DOES A V-SHAPE IMPROVE PERFORMANCE OF A MULTI-PURPOSE CONSTRUCTION VESSEL?

A comparison study of V-shape versus conventional hullform

Master thesis
Wouter Axel Oosterlaak

PUBLIC VERSION

DOES A V-SHAPE IMPROVE PERFORMANCE OF A MULTI-PURPOSE CONSTRUCTION VESSEL?

A comparison study of V-shape versus conventional hullform

Master Thesis

For the degree of Master of Science in Offshore engineering at Delft University of Technology

Conducted at Huisman Equipment BV, Schiedam

Wouter Axel Oosterlaak

[redacted]

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Faculty of Mechanical, Maritime and Materials Engineering (3mE)

Thesis committee

DELFT UNIVERSITY OF TECHNOLOGY

Prof.ir. J.J. Hopman
Ir. J.W. Frouws
Dr. ir. S.A. Miedema

HUISMAN EQUIPMENT BV

Ir. F. van Heerd

Executive summary

The report presents a comparison study of a multi-purpose construction with a V-shape hullform versus one with a conventional (U-shape) hullform. This research considers multi-purpose construction vessels which have the capability to perform heavy lift and pipe lay operations. These different functions typically result in contradicting design requirements. In general heavy lift crane operations require a large breadth to provide sufficient stability. However, during other operations (such as pipe lay or light lifts) the high stability results in relatively short natural heave and roll periods with high accelerations. In fact, this kind of operations would benefit from a reduced breadth. In an attempt to improve the operational performance of these type of vessels, heavy lifting equipment specialist Huisman Equipment BV has proposed a new hullform. This new concept is a mono hull with a V-shape hull, the vessel's breadth varies over the depth, see Figure A. By means of adjusting the draft, the stability and the motion behaviour can be adapted to different operational conditions

Figure A: Impression of a the V-shape concept in comparison with a conventional hull shape.

In this research the properties and operational performance of the multi-purpose construction vessel with a V-shape hull and one with a U-shape hull are compared. In order to perform this comparison, two vessel concepts have been designed and studied for the same operational requirements.

Prior to the design and phase, the potential of a V-shape hull with respect to a U-shape hull was studied. Based on this study, it is expected that the following operational aspects will benefit by a V-shape:

[redacted]

- [redacted].

Both vessel concepts are designed for the, same, requirements:

- Lift capacity up to 5000 metric tons at a radius of 34 meter, over the vessel side,
- Installation of a pipeline on the seabed up to 46 inch, by means of a S-lay system,
- Minimum transit speed of 15 knots.

No requirements were set for the vessel dimensions, this enable to develop the most optimal vessel design for each concept.

A parametric study was conduct on the V-shape mid-ship geometry, in order to support the design process. With the objective to determine a V-shape geometry which is featured with maximum roll natural period possible for each different operation type.

The vessel stability for the most common and critical load cases are calculated and evaluated with applied stability criteria. By means of the stability program Delftship. The still water resistance, motion behaviour, workability and the dynamic positing (DP) capability of two created concept designs were analysed using the following methods:

- <u>Still water resistance</u>
  The Holtrop & Mennen theory was used for both concept designs. The method was validated, for the U- and V-shape, against model tests results of reference vessels. The hull-resistance prediction based on the Holtrop & Mennen method shows sufficient similarity to the measured data coming from the model tests. However, the Holtrop & Mennen method results in an overestimation of the hull-resistance for the V-shape reference vessel. To account for this over estimation a correction was applied for still water resistance of the V-shape concept design.
- <u>Motion behaviour and workability</u>
  A motion analysis was performed to determine the workability of each concept, for several operational cases and three areas. Using diffraction software (AQWA), the linear RAO's are calculated and used as input data for the irregular wave calculations. The roll damping is estimated using the Ikeda prediction method. The Ikeda method and RAO's (from AQWA) were validated against model test results to determine applicability of these methods. It was concluded that both the Ikeda method and AQWA show sufficient similarity with the model tests results.
- <u>DP Capability</u>
  Both current, wind as well wave forces acting on the vessel during DP operations were considered. By means of comparing the forces, the differences in DP capability was estimated

The main conclusion drawn from this research is:

[redacted]

This follows from the still water resistance, DP capability, motion behaviour and the workability analysis on both concept designs. The newly designed V-shape hull geometry results in:

[redacted]

It is recommended that future research is focused on the following topic:

- A detailed comparison study on hull resistance between both concept designs,
- A detailed comparison study on the economic performance of both concept designs,
- A detailed DP performance analysis of a vessel with a V-shape hull,
- Investigate the, practical, vessel handling of the V-shape concept by interviewing marine contractors and crew.

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Preface

This thesis is my final result of my graduation project of the master 'Offshore and Dredging Engineering' at the Delft University of Technology. I would like to thank Ferdinand van Heerd for helping and supervising me on this project especially in the last phase of the project; spending his Sunday evening reviewing my work. Special thanks to Hans Hopman my supervising professor and Volkert.

I want to thank the entire Naval Department for making me feel welcome and making time to answer questions. Especially thanks for the hydrodynamic team for their patience when I occupied the licences of AQWA and Delftship. I would also like to thank all the students that were graduating alongside me at Huisman Equipment for helping me with the practical matters and support. But mainly for giving me a great time at the Tackebar and during coffee breaks at "our" own coffee machine.

Besides I like to thanks the more direct guidance of Ancella and friends for their support and relaxing moments during the last year. Last but not least I would like to thank my parents for giving me guidance throughout my life.

Wouter Axel Oosterlaak

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Glossary

acronyms

A_{pp}	Aft perpendicular
A&R	Abandoned and Recovery operation
BL	Base Line
BM	Distance: centre of buoyancy to meta centre
CAPEX	Capital Expanse
CB	Block coefficient
COB	Centre Of Buoyancy
COG	Centre Of Gravity
CFD	Computational Fluid Dynamics
CL	Centre Line
DNV	Det Norske Veritas
DP	Dynamic Positioning
FIDD	Frequency Independent Damping
GM	Distance: centre of gravity to meta centre
GZ	The righting lever
KB	Height centre of buoyancy above base
KG	Height centre of gravity above base
Meta centre (M)	Intersection point of: 2 successive line of buoyancy with very small increased of angle of inclination
MPM	Most Probable Maximum
NDT	Non Destructive Testing
IMO	International Maritime Organization
JONSWAP	Joint North Sea Wave Observation Project
Lpp	Length between perpendiculars
LCG	Longitude Centre of Gravity
LSW	Light Ship Weight
OMC	Offshore Mast Crane
OPEX	Operation Expense
PS	Port Side
RAO	Response Amplitude Operator
SP	Starboard
SWL	Safe Working Load
TCG	Transversal Centre of Gravity
VCG	Vertical Centre of Gravity

Table of Symbols

Latin Letters	Name	Unit
$\boldsymbol{a}$	Added mass	kg
$\boldsymbol{A_{wl}}$	Waterline surface	m^2
$\boldsymbol{A_{s\ bw}}$	Projected side area of a vessel	m^2
$\boldsymbol{A_{1/3}}$	Significant response amplitude	m
B	Overall breadth of a vessel	m
$\boldsymbol{C_f}$	Friction coefficient	-
$\boldsymbol{C_y}$	Current coefficient in y-direction	-
$\boldsymbol{c_{nn}}$	Spring term	N/m
$\boldsymbol{F_n}$	Froude number	-
$\boldsymbol{F}$	Force	N
$\boldsymbol{g}$	Gravitational acceleration	m/s^2
$\boldsymbol{k_{nn}}$	Radius of gyration	m
L	Overall vessel length	m
$\boldsymbol{H_{sig}}$	Significant wave height	m
$\boldsymbol{I_{xx}}$	Mass moment of inertia, with respect to the x-axis	m^4
$\boldsymbol{I_{yy}}$	Mass moment of inertia, with respect to the y-axis	m^4
$\boldsymbol{I_{zz}}$	Mass moment of inertia, with respect to the y-axis	m^4
$\boldsymbol{m}$	Mass	Kg
$\boldsymbol{M}$	Moment	Nm
$\boldsymbol{T_p}$	Wave peak period	s
$\boldsymbol{T_{roll}}$	Natural roll period	s
$\boldsymbol{\rho}$	Density	Kg/m^3
$\boldsymbol{R_f}$	Friction resistance	N
$\boldsymbol{v}$	Damping coefficient	$[Nm\frac{rad}{sec}]$
$\boldsymbol{V}$	Ship velocity	Kn
$\boldsymbol{S}$	Wetted surface	m^2
$\boldsymbol{U}$	Mean current speed	m/s
$\boldsymbol{\ddot{z}}$	Heave acceleration	m/s^2
$\boldsymbol{\dot{z}}$	Heave velocity	m/s
$\boldsymbol{z}$	Heave motion	m

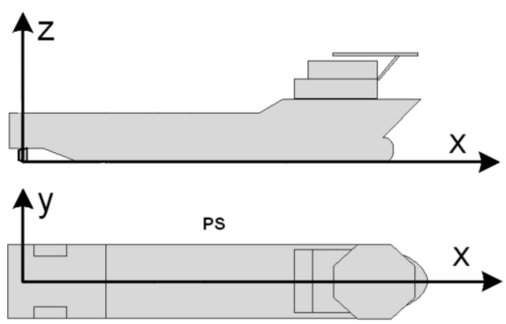
Greek Letters

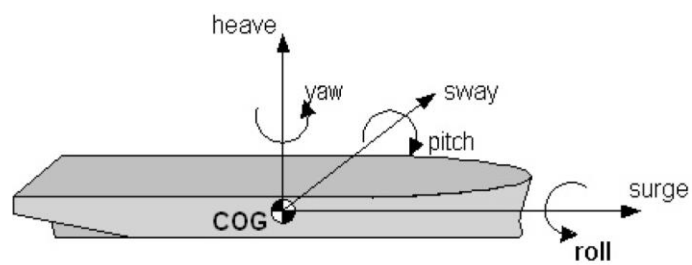
$\boldsymbol{\alpha}$	Side shell slope	deg
$\boldsymbol{\rho}$	Density	Kg/m^3
$\boldsymbol{\nabla}$	Displacement of volumr	m^3
$\boldsymbol{\omega}$	Wave frequency	rad/s

$\ddot{\varnothing}$	Roll acceleration	rad/s^2
$\dot{\varnothing}$	Roll velocity	rad/s
$\varnothing$	Roll motion	rad
$\ddot{\boldsymbol{\theta}}$	Pitch acceleration	rad/s^2
$\dot{\boldsymbol{\theta}}$	Pitch velocity	rad/s
$\boldsymbol{\theta}$	Pitch motion	rad

Convention

This thesis uses the coordinate system and motion conventions as given below





The origin of coordinate system:	
At App	X positive to bow
At CL	Y positive to PS
At BL	Z positive upwards

The vessel motions are defined as follows:
Three translation of the COG in axes direction
Three rotations about the axes

1 Introduction & Background

In this report a research is described with the goal to investigate if the V-shape hull concept improves the operational performance of multi-purpose construction vessels. The improvement is qualified with respect to multi-purpose construction vessels based on conventional hullforms. This study considers vessels which have the capability to perform heavy lift and pipe lay operations. These operations are incompatible, as heavy lift operations require a large vessel's breadth while pipe lay operations demand a relative small breadth to obtain good motion characteristics. This type of multi-purpose construction vessel do exist, these designs are dictated by the lifting stability resulting in relatively wide vessels. Hence, the vessels are featured by moderate motion behaviour during pipe lay operations. Due to this limitation, they can perform pipe lay operations only in a small weather window, i.e. small workability.

In an attempt to improve the operational performance of this type of vessels, heavy lifting equipment specialist Huisman Equipment B.V. has proposed a new hullform. The V-shape concept: a mono hull where the vessel's breadth varies over the depth. This makes it possible to change the motion behaviour by adjust the vessel's draft, for different type of operations. In deep draft the vessel has sufficient stability for lifting operations, while in light draft improved motion behaviour for pipe lay operations.

This research investigates the potential of multi-purpose construction vessels with a design based on the V-shape concept. To achieve this, two concept designs are created, a conventional and a V-shape design. Both designs are based on the same operational requirements and will be compared to each other. The designs will be compared by the following points: still water resistance, dynamic position performance capability, motion behaviour, and workability. This research and report is divided in four different phases, knowing to be:

- **Phase 1:** V-shape concept implementation
  Study the effect on an integral vessel design of a multi-purpose construction vessel with a V-shape hullform.
- **Phase 2:** Concept design process
  Two concept designs are made; one U-shape and one based on the V-shape concept
- **Phase 3:** Performance analysis
  Analyse the performance aspects which are expected to have significant deviation between the two concepts.
- **Phase 4:** Conclusions & recommendations
  This phase includes the conclusions of the conducted research and recommendations for further research. Furthermore, the applicability of the V-shape concept for vessels with other mission statements and operation profiles is described.

More on the problem statement, V-shape concept, research objective and method is described in the following sections of chapter 1.

1.1 Background

The exploration for offshore oil and gas resources began in the late 1800's. In 1896, an offshore well was drilled off the coast of California. These were drilled from piers generally 100 to 150 meter long, some producing from as deep as 200 meter. The 1938 discovery of the Creole field 2 kilometre off the coast of Louisiana in the gulf of Mexica marked the first venture into open, unprotected waters. The discovery well was drilled from a 20 by 90 meter drilling platform secured to a foundation of timber piles set in 4 meter of water. In the search for oil and gas in offshore areas the oil industry has continually extended and improved drilling and production technology. The early schemes utilising fixed structures tied to the sea bed which evolved into large bottom founded steel jacket production platforms in water depths up to 300 meter [ref. 1]. The driving necessities of cost reduction and the need to develop fields at ever increasing water depths has to led to other concepts including:

- Bottom founded platforms
- Floating production vessels
- Tension leg platforms
- Floating storage units
- Floating production storage and offloading units
- Spars

Over the last decade offshore oil production continued to increase globally in all areas. The wordwide offshore expenditure is massive. In order to keep up with the ever increasing worldwide energy demand new oil and gas reservoirs are developed. This, together with the recent rapid development of offshore wind energy market results in an increasing amount of offshore construction activities within the industry. Offshore operations come with significant challenges, many special vessels are developed to perform these operations. A general trend in the offshore industry is the application of the so called multi-purpose construction vessels. This type of vessels combines different functions. For instance, offshore crane vessels may also be outfitted with pipe laying systems. The goal is to improve their workability and versatility, however combining these different functions in one vessel design typically results in contradicting design requirements. Huisman equipment attempts to provide a solution for these contradicting requirements with the V-shape concept.

Workability and versatility in this research is defined as follows:

- <u>Workability</u>
  The capability of a vessel to being put in effective operation. Quantified by: the percentage of time that a vessel is able to conduct an operation (within its restrictions) in a particular area.
- <u>Versatility</u>
  The capability of a vessel to conduct different type of operations. For example, the capacity of a pipe lay vessel to install a large range of pipeline diameters.

1.2 Why the choice for a multi-purpose construction vessel?

The development of a vessel begins with a clear definition of the operations that it has to perform, i.e. its missions. Within the maritime cluster different vessel types are developed to perform a particular construction operation. These very specialized vessels are designed and optimized to perform complex and specific missions within the offshore oil & gas and renewable energy industry. Their special capability can be a hazard, as they operate in this niche market insufficient demand of work can occur, i.e. a long down-time. Potentially results in considerable financial losses. An often observed mitigation is that the offshore construction vessels are designed for two different missions, the so-called multi-purpose construction vessels. For example outfitting crane vessels with a pipe lay system is often observed in current offshore industry. This double functionality can improve the vessel's availability and therefore its financial potential. The current research considers multi-purpose construction vessels developed for the missions: lift and pipe lay operations.

Designing a vessel for multiple purposes can be difficult, as each mission has particular requirements on the vessel design, which can be contradicting. In more explicit words: workability and availability for pipe lay operations can negatively be affected by the requirements for lifting operations. This with respect to a vessel specially optimized for pipe lay operations. Before posing the research objectives, understanding needs to be gain about the requirements imposed by each mission type. The characteristics of lift and pipe lay operations is analysed below.

1.2.1 Lift operations

Offshore construction - such as involved in building a bottom founded production platform - requires a large number of crane operations in which the modules are lifted onto the facility structure. In the industry it is recognized that it is beneficial to reduce the number of modules to be connected up offshore to the smallest possible number [ref. 7]. As offshore construction operations are significantly more challenging than onshore. This results in demand of an extreme lifting capacity of floating cranes. Heavy lift units with a lift capacity up to 14,000 metric tons exist. The stability is one of the most critical design aspects for vessels with a large lifting capability.

The dimensions of crane vessels are dictated by the required high stability to support the maximum crane lift capacity. Crane vessels are typically characterized by a low L/B ratio, to fulfil with stability demand. Because this is an effective manner to obtain a high stability, with a limited vessel dimensions. Equation 1-1 describes how the intact metacentric height of a vessel is calculated. According this equation, increasing the vessel's breadth is an effective manner to improve the intact stability. As the transversal moment of inertia – and so BM, is a second order function of the vessels breadth.

Good motion characteristics of a crane vessel when performing lift operations is required to improve workability and for proper crane load handling. Due to moderate motion behaviour, the maximum allowable motion response (where it still safe is to operate) is reached at relatively low sea-states. Furthermore, as offshore crane vessels operate in the world-wide market, contractors have also placed emphasis on a significant transit speed [ref. 2].

Design requirements for crane vessels imposed by offshore crane with substantial lift are point out below:

- High stability and counter ballast: to compensate the crane overturning moment
- Good motion behaviour for a proper crane load handling and large as possible workability, more precise:
    - o Large workability is obtained, as at large sea-states the maximum motion responses imposed by the crane specification is researched.
    - o Low roll motion response: accurate horizontal hook motion
    - o Low heave and pitch motion response: vertical crane hook motions in waves
- Considerable transit speed
- Ability to keep accurate position
- Structural crane integration: large crane load

The initial metacentric height GM is calculated by

$$GM_{T,L} = BM + KB - KG \qquad [m] \qquad \{1\text{-}1\}$$

Where:

KB	:	$Centre\ of\ floatation$	[m]
KG	:	$Centre\ of\ gravity$	[m]
$BM = \frac{I_t}{\nabla}$	:	$I_t = 2 \cdot \int_0^L \frac{1}{3} y^3 dx$	[m]
y	:	$Distance\ from\ centre\ of\ surface\ dx$	[m]

1.2.2 Pipe lay operations

The installation of pipelines and flowlines and their connection to offshore facilities are considered to be as some of the most challenging operations within the industry. The amount of required engineering, work, cost and the size of different pipeline installation vessels are substantial, because of this the pipe lay industry has developed in to a special engineering discipline [ref. 3]. Especially since the offshore oil and gas developments move into (ultra) deep water depths, the greater the engineering challenges on the pipeline design and installation procedures become.

One of the problems occurring during offshore pipe-lay process is a controlled pipe handling. The limited capacity of the pipe to withstand bending stresses is a hazard for the pipe handling. It is required to keep a pipe line under constant significant axial tension during the operation. Otherwise the pipe weight, from the section between vessel and sea bottom, can cause pipe failure (buckling). The tension is delivered by mooring lines or by constant thrust provided by the Dynamic Positing (DP) system. This tension is transferred to the pipe line through a tensioner system. Furthermore, low motion response of the pipe lay vessel keep the bending and axial stress in the pipe at an acceptable level. Hence, a good motion characteristic in seas during operating is required, as it will increase the workability for pipe lay operations.

Pipe lay operations are sometimes performed near an offshore facility - during start-up and hand over procedures- this requires a reliable station keeping system. Therefore the highest DP classification system (DP 3 class) is typically mandatory for pipe lay vessels. Different methods to install pipelines have been developed by the offshore Oil and Gas industry, the most common methods considering S-lay, J-lay and Reel lay method are explained below.

S-lay method

The most common installation method in shallow waters is the S-lay method. This method is characterized by a horizontal production line and stinger. On board the vessel pipe joints (pipe segment of 40 feet or 12.2 meter in length) are assembled in a horizontal working plane, the so called firingline [ref. 3]. The stinger construction supports the pipe over bend and the departure angle of the pipe line. See Figure 1-1 for an impression of the system configuration.

The tensioners on the vessel pull on the pipeline, keeping the pipeline part to the seabed at tension. The reaction force to compensate the pipeline tension is delivered by thrusters or by pulling on the anchor lines. Below the main components of the firing line, to weld pipe joints on the pipeline, is briefly discussed:

- Transport systems
- Production systems

  The pipe joints are constructed on the pipeline in the firing-line, by the following main steps:
  - Bevelling
  - Pre-heating
  - Welding the joints
  - Non-destructive testing
  - Field joint coating
- Tensioners

  The tensioners provide the connection between pipeline and vessel, and is featured by:
  - Maintain tension in pipeline to avoid buckling
  - Compensate surge motion
  - Pay out during pipe pull
- Stinger

  The stinger is characterised by:
  - Stinger support over bend of pipe
  - Stinger determines departure angle
  - Stinger has adjustable and radius

The production rate of an S-lay system is typical higher than a J-lay method, the description of this system is given in the next section. This because the horizontal firing line consists of multiple working stations, this allows for larger crew numbers working on the pipeline. Due to the adjustable stinger pipelines can be installed for a large range of water depths. On the other hand, the pipeline can be subjected to significant axial tension, especially in very deep waters where the length of pipe suspended from the installation vessel to the sea floor becomes large. The result of this tension is that pipelines have some level of plastic deformation when passing over the stinger. This leads to additional requirements on strength strain and fatigue behaviour of the pipeline.

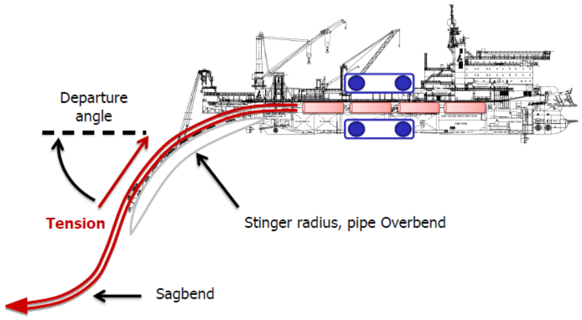


Figure 1-1: Schematic impression of the S-lay pipe lay method [ref. 4]

Using S-lay method in very deep waters is possible however it requires a long stinger and a large vessel to support it. Despite this, the industry considers S-lay methods suitable for (ultra) deep water, as it is attractive by its high production rate.

- <u>J-lay method</u>

  This method is developed in order to keep up with the development of deeper oil and gas fields. In the J-lay method the pipes are welded in vertical orientation and then vertically lowered toward the seabed. Figure 1-2 and 1-3 presents a typical J-lay configuration. The pipe line makes one bend, from the vertical departure to the seabed. This result typically in lower stresses during the pipe lay process in contrast to the S-lay method for similar water depth. Because no pipe overbend is exist in a J-lay configuration. Furthermore, lower pulling force is required to maintain the pipe in a J-lay shape than with the S-lay method. This makes the J-lay method especially suitable for deep and ultra-deep water. The pipeline fabrication method is in principle similar to that of the S-lay method but the firing line lay-out is vertically. Section 1.2.2, describes briefly the main firing-line components. Due to the vertical assembly configuration, a significantly smaller amount of working stations is located in the J-lay tower.

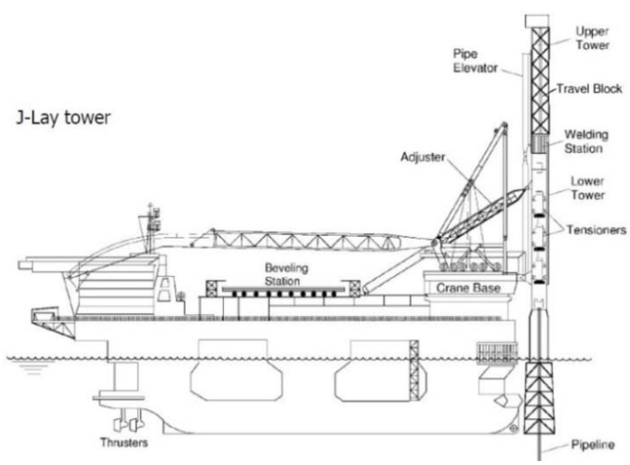


Figure 1-2: Impression of J-lay method

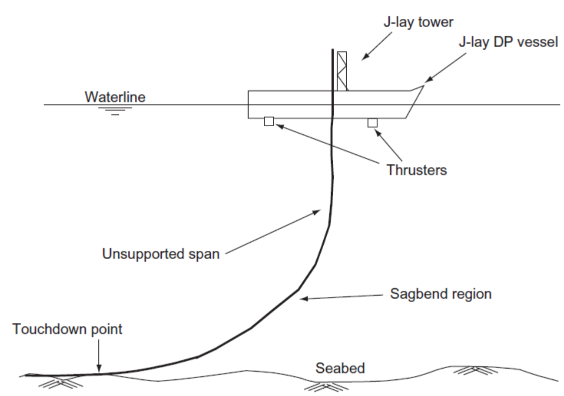


Figure 1-3: Schematic impression J-lay configuration

- <u>Reel lay method</u>

  By the use of a giant reel mounted on offshore vessel, pipelines can be installed. Pipelines are assembled at an onshore spool-base facility and spooled onto a vessels reel. Reeled pipelines can be installed significantly faster than the methods where pipe joint are used, since offshore welding of pipe joints is unnecessary. The installation of a reel pipe can be performed in S-lay or J-lay configuration depending on water depth. The installation method is briefly described below:

- o The pipe is reeled, straightened, de-ovalized and connected to the wire rope from the seabed pre-installed anchor. The axial tension is controlled by the tensioning system on the reel vessel. The vessel moves ahead while it slowly unreels the pipeline from the reel.

A drawback of this method: the limited maximum pipe line diameter, up to ± 18 inch. Furthermore: note that the production rate largely depends on distance, between spool-base and location where the pipeline will installed. If this distance is large the advantage of a high lay-rate will decrease rapidly.

Additional to pipe lay operation the vessel must be able to perform related operations during the construction of a marine pipeline. This applies for the three previously explained pipe lay methods and is explained below.

- <u>A&R operations</u>
  Stands for Abandoned and Recovery operations. In abandoned operation the pipeline is lowered to the seabed through a steelwire, if the pipe lay operations is finished or stopped due to severe weather or system failure. The pipeline is hoisted from the seabed in a recovery operation to continue.
- <u>Cable laying operations</u>
- <u>Retrieval of rigid pipe to connect flexible pipe</u>

By analyse the specific demands for a vessel to support pipe lay operations, the following design requirements can be noted:

- Good motion behaviour to gain a considerable workability as:
  - o A limited vessel response leads to small pipe displacement, velocity, acceleration and stresses
- Sufficient pulling force to maintain the pipe line in the correct - S or J - configuration
- Significant storage capacity: to facilitate storage of pipe joints (on deck or in the cargo hold)
- Accurate and reliable station keeping capabilities

1.2.3 Multi-purpose construction vessels

A multi-purpose construction vessel with a U-shape hullform has been designed and built to perform lift and pipe lay operations. However, these different functions typically result in contradict design requirements. Heavy lift crane operations require a large hull breadth in order to provide sufficient stability. However, when the crane is not in operation (light lifts and pipe lay operations) the large hull breadth (i.e. high stability) leads to a relatively short unfavourable natural roll period, resulting in high accelerations. Hence, this kind of operation would benefit from a reduced breadth. A more in depth analysis by considering the stability and motion behaviour of different operation conditions is given below:

- Heavy lift operation

  This is a lift operation where the maximum hoist capacity of the crane is used. To support this operation, the initial stability is maximized by filling the lowest ballast tanks to reduce the KG and so the GM value increases. The hook load acts at the crane tip and reduces KG which results in a relatively small GM. Which in turn causes in a large natural roll period as the spring term c_{44} is small, as given in equations 1-2 until 1-7. A natural roll period beyond typical range of encountered wave periods is beneficial for the vessel's workability. As, this effect reduces the vessel's response during heavy lift operations.
- Light lift operation

  During this operation a lift of approximately 5% of the maximum crane capacity is considered. The relatively small crane load reduces the KG only for a small part causing a large GM. Hence, a short natural roll period results in moderate roll motion behaviour. This can partly be reduced by adjusting the vessel's loading condition. By means of ballast located above KG the GM height can be reduced and the radius of gyration will increase. Hence, the natural roll period is enlarged by a small amount.
- Pipe lay operations

  During this operation no crane load is present resulting in a high GM. The loading condition can be adjusted as much as possible, but due to the large breadth this vessel will stay relative stiff. This leads to moderate heave, roll and pitch motions response which is undesirable for pipe lay operations, as it introduces extra pipeline loading. This leads to reduced workability, as the pipe lay operations only be performed in mild environmental conditions. Or conduct pipe lay operations can only in benign areas, or the installation of small pipe diameters, to keep the workability at an acceptable level.

$$Heave \quad : \quad (m + a_{33})\ddot{z} + b_{33}\dot{z} + c_{33}z = 0 \tag{1-2}$$
$$Roll \quad : \quad (I_{44} + a_{44})\ddot{\emptyset} + b_{44}\dot{\emptyset} + c_{44}\emptyset = 0 \tag{1-3}$$
$$Pitch \quad : \quad (I_{55} + a_{55})\ddot{\theta} + b_{55}\dot{\theta} + c_{55}\theta = 0 \tag{1-4}$$

Where:

m	:	Displacement
I_{nn}	:	Mass moment of inertia of the vessel
a_{nn}	:	Added mass
b_{nn}	:	Damping terms
c_{33}	:	Spring terms: $\rho g A_{wl}$
c_{44}	:	Spring terms: $\rho g \nabla GM$
c_{44}	:	Spring terms: $\rho g \nabla GM_L$

These equations can be rewritten as:

$$\ddot{z} + 2v_{33}\dot{z} + \omega_{03}^2 z = 0 \tag{1-5}$$
$$\ddot{\emptyset} + 2v_{44}\dot{\emptyset} + \omega_{04}^2 \varphi = 0 \tag{1-6}$$
$$\ddot{\theta} + 2v_{55}\dot{\theta} + \omega_{05}^2 \varphi = 0 \tag{1-7}$$

Where:

Damping coefficient : $2v_{33} = \dfrac{b}{m+a}$ $\quad 2v_{44} = \dfrac{b}{m+a}$ $\quad 2v_{55} = \dfrac{b}{m+a}$

Undamped natural Freq. : $\omega_{03}^2 = \dfrac{c_{44}}{m+a}$ $\quad \omega_{04}^2 = \dfrac{c_{44}}{I_{44}+I_{4a}}$ $\quad \omega_{05}^2 = \dfrac{c_{44}}{I_{55}+I_{5a}}$

To conclude, multi-purpose construction vessels are always a compromise between the required stability for the maximum lift operation, the good motion characteristics and a high transit speed. It is observed that the workability and availability of multi-purpose construction vessels for pipe lay operations is smaller with respect to vessels which are only designed for pipe lay operations. Furthermore, the low length over breadth ratio impedes the demand of a high transit speed.

