

Master's degree thesis

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**A Generic Modelling Approach for Heavy Lifting Marine
Operation**

Jiafeng Xu

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MASTER THESIS 2014 FOR STUD. TECHN. JIAFENG XU

A Generic Modelling Approach for Heavy Lifting Marine Operation

Background

Vessel-crane simulation has always been a helpful tool for marine operation training, operation planning and product designing. Most designing works of crane and vessel require simulation results of their hydrodynamic behaviour, while most simulations so far regard crane and vessel isolated, without physical interaction with each other: the vessel always stays on its weight distribution equilibrium and the crane always experiences a given vessel motion no matter changes on loaded weight and operating posture. This issue often lowers the final performance of AHC (Active Heave Compensation) and ballast water control, hence worse positioning of the submerged load.

Objective

Research Study

- Lagrangian physics
- Maritime Guidance, Navigation and Control theory
- Specification rules
- Vibration dynamics
- 20SIM Modeling and programming skills
- CFD

Work

- Dynamic modelling of vessel, crane, cable, load and other appurtenance.
- Implementation of AHC control, joystick control
- Load hydrodynamics correlation analysis
- Integration of all sub-models
- Simulation test for different vessel, crane parameters
- Result analysis

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Supervision at Aalesund University College: Karl H. Halse,

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PREFACE

This thesis is submitted in partial requirements for a Master's Degree in Ship Design for the author. It contains work done from January to May 2014 at Aalesund University College. The supervisor of on the project has been Associate Professor Karl Henning Hals, Faculty of Maritime Technology and Operations (AMO). The thesis has been made solely by the author; some contents used in the project, however, is based on previous works of others. Thorough reference has been made accordingly.

The thesis project is part of the ongoing research in the Ship Operation Lab at Aalesund University College, whose activities support the ongoing development of the activities at the Offshore Simulator Centre (OSC). The research outline was introduced in one of the previous papers *Lifting Operations for Subsea Installations Using Small Construction Vessels and Active Heave Compensation Systems – A Simulation Approach* from Ship Operation Lab. The target is to develop generic modelling method for maritime simulation as a platform for both training and design purpose. The thesis project is also an upgrading of my earlier model developed in the summer of 2013.

The project is an interdisciplinary task requiring different aspects of knowledge. One of the biggest challenge in the research development is to integrate sub-systems from various sources into one compatible system. This is also the keypoint for the standardization and expansion of a generic modelling method. I would like to thank my supervisor Associate Professor Karl Henning Halse for his guidance and support throughout the entire thesis composition. The new ideas inspired and new fields of knowledge introduced by him has always been fruitful to me.

I would also like to thank Associate Professor Vilmar Ærøy for his assistance on 20-sim modelling and project ideas. I enjoy his teaching very much.

The gratitude also goes to Yingguang Chu for his crane model and assistance on 3D Mechanics modelling. The project is far from over and I am looking forward to our cooperation in the future.

Finally, the thesis is also to commemorate my amazing two years in Norway.

Aalesund University College

May. 2014

Jiafeng Xu

ABSTRACT

This thesis project introduced a generic modelling approach for heavy lifting marine operation based on 20-sim simulation and Matlab control. The model is a multi-body dynamic system which can be divided into vessel, crane, cable, load, and control system. Physical entities are modelled either in bond graph or directly using 3D Mechanics toolbox and connected by interactive power port. All control scheme is modelled as signal flow separated from physical entities. The vessel is modelled as 6-DOF bond graph using parameters from SHIPX data, then connected to the crane model inside 3D Mechanics unit. Crane model is controlled by outside manual/auto control scheme. Cable and load are modelled inside 3D Mechanics with hydrodynamic behaviour represented by actuators. The performance of each system is evaluated respectively by regulations and analysis. The simulation examples of different combinations of sub-systems are given at the end. The project is aiming at developing a generic modelling method serves as a multi-user training and design platform. Standardization and potential for upgrading are to be expected in the future.

Keywords: generic modelling, simulation, vessel-crane interaction, hydrodynamics, 20-sim.

CONTENTS

ASSIGNMENT.....	1
PREFACE.....	3
ABSTRACT.....	4
CONTENTS.....	5
LIST OF FIGURES.....	7
LIST OF TABLES.....	8
TERMINOLOGY.....	9
1 INTRODUCTION.....	10
1.1 BACKGROUND.....	10
1.2 MOTIVATION AND OBJECTIVE.....	10
1.3 ORGANIZATION OF THESIS.....	11
1.4 METHODOLOGY.....	11
1.5 LITERATURE AND PREVIOUS WORK.....	12
2 MULTI-BODY DYNAMICS.....	13
2.1 LAGRANIAN MECHANICS.....	13
2.1.1 <i>Lagrange Equations of the Second Kind</i>	13
2.1.2 <i>Kirchhoff Equations</i>	13
2.2 BOND GRAPH MODELLING.....	14
2.2.1 <i>Standard Bond Graph Modelling</i>	14
2.2.2 <i>Modified Bond Graph Modelling</i>	14
2.2.3 <i>20-sim and 3D Mechanics Toolbox</i>	14
3 VESSEL MODEL.....	17
3.1 REFERENCE FRAMES.....	17
3.1.1 <i>Reference Frame in ShipX</i>	17
3.1.2 <i>Reference Frame in 20-sim</i>	17
3.2 MOTION EQUATION.....	18
3.2.1 <i>Equation of Vessel Hydrodynamics</i>	18
3.2.2 <i>Coordinate Transformation</i>	19
3.3 BOND GRAPH MODEL.....	20
3.4 PARAMETERS.....	22
3.4.1 <i>Calculation in ShipX</i>	22
3.4.2 <i>MSS Post-process</i>	23
3.4.3 <i>20-sim Parameter Input</i>	23
3.5 MODEL ASSESSMENT.....	24
3.5.1 <i>Results</i>	24
3.5.2 <i>Discussion</i>	26
4 CRANE MODEL.....	27
4.1 DYNAMIC SYSTEM.....	27
4.1.1 <i>Solidworks Modelling</i>	27
4.1.2 <i>Converting Steps and Tips</i>	28
4.2 HYDRAULIC SYSTEM.....	28
4.2.1 <i>Main Derrick Sketch</i>	29
4.2.2 <i>20-sim Modelling</i>	29
4.3 CONTROL SYSTEM.....	30
4.3.1 <i>Manual Control</i>	30
4.3.2 <i>Auto Control</i>	31
5 CABLE MODEL.....	32
5.1 CABLE CHARACTERISTICS.....	32
5.2 CATEGORIES OF CABLE MODEL.....	32

5.2.1	<i>Method of Characteristics</i>	32
5.2.2	<i>Finite Element Methods</i>	33
5.2.3	<i>Linearization Method</i>	34
5.3	CABLE MODEL IN 3D MECHANICS.....	34
6	LOAD MODEL	37
6.1	LIFTING OPERATION	37
6.2	LOAD CFD.....	37
6.2.1	<i>Motion Function of the Load</i>	37
6.2.2	<i>Velocity & Acceleration Dependent Coefficient</i>	38
6.3	20-SIM MODELLING	38
6.4	MODEL ASSESSMENT	41
6.4.1	<i>DNV Rules Calculation</i>	41
6.4.2	<i>FEM in 3D Mechanics</i>	42
6.4.3	<i>Cable Parameters</i>	42
6.4.4	<i>Results</i>	43
6.4.5	<i>Discussion</i>	47
7	SIMULATION EXAMPLE	49
7.1	VESSEL + CRANE	49
7.1.1	<i>Model</i>	49
7.1.2	<i>Results</i>	50
7.1.3	<i>Discussion</i>	52
7.2	VESSEL + CRANE + CABLE + LOAD	52
7.2.1	<i>Model</i>	52
7.2.2	<i>Results</i>	54
7.2.3	<i>Discussion</i>	55
7.3	VESSEL + CRANE + CABLE + LOAD + AHC	56
7.3.1	<i>Model</i>	56
7.3.2	<i>Results</i>	56
7.3.3	<i>Discussion</i>	58
8	CONCLUSION & FUTURE WORK	59
8.1	CONCLUSION	59
8.2	FUTURE WORK.....	59
	REFERENCES	60
	APPENDIX	61
APPENDIX A	USER MANUAL	61
APPENDIX B	MATLAB SCRIPTS	62
APPENDIX C	20-SIM MODELS (DIGITAL)	64
APPENDIX D	VIDEO FOOTAGE OF SIMULATION (DIGITAL).....	64
APPENDIX E	20-SIM SCRIPTING LIBRARY	64

LIST OF FIGURES

FIGURE 1.1 WORLD CONSUMPTION (MILLIONS TONNES OIL EQUIVALENT) (BP 2013).....	10
FIGURE 2.1 20-SIM PANEL.....	15
FIGURE 2.2 3D MECHANICS PANEL.....	15
FIGURE 3.1 BODY-FIXED REFERENCE IN SHIPX AND 20-SIM (T=T1)	18
FIGURE 3.2 BOND GRAPH OF SIMPLE VESSEL.....	20
FIGURE 3.3 VESSEL MODEL IN 3D MECHANICS	21
FIGURE 3.4 ACTUATOR IN 3D MECHANICS TOOLBOX.....	22
FIGURE 3.5 USER PANEL FROM *.MAT FILE TO RUNNING 20-SIM MODEL.....	23
FIGURE 3.6 PARAMETERS PROCESS FROM SHIPX TO 20-SIM.....	24
FIGURE 3.7 MOTION RAO FROM 20-SIM	25
FIGURE 3.8 MOTION RAO FROM SHIPX.....	25
FIGURE 3.9 ERROR BETWEEN 20-SIM AND SHIPX.....	26
FIGURE 4.1 SOLIDWORKS MODEL OF SIMPLIFIED ROLLS-ROYCE PSV-100.....	27
FIGURE 4.2 STEPS FROM SOLIDWORKS TO 20-SIM	28
FIGURE 4.3 CRANE MODEL IN 3D MECHANICS	28
FIGURE 4.4 SIMPLIFIED MAIN DERRICK HYDRAULIC SKETCH.....	29
FIGURE 4.5 BOOM CYLINDER HYDRAULICS MODEL.....	29
FIGURE 4.6 BOOM CYLINDER SETUP IN 3D MECHANICS.....	30
FIGURE 4.7 CRANE TIP VELOCITY CONTROL	31
FIGURE 4.8 DIRECT CYLINDER VELOCITY CONTROL	31
FIGURE 4.9 CYLINDER VELOCITY INPUT AND PID CONTROL.....	31
FIGURE 4.10 COMPENSATING MOTION/VELOCITY ON OPPOSITE DIRECTION.....	31
FIGURE 5.1 FOUR TYPES OF FEM CABLE MODEL.....	33
FIGURE 5.2 FEM CABLE MODEL IN 3D MECHANICS.....	34
FIGURE 5.3 EXTERNAL FORCE ACTING AS ACTUATOR	35
FIGURE 5.4 CABLE MODEL (N=3, L=10).....	35
FIGURE 5.5 EXTERNAL FORCE INPUT.....	36
FIGURE 6.1 CFD EXPERIMENT TO TEST & VERIFY VELOCITY-ACCELERATION DEPENDENT COEFFICIENT	38
FIGURE 6.2 LOAD MODEL IN 3D MECHANICS.....	38
FIGURE 6.3 BUOYANCY FORCE ACTUATOR	39
FIGURE 6.4 ADDED MASS AND DAMPING FORCE ACTUATOR	40
FIGURE 6.5 CABLE + LOAD MODEL.....	41
FIGURE 6.6 HORIZONTAL OFFSET WITH CURRENT FORCE ON LOAD V=1M/S.....	43
FIGURE 6.7 HORIZONTAL OFFSET WITH CURRENT FORCE ON LOAD V=2M/S.....	44
FIGURE 6.8 HORIZONTAL OFFSET WITH CURRENT FORCE ON LOAD V=3M/S.....	44
FIGURE 6.9 HORIZONTAL OFFSET WITH CURRENT FORCE ON LOAD V=4M/S.....	45
FIGURE 6.10 HORIZONTAL OFFSET WITHOUT CURRENT FORCE ON LOAD V=1M/S.....	45
FIGURE 6.11 HORIZONTAL OFFSET WITHOUT CURRENT FORCE ON LOAD V=2M/S.....	46
FIGURE 6.12 HORIZONTAL OFFSET WITHOUT CURRENT FORCE ON LOAD V=3M/S.....	46
FIGURE 6.13 HORIZONTAL OFFSET WITHOUT CURRENT FORCE ON LOAD V=4M/S.....	47
FIGURE 7.1 SIMULATION EXAMPLE: VESSEL + CRANE.....	49
FIGURE 7.2 ANIMATION: VESSEL + CRANE	49
FIGURE 7.3 VESSEL MOTION: VESSEL + CRANE AT TRANSIENT STATE	50
FIGURE 7.4 VESSEL MOTION: VESSEL + CRANE ROLL AT STEADY STATE	51
FIGURE 7.5 VESSEL MOTION: VESSEL + CRANE FROM SHIPX.....	51
FIGURE 7.6 SIMULATION EXAMPLE: VESSEL + CRANE + CABLE + LOAD.....	52
FIGURE 7.7 ANIMATION: VESSEL + CRANE + CABLE + LOAD (100 T).....	53
FIGURE 7.8 ANIMATION: DYNAMIC BEHAVIOUR OF CABLE + LOAD	53
FIGURE 7.9 VESSEL MOTION: VESSEL + CRANE + CABLE + LOAD (50 T) AT TRANSIENT STATE.....	54
FIGURE 7.10 VESSEL MOTION: VESSEL + CRANE + CABLE + LOAD (50 T) ROLL AT STEADY STATE	54
FIGURE 7.11 VESSEL MOTION: VESSEL + CRANE + CABLE + LOAD (100 T) ROLL AT STEADY STATE	55
FIGURE 7.12 SIMULATION EXAMPLE: VESSEL + CRANE + CABLE + LOAD + AHC	56
FIGURE 7.13 LOAD & PEDESTAL HEAVE: VESSEL + CRANE + CABLE + LOAD (100 T).....	57
FIGURE 7.14 LOAD & PEDESTAL HEAVE: VESSEL + CRANE + CABLE + LOAD + AHC (100 T).....	58

LIST OF TABLES

TABLE 2.1 EFFORT AND FLOW VARIABLES IN DIFFERENT PHYSICAL DOMAINS..... 14
TABLE 3.1 VESSEL AND WAVE DATA..... 24
TABLE 5.1 COMPARISON OF FEM CABLE MODELS 33
TABLE 6.1 6X36 WS+IWRC PARAMETERS..... 42
TABLE 6.2 PARAMETERS OF CABLE AND LOAD 42
TABLE 6.3 VERTICAL OFFSET WITH LIFTED LOAD [M] 43

TERMINOLOGY

A	added mass
B	damping tensor
C	restoring force tensor
C_a	added mass coefficient
C_d	drag force coefficient
C_A	coriolis and centripetal forces of added mass
C_{RB}	coriolis and centripetal forces of rigid body
g	gravitational acceleration
\tilde{I}	moment of inertia tensor
J_b^h	rotation matrix for 6 degree of freedom
L	lagrangian function
M	inertia tensor
m	mass
ρ	density
$p.e$	effort in bond graph
$p.f$	flow in bond graph
R_b^h	rotation matrix for linear velocity
r_G	centre of gravity vector
T	kinetic energy
T_b^h	rotation matrix for angular velocity
t	time
τ	external force
U	velocity of the submerged object
v	linear velocity
V	potential energy
ω	angular velocity
η	motion displacement

1 INTRODUCTION

1.1 Background

Modern human civilization is driven by the gigantic energy consumption majorly dependent on fossil fuels discovery and production, with a market share of 87%. Oil remains the world's leading fuel, accounting for 33.1% of global energy consumption, where natural gas follows by 24.2%.

