ft	Torsional Lton-ft
Naval, Operational	53,052.8	22,708.3	30,344.4	0.0	-38,063.8	33,433.6	42,026.5	89,937.5
Naval, Survival	53,052.8	22,708.3	30,344.4	0.0	-38,063.8	27,501.8	34,570.3	73,980.8
Coastal, Operational	53,052.8	22,708.3	30,344.4	0.0	-38,063.8	31,546.3	39,653.8	84,860.1
Coastal, Survival	53,052.8	22,708.3	30,344.4	0.0	-38,063.8	27,501.8	34,570.3	73,980.8

Note: * The Maximum Vertical Moment is the worst-case absolute value moment comparing |SW Hogging + WI Hogging| and |SW Sagging + WI Sagging| where SW = Still Water and WI = Wave Induced.

5.3.3 ABS Secondary Slam Loads

For a given ship and design speed, the most important variable in consideration of secondary slam loads is the vertical acceleration at the center of gravity, n_{cg}. This is a critical variable for a number of the global loads shown above as well as all secondary slam loads. For preliminary design, this variable is presented in ABS 3-2-2/1.1 [1] and is calculated as:

$$n_{cg} = N_2 \left[\frac{12h_{1/3}}{N_h B_w} + 1.0 \right] \tau \left[50 - \beta_{cg} \right] \frac{V^2 (N_h B_w)^2}{\Delta}$$

All variables are defined in the List of Symbols included in the Front Matter of this report. For quick reference, it is worth repeating that $h_{1/3}$ is the Design Significant Wave Height and V is the speed of the vessel in knots. The values for n_{cg} are developed using the input from ABS 3-2-2/Table 1 [1] which defines criteria for Operational and Survival conditions for the various craft notations. This table is repeated below as Figure 5-2:

	Operational Condition		***Survival Condition***	
	$h_{1/3}$	***V***	***$h_{1/3}$***	***V***
Naval Craft	4 m (13 ft)	V_m [2]	6 m (20 ft) [1]	10 knots [3]
Coastal Naval Craft	2.5 m (8.5 ft)	V_m [2]	4 m (13 ft)	10 knots [3]
Riverine Naval Craft	0.5 m (1.75 ft)	V_m [2]	1.25 m (4 ft)	10 knots [3]

Notes 1. Not to be taken less than L/12
2. V_m = maximum speed for the craft in the design condition specified in 3-2-2/1
3. This speed is to be verified by the Naval Administration

Figure 5-2. Design Significant Wave Heights, $h_{1/3}$, and Speeds, V from ABS 3-2-1/Table 1

Using the input from this table and the associated vessel particulars, results in the following values for n_{cg} for this design:

Naval Craft, Operational ⇒ n_{cg} = 0.24 g's
Naval Craft, Survival ⇒ n_{cg} = 0.02 g's
Coastal Naval Craft, Operational ⇒ n_{cg} = 0.17 g's
Coastal Naval Craft, Survival ⇒ n_{cg} = 0.02 g's

ABS 3-2-2/3.1, 3.3 and 3.5 [1] provides equations for the determination of Bottom, Side and Transom, and Wet Deck Slamming, respectively. These equations are given as:

- 3.1.1 Bottom Slamming Pressure: (ABS 3-2-2/3.1)

$$p_{bxx} = \frac{N_1 \Delta}{L_w N_h B_w} \left[1 + n_{xx} \right] \left[\frac{70 - \beta_{bx}}{70 - \beta_{cg}} \right] F_d$$

- 3.3.1 Side and Transom Slamming Pressure: (ABS 3-2-2/3.3)

$$p_{sxx} = \frac{N_1 \Delta}{L_w N_h B_w}\left[1 + n_{xx}\right]\left[\frac{70 - \beta_{sx}}{70 - \beta_{cg}}\right] F_d$$

- 3.5 Wet Deck or Cross Structure Slamming Pressure: (ABS 3-2-2/3.5)

$$p_{wd} = 30 N_1 F_D F_I V V_I \left(1 - 0.85 h_a / h_{1/3}\right)$$

In all these equations, n_{xx}, is the vertical acceleration at section x – x of the ship which can be determined from the relationship:

$$n_{xx} = n_{cg} K_v$$

Where K_v is the vertical acceleration distribution factor from ABS 3-2-2/Figure 7 [1], shown in Figure 5-3:

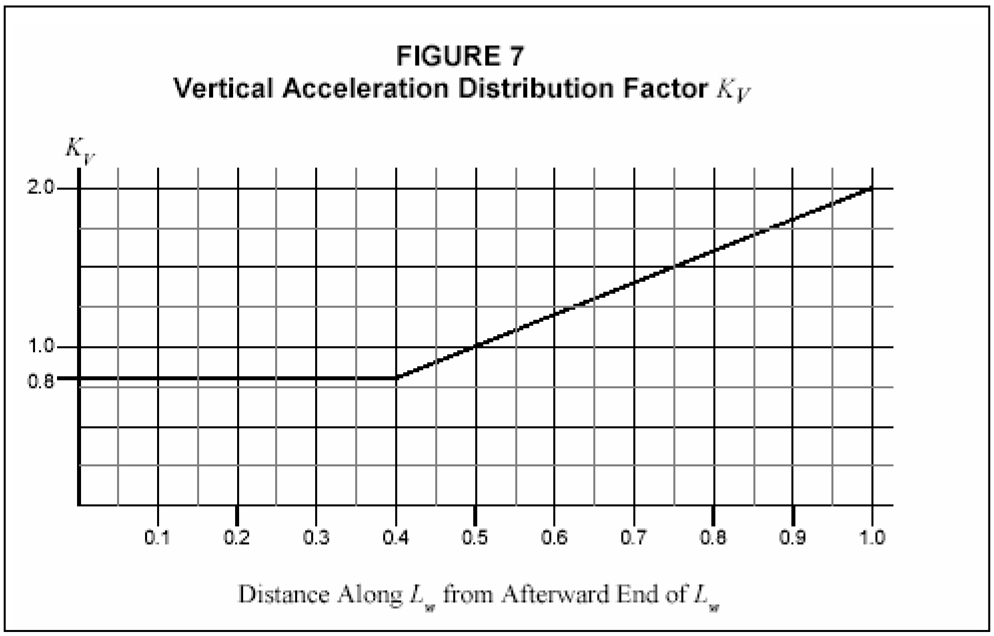


Figure 5-3. ABS Vertical Acceleration Distribution Factor K_V, from ABS 3-2-2/Figure 7

F_I, shown above, is the Wet Deck pressure distribution factor and is obtained from ABS 3-2-2/Figure 9 [1] and shown below in Figure 5-4:

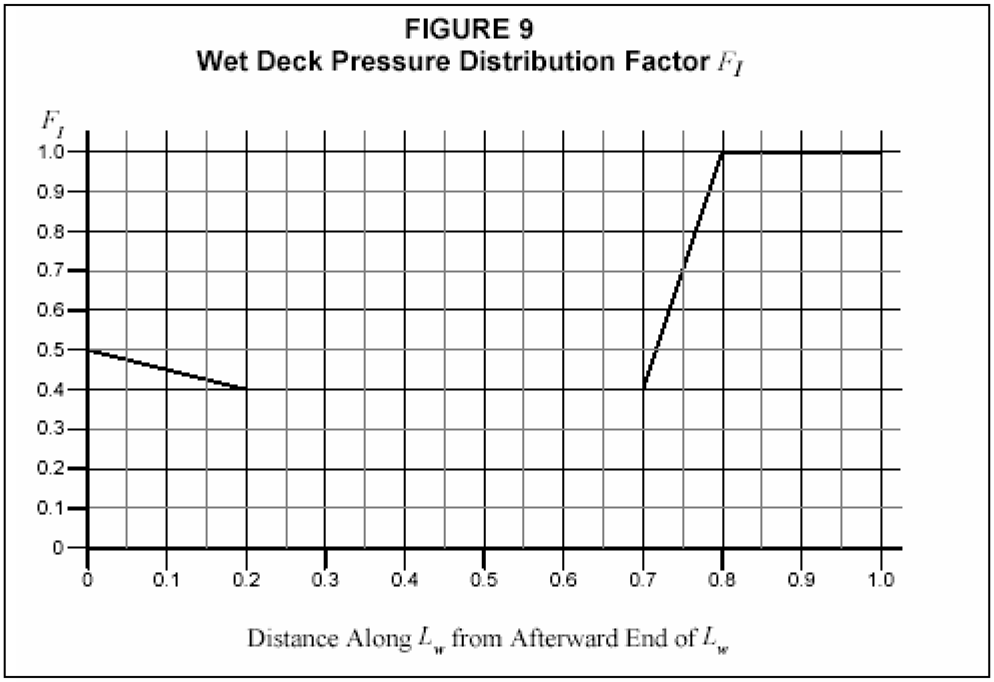


Figure 5-4. ABS Wet Deck Pressure Distribution Factor F_I, from ABS 3-2-2/Figure 9

Based on all the input and equation information provided above, Table 5-4 provides the ABS slam load data for the Naval Craft in Operational and Survival conditions in Metric units. Table 5-5 provides the slam load data for the Coastal Naval Craft in Operational and Survival conditions in Metric units. Table 5-6 and Table 5-7 present the same information using English units of measure.

The same effect to the secondary slam loads is realized as the global bending moments, i.e., the survival loads are lower than the operational loads. The explanation is the same as that provided above.

<u>NOTE TO READERS</u>

The correlation between the "Location" references throughout the rest of this report and actual location along the ship length is shown below in Table 5-3.

Table 5-3. Correlation of Table Locations & Ship Frames

Location	Frame Number	Location	Frame Number
1	88 (FP)	7	29.2
2	78.2	8	19.4
3	68.4	9	9.6
4	58.6	10	-0.2 (AP)
5	48.8	11	-10.0
6	39.0		

Table 5-4. ABS Secondary Slam Load Pressures for Naval Craft Operational & Survival Conditions - Metric Units

All pressures are kN/m^2

	Naval Craft Operational			Naval Craft Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	243.3	94.05	138.3	171.0	66.09	67.6
2	235.4	91.00	138.3	170.3	65.84	67.6
3	227.5	87.95	138.3	169.6	65.58	67.6
4	219.6	84.90	55.3	169.0	65.33	27.0
5	211.7	81.85	55.3	168.3	65.07	27.0
6	203.8	78.80	55.3	167.7	64.82	27.0
7	195.9	75.75	55.3	167.0	64.56	27.0
8	226.3	75.75	55.3	192.9	64.56	27.0
9	265.0	75.75	55.3	225.9	64.56	27.0
10	293.9	75.75	62.2	250.5	64.56	30.4
11	293.9	75.75	69.2	250.5	64.56	33.8

Table 5-5. ABS Secondary Slam Load Pressures for Coastal Naval Craft Operational & Survival Conditions - Metric Units

All pressures are kN/m^2

	Coastal Naval Craft Operational			Coastal Naval Craft Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	220.3	85.15	33.9	171.0	66.09	37.4
2	214.7	82.99	33.9	170.3	65.84	37.4
3	209.1	80.83	33.9	169.6	65.58	37.4
4	203.5	78.67	13.6*	169.0	65.33	15.0
5	197.9	76.51	13.6*	168.3	65.07	15.0
6	192.3	74.35	13.6*	167.7	64.82	15.0
7	186.7	72.19	13.6*	167.0	64.56	15.0
8	215.7	72.19	13.6*	192.9	64.56	15.0
9	252.6	72.19	13.6*	225.9	64.56	15.0
10	280.1	72.19	15.3*	250.5	64.56	16.8
11	280.1	72.19	17.0*	250.5	64.54	18.7

*These slam loads are less than other design minimum loads that are typically applied to shell structure, i.e., 500 psf = 3.5 psi = 25.8 kN/m^2 for the design of shell plate and stiffening.

Table 5-6. ABS Secondary Slam Load Pressures for Naval Craft Operational & Survival Conditions - English Units

All pressures are PSI

	Naval Craft Operational			Naval Craft Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	35.3	13.64	20.1	24.8	9.58	9.8
2	34.1	13.19	20.1	24.7	9.55	9.8
3	33.0	12.75	20.1	24.6	9.51	9.8
4	31.8	12.31	8.0	24.5	9.47	3.9
5	30.7	11.87	8.0	24.4	9.44	3.9
6	29.6	11.43	8.0	24.3	9.40	3.9
7	28.4	10.98	8.0	24.2	9.36	3.9
8	32.8	10.98	8.0	28.0	9.36	3.9
9	38.4	10.98	8.0	32.8	9.36	3.9
10	42.6	10.98	9.0	36.3	9.36	4.4
11	42.6	10.98	10.0	36.3	9.36	4.9

Table 5-7. ABS Secondary Slam Load Pressures for Coastal Naval Craft Operational & Survival Conditions - English Units

All pressures are PSI

	Coastal Naval Craft Operational			Coastal Naval Craft Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	31.9	12.35	4.9	24.8	9.58	5.4
2	31.1	12.03	4.9	24.7	9.55	5.4
3	30.3	11.72	4.9	24.6	9.51	5.4
4	29.5	11.41	2.0*	24.5	9.47	2.2*
5	28.7	11.09	2.0*	24.4	9.44	2.2*
6	27.9	10.78	2.0*	24.3	9.40	2.2*
7	27.1	10.47	2.0*	24.2	9.36	2.2*
8	31.3	10.47	2.0*	28.0	9.36	2.2*
9	36.6	10.47	2.0*	32.8	9.36	2.2*
10	40.6	10.47	2.2*	36.3	9.36	2.4*
11	40.6	10.47	2.5*	36.3	9.36	2.7*

*These slam loads are less than other design minimum loads that are typically applied to shell structure, i.e., 500 psf = 3.5 psi = 25.8 kN/m^2 for the design of shell plate and stiffening.

5.4 DNV STRUCTURAL LOADS

In accordance with DNV the global hull girder and local slamming loads acting on a high-speed vessel are a function of its service restrictions and the type of craft.

5.4.1 DNV Service Restrictions and Vessel Types

An important consideration for the development of loads in accordance with DNV is the type and service of the vessel to be designed. The types of vessels include:

- Passenger
- Car Ferry
- Cargo
- Crew
- Yacht
- Patrol
- Naval

The areas of service of these vessels includes, see Figure 5-5:

- Unrestricted
- Ocean
- Offshore
- Coastal
- Inshore
- Inland
- Sheltered

As part of these definitions, DNV also includes service area restrictions, which relates to the distance from safe harbor that a vessel is allowed to operate during different times of the year. The service area restriction definitions allow an owner to design a vessel tailored to the environment expected for the operation of the vessel. The service area restrictions range from **R0** to **R6** and are shown in Figure 5-5 below, which is reprinted from DNV Part 1, Chapter 1, Section 2/401 [2]. Each of the service area restrictions requires the calculation for global loads and slamming loads. The values of various coefficients that are used for the load calculations have reduced values as the service area restrictions become more severe, i.e., the design loads decrease as the allowable, maximum distance from safe harbor decreases. This reduces the loads acting on the ship structure and the subsequent cost for the structural design and fabrication. Figure 5-5 summarizes the various service area restriction notations and defines the maximum distances from safe harbor, in nautical miles, that a vessel can operate for the given notation.

Table B1 Service restrictions, general				
Condition	*Notation*	*Winter*	*Summer*	*Tropical*
Ocean	None	1)	1)	1)
Ocean	**R0**	300	1)	1)
Ocean	**R1**	100	300	300
Offshore	**R2**	50	100	250
Coastal	**R3**	20	50	100
Inshore	**R4**	5	10	20
Inland	**R5**	1	2	5
Sheltered	**R6**	0.2	0.3	0.5

Guidance Note:

1) Unrestricted service notation is not applicable to craft falling within the scope of the HSC Code, i.e., service and type Notations **Passenger**, **Car Ferry** or **Cargo**.

Figure 5-5. DNV Service Restrictions from DNV 1-1-2/401

As called out in Note 1 in Figure 5-5, the unrestricted service notation is not applicable to craft designed for the HSC Code. Therefore, it was necessary to check the requirements for Patrol and Naval craft to see if the criteria for either one of these classes allows for the unrestricted operation notation.

DNV Table F1 and Table F2, shown below in Figure 5-6, are from DNV Part 0, Chapter 5, Section 1 [2]. They imply that the unrestricted service notation may be available for both the Patrol and Naval craft notations.

Table F1 Overview of class notations		
Main class requirements	*Craft for special service*	*Special equipment and systems*
Mandatory requirements for all craft (military and merchant) Rules Pt. 1, Pt. 2, Pt. 3 and Pt.4	Mandatory requirements for each craft type Rules Pt. 5	Optional requirements for special equipment or features including naval performance Rules Pt. 6
✠ **1A1** **R0** **R1** **R2** **R3** **"unrestricted"**	**Patrol** **Naval**	**E0** **NAUT** **HMON** **ICS** **ELT** **NBC** **N, SV, MV**
Abbreviations used are explained in Table F2 to F6.		

F 400 Main class notations (Pt. 1 to Pt. 5 of the rules)

401 The main class notations for naval surface craft are Explained in Table F2.

Table F2 Main class notations for naval surface craft	
Class notation (See Pt.1, Pt. 2, Pt.3, Pt. 4 *and* Pt. 5*)*	*Description*
✠	Construction symbol indicating that the craft is built under supervision by the Society.
1A1 (Pt. 1 to Pt.4)	Mandatory minimum requirements to accommodate for naval operations with respect to: - freeboard, stability and watertight integrity - structural strength - propulsion system - electrical power supply - steering, navigation and communication - fire safety - equipment - accommodation and lifesaving equipment.
"Unrestricted"	Unlimited service worldwide when no service restriction is given.
R0 to R3	Service restriction specifies maximum distance from safe place of refuge/harbour/anchorage.
Naval (Pt. 5 Ch. 7)	Mandatory minimum requirements to accommodate for naval operations with respect to: - wave loads - stability and watertight integrity - loads from weapon systems - survivability - damage resistance to the hull, machinery and systems The requirements can be n the form of additions and exemptions to Pt. 1, Pt. 2, Pt. 3 and Pt. 4

Figure 5-6. DNV Patrol & Naval Craft Notations from DNV 0-5-1

The Patrol Craft criteria are contained in DNV Part 5, Chapter 6. Section 1/301 [2] of this portion of the rules states:

> "*Craft with the class notation* ***Patrol*** *will be assigned one of the service restrictions* ***R0****,* ***R1****,* ***R2****, or* ***R3****.*"

This provides a concrete definition that Patrol craft cannot be granted an unrestricted notation without seeking a deviation from the rules. Since that is not the intent of this project it became necessary to investigate the Naval Craft criteria to determine if unrestricted notation is allowed within the rules for these craft.

The Naval Craft criteria are contained in DNV Part 5, Chapter 7 [2]. Table A1 from Section 3/A201, shown below as Figure 5-7 provides the definition that confirms that Naval Surface Craft can be designed for unrestricted operation.

Table A1 Acceleration factor f_g

Type and service notation	*Service area restriction notation*				
	None	**R0**	**R1**	**R2**	**R3**
Naval	8	7	5	3	1
None = unrestricted service					

Guidance note:

Vertical design acceleration a_{cg} is to be considered as a design Parameter used for calculation of local and global loads and for Specification of operational restrictions for the craft.

Figure 5-7. DNV Acceleration factor f_g for Naval Surface Craft from DNV 5-7-3/A201

It is particularly important to recognize two impacts of Table A1, Figure 5-7:

1. It does confirm the allowance of Naval craft to be designed for unrestricted notation.
2. The values given in Table A1 are for the acceleration factor, f_g, which is used to calculate a_{cg}, the design vertical acceleration at the center of gravity of the vessel, which is used as a variable for all slam load and global hull girder load calculations. The value of $f_g = 8$ is required for unrestricted notation. The value of $f_g = 1$ was used for the original **R4** PacifiCat design. The value used for the original PacifiCat design was taken from DNV Part 3, Chapter 1, Section 2 B/201 Table B1 [2] as shown below in Figure 5-8.

Table B1 Acceleration factor f_g						
Type and service notation	*Service area restriction notation*					
	R0	**R1**	**R2**	**R3**	**R4**	**R5-R6**
Passenger	1)	1	1	1	1	0.5
Car Ferry	1)	1	1	1	1	0.5
Cargo	4	3	2	1	1	0.5
Patrol	7	5	3	1	1	0.5
Yacht	1	1	1	1	1	0.5
1) Service area restriction **R0** is not available for class notations **Passenger** and **Car Ferry**.						

Figure 5-8. DNV Acceleration factor f_g for Other Service Types from DNV 3-1-2 B/201

The increase of f_g from 1 to 8 plays a very significant role increasing the loads required to convert the vessel in accordance with the DNV HSLC&NSC criteria. The equation for a_{cg} is presented in various locations of the DNV rules for consideration of commercial and naval craft. Regardless, the same equation is used for the **R4**, **R1** and unrestricted calculations in this project and is given below as:

$$a_{cg} = \left(\frac{V}{\sqrt{L}}\right)\left(\frac{3.2}{L^{0.76}}\right) f_g g_0 \quad (\text{m/s}^2)$$

Where: $\frac{V}{\sqrt{L}}$ need not be taken greater than 3.0

V, L defined in the List of Symbols
$g_0 = 9.81$ m/s^2
f_g = Acceleration factor discussed above
a_{cg} = Vertical design acceleration

For the Operational condition using V = 37 knots, L = 96 meters and $V/\sqrt{L} \leq 3.0$, results in:

a_{cg} = 2.8 m/s^2 for Car Ferry, **R4** notation, f_g =1
a_{cg} = 15.1 m/s^2 for **R1**, f_g = 5
a_{cg} = 23.5 m/s^2 for Naval Surface Craft, Unrestricted, f_g = 8

A requirement in DNV Part 3, Chapter 1, Section2/B201 [2] states that a_{cg} cannot be taken as less than $1g_0$ for service notations **R0** – **R4.** This redefines the value shown above from a_{cg} = 2.8 m/s^2 to a_{cg} = 9.8 m/s^2 for the **R4** notation. Regardless, this represents an increase of approximately 240% in a variable that is directly used in the calculation of all hull girder and slam loads required for the structural design of the vessel in the operational mode.

In the survival condition, V = 10 knots is substituted above, for comparison to ABS.

For the Survival condition, V = 10 knots, L = 96 meters results in:

$a_{cg} =$ 1.0 m/s^2 for Car Ferry, **R4** notation, $f_g = 1$
$a_{cg} =$ 5.0 m/s^2 for **R1**, Unrestricted, $f_g = 5$
$a_{cg} =$ 8.0 m/s^2 for Naval Surface Craft, Unrestricted, $f_g = 8$

All of these values default to $a_{cg} = 9.81$ m/s^2.

5.4.2 DNV Global Loads

As stated above, the Naval Surface Craft design criteria are presented in DNV HSLC & NSC Part 5, Chapter 7 [2]. For calculation of the Midship vertical wave bending moment DNV 5-7-3/B 201 refers to the earlier section of the rules that deals with all vertical moment calculations, DNV Part 5, Chapter 7, Section 3 and Part 3, Chapter 1. Most of the actual load calculations appear in Part 3, Chapter 1. The most important aspect from DNV 5-7-3 is the calculation for a_{cg}, the design vertical acceleration, which, as stated above, uses a value of $f_g = 8$ for Naval Surface Craft, resulting in the value of $a_{cg} = 23.5$ m/s^2.

The DNV HSLC & NSC Rules presents the following equations for the calculations of the global loads acting on a twin hull vessel:

- $M_{tot\ hog} = M_{sw} + 0.19 C_w L^2 (B_{wL2} + k_2 B_{tn}) C_B$ (DNV 3-1-3/A 503)
- $M_{tot\ sag} = M_{sw} + 0.14 C_w L^2 (B_{wL2} + k_3 B_{tn}) (C_B + 0.7)$ (DNV 3-1-3/A 503)
- M_{sw} = Still water bending moment (DNV 3-1-3/A 503)
  = $0.5\Delta L$ in hogging, if not known
  = 0 in sagging, if not known

The slam induced bending moments in DNV are generated from two different conditions:

- Crest Landing

$$M_B = \frac{\Delta}{2}(g_0 + a_{cg})\left(e_w - \frac{l_s}{4}\right) \quad \text{(DNV 3-1-3/A 200)}$$

- Hollow Landing

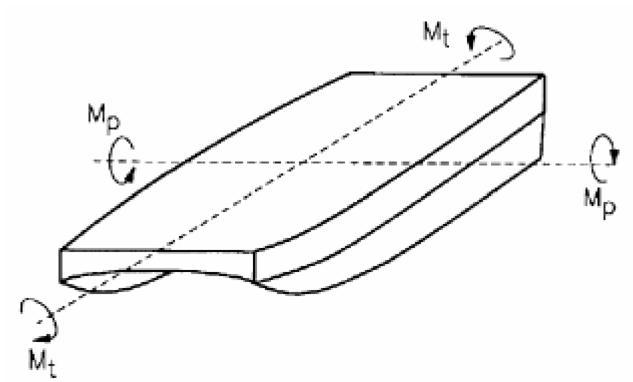
$$M_B = \frac{\Delta}{2}(g_0 + a_{cg})(e_r - e_w) \quad \text{(DNV 3-1-3/A 300)}$$

The transverse hull girder bending moment is to be taken as the worst of the following two conditions:

$$M_s = M_{so}\left(1 + \frac{a_{cg}}{g_0}\right) \qquad \text{(DNV 3-1-3/B 202)}$$

$$M_s = M_{so} + F_y(z - 0.5T) \qquad \text{(DNV 3-1-3/B 202)}$$

DNV defines pitch connecting and torsional moments in accordance with DNV 3-1-3/B 301/Figure 7:



The pitch connecting moment is determined as:

$$M_p = \frac{\Delta a_{cg} L}{8} \qquad \text{(DNV 3-1-3/B 301)}$$

The torsional moment is defined as:

$$M_t = \frac{\Delta a_{cg} b}{4} \qquad \text{(DNV 3-1-3/B 400)}$$

DNV 3-1-3/A 800 [2] requires the following hull girder load combinations be considered:

- 80% longitudinal bending and shear + 60% torsion
- 60% longitudinal bending and shear + 80% torsion
- 70% transverse bending + 100% pitch connecting
- 100% transverse bending + 70% pitch connecting

Similar to the ABS load calculations, the DNV vertical bending moments for wave induced and still water do not include any effect of vessel speed or vertical acceleration. All other global moment loads do include an effect from vertical acceleration of the vessel.

Table 5-8 (Metric units) and Table 5-9 (English units), present the DNV global moments required for the **R1** and Unrestricted service notations for the vessel in accordance with the DNV HSLC&NSC Rules.

In Table 5-8, the Crest Landing, **R1** Survival Moment (a_{cg} =9.81 m/s^2), is greater than the Crest Landing, **R1** Operational Moment, (a_{cg} = 15.1 m/s^2) and the Hollow Landing, **R1** Operational

Moment (a_{cg} = 15.1 m/s^2), is greater than the Hollow Landing, Unrestricted Operational Moment (a_{cg} = 23.5 m/s^2) because of the manner in which these values are calculated. The equations for Hollow and Crest Landing require definition of e_w, l_s, and e_r, which in turn require the definitions for A_R and b_s. These relationships are presented below:

e_w = one half the distance from the LCG of the fore half body to the LCG of the aft half body of the vessel, in m = 0.25L for crest landing if not known (0.2L for hollow landing)

e_r = mean distance from the center of the A_R/2 end areas to the vessels LCG, m. (Refer to sketch for hollow landing.)

l_s = longitudinal extension of the slamming reference area given as:

$$l_s = \frac{A_R}{b_s}$$

where b_s is the breadth of the slamming reference area.

$$A_R = k\Delta \frac{\left(1 + 0.2\frac{a_{cg}}{g_0}\right)}{T} \ (\text{m}^2)$$

where k = 0.7 for crest landing and 0.6 for hollow landing.

Investigation shows that as a_{cg} increases, so do the values of A_R and l_s. Therefore, the value of Crest landing decreases as a_{cg} increases because the term ($e_w - l_s/4$) decreases.

Similarly, the term ($e_r - e_w$) for hollow landing decreases as the value of a_{cg} increases because e_r decreases as a_{cg} and A_R increase.

Table 5-8. DNV Global Hull Girder Loads for R1 & Unrestricted Notations - Metric Units

	MaxVert Moment* kN-m	SW Hogging kN-m	WI Hogging kN-m	SW Sagging kN-m	WI Sagging kN-m	Crest Landing kN-m	Hollow Landing kN-m	Transverse kN-m	Pitch Connecting kN-m	Torsional kN-m
Unrestricted Operational	245,774	90,835	122,825	0	245,774	120,939	169,564	130,527	533,093	241,633
Unrestricted Survival	245,774	90,835	122,825	0	245,774	118,316	165,031	76,901	222,547	100,873
R1 Operational	245,774	90,835	122,825	0	245,774	113,914	177,062	96,006	333,183	151,021
R1 Survival	245,774	90,835	122,825	0	245,774	118,316	165,031	76,901	222,547	100,873

Note: * The Maximum Vertical Moment is the worst-case absolute value moment comparing |SW Hogging + WI Hogging| and |SW Sagging + WI Sagging| where SW = Still Water and WI = Wave Induced.

Table 5-9. DNV Global Hull Girder Loads for R1 & Unrestricted Notations - English Units

	Max Vert Moment* Lton-Ft	SW Hogging Lton-Ft	WI Hogging Lton-Ft	SW Sagging Lton-Ft	WI Sagging Lton-Ft	Crest Landing Lton-Ft	Hollow Landing Lton-Ft	Transverse Lton-Ft	Pitch Connecting Lton-Ft	Torsional Lton-Ft
Unrestricted Operational	80,925.7	29,909.1	40,442.4	0	80,925.7	39,821.4	55,832.1	42,978.5	175,530.9	79,562.2
Unrestricted Survival	80,925.7	29,909.1	40,442.4	0	80,925.7	38,957.8	54,339.6	25,321.1	73,277.8	33,214.3
R1 Operational	80,925.7	29,909.1	40,442.4	0	80,925.7	37,508.3	58,301.0	31,611.8	109,706.7	49,726.5
R1 Survival	80,925.7	29,909.1	40,442.4	0	80,925.7	38,957.8	54,339.6	25,321.1	73,277.8	33,214.3

Note: * The Maximum Vertical Moment is the worst-case absolute value moment comparing $|\text{SW Hogging + WI Hogging}|$ and $|\text{SW Sagging + WI Sagging}|$ where SW = Still Water and WI = Wave Induced.

5.4.3 DNV Secondary Slamming Loads

Similar to the global loads, DNV 5-7-3 [2] refers to DNV 3-1 [2] for the calculation of secondary slam loads. More specifically, the bottom slamming and bow impact pressures are to be obtained from DNV 3-1-2/C 200 and C 300 [2]. The Wet Deck slamming loads are to be obtained from DNV 3-1-2/C 400 [2]. The vertical acceleration, a_{cg}, to be used in those calculations is obtained from the Naval Surface Craft requirements in DNV 5-7-3 [2] for the unrestricted notation. For service restriction **R1** the vertical acceleration is obtained from DNV 3-1-2/B 200 [2].

The bottom slamming pressure is calculated as:

$$P_{sl} = 1.3 k_l \left(\frac{\Delta}{nA} \right)^{0.3} T_O^{0.7} \left(\frac{50 - \beta_x}{50 - \beta_{cg}} \right) a_{cg} \qquad \text{(DNV 3-1-2/C 200)}$$

The forebody side and bow impact pressures are calculated from DNV 3-1-2/C 300 as:

$$P_{sl} = \frac{0.7 L C_L C_H}{A^{0.3}} \left(0.6 + 0.4 \frac{V}{\sqrt{L}} \sin\gamma \cos(90-\alpha) + \frac{2.1 a_0}{C_B} \sqrt{0.4 \frac{V}{\sqrt{L}} + 0.6} \sin(90-\alpha) \left(\frac{x}{L} - 0.4 \right) \right)^2$$

The Wet Deck slamming pressures are calculated using:

$$P_{sl} = 2.6 k_t \left(\frac{\Delta}{A} \right)^{0.3} a_{cg} \left(1 - \frac{H_C}{H_L} \right) \qquad \text{(DNV 3-1-2/C 400)}$$

Table 5-10 and Table 5-11 present the DNV **R1** and Unrestricted slam loads, respectively, in Metric notation. Table 5-12 and Table 5-13 present the same information in the English system of units.

Table 5-10. DNV Secondary Slam Load Pressures for R1 Notation Operational & Survival Conditions - Metric Units

All pressures are kN/m^2

	R1 Operational			R1 Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	375.8	103.6	164.0	251.0	103.6	55.8
2	375.8	97.1	164.0	251.0	97.1	55.8
3	375.8	90.6	164.0	251.0	90.6	55.8
4	375.8	84.1	164.0	251.0	84.1	55.8
5	375.8	77.6	82.0	251.0	77.6	27.9
6	375.8	71.2	82.0	251.0	71.2	27.9
7	338.3	71.2	82.0	225.9	71.2	27.9
8	393.9	71.2	82.0	263.1	71.2	27.9
9	448.6	71.2	82.0	299.6	71.2	27.9
10	451.0	71.2	82.0	301.2	71.2	27.9
11	375.8	71.2	82.0	251.0	71.2	27.9

Table 5-11. DNV Secondary Slam Load Pressures for Unrestricted Notation Operational & Survival Conditions - Metric Units

All pressures are kN/m^2

	Unrestricted Operational			Unrestricted Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	601.3	103.6	262.4	251.0	103.6	89.3
2	601.3	97.1	262.4	251.0	97.1	89.3
3	601.3	90.6	262.4	251.0	90.6	89.3
4	601.3	84.1	262.4	251.0	84.1	89.3
5	601.3	77.6	131.2	251.0	77.6	44.6
6	601.3	71.2	131.2	251.0	71.2	44.6
7	541.2	71.2	131.2	225.9	71.2	44.6
8	630.2	71.2	131.2	263.1	71.2	44.6
9	717.7	71.2	131.2	299.6	71.2	44.6
10	721.6	71.2	131.2	301.2	71.2	44.6
11	601.3	71.2	131.2	251.0	71.2	44.6

Table 5-12. DNV Secondary Slam Load Pressures for R1 Notation Operational & Survival Conditions - English Units

All pressures are PSI

	R1 Operational			R1 Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	54.5	15.0	23.8	36.4	15.0	8.1
2	54.5	14.1	23.8	36.4	14.1	8.1
3	54.5	13.1	23.8	36.4	13.1	8.1
4	54.5	12.2	23.8	36.4	12.2	8.1
5	54.5	11.3	11.9	36.4	11.3	4.0
6	54.5	10.3	11.9	36.4	10.3	4.0
7	49.1	10.3	11.9	32.8	10.3	4.0
8	57.1	10.3	11.9	38.1	10.3	4.0
9	65.0	10.3	11.9	43.4	10.3	4.0
10	65.4	10.3	11.9	43.7	10.3	4.0
11	54.5	10.3	11.9	36.4	10.3	4.0

Table 5-13. DNV Secondary Slam Load Pressures for Unrestricted Notation Operational & Survival Conditions - English Units

All pressures are PSI

	Unrestricted Operational			Unrestricted Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	87.2	15.0	38.0	36.4	15.0	12.9
2	87.2	14.1	38.0	36.4	14.1	12.9
3	87.2	13.1	38.0	36.4	13.1	12.9
4	87.2	12.2	38.0	36.4	12.2	12.9
5	87.2	11.3	19.0	36.4	11.3	6.5
6	87.2	10.3	19.0	36.4	10.3	6.5
7	78.5	10.3	19.0	32.8	10.3	6.5
8	91.4	10.3	19.0	38.1	10.3	6.5
9	104.1	10.3	19.0	43.4	10.3	6.5
10	104.6	10.3	19.0	43.7	10.3	6.5
11	87.2	10.3	19.0	36.4	10.3	6.5

5.4.4 Direct Comparison of ABS and DNV Hull Girder and Secondary Slam Loads

The following tables present the respective hull girder and slamming loads from ABS and DNV in the same tables for ease of comparison.

Table 5-14 compares the ABS and DNV global loads. It only includes the ABS Naval and DNV Unrestricted classification loads. It does not include the ABS Coastal Naval and DNV **R1** Service Restriction loads. The values shown in Table 5-14 address the loads that represent the final conversion objective of the project.

Table 5-15 and Table 5-16 are copies of Table 5-4 and Table 5-11, respectively. They present the ABS and DNV slam loads for the ABS Naval and DNV Unrestricted classifications. Similar to the global loads, the comparison tables are only presented for the final objective design loads of this project.

Table 5-14. Comparison of ABS & DNV Global Hull Girder Loads - Metric Units

	Max Vertical Moment* kN - m	Transverse kN - m	Torsional kN - m	Slam Induced kN - m	Pitch Connecting kN-m	Crest Landing kN-m	Hollow Landing kN-m
ABS Naval, Operational	161,123	127,636	273,143	101,539	NA	NA	NA
ABS Naval, Survival	161,123	104,991	224,682	83,524	NA	NA	NA
DNV Unrestricted Operational	245,774	130,527	241,633	NA	533,093	120,939	169,564
DNV Unrestricted Survival	245,774	76,901	100,873	NA	222,547	118,316	165,031

Note: * Maximum value of Wave Induced plus Still Water, hogging and sagging conditions.

Table 5-15. ABS Secondary Slam Load Pressures for Naval Craft Operational & Survival Conditions - Metric Units

All pressures are kN/m^2

	Naval Craft Operational			Naval Craft Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	243.3	94.05	138.3	171.0	66.09	67.6
2	235.4	91.00	138.3	170.3	65.84	67.6
3	227.5	87.95	138.3	169.6	65.58	67.6
4	219.6	84.90	55.3	169.0	65.33	27.0
5	211.7	81.85	55.3	168.3	65.07	27.0
6	203.8	78.80	55.3	167.7	64.82	27.0
7	195.9	75.75	55.3	167.0	64.56	27.0
8	226.3	75.75	55.3	192.9	64.56	27.0
9	265.0	75.75	55.3	225.9	64.56	27.0
10	293.9	75.75	62.2	250.5	64.56	30.4
11	293.9	75.75	69.2	250.5	64.56	33.8

Table 5-16. DNV Secondary Slam Load Pressures for Unrestricted Notation Operational & Survival Conditions - Metric Units

All pressures are kN/m^2

	Unrestricted Operational			Unrestricted Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	601.3	103.6	262.4	251.0	103.6	89.3
2	601.3	97.1	262.4	251.0	97.1	89.3
3	601.3	90.6	262.4	251.0	90.6	89.3
4	601.3	84.1	262.4	251.0	84.1	89.3
5	601.3	77.6	131.2	251.0	77.6	44.6
6	601.3	71.2	131.2	251.0	71.2	44.6
7	541.2	71.2	131.2	225.9	71.2	44.6
8	630.2	71.2	131.2	263.1	71.2	44.6
9	717.7	71.2	131.2	299.6	71.2	44.6
10	721.6	71.2	131.2	301.2	71.2	44.6
11	601.3	71.2	131.2	251.0	71.2	44.6

5.5 LOADS USED FOR THE DESIGN OF THE ORIGINAL PACIFICAT

The original design for the PacifiCat ferries was done using the DNV HSLC & NSC Rules [2] with an **R4** service restriction for a car ferry. All the local slamming and global load calculations were developed using the same equations presented above with certain coefficients significantly reduced for the **R4** notation compared to the unrestricted values used above. In particular, the two variables that have the most immediate impact are design vertical acceleration, a_{cg} and the wave coefficient, C_w. These values for **R4** and unrestricted usage are:

$a_{cg} = 9.8$ m/s^2	**R4** Car Ferry
$a_{cg} = 23.5$ m/s^2	Unrestricted Naval Craft
$C_w = 4.61$	**R4** Car Ferry
$C_w = 7.68$	Unrestricted Naval Craft

Table 5-17 and Table 5-18 present the DNV hull girder and slam loads, respectively using the Metric system of measure. Table 5-19 and Table 5-20 present the same information using the English system of notation. These can only be considered as the preliminary design, rule-book based load values used for the design of the PacifiCat. To date, it has not been possible to obtain the actual design loads from the builder and so the values presented below can be used for comparison.

Table 5-17. DNV Rule Book Global Hull Girder Loads for PacifiCat R4 Notation - Metric Units

	Max Vert Moment* kN-m	SW Hogging kN-m	WI Hogging kN-m	SW Sagging kN-m	WI Sagging kN-m	Crest Landing kN-m	Hollow Landing kN-m	Transverse kN-m	Pitch Connecting kN-m	Torsional kN-m
R4	165,031	90,835	158,218	0.0	137,065	118,315	165,031	76,900	222,547	100,873

Note: * The Maximum Vertical Moment is the worst-case absolute value moment comparing |SW Hogging + WI Hogging| and |SW Sagging + WI Sagging| where SW = Still Water and WI = Wave Induced.

Table 5-18. DNV Rule Book Secondary Slam Load Pressures for PacifiCat R4 Notation

All pressures are kN/m^2

	R4		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	251.0	70.0	32.8
2	251.0	66.1	32.8
3	251.0	62.2	32.8
4	251.0	58.3	32.8
5	251.0	54.4	16.4
6	251.0	50.5	16.4
7	225.9	50.5	16.4
8	263.1	50.5	16.4
9	299.6	50.5	16.4
10	301.2	50.5	16.4
11	251.0	50.5	16.4

Table 5-19. DNV Rule Book Global Hull Girder Loads for PacifiCat R4 Notation - English Units

	Max Vert Moment* Lton-Ft	SW Hogging Lton-Ft	WI Hogging Lton-Ft	SW Sagging Lton-Ft	WI Sagging Lton-Ft	Crest Landing Lton-Ft	Hollow Landing Lton-Ft	Transverse Lton-Ft	Pitch Connecting Lton-Ft	Torsional Lton-Ft
R4	54,339.6	29,909.1	52,096.2	0.0	45,131.2	38,957.4	54,339.6	25,320.8	73,277.8	33,214.3

Note: * The Maximum Vertical Moment is the worst-case absolute value moment comparing |SW Hogging + WI Hogging| and |SW Sagging + WI Sagging| where SW = Still Water and WI = Wave Induced.

Table 5-20. DNV Rule Book Secondary Slam Load Pressures for PacifiCat R4 Notation

All pressures are PSI

	R4		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	36.4	10.1	4.8
2	36.4	9.6	4.8
3	36.4	9.0	4.8
4	36.4	8.5	4.8
5	36.4	7.9	2.4
6	36.4	7.3	2.4
7	32.8	7.3	2.4
8	38.1	7.3	2.4
9	43.4	7.3	2.4
10	43.7	7.3	2.4
11	36.4	7.3	2.4

5.6 LOADS DUE TO PAYLOAD OF MILITARY VEHICLES

Discussion presented below uses the English system of units, inches & pounds, for all tire and vehicle data.

One of the critical loads to be considered for the structural modifications required for the conversion of the vessel are the tire footprint loads associated with the military vehicles. Typical vehicle deck structure is checked against the matrix of vehicles scheduled to access the deck and the worst footprint is used to design the deck plate and stiffening. In the case of extreme loads that may only result from a few select vehicles, it is standard practice to reinforce a portion of the deck as necessary to accommodate these loads and require that the governing vehicle park in the reinforced area. Of course, all deck structure leading to the reinforced area would also have to be reinforced to support the vehicle as it transits to it designated area. The reinforcement of the travel lanes may not be as demanding because such transit will occur with the ship pierside and the design vertical acceleration will reflect static 1g with no increase from ship motion effects, i.e., $n_{xx} = 0$.

A typical vehicle footprint is shown in Figure 5-9. Other footprints are shown in Appendix C.

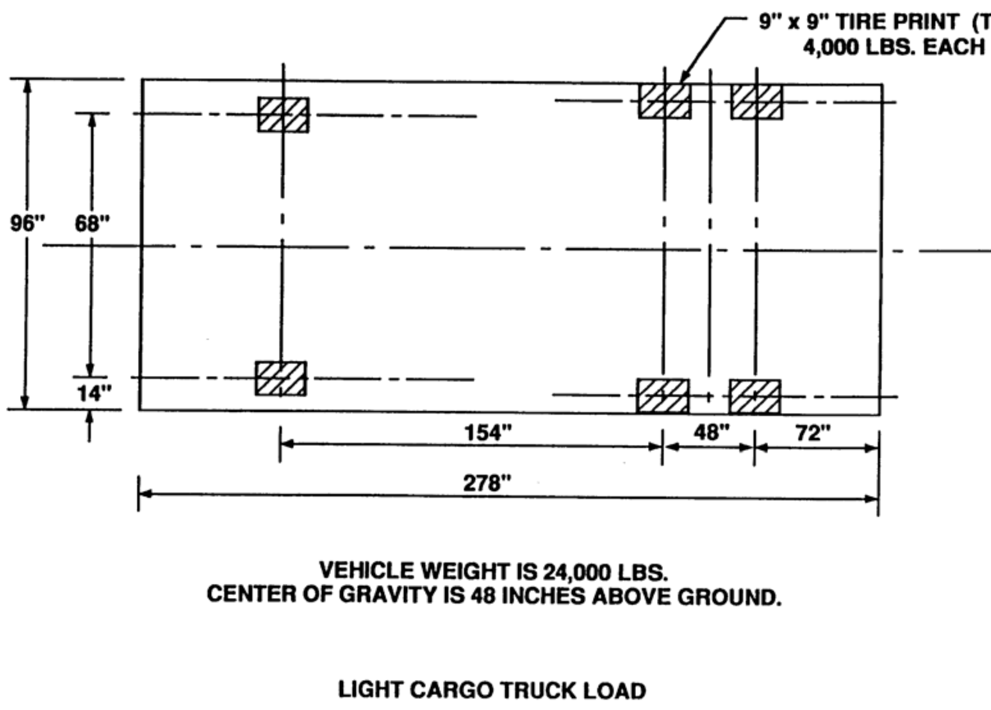


Figure 5-9. Typical Tire Footprint for Vehicle Deck Loading Data

Unfortunately, it was not possible to find all this footprint data for the MEU loadout selected for the conversion. Therefore, various approximations were made to estimate the footprints and associated tire pressures with the MEU loadout. The estimates are based on typical vehicle information used for sealift programs and originate from the technical manuals of the United States Marine Corps as well as other sources of information.

Assessment of the deck structure subjected to the vehicle loads was undertaken using both the ABS and DNV criteria. Both ABS and DNV present design criteria that directly addresses deck structure subjected to wheel loads.

5.6.1 Investigation of Tire Footprints on Deck Structure

The typical structural analysis of vehicle handling decks will investigate vehicles and their tires oriented in both the transverse and longitudinal direction, defining the more severe of the two conditions as the design governing criteria. The PacifiCat has a relatively long, narrow vehicle deck conducive to longitudinal travel for the vehicles. Also, it is not required that vehicles turn for off-loading, they can proceed "straight ahead" to off load at the end of the ship opposite from that which they entered due to the bow and stern access available on this vessel. In addition, the longitudinal stiffener spacing of the vehicle deck is relatively tight, either 210 mm or 200 mm, and the following discussion addresses this arrangement

The Main Vehicle Deck of the PacifiCat is fabricated using two different extrusions, a heavier extrusion from ship centerline to 3375 mm off centerline, P/S, and a lighter extrusion that covers the rest of the deck. Some basic characteristics of these two extrusions are summarized below:

	Plate Thickness (mm)	Stiffener Spacing (mm)	Stiffener Depth (mm)
Heavy Extrusion	9.8	210	140
Light Extrusion	8.0	200	103

The transverse web frame spacing is 1200 mm throughout the entire Main Vehicle Deck. This results in two similar deck panels that may require investigation subjected to vehicle tire footprint loads. These are shown schematically in Figure 5-10 as:

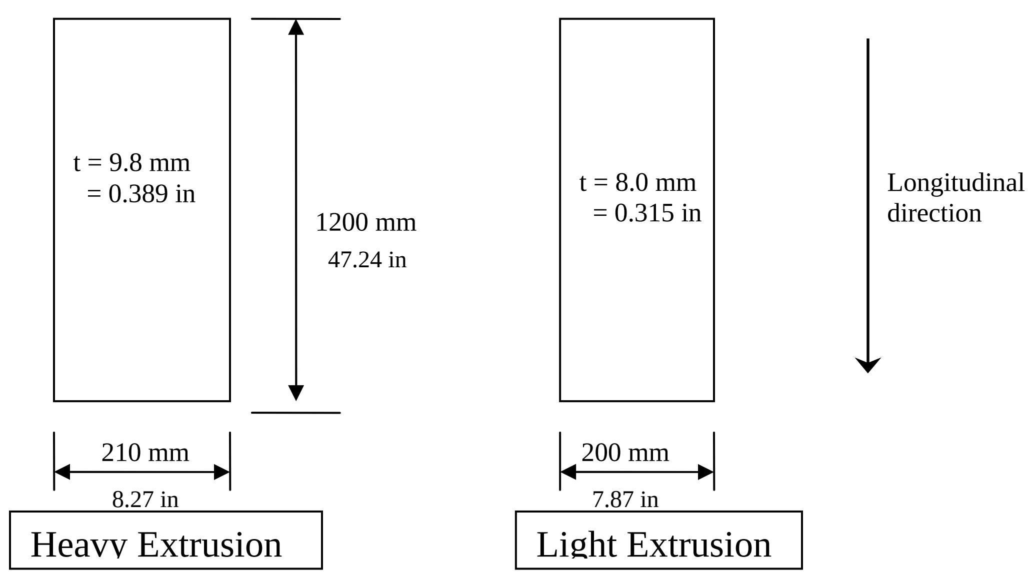


Figure 5-10. Schematic Representation of Heavy & Light Deck Extrusions

As seen in Appendix C, most of the tire footprints have a minimum dimension that exceeds the stiffener spacing of these extrusions. This has some interesting implications regarding the assumptions inherent in the required plate thickness equations of ABS and DNV and further discussion on this issue is provided below. Also, it was not possible to gather all the tire footprint data for all the vehicles contained in the chosen load-out. The only vehicle for which this is thought to be a concern is the 25,000 pound forklift truck, shown below in Table 5-23. Realizing that the forklift truck will only be stowed and not operational reduces the concern regarding this vehicle since most forklift trucks only present design governing wheel loads when they are carrying a maximum payload and their front axle is carrying its maximum design load.

5.6.2 Vehicle Data – Tire Footprints

The typical requirement for the design of deck structure subjected to wheel loads is knowledge of both the load and the footprint associated with the tire or track load of the vehicles. No single source of information was found that provided this information for the current project. Data was gathered from earlier Sealift projects, an MTVR study (Medium Tactical Vehicle Replacement) [5] and Globalsecurity.org. None of these sources was able to provide exact data for any of the vehicles in the proposed loadout. Estimates using the available data were made to assess the impact of the military vehicle payload on the existing deck structure in accordance with the procedure discussed below.

It was the intent of this project to have footprint data for all of the vehicles in the proposed loadout. This was severely complicated by the lack of required data for vehicles. As work progressed, it became apparent that the original list of vehicles could be broken into three categories based on the data that became available:

1. High Confidence Data for Vehicles, Table 5-22, – This is a list of vehicles and associated footprint data that has a high degree of confidence. The information presented in Table 5-22 forms the baseline for the calculations determining the effect to deck structure to accommodate the MEU loadout.
2. Lower Confidence Data for Potentially Critical Vehicles, Table 5-23, – This is a list of vehicles that could govern the design of deck structure but for which very little direct information was available. These vehicles and the data assumed for their footprint is provided in Table 5-23. Estimates on footprint size and load were based on data from other, similar vehicles, such as those in Table 5-22.
3. Lower Confidence Data for Non-Critical Vehicles, Table 5-24, – There are a number of vehicles which, based on intuition and other calculations, will not govern the design of deck structure but also had very little direct information available. The appropriate data for these vehicles is included in Table 5-24 but no calculations will be required for these vehicles.

The data in Table 5-22, Table 5-23 and Table 5-24 is presented in English units only. This is raw data, unlike the calculated data reflecting ABS and DNV loads, which can be calculated in either English or metric units for both rule sets. It was considered appropriate to leave the raw data in its original units.

All of the vehicles included in the original table for the proposed loadout are included in one of the three tables shown below. The original table (Table 2-1) is presented below, in Table 5-21, for ease of reference.

