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ABSTRACT

The remotely operated vehicle represents a key element in cost effective development, maintenance and decommissioning of various subsea installations. The ROV is in particular useful when performing deepwater operations. In many cases unique and highly advanced tooling are developed to perform complicated operations on subsea structures. Without the ROV it would be very hard, if not impossible to develop deepwater subsea fields.

This thesis investigates how remotely operated vehicles are launched and recovered during subsea operations. The report takes a basic look at several common launch and recovery systems that are in use. The software SolidWorks have been used to design a 3D model of an improved ROV launch and recovery system.

On older drill rigs the ROV requires extensive guiding through the ROV moon pool. Usually the guiding is done physically by the ROV crew, using ropes, chains or “båtsake” (stick) to control the ROV.

The main focus when designing the improved launch and recovery system is to separate “man and machine”. This is done by integrating a Hydraulic Rotary Table on a Guide Beam. The Beam is guided by two 6 inch Guide Pipes running from the substructure of the moon pool and all the way up to the workshop area.

The Guide System together with the Hydraulic Rotary Table will allow the ROV to be mechanically rotated and adjusted before and during launch and recovery. The benefits of such a system will be less contact between the ROV and the crew. This concludes a safe and more efficient launch and recovery.

PREFACE

This master thesis is the concluding part of my master degree at the University of Stavanger. The thesis is carried out at the department of Deepwater Technical Solutions at Oceaneering AS in the period between 01.February and 16.June 2008.

The core aim of the thesis is to design a simplified ROV Launch and Recovery System that can be used on various oilrigs.

I would like to thank my Supervisors Jan I. Johanson at Oceaneering AS and Project Advisor Hirpa L. Gelgele at the University of Stavanger. They have both provided me with useful ideas and feedback during the whole research and design process. I would also like to thank the engineers and the ROV personnel at Oceaneering AS. They have provided me with great advice and assistance. Without their insight and patience the research process would have been much harder.

Stavanger, June.16.2008

Leiv-Ove Nordmark

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ABBREVIATIONS

The following definitions, abbreviations and symbols will be used throughout the document:

HPU	Hydraulic power unit
Capex	Capital costs
Opex	Operating costs
ROV	Remotely operated underwater vehicle
ROT	Remotely underwater operated tool
WROV	Work remote operated underwater vehicle
EOR	Enhanced oil recovery
IOR	Improved oil recovery
WOW	Waiting on weather
DTS	Deepwater technical solutions
Splash Zone	Interface between the air and the sea
Hs	Significant wave height
FPSO	Floating production storage and offloading unit
BOP	Blow out preventer
X-mas tree	Well head valve configuration
Workover	Total well work over
rpm	Rounds per minute
I	Mass Moment of Inertia
m	Mass
τ	Torque
α	Angular Acceleration
R, r	Radius
m	Gearwheel module
POM	Polyacetal Plastic Material
Båtsake	Stick with a hook on. Used to manually guide the ROV

TABLE OF FIGURES

1. INTRODUCTION

In recent years the subsea field developments have increased in both number and size. This is due to the technology progress and the growing demand of cost efficient solutions. Today approximately 50% of the petroleum production in the Norwegian offshore sector comes from subsea wells [9].

Maintaining subsea installations and equipment is often categorized as complicated and in some cases quite expensive. In the early days of the subsea field development, huge subsea structures would be detached from the seafloor, raised to the surface and transported onshore for maintenance work. This was both time consuming and very expensive. The operating expenses (OPEX) of a subsea development were then often categorized as rather large. New and redesigned "smarter" equipment allow for a simpler and more cost efficient maintenance procedure. At present time most of the maintenance work is done subsea while the equipment still is in production mode.

The combination of intelligent design, subsea structures, highly advanced remotely operated vehicles and sophisticated tooling technology makes the subsea maintenance work possible.

Due to the above the business market for remotely operated operations has never been better. This allows for great income and further development of the subsea ROV technology.

Because of the increased activity in the offshore sector, rig and vessel rates have in recent years been stratospheric. The rig utilization in the North Sea is today estimated to be at 95% [13]. The cost of renting drill rigs is huge. Even if operations are halted due to waiting on weather or similar effects, money is still being spent by the operator. This creates an increased interdependence of all rig related activities. Due to this, improved systems and back up systems are often developed so that there will be as little waiting on weather (WOW) as possible. This is also partly the topic for the thesis which is presented here.

The thesis is written in conjunction with Oceaneering AS which is one of the leading contractors of subsea ROV operations. The Norwegian main office, located in Stavanger, is in charge of tool development and ROV operations. Oceaneerings department of Deepwater Technical Solutions (DTS) have in recent years redesigned and improved several launch and recovery systems used on oil rigs all over the world.

The main objective of this thesis work involves designing and improving a basic ROV launch and recovery system. The system is designed so that it has guiding through and above the

ROV moon pool. The benefits of such a system will be safer and more reliable lunch and recovery. One of the main aspects of the design will be to ensure minimal contact between the ROV crew and the ROV.

The SolidWorks software has been used extensively to design a 3D model of the launch and recovery system. This has also been the main part of the thesis. Some basic calculations have been performed to determine the loads and stresses acting on the system. Especially concerning the design of the Hydraulic Rotary Table.

The thesis first highlights the related theories and previous works in this area. The review of the literature shows that the intent of this project is well founded and the result contributes to an effective and safer implementation of ROV system. The major part of the work in this project is reported in chapter 5 to 10 where the design solution is discussed, supported with drawings from SolidWorks, and the necessary strength calculations are presented.

Several figures and illustrations have been used to explain and simplify the understanding of the various topics and parts included in the launch and recovery system.

Finally, chapter 11 presents the concluding remarks.

2. SUBSEA OPERATIONS

2.1 Offshore oil and gas production

The offshore oil and gas production is often considered as more challenging and less cost efficient compared to the land based production. However, new and highly innovative technology is constantly pushing the limits of the offshore field development. A lot of the innovation in the offshore sector revolves around overcoming these challenges. This might be challenges such as dealing with harsh environment, ultra deep water, producing at great distances and extreme temperatures, improving already existing field developments and increasing the petroleum production with the help of IOR and EOR.

In recent years the oil price has been increasing dramatically. In 1996 price for crude oil was approximately US $20 per barrel. During the second quarter of 2008, the oil price has reached its highest level ever. One barrel of oil is now traded for approximately US $130. The extreme increase in the oil price allows for new and improved technology to be developed. This enhanced technology takes place in all aspects and areas of the oil and gas industry, and allows for smart and complex solutions to be designed. The increase in profit, due to the extremely high oil price, combined with new and improved technology, allow for smaller and complicated oil and gas reservoirs to be cost effective and hereby developed.

Irrespective of size and water depths, the subsea solution with subsea wells, tie back to shore [Fig 1B] or tie back to an existing installation/FPSO [Fig 1A], is strongly recommended. Smaller reservoirs are often developed using subsea solutions.

The recent increase of subsea installations has allowed equipment and concepts to mature to a very high standard. New and superior experience has been made from several recent projects such as Ormen Lange [Fig 1B], Snøhvit and Tordis IOR. This allows for a more cost efficient developments. What the oil industry do today was not even considered as a solution, just a few years back.

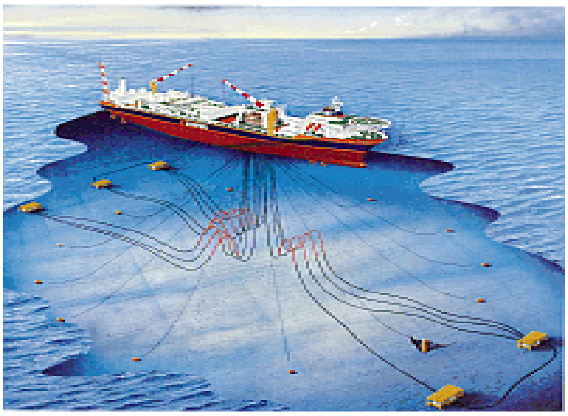
2.2 Subsea production

In general we choose between two types of offshore field developments, subsea or topside developments. The topside field development consists of either a fixed concrete installation, tension leg platform or a steel jacket. Most of the production facilities are in this case placed on top of the installation and well above the sea level. A topside field development is usually, in economic terms, characterized as a development with large capital expanses (CAPEX) and low operating expanses (OPEX) [9].

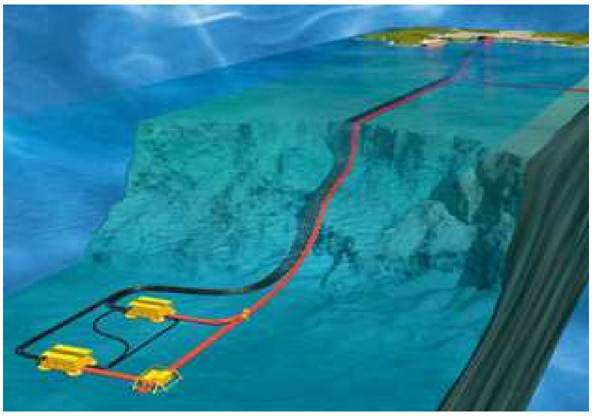
The subsea field development consists of advanced technology allowing most of the production equipment to be placed subsea. This includes subsea wells, X-mas trees, manifolds, flow lines, injection lines, boosting stations and in some cases even the process system is placed in special templates on the seabed.

With today's technology the subsea solution is defiantly the leading alternative compared to a full topside solution. This is due to the low CAPEX. Oil and gas production from subsea fields have in recent years increased dramatically. Today approximately 50% of all offshore oil and gas production in the Norwegian sector comes from subsea developments [9]. The subsea developments have in general a lower CAPEX compared to the huge topside structures. However, the subsea operating costs might in some cases be higher than with a topside development.

In recent years the ROV technology has allowed a lot of complex maintenance and repair work to be done subsea. This has definitely helped reduce some of the OPEX costs of a subsea development.



A) *Nornen subsea to FPSO development (Statoil)*



B) *Ormen Lange subsea to shore development (Hydro)*

Figure 1 *A and B represent advanced subsea field developments located in the Norwegian sector*

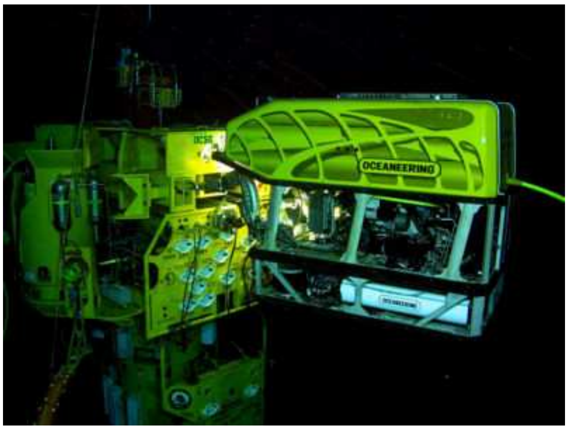
2.3 Remote operated operations

The US Navy was the pioneers behind the ROV technology. During the 1960 they developed technology used for deep sea diving. Their main focus was to develop the ROV so that it could perform deep sea rescue operations and recover various objects from the ocean floor.

In the 1980`s, when the offshore development exceeded the reach of human divers, the ROV technology was further developed and adapted into the offshore and subsea industry. Today the oil industry has moved into even deeper waters. Due to the ROV we are now capable of developing subsea fields located at 2500-3000 meters sea depth. [14]

The ROV is used in all aspects of the subsea development. The ROV can help carry out inspection, installation, repair, maintenance and decommissioning of all kinds of subsea structures (depending on size). Different tools are often used during the ROV operation. There are several tools on the market, both standard tools and tools specifically made for independent operations. The most common tools are the torque tool, seal replacement tool, cutting tool, and the cleaning tool.

The ROV is often used to connect different units together with the help of stabbers, which can transmit hydraulic power from one subsea unit to another. The ROV uses its manipulators (arms and fingers) to install and operate both the tooling and the stabbers. Visuals of the operation are transmitted from the ROV camera to the topside control room where the ROV pilot is located. The tetra cable or the umbilical transmits the signals. Usually a crew of three is needed for operating and maintaining the ROV during operation.



A) *ROV performing work on a subsea X-mas tree*

B) *Picture of torn hose provided by ROV*

Figure 2 *A and B illustrates different work scenarios of the ROV*

2.4 Remote operated vehicle

There are several types of Remote Operated Vehicles on the market. Some are specifically used for inspection and transmitting visuals from subsea to the surface [Fig 2 B]. Some are strictly used to carry out special subsea operations [Fig 2 A] where large tooling, equipment and remote operated tools (ROT) are involved. In general, the ROV and the ROT is vastly used during all subsea operations. Both cases are used during the entire life aspects of the subsea field development.

The ROVs are normally classified into categories based on their size, weight, ability and power. Some of the common ratings are:

- **Micro**
- **Mini**
- **General**
- **Light works class**
- **Heavy work class**

The difference among the classes is the power and size. The micro ROV class is very small, and typically less then 3kg [Fig 3 B]. It is used as an alternative to a diver. The moderate size is beneficial when it is necessary to gain visuals from the inside of a small structure where no diver would be able to enter. This might be structures such as a pipe or some kind of tubing.

The heavy work class ROV [Fig 3 A] is on the opposite end of the scale. Even though it has typically less then 200 hp (propulsion), it has the ability to carry at least two manipulators.

The heavy works class ROV has a working depth up to 3500meters [12]. During launch, the ROV sits safely inside a cage [Fig 4 A and B]. The cage is lowered to the desired operating depth, where the ROV leaves the cage. In some cases a tool basket might be added to the ROV cage so that extra tooling can be carried subsea. According to [1] and [16] this might save time and improve operational efficiency. Without the highly advanced ROV, most of the subsea developments would not be possible.

A) *The Magnum work ROV*

B) *The Minimum observation ROV*

Figure 3 *A and B represent different ends of the ROV size chart*

2.5 The ROV cage

The ROV cage works as a garage for the ROV. As depicted in figure 4 A and B, the cage has robust steel pipes designed to protect the ROV from bumping [Fig 4 C] into the moon pool during launch and recovery. There are several types of ROV cages used for launching. Typical for all types is that they are all designed to be extremely heavy and possesses as great of a weight to drag ratio as possible. Figure 4 A and B shows two of the typical ROV cage types. A very low center of gravity is achieved by adding lead to the bottom of the ROV cage.

If a cursor is used, both of these characteristics allow the cage and the ROV to contribute to the cursor performance. Additionally, once the cage and ROV have left the cursor and traveled to the works site several hundred meters below sea level, the low centre of gravity of the cage will help minimize the effect of upper and intermediate currents.

Since the cage is attached to the rig/boat through the umbilical, the heave of the cage will be the same as the heave of the rig/boat (if not heave compensated).

A) *Typical ROV cage*

B) *Typical ROV cage*

C) *ROV cage bumping into moon pool structure*

Figure 4 *Typical ROV cages*

2.6 ROV launch and recovery systems

Oceaneerings experience with the use of heavy weather launch and recovery systems began in the Gulf of Mexico in the 1960`s. The concept of launch and recovery was first applied to manned diving bells. The purpose then, as well as now, was to increase the weather window for launch and recovery.

A lot of knowledge was gained from these early experiences, and this was later applied to the ROV operations in the early 1980`s.

A number of variations and enhancements have been developed to fit all kinds of oilrigs and vessels individual needs. These variations in launch and recovery design allow Oceaneering to suite any drill rig or vessel with the proper launch and recovery system.

In the past years Oceaneering has installed and modified several heavy weather launch systems. This is due to the increase of vessels and oilrigs moving into deeper water and harsher areas. The escalating rig and vessel rates, as well as the companies` policy of less tolerance of injury (increased health and safety focus) can also be partly responsible for the improved launch and recovery systems. However, the most important reason is most likely the increased dependence on the ROV during subsea operations.

The ROV is today part of almost every subsea operation. It is used in all aspects throughout the field lifetime. The ROV is in use whether it is during survey, installation, commissioning, repair, maintenance or decommissioning of all kinds of subsea installations and equipment. This dependence on the ROV, and the enormous cost of vessel downtime, has made the launch and recovery system a crucial part of the ROV operation [1]. A proper launch and recovery system will allow for a safer and more efficient deployment of the ROV. The system will also permit launching during a greater significant wave height. This will allow for less waiting on weather and hereby a more cost efficient operation (less WOW).

