Coupling of marine riser and tensioner system

Master's Thesis

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Abstract

A coupled model of a marine riser and a tensioner system is built. The riser is modeled using the multi-body dynamics program MSC Adams, and the tensioner system using the powerful controls and systems simulation tool, MSC Easy5. The hydrodynamic forces on the marine riser are calculated according to linear wave theory, and implemented in the model using a custom made subroutine. The riser is modeled using flexible beam elements according to Timoshenko beam theory.

The tensioner system is modeled using moments of fluid system and body mechanics. The system is solved using equations of equilibrium of fluid momentum, mass and energy. For the unfamiliar reader, an introduction to deep sea oil exploration and production is given. Each subsystem included in the model is explained. The mode of working of the more complex subsystems is presented, followed by a discussion of the simplifications made to include them in the model. The hydrodynamic force module is based on linear potential theory of long waves, and Morison's equation for oscillating structure in oscillating flow. An introduction to wave theories with derivation of the necessary equations is given, together with a thorough discussion of hydrodynamic coefficients.

An introduction to the mechanics of fluid systems is presented, to give the reader the knowledge to understand the underlying principles of the Easy5 tensioner model.
A rough guide to MSC Adams and MSC Easy5 explaining normal usage of the programs, alterations made to them and use of the 4subsea interface, is presented. Theory regarding computational methods of solving ordinary differential equations is also offered.
Finally all results are verified by analyzing similar models in the widely used and highly acknowledged global analysis program for slender marine structures RIFLEX. Excellent compliancy is found for the hydrodynamic force module and the forced vessel motion subroutine. The coupled marine riser and tensioner model shows how tension between lower marine riser package (LMRP) and blow-out preventer (BOP) becomes a critical parameter.

Preface

This master thesis is part of the Master of Science program in Marine Constructions, at the Department of Marine Technology of the Norwegian University of Science and Technology, NTNU. The work presented in this paper started over a year back at the beginning of my internship at 4subsea, the summer of 2010. I also worked with the subject as part of my project thesis the fall of 2010. I had no knowledge of analysis of marine risers and offshore drilling, nor of the expressions and terminology related to it, prior to my internship. Although offering headaches and great frustration at times, being given the opportunity to approach a problem with an innovative and future oriented set of tools, sparked my engineering heart. The pleasure of succeeding and acquiring new knowledge in the interesting field of marine structure dynamics was very gratifying.

With the world's petroleum reservoirs becoming ever more remote, entering into the ultra-deep waters of Brazil, the Gulf of Mexico and West-Africa and the harsh environment of the Barents Sea, the world of the marine riser most certainly has not stopped to evolve. It has been fascinating to study, and maybe even contribute to, the innovative and genius solutions to the problems encountered.

I would like to thank 4subsea for the opportunity to be part of such an interesting project, and especially my advisor Ignacio Rudolfo Marré for generously contributing time and energy to stand by with guidance, explanations and words of motivation. I owe Gunnar Axelsson at 4subsea great thanks for help, support and guidance with questions regarding software and simulations. I want to thank Martin Olsson and Fredrik Sjögren at MSC Software for excellent guidance in the sometimes complicated world of computer engineering. Last I would also like to thank my supervisor at NTNU, Bernt Johan Leira, for being available for questions and support the month I spent in Trondheim while working on the thesis, and the fall of 2010 when working on the project thesis.

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13-06-2011, Asker

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Nomenclature

Symbols

a	Acceleration
c	Wave speed
d	Depth
dA	Infinitesimal area
dF	Infinitesimal force
dV	Infinitesimal volume
dx	Infinitesimal distance in x-direction
dy	Infinitesimal distance in y-direction
dz	Infinitesimal distance in z-direction
e	Mass derivative of internal energy
$f_i(\omega)$	Frequency dependent function
g	Gravitational acceleration
i	Electric current
k	Wave number, roughness height
m	Mass
$\dot{m}$	Mass flux
$\vec{r}$	Directional coordinate
s	Stroke
t	Time, pipe wall thickness
u	Velocity in local x-direction
v	Velocity in local y-direction , electric voltage
w	Velocity in local z-direction, mass flux
x	Coordinate along x-axis
y	Coordinate along y-axis
z	Coordinate along z-axis

A	Area
A_i	Amplitude of wave j
A_{kj}	Added mass coefficient in direction k due to motion in direction j
B	Fluid system property
B_{kj}	Damping coefficient in direction k due to motion in direction j
C	Electric and hydraulic capacitance
C_D	Drag coefficient
C_{Dn}	Normal drag coefficient
C_{Dt}	Tangential drag coefficient
C_{kj}	Spring coefficient in direction k due to motion in direction j
C_M	Mass coefficient
C_{Mna}	Normal mass coefficient
C_{Mt}	Tangential mass coefficient
CS	Control surface
CV	Control volume
D	Diameter
E	Module of elasticity
F_k	Force component in k-direction
H_s	Significant wave height
I	Hydraulic inertance
I_{xx}	Second area moment around x-axis
I_{yy}	Second area moment around y-axis
I_{zz}	Second area moment around z-axis
KC	Kuelegan-Carpenter number
L	Length, electric inductance
Q	Heat added system, electric charge
R	Gas constant, electric and hydraulic resistance
R_N	Reynolds number
S	Wave steepness parameter
$S(\omega_j)$	Wave spectrum
T	Wave period, tension force, temperature
T_p	Mean wave period
U_R	Ursell number
V	Volume
W	Work performed by system

α	Angle from z-axis
β	Constant in integration method
β_{syst}	Mass derivative of fluid system property
δ	Wave lag
$\Delta\omega$	Finite wave frequency interval
$\dot{\eta}$	Structure velocity
$\ddot{\eta}$	Structure acceleration
Ψ	Stream function
λ	Wave length
∇	Differential operator
ω	Wave frequency
$\vec{\omega}$	Curl
ϕ	Wave potential
ρ	Fluid density
ξ	Structural damping parameter
ζ	Instantaneous wave surface elevation
ζ_a	Surface elevation amplitude

Abbreviations

BOP	Blow-out preventer
DAT	Direct acting tensioner
DNV	Det norske veritas
DP	Dynamic positioning
GD	Easy5 gas dynamics library
GP	Easy5 general purpose library
HC	Easy5 thermal hydraulics library
I/O	Input/Output
LMRP	Lower marine riser package
USD	United States dollars
WH	Well head
XI	Easy5 external interface library
XT	Christmas tree subsea installation
RAO	Response amplitude operator

1 Introduction

1.1 Motivation

This report is the result of a Master Thesis at the Norwegian University of Science and Technology the spring of 2011. The work initiated as part of an internship at 4subsea, the summer of 2010, and developed further into a project thesis, the fall of 2010.

Exploration and production of ultra-deep waters (up to 3000m) introduces problems not contained in the marine industry's experience database, making computer aided modeling and analysis of increasing significance. As costs related to time and effort required to drill, complete and workover wells make up the greater part of the development costs, the demand for exact and efficient computer modeling is great. For this reason the technologists at 4subsea are eager to develop a tool to more accurately model the coupling between drill rig and riser. As none of the existing riser analysis programs have options for creation of fluid systems, other solutions are studied.

Coupled models of body dynamics and fluid system dynamics may be created by combining MSC Easy5 and MSC Adams. Easy5 offers great flexibility to create fluid systems, while Adams is a multi-body dynamics program. The biggest challenge with Adams is programing and implementing the lacking hydrodynamic force module. With a coupled model, we can get a more complex and accurate model of the marine riser and tensioner system. New and innovative solutions may be modeled and tested, and the operation window can be estimated with finer margins.

1.2 Background

Petroleum has been useful to mankind since ancient times, as construction material (asphalt) and for lightening purposes, but its real potential was not exploited before the latter part of the industrial revolution brought us the combustion engine. Oil wells have been drilled as far back as year 347 in China, where bamboo "drill pipes" were used to reach depths of about 240 meters. The first ever commercial oil well was drilled in Poland in 1853.

With the increasing exponentially consummation over the last century, the numbers of easily reachable oil fields have decreased (Maugeri, 2007). The timeline of offshore Exploration and Production (E&P) started in the late 1800's (Yamamoto, Morooka, & Ueno, 2007). The first ever subsea wells were drilled from piled platforms at shallow freshwater in Ohio, USA, 1891. Simultaneously, the Summerland Oil Field was discovered outside the coast of California, which led to the world's first ever offshore oil well, completed in 1896. As the timeline of the offshore oil well continued, the accessibility yet again decreases. As oil fields on shallow waters empty, deeper waters in harsher environments are explored.

To access the oil reservoirs at ever increasing depth, the technology and science have been forced to develop. Scientific study of the interaction between fluids and solid bodies has been documented since Archimedes (250 B.C.). It has intrigued great minds as those of Isaac Newton, Leonardo da Vinci, Daniel Bernoulli, Jean le Rond d'Alembert, Leonhard Euler, Claude-Louis Navier and George Gabriel Stokes. By combining mechanics of bodies and fluids with modern day computer technology and innovational engineering, modeling and simulation may be carried out before construction. This gives a better idea of the structure's capability and performance.

A variety of programs have been developed to analyze the behavior of offshore structures, such as RIFLEX, Orcaflex and Flexcom, and just as many exist to simulate fluid system mechanics. None of them are capable of analyzing coupled models combining the two domains.

1.3 Approach

MSC Software offers a broad spectrum of engineering software, including the multi-body dynamics program MSC Adams and the control system simulation program Easy5. MSC Adams allows for modeling of flexible beams with distributed mass, a valid representation of a marine riser. A wave force subroutine is programmed, making Adams able to computing hydrodynamic forces based on Morison's equation for oscillating slender cylinder in oscillating flow, and Archimedes' law of buoyancy. Assuming marine risers to be slender structures is realistic.

The tensioner system is modeled in Easy5 by a suitable number of components (pressure sources, pipes, valves, storage volumes, cylinders and pistons). The equilibrium states are calculated by solving equations of equilibrium for fluid mass, momentum and energy.

1.4 Scope of work

Work description

Marine risers are widely used in offshore industry when drilling subsea wells, and for accessing wells during completion and service. The marine riser is deployed from floating vessels and is thus exposed to loads from vessel motions as well as environmental loads from waves and current. The riser is top tensioned through a tensioner system which typically consists of mechanical and hydraulic subsystems.

The dynamic response caused by loads imposed on the riser is important when considering fatigue damage accumulation on the subsea wellhead systems. Tensioner system hysteresis is believed to affect the loads on the wellhead, and there has therefore been an increased focus on the interaction between riser response and tensioner system dynamics. Global riser models used in the industry have in general not incorporated the dynamics of the tensioner system, only including riser tension through a simplified approach.

The task for this master thesis is to develop a model of a coupled marine riser and tensioner system. The model shall be developed in the software package ADAMS by MSC Software.

ADAMS does not include a hydrodynamic module, but is designed for multibody dynamics and motion analysis. Developing a hydrodynamic module for the program has been done in a pre project together with a thorough literature survey. The hydrodynamic module developed in the pre project is basic, and will need to be further developed. The goal is to use ADAMS to model a coupled system simulation of a marine riser and tensioner system.

Scope of work

1. Further literature survey within the field of marine riser systems, with special depth on tensioner systems.
2. Further develop a hydrodynamic module for slender structures. Build a marine riser model and verify the hydrodynamic module by comparing output from ADAMS to provided results from an established global riser analysis software package (e.g. Riflex, Orcaflex).
3. Expand the marine riser model in ADAMS to include a tensioner system. The tensioner system shall be modeled as a multi physical domain system, comprised of mechanical, hydraulic and pneumatic components.

The report shall be written in English and outlined as a research report including literature survey, description of mathematical models, simulation results, model test results, discussion and a conclusion including a proposal for further work. Source code should be provided on a CD with code listing enclosed in appendix. It is assumed that Department of Marine Technology, NTNU, can use the results freely in its research work, unless otherwise agreed upon, by referring to the student’s work. The part of the report using detailed knowledge obtained from 4Subsea will be confidential in nature.

The thesis should be submitted in three copies within 14th of June.

1.5 Report organization

- *Chapter 2* gives an introduction to the challenges of deep water subsea exploration and production, and the equipment and tools required to make it possible.
- *Chapter 3* gives a thorough description of the marine riser with all of its components.
- *Chapter 4* goes in depth of how tensioner systems work and the offers a presentation of the most common tensioner systems.
- *Chapter 5* explains the basics of wave theories and exciting wave forces, including concepts as added mass and drag force.
- *Chapter 6* gives a short introduction to the basic theory of hydraulics and pneumatics.
- *Chapter 7* offers a clarifying conceptual overview of the path to solving the thesis.
- *Chapter 8* provides an overview of the multi body dynamics program MSC Adams, and gives an explanation for choosing this exact program. Further, discussing advantages and disadvantages with the choice.
- *Chapter 9* provides an introduction to the control system program MSC Easy5, and the theory behind.
- *Chapter 10* gives a short introduction to RIFLEX.
- *Chapter 11* describes how the marine riser and tensioner system are modeled, assumptions made, and the simplifications made to be able to run the analysis while achieving sufficiently accurate results.
- *Chapter 12* evaluates the results of the analysis as well as comparing them with results of similar analysis in RIFLEX.
- *Chapter* 13 gives a summary of the thesis and discusses the results.

2 Subsea Exploration

Offshore oil fields are reached from drill rigs equipped with drilling risers. Depending on environmental conditions as wave, current and wind loads, and water depth, the drill rig may have one of many shapes or sizes. On shallower waters (30-100m), the rigs are normally standing on the seabed, while for intermediate water operations (100m-300m) tension leg platforms are widely used. For deep and ultra-deep waters, semi-submersibles are employed (AIP, 2009). An excerpt of drill rigs are shown in Figure 2-1.

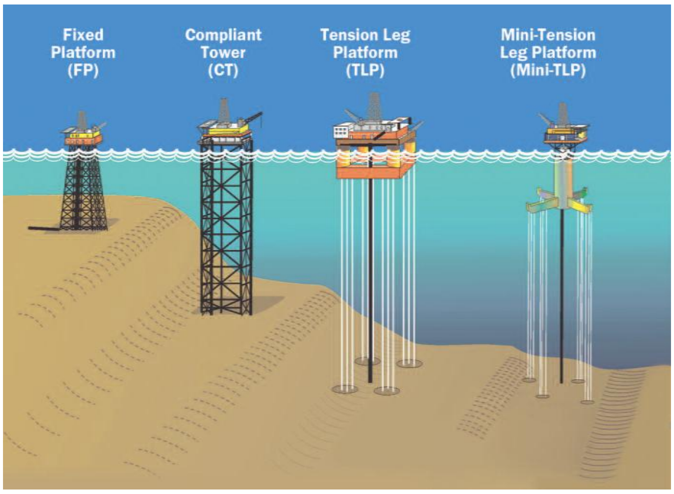

Figure 2-2 Platform types (Shallow and intermediate waters)

Over the last 30 years, the discovery of giant oil fields at deep (300-1500m) and ultra-deep (deeper than 1500m) has drastically increased the demand for floating drill rigs (Yamamoto, Morooka, & Ueno, 2007). The reservoir is reached by a long and slender pipe referred to as drill string or drill pipe. At the reservoir end of the drill pipe sits the drill bit, and the other is connected to the rotary of the drill rig. As the rig is subject to oscillating wave forces, and only restricted by anchors or dynamic positioning (DP), it moves relative the seabed and the reservoir. To avoid smashing the drill bit against the bottom of the borehole and tearing off the riser, flex and telescope joints, and compensator and tensioner systems are installed. This allows the oscillating rig to be connected to the more static drill riser. A situational description of drilling from floating rig is shown in Figure 2-1.

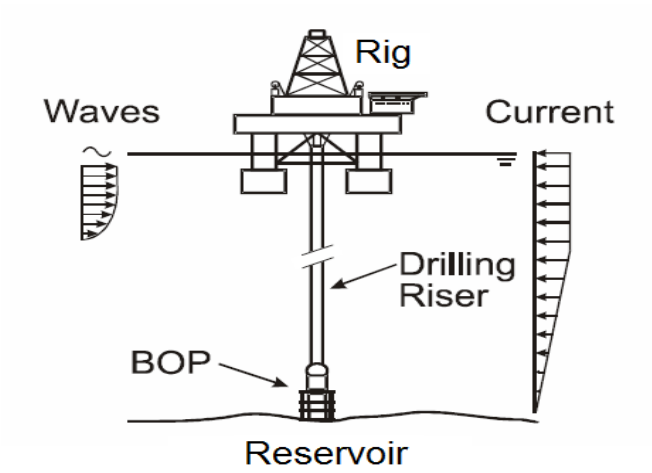

Figure 2-1 Semi-sub (deep and ultra-deep waters)

2.1 Petroleum reservoir

Petroleum is the result of organic waste exposed to millions of years of extreme pressure and high temperatures. A typical subsea petroleum reservoir is created of dead plankton and algae, accumulated on the seabed and buried, and over millions of years exposed to the right temperature and pressure (Nurmi, 2010). There are, from nature's side, three main factors that need to be present in order for a reservoir to be created.

- *Source rock*, a rock rich on hydrocarbons, deeply buried in order for the right pressure and temperature to be present.
- *Reservoir rock*, a porous rock where the petroleum may be contained.
- *Cap rock*, a rock sealing off the reservoir and this way preventing the petroleum to escape.

2.2 Challenges

The challenges of deep sea exploration can roughly be divided into economic, environmental and technological;

- *Economic:* The economic viability is strongly dependent on the crude oil prices versus production cost. Production costs depend on among other things rig rental costs, distance from the coast and drilling depth. With increasing depth comes increasing expenses expense with drilling rigs, production equipment and limited knowledge. Some subsea oil fields are only viable at a crude oil price over 80-90 USD per barrel. As a result of the financial crisis of 2007-2010, where oil prices plummeted from almost 150 to mere 60 USD in a couple of months, awareness of the field's viability has increased.

- *Physical and technological:* The physical challenges with deep sea oil wells are closely related to hydrostatic pressure and displacement related forces. The motion of the riser increases with the length of it. The equipment has to withstand hydrostatic pressure at depths of up to 3000 meters, before drilling through different types of rock for up to 10 000 meters.

  Hardened steel is used to develop drill crowns able to crush solid rock, and pressure systems are installed to remove the gravel. Closer to the surface, wave and wind forces give the toughest challenges. The effects of these are minimized by employing heave compensator and tensioner systems, as well as dynamic positioning and strategic hull shapes (Framnes & Gleditsch, 1994). When finally the well is connected to production facilities by sealed risers, the fraction of water in the crude oil prompts other challenges, by turning into ice and clogging the pipelines. This is usually the result of the extreme pressure gradients experienced. The crude's viscosity also introduces challenges, as it is difficult to transport high viscous liquids. This can be solved by preprocessing the oil at subsea preprocessing stations.

- *Environmental:* The environmental aspect of petroleum exploration and production is becoming increasingly important. Even though oil and gas will remain the greatest source of energy for some time yet, greener energy sources are developing. Every oil spill, significant or insignificant, not only harms the environment, but stains the reputation and the environmental conscience of the parties involved, and of the industry as a whole.

Figure 2-3 Oil spill (Think quest)

2.3 Accessing subsea reservoirs

Offshore drilling is usually separated into two domains; exploration drilling and production drilling. Exploration wells can be either research wells to determine the existence of petroleum, or appraisal wells to determine the size of the reservoir. Production wells are either petroleum extraction points, or production enhancing wells. The production enhancing wells are used for gas and/or air injection to force the crude out of the extraction wells (Framnes & Gleditsch, 1994).

Once a reservoir has been located, sized off and labeled as commercially viable, one or more production wells are installed. The well is opened to the largest diameter at the seabed, fitting casings of up to 36”. The diameter of the drill bit and casings decreases towards the reservoir, with the smallest diameter being 5”. Near the sea bed, each section of casing is cemented into place to increase structural stability, distribute the loads, and prevent leakage. When drilling in areas with high probability of striking shallow gas, a leading hole of diameter 9 7/8” should be drilled first, before opening the well to the right diameter (Gleditsch & Framnes, 2000).

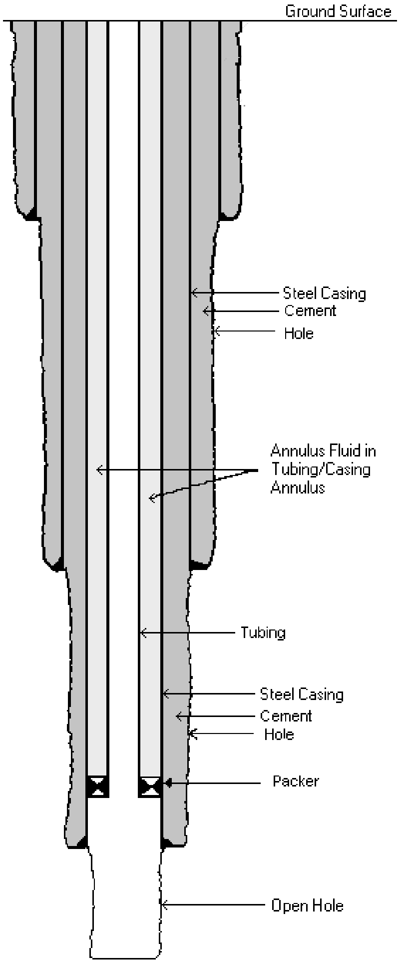

Figure 2-4 Well casings

2.4 Equipment and systems

A lot of equipment and many systems are required to ensure safe and sufficient offshore E&P. This subchapter will give a brief introduction to the equipment with key importance. Deep-water offshore wells are drilled by tensioned marine risers. The tensioning is vital for stability reasons and because the riser is not self-buoyant. With increasing water depths, to maintain a tensile force lower than the material’s yield strength, buoyancy joints are added.

Before completion of the well, the drilling may be done by a bare drill, but as soon as the well is completed, the full marine riser is used (Yamamoto, Morooka, & Ueno, 2007). This is done to avoid polluting the ambient environment when the reservoir is reached. With the marine riser in place, there is a pipe-in-pipe situation, with the drill pipe inside the outer casing. To enable motion of the riser, flex and telescope joints are installed at the top and bottom of the marine riser. Directly on top of the wellhead (WH), sits the blowout preventer (BOP) to prevent leakage. It is connected to the bottom most section of the marine riser called lower marine riser package (LMRP).

2.4.1 Drill string

The drill string's main objective is to transfer the torsion from the rotary to the drill bit. The string consists of drill pipe, tool joints and drill collars. A typical drill pipe section is hollow for liquid transportation and 7-15 meter of length. Tool joints are situated at the end of each drill pipe section for easy assembly and disassembly. Drill collars are added closer to the drill bit to create sufficient force for the boring, sufficient tension in the drill string and to stabilize the direction.

2.4.2 Marine Riser

The marine riser is a complex pipe of various sections connecting the drill rig to the well head. It serves as a well access point for the rig.
The marine riser is described to the full extent in chapter 3.

2.4.3 Tensioner system

The tensioner system is meant to uphold a constant marine riser tension, avoiding large tension and displacement gradients.
The tensioner system is described to the full extent in chapter 4.

2.4.4 Drill string compensator system

The main task of the drill string compensator system is to eliminate any vertical motion of the drill string. This is done to maintain constant distance from the rotary to the sea bed and constant force on the drill bit (Framnes & Gleditsch, 1994). This will hinder the drill bit from smashing against the end of the bore hole, and ensure a higher penetration rate.

Wire line compensators are widely used. Starting from the hooking point of the drill string, wires or chains are guided through guide-wheels and over the top of a large hydraulic piston-cylinder installation. The *high-pressure* side of the piston is connected to a large volume of high pressure gas, so that a heaving motion of the rig, results in a stroke of the piston, and a compression of the gas. As gas is compressible, and the volume is large, the pressure force and the drill string remains close to constant. On the *low-pressure* side of the piston, hydraulic oil in collaboration with a speed sensitive regulating valve prevents the piston to smash into the cylinder in the case of a recoil.

We differentiate between active and passive heave compensation, AHC and PHC. The passive system is as described above, floating on a cushion of air. It has an efficiency of 40-85% depending on the rigs motion. ACH employs a computer controlled system of hydraulic cylinders to force the contra motion, and delivers an efficiency of about 95%, independent of the rigs heave translation (Gleditsch & Framnes, 2000). A third group assisting the elimination of vertical movement of the drill string is *bumper subs*. Bumper subs are telescope joints in the drill string.

2.4.5 Christmas tree (XT)

The final installation before production, after well completion and installation of production risers, is the Christmas tree (XT). The Christmas tree sits on top of the well head and controls the hydrocarbon and chemical flow to and from the reservoir. It controls both injection of chemicals to solve blockage and clogging problems, injection of water or gas to further increase the economic efficiency, and is also the amount of hydrocarbons extracted from the reservoir (Gleditsch & Framnes, 2000).

A typical subsea XT consists of 4-5 valves, and is operational on depths of up to 3000 meters. Situated at the bottom of the XT are the two master valves (upper and lower), which lie directly in the flow way. The lower master valve is usually manually operated, while the upper is hydraulic. Next we have wing valves; production wing valve and kill wing valve. The production wing valve is where the production unit taps in, while the kill wing valve is for injection of chemicals. Last there is the swab valve. This valve is situated at the top of the XT, and is used for well intervention.

