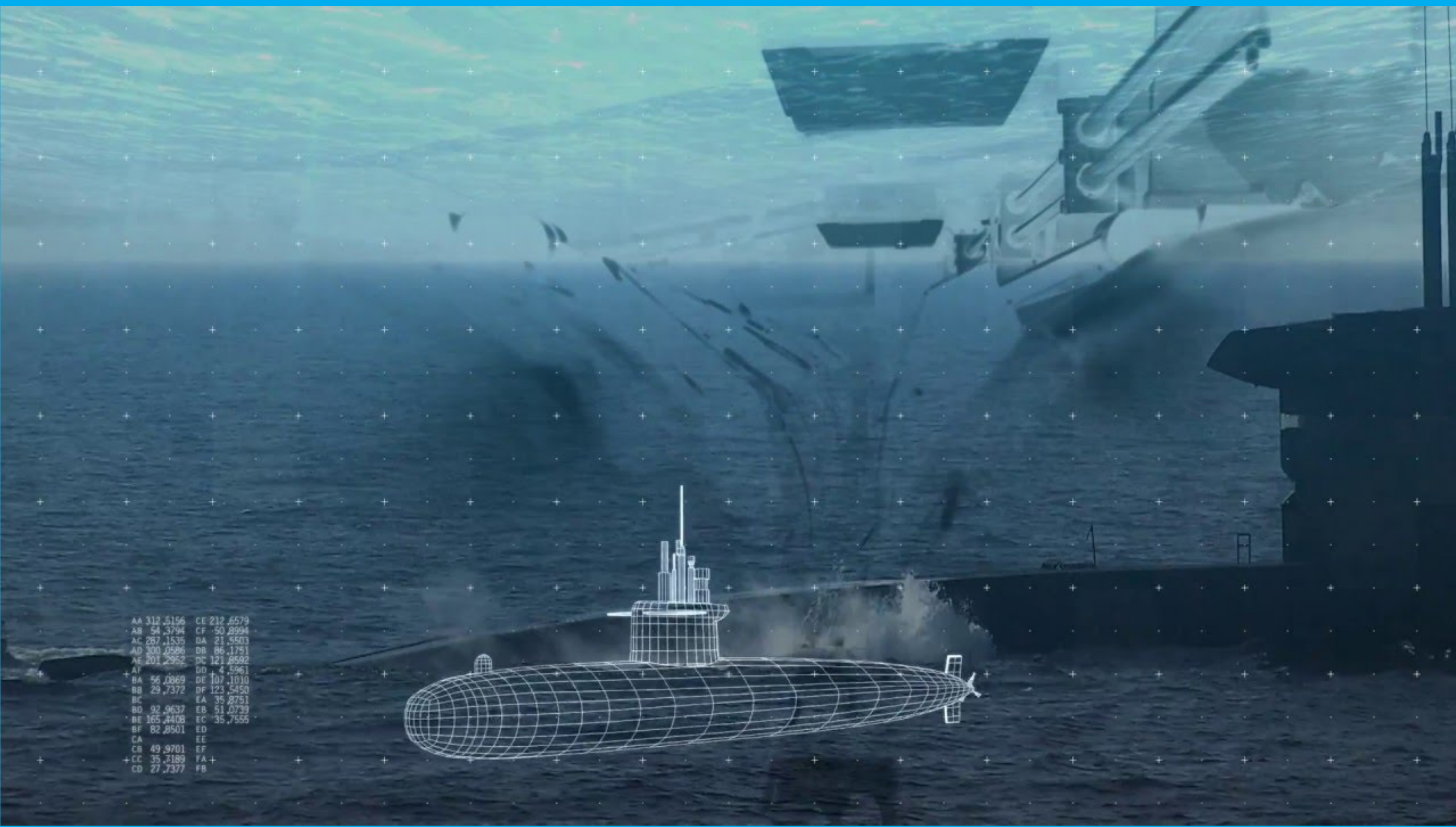
Ballast trim system for a modern SSK submarine

Master thesis

J.H. Verbaan

A research project on ballast trim systems for a modern SSK submarine based on the BB2 submarine hull by comparing and modelling different solutions

SDPO.18.006.m



TU Delft

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Thesis for the degree of MSc in Marine Technology in the specialisation of *Marine Engineering*

Ballast trim system for a modern SSK submarine

By

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Abstract

Keeping a submarine stable underwater is important to be able to execute missions. Therefore, one of the main systems on board of submarines is therefore the ballast trim system. This system is able to maintain the submarine balanced. There must be equilibrium in weight and buoyancy as well as in the trimming moment. This makes the ballast trim system a vital system on board of submarines.

The world around us changes, causing new mission specifications and requirements. This results in a different operational profile than before, which has consequences for the ballast trim system. For example, for a modern submarine hovering is desired, resulting in a faster and more accurate ballast trim system. Therefore, the existing systems must be reconsidered for modern SSK submarine applications. Next to the changed environment, new technologies are available, which might be applicable for a modern SSK submarine. The main goals of this research are: give insight in the ballast trim system and see which technologies are best to use in the ballast trim system. The technologies are tested on the 4000 and 2000 ton BB2 submarine.

In this research different empirical models were set up in order to simulate the behaviour of the different ballast trim systems. The results of these models are compared to each other based on energy, noise and redundancy. As an input, different mission profiles were made, which are likely to be executed by a modern SSK submarine. The ballast trim system was divided into three separate systems for the comparison, namely: trim, compensation and hover.

From the simulation made there was seen which modelled solutions were the best capable to use as a ballast trim system. In this research also the second order effects of waves were included. There is seen from the simulation that it is possible to compensate for the motions due to these second order wave effects. The simulations also showed that a new technique, the so called 'variable buoyancy system' is very promising.

With the results from this research it can be seen which system can be best used as ballast trim system for the three different tasks on board of the BB2 submarine. To maintain longitudinal equilibrium, it is best to use a centrifugal pump to transfer ballast water between tanks. To maintain balance between the weight and buoyancy forces of the submarine, it is best to use a plunger pump. It is also seen that a centrifugal pump in a system with a very large static head compared to dynamic head, is hard to operate and can even become incapable of expelling water from the tank. For hovering, a special dedicated system is needed. For the 4000 ton BB2 submarine, it is best to use a pre-pressurised tank to lose weight very fast in case of a change in seawater density. The 2000 ton BB2 submarine can use its snorkel mast to maintain equilibrium. For low frequency forces such as the second order forces due to waves, both studied BB2 submarines can use their snorkel mast to keep the submarine at depth.

Preface

My passion for ships developed young age. As a result of this passion I started to study Maritime Technology. The final results results of this study lies before you. My thesis, 'Ballast trim system for a modern SSK submarine', is the result of a year 'in depth' research in the ballast trim system of submarines. This thesis is written in order to obtain my master degree in Maritime Technology / Marine Engineering.

This thesis is written for the Defence Material Organisation in cooperation with the TU Delft. Together with my responsible supervisor of the Defence Material Organisation, Isaac Barendregt, and my supervisors of the TU Delft, Klaas Visser and Peter de Vos, we formulated my research question. Sometimes researching the field of ballast trim systems for submarines can be difficult but luckily most of the times the topic is interesting enough to keep motivated.

I would like to thank my supervisors for helping during the whole period with feedback and answers which helped me to keep on track. I would also like the submarine division of the Royal Netherlands Navy and particularly Bertrand Wajer who was always willing to answer my questions. I would also like to thank my colleagues at DMO for the advices and the cooperation. I hope this can be continued in the coming years. A special thanks to my direct supervisors, Feiko and William who helped me with my grammar.

Final a special thanks to my parents, Bert and Marieke, and my girlfriend Josien who helped me to keep focussed and supported me during my studies and especially during the final graduation process. The graphical skills of Josien can also be found back in this thesis, because she helped me with creating some of the figures. Without their help I would probably not have finished my studies.

To the readers, I hope this thesis can give the insight of the considerations that needs to be made when designing the ballast trim system for a submarine. Submarine design and technology is a small part of the broader field of shipbuilding. Nevertheless the functions of submarines differ far from the conventional surface ships. I hope that the reader will get an understanding of depth and trim keeping. This thesis is written between two different field of submarine design, namely marine engineering and submarine hydromechanics. There is tried to make a connection between these two fields. These fields of expertise have more overlap than acknowledged. After reading the reader is able to understand more about submarine movements and the systems that are able to balance the submarine.

J.H. Verbaan
Delft, February 2018

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Nomenclature

Abbreviations

ATT	Aft Trim Tank
AUV	Autonomous Underwater Vehicle
CT	Compensation Tank
D	Studied operational depth
D	the studied diving Depth
DMO	Defence Material Organisation
FASC	Full Authority Submarine Control
FTT	Forward Trim Tank
HNLMS	His/Her Netherlands Majesty's Ship
HT	Hover tank
HVO	hydrotreated vegetable oil
LOX	Liquefied Oxygen
MCA	Multi Criteria Analysis
meH	Metal Hybrid Hydrogen storage
meOH	Methanol
NATO	North Atlantic Treaty Organisation
PD	Periscope Depth
PD	Periscope depth
PEMFC	Proton Exchange Membrane Fuel Cell
RN	Royal Navy, navy of the united kingdom
RNLN	Royal Netherlands Navy
SF	Air Independent Propulsion
SF	Special Forces
SSK	conventional / diesel-electric
TT	Trim tank
TU	Technical University
USA	United States of America

Greek variables

α	Scaling factor	–

α_0	Proportion coefficient	$-$
β_0	Proportion coefficient ellipsoid	$-$
Δ	Displacement, weight	kg
ϵ	Elasticity of the material, in case of compression and extension of materials	$[-]$
ϵ	Pipe Roughness, in case of pipe resistance	mm
η	Efficiency	$-$
λ	Wave length	m
μ	Dynamic viscosity	Pa/s
ω	Radial velocity	rad/s
ϕ	Roll motion	$^\circ$
ψ	Yaw motion	$^\circ$
ρ	Density	kg/m^3
θ	Pitch motion	$^\circ$
ζ_a	Wave height amplitude	m
ζ_t	Wave height at time t	m
$\zeta_x x$	Resistance factor in pipe flows of element xx	$-$

Roman variables

Δz	Delivered head	m
$\dot{V}$	Volume flow	m^3/s
∇	Displacement, volume	m^3
ξ	Temporal relative volume flow	m
A	Area	m^2
a	Half length of ellipsoid	m
A_f	Frontal area	M^2
A_w	Wetted area	m^2
a_{33}	Added mass coefficient in z-direction	kg
a_{44}	Added inertia coefficient around the x-axis	kgm^3
a_{55}	Added inertia coefficient around the y-axis	kgm^3
A_w	Wetted area	m^2
B	Width of the submarine	m
b	Half width of ellipsoid	m
b_{33}	Damping coefficient in z-direction	kg/m
b_{44}	Damping coefficient around the x-axis	kgm^2
b_{55}	Damping coefficient around the y-axis	kgm^2

BM	Distance between centre of buoyancy and metacentre	m
c	Half height of ellipsoid	m
c	Speed of sound in certain fluid	m/s
C_cf	Static coulomb friction coefficient	–
C_cf^*	Dynamic coulomb friction coefficient	–
C_d	Drag coefficient	–
C_s	Static slip coefficient	–
C_s^*	Dynamic slip coefficient	–
C_vf	Static viscous friction coefficient	–
C_vf^*	Dynamic viscous friction coefficient	–
D	Diameter	m
d	Pipe diameter	m
E	E-modulus	Pa
e	Eccentricity coefficient	–
EN	Equipment number	–
F	Force	N
f	Friction factor of pipe	–
F_a	Amplitude of the force	N
F_a	amplitude of the force	m/s
F_t	Force at time t	N
F_z	Force in z-direction	N
g	Gravitational constant	$9.81 m/s^2$
GM	Distance between centre of gravity and metacentre	m
H	Height	m
H	Pump head	m
h	Height	m
H_{hull}	Height of the hull	m
H_{sail}	Height of the sail	m
I	Moment of inertia	m^4
k	Wave number	$1/m$
k'	Lambs' factor for trim	–
k_2	Lambs' factor for heave	–
K_v	Flow coefficient valve	–
KB	Distance between keel and centre of buoyancy	m

KG	Distance between keel and centre of gravity	m
L	Length of the submarine	m
L_{sail}	Length of the sail	m
L_{wl}	Sound Power Level, total sound power in the source	dB
LCB	Longitudinal centre of buoyancy	
M	Moment	Nm
m	Mass	kg
M_{xx}	Moment around x-axis	Nm
M_{yy}	Moment around y-axis	Nm
N	Amount of item	#
n	Pump speed	rpm
p	Pressure	Pa
p^{++}	Delivered pressure by pump	Pa
P_b	Break power	kW
P_b	Engine brake power	kW
P_t	Power at time t	kW
Q	Flow	m^3/h
Re	Reynold number	$-$
sfc	Specific fuel consumption	g/kWh
T	Depth of the submarine	m
T	Period	s
t	Thickness	m
t	Time	s
u	Velocity in x-direction	m/s
V	Volume	m^3
v	Speed	m/s
V_{xx}	Volume of item xx	m^3
x	Surge motion, distance	m
x_{xx}	X-position of item xx	m
y	Sway motion, distance	m
z	Heave motion, distance	m

Contents

1

Introduction

This master's thesis is written for the TU Delft and the Defence Material Organisation (DMO). The main goal of this thesis is to give insight into the ballast trim system of a diesel-electric, SSK, submarine. The hull notation SSK originates from the USA hull classification for hunter-killer submarines but is nowadays used as classification for diesel-electric submarines within NATO (NATO, 2013). The principle of having a ballast trim system on board a submarine is as old as the submarine itself. However technology evolves and new techniques with different actuators, and based on different principles, are available. In designing such a complex weapon platform as the SSK submarine, the designer should not only consider readily-used technologies but also innovative technologies. The challenge with submarine design is that because of the consequences of a vital system failure a proven concept is required.

Besides the evolution in technology, the world around us also changes. Originally the main task of a SSK submarine was to attack other ships. Nowadays the ability to gather intelligence, conduct covert special forces operations and being a platform for AUV operations is also expected of a SSK submarine. This requires other capabilities of the platform such as a very stable underwater position keeping. To this end the Defense Material Organisation, DMO, is interested in options for a very accurate ballast trim system on board of future SSK submarines, with a low noise signature.

1.1. Background

A ballast trim system is as old as the submarine itself. The basic principle of a ballast trim system has not changed despite the development of submarines over time. The missions executed by the submarine, however, have changed. A brief history of the submarine will be given first, followed by the plan for new submarines for the Royal Netherlands Navy, RNLN.

1.1.1. Historical overview

Cornelus Drebbel, a born Dutchman, is attributed to be the inventor of the submarine (Joly, 1917) (Hart-Davis, 2012). In 1620, he succeeded in submerging into the Thames, taking several people with him. He reached a depth of 5 meters and sailed underwater for 3 hours. Ballast water was taken from outside the hull and stored in leather bags. These leather bags could be filled and emptied from the inside in order to obtain a neutral buoyancy. The propulsion of this first submarine was supplied by oarsmen. An artist impression of the Drebbel submarine can be found in figure 1.1. In figure 1.1 the ballast system is also shown. However the admiralty of England at the time did not see the benefits of such an invention and so did not support further development.

Over time, the Drebbel submarine evolved into a modern weapon platform which is capable of staying below the surface for multiple days. One of the areas in development since the Drebbel submarine is in power configuration. Nowadays submarines have electric power on board. This results in different propulsion but also in different ways of submerging and surfacing. Electric machines have made it easier to manage pumps and control electronics have made it possible to automate the systems on board. The downside of having all kinds of actuators and electrical/mechanical systems is the noise production when used. This should be taken into account since a submarine's signature is operationally critical.

Figure 1.1: The Drebbel submarine

1.1.2. A modern SSK submarine for the RNLN

The last submarine to be built by the RNLN, HNLMS Bruinvis, was launched in 1992 (Ministerie van Defensie). HNLMS Bruinvis is a Walrus class submarine sailing under the Dutch flag. The submarine division of the RNLN currently consists of four Walrus class submarines. These submarines are planned to be replaced from 2027. Therefore a replacement for the Walrus class submarines is desired by the RNLN (Hennis-Plasschaert, 2016). The decision to start the replacement programme still has to be made. However, DMO has already started the study phase for several reasons. The first reason is to refresh their knowledge about designing a submarine, since the last submarine was launched 24 years ago, and new technologies have to be studied. The second reason why the RNLN is interested in research in submarines is that when the decision about building new submarines becomes positive, and the Walrus class is going to be replaced by new submarines, they will be able to come up with conceptual designs and specification quickly.

The RNLNs' submarines belong among the most effective weapon systems at sea (Hennis-Plasschaert, 2016). At this moment there is a shortage in effective weapon systems at sea within the NATO. This includes a shortage in multifunctional submarines. The Walrus class submarine fills a gap in required capability within the NATO. The Walrus class submarine distinguishes itself compared to submarines of other nations by its capabilities and resulting size. It is often said that the Walrus class submarine is a unique piece of weapon system capable of multiple important tasks worldwide in deep and shallow waters. The NATO countries are not obligated to invest in niche weapon systems, however it is advised to keep the current NATO inventory in mind when investing in new weapon platforms. Replacing the Walrus class submarines by submarines with at least the same capabilities instead of scraping them does fit in this philosophy.

The submarine of the future will differ from current designs (De Boer, 2015). Flexibility is an important issue these days. A submarine should be able to do a variety of tasks. Therefore thinking and designing modular is important. The tasks of submarines could change on short notice and the structure and main systems must be able to deal with these different tasks. Having a more modular design which can be equipped with different equipment for every task results in for example a ballast trim system with a wide operational profile. The future submarine should also be more silent since sonars are getting better increasing the risk of identification. A large autonomous unmanned submarine is not realistic for the near future, however smaller autonomous unmanned vehicles, AUVs, are already available and will be operated using the the submarine as main platform, which is challenging for the ballast trim system.

A modern SSK submarine must be able to execute the latest tasks for modern submarines. These modern tasks do differ slightly from the older ones (Hennis-Plasschaert, 2016). These tasks are as follow:

- Offer strategic influence
- Offer great and precise maritime strength
- Worldwide gathering, analysing and sharing of intelligence
- Perform special operations

One of the systems on board of a submarine is the ballast trim system. A ballast trim system can be found in almost every conventional ship. However on a submarine this system has a key task. The main feature of a submarine is that it can sail under the waterline. This is what distinguishes a submarine from 'surface' ships. The ballast tanks of a submarine are therefore important. The ballast tanks allow the submarine to maintain hydrostatic equilibrium underwater. This is a very delicate operation and therefore needs to be thought through well since the ballast trim system needs to be very accurate and reliable operated.

These new tasks for submarines results in different requirements for the ballast trim system. Also technology has evolved since the last build submarine by the RNLN. Therefore research is needed to see what options are possible for ballast trim systems and to look for any improvements compared to earlier classes. This research will give a better overview of possibilities for a ballast trim system of a modern SSK submarine.

1.2. Problem description

DMO is at the moment conducting studies into operational and technical aspects of modern SSK submarines. As seen, the ballast trim system is an important system on board of submarines. Since specifications for modern submarines are slightly different to those of the current class of submarines, research in the ballast trim system is needed. A few differences between the operational requirements of a modern SSK submarine and the Walrus class submarine which have consequences on the design of ballast trim system are:

- It will be more often required to trim on depths between periscope depth and deeper.
- The trim system must act faster and more accurate than the trim system on the current class submarines, it must have a lower response time.
- A modern SSK submarine may be equipped with anechoic tiles or coating which are compressible. The consequences for the ballast system need to be investigated.
- The requirements for noise production will be more strict which may result in other system solutions
- The submarine will be more often stopped while submerged and should maintain equilibrium, hovering. Hovering is at the current class very difficult

The design of a ballast trim system should be reconsidered. New technologies can probably be applied that could perform better than the existing system. Especially the development of AUVs' goes fast and in this field of expertise new insights can be used on the development of a military man controlled submarine.

Next to having new tasks, it is also not yet clear what impact the new requirements have on the ballast trim system. This study will give insight in the consequences of the new requirements on the ballast trim system. For example some missions could require that the submarine hovers while executing its mission at zero speed. By having zero speed the hydroplanes of the submarine are not able to create lift and the total hover capacity is delivered by the ballast trim system. Outside disturbances can be expected during hovering. These disturbances should be taken into account and quantified to see for which disturbances the ballast trims system should compensate.

Investigate the behaviour of a ballast trim system ,there should be first of all a functional specification of different systems including the expecting disturbances. Since the subject of submarines is sensitive information, detailed technical information on the ballast trim system of a submarine is hard to obtain.

1.3. Significance

This thesis will investigate the possible design solutions for a ballast trim system. The objectives in this research are as follows:

- Investigate design considerations and design solutions in the earlier class submarines.
- Formulate expected changes to the previous design considerations
- Quantify the consequences of these changes for the design of the ballast trim system.
- Propose alternative and innovative solutions for the new ballast trim system
- Analyse the proposed solutions of the possible ballast trim systems.

These goals are proposed by DMO in accordance with the different internal stakeholders. The information gathered about ballast trim systems on board of submarines must be helpful to make the right decision in the future. Giving insight in the multiple solutions is even more important than finding the best solution.

Having the results of this research will give a proper overview of different options that can be used as ballast trim system and how these work. It will also provide knowledge for the formulation of specifications for the ballast trim system of the new submarines.

1.4. Main- and sub-questions

To come up with a research that meets all the goals, a main-question and sub-questions are asked. The main question is: *What is the best design, new or existing, for a ballast trim system for a 2000 and 4000 ton modern SSK submarine based on existing and new operational and technical requirements, noise production and energy?*

To be able to answer this main-question, sub-questions are defined which will help to give an answer to the main question.

1. What were the design considerations and requirements of the ballast trim system of earlier class submarines of the Royal Netherlands Navy?
2. Which different design concepts of ballast trim systems are applied on board of the submarines of the Royal Netherlands Navy?
3. What are in general the different technical solutions for a ballast trim system?
4. What are the design requirements for a modern SSK submarine with respect to the position accuracy and how are they different from before?
5. Which trim disturbances could a modern submarine encounter in terms of loads and frequency?
6. How can the different ballast trim systems be as silent as possible without noise reduction methods?
7. What is the best solution as ballast trim system for a modern SSK submarine in terms of efficiency, redundancy, capability and noise production?
8. What are the sources of noise produced by the ballast trim system and how to reduce this noise?

The method of finding the answer to a sub-question is different for every question. The biggest part of this project will consist of modelling the different solutions of ballast trim systems in a analytical models . To be able to make these models, there is a need of information. This information is gathered in prior. Part of this gathered information is crucial as an end result. One of the goals of this project is to give insight in solutions for a ballast trim system. With the models the different ballast trim systems could be compared to each other.

1.5. Outline

This thesis is separated in nine chapters. Every chapter will provide step by step the needed information to in the end see which solution as ballast trim system is the best suitable. This thesis is set up as follow:

- Chapter 2: Submarine hydrodynamics
  In this chapter the basic understanding of submarine hydrodynamics will be explained. A basic understanding of the main hydrostatic principles will be described in order to understand the difficulties in maintain equilibrium underwater.
- Chapter 3: Designs and considerations in earlier classes
  This chapter research the options that were used in earlier Dutch designs of submarines. Not only the system itself is important, but also the main considerations behind the design are important. In this Chapter the first two sub-questions will be answered.
- Chapter 4: Different options in ballast trim systems
  This chapter will investigate the different technical solutions for a ballast trim system. This will mainly consist of a literature review about the different options as a ballast trim system in multiple applications. This chapter will give an answer to sub-question 3.
- Chapter 5: Disturbances and consequences for the ballast trim system
  To be able to come up with a solution of a ballast trim system, the required performance of the ballast trim system for the new submarines have to be studied. As a result a set of specifications for the ballast trim system will be formulated. This will lead to an answer to question number 4 and 5.
- Chapter 6: Suitable options for the ballast trim system
  Based on the requirement and possible systems, there can be set-up final options for a ballast trim system which do comply. In this chapter a model will be build to simulate the different options. sub-question 6 will be answered in this chapter.
- Chapter 7: Comparison of the different options
  This chapter will include a comparison of the different solutions. From this comparison an advice for the best suitable solution can be given. Then sub-question 7 can be answered.
- Chapter 8: Noise within the ballast trim system
  Noise production has to be analysed in order to quantify the system. Noise can be reduced. The different solutions of sound reduction should be considered if feasible. Sub-question 8 is answered in this chapter. This should also be done with a model that simulated the noise production.
- Chapter 9: Conclusions
  The conclusions will be summarised in this chapter to see if the key-question is properly answered.
- Chapter 10: Discussion
  In the last chapter all the assumptions that are made will be discussed. The result of the research project will also be tested and discussed.

2

Submarine hydromechanics

Submarine hydromechanics differ from surface ships hydromechanics. This chapter will briefly go into the subject of submarine hydromechanics. Since a submarine is able to sail above as well as below the waterline, both surfaced and submerged hydromechanics have to be studied for a submarine. First the hydrostatics of submarines will be discussed. After that, the hydrodynamics of submarines will be explained. Finally it will be researched how these hydromechanics are applied to maintain stability.

2.1. Submarine hydrostatics

To explain all motions of the submarine, first a system of axes needs to be determined. The system of axes which is used throughout this thesis, is the system of axes that is usually used for surface ships. This system of axis can be found in figure 2.1. The motions that are shown in figure 2.1 are as follows:

- x surge
- y sway
- z heave
- ϕ roll
- θ pitch
- ψ yaw

The motions which are important in this research are the heave motion, the pitch motion and the roll motion. The other three motions are not taken into account for this research.

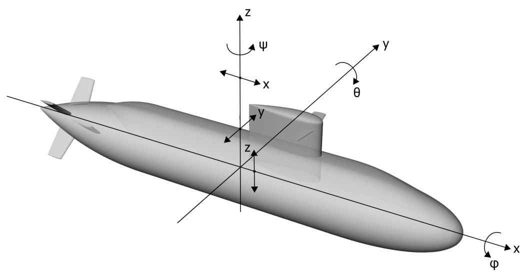


Figure 2.1: Coordinate system

The way objects float is described by Archimedes law, see 2.1. Archimedes law stated that the upward buoyancy force is equal to the weight of the displaced water. This means that when the weight of the submarine is less than the weight of of the displaced water, the submarine will float.

$$\Delta = \rho * V \tag{2.1}$$

With: Δ is the displacement in kg; ρ is the density of the fluid in kg/m^3; V is the volume in m^3.

To let the submarine float, the weight of the displaced water has to be higher than the weight of the submarine. This is not hard to obtain. Taking enough margin into account should avoid disasters while surfaced.

However a submerged submarine gives difficulties. A submarine in equilibrium underwater should have exactly the same weight as its displaced volume of seawater. If not in equilibrium, the submarine will tend to surface or submerge.

When a submarine is sailing, it is able to use its fins to create lift force(Burcher and Rydill, 1994). These lift forces can be applied downwards as well as upwards. Normally a submarine is ballasted in such a way that there is a little more buoyancy force than displacement force. This way the submariners ensure that the submarine will always tend to float to the surface and will not dive deeper. While in steady state operation submerged, the buoyancy should be neutral without the assistance of the hydroplanes.

2.1.1. Surfaced stability

The main parameters of a ship sailing on the surface are the centre of buoyancy and the centre of gravity. The centre of buoyancy, B, is the point where the buoyancy force acts. This point is determined by the underwater shape of the hull. It is actually the centre of the submerged part of the hull. The centre of gravity, G, is determined by all the weights in the submarine including the submarines' own weight. If the submarine becomes angled and all mass remains in place, the centre of gravity will not change. The centre of buoyancy however does change since the submerged part of the ship changes. The centre of gravity is assumed to be on the centreline of the submarine. The metacentre almost does not change if the submarine becomes angled. The centre of buoyancy rotates around this metacentre when the ships gets an angle. Initially when the submarine has no list, the centre of buoyancy is also located in the centreline of the submarine below the centre of gravity. With equation 2.2 the value for the initial BM can be determined. GM says more about the stability and the up righting moment of the submarine. Equation 2.3 shows how to get to a value for GM.

$$BM = \frac{I}{\nabla} \tag{2.2}$$

$$GM = KG - BM + KB \tag{2.3}$$

With: BM is the distance between centre of buoyancy and metacentre in m; I is the moment of inertia in m^4; ∇ is the displacement in m^3; GM is the distance between centre of gravity and metacentre in m; KG is the distance between keel end centre of gravity in m; KB is the distance between keel and centre of buoyancy in m.

As long as the metacentre is above the centre of gravity, the buoyancy force works as a restoring force if the ship becomes angled. If this is not the case the buoyancy force tends to make the angle even larger. Therefore it is necessary to make sure that while the submarine is surfaced the metacentre is always above the centre of gravity, i.e. a positive GM. An overview of the metacentre, centre of gravity and centre of buoyancy can be found in figure 2.2

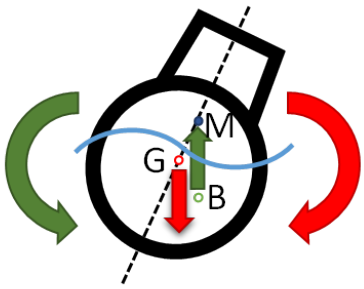


Figure 2.2: Surfaced submarine stability

Longitudinal stability above the waterline is applied in the same way. However since the longitudinal position of the metacentre is in this case proportional to length^3, the longitudinal stability is almost certain to be right. GM is in this case so high that the restoring moment for pitch becomes very high.

2.1.2. Submerged stability

When a submarine is fully submerged other phenomena appear. First of all a submerged submarine has no waterline and therefore no metacentre. Since there is no metacentre GM does not play a roll. In submerged conditions only BG is considered. BG must be negative to obtain a stable submerged submarine. This differs from the surfaced stability. A negative BG implies that the centre of buoyancy is above the centre of gravity, figure 2.3. Having a positive BG results in an unstable situation, figure 2.4.

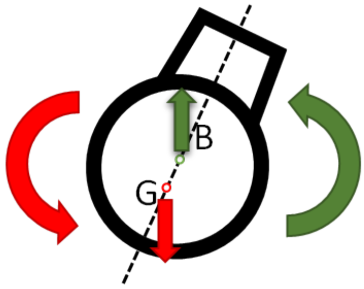


Figure 2.3: Stable submerged submarine

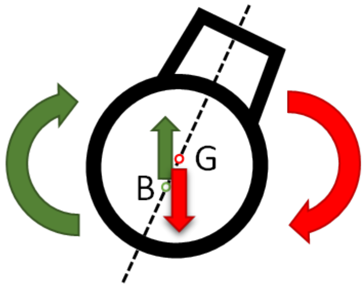


Figure 2.4: Unstable submerged submarine

Longitudinal stability in the submerged condition is more critical than in the surfaced condition. Since there is no metacentre, there is no reserve buoyancy that can work as a spring. The pitch in submerged condition depends on the longitudinal centre of buoyancy and the longitudinal centre of gravity. Having no waterline results in no restoring moment when having an angle. This results in that the submarine is only longitudinally in balance if the centre of buoyancy and the centre of gravity are lined up. Sailing submerged while having a pitch angle can result in a change of depth. This is not always desired and therefore the pitch angle should always be controlled in the right way. Another phenomena of having no equilibrium underwater is that the pitch angle will increase due to the lack of no restoring moment.

2.2. Submarine hydrodynamics

The submarine submerged hydrodynamics are also very different to the hydrodynamics of surface ships. As seen in the previous section the submerged submarine has no restoring force. The restoring force usually works as a spring resulting in stability. Therefore this term is left out of the ship's motion equation. The submarines' equations of motion are for this research only used for the heave, roll and pitch motions. Other motions of the submarine are present but not important for the study of the ballast trim system previously stated.

Heave will occur because of the external forces on the submarine in the z-direction and is described by equation 2.4. Roll will occur due to the asymmetry of tank positions and is calculated with equation 2.5. Finally pitch will occur due to moment on the submarines' according relation 2.6. The details of the system of axes can be found in figure 2.1.

$$F_z = (m + a_{33})\ddot{z} + b_{33}\dot{z}^2 \tag{2.4}$$

$$M_{xx} = (I_{xx} + a_{44})\ddot{\phi} + b_{44}\dot{\phi}^2 \tag{2.5}$$

$$M_{yy} = (I_{yy} + a_{55})\ddot{\theta} + b_{55}\dot{\theta}^2 \tag{2.6}$$

With: F_z is the sum of the forces in z-direction in N; M_{xx} and M_{yy} are the sum of the moments around the x- and y-axes in Nm; m is the mass of the submarine in kg; a_{33} is the added mass in kg; I_{xx} and I_{yy} are the moments of inertia around the x- and y-axes in kgm^3; a_{44} and a_{55} are the added moments of inertiaaround the x- and y-axes in kgm^3; b_{33} is the damping coefficient in kg/m; b_{44} and b_{55} are the damping coefficients around the x- and y-axes in kgm^2;

The coefficients in the equations of motion are usually obtained by model tests. When it is not possible to do model tests, literature gives approximations of these coefficients. However good approximations of these coefficients can only be made for simple shapes. Therefore it is not always possible to give an approximation. Chapter 6 will later give the approximation for the coefficients of motions for a submarine. As can be seen in the equations of motion, having an external force on the submarine will result in a continuous motion of the submarine. This shows again the impact of no restoring force.

2.3. Dealing with hydromechanics

The ballast trim system is crucial in submarines since they are capable of maintaining the submarine in equilibrium. Disturbances in the water can cause a pitching moment. Not only forces from the outside, but also changes in the centre of gravity can cause a pitching moment. For example men walking around the submarine causes a change in the centre of gravity. The trim system should be able to deal with this and keep the submarine in a stable position.

Keeping the submarine longitudinally stable can be done in theory in two ways. The first manner is to change the position of the centre of buoyancy. This can be done by changing the hull form. By decreasing the displaced volume at the front end of the submarine, the centre of buoyancy will move to the aft. By increasing the displaced volume at the front, the centre of buoyancy will move back to the front. In theory shifting the centre of buoyancy is a way of keeping the submarine stable, however in practice this is not seen. The second way of balancing the submarine longitudinally, is by shifting the centre of gravity. By transferring weight, the centre of gravity can be positioned in line with the centre of buoyancy, see figure 2.5. Normally submarines keep balanced by changing their centre of gravity.

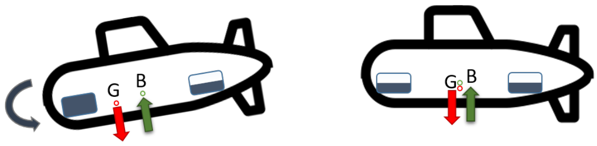


Figure 2.5: Longitudinal balance of a submarine

On the other hand there is the ballast system. This system is intended to make sure Archimedes law is fulfilled, equation 2.1. By ballasting the submarine one can make sure that the ship is in neutral buoyancy. Neutral buoyancy is necessary for a submarine to maintain depth as seen earlier. As seen, buoyancy is a very important parameter for the ballast system. The buoyancy force depends on the displacement of the submarine. For a hull that is fully submerged and not changing over time the total displaced volume is easy to determine. But when a submarine is diving to greater depths, the submarine is exposed to higher pressures(Burcher and Rydill, 1994)(NDRC, 1946). This increased pressure causes the submarine to compress. It is also expected that when using anechoic tiles, which will be discussed later, this effect will be larger. The buoyancy force becomes lower when the hull of the submarine is compressed. The ballast system should cope with this effect and must be able to compensate. In figure 2.6 the equilibrium between weight and buoyancy is graphically shown. If the buoyancy force is larger than the gravitational force, the submarine will surface. Therefore the ballast tank has to be filled which makes the submarine heavier. If the submarine becomes heavy, the submarine will sink. In this case the ballast tank needs to be emptied.

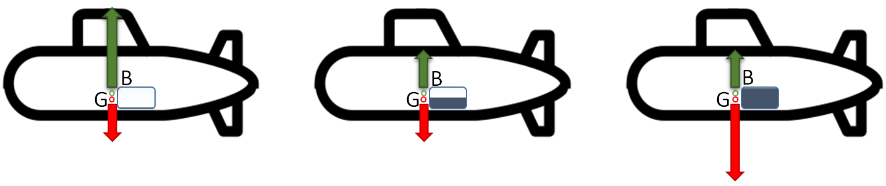


Figure 2.6: Weight and buoyancy of a submarine

2.3.1. Ballast tanks

One way of compensating the motions of a submarine is to use ballast water. Therefore ballast tanks are needed. The literature distinguishes two types of ballast tanks. The different types are main ballast tanks and trim and compensation tanks.

Main ballast tanks

The main ballast tanks are those tanks that are used for major adjustments of the submarine mass to allow it to operate submerged (Renilson, 2015). The main ballast tanks are usually placed outside the pressure hull. In order for the submarine to submerge, the main ballast tanks are flooded and the total weight of the submarine will increase. This will cause a downward force and the submarine is able to submerge. The most useful floodable tankage is situated in the lower part of the submarine, since this will give buoyancy while surfaced(Burcher and Rydill, 1994). The main ballast tanks are divided by bulkheads in the transverse direction and sometimes also in longitudinal direction(Burcher and Rydill, 1994). These bulkheads are sometimes naturally formed by structural members and sometimes by design. By having multiple main ballast tanks, the submarine is able to let some tanks flood and some not. This way the submarine is able to already have some tanks flooded in order to dive. Longitudinal separation is mainly meant to create a difference in starboard and portside. In this way the submarine is able to compensate for the heel angle. The main ballast tanks have a volume of 10-20% of the total volume of the submarine(Burcher and Rydill, 1994).

Trim and compensation tanks
The trim and compensation tanks are intended to deal with the minor adjustments that keep the submarine balanced while submerged. As seen previously, minor disturbances in the longitudinal centre of gravity will disturb the submarine's longitudinal stability. To maintain the longitudinal equilibrium, trim tanks are installed. This can be seen in figure 2.5 It is obvious to place these trim tanks at the extremities of the submarine. This way shifting ballast water from one to another tank has the largest effect. In most modern cases the trim tanks are not directly connected to the sea surrounding and therefore not subjected to high sea pressures. If the pressures inside the tank are low, the tanks can be considered as soft tanks.

An other effect during sailing is the change of weight inside the submarine. When using fuel, the fuel inside the tanks is replaced by seawater in case of a double hull submarine. Since seawater has a higher density than fuel, the submarine becomes heavier. Traditionally the compensation tanks are filled and emptied to compensated for these disturbances, figure 2.6. The compensation tanks are filled directly from the sea. Therefore they are called hard tanks. If the pressure is first brought down to atmospheric they can be considered soft tanks.

2.4. Functions

A distinction is made between three different functions in the ballast trim system. These modi are: trimming, compensating and hovering. With trimming all the pitch movements are compensated as seen before.

For compensating, the changes in weight of the submarine are compensated. Some argue that complete equilibrium is not necessary in order to operate safely (Burcher and Rydill, 1994). Having a slight error in the weight buoyancy equilibrium can be compensated by the hydroplanes. However, submarines usually have the a requirement to stop within a certain timespan. When stopping, the weight buoyancy equilibrium must be balanced. Another issue with submarines is that the speed of the submarine is not constant. Therefore it is not possible to create lift in the same way as aeroplanes do. Therefore it is important to maintain vertical equilibrium with the compensating system. Having some heave motions is not corrected by the compensation system

Hovering is in principle the same as compensating. However it it not expected that the hover system is operated as often as the compensating mode. Hovering is maintaining equilibrium while having a speed of zero. In this case hydroplanes, which are capable of delivering a lift force, can not be used. Sudden disturbances can cause the submarine to surface or submerge. The compensation system will compensate for the major disturbances. The hover system will compensate for another set of disturbances. Literature gives two kind of disturbances for which the hover system must compensate (Burcher and Rydill, 1994). These are: change of density and movements due to the waves. Some movement due to the waves is in normal transit mode compensated by the hydroplanes or not at all compensated. The change of density is in normal transit mode initially compensated by the hydroplanes after which the compensation system will slowly correct.

3

Designs and considerations in earlier classes of submarines

The RNLN built their first submarine at the end of the 20^{th} century. There will be a detailed overview of the ballast trim system on board of the Walrus class submarine. Also the design of the ballast trim system of the recently built Astute class of the Royal Navy, UK, will be elaborated in this chapter. Other examples of ballast trim systems are also discussed.

3.1. Walrus class

The Walrus class submarine is the current class of submarines of the RNLN. The design of this submarine is considered very effective within the category of SSK submarines. The last Walrus class submarine was launched in 1992 (Ministerie van Defensie), although the project 'Walrus class' started in 1978 (Naval Technology, b). Since the Walrus class submarines are operational, the detailed information is classified and therefore not available to all.

The ballast trim system of the Walrus class submarine will be discussed in appendix A.[1]

3.2. Potvis class

The Potvis class submarines of the RNLN are currently out of duty. The Potvis class submarine is often referred to as the 'three cylinder' because of the construction which contained three stacked cylinders. The ballast trim system of the Potvis class submarine is separated into two system. First is the trim system which compensates for the longitudinal trim. Second there is the compensation system which compensates for the weight buoyancy disturbances.

The trim system is a closed system. This means that the trim system is not connected to a seawater tank. The trim system is therefore also operated with fresh water. The trim is compensated with the so called 'helling tanks'. These four tanks are situated in the front and in the aft of the submarine. Between these two tanks, water is transported due to a pressure difference between tank in the aft and front of the submarine, shown in figure 3.1. There are two trim lines inside of the submarine. One on starboard and one on port side. One tank in front and one tank in the aft of the submarine is brought onto 5 bar pressure. This way water in transported to adjust the trim while sailing. When one pressurised tank is below its minimum water level, the process is reversed.

The compensation system works with a plunger pump. This plunger pump is able to overcome the pressure at large depths. This can be seen in figure 3.2.

[1] Not available on-line

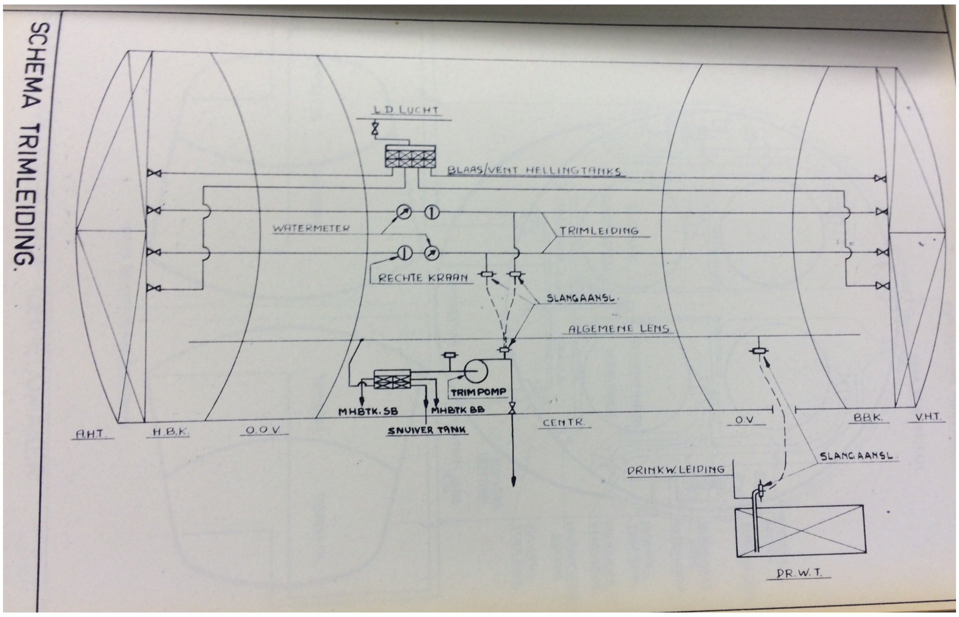


Figure 3.1: trim system of the Potvis class

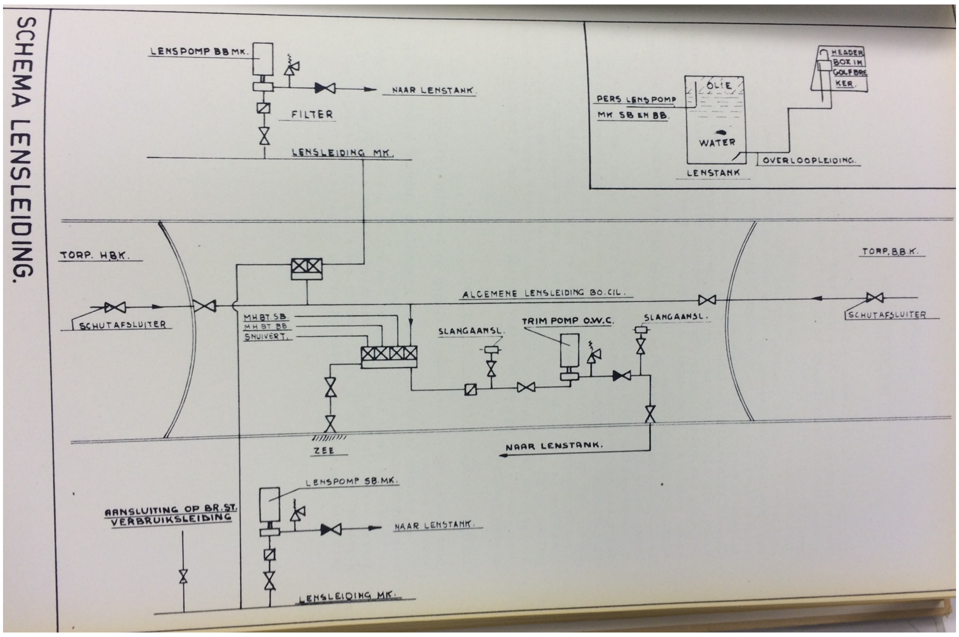


Figure 3.2: Ballast system of the Potvis class

3.3. Zwaardvis class

The Zwaardvis class of submarines is a retired class of the RNLN.

The ballast trim system of the Zwaardvis class submarine is combined. This means that all the tanks are connected to each other by the trim manifold.

The trim system consists of two trim tanks. These tanks are situated in the front and in the aft of the submarine. The water inside these tanks is transported with a centrifugal pump. This centrifugal pump can be switched in series or parallel operation. When switched to series the pump is able to pump water from the trim tanks to the sea. When switched to parallel, the trimming system has a high capacity for transporting water between the trim tanks.

For the compensation system the same principle of pumps is used as seen in the trim system. This allows the submarine to be compensated with a high capacity at low depths. For large depths the pumps can be switched into series creating a higher head but having a lower capacity. Figure 3.3 shows the ballast trim system of the Potvis class submarine.

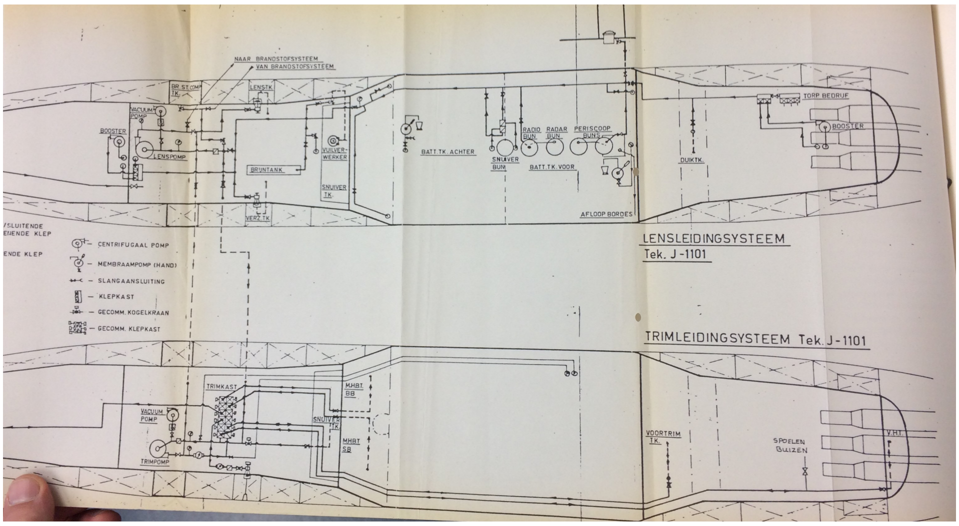


Figure 3.3: ballast trim system of the Zwaardvis class

3.4. Trafalgar/Astute class

The Trafalgar class submarine and the Astute class submarine are SSN, nuclear powered attack, submarines of the Royal Navy, UK. The Trafalgar class submarine was launched in 1983 (Naval Technology, a). The Astute class submarine is the newest asset to the RN and was launched in 2010. The Astute class consists therefore of the latest technology in the submarine field of the RN. The Trafalgar class is discussed first because the design of the Astute class is based on the Trafalgar class submarine. These two nuclear submarines are significantly larger than the Walrus class submarine, however the used techniques could still be applicable.

3.4.1. Design of the ballast trim system

The ballast trim system is separated into two main systems. There is the ballast system and the trim system. These two systems are normally operated independently, but can be cross-connected. This is done to fill or empty the trim tanks in such a way that the trim and compensation tanks can be used over a long time.

The ballast system consists of four tanks at two longitudinal positions; they both have a port and starboard tank. Two of these tanks are called the O-tanks and the other two are called the M-tanks. These tanks can be filled with seawater from outside. The ballast pump is able to pump out the ballast water to the outside. An overview of the ballast system of the Trafalgar class can be seen in figure 3.4. The bilge system is also connected to the compensation system. This way the bilge tanks can be emptied into the compensation tanks and from there expelled to the sea. This operation ensures that when emptying the bilge tanks, there is no change of total weight of the submarine.

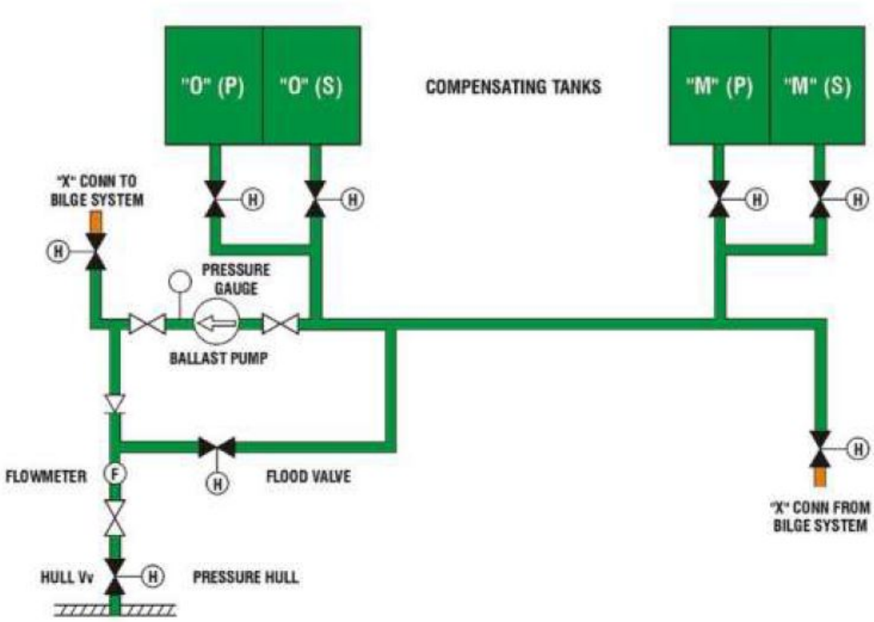


Figure 3.4: Ballast system of the Trafalgar class

The trim system consists of four types of tanks. These tanks are positioned on port side as well as starboard. This results in a total of eight tanks. The tanks in the trim system are:

- WRT tank, water round torpedo tank which contains the extra water to fill inside the torpedo tube when the torpedo is placed inside the tube;
- TOT, torpedo operating tank which is filled from the torpedo tube after a torpedo is fired;
- Forward trim tank, The forward trim tank to compensate for the trim angle;
- Aft trim tank, the tank that compensates for the trim angle at the aft.

In the trim system there are two pumps available to control the trim. Sanitary tanks are also connected to the trim system. When emptying the sanitary tanks, the trim system and the compensating/ballast system tanks are connected. The sanitary water will flow through the trim system into the compensation system before being expelled to the sea. An overview of the total ballast trim system of the Trafalgar class can be found in figure 3.5.

As an addition the Astute class submarines maintain one of their M-tanks at high pressure. The tank is brought pressurised by air. This way the Astute class is able to lose weight very fast in case of losing control of the submarine or in case of hovering. This is mainly done because the anechoic tiles, with which the Astute class is covered, can cause an uncontrolled descent of the submarine due to the compression. The pressurised M tank can lose weight very rapidly and prevent an uncontrolled descent.

3.4.2. Discussion Astute class

The ballast trim system of the Astute class is in general the same as the ballast trim system of the Trafalgar class. However the addition of the pressurised M-tank does give this type of submarine a broader operational profile. The pressurised tank allows the submarine to hover. In case of the Trafalgar class, losing ballast water very quickly was done by the compensation/ballast pumps. Using the compensation pump for hovering results in a too low flow rate for safe hovering operation. Therefore hovering is not possible with the Trafalgar class since it always needs lift from the hydroplanes to maintain its depth. Making the submarine heavier is done by opening the free flood valve of the compensation system. In this case the seawater pressure is high enough to fill the compensation tank.

Having the compensation tank on starboard as well as port side means that especially during a hover operation the submarine will list. Therefore during a hover operation the submarine must always pump

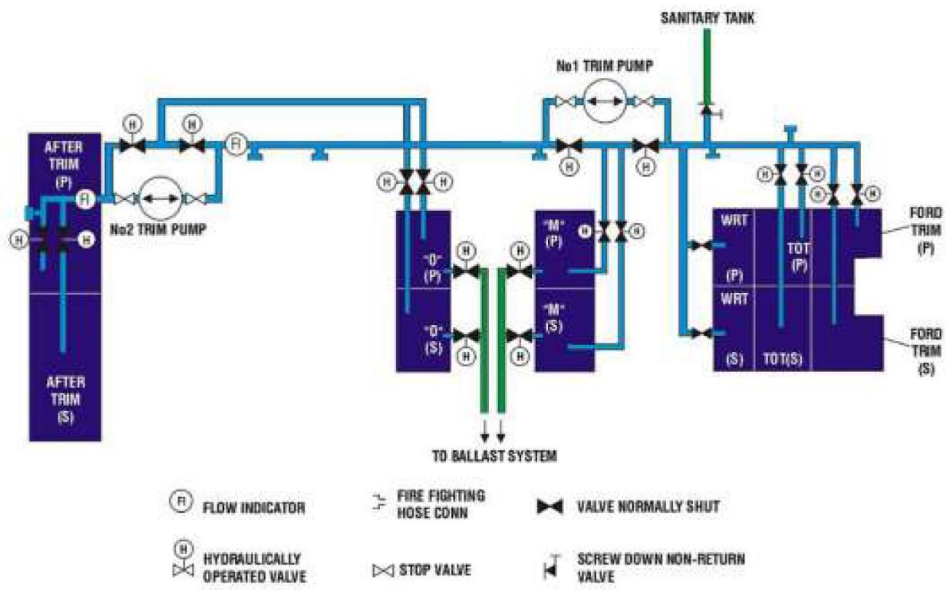


Figure 3.5: Total ballast trim system of the Trafalgar class

water from a starboard tank to a port side tank or vice-versa. When considering the system diagram, see figure 3.4 and 3.5, it is not possible to transfer water between two tanks of the same type.

The two systems, compensating and trim, are separated from each other by valves. This way the submarine is able to trim and compensate at the same time. However the valves can be opened allowing the trim system to use the compensation tanks as trim tanks if needed. It is not possible to use the trim pumps for compensation or the compensation pump for trimming. This way the trim tanks can not be emptied directly into the sea. Emptying the trim tank in the sea is very difficult since the operators have to compensate for trim as well as weight at the same time. For safety reasons it is better not to empty the trim tanks directly into the sea.

3.5. Other examples of ballast trim systems

Lately the same system as seen on the Potvis class submarine is used on modern submarines. The system was also a fully closed system. There was chosen for this system because it is more quiet. With four tanks, two of them pressurised, the right trim can be obtained. The tanks can be pre-pressurised in such a way there is no need for intermediate air supply in the tanks. It is known that when expelling air into the tanks, a lot of noise is produced. Therefore, to avoid this, the tanks can be pre-pressurised. With the pre-pressurised tanks, water can be expelled from the tanks very fast.

By dividing these two trim direction lines, the tanks don't have to be vented all the time when deciding to trim in the other direction. If one tank over time becomes empty, it can be decided to change the lines. This system is fully closed so there is no possibility to use the tanks also as compensation tanks. The distance between the two tanks in the front should be kept equal to the distance between the two tank in the aft. Otherwise when transporting the ballast water the submarine could capsize. This is not always possible and therefore compensating for trim can cause the submarine to list.

Another feature seen in the trim system of submarines is to pressurise both of the trim tanks to an equal pressure. A centrifugal pump transfers the water between the trim tanks. The tanks are always kept at equal pressure. This allows for the trim tank to also be used as a fire-fighting tank. Another advantage of having the tanks pressurised is that in case of, for example, an emergency dive, the forward trim tank can be punctured allowing trim water to flow rapidly from the aft to the front of the submarine. In this case the tanks were pressurised at 5 bar.

4

Different options in ballast trim systems

Historically, different configurations of ballast trim systems have been used on board of submarines. This chapter will give an overview of these systems including systems which are used on AUVs'. Research is conducted into systems that could be used for trimming, compensating and hovering. These three functions are crucial for the submarine to maintain equilibrium while executing its mission.

4.1. Trimming

In general there are two types of trimming systems found on board of submarines, the open and the closed trim system. These two trimming systems will be explained in this section. The systems that were found all use fluid as ballast. There can be a difference in transporting the fluid by blowing or by pumping.

4.1.1. Closed trimming system

In a closed trim system the trim tanks are only connected with each other(Burcher and Rydill, 1994). In most cases of a closed trim system there are two tanks, one situated at the front and one situated at the back of the submarine. By transferring water between these two trim tanks, the weight of the submarine remains the same and therefore the buoyancy force does not change. Usually seawater is pumped, but also fresh water is seen in closed systems. It is also seen that AUVs used fluids with a higher density such as mercury.

4.1.2. Open system

An open trim system always uses seawater as the medium. These systems can be fed by the compensation tanks or directly by the sea. In case there is a connection between the compensation tank and the sea, seawater first is transported into the compensation tanks and after that distributed to the trim tanks. Some configurations consists of multiple compensation tanks in the aft as well as in the front of the submarine, making it possible to use these tanks as trim tanks as well as compensation tanks. If the water is not directly flooded into the trim tanks, the water first floods into the main compensation tank and is pumped to the trim tanks and the trim tanks can be soft. The trim tanks can also be flooded directly by seawater from outside the pressure hull. In this case the tanks need to be hard in order to withstand the higher pressures. In some other cases it is seen that the trim pump and the compensation pump are the same pump. This saves space and weight, but means that only one function can be executed at the time.

4.1.3. Pumping

In most cases the trim system is operated by a pump, which displaces the fluid between the trim tanks (Burcher and Rydill, 1994). If operated by pumps and having the tanks at atmospheric pressure, the advantage is that the pump pressure only has to be high enough to overcome the internal system losses. The speed of transported fluid depends mainly on the pump capacity. However choosing a pump that is suitable as a trim pump is complicated. Since the pump is operated in a submarine, noise is an important parameter. Cavitation can occur in pumps which produces a lot of noise. The pump characteristics should therefore be carefully studied in order to make sure that the specific pump is capable of operating in its design point, given the conditions.

4.1.4. Blowing

Another way of transporting fluid between tanks is by exploiting a pressure difference between the tanks (Burcher and Rydill, 1994). If the pressure in, for example, the aft trim tank is higher than in the forward trim tank, the fluid will flow from the aft to the forward trim tank. This pressure difference can be created by air. High pressurised air can be blown in one of the trim tanks. This creates a high pressure inside the trim tank. If the pressurised trim tank has to receive water, it has to be vented first. This can be done by opening the vents. Normally the trimming tanks are vented within the pressure hull. This will cause pressure increase within the pressure hull. The downside of having a high pressure in the pressure hull is that toxic gasses can be formed. Therefore the submarine needs to snorkel in these cases to lower the pressure inside the pressure hull. Usually high pressurised air is stored in gas bottles. If the high pressure air is needed somewhere in the submarine, the closing valve can be opened. The empty air bottles can be filled with a compressor when snorkelling.

An other system that uses air to transport the ballast water is seen in chapter 3. This system consists of four trim tanks, two in the aft and two in the front of the submarine. One tank in the aft is connected to one tank in the front. one of these tanks is pressurised and the other is at atmospheric pressure. By opening the valve, ballast water will flow from the pressurised tank to the vented tank. By having one trim tank in the front and one in the aft at pressure, the trim can be adjusted by opening the right valve. When the pressurised tanks are at their minimum water level, the process is reversed.

4.2. Compensating and Hovering

Compensating is in general the same as hovering. The distinction in this thesis is that the hover system will react to forces due to waves and due to a change of density. These are in general disturbances for which a high flow is needed. Hovering is very desired as there will be seen in chapter 5. In the past hovering was not always possible or at least not for long. Deployment of the submarine in an other way than in the past makes it very important to enable hovering. On board the Astute class and the Trafalgar class submarines there are multiple compensation tanks. All these tanks can be filled by opening the flood valve. All the tanks can be emptied with the use of a pump. However one tank of the new Astute class is always pressurised and can therefore be emptied by just opening the main hull valve (Purvis and Philip, 2005).

4.2.1. Propelled

Autonomous unmanned vehicles are usually initially ballasted on the surface until they have a neutral buoyancy (Zhao et al., 2016). After that the vertical movements are controlled by propellers. The hovering capability is also delivered by propellers that can deliver a trust in vertical direction. A good example of the vertical propellers can be found in figure 4.1. These kind of AUVs'are known for their high performance in position keeping.

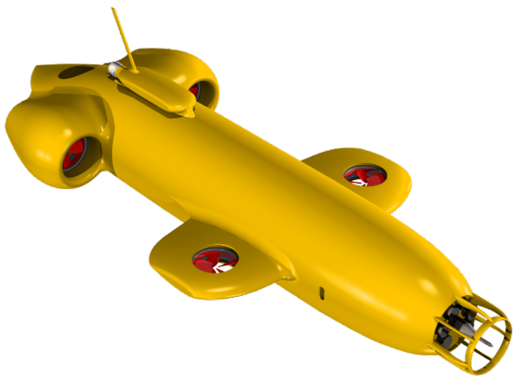


Figure 4.1: Vertex AUV of Hydromea with vertical propellers

For executing these hovering manoeuvres one could also use thrusters instead of propellers. Thrusters are able to deliver thrust in multiple directions. In case of a submarine, a thrusters can be placed on the side. This way they are able to deliver thrust in horizontal direction as well as in vertical direction.

Having the thrusters installed at the side of the submarine results in an odd shape of submarine. This leads to a higher resistance and increases sonar-detection vulnerability.

4.2.2. Ballast water

Hover control can also be done by taking in and letting out ballast water. Disturbances in the water cause a change in the buoyancy side of the Archimedes equilibrium, see 2.1. By changing the mass of the submarine the equilibrium can be regained. Taking seawater in underwater is done by free flood holes in the hull. The underwater pressure is sufficient enough to let the water flow into the tanks. For now there are two ways of ejecting the ballast water back to the sea. The first option is by pumps and the second option is by pressurised air(Purvis and Philip, 2005).

Pumping

Most common is the use of pumps to discharge the ballast water. The pump must be able to overcome at least the pressure difference between the tank and the outside seawater.

There are different types of pumps; the most commonly used pumps on board of submarines are the centrifugal pump and the plunger pump. In the case of the centrifugal pump, the water flows in at the centre of the impeller. The blades inside the centrifugal pump accelerate the fluid causing a higher pressure at the exit of the pump in accordance with Bernoulli. Figure 4.2 gives an overview of a centrifugal pump. The advantage of the centrifugal pump is that it is able to deliver a high flow. However they are more common in low pressure systems. The noise production of the centrifugal pump is usually low. The largest component in the noise production of the centrifugal pump is the air borne noise.

The plunger pump is also used in submarine applications. The advantage of this piston pump is that it is capable of having a high output pressure. At certain depths the submarine's compensation system and hover system need high output pressure. For every ten meters of water depth, there is a need of approximately 1 bar pressure. When the piston moves outwards it creates a lower pressure in the chamber causing the inlet valve to open and let the fluid flow inside. If the piston is moved inwards it creates a high pressure in the chamber which opens the outlet valve and forces the fluid out of the chamber. The working principle of the piston pump is shown in figure 4.3. An issue with the plunger pump is the noise production. Pressure fluctuation will occur due the the moving plunger. This is usually compensated by noise measurements.

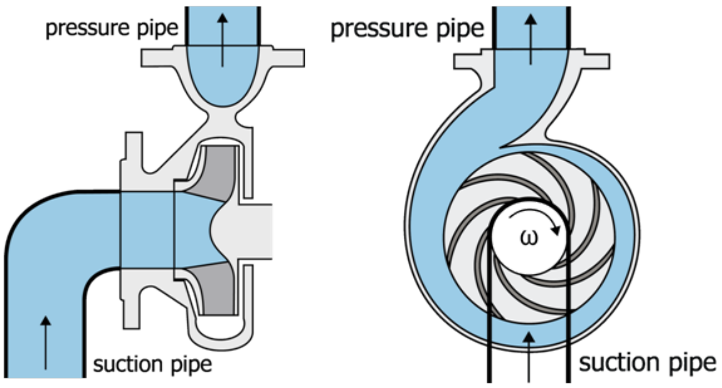


Figure 4.2: Working of a centrifugal pump

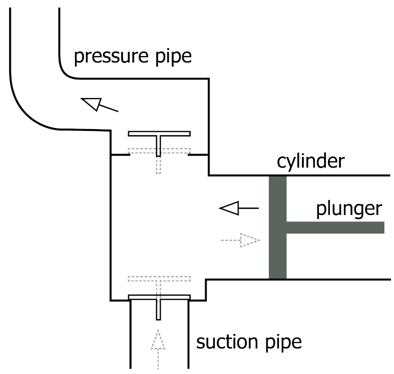


Figure 4.3: Working of a piston pump

Blowing

Other research also considered the possibilities of an open system using blowing and venting for hovering control. The main advantage of using pressurised tanks for hovering is the fast response time and the accuracy that can be achieved (Font and García-Peláez, 2013). The tank can be free flooded, and controlled by the air pressure in the top of the tank. The deeper the submarine submerges in the water the higher the air pressure must be to prevent more seawater flow into the tank. This shows also directly the downside of this system. The blowing system will need a lot of pressurised air. Pressurised air on submarines is transported in bottles. Therefore having more pressurised air on board results in a heavier submarine. Another downside of this system is the noise. Venting air causes a lot of noise. Not only when venting the tank noise is produced, but also when filling the tank with air. This air is stored at high pressures. Another disadvantage is that transporting high pressure air through pipes produces a lot of noise. The layout of this hover tank based on blowing and venting can be found in figure 4.4.

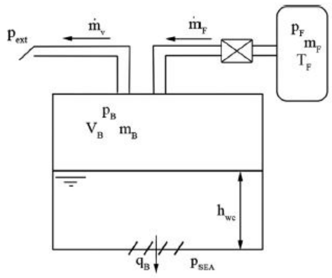


Figure 4.4: A hover tank based on blowing and venting, (Font and García-Peláez, 2013)

On board of the Astute class of the Royal Navy there is an a option that includes air pressurised compensation tanks in which the submarine is able to discharge the water fast down to intermediate depths (Purvis and Philip, 2005). The principle is to pressurise the compensation tank with the air. This way by opening the the outlet valves the water will flow to the outside because of the pressure difference. The tank on board the Astute class is pre-pressurised. This way the submarine can be silent when needed. Only when the tank needs to be pressurised the system produces noise. This system is also not directly coupled to the sea.