3. HEAVY WEATHER LAUNCH AND RECOVERY SYSTEMS

The main purpose of the heavy weather launch and recovery system is to stabilize and centralize the ROV and the ROV cage with a device that restricts horizontal movement. It is also important to speed up the transition time through the splash zone. The air to sea interface presents the greatest risk of damage to the ROV, cage, equipment and potentially the vessel/rig. Large waves and winds of great magnitude can cause the ROV and cage to swing uncontrollably, with a chance of hitting the vessel or rig structure. As the ROV is raised out of the sea [Fig 5], the motion is amplified many times. This is due to the shortening of the umbilical as well as the lack of damping (running in air instead of water), as shown in figure 6 D. This will be similar to shortening the string on a pendulum, causing the motion to speed up. With the absence of Guide Wires or Guide Rails, this greatly amplified swinging can in some cases make it difficult if not impossible to recover the ROV. Normal procedure in that case would be to wait out the weather and to recover the ROV when the sea is calm.

Figure 5 *Typical A frame is shown inside the red circle*

Furthermore the swinging during the launch and the recovery can cause excessive side loading on the A-frame [Fig 5] and this might cause the A-frame to fail. Another hazard, on some vessels, is the close proximity of the ROV to the vessel thrusters. A lot of the new deepwater vessels use dynamic positioning systems that use thrusters rather than anchors to hold the vessel in place. These thrusters are very powerful and can easily damage or destroy the ROV system during a regular launch or recovery.

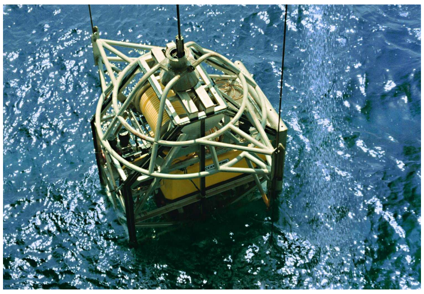
When using a heavy weather launch system, the pivot point will be moved form the A-frame to the bottom of the vessel. This minimizes the chance of having the ROV being pulled into, and damaged by the thrusters.

However, the single and most disruptive failure that can occur to an ROV system is the failure of the umbilical. A failure of the umbilical will cause a long period of downtime to repair or change the umbilical offshore. The time and cost of transporting a new umbilical offshore will also be of main concern. Though a failure of the umbilical rarely happens, it is most likely to happen if heavy wave action or high currents should drive or push the cage onto its side. This might cause the cable on top of the cage to be bent at a sever angle, causing the umbilical to fail. Connecting the umbilical after termination would be time consuming and expensive.

For this reason one of the main important parts of the heavy weather launch system is to maintain the vertical orientation of the ROV cage. The heavy weather launch system will assure a safe and efficient ROV launch and recovery. However, some restrictions must be followed. This might be proper launch and recovery procedures for the exact ROV in hand, as well as procedures for the right vessel used for the operation.

3.1 Three main heavy weather launch and recovery systems

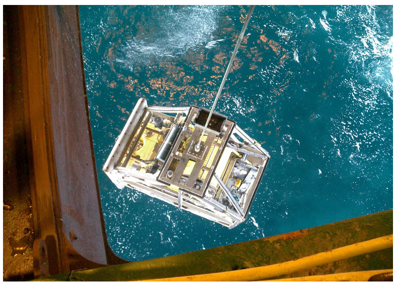
We may choose from three heavy weather launch and recovery systems. The three systems are the Guide Wire Cursor System, the Rail Cursor System and the Cursor Moon Pool System (boat). These Systems are shown in figure 6 A, B and C respectively.



A) *Guide Wire Cursor System*

B) *Rail Cursor System*

C) *Launching the ROV through moon pool on a boat*



D) *No guiding. Free pendulum from rig*

Figure 6 *Numerous methods are used when launching the ROV*

In some cases, and especially on older drill rigs, a primitive launch system with absolutely no guiding is used when launching the ROV. Such a system is shown in figure 6 D.

The main objective of the launch and recovery system is to assure a safe and efficient launch of the ROV. As long as the ROV is secured in the Cursor, the Guide Wires or the Rail System will safely guide the Cursor and the ROV up to the workshop/deck or down through the splash zone. The main objective of the Cursor is to encompass the top half of the ROV. This connects the ROV with the guide structure and allows the ROV to be safely guided.

3.2 Guide wire cursor system

The most commonly used cursor system in Oceaneerings fleet uses Guide Wires. This is a well known technology used to constrain the cursors path.

The system is typically used on drill rigs, tension leg platforms, and other similar installations. This method is useful when some horizontal movement is acceptable and there is no structure present to attach rails too. Cursor Guide Wire lengths range from 45-50 meters as shown in figure 6 A. The Guide Wire System includes a pair of 2 inch parallel wires, strung from the fixed A-frame on deck to the lower cursor arms. The cursor arms are either bolted or welded to the pontoon. Turnbuckles, which are used to tension the wires, are connected to the A-frame. The turnbuckles are used to increase the tension in the wires.

A specially designed and tested breakaway joint connects the wires to the lower cursor arms. This breakaway joint prevents damage to the A frame, vessel, and most importantly personnel in case a wire becomes overloaded.

The Guide Wire System enables Oceaneering to offer a heavy weather launch system with excellent operational characteristics for semi-submersible drilling rigs, and other similar vessels.

3.3 Rail cursor system

Guide rails [Fig 6 B] are used to constrain the path of the cursor. This method is often used on drill ships, FPSO`s, dive support vessels, intervention vessels and other vessels where minimal horizontal movement is desired. The cursor encompasses the ROV [Fig 7 A], and runs up and down the rail path. The rails are placed vertically at the side of the ship [7 A]. This assures a safe and ridged deployment and recovery of the ROV.

The guide rail system is also often used on vessels that launch the ROV through a moon pool [Fig 6 C]. The rails are typically 20-25 meters in length and they usually run from deck or the ROV workshop, and all the way down to the boats keel where they are terminated. The rails assure a safe and efficient heavy weather launch and recovery of the ROV.

3.4 Launch and recovery through moon pool (boat)

Launching the ROV through the moon pool of a boat [Fig 6 C and 7 C] allows for launching in extreme conditions. This is typically not possible in other ROV installations.

Usually the ROV workshop area is located inside the boat. The guide rails run from the workshop area and all the way down to the keel of the boat, where the rails are terminated.

The ROV is encompassed and stabilized by the Cursor. The Cursor is horizontally restricted and guided by the guide rails. The Cursor will carefully deploy or recover the ROV from a safe distance below the boat. A special feature of the internal moon pool is that it allows for a launch and recovery in ice infested waters.

3.5 The cursor

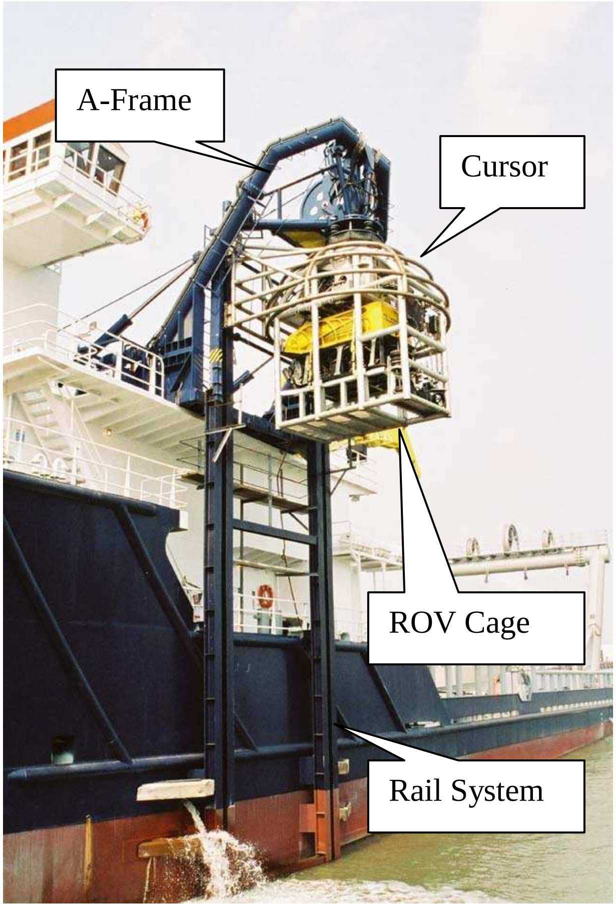
Oceaneering has several different types of heavy weather launch systems. They all have one piece in common. This piece is called a Cursor. The Cursor is shown in figure 7 A, B and C.

The Cursor is the essential piece of equipment that allows the heavy weather launch system to safely deploy and recover the ROV.

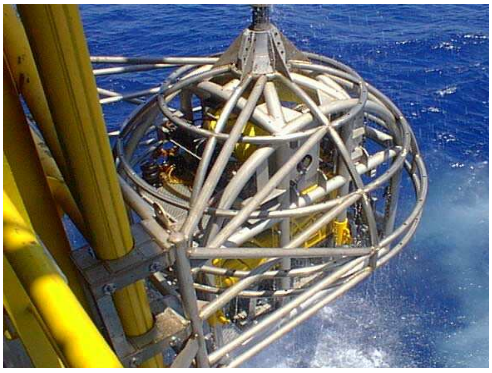
The Cursors are fabricated from stainless steel pipe with very few moving parts and almost no maintenance. It is in the shape of an upside down bowl that encompasses the top half of the ROV and cage.

The cursor travels a constrained path down the side of the vessel on guide wires or rails. The cursor travels with the ROV cage until the Cursor stops at the point where the wires or rails are terminated subsea. The wires or rails are usually terminated at the deepest possible level on the vessel structure.

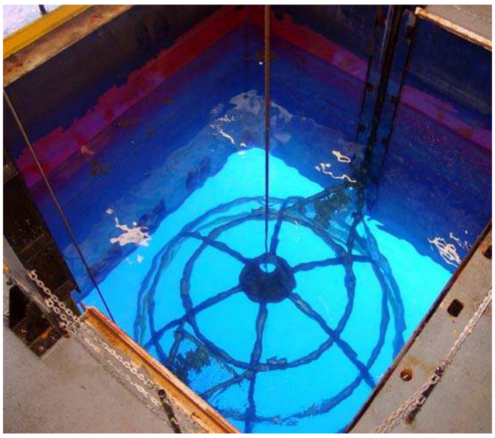
The Cursor is also perforated with 1 inch holes to allow it to flood as fast as the winch can lower it. This combined with the weight of the Cursor helps transit the cage through the waves as quickly as possible. This reduces "hesitation" and the chance of damage to the cage/ROV.



A) *Rail Cursor System*



B) *Typical Rail Cursor System*



C) *Typical Moon Pool Cursor System (boat)*

Figure 7 *The Cursor is shown in picture A, B and C*

4. ROV WEATHER TOLERANCE DIAGRAM

The ROV Tolerance Diagram illustrates the allowed significant wave height when launching the ROV. The allowed significant wave height might vary depending on the launch system that is being used. Figure 8 and figure 9 illustrate the difference in use of launch system. Figure 8 shows allowed significant wave height when using a launch system without the Cursor Guide System. Figure 9 shows the allowed significant wave height when launching with the Cursor Guide System. The column on the right illustrates the maximum allowed heave on the cage. Since the cage is connected to the rig, the heave will be the same as the heave experienced by the rig. Usually the heave of an operating rig will be approximately half of the wave height. This will off course depend on the rig design and rig mode. A boat will have a larger surface area, and hence a larger heave.

4.1 ROV launch with no guiding

When launching without any form of guiding the ROV can only be launched safely from the oilrig as long as the significant wave height, H_S does not exceed 5meters (see figure 8). In case H_S exceeds this maximum limit, then the ROV launch or recovery will have to wait till the sea state is reduced and H_S is less than 5meters. [1]

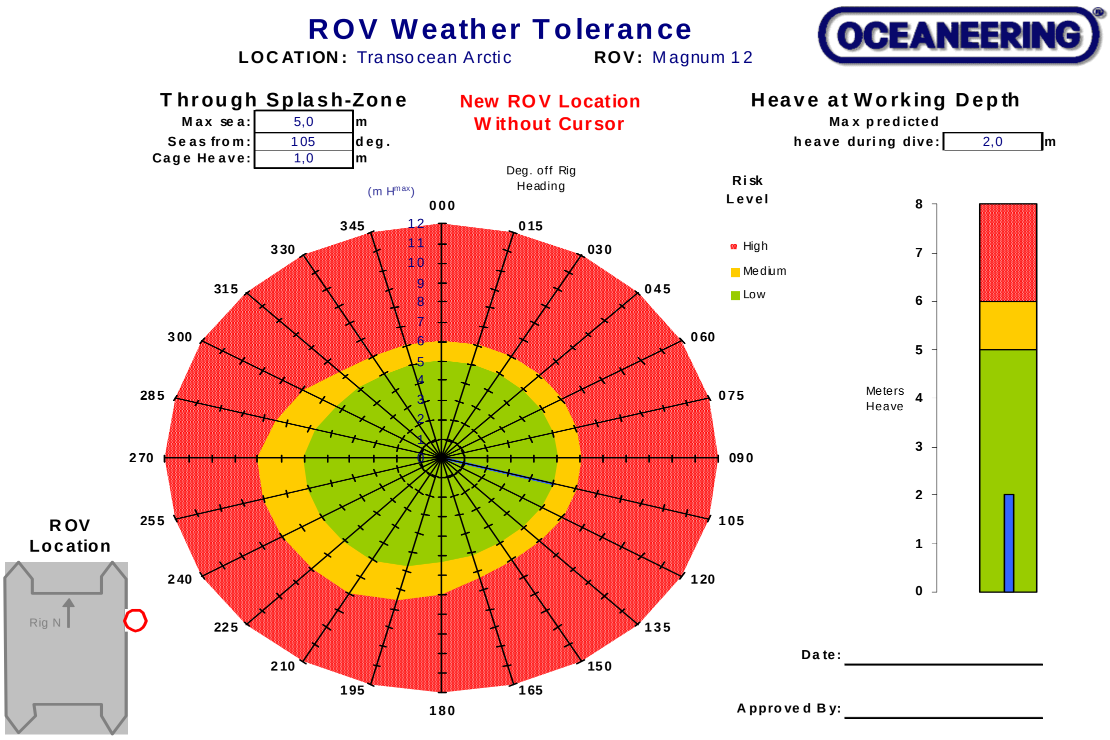


Figure 8 *ROV Weather Tolerance Diagram for ROV launching without Cursor System, Transocean Arctic*

4.2 ROV launch with guiding

When launching with a heavy weather launch system such as the Guide Wire Cursor System or the Rail Cursor System, the ROV can be launched safely from the oil rig as long as the significant wave height H_S does not exceed 9 meters (see figure 9.)

In case the significant wave height exceeds this maximum limit, then the ROV launch or recovery will have to wait till the sea state is reduced and H_S is less then 9 meters. [1]

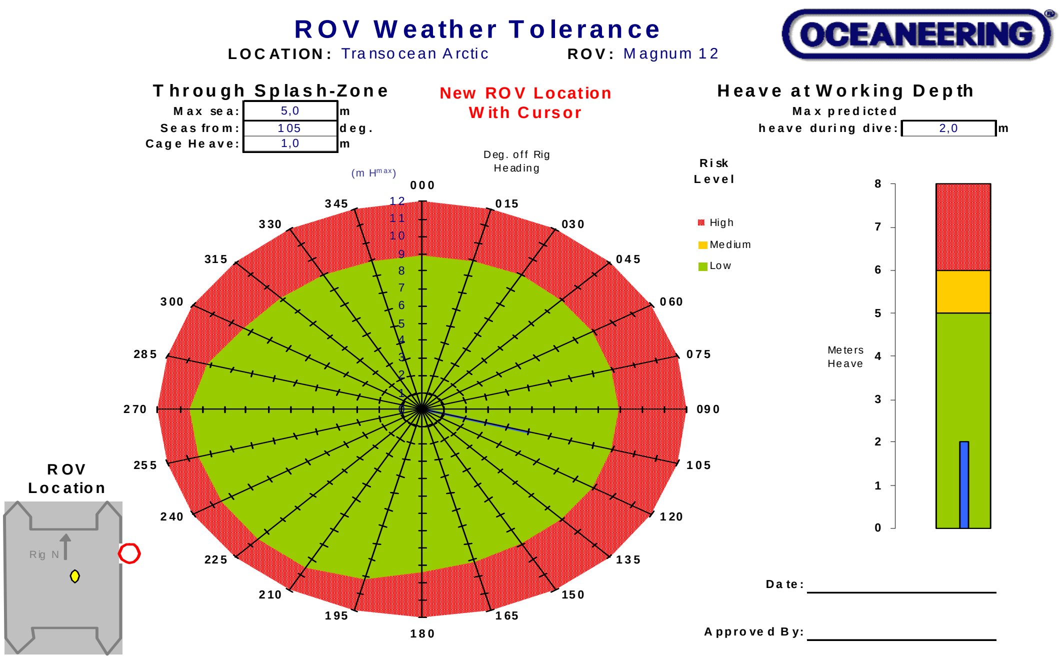


Figure 9 *ROV Weather Tolerance Diagram for ROV launching with Cursor System, Transocean Arctic*

If we put this into perspective and compare the two launch and recovery methods in a sea state such as the one found in the North Sea, we clearly see that a heavy weather launch system will be much more efficient.