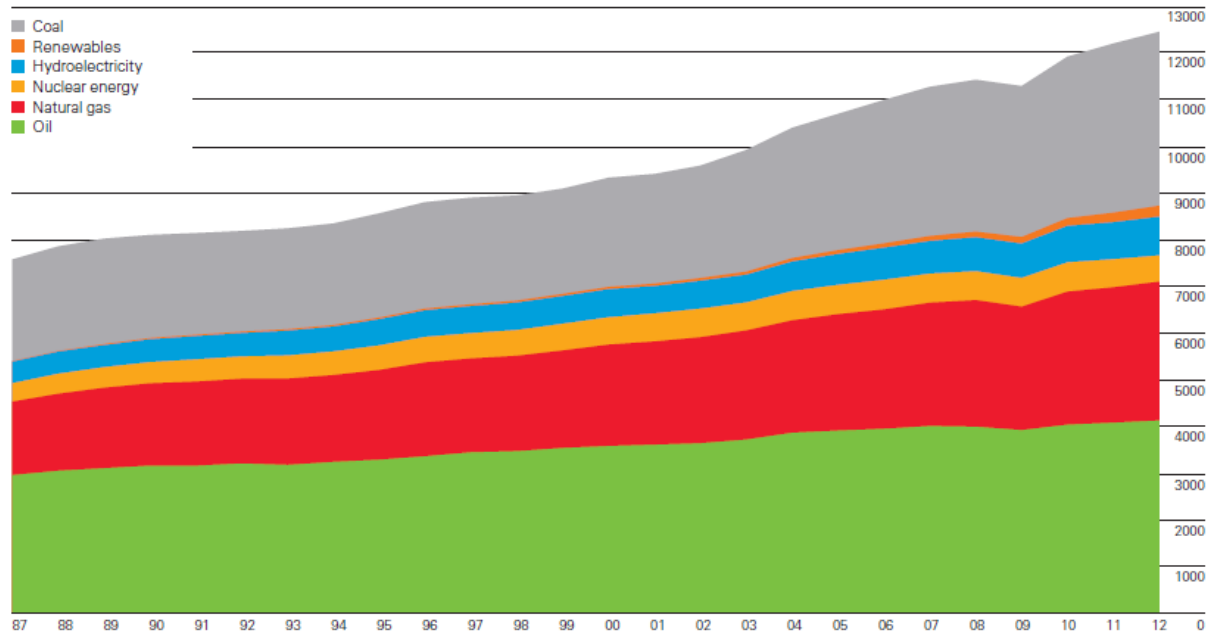


Figure 1.1 World Consumption (Millions tonnes oil equivalent) (BP 2013)

Currently, approximately 30% of world oil and gas production comes from offshore and a continuous increase is expected in the future.

The Petroleum/Oil & Gas sector is Norway's largest industry, accounting for 22% of national value creation. By 2010, Norway has produced and delivered about 40% of the expected total oil & gas resources on the Norwegian continental shelf. While 35% are reserves yet to be developed, 25% are undiscovered resources, two thirds of which are expected to be gas and one third oil (Norwegian Petroleum Directorate 2010). The easiest barrels have been found and produced, so that the way ahead will be demanding in terms of expertise, technology and costs.

Over the last six decades, offshore industry has witnessed a tremendous development. Its activities has been expanded from conventional fossil fuel production to areas such as renewable energy development, telecommunication infrastructure construction, fisheries installation and maritime search and rescue. The worksite have also been expanded to deeper water thanks to the advancement in technology. Fixed platforms were initially used for the offshore development, but as the fields have gone deeper, floating production facilities have become the main solution for the offshore production. Also for safety and cost concerns, subsea platform, pipeline and ROV (Remotely Operated Vehicles) are increasingly being applied into offshore operations. Higher requirement for precision, accuracy and operability are thusly being raised towards OSV (Offshore Supply Vessel) and its lifting device, along with other relevant equipment on-board.

1.2 Motivation and Objective

Marine operations are usually multi-system involved activities with interaction and coordination behind. Heavy lifting operation is a typical example often used for ROV deployment and subsea installation /demolition, which normally involves vessel, crane, cable, load and the manual/auto control system. Because of the complexity and diverse research focus of its nature, people tend to isolate the issue by neglecting the insignificant and simplifying the complicated, e.g. the researchers who intend to study the sea-keeping feature of the PSV normally regard the dynamic behaviour of the crane and the load as negligible (Lloyd 1989), while people who study the AHC control treat vessel motion as unaffected

by the crane hence an independent variable outside the equation. Moreover, people who study the hydrodynamic behaviour of a submerged load usually only apply simple motion to the load in CFD calculation (Fackrell 2011) or experiment. Also, cables are sometimes a one dimension spring characterized with only heave motion.

However, ordinary computers have long reached the capability of processing complex mathematical model of a marine operation, if not in real-time at least. Bond graph modelling technique enables a generic approach to build a physical system whereas control system can be modelled by traditional signal units. Also, the calculated results can be demonstrated by 3D visualization. Therefore, a system integrated with major subsystems can now be established and simulated, expectedly to have a better approximation of the reality.

This thesis introduces a generic modelling approach for heavy lifting operation by using bond graph technique, signal blocks and 3D mechanics toolbox in simulation software 20-sim. The input data resources are from ShipX (strip theory), DNV rules and CFD calculation, managed in MATLAB panel and communicated with 20-sim. A fully interactive system of vessel, crane, cable, and control system is realized and able for easy parameter selection. If real-time simulation can be reached, manual control can also be added as well as auto control.

The assembled system shall have a better approximation to the reality also ensure the functionality of all sub-systems. Thus simulation shall be monitored in 3D visualization and digital records. Analysis shall also be made comparing simulation results with regulations and experiments.

Ultimately, the model shall have the potential for quick adjustment and compatibility for simulators. Different sub-system shall be able to be developed by different parties with standardized interface and 'Plug and Play' function.

1.3 Organization of Thesis

The thesis is mainly divided into three parts. Firstly, the principle of Lagrangian mechanics and bond graph modelling technique are introduced, which are applied in multi-body dynamics modelling in 20-sim. Secondly, detailed modelling method of vessel, crane, cable, load, control system and data panel are introduced separately with argument of modifications from previous works and interfaces between sub-systems. Alternative methods and potential improvements are also discussed in each chapter, followed by evaluation of each subsystem's performance. Finally, simulation examples of different combinations of sub-systems are given with arguments supported by simulation results. A user manual is also given as reference.

1.4 Methodology

The main method used in this thesis is virtual simulation and the primary target of this thesis is to build a realistic real-time simulation model for marine operation. As for economic and safety reasons, a full size experiment of marine operation is practically impossible for the industry. A virtual environment of maritime operation shall thusly be developed in the computer which allows users to interact with. Physical objects are being represented as components with different functionality in the system. The system can accept input from the user (e.g., wave spectrum, vessel inertia matrix, hydrodynamic damping parameters, crane model, CFD data of the load) and produce output to the user (e.g., vessel motion during the operation, crane hydraulics performance, load motion in the water, power output, mechanical behaviours). The model is built based on studies about realistic feature of the vessel and hydrodynamic statistics, and coded according to general physical principles with simplification for faster performance. The parameters the model chooses depend on field measurement but are also tuned for better approximation to the reality. Each sub-system is modelled separately whose accuracy and applicability are tested before being integrated into bigger system. During the simulation, variables are controlled to test the influence of each. Although virtual simulation has the flexibility and compatibility to adjust and to replace input with minimum cost and maximum fidelity, it always stands on certain assumptions that loses reality in some extent. Thusly simulation results are recorded and analysed both theoretically and statistically. DNV rules checking will be conducted to make sure its correspondence with legitimate requirements.

1.5 Literature and Previous Work

Currently there are many types of marine simulators on the market. Kongsberg Maritime provides Offshore Vessel Simulator for education and procedure training of navigators and winch-operators of anchor-handling vessels, along with the MasterLift™ (ML) line of crane simulators, dynamic positioning simulators, liquid cargo handling simulator, ships bridge simulators, etc. NAUTIS provides DNV certified maritime training solutions for the military & civilian maritime industry. TRANSAS provides navigational simulators, GMDSS simulator, engine room and cargo handling simulators, crane simulators and simulator development tools. CSMART provides two full mission bridge simulators, six part-task bridge simulators, with the ability to simulate fixed propeller and azipod simulation. Prof. Thro I. Fossen and Prof. Tristan Perez from NTNU have also developed a Marine System Simulator (MSS) tool box in Matlab/Simulink library, which includes models for ships, underwater vehicles, and floating structures. The library also contains guidance, navigation, and control (GNC) blocks for real-time simulation.

This thesis is part of the ongoing research in the Ship Operation Lab in Aalesund University College, whose activities support the ongoing development of the activities at the Offshore Simulator Centre (OSC). The model consists of several modelling methods from other researchers yet with necessary modification to make a compatible system. The general second order differential equation of vessel hydrodynamics is from (Fossen and Fjellstad 2011). MATLAB files for ShipX data transformation is from (Fossen and Perez 2014). Bond graph modelling technique is from (Pedersen 2008). Crane hydraulics and control model is from (Chu 2013). Cable model inspired by (Johansson 1976). Load model inspired by (Halse, et al. 2014)

2 MULTI-BODY DYNAMICS

2.1 Lagrangian Mechanics

Lagrangian mechanics is a re-formulation of classical mechanics using the principle of stationary action (also called the principle of least action) (Goldstein 2001). Lagrangian function is the core of Lagrangian mechanics, where the kinetic and potential energies of the constituents of the system in a generalized coordinates form the equation defined as

$$L = T - V$$

Where T is the total kinetic energy and V is the total potential energy of the system (Torby 1984).

2.1.1 Lagrange Equations of the Second Kind

For any system with m degrees of freedom, the Lagrange equations include m generalized coordinates and m generalized velocities. Euler-Lagrange equations, also known as the Lagrange equations of the second kind (Euler 1744) state that for a holonomic system

$$L(q, \dot{q}, t)$$

$$\frac{d}{dt} \left(\frac{\partial L}{\partial \dot{q}_j} \right) - \frac{\partial L}{\partial q_j} = 0$$

where $j = 1, 2, \dots, m$ represents the j th degree of freedom, q_j are the generalized coordinates and \dot{q}_j are the generalized velocities. The equations can be applied into multi-body dynamics modelling where Newton Laws of each physical body can be expressed in its own body coordinates.

2.1.2 Kirchhoff Equations

In fluid dynamics, the Kirchhoff equations, named after Gustav Kirchhoff, is derived from generalized Lagrangian equations, describing the motion of a rigid body in an ideal fluid. (Kirchhoff 1877)

$$T = \frac{1}{2} (\vec{\omega}^T I \vec{\omega} + m \vec{v}^2)$$

$$\frac{d}{dt} \frac{\partial T}{\partial \vec{\omega}} + \vec{\omega} \times \frac{\partial T}{\partial \vec{\omega}} + \vec{v} \times \frac{\partial T}{\partial \vec{v}} = \vec{Q}_h + \vec{Q}$$

$$\frac{d}{dt} \frac{\partial T}{\partial \vec{v}} + \vec{\omega} \times \frac{\partial T}{\partial \vec{v}} = \vec{F}_h + \vec{F}$$

$$\vec{Q}_h = - \int p \vec{x} \times \hat{n} d\sigma$$

$$\vec{F}_h = - \int p \hat{n} d\sigma$$

Where:

$\vec{\omega}$ and \vec{v} are angular and linear velocity vectors at point \vec{x} respectively.

I is the moment of inertia tensor, m is body's mass.

\hat{n} is a unit normal to the surface of the body at point \vec{x} .

p is a pressure at the point \vec{x} .

\vec{Q}_h and \vec{F}_h are the hydrodynamic torque and force acting on the body respectively.

\vec{Q} and \vec{F} likewise denote all other torques and forces acting on the body.

The integration is performed over the fluid-exposed portion of the body surface.

2.2 Bond Graph Modelling

Bond graph is an explicit graphical tool for capturing the common energy structure of systems. Complex systems can be described concisely by bond graph in vector form, and the notations of causality provides a tool not only for formulation of system equations, but also for intuition based discussion of system behaviour, such as controllability, observability, fault diagnosis, etc. (Bond Graph 2014)

2.2.1 Standard Bond Graph Modelling

Readers hereby are presumed to have basic knowledge about bond graph modelling, a brief introduction of standardized bond graph modelling are given as follows.

The language of bond graphs expresses general class physical systems through power interactions. The factors of power, Effort and Flow, have different interpretations in different physical domains.

In the following table, effort and flow variables in some physical domains are listed.

Table 2.1 Effort and Flow Variables in Different Physical Domains

Systems	Effort (e)	Flow (f)
Mechanical	Force (F)	Velocity (v)
	Torque (t)	Angular velocity (w)
Electrical	Voltage (V)	Current (i)
Hydraulic	Pressure (P)	Volume flow rate (dQ/dt)
Thermal	Temperature (T)	Entropy change rate (ds/dt)
	Pressure (P)	Volume change rate (dV/dt)
Chemical	Chemical potential (m)	Mole flow rate (dN/dt)
	Enthalpy (h)	Mass flow rate (dm/dt)
Magnetic	Magneto-motive force (em)	Magnetic flux (f)

The standard elements in bond graphs include but are not limited to: R-Elements, C-Elements, I-Elements, Effort and Flow Sources, Transformer, Gyration, 1 junctions, 0 junctions. Each element is designed to be coded representing specific relations between Effort and Flow, with only variation in parameters and vector size.

2.2.2 Modified Bond Graph Modelling

In conventional bond graph modelling, each elements are fixated and constrained by its own function. But thanks to software like 20-sim and SimulationX, bond graph elements are more flexible and in some extent can be regarded as a coding language. For example, several elements can be modelled in a single block with extra power ports and equations. New types of element can also be invented and named by user's preference. In this thesis, three translational degrees of freedom and three rotational degrees of freedom of the vessel and for other objects are being treated homogeneously as six elements in motion vectors. Standard elements such as R-element, C-element and I-element are being recoded to cope the expanded expression of degree of freedom. Other Transformers and user defined elements will be explained in the following chapters. Names of each element are simply to reveal the functionality of which in real physical system, for reader's convenience.

2.2.3 20-sim and 3D Mechanics Toolbox

20-sim is a commercial modelling and simulation software developed by Controllab Products B.V. It enables user to enter model graphically, similar to drawing an engineering scheme, and to simulate and analyse the behaviour of multi-domain dynamic systems and create control systems. It also provides tools that allow users to create models using equations, block diagrams, physical components and bond graphs. C-code are also compatible in this software to use on hardware for rapid prototyping and HIL-simulation.

3D mechanics toolbox is a packaged module in 20-sim for multi-body system modelling in a 3D workspace. Rigid bodies are represented by 3D graphics and physical properties, connected by joints, monitored by sensors and actuated by actuators. 3D mechanics toolbox can generate its model into 20-sim code ready for bond graph connection. Forces can hence be applied to the joints or on each body directly, and 3D mechanics toolbox itself provides options to define springs and dampers to the joint. Finally, the complete model can be shown by 3D animation toolbox.