4.2.3. Changes of buoyancy

In some cases for hovering only a small amount of force is needed for a short time to compensate the movements of the submarine. It is known that an experienced submarine commander is able to compensate these by using the submarines' periscope or snorkel mast (Burcher and Rydill, 1994). In principle the commander is in this case changing the submarines' total volume and therefore the buoyancy force. By forcing the periscope or snorkel upwards the submarine becomes more buoyant and obtains more upwards forces. By moving the periscope or snorkel downwards, the submarine becomes less buoyant. Since only a few kilograms can already make a difference in hover operations, controlling the submarine this way can be executed very accurately.

It is known that fish use the buoyancy principle to have a neutral submerged buoyancy. Fish have an organ called the swim bladder. This organ works like a lung. The swim bladder can be filled with oxygen from the blood and emptied with a muscle. This way a fish is able to control its own buoyancy. Figure 4.5 shows this organ. Some AUVs use hovering systems that are based on the principle of the swim bladder of a fish (jun Xu et al., 2008). This so called bionic swim bladder is able to mechanically create more buoyancy. In figure 4.6 this system is shown. This is done by a moving piston. It looks like a piston pump, but the chamber is directly coupled to the sea. The piston is controlled by a linear electric motor. By moving the piston inwards, the total volume of the submarine decreases. By moving the piston outwards the volume of the submarine is increased. This way the AUV is able to change its buoyant forces.

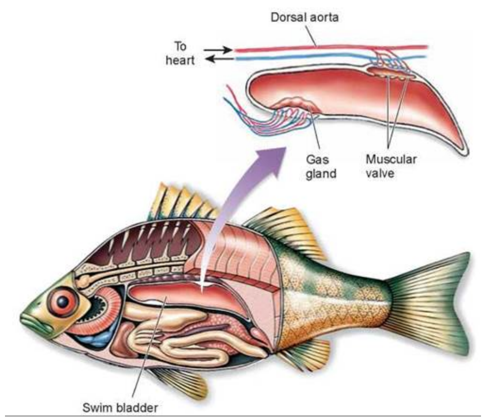


Figure 4.5: Swim bladder, buoyancy control of fishes

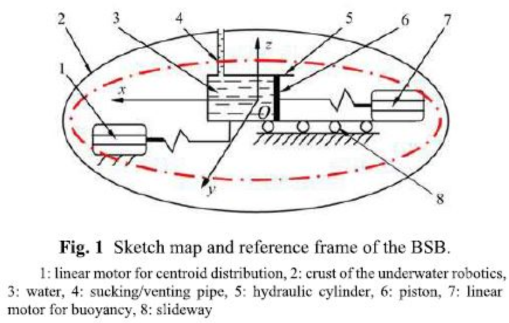


Figure 4.6: Bionic bladder, (jun Xu et al., 2008)

4.3. Automation

A modern ballast trim system should be able to control itself. Most of the ballast trim systems can be divided into three operation modes: automatic, semi-automatic and manual(Mansfield and Venn, 2011). These three operation modes make it possible for the submariners to operate more flexibly.

- Automatic
  The automatic mode on the control system makes it possible for the submarine to operate on autopilot. In this case a computer makes the decision about how to control the submarine. The operators are not involved in this mode.

- Semi-automatic
  In this cases the operator gives the commands to the system. The system will make sure that all input given by the operator will be translated into commands to the ballast trim system.

- Manual
  In case of manual operating, the system will not listen to the computer algorithms. The system listens in this case only to the operator and relies on his expertise. The system can be controlled by operating the valves and pumps manually.

The main focus in this thesis will be on the automatic system. The system should in practice also be operated in semi-automatic and manual mode. A fully reliable automated system is difficult to realise. A good model is needed which can reflect the submarines' behaviour.

Since the submarine is only in an early stage of design and the final design is far from ready, the ballast trim system should be modelled on a computer. With computer software disturbances can be simulated. The ballast trim system will give its response to these disturbances and the response will be used to see which fits best.

Conventional submarine control divides the different tasks the same as seen in the beginning of this chapter (Mansfield and Venn, 2011). There are in general three compensation modi which need to be considered. These three modi are submerging, compensating and trimming. In most cases the systems for these modi are physically separated. It therefore makes sense to separate the control sequences as well. A traditional model of a control system on board of submarines can be found in figure 4.7.

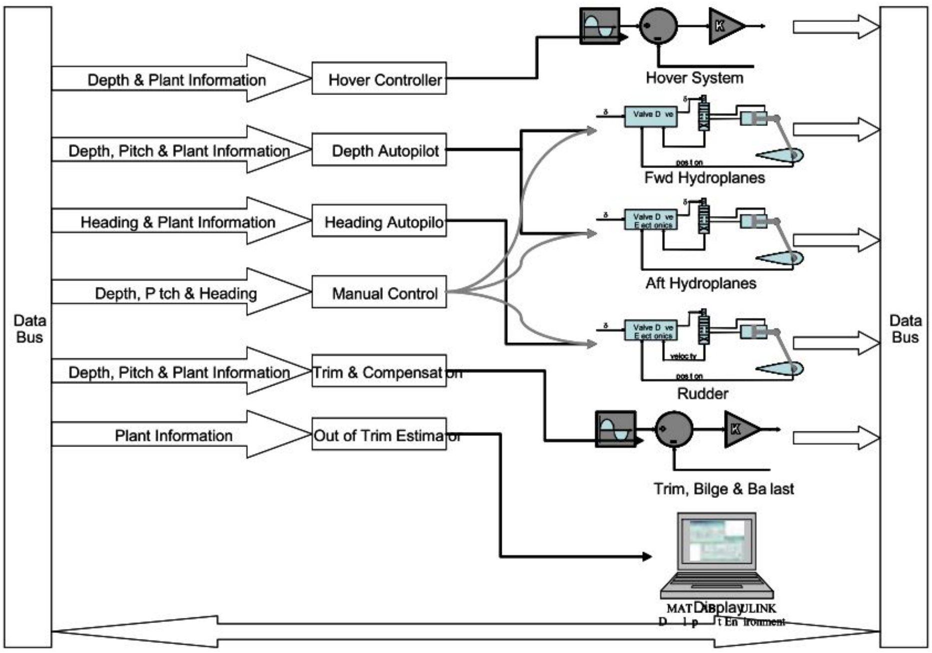


Figure 4.7: Traditional submarine control, (Mansfield and Venn, 2011)

For modern applications these systems are more combined. In the UAVs' for example, tanks and pumps are combined for the different tasks (Shawn A. Woods, 2012). A two-tank configuration forces the designer to create a system in which the different tasks are executed by the same operating system. First the system ensures a neutral buoyancy. This is done by initially filling the ballast trim tanks. There are two tanks that can be partially filled with ballast water. This ballast water can be pumped between the tanks. By this the trim can be adjusted. There is also the option to empty the tanks. By filling and emptying the tanks the submarine can be compensated and there is the ability to hover.

On the new Astute class submarine these systems are physically separated from each other, but the operating system is combined (Mansfield and Venn, 2011). The operating system does not only consider one solution to compensate for a disturbance, but tries to see what consequences it has for the other functions of the submarine. For example, compensating the weight of the submarine by filling a tank that is not exactly at the centre of gravity results in a trimming moment. This needs to be compensated by the trim system. Therefore operating one system results also in operating the other system.

New developments in position control of submarines are combining all the different tasks in one operating system (Mansfield and Venn, 2011) (Thomas, 2014). A system like this should deal with multiple situations. The system is able to make the right decision for when the submarine is sailing but also for when the submarine has to hover at zero speed. In this case already the difference in actuators can be found. In the first case the hydroplanes on the side of the submarine need to reposition and in the second case the submarine can for example take in and let out ballast water. If the ballast trim system is able to maintain equilibrium at zero speed, it is also able to maintain equilibrium when the hydroplanes are able to help. Figure 4.8 shows an example of an operating system called FASC. FASC, Full Authority Submarine Control, is developed by Stirling Dynamics. The FASC operating system combines all the different tasks of the submarine in one system. These new developments in operating systems result in a faster and more accurate system especially when operating in autopilot mode. Stirling Dynamics has already developed the operating system for the hover control and ballast control of the Astute class submarines of the RN. The FASC is not yet tested in real life, but the simulations that were done look very promising.

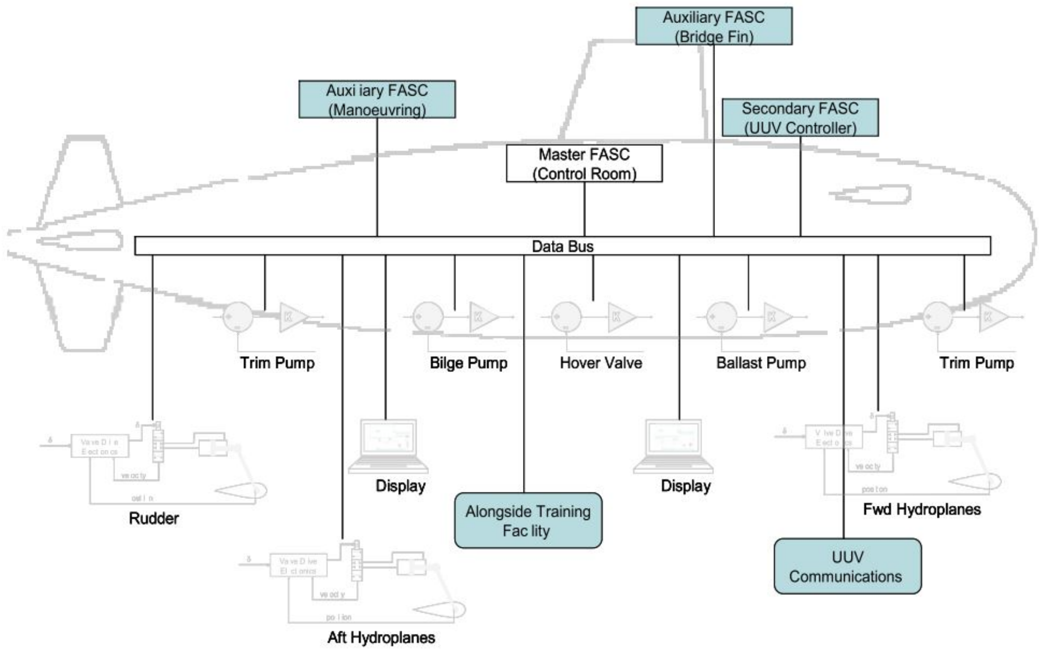


Figure 4.8: FASC operating system, (Mansfield and Venn, 2011)

Stirling Dynamics is currently still working on improving the hover, ballast and trim system on the Astute class submarine. They are updating the operating system, which should result in more safe operations.

5

Disturbances and consequences for the ballast trim system

Rapid developments in technology make it difficult to anticipate the possibilities for a new and modern SSK submarine suited for missions after 2025 (De Boer, 2015). First of all sonar technology becomes more effective allowing navies to detect submarines more easily. To avoid detection, one of the counter measures is to cover the hull of the submarine with anechoic tiles or coating. The downside of these tiles or coatings is that they are very compressible, and therefore change the volume of the hull with changing depth.

A modern SSK sails more often in littoral waters and has to execute new types of missions compared to the past. All these new tasks and design changes result in different considerations for the on board systems. This chapter elaborates on the specifications of a modern SSK submarine and what implications this has on the ballast trim system. First the generic hull design will be discussed. This generic hull will be used to find different disturbances that the submarine can encounter. The three different types of disturbances that are considered are: those impacting every submarine, disturbances due to the sea and specific mission related disturbances. Also consequences of design considerations are discussed in this chapter.

5.1. Submarine hull design

A generic submarine hull is used to study the disturbances on a submarine. Designs of submarines are often highly classified. This makes it to hard conduct tests on submarines. In order to enable studies in an early design stage, a generic hull model is used. A generic hull is a great tool to do in-depth research on submarines.

Drawings and test results of modern SSK submarines are confidential, therefore a non-existing hull is used (Overpelt et al., 2015). The Australian Defence Science and Technology Group and the DMO cooperate to study submarine behaviour. The BB2 design is set-up to enable this international research. The BB2 hull is based on the Joubert hull design. The original Joubert hull form is a modern submarine hull in which lessons learned from the past are applied (Joubert, 2004) (Joubert, 2006). The original design is a three deck submarine which has a displacement of 4000 ton when submerged. The BB2 variant of the Joubert design has a two deck lay-out but the same hull shape. The main particulars of this hull can be found in table 5.1. The BB2 design will be dealt with as a double hull submarine. This results in a submarine with an inner pressure hull covered by a visible hull which is not pressure resistant.

Table 5.1: Main particulars Joubert

Length	70.2	[m]
Width	9.6	[m]
Height hull	10..6	[m]
Displacement (submerged)	4000	[t]
Fullness parameter (CQ)	0.85	[-]

A sketch of the BB2 submarine based on the 'Joubert' hull can be found in figure 5.1

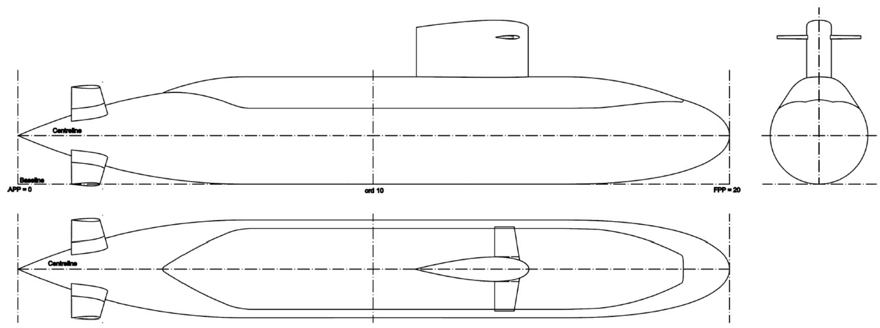


Figure 5.1: BB2 Joubert hull

Multiple tests have been conducted on the BB2 hull to study its free sailing capabilities(Overpelt et al., 2015). These tests were performed to see if the hydroplanes were effective to compensate the behaviour of the submarine when sailing. In these tests the submarine had a velocity and therefore lift on its hydroplanes. The tests are especially done to see how the hydroplanes react. A submarine at zero velocity results in having no lift on the hydroplanes.

For this study the BB2 model will also be used. The ballast trim system will be designed as if it is installed on the BB2. The BB2 hull design is for now the best available hull to test for the design of the ballast trim system. Since all exploratory studies for the new build submarine are done with the BB2, the study for a ballast trim system will be no exception. The behaviour due to disturbances of the BB2 will be used as input for the ballast trim system. The geometry of the submarine determines where tanks can be placed. The position of the trimming tanks determines the reaction of the ballast trim system to disturbances and therefore their position is needed.

The model of the BB2 submarine hull is available online. This model is used to find all characteristics of the submarine. For this research there are two models used. First is the original 4000 ton BB2 Joubert hull to which will be referred to as submarine 1. The second model is the same BB2 joubert hull, but scaled to a 2000 ton BB2 submarine. The 2000 ton submarine will be referred to as submarine 2. This scaled down submarine is used to see how the size of the submarine influences the ballast trim system. A smaller submarine is expected to have also a smaller ballast trim system in terms of tanks and pumps. To verify this assumption submarine 2 is studied.

The basic hull of the submarine is the BB2 Joubert hull, which can be found in appendix B. In Rhino[1] the required characteristics are measured. To make the model for submarine 2, a scaling factor needs to be determined. This is done with equation 5.1. This results in a scaling factor of 0.794.

$$\alpha^{1/3} = \frac{\Delta_{submarine}}{\Delta_{model}} \tag{5.1}$$

With: α is the scaling factor; Δ is the displacement in tonnes.

With the scaling factor all the characteristics of the BB2 Joubert submarine can be scaled to a submarine with the same shape, but a displacement of 2000 ton. The characteristics of both submarines are shown in table 5.2. For study purposes, a diving depth of 300 meters is taken (Ministerie van Defensie). The calculation for the power is followed from Van Buren (2016).

Not only the main characteristics of the submarine are needed to make a comparison. Some of the solutions determined in chapter 6 include using air to expel water from the tanks. The pressure in the tank used to expel the water from the tanks, strongly depends on the volume of the tank. Therefore there is a need for a indication of the volumes of the tank. These volumes are determined according the design parameters of Kormilitsin (2001). The trim tanks usually have a volume of 1% of the total displaced volume of the submarine each. The main compensation tank has a volume around 2% of the total displaced volume. For the hovering tank there do not consist parameters for the volume. The volume of the hover tank is taken at 1/3 of the

[1] 3D modelling program

Table 5.2: Characteristics of used submarines

Characteristics	**Symbol**	**Order**	**Submarine 1**	**Submarine 2**
Displacement	Δ	[ton]	4000	2000
Scaling factor	α	[-]	1.000	0.794
Length	L	[m]	70.20	55.72
Width	B	[m]	9.60	7.62
Height hull	H_{hull}	[m]	10.60	8.41
Length sail	L_{sail}	[m]	10.95	8.69
Height sail	H_{sail}	[m]	5.60	4.44
Wetted area	A_w	[m^2]	2170	1367
Centre of buoyancy	LCB	[m]	37.85	30.04
Centre of buoyancy forship	LCB_{for}	[m]	51.39	40.79
Centre of buoyancy aftship	LCB_{aft}	[m]	23.73	18.83
Periscope depth	PD	[m]	21.20	17.86
Studied depth	D	[m]	300	300
Speed slow	v	[m/s]	10	10
Speed transit	v	[m/s]	20	20
Power slow	P_b	[kW]	880	622
Power transit	P_b	[kW]	3520	2489

volume of the compensation tank. Later it will be shown that this results in enough volume to compensate. The position of the tanks is also important since that determines the trimming moment caused by the fluid volume of the tanks. To be able to make an estimation of the position of the tanks, the Rhino model is used. In this model the origin of the system of axes is shifted the aft of the submarine. This is a normal convention in shipbuilding. In this Rhino model the tanks are implemented. The compensation tanks are positioned on the longitudinal centre of buoyancy, LCB. This way compensation does not result in a trimming moment. The same is valid for the hovering tanks. The Aft trim tank is positioned as far as possible at the aft of the submarine. The forward trim tank is positioned as far as possible in the front of the submarine. Since some of the solutions consist of a piping system with an actuator in between, the length of the pipes also have to be determined. Usually a submarine can be divided into three parts. From the aft is the engine room, close to the propeller. In the mid of the submarine are the control room and the living quarters. In the front of the submarine is the torpedo room. These three parts of the submarine are assumed to be of equal length. This gives that the engine room is positioned in the first third of the submarine, and the pumps are chosen to be positioned in the middle of the engine room. The resulting values for the two used submarines can be found in table 5.3. An overview of the tanks in the Rhino model can be found in appendix B.

In the design stage it can be assumed that the submarine is designed in such a way that there will be no initial trim (Kormilitsin, 2001). Initial trim is assumed to be compensated by fixed ballast inside the submarine. All items should be listed throughout the design process of a submarine. It is assumed that the submarine is in equilibrium when all the ballast trim tanks are half filled.

Table 5.3: Main components position and size

Characteristics	Symbol	Order	Submarine 1	Submarine 2
Position aft trim tank	x_{ATT}	[m]	12.88	10.22
Position for trim tank	x_{FTT}	[m]	63.97	50.77
Posistion compensation tank	x_{CT}	[m]	38.00	30.16
Postion hover tank	x_{HT}	[m]	40.69	32.30
Postion variable buoyancy	x_{VB}	[m]	43.63	34.63
Position pump	x_{pump}	[m]	17.55	13.93
Volume trim tank	V_{TT}	[m^3]	39.02	19.51
Volume compensation tank	V_{CT}	[m^3]	78.05	39.02
Volume hover tank	V_{HT}	[m^3]	26.02	13.01

Table 5.4: Record of all weights on board of the submarine

Item	x-position [m]	weight [ton]	rate of change [ton/s]
...	...	...	...
...	...	...	...
...	...	...	...

Weights inside the submarine are usually tracked in a static way, only their position and their weight. For the ballast trim system it is also useful to consider the rate of change. Some components on board will shift position or even be pumped out of the submarine. An example of how to keep track of the weights on board can be found in table 5.4.

5.2. General disturbances of a SSK submarine

A SSK submarine will always encounter disturbances during its voyage. General design parameters are taken into account. This section will investigate the general disturbances that a submarine can encounter.

The trim polygon of a submarine shows if the submarine is able to maintain neutral buoyancy during its voyage. Two situations are considered, when the submarine is at full load and when the submarine is empty. These two conditions describe when the submarine is departing and when the submarine is arriving or re-supplied. Within the trim polygon the effect of the density of water can also be found. Since seawater can differ in salinity, letting in ballast water gives a different weight over time. An example of a trim polygon can be found in figure 5.2. When folowing the ballast trim system clockwise, there can be seen which tanks have to be filled or emptied to obtain equilibrium. The trim polygon gives the capability of the submarine to maintain trim. It should be noted that the trim polygon does not influence the ballast trim system, but it does influence the size of the tanks. The blue window gives the moment that can be achieved with filling and emptying the specific tanks. The red line gives the departure condition. This is when all storages and consumption tanks are filled. The upper and lower dot give the condition for the difference in water density. The yellow line gives the arrival condition. In this case all the storages and consumption tanks are empty. By plotting these extreme values for the submarine weight condition in the trim polygon, there can be checked if the condition can be compensated by the trim and compensation tanks.

The size of the internal disturbances due to a change of the consumables does strongly depend on the design of the submarine. The position of storage mainly determines the trimming moment. This makes it hard to estimate the disturbances in an early design stage. Having the storages close to the centre of buoyancy results in smaller trimming moments and therefore smaller trimming tanks. Space is valuable in a submarine,

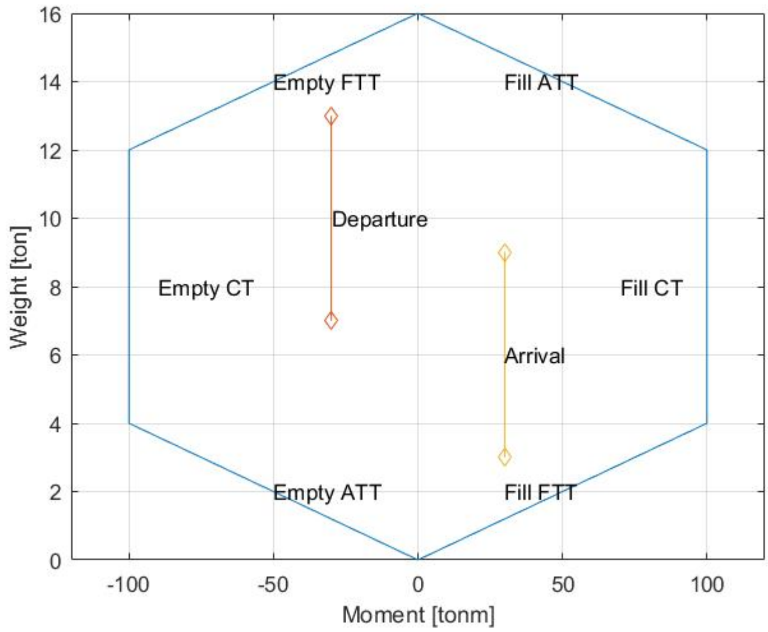


Figure 5.2: Trim polygon

and having large trimming tanks that are not used in an effective manner is therefore a waste of this valuable space.

5.2.1. People

Submariners will walk around the submarine. The effect on trim of one person walking from the front to the aft is negligible. Multiple people walking around the submarine however, is not. The number of people walking around the submarine is dependent on the total number of crew. Figure 5.3 shows a relation between the displacement of the submarine and the number of crew on board.

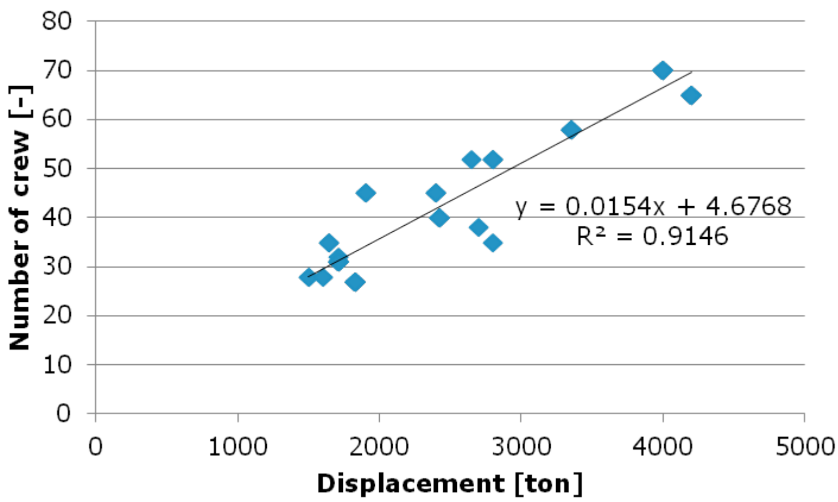


Figure 5.3: Displacement vs submariners on board

When knowing the number of submariners on board an estimation is made of the number of submariners walking at the same time. Assumed is that 10% of the total crew is walking around at the same time. This will give a disturbances in the longitudinal trim. The other assumption is that these people will walk in the same direction with a speed of 5 km/h. They will walk for half of the length of the submarine and will then return back. The average weight of the submariner is taken at 80 kg. These assumptions will lead to equation 5.2 and 5.3.

$$M_{walking,max} = \frac{0.1 * N_{submariners} * 80 * g}{\Delta_{submarine}} * 0.5 L_{submarine} \tag{5.2}$$

$$\dot{M}_{walking} = \frac{0.1 * N_{submariners} * 80 * g}{\Delta_{submarine}} v_{walking} \tag{5.3}$$

With: M is the moment in Nm; $N_s ubmarines$ is the number of submariners on board; Δ is the ships' displacement in kg; L is the length in meters.

5.2.2. Fuel

The fuel that the RNLN uses is F76. In case of a double hull submarine the fuel tanks are connected to the sea. This way the fuel transported to the internal tanks, fuel consumption tanks, is directly replaced by seawater. Seawater has a higher density than fuel. Therefore the fuel will float in the top of the fuel tanks. Because of the difference in density between the two fluids, taking out fuel of the fuel tanks results in a heavier submarine. The resulting force depends on the density difference between the fuel and the seawater. This can be seen in equation 5.4 and 5.5. This force can be based on the produced power and the sfc of the submarines' engine. This will lead to equation 5.7.

$$m_{added} = \frac{\rho_{seawater} - \rho_{fuel}}{\rho_{fuel}} * m_{fuel} \tag{5.4}$$

$$\dot{m}_{added} = \frac{\rho_{seawater} - \rho_{fuel}}{\rho_{fuel}} * \dot{m}_{fuel} \tag{5.5}$$

$$\dot{F}_{fuel} = \frac{\rho_{seawater} - \rho_{fuel}}{\rho_{fuel}} * \frac{sfc}{3.6 * 10^6} * g * P(t) \tag{5.6}$$

With: m is the mass in kg; ρ is the density in kg/m^3; F is the force in N; sfc is the specific fuel consumption in g/kWh; P is the power in kW.

Fuel is only burned if a engine is running. In conventional designs the engines were only running when a part of the energy is used for charging the batteries and the other part is used for the sailing and hotel load. The main parameters that influence the brake power of the engine are:

- resistance during snorkelling;
- velocity of the submarine;
- hotel load;
- battery load;
- specific fuel consumption.

By assuming the engine(s) running at full power when the submarine is snorkelling, the biggest disturbance is found for the compensation of the fuel. Snorkelling is done while sailing. When trim due to fuel usage occurs, the trim can be compensated with the hydroplanes. Usually the fuel tanks are situated along the length of the submarine. When choosing wisely which fuel tank to use, trim will not occur. The sfc of a diesel engine is dependent on the kind of diesel engine (Stapersma, 2010). With the SFC, the total fuel consumption can be found when knowing the total used power.

For a submarine a diesel generator is needed. For generator purpose on board submarines a medium speed engine can be used. Those engines have a sfc between 170 and 195 g/kWh. A submarine's exhaust pipe is under the waterline, which results in a back pressure. Because of this back pressure turbo-charger can not be used optimally. Therefore the specific fuel consumption increases. MTU has special submarine engines (Von Drathen, 2014). The specific fuel consumption of those engines is around 225 g/kWh when snorkelling.

The average density of the fuel used nowadays, F76, is 847.4kg/m^3. Currently the RNLN is interested in alternative fuels. One of the options is the use of hydrotreated vegetable oil, HVO, which reduces the toxic exhaust gasses. This HVO can be mixed with the F76 with up to 50% HVO. Mixing these two oils will lead to a change in density of the fuel. These 50% mixed fuels have a density of 815.1 kg/m^3 (Boumeester, 2017). For future SSK submarines this fuel density will be used because of the density change of the fuel, the disturbance due to pumping out fuel of the fuel tanks becomes larger. It can be expected that a future SSK submarine must also be operated on HVO because of NATO guidelines.

The oil used by the submarine because of the diesels running is assumed to be very low compared to the fuel consumption and is therefore not taken into account.

$$\dot{F}_{fuel} = \frac{\rho_{seawater} - \rho_{fuel}}{\rho_{fuel}} * \frac{225}{3.6 * 10^6} * g * P(t) \tag{5.7}$$

With: F is the force in N; ρ is the density in kg/m^3; P is the engine power in kW.

Modern submarines can also be equipped with an air independent propulsion, AIP, installation. This way the endurance underwater is extended with a constant battery size. For AIP three different systems can be employed (Van Buren, 2016):

- Closed cycle diesel;
- Stirling engine;
- Fuel cell.

For an AIP installation, 6 knots is seen as the maximum speed. The higher the speed, the higher the consumption of fuel and air. The specific consumption of AIP systems is not the same for all types. It is seen that fuel cell systems have a higher efficiency at a low load (Van Buren, 2016). The Stirling engine and the closed cycle diesel obtain a better efficiency with increasing load until 100%. An AIP system needs fuel and liquefied oxygen to make energy. Therefore the consumption of both of these components are important for the ballast trim system. Four different known systems are studied (Van Buren, 2016):

- PEMFC + meH_2, fuel cell with hydrogen from metal hydrides and LOX;
- PEMFC + MeOH, fuel cell with hydrogen from methanol and LOX;
- Stirling, which works on the expansion of gas by heating. Gas is heated by diesel fuel and LOX;
- Closed cycle diesel, a closed system which works on LOX and diesel fuel.

It is assumed that all exhaust gasses will be expelled from the submarine. That said, it is possible to store them within the submarine in the case of using hydrogen as fuel, which will result in no change of total weight of the submarine. (Van Buren, 2016).

Table 5.5: Specific consumptions AIP systems

	sfc [g/kWh]	**soc [g/kWh]**	**total [g/kWh]**
PEMFC + meH	65	519	584
PEMFC + meOH	434	652	1086
Stirling	216	823	1039
Closed cycle diesel	289	971	1260

The use of an AIP installation on board is very interesting for the ballast trim system. Without AIP energy could be taken from the battery having (almost) no effect on the equilibrium of the submarine. By using AIP like the Stirling motor, fuel and gasses are used and expelled in order to drive the engine. This results in a change of internal weight in the submarine. The AIP system considered is the Stirling motor. This system is already in use in the Swedish submarines of Saab Kockums. The Stirling motor uses fossil fuels and LOX. The fossil fuels are stored outside the pressure hull and are replaced by seawater when used. Equation 5.7 can be used to determine the disturbances, but the sfc should be changed. This results in equation 5.8. The LOX is stored in closed tanks. Therefore using LOX is directly related to a change of weight of the submarine and should be compensated by the compensation system.

$$\dot{F}_{fuel} = \frac{\rho_{seawater} - \rho_{fuel}}{\rho_{fuel}} * \frac{216}{3.6 * 10^6} * g * P(t) \tag{5.8}$$

$$\dot{F}_{stirling} = \frac{1.732}{3.6 * 10^6} * g * P(t) \tag{5.9}$$

With: F is the force in N; ρ is the density in kg/m^3; P is the engine power in kW.

5.2.3. Waste water tanks

Within the submarine there are several tanks that can be filled or emptied. Waste water is produced on board and needs to be retained, at least temporarily, since it is in some regions prohibited to discharge (Kormilitsin, 2001). On average a person produces 60 litres/day of waste water (following the design parameters). The total amount of waste water is not important but the rate of production and the position of the tanks is. The assumption is made that the distance between the fresh water tanks and the waste water tanks is very low, $\frac{1}{12} L_{submarine}$. This way they are stretched under the living quarters. It can be assumed that the biggest use of fresh water occurs during the morning rush. The assumption is made that $\frac{1}{3}$ of the water is used in the morning in a time span of an hour. This leads to the equations 5.10 and 5.11.

$$M_{wastewater,max} = 20 * g * N_{submariners} * \frac{1}{12} * L_{submarine} \tag{5.10}$$

$$\dot{M}_{wastewater} = \frac{20 * g * N_{submariners}}{60} * \frac{1}{12} * L_{submarine} \tag{5.11}$$

With: M is the moment in Nm; $N_{submariners}$ is the number of submariners on board; L is the length in m.

It is seen in the design of the Astute class that the submarine empties its sanitary water and bilge water in the ballast trim system, see figure 3.5. By doing so, there will be no change of weight of the submarine. Oil from the bilge water can pumped into the fuel tanks. This way the oil can be used by the diesel engines when snorkelling. This results that emptying these tanks does not influence the ballast trim system. Pumping out the water of the tanks is done by the compensation pump and there is no need for minimum speed of this. Waste water can contain large particles which, in case of using the compensation pump, the pump must be able to handle.

5.2.4. Masts

A periscope will work as an extra floating body. When the periscope is forced upwards in the direction of the free surface, it will increase the total volume of the submarine. By increasing the total volume of the submarine, the submarine will become more buoyant, as seen in chapter 2. Usually a submarine rises its periscope only for a few seconds because when the periscope is above the waterline, it can be spotted.

The periscope is modelled as a cylindrical body. Periscope depth is around 15 meters measured from the keel of the submarine (Vego, 2015). Periscope depth, PD, is taken at 21.6 meters measured from the keel for submarine 1 and 17.45 meters for submarine 2. This is based on the assumption of having a periscope with an effective length of 5 meters. Equation 5.12 shows the force due to raising the periscope.

$$F_{periscope} = (T_{PD} - H_{submarine}) \frac{1}{4} D^2_{periscope} \pi \rho_{sea} g \tag{5.12}$$

With: F is the force due to the periscope in N; T_{PD} is the depth of the submarine at periscope depth measured from the keel in m; $H_{submarine}$ is the total height of the submarine in m; D is the diameter in m; ρ is the density in kg/m^3.

As well as the size of the periscope, the raising, looking and lowering speed is important. With these times and forces the profile of the disturbances over time is determined. This profile is displayed in figure 5.4.

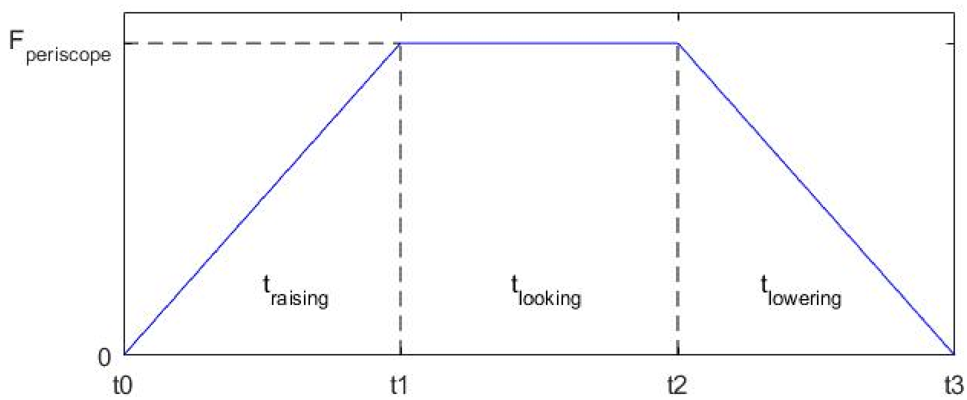


Figure 5.4: Profile of Buoyancy forces caused by the periscope

$$\dot{F}_{periscope} = \begin{cases} \dfrac{F_{periscope}}{t_{raising}} & t_0 < t \leq t_1 \\ 0 & t_1 < t \leq t_2 \\ \dfrac{F_{periscope}}{t_{lowering}} & t_2 < t \leq t_3 \end{cases} \tag{5.13}$$

The times for rising, looking and lowering are $t_1 = 7.5$, $t_2 = 12.5$, $t_3 = 7.5$. These times do comply with real data. The diameter of the periscope is taken as 500mm. This results in equation 5.14. The raising length of a periscope is taken as 5 meter. This gives a raising and lowering speed of 0.67 m/s. Equation 5.15 gives the force by the periscope per second.

$$F_{periscope,max} = (5) * \frac{D_{periscope}\pi}{4} \rho_{seawater} g \tag{5.14}$$

$$\dot{F}_{periscope} = (T - H_{submarine}) * \frac{D_{periscope}\pi}{4} v_{raising} \rho_{seawater} g \tag{5.15}$$

With: F is the force in N; t is the time in s.

Another periscope that is on board of the submarine is the attack periscope. The attack periscope has smaller diameter and has therefore less influence on the buoyancy of the submarine. These two periscopes will not be used at the same time. Other masts on the submarine are assumed the same size as the periscope. These masts are:

- Electronic warfare, EOV, mast
- Radar mast
- Radio mast
- Satelite communication, Satcom, mast

Last but not least is the Snorkel mast. The snorkel on a SSK submarine is needed to supply fresh air for the diesel engine as well as refresh the air on board of the submarine. By raising or lowering the snorkel, the buoyancy of the submarine will change as seen before. The forces will be described the same as the periscope forces, see equation 5.14. The speed for lowering and raising the mast is taken the same as for the periscope. The diameter of the snorkel is assumed to have twice the diameter of the periscope. The length of the snorkel is taken as 5 meters. Equation 5.16 and 5.17 describes the force caused by the snorkel.

$$F_{snorkel,max} = (5) * \frac{D_{snorkel}\pi}{4} \rho_{seawater} g \tag{5.16}$$

$$\dot{F}_{snorkel} = (T - H_{submarine}) * \frac{D_{snorkel}\pi}{4} v_{raising} \rho_{seawater} g \tag{5.17}$$

With: F is the force in N; ρ is the density in kg/m^3; T is the depth measured from the keel in m; $H_{submarine}$ is the total height of the submarine in m.

5.3. Sea

The submarine is subjected to sea variations and disturbances in terms of waves and density. The general requirements give to what extent the system must be operational, including the maximum sea state and expected range of density. The influence of the sea state and the density of seawater will be explained in this section.

5.3.1. Sea state

Waves are different all over the world, but their patterns are in some way predictable. This is given by the wave scatter of a certain area. The wave scatter gives the probability of having certain waves. The North Atlantic is one of the most heavy seas, known for the high occurrence of high waves. For the highest sea states it can be said that the submarine is not able to completely fulfil all its tasks. For example, if the requirement is that the ship must be able to operate in sea state 6, it will be able to operate 97% of the time in the North Atlantic. Table 5.6 shows the distribution of the expected waves in the North Atlantic from which the probability of a certain seastate can be determined (Gerritsma, 2004).

Table 5.6: North Atlantic wave scatter

		Wave period [s]										
Sea state	**Wave height [m]**	**2.5**	**6.5**	**8.5**	**10.5**	**12.5**	**14.5**	**16.5**	**18.5**	**20.5**	**>21.0**	**total**
0-1	**<0.5**	5.19	0.39	0.14	0.08	0.03	0.01	0.01	0.01	0.06	0.15	*6.07*
3	**0.50 - 1.25**	13.93	5.41	1.25	0.43	0.15	0.05	0.02	0.02	0.03	0.27	*21.56*
4	**1.25 - 2.50**	7.43	18.12	10.44	3.45	1.03	0.33	0.10	0.04	0.02	0.03	*40.99*
5	**2.50 - 4.00**	0.79	5.00	7.60	4.92	2.01	0.66	0.19	0.06	0.01	0.01	*21.25*
6	**4.00 - 6.00**	0.13	0.92	2.07	2.00	1.16	0.48	0.18	0.05	0.01	0.01	*7.01*
7	**6.00 - 9.00**	0.04	0.25	0.64	0.79	0.56	0.26	0.12	0.02	0.00	0.01	*2.69*
8	**9.00 - 14.00**	0.00	0.02	0.07	0.12	0.10	0.06	0.03	0.02	0.01	0.00	*0.43*
9	**>14.00**	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	*0.00*
	Total	*27.51*	*30.11*	*22.21*	*11.79*	*5.04*	*1.85*	*0.65*	*0.22*	*0.14*	*0.48*	*100*

A submarine is a high-end weapon platform. Missions have to be executed as readily as possible. Sea state 5 is taken as maximum operational sea state. This means that for the North Atlantic, the submarines ballast trim system is able to be fully operational 90% of the time. In other parts of the world the submarine can operate even more often since the North Atlantic is known as one of the roughest waters. In case of a higher sea state the submarine will not fail, but will be less operational on the surface as well as underwater.

5.3.2. First order effects

An ocean wave is modelled as a propagating cosine wave. This cosine wave gives disturbances to the submarine which are called the first order wave effects. The wave period is an important parameter within waves. From this the radial speed of the wave can be determined, see equation 5.18. This radial speed is later on necessary to find the force of the wave on the submarine.

$$\omega = \frac{2\pi}{T} \tag{5.18}$$

With: ω is the radial speed of the wave in rad/s; T is the wave period in s.

Deep water is assumed which makes it possible to define all the needed parameters to describe the waves. For deep water the linear wave theory can be used and the wave number can be determined according to 5.19. The relation between the wavenumber and and wave length can also be determined with equation 5.20. If k according to 5.19 is filled in equation 5.20, a relation between the wave length and the radial speed of the wave is obtained, 5.21.

$$\omega^2 = kg \tag{5.19}$$

$$\lambda = \frac{2\pi}{k} \tag{5.20}$$

$$\lambda = \frac{2\pi g}{\omega^2} \tag{5.21}$$

With: ω is the radial speed in rad/s; k is the wave number in 1/m; λ is the wave length in m.

The main parameters of the waves in sea state 5 can be determined. These parameters are used to determine the disturbances caused by waves. In table 5.7 this data is listed.

Table 5.7: Wave data sea state 5

T [s]	2.5	6.5	10.5	12.5	14.5	16.5	18.5	20.5
ω [rad/s]	2.51	0.97	0.60	0.50	0.43	0.38	0.34	0.31
k [1/m]	0.6439	0.0952	0.0365	0.0258	0.0191	0.0148	0.0118	0.0096
λ [m]	9.76	65.97	172.13	243.95	328.27	425.07	534.36	656.14

With this wave data, disturbances due to the waves can be determined. For sea state 5 different waves are considered. The equation for a wave can be found in equation 5.22. This equation describes the wave pattern at the free surface.

$$\zeta(t) = \zeta_a cos(kx - \omega t) \tag{5.22}$$

With: ζ is the wave height in m; ζ_a is the wave amplitude in m.

In case of long waves and deep water, the pressure can be described according to equation 5.23 (Gerritsma, 2004). This equation already takes into account non linear effects (Gerritsma, 2004). With this equation the pressure at every point of the submarine is calculated. The first two terms in equation 5.22 are for a single wave constant. It is assumed that the submarine is balanced for this static forces due to single waves. Therefore only the last term of equation 5.23 is needed to study the first order wave effects. The forces due to the first order effects of the waves are studied by assuming the submarine cylindrical. This cylinder has a diameter which is equal to the width of the submarine and a length that is equal to the length of the submarine. The forces due to the first order effect can be calculated with equation 5.24.

$$p = -\rho g z + \frac{1}{2}\rho\zeta_a^2\omega^2 e^{2kz} + \rho g \zeta_a e^{kz} cos(kx - \omega t) \tag{5.23}$$

$$\int_{x=0}^{L}\int_{y=-0.5D}^{0.5D} \rho g \zeta_a e^{k(\sqrt{(D/2)^2 - y^2} + z_0)} cos(kx - \omega t) \tag{5.24}$$

With: ρ is the density in kg/m^3; z is the depth in m measured from neutral axis; ζ_a is the wave amplitude in m; ω is the radial wave velocity in rad/s; k is the wave number in 1/m; x is the position in x-direction in m; y is the position in y-direction in m; D is the diameter in m; z_0 is the depth at the centre of the cylinder.

With the pressure fluctuation known at all points of the submarine, the force of the first order effects can be determined. These first order effects are determined for the submarine 1 and 2. The forces due to the different waves for submarine 1 and submarine 2 can be found in figure 5.5 and 5.6. The maximum amplitudes of these forces for the different waves can be found in figure 5.7 and 5.8.

It is known that submarines do not compensate for the first order wave effects. Simulation further on will investigate the effects of the wave on the submarine. It could be possible that when hovering close to the surface there is a need to compensate for the first order effects of waves. If not, simulation of these waves on the submarine can show this.

5.3.3. Second order effect

Second order effects can be felt when travelling close to the surface. Literature gives different definitions of the second order effect. One is a suction force due to the geometry of the submarine (Renilson, 2015) . The suction force is caused by the difference in the velocity of the water above and under the submarine. The water above the submarine will flow faster than under the submarine, 5.9. This results in a lower pressure at the top of the submarine which causes the submarine to surface. Extra ballast is needed to make the

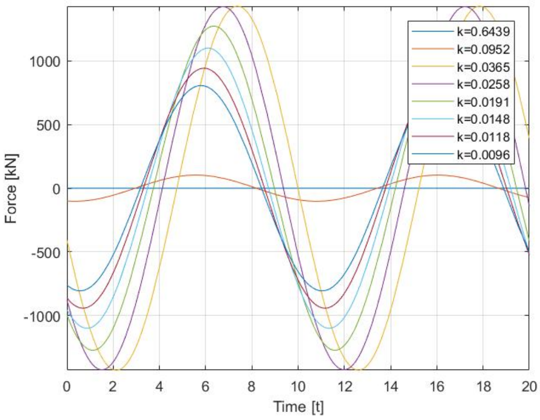


Figure 5.5: 1^{st} order wave force submarine 1

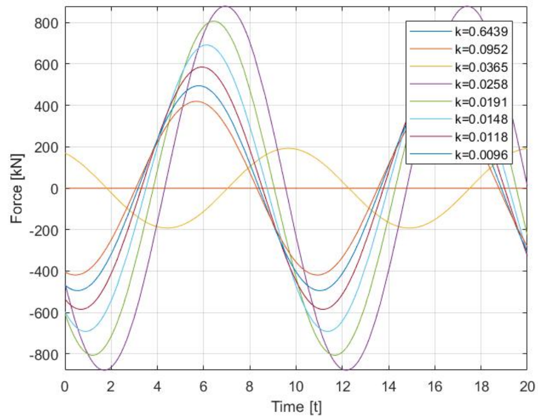


Figure 5.6: 1^{st} order wave force submarine 2

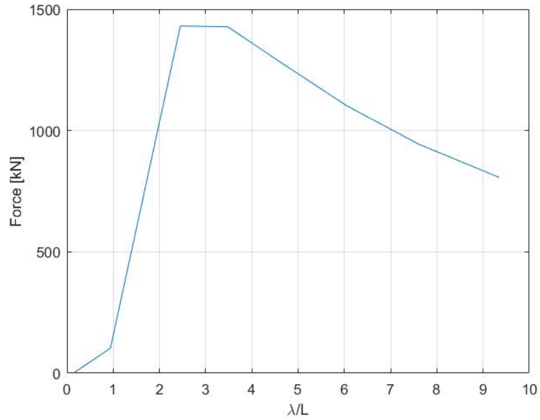


Figure 5.7: Maximum 1^{st} order wave force submarine 1

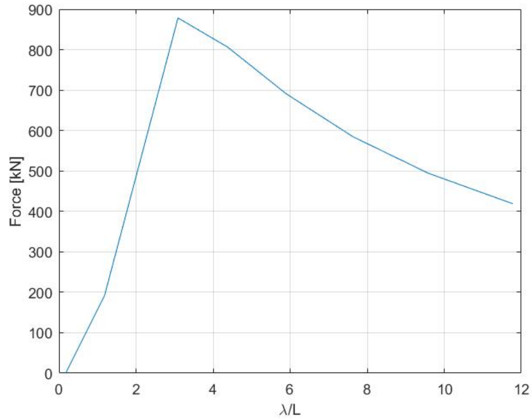


Figure 5.8: Maximum 1^{st} order wave force submarine 2

submarine more heavy and compensate for this second order effect. This should be done carefully to avoid uncontrolled descending.

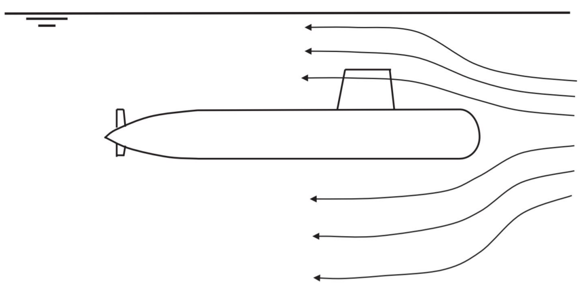


Figure 5.9: 2^{nd} order effect in calm water due to flow

Another definition is that open water waves are a superposition of multiple ideal waves, equation 5.22. Therefore the second term of equation 5.23 changes over time (Renilson, 2015). The second order effect close to the surface is also referred to as a suction force. Due to the superposition of the different waves, a low frequency force occurs. This force is moving the submarine in a absolute way. Earlier for the first order effects it is concluded that the submarine is following the oscillating water surface. The second order effects cause a movement with respect to the water surface.

When having these two definitions it can be concluded that the second order effects in waves are different than those in calm water. The second order wave force depends on the speed of the submarine, the sea state, the depth and the heading of the waves. These forces can become very large. Some even say that for a 10.000 ton submarine at a depth of 50 meter it is possible that there is a need of 30 ton extra ballast to stop the submarine from being surfaced (Renilson, 2015).

The Forces of a single wave are already determined. However, the sea consists of multiple small waves added together. For two waves with the same wave height the elevation of the surface can be determined by equation 5.25. Equation 5.23, the second term in the equation takes into account the second order effects. In order to determine the second order effects due to the superposition of waves, it is important to know were this equation originated. This second order term is related to the speed of the water particulars at the free surface. For regular waves this speed at the free surface is determined by equation 5.26 and 5.27. The second order pressure can be determined by equation 5.28.

$$\zeta_t = 2 * \zeta_a \cos\left(\frac{(k_1 - k_2)x}{2} - \frac{(\omega_1 - \omega_2)t}{2}\right) * \cos\left(\frac{(k_1 + k_2)x}{2} - \frac{(\omega_1 + \omega_2)t}{2}\right) \tag{5.25}$$

$$u = \omega e^{kz} \zeta_a \cos(kx - \omega t) \tag{5.26}$$

$$w = \omega e^{kz} \zeta_a \sin(kx - \omega t) \tag{5.27}$$

$$p_2 = \frac{1}{2}(u^2 + w^2) \tag{5.28}$$

For two waves with the same height it is seen that the wave elevation changes. Equation 5.25 can be split into two waves. In equation 5.25, the first cosine describes the group wave. The second cosine factor describes the wind waves. This results in equation 5.29. The speed of the water particulars can be described by equation 5.30 and equation 5.31. When these new relations for the speed of the waves at time t is used to determine the second order wave force, equation 5.32 is obtained. Here the term $\cos^2(k_g x - \omega_g t)$ is found. This function describes the fluctuation of the suction force over time.

$$\zeta_t = 2 * \zeta_a \cos(k_g x - \omega_g t) * \cos(k_w x - \omega_w t) \tag{5.29}$$

$$u = \omega e^{kz} \zeta_a \cos(k_g x - \omega_g t) * \cos(k_w x - \omega_w t) \tag{5.30}$$

$$w = \omega e^{kz} \zeta_a \cos(k_g x - \omega_g t) * \sin(k_w x - \omega_w t) \tag{5.31}$$

$$p_2 = \frac{1}{2}\omega^2 e^{2kz} 2 * \zeta_a \cos^2(k_g x - \omega_g t) \tag{5.32}$$

With: ζ_t is the wave height at time t in m; ζ_a is the wave amplitude in m; k is the wave number; x is the position in m; ω is the radial velocity of the wave in rad/s; t is the time; u is the speed of a water particle in x-direction in m/s; w is the speed of a water particle in z-direction in m/s; p_2 is the second order wave pressure in N/m^2; z is the depth in m.

The best way to find the second order effects is by model tests. This is done for three test cases. In these test cases the radial velocity of the waves is changed. It is seen that the fluctuation of the suction force due to the waves is described by $cos^2(\omega_g t)$ in equation 5.32. That knowledge is used to find the second order suction force from the measurement data. The total force due to bichromatic waves can then be described by equation 5.33. In here the first order forces due to the wind waves and the second order forces due to the group wave are split from each other. With the output of these tests, the group wave is determined. After that the maximum second order force is determined, this is the amplitude of the fluctuating force. This is done by equation 5.34. This equation is based on the knowledge that the first order forces are characterised by the product of 2 cosines functions. The amplitude of the second order forces can be determined by equation 5.34. The characteristics of the waves and their forces can be found in table 5.8. This data is based upon bichromatic waves and submarine 2 having a speed of 4 knots.

$$F_t = F_{2,a} \cos^2(\omega_g t) + F_{1,a} \cos(\omega_w t) \tag{5.33}$$

Table 5.8: Bichromatic wave data

	$\omega_{\mathbf{w,1}}$ [**rad/s**]	$\omega_{\mathbf{w,2}}$ [**rad/s**]	$\omega_{\mathbf{group}}$ [**rad/s**]	$\mathbf{F_2, a}$ [**kN**]
Test 1	0.5	0.525	0.0152	-5
Test 2	0.7	0.75	0.0327	-15
Test 3	0.95	1	0.0355	-8.5

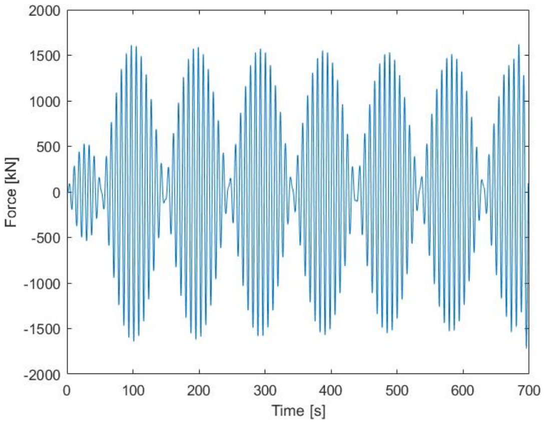


Figure 5.10: Total wave force test 2 submarine 1

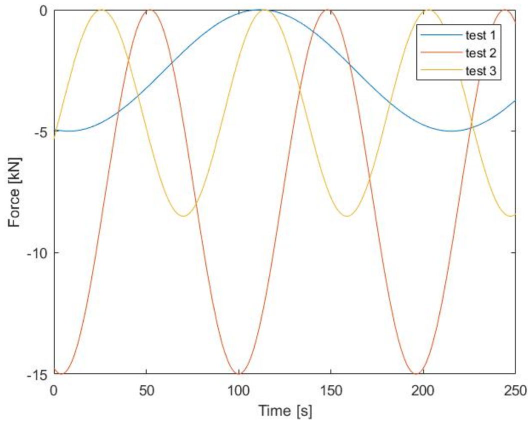


Figure 5.11: Second order wave force submarine 1

$$F_{2,a} = \frac{F_{max}^{tot} + F_{min}^{tot}}{2} \tag{5.34}$$

With: F_t is the force at time is t in N; $F_{2,a}$ is the amplitude of the force for the second order term in N; $F_{1,a}$ is the amplitude of the force for the first order term in N; ω_g is the radial velocity of the group wave in rad/s; ω_w is the radial velocity of the wind waves in rad/s;

With these parameters it is possible to sketch the profile of the second order disturbances. These second order forces are described by equation 5.35. The maximum second order force is found in test 2. The total force during this test can be found in figure 5.11. All the second order forces for the different tests can be found in figure 5.11.

$$F_{2,t} = F_{2,a} * \cos^2(\omega_g t) \tag{5.35}$$

The forces scale according Froude. The forces were measured for submarine 1. The forces scale with α^3. This implies that the second order wave forces for submarine 2 are lower by a factor of 0.5. The forces for submarine 2 can be found in table 5.8. For the simulations the largest disturbance, test 2, is used to see how the submarine behaves due to the second order effects of waves.

5.3.4. Density

The seawater density can change during a voyage since seawater density differs all over the world. When the density of seawater changes, the buoyancy force changes as seen in chapter 2. The change of density can occur quickly. In seawater there are pycnoclines, a density front in the sea (NDRC, 1946). When sailing through a pycnocline the density changes instantly. The North Sea for example is more saline than the Baltic Sea. Near the coast of Schagen in Denmark these two seas are connected to each other and merge. This difference of density is even visible in figure 5.12. At such points the salinity changes quickly and therefore also the density of the seawater. Normally salinity is measured in parts per thousand. A change of salinity of 1‰ results in a density change of 0.78 kg/m3.

Not only is the salinity of seawater important in determining the density of seawater, but also the temperature. The temperature of seawater is usually warmer on the surface than it is a few meters below because of

Figure 5.12: Mixture zone of the Baltic Sea and the North Sea, citep

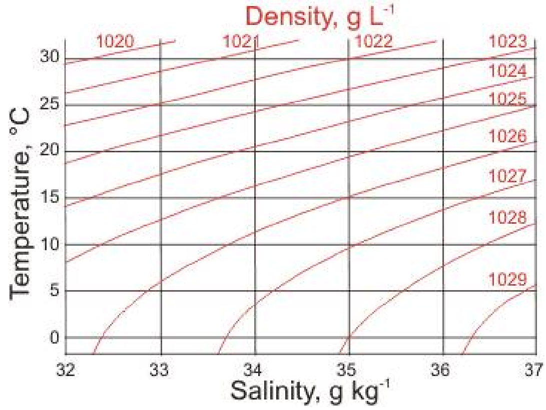


Figure 5.13: Density of seawater of 35 versus the temperature

the heating of the sun. The density as a function of the temperature of seawater with a salinity of 35‰ can be found in figure 5.13

Combinations of difference in temperature and salinity can compound the effect. It can be expected that a stream that passes has a different temperature as well as a different salinity. This together will create the difference in density and therefore the difference in buoyancy 2.1. For submarine 1, when the density of the seawater changes from 1025 kg/m^3 to 1026 kg/m^3, there is a need for 4 ton extra ballast.

Pycnoclines are mainly found horizontally but can also be found vertically. Therefore when hovering, due to currents there can be a sudden change of density. Earlier research uses 2 kg/m^3 of density change for a density front. This means that for example the density at 199 meters depth is 1025 kg/m^3 and at 201 meters depth 1027 kg/m3. It is expected that every 200 meters such a density front is present (Mansfield and Venn, 2011). Since the pycnoclines can also be vertical, this change of density is also taken into account when the submarine is keeping depth.

5.4. Mission related

Disturbances in the equilibrium of the submarine can in many cases relate to a certain mission or requirement of the submarine. This chapter will elaborate on the requirements for a modern SSK submarine and the missions a modern SSK submarine must execute. These have implications on the ballast trim system which will be explained in this chapter.

5.4.1. Littoral waters

A modern submarine is expected to operate more in littoral waters than earlier classes of submarines (De Boer, 2015). Littoral waters are defined as waters close to the shore (Vego, 2015). In the littoral zone the shore starts. Coming from open sea, the water becomes more shallow. In practise the littoral zone mostly refers to the zone stretching from the edge of the continental plate towards the end of dunes. In a military sense the littoral zone is typically the zone in which a mission changes. Littoral warfare is often referred to as the landing of land forces. But also close to shore surveillance missions are executed in the littoral zone. An overview of the geographically defined littoral zone can be found in figure 5.14.

Lately navies are focussing more and more on littoral warfare. This includes all types of missions within the littoral zone. Operating in the littoral zone brings different implications, as discussed in the following subsections.

Density

In the littoral zone fresh water inlets are present. For example rivers and glaciers supply the sea with fresh water (NDRC, 1946). Mixture of fresh and salt water results in more pycnoclines. The possibility of sudden density changes therefore increases in littoral waters. The importance of being able to maintain equilibrium due to density changes therefore becomes bigger for submarines operating more often in the littoral zone.

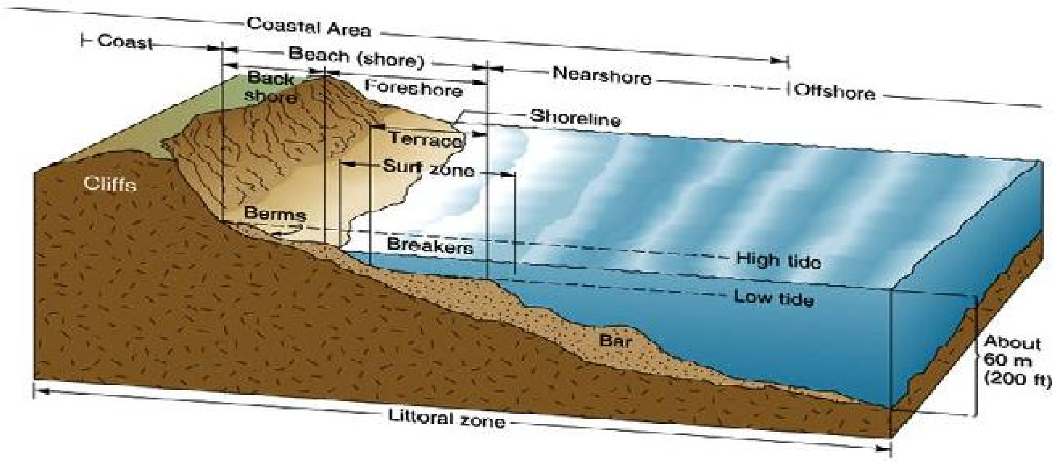


Figure 5.14: overview of the littoral zone

Mines
The littoral zone is close to shore and is generally shallow. This means that the littoral zone is ideal to place mines (Vego, 2015). Mines are a budget solution to avoid enemies from entering specific waters. The impact of mines is catastrophic. In the gulf war three American ships were hit by mines. The damage these mines caused was around 125 million dollars, while the costs of these mines was approximately 30,000 dollars (National Research Council, 2000). This underlines the cost effectiveness of sea mines. Different kinds of mines are available and can be classified in different ways. The most common way to differentiate mines is by detonation method. Three such detonation methods are:

- Influence
- Contact
- Controlled

Contact and controlled mines do not relate to the design of the ballast trim system. Therefore these mines are not important to take into account when considering the ballast trim system. The treat of influence mines, however, do have consequences for the ballast trim system.

A sea mine can be influenced in in three different ways (Bennet, 1998):

- magnetically
- by pressure
- acoustically

The acoustic signature of a submarine is particularly interesting regarding the ballast trim system. The noise production of the submarine must be low when operating in an area in which a threat of sea mines is present. Since sea mines can be triggered by pressure, hovering can be dangerous since pumping out and letting in water causes large pressure fluctuations. The best solutions against sea mines for submarines remains to avoid areas in which sea mines are resent. Next to this the equipment on board should be shock proof to a certain extend. If the treat of acoustic mines becomes high, a super silent mode of the submarine is desired. This means that the ballast trim system should be able to maintain depth while producing very little or no noise. Dealing with mines that detonate due to pressure, the submarine should slow down. Sailing slower means that fewer disturbances can be dealt with by using the hydroplanes. This way the operation depends more on the ballast trim system. Magnetic mines can react to electromagnetic signals. By shutting off engines the electromagnetic signature is reduced. This implies also slowing down the submarine. Therefore also in this case the hydroplanes can not be used in an optimal way.

A mine exploding close to the submarine causes disturbances in the trim of the submarine. If the submarine is intact after the blast, it is already compromised. Sailing away as fast as possible is important. This will reduce the consequences for the ballast trim system.

Exposure by air

The littoral zone is easy to control by aircraft. An aircraft is able to spot a submarine from the air. The most critical way for a submarine to be spotted by an aircraft is visually. The visibility of a submarine in water depends strongly on the transparency of the water. This is different for every sea. It is therefore hard to determine in this stage what the minimum depth is for a submarine to avoid detection by air. However, what is known is that it is important to maintain depth. For some operations it is desired to operate just under the water. If the submarine tends to surface during such an operation there is a risk of exposure. Maintaining the desired depth is therefore very important.

Air bubbles can tell an aircraft if something is in the water. One of the given solutions for ballast trim systems implies having pressurised tanks on board. If tanks needed to be vented, it is smart to vent them inside the pressure hull and let no air escape the submarine. This gives a pressure build up inside the pressure hull of the submarine which must be studied.

5.4.2. Increased special forces operations

A task of a modern SSK submarine is to embark and disembark special forces (Hennis-Plasschaert, 2016) (Binns, 2008). These special forces can be embarked and disembarked underwater. Especially with the threat of terrorism special operations become more important (Binns, 2008). There are several ways for these special forces to leave the submarine. Commonly used is the emergency escape hatch. Since the emergency escape hatch is already available, it is easy to use it as an exit route for the special forces. Missions with special forces are expected to increase in frequency, so special designed exit routes are found on modern submarines. Diver rooms can for example be found in the USS Virginia class submarines (Geelhoed-Bakiu, 2016). From these diver chambers nine fully-equipped special forces can be disembarked. Embarking and disembarking special forces with their gear gives a disturbance to the equilibrium of the submarine. But also other aspects play their role in these kind of special operations. Special forces, waiting outside the pressure hull until they can disembark, are exposed to the sea pressure. A quick change in pressure is not desirable for a human body. 18 ±3 meters water column/min is the guideline for the pressure change on a human body (DCIEM, 1992). Therefore it is important to keep the submarine at desired depth while executing a SF mission. Besides the pressure change, when swimming in and out of a submarine it is desired that the submarine is in total rest to avoid collisions underwater.

Embarking and disembarking special forces cause a change in weight. In a short time span weight is lost or gathered. The compensation system should take this into account. If the special forces leave the submarine it is most likely that they also take a lot of gear with them. However a good diver and diving equipment is buoyant. The special forces will tend to be in a neutral buoyancy. Therefore when the diver swims out of the submarine and his volume fills with water there will be no change of weight. The same is valid for the equipment the divers take with them.

There exists different systems that allow embarking and disembarking of special forces. These systems are:

- Escape tower;
- Horizontal escape;
- Cofferdam;
- Dry dock shelter.

Some of these systems work with ballast water which is already inside the submarine. The cofferdam is the most sophisticated solution for special forces operations. An overview of this cofferdam can be found in appendix C together with the other solutions. This so-called cofferdam is a built-in feature with which multiple divers can leave the submarine safely (Geelhoed-Bakiu, 2016). This cofferdam uses ballast water which is already in the submarine to fill the cofferdam with water. By using the ballast water from the internal tanks there will be no weight change due to the intake of water. The escape chamber is then pressurised with valves connecting it to the sea. As stated before, the divers are at a neutral buoyancy so if they swim out of the submarine, their volume will be filled water water and causes therefore almost no change in weight. Depending on the place of the cofferdam and the place of the connected ballast tank, this operation could give a trimming moment. After the divers exit of the submarine, the water inside the cofferdam is transferred back to the ballast tank.