2.4.6 Wellhead (WH)

The wellhead is the top most point of the well, and serves as a connection point for the BOP during drilling and Christmas tree (XT) during production. It is also a point of suspension for the casings and the production pipe leading down into the reservoir. The first casings installed, are cemented into place, and the wellhead is then welded directly onto them.

3 Marine Risers

The marine riser connects the rig to the blow-out preventer (BOP). It is used for well intervention and drilling. The riser serves, not only as a barrier for the high pressure of the reservoir, but also as a barrier against the harsh surrounding environment.

To endure the rough task of being connected both to a moving point in one end, and a fixed point in the other, the marine riser consists of several joints, each with a special contribution to make the connection possible. As the riser is not positively buoyant, a tension force is applied at the rig. To withstand the extreme tension force, the casings of the marine riser are preloaded in axial pressure before they are welded together.

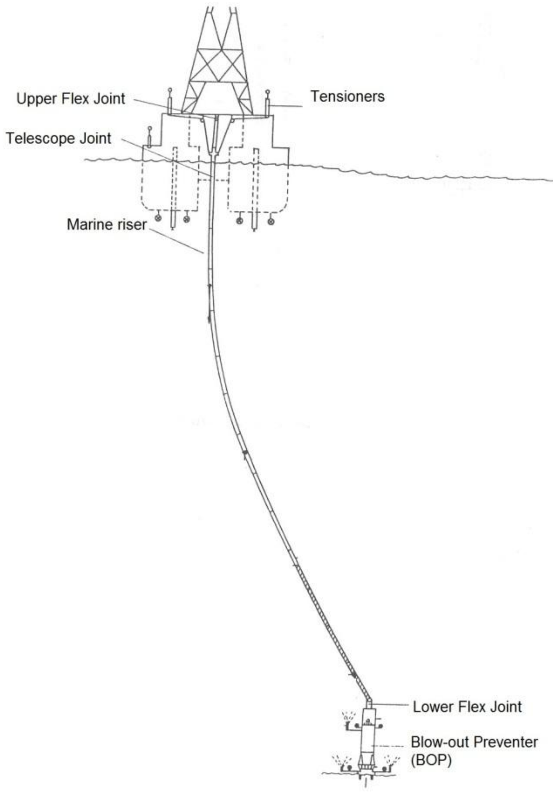

Figure 3-1 Drawing of drilling situation (Framnes & Gleditsch, 1994)

3.1 Joints

There are many different joints in the topology, including flex joints, pup joints, slick joints, slip joints and buoyancy joints.

- The *flex joint* is a short (1-2m) and elastic section.
  A ball makes it able to rotate and flex, having bending stiffness bound by the extent of rotation to be allowed and the maximum amount of force to be transferred to the BOP. From fundamental mechanics, we know that the stiffer the joint, the more forces are transferred. During drilling the flex joint's objective is to avoid forces being transferred to the BOP and therefore damaging the well head and at the same time avoid deformation of the drill pipe. Normally the flex joint allows +-10 degrees rotation.

- The *slip/telescope joint* consists of two pipes (inner and outer barrel), one inside the other. It allows vertical rig motion relative the seabed, by letting the inner barrel stroke in and out of the outer barrel. The inner barrel has the same dimensions (diameter and wall thickness) as the rest of the riser, while the outer barrel is bigger (Framnes & Gleditsch, 1994). Welded to the top of the outer barrel is the tensioner ring, which serves as a connection point for the tensioner system. The slip joint is usually equipped with two seals to ensure a pressure lock between the two pipes, and keep the drill fluid from escaping.

- *Buoyancy joints* make up the major part of the marine drill riser, and contribute to the global decompression of the riser. The buoyancy joint is a slick joint with a buoyancy volume attached to it. There are many different buoyancy joints on the market today, with either foam elements or air filled aluminum tanks serving as buoyancy volumes.

- *Slick joints* are pipe sections with the same dimensions as the buoyancy joints, but without the buoyancy elements.

- *Pup joints* are steel pipes used to adjust the overall length of the marine riser.

3.2 Lower Marine Riser Package (LMRP)

In harsh weather the rig motion may exceed the limitations of the heave compensator and the tensioner systems. When the rig motion becomes larger than what the systems are dimensioned for, they are no longer capable of maintaining constant force or position as they require. As there is no linearity in the behavior of fluid systems, the tension force may pulsate violently. The forces may be transferred to the BOP and also be large enough to inflict damage on the well head. To be able to swiftly disconnect the riser from the BOP, the lower marine riser package (LMRP) was developed. The LMRP connects the marine riser to the BOP. It is equipped with valves for easy connection and disconnection (Framnes & Gleditsch, 1994).

3.3 BOP

During the drilling phase, one depends on the hydrostatic pressure of the slag to be greater than the ambient pressure, and therefore prevent oil and gas to enter into the well. The drill slag is fed into the marine riser by the *kill*, *choke* and *booster lines*. These lines are strapped or clamped to the outside of the marine riser. On the buoyancy joints they are covered by buoyancy elements and therefore invisible. At some point the drill crown may pass through pockets of the reservoir with extreme pressure, and cause gas or crude to enter into the well at high velocity. This is called a kick, and may develop into a blowout if not brought under control. To monitor the pressure and provide a solid anchor for the riser and drilling equipment, a Blowout Preventer (BOP) is installed.

The BOP is basically a controllable valve, able to close or shut of the well in several ways. It is controlled from the surface by a *pod*. The pod is either an electric or hydraulic signal system, or a combination of the two. For redundancy purposes, several BOPs are installed, one on top of the other, making up a BOP stack. The stack is typically of height 10-15m, and about 100-400 tons.

The BOP stack is rigged with high pressure accumulators, which are controlled by a signal from the surface. In principal there are two types of BOPs; annular and ram. The *annular* safety valve works in the longitudinal direction and consists of an elastic steel reinforced “dough-nut” and a hydraulic cylinder. At the occurrence of a kick, the “dough-nut” will be given an up-wards and in-wards motion, sealing off the annulus in a wedge cork fashion. This type of BOP consists of only two parts, a hydraulic cylinder and an elastic seal. It is also capable of sealing the void around noncircular shapes. Because of its simplicity and flexibility, it is commonly used, but normally in combination with other ram and annular valves.

Ram BOPs are normally what makes up the lower part of the BOP stack. They act transverse to the riser, and are, depending on the type, able to both seal off the well and cut the drill string.

1. *Pipe rams*; make a seal around the drill pipe, restricting flow in annulus, but not within the drill pipe.
2. *Blind rams;* seal off void with no drill string or equipment in the way.
3. *Shear rams;* are able to cut drill pipe to free drill rig from the well.
4. *Blind shear rams;* as shear rams, but can also seal to do tests on valves beneath.

3.4 Diverter

During drilling and work over operations, gas may accumulate in the annulus of the marine riser. This usually occurs in situations before the BOP has been installed, but also after gas may migrate past and accumulate in the riser. The diverter system is normally situated right beneath the rotary ensures pressure to be relieved safely and a way to divert the gas away from the rig. To force the gas out of the annulus, it is filled with drill fluid. The mixture of gas and mud exiting the riser must be separated, as the mud is reused. Depending on the wind direction, the diverter system may be set to discharge gas at different orifices around the rig (Framnes & Gleditsch, 1994).

4 Tensioner Systems

4.1 Objective

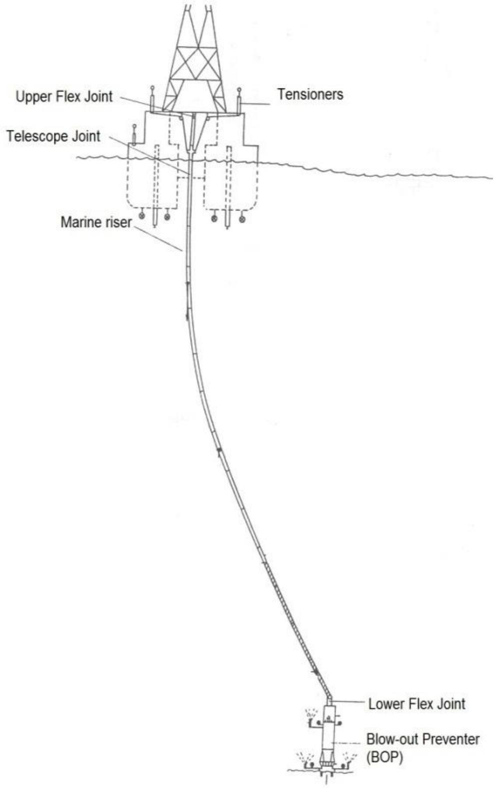

Figure 4-1 Semi sub with marine riser (Framnes & Gleditsch, 1994)

The marine drilling riser is the physical connection between the rig and the sea bed. The rig's wave induced motion relative the static sea bed introduces problems. The tensioner system is the mechanical link between the drilling rig and the marine riser, and is intended to deliver a constant tension force on the tension ring (Framnes & Gleditsch, 1994).

The tension force is needed primarily because of the riser's negative effective weight, but also to ensure small horizontal displacements. Too small tension force would mean buckling of the riser.

4.2 Principle of working

Although tensioners come in different forms and sizes, the general concept of working is usually the same; the tensioners "float" on a pneumatic-hydraulic cushion. The pneumatics can be viewed as a spring and the hydraulics as a damper (Framnes & Gleditsch, 1994). The two most common tensioner systems are; wire line tensioners and direct acting tensioners (DAT).

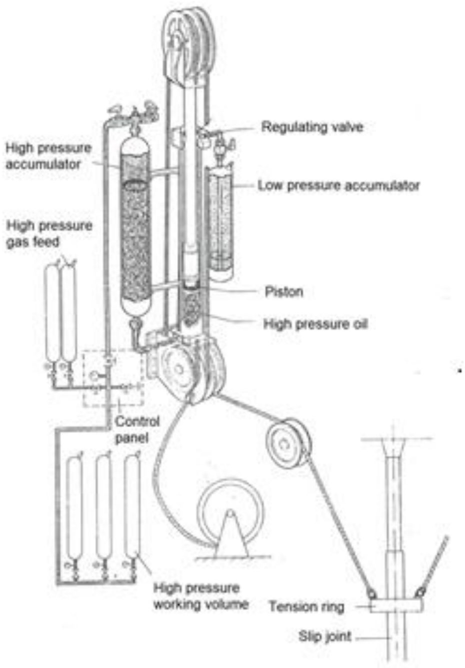

Figure 4-2 Direct acting tensioner (DTI, 2011) and wire line tensioner (Framnes & Gleditsch, 1994)

Both tensioner systems have large volumes of high pressure gas, gas-oil accumulator, and high pressure feed gas on one side of the piston. This side is referred to as the pressure side. The other side is supported by a low pressure gas reservoir. To utilize the full stroke potential, the tensioners should be at mid position at calm sea. This means that they can stroke $\pm s/2$.

From equilibrium, with no sea, the pressure on the high pressure side is increased by letting gas from the feed reservoir enter the system, until wanted tension is achieved. Then, as the rig starts to move, hydraulic oil is forced in and out of the cylinders pressure side. The oil then forces gas in and out of the accumulator, as oil is considered incompressible. The gas on the other hand, is very compressible, and as the high pressure reservoir is large, the gas mass fraction entering the reservoir is small, and therefore the pressure is close to constant.

On the conventional wire line tensioner version, the tension force is applied to the riser by wires, while on the DAT version the cylinders are directly connected to the deck and the marine riser. The possibility to stop the vertical motion after an eventual snap of the riser has made the DAT system preferable in later years because of the increasing water depth, when the riser is subject to greater forces.

5 Theory of Hydrodynamics

5.1 Approaching wave dynamics

In this chapter the basis for theory of the hydrodynamic force implemented in Adams is laid. To derive an expression for the hydrodynamic forces acting on a solid in water, we first need a way to describe the motion of the water particles. Describing the exact motion of a real flow is an impossible task, but there are some tools that may help us to get satisfactory results. Depending on the conditions at hand and the need for accuracy, we may employ various wave theories.

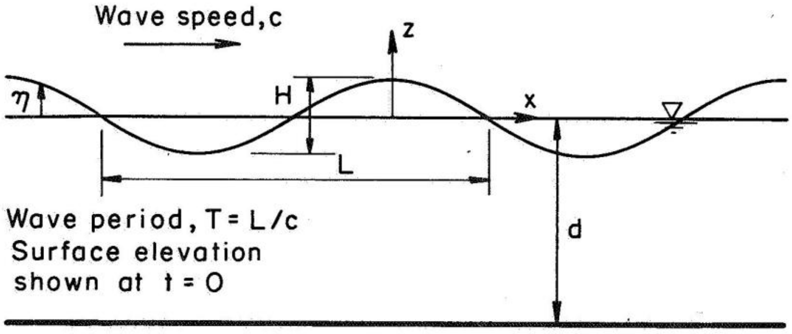

Figure 5-1 Characteristic wave parameters (DNV, 2007)

We have three parameters to help us select the correct theory.

- Steepness parameter: $S = \dfrac{H}{\lambda_0}$
- Shallow water parameter: $\mu = \dfrac{d}{\lambda_0}$
- Ursell number: $U_R = \dfrac{H\lambda^2}{d^3}$

These parameters are not uniquely defined. If two of them are given, the third can be found. The relation between them is defined as $U_R = S/\mu^3$.

5.1.1 Choice of theory

To find the appropriate wave theory for a specific problem, there are two important moments: validity (ref. Figure 5-2) and complexity. It is wise to always choose the least complex theory valid for the problem. The deep water condition is fulfilled in most offshore situations, and if we assume the wave height to be sufficiently small compared to the wave period, we may use linear theory. As linear theory gives a reasonable description of our problem, in addition being the mathematically least demanding, it is the natural choice. In the following chapter the background for linear theory is therefore closely studied, while the others are superficially presented.

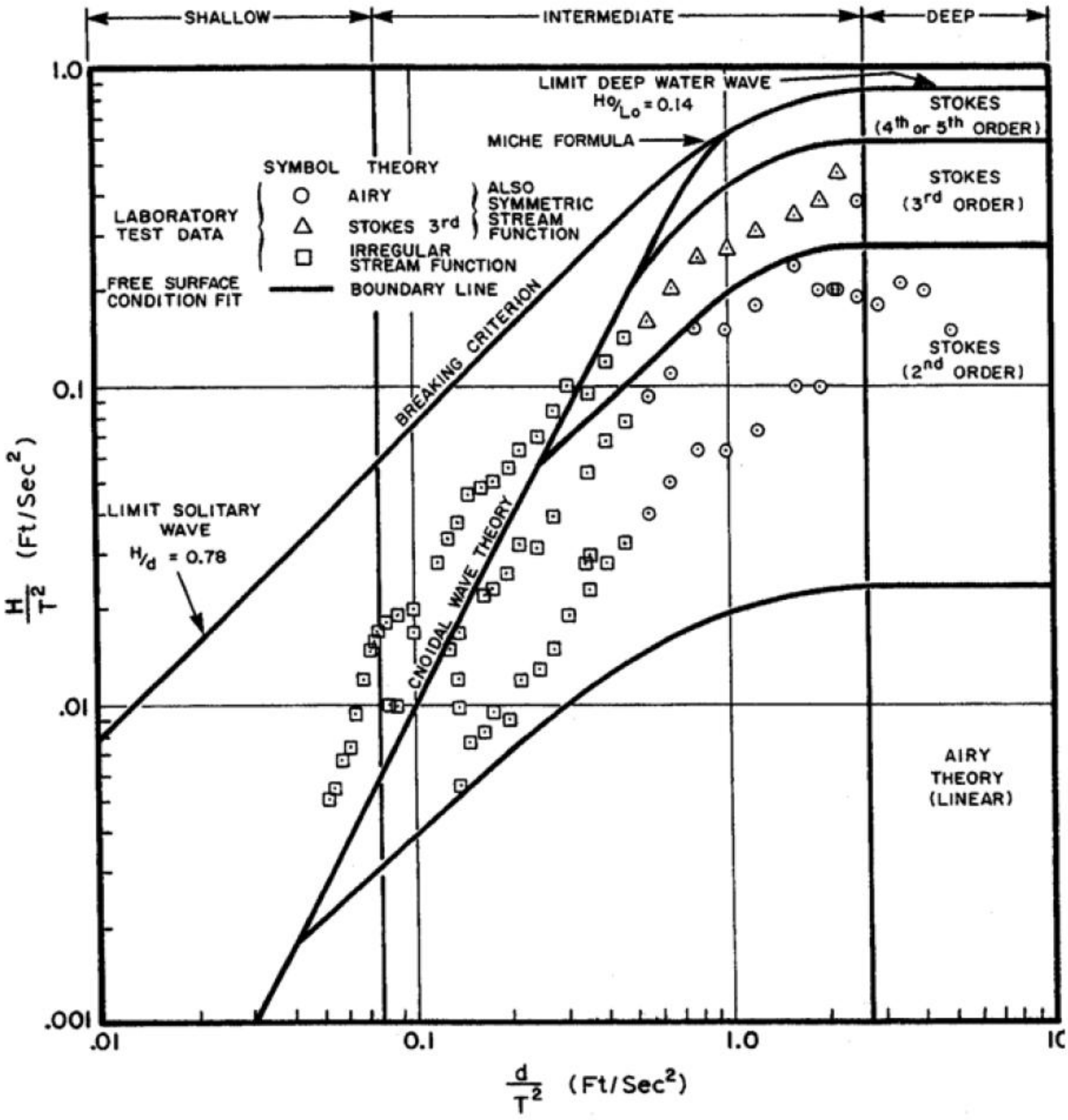

Figure 5-2 Wave theory domain. Horizontal axis measures shallowness, vertical axis measures steepness (DNV, 2007)

5.2 Wave theories

5.2.1 Linear wave theory

The velocity potential of an undisturbed linear wave, also called Airy wave, may be derived from the following boundary conditions and assumptions:

- Irrotational and inviscid
- Incompressible
- No vertical velocity at sea bed
- Atmospheric pressure at the surface
- Low waves

5.2.1.1 *Laplace's equation*

Laplace's equation is fulfilled for any incompressible potential flow. In potential flow, the velocity components are described by

$$\vec{V} = \left\langle \frac{\delta\Phi}{\delta x}, \frac{\delta\Phi}{\delta y}, \frac{\delta\Phi}{\delta z} \right\rangle$$

Equation 5-1 (Faltinsen, 1990)

And the curl of the flow is assumed to be zero

$$\vec{\omega} = \nabla \times \vec{V} = \vec{0}$$

Equation 5-2 (Faltinsen, 1990)

Sea water is assumed to be incompressible, a realistic assumption.

$$\nabla \bullet \vec{V} = \vec{0}$$

Equation 5-3 (Faltinsen, 1990)

By combining these equations, we see that the velocity potential fulfills the Laplace equation

$$\frac{\delta^2\Phi}{\delta^2 x} + \frac{\delta^2\Phi}{\delta^2 y} + \frac{\delta^2\Phi}{\delta^2 z} = 0$$

Equation 5-4 (Faltinsen, 1990)

5.2.1.2 *Zero velocity at sea bed*

If we assume no vertical velocity over a non-inclined, plane and infinite sea bed, we get the following condition;

$$\left.\frac{\delta\Phi}{\delta z}\right|_{z=-h} = 0$$

Equation 5-5 (Faltinsen, 1990)

5.2.1.3 *Dynamic free-surface condition*

By assuming atmospheric pressure anywhere on the surface, the dynamic free-surface condition is derived. Introducing Bernoulli's equation and assuming non-linear contributions to be zero, gives

$$g\zeta + \frac{\delta\Phi}{\delta t} = 0$$

Equation 5-6 (Faltinsen, 1990)

5.2.1.4 *Kinematic boundary condition*

The kinematic boundary condition is a result of presume that the wave height is small compared to the wave length, and that water particles at the surface will stay at the surface

$$\frac{\delta\zeta}{\delta t} = \frac{\delta\Phi}{\delta z}$$

Equation 5-7 (Faltinsen, 1990)

5.2.1.5 *Velocity potential*

From these conditions we can derive the velocity potential

$$\Phi = \frac{g\zeta_a}{\omega}\frac{\cosh(k(z+h))}{\cosh(kh)}\cos(\theta)$$

Equation 5-8 (Faltinsen, 1990)

$$\theta = k(x\cos(\beta) + y\sin(\beta))$$

Equation 5-9 (Faltinsen, 1990)

That, for deep water, converges towards the following;

$$\Phi = \frac{g\zeta_a}{\omega}e^{kz}\cos(\theta)$$

Equation 5-10 (Faltinsen, 1990)

This is a reasonable assumption when applied to describe an offshore sea state.

5.2.2 Stokes wave theory

In stokes wave theory the surface elevation is described by adding higher orders of the wave height. First order Stokes is identical with an Airy wave, whilst second order is described by:

$$\eta = \frac{H}{2}\cos(\theta) + \frac{\pi H^2}{4\lambda}\cos(2\theta)$$

Equation 5-11 (DNV, 2007)

$$\theta = k(x\cos(\beta) + y\sin(\beta))$$

Equation 5-12 (DNV, 2007)

Second order Stokes theory makes the crest sharper and the troughs wider. The linear dispersion relation holds up, and therefore the wavelength λ and the phase velocity c remain independent of the wave. This gets increasingly complicated for higher order theory, as shown below for third order.

$$c^2 = \frac{g}{k}\left[1+\left(\frac{kH}{2}\right)^2\right]$$

Equation 5-13 (DNV, 2007)

5.2.3 Cnoidal wave theory

Cnoidal waves are nonlinear and periodic, with very broad troughs and sharp crests. The cnoidal wave function is an exact solution of the Korteweg-de Vries equation (Drazin & Johnson, 1989), an equation for waves with large wavelength at shallow waters.

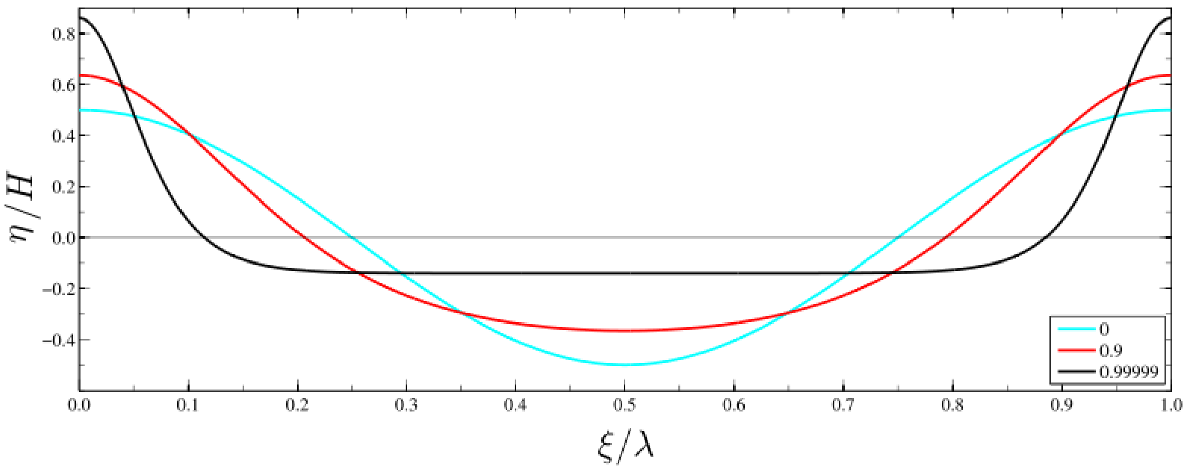

Figure 5-3 Cnoidal wave (Drazin & Johnson, 1989)

5.2.4 Solitary wave theory

Solitary wave theory describes the situation of a single wave travelling through a liquid. It was discovered in 1834 by the naval architect Scott Russell while studying canal boats. He noticed that the solitary wave created by towing a barge through a canal, would, when the barge had been stopped, continue for 2-3 kilometers with the same height and velocity. Characteristic is that the wave elevation is exclusively wave crest.

$$\eta(x,t) = H\cosh^{-2}\left[\frac{\sqrt{3\frac{H}{d}}}{2d}\left(1-\frac{5}{8}\frac{H}{d}\right)\left(x-1.33\sqrt{gd}\,t\right)\right]$$

Equation 5-14 (DNV, 2007)

5.2.5 Stream function wave theory

When a broader range of validity is needed, the stream function wave theory can give satisfying results. This method is computationally easier than stokian, as can be seen from the stream function below.

$$\Psi(x,z) = cz + \sum_{n=1}^{N} X(n)\sinh(nk(z+d))$$

Equation 5-15 (DNV, 2007)

The number of terms needed depends on the steepness parameter S and the shallow water parameter μ . N increases with increasing steepness, and N=1 gives an Airy wave.