Table 5-21. Notional Vehicle Load for Conversion Vessel

NOMENCLATURE	LHA(R) QTY	Conv QTY	LN	WD	HT	WT	Area (SQ FT)	TOTAL WT (LBS)
AN/MLQ-36	1	1	255	99	126	28,000	175	28,000
AN/MRC-138B	8	2	180	85	85	6,200	213	12,400
AN/MRC-145	8	2	185	85	83	6,200	218	12,400
ARMORED COMBAT EXCA	2	1	243	110	96	36,000	186	36,000
TRK, FORKLIFT	1	1	315	102	101	25,600	223	25,600
TRK, FORKLIFT, 4K	1	1	196	78	79	11,080	106	11,080
TRAM	1	1	308	105	132	35,465	225	35,465
CONTAINER, QUADRUPLE	70	20	57	96	82	5,000	760	100,000
CHASSIS TRLR GEN PUR, M353	1	1	187	96	48	2,720	125	2,720
CHASSIS, TRAILER 3/4 T	2	2	147	85	35	1,840	174	3,680
POWER UNIT, FRT (LVS) MK-48	2	1	239	96	102	25,300	159	25,300
TRLR CARGO 3/4 T (M101A1)	4	2	145	74	50	1,850	149	3,700
TRAILER CARGO M105	6	2	185	98	72	6,500	252	13,000
TRLR, MK-14	2	2	239	96	146	16,000	319	32,000
TRLR TANK WATER 400 GL	2	1	161	90	77	2,530	101	2,530
TRK AMB 2 LITTER	1	1	180	85	73	6,000	106	6,000
TRK 7-T MTVR	24	8	316	98	116	36,000	1,720	288,000
TRK 7-T M927 EXTENDED BED	1	1	404	98	116	37,000	275	37,000
TRK 7-T DUMP	1	1	315	98	116	31,888	214	31,888
TRK TOW CARRIER HMMWV	8	4	180	85	69	7,200	425	28,800
TRK, MULTI-PURPOSE M998	45	15	180	85	69	6,500	1,594	97,500
TRK,AVENGER/CLAWS	3	1	186	108	72	7,200	140	7,200
TRK ARMT CARR	10	2	186	108	72	7,000	279	14,000
TRK LIGHT STRIKE VEHICLE	6	2	64	132	74	4,500	117	9,000
155MM HOWITZER	6	2	465	99	115	9,000	639	18,000
LAV ANTI TANK (AT)	2	2	251	99	123	24,850	345	49,700
LAV C2	1	1	254	99	105	26,180	175	26,180
LAV ASSAULT 25MM	4	2	252	99	106	24,040	347	48,080
LAV LOGISTICS (L)	3	1	255	98	109	28,200	174	28,200
LAV, MORTAR CAR	2	1	255	99	95	23,300	175	23,300
LAV, MAINT RECOV	1	1	291	99	112	28,400	200	28,400
TANK COMBAT M1A1	4	-	387	144	114	135,000	-	-
MAINT VAN	4	2	240	96	96	10,000	320	20,000
RECOVERY VEH, M88	1	-	339	144	117	139,600	-	-
Totals:	**238**	**87**					**10,629**	**1,105,123**
Total STons								553

Table 5-22. Vehicle Footprint Data used for Deck Structural Requirements (High Confidence)

Vehicle	Total Weight (LBS)	No of Tires	Max Tire/Track Load (LBS)	Tire Footprint (IN x IN)	Deck Load (PSI)	Source of data
Armored Combat Excavator	36,000	2 tracks	18,000 lb/track	180x15	6.67	Sealift (Fig A-5)
MK-48 LVS, Front Power Unit	25,300	4 – 16R21	6830/tire	16x27	15.8	Globalsecurity.org
Trailer, Cargo 3/4 T (M101A1)	1850	2	925	7x12	11.0	Globalsecurity.org
Truck Ambulance 2 Litter (HMMWV)	6000	4 – 37x12.5R-16.5	1600	12.5x14	9.2	Globalsecurity.org
Truck 7-T, MTVR	36,000	6 – 16R20	6100	16x25	15.3	MTVR data
Truck 7-T M927, Extended Bed	37,000	6 – 4 rear are 16R20	6100	16x25	15.3	MTVR data
Truck Tow Carrier, HMMWV	7200	4 – 37x12.5R-16.5	2000	12.5x14	11.4	Globalsecurity.org
Truck, Multi Purpose M998, HMMV	6500	4 – 37x12.5R-16.5	1700	12.5x14	9.7	Globalsecurity.org
Truck, Avenger/Claws, HMMWV	7200	4 – 37x12.5R-16.5	2000	12.5x14	11.4	Globalsecurity.org
Truck, Army Carr, HMMWV	7000	4 – 37x12.5R-16.5	1950	12.5x14	11.1	Globalsecurity.org
Truck, Light Strike Vehicle (LSV)	4500	4	1200	9x9	14.8	Sealift (Fig A-1)
155MM Howitzer	9000	8	1200	8x12	12.5	Globalsecurity.org
LAV Anti Tank (AT)	24,850	8	3299	22x11	13.6	Sealift (Fig A-2)
LAV C2	26,180	8	3475	22x11	14.4	Sealift (Fig A-2)
LAV Assault 25MM	24,040	8	3191	22x11	13.2	Sealift (Fig A-2)
LAV Logistics (L)	28,200	8	3743	22x11	15.5	Sealift (Fig A-2)
LAV, Mortar	23,200	8	3080	22x11	12.7	Sealift (Fig A-2)
LAV, Maintenance Recovery	28,400	8	3770	22x11	15.6	Sealift (Fig A-2)

Table 5-23. Vehicle Footprint Data (Low Confidence – Potential Design Governing)

Vehicle	Total Weight (LBS)	No of Tires	Max Tire/Track Load (LBS)	Tire Footprint (IN x IN)	Deck Load (PSI)	Source of data
AN/MLQ-36	28,000					
AN/MRC-138B	6200					
AN/MRC-145	6200					
Truck, Forklift	25,600	TBD	TBD	TBD	TBD	
Truck, Forklift	11,080	TBD	TBD	TBD	TBD	
Tram	35,465	6 – 16R21	5911/tire	16x27	13.7	
Container, Quadruple	5000					
Trailer, Cargo M105	6500					
Trailer, MK-14	16,000					
Truck, 7-Ton Dump	31,888	6 – 16R20	4234	16x25	10.6	Assume MTVR
Maintenance Van	10,000					

Table 5-24. Vehicle Footprint Data (Low Confidence – Non-Design Governing)

Vehicle	Total Weight (LBS)	No of Tires	Max Tire/Track Load (LBS)	Tire Footprint (IN x IN)	Deck Load (PSI)	Source of data
Chassis, Trailer 3/4 T	1840	2	920	7x12	11.0	
Chassis, Trailer, Gen purp M353	2720	2	1360	7x12	16.2	
Trailer, Tank Water 400 Gallon	2530	2	1265	7x12	15.1	

5.6.3 Deck Structure Required – ABS HSNC Rules

Section 3-2-3/1.9 of the ABS HSNC Rules [1] addresses "Decks Provided for the Operation or Stowage of Vehicles." It presents the following equations for deck thickness requirements in metric and English consideration:

$$t = \sqrt{\frac{\beta W\left(1 + 0.5 n_{xx}\right)}{1000\sigma_a}} \text{ mm} \qquad\qquad t = \sqrt{\frac{\beta W\left(1 + 0.5 n_{xx}\right)}{\sigma_a}} \text{ in}$$

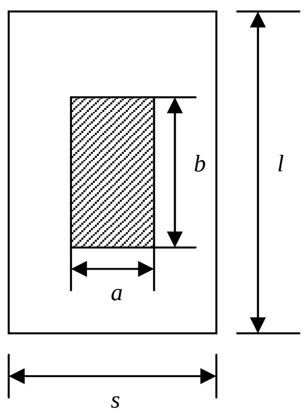
Where: W = static wheel load in kN (lbf)

n_{xx} = average vertical acceleration at the location under consideration as defined in ABS 3-2-5/1.1 (misprint in HSNC – should actually read 3-2-2/1.1)

β = as given in ABS 3-2-3/Figure 1, which is presented below as Figure 5-11

σ_a = design stress for decks, N/mm^2, (kgf/mm^2, psi), as given in ABS 3-2-3/Table 2, presented below as Figure 5-12

FIGURE 1
Values for β



B/s	*l/s* = 1						*l/s* = 1.4						*l/s* ≥ 2					
a/s	0	0.2	0.4	0.6	0.8	1	0	0.2	0.4	0.8	1.2	1.4	0	0.4	0.8	1.2	1.6	2
0		1.82	1.38	1.12	0.93	0.76		2.00	1.55	1.12	0.84	0.75		1.64	1.20	0.97	0.78	0.64
0.2	1.82	1.28	1.08	0.90	0.76	0.63	1.78	1.43	1.23	0.95	0.74	0.64	1.73	1.31	1.03	0.84	0.68	0.57
0.4	1.39	1.07	0.84	0.72	0.62	0.52	1.39	1.13	1.00	0.80	0.62	0.55	1.32	1.08	0.88	0.74	0.60	0.50
0.6	1.12	0.90	0.74	0.60	0.52	0.43	1.10	0.91	0.82	0.68	0.53	0.47	1.04	0.90	0.76	0.64	0.54	0.44
0.8	0.92	0.76	0.62	0.51	0.42	0.36	0.90	0.76	0.68	0.57	0.45	0.40	0.87	0.76	0.63	0.54	0.44	0.38
1	0.76	0.63	0.52	0.42	0.35	0.30	0.75	0.62	0.57	0.47	0.38	0.33	0.71	0.61	0.53	0.45	0.38	0.30

Note: *s* = spacing of deck beams or deck longitudinals in mm (in.)

l = length of plate panel in mm (in.)

a = wheel imprint dimension, in mm (in.), parallel to the shorter edge, *s*, of the plate panel, and in general the lesser wheel imprint dimension

b = wheel imprint dimension, in mm (in.), parallel to the longer edge, *l*, of the plate panel, and in general the longer wheel imprint dimension

Figure 5-11. Aspect Ratio Data for Design of Wheel Loaded Deck Plate from ABS 3-2-3/Figure 1

TABLE 2
Design Stress, σ_a, Aluminum and Steel

Location		*Design Stress, σ_a* [1]	
Bottom Shell		Slamming Pressure	$0.90\sigma_y$ [2]
		Hydrostatic Pressure	$0.55\sigma_y$
Water Jet Tunnels		Slamming Pressure	$0.60\sigma_y$
		Hydrostatic Pressure	$0.55\sigma_y$
Side Shell	Below Bulkhead Deck	Slamming Pressure	$0.90\sigma_y$
		Hydrostatic Pressure	$0.55\sigma_y$
	Above Bulkhead Deck (i.e., foc'sles)	Slamming Pressure	$0.90\sigma_y$
		Hydrostatic Pressure	$0.55\sigma_y$
Deck Plating	Strength Deck	$0.60\sigma_y$	
	Lower Decks/Other Decks	$0.60\sigma_y$	
	Wet Decks	$0.90\sigma_y$	
	Superstructure & Deckhouse Decks	$0.60\sigma_y$	
Bulkheads	Deep Tank	$0.60\sigma_y$	
	Watertight	$0.95\sigma_y$	
Superstructure aft of 0.25L from F.P. & Deckhouses	Front, Sides, Ends, Tops	$0.60\sigma_y$ [3]	

Notes:

1 σ_y = yield strength of steel or of welded aluminum in N/mm^2 (kgf/mm^2, psi), but not to be taken greater than 70% of the ultimate strength of steel or welded aluminum.

2 The design stress for bottom shell plates under slamming pressure may be taken as σ_y for plate outside the midships 0.4L.

3 The design stress for steel deckhouse plates may be taken as $0.90\sigma_y$.

Figure 5-12. ABS Allowable Design Stress for Wheel Loaded Decks from ABS 3-2-3/Table 2

The calculation for n_{xx} was shown above to be determined as:

$$n_{xx} = n_{cg}K_v$$

where K_v is determined from ABS 3-2-2/Figure 7 [1], which is shown above but is also repeated below in Figure 5-13 for ease of reference.

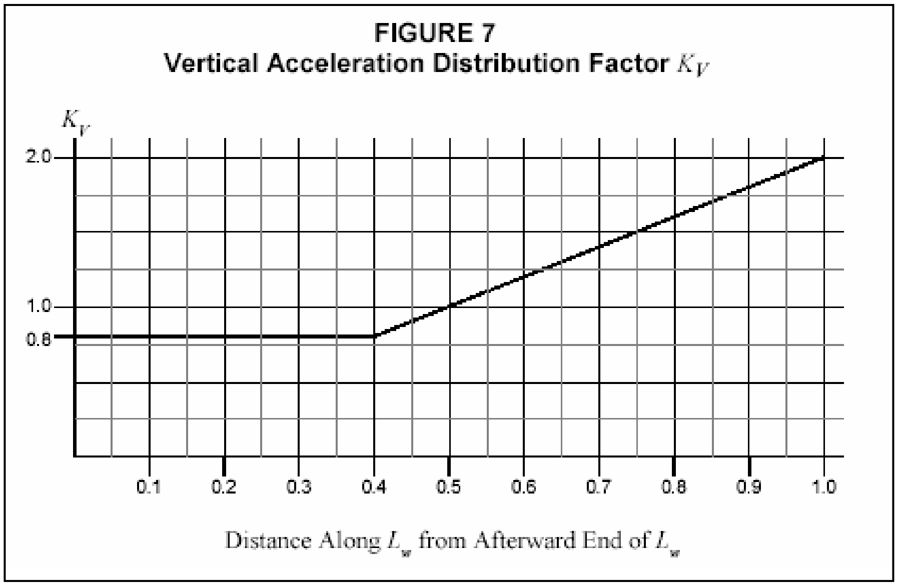


Figure 5-13. ABS Vertical Acceleration Distribution Factor K_V, from ABS 3-2-2/Figure 7

This figure implies that the vertical acceleration from the stern to $0.5L_w$ forward of the stern will be less than the design vertical acceleration determined at the center of gravity of the vessel, n_{cg}, i.e., it can be reduced to 80% of n_{cg} from the stern to $0.4L_w$. None of the calculations performed will utilize $K_v < 1.0$.

As shown in the equations for the plate thickness, the vertical acceleration, n_{xx}, is required input to the analysis. To summarize from above, there are four different conditions to be investigated that result in the following n_{cg} values:

Naval Craft, Operational ⇒	n_{cg} = 0.24 g's
Naval Craft, Survival ⇒	n_{cg} = 0.02 g's
Coastal Naval Craft, Operational ⇒	n_{cg} = 0.17 g's
Coastal Naval Craft, Survival ⇒	n_{cg} = 0.02 g's

The simplest approach is to determine if there are any impacts when using the most severe conditions. Therefore, all loading and structural requirements for ABS will be initially investigated using the Naval Craft, Operational scenario with n_{cg} = 0.24 g's. Additional calculations will be developed below, as required, so that the full range of impacts can be assessed when investigating final locations on the Vehicle Deck for payload items that would overstress the existing deck structure if located at the worst location for n_{xx}, i.e., at the bow of the vessel. If the finite element analyses developed later in this project confirm that all vehicle deck structure is acceptable using the maximum value for n_{cg} then no additional analyses will be developed using the smaller values. It is recognized that this is an area for potential structural optimization that could be investigated for a new build project and it is recommended that designers recognize that all FEA is done with maximum design value for vertical acceleration, i.e., for vehicles located in the bow of the ship. It would be possible to develop lighter structure for other locations in the ship, other criteria not withstanding.

5.6.4 Boundary Conditions - Discussion on the ABS Plate Thickness Requirements & Loading Conditions in the Current Conversion

Before proceeding with specific deck plate requirements, it is important to present the following discussion regarding the boundary conditions assumed by the plate thickness requirements in the ABS equation and the loading conditions resulting from the tire footprints in the MEU loadout.

The boundary conditions assumed for the analysis of any plate panel have a direct impact on the requirements for the panel thickness. There are two extreme conditions that can generally be assumed to bracket most problems associated with the design of typical ship structure plating:

1. Fully Fixed. – These boundary conditions assume that all six degrees of freedom are resisted around all four boundaries of the plate panel. The six degrees of freedom are translation in the X, Y and Z directions and rotation about the same three axes. Under fully fixed conditions, a reaction will occur for each of these DOF's.
2. Simply Supported. – These boundary conditions assume that five of the six degrees of freedom are restrained. The released DOF is the rotational restraint about the axis parallel to each edge of the plate. This implies that there is no resistance to bending, i.e., no bending moment at the boundary of the plate about the axis of primary concern.

The assumption of fully fixed boundary conditions will result in thinner, lighter plate than a plate experiencing the same load but with simply supported boundary conditions, which is why this consideration is important.

These two conditions are generally considered in the analysis of plate panels subjected to loads that are perpendicular, i.e., normal, to their surface, such as a tire load. When an individual plate panel is subjected to a load normal to its surface it is typically analyzed as simply supported. For the situation when adjacent panels are simultaneously subjected to normal loads then fixed conditions are often considered. Fixed boundary conditions may be assumed for a large area of stiffened panels subjected to hydrostatic loads, for instance. The loads imparted by tire footprints typically suggest analysis using simple support boundary conditions.

A comparison of the coefficients used by ABS and presented in Figure 5-11 to those presented in classical textbooks such as Roark, [6], confirms that the ABS equation assumes simply supported boundary conditions. This is perfectly acceptable and is consistent with sound engineering practice because most tire footprints are narrower than the deck stiffener spacing. However, this is not always true for the current conversion study and the assumption of simple supports may not be completely accurate which could result in greater, or lesser, plate thickness requirements than those resulting from the straight-forward application of the ABS equations.

The equation presented by Table 26, Case 1c in Roark [6] is for a simply supported plate panel with a centrally located rectangular patch load, similar to the tire footprint for the current condition. The equation in [6] is:

$$Max\sigma = \sigma_b = \frac{\beta W}{t^2}$$

Where: σ_b = the maximum bending stress, located at the center of the panel
W = central patch load
t = plate thickness
β = coefficient based on the plate and load footprint and aspect ratios

Roark provides a table of vales for β that is identical to the coefficients presented by ABS in their table, Figure 5-11. Simple manipulation of the Roark equation yields:

$$t = \sqrt{\frac{\beta W}{\sigma_a}}$$

Where: σ_a = the allowable stress, replacing σ_b.

The only difference between this equation and that presented in the ABS rules is the factor for vertical acceleration. Obviously, the ABS equation reduces to the exact same form as that presented by Roark if the design acceleration, $n_{xx} = 0$ so that both equations are using static 1g acceleration.

The reason that simple support boundary conditions are typically and accurately assumed for loading a single plate panel relates to the rotational restraint provided by the adjacent panel, i.e., it is assumed to be zero. Therefore, the boundary of the loaded panel will have a nonzero slope as shown below in Figure 5-14 (a), i.e., the slope of the deck plate at the stiffeners is non-zero. When two adjacent panels are equally loaded the rotation along the boundary between the two plates are offset by each other resulting in a zero slope at the boundary and effecting a fixed boundary condition.

A quick review of the tire footprints for the vehicles in Appendix C and Table 5-22, Table 5-23 and Table 5-24 shows that there are three different relationships between the tire widths of the loadout and the stiffener spacing on the vehicle deck:

1. Width of tire less than stiffener spacing, Figure 5-14 (a), – this is the typical scenario and will allow for the straight forward application of the equations presented by ABS.
2. Width of tire nominally greater than the stiffener spacing, Figure 5-14 (b), – deck thickness requirements may be reduced compared to those predicted from the ABS equations because portion of tire footprint that overlaps into adjacent panel will resist rotation of plate panel along the longitudinal stiffener and shift the response of the deck structure. This potential reduction will also have to be investigated for these same footprints when they are centered on the stiffener instead of the plate panel. This may be the governing condition that eliminates the possibility for plate reduction below the ABS formulation requirements.
3. Width of tire footprint significantly greater than stiffener spacing, Figure 5-14 (c), – Some tire footprints are approximately twice the stiffener spacing and can result in two adjacent plate panels simultaneously loaded. The reaction in the plating along the stiffener that separates these two panels will be significantly higher than predicted by the ABS equations assuming simple support.

Therefore, the end rotations resulting in the plate panels of Figure 5-14 (b) & (c) will differ from those assumed by the ABS equations, which will cause a different structural response than assumed by the ABS equations. The change in structural response results in a different set of moment distributions acting through the plate panels. This will be demonstrated by a few textbook examples illustrated below, see Figure 5-15 and Figure 5-16. Complete investigation of the various relationships between tire footprint, stiffener spacing and the resulting impact to the deck plating are investigated later in this project.

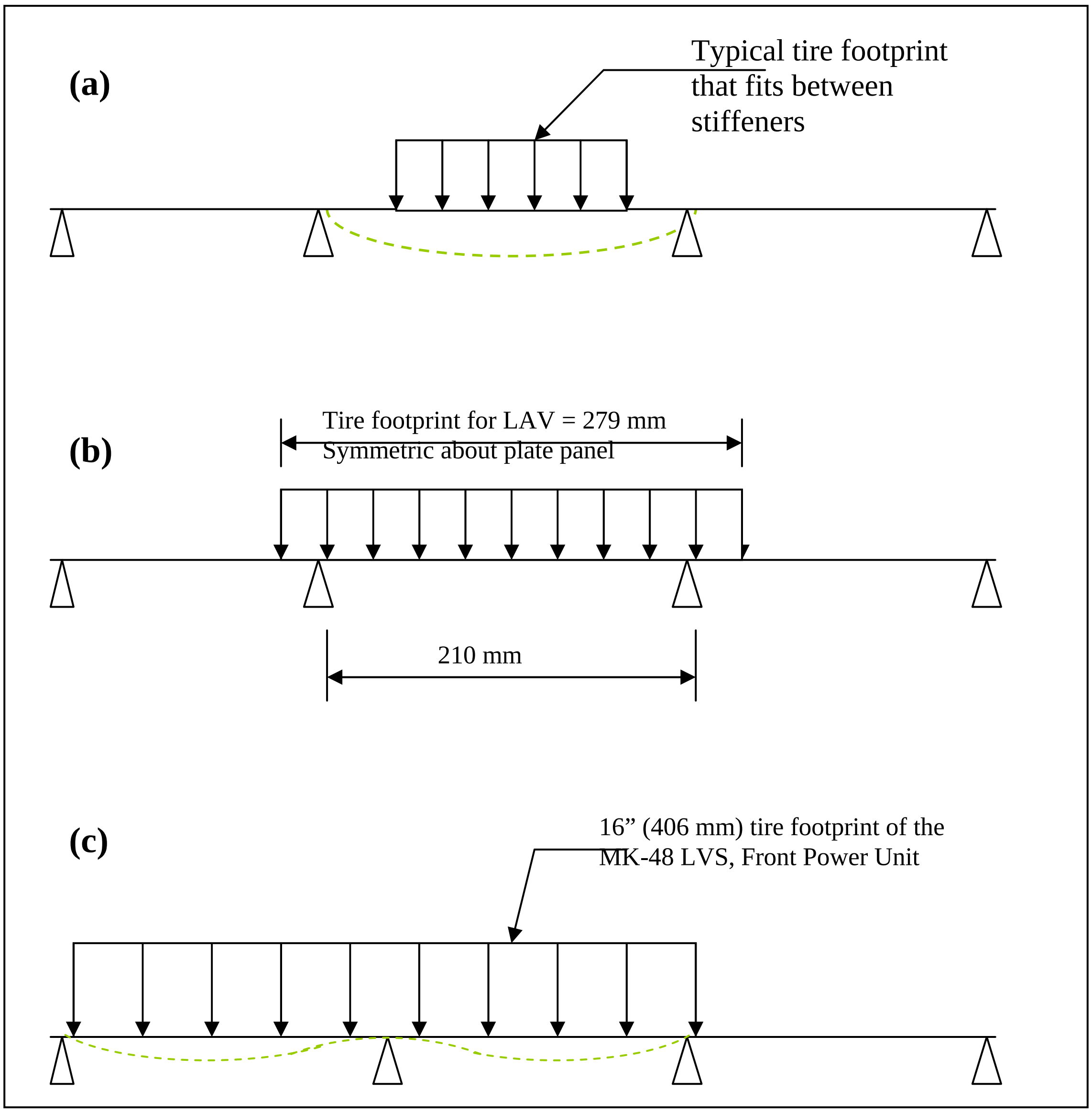


Figure 5-14. Relationship of Tire Footprint Width to Stiffener Spacing

5.6.5 Continuous Span Beams with Various Load Scenarios

Detailed analysis will have to be conducted for the situations that arise from circumstances similar to Figure 5-14 (b). This type of overlap, it may not be symmetric as shown in the figure above, is not covered as a typical situation in any textbook and direct analysis will be conducted later in this project. The results of these analyses are included in this report.

However, the loading conditions in Figure 5-14 (a) & (c) can be approximated as shown below in Figure 5-15 and Figure 5-16, respectively.

For a continuous span beam with only one span loaded, Figure 5-15 presents the resulting load, shear and moment diagrams:

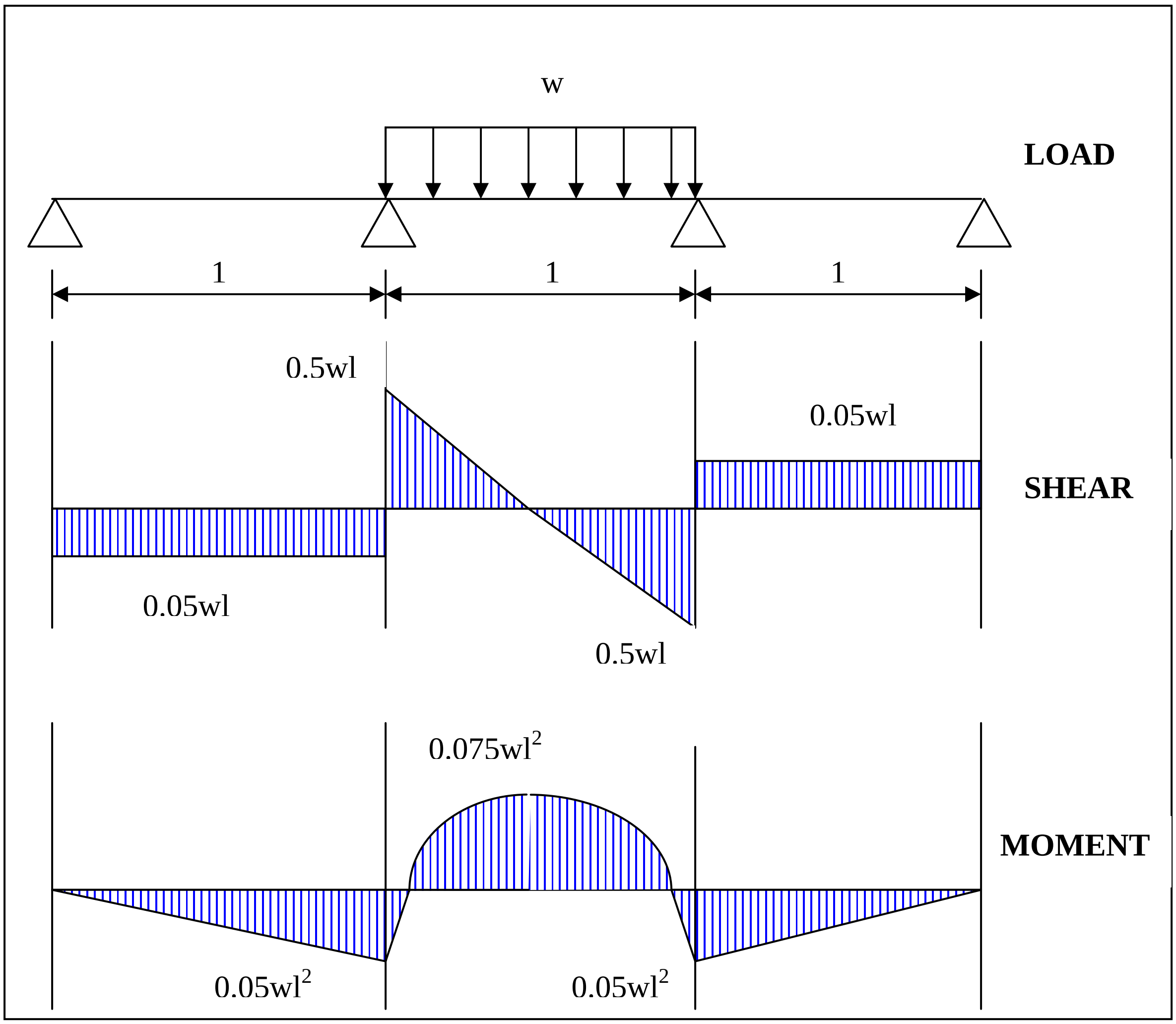


Figure 5-15. Load, Shear & Moment Diagrams for Continuous Span Beam – One Span Loaded

For a continuous span beam with two spans loaded, Figure 5-16 presents the resulting load, shear and moment diagrams:

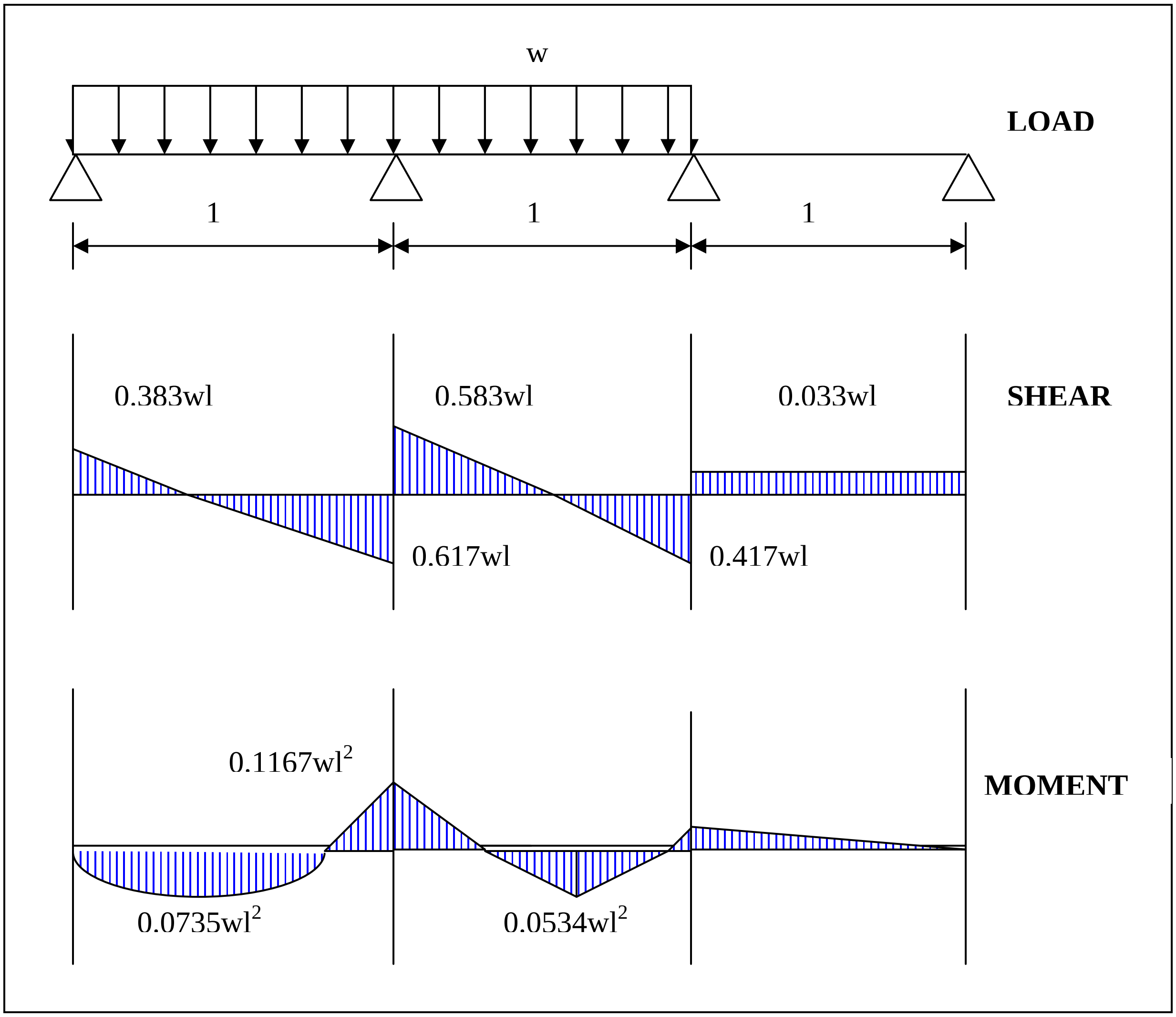


Figure 5-16. Load, Shear & Moment Diagrams for Continuous Span Beam – Two Adjacent Spans Loaded

Figure 5-15 illustrates the moment diagram resulting when there is only a single span loaded in a continuous span beam. Most tire footprints would not load the entire span, only a portion. Regardless, the same basic diagrams would result for a centrally loaded span, i.e., the maximum moment is at the center of the plate panel. The maximum moment in Figure 5-15 is equal to 0.075 wl^2. The moment will be less than this maximum value if only a portion of the span is loaded. Figure 5-16, on the other hand, shows that the maximum value for two adjacent spans uniformly loaded is equal to 0.1167 wl^2, which is greater than the value of 0.075wl^2. The location of the maximum moment has also shifted from the center of the plate to the support, i.e., the stiffener. This is a clear demonstration that the assumption of simple support boundary conditions (assumed in the ABS equations) will not be sufficient for tire footprints that are twice the value of the stiffener spacing. It also suggests that there will be some intermediate value of tire footprint that is greater than the stiffener spacing but less than twice the spacing that might benefit the moment distribution in the plate panel by reducing the moment at the center of the plate but has not yet caused the moment at the supports to be the governing value. This will be further investigated later in this project.

5.6.6 ABS Aluminum Extrusion Properties for 6061-T6

As stated above, the Main Vehicle Deck is fabricated using extrusions that are 6082-T6 (or 6061-T6). ABS does not provide properties for any 6082 aluminum alloys but does provide data in Table 3.3.2 for 6061-T6 as follows:

6061-T6 Ultimate Tensile Strength = U_{al} = 16.9 kg/mm^2 = 165.5 N/mm^2 = 24,000 psi
Welded Yield Strength = Y_{al} = 14.13 kg/mm^2 = 137.9 N/mm^2 = 20,000 psi

These strengths are for the welded alloy using 5183, 5356 or 5556 filler wires and can be used in the ABS equations provided above. In the unwelded condition, 6061-T6 has a minimum yield strength of 24.6 kg/mm^2 = 241.3 N/mm^2 = 35,000 psi. ABS Table 3.3.2 also presents welded material properties using other filler wires, which are lower than those shown above. ABS suggests that the superior filler wires and corresponding properties shown above are the most commonly used.

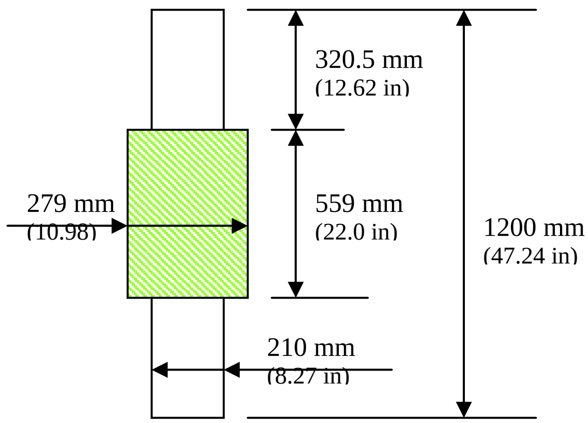
5.6.7 Loads in Way of the Light Armored Vehicles (LAV) & Required Deck Plate

The military vehicle payload for the converted vessel specifies eight total LAV's with weights ranging from 23,300 pounds to 28,400 pounds. As shown in Table 5-22 this does not represent the worst pressure load resulting from the tire loads however, with eight LAV's in the loadout this is a good vehicle to investigate as it represents a significant portion of the loadout. As such, this represents a good baseline vehicle for determination of the tire footprint loads and the resulting structural requirements. It also represents the intermediate tire width depicted in Figure 5-14 (b) and allows for discussion on the preliminary design approach for the deck plate in way of these vehicles.

Four of the LAV's are pictured below in Figure 5-18. They all have the same basic configuration including eight tires distributed over four axles. It has been assumed that their tires have individual footprints approximated by the Medium Cargo Truck Load as shown in Appendix C, Figure C-2. Assuming that the tires are loaded in the same ratio to the total vehicle weight as shown in Appendix C, it is approximated that the maximum tire load associated with the 28,400 pound LAV (R) is 3770 pounds, 16,770 N. It is also assumed that the tires have footprints that are similar to those of the Medium Cargo Truck, i.e., 22" x 11" or 559 mm x 279 mm.

In all likelihood, the LAV's will have a predominantly longitudinal orientation on the PacifiCat and never have a transverse orientation. Even if they do have a transverse orientation the tire footprint is 559 mm, which means that the 3770 pound tire load would be distributed over more than two plate panels, i.e., the tire load on the plate panel would be a lot smaller than the longitudinal orientation. Therefore, this situation can be ignored for deck plating requirements.

A schematic of the Heavy Extrusion plate panel with the tire located at its center is shown below in Figure 5-17. Note – it is not possible for the complete tire load to be on one plate panel. This is accounted for in the 2nd calculation shown below which uses a tire load equal to (210/279) x 3770 pounds = 2838 pounds.



For this arrangement:

$a/s = 210/210 = 1.0$

$b/s = 559/210 = 2.66$

$l/s = 1200/210 = 5.7$

From ABS: $\beta = 0.30$

Figure 5-17. Schematic Diagram of LAV Wheel Load on Heavy Extrusion

For:

$\beta = 0.30$, $W = 3770$ pounds, $n_{xx} = 0.24 \times 2$ (at the bow) and $\sigma_a = 20{,}000 \times 0.6 = 12000$ psi

and using the equation provided above for the English units, the required ABS plate thickness becomes:

$$t = \sqrt{\frac{0.3(3770)(1+0.5(0.48))}{12000}} = 0.342 \text{ in} = 8.68 \text{ mm}$$

This is slightly less than the 9.80 mm plate thickness available in the heavier extrusion. However, if it is considered that the actual load on the plate panel is only (210/279) x 3770 = 2838 pounds then the required plate thickness becomes:

$$t = \sqrt{\frac{0.3(2838)(1+0.5(0.48))}{12000}} = 0.297 \text{ in} = 7.54 \text{ mm}$$

Also, recalling from the discussion presented above, the tire load on the adjacent panels may reduce the stress in the centrally loaded panels by helping to restrain rotation about that boundary of the plate. This would further reduce the required plate thickness determined above, effectively adding to the margin already seen in the last calculation.

Additional calculations will be performed for the LAV's to investigate the deck plate requirements when the tire is centered on the stiffener instead of the plate panel. These calculations and results are developed later in this project and presented in the final report.

These calculations demonstrate that the LAV's (Figure 5-18) could be parked at any location of the heavier extrusion. The last calculation also indicates that the LAV's could be parked on the lighter extrusion.

LAV Logistics (L)

LAV Command & Control (C2)

LAV Anti-Tank (AT)

LAV Recovery (R)

Figure 5-18. Various Configurations of the Light Armored Vehicle, LAV

5.6.8 Other Vehicle Deck Plate Requirements IAW ABS HSNC Rule

Brief calculations shall be presented for some of the other critical vehicles included in the loadout.

5.6.8.1 Armored Combat Excavator

This is a tracked vehicle with a footprint similar to that shown in Appendix C, Figure C-5. The total track area is 2 x (25" x 180") = 9000 in^2. With a vehicle weight of 36,000 pounds the average deck pressure = 36,000/9000 = 4 psi. Since this will certainly load numerous deck plate panels simultaneously, it can be assumed that the panel experiences a uniform load of 4 psi over its entire surface. The plate panel area is (210 mm x 1200 mm)/(25.4^2) = 391 in^2 resulting in a load W = 391 x 4 = 1564 pounds. Plugging this into the ABS equation with β = 0.5 results in a required plate thickness of t = 0.328 in = 8.4 mm. This is a very conservative estimate of the required plate thickness, which could be more accurately determined using the equations for a uniformly loaded plate over its entire surface. Regardless, the deck plate in way of the Heavy Extrusion is acceptable and, in all likelihood, so is the deck plate of the Light Extrusion.

5.6.8.2 MK-48 LVS, Front Power Unit

The tire loads resulting from this unit impart the largest pressure load on the deck, 15.8 psi over an area of 16 in x 27 in as presented in Table 5-22. The maximum tire load is 6830 pounds assumed to be uniformly distributed over the 16 in x 27 in (406.4 mm x 686 mm) footprint. Based on the 210mm (8.25 in) plate panel, the load on the panel is (8.27/16) x 6830 = 3530 pounds, which is equal to the load W in the ABS equation. With a/s = 210/210 = 1 and b/s = 686/1200 = 0.57 and l/s = 1200/210 = 5.7 implies that β = 0.57. Using these values results in an ABS required plate panel thickness of:

$$t = \sqrt{\frac{0.57(3530)(1+0.5(0.48)\ )}{12000}} = 0.456\text{in} = 11.58 \text{ mm}$$

This thickness assumes that the boundary conditions for the plate panel are simply supported. As demonstrated by the loading and moment diagram information in Figure 5-16, this is not the case. As shown in the ABS equation for plate thickness, it can be demonstrated that the required thickness will vary as the square root of the moment, i.e., if the load W is doubled, the moment acting on the plate panel will double but the required plate thickness will only increase as $(2 \text{ x } W)^{1/2} = 1.414W$. This is further reinforced recognizing that the section modulus for a unit width plate strip is calculated as:

$$SM_{plate} = t^2/6$$

Where t is the thickness of the plate panel. This confirms that the ability of a plate panel to resist an increasing moment varies as the square of its thickness. Using this information, and the calculated required plate thickness of 0.456 inches, it can be calculated that the actual required deck plate thickness in way of the tire footprint on the MK-48 LVS will be:

$$[(0.1167wl^2)/(0.075wl^2)]^{1/2} \text{ x } 0.456 = 0.568 \text{ in} = 14.4 \text{ mm}$$

This represents a significant increase to the existing deck plate, which is only 9.8 mm or 8.0 mm, depending on the location of the vehicle deck. (Actual requirement in way of 8.0 mm deck plate would be recalculated to account for the closer stiffener spacing.) Various options will be investigated to minimize this impact in way of this severe tire load. These will include:

- The use of locally thickened deck plate to accommodate these vehicles. Other vehicles having similar, severe requirements will be determined and accommodated in a localized area to minimize the amount of structural modifications.

- Re-locate the vehicles so that they are closer to the center of gravity of the ship and will not experience the maximum value of design vertical acceleration, n_{xx}, as assumed for these calculations.
- Investigate the use of relaxed allowable stresses, which are currently based on the typical ABS design values for the design of a new vessel. Argument can be made that, for military application, there would be little damage allowing the plate to go to yield and perhaps experience slight permanent deformation.
- Finite element analysis to more accurately predict the structural response of the vehicle deck structure.

The weight penalties associated with the various options will be investigated along with approximate cost and payload impacts.

5.6.8.3 Trailer, Cargo 3/4 T (M101A1)

This is one of the few vehicles for which the entire footprint can be contained on a single plate panel. The plate requirements for this wheel load will be checked for both of the extrusions. The footprint is 7 in x 12 in (178 mm x 305 mm).

- Heavy Extrusion

a/s = 178/210 = 0.85 b/s = 305/210 = 1.45 l/s = 1200/210 = 5.7 $\Rightarrow \beta = 0.46$
W = 925 pounds n_{xx} = 2 x 0.24 = 0.48 σ_a =20,000 psi

$$t = \sqrt{\frac{0.46(925)(1+0.5(0.48))}{12000}} = 0.210\text{in} = 5.33\text{ mm}$$

- Light Extrusion

a/s = 178/200 = 0.89 b/s = 305/200 = 1.53 l/s = 1200/200 = 6.0 $\Rightarrow \beta = 0.44$
W = 925 pounds n_{xx} = 2 x 0.24 = 0.48 σ_a =20,000 psi

$$t = \sqrt{\frac{0.44(925)(1+0.5(0.48))}{12000}} = 0.205\text{in} = 5.21\text{ mm}$$

In all cases, the deck plate structure in way of the Trailer, Cargo 3/4 T (M101A1) tire load is acceptable.

5.6.9 Vehicle Deck Longitudinal IAW ABS

The Vehicle Deck longitudinal stiffeners are designed in accordance with ABS HSNC Rules 3-2-4 [1]. In accordance with Table 1 from this section of the ABS Rules the allowable design stress for aluminum deck longitudinal stiffeners that form a portion of an effective strength deck is:

$$\sigma_a = 0.33\sigma_y = 0.33 \text{ x } 20{,}000 = 6600 \text{ psi}$$

This is a relatively small value for allowable stresses in deck longitudinals, which although they form a portion of a strength deck, may not be a portion of the uppermost strength deck and therefore may not be subjected to the extreme fiber stresses of the uppermost strength deck.

The ABS rules do not specify any calculation procedures, requiring only that all possible loading combinations be investigated for vehicle deck longitudinals. This implies that ABS is accepting a first principles approach to the determination of the shear and bending moment acting in the deck longitudinals. Therefore, a simplified analysis using a single span beam with a load along its length will be used to generate the shear and moment acting in the longitudinals. The load acting along the length of the beam represents the tire footprint acting on the deck.

Regardless, for this phase of the project the shear and moment acting in the longitudinal stiffeners of the Vehicle Deck will be calculated from the scenario shown in Figure 5-19.

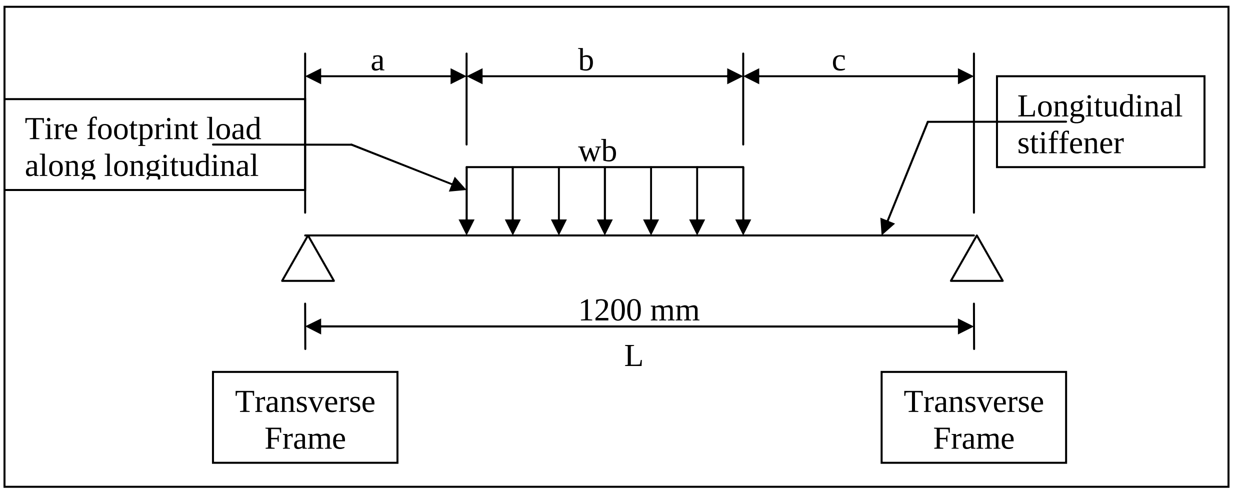


Figure 5-19. Deck Longitudinal Loading Diagram for ABS Calculations

The maximum moment occurs at the midspan of the beam when a = c and is calculated as [7]:

$$M_{\max} = \frac{wb}{2}\left(a + \frac{b}{4}\right)$$

The maximum reaction/shear force will occur when either “a” or “c” is equal to zero and can be calculated as [7]: (Assuming a = 0)

$$V_{\max} = \frac{wb}{2L}\left(2c + b\right)$$

These values for M_{max} and V_{max} are both conservative because they are based on a single span beam, not the continuous span arrangement that actually exists on the vehicle deck. This estimate is provided for now in order to assess the fundamental requirements for the ABS deck longitudinals and establish the order of magnitude of the strength requirements for these elements. More refined analysis will be performed as the project continues. The results of these analyses are included in the final report.

An important consideration is to determine the number of longitudinal stiffeners that will resist a given tire load. For instance, the tire footprint on the MK-48 LVS Front Power Unit have a 16 inch width which, in accordance with Figure 5-16, will never engage less than three longitudinal stiffeners.

Table 5-25 presents the shear force and bending moment to be resisted by the deck longitudinals by various tire footprints. The table also includes the number of deck longitudinals that are assumed to be engaged in resisting these forces.

The shear and moment values presented in Table 5-25 use the single span beam configuration shown in Figure 5-19. As discussed above, this produces conservative results compared to the continuous span beams that can actually be assumed for these analyses. As a comparison, a single span beam uniformly loaded along its entire span will develop a maximum moment of $wl^2/8 = 0.125\ wl^2$. From above, Figure 5-15, it can be seen that a continuous span beam with a single span uniformly loaded will develop a maximum moment of $0.075wl^2$. Thus, the continuous span beam develops a maximum moment that is only 60% of the maximum moment in a single span beam subjected to the same load. It is expected that similar reductions will result upon detailed analysis, i.e., finite element analysis, of the current loading scenarios.

Table 5-25. Vehicle Deck Longitudinal Stiffener Shear & Moment Loads – ABS Design – English Units

Vehicle	Tire Load (lbs)	Footprint on long'l (in)	Load, *w* (lb/in)	Total Shear, V (lb)	Total Moment, M (in-lb)	Number of deck long's	Shear/long'l (lb)	Moment/long'l (in-lb)	Req'd Section Modulus (in^3)
MK-48 LVS Front Power Unit	6830	27	253	4879	57,606	3	1627	19,202	2.9
Truck 7-T, MTVR	6100	25	244	4486	52,966	2	2243	26,483	4.1
Truck Tow Carrier, HMMWV	2000	14	143	1705	20,135	1	1705	20,135	3.1
Truck, Light Strike Vehicle (LSV)	1200	9	134	1091	12,883	1	1091	12,883	2.0
LAV Logistics (L)	3743	22	171	2886	34,074	1*	2886	34,074	5.2

Note: * The 11 inch breadth of this footprint, if centered on a specific deck longitudinal, will overlap into the effective breadth of deck plate of the two adjacent longitudinals, engaging each of them in the reaction to resist the imposed loads. Specifying 1 deck longitudinal in this table is very likely conservative. Detailed analysis of this loading arrangement will be investigated as the project continues.

ALL OF THE LOAD, SHEAR AND MOMENT VALUES IN THIS TABLE REFLECT A STATIC 1G LOAD.

The available section modulus of the longitudinal stiffeners in the Heavy Extrusion has been estimated as 4.9 in^3 (80.3 cm^3). With an allowable stress of 6600 psi this suggests that the maximum moment that can be tolerated by the longitudinal stiffeners on the Heavy Extrusion is:

Maximum Allowable Moment, Heavy Extrusion = 4.9 x 6600 = 32,340 in-lb.

As shown in Table 5-25, the moment resulting from the LAV Logistics vehicles exceeds the capacity of the existing deck stiffening. Final decision regarding the capacity of the deck stiffening will be deferred until more detailed analysis can be performed that will include the beneficial effects of the continuous span as well as the penalizing effects of the design vertical acceleration.

The maximum allowable shear strength will be taken as 60% of the allowable bending stress:

$$\sigma_{v\ all} = 0.6 \times 6600 = 3960 \text{ psi}$$

The available shear area in stiffening of the Heavy Extrusions is 1.31 in^2 resulting in a maximum allowable shear force of:

Maximum Allowable Shear, Heavy Extrusion = 1.31 x 3960 = 5188 lb.

From Table 5-25 above, it is seen that the maximum shear force is 2886 pounds and therefore the existing deck longitudinal stiffeners can satisfactorily resist all shear forces.

In general, it looks like the deck stiffening will be strong enough to accommodate the wheel loads resulting from the military vehicle payload specified for this conversion study.

5.6.10 Vehicle Deck Structure – DNV HSLC and Naval Vessel Rules

The following calculations will determine the Vehicle Deck plate and stiffening requirements using the same vehicles with DNV criteria as just used for the ABS calculations.

5.6.10.1 Vehicle Deck Plating – DNV Criteria

The DNV criteria for the design of deck structure subjected to vehicle wheel loads are contained in DNV Part 5, Chapter 2, Section 3 [2] of their rules. The design pressure to be taken over the area of the footprint is calculated as:

$$p = \frac{Q}{n_o ab}\left(9.81 + 0.5a_v\right) \qquad (\text{kN/m}^2)$$

Where: Q = maximum axle load, metric tones
n_0 = number of tires on the axle
a = tire footprint dimension parallel to stiffeners, meters
b = tire footprint dimension perpendicular to stiffeners, meters
a_v = design vertical acceleration for the craft, m/s^2

For the consideration of this design the value of a_v will be limited to the design values determined above and repeated below. For general DNV design practice it would be necessary to also check values of $a_v = 6/\sqrt{Q}$ for moving cargo handling apparatus in port. This is not a consideration for the current conversion project since no cargo handling operations are assumed for this project.

$a_{cg} = 23.5$ m/s^2 for Naval Surface Craft, Unrestricted notation

Once the design pressure, p, is determined it is used in the equation to determine the required plate thickness from DNV 5-2-3/A501 [2] which refers to the same formula used for steel deck plating in A201 with different definitions for the bending/stress parameter, mσ, which is all defined below:

$$t = \frac{77.4 k_a \sqrt{k_w csp}}{\sqrt{m\sigma}} + t_k \qquad \text{(mm)}$$

Where: $k_a = 1.1 - 0.25 s/l$
maximum 1.0 for $s/l = 0.4$
minimum 0.85 for $s/l = 0.1$

$k_w = 1.3 - \dfrac{4.2}{\left(\dfrac{a}{s} + 1.8\right)^2}$, maximum 1.0 for $a \geq 1.94s$, a = tire footprint dimension parallel to the deck stiffening

c = b for $b < s$
= s for $b > s$

s = stiffener spacing

$$m = \frac{38}{\left(\frac{b}{s}\right)^2 - 4.7\frac{b}{s} + 6.5} \quad \text{for } b/s \leq 1.0$$

m = 13.57 for b/s > 1.0

From DNV 5-2-3/A501 [2], the value of σ is to be determined as:

$$\sigma = \sigma_0 f_1$$

Where: σ_0 = 180 N/mm^2 (maximum) in general for seagoing conditions
= 210 N/mm^2 (maximum) in general for harbor conditions
f_1 = as given in DNV 3-3-1/A200 [2] with respect to plate material yield stress given in 3-3-2/Table B1, B2, B3 and B4

For these applications for deck plate: σ_0 = 180 N/mm^2 f_1 = 0.48 (welded condition Table B4) ⇒ σ = 86.4 N/mm^2

For the most direct comparison to ABS, the calculations for the Trailer, Cargo 3/4 T (M101A1) will be performed for both the Heavy and Light Extrusion. This was the only tire footprint that fit within the stiffener spacing and is therefore felt to yield the most accurate comparison using the ABS and DNV equations because it satisfies the assumed boundary conditions, as discussed above.

5.6.10.2 Trailer, Cargo 3/4 T (M101A1)

This is one of the few vehicles for which the entire footprint can be contained on a single plate panel. The plate requirements for this wheel load are checked for both of the extrusions. The footprint is 7 in x 12 in (178 mm x 305 mm).

Q = 1850 pounds = 1850/2204 = 0.84 metric tonnes (1850 lbs from Table 5-22)
n_0 = 2
a_{cg} = 23.5 m/s^2

- Heavy Extrusion

a = 0.305 meters b = 0.178 meters s = 0.210 meters l = 1.2 meters

$$p = \frac{0.84}{2(0.305)(0.178)}(9.81 + 0.5(23.5)) = 166.8 \text{ kN/m}^2$$

To determine the required deck plate, the corresponding variables are calculated as:

k_a = 1.1 – 0.25(.210)/(1.200) = 1.06 use max of 1.0

$$k_w = 1.3 - \frac{4.2}{\left(\frac{0.305}{0.210} + 1.8\right)^2} = 0.903$$

c = 0.178 meters s = 0.210 meters p = 166.8 kN/m^2 b/s = 0.8476 ⇒ m = 11.75

From which the required deck plate becomes:

$$t = \frac{77.4(1.0)\sqrt{(0.903)(0.178)(0.210)(166.8)}}{\sqrt{11.75x86.4}} + t_k = (5.76 + t_k) \text{ mm}$$

- Light Extrusion

a = 0.305 meters b = 0.178 meters s = 0.200 meters l = 1.2 meters

$$p = \frac{0.84}{2(0.305)(0.178)}(9.81 + 0.5(23.5)) = 166.8 \text{ kN/m}^2$$

To determine the required deck plate, the corresponding variables are calculated as:

k_a = 1.1 – 0.25(.200)/(1.200) = 1.04 use max of 1.0

$$k_w = 1.3 - \frac{4.2}{\left(\frac{0.305}{0.200} + 1.8\right)^2} = 0.920$$

c = 0.178 meters s = 0.200 meters p = 166.8 kN/m^2 b/s = 0.89 $\Rightarrow$ m = 12.22

From which the required deck plate becomes:

$$t = \frac{77.4(1.0)\sqrt{(0.920)(0.178)(0.200)(166.8)}}{\sqrt{12.22x86.4}} + t_k = (5.57 + t_k) \text{ mm}$$

The only reference for t_k was found for pneumatic tires in DNV 5-2-3/A202 [2], where it is defined as a corrosion addition with a value of t_k = 0 mm. Therefore, nothing will be added to the deck plate values calculated above. Discussion with DNV confirms that this variable is left over from steel vessel application and need not be considered for aluminum HSC.

From above, it is seen that this is close to, but slightly heavier than the deck plate calculated by the ABS equation, which determined a required deck plate of 5.33 mm for the Heavy Extrusion and 5.21 mm for the Light Extrusion. Both societies use similar allowable stresses based on the welded condition of the material. The ABS design uses an allowable stress of 12,000 psi whereas the DNV design uses the value of 12,530 psi (0.48 x 180 N/mm^2). In practical application, both rule set calculations would result in the use of 6 mm plate for this vehicle although some builders might select 5.5 mm plate if designing to the ABS criteria.

5.6.10.3 Vehicle Deck Plate – Light Armored Vehicle

The coefficient "m' used in the DNV plate formulation includes an option for width of tire greater than the stiffener spacing, i.e., b/s > 1 $\Rightarrow$ m = 13.57, as shown above. The following information is also used to generate the required vehicle deck plate in way of the LAV using the DNV criteria.

$$p = \frac{Q}{n_o ab}(9.81 + 0.5a_v)$$

Q = 3770 x 2/2204 = 3.421 m tonnes (3770 lbs from Table 5-22)
n_0 = 2 a = 0.559 meters (22") b = 0.279 meters (11") a_v = 23.5 m/s^2

$$p = \frac{3.421}{2(0.559)(0.279)}(9.81 + 0.5(23.5))$$ = 236.5 kN/m^2

To determine the required deck plate, the corresponding variables are calculated as:

k_a = 1.1 – 0.25(.210)/(1.200) = 1.06 use max of 1.0

$k_w = 1.3 - \frac{4.2}{\left(\frac{0.559}{0.210} + 1.8\right)^2}$ = 1.09 ⇒ use maximum value of 1.0 for a ≥ 1.94s

c = 0.210 meters s = 0.210 meters p = 236.5 kN/m^2 b/s = 1.33 ⇒ m = 13.57

From which the required deck plate becomes:

$$t = \frac{77.4(1.0)\sqrt{(1.0)(0.210)(0.210)(236.5)}}{\sqrt{13.57x86.4}} + t_k$$ = (7.30 + t_k) mm

The deck plate required using the ABS criteria is 7.54 mm, again almost identical to that required using DNV.