Both of the two heavy weather launch systems, the Guide Wire and the Guide Rail, will reduce the waiting on weather (WOW) substantially compared to a system with no guiding. By other means, the ROV will be more operative when using a heavy weather launch system.

"The price you pay for not being able to operate during a seasonal storm is the cost of installing the heavy weather launch system". [16]

5. DESIGN CONSIDERATIONS

When designing a ROV launch and recovery system, a lot of different considerations need to be taken into account. First of all it is very important to know the exact environmental conditions. This might be the depth, sea state, vessel type and general vessel limitations. It will also be important to equip the vessel with the correct ROV system. The system needs to be suitable for the desired operation. Occasionally several ROVs are required. Some of the ROVs might carry tooling and some might only be used to observe and transmit visual information from the seabed to the surface. A proper launch and recovery system, which is able to perform in the operating sea state, is most important.

Several miscellaneous systems are available that are suited for the sea state in consideration. For example for a heavy sea state similar to the one found in the North Sea, either a heavy weather launch system such as the Guide Wire Cursor System or the Rail Cursor System would be to prefer. On the other hand, a regular A-frame combined with a winch structure might be all that is needed to perform launch and recovery of the ROV, in a fairly calm sea state.

A lot of the older oilrigs where designed to launch manned diving bells. Some of these rigs are still in use. However, the diving bells have in recent years been replaced by ROV systems. The ROV systems are usually bigger in size and occupy more space of the moon pool area.

Due to this, a lot of the oilrigs do not have proper ROV guide systems, such as the Guide Wire Cursor System or the Rail Cursor System. When launching from these rigs the ROV is carefully winched from the ROV workshop, through the rig moon pool and into the sea 15-20 meters beneath the moon pool. During this maneuver, the ROV is manually guided through the moon pool with the help of chains, ropes and sticks (båtsake).

At some rigs the ROV is actually winched through several moon pools before it reaches the sea. This makes for a very complicated launch and recovery. During both launch and recovery heavy winds and sea will cause the ROV to swing and rotate.

During the launch and recovery procedure the ROV crew member is encountered with many hazards. The moon pool area is frequently small and hectic. Various equipment and structures often cause launch and recovery problems. This might be gangways or other rig structures such as tools, various machinery and equipment that are located close to or in the actual launch path of the ROV.

Throughout this type of launching, the 4-ton ROV can swing and rotate freely with barely any restraints. Due to no guiding the pivot point of the pendulum behavior of the umbilical is located all the way up on the A-frame. This can be 6-10meters over the moon pool and 21-30 meters over the surface. This pendulum behavior is one of the reasons for a complicated launch and recovery. Wind and heavy seas cause the ROV to swing. However the biggest problem is that the ROV is constantly rotating and spinning without any form of restriction. This makes it very hard to lower or raise the ROV through the tight moon pool.

An important part of the design will be to improve the overall health and safety situation of the ROV personnel. The ROV crew is in charge of all ROV related issues during the entire ROV operation. This includes both launch and recovery of the ROV.

Typical personnel injuries during launch and recovery would be squeeze of hand, fingers and toes. More serious injuries to personnel might be man over board or snap of the umbilical or the A-frame. This can potentially cause heavy equipment hitting personnel. In both cases death or serious injury might be the outcome. Other less serious consequences might be damage to moon pool equipment, ROV tooling, rig structure and the ROV itself.

To deal with the above, a full guide system such as the Guide Wire Cursor System or the Rail Cursor System would be to prefer. However such a system is very complicated and not suitable for offshore installation. To be able to install a full cursor system, with guiding of the ROV below and above the moon pool, the oilrig will have to undergo a fairly complicated installation work. This can only be done onshore in a docking facility. At present time, and with the oil industries huge activity level, this is not an option. The installation work will have to be done offshore when the rig is in drill/production mode.

Based on the above stated problems, the objective of this thesis is to design and develop a system that can guide the ROV through and above the moon pool. The Guide Structure will have an integrated Hydraulic Rotary Table that will permit rotary motion of the ROV before it enters the moon pool. This will assure that the ROV is parallel to the moon pool opening without physically involving any of the crew. It will also allow for rotation of the ROV incase the launch path is blocked. The design will ensure that the ROV launch and recovery system is easy to install and can be installed offshore while the rig is in drill/production modus. The installation work will only require work performed above the moon pool.

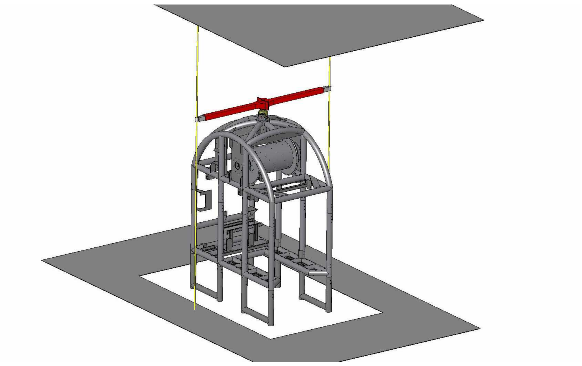
I believe the ROV crew will have huge benefit from such a system. The launch process will be safer and the ROV launch and recovery will be smarter and more efficient.

6. LAUNCH AND RECOVERY SYSTEM

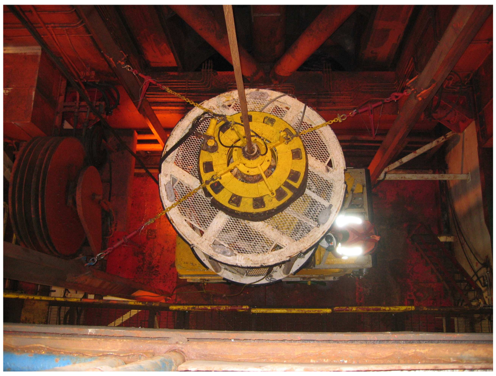
The main part of this project is to design a smart ROV launch and recovery system that is easy to install, safe to operate and most of all a system that will improve the ROV pilots work situation substantially.

When launching through a rig moon pool, a special Guide Wire System or a Rail System is usually used in combination with a Cursor. Both the Guide Wire Cursor System and the Rail Cursor System are used to restrain the horizontal movement of the ROV while it is lowered into, or pulled out of the sea. The two systems are well known and widely used all over the world [Fig 6 A and B]. Nevertheless both systems are complex and cannot be installed while the rig is offshore. This consents for a new system to be developed. Due to the cost of “rig down time” the new launch system has to be installed offshore, while the rig is in production/drilling mode. The new system has to be a robust and simple system that is easy to install. Due to lack of space the system has to be compact and smart. One of the main goals will be to implement health and safety issues and improve the work condition for the ROV personnel.

The new launch system will not be a full cursor system with complete guiding. The designed system will only have guiding above the moon pool. However, the new guide system will lower the pivot point of the umbilical from the A-frame to the lowest point of the moon pool. The system will also have full guiding from the moon pool and all the way up to the A-frame. This will allow for less pendulum behavior of the ROV, as well as a guided and smart launch and recovery through moon pool.



A) *Concept study of ROV launch and recovery system with integrated hydraulic rotary table. Guide Wires are in this case used to restrain the rotation* of the Guide Beam while rotating the ROV.



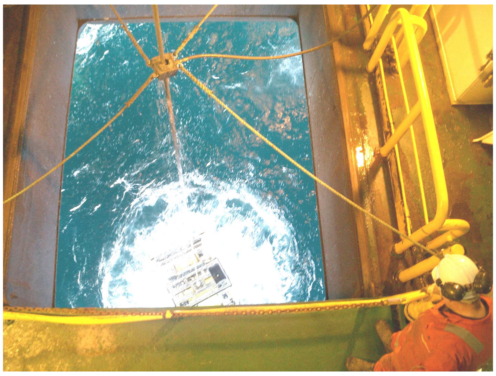
B) *Old Hydraulic Rotary Table concept used on the rig Deep Sea Delta. Chains are used to restrain the rotation of the Rotary Table Structure, while rotating the ROV*

Figure 10 *A and B represent similar concept for guiding the ROV through moon pool.*

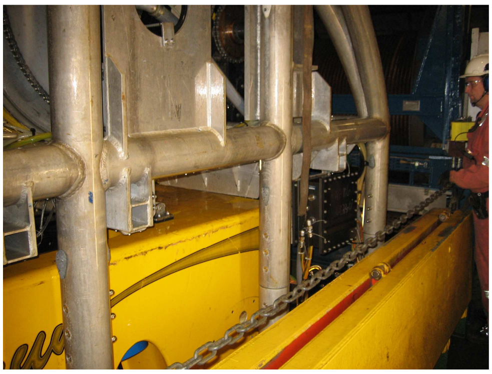
Since there will be no guiding beneath the moon pool, a special system has to be developed to be able to align the ROV before it is being pulled through the tight moon pool. This system will consist of a Hydraulic Rotary Table design [Fig 10 A and B].

The Hydraulic Rotary Tables main objective will be to align the ROV cage before it runs through the moon pool. This will be a great improvement compared to the older system where the ROV crew had to guide the ROV cage manually by hand with the help of sticks, rope or chain [Fig 11 A and B].

Furthermore the new system permits a safer operation. When installed, there will be no need for personnel to manually guide the ROV cage when it runs through the moon pool. This will improve the human environmental safety factor substantially.



A) ROV being manually aligned with ropes during launch and recovery



B) ROV is launching through a tight moon pool. Chains are used to guide the ROV

Figure 11 *Manually Guiding the ROV through moon pool*

7. DEVELOPMENT OF LAUNCH AND RECOVERY SYSTEM

The design process started of with a concept meeting with some of the engineers at Oceaneering AS. After briefly studying a number of concepts, we all agreed on one particular concept. A basic execution plan for the design stage was put together and potential problems where defined and discussed how they best could be solved. The development of the Launch and Recovery System was split into four main components.

1. **Design of Hydraulic Rotary Table**
2. **Design of Guide Beam**
3. **Design of Guide Structure**
4. **Design of Interface between ROV cage and Rotary Table**

It was decided that the Hydraulic Rotary Table would have to be an integrated part of the Guide Beam. The Guide Beam would then be serving as a base structure where the Hydraulic Rotary Table was to be mounted. With the help of a friction surface between the ROV cage and the Rotary Table, the Rotary Table should be able to rotate the ROV. The Guide Structure would consist of two parallel pipes guiding the Guide Beam through the moon pool area. The guiding would stop when the ROV leaves the moon pool (when launching). The advantage with this system is that it allows the ROV to be rotated mechanically during launch and recovery. The pivot point of the pendulum behavior will also be substantially lower. This will cause the ROV-swing amplitude to decrease (less swing of the ROV). Though the ROV will not be guided through the splash zone, it will have full guiding through and above the moon pool. This will improve the ROV crews work situation considerably. Less contact between ROV and its personnel is expected.

Improvements during Launch and Recovery:

1. **Less contact between ROV and crew**
2. **No impact between ROV and moon pool**
3. **Lower pivot point of the umbilical (reduced swing)**

8. DESIGNING THE LAUNCH AND RECOVERY SYSTEM

When designing the ROV Launch and Recovery System a lot of different aspects need to be taken into consideration. Details such as weight and size of the ROV and ROV cage, size of the moon pool area, size of the moon pool itself, choice of material and so on are all important factors. Launch Systems are seldom identical.

Due to the great expenses of disconnecting the umbilical, all parts must be designed so that they can be mounted without disconnecting the umbilical. This is mainly done by splitting all parts of the rotary table into two halves.

It is at its most important that all parts must be designed so that they will fit perfectly together and manage estimated loads exerted to the system. During this process several calculations have been made so that the design will be acceptable and strong enough to withstand any of the expected loads. Most of the calculations are found in chapter 9.

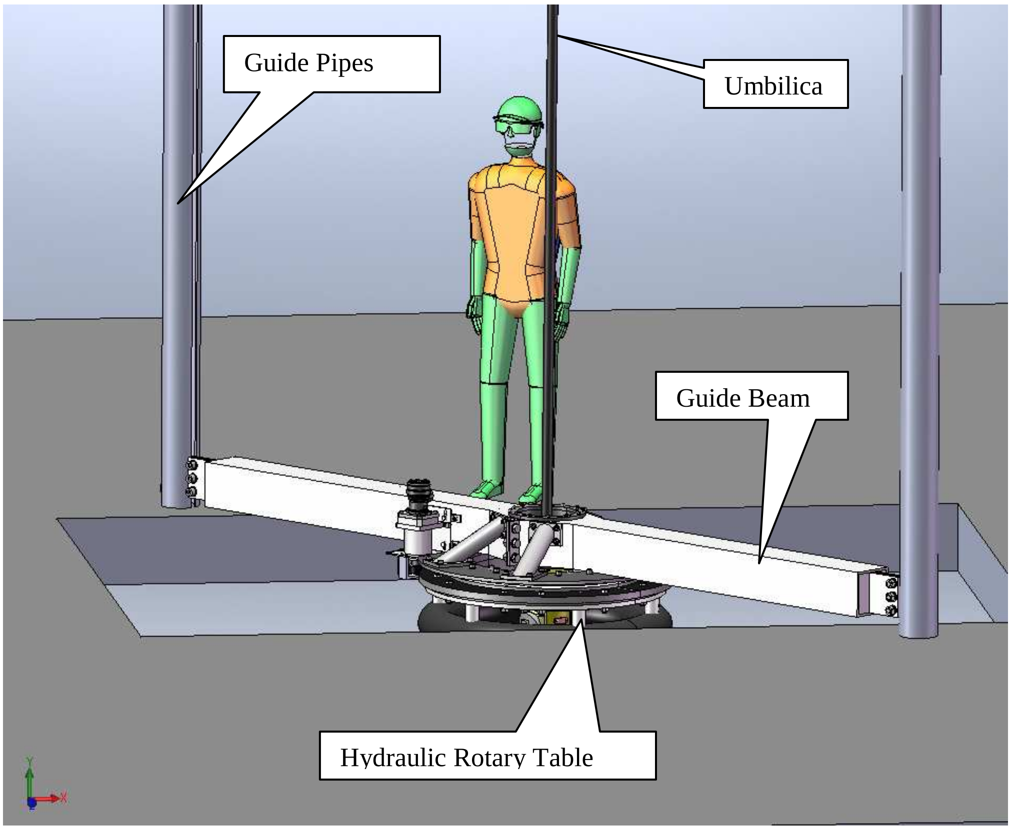


Figure 12 *ROV Launch and Recovery Design.*

8.1 Hydraulic rotary table

The Hydraulic Rotary Table is assumed to be the most complicated part of the launch and recovery system. It will allow the ROV to be mechanically rotated and aligned with the edge of the moon pool, before it is launched or recovered. The Rotary Table will furthermore correct the ROV while it runs through the moon pool area where personnel, equipment or various structures might be located. This might be structures such as gangways, rig structure, pipe configurations or similar. Structures such as these are quite common on older oilrigs that have been refit with new and bigger ROV equipment.