5.2.6 Surface stretching

To correct for some of the simplifications done when deriving the different theories we "stretch" the instantaneous wave surface. There exist three methods for this correction;

- Grue's method
- Wheeler's method
- Second-order kinematics model

As the *Wheeler stretching method* is simple and widely used, it will be presented and used. The method is based on the observation that the fluid velocity at the still water level is reduced compared with linear theory. The basic principle is that from a given free surface elevation record, one computes the velocity for each frequency component using linear theory and for each time step in the time series, the vertical coordinate is stretched according to Equation 5-16.

$$z = \frac{z_s - \eta}{1+\eta/d} \quad ; \quad -d < z < 0 \;\; ; \;\; -d < z_s < \eta$$

Equation 5-16 Wheeler stretching (DNV, 2007)

Where η the free surface, d is the water depth and z_s is the stretched z-coordinate. From Figure 5-4 below, we see how the new z-coordinate z_s is valid up to the dynamic free surface (DNV, 2007).

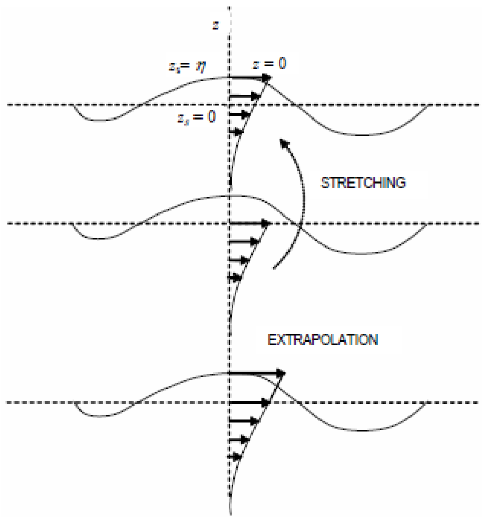

Figure 5-4 Stretching velocity profile (DNV, 2007)

5.2.7 Irregular and regular sea state

Real-life ocean waves are not smooth geometrical functions, but rather a random signal described by mean values of wave height, period and lag. The wave characteristic value of wave height used in irregular sea is called significant wave height (H_s), and is given by the mean value of the 1/3 highest waves. The characteristic wave period (T_p) is given as the mean wave period. It is possible to describe this signal by a sum of 20 or more regular waves with parameters distributed according to wave spectra (Faltinsen, 1990). Wave spectra are functions that give the wave energy's dependency of the wave frequency. The wave energy is denoted by the wave height squared.

$$\frac{1}{2}A_j^2 = S(\omega_j)\Delta\omega$$

Equation 5-17 (Faltinsen, 1990)

It is derived from measured data. The wave spectrum indicates for which frequency interval the energy of the sea state is concentrated. When approximating an irregular sea state by a single regular wave, the following formulas can be applied;

$$T = T_p; \qquad H = 1{,}92H_s$$

Equation 5-18

Wave data is often measured for specific areas and for specific projects. They are usually tabled values of wave height and period.

Tp (s)			2	4	6	8	10	12	14	16	18	20	>=
Hm0 (m)			4	6	8	10	12	14	16	18	20	22	22
0	-	0,5	0,005	0,043	0,145	0,156	0,061	0,029	0,011	0,007	0,007	0	0
0,5	-	1	0,179	1,465	3,15	2,708	1,157	0,32	0,07	0,034	0,023	0,009	0,011
1	-	1,5	0,045	2,524	5,101	4,903	2,551	0,959	0,283	0,091	0,027	0,011	0,002
1,5	-	2	0,002	1,613	4,497	5,205	2,851	1,39	0,513	0,129	0,027	0,011	0,005
2	-	2,5	0	0,438	3,332	4,368	3,023	1,188	0,701	0,111	0,018	0,007	0,002
2,5	-	3	0	0,113	2,15	3,565	2,672	1,225	0,637	0,113	0,027	0,014	0
3	-	3,5	0	0,007	1,222	2,969	2,406	1,077	0,624	0,206	0,032	0,005	0,002
3,5	-	4	0	0	0,531	2,252	2,123	1,129	0,506	0,161	0,043	0	0
4	-	4,5	0	0	0,188	1,642	1,753	0,991	0,451	0,129	0,02	0,005	0,005
4,5	-	5	0	0	0,054	0,978	1,404	0,826	0,329	0,147	0,018	0,005	0
5	-	5,5	0	0	0,02	0,494	1,182	0,651	0,268	0,147	0,032	0,002	0
5,5	-	6	0	0	0	0,215	0,86	0,59	0,245	0,098	0,027	0,007	0
6	-	6,5	0	0	0	0,098	0,553	0,515	0,172	0,052	0,016	0	0
6,5	-	7	0	0	0	0,029	0,39	0,415	0,181	0,059	0,007	0	0
7	-	7,5	0	0	0	0,011	0,225	0,324	0,159	0,048	0,014	0	0
7,5	-	8	0	0	0	0	0,134	0,236	0,141	0,039	0,002	0	0
8	-	8,5	0	0	0	0	0,077	0,175	0,118	0,02	0,005	0,002	0
8,5	-	9	0	0	0	0	0,032	0,129	0,084	0,014	0,005	0,002	0
9	-	9,5	0	0	0	0	0,007	0,082	0,061	0,011	0,002	0	0
9,5	-	10	0	0	0	0	0	0,034	0,034	0,011	0,002	0	0
10	-	10,5	0	0	0	0	0	0,02	0,039	0,009	0	0	0
10,5	-	11	0	0	0	0	0	0,007	0,02	0,014	0,002	0	0
11	-	11,5	0	0	0	0	0	0,005	0,009	0,002	0	0	0
11,5	-	12	0	0	0	0	0	0	0,002	0,007	0	0	0
12	-	12,5	0	0	0	0	0	0	0,002	0,007	0	0	0
12,5	-	13	0	0	0	0	0	0	0	0,007	0	0	0
13	-	13,5	0	0	0	0	0	0	0	0	0	0	0
13,5	-	14	0	0	0	0	0	0	0,002	0,002	0	0	0
>=		14	0	0	0	0	0	0	0	0	0	0	0

Figure 5-5 Sea state scatter diagram

In the scatter diagram in Figure 5-5 the wave energy is concentrated around $4{,}0s \le T_p \le 14{,}0s$ and $0{,}5m \le H_s \le 4{,}0m$. The data in the colored part of the diagram is in percent and sum up to 100.

5.3 Response in regular waves

The analytic approach to calculating the oscillating forces on marine structures starts by assuming the forces to be dividable in two sub-problems; A and B

A. *The forces and moments on the body when the structure is restrained from oscillating and there are incident regular waves. The hydrodynamic loads are called wave excitation loads and composed of so-called Froude-Kriloff and diffraction forces and moments.*
B. *The forces and moments on the body when the structure is forced to oscillate with the wave excitation frequency in rigid-body motion mode. There are no incident waves. The hydrodynamic loads are identified as added mass, damping and restoring terms.* (Faltinsen, 1990)

Due to the linearity of regular waves (one wave does not interact with the other) these contributions can be added to give the total hydrodynamic forces. Another advantage with linear theory is that the structure's motion will be proportional to the oscillating forces and therefore also the incident wave.

5.3.1 Sub-problem A

Sub-problem A, exciting forces, describes the forces acting on a structure when it is denied from moving. The sub-problem is a combination of two contributions. The first contribution is a result of the dynamic pressure field the incident wave induces on the body. The second contribution is found by demanding no fluid flow through the body surface. This is done by setting up a velocity potential at the body surface identical to that of the incident wave, but in the opposite direction. This results in zero flux through the surface, which physicality demands (Faltinsen, 1990).

5.3.1.1 *Froude-Kriloff force*

The first contribution of sub-problem A is called a Froude-Krylov force, and is derived from the pressure beneath an undisturbed wave. For a circular cylinder we find the Froude-Kriloff force by integrating dynamic pressure over an infinitesimal cylinder height of dz, and assuming the wave length to be large compared to the diameter:

$$dF_{Froude-Krylov} = \rho\pi \frac{D^2}{4} a_1\big|_{x=0} dz$$

Equation 5-19 (Faltinsen, 1990)

5.3.1.2 *Diffraction force*

The second contribution is called the diffraction force, is the result of introducing the physicality of the solid cylinder. For a circular cylinder, by demanding zero fluid transport through the cylinder wall, we derive the following diffraction force:

$$dF_{Diffraction} = A_{11}\, a_1\big|_{x=0}$$

Equation 5-20 (Faltinsen, 1990)

Where A_{11} is the proper added mass coefficient.

5.3.2 Sub-problem B

Sub-problem B consists of added mass, damping and restoring forces and moments. These forces are due to rigid body motion in still water (no wave). The added mass and damping is due to the pressure field the body's motion induces in the surrounding fluid. This pressure field is then integrated to find the forces and moments acting (Faltinsen, 1990).

$$F_k = -A_{kj}\frac{d^2\eta_j}{dt^2} - B_{kj}\frac{d\eta_j}{dt}$$

Equation 5-21 (Faltinsen, 1990)

There are a total of 36 added mass and damping coefficients, though for symmetric bodies most of them are zero. The restoring forces and moments are due to the body's displacement from the equilibrium position, and re given by:

$$F_k = -C_{kj}\eta_j$$

Equation 5-22 (Faltinsen, 1990)

5.3.3 Buoyancy

The buoyancy force acting on the body is due to a pressure difference in the gravitational direction. The force is calculated by integrating the static pressure over the wetted surface of a submerged body. For surface piercing bodies, the force is dynamic because of its varying wetted surface.

5.3.4 Response Amplitude Operator (RAO)

The RAO gives us the response of a floating structure as function of wave height and period. The response is usually a fraction of the amplitude of the incoming wave, with the same period, but with a faze angle. It may be derived by analyzing a hydrodynamic computational model, a scaled model or empirically. The RAO is different for every degree of freedom.

$$\eta_i(t) = \zeta_A f_i(\omega)\sin(k\vec{r} - \omega t + \delta_i) \qquad ; \qquad \text{i=1,2,3}$$

Equation 5-23 (DNV, 2007)

Rotational degrees of freedom are usually given as degree per degree. This means number of degrees the structure is to rotate when given the number of degrees the wave is inclined.

$$\eta_i(t) = \zeta_A f_i(\omega)k\sin(k\vec{r} - \omega t + \delta_i) \qquad ; \qquad \text{i=4,5,6}$$

Equation 5-24 (DNV, 2007)

This way it is possible to easily find the boundary conditions at the rig end of the riser.

5.4 Experimental forces – Morison's equation

For slender structures, like risers, the complexity of chapter 5.3 is reduced to a far simpler expression. The most commonly used equation for calculating the forces exerted on slender cylinders, is known as Morrison's equation, and was first published in 1950.

$$dF_{Morrison} = dF_{Drag} + dF_{Mass} = \frac{\rho}{2} C_D D u \left| u \right| dz + \rho \pi \frac{D}{4} C_M a_1 dz$$

Equation 5-25 (DNV, 2007)

The equation states that the force exerted by the fluid on the cylinder can be decomposed into one part proportional to the acceleration and one proportional to the square of the velocity of the incoming flow. It is valid provided velocity and acceleration gradients are small across the diameter, limiting at $\lambda > 5D$.The equation can be modified for moving cylinders, by introducing relative velocities and accelerations between oscillating cylinder and undisturbed flow.

$$dF_{Morrison} = \frac{1}{2} \rho C_D D (u - \dot{\eta}_1) dz \left| u - \dot{\eta}_1 \right| + \rho C_M \pi \frac{D^2}{4} a_1 dz - \rho (C_M - 1) \pi \frac{D^2}{4} \ddot{\eta} dz$$

Equation 5-26 (DNV, 2007)

In the equations above, the cylinder is assumed with the z-axis through its center and the inbound flow in positive x-direction. When this is not true, velocities and forces have to be decomposed. Decomposition is usually done by Euler angles.

$$dF_n = \rho \pi \frac{D^2}{4} C_{Mn} a_n ds + \frac{\rho}{2} C_{Dn} u_n \left| u_n \right| ds$$

Equation 5-27 (DNV, 2007)

$$dF_t = \rho \pi \frac{D^2}{4} C_{Mt} a_t ds + \frac{\rho}{2} C_{Dt} v^2 ds$$

Equation 5-28 (DNV, 2007)

5.4.1 Coefficients

Slender cylinders are normally categorized as mass dominated. This means that the force component in the Morison equation proportional to the acceleration is large compared to the component proportional to the velocity. This is shown in the figure below.

h/λ						
		Water depth, h				
		75	150	300	350	500
Wave period, T	8	0.8	1.5	3.0	3.5	5.0
	12	0.3	0.7	1.3	1.6	2.2
	16	0.2	0.4	0.8	0.9	1.3

H/D						
		Pipe outer diameter, D [inch]				
		5	8	15	21	26
Wave height, H	1	7.9	4.9	2.6	1.9	1.5
	3	23.6	14.8	7.9	5.6	4.5
	5	39.4	24.6	13.1	9.4	7.6
	7	55.1	34.4	18.4	13.1	10.6

H/D						
		Pipe outer diameter, D [mm]				
		127	203	381	533	660
Wave height, H	1	7.9	4.9	2.6	1.9	1.5
	3	23.6	14.8	7.9	5.6	4.5
	5	39.4	24.6	13.1	9.4	7.6
	7	55.1	34.4	18.4	13.1	10.6

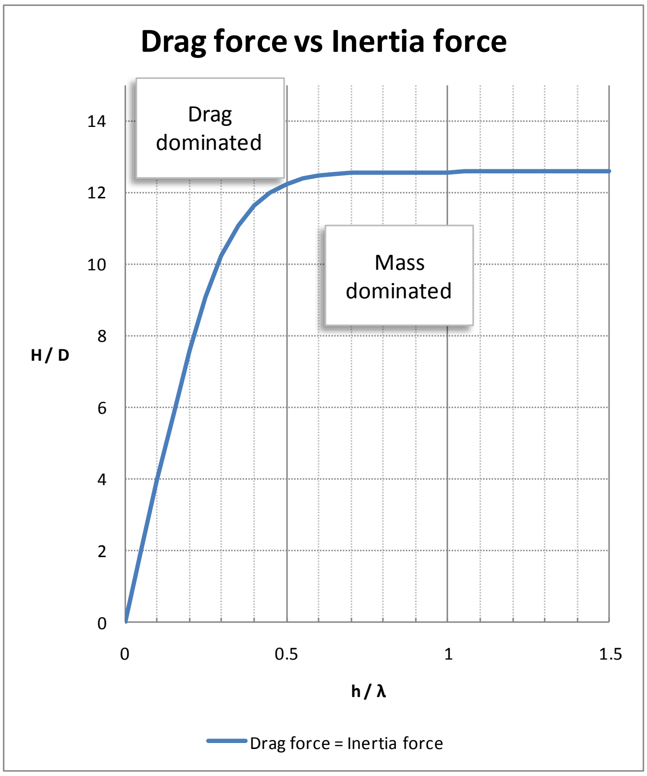

Figure 5-6 Drag and mass force dominated areas

Estimating the drag and added mass coefficients for use in the Morison equation, is done by studying experimental data, and is often a subject to uncertainties. These coefficients may be strongly dependent on Reynolds number R_N , Kuelegan-Carpenter number (*KC)* and the roughness height (*k*). Defining

- $$R_N = \frac{vD}{\nu}$$
- $$K_C = \frac{v_m T}{D}$$
- $$\Delta = \frac{k}{D}$$

Equation 5-29 (DNV, 2007)

The coefficients may also vary greatly with the distance from the free surface to the object. An educated guess will normally give satisfactory results. Usually one uses conservative values for the coefficients. It may be counter intuitive, but for the bottom part of the riser, where the wave forces have decayed, it is conservative to use a lower number for the coefficients. This is because this part of the riser is motion governed, and the added mass and damping forces represent damping.

5.4.2 Drag

As mentioned earlier, the drag coefficient is strongly dependent of the Reynolds number R_N. The flow domain is usually separated into four; subcritical, critical, super critical and trans-critical (also post critical is used about the area super through trans-critical), critical being the lowest point of the $C_D - R_N$ curves in Figure 5-7. The figure also shows the drag coefficient's dependency of the pipe's roughness height k. We see that higher roughness means shifting the critical area down the R_N -axis, which is natural because turbulence will occur earlier. It also means an asymptotic higher resistance for high R_N, which is natural.

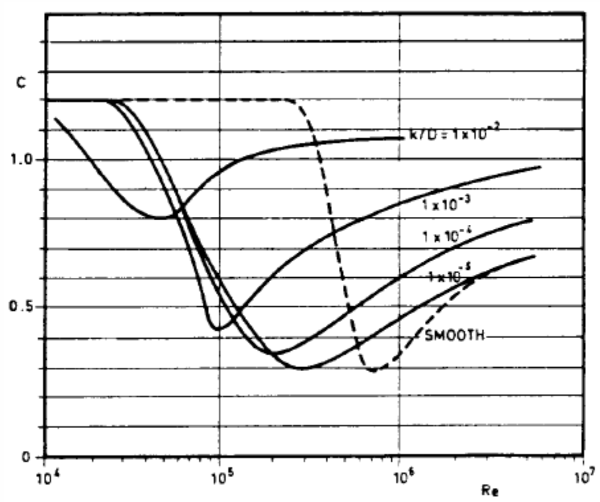

Figure 5-7 Drag dependency of roughness height (DNV, 2007)

The roughness heights for different materials are given in below.

Material	k [m]
Steel, uncoated	5×10^{-5}
Steel, painted	5×10^{-6}
Steel, highly corroded	3×10^{-3}
Concrete	3×10^{-3}
Marine growth	$5\times10^{-3} - 5\times10^{-2}$

Table 5-1 Typical roughness heights (DNV, 2007)

Uncoated and painted steel risers may be considered as smooth, and to get conservative results, we should choose a drag coefficient between the asymptotes $C_D = 1,2$ and $C_D = 0,7$. Marine growth does not only affect the surface roughness of the cylinder, but should also be considered when choosing the effective pipe diameter.

The Keulegan-Carpenter number (KC) $0,65 \le C_D \le 1,05$.

5.4.3 Added mass

The added mass coefficient is strongly dependent on the distance to the free surface, the roughness height k, and the Keulegan-Carpenter number K_C. By collecting experimental data for tests done for a rough and smooth cylinder deeply submerged, it is found that for high K_C (>3), the added mass coefficient can be described by

$$C_A = \max\begin{Bmatrix} 1.0 - 0.044(K_C - 3) \\ 0.6 - (C_{DS} - 0.65) \end{Bmatrix}$$

Equation 5-30

Where C_{DS} is a function of K_C ($C_{DS_{rough}} = 1,05$ and $C_{DS_{smooth}} = 0,65$) that can be found in the appendix. The asymptotic values of C_A is 0.6 for smooth and 0.2 for rough cylinder.

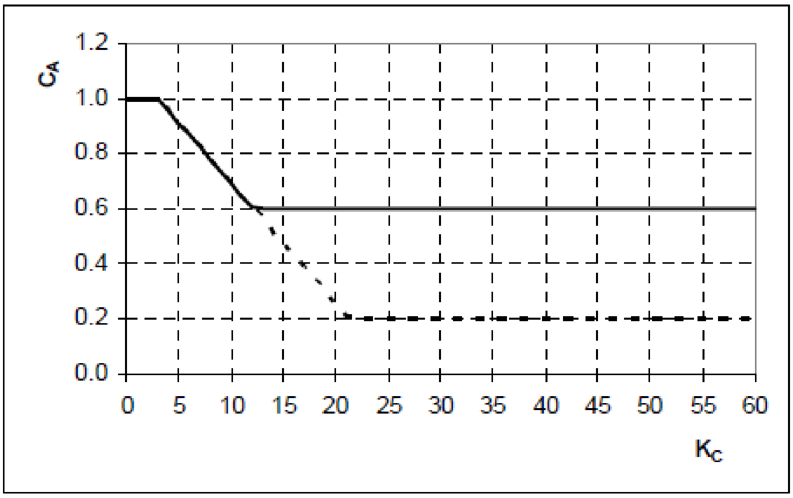

Figure 5-8 Added mass dependency of KC (DNV, 2007)

For elements closer to the surface, C_A may vary greatly.

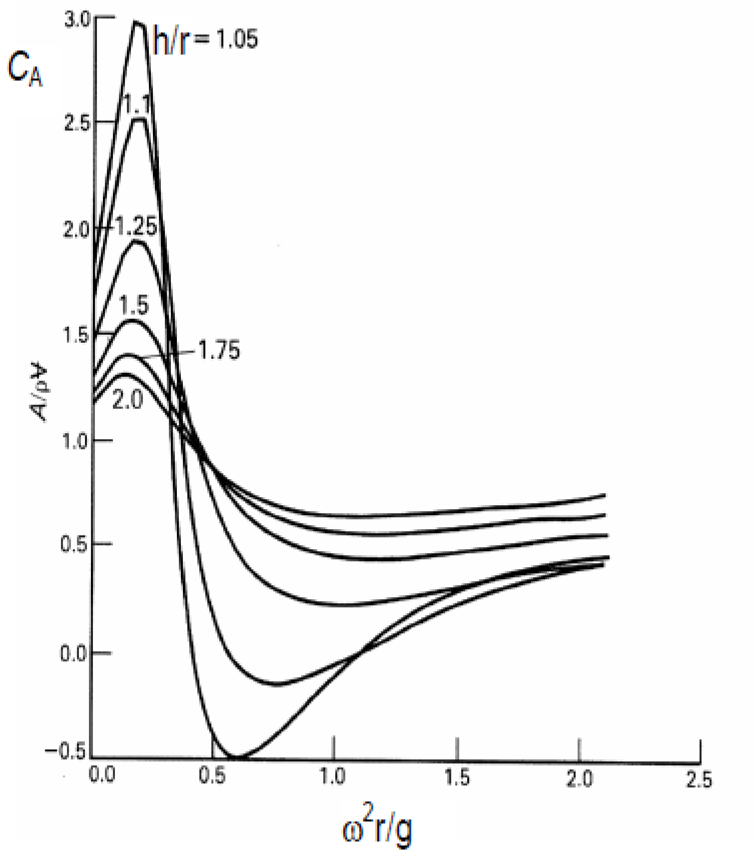

Figure 5-9 Added mass dependency of distance to surface (DNV, 2007).

5.4.4 Geometrical challenges

As explained in chapter 2.4, controlling well pressure and BOPs is done by kill/choke and booster lines. These lines are normally attached to the outside of the joints, so the hydrodynamic cross section deviates from a circle. On a marine riser the only parts that can be considered to have a circular cross section are the buoyancy joints, because the control lines are covered by the buoyancy elements. There are three important changes when considering a cross section with auxiliary lines;

1. The inertia force is changed: $\rho A a = \rho(D_1^2 + D_2^2 + D_3^2 + D_4^2)a$
2. The added mass force is changed: $\rho C_{A,nom} A a_r = \rho C_{A,nom} \frac{\pi}{4}(D_1^2 + D_2^2 + D_3^2 + D_4^2)a_r$
3. The drag force is changed: $\frac{1}{2}\rho C_{D,nom} D v_r |v_r| = \frac{1}{2}\rho C_{D,nom}(D_1 + D_2 + D_3)$

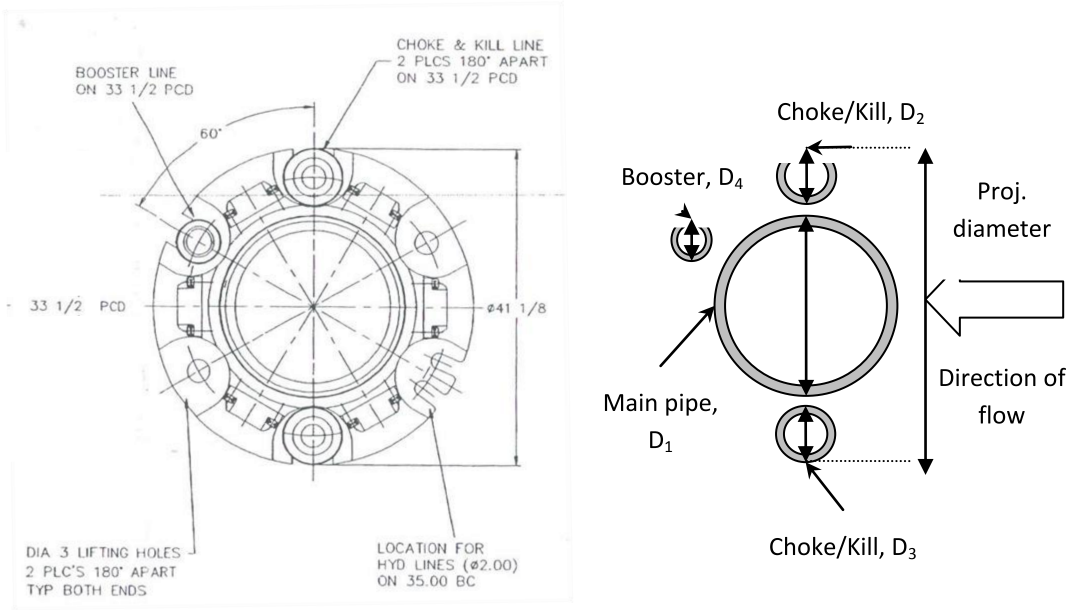

Figure 5-10 Equivalent hydrodynamic diameter (Marré, 2009)

Lines hidden in the flow direction, as the booster line in the situation above, are not considered to contribute to the drag force. For computational use, it is normal to define a hydrodynamic diameter, found by summing the pipe areas;

$$D_h = \sqrt{D_1^2 + D_2^2 + D_3^2 + D_4^2}$$

Equation 5-31 (Marré, 2009)

When the diameter is changed, the added mass and drag coefficients should be corrected;

$$C_{mn} = \frac{C_{A,nom}(D_1^2 + D_2^2 + D_3^2 + D_4^2)}{D_h^2} = \frac{C_{A,nom}(D_1^2 + D_2^2 + D_3^2 + D_4^2)}{\left(\sqrt{D_1^2 + D_2^2 + D_3^2 + D_4^2}\right)^2} = C_{A,nom}$$

Equation 5-32 (Marré, 2009)

$$C_{dn} = \frac{C_{D,nom}(D_1 + D_2 + D_3)}{D_h} = \frac{C_{D,nom}(D_1 + D_2 + D_3)}{\sqrt{D_1^2 + D_2^2 + D_3^2 + D_4^2}}$$

Equation 5-33 (Marré, 2009)

Other components are also represented by a hydrodynamic diameter, such as BOP, XT and LMRP.