5.6.10.4 Vehicle Deck Plate - MK-48 LVS, Front Power Unit

This represents the extreme case analyzed above using a combination of the ABS design equations and other factors to estimate the Vehicle Deck plate thickness in way of this vehicle. The following calculations present the Vehicle Deck plate required in way of this vehicle using DNV criteria.

$$p = \frac{Q}{n_o ab}(9.81 + 0.5a_v)$$

Q = 6830 x 2/2204 = 6.198 m tones (6830 lbs from Table 5-22)
n_0 = 2 a = 0.686 meters (27”) b = 0.406 meters (16”) a_v = 23.5 m/s^2

$$p = \frac{6.198}{2(0.686)(0.406)}(9.81 + 0.5(23.5))$$ = 239.9 kN/m^2

To determine the required deck plate, the corresponding variables are calculated as:

k_a = 1.1 – 0.25(.210)/(1.200) = 1.06 use max of 1.0

$k_w = 1.3 - \frac{4.2}{\left(\frac{0.686}{0.210} + 1.8\right)^2}$ = 1.14 ⇒ use maximum value of 1.0 for a ≥ 1.94s

c = 0.210 meters   s = 0.210 meters   p = 239.9 kN/m^2 b/s = 1.93 $\Rightarrow$ m = 13.57

From which the required deck plate becomes:

$$t = \frac{77.4(1.0)\sqrt{(1.0)(0.210)(0.210)(239.9)}}{\sqrt{13.57x86.4}} + t_k \; = (7.35 + t_k)\text{ mm}$$

This is significantly lighter than the deck plate determined using the modified ABS approach, 11.57mm. Additional work will be done to investigate the required deck plate in way of this vehicle. This will include the development of simple finite element models to assess the deck requirements when subjected to this wheel load.

It will be assumed for now that similar results will happen using the DNV analysis for the other wheel loads, i.e., the DNV equations will result in smaller deck thickness requirements than those from ABS. More detailed development of these calculations are preformed later in this project with the results included in the final report.

Noting that the plate thickness required from the DNV formulations is less severe than that resulting from ABS can be attributed to various factors, perhaps the most obvious being the allowable stress used in the two different rules. Regardless, this result also confirms that the DNV equations are based primarily on the assumption of simple support boundary conditions for the plate, except for the consideration of the coefficient, “m”.

Table 5-26 presents a summary of the ABS and DNV Vehicle Deck plate thicknesses calculated in this report.

Table 5-26. Comparison of ABS & DNV Vehicle Deck Plate Requirements

Vehicle	ABS Req’d Plate (mm)	DNV Req’d Plate (mm)
Light Armored Vehicle, LAV*	7.54	7.27
MK-48 LVS, Front Power Unit*	11.57	7.35
Trailer, Cargo 3/4 T (M101A1) Heavy Extrusion	5.33	5.76
Trailer, Cargo 3/4 T (M101A1) Light Extrusion	5.21	5.57

Note * These vehicles have tire footprints where the smaller footprint dimension exceeds the stiffener spacing. Detailed calculations will be performed to more accurately determine the Vehicle Deck plate requirements. ABS does not include any contingency in their design algorithms for such a tire footprint.

5.6.11 Vehicle Deck Stiffening – DNV Criteria

The criteria for aluminum deck stiffening is presented in DNV 5-2-3/A600 [2], which refers to 5-2-3/A300 for use of the same equations as presented for steel, substituting different allowable stress in the form of $\sigma = \sigma_0 f_1$. The equation to be used for section A300 is given as:

$$Z = \frac{1000 k_z l c d p}{m\sigma} + Z_k$$

for use in this application:

$\sigma_0 = 160$ N/mm^2 $f_1 = 0.48 \Rightarrow \sigma = 76.8$ N/mm^2 (11,140 psi compared to 6600 psi for ABS)

$k_z = 1.0$ for $b/s < 0.6$ and $b/s > 3.4$

$k_z = \left(1.15 - 0.25\frac{b}{s}\right)$ for $0.6 < b/s < 1.0$

$k_z = \left(1.15 - 0.25\frac{b}{s}\right)\frac{b}{s}$ for $1.0 < b/s < 3.4$

$c =$ b for $b < s$
$=$ s for $b > s$

d = a for a < l
= l for a > l

a, b and p as given above

$$m = \frac{r}{\left(\frac{a}{l}\right)^2 - 4.7\frac{a}{l} + 6.5} \text{ for } a/l \le 1.0$$

$$m = \frac{87}{\left(\frac{a}{l}\right)^2\left[\left(\frac{a}{l}\right)^2 - 6.3\frac{a}{l} + 10.9\right]} \text{ for } 1.2 < a/l \le 2.5$$

$m = 12$ for $a/l \ge 3.5$

r = factor depending on the rigidity of the girders supporting the continuous stiffeners, taken as 29 unless better support conditions are demonstrated.
= 38 when continuous stiffener may be considered as rigidly supported at each girder.

r shall be taken as 29 for these calculations.

No definition for Z_k was available in the DNV rules [2].

Table 5-27 provides a summary of the relevant input data and load calculation, p, for the design pressure acting on the vehicle deck to be used as input for the required section modulus calculation, Z.

Table 5-28 is provided below to show the DNV section modulus requirements for the same vehicles that were used for ABS and presented in Table 5-25. In order to provide a simple comparison the value of a_v, design vertical acceleration, was taken equal to zero so that both the ABS and DNV stiffener requirements are currently based on static 1g loads. Similar to ABS, a_v will be included as the design progresses.

Table 5-27. Vehicle Data & Load Calculation for DNV Stiffeners

Vehicle	Axle Load, Q (tonnes)	"a" (m)	"b" (m)	No of tires, n_o	Design Pressure, p (kN/m^2)
MK-48 LVS Front Power Unit	6.202	0.686	0.406	2	109.2
Truck 7-T, MTVR	5.539	0.635	0.406	2	105.4
Truck Tow Carrier, HMMWV	1.816	0.356	0.318	2	78.7
Truck, Light Strike Vehicle (LSV)	1.090	0.229	0.229	2	102.0
LAV Logistics (L)	3.399	0.559	0.279	2	106.9

For the balance of the DNV stiffener and required section modulus calculations:

s = 0.210 meters l = 1.200 meters σ = 76.8 N/mm^2

Table 5-28. Vehicle Deck Longitudinal Stiffener Section Modulus Requirements – DNV Design – METRIC UNITS

Vehicle	"a" (m)	"b" (m)	"b/s"	k_z	"c"	"d"	"a/*l*"	m	Z (cm^3)
MK-48 LVS Front Power Unit	0.686	0.406	1.933	1.289	0.210	0.686	0.572	7.007	45.2
Truck 7-T, MTVR	0.635	0.406	1.933	1.289	0.210	0.635	0.529	6.754	41.9
Truck Tow Carrier, HMMWV	0.356	0.318	1.514	1.168	0.210	0.356	0.297	5.585	19.2
Truck, Light Strike Vehicle (LSV)	0.229	0.229	1.090	0.956	0.210	0.229	0.191	5.143	13.2
LAV Logistics (L)	0.559	0.279	1.329	1.087	0.210	0.559	0.466	6.406	33.3

These calculations assume that the vehicles are parked on the deck oriented longitudinally as shown in the parking arrangement earlier in this report. This will place the long tire footprint dimension in the longitudinal direction, i.e., parallel to the deck stiffening and equal to dimension "a", in accordance with the DNV definitions. The short dimension is "b", perpendicular to the stiffener.

ALL OF THE LOAD, SHEAR AND MOMENT VALUES IN THIS TABLE REFLECT A STATIC 1G LOAD.

Table 5-29 and Table 5-30 present comparisons of the required section modulii for the Vehicle Deck stiffening using the ABS and DNV design criteria. Table 5-29 presents the comparison using the actual section modulus values calculated above, which were all based on static 1g loads. Table 5-30 modifies the required section modulii to account for the maximum values of design vertical acceleration, as would be required for the actual conversion. The values in Table 5-30 were increased by 1.48g's to account for the ABS requirements and 3.4 g's to account for the DNV requirements. The DNV required section modulii are still below those required by ABS. Further explanation for this is provided in Table 5-31, which shows the allowable stresses for the ABS and DNV design for both plate and stiffener considerations. As indicated in this table, the allowable stress for DNV stiffener design is approximately 225% greater than the ABS allowable stiffener stress. Since all of these allowable stresses are based on the welded yield strength of the aluminum, the reduced values used by ABS imply a greater factor of safety, perhaps accounting for material degradation due to welding.

Table 5-29. Comparison of ABS & DNV Vehicle Deck Section Modulus Requirements – Static 1g Loading

Vehicle	ABS Req'd SM (cm^3)	DNV Req'd SM (cm^3)
MK-48 LVS Front Power Unit *	47.9	15.1
Truck 7-T, MTVR*	66.4	21.0
Truck Tow Carrier, HMMWV	50.4	19.2
Truck, Light Strike Vehicle (LSV)	32.0	13.2
LAV Logistics (L)	84.8	33.3

Note: * These values have been reduced from Table 5-28 to account for the number of deck longitudinal stiffeners that, as shown in Table 5-25, will resist the loads applied by these vehicles.

Table 5-30. Comparison of ABS & DNV Vehicle Deck Section Modulus Requirements with Respective Design Accelerations

Vehicle	ABS Req'd SM (cm^3)	DNV Req'd SM (cm^3)
MK-48 LVS Front Power Unit	140.0	51.3
Truck 7-T, MTVR	193.9	71.4
Truck Tow Carrier, HMMWV	147.3	65.3
Truck, Light Strike Vehicle (LSV)	93.2	44.9
LAV Logistics (L)	247.8	113.2

Table 5-31. Comparison of ABS & DNV Allowable Stresses for Plate & Stiffener Design

Classification Society	Allowable Stress, Plate N/mm^2/(psi)	Allowable Stress, Stiffener N/mm^2/(psi)
ABS	62.1/(12000)	34.1/(6600)
DNV	86.4/(12,530)	76.8/(11,140)

5.6.12 Vehicle Tie-Down Loads

It was originally expected that a typical calculation for the tie down requirements based on actual vehicle weight and ship motions would be required for this portion of the project. Reference [3], USMC Marine Lifting and Lashing Handbook, simplifies all of this. It presents tie down and lashing requirements based on vehicle weight and the type of ship used for transport. Large ships are shown to have lower requirements due to the reduced vessel motions in a seaway.

Therefore, all tie down and lashing requirements are developed using reference [3].

It was anticipated that the tie-down requirements would be similar for the two class societies and that accommodating the tie-downs would not be a big discriminator for one set of rules over the other. Using reference [3] will result in identical tie-down requirements for both conversions. Tie down arrangements and details are discussed later in the report and all tie down procurement and installation costs are included in the final cost estimate included with the report.

5.7 CONCLUSIONS TO STRUCTURAL LOADS REQUIRED FOR CONVERSION

As shown throughout the tables presented in Section 5 of this report, the ABS and DNV rules result in different hull girder and secondary slam loads to be resisted by ship structure. They also result in different Vehicle Deck requirements to satisfy the local strength criteria in this area. Many of the global and secondary slam loads predicted by DNV are larger than the corresponding loads determined through ABS. With different allowable stresses, it is not obvious which of these loading/allowable stress systems will result in the heavier hull girder and local structure required to resist the respective loads. These calculations are developed in the next section of this report to confirm the impacts from each of the class societies.

Additional calculations are also developed in the next section to finalize the impacts to the Vehicle Deck structure to accommodate this portion of the mission of the vessel.

The structural drawings of the PacifiCat are marked-up and presented in the next section of the report indicating the different structural schemes required by each of the societies. The weight estimates resulting from each of the structural modifications are also developed and presented in the next section of the report.

6.0 STRUCTURAL MODIFICATIONS REQUIRED FOR CONVERSION

6.1 INTRODUCTION

This section presents the structural modifications that are required to resist the ABS and DNV loads developed in the previous section of this report. Hull girder and local structural properties were calculated and are presented below. No strength data was available from the original design so all strength properties were calculated for this project and may differ slightly from those used in the original design.

The strength data presented below is used with the global hull girder and secondary slam loads to determine the structural modifications required for the PacifiCat to operate in compliance with the unrestricted, open-ocean criteria for both ABS and DNV. Marked-up drawings are presented that show the structural modifications required for both the ABS and DNV conversions.

Structure on the Main Vehicle Deck is also evaluated for modifications required to accommodate the military vehicle payload for both ABS and DNV. A sketch is provided that shows the installation detail for the typical vehicle tiedown.

Finally, this section also provides estimates to the range, speed and endurance for the PacifiCat after the modifications have been incorporated. This is done using the weight estimates that have also been developed for the structural conversion requirements.

6.2 HULL GIRDER PROPERTIES & REQUIRED STRUCTURE

There was no clear definition of a strength deck provided on the drawings or within the notes that were available for this task. Engineering judgment determined that the uppermost strength deck of the PacifiCat is the TIER 3 Deck, directly above the Upper Vehicle Deck, approximately 14.1 meters above baseline. The TIER 3 Deck is shown as the uppermost deck on the midship section of the PacifiCat and all side shell plating and structure appears to be continuous up to this level on the shell expansion. The structure appears to be discontinuous above TIER 3, which is consistent with the knowledge that the upper passenger deck structure is not continuous with the hull girder. These observations suggest that the TIER 3 Deck is the uppermost strength deck of the ship. The only confirmation of the strength deck is the PacifiCat brochure information shown in **Section 3.5.2 Structural**/Superstructure of this report, which cites the T3 strength deck.

Various models of the PacifiCat structure were developed for use in the HECSALV software package, where the hull girder properties were calculated. Using the appropriate effective material, hull girder properties were calculated for both longitudinal and transverse bending. The ineffective material for openings and their shadow areas was not included in the section property calculations. Transverse bending is the action of prying apart or squeezing together of the two hulls under the action of global transverse loads. The hull girder properties are shown in Table 6-1 and Table 6-2. Figure 6-2 provides a schematic that identifies various areas of the hull from the Main Vehicle Deck to the Keel of the vessel.

Table 6-1. PacifiCat Hull Girder Properties to Resist Vertical Bending at Midship

Property	Value	Value
Cross Sectional Area	13,339.97 cm^2	
Moment of Inertia	236,743.70 cm^2 m^2	
Section Modulus, Deck	34,685.58 cm^2 m	3,468,558 cm^3
Section Modulus, Keel	32,575.32 cm^2 m	3,257,532 cm^3
Shear Area	5409.67 cm^2	
NA, deck	6.83 m	
NA, keel	7.27 m	

Table 6-2. PacifiCat Hull Girder Properties to Resist Transverse Bending

Property	Value	Value
Cross Sectional Area	39,441.75 cm^2	
Moment of Inertia	350,509.90 cm^2 m^2	
Section Modulus, TIER 3	60,685.73 cm^2 m	6,068,573 cm^3
Section Modulus, Wet Deck	125,276.80 cm^2 m	12,527,680 cm^3
Shear Area	10,255.48 cm^2	
NA, deck	5.78 m	
NA, keel	2.80 m	

The information in Table 6-3 presents a summary of the global loads that the hull girder needs to resist for its original design as well as the ABS and DNV conversion designs.

Table 6-3. Summary of Global Loads Required for Original Design and Conversions

	Max Vertical Moment kN - m	Transverse kN - m	Torsional kN - m	Slam Induced kN - m	Pitch Connecting kN-m	Crest Landing kN-m	Hollow Landing kN-m
ABS Naval, Operational	161,123	127,636	273,143	101,539	NA	NA	NA
ABS Naval, Survival	161,123	104,991	224,682	83,524	NA	NA	NA
DNV Unrestricted Operational	245,774	130,527	241,633	NA	533,093	120,939	169,564
DNV Unrestricted Survival	245,774	76,901	100,873	NA	222,547	118,316	165,031
R4 – Original Design	165,031	76,900	100,873	NA	222,547	118,316	165,031

6.2.1 Hull Girder Stresses in the Original R4 Design

From Table 6-3 it is seen that the vertical bending moment for the original **R4** design was calculated as 165,031 kN-m. Using the hull girder properties from Table 6-1 results in the following stresses:

- Stress at TIER 3 Deck = 165,031,000/3,468,558 = 47.58 N/mm^2
- Stress at Keel = 165,031,000/3,257,532 = 50.66 N/mm^2

In accordance with the DNV criteria (3.3.4/B101), the allowable primary stress, σ, is taken as 175 f_1 where f_1 is a material reduction factor determined from tables within DNV to help determine the actual allowable stress for a given alloy/temper in a given function, i.e., primary or secondary. The midship section drawing indicates that the TIER 3 Deck consists of both extruded and rolled structure. As scaled from the midship section, TIER 3 is fabricated from extrusions from ship centerline to 9.4 meters off center, P/S. The outboard 2.75 meters of TIER 3 is fabricated from rolled structure. Both the Upper Vehicle Deck and the Main Vehicle Deck are fabricated from extrusions. In accordance with Drawing No. 1811/2-201 Rev D all extrusions for this ship are either 6061 T6 or 6082 T6, each with a yield strength of 115 MPa. Review of hull bottom plating and Wet Deck structural drawings does not specifically define the material used for fabrication, however review of Dwg No 1811/2-201 Rev D suggests that all these structural components are fabricated from either 5083 H321 or H116, 5086 H32 or H116 or 5383 H321 or H116. The f_1 factors for all of these materials are presented below in Table 6-4, which is excerpted from the DNV rules. The values for f_1 shown in Table 6-4 reflect the material in its welded condition. DNV also presents values of f_1 for all these materials in their unwelded conditions.

Table 6-4. Information Extracted from DNV 3.3.2/Table B4

DNV Material Designation	*Temper*	*Filler*	f_1
NV 5083	H116, H321 H116, H321	5356 5183	0.53[1] 0.60[1]
NV 5086	0, H111, H116, H32, H34	5356-5183	0.42
NV 5383	H116, H34	5183	0.64[2]
NV 6061	T4 T5/T6	5356-5183	0.48 0.48
NV 6082	T4 T5/T6	5356-5183	0.46 0.48

Notes: 1 The utilization of the material is higher than given by the f_1 factor as given in Sec 1A. This is due to extended utilization in Rules for HS, LC and NSC, $f_1 = (\sigma_1/240) \times 1.10$.

2 The utilization of the material is higher than given by the f_1 factor as given in Sec 1A. This is due to extended utilization in Rules for HS, LC and NSC, $f_1 = (\sigma_1/240) \times 1.10$.

Based on Table 6-4 it is easy to define $f_1 = 0.48$ for the entire Upper Vehicle Deck, Main Vehicle Deck and the extruded portions of TIER 3. The appropriate values for other areas of the ship are not as easily defined. Regardless, it can be seen that only one material results in a value of f_1 less than that used for the extrusions with all others being greater. As a conservative check it is assumed that the bottom shell plate and outboard strake of TIER 3 are fabricated from 5086 with $f_1 = 0.42$ resulting in an allowable stress of $\sigma = 175 \times 0.42 = 73.5$ N/mm^2.

Based on the stresses calculated above, this would indicate that the hull girder properties and loads determined for the original **R4** PacifiCat are consistent and result in an acceptable design.

Neither the original loads nor the original hull girder properties were available for this project. Similarly, it was not possible to uniquely define all the alloys in the different areas of the ship. Therefore, these calculations help to confirm the accuracy the work presented in this report.

The calculations demonstrated above can be rewritten to confirm compliance with DNV 3.3.4/B101 where:

$$Z = \frac{M}{\sigma} x 10^3 \left(cm^3\right)$$

Where: Z = Required hull girder section modulus, cm^3
M = Governing vertical bending moment, kNm
σ = allowable stress = 175 f_1, N/mm^2

From this it is seen that:

$$Z = (165{,}031/84) \times 1000 = 1{,}964{,}655 \text{ cm}^3$$

From Table 6-1 it is seen that the minimum section modulus is 3,257,532 cm^3, satisfying the DNV criteria.

To determine the hull girder stresses when the original **R4** design was subjected to transverse bending the loads from Table 6-3 and the hull girder properties from Table 6-2 are used:

- Stress at TIER 3 Deck = 76,900,000/6,068,573 = 12.67 N/mm^2
- Stress at Wet Deck = 76,900,000/12,527,680 = 6.14 N/mm^2

DNV 3-3-4/E100 defines typically acceptable stresses for transverse hull girder strength as $160f_1$. Table 6-4 shows the value for f_1 for 6061/6082 as $f_1 = 0.48$ resulting in an allowable normal stress of 76.8 N/mm^2 for the TIER 3 Deck, well above the actual stresses predicted for transverse bending. For the Wet Deck, the minimum possible value is $f_1 = 0.42$ resulting in an allowable stress of 67.2 N/mm^2, also well above the actual stresses determined above.

6.2.2 ABS and DNV Buckling Criteria

An important consideration in any design, especially a conversion that contains a lot of relatively light plating, is the buckling capacity of the structure subjected to the new loads. Both ABS and DNV use the same "first principle's" approach to this aspect of the design and, as such, they both contain the same equations and procedures to check for buckling. Although slightly redundant, both of those procedures are presented below. This is done for completeness within this report and also to provide definitions for the terms used by each society, which differ in each rule set, although the equations are all the same.

Only a portion of the buckling criteria from each rule set is reproduced in this report. Both rule sets include the calculation procedures for determining the buckling capacity for numerous situations that are not being checked in the current design, such as plate panels subjected to bi-axial compression and shear. The nature of this task allows for the determination of the principal buckling capacities. These include uni-axial compression of the strength deck plate subjected to longitudinal and transverse bending along with lateral stability of the stiffeners that support the deck. It is that information which is reproduced below.

This section of the report also includes calculations for the buckling capacity of the TIER 3 strength deck structural components, which consists of very light scantlings, deck plate that is 3.70 mm thick with 70 x 40 IT longitudinal stiffeners on 200 mm centers. Buckling of the lower flange of the vessel, i.e., the Wet Deck or Bottom Shell, will not be considered if TIER 3 is acceptable because, similar to most ships, the structure along these lower flanges is more robust than associated with the strength deck.

All calculations performed below assume uniform compression of the strength deck when subjected to transverse and longitudinal hull girder bending.

6.2.2.1 ABS Buckling Criteria

ABS presents their buckling criteria in two different areas of the rules. Plate buckling is covered within ABS HSNC 3-2-3/1.5 “Buckling Criteria” whereas stiffeners are covered in 3-2-4/1.5 “Elastic Buckling of Longitudinal Members.”

- **ABS 3-2-3/1.5.1 Uni-Axial Compression**

The ideal elastic stress is given as:

$$\sigma_E = 0.9 m_1 E \left(\frac{t_b}{s} \right)^2 \text{ (N/mm}^2\text{)}$$

Where: σ_E = ideal elastic buckling stress, N/mm^2
m_1 = buckling coefficient as given in ABS 3-2-3/Table 3
E = modulus of elasticity = 69,000 N/mm^2 for aluminum
t_b = thickness of plating, mm
s = shorter side plate panel, mm
l = longer side plate panel, mm

For the current design of PacifiCat and the TIER 3 strength deck:

For uni-axial compression on the short edge of plate (longitudinal bending):

$$t_b = 3.70 \text{ mm}; \quad s = 200 \text{ mm}; \quad l = 1200 \text{ mm}; \quad m_1 = 4.0 \Rightarrow \sigma_E = 85.01 \text{ N/mm}^2$$

For uni-axial compression on the long edge of plate (transverse bending):

$$m_1 = C_2\left[1+\left(\frac{s}{l}\right)^2\right]^2 = 1.21\left[1+\left(\frac{200}{1200}\right)^2\right]^2 = 1.278 \text{ and}$$

with C_2 = 1.21 for angle and T profile stiffeners.

t_b = 3.70 mm; s = 200 mm; l = 1200 mm; m_1 = 1.278 $\Rightarrow$ σ_E = 27.16 N/mm^2

The critical buckling stress, σ_c, is defined as:

$$\sigma_c = \sigma_E \text{ when } \sigma_E \leq 0.5\sigma_y$$

$$\sigma_c = \sigma_y\left(1-\frac{\sigma_y}{4\sigma_E}\right) \text{ when } \sigma_E > 0.5\sigma_y$$

Where: σ_y = yield stress of material, N/mm^2. Generally the unwelded yield strength may be used but due account should be made for critical or extensive weld zones.

For TIER 3 extrusions fabricated from 6061 T6, the ABS value for $\sigma_{y\ welded}$ = 137.9 N/mm^2 (20,000 psi) and the value for $\sigma_{y\ unwelded}$ = 241.3 N/mm^2 (35,000 psi). (See Table 6-5 for material properties.)

For longitudinal bending: σ_E = 85.01 N/mm^2 > $\sigma_{y\ welded}/2 \Rightarrow \sigma_c = 137.9\left(1-\frac{137.9}{4(85.01)}\right)$ = 82.0 N/mm^2 (worst case assumption, i.e., welded values for the strength of material).

For transverse bending: σ_E = 27.16 N/mm^2 < $\sigma_{y\ welded}/2 \Rightarrow \sigma_c$ = 27.16 N/mm^2 (worst case assumption).

The buckling capacity of the longitudinal stiffeners subjected to axial compression is developed in ABS 3-2-4/1.5:

- **ABS 3-2-4/1.5 Elastic Buckling of Longitudinal Members**

$$\sigma_E = \frac{EI_a}{C_1 A l^2} \text{ (N/mm}^2\text{)}$$

Where: I_a = moment of inertia, including plate flange, cm^4
C_1 = 1000 for N/mm^2
A = cross-sectional area including the plate flange, cm^2
l = stiffener span, m

The properties of the longitudinal stiffener on the TIER 3 strength deck are approximated from the 70 x 40 IT as:

I_a = 99.9 cm^4 A = 11.23 cm^2 $\Rightarrow$ σ_{el} = 426.3 N/mm^2 $\Rightarrow$ significantly greater than yield strength or allowable stress.

6.2.2.2 DNV Buckling Criteria

DNV 3-3-10 provides guidance on Buckling Control for both plate and stiffening.

The allowable, or critical compressive stress, σ_c, is a function of the ideal elastic, Euler, buckling stress, σ_{el}. These quantities are calculated below as reproduced from the appropriate sections of the DNV HSLC & NSC rules.

- **DNV 3-3-10/A102 – Relationships**

$$\sigma_c = \sigma_{el} \text{ when } \sigma_{el} < \frac{\sigma_f}{2}$$

$$\sigma_c = \sigma_f \left(1 - \frac{\sigma_f}{4\sigma_{el}}\right) \text{ when } \sigma_{el} > \frac{\sigma_f}{2}$$

Where: σ_c = critical compressive buckling stress, N/mm^2
σ_{el} = ideal elastic Euler compressive buckling stress, N/mm^2
σ_f = minimum upper yield stress of material, N/mm^2. Usually base material properties are used, but critical or extensive weld zones may have to be taken into account.

For the current design, the yield strength of the strength deck is taken from Dwg 1811/2-201D which shows all extrusions having a yield strength of σ_f = 115 N/mm^2, which varies from the value used for ABS and is used below to provide a different set of allowable stress values.

- **DNV 3-3-10/D100 – Plate Panel in uniaxial compression**

$$\sigma_{el} = 0.9kE\left(\frac{t}{1000s}\right)^2 \text{ (N/mm}^2\text{)}$$

Where: E = modulus of elasticity = 69,000 N/mm^2 for aluminum
t = plate thickness, mm
s = shortest side of plate panel, m
k = $k_l = \dfrac{8.4}{\Psi + 1.1}$ for ($0 \le \Psi \ge 1$) for plating with stiffener in direction of compression.
k = $k_s = c\left[1 + \left(\dfrac{s}{l}\right)^2\right]^2 \dfrac{2.1}{\Psi + 1.1}$ for ($0 \le \Psi \ge 1$) for plating with stiffener perpendicular to compression.
c = 1.21 for angle or T stiffeners

l = longest side of plate panel, m
Ψ = ratio of smaller to larger compressive stress acting on the panel.

For these calculations of the TIER 3 strength deck capacity, the following values are used:

t = 3.70 mm s = 0.2 m l = 1.2 m Ψ = 1.0 c = 1.21

which results in: $k_l = 4.0$ and $\sigma_{el\ longitudinal} = 85.01\ \text{N/mm}^2$ and
$k_s = 1.278$ and $\sigma_{el\ transverse} = 27.16\ \text{N/mm}^2$

With the information developed so far the following values are determined:

A. For longitudinal bending considerations of the TIER 3 strength deck:

$\sigma_f = 115\ \text{N/mm}^2$, $\sigma_{el\ longitudinal} = 85.01\ \text{N/mm}^2 \Rightarrow \sigma_{el\ longitudinal} > \sigma_f/2$ therefore:

$$\sigma_{c\ longitudinal} = 115\left(1-\frac{115}{4(85.01)}\right) = 76.11\ \text{N/mm}^2$$

B. For transverse bending considerations of the TIER 3 strength deck:

$\sigma_f = 115\ \text{N/mm}^2$, $\sigma_{el\ transverse} = 27.16\ \text{N/mm}^2 \Rightarrow \sigma_{el\ transverse} < \sigma_f/2$ therefore:

$\sigma_{c\ transverse} = 27.16\ \text{N/mm}^2$

- **DNV 3-3-10/E100 – Lateral buckling mode** (of longitudinal stiffeners)

$$\sigma_{el} = 10\frac{E}{\left(100\frac{l}{i}\right)^2}\ (\text{N/mm}^2)$$

Where: $i = \sqrt{\frac{I_A}{A}}$
I_A = moment of inertia of stiffener, cm^4
A = cross sectional area of stiffener, cm^2

The longitudinal stiffener on TIER 3 deck is approximated from the 70 x 40 IT from the Pc No 2 on Drawing No. 1811/2-201 Rev D.

The properties are: $I_A = 99.9\ \text{cm}^4$ $A = 11.23\ \text{cm}^2$ $\sigma_{el} = 426.3\ \text{N/mm}^2$

The stresses calculated above, along with appropriate factors of safety, will be used to check the buckling capacity of the ship structure subjected to the ABS and DNV loads required for the conversion.

6.2.3 Hull Girder Subjected to ABS Conversion Design Primary Loads

In accordance with ABS practice, the allowable stress for an aluminum alloy subjected to primary hull girder bending loads, is given by the following formula:

$$\text{Allowable stress} = f_p = 0.9 \text{ x } (f_p/Q)$$

Where: f_p = the allowable stress for mild steel = 175 N/mm^2, 25,380 psi
$Q = 0.9 + q_5$ but not less than Q_0
$q_5 = 115/\sigma_y$ N/mm^2 , $17{,}000/\sigma_y$ psi
$Q_0 = 635/(\sigma_y + \sigma_u)$ kN/mm^2, $92{,}000/(\sigma_y + \sigma_u)$ psi
σ_y = minimum yield strength of unwelded aluminum in N/mm^2, psi
σ_u = minimum ultimate strength of welded aluminum in N/mm^2, psi

The material properties and allowable stresses for use in the ABS equation are presented in Table 6-5 (Metric) and Table 6-6 (English). Table 6-7 presents a comparison of the ABS and DNV allowable stresses for primary hull girder consideration. It is noted that the ABS allowable stresses are greater than the DNV allowable stresses.

Table 6-5. ABS Material Properties for Primary Bending – Metric Units

Material	**σ_y unwelded N/mm^2**	**σ_u welded N/mm^2**	**q_5**	**Q**	**Q_0**	**f_p N/mm^2**
5083-H116/H321	213.7	275.8	0.538	1.438	1.297	109.5
5086-H116/H32	193.1	241.3	0.596	1.496	1.462	105.3
5383-H116/H321	NA	NA	NA	NA	NA	NA
6061-T6	241.3	137.9	0.477	1.377	1.675	114.4
6082-T6	NA	NA	NA	NA	NA	NA

Table 6-6. ABS Material Properties for Primary Bending – English Units

Material	**σ_y unwelded psi**	**σ_u welded psi**	**q_5**	**Q**	**Q_0**	**f_p psi**
5083-H116/H321	31,000	40,000	0.548	1.448	1.296	15,775
5086-H116/H32	28,000	35,000	0.607	1.507	1.460	15,158
5383-H116/H321	NA	NA	NA	NA	NA	NA
6061-T6	35,000	20,000	0.486	1.386	1.673	16,481
6082-T6	NA	NA	NA	NA	NA	NA

Table 6-7. ABS & DNV Hull Girder Allowable Stresses - Primary Bending

Material	DNV Filler Metal	DNV Factor, f_1	DNV Allowable N/mm^2	ABS Allowable N/mm^2
5083 H116, H321 **H116, H321**	5356 5183	0.53 0.60	92.8 105.0	109.5
5086, H116, H32	5356-5183	0.42	73.5	105.3
5383 H116	5183	0.64	112.0	NA
6061-T6	5356-5183	0.48	84.0	114.4
6082-T6	5356-5183	0.48	84.0	NA

6.2.3.1 ABS Vertical Bending Moment

From Table 6-3 it is seen that the largest ABS vertical bending moment acting on the vessel in accordance with the ABS Rules is 161,123 kN-m. Using the hull girder properties from Table 6-1 results in the following stresses acting on the hull girder:

Stress at Strength Deck = (161,123/3,468,558) x 1000 = 46.45 N/mm^2
Stress at Keel = (161,123/3,257,532) x 1000 = 49.46 N/mm^2

The actual stresses are well within the allowable stresses defined in Table 6-5. Similarly, these stresses are lower than the ABS allowable buckling stress for longitudinal bending, 82.0 N/mm^2.

6.2.3.2 ABS Minimum Hull Girder Requirements

As demonstrated above, the actual stresses resulting from vertical bending are within the allowable stresses defined for ABS. In addition to the allowable stresses, ABS also has minimum hull girder property requirements. These are shown in Table 6-8 and also confirm that the conversion would be acceptable for vertical bending moment when compared to the actual hull girder properties shown in Table 6-1. In accordance with the ABS definitions used for this project, the following requirements are determined from ABS 3-2-1:

Table 6-8. ABS Minimum Hull Girder Requirements for Vertical Bending

	Required Minimum Section Modulus cm^2 - m	Required Minimum Inertia cm^2 - m^2
NAVAL & COASTAL, Full Speed	15,905.98	124,866
NAVAL & COASTAL, Survival	13,083.95	102,712

6.2.3.3 ABS Transverse Bending Moment

From Table 6-3 it is seen that the largest ABS transverse bending moment acting on the vessel in accordance with the ABS Rules is 127,636 kN-m. Using the hull girder properties from Table 6-2 results in the following stresses acting on the hull girder:

$$\text{Stress at Strength Deck} = (127{,}636/6{,}068{,}573) \times 1000 = 21.03 \text{ N/mm}^2$$
$$\text{Stress at Wet Deck} = (127{,}636/12{,}527{,}680) \times 1000 = 10.19 \text{ N/mm}^2$$

ABS 3-5-3 defines the allowable primary transverse bending stress as $0.66\sigma_{y\ welded}$, which from Table 6-5 (Table 6-6) has a minimum value of 193.1 N/mm^2 (28,000 psi). This results in an allowable stress of 127.4 N/mm^2, demonstrating the acceptable stress levels for transverse bending. Similarly, the allowable buckling stress for ABS transverse bending was determined as 27.16 N/mm^2, showing acceptability of the conversion for transverse buckling.

6.2.4 Hull Girder Subjected to DNV Conversion Design Primary Loads

From Table 6-3 it is seen that the largest DNV vertical bending moment acting on the vessel in accordance with the DNV Rules is 245,774 kN-m. Using the hull girder properties from Table 6-1 results in the following stresses acting on the hull girder:

$$\text{Stress at Strength Deck} = (245{,}774/3{,}468{,}558) \times 1000 = 70.86 \text{ N/mm}^2$$
$$\text{Stress at Keel} = (245{,}774/3{,}257{,}532) \times 1000 = 75.45 \text{ N/mm}^2$$

From Table 6-7 it is seen that the strength deck satisfies all allowable stresses but the stress at the keel is marginally unacceptable if it is fabricated from 5086 material. If the keel is fabricated from 5083 or 5383, then the stress is acceptable.

At worst, the section modulus to the keel would have to be increased slightly to satisfy the allowable stress criteria. The increase in section modulus would be ((75.45 – 73.5)/73.5)) x 100 = 2.66% in order to reduce the bending stress to the allowable level. The new section modulus required at the keel becomes:

$$SM_{DNV\ Keel} = 1.0266 \times 3{,}257{,}532 = 3{,}344{,}182 \text{ cm}^3$$

Obviously, this is not a significant increase and could be accommodated through some minor structural modifications that could include replacement of bottom shell plate with heavier plating or replacement of bottom shell stiffening with heavier stiffening. It is possible that heavier plate and stiffening will be required to satisfy the slam load requirements and that these increased scantlings will also increase the hull girder inertia/section modulus sufficiently so that it will not be necessary to replace any additional structure for primary hull girder considerations. Calculations presented below will determine the conversion requirements to satisfy the slam loads and the subsequent impact to hull girder strength.

From DNV 3-3-10/102 the allowable buckling stress for the bottom plate has to be reduced by a factor of $\eta = 0.9$ resulting in an allowable buckling stress at the keel of 76.11 x 0.9 = 68.50 N/mm^2. This exceeds the stress at the keel calculated above by approximately 10% and represents a more critical overstress than the nominal stress calculated above. Again, the results of the slam load calculations will determine if any additional structure is required at the bottom of the ship and its impact on hull girder properties before any recommendations to solve this specific problem are addressed.

The allowable buckling stress at the deck does not have to be reduced, i.e., DNV defines $\eta = 1.0$ for the deck resulting in an allowable buckling stress to the deck of 76.11 N/mm^2 compared to the actual stress of 70.86 N/mm^2.

6.2.5 DNV Transverse Bending Moment

From Table 6-3 it is seen that the largest DNV transverse bending moment acting on the vessel in accordance with the DNV Rules is 130,527 kN-m. Using the hull girder properties from Table 6-2 results in the following stresses acting on the hull girder:

Stress at Strength Deck = (130,527/6,068,573) x 1000 = 21.51 N/mm^2
Stress at Wet Deck = (130,527/12,527,680) x 1000 = 10.42 N/mm^2

DNV 3-3-4/E100 defines typically acceptable stresses for transverse hull girder strength. The allowable normal stress is 160f_1. Table 6-4 shows the minimum value for f_1 for this design as $f_1 = 0.42$ resulting in an allowable normal stress of 67.2 N/mm^2, well above the actual stresses predicted for transverse bending. Similarly, these stresses are lower than those that have already been determined as acceptable for compressive buckling considerations in accordance with DNV criteria, i.e., 27.16 N/mm^2.

6.2.6 Global Loads – Torsional and Pitch Connecting Moments

A clarification of the terminology used by ABS and DNV needs to be provided for these quantities. DNV 3-1-3/B300, Figure 7 is copied below and shown in Figure 6-1. It provides clear definition for both torsional and pitch connecting moments. The torsional moment acts about the longitudinal axis of the ship and is typically a concern in monohulls with large deck openings. The pitch connecting moment refers to the action of the ship where the bow of one hull is pitching up while the other pitches down, or vice versa.

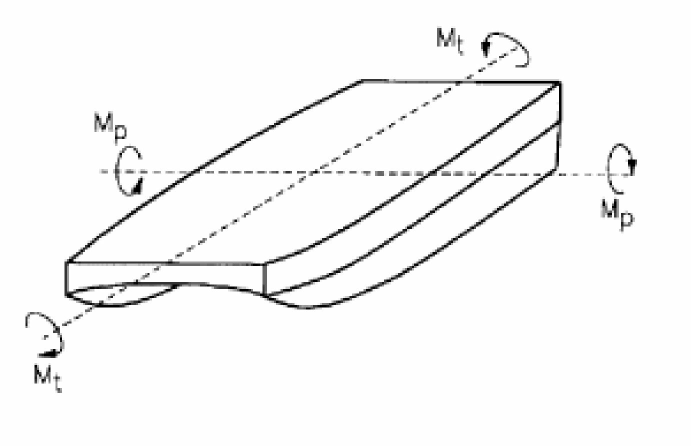


Fig. 7
Pitch connecting moment and torsion moment on twin hull connection

Figure 6-1. DNV Torsional & Pitch Connecting Moments from DNV 3-1-3/B300

The ABS torsional moment accounts for the same vessel motion as the DNV pitch connecting moment. ABS does not include the action of a torsional moment about the longitudinal axis of the ship in their nominal design criteria for twin-hulled craft.

With the relatively large hull girder offered by PacifiCat it is not expected that either of these quantities will govern the design. Regardless, proper calculation of the stresses acting in the hull girder as a result of the pitch connecting or torsional moments requires the development of a full ship finite element analysis, which is outside the scope of this project. The relatively light strength deck plate (TIER 3 deck) is only 3.70 mm thick and raises the concern of its stability under the combined shear and compressive forces that would be acting when the ship is subjected to pitch connecting or combined pitch connecting and vertical bending load cases. This analysis would be required if the actual conversion were to be completed.

6.3 STRUCTURAL MODIFICATIONS FROM SECONDARY SLAM LOADS

Secondary slam loads factor into the design of numerous locations around the vessel. In accordance with the ship structural drawings there are a variety of different plate and stiffener profiles that resist these loads as a function of location. These structural combinations are listed in Table 6-9. Figure 6-2 provides a Key Plan for the slam load locations throughout the ship. The Main Vehicle Deck is shown in Figure 6-2 as a point of reference. There are no slam loads acting on the Main Vehicle Deck.

Table 6-9 includes a column for the section modulus of the plate/stiffener combination resisting the slam load. The section modulus is calculated assuming an effective breadth of plate equal to the stiffener spacing at the location under consideration. This reflects typical commercial practice and the shear lag phenomenon associated with secondary bending, i.e., bending under a normal load (the load is perpendicular to the plate). In virtually all instances, using the stiffener spacing as the effective breadth of plate produces the same results that would have been obtained using the US Navy post-buckling approach to effective width of flange. For aluminum in these calculations the effective width would have been taken as 35t, i.e., the effective width of the plate flange is equal to 35 times the thickness of the plating to which the stiffener is welded. Because the stiffener spacing throughout most of the ship is relatively tight the product of 35t equals or exceeds most stiffener spacings, the T3 strength deck is an exception. This would result in the same plating flange and mechanical properties for the plate/stiffener combination using either approach. In no case does the effective breadth used for the calculations exceed the stiffener spacing. It was not possible to get the properties for the 185 x 35 bulb or 150 x 55 IT. Since these are among the stronger sections and the calculations for these areas did not result in severe requirements it is not expected that they would need replacement.

Table 6-9. Secondary Stiffeners Subjected to Slam Load

Location on ship	Scantlings	Section Modulus cm^3	Effective Breadth mm
Wet Deck Bow to Frame 60	185 x 35 Bulb Flat on 10 mm plate	NA	230
Wet Deck Frame 60 to Transom	120 x 50 IT on 7 mm plate	67.2	230
Hull Bottom Bow to Frame 69	120 x 50 IT on 20 mm plate	78.2	240
Hull Bottom Frame 69 to Frame 55	140 x 50 IT on 12 mm plate	99.2	270
Hull Bottom Frame 55 to Frame 43	140 x 50 IT on 8 mm plate	94.5	270
Hull Bottom Frame 43 to Frame 31	150 x 55 IT on 8 mm plate	104.1*	270
Hull Bottom Frame 31 – Aft	140 x 50 IT on 8 mm plate	94.5	270
Inner Hull Bow to Frame 73	120 x 50 IT on 16 mm & 20 mm plate	75.6/78.7	270
Inner Hull Frame 73 – Aft	80 x 40 IT on 8 mm & 12 mm plate	28.8/30.5	270
Inner Hull Frame 73 – Aft	70 x 40 IT on 8 mm & 10 mm plate	23.8/24.4	270
Haunch Bow to Frame 60	185 x 35 Bulb Flat on 10 mm plate	NA	NA
Haunch Frame 60 – Aft	120 x 50 IT on 10 mm plate	69.9	210
Outer Hull Bow to Frame 53	120 x 50 IT on 8 mm, 10 mm, 12 mm & 16 mm plate	69.0/71.0 72.5/75.6	270
Outer Hull Frame 53 – Aft	80 x 40 IT on 6 mm plate	27.9	270
Outer Hull Frame – 53 Aft	70 x 40 IT on 6 mm plate	23.0	270

*Since there was no information on the drawings which could be used to determine the web and flange dimensions for the these profiles, the same section was used as the 140 x 50 IT with the depth of the web increased by 10 mm.

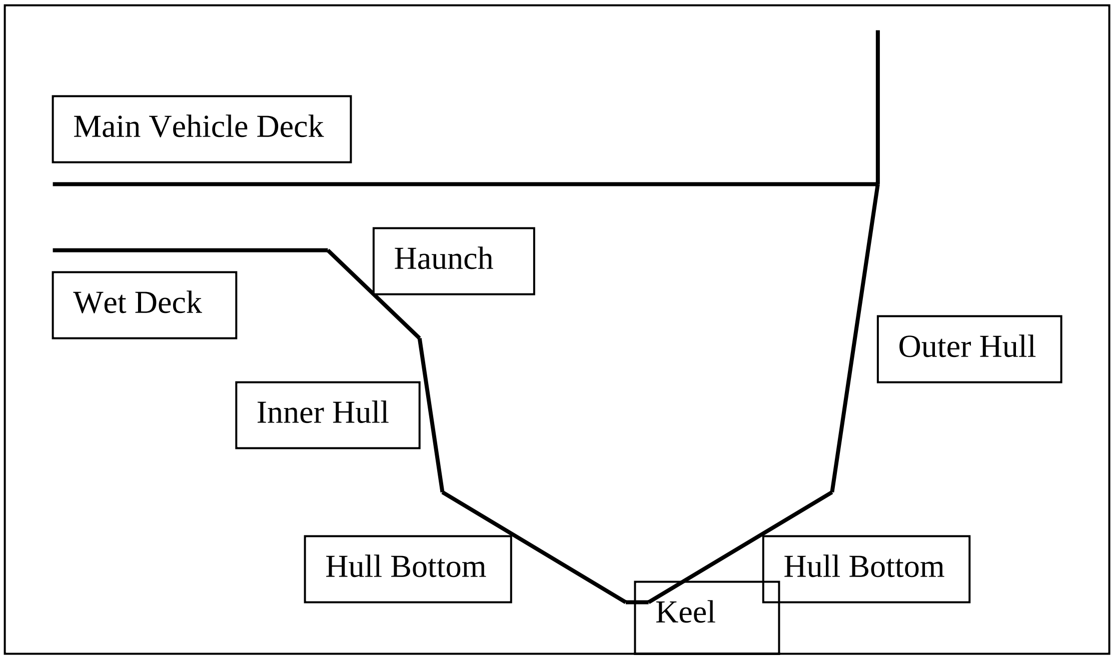


Figure 6-2. Key Plan for Stiffener Locations Subjected to Slam Loads

6.3.1 Slam Loads Used for Calculations

Earlier in this report a variety of calculations and tables were presented for slam loads under different vessel classing scenarios. These corresponded to ABS classing requirements for Coastal Naval Craft and Naval Craft, which were taken to correspond to DNV service area restrictions **R1** and unrestricted, respectively. Slam loads were also presented for the original PacifiCat designed to its **R4** service restrictions. The calculations developed for this report will only investigate the most severe ABS and DNV scenarios, i.e., ABS Naval Craft and DNV unrestricted slam loads in the operational condition. For ease of reference, the tables containing these loads (Table 5-4 and Table 5-11), along with the table containing the original **R4** loads (Table 5-18) are repeated below in Table 6-10, Table 6-11, and Table 6-12, respectively.

The original intent to increment the structural modifications for slam loads from **R4** to **R1** and then from **R1** to Unrestricted was not required by the SOW for this project and it has been decided that this intermediate step is no longer necessary. This was further justified by the lack of any structural modifications resulting from the primary hull girder loads required for the conversion from **R4** to Unrestricted.

Table 6-10. ABS Slam Loads for Naval Craft Operational & Survival Conditions

All pressures are kN/m^2

	Naval Craft Operational			Naval Craft Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	243.3	94.05	138.3	171.0	66.09	67.6
2	235.4	91.00	138.3	170.3	65.84	67.6
3	227.5	87.95	138.3	169.6	65.58	67.6
4	219.6	84.90	55.3	169.0	65.33	27.0
5	211.7	81.85	55.3	168.3	65.07	27.0
6	203.8	78.80	55.3	167.7	64.82	27.0
7	195.9	75.75	55.3	167.0	64.56	27.0
8	226.3	75.75	55.3	192.9	64.56	27.0
9	265.0	75.75	55.3	225.9	64.56	27.0
10	293.9	75.75	62.2	250.5	64.56	30.4
11	293.9	75.75	69.2	250.5	64.56	33.8

Table 6-11. DNV Slam Loads for Unrestricted Operational & Survival Conditions

All pressures are kN/m^2

	Unrestricted Operational			Unrestricted Survival		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	601.3	103.6	262.4	251.0	103.6	89.3
2	601.3	97.1	262.4	251.0	97.1	89.3
3	601.3	90.6	262.4	251.0	90.6	89.3
4	601.3	84.1	262.4	251.0	84.1	89.3
5	601.3	77.6	131.2	251.0	77.6	44.6
6	601.3	71.2	131.2	251.0	71.2	44.6
7	541.2	71.2	131.2	225.9	71.2	44.6
8	630.2	71.2	131.2	263.1	71.2	44.6
9	717.7	71.2	131.2	299.6	71.2	44.6
10	721.6	71.2	131.2	301.2	71.2	44.6
11	601.3	71.2	131.2	251.0	71.2	44.6

Table 6-12. DNV Secondary Slam Load Pressures for R4 Notation

All pressures are kN/m^2

	R4		
Location	**Bottom Slamming**	**Side & Transom Slamming**	**Wet Deck Slamming**
1	251.0	70.0	32.8
2	251.0	66.1	32.8
3	251.0	62.2	32.8
4	251.0	58.3	32.8
5	251.0	54.4	16.4
6	251.0	50.5	16.4
7	225.9	50.5	16.4
8	263.1	50.5	16.4
9	299.6	50.5	16.4
10	301.2	50.5	16.4
11	251.0	50.5	16.4

As seen in Table 6-10 and Table 6-11 the operational slam loads are always more severe than the survival slam loads and will be taken to govern the design. No calculations for the survival condition will be developed for this report.

6.3.2 ABS Structure to Resist Slam Loads

These calculations shall be further subdivided for plates and stiffeners.

6.3.2.1 ABS Plating to Resist Slam Loads

In accordance with ABS HSNC 3-2-3/1.3, plating subjected to lateral loading is designed in accordance with the following formula:

$$t = s\sqrt{\frac{pk}{1000\sigma_a}} \text{ (mm)}$$

Where: s = stiffener spacing, mm
p = design pressure, kN/m^2
k = plate panel aspect ratio factor from ABS 3-2-3/Table 1
σ_a = design stress, N/mm^2 from ABS 3-2-3/Table 2

For all instances, the factor k resulting for this design is k = 0.5.

From ABS 3-2-3/Table 2 the design stress, σ_a, is taken as:

- Bottom Shell $\sigma_a = 0.90\sigma_y$
- Side Shell $\sigma_a = 0.90\sigma_y$
- Wet Deck $\sigma_a = 0.90\sigma_y$

Where σ_y is taken as the welded yield strength of aluminum. Also, for bottom shell outside of 0.4L, the design stress may be taken as σ_y.

In accordance with INCAT Designs Dwg 1811/2-201 the structure subjected to slam loads, Figure 6-2, could be a variety of different aluminum alloys/tempers. Once again, specific definition was not available for this project and the calculations shall be presented assuming 5083 and 5086. Using the strength data from ABS Table 3.3.2 presents the following material strengths:

- 5083-H116(H321) - $\sigma_{y\ welded}$ = 24,000 psi = 165.5 N/mm^2
- 5086-H32(H116) - $\sigma_{y\ welded}$ = 19,000 psi = 131.0 N/mm^2

There is a strong possibility that the actual ship was fabricated using 5383, a new aluminum alloy developed specifically for marine applications. It is recognized that the strength of this material is very similar to 5083 and any results would be similar for these two alloys.

The results of the ABS plate calculations for slam load are summarized in Table 6-13. In all the tables that follow for the adequacy of plate and stiffening, inadequate existing structure is shown in a **red, bold** font.

Table 6-13. ABS Naval Craft Operational Slam Loads – New Hull Plating Requirements & Existing Thicknesses

Location	Bottom Slam kN/m^2	Bottom Plate Reqd 5083/5086 mm	Bottom Plate Actual mm	Side Slam kN/m^2	Side Plate Reqd 5083/5086 mm	Side Plate Actual mm	Wet Deck Slam kN/m^2	Wet Deck Plate Reqd 5083/5086 mm	Wet Deck Plate Actual mm
1	NA	NA	NA	NA	NA	NA	NA	NA	NA
2	235.4	6.40/7.19	20	91.00	4.72/5.30	16	138.3	4.96/5.57	10
3	227.5	6.29/7.07	12	87.95	4.64/5.21	10	138.3	4.96/5.57	10
4	219.6	7.33/8.24	12	84.90	4.56/5.12	10	55.3	3.13/3.52	10
5	211.7	7.20/**8.09**	8	81.85	4.48/5.03	8	55.3	3.13/3.52	7
6	203.8	7.06/7.94	8	78.80	4.39/4.94	6	55.3	3.13/3.52	7
7	195.9	6.92/7.78	8	75.75	4.31/4.84	6	55.3	3.13/3.52	7
8	226.3	7.44/**8.36**	8	75.75	4.31/4.84	6	55.3	3.13/3.52	7
9	265.0	7.64/8.59	12	75.75	4.31/4.84	6	55.3	3.13/3.52	7
10	293.9	8.05/9.04	14	75.75	4.31/4.84	10	62.2	3.32/3.74	7
11	NA	NA	NA	NA	NA	NA	NA	NA	NA

As seen in Table 6-13, the only plate that would need to be replaced are two local areas of bottom shell, which are nominally overstressed if they are fabricated from 5086 aluminum. All other existing PacifiCat bottom shell, side shell and Wet Deck plating is adequate to resist the slam loads predicted by the ABS HSNC rules for unrestricted, open-ocean operation.

6.3.2.2 ABS Stiffener Requirements to Satisfy Slam Loads

The nominal stiffening requirements need to satisfy ABS 3-2-4/1.3.1 for section modulus:

$$SM = \frac{83.3 psl^2}{\sigma_a} \text{ (cm}^3\text{)}$$

and ABS 3-2-4/1.3.2 for moment of inertia:

$$I = \frac{260 psl^3}{K_4 E} \text{ (cm}^4\text{)}$$

Where: s = stiffener spacing, m
l = stiffener span, m
K_4 = 0.0021 for shell and deep tank stringers and transverse webs constructed of aluminum
= 0.0018 for deck girders and transverses constructed of aluminum
E = modulus of elasticity = 6.9 x 10^4 N/mm^2 for aluminum

p is the same as defined above. The design stress, σ_a, is assigned different values than used for plate analysis. These values are taken from ABS 3-2-4/Table 1 and are summarized below for use in this project. All values reflect the slam load condition.

- Bottom longitudinal $\sigma_a = 0.65\sigma_y$
- Side longitudinal $\sigma_a = 0.60\sigma_y$
- Wet Deck longitudinal $\sigma_a = 0.75\sigma_y$

Where: σ_y can be taken as above, Section 6.3.2.1, for welded aluminum

The results of the ABS slam load requirements for stiffener section modulus and inertias are summarized in Table 6-15 and Table 6-16, respectively. Table 6-14 presents a summary of the ABS allowable secondary stresses.