The Hydraulic Rotary Table will be an integrated part of the Guide Beam. The umbilical which is connected to the ROV cage will be running through the Rotary Table. All components of the Hydraulic Rotary Table must therefore be designed so that they can be installed without detaching the umbilical. Basically this means that most of the parts will have to be split into two equal halves and then coupled around the umbilical.

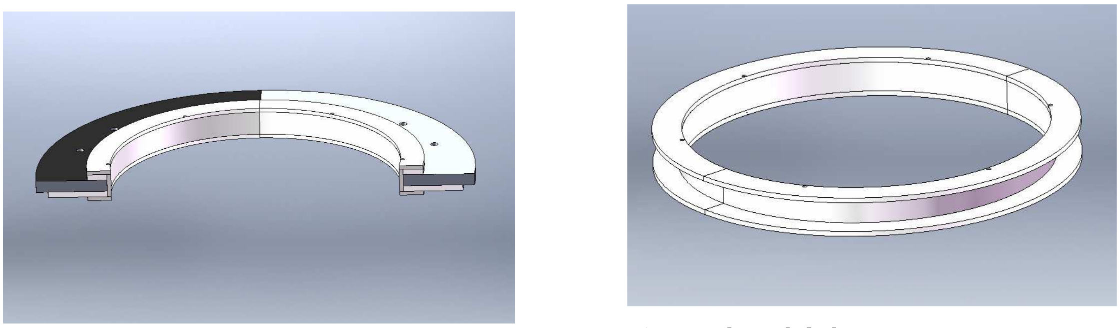
The Hydraulic Rotary Table design will consist of several parts. Main components in the system are the Hydraulic motor, Gearbox, d = 1120 mm HPC Gearwheel and the d = 760 mm Polyacetal Slide Bearing.

8.1.1 Polyacetal slide bearing

The POM slide bearing is produced by Astrup AS, which is one of Norway’s leading companies within plastic and metal distribution.

The POM slide bearing will have an inner diameter of 740 mm and an outer diameter of 760mm. The slide bearing is only 10 mm thick. The slim POM bearing allows for a rather large heat transfer. This will cause the cooling process of the bearing to be rapid. The ROV is only turned 1-2 revolutions during either launch or recovery, so the rotational speed of the bearing is extremely low. This combined with the rapid cooling makes it unnecessary to calculate the wear out of the bearing (PV factor). The wear out of the bearing will be very low due to the extremely low rotational speed.

Since we cannot detach the umbilical, due to a cost and time consuming operation, we will have to design the slide bearing so that it can be split into two halves. The two equal parts of the slide bearing will be latched together with the help of a top and bottom slide bearing flange. The slide bearing flange will connect the two slide bearing parts. The bottom flange will support the weight of the “hang off load”, F_Z. F_Z is a force that represents the weight of all parts that are supported in the slide bearing. This meaning that the force, $F_{Z,}$ is acting on the Slide bearing. The total weight of the "hang off load" is 269 kg ($F_Z \approx 2640N$). The surface between the steel Latch Plate (latching the big Gearwheel together) and the plane bottom Slide bearing flange causes friction in the Slide bearing. The friction force is calculated in chapter 9.1.2. The transfer of power from the small gearwheel to the big driven gearwheel also produces friction in the slide bearing. This is calculated in chapter 9.1.3 and 9.1.4.



A) *The driven Gearwheel, upper and lower Latch Plate is supported in the Slide bearing.*

B) *Complete Slide bearing*

Figure 13 *Details of POM Slide bearing, Gearwheel, upper and lower Latch Plates are shown in figure A and B*

The Slide bearing will be installed onto the Rotary Table Support Structure shown in figure 14 below. The Slide bearing will sit fairly loose around the Support Structure. This will allow either the Slide bearing to rotate around the Support Structure, or the Gearwheel with its Latch Plates [Fig 13 A and Fig 15 A and B] to rotate around the POM Slide bearing [Fig 13 B]. Either way, there will be a rotation causing the ROV cage to rotate.

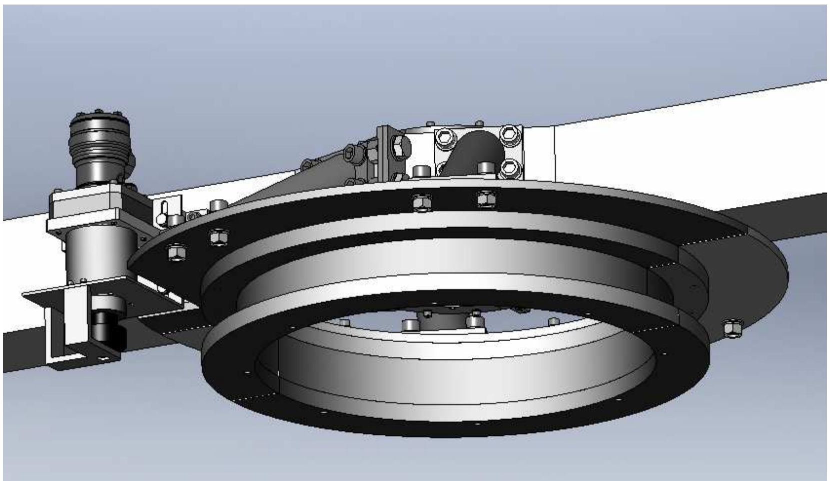


Figure 14 *Rotary Table Support Structure.*

8.1.2 Gearwheel

Both the big Gearwheel (pitch circle diameter d_1 = 1120mm) and the small Gearwheel (pitch circle diameter d_2 = 80mm) is produced by HPC Gears. HPC Gears is a company based in the UK, and is specialized in producing gears for various size and purpose. Both gearwheels will have a thickness of 30mm and module 4. Thus the number of teeth (Z) of the gears is:

Driven Gearwheel:

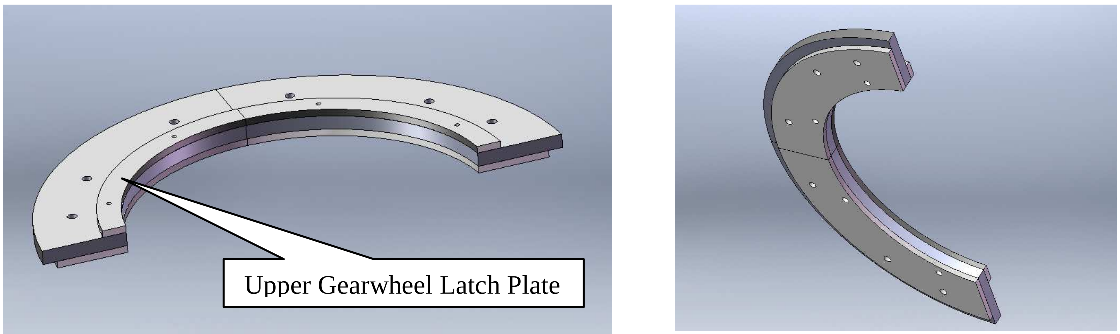
$$Z_1 = \frac{d_1}{m} = \frac{1120}{4} = 280 \text{ tooth}$$

Small Gearwheel:

$$Z_2 = \frac{d_2}{m} = \frac{80}{4} = 20 \text{ tooth}$$

Since the umbilical runs through the centre of the big Gearwheel, as shown in figure 12, the big Gearwheel will have to be split into two equal parts and latched together with Latch Plates on the top and on the bottom of the gearwheel. The Latch Plates will connect the two gearwheel parts together with the help of 12 bolts of size M20 x 90 mm. The Latch Plate will also perform as the sliding surface in the POM Slide bearing. Both the big Gearwheel and the Latch Plate material will be stainless 316 steel. The friction between steel and the POM Slide bearing is μ = 0.32. [4]

As shown in figure 16, the ROV Interface is connected to the lower Latch Plate. During launch and recovery through moon pool, as shown in figure 20 A, the ROV interface is in direct contact with the ROV cage.



A) *Cross section view of Latch Plate on top of Gearwheel*

B) *Cross section view of Latch Plate on bottom of Gearwheel*

Figure 15 *A and B show upper and lower Latch Plate of the Gearwheel assembly*

As stated above (see also figure 19 A and figure 20 A) the form and the function of the umbilical necessitates that the big Gearwheel is split into two equal parts.

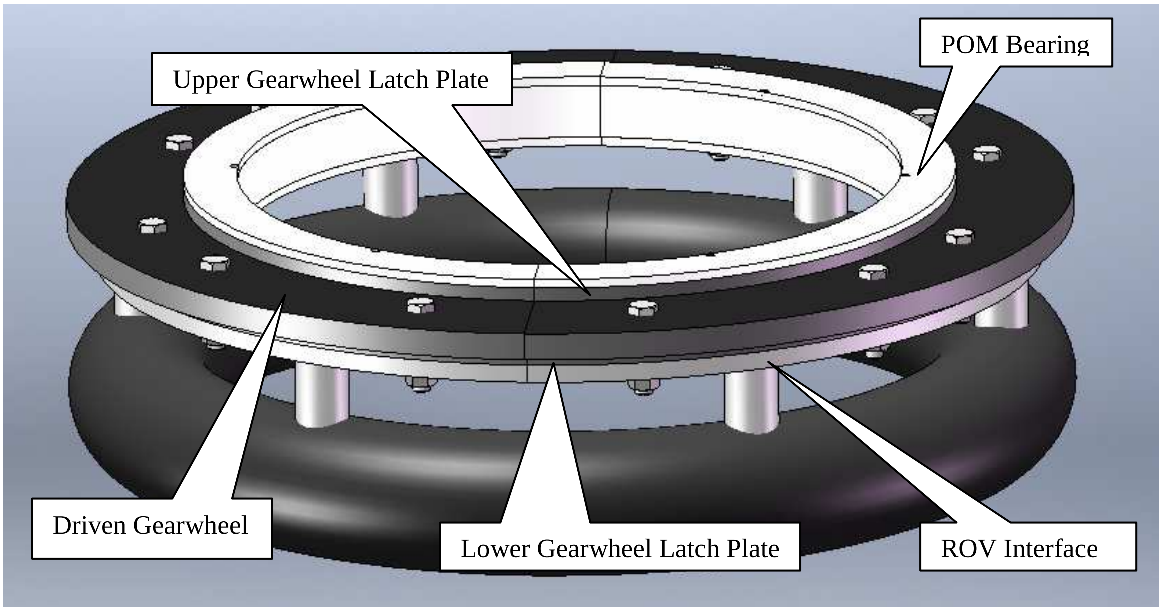


Figure 16 *POM slide bearing assembly*

8.1.3 Hydraulic motor and Gearbox

There is a great assortment of hydraulic motors and gearboxes on the market. In this design study a lot of different brands and sizes would work perfectly fine. However there will always be some preferences with one over the other. For this design the selection of hydraulic motor and gearbox was done in conjunction with the engineers at Oceaneering AS.
Oceaneering AS has great experience with the use of Sauer Danfoss OMR hydraulic motors. They use the brand Sauer Danfoss in many of their tool designs. The criteria for my selections are as follows:

The hydraulic motor component will be powering the rotation of the Rotary Table. The essential torque, speed and acceleration required will be achieved by combining the use of a large driven Gearwheel (d =1120mm), Ondrives PGE1001-10:1 ratio Gearbox, hydraulic fluid pressure, hydraulic fluid flow and the proper size of the Sauer Danfoss OMR hydraulic motor.

During a constant pressure gradient, it is common knowledge that a larger hydraulic motor will produce more torque than a small hydraulic motor. This is due to the larger area of the impellers/blades which produces the force and herby the torque. Selecting the right motor size and pressure will be crucial to the design.

The acceleration of the Hydraulic Rotary Table System is quite critical. If the acceleration of the System is large, torque will also be large (see chapter 9.1.1). Large acceleration will cause torque to build up and the torque may exceed the maximum allowed design torque. As a result of this “torque build up” components might brake down.

To assure a controlled acceleration and less chance of components braking down, a flow control valve will be used. Other, more basic, hydraulic components will be standard 3/8” hydraulic tubes, valves, couplings and an adjustable check valve. The flow control valve will make sure that the angular acceleration of the hydraulic motor, and hereby the ROV, is restrained and does not exceed 0.2 rad/s^2. This will ensure a smooth rotational acceleration of the ROV and less chance of exceeding the maximum allowed design torque. The check valve will make sure that the pressure is constant and that it does not exceed the set level.

Another important part in designing the speed of the system is to choose correct size of the hydraulic pump. It is the pump that is producing the desired flow and pressure. However it is common on oilrigs that huge hydraulic power units (HPU`s) are supplying smaller hydraulic units and equipment with both flow and pressure. Even thought there are no standardized flow rate, topside hydraulic power units (HPU`s) might vary from anywhere between 30 l/min to 120 l/min. However a flow rate of 60 l/min is often utilized on topside equipment. To control the flow rate further more, adjustable nozzle valves are used to decrease or increase the hydraulic flow.

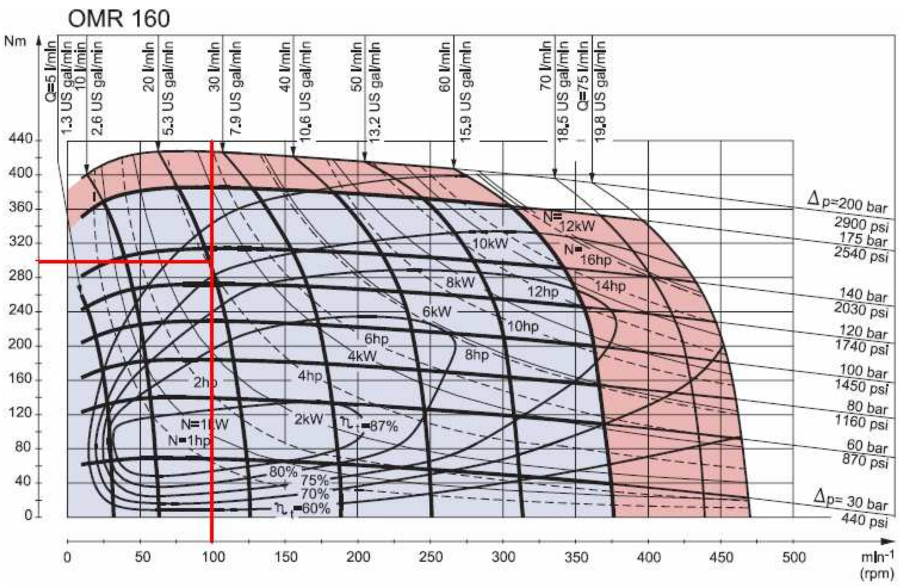


Figure 17 *Function diagram for Sauer Danfoss OMR160 hydraulic motor*

To be able to decide which hydraulic motor to use in this system, the torque, angular acceleration and the angular speed/rotational speed of the Hydraulic Rotary Table is taken into consideration. The Sauer Danfoss OMR160 hydraulic motor function diagram is used to select the desired flow and pressure. From figure 17 we see that the pressure decides the torque and the flow decides the number of revolutions per minute (rpm).

If we set the check valve to 130 bar pressure and choke the flow rate to 28 l/min we can estimate both the rpm and torque from the OMR 160 function diagram [Fig 17].

When using the OMR 160 hydraulic motor together with the Ondrives PGE 1001-10:1 ratio Gearbox we can further study, from the OMR160 function diagram [Fig 17], the actual torque and rotational speed (rpm) applied to the driven Gearwheel. Since the Gearbox ratio is 10:1 we estimate that the output torque will be 10 times larger than the input torque, and that the output rotational speed will be 10 times less than the input rotational speed.

The maximum speed of the system (speed of the rotation) is not essential to the design. It dose not make a difference if it takes the operator 30-60 or even 90 seconds to fully rotate the ROV to the desired position. This is because the total duration of the launch operation usually takes substantially more time than the time that is spent rotating the ROV. For that reason, the maximum angular speed of the rotation is not a main design factor.