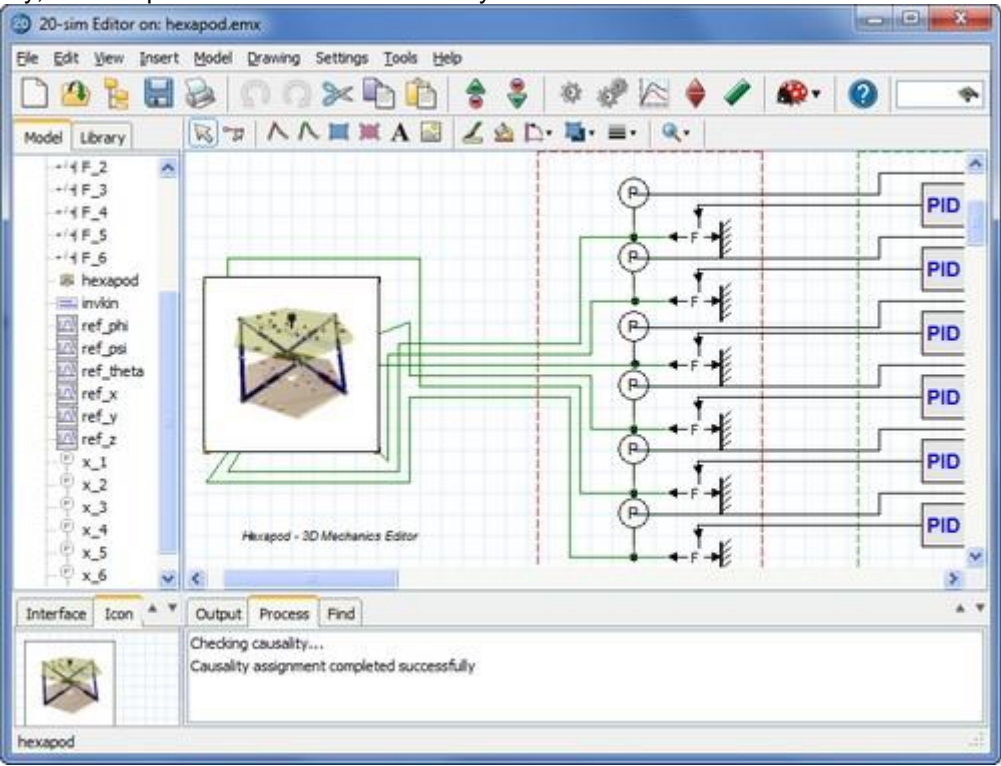


Figure 2.1 20-sim Panel

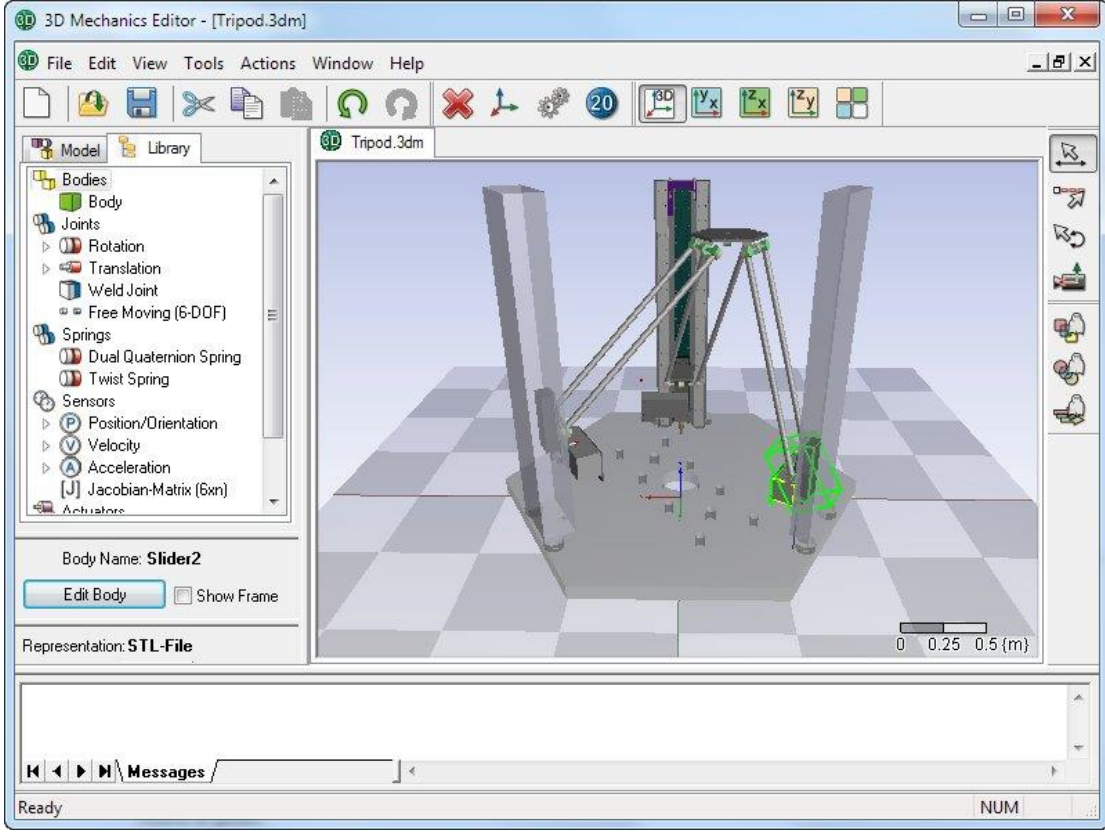


Figure 2.2 3D Mechanics Panel

Comparing with direct bond graph modelling (Fagereng 2011), which is unintuitive but with less code and consumes less computer memory, 3D Mechanics Toolbox generates enormous lines of code and computer resources. However the biggest advantage of 3D Mechanics is its user friendly interface and modelling accuracy. For efficiency and feasibility purpose, 3D Mechanics Toolbox is used in this thesis.

3 VESSEL MODEL

The following chapter is based on Fossen's equation of vessel hydrodynamics (Fossen and Fjellstad 2011) and Pederson's bond graph model (Pedersen 2008).

3.1 Reference Frames

Following the guiding spirit of Lagrangian Mechanics, it is convenient to use different reference frames in each particular case. In this thesis, the Earth surface is assumed to be flat in the range of where the vessel moves. Only three right-handed orthogonal reference frames are used in this thesis.

- **NED:** The North-east-down frame $E(x_e, y_e, z_e)$, is fixed to the Earth. The positive x_e -axis points towards the North, the positive y_e -axis towards the East and the positive z_e -axis towards the centre of the Earth. The origin, o_n , is located on mean water free surface and coincides with body-fixed coordinate system $B(x_b, y_b, z_b)$ at time t_0 . This frame is considered inertial because the force acting on the vessel due to the Earth rotation is negligible compared to the hydrodynamic forces acting on the vessel.
- **Body-fixed:** The body-fixed coordinate system $B(x_b, y_b, z_b)$ is fixed to the hull. The directions of the positive axes are: x_b -axis (forward); y_b -axis (starboard); z_b -axis (downward). The origin o_b at the vessel's water plane and the z_b -axis pointing through CoG.
- **Hydrodynamic:** The hydrodynamic frame $H(x_h, y_h, z_h)$ is not fixed to the hull; it moves at the average speed of the vessel following its path. The x_h - y_h plane is located on mean water free surface and the origin o_h coincide with o_b at vessel's stationary status. The positive x_h -axis points forward and it is aligned with the low-frequency yaw angle. The positive y_h -axis points towards starboard and the positive z_h -axis points downwards. In this thesis, the hydrodynamic frame moves with the harmonic motion of surge, sway and heave that cannot be considered inertial. But the wave induced hydrodynamic force is simplified as constant in hydrodynamic frame which means the vessel is assumed to take a straight course (low-frequency yaw angle is zero) so that wave heading change is negligible, and so does encounter wave period change due to surge/sway.

3.1.1 Reference Frame in ShipX

In this simulation, all vessel data directly comes from ShipX calculation. ShipX is possible to compute the hydrodynamic data in the following points:

CoG – centre of gravity

CO – same as o_b , in the still water plane line on the centreline with the z -axis pointing through CG.

The preferred point is o_b because it coincides with o_n at time t_0 . The ShipX axes are x -axis (backward); y -axis (starboard); z -axis (upward).

3.1.2 Reference Frame in 20-sim

In 20-sim, the NED $E(x_e, y_e, z_e)$ coincides the hydrodynamic reference frame $H(x_h, y_h, z_h)$ at time t_0 . The axes are x -axis (forward); y -axis (starboard); z -axis (downward), the same as in Marine control toolbox. It is different from ShipX axes because all ShipX data must be transformed by Marine control toolbox (Fossen and Perez 2014) into MATLAB data file before being input in to 20-sim.

In 20-sim 3D animation, the world reference only support a camera view of z -axis upwards, so the world reference shall be ignored and the direction of the Earth-fixed reference is determined by the stationary position of body-fixed $B(x_b, y_b, z_b)$ at time t_0 .

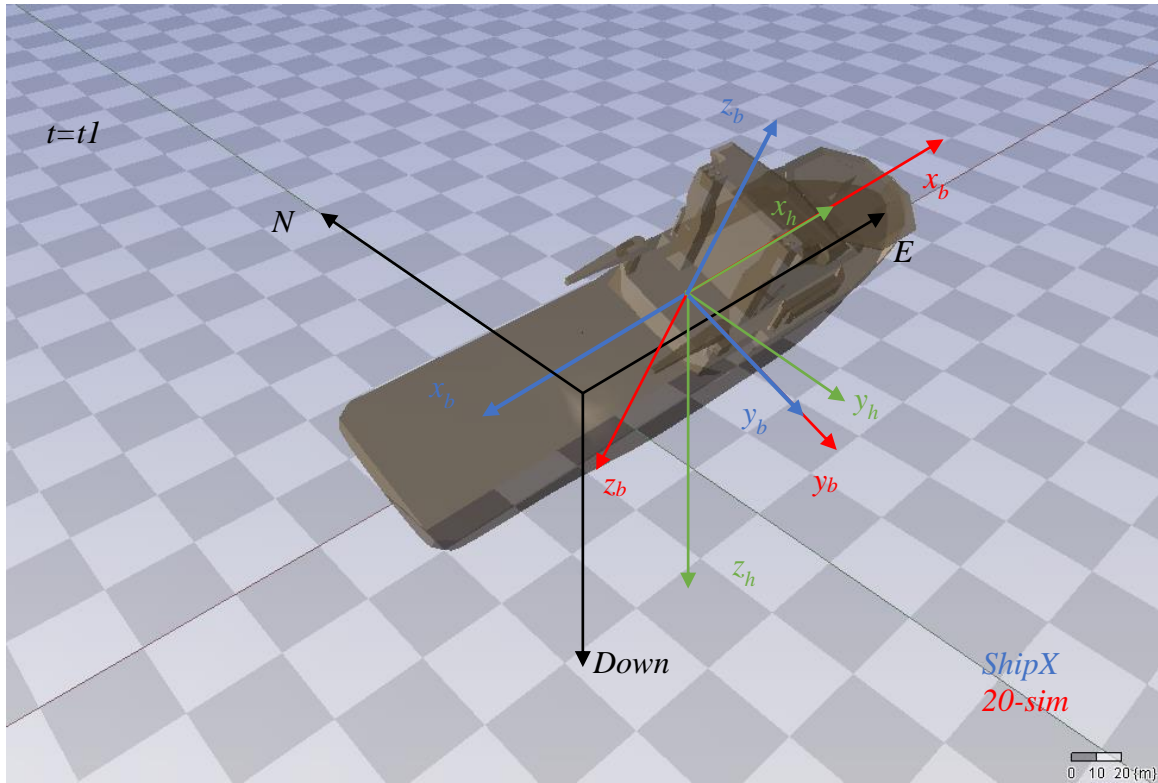


Figure 3.1 Body-fixed reference in ShipX and 20-sim ($t=t1$)

3.2 Motion Equation

3.2.1 Equation of Vessel Hydrodynamics

Fossen presented the hydrodynamic behaviour of a vessel in traditional quadratic linear differential equations by utilizing the Kirchhoff equations (Fossen and Fjellstad 2011).

Take the second and third equations from Kirchhoff equations

$$\frac{d}{dt} \frac{\partial T}{\partial \vec{\omega}} + \vec{\omega} \times \frac{\partial T}{\partial \vec{\omega}} + \vec{v} \times \frac{\partial T}{\partial \vec{v}} = \vec{Q}_h + \vec{Q}$$

$$\frac{d}{dt} \frac{\partial T}{\partial \vec{v}} + \vec{\omega} \times \frac{\partial T}{\partial \vec{v}} = \vec{F}_h + \vec{F}$$

Also we have two equations from Momentum Conservation Principle.

$$\frac{\partial T}{\partial \vec{v}} = m \cdot \vec{v} - m \cdot \vec{r}_G \times \vec{\omega} = m \cdot \vec{v} - (m \times \vec{r}_G) \cdot \vec{\omega}$$

$$\frac{\partial T}{\partial \vec{\omega}} = -m \cdot (\vec{v} \times \vec{r}_G) + \vec{I} \cdot \vec{\omega} = (m \times \vec{r}_G) \cdot \vec{v} + \vec{I} \cdot \vec{\omega}$$

Restate into standard form of the equation of motion.

$$\mathbf{M}(m, r_G, \vec{I}) \cdot \dot{\vec{\eta}} + \mathbf{C}'(M, \vec{U}) \cdot \dot{\vec{\eta}} = \vec{\tau}_h + \vec{\tau}$$

Where $\vec{\eta} = [v, \omega]^T$, $\vec{\tau}_h = [F_h, Q_h]^T$ and $\vec{\tau} = [F, Q]^T$.

The inertia matrix M is recognized to be

$$\mathbf{M} = \begin{bmatrix} m & -m \times r_G \\ m \times r_G & \tilde{I} \end{bmatrix}$$

And the remaining terms which are known to contribute the Coriolis and centripetal forces can be state as

$$[a_1, a_2, a_3, b_1, b_2, b_3]^T = \mathbf{M} \cdot \dot{\boldsymbol{\eta}}$$

$$\mathbf{C}' = - \begin{bmatrix} 0 & 0 & 0 & 0 & -a_3 & a_2 \\ 0 & 0 & 0 & a_3 & 0 & -a_1 \\ 0 & 0 & 0 & -a_2 & a_1 & 0 \\ 0 & -a_3 & a_2 & 0 & -b_3 & b_2 \\ a_3 & 0 & -a_1 & b_3 & 0 & -b_1 \\ -a_2 & a_1 & 0 & -b_2 & b_1 & 0 \end{bmatrix}$$

The hydrodynamic force can also be separated into two components.

$$\boldsymbol{\tau}_h = \boldsymbol{\tau}_{rad} + \boldsymbol{\tau}_{exc}$$

Where $\boldsymbol{\tau}_{rad}$ is the hydrodynamic radiation forces and $\boldsymbol{\tau}_{exc}$ is environmental excitation forces from wind and waves. The radiation forces can be expressed in frequency domain as follows.

$$\boldsymbol{\tau}_{rad} = -\mathbf{A}(\omega)\ddot{\boldsymbol{\eta}} - \mathbf{B}(\omega)\dot{\boldsymbol{\eta}} - \mathbf{C}\boldsymbol{\eta}$$

Where $\mathbf{A}(\omega)$ and $\mathbf{B}(\omega)$ are the frequency – dependent added mass and damping matrices. The restoring forces are assumed to be a linear formulation $\mathbf{C}\boldsymbol{\eta}$. The resulting motion can be stated as $\boldsymbol{\eta} = \hat{\boldsymbol{\eta}}e^{i\omega t}$ and the dynamic equation in frequency domain becomes

$$-\omega^2[\mathbf{M} + \mathbf{A}(\omega)] \cdot \boldsymbol{\eta}(j\omega) + j\omega\mathbf{B}(\omega) \cdot \boldsymbol{\eta}(j\omega) + \mathbf{C} \cdot \boldsymbol{\eta}(j\omega) = \boldsymbol{\tau}_{exc} + \boldsymbol{\tau}$$

Naval architects usually write the equation in a mixed frequency – time domain formulation

$$(\mathbf{M}_{RB} + \mathbf{M}_A(\omega)) \cdot \ddot{\boldsymbol{\eta}}(t) + (\mathbf{C}_{RB} + \mathbf{C}_A(\omega)) \cdot \dot{\boldsymbol{\eta}}(t) + \mathbf{B}(\omega) \cdot \dot{\boldsymbol{\eta}}(t) + \mathbf{C} \cdot \boldsymbol{\eta}(t) = \boldsymbol{\tau}_{exc} + \boldsymbol{\tau}$$

Strictly speaking, this equation is not correct because the real vessel motion is not purely harmonic, but it is tolerated in a dominated frequency motion.

The excitation forces include forces from wave, current and wind can be derived from force RAO in ShipX and other forces from appurtenances such as rudder, propeller, foil and fin stabilizer can be either assumed or through classical calculation.

The motion equation is established in body-fixed coordinate system $B(x_b, y_b, z_b)$ in order to keep the rigid body inertia and added mass as constants. The restoring force and damping force are treated with respect to the hydrodynamic frame $H(x_h, y_h, z_h)$. Therefore a coordinate transformation is required before those two elements are being put into the equation.

3.2.2 Coordinate Transformation

To transform hydrodynamic excitation and restoring forces from $H(x_h, y_h, z_h)$ to $B(x_b, y_b, z_b)$, a coordinate transformation matrix is established. The principle rotation matrices (one axis rotations) can be obtained by setting x , y and z axes rotation as:

$$\mathbf{R}_{x,\phi} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\phi & -\sin\phi \\ 0 & \sin\phi & \cos\phi \end{bmatrix}$$

$$\mathbf{R}_{y,\theta} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\theta & -\sin\theta \\ 0 & \sin\theta & \cos\theta \end{bmatrix}$$

$$\mathbf{R}_{z,\psi} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\psi & -\sin\psi \\ 0 & \sin\psi & \cos\psi \end{bmatrix}$$

The rotation matrix for linear velocity is

$$\mathbf{R}_b^h = \mathbf{R}_{x,\phi} \cdot \mathbf{R}_{y,\theta} \cdot \mathbf{R}_{z,\psi}$$

The rotation matrix for angular velocity is

$$\mathbf{T}_b^h = \begin{bmatrix} 1 & \sin\phi \cdot \tan\theta & \cos\phi \cdot \tan\theta \\ 0 & \cos\phi & -\sin\phi \\ 0 & \sin\phi/\cos\theta & \cos\phi/\cos\theta \end{bmatrix}$$

The overall 6DOF kinematic equation between $H(x_h, y_h, z_h)$ and $B(x_b, y_b, z_b)$ is:

$$\dot{\boldsymbol{\eta}}_h = \mathbf{J}_b^h \cdot \dot{\boldsymbol{\eta}}_b; \quad \mathbf{J}_b^h = \begin{bmatrix} \mathbf{R}_b^h & \mathbf{0}_{3 \times 3} \\ \mathbf{0}_{3 \times 3} & \mathbf{T}_b^h \end{bmatrix}$$

And the final equation used in bond graph model is

$$\begin{aligned} (\mathbf{M}_{RB} + \mathbf{M}_A(\omega)) \cdot \ddot{\boldsymbol{\eta}}(t) + (\mathbf{C}_{RB} + \mathbf{C}_A(\omega)) \cdot \dot{\boldsymbol{\eta}}(t) + \mathbf{J}^{-1h} \cdot \mathbf{B}(\omega) \cdot \mathbf{J}_b^h \cdot \dot{\boldsymbol{\eta}}(t) + \mathbf{J}^{-1h} \cdot \mathbf{C} \cdot \int \mathbf{J}_b^h \cdot \boldsymbol{\eta}(t) \\ = \mathbf{J}^{-1h} \cdot \boldsymbol{\tau}_{exc} + \boldsymbol{\tau} \end{aligned}$$