Table 6-14. Summary of ABS Allowable Secondary Bending Stresses

ABS		**5083 (N/mm^2)**	**5086 (N/mm^2)**	**5083 (psi)**	**5086 (psi)**
Bottom Shell	**Plate**	149	118	21,600	17,100
	Stiffener	107.6	85.2	15,600	12,350
Side Shell	**Plate**	149	118	21,600	17,100
	Stiffener	99.3	78.6	14,400	11,400
Wet Deck	**Plate**	149	118	21,600	17,100
	Stiffener	124.1	98.3	18,000	14,250

Note: * This table does not include the higher allowable stresses for bottom shell plate outside of 0.4L.

Table 6-15. ABS Naval Craft Operational Slam Loads– Section Modulus Requirements & Existing Strength

Location	Bottom Slam kN/m^2	Bottom SM$_{reqd}$ 5083/5086 cm^3	Bottom SM Actual cm^3	Side Slam kN/m^2	Side SM$_{reqd}$ 5083/5086 cm^3	Side SM Actual cm^3	Wet Deck Slam kN/m^2	Wet Deck SM$_{reqd}$ 5083/5086 cm^3	Wet Deck SM Actual cm^3
1	NA	NA	NA	NA	NA	NA	NA	NA	NA
2	235.4	63.0/**79.6**	78.2	91.00	29.68/37.50	72.5	138.3	30.7/38.8	NA
3	227.5	60.9/76.9	99.2	87.95	28.69/36.24	72.5	138.3	30.7/38.8	NA
4	219.6	66.1/85.5	99.2	84.90	27.69/34.98	72.5	55.3	12.3/15.5	67.2
5	211.7	63.7/80.5	94.5	81.85	26.70/**33.73**	27.9	55.3	12.3/15.5	67.2
6	203.8	61.4/77.5	104.1	78.80	25.70/**32.47**	27.9	55.3	12.3/15.5	67.2
7	195.9	59.0/74.5	94.5	75.75	24.71/**31.21**	27.9	55.3	12.3/15.5	67.2
8	226.3	68.1/86.1	94.5	75.75	24.71/**31.21**	27.9	55.3	12.3/15.5	67.2
9	265.0	79.8/**100.8**	99.1	75.75	24.71/**31.21**	27.9	55.3	12.3/15.5	67.2
10	293.9	88.5/**111.8**	101.3	75.75	24.71/**31.21**	27.9	62.2	13.8/17.5	67.2
11	NA	NA	NA	NA	NA	NA	NA	NA	NA

Table 6-16. ABS Naval Craft Operational Slam Loads– Inertia Requirements & Existing Strength

Location	Bottom Slam kN/m^2	Bottom I$_{reqd}$ cm^4	Bottom I Actual cm^4	Side Slam kN/m^2	Side I$_{reqd}$ cm^4	Side I Actual cm^4	Wet Deck Slam kN/m^2	Wet Deck I$_{reqd}$ cm^4	Wet Deck I Actual cm^4
1	NA	NA	NA	NA	NA	NA	NA	NA	NA
2	235.4	175.2	885	91.00	76.2	757	138.3	98.6	NA
3	227.5	169.3	1164	87.95	73.6	757	138.3	98.6	NA
4	219.6	183.8	1164	84.90	71.1	757	55.3	39.4	606
5	211.7	177.2	1014	81.85	68.5	187	55.3	39.4	606
6	203.8	170.6	1187	78.80	66.0	187	55.3	39.4	606
7	195.9	164.0	1014	75.75	63.4	187	55.3	39.4	606
8	226.3	189.5	1014	75.75	63.4	187	55.3	39.4	606
9	265.0	221.9	1164	75.75	63.4	187	55.3	39.4	606
10	293.9	246.0	1225	75.75	63.4	187	62.2	44.4	606
11	NA	NA	NA	NA	NA	NA	NA	NA	NA

6.3.2.3 ABS Structure Required to Satisfy Slam Load Criteria

Table 6-16 shows that all of the existing stiffening satisfies the ABS inertia requirements while Table 6-15 shows that some of the existing stiffening on the bottom shell and side shell would have to be replaced if it's fabricated out of 5086 aluminum. Close investigation of the undersized stiffening identified in Table 6-15 reveals that the existing stiffeners are only nominally undersized compared to the new requirements. The most undersized stiffeners are at Location 10 on the bottom shell, which are currently 140 x 50 I/T. The new, required section modulus is only 10.5 cm^3 (0.65 in^3) greater than the capacity of the existing stiffener. The overstressed stiffener, 140 x 50 I/T has a 50 mm x 4 mm (2" x 0.157") flange and the actual modifications to this structure could be accomplished by welding a 32 mm x 6 mm (1.25" x 0.236") doubler to the existing flange. This increases the section modulus from 101.3 cm^3 (6.2 in^3) to 114.1 cm^3 (7.0 in^3), satisfying the required section modulus criteria of 111.8 cm^3. This represents the most severe modification that would have to be applied, all others being even less severe. In addition, it is also noted that the doubler added to the flange would only need to be applied to the local portion of the stiffener that is overstressed, i.e., the entire span of the stiffener is not overstressed and more detailed calculation would quickly reveal that only a small portion of the overstressed stiffener spans would have to be reinforced with flange doublers. The cost estimates developed for this project will assume the entire span of the stiffener has the doubler added.

As noted in Table 6-15 and Table 6-16, there is only minimal stiffening that would need reinforcement along the bottom shell, side shell or Wet Deck to satisfy the ABS HSNC criteria for open-ocean operation. Most of the existing stiffeners throughout these areas are acceptable. The original PacifiCat structure was designed to satisfy DNV **R4** service area restrictions, a classification that requires the craft never be more than 20 nautical miles from safe harbor. Combined with the earlier results for global loads and shell plate subjected to slam loads, it is a surprising and unexpected result that the structure for this craft is essentially acceptable to operate in the open-ocean, unrestricted environment in accordance with the criteria presented in the ABS HSNC, requiring only minimal modification.

All of the ABS structural modifications are shown on the drawings included in this report. In summary, the ABS modifications to resist the slam loads are:

- Increase a portion of the bottom hull plate from 8mm to 9mm plate.
- Add 32mm x 6mm flange doublers throughout a few forward and aft stiffeners.
- Add 25mm x 6mm flange doublers from Frame 48 aft on the inner side shell and outer side shell stiffeners.

All structural modifications assume the use of the stronger alloy, 5083.

6.3.2.4 Weight Estimate of ABS Structural Modifications for Slam Loading

The following weight estimate, summarized in Table 6-17, is associated with the ABS structural modifications discussed above. All plate areas where electronically measured off the AutoCad drawings available for the ship. The stiffener lengths were estimated from these same drawings with approximations for stiffener termination points to simplify the weight impacts for the purposes of this project.

Table 6-17. Weight Increase to Accommodate ABS Structural Modifications – Open-Ocean, Unrestricted

Component	Existing Structure	Weight of Existing Structure (kg)	New Structure	Weight of New Structure (kg)	Weight Increase (kg)
Bottom shell plate, P/S	288.44 m^2 @ 8mm	6265	288.44 m^2 @ 9mm	7048	783.0
Bottom Shell Stiffening IWO Removed Plate	140 x 50 I/T in both locations	3284	140 x 50 I/T in both locations	3284*	0
Bottom Shell stiffening, P/S	Remains Unchanged	N/A	432 meters of 32mm x 6mm flange doubler	225	225
Outer hull stiffening, P/S	Remains Unchanged	N/A	2030 meters of 25mm x 6mm flange doubler	827	827
Inner hull stiffening, P/S	Remains Unchanged	N/A	1320 meters of 25mm x 6mm flange doubler	538	538
					Σ 2373

*The new stiffening on the new plate will be the same as the existing stiffening on the removed plate, i.e., no weight change for the stiffening component.

The results presented in Table 6-17 indicate a total structural weight increase of just over 2 metric tonnes to accommodate the ABS structural modifications.

6.3.3 DNV Structure to Resist Slam Loads

These calculations shall be further subdivided for plates and stiffeners.

6.3.3.1 DNV Plating to Resist Slam Loads

In accordance with DNV HSLC & NSC 3-3-5/B300, the hull plate subjected to slam loads needs to satisfy the following:

$$t = \frac{22.4 k_r s \sqrt{p_{sl}}}{\sqrt{\sigma_{sl}}} \text{ (mm)}$$

where: k_r = correction for curved plates = (1-0.5(s/r))
s = stiffener spacing, m
p_{sl} = slam pressure from 3-1-2, kN/m^2
σ_{sl} = allowable bending stress due to lateral load, N/mm^2

The slam pressures are shown above in Table 6-11, again using only the operational slam loads for these calculations since they govern in all instances compared to survival. The allowable stresses are taken from DNV 3-3-5/Table A1 with only the appropriate values shown below:

- Bottom plate σ_{sl} = 200f_1
- Side plate σ_{sl} = 180f_1
- Wet Deck plate σ_{sl} = 200f_1

The values for f_1 are summarized below and are the same as those shown above in Table 6-4.

- 5083; f_1 = 0.60
- 5086; f_1 = 0.42
- 5383; f_1 = 0.64

Similar to PacifiCat Dwg 1811/2-201, the strength for 5383 is slightly less than, but similar to 5083. As a result, all the calculations for DNV slam loads will only be done for 5083 and 5086, the same as ABS.

The allowable bending stresses, based on welded properties, will be used for the plate design:

- 5083 0.60 x 200 = 120 N/mm^2 for Bottom & Wet Deck plate
  0.60 x 180 = 108 N/mm^2 for Side plate

- 5086 0.42 x 200 = 84 N/mm^2 for Bottom & Wet Deck plate
  0.42 x 180 = 75.6 N/mm^2 for Side plate

The results of these calculations are presented in Table 6-18.

Table 6-18. DNV Unrestricted Operation Slam Loads– New Hull Plating Requirements & Existing Thicknesses

Location	Bottom Slam kN/m^2	Bottom Plate Reqd 5083/5086 mm	Bottom Plate Actual mm	Side Slam kN/m^2	Side Plate Reqd 5083/5086 mm	Side Plate Actual mm	Wet Deck Slam kN/m^2	Wet Deck Plate Reqd 5083/5086 mm	Wet Deck Plate Actual mm
1	NA	NA	NA	NA	NA	NA	NA	NA	NA
2	601.3	12.03/14.38	20	97.1	5.76/6.88	16	262.4	7.62/9.11	10
3	601.3	12.03/14.38	12	90.6	5.56/6.65	10	262.4	7.62/9.11	10
4	601.3	**13.54/16.18**	12	84.1	5.36/6.40	10	262.4	7.62/9.11	10
5	601.3	**13.54/16.18**	8	77.6	5.15/6.15	8	131.2	5.39/6.44	7
6	601.3	**13.54/16.18**	8	71.2	4.91/5.87	6	131.2	5.39/6.44	7
7	541.2	**12.84/15.35**	8	71.2	4.91/5.87	6	131.2	5.39/6.44	7
8	630.2	**13.86/16.57**	8	71.2	4.91/5.87	6	131.2	5.39/6.44	7
9	717.7	**14.79/17.68**	12	71.2	4.91/5.87	6	131.2	5.39/6.44	7
10	721.6	**14.83/17.73**	14	71.2	4.91/5.87	10	131.2	5.39/6.44	7
11	NA	NA	NA	NA	NA	NA	NA	NA	NA

6.3.3.2 DNV Stiffener Requirements to Satisfy Slam Loads

In accordance with DNV 3-3-5/C201, stiffeners subjected to slam loads shall satisfy the following criteria for section modulus, Z, and shear area, A_s:

$$Z = \frac{ml^2 sp_{sl}}{\sigma_{sl}} \text{ (cm}^3\text{)}$$

$$A_s = \frac{6.7(l - s)sp_{sl}}{\tau} \text{ (cm}^2\text{)}$$

where: m = factor from DNV Table C1
= 85 for side, bottom and deck longitudinal members
l = stiffener span, m
s = stiffener spacing, m
p_{sl} = design slam pressure, kN/m^2
σ_{sl} = allowable bending stress, =180f_1, N/mm^2
$\tau = \tau_{sl}$ = allowable shear stress = 90f_1 for slam load considerations

The allowable stresses become:

- 5083 σ_{sl} = 0.60 x 180 = 108 N/mm^2
  τ_{sl} = 0.60 x 90 = 54 N/mm^2

- 5086 σ_{sl} = 0.42 x 180 = 75.6 N/mm^2
  τ_{sl} = 0.42 x 90 = 37.8 N/mm^2

The calculations for section modulus and shear area are presented in Table 6-20 and Table 6-21, respectively. Table 6-19 presents a summary of the DNV allowable secondary bending stresses.

Table 6-19. Summary of DNV Allowable Secondary Bending Stresses

DNV		**5083 (N/mm^2)**	**5086 (N/mm^2)**	**5083 (psi)**	**5086 (psi)**
Bottom Shell	**Plate**	120	84	17,400	12,180
	Stiffener	108	75.6	15,660	10,960
Side Shell	**Plate**	108	75.6	15,660	10,960
	Stiffener	108	75.6	15,660	10,960
Wet Deck	**Plate**	120	84	17,400	12,180
	Stiffener	108	75.6	15,660	10,960

Table 6-20. DNV Unrestricted Operation Slam Loads– Section Modulus Requirements & Existing Strength

Location	Bottom Slam kN/m^2	Bottom SM_{reqd} 5083/5086 cm^3	Bottom SM Actual cm^3	Side Slam kN/m^2	Side SM_{reqd} 5083/5086 cm^3	Side SM Actual cm^3	Wet Deck Slam kN/m^2	Wet Deck SM_{reqd} 5083/5086 cm^3	Wet Deck SM Actual cm^3
1	NA	NA	NA	NA	NA	NA	NA	NA	NA
2	601.3	163.6/233.7	78.2	97.1	29.8/42.6	72.5	262.4	68.4/97.7	NA
3	601.3	163.6/233.7	99.2	90.6	27.8/39.8	72.5	262.4	68.4/97.7	NA
4	601.3	184.0/262.9	99.2	84.1	25.8/36.9	72.5	262.4	68.4/97.7	67.2
5	601.3	184.0/262.9	94.5	77.6	23.9/34.1	27.9	131.2	34.2/48.9	67.2
6	601.3	184.0/262.9	104.1	71.2	21.8/31.1	27.9	131.2	34.2/48.9	67.2
7	541.2	165.6/236.6	94.5	71.2	21.8/31.1	27.9	131.2	34.2/48.9	67.2
8	630.2	192.8/275.5	94.5	71.2	21.8/31.1	27.9	131.2	34.2/48.9	67.2
9	717.7	219.6/313.7	99.1	71.2	21.8/31.1	27.9	131.2	34.2/48.9	67.2
10	721.6	220.8/315.4	101.3	71.2	21.8/31.1	27.9	131.2	34.2/48.9	67.2
11	NA	NA	NA	NA	NA	NA	NA	NA	NA

Table 6-21. DNV Unrestricted Operation Slam Loads– Shear Area Requirements & Existing Strength

Location	Bottom Slam kN/m^2	Bottom A_s Reqd cm^2	Bottom A_s Actual cm^2	Side Slam kN/m^2	Side A_s Reqd cm^2	Side A_s Actual cm^2	Wet Deck Slam kN/m^2	Wet Deck A_s Reqd cm^2	Wet Deck A_s Actual cm^2
1	NA	NA	NA	NA	NA	NA	NA	NA	NA
2	601.3	17.2/24.6	7.8	97.1	3.0/4.3	7.8	262.4	7.3/10.4	NA
3	601.3	17.2/24.6	10.5	90.6	2.8/4.0	7.8	262.4	7.3/10.4	NA
4	601.3	18.7/26.8	10.5	84.1	2.6/3.8	7.8	262.4	7.3/10.4	7.8
5	601.3	18.7/26.8	10.5	77.6	2.4/3.5	3.6	131.2	3.6/5.2	7.8
6	601.3	18.7/26.8	10.6	71.2	2.2/3.2	3.6	131.2	3.6/5.2	7.8
7	541.2	16.9/24.1	10.5	71.2	2.2/3.2	3.6	131.2	3.6/5.2	7.8
8	630.2	19.6/28.1	10.5	71.2	2.2/3.2	3.6	131.2	3.6/5.2	7.8
9	717.7	22.4/31.9	10.5	71.2	2.2/3.2	3.6	131.2	3.6/5.2	7.8
10	721.6	22.5/32.1	10.5	71.2	2.2/3.2	3.6	131.2	3.6/5.2	7.8
11	NA	NA	NA	NA	NA	NA	NA	NA	NA

6.3.3.3 DNV Structure Required to Satisfy Slam Load Criteria

As shown in Table 6-18, Table 6-20 and Table 6-21 by the **red, bold** inputs, there are a number of plate thickness and stiffener requirements that are not satisfied by the existing structure for a DNV open-ocean, unrestricted notation. Most of the problems are in the bottom hull although there are a few areas of concern in the side shell and Wet Deck.

Again, all structural modifications will be made using the stronger aluminum alloy, 5083, with no structural modifications developed for 5086.

All of the DNV structural modifications are shown on the drawings included in this report. In summary, the DNV modifications to resist the slam loads are:

- Increase a significant percentage of the bottom hull plate from 8mm, 10mm or 12mm to 14mm plate.
- Increase the bottom hull stiffening to 10 x 5.75#T forward of frame 8 and to 10 x 7.25#T aft of Frame 8. The requirements for shear area drove the design of these members and resulted in significantly heavier stiffening than would have been required to satisfy the section modulus criteria, which would have been satisfied with 7 x 3.5#T for all stiffening on the bottom hull.
- Similar to the modifications required for the ABS Inner and Outer side shell, add 25mm x 6mm flange doublers from Frame 48 aft on the inner side shell and outer side shell stiffeners. The same flatbar doubler is also assumed for Location 4 on the Wet Deck, which is marginally overtstressed.

6.3.3.4 Increased Hull Girder Properties as a Result of New Slam Load Structure

As discussed in Section 6.2.4 above, it was necessary to increase the section modulus to the keel of the vessel to satisfy the DNV allowable stress and buckling criteria. The scantlings resulting from the conversion slam loads are significantly greater than the original scantlings with 14mm plate replacing 8mm original plate and 10 x 5.75#T stiffeners (Area = 32.2 cm^2) replacing 140 x 50 IT (Area = 14.32 cm^2) original stiffeners. This has a significant impact to the hull girder properties as shown in Table 6-22.

Table 6-22. Hull Girder Properties with Revised DNV Slam Required Scantlings

Property	Value	Value
Cross Sectional Area	14,713.96 cm^2	
Moment of Inertia	287,695.90 cm^2 m^2	
Section Modulus, Deck	38,769.54 cm^2 m	3,876,954 cm^3
Section Modulus, Keel	43,117.75 cm^2 m	4,311,775 cm^3
Shear Area	6273.71 cm^2	
NA, deck	7.42 m	
NA, keel	6.67 m	

From Table 6-1 it is seen that the original section modulus to the keel was 32,575.32 cm^2. The new section modulus is 43,117.75 approximately 32% greater than the original strength – far greater than required by the calculations shown above to satisfy the stress and buckling criteria.

6.3.3.5 Weight Estimate of DNV Structural Modifications for Slam Loading

The following weight estimate, summarized in Table 6-23, is associated with the DNV modifications to the structure. The drawings of the bottom hull are not expanded and so the plate areas were estimated from the midship section and hull bottom drawings to approximate the required area of plate replacement. The 25 mm flat keels in the port and starboard bottom hulls was accounted for and not included in the calculation of plate area that needs to be replaced. The stiffener lengths were estimated from these same drawings with approximations for stiffener termination points to simplify the weight impacts for the purposes of this project.

Table 6-23. Weight Increase to Accommodate DNV Structural Modifications – Open-Ocean, Unrestricted

Component	Existing Structure	Weight of Existing Structure (kg)	New Structure	Weight of New Structure (kg)	Weight Increase (kg)
Bottom hull plate, P/S	449.5 m^2 @ 8mm 27.8 m^2 @ 10mm 248.7 m^2 @ 12mm	9558 739 7933	726.0 m^2 @ 14 mm	27,016	8786
Bottom hull stiffening, P/S	3432.0 m of 140 x 50 IT 633.6 m of 150 x 55 IT	13,064 3032	3484.8 m of 10 x 5.75#T 580.8 m of 10 x 7.25#T	29,822 6275	20,001
Outer hull stiffening, P/S	Remains Unchanged	N/A	2030 meters of 25mm x 6mm flange doubler	827	827
Inner hull stiffening, P/S	Remains Unchanged	N/A	1320 meters of 25mm x 6mm flange doubler	538	538
Wet Deck stiffening, P/S	Remains Unchanged	N/A	441 meters of 25mm x 6mm flange doubler	180	180
					Σ 30,332

As seen in Table 6-23, it requires approximately 30 metric tonnes of structural modifications to upgrade the PacifiCat to satisfy DNV open, ocean requirements. ABS requires approximately 1 metric tonne of modifications for the same notation. As discussed above, the shear requirements for DNV stiffening present governing criteria that far surpass the criteria for section modulus. ABS does not have any specific criteria for the shear area of stiffening subjected to slam loads.

6.4 VEHICLE DECK STRUCTURE & FINITE ELEMENT ANALYSIS

As demonstrated in Section 5, it was necessary to develop various finite element analyses to confirm the vehicle deck structure in way of the different wheel loads acting on the deck. The primary purpose of the FEA is to analyze the vehicle deck plating subjected to the tire loads. Analysis is also provided for the deck stiffening but it is not proposed that the stiffeners need the FEA – their design can be accomplished using the rule procedures for both ABS and DNV.

To analyze the plate structure FEA was developed for tire footprints that have a breadth greater than the vehicle deck stiffener spacing. Of these tires, there are two basic scenarios (please refer to Figure 6-3):

1. Tire A - Tires with a breadth that is only nominally greater than the stiffener spacing,

2. Tire B - Tires with a breadth significantly greater than the stiffener spacing and, in some instances, approaching twice the stiffener spacing.

In order to determine the maximum plate bending stresses it was necessary to investigate two fundamental tire locations relative to the structural arrangement of the vehicle deck. These are shown schematically in Figure 6-3 and summarized as:

1. Tire A/Tire B centered along the length of the deck longitudinal with the centerline of the tire centered between deck longitudinals.

2. Edge of the Tire A/Tire B footprint at the end of the deck longitudinal with the centerline of the tire centered between the deck longitudinals.

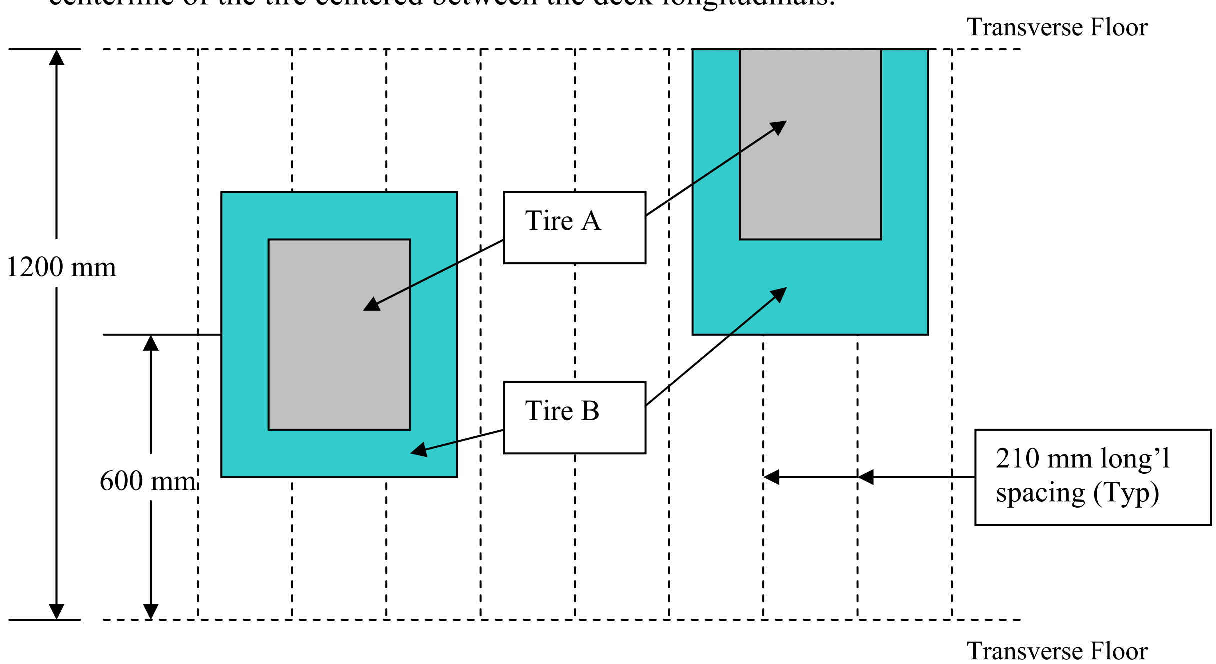


Figure 6-3. Tire Footprint Schematic for Plate Design on Vehicle Deck

In order to fully analyze the vehicle deck structure, the footprints shown in Figure 6-3 had to be analyzed on both the light and heavy vehicle deck extrusions using both the ABS and DNV design accelerations. The results of these analyses are shown below.

The schematic in Figure 6-4 presents the tire footprint locations studied to investigate the stiffener requirements. These are summarized as:

1. Tire A/Tire B centered along the length of the longitudinal with the centerline of the tire aligned with a deck longitudinal. This will result in the maximum moment acting in the deck longitudinal directly under the center of the tire.

2. Edge of the Tire A/Tire B footprint at the end of the longitudinal with the centerline of the tire aligned with the deck longitudinal. This will produce the maximum shear in the stiffener.

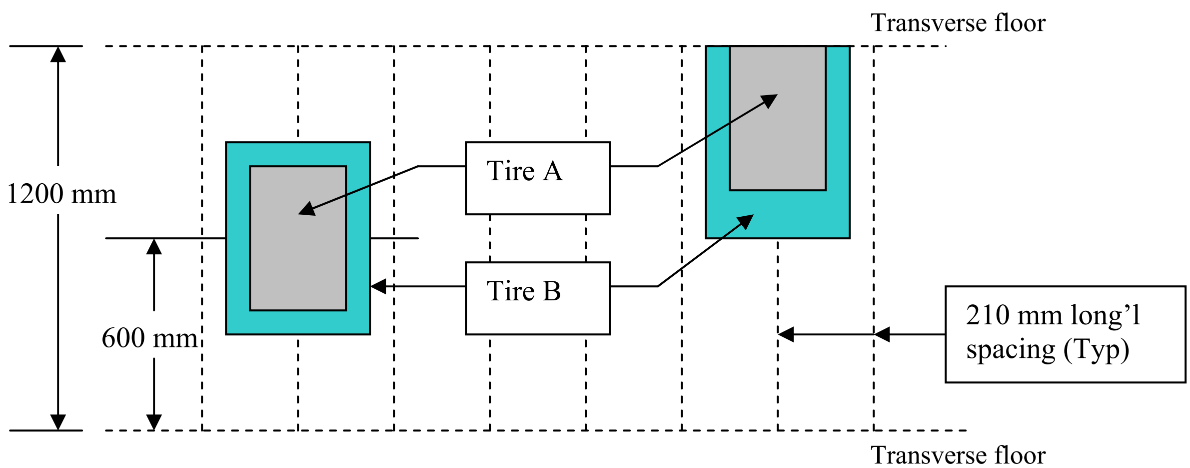
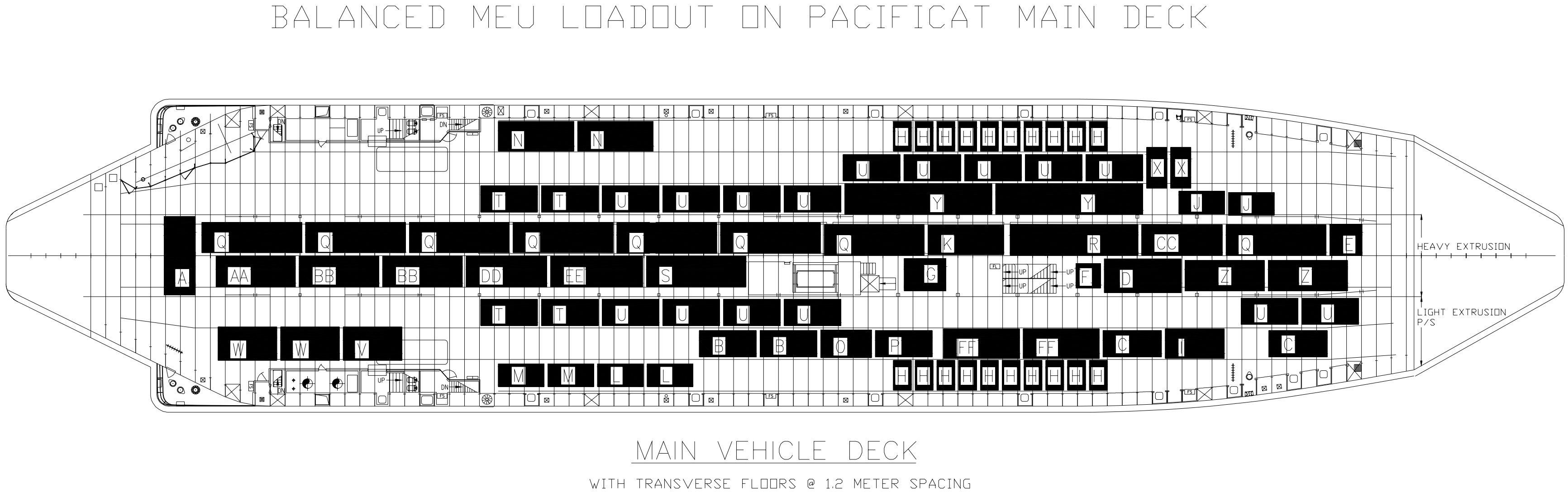


Figure 6-4. Tire Footprint Schematic for Stiffener Design on Vehicle Deck

The tire loads were analyzed for the following vehicles

- MK-48 LVS, Front Power Unit
- HMMWV
- LAV, Maintenance Recovery
- Chassis, Trailer, General Purpose M353 (footprint fits between deck longitudinals)

As a result of the structural analysis performed for the vehicle deck a new vehicle arrangement has been developed and is shown in Figure 6-5. The vehicles are arranged to include 30 cm (approximately 12 inches) between their bumpers. The US Marine Corps [3] only require 10 inches so additional space could be reclaimed if necessary. The loadout shown in Figure 6-5 has been developed with attention to balancing the load port/starboard, i.e., the balanced MEU loadout. The trim of the vessel needs to be addressed at the time of loadout and vehicles loaded in conjunction with other factors contributing to the trim of the vessel.



DES	ITEM	UNIT WEIGHT [LBS]
A	AN/MLQ-36 (1)	28,000
B	AN/MRC-138B (2)	6,200
C	AN/MRC-145 (2)	6,200
D	ARMORED COMBAT EXCAVATOR (1)	36,000
E	TRK, FORKLIFT (1)	25,600
F	TRK, FORKLIFT 4K (1)	11,080
G	TRAM (1)	35,465
H	CONTAINER, QUADRUPLE (20)	5,000
I	CHASSIS TRAILER, GENERAL PURPOSE M353 (1)	2,720
J	CHASSIS TRAILER, 3/4T (2)	1,840
K	POWER UNIT, FRT (LVS) MK-48 (1)	25,300
L	TRLR CARGO 3/4T (M101A1) (2)	1,850
M	TRAILER CARGO M105 (2)	6,500
N	TRLR, MK 14 (2)	16,000
O	TRLR TANK WATER 400 GL (1)	2,530
P	TRK AMB 2 LITTER (1)	6,000
Q	TRK 7-T MTVR (8)	36,000
R	TRK 7-T M927 EXTENDED BED (1)	37,000
S	TRK 7-T DUMP (1)	31,888
T	TRK TOW CARRIER HMMWV (4)	7,200
U	TRK, MULTI-PURPOSE M998 (15)	6,500
V	TRK, AVENGER/CLAWS (1)	7,200
W	TRK ARMT CARR (2)	7,000
X	TRK LIGHT STRIKE VEHICLE (2)	4,500
Y	155MM HOWITZER (2)	9,000
Z	LAV ANTI TANK (AT) (2)	24,850
AA	LAV C2 (1)	26,180
BB	LAV ASSAULT 25MM (2)	24,040
CC	LAV LOGISTICS (1)	28,200
DD	LAV, MORTAR CAR (1)	23,300
EE	LAV, MAINT RECOV (1)	28,400
FF	MAINT VAN (2)	10,000

Figure 6-5. New Military Vehicle Arrangements with Transverse Floor Locations

6.4.1 Transverse Vehicle/Tire Orientation

All analyses presented in this project investigate the requirements for the Vehicle Deck structure assuming longitudinal orientation of the vehicles on the deck. Most studies of any vehicle deck structure also include analysis for the athwartships orientation of the proposed vehicles in the loadout. There are a number of reasons why the transverse orientations have been eliminated from the studies of this project:

1. The traffic flow of vehicles on this ship is different from typical US Navy displacement vessels. The PacifiCat has centerline pillars and obstacles that make traversing this portion of the ship difficult and defines longitudinal travel lanes along the length of the ship.

2. There is no need to turn vehicles to load or off-load. Access to the Main Vehicle Deck is provided through both the bow and the stern of the ship. Vehicles can be loaded through the stern and off-loaded from the bow (or vice versa) without having to turn any vehicles.

3. Even if a design governing vehicle were to be transversely oriented it will only happen during the loading or off-loading operation, i.e., in port, not at sea where ship motion accelerations greatly increase the design loads seen by the deck structure.

Recognition of this aspect of the PacifiCat helps to minimize the work required for conversion of the ship to support military vehicles. Other HSV conversions may still require the investigation of both transverse and longitudinal orientations of the vehicles. As shown in Figure 6-5, all vehicles are longitudinally oriented.

6.4.2 Structural & Material Properties of the Main Vehicle Deck Extrusions

As discussed above, there are two basic extrusions that comprise the Main Vehicle Deck; a heavy extrusion from ship centerline to 3375 mm off centerline P/S, and a light extrusion that makes up the balance of the deck. Per INCAT DESIGNS Dwg No. 1811/2-201 Rev D, all extrusions on the ship are either 6061-T6 or 6082-T6. ABS only presents material properties for 6061-T6 and DNV uses the same f_1 factor for both alloys. Both ABS and DNV note two different sets of material properties depending on the filler metal used in the welding process. All indications suggest the use of the stronger set of properties is appropriate and so that data will be used in this project.

In accordance with the ABS and DNV Rules, the allowable stresses for the Main Vehicle Deck structure subjected to wheel loads is determined as:

- ABS allowable stress, deck plate = $0.60\sigma_y$ from ABS 3-2-3/Table 2, where σ_y is the welded strength of the material.
- ABS allowable stress, stiffening = $0.33\sigma_y$ from ABS 3-2-4/Table 1, where σ_y is the welded strength of the material.

Where: ABS uses a value of $\sigma_{y\ welded}$ = 20,000 psi = 138 N/mm^2

- DNV allowable stress, deck plate = 180f_1
- DNV allowable stress, stiffening = 160 f_1

With f_1 = 0.48 for both 6061-T6 and 6082-T6.

This information is used to develop the allowable stresses presented in Table 6-24, which demonstrates the higher allowable stresses used by DNV for the secondary loads/stresses in this area.

Table 6-24. ABS & DNV Allowable Stresses for Vehicle Deck Structure

	ABS Allowable Stress (6061-T6) (N/mm^2)	DNV Allowable Stress (6061-T6) (N/mm^2)	ABS Allowable Stress (6061-T6) (psi)	DNV Allowable Stress (6061-T6) (psi)
Vehicle Deck Plate	82.7	86.4	12,000	12,530
Vehicle Deck Stiffening	55.2	76.8	6600	11,140

The data in Table 6-25 summarizes the physical dimensions and resulting properties of the extrusions on the Vehicle Deck. The physical properties were estimated from Catamaran Ferries International Dwg No S101 Midship, Rev 0 Midship Section – Ship #3 At Frames 28 and 43, which was available as an AutoCad file. While some of the dimensions could be measured directly from the drawing, such as deck plate thickness, web thickness and breadth of flange, others had to be estimated based on other data ascertained from the drawing. For instance, the enclosed area of the stiffener was determined from the AutoCad drawing and used to determine a reasonable balance of flange and web dimensions to approximate the flange area of the stiffener. Efforts to obtain this information directly from the builder or designer were not successful.

Table 6-25. Mechanical Properties - Heavy & Light Extrusions on Vehicle Deck

Property	Heavy Extrusion	Light Extrusion
Web Height, mm (in)	127.8 (5.032)	95.0 (3.740)
Web Thickness, mm (in)	7.0 (0.276)	4.3 (0.167)
Flange Breadth, mm (in)	50 (1.969)	50 (1.969)
Flange Thickness, mm (in)	9.6 (0.380)	8.7 0.341)
Deck Plate Thickness, mm (in)	9.8 (0.386)	8.0 (0.315)
Stiffener Spacing, mm (in)	210 (8.268)	200 (7.874)
Moment of Inertia, cm^4 (in^4)	981 (23.6)	426.6 (10.2)
Section Modulus, Flange, cm^3 (in^3)	93.3 (5.7)	52.9 (3.2)
Section Modulus, Deck cm^3 (in^3)	233.2 (14.2)	137.6 (8.4)

6.4.3 Summary of the Finite Element Models

There were two basic finite element models developed for this task, one each for the light and heavy extrusion. They are very similar and are shown below in Figure 6-6 and Figure 6-7, respectively. A very fine mesh was developed for these models in order to more easily accommodate all the different tire footprints that needed to be placed into the model. The models were built and analyzed using the FEMAP/NE NASTRAN package. The metric dimensions of the extrusions were converted into English measure for analysis using pounds and inches for all modeling and analysis. Each of the models has a breadth equal to the breadth of eight extrusions and a span of 4.8 meters, i.e., the length of four extrusions. The outer perimeter of the models is supported by simple support boundary conditions. The presence of the transverse floors, spaced at 1.2 meters, are developed in the model using simple support boundary conditions, i.e., they allow rotation about the transverse axis at the ends of the longitudinal stiffener spans and are fixed against all translation. No structure is included in either model to analyze the transverse floors, which have an upper strake of 10mm plate, 252 mm high. The upper strake of plate is supported by a 100 x 10mm flatbar that is used to land the extrusion. See discussion below on Fabrication of Main Vehicle Deck and Transverse Floors.

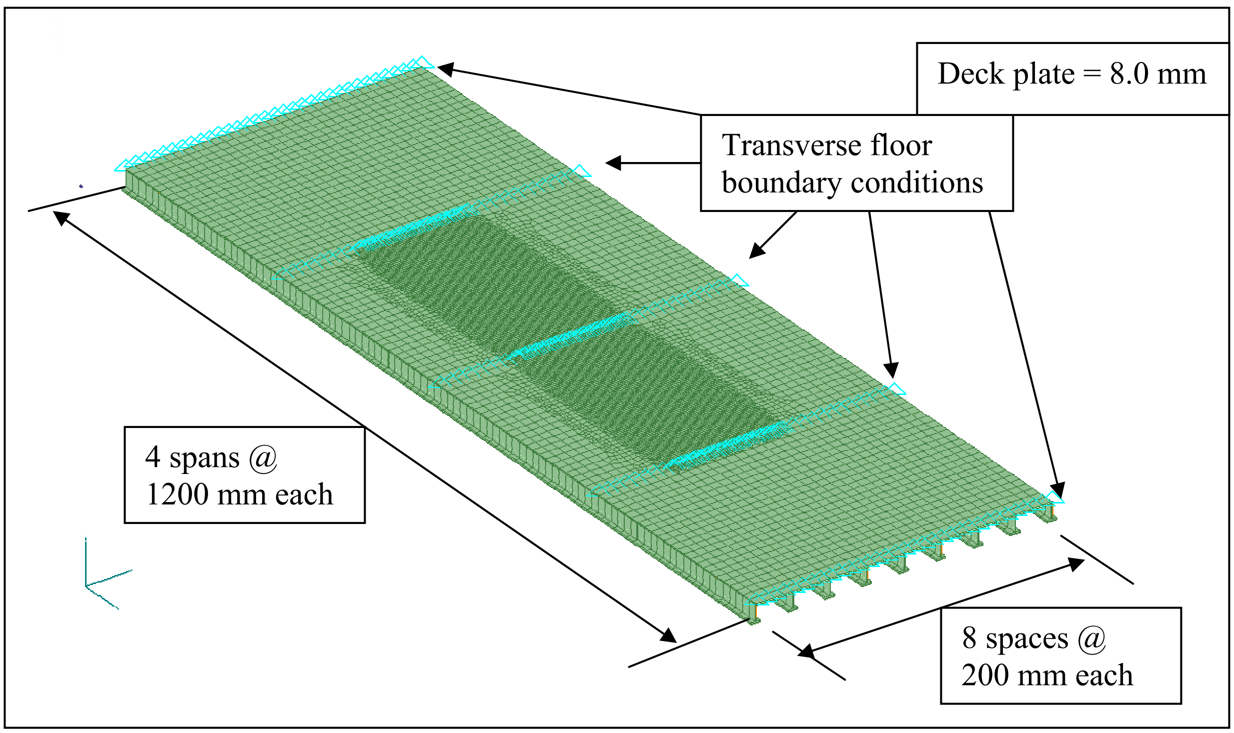


Figure 6-6. Finite Element Model Used for Light Extrusion

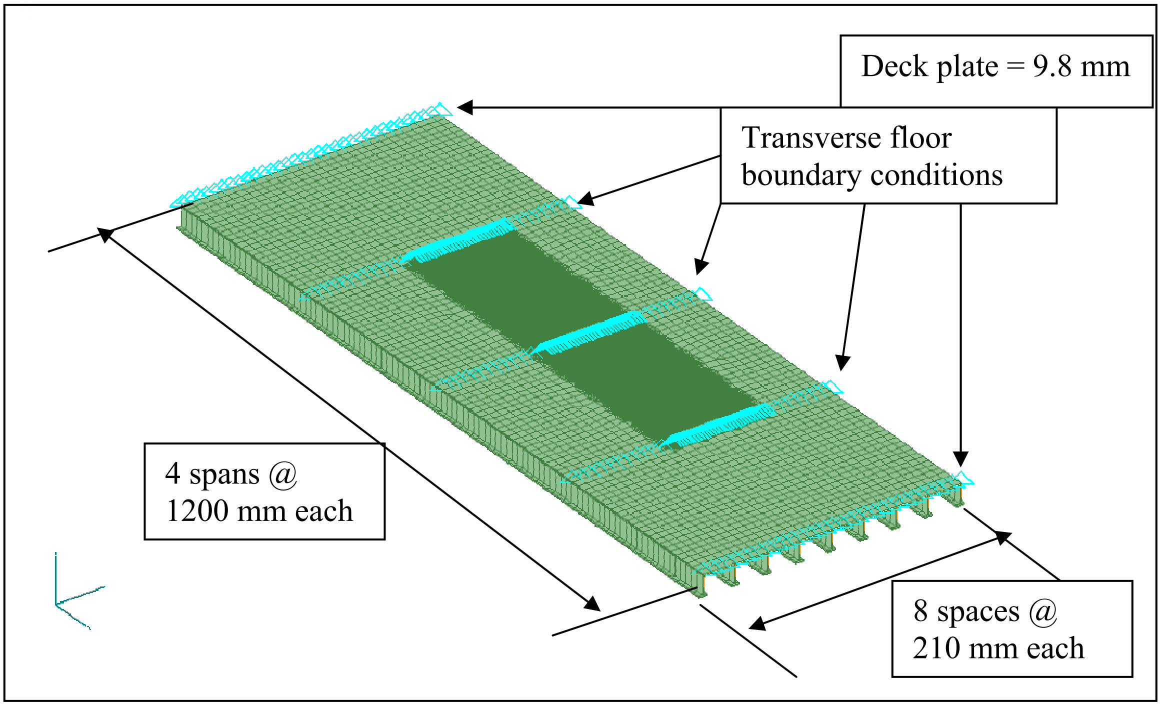


Figure 6-7. Finite Element Model Used for Heavy Extrusion

6.4.4 Fabrication of Main Vehicle Deck and Transverse Floors

Although not specifically relevant to the finite element model, this is a good location to present discussion on the actual fabrication of the Main Vehicle Deck, which consists of the light and heavy extrusions discussed above and transverse floors every 1.2 meters. Each of the two extrusions has a length of 1.2 meters and is welded into the transverse floor at each end. The extrusions are intercostal between the floors, i.e., they are not continuous as typically envisioned of vehicle deck structure in a US Navy ship. The flanges of the stiffeners on the extrusion rest on the 100 x 10 mm flatbar mentioned above, which is approximately 162 mm below the upper edge of the floor. The flatbars for both extrusions are located such that the upper edge of the transverse floor is approximately 10mm above the upper surface of the extrusion, i.e., the floor "sticks up" above the deck surface enough to accommodate the weld required for the support of the extrusion. Details of this fabrication are taken from the midship section drawing, Figure 6-8, and in more detail in Figure 6-9. This implies that the upper edges of the transverse floors actually protrude above the road surface of the Main Vehicle Deck and are part of the roadway surface contacted by the vehicle tires. Figure 6-23 provides an additional close-up view of this detail.

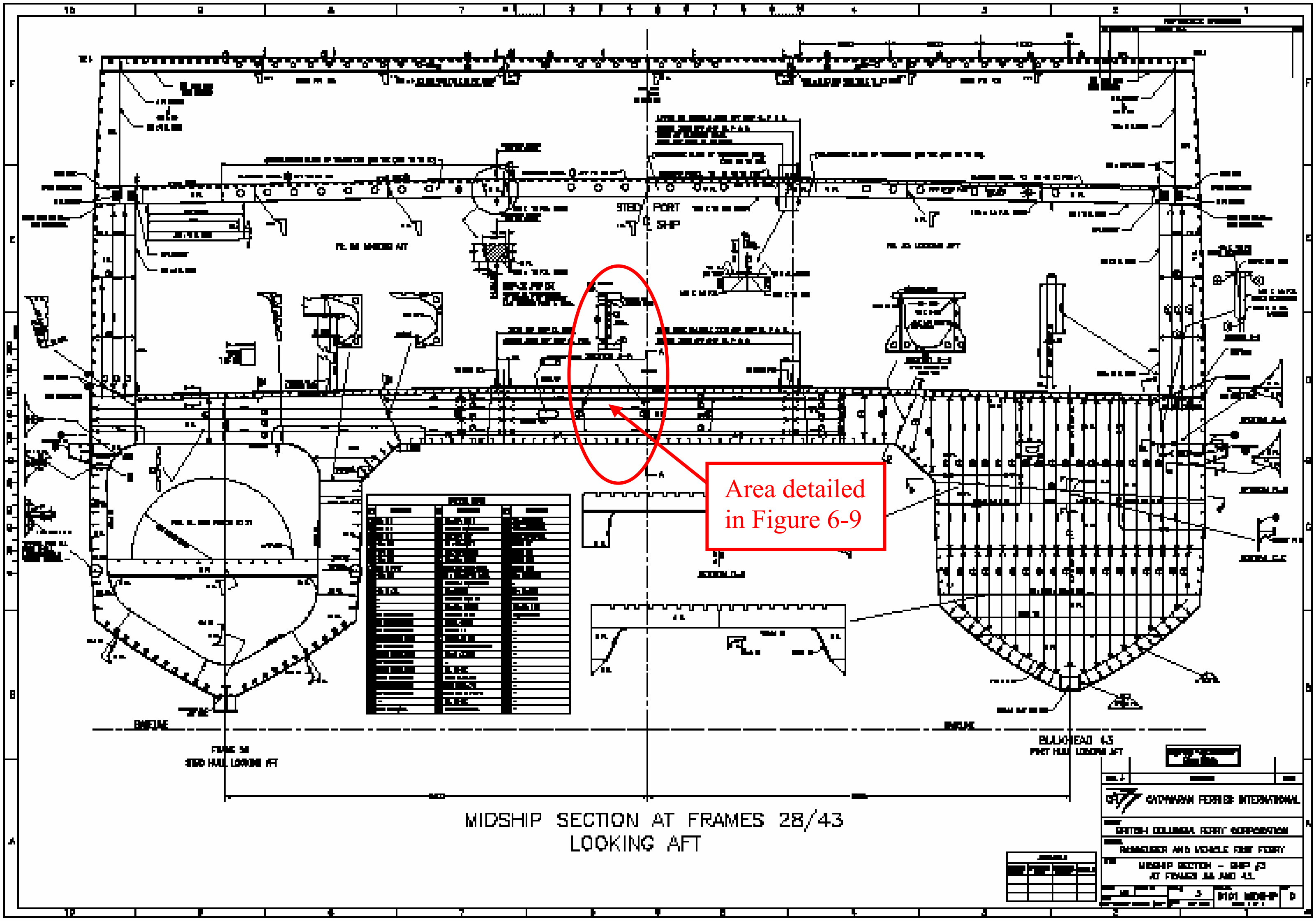


Figure 6-8. Midship Section of PacifiCat

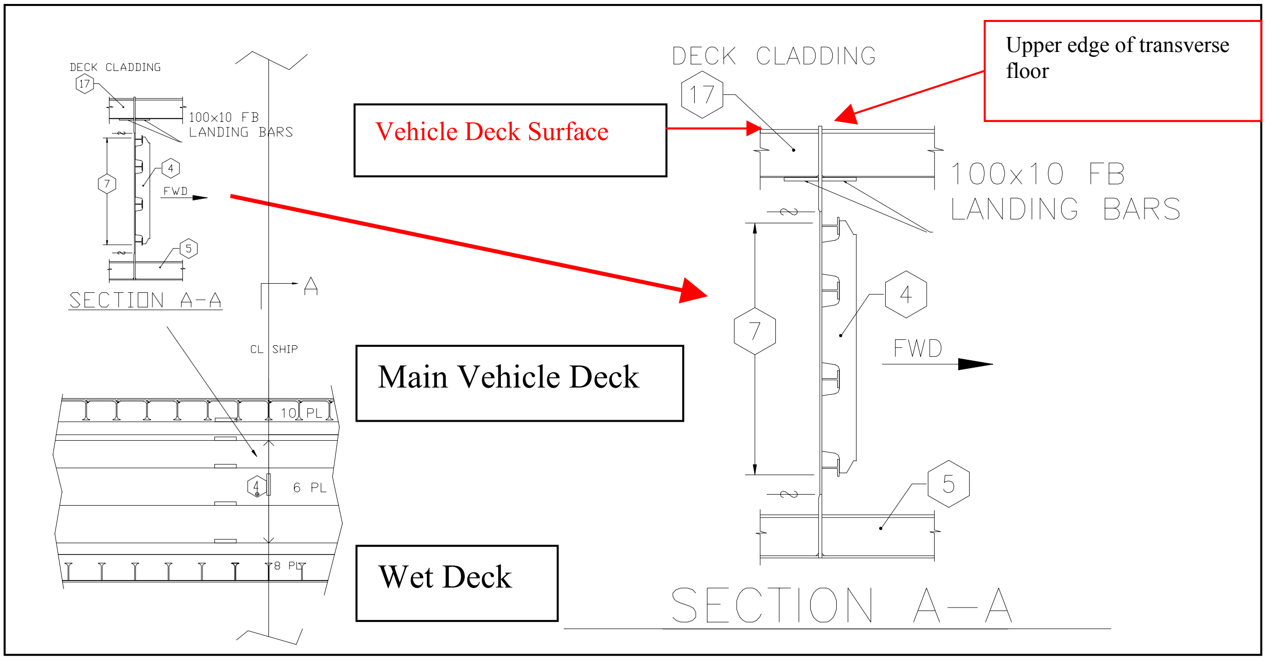


Figure 6-9. Midship Section Details IWO Main Vehicle Deck Fabrication

6.4.5 Summary of Vehicle Deck FEA Results

The finite element analyses were developed to investigate the response of the vehicle deck plate to the tire loads. It was felt that this was necessary because many of the tire footprints exceed the stiffener spacing, which violates the assumptions in the ABS deck plate formulations for wheel loaded decks. The DNV rules allow for this situation but the FEA also includes DNV accelerations and comparison to DNV allowable stresses for completeness and consistency of the work within this project.

There were a large number of analyses that were performed for this FEA work. The tire locations in Figure 6-3 and Figure 6-4 were analyzed for ABS and DNV loads on both the heavy and light extrusions. The results of all the analyses are presented below in Table 6-26 through Table 6-33.

The loads in the tables represents the static tire footprints increased by the vertical acceleration for either ABS or DNV assuming their maximum vertical accelerations, i.e., assuming the vehicle is located in the bow of the ship. As such, many of these results are conservative. These static tire footprint loads are shown in Table 5-22 and Table 5-24.

All of the FEA work was done using inch/pound units. The data available from the vehicles was all in English units and it simplified the analysis to use the English system of measure although all the ship structure is metric. The plate bending stresses are calculated in psi and converted to N/mm^2 for presentation in Table 6-26 through Table 6-33.

The Load Case shown in the tables below is used as a reference for discussion of specific governing load cases. There are two numbers in each Load Case cell. The left hand number refers to the Load Case for the Tire @ Midspan while the right hand number refers to the Load Case for the Tire @ End of stiffener.

To summarize from above, the static footprints for each of the tires used in the analysis are:

- LAV 11" x 22" with a total tire load = 3770 pounds
- HMMWV 12.5" x 14" with a total tire load = 1950 pounds
- MK48 LVS 16" x 27" with a total tire load = 6830 pounds
- M353 Chassis 7" x 12" with a total tire load = 1360 pounds

With extrusion properties:

- Light Extrusion: 8 mm plate with 200 mm stiffener spacing (0.315" plate with stiffener spacing = 7.87")
- Heavy Extrusion: 9.8 mm plate with 210 mm stiffener spacing (0.386" plate with stiffener spacing = 8.27")

As such, the M353 chassis will fit between the stiffeners of both extrusions when centered on the plate. None of the other tires fit within the stiffener spacing, regardless of location.

Figure 6-10 through Figure 6-13 are provided as a few examples of some of the load input and respective output files for the plate bending stress. All of these figures provide input and results for the tire load at the center of the plate panel using ABS vertical accelerations.

The stresses shown in Table 6-26 through Table 6-33 compare directly to the allowable stresses associated with both ABS and DNV. Both rule sets define allowable stresses associated with plates subjected to normal loads. These stresses are referred to as normal bending stresses, i.e., they are oriented along the principal bending axes of the element being investigated and are parallel to the longitudinal and transverse axes of the ship. Many FEA results calculate and report the Von Mises stresses acting in plate elements. This type of stress represents different criteria than the nominal allowable stresses cited by ABS and DNV for typical design associated with rule based calculations. The stresses presented in the tables below, i.e., the "Plate Bending Stress", are normal bending stresses parallel to the transverse axis of the ship, i.e., they are acting perpendicular to the longitudinal deck stiffeners across the short dimension of the plate panel.