6 Fluid systems

6.1 Introduction

To get a better understanding of the tensioner system and the computational modeling of it (see chapter 9), a brief introduction to theory of fluid systems is given. The study of fluid systems can be divided into *hydraulics* and *pneumatics* (Palm, 2005). Hydraulic systems transfer force or movement by a limited amount of liquid, while pneumatic systems have gas as work medium (Framnes & Gleditsch, 1994).

Hydraulic and pneumatic systems have many favorable characteristics, as they, if used in the right way, can eliminate the need for complicated systems of gears, cams and levers (Introduction to Fluid Power, 2005). Forces and movements can be transferred around corners and over relative large distances with small loss in efficiency. Aboard any drill rig, there are various systems, both hydraulic and pneumatic. The main difference between them is that liquids are considered incompressible, while gases are not. An exact rule for when to use liquid and when to use gas does not exist, but from experience the following rule of thumb has been derived.

"If the application requires speed, a medium amount of pressure, and only fairly accurate control, a pneumatic system may be used. If the application requires only a medium amount of pressure and a more accurate control, a combination of hydraulics and pneumatics may be used. If the application requires a great amount of pressure and/or extremely accurate control, a hydraulic system should be used." (Introduction to Fluid Power, 2005)

For this reason a tensioner system which requires both great pressure and accuracy is governed by hydraulics. The heave compensator system on the other hand, requires a lot lower pressure, and is therefore usually governed by pneumatics.

6.2 Theory

Although hydraulics and pneumatics may differ in areas of application, the fundamental principles used to model them are the same. They are modeled by the basic principles of *fluid systems*; *conservation of energy, conservation of mass* and *conservation of momentum.* As liquids are considered incompressible, their equations differ to some extent from those of gases. Models can be more or less complex depending on the level of accuracy needed, and may consider both losses due to friction, leakage, condensation and heat transfer to surroundings and temperature variations. The latter is part of *thermal systems theory*, and will not be treated in this thesis.

6.2.1 Governing equations

When studying a control volume, Reynolds' transport theorem can be applied to find the rate of change of an arbitrary gross fluid property. It can be applied to mass, momentum and energy to derive the conservation laws for these properties. Reynolds' theorem is given as

$$\frac{d}{dt}(B_{syst}) = \frac{d}{dt}\left(\int_{CV} \beta\rho dV\right) + \int_{CS} \beta\rho(\vec{v}\bullet\vec{n})dA$$

Equation 6-1 Reynolds' transport theorem (White, 2003)

Where B_{syst} is a fluid property and $\beta = dB / dm$. It states that the rate of change of a fluid property of a system is given by the rate of change of the fluid inside the control volume plus the sum of the fluxes in and out.

- Applied to the system's mass, we can find the equation for *conservation of fluid mass*.

$$\frac{d}{dt}(m)_{syst} = \frac{d}{dt}\left(\int_{CV} \frac{dm}{dm}\rho dV\right) + \int_{CS} \frac{dm}{dm}\rho(\vec{v} \bullet \vec{n})dA$$

$$0 = \frac{d}{dt}\left(\int_{CV} \rho dV\right) + \int_{CS} \rho(\vec{v} \bullet \vec{n})dA$$

$$\frac{d}{dt}(\rho V) = \dot{m}_{in} - \dot{m}_{out}$$

Equation 6-2 Conservation of mass (White, 2003)

The rate of change of mass is a fluid volume is the sum of outgoing and incoming flow.

- In the same manner we derive the equation for *conservation of fluid momentum*

$$\frac{d}{dt}(m\vec{v})_{syst} = \frac{d}{dt}\left(\int_{CV} \vec{v}\rho dV\right) + \int_{CS} \vec{v}\rho(\vec{v} \bullet \vec{n})dA$$

Equation 6-3 Conservation of momentum (White, 2003)

$m\vec{v}$ is the linear momentum of the system. The principle of conservation of fluid momentum is an application of Newton's law of motion to a fluid volume.

- By applying Reynolds' transport theorem to the fluid energy, we may derive the equation of *conservation of fluid energy*

$$\frac{dQ}{dt} - \frac{dW}{dt} = \frac{d}{dt}(E)_{syst} = \frac{d}{dt}\left(\int_{CV} e\rho dV\right) + \int_{CS}\left(e + \frac{p}{\rho}\right)\rho(\vec{v} \bullet \vec{n})dA$$

Equation 6-4 Conservation of energy (White, 2003)

E is internal energy, Q is the heat transferred to the system, and W the work it performs.

These are the conservation equations in their most basic forms. When applied to fixed volumes and incompressible fluids, they simplify a lot. When considering compressible fluids, the ideal gas law is also needed.

$$p = Z\rho RT$$

Equation 6-5 Ideal gas law (White, 2003)

For an ideal gas Z is set to unity, otherwise it may be set to experimental values.

6.2.2 Resistance, capacitance and inertance

It may be convenient to draw analogies between fluid systems and electric systems. Loss and resistance are important factors to include when studying fluid systems. Fluid meets *resistance* when flowing through conduit such as a pipe, through a component such as a valve, or even a simple opening or orifice, such as a whole (Palm, 2005). A fluid resistance is related to the mass flow through it and the pressure drop across is.

Fluid quantity	Mass	Mass flux	Pressure	Fluid resistance	Fluid capacitance	Inertance
	m	$\dot{m}$	p	$R = p/\dot{m}$	$C = m/p$	$I = p/(d\dot{m}/dt)$
Electric quantity	Charge	Current	Voltage	Electric resistance	Electric Capacitance	Electrical inductance
	Q	i	v	$R = v/i$	$C = Q/v$	$L = v/(di/dt)$

Table 6-1 Electric and fluid analogy (Palm, 2005)

Fluid *capacitance* is the relation between *pressure* and *stored mass*, and gives the size of the potential energy. Pneumatic capacitance differs from hydraulic capacitance in the way that it is dependent on fluid properties and temperature. The fluid *inertance* relates to the fluid acceleration and kinetic energy. In systems of steady state we do not regard fluid inertance.

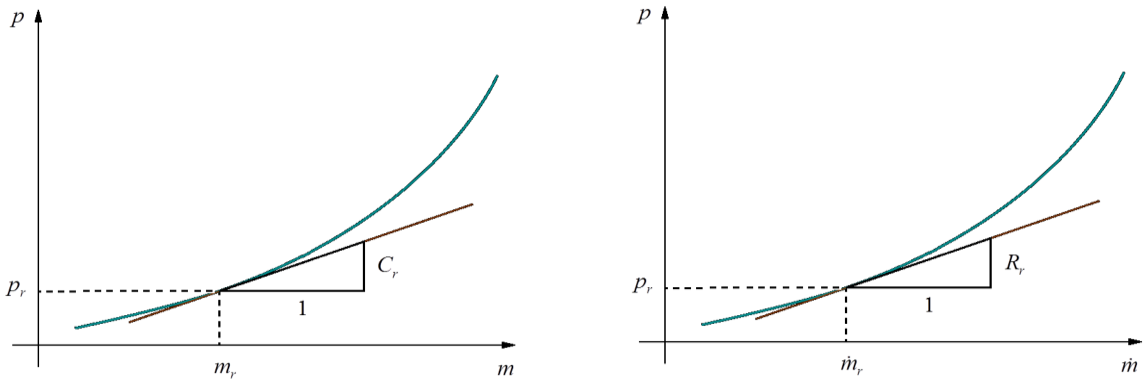

Figure 6-1 Fluid capacitance and resistance (Palm, 2005)

By drawing an electric analogy, the laws of *continuity* and *compatibility* are equivalent to Kirchoff's laws of current and voltage.

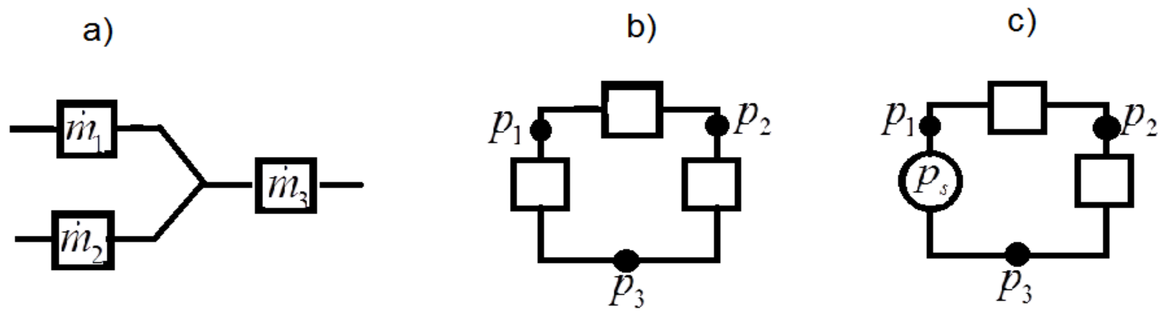

Figure 6-2 Kirchoff's current (a) and voltage (b and c) laws applied to fluids (Palm, 2005)

The current law (Figure 6-2-a) in fluid terms states that the mass is conserved;

$$a: \quad \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Equation 6-6 (Palm, 2005)

While the current law (Figure 6-2-b and c) in fluid terms states that the pressure differences around a closed circuit is zero.

$$b: \quad (p_1 - p_2) + (p_2 - p_3) + (p_3 - p_1) = 0$$

$$c: \quad (p_1 - p_2) + (p_2 - p_3) - p_s = 0$$

Equation 6-7 (Palm, 2005)

The fluid flow may be considered either *laminar* or *turbulent.* Laminar flow is described as "smooth", as the particle velocity is the same size as the average fluid velocity. Turbulent flow is described as "rough", as the average fluid velocity is less than the actual particle velocity. The higher velocity in the turbulent flow is a result of particle meandering. The practical importance laminar and turbulent flow, is that laminar flow resistance may be described by a linear relation, while turbulent resistance is described by a nonlinear relation (Palm, 2005).

$$\dot{m}_{\text{laminar}} = \frac{p}{R} \qquad \dot{m}_{\text{tubulent}} = \sqrt{\frac{p}{R}}$$

Figure 6-3 Laminar and turbulent resistance (Palm, 2005)

The resistance parameter R of turbulent flow is usually found empirically.

7 Model description

7.1 Introduction

The main elements of subsea exploration and production have been presented. The governing theory and methods used to analyze and model the problems encountered have also been introduced. This chapter is going to give an overlook of the main elements the previous chapters and clarify the core essence of what we want to accomplish.

The main goal of the theses is to develop an analysis tool capable of coupling the global dynamics of marine structures with the fluid mechanics of tensioners and heave compensators. A coupled model introduces a new aspect of complexity that has not yet been modeled, and is highly desired by the industry. Existing marine structure programs go no further than to model the tension force as a tabled function of displacement or velocity. On the other side, existing fluid system programs have no option for more advanced modeling of the marine riser dynamics than simple spring-mass-damper model.

7.2 Model

MSC Adams is multi-body dynamics program that is capable of modeling the body-force interaction of a marine riser. MSC Easy5 is an analysis tool for fluid system dynamics that is compatible with MSC Adams. By modeling the tensioner system in Easy5 and the marine riser in Adams, we may create a coupled model. One of the greatest challenges is implementing hydrodynamic forces in Adams. This is done by the wave force subroutine (chapter 8.3).

The communication between the two programs is done by passing the displacement and velocity of the marine riser from Adams to the model of the tensioner system in Easy5. The resulting force of the tensioner system is then calculated in Easy5, and passed back to Adams. The dynamics of the platform is given by a response of amplitude (RAO) table. Inducing the right rig motion is vital to accomplish the correct relative displacement and velocity of the riser and rig, as these are the parameters passed to the hydraulic tensioner model.

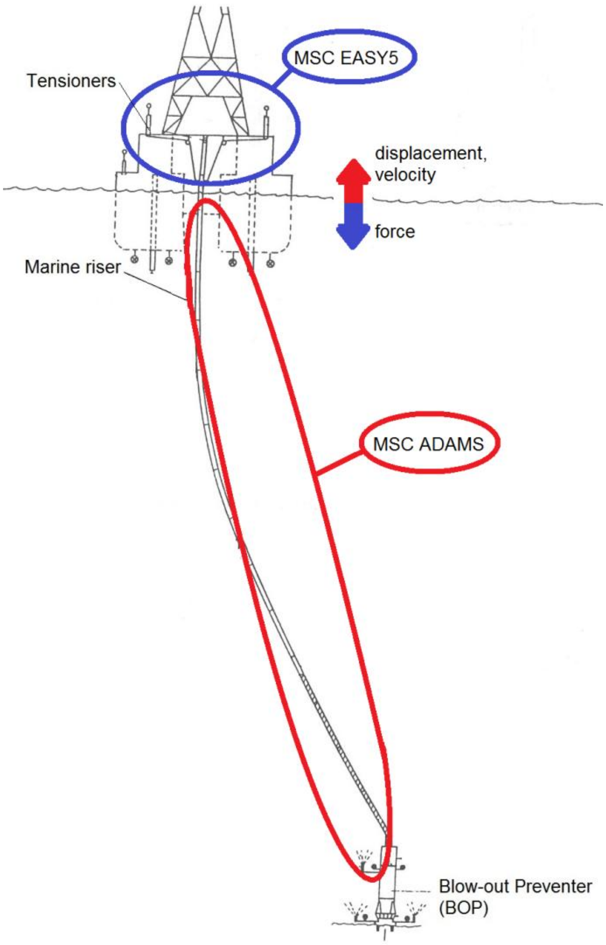

Figure 7-1 Coupling models

8 MSC Adams

8.1 Why Adams?

The existing tools for riser analysis are well suited for analyzing the global dynamics of marine structures, but lack complexity when implementing hydraulic and pneumatic systems. MSC Adams is a multi-body dynamics program, developed and utilized for the automobile industry. Adams makes it possible to create complex models like suspension, steering, and breaking systems of automobiles, and with the coupling to Easy5, include hydraulics, pneumatics and electric motors and control systems. The program's algorithm for calculating contact forces is also very accurate, so a pipe-in-pipe situation would be treated with more accuracy than in existing tools. The possibility to model a complete marine riser and include the complexity of a tensioner system or a heave compensator system, may offer deeper insight and more accurate results.

8.2 Challenges with Adams

One of the biggest challenges when applying MSC Adams to analyze a body's motion in liquid is the fault of a hydrodynamic force module. Another problem is that the interface is designed to make it easy to build models of systems consisting of many different parts, like i.e. a car suspension. A typical riser model consists of multiple elements of very similar geometry and material properties.

8.3 Wave force subroutine

Integrating hydrodynamic forces in Adams is made possible by manipulation of the internal general force command (G-Force). The general force is a point force with six degrees of freedom, and the force vector may be given either as a function or as a subroutine. The subroutine option was utilized to calculate the hydrodynamic forces.

For each time step the subroutine is called, calculating the Morrison force, gravity force and buoyancy force on each element according to the given parameters. To calculate the Morison forces as accurately as possible, the effects of relative velocity and acceleration has also been considered. The wave force subroutine calls a built-in Adams subroutine (MOTSUB) to return the displacements and displacement derivatives to be used in the Morison equation. As an element may be oriented or moving in any direction, the velocities and accelerations are decomposed by Euler decomposition. To avoid shocking the structure into violent vibrations, the forces are applied with increasing magnitude over the period defined by the input variable "RAMP". This is done by a linear ramping function.

Close to the surface, the magnitude of the wave force is calculated at the center of projected area of the given element, and multiplied by the fraction of the wetted surface. The surface elevation is calculated by "Wheeler-stretching" (see chapter 5).

8.4 4subsea interface

As mentioned above, one of the problems with Adams is the lack of an option to rapidly create multiple parts. The tailor made 4subsea interface is the solution to this problem, and makes it possible to swiftly create a riser model of a suitable number of elements. Material and cross section properties can also be entered as suited. The wave forces, exerted at the center of mass of each element, may be created all at once by utilizing the “G-force” menu in the 4ubsea interface. This way the modeler will not have to apply the force on each and every element of the model.

8.5 RAO motion subroutine

The RAO, as seen in chapter 5, gives the relation between the wave’s motion and the structure’s motion. The RAO motion subroutine gives the part in Adams representing the platform deck the right motion for a given wave. This is done by manipulating a general point motion, which has six degrees of freedom. The subroutine has wave height, period and degree of freedom (X=1, Y=2, …rotZ=6) as inputs. It reads the right values from a text file, and returns the motion. If the wave period passed to the RAO motion subroutine does not exist in the text file, the value is found by linear interpolation of the closest values. If the value is out of the domain of the text file, the value is sat to zero, and the platform deck will stand still.

8.6 Solvers

Flexible parts in Adams are modeled by Timoshenko 3-D beam theory. Each element has 12 degrees of freedom (DOF) when including rotation and displacement at each end node. Every beam element has a corresponding differential equation and boundary conditions to satisfy. Obtaining an exact solution of these differential equations is time consuming and in many cases impossible and computational methods are therefore widely used. We distinguish between numerical integration methods and numerical differentiation methods. Mutual for the methods is that they stepwise approximate the solution of the differential equations, where the initial unknowns are replaced by simpler expressions.

$$M\ddot{r} + C\dot{r} + Kr = F(t)$$

Equation 8-1 (Langen & Sigbjørnsson, 1986)

8.6.1 Numerical differentiation methods

In the numerical differentiation methods, the derivatives in the dynamic equilibrium equation are replaced by a difference expression of the required order. The most common methods in this category are the second central difference method and Houbolt’s method.

The *central difference method* is given as

$$(m + \frac{h}{2}c)u_{k+1} = h^2 Q_k + (2m - h^2 k)u_k - (m - \frac{h}{2}c)u_{k-1}$$

Equation 8-2 (Langen & Sigbjørnsson, 1986)

With starting conditions

$$u_1 = u_0 + h\dot{u}_0 + \frac{h^2}{2m}(Q_0 - c\dot{u}_0 - ku_0)$$

Equation 8-3 (Langen & Sigbjørnsson, 1986)

As we can see it is an explicit method, as the next time step is calculated on behalf of previous time steps only. The method suffers instability for large time steps, h.

$$h_{cr} \leq \frac{T}{\pi}$$

Equation 8-4 (Langen & Sigbjørnsson, 1986)

Houbolt's method, on the other hand, is unconditionally stable and will not diverge for any step length. It is subject to an artificial damping, which means that if applied to a free undamped oscillation, the amplitude of oscillation will slowly decay. Houbolt's method is implicit; the next time step may be a function containing values from both previous and next time step. It is defined as follows

$$(h^2k + \frac{11}{6}hc + 2m)u_{k+1} = h^2Q_{k+1} + (3hc + 5m)u_k - (\frac{3}{2}hc + 4m)u_{k-1} + (\frac{1}{3}hc + m)u_{k+1}$$

Equation 8-5 (Langen & Sigbjørnsson, 1986)

8.6.2 Numerical integration methods

In numerical integration methods, displacements are found by integrating the acceleration twice. The expression for the acceleration is an assumed function depending on the method. The Euler method assumes constant acceleration, but can be improved by assuming constant average acceleration over two neighboring points. Time varying acceleration methods are also common in dynamic analysis, mostly are Runge-Kutta and Newmark's β -family applied. Runge-Kutta methods can be either implicit or explicit, where only the implicit method is unconditionally stable. The *Runge-Kutta-4 method* is stated below

$$\dot{u}_{k+1} = \dot{u}_k + \frac{h}{6}(a_1 + 2a_2 + 2a_3 + a_4)$$
$$u_{k+1} = u_k + \frac{h}{6}(b_1 + 2b_2 + 2b_2 + b_4)$$

Equation 8-6 (Langen & Sigbjørnsson, 1986)

Where a_i and b_i corresponds to velocities and accelerations at the beginning middle and end of a time-step (Langen & Sigbjørnsson, 1986). Analysis of cases with time-varying acceleration is preferably done with the *Newmark-* β *method*. The method is conditionally stable, which in special cases reduce to some of the methods previously discussed.

$$\dot{u}_{k+1} = \dot{u}_k + (1+\lambda)h\ddot{u}_k + \lambda h\ddot{u}_{k+1}$$
$$u_{k+1} = u_k + h\dot{u}_k + (\frac{1}{2} - \beta)h^2\ddot{u}_k + \beta h^2\ddot{u}_{k+1}$$

Equation 8-7 (Langen & Sigbjørnsson, 1986)

These equations are obtained by Taylor expanding the accelerations, velocities and displacements. The values of λ and β are important for the stability and damping;

Stability: $\lambda \geq \frac{1}{2}$ and $\beta \geq \frac{1}{4}(\lambda + \frac{1}{2})^2$

Damping: $\lambda < \frac{1}{2}$ gives artificial damping, while $\lambda = \frac{1}{2}$ gives no artificial damping.

For this reason it is common to set $\lambda = \frac{1}{2}$.

The Newmark-β method is used by RIFLEX and Orcaflex, and is therefore also the preferable method in Adams.

9 MSC Easy5

9.1 Introduction

MSC Easy5 is a powerful controls and systems simulation tool with the ability to model and analyze complex fluid systems. Subsystems from different domains, as electrical, mechanical and fluid mechanical, may be combined. As traditional building and testing of scaled models and prototypes become increasingly expensive, both time and costs can be saved. Easy5 has the joint potential of five libraries, containing hydraulics, pneumatics, electronics, multi-phase fluids, computer signals and fuel cells (MSC Software, 2010). At the same time connections to other MSC programs, as to the multi-body dynamics program MSC Adams, may be created effortlessly. The automobile industry has made use of the powerful combination for many years already, and it is time to adapt it for the offshore industry.

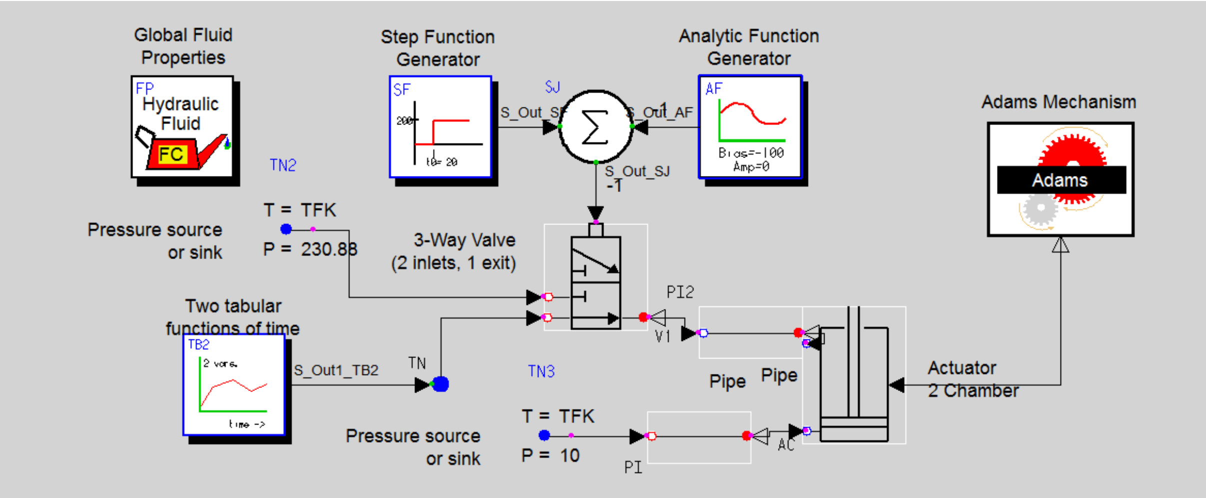

Figure 9-1 Example of Easy5 model

9.2 Libraries

The components in Easy5 are sorted by their domain into libraries. Components which model an i.e. an electric motor, belongs to the electronics library. Components from the electronics library do not automatically connect with any component from i.e. the gas dynamics library, like a pipe. Some components are inter-connectable between the different libraries, like an electric motor and a hydraulic pump. It may seem counter-intuitive that the components of the gas dynamics library do not automatically connect with those of the thermal hydraulics library until it is understood that the governing theory of the two are different (see chapter 6). When connecting components, the program sets up differential and algebraic equations between them. Therefore the output of one component cannot be a compressible gas, while the input to the next is an incompressible fluid. The equations and therefore the communication between the components fail. A tensioner system would for the most part consist of components from four libraries; *General Purpose (GP), Gas Dynamics (GD), Thermal Hydraulics (HC) and External Interface (XI).*

9.2.1 General Purpose (GP)

The general purpose library contains integrators, sum functions, logic ports, signal constructors, controllers, switches and many more. These components are connectable to most components in the other libraries, as their input and output (I/O) are only signals. It also offers components with options to write FORTRAN and C code with multiple inputs and multiple outputs to create user defined components (MSC Software, 2010).