A SSK submarine is always equipped with an emergency escape tower. Special forces can use this to disembark and embark. If the escape possibility for the special forces is filled with water from outside the submarine, there will be a change of weight for which the trim system has to compensate. The disturbances caused by filling the escaperoom with seawater can be determined with following parameters:

- size of the escaperoom;
- the number of people in the escaperoom;
- the position of the escaperoom;
- the filling rate;

At the start of disembarkation special forces, SF, they will first walk towards the escaperoom. This will cause a moment. When the SF are inside the room, the room will be filled with seawater from outside. Filling the room will therefore cause moment and change in weight. The volume of water in the escape tank can be determined by extracting the volume of one person including gear from the total volume of the escape room. The volume of a diver including gear is assumed to be 0.2 m^3 (Geelhoed-Bakiu, 2016). Equation 5.38 gives the force due to filling the escape room with water and equation 5.37 gives the moment that filling the escape room results in. In both equations it is assumed that the escape rooms are filled with seawater from the outside.

$$F_{waterescaperoom} = (V_{escaperoom} - V_{diver} * N_{divers}) * \rho_{seawater} \tag{5.36}$$

$$M_{waterescaperoom} = (V_{escaperoom} - V_{diver} * N_{divers}) * \rho_{seawater} * (x - LCB) \tag{5.37}$$

With: F is the force in N; V is the volume in m^3; N_{diver} is the number of divers; ρ is the density in kg/m^3; M is the moment in Nm; x is the position of the escape room in m; LCB is the centre of buoyancy in m.

The compensation system must expel the same weight from the submarine as that which is needed for filling the escape room. The moment caused by the the special forces escape room depends on the position of the tank. The closer it is positioned towards the centre of buoyancy, the smaller the trimming moment.

The horizontal escape, like built in the A26 of SAAB Kockums, is positioned in the front of the submarine (Saab Kockums) and is called multi-mission portal. This solution results in the biggest trimming moment. However, this system allows multiple people to escape at once, which is not possible with an emergency escape tower. Another advantages is that it is a built-in system. The dry-dock shelter comes on top of the submarine and effects the signature of the submarine. The cofferdam is placed in the middle of the submarine and is therefore less practical with AUV operation since this is where the living quarters and the command centre are situated. Usually places where AUVs are stored is in the front of the submarine. When having the escape possibility in the middle of the submarine, the AUV must be transported through the living quarters and command centre. Having the escape possibility in the front of the submarine is therefore the best option which is evaluated in this research. The specifications of the multi-mission portal, MMP, can be found in table 5.9. The filling rate is taken as 60m^3/h.

Table 5.9: Specification multimission portal

# people	**L [m]**	**D [m]**	**V[m^3]**	**Q [m^3/h]**	x **[m]**
8	6	1.5	10.6	60	$L_{submarine}$-3

With this data the disturbance from filling a multi-mission escape room can be calculated. There is a maximum force which is described by equation 5.38. For the force per second, the filling rate of the tube is to be taken into account with equation 5.39. Because the tube is not located at the centre of buoyancy, filling the escape room will also give a trimming moment. This moment is calculated with equation 5.40.

$$F_{escaperoom,max} = (10600 - 200 * N_{divers}) * \rho_{seawater} g \tag{5.38}$$

$$\dot{F}_{escaperoom} = \frac{60}{3600} * \rho_{seawater} g \tag{5.39}$$

$$\dot{M}_{escaperoom} = \dot{F}_{escaperoom} * (x - LCB) \tag{5.40}$$

With: F is the force in N; N_{divers} is the number of divers going in the escape room; ρ is the density in kg/m^3; x is the position of the escape room in m; LCB is the centre of buoyancy in m.

It should be considered that even though the filling rates are equal, the maximum disturbance is when there are no people inside the escape room. This is because the maximum force will in this case be higher and therefore the filling time will be longer.

Before the special forces enter the MMP, they have to walk fully equipped towards the front of the submarine. This is calculated the same way as walking people in the submarine, equation 5.2 and 5.3. Only the weight of the persons is changed. It is expected that the weight of a fully equipped diver is higher than that of a submariner. 200 kg is taken as the weight which is also used in earlier research (Geelhoed-Bakiu, 2016).

5.4.3. AUVs

AUVs have become more important for underwater missions, with technology becoming more advanced and their uses more varied. However an AUV needs a main platform to operate from. Using a submarine as a main platform is for some operations practical (De Boer, 2015). This results in the requirements that a submarine should be able to deploy and retrieve an AUV.

The retrieval and deployment of an AUV looks very similar to the missions done with the special forces, and the multi-mission portal used for SF can also be used for AUVs. Therefore the disturbances can be calculated in the same way. Equation 5.38 is modified to equation 5.41 to obtain the right disturbance.

$$F_{escaperoom} = (V_{escaperoom} - V_{AUV} * N_{AUV}) * \rho_{seawater} g \tag{5.41}$$

With: F is the force in N; V is the volume in m^3; N_{AUV} is the number of AUVs disembarking; ρ is the density in kg/m^3.

For the deployment of divers there is a maximum depth based on the maximum pressure for a diver. AUVs' can dive deeper and are therefore most likely to be retrieved and deployed at greater depths. Therefore the disturbances caused by the surface waves are lower.

5.4.4. Anchoring

Submarines can have an anchor (McLaughlin et al., 2010). This anchor can be used in harbours but also when submerged to maintain position. Especially within littoral water missions, underwater anchoring can have an advantage. This way the submarine can stay at position without driving its propeller.

The size of an anchor depends on the size of the ship. For surface ships the so-called equipment number determines the size of the anchor. The equipment number can be calculated with equation 5.42. When knowing the equipment number the weight of the anchor can be determined according to the DNV rules (DNV, 2010). In here BH stands for the frontal surface of the ship above the waterline and A for the side surface of the ship. For a submarine the superstructure can be considered as only the sail. The part of the hull that is above the waterline is assumed 10% of the height of the submarine based on submarine characteristics (Renilson, 2015). This will lead to equation 5.43 as approximation of the equipment number. The weight of the anchor can be found in a table according to DNV (2010).

$$EN = \Delta^{2/3} + 2BH_{superstructure} + 0.1A \tag{5.42}$$

$$EN = \Delta^{2/3} + 2B * (0.1 * H_{hull} + H_{sail}) + 0.1 * (0.1 * H_{hull} * L_{submarine} + H_{sail} * L_{sail}) \tag{5.43}$$

With: EN is the equipment number; Δ is the displacement in t; B is the width in m; H is the height in m; A is the side area in m^2; L is the length in m.

The biggest disturbances that the anchor gives are during lowering. When the anchor is lowered and touches the seabed, the weight of the anchor can be subtracted from the total weight of the submarine. If the anchor is positioned at the front of the submarine, it will cause the nose of the submarine to move upwards. The force to which the submarine is subjected to when the anchor hits the bottom can be found with equation 5.44. The anchor is expected to be situated at the front of the submarine. The resulting moment can be found with equation 5.45. Since the force and the moment will only occur when the anchor hit the bottom, which is in a split second, the force and the moment caused by the anchor are time independent.

$$F_{anchor,max} = m_{anchor} g \tag{5.44}$$

$$M_{anchor,max} = m_{anchor} g * (L_{submarine} - LCB) \tag{5.45}$$

Width: F is the force in N; m is the mass in kg; M is the moment in Nm; L is the length in m; LCB is the longitudinal centre of gravity in m.

5.4.5. Rescue

A rescue mission is performed when another submarine has sunk. During a rescue mission, the operational submarine will sail to the sunken submarine. The operational submarine will then deploy a rescue capsule. This capsule is the transfer between the two submarines. the capsule will have to land on top of the submarine at a hatch to allow the crew to get in and out of the rescue capsule. Since the rescue capsule also is at neutral buoyancy, landing the capsule on the operational submarine does not give a force or moment on the submarine. Therefore this specific mission is not taken into account for the ballast trim system of the submarine

5.5. Anechoic tiles or coatings

New techniques and developments make it interesting to cover the submarine with anechoic tiles or coating. These anechoic tiles or coating reduce the signature of the submarine, making it less detectable by sonar. The downside of these tiles on the submarines' hull is that the submarine becomes more compressible. The type of anechoic tiles influences the compressibility. For example rubber tiles are more compressible than sandwich tiles.

When a submarine dives, the pressure acting on the submarine increases. This causes the anechoic tiles to compress which results in less buoyancy. Therefore submarines equipped with anechoic tiles or coatings become more unstable. To avoid undesired movements of the submarine, the ballast trim system should be able to maintain equilibrium of the more unstable submarine.

The BB2 hull is used to study the effects of the anechoic tiles. By varying the total area of the anechoic tiles and the E-modulus of these tiles, a relation is found between the size of the submarine, the E-modulus of the anechoic tiles and the disturbance.

When the submarine experiences a positive trim angle, the front of the submarine is subjected to a higher pressure than the aft of the submarine. Due to the compression of the submarines' anechoic tiles, the buoyancy force at the front of the submarine will become lower than the average buoyancy force. Vice versa, the buoyancy force at the aft of the submarine becomes more buoyant. This causes a trimming moment. Figure 5.15 shows the working principle of this phenomenon.

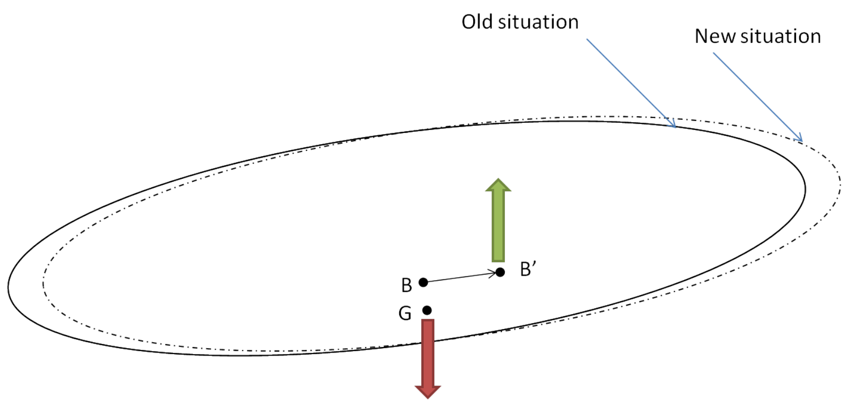


Figure 5.15: Trimming moment due to compression of the hull and trim angle

A few assumptions are made in order to model the anechoic tiles. The first assumption is that the submarine is an ellipsoid. This way the sail is left out and the submarine is symmetrical in xy-plane. The hydrostatic coefficients are taken from the model and not based on an ellipsoid. The other assumption is that the change in volume is linear. The E-modulus is taken to model the volume change of the submarine. It is assumed that the change in volume will be small and will therefore not influence the LCB. The last assumption is that the trim angle is small. With these assumptions a model can be made, which takes into account the change of volume due to the trim angle at a certain condition. The change of volume of a certain point along the x-axis of the submarine is the same at the top of the submarine as it is at the bottom of the submarine. An overview of how to obtain a model for the compressibility of the submarine can be found in figure 5.16.

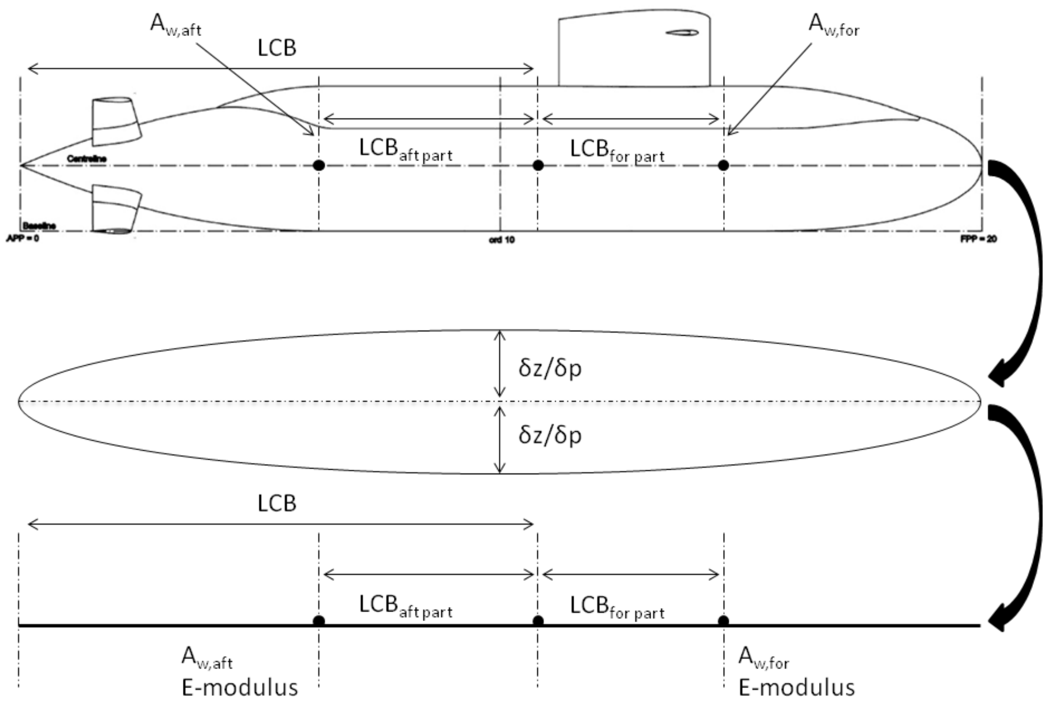


Figure 5.16: Determination of model submarine to calculate compressibility

To make an indication of the disturbance caused by the anechoic tiles the following parameters are needed:

- size of the submarine, especially wetted surface;
- E-modulus of the anechoic tiles;
- hydrostatics of the submarine.

The moment for different E-moduli for a submarine with a 10° trim angle can be found in figure 5.17. Figure 5.17 shows that both the size of the submarine and the elastic modulus influence the extra trimming moment because of having anechoic tiles.

More obvious is the change in buoyancy due to the compressibility of the submarine. The change of the buoyancy force for a 100 meter dive can be found in figure 5.18. The dependency of the size of the submarine and the E-modulus of the anechoic tiles can be seen here. The bigger the submarine, the bigger the wetted area, the bigger the loss in buoyancy due to a descent.

The disturbance due to the anechoic plates can be determined by equations 5.46, 5.47 and 5.48.

$$dF_{buoyancy} = \frac{\rho^2 g^2 (T - 0.5H_{hull}) * t_{tiles}}{E} * A_w \tag{5.46}$$

$$dM_{buoyancy,for} = \frac{\sin(\theta) * (LCB_{for} - LCB)^2 \rho^2 g^2 (T - 0.5H_{hull}) * t_{tiles}}{E} * A_{w,for} \tag{5.47}$$

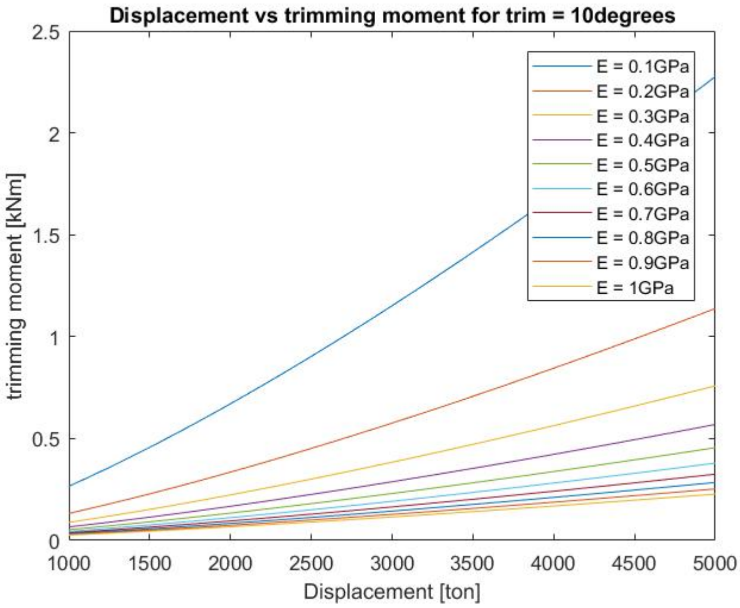


Figure 5.17: Displacement versus trimming moment

$$dM_{buoyancy,aft} = -\frac{\sin(\theta) * (LCB_{aft} - LCB)^2 \rho^2 g(T - 0.5H_{hull}) * t_{tiles}}{E} * A_{w,aft} \tag{5.48}$$

With: dF is the change of force in N; ρ is the density in kg/m^3; T is the depth of the submarine measured from the keel in m; H is the height of the ellipsoid in m; E is the E-modulus of the anechoic tiles in N/m^2; A_w is the wetted area of the submarine in m^2; ϕ is the trim angle in degrees; LCB is the centre of buoyancy in m; t_{tiles} is the thickness of the anechoic tiles in m.

In this research the E-modulus of the anechoic tiles is taken as 0.1 GPa. This gives the largest disturbance as seen in figure 5.17 and 5.18. The thickness of anechoic tiles are usually 2.5 cm (Charles Q. Choi, 2015). This is taken into account in this thesis. An anechoic coating can also reduce the submarines' signature. These coatings are very thin resulting in a more stable submarine. This should be taken into account when designing the submarine.

5.6. Overview

When all the disturbances are determined, these can be added to obtain the total disturbance on the submarine. A few parameters still have to be determined in order to obtain the right order of disturbances. An overview of all the disturbances can be found in table 5.10 and table 5.11. In Appendix 5.1 an overview of the position of the tanks can be found.

In the previous sections also values are determined for positions, sizes and other detailed data for the specific disturbances. With these parameters the disturbances for submarine 1 and 2 can be determined. An overview of these specific parameters can be found in table 5.12.

5.7. Load Cases

With these disturbances there are five load cases set up that are studied. These five load cases take into account specific disturbances which influence the ballast trim system. The profiles that are studied are:

- transit snorkelling
- transit AIP
- disembark SF
- disembark AUV

Table 5.10: Disturbance forces

Forces	$F[N]$	$\dot{F}[N/s]$
Fuel	continuous	$\frac{\rho_{seawater}-\rho_{fuel}}{\rho_{fuel}} * \frac{sfc}{3.6*10^6} * g * P(t)$
LOX	continuous	$\frac{soc}{3.6*10^6} * g * P(t)$
Periscope / mast	$L_{perscope} * \frac{1}{4} D^2_{periscope} \pi \rho_{seawater} g$	$\frac{1}{4} D^2_{periscope} \pi \rho_{seawater} g v_{raising}$
Snorkel mast	$L_{snorkelmast} \frac{1}{4} D_{snorkelmast} \pi \rho_{seawater} g$	$\frac{1}{4} D_{snorkelmast} \pi \rho_{seawater} g v_{raising}$
Disembark SF	$(V_{Escaperoom} - V_{SF} * N_{SF}) * \rho_{seawater}$	$\frac{60}{3600} \rho_{seawater}$
Disembark AUV	$(V_{Escaperoom} - V_{AUV}) * \rho_{seawater}$	$\frac{60}{3600} \rho_{seawater}$
Anchor	$m_{anchor} g$	at once
Density	$-2g * \Delta_{submarine}$	at once
Second order effect	continuous	$F_a^2 \cos^2(\omega_g t)$

Table 5.11: Disturbances Moments

Moments	$M[N]$	$M[\dot{N}/s]$
Walking submariners	$\frac{10\% N_{submariners} * 80 * g}{\Delta_{submarine}} * 0.5 L_{submarine}$	$\frac{10\% N_{submariners} * 80 * g}{\Delta_{submarine}} * v_{walking}$
Waste water	$20 * N_{submariners} * g * \frac{1}{12} L_{submarine}$	$\frac{20 * N_{submariners}}{3600} * g * \frac{1}{12} L_{submarine}$
Disembark SF	$(V_{Escaperoom} - V_{SF} * N_{SF}) * \rho_{seawater} * (\overline{x} - LCB)$	$\frac{60}{3600} \rho_{seawater} * (\overline{x} - LCB)$
Walking SF	$m_{SF} 0.5 L_{submarine} g$	$\frac{m_{SF}}{1000} v_{walking} g$
Disembark AUV	$(V_{Escaperoom} - V_{AUV}) * \rho_{seawater} * (\overline{x} - LCB) g$	$\frac{60}{3600} \rho_{seawater} * (\overline{x} - LCB)$
AUV transporting	$m_{SF} * x * AUV g$	$m_{SF} * v_{transporting} * g$
Anchor	$m_{anchor} * (L_{submarine} - LCB) * g$	at once

Table 5.12: Parameters that influence the disturbances

Characteristics	**Symbol**	**Order**	**Submarine 1**	**Submarine 2**
Length mast	L_{mast}	[m]	5	5
Diameter mast	D_{mast}	[m]	0.5	0.5
Length snorkel	$L_{snorkel}$	[m]	5.0	5.0
Diameter snorkel	$D_{snorkel}$	[m]	1.0	1.0
Raising/lowering speed	$v_{raising}$	[m]	0.67	0.67
Number submariners	N	[#]	67	36
weight submariner	$m_{submariner}$	[kg]	80	80
Number SF	N_{SF}	[#]	8	8
weight SF	m_SF	[kg]	200	200
position MMP	x_{MMP}	[m]	67.20	52.72
Filling speed MMP	Q_{MMP}	[m^3/h]	60	60
weight of anchor	m_{anchor}	[kg]	1140	780
Weight AUV	m_{AUV}	[kg]	540	540
Travel distance AUV	x_{AUV}	[m]	8	6.97
density fuel	ρ_{fuel}	[ton/m^3]	0.8151	0.8151
density water	$\rho_{seawater}$	[ton/m^3]	1.025	1.025
sfc diesel	sfc	[g/kWh]	225	225
sfc stirling	sfc	[g/kWh]	216	216
soc stirling	soc	[g/kWh]	823	823
Equipment number	EN	[-]	393	248

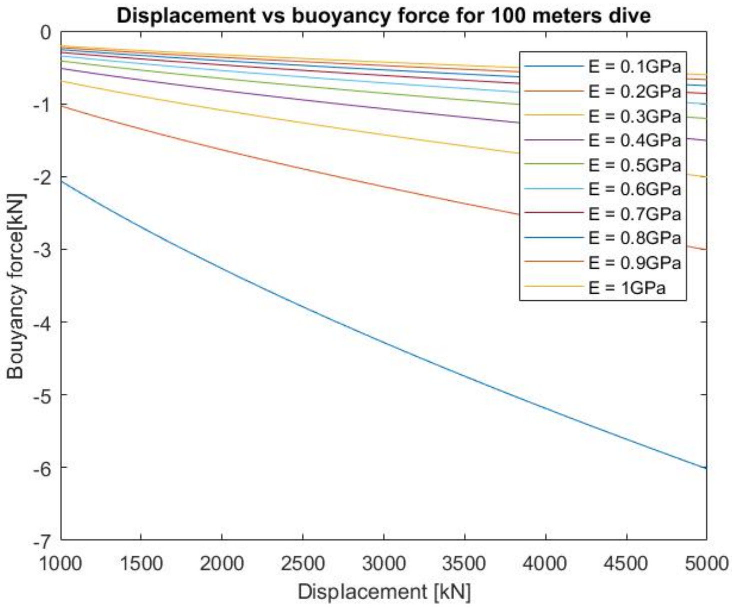


Figure 5.18: Displacent versus buoyancy force

- underwater anchoring

These five different profiles distinguish themselves in the expected disturbances as well as in requirements to the system. Some moments that the submarine can expect are always there. These moments are:

- submariners walking
- water usage

First of all the disturbances determined in earlier sections are calculated for submarine 1 and submarine 2. This is done by using the submarine's main parameters and using the specific needed information which is determined in section 5.6.

When all the needed parameters are filled all the disturbances are quantified. This result in two kinds of disturbances. The first set of disturbances are in the z-direction and are given in table 5.13. The second set of disturbances are the moments around the y-axis. These can be found in table 5.14.

Table 5.13: Disturbance forces

	Submarine 1		**Submarine 2**	
	Force max [kN]	Force/s [kN/s]	Force max [kN]	Force/s [kN/s]
Fuel	continuous	-4.817E-4	continuous	-3.406E-4
Fuel stirling	continuous	-1.334 E-4	continuous	-9.430 E-5
LOx stirling	continuous	1.974E-3	continuous	1.395E-3
Second order effects				
Density	78.480	at once	39.240	at once
Mast	9.872	1.316	9.872	1.316
Snorkel	39.487	5.265	39.487	5.265
Escape SF	-90.497	-0.168	-90.497	-0.168
Anchor	11.183	at once	7.652	at once

Table 5.14: Disturbance moments

	Submarine 1		Submarine 2	
	Momant max [kNm]	Moment/s [kNm/s]	Momant max [kNm]	Moment/s [kNm/s]
Submariners walking	184.561	7.303	78.709	3.924
Water usage	-76.901	-0.021	-32.795	-0.009
Escape SF	2656.004	4.919	2052.063	3.800
Walking SF	550.930	21.800	437.273	21.800
Anchor	-361.772	at once	-196.463	at once

The resulting loadcases for submarine 1 will be shown in the following subsections. For submarine 2 and the enlarged figures for submarine 1 can be found in appendix D.2.

5.7.1. Transit snorkelling

For snorkelling, it is assumed that all the masts are raised. The total force for this load case will therefore be modelled with the four small masts of periscope size, and the snorkel mast. Furthermore, during snorkelling fuel is used which must be compensated. The moments that can be expected are the moments that are always present. The first moment is due to submariners walking in the submarine. The second moment which is considered is the moment due to water usage. Adding these forces and moments will result in a simulated submarine profile. This profile can be found in figure 5.19 for the forces and figure 5.20 for the trimming moment. The snorkelling profile will occur at periscope depth such that the snorkel and other masts are above the waterline.

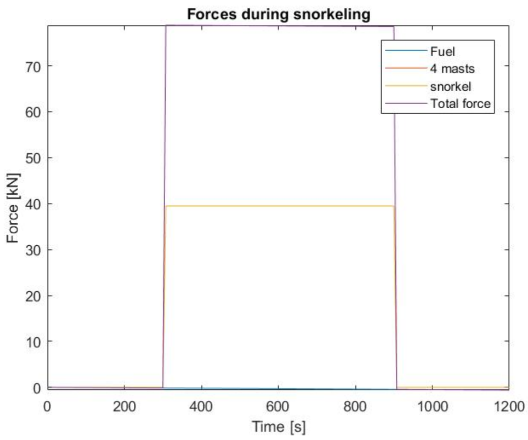


Figure 5.19: Forces during snorkelling submarine 1

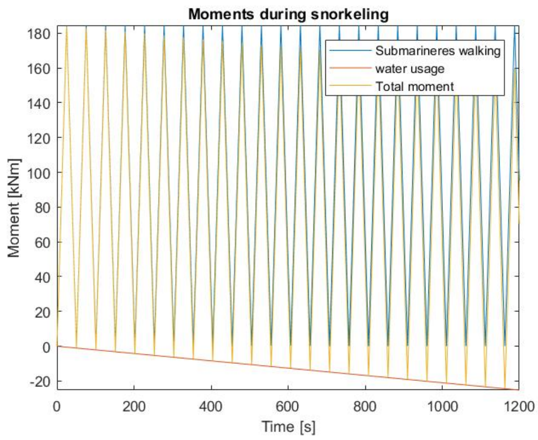


Figure 5.20: Moments during snorkelling submarine 1

5.7.2. Transit AIP

Transit on AIP is assumed to occur at large depths. For simulations 300 meter depth is taken. The moments taken into account during this profile are the walking people on board and the water usage.

The disturbances that occur during this profile are the use of fuel for the AIP system and the use of LOX. Since the depth is set at 300 meters the second order effects are not taken into account, as they are not present. The change in density can occur and is therefore taken into account. The forces and moments during this load case are given in figure 5.21 and 5.22.

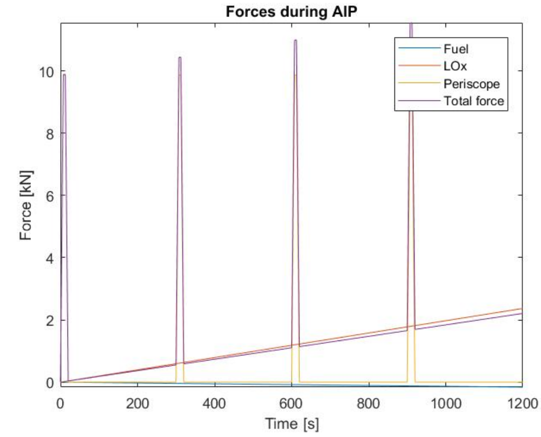


Figure 5.21: Forces during AIP transit submarine 1

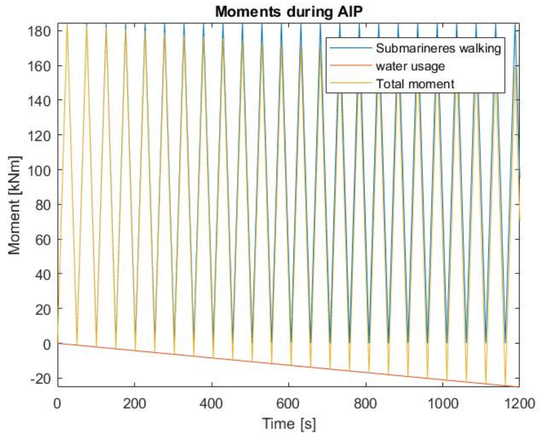


Figure 5.22: Moments during AIP transit submarine 1

5.7.3. Disembark SF

The disembarkation of SF is usually done at periscope depth. This is done because the SF can not be deployed/recovered at extremely high pressures. When at periscope depth, the area can also be scanned from the submarine. For the disturbance due to disembarking SF the forces of the periscope are taken into account as well as the forces due to filling the multi-mission portal. For the moments, those due to submariners walking, water usage and the moment due to filling the multi-mission portal is taken into account. figure 5.23 and 5.24 show the profile for this load case for submarine 1. For submarine 2 these figures can be found in appendix D.2.

The depth is taken as periscope depth which is different for both studied submarines. At periscope depth the second order effect due to waves can be felt and should therefore be taken into account. A change in density can also occur at this depth and must therefore be compensated when occurring.

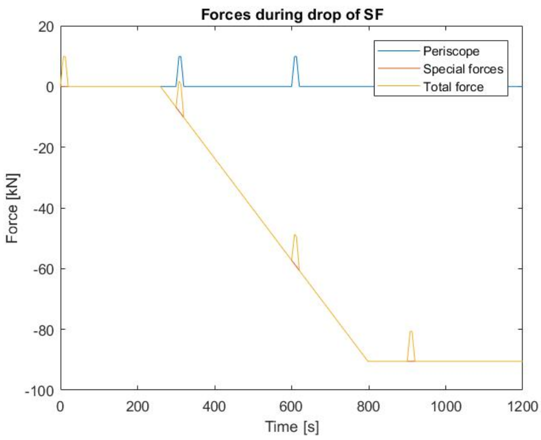


Figure 5.23: Forces during disembarking SF submarine 1

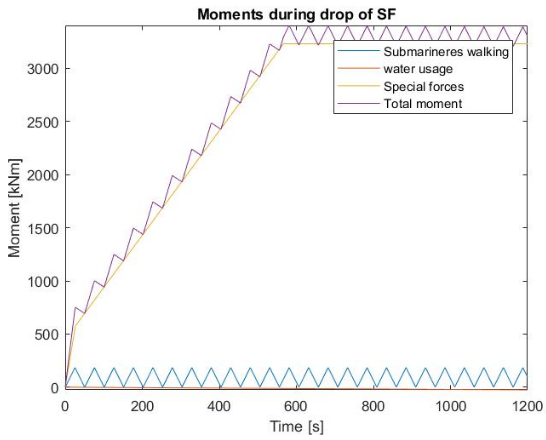


Figure 5.24: Moments during disembarking SF submarine 1

5.7.4. Disembark AUV

One of the advantages is that AUVs can be disembarked at large depths. Therefore the depth for this profile is taken as 300 meters. The speed during this mission is taken as 0 kts. The moments that are taken into account are the moments that always are present: walking submariners and water usage. For this profile the extra moment of disembarking the AUV is also taken into account

The forces that are present during this profile are the forces due to the filling of the MMP before the AUVs can be disembarked. Furthermore at this depth only density changes are taken into account The forces and moments during this load case are given in figure 5.25 and 5.26.

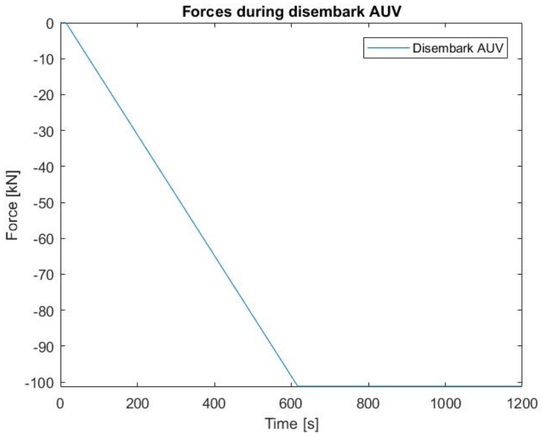


Figure 5.25: Forces during disembarking SF submarine 1

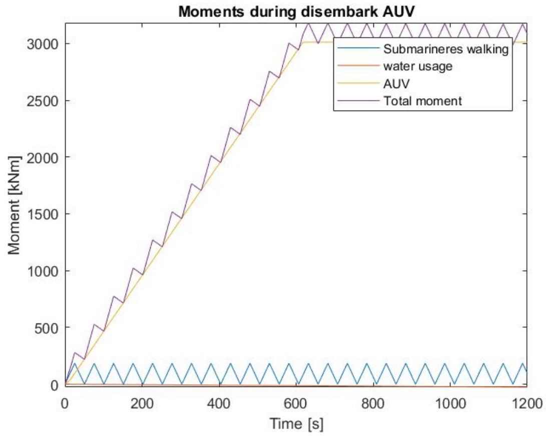


Figure 5.26: Moments during disembarking SF submarine 1

5.7.5. Anchoring

For the profile of anchoring, underwater anchoring is considered. This underwater anchoring is simulated at 300 meters. It is expected that during this profile the submarine has a speed of 0 knots. The moments that are taken into account for this profile are the walking people on board and the usage of water. The specific profile related moment is the drop of the anchor.

The forces that are taken into account for this profile are only the forces due to the loss of weight of the anchor. Furthermore it is expected that the hover system must deal with a change in density of the seawater. The forces and moments during this load case are given in figure 5.27 and 5.28.

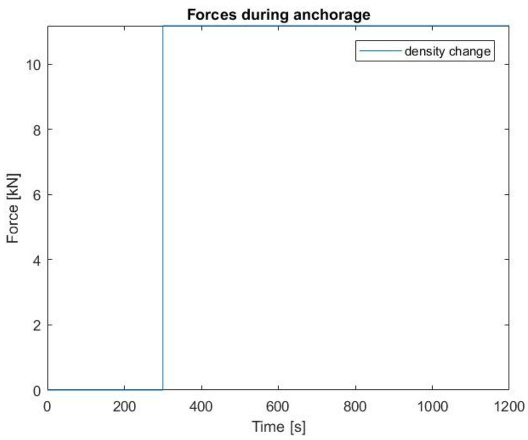


Figure 5.27: Forces during anchoring submarine 1

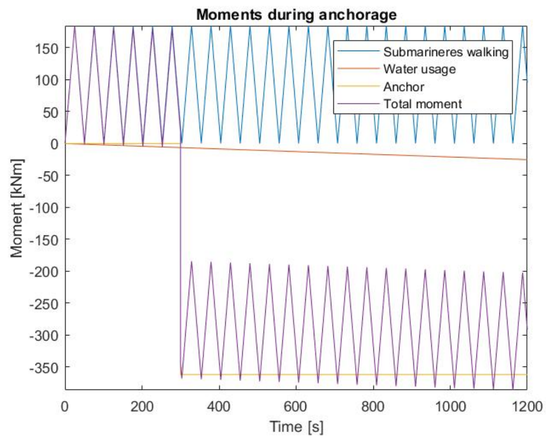


Figure 5.28: Moments during anchoring submarine 1

5.7.6. Change of density

During hovering the density can change. This change in density can be found at periscope depth as well as maximum depth. Especially when the submarine has zero velocity this disturbance is very important. The change in density is modelled for a negative change of density. This will result in less buoyancy force and therefore a descent of the submarine. When having a positive change in density, a tank can be filled with ballast water and has therefore no effect on the actuators size. The profile of this disturbance can be found in figure 5.29.

5.7.7. Second order wave effects

When operating close to the water surface, as for example at PD, the second order effects due to the waves can be felt. Due to the second order forces an harmonic disturbance can be felt by the submarine. This is in fact only a suction force, only positive. For the disturbance profile it is expected that the submarine will first compensate for the average disturbance, taking in water is such a way the submarine has on average zero

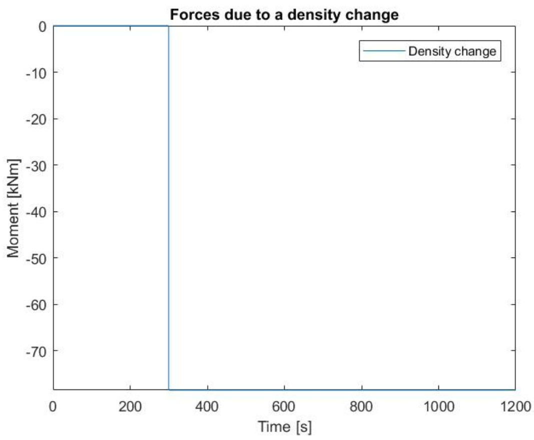


Figure 5.29: Forces during hovering submarine 1

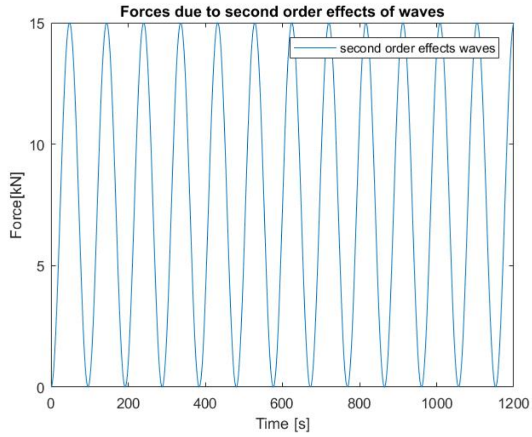


Figure 5.30: Forces during hovering due to second order effects submarine 1

disturbance. This results in that the disturbance profile of the second order wave forces can be described by a harmonic force around the zero axis. This is shown in figure 5.30.

6

Options and models of the ballast trim system

In chapter 6, multiple solutions were identified which could be used as ballast trim system. Some of these solutions will be modelled and compared. Not all solutions are expected to be suitable for use as a ballast trim system on board a SSK submarine. Therefore a multi criteria analysis, MCA, is made to investigate the solutions to be modelled for the three modes: trimming, compensating and hovering. The selected solutions are modelled and discussed in this chapter. The solutions are modelled as a chain of smaller models. The total chain of these sub models will then describe the total solution based on the parameters of submarine 1 and 2.

6.1. Options to model

Chapter 4 gives an overview of all the available solutions used for trimming, compensating and hovering. First for the three different modes a MCA is executed to see which systems are interesting to compare to each other. Some options will therefore not be selected to model since it is expected that they will in the end not work proper enough to use on board of a modern SSK submarine.

6.1.1. Trim

Trim can occur due to moments that act on the submarine as discussed in chapter 5. Due to these moments the submarine will trim as seen in chapter 2. The trim systems from chapter 4 are all capable of maintaining equilibrium underwater. However it can be seen in an early stage that not all systems will work as well as others. A global overview of the trim system can be found in figure 6.1. As seen, all the trim systems from chapter 4 transport fluid between the tanks to create a counter moment. The system is considered a closed system. This is because having an open system results in a combination of trimming and compensating. With an open system, it is not possible to compare the effectiveness of the different options with respect to trim.

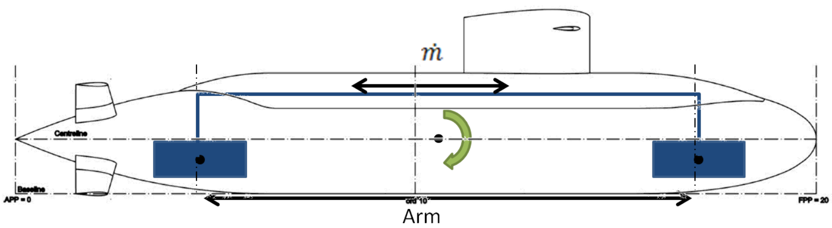


Figure 6.1: The forces taken into account for the trim system

The MCA will help to make the decision of which systems are going to be modelled. The MCA can be found in table 6.1. From this MCA it was chosen to model the system including a centrifugal pump and the system which uses pre-pressurised tanks.

Table 6.1: MCA trim system

Criteria	Factor	Centrifugal pump	Piston pump	Blowing and venting	Pre-pressurised tank
Reliancy	*2*	3	3	4	4
Noise	*3*	3	2	1	4
Energy	*3*	3	4	1	1
Reaction time	*1*	3	3	3	4
Cost	*2*	3	2	4	4
score		**33**	**31**	**25**	**35**

For the systems that are modelled, system diagrams are made. The symbols that are used in these diagrams can be found in figure 6.2. The system models for the trim system can be found in 6.3 and 6.4. Appendix F.4 give an overview of the used input as system in the model such as pipe lengths and diameters.

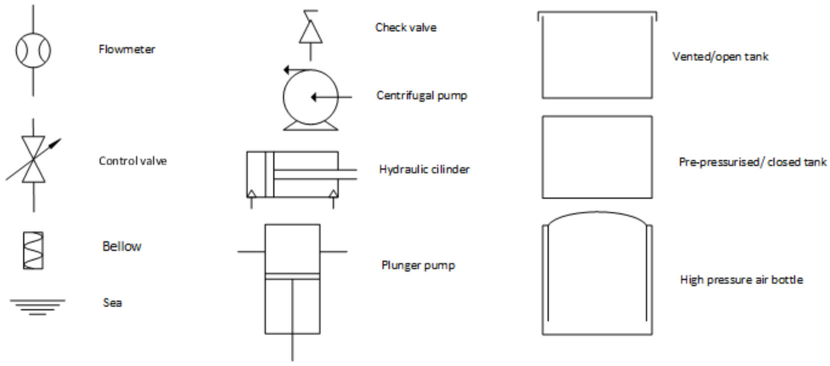


Figure 6.2: Legend for the system diagrams

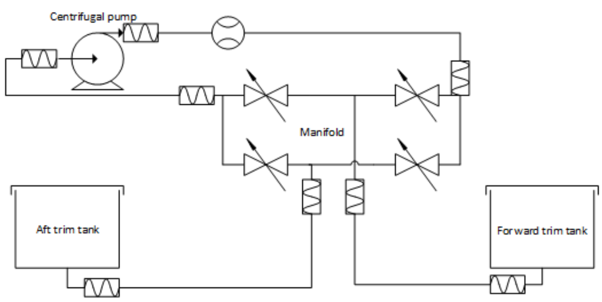


Figure 6.3: Trim system with centrifugal pump

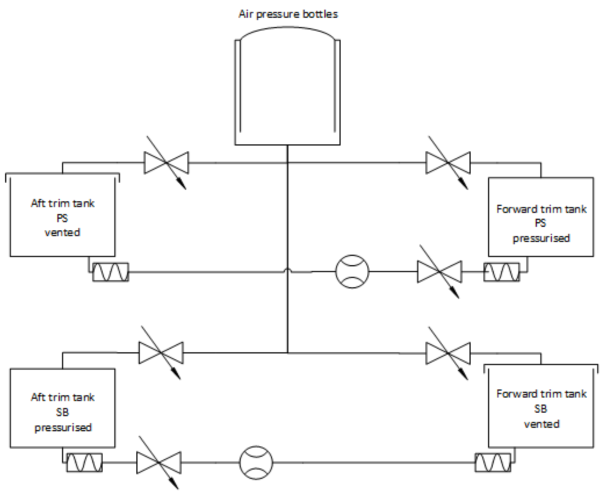


Figure 6.4: Trim system with pre-pressurised tanks

6.1.2. Compensation

Due to disturbances in the equilibrium, the submarine can start heaving. This is already discussed in chapter 2. The systems that can be used to compensate for this motion are discussed in chapter 4. Figure 6.5 gives a brief overview of how these forces work and what counter measures there must be initiated from the compensating system. To compare the different solutions for compensating, the compensation system is also assumed to be not connected to the trim tanks. This way compensating can only be done by the compensating system and the compensating tanks.

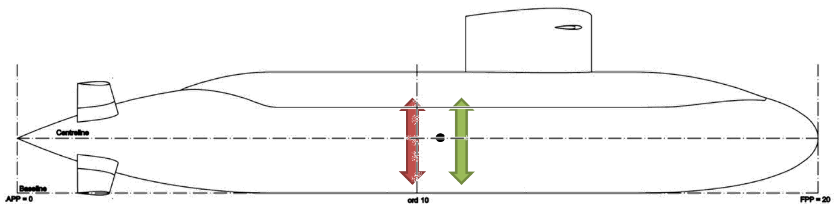


Figure 6.5: Forces considered during compensation and hovering

To see which models need to be simulated in Simulink, a MCA is made for the compensationo system solutions. This way there is first a selection of which solutions are taken into account in the comparison. The MCA made can be found in table 6.2. From here there can be seen that the solutions that will be implemented in a Simulink model are the pumping systems. A centrifugal pump, a centrifugal pumps in series and a plunger pump will be modelled.

Table 6.2: MCA compensation system

Criteria	Factor	Centrifugal pump	Centrifugal pump series	Piston pump	Blowing and venting	Pre-pressurised tank	Variable buoyancy	propellor
Reliancy	*2*	3	2	3	4	4	3	3
Noise	*2*	3	3	2	1	4	4	2
Energy	*3*	3	4	4	1	1	3	4
Reaction time	*1*	3	3	3	3	4	4	3
Price	*3*	3	3	2	4	2	1	1
score		**33**	**34**	**31**	**28**	**29**	**30**	**28**

For the three models that are made for the compensation system, system diagrams are made. The first system diagram is for the compensation system using the centrifugal pump in figure 6.6. The second system diagram is for the model with the centrifugal pumps in series, figure 6.7. The third system diagram consist the model with the plunger pump. This diagram can be found in figure 6.8. All these diagrams can also be found in appendix F.4 with the more detailed data of the modelled systems.

6.1.3. Hovering

The main advantages of being able to hover is that it is safe for the submarine to execute missions without using the hydroplanes. The disturbances that the hover system has to compensate for are already discussed in chapter 4. The systems seen for hovering are the same systems as used for compensating. This because in principle the way of compensating the heave motion is the same, see figure 6.5. However the criteria of the hover system are different and therefore the solutions will differ. For hovering there is also made an MCA to see which solutions have to be modelled and which not. The MCA for hovering can be found in table 6.3. The solutions that are modelled are: the centrifugal pump, the centrifugal pumps in series, the piston pump, the pre-pressurised tank and the variable buoyancy. The pump systems are taken the same is for the

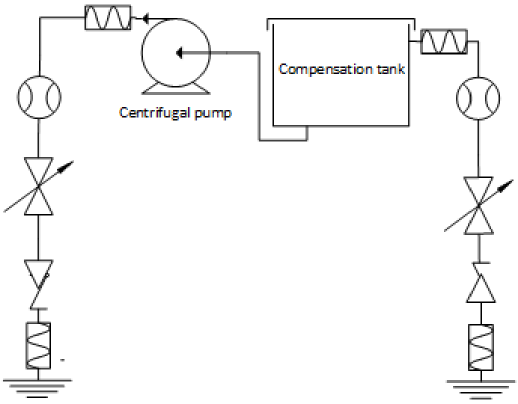


Figure 6.6: Compensation system with centrifugal pump

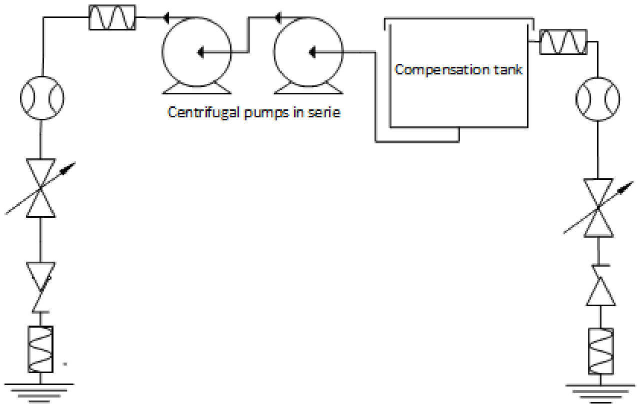


Figure 6.7: Compensation system with centrifugal pumps in series

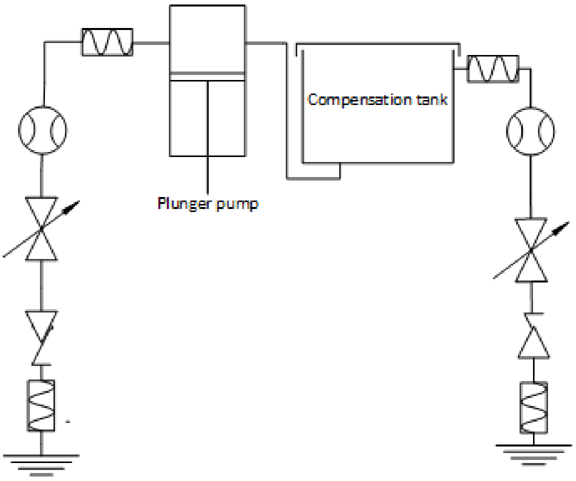


Figure 6.8: Compensation system with piston pump

compensation system. In practise this means that the compensation system is used for hovering.

Table 6.3: MCA hovering system

Criteria	Factor	Centrifugal pump	Centrifugal pump series	Piston pump	Blowing and venting	Pre-pressurised tank	Variable buoyancy	propellor
Reliability	*1*	3	2	3	4	4	3	3
Noise	*3*	3	3	2	1	4	4	2
Energy	*2*	3	4	4	1	1	3	4
Reaction time	*3*	3	3	3	3	4	4	3
Price	*1*	3	3	2	4	2	1	1
score		**30**	**31**	**28**	**22**	**32**	**34**	**27**

The first model for the hovering system consist of a large centrifugal pump. This can be found in figure 6.6. The second model is the model with two centrifugal pumps in series. The system diagram of this solution can be found in figure 6.7. The third system diagram, figure 6.8, consists of the model of the piston pump.

In figure 6.10 the system diagram for the pre-pressurised tanks can be found. To conclude, in figure 6.9 the hovering system which uses the variable buoyancy solution is displayed. All the modelled systems for hover can be found in appendix F.4. In here is also an overview of the pipe lengths and resistances added to the systems.

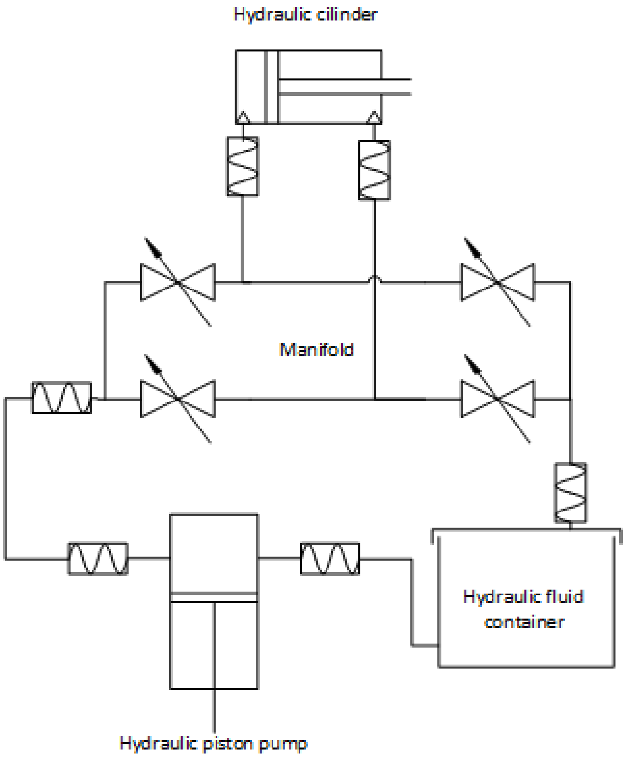


Figure 6.9: Hover system with variable buoyancy

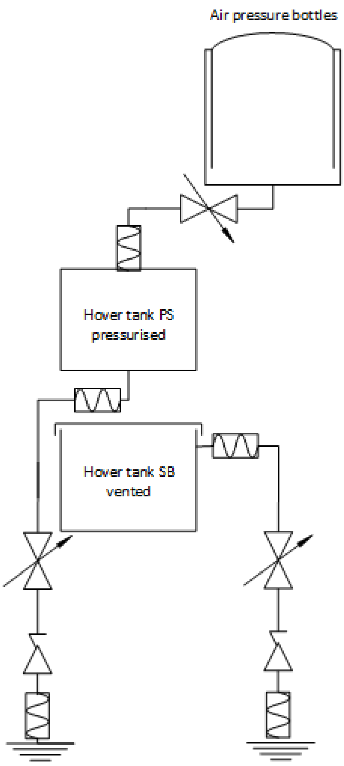


Figure 6.10: Hover system with pre-pressurised tanks

6.2. Models

To make a comparison of the different options, models have been made in Simulink to see the behaviour of the different systems discussed in section 6.1. The models in Simulink for the ballast trim system are made by setting up seven main blocks which take into account the different actuators and parts of the system. These seven blocks will be discussed and explained in this section. The seventh block is not directly part of the ballast trim system, but is the submarine itself. The motions of the submarine are studied and therefore also a block for the submarine is needed. The models that are made are:

- pipe flow model;
- centrifugal pump model;
- plunger pump model
- variable buoyancy;
- valve model;
- tank model;
- submarine.

The following subsections will give the mathematical description of the models. Also the assumptions made in the models will be discussed. With these blocks the total solution for a ballast trim system can be created. The solutions are than compared to each other in chapter 7.

6.2.1. Pipe flow system

Several pipes can be found in the different solutions for the ballast trim system. The main method to determine the size of a pipe flow system is to start with the maximum flow velocity through the pipe. The velocity of the fluid in the pipe is in most cases given. For water pipe lines 1.5 m/s is used as maximum velocity and for hydraulic pipes 4.5 m/s is used as maximum velocity. This velocity is based upon the restrictions of noise production. When the flow velocity is known, the pipe can be dimensioned based on the required mass flow. The resistance in the pipe can be determined by knowing the lay-out of the pipe. Arcs and appendages on the pipework result in extra resistance.

Energy

The pipe gives a certain resistance which causes a pressure drop in the fluid flowing through the pipe. This resistance depends on:

- the pipe diameter;
- the length of the pipe;
- the roughness of the pipe;
- difference in height between inlet and outlet.
- the density of the fluid;
- the viscosity of the fluid;
- the velocity of the fluid.

The fluid is considered to be homogeneous. This will lead to a model that consists of the pipe information and the fluid properties. The model must be dynamic since the velocity of the fluid changes. Not only does the velocity of the fluid change but also static height of pipe lines which are longitudinally placed. The submarine can trim during operations resulting in a static height difference in the pipe.

The losses in a pipe system can be calculated with the use of equation 6.1. ζcan be determined with equation 6.2. In here coefficient for the friction losses in a pipe can be found with the Moody chart or equation 6.3. For the model an approximation of the moody diagram is used to calculate the friction factor of the pipe. The Haaland equation, equation 6.3 , gives a good approximation of the the friction factor for Reynolds number above 2300 (White, 2008). For lower Reynolds numbers, and thus laminar flow, the friction factor is can be determined with a linear correlation. Bends within a pipe system produce extra resistance. Besides bends, valves and bellows also give extra resistance in the pipe flow system. The resistance coefficients are added to the total resistance coefficient.

$$\Delta p_{loss} = \sum_{i=1}^{n} \zeta_n \frac{1}{2} \rho v^2 \tag{6.1}$$

$$\zeta_{pipe} = f \frac{L}{d} \tag{6.2}$$

$$\begin{aligned} \frac{1}{\sqrt{f}} &= \frac{1}{-1.8 * log\left[\left(\frac{\epsilon/D}{3.7}\right)^{1.11} + \frac{6.9}{Re}\right]} \quad for: Re > 2300 \\ f &= \frac{64}{Re} \quad for: Re \leq 2300 \end{aligned} \tag{6.3}$$

With: Δp_{loss} are the pressure losses in Pa; ζ resistance coefficient; ρ is the density of the fluid in kg/m^3; v is the velocity of the fluid in m/s; f is the friction factor of the pipe; L is the pipe length in m; D is the pipe diameter in m; ϵ is the pipe roughness factor (0.02 for metal); Re is the Reynolds number.

The pipe could also have a static height difference. When the submarine is horizontal, it is assumed that there is no height difference in the piping. However when the submarine is trimmed there will be a static height difference. The static pressure difference due to the longitudinal trim can be found in equation 6.4.

$$p_{static} = (x_{tankB} - x_{tankA}) * -sin(\theta) \tag{6.4}$$

With: p is the pressure in Pa; x is the position in m; θ is the pitch angle in degrees.

An overview of how this implemented into a model can be found in figure 6.11.

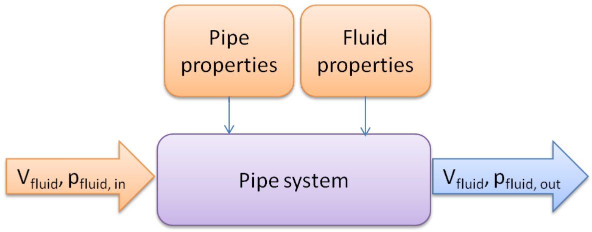


Figure 6.11: Model of a pipe flow system

Noise

A fluid flowing through a pipe produces noise. This is not taken into account because for the pipe flows, a maximum flow is taken into account which is defined in such a way that the noise production of the pipes fulfils the requirements.

6.2.2. Centrifugal pump

It is possible physically model of the centrifugal pump. However a lot of detailed information of the pump is needed. These pump parameters can be obtained after testing or from the manufacturers. Because of the details of the pump construction, manufacturers do not like to give this information. Therefore it was chosen to create an analytical model. The advantage of having an analytic model is that different centrifugal pumps can be plugged in based on basic pump head curves that are available.

Energy

The dimension of a centrifugal pump can be determined when the total pressure difference that the system has to overcome is known. Therefore by having the system behaviour, the pump size can be determined. This must by done first by determine the needed flow capacity. With this flow capacity the pressure losses in the system can be determined. With the flow and the needed pressure to overcome, a pump can be selected.

The pump head curve can be assumed quadratic (Klein Woud and Stapersma, 2016). The best fit for a centrifugal pump can be found in equation 6.5. With actual pump head curves from the manufacturer, an expression can be made of the pump head curve for the specific centrifugal pump. The actual pump head curve is fitted by the Matlab curve fitting tool. Now there is an expression of the pump head curve at nominal speed.

$$\Delta z^{++} = c_3 - c_1 \dot{V}^2 - c_2 \dot{V} \tag{6.5}$$

With: Δz^{++} is the delivered head by the pump in m; $\dot{V}$ is the volume flow in m^3/s.

By decreasing the speed of the pump both the volume flow and the delivered pressure will decrease. The diameter of the pump will stay the same and the pump will only be dependent on the speed. Equation 6.6 and 6.7 give the relations between changing the speed of the pump, the volume flow and the head. These so called pump affinity laws can be used to determine the operation point of the pump at different speeds. The affinity laws are used to modify equation 6.5 into equation 6.8. When knowing the operational point of the pump, the intersection between pump curve and system curve, the pump speed can be determined with equation 6.8. The system curve can be determined with previous subsection. The pump curve is also a quadratic curve. In a graphical display of these two curves, the delivered head and flow by the pump can be red at the intersection point.

$$\frac{\dot{V}_2}{\dot{V}_1} = \frac{n_2}{n_1}\left(\frac{D_2}{D_1}\right)^3 \tag{6.6}$$

$$\frac{H_2}{H_1} = \left(\frac{n_2}{n_1}\right)^2\left(\frac{D_2}{D_1}\right)^2 \tag{6.7}$$

$$\Delta z_{pump} = c_3 * n^2 - c_1 \dot{V}_{pump}^2 - c_2 * \dot{V}_{pump} * n \tag{6.8}$$

With: $\dot{V}$ is the volume flow in m^3/s; n is the pump speed in rpm; D is the impeller diameter in m; z pump is the delivered pressure of the pump in m.

The mechanical power can be determined by equation 6.9. When equation 6.10 is substituted into equation 6.9, an analytic expression for the mechanical power is obtained (Klein Woud and Stapersma, 2016). This expression can be found in equation 6.11. In these equations all losses are taken into account. An overview of these losses can be found in figure 6.12. The advantages of the analytical model is that the losses do not have to be taken into account separately. They are already included in the pump curve.

$$P_m = \frac{P_{ideal}}{\eta_m} = \frac{\dot{V}_{ideal} * \Delta p^{++}_{ideal}}{\eta_m} \tag{6.9}$$

$$\Delta p^{++}_{ideal} = k * \Delta p^{++}_E = k * c_4 * n^2 - c5 * \dot{V}_{ideal} * n \tag{6.10}$$

$$with: \quad \dot{V}_{ideal} = \frac{\dot{V}_{pump}}{\eta_v}$$

$$P_m = c_6 * \dot{V}_{pump} * n^2 - c_7 * \dot{V}^2_{pump} * n \tag{6.11}$$

With: P is the power in W; η_m is the mechanical efficiency; Δp^{++} is the delivered pump pressure in Pa; k is

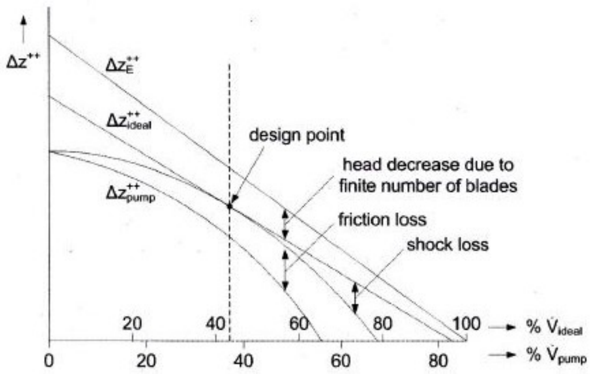


Figure 6.12: Losses inside a centrifugal pump, (Klein Woud and Stapersma, 2016).

An overview of the model can be found in figure 6.13. Still a lot information of the pump itself is needed before the model will work.

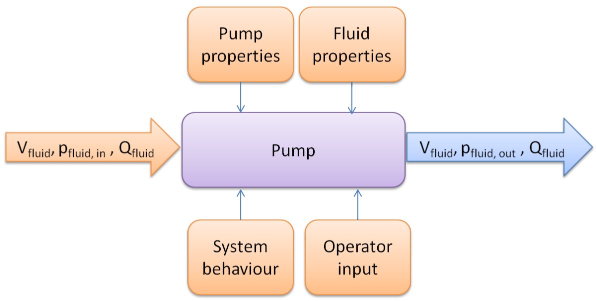


Figure 6.13: Model of a Pump

Noise

The centrifugal pump produces noise when operating. The noise production of the centrifugal pump is modelled with a parametric model to get an indication of the noise production of the centrifugal pump.

The noise of the centrifugal pump can be estimated based on the power of the centrifugal pump (Irwin and Graf, 1979). This approximation is very rough but will give an indication of the noise produced by the pump. Since the centrifugal pump is a flow pump, the noise will not change drastically at different operational points. Also due to the centrifugal pump characteristics, it is not likely that the centrifugal pump will operate far from its best efficiency point. The pump noise is based on the nominal power of the pump. Equation 6.12 give the estimation for the sound power level produced by a centrifugal pump.[1]

$$L_{Wl} = 95 + 10log(P_b) \tag{6.12}$$

With: L_{WL} is the sound power level in dB; P_b is the mechanical power in kW.

For the system that operates with two centrifugal pumps in series, 3 dB can be added to equation 6.12. This because the two pumps will be operated at the same speed. The two pumps will also have the same size and are identical. When having two sources producing the same sound, 3 dB should be added to the total sound level.

The noise produced by the centrifugal pump is mainly emitted to the outside by the casing of the pump (Müller and Möser, 2013). This results mainly in airborne and structure-borne noise. Fluid-borne noise is almost not present in case of a centrifugal pump. More details of noise and how this noise can be reduced can be found in appendix G.

6.2.3. Piston/plunger pump

In the literature the pump modelled is referred to as a piston pump as well as a plunger pump. The difference between these pumps can be found in the pump size. Piston pump is in most cases smaller and more often used in hydraulic systems. The plunger pump is bigger and used in water systems. The working principle of these two pumps is the same.

Energy

In most cases a plunger pump is connected to a constant speed electric motor. The volume flow through the plunger pump is proportional to the speed of the plunger pump in an ideal case. This can be seen in figure 6.14 where the pump curve for a plunger pump at a constant speed is shown. The torque of the plunger pump determines the pressure rise. By adjusting the pump speed, the volume flow is adjusted.

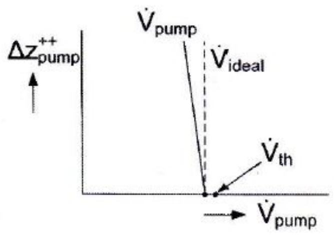


Figure 6.14: Volume flow - head curve for a plunger pump, (Klein Woud and Stapersma, 2016)

The Dorey model, evaluated by Grandall gives a good approximation of all the losses in a piston pump (Grandall, 2010). The Dorey model is based on the Wilson physical piston pump model (Hall, 2014). Dorey took the physical description of Wilson and adjusted it so it could be fitted on all piston pump curves. This makes the model of Dorey an analytical model. When knowing the main characteristics of the pump and the pump curve, the behaviour of the piston pump at different operational points can be determined. Equation 6.13 gives the volume flow of the pump at a certain operation point according to Dorey. The first term in this equation is the ideal volume flow of the pump. The second term takes into account the slip. Not all fluid will be displaced. Therefore there is a little slip in every piston pump which depends on the pressure rise in the pump. The last term takes into account for the compressibility of the fluid. The slip factor C_s^* in equation 6.13 can be taken as a constant. However for a more precise model the slip factor needs to be taken as a variable which depends on the speed of the pump, equation 6.14.

$$\dot{V}_{pump} = \frac{n * V_{stroke} * X}{1000} - C_s^* \frac{1000 * 60 * \Delta p * V_{stroke}}{2\pi * \mu} - \frac{\Delta p * V_{stroke} * \omega)}{10^6 * B} * \left(\eta_{clearance} + \frac{1+X}{2}\right) \tag{6.13}$$

$$C^*_{s,piston} = C_s * \Delta p \left(c_1 + c_2 \frac{n}{n_{max}}\right) \quad with: \quad C_s = \frac{d\dot{V}}{dp} * \frac{\mu}{V_{stroke}} * 60 * 2\pi \tag{6.14}$$

[1]This equation is only used to make a comparison of the systems based on noise, not to obtain a absolute value for the noise production

With: n is pump speed in rpm; V_{stroke} is the stroke volume of the pump; p is the pressure in Mpa; X is the fractional displacement; μ is the dynamic viscosity in Pa/s; B is the Bulk modulus in GPa; C_s is the slip coefficient; $\dot{V}$ is the volume flow in m^3/s.

In equation 6.13 the last term can be neglected, because water is assumed to be incompressible. This assumption implies that the bulk modulus will go to infinity leaving this last term zero. The efficiency of a plunger pump is also different in the range of working points, just like in a centrifugal pump. Figure 6.15 shows the efficiency versus the delivered head by the pump. With equation 6.14 the volumetric efficiency is taken into account, the so called slip losses. When also determining the Hydraulic losses and the mechanical losses, the mechanical power can be determined.