Table 6-26. ABS Load, Light Extrusion, Tire Centered on Plate

Vehicle	Load Case	Load psi	Plate Bending Stress Tire @ Midspan psi (N/mm^2)	Plate Bending Stress Tire @ End psi (N/mm^2)
LAV	1/5	19.3	5941 (44.3)	5128 (41.2)
HMMWV	2/6	14.1	3680 (25.4)	2848 (19.6)
Mk48 LVS	3/7	19.6	5236 (36.1)	4780 (33.0)
Chassis Tri GEN purpose m353	4/8	20.1	5105 (35.2)	3965 (27.3)

Table 6-27. ABS Load, Light Extrusion, Tire Centered on Stiffener

Vehicle	Load Case	Load psi	Plate Bending Stress Tire @ Midspan psi (N/mm^2)	Plate Bending Stress Tire @ End psi (N/mm^2)
LAV	9/13	19.3	4210 (31.1)	3671 (29.1)
HMMWV	10/14	14.1	3044 (21.0)	3852 (26.6)
Mk48 LVS	11/15	19.6	5282 (36.4)	5357 (36.9)
Chassis Tri GEN purpose m353	12/16	20.1	2429 (16.7)	2850 (19.6)

Table 6-28. ABS Load, Heavy Extrusion, Tire Centered on Plate

Vehicle	Load Case	Load psi	Plate Bending Stress Tire @ Midspan psi (N/mm^2)	Plate Bending Stress Tire @ End psi (N/mm^2)
LAV	17/21	19.3	4171 (30.8)	3760 (28.8)
HMMWV	18/22	14.1	2547 (17.6)	2049 (14.1)
Mk48 LVS	19/23	19.6	3672 (25.3)	3380 (23.3)
Chassis Tri GEN purpose m353	20/24	20.1	3543 (24.4)	2825 (19.5)

Table 6-29. ABS Load, Heavy Extrusion, Tire Centered on Stiffener

Vehicle	Load Case	Load psi	Plate Bending Stress Tire @ Midspan psi (N/mm^2)	Plate Bending Stress Tire @ End psi (N/mm^2)
LAV	25/29	19.3	2796 (17.2)	3357 (21.9)
HMMWV	26/30	14.1	2539 (17.5)	2914 (20.1)
Mk48 LVS	27/31	19.6	1569 (10.8)	2163 (14.9)
Chasis Tri GEN purpose m353	28/32	20.1	3653 (25.2)	4787 (33.0)

Table 6-30. DNV Load, Light Extrusion, Tire Centered on Plate

Vehicle	Load Case	Load psi	Plate Bending Stress Tire @ Midspan psi (N/mm^2)	Plate Bending Stress Tire @ End psi (N/mm^2)
LAV	33/37	37.4	11,498 (86.1)	9925 (79.8)
HMMWV	34/38	27.3	7122 (49.1)	5513 (38.0)
Mk48 LVS	35/39	37.9	10,133 (69.9)	9251 (63.8)
Chasis Tri GEN purpose m353	36/40	38.8	9880 (68.1)	7674 (52.9)

Table 6-31. DNV Load, Light Extrusion, Tire Centered on Stiffener

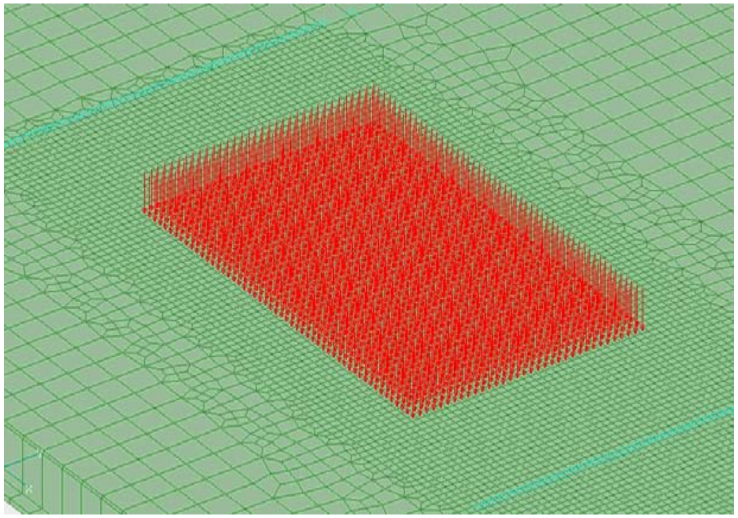
Vehicle	Load Case	Load psi	Plate Bending Stress Tire @ Midspan psi (N/mm^2)	Plate Bending Stress Tire @ End psi (N/mm^2)
LAV	41/45	37.4	8148 (60.2)	7105 (56.3)
HMMWV	42/46	27.3	5892 (40.6)	7455 (51.4)
Mk48 LVS	43/47	37.9	10,224 (70.5)	10,369 (71.5)
Chasis Tri GEN purpose m353	44/48	38.8	4701 (32.4)	5515 (38.0)

Table 6-32. DNV Load, Heavy Extrusion, Tire Centered on Plate

Vehicle	Load Case	Load psi	Plate Bending Stress Tire @ Midspan psi (N/mm^2)	Plate Bending Stress Tire @ End psi (N/mm^2)
LAV	49/53	37.4	8073 (59.6)	7104 (55.8)
HMMWV	50/54	27.3	4930 (34.0)	3966 (27.3)
Mk48 LVS	51/55	37.9	7107 (49.0)	6543 (45.1)
Chassis Tri GEN purpose m353	52/56	38.8	6858 (47.3)	5468 (37.7)

Table 6-33. DNV Load, Heavy Extrusion, Tire Centered on Stiffener

Vehicle	Load Case	Load psi	Plate Bending Stress Tire @ Midspan psi (N/mm^2)	Plate Bending Stress Tire @ End psi (N/mm^2)
LAV	57/61	37.4	5413 (39.9)	4838 (42.4)
HMMWV	58/62	27.3	4566 (31.5)	5640 (38.9)
Mk48 LVS	59/63	37.9	3037 (20.9)	4186 (28.9)
Chasis Tri GEN purpose m353	60/64	38.8	7071 (48.8)	9265 (63.9)



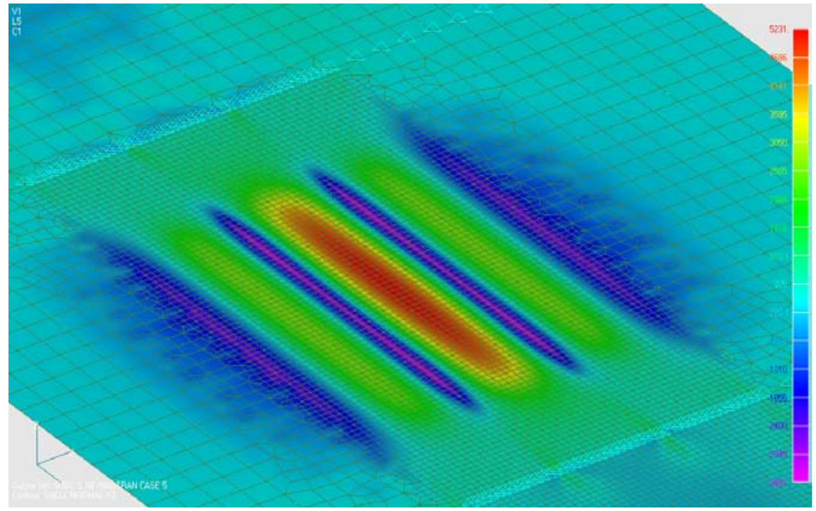
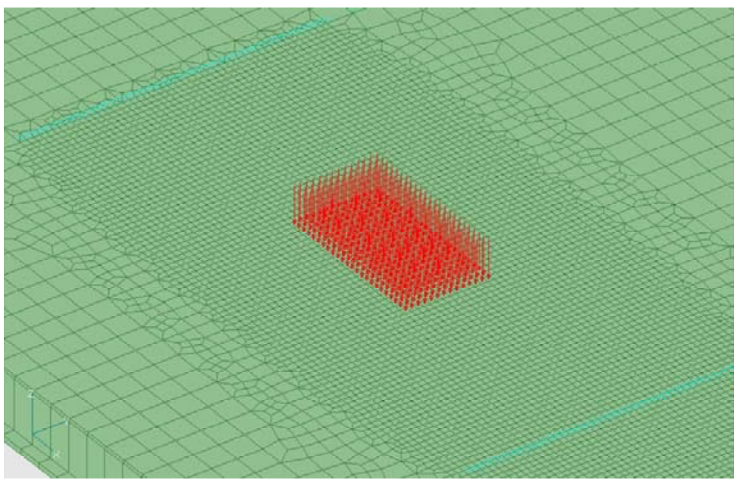


Figure 6-10. Mk 48 Applied Load and Plate Bending Stresses on Light Extrusion



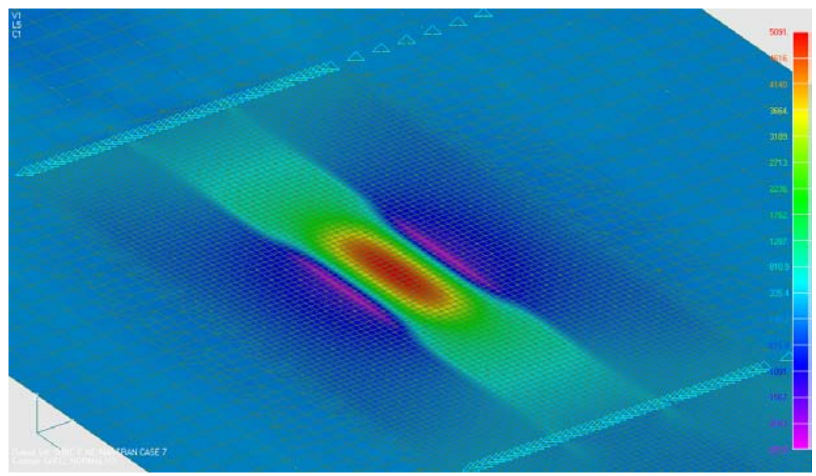


Figure 6-11. 353 Chassis Applied Load and Plate Bending Stresses on Light Extrusion

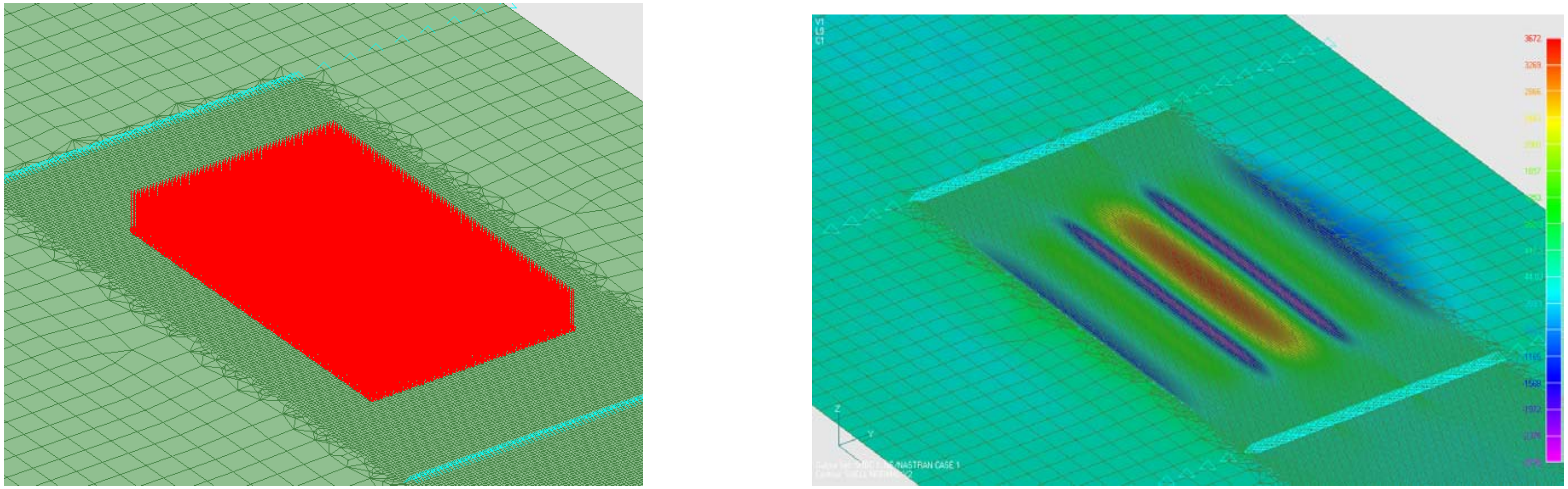


Figure 6-12. Mk 48 Applied Load and Plate Bending Stresses on Heavy Extrusion

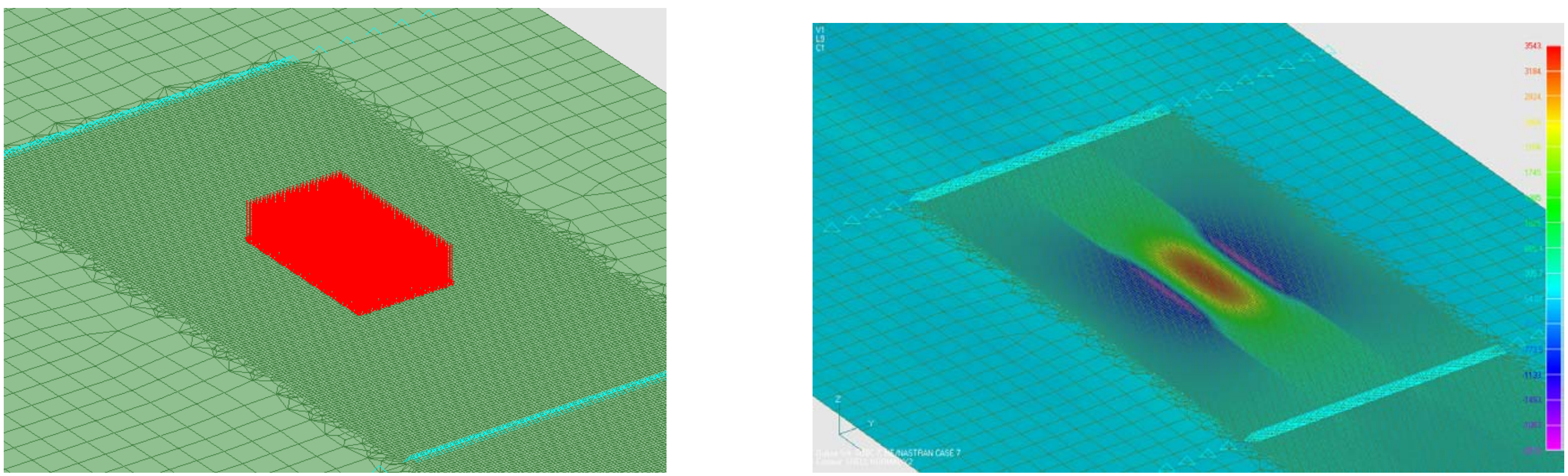


Figure 6-13. 353 Chassis Applied Load and Plate Bending Stresses on Heavy Extrusion

From Table 6-24 it is seen that the ABS and DNV allowable plate stresses are 12,000 psi and 12,530 psi, respectively. In no case are these stresses exceeded by the military vehicle loadout proposed for this project.

A special case was also run using the Mk 48 centered on the plate panel but straddling a transverse floor resulting in a load on two adjacent panels in the longitudinal direction. The analysis was done using the DNV vertical acceleration loads as input. This was of interest due to the possibility of higher Von Mises stresses due to the bi-directional stress field at the end of the plate panel compared to the center of the panel. While the results of this analysis do not define a governing condition it was interesting to note the actual results:

- Plate bending stress Y direction 6671 psi
- Plate bending stress X direction 6958 psi
- Von Mises stress 7159 psi

From Table 6-32 it is seen that the stresses are greater from that loading scenario, but not much.

6.4.5.1 Discussion of Results of Analysis on ABS and DNV Loads on Heavy and Light Extrusions

As stated above, the deck plate requirement for wheeled vehicles in the ABS rules are formulated on the assumption of simple support boundary conditions, i.e., no resistance to rotation around the perimeter of the plate panel. The M353 satisfies these conditions because the entire footprint lands on a single plate panel when centered on the panel. None of the other footprints fit within a single panel and the portion of the footprint, which overhangs into the adjacent panel will affect the stresses in the most heavily loaded, central panel.

6.4.5.2 Simple Support & Fixed Boundary Conditions

The two classic boundary conditions for much of structural analysis are simple support and fixed support. Simple support boundary conditions allow rotation about the principal bending axis along the side of the plate considered to be simply supported, in many cases, this includes all four edges of the plate panel. The other five degrees of freedom are considered fixed.

With fixed boundary conditions, all six degrees of freedom are fixed implying that there is resistance to rotation about the principal bending axis, it is not free to rotate as in simply supported conditions.

When the load on a plate panel is contained completely within that panel it is assumed to be free to rotate about its edges similar to simple support conditions. The portions of any tire load that overlaps onto adjacent panels effectively restrains the edge of the centrally loaded panel from rotating and helps to affect a fixed condition. This not only changes the stress magnitudes in the various plate panels, it also redistributes the stresses.

In a simply supported plate panel the plate bending stresses are zero along the boundary of the panel and maximum in the center of the panel. In a fixed plate, the bending stresses are maximum at the center of the long edge and half that value at the center of the panel. In accordance with Roark and Young, "Formulas for Stress and Strain" [6], the values for plate bending stress for simple and fixed support plate panels are given as follows:

- Simple support $$Max\sigma = \sigma_b = \frac{\beta q b^2}{t^2}$$ at center of plate panel
- Fixed Support $$Max\sigma = \sigma_b = \frac{\beta_1 q b^2}{t^2}$$ at center of long edge

$$\sigma = \sigma_b = \frac{\beta_2 q b^2}{t^2}$$ at center of plate panel

where: β, β_1 β_2 are constants given below, see Table 6-34
q = uniform pressure load over the entire plate panel
b = short dimension of the plate panel
a = long panel dimension
t = panel thickness

Table 6-34. Plate Bending Stress Coefficients From Roark & Young

a/b	1.0	1.2	1.4	1.6	1.8	2.0	∞
β	0.2874	0.3762	0.4530	0.5172	0.5688	0.6102	0.7500
β_1	0.3078	0.3834	0.4356	0.4680	0.4872	0.4974	0.5000
β_2	0.1386	0.1794	0.2094	0.2286	0.2406	0.2472	0.2500

It is obvious from these equations that for a given support condition and location the stresses will vary as a function $(b/t)^2$. For the light extrusion $(b/t)^2 = (200/8.0)^2 = 625$ while for the heavy extrusion $(b/t)^2 = (210/9.8)^2 = 459.2$. Applying these ratios to the stresses in Table 6-26 through Table 6-33 results in fairly close prediction of one stress from an initial stress. For instance starting with Load Case 3, Table 6-26, and trying to predict the stress from Load Case 19, Table 6-28, one would expect a value of 5236 x (459.2/625) = 3847 psi. The actual stress from Load Case 19 is 3672 psi, slightly less than predicted. This is because the Roark equations assume a uniform load over the entire plate, not just a portion of the plate. Regardless, this is fairly good correlation and helps to confirm the accurate behavior of the FEA.

From the equations presented above and their respective values, it is seen that the plate bending stresses at the center of a plate panel should be one-half the value at the side for load on a single panel of fixed plating, i.e., not adjacent plate panels. The original inspiration for developing the FEA was the fact that the footprints that extend over more than one panel would start to induce fixed boundary conditions between the adjacent panels thereby resulting in stress redistributions and reductions compared to the simple support boundary conditions assumed by the ABS design algorithms, shown in Figure 5-15 and Figure 5-16. Table 6-35 and Table 6-36 present summaries of the stresses at the center of the plate panel and the side of the plate panel for the heavy and light extrusion, respectively, for the ABS tire loads in the finite element analysis. The effects are the same for the DNV loads and so the actual data is not repeated for them. These tables also include the ratios of the stresses at these two points of interest.

Table 6-35. ABS Plate Bending Stress @ Center & Edge of Light Extrusion (FEA Based)

Vehicle	Bending Stress @ Plate Edge (psi)	Bending Stress @ Plate Center (psi)	Ratio (Stress @ Center /Stress @ Edge)
LAV	2291	5941	2.59
HMMWV	2064	3680	1.78
MK 48	3491	5236	1.50
M353	2518	5105	2.03

Table 6-36. ABS Plate Bending Stress @ Center & Edge of Heavy Extrusion (FEA Based)

Vehicle	Bending Stress @ Plate Edge (psi)	Bending Stress @ Plate Center (psi)	Ratio (Stress @ Center /Stress @ Edge)
LAV	1956	4171	2.13
HMMWV	1704	2547	1.49
MK 48	2788	3672	1.32
M353	2212	3543	1.60

The results presented in Table 6-35 and Table 6-36 do not agree with the predicted results, i.e., the stresses at the center of the plate panel are not less than the stresses along the edge, in fact, they are greater, indicative of a simple support response.

An interesting relationship is presented in Figure 6-14, which shows the stresses at the center and edge of the plate panel using the heavy extrusion FEA model with the MK 48 tire load and varying the plate thickness from the original 9.8m down to 8mm, 7mm, 6mm and 5mm. This figure shows that as the plate gets lighter the stresses along the edge go up more rapidly than the stresses in the center of the panel. This indicates that as the plate gets lighter it tends to approach the stress distribution anticipated by the β_∞ coefficients presented in Table 6-34, i.e., it starts to behave more like a fixed plate panel as the plate gets lighter. This suggests that the (b/t) ratio for the original heavy extrusion is fairly robust and that the tire load that overlaps onto the adjacent panels is not enough to counteract the rotation induced in the plate from the portion of the tire load on the central panel. As the plate thickness is reduced it is more flexible and the overlapping tire load more capable of offsetting the rotations induced within the central panel.

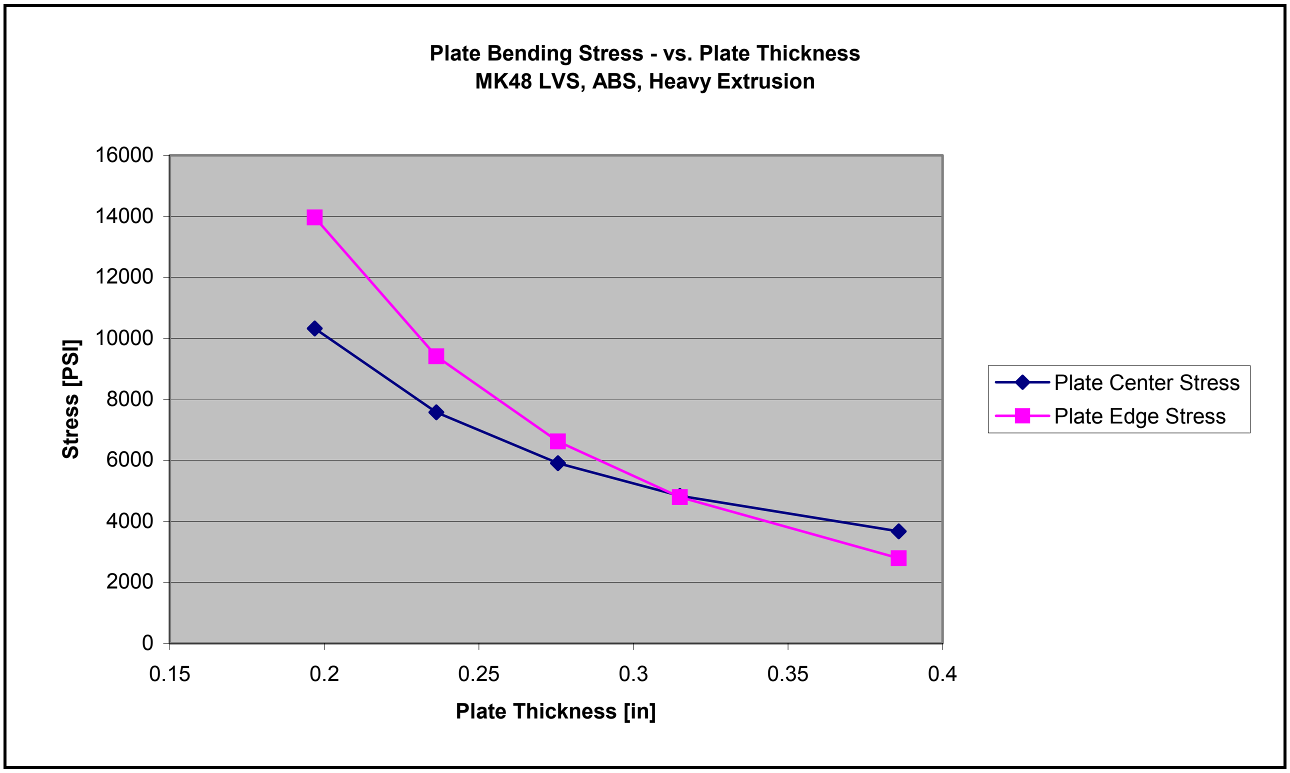


Figure 6-14. Plate Bending Stress vs. Plate Thickness

6.4.6 Structural Optimization Using the ABS Mk 48 Load

A quick comparison is presented using the vehicle deck plate that would be developed using the ABS rules and through the FEA. This is done for the Mk 48 only as a means for developing the comparison. All units of measure in this section are English.

ABS 3-2-3/1.9 presents deck plating requirements for decks subjected to wheel loads. The governing equation is:

$$t = \sqrt{\frac{\beta W\left(1 + 0.5 n_{xx}\right)}{\sigma_a}} \text{ (inches)}$$

where: β = coefficient from 3-2-3/Figure 1
W = static wheel load, pounds
n_{xx} = vertical acceleration at location under consideration = 0.24 g's x 2 per Figure 5-13 for maximum value in the bow.
σ_a = allowable stress from 3-2-3/Table 2 = $0.6\sigma_y$ = 0.6 x 20,000 = 12,000 psi

The Mk 48 tire load is taken from Table 5-22 as 6830 pounds acting over a footprint that is 16" x 27". The stiffener spacing on the plate panel is 8.27" (210 mm). This implies that, when the tire is centered on a given plate panel only (8.27/16) x 6830 = 3530 pounds is on the centrally loaded panel. This is the value that will be taken for W. β = 0.57 is determined from the panel dimension of 8.27" x 47.24" (210 mm x 1200 mm) and the tire footprint of 16" x 27". Using these values results in the following required deck plate thicknesses for the Mk 48 located in the bow and at midship, respectively, within the ship:

$$t = \sqrt{\frac{0.57(3530)(1+0.5(0.48))}{12{,}000}} = 0.456 \text{ inches} = 11.58 \text{ mm for bow location of Mk 48.}$$

$$t = \sqrt{\frac{0.57(3530)(1+0.5(0.24))}{12{,}000}} = 0.433 \text{ inches} = 11.01 \text{ mm for midship location of Mk 48.}$$

There is not a significant difference for the required plate thickness by moving the vehicle from the bow to the midship area of the ship.

From Table 6-28, Case 19, it is seen that the stress in the heavy extrusion with a thickness of 0.386 inches (9.8 mm) is 3672 psi, significantly below the ABS allowable stress of 12,000 psi. To investigate the reduction possible in the deck plate through FEA the model was run with the ABS Mk 48 loads using 5 mm and 6 mm plate. The results of this data are plotted in Figure 6-14 without any specific values for the plate thicknesses. For 6 mm plate the maximum stress is 9410 psi while the 5mm plate experiences a maximum stress of 13,961 psi, the latter exceeding the allowable stress of 12,000 psi. This suggests that, in comparison to the ABS plate design algorithm, which would require 11mm/12mm deck plate, the FEA would allow the vehicle deck to be designed using 6mm plate. Minimum thickness criteria for ruggedness would still have to be checked but this is a good indication of the optimization that could be attained using FEA in this instance. This is particularly noteworthy for new design consideration.

6.4.7 Verification of the Finite Element Results

One model, analyzed with two sets of boundary conditions, was used to verify that the finite element models being used for these analyses were correctly developed and their results correctly interpreted. The analyses are:

- Plate panel, simply supported, (pinned) around its entire perimeter and subjected to a 20 psi uniform load over the entire panel and,
- Plate panel, fixed around its entire perimeter and subjected to the same 20 psi load.

The plate panel was extracted from the model used for the analyses of the heavy extrusion and has the following properties:

s = 8.25”, l = 47.25” t = 0.3858”, E = 9.9×10^6 psi

The results of the FEA verification runs were compared to the theoretical predictions using the equations presented by Roark & Young, “Formulas for Stress and Strain” [6]. The results of the comparison are provided in Table 6-37. All comparisons are acceptable.

Table 6-37. FEA Results vs. Roark Predictions

Case	Finite Element Value	Roark Stress	% Difference
Pinned – Stress at Center	6844 psi	6859 psi	-0.22%
Fixed Support - Stress @ Center	2290 psi	2286 psi	0.17%
Fixed Support - Stress @ Edge	4485 psi	4572 psi	-1.90%
Pinned - Deflection @ Center	0.0228”	0.0231”	-1.30%
Fixed Support, Deflection @ Center	0.00468”	0.00463”	1.08%

6.5 DRAWINGS FOR STRUCTURAL MODIFICATIONS

The ABS and DNV required structural modifications are provided on marked-up drawings in Figure 6-15 through Figure 6-20. A brief summary of the changes is included on the appropriate figure. The modifications shown on these drawings are the same as those used for the weight increase calculations presented in Table 6-17 and Table 6-23.

These drawings show the structural modifications required to meet the slam load requirements for the conversion. There are no modifications required for either global hull girder loads or vehicle deck operation with the MEU loadout defined for this project. None of these drawings are intended to address the local requirements for installation of the tiedowns, details for which are provided later.

The ABS structural modifications are shown in Figure 6-15 through Figure 6-18 and the DNV modifications are shown in Figure 6-19 and Figure 6-20. All modifications are indicated by the red text that has been added to these drawings. There is no separate drawing provided for the side shell flange doublers that need to be added for DNV since they are effectively the same as those required for ABS. These modifications are shown in Figure 6-15 and Figure 6-16.

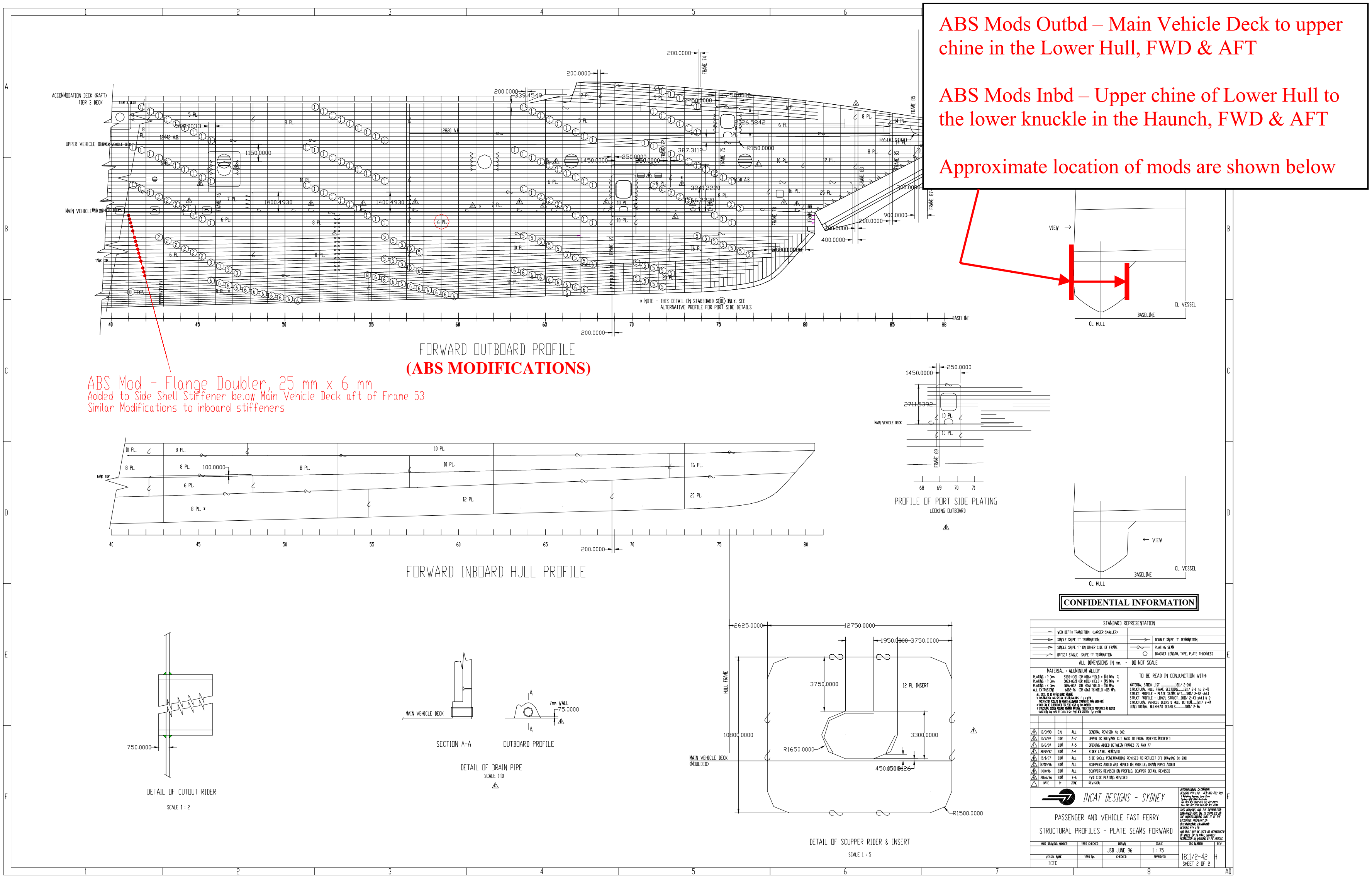


Figure 6-15. ABS Structural Modifications – Forward Portion of Side Shell

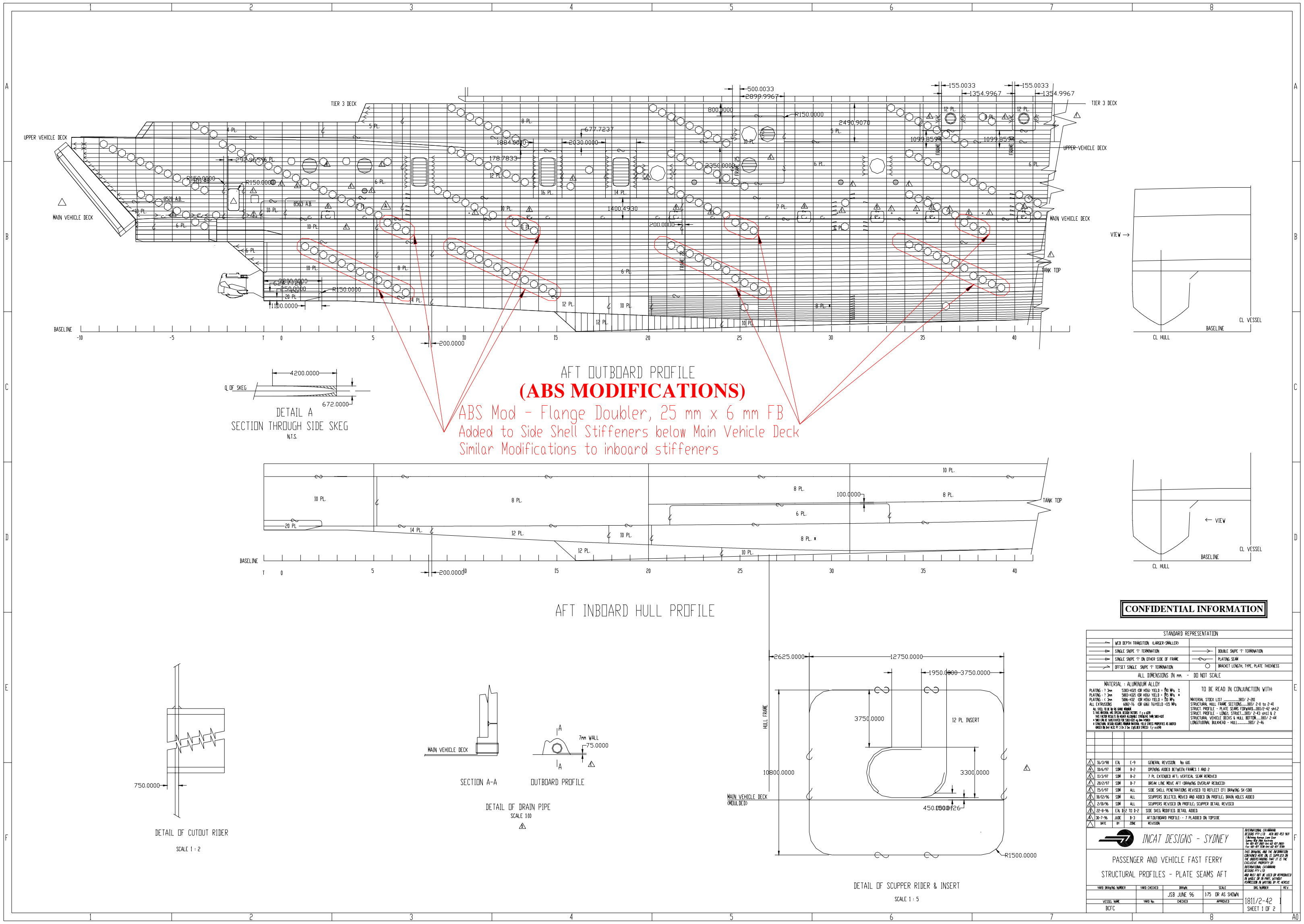


Figure 6-16. ABS Structural Modifications – Aft Portion of Side Shell

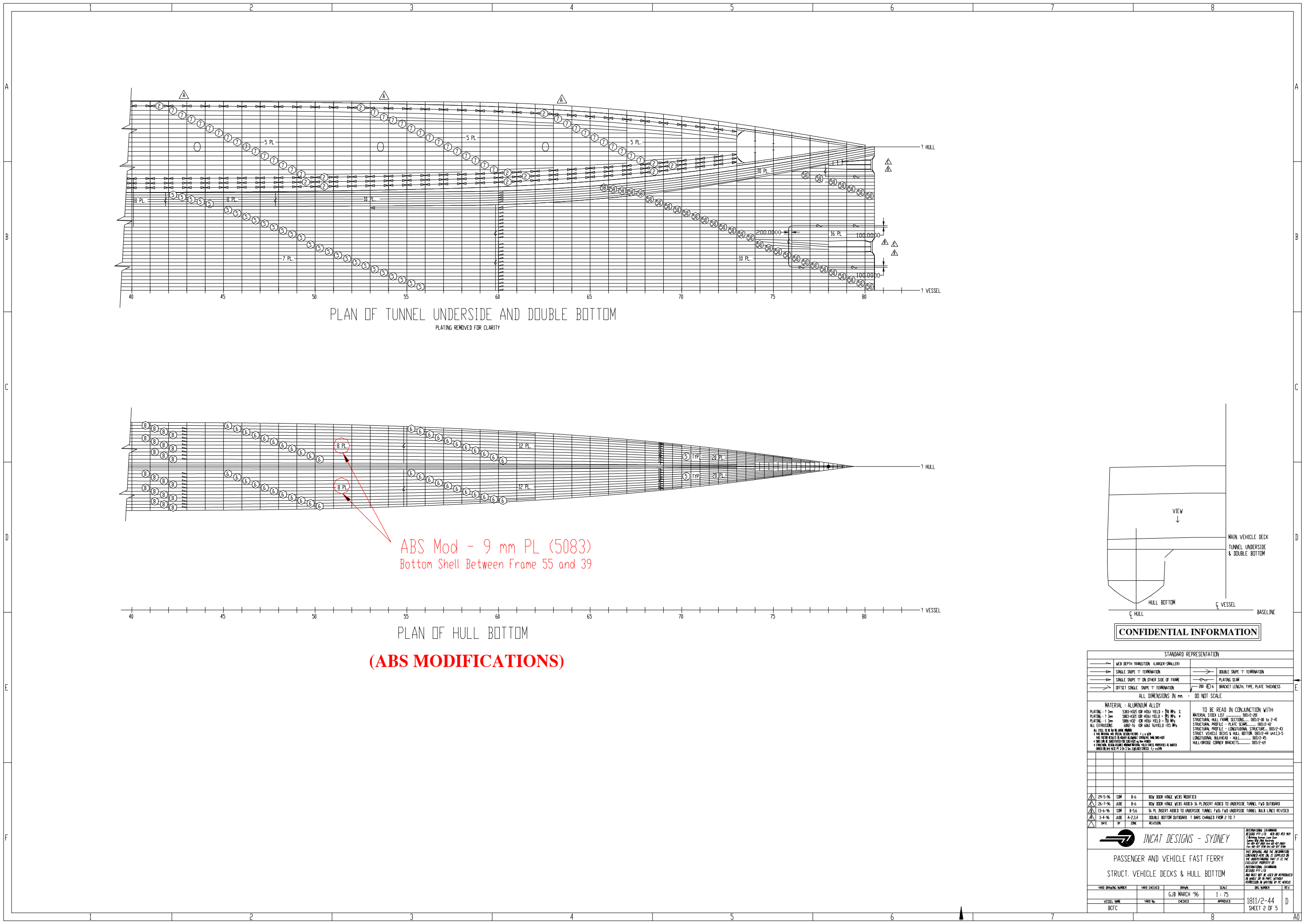


Figure 6-17 ABS Structural Modifications – Forward Portion of Bottom Hull

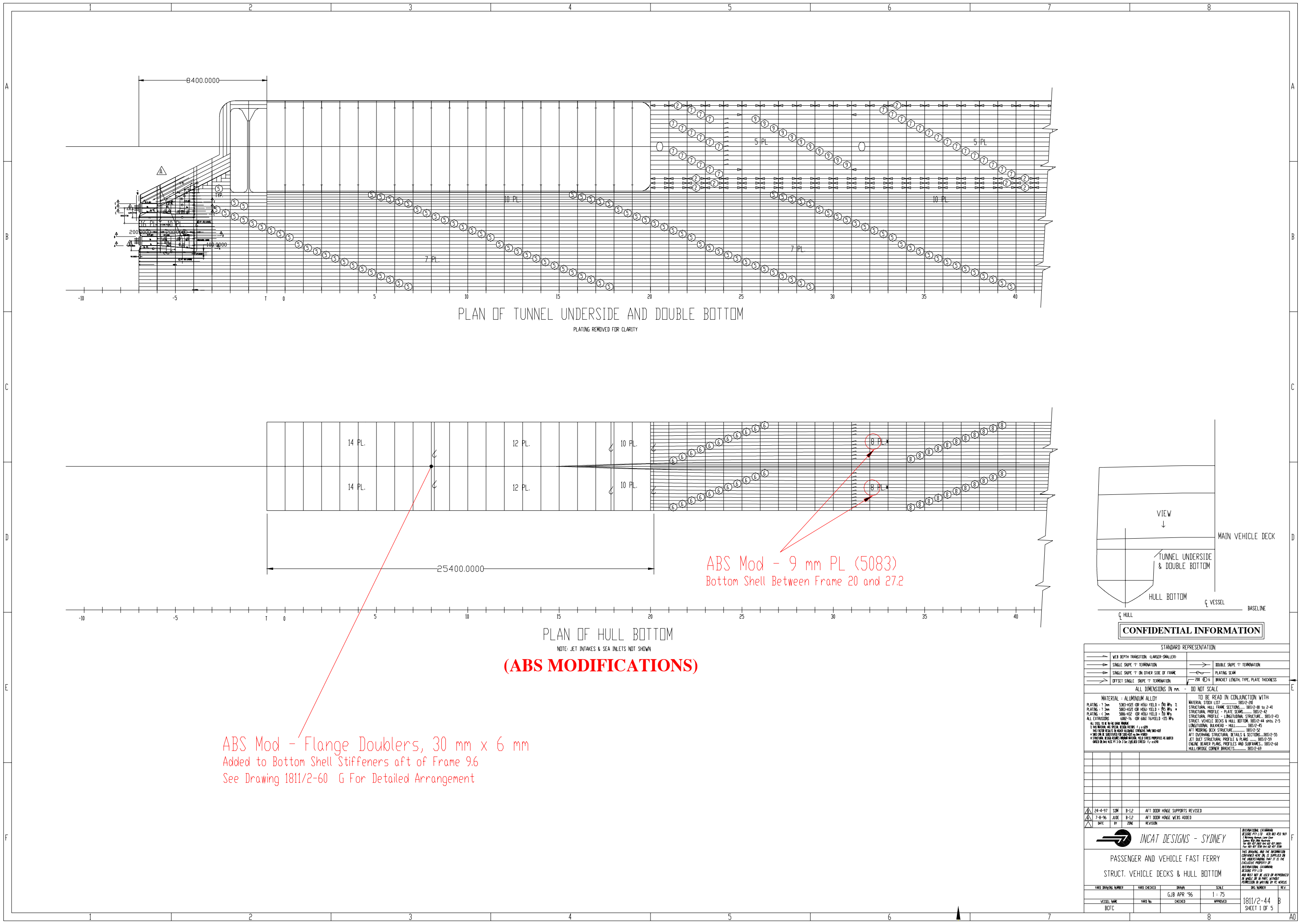


Figure 6-18 ABS Structural Modifications – Aft Portion of Bottom Hull

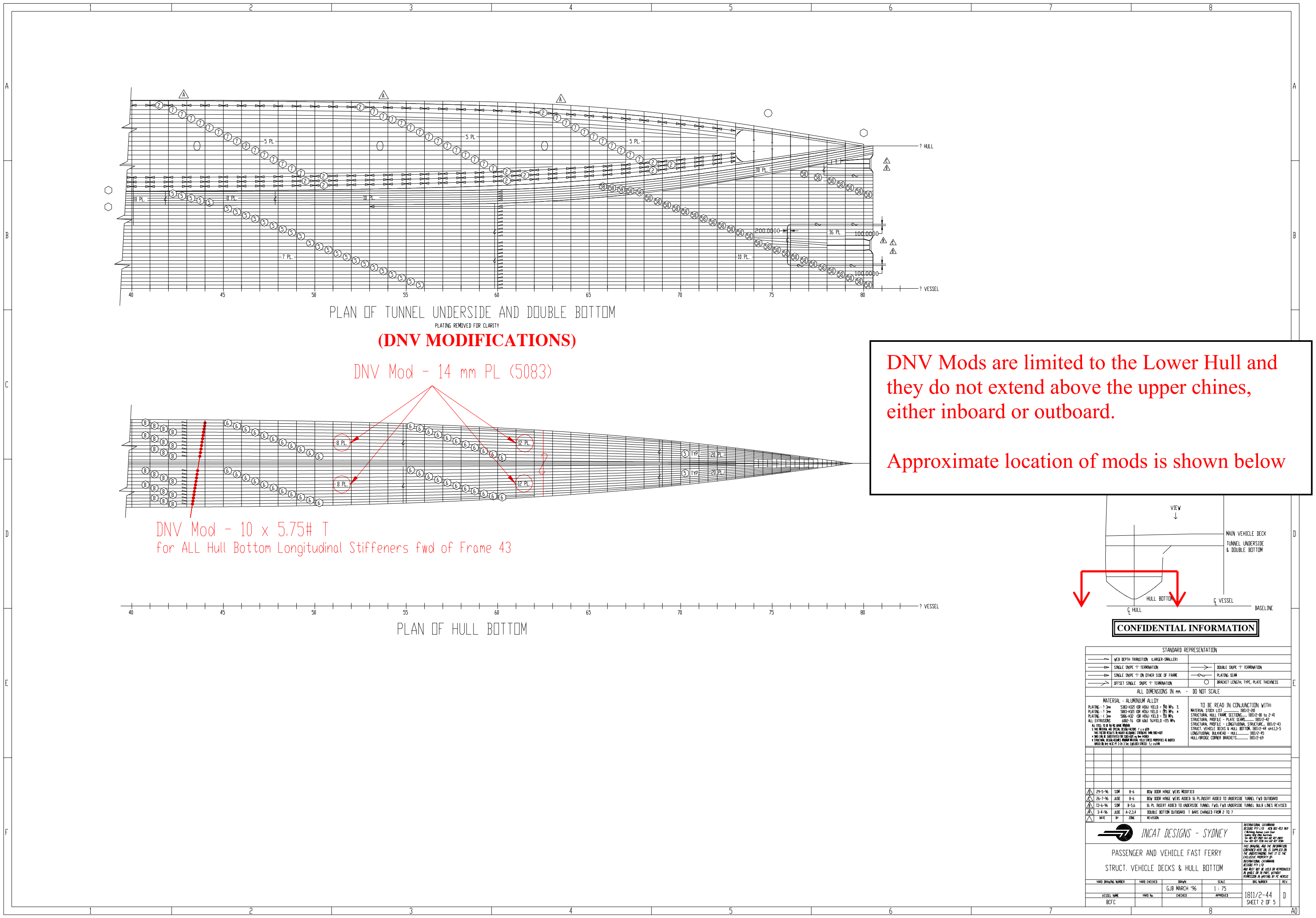


Figure 6-19. DNV Structural Modifications – Forward Portion of Bottom Hull

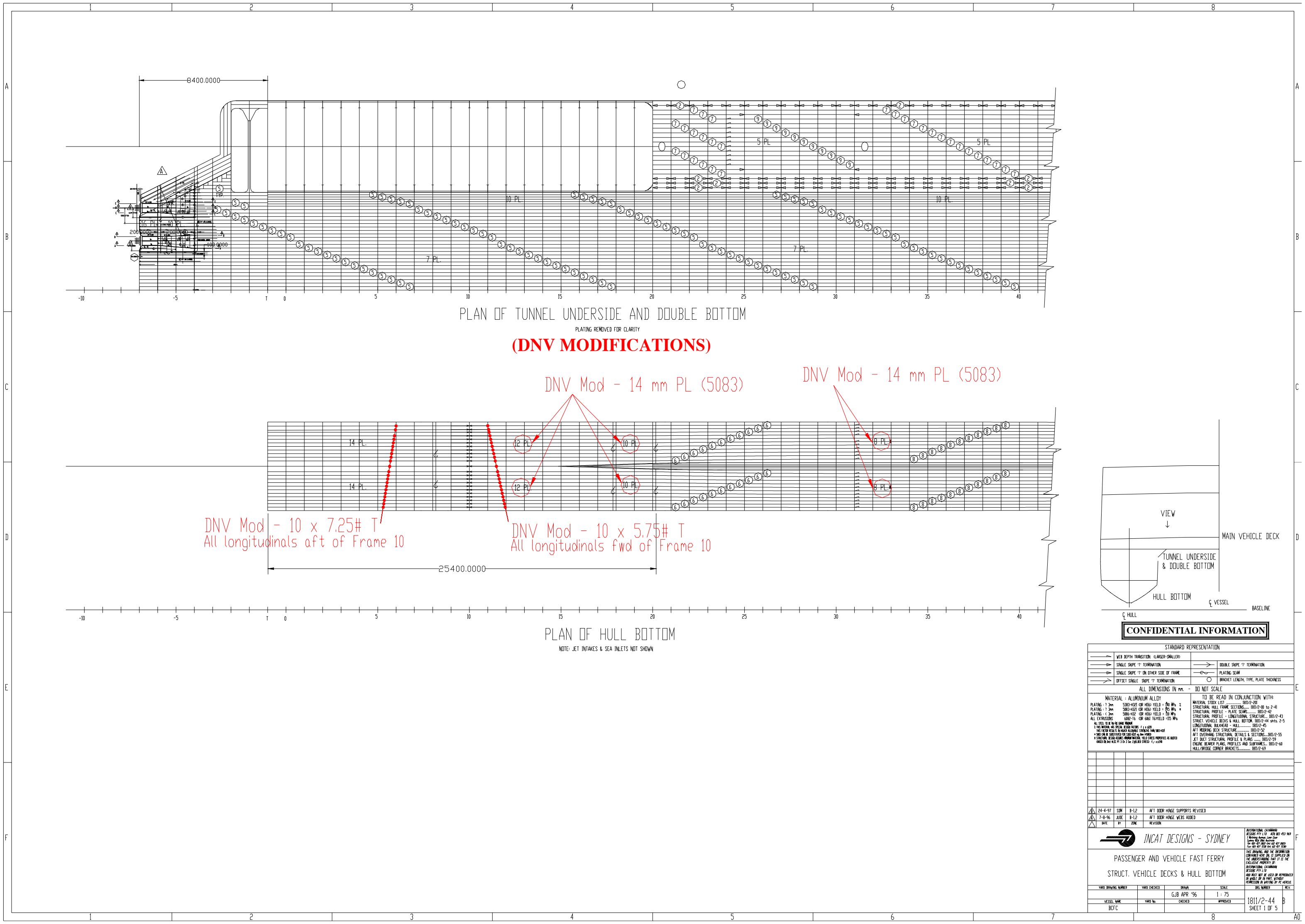


Figure 6-20. DNV Structural Modifications – Aft Portion of Bottom Hull

6.6 VEHICLE DECK TIE DOWNS

A primary concern that was soon dispelled involved the use of steel tiedowns in aluminum deck structure. Previous experience suggested that tiedown fittings were only available in steel and typical practice would discourage the use of steel fittings in an aluminum deck. It also did not seem likely that aluminum fittings for this function would be available due to the rugged nature of the fittings operation – not an operation thought to be well suited to a relatively malleable material like aluminum.

Inquiry to various fittings manufacturers, Peck & Hale and Pacific Marine and Industrial, quickly revealed that there is a common application for vehicle tiedown fittings in aluminum decks. Indeed, the fittings are steel and isolated from the aluminum deck structure with gaskets to prevent galvanic action between the dissimilar metals. A sealant is applied around the perimeter of the tiedown after it is bolted into place to provide a watertight boundary. This is typical practice and an application of a typical cloverleaf fitting on HSV-X1 is shown below in Figure 6-21, which also include one of the original tiedown tubes included on the HSV-X1. The tiedown tubes have very limited capacity, approximately 0.8 short tons and are not usually included in tiedown applications for military vehicles and will not be considered effective or included on the PacifiCat conversion.

Figure 6-21. Typical Cloverleaf Tiedown
Shown on HSV-X1 (With Original Tiedown Tube)

6.6.1 Arrangement of Vehicle Tiedowns

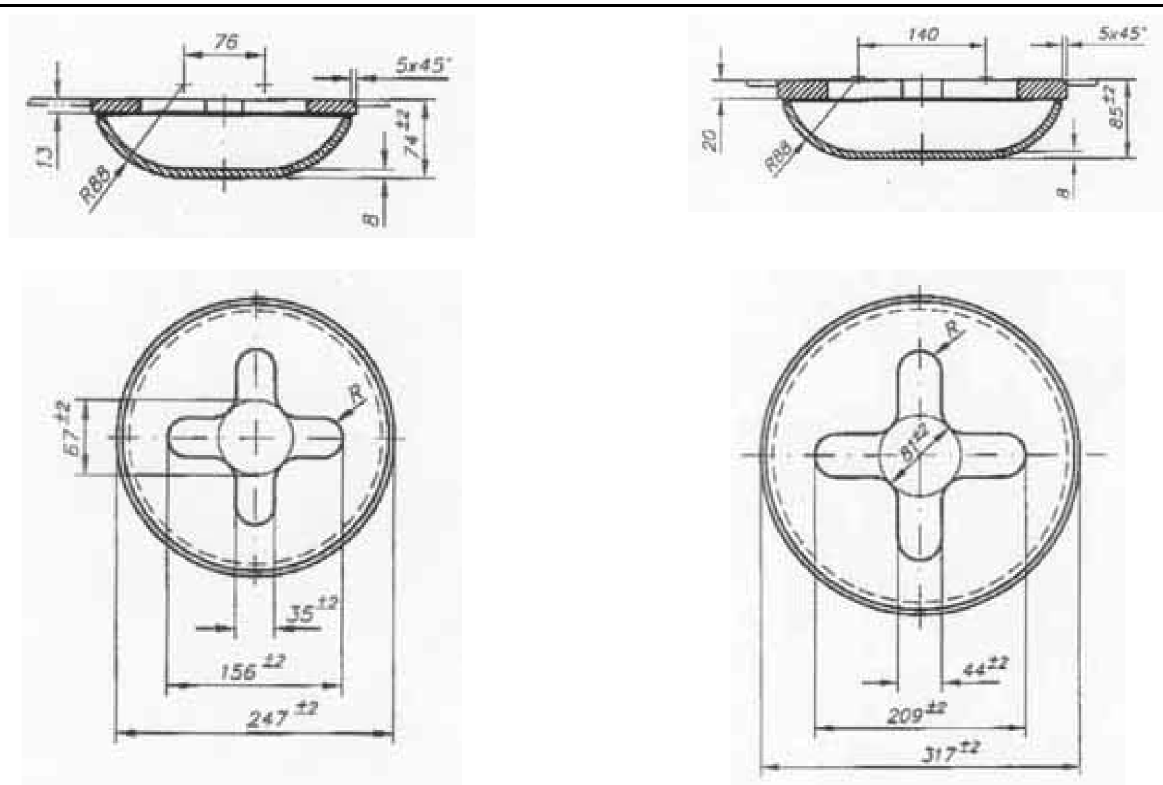
A typical vehicle deck on a military vessel includes a grid of tiedowns on 4-foot centers. There are two reasonable possibilities for vehicle deck tiedowns on the PacifiCat:

1. Provide a general grid of tiedowns on 4-foot centers similar to that provided on other vehicle decks.
2. Provide a tiedown arrangement that specifically reflects the vehicle arrangement shown in Figure 6-5.