1.3 Variable draft concept

In an attempt to improve the motion characteristics of multi-purpose construction vessels, heavy lifting equipment specialist Huisman Equipment BV has proposed a new hullform. This new concept: the V-shape is a mono hull where the vessel's breadth varies with its depth. The key feature is that through adjusting the draft the waterline surface will change significantly and so does the stability and motion characteristics. Due to this hullform, the workability and versatility for lift and pipe lay operations should improve. By means of changing the vessels draft the stability and motion behaviour can be adjusted to a particular operation condition. Figure 1-4 includes an illustration of this concept next to a conventional U-shape mono hull.

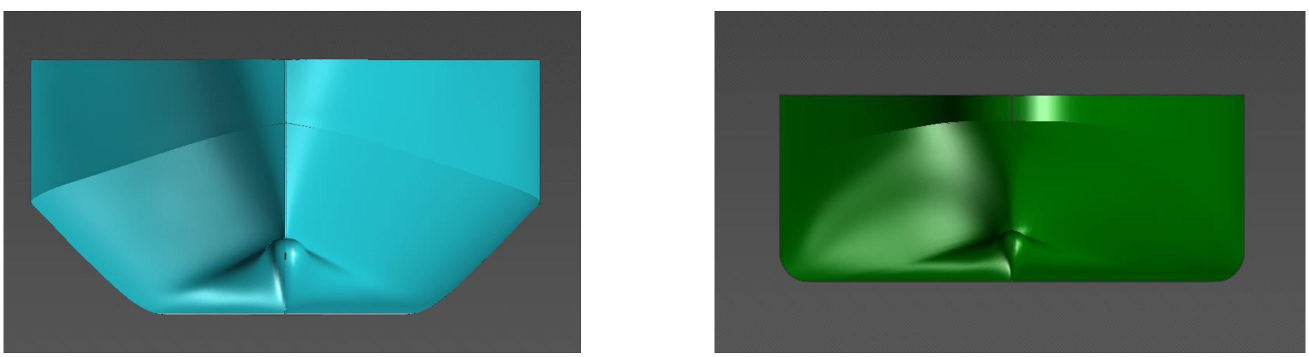


Figure 1-4: Impression of a the V-shape concept compared to a conventional hull shape

A model test program was conducted for Huisman Equipment in order to verify the V-shape concept [ref. 14]. This concept is developed with the objective to improve the motion behaviour for an offshore crane vessel. However, the properties and operational performance of an integral designed multi-purpose construction vessel with a V-shape hullform is not investigated.

1.4 Research question and objectives

In the previous sections the reason for this thesis is explained. It observed that combining pipe lay and lifting missions in one vessel design results in contradict design requirements. A V-shape hullform concept is developed in an attempt to provide a solution for these contradict design requirements. This has led to the following research question:

> **'Does the operational performance of a multi-purpose construction vessel improve with a design based on the V-shape concept, compared to a conventional design?'**

In order to answer the research question, the following three objectives are defined:

I. 'Create two multi-purpose construction vessel concept designs, one U-shape and one V-shape, both developed for the same operational requirements'
II. 'Compare the two developed vessels on their particulars and operational performances, for several operational conditions'
III. 'Investigate whether the V-shape concept is applicable for other vessel types'

1.5 Method of research

In order to achieve the research objectives, the method of research is divided in four phases:

I. Study the effect of implementing a multi-purpose construction vessel with a V-shape concept on the integral vessel design.
II. Two concept designs are made; one U-shape and one based on the V-shape concept.
III. Analysis the performance aspects which appears to have significant differences between the two concepts. The calculated characteristics are compared which each other.
IV. Conclusions are drawn based on this research. Furthermore, recommendations for future research are posed.

In order to make a qualitative and fair comparison between the two concepts, one set of operational requirements is set. Both concept designs must fulfil these (minimum) requirements. The two created concept designs and their operational performance are compared to each other. The vessel performance aspects which are expected to diverge significantly between U- and V-shape (according to phase 1) are analysed and compared which each other in phase three. Phase four use the obtained knowledge to answer the defined research question.

1.6 Limitations

To make the previously set objectives reachable some limitations have to be stated, which define the outlines of the shape of this research. The limitations of this thesis are given below:

- No detailed vessel designs were made, but two concept designs,
- Only mono hull vessels are considered in this research,
- The economic performance of the both created concepts is not evaluated.

PHASE I

ANALYSIS OF THE V-SHAPE CONCEPT

Prior to design phase and the performance analysis of the two concept designs, the potential of the V-shape hull was studied. This chapter studies the expected particulars of multi-purpose construction vessel with a V-shape hull compared to a U-shape hull. The following topics will be covered:

- General vessel design
- Stability properties
- Dynamic behaviour
- Vessel handling

This chapter use basic equations and reasoning. Furthermore, model tests results are made available by Huisman equipment of the V-shape concept [ref. 14] and of a reference multi-purpose construction vessel with a U-shape.

2.1 General vessel design

Vessel arrangement

The V-shape geometry has consequences for the inner vessel's arrangement. At lower decks the inner space will be smaller, as the vessel breadth here is smaller. The lack of available space results in the fact that equipment and other support systems need to be located at higher decks, see Figure 2-1.

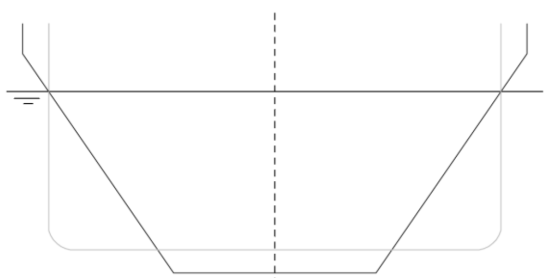


Figure 2-1: Mean section U-shape versus V-shape

The same holds for the tank arrangement, as at the lower decks reduced space is available to locate all the necessary tanks. As a consequence more tanks must be located at higher decks, which results in an higher centre of gravity. This is unfavourable for the stability, because it reduces GM. On the other hand, the smaller immersion rate requires less water ballast with respect to a U-shape.

Accommodation

Offshore operations require a large amount of crew, therefore a substantial sized superstructure is needed to accommodate them. However, it can be assumed that the crew size is independent of the hull hullform. Moreover, the available area to place a superstructure is comparable, as it is likely that the overall breadth and length of the vessels are comparable. This because section 1.2.1 stated that the breadth is dictated by the required high stability to support lift operations.

Deadweight and vessel dimensions

If one considers two designs, U- and V-shape, that both has to comply with the same amount of deadweight. In order to achieve this, the length, breadth or the draft (so the depth) of a V-shape must be larger than a U-shape vessel, to satisfy the deadweight requirement. The V-shape hull breadth on the waterline is likely to be equal to a U-shape vessel. As a larger breadth will reduce the aimed effect of a V-shape; slender waterline in light draft conditions. It is likely that a longer vessel length is the preferred design option, as the waterline is than small as possible for conditions with a light draft. Figure 2-1 presents the difference between the main sections.

Lightship weight

No significant difference in lightship weight is expected. As limited difference in vessel arrangement, as mentioned above, is expected for construction vessel based on a V-shape concept. Besides, the same equipment must be accommodated, as the V-shape vessel will be designed for the same operational requirements. No significant difference between the construction steel weight of a V-shape hull is expected with respect to a U-shape. If the vessel length and depth not diverge too much, the longitude bending moment and shear forces will be in the same order of magnitude. Resulting in a comparable construction and therefore weight.

Still water hull resistance

The still water hull resistance of multi-purpose construction vessels is an important design aspect, as was described in section 1.2.1. The V-shape concept have a relative small waterline and likely a smaller wetted surface (V-shape approximate a circle) in transit condition. This could probably leads to a lower hull resistance. This statement is based on the following:

The total hull resistance consists of the following main parts:

- Friction resistance [N]
- Wave making resistance [N]

Below each hull resistance aspect is discussed:

- Friction resistance

  The wetted surface, as mentioned above, is expected to be smaller so will the friction resistance. Equation 2-1 presents that the friction resistance depends linear on the wetted surface. Note that this gain mainly occurs at low and intermediate Froude numbers. Because the resistance at higher Froude numbers is dominate by the wave making resistance.

$$R_f = \frac{1}{2}\rho V^2 \cdot C_f \cdot S \qquad \{2\text{-}1\}$$

Where:

R_f	:	Friction resistance	[N]
ρ	:	Density of water	[kg/m³]
V	:	Sailing speed	[m/s]
C_f	:	Friction coefficient	[-]
S	:	Wetted surface	[m²]

- Wave making resistance

It is likely that the a more narrow waterline of the V-shape hull will reduce the wave making resistance. If one considers a catamaran that sails with a high Froude number; in general the vessel produces relatively small waves without it is planing. The extreme low L/B ratio of catamarans cause the low wave making resistance. With this extreme example in mind it is expected the V-shape hull (in shallow draft) has a lower wave making resistance. This leads mainly to a lower resistance at forward speeds at higher Froude numbers, as the wave making resistance is than governing.

The still water resistance was studied during model tests of the reference U-shape construction vessel and of Huisman V-shape concept. Figure 2-2 presents still water hull resistance of measured data coming from model tests, both during transit operation. These vessels are both designed to accommodate a 5000 metric tons crane, see appendix A for their particulars. Figure 2-2 presents a significant lower resistance for the Huisman V-shape concept. Note that these vessels are not exact designed for the same missions or operation profile. Although, the data presented in Figure 2-2 confirms the expected lower hull resistance of a V-shape hull, as previously described in this section.

Confidential

Figure 2-2: Still water resistance during transit

Financial performance

The financial performance is of great importance for an vessel owner. Even though a vessel has excellent technical specifications but moderate financial performances, it will never be build. The expected strengths and weakness of the financial characteristics for a V-shape is described below:

- <u>Capex</u>
  It is expected that the CApital EXpense (CAPEX) of conventional and V-shape construction vessels is comparable. Because the required amount of construction steel weight is expected comparable. Moreover, the CAPEX of the mission equipment (crane pipe lay systems) is in the basic independent of the hull geometry.
- <u>Running costs</u>
  The running costs are those costs required to have the crew on board and ready to sail.[ref. 5] The main components: crew expenses, insurance costs, maintenance, docking and repair costs. It can be expected that these costs are comparable, as this depends more on the size, mission type and operation profile than the hullform.

- Voyage costs

  This includes: mobilisation, harbour and fuel costs for a particular voyage and dependents on the operation profile. As mentioned above, the hull resistance is potential lower of a V-shape vessel. Therefore; lower fuel consumption. Furthermore, the workability for light lift and pipe lay operations is expected to be larger. Hence, a lower voyage costs for a V-shape concept can be expected.

By summing up the expected financial characteristics of the vessel with a V-shape, it can be concluded that it likely will have a positive effect. A financial analysis of this vessel type is here not conduct in this research. In order to make a qualitative estimation a financial analysis must be performed on multiple operation profiles.

2.2 Intact stability

This section analysis the expected stability properties of multi-purpose construction vessel based on the V-shape concept in comparison to a U-shape hullform. The stability properties for two “extreme” stability conditions; heavy lifting (deep draft) and transit (shallow draft) operations are considered. In shallow draft condition, the V-shape is results in a smaller breadth on the waterline than a U-shape. While for heavy lift operation, the breadth on the waterline is likely to be similar in magnitude. Per operation condition the deadweight for each hullform concept is considered to be identical.

Initial stability

By means of considering the initial GM height, according to equation 1-1 given in chapter 1, the (expected) stability properties for a V-shape vessel is evaluated:

- Heavy lifting operation

  During heavy lifting operations, the KM is made as large as possible to compensate for the high KG, introduced by the heavy crane load. Water ballast is needed to enhance the initial stability and to compensate the overturning moments generated by the load suspended from crane (i.e. to keep the vessel upright). By means of water ballast the draft is increased such that the V-shape waterline breadth is maximized i.e. equal to the overall breadth. Due to this the transversal surface moment of inertia of the waterplane area and therefore the BM is comparable to a U-shape vessel with similar particulars. The KB height is larger, as the draft of V-shape is expected to be larger, to gain sufficient buoyancy. Besides, the centre of buoyancy of the “triangle” shape underwater ship is higher than the “box” shape of a U-shape hullform. The KG height of a V-shape vessel is assumed to be comparable. Hence, the GM is expected to be higher than a U-shape vessel, with comparable dimensions and deadweight.

  Although, heavy lifting operations are the most critical conditions regarding the intact stability. Therefore, multi-purpose construction vessels are in general optimized for this condition. Due to this, it is likely that each concept is designed for the minimum required GM value during lift operations, according to class societies or/and specifications of crane. Hence, the GM value of U – and V-shape vessels are assumed comparable during heavy lifting operations.

- Transit operation

In this condition the deadweight of the vessel is assumed small and the amount of water ballast limited to reduce the draft, resulting in narrow waterline for the V-shape concept. Due to this, the transverse surface moment of inertia of the waterplane area is significantly reduced, with respect to the deep draft condition. While the breadth on the waterline for a U-shape hullform is the same as in deep draft condition. Hence, the BM height of a U-shape hullform will be significantly larger as V-shape hullform during transit operations. The KG and KB of multi-purpose construction vessel with a design based on the V-shape concept will be little higher. This due to the geometry of the V-shape hullform. With this in mind, it is expected that the GM value of a V-shape is significantly lower than for a vessel with a U-shape, during transit.

Range of stability

The range of stability is qualified by the righting lever over the heeling angle, the so called GZ curve. For small heeling angles – up to circa 5° - the GZ is calculated according to equation 2-2, see also Figure 2-3.

$$GZ = GM \cdot \sin\varphi \qquad \{2\text{-}2\}$$

As for larger heeling angles the waterline area and centre of buoyancy changes significantly, therefore equation 2-2 becomes inaccurate. So, the corrected GM must be calculated, the so called GN. This is determined by, calculating the BM and KB for every heeling angle separately. The expected range of stability is evaluated for a multi-purpose construction vessel with a design based on the V-shape concept with respected to one based on a U-shape. The range of stability during heavy lifting and transit operation is described next.

- Heavy lifting operation

  In this condition the waterplane area is maximum i.e. the draft is almost identical to the height of the node in the hull (the point where the inclined side shell goes to vertical). This results in a larger KM height, when the vessel is in upright condition. If the heel angle increases the breadth on the waterline will significantly reduce. While for a U-shape the breadth on the waterline will increase with its heel angle. Hence, the range of stability for a V-shape is expected to be significantly smaller.

  As a consequence the V-shape hullform has a reduced capability to fulfil criteria regarding the range of stability. For example, the stability criteria of class societies on the effect of accidental drop of crane load, see appendix E for the criteria. By means of a larger breadth or depth the range of stability can increased. Or the V-shape should be designed such that the draft during heavy lift conditions is lower as the height of the node. The range of stability will increase, as the waterplane area increase with the heel angle.
- Transit operation

  Due to reduced breadth on the waterline, the range of stability is smaller with respect to a U-shape vessel. However, in this condition the waterplane area increases with the heel angle also for a V-shape hull. A smaller range of stability is expected to have no negative consequence for the capability to comply with stability criteria, as stability is still relative high.

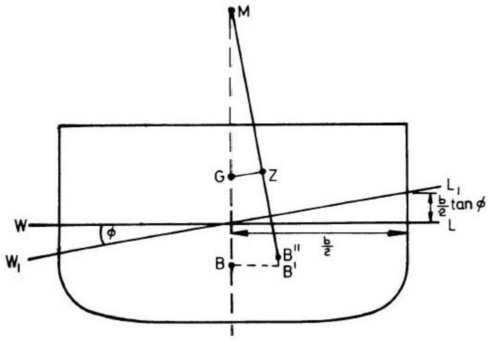


Figure 2-3: Geometric properties of hull form

Damaged stability

In order to assure the safety of personal and vessel itself, the stability of damaged vessels needs to be checked. The IMO made an effort to introduce Quantitative Risk Assessment and risk assessment, the so called probabilistic approach. These risk-based regulations consider a certain amount of damage submitted to the vessel and the probability of that damage occurring times the probability of loss of stability. This is known by the following equation:

$$A = p \cdot s \qquad \{2\text{-}3\}$$

Where:

p : Is the probability of the extent of flooding
s : Is the probability of surviving the flooding

$$^{i}A = \sum_{i=1}^{I} p_i \cdot s_i \qquad \{2\text{-}4\}$$

Where:

i	:	Represents the damaged compartments under consideration within the watertight subdivision of the ship. The subdivision is viewed in the longitudinal direction, starting with the aft most compartment
I	:	Set of all feasible flooding scenarios comparing a single compartment of adjacent compartments
p_i	:	Represents the probability that on the compartment *i* under consideration will be flooded, disregarding any horizontal subdivision, but taking transverse subdivision into account, longitudinal subdivision within the zone will result in additional flooding scenarios, each with its own probability of occurrence
s_i	:	Represents the probability of survival after flooding of the considered compartment

Damage stability is considered sufficient, if the attained subdivision index *A* is not less than the required index *R*:

$$A \geq R$$

Avoiding a small index A is sufficiently achieved by increasing the amount of transversal compartments. As small flooded compartment consequence on the overall stability is smaller over larger compartments. Hence the probability of survival after flooding of the compartment increases, with this index A.

It can be expected that the transversal compartment arrangement typical not differ for a V-shape concept. Therefore, capability for a V-shape to meet the damage stability regulations is comparable to U-shape vessel.

2.3 Dynamic behaviour of a V-shape hull

This section describes the expected roll damping, motion characteristics, hull slamming and the dynamic positioning capability of multi-purpose construction vessel with a V-shape hullform.

Roll Damping

The potential and viscous roll damping for a V-shape concept is described below.

- Potential roll damping
  The V-shape hull approaches the shape of a half circle. If one considers a circular cylinder, rotating about its centre, it will produce no waves. Hence, the potential roll damping of a circular cylinder is negligible small. With this in mind, the potential roll damping of the V-shape concept is expected to be small with respect to a U-shape. The potential damping of a U-shape is small as well, as roll potential damping is small in general.
- Viscous roll damping
  The viscous roll damping can be significant for rolling vessels, as the potential damping is almost zero. The viscous damping is made up of the following components: bilge keel, friction, lift (if forward speed is present) and eddy making damping [ref. 15.]. The bilge damping has the largest contribution. However the bilge keels are less effective for vessels with a V-shape hull. Since the distance between the rotation point and base of the bilge keel is substantial smaller in comparison to U-shape vessels, resulting in a lower rotation velocity of the bilge keels. See also Figure 2-4.

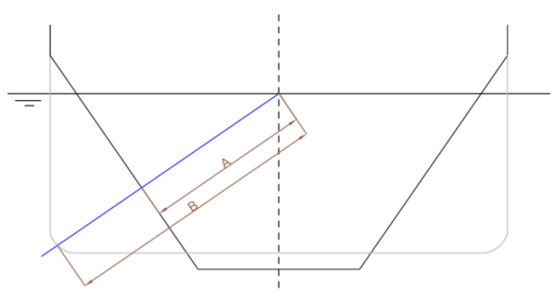


Figure: 2-4: The difference in radius of bilge keels

The V-shape concept allows to accommodate large bilge keels, without they extending the waterline. By means of equipping a V-shape with large bilge keels the roll damping can be significantly larger in comparison to a U-shape vessel. Despite that the potential roll damping is assumed to be smaller and the average roll velocity of the bilge keels is lower. Note that increased roll damping is preferable, as it reduced the roll response. On the other hand, larger bilge keels lead to increased wetted hull surface and therefore increased hull resistance.

Motion behaviour

The vessel motion with restoring terms (heave, roll and pitch) are only considered. The uncoupled motion equations are given by the equation 1-2 until 1-7, presented in chapter 1. A small GM results in a long natural roll period according to equation 1-3. Besides increasing the mass moment of inertia (I_{44}) will result in a longer natural roll period, as follows from equation 1-6. The longitudinal GM value and mass moment of inertia (I_{55}) has the same effect on the natural pitch period. Hence, these two parameters have considerable influence on the motion characteristics of a vessel and depend to a great extent on the loading conditions of the vessel, see section 1.2 for more information.

The typically motion characteristics of a V-shape hull during heavy lifting and transit operations in comparison with a U-shape vessel is discussed next.

- <u>Heavy lifting operation</u>
  During this operation the GM is relatively small and comparable to a U-shape vessel, see section 2.2. No significant difference of transversal mass moment of inertia is expected. Since the displacement and vessel dimensions are assumed comparable, see section 2-1. Based on this the natural roll period for a V-shape hull is assumed to be similar to a U-shape, see equation 1-3. However, the roll damping is larger, in case larger bilge keels are installed, as mentioned above. This improves the motion behaviour, as a large damping result in reduced roll response. The natural periods for heave and pitch are assumed comparable, since water plane area is expected to have the same length and breadth.
- <u>Transit operation</u>
  In transit operations tender roll motions are preferred as it will enhance crew comfort during voyages. Tender roll motions require a long natural roll period, small restoring force and large damping. During transit operations V-shape hull is expected to be featured by a lower GM value, with respect to a U-shape hull, see section 2.2. This obtained by reducing the draft, so a limited the amount of waterballast. While for a U-shape the roll behaviour is enhanced by filling high positioned water ballast tank. This will reduce the KG and increase the mass moment of inertia. However, to much water ballast in transit is unwanted, as it will have a negative influence on the hull resistance. It is expected that the V-shape natural roll period is longer as a U-shape, due to the significant smaller GM value.

  The typical smaller waterplane area results in reduced restoring term for heave and pitch and longer natural periods.

Linear calculation methods are used to determine the vessel response in waves, if experimental data is not available. The Response Amplitude Operators (RAO's) can be used as input for the (ir)regular wave response calculations. This method considers that the hydrostatic parameters, $GM_{L,T,}$ KB waterline area, etc., are constant with the vessel motions. However, in reality these parameters will change with vessel motion, i.e. it is actually non-linear. It is likely that larger non-linear effect for a V-shape hullform occurs. It is assumed that a waterline area during a heave motion for a V-shape will vary more than for a U-shape. The applicability of the linear RAO must be analysed.

Hull slamming

As well know from ships with considerable bow flare, a potential problem of sloping ships sides, a V-shape hull, is the occurrence of slamming impacts in steep incident waves. Typically the impulsive pressures are very sensitive to the angle between the slope of the water surface and the dead rise of the hull. If the slope of the face of the wave approaches the deadrise of the hull, slamming can be expected. [ref.7] Slamming is unwanted phenomena, as it influences the local pressures on the hull plating and a local damage can be the result. The impulse nature of the impact also causes internal vibrations which can contribute to structural fatigue in the ship [ref. 7]. Hull slamming hardly occurs for vessel with a U-shape hull.

During the model tests, wave impacts were observed at the sloped side shell in beam seas. This in very steep, and therefore rare, wave conditions [ref. 14]. It can be assumed that this is an unlikely condition as one would try to avoid working in this condition. In other headings tested, no slamming was observed. According the model test observations it was stated: the probability of significant hull slamming, in steep incident waves, is limited if the sloped ship sides are not smaller than (circa) [redacted] [ref. 14].

Dynamic positioning capability

Offshore construction vessels stay typically at position using a Dynamic Position (DP) system. The environmental loading - wave, current and wind forces – are balanced by the thruster forces. The environmental loading depends on the vessel particulars, vessel draft and hullform. Because the V-shape differs form a U-shape, the DP capability of this new hull concept needs to be considered. The current loading on a floating structure can be calculated from equation 2-5 [ref. 8].

$$F_{y\,current} = \frac{1}{2}\rho U^2 A_s C_y \qquad \{2\text{-}5\}$$

Where:

F_{cy}	:	Static current load in y-direction	[N]
ρ	:	Density	[kg/m^3]
U	:	Mean current speed	[m/s]
$A_{s\,bw}$	:	Projected side area of vessel	[m^2]
C_y	:	Non-dimensional static coefficients for forces	[-]

It can be expected that a nicely lateral V-shaped underwater ship has a lower static force coefficient, C_y, for current loading, see equation 2-5. This equation shows that this will result in a lower current loading and therefore less thruster force is required. On the other hand, if the projected side area is significantly larger (due to a large vessel length and/or draft) the current force might be larger. The same reasoning goes for the environmental loading in x-direction. Note that a more detailed analyse is required prove this statement.

2.4 Vessel handling

The practical vessel handling of a V-shape concept is briefly discussed below.

Side by side mooring and harbour moored

A potential problem of sloping vessels sides is inconvenient side by side mooring with other vessels. For example, supply vessels or a cargo barges delivering pipe joints. These vessels with lower freeboard can, possibly, 'slide' underneath node of the V-shape hull. Using large fenders or kind of steel framework filling the "gap", can possible provide a solution of this practical problem.

2.5 Conclusion on the potential of V-shape concept

The expected particulars and performance capability of a multi-purpose construction vessel with a V-shape hull is analysed in this section. By interpreting this analyse several points can be noted:

- It is expected that the V-shape concept improve the vessels capability to adapt its motion characteristics toward a particulars operation i.e. a wider operating envelope.
- The roll damping of V-shape concept is in general smaller. However, the V-shape concept can be accommodated with large bilge keels. Resulting in a larger roll damping and through this a improved roll motion behaviour.
- A lower still water resistance in transit operation for a V-shape is expected, allowing faster sailing or a reduced fuel consumption with respect to a U-shape hullform.
- During heavy lifting operations, i.e. deep draft, the V-shape has a reduced capability to fulfil criteria related to the range of stability with respect to U-shape. This may have a consequence for the vessel (larger) depth and/or breadth.
- The DP capability of a V-shape hull differs probably to a U-shape hull, due to a difference in environmental force acting on the vessel.
- When considering deadweight; larger vessel dimensions are required to obtain the same deadweight capacity compared to a U-shape.
- The non-linear behaviour of heave, roll and pitch motions seems to be larger, with respect to a U-shape.

Based on the above, it expected that the V-shape concept enhance the following performance aspects of multi-purpose construction vessels:

- Workability
  It is expected that the V-shape concept improve motion behaviour for light lift and pipe lay operations. Due this, multi-purpose construction vessels with a design based on the V-shape concept can perform these operations in higher sea-states. Hence, the workability will improve.
- Hull resistance
  A lower hull resistance is expected, allows higher transit speeds resulting in a larger versatility. The lower hull resistance can be used to obtain reduced fuel consumption.
- Dynamic positioning (DP)
  The potential lower current force of V-shape hull can leads to an lower fuel consumption when the vessel is in DP operation modes.

Phase three determine these performance aspects for the two concept designs, which are created in phase two. The U-shape and the V-shape are compared by means of evaluating these performance capabilities, in phase three. To determine if these statement, as mentioned above, are correct and it enables to answers the set research question.

PHASE II

CONCEPT DESIGN PROCESS

As in each elaboration of a vessel design its missions have to be defined. This chapter sets the missions, functions and operational profile. Furthermore, the selected mission equipment, design and operational requirements, imposed the mission equipment, is presented. These are applicable for both U- and V-shape concept designs.

This chapter unfolds as follows; first the missions are set and the mission equipment is selected. Based on the selected mission equipment, the vessels functions, certain design and operational boundaries are defined, Figure 3-1 visualize this. In the following sections each aspect of Figure 3-1 is described.

The operational requirements in this thesis is based on: the offshore multi-purpose construction vessel of Huisman Equipment BV client. The operational requirement of this client is used to gain qualitative and realistic requirements, as offshore contractors has a clear understanding about what missions, maximum sea state, operation profile etc. are realistic and suiting the current market demand. This makes it unnecessary to perform a market research for this thesis.

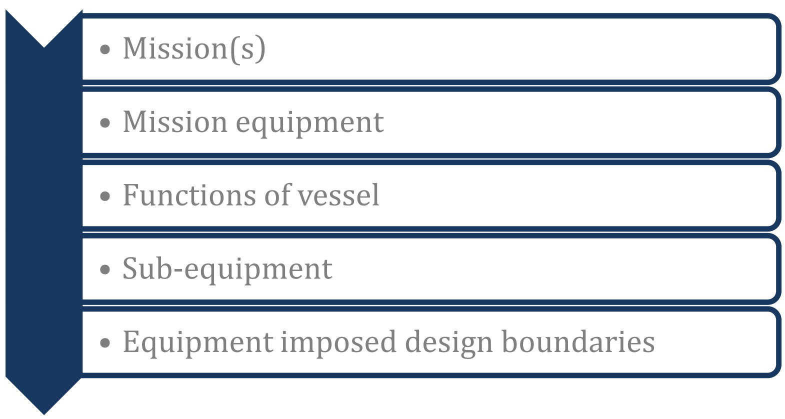


Figure 3-1: Structure design process

Not that no requirements regarding the vessel dimensions are. This enables to create the most optimum vessel design for each concept, without being constrained by any pre-defined dimensions.

3.1 Mission requirements

The offshore construction vessels are designed to fulfil the following missions:

- Installation of offshore facilities such as bottom founded support structures and topsides by means of crane lifts, for weights up to 5000 metric tons.
- Installation of pipelines, with a maximum pipe diameter of 46 inch, in S-lay configuration
- Long transit operations
- Transport operations

Pipe lay operations requires a vessel to perform additional missions:

- Abandonment and recovery operations

3.2 Mission equipment

In order to perform the mission(s) as mentioned above, the following mission equipment is needed:

- Accommodate of a fully revolving crane with a capacity of: 5000 metric tons at a 34 meter outreach.
- An S-lay pipe lay system should meet minimal the requirement

S-Lay system		Unit
Max. tensioners capacity	600	[mt]
Pipe diameter range	6 – 46	[inch]
Operating water depth	20 - 2000	[m]
Max. A&R winch capacity	600	[mt]
Layrate	1 pipe joint every 4 minutes	
Pipe storage capacity		
The S-lay system can handle single and double pipe joints		

Table 3-1: requirements of S-lay system

- Two pipe loading cranes on the main deck to lift pipe joints from supply vessels onto the main deck. This to allow for continuous production. Both cranes must have a swl of 100 metric tons.

The mission specific requirements as mentioned above will be discussed in further detail in section 3.5.

3.3 Functions

In order to successfully perform the missions, the vessel must have certain specific functions. The key vessel functions are given below:

- Provide a stable platform for the mission equipment
- Provide a platform for the mission equipment which can stay in position
- Provide necessary crew accommodation and facilities
- Provide storage capacity
- Provide energy supply to all the equipment

3.4 Additional requirements

In addition to the main missions the vessel will be designed for the following requirements:

- Transport capacity minimum 9,000 metric tons
- High transit speed minimum 15 knots
- Dynamic positioning class 3 system
- Compact hull design (able to sail through the ‘new’ Panama canal)
- Accommodation for a crew of 398 people
- Helicopter deck
- Ballast system

  This system will have two functions. Function 1: ballasting the vessel to the best suitable loading condition for each particular operation. Function 2: heel compensation system during lift operations. The system must allow slewing of the crane, with a slewing speed at maximum load of: 90 degree within 10 minutes.