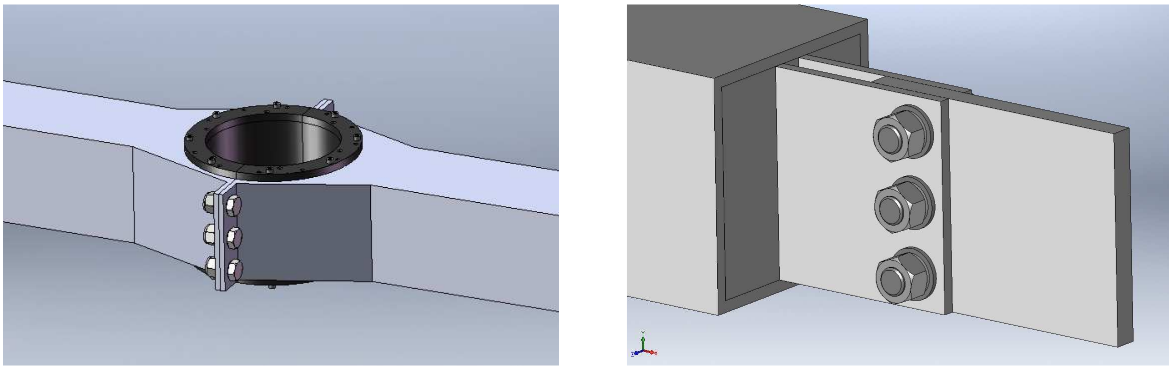
On the basis of these assumptions it will be preferred to have a slow angular rotation of the ROV. The moderate angular speed of the rotation will reduce the torque that is exerted to the system when accelerating or retarding the rotation of the system. The whole process of rotating the ROV will be easy to monitor and control.

The retarding of the rotation will be done by controlling the flow. In this case the flow will be gradually choked and the rotary table will eventually come to a stand still. When accelerating the Rotary Table, the flow will gradually increase so that the Rotary Table gets a smooth acceleration.

8.2 Guide Beam

As shown in figure 18, the Guide Beam consists of two equal parts that are bolted together with six M24 x 55mm Din 933 bolts. The base structure of the Guide Beam is 200mm wide and 200mm tall. The thickness of the beam is 10mm. The main purpose of the Guide Beam is to support and guide the Hydraulic Rotary Table. When the ROV is guided through and above the moon pool, it is actually the umbilical that holds the load of the Guide Beam and its integrated Hydraulic Rotary Table. However when the ROV cage is being launched, and the ROV cage leaves the moon pool, the Guide Beam is left on the lowest level of the moon pool area [Fig 19 A]. The ROV is lowered into the sea while the Guide Beam sits in the moon pool. In this case, when the ROV leaves the Guide Beam, the Guide Beam is supporting the total load of itself and the Hydraulic Rotary Table.

Another important part of the Guide Beam is to guide and restrain the umbilical. Since the Guide Beam is enclosing the umbilical (vertically the umbilical runs free), it will control the horizontal movement of the umbilical. The umbilical will be forced to run through the centre of the Guide Beam and hereby the moon pool. Due to the danger of damaging the steel umbilical in case of contact between the steel Guide Beam and the umbilical, a POM Ring is installed in the centre of the Guide Beam. The POM Ring will sit inside the Guide Beam almost like a bucket, shown in figure 18 A.



A) *Guide Beam with POM ring* **B)** *20mm Guide Plate*

Figure 18 *Guide Beam details*

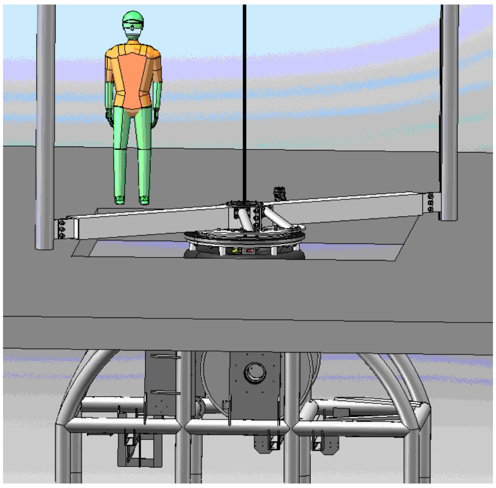
The POM Ring, shown in figure 18 A, will decrease the chance of damage to the umbilical.

The Guide Beam may be designed to have various length and sizes so that it fits the desired moon pool measures. In some cases, the area of the moon pool opening measures only 3 x 2 meters. [1], [16]

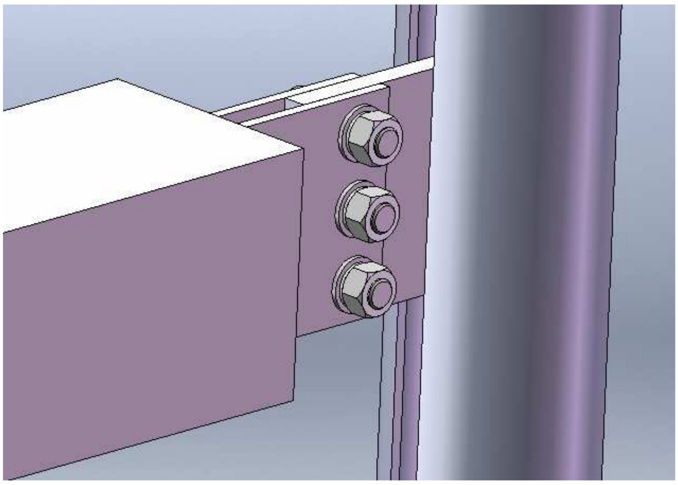
The Guide Structure is placed diagonal over the moon pool. On each side of the Guide Beam, two Guide Plates will be installed. The Guide Plates will attach the Guide Beam to the Guide Structure (6 inch pipes with guide path), as shown in figure 19 B. This will ensure no horizontal movement of the Guide Beam. However, the umbilical and ROV cage will easily be able to lift the Guide Beam and Hydraulic Rotary Table when the ROV is recovered through the moon pool opening, as shown in figure 19 A.

8.3 Guide Structure

The Guide Structure will consist of two 6 inch pipes made of stainless 316 steel. A 30 mm guide path will be machined out in the longitudinal direction of both pipes. The pipes will be placed vertically and diagonal over the moon pool opening, shown in figure 19 A. Both guide paths will be facing each other. The Guide Beam will then be installed so that the Guide Plates are inside the Guide Pipes, shown in figure 19 B. This makes the guide Beam horizontally restrained and only able to move up or down the Guide Pipe Structure. The Guide Pipes will have to be bolted or welded to the base of the moon pool. The Guide Plates are bolted to both ends of the Guide Beam.



A) *Guide Beam and Guide Structure*

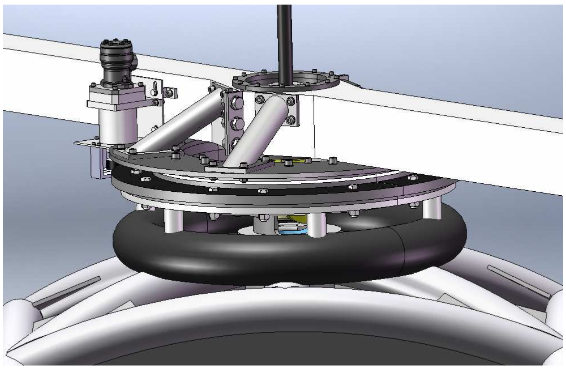


B) *20mm Guide Plate slides inside the Guide Pipe Structure*

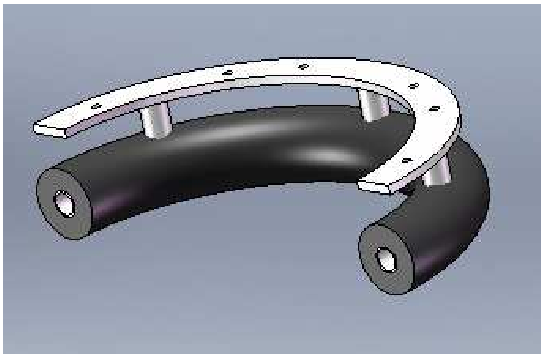
Figure 19 *Guide Structure details*

8.4 Interface between ROV cage and Rotary Table

When the ROV is being recovered from the sea, it will eventually reach the moon pool area where it will come in contact with the Hydraulic Rotary Table, integrated on the Guide Beam. The ROV cage will then lift the Guide Beam up. To be able to rotate the ROV, some kind of connection will have to be made between the ROV cage and the Hydraulic Rotary Table. Since the umbilical is pulling the ROV cage, as shown in figure 20 A, forcing the ROV cage to lift the Guide Beam, we can assume that the total weight of the Guide Beam, including the Hydraulic Rotary Table will be acting on the ROV cage.



A) *ROV Interface in contact with ROV cage*



B) *ROV interface coated with 50mm rubber coating*

Figure 20 *ROV Interface details*

The weight of the Guide Beam and the integrated Hydraulic Rotary Table is approximately 930kg (not including hydraulic hoses, valves and other hydraulic equipment).

The ROV Interface, which is part of the Hydraulic Rotary Table, and in direct contact with the ROV cage, has a 50 mm thick rubber coating. The rubber coating is fairly soft and generates a lot of friction between the two parts.

When in contact with the ROV cage, the rubber coating will squeeze together and partly circumference the pipe structure of the ROV cage as shown in figure 20 A. This will help create enough friction between the two surfaces so that the ROV cage can slowly rotate as one with the Rotary Table.

When the rubber coating is squeezed together so that part of the ROV cage pipe configuration is covered by the rubber coating, it is assumed that the static friction coefficient (μ) is 0.6. This is just an assumption. In case more friction is required, the ROV cage can easily be modified so that the connection point between the two surfaces has a static friction coefficient (μ) of 0.6 or similar. The radius of the intersection point between the ROV cage and the ROV Interface is 0.480 m.

Calculating the friction force F_F

We know from the torque calculations that the rotary momentum of the ROV alone is 2312Nm (se chapter 9.1.1). This tells us that the momentum caused by the friction force must be larger than the momentum trying to rotate the ROV cage. If not the Hydraulic Rotary Table will start to spin and the ROV cage will not rotate.

The friction force (F_F) is directly proportional to the normal force (F) where the proportionally constant is the coefficient of friction (μ). This relation is given by: $F_F = F \cdot \mu$ where $F_F = 930kg \cdot 9.81m/s^2$ and $\mu = 0.6$. Substituting values we get $F_F = 5.47$ kN.

$$F_F = F \cdot \mu$$

$$F_F = 930Kg \cdot 9.81m/s^2 \cdot 0.6 = 5474N$$

This friction force between the ROV Interface rubber and the stainless steel pipe configuration of the ROV cage causes a momentum (MF_F) given by: $MF_F = F_F \cdot R$, where R = 0.480m. Substituting values, we get $MF_F = 2628$ Nm, which is larger than the rotary momentum of the ROV alone ($\tau_1 = 2312$ Nm).

This means that there is enough friction ($\mu = 0.6$) between the two surfaces so that the ROV cage will rotate together with the ROV interface and the Hydraulic Rotary Table.

Further more we can calculate the minimum friction coefficient (μ_{min}) needed between the two surfaces to be able to rotate the ROV, as follows:

$$\mu_{\min} = \frac{F_F}{F} = \frac{\tau_1}{R} \cdot \frac{1}{F}$$

$$\mu_{\min} = \frac{2312Nm}{0.48m} \cdot \frac{1}{930kg \cdot 9.81m/s^2} = 0.52$$

Minimum friction coefficient (μ_{min}) between the two surfaces must be at least 0.52.

9. MAIN CALCULATIONS

Basic calculations have been performed to estimate the main loads and stresses in the ROV launch and recovery design.

9.1 Torque calculations

To be able to pull or lower the ROV through the moon pool, the ROV has to be rotated so that it is correctly aligned to the moon pool opening. The rotary motion of the ROV will be done by a Hydraulic Rotary Table. The rotation will require a certain torque transmitted from the rotary table to the 4 ton ROV. The total torque that needs to be overcome will be decided by four main components. The four components are as follows:

1. **The ROV moment of inertia (4 ton mass, ROV radius = 1.7meters)**
2. **The applied acceleration α (The angular rotation of the ROV)**
3. **The resistance caused by axial friction in the slide bearing**
4. **The resistance caused by radial friction in the slide bearing**

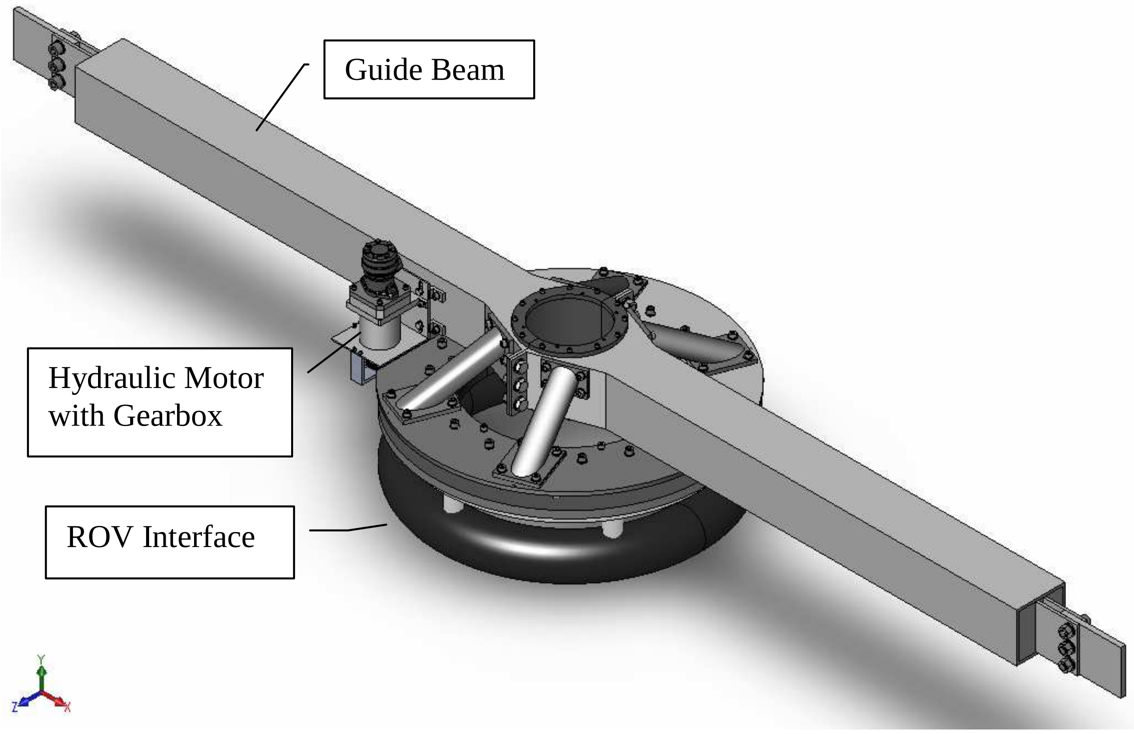


Figure 21 *Hydraulic Rotary Table System*

9.1.1 Rotary momentum of the ROV

Maximum torque will occur when accelerating the hydraulic rotary table from a stand still position. If we combine point 1 and 2 in chapter 9.1, and assume a maximum angular acceleration $\alpha = 0.2 \text{ rad/s}^2 \approx 11.45°/\text{s}^2$, the maximum torque generated during the acceleration phase (rotation), can be estimated as follows:

I = Mass Moment of Inertia

m = 4000kg (The mass of cage and ROV together)

τ = Torque

α = Maximum Angular Acceleration (assumed to be $0.2\text{rad/s}^2 \approx 11.45°/\text{s}^2$)

r = 1.7meters (radius of the ROV cage)

$$I = m \cdot r^2 \tag{1}$$

$$I = 4000\text{kg} \cdot (1.7\text{m})^2 = 11560 \text{ kgm}^2$$

$$\tau_1 = I \cdot \alpha \tag{2}$$

$$\tau_1 = 11560\text{kgm}^2 \cdot 0.2\,{}^{\text{rad}}/{}_{\text{s}}{}^2 = 2312 \text{ Nm}$$

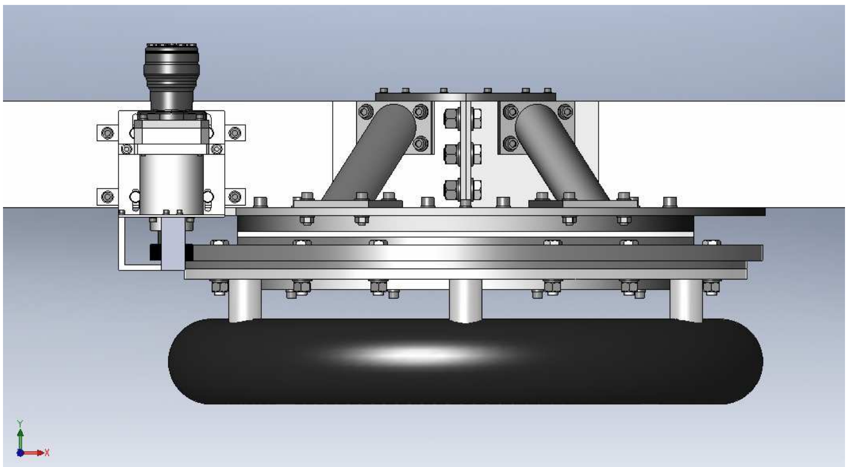


Figure 22 *Hydraulic Rotary Table details*

To be able to rotate the Rotary Table [Fig 22], torque is transferred from the Hydraulic Motor and the Gearbox to the Gearwheel.