Providing the specific tiedown arrangement to reflect Figure 6-5 has the advantage of minimizing the cost impact for converting the vessel. However, it may also become too restrictive for overall use of the vessel. Regardless, the requirement for the number of tiedown fittings to suit the conversion is fairly extensive and providing an arrangement for the conversion will still satisfy the academic nature of this project. Typical structural details are shown below, Figure 6-23, for the converted deck structure in way of the tiedowns.

The spacing of the transverse floors below the Main Vehicle Deck is 1.2 meters throughout the deck, which is approximately 47.2 inches, i.e., 4 feet, the same spacing typically used for a tiedown arrangement. This lends itself very nicely to arranging the tiedowns in a manner that is coordinated with the floors and consistent with typical tiedown arrangements.

Two typical tiedowns with capacities required for this project are shown in Figure 6-22. These tiedowns mount into the deck similar to most US Navy vehicle decks, i.e., they are flush mounted tiedowns. Protruding tiedowns will have a bolting flange that allows bolting to the vehicle deck structure. The typical detail proposed for the PacifiCat is shown in Figure 6-23.



Breaking Load: 160kN (36,000 lb)	Breaking Load: 315Kn (70,800 lb)

Figure 6-22. Typical Flush Cloverleaf Tiedown Fittings

The following information, Table 6-38, for the number and size/capacity of tie down fittings required for each vehicle is taken from Marine Lifting and Lashing Handbook, MTMCTEA REF 97-55-22, Table 4-1 [3] for “Other Ships”, referring to all ships other than FSS and large, medium speed RO/RO ships, which have reduced lashing requirements due to the large sizes of these ships and their relatively low motions in extreme seas.

Table 6-38. Vehicle Lashing Requirements, US Marines

Vehicle Weight (Lbs)	Lashing Strength (Lbs)	Total Number of Required Lashings
Up to 5260	5000	4
Up to 10,530	10,000	4
Up to 14,850	14,100	4
Up to 17,900	17,000	4
Up to 36,860	35,000	4
Up to 73,720	70,000	4
Up to 147,450	70,000	8

Based on the vehicle loadout proposed for this project and their respective weights, Table 6-39 summarizes the tiedowns that are nominally required for the military loadout based on vehicle weight.

As seen in Table 6-39, the conversion only includes one vehicle that exceeds 36,860 pounds and that is the 7-T M927, Extended Bed Truck, with a weight of 37,000 pounds. The loadout only includes one of these trucks.

Table 6-39. Tiedown Requirements for the Conversion

Vehicle Weight (Lbs)	Number of Vehicles in Weight Range	Total Number of Tiedowns Required at Given Strength
Up to 5260	30	120 @ 5000 lb
5260 to 10,530	31	124 @ 10,000 lb
10,530 to 14,850	1	4 @14,100 lb
14,850 to 17,900	2	8 @ 17,000 lb
17,900 to 36,860	22	88 @ 35,000 lb
36,860 to 73,720	1	4 @ 70,000 lb
73,720 to 147,450	0	0 @ 70,000
		Σ = 348 tiedowns

As shown in Table 6-39, a total of 348 tiedowns are required to satisfy the nominal requirements for this conversion study. Practical considerations prohibit using the entire array of tiedowns reflected in the table. All of the vehicles heavier than 17,900 pounds are arranged on the heavy extrusion, Figure 6-5. The TRLR, Mk 14 weighs 16,000 pounds and accounts for the two vehicles in the weight category of 14,850 to 17,900 pounds. As such, it is easy to select two tiedown sizes to accommodate the entire conversion effort. 70,000 pound tiedowns will be used for all the vehicles on the heavy extrusion and 17,000 pound tiedowns will be used for all other vehicles. This simplifies installation and increases flexibility for arranging vehicles within the deck.

The cost estimate for the task will be based on the following number of tiedowns:

- Number of 17,000 pound tiedowns = 1.25 x (120 + 124 + 4 + 8) = 320

- Number of 70,000 pound tiedowns = 1.25 x (88 + 4) = 115

The tiedowns are increased by 25% to allow for additional tiedown installations beyond the minimum for increased loading flexibility. It also provides a nominal margin on a portion of the conversion cost that is relatively inexpensive.

6.6.2 Installation of Vehicle Tiedowns

The vehicle tiedowns will be installed in the PacifiCat in a manner similar to the HSV-X1, i.e., they will be bolted into the deck structure, not welded into the deck plate and stiffening as is typical with many US Navy vehicle decks. As a result, they will be raised tiedowns, which protrude above the deck surface, not flush tiedowns typical of a deck fabricated specifically for transport of military vehicles. A typical raised tiedown is shown on the HSV-X1 in Figure 6-21. A sketch showing the basic installation envisioned for the PacifiCat is provided in Figure 6-23. The tiedowns will be centered on the transverse floors and the longitudinal stiffeners associated with the extrusions. This will allow each tiedown to be aligned with the deck stiffener upon which it is centered and not interfere with the stiffeners to either side, i.e., it will keep the bolts clear of the longitudinal stiffening. The floor and the stiffener provide good local support to help resist the bolt forces developed in the tiedown. The portion of the transverse floor that protrudes above the vehicle deck will be ground smooth with the surrounding deck so that there is a flush bolting surface. The grinding and installation process will be accomplished with smooth transitions in way of all tiedown installations to ensure there are no stress raisers causing early cracking problems or stress concentrations. The local strength and tightness of the extrusion in way of the area ground smooth will be inspected. If necessary, additional welding in way of the ground area will be performed to ensure that the required strength and tightness in way of this connection is maintained.

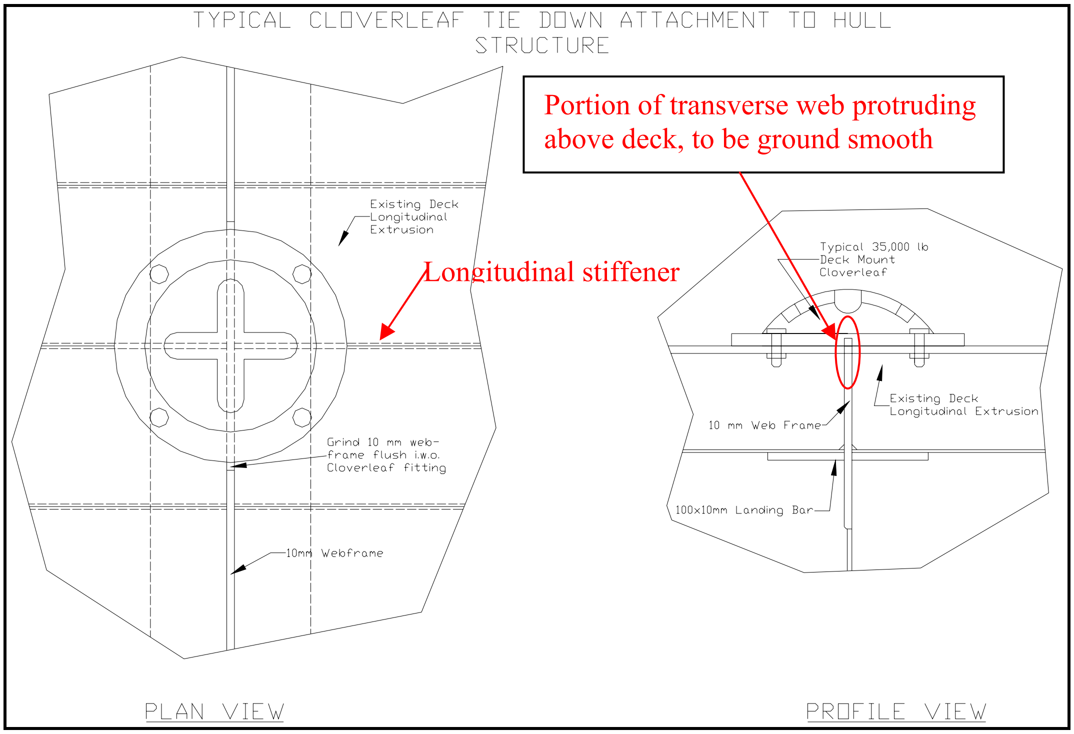


Figure 6-23. Detail of Vehicle Tiedown Installation Proposed for PacifiCat

The cost estimates for the conversion of these vessels will assume that they are bolted through the Main Vehicle Deck plate using four bolts capable of withstanding the tiedown loads, as calculated below.

6.6.3 Bolting Requirements for the Tiedowns

The bolts used to secure the tiedowns to the deck structure need to be able to withstand the full load exerted by the lashing. The force in the lashing will be resisted by a combination of shear and tension in the tiedown bolts. The deck plate also needs to be checked to ensure that the bolts will not cause pull-through failure as the tensile load they support bears against the deck plate.

6.6.3.1 Size of Tiedown Bolts

These calculations assume that all bolts are Grade 5 (ASTM A 325). For bolts up to ¾" these can sustain a proof load of 85,000 psi with a tensile strength of 120,000 psi. The allowable tensile stress in the bolt shall be taken as 60% of the proof load = 51,000 psi and the allowable shear stress shall be taken as 60% of the yield in shear = 0.6 x (85,000/√3) = 29,500 psi.

Neither ABS nor DNV has any criteria immediately available for bolt sizing. This report will use the procedures typical for US Navy bolt calculations for foundation design where the tensile and shear stresses are determined as:

$$F_t = \frac{0.5P}{A_t} + \sqrt{\left(\frac{0.5P}{A_t}\right)^2 + \left(\frac{V}{A_s}\right)^2}$$

and

$$F_v = \sqrt{\left(\frac{0.5P}{A_t}\right)^2 + \left(\frac{V}{A_s}\right)^2}$$

where: F_t = tensile stress in the bolt
F_v = shear stress in the bolt
P = tensile load
V = shear load
A_t = bolt tensile area at threads when checking tension only and at shank when combining with shear
A_v = bolt shear area through shank

The calculations performed for the bolts are simplified and assume that the load in the bolts is the result of a load in the lashing equal to the capacity of the tiedown acting at an angle of 45^0 to the deck.

With these assumptions, the 70,000 pound tiedowns can be supported on the heavy extrusion using four, 5/8" Grade 5 bolts. The bolts will develop axial and shear stresses of F_t = 38,051 psi and F_v = 24,363 psi, respectively. This study assumes that all tiedowns with a rating of less than 70,000 pounds will use four, 1/2" bolts, which when supporting a 17,000 pound tiedown, develop their maximum stresses of F_t = 29,211 psi and F_v = 18,630 psi. Using different bolt sizes for each different tiedown rating would complicate fabrication and increase the likelihood of installing the incorrect bolt/tiedown combination. This is justified because the cost of using the 1/2" bolts for all smaller installations will be negligible in comparison to the other costs associated with the conversion.

The total number of bolts purchased for the conversion is shown in Table 6-40.

Table 6-40. Grade 5 Bolts Required for Tiedowns

Total Number of Tiedowns Required at Given Strength	Number and Size of Grade 5 Bolts
120 @ 5000 lb	480 – 1/2"
124 @ 10,000 lb	496 – 1/2"
4 @14,100 lb	16 – 1/2"
8 @ 17,000 lb	32 – 1/2"
88 @ 35,000 lb	352 – 5/8"
4 @ 70,000 lb	16 – 5/8"
0 @ 70,000	0
Σ = 348 tiedowns	**Σ = 1024 – 1/2" bolts** **Σ = 384 – 5/8" bolts**

The cost estimate for the task will be based on the following number of bolts:

- Number of 1/2" bolts = 1.25 x (480 + 496 + 16 + 32) = 1280
- Number of 5/8" bolts = 1.25 x (352 + 16) = 460

The bolts are increased by 25% to allow for the same margins as the tiedown installations cited above.

6.6.3.2 Pull-Through Stresses on Vehicle Deck Plate

The pull-through stress is calculated using the following equation:

$$\sigma_{pull-through} = \frac{P}{1.5\pi dt}$$

where: d = nominal bolt diameter
t = deck plate thickness

The allowable shear stress for the deck plate shall be assumed to be 40% of the welded yield strength for ABS and taken as 90f_1 for DNV from DNV 3-3-5/C200. Neither regulatory body has specific criteria for this loading but such allowable stresses are consistent with typical practice for similar designs.

The welded yield strength of the 6061-T6 extrusion is 20,000 psi resulting in an ABS Allowable stress of 8000 psi.

For 6061-T6 DNV defines f_1 = 0.48 resulting in an allowable shear stress of 43.2 N/mm^2 = 6266 psi.

As shown in Figure 6-5, all of the vehicles using the 70,000 pound tiedown fittings have been located within the heavy extrusion of the Main Vehicle Deck. For the 70,000 pound tiedown with 5/8" bolts in the heavy extrusion the pull-through stress is 5442 psi and the pull-through stress of the 1/2" bolts supporting the 17,000 pound tiedown in the light extrusion is 4049 psi. Both stresses are acceptable with the criteria defined above.

No calculations are performed for the shear reaction of the bolts against the deck plate, what would typically be referred to as "Tearout Failure" of a bolt ripping through the flange of a foundation installation. These calculations are typically required to ensure that there is enough shear area in the bolting flange to resist the shear force developed in the bolt. In the current installation the shear force developed in the bolts will cause them to bear against the vehicle deck plating, which is regarded as having a large amount of deck plate effective in resisting this force.

6.6.4 Vehicle Lashing Requirements

The lashing requirements are only addressed to determine an approximate weight impact to account for this component of the loadout. Reference [3], USMC Lifting and Lashing Handbook, does not contain any information regarding the weight of the lashing hardware. To estimate the weights for this report the vehicles included in the loadout are grouped by weight in accordance with Table 4-1, from reference [3]. This grouping is presented below in Table 6-41.

The following unit weights are used to develop the lashing weight estimates provided below. Each lashing consists of:

- 1 Level/ratchet load binder @ 7.75 pounds
- 10 feet of chain @ 0.63lb/ft for 1/4” chain; 1.41 lb/ft for 3/8” chain & 2.50 lb/ft for 1/2” chain
- 2 hooks @ 0.85lb/1/4” hook; 1.38lb/3/8” hook & 3.00 lb/1/2” hook

As shown in Table 6-41, each vehicle requires four lashings. The lashing components described above correspond to the “Light”, “Medium” and “Heavy” lashings typically referred to in this application. The Heavy lashings are used for much larger/heavier vehicles than included in the current loadout, i.e., M1A1 tanks which weigh in at approximately 130,000 pounds per tank. These calculations assume that the two lightest categories in Table 6-41 are secured by “Light” lashings, i.e., 1/4” and the three heaviest categories by “Medium”, 3/8”, lashing components. As such the unit weight of each lashing is determined as:

- 1/4” lashing = 7.75 + 10(0.63) + 2(0.85) = 15.75 pounds per lashing
- 3/8” lashing = 7.75 + 10(1.41) + 2(1.38) = 24.61 pounds per lashing

Table 6-41 Weight of Vehicle Lashing Hardware

Vehicle Weight	Required Lashing Strength	Total Number of Required Lashings	Number of Vehicles in Loadout	Assumed Weight per Vehicle Lashing Set (Lb)	Total Lashing Weight (Lb)
Up to 8930 lb	5000 lb	4	57	15.75 x 4 = 63.0	3591.0
Up to 17,860 lb	10,000 lb	4	7	63.0	441.0
Up to 25,180 lb	14,100 lb	4	5	24.61 x 4 = 98.44	492.2
Up to 30,360	17,000 lb	4	8	98.44	787.5
Up to 62,510	35,000 lb	4	12	98.44	1181.3
					Σ 6493.0

The total lashing weight shown in Table 6-41 of 6493 pounds is equal to 3.25 short tons = 2.95 metric tonnes.

6.7 IMPACT ON RANGE, SPEED AND ENDURANCE FROM CONVERSION

Earlier work done with the PacifiCat was used to develop the curves for range and speed. Figure 6-24 shows some of the original predictions and test results for the speed – power performance of the ship. The original design has an installed power of 26,000 kW and a displacement of 1885 metric tonnes. This includes a deadweight tonnage of 518 metric tonnes with 66 metric tonnes of fuel oil.

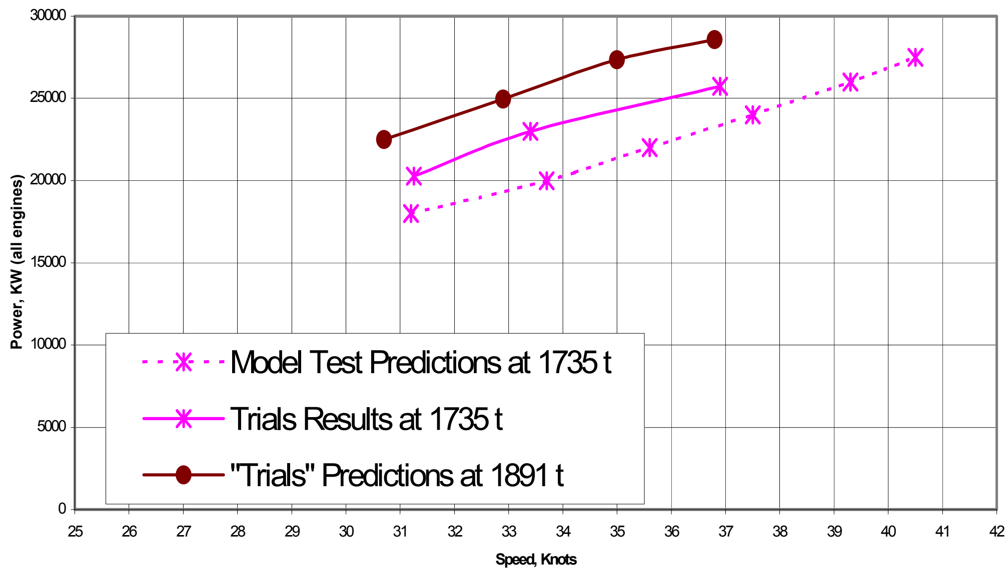


Figure 6-24. Estimate of PacifiCat Speed – Power

Figure 6-25 shows the estimates on range based on speed and fuel load developed for this task. It is expected that the fuel load could be increased by judicious removal of structure and outfitting that would no longer be required on the converted vessel. The original design included accommodations for 1000 crew and passenger. The converted vessel would have a nominal crew with zero passengers, although overnight berthing/accommodations and a full galley would also be required on the converted vessel. A complete study of this aspect of the conversion would have to be made to determine the exact additions and removals that could be incorporated into the converted vessel. The varying fuel loads shown in Figure 6-25 refers to the total weight of fuel carried on-board to achieve the speed/range profiles in the figure.

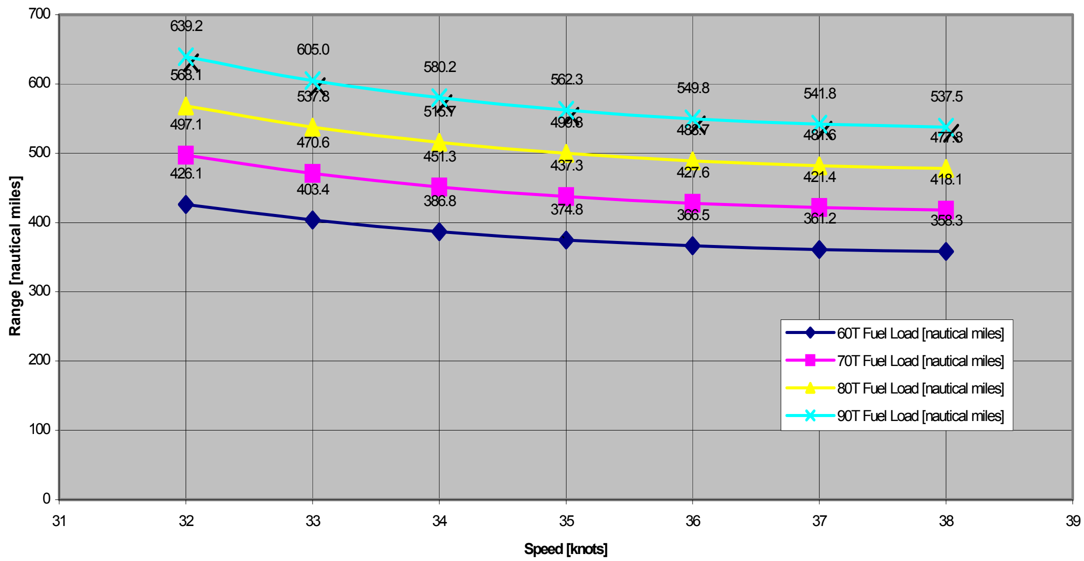


Figure 6-25. PacifiCat Range vs. Speed for Varying Fuel Loads

Figure 6-26 provides an estimate of the fuel loads required for a 4000 nautical mile transit at various speeds, i.e., it is necessary to carry nearly 500 metric tonnes of fuel to transit 4000 nautical miles at 37 knots. In accordance with the brochure information provided in Section 3.5.1 of this report, the baseline fuel load is 66 metric tonnes. The vessel is not configured to carry 500 tonnes of fuel and it would be necessary to add tankage, i.e., fuel trailers, to achieve this loadout, which would also reduce all other payload capacity to zero. Realistic use of the PacifiCat as a sealift vessel would probably be more beneficial in a theater support or sea-basing operation. The modification requirements and fuel load constraints on an **R4** are probably too great to allow for its efficient use in a trans-oceanic mode. For local, in-theater service, the PacifiCat would make for a good vessel.

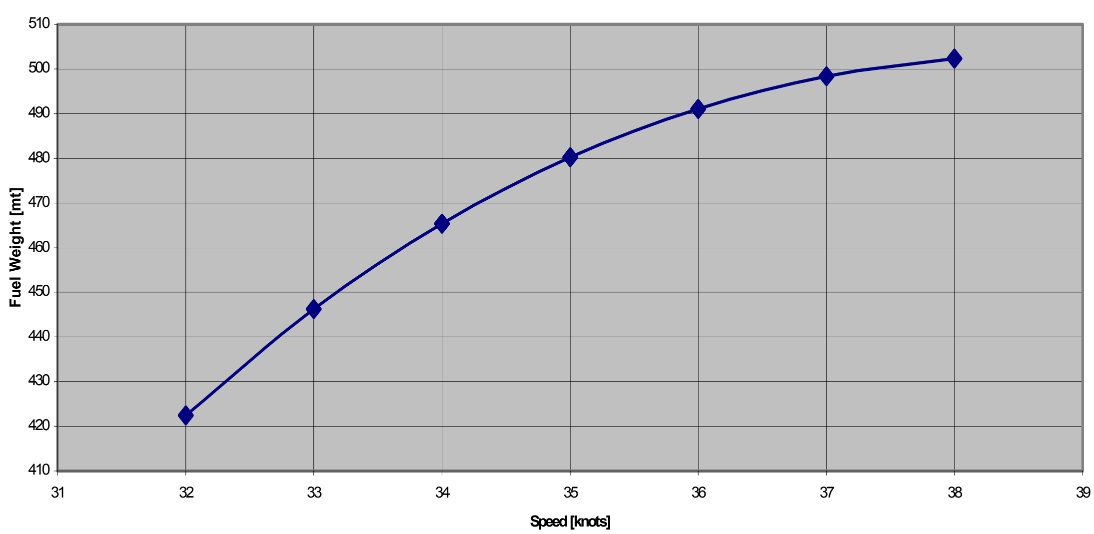


Figure 6-26. Fuel Load vs. Speed – 4000 Nautical Mile Transit

6.8 FATIGUE ANALYSIS

The Statement of Work for this project does not require any fatigue analysis. The following discussion is presented for general consideration.

The requirements for fatigue analysis would have to be specially developed for any conversion. While fatigue is important for any design, and for high speed aluminum craft, in particular, the conversion aspect requires augmenting the considerations that would typically be addressed for a new design to those that would be more specifically developed for the actual vessel being converted.

Considerations would include the age of the vessel being converted and its own fatigue history or fatigue related problems and the anticipated loading profile/histogram for the converted vessel. The loading histogram developed for any ship converted into the mission required for this project would consist of the following components:

1. Loading history prior to conversion, i.e., commercial operation,
2. Loading history subsequent to conversion, i.e., military operation,
3. Loading history subsequent to military mission, i.e., return to commercial operation.

The third portion of this loading history may be unknown at the time of conversion and it may be assumed that the balance of the service life of the vessel will be spent performing the military mission. This would probably lead to a conservative assessment of the loading histogram since it is expected that the loading spectra associated with the military mission would be more severe than those experienced during commercial operation.

The loading history prior to the conversion would not include any loads due to the open ocean environment. The loading history subsequent to the conversion would include open ocean loads and a realistic assessment of the total exposure time in the open ocean would have to be determined in order to assess the load magnitudes and cycles, i.e., loading spectra, for the global hull girder and secondary slam loads. It is not expected that the converted vessel would spend significant time in the open ocean, only be required to transit to get in theatre.

After assessing the loads required for the conversion by both ABS and DNV and also developing the local loads required for the vehicle deck, it can be seen that the loading histogram for the ship would be increased compared to any originally defined histogram however, the load cycles associated with this new mission of the vessel would also have to be considered. As discussed in Section 2 of this report, it will take between 20 and 30 sorties to complete the delivery of a single MEU. The load cycles associated with this would have to be determined along with the actual exposure time anticipated in the open-ocean environment to accurately determine the new loading profile for primary hull girder and slamming loads. These considerations would all have to be quantified as input to the load histogram.

Regarding ABS and DNV criteria for fatigue analyses in their respective high speed rules:

- The ABS HSNC rules do not contain any specific requirements for fatigue design. Development of any fatigue analysis would be as required for the conversion specification although the requirements for any fatigue analysis may be tempered, or more strongly reinforced, depending on the fatigue history/performance of details prior to the conversion.

- The DNV HSLC & NSC covers fatigue analysis through their Direct Calculation Methods in Part 3, Chapter 9. Section 6 addresses Fatigue Analysis and presents only brief discussion regarding the procedures to be followed for the analysis. This includes defining a minimum service life not less than 20 years and using the Palmgren-Miner linear cumulative damage assessment to define acceptance.

6.9 CONCLUSIONS REGARDING STRUCTURAL MODIFICATIONS

The structural modifications required for the PacifiCat's, originally designed for DNV **R4** Service Area Restrictions, are not as severe as originally anticipated. The relatively deep hull girder helps to limit the modifications necessary to satisfy global requirements and the relatively tight stiffener spacing yields a fairly robust structure for resisting the local slam loads as well as the tire loads on the vehicle deck extrusions.

7.0 STRUCTURAL MODIFICATION COST ESTIMATES

This section of the report presents the cost estimates for the structural conversions required to incorporate the ABS and DNV structural modifications defined above. Each of the estimates also includes the cost for outfitting the Main Vehicle Deck with the tiedown sockets and supporting structure identified above. There are no costs included for the lashing equipment that would be required to actually secure the vehicles to the fittings.

The cost estimate information provided below is copied from a spreadsheet used to assemble and simplify all information associated with the estimate.

7.1 CONCLUSION

As detailed in the cost estimates shown below, the cost for the ABS modifications are less than the DNV modifications. The total costs including Outfit and Yard Services are:

- ABS $1,136,531
- DNV $1,927,699

The breakdown of the cost estimates is provided below.

APPROACH

This worksheet presents a summary of the conversion cost estimate for an Aluminum Car Ferry to support the efforts for this task.

The estimate was developed using material take-offs as calculated in Section 6. Costs for the plate and shapes were based on normalized market costs for equivalent grade materials, as available from public resources. Labor hours were based on JJMA experience with marine aluminum-based conversion work; cross-checked where practicable with industry and cost analyst data for like activities and motions. Labor costs were based on commercially oriented, 2nd tier yards, and were set at $45/Hour without fee.

CALCULATIONS

Bottom Plate

Removal of the bottom plate was estimated using available algorithms for cutting of existing plate; removal of identified stiffening for replacement with more robust structure, preparation of the remaining stiffening to support welding, and installation of the new plate. It should be noted that the removal and installation of the bottom stiffening is documented in the following tables.

All plate is 5083 or equal per the technical report. This plate is estimated at a nominal $1.25/pound ($2.75/kilogram) based on July 2004 spot market rates for aluminum; and modified to reflect the additional costs for preparing this particular alloy. A wastage factor of 15%, unless noted otherwise, was applied to the weight of the plate to account for losses due to nesting. This factor is based on experience with nesting of similar simple shapes as provided by representative 2nd tier yards.

All shapes are also from 5083 or equal per the technical report. The costs for these shapes was estimated at 20% over the regular plate cost identified above. This estimating technique was used in place of spot market quotes for the identified shapes as costs for these shapes were not readily available. Other costs for shapes in this size and material range were available, and that information was used to calculate the cost growth associated with the shapes. The 20% is what resulted.

All welding was assumed to be continuous heliarc or similar welding. Materials including gases and welding materials are estimated and accounted for in this element.

To clarify the plate and stiffener removal process:

Under the heading **Remove bottom plate**

Perimeter cutting of plate – cut around the entire perimeter of the plate panel being removed.

Stiffener cutting – cut thru the welds that connect the stiffening to the plate so the plate comes off and the stiffening remains in place.

Under the heading **Remove Bottom Structure**

Stiffener cuts – cuts are now made where the transverse and longitudinal members need to be separated from the ship to complete the removal.

SUMMARY

The following tables summarize the two cost estimates, one for ABS, the second for the DNV. Cost calculations were based on the material take-offs specific to the rules being applied. Tie down costs are identical for both studies as the deck arrangement and quantities were the same. Outfitting items are based on material take-offs, predominately painting. Yard services were based on a predicted timeline, which was identical for both studies.

A 20% factor was applied to the direct work to account for on-site engineering and program management. This is representative of costs for similar programs.

Fee was applied at 10%, representative of a normal 2nd tier repair-yard fee structure.

	ABS	DNV
ACTION	**Cost, FY05**	**Cost, FY05**
Remove Plate	$ 8,219	$ 14,389
Remove Stiffening	$ 12,220	$ 30,850
Install Plate	$ 37,855	$ 120,805
Install Flange Doublers	$ 160,378	$ 160,683
Install Stiffening	$ 71,662	$ 311,182
Install Tie Downs	$ 146,369	$ 146,369
Outfit Services	$ 76,769	$ 328,564
Yard Services	$ 347,536	$ 347,536
Sub-Total	*$ 861,008*	*$1,460,378*
Engineering/Supervision @ 20%	$ 172,202	$ 292,076
Sub-Total	*$1,033,210*	*$1,752,454*
Fee @ 10%	$ 103,321	$ 175,245
TOTAL	**$1,136,531**	**$1,927,699**

ABS SLAM LOAD - OPEN OCEAN, UNRESTRICTED

Remove bottom plate	**6,265 kg**		
Perimeter Cutting -- equivalent of 2.25m-6.10m sheets each weighs an average of 525kg.		12.00	Equiv Sheets
Perimeter cutting takes an average of 15 cm/minute with a 2-person team		3.71	Hours/Sheet
Stiffener cutting -- equivalent of 9 stiffener runs per panel Each assumed to have intermittent welds [50%],		10.13	meters of weld
Cutting is equivalent to perimeter cutting		2.25	Hours/Sheet
Net Labor Effort		***72***	***Hours***

Item	**Units**		**Rate**	**Cost**
Labor Effort	72	Hrs	$ 45	$ 3,219
Gases	1	Lot	$ 5,000	$ 5,000
Net Cost, Bottom Panels Removal				***$ 8,219***

Remove Bottom Structure	**3,284 kg**		
* Assumes perimeter cuts only required to break structure free of the vessel. At 9 members per panel, and 2 edges per member, we see 18 cuts per panel.		18.00	Cuts/Panel
At the identified # of panels		12.00	Equiv Sheets
Total # of cuts		216.00	Cuts

Using available references, identify 30 minutes per cut
to prepare, execute, and take-down. Assumes a 2 person team

Net Labor Effort	***216***	***Hours***

Item	Units		Rate	Cost
Labor Effort	216	Hrs	$ 45	$ 9,720
Gases	0.5	Lot	$ 5,000	$ 2,500
Net Cost, Bottom Stiffeners Removal				***$ 12,220***

ABS SLAM LOAD - OPEN OCEAN, UNRESTRICTED

Add New Bottom Structure **7,048** kg

Plate preparation, nesting, cutting and edge preparation

* Assume 8 hours to prepare nesting, check CNC data, and position the plate
* Cut Plate, prepare edges -- 4 hours per plate/panel
* Weld plate in place; assumes backing structure was positioned prior
  Welding set at 6 meters/hour for a 2-person team ($45/hr-person * 2 people/6M-Hr).
* Assumes a 15% nesting factor.
* Stiffening welds-determined stiffener perimeter based on shape [AVG] 0.628
  Determined # of welds based on diagram as approximately 70
  Total linear meters of stiffening welds -- edges 43.96
  Total linear meters of stiffening welds -- runs 1360

	# of Units		Unit Cost	Total Cost
Plate Cost	8,105	kg	$ 2.75	$ 22,289
Plasma Cutting set-up Labor [$45/Hr]				
Assumes 12.00 panels	14	panels	$ 540.00	$ 7,560
Weld Plate In Place [2 man team; 6m/Hour]				
Perimeter Weld	200	meters	$ 15.00	$ 3,006
assumes continuous welds				
Gases and materials	1.00	lot	$ 5,000.00	$ 5,000
Net Material Cost, Bottom Panels				***$ 37,855***

Add New Flange Doublers, cut from pl **1,590** kg

Plate preparation, nesting, cutting and edge preparation

* Assume 8 hours to prepare nesting, check CNC data, and position the plate
* Cut Plate, prepare edges -- 4 hours per plate/panel
* Multi-head cutting tool cuts flange doubler strips- 8 hrs per panel

Field install all doublers - provide gasses and ventilation on ship

* Assumes a 15% nesting factor.
* Flange doubler welds based on 360 pcs @ 32mm and 2790 pcs @ 25 mm 3150

Each pc is 1.2m, intercostal between frames, continuous weld both edges 2.4

Total linear meters of flange doubler welds -- edges 7560

Two pcs per hour welded in place in ship includes clamping time (2 man team @ $45/hr-person)

	# of Units		Unit Cost	Total Cost
Plate Cost - to cut flange doubler strips	1,829	kg	$ 2.75	$ 5,028
Plasma Cutting set-up Labor [$45/Hr]				
Assumes 4.00 panels	4	panels	$ 900.00	$ 3,600
Weld flange doubler in place on ship [2 man team; 2pcs/Hour]				
Perimeter Weld	3,150	pieces	$ 45.00	$ 141,750
assumes continuous welds				
Gases and materials	2.00	lot	$ 5,000.00	$ 10,000
Net Material Cost, Bottom Panels				***$ 160,378***

Add New Bottom Stiffening **3,284** **kg**

Shape preparation, nesting, cutting and edge preparation

* Assume 2 hours to prepare nesting, check CNC data, and position the shape
* Cut Plate, prepare edges -- 1 hours per shape (15% nesting factor)
* Weld plate in place; assumes backing structure was positioned prior

Welding set at 6 meters/hour for a 2-person team.

	# of Units		Unit Cost	Total Cost
Shape Cost	3,777	kg	$ 3.30	$ 12,463

Plasma Cutting					
Assumes 108.00 Shapes	108	shapes	$ 135.00	$	14,580
Weld Plate In Place					
Line Weld	2,720	meters	$ 15.00	$	40,800
Edge Welds	88	meters	$ 15.00	$	1,319
assumes continuous welds					
Gases and materials	0.50	lot	$ 5,000.00	$	2,500
Net Material Cost, Bottom Stiffening				***$***	***71,662***

DNV SLAM LOAD - OPEN OCEAN, UNRESTRICTED

Remove bottom plate	**18,230**	**kg**		
Perimeter Cutting -- equivalent of 2.25m-6.10m sheets				
each weighs an average of 525kg.			35.00	Equiv Sheets
Perimeter cutting takes				
an average of 15 cm/minute with a 2-person team			3.71	Hours/Sheet
Stiffener cutting -- equivalent of 9 stiffener runs per panel				
Each assumed to have intermittent welds [50%],			10.13	meters of weld
Cutting is equivalent to perimeter cutting			2.25	Hours/Sheet
Net Labor Effort			***209***	***Hours***

Item	**Units**		**Rate**		**Cost**
Labor Effort	209	Hrs	$ 45	$	9,389
Gases	1	Lot	$ 5,000	$	5,000
Net Cost, Bottom Panels Removal				***$***	***14,389***

Remove Bottom Structure **16,096** **kg**

* Assumes perimeter cuts only required to break
structure free of the vessel. At 9 members per panel,

and 2 edges per member, we see 18 cuts per panel.	18	Cuts/Panel
At the identified # of panels	35	Equiv Sheets
Total # of cuts	630	Cuts

Using available references, identify 30 minutes per cut for the "baseline"
used on the ABS analysis; assumes a 2 person team.
Same assumption used for ABS

Times 630 cuts = 630 hrs

Net Labor Effort	***630***	***Hours***

Item	**Units**			**Rate**		**Cost**
Labor Effort	630	Hrs	$	45	$	28,350
Gases	0.5	Lot	$	5,000	$	2,500
Net Cost, Bottom Stiffeners Removal					***$***	***30,850***

DNV SLAM LOAD - OPEN OCEAN, UNRESTRICTED

Add New Bottom Structure **27,016** **kg**

Plate preparation, nesting, cutting and edge preparation

* Assume 8 hours to prepare nesting, check CNC data, and position the plate
* Cut Plate, prepare edges -- 4 hours per plate/panel
* Weld plate in place; assumes backing structure was positioned prior
  Welding set at 6 meters/hour for a 2-person team ($45/hr-person * 2 people/3M-Hr).
* Assumes a 15% nesting factor.

* Stiffening welds -- determined perimeter of stiffener based on shape	1.2
Determined # of welds based on diagram as approximately	200
Total linear meters of stiffening welds -- edges	240
Total linear meters of stiffening welds -- runs	4065.6

	# of Units		Unit Cost	Total Cost
Plate Cost	31,068	kg	$ 2.75	$ 85,438
Plasma Cutting set-up Labor [$45/Hr]				
Assumes 35.00 panels	40	panels	$ 540	$ 21,600
Weld Plate In Place [2 man team; 10m/Hour]				
Perimeter Weld	585	meters	$ 15	$ 8,768
assumes continuous welds				
Gases and materials	1.00	lot	$ 5,000	$ 5,000
Net Material Cost, Bottom Panels				***$ 120,805***

Add New Flange Doublers, cut from pl **1,544** kg

Plate preparation, nesting, cutting and edge preparation

* Assume 8 hours to prepare nesting, check CNC data, and position the plate
* Cut Plate, prepare edges -- 4 hours per plate/panel
* Multi-head cutting tool cuts flange doubler strips- 8 hrs per panel

  Field install all doublers - provide gasses and ventilation on ship
* Assumes a 15% nesting factor.
* Flange doubler welds based on 3160 pcs @ 25mm 3160

  Each pc is 1.2m, intercostal between frames, continuous weld both edges 2.4

  Total linear meters of flange doubler welds -- edges 7584

  Two pcs per hour welded in place in ship includes clamping time (2 man team @ $45/hr-person)

	# of Units		Unit Cost	Total Cost
Plate Cost - to cut flange doubler strips	1,776	kg	$ 2.75	$ 4,883
Plasma Cutting set-up Labor [$45/Hr]				
Assumes 4.00 panels	4	panels	$ 900.00	$ 3,600
Weld flange doubler in place on ship [2 man team; 2pcs/Hour]				
Perimeter Weld	3,160	pieces	$ 45.00	$ 142,200
assumes continuous welds				

Gases and materials	2.00	lot	$ 5,000.00	$ 10,000
Net Material Cost, Bottom Panels				***$ 160,683***

Add New Bottom Stiffening **36,097** **kg**

Shape preparation, nesting, cutting and edge preparation
* Assume 2 hours to prepare nesting, check CNC data, and position the shape
* Cut Plate, prepare edges -- 1 hours per shape (15% nesting factor)
* Weld plate in place; assumes backing structure was positioned prior
  Welding set at 6 meters/hour for a 2-person team.

	# of Units		**Unit Cost**	**Total Cost**
Shape Cost	41,512	kg	$ 3.30	$ 136,989
Plasma Cutting				
Assumes 315.00 Shapes	315	shapes	$ 135	$ 42,525
Weld Plate In Place				
Line Weld	8,131	meters	$ 15	$ 121,968
Edge Welds	480	meters	$ 15	$ 7,200
assumes continuous welds				
Gases and materials	0.50	lot	$ 5,000	$ 2,500
Net Material Cost, Bottom Stiffening				***$ 311,182***

DECK TIE-DOWN ADDITIONS

115 @ 70,000# capacity each
320 @ 17,000# capacity each
460 Grade 5 bolts, 5/8" Dia, 60mm or greater length
1300 Grade 5 bolts, 1/2" Dia, 60mm or greater length
435 Gaskets

	# of Units		Unit Cost	Total Cost
Lay-out Deck for tie-downs	40.00	Hours	$ 45.00	$ 1,800
Set-up, Drill 5/8" Holes 10mm Pl	460	Holes		
	15	Minutes/Hole		
	1	Person/Team		
	115	Hours	45	$ 5,175
Set-up, Drill 1/2" Holes 10mm Pl	1300	Holes		
	15	Minutes/Hole		
	1	Person/Team		
	325	Hours	45	$ 14,625

Set-up assumed to use cloverleaf as template
and drill guide; pilot hole then master hole.
Each hole is a one person operation; 15 minutes per hole

Install Cloverleafs	435	Cloverleafs		
	40	Minutes/Cloverleaf		
	2	Person/Team		
	580	Hours	45	$ 26,100

Assumed as a 2 person operation, one below decks to affix
washer and nut. Assumed normal torque and tools.
Assumed 30 minutes for set-up; bolt; check-out
An additional 10 minutes per cloverleaf, average, was assigned to perform
required 10mm webframe grinding and inspection prior to cloverleaf installation

Materials -- 70k# Cloverleaf	115	$	500.00	ea	$	57,500
Materials -- 17k# Cloverleaf	320	$	121.43	ea	$	38,857
Materials -- 5/8" Grade 5 Bolt	460	$	0.50	ea	$	230
Materials -- 1/2" Grade 5 Bolt	1,300	$	0.29	ea	$	377
Materials -- Gaskets	435	$	2.50	ea	$	1,088
Materials -- 5/8" Grade 5 Nut	460	$	0.21	ea	$	97
Materials -- 1/2" Grade 5 Nut	1,300	$	0.17	ea	$	221
Materials -- 5/8" Washers	460	$	0.28	ea	$	129
Materials -- 1/2" Washers	1,300	$	0.13	ea	$	170
Sub-Total, Materials					***$***	***98,669***
TOTAL, DECK TIE DOWN INSTALLATION					***$***	***146,369***

ABS SLAM LOAD - OPEN OCEAN, UNRESTRICTED

OUTFITTING SERVICES

* Bottom preparation and painting after installation of the new plating.
* Interior tank treatments
* Assumes no hull insulation in way of the tanks or deck mounted tie-downs is disturbed.
* Area of the stiffeners calculated from the shapes 855 M2
* Assume paint 50M2/Hour per team
* Paint cost of $25/sq m is based on military naval bottom paint based on ablative technologies. The cost per gallon is derived from HAYSTACK [subscription account for online access to DoD inventory and procurement costs] and averaged $500/gallon. Coverage is based on the conservative end of vendor data.

BOTTOM PAINT

	# of Units			Unit Cost		Total Cost
Bottom painting - 2 coats; 2 man spray team						
Mobile Lift	2	Days	$	250.00	$	500
Plate Paint, 2 coats, 20 sq m/Gall	300	sq m	$	25.00	$	7,500
	300	sq m	$	25.00	$	7,500
Labor	32	Hours	$	45.00	$	1,440
Sub-Total					***$***	***16,940***

TANK PREPARATION

	# of Units		Unit Cost	Total Cost
Painting, 2 coats epoxy or equal				
Plate Paint, 2 coats, 20 sq m/Gall	600	sq m	$ 25.00	$ 15,000
Stiffener Paint, 2 coats, 20 sq m/Gall	1,710	sq m	$ 25.00	$ 42,750
Labor	46	Hours	$ 45.00	$ 2,079
Sub-Total				***$ 59,829***
TOTAL, OUTFITTING SERVICES				***$ 76,769***

DNV SLAM LOAD - OPEN OCEAN, UNRESTRICTED

OUTFITTING SERVICES

* Bottom preparation and painting after installation of the new plating.
* Interior tank treatments
* Assumes no hull insulation in way of the tanks or deck mounted tie-downs is disturbed.
* Area of the stiffeners calculated from the shapes 4,879 M2
* Assume paint 50M2/Hour per team

BOTTOM PAINT

	# of Units		Unit Cost	Total Cost
Bottom painting - 2 coats; 2 man spray team				
Mobile Lift	2	Days	$ 250.00	$ 500
Plate Paint, 2 coats, 20 sq m/Gall	726	sq m	$ 25.00	$ 18,150
	726	sq m	$ 25.00	$ 18,150
Labor	32	Hours	$ 45.00	$ 1,440
Sub-Total				***$ 38,240***

TANK PREPARATION

	# of Units		Unit Cost	Total Cost
Painting, 2 coats epoxy or equal				
Plate Paint, 2 coats, 20M2/Gall	1,452	sq m	$ 25.00	$ 36,300
Stiffening Paint- 2 coats, 20M2/Gall	9,757	sq m	$ 25.00	$ 243,936

Labor	224	Hours	$	45.00	$	10,088
Sub-Total					***$***	***290,324***
TOTAL, OUTFITTING SERVICES					***$***	***328,564***

FACILITIES COSTS

This element of the cost estimate captures the costs of dry-docking the vessel, crane and rigging operations to lift out and install the stiffening and plating, firewatch and security functions, waste removal, and other services.

DRY-DOCKING

* Assumed a 3 week dry-dock period; 1 week to remove the plate, 2 weeks to install the new stiffening and plate, as well as bottom preparation and float-off.
* Assume interior work will be accomplished after float-off
* Assumes a rather large dry-dock given the beam of the subject vessel. A dock similar to the Alabama at Alabama Dry-Dock was used in this estimate.

Item	Units			Rate		Cost
Docking Calculation	80	Hrs	$	80	$	6,400
Dry-Docking	21	Days	$	10,000	$	210,000
Sub-Total					***$***	***216,400***

OUTFITTING PIER

* Assumed a 3 week post dry-dock period to refurbish interior compartments

Item	Units			Rate		Cost
Pier Side, Post Docking	21	Days	$	1,000	$	21,000
Sub-Total					***$***	***21,000***

SERVICES

* Electrical Power, Fluids services are assumed to be negligible as "normal" loads and services are contained within the daily rates.
* 2 man roving firewatch assumed for nite shift; day-shift firewatch is integral to the working teams.

* 2 man security detail for nite shift, dedicated to the ship/dry-dock
* Weekly removal of 1 commercial dumpster with typical waste, $600/week
* Material scrapping assumed to be within the yard and a net $0 cost.
* Also covers tank gas-free and inerting functions

Item	Units			Rate		Cost
Firewatch	504	Hrs	$	45	$	22,680
Security	504	Hrs	$	45	$	22,680
Waste Removal	5	Weeks	$	599	$	2,995
Tank Gas Free Service	12	tanks	$	500	$	6,000
Sub-Total					***$***	***54,355***

RIGGING OPERATIONS

* Assumes a mobile Lattice-Boom Crane of sufficient capacity to reach out with a 10LT Load. This would come with a driver, spotter and 2 riggers.
* Estimated as a commercial daily rental; assumes 3 weeks of use.
* Costs taken from MEANS Estimating handbooks, scaled to 2004/2005 rates

Item	Units			Rate		Cost
Crane	21	Days	$	1,216	$	25,541
Crew	21	Days	$	1,440	$	30,240
Sub-Total					***$***	***55,781***

TOTAL, FACILITIES COSTS	***$***	***347,536***

8.0 CONCLUSIONS ON REQUIRED ABS AND DNV STRUCTURAL MODIFICATIONS

The nominal hull girder and vehicle deck structure is adequate to handle the increased loading requirements for the unrestricted, open-ocean notation for both ABS and DNV. ABS HSNC requires only limited modifications to resist slam loads whereas DNV criteria require nearly 30 metric tonnes of additional structural weight to satisfy their secondary slamming criteria. The structural modifications to satisfy ABS is approximately 2 metric tonnes.

This report recognizes that both ABS and DNV typically require model tests to confirm final loads for the structural design of any high speed craft. It is certainly possible that such tests would produce design loads more demanding than those predicted by either set of design algorithms and that all conclusions developed through this report need to be tempered with this recognition.

Another factor in the performance of the PacifiCat hull girder structure is a relatively deep hull girder resulting from two vehicle decks. Many fast ferries only have one vehicle deck and the presence of the second vehicle deck results in a strength deck that is one deck higher than usual and an associated hull girder that is also that much deeper than usual. Naturally, this results in greater hull girder inertias and section modulii to resist longitudinal, transverse and torsional bending moments acting on the vessel.

Structural optimization is possible as was demonstrated through the FEA developed for this portion of the task. If an owner should consider increasing class to handle the open-ocean, unrestricted operation it is possible that any weight gain nominally included through application of the rule loads could be offset by optimizing the structure with more refined analysis and FEA.

9.0 REFERENCES

1. ABS Rules for Building and Classing High Speed Naval Craft, 2003.

2. DNV Rules for Classification of High Speed, Light Craft and Naval Surface Craft, January 2002.

3. Marine Lifting and Lashing Handbook, Second Edition, MTMCTEA REF 97-55-22, October 1996.

4. ABS Guide for Building and Classing High-Speed Craft, February 1997.

5. Medium Tactical Vehicle Replacement (MTVR) Report, John J. McMullen Associates, Inc., June 2000.

6. 1997 Formulas for Stress and Strain, Roark, Raymond J., Young, Warren C., McGraw-Hill Book Company, Fifth Edition, 1982.

7. Design of Welded Structures, Blodgett, Omar, W., The James F. Lincoln Arc Welding Foundation, 1966.

10.0 DRAWING REFERENCS

8. Midship Section – Ship #3 At Frames 28 and 43, Catamaran Ferries International, Drawing # S101 MIDSHIP, REV O.

9. Structural Profiles – Plate Seams Aft, INCAT Designs – Sydney, Drawing # 1811/2-42 Sheet 1 of 2, REV I.

10. Structural Profiles – Plate Seams Forward, INCAT Designs – Sydney, Drawing # 1811/2-42 Sheet 2 of 2, REV H.

11. Structural Profile – Longitudinal Structure, Aft End, INCAT Designs – Sydney, Drawing # 1811/2-43 Sheet 1 of 2, REV F.

12. Structural Profile – Longitudinal Structure, Forward End, INCAT Designs – Sydney, Drawing # 1811/2-43 Sheet 2 of 2, REV F.

13. Struct, Vehicle Decks & Hull Bottom, INCAT Designs – Sydney, Drawing # 1811/2-44 Sheet 1 of 5, REV B.

14. Struct, Vehicle Decks & Hull Bottom, INCAT Designs – Sydney, Drawing # 1811/2-44 Sheet 2 of 5, REV D.

15. Struct, Vehicle Decks & Hull Bottom, INCAT Designs – Sydney, Drawing # 1811/2-44 Sheet 3 of 5, REV H.

16. Struct, Vehicle Decks & Hull Bottom, INCAT Designs – Sydney, Drawing # 1811/2-44 Sheet 4 of 5, REV I.

17. Struct, Vehicle Decks & Hull Bottom, INCAT Designs – Sydney, Drawing # 1811/2-44 Sheet 5 of 5, REV E.

18. Jet Duct Structural Profile and Plan, INCAT Designs – Sydney, Drawing # 1811/2-59, REV C.

19. Material Stock List, INCAT Designs – Sydney, Drawing # 1811/2-201, REV D.

APPENDIX A.