- Minimum self-supporting requirements
  The vessel shall be fully self-supporting on consumables for the duration given in Table 3-2.

Endurance		
DP operation duration	40	[days]
Transit duration	60	[days]
Stores and provisions	50	[days]

Table 3-2: Requirement regarding endurance

3.5 Requirements imposed by mission equipment

The mission equipment imposes specific design requirements on the vessel design. The specification of the mission equipment sets the design and the operational boundary conditions, i.e. the maximum condition at which the vessel and its equipment may be operated. The most governing requirement related to the vessel design is given in this section. Additionally it must be noted that the vessel will be equipped with a Huisman offshore crane and S-lay system. Extensive technical data has been made available by Huisman Equipment for this thesis work.

3.5.1 Particulars and requirements of lifting equipment

The vessel will be equipped with a Huisman 5000 metric tons Offshore Mast Crane (OMC), the crane is capable of fully revolving with 5000 metric tons, and is characterized by the technical specifications [ref. 9] of Table 3-3.

Crane dimensions (in stowed position)		Unit
Total length to store the jib	[redacted]	[m]
Distance of begin pedestal to boomrest	[redacted]	[m]
Height of jib (above main deck)	[redacted]	[m]

Table 3-3: OMC dimensions

Interface information			Unit
Dimensions			
	Pivot height from deck level	[redacted]	[m]
	Square footprint pedestal	[redacted]	[m]
	Radius tailswing	[redacted]	[m]
	Demarcation level (above deck level)	[redacted]	
	Height crane mast (above deck level)	[redacted]	[m]
Weight			
	Total construction weight of crane, (including lower blocks and wires)	[redacted]	[mt]
	2 falls AH lowerblock	[redacted]	[mt]
	Total construction deep water equipment below deck (including traction winch, storage winch, and heave compensator, excluding wire rope)	[redacted]	[mt]
	Weight 6300 [m] wire rope	[redacted]	[mt]

Table 3-4: OMC interface information

The crane capacity depends on the static vessel heel and trim and vice versa, Table 3-5 presents the crane capacity with respect to trim and heel condition.

Capacity, revolving		Unit
SWL at 34m radius	[redacted]	[mt]
SWL at 40m radius	[redacted]	[mt]
SWL at 58m radius	[redacted]	[mt]
SWL at 73m radius	[redacted]	[mt]
5000 [mt]lift capacity given for the following condition (criteria's)		
Fduty	[redacted]	[-]
Fhoist	[redacted]	[-]
Offlead (including 1° static)	[redacted]	[deg]
Sidelead (including 0.5° static)	[redacted]	[deg]

Table 3-5: Crane capacity and vessel requirements

Where:

- Fduty : Dynamic hoist factor
- Fhoist : Static hoist factor
- Offlead : Cranes pulling angle (in longitudinal plane of the crane)
- Sidelead: Cranes pulling angle (in transversal plane of the crane)

3.5.2 Particulars and requirements of S-lay system

Additional to the lift equipment a 600 [mt] S-lay system will be installed on the vessel with the technical specifications of Table 3-6 [ref. 10].

Pipe specifications				Unit
Maximum pipe diameter	46.0	[inch]	11684	[mm]
Minimum pipe diameter	6	[inch]	15	[mm]
Nominal single joint length			12.2	[m]
Nominal double joint length			24.4	[m]
Maximum single joint weight			30	[mt]
Maximum double joint weight			40	[mt]

Table 3-6: Specification of pipe

Systems specifications			Unit
Tensioners	Maximum pipe tension (including dynamics)	600	[mt]
	Nominal pipe tension	500	[mt]
Primary A&R system	Maximum A&R pipe tension (including dynamics)	600	[mt]
	Nominal A&R pipe tension	500	[mt]
	Delivered wire length (Ø 135mm)	3900	[m]
Secondary A&R system	Maximum pipe tension (including dynamics)	200	[mt]
	Nominal; A&R pipe tension	160	[mt]
	Delivered wire length (Ø 76mm)	3000	[m]

Table 3-7: S-lay system specifications

Limiting motion criteria (pipe lay)	Max heel [deg]	Max trim [deg]	Max roll [deg]	Max pitch [deg]	Max heave [m]	Max heave [m/s]
Pipelay mode*	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	**
A&R mode	[redacted]	[redacted]	[redacted]	[redacted]		
Survival mode (transit)	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	
Survival mode (maximum)						

Table 3-8: Criteria's on vessels floating condition and motions of S-lay system

*These criteria are defined to restrain the maximum stresses in the pipe, to prevent pipe failure.

** To be determined (will be based on existing motion and pipe strain analyse of Seven Borealis)

The vessel will be equipped with two Huisman 100 metric tons pedestal mounted cranes, this crane type is designed specifically for larger outreach. They are characterized by the technical specifications given in Table 3-9.

Pipe loading cranes		Unit
SWL	100	[mt]
Pedalstal length (with respect to vessel's coordinate system)	4.3	[m]
Pedalstal breadth (with respect to vessel's coordinate system)	2.9	[m]

Table 3-9: specifications of pipe loading cranes

3.6 Operational profile

The vessels are designed for worldwide operability. Arctic regions are however not considered. The vessel design will be optimized for areas were the offshore industry is the most intense. Hence, the concept designs are designed for the following areas:

- West coast of Africa
- East coast of Brazil
- Gulf of Mexico

The vessel to be designed is developed for the following operational profile:

- 40% Lift operations
- 30% Transit
- 25% Pipe lay operations
- 5% Harbour condition

3.7 Environmental conditions

This section sets the minimum environmental conditions at which the vessels still can perform pipe lay and lift operations.

3.7.1 Environmental conditions during operations

Vessel in transit condition

Because of the vessel's worldwide operability, the environmental conditions are according to classification for unrestricted service.

Lifting operation

The minimum environmental condition at which the vessel should be able to lift 5000 metric tons is set in Table 3-10.

Item		Unit
Significant wave height ($H_{1/3}$)	1.0	[m]
Wave peak period (T_p)	3.0 - 14.0	[s]
Wind speed	10.0	[m/s]
Current speed surface	3.0	[kn]
Headings	180 135 90	[°]
Spectrum type	Pierson-Moskowitz	

Table 3-10: Environmental condition for 5000 [mt] lift operation

During lift operations the vessels angle and motion response may not exceed the crane criteria, as defined in Table 3-5.

Pipe lay operation

No environmental condition for pipe lay operations is defined. Because, stated in chapter 1, the multi-purpose construction vessel design is dictated by the high stability demand. Due this it is difficult to optimize a vessel for two different operation types i.e. lift and pipe lay operations. For this reason the environmental condition for lift operation is defined only. The limited motion characteristic for pipe lay operations will be calculated for each concept design. During the pipe lay operations the vessel motions may not exceed the limiting motion criteria as stated in Table 3-9. Based on these criteria the maximum environmental condition in which pipe lay operations can be performed will be calculated.

3.7.2 Environmental conditions for DP system

The construction vessel remains at position by means of a DP 3-class system, the maximal environmental conditions are different per case. The two following two DP cases are considered:

- Intact — Vessel standby condition (no operation is conducted), fully operating DP system
- DP 3 class — Vessel is able to remain its position - at set environmental conditions - when worst single failure of the DP system occurs

Intact

The vessel should remain its position with an intact DP system for the environmental condition per heading defined in Table 3-11 to 3-12.

All headings		Unit
H_{sig}	3.0	[m]
T_p	3.5-14.0	[s]
Spectrum	JONSWAP	[-]
Wind speed	15.4	[m/s]
Current speed	2.0	[m/s]
All environments are collinear		

Table 3-11: Environmental conditions for DP system

Head seas+/-30 degrees		Unit
H_{sig}	5.0	[m]
T_p	13.5	[s]
Spectrum	JONSWAP	[-]
Wind speed	17.0	[m/s]
Current speed	2.0	[m/s]
All environment is collinear		

Table 3-12: Environmental conditions for DP system

DP 3 class

The design requirement for the DP 3 class system is based on the maximum environmental condition at which the vessel still is capable to perform crane or pipe lay operations, according to the technical specifications. Hence, the governing environmental condition for this DP case will be determined in a later design process phase.

The missions, operational requirements and the vessel requirements imposed by the mission equipment are set, as previously described in chapter 3. This chapter describes the design methodology and the design goals applied on each concept design. This is used as guidance or help by making design decisions.

4.1 Used design philosophy and approach

The used design philosophy for both concepts is specified below:

- The aim of this study is to design a cost efficient vessel which has as large as possible workability for pipe lay and for lift operations. Where the vessel's key function is providing a platform to accommodate the mission equipment. Furthermore, the vessel capacity must be balanced with the capability of the mission equipment.

The objective of the design philosophy is to develop a cost efficient vessel; this can be achieved by a compact vessel design. In general a compact vessel leads to reduced CAPital EXpense (CAPEX), as less construction steel is needed for example. Besides it results in lower fuel consumption during transit and DP operations. Furthermore, better vessel handling and manoeuvrability can be expected. This led to a low OPeration EXpense (OPEX). The same reasoning goes for the objective to design a balanced vessel. If the vessel capability does not meet or exceeds the requirement imposed by the mission equipment, the potential of the equipment or vessel is not fully used i.e. unnecessary expansive equipment or vessel.

The same design philosophy and approach is applied for each concept design, to obtain an objective comparison as possible. Figure 4-1 visualizes the used design methodology.

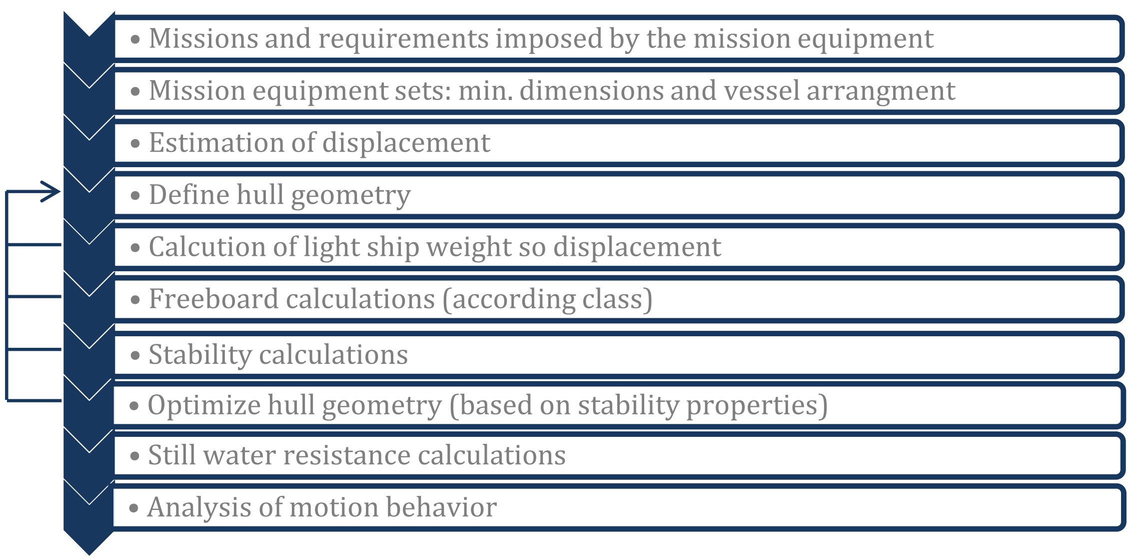


Figure 4-1: Used design methodology of both concepts designs

4.2 Implementation of equipment

In the design a strict separation will be made between both main missions (lifting and pipe lay), in order to develop a vessel where the missions affect each other as little as possible. Therefore; the firing line will be located underneath the main deck. Resulting in a clean main deck, this has a positive effect on offshore construction work besides it enables to transport large sized modules and provides a sufficient space to store the pipe joints. This with respect to a configuration where the S-lay system is located on the main deck, by means of a large duct. On the other hand, placing the firing line underneath the main deck leads to a larger depth thus extra steel weight i.e. a higher LSW.

4.3 Requirements and considerations for operations

This section describes the requirements on the motion behaviour imposed by the missions and environmental conditions. Each concept is designed for the areas West Africa, Brazil and the Gulf of Mexico, as stated in section 3.6. Using the global wave statistics [ref. 20] the environmental conditions is known. For a significant wave height of 1.0 meter the following range of wave peak periods was observed, per area:

- West coast of Africa
  3.0 - 13.5 seconds
- East coast of Brazil
  5.0 - 13.5 seconds
- Gulf of Mexico
  4.0 - 12.0 seconds

If encountered wave periods are around the resonant period of the vessel (natural period), the waves exerting on the vessel is able to produce large amplitude oscillations. The influences of the natural period on the vessel response is illustrated by means of an example, see Figure 4-2. A roll RAO of a vessel with a natural roll period equal and one with a natural period beyond the present wave periods is presented in Figure 4-2. The range of wave peak periods for a significant wave height up to 1.0 meter (in considered areas) is presented as shaded area in Figure 4-2.

Based on the example of Figure 4-2 and the observed wave conditions, as presented above, it is concluded that a natural periods for heave, roll and pitch beyond (at least) 14.0 seconds is preferred during lift, pipe lay and transit operations. If this condition is satisfied the vessel response due to wave will be significantly reduced. Hence, the workability for pipe lay and lift operations and the motion behaviour during transit will benefit by this effect. The workability will enhance because the maximum allowable vessel motions during pipe lay and lift operations, imposed by the mission equipment, will be research at higher sea-states.

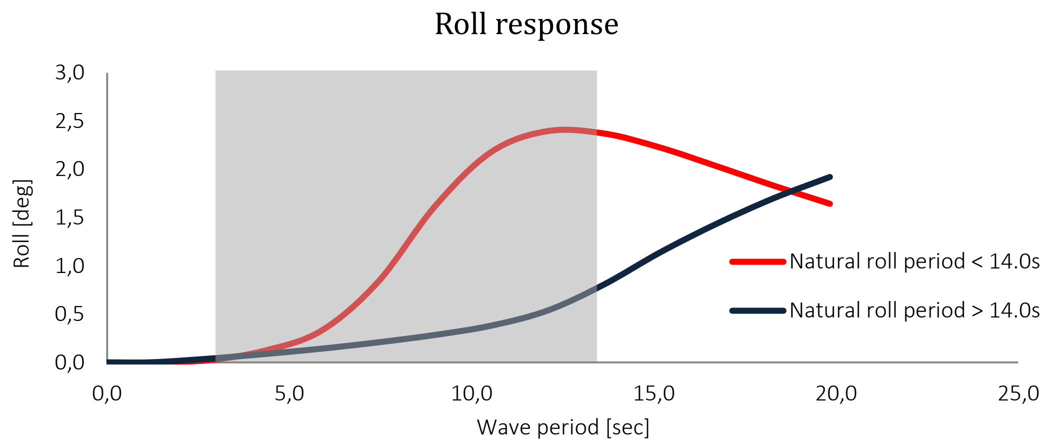


Figure 4-2: Example of the influence of natural roll period on the roll response in beam waves of Hs: 1.0 meter

Note that Figure 4-2 is only presented as an illustration and has no relation with the created concept designs of phase two.

4.4 Design goals

Design goals are set by analysing the operational requirements, limiting motion criteria, the operation profile and the design philosophy. The design goals, in order of importance, are listed below:

- High stability, to support the 5000mt lift operations
- Good motion behaviour, to gain a large workability for all types of lift and pipe lay operation.
- Significant transit speed
- A compact vessel, reduce of CAPEX and OPEX

This order is used during the design process to determine which design aspect is more important over another aspect. This order is based on the set operational profile of section 3.6.

It is important to note that the purpose of this research is to compare the different concepts – U and V shape - on a certain essential design and performance aspects. The emphasis of this design progress is therefore on creating two global concept designs rather than developing the designs in detail.

5 Estimation main dimensions

In this research the properties and operational performance of the multi-purpose construction vessel with a V-shape hull and one with a U-shape hull are compared. In order to perform this comparison, two vessel concepts are designed and studied for the same operational requirements as defined in chapter 3. The operational requirements are leading in this design process i.e. the vessel particulars results from the operational requirements. The design process helps gain understanding about the integral consequence of construction vessel with a V-shape hull.

This chapter unfolds as follows: the design approach is set, relevant reference vessel data is collected, general arrangement is defined, vessel dimensions and displacement are estimated. Using this data, the both hullform designs are defined.

5.1 Reference designs

A selection of reference designs are made to investigate whether certain trends can be found which can be used to estimate vessels dimensions. This data helps the design process, as it enables to start with realistic initial dimensions, arrangements, etc. For example: the typical waterline breadth of vessels with large offshore crane or the amount of fuel consumption. Despite construction vessels with a V-shape hull have never been build or designed, it is useful to monitor vessels with comparable operational requirements. Design aspects, which are not strictly related to the hull geometry, can be used. For example, to determine the required super structure area to provide accommodation for the crew. Only mono hull vessels with comparable functions and mission equipment are considered. Within the sponsor company Huisman Equipment a reference database is available which contains – circa 80 ships - useful information of construction vessels and is divided in the following four groups:

- Heavy Lift and Transport
- Reel lay vessels
- Pipe lay vessels
- Crane vessels

The most relevant vessels – only mono hull construction vessel – used in this study is presented in Table 5-1. The complete data of these vessels is included in appendix A.

Reference vessel	Missions
Seven Borealis	[redacted]
Oleg Strashnov	[redacted]
Aegir	[redacted]
Sapura 3000	[redacted]
Huisman crane vessel	[redacted]

Table 5-1: Most relevant and used reference vessels

The knowledge of studying the reference vessels, presented in Table 5-1, is used to make an estimation of the following design parameters:

- Waterline breadth

- Displacement
- General arrangement (firingline, superstructure)
- Required amount of consumables

The data and certain trends of the reference vessels are used and discussed in each section where a particular design aspect is considered.

5.2 Estimation of main dimensions

This section estimates the minimum required vessel dimensions to accommodate and support the mission equipment. Estimation is based on mission equipment specifications and reference vessels, per aspect as follows:

- Breadth:	Specified by required high stability for heavy lift operations
- Overall length	Based on length firing-line, OMC, pipe racks and superstructure
- Depth	Based on min. height of: firling line, engine room, tanktop and freeboard
- Displacement	Specified by the dimensions of above

These main particulars are based on the following:

- Breadth	Based on data of reference vessels
- Length	Based on data of reference vessels and technical specification of equipment
- Depth	Required vertical height of each mission equipment and freeboard requirements
- Displacement	Based on data of reference vessels

No separation is made between dimensions estimation of the different concept designs. Because this estimation is driven by needed space or support the mission equipment which is the same for each concept. In a later design phase the dimensions are optimized towards their typically characteristics.

5.2.1 Estimation of breadth

The initial stability of a vessel depends significantly on the waterline breadth, see also section 2.2. Owing to this the breadth of vessels which accommodates crane(s) with large lift capacities, are typically dictated by required high stability to support these lift operations. Hence, to determine the breadth the most critical loading conditions regarding the intact stability must be calculated, for every concept design individually. Prior to these stability calculations, the breadth was estimated to have a start value for the iterative study of the breadth. The breadth for both concept designs is set to 47.0 meter, this estimation is based on the lift capacity of 5000 metric tons and the data of reference vessels, as presented Figure 5-1. Note that only the overall breadth is estimated in this chapter, the breadth variation over depth of the V-shape is still undefined.

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Figure 5-1: Waterline breadth versus maximum lift capacity

Note that Figure 5-1 shows no clear trend between lift capacity and the breadth. It is likely to be that the difference is caused by the difference in vessel missions. Besides, the Figure contains a limited amount of data points. Therefore it is difficult to make a qualitative estimation. The breadth of 47.0 meter is based on the average value of the vessels which are designed to lift 5000 metric tons. This estimation is considered sufficient enough, as the estimated breadth is only used as start value for the iterative stability calculation.

5.2.2 Estimation of the length

The aspects which dominate the vessel length are given below.

- S-lay system
  In order to determine the length of the firing line, the arrangement of the total S-lay system is determined first. The firing line will be located underneath main deck on top of the freeboard deck and can handle single and double pipe joints. The produced pipeline will be pay-out through an opening in the vessel transom onto a stinger. Because the firing line is located on top of the freeboard the opening in the transom does not harm the enclosed volume of the hull.

  The pipeline production is divided over two deck layers, to gain an efficient as possible system within a small as possible vessel dimensions. The amount of working stations in the firing line is based on reference S-lay systems of the multi-purpose construction vessels: Sapura 3000 and Seven Borealis. Based on these systems it was defined that the firingline will have 4 welding stations, 1 Non Destructive Testing (NDT) and 1 coating station. Below the two different configurations, single and double joint, explained. See Figure 5-2 for a schematic visualization of the S-lay system.

  - <u>Single joint configuration</u>
    The production of the 40 inch single pipe joints is divided over two decks. At main deck level a workshop is located of 42.0 meter long and 6.2 meter high. These dimensions are based on the required space to fit two joints in longitudinal direction and with sufficient working space around it. In the workshop single joints are bevelled, pre-heated lined-up

and welded by 3 different welding stations into double joints. By means of an elevator the double joints are transfer to one deck lower where the firing line is located. Before entering the firing line the double joints will pass the through the last two welding stations of the total five stations. The firing line itself consists of a line-up Table, four welding stations, one NDT for the double joint welds and one for the welds created in the firing line itself. The last two workstations are the coating stations, one for the welds made at main deck level and one for the welds created in the firing line. This configuration allows a pay-out of two pipe joints once at a time. To ensure a constant production, the double joints factory has an extra welding station with respect to the firing line, to create stock for the firing line production.

A pipe kick out line to transport the pipe out of the firing line, in case the pipe has to be cut back is located at the starboard side. The damaged pipe can be stored in the area starboard of the firing line and by using the starboard elevator damaged pipe is transferred to main deck. Behind the line up Table there is a space with the length of a single pipe joint for in case the pipe is pulled back. See Figure 5-2 for a schematic visualization of the S-lay system with a single joint configuration.

- Double joint configuration
  In this configuration the vessel is supplied with prefabricated double pipe joints. The consequence of double joint supply on the S-lay system is minor. Instead of two NDT and coating stations, one of each station is sufficient in this configuration. Besides, at maindeck level only bevelling of the double joints is performed, instead the production of double joints. The pipe lay rate in the double joint configuration is comparable to the single joint configuration, because the production process in firing line is the same for both configurations. Though, a smaller overall production crew is needed, as the production of double joint is conducted onshore instead onboard.

The required vessel length to accommodate the above described S-lay system is determined, based on the set amount working stations and the distances between them. The location of the work stations are based on the fact that no station is placed underneath the main crane pedestal due to the limited space. This all results in the following initial particular:

The length of the firing line: 178.0 meter

Figure 5-2 visualise the pipeline production in a single joint configuration.

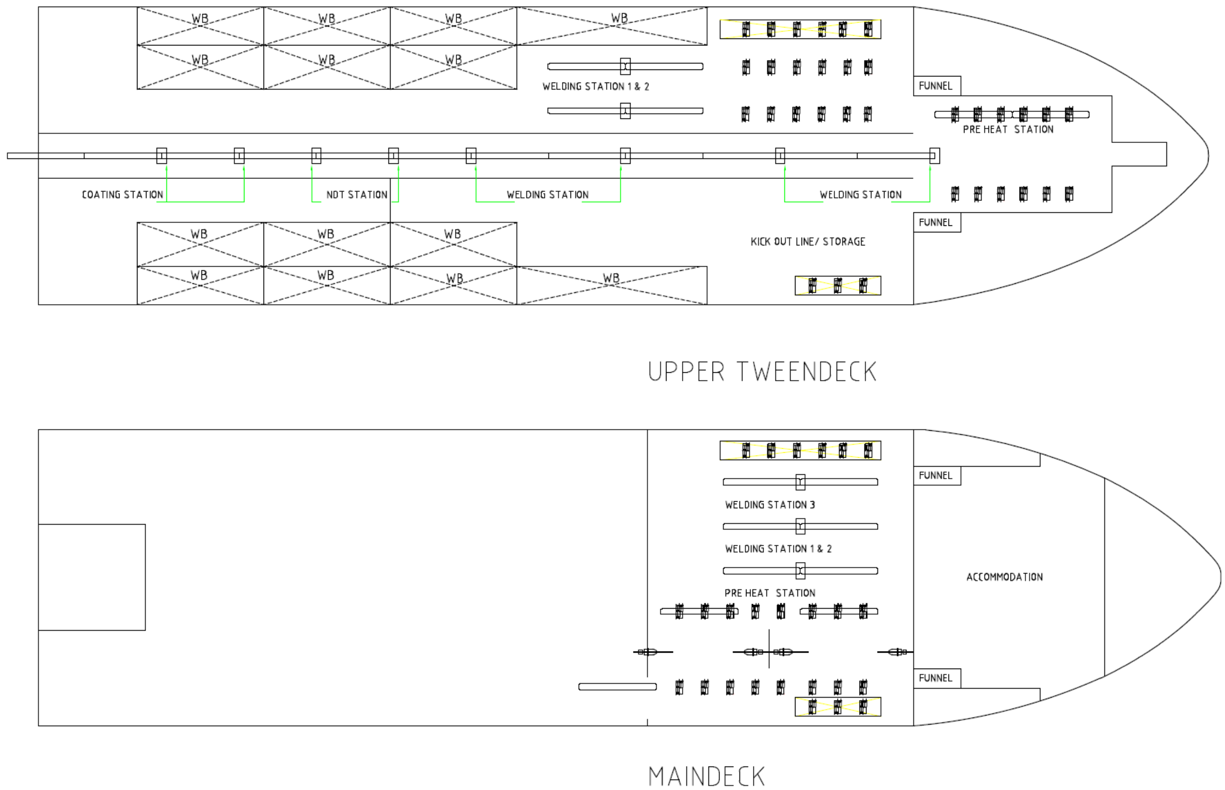

Figure 5-2: Firing line configuration, handling single pipe joints

- **Crane dimensions**
  The interface and dimensions of the OMC are given in table 3-4. The crane pedestal is located on centreline aft of the vessel, such that the crane outreach - over stern - is maximized. The distance between the square pedestal and the boomrest is 98.0 meter, the horizontal length of the boom in stowed position is 132.0 meter. The height of the boomrest enables to locate a low superstructure after the boom rest underneath the tip of the boom. Though, the height of the superstructure between the boomrest and tip of the jib is restricted, by the necessary space to place the block catcher of the auxiliary hoist. See Figure 5-2 and 5-3 for the vessel layout and see appendix B for the general arrangement of the crane.

- **Main deck**
  The length of the main deck is dictated by required deck surface to storage sufficient amount of pipe joints. Using the operational requirement as described in chapter 3, the pipe joint storage capacity on maindeck can be defined. It is required to storage a minimum amount of pipe joints to facilitate a production of 24-hours. In the operational requirement it is stated that the production rate should be minimum 1 pipe joint of 46 inch every 4 minutes. Hence, 360 pipe joints should be stored at least on main deck. As the vessel breadth is estimated the minimum main deck length can be calculated.

  The available deck space is impeded by the footprint of: main crane, pipe loading cranes and the boom rest. The 47.0 meter wide main deck allows a pipe rack which contains 26 pipes per layer of 46 inch in the transversal direction. In the area next to the boom rest a smaller racks - containing

8 pipe- per side can fitted. Between the main crane pedestal and the boom rest 4 pipe racks, in longitudinal direction, can be placed. If each rack contains 3 layers a total amount of 360 pipe joints can be stored on the main deck. Figure 5-3 presents pipe rack configuration in case 360 pipe joints are stored with a diameter of 46 inch. This configuration is designed such that a pipe transporting system can be fitted along racks, to feed the pipes into the production area. Based on this analyses it can be stated that the, from pipe lay modus perspective, main deck should be at least 96.0 meter in length.

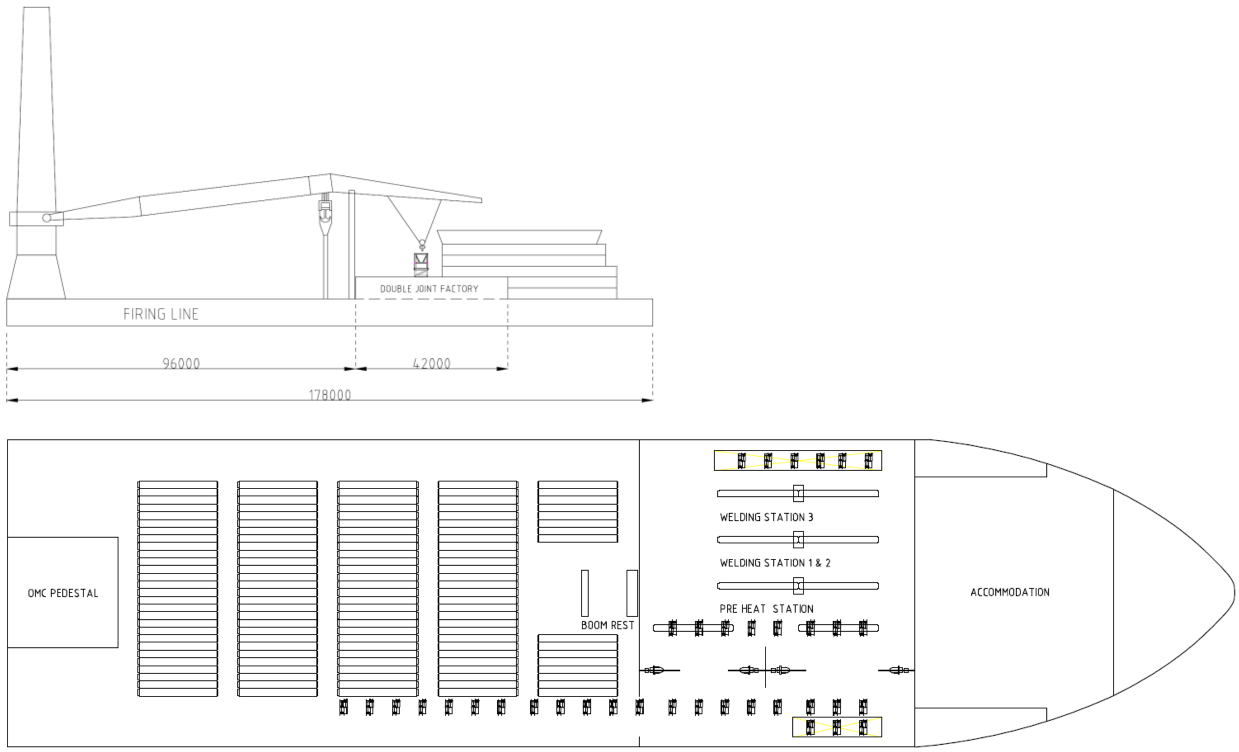

Figure 5-3: Schematic drawing of components determining vessel dimensions. And the pipe racks configuration.