3.3 Bond Graph Model

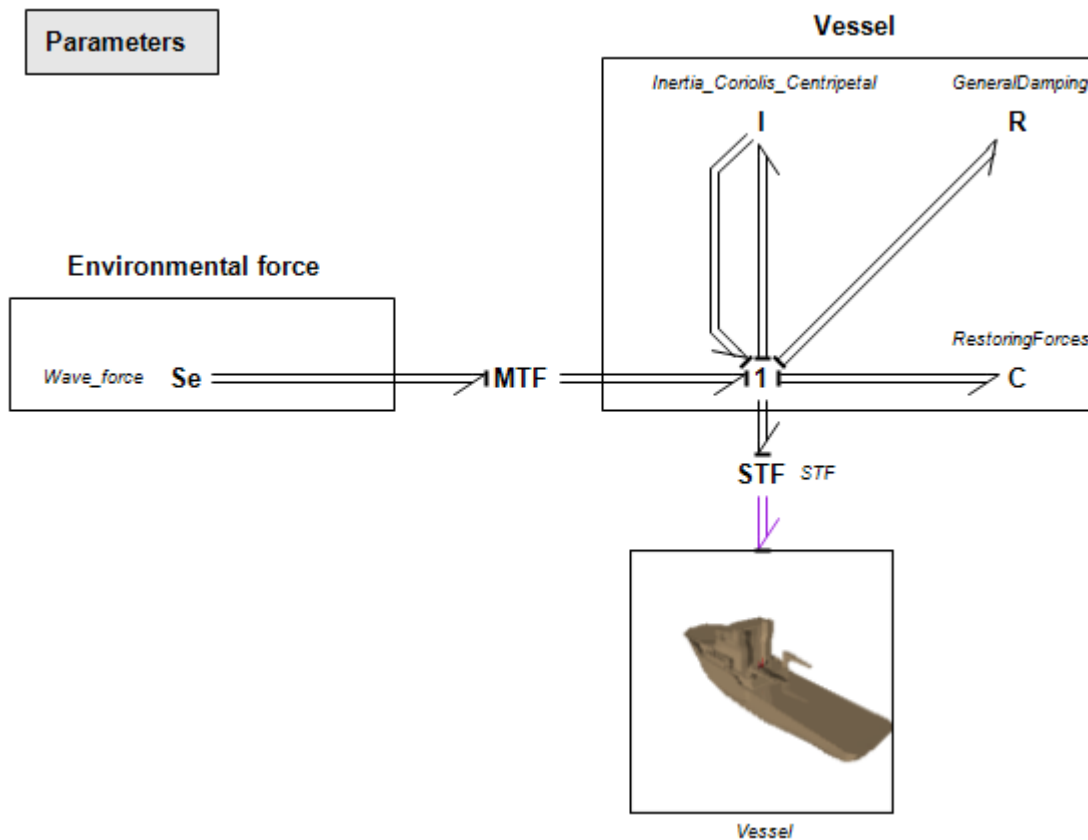


Figure 3.2 Bond Graph of Simple Vessel

3D Mechanics toolbox does not have the option for a full inertia/damping/spring tensor property, so all elements of ship motion equation are defined outside the 3D Mechanics block.

The **I** element in the model represents the Inertial force and Coriolis force:

$$(\mathbf{M}_{RB} + \mathbf{M}_A(\omega)) \cdot \dot{\boldsymbol{\eta}}(t) + (\mathbf{C}_{RB} + \mathbf{C}_A(\omega)) \cdot \dot{\boldsymbol{\eta}}(t)$$

The **R** element represents the damping force:

$$\mathbf{J}_b^{-1h} \cdot \mathbf{B}(\omega) \cdot \mathbf{J}_b^h \cdot \dot{\boldsymbol{\eta}}(t)$$

The Effort Source **Se** and a **MTF** acting as coordinate transformation Jacobian matrix. The Jacobian matrix is set as a global variable J . Only wave force is included in this model, but other hydrodynamic forces can be added by the same modelling approach.

$$\mathbf{J}_b^{-1h} \cdot \boldsymbol{\tau}_{exc}$$

The **C** element represents the restoring force with Jacobian matrix J in the code:

$$\mathbf{J}_b^{-1h} \cdot \mathbf{C} \cdot \int \mathbf{J}_b^h \cdot \boldsymbol{\eta}(t)$$

The vessel in the 3D Mechanics block has a negligible inertia, thusly can be called as shadow vessel. By connecting the bond graph to an actuator located on the origin of body-fixed coordinate system, the outside bond graph is acting equivalent to a Flow Source to determine the vessel motion inside the 3D Mechanics block.

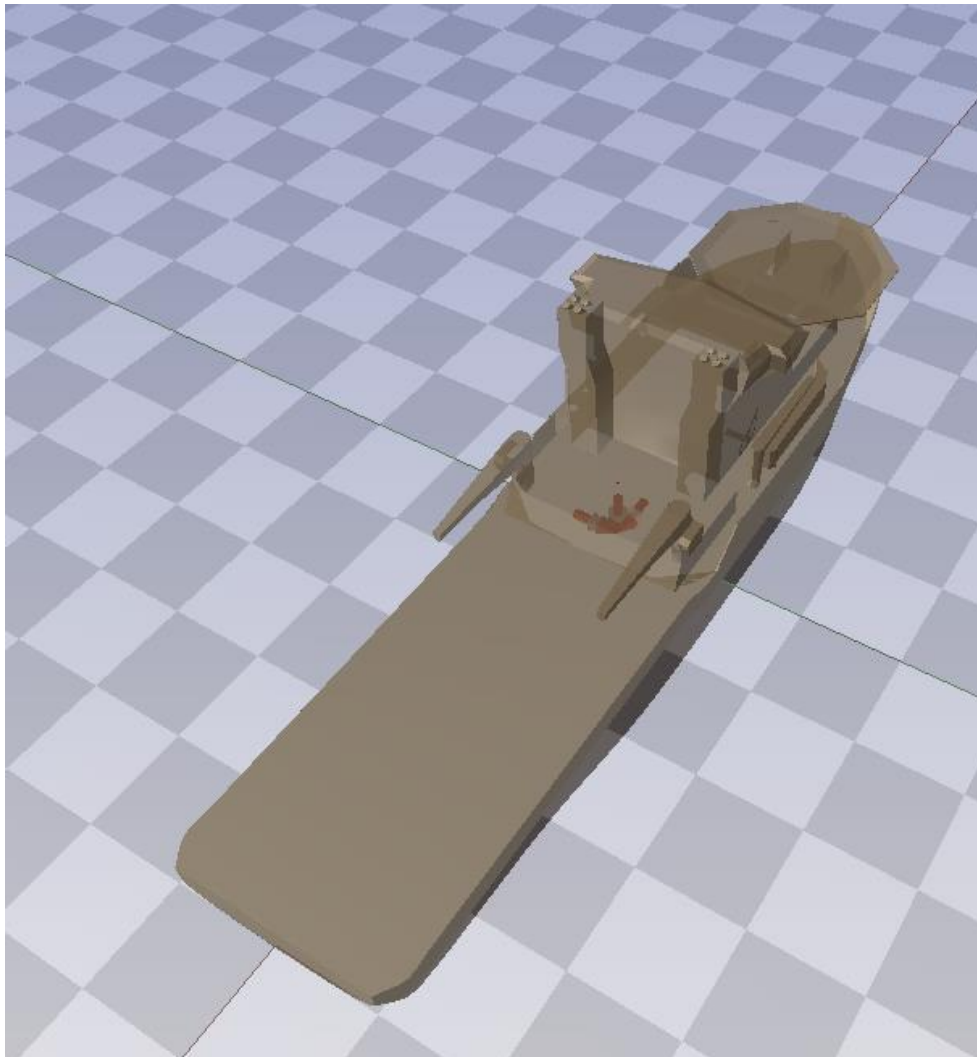


Figure 3.3 Vessel Model in 3D Mechanics

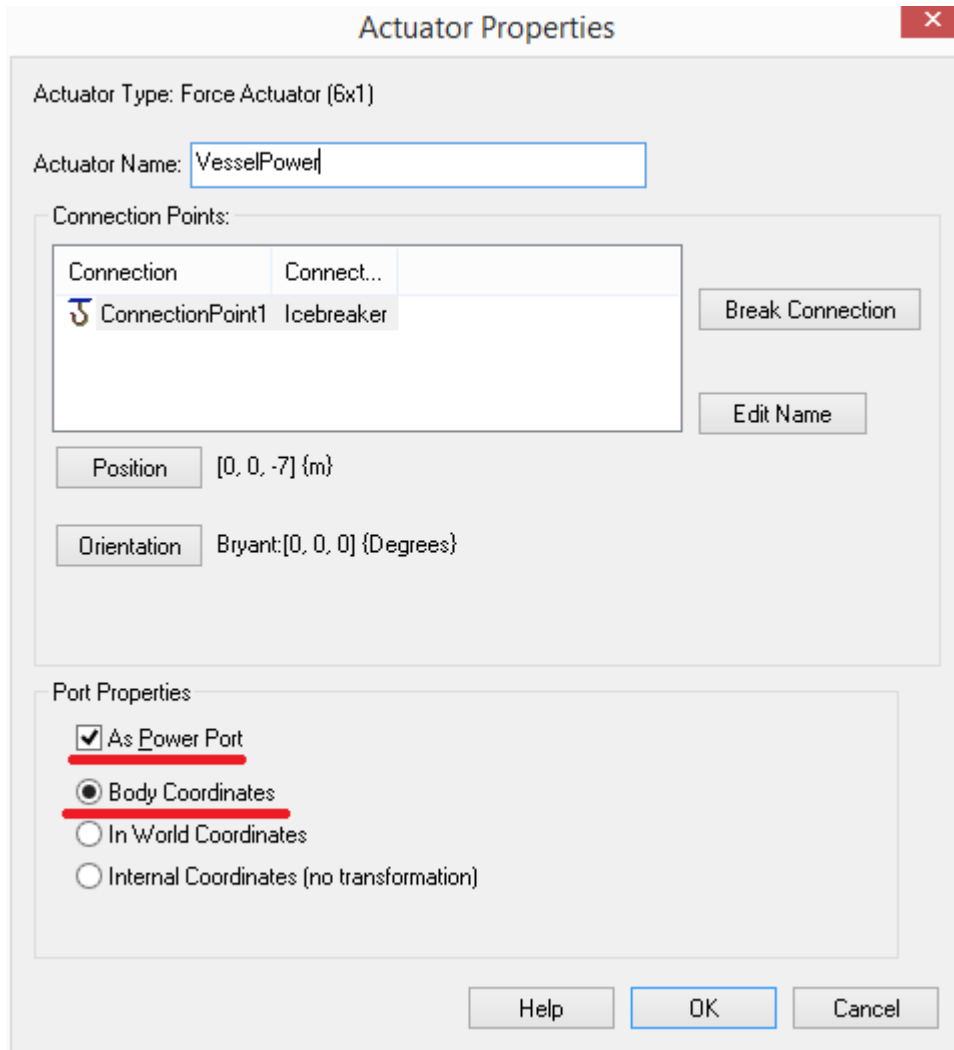


Figure 3.4 Actuator in 3D Mechanics Toolbox

In 3D Mechanics block, an actuator puts torque before translational force, as a result, to connect bond graph model to 3D Mechanics block, a transformer **STF** is added to shift the translational force above torque.

$$3D\ Mechanics \begin{bmatrix} Torque \\ Force \end{bmatrix} \leftarrow STF \leftarrow \begin{bmatrix} Force \\ Torque \end{bmatrix} Bond\ Graph$$

Moreover, to solve rigid body causality problem, the causality of two ports in **I** element shall be both changed to 'indifferent'.

3.4 Parameters

To input realistic data of rigid body inertia, added mass, hydrodynamic damping, restoring force and environmental forces. A MATLAB-20-sim connection is established to transform vessel data from ShipX to 20-sim using Marine control (MSS) toolbox and the new 'transcript' feature in 20-sim.

3.4.1 Calculation in ShipX

In ShipX, if the reference origin is set on the water plane, the equivalent position vector R44, R55, R66, R64 are with respect to the origin on water plane instead of to the centre of gravity.

In order to generate ShipX data for MSS, the vessel hull must be represented by table-of-offsets (Fossen and Perez 2014). Normally 20 offset points on each half section will provide an adequate description of the section shape and assure that correct added mass and damping coefficients are

obtained. In the pre-process of VERES vessel response calculation, the following *calculation options* must be chosen to comply with the MSS data structure.

- Ordinary strip theory (recommended but other methods can be used)
- Added resistance – Gerritsma & Beukelman
- Generate hydrodynamic coefficient files (*.re7 and *.re8)
- Calculation options: choose z-coordinates from CO (O_b).

The condition information for frequency-domain simulations must be chosen according to:

- Vessel velocities must always include the zero velocity: it is optional to add more velocities that are needed for manoeuvring.
- Wave periods: it is recommended to use values in the range 2.0s to 60.0s.
- The wave heading must be chosen every 10 deg starting from 0 deg.

3.4.2 MSS Post-process

After ShipX calculation, put all five output result files: *.hyd, *.re1, *.re2, *.re7, *.re8 into current folder in MATLAB. Install MSS toolbox and run

```
>>vessel=veres2vessel('input')
```

MSS is using x (forwards), y (starboard) and z (downwards) by introducing a transformation matrix [-1, 1, -1, -1, 1, -1] in the function. And the position of origin depends on the *calculation option* defined in ShipX. After abovementioned process, a MATLAB data file *.mat will be created and a data structure of vessel will be given (saved to *.mat).

3.4.3 20-sim Parameter Input

20-sim model uses data from MSS data file *.mat. In newly released 20-sim 4.4 version, a library of 20-sim-MATLAB connection toolbox is published using TCP/IP protocol for data exchange and control. In MATLAB, a short script can be written for users to select the target MSS data file *.mat, 20-sim model file *.emx and desired forward velocity, wave heading, wave period and amplitude. Several functions from 20-sim-MATLAB connection toolbox are being used here.

```
>> xxsimConnect() % MATLAB connects to 20-sim
>>xxsimOpenModel(filename) % Open 20-sim model
>>xxsimProcessModel() % Process the model
>>xxsimSetParameters(parameternames, values) % Input variables from Matlab to 20-sim
>> xxsimRun() % Run the 20-sim model
```

Full script and user manual can be found in Appendix.

```
Command Window
>> simulation
Please select a 20SIM model,press [Enter] to continue

User selected:C:\Users\Jiafeng\Documents\Master in Ship Design\Master Thesis\Appendix\Model\Vessel\S
Please select a SHIPX data, press [Enter] to continue

User selected:C:\Users\Jiafeng\Documents\Master in Ship Design\Master Thesis\Result\run5.mat

The vessel speeds in knots are [0].
Choose target vessel speed in knots:0

The wave periods are [30 25 20 19 18 17 16 15.5 15 14.5 14 13.5 13 12.5 12 11.5 11 10.5 10 9.5 9 8.5]
Choose wave period:10

The wave headings are [0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210]
Choose wave heading:60
```

Figure 3.5 User panel from *.mat file to running 20-sim model

The entire process from ShipX to 20-sim can be described as follows:

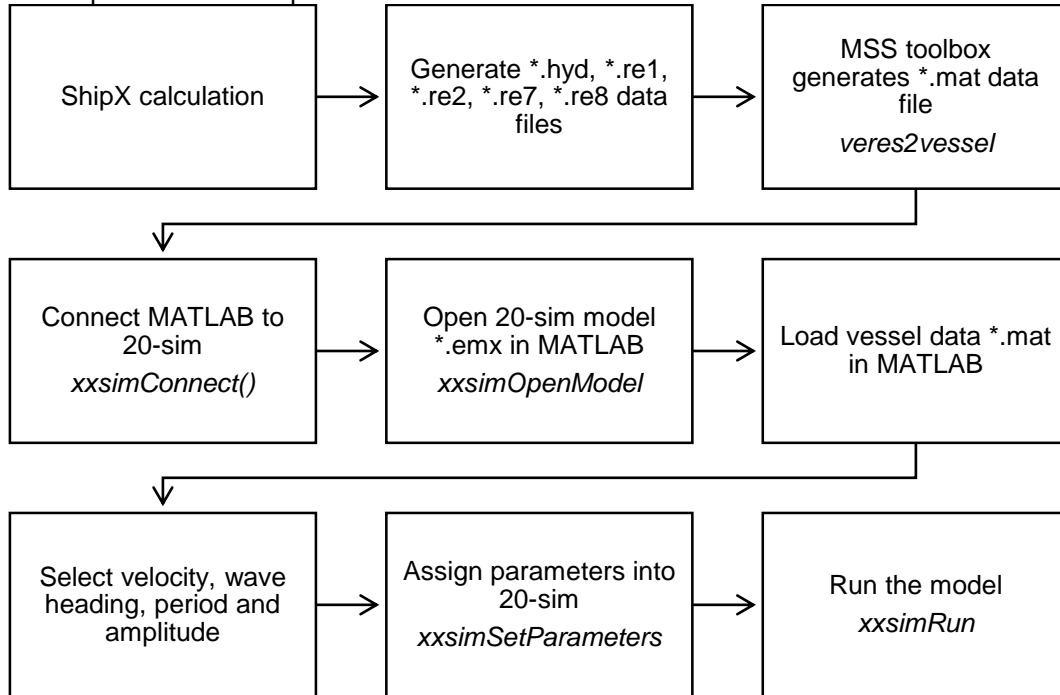


Figure 3.6 Parameters Process from ShipX to 20-sim

3.5 Model Assessment

To assess the accuracy and authenticity of the bond graph model, an evaluation between 20-sim simulation results and ShipX motion RAO is conducted. The default ship 's175' from ShipX database is used. The time domain 6 DOF motion are compared in amplitude.