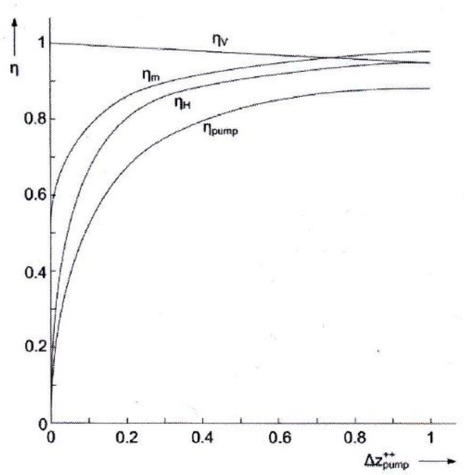


Figure 6.15: Efficiency of a plunger pump

The manufacturer gives the curves of the mechanical power absorbed by the pump in a certain range. By dividing this power by the radial speed of the pump, the torque is obtained. Dorey also gives an analytic model for the torque losses of a piston pump (Grandall, 2010). Equation 6.15 gives the mechanal torque for the pump. The first term in equation 6.15 is the ideal torque without losses. The second term accounts for the viscous friction losses which are related to the speed of the pump. The last term accounts for the Coulomb friction losses which are related to the pressure rise inside the pump. The viscous friction factor C_v^* can analytically solved by equation 6.16. The data of the manufacturer is fitted with Matlab. Coulomb friction factor can be solved with equation 6.17. Grandall gives as advise to neglect the factor $c_7\left(\frac{n}{n_{max}}\right)^2$ since it is very small and a better fit is obtained without this factor. Finally the mechanical power can be obtained by equation 6.18.

$$T_{mechanical} = \frac{\Delta p * D * X}{2\pi} + \frac{C_{vf}^* * \mu * n * D}{60 * 10^6} + \frac{C_{cf}^* * \Delta p * D}{2\pi} \tag{6.15}$$

$$C_{vf}^* = C_{vf}(c_3 + c_4 X) \tag{6.16}$$

$$with: \quad C_{vf} = \frac{dT}{dn}\frac{1}{\mu * D} * 60 * 10^6$$

$$C_{cf}^* = C_{cf}\left(c_5 + c_6\frac{n}{n_{max}} + c_7\left(\frac{n}{n_{max}}\right)^2\right)(c_8 + c_9 X) \tag{6.17}$$

$$with: \quad C_{cf} = \frac{dT}{dp} * \frac{1}{D * 2\pi}$$

$$P = \omega T = n * 2\pi T \tag{6.18}$$

With: n is pump speed in rpm; V_{stroke} is the stroke volume of the pump; p is the pressure in Mpa; X is the fractional displacement; μ is the dynamic viscosity in Pa/s; B is the Bulk modulus in GPa; C_{vf} viscous friction coefficient; C_{cf} is the coulomb friction factor; $\dot{V}$ is the volume flow in m^3/s; T is the torque in Nm;

The model of the plunger pump will work the same as the model for the centrifugal pump, figure 6.13. When the operational working point is known, the pumps speed can be determined. With this pump speed at a certain time, the mechanical power is calculated.

Noise

The plunger pump is known for its excessive noise production. The plunger pumps noise production does depend on the flow and delivered pressure of the pump (Müller and Möser, 2013). The flow of a plunger pump fluctuates over time, there is no constant flow. This flow fluctuation is given by ξ. ξ is the change of the temporal relative volume flow according equation 6.19. ξ for piston pumps is usually between 0.02 and 0.005. The higher this coefficient, the higher the noise production. 0.01 is taken because it is assumed that for modern plunger pumps the noise will be reduced in comparison with older plunger pumps. The sound power level is estimated with equation 6.20.[2]

A large contributor to the total sound power level of the plunger pump is the fluid-borne noise. Fluid-borne noise is not desired for submarine purposes since it can cause detection. Next to the fluid-borne noise also airborne and structure-borne noise are present inside the plunger pump. More about noise can be found in appendix G.

$$\xi = \frac{\dot{V}_t}{\dot{V}_{average}} \tag{6.19}$$

$$L_{WL} = 117 + 10\log\left(\rho c_0 \frac{v^2}{D}\left[\xi + 40\left(\frac{\Delta p}{\rho c_0^2}\right)^2\right]\right) \tag{6.20}$$

With: $\dot{V}$ is the volume flow in m^3/s; L_{WL} is the sound power level in dB; ρ is the density of the pump fluid in kg/m^3; c_0 is the speed of sound in the fluid in m/s (c = 1435 m/s for water); v is the velocity of the fluid of m/s; Δp is the pressure rise of the pump in Pa; D is the pipe diameter in m; ξ is the temporal relative volume flow.

6.2.4. Variable buoyancy/ hydraulic cylinder

An option used for hovering is to adjust the buoyancy of the submarine. This is modelled by moving a tube through the water. By moving the tube up- and downwards the volume of the submarine changes which changes the buoyancy force. The tube is moved by a hydraulic system.

To move a tube through the water, a hydraulic cylinder is needed, fed by a hydraulic system. The hydraulic system consist in most cases of a piston pump which delivers the pressure to the hydraulic system. The hydraulic fluid is transported through pipes to the actuators. A model of a piston pump and a model of the piping are already present. In these models only the fluid properties and the size is changed in order to make the model for the hydraulic system. A model for the hydraulic cylinder and the tube moving through the water must be included to model the variable buoyancy system.

A hydraulic cylinder, figure 6.16, is filled with hydraulic fluid. This causes the piston rod to move. The hydraulic fluid in the opposite chamber in the cylinder is forced out of the cylinder. A multi-stage hydraulic cylinder works according to the same principles, only with multiple stages and can therefore be more extended.

The volume flow is controlled by the piston pump in the hydraulic system. The hydraulic oil can be assumed incompressible. Therefore the pressure delivered to the chamber inside the hydraulic cylinder is equal to the p^{++} delivered by the pump minus the losses in the hydraulic system. There are two chambers inside a hydraulic cylinder. The pressure difference between these two results in the total force delivered by the hydraulic cylinder as can be seen in equation 6.21. This force has to overcome the resistance caused by the (moving) tube and the losses inside the cylinder. This can be found in equation 6.22. The losses and therefore the resistance forces in the hydraulic cylinder are very small and therefore neglected (Bo et al., 2012).

$$F_{cylinder} = p_1 A_1 - p_2 A_2 \tag{6.21}$$

$$F_{cylinder} = F_{R,tube} + F_{R,cylinder} \tag{6.22}$$

[2]This equation is only used to make a comparison of the systems based on noise, not to obtain a absolute value for the noise production

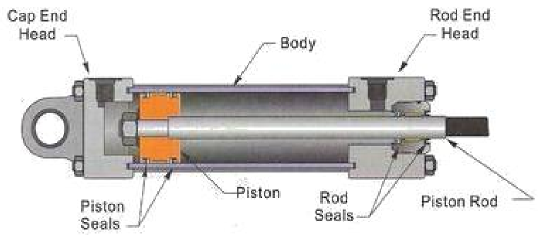


Figure 6.16: Overview of a hydraulic cylinder

With: F is the force in N; p is the pressure in Pa; A is the inner area in m^2.

The tube is moved through the water. Moving this tube through the water cost energy. The force needed to move the tube through the water can be found in equation 6.23. This force is dynamic and depends on the depth and the raising/lowering speed of the the tube.

$$F_{R,tube} = (m+a)\ddot{z} + C_d \frac{1}{2}\rho \dot{z}^2 + A_{tube}\rho g z \tag{6.23}$$

With: m is the mass of the tube in kg; a is the added mass of the tube in kg; C_d is the drag coefficient of the tube; A_{tube} is the top area of the tube; ρ is the density; z is the position of the top of the tube.

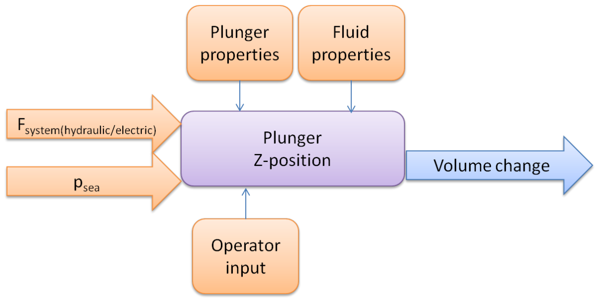


Figure 6.17: Model of a plunger causing buoyancy

6.2.5. Valve

With a valve the flow through the system can be controlled. It is assumed that the valve is controlled ideally. The flow regulated by the valve is equal to the flow demanded. A valve regulates the flow by increasing or decreasing the resistance. By decreasing the pressure drop over the valve, the flow will increase. By increasing the pressure loss over the valve, the flow will decrease. The way a valve is operated and controlled can be seen in figure 6.18. Figure 6.18 shows a butterfly valve. By closing the valve, the open area of the valve decreases and the resistance will increase.

The resistance of the valve at time t is therefore dependent on the desired flow. A valve can be described by its K_v value. The K_v can be assumed linear, depending on the valve opening. When knowing the K_v value, the position of the valve can be determined. In documentation of valves it is common to give the K_{vs} value. This is the K_v value at a pressure difference of 1 bar with the valve fully opened. The K_v can be calculated with equation 6.24. Therefore the pressure drop can be be calculated with the information given by the manufacturer. This can be done with equation 6.25.

$$K_v = \frac{Q}{\sqrt{\Delta p_{valve}}} \tag{6.24}$$

$$\Delta p_{valve} = \frac{Q^2}{K_v^2} \tag{6.25}$$

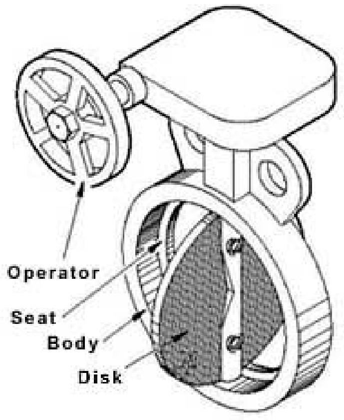


Figure 6.18: Overview of a butterfly valve

With: Δp is the pressure losses over the valve in bar; K_v is the flow coefficient of a valve; Q is the flow in m^3/h.

Noise

The flow through a valve produces noise. An approximation is made of the noise based on the pressure losses and the valve opening. Two situations are considered in which noise can be produced. The first situation is in normal operation when having no cavitation in the valve. The noise can in this case be calculated according to 6.26 (Müller and Möser, 2013). The pressure losses can become very high in the systems with the valve. This can cause cavitation in the fluid. When having cavitation, the noise will be higher. Therefore the noise produced by the valve when having cavitation is given by equation 6.27. These two equation are based on a velocity of the fluid through the valve. This velocity changed by increasing and decreasing the valve opening. The velocity of the fluid flowing through the valve is approximately equal to equation 6.28.[3] In the model there is assumed that cavitation will not occur.

The noise produced by the valve is mainly emitted through the structure and air. However there is a possibility for high fluid-borne noise when the cavitation occurs in the valve, which is not taken into account in this research. For about noise can be found in appendix G

$$L_{WL} = 41 + 10 \log \left(\frac{\frac{\dot{V}}{c} u_{max}^3}{10^{-12}} \right) \tag{6.26}$$

$$L_{WL} = 88 + 10 \log \left(\frac{\frac{\dot{V}}{c} u_{max}^3}{10^{-12}} \right) \tag{6.27}$$

$$With: u_{max} = \sqrt{1.4 \frac{2}{\rho} * (p_1 - p_2)}. \tag{6.28}$$

With: L_{WL} is the overall sound power level in dB; $\dot{V}$ is the volume flow in m^3/s; c is the speed of sound in the fluid in m/s; u_{max} is the maximum velocity of the fluid in the valve in m/s; ρ is the density of the fluid; p is the pressure in Pa.

6.2.6. Tanks

In the ballast trim system are multiple tanks. The distinction between these tanks is in if they are pressurised or at atmospheric pressure, vented. Once again, water is seen as an incompressible fluid. For air, the compressibility is taken into account. The volume of the tank is only important if the tank is pressurised by air.

[3]This equation is only used to make a comparison of the systems based on noise, not to obtain a absolute value for the noise production

The volume of air inside the tank can be calculated according to equation 6.29. The fluid inside the tanks is moved from one to another. Therefore the amount of water inside the tank changes over time according to equation 6.30. The piping of the tank is connected in the bottom of the tank. It is assumed that the inlet and outlet of the tank are the same. By placing the inlet and outlet of the tank in the bottom, the tank can always be emptied. This leads to the expression for the pressure inside the tank according equation 6.31. When the tank is not pressurised the term p_{air} becomes 1 bar, atmospheric pressure, and the pressure only depends on the water level inside the tank. .

$$V_{air} = V_{tank} - V_{water} \tag{6.29}$$

$$V_{water} = V_{water,initial} + \int \dot{V}_{water,system} \tag{6.30}$$

$$p_{tank} = p_{air} + \rho g h_{water} \tag{6.31}$$

With: V is the volume in m^3; p is the pressure in Pa; h_{water} is the water level in m.

The assumption is made that for pre-pressurised tanks, the air will react as an ideal gas, see equation 6.32. Within all the air systems it is assumed that the temperature stays constant which results in equation 6.33.

$$pV = nRT \tag{6.32}$$

$$pV = Constant \tag{6.33}$$

With: p is the pressure in Pa; V is the volume in m^3; n is the amount of gas in mol; R is the gas constant in J K^{-1} mol^{-1}; T is the temperature in K.

When using a pre-pressurised tank, the required energy of emptying the tank depends on the inside pressure. The pressure inside the tank changes over time. When water is expelled from the tank, the volume of the air increases leading to a decrease of the air pressure, equation 6.33. The energy stored in a water particle depends on the pressure of the particle and its volume. By expelling water out of the tank, the energy in the water particles is also expelled. The loss of the energy is equal to the losses in the pipe flow. The pressure due to the water level in the tank is not taken into account to determine the energy that is lost by expelling water. Over time the water level in the pressurised tank decreases with the flow. The water level in the other tank increases leading to an increase of stored energy in the other tank. Therefore the total static pressure in the remains the same over time. The pressure that is lost is the pressure due to the air pressure inside the tank. Now the loss of energy can be determined by equation 6.34. For the value of p_{air}, a average value must be taken. The simulations will not run until the tanks are at their minimum water level. The air pressure, and therefore the loss, is at $t = 0$ higher than when the tank is at its minimum water level. Therefore it is not realistic to use the air pressure at time t. The average air pressure can be calculated with equation 6.35. When calculating the total mechanical power needed, the mechanical losses of the compressor must be taken into account and the losses due to venting a pressurised tank. This can be found in equation 6.36. A graphical display of the stored energy in a water particle can be found in figure 6.19.

$$P_{loss,pipe} = p_{air} * \dot{V}_{water} \tag{6.34}$$

$$p_{air,av} = \frac{p_{air,max} - p_{air,min}}{2} \tag{6.35}$$

$$P_{mechanical} = \frac{P_{loss,pipe}}{\eta_{compressor}\eta_{venting}} = \frac{p_{air} * \dot{V}}{\eta_{compressor}\eta_{venting}} \tag{6.36}$$

The compressor efficiency is estimated based on compressor data. Before storage, the air is compressed. This process is assumed to be isothermal. For high pressure ratios the isothermal efficiency goes to 0.84 (Van Buijtenen and Visser, 2007). When the air expands from the air bottles in to the system an isothermal efficiency of 0.94 can be found for high pressure ratios. These two efficiencies give the total efficiency of the compression and expansion of air. 0.79 is the resulting efficiency and is called the compressor efficiency, $\eta_{\text{compressor}}$.

The pressurised tanks needs to be pressurised with an initial pressure in an initial volume. The volume of air inside the tank can be determined when assuming that the volume of trapped air inside the tank is equal

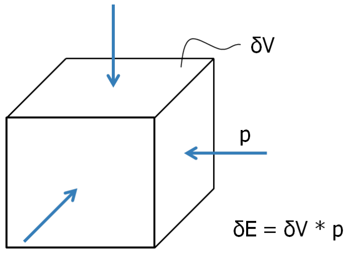


Figure 6.19: Stored energy in an infinite small water particle

to the minimum water volume inside the tank. First the initial pressure can be calculated with equation 6.31. When the expansion volume is written in terms of the initial air volume with equation 6.37, the initial air pressure can be determined with equation 6.38 when knowing the initial air volume and the minimum pressure needed. Due to the valve, which regulates the flow through the system, there is a loss. Assuming that the minimum pressure is needed when the valve is fully open, the valve efficiency is 1 when the average pressure over time is equal to the minimum pressure. This results in a vented tank. The average air pressure in the tank can be calculated with equation 6.39. When having these pressures the efficiency of the valve assuming fully open for the minimal pressure is calculated with equation 6.40. The efficiency for venting can be calculated with equation 6.41. This efficiency takes into account the losses because of the loss of trapped air when venting. These two efficiencies together give the efficiency of the pre-pressurised tank according to equation 6.42. These efficiencies are plotted and there can be found a relation between the initial air volume and efficiency, figure 6.20. In figure 6.20 it can be seen that when increasing the initial air volume the efficiency for venting decreases while the efficiency of the valve increases. The optimal combination results in an initial air volume of 29%. In that case it becomes more clear that the initial air pressure must be 2.41 times higher than the minimum pressure.

$$V_{air,expansion} = 1 - 2 * V_{air,initial} \tag{6.37}$$

$$p_{air,initial} = \frac{V_{air,expansion}}{V_{air,initial}} * p_{air,minimal} + p_{air,minimal} \tag{6.38}$$

$$p_{air,average} = \frac{p_{air,initial} - p_{air,minimal}}{2} + p_{air,minimal} \tag{6.39}$$

$$\eta_{valve} = p_{min} / p_{air,average} \tag{6.40}$$

$$\eta_{vent} = 1 - \frac{p_{air,minimum}}{p_{air,initial}} \tag{6.41}$$

$$\eta_{tank} = \eta_{vent} * \eta_{valve} \tag{6.42}$$

With: V is the volume in m^3; p is the pressure in Pa; η is the efficiency.

An overview of the model can be found in figure 6.21

6.2.7. Submarine

In chapter 2 the movement of the submarine has already been explained. For the understanding of this subsection, the most important characteristics will shortly be explained. In figure 6.22, the used coordinate system is shown.

The equations 2.4, 2.5 and 2.6 consist of parameters that can be determined from the 3D-model and other parameters that can be calculated assuming the submarine is an ellipsoid. The weight of the submarine and the mass moments of inertia around the x-axis and y-axis can be obtained from the rhino model. These values can be found in table 6.4.

In equation 2.4, the factor a_{33} is the added mass for the heave motion. This added mass is the extra weight of the water that is moved when moving the submarine in z-direction. For the assumption that the submarine is an ellipsoid, the added mass can be determined without any tests. The same accounts for a_{55} in equation 2.6. A drawing of a ellipsoid with its parameters can be found in figure 6.23. The added mass coefficients are determined by using the Lambs' coefficients (Lamb, 1945). Lamb studied the added mass coefficients of an ellipsoid and gave an mathematical expression of these coefficients based on the form factors of the ellipsoid.

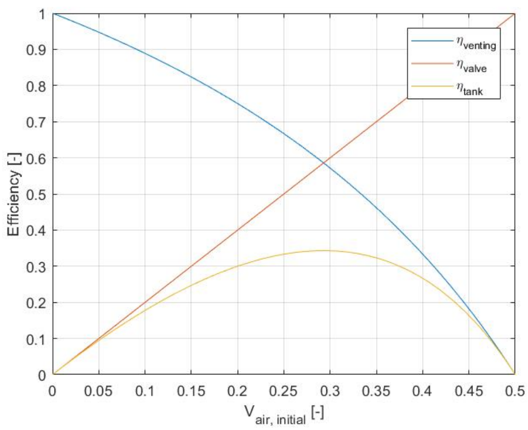


Figure 6.20: Efficiency of pre-pressurised tank

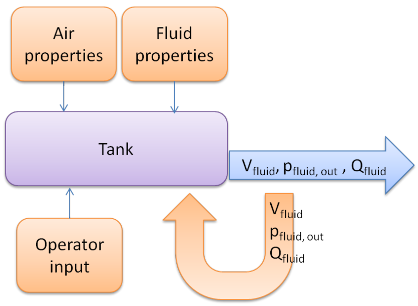


Figure 6.21: Model of the tanks inside the submarine

These factors are recently verified with computational fluid dynamics, CFD, and measurements. For the pure heave motion and the pure pitch motion the approximation of Lamb is accurate enough (Lee et al., 2011). The Lamb added mass coefficients are determined based upon a ellipsoid with the sizes of the BB2 submarine. The Lamb's factors can only be determined for a prolate ellipsoid, $b = c$ and $a > b$. For this research only the added mass for the heave motion and pitch motion are important. The added mass coefficient for the roll motion of an ellipsoid is zero and therefore not taken into account. This gives a small error in the roll motion because the submarine is not a perfect round ellipsoid and in reality the mass moment of inertia around the x-axes of the submarine will be present.

To calculate the added mass coefficients k_2 and k', equations 6.43 and 6.44, the eccentricity of the ellipsoid and the coefficients α_0 and β_0, which describe the proportion of the ellipsoid, are needed. The eccentricity of the ellipsoid can be determined with equation 6.45 in which a is half the length of the ellipsoid and b the radius. α_0 and β_0 can then be calculated knowing the eccentricity with equations 6.46 and 6.47. Figure 6.24 shows the relation between the b/a and the Lamb's k-factors. Since the k-factors are based upon the relation b/a, the k-factor for the BB2 hull are the same despite the difference in the size of the submarine 1 and 2. For the BB2 hull it is found that $k_2 = 0.94$ and $k' = 0.82$. Finally the added mass coefficients can be determined with equations 6.48.

$$k_2 = \frac{\beta_0}{2-\beta_0} \tag{6.43}$$

$$k' = \frac{e^4(\beta_0-\alpha_0)}{(2-e^2)\left[2e^2-(2-e^2)(\beta_0-\alpha_0)\right]} \tag{6.44}$$

$$e = \sqrt{1-(b/a)^2} \tag{6.45}$$

$$\alpha_0 = \frac{2(1-e^2)}{e^2}\left(\frac{1}{2}ln\left(\frac{1+e}{1-e}\right)-e\right) \tag{6.46}$$

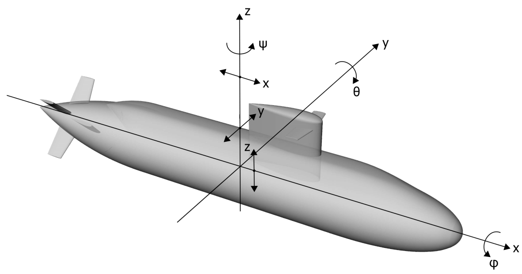


Figure 6.22: Coordinate system

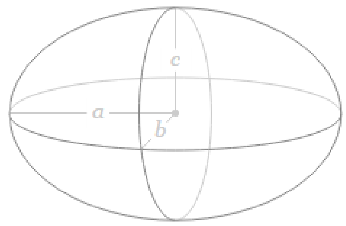


Figure 6.23: Ellipsoid with used parameters

$$\beta_0 = \frac{1}{e^2} - \frac{1-e^2}{2e^2} ln\left(\frac{1+e}{1-e}\right) \tag{6.47}$$

$$a_{33} = k_2 * m \quad \text{and} \quad a_{55} = k' * I_{yy} \tag{6.48}$$

With: k is the Lambs' factor; β and α are proportion coefficients; e is the eccentricity coefficient; a_{33} is the added mass for heave in kg; a_{55} is the added mass for pitch in kgm^3.

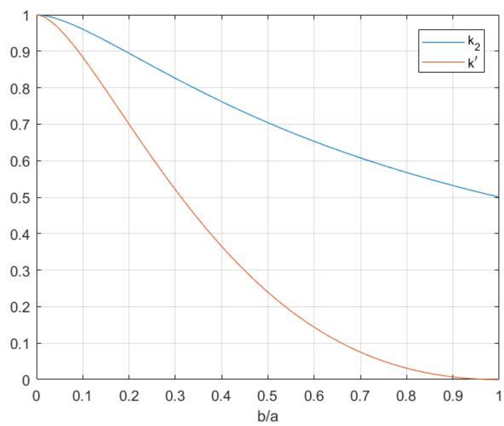


Figure 6.24: Lamb's k-factors

In equation 2.4 there is also the coefficient b_{33}. This coefficient depends on the speed of the heave motion. This term stands for the resistance that is usually described by $F_d = 0.5 C_d A_w \rho v^2$. Therefore b_{33} is determined by equation 6.49. C_d is the form factor. The submarine is taken as a long ellipsoid. This gives that C_d can be taken as 0.5(White, 2008). The flow can be assumed laminar around the submarine since it is not expected or desired to have high velocities, and thus turbulent flow, in z-direction.

$$b_{33} = C_d A_w \frac{1}{2} \rho \tag{6.49}$$

The coefficient b_{44} relates to the drag of a rolling submarine. The drag of a rolling submarine is described as only the friction of a rolling ellipsoid. The average velocity of the flow passing the ellipsoid when rolling can be determined in the centre of area of half the ellipsoid. At $\overline{x}$, equation 6.50, the flow will pass the ellipsoid with the average velocity. To use this in the final relation of the motion of the submarine, this velocity needs to be written as function of the radial velocity around the x-axis. The radius can be determined by the ellipse equation, equation 6.51. Now b_{44} can be calculated with equation 6.52. The friction coefficient, c_f,that will be used is the friction coefficient of a flat plate assuming a laminar flow, 0.00493 (White, 2008). A rolling submarine submarine will result in a new equilibrium when having the trim system with pre-pressurised tanks. When the starboard side is getting heavier, the submarine will roll in the starboard direction. This

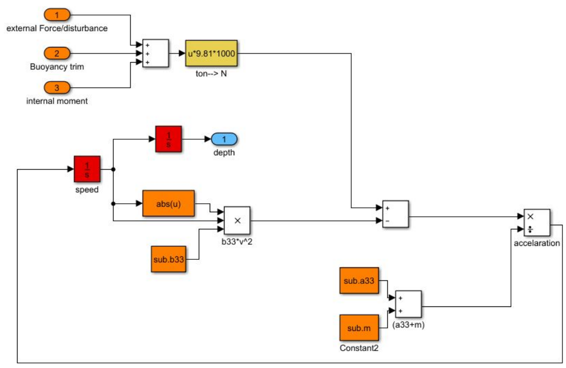


Figure 6.25: Simulink model of the heave motion

results in that the moment caused by the starboard tank is getting less and the moment caused by the port side tank is getting larger due to the position of the tanks. Therefore a new equilibrium will be found and the rolling motion is dampened.

$$\overline{x} = \frac{3}{8}a \tag{6.50}$$

$$\left(\frac{\overline{x}}{a}\right)^2 + \left(\frac{\overline{y}}{b}\right)^2 = 1 \tag{6.51}$$

$$b_{44} = C_f A_w \frac{1}{2}\rho\overline{y}^2 \tag{6.52}$$

With: $\overline{x}$ is the distance to the centre of area in m; $\overline{y}$ is the radius of the ellipsoid at the centre of area in m; a is half the length of the ellipsoid in m; b is the radius of the ellipsoid in m; b_{44} is the roll friction factor in kg m^2; C_f is the friction factor of the ellipsoid; A_w is the wetted area in m^2; ρ is the density in kg/m^3.

The coefficient b_{55} can be determined in almost the same way as b_{33}. The velocity at any point of the ellipsoid can be described as $v = \omega x$. The average velocity of the surface of the ellipsoid can be found in the centre of area, equation 6.50. The resistance due to trimming will work in this point. Since the ellipsoid is symmetrical the arm can be found in $-\overline{x}$ and $\overline{x}$. The drag coefficient will remain the same as in b_{33}. this will result in equation 6.53.

$$b_{55} = C_d A_f \frac{1}{2}\rho\overline{x}^3 \tag{6.53}$$

With all the motions coefficients of the submarine, a model is made which describes the movements of the submarine. The coefficients for the motions of the submarine can be found in table 6.4. The model of the submarine should have as an input the forces and moments on the submarine. The output of the ships model is the position. The simulink model that is made for the heave motion can be found in figure 6.25.

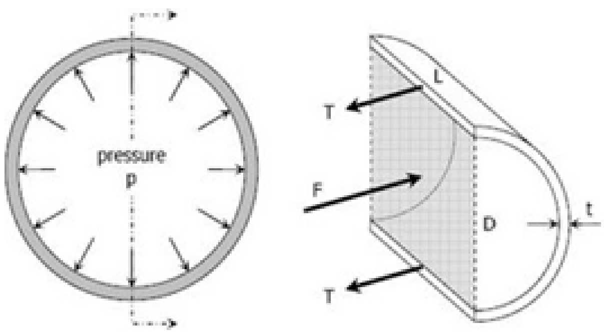


Figure 6.26: Internal stresses due to pressure

Next to having the models for the movement of the submarine, the submarine will also be compressed when diving deeper. The compression of the submarine will lead to a smaller buoyancy force. The model this effect, the submarine is again assumed to be an ellipsoid. This way the effects of compression can be taken into account. The compression of the submarine can be determined by cylindrical pressure vessel laws (Hibbeler, 2011). Usually these laws are used in combination with an internal pressure, see figure 6.26. The principle can of course be used in the opposite way. Important parameters are the thickness of the hull and the E-modulus of the used steel. The thickness for the hull is for now assumed to be 5.5 cm (NZHR, 1979). A common steel to use for submarines is the high strength steel HY-100. This steel has an elasticity modulus, E, of 205 GPa (Matweb). First the stress inside the steel plates can be calculated with equation 6.54. With this stress due to the underwater pressure, the elongation of the round plate can be determined with equation 6.55. Than the new radius of the submarine hull follows according 6.56. Finally the buoyancy force can be calculated with equation 6.57. The result of the change in buoyancy force for a 100 meter dive can be found in figure 6.27.

$$\sigma_1 = \frac{pr}{t} \tag{6.54}$$

$$\epsilon = \frac{\sigma_1}{E} \tag{6.55}$$

$$r_{compressed} = r - \frac{\epsilon}{2\pi} \tag{6.56}$$

$$F_{buoyancy} = \int_{x=0}^{L} \pi r_{compressed}^2 \tag{6.57}$$

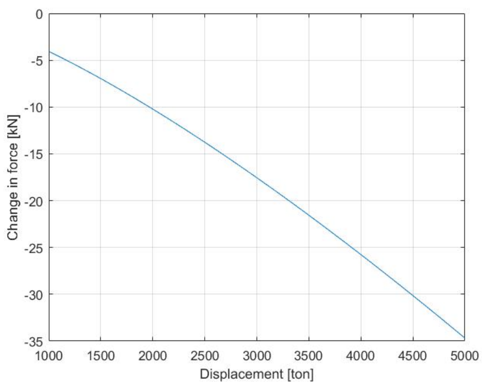


Figure 6.27: Change in buoyancy 100 meter dive

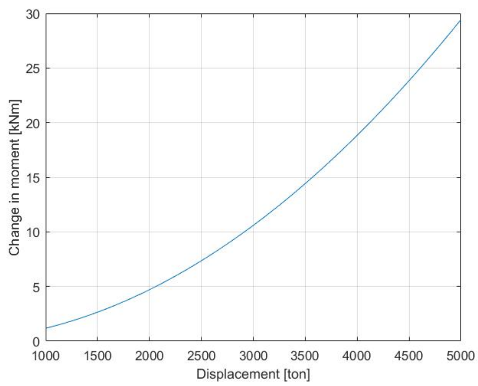


Figure 6.28: Trimming moment due to compression for ϑ= 10°

With: σ_1 is the stress in Pa; p is the outside water pressure; r is the radius of the submarine; t is the plate thickness; E is the E-modulus in Pa; ϵ is the elasticity; F is the force in N.

For a trim angle, the compression of the submarine also results in a small trimming moment. This moment is taken into account the same way as the anechoic tiles in chapter 5. In this case the trimming moment will follow when finding the buoyancy force at the front and the buoyancy force at the aft of the submarine. The extra trimming moment due to the compression of the submarines hull when having a trim angle of 10° can be found in figure 6.28

$$M_{compression} = dF_{buoyancy,for} * LCB_{for} + dF_{buoyancy,aft} * LCB_{aft} \tag{6.58}$$

When this is compared to the effects of the anechoic tiles, it can be seen that the effect of the anechoic tiles is not as large as the compressibility of the submarine itself. The effect of the anechoic tiles can almost be neglected compared to the compressibility of the hull of the submarine.

Table 6.4: Submarine motion coefficients

Characteristics	**Symbol**	**Order**	**Submarine 1**	**Submarine 2**
Mass	m	[kg]	4,000E+06	2,000E+06
Moment of inertia	I_{xx}	[kg m^2]	5,700E+07	1,795E+07
Moment of inertia	I_{yy}	[kg m^2]	1,204E+09	3,793E+08
Lambs' factor heave	k_2	[-]	0,94	0,94
Lambs' factor pitch	k'	[-]	0,82	0,82
Added mass heave	a_{33}	[kg]	3,760E+06	1,880E+06
Added mass roll	a_{44}	[kg m^2]	0	0
Added mass pitch	a_{55}	[kg m^2]	9,873E+08	3,110E+08
Damping heave	b_{33}	[kg /s]	1,356E+05	5,383E+04
Damping roll	b_{44}	[kg m^2/s]	4,831E+05	1,917E+05
Damping pitch	b_{55}	[kg m^2/s]	3,093E+08	1,227E+08

6.3. Controller

The submarine is able to measure its depth, roll and pitch angle. Based on these measurements the controller gives an output. This output is a state which is fed in the ballast trim system. For example, the controller gives as input for the system that it must create a total down force of 2 kN. The compensation tank gives at that given time a total down force of 1.8 kN. This means that the system should take in extra weight to fulfil the controller's request to deliver a down force of 2 kN, which is 0.2 kN.

Simulink has an implemented PID auto tune function. This function is used to design the PID controllers. This controller gives a state order to the system. This state gives for example the needed water level of the tank. The input for the controller is the out of depth measurement or the out of trim measurement. The submarine will measure its depth and trim angle. These values are compared to the desired depth and trim angle. The error between these values is used as input for the PID controller.

It is assumed that the pumps and valves come with a proper controller. Therefore the needed flow is considered to be equal to the desired flow. An overview of the made model can be found in figure 6.29.

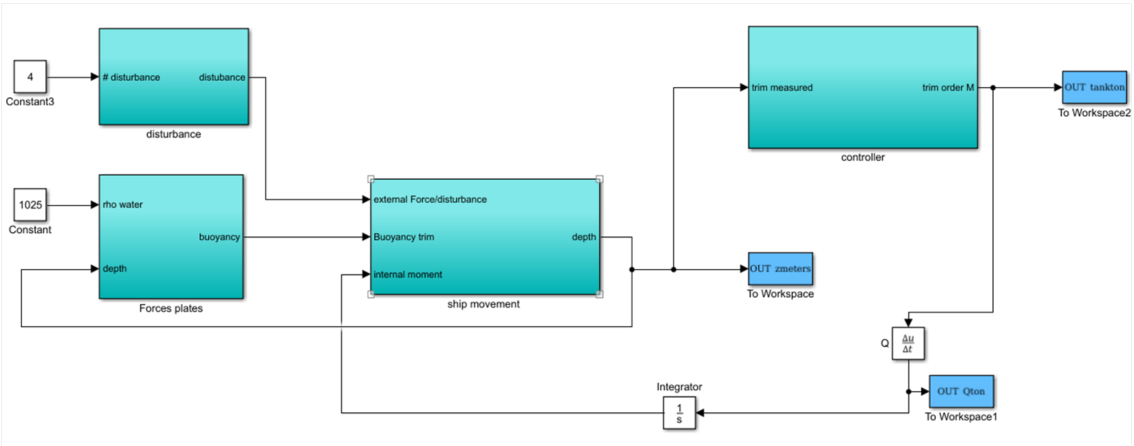


Figure 6.29: Model overview for tuning the systems

7

Comparison of the different options

To see which system is the best suitable solution for a modern SSK submarine, two different sizes of SSK submarines are studied. This research is based on the BB2 Joubert hull form, a 4000 ton submarine. This can be scaled down to see how the performance of the systems on board a submarine changes by changing the size of the submarine. Therefore two case studies have been carried out, one with the 4000 ton hull and one hull that is scaled down to 2000 ton. The studied disturbances as determined in chapter 5 are used to see the movement of the submarine. With these disturbances the performance of the ballast trim system, as determined in chapter 6, is studied. The different systems will be compared in:

- energy consumption
- capability
- redundancy
- noise production

The energy consumed by the systems can be obtained from the models. The models give also an answer to whether the system is capable of operating in the specified operation window.

Redundancy is compared afterwards. There is investigated to which extend the systems can be replaced by another system. This answers the question of how redundant the complete ballast trim system is.

In chapter 5 the disturbances to which the submarine is subjected are studied. The first order effects of waves were also given in this chapter. The statement was made that the first order effects can not be compensated. This was based on the knowledge of earlier designs of submarines. There will be first given prove of this statement.

7.1. First order effects of waves

The assumption was made that the first order waves can not be compensated. This can be shown with the model for the submarine and the disturbances due to the waves. The movement of the submarine due to the waves is relatively small. The forces caused by the waves are very large but, due to the fast fluctuation of the force, the resulting movement is relatively small. This movement can be reduced by fast compensation for the first order effects of the waves. This will lead to a system that must be able to loose weight or take in weight very fast. For sea state 5, the highest disturbance was found for waves with a wave number of $k = 0.365$ for submarine 1 and $k = 0.268$ for submarine 2. Sea state 5 was also said to be the highest sea state for which the submarine has to be fully operational. The movement due to these waves for submarine 1 and submarine 2 can be found in figure 7.1 and 7.2.

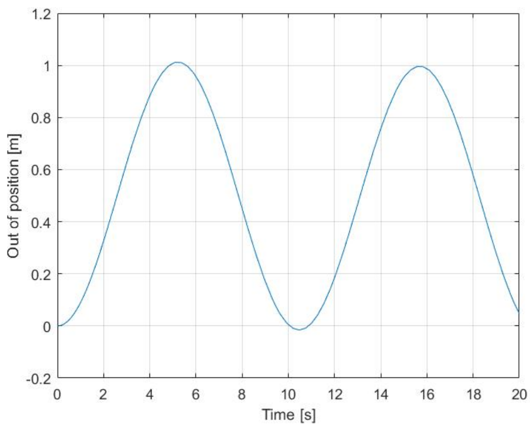


Figure 7.1: Heave motion submarine 1 due to waves

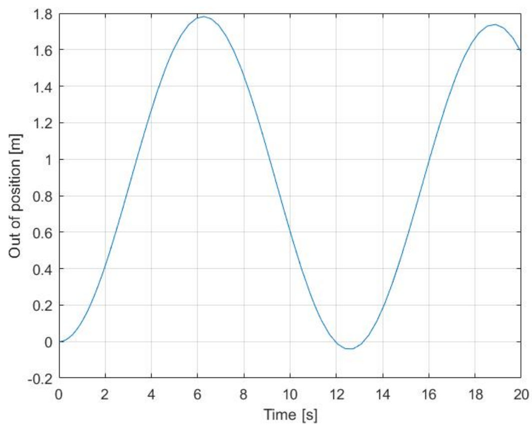


Figure 7.2: Heave motion submarine 2 due to waves

When trying to reduce this heave motion, a high flow is needed while the effect remains low. The effect of compensating the movements due to the waves for submarine 1 and submarine 2 can be found in figure 7.3 and 7.4. To obtain these results, there is a need for a large flow of the compensation system shown in figure 7.5 and 7.6. The needed capacity of the system which follows is not realistic. Therefore the first order effect of waves can not be compensated by the ballast trim system.

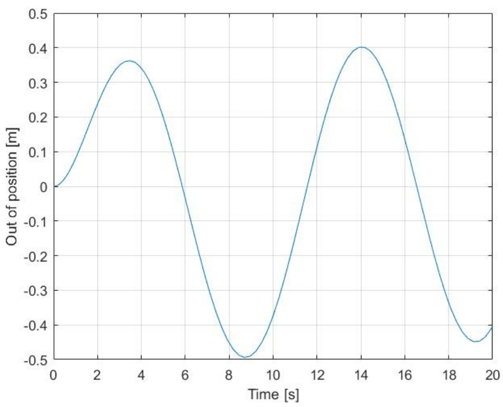


Figure 7.3: Heave motion submarine 1 due to waves when compensated

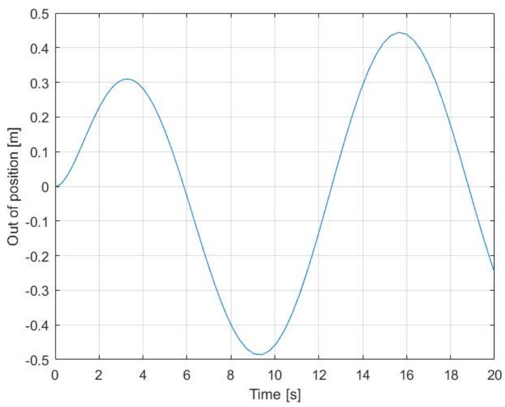


Figure 7.4: Heave motion submarine 2 due to waves when compensated

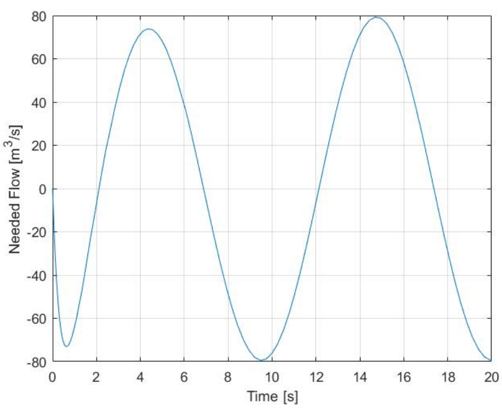


Figure 7.5: Needed flow for compensation submarine 1

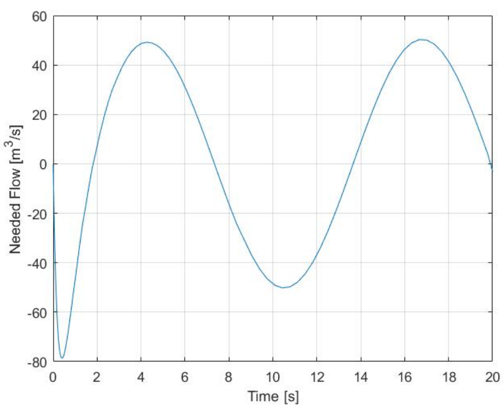


Figure 7.6: Needed flow for compensation submarine 2

Table 7.1: Absolute flows needed for trim, submarine 1

	$\theta_{0.1}$	$\theta_{0.5}$	$\theta_{1.0}$	$\theta_{1.5}$	$\theta_{2.0}$
Disembarking SF	0.060	0.060	0.060	0.060	0.060
Anchoring	0.121	0.051	0.035	0.028	0.025
Transit	0.010	0.008	0.006	0.005	0.004
Disembarking AUV	0.018	0.017	0.017	0.017	0.017

7.2. Dimensions of the systems

Different maximum errors are studied to see what this means for the system. To see the implications of the different maximum overshoots, the signal from the PID controller which gives a certain state for the ballast system, is directly fed into the model of the submarine. For the three different modes there are taken maximum out of position, assuming that this directly leads different flows for the system.
The model used to see what the effect of changing PID values is can be found in figure 6.29 for the compensation system. This model lay-out is also valid for the two other modes.

7.2.1. Trim system

By setting a maximum overshoot of 0.1°, 0.5°, 1.0°, 1.5°and 2.0°, The trim system can be dimensioned. There is looked how the maximum overshoot influences the trim system. The maximum overshoot depends on the PID controller output, so by tuning the controller, the maximum overshoot can be controlled. The PID controller also gives a flow order. Therefore the different maximum overshoots can result in different system capacities.

The absolute values for the needed flow for the trim system of submarine 1 can be found in table 7.1. The moments during transit on AIP and transit while snorkeling are the same. Therefore there is looked into transit in general. What is interesting to see is that for the case of disembarking SF, the maximum needed flow for the trim system does not change. Also in transit mode the maximum needed flow does not change much. However for the case of anchoring, the maximum needed capacity of the system does change. This gives that only for sudden disturbances a higher capacity of the system, results in a better trim keeping. Figure 7.8 shows these maximum trim angles during the disembarking of SF for submarine 1. Figure 7.7 shows the needed flow over time for this operation for submarine 1. The profiles for disembarking SF, disembarking an AUV and Achoring for submarine 1 and 2 are shown in appendix H.3.2.

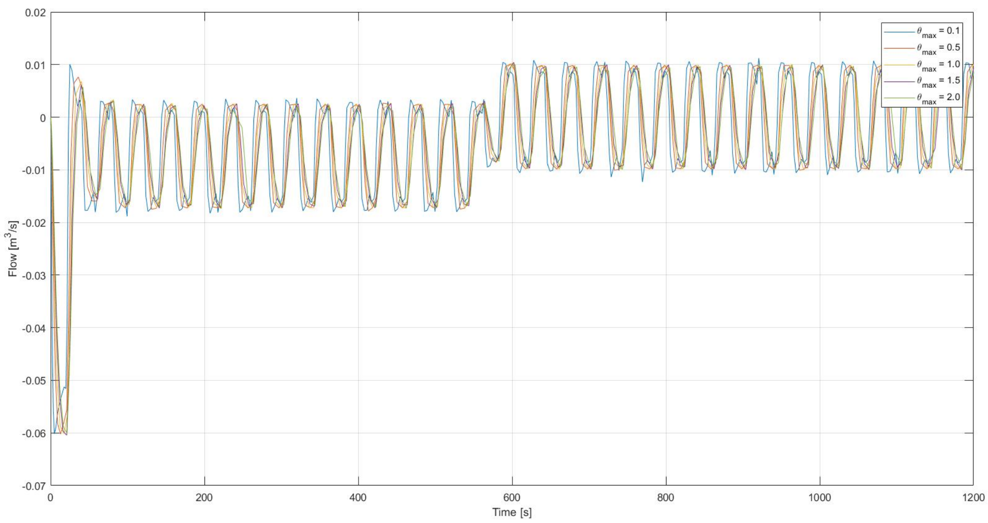


Figure 7.7: Needed flow for trimming while disembarking SF

Disembarking of SF is appointed to be the most critical mission for which longitudinal trim must held. This because not keeping trim when disembarking people is dangerous. While debarking SF the needed flow for the trim system is 0.060 m^3/s according to table 7.1. Only for a small amount of time this flow is

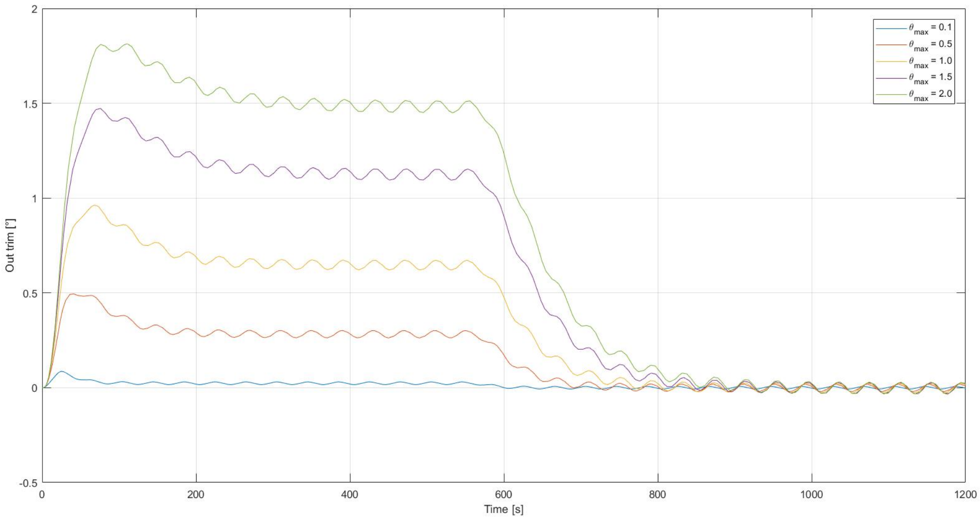


Figure 7.8: Pitch motion during disembarking SF

needed, and therefore it is expected that with a flow of 0.060 m^3/s better results can be obtained. This will result in a trim angle which is larger than 2.0° while anchoring. This does not seems to be an problem since the submarine is not vulnerable during anchoring and therefore larger trim angles can be allowed. During transit, this flow will result in a very small trim angle.

For submarine 2 the maximum flow is set to 0.079 m^3/s for the same reasons as for submarine 1. The absolute needed flows for the different profiles can be found in table 7.1. With this maximum flow the trim angle while disembarking SF can be around 0.1° when tuned properly. The trim angle while anchoring can become very high with this maximum flow. However, as stated before, the trim angle during anchoring is not a problem. The trim angle when in transit will remain very low all the time.

When knowing the maximum flow parameters, the next step is to look what impact this maximum flow has on the maximum trim angle. The maximum flow that is determined is added to the model. Than the PID controller is tuned in such a way that it will give the lowest trim angle in all cases. The results can be found in figure 7.8 for the trim during disembarking SF from submarine 1. The profiles for disembarking SF, Anchoring and disembarking an AUV can be found in appendixidsystem. Next to a as low as possible trim angle, it must also deliver a steady flow demand to the actuator. Fast fluctuation in the flow order are not desired.

To choose the right actuator, the head that needs to be delivered is important. This head can be determined by letting the earlier determined flow through the system and see what the pressure losses are when transported between the tanks. When knowing this pressure losses the needed system can be determined in terms of needed flow and head. The dynamic pressure losses depend on the piping. The static head losses depend on situation of the submarine. To obtain the static head the system has to overcome, the trim angle is set at 3°. The maximum tank height in the model is 2.7 meters. This 2.7 meters is also taken into account as maximum head difference due to the tank levels in submarine 1. For submarine 2 this result in 2.14 meters. For the case of having pre-pressurised tanks, the minimum water level was set at 29% of the total tank volume. This results in a static pressure difference of 42% of the volume of the trim tank. This means a water level difference of 1.13 meters for submarine 1 and 0.90 meters for submarine 2. The pressure losses at the maximum flow for submarine 1 and submarine 2 can be found in table 7.2.

For the trim system there are two different systems that will be studied. The first system uses a centrifugal pump to displace the fluid between the tanks. The centrifugal pump is selected from the brochures of KSB. The centrifugal pump that fulfils these requirements is the KSB Etanorm 150-125-200 at 1450 rpm (KSB, a). The pump characteristics can be found in figure 7.10. The whole product sheet of this pump can be found in appendix I.

The chosen centrifugal pump for submarine 2 is also the KSB Etanorm 150-125-200 with 1450 rpm. This centrifugal pump is able to fulfil the requirements of having a flow of 0.060 m^3/s and a head of 11.7 meters. The modelled pump characteristics can be found in figure 7.10.

For the model it is assumed that the pressurised tank is filled with water for 71%. This is in detail explained

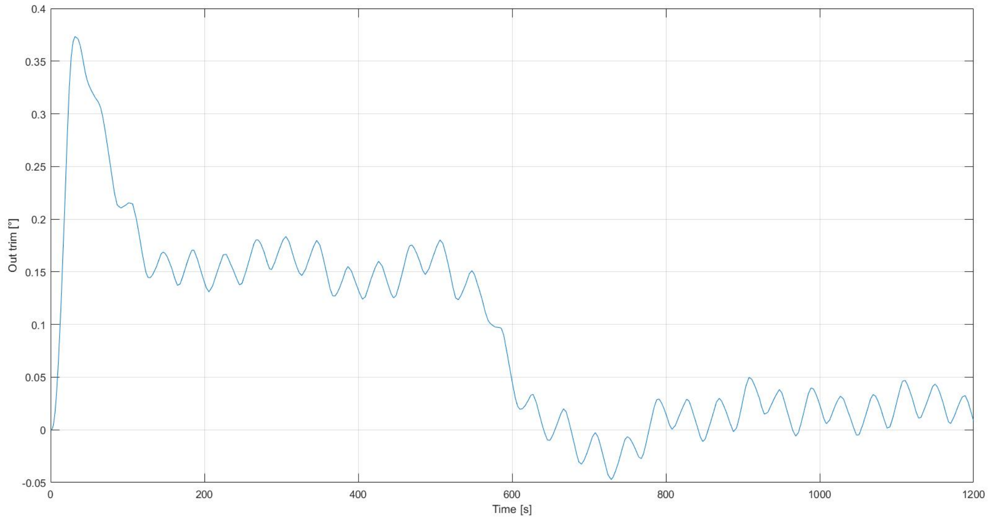


Figure 7.9: Pitch motion during disembarking SF, tuned trim system submarine 1

Table 7.2: Pressure losses in systems at maximum flow

		Pre-pressurised tanks	Centrifugal pump
Submarine 1 $\dot{V} = 0.060$	Pipe losses [bar]	0.49	0.73
	δp tanks [bar]	0.11	0.27
	Total losses [bar]	**0.60**	**1.00**
Submarine 2 $\dot{V} = 0.079$	Pipe losses [bar]	0.48	0.80
	δp tanks [bar]	0.09	0.22
	Total losses [bar]	**0.57**	**1.02**

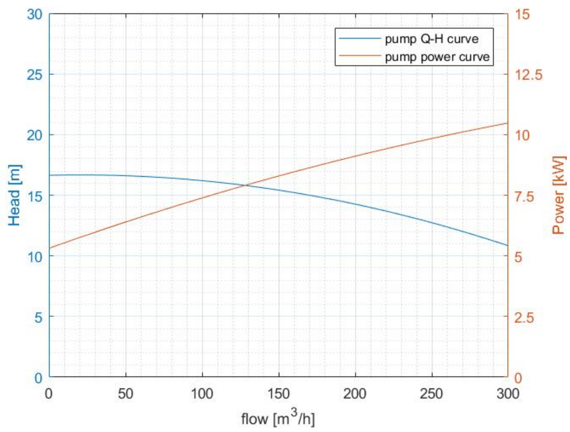


Figure 7.10: Pump characteristics of modelled KSB Etanorm 150-125-200 1450 RPM centrifugal pump

in chapter 6. For the initial pressure inside the tank there is need of a minimum pressure. That thee initial pressure is equals 2.41 times the minimum pressure. This results in an initial air pressure of 3.86 bar for submarine 1 and 3.78 bar for submarine 2 when the vented tanks are at atmospheric pressure. The average pressure loss equals than 2.73 bar for submarine 1 and 2.68 bar for submarine 2.

7.2.2. Compensation system

The maximum overshoot is set at 0.1, 0.5, 1.0, 1.5 or 2.0 meters. The maximum overshoot depends on the PID controller output. Determining the right capacity of the actuator is done the same as for the trim system

What can be seen from the results is that when the tolerances for the overshoot are tighten, the compensation system has to deliver larger forces, resulting in a higher flow of ballast water in and out of the submarine. The absolute maximum flows for the different operational profiles can be found in table 7.3 for submarine 1 and in table 7.4 for submarine 2. For submarine 1, figure 7.12 shows the heave motion of the submarine during the debarking of SF. Figure 7.11 shows the needed flow over time for this operation.

Table 7.3: Absolute flows needed for compensating, submarine 1

	$z_{0.1}$	$z_{0.5}$	$z_{1.0}$	$z_{1.5}$	$z_{2.0}$
Dropping off SF	0.160	0.119	0.097	0.088	0.083
Anchoring	0.237	0.103	0.075	0.059	0.052
Transit AIP	0.102	0.044	0.032	0.027	0.024
Transit snorkeling	1.205	0.961	0.796	0.714	0.654

Table 7.4: Absolute flows needed for compensating, submarine 2

	$z_{0.1}$	$z_{0.5}$	$z_{1.0}$	$z_{1.5}$	$z_{2.0}$
Dropping off SF	0.191	0.146	0.115	0.098	0.087
Anchoring	0.070	0.070	0.048	0.040	0.035
Transit AIP	0.150	0.042	0.023	0.016	0.012
Transit snorkeling	1.323	1.152	0.988	0.863	0.782

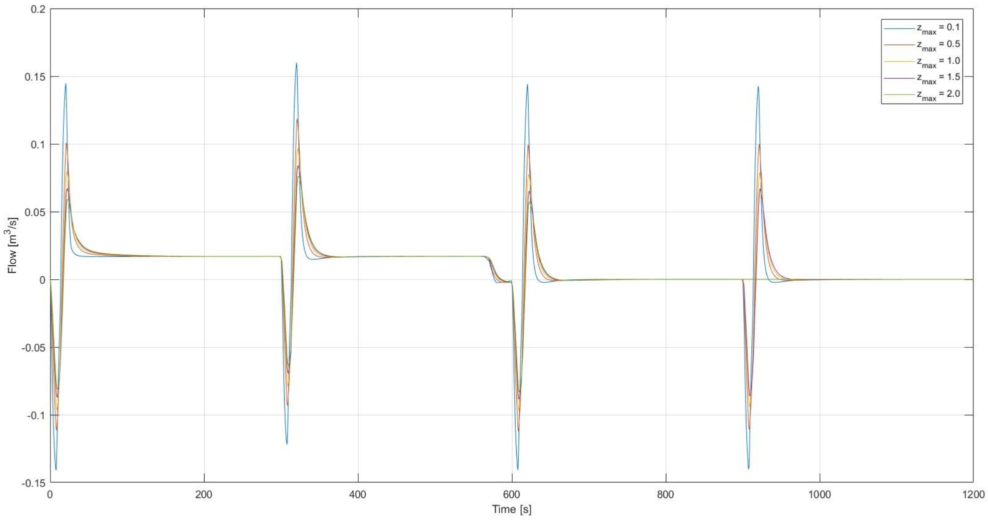


Figure 7.11: Needed flow for compensating while debarking SF, submarine 1

What can be seen in figure 7.11 is that only shortly the maximum flow is needed. This is because for now there is no restriction in the maximal delivered flow. For a real case there is a maximal flow that can be delivered due to the system capacity. In figure 7.12 there can be seen that the maximum overshoot is not equal to the maximum overshoot restriction. This is because the controller, that is used, is the automatic PID controller is Simulink. In some cases loosen the controller with one step resulted in a to high overshoot. That can be seen in for example the line $z_{2.0}$. The maximum overshoot was set to 2 meters. However the settings for a overshoot of 1.8 meters were the first settings found for which the overshoot is less than 2 meters. The final out of position during disembarking SF can be found in figure 7.13. Other profiles for submarine 1 and 2 can be found in appendix H.3.2.

What can be seen in table 7.3 and table 7.4 is that in almost all cases the compensation system for submarine 2 must deliver a higher flow than the compensation system for submarine 1. This is a result of the disturbances that are put on the submarines. The disturbances for submarine 2 are in almost all cases equal

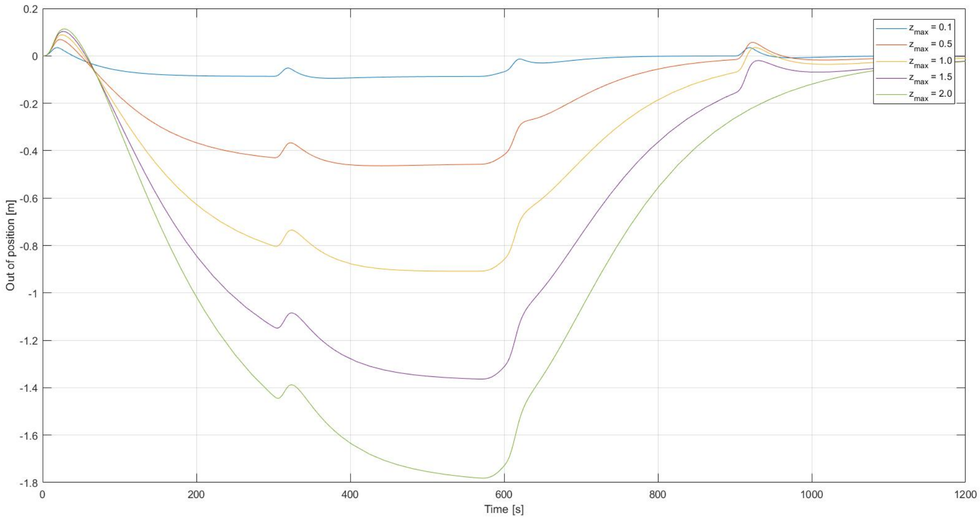


Figure 7.12: Heave motion during debarking SF, submarine 1

to the disturbances on submarine 1. Having the same disturbances on a submarine that is smaller, and therefore easier to move through the water, results in a larger capacity system.

There is choosen to set the maximum flow to 0.1 m^3/s for submarine 1. This is because it can be expected that especially for SF missions the submarine must be able to keep depth. Having a maximum overshoot of 0.5 meters seems to be enough to conduct the operation safely. In table 7.3 there can be seen that for a maximum overshoot of 0.5 meters a flow of 0.12 m^3/s is needed. It is expected that the same results or better results can be obtained with this flow when tuning the controller in the right way. For submarine 2 the maximum needed flow is set to 0.12m^3/s. This because of the same reasons as for submarine 1.

For snorkeling it can be seen that there is a need of a large flow. rising all the masts at once causes a large disturbance. When to submarine is snorkeling, the hydroplanes can compensate for the movements. Therefore there is no need for the system to have such a large capacity

The pressure that the system has to overcome needs to be studied to see which actuator is able to deliver enough head to operate as expected. The main component of the needed head depends on the depth of the submarine. The studied diving depth of the submarine is 300 meters below the surface. 300 meters water column is approximately equal to 30 bar. It can be assumed that the dynamic pressure losses in the compensating system are very small compared to the static pressure the system has to overcome and can can be neglected for now. This maximum pressure that the system has to overcome is equal for both submarines.

As discussed previously, the models that are made for the compensations system are:

- one large centrifugal pump
- Two centrifugal pumps in series
- piston pump

The system must be able to deliver 0.1 m^3/s and overcome a pressure difference of approximately 30 bar.

The first model is a model in which a large centrifugal pump is able to overcome the total needed pressure difference while allowing enough flow. The centrifugal pump that is chosen for submarine 1 to model is the KSB Multitec 150 12.1. This centrifugal pump is able to deliver enough head and has a volume flow that is large enough. The reason of the large head that the pump can deliver is because it is a multi stage centrifugal pump. This means that the pump in theory consist of multiple impellers combined in one pump. The Multitec 150 12.1 delivers enough head when choosing the six stage variant (KSB, b). The pump characteristics of this pump can be found in figure 7.14. The important pages from the product catalogue can be found in appendix I. For submarine 2 the same pump is used since this pump is also able to full fill the requirements for submarine 2

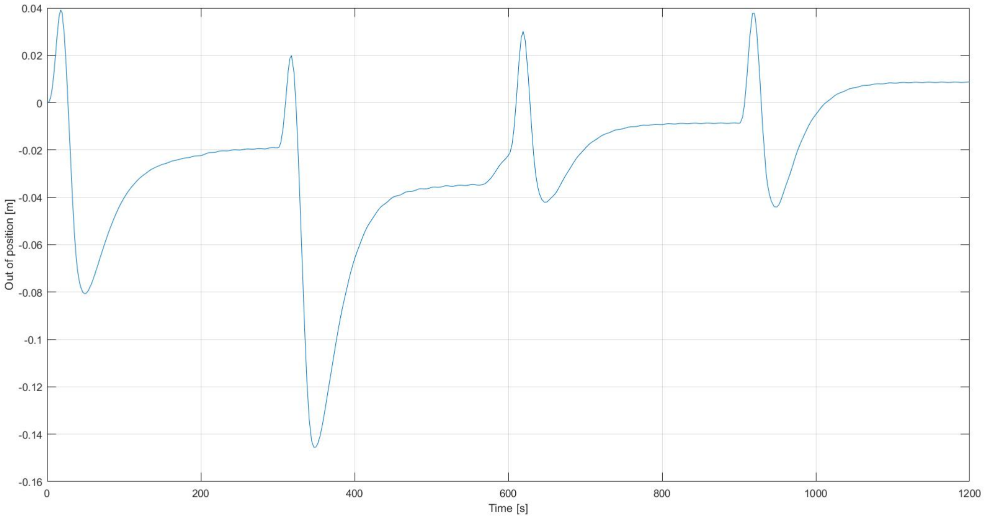
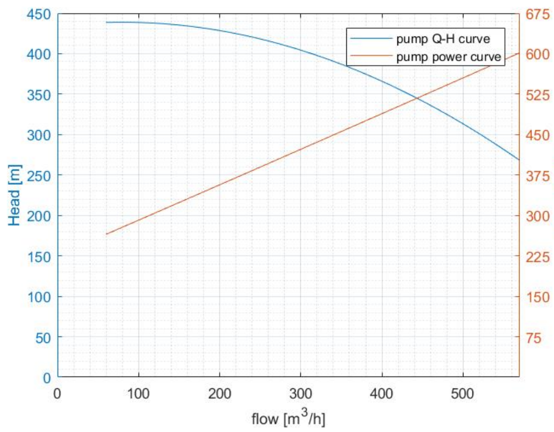


Figure 7.13: Heave motion during disembarking SF, tuned trim system submarine 1



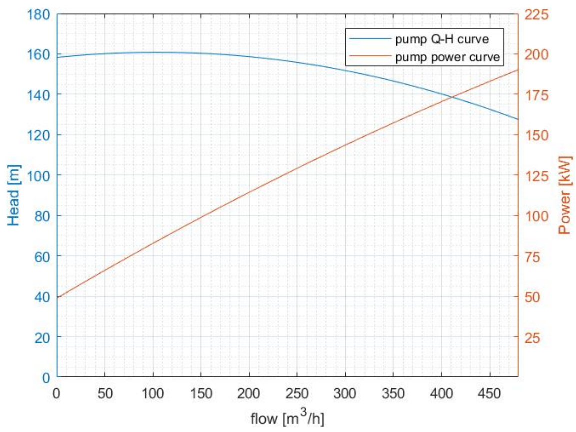


Figure 7.14: Pump characteristics of the KSB Multitec 150 12.1 with 1750 RPM centrifugal pump

Figure 7.15: Pump characteristics of the KSB Etanorm 125-100-315 with 2900 RPM centrifugal pump

The second model, consist of two identical centrifugal pumps in series. This because the largest head produced by one single centrifugal pump is 160 meters. The two combined deliver a head of 320 meters. The centrifugal pump selected is the KSB Etanorm 125-100-315 (KSB, a). The pump characteristics can be found in figure 7.15. The page from the KSB cathelogue can be found in appendix I. For submarine 2 the same pumps are used.

The last option is the model which consist of a piston pump. The piston pump needed could not be found in the catalogues. One of the disadvantages of the piston pump is that it usually delivers a low flow. Therefore a smaller piston pump is scaled up in order to achieve the right flow. The pressure of this pump is kept the same. The piston pump model is based on the Bosch Rexroth A4VSO size 1000 piston pump (Bosch Rexroth AG, 2009). This pump is scaled up with a factor 5 to fulfil the criteria for the compensation system. The characteristics of this pump can be found in figure 7.16. For submarine 2 this pump is scaled up with a factor 6 to fulfil the requirements which can been seen in figure 7.17. The original pump characteristic can be found in appendix I.