- Accommodation

  A maximum crew of 398 people will work – in a double shift system – on the vessel and needs accommodation on board. The amount of crew is set in the operational requirement, see section 3.4. The superstructure dimensions are based on a reference data of the Huisman crane vessel concept, as amount of crew is equal. The accommodation has a total floor space of: 5700.0 square meters - excluding the bridge deck – and 620.0 square meters offices. This area was used to estimate the superstructure dimensions of both concept designs. The superstructure will consist of 5 floors with the bridge deck on top, as a sixth layer. This superstructure configuration requires 30.0 meter vessel length after the workshop. Hence, the required length of the main deck plus the accommodation falls within the firingline length. See Figure 5-3 for a schematic representation of the superstructure.

Conclusion on the length estimation:

Based on the mission equipment - firing line length, superstructure, pipe storage capacity and crane in stowed position - the minimum vessel length was estimated. From the analysis of this section it was concluded that the firing line length dictates the vessel length. The length of the firing line is set to 178.0 meter, the base of the firing line is located just above the waterline, hence the overall length should be longer. Although, the overall length does not have to be substantially longer than the firing line, because the last part - pipe overshoot – is narrow. This all results in the following length estimation:

- The waterline length is set as: 178.0 meter

5.2.3 Estimation of the depth

The minimum required depth depends on required freeboard height and height to implement all the mission equipment is governed by height of: firing line, engine room(s), crane winch room and height of the tanktop. The pipe line production area is located underneath main deck and stretches out over the total vessel length. The crane winch and engine rooms must be located underneath the firing line. The longitudinal configuration of the engine and crane winch room(s) is not defined yet however, it can be assumed that they will be situated at the same deck. The required winch room height is known by the technical specifications of the OMC. Where, the firing line height is defined by the size of the pipe tensioners that clamp the pipe. The size of this equipment is known by the technical specifications. The tanktop and engine room height is based on data of references vessels. Resulting in the following heights:

- Firing line: [redacted] [m]
- Crane winch: [redacted] [m]
- Engine room: [redacted] [m]
- Tanktop: [redacted] [m]

Based on the values of above the minimal depth is set to: [redacted]0 meter

5.2.4 Estimation of displacement

The displacement of both concept vessels is based on the reference vessels as presented in Table 5-1. The relation between the Lpp/breadth ratio and displacement is plotted in Figure 5-4. The Lpp/breadth ratio for both concepts is 3.78, the displacement is [redacted] metric tons. In this design phase it is assumed that the displacement of the V-shape concept is comparable. The draft will be large of the V-shape concept design, as the block coefficient is smaller in comparison to the U-shape concept design.

Note that no clear relation between the Lpp/B ratio and displacement is shown in Figure 5-4 and a limit amount data point is present. However, the vessels Oleg strashnov, Huisman crane vessel and Seven borealis appears to have a comparable displacement Lpp/B ratio. Based the comparable Lpp/ratio and missions of these vessels it assumed that a displacement of [redacted] tons is a sufficient estimation.

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Figure 5-4: Displacement as a function of the ratio Lpp/B

5.2.5 Result main dimension estimation

The minimum vessel dimensions in order to accommodate the mission equipment within a vessel were estimated. These minimum dimensions are considered as a starting point for both U- and V-shape concepts and are included in Table 5-2. It is assumed that the required vessel breadth to support maximum lift operations is the same for both vessels. However, it is expected that the stability property depends on the hullform and could be different; stability calculations will determine the breadth more precise.

Item		Unit
Min waterline length	[redacted]	[m]
Waterline breadth	[redacted]	[m]
Min depth	[redacted]	[m]
Displacement (transit)	[redacted]	[mt]

Table 5-2: Minimum vessels dimensions to implement mission equipment

6 U-shape design

The design of the U-shape concept is presented in this section. The concept design includes the hullform design, displacement and freeboard calculation. The hullform and tank arrangement is modelled in Delftship V7.11, this program is developed by Delftship Marine Software in the Netherlands. Using this software, the displacement can be calculated and in later state the intact stability.

Note that the goal of this research is to create two global concept designs, no detailed vessel designs are created in this research, as previously in the research limitations; section 1.6.

6.1 Hullform design U-shape

The defined minimum main dimensions, imposed by the mission equipment, are used as input for the U-shape hullform design. The set design goals are used as guidance in the design process. This section considers the stern, mid-ship, and stern separately and are assembled afterwards. This approach enables to focus first on the key design elements of each section, rather than the total hull design at once.

6.1.1 Stern design

By studying the operational requirements and the operation profile regarding vessel's stern the following can be pointed out:

- Sufficient buoyancy: To support the OMC and heavy stinger and pipe loading
- Slender stern: To reduce (flow separation) the hull resistance
- No immerse of transom: To reduce the hull resistance
- Sufficient height to mount thrusters: A number of thrusters is required for station keeping

Based on these the following geometry boundaries for stern are set:

- A slender stern with no submerged transom in transit condition. In pipelay and lift condition it will be submerged to gain buoyancy.
- Min. vertical distance to mount a thrusters (distance baseline waterline): 6.2 [m]

  This value is based on the specification of 5.5 [MW] Rolls-Royce thrusters. This thruster is installed at the Seven Borealis for propulsion and DP. This gives insight in the required vertical space to mount a thruster.
- In order to avoid severe flow separation the stern rise should be in the range of: 10.0 – 15.0 degrees.

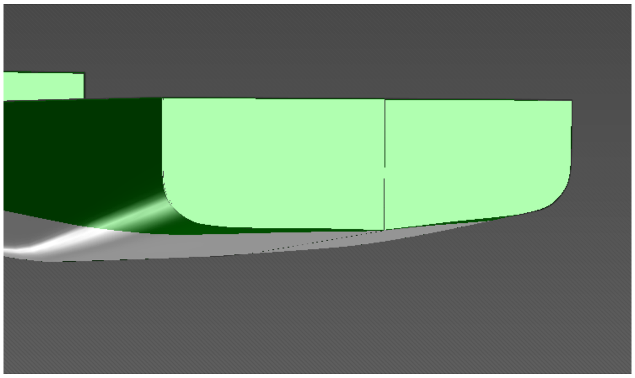


Figure 6-1: Impression of the vessel aft of the U-shape concept design

According these boundaries, a stern was designed; Figure 6-1 gives an impression of the stern. In order to improve the directional stability a skeg is installed. Besides, the skeg adds roll damping and extra buoyancy at the vessels aft. On the other hand, a skeg introduces extra friction resistance and distortion of the inflow into the thruster(s).

6.1.2 Mid-ship design

The mid-ship (main-section) is defined as the section where geometry is constant with respect to the longitude axis. The mid-ship section characterizes the key difference between the two vessel types. Since the breadth and depth are estimated in the previous section, no design parameters remain to define apart from the bilge radius. The bilge keel radius is set to 2.5 meter and is based data of reference vessels.

The vessel will accommodate bilge keels over the entire length of the mid-ship, to improve the roll damping. The bilge keel radius of 2.5 meter allows to install bilge keels with a cord length of 0.75m, without exceeding the overall breadth.

6.1.3 Bow design

For the bow design, the emphasis is on a low resistance. To fulfil the requirement of 15 knots transit speed, in an efficient as possible manner. However, according to priority of the design goals, section 3.4, lift and pipe lay capability may not suffer too much by the bow design. The bow section starts minimal 125.0 meter from the origin of the axis convention. This to implement the S-lay system as defined in section 5.2.

The second key design aspect is the choice for a normal or a bulbous bow. Based on the hydrodynamic optimization of ship hull forms [ref. 9] it was decided to design a bulbous bow. In this study a series of eight different hull forms was considered, 7 vessels with a unique bulb shapes and one with a no-bulb bow. The study, shows for low Froude numbers vessels with no-bulb have the lowest resistance but the highest resistance at intermediate and high Froude numbers. The “break-even-point” is circa at Froude number of 0.17, depending on bulb type. As the transit speed for the U-shape concept is higher than a Froude number of 0.17, it is beneficial to implement the bow with bulb. The Froude number for the transit speed of 15 knots is given below:

- Froude number (for both concepts): 0.185 [-]

Because the transit speed is relative low a bulb design is chosen which preforms best at relative low Froude numbers. The so called “Baseline bulb” is selected, according hydrodynamic optimization for ship hull forms [ref. 11]. This elliptical shaped bulb moulded totally underneath the waterline. A more in depth optimization of the bulb is not conducted in this thesis, as a detailed design is out of the research scope. Figure 6-2 shows the bow section design, as in the software Delftship.

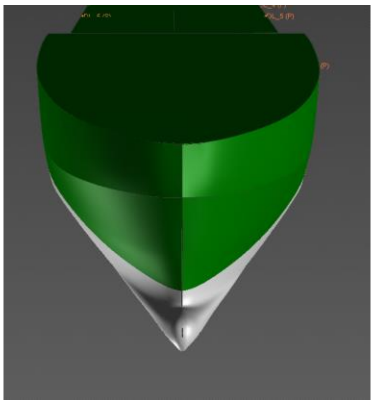


Figure 6-2: Impression of the bow of the U-shape concept design

6.2 Weight calculation

Using the defined hullform, the displacement was more precisely calculated, with respect to section 5.2.4. The lightship weight and deadweight are considered separately below.

6.2.1 Deadweight

Deadweight is based on; operational requirements and data of reference vessels. Distinction is made between the weight groups as follows:

- Pipe joints storage
  Based on the operation requirements the amount to storage pipe joints is set to: 360. Based on Huisman S-lay system technical specification the maximum single joint weight is set to 30 metric tons. Resulting in maximal pipe joint storage weight of: 10,800 metric tons.
- Deck load
  The amount of required cargo on the maindeck is define in operational requirements
- Storage capacity of consumables
  Depends on the specific fuel consumption of the propulsion system, to be designed. In this stage, the estimation is based on: data of the reference vessel Seven Borealis. This is assumed to be useful data, as the dimension and the functions of this vessel are comparable. The following capacities are used:
  - Fuel oil 3600 [m³]
  - Fresh water 1800 [m³]
  - Food 400 [mt]

- Crew
  A weight of 0.2 metric tons per crew member is assumed. The weight includes personal weight and personal belongings.
- Stinger
  The stinger is considered as deadweight, as it is only connected during pipe lay operations. The vessel will be equipped with same stinger as the Seven Borealis, as detailed data is available within Huisman Equipment.

The deadweight is summed-up in Table 6-1. Note that the amount of deadweight depends significantly on the type of operation. For example, The main deck is not loaded with 9000 metric tons in case it is fully loaded with pipe joints for example.

Groups	Mass	Unit
Pipe joint storage	[redacted]	[mt]
Deck load	[redacted]	[mt]
Consumables (100%)	[redacted]	[mt]
Crew	[redacted]	[mt]
Stinger and frame (dry weight)	[redacted]	[mt]
Stinger and frame (Submerged weight)	[redacted]	[mt]

Table 6-1: Deadweight summery

6.2.2 Light ship weight

Distinction is made between different light weight groups. The different weight groups and the used estimation method is described in Table 6-2.

Weight groups	Contains	Estimation method
Hull, Superstructure	All containing construction steel weight and welding of steel	Huisman in house developed volumetric weight coefficients.
Accommodation	All outfitting for: accommodation, excluding steel weight of the superstructure	By Huisman in house developed volumetric weight coefficients.
Mission equipment	OMC, S-lay system and pipe loading cranes	Known by technical specification
Vessel equipment	Life and rescue boats, mooring systems, etc.	Based on reference vessels
Propulsion	Main engines and thrusters	Based on reference vessels
Rest	Electrical, marine and safety systems	Based on reference vessels

Table 6-2: Light weight groups

Table 6-3 includes the calculated light weight per group.

Groups	U-shape	Unit
Hull, superstructure	[redacted]	[mt]
Accommodation	[redacted]	[mt]
Mission equipment	[redacted]	[mt]
Equipment deck	[redacted]	[mt]
Propulsion	[redacted]	[mt]
Rest	[redacted]	[mt]
Totals for lightship (incl. 5% contingency)	[redacted]	[mt]
VCG	[redacted]	[m]
LCG	[redacted]	[m]
TCG	[redacted]	[m]

Table 6-3: Light weight summery

The LSW and its centre of gravity (COG) as presented in Table 6-3 is calculated according to the input, assumptions and method as described below.

- Hull, superstructure and Accommodation
  This weight group is volume driven, the volumetric centre is assumed as the COG. The centre of buoyancy of the submerged hull – until main deck – according Delftship is assumed as the COG of this weight group. The COG of the superstructure is per layer calculated. Each superstructure layer is considered as a solid body, the volumetric centre is assumed as the COG.
- Mission equipment
  The individual centre of gravity of each mission equipment member is known by the technical specification.
- Propulsion
  This weight group includes six main engines, five thrusters and one bow thruster.
- Equipment deck
  The COG is taken vertically one meter above main deck. Half of the overall length is assumed as the longitudinal COG.
- Rest
  This weight group includes the electrical, marine and safety systems. The main part of this equipment is assumed to be located in the technical room. This room is located prior to the engine room, see appendix C for the general arrangement.

The calculation light weight is used as input for the freeboard and stability calculations. Note that the displacement significantly depends on the type of operation.

6.3 Tank arrangement

The tank arrangement includes fuel, fresh water and water ballast tanks. The water ballast system contains the following two main functions.

- Counter weight to keep the vessel upright during lift operations

To withstand the large overturning moments generated by the cargo load hanging in the crane and to keep the vessel upright counter ballast is needed. By filling the wing tanks with water, at the opposite side of the crane load, the overturning moment is compensated.

- <u>Improve initial and range of stability, during operation process</u>
  Through filling the lower positioned water ballast tanks, the vessel's KG will be reduced, resulting in an increased GM, see equation 2-1.

The tank arrangement design has significant influences on displacement and weight distribution, therefore on the motion behaviour and workability. The broad operational profile makes the tank arrangement complex, as each operation type requires different sized tanks. This statement can be clarified with the example below.

- <u>Lift operation</u>
  To compensate for the high and heavy hook load, high vessel stability is needed. In this critical condition free surface moment is therefore unwanted and needs to be avoided. Hence, in the most ideal situation, a tank is fully or not filled. The required amount of water depends on the cargo load and crane outreach for example. However the free surface effects should be avoided for all lift operations, requires different sized water ballast tanks.

Based on the operation profile and the critical stability condition during heavy lift operations, it is decided that the 5000 metric tons lift over side operation is governing for the tank arrangement design.

The stability can be improved by lowering the vessel's KG, the most efficient way to achieve this is filling the low located water ballast tanks Therefore, the entire tanktop is equipped with multiple water ballast tanks. To compensate the overturning moment introduced by the OMC, the ballast tanks will be located along the vessel side shell (wing tanks), to gain the maximum transversal distance between hook load and counter weight. The tanks are situated in the available space around the mission equipment.

The required tank capacity for the fuel and fresh water consumables is set in the deadweight definition in Table 6-1. The freshwater tanks are located near the superstructure and the fuel tanks close to the engine rooms. The tanks are located at the upper tweendeck (one deck below the freeboard deck), this relative high vertical position will increase the KG resulting in lower GM during transit operations. These tanks cannot be located on the freeboard deck. As on the freeboard deck around the engine and superstructure no space is available due to the S-lay system configuration. Figure 6-3 shows the tank arrangement for the U-shape concept. Note that the freeboard deck is the deck where the firingline is located, see the general arrangement included in appendix C for more information for the tank arrangement.

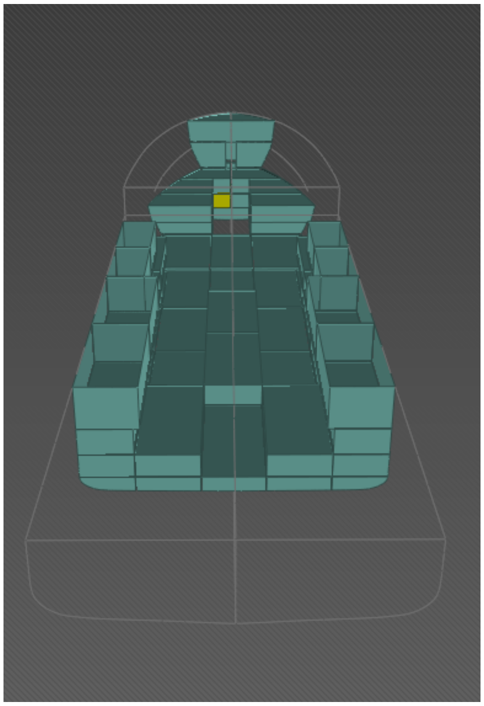


Figure 6-3A: Water ballast tank arrangement

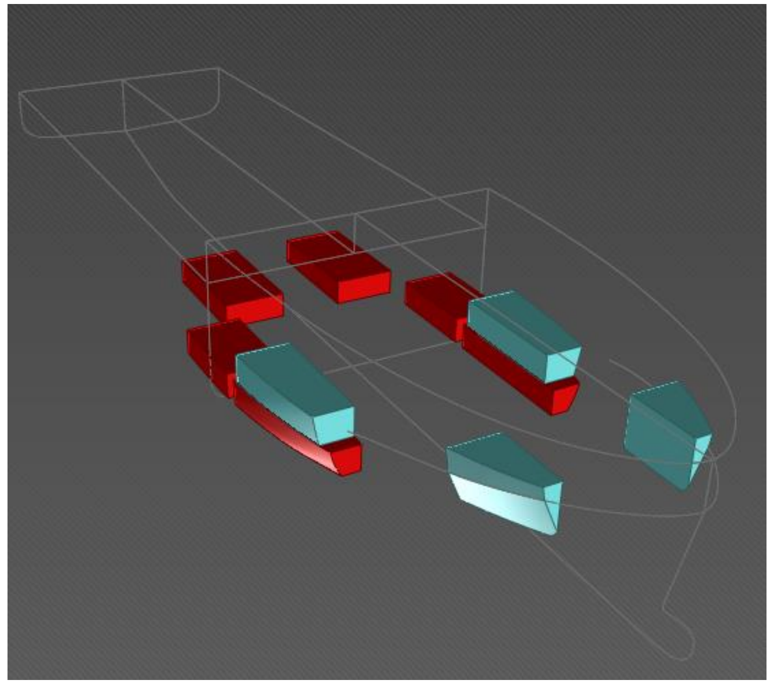


Figure 6-3B: The red coloured tanks represent the fuel and the green tanks the freshwater tank arrangement.

6.4 Freeboard calculations

The freeboard height was calculated, to check if the U-shape concept fulfils the IMO freeboard criteria [ref. 6]. In order to calculate the freeboard height of the U-shape concept, the following data is needed:

- Input data
  The most critical loading conditions
  Tank arrangement
  Draft

An important aspect for the design is that IMO allows an exception of the freeboard criteria for vessel undergoing special operations as crane lift conditions. This is permitted since; (heavy) lift operations are only conduct at restricted areas or in mild sea states, allowing accepting reduced reserve buoyancy. This exception is not valid if the vessel conducts pipe lay operations. Despite that these operations are performed in relatively mild sea-states. In case the pipeline suddenly must be abandoned, due to upcoming severe weather, the vessels loading conditions is unaltered and needs to withstand the present environmental conditions, i.e. sufficient reserve buoyance is required. This section describes; the input data, the results and the hullform modification in order to comply with the criteria.

6.4.1 Loading conditions

The loading conditions must be defined in order to know which loading condition is governing – i.e. the condition with the largest draft – for the freeboard calculation. Table 6-4 presents the most governing and common loading conditions. The presented weights are according to the deadweights as was presented in section 6.2. Note that the stinger is only attached during pipe lay operations or in transit conditions towards the area where the pipeline will be installed. During the other operation conditions it is assumed that the stinger is not on board.

No.	Operation conditions	Cons weight	Specific weight item		Unit
1	5000 mt 34m over side* (100% cons)	5800	Hook load:	5000	[mt]
2	5000 mt 40m over stern* (100% cons)	5800	Hook load:	5000	[mt]
3	Pipelay (max pipe storage & 100% cons)	5800	Pipe joint & stinger:	10,800	[mt]
4	Transit (50% cons)				[mt]
5	Transit (pipelay ready and 100% cons)	5800	Stinger:	1932	[mt]
6	Transit (deckload and 100% cons)	5800	Deck load:	10,800	[mt]

Table 6-4: Considered loading conditions

Table 6-4 shows, load case three and six have the largest amount of deadweight. To calculate the displacement, the amount of water ballast needs to be determined.

6.4.3 Freeboard height calculation

Using the software Delftship, the amount of water ballast was determined and so the draft was calculated, for the load case as stated in Table 6-4. Per case the amount of water ballast was determined according to the approach as described in section 8.4. The aim is to ballast the vessel such that the list, trim, stability and motion characteristics are optimal as possible with a minimum as possible amount ballast.

The calculations show that the freeboard height does not comply for load case three. The depth was therefore increased with 0.2 meter. This allows to place the freeboard 0.2 higher in the vessel with respect to the baseline, such that the S-lay system can be implemented between the freeboard and main deck. The light ship weight, draft, and required freeboard height was recalculated and included in Table 6-5.

Item	U-shape	Unit
Draft aft pp	8.8	[m]
Mean draft	8.8	[m]
Draft forward pp	8.9	[m]
Freeboard height	3.3	[m]

Table 6-5: Draft and freeboard height

For the changed hull geometry the minimum freeboard according IMO:

- U-shape: Minimum required winter freeboard height: 3237 [mm]

Hence, the freeboard height satisfies the criteria. Moreover; the transom is not submerged in transit condition, despite the extra construction steel weight.

6.5 Results U-shape concept design

Table 6-6 includes the main particulars of the U-shape concept. In chapter 8 the stability properties are calculated, to determine if this concept fulfil the operational requirements, as previously described in chapter 3.

Item	U-shape	Unit
Overall length	[redacted]	[m]
Overall breadth	[redacted]	[m]
Depth	[redacted]	[m]
Draft (transit)	[redacted]	[m]
Draft (lift operation)		[m]
Min slope of vessel sides	[redacted]	[deg]
Light ship weight	[redacted]	[mt]
VCG	[redacted]	[m]
LCG	[redacted]	[m]
TCG	[redacted]	[m]

Table 6-6: Main particulars the U-shape concept design

The design process of the V-shape concept is described in this section. The V-shape concept is developed for the same missions and operational requirements as the U-shape concept. The minimum dimensions imposed by the mission equipment, as set in chapter 5, are therefore the same.

In this chapter: a parametric study for the V-shape mid-ship geometry was performed in order to support the design process. Using this knowledge the hull of the V-shape concept was defined. Furthermore, the displacement, freeboard height and tank arrangement are determined in this chapter.

7.1 Parametric study mid-ship

This parametric study on the mid-ship section is performed to gain insight about how V-shape geometry affects the initial and range of stability of a vessel. Furthermore, to investigate which V-shape geometric as the large as possible capability to adapted the initial stability and motion characteristics to different operation conditions. In order to gain this knowledge, the vertical node position and side shell slope was systematically varied. The definition of these parameters is given below. This section presents the approach, input and results of this study.

- Node position
  The hull goes from vertical to a sloped side shell at this point; the node, see also Figure 7-1.
- Side shell slope
  The angle between baseline and the vessel side shell, see Figure 7-1.

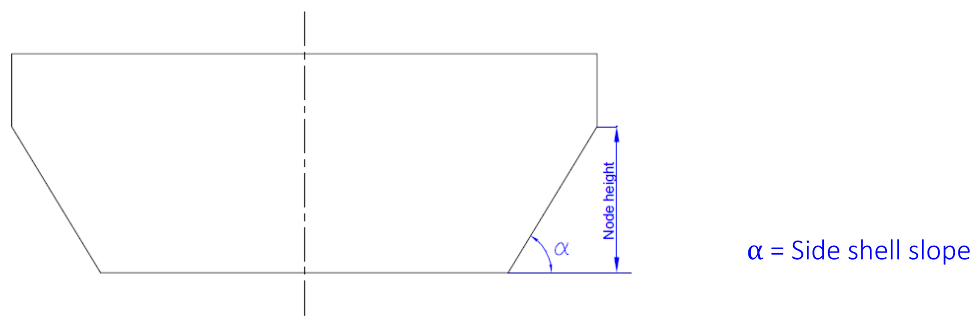


Figure 7-1: Definition of the V-shape hull parameters

7.1.1 Approach

The geometry parameters are varied one at the time, such that the influence of each parameter on the hydrostatic properties can be individually analysed. Each parameter is varied four times according to the steps given below:

- Node height: [redacted] meter
- Side shell slope: [redacted] degree

The stability characteristics of all sixteen hull geometries are calculated by using DelftShip V7.11, this program is developed by DelftShip Marine Software in the Netherlands. In the software the mid-ship section was modelled only, to research only the effects of these two parameters on the stability properties. This approach is simple but effective, since it provides insight about the fundamental effects. The vessel's stability is calculated for the load cases given below.

- Maximum lift capacity over side : The vessels has a high KM and KG, hence a low GM
- Transit operation with no cargo: The vessel has a high KM and low KG, hence a high GM

These two cases are considered, as the difference in initial stability between these cases is large i.e. the upper and lower bound case. Using this data, the capability of each model to adjust its initial stability can be determined.

Monitored stability parameters

Each model is evaluated by the stability properties, as following:

- <u>The stability parameters KM, GM and GZ curve</u>
  GM: Obtain understanding about the influence of the geometry on the initial stability including the effect of changed tank arrangement and the COG of the light ship weight.
  GZ: Determine the effect of the parameters on the range of stability
- <u>ΔGM</u>
  The difference in GM value between the two considered load cases. Note that a small ΔGM is beneficial over a large ΔGM! As the initial stability in max lift capacity over side is small and when the ΔGM is small the GM in transit load cases is relative small as well.

7.1.2 Input and models

A base model was made. With respect to this base model the parameters were varied resulting in a total of sixteen different models. Each model was implemented in the software Delftship. The base model is based on the dimensions shown in Table 5-2. All the dimensions of the base case model are kept the same except for the length. The main section of the vessel was extended over the total vessel length. Because the models include only a parallel mid-ship the length was reduced to get the same displacement. The node height is set at 10.5 meter and the side shell slope is set at 45.0 degree for the base model. The particulars of the base model are included in Table 7-1.

Item	Particulars	Unit
Length	[redacted]	[m]
Depth	[redacted]	[m]
Breadth	[redacted]	[m]
Displacement	[redacted]	[mt]
Node height	[redacted]	[m]
Side shell slope	[redacted]	[deg]

Table 7-1: Particulars base model

The lightship weight and the KG are calculated for each model separately, as they change by each systematic parameter variation. Section 6.2 describes how the displacement is calculated. The dimensions of the water ballast tanks change with systematic variation of the model. This allows to determine the effect of the two geometric parameters on the KG and therefore the GM. For example; A mid-ship with a side shell slope of 45.0 degrees and a node of 11.5 meter leads to a relative small baseline breadth. Therefore a limited amount of water ballast can be located in the tanktop and lower wing tanks, resulting in a relative high KG. Whereas a V-shape with a side angle of 60.0 degree and node height of 8.5 meter led to relative low KG. In Delftship, the water ballast tanks located along the hull are defined by the shell of the model. So the dimensions of these tanks change automatically with the varied hull parameters, whereas the inner configuration stays the same.

7.1.3 Results

The initial and range of stability for the sixteen different V-shape mid-ship sections was calculated. Figure 7-2A presents the ΔGM as a function of node height and side shell angle. All obtained results are included in appendix D. Note that this result was not evaluated with stability criteria.

Confidential	Confidential
Figure: 7-2A: Delta GM as a function of node height and side shell inclination	Figure 7-2B: GZ curves for different node height

The parameter ΔGM is considered leading in the selection of the geometric parameters. Because this quantifies the capability to adjust the stability properties depending on operational requirements and therefore the motion characteristics of a vessel. A V-shape mid-ship with a node of 11.5 meter and a side shell angle of 45.0 degree proves to have the smallest ΔGM, as shown in Figure 7-2.

On the other hand, the range of stability for slender geometries is significant smaller, as a negative effect the vessel could have difficulties with [redacted] side of a “too” slender vessel is the limited amount of displacement. Because of this all a node height of 10.5 meter is selected.

7.2 Hullform design V-shape

The design process considers the stern, mid-ship and stern separately and assembled them in a hull afterwards. Table 7-4 presents the determined particulars of the V-shape concept design.

7.2.1 Stern and bow design

The design of these sections is based on the same requirements and boundaries as for the U-shape concept, as described in section 6.1. Figure 7-3 gives an impression of the stern and bow sections.

- Stern section
  In order to improve the directional stability a skeg is added. Besides, the skeg adds roll damping and extra buoyancy at the vessels aft. However, a skeg introduces extra friction resistance and distortion of the inflow into the thruster(s)
- Bow section
  The bow section starts at 125.0 meter from the stern; the same distance as the U-shape bow. Although the smaller breadth on the waterline (at transit draft), enables to design a more slender bow within the same length with respect to U-shape hull. Furthermore, the V-shape concept will be equipped with a bulb and with the same type as of the U-shape. As the Froude numbers are identical for both concepts design, for a design speed of 15.0 knots.

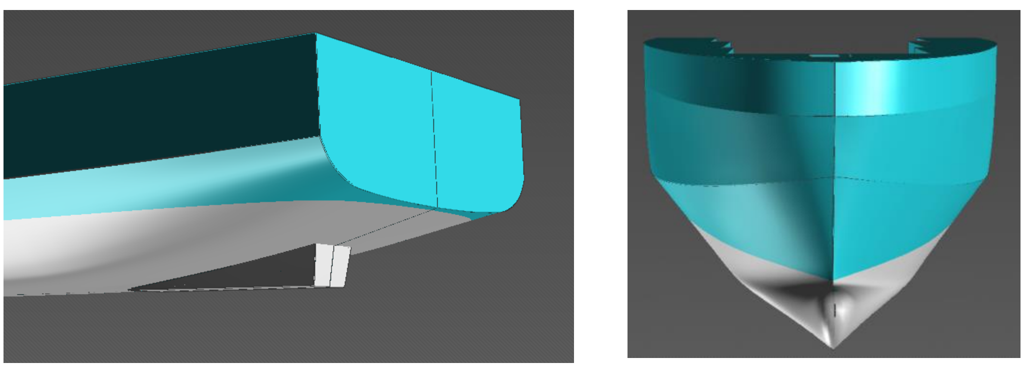


Figure 7-3: Impression of the stern and bow section of V-shape concept design

7.2.2 Mid-ship design

The mid-ship geometry (main-section) of the V-shape concept presents the key difference between the concept designs. The mid-ship design is based on the knowledge of the parametric study of section 7-1 and as follows:

- Node height: confidential Meter
- Side shell slope: confidential Degree

These parameters together with the estimated depth and breadth define the geometry of this section.