3.5.1 Results

Table 3.1 Vessel and Wave Data

Vessel name	s175	Lwl/m	175
Lpp/m	175	Vessel velocity/kn	0
T/m	9.5	Wave height/m	1
B/m	25.4	Wave heading/deg	0:10:180
CoG from Lpp/2 and keel line/m	[-2.55, 0, 9.55]	Wave Period/s	4:0.5:16,16:1:20,25,30
Displacement/t	24589470	Sampling section	t = 9900-10000 sec*
C_B	0.568	Integration method	Modified BDF
CoB from Lpp/2 and keel line/m	[-2.55, 0, 5.21]	Integration error	Abs 1e-5; Rel 1e-5

*The motion RAO in ShipX only gives one amplitude for one specific wave period and heading, meaning there is no transient dissipating vessel motion in its natural period but only steady wave induced motion in wave period. However, due to the equation 20-sim model based on, it allows vessel have transient dissipating motion in its natural period. Thus the assessment will be conducted after the vessel in 20-sim reaching the steady state.

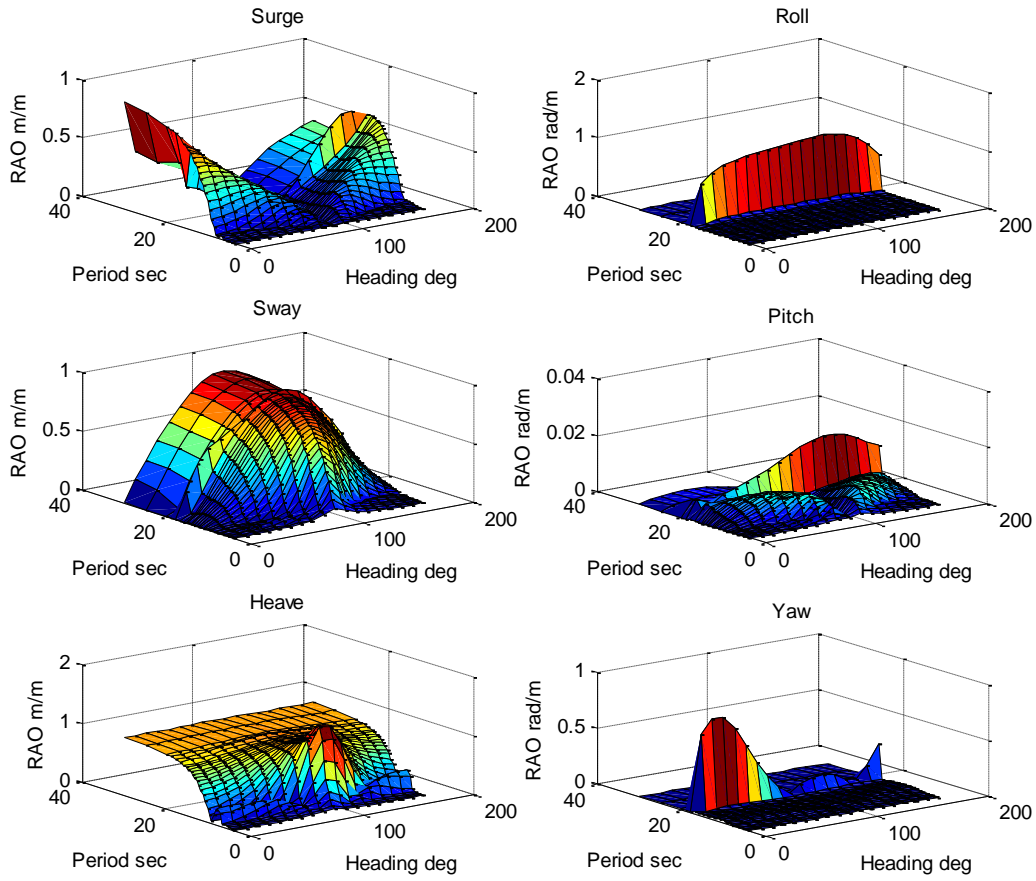


Figure 3.7 Motion RAO from 20-sim

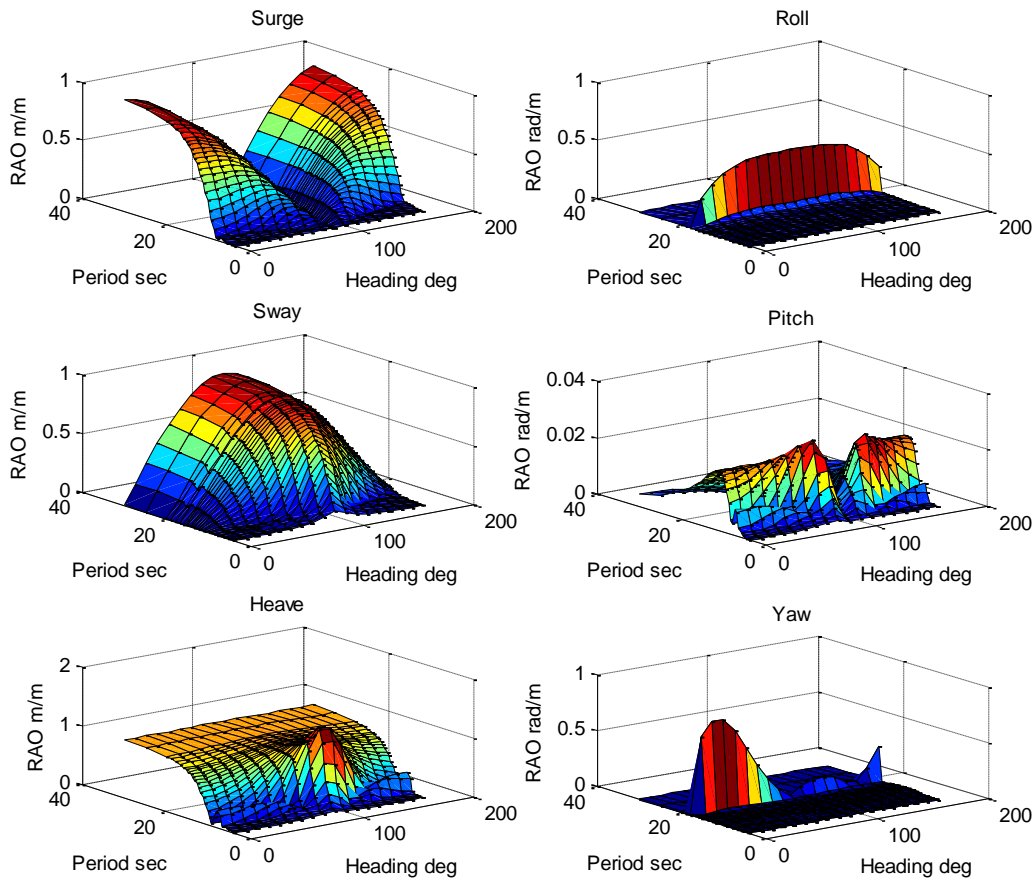


Figure 3.8 Motion RAO from ShipX

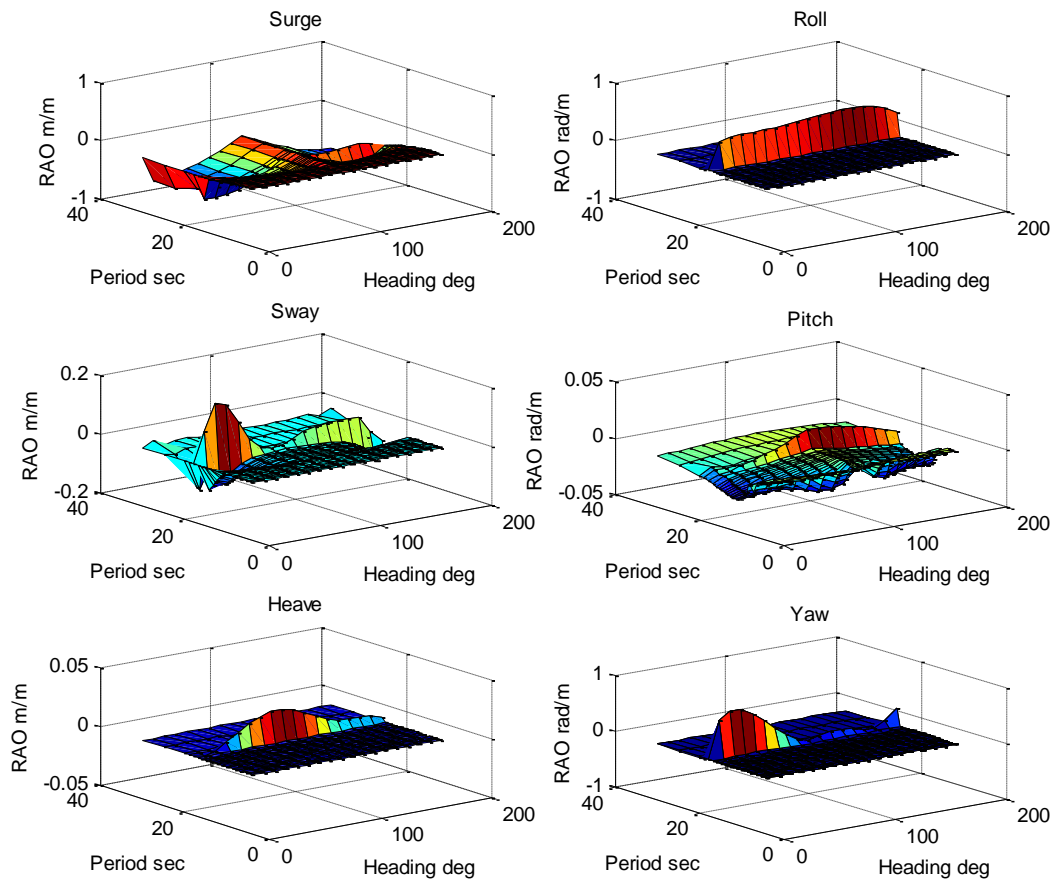


Figure 3.9 Error between 20-sim and ShipX

3.5.2 Discussion

The results show good matching between 20-sim and ShipX in most wave periods and headings yet poor accuracy in vessel's natural rolling period (18 sec). There is also relatively high error of surge and pitch in extremely long and short period wave. The reason is that all hydrodynamic coefficients in 20-sim are from potential theory, but ShipX has additional viscous damping coefficient in its calculation. The error exaggerates itself when high velocity occurs in natural rolling frequency. Also, the motion RAO values at where high surge and pitch errors exist (e.g. surge in beam sea, pitch in 4 sec period wave) are extremely low, hence higher potential for large relative error mathematically speaking.

One way to improve the model quality is to add linear viscous damping coefficient for natural rolling frequency, and to tune the motion RAO in 20-sim closer to that in ShipX. The reason not using non-linear viscous damping is that non-linear viscous damping will cause the vessel yaw drastically because of the coupling effect between degrees of freedom. A linear approximation is more favourable for having a stable course. The user can tune the damping coefficient for each wave period and heading to have a better approximation towards ShipX motion RAO.

4 CRANE MODEL

The following chapter is based on Yingguang Chu's crane model in 20-sim, including kinematics system, hydraulics system and control system (Chu 2013).

4.1 Dynamic System

The 3D Mechanics toolbox in 20-sim provides 3D animation environment for rigid body dynamics modelling, where the crane can be modelled as well as the vessel. In Chu's work, the 3D modelling of crane was firstly done in Solidworks and then transferred into 3D Mechanics by using a converter called "COLLADAto20-sim". "COLLADAto20-sim" is developed by the Controllab group using C++ and COLLADA, an inter application exchange file format (Janssen 2013). There are special requirement for this kind of conversion however.

4.1.1 Solidworks Modelling

Solidworks is a well-developed 3D modelling tool which supports many general formats among which *.STL files can be imported into 20-sim 3D Mechanics toolbox and animation window. In current model, a Rolls-Royce PSV-100 Crane model was simplified and modelled in Solidworks.

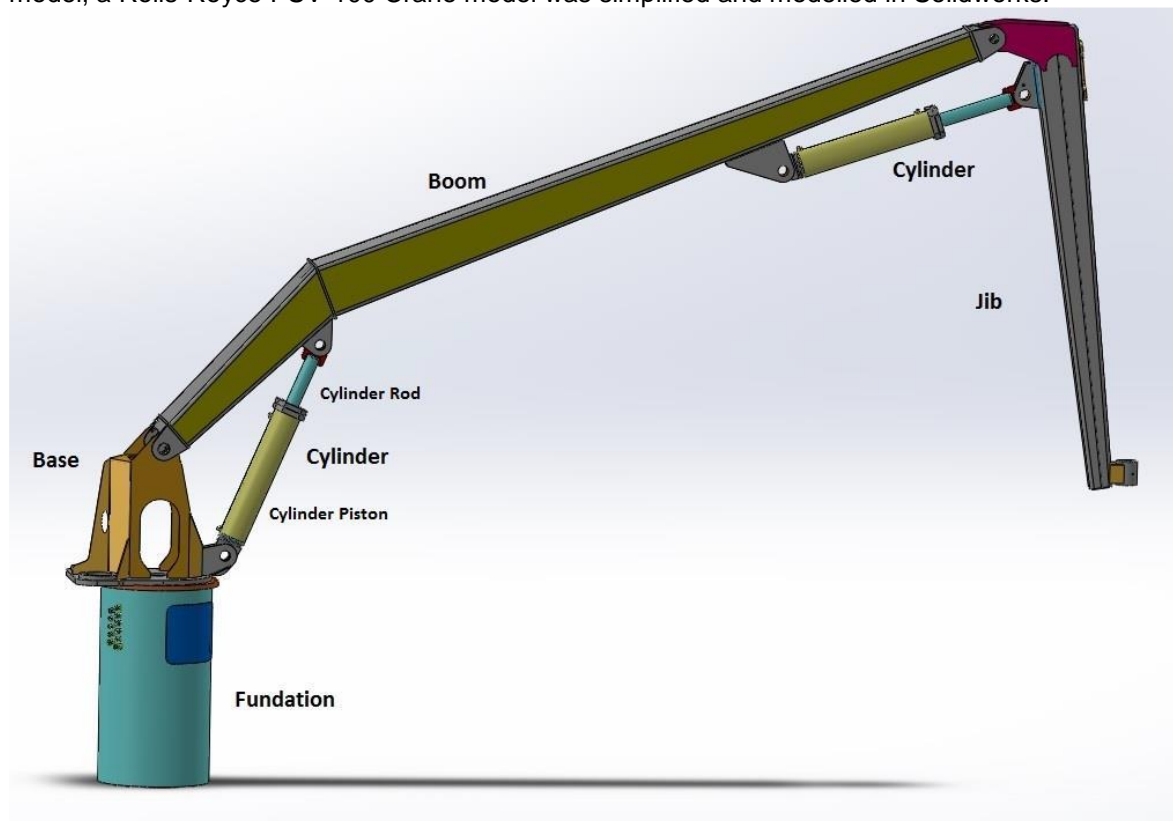


Figure 4.1 Solidworks model of simplified Rolls-Royce PSV-100

This knuckle boom crane, Rolls-Royce PSV-100, has eight assemblies (parts) named as in the picture. Each assembly contains several sub-level assemblies (parts), e.g. one cylinder set has two parts: a piston and a rod. Solidworks mates combine those assemblies and parts to form a complete structure. Mates can only be created in where physical mechanical connection exists, otherwise unexpected bodies and joints will be generated because each mate represents one joint in 3D Mechanics model. There are three types of mates provided in Solidworks: standard mates, advanced mates and mechanical mates. Only standard mates are allowed if the model needs to be converted into 20-sim through COLLADA.

After modelling, Solidworks can export its assembly into COLLADA format *.DAE through a plug-in which preserves physical and visual information of the model.