7.2.3. Hover system

The dedicated hover system has to compensate for two different disturbances. These two disturbances are: a change in density and second order effect due to waves. These two disturbances are first considered sep-

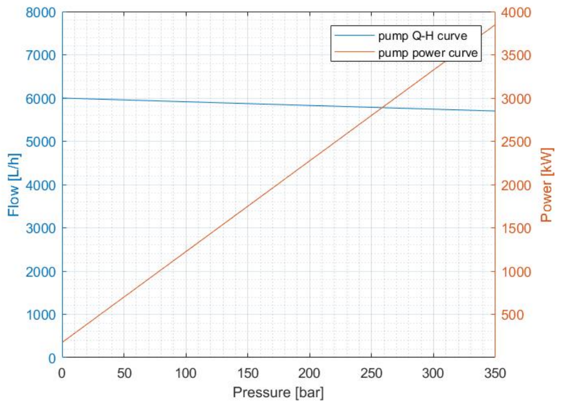


Figure 7.16: Modelled piston Pump characteristics Bosch Rexroth A4VSO size 1000 1200 rpm scale 5

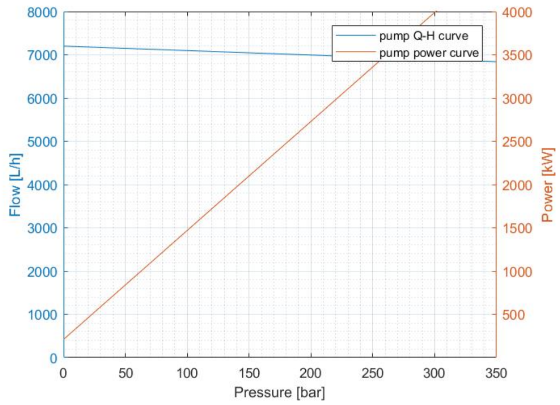


Figure 7.17: Modelled piston Pump characteristics Bosch Rexroth A4VSO size 1000 1200 rpm scale 6

arately. This because it can be possible that the two disturbances are best compensated by two different systems. The change in density occurs suddenly and cause the submarine to surface or dive deeper. The second order effect due to the waves is harmonic disturbance. This causes that on average the submarine will stay in position without compensation while over time there are depth fluctuations.

As seen in chapter 5, the change in density occurs instantly and is relatively large. The same maximum overshoots are used as for the compensation system; 0.1, 0.5, 1.0, 1.5 and 2.0 meters. Table 7.5 and 7.6 give the needed flows of the system for submarine 1 and submarine 2. The disturbance for submarine 1 is two times larger than the disturbance for submarine 2, which also results in a flow of approximately 2 times larger.

Table 7.5: Absolute flows needed for hovering, submarine 1

	$z_{0.1}$	$z_{0.5}$	$z_{1.0}$	$z_{1.5}$	$z_{2.0}$
Density change	4.38	2	1.39	1.15	0.96

Table 7.6: Absolute flows needed for hovering, submarine2

	$z_{0.1}$	$z_{0.5}$	$z_{1.0}$	$z_{1.5}$	$z_{2.0}$
Density change	2.06	0.94	0.65	0.54	0.47

For submarine 1, the graphs for the needed flow and the out of position track are shown in figure 7.18 and 7.19. All the other graphs, including those for submarine 2 can be found in appendix H.3.2. To define the needed actuators, the maximum overshoot is set at 1 meter. This results in a needed flow of 1.39 for submarine 1 and 0.41 for submarine 2. Because the variable buoyancy system will not expel water from tank, this flow can also be defined as 14.98 kN/s and 4.12 kN/s.

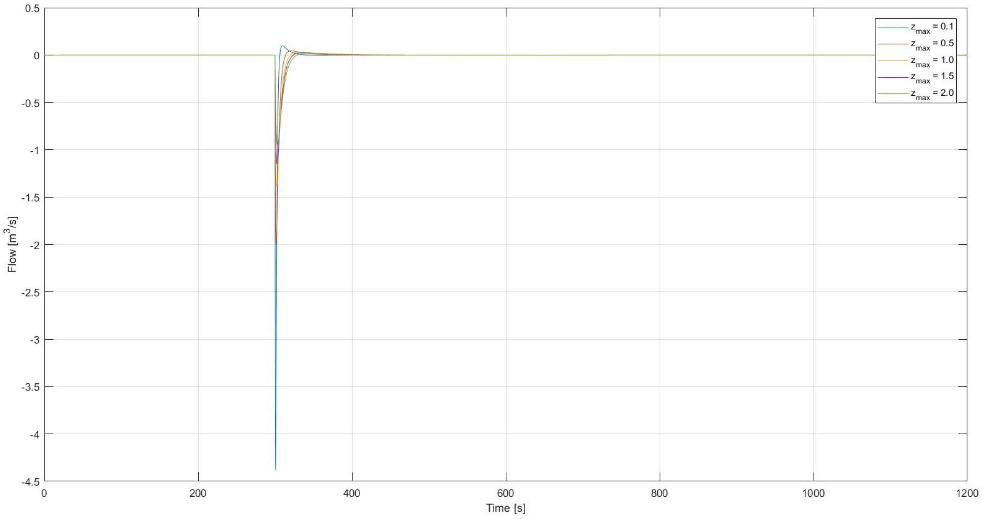


Figure 7.18: Needed flow for compensating while density changes, submarine 1

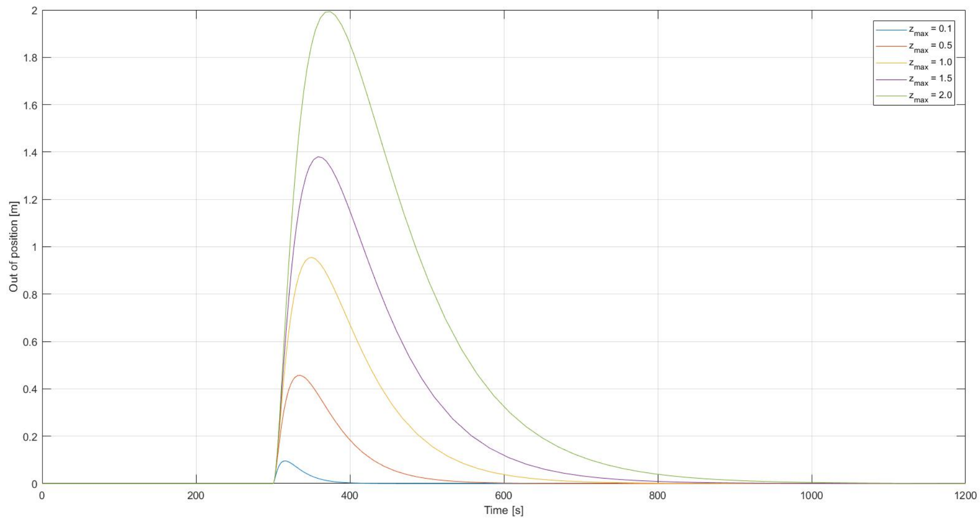


Figure 7.19: Heave motion during a density change, submarine 1

With the maximum flows defined, the pressures that the actuators have to deliver can be determined. This is done by letting the needed flow as calculated before, go through the system and look at the pressure losses. This is only done for the pre-pressurised tanks and the variable buoyancy, because the pump systems that are also included will be the pump systems which are used for the compensation system. The pressure head for the variable buoyancy is calculated different. This will be discussed later because it is part of sizing the total hydraulic system.

The second order effects are a harmonic disturbance. This gives that the forces over time will be zero resulting in a heaving submarine but on average, it keeps its position. However the anechoic tiles on the hull of the submarine and the compressibility of the submarine cause the submarine to surface. Figure 7.20 gives the difference in heave motion for submarine 1. Due to the effect of the anechoic tiles and compression of the hull, a submarine must compensate for second order effects. When the hull of the submarine is not covered with anechoic tiles and is very stiff, the second order effects of waves do not have to be compensated. The submarine considered is covered with anechoic tiles and therefore there is a need of compensation.

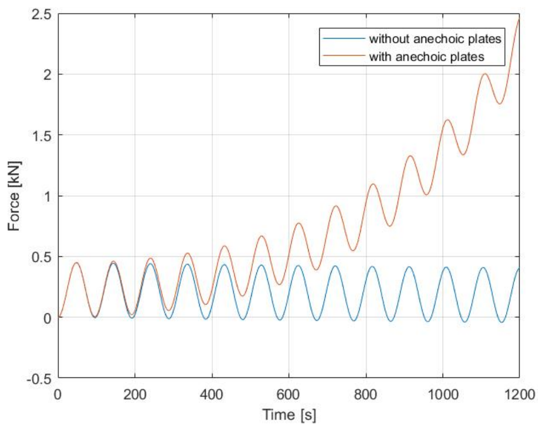


Figure 7.20: Have motion due to second order effects of waves without compensation for submarine 1

Table 7.7: Absolute flow needed for hovering for second order wave effects submarine 1

	$z_{0.1}$	$z_{0.2}$	$z_{0.3}$	$z_{0.5}$
Second order wave effects	0.113	0.053	0.019	0.003

As can be seen in figure 7.20, the disturbance of the second order effect is not large. When only compensate for the compression effects, the heave amplitude remains under 0.5 meter. The heave amplitudes that are studied for the second order effects are therefore 0.5, 0.3, 0.2 and 0.1 meters. In case of a maximum heave amplitude of 0.5 meters, only the compression effects are compensated. For the other maximum amplitudes there is considered an optimal tuning of the PID controller is such a way that the heave motion is reduced. The resulting heave motion for submarine 1 can be found in figure 7.21. The needed flows for compensation can be found in figure 7.22. Table 7.7 gives the needed flows to compensate for the second order effects of waves for submarine 1 and table 7.8 gives these details for submarine 2. These flows can also be translated into an extension in case of the system with variable buoyancy.

Depending on the desired maximum out of position, the delivered flow can be determined. The maximum heave amplitude is taken as 0.2 meters. This is seen as enough for safe operations. For submarine 1 the capacity of the hover system is taken at 0.050 m^3/s. For submarine 2 the capacity is taken at 0.025 m^3/s. It is expected that although these capacities are lower than found in table 7.7 and 7.8 the results will be the same or even better. The results of tuning for hovering for submarine 1 and submarine 2 can be found in appendix H.3.2.

The hover system to compensate for the second order wave effects is modelled with four models. These models are:

- a centrifugal pump
- a piston pump
- a pre-pressurised tank
- the variable buoyancy system

For the option of hovering with the pumps, the compensation pumps are used. The ability of using the compensation system to also function as hover system is considered. This means that for the hover system which uses pumps, the same pumps are used as previously for the comparison of the compensation systems. The centrifugal pumps in series is taken out. This because of the centrifugal pump only operates close to the

Table 7.8: Absolute flow needed for hovering for second order wave effects submarine 2

	$z_{0.1}$	$z_{0.2}$	$z_{0.3}$	$z_{0.5}$
Second order wave effects	0.055	0.026	0.013	0.002

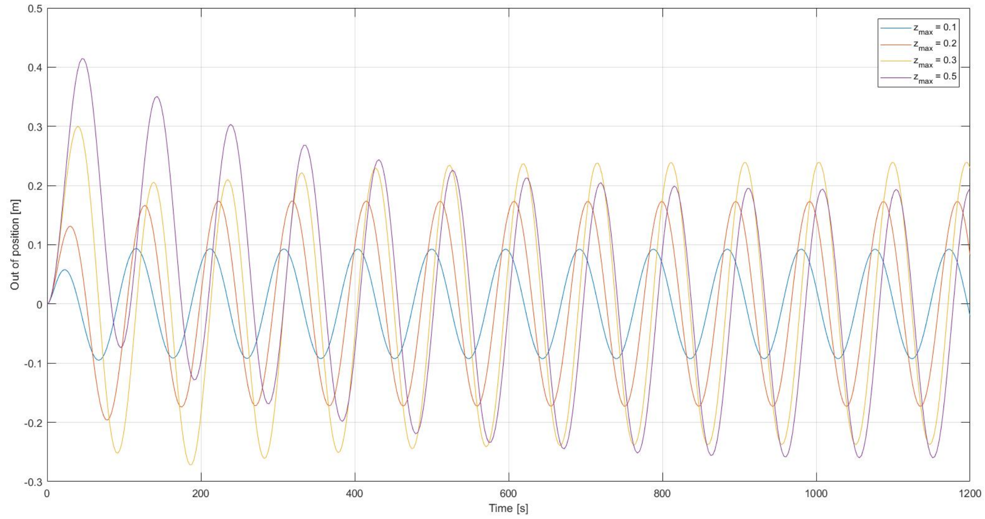


Figure 7.21: Have motion due to second order wave effects for submarine 1

surface which results in a smaller pressure to overcome. Therefore there is no need to have two centrifugal pumps in series to achieve a high output head.

For the dedicated hover systems, pre-pressurised tank and variable buoyancy, new models are made. The pre-pressurised tank uses pressurised air to expel the water from the tank. The hover tanks are assumed to have a total size of 25% of the compensation tank. This results in a hover tanks size of 19.5 m^3 for submarine 1 and 9.75 m^3 for submarine 2. When using the the calculated optimal air volume as determined in chapter 5, the hover system empties its hover tank completely when travelling through a density front. The initial air volume is for submarine 1 equal to 5.66 m^3 and 2.83 m^3 for submarine 2. The maximum pressure that the system has to overcome is when the submarine is at the studied depth of 300 meters. This will result in a initial pressure of 72.3 bar. The average pressure that is lost due to expelling the ballast water equals than 51.15 bar according to equation 6.35.

For the system using variable buoyancy the characteristics of the system can be found in table 7.9. The diameter of the tube is determined by the maximum disturbance and the length of the tube of 5 meters. This 5 meters is taken because this length is also seen for the maximum length of snorkel mast. Hydraulic cylinders usually have a maximal operating pressure of 270 bar. With the needed speed for extending determined earlier, the maximum force per second can be determined. By varying the hydraulic cylinder's inner area until the operating pressure of the hydraulic cylinder is 270 bar, the cylinder area is found. This result in an area of 0.48 m^2 for the hydraulic cylinder of submarine 1 and 0.34 m^2 for submarine 2. With the hydraulic cylinder known, the hydraulic pipe lines can be determined. The volume flow through the system can be determined. With the maximum volume flow and the restriction of a speed of 4.5 m/s, the diameter of the hydraulic lines can be determined.

Table 7.9: Characteristics of the hydraulic system components used

Item	**Symbol**	**unit**	**Submarine 1**	**Submarine 2**
Length tube	L_{tube}	[m]	5	5
Diameter tube	D_{tube}	[m]	1.45	1.02
Stroke hydraulic cylinder	$L_{hydr.cyl.}$	[m]	5	5
Inner diameter hydraulic cylinder	$D_{hydr.cyl.}$	[m]	0.48	0.34
Pipe diameter	D_{pipe}	[m]	0.208	0.143

The maximum pressures needed for submarine 1 and submarine 2 are 270 bar.

The flow for the piston pump for submarine 1 equals then 0.152 m^3/s. For submarine 2 the flow must be

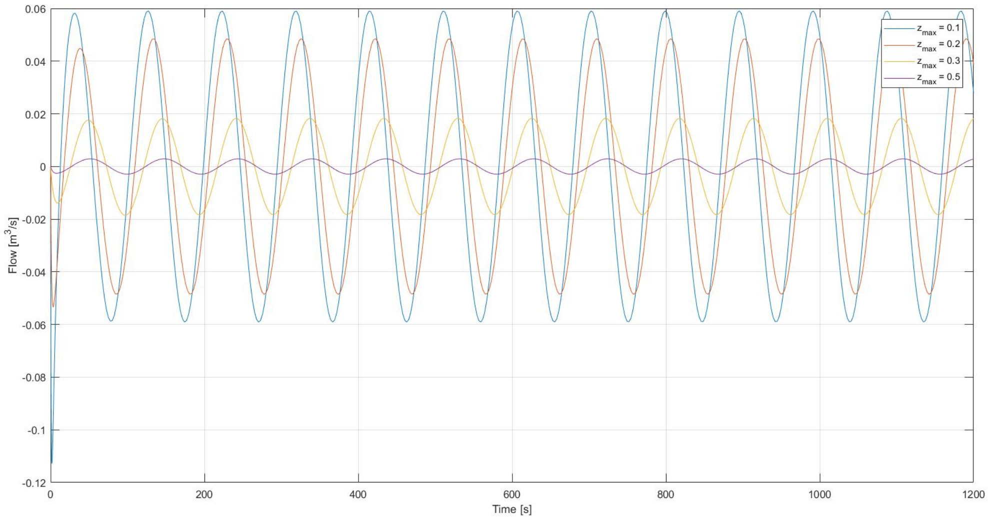


Figure 7.22: Needed flow for compensation of second order wave effects for submarine 1

equal to 0.072 m^3/s. The same piston pump as used as plunger pump earlier is used. This leads also to a scale up of the used pump. For submarine 1, the piston pump is scaled up with a factor 8. For submarine 2 this factor is 2.5.

7.3. Performance of the systems

In the previous section the systems models are sized. These models are used in this section to compare the different solutions. The different comparisons are made for the trim, compensation and hover system. The systems are compared to each other based on energy and sound power level. An overview of the compensation model including a piston pump can be found in figure 7.23. The models made for the other systems have the same lay-out.

7.3.1. Trim system

The two modelled solutions for the trim system are compared to each other. The two options are:

- a trim system with centrifugal pump
- a trim system with pre-pressurised tanks

These systems are compared based on energy consumption and noise production. A major drawback of having the system with pre-pressurised tanks is that it gives a roll angle. This because of the a-symmetry of the trim tanks which can be seen in figure 6.4. This roll angle over the time can be found in figure 7.24. In the solution with the centrifugal pump there are only two tanks and therefore displacing water between the tanks will not result in a rolling moment. The final results for energy and noise for the different profiles can be found in table 7.10.

Energy

There can be seen from the result of the simulation that the centrifugal pump is very much over dimensioned. The centrifugal pump will never run at full speed in the simulations. This is because of the margin which is taken in the maximum trim. The centrifugal pump is selected based upon the assumption that it must also be able to overcome the static pressure difference when having a trim angle of 3°. This results in that the centrifugal pump will run in most cases in off design situations, but still around its best efficiency point.

The downside of having the system with the pre-pressurised tanks is that in almost all cases a lot of energy is lost due to the valve. The valve is controlling the flow and must, because of the high pressures inside the tank, reduce the high pressure in order to achieve the right flow.

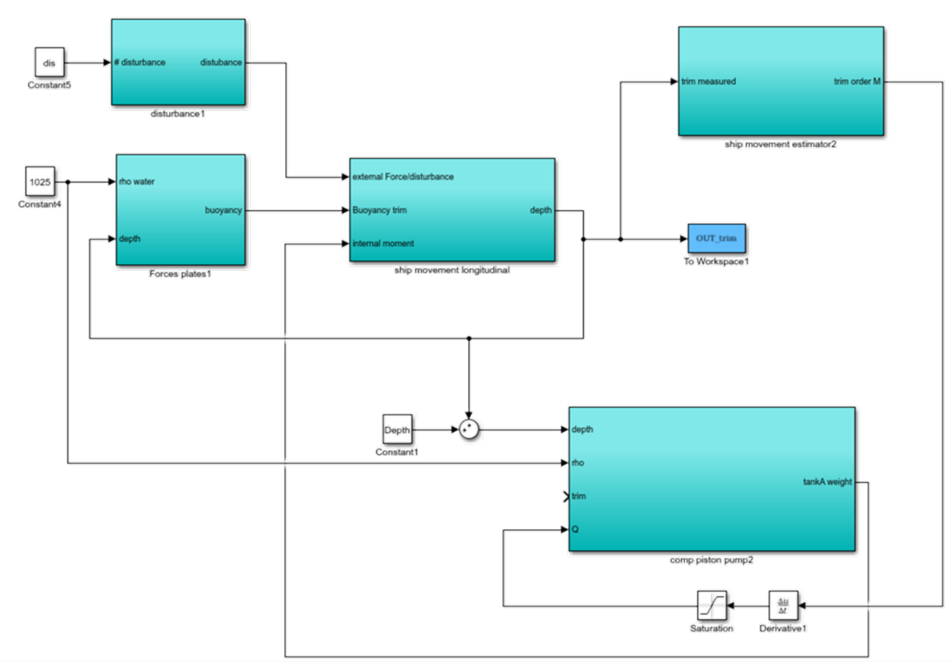


Figure 7.23: Simulink model for compensation system with piston pump

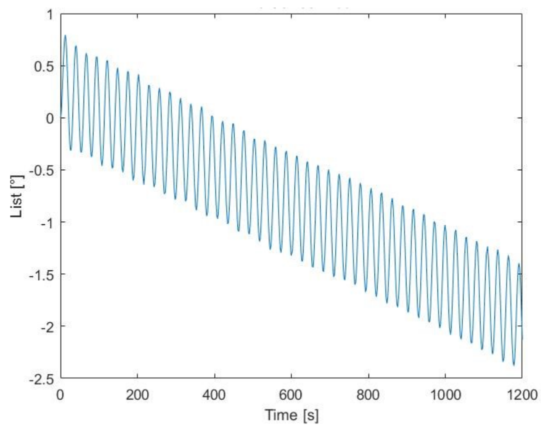


Figure 7.24: Roll angle during trimming with pre-pressurised tanks

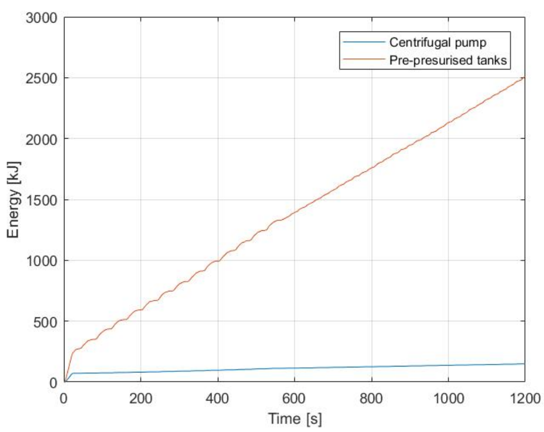


Figure 7.25: Energy disembarking SF trim system submarine 1

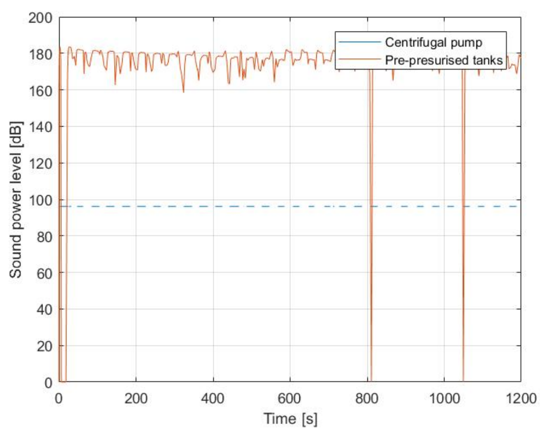


Figure 7.26: SPL disembarking SF trim system submarine 1

Table 7.10: Results for trim system

		Disembark SF		Anchoring		Transit AIP		Snorkelling		Disembark AUV	
		Centrifugal pump	Pre-pressurised tanks	Centrifugal pump	Pre-pressurised tanks	Centrifugal pump	Pre-pressurised tanks	Centrifugal pump	Pre-pressurised tanks	Centrifugal pump	Pre-pressurised tanks
sub 1	L_{wl} [dB]	96	183	96	182	96	182	96	182	96	183
	E [kJ]	123	2268	44	2035	15	1970	15	1970	57	2017
	P_{max} [kW]	4	12	6	14	0	2	0	2	1	5
sub 2	L_{wl} [dB]	107	184	107	183	107	183	107	183	107	184
	E [kJ]	235	2405	96	2282	24	2170	24	2170	124	2240
	P_{max} [kW]	7	13	8	38	0	3	0	3	1	5

Noise

The centrifugal pump is known for its low noise production. The noise produced by the valve depends on the flow and the pressure losses in the valve as seen in chapter 6. Because of the high pressure in the pre-pressurised tanks, the valve is in many cases slightly open resulting in a high pressure loss. This results in a high noise production when controlling the flow by the valve. For the centrifugal pump, the noise level is constant over time. It is seen that the noise production of the centrifugal pump when controlled with variable speed.

In appendix J the results for disembarking SF, anchoring and disembarking an AUV are shown. This is done for submarine 1 and 2

7.3.2. Compensation system

The compensation system is modelled with three pump models which are compared to each other. The models which are compared to each other are:

- a compensation system with a large head multistage centrifugal pump
- a compensation system with centrifugal pumps in series
- a compensation system with a plunger pump

In chapter 5 there was taken a disturbance for snorkelling. This disturbance took into account a large force due to raising the masts. Since during this disturbance the submarine is sailing, the hydroplanes can help to keep the submarine stable. Therefore this disturbance will not be considered for the compensation system. all the results considering noise and energy are listed in table 7.11.

Energy

It can be seen from the simulations is that the piston pump scores low when operated at low pressures see figure 7.27. This was expected because the piston pumps' efficiency is low when operated at low torque. This can be seen in figure 6.15. When operated at large depths, the efficiency of the plunger pump becomes very high, resulting in a good performances compared to the centrifugal pumps, see figure 7.28.

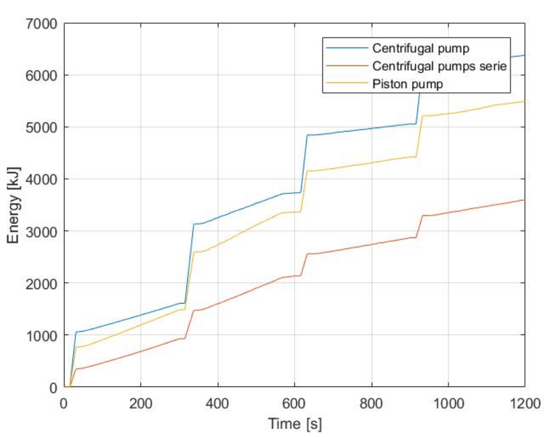


Figure 7.27: Energy disembarking SF compensation system submarine 1

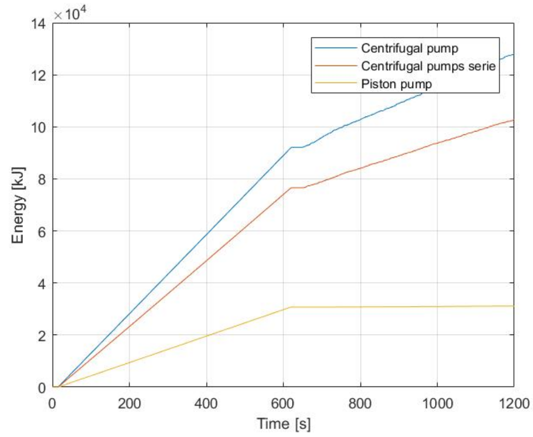


Figure 7.28: Energy disembarking AUV compensation system submarine 1

The efficiency of the multistage centrifugal pump decreases when operated at greater depths, see figure 7.28. The efficiency of the centrifugal pumps in series also decreases when operated deeply submerged. However the centrifugal pumps can be operated separately, increasing the overall efficiency when compared to the multistage centrifugal pump.

Based on energy, the plunger pumps is better than the centrifugal pumps. This is because the low efficiency of the centrifugal pump when operated outside its operating envelope. Especially when the system has a high head difference to overcome, the centrifugal pump becomes inefficient.

One of the disadvantages of the centrifugal pump is that when it is operated in off design situations, the efficiency decreases. This is shown in figure 7.29 and 7.30. At certain points it is even physical impossible for the centrifugal pump to work. this because the centrifugal pump needs a certain flow to work. Therefore the centrifugal pump is not suited for relatively low flow rates. This is especially the case when the system curve is almost a flat line. For continuously control, The pump must also be able to operate at low flow rates.

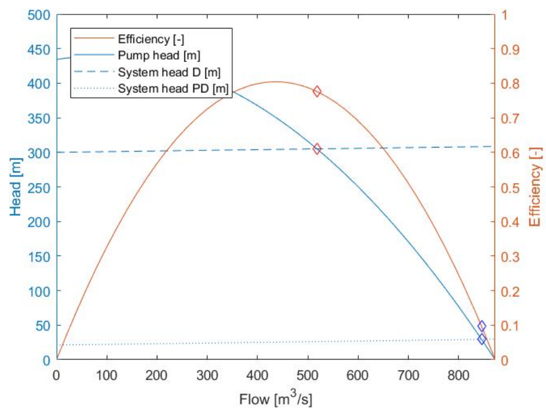


Figure 7.29: Centrifugal pump efficiency with static head change

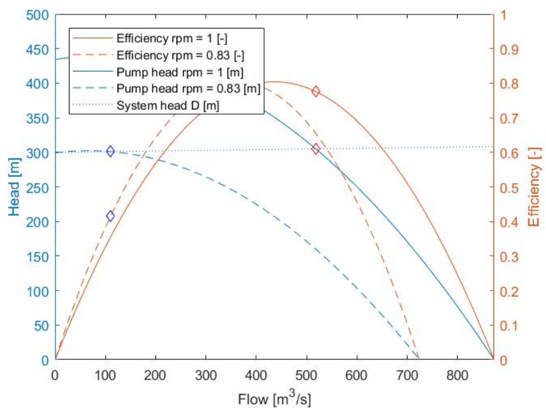


Figure 7.30: Centrifugal pump efficiency with variable speed control

Based on the results of the centrifugal pump when operated far from its optimal efficiency point, there can be concluded that a centrifugal pump, in series or single, is not suitable for the compensation system. This because the system operates in many cases outside the best operational window. Next to that, the centrifugal pump operated far from its operational point can give cavitation. This will damage the pump and at certain operational points, the pump will not be able to deliver flow.

Noise

Based on noise production the plunger pump scores very bad. Due to multiple moving parts, the noise of the plunger pump is high.

Table 7.11: Results for the compensation system

		Disembarking SF			Anchoring			Transit AIP			Disembarking AUV		
		Centrifugal pump	Centrifugal pump series	Piston pump	Centrifugal pump	Centrifugal pump series	Piston pump	Centrifugal pump	Centrifugal pump series	Piston pump	Centrifugal pump	Centrifugal pump series	Piston pump
sub 1	L_{wl} [dB]	124	119	146	124	122	143	124	119	146	124	122	135
	E [MJ]	6.2	3.5	5.0	67.6	29.6	3.5	46.1	25.2	36.1	133.7	106.9	31.1
	P_{max}[kW]	67	26	48	245	216	186	201	161	164	160	136	66
sub 2	L_{wl}[dB]	124	119	145	124	122	145	124	119	147	124	122	133
	E [MJ]	10	4	21	40	28	3	104	73	157	138	107	33
	P_{max}[kW]	111	34	222	302	301	415	500	461	623	165	142	82

The centrifugal pumps in series scores the best based on noise production, see figure 7.31. However it is seen that the centrifugal pump works often far from its working point which can result in cavitation. Noise due cavitation is not taken into account in the calculation but is very undesired due to the high noise that it produces.

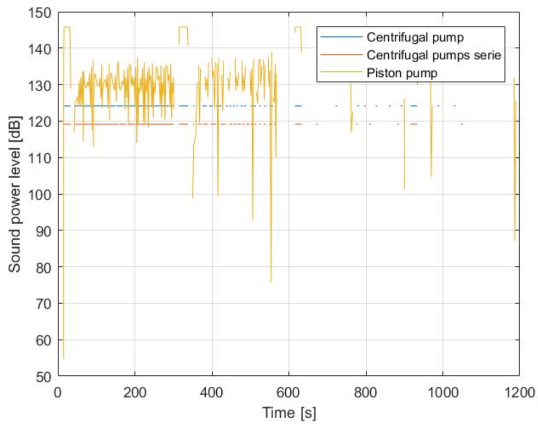


Figure 7.31: SPL during disembarking SF compensation system submarine 1

The results of the simulation of other profiles can be found in appendix J.

7.3.3. Hover system for change of density

There can be chosen to hover with the compensation system. This saves space and money. However since the compensation system is smaller, deliver less flow, the out of position will increase. It is seen that for submarine 2 the out of position when using the compensation system becomes 7 meters. For submarine 1 the out of position becomes 31 meters. These out of trims are unacceptable. The difference is caused by the larger compensation system on board of submarine 2. The out of positions plots when using the compensation system for hovering can be found in figure 7.32 and 7.33.

The comparison for the compensation system showed that the plunger pump scores best based on energy. Therefore the comparison between the pump systems will not be made in here. The centrifugal pump appeared not to be fully functional at all the depths. The results for the studied systems can be found in table 7.12.

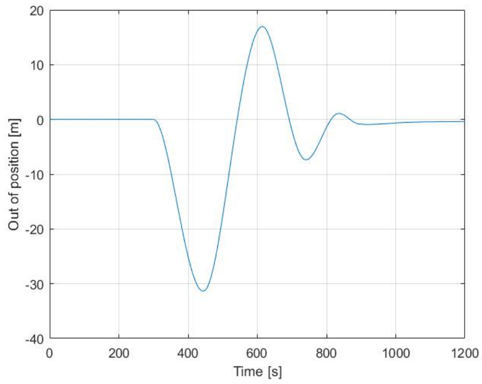


Figure 7.32: Out of position submarine 1 using compensation system to hover

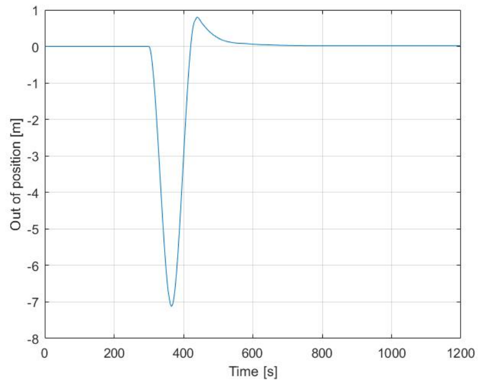


Figure 7.33: Out of position submarine 2 using compensation system to hover

Energy

For the dedicated hover systems the comparison based on energy is made in here. The density change for which the hover system has to compensate for, is simulated at two depths. The first case is at periscope depth and the second case is at 300 meters. The system using variable buoyancy scores the best based on energy. Moving the tube through the water does not cost much energy. Especially when at periscope depth, see figure 7.34. The losses due to the valve become very high at periscope depth for the pre-pressurised tank system. The energy for the pre-pressurised tanks is the same at periscope depth as it is for maximum depth, see figure 7.35.

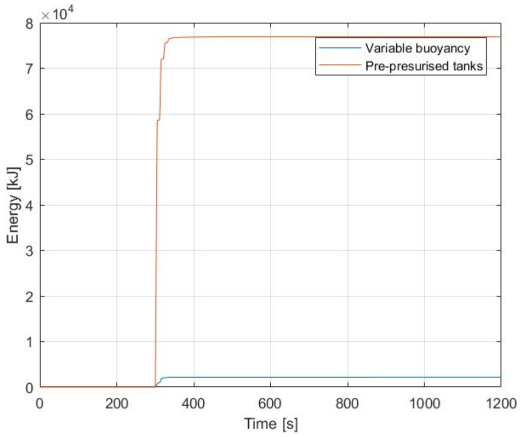


Figure 7.34: Energy disembarking SF hover system submarine 1

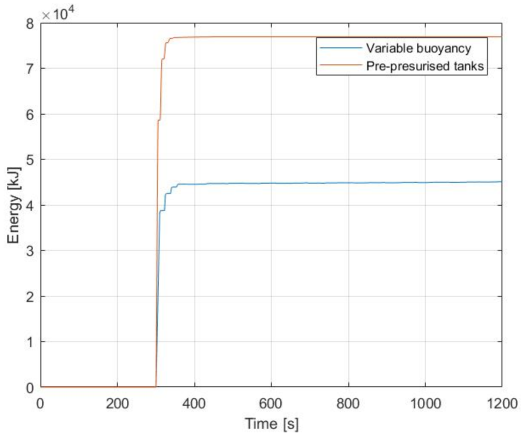


Figure 7.35: Energy disembarking AUV hover system submarine 1

Based on energy the variable buoyancy solution scores the best. The pre-pressurised tanks scores low because to control the flow, lots of energy is lost due to the energy stored in the pressurised water particles. Therefore, especially at lower depths the variable buoyancy options scores better. The variable buoyancy option uses energy based upon the situation, i.e. the outside pressure. Therefore the energy used by the system at PD is lower than at D.

Noise

The piston pump in the hydraulic system produces noise. The main noise components of the total sound power level of the piston pump, is the fluid borne noise. The piston pump is not connected to the sea. Therefore the fluid borne is not as important as it is for systems connected to the sea. Based on noise, the variable buoyancy scores the best. For submarine 1 the sound power level for the change in density at PD can be found in figure 7.36.

Table 7.12: Results for hover system for change of density

		Density change z = PD		Density change z = D	
		Variable buoyancy	Pre-pressurised tanks	Variable buoyancy	Pre-pressurised tanks
sub 1	L_{wl} [dB]	157	225	175	222
	E [kJ]	2174	76855	45156	76855
	P_{max} [kW]	310	12676	4096	12676
sub 2	L_{wl} [dB]	161	222	178	219
	E [kJ]	1789	53169	19589	53169
	P_{max} [kW]	216	5928	2133	5928

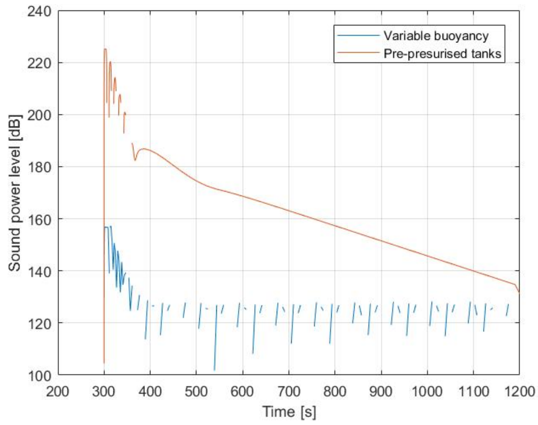


Figure 7.36: SPL during hovering for change of density at PD, submarine 1

In the dimensioning of the tube there is seen that this leads to large cylinder in case of submarine 1. For submarine 2 this cylinder is more like the dimensions of the snorkel mast. Using the snorkel mast as variable buoyancy results in no extra structural members. This is possible for the smaller submarine but not for the large submarine. This must be taken into account when making a final decision. More results for the change in density can be found in appendix J.

7.3.4. Hover system for second order wave effects

Because of the needed flow for compensating the second order wave effects is lower than the delivered flow by the compensation system, the second order wave effects can be compensated by the compensation system for both submarines. Of course there can also be chosen to use a dedicated system to compensate for the second order wave effects. The results from the simulations are listed in table 7.13.

Energy

Based on energy the best solution is to use the variable buoyancy system. This system scores best based on energy. However it can be seen that it is possible to compensated for the second order wave effects with the already installed compensation system. In here there is also taken into account a compensation system working with a centrifugal pump. There was seen that this was not the best option for compensation. However an extra centrifugal pump can be installed which could compensate for the second order wave effects. This is because the second order effects can only be felt close to the surface, resulting in a low static head to overcome. Therefore a centrifugal pump can be used.

Table 7.13: Results for hover system for second order wave effects

		Second order effect			
		Variable buoyancy	Pre-pressurised tank	Piston pump	Centrifugal pump
sub1	P_{max} [kW]	128	209	140	124
	E [kJ]	1164	104984	6303	5141
	P_{max} [kW]	3	275	14	11
sub 2	L_{wl} [dB]	132	208	131	124
	E [kJ]	731	110183	4872	1085
	P_{max} [kW]	2	228	11	2

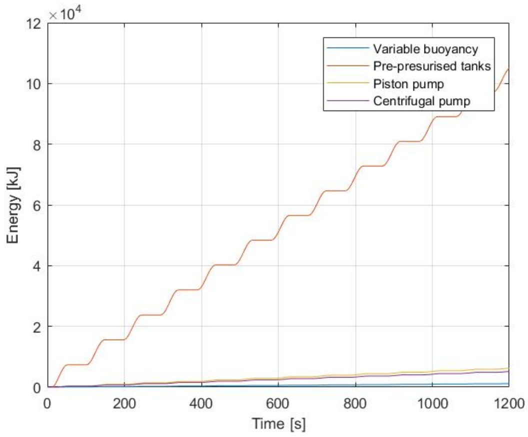


Figure 7.37: Energy for compensating second order effects submarine 1

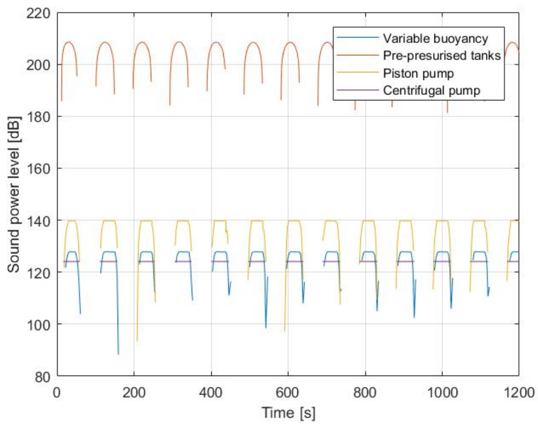


Figure 7.38: SPL during hovering for second order wave effects submarine 1

Noise

Based on noise it is the best to use the variable buoyancy. The centrifugal pump scores the best based on the sound power level. However, since the piston pump in the variable buoyancy system is not connected to the sea, the fluid borne noise is not an issue. The fluid borne noise is included in the overall sound power level.

For both submarines the snorkel mast can be used to compensate for the second order wave effects. This is an advantage since there is no need for extra structural members. Both the systems can be operated with the already installed hydraulic system. For submarine 2 the results are plotted in appendix J

7.4. Combined model

The three best options are put in one model. This because the different systems influence each other. When having a trim angle, the static height in the pipes of the compensation system and the hover system changes. When the hover tank and the compensation tank are not located in the centre of buoyancy, compensating and hovering will bring an extra trimming moment to the system. The systems that are in the combined model are:

- the trim system with the centrifugal pump;
- the compensation system with the plunger pump;
- the hover system using variable buoyancy.

These models together give an overview of the total movement of the submarine. The depth when placing these systems on submarine 2 can be found in figure 7.40. The trim of the submarine can be found in figure 7.39. The rest of the results for different profiles can be found in appendix J.

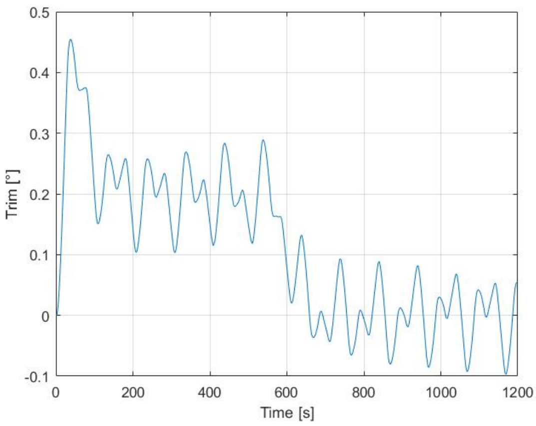


Figure 7.39: Trim while disembarking SF

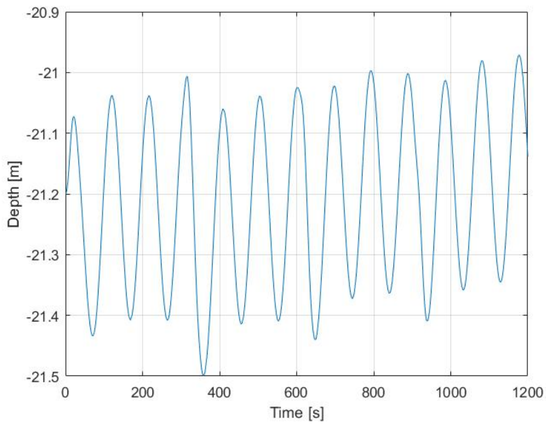


Figure 7.40: Depth while disembarking SF

7.5. Redundancy

Redundancy gives an idea till what extend the system is operational when one system breaks down. Redundancy can be achieved by making sure there are spare parts connected to the system. In order to reduce spare part, it is wise to see until which extend systems can have the the same spare parts.

Some systems are most likely more reliable than others. For example the systems working with pre-pressurised tanks have less change on failure because of there simple design. For pumps on the other hand, mechanical failure occurs more often and pumps are therefore less reliable than the pre-pressurised tanks. This is for now not taken into account. Only redundancy is taken into account. This is done with the selected actuators. All the modelled actuators can break down. There is investigated if there is another actuator, used for a different mode, which is able to take over the the task for the broken actuator. Some actuators are not able to fully take over the task of a system. In this case the broken actuator can be replaced by the other till certain extend. Table 7.14 shows if the actuator can be replaced by another actuator. The numbers in the table shows to which extend they can be replaced by that certain system. 0 means that it can not be replaced. 1 indicates that the system can only replace until a certain extend, the broken system is not able to fully operate. 2 stands for the same as 1, but in this case the replacement is more capable of taking over. 3 stands for that the broken system can be fully replaced by its replacement.

From this table it can be concluded that for the trim system the best choose is to use the centrifugal pump. The pre-pressurised tanks can not be replaced by another system. This is because that will mean that there is another trim tank. However the pre-pressurised tank system is most likely more reliable. When the system does not fail, it does not have to be replaced at all.

For the compensation system, all solutions score equal based on redundancy when replaced by an actuator from the hover system. These solutions are all pump systems with the same head and flow demands. Therefore they can be replaced by the same systems. However the systems with which they can be replaced and be fully operational, are the pump systems from the hover system. There is said that the pump systems of the hover system are the same systems as used for compensating. When replaced by the centrifugal pump from the trim system, there can be seen that the centrifugal pump in series scores the best. This is because this systems has two smaller centrifugal pumps within the system. When one of the centrifugal pump breaks down, it can be replaced by the trim pump. Because these pumps are more similar to each other than the large centrifugal pump or the plunger pump, the system will be more operational. Based on redundancy the best option for the compensating system is the system with the centrifugal pumps in series.

The hovering systems can not be well replaced by the trim system. When using a centrifugal pump in the trim systems, the pre-pressurised hover tank can be connected to the centrifugal pump. This allows emptying the hover tank by the trim system until certain extend. When using the compensation system as hover system, the same results are obtained as seen for the compensation system. Therefore these hover systems will not be elaborated furthermore. The system using pre-pressurised tanks can also be connected to the compensation

Table 7.14: Redundancy of the ballast trim system

		Replacement									
		Trim		Compensation			Hovering				
Actuators		Centrifugal pump	Pre-pressurised tanks	Centrifugal pump	Centrifugal pump	Piston pump	Centrifugal pump	Centrifugal pump series	Piston pump	Pre-pressurised tank	Variable buoyancy
Trim	Centrifugal pump	na	na	3	3	3	3	3	3	0	0
	Pre-pressurised tanks	na	na	0	0	0	0	0	0	0	0
Compensation	Centrifugal pump	1	0	na	na	na	3	3	3	0	0
	Centrifugal pump	2	0	na	na	na	3	3	3	0	0
	Piston pump	1	0	na	na	na	3	3	3	0	0
Hovering	Centrifugal pump	1	0	3	3	3	na	na	na	na	na
	Centrifugal pump series	2	0	3	3	3	na	na	na	na	na
	Piston pump	1	0	3	3	3	na	na	na	na	na
	Pre-pressurised tank	1	0	2	2	2	na	na	na	na	na
	Variable buoyancy	0	0	0	0	0	na	na	na	na	na
Average score as replacement		**0.9**	**0**	**2**	**2**	**2**	**2.4**	**2.4**	**2.4**	**0**	**0**

system. Therefore there can be hovered till a certain level when the pre-pressurised tanks fails and is replaced by the compensation system. It is better to use to compensation system to replace the pre-pressurised tank system because of the higher pressure and higher flow it can deliver.

To reduce spare parts it is wise to choose a spare actuator that is able to fulfil as many tasks as possible. When looking back in table 7.14, there can be seen that having as spare part the actuator of the compensation system scores best. This is when not taking into account the scores of the hovering system using the compensation system. That one scores based, as discussed earlier, because in that case the hovering system and compensation system are the same.

7.6. Difference submarine 1 and 2

There was made a model for the 4000 ton BB2 submarine and the 2000 ton BB2 submarine. This is done to see if there are differences in the ballast trim system between a large and small size SSK submarine. These two different types of submarines was refereed to as submarine 1 and submarine 2.

For the trim system the best option was to use a centrifugal pump to pump water between the trim tanks. This was mainly because of the energy usage of the system using pre-pressurised tanks. For both of the submarines using a centrifugal pump to transfer ballast water between the trim tanks is the best option. The size of the trim system does depend mainly on the installed systems on-board that needs to be compensated. This resulted in a larger trim system for submarine 2 than for submarine 1. This does concluded that a smaller submarine does not have necessary a smaller trim system. It can be expected that the mission for a smaller submarine do differ from a large submarine. A choice could be to make the multi-mission portal smaller. This will result in less water to take in for filling the portal. This will not lead to a trim system with less capacity. This because there is seen that especially the filling is important to size the trim system.

For the compensation system there was found that a system using a centrifugal pump was not suited as compensation system. This resulted directly in using a plunger pump for the compensation system. This is valid for both the submarines. Therefore there can not be found any difference in used system on board of a small and large submarine. The size of the pump does mainly depend on the installed systems on board which could give a difference in weight. The profile that gave the largest disturbance was disembarking SF. The capacity of the compensation system mainly depends on the filling rate of installed systems on board. Since submarine 2 is smaller and therefore easier to move through the water, the same disturbance results in a slightly larger compensation system.

The hover system used for compensating for a sudden change in density differs for the two submarines. For both submarines the variable buoyancy systems scored best. For practical reasons there is chosen to compensate for the change in density on board of submarine 1 with the pre-pressurised tank. This is because the variable buoyancy system becomes very large when installed on submarine 1. For submarine 2 the variable buoyancy system can be used to compensate for the change in density. In this case the snorkel mast can be used. This results in no extra structural members.

The hover system for the second order wave effects is for both submarines the same. The snorkel mast can in both cases be used to compensate for these effects. This way the submarine is able to compensate very accurately for the movements due to the second order waver effects.

The final proposed system lay-outs can be found in appendix F.4. In here the items as bellows and flow meter are left. This is done in order to make the systems for submarine 1 and submarine 2 more clear.

8 Conclusion

The main conclusion of this study is that the best ballast trim system for a modern SSK submarine is the option using a centrifugal pump to compensate for the trimming moment of the submarine, a plunger pump to compensate for the vertical equilibrium in order to enable safe hovering, the best options are to use the variable buoyancy and the pre-pressurised tanks. Employment of these systems on board a modern SSK submarine best enables the submarine to operate safely in a modern warfare zone. The simulations show that from the modelled possibilities, these options are the best (i.e. most suitable for performing their tasks) based on capability, energy, noise and redundancy. Further conclusions that can be drawn, based on the sub-questions specified in chapter 1 can be found below.

- Sub-question 1 is discussed in chapter 3. From the general requirements it can be seen what the considerations were when designing the ballast trim system of the earlier classes of submarines of the RNLN. Due to the classification of this information, it is not available publicly.

- Sub-question 2 took into account the final design of the ballast trim system on board of earlier classes of submarines of the RNLN. The current class of submarines of the RNLN has a well-functioning ballast trim system. This is discussed earlier in chapter 3. Classification of this information means that details of this ballast trim system are not open to the public. Retired submarines of the RNLN also have interesting ballast trim systems on board. The designs of these previously realised systems provided insight required for setting up the models of the system investigated in this study.

- Sub-question 3 included the ballast trim systems on board the submarines of foreign nations and ballast trim systems on board AUVs. These solutions can be found in chapter 4. From AUVs the system using variable buoyancy is very promising. Also detailed information of the Astute class submarine is found which uses a pre-pressurised tank for hovering.

- Sub-question 4 is discussed in chapter 5. A modern SSK submarine is deployed in different ways to in the past. Modern operations,suck as the disembarkation of special forces, result in other considerations for the ballast trim system. The capability of hovering is even more desirable for a modern SSK submarine. This requires a more accurate, faster, but also larger ballast trim system since the disturbances can not be compensated with the hydroplanes while hovering.

- Sub-question 5 considered the disturbances on a modern SSK submarine. These disturbances were all discussed in chapter 5. With these disturbances load cases are made which are used to compare the ballast trim systems. Five different load cases are important. These are: disembarking special forces, disembarking an AUV, snorkelling, transit on AIP and anchoring. For the hover system two important lead cases are considered: change in density and second order wave effects.

- Sub-question 6 resulted in different systems with different characteristics to fulfil the different functions. These systems for trim compensation and hovering can be found in chapter 6. Appendix 4 gives the system diagrams of the modelled ballast trim systems.

- Sub-question 7, what is the best solution as ballast trim system, is discussed in chapter 7. In this chapter the different solutions are compared to each other using the models of chapter 6. The different systems are compared to each other based on: energy, noise production, redundancy and capability. Further conclusions from this sub-question will be discussed later since it also gives an answer to the main question.

- Sub-question 8 is discussed because noise production in the ballast trim system must be low. The noise production of the different systems used as ballast trim system is investigated. This is done based on an estimation of the total sound power level. For now these values should only be taken as an indication of the total sound power level. This produced noise can be reduced by taking extra precautions. These different precautions can be found in appendix G.

The answers to the sub-questions help to form an answer to the main question: *What is the best design, new or existing, for a ballast trim system for a 2000 and 4000 ton modern SSK submarine based on existing and new operational and technical requirements, noise production and energy?*

From an energy perspective the centrifugal pump scores the best for the trim system. The system using pre-pressurised tanks scores low because when controlling the flow with a valve, a large amount of energy is lost. The centrifugal pump on the other hand can be chosen in such a way that it operates around its optimal working point. Therefore the centrifugal pump is energy efficient. A side effect of having the pre-pressurised tanks as trim system is that the submarine will list during the operation. This must than be compensated in some way. When having the centrifugal pump and only two trim tanks, list will not occur.

The trim systems are also compared to each other based on redundancy. It can be concluded that the redundancy of the centrifugal pump is better than the redundancy of the system using pre-pressurised tanks.

It is seen that the trim system for the 2000 ton submarine becomes larger than that of the 4000 ton submarine. This is because when having the same specifications as for example being able to disembark six special forces at the time, the resulting disturbance for the 2000 ton submarine is relatively larger. This of course means that the trim system installed increases in size. Therefore it can be said that decreasing the size of the submarine results in a significant relative volume increase of the trim system.

For the compensation system it is shown that the systems containing centrifugal pumps are not able to fully operate at all depths and volume flows. This is because of physical inability of the centrifugal pumps when operated in off-design conditions. Especially the change in static head the pump has to overcome results in low performance of a centrifugal pump used for the compensation system. This can be seen in figure 7.29. The dynamic losses in the pipe can almost be ignored compared to the static head. This results in a very flat system curve. The piston pump on the other hand, is able to operate at the different off-design conditions. The efficiency of the piston pump mainly relies on the delivered pressure. As long as the static head is high enough, the efficiency of the piston pump will be high. However the piston pump has a high sound power level. This is the major drawback of the system using the plunger pump to compensate. Noise reduction measures are needed to make sure the submarine remains undetected. Especially the fluid born noise of the plunger pump must be dampened. Based on redundancy, all pumps score equal.

For the same reasons as for the trim system, the compensation on board the 2000 ton submarine becomes larger than the compensation system of the 4000 ton submarine. This results also in a significantly larger compensation system when decreasing the size of the submarine having the same operational specifications.

For the hovering system, two options are compared to each other. One is to use the compensation system for hovering operations and the second is to use a dedicated hovering system. When using the compensation system for hovering, the maximum out-of-position increases. This is because the hover system need a larger flow in order to keep the submarine in position. When using the compensation system for hovering, the 4000 ton submarine had an out of position of 31 meters and the 2000 ton submarine had an out of position of 7 meters. These out of positions are unacceptable.

For the dedicated hover system it is seen that the system with the variable buoyancy scores best. This is mainly due to the high efficiency of a piston pump but also that with pre-pressurised tanks lots of energy is lost due to the control of the flow. This is discussed in chapter 5. However for the 4000 ton submarine, hovering with the variable buoyancy system results in a large tube to compensate for the change in density. For the 2000 ton submarine, the variable buoyancy system result in a tube that is of the same size as the snorkel mast.

Therefore a distinction is made between the optimal hover system for the 2000 and 4000 ton submarine. For the 4000 ton submarine the pre-pressurised tank will be used as hover system for the change in density. The second order effects of waves can then be compensated with the variable buoyancy system. The 2000 ton submarine is able to hover for both disturbances with its snorkel mast.

Ballast trim system 4000 ton submarine

To conclude the final ballast trim system for the 4000 ton submarine consists first of all of a trim system with a capacity of 0.060 m^3/s. The best option is to use a centrifugal pump and transfer water between two tanks. The centrifugal pump does not have to overcome a very large head in case of trimming. This is because the pressure difference that needs to be overcome mainly depends on the pressure losses in the trim system's piping.

For the compensation system, a plunger pump is needed. The plunger pump must be able to deliver enough head to overcome the static pressure at large depths. With a capacity of 0.10 m^3/s the plunger pump is able to keep correct depth. Due to the high fluid-borne noise produced by the plunger pump, noise reduction measurements are needed.

For the hover system it is best to use two dedicated hover systems. For safety reasons, a pre-pressurised tank is needed to compensate for the change in density. For second order wave effects, the snorkel mast can be used to maintain depth while being close to the surface.

The final system sketch of the ballast trim system of the 4000 ton submarine can be found in appendix 4.

Ballast trim system 2000 ton submarine

The trim system of the 2000 ton submarine uses a centrifugal pump to displace water between the trim tanks. To maintain longitudinal trim, the capacity of this pump should be 0.079 m^3/s. The delivered pressure of the pump can be very low since there is no large static pressure difference to overcome.

A plunger pump is used in the compensation system. To maintain depth, a flow of 0.12 m^3/s is needed. The pressure delivered by the pump mainly depends on the depth to which the submarine can dive. The noise however produced by the plunger pump must be reduced.

For the 2000 ton submarine, a dedicated hover system must be installed. On the submarine already a snorkel is installed. This snorkel is used to make sure that when a change in density occurs, the submarine is able to keep depth. This snorkel can also be used when operating close to the surface to compensate for the second order effects of waves. The final system lay-out can be found in appendix 4.

9
Discussion and recommendations

A comparison was made based on actual pump curves, with a quadratic pump curve assumed. This results in a low pump efficiency when the pump is operated far from its best working point. This could be seen especially when using the centrifugal pump for pumping out water from the compensation tank. All the pumps studied are as variable speed pumps. This means that the flow of the pump system is controlled by the speed of the pump. It is expected that this results in higher efficiencies. However when having a large system head, the operational point of the pump shifts. Due to this shift, the efficiency decreases very quickly. Perhaps when using by-passing, draining delivered flow from the pump to control the flow through the system, the efficiency of the centrifugal pump is better than when using the variable speed.

Another effect which is not taken into account in this research is the need for a minimum flow of the centrifugal pump. Centrifugal pumps are flow machines. Therefore a flow is needed to increase the pressure. Usually a manufacturer gives the minimum flow for a pump. This flow is based on tests. By having this minimum flow, the user makes sure the pump will not be damaged. For now the flow equals the demand. The effects of having a minimum flow should be studied in order to make a better comparison between the different pumps.

To model a plunger pump, an existing smaller plunger pump is scaled up to be able to deliver enough flow for the compensation and hover system. This is because no data of plunger pumps with the needed size were available. It needs to be validated if the assumptions made in scaling the losses are correct. This is not done in this research because due to the lack of information. When having a plunger pump with a large flow, the losses can be measured. Especially when using Dorey's model, the measurement data can be compared to the scaled-up version of the plunger pumps which is used here. The leakage losses in the piston pump are very small. This is understandable when looking at the equations given by Dorey. The leakage of the plunger pump depends on the delivered pressure. The delivered pressure is in this case lower than what the plunger pump could deliver in theory. This results in very little leakage losses in the pump. The effect of scaling up the pump is that this leakage losses will be increased. This needs to be studied.

For now an estimation is made of the sound power levels of the different actuators. When possible, it is better to measure the real actuator sound power level and split this total sound power level into the three main sound propagations: airborne, fluid-borne and structural-borne. In submarine specifications usually a maximum airborne noise level is given. However we have seen that for the signature of the submarine, not only airborne noise is important. Fluid-born noise is as important as air-borne noise in the subject of submarines. A estimation method for the fluid-borne noise would therefore be a good addition to this work. Structure-borne noise is usually easier to reduce by applying dampers between the source and the structure.

Within the disturbances that are set up, the disembarkation of special forces will result in the largest pitching moment as well as the largest heave force. The pumps are chosen based on these disturbances. For the trim it is seen that the special forces walking towards the multi-mission portal results in the largest moment on board of the submarine. It may be chosen to allow these special forces to only walk when the submarine is sailing so that the hydroplanes can be used to trim the submarine. If the special forces only walk when the submarine is sailing, the trim pump can be smaller. The same is true for other submariners walking on board.

When having the submarine at zero speed, there can be restriction for walking on board. If so the pumps can also be smaller and this will result in less noise since the trim pumps do not have to continuously pump ballast water. Therefore regulations are very important when considering the ballast trim system. Regulation on board for the specific tasks of the submarine should be taken into account when designing a ballast trim system.

For now it was also assumed that the multi-mission portal was filled with water from the sea. If a dedicated tank, as seen in chapter 3 for the astute class torpedo tube, is made the disturbance due to the filling of the multi-mission portal can be reduced. In advance a dedicated tank located above the multi-mission portal can be filled with water. This way the trim and weight can already be controlled in conjunction with the disembarkation of the special forces. When the special forces enter the multi-mission portal, the portal can be filled from the dedicated tank with no extra moment or weight change due to the filling of the portal.

One of the options studied for compensating consists of two centrifugal pumps in series. It was concluded that this option was not the best compared to the other studied options for the compensation system. When having smaller centrifugal pumps a system can be designed in which the pumps can work in series and parallel. This allows a broader operational profile; At low depths the pumps can be set parallel allowing high flows for compensation. When at large depths the pumps can be switched in series allowing a high pressure head. It can be studied how this set-up will perform.

The studied depth of the submarine is for this thesis set to 300 meters. Based on this depth the simulations are done and the results are therefore also based on a operational depth of 300 meters. The real operational depth of a submarine is highly classified, and therefore it can not be concluded whether the proposed ballast trim system will be well operational at the real operational depth. Therefore to see the performance of the submarine at the real operational depth, new simulations must be done. It may be that another depth will result in different systems. For example, when the depth is increased, the pressure the pump has to overcome must be larger.

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A

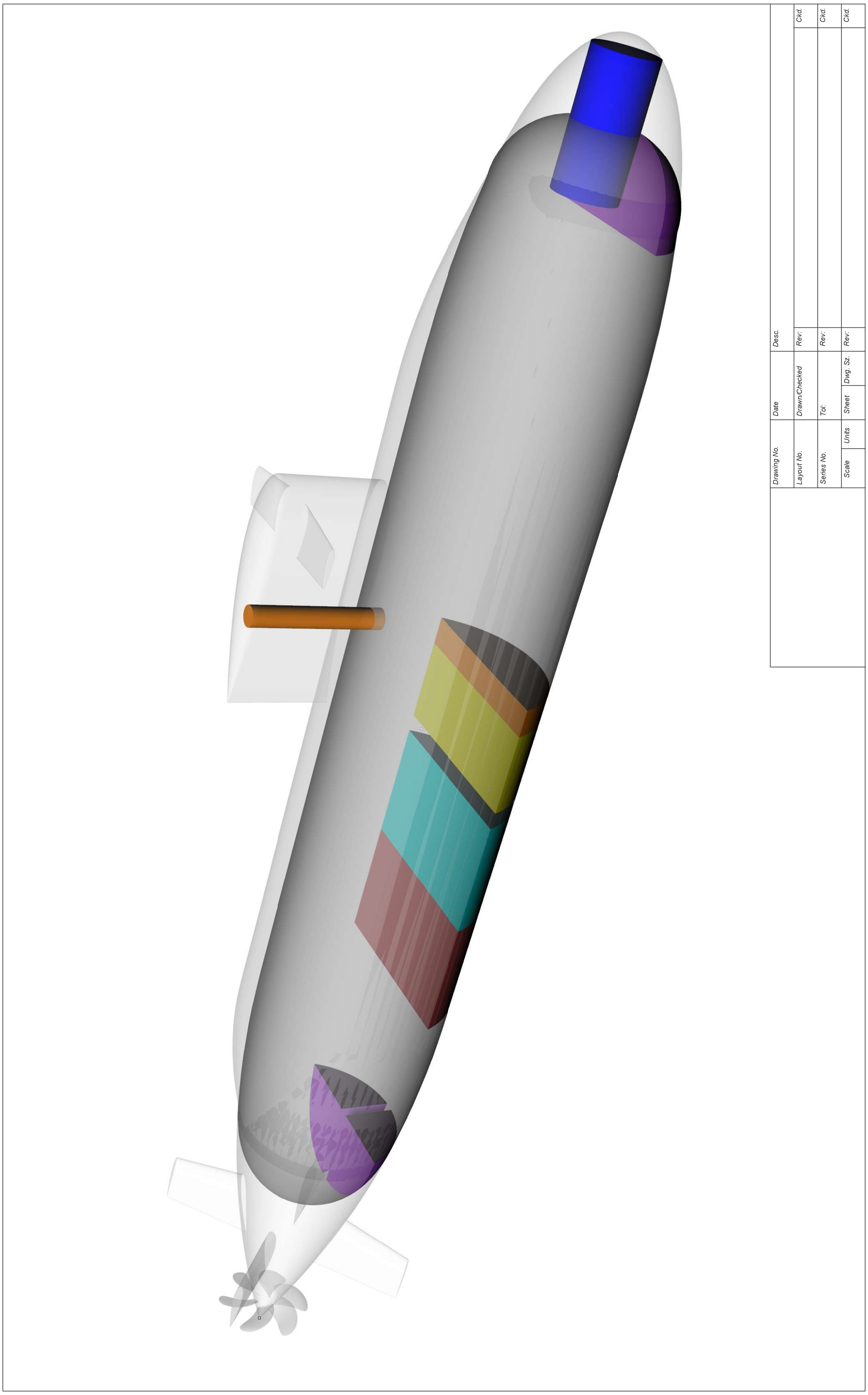
Walrus class

Initially left out

B
Rhino model

For this research there was a need for a generic submarine hull. The hull used is the BB2 Joubert hull. This hull is used in earlier research by Marin. The standard Rhino drawings can be downloaded from the Marin website. This model is used and items are added to the model which were needed for the research to a ballast trim system on board of submarines. The renders of these added information can be found below. Figure B.1 gives the legend for the items that are added in this model.