The vessel will accommodate bilge keels over the entire length of the mid-ship, to improve the roll damping. The V-shape geometry allows to install bilge keels with a cord length of 2.2 meter, without exceeding the breadth on the waterline in transit draft. The exceptional large bilge keels enhance the roll damping significantly. On the other hand, large wetted surface of the bilge keels is a drawback, as it increases the friction resistance in comparing with bilge keels of the U-shape concept. The drawback of a larger wetted surface can be accepted, as good motion behaviour is considered more important, according to the design goals, than a low hull resistance. In section 10.2.5 analysis if the roll damping will improve by equipping the V-shape with large bilge keels.

7.3 Weight calculation

Using the defined hullform, the displacement can be calculated according the method described in section 6-2.

- Deadweight
  The amount of deadweight is set in the operational requirements and therefore the same for both design concepts. Section 6.2.1 specifies each deadweight group. Note that the fuel storage capacity for the V-shape is assumed to be the same. If the resistance calculations show a significant difference, the required fuel capacity will recalculated.
- Light ship weight
  As described in chapter 6.2.2 the light weight contains the groups; hull & superstructure, accommodation, missions equipment, vessel equipment propulsion and rest. It is assumed that only the hull & superstructure weight group causes the difference between the two concept designs. As, the other weight groups are driven by the, same, mission requirements. For example, the weight of the accommodation is comparable as it will be designed for the same crew size. The same goes for the selected mission equipment, which is the same for each concept. Note that it is assumed that the propulsion system is equal. Resistance calculations will show if this assumption is correct.

  The hull & superstructure weight is calculated according the same volumetric weight coefficients. Huisman Equipment in house developed coefficient, for construction vessels.

The calculated displacement is presented, together with the U-shape concept, in Table 7-2. Note that the displacement depends on the operational condition.

Groups		U-shape	V-shape	Unit
Deadweight				
	Pipe storage	[redacted]	Idem	[mt]
	Deck load	[redacted]	Idem	[mt]
	Consumables (100%)	[redacted]	Idem	[mt]
	Crew	[redacted]	Idem	[mt]
	Stinger			
Lightship (incl 5% contingency)		[redacted]	30793	[mt]
Displacement (100% consumables, no loading)		[redacted]	36673	[mt]

Table 7-2: Calculated displacement

7.4 Tank arrangement

The tank arrangement includes fuel, fresh water and water ballast tanks. The amount of fuel and fresh water of the V-shape concept is equal to the U-shape. The water ballast system as the same two main function as the system of the U-shape, as was described in section 6.3. However, the required amount of water ballast for the V-shape concept is unequal. As the effect of water ballast on the stability of a V-shape hull is differently as for a U-shape:

- Additional to the effect of water ballast to act as a contra weight to the cargo load in the crane and to reduce the vessel's KG. The draft increases and the water plane area increases as the breadth increases significantly with the draft. Resulting in an increased GM value.

Despite this "extra" function of the water ballast, the same design considerations are used for the tank arrangement as of the U-shape. Figure 7-4 shows the created tank arrangement of the V-shape concept.

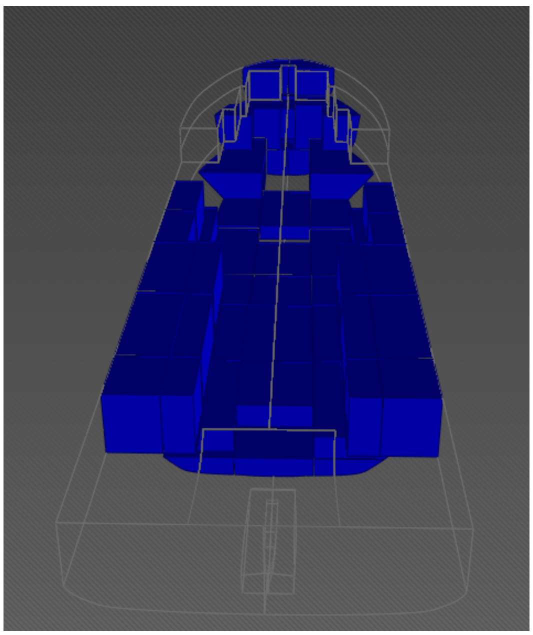


Figure 7-4A: Water ballast tank arrangement of V-shape

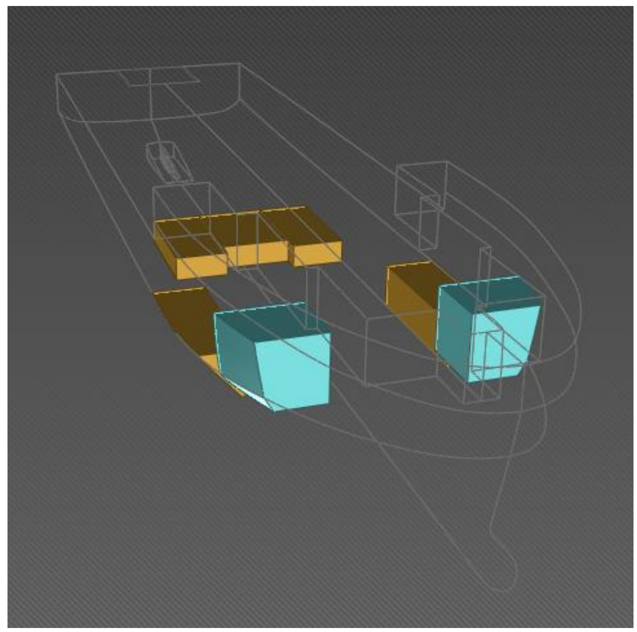


Figure 7-4B: The brown coloured tanks represent the fuel and the green tanks the fresh water tank arrangement, of V-shape

7.5 Freeboard calculations

The freeboard height was calculated, to check if the V-shape concept fulfils the IMO minimum freeboard criteria [ref. 6]. The freeboard height did not comply for the governing pipe lay load case, as defined in Table 6.4.

In order to satisfy this IMO regulation the depth was increased with 1.1 meter upward of the node. As a consequence the hull construction weight will increase and therefore the draft. The vessel length, mid – ship section, was enlarged with 2.5 meter to gain extra buoyancy, to compensate for the extra steel weight. This can also be achieved by increasing the vessel side angle, vessel wide or lowering the vessel node. Extending the vessel length is the preferred option, this is explained by considering the effect of each modification below:

- <u>Vessel length</u>
  By extending the mid ship section extra buoyancy is created by the same waterline width. More additional steel weight is needed to gain buoyancy with respect to the design options (described below) By adding 1.0 metric tons steel weight 3.4 metric tons buoyancy is gained.
- <u>Side shell slope</u>
  A larger side shell slope leads to a larger blockcoefficient, and therefore more buoyancy at the same draft and main dimensions. With respect to steel weight this is an effective way to increase buoyancy. By adding 1.0 metric tons steel weight 9.2 metric tons extra buoyancy is gained. However, a larger side shell slope has the consequence of a wider waterline in lighter draft conditions and therefore less favourable motion behaviour. This is shown in the Figure 4-7, where models with a larger side shell angles have a large ΔGM, between deep and light draft condition.
- <u>Lower the node position</u>
  A lower node position leads to enlarged blockcoefficient, and therefore more buoyancy at the same draft and main dimensions. The consequence of this modification on the motion characteristic is comparable to increasing the side shell slope. Based on Figure 7-8 it can be conclude that lowering the node position is slightly better that increasing the side angle regarding the motion behaviour. By adding 1.0 metric tons steel weight 9.5 metric tons buoyancy is gained.

Extending the mid ship section is considered as the best option since it complies with the design goals (see section 4.4)

The recalculated draft and freeboard of the modified V-shape is given in Table 7-3.

Items	V-shape	Unit
Draft aft pp	9.8	[m]
Mean draft	9.8	[m]
Draft forward pp	9.7	[m]
Freeboard height	3.3	[m]

Table 7-3: Draft and freeboard height

For the changed hull geometry the minimum freeboard regarding the criteria was calculated:

- V-shape: Minimum required winter freeboard height: 3291 [mm]

Hence, the V-shape concept complies with the criteria. The lightship weight is changed and recalculated, according the approach described in the section 6.3. The main particulars of the modified V-shape concept are included in Table 7-4.

7.6 Results V-shape concept design

The main particulars of both concept designs are presented in Table 7-4. Both concepts fulfil the required dimensions imposed by the mission equipment and the freeboard criteria. Table 7-4 presents that the length of the V-shape concept is longer, due to the smaller blockcoeffiecient the vessel was enlarger to meet the required deadweight. Phase 3 analyses the properties and the performance characteristics of both concepts.

Modified vessel particulars	U-shape	V-shape	Unit
Overall length	[redacted]	[redacted]	[m]
Overall breadth	[redacted]	[redacted]	[m]
Depth	[redacted]	[redacted]	[m]
Draft (transit)	[redacted]	[redacted]	[m]
Draft (lift operation)			[m]
Min slope of vessel sides	[redacted]	[redacted]	[˚]
Light ship weight	[redacted]	[redacted]	[mt]
VCG	[redacted]	[redacted]	[m]
LCG	[redacted]	[redacted]	[m]
TCG	[redacted]	[redacted]	[m]

Table 7-4: Main particulars of both concept designs

8 Intact stability

The stability is one of the most challenging design aspects of construction vessels which are equipped with a crane which has a large lifting capability. The vessel needs sufficient amount of stability to withstand the substantial overturning moment generated by the crane load. The stability calculations are performed in the software DelftShip. In this section the most governing and common load cases for both concept designs are considered. The stability calculations require the following input data:

- Input data
    - Definition of load cases
    - Definition of restrictions and stability criteria
    - Definition of weight and centre of gravity (KG)

The chapter unfolds as follows: first the load cases and restrictions are defined, with this the necessary amount of water ballast and centre of gravity of each case is determined. This input data is used to calculate and evaluated the intact stability. Based on these results, the V-shape hullform was modified and the stability recalculated. The obtained GM values were used to estimate the natural roll periods of both concepts.

8.1 Definition of load cases

The considered load cases are stated below.

Case no.	Operation	Loading	Consumables
	Lifting		
1		5000 [mt] at 34m over side	100 [%] cons
2		5000 [mt] at 34m over side	50 [%] cons
3		5000 [mt] at 40m over stern	100 [%] cons
4		5000 [mt] at 40m over stern	50 [%] cons
5		2000 [mt at 34 m over side	100 [%] cons
6		500 [mt] at 34 m over side	100 [%] cons
	Pipe lay (deep water)		
7		10,000 [mt] of 46 [inch] pipe storage	100 [%] cons
8		2,500 [mt] of 24 [inch] pipe storage	50 [%] cons
	Transit		
9		None	100 [%] cons
10		None	50 [%] cons
11		Attach stinger (pipelay ready)	100 [%] cons
12		9,000 [mt] cargo on maindeck	100 [%] cons

Table 8-1: Load case for which the stability is calculated and evaluated, used for both concepts designs

The choice for the load cases of Table 8-1 is discussed below.

- Load case 1 and 2:
  These load cases are the most critical for the transverse stability, as it includes the maximum OMC lift capacity over side. This generates the largest occurring overturning moment combined with the highest centre of gravity of the crane load.
- Load case 3 and 4:
  These load cases are the most critical cases for the longitude stability, as it includes the maximum OMC lift capacity over the stern. This generates the maximum occurring longitudinal moment and the highest centre of gravity of the crane load.
- Load case 5 and 6
  These cases are not governing for the stability but are considered to gain understanding about the stability properties during intermediate and light lift operations.
- Load case 7 and 8:
  Load case 7 is the most governing for the intact stability during pipe lay operations, as it includes the maximal pipe joint loading. Load case 8 is considered to gain understanding about the stability properties during pipe lay operations with a low pipe joint loading.
- Load case 9 and 12:
  Load case 9 and 10 includes the most common load cases during transit operations. Where, load cases 11 and 12 describes the most governing cases regarding the intact stability during transit operations.

8.2 Restrictions and stability criteria

This section presents restrictions imposed by the mission equipment and the applied stability criteria.

8.2.1 Restriction imposed by mission equipment

The vessel's maximum roll and pitch angle during lift and pipe lay operations are limited by the technical specification of the mission equipment. This criterion is defined by the equipment manufacture, to ensure that the maximum loading capacity during operating is not exceeded.

- Criteria of OMC
  Maximum allowable off and side lead are presented in Table 8-2

Item	Value	Unit
Offlead (including 3˚ static)	3.5	[deg]
Sidelead	3.5	[deg]

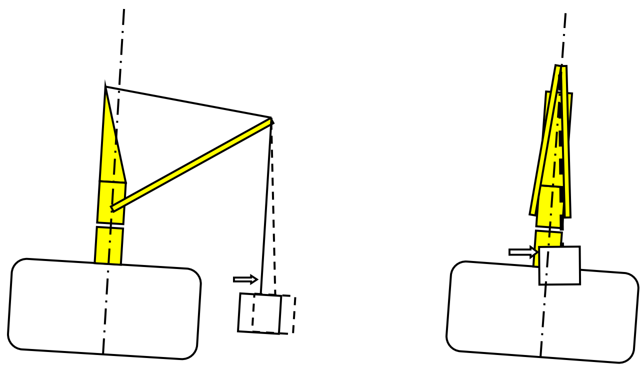
Table 8-2: OMC operational criteria

Where:

- Offlead: Cranes pulling angle (in longitude plane)
- Sidelead Cranes pulling angle (in transversal plane)

The maximum roll and pitch angle depends on the crane revolving angle as explained below.

- Crane revolving 90 degree (lift over side)
    - Maximum allowable roll is equal to: offlead angle
    - Maximum allowable pitch angle is equal to sidelead angle
- Crane revolving 180 degree (lift over stern)
    - Maximum allowable roll is equal to: sidelead angle
    - Maximum allowable pitch angle is equal to offlead angle



Offlead [deg] load away from the crane   Sidelead [deg] load transverse to the cranetip

Figure 8-1: Principle of offlead and sidelead

- Criteria by S-lay system
  Maximum vessel motions during pipe lay operations are presented in Table 8-3.

Operation mode	Max heel	Max trim	Max roll	Max pitch	Unit
Pipelay mode	[redacted]	[redacted]	[redacted]	[redacted]	[deg]
A&R mode	[redacted]	[redacted]	[redacted]	[redacted]	[deg]
Survival mode (no pipe lay)	[redacted]	[redacted]	[redacted]	[redacted]	[deg]

Table 8-3: Operation criteria of 600 [mt] S-lay system

To gain a large as possible workability, the vessels should be ballasted such that the initial heel and trim angle are zero or as small as possible. In order to maximize the vessel response during lift and pipe lay operations, within the criteria imposed by the mission equipment.

8.2.2 Intact stability criteria

The vessel should fulfil the intact stability criteria of the International Code in Stability (2008) and Det Norske Veritas (DNV). The applied criteria depend per operation, as explained below.

- During pipe lay operations: IMO special purpose ship (SPS) alternative code
  Because more than 12 persons "special personnel" on board. Special personnel are neither crew members nor passengers. Due to the shape of the vessel, a high B/D ratio, the alterative criteria, within the code, for the maximum GZ is used.
- During transit operations: IMO special purpose ship (SPS) alternative code
  Idem for a vessel undergoing pipe lay operations
- During lift operations: DNV alternative intact stability criteria during heavy lift operations

  In addition to this criteria Huisman Equipment requires a minimum GM: 3.5 [m]

Note that the entire stability criteria are included in appendix E.

8.3 Description of both models

This section describes the implementation of both concept designs in the software Delft ship. The dimensions presented in Table 7-7 are used as input for Delftship. The tank arrangements as set in section 6.3 for the U-shape and 7.4 for the V-shape are implemented in the software. The funnels of the bow and retractable thrusters are implemented as non-buoyancy tanks in the software.

A wind silhouette per load case was made in the program Delftship, to evaluate if the applied stability criteria are fulfilled. Figure 8-2 shows wind silhouette of the V-shape concept during a lift operation of 5000 metric tonnes over the stern.

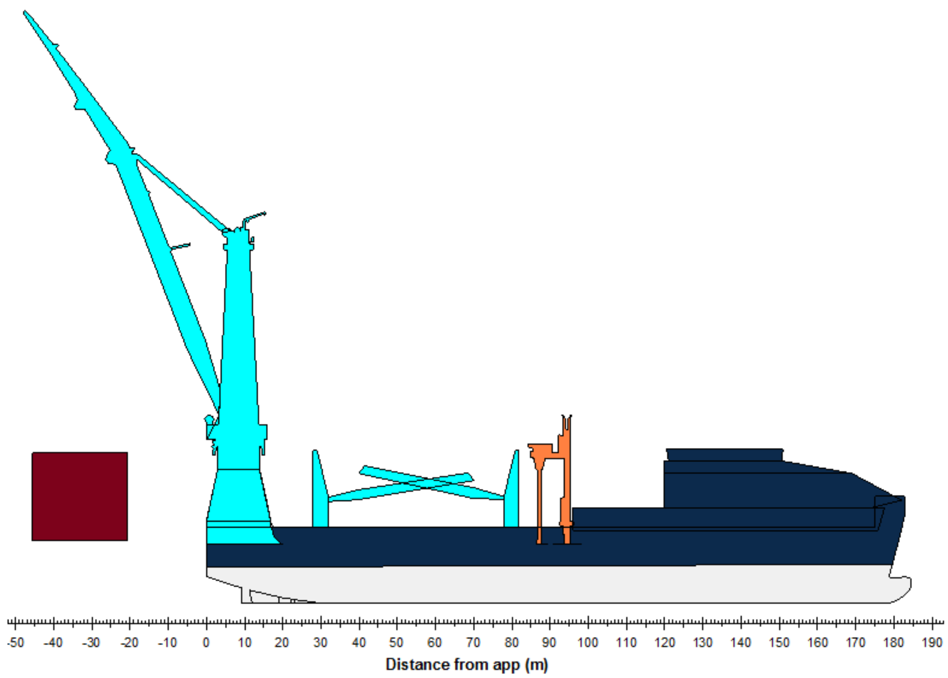


Figure 8-2: Wind silhouette as defined in the software Delftship, for lift over stern

8.4 Intact stability calculations

This section discusses the results of the intact stability of the V and U shape concept design. The intact stability is calculated for all load cases, as defined in Table 8-1, and evaluated with the stability criteria given in appendix E.

8.4.1 Intact stability during lift operations

During lift operations the vessel keeps position by use of the DP system and zero forward speed is assumed. Furthermore, the stinger is not connected, no deckload is present and the pipe loading cranes are in stowed position. The input data and results of the stability calculations are given below.

- **Input data for the stability calculations**

  To lower the vessels KG the lowest located water ballast tanks are particularly filled. The lowest tanks are used as this is the most efficient regarding the necessary amount of water ballast. Furthermore, to minimize the free surface effects it is intend to fill the water ballast tanks fully as much as possible. This philosophy was applied for both concepts. The amount of water ballast per considered lift case is included in Table 8-4. The displacement is included in Table 8.5 for the U-shape concept and in Table 8.6 for the V-shape concept.

Table 8-4, 8-5 and 8-6 are Confidential

Appendix F presents the deadweight and centres of gravity per weight group for both concepts.

- **Results**

  The results of the intact initial stability of both vessel concepts according to Delftship are presented in Table 8-7. The range of stability of all lift cases for the U-shape vessel is given in Figure 8-3 and in Figure 8-4 for the V-shape concept.

  For all considered lifting cases both vessels comply with the applied stability criteria.

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Figure 8-4: GZ curves of V-shape for all considered lift cases

8.4.2 Intact stability during pipe lay operations

In these operations the stinger is connected to the vessel and pipe joints are stored on main deck. And the offshore mast crane is not used and in stowed position During the production the pipe joints are supplied by vessels and transmited to the deck by the pipe loading cranes. The amount of storage is variable and depends on the pipe supply intensity. For pipe lay conditions the initial stability is reduced as much as possible, to gain good motion behaviour, as was explained in section 1.2.2. The input data and results of the stability calculations are given below.

- **Input data for the stability calculations**

  Both vessels are ballast to keep the trim angle as small as possible, to compensate the stinger weight. Reducing the initial stability is achieved for each concept in a different manner, as explained below.
  - U-shape concept

    Ballast tanks located above the KG are filled to reduce the vessels KG and with this the GM height.
  - V-shape concept

    The draft is reduced as much as possible to gain a narrow as possible waterline, to reduce the KM and so the GM value.

  The deadweight loading per considered pipe lay case is given in the Table 8-8. The displacement is known and included in Table 8-9 for the U-shape concept and in Table 8-10 for the V-shape concept. Tables: 8-8, 8-9 and 8-10 are confidential

Appendix F includes the deadweight and centres of gravity per weight group for both concept designs.

- **Results**
  The results of the intact initial stability of both vessel concepts according to Delftship are presented in Table 8-11. The range of stability of all lift cases for the U-shape vessel is given in Figure 8-5 and in Figure 8-6 for the V-shape concept.

  For all considered pipe lay operations both vessels comply with the applied stability criteria.

Case no.	U-shape			V-shape		
	Heel angle [deg]	Trim [deg]	GM [m]	Heel angle [deg]	Trim [deg]	GM [m]
7	0.0	-0.5	[redacted]	0.0	-0.1	[redacted]
8	0.0	0.1	[redacted]	0.0	-0.1	[redacted]

Table 8-11: Intact stability of both concept designs during pipe lay operations

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Figure 8-5: GZ curves of U-shape for all considered pipelay cases

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Figure 8-6: GZ curves of V-shape for all considered pipelay cases

8.4.3 Intact stability during transit operations

During transit the 5000 metric tonnes offshore mast crane and the pipe loading cranes are in stowed position. The stinger is only attached in transit operations from or towards a pipe lay operations. For other transit cases the stinger is not connected to the vessel. During transit the initial stability is reduced as much as possible by tuning the loading condition, according the approach given in section 8.4.2. However the amount of water ballast is limited by the design draft, to enhance the hull resistance.

- **Input data for the stability calculations**
  The amount water ballast per considered transit case is given in the Table 8-12.

Case no.	Loading	U-shape Ballast [mt]	V-shape Ballast [mt]
9	None	[redacted]	[redacted]
10	None	[redacted]	[redacted]
11	Attached stinger	[redacted]	[redacted]
12	9,000 [mt] cargo load	[redacted]	[redacted]

Table 8-12: Amount of water ballast for the considered transit cases

The deadweights and centres of gravity is determined and used as input for the software Delftship. Hence the displacement is known and included in Table 8-13 for the U-shape concept and in Table 8-14 for the V-shape concept. Appendix F includes the deadweight and centres of gravity per weight group for both concept designs.

Case no.	Item	Weight [mt]	LCG [m]	TCG [m]	VCG [m]
9	Displacement	[redacted]	[redacted]	0.0	[redacted]
10	Displacement	[redacted]	[redacted]	0.0	[redacted]
11	Displacement	[redacted]	[redacted]	0.0	[redacted]
12	Displacement	[redacted]	[redacted]	0.0	[redacted]

Table 8-13: Definition of displacements and centres of gravities during transit for U-shape concept

Case no.	Item	Weight [mt]	LCG [m]	TCG [m]	VCG [m]
9	Displacement	[redacted]	[redacted]	[redacted]	[redacted]
10	Displacement	[redacted]	[redacted]	[redacted]	[redacted]
11	Displacement	[redacted]	[redacted]	[redacted]	[redacted]
12	Displacement	[redacted]	[redacted]	[redacted]	[redacted]

Table 8-14: Definition of displacement and centres of gravities during transit for V-shape concept

- **Results**

  The results of the intact initial stability of both vessel concepts according Delftship are presented in Table 8-15. The range of stability of all lift cases for the U-shape vessel is given in Figure 8-7 and in Figure 8-8 for the V-shape concept.

  For all considered transit cases both vessels comply with the applied stability criteria.

Case no.	U-shape			V-shape		
	Heel angle [deg]	Trim [deg]	GM [m]	Heel angle [deg]	Trim [deg]	GM [m]
9	0.0	-0.0	[redacted]	0.0	-0.0	[redacted]
10	0.0	-0.0	[redacted]	0.0	-0.0	[redacted]
11	0.0	-0.0	[redacted]	0.0	-0.2	[redacted]
12	0.0	-0.1	[redacted]	0.0	-0.2	[redacted]

Table 8-15: Intact stability of both concepts during transit operations

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Figure 8-7: GZ curves of U-shape for all considered transit cases

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Figure 8-8: GZ curves of V-shape for all considered transit cases

8.4.4 Conclusions on intact stability calculations

By analysing the results, the following points are noted:

- U-shape concept
  The GM height during 5000 metric tonnes lifts operations remains just within the Huisman GM requirement of [redacted] meter. Due to this, optimization is not needed and possible for the U-shape concept from stability perception. The range of stability complies easily with the applied stability criteria.
- V-shape concept
  The range of stability during 5000 metric tonnes lifts operations remains just within the stability criteria. In order to satisfy these criteria the stability is maximized, resulting in a relative high GM, with respect to the Huisman GM requirement.

The V-shape dimensions will be modified, to improve the stability property in heavy lift conditions. Section 8.5 presents the modification of V-shape concept.

8.5 Modification of the V-shape concept

The aim of the modification is to reduce the KM and with this the GM during lift operations. This section describes the modification and presents the vessel particulars and the calculated intact stability properties of the changed hullform.

By reducing the overall breadth and lowering the node height, the initial stability and range of stability will be reduced. In the pipe lay and transit operations the waterline breadth and the buoyancy is equal to the unmodified V-shape. The modification is visualized in Figure 8-9, where the dash line presents the more slender hullform.

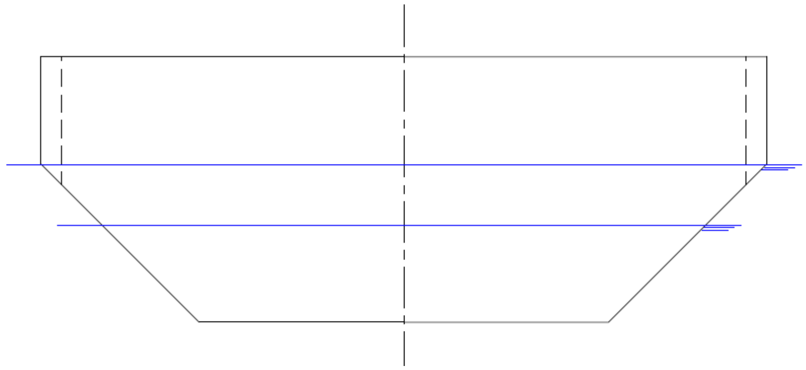

Figure 8-9: The dash line visualise the modification of mid-ship section of the V-shape concept

As a consequence of this modification the lightship weight, arrangement and the KG is changed and therefore recalculated. Section 8.5.1 determines if the changed V-shape concept still complies with the applied stability criteria. Table 8-16 presents these main particulars.

Vessel particulars	U-shape	V-shape	Unit
Overall length	[redacted]	[redacted]	[m]
Overall breadth	[redacted]	[redacted]	[m]
Depth	[redacted]	[redacted]	[m]
Draft (transit)	[redacted]	[redacted]	[m]
Draft (lift operation)			[m]
Min slope of vessel sides	[redacted]	[redacted]	[deg]
Light ship weight	[redacted]	[redacted]	[mt]
VCG	[redacted]	[redacted]	[m]
LCG	[redacted]	[redacted]	[m]
TCG	[redacted]	[redacted]	[m]

Table 8-16: Modified vessel based on freeboard calculations

The lightship is calculated according the method as presented in section 5.5. Note that the U-shape concept is unmodified with respect to the Table 7-8 and is only presented to show the difference between them.

8.5.1 Intact stability of modified V-shape concept

This section presents the input data and results of the intact stability calculations for all load cases as included in Table 7-1, is given below.

- <u>Input data for the stability calculations</u>
  The amount of water ballast for all load cases is presented in Table 8-17 and is determined according the approach stated in section 8.4. The displacement and the centres of gravities is given in Table 8-18.

Case no.	Operation	Loading	U-shape Ballast [mt]	V-shape Ballast [mt]
	Lifting			
1		5000 at 34m	[redacted]	[redacted]
2		5000 at 34m	[redacted]	[redacted]
3		5000 at 40m	[redacted]	[redacted]
4		5000 at 40m	[redacted]	[redacted]
5		2000 at 34m	[redacted]	[redacted]
6		500 at 34m	[redacted]	[redacted]
	Pipe lay			
7		10 000 [mt] pipe joints	[redacted]	[redacted]
8		2500 [mt] pipe joints	[redacted]	[redacted]
	Transit			
9		None	[redacted]	[redacted]
10		None	[redacted]	[redacted]
11		Attached stinger	[redacted]	[redacted]
12		9000 [mt] cargo load	[redacted]	[redacted]

Table 8-17: Amount of water ballast for all the considered stability cases

Case no.	Operation	Weight [mt]	LCG [m]	TCG [m]	VCG [m]
	Lifting				
1		[redacted]	85.8	0.0	21.5
2		[redacted]	84.3	0.0	21.5
3		[redacted]	85.1	0.0	23.7
4		[redacted]	87.7	0.0	23.5
5		[redacted]	86.4	0.0	22.3
6		[redacted]	86.8	0.0	19.7
	Pipe lay				
7		[redacted]	85.3	0.0	17.5
8		[redacted]	85.3	0.0	17.5
	Transit				
9		[redacted]	88.3	0.0	16.9
10		[redacted]	88.3	0.0	17.5
11		[redacted]	87.3	0.0	17.3
12		[redacted]	86.4	0.0	21.0

Table 8-18: Definition of displacement and centres of gravities for the modified V-shape concept

- Results
  The results for of intact stability calculations of the modified V-shape concept design are presented in Table 8-19. The calculated range of stability is shown in figures 8-10, 8-11 and 8-12. The vessels complies for all load cases with the applied stability criteria. Through comparing the GM value during lifting 5000 metric tonnes (Table 8-7) of the V-shape concept before the modification with GM values presented in table 8-19. It is concluded that the intact stability is reduced as a result of the modification.

Case no.	Operation	U-shape	V-shape		
		GM [m]	Heel angle [deg]	Trim [deg]	GM [m]
	Lifting				
1		[redacted]	0.0	0.2	[redacted]
2		[redacted]	0.0	-0.2	[redacted]
3		[redacted]	0.0	-0.3	[redacted]
4		[redacted]	0.0	0.1	[redacted]
5		[redacted]	-0.1	-0.0	[redacted]
6		[redacted]	0.0	-0.0	[redacted]
	Pipe lay				
7		[redacted]	0.0	-0.3	[redacted]
8		[redacted]	0.0	0.0	[redacted]
	Transit				
9		[redacted]	0.0	0.1	[redacted]
10		[redacted]	0.0	0.0	[redacted]
11		[redacted]	0.0	-0.0	[redacted]
12		[redacted]	0.0	-0.0	[redacted]

Table 8-19: Intact stability of modified V-shape concept of all considered cases

Note that the results of the U-shape vessel in this section are the same as prior presented in section 5-4. This data is only added in Table 8-19 to present the difference in stability properties.