9.2.2 Gas Dynamics (GD)

The gas dynamics library contains all components designed to model compressible fluid systems and a database of the 30 most common engineering gases. The components can be used to model a variety of devices such as heat exchangers, ducts, valves, orifices, actuators, compressors, turbines, pumps, and fans. Both ideal and real gases may be modeled, with humidity and condensation effects considered. The system can either be steady-state or transient. Behind the transient model lie transient forms of the energy and species mass conservation equations (MSC Software, 2010).

9.2.3 Thermal hydraulics (HC)

The thermal hydraulics library consists of all the components used to model incompressible fluid flow of any kind. The components are designed to model both transient and steady-state behavior of hydraulic/mechanic systems, including effects as energy loss due to friction and heat transfer to surroundings. The property database includes 20 common liquids such as water, SAE Oil and gasoline. The fluid properties are expressed as analytical functions of fluid pressure and temperature (MSC Software, 2010).

9.2.4 Extensions (XI)

The extensions library contains the component that enables the communication with Adams. It allows information to flow between the programs during analysis.

9.3 Components

9.3.1 Connecting components

Connecting components in Easy5 is simple. It is done by clicking on the preferred pin (port) on the upstream component, before selecting the preferred pin (port) downstream. As mentioned above, not all components in Easy5 are connectable because they belong to different libraries, and therefore have different states, variables and parameters (see chapter 9.4). This is not the only problem in connecting components as we also separate between storage and resistive ports, and upstream and downstream ports. A downstream port cannot be connected to another downstream port as they both depend on mass inflow.

Class	at In or inlet port	pin design	at Out or exit port	pin design
Resistive	P is input, w is output		P is input, w is output	
Storage	P is output, w is input		P is output, w is input	
Storage/Resistive	P is output, w is input		P is input, w is output	

Table 9-1 Classification of component (MSC Software, 2010)

As we understand from Table 9-1, a storage port has mass flux (w) as input and pressure (P) as output, while a resistive port has mass flux (w) as output and pressure (P) as input. Depending on what ports a component has, it is defined as *storage, resistant* or *storage/resistant.*

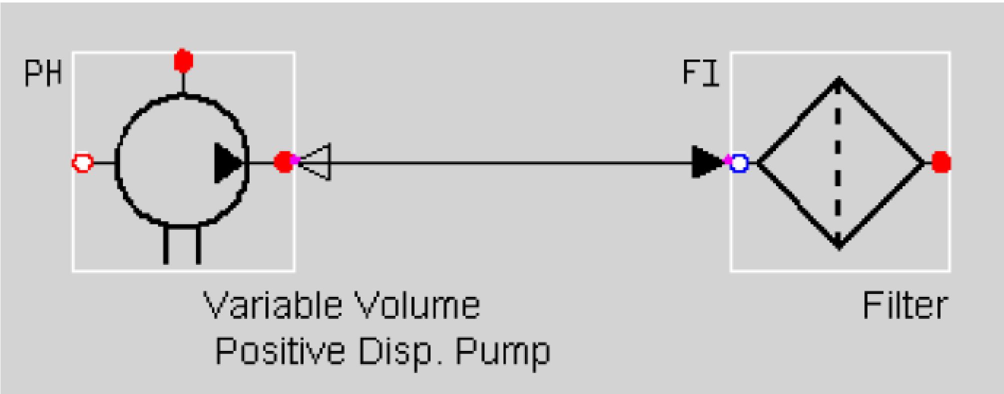

Figure 9-2 Connection between a hydraulic pump and a filter (MSC Software, 2010)

9.4 Variables, States, Parameters and Tables

The data traffic in and out of components fall into one of four groups: *Variables, states, parameters or tables.*

9.4.1 Variables

Variables are output quantities in the system model that are a function of algebraic relationships. Simply put, variables are defined by algebraic equations, and may be a function of states, variables, parameters or table data. (MSC Software, 2010)

9.4.2 States

States are variable output quantities in the system model that are functions of first order differential or difference equations. For example, a state variable occurs when position is calculated by using an integrator to integrate velocity data. The output of the integrator (position) is a state. There are four types of state variables: continuous, delay, sample (also called sample and hold states), and switch states. These states are used in many standard components and can be added to any User Code and Library components you write. (MSC Software, 2010)

9.4.3 Parameters

A parameter is an input to a component that is not connected to the output of another component. You determine what will and what will not be a parameter when connecting the components in your model. Parameters are assigned constant values before performing an analysis. These constants are derived from data associated with the system you are modeling. (MSC Software, 2010)

9.4.4 Tables

A table is a set of constant tabular data used by any of several components that produce outputs using a table look up algorithm. In general, tables are used to represent algebraic functional relationships, containing from one to nine independent variables, and they allow you to include a set of "real world" data into your model. You can also design User Code and Library components which use table look up algorithms. (MSC Software, 2010)

10 RIFLEX

10.1 Introduction

As RIFLEX is a widely used finite element program for static and dynamic analysis of slender marine structures, it is an ideal tool for benchmarking the results from the Adams analysis and verifying the custom made subroutines. It was tailor-made by MARINTEK for calculations on flexible risers, but is applicable on any kind of slender marine structure. Representing state-of-the-art technology for riser analysis suitable for flexible, metallic or steel catenary riser applications, RIFLEX is considered to simulate both regular and irregular wave response very efficiently. It is able to model a wide range of structures and environments, and is verified against both model and full-scale tests (MARINTEK, 2011).

10.2 Theory

The theory on which the RIFLEX code is based is given in the RIFLEX theory manual. A riser model in RIFLEX consists of a finite number of beam elements. The model is first solved statically, this means summing up buoyancy, mass and tension forces, excluding dynamic forces. Then, when equilibrium is reached, the hydrodynamic forces are ramped on. The finite beam element formulation is derived especially for analysis of slender flexible bodies, this means;

- *A plane section of the beam initially normal to the x-axis, remains plane and normal to the x-axis during deformations.*
- *Lateral contraction caused by axial elongation is neglected.*
- *The strains are small.*
- *Shear deformations due to lateral loading are neglected, but St. Venant torsion is accounted for.*
- *Coupling effects between torsion and bending are neglected. Thus, warping resistance and torsional stability problems are not considered.* (MARINTEK, 2011)

The waves may either be irregular or regular, where the irregular wave state is based on an Airy wave, while the regular wave may either be an Airy wave or a Stokes 5th order wave. Surface stretching may be done by three different methods, one being Wheeler stretching method. The hydrodynamic forces are calculated based on Morison's equation for oscillating slender structure in oscillating flow. Forced vessel motion is assumed to be a linear multiple of the incoming wave (RAO), and is introduced as a boundary condition in the beam element model. The tensioner force is modeled by a nonlinear spring where the spring stiffness is dependent on the displacement.

11 Case modeling

This chapter has two goals. One is showing the stepwise verification of MSC Adams as an analysis tool for slender marine structures, the other is studying the effect of having a hydraulic tensioner system. The complexity of the models is increased gradually, and each new feature is analyzed separately, for easy troubleshooting.

Every model is analyzed for three sea states ((T_A, H_A), (T_B, H_A) and (T_B, H_B)), and verified against RIFLEX. The comparison between the Adams and RIFLEX models is given in chapter 12. This is done to verify the models both when varying the period and the wave height. From the data in Figure 5-5 it would be natural to choose the wave data as follows;

	T_p	H_s	T	H	d	d/T^2	H/T^2
	Period	Sign. height	Period	Height	Depth	Shallowness	Steepness
$Wave-I$	10,0s	1,5m	10,0s	3,0m	300,0m	3,00	0,030
$Wave-II$	14,0s	1,5m	14,0s	3,0m	300,0m	1,53	0,015
$Wave-III$	14,0s	3,0m	14,0s	6,0m	300,0m	1,53	0,031

Table 11-1 Wave data

By verifying the wave steepness and shallowness parameters in Figure 5-2, it is clear that the waves chosen are within the limits of linear wave theory.

11.1 Uniform riser model

To make sure that the wave force subroutine works, a simple model is built and analyzed. The results are then compared with a similar analysis in RIFLEX. The riser is modeled as a smooth pipe cantilever beam to be as uncomplicated as possible. The riser is positively buoyant, to avoid the need for a tensioner force. The dimensions and stiffness parameters as well as wave data are arbitrary, but of sensible dimensions for such a model. To verify that the force module also considers the forces on a surface piercing structure in a correct manner, the top of the riser is set 10 meters above the mean level water surface.

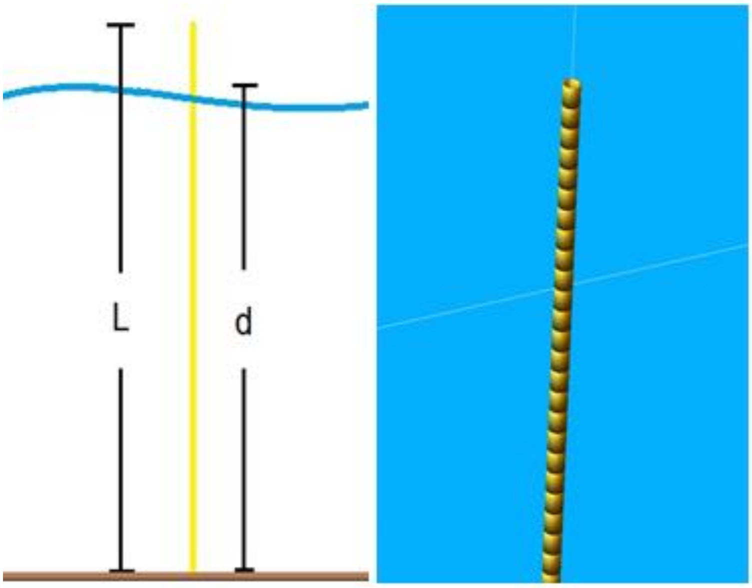

Figure 11-1 Concept drawing of uniform riser (left) and riser model in Adams (right).

The top node is free in all six degrees of freedom, while the bottom node is locked in all.

D	t	L	L_{Elem}	ξ
Diameter	Thickness	Length	Element length	Damping
$1.0m$	$0.1m$	$310.0m$	$1.0m$	0.02
I_{xx}	I_{yy}	I_{zz}	A	E
Inertia	Inertia	Inertia	Area	E-modulus
$0.02m^4$	$0.02m^4$	$0.04m^4$	$0.149m^2$	$2.07E11\,N/m^2$

Table 11-2 Structural properties uniform cantilever

$C_{D,N}$	$C_{D,A}$	$C_{M,N}$	$C_{M,A}$	d	$D_{buoyancy}$	D_{hydro}
Normal drag coef.	Axial drag coef.	Normalass coef.	Mass coef.	Depth	Buoyancy diameter	Hydrodynamic diameter
1.2	0.001	2.0	0.0	$300.0m$	$1.0m$	$1.0m$

Table 11-3 Hydrodynamic properties uniform cantilever

11.2 Marine riser model

To ensure that a more complex model consisting of joints with varying stiffness and hydrodynamic parameters does not introduce any problems or inaccuracies, a full marine riser model is built. The model has both upper and lower flex joint and a telescope joint.

All dimensions and stiffness parameters as well as wave data are identical to that of an undisclosed existing North Sea marine riser. The rig-movement is set to zero. As this riser is not positively buoyant, we have to include a tension force. The tension force is set to be constant, and oriented according to a global axis system. The force is calculated in excel by demanding an over pull at the bottom of the LMRP of 20 tons;

$$T = (m_{riser} - m_{water} + m_{overpull}) \bullet g$$

Equation 11-1

Where m_{water} is the mass of the water displaced by the riser. As the marine riser consists of a number of parts with different characteristics, a detailed table of properties will not be disclosed ere. To get an idea of the composition, a summation table is presented instead.

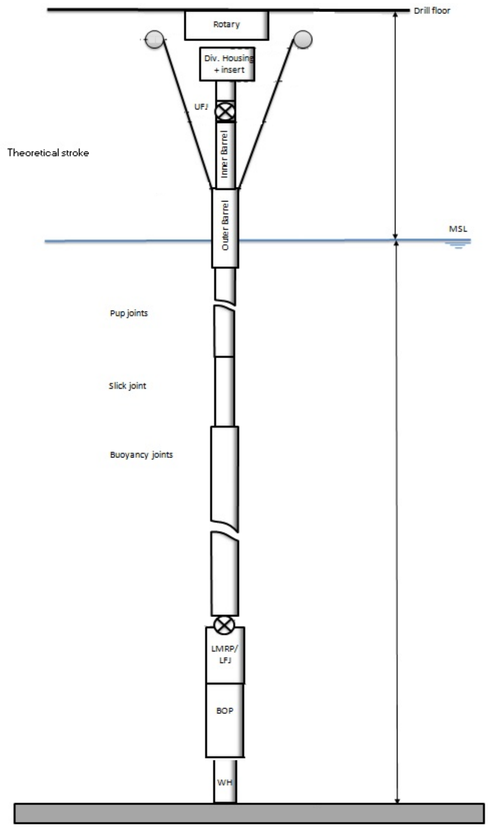

Figure 11-2 Concept drawing of marine riser

Marine riser case		Length	Weight in air (fluid filled)	Top node position	Quad drag Normal	Mass Normal	Hydrodyn diameter	Buoyancy diameter
Number of lenghts	Component	L	M	TNP	$C_{D,N}$	$C_{M,N}$	D_{hydro}	$D_{buoyancy}$
[-]		[m]	[kg]	[m]	[-]	[-]	[m]	[m]
1	WH	2,38	0,00	-297,62	0,00	1,00	0,00	0,00
1	BOP	6,87	164754,0	-290,75	2,00	2,00	4,97	2,03
1	LMRP with LFJ	4,38	37799,0	-286,38	2,00	2,00	4,97	1,28
1	Pup joint	6,10	6152,6	-280,28	1,00	1,80	0,79	0,70
1	Slick joint	22,86	13751,2	-257,42	1,00	1,80	0,79	0,56
11	Buoyancy joint	251,46	204943,0	-5,96	1,00	2,10	1,12	0,99
1	Pup joint	6,10	6152,5	0,14	1,00	1,80	0,79	0,70
1	Slip joint outer	21,86	21534,4	22,00	1,00	1,82	0,84	0,66
1	Tensinoer ring	1,00	15985,1	23,00	1,00	2,10	0,59	1,44
0,41	Slip joint inner	8,36	1694,4	31,36	1,00	2,10	0,53	0,50
1	UFJ	1,83	399,8	33,19	1,00	2,10	0,55	0,50
1	Diverter	1,81	414,6	35,00	1,00	2,10	0,64	0,53
	SUM	**335,0**	**280171,0**					

Table 11-4 Properties marine riser

As we see from Table 11-4 the wellhead (WH), is situated at the sea bed at a depth of 300 meters. The top node of the rotary is at 35 meter above the mean water level, making the riser 335 meters of length. Roughly 75% of the length is buoyancy joints. Both upper and lower flex joints are included, together with a telescope joint.

11.3 Forced vessel motion (RAO)

The RAO subroutine calculates platform motion depending on the incoming wave. As we are going to run analysis comparing results with RIFLEX, we first want to ensure that the motion subroutine gives identical results. This is done by modeling the platform deck as a single beam, and imposing RAO-motion.

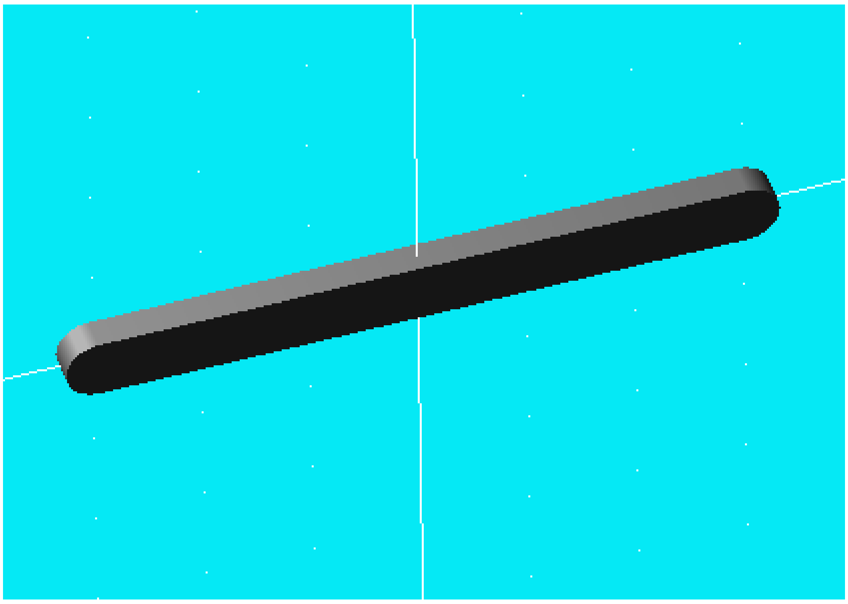

Figure 11-3 RAO motion link

The test is done with three combinations of wave height and period. The translational displacement of the center of mass and the rotational displacement about its axis are compared with the result of the RIFLEX analysis.

11.4 Marine riser model with rig motion

Even though tests should show that the wave force subroutine and the RAO motion subroutine works isolated, we still have to test them together. Therefore, as a last step in verifying Adams, an analysis of the marine riser with forced deck motion is done. The RAO motion is applied to the top node of the diverter (z=35m), representing the rig deck. For simplicity the same riser as in chapter 11.2 is used for the analysis. To make the model more realistic, the tensioner force is no longer a single force component directed in global z-direction. Instead, two forces are applied with an initial angle of $5°$ from the z-axis. The forces follow the orientation of the deck. The telescope joint is modeled by allowing only displacement in local z-direction between the inner and outer barrel (see Figure 11-4). It is modeled frictionless. The deck is modeled by a square plate, rigidly connected to the diverter.

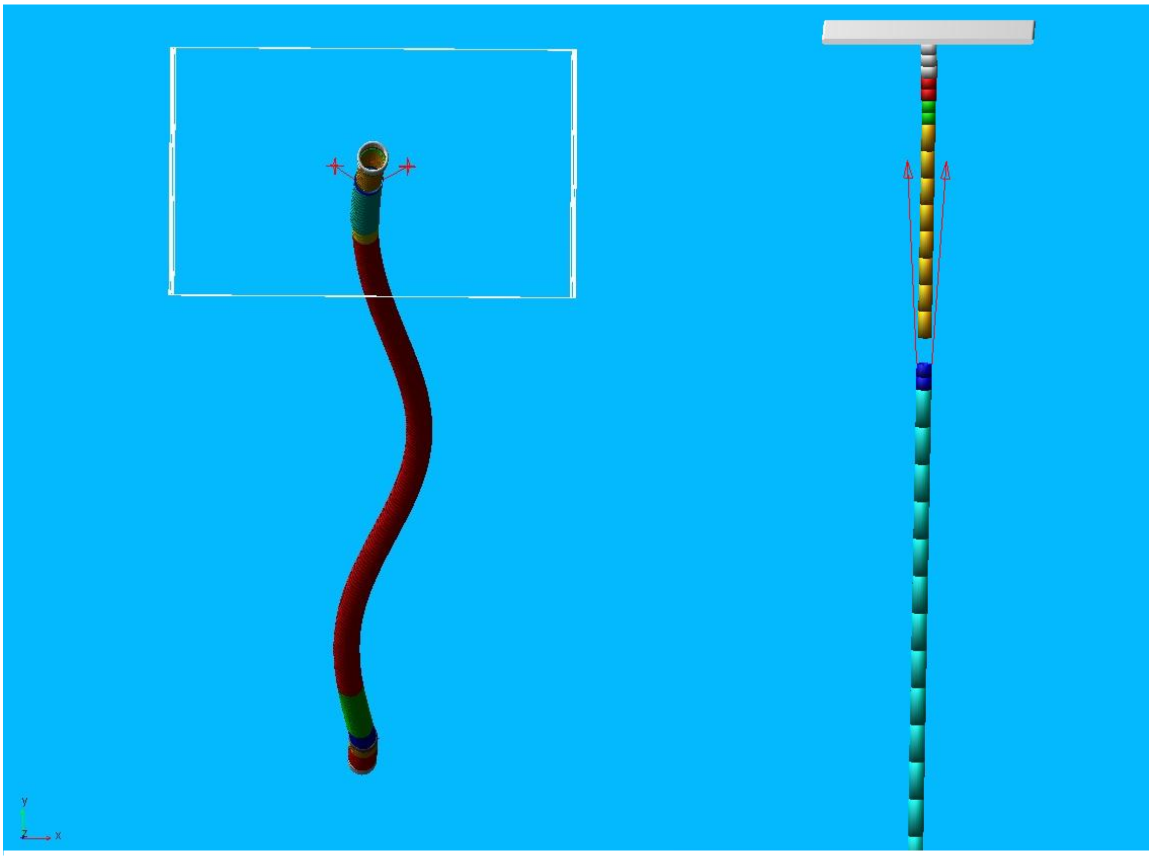

Figure 11-4 Marine riser model with deck motion and two tensioner forces

11.5 Tensioner system

To make sure that the tensioner system works as according to what we intuitively expect from physicality. At the same time it is verified that the variables are passed in the right manner from Adams to Easy5 and back.

To make troubleshooting easier, we want a simple model. As marine riser tensioners are governed by hydraulics, we chose to represent the pneumatics of the system by a constant pressure source. Two pressure sources are included, representing the high and low pressure gas volumes. The piping with bend and frictional losses are included, together with a shutoff valve. To not shock the piston out of place, the pressure at the high pressure reservoir is ramped on over the first 20 seconds by a linear ramping function. The piston is given mass and damping constant of realistic proportions. Pipe and valve resistance values are found in the literature ((White, 2003) and (Marré, 2009)).

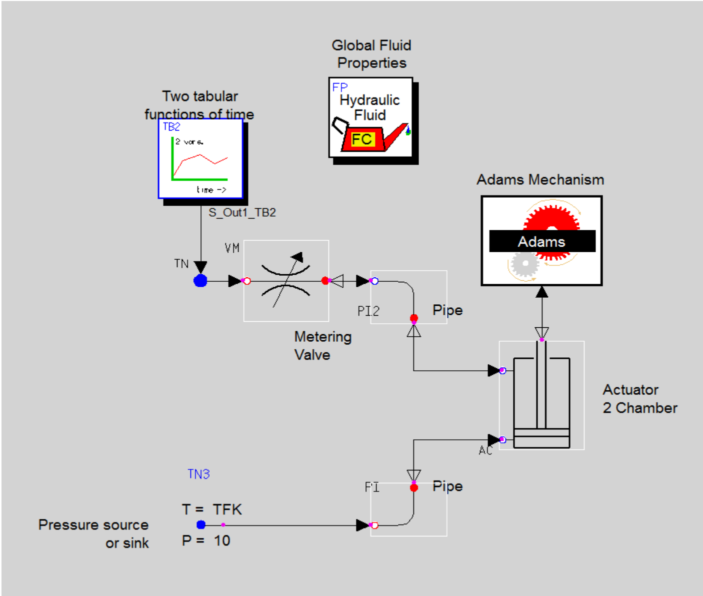

Figure 11-5 Easy5 tensioner model

- Actuator: $A_{high} = 763\text{cm}^2$, $A_{low} = 905cm^2$ and $C_{friction} = 10Ns/cm$
- Boundary conditions: $P_{low} = 10bar$ and $P_{high} = 85bar$
- Metering/Shutoff valve: $A_{orifice} = 100cm^2$ and $C_{\text{discharge}} = 0.61$
- High pressure pipe: $D = 15cm$, $L = 2500cm$, $\varepsilon_{roughness} = 0.005cm$ and $N_{bends} = 8$.
- Low pressure pipe: $D = 12cm$, $L = 2000cm$, $\varepsilon_{roughness} = 0.005cm$ and $N_{bends} = 4$.
- Function generator: $P_{start} = 10bar$, $P_{stop} = 85bar$ and $T_{ramp} = 20s$.
- Adams parameters: s_{piston}, v_{piston} (input) and $F_{reaction}$ (output).

A rigid link acted on by a force is created in Adams. The force has action point on the center of mass of the link and reaction point at the origin. The link is forced into motion by the same RAO subroutine as the link in chapter 11.3 and the analysis are done for the same waves. The magnitude of the displacement offset and velocity between the two points are passed as parameters to the easy5 model. As easy5 is based on centimeters, and Adams on SI-units, it is necessary to convert the output from centimeters to meters. This is done by multiplying the displacement and velocity in Adams with 100. The piston rod in the tensioner model is then given the same velocity and displacement. This forces hydraulic fluid in and out of the chambers of the hydraulic cylinder, creating a variation of the force.