Figure B.1: Legend for renders BB2 Joubert hull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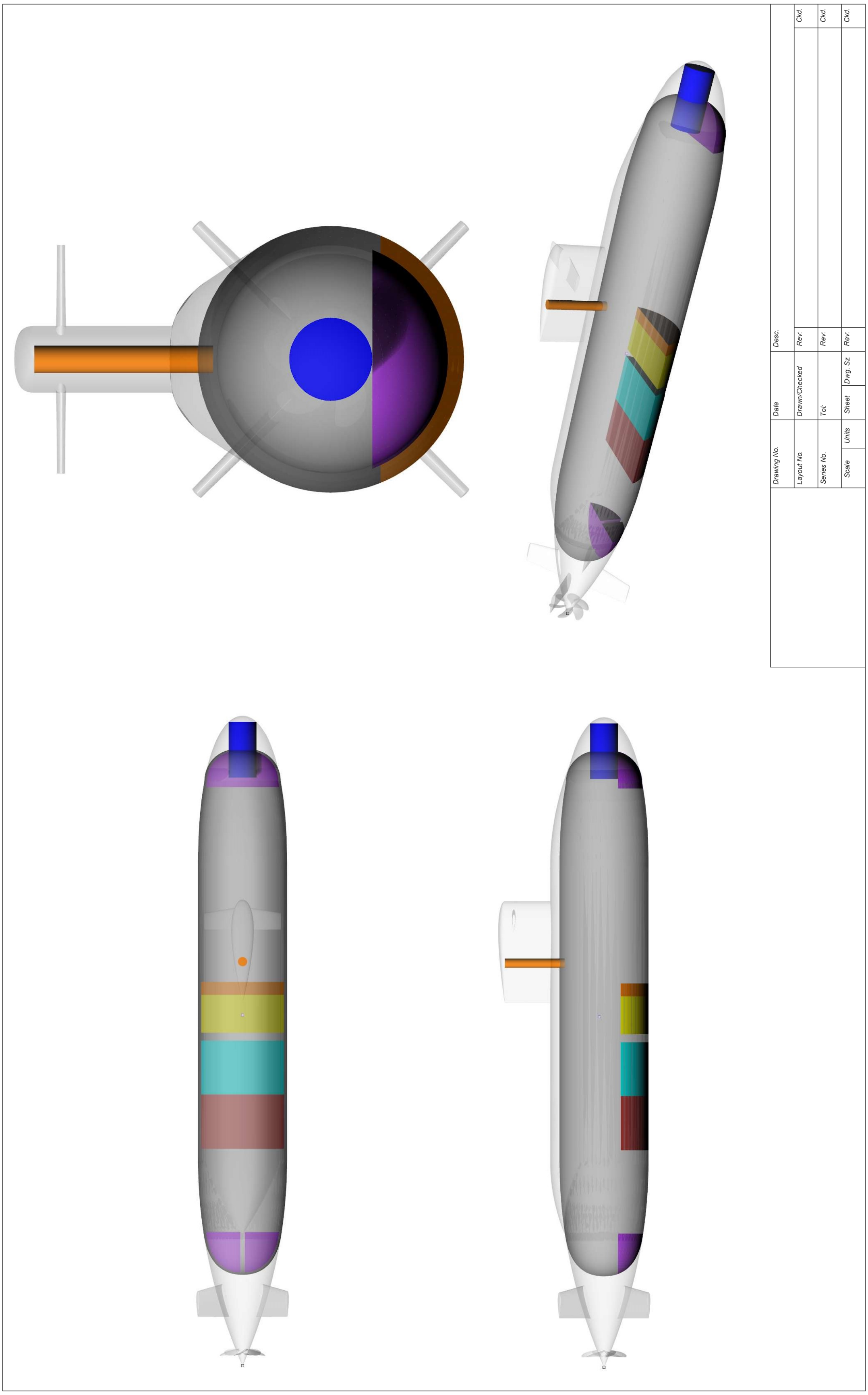
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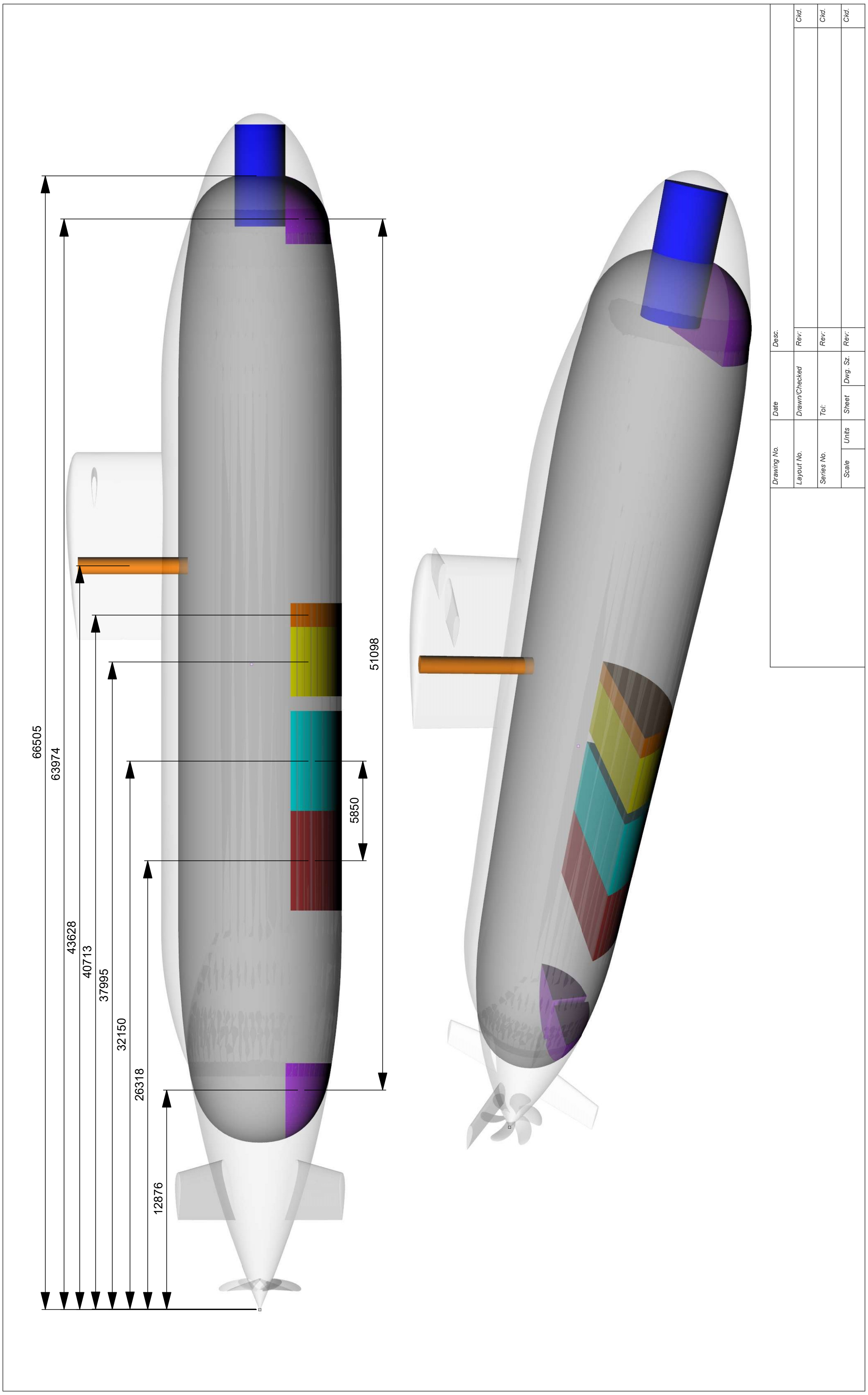
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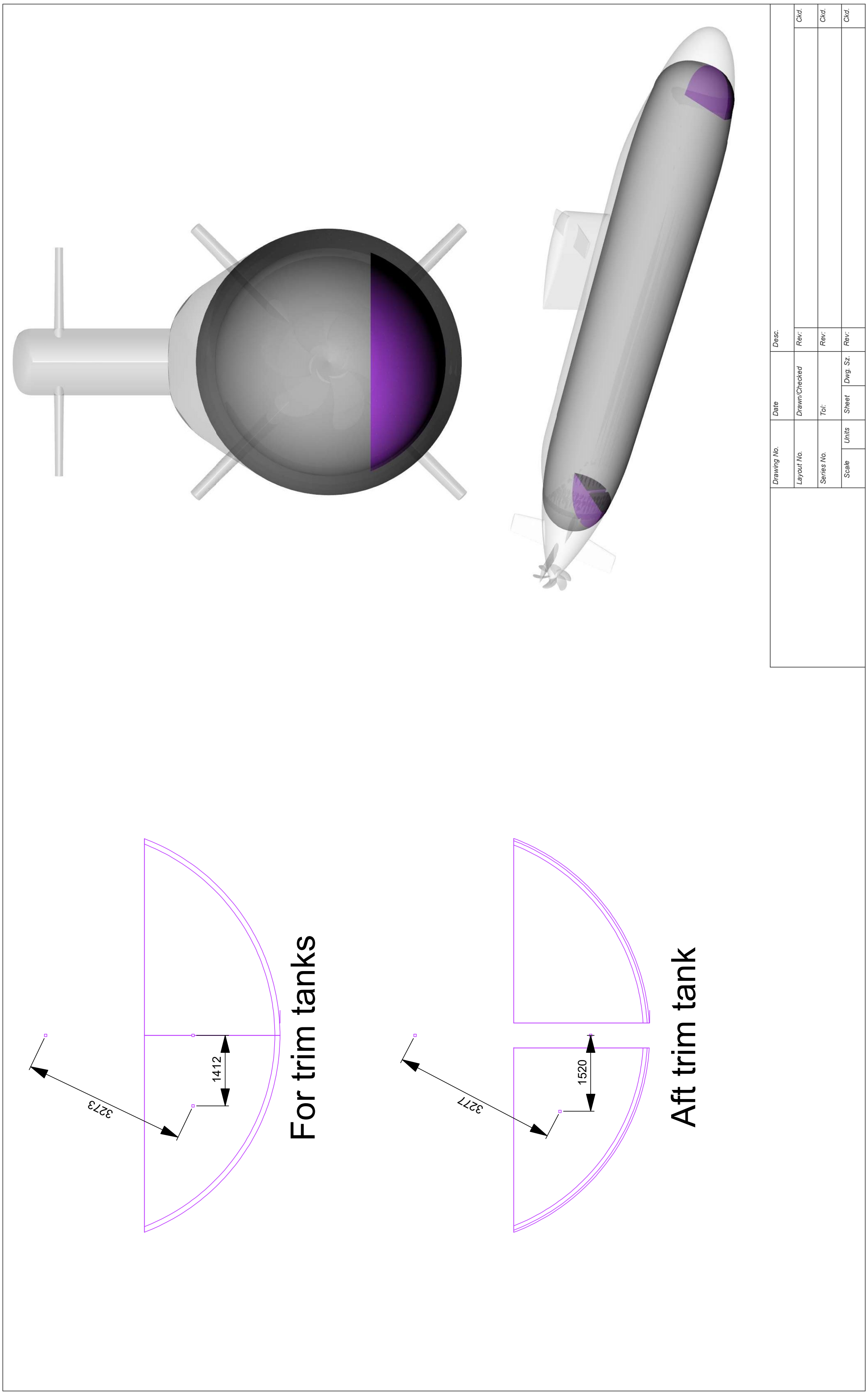
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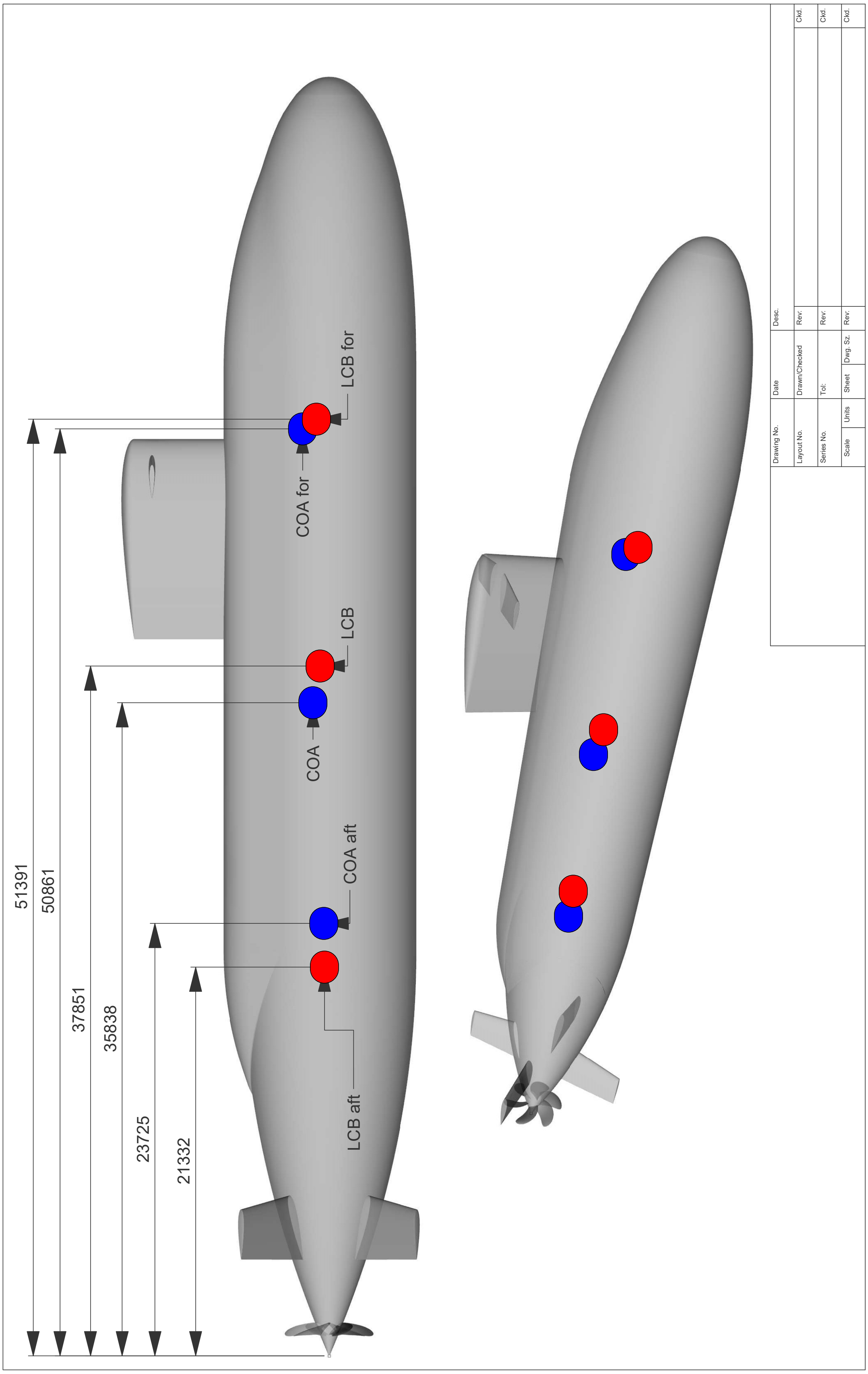
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C

Options special forces escape

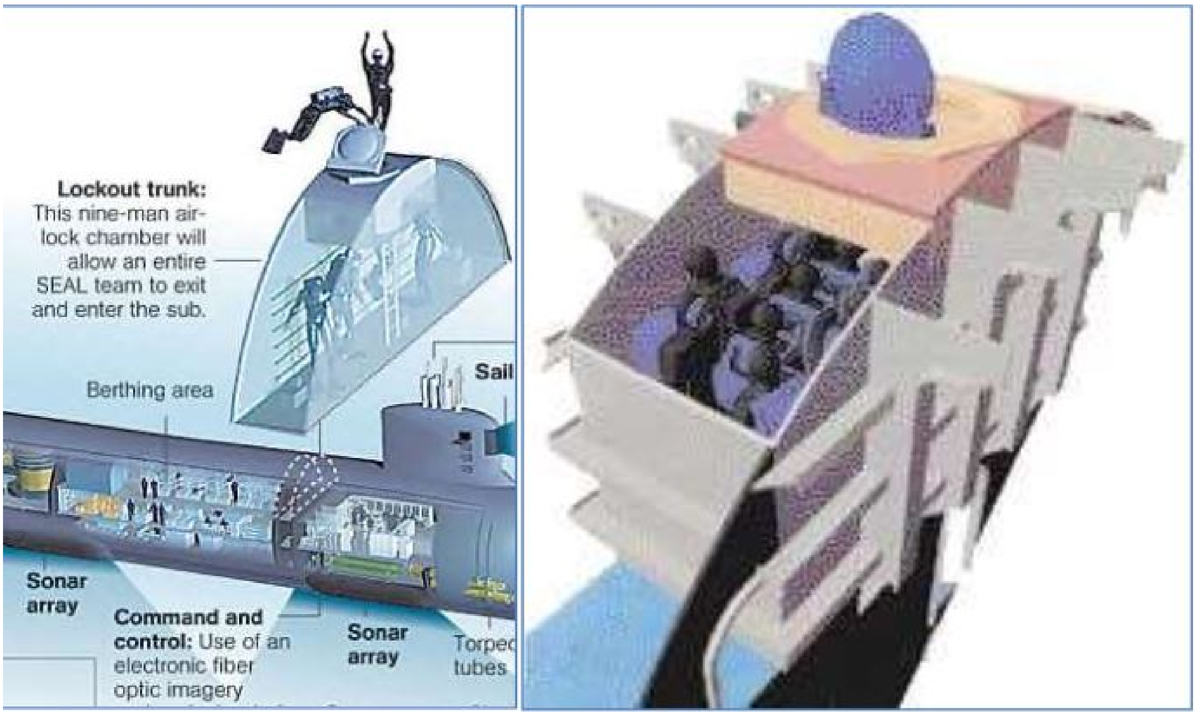


Figure C.1: Overview of the cofferdam as installed on the Virginia class

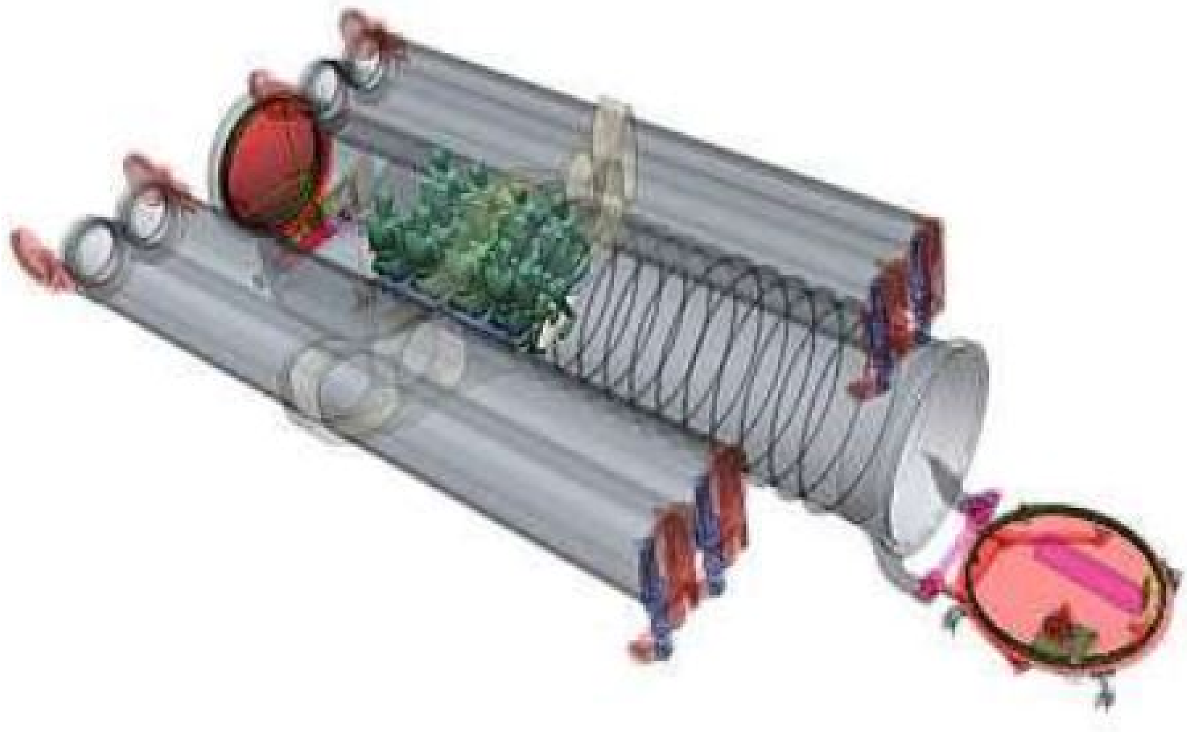


Figure C.2: Overview of the multimission portal

Figure C.3: Overview of the multimission portal

Figure C.4: Dry dock shelter

Figure C.5: Veritcal escape possibility

D

Load cases

Table D.1: Moments of disturbances considered

	Transit snorkeling	**Transit AIP**	**Dropping off SF**	**Setting out AUV**	**Anchoring**
Submariners walking	x	x	x	x	x
Water usage	x	x	x	x	x
Escape SF			x		
Walking SF			x		
AUV				x	
Anchor					x

Table D.2: Forces of disturbances considered

	Transit snorkeling	**Transit AIP**	**Dropping off SF**	**Setting out AUV**	**Anchoring**
Fuel	x				
Fuel stirling		x			
LOx stirling		x			
Second order effects	x		x		
Density	x	x	x	x	x
Mast	x	x	x	x	
Snorkel	x				
Escape SF			x		
AUV				x	
Anchor					x

D.1. Profiles submarine 1

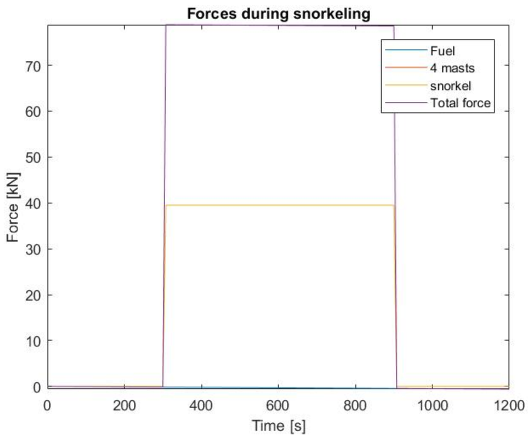


Figure D.1: Forces while snorkelling

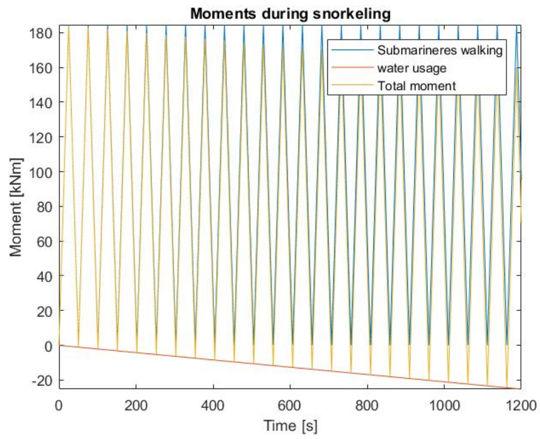


Figure D.2: Moments while snorkelling

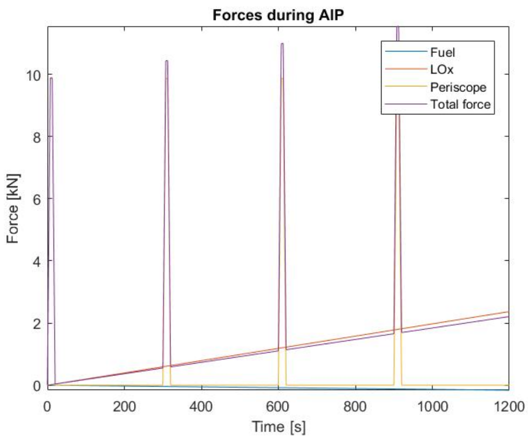


Figure D.3: Forces while transit on AIP

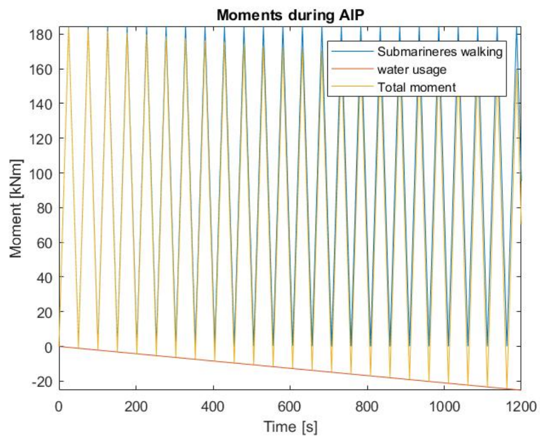


Figure D.4: Moments while transit on AIP

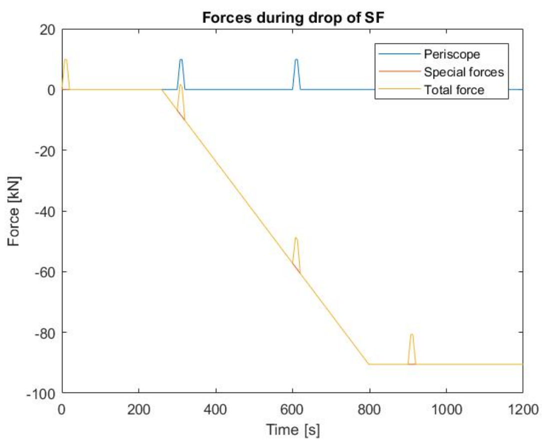


Figure D.5: Forces while disembarking SF

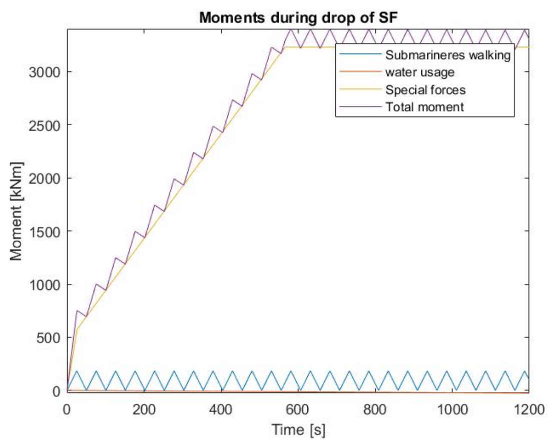


Figure D.6: Moments while disembarking SF

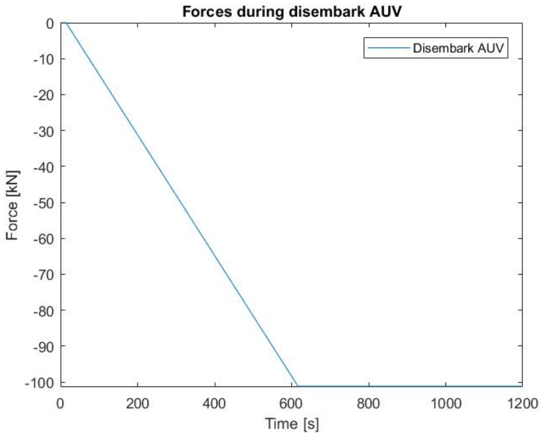


Figure D.7: Forces while disembarking AUV

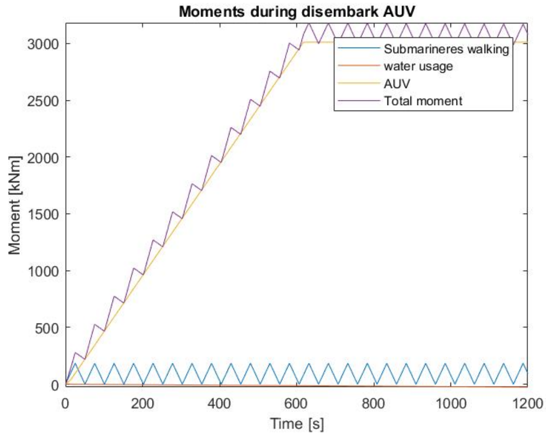


Figure D.8: Moments while disembarking AUV

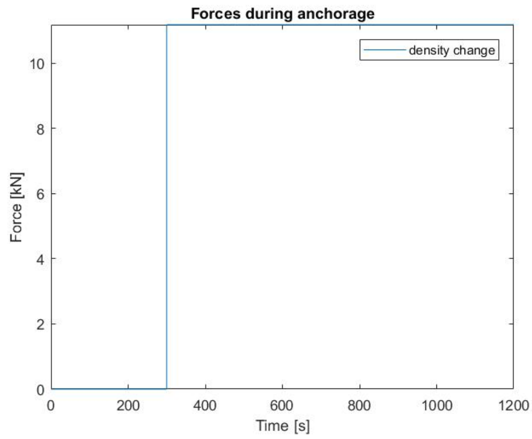


Figure D.9: Forces while disembarking SF

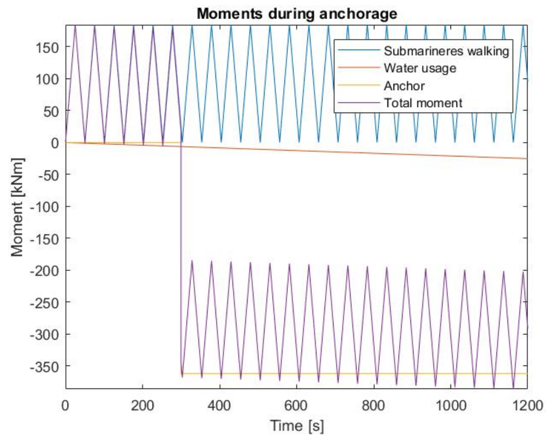


Figure D.10: Moments while disembarking SF

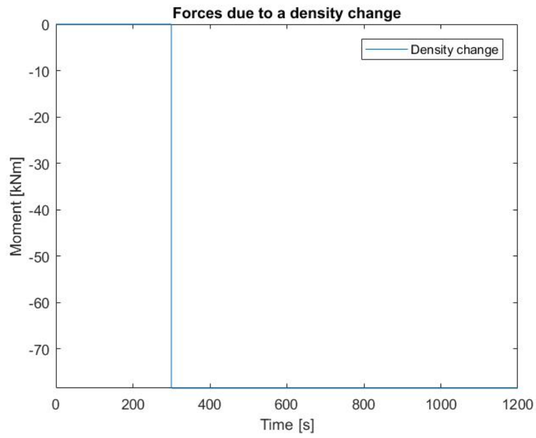


Figure D.11: Forces due to change of density

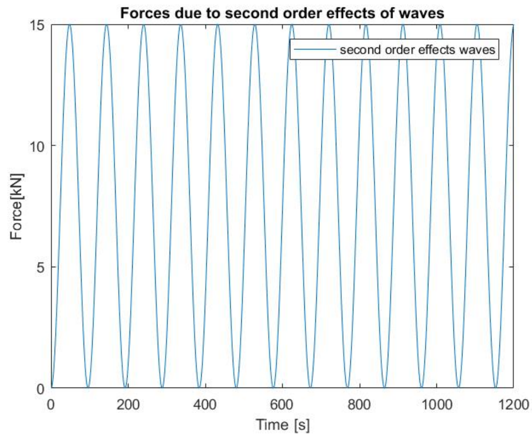


Figure D.12: Forces due to second order effects waves

D.2. Profiles submarine 2

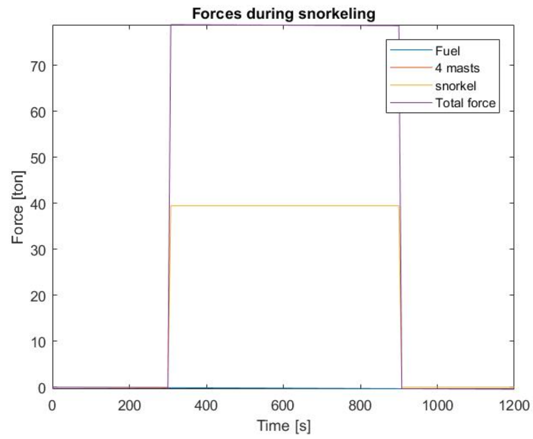


Figure D.13: Forces while snorkelling

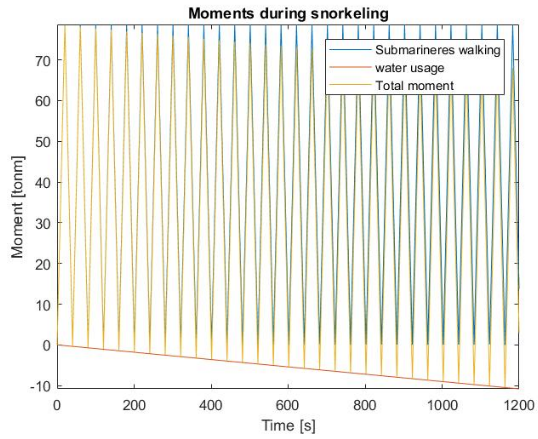


Figure D.14: Moments while snorkelling

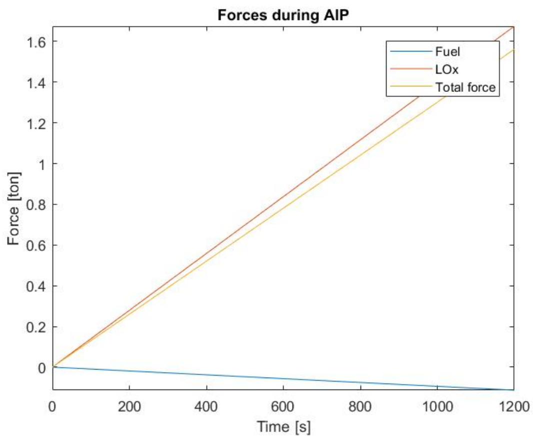


Figure D.15: Forces while transit on AIP

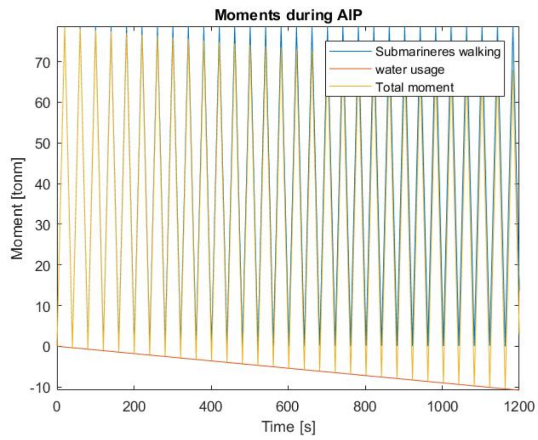


Figure D.16: Moments while transit on AIP

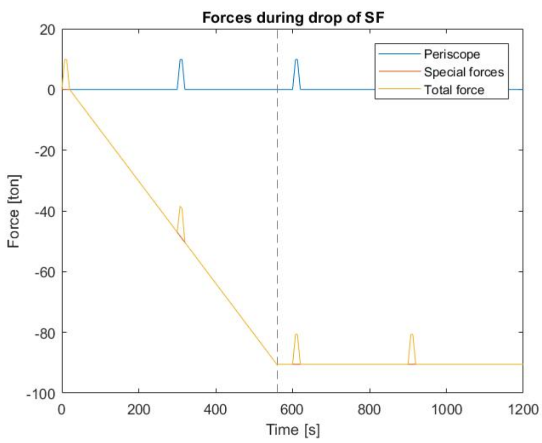


Figure D.17: Forces while disembarking SF

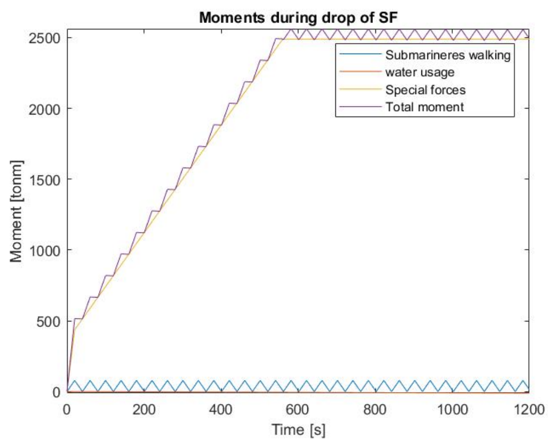


Figure D.18: Moments while disembarking SF

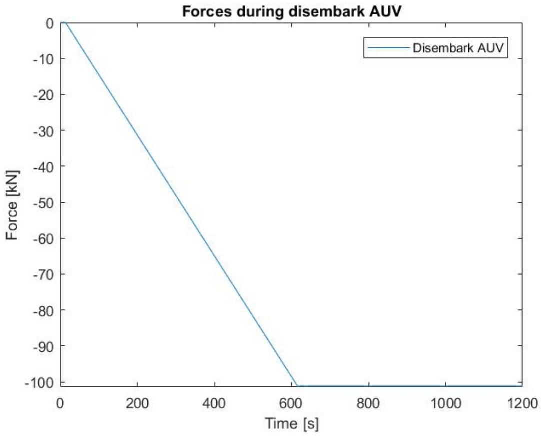


Figure D.19: Forces while disembarking AUV

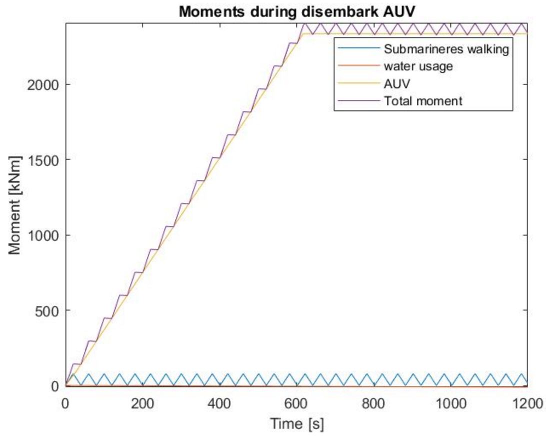


Figure D.20: Moments while disembarking AUV

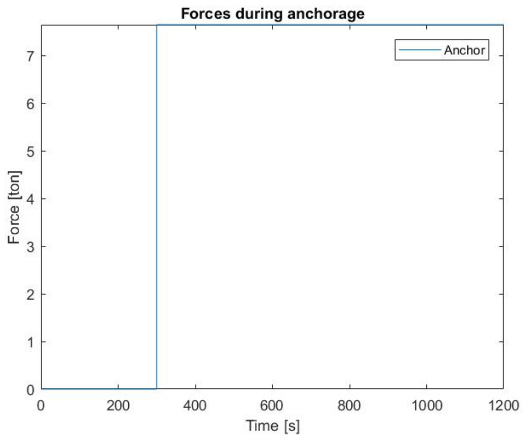


Figure D.21: Forces while disembarking SF

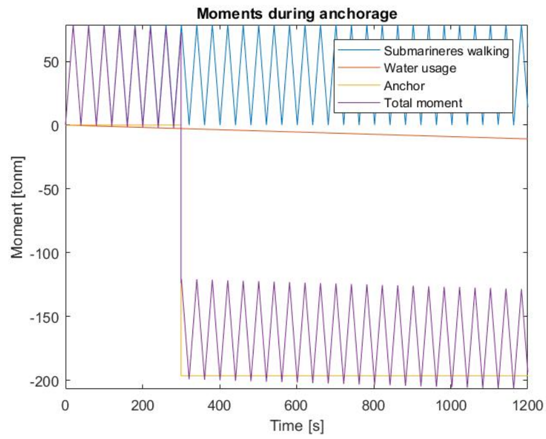


Figure D.22: Moments while disembarking SF

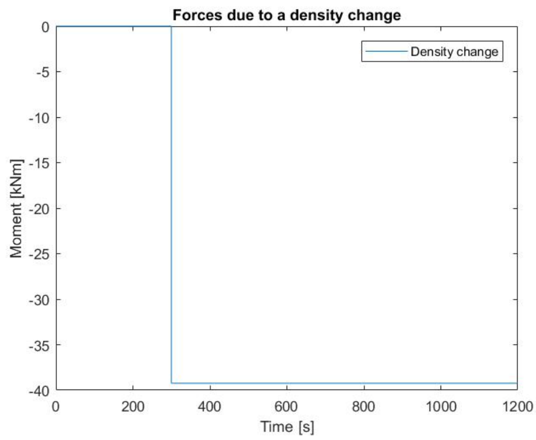


Figure D.23: Forces due to change of density

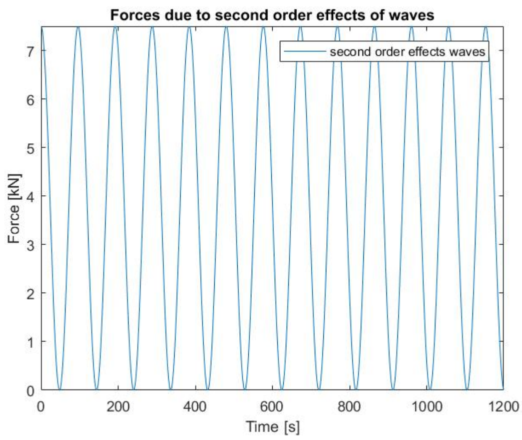


Figure D.24: Forces due to second order effects waves

E

Validation models

This appendix will look into the models made in Simulink and Matlab. These models contain idealisation which needs to be validated by actual data. This is done in different ways for the different actuators. Data of manufacturers is used to see if the results of the models are accurate enough to use for this research. Other models are compared to earlier results of other widely used methods. The different models that are validated in this appendix are the pipe flow model, the centrifugal pump model and the plunger pump model.

E.1. Pipe flow

In the Simulink model that made to model the pipe flow, the dynamic resistance factor for turbulent flow is determined with the Haaland equation. This is done to speed up the model and avoid algebraic loops. The Colebrook-White equation can only be solved iterative or by using the moody diagram. The Haaland equation is a approximation of the Colebrook-White equation and can be solved algebraic. With the use of the Haaland equation there will be a small error in the friction factor for the dynamic flow in the pipe. The absolute values for the pipe friction factors for different Reynolds numbers calculated with both methods can be found in figure E.1. The error in percentages for different Reynolds numbers can be found in E.2. In here a pipe is used with a diameter of 0.2 meters and a roughness coefficient of 0.02.

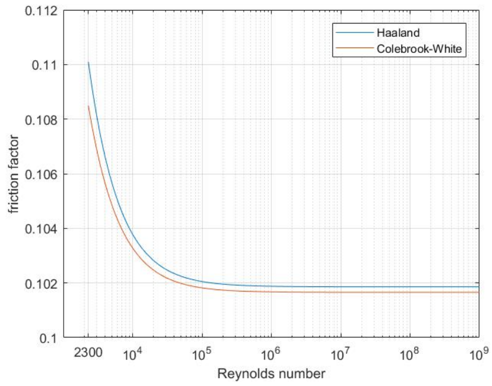


Figure E.1: Friction factors dynamic pipe flow losses

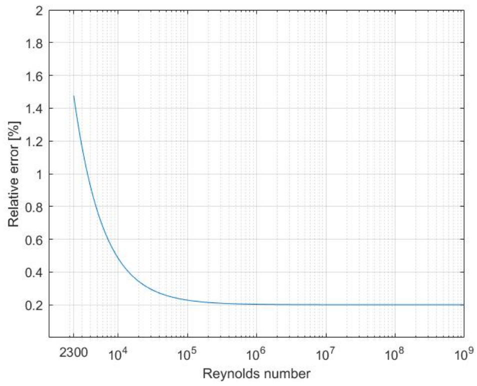


Figure E.2: relative error Haaland equation

As can be seen in figure E.1 and figure E.2 the error between the Haaland equation and the Colebrook-White equation is small. Therefore using the Haaland equation to calculate the friction factor for the dynamic losses seems valid.

To validate the model of the pipe flow, an exercise is used. This exercise is an exercise that is commonly used at the TU Delft. Therefore it can be assumed that the solutions of the exercise are precise and can be used for validation of the pipe flow model The pipe flow from the exercise is a cooling system that consist of

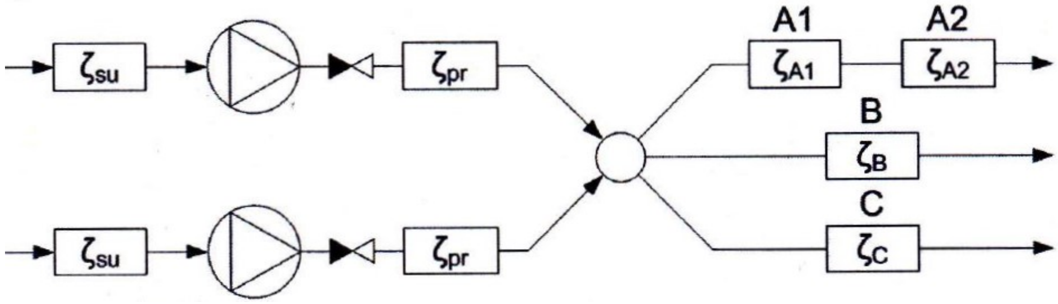


Figure E.3: Cooling system exercise

Table E.1: Excercise conversion for simulink model

	D[**m**]	Δz_{loss}[**m**]	$\dot{V}$[m^3/h]	**v**[**m/s**]	ζ[−]
A1	0.05	5	10	1.41	49.34
A2	0.05	3	10	1.41	29.61
B	0.06	8	15	1.47	72.64
C	0.07	8	20	1.44	75.69
$pipe_{suction}$	0.09	1	45	1.96	5.11
$pipe_{pressure}$	0.09	5	45	1.96	25.54

multiple pipes and two pumps. All this pipes have different particulars and therefore the flow differs from pipe to pipe. Figure E.3 shows the system. The fluid that is pumped has a density of 1016 kg/m^3.

The first step in the model is to calculate the system resistance of every pipe. This is done in the exercise by giving the head loss at a certain flow. This can be written as a friction factor, ζ as is used in the Simulink model. In the exercise the flow losses are quadratic. This conversion is shown in table 7.11.

These friction factors can now be implemented in the made model. The results from the model can than be compared to the answers of the exercise. The questions of the exercise that are done are:

1. Determine the pumps flow and head with one pump in operation
2. Determine the volume flows through the branches A1, A2, B, C
3. Determine the volume flow through the system with both pumps in operation

The answers of the exercise are compared to the outcome of the model. This is shown in table E.2.

Table E.2: Answers of the exercise versus the outcome of the model

	δz [m]		**Q [m³/h]**	
exercise	*answer*	*model*	*answers*	*model*
2	18	17.7	51	50.25
3			11.3 17.0 22.7	11.16 16.75 22.34
5	21	21.22	67	66.7

There can be seen that there are small differences between the answers of the exercise and the outcome of the model. These differences are because of rounding errors. also there is a small error in the pipe flow resistance since the model also takes into account the dynamic pressure losses of the pipe based on the speed of the fluid through the pipe. Therefore the resistance of the pipes is slightly higher than in the exercise.

With this exercise there can also be seen that the actuator can be placed correctly inside the piping system. The model will find the working point from which the flow and the pressure head can be determined.

E.2. Centrifugal pump

The centrifugal pump is modelled assuming a quadratic pump characteristic. An ideal centrifugal pump will have in the ideal situation indeed a quadratic pump curve. In pump manufacturers manuals however there can be found pump curves that are far from quadratic. There could be a small error in assuming the all centrifugal pump being quadratic. Several pump curves from the KSB manual are compared to each other. The centrifugal pump model needs as input a few working point of the centrifugal pump from the actual pump curve. A third order polygonal will be fitted through these points. It is important to see if this polygonal can describe the pump characteristics accurate enough to use in the Simulink model. This is shown in the figures E.4 to E.7.

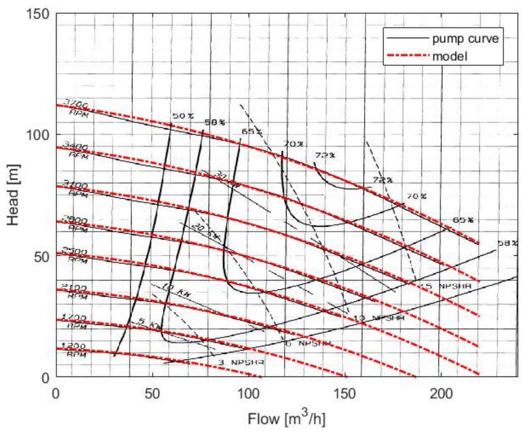


Figure E.4: Model output vs specification centrifugal pump 1

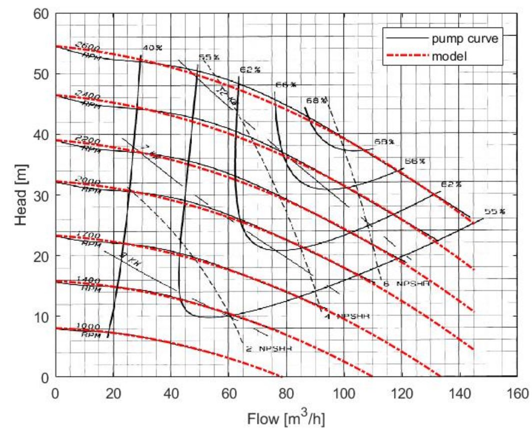


Figure E.5: Model output vs specification centrifugal pump 2

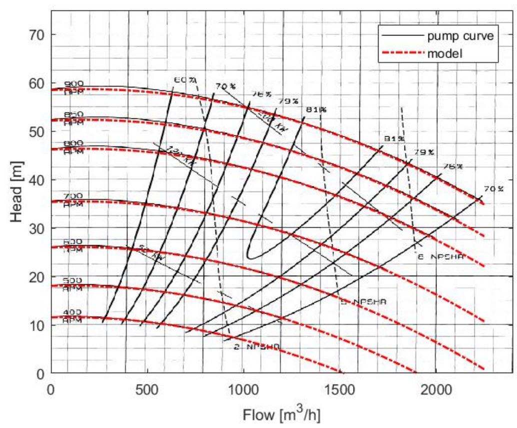


Figure E.6: Model output vs specification centrifugal pump 3

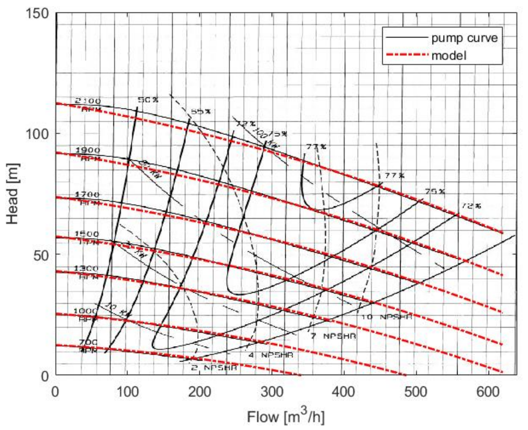


Figure E.7: Model output vs specification centrifugal pump 4

A centrifugal pump in a system has its working point in the place where the pump curve intersects with the system curve. In this working point the energy losses due to the system are equal to the delivered energy by the pump. The pump curve is obtained from manufacturers. The system curve can be calculated by adding up all resistances. That the working point of the pump, the intersection between the pump curve and system curve, can be determined is already shown in previous section.

E.3. Plunger pump

The plunger pump is modelled according to the Dorey pump model. This pump model can be fitted on existing pump curves. The used data for the fitting comes the Bosch Rexroth A4VSO1000 piston pump. The plunger pump model is fitted on existing an existing plunger pump. The result from the curve fitting can be compared to the manufacturers data. This is done by plotting The pump curve obtained by the curve fitting over the pump manufacturers data. This is shown in figure E.8. It can be seen that the dorey model is very accurate in describing the pump curve. With a few points the pump curve can be described accurately. This

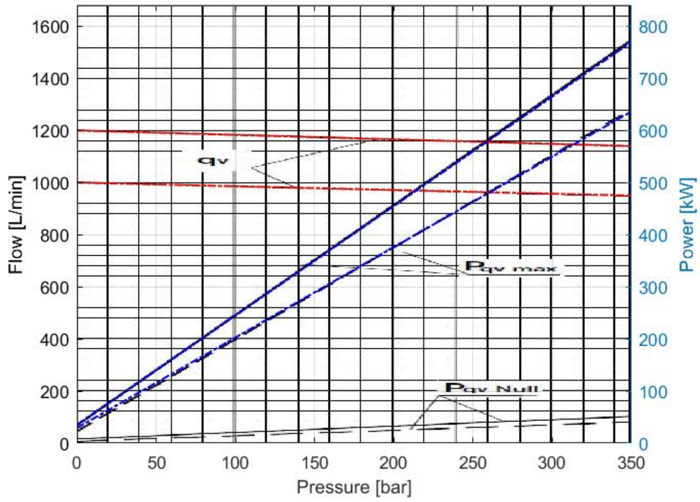


Figure E.8: Model output vs specification piston pump

is also a result of the simplicity of the plunger pump it self. The head flow relation of the plunger is almost a first order relation. The same is valid for the power input of the plunger pump.

The working point of the piston pump can be determined in the same way as for the centrifugal pump. There is already seen in the first section that solving the equations is accurate enough.

E.4. Submarine

The model made for the submarine is not validated. This due to a lack of data. When possible it would be great to validate the model. This because the model made for the submarine is very simple and easy to use in an early stage of designing a submarine. Assuming the submarine to be a perfect ellipsoid simplifies the calculations but it it possible that the error made is larger than desired.

F

System models

In this appendix the made models will be discussed. First the legend for the system diagrams

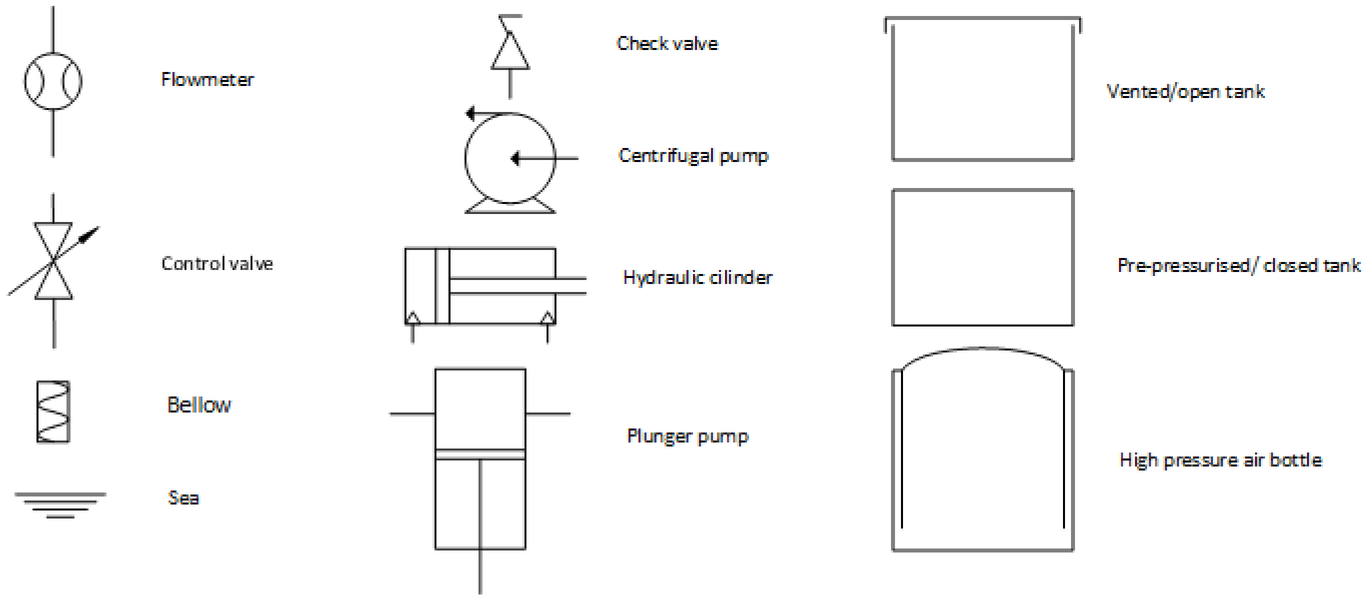


Figure F.1: Legend for the system diagrams

F.1. Trim system

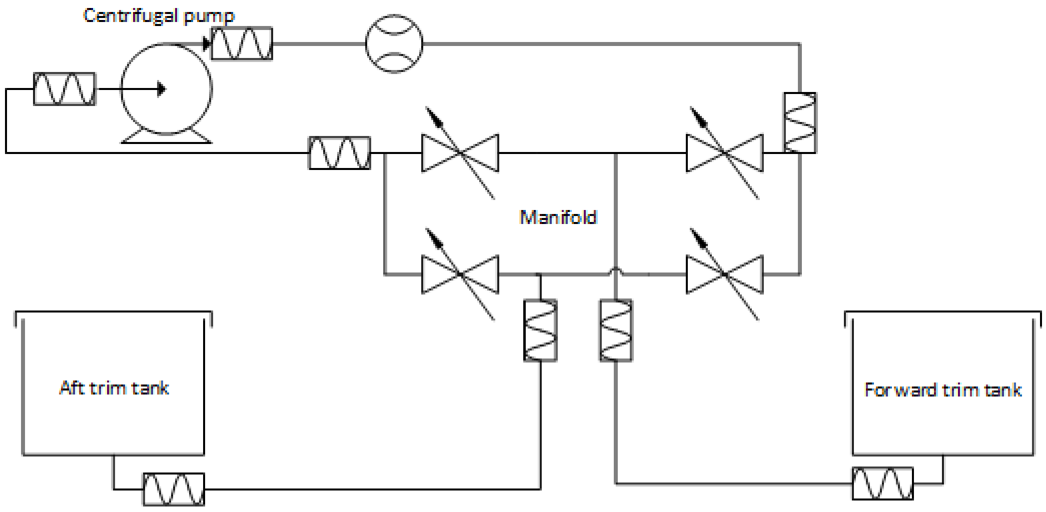


Figure F.2: Trim system with centrifugal pump

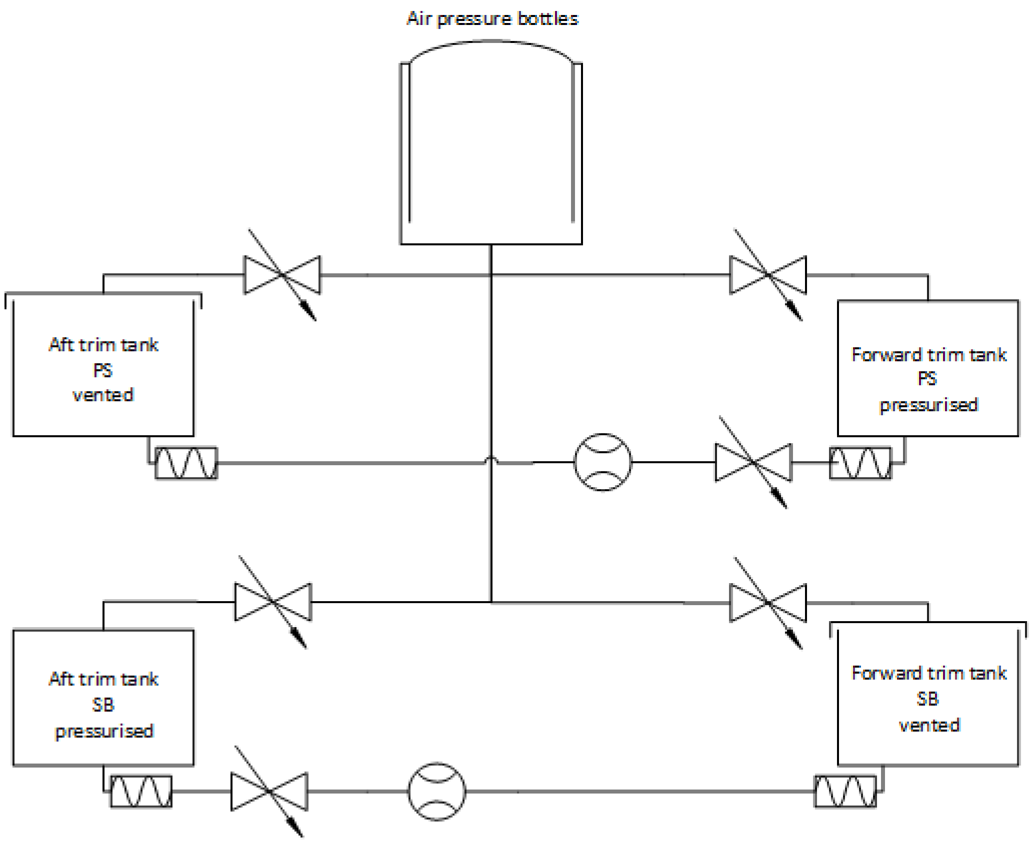


Figure F.3: Trim system with pre-pressurised tanks

Table F.1: Hover system piping information

Centrifugal pump

	Item	Submarine 1	Submarine 2	Unit
Pipe suction	L_{pipe}	5.14	4.08	[m]
	D_{pipe}	0.24	0.26	[m]
	ε	0.02	0.02	[-]
	ζ_{extra}	23.52	23.52	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	1	2	2
exit tank	1	1	1
arcs	3	0.72	2.16
bellow	4	2	8
manifold	1	10.36	10.36
total			**23.52**

	Item	Submarine 1	Submarine 2	Unit
Pipe pressure	L_{pipe}	51.06	40.53	[m]
	D_{pipe}	0.24	0.26	[m]
	ε	0.02	0.02	[-]
	ζ_{extra}	24.42	24.42	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	0	2	0
entrance tank	1	1	1
arcs	3	0.72	2.16
bellow	4	2	8
manifold	1	10.36	10.36
check valve	1	2.9	2.9
total			**24.42**

Pre-pressurised tanks

	Item	Submarine 1	Submarine 2	Unit
Pipe suction	L_{pipe}	56.20	44.61	[m]
	D_{pipe}	0.24	0.26	[m]
	ε	0.02	0.02	[-]
	ζ_{extra}	19.64	19.64	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	1	1	1
entrance	1	1	1
arcs	2	0.72	1.44
valve	1	8.2	8.2
bellow	4	2	8
total			**19.64**

F.2. Compensation system

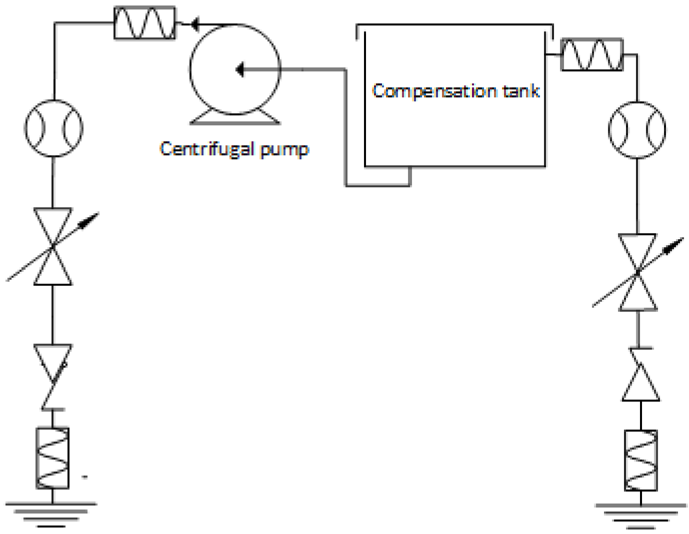


Figure F.4: compensation system with large centrifugal pump

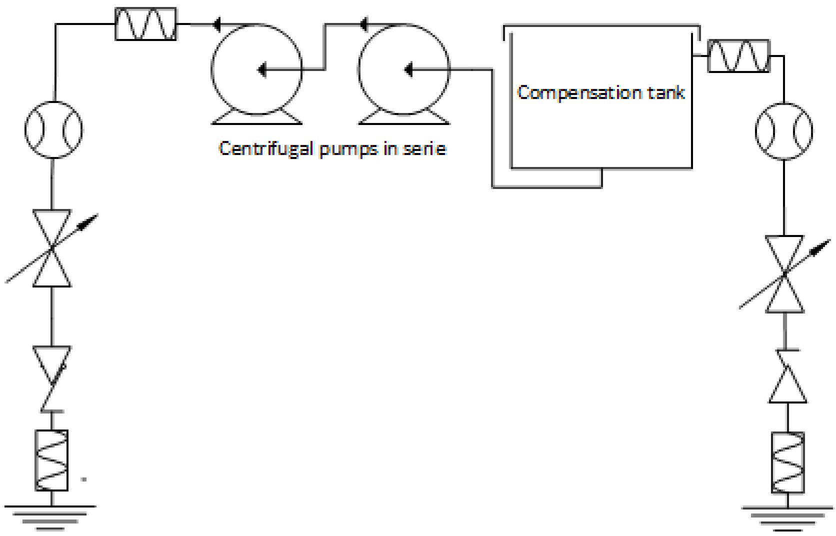


Figure F.5: compensation system with centrifugal pumps in series

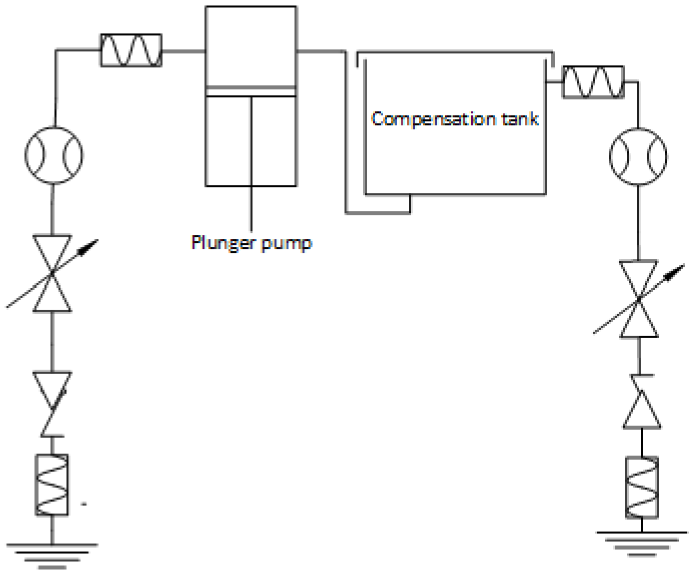


Figure F.6: compensation system with plunger pump

Table F.2: Compensation system piping information

Centrifugal pump series

Pipe suction

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	2.00	1.59	[m]
D_{pipe}	0.32	0.36	[m]
ε	0.02	0.02	[-]
ζ_{extra}	18.82	18.82	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	1	2	2
exit tank	1	1	1
arcs	1	0.72	0.72
bellow	2	2	4
check valve	1	2.9	2.9
valve	1	8.2	8.2
ζ_{extra}			**18.82**

Pipe pressure

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	22.50	17.85	[m]
D_{pipe}	0.32	0.36	[m]
ε	0.02	0.02	[-]
ζ_{extra}	23.98	23.98	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	0	2	0
entrance tank	1	1	1
arcs	4	0.72	2.88
bellow	4	2	8
valve	1	8.2	8.2
check valve	1	2.9	2.9
exit tank	1	1	1
ζ_{extra}			**23.98**

Centrifugal pump large

Pipe suction

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	2.00	1.59	[m]
D_{pipe}	0.32	0.36	[m]
ε	0.02	0.02	[-]
ζ_{extra}	18.82	18.82	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	1	2	2
exit tank	1	1	1
arcs	1	0.72	0.72
bellow	2	2	4
check valve	1	2.9	2.9
valve	1	8.2	8.2
ζ_{extra}			**18.82**

Pipe pressure

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	22.50	17.85	[m]
D_{pipe}	0.32	0.36	[m]
ε	0.02	0.02	[-]
ζ_{extra}	23.98	23.98	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	0	2	0
entrance tank	1	1	1
arcs	4	0.72	2.88
bellow	4	2	8
valve	1	8.2	8.2
check valve	1	2.9	2.9
exit tank	1	1	1
ζ_{extra}			**23.98**

Plunger pump

Pipe suction

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	2.00	1.59	[m]
D_{pipe}	0.32	0.36	[m]
ε	0.02	0.02	[-]
ζ_{extra}	18.82	18.82	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	1	2	2
exit tank	1	1	1
arcs	1	0.72	0.72
bellow	2	2	4
check valve	1	2.9	2.9
valve	1	8.2	8.2
ζ_{extra}			18.82

Pipe pressure

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	22.50	17.85	[m]
D_{pipe}	0.32	0.36	[m]
ε	0.02	0.02	[-]
ζ_{extra}	23.98	23.98	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	0	2	0
entrance tank	1	1	1
arcs	4	0.72	2.88
bellow	4	2	8
valve	1	8.2	8.2
check valve	1	2.9	2.9
exit tank	1	1	1
ζ_{extra}			**23.98**