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Figure 8-10: GZ curves of modified V-shape concept for all considered lift cases

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Figure 8-11: GZ curves of modified V-shape concept for all considered pipe lay cases

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Figure 8-12: GZ curves of modified V-shape concept for all considered transit cases

8.6 Natural roll period estimation

Using the calculated initial stability, GM height, the natural roll periods for all considered load cases was estimated. This gain understanding if the vessels have different motion characteristics with respect to each other. The natural roll period is estimated by using the equation 8-1. Note that in equation 8-1 the radius of gyration and the added mass are estimates.

$$T_{roll} = 2\pi\sqrt{\frac{(1+a)\cdot k_{xx}^2}{GM\cdot g}}$$

Where:

k_{xx}	:	Radius of gyration as 39% of the breadth
a	:	Added mass assumed as 40% of displacement

Results of the roll estimation for all twelve load cases are given in Table 7-18.

Case no.	Operation	Roll period U-shape [sec]	Roll period V-shape [sec]	Deviation [%]
	Lifting			
1		[redacted]	[redacted]	[redacted]
2		[redacted]	[redacted]	[redacted]
3		[redacted]	[redacted]	[redacted]
4		[redacted]	[redacted]	[redacted]
5		[redacted]	[redacted]	[redacted]
6		[redacted]	[redacted]	[redacted]
	Pipe lay			
7		[redacted]	[redacted]	[redacted]
8		[redacted]	[redacted]	[redacted]
	Transit			
9		[redacted]	[redacted]	[redacted]
10		[redacted]	[redacted]	[redacted]
11		[redacted]	[redacted]	[redacted]

Table: 8-20: Estimated natural roll period of both concepts and the deviation of V-shape with respect to the U-shape

By analysing the results of Table 8-20 it can be noted that the roll motion characteristics for intermediate and light lift operations and transit operations are significantly different for each concept design. In these cases the operational performance is probably differently and could distinct the workability of each concept.

8.7 Conclusion

The stability of both concept designs has been calculated and evaluated for the most common and governing cases. Both vessels comply for all considered load cases with the applied stability criteria. Based on the stability analysis, the V-shape concept has been modified. By analysing the results, certain points can be noted:

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- Significant more water ballast is necessary to adjust the stability properties of the U-shape concept in pipe lay and transit operations with respect to the V-shape concept.
- It is observed that the draft of V-shape concept design significantly larger during lifting operations than for a U-shape vessel. This result most likely in a smaller versatility for V-shape, as lifting operations in shallow water areas is impossible. However, in reality lifting operations are mainly offshore performed i.e. deep draft areas
- The estimate natural roll periods presented in Table 8-20 shows a significant difference between the concepts occur in the lift load case 5 and 6 and for all transit cases.

PHASE III

PERFORMANCE ANALYSIS

The still water resistance of both concepts is calculated. To determine if a multi-purpose construction vessel with a V-shape has a lower hull resistance than one with a U-shape hullform. The still water resistance is compared as it is expected, according to phase I, that a V-shape reduces the hull resistance characteristics. The Holtrop & Mennen theory is used is to determine the hull resistance. This section unfolds as follow: the Holtrop & Mennen theory is validated against model test results to investigate if this method is applicable for the considered vessel types. The still water resistance characteristic of both vessel concepts is predicted according to this method and presented in this chapter.

9.1 Validation of used prediction method

The Holtrop & Mennen theory is a statistical power prediction method, based on the regression analysis of random model and full-scale test data together with, in the latest version of the method, the published results of the Series 64 high speed displacement hull terms [ref. 12]. The total resistance coefficient is divided into five components, see equation 9-1.

$$C_T = (1+k)C_F + C_w + C_{tr} + C_b + C_a \tag{9-1}$$

Where:

C_f	:	Friction resistance coefficients	[-]
C_w	:	Wave resistance coefficients	[-]
C_{tr}	:	Additional resistance due to transom immersion	[-]
C_b	:	Additional resistance due to bulbous bow	[-]
C_a	:	Appendage drag (bilge keels, rudder etc.)	[-]

The Holtrop & Mennen theory provides a useful estimation tool for designers. However, like many analysis procedures it relies to a very large extent on traditional naval architectural parameters. As these parameters cannot fully act as a basis for representing the hull curvature and its effect on the flow around the vessel there is natural limitation on the accuracy of the approach without using more complex hull definition parameters [ref. 13]. Moreover, the developed concept designs are both characterized by a low length breadth ratio, besides the V-shape concept has special hullform. It is therefore analysed if the statistical Holtrop & Mennen theory is applicable for the created concept designs. This is achieved by means of a validation of the analytical results against measured data coming from the model tests. This is done for both hull type based on reference vessels as presented below.

- <u>U-shape</u>
  The still water resistance model test of reference U-shaped construction vessel available within Huisman Equipment – with comparable mission and dimensions - is used to validate the Holtrop & Mennen prediction method.
- <u>V-shape</u>
  The still water resistance model test of the Huisman Equipment V-shape concept is used to validate the prediction method for this type of vessels.

Only the still water resistance of the vessel in design draft is considered. The used input data for the validation analysis is given in Table 9-1. The analytical results versus the model test results are presented in Figures 9-1, 9-2 and 9-3. Note that the breadth on the waterline is used in the Holtrop & Mennen theory if the hull resistance is calculated for a V-shape hull.

Item	U-shape	Huisman crane vessel	Unit
Length overall	[redacted]	[redacted]	[m]
Water line length	[redacted]	[redacted]	[m]
Position of COB	[redacted]	[redacted]	[m]
Breadth moulded	[redacted]	[redacted]	[m]
Breadth waterline	[redacted]	[redacted]	
Draught fore	[redacted]	[redacted]	[m]
Draught aft	[redacted]	[redacted]	[m]
Block coefficient	[redacted]	[redacted]	[-]
Volume displacement	[redacted]	[redacted]	[m^3]
Water plane area	[redacted]	[redacted]	[m^2]
Water plane coefficient	[redacted]	[redacted]	[-]
Mid-ship coefficient	[redacted]	[redacted]	[-]
Mid-ship section area	[redacted]	[redacted]	[m^2]
Immersed part of transom	[redacted]	[redacted]	[m^2]
Wetted area	[redacted]	[redacted]	[m^2]
Half angle of entrance	[redacted]	[redacted]	[deg]
Stern shape	[redacted]	[redacted]	[-]
Abt surface*	[redacted]	[redacted]	[m^2]
Hb**	[redacted]	[redacted]	[m]
Skeg surface	[redacted]	[redacted]	[m^2]
Bilge keel	[redacted]	[redacted]	[m^2]
Numbers of bow thrusters	[redacted]	[redacted]	[-]
Diameter of bow thrusters	[redacted]	[redacted]	[m]

Table 9-1: Input data of the Holtrop & Mennen validation analysis

*Transverse sectional area of bulb at position where the stillwater surface insects stern
**Position of the centre of transverse area Abt above the keel line

The still water hull resistance of the U-shape vessel predicted by the Holtrop & Mennen method, corresponds well to the hull resistance of the model test, see Figure 9-1. Furthermore, the hullform and main particular of the reference vessel show sufficient similarity with the U-shape concept design. Based on this comparison, it is concluded that the Holtrop & Mennen method can be used to estimate the still water hull resistance of the U-shape concept.

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Figure 9-1: Resistance characteristics prediction of different methods, U-shape

[kn]	Holtrop [kN]	Model test [kN]	Deviation [%]
11	[redacted]	[redacted]	[redacted]
12	[redacted]	[redacted]	[redacted]
13	[redacted]	[redacted]	[redacted]
14	[redacted]	[redacted]	[redacted]
15	[redacted]	[redacted]	[redacted]
16	[redacted]	[redacted]	[redacted]
17	[redacted]	[redacted]	[redacted]

Table 9-2: Deviation between the two different prediction methods

The hull resistance prediction based on the Holtrop & Mennen method of the Huisman V-shape is presented in Figure 9-2. For sailing velocities up to circa 13 knots, good similarity with the model test results is shown. However, the Holtrop & Mennen method overestimates the hull resistance for velocities faster than, circa, 13 knots. In order to investigate the origin of this deviation, the hull resistance of the Huisman V-shape for deep draft is considered. In this loading condition the waterline breadth is almost the same as the overall breadth. Figure 9-3 compares the measured the hull resistance of the Huisman V-shape at a deep draft of 9.0 meter to the Holrop & Mennen method.

The hull resistance prediction according to the Holtrop & Mennen method shows good agreement to the measured data of the model test, see Figure 9-3. The effect of the smaller V-shape midship section area compared to the L/B ratio on the wave resistance, seems under estimated by the Holtrop & Mennen method. This is most likely because the ratios for the V-shape concept are outside the applicability range of the Holtrop & Mennen method.

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[kn]	Holtrop [kN]	Model test [kN]	Deviation [%]
8	[redacted]	[redacted]	[redacted]
10	[redacted]	[redacted]	[redacted]
12	[redacted]	[redacted]	[redacted]
14	[redacted]	[redacted]	[redacted]
16	[redacted]	[redacted]	[redacted]

Table 9-3: Deviation between the two different prediction methods

Figure 9-2: Resistance characteristics prediction of different methods, V-shape

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Figure 9-3: Resistance characteristics prediction of two methods, V-shape (deep draft)

	Holtrop	Model test	Deviation
[kn]	[kN]	[kN]	[%]
8	[redacted]	[redacted]	[redacted]
10	[redacted]	[redacted]	[redacted]
12	[redacted]	[redacted]	[redacted]
14	[redacted]	[redacted]	[redacted]
16	[redacted]	[redacted]	[redacted]

Table 9-4: Deviation between the two different prediction methods

To account for this over estimation a correction factor was determined and applied for the still water hull resistance of the V-shape concept design. The correction factor describes the deviation in percentage as a function of the velocity. By means of multiplying this percentage with the estimated resistance, the Holtrop & Mennen method was corrected. Note that this correction is based on one resistance test result. Therefore the accuracy of the correction factor for the V-shape concept design was not verified. However, it is assumed that the correction factor is applicable for the V-shape design concept. Since, the dimensions and the hullform of V-shape concept show significant similarity with the Huisman V-shape concept.

9.2 Calculation of still water resistance

The still water hull resistance is calculated for the transit operations (shallow draft) i.e. load case 10. The resistance due to wind and current is neglected in the calculations. The still water hull resistance characteristic of both concepts designs is presented in Figure 9-4.

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Figure 9-4: Still water resistance characteristics of U-shape concept design versus V-shape concept designs

The Table 9-5 presents the difference in still water hull resistance between the U-and V-shape concept design.

Sailing speed [kn]	U-shape [kN]	V-shape [kN]	Deviation [%]
8	[redacted]	[redacted]	[redacted]
10	[redacted]	[redacted]	[redacted]
12	[redacted]	[redacted]	[redacted]
14	[redacted]	[redacted]	[redacted]
15	[redacted]	[redacted]	[redacted]
16	[redacted]	[redacted]	[redacted]
17	[redacted]	[redacted]	[redacted]

Table 9-5: Calculated still water resistance of both concept designs

9.3 Conclusion on the still water characteristics

By analysing the results of Figure 9-4, it can be concluded that the still water resistance for the V-shape concept is lower for sailing speed upward of ± 13.0 knots, than for the U-shape concept design. This enables to sail faster if the same amount of propulsion power is installed. The reduced resistance also results in lower fuel consumption during the transit operations at the design velocity of 15.0 knots. Based on the present analysis, the still water resistance at the design speed is approximately [redacted] lower for the V-shape concept.

The preference for a higher sailing speed or reduced fuel consumption during a transit operation depends on the operation profile or the preference of a marine contractor. It is assumed that the lower still water resistance does not lead to a reduced installed engine power. This in general defined by the required power to fulfil the DP capability requirements.

In this chapter the results of the performed motion behaviour and workability analysis of the two concept designs, as developed in phase two, are presented. This analysis is conducted, as the estimated natural roll periods shows a significant difference between the U- and V-shape concept designs. The estimated natural roll periods are based on the calculated GM values, as previously described in chapter 8. Additionally, a larger roll damping is expected for the V-shape concept design, as previously described in section 2.3

Linear Response Amplitude Operators (RAO's) are used to calculate the response of the vessel in irregular waves. The RAO's are calculated with AQWA-LINE, a diffraction radiation program in the frequency domain. This chapter unfolds as follow:

Using model test results of irregular wave experiments, the applicability of linear RAO's and analytical calculated viscous roll damping for vessels with a V-shape hull was validated. The input data and assumptions for the diffraction radiation software AQWA-line is set and presented. The most probable maximum response for both concepts per load case and environmental condition was calculated. By means of applying the set operational criteria, the workability for the Gulf of Mexico, West coast of Africa and East coast of Brazil was analysed.

10.1 Validation of linear RAO's

Although the linear RAO's of a vessel are calculated for the initial position of the vessel, the geometric of water plane area changes with the vessel motion. The motion characteristics depends therefore on the vessel motion i.e. RAO's are not linear. However in general it is assumed that this nonlinear effect is small and can be neglected. It needs to be determined if the assumption holds also for a V-shape hull. The applicability of linear RAO's calculated by AQWA-LINE for vessels with a V-shape was therefore validated, using model test results of the Huisman V-shape crane vessel.

The Huisman V-shape crane vessel was modelled in the AQWA-LINE diffraction radiation software to calculate the RAO's. By means of comparing these obtained RAO's against RAO's calculated from measured data coming from model tests [ref. 14] the linear RAO's of AQWA-LINE was validated. This section presents the input, assumptions and result of this analysis.

Note that this analysis was only conducted for the V-shape concept, as calculating the motion response for vessels with a U-shape hullform by means of linear RAO's is a common engineering practice. Linear RAO's are therefore assumed applicable for the U-shape concept design.

10.1.1 Input and assumptions

The dimensions of the Huisman V-shape crane vessel are included in Table 10-5 and used as input data for the AQWA-LINE model. The vessel is modelled on full scale, as the model test results are also presented on full scale. Only the transit loading condition was considered, as in this condition the waterline area changes most with the vessel motion, i.e. the non-linear effect is maximum. The used loading condition is given in Table 10-1.

The Huisman V-shape crane vessel has large bilge keels, which introduces a significantly amount of viscous damping. As AQWA-LINE does not account for viscous damping this will lead to an overestimation of the responses near the natural periods. However, Frequency Independent Additional Diagonal Added Damping (FIDD), can be added in to the AQWA-LINE. The additional damping obtained from the model test [ref. 14] was used and directly added in to AQWA-LINE. The following amount of additional damping was added:

- FIDD: 1.53E+9 $[\mathrm{Nm}\frac{\mathrm{rad}}{\mathrm{sec}}]$

Item		Unit
Draft	[redacted]	[m]
Displacement		[mt]
GM		[m]
Kxx		[m]
Kyy		[m]
kzz		[m]

Table 10-1: Loading condition Huisman V-shape crane vessel

All used model tests are listed in Table 10-2. The presented test numbers refers to the model tests and wave conditions.

Marin Test no.	Significant height [m]	Peak Period [s]	Heading [deg]
214001	[redacted]	[redacted]	90.0
213001			90.0
206001			180.0

Table 10-2: Description of model tests

The goal of each performed tests is given in Table 10-3.

Marin Test no.	Goal of tests
214001	Validate heave RAO’s
213001	Validate roll RAO’s
206001	Validate pitch RAO

Table 10-3: Used model tests

10.1.2 Results models

Using AQWA-LINE, the RAO's were calculated and compared against RAO's obtained in the seakeeping model tests. The free floating natural periods according to each calculation method is given in Table 10-4. Figures 10-1 to 10-3 presents the RAO's derived by AQWA versus model test results.

Motion	Model test results [sec]	AQWA-LINE [sec]	Heading [deg]
Heave	[redacted]	[redacted]	[redacted]
Roll	[redacted]	[redacted]	[redacted]
Pitch	[redacted]	[redacted]	[redacted]

Table 10-4: Natural periods according to model test versus AQWA-LINE

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Figure 10-1: Heave RAO derived by AQWA versus model test results, heading: 90 degree.

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Figure 10-2: Roll RAO derived by AQWA versus model test results, heading 90 degree

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Figure 10-3: Pitch RAO derived by AQWA versus model test results, heading 180 degree.

10.1.3 Conclusion on validation of RAO's

The Figures 10.1 to 10.3 show sufficient similarity between the AWAQ-LINE and model test results. Based on this; linear RAO's calculated by AQWA-LINE are considered suitable as input for the irregular wave response calculations for a V-shape hull.

10.2 Validation roll damping prediction method

Both concept designs are equipped with bilge keels which introduces a significant amount of viscous damping. The bilge keels are not modelled in the software. Furthermore, AQWA-LINE does not account for viscous damping of the vessel. This damping needs to be predicted and added in the software, to gain results as accurately as possible.

The predicting method developed by Ikeda et al. – published in four papers in (1976 – 1978) – is used to predict the roll damping. This empirical method is well known as the "Ikeda method". The original method for predicting the roll damping of ships was developed for conventional hull shape of cargo ships, with block coefficients of around 0.56-0.85, and Froude number up to 0.25 [ref. 15]. The method have been modified to improve the accuracy and to extend their applicability to other vessel types. However, the concept designs considered in this study are featured by specific hull dimensions. Due to this, the predicated roll damping is validated against model test results, to determine the applicability of the Ikea method for the considered vessel types.

10.2.1 Used numerical prediction method of roll damping

The Ikeda method assumes that the total roll damping can be divided in to the following components [ref. 16]:

$$^{\text{ii}}B_{\varphi\varphi} = B_F + B_E + B_L + B_W + B_{BK} \qquad [\mathrm{Nm}\frac{\mathrm{rad}}{\mathrm{sec}}] \qquad \{10\text{-}1\}$$

Where:

$B_F =$ Friction damping
$B_E =$ Eddy damping
$B_L =$ Lift damping
$B_s =$ Correction of the potential roll damping in presence of forward velocity
$B_{BK} =$ Bilge keel damping
$B_W =$ Wave making component

All these components are empirical and can be derived from equations, the total method is included appendix G. Ikeda, Himeno and Tanaka claim a good similarity between their method and experimental results [ref. 15]. The components are briefly described below.

- <u>Friction damping</u>
  This damping is obtained by the friction caused by the skin of the hull. The contribution of the bilge keel and the waves, obtained by the vessel was ignored. The friction is considered constant with the roll amplitude, but linear depended on the angular velocity. The friction component accounts for between 1 and 3 percent of the total roll damping.
- <u>Eddy damping</u>
  At zero forward speed, the eddy making component for a naked hull is mainly caused by sectional vortices around the bow, stern and bilge radius [ref. 17]. Eddy damping is proportional to the square of both roll frequency and the roll amplitude. The eddy damping generated by the bilge keel is here not included. This damping depends on the hull shape – B/D ratio and section area coefficient – and is determined through using the strip theory.
- <u>Lift damping</u>
  Since the lift force acts on the vessel hull moving forward with sway motion, it can therefore be concluded that a lift effect occurs for vessels during roll motion as well [ref. 16]. This damping component is zero at absence of forward speed. This component is not considered, as the vessel operates at zero or small forward velocities. Note that transit operations are not included in this chapter.
- <u>Bilge keel damping</u>
  Bilge keels contribute to the major roll damping at zero forward speed. The bilge keel damping represents the increment of pressure damping due to the presence of bilge keels [ref. 16]. A distinction is made between the following three components:
    - The normal force on the bilge keels themselves.
    - Hull pressure caused by the bilge keels: including pressure change on the hull when bilge keels are installed (interaction).
    - Wave damping of the bilge keels: represents the change of above terms due to waves.

  The strip theory was used to determine this damping, as bilge keel hull interaction depends on the hull shape.

10.2.2 Method of validation

The validation is performed using model test results:

- <u>U-shape design</u>
  Model test results were made available by Huisman Equipment of the U-shaped drill vessel, Globetrotter, [ref. 18]. Note that the L/B ratio of this vessel is larger as of U-shape concept design in this study but represents the best available model test data within the company Huisman Equipment.
- <u>V-shape design</u>
  The model test data of Huisman V-shape crane vessel concept [ref. 14] was used. The dimensions of this vessel corresponds well with the develop V-shape concept design.

Roll damping depends on the roll amplitude and roll velocity. However, roll damping must be linearized in frequency domain calculations. The roll damping value is linearized per sea spectrum and per loading condition as follows:

- Using AQWA-LINE, the RAO's without additional linear damping *FIDD* are calculated. These RAO's are used as input data for the frequency domain motion response calculations of AQWA-FER. The obtained, overestimated, roll amplitude is used to calculate the roll damping for that particular amplitude. This linear roll damping value is added into AQWA-LINE and AQWA-FER recalculates the roll amplitude for a particular wave heading and spectrum. This iterative progress is repeated until the roll damping converts to a constant value. This linearized roll damping value is only applied for one particular loading condition and wave spectrum. This approach is visualized in Figure 10-4.

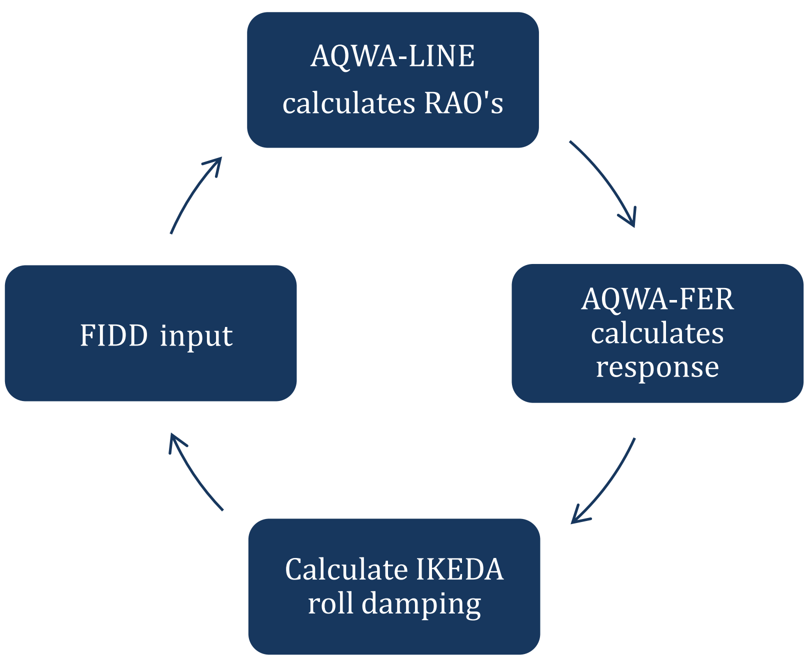

Figure 10-4: The progress of obtaining linear roll damping

The V-shape the roll damping is linearized using the significant roll response. The U-shape vessel the roll damping is linearized using the probable maximum roll response during 3 hours simulation. Because it was observed that the Ikeda method under estimated the roll damping for the U-shape drill vessel. According to the validation analysis it is decided to use different method of prediction for the roll damping. An accurate prediction method is preferred over the use of the same prediction method.

10.2.3 Input data

Table 10-5 presents the particulars of the vessels used to validate the RAO's and Ikeda roll damping. Both are modelled on full scale in AQWA-LINE as the model test results are also presented on full size scale.

Item	Globetrotter	Huisman crane vessel	Unit
LOA	[redacted]	[redacted]	[m]
Breadth	[redacted]	[redacted]	[m]
Depth	[redacted]	[redacted]	[m]
Draft	[redacted]	[redacted]	[m]
Deadweight	[redacted]	[redacted]	[mt]
Lightweight	[redacted]	[redacted]	[mt]
Displacement	[redacted]	[redacted]	[mt]

Table 10-5: Vessel dimensions as used for the validation of the Ikeda roll damping

10.2.4 Results and conclusions of predicted roll damping

Figure 10-5 and 10-6 present the roll RAO's with additional Ikeda damping versus the RAO's obtained by the model tests. The analytical results show some difference with the natural roll period. This difference is presumably caused by the absence of bilge keels in the AQWA model, as they introduce a significant amount of added mass. Hence, the lack of added mass in the model results in a longer natural roll period. Figure 10-5 and 10-6 shows sufficient similarity with the model test results. Accordingly, the linearized damping by Ikeda was considered sufficient accurate, to calculate the roll responses imposed by irregular waves.

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Figure 10-5: U-shape roll RAO, model test versus AQWA-LINE with linearized analytical damping

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Figure 10-6: V-shape roll RAO, model test versus AQWA-LINE with linearized analytical roll damping

10.2.5 The effect of bilge keels on roll damping.

In phase I it was stated that the roll damping of V-shape hull is smaller compared to a U-shape hull. A suggested solution, see section 2.3, is to improve the roll damping by accommodate the V-shape hull with large bilge keels, since the geometry allows this. The roll damping for the V-shape concept design during transit operations (loadcase 10) is calculated with 'normal' sized and larger bilge keels and presented in Table 10-6. The roll damping is predicted according to the method previously described in this section. The bilge keels have the following dimensions:

- "Normal" sized bilge keels: cord length of 0.75 meter and length of 115 meter
- "Large" sized bilge keels: cord length of 2.2 meter and length of 115 meter

Bilge keel configuration	Visc. damping*	Pot. Damping**	Unit
V-shape (normal bilge keels)	[redacted]	[redacted]	[Nm·rad/s]
V-shape (large bilge keels)	[redacted]	[redacted]	[Nm·rad/s]
U-shape (normal bilge keels)	[redacted]	[redacted]	[Nm·rad/s]

Table 10-6: Difference in roll damping for different bilge keel configurations

*Viscous damping according to Ikeda method
**Potential damping according to AQWA

Based on the presented results in Table 10-6 it is concluded that the roll damping of the V-shape concept significantly improves if bilge keels with a cord length of 2.2 meter are installed. Furthermore, Table 10-6 show that the potential damping of the V-shape concept is significantly smaller compared to the U-shape concept. This give the reason to install large bilge keels, as was expected in section 7.2.

10.3 Motion behaviour analysis of both concept designs

This section presents the input, assumptions, used calculation method and results of the performed motion behaviour analysis of both concept designs.

10.3.1 Input and assumptions

The motion behaviour is analysed and considers the following modes of operations:

- Lifting 5000 metric tons at 34 meter over side
  To check if both vessel concepts comply with the operational requirements, as stated in section 3.7.2

- Lifting 2000 metric tons at 34 meter over side
  Represents an intermediate lift operation, i.e. load case: 5 as stated in section 6.1

- Lifting 500 metric tons at 34 meter over side
  Represents a light lift operation, i.e. load case: 6 as stated in section 6.1

- Pipe lay operation with a 100% pipe joint storage
  Presents the heaviest pipe lay loading condition, i.e. load case: 7 as stated in section 6.1

- Pipe lay operation with a 80% pipe joint storage
  Presents a typical pipe lay operation, i.e. load case 8 as stated in section 6.1

- Transit operation, loaded with 50% consumables
  Presents a typical transit operation if the vessel "act" as crane vessel

The capability to lift 5000 metric tons in the environmental conditions of Table 10-7 was required for both concept designs, as described in chapter 3.

Items		Unit
Significant wave height (H⅓)	1.0	[m]
Wave peak period (T_p)	3.0 - 13.0	[s]
Headings	180 135 90	[deg]

Table 10-7: Environmental condition for 5000 [mt] lift operations

Mass moment of inertia

The mass moment of inertia is calculated for each concept design and per load case. It includes the mass moment of inertia of the crane, deckload, water ballast, consumables and vessel structures. This is calculated per member as follows:

- <u>Mission equipment</u>
  Inertia of individual components is known from the technical specifications [ref. 9] and [ref. 10]. Their contribution with respect to total vessel is calculated.
- <u>Deckload, water ballast and consumables</u>
  The weight and centre of gravity of each weight component is known by the vessel models in the Software Delftship. This enables to calculate the contribution of each weight item on the total mass moment of inertia.
- <u>Vessel structures</u>
  The weight and location of each lightweight component is known by the lightship weight calculations, except for the structural steel weight (hull plating and scantling). To calculate the mass moment of inertia of the structural steel is difficult, as no scantling arrangement is made nor calculated. However, the software AQWA can calculate the mass moment of inertia of the modelled body. The mass moment of inertia of the AQWA model with no loading (LSW draft) is considered representative for the structural contribution.

It is assumed that the crane load does not contribute to the mass moment of inertia of the vessel. Because the crane load is assumed to be behave like a pendulum i.e. the vessel moves around the unconstraint crane load.

AQWA models

The concept designs are modelled on full scale in AQWA-LINE according to the particulars given in Table 8-16. The linearized roll damping is calculated for each model per direction, loading condition and environmental condition. The used input data for the diffraction radiation software is given in table 10-8.

Load case	Concept	GM [m]	Ixx [kg·m²]	Iyy [kg·m²]	Izz [kg·m²]	Kxx [m]
Lift 5000 [mt], 34[m]	U-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
	V-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Lift 2000 [mt], 34[m]	U-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
	V-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Lift 500 [mt], 34[m]	U-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
	V-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Pipe lay (load case 7)	U-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
	V-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Pipe lay (load case 8)	U-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
	V-shape	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Table 10-8: Used input data for the calculations in AQWA-LINE

10.3.2 Calculation of irregular significant responses

Using the calculated RAO's by AQWA-LINE the most probable maximum irregular response can be derived for a particular wave spectrum. The used calculation method is described below [ref. 19].

The JONSWAP spectrum is defined by:

$$S_\tau(\omega) = \frac{320 H_{1/3}^2}{T_p^4} \cdot \omega^{-5} \cdot exp\left[\frac{-1950}{T_p^4} \cdot \omega^{-4}\right] \cdot \gamma^A \qquad \{10\text{-}2\}$$

$$T_p = T_z(0.327 e^{(-0.315\gamma)} + 1.17) \qquad \{10\text{-}3\}$$

Where for a Pierson-Moskowitz spectrum:

$$\gamma = \quad 1.0$$

$$A = exp\left(-\left(\frac{\frac{\omega}{\omega_p} - 1}{\sigma\sqrt{2}}\right)^2\right) \qquad \{10\text{-}4\}$$

$$\omega_p = \frac{2\pi}{T_p} \qquad \{10\text{-}5\}$$

With:

$$\sigma = \begin{cases} 0.07 & \omega < \omega_p \\ 0.09 & \omega > \omega_p \end{cases}$$

The response spectrum for a response is given by:

$$S_z(\omega) = \left|\frac{z_a}{l^2}(\omega)\right|^2 \cdot S_t(\omega) \qquad \{10\text{-}6\}$$

The significant response amplitude is:

$$A_{1/3} = 2 \cdot \sqrt{\int_0^\infty S_z(\omega) d\omega} \qquad \{10\text{-}7\}$$

The significant response single amplitude is calculated as:

$$A_{1/3} = 2 \cdot \sqrt{\sum \left| \frac{z_a}{l^2}(\omega) \right|^2 \cdot S_l(\omega) \cdot \Delta\omega} \qquad \{10\text{-}8\}$$

The maximum response single amplitude for a three hour period is:

$$A_{max} = f \cdot \bar{z}_{a1/3} = \frac{1}{2} \cdot \sqrt{2ln\left(\frac{3 \cdot 3600}{T_z}\right)} \cdot A_{1/3} \qquad \{10\text{-}9\}$$

10.3.3 Limiting operational conditions and criteria

The maximum allowable motion response during pipe lay and lift operations imposed by the mission equipment (motion criteria) are given in Table 10-9. By applying these criteria, the maximum environmental conditions at which the vessels are still able to operate, can be determined.