COMPREHENSIVE LOAD LISTS

UNITED STATES MARINE CORPS MARINE EXPIDTIONARY UNIT ASSETS

TAMCN	NOMENCLATURE	QTY	LN	WD	HT	WT	TOTAL SQ. FT.
MEU COMMAND ELEMENT							
A0966	AN/MLQ-36 MOBILE ELECTRONIC	1	255	99	126	28,000	175
A1935	RADIO SET AN/MRC-138B	4	185	85	82	5,190	437
A1957	AN/MRC-145	4	185	85	93	5,190	437
C4154	INFLATABLE BOAT RIGID HULL	2	468	98	80	6,000	637
C4433	CONTAINER, QUADRUPLE	15	57	98	82	6,500	582
D0080	CHASSIS TRLR GEN PURPOSE, M353	1	187	96	48	2,720	125
D0085	CHASSIS, TRAILER 3/4 TON	2	147	85	35	1,340	174
D0850	TRAILER CARGO 3/4 TON M101	4	147	75	35	1,340	302
D1059	TRUCK, CARGO	1	316	98	116	28,950	215
D1158	TRUCK UTIL, HMMWV	6	180	85	72	5,200	638
D1180	TRUCK UTILITY	2	180	85	72	5,200	213
TOTAL SQFT							**3,935**

GROUND COMBAT ELEMENT							
A1935	RADIO SET AN/MRC-138B	4	185	85	82	5,190	437
A1937	RADIO SET, MRC145A	9	185	85	83	5,190	983
B0589	AMORED EARTHMOVER, ACE M-9	2	243	110	96	54,000	371
B2482	TRACTOR, ALL WHL DRV W/ ATACH	1	277	94	141	13,000	181
B2566	FORKLIFT	1	196	78	79	11,080	106
B2567	TRAM	1	308	105	132	35,465	225
C4433	CONTAINER, QUADRUPLE	84	57	98	82	6,500	3,259
D0085	CHASSIS, TRAILER, 3/4 TON	1	147	85	35	1,340	87
D0850	TRAILER CARGO 3/4 TON M101	13	147	74	35	1,340	982
D0860	TRAILER CARGO	10	167	83	53	2,670	963
D0880	TRAILER TANK WATER 400GL	1	161	90	77	2,530	101
D1059	TRK, CARGO	17	316	98	116	28,950	3,656
D1072	DUMP TRK	2	316	98	116	31,888	430
D1161	VEH, INTER TRANSPORTABLE	8	194	65	73	4,200	701
D1125	TRK, UTILITY	8	180	85	69	7,098	850
D1158	TRK, UTIL, HMMWV M1038	11	180	85	72	5,200	1,169
D1158	TRK, UTIL, HMMWV M998	39	180	85	72	5,200	4,144
D1159	TRK, UTIL, ARMT CARR HMMV	12	180	85	69	5,977	1,275
E0665	155MM HOWITZER	6	465	99	115	15,400	1,918
E0796	AAAV COMMAND (C-7)	1	360	144	126	72,500	360
E0846	AAAV PERSONNEL (P-7)	14	360	144	126	72,500	5,040
E0942	LAV ANTI TANK (AT0	2	251	99	123	24,850	345
E0946	LAV C2	1	254	99	105	26,180	175
E0947	LAV ASSULT 25MM	11	252	99	106	24,040	1,906

TAMCN	NOMENCLATURE	QTY	LN	WD	HT	WT	TOTAL SQ. FT.
E0948	LAV LOGISTICS (L)	1	255	98	109	28,200	174
E0949	LAV, MORTAR CAR	2	255	99	95	23,300	351
E0950	LAV, MAINT RECOV	1	291	99	112	28,400	200
E0996	BLADE MINE CLEARING	1	179	115	29	9,000	143
E1888	TANK, M1A1	4	387	144	114	128,600	1,548
TOTAL SQ FT							**32,080**

COMBAT SERVICE SUPPORT EQUIPMENT							
A1935	RADIO SET AN/MRC-138B	2	185	85	82	5,190	218
A1957	AN/MRC-145	2	185	85	82	5,190	328
B0215	BUCKET, GEN PURP, 2 1/2 YD	2	108	50	50	2,910	75
B0395	COMPRESSOR AIR 250CFM	3	214	97	83	8,390	433
B0579	LOAD BANK, DA543/G	1	187	96	67	4,800	125
B0591	EXCAVATOR HYD WHL 1085C	1	365	97	154	40,960	246
B0635	FLOODLT SET SKID MTD SM 4A30	4	134	43	66	2,000	160
B1220	KIT ASSAULT TRACKWY (MOMAT)	12	96	60	60	4,000	480
B2561	TRUCK, FORKLIFT	2	315	102	101	25,600	446
B2566	FORKLIFT 4000 LBS	2	196	78	79	11,080	212
B2567	TRAM	4	308	105	132	35,465	898
B2685	WELD MACHINE, ARC, TRL-MTD	1	186	96	88	8,100	124
C4433	CONTAINER, QUADRUPLE	50	57	98	82	6,500	1,940
D0080	CHASSIS TRLR GEN PURPOSE, M353	6	187	96	48	2,720	748
D0085	CHASSIS, TRAILER, 3/4 TON	6	147	85	35	1,340	521
D0209	POWER UNIT, FRONT (LVS) MK-48	13	239	96	102	25,300	2,072
D0850	TRAILER CARGO 3/4 TON M101	2	147	74	35	1,340	151
D0860	TRAILER CARGO 1-1/2 TON M-105	6	167	83	53	2,670	578
D0860	TRAILER CARGO 1-1/2 TON M-105	1	167	83	53	2,670	96
D0876	TRLR, POWERED, MK-14	7	239	96	50	16,000	1,115
D0877	TRLR, POWERED, MK-15	1	240	96	96	26,000	160
D0879	TRLR, POWERED, MK-17	2	240	96	96	22,650	320
D0880	TRAILER TANK WATER 400 GL	4	161	90	77	2,530	403
D1001	TRUCK AMB 4 LIT ARMD 1 1/4 TON	4	180	85	73	4,960	425
D1059	TRUCK, CARGO	25	316	98	116	28,950	5,376
D1061	TRUCK M- 927 5 TON EXT BED	4	404	98	116	31,500	1,100
D1072	DUMP TRUCK	1	316	98	116	31,888	215
D1158	TRUCK UTIL HMMWV	5	180	85	72	5,200	531
D1159	TRUCK UTIL, HMMWV M1044	3	180	85	69	7,098	319
D1159	TRUCK UTIL, HMMWV M1043	2	180	85	69	7,098	213
D1193	TRUCK, 5-TON, M934A1	1	346	98	114	38,466	236
D1212	TRUCK WRECKER 5 TON 6X6	2	346	98	114	38,466	471
E1377	REC VEH, FULL TRACK M88A1	1	339	144	117	139,600	339

TAMCN	NOMENCLATURE	QTY	LN	WD	HT	WT	TOTAL SQ. FT.
E1712	SHOP SET, MOBILE ARTY	1	240	96	96	8,000	160
E1713	OPT MAINT SHELTER 20 FT	1	240	98	98	10,000	163
TOTAL SQ FT							**21,397**

AVIATION COMBAT ELEMENT

A1935	RADIO SET AN/MRC-138B	2	185	85	82	5,190	218
A1957	AN/MRC-145	1	185	85	82	5,190	109
C4433	CONTAINER, QUADRUPLE	19	57	98	82	6,500	738
D0085	CHASSIS, TRAILER, 3/4 TON	2	147	85	35	1,340	174
D1158	TRUCK UTIL, HMMWV M1038	4	180	85	72	5,200	425
E1836	AVENGER VEHICLE (CLAWS)	3	185	109	98	9,000	420
D1061	MTVR 7 TON EXT BED	3	404	98	116	31,500	825
TOTAL SQ FT							**2,909**

MAGTF TOTALS

	MAGTF ELEMENT	TOTAL
	COMMAND ELEMENT	3,935
	GROUND COMBAT ELEMENT	32,080
	COMBAT SERVICE SUPPORT ELEMENT	21,397
	AVIATION COMBAT ELEMENT	2,909
	TOTAL SQ FT	**60,321**

UNITED STATES ARMY STRYKER BRIGADE COMBAT TEAM

Model	Quantity Veh	Veh	Length (IN)	Width	Height	Weight	SqFt	STons
1650-EH-MS	3		72	10	72	110	15	0
OE-239/GSQ	4		42	40	40	45	47	0
AN/TPQ-37V	1		90	78	72	2196	49	1
AN/TPQ-37V	1		196	96	92	10855	131	5
AN/TPQ-37V	1		181	96	64	7866	121	4
M3 (SUMMA)	2		234	92	51	11040	299	11
M3 (SUMMA)	19		234	92	85	18400	2841	175
MED GIRDER	1		205	93	67	6306	132	3
MED GIRDER	5		179	83	53	4756	516	12
MED GIRDER	1		161	87	65	4356	97	2
MED GIRDER	1		125	63	69	6212	55	3
MED GIRDER	1		124	68	43	5841	59	3
MED GIRDER	1		134	87	43	2273	81	1
MED GIRDER	1		156	74	70	6720	80	3
100-FT LG	1		186	82	82	7200	106	4
100-FT LG	1		186	82	83	7400	106	4
100-FT LG	1		176	82	86	6500	100	3
100-FT LG	1		176	82	82	6400	100	3
100-FT LG	1		176	82	80	6400	100	3
100-FT LG	5		176	82	77	9000	501	23
100-FT LG	1		186	82	72	7200	106	4
100-FT LG	1		176	82	89	7200	100	4
100-FT LG	1		176	82	83	7600	100	4
100-FT LG	1		178	89	68	6000	110	3
100-FT LG	1		178	82	86	6300	101	3
100-FT LG	1		178	82	90	7100	101	4
100-FT LG	1		178	82	40	6000	101	3
100-FT LG	1		186	83	62	7000	107	4
SET AN/TSM-153	1		147	87	84	2501	89	1
NONE	6		239	96	96	8000	956	24
NONE	13		96	96	46	3020	832	20
NONE	13		44	43	39	1755	171	11
M139	3		94	32	15	500	63	1
M139	3		94	32	15	425	63	1
WTR 500 GA	36		78	36	25	400	702	7
AN/TSM-210	1		123	89	72	3782	76	2
1/2 T M200A1	9	R	169	94	41	2445	993	11
C-20X-8016	3		65	25	40	610	34	1
40 AM NONE	3		60	36	36	400	45	1
60 AM NONE	10		60	36	36	400	150	2
AN/GYK-37V	13		147	87	84	5893	1155	38

Model	Quantity Veh	Veh	Length (IN)	Width	Height	Weight	SqFt	STons
AN/GYK-37V	13		147	87	84	3911	1155	25
NONE	1		84	48	48	700	28	0
MEP-802A	25		50	32	36	825	278	10
MEP-806B	1		87	36	59	3992	22	2
MEP-804A	1		70	36	54	2500	18	1
MEP-804A	2	R	139	87	85	4080	168	4
PU-806B/G	1	R	165	95	87	6813	109	3
MEP-816B	1		87	36	59	4042	22	2
PU-803B/G	4	R	165	95	84	5700	435	11
PU-751/M	1	R	144	74	79	2840	74	1
PU-753/M	1	R	144	74	78	2525	74	1
PU-798	1	R	147	84	76	2570	86	1
PU-798A	15	R	135	86	67	2480	81	19
PU-798	2	R	147	84	78	2830	172	3
PU-799A	1	R	135	86	67	2510	81	1
PU-799	1	R	147	85	77	3040	87	2
PU-802	17	R	165	95	84	4920	1851	42
500 GAL CA	42		74	35	18	233	755	5
MEP-803A	19		62	32	37	1250	262	12
MEP-813A	1		62	32	36	1280	14	1
NONE	13		65	40	58	355	235	2
AN/ASM-146	10		147	87	84	3664	888	18
AN/ASM-147	4		147	87	84	3962	355	8
AN/GRC-193 RADIO	2		27	20	40	130	8	0
NONE	14		81	39	35	840	307	6
148002-1	1		73	30	37	1720	15	1
AN/TSC-116	3		83	82	39	950	142	1
MK-2727/G	15		96	53	108	201	530	2
TYPE-2	20		48	40	48	330	267	3
TYPE-2	20		48	40	48	422	267	4
M198	12	R	496	111	117	15750	4588	95
MKT-95	1	R	201	152	132	5260	212	3
NONE	6	R	187	96	71	6109	748	18
M121	36	R	95	60	45	720	1425	13
AN/MJQ-35A	2	R	135	86	66	3140	161	3
M59	22		27	24	42	253	99	3
M93A1 FOX	3	R	288	118	105	38500	708	58
CE-11	270		6	24	36	32	270	4
NONE	2		41	31	55	384	18	0
NONE	14		50	36	36	530	175	4
LP/PD1-90	22		102	84	71	3930	1309	43
S250	12		87	79	70	770	573	5
TY3	8		102	89	67	680	504	3

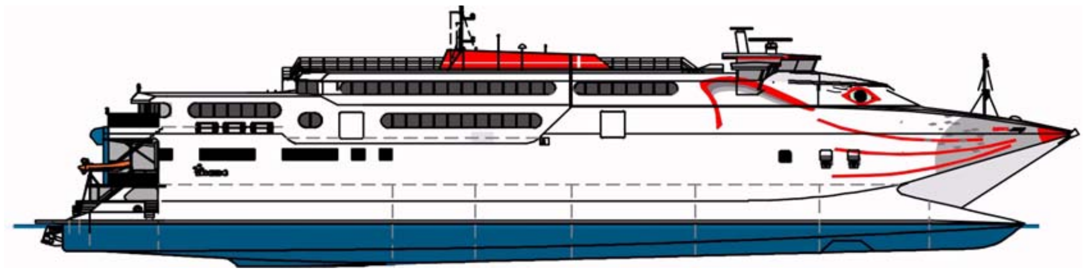
Model	Quantity Veh	Veh	Length (IN)	Width	Height	Weight	SqFt	STons
TY3	1		140	84	70	3249	82	2
NONE	17	R	191	89	99	9200	2007	78
NONE	11		49	27	38	41	101	0
NONE	11		56	26	38	57	111	0
NONE	1		73	20	34	150	10	0
MST F/VOLC	2		20	17	43	19	5	0
M1097	1	R	191	86	72	5600	114	3
M1097	3	R	197	86	102	9220	353	14
M1097	3	R	197	86	102	9165	353	14
M1097	1	R	199	89	82	7980	123	4
M1097	1	R	216	86	102	8620	129	4
M1097	1	R	202	94	102	9236	132	5
M1097	1	R	207	84	102	8326	121	4
M1097A2	76	R	191	91	72	5900	9173	224
M1097A2	1	R	191	86	56	5900	114	3
M1097A2	1	R	191	86	102	10214	114	5
NONE	2		101	79	70	2740	111	3
SET N	2		73	26	36	225	26	0
SET N	2		101	12	8	183	17	0
FIELD MAIN	2		239	96	96	18687	319	19
FM SUPPL N	2		148	88	84	9219	181	9
ADC 1000	1	R	151	79	76	2860	83	1
NONE	1		239	96	96	8490	159	4
FLU-419	6	R	250	96	102	15920	1000	48
NONE	2		55	25	37	268	19	0
NONE	2		25	21	36	158	7	0
M997A2	27	R	205	86	102	7660	3306	103
M1083A1 WW	26	R	274	96	112	23700	4749	308
M1084A1 WO	15	R	306	96	112	25373	3060	190
AN/TSM-174	1		67	55	40	485	26	0
LME	13		96	49	35	990	425	6
M978 WWN	4	R	401	96	112	38165	1069	76
NONE	2		27	42	50	70	16	0
M998	1	R	187	84	53	5280	109	3
M998	1	R	187	84	53	5280	109	3
M998A1	294	R	180	86	72	5380	31605	791
M1038A1 WW	2	R	186	86	72	5507	222	6
M1113 WWN	26	R	191	98	72	6380	3380	83
M1113 WWN	2	R	220	89	102	10066	272	10
M1085A1 WO	2	R	349	96	112	23973	465	24

Model	Quantity Veh	Veh	Length (IN)	Width	Height	Weight	SqFt	STons
M1083A1 WO	120	R	274	96	112	22723	21920	1363
M984A1 WWN	17	R	402	102	112	51300	4841	436
(ATLA 10000 LBS	8	NR	349	101	107	33120	1958	132
HI SP DUECE	6	NR	230	107	109	35750	1025	107
AN/TSC-154	3		111	85	50	2530	197	4
M978 WOWN	10	R	401	96	112	38165	2673	191
M1977 WOWN	4	R	395	97	113	37240	1064	74
M1025A2	8	R	191	86	74	6780	913	27
M1114	7	R	197	86	72	9800	824	34
M1076	58	R	324	96	124	16530	12528	479
M1095	51	R	230	96	82	9202	7820	235
M1102	55	R	136	86	100	1400	4467	39
M1102	1	R	136	86	86	4114	81	2
M1102	1	R	136	86	86	3652	81	2
M1101	191	R	136	86	100	1400	15513	134
XM1120 WOW	56	R	401	96	129	35300	14971	988
M1082	1	R	210	96	79	6860	140	3
COMMON NO	1		167	87	84	4460	101	2
NONE	1	R	179	96	97	7355	119	4
NONE	1		76	33	8	157	17	0
NONE	1		75	28	41	515	15	0
NONE	1		76	28	41	530	15	0
M105A2	1	R	166	83	82	6280	96	3
M1112	67	R	163	98	84	3945	7432	132
2-1/2-TON	1	R	281	100	128	19480	195	10
M116A2	1	R	147	85	77	2940	87	1
M1102	3	R	170	85	86	3710	301	6
ATG	1	R	161	85	87	3149	95	2
AVT	1	R	217	85	105	11431	128	6
GCS1	1	R	208	85	105	11310	123	6
LRT	1	R	166	87	70	4188	100	2
TC1	1	R	208	85	72	9775	123	5
GCS2	1	R	208	85	105	9360	123	5
ET	1	R	131	85	96	3842	77	2
MMF	1	R	218	85	105	10562	129	5
MMFT	1	R	131	85	96	3831	77	2
TC2	1	R	208	85	72	9775	123	5
NONE	3		63	2	63	3	3	0
MECC	2		240	94	96	4000	313	4
NONE	5		240	96	96	15000	800	38
ESV	9	R	283	110	106	42000	1946	189
CV	27	R	283	110	106	42000	5837	567
FRS-L	12		240	96	96	24000	1920	144

Model	Quantity Veh	Veh	Length (IN)	Width	Height	Weight	SqFt	STons
STRYKER	108	R	288	113	106	42000	24408	2268
MGS	27	R	301	105	106	41300	5926	558
MC	36	R	277	117	104	42000	8102	756
AN/GRC-229	10		45	2	45	5	6	0
FSV	13	R	283	111	106	42000	2836	273
MEV	16	R	277	117	94	42000	3601	336
AN/TSQ158-	3		206	87	90	6235	373	9
RCV	48	R	283	113	106	42000	10660	1008
ATGM	9	R	283	111	104	42000	1963	189
Supply						42475	264	21
----------						10578	28	5
TOE *						1375544	8085	688
						2969617	272607	14403
						148481	13630.4	720.15

APPENDIX B.

DETAILS OF CANDIDATE VESSELS



INCAT 91m – Spec Sheet provided by INCAT website as
"63250_2091mwpc-ropax-onepage-001.doc"

General Particulars

Builder	Incat Tasmania Pty Ltd.
Class Society	Det Norske Veritas
Certification	DNV +1A1 HSLC R1 Car Ferry "B" EO
Length overall	91.30m
Length waterline	81.33m
Beam overall	26.00m
Beam of Hulls	4.50m
Draft	3.73m approx. in salt water
Service Speed	42 knots
Lightship Speed	48 knots

*Note - All speeds quoted at 100% MCR (4 x 7080kw @ 1030 rpm) and excluding T-foil.

Capacities

Max Deadweight 510 tonnes (maximum available dependent on building specification).
Total persons up to 900 persons
Vehicle Capacity 220 cars at 4.5m length x 2.3m wide or combination of cars and up to 4 buses.
Axle loads - Transom to Frame 14 - 9 tonnes per dual wheel axle, Frame 14 to 35 - 2 tonnes per axle group and fwd of Frame 35 Ramps A to D - 0.8 tonnes per axle group.
Fuel Capacity - 4 x 14m3 integral aluminium tank and additional long-range tank of minimum 170m3 capacity provided in each hull.
Fresh Water 1 x 5.0 m3 GRP tank.
Sewage 1 x 5.0 m3 GRP tank.

Construction

Design - Two slender, aluminum hulls connected by a bridging section with center bow structure at fwd end. Each hull is divided into eight vented, watertight compartments divided by transverse bulkheads. One compartment in each hull prepared as short-range fuel tanks and one as a long-range fuel tank.

Welded and bonded aluminium construction using longitudinal stiffeners supported by transverse web frames and bulkheads. Aluminum plate grade 5383 H321 or H116. Aluminium extrusions grade 6082 T6 and 5083 H112.

Passenger Accommodation

Superstructure - Passenger accommodation supported above the vehicle deck on anti-vibration mounts.

Outfit – All materials comply with IMO standards for fire, low flame spread, smoke and toxicity.

Public Address - Builder's standard, marine, public address system supplied and fitted to cover all passenger and crew areas, vehicle decks, stairwells and ante rooms. Colour tvs fitted throughout the passenger cabin to enable seated view of safety messages and video.

Life Saving and Evacuation

Escape - Four Marine Evacuation Stations, two port and two starboard, and two external stairs aft. The two forward MES serve a total of 100 persons each, the two mid MES serving a total of 200 persons each and one aft stair serving 200 persons and one aft stair serving 100 persons. A total of ten x 100 person rafts are fitted.

Rescue - Two SOLAS inflatable rescue dinghy with 30 hp motor and approved launch / recovery method.

Lifejackets with lights for full compliment fitted under seats and in storage cabinets.

Safety Equipment - Liebuoys with lights and lines, smoke flares, Immersion suits, flares and lines throwing device fitted in accordance with international regulations

Fire Safety

Fire Detection - An addressable fire detection system covers at minimum all high and moderate risk spaces (other than the wheelhouse) with alarm panel situated in the wheelhouse. CCTV system covers at a minimum, engine rooms, ante rooms, vehicle spaces, jet rooms, MES and liferafts stations (serving as mooring cameras) with monitors in the wheelhouse.

Fire Sprinklers - Vehicle deck and passenger cabin are protected by drencher systems with overhead sprinklers. Pump control is from the wheelhouse and anterooms.

General Equipment - Portable fire extinguishers, Fireman's outfits and equipment, water fog applicators, breathing apparatus, international connections and fire control plans fitted in accordance with international regulations.

Machinery Installations

Main Engines - 4 x resiliently mounted Ruston 20RK270 or Caterpillar 3618 marine diesel engines.

Water Jets - 4 x Lips LJ145D waterjets with steering and reverse.

Transmission - 4 x Renk ASL60 gearboxes, approved by the engine manufacturer, with reduction ratio for optimum jet shaft speed.

Ride Control - A 'Maritime Dynamics' active ride control system is fitted to maximise passenger comfort. The system combines active trim tabs aft and optional bolt-on T-foil located at the forward end of each hull.

Electrical Installations

Alternators – 4 x Caterpillar 3406B 230kw

Distribution - 415V, 50 Hz. 3 phase. 4 wire distribution with neutral earth allowing 240 volt supply using one phase and one neutral. Distribution via distribution boards adjacent to or within the space they serve.

INCAT 96m - ***EVOLUTION 10* - 96m Wave Piercing Ro/Pax Catamaran**

Yard No.	-	051
Builder	-	Incat Tasmania Pty Ltd
Class Society	-	Det Norske Veritas
Applicable Regulations	-	DNV HSLC Rules current at the date of contract.
	-	IMO HSC Code and applicable IMO Regulations at date of contract.
Certification	-	DNV ✠1A1 HSLC R1 Car Ferry "B" EO Certificate
Length overall	-	95.47m
Length waterline	-	86.00m
Beam overall	-	26.60m
Beam of Hulls	-	4.50m
Draft loaded	-	4.03m
Speed	-	38 knots at 868 tonnes deadweight
	-	47 knots at Lightship

*Note - All speeds quoted at 100% MCR 4 x 7080 KW @ 1030 rpm excluding T-foil.

Max Deadweight	-	868 tonnes
Total persons	-	750 maximum
Vehicle Deck Capacity	-	330 truck lane metres at 3.1m wide x 4.0m/4.35m clear height plus 80 cars at 4.5m length x 2.3m width or maximum 230 cars only.
Axle loads	-	Transom to Frame 47 - 10 tonnes per dual wheel axle or axle groups to suit European standards. Fwd of Frame 47 Ramp A to D and Optional Mezzanine Decks - 0.8 tonnes per axle.
Fuel Capacity	-	4 x 43,720 litre integral aluminium tanks and 2 x 196,428 litre long-range tanks.
Fresh Water	-	1 x 5000 litre GRP tank.
Sewage	-	1 x 5000 litre GRP tank.
Lube Oil	-	2 x 465 litre aluminium tanks.

Structures

Design	-	Two slender, aluminum hulls connected by a bridging section with center bow structure at fwd end.
Subdivision	-	Each hull is divided into eight vented, watertight compartments divided by transverse bulkheads. Two compartments in each hull prepared as short range fuel tanks and one as a long range fuel tank.
Fabrication	-	Welded and bonded aluminum construction, with longitudinal stiffeners supported by transverse web frames and bulkheads. Aluminum plate grade 5383 H116 or 5518 H116 and extrusion grade 6082 T6 and 5083 H112.

Passenger Accommodation and Escape

Wall Coverings	-	Ayrlite 2005 laminated pre-finished composite board.
Floor Coverings	-	80:20 tufted carpet and selected chlorine-free vinyl type flooring adhered to decks with epoxy adhesive.
Ceilings	-	Luxalon 300C aluminium linear ceiling, cotton white, with 75mm semi-circular trim between every three panels or Luxalon 180 B aluminium linear ceiling.
Windows	-	Combination of Incat "Glass Only" window installation system with toughened glass and aluminium framed sliding windows where required.
Lighting	-	Peirlite 18W PL tube recessed downlights and Staff recessed low voltage 12v 50W downlight with dichoric lamps.

Air Conditioning	-	Sanyo model SPW-XC 483 throughout capable of maintaining between 20-22 deg C and 50% RH with a full passenger loads and ambient temperature of 32 deg C and 50% RH
Ventilation System	-	Supply fans will provide fresh air into the Pax area at a rate of 3 air changes per hour. Pantry, Kiosk, Bar, and Toilet exhaust fans provide 30 air changes per hour within the space. Purge and exhaust fans will purge the air from the Pax space at the rate of 6 air changes per hour
Aircraft Seating	-	Beurteaux Ocean Tourist high back reclining and fixed seats with open arm rest, magazine holders, folding meal table, under-seat life jacket holders and wool fabric upholstery with leather trim.
Lounges	-	Beurteaux Tub seats with wool fabric upholstery and leather trim.
Bar Stools	-	Incat Bar stools with selected wool fabric upholstery, stainless steel pillar and footrest.
Passenger Access	-	Four stairways from the vehicle deck offer entry to Pax area. Two fwd and two amidships plus disability access ramp from forward vehicle deck.
	-	Shore access via two dedicated passenger gangway gates port and stbd at the pax level aft.
Public Address	-	Builder's standard public address system to cover all passenger and crew areas, vehicle decks, stairwells and anterooms. Colour televisions fitted throughout the passenger cabin configured to receive video, safety messages and input from the electronic chart system.
Alarm	-	Two tone general alarm (seven short and one long) signal generator activated from wheelhouse.
Escape	-	Escape is via Four Marine Evacuation Stations, two port and two stbd. A total of nine 100person rafts are fitted.
	-	2 x SOLAS inflatable dinghy with 30 hp motor and approved launch / recovery method.

Fire Safety

Fire Detection	-	An addressable fire detection system covers at minimum all high and moderate risk spaces (other than the wheelhouse) with alarm panel situated in the wheelhouse with CCTV cameras.
Fire Protection	-	Lightweight structural fire protection protects all moderate and high risk spaces.
ER Fire Control	-	CO2 system for each engine room together with second shot cross connection.
Drenchers	-	Vehicle deck is protected by a zoned drencher system capable of operating two zones simultaneously. Pump control is from the wheelhouse and anterooms.
	-	Pax area is protected by a zoned, dry closed bulb drencher system interconnected with control valves to a single vehicle deck drencher pump.
Hydrants	-	Two electric motor driven pumps, one in Void 2 port and stbd, feed into a common loop which feed fire hydrants distributed throughout the ship.
General Equipment	-	Portable fire extinguishers, Fireman's outfits and equipment, water fog applicators, breathing apparatus, international connections and fire control plans included to meet rule requirements.

Machinery Installations

Main Engines	-	4 x resiliently mounted Ruston 20RK270 marine diesel engines, each rated at over 7080 kW at 1030 rpm.
Water Jets	-	4 x Lips LJ150D waterjets configured for steering and reverse.
Transmission	-	4 x Reintjes VLJ 6831 gearboxes, approved by engine manufacturer, with reduction ratio suited for optimum jet shaft speed.
Hydraulics	-	Three hydraulic power packs, one forward and two aft, for running of mooring capstans, anchor winch, ride control, steering/reverse and rescue boat cranes.
Ride Control	-	A 'Maritime Dynamics' active ride control system is fitted to maximise passenger comfort. This system combines, active trim tabs aft and optional fold-down T-foil located at aft end of centre bow fitted with active fins. The structural abutment, electrical and hydraulic services to receive the fwd T-foil will be fitted as standard to the vessel.
Trim Tabs	-	A hydraulically operated trim tab is hinged at the aft end of each hull.
Monitoring	-	An electronic alarm and monitoring system with dual central VDU displays, keyboards and printer fitted in the wheelhouse. Alarm and monitoring to meet the requirements of the HSC Code, the HSLC Rules and EO requirements.

Communication	-	A 'David Clark' system is fitted to allow communication between the Wheelhouse helm position, Aft vehicle deck, Anchor area, Anterooms, Jet rooms, Engine rooms, T-foil void. All points have call facilities to the wheelhouse via headset stations with volume control.

Electrical Installations

Alternators	-	4 x Caterpillar 3406B 245 kW (nominal) marine, brushless, self-excited alternators.
Distribution	-	415V, 50 Hz. 3 phase. 4 wire distribution with neutral earth allowing 240 volt supply using one phase and one neutral. Distribution via distribution boards adjacent to or within the space they serve.
Switchboards	-	Main switchboards fitted with a load preferential trip system which automatically sheds non essential loads whilst still maintaining one alternator as a standby set. Each switchboard fitted with a bus coupler breaker to allow the main bus bars to be split in the event of a fault condition.
Essential Distribution	-	Distribution to essential services from independent distribution boards supplied from both switchboards.
Shore Power	-	60 amp 415V 3 phase outlet fitted in port and stbd anterooms.
24v DC Systems	-	Separate systems for automation and to power ship's radio communication.
Essential Lighting	-	10% of the main light fittings are powered from the essential services distribution board. Essential lights and exit signs fitted as required and indicated by red dot.
Navigation Lights	-	Dual power supply (Main and essential services) controlled from the wheelhouse for all navigation lights including NUC and anchor lights.
Cathodic Protection	-	Sea inlets and jet area protected by high capacity anodes. Hull potential monitoring system, alarmed to the wheelhouse fitted.

Operating Compartment

Operation	-	There are three forward facing seats around the centre line. Captain in the centre, Navigator to starboard and Engineer to port. Main Console contains all required navigation, communication and monitoring equipment.
Communication	-	The three onboard communication systems are operable from the wheelhouse, enabling communication to all machinery, mooring and passenger spaces.

Navigational Equipment

GPS	-	2 x Leica Differential GPS
Radars	-	Captain - Bridgemaster X band with 15" True motion performance monitor inc. auto track and geographics
	-	Navigator - Bridgemaster S band with 15" Arpa performance monitor inc. auto track and geographics (Radar interswitching)
Autopilot	-	Lips
Gyro Compass	-	An Schutz
Magnetic Compass	-	Plath
Electronic Chart System	-	Transis
Echo Sounder	-	Skipper
Speed / Distance Log	-	Walker electromagnetic with interface to radar's, GPS and autopilot.
Wind Speed/Direction	-	Walker
Weather Fax / Navtex	-	Furuno
Barometer / Clock	-	Builders standard
Air Horn	-	Ibuki
Daylight Signal Lamp	-	Aldis Francis
Search Light	-	Mounted on fwd mast with remote control - Den Hann

Radio Communications

MF / HF Radios	-	}	
HF DSC inc. 2187.5 kHz	-	}	
Simplex / Semi Duplex VHF Transceivers	-	}	
VHF / DSC Controller with Ch.70 Receive	-	}	To comply with GMDSS Sea Area 1 and 2
Hand held transceivers inc. chargers.	-	}	
EPIRB (406 Mhz)	-	}	
SART	-	}	
Satcom C	-	}	

INCAT Evolution 10 – Spec Sheet provided by INCAT website as
"96mwpc-ropax-onepage-004.doc"

General Particulars

Builder	Incat Tasmania Pty Ltd.
Class Society	Det Norske Veritas
Certification	DNV +1A1 HSLC R1 Car Ferry "B" EO
Length overall	96.00m
Length waterline	86.00m
Beam overall	26.60m
Beam of Hulls	4.50m
Draft	4.00m approx. in salt water
Speed	38 knots @ 1650 tonnes displacement
	42 knots @ 1400 tonnes displacement

*Note - All speeds quoted at 100% MCR (4 x 7080kw @ 1030 rpm) and excluding T-foil.

Capacities

Max Deadweight 675 tonnes (estimate based on building specification).

Total persons up to 900 persons

Vehicle Capacity 380 truck lane metres at 3.1m wide x 4.35m clear height plus 90 cars at 4.5m length x 2.3m wide or 260 cars only using optional mezzanine decks.

Axle loads - Transom to Frame 47 - 10 tonnes per dual wheel axle or axle groups to suit European standards. Forward of Frame 47 - Ramp A to D and Optional Mezzanine Decks - 0.8 tonnes per axle.

Fuel Capacity - 4 x 40m3 integral aluminum tank and additional long-range tank of minimum 170m3 capacity provided in each hull.

Fresh Water 1 x 5.0 m3 GRP tank.

Sewage 1 x 5.0 m3 GRP tank

Construction

Design - Two slender, aluminum hulls connected by a bridging section with center bow structure at fwd end. Each hull is divided into eight vented, watertight compartments divided by transverse bulkheads. Two compartments in each hull prepared as short-range fuel tanks and one as a long-range fuel tank.

Welded and bonded aluminium construction using longitudinal stiffeners supported by transverse web frames and bulkheads. Aluminum plate grade 5383 H116 or 5518 H116. Aluminium extrusions grade 6082 T6 and 5083 H112.

Air Conditioning

Sanyo model SPW-XC 483 throughout capable of maintaining between 20-22 deg C and 50% RH with a full passenger loads and ambient temperature of 32 deg C and 50% RH

Evacuation

Escape is via Four Marine Evacuation Stations, two port and two starboard, and two external stairs aft. The two forward MES serve a total of 200 persons each (4 x 100), the two mid MES serve a total of 200 persons each (4x100) and one aft stair serving 100 persons. A total of ten x 100 person rafts are fitted.

2 x SOLAS inflatable dinghy with 30 hp motor and approved launch / recovery method.

Machinery Installations

Main Engines - 4 x resiliently mounted Ruston 20RK270 or Caterpillar 3618 marine diesel engines

Water Jets - 4 x Lips LJ150D waterjets configured for steering and reverse.

Transmission - 4 x Reintjes gearboxes, approved by the engine manufacturer, with reduction ratio suited for optimum jet shaft speed.

Hydraulics - Three hydraulic power packs, one forward and two aft, all alarmed for low level, high temperature and filter clog and low pressure. One pressure line filter and two return line filters fitted. An off-line filter / pump provided.

Ride Control - A 'Maritime Dynamics' active ride control system is fitted to maximise passenger comfort. This system combines active trim tabs aft and optional fold-down T-foil located at aft end of centre bow fitted with active fins. The structural abutment, electrical and hydraulic services to receive the fwd T-foil will be fitted as standard to the vessel.

Electrical Installations

Alternators – 4 x Caterpillar 3406B 230kw (nominal) marine, brushless, self-excited alternators.

Distribution - 415V, 50 Hz. 3 phase. 4 wire distribution with neutral earth allowing 240 volt supply using one phase and one neutral. Distribution via distribution boards adjacent to or within the space they serve.

INCAT Evolution 10B

General Particulars

Builder	Incat Tasmania Pty Ltd.
Class Society	Det Norske Veritas
Certification	DNV +1A1 HSLC R1 Car Ferry "B" EO
Length overall	97.22m
Length waterline	92.00m
Beam overall	26.60m
Beam of Hulls	4.50m
Draft	3.42m maximum
Speed	36 knots @ 750 tonnes deadweight
	40 knots @ 375 tonnes deadweight

*Note - All speeds quoted at 100% MCR (4 x 7080kw @ 1030 rpm) and excluding T-foil.

Capacities

Max Deadweight - 750 tonnes (to be confirmed at time of contract)

Total persons - up to 900 persons

Vehicle Capacity - 380 truck lane metres at 3.1m wide x 4.35m clear height plus 80 cars at 4.5m length x 2.3m wide or 260 cars only using optional mezzanine decks.

Axle loads – Transom to Frame 49 - 10 tonnes per dual wheel axle or axle groups to suit European standards. Forward of Frame 49 - Ramp A to D and Optional Mezzanine Decks - 0.8 tonnes per axle.

Fuel Capacity - 4 x 40m3 integral aluminum tank and additional long-range tank of minimum 170m3 capacity provided in each hull.

Fresh Water	1 x 5.0 m3 GRP tank.
Sewage	1 x 5.0 m3 GRP tank

Construction

Design - Two slender, aluminum hulls connected by a bridging section with center bow structure at fwd end. Each hull is divided into nine vented, watertight compartments divided by transverse bulkheads. Two compartments in each hull prepared as short-range fuel tanks and one as a long-range fuel tank. Welded and glued aluminum construction using longitudinal stiffeners supported by transverse web frames and bulkheads. Aluminum plate grade 5383 H116 or 5518 H116. Aluminium extrusions grade 6082 T6 and 5083 H112.

Air Conditioning

Sanyo model SPW-XC 483 throughout capable of maintaining between 20-22 deg C and 50% RH with a full passenger loads and ambient temperature of 32 deg C and 50% RH

Evacuation

Escape is via Four Marine Evacuation Stations, two port and two starboard, and two external stairs aft. The two forward MES serve a total of 200 persons each, the two mid MES serve a total of 200 persons each and one aft stair serving 100 persons. A total of ten 100person rafts are fitted.

2 x SOLAS inflatable dinghy with 30 hp motor and approved launch / recovery method.

Machinery Installations

Main Engines - 4 x resiliently mounted Ruston 20RK270 or Caterpillar 3618 marine diesel engines, each rated at 7080 KW.

Water Jets - 4 x Lips 120E waterjets configured for steering and reverse.

Transmission - 4 x Reintjes gearboxes, approved by the engine manufacturer, with reduction ratio suited for optimum jet shaft speed.

Hydraulics - Three hydraulic power packs, one forward and two aft, all alarmed for low level, high temperature and filter clog and low pressure. One pressure line filter and two return line filters fitted. An off-line filter / pump provided.

Ride Control - A 'Maritime Dynamics' active ride control system is fitted to maximise passenger comfort. This system combines active trim tabs aft and optional fold-down T-foil located at aft end of centre bow fitted with active fins. The structural abutment, electrical and hydraulic services to receive the fwd T-foil will be fitted as standard to the vessel.

Electrical Installations

Alternators – 4 x Caterpillar 3406B 230kw (nominal) marine, brushless, self-excited alternators.

Distribution - 415V, 50 Hz. 3 phase. 4 wire distribution with neutral earth allowing 240 volt supply using one phase and one neutral. Distribution via distribution boards adjacent to or within the space they serve.

AUSTAL AutoExpress 101 – from Austal file “ae101-euroferry.pdf”

AUTO EXPRESS 101

"EUROFERRYS PACIFICA"

AUSTAL YARD NO: 114

STYLING AND INTERIOR BY OLIVER DESIGN

DESTINATION: SPAIN

DELIVERED: APRIL 2001

CLIENT: EUROFERRYS

PRINCIPAL DIMENSIONS

Length Overall	101.0 metres
Length (Immersed hull)	88.70 metres
Beam (Moulded)	26.65 metres
Depth (Moulded)	9.4 metres
Hull Draft (Approx.)	4.2 metres
Vehicle Deck Clear Heights	
centre lane	4.6 metres
side lanes	2.7 metres
mezzanine lanes	2.0 metres

PAYLOAD & CAPACITIES

Passengers	951 (~60 non-revenue outdoor)
Vehicles	251 cars or 16 trucks and 96 cars
Maximum Deadweight	750 tonnes
Maximum Axle Loads	
centre lanes	(dual wheels) 15.0 tonnes (single wheels) 12.0 tonnes
side lanes	3.0 tonnes
mezzanine lanes	1.0 tonne
Fuel	160,000 litres

PROPULSION

Main Engines	4 x Caterpillar 3618 4 x 7,200 kW @ 1050 rpm
Gearboxes	4 x Reintjes VLJ 6831
Waterjets	4 x KaMeWa 125 SII

PERFORMANCE

Speed (500 t DWT, 90% MCR)	37 knots
Fuel Consumption (approx.) @ 90% MCR	5.3 tonnes/hr

SURVEY

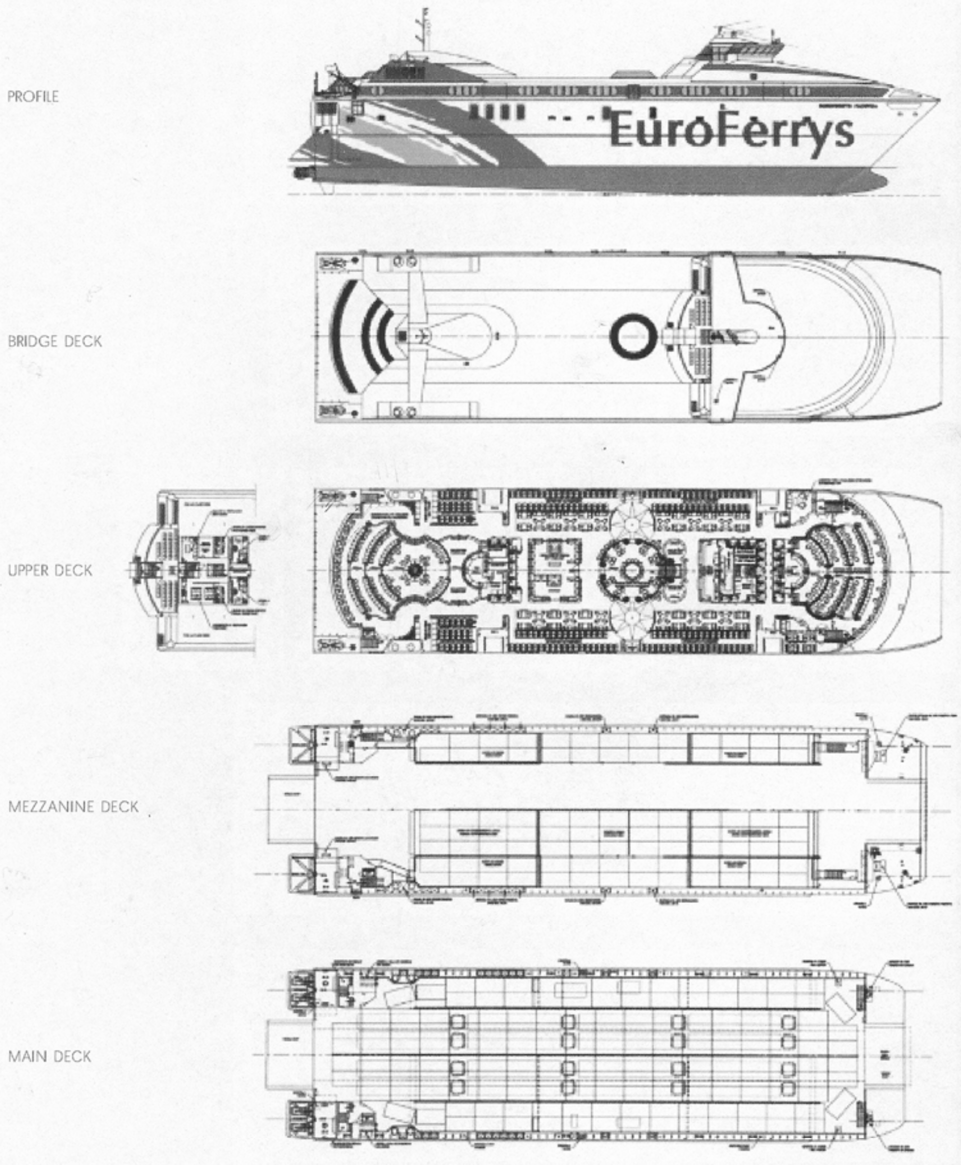
Germanischer Lloyd	✠ 100A5, HSC-B OC3

Styling and Interior by Oliver Design

AUSTAL SHIPS

AUTO EXPRESS 101

Vessel Type: 101m High Speed Vehicle-Passenger Catamaran



"EUROFERRYS PACIFICA"

AUSTAL YARD NO: 114

DESTINATION: SPAIN

DELIVERED: APRIL 2001

CLIENT: EUROFERRYS

100 Clarence Beach Road | Tel +61 8 9410 1111
Henderson | Fax +61 8 9410 2564
Western Australia 6166 | www.austal.com

AUSTAL SHIPS

INCAT Evolution 112 – Spec Sheet provided by INCAT website as "112mwpc-ropax-onepage-0.doc"

General Particulars

Designer	Incat Tasmania Pty Ltd.
Builder	Incat Tasmania Pty Ltd.
Class Society	Det Norske Veritas
Certification	DNV ✠1A1 HSLC R1 Car Ferry "B" EO
Length overall	112.63 m
Length (hulls)	105.60 m
Beam (moulded)	30.20 m
Beam (hulls)	5.80 m
Draft	3.30m approx. in salt water
Speed	40 knots @ 1000 tonnes deadweight 45 knots @ 500 tonnes deadweight

*Note - All speeds quoted at 100% MCR (4 x 9000 kW) and excluding T-foil.

Capacities

Deadweight - 1000 tonnes (1500 tonnes 'cargo only' at reduced speed).

Passengers capacity - up to 1000 persons

Vehicle Capacity - 589 truck-lane metres at 3.5m wide plus 50 cars at 2.3m wide or 321 cars only.

Vehicle Deck Clear Heights - 6.30m centre vehicle deck, 5.95m mezzanine decks raised, 2.10m upper mezzanine decks and 3.90m under mezzanine decks.

Axle loads - 12 tonnes per single axle (European standard) transom to frame 63/72, 0.8 tonnes per axle forward of frame 63 outboard and 0.8 tonnes on mezzanine decks.

Fuel Capacity - Six integral aluminium tanks (three in each hull) to provide short and long range capacity.

Fresh Water - 5000 litres

Construction

Design - Two slender, aluminum hulls connected by a bridging section with center bow structure at fwd end. Each hull is divided into nine vented, watertight compartments divided by transverse bulkheads. Three compartments in each hull prepared as fuel tanks with additional strengthening on each of the end bulkheads and intermediate tank top. Welded aluminium construction using predominantly grade 5383-H116 or 5518-H321, and extrusion grade 6082-T6, 5083-H112, 5383-H112 or 5518-H112. Longitudinal stiffeners supported by transverse web frames and bulkheads.

Air Conditioning

Sanyo ceiling mounted reverse cycle units capable of maintaining between 20-22 deg C and 50% RH with a full passenger loads and ambient temperature of 32 deg C and 50% RH

Evacuation

Escape - Four Marine Evacuation Stations, two port and two starboard, and two external stairs aft. The two forward MES serve a total of 200 persons each (4 x 100), the two mid MES serve a total of 200 persons each (4 x 100) and one aft stair serving 200 persons. A total of eleven 100-person liferafts are fitted.

Rescue - Two SOLAS inflatable dinghy with 30 hp motor and approved launch / recovery method.

Machinery Installations

Main Engines - Four resiliently mounted Ruston 20RK280 each rated at 9000kW at 100% MCR at 25 deg C ambient temperature.

Water Jets - Four Lips 150E waterjets configured for steering and reverse.

Transmission - Four Gearboxes approved by the turbine manufacturer, with reduction ratio suited for optimum jet shaft speed.

Hydraulics - Three hydraulic power packs, one forward for operating anchor winch, capstans and ride control. Two aft for operation of waterjet steering and bucket movement, capstans and ride control.

Ride Control - A 'Maritime Dynamics' active ride control system is fitted to maximise passenger comfort. This system combines, active trim tabs aft and optional fold-down T-foil located at aft end of centre bow fitted with active fins. The structural abutment, electrical and hydraulic services to receive the fwd T-foil will be fitted as standard to the vessel.

Electrical Installations

Alternators - A combination of marine diesel driven, brushless, self-excited alternators (total combined output of 1200kw).

Distribution - 415V, 50 Hz 3 phase 4-wire distribution system with neutral earth allowing 240 volt supply, using one phase and one neutral. Distribution via distribution boards adjacent to or within the space they serve.

STENA HSS 1500

Le type HSS 1500

HSS : **H**igh-speed **S**ea **S**ervice

Trois HSS 1500 ont été commandés par la Stena Line aux chantiers finlandais Finnyards en Juillet 1993 au coût unitaire d'environ 65 millions de Livres Sterling. Le premier, le STENA EXPLORER est entré en service au printemps 1996 entre le Pays de Galles et l' Irlande sur un parcours de 55 milles parcouru en 99 minutes.Le second est le STENA VOYAGER et le troisième le STENA DISCOVERY a été livré fin mai 1997 pour la ligne Harwich (UK) - Hook of Holland.

	**HSS STENA EXPLORER** **Caractéristiques principales**
Type	HSS 1500 Passagers/Véhicules Catamaran Ferry
Classification	DNV +1A1 HSLC R1 Car Ferry A EO ICS NAUT
Armateur	Stena Rederi
Chantiers	Finnyards
Construction	Aluminium et composite
Longueur hors-tout	126,6 m
Largeur	40 m
Tirant d'eau	4,5 m
Déplacement	1500 t
Machines	2 turbines à gaz LM 2500 soit 2x20500 KW 2 turbines à gaz GE LM1600 soit 2x13500 KW
Réducteurs	2 MAAG HPG 185/C
Propulseurs	4 WJ KAMEWA S160
Auxiliaires	4 CUMMINGS-STANFORD de 747 KW chaque

Vitesse	25 nds avec 2 LM 1600 soit 27 MW 32 nds avec 2 LM 2500 soit 41 MW 40 nds avec les 4 turbines soit 68 MW
Consommation	8 m3/h avec 2 LM 1600 15 m3/h avec 2 LM 2500 50 m3/h avec les 4 turbines
Autonomie	
Capacité GO	235 m3
Capacité Eau douce	20 m3
Passagers	1500
Véhicules	275 ou 120 + 50 camions
Equipage	48 à 75
Stabilisation	NIL

PacifCat On-Board Ramp Feasibility Study

➢ *Technical Memorandum*

PacifiCat On-Board Ramp Feasibility Study

August 17, 2000

Performed by:

Jeff Kelton, Sr. Naval Architect
John J. McMullen Associates, Inc.
400 Warren Ave., Suite 320
Bremerton, Washington 98337
Tel: (360) 782-1112
Fax: (360) 782-1115

Performed for:

PRICEWATERHOUSECOOPERS

TECHNICAL MEMORANDUM

From: JJMA – Jeff Kelton, Senior Naval Architect
To: PricewaterhouseCoopers - Mr. Mark Hodgson

Subj: PacifiCat On-Board Ramp Feasibility Study

EXECUTIVE SUMMARY

John J. McMullen Associates, Inc., was tasked by PricewaterhouseCoopers to investigate the feasibility of adding vehicle ramps to the existing PacifiCats to facilitate on-board vehicle passage between the upper and lower car decks. The PacifiCats are presently in double-ended operation and dock at two-brow terminals in British Columbia, having the capability of loading vehicles on the two levels simultaneously with two parallel lanes per deck. It is rare to find such two-level terminals anywhere else in the world. Thus this study, while not being a detailed parking or traffic flow study, investigates the feasibility of adding onboard ramps to facilitate the boarding of vehicles on only the lower deck at more conventional ferry terminals.

The study investigated three on-board single-ramp configurations, one for double-ended operation / drive-through loading, one for single-ended operation / bow-to loading, and one for single-ended operation / stern-to loading. Sketches for each of the three configurations confirms the feasibility of the ramps from an arrangement standpoint. An order-of-magnitude estimate of the impacts of the modification upon the ship and the dollar cost thereof are presented in table 1

Table 1: Comparison of Modification Configurations

Configuration	*Car Capacity*	*Weight (tonne)*	*Terminal Time*	*Performance*	*Costs (US)*
Double-Ended / Drive-Through	Max: 233 Real: 228	Decrease of 23.5 - 31	Small Lengthening	Trim: Negligible Change Full Load Speed: Appreciable Increase	$1.6M to $2.4M
Single-Ended / Bow-To Ops	Max: 223 Real: 213	Decrease of 39.5 – 54.5	Considerable Lengthening	Trim: Positive Change (aft trim) Full Load Speed: Significant Increase	$0.8M to $1.2M
Single-Ended / Stern-To Ops	Max: 230 Real: 220	Decrease of 29 - 44	Considerable Lengthening	Trim: Negative Change (fwd trim) Full Load Speed: Slight Increase	$0.8M to $1.2M

Ramp placement was determined to have the least impact on ship systems if located in the lane just to port of ship's centerline, rather than at the side shell. Having the ramps hoistable preserves the weathertight integrity of the upper deck. Having the ramps as much as possible within the bounds of the weathertight roller doors preserves the existing fire boundaries and the operation of the water fog fire suppression system.

The scantlings of the PacifiCats were painstakingly optimized to the maximum during the design phase, and probably very little margin exists currently on global longitudinal strength. The cutouts in the decks for the ramps will have a very significant deleterious effect on the global strength of these vessels, and will require extensive structural modifications to provide adequate reinforcements. For these reasons, only single-lane ramps, as opposed to double-lane ramps, were investigated. Also, certain pairs of existing stanchions will require removal for adequate vehicle turning clearances, and compensating structure designed and installed, to carry the loads the removed stanchions supported.

An extensive 3D finite element engineering analysis will be required to fully prove the feasibility of installing vehicle ramps on the PacifiCats, as well as a detailed design effort to eventually construct them. These costs are reflected in table 1.

BACKGROUND & ORGANIZATION OF REPORT

The three PacifiCats were designed for the specialized two-deck loading terminals of B.C. Ferries at Horseshoe Bay, north of Vancouver, and at Departure Bay, north of Nanaimo on Vancouver Island. Two of the PacifiCats, the Explorer and Discovery, are shown in figure 1 docked bow-to and stern-to at the Departure Bay Terminal.

Figure 1: Departure Bay Terminal

The PacifiCats are presently capable of loading vehicles on two levels simultaneously with two parallel lanes per deck. There is presently no way for vehicles to move between the upper and lower vehicle decks onboard. This configuration requires that the shore facility be equipped with a double-deck vehicle brow. The offloading of passenger cars from the upper deck, and a commercial truck and cars from the lower deck, is shown in figure 2.

Figure 2: Passenger Cars Disembarking Upper Deck / Truck Disembarking Lower Deck

While such vehicle brows are common in British Columbia, they are not common elsewhere. It is expected that many prospective buyers will be interested in what it would take to modify the ships such that all vehicle loading and unloading is accomplished from the lower vehicle deck.

This memorandum presents the results of John J. McMullen Associates' brief feasibility study of this subject. The study investigated three on-board ramp configurations, one for double-ended operation / drive-through loading, one for single-ended operation / bow-to loading, and one for single-ended operation / stern-to loading. For each of the three configurations we present sketches of the studied configuration, and order-of-magnitude estimates of the impacts of the modification upon the ship and the dollar cost thereof. All costs are estimated in US Dollars, assuming performance of the work by a North American shipyard. All such cost estimates are necessarily rough, as the variance in price between shipyards in different parts of the world and with different skill levels in aluminum fabrication, can of course be considerable.

RAMP ARRANGEMENT CONSIDERATIONS

Flexible Arrangements: Vehicle ramps aboard ferries, whether fixed or hoistable, are a very mature technology with several ferry-building shipyards worldwide having experience installing them during new construction or as retrofits to add increased flexibility and vessel usefulness. Often such ramps are just one element of on-board vehicle loading systems which may include stern, bow, and/or side loading doors, hoistable mezzanine decks, ramp covers, and flood control doors as shown in figure 3 below.

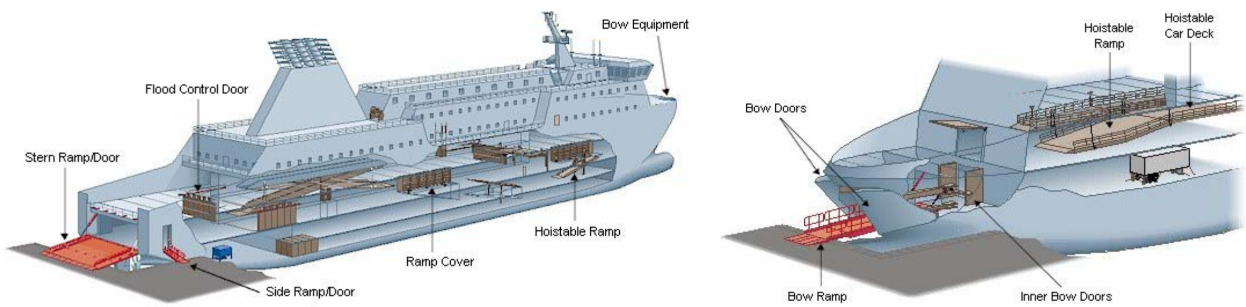


Figure 3: Flexible Vehicle Loading Systems

The flexibility provided by such vehicle loading systems, and vehicle ramps in particular, can be vitally important to commercial operators with conventional ferry slips which normally would have a single brow transfer span for loading and unloading at the lower vehicle deck level. This flexibility is indispensable for naval sealift applications of the PacifiCats where the loading and offloading of military vehicles must be accomplished, in an efficient manner anywhere in the world where shore-side infrastructure may range from very good at to very primitive.

Ramp Placement: Maximum utility and efficiency would dictate that the present drive-thru double-ended capabilities of the PacifiCats be preserved. This can be accomplished by the installation of two ramps, one in the fore part of the ferry and another in the aft part. If end-hinged, and hoistable, this allows for the full load-out of the lower vehicle deck when the ramps are in the raised position flush with the upper vehicle deck. Backing and filling of vehicles would be required on the upper deck to take full advantage of its available vehicle deck space.