4.1.2 Converting Steps and Tips

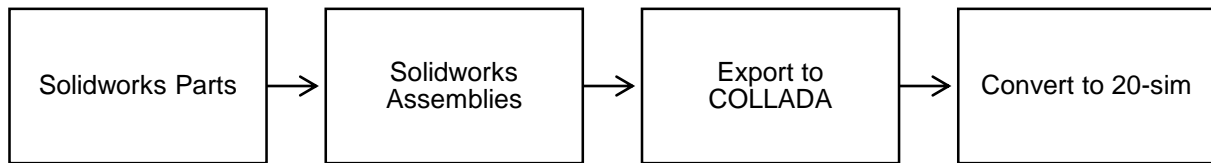


Figure 4.2 Steps from Solidworks to 20-sim

Tips:

- The model shall be simplified as much as possible. Unnecessary parts that have no effect on dynamic behaviour of the model should not be included.
- Mates of parts shall be designed correctly. Every mate will result in a joint in 3D Mechanics. Only standard mates are allowed.
- Mass and inertia need to be assigned to each body manually after the conversion.
- The setup of the coordinate system will need to be adjusted if one would like to use, e.g. the Denavit-Hartenberg method (D-H method) in the control scheme.
- After converting the crane model into 20-sim 3D Mechanics, connect the foundation of the crane with the vessel hull on the desired location by using a welded-joint.
- The Solidworks COLLADA exporter can be downloaded at: <http://labs.solidworks.com/Products/Product.aspx?name=colladaexport>
Prerequisites for installation and instructions can be found through the same link as well.
- NVIDIA PhysX System Software may be required if installation failed in some cases. PhysX 9.12.1031 can be downloaded at: <http://www.nvidia.com/object/physx-9.12.1031-driver.html>

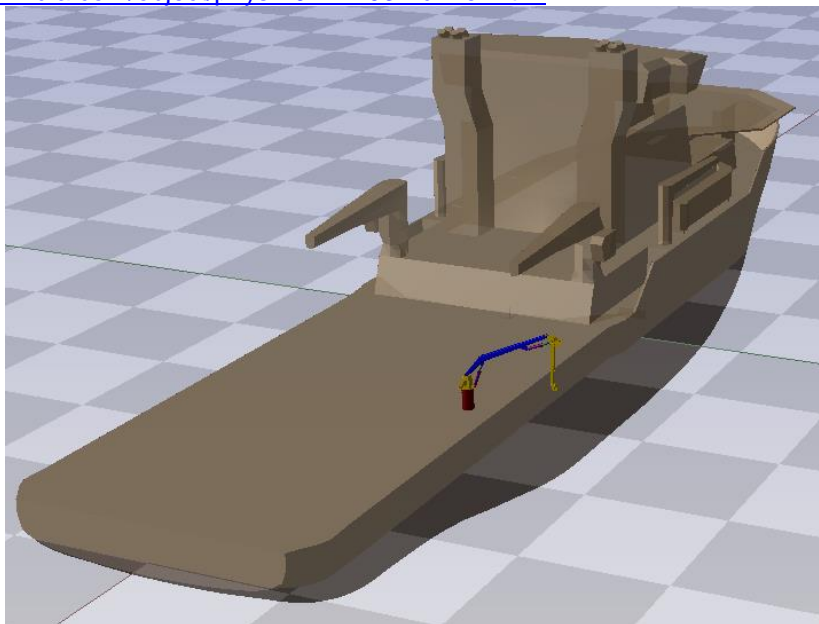


Figure 4.3 Crane Model in 3D Mechanics

4.2 Hydraulic System

20-sim provides hydraulic components library for modelling and simulation hydraulic systems. In Chu's work, the model is designed as close as possible to the Modelica hydraulic library (Controllab Product BV 2013). However the library does not provide all components that are required. So some components are self-designed and added into the library later. An example of main derrick hydraulic is briefly introduced below.

4.2.1 Main Derrick Sketch

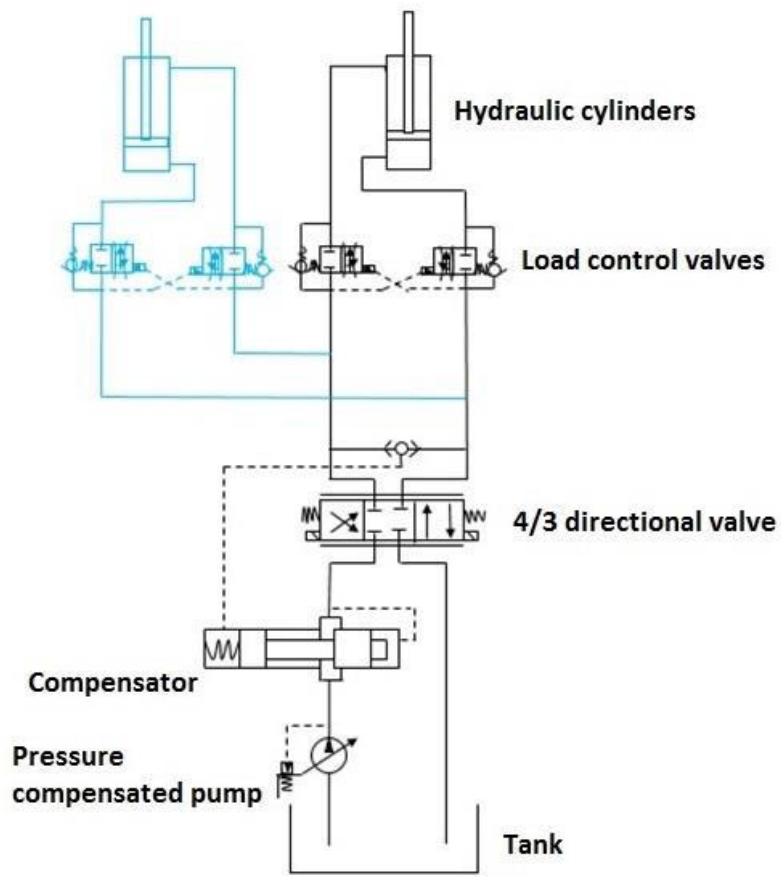


Figure 4.4 Simplified Main Derrick Hydraulic Sketch

4.2.2 20-sim Modelling

According to the simplified sketch of main derrick hydraulic, a 20-sim hydraulic model can be built using the hydraulic components library except the cylinder. The parameters can be adjusted according to the crane technical specification. The 20-sim model is equipped with input signal port for control system and output power port for dynamic model in 3D Mechanics. In 3D Mechanics, the joint where hydraulic system is deployed acts as Power Interaction Port (1x1) and outputs position and velocity of itself.

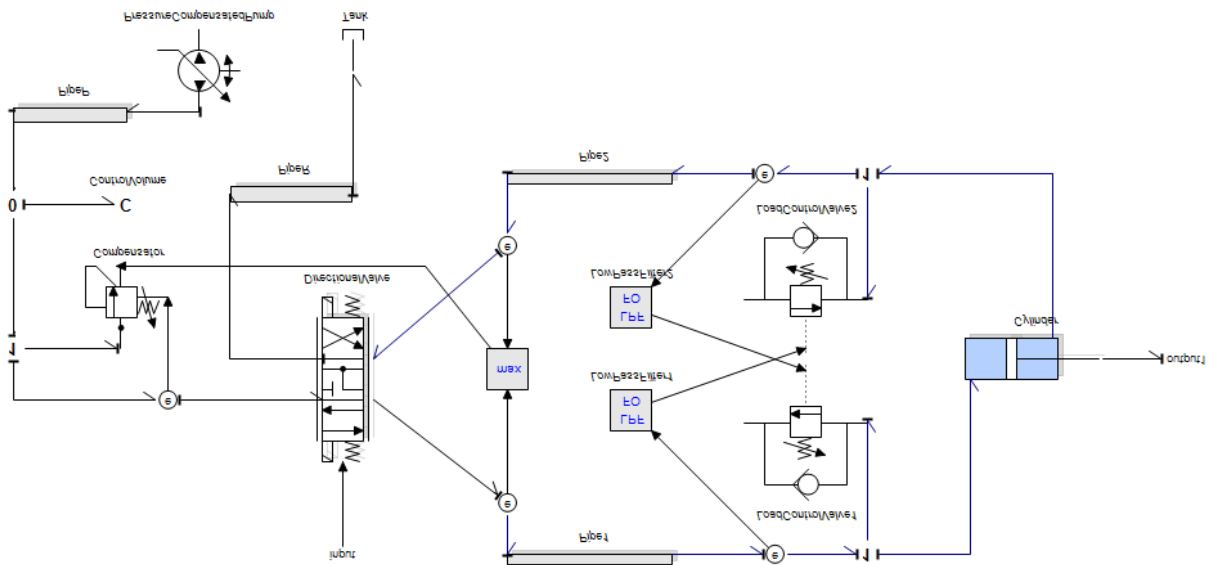


Figure 4.5 Boom Cylinder Hydraulics Model

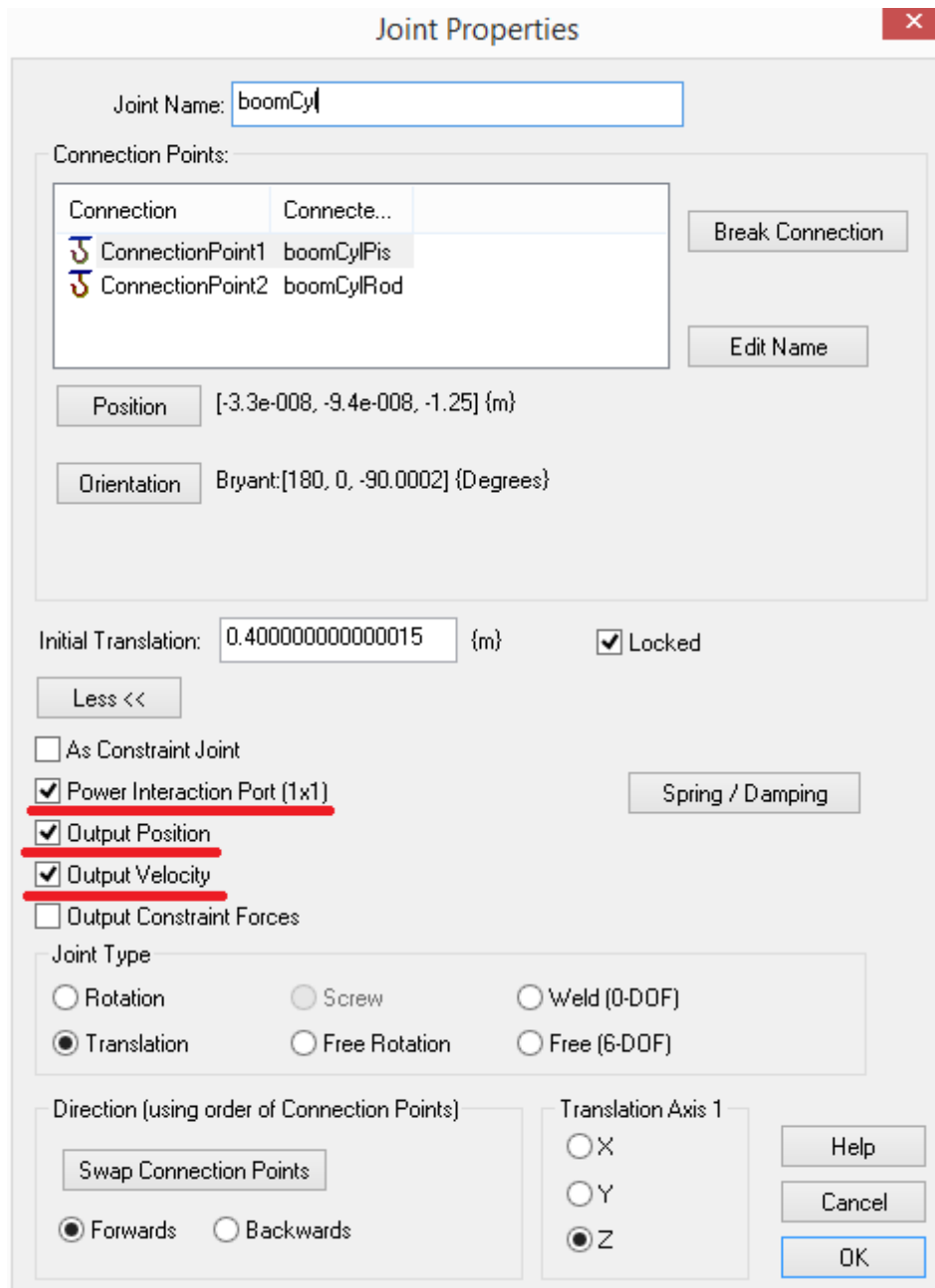


Figure 4.6 Boom Cylinder Setup in 3D Mechanics.

4.3 Control System

The control system of an offshore crane consists of two kinds – manual control and auto control. The crane can be controlled by solving its kinematic model. The resulting signal for each hydraulic system can be input into hydraulic model hence the crane model.

4.3.1 Manual Control

The manual control part can be realised by joystick or other input device, e.g. haptic control (Sanfilippo, et al. 2013). The joystick sends signals of the crane tip velocities or cylinder velocities into kinematic model where corresponding joint velocities and cylinder velocities are calculated by jacobians and fed to the hydraulic system using PID controller. Joint angles and vessel motions can be obtained by sensors in 3D Mechanics model.

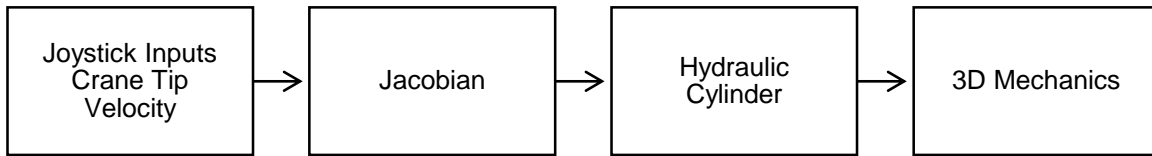


Figure 4.7 Crane Tip Velocity Control

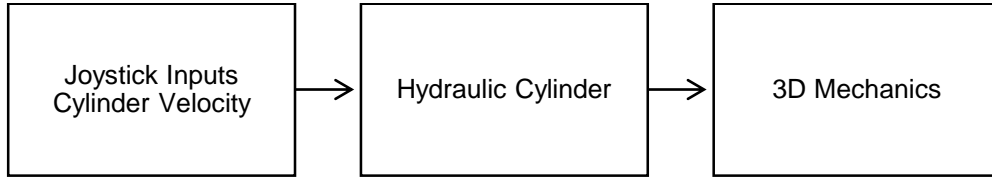


Figure 4.8 Direct Cylinder Velocity Control

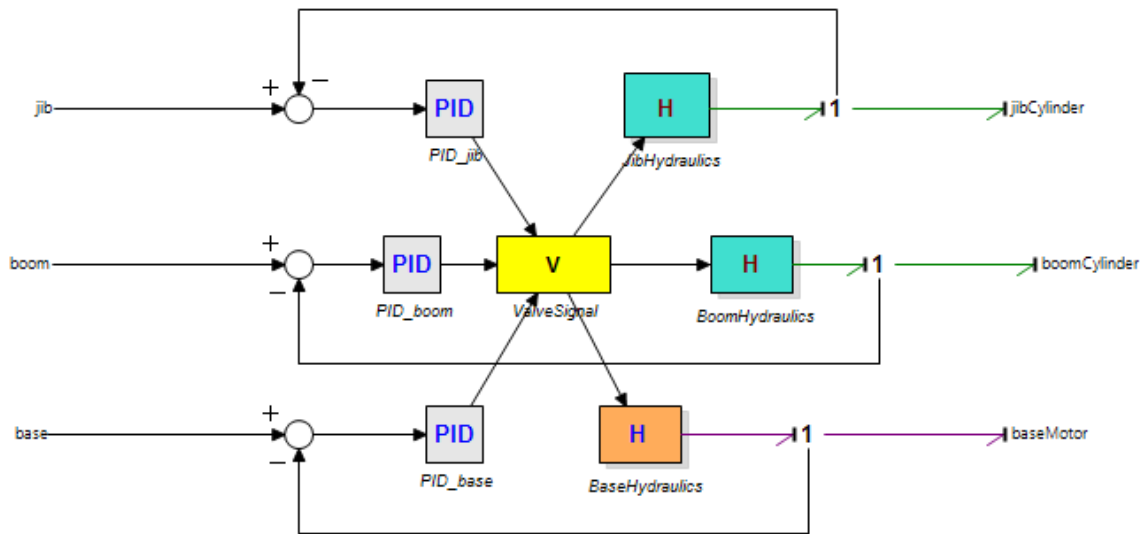


Figure 4.9 Cylinder Velocity Input and PID Control

4.3.2 Auto Control

The auto control system may include heave compensation system, anti-sway system or the combination for all three translational degrees of freedom. The heave motion of the vessel can be compensated by crane kinematics or winch. The sway motion can only be compensated by crane kinematics. There are many algorithms to realise the function of compensation. One simple solution is to add compensating motion/velocity on opposite direction into crane kinematics or winch.