9.1.2 Friction between the Latch Plates and the Slide bearing

Upper and lower steel Latch Plates are used to connect the big Gearwheel around and inside the Slide bearing. Due to friction between the upper and lower Latch Plates and the Slide bearing, which is made of POM material, extra resistance is created in the bearing. This resistance will have to be considered when calculating the total torque required to rotate the ROV.

Together with the selection of material used for the Slide bearing and the upper and lower Latch Plates (sliding inside the Slide bearing), the "hang off load" (F_Z), shown as colored green in figure 23, is part of what creates the friction force (F_R) and the resistance in the slide bearing. Large "hang off load" (F_Z) will generate a larger friction force (F_R) and herby more resistance in the slide bearing. The "Hang off load" (F_Z) in the Rotary Table design consists of several parts. The total weight of the "hang off load" (F_Z) is 269kg.

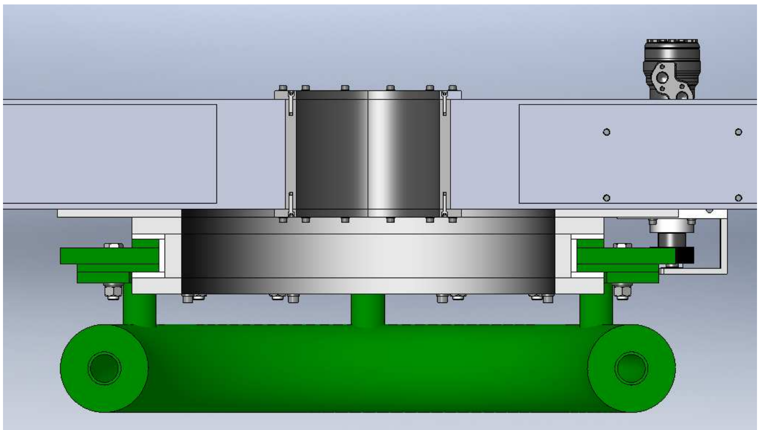


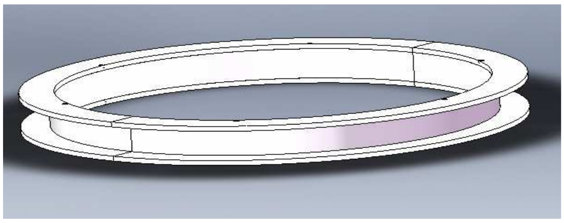
Figure 23 *Hydraulic Rotary Table cross section*

Latch plate material:	Stainless 316 Steel
Slide bearing material:	POM
Friction coefficient between Steel and POM:	$\mu = 0.32$
Hang off load, (F_Z):	$269kg \cdot 9.81m/s2 \approx 2640N$

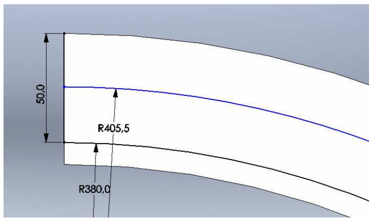
$$F_R = F_Z \cdot \mu$$

$$F_R = 2640N \cdot 0.32 = 845N$$

Since F_R is a force created due to friction between the “hang off load” (F_Z), and the bottom Slide bearing surface, we know that (F_R) together with the arm R [Fig 24 B], will create a torque that counteracts the rotation of the ROV/Rotary Table. To calculate this torque, we must first decide the arm, R, which the force F_R is acting on. This will be the arm R that divides the slide bearing face into two equal areas [Fig 24 B].



A) *POM Slide bearing*



B) Slide plane of the slide bearing

Figure 24 *POM Slide bearing details*

To calculate R we use the formula

$$R = \frac{2}{3}\left[\left(\frac{Ro^3 - Ri^3}{Ro^2 - Ri^2}\right)\right] \tag{3}$$

Where R_O is the outer radius and R_i is the inner radius.
Substituting numerical values we get:

$$R = \frac{2}{3}\left[\left(\frac{430^3 - 380^3}{430^2 - 380^2}\right)\right] = 405.5\text{mm}$$

The friction force F_R is acting on the bottom exposed plane face of the slide bearing [Fig 24 A and B]. The arm, which creates the counter torque, is calculated from equation nr. 3. The length of the arm is 405.5mm as shown in figure 24 B. This is a counter clockwise torque that is slowing down and working against the rotation of the rotary table/ROV. The magnitude of the torque can easily be calculated:

$$\tau_2 = F_R \cdot R \tag{4}$$

$$\tau_2 = 845N \cdot 0.4055m = 343 \text{ Nm}$$

9.1.3 Calculating the force applied to the driven gearwheel

In addition to the friction force F_R, which is creating a torque that is acting in the plane face of the Slide bearing, there will also be a radial friction force in the bearing. The radial force can be estimated by decomposing and calculating the force acting on the pitch circle, as shown in figure 25. To perform such calculations we first need to calculate the force F_Y. F_Y and F_X are components of the driven force F, as shown in figure 25, which is transferred from the small Gearwheel to the big Gearwheel (driven gearwheel).

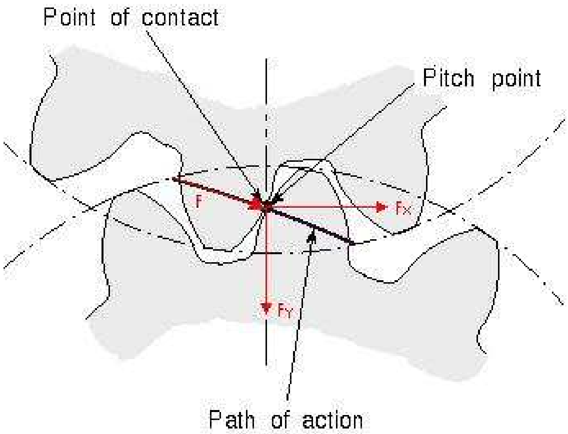


Figure 25 *Force is transferred from small gearwheel to big gearwheel*

On the pitch circle the tangential force (F_X) and the radial force (F_Y) is transferred from one gearwheel to the other. The pitch point is the contact point between the two gearwheels. The path of action or also known as the angle of attack is at a 20-degree angle compared to the tangential force (F_X).

Calculations

$$F = \frac{\tau_1 + \tau_2}{R_{PitchCirle}} \qquad (5)$$

$\tau_1 = 2312\text{Nm}$

$\tau_2 = 343\text{Nm}$

$R_{PitchCircle} = 0.560\text{m}$

$$F_X = \frac{2312Nm + 343Nm}{0.560m} = 4741\text{N}$$

The force (F_X) is tangential to the gearwheel pitch circle. We use the force component (F_X) together with the resultant force (F) to find and estimate the radial force component (F_Y). Since the resultant force (F) has a 20° angle of attack or path of action compared to the gearwheel pitch circle, we use this to calculate the radial force (F_Y). When decomposing the resultant force (F) into the two components (F_X) and (F_Y), using formula 6, we can easily calculate the radial force (F_Y) exerted to the Slide bearing:

$$\text{Tan}(20°) = \frac{F_y}{F_x} \tag{6}$$

$$F_Y = \text{Tan}(20°) \cdot F_X$$

$$F_Y = \text{Tan}(20°) \cdot 4741\text{N} = 1726\text{N}$$

9.1.4 Torque caused by radial friction in slide bearing

The force (F_Y), is in this case, a radial force causing friction between the steel Latch Plate/Gearwheel combination and the POM bearing. The friction force will generate even more torque counteracting the rotation of the rotary table. This will be the third and last torque that the system will have to overcome. The torque τ3 can be estimated by basic calculations.

$$\tau_3 = F_y \cdot \mu \cdot R_{\text{(OUTER SLIDE BEARING FLANGE)}} \tag{7}$$

$$\tau_3 = 1726\text{ N} \cdot 0.32 \cdot 0.380\text{m} = 210\text{ Nm}$$

9.1.5 Torque calculation conclusion

If we combine all three momentums, τ1, τ2 and τ3 we can estimate the total torque needed to rotate the ROV as follows:

Total torque needed to rotate the ROV

$$\tau_{\text{Total}} = \tau_1 + \tau_2 + \tau_3 \tag{8}$$

τ_{Total} = 2312Nm + 343Nm + 210 Nm = 2865 Nm

This tells us that the Hydraulic Motor, the Gearbox, the small and big Gearwheel combined together as one unit will have to produce a total torque of 2865 Nm to be able to accelerate and rotate the ROV during launch and recovery.

9.2 Calculating main bolts in the ROV launch system

The Hydraulic Rotary Table consists of several parts that are primarily bolted together. The main bolts will have to be calculated to be sure that the Rotary Table is designed for the proper loads. The main bolts are shown in figure 26.

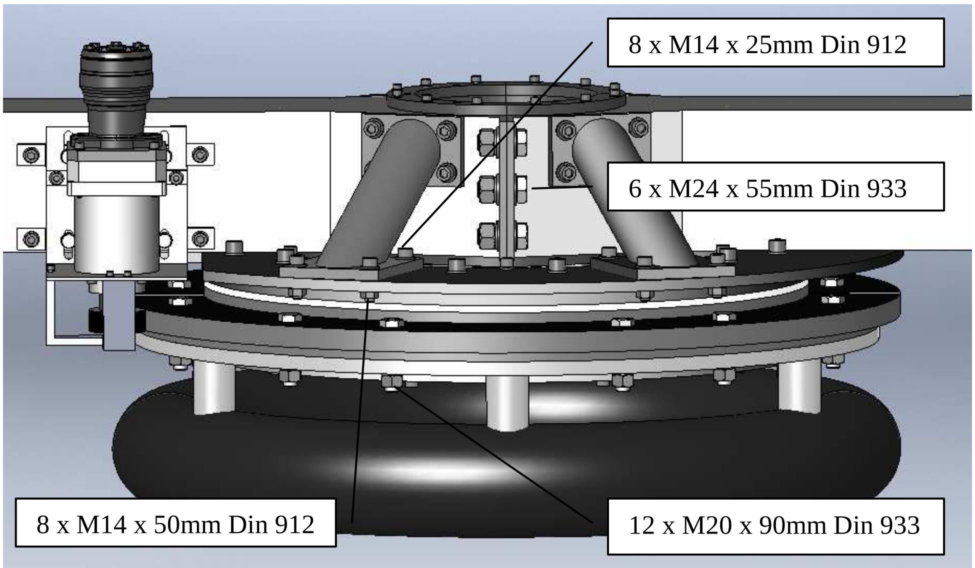
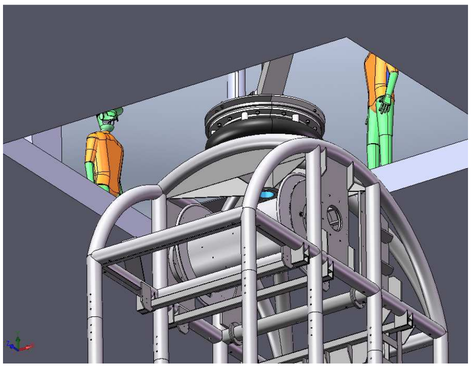
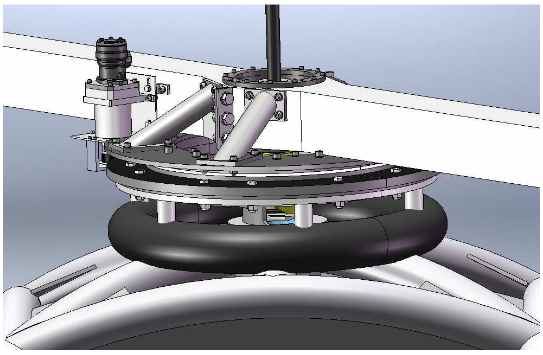


Figure 26 *Main bolts of the design*

9.2.1 M24 x 55 mm din 933 bolts

Both ends of the Guide Beam are horizontally fixed. Vertically the Guide Beam is free to move up and down inside the Guide Pipes. This tells us that when there is contact between the ROV Interface and the ROV cage, the Guide Beam will move along with, and at the same speed as the ROV cage. The Guide Beams vertical speed is therefore the same as the winch speed [Fig 27 A and B].





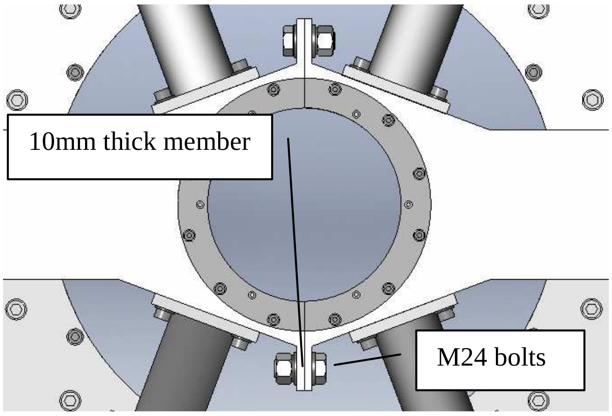
A) *ROV runs through moon pool*

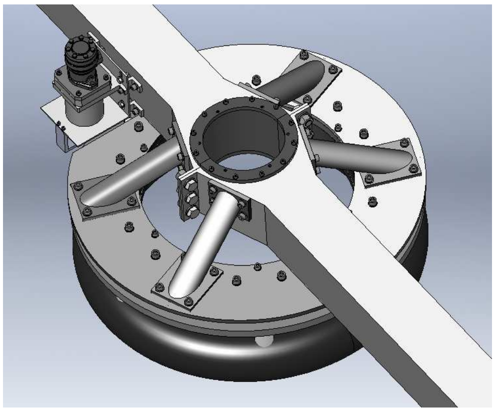
B) *ROV connects with ROV interface*

Figure 27 *A and B show details during launch and recovery*

The two Guide Beam parts are bolted together with the help of six M24 x 55 mm Din 933 bolts [Fig 28 A and B]. Under normal circumstances the connection between the ROV cage and the ROV Interface will be done at a very low winch speed. Due to this, there will be barley any force trying to split the two 170 kg Guide Beam parts from each other.

However if the ROV cage hits the ROV Interface at full winch speed, the Guide Beam will try to separate, and the bolts latching the two Guide Beam parts together will endure large stress.





A) *Guide Beam seen from above*

B) *View of Guide Beam with bolts*

Figure 28 *Guide Beam details*

I assume that the hook up between the ROV Interface and the Guide Beam will always be done at a very low speed. When calculating the M24 bolts it is therefore assumed that there is no extern forces acting on the bolts. The bolts only clamp the two Guide Beam members together. Further more I assume that the total tightening torque of the M24 bolts are 829.92 Nm [11], and that the flange (member) of the Guide Beam is 10 mm thick [Fig 28 A].

To calculate the bolt stress, we have to first find out the portion that is taken up by a given bolt. The bolt load depends on the level of bolt preload and the relative stiffness of the bolt with respect to the members that the bolt connects. In this case we have six M24 bolts clamping together two 10 mm thick steel members [Fig 28 A and B].

Since we assume that the bolts are only preloaded and that there is no external force acting on the bolts, we can simplify the bolt calculations for the M24 bolts by only calculating the tension caused by preloading of the bolts. The preloading of the M24 bolts is set to be 829.92 Nm [11].

Fastening moment of the M24 bolts

The M24 bolts are assumed to be threaded only where the M24 nut sits. The bolts are fastened with a total tightening torque of 829.92Nm [11].