F.3. Hover system

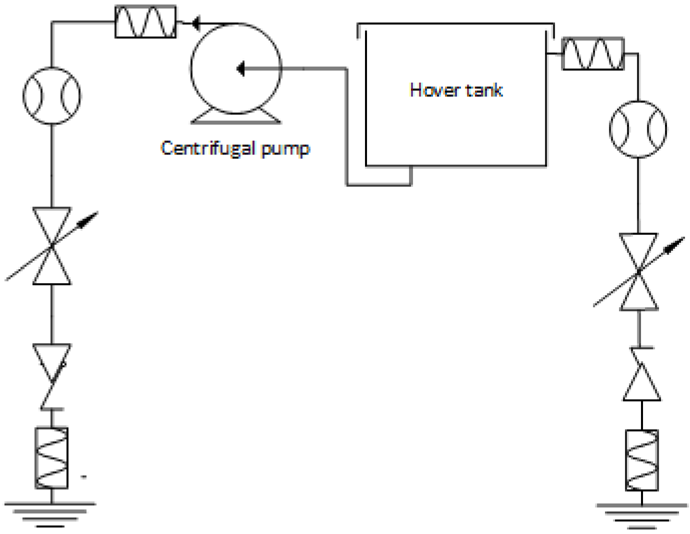


Figure F.7: Hover system with large centrifugal pump

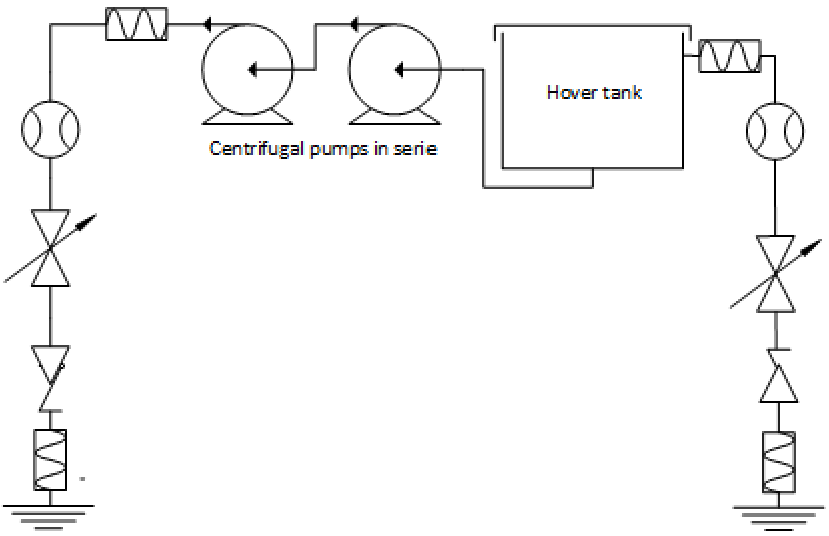


Figure F.8: Hover system with centrifugal pumps in series

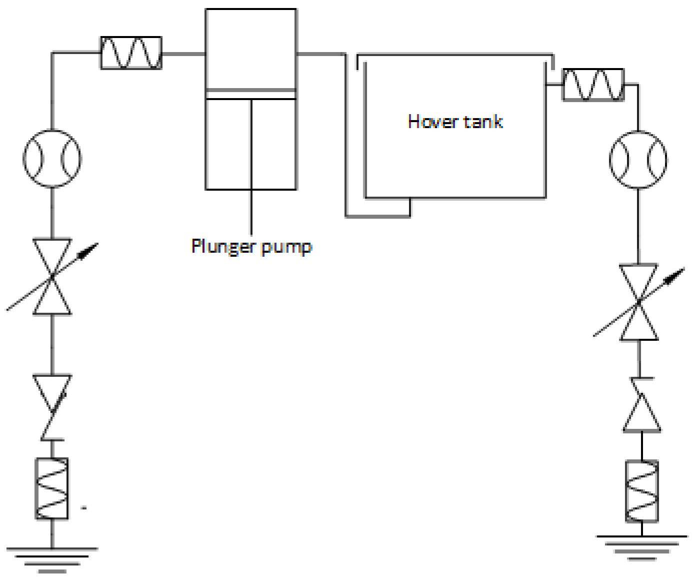


Figure F.9: Hover system with plunger pump

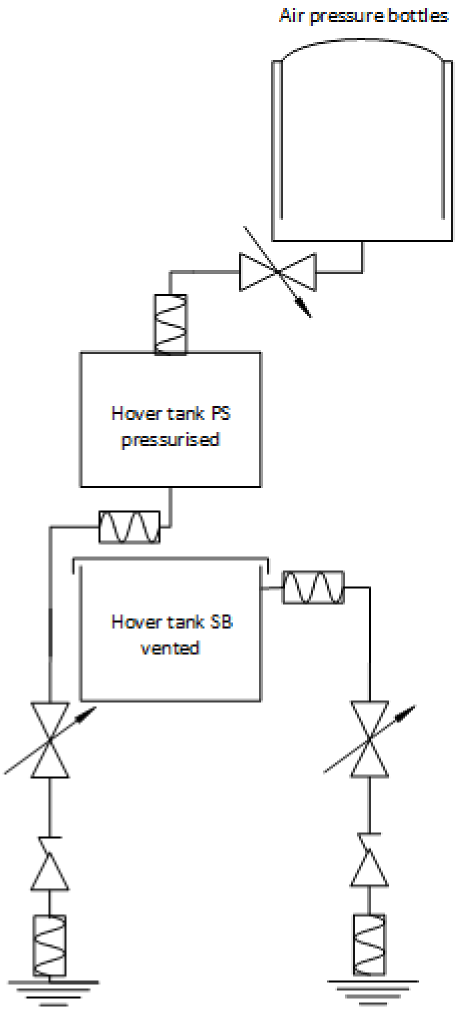


Figure F.10: Hover system with pre-pressurised tank

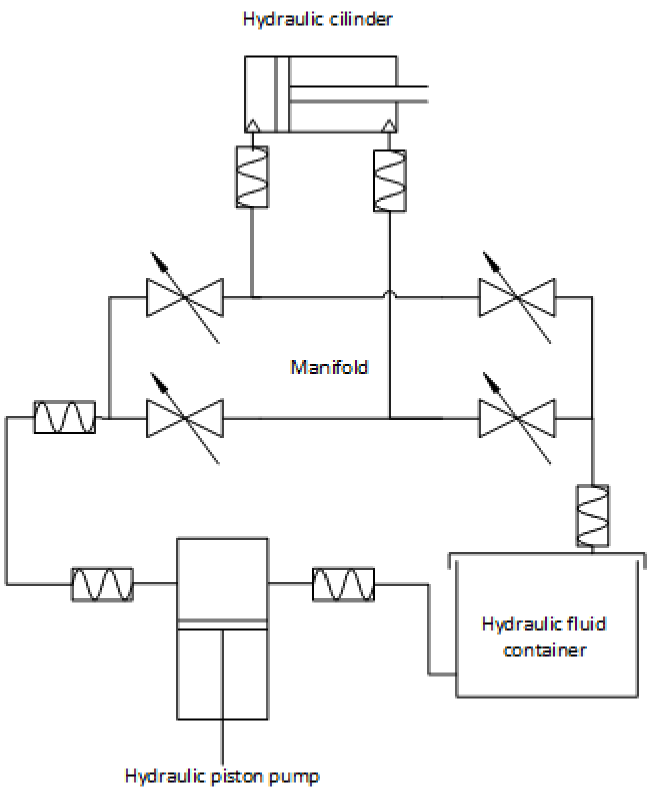


Figure F.11: Hover system with variable buoyancy system

Table F.3: Hover system piping information

Hydraulic system

Pipe suction

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	19.31	15.32	[m]
D_{pipe}	0.208	0.143	[m]
ε	0.02	0.02	[-]
ζ_{extra}	19.16	19.16	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	1	2	2
entrance tank	1	1	1
arcs	3	0.72	2.16
bellow	6	2	12
manifold	1	10.36	10.36
entrance cilinder	1	1	1
exit pump	1	1	1
ζ_{extra}			29.56

Pipe pressure

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	19.31	15.32	[m]
D_{pipe}	0.208	0.143	[m]
ε	0.02	1.02	[-]
ζ_{extra}	14.44	14.44	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	1	2	2
entrance tank	1	1	1
arcs	2	0.72	1.44
bellow	4	2	8
manifold	1	10.36	10.36
exit cilinder	1	1	1
entrance tank	1	1	1
ζ_{extra}			**23.98**

Pre-pressurised tanks

Pipe suction

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	2.00	1.58	[m]
D_{pipe}	0.235	0.235	[m]
ε	0.02	0.02	[-]
ζ_{extra}	18.82	18.82	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	1	2	2
entrance tank	1	1	1
arcs	1	0.72	0.72
bellow	2	2	4
valve	1	8.2	8.2
check valve	1	2.9	2.9
ζ_{extra}			**18.82**

Pipe pressure

Item	Submarine 1	Submarine 2	Unit
L_{pipe}	2.00	2.00	[m]
D_{pipe}	0.235	0.235	[m]
ε	0.02	0.02	[-]
ζ_{extra}	18.82	18.82	[-]

ζ_{extra}	#	ζ	ζ_{items}
flow meter	1	2	2
exit tank	1	1	1
arcs	1	0.72	0.72
bellow	2	2	4
valve	1	8.2	8.2
check valve	1	2.9	2.9
ζ_{extra}			**18.82**

F.4. Final model as ballast trim system

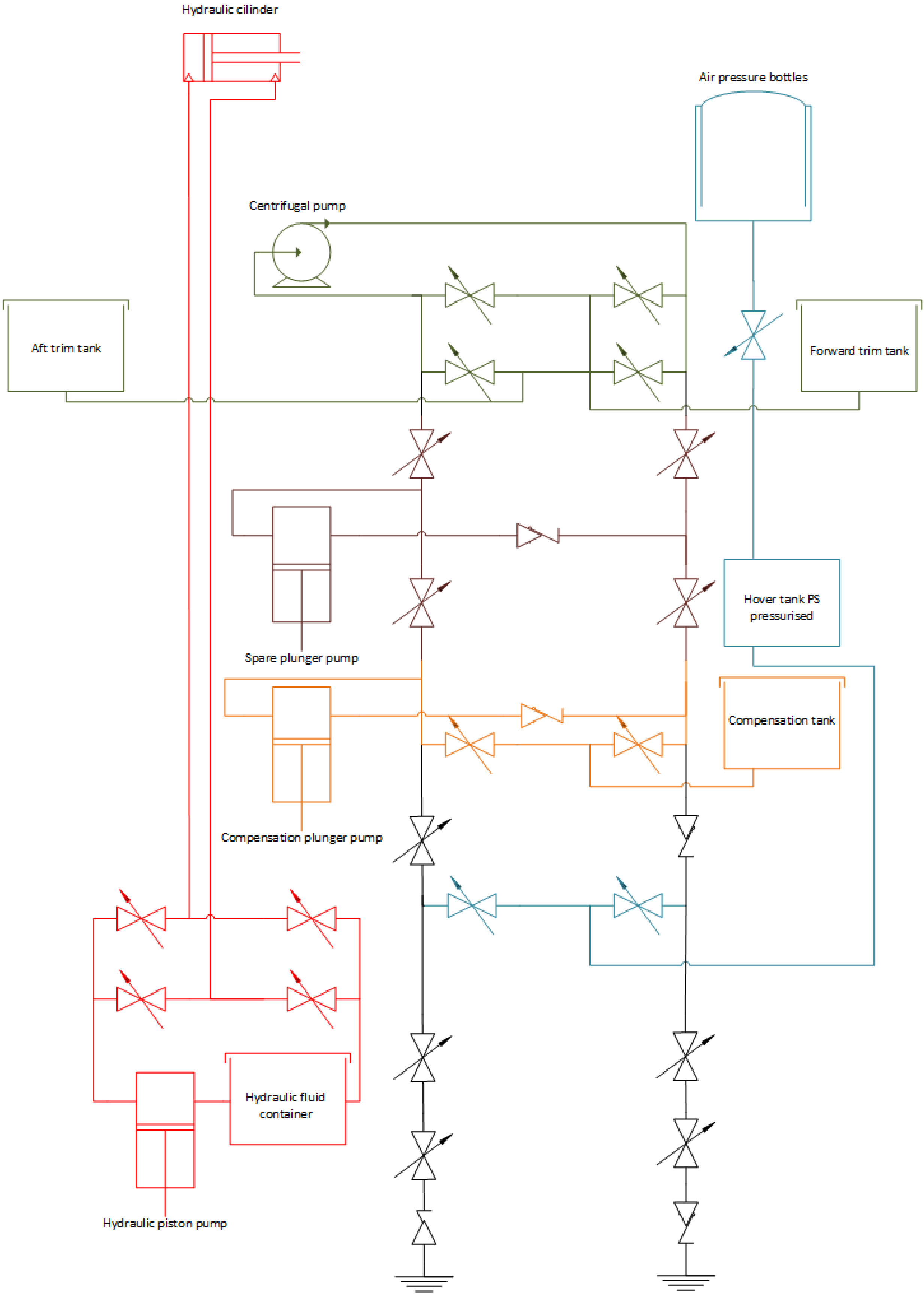


Figure F.12: Final proposed model of the ballast trim system for the 4000 ton submarine

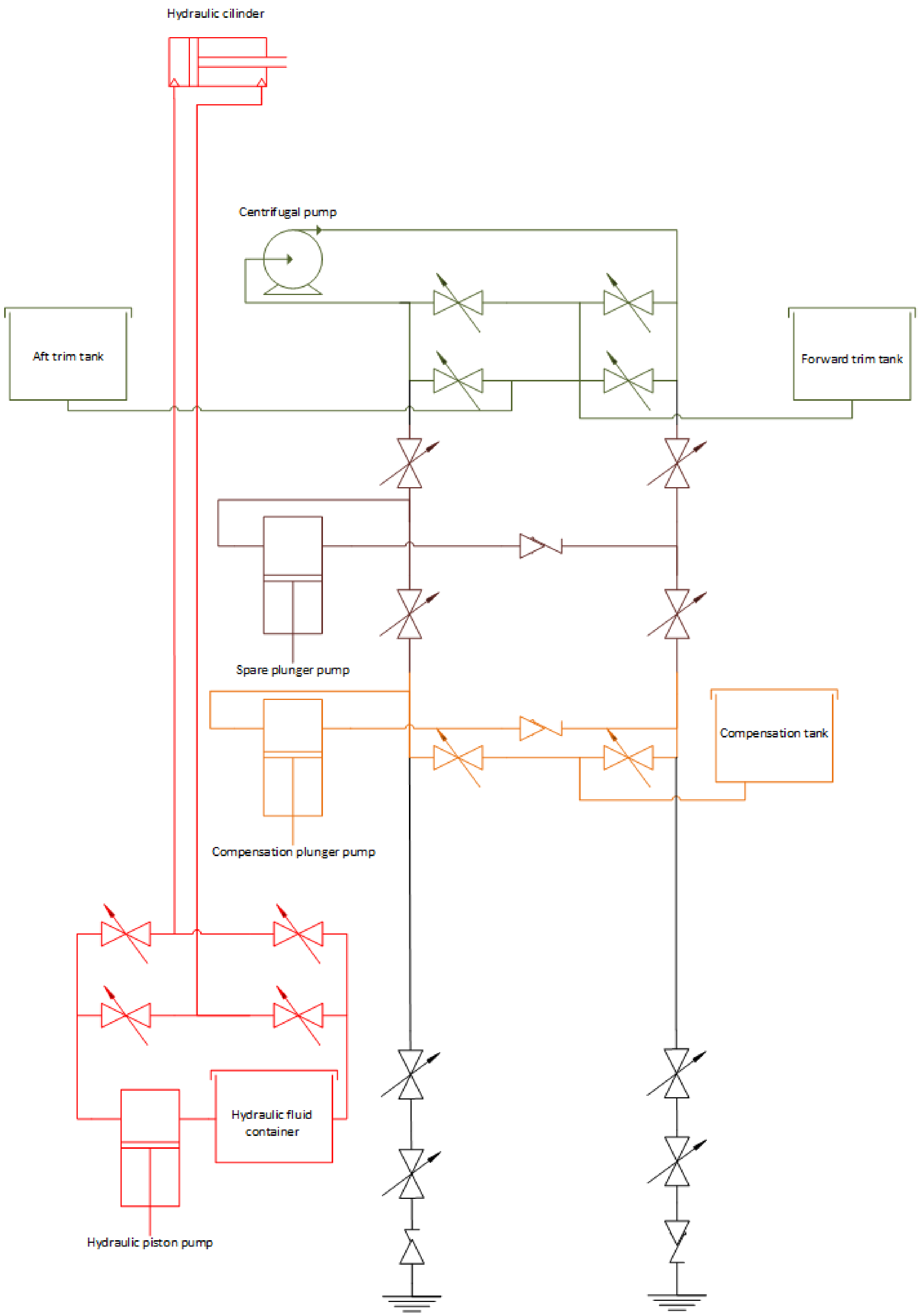


Figure F.13: Final proposed model of the ballast trim system for the 2000 ton submarine

G

Noise within the ballast trim system

Earlier in chapter 5 there are given methods to determine the sound power level of a valve, centrifugal pump or plunger pump. These methods give an indication of the produced noise at certain operational points. In practise this noise consists of three main types based upon their propagation. These three noise propagations are:

- structure born noise
- fluid born noise
- air borne noise

The overall sound power level is taken into account when comparing the different options as ballast trim system in chapter 6. Even though the production of noise by the ballast trim system is low, it can be more reduced by taking the right counter measurements. In this chapter the different propagations of the noise will be evaluated. There is also investigated how to reduce the noise production for the different types of noise. Modern submarines will most likely also have restrictions in narrow band tonal noise. The definition narrow band tonal noise will also be considered. First a small overview of the noise that was discussed in chapter 4 which can be found in table G.1.

Table G.1: Noise predictions used

Source	Sound power level [dB]	Dependancy
Centrifugal pump	$L_{Wl} = 95 + 10log(P_b)$	Power input
Piston pump	$L_{WL} = 117 + 10\log(\rho c_0 \frac{v^2}{D} \left[\epsilon + 40(\frac{\Delta p}{\rho c_0^2})^2 \right])$	Pressure fluctuations
Valve	$L_{WL} = 41 + 10\log(\frac{\frac{\dot{V}}{c_0} u^3}{10^{-12}})$	Pressure difference

Source	**Main noise type**	**Reference**
Centrifugal pump	Airborne/structure borne	(Irwin and Graf, 1979)
Piston pump	Fluid borne	(Müller and Möser, 2013)
Valve	Fluid-borne	(Müller and Möser, 2013)

G.1. Structure born noise

Structure born noise is the vibration that is transmitted through structural members. Figure G.1 shows structure born noise. Mechanical driven actuators vibrate. This vibration causes other structural member to vibrate with the same frequency. Structure born noise is hard to isolate. To only method to reduce structure born noise is to isolate the vibrating machine from its surroundings. This can be done by dampers. As seen

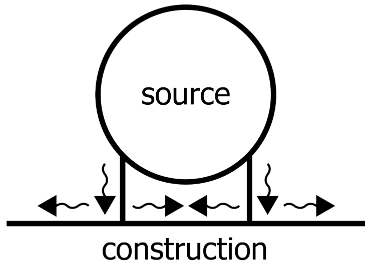


Figure G.1: Structure borne noise

in the simulated solutions, there where bellows. These bellows isolate the actuator from piping. This way the vibrations of the actuator is damped and less vibrations are transmitted through the pipe.

Machinery its produced structure borne noise can be reduced by installing the machinery on top off elastic materials or springs (Nilsson, 1978). These materials and springs make that the vibration is damped and less noise is transmitted through the structure. This is needed because the loss factor in steel plates used in submarine structures is low. The loss factor is the factor which determines the amount of sound waves that is converted into heat (Müller and Möser, 2013). for steel this loss factor is around $3 * 10^{-3}$ for frequencies below 500Hz and $1 * 10^{-3}$ for frequencies above 1000Hz. This results in that almost all the energy due to the noise is transmitted through the hull of the submarine.

The structure borne noise can reduced by applying material with lower loss factors. For example fiber

When determining the the right material or damper, the noise produced by the machine is considered first. The total noise produced by the machine can be split into the different noise levels within the frequency spectrum. This way there is made an overview of the noise within a certain frequency domain. The damper is effective for frequencies 1.4 times higher than the eigenfrequency of the damper (Müller and Möser, 2013). For frequencies below the eigenfrequency, the damper will not reduce the oscillation. For frequencies around the eigenfrequency of the damper, the oscillation will be intensified. Determining the minimum frequency for which the damper has to work is therefore important. The tuned frequency is a characteristic of the damper. The tuned frequency can be determined by equation G.1. This tuned frequency must be 1.4 times lower than the frequency of interest for damping.

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{s}{m}} \tag{G.1}$$

With: f are the eigenfrequencies of the system in Hz; s is the spring coefficient in $\mathrm{N/m^2}$; m is the mass in kg.

For pumps it is common to use a double damping system. This because there high structure born noise. This will lead to two tuning frequencies for a systems with a double mounted spring. These tuning frequencies for double mounted dampers can be found with equation G.2.

$$f_{\mathrm{I,II}} = \frac{1}{2}\left[(f_1^2 + f_2^2 + f_3^2) \pm \sqrt{\left(f_1^2 + f_2^2 + f_3^2\right)^2 - 4 f_1^2 f_2^2}\right] \tag{G.2}$$

$$f_1 = \frac{1}{2\pi}\sqrt{\frac{s_1}{m_1}}, \quad f_2 = \frac{1}{2\pi}\sqrt{\frac{s_2}{m_2}}, \quad f_3 = \frac{1}{2\pi}\sqrt{\frac{s_1}{m_2}}$$

With: f are the eigenfrequencies of the system in Hz; s is the spring coefficient in $\mathrm{N/m^2}$; m is the mass in kg.

G.2. Fluid born noise

Flow fluctuations causing vibrations in the fluid, shown in figure G.2. Methods that reduce the vibration in the fluid influence the system. By adding noise reduction measurement in the pipe, the resistance in the pipe will increase. It is important to keep these resistances in mind cause for some actuators studied, the delivered pressure increases the noise of the actuator.

Fluid borne noise can be found in the different solutions that are modelled in chapter 6.Fluid borne noise is especially a problem in the solutions with a plunger pump. The plunger pump creates large pressure fluctuations.

There do exist several methods to reduce this fluid borne noise. The best method is to modify the source of the noise (Cudina, 2008). The most critical source for fluid borne noise is the plunger pump. The pressure

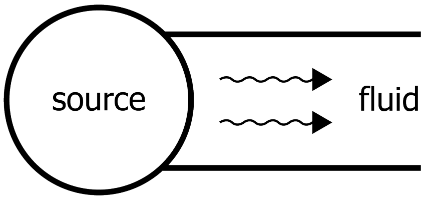


Figure G.2: Fluid born noise

pulsation in the fluid can be changed by increasing the number of plungers inside the pump. having an odd number of plunger does also help to reduce the noise. The pressure pulsation inside the fluid can be reduced further inside the piping. this can be done be placing the following silencer inside the pipe line:

- Expansion chamber;
- Side branch resonator;
- Metal compensator;
- Molded neoprene;
- Nylon chord reinforcement.

Figures of these silencers can be found in figure G.3.

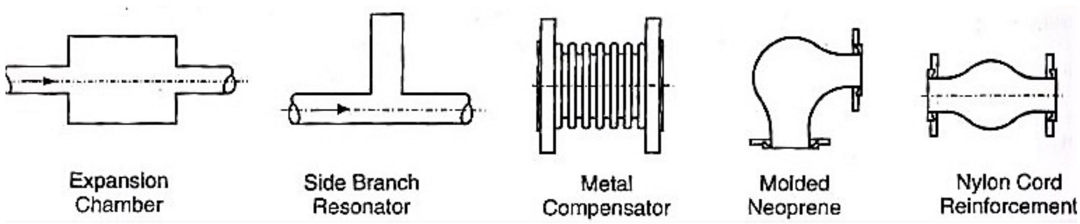


Figure G.3: Fluid-borne noise measures (Cudina, 2008)

The expansion chamber can be inserted in the pipe line. Due to sudden expansion of the pipe line the shock waves will propagated through the whole volume. This result in that in at the end of the expansion chamber less pressure fluctuations will be emitted to the pipeline

The side branch resonator can be found in most cases close to a plunger pump. The pressure fluctuations in the fluid causes noise. The side branch resonator is filled with are. When the pressure fluctuates in a pipe line, the air inside the resonator will compress and expand. The air inside the resonator works therefore as a spring and damps the pressure fluctuations in the pipe line.

The molded neoprene makes sure that the bend of the pipeline is very smooth. Therefore the fluid inside the bend will also flow smooth through the pipe resulting in no noise production inside bends of pipelines.

G.3. Air born noise

Air born noise is the best known kind of noise. Airborne noise is always around us. Air born noise are vibrations transmitted by air as can be seen in figure G.4. Counter measurements can be found in enclose the space around the source. For submarines this can be done by placing anechoic tiles of coating on the hull of the submarine. The effects of these anechoic tiles are already discussed.

Sound travelling through air hitting an obstacle is split into four energy transfers. These three transfers are:

- reflection
- absorption
- transmitted
- transformed

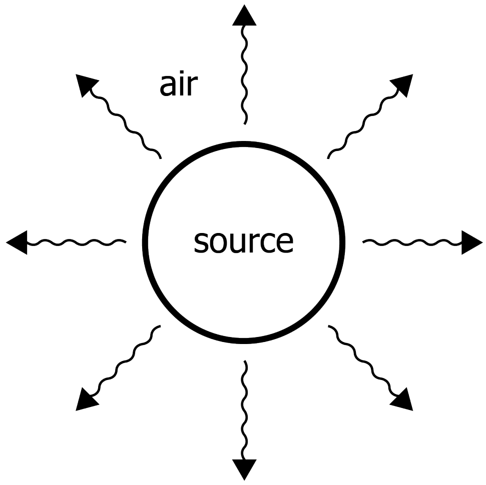


Figure G.4: Air borne noise

The reduce the air borne noise, the amount of energy that is adsorbed is the most interesting. Materials have an absorption coefficient. With this coefficient the total energy absorbed by the material can be determined. Materials with high absorption coefficient are used in noisy areas. These materials can be placed over the noise source to make sure the sound is not transmitted through the air.

The hull of the submarine and the anechoic tiles absorb the air borne noise energy. This absorption is only to a certain level. Therefore the systems on board of a submarine all have to comply with a maximum sound power level. This will be given by the general requirements of the submarine

G.4. Narrow band tonal noise

Modern submarines do have restrictions in the produced narrow band tonal noise. In the past these restrictions were not present. Narrow band tonal noise is a peak in the noise production in a very small band width. This gives that the overall noise production can be low but at certain frequencies the noise production is very high. Machinery usually have at certain frequencies a very high noise level. Measuring the noise in small band frequencies gives an identification of the tonal noise. The centrifugal pump is for example known for its narrow band tonal noise. There is seen that compared to other systems, the centrifugal pump scores good based on noise production. However when also taking the tonal noise in consideration it is possible that the centrifugal pump scores worst.

H

Tuned system

H.1. Trim system

H.1.1. Submarine 1

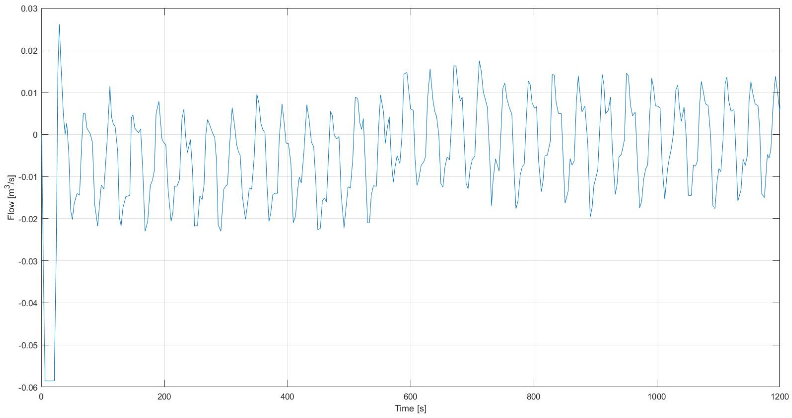


Figure H.1: Needed flow for trimming while debarking SF

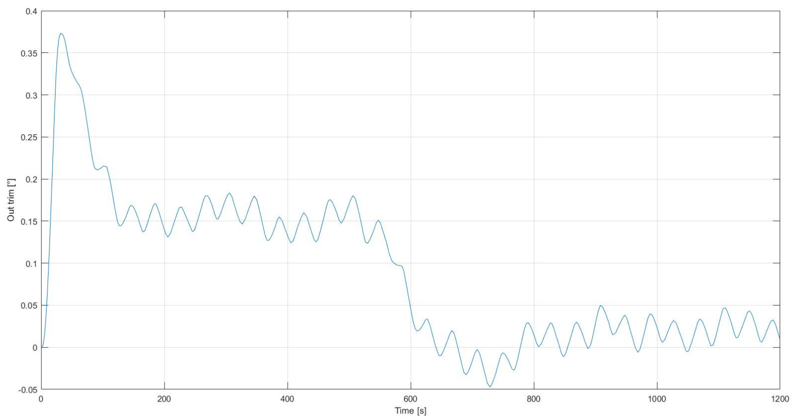


Figure H.2: Pitch motion during debarking SF

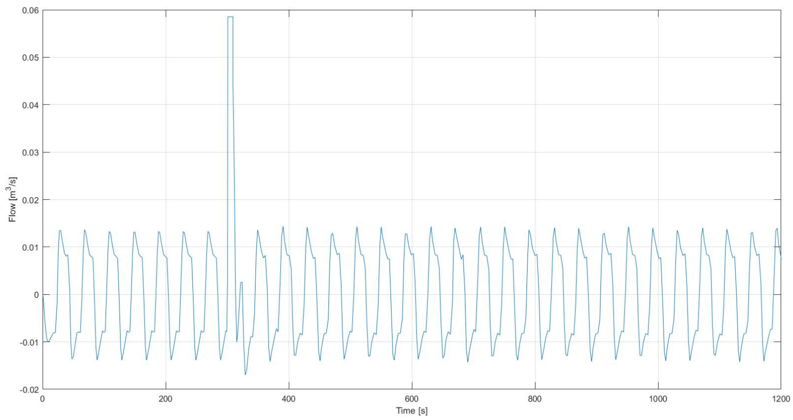


Figure H.3: Needed flow for trimming while anchoring

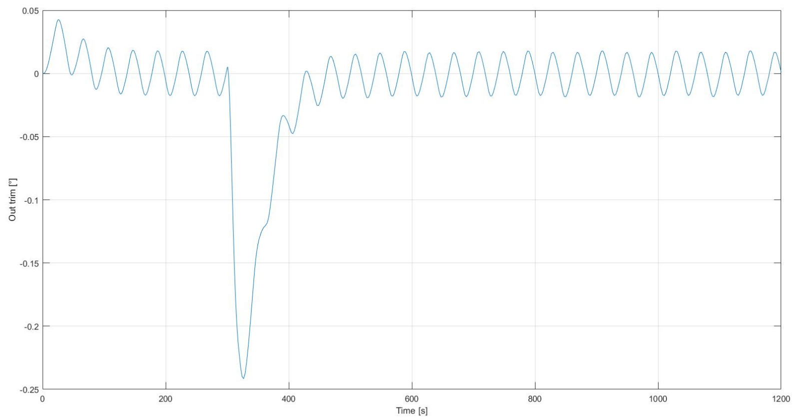


Figure H.4: Pitch motion during anchoring

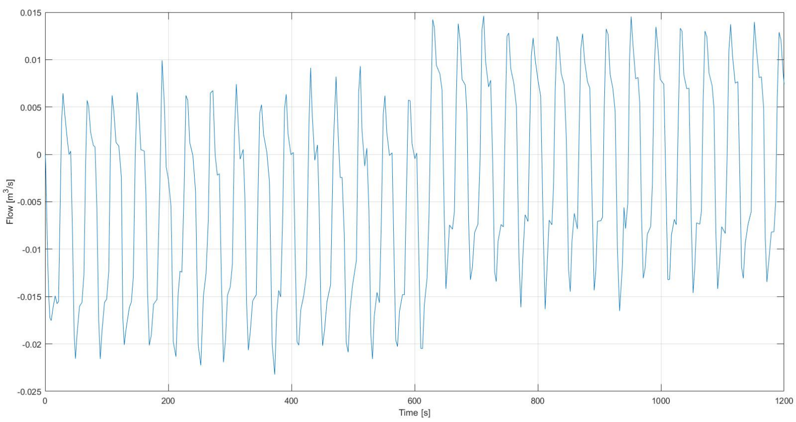


Figure H.5: Needed flow for trim during disembarking AUV

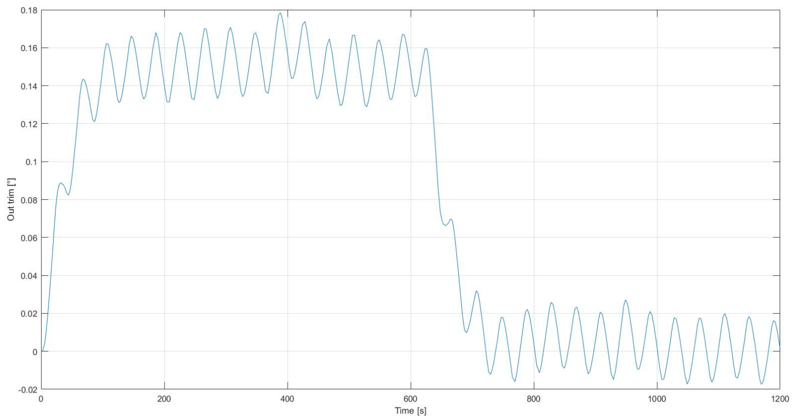


Figure H.6: Pitch motion during disembarking AUV

H.1.2. Submarine 2

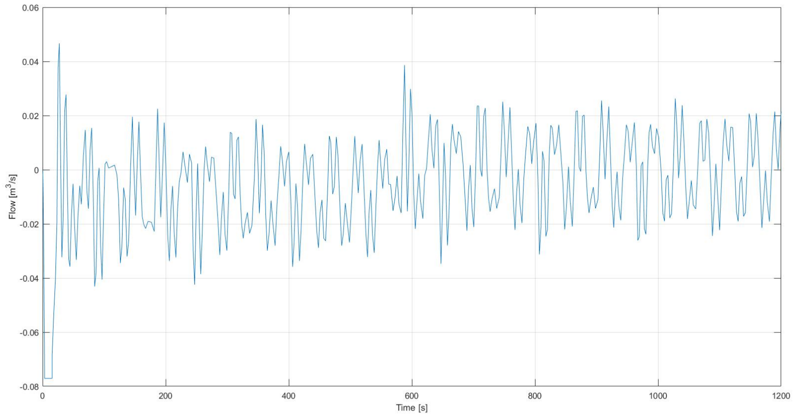


Figure H.7: Needed flow for trimming while debarking SF

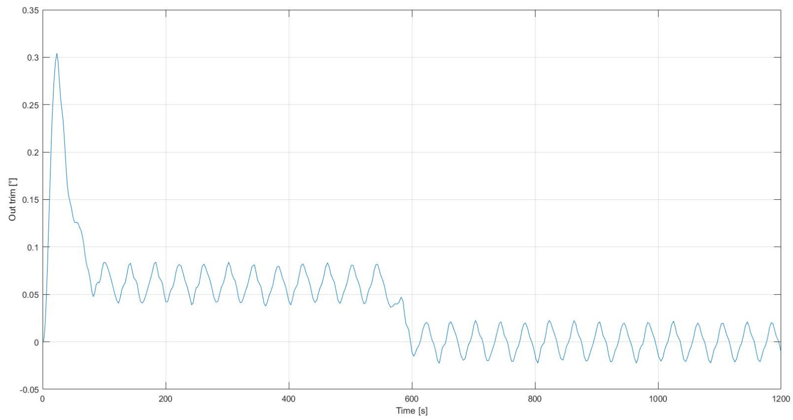


Figure H.8: Pitch motion during debarking SF

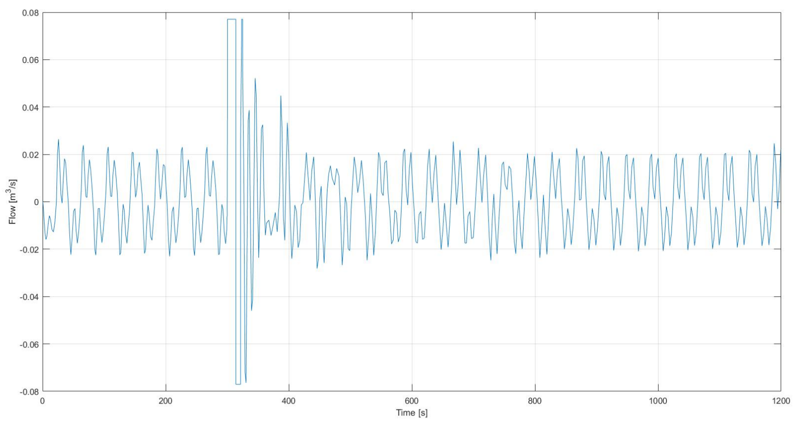


Figure H.9: Needed flow for trimming while anchoring

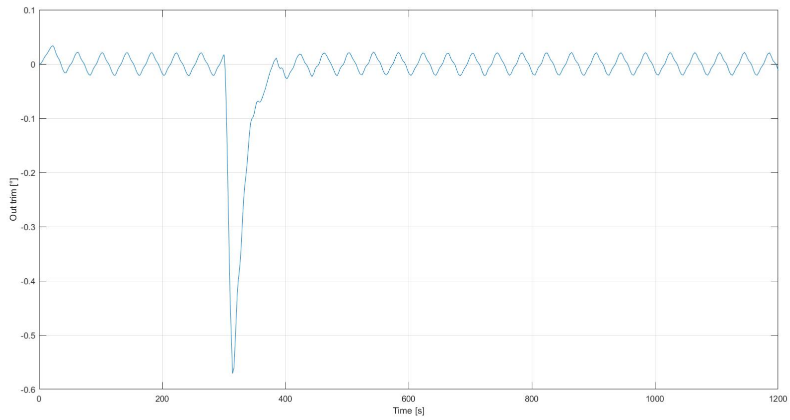


Figure H.10: Pitch motion during anchoring

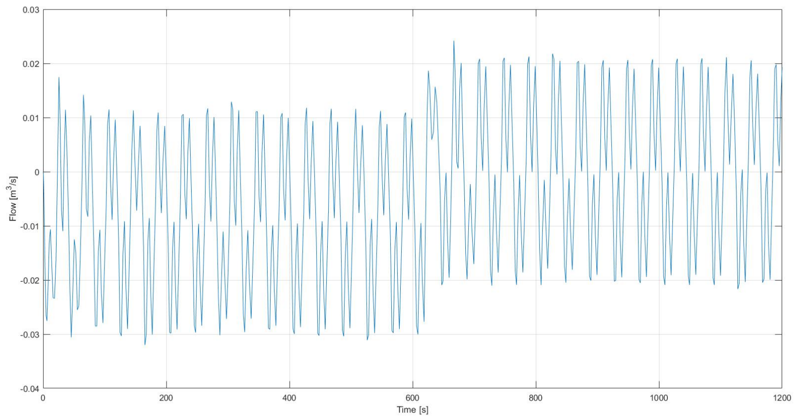


Figure H.11: Needed flow for trimming disembarking AUV

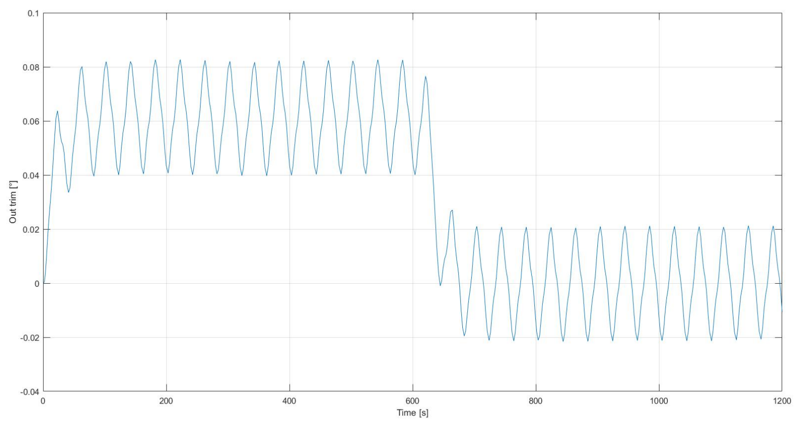


Figure H.12: Pitch motion during disembarking AUV

H.2. Compensation system

H.2.1. submarine 1

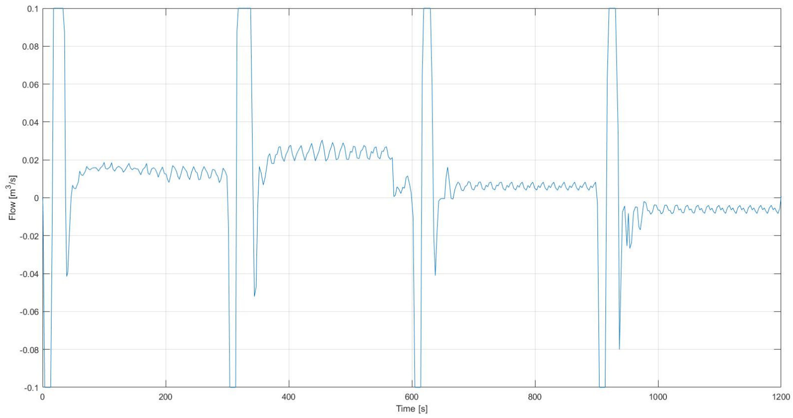


Figure H.13: Needed flow for compensating while debarking SF

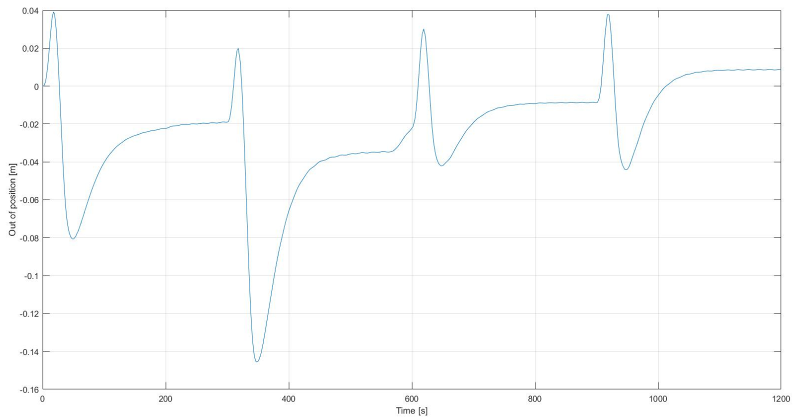


Figure H.14: Heave motion during debarking SF

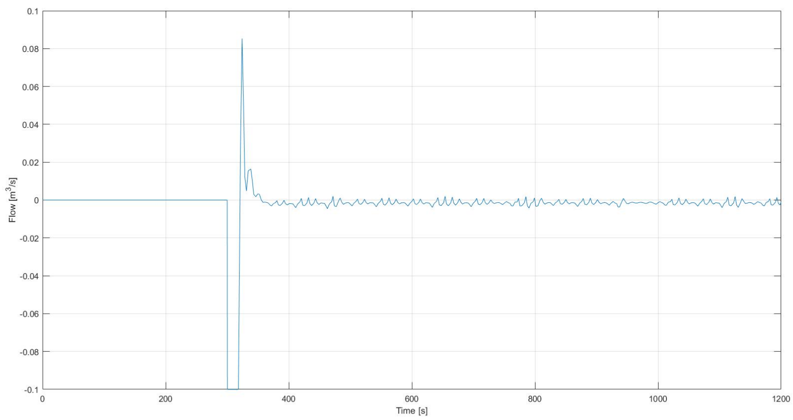


Figure H.15: Needed flow for compensating while anchoring

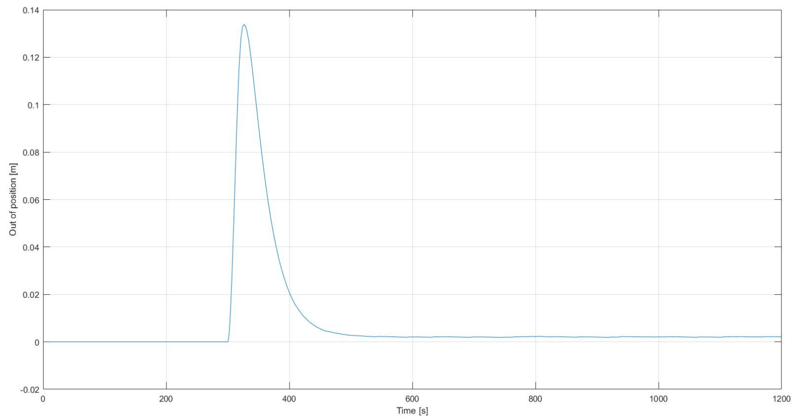


Figure H.16: Heave motion during anchoring

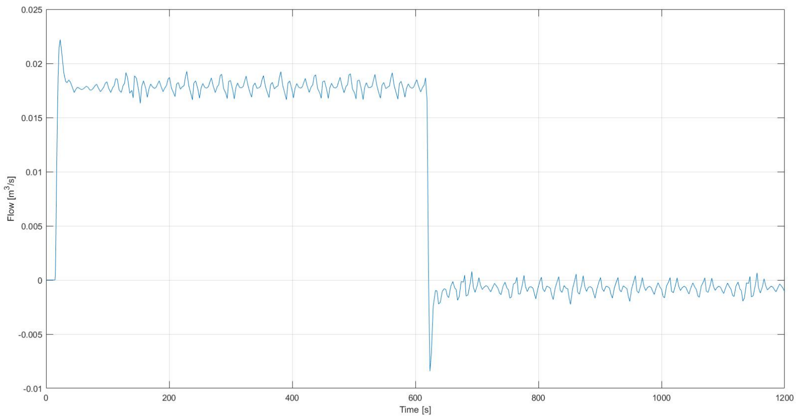


Figure H.17: Needed flow for compensating while disembarking AUV

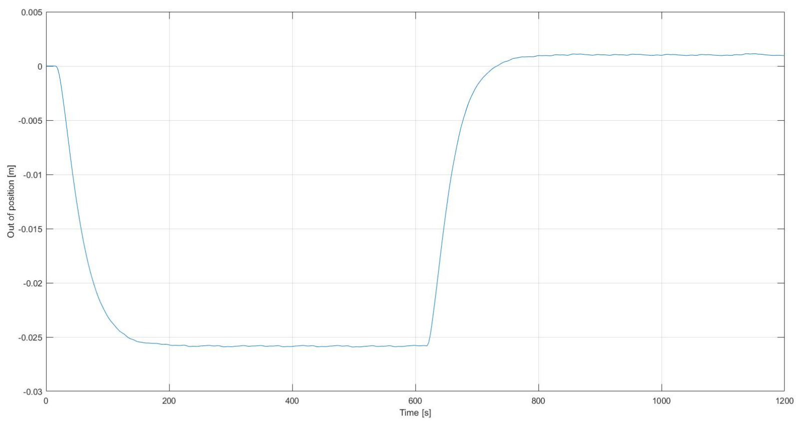


Figure H.18: Heave motion during disembarking AUV

H.2.2. submarine 2

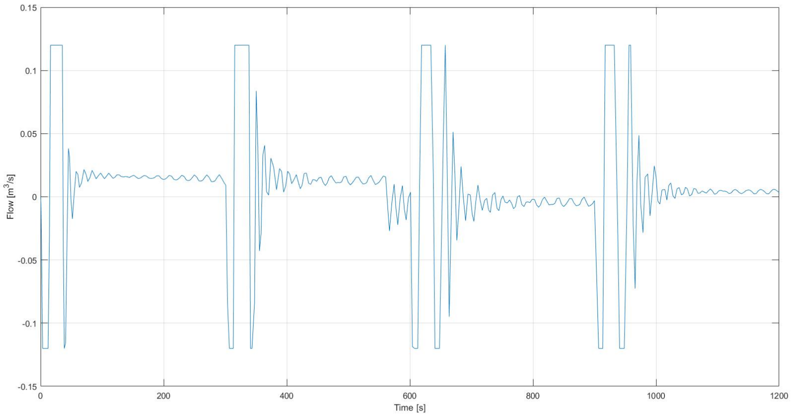


Figure H.19: Needed flow for compensating while debarking SF

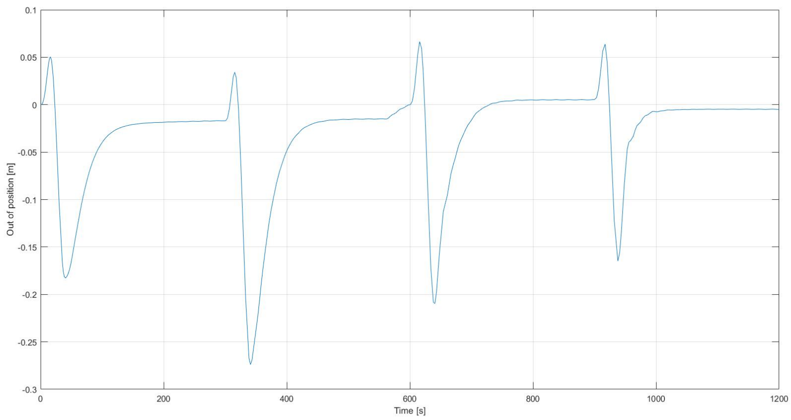


Figure H.20: Heave motion during debarking SF

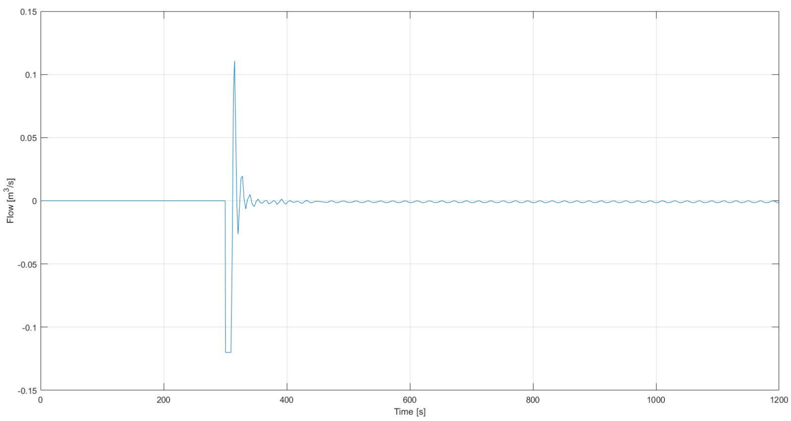


Figure H.21: Needed flow for compensating while anchoring

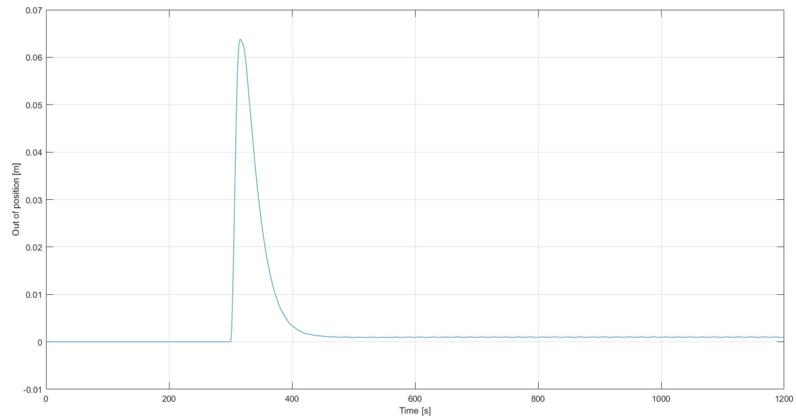


Figure H.22: Heave motion during anchoring

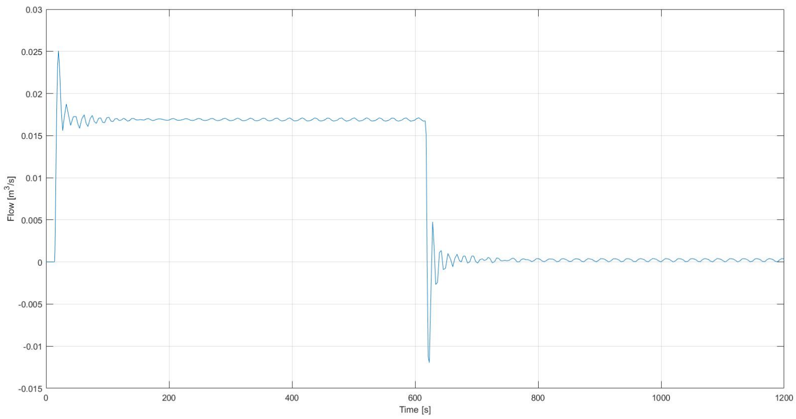


Figure H.23: Needed flow for compensating while disembarking AUV

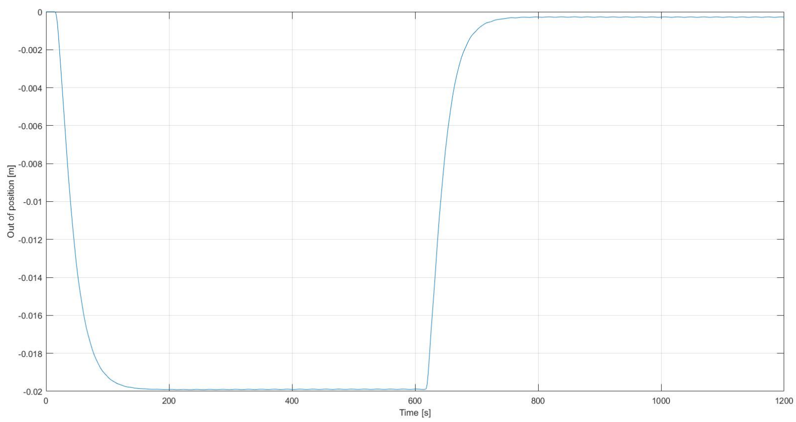


Figure H.24: Heave motion during disembarking AUV

H.3. Hovering system

H.3.1. submarine 1

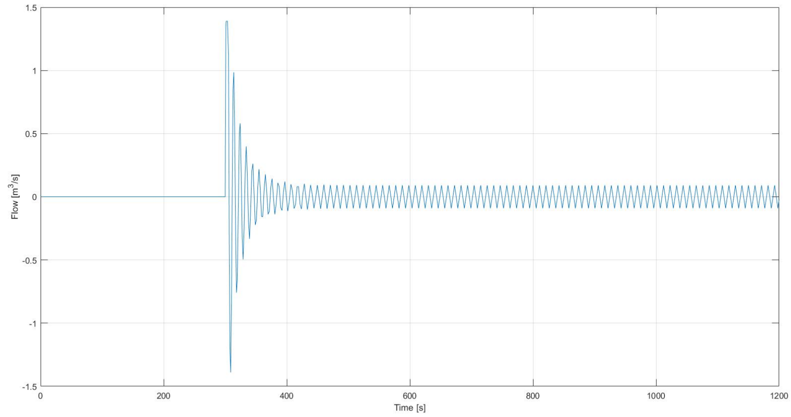


Figure H.25: Needed flow for hovering with change in density

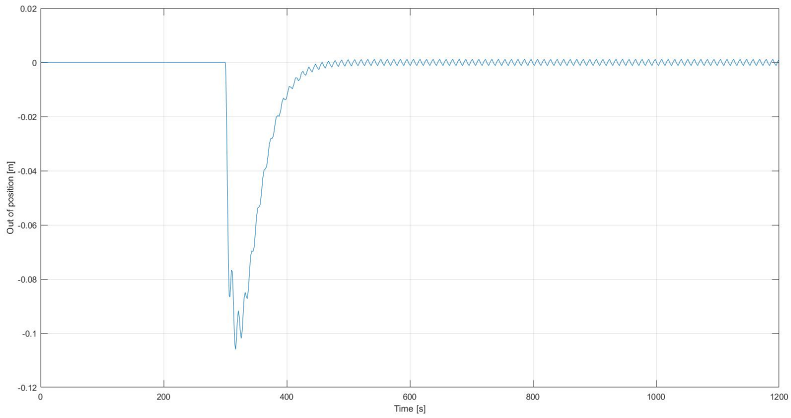


Figure H.26: Heave motion when density changes

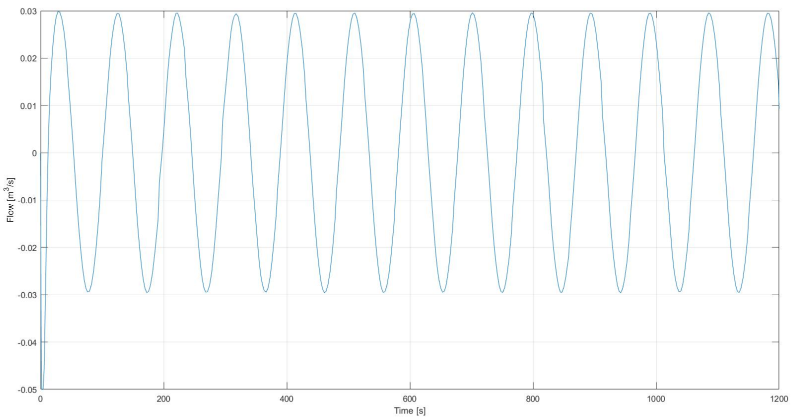


Figure H.27: Needed flow for hovering due to second order effects waves

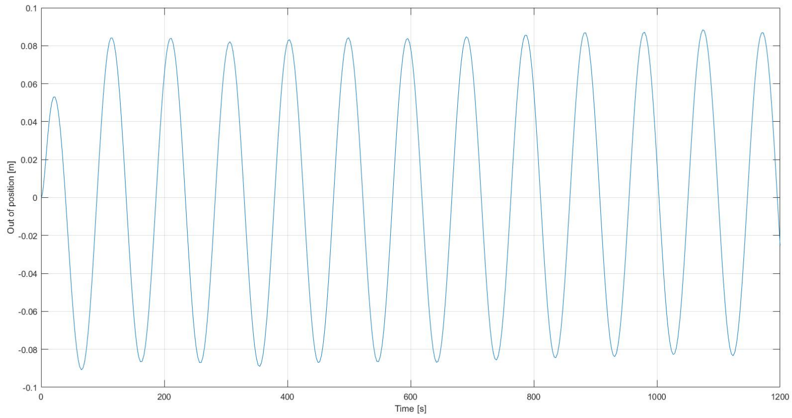


Figure H.28: Heave motion due to second order effects waves

H.3.2. submarine 2

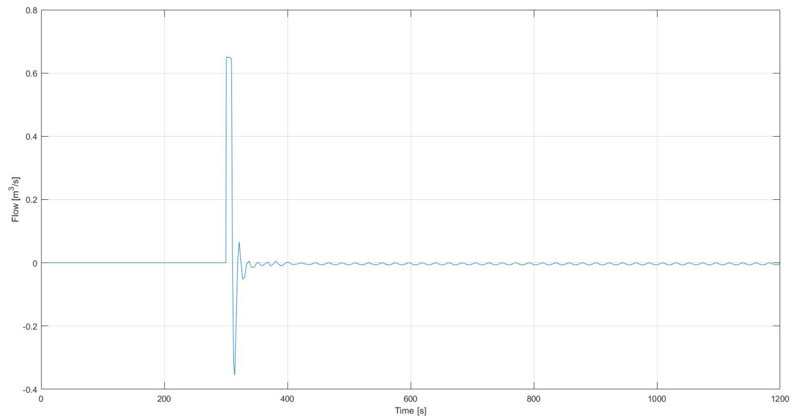


Figure H.29: Needed flow for hovering with change in density

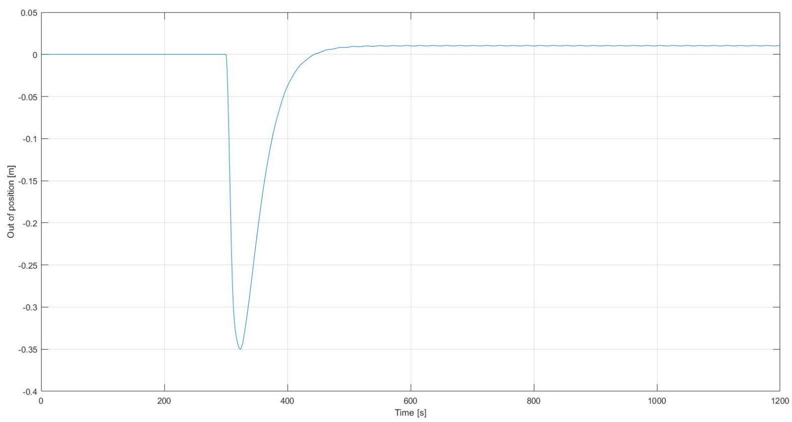


Figure H.30: Heave motion when density changes

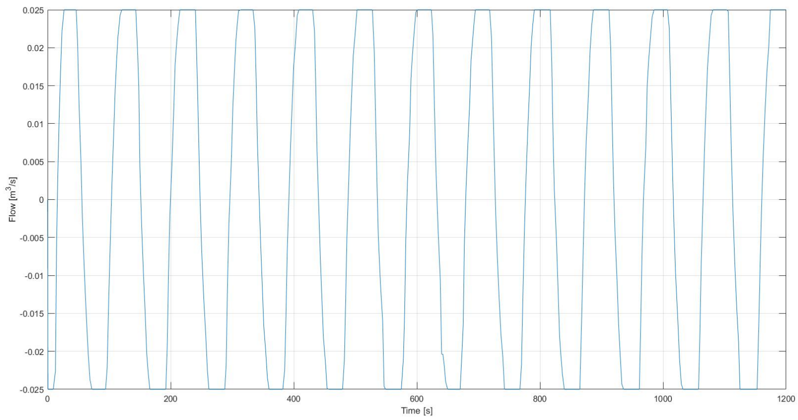


Figure H.31: Needed flow for hovering due to second order effects waves

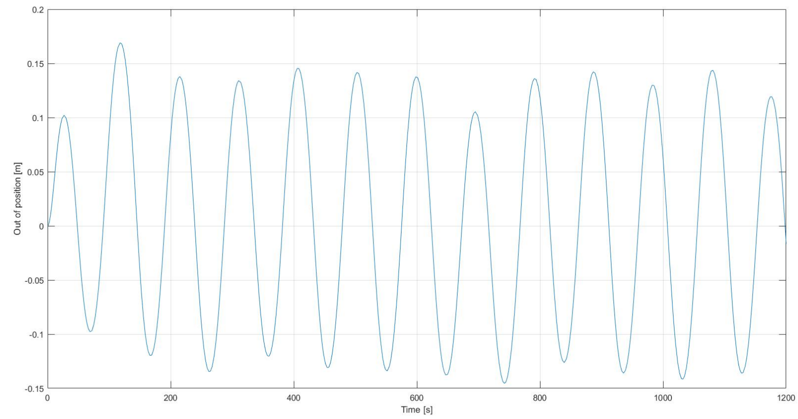


Figure H.32: Heave motion due to second order effects waves

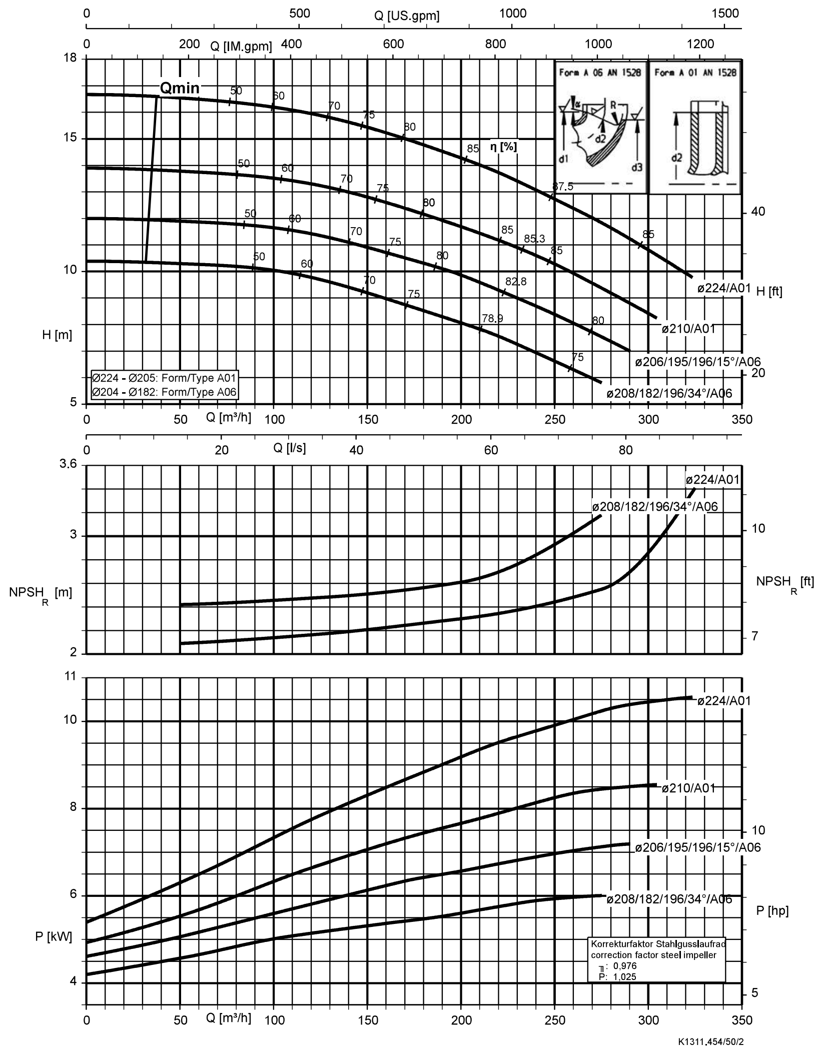
I

Pump information

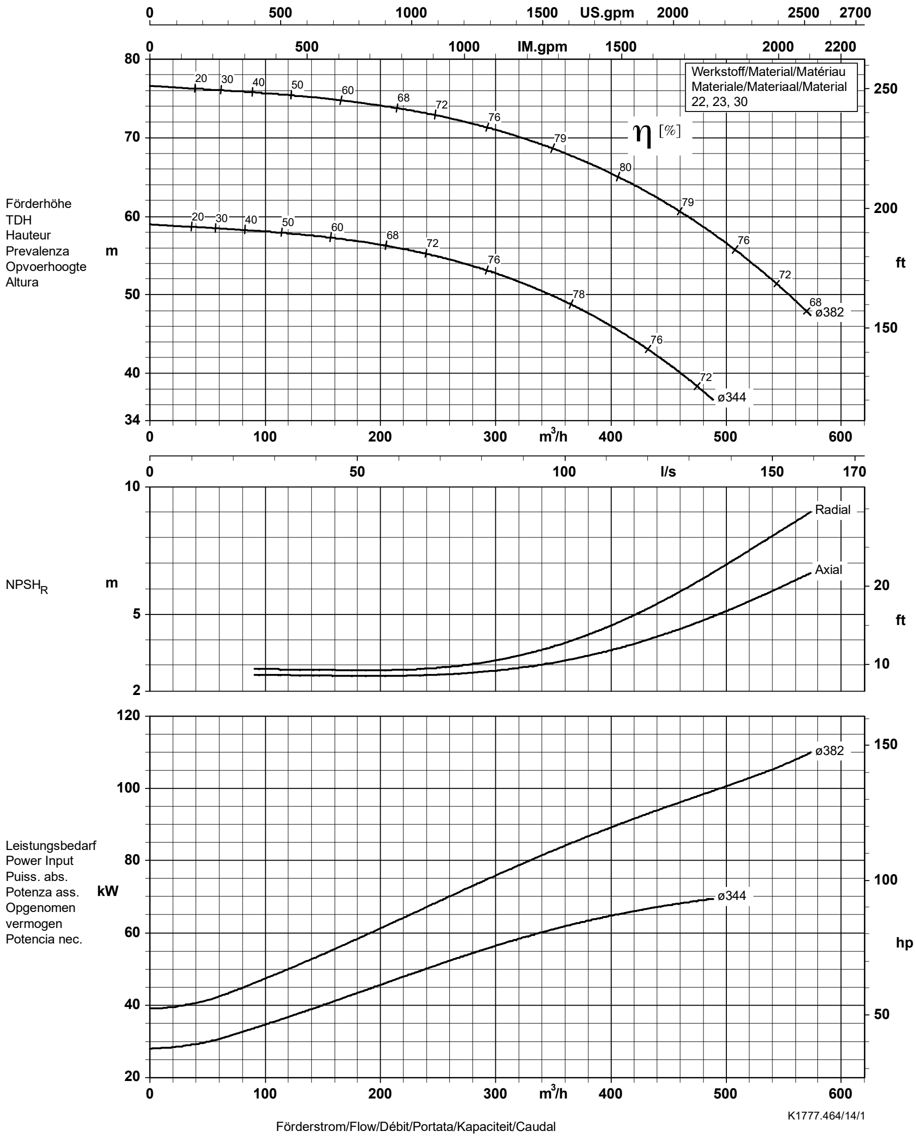
Centrifugal Pumps with Shaft Seal
Standardised Water Pump / Thermal Oil and Hot Water Pump