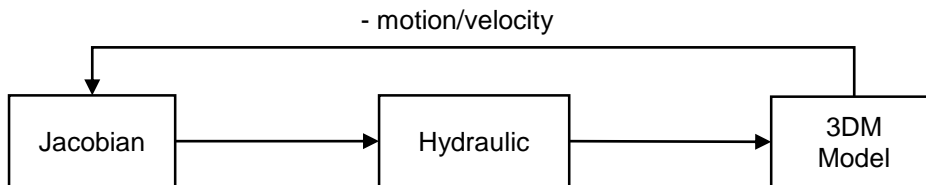


Figure 4.10 Compensating Motion/velocity on Opposite Direction

One thing should be noticed is that all manual/auto control algorithms are signal flow model or simple codes in Jacobian (Control Box). The physical entities, hydraulics and dynamics, are modelled as power flow model (hydraulic library, 3D Mechanics, bond graph) which remains intact from the variations of control system.

5 CABLE MODEL

5.1 Cable Characteristics

Offshore cable is usually treated as flexible body with infinite degrees of freedom. It has physical features of elasticity and plasticity in both axial direction and radial direction. In practice, the external forces offshore cable may experience includes concentrated force on both ends, gravity force along the entire length, buoyancy force on the submerged part and damping forces caused by relative motion between cable and the fluid around. Those external forces can cause axial tension, axial torque, radial shear and in extreme cases bending moment in local area. Because of the compound material inside the cable, some may have non-linear elastic modulus and unevenly distributed stress level in local area.

Offshore crane often use multi-strand steel wires as lifting cable in order to enhance its strength and minimize axial rotation for operational purpose. When loaded, steel wire will generate torque if both ends are fixed and turn if one end is unrestrained. The torque or turn generated will increase as the load applied increases. The degree to which a cable generates torque or turn will be influenced by the construction of the rope. All cables will rotate to some degree when loaded.

5.2 Categories of Cable Model

Choo and Masarella classified current analytical and numerical methods of solving the equations of cable motion in four categories (Johansson 1976).

- Method of characteristics
- Finite element methods
- Linearization methods
- Other Methods

Different categories suit different modelling scenarios and in this thesis, finite element methods are used in 3D Mechanics.

5.2.1 Method of Characteristics

The method of characteristics is a direct mathematical interpretation of the physical behaviour of the cable. E. J. Routh proposed an early formulation of a dynamic cable model in 1905 (Johansson 1976). A small segment ds of the cable is considered with its parametric trajectory $x(s), y(s), z(s)$ in terms of Cartesian coordinates where s is the trajectory curve length. $d\delta$ is the unstretched length of ds so that $ds = d\delta(1 + \epsilon)$ where ϵ is the local strain. R is the tension at the lower end of ds , and the resolve components along x, y, z axes are

$$R \cdot \frac{dx}{ds}, R \cdot \frac{dy}{ds}, R \cdot \frac{dz}{ds}$$

The resolved tension components at the upper end are

$$R \cdot \frac{dx}{ds} + \frac{d}{ds} \left(R \cdot \frac{dx}{ds} \right) ds, \text{ etc.}$$

The net tension force components thusly are

$$\frac{d}{ds} \left(R \cdot \frac{dx}{ds} \right) ds = \frac{d}{d\delta} \left(R \cdot \frac{dx}{ds} \right) d\delta, \text{ etc.}$$

Let m be the mass per unit length of unstretched cable, and X, Y, Z external force components per unit of unstretched length. Also, let u, v, w be the components of the segment velocity. The equations of motion can then be written as

$$m \cdot \frac{\partial u}{\partial t} = \frac{\partial}{\partial \delta} \left(R \cdot \frac{dx}{ds} \right) + X$$

with corresponding equations for the y and z directions. t represents time. For the submerged cable, the hydrodynamic forces are included in X, Y, Z .

For two-dimensional problems, the equations may be expressed in terms of local tangential and normal components in the form

$$m \cdot \left(\frac{\partial u}{\partial t} - v \cdot \frac{\partial \phi}{\partial t} \right) = P + \frac{\partial T}{\partial s}$$

$$m \cdot \left(\frac{\partial v}{\partial t} + u \cdot \frac{\partial \phi}{\partial t} \right) = Q + \frac{T}{\rho} \cdot \frac{\partial s}{\partial \delta}$$

where P, Q are tangential and normal components of the external force, u, v represents the tangential and normal velocities and ϕ is the angle between the tangent and the horizontal plane or some other fixed direction in the cable plane.

Through the integration of Routh's equation along characteristic lines in x, y, z, t space, for some simple examples like straight strings, exact analytical solutions can be given. However, for problems like lifting cable or mooring line, when cable experiences unsteady irregular external forces, a stepwise computerized solution method is required but both over-detailed and time consuming. Most of the time approximation is involved so that the accuracy is compromised.

5.2.2 Finite Element Methods

Finite Element Methods enable the cable to be conceptually modified (Johansson 1976) before mathematical formulation. Comparison of different FEM cable models are given below.

- **Mass-String:** The cable is modelled as an inelastic string with mass lumped at several nodes.
- **Mass-Spring:** The cable is modelled as springs with mass lumped at several nodes.
- **Bars:** The cable is modelled as a series of straight bars with distributed mass, which are jointed at both ends.
- **Beams:** The cable is modelled as a series of beams with distributed mass, which are jointed at both ends and with additional curvature.

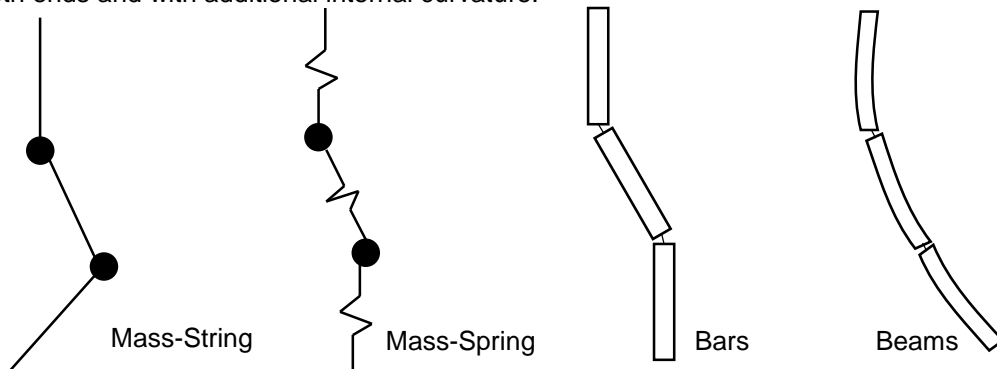


Figure 5.1 Four Types of FEM Cable Model

Table 5.1 Comparison of FEM Cable Models

Models	Mass-String	Mass-Spring	Bars	Beams
Elasticity		Y	Y	Y
Mass	Y	Y	Y	Y
Moment of Inertia			Y	Y
Plasticity				
Slope Continuity				Y
Structural Damping			Y	Y
Tension Induced Torque				

5.2.3 Linearization Method

In some stability and frequency response studies of cable vehicle systems, the motion equations are linearized and assumed to only have small deviations from the state of equilibrium. The linear partial differential equations will then be reduced to linear partial differential equations with curve length as the space variable. Moreover, sometimes the perturbations are even assumed to be harmonic. Then the time derivatives can be further eliminated from the equations, which become ordinary differential equations solely about the cable length. In this thesis, the maximum length of lifting cable can reach thousands of meters, which makes deformation along the cable significant and irregular. The linearization method does not suit for this purpose.

5.3 Cable Model in 3D Mechanics

In this model, a combination of different types of FEM cable model is used in 3D Mechanics. The cable is modelled as rigid bars with free rotational joint and 1-DOF translational joint on two ends respectively. The rigid bars are represented by a inertia matrix which has both mass and moment of inertia. The free rotational joint has no spring or damping but constrained all translational motions between bars. The 1-DOF translational joint has spring and damping property which represent the elasticity and structural damping of the cable respectively. The joint connecting the crane tip is free rotational and the joint connecting to the load can be either free rotational or translational depending on the structure of the connecting point. The elasticity, mass, moment of inertia and structural damping of the cable are all realized in this modelling method.

If the number of discretized parts in FEM is n , the number of rigid bars, free rotational joint and 1-DOF translational joint shall be $2n, n, n$ respectively, not including the joint on crane tip and load.

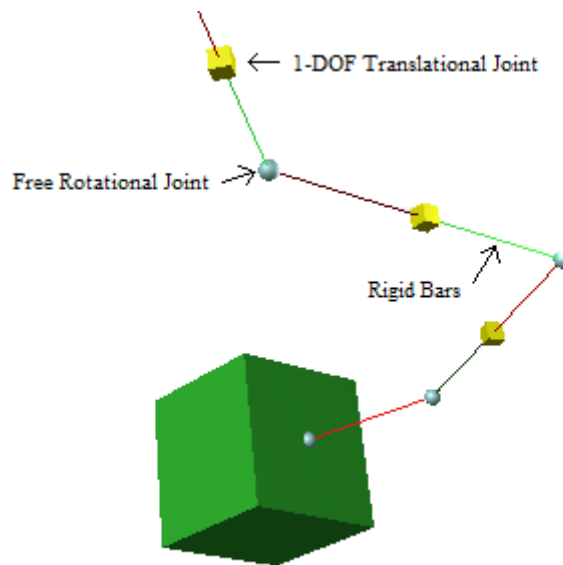


Figure 5.2 FEM Cable Model in 3D Mechanics

Using the same approach as in vessel motion input, the external forces such as current force and buoyancy force can be applied as bond graph model on each rigid bar as a constant or the function of motion and velocity. The port power should be defined in world coordinates, because of the coordinate where current force is defined.

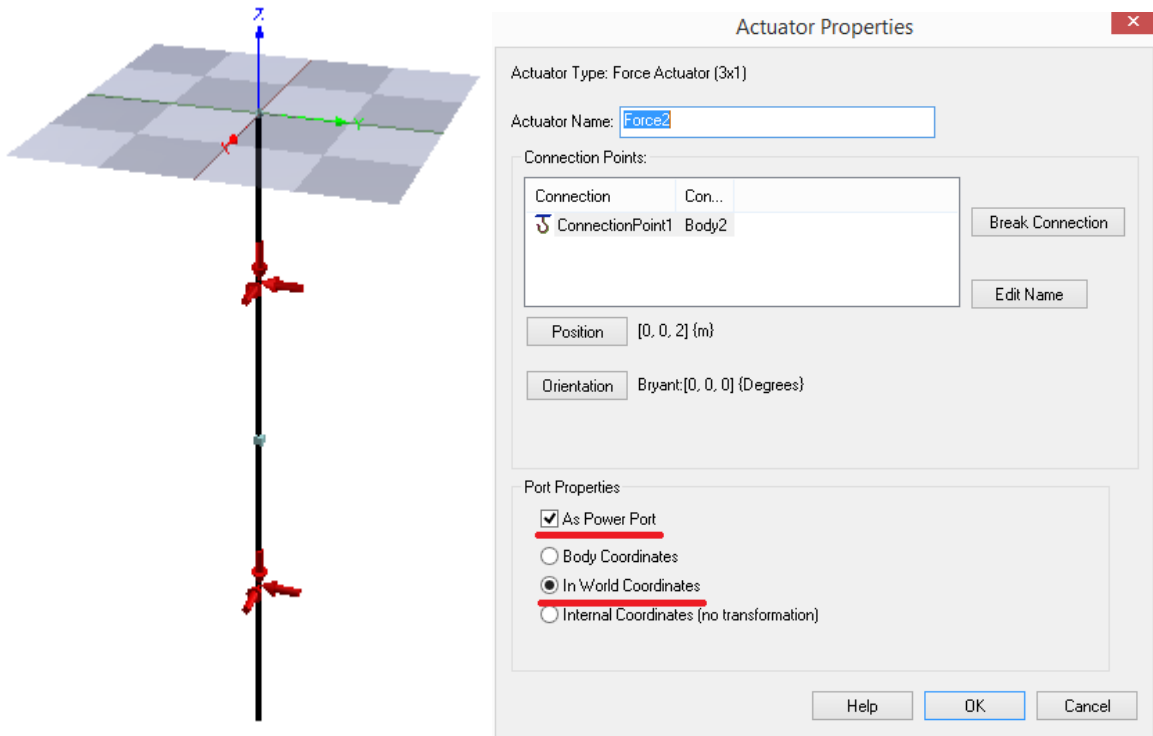


Figure 5.3 External Force Acting as Actuator

Outside the 3D Mechanics, each actuator is linked with a bond graph effort source where current force and buoyancy force can be defined by simple Morison equation and Archimedes law or complex expressions. The current speed is defined in a signal block by an array with length of n or $2n$ depending on if two bars connected by translational joint shall have the same current speed.

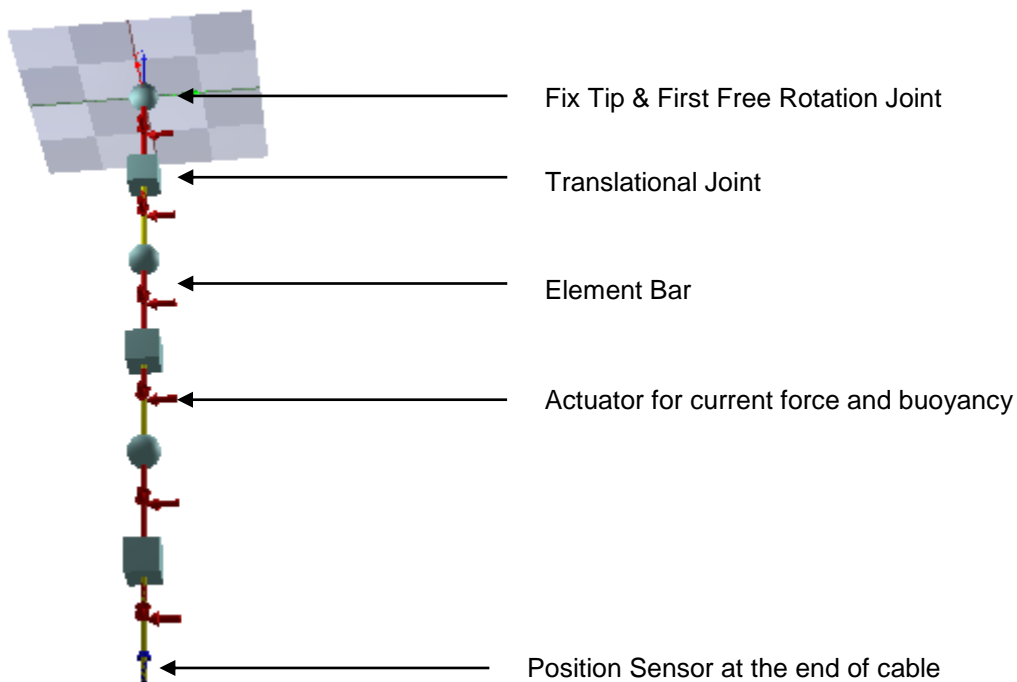


Figure 5.4 Cable Model ($n=3$, $L=10$)

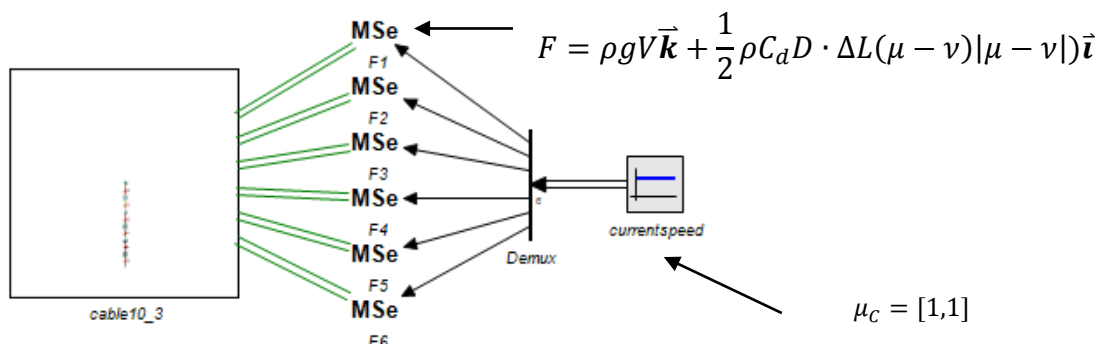


Figure 5.5 External Force Input

6 LOAD MODEL

6.1 Lifting Operation

Offshore crane is designed to be multifunctional. The loading objects include subsea construction materials, ROV, platform supplies, personnel carrier and all other different objects. Each loading object has its unique physical property and installation requirement. The lifting operation can normally be divided in five phases.

- **Lift Off:** The object is fixed to the cable and snatched from the deck. The major external forces are friction from the deck, supporting force and lifting force.
- **In Air:** The object has been lifted and being moved to the designated position before being lowered down into the water. The major external forces are air resistance, inertia force and lifting force.
- **Splashing:** The object has been lowered close to the free water surface and experiencing splashing impact until it is totally submerged and unaffected by the water surface. The major external forces are splashing force, buoyancy force and lifting force.
- **Deeply Submerged:** The object is deeply submerged in the water yet still has a certain distance from the seabed. The major external forces are buoyancy force, hydrodynamic force and lifting force.
- **Landing:** The object has been lowered close to the seabed and ready to land. The major external forces are buoyancy, hydrodynamic force, lifting force and possibly landing impact force.