Load case	Heave [m]	Roll [deg]	Pitch [deg]
Heavy lifting (5000 mt)	[redacted]	[redacted]	[redacted]
Intermediate lifting (2000 mt)	[redacted]	[redacted]	[redacted]
Light lifting (500 mt)	[redacted]	[redacted]	[redacted]
Pipe lay (2500 mt pipe storage)	[redacted]	[redacted]	[redacted]
Pipe lay (10,000 mt pipe storage)	[redacted]	[redacted]	[redacted]

Table 10-9: Motion criteria during operations

The heave limitation is determined by the maximum effective stroke of the active heave compensator. The heave motion criteria are therefore applied to the allowable vertical motions in the crane tip. The maximum vertical motion depends on the amount falls on the block and depends therefore on the crane load. For example, if for a heavy lift operation 4 falls are used this will reduce the maximum stroke of the heavy compensator from 8 meter, to 2 meter. The roll and pitch limits are imposed by the dynamic part of the off- and sidelead capacity of the crane, see section 8.2.1.Additional criteria per operational type are given below:

- Lift operations
  The maxim allowed vertical acceleration of crane tip: 0.1 gravity
- Pipe lay operations
  The stinger is connected to the vessel transom by a bearing and is held in place by two tensioned tethers fixed to the stinger support frame, see figure 10-7 and 10-8. Due to this configuration the vertical velocity of the stinger is restricted, to prevent uplifting of the stinger. This effect needs to be avoided, as it can cause damage to the pipeline and S-lay system. In case the vertical velocity is high, the stinger drag will reduce the tension in the tethers such that they will be slack. If this occurs a very high impulse load is present when the tension is resorted. To avoid this, the stinger

on the Seven Borealis has a maximum allowable vertical velocity during operation, as follows:

3.0 meter per second

Hence, same criteria are also applied for the two considered design concept in this research.

Figure 10-7: Stinger of Seven Borealis connect to the vessel transom

Figure 10-8: Side view on connected stinger of Seven Borealis

10.4 Vessels motion response in irregular waves

This section provides the most probable maximum response (MPM) during 5000 metric tons overside lifting operations. This analysis determines if the concept designs fulfil the operational requirements. Furthermore, the motion behaviour during pipe lay operations are analysed considered.

10.4.1 Motion response during 5000 metric tons lift operation

Figure 10-9 presents the MPM and vertical crane tip acceleration during 5000 metric tons lifting operation for the environmental condition given in table 10-7. Figure 10-9 presents only the wave heading with is featured by the most governing vessel response. The operating range of the wave peak periods, as defined in operational requirements of chapter 3, is presented as shaded area in the graphs.

Based on Figure 10-9 it is concluded that both concept designs do comply with the applied operational requirements for headings of 90.0 degree. For the other considered headings it is concluded that both concept designs do not comply with the applied operational requirements. As the heave motion of the crane tip exceeds the criteria, for waves with a peak period longer as 12.0 seconds.

Although waves with a significant wave height of 1.0 meter with a wave peak periods beyond the 12.0 seconds rarely occurs at considered operation areas (see section 3.6), according to global wave statistics [ref. 20]. Furthermore, in general the most favourable vessel heading can be selected, such that lift operations can be performed at the design wave spectrum. Therefore it is assumed that not fulfilling the operational requirements has a limited consequence on aimed operational performance.

In order to fulfil the operational requirements, the vessels should be modified or another active heave compensator can be installed with larger capability to compensate vertical crane tip motions. This modification is not conducted in this research.

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Figure 10-9: Above the most probable maximum (MPM) response during 5000 metric tons lift for a HS: 1.0 meter is presented.

10.4.2 Motion response during pipe lay operations

No minimum environmental conditions for pipe lay operations are specified in the operational requirements, as described in section 3.7. Using the motion criteria of Table 10.9 and the RAO's obtained by AWQA-LINE the maximum environmental condition at which the vessel can to perform pipe lay operations is determined. The vessel response during pipe lay operations is evaluated per sea-state, with a step of 1.0 meter per wave heading. The environmental condition at which each concept design still can perform pipe lay operations is given in table 10-10. The presented environmental is based on the load case which is feature by the most unfavourable motion behaviour, load case 7.

Item	V-shape	U-shape	Unit
Significant wave height ($H_{1/3}$)	0.0 - 2.0	0.0 - 2.0	[m]
Wave peak period (T_p)	3.0- 17.0	3.0 – 12.0	[s]
Headings	All	All	[deg]
Spectrum type	Pierson-Moskowitz		

Table 10-10: Environmental condition for pipe lay operations

Note that the each concept design can perform pipe lay operations with larger wave height, but only for small wave peak period range. The workability analysis of section 10.5, considers each wave height the concept design are capable to perform pipe lay operations within the range of motion criteria.

10.4.3 Motion behaviour during transit operations

The motion behaviour during transit operation (load case 10) are considered. In order to determine if a V-shape hull has a positive effect on roll behaviour of multi-purpose construction vessels during transit operations. The roll RAO of the U-and V-shape is illustrated in Figure 10-10. The MPM roll response for transit operation with a Pierson-Moskowitz wave spectrum with $H_{1/3}$ = 1.0 meter is included in the Figures 10-11 and 10-12. Furthermore, the Figures 10-13 and 10-14 present the most probable maximum acceleration in y-direction at the navigation bridge.

Confidentail

Figure 10:10: Roll RAO for transit operations

The analysis of this section clearly shows that the V-shape concept design has significantly improved roll motions compared to the U-shape concept design. For the V-shape concept the roll response and the acceleration in y-direction is smaller enhancing the crew comfort during transit operations.

Confidentail	Confidentail
Figure 10-11: MPM roll response during transit, heading 90°	Figure 10-12: MPM roll response during transit, heading 135°

Confidentail	Confidentail
Figure 10-13: MPM acceleration in y-direction at navigation deck	Figure 10-14: MPM acceleration in y-direction at navigation deck

10.5 Workability for lift and pipe lay operations

The maximum vessel responses and the operational restrictions are used to calculate the workability of each design concept. Three lift and two pipe lay operations, presented in section 10.3.1, are considered. The workability analysis is performed for the areas: Gulf of Mexico, east coast of Brazil and West coast of Africa, as these areas are set in the operational requirements of section 3.6. Global wave statistics [ref. 20] are used to obtain the workability. The scatter diagrams of the nautical areas, presented in table 10-11, are applies as input.

Area	Nautical area
West coast of Africa	68
East coast of Brazil	74
Gulf of Mexico	32

Table 10-11: Considered operational areas

The vessel response is calculated for all encountered wave periods and significant wave heights present in considered areas, for three wave headings, i.e. 90, 135 and 180 degrees.

The calculated vessel response per operation type is given below:

- Lift operations
  The most probable maximum response for heave, roll, pitch and the vertical acceleration of crane tip is calculated.
- Pipe lay operations
  The most probable maximum response for heave, roll, pitch and the vertical velocity of the stinger is calculated.

10.5.1 Workability for considered lift operations

Table 10-12 presents the workability per wave heading operating at the west coast of Africa for all considered over side lift cases for both concept designs.

Load case	Heading [deg]	Workability [%] U-shape	Workability [%] V-shape	Deviation [%]
Lifting (5000mt)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
Lifting (2000mt)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
Lifting (500 mt)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]

Table 10-12: Workability for a considered lifting cases for the west coast of Africa

Table 10-13 presents the workability per wave heading operating at the east coast of Brazil for all considered over side lift cases for both concept designs.

Load case	Heading [deg]	Workability [%] U-shape	Workability [%] V-shape	Deviation [%]
Lifting (5000mt)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
Lifting, (2000mt)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
Lifting (500 mt)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]

Table 10-13: Workability for a considered lifting cases for the east coast of Brazil

Table 10-14 presents the workability per wave heading operating at the Gulf of Mexico for all considered over side lift cases for both concept designs.

Load case	Heading [deg]	Workability [%] U-shape	Workability [%] V-shape	Deviation [%]
Lifting (5000mt)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
Lifting (2000mt)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
Lifting (500 mt)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]

Table 10-14: Workability for a considered lifting cases for the Gulf of Mexico

10.5.2 Workability for considered pipe lay operations

Table 10-15 presents the workability per wave heading operating at the west coast of Africa for the two considered pipe lay cases for both concept designs.

Load case	Heading [deg]	Workability [%] U-shape	Workability [%] V-shape	Deviation [%]
Pipe lay (100% pipe storage)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
	All	[redacted]	[redacted]	[redacted]
Pipe lay (25% pipe storage)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
	All	[redacted]	[redacted]	[redacted]

Table 10-15: Workability for a considered lifting cases for the west coast of Africa

Table 10-16 presents the workability per wave heading operating at the east coast of Africa for the two considered pipe lay cases for both concept designs.

Load case	Heading [deg]	Workability [%] U-shape	Workability [%] V-shape	Deviation [%]
Pipe lay (100% pipe storage)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
	All	[redacted]	[redacted]	[redacted]
Pipe lay (25% pipe storage)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
	All	[redacted]	[redacted]	[redacted]

Table 10-16: Workability for a considered pipe lay operations for the east coast of Brazil

Table 10-17 presents the workability per wave heading operating at the Gulf of Mexico for the two considered pipe lay cases for both concept designs.

Load case	Heading [deg]	Workability [%] U-shape	Workability [%] V-shape	Deviation [%]
Pipe lay (100% pipe storage)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
	All	[redacted]	[redacted]	[redacted]
Pipe lay (25% pipe storage)	90.0	[redacted]	[redacted]	[redacted]
	135.0	[redacted]	[redacted]	[redacted]
	180.0	[redacted]	[redacted]	[redacted]
	All	[redacted]	[redacted]	[redacted]

Table 10-17: Workability for a considered pipe lay operations for the Gulf of Mexico

During pipe lay operations the vessel heading is fixed, the vessels encounters wave from all directions. In this research it is therefore assumed that the vessel encounters each analysed wave heading even often in its life time. Based on this assumption the average workability for all considered pipe lay operations calculated and presented in table 10-18.

Load case	Area	Workability [%]		Deviation [%]
		U-shape	V-shape	
Pipe lay (100% pipe storage)	West coast of Africa	[redacted]	[redacted]	[redacted]
	East coast of Brazil	[redacted]	[redacted]	[redacted]
	Gulf of Mexico	[redacted]	[redacted]	[redacted]
Pipe lay (25% pipe storage)	West coast of Africa	[redacted]	[redacted]	[redacted]
	East coast of Brazil	[redacted]	[redacted]	[redacted]
	Gulf of Mexico	[redacted]	[redacted]	[redacted]

Table 10-18: Average workability for a considered pipe lay operations and areas

10.7 Conclusion on motion behaviour

By analysing all the results in this chapter; certain points can be noted and concluded:

- Linear RAO's can be used to calculated the motion behaviour of the V-shape
  According to the validation of linear RAO's, obtained by AQWA-LINE, against RAO's based on measured data coming from model tests.
- Ikeda method is applicable for both concept designs
  The analytic RAO's including roll damping according to the Ikeda prediction method shows sufficient similarity with RAO's determined by model tests.
- Lifting 5000 metric tons overside the operation
  Both design concepts do not comply with the operational requirements for the 5000 metric tons over side lift operation. The maximum allowable vertical displacement of the crane tip is exceed for wave heading of 135.0 and 180 degree for wave peak periods beyond 12.0 seconds. While it was required that this lift operation needs to performed in a sea state up to 1.0 meter significant wave height and a peak period up to 13.0 seconds. Modification on both concept designs is therefore required, as described in section 10.4.

[redacted]

[redacted]

The DP performance capability of the two concept designs are compared in this chapter. As it was noted in phase I; it is likely that the current forces acting on the V-shape concept design diviates from the U-shape concept design. By means of comparing the environmental forces exerting on the vessel during DP operations; the differences in DP performance capability is estimated. Both current, wind as well wave forces are considered in this chapter. All environmental forces are assumed to be collinear. Note that no detailed DP performance analysis is performed in this research.

11.1 Current forces

The current forces can be calculated using current coefficients; in general these are measured in model tests. With these coefficients the full scale forces can be calculated by equations 11-1 to 11-3 [ref. 8].

$$F_{x\ current} = \frac{1}{2}\rho U^2 A_f C_x \tag{11-1}$$

$$F_{y\ current} = \frac{1}{2}\rho U^2 A_s C_y \tag{11-2}$$

$$M_{z\ current} = \frac{1}{2}\rho U^2 A_s L_{oa} C_N \tag{11-3}$$

Where:

$F_{x\ current}$	:	Current force in x-direction	[N]
$F_{y\ current}$	:	Current force in y-direction	[N]
$M_{z\ current}$	:	Current moment about z-axis	[Nm]
ρ	:	Density	[kg/m^3]
U	:	Mean current speed	[m/s]
A_f	:	Projected frontal area of vessel below waterline	[m^2]
A_s	:	Projected side area of vessel below waterline	[m^2]
L_{oa}	:	Length overall	[m]
C_x	:	Current coefficient in x-direction	[-]
C_y	:	Current coefficient in y-direction	[-]
C_N	:	Current coefficient about the z-axis	[-]

The current coefficients of the Huisman crane vessel [ref. 14] and of a U-shape reference vessel [ref. 8] are determined from data coming from model tests. These coefficients are presented in Figure 11-1. Based on Figure 11-1 it is concluded that the current coefficients of the U- and V-shape are almost identical. Hence, significant deviation in current forces can only result from the difference in projected side area. Table 11-1 presents the projected side and frontal area below the waterline for both concepts, for different modes of operations.

Confidentail	Confidentail
Confidentail	Confidentail

Figure 11-1: The current coefficient of reference U-shape versus V-shape

Load case	Projected side area		Projected frontal area		Unit
	U-shape	V-shape	U-shape	V-shape	
Lifting (5000mt)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]
Lifting (2000mt)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]
Lifting (500mt)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]
Pipe lay (100% pipe storage)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]
Pipe lay (25% pipe storage)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]

Table 11-1: The projected side- and frontal- areas of the U- and V-shape for current loading

Based on the Figure 11-1 and Tables 11-1, the following can be concluded:

- Current forces in x-direction
  The projected front areas differ significantly between the U- and V-shape concepts and therefore the current forces.
- Current forces in y-direction
  The projected side areas are comparable and therefore also the current forces, except for the 5000 metric tons lift operations. During this operation the current force in y-direction of the V-shape vessel is higher, caused by the larger draft.

Note that the current forces on the stinger, during pipe lay operations, are not considered. As both concept designs are accommodated with the same stinger i.e. no difference in current loads.

11.2 Wind forces

The wind loads can be calculated using wind coefficients; in general these are measured in model tests. With these coefficients the full scale forces can be calculated by equations to the equations 11-4 to 11-6 [ref. 8].

$$F_{x\,wind} = \frac{1}{2}\rho U^2 A_f C_x \tag{11-4}$$

$$F_{y\,wind} = \frac{1}{2}\rho U^2 A_s C_y \tag{11-5}$$

$$M_{z\,wind} = \frac{1}{2}\rho U^2 A_s L_{oa} C_N \tag{11-6}$$

Where:

$F_{x\,wind}$	:	Wind force in x-direction	[N]
$F_{y\,wind}$	:	Wind force in y-direction	[N]
$M_{z\,wind}$	:	Wind moment about z-axis	[Nm]
ρ	:	Density	[kg/m^3]
U	:	Mean wind speed	[m/s]
A_f	:	Projected frontal area of vessel above waterline	[m^2]
A_s	:	Projected side area of vessel above waterline	[m^2]
L_{oa}	:	Length overall	[m]
C_x	:	Wind coefficient in x-direction	[-]
C_y	:	Wind coefficient in y-direction	[-]
C_N	:	Wind coefficient about the z-axis	[-]

Table 11-2 presents the projected side and frontal area of the hull above the waterline for both concepts for different operational cases.

Load case	Projected side area		Projected frontal area		Unit
	U-shape	V-shape	U-shape	V-shape	
Lifting (5000mt)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]
Lifting (2000mt)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]
Lifting (500mt)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]
Pipe lay (100% pipe storage)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]
Pipe lay (25% pipe storage)	[redacted]	[redacted]	[redacted]	[redacted]	[m^2]

Table 11-2: The projected side- and frontal- areas of the U- and V-shape for wind loading

Note that Table 11-2 only includes the areas of the hull, no superstructure equipment, etc. It is assumed that the wind force on the exposed areas of the V-shape concept design has the same order of magnitude for the U-shape concept design. Because, the same equipment is installed and the superstructure is almost identical. By analysing the presented data in table 11-2, the following can be noted:

- The projected side area deviates between the U- and V-shape concepts, due to the difference in draft,
- The projected front area of the V-shape shows some minor differences compared to the U-shape concept design,
- Wind forces on superstructure and equipment is assumed to be equal

11.3 Wave forces

In order to keep position, the mean wave drift forces needs to be compensated by the DP system. The mean wave drift forces are calculated using the software AWQA-LINE. In the Figures 11-2 to 11-5 the wave drift forces of both concept designs during 5000 metric tons lift operation as presented. This condition is compared towards each other as it represents the conditions which has the largest difference in mean drift forces.

Confidentail	Confidentail
Figure 11-2: Mean drift force in x-direction, wave heading: 180 degree	Figure 11-3: Mean drift force in x-direction, wave heading 90 degree
Confidentail	Confidentail
Figure 11-4: Mean drift force in y-direction, wave heading: 180 degree	Figure 11-5: Mean drift force in y-direction, wave heading 90 degree

Based on this and the Figures 11-2 – 11-5 it is concluded that the wave force on both concept designs is comparable for all considered load cases.

11.4 Difference in DP performance

The current and wind forces on the V-shape concept differ significantly with respect to the U-shape concept, as earlier observed in this chapter. The consequence of this on the overall DP performance is estimated in this section. The DP performance analysis of a reference multi-purpose construction vessel (with a V-shape hull) is made available by Huisman equipment. Using this data the total required (reference) forces to keep the concept designs at position is estimated. With this the influence of difference in current and wind forces on total required thrust power can be determined, hence on the DP performance.

The DP feasibility plots of the reference vessel present the required engine power to keep position during a particular operation and environmental condition. By means of using feasibility plots the required (average) force to keep position during a particular operation and environmental condition is calculated. A thruster-power ratio of [redacted] kilo Newton per kiloWatt for the tunnel thrusters was applied. Note that only the load cases with the largest deviation in projected side and front area are considered.

Lift operations

The required force to keep the vessel on position during lift operations is obtained from the feasibility plots, see Figure 11-5 and 11-6. Using the Tables 11-1 1 - 11.2 and the current and wind coefficients, the difference between the environmental forces (Δ) of the V-shape with respect to the U-shape is calculated.

In order to calculate the wind force, wind coefficients according to the class society ABS [ref. 21] is used, see Figure 11-8. The wind and current velocity, presented in Figure 11-6 and 11-7, are used to calculate the forces. Table 11-3 presents the deviation of the environmental forces with respect to the total estimated force.

Load case	Heading [deg]	Total force [kN]	Δ environmental load [kN]	Deviation [%]
Lifting (5000mt)	90	[redacted]	[redacted]	[redacted]
	180	[redacted]	[redacted]	[redacted]
Lifting (500mt)	90	[redacted]	[redacted]	[redacted]
	180	[redacted]	[redacted]	[redacted]

Table 11-3: The estimated difference in required force, between the U-and V-shape

Confidentail

Figure 11-6: Feasibility plot heavy lifting, intact

Confidentail

Figure 11-7: Feasibility plot light lifting, intact

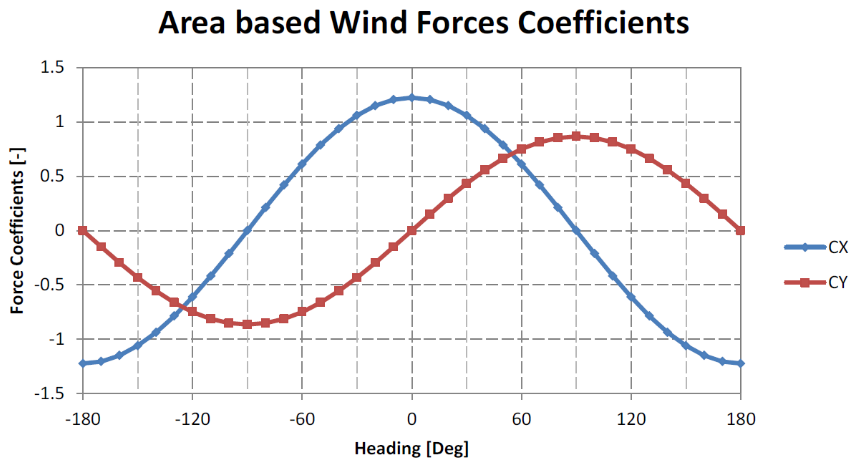

Figure 11-8: Wind coefficients according to ABS [ref.21]

Pipe lay operations

During pipe lay operations the DP system must provide thrust to keep the pipeline (pipe span) in a S-configuration, additional to the environmental forces exerting on the vessel. Using the Tables 11-1 1 and 11.2 and the current and wind coefficients, the difference between current and wind forces (Δ) of the V-shape with respect to the U-shape is calculated. The wind and current velocity, presented in Figure 11-9 ,is used to calculate the forces. Table 11-4 presents the deviation of the environmental force with respect to total estimated force.

Load case	Heading [deg]	Total force [kN]	Δ environmental load [kN]	Deviation [%]
Pipe lay (100%)	90	[redacted]	[redacted]	[redacted]
	0	[redacted]	[redacted]	[redacted]
Pipe lay (25%)	90	[redacted]	[redacted]	[redacted]
	0	[redacted]	[redacted]	[redacted]

Table 11-4: The estimated difference in required force, between the U-and V-shape

Confidentail

Figure 11-9 Feasibility plot deep water S-lay, of reference multi-purpose construction vessel

11.5 Conclusions on DP capability

In this chapter, the environmental forces exerting on the concept designs during lift and pipe lay operations are analysed. The difference in environmental load between the two concept designs is determined and evaluated with respect to total force exerting on the vessel during DP operations. By interpreting the results, the following points can be noted:

[redacted] The environmental forces, exerting on the V-shape concept with a heading of 0 degrees is smaller for all considered lifting operations. [redacted]

[redacted] The environmental forces exerting on the V-shape concept with a heading of 90 degree is larger for all considered lifting operations. It [redacted]

- No significant difference between the environmental forces exerting on the concept designs [redacted].

According to the above, the following is concluded:

- The DP system of the V-shape requires [redacted]

[redacted] The V-shape requires [redacted]

[redacted] During pipe lay operations the DP performance [redacted]

*In the presented analysis, the favourable vessel heading for DP operations is equal to the most favourable heading from the motion behaviour point of view.

PHASE IV

CONCLUSIONS AND RECOMMENDATIONS

The research is concluded by answering the research question.

'Does the operational performance of a multi-purpose construction vessel improve with a design based on the V-shape concept, compared to a conventional design?'

[redacted]

This follows from the still water resistance, DP capability, motion behaviour and the workability analysis on both created concept designs. The newly designed V-shape hull geometry results in:

[redacted] The workability of the V-shape concept [redacted]

Load case	Area	Improved workability* [%]
Lifting (5000mt)	West coast of Africa	[redacted]
	East coast of Brazil	[redacted]
	Gulf of Mexico	[redacted]
Lifting (2000mt)	West coast of Africa	[redacted]
	East coast of Brazil	[redacted]
	Gulf of Mexico	[redacted]
Lifting (500mt)	West coast of Africa	[redacted]
	East coast of Brazil	[redacted]
	Gulf of Mexico	[redacted]
Pipe lay (100% pipe storage)	West coast of Africa	[redacted]
	East coast of Brazil	[redacted]
	Gulf of Mexico	[redacted]
Pipe lay (25% pipe storage)	West coast of Africa	[redacted]
	East coast of Brazil	[redacted]
	Gulf of Mexico	[redacted]

Table 12-1: improved workability of the V-shape concept design with respect to the U-shape concept design

*Note that Table 12-1 includes the average gained workability of the V-shape design concept with respect to the U-shape concept design. It is assumed that the vessel encounters each analyzed wave heading even often in its life time, see section 10.5.2 for more details.

- Based on the present analysis, the still water resistance (for design speed 15 knots) is [redacted]
- The DP capability [redacted].

[redacted] Based on the present analysis, it is expected that financial performance of a multi-purpose construction [redacted] a V-shape hull. [redacted]

[redacted]

- [redacted].

Key feedings on the particulars of multi-purpose construction vessel with a V-shape compared to a U-shape is described in this chapter. Furthermore, the possible applicability of the V-shape concept on other vessel types is discussed.

13.1 Key findings

Based on the two developed concept design, certain findings are observed on the particulars of multi-purpose construction vessel with a V-shape hull compared to a U-shape:

- V-shape vessels are typically longer and have a larger draft than U-shape vessels, to gain sufficient buoyancy. The larger draft leads to a larger depth in order to comply with freeboard criteria's.
- The draft of V-shape vessels is typically larger during lifting operations than for a U-shape vessel. This result in a smaller versatility for V-shape, as lifting operations in shallow water areas is impossible. However, in reality lifting operations are mainly offshore performed i.e. deep water areas.

[redacted]

- [redacted] Note that it is assumed that the heavy lift operations are a small part of the operational profile.

[redacted] The roll damping of a V-shape is in general smaller, [redacted]

- [redacted].

[redacted] Due to the large deadweight during pipe lay operations (stinger & pipe joints) the draft increases rapidly resulting in large breadth on the waterline. [redacted]

13.2 V-shape applicability

This section discuss the possibilities to implementation the V-shape concept on other vessel types than multi-purpose construction vessels, using the knowledge of the presented research.

13.2.1 Characteristics which potentially benefit by a V-shape hull

The operational and design aspects of vessels which can potentially benefit by a V-shape hull, as observed in this research, is listed below:

- Larger range in required amount of stability,

- Vessels which needs to have a high stability to support certain operations in combination with a high transit speed and a considerable part of operation profile exist of transit operations,
- The operational profile contains of a wide range of different operational condition. And for each operation good motion characteristics are required.

13.2.2 Applicability of the V-shape on other vessel types

Based on the previous section it is likely to be that the following vessel types will benefit from a design based on the V-shape concept:

- Heavy cargo vessels
  [redacted] vessel is deballasted such that it sails with a reduced waterline breadth, enhancing the hull resistance and roll motion behaviour.

  In case heavy cargo vessels with a V-shape sail at its maximum allowable draft, the waterline breadth will be almost equal to the overall breadth. In the condition the possible advantage of a V-shape hull is reduced. [redacted]
- Reel lay vessels
  Reel lay vessels are designed to installed reeled pipes, see section 1.2.2 for more details. In general this type [redacted] high centre of gravity of the pipes. Due to the high deadweight the draft and the breadth on waterline increases, resulting in a high stability. If the vessel sails without pipe loading, for example toward the pipe spool base, the displacement is significantly reduced. [redacted]
- Multi-purpose construction vessel combining lifting and pipe lay (J-lay method) capability
  In general this type of vessels are characteristic by the similar contradicting design requirements as the multi-purpose construction vessel considered in this research. See section 1.2.3 for more information. The workability for both intermediate and light weight lifting operations will probably improve by a V-shape hull compared to U-shape. [redacted] operational improvement is expected for this kind of vessels as for the concept studied in this research. Because the difference between the displacement in light and heavy lifting operations are smaller, with respect to the concept designs developed in this research. As the J-lay lay system is permanent accommodate on the vessel, while for the stinger is only connected during pipe lay operations.

- <u>Crane vessels</u>

  Crane vessels with the capability to perform long transits with a relative high speeds. The design and operational requirements of this type of crane vessels is comparable to the multi-purpose construction vessel considered in this research. Hence, the V-shape is featured by a small waterline breadth for transit and light lifting operations and a high stability if the draft is increased.

Based on the brief analysis of above it can stated; the V-[redacted] Note that this chapter only includes a first estimated. Further research is recommend to investigated if a V-shape hull improves the operational performances of the above mentioned vessel types.

14 Recommendations

It is recommended that future research is focused on the following topics:

- A detailed comparison study on hull resistance between both concept designs,
- An extensive parametric study on the V-shape hull, which considers more loading conditions and investigates the relation between the displacement and motion behaviour more in detail. In order to improve the effect of the V-shape (a reduced waterline breadth) during pipe lay operations,
- A detailed comparison study on the economic performance of both concept designs. To determine the consequences of a V-shape hull on the CAPEX and OPEX,
- A detailed DP performance analyse of a vessel with a V-shape hull, to determine more in detail the difference in DP performance,
- Calculated the total electrical load balance of the vessel in order to determine the required engine power,
- Investigate the, practical, vessel handling of the V-shape concept by interviewing marine contractors and crew.

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APPENDICES

A	Reference vessels
B	General arrangement OMC
C	General arrangement of U-and V-shape concept design
D	Parametric study V-shape hull
E	Intact stability criteria
F	Definition of weights
G	Short summary of the Ikeda Method

A. Reference vessels

The particulars of the used reference vessels in this research are in this appendix. Their missions and installed mission equipment is presented below:

Vessels	Mission(s)
Seven Borealis	Offshore installation by means of 5000 [mt] crane and installation of pipeline with J-lay of S-lay method.
Oleg Strashnov	Offshore installation by means of 5000 [mt] crane. A firing line duct underneath main deck is presented.
Aegir	Offshore installation by means of 4000 [mt] crane and installation of pipeline with J-lay of R-lay method.
Sapura 3000	Offshore installation by means of 3000 [mt] crane and installation of pipeline with S-lay and J-lay method
Yantai salvage (concept)	Offshore installation by means of 5000 [mt] crane and S-lay ready

The vessels have the following main particulars:

Particulars	Seven Borealis	Oleg Strashnov	Aegir	Sapura 3000	Yantai salvage (concept)	Units
LOA	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[m]
Lpp	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[m]
Breadth	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[m]
Depth	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[m]
Draft	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[m]
Deadweight		[redacted]	[redacted]	[redacted]	[redacted]	[mt]
Lightweight	[redacted]		[redacted]	[redacted]	[redacted]	[mt]
Displacement	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	
Fuel oil	[redacted]	[redacted]		[redacted]	[redacted]	[m³]
Ballastwater	[redacted]	[redacted]		[redacted]	[redacted]	[m³]
Fresh water	[redacted]	[redacted]		[redacted]		[m³]
HP total	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[kW]
Speed	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[kn]
Max. crew	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	POB

B. General arrangement OMC

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C. General arrangement of both concept designs

Confidentail

D. Parametric study V-shape hull

The node position and side shell slope are systematic varied in order to investigate their influence on hydro – static and dynamics characteristics. The goal is to investigate which V-shape geometric as the large as possible capability to adapted the initial stability and motion characteristics to different operational conditions. This appendix describes the performed parameter study, and unfolds as follows: the approach, input and assumptions are set and next the results are presented and discussed.