Single-ended operation of the PacifiCats is also feasible. The accommodation of a single end-hinged hoistable vehicle ramp located at the end of the ferry where the loading/unloading occurs works best in these configurations. Bow loading/unloading, as opposed to the stern would experience easier dock maneuvering, especially in strong wind, as there is greater control with the waterjets in the forward mode than in reverse mode. Both single-ended configurations would experience a significantly greater requirement for the backing and filling of vehicles on the upper deck to maximize that deck's utilization as compared with double-ended operation, and thus terminal turnaround times will be longer in the case of the single-ended configurations.

Especially in the case of naval sealift applications of the PacifiCats, placement of these ramps near the center of the vessel insures the greatest possibility of continued operation of the ramp if the ship sustains battle damage. In the case of commercial applications, placement of the ramps near the centerline of the ferry minimizes the impacts to existing ship systems as compared to placement of the ramps near the side shell of the ferry where many such systems exist.

In order to maintain the integrity of the PacifCat's water fog fire suppression system, the ramps should be placed as far as practicable within the boundaries defined by the upper vehicle deck's fore and aft weathertight roller doors,

shown in figure 4. As the sea states in which the ferries may be operated in the future may well exceed those experienced between Nanaimo and Vancouver, it is also important that the weather tightness of the upper vehicle deck exterior to the fore and aft roller doors be maintained.

Forward Roller Doors

Aft Roller Doors

Figure 4: Upper Vehicle Deck Weathertight Boundaries

Island structures, which enclose passenger access means to the lounge deck above, exist in the lane on the upper vehicle deck just to starboard of centerline. One such structure is shown in figure 5. The location of these island structures limits how far toward amidships a vehicle ramp may be situated and still have enough turning clearance for vehicles to avoid these enclosures.

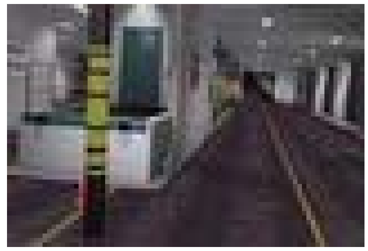


Figure 5: Island Structures

Structure: Numerous stanchions exist on the upper vehicle deck as shown in figure 6. The longitudinal spacing of these stanchions is approximately one car-length, necessitating the removal of one pair of these stanchions at the top of the ramp to provide enough vehicle turning clearance. Compensating structure will need to be added to the adjacent remaining stanchions, and to the overhead, to carry the loads that the removed pair of stanchions supported.

Figure 6: Upper Deck Stanchions

This stanchion removal and compensating structure replacement process may also be required elsewhere on the upper vehicle deck, especially in the case of single-ended ferry operations where a stanchion pair would inhibit the turning of cars to face in the direction in which unloading would occur.

The structural design of the PacifiCats was optimized for lightest scantlings to achieve the best balance among operational requirements, weight, powering and running costs. Complex 3D finite element analyses were performed to achieve these light optimized scantlings. Various load case stresses were investigated including the wave crest landing case shown in figure 7. Because of the overall slender length-to-beam proportions of the PacifiCats dictated by B.C. Ferries' terminal requirements, longitudinal and global strength considerations were uppermost in the structural design of the vessels. Deck penetrations such for the elevator, stairways, hatches, ventilation, and exhaust, were minimized as much as possible. It is clear that to provide cutouts in the upper vehicle deck that are the equivalent of more than 5 standard cars in length and 1 car in breadth will have a profound negative effect on the global strength of the vessel. Fortunately, the cutouts will occur near the neutral axis of the box girder, not close to the tunnel top or the superstructure support deck (raft deck in the midship section shown in figure 8) where the most extreme stresses occur. Corner stresses in the deck cutouts will be high necessitating significant doubler plate deck reinforcing. Major structural reinforcement will be required to recover the longitudinal strength lost to the cutouts. Further 3D finite element analyses will be required to properly design the modifying structure, especially where the vessel may be operated in sea states with more than a 2.5m significant wave height, which was one of the governing criteria in the design of the PacifiCats.

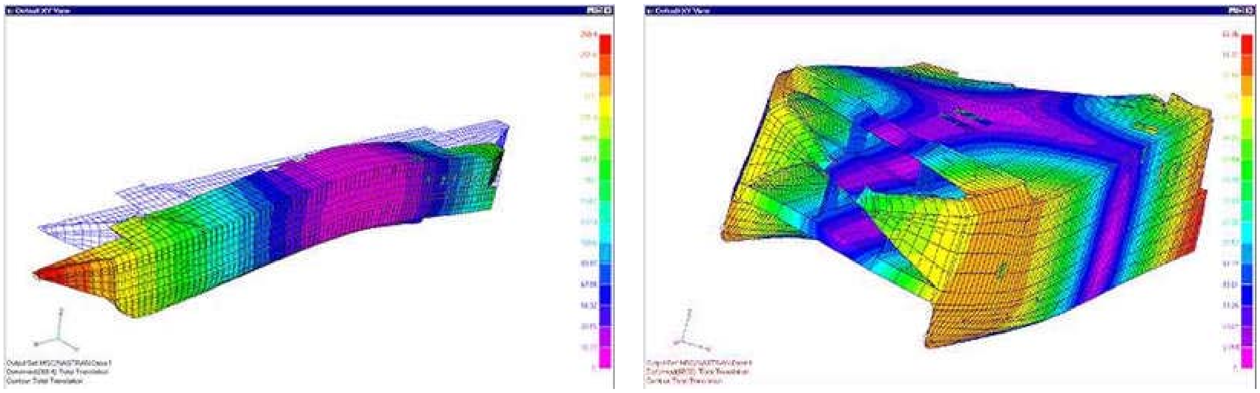


Figure 7: Wave Crest Landing Structural Stresses

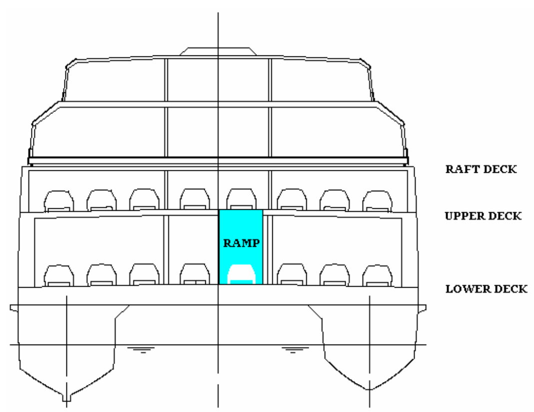
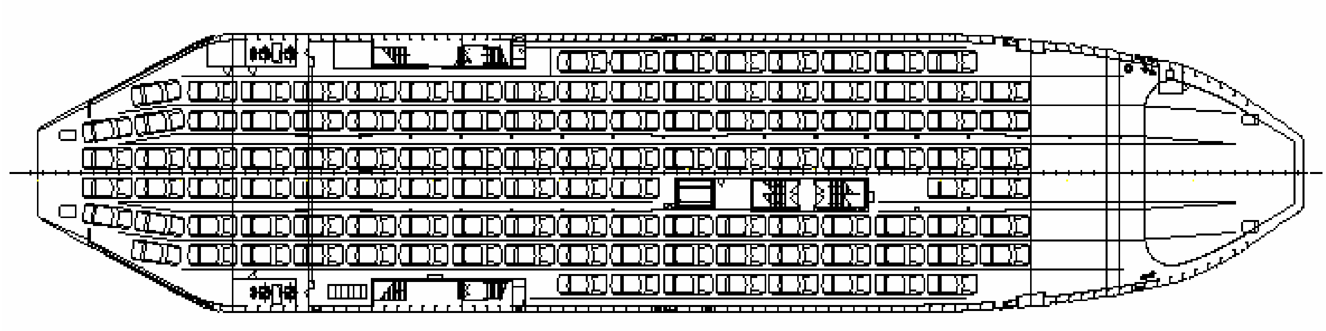


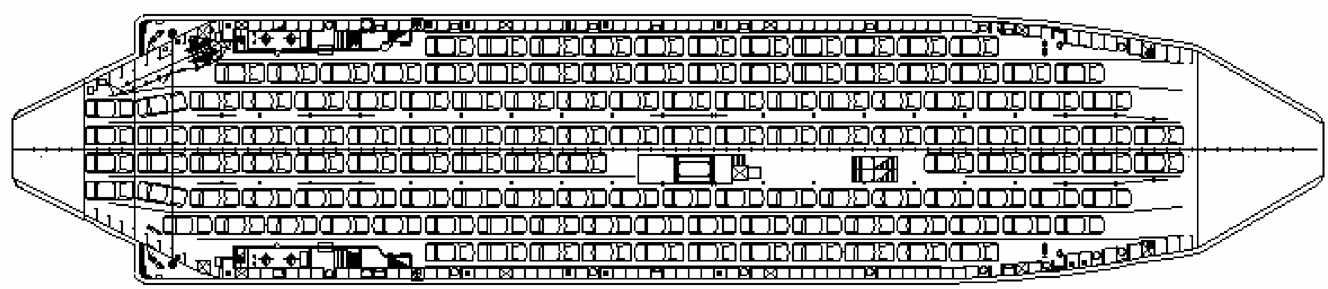
Figure 8: Midship Section With Ramp

THE 'AS-IS' CONFIGURATION

Shown below in figure 9 is the vehicle arrangement of the PacifiCats as they exist presently. The 250 vehicle capacity is comprised of 133 vehicles on the lower deck and 117 vehicle on the upper deck. This 'As-Is' configuration serves as the starting point in the planning of all other possible ramp-incorporating arrangements.



Upper Deck – 117 Vehicles



Lower Deck – 133 Vehicles

Figure 9: PacifiCat ‘As Is’ Configuration - 250 Vehicles Total

Interesting features to note in this ‘As-Is’ configuration are the following:

- Although being an efficient double-ended drive-through arrangement, some backing and filling of cars is still required to fully load out the two vehicle decks.

- On the lower deck to a certain degree, but on the upper deck to a much greater degree all cars are loaded as far aft as possible to maintain the most optimal vessel trim condition for the highest service speeds.

- On the upper deck, vehicles are shown underneath the aft roller doors. In actual operation, these roller doors would be closed so that the water fog fire suppression system could function properly. Thus, either 6 fewer cars would be carried, or a more forward placement of all cars forward of the aft roller doors would need to occur (with a slightly less favorable trim condition) if the full 117 vehicle capacity was to be maintained.

FIRST CONFIGURATION - DOUBLE-ENDED/DRIVE-THROUGH LOADING

Arrangement and General Description: The PacifiCats, as presently configured, use a double-ended / drive-through loading scheme, where vehicles drive aboard the ship over one end, and drive off the ship at the other end. This is an efficient loading scheme, which results in the fastest possible terminal turn-around times, since there is very little maneuvering of vehicles required. To accomplish double-ended drive-through loading using on-board ramps requires at least two ramps. Such a configuration, with maximum load-out is depicted in figure 11. In a stern-to docking scenario, vehicles boarding the ship over the stern will drive up the aftbody ramp onto the upper vehicle deck. They will park as also depicted in figure 11. Note that in order to fill in the parking some backing of vehicles will be necessary, as is presently the case.

Disembarking of vehicles is the reverse of loading, starting with the vehicles at the stern. These vehicles will drive down the forebody ramp and ashore. After the stern end of the upper deck is empty, then the vehicles toward the bow can back up, and proceed down the forebody ramp themselves.

In this system the ramps are hoistable. Once the upper deck is loaded the ramp is hoisted up until it is flush with the upper deck. This clears access to all lanes of the lower deck. Note that in this scheme the lower deck does not require any backing or maneuvering, which means that the loading of large vehicles such as trucks or coaches is not unduly complicated. Further, since the depicted concept only uses a portion of the available width for ramps, it would be possible to have cars loading on the lower deck via the outermost lanes, while vehicles arriving in the center lanes are moving to the upper deck. This should help minimize the impact upon turn-around time.

This configuration is only one of myriad possible ramp configurations. Other possible configurations may be preferred due to some operator's particular needs and / or expected vehicle mix. This configuration, however, will be generally typical in cost and impact of any other ramp arrangement.

Ship Impacts: Having the ramps located in the lane just to port of centerline minimizes impacts to ship systems, which would not be the case if the ramps were near the side-shell with its many ventilation and exhaust ducts, as shown in figure 10.

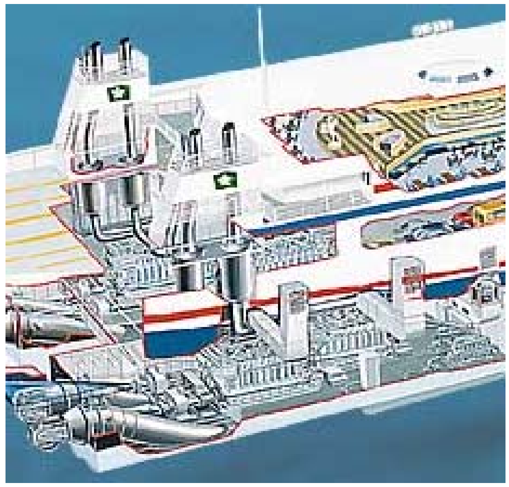


Figure 10: Side-Shell Systems

Near centerline and under the upper deck, there may be some impacts to lighting, electrical wireways, firefighting and other fluid piping systems, but these should be readily reroutable. As noted in a previous section of this study the structural impacts will be considerable. Restoration of the structural fire protection in impacted areas will also be required. Electrical and hydraulic requirements of ramp systems should not significantly burden the vessels existing distributive system capacities.

Vehicle Capacity: The direct footprint of the ramp is approximately 5 AEQ on the upper deck for a ramp with a slope of 1:6 (which prevents vehicles from 'bottoming-out'). This assumes that no vehicles park on the ramp, but that they do park under the ramp when the ramp is in the raised position. An athwartships clear buffer zone on the upper deck of 1 car-length near amidships will be necessary to allow for adequate vehicle turning clearances when disembarking. Figure 11 shows that the upper deck will accommodate a maximum of 100 vehicles. The maximum lower deck capacity should remain unchanged at 133 vehicles.

In the proposed configuration, therefore, the ship will have 2 ramps and a maximum capacity of 233 vehicles. A more realistic loading capacity for minimum terminal turnaround times may be 5 fewer AEQ, or 228 vehicles total.

Weight: The net weight increase of the ramps themselves is minimal, if constructed out of aluminum. Each ramp should weigh approximately 2.5 tonnes, but will replace existing deck structure of about 2 tonnes. With structural reinforcement added in, the net increase in weight due to one ramp will be approximately 1 tonne. Note that the weight of 17 to 22 fewer AEQ, as compared with the vessel's current configuration, is about 25.5 to 33 tonnes.

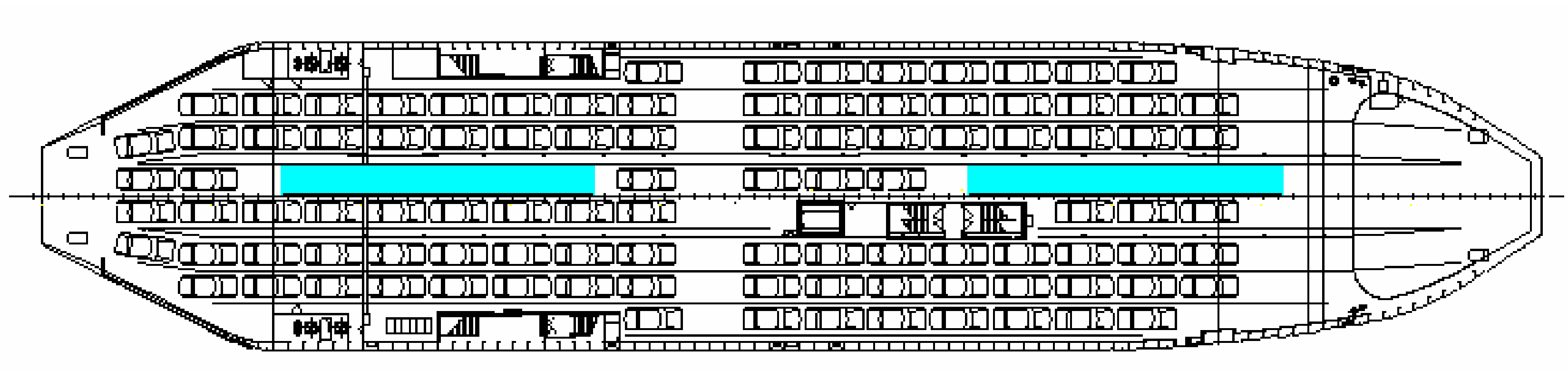
Thus the overall effect of adding two ramps is a weight reduction of approximately 23.5 to 31 tonnes. As the ramps are well distributed fore and aft, the trim of the vessel should not be significantly affected either fore or aft, from what it is presently.

Terminal Turn-Around Time: The two-ramp drive-through configuration will have small negative impact on the present loading and unloading times due to an increase in backing and filling required. B. C. Ferries determined that the best practice for loading of the PacifiCats is to flood the upper deck first, loading the lower deck more slowly. Once the upper deck was full, then load "fine tuning" is accomplished via the latter phases of loading on the lower deck. This same scheme will apply when the ramps are fitted. The ramps will require that the upper deck be loaded first and unloaded last, because lowering the ramps for access requires that the lower deck be clear. Note that there is some degree of simultaneous loading that is still possible, since the ramps do not block all of the loading lanes. Thus, while one line of traffic rolls to the upper deck, other lines may be loading outboard on the lower deck.

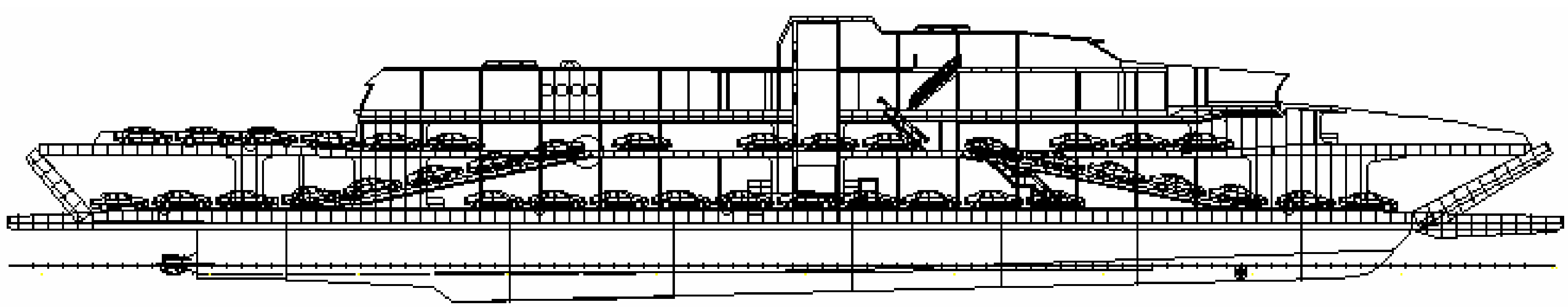
In consideration of all of the above facts, it is expected that the addition of two ramps on order to preserve the drive-through feature of the PacifiCats, will have a small negative impact on the present load and unload times.

Performance: Because of the likely weight impact of a reduction of 23.5 to 31 tonnes, combined with a negligible effect on trim, there will be an appreciable increase in performance in the full load condition. It may be worth noting that the weight impact is negligible upon light ship weight. This means that performance in the light condition will not be changed significantly. This, in turn, suggests that the range of performance between full and empty will be increased appreciably once the ramps are fitted.

Cost Estimate: The unit cost per hoistable vehicle ramp, without structural reinforcement included, is likely to be between $0.4 and $0.6M (US). With structural reinforcement modifications included, the cost per ramp would likely be between $0.8M and $1.2M (US). Therefore, the total cost of this configuration is estimated to be between $1.6M and $2.4M (US). Costs are based on the assumption of the work being performed by a North American shipyard. The cost estimate is necessarily rough, as the variance between shipyards in different parts of the world and with different skill levels in aluminum fabrication, can be considerable.



Upper Deck – 100 Vehicles



Lower Deck – 133 Vehicles

Figure 11: PacifiCat Double-Ended / Drive-Through Configuration – 233 Vehicles

SECOND CONFIGURATION – SINGLE ENDED/BOW-TO LOADING

Arrangement and General Description: For operators which require single-ended operation, the PacifiCats will lend themselves better to docking bow-to, as maneuverability is greater in the forward direction, especially in strong winds. To accomplish single-ended bow-to loading and unloading, one ramp is required in the forebody. Such a configuration, with maximum load-out is depicted in figure 12. Vehicles boarding the ship over the bow will drive up the ramp onto the upper deck, execute a u-turn and park facing the bow. Backing and filling of vehicles will be necessary - much more so than in a drive-through configuration. Disembarking of vehicles is the reverse of loading, starting with the vehicles at the stern. These vehicles will drive down the forebody ramp and ashore, with the vehicles toward the bow backing up and then proceeding down the forebody ramp themselves.

Once the upper deck is loaded, the ramp is hoisted up until it is flush with the upper deck, clearing access to all lanes of the lower deck. Note that in this scheme the lower deck also requires u-turns and backing, which means that the loading of large vehicles such as trucks or coaches is more complicated. It is possible to have cars loading on the lower deck via the outermost lanes, while vehicles are moving to the upper deck, helping to somewhat minimize the impact upon terminal turn-around times.

Ship Impacts: Impacts to structure will be considerable, but impacts to lighting, electrical wireways, piping, and other ship distributive systems will be minimal as these should be readily re-routable in the affected area.

Vehicle Capacity: An impact of 5 AEQ will be felt on the total vehicle capacity on the upper deck for the ramp itself, plus approximately 6 more for space utilization inefficiencies on the lower deck. In addition, it will not be possible to fully load-out the lane that contains the ramp. An athwartships clear buffer zone on the upper deck of 1 car-length near amidships will be necessary to allow for adequate vehicle turning clearances when disembarking. Figure 12 shows that the upper deck will accommodate a maximum of 96 vehicles. The maximum lower deck capacity, with impacts accounted for, is 127 vehicles.

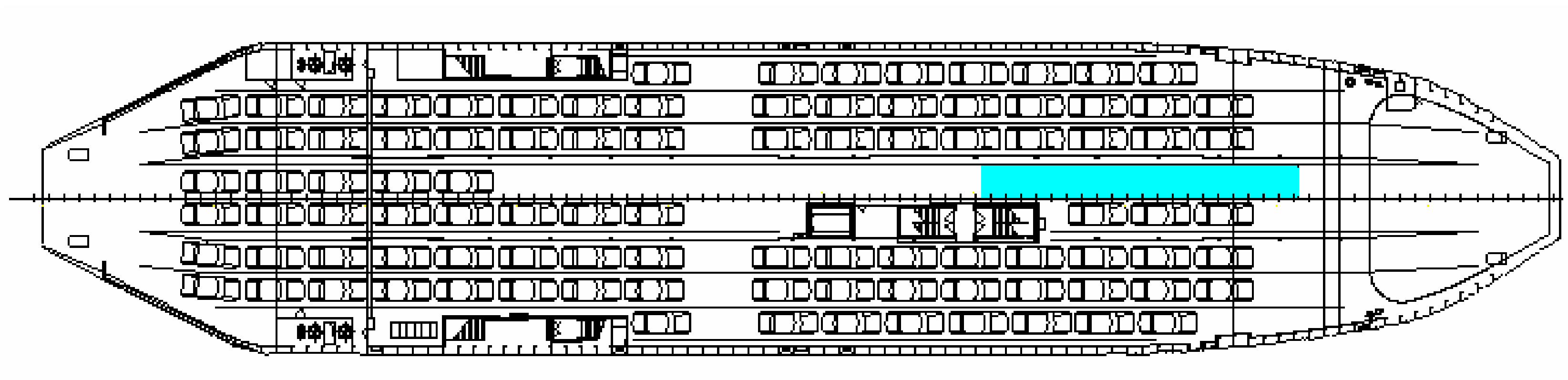
Therefore, in the proposed configuration, the ship will have a maximum capacity of 223 vehicles. A more realistic loading capacity for minimum terminal turnaround times may be 10 fewer AEQ, or 213 vehicles total.

Weight: The net weight increase of the ramps themselves is minimal, if constructed out of aluminum. Each ramp should weigh approximately 2.5 tonnes, but will replace existing deck structure of about 2 tonnes. With structural reinforcement added in, the net increase in weight due to one ramp will be approximately 1 tonne. Note that the weight of 27 to 37 fewer AEQ, as compared with the vessel's current configuration, is about 40.5 to 55.5 tonnes. Thus the overall effect of adding the ramp is a weight reduction of approximately 39.5 to 54.5 tonnes. There will be a net decrease in weight forward, which increases the trim aft slightly, which is a beneficial effect for vessel speed.

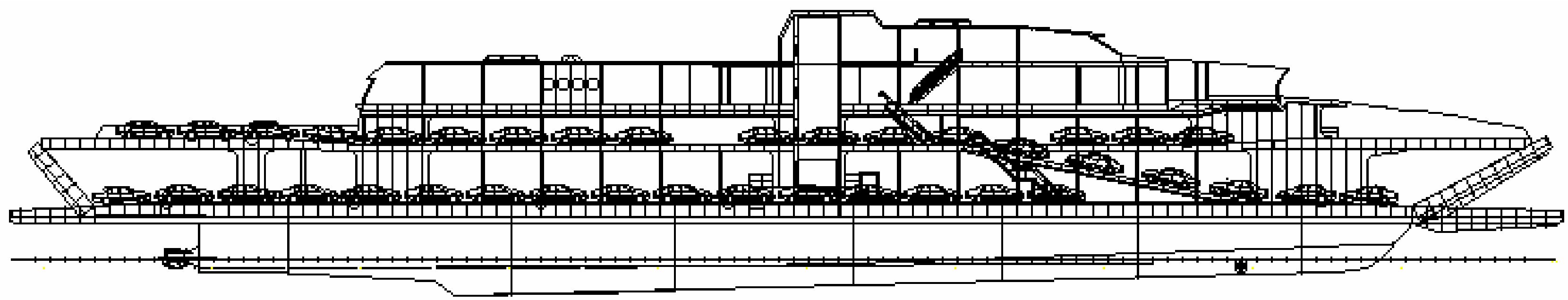
Terminal Turn-Around Time: Loading will proceed by flooding the upper deck first with cars, while loading the lower deck more slowly, followed by "fine tuning" on the lower deck in the latter stages. Some degree of simultaneous loading will still be possible since the ramp blocks only one loading lane of the lower deck. Added terminal time will be required for the u-turns and the backing and filling on both decks. Terminal turn-around time is significantly increased primarily due to the nature of single-ended operations – not the presence of the ramp.

Performance: Because of the net decrease in weight of 39.5 to 54.5 tonnes, combined with a more favorable aft trim in the full load condition, there will be a significant increase in performance. Light ship performance should be close to what it is presently.

Cost Estimate: The total cost of this configuration is estimated to be between $0.8M and $1.2M (US). Costs are based on the assumption of the work being performed by a North American shipyard.



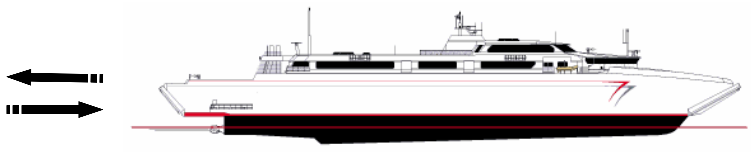
Upper Deck – 96 Vehicles



Lower Deck – 127 Vehicles

Figure 12: PacifiCat Single-Ended / Bow-To Loading Configuration – 223 Vehicles

THIRD CONFIGURATION - SINGLE-ENDED/STERN-TO LOADING



Arrangement and General Description: For operators which require single-ended operations with stern-to docking, one ramp is required in the aftbody. Such a configuration, with maximum load-out, is depicted in figure 13. Vehicles boarding the ship over the stern will drive up the ramp onto the upper deck, execute a u-turn, and park facing the stern. Backing and filling of vehicles will be necessary - much more so than in a drive-through configuration. Disembarking of vehicles is the reverse of loading, starting with the vehicles at the stern. These vehicles will drive down the forebody ramp and ashore, with the vehicles toward the bow backing up and then proceeding down the forebody ramp themselves.

Once the upper deck is loaded, the ramp is hoisted up until it is flush with the upper deck, clearing access to all lanes of the lower deck. Note that in this scheme the lower deck also requires u-turns and backing, which means that the loading of large vehicles such as trucks or coaches is more complicated. It is possible to have cars loading on the lower deck via the outermost lanes, while vehicles are moving to the upper deck, helping to somewhat minimize the impact upon terminal turn-around times.

Ship Impacts: Impacts to structure will be considerable, but impacts to lighting, electrical wireways, piping, and other ship distributive systems will be minimal as these should be readily re-routable in the affected area.

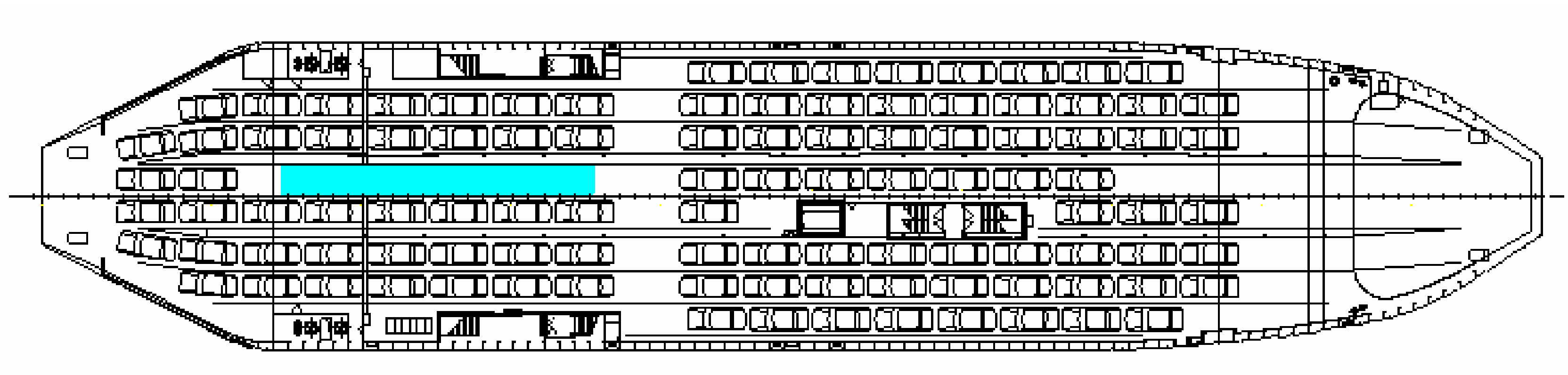
Vehicle Capacity: An impact of 5 AEQ will be felt on the total vehicle capacity on the upper deck for the ramp itself, plus approximately 6 more for space utilization inefficiencies on the lower deck due to having to turn vehicles. In addition, it will not be possible to fully load-out the lane that contains the ramp. An athwartships clear buffer zone on the upper deck of 1 car-length near amidships will be necessary to allow for adequate vehicle turning clearances when disembarking. Figure 13 shows that the upper deck will accommodate a maximum of 103 vehicles. The lower deck capacity, with impacts accounted for, is 127 vehicles. Therefore, in the proposed configuration, the ship will have a maximum capacity of 230 vehicles. A more realistic loading capacity for minimum turnaround times may be 10 fewer AEQ, or 220 vehicles total.

Weight: The net weight increase of the ramps themselves is minimal, if constructed out of aluminum. Each ramp should weigh approximately 2.5 tonnes, but will replace existing deck structure of about 2 tonnes. With structural reinforcement added in, the net increase in weight due to one ramp will be approximately 1 tonne. Note that the weight of 20 to 30 fewer AEQ, as compared with the vessel's current configuration, is about 30 to 45 tonnes. Thus the overall effect of adding the ramp is a weight reduction of approximately 29 to 44 tonnes. There will be a net decrease in weight aft, which will increase the trim forward slightly - a non-beneficial effect for vessel speed.

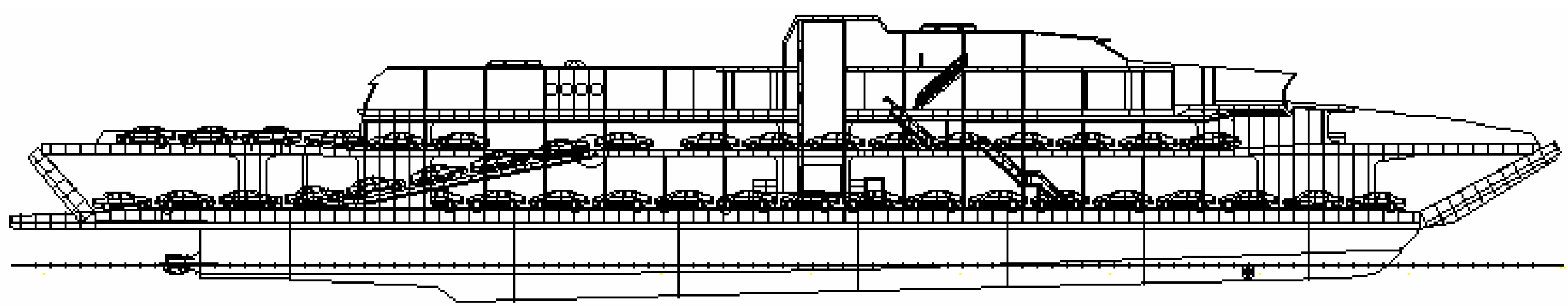
Terminal Turn-Around Time: Loading will proceed by flooding the upper deck first with cars, while loading the lower deck more slowly, followed by "fine tuning" on the lower deck in the latter stages. Some degree of simultaneous loading will still be possible since the ramp blocks only one loading lane of the lower deck. Added terminal time will be required for the u-turns and the backing and filling on both decks. Terminal turn-around time is significantly increased primarily due to the nature of single-ended operations – not the presence of the ramp.

Performance: Because the benefit of a net decrease in weight of 29 to 44 tonnes, may be offset somewhat by a less favorable forward trim in the full load condition, there will be a slight increase in performance. Light ship performance should be close to what it is presently.

Cost Estimate: The total cost of this configuration is estimated to be between $0.8M and $1.2M (US). Costs are based on the assumption of the work being performed by a North American shipyard.



Upper Deck – 103 Vehicles



Lower Deck – 127 Vehicles

Figure 13: PacifiCat Single-Ended / Stern-To Loading Configuration – 230 Vehicles

CONCLUSIONS AND RECOMMENDATIONS

Installing on-board ramps from an arrangement standpoint has been shown by this study to be feasible. Any ramp configuration must maintain the integrity of the superstructure’s fire protection systems, and allow for vessel operations in higher sea states. Hoistable end-hinged ramps, as opposed to fixed ramps are best for maintaining this integrity.

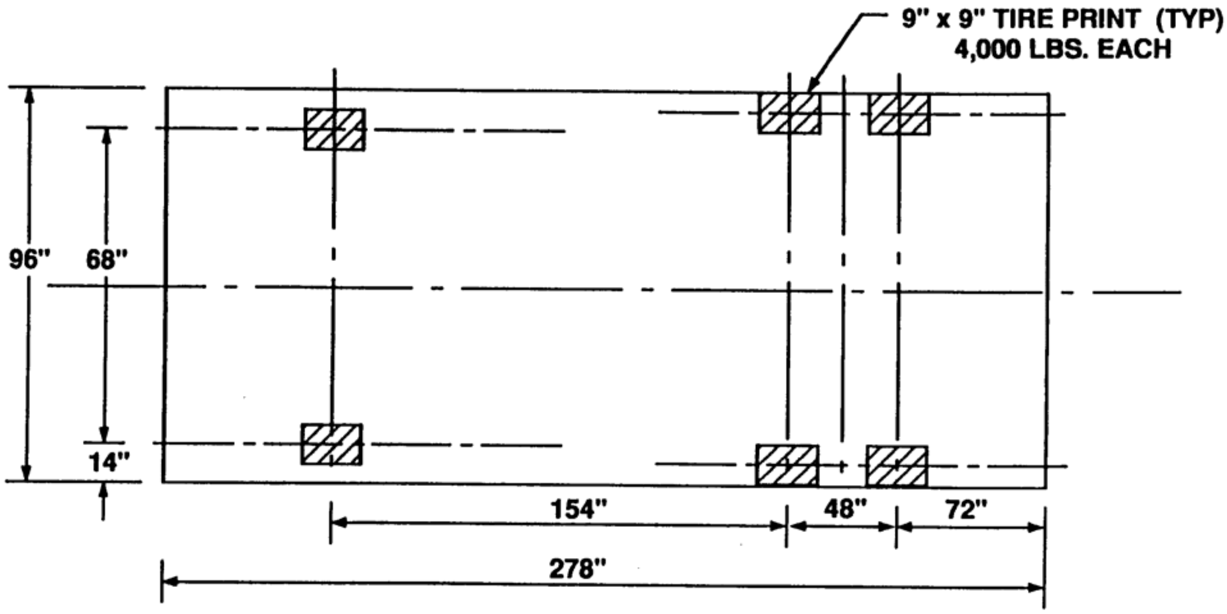
Clearly, any on-board ramp configuration has a significant cost, both in terms of initial dollar outlay and in terms of revenue impact due to car spaces lost. It will also increase the turn-around time of the ship. Whether such costs are acceptable will depend upon the owner's intended route and service.   Offsetting these impacts to some degree however, is the advantage of an increase in full load performance due to the decrease in full load displacement caused by the lost car spaces.

For minimum terminal turnaround times, double-ended operation with two hoistable ramps is the best option. Operational savings will clearly offset the increased capital costs of the conversion work.   For operators that choose single-ended operations with one hoistable ramp, bow-to loading has advantages over stern-to loading with regard to vessel maneuverability, optimum trim conditions, powering, and speed.

Installing on-board ramps from a strength standpoint has not yet been shown to be feasible by this study due to its limited scope. Further extensive 3D finite element analyses to properly design the compensating structure are recommended in concert with Incat Designs Sydney (the designers of the PacifiCats) and Det Norske Veritas (the classification society). Coordination with Robert Allan Ltd. (the systems designers) is recommended to re-route and re-design impacted ship systems.

APPENDIX C.

VEHICLE FOOTPRINT DATA



VEHICLE WEIGHT IS 24,000 LBS.
CENTER OF GRAVITY IS 48 INCHES ABOVE GROUND.

LIGHT CARGO TRUCK LOAD

Figure C-1

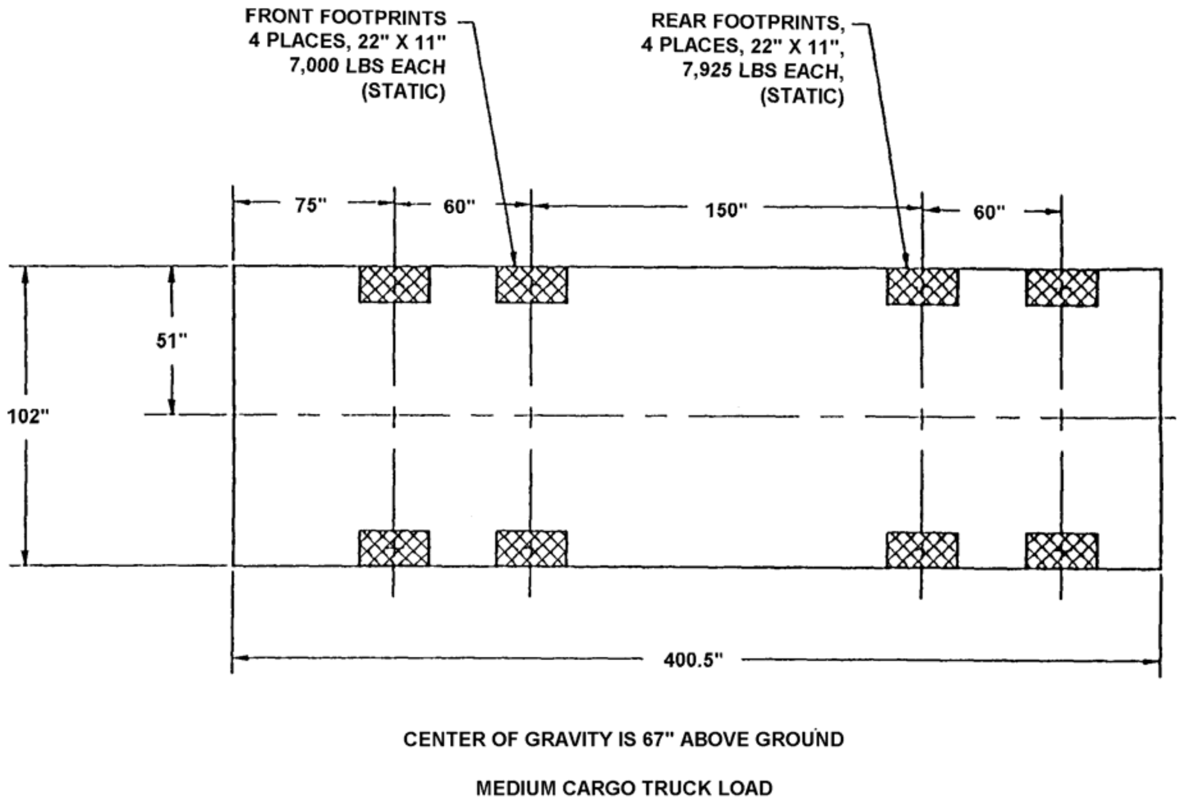
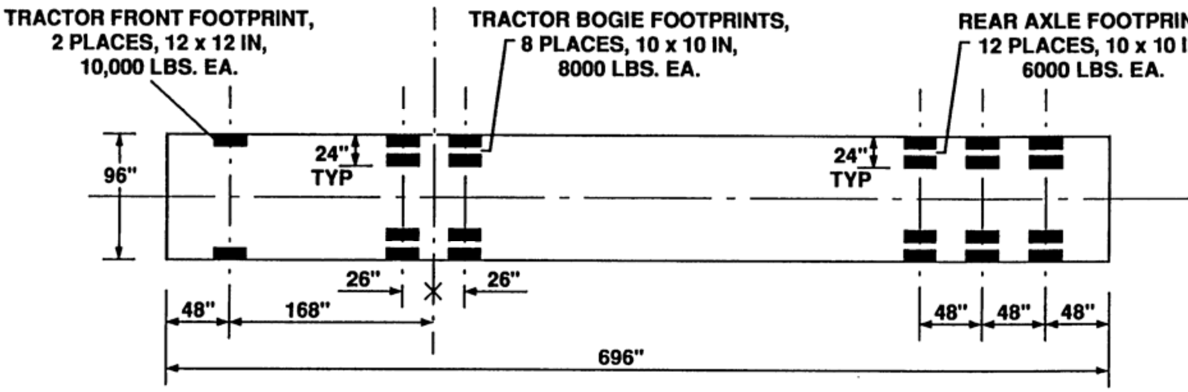

FRONT FOOTPRINTS
4 PLACES, 22" X 11"
7,000 LBS EACH
(STATIC)
REAR FOOTPRINTS,
4 PLACES, 22" X 11",
7,925 LBS EACH,
(STATIC)
75"
60"
150"
60"
51"
102"
400.5"
CENTER OF GRAVITY IS 67" ABOVE GROUND
MEDIUM CARGO TRUCK LOAD

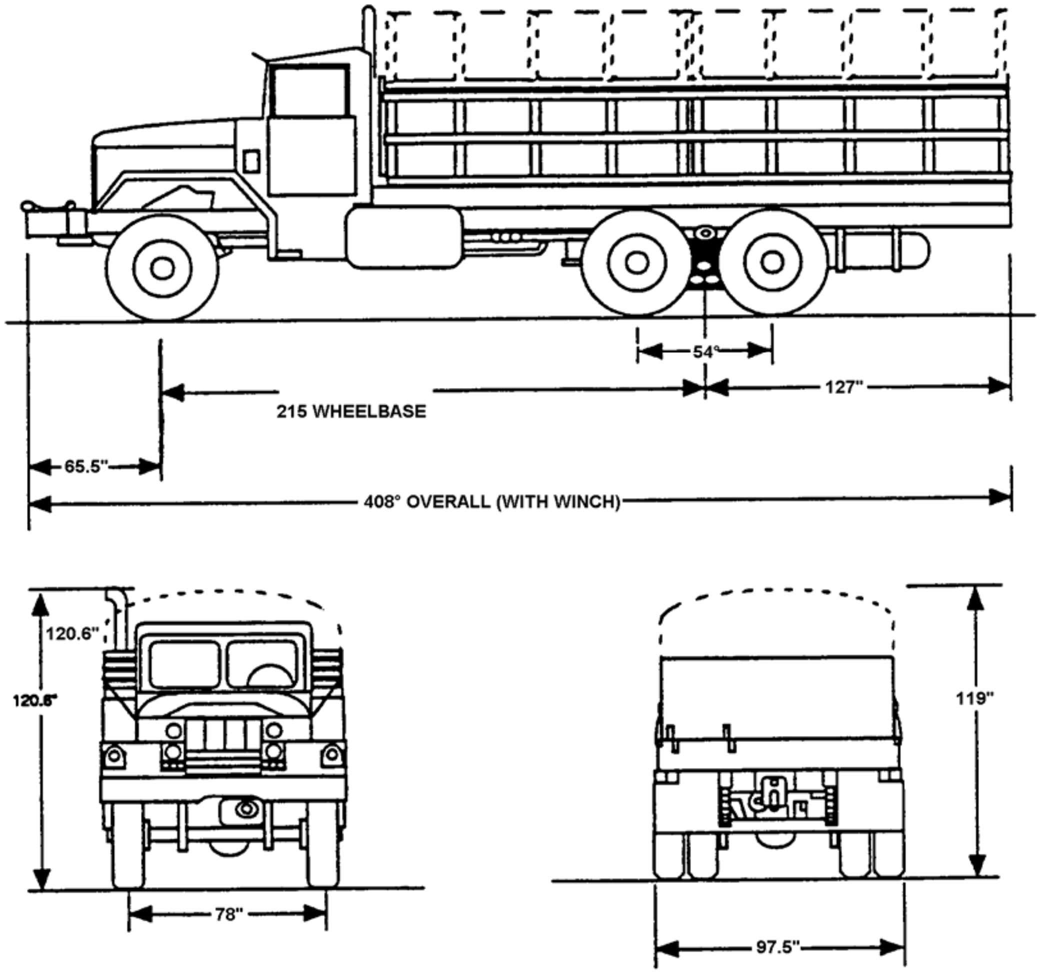
Figure C-2



VEHICLE WEIGHT IS 144,000 LBS.
CENTER OF GRAVITY IS 86 INCHES ABOVE GROUND.

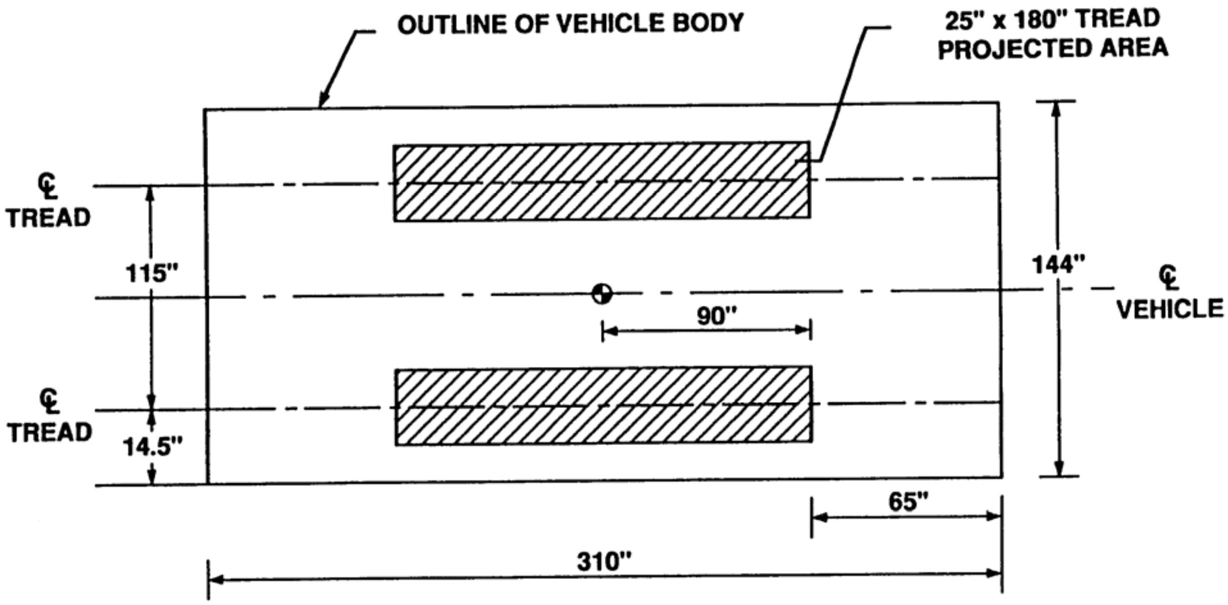
HEAVY CARGO TRUCK LOAD

Figure C-3



TRUCK, CARGO 5-TON, 6X6
M927A1

Figure C-4



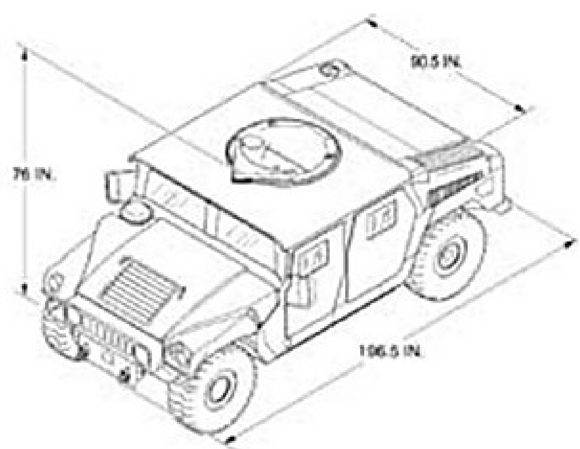
VEHICLE WEIGHT IS 160,000 LBS.
CENTER OF GRAVITY IS 53 INCHES ABOVE GROUND.

TRACKED VEHICLE LOAD

Figure C-5

M927, Truck Extended Bed, 7 ton 16R20 radial tires (6)

High Mobility Multipurpose Wheeled Vehicle (HMMWV) – Typical – 4 tires, 37 x 12.5R-16.5

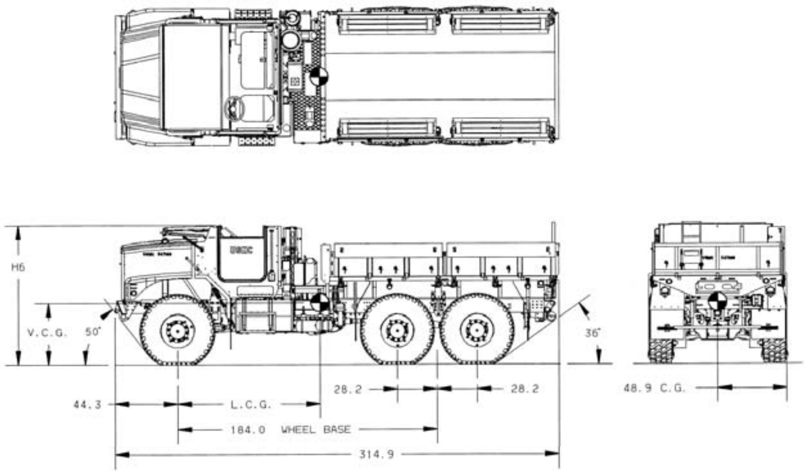


M101A1 – Trailer, 2 tires unknown size

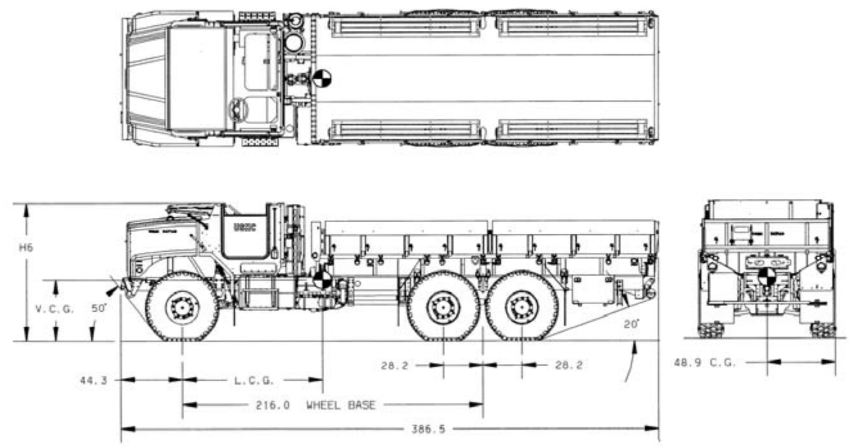
155MM Howitzer – 8 tires, large but size is unknown.

Note: This howitzer is reported to have a weight of 5765 pounds. The 9000 pound howitzer specified in the military vehicle payload for the conversion is only shown with two tires but additional anchoring/bracing locations. Additional work is underway to determine the appropriate characteristics of the howitzer to be included in the loadout.

Medium Tactical Vehicle Replacement (MTVR) – Replaces M923, M925, M927 and M928 – All tires are 16R20's, single radial tire on each axle



Standard Wheel Base



Extended Wheel Base

	M923A1			M927A1			MK23 Std Cargo			MK27 XLWB Cargo		
Loaded condition	Empty	Cross-country	* Highway	Empty	Cross-country	* Highway	Empty	Cross-country	* Highway	Empty	Cross-country	* Highway
Gross Weight (lbs)	21740	31740	42740	26135	36135	46135	27882	42650	58361	30195	44666	60466
#1 Axle Weight (lbs)	9275	9740	10000	N/A	11735	N/A	12640	12665	12433	13358	13342	13147
#2 Axle Weight (lbs)	6232	11000	16370	N/A	12200	N/A	8473	14951	21902	9161	15695	22808
#3 Axle Weight (lbs)	6232	11000	16370	N/A	12200	N/A	6763	14945	24026	7676	15692	24511
Maximum Height (in)	122.3	122.3	122.3	N/A	120.6	N/A	142.1	140	139.2	141.1	139.4	139
Reduced Height (in)	93.9	93.9	93.9	N/A	93.5	N/A	99.1	99	99.1	98.4	98.3	99.4
CG from Front (in)	N/A	N/A	N/A	N/A	N/A	N/A	98.8	129.3	145.8	119.1	151.4	169.8
CG from Ground (in)	N/A	N/A	N/A	N/A	N/A	N/A	39.7	53.8	60.6	38.8	52.8	59.6
Max Tire Press (psi)*	80	60	80	80	60	80	47	35	96	47	35	96

Notes: N/A (Not Available)

* Highway Loads for MTVR provided for information only. Since the only load that consistently gets the vehicle up to its full rated 15 tons is ammunition, it is very unlikely a vehicle would be mobile loaded at 15 tons aboard ship.

** Tire pressure would be set for the appropriate load and terrain. Aboard ship it would nomiN/Ally be set at cross country (or the lower sand, mud & snow) setting to match an over the beach load (7.1T) and the terrain expected on the beach. Highway psi

MK-48 Logistics Vehicle System (LVS), Front Power Unit

This photograph shows the front power unit with a trailer. The military vehicle payload includes the front power unit only. All four tires are 16R21 radials.

Rough Terrain Container Handler (RTCH) – Not included in the MEU Loadout for Conversion Vessel. Presented for information and to demonstrate large tire footprints that various vehicles may have.

PROJECT TECHNICAL COMMITTEE MEMBERS

The following persons were members of the committee that represented the Ship Structure Committee to the Contractor as resident subject matter experts. As such they performed technical review of the initial proposals to select the contractor, advised the contractor in cognizant matters pertaining to the contract of which the agencies were aware, performed technical review of the work in progress and edited the final report.

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Executive Director Ship Structure Committee:

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Welding Research Council	Dr. Martin Prager
Worchester Polytechnic Institute	Dr. Nick Dembsey
World Maritime Consulting, INC	VADM Gene Henn, USCG Ret.
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