Etanorm 150-125-200, n = 1450 rpm

Etanorm SYT, Etanorm V, Etabloc



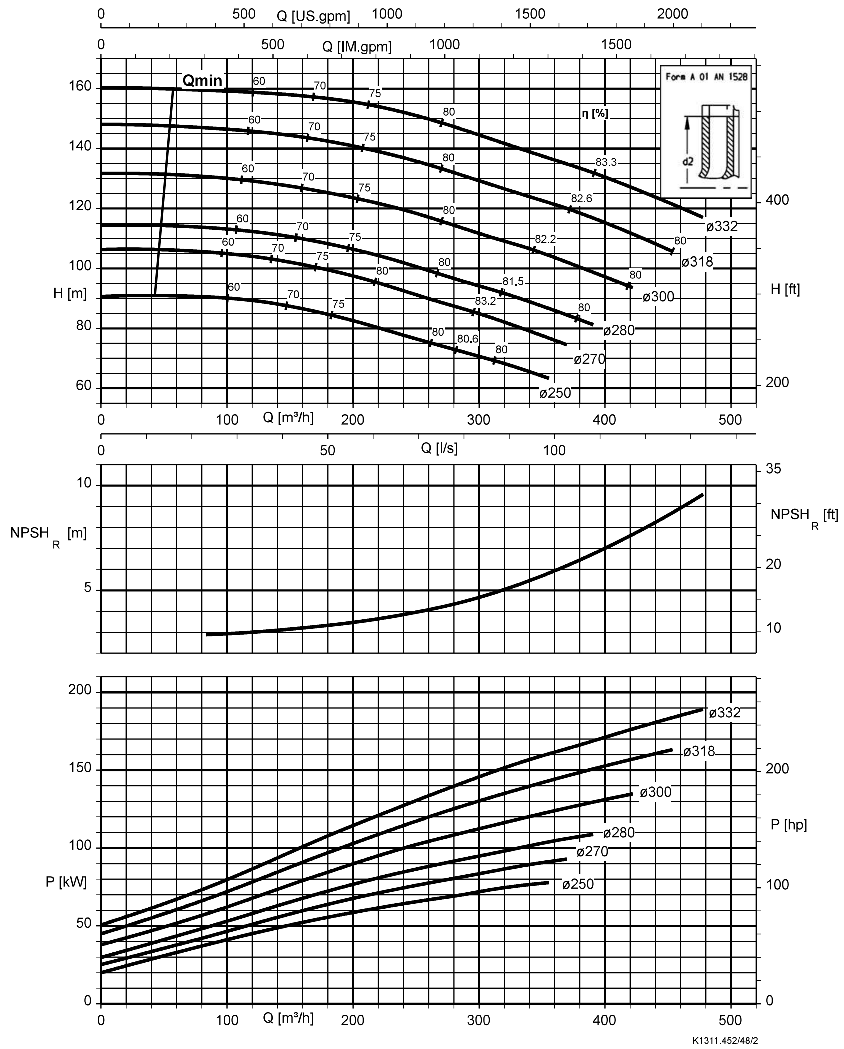
Baureihe-Größe / Tipo Type-Size / Serie Modèle / Tipo	Nenndrehzahl / Velocità di rotazione nom. Nom. speed / Nominaal toerental Vitesse nom. / Revolutiones nom.	Laufrad-ø / ø Girante Impeller Dia. / ø Waaier Diamètre de roue / ø Rodete	KSB
Multitec 150 12.1 1.4408	1750 1/min		
Projekt / Progetto Project / Projekt Projet / Proyecto	Angebots-Nr. / Offerta-No. Project No. / Offertenr. No. de l'offre / Offerta-No.	Pos.-Nr. / Pos.-Nr. Item No. / Positienr. No. de pos. / Pos.-Nr.	Division Pompes Industrie et Eau KSB S. A. Allée de Sagan - B.P. 189 36004 Châteauroux



Laufradaustrittsbreite/Impeller outlet width/Largeur à la sortie de la roue 26,5 mm
Luce della girante/Waaier uittredebreedte/Anchura de salida rodete 26,5 mm

108

Etanorm 125-100-315, n = 2900 rpm



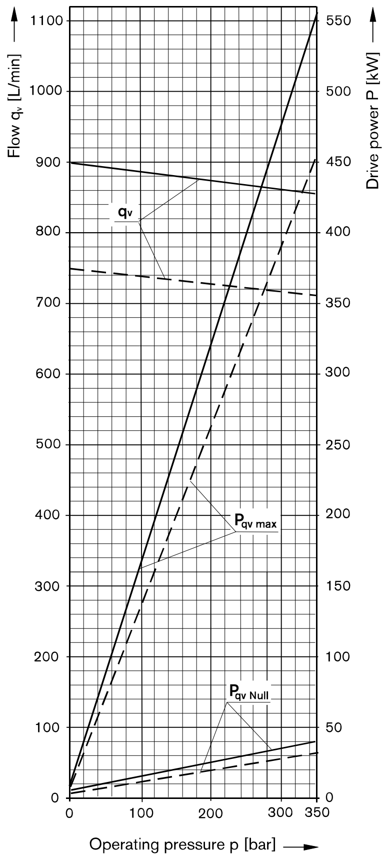
Characteristics

Drive power and flow

(Fluid: Hydraulic oil ISO VG 46 DIN 51519, t = 50°C)

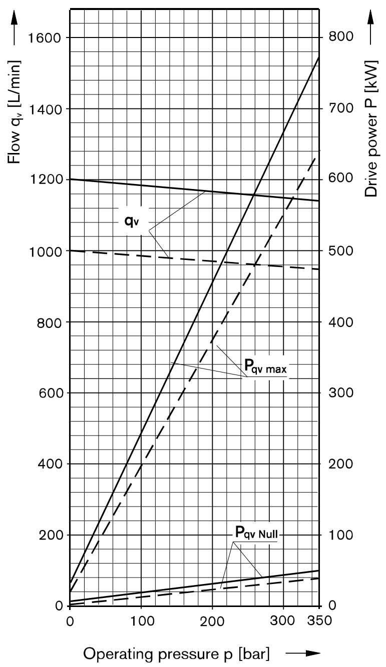
Overall efficiency: $\eta_t = \frac{q_v \cdot p}{P_{q_v\,max} \cdot 600}$ Volumetric efficiency: $\eta_v = \frac{q_v}{q_{v\,theor}}$

Size 750



n = 1200 rpm
n = 1000 rpm

Size 1000



n = 1200 rpm
n = 1000 rpm

J

Results simulations

J.1. Trim

J.1.1. Submarine 1, 4000 ton

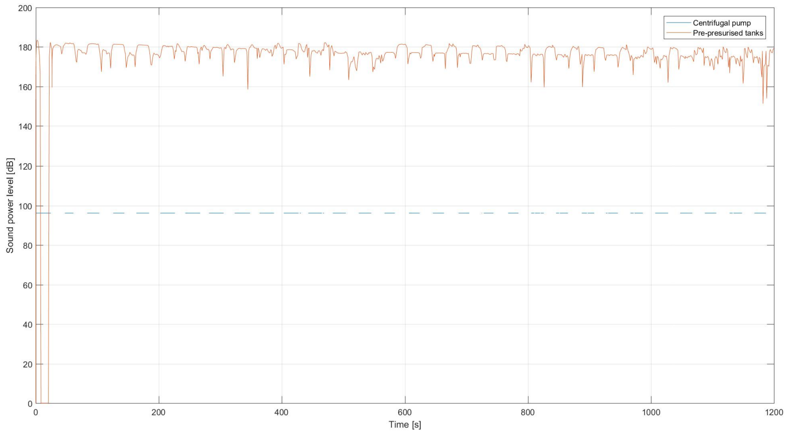


Figure J.1: Sound power level due to disembark SF submarine 1

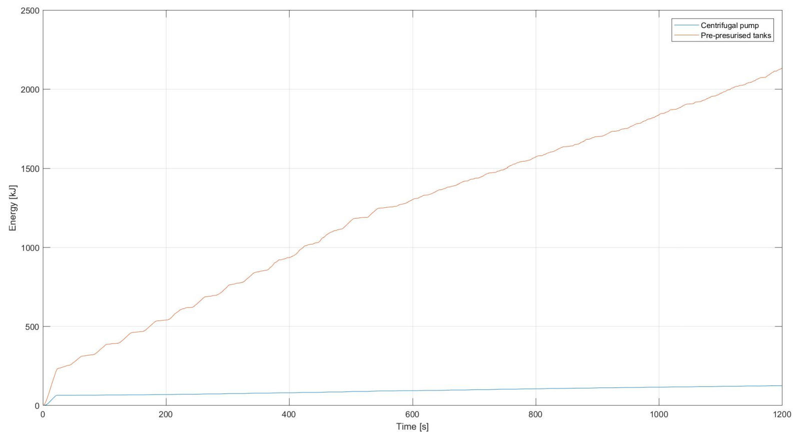


Figure J.2: Energy due to disembark SF submarine 1

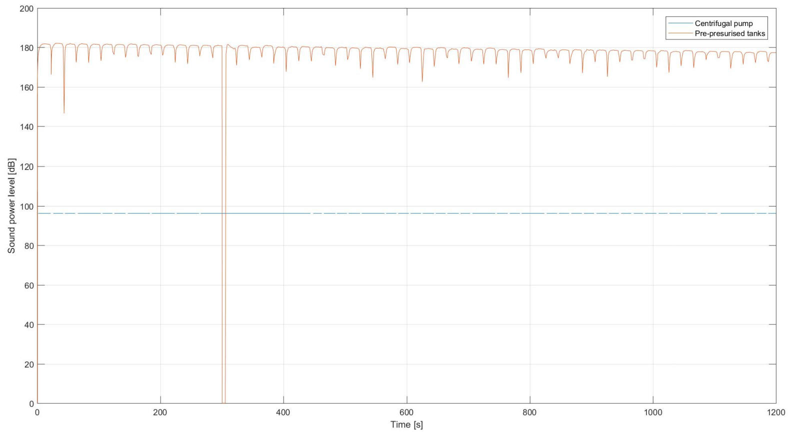


Figure J.3: Sound power level due to Anchorage submarine 1

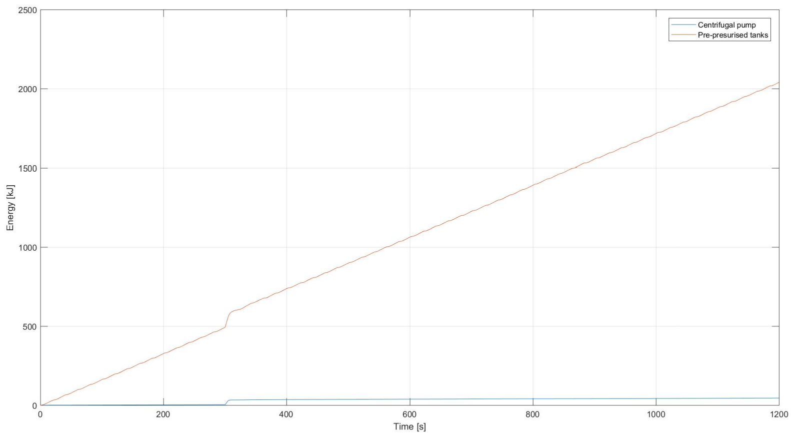


Figure J.4: Energy due to Anchorage submarine 1

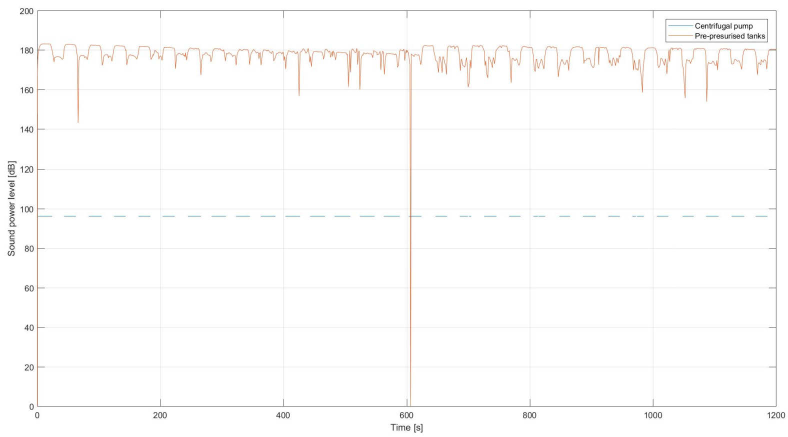


Figure J.5: Sound power level due to disembark AUV submarine 1

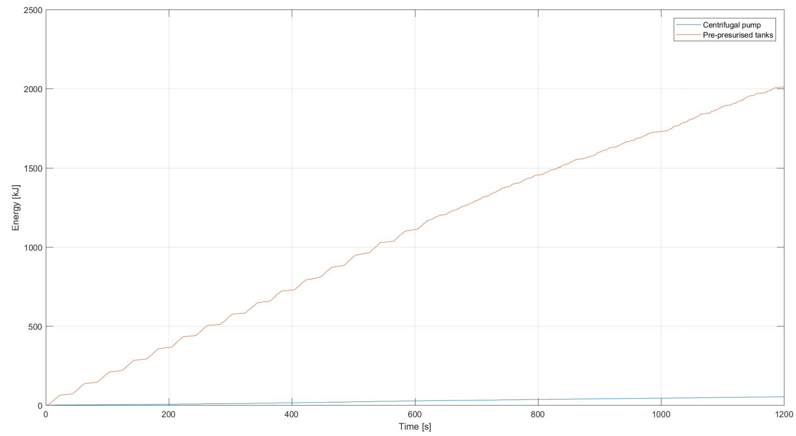


Figure J.6: Energy due to disembark AUV submarine 1

J.1.2. Submarine 2, 2000 ton

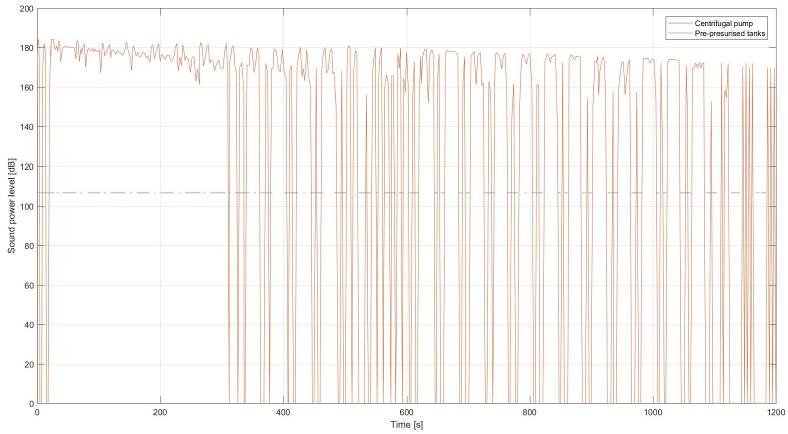


Figure J.7: Sound power level due to disembark SF submarine 2

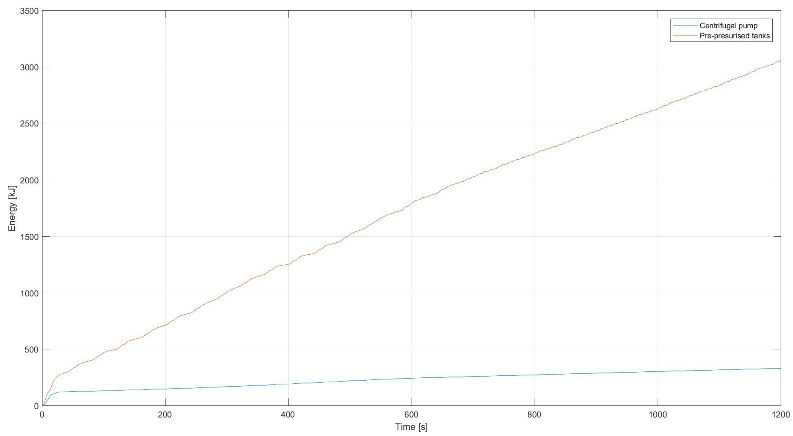


Figure J.8: Energy due to disembark SF submarine 2

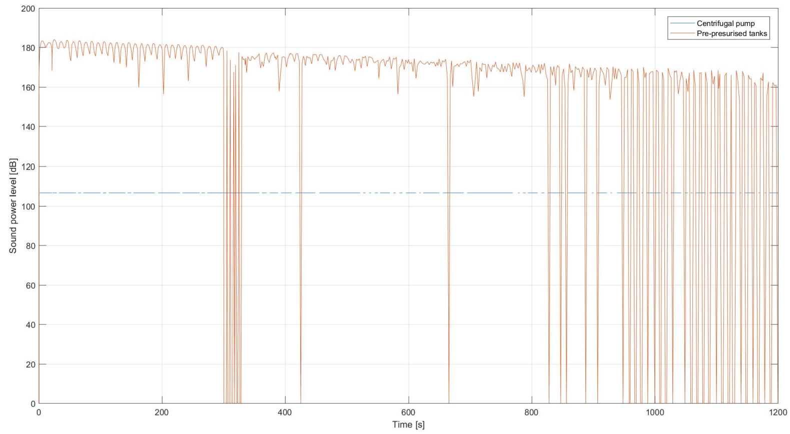


Figure J.9: Sound power level due to Anchorage submarine 2

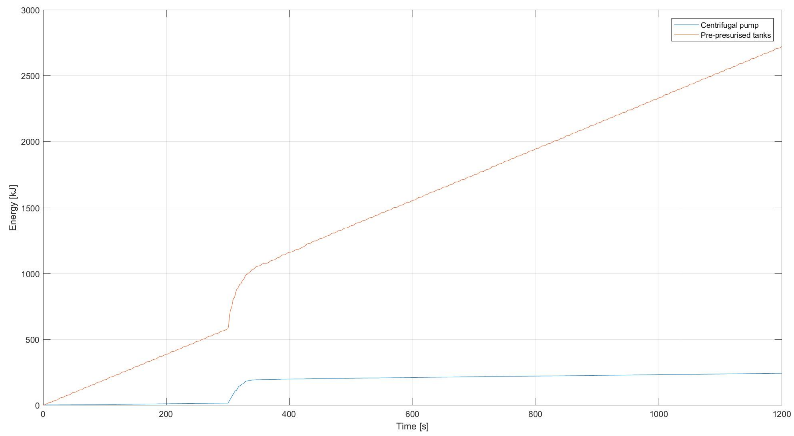


Figure J.10: Energy due to Anchorage submarine 2

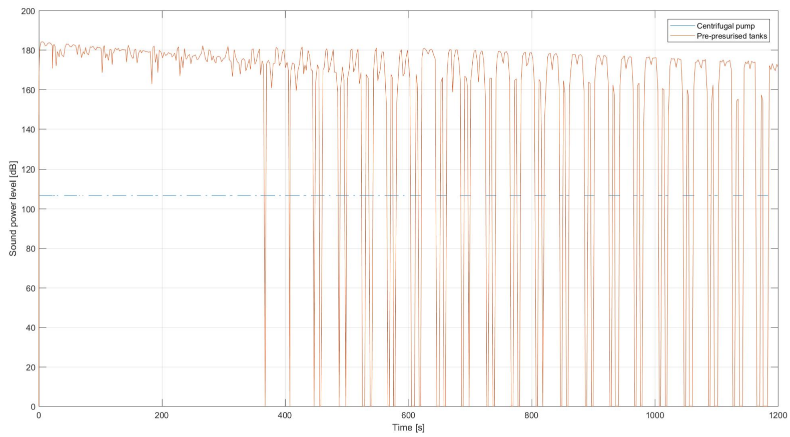


Figure J.11: Sound power level due to disembark AUV submarine 2

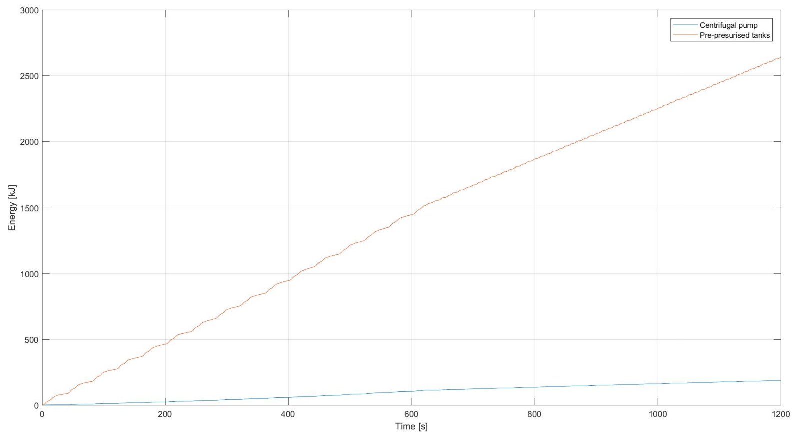


Figure J.12: Energy due to disembark AUV submarine 2

J.2. Compensation

J.2.1. Submarine 1, 4000 ton

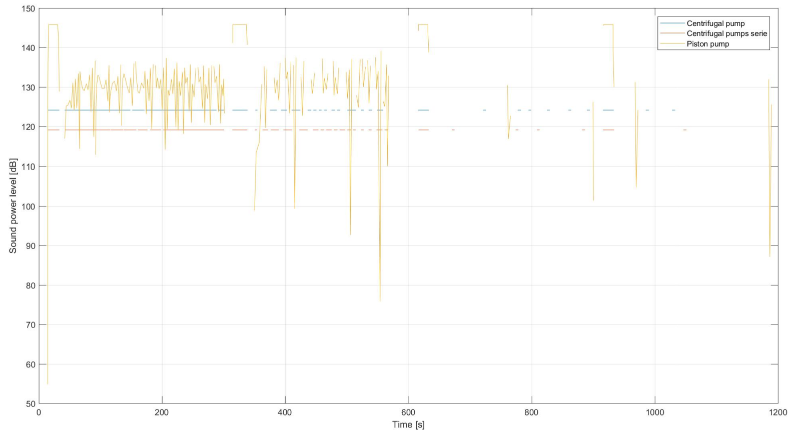


Figure J.13: Sound power level due to disembark SF submarine 1

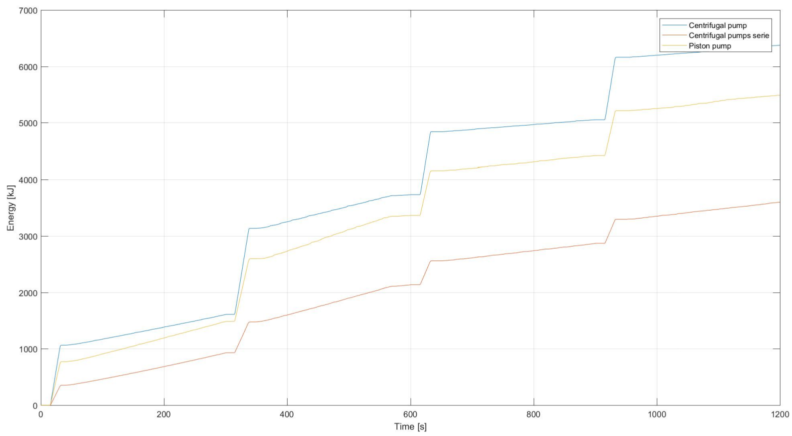


Figure J.14: Energy due to disembark SF submarine 1

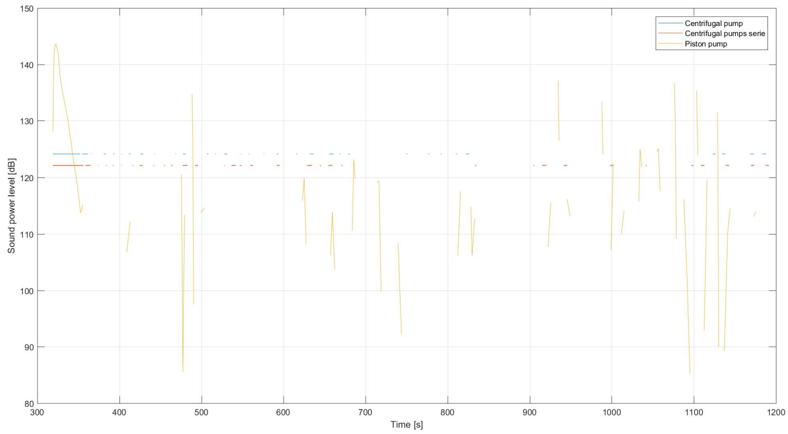


Figure J.15: Sound power level due to Anchorage submarine 1

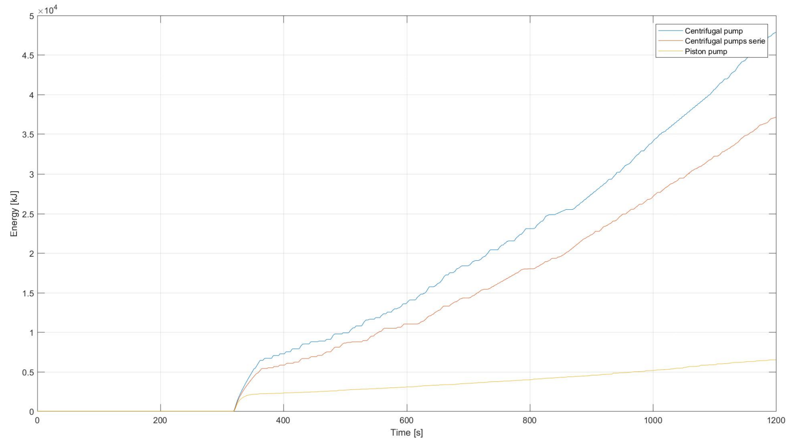


Figure J.16: Energy due to Anchorage submarine 1

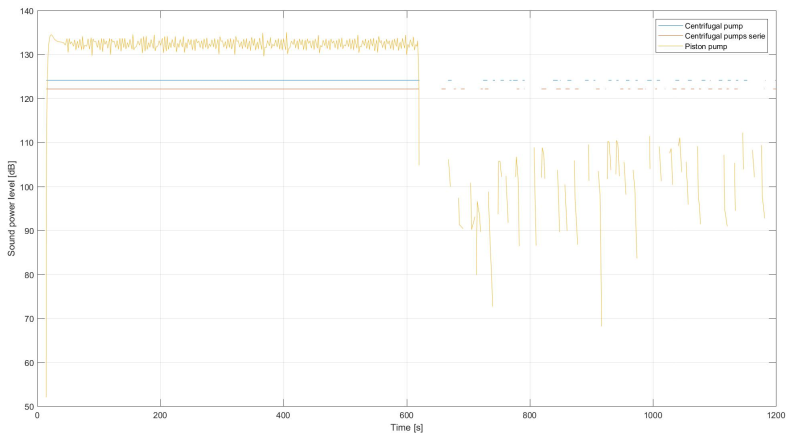


Figure J.17: Sound power level due to disembark AUV submarine 1

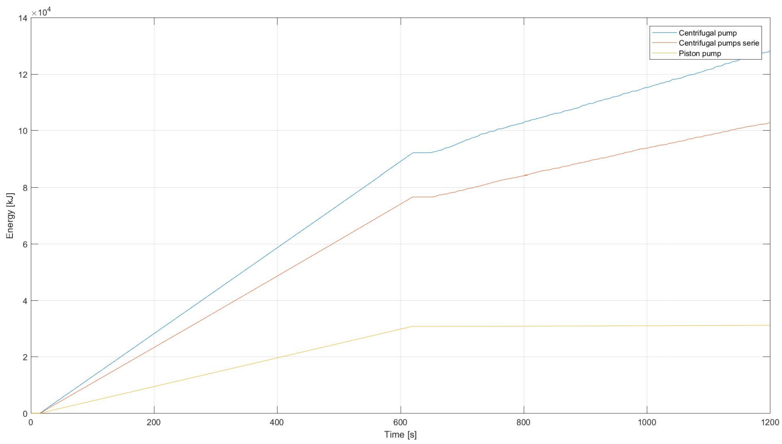


Figure J.18: Energy due to disembark AUV submarine 1

J.2.2. Submarine 2, 2000 ton

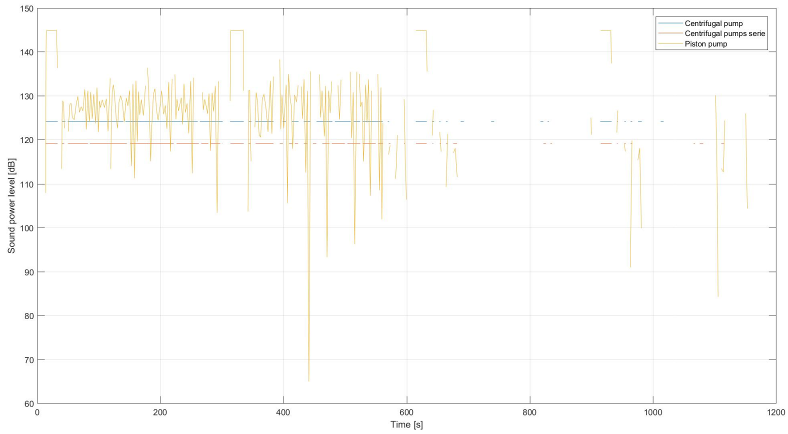


Figure J.19: Sound power level due to disembark SF submarine 2

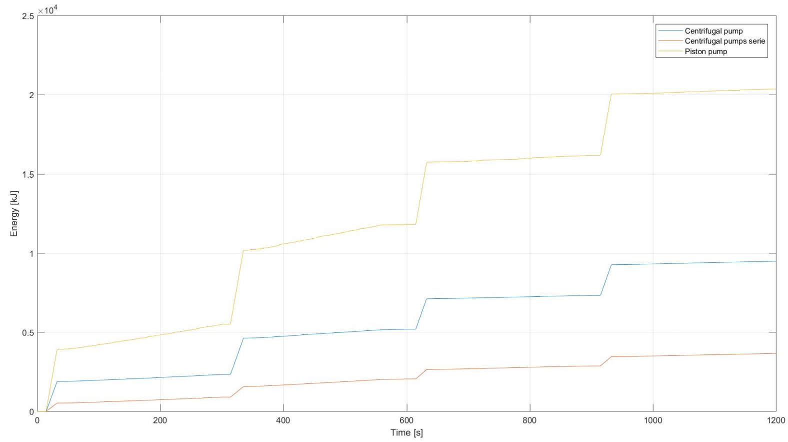


Figure J.20: Energy due to disembark SF submarine 2

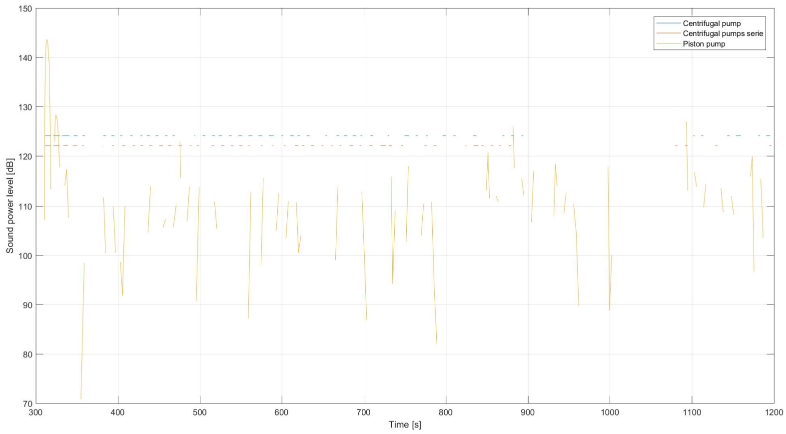


Figure J.21: Sound power level due to Anchorage submarine 2

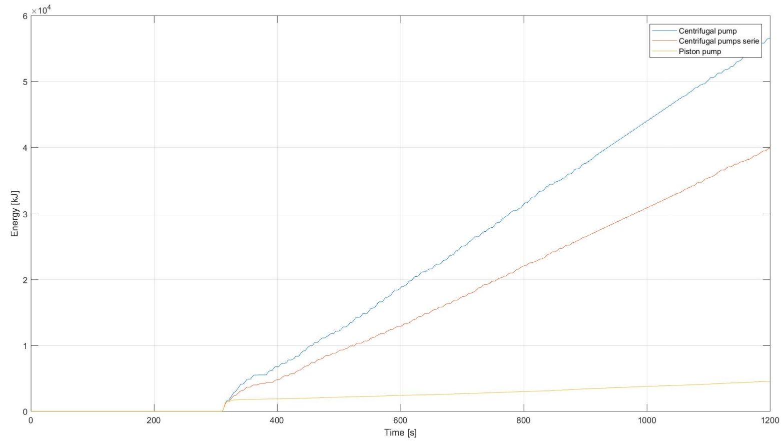


Figure J.22: Energy due to Anchorage submarine 2

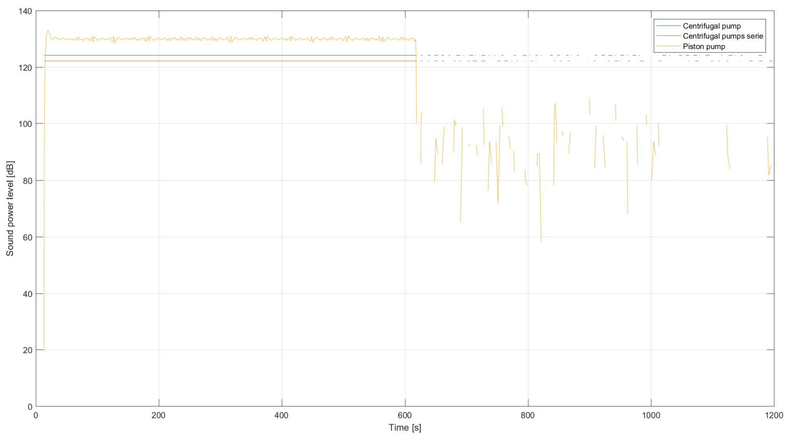


Figure J.23: Sound power level due to disembark AUV submarine 2

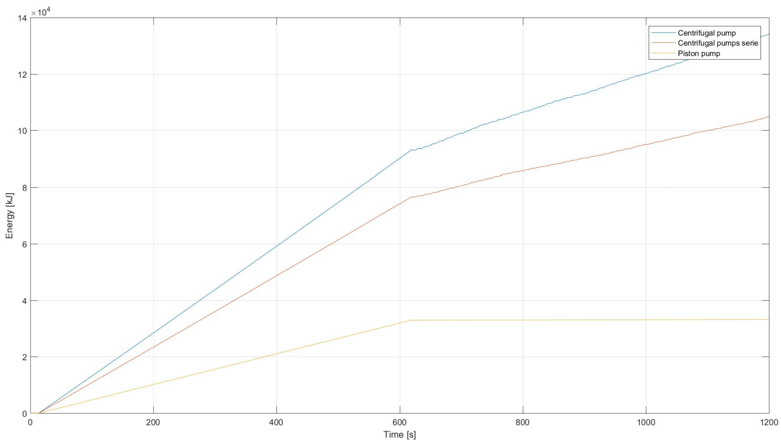


Figure J.24: Energy due to disembark AUV submarine 2

J.3. Hovering

J.3.1. Submarine 1, 4000 ton

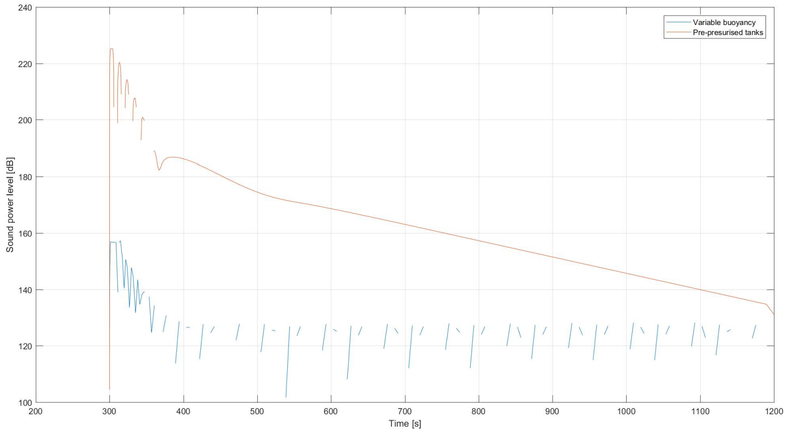


Figure J.25: Sound power level due to density change z=PD submarine 1

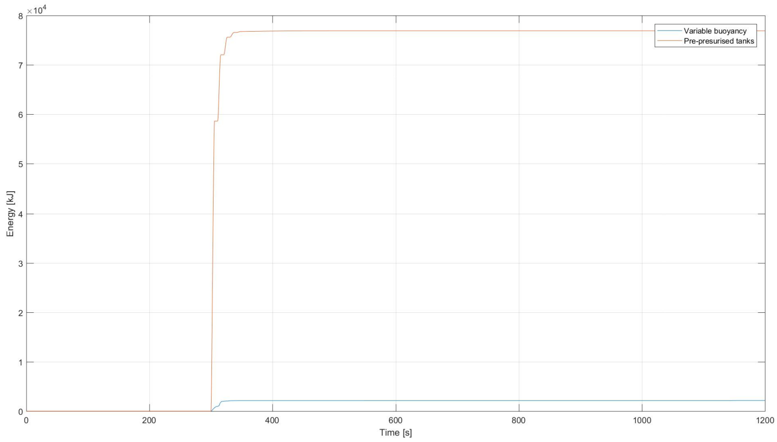


Figure J.26: Energy due to density change z=PD submarine 1

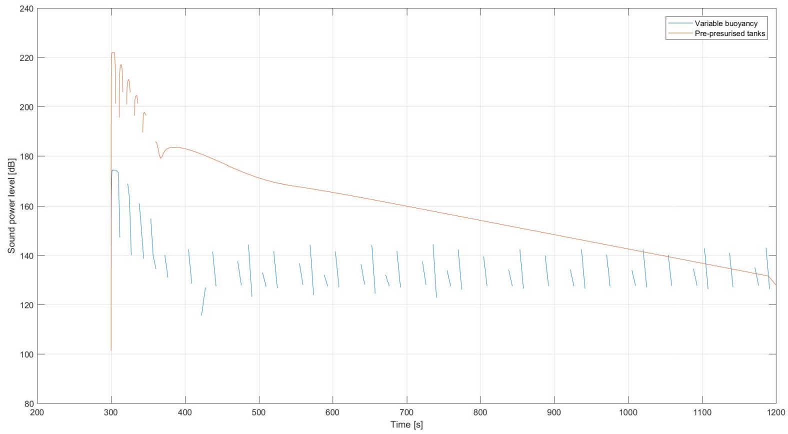


Figure J.27: Sound power level due to density change z=D submarine 1

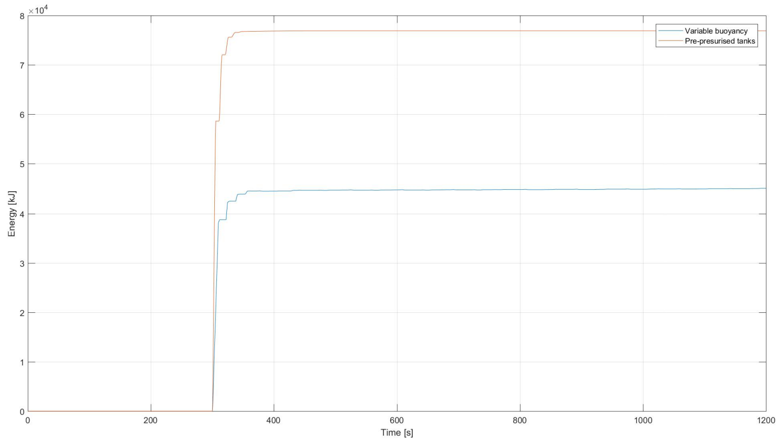


Figure J.28: Energy due to density change z=D submarine 1

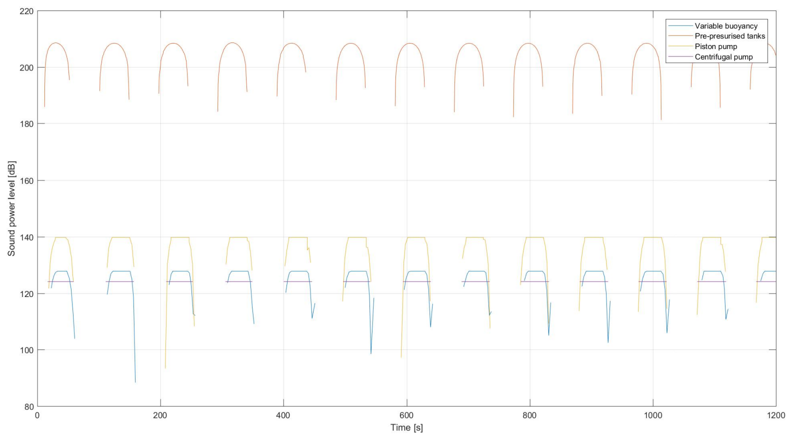


Figure J.29: Sound power level due to second order effects submarine 1

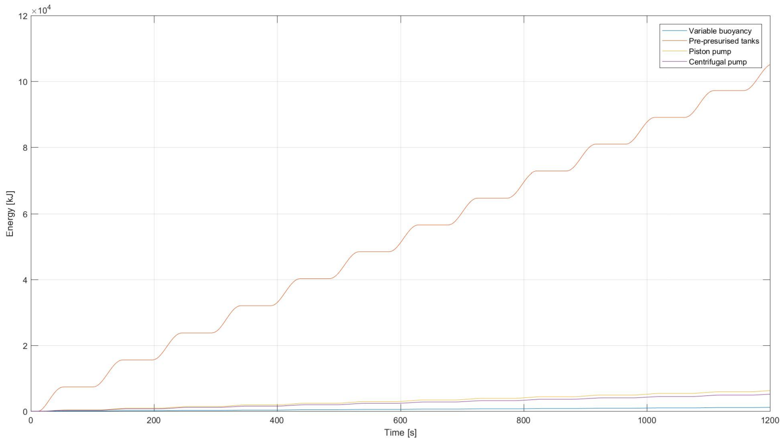


Figure J.30: Energy due to second order effects submarine 1

J.3.2. Submarine 2, 2000 ton

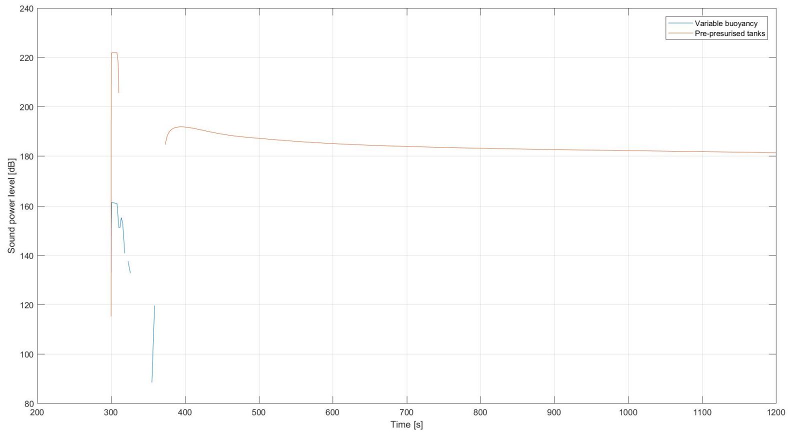


Figure J.31: Sound power level due to density change z=PD submarine 2

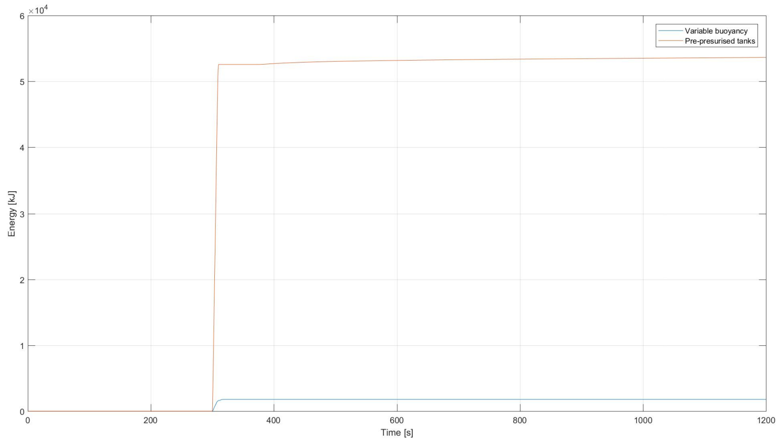


Figure J.32: Energy due to density change z=PD submarine 2

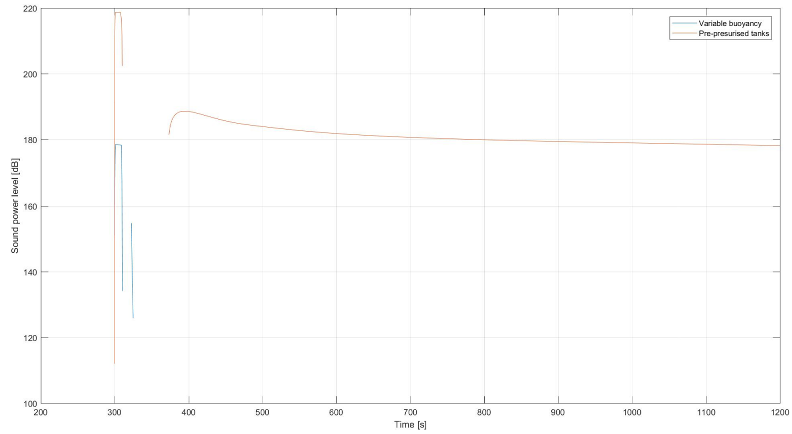


Figure J.33: Sound power level due to density change z=D submarine 2

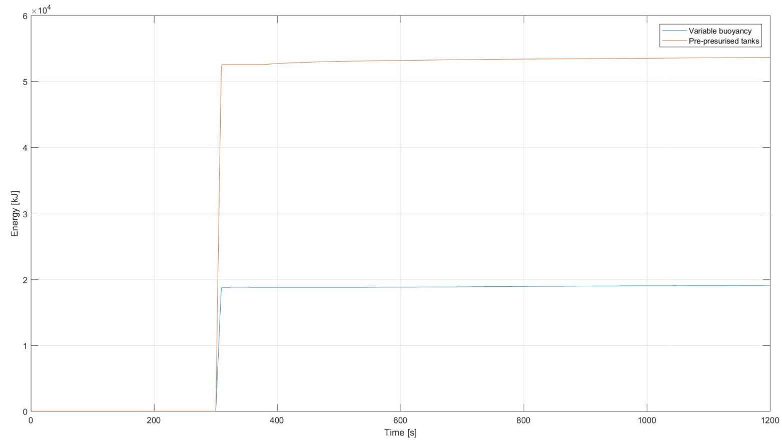


Figure J.34: Energy due to density change z=D submarine 2

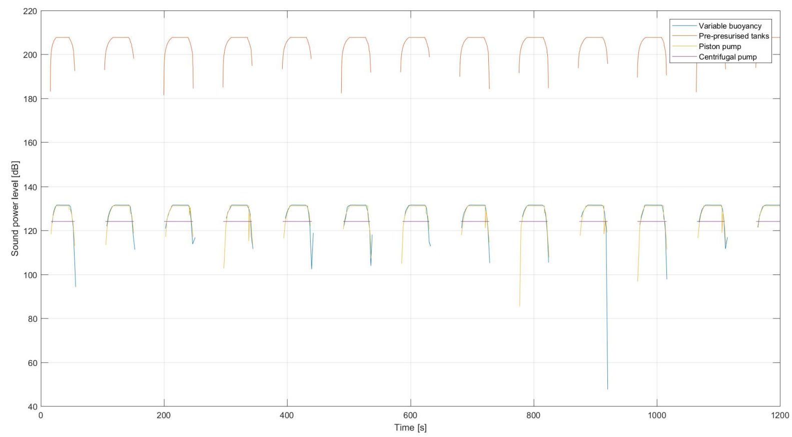


Figure J.35: Sound power level due to second order effects submarine 2

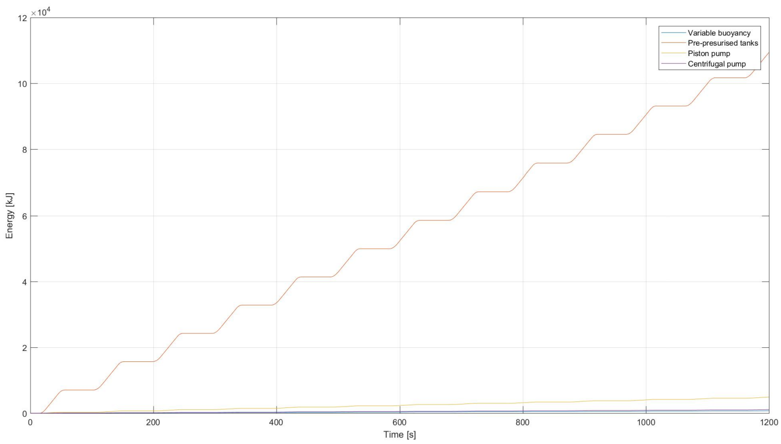


Figure J.36: Energy due to second order effects submarine 2

J.4. Combined

J.4.1. Submarine 1, 4000 ton

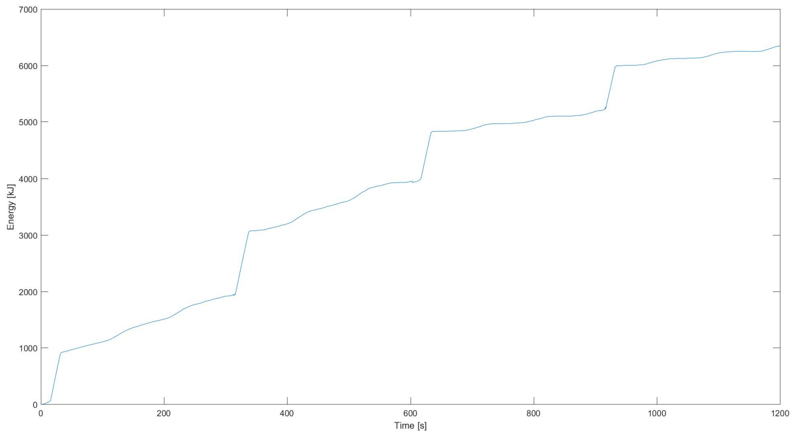


Figure J.37: Energy due to disembark SF submarine 1

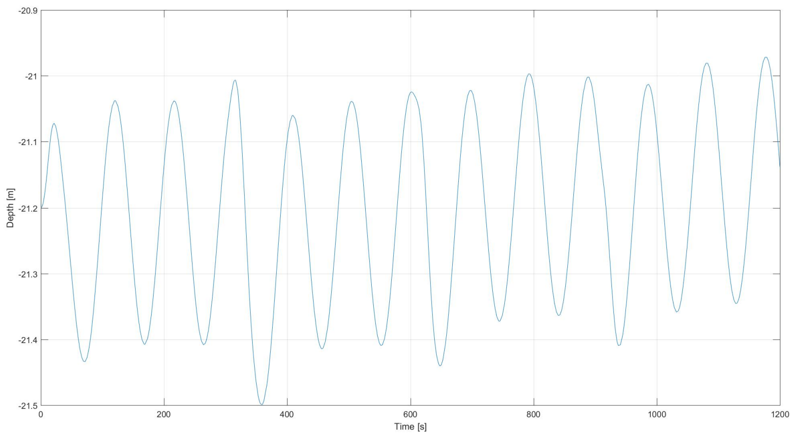


Figure J.38: Depth due to disembark SF submarine 1

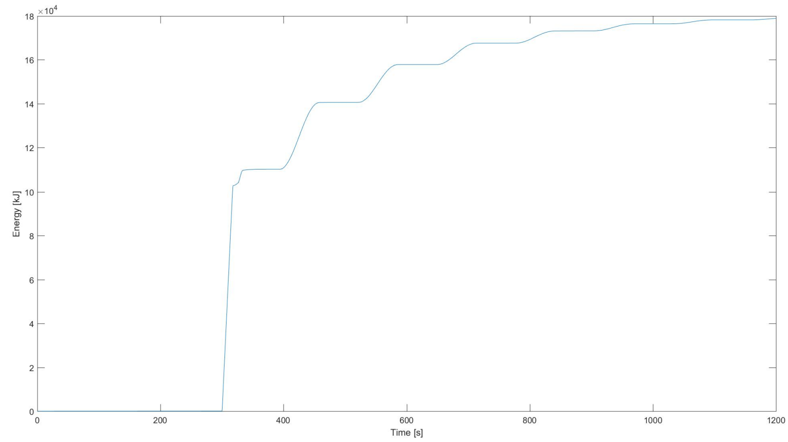


Figure J.39: Energy due to Anchorage submarine 1

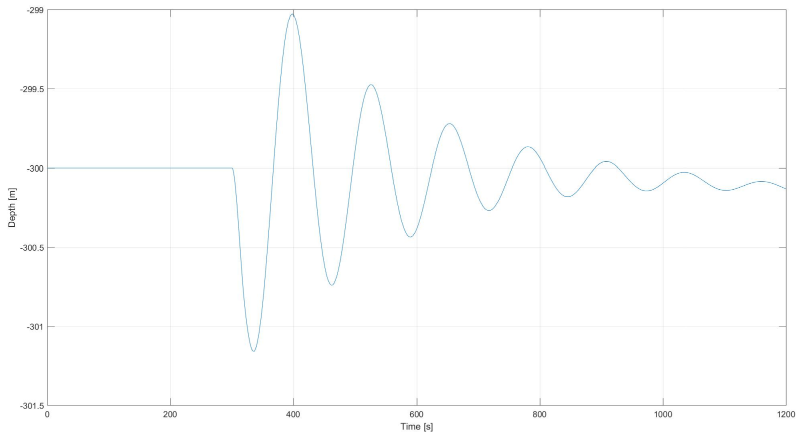


Figure J.40: Depth due to Anchorage submarine 1

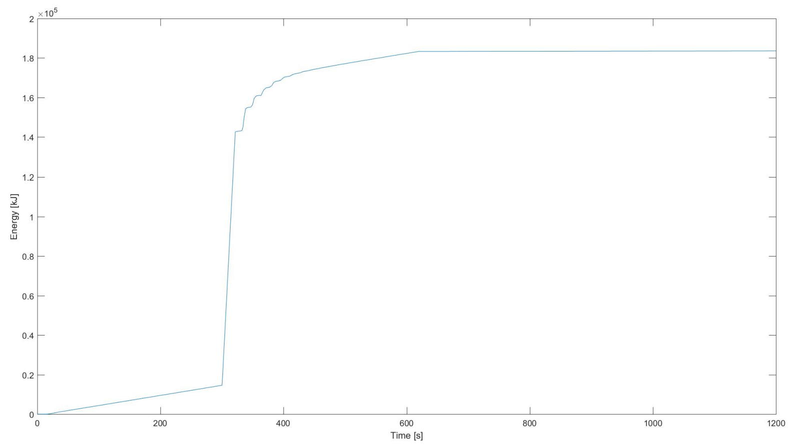


Figure J.41: Energy due to disembark AUV submarine 1

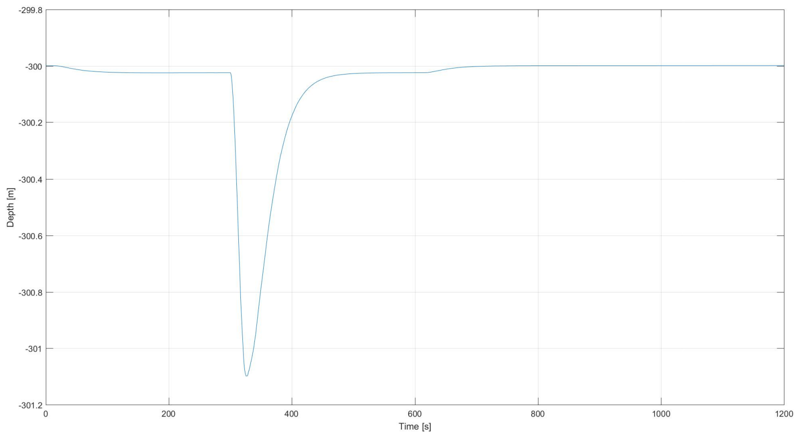


Figure J.42: Depth due to disembark AUV submarine 1

J.4.2. Submarine 2, 2000 ton

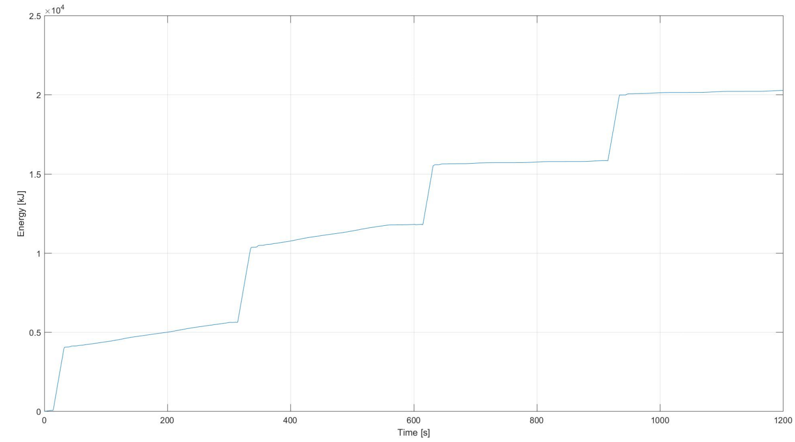


Figure J.43: Energy due to disembark SF submarine 2

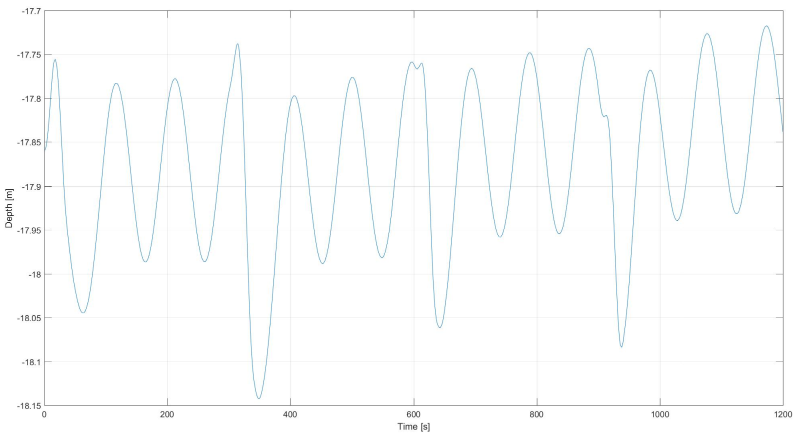


Figure J.44: Depth due to disembark SF submarine 2

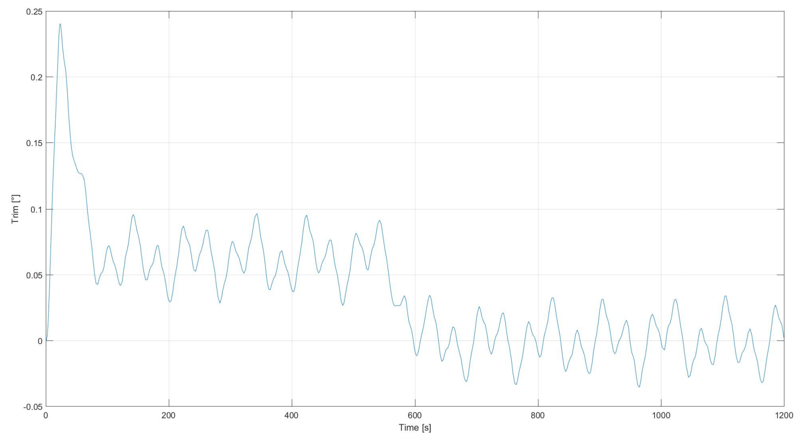


Figure J.45: Trim due to disembark SF submarine 2

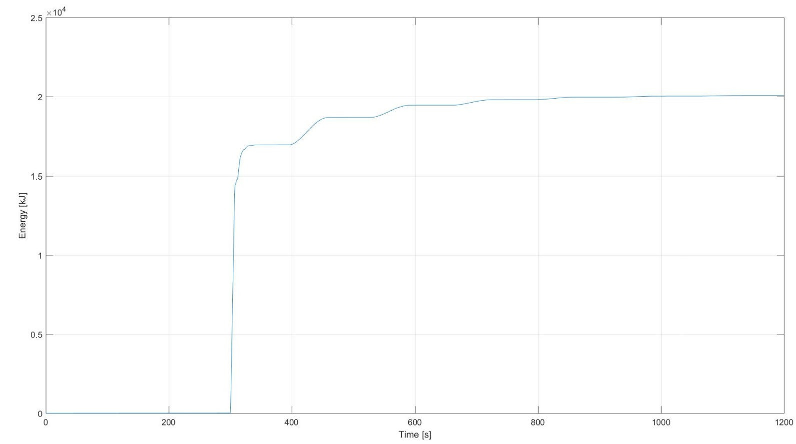


Figure J.46: Energy due to Anchorage submarine 2

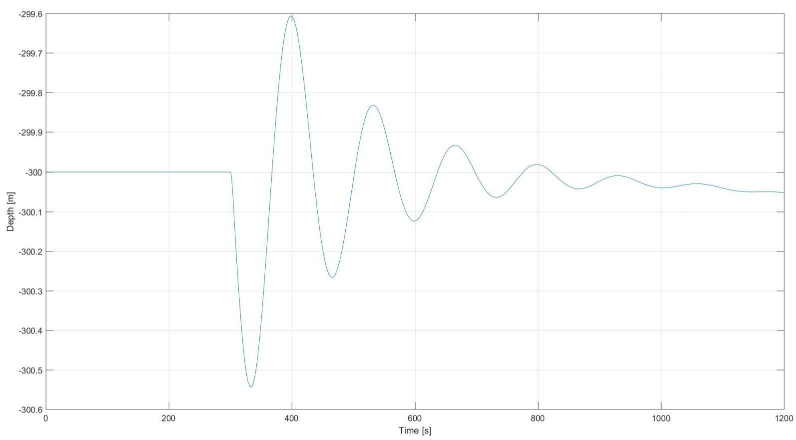


Figure J.47: Depth due to Anchorage submarine 2

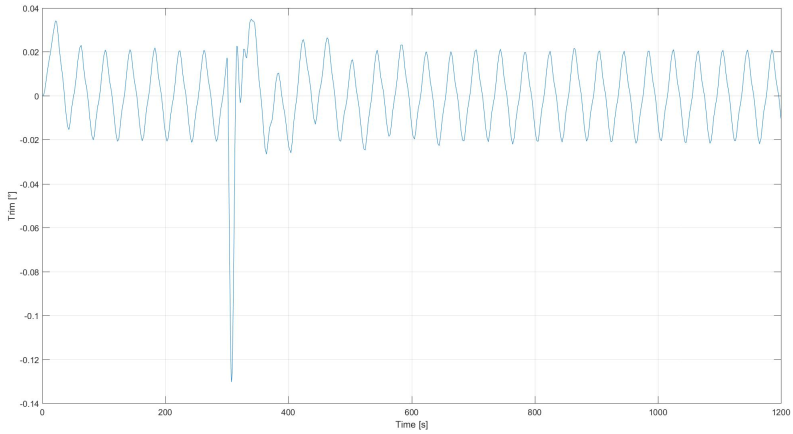


Figure J.48: Trim due to Anchorage submarine 2

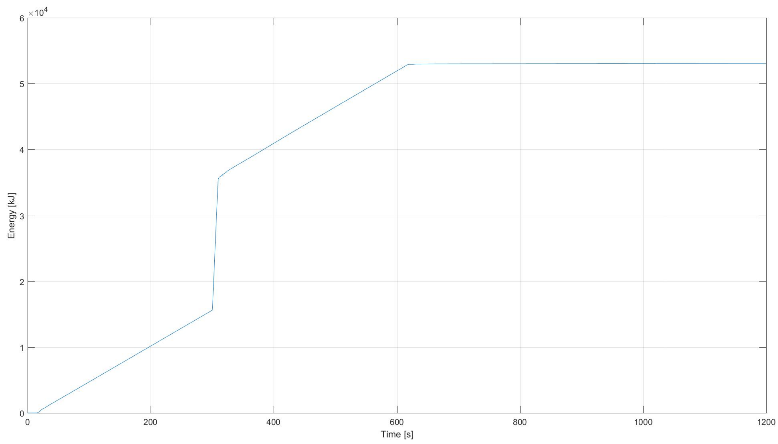


Figure J.49: Energy due to disembark AUV submarine 2

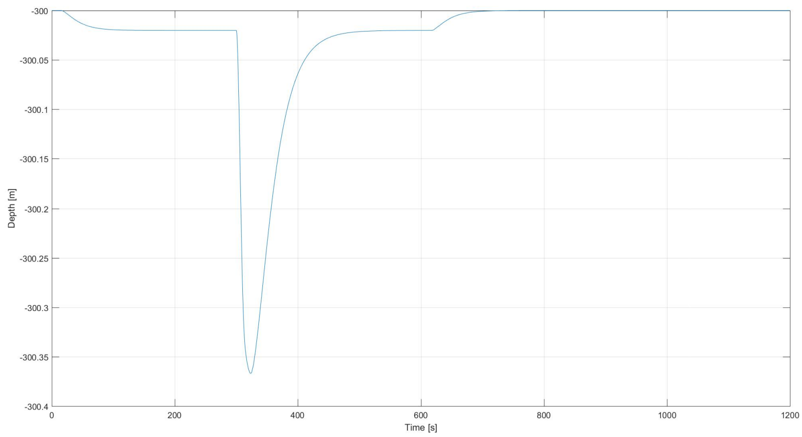


Figure J.50: Depth due to disembark AUV submarine 2

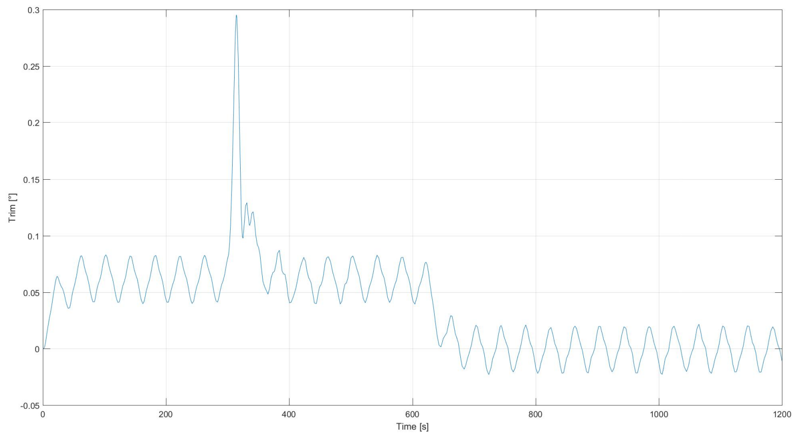


Figure J.51: Trim due to disembark AUV submarine 2