The core diameter (ds) of the bolt can be estimated using formula nr 15 below:

$$ds = \sqrt{\frac{4 \cdot As}{\pi}} \tag{15}$$

$$ds = \sqrt{\frac{4 \cdot 352.5mm^2}{\pi}} = 21.18mm$$

The force F_0, which is acting in the bolt, due to the preload is calculated from formula nr 16.

$$F_0 = \frac{M_T}{0.2 \cdot ds} \tag{16}$$

M_T = Total tightening torque = 829.92 Nm

$$F_0 = \frac{829.92Nm}{0.2 \cdot 0.2118m} = 19592N$$

The total force acting in the bolt (Fb) can be calculated from formula nr 17.

$$Fb = F_0 + \phi \cdot F \tag{17}$$

Since it is assumed that it is only the preloading of the bolts that generates the stress in the M24 bolts, we can study from formula nr 17 that F_0 = Fb (F is ignored and assumed to be zero).

It could be discussed if this is correct or not. Most likely the net weight of the Guide Beam and Rotary Table will also generate stress in the bolts. However, in this case only the tightening torque of the M24 bolts that has been considered. All other factors have been ignored. From assumptions stated above, we can estimate Fb:

$$Fb = 19592N + \phi \cdot 0 = 19592N$$

The force Fb is distributed equally to all six of the M24 bolts. The stress in each bolt is estimated from formula nr 18:

$$\sigma b = \frac{Fb}{As} \tag{18}$$

$$\sigma b = \frac{19592N}{6 \cdot 352.5mm^2} = 9.3N/mm^2$$

Since F_0 = Fb, $\sigma_0 = \sigma b = 9.3$ N/mm^2

Due to the preloading, torsion (τ_0) will be transferred to each bolt. The torsion (τ_0) can be estimated from formula nr 22:

$$\tau_0 = \frac{9.6 \cdot M_T}{\pi \cdot d_S^{\,3}} \tag{22}$$

$$\tau_0 = \frac{9.6 \cdot 829.29Nm}{\pi \cdot 21.18^3} = 0.26N/mm^2$$

As we can se from formula nr 22, the shear stresses caused by the preloading is very small. The equivalent stresses are calculated using the Von Mises criteria and formula nr 20:

$$\sigma_{eq} = \sqrt{\sigma b^2 + 3((0.5 \cdot \sigma_0) + \tau_v)^2} \tag{20}$$

$$\sigma_{eq} = \sqrt{9.3^2 + 3(0.5 \cdot 9.3 + 0.26)^2} = 12.6 N / mm^2$$

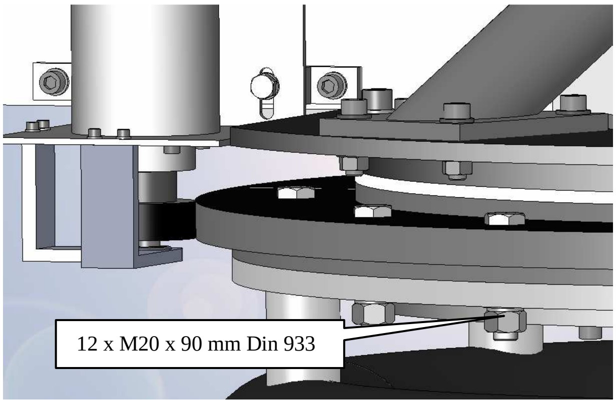
The total stress acting in each of the six bolts is 12.6 N/mm^2. The bolts that will be used are of 8.8 stainless steel qualities. The tensile stress limit for the 8.8 bolts is 640N/mm^2.
This meaning that the bolts can theoretically be loaded with 640 N/mm^2 before they start to deform.

Conclusion

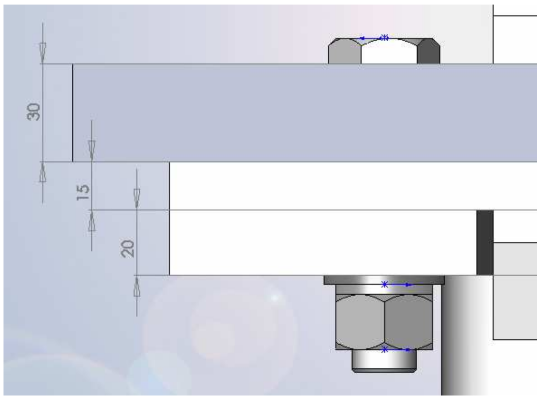
The stress in the M24 bolts is only 12.6 N/mm^2. This tells us that the bolts connecting the two Guide Beam parts are extremely over dimensioned. However, in this case and especially since the total force acting on the Guide Beam latch is uncertain and partly unknown, it would be safest to scale the bolts so that there will be no chance of failure. Since the launch system will be used in an offshore environment, all bolts will have to be of corrosion resistant quality (stainless steel).

9.2.2 M20 x 90 mm din 933 bolts

The driven Gearwheel is connected together with the ROV Interface with the help of twelve M20 x 90 mm Din 933 bolts. The bolts are assumed to have normal 8.8 qualities and are made out of stainless steel. The bolts are somewhat evenly spread around the Gearwheel [Fig 29 A]. As shown in figure 29 A and B, their main purpose is to connect the Gearwheel, the lower Latch Plate and the ROV Interface. This allows the ROV Interface to rotate as one with the Gearwheel. Since torque is transferred from the Gearwheel to the ROV Interface, it is expected some shear stress in the bolts. The bolts will also have axial force due to the weight of the ROV Interface. When calculating the 20 mm bolts a fastening torque of 533.7 Nm is assumed [11].



A) *Clamp details*



B) *Cross section view of clamp area*

Figure 29 *Gearwheel, lower Latch Plate and the ROV Interface clamped together*

As seen in chapter 9.2.1, the shear force in the bolts, due to the preloading of the bolts, is very small. In the remaining bolt calculations the shear force due to the preloading has for that reason been neglected.

Formulas used for calculating the total stress in the M20 bolts:

dh = diameter of the bolt hole

The nut of the clamped area is estimated to cover twice the diameter of the bolt hole. This is shown in formula nr 9. The diameter of the clamped area can then be estimated as follows:

$$D = 2 \cdot dh \qquad (9)$$

$$D = 2 \cdot 20mm = 40mm$$

The area of the clamped material (Am) can then be calculated from formula nr 10.

$$Am = \left(D^2 - dh^2\right)\frac{\pi}{4} \qquad (10)$$

$$Am = \left(40^2 - 20^2\right)\frac{\pi}{4} = 942.5mm^2$$

The stiffness of the clamped area (Km) can be estimated when using formula nr 11:

$$Km = \frac{Am \cdot E}{L} \tag{11}$$

The stiffness of the bolt (Ks) can be estimated when using formula nr 12:

$$Ks = \frac{A \cdot E}{L} \tag{12}$$

$As =$ Tensile Stress Area of M20 bolt = 244,79mm^2

Calculating the stiffness of the clamped connection

Part	Clamp Length	Material	E-Module	Km	Ks
Gearwheel	30mm	316 steel	193000 N/mm^2	Km1 6063417 N/mm	
Lower Latch Plate	15mm	316 steel	193000 N/mm^2	Km2 12126833 N/mm	
ROV Interface	20mm	Al 6082 T6	70000 N/mm^2	Km3 3298750 N/mm	
M20 Bolt 8.8 Quality	65mm	316 steel	193000 N/mm^2		932932,3 N/mm

When clamping several parts together, the total stiffness of the clamped area can be estimated from formula nr 13.

$$\frac{1}{Km} = \frac{1}{Km_1} + \frac{1}{Km_2} + \ldots\ldots\frac{1}{Km_n} \tag{13}$$

$$Km = \frac{1}{\left(\frac{1}{6063417} + \frac{1}{12126833} + \frac{1}{3298750}\right)} = 1816430.1 N/mm$$

Formula nr 14 is used to estimate how much of the force (F) that is transferred to the bolts, and how much of the force (F) which is used to relieve the tension in the clamped area.

$$\phi = \frac{Ks}{Ks + Km} \qquad (14)$$

$$\phi = \frac{932932.3}{932932.3 + 1816430.1} = 0.34$$

This meaning that 34% of the force (F) is absorbed by the bolts and that 66% of the force (F) is used to relieve the clamped area. F is in this case the weight of the ROV Interface (24.1 kg).

Fastening moment of the M20 bolts

The M20 bolts are assumed to be threaded only where the M20 nut sits. The bolt is fastened with a total tightening torque of 533.67 Nm [11].

The core diameter of the bolt can be estimated using formula nr 15 below:

$$ds = \sqrt{\frac{4 \cdot As}{\pi}} \qquad (15)$$

$$ds = \frac{\sqrt{4 \cdot 244.79}}{\pi} = 17.65mm$$

M_T = Total Tightening Torque

The M20 Din 933 bolts are fastened with a total tightening torque of 533.67 Nm. [11]

The force F_0, which is acting in the bolt, is calculated from formula nr 16.

$$F_0 = \frac{M_T}{0.2 \cdot ds} \qquad (16)$$

$$F_0 = \frac{533.67Nm}{0.2 \cdot 0.01765m} = 151181.3N$$

F is the load trying to stretch the bolts causing axial stress in the bolts. In this case F is the weight of the ROV Interface. The ROV interface is made of 6082 T6 Aluminum and has a total weight of 24.1 kg.

$$F = 24.1kg \cdot 9.81m/s^2 = 236.4N$$

$$Fb = F_0 + \phi \cdot F \tag{17}$$

$$Fb = 151181.3N + 0.34 \cdot 236.4N = 151261.7N$$

The force Fb is distributed equally to all twelve M20 bolts. The stress in each bolt is calculated from the formula nr 18 and 19.

$$\sigma b = \frac{Fb}{As} \tag{18}$$

$$\sigma b = \frac{151261.7N}{12 \cdot 244.79mm^2} = 51.5N/mm^2$$

$$\sigma_0 = \frac{F_0}{As} \tag{19}$$

$$\sigma_0 = \frac{151181.3N}{12 \cdot 244.79mm^2} = 51.5N/mm^2$$

Since the Gearwheel is transmitting torque to the ROV Interface, it is expected shear stress in the M20 bolts. The torque that is transferred is calculated in chapter 9.1.5. The total torque that is transferred is 2865 Nm and the twelve bolts sits at a radius R = 0.480m.
We can then estimate that the shear force exerted to the twelve bolts will be as follows:

$$F_S = \frac{2865Nm}{0.480m} = 5969N$$

The shear force F_S is acting tangential to the bolt path. The force is equally distributed to all twelve bolts. Formula nr 21 is used when calculating the shear stress in the bolts due to the torque transfer:

$$\tau_V = \frac{F_S}{A_{Bolts}} \qquad (21)$$

$$\tau_V = \frac{5969N}{12 \cdot \left(\frac{\pi \cdot 20^2}{4} \right) mm^2} = 1.58N / mm^2$$

Furthermore the equivalent stresses are calculated using the Von Mises criteria and formula nr 20.

$$\sigma_{eq} = \sqrt{\sigma b^2 + 3((0.5 \cdot \sigma_0) + \tau_v)^2} \qquad (20)$$

$$\sigma_{eq} = \sqrt{51.5^2 + 3((0.5 \cdot 51.5) + 1.58)^2} = 70N / mm^2$$

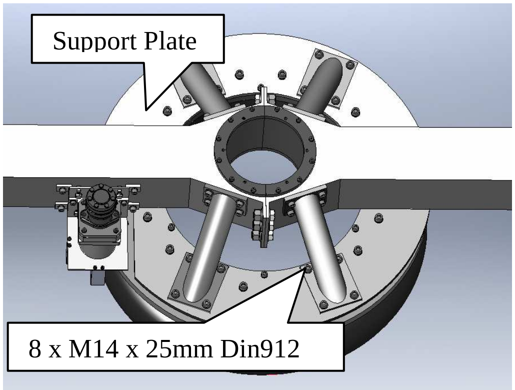
The bolts that are used are of 8.8 stainless steel qualities. The tensile stress limit for the bolts is 640N/mm^2.

Conclusion

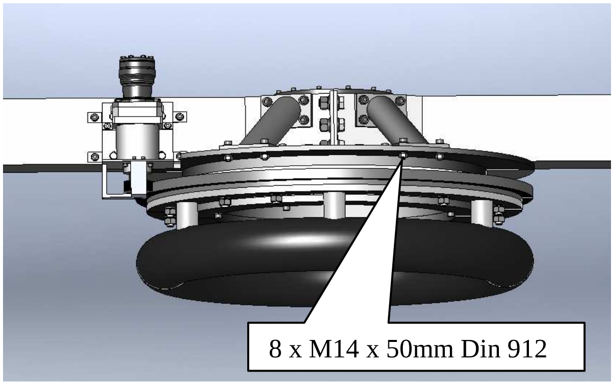
The stress in the M20 bolts is 70 N/mm^2. This tells us that the bolts connecting the Gearwheel, lower Latch Plate and the ROV Interface are over dimensioned. A smaller bolt diameter or a lower bolt property could be used. Since the launch system will be used in an offshore environment, all bolts will have to be of corrosion resistant quality.

9.2.3 M14 x 50 mm din 912 bolts

The Hydraulic Rotary Table is bolted to the Guide Beam with the help of sixteen M14 Din 912 bolts of various lengths. Eight of the sixteen bolts are running through the Support Arm [Fig 32], Support Plate [Fig 30 A], and are connected to M14 nuts. The remaining eight bolts are bolted through the Support Arm and directly to the Support Plate. The bolts used in the two fastening principals are of different lengths. The bolts that are directly connected to the support plate are 25 mm long. The bolts that run through the Support Plate and connect with the M14 nuts are 50 mm long. Washers are used with all bolts and nuts. The bolts are made of stainless steel and assumed to have 8.8 qualities. The main purposes of the bolts are to connect the Rotary Table to the Guide Beam and bear the load of the Rotary Table.



A) Top view



B) Front view

Figure 30 *Guide Beam and Hydraulic Rotary Table*

The M14 bolts, shown in figure 30 A and B, will have axial stress due to the total weight of the Rotary Table [Fig 31]. The Rotary Table weighs 499.74kg. When calculating the M14 bolts a tightening torque of 154.74 Nm is assumed. [11]

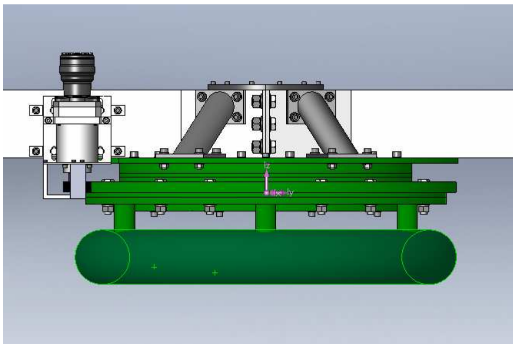


Figure 31 *Rotary Table hang off load*

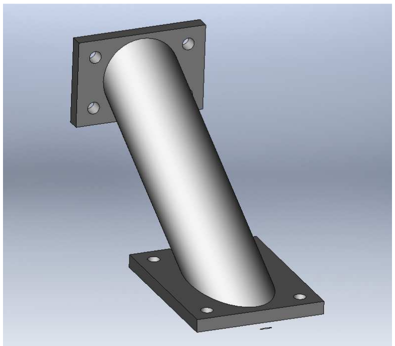


Figure 32 *Support Arm*

The M14 x 50 mm bolt is assumed to have threads only where the nut sits. The M14 x 25mm is assumed to have threads all along the bolt stem. As mentioned in chapter 9.2.2, the shear forces due to the preloading of the bolts are small, and have in these calculations been neglected.