11.6 Coupled marine riser and tensioner system

We continue with the marine riser from chapter 11.4, but with the constant tensioner forces replaced by the hydraulic system of chapter 11.5 modified for two forces. As input parameters in the tensioner model is the relative displacement offset and velocity between the tension ring and the point on the rig deck where the force is applied. Output parameters of the tensioner model are the tension forces, applied in $5°$ angle from the local z-axis. In the tensioner system model, we need two actuators, one for each tension force. Pressure is ramped on, to avoid unrealistic initial conditions.

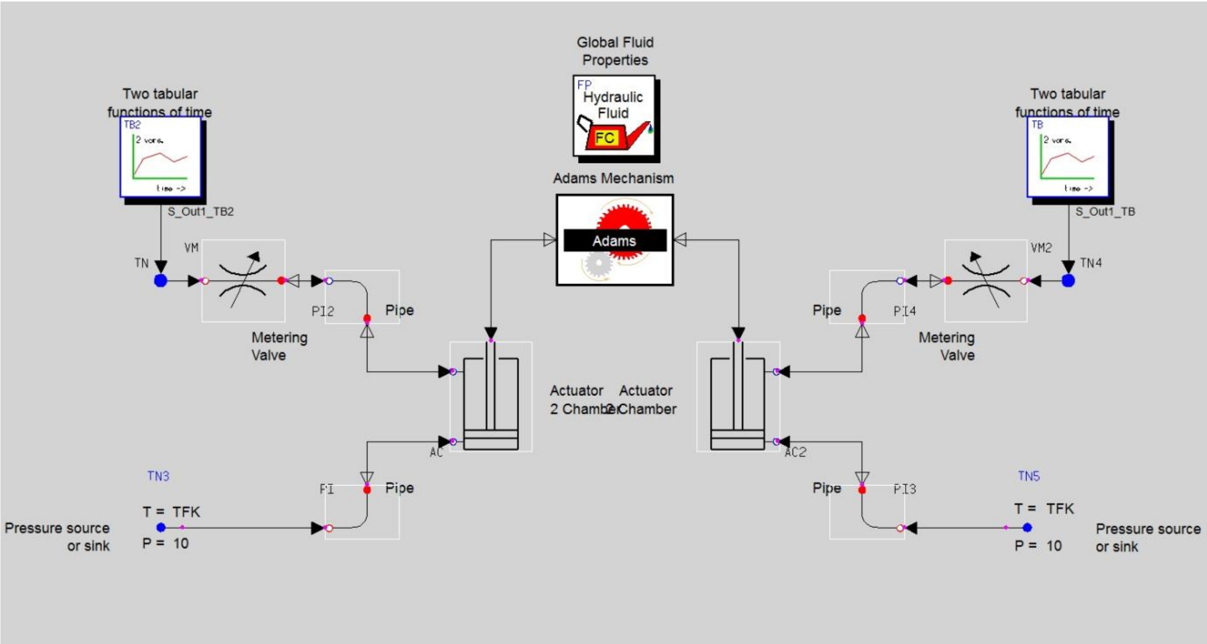

Figure 11-6 Double tensioner model in Easy5

The pressure difference over the piston is decided by the area of the piston, and the size of the vertical force component required.

$$\sum F_z = 2T\cos\alpha - F_{mass} + F_{overpull-LMRP} + F_{buoyancy} = 0 \qquad ; \qquad T = \Delta p A$$

Equation 11-2

The hydraulic parameters are as described in chapter 11.5.

12 Results

This chapter presents an excerpt of the key results of the analysis. When checking for compliance between Adams and RIFLEX for the riser models, the reaction moment at the sea bed is seen as the most important parameter of verification, as it represents a summation of the global picture, both forces and displacements. It is also one of the few parameters possible to extract at the exact same location in both programs, as the positioning of nodes differs. In areas with significant dynamics, the force and moment gradients may be great, and therefore the compliance is expected to be lower. Mainly the results of the analysis with the highest wave are studied, as it gives largest forces and displacements, and potentially largest deviance. A collection of all measured results is found on the CD in Appendix A.

12.1 Uniform riser model

Analysis of the uniform riser model is the first step in verifying the wave force subroutine. The riser is made as uncomplicated as possible, for easy troubleshooting, but at the same time having realistic dimensions. The riser is a positively buoyant cantilever pipe, and needs therefore no tension force to avoid buckling. The lack of a tension force gives large horizontal displacements, and in that manner testing the wave force subroutine to a greater extent. Reaction forces, beam forces and displacements at key locations are plotted against a similar analysis in RIFLEX.

12.1.1 Sea bed

The reaction forces at the sea bed are seen as most important, as they represent a summation of the forces acting on the rest of the riser.

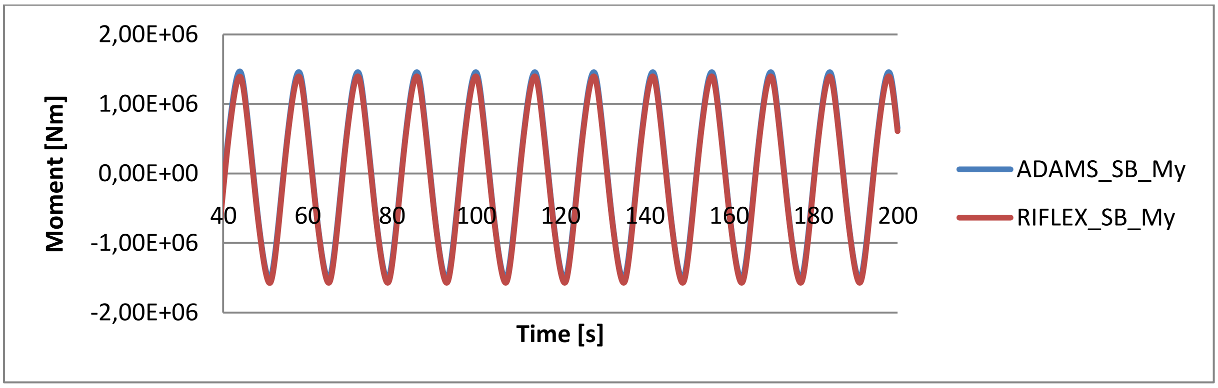

Figure 12-1 Reaction moment around global y-direction at sea bed (H=06m, T=14s)

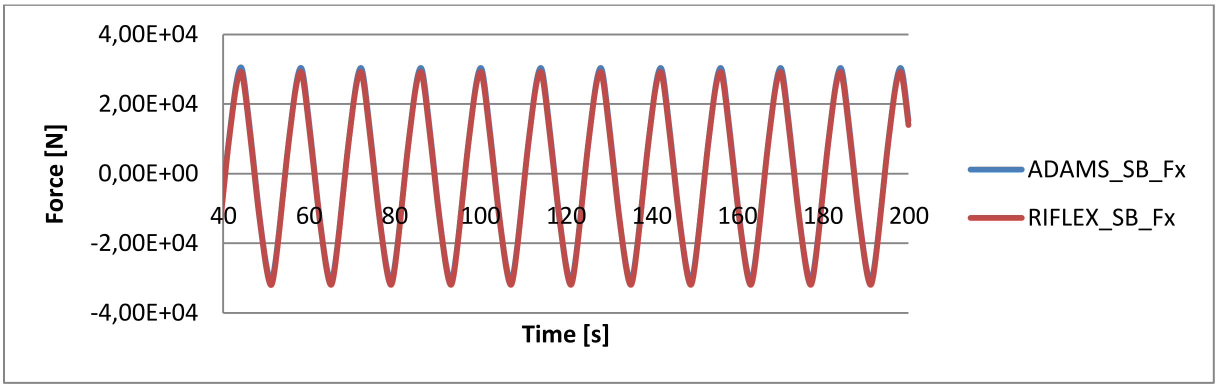

Figure 12-2 Reaction force in x-direction at sea bed (H=06m, T=14s)

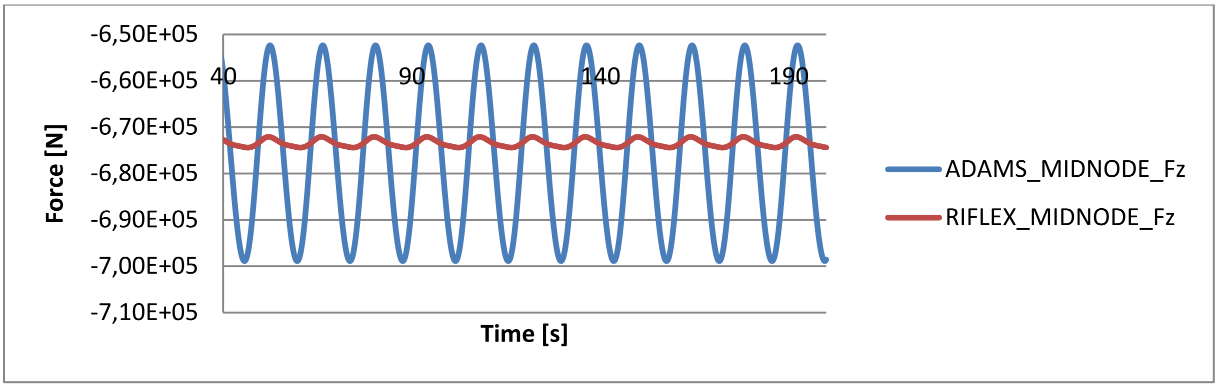

Figure 12-3 Reaction force in global z-direction at seabed (H=06m, T=14s)

The compliance with RIFLEX is found satisfactory. The tension force in RIFLEX is extracted from the static initial conditions run. It is applied according to a global coordinate system, but extracted according to the local element coordinates, and oscillates therefore with the angle of rotation. The slight offset from the mean value is because of the different node location in the models. The offset is equal to the half the element's buoyancy minus mass. It is negative as it counter acts the riser's positive buoyancy, and pulls it in negative z-direction. The Adams tension force is seen to oscillate. This is because the buoyancy force in Adams is dependent on the instantaneous sea level.

12.1.2 Mid water depth

Secondly, the forces at mid water depth are compared. Yet again the compliance is excellent.

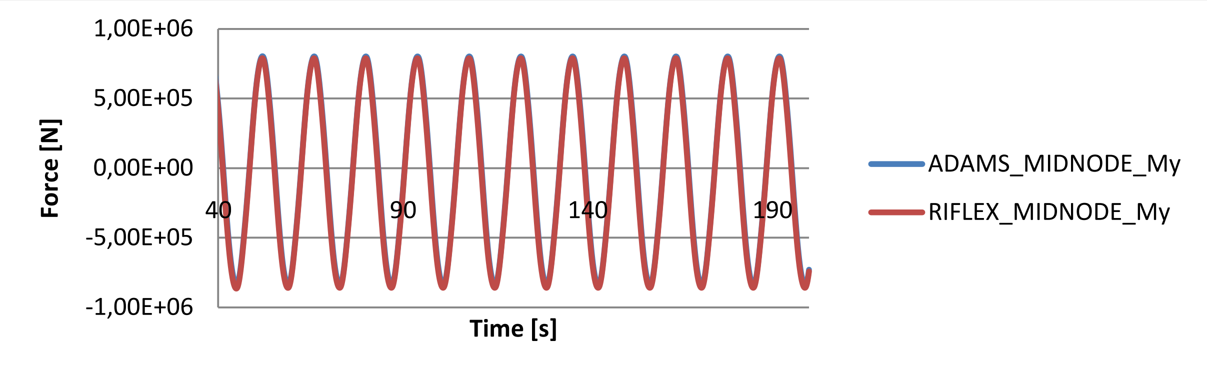

Figure 12-4 Beam moment around global y-axis at Z=-150m (H=06m, T=14s)

12.1.3 Top node

Finally the horizontal displacement of the top node gives the boundary conditions in the other end.

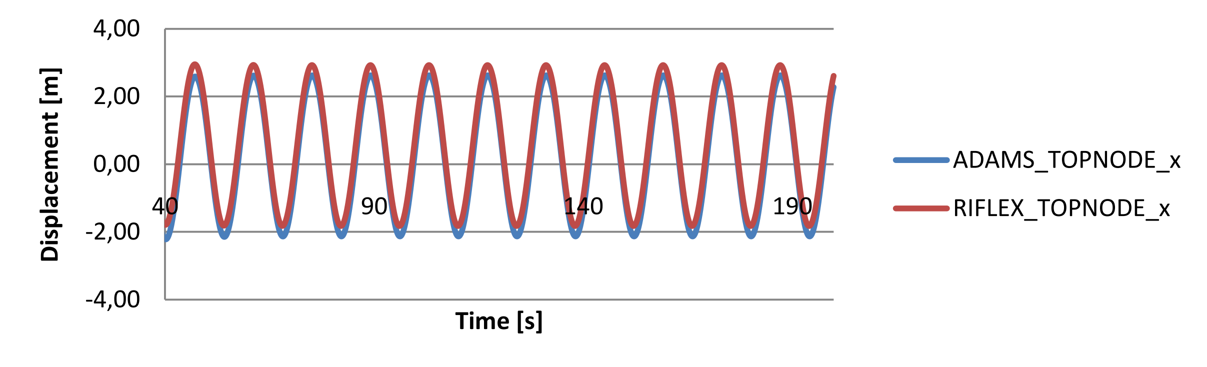

Figure 12-5 Horizontal displacement of top node (H=06m, T=14s)

We notice good compliance at all locations. As expected the largest deviance is found at the top node, the node with most dynamics.

12.2 Marine riser model

Now that the wave force module is verified, the second step in verifying MSC Adams as a riser analysis software, is to confirm compliance with RIFLEX on a full marine riser model. Staying with the stepwise verification pattern, the riser is modeled only up to the tensioner ring, with a constant globally oriented tensioner force. Again we start by considering the reaction forces at the sea bed.

12.2.1 Sea bed

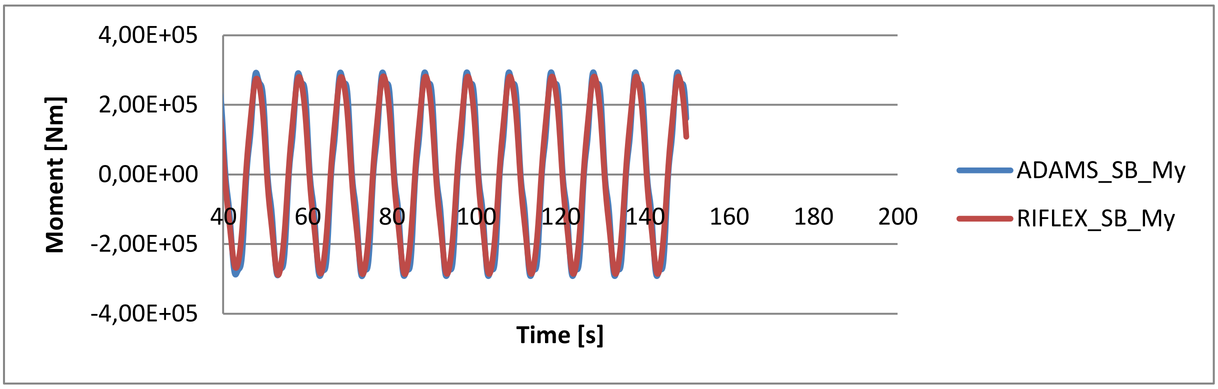

Figure 12-6 Reaction moment around global y-axis at sea bed (H=06m, T=14s)

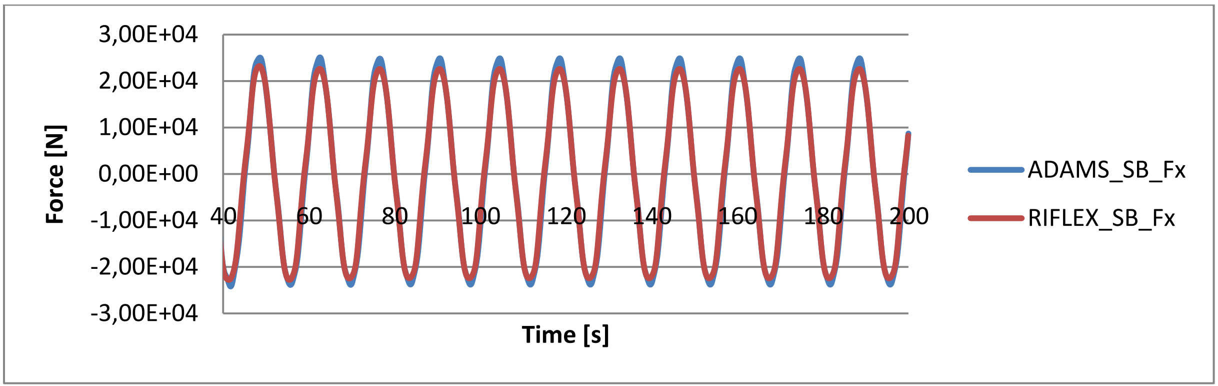

Figure 12-7 Reaction force in global x-direction at sea bed (H=06m, T=14s)

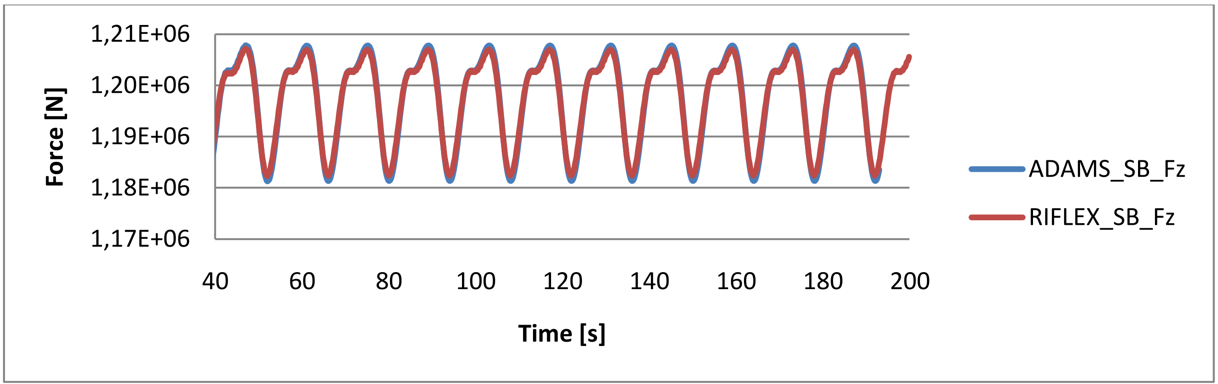

Figure 12-8 Dynamic tension force at sea bed (H=06m, T=14s)

We find good compliance of the reaction forces at the sea bed. A slight over estimation of the Adams model is noted.

12.2.2 Mid water depth

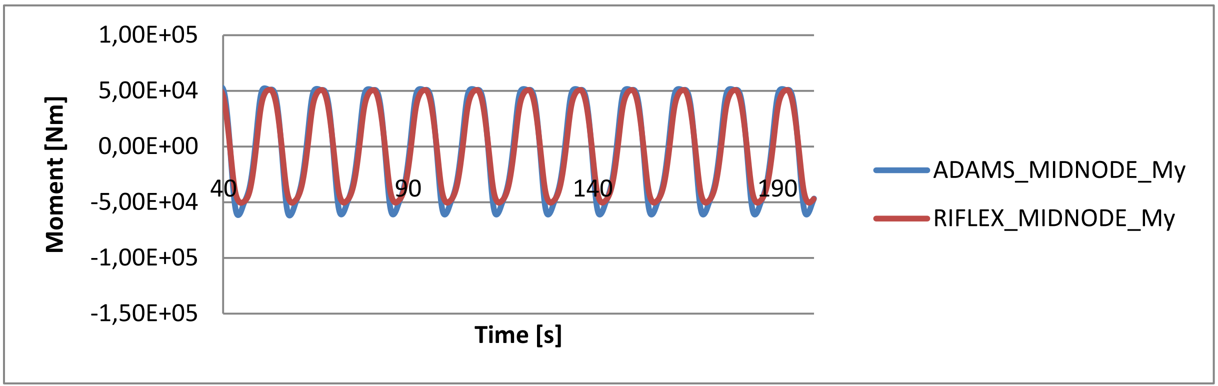

Figure 12-9 Moment around global y axis at mid water depth (H=6m, T=14s)

12.2.3 Tension ring

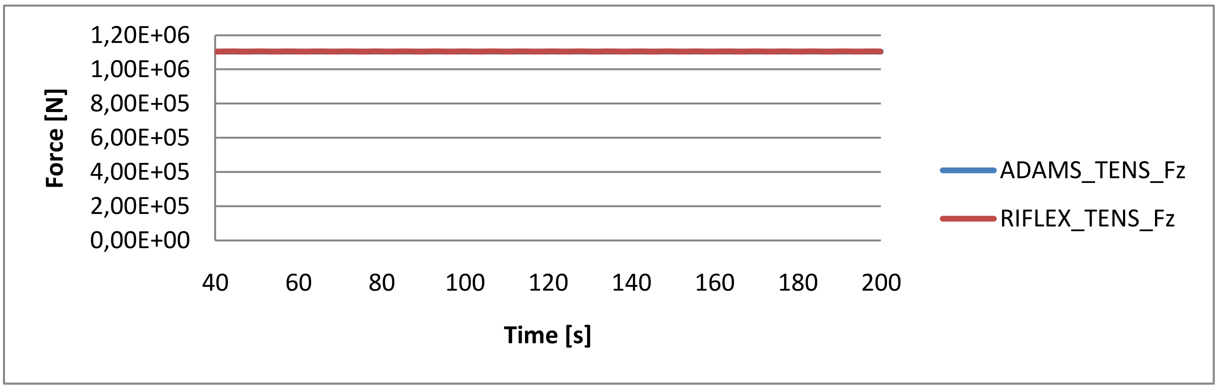

Figure 12-10 Tension force applied in global z- direction at tension ring (H=6m, T=14s)

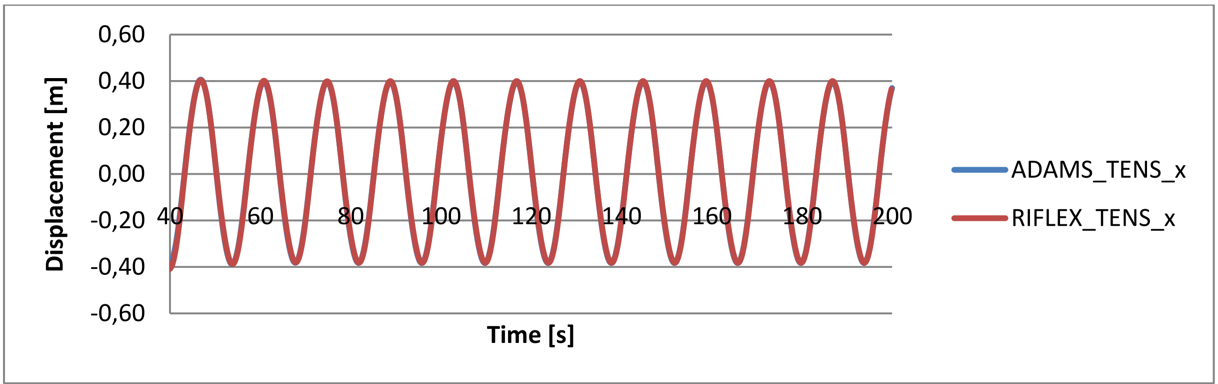

Figure 12-11 Displacement in global x-direction at tension ring (H=6m, T=14s)

Over all very good compliance in the marine riser model as well, although the results are closer to each other close to boundary conditions than in areas with significant dynamics.

12.3 Forced vessel motion (RAO)

Before applying rig motion to the marine riser, it is seen as vital to first verify that the RAO motion subroutine works. This is done by measuring the translational displacement in x- and z-direction and rotational displacement around the y-axis of a bar element set in motion by the subroutine. The analysis is done for all three wave cases, and the results are plotted against a similar analysis in RIFLEX. The RAO is found on the “RAO.txt” file on the CD in appendix A.