Approach

The two geometry parameters, node position and vessels side shell slope, are varied one at the time. So the influence of each parameter on the hydrostatic and hydrodynamic properties can solely be analysed. Each [redacted]

[redacted]

- [redacted] [°]

In order to investigate the influences of these parameters in the most valuable as possible manner, the weight distribution and load cases are used as defined in section 6.4 Table 6.4. The two following loading conditions are considered:

- Maximum lift capacity over side [case 1]
- Transit (50% consumables)

These two conditions are considered, as the difference in GM height between the two conditions are maximum. The V-shape geometry with is featured in both condition with a relative small GM height is the most optimal V-shape. As this geometry has the largest capability to adapted its initial stability and motion behaviour.

This study includes only the parallel mid-ship section. This allows to determine pure the influence of the parameters on the midship. In reality, the variation of the hull parameters has consequences on the bow and aft geometry, hence the total vessel properties. For example a node height of 12.5 [m] will result in a more slender bow section as a geometry with a node height of 9.5 [m]. However, it is assumed that this influence is minor on the stability properties. The applied approach is simple however it is considered accurately enough for this design phase. The four-time variation of each parameter results in total 16 different geometries. Each of the geometry was modelled in the software Delftship, which can calculate the stability properties. The results of the stability calculations are not evaluated with criteria related to stability.

- The following aspects are monitored and evaluated in this study:
    - GM
    - Range of stability
    - Delta GM between the two different load cases

Input

The used input data of this systemic variation study is as follow:

- <u>Used input data</u>
  The used model is based on the minimum vessel dimensions to accommodate the mission equipment, as presented Table 4-2. The main particulars are presented in the Table A-1.

Item	V-shape	Unit
Length	[redacted]	[m]
Depth		[m]
Waterline breadth		[m]
Node height		[m]
Displacement		[mt]
Slope vessel side		[˚]

Table A-1: V-shape concept particulars (based on mission equipment)

Description of the model

The 16 different V-shape mid-ship geometries are modelled in the software Delftship, below their implementation in the software is discussed.

1. <u>Dimensions</u>
   A base case model was created with respected to table A-1. All the dimensions of the base case model are kept the same except for the length. The main section of the vessel was extended over the total vessel length. Because the models include only a parallel mid-ship the length was reduced to get the same displacement. The node height is set [redacted] and the side shell slope is [redacted] for the base model. The particulars of the base model are included in Table A-2.

Item	V-shape	Unit
Length	[redacted]	[m]
Depth		[m]
Waterline breadth		[m]
Node height		[m]
Light displacement		[mt]
Slope vessel side		[˚]

Table A-2: Base case model (used for systematic variation study)

The lightship weight and the centre of gravity of each model are separately calculated, as volume of each model is different. The LSW calculation is performed according the same method as discussed in section 6.2.

2. Arrangement
   The vessel's loading condition is adjusted, in order to tune the vessel characteristics towards a particular operation condition, using water ballast. The amount water ballast and tank arrangement has a significant influence on the stability and motion characteristic. Its arrangement is therefore kept as constant as possible between the different models and is based on the arrangement as presented in section 6.3. The water ballast and consumable tank are defined in Delftship such that they automatic will be adjusted with hull when the geometric is modified, i.e. the inner vessel space is constant but the tank volume will change. The correction, on the GM height, for free surface moment is neglected, as this correction is not a pure caused by the parameter considered parameters.

Results

The results of the stability calculations for all the 16 different geometries are presented by the below Figures.

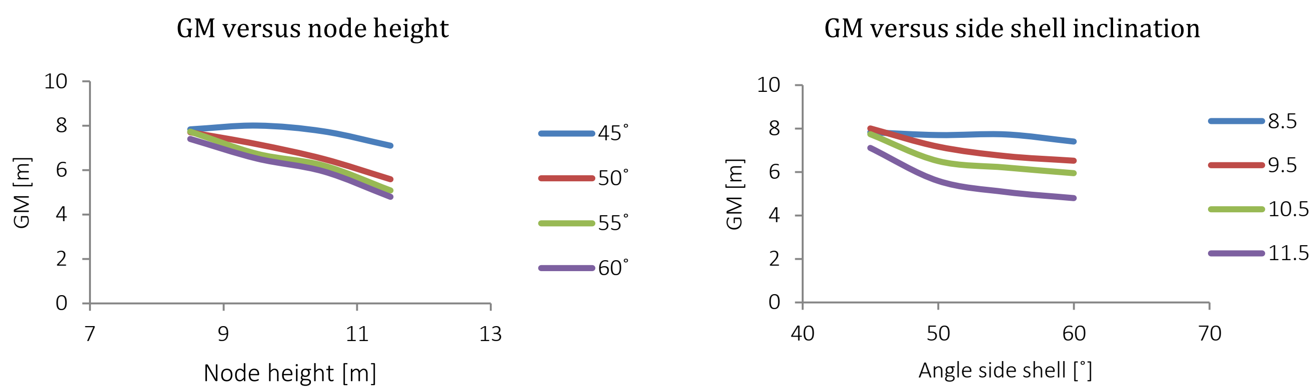

Figure A-1: GM (solid) value for maximum lift overside operation, 100% cons

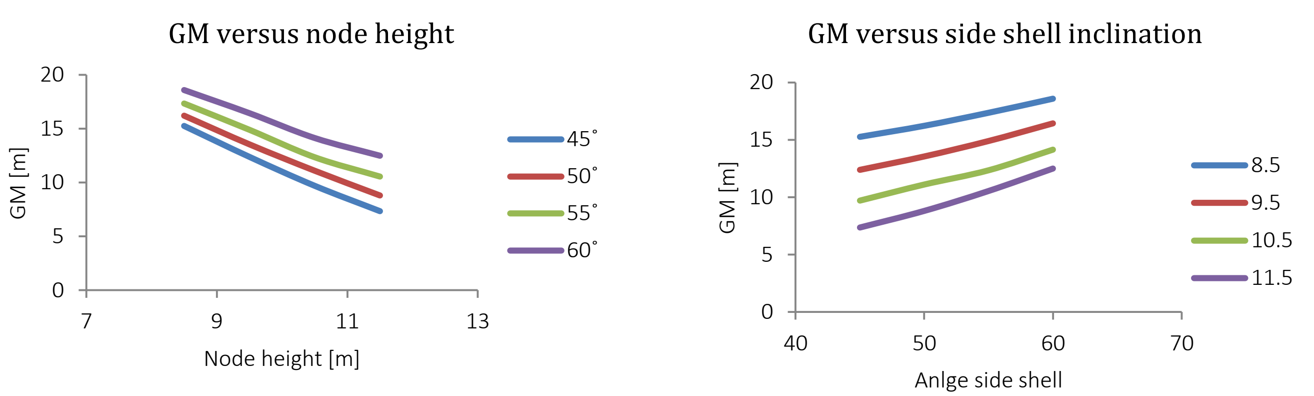

Figure A-2: GM (solid) value for transit operation, 50% cons

Figure A-3: GZ curves for 5000 [mt] lift over side of all varied geometries

Figure A-4: Difference in GM between lift and transit operation as a function of node height and side shell inclination

Conclusion

[redacted]

[redacted]nt.

E. Intact stability criteria

The intact stability criteria to be applied for "Special Purpose Ships" are as follows based on International Code in Intact Stability.

I. Criteria regarding righting lever curve properties

1. The area under the righting lever curve (GZ-value) should not be less than 0.055 metr-radians up to an angle of 30 ˚;
2. The area under the righting lever curve (GZ-value) should not be less than 0.09 metre-radians up to an angle of 40 ˚ degrees or the angle of downflooding ϕf if this angle is less than 40 ˚ degrees.
3. The area under the righting lever curve (GZ) between the angles of heel of 30˚ and 40˚ or between 30 ˚ and ϕf, shall not be less than 0.03 metre-radians;
4. The righting lever (GZ) should be at least 0.2 metre at an angle of heel equal to or greater than 30˚;
5. The maximum righting lever (GZ) shall occur at an angle of heel not less than 25˚. If this is not applicable, alternative criteria, based on a equivalent level of safety, may be applied (see 2).
6. The initial metacentric height GM_0 shall not be less than 0.15m.

II. Alternative criteria

The alternative criteria make reference to the Explanatory Notes to the International Code on Intact Stability, 2008 (MSC.1/Circ.1281). The alternative criteria have been used instead of the criterion stated in 1.5 because the vessel has a wide beam – 47.2 and 46.8 [m] - and low depth – 19.5 and 19.9 [m]- , which results in the beam-to-depth ratio (B/D) of 2.42 and 2.23.

Guidance for the application of the IS Code

Criteria regarding righting lever curve properties:

1. The maximum righting lever (GZ) should occur at an angle of heel not less than 15°
2. The area under the righting lever curve (GZ curve) should not be less than 0.07 metre-radians up to an angle of 15° when the maximum righting lever (GZ) occurs at 15° and 0.055 metre-radians up to an angle of 30° when the maximum righting lever (GZ) occurs at 30° or above. Where the maximum righting lever (GZ) occurs at angles of between 15° and 30°, the corresponding area under the righting lever curve should be:
   $0.055 + 0.001\,(30° + \phi_{max})$ metre-radians

Where ϕ_{max} is the angle of heel in degrees at which the righting lever curve reaches its maximum.

Severe wind and rolling criterion (weather criterion)

The ability of a ship to withstand the combined effects of beam wind and rolling shall be demonstrated as follows:

1. The ship is subjected to a steady wind pressure acting perpendicular to the ship's centreline which results in a steady wind heeling lever (Lw1);
2. From the resultant angle of equilibrium (ϕ0), the ship is assumed to roll owing to wave action to an angle of roll (ϕ1) to windward. The angle of heel under action of steady wind (ϕ0) should now exceed 16° or 80% of the angle of deck edge immersion, whichever is less;
3. The ship is then subjected to a gust wind pressure which results in a gust wind heeling lever (Lw2); and
4. Under these circumstances, area b shall be equal to or greater than area a, as indicated in Figure A-5 below:

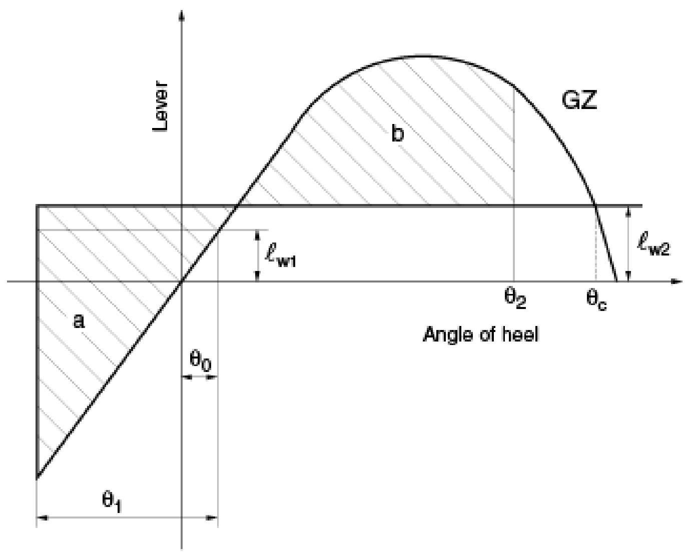

Figure A-5: Severe wind and rolling

Where the angles in Figure A-5 are defined as follows:

ϕ0	=	angle of heel under action of steady wind
ϕ1	=	angle of roll to windward due to wave action
ϕ2	=	angle of down-flooding ϕf or 50° or ϕcr, whichever is least
ϕf	=	angle of heel at which openings in the hull, superstructures or deck-houses which cannot be closed weather tight immerse. In applying this criterion, small openings through which progressing flooding cannot take place need not be considered as open.
ϕcr	=	angle of second intercept between wind heeling lever Lw2 and GZ curve.

III. Alternative criteria during heavy crane lift (DNV D200)

D 200 Accidental load drop

The effect of accidental drop of crane load shall be investigated and shall meet the following criteria:

1. The restoring energy represented by Area A2 in Figure A-6 is to be at least 40 % in excess of potential energy represented by area A1.
2. The angle of static equilibrium Θ_e after loss of crane load shall not be more than 15 degreed from the upright.

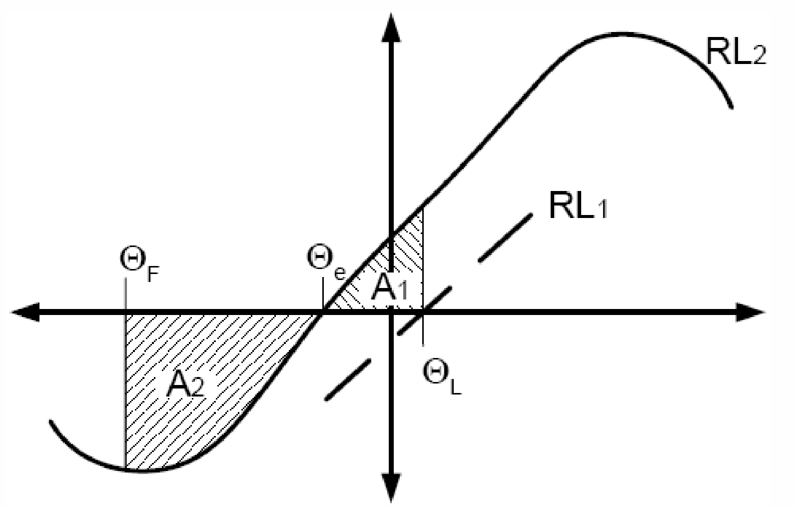

Fig. 1
Stability with loss of crane load

Figure A-6: GZ curve after crane load is loss

Where parameters in Figure A-6 are defined as follows:

RL1 = Net righting lever (GZ) curve for the condition before loss of crane load, corrected for crane heeling for crane heeling moment and for the righting moment provided by the counter ballast if applicable.
RL2 = Net righting lever (GZ) curve for the condition after loss of crane load, corrected for the transverse moment provided by the counter ballast if applicable.
ΘL = Static angle of equilibrium before loss of crane load

Θ_L = Static angle of equilibrium before loss of crane load.

ΘL may alternatively be determined by the equation

ΘL = arctan (TCG/GMt)

if this results in a small angle of heel. TCG is then to be taken as the vessel's transverse Centre of gravity before loss of crane load, and GMt is the corrected transverse metacentric height in the same condition.

Θe = Static angle of equilibrium after loss of crane load

ΘF = Angle of down flooding as defined in Pt.3 Ch.3 Sec.9.

D 300 Alternative intact stability criteria during heavy crane lift

301. The criteria given in 304 may be applied in lieu of the intact statbility criteria for the crane loading conditions when operational and environmental limitations are imposed.

302. The environmental limitation shall at least be specified as follow:
   a. Maximum wind speed (1 minute sustained at 10m above sea level)
   b. Maximum significant wave height.

303. The operational limitations shall at least be specified as follows:
   a. Maximum duration of the lift (operations reference period)
   b. Limitations in vessel speed
   c. Limitations in traffic/traffic control

304. The following criteria shall be met when the crane load is at the most unfavorable position:
   a. The deck edge shall not be submerged
   b. With the wind superimposed from the most unfavorable direction the are (A+B) ≥1.4(B+C) in accordance with Figure A-7.
   c. The area under the GZ curve measured from the equilibrium position ΘL and to the down flooding angle Θf or 20˚, whichever is less shall be at least 0.03 mrad.

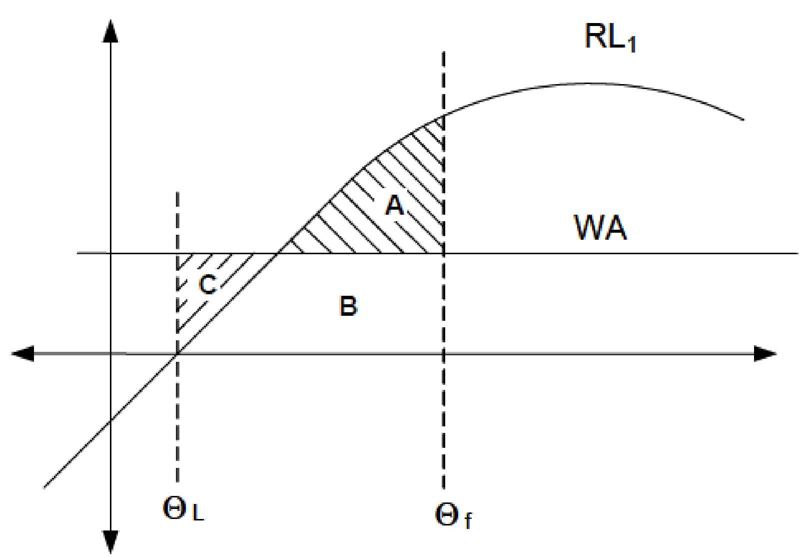

Figure A-7: Alternative intact criteria

The criteria given in 304 may be applied in lieu of the intact stability critertia for the crane laoding conditions when operational and environmental limitations are imposed.

IV. Free surface effects and liquid fillings in tanks

1. For all loading conditions, the initial metacentric height and the righting lever curve are corrected for the effects of free surfaces of liquids in tanks.
2. The free surface effects are considered whenever the filling level in a tank is less than 98% of full condition.
3. Tanks which are taken into consideration when determining the free surface correction are in one of the two categories:
   a. Tanks with fixed filling levels (i.e. liquid cargo, ballast). The free surface correction is defined for the actual filling level to be used in each tanks;
   b. Tanks with variable filling levels (i.e. consumable liquids such as fuel oil, diesel oil, fresh water, as well as liquid cargo and water ballast during transfer operations). The free surface correction is the maximum value attainable between the filling limits envisaged for each tank, consistent with any operating instructions.
4. In calculating the free surface effects in tanks containing consumable liquids, for each type of liquid at least one transverse pair or a single centreline tank has a free surface and the tank or combination of tanks taken into account should be those where the effect of free surfaces is the greatest.
5. Where water ballast tanks, including anti-rolling tanks and anti-heeling tanks, are to be filled or discharged during the course of a voyage, the free surface effects is calculated to take into account of the most onerous transitory stage relating to such operations.
   1. All Fuel tanks are filled up to a maximum of 98%

F. Definition of weights

Confidentail

G. Short summary of IKEDA method

4.2.4.2 Effect of Forward Speed, B_{44S}

$$B_{44S} = B_{44} \cdot \left\{ 0.5 \cdot \left(\begin{array}{l} A_2 + 1 + (A_2 - 1) \cdot \tanh\left[20 \cdot (\Omega - 0.3)\right] \\ + (2 \cdot A_1 - A_2 - 1) \cdot e^{-150 \cdot (\omega - 0.25)^2} \end{array} \right) - 1.0 \right\}$$

Equation 4.2–19

with:

B_{44}	potential roll damping coefficient of the ship (about G)
$\Omega = \omega \cdot V/g$	non-dimensional circular roll frequency
$\xi_D = \omega^2 \cdot D/g$	non-dimensional circular roll frequency squared
$A_1 = 1.0 + \xi_D^{-1.2} \cdot e^{-2 \cdot \xi_D}$	maximum value of B_{44} at $\omega = 0.25$
$A_2 = 0.5 + \xi_D^{-1.0} \cdot e^{-2 \cdot \xi_D}$	minimum value of B_{44} at large ω

4.2.4.3 Frictional Roll Damping, B_{44F}

Kato deduced semi-empirical formulas for the frictional roll damping from experimental results of circular cylinders, wholly immersed in the fluid. An effective Reynolds number of the roll motion was defined by:

$$Rn = \frac{0.512 \cdot \left(\frac{r_f}{\phi_a} \right)^2 \cdot \omega}{\nu}$$

Equation 4.2–20

In here, for ship forms the average distance between the roll axis and the hull surface can be approximated by:

$$r_f = \frac{(0.887 + 0.145 \cdot C_B) \cdot \frac{S_f}{L} + 2 \cdot \overline{OG}}{\pi}$$

Equation 4.2–21

with a wetted hull surface area S_f, approximated by:

$$S_f = L \cdot (1.7 \cdot D + C_B \cdot B)$$

Equation 4.2–22

4.2.4.4 Eddy Making Damping, B_{44E}

When using a simple form for the pressure distribution on the hull surface it appears that the pressure coefficient C_p is a function of the ratio γ of the maximum relative velocity U_{max} to the mean velocity U_{mean} on the hull surface:

$$\gamma = \frac{U_{max}}{U_{mean}}$$

Equation 4.2–28

The $C_p - \gamma$ relation was obtained from experimental roll damping data of two-dimensional models. These experimental results are fitted by:

$$C_p = 0.35 \cdot e^{-\gamma} - 2.0 \cdot e^{-0.187 \cdot \gamma} + 1.50$$

Equation 4.2–29

The value of γ around a cross section is approximated by the potential flow theory for a rotating Lewis form cylinder in an infinite fluid.
An estimation of the local maximum distance between the roll axis and the hull surface, r_{max}, has to be made.
Values of $r_{max}(\psi)$ have to be calculated for:

$$\psi = \psi_1 = 0.0 \qquad \text{and} \qquad \psi = \psi_2 = \frac{0.5}{\cos\left(\frac{a_1 \cdot (1 + a_3)}{4 \cdot a_3}\right)}$$

Equation 4.2–30

The values of $r_{max}(\psi)$ follow from:

$$r_{max}(\psi) = M_s \cdot \sqrt{\begin{Bmatrix} ((1+a_1) \cdot \sin(\psi) - a_3 \cdot \sin(3 \cdot \psi))^2 + \\ ((1-a_1) \cdot \cos(\psi) + a_3 \cdot \cos(3 \cdot \psi))^2 \end{Bmatrix}}$$

Equation 4.2–31

With these two results a value r_{max} and a value ψ follow from the conditions:

- For $r_{max}(\psi_1) > r_{max}(\psi_2)$: $r_{max} = r_{max}(\psi_1)$ and $\psi = \psi_1$
- For $r_{max}(\psi_1) < r_{max}(\psi_2)$: $r_{max} = r_{max}(\psi_2)$ and $\psi = \psi_2$

Equation 4.2–32

$$B_{44E_0}{}^{(2)'} = \frac{1}{2} \cdot \rho \cdot D_s{}^4 \cdot \left(\frac{r_{max}}{D_s}\right)^2 \cdot C_p \cdot \left\{\left(1 - \frac{f_1 \cdot r_b}{D_s}\right) \cdot \left(1 + \frac{\overline{OG}}{D_s} - \frac{f_1 \cdot r_b}{D_s}\right) + f_2 \cdot \left(H_0 - \frac{f_1 \cdot r_b}{D_s}\right)^2\right\}$$

with:

$$f_1 = 0.5 \cdot \{1 + \tanh[20 \cdot \sigma_s - 14]\}$$
$$f_2 = 0.5 \cdot (1 - \cos(\pi \cdot \sigma_s)) - 1.5 \cdot (1 - e^{5 - 5 \cdot \sigma_s}) \cdot \sin^2(\pi \cdot \sigma_s)$$

Equation 4.2–34

The approximations of the local radius of the bilge circle r_b are given as:

for $r_b < D_s$ and $r_b < \frac{B_s}{2}$: $\quad r_b = 2 \cdot D_s \cdot \sqrt{\frac{H_0 \cdot (\sigma_s - 1)}{\pi - 4}}$

for $H_0 > 1$ and $r_b > D_s$: $\quad r_b = D_s$

for $H_0 < 1$ and $r_b > H_0 \cdot D_s$: $\quad r_b = \frac{B_s}{2}$

Equation 4.2–35

For three-dimensional ship forms the zero forward speed eddy making quadratic roll damping coefficient is found by an integration over the ship length:

$$B_{44E_0}{}^{(2)} = \int_L B_{44E_0}{}^{(2)'} \cdot dx_b$$

4.2.4.5 Lift Damping, B_{44L}

The roll damping coefficient due to the lift force is described by a modified formula of Yumuro:

$$B_{44L} = \frac{1}{2} \cdot \rho \cdot S_L \cdot V \cdot k_N \cdot L_O \cdot L_R \cdot \left(1.0 + 1.4 \cdot \frac{\overline{OG}}{L_R} + 0.7 \cdot \frac{\overline{OG}^2}{L_O \cdot L_R}\right)$$

Equation 4.2–38

The slope of the lift curve C_L/α is defined by:

$$k_N = \frac{C_L}{\alpha} = \frac{2 \cdot \pi \cdot D}{L} + \chi \cdot \left(4.1 \cdot \frac{B}{L} - 0.045\right)$$

Equation 4.2–39

in which the coefficient χ is given by Ikeda in relation to the amidships section coefficient C_M :

$$\begin{array}{rl} C_M < 0.92: & \chi = 0.00 \\ 0.92 < C_M < 0.97: & \chi = 0.10 \\ 0.97 < C_M < 0.99: & \chi = 0.30 \end{array}$$

Equation 4.2–40

These data are fitted here by:

$$\chi = 106 \cdot (C_M - 0.91)^2 - 700 \cdot (C_M - 0.91)^3$$

with the restrictions:

- if $C_M < 0.91$ then $\chi = 0.00$
- if $C_M > 1.00$ then $\chi = 0.35$

Equation 4.2–41

4.2.4.6 Bilge Keel Damping, B_{44K}

The quadratic bilge keel roll damping coefficient has been into two components:

- a component $B_{44K_N}{}^{(2)}$ due to the normal force on the bilge keels
- a component $B_{44K_S}{}^{(2)}$ due to the pressure on the hull surface, created by the bilge keels.

$$B_{44K_N}{}^{(2)} = \rho \cdot r_k^{\ 3} \cdot h_k \cdot f_k^{\ 2} \cdot C_D \qquad \text{with:} \quad \begin{array}{l} C_D = 22.5 \cdot \dfrac{h_k}{\pi \cdot r_k \cdot \phi_a \cdot f_k} + 2.40 \\ f_k = 1.0 + 0.3 \cdot e^{-160 \cdot (1.0 - \sigma_s)} \end{array}$$

Equation 4.2–42

The approximation of the local distance between the roll axis and the bilge keel r_k is given as:

$$r_k = D_s \cdot \sqrt{\left(H_0 - 0.293 \cdot \frac{r_b}{D_s}\right)^2 + \left(1.0 + \frac{\overline{OG}}{D_s} - 0.293 \cdot \frac{r_b}{D_s}\right)^2}$$

Equation 4.2–43

The approximation of the local radius of the bilge circle r_b in here is given before.

Assuming a pressure distribution on the hull caused by the bilge keels, a quadratic sectional roll damping coefficient can be defined:

$$B_{44K_S}{}^{(2)} = \frac{1}{2} \cdot \rho \cdot r_k^{\ 2} \cdot f_k^{\ 2} \cdot \int_0^{h_k} C_p \cdot l_m \cdot dh$$

Equation 4.2–44

Ikeda defines an equivalent length of a constant negative pressure region S_0 over the height of the bilge keels, which is fitted to the following empirical formula:

$$S_0 = 0.3 \cdot \pi \cdot f_k \cdot r_k \cdot \phi_a + 1.95 \cdot h_k$$

Equation 4.2–45

The pressure coefficients on the front face of the bilge keel, $C_p^{\ +}$, and on the back face of the bilge keel, $C_p^{\ -}$, are given by:

$$C_p^{\ +} = 1.20 \qquad \text{and} \qquad C_p^{\ -} = -22.5 \cdot \frac{h_k}{\pi \cdot f_k \cdot r_k \cdot \phi_a} - 1.20$$

Equation 4.2–46

and the sectional pressure moment is given by:

$$\int_0^{h_k} C_p \cdot l_m \cdot dh = D_s^{\ 2} \cdot \left(-A \cdot C_p^{\ -} + B \cdot C_p^{\ +}\right)$$

with:

$$A = (m_3 + m_4) \cdot m_8 - m_7^{\ 2}$$

$$B = \frac{m_4^{\ 3}}{3 \cdot (H_0 - 0.215 \cdot m_1)} + \frac{(1 - m_1)^2 \cdot (2 \cdot m_3 - m_2)}{6 \cdot (1 - 0.215 \cdot m_1)} + m_1 \cdot (m_3 \cdot m_5 + m_4 \cdot m_6)$$

$$m_1 = \frac{r_b}{D_s}$$

$$m_2 = -\frac{\overline{OG}}{D_s}$$

$$m_3 = 1.0 - m_1 - m_2$$

$$m_4 = H_0 - m_1$$

$$m_5 = \frac{0.414 \cdot H_0 + 0.0651 \cdot m_1^{\ 2} - (0.382 \cdot H_0 + 0.0106) \cdot m_1}{(H_0 - 0.215 \cdot m_1) \cdot (1 - 0.215 \cdot m_1)}$$

$$m_6 = \frac{0.414 \cdot H_0 + 0.0651 \cdot m_1^{\ 2} - (0.382 + 0.0106 \cdot H_0) \cdot m_1}{(H_0 - 0.215 \cdot m_1) \cdot (1 - 0.215 \cdot m_1)}$$

$$m_7 = \frac{S_0}{D_s} - 0.25 \cdot \pi \cdot m_1 \qquad \text{for} : S_0 > 0.25 \cdot \pi \cdot r_b$$

$$= 0.0 \qquad \text{for} : S_0 < 0.25 \cdot \pi \cdot r_b$$

$$m_8 = m_7 + 0.414 \cdot m_1 \qquad \text{for} : S_0 > 0.25 \cdot \pi \cdot r_b$$

$$= m_7 + 1.414 \cdot m_1 \cdot \left\{1 - \cos\left(\frac{S_0}{r_b}\right)\right\} \qquad \text{for} : S_0 < 0.25 \cdot \pi \cdot r_b$$

The equivalent linear total bilge keel damping coefficient can be obtained now by integrating the two sectional roll damping coefficients over the length of the bilge keels and linearizing the result:

$$B_{44K} = \frac{8}{3 \cdot \pi} \cdot \phi_a \cdot \omega \cdot \int_{L_k} \left(B_{44K_N}{}' + B_{44K_S}{}'\right) \cdot dx_b$$

REFERENCE AND LITERATURE LIST

[i]International Maritime Organization IMO (2009) *SOLAS Consolidated edition.* Londen United Kingdom
[ii] Prof. Yoji Himeno, (1981). *Prediction of Ship Roll Damping state of the Art.* Michigan: University of Michigan College of Engineering.