Formulas used for calculating the total stress in the M14 Bolts

dh = diameter of the bolt hole

The nut of the clamped area is estimated to cover twice the diameter of the bolt hole. This is shown in formula nr 9. In this case the bolt hole has a diameter of 14 mm:

$$D = 2 \cdot dh \tag{9}$$

$$D = 2 \cdot 14mm = 28mm$$

The area of the clamped material (Am) can then be calculated from formula nr 10.

$$Am = \left(D^2 - dh^2\right)\frac{\pi}{4} \tag{10}$$

$$Am = \left(28^2 - 14^2\right)\frac{\pi}{4} = 461.8mm^2$$

The stiffness of the clamped area (Km) can be estimated when using formula nr 11:

$$Km = \frac{Am \cdot E}{L} \tag{11}$$

The stiffness of the bolt (Ks) can be estimated when using formula nr 12:

$$Ks = \frac{A \cdot E}{L} \tag{12}$$

$As =$ Tensile Stress Area of M14 bolt = 118mm^2

Calculating the stiffness of the clamped connection

Part	Clamp Length	Material	E-Module	Km	Ks
Support Plate	15mm	316 steel	193000 N/mm^2	Km1 5939253.3 N/mm	
Support Arm (3"Pipe)	15mm	316 steel	193000 N/mm^2	Km2 5939253.3 N/mm	
M14 8.8 Bolt without nut	15mm	316 steel	193000 N/mm^2		Ks1 1980180 N/mm
M14 8.8 Bolt with Nut.	30mm	316 steel	193000 N/mm^2		Ks2 989833 N/mm

When clamping several parts together, the total stiffness of the clamped connection can be estimated from formula nr 13.

$$\frac{1}{Km} = \frac{1}{Km_1} + \frac{1}{Km_2} + \ldots\ldots\frac{1}{Km_n} \tag{13}$$

$$Km = \frac{1}{\left(\frac{1}{5939253.3} + \frac{1}{5939253.3}\right)} = 2969626.7N/mm$$

If several different bolts are used, the resultant stiffness of all bolts can be estimated using formula nr 23.

$$\frac{1}{K_S} = \frac{1}{Ks_1} + \frac{1}{Ks_2} + \ldots\ldots\frac{1}{Ks_n} \tag{23}$$

$$K_S = \frac{1}{\left(\frac{1}{1980180} + \frac{1}{989833}\right)} = 659945.8N/mm$$

Formula nr 14 is used to estimate how much of the force (F) that is transferred to the bolts, and how much of the force (F) which is used to relieve the tension in the clamped area.

$$\phi = \frac{Ks}{Ks + Km} \tag{14}$$

$$\phi = \frac{659945.8}{659945.8 + 2969626.7} = 0.18$$

This meaning that 18% of the force (F) is absorbed by the bolts and that 82% of the force (F) is used to relieve the clamped area. The force (F) is in this case the weight of the Rotary Table (24.1 kg).

Fastening moment of the M14 bolts.

I assume the M14 Din 912 bolts have been fastened with a total tightening torque of 154.75Nm [11].

The core diameter of the bolt can be estimated using formula nr 15:

$$ds = \sqrt{\frac{4 \cdot As}{\pi}} \tag{15}$$

$$ds = \sqrt{\frac{4 \cdot 118}{\pi}} = 12.25mm$$

The force F_0, which is acting in the bolt, is calculated from formula nr 16.

$$F_0 = \frac{M_T}{0.2 \cdot ds} \tag{16}$$

M_T = Total tightening torque = 154.75

$$F_0 = \frac{154.75Nm}{0.2 \cdot 0.01225m} = 63159.2N$$

$$Fb = F_0 + \phi F \qquad (17)$$

The force F is trying to stretch the bolts. This is causing axial stress in the bolts. In this case F is the total weight of the Rotary Table multiplied with the gravity force ($9.81m/s^2$). The Rotary Table weighs 499.74kg.

$$F = 499.74kg \cdot 9.81m/s^2 = 4906.2N$$

$$Fb = 63159.2N + 0.18 \cdot 4906.2N = 64042.3N$$

$$\sigma b = \frac{Fb}{As} \qquad (18)$$

$$\sigma b = \frac{64042.3N}{16 \cdot 118mm^2} = 33.9N/mm^2$$

$$\sigma_0 = \frac{F_0}{As} \qquad (19)$$

$$\sigma_0 = \frac{63159.2N}{16 \cdot 118mm^2} = 33.5N/mm^2$$

The equivalent stresses are calculated using the Von Mises criteria and formula nr 20. The total stress in the bolts is as follows.

$$\sigma_{eq} = \sqrt{\sigma b^2 + 3(0.5 \cdot \sigma_0)^2} \qquad (20)$$

$$\sigma_{eq} = \sqrt{33.9^2 + 3(0.5 \cdot 33.5)^2} = 44.6N/mm^2$$

The bolts that are used are of 8.8 stainless steel qualities. The tensile stress limit for the 8.8 bolts is 640N/mm^2. This meaning that the bolts can theoretically be loaded with 640 N/mm^2 before they start to deform.

Conclusion

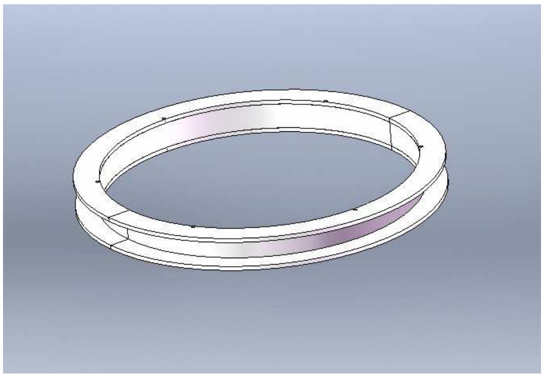
The stress in the M14 bolts is 44.6 N/mm^2. This tells us that the bolts connecting the Rotary Table to the Support Plate and Guide Beam are over dimensioned.

A smaller bolt diameter, less bolts or a lower bolt property could be used. Since the launch system will be used in an offshore environment, all bolts will have to be of corrosion resistant quality.

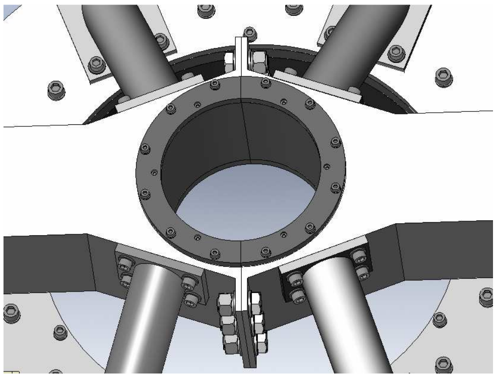
10. MATERIAL SELECTIONS

10.1 Polyacetal used as slide bearing and POM ring

The polyacetal material (POM) will be used in the Slide bearing design and in the POM Ring design. The two parts are shown in figure 33 A and B.



A) *POM Slide bearing*



B) *POM Ring protecting the umbilical from the steel Guide Beam*

Figure 33 *POM material is used for both the Slide bearing and the POM Ring*

In the last 30 years some of the thermoplastics have shown that they are highly useful materials. The plastic materials are now conquering more and more of the market. Polyacetal or better known as POM is among this group. It has proven that in certain areas, it can replace better known materials such as steel, copper, brass and aluminum. All the different acetal plastic types are designed to have various qualities. They are often compared to and favored over other thermoplastic materials. When mixing different types of acetal plastics we get a plastic material that has great properties. The acetal material has great strength and stiffness. Due to the low friction coefficient, $\mu = 0.32$ against steel, and a great wear out coefficient, the material polyacetal is quite suitable for the use in slide bearings. As a bonus, the polyacetal is very easy to work with. When working with the material in a cutting machine or a lath, same tools are used for polyacetal as for steel.

According to Astrup AS, all plastic materials are poor heat transmitters. This is also the case for polyacetal. Due to poor heat transmitting qualities, there will always be less wear out in a plastic slide bearing compared to a conventional slide bearing (conventional material such as brass or bronze). Poor heat transmission equals less PV factor. However this also depends on the slide bearing design. To make sure of minimal heat build up in the slide bearing, the

bearing needs to be designed as slim as practical possible. This will assure fast and efficient transfer of heat from the bearing to the surroundings.
Further more the surface which is sliding inside the slide bearing needs to be fairly smooth and as hard as possible. To assure this, the slide surface needs to have a property of minimum 50 Rockwell C. A hard and smooth material sliding inside the slide bearing will assure a solid and wear out resistant slide bearing design. The acetal material also allows for a high pressure/force perpendicular to the sliding face of the slide bearing. This means that a polyacetal slide bearing can deal with fairly high loads. [4]

Polyacetal (POM) Properties [4]

Rm = 65 MPa
ρ = 1410 Kg/m^3
E-module = 3000 MPa

10.2 Aluminum 6082 T6

Aluminum 6082 T6 material is intended for use in the ROV Interface design shown in figure 34. The ROV Interface is the part that is connecting the ROV and the Hydraulic Rotary Table. Due to friction between the ROV cage and the ROV Interface, the Rotary Table will be able to rotate the ROV cage.

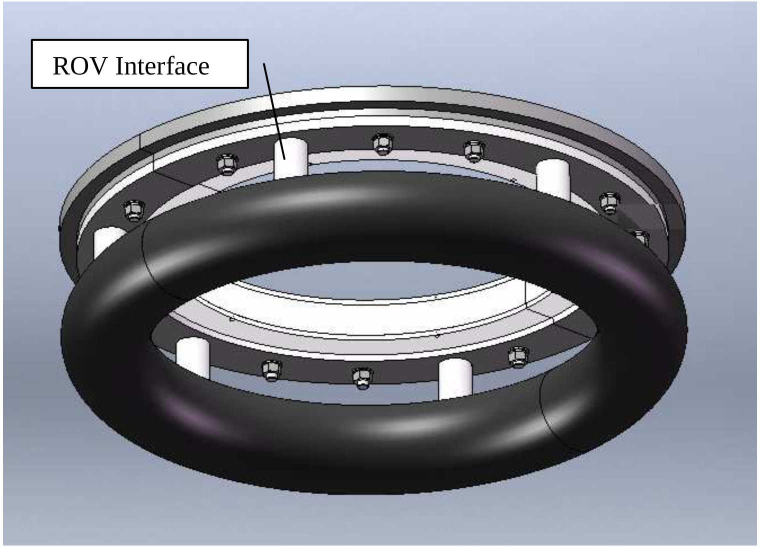


Figure 34 *The aluminum ROV Interface is bolted together with the Lower Latch Plate*

The 6000 aluminum alloy series contains magnesium and silicon as main alloys. Many strong and reliable alloys have been produced from the 6000 series. It is probably the most important aluminum alloy around, and it is widely used in all kinds of industry. The alloy has a great balance between ductility, strength, corrosion resistance and welding abilities. More than 90% of all extruded aluminum in Europe comes from the 6000 series.

The aluminum alloy 6082 is a medium strength alloy with excellent corrosion resistance. It has the highest strength of the 6000 series alloys. The 6082 is known as a structural alloy. Due to its higher strength, it has replaced the 6061 alloy in many applications. The aluminum alloy 6082 machines well and is commonly used in producing various parts where medium strength is required [5].

Aluminum 6082 T6 Properties [5]

Si, 0.7-1.3%	Cu, 0.1%
Mg, 0.6-1.2%	Ti, 0.1%
Fe, 0.5%	Mn, 0.4-1.0%
Zn, 0.2%	Cr, 0.25%

Re (0.2%) = 310 MPa

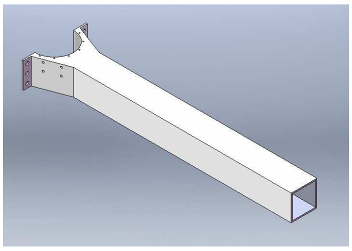
Vickers (HV) = 100

ρ = 2700 Kg/m^3

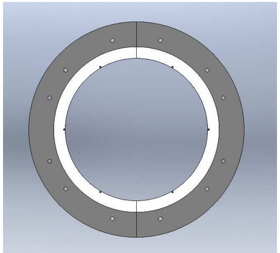
E-Module = 70000 MPa

10.3 Stainless 316 steel

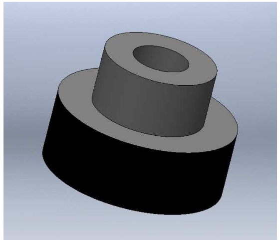
Stainless 316 steel will be used as material for the Guide Beam, both the Gearwheels, the Guide Plates, the 3 inch Support Pipes and the Support Structure. All parts are shown in figure 35 A, B, C, D, E and F.



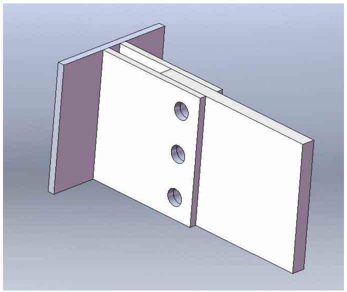
A) *Guide Beam*
(200 mm x 200 mm x L)



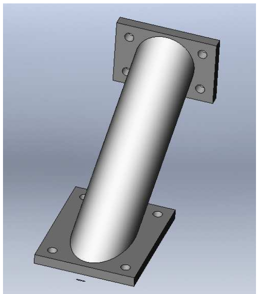
B) *Big Gearwheel*
(d=1120 mm)



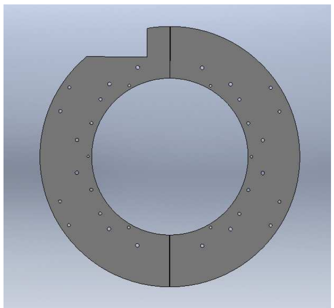
C) *Small Gearwheel*
(d=80 mm)



D) *Guide Plate*



E) *Support Pipe*



F) *Support Structure*

Figure 35 *Essential parts made of 316 Stainless steel*

Stainless steel grade 316 is the standard molybdenum grade, second in importance to the 304 amongst the austenitic stainless steels. The 304 steel contains no molybdenum. The 316 contains about 2.2-2.7% molybdenum. The molybdenum gives the 316 better overall corrosion resistance compared to the grade 304. This is particularly when it comes to pitting. The 316 has excellent forming and welding characteristics. Even when welding thin sections, post-weld annealing is not required. Due to its corrosion resistance, ductility and welding abilities, stainless 316 steel is highly used in all kinds of industrial fields. The austenitic structure gives the stainless 316 steel excellent toughness even at low temperatures. The stainless 316 steel is often used in offshore constructions [5].

Stainless 316 Steel Properties

Fe, <0.03%	Mo, <2%
C, 16-18.5%	Mn, <1%
Cr, 10-14%	Si, <0.045%
Ni, 2-3%	P, <0.03% S

Re = 210 MPa

Rockwell B (HR-B) = 95

ρ = 8000 Kg/m^3

E-Module = 193000 MPa

11. CONCLUSION

The idea behind the thesis was to take a basic look at already existing ROV Launch and Recovery Systems and to design a new and simplified System. The improved System was to be partly guided and designed so that it could be installed offshore on oilrigs with ROV moon pools. The main focus of the design was to improve the ROV crewmembers human, environmental and safety factors during launch and recovery.

During the early stage of the concept study it was suggested that some type of mechanical guiding of the ROV, through and above the moon pool, would satisfy the desired criteria. The Launch and Recovery concept was then decided to include a Hydraulic Rotary Table, Guide Beam and a Guide Pipe Structure guiding the Guide Beam. The Hydraulic Rotary Table was to be integrated on the Guide Beam.

SolidWorks have been used comprehensively to design a three dimensional model of the simplified Launch and Recovery System. This has also been the main part of the thesis.

The thesis substantiates the concept of the simplified ROV guiding through and above moon pool, and explains the benefits and challenges with such a system.

Remaining work with the design will be to calculate the fit and tolerances of the various Guide Beam parts, and to decide on who would be the proper manufacturer. Desired quality, price and lead time will all have an influence on the final result.

If the system is manufactured and installed, it will defiantly improve the ROV crewmembers works situation substantially. The design permits by far less human involvement during the ROV launch and recovery. This will to a great extent reduce the potential of injury to crewmembers and personnel who are working in the ROV moon pool area.

The designed Launch and Recovery System can be viewed using the SolidWorks software eDrawings. Two eDrawing files are attached on CD.

12. LITERATURE LIST

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[16] Interview of ROV personnel, Oceaneering AS, 24.01.2008.

ATTACHMENTS

A.	**Guide Beam Assembly (eDrawings file)**
B.	**Guide Beam Assembly with ROV Cage (eDrawings file)**
C.	**Pre Studies of Master Thesis**
D.	**Master Thesis Execution Plan**
E.	**ROV Handling Document**
F.	**Technical Data Sheet, OMR 160 Hydraulic Motor**
G.	**Technical Data Sheet, Ondrives PGE 1001-10:1 ratio Gearbox**