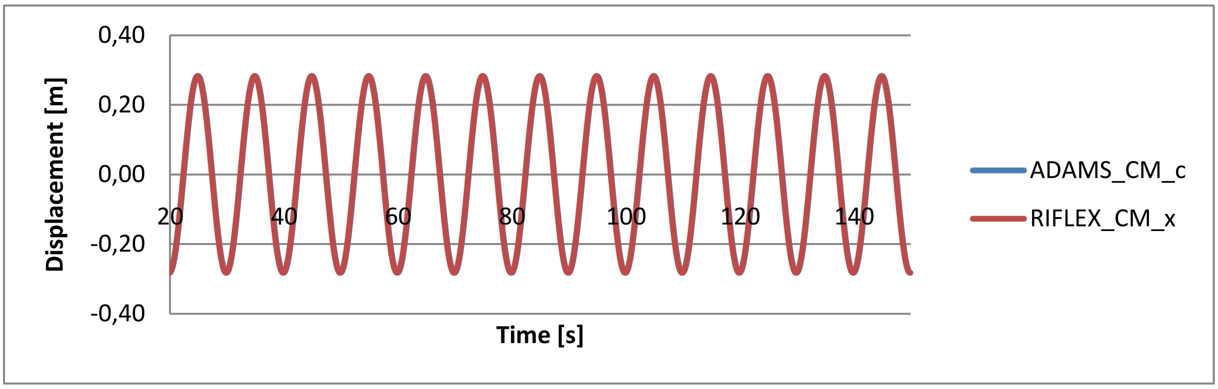

Figure 12-12 Translational displacement of center of mass (CM) in global x-direction (H=3m, T=10s)

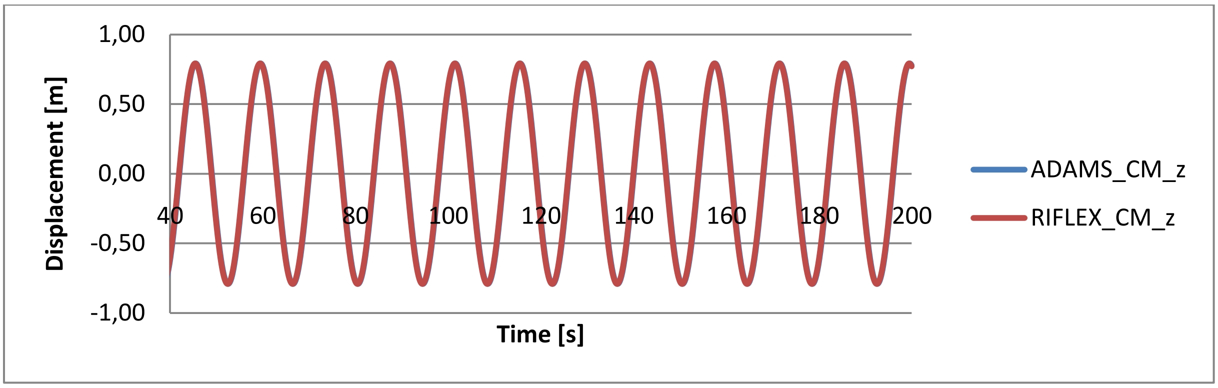

Figure 12-13 Translational displacement of center of mass (CM) in global z-direction (H=3m, T=14s)

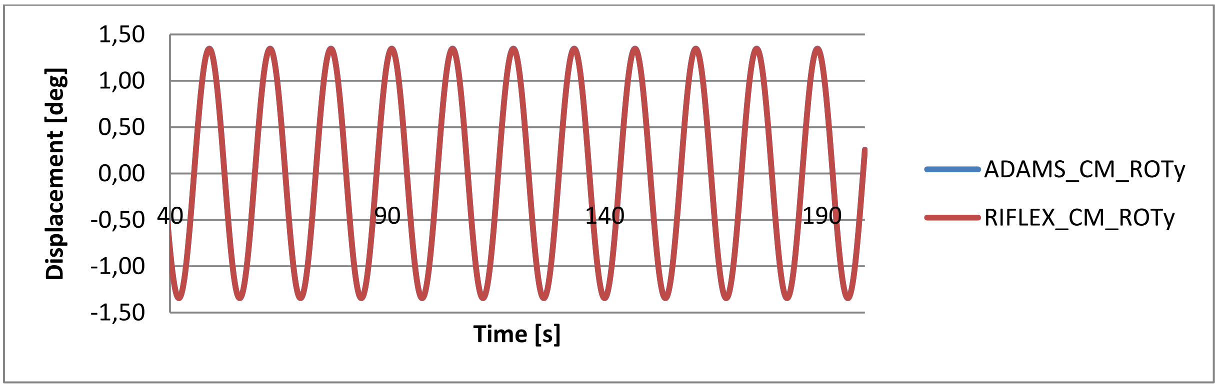

Figure 12-14 Rotational displacement of center of mass (CM) around global y-axis (H=6m, T=14s)

The displacements have excellent compliance as expected. One displacement from each of the three waves is presented.

12.4 Marine riser model with rig motion

As satisfactory compliance between Adams and RIFLEX on the marine riser was found in chapter 12.2 and the RAO subroutine is verified in chapter 12.3, we are ready to simulate a full marine riser with rig motion. Unfortunately, the Newmark solver in Adams fails for this analysis, and after consulting the support at MSC, the GSTIFF solver in Adams in used. It was also advised to increase the error acceptance from 10^{-5} to 10^{-2}. Higher deviation of the results is therefore expected.

12.4.1 Reaction at sea bed

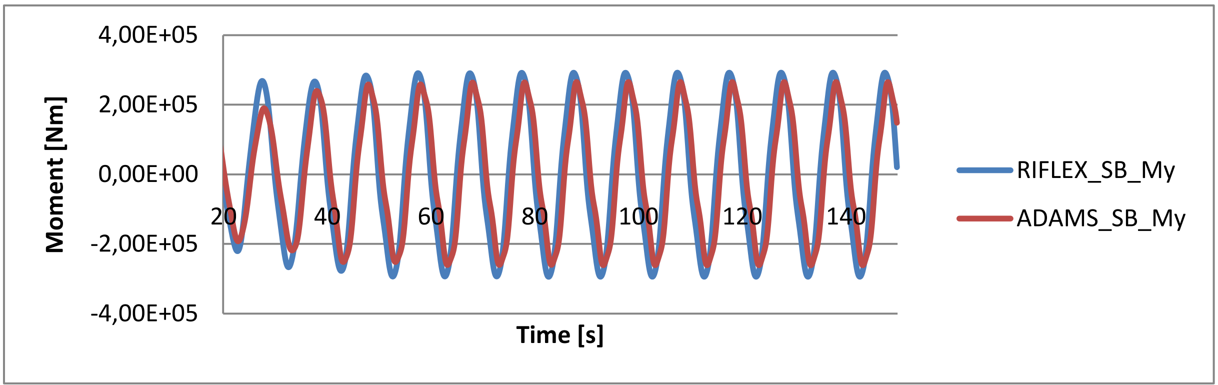

Figure 12-15 Reaction moment around global y-axis at sea bed (H=03m, T=10s)

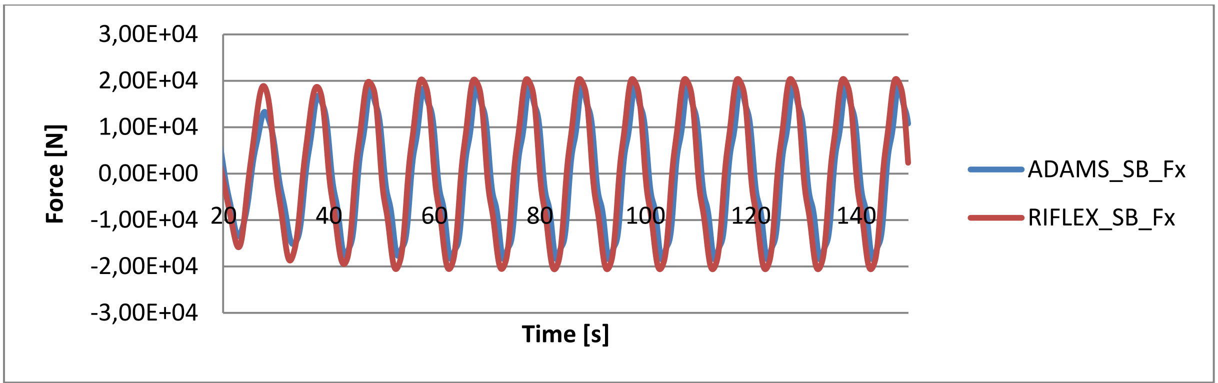

Figure 12-16 Reaction force in global x-direction at sea bed (H=3m, T=10s)

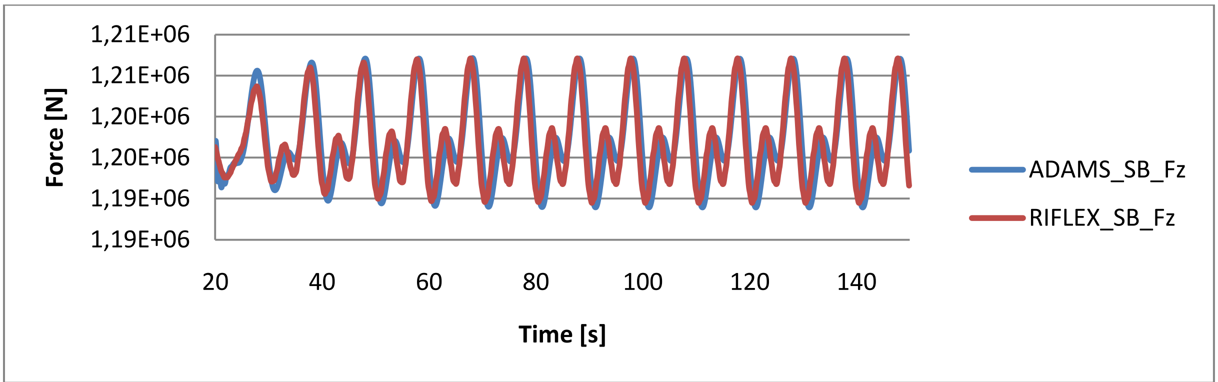

Figure 12-17 Global Reaction force in global z- direction at sea bed (H=3m, T=10s)

12.4.2 Mid water depth

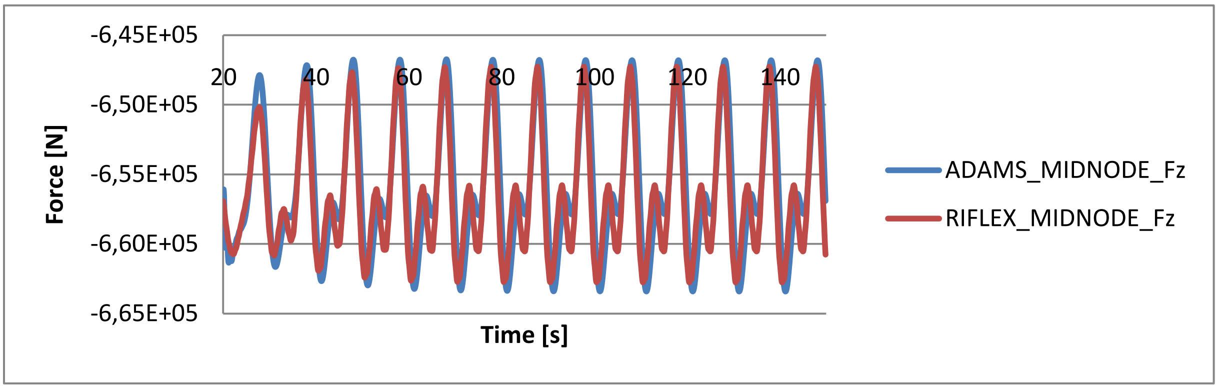

Figure 12-18 Tension force in global z-direction at mid water depth (H=3m, T=10s)

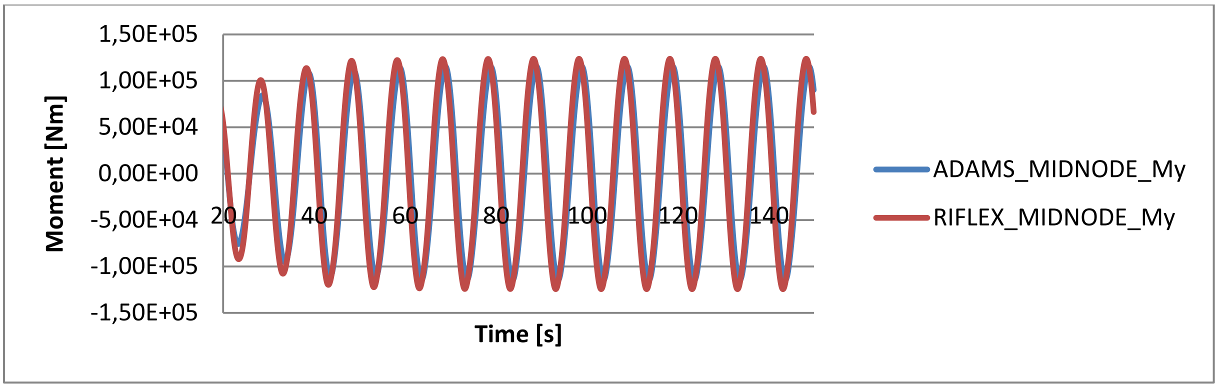

Figure 12-19 Beam moment around global y-axis at mid water depth (H=3m, T=10s)

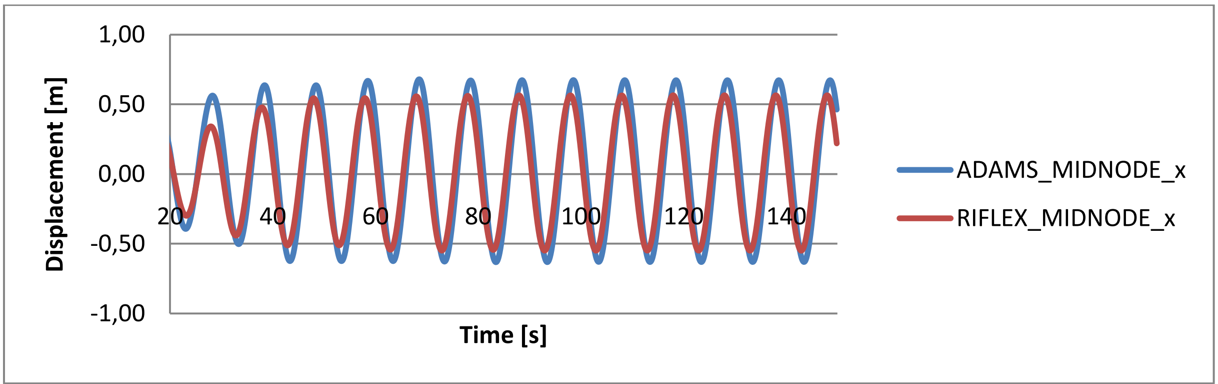

Figure 12-20 Displacement in global x-direction at mid water depth (H=3m, T=10s)

12.4.3 Deck

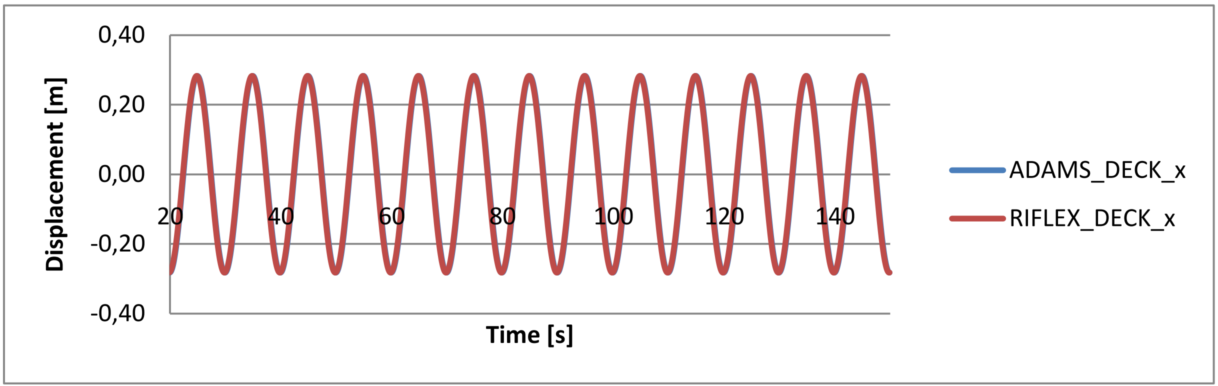

Figure 12-21 Deck displacement in global x-direction (H=3m, T=10s)

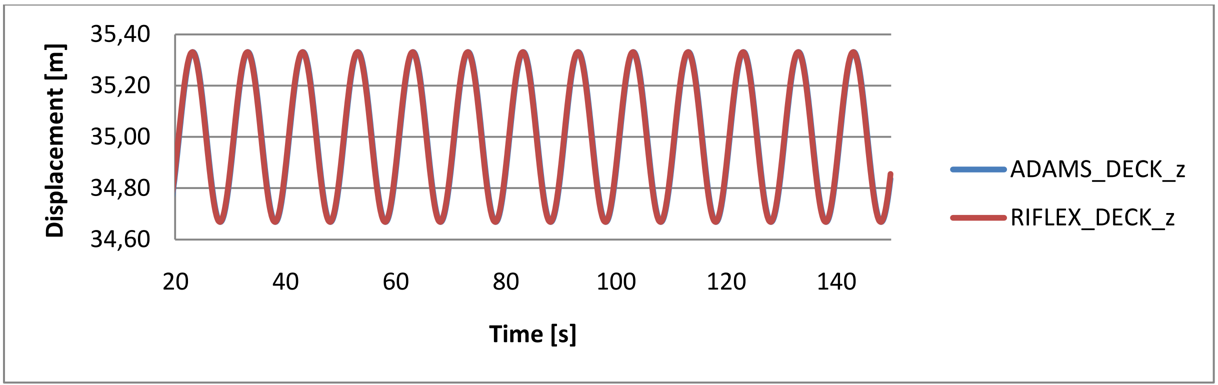

Figure 12-22 Deck displacement in global z-direction (H=3m, T=10s)

The results with the GSTIFF solver are not compliant to the same extent as with the Newmark solver, but they are of physical size and shape. A small delay in the Adams force results is noted.

12.5 Tensioner system

Before implementing the hydrodynamic tensioner on the Adams marine riser, it is verified in isolation. The interesting parameters to study would be pressure, volume flux, velocity and displacement of piston and resulting piston force. It is interesting to find out whether the tension force is in phase with the displacement or velocity, or both. Again we chose to concentrate on the results from the analysis with the most dynamics.

12.5.1 Input and output (I/O)

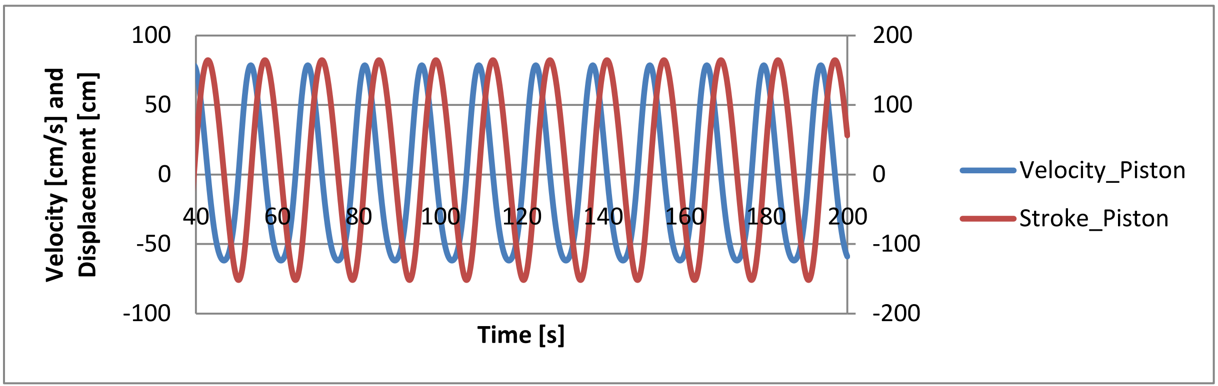

Figure 12-23 Velocity and displacement of piston in tensioner (H=6m, T=14s)

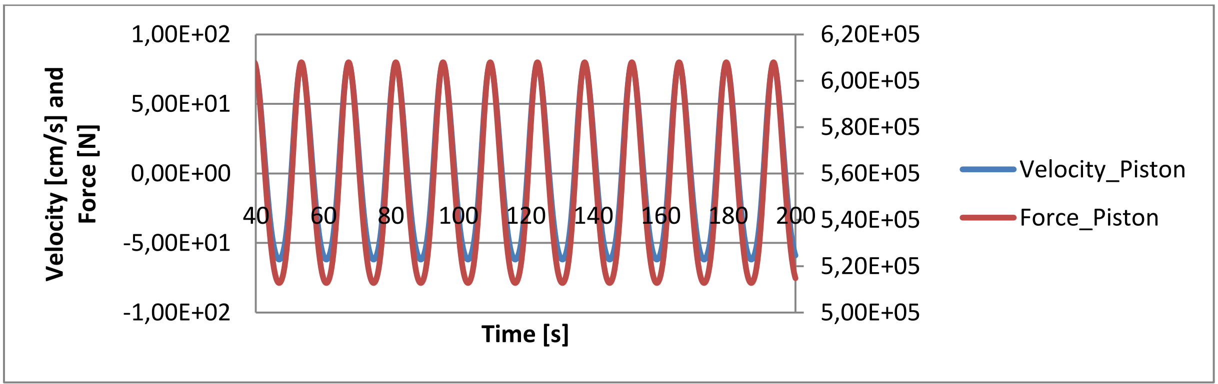

Figure 12-24 Velocity and force of piston (H=6m, T=14s)

The tension force is clearly in phase with the velocity, and not the displacement. This is natural as the pipes and valves in the hydraulic model include losses dependent on Reynolds number.

12.5.2 Pressure

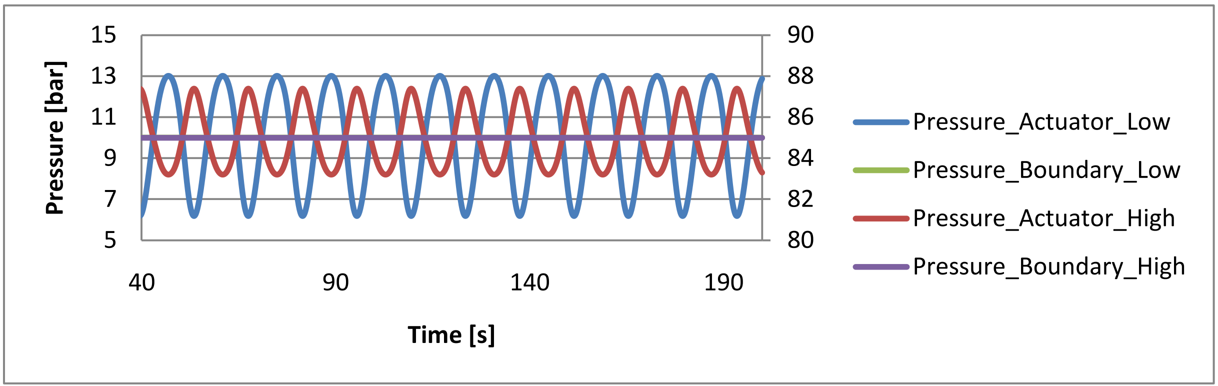

Figure 12-25 Pressure in actuator oscillating about the boundary pressure (H=6m, T=14)

The forced motion of the piston requires the hydraulic oil to move through pipes and lose pressure to the pipe resistance. The variation of the pressure gives a varying tension force.

12.6 Full marine riser model with hydraulic tensioning

In this chapter the effect of having a hydraulic tensioner system is studied. The constant tension force riser is identical to the riser in chapter 12.4. The tensioner system is identical to the one in chapter 12.5, but with twice as many components, to be compatible for two tensioners. It is interesting to study how having a varying tension affects both the tension in the riser, but also other forces, moments and displacements.

12.6.1 Reaction at sea bed

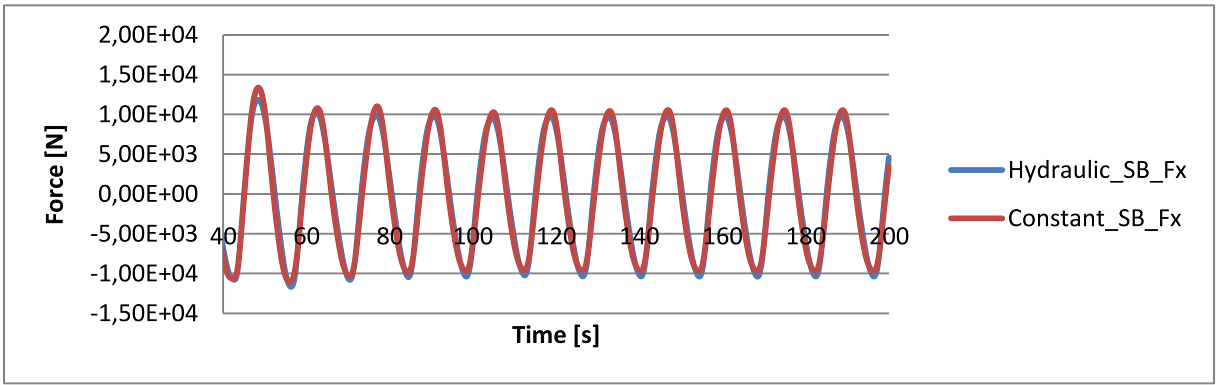

Figure 12-26 Reaction force at sea bed in global x-direction (H=6m, T=14s)

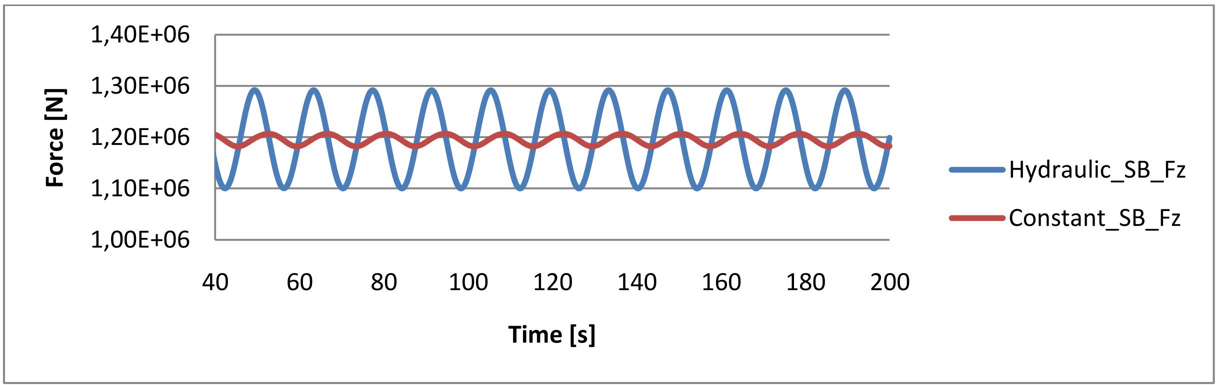

Figure 12-27 Reaction force at sea bed in global z-direction (H=6m, T=14s)

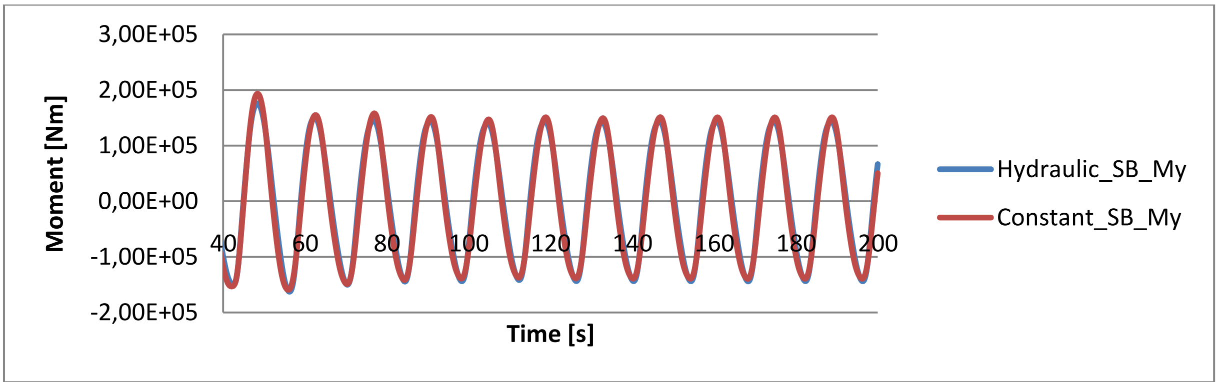

Figure 12-28 Reaction moment at sea bed about global y-direction (H=6m, T=14s)

12.6.2 LMRP

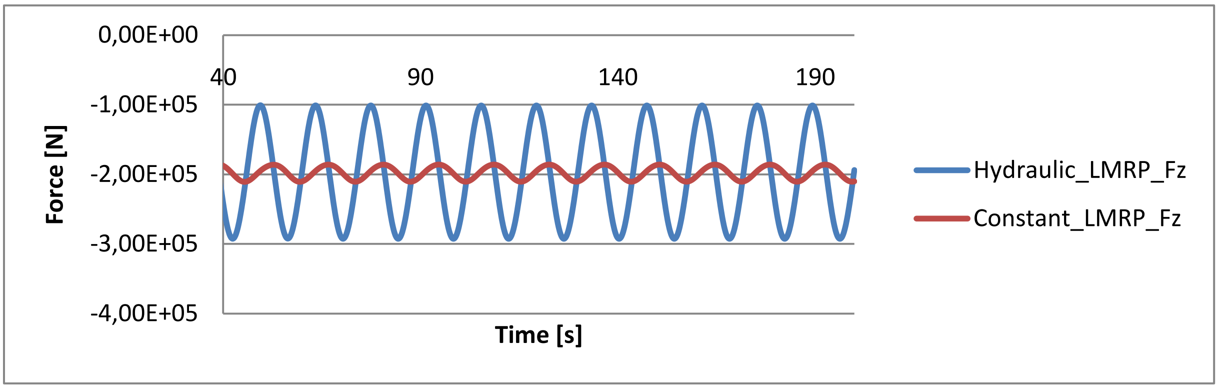

Figure 12-29 Tension overpull between LMRP and BOP (H=6m, T=14s)

12.6.3 Mid water depth

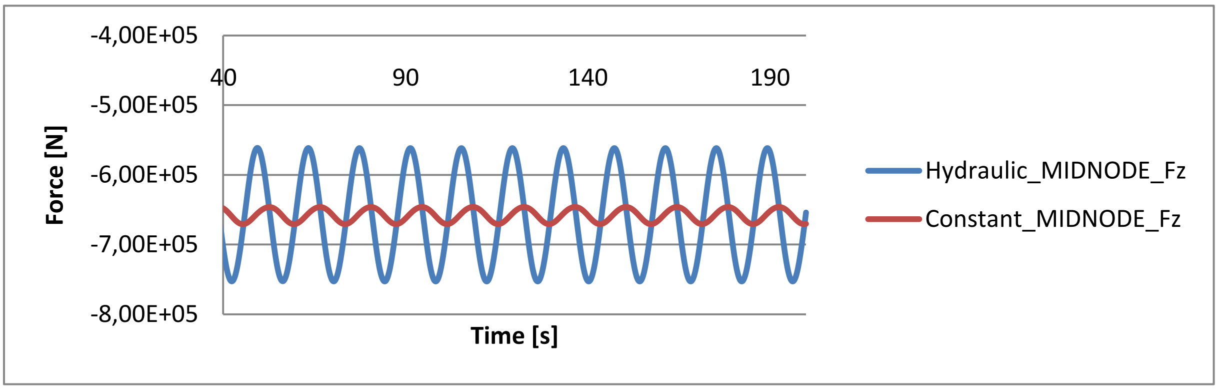

Figure 12-30 Tension force at mid node in global z-direction (H=6m, T=14s)

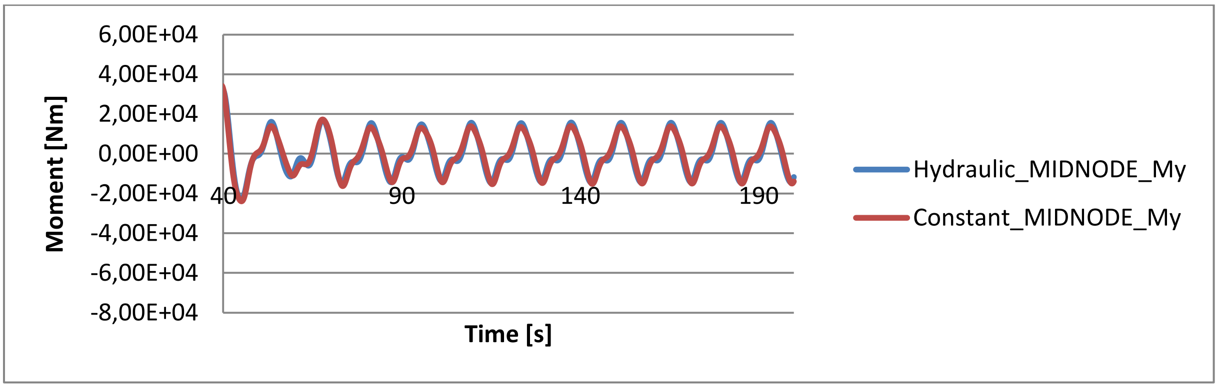

Figure 12-31 Beam moment around y-axis at mid node (H=6m, T=14s)

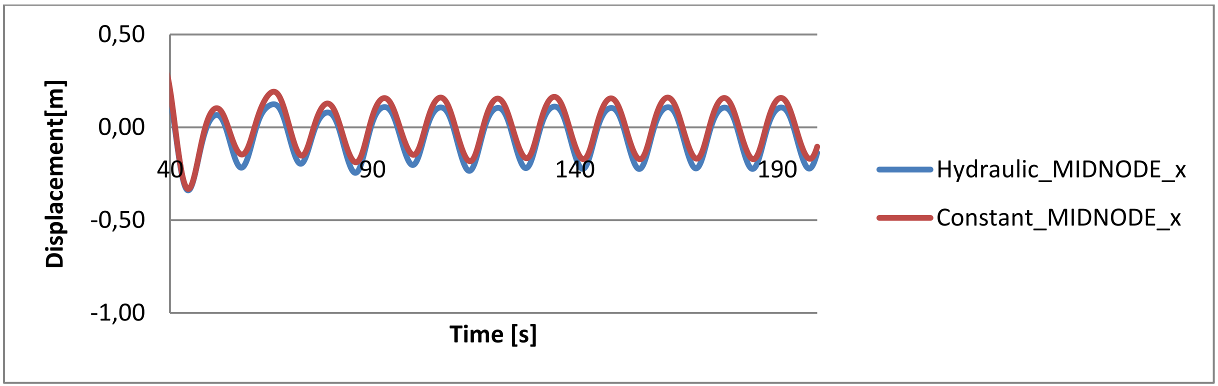

Figure 12-32 Displacement in global x-direction at mid node (H=6m, T=14s)

12.6.4 Tension ring

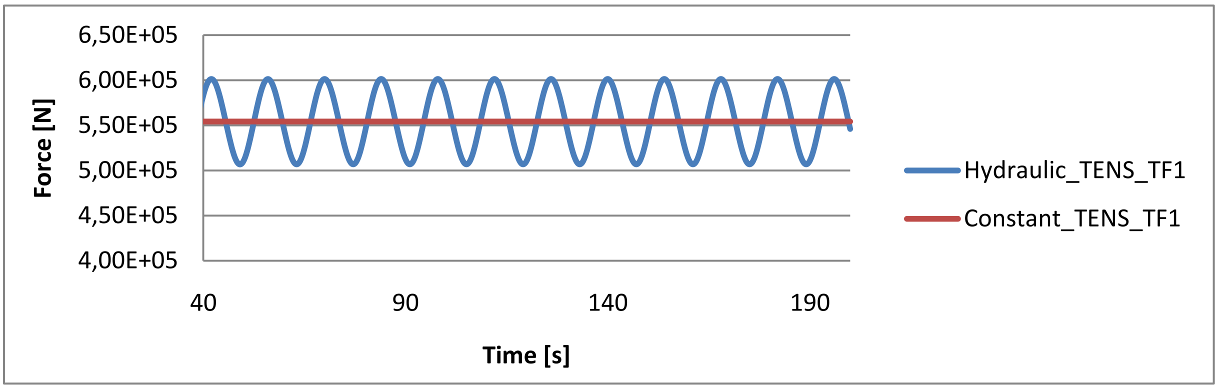

Figure 12-33 Applied tension force (H=6m, T=14s)

12.6.5 Tensioner parameters

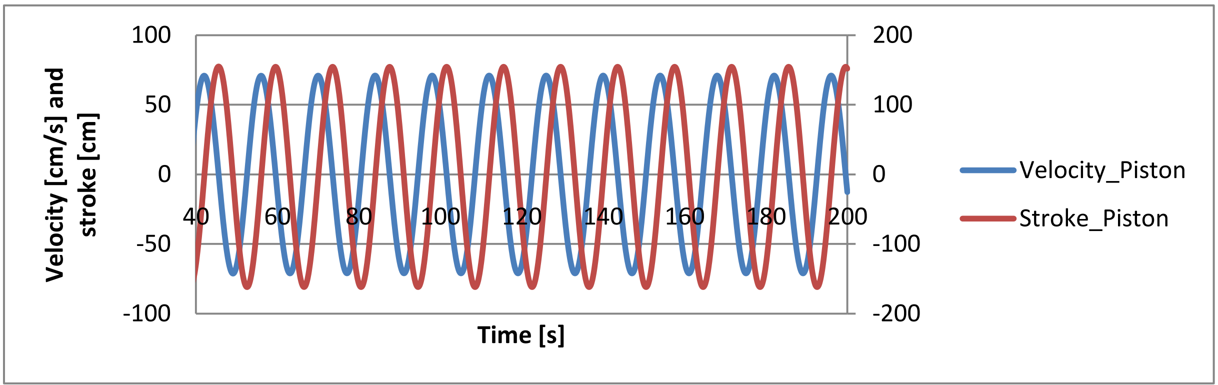

Figure 12-34 Velocity and displacement of piston in hydraulic system (H=6m, T=14s)

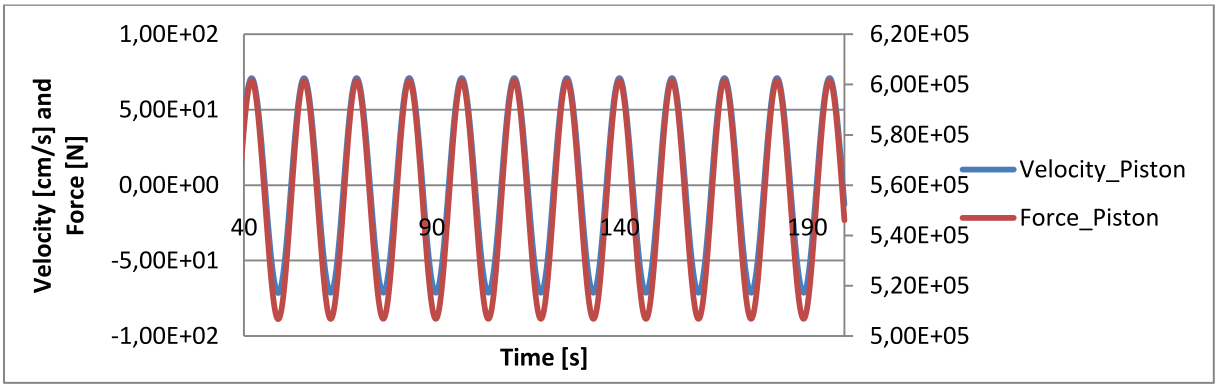

Figure 12-35 Velocity of tensioner piston and delivered tensioner force (H=6m, T=14s)

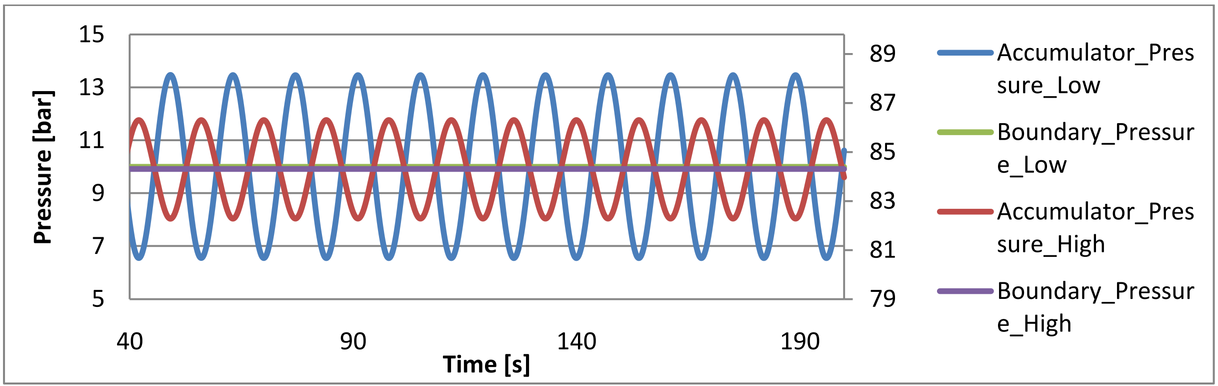

Figure 12-36 Pressure of boundary conditions and accumulator volumes (H=6m, T=14s)

The results of the two analyses show that the global dynamics of the riser is affected by having a more complex tensioner system. Although the effect in most areas is slight for average waves, as the ones in this analysis, the effect in extreme situations may be significant. It is also noted that the overpull tension at between the LMRP and BOP is reduced to a mere 50% at the minimum values.

13 Summary and discussion

An analysis tool for modeling the coupled interaction between marine riser and tensioner system is developed by implementing hydrodynamic forces and RAO motion in MSC Adams. Adams is verified as an analysis tool for slender marine structures by checking for compliance with RIFLEX. The verification is done in a step-wise manner by analyzing riser models of gradually increasing complexity.

The hydrodynamics are calculated based on Morison's equation for slender oscillating structures in oscillating flow, and are implemented into Adams as a subroutine. The wave dynamics are based on linear regular waves, and the rig motion is assumed to be a function of the wave height and period. The dynamics of the hydraulic tensioner system are based on equations of equilibrium of mass, momentum and energy of fluid systems and are calculated by the implemented routines of MSC Easy5. The dynamics of a marine riser model with a hydraulic tensioner system is compared to one with constant tension. The marine riser includes the most common components and joints, and is represented by a finite number of Timoshenko beams with distributed mass and hydrodynamic properties. The tensioner system is represented by hydraulic actuators connected through pipes to constant pressure boundary conditions.

Various assumptions and simplifications have been made in order to reduce the complicity of the models. In the tensioner system, the gas accumulators are reduced to a constant pressure source. As the analysis is done for a maximum wave height of 6 meters, this may be seen as a valid assumption. For extreme condition analysis, the tensioner model is recommended to include gas dynamics, as nonlinearities may occur. The six tensioners of the real system are reduced to two. This means manipulating pipe and actuator cross sectional areas, to acquire the right mean tension force. The pressure loss through a pipe is Reynolds number dependent, and manipulation of the pipe diameter introduces inaccuracies. Assumptions of other parameters of the hydraulic system, as roughness height, frictional resistance of piston-cylinder assembly and viscosity may also introduce errors. These are hard to find, and are recommended to be chosen from a broad collection of empiric data for resembling systems. Thermal energy considerations and temperature variations are also assumed to be nonexistent, as the effect on the hydraulic fluid is small. But for systems including gas dynamics it is recommended to include, high because of the temperature dependency of the gas state.

The marine riser dynamics are also subject to many assumptions. The rig motion is assumed unaffected of the tension forces as they represent less than 1% of the gravitational forces acting on a drill rig. The hydraulic forces are applied in the center of mass of each element, instead of being evenly distributed, an assumption valid for a reasonable number of elements. The force module is meant for slender structures, Morison's equation is assumed valid, though greater accuracy would be acquired with a more complex wave theory. Airy wave theory gives reasonable accuracy for the wave cases in this thesis, though higher order Stokes theory is preferred. The representation of the riser as Timoshenko beam elements, may introduce inaccuracies and unnecessary calculation time. This beam theory was developed for short beam analysis subject to excitation of high frequency loads, with wavelength approaching the thickness of the beam. It takes into account shear deformation and rotational inertia effects. The fact that the RIFLEX beam elements are long beam elements, may introduce may lead incompliance between the two programs. All components of the riser are assumed to have circular cross section when considering hydrodynamic forces. As the components of the marine riser with noncircular cross sections normally are situated closer to the sea bed, where the dynamics are small, the assumption is considered good.

The compliance of the Adams model with RIFLEX was of varying degree. The results from the analysis of the models in 11.1, 11.2 and 11.3 all gave excellent agreement with the RIFLEX analysis. The results of the marine riser in 11.2 deviate the most at mid water depth. This may be a result of the difference in the way the beam models are constructed. While the RIFLEX model is flexible all the way to the sea bed, the Adams model is rigid the last half element length. This means that the model nodes, where the results are extracted, have an offset of a half element length compared to RIFLEX. The node at mid water depth also has the riser with the least restrictions against movement, so a small difference in the computed eigenvalue, may lead to large differences in forces, moments and displacements locally. The compliance between the riser models at the top node and the reaction forces at the sea bed are therefore seen as the most important evaluating parameters and since the compliance on this point is very good, the wave force subroutine is assumed to be correct. The small offset of the tension force seen will differ by half the weight of the top element of the tension ring, as Adams measures the force applied, while RIFLEX measures the beam reaction tension.

After encountering problems running the full marine riser model in Adams with forced vessel motion (ref. 11.4), the solver was switched from Newmark to the “SI2” formulation of “GSTIFF” as advised by MSC Software support. At the same time, it was recommended to lower the error tolerance from 10^{-5} to 10^{-2}. This meant that a great part of the basis for comparison with RIFLEX disappeared. As seen from the results presented in 12.4, the results now suffer a lag, not present in the earlier analysis. The amplitudes of the forces and displacements also differ 3-7%. The compliance continues to be best at the boundary conditions. It may also be that the difference in beam element formulations comes clearer in model 11.4 than the earlier three, because of the increased dynamics introduced by the forced vessel motion. The fact that RIFLEX finds static equilibrium of mass, tension and buoyancy forces first, before ramping on the wave forces, while Adams ramps on all forces both mass, tension, buoyancy and wave forces all at the same time, may introduce differences. Early in the ramping period, the tension force is low, so the effect of the horizontal wave forces leads to greater displacements.

The tensioner system model created in Easy5 (ref. 11.5) worked well. As the pipes, the valve and the actuator all have resistances only dependent on the flow’s Reynolds number, the tension force is in phase with the velocity of the piston. As the hydraulic resistance parameters are found from mean values in the literature and by educated guessing, they should not be given too much attention, but it is noticed that the coupling between the two programs works as intended.

The coupled marine riser and hydraulic tensioner model shown in 11.6 is created based on the riser model the riser model in 11.4 and a modified version of the tensioner in 11.5. When the results of the hydraulically tensioned model are compared to the model with constant tension forces, it is noted that the dynamics of the riser changes slightly. As expected the only results suffering great deviation are the ones of the beam tension. It is noted that the tension force between LMRP and BOP is a mere 50% of the intended value at the minimums. In higher waves or with a more complex tensioner model, this may lead to a negative overpull. In disconnection situations this may mean slamming the LMRP into the BOP, a situation far from ideal.

The further work with the thesis will be to solve the solver problems, so the Newmark solver may be applied to all models. Also further development of the wave force subroutine to include irregular sea, and potentially higher order Stokes waves, should be done.

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A Scatter diagram

Tp (s)			2	4	6	8	10	12	14	16	18	20	>=
Hm0 (m)			4	6	8	10	12	14	16	18	20	22	22
0	-	0,5	0,005	0,043	0,145	0,156	0,061	0,029	0,011	0,007	0,007	0	0
0,5	-	1	0,179	1,465	3,15	2,708	1,157	0,32	0,07	0,034	0,023	0,009	0,011
1	-	1,5	0,045	2,524	5,101	4,903	2,551	0,959	0,283	0,091	0,027	0,011	0,002
1,5	-	2	0,002	1,613	4,497	5,205	2,851	1,39	0,513	0,129	0,027	0,011	0,005
2	-	2,5	0	0,438	3,332	4,368	3,023	1,188	0,701	0,111	0,018	0,007	0,002
2,5	-	3	0	0,113	2,15	3,565	2,672	1,225	0,637	0,113	0,027	0,014	0
3	-	3,5	0	0,007	1,222	2,969	2,406	1,077	0,624	0,206	0,032	0,005	0,002
3,5	-	4	0	0	0,531	2,252	2,123	1,129	0,506	0,161	0,043	0	0
4	-	4,5	0	0	0,188	1,642	1,753	0,991	0,451	0,129	0,02	0,005	0,005
4,5	-	5	0	0	0,054	0,978	1,404	0,826	0,329	0,147	0,018	0,005	0
5	-	5,5	0	0	0,02	0,494	1,182	0,651	0,268	0,147	0,032	0,002	0
5,5	-	6	0	0	0	0,215	0,86	0,59	0,245	0,098	0,027	0,007	0
6	-	6,5	0	0	0	0,098	0,553	0,515	0,172	0,052	0,016	0	0
6,5	-	7	0	0	0	0,029	0,39	0,415	0,181	0,059	0,007	0	0
7	-	7,5	0	0	0	0,011	0,225	0,324	0,159	0,048	0,014	0	0
7,5	-	8	0	0	0	0	0,134	0,236	0,141	0,039	0,002	0	0
8	-	8,5	0	0	0	0	0,077	0,175	0,118	0,02	0,005	0,002	0
8,5	-	9	0	0	0	0	0,032	0,129	0,084	0,014	0,005	0,002	0
9	-	9,5	0	0	0	0	0,007	0,082	0,061	0,011	0,002	0	0
9,5	-	10	0	0	0	0	0	0,034	0,034	0,011	0,002	0	0
10	-	10,5	0	0	0	0	0	0,02	0,039	0,009	0	0	0
10,5	-	11	0	0	0	0	0	0,007	0,02	0,014	0,002	0	0
11	-	11,5	0	0	0	0	0	0,005	0,009	0,002	0	0	0
11,5	-	12	0	0	0	0	0	0	0,002	0,007	0	0	0
12	-	12,5	0	0	0	0	0	0	0,002	0,007	0	0	0
12,5	-	13	0	0	0	0	0	0	0	0,007	0	0	0
13	-	13,5	0	0	0	0	0	0	0	0	0	0	0
13,5	-	14	0	0	0	0	0	0	0,002	0,002	0	0	0
>=		14	0	0	0	0	0	0	0	0	0	0	0

B CD-ROM

The following is enclosed on the CD-ROM

- C-code for wave force module
- C-code for RAO motion module
- Adams models
- Easy 5 models
- Results