The object has its inertia matrix constantly through all phases, but different damping and restoring functions respectively, e.g. the air resistance is mostly negligible while the object is in air but the hydrodynamic resistance is significant in the water; the buoyancy force varies throughout the splashing phase yet stays constant after the object is full submerged. In this thesis, a general method of expressing the physical properties of the object in deeply submerged phase is given which can be without loss of generality applied in other phases.

6.2 Load CFD

6.2.1 Motion Function of the Load

When an object is deeply submerged, it experiences a concentrated/scattered lifting force from the cable, an evenly distributed gravity force downwards and buoyancy force upwards under the premise of a constant submerged volume of which. It also experiences hydrodynamic forces including the inertia force and the damping force by the definition in the Morison equation.

Similar to the ship motion equation, for a harmonically oscillating object, the motion function of an submerged object in its own body fixed coordinate can be written as

$$\mathbf{M}_{RB} \cdot \ddot{\boldsymbol{\eta}}(t) + \mathbf{M}_A(\omega, \zeta) \cdot \ddot{\boldsymbol{\eta}}_r(t) + \mathbf{C}_{RB} \cdot \dot{\boldsymbol{\eta}}(t) + \mathbf{C}_A(\omega, \zeta) \cdot \dot{\boldsymbol{\eta}}_r(t) + \mathbf{B}(\omega, \zeta) \cdot \dot{\boldsymbol{\eta}}_r(t) |\dot{\boldsymbol{\eta}}_r(t)| = \boldsymbol{\tau}_l + \boldsymbol{\tau}_g + \boldsymbol{\tau}_b$$

where

\mathbf{M}_{RB}	= rigid body inertia matrix of the object
ω	= oscillation frequency of the object
ζ	= oscillation amplitude of the object
$\mathbf{M}_A(\omega, \zeta)$	= added mass matrix for a specific ω and ζ .
\mathbf{C}_{RB}	= coriolis force matrix of the object
$\mathbf{C}_A(\omega, \zeta)$	= coriolis force matrix of the added mass for a specific ω and ζ
$\boldsymbol{\eta}(t)$	= object motion
$\boldsymbol{\eta}_r(t)$	= object relative motion to the fluid
$\mathbf{B}(\omega, \zeta)$	= damping force matrix for a specific ω and ζ
$\boldsymbol{\tau}_l$	= lifting force from the cable
$\boldsymbol{\tau}_g$	= gravity force of the object
$\boldsymbol{\tau}_b$	= buoyancy force of the object

The added mass matrix and the damping force matrix can be transformed into added mass coefficient matrix $\mathbf{M}_A(\omega, \zeta)$ and damping force coefficient $\mathbf{B}(\omega, \zeta)$ for a specific object with its unique shape. The damping force here is by definition of Morison equation proportional to the square of velocity. The

calculation for a set of different frequencies and amplitudes can be done using CFD method (Halse, et al. 2014)

6.2.2 Velocity & Acceleration Dependent Coefficient

The motion of the vessel model is assumed to be only periodic with constant added mass and damping matrices for each frequency and amplitude. But the load could be lowered, lifted, dragged and rotated without periodic feature. As the simulation is on a time-domain basis, the hydrodynamic coefficients of the load should be transformed from frequency-amplitude dependent into velocity-acceleration dependent $M_A(\dot{\eta}, \ddot{\eta})$ and $B(\dot{\eta}, \ddot{\eta})$. Although the function is not strictly correct because the fluid has memory effect, i.e. the historical motion of the fluid will affect the presence. But if a good correlation can be shown by CFD calculation, the trajectory of the load can then be removed from the variables.

To investigate the feasibility of velocity-acceleration dependent coefficients, a CFD experiment can be conducted to compare the difference of coefficients under same velocity and acceleration condition but with and without periodic movement.

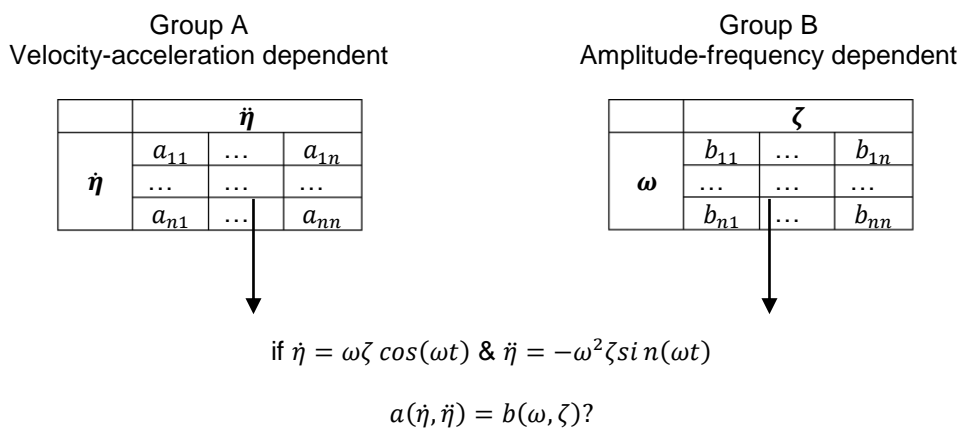


Figure 6.1 CFD Experiment to Test & Verify Velocity-acceleration Dependent Coefficient

If valid, the velocity-acceleration dependent coefficients can then be fitted and input into 20-sim model as functions of velocity and acceleration.

6.3 20-sim Modelling

In 20-sim 3D Mechanics, the loading object can be simply modelled as an object attached to the end of the rope with an 6x1 actuator in body fixed coordinate representing extra added mass inertia force, damping force and another 1x1 actuator in world coordinate representing buoyancy force.

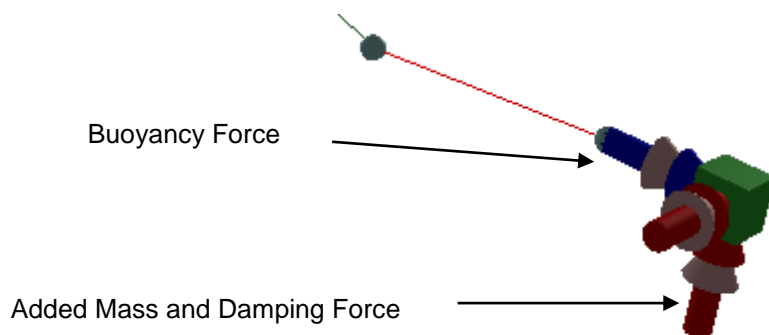


Figure 6.2 Load Model in 3D Mechanics

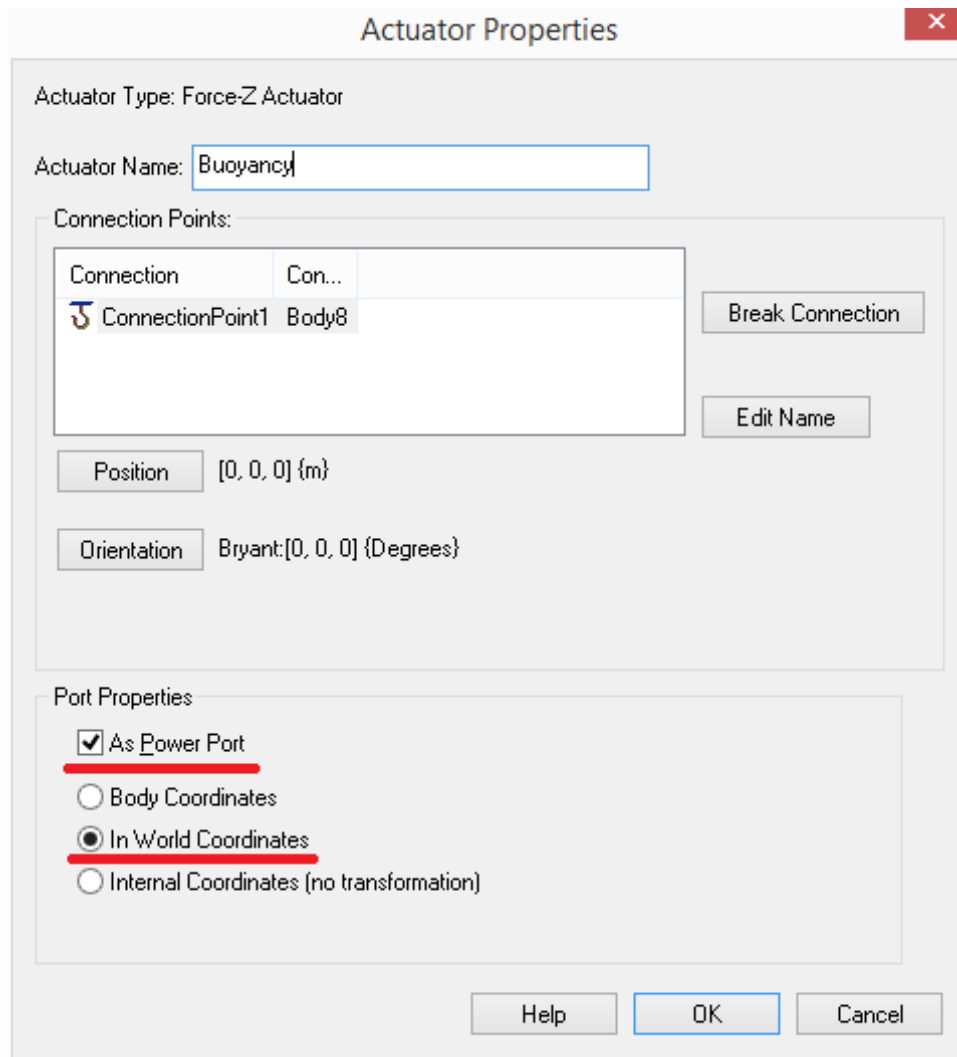


Figure 6.3 Buoyancy Force Actuator

The equation for the 1x1 buoyancy actuator is

$$p.e = \rho g V$$

Where ρ is the water density, g is the gravitational acceleration and V is the submerged volume of the load.

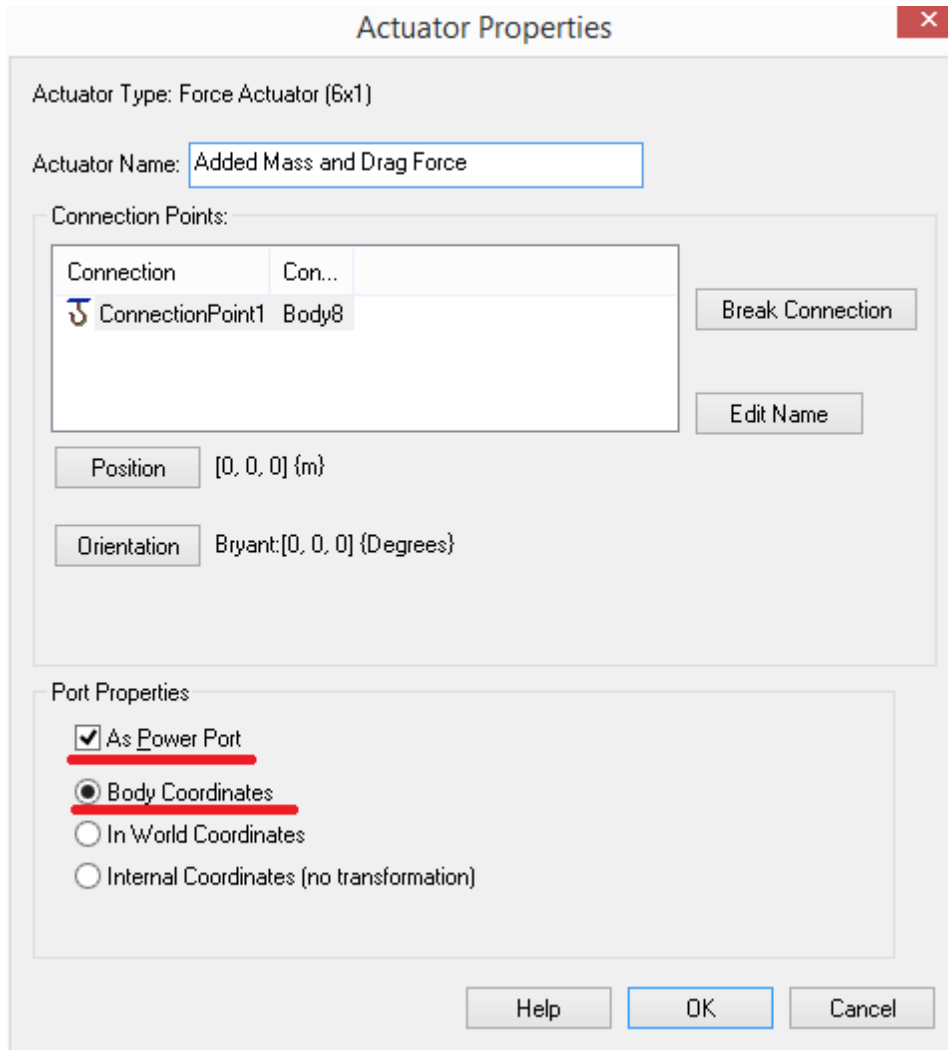


Figure 6.4 Added Mass and Damping Force Actuator

The setting inside 3D Mechanics only allows user to define diagonal inertia matrix for the object body, while added mass is not always diagonal, so the added mass inertia force and coriolis force have to be defined outside the 3D Mechanics. The equation for the 6x1 added mass and damping force actuator is

$$\mathbf{p} \cdot \mathbf{e} = -\mathbf{M}_a(\dot{\boldsymbol{\eta}}, \ddot{\boldsymbol{\eta}}) \cdot (\mathbf{p} \cdot \dot{\mathbf{f}} - \dot{\mathbf{U}}) - \mathbf{C}_A(\dot{\boldsymbol{\eta}}, \ddot{\boldsymbol{\eta}}) \cdot (\mathbf{p} \cdot \mathbf{f} - \mathbf{U}) - \mathbf{B}(\dot{\boldsymbol{\eta}}, \ddot{\boldsymbol{\eta}}) \cdot (\mathbf{p} \cdot \mathbf{f} - \mathbf{U}) |\mathbf{p} \cdot \mathbf{f} - \mathbf{U}|$$

with \mathbf{U} as the current velocity vector acting on the object.

The reason transformation jacobian \mathbf{J}_b^h is not used here is because added mass and damping coefficient for an submerged object is constant with respect to the body fixed coordinate, while floating object has a constant added mass and damping coefficient with respect to the steadily translating coordinate.

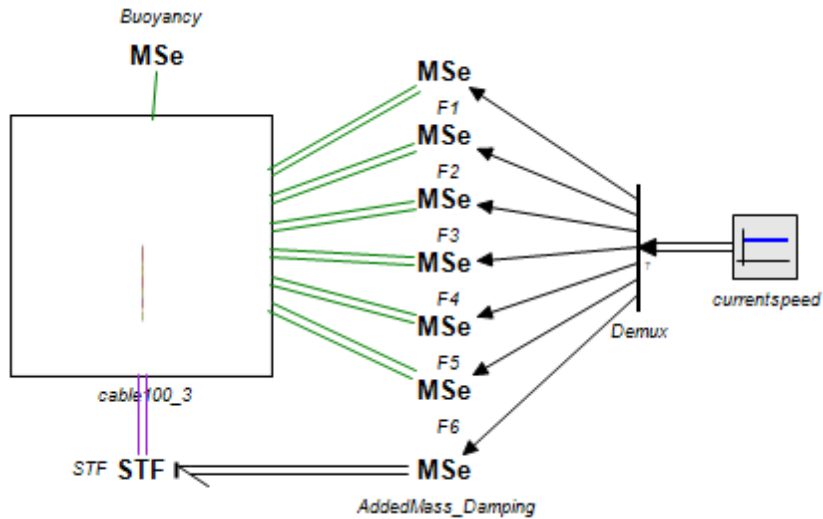


Figure 6.5 Cable + Load Model

6.4 Model Assessment

To assess the accuracy and authenticity of the cable and load model, an evaluation between 3D Mechanics model and standard DNV rules calculation is conducted. The 3D Mechanics model will have a series number of discretized FEM parts n and all parameters are set according to the distribution of physical properties without manual tuning.

6.4.1 DNV Rules Calculation

According to DNV-RP-H103 5.2 (DNV 2010), the stretched length L_s of a cable L is

$$L_s = L \left[1 + \frac{W + \frac{1}{2}wL}{EA} \right] [m]$$

where

- L_s = stretched length of cable [m]
- L = original length of cable [m]
- W = $Mg - \rho gV$ = fully submerged weight of lifted object [N]
- w = $mg - \rho gA$ = fully submerged weight per unit length of cable
- M = mass of lifted object [kg]
- m = mass per unit length of cable [kg/m]
- g = acceleration of gravity = 9.81 m/s²
- ρ = density of water [kg/m³]
- E = modulus of elasticity of cable [N/m²]
- A = nominal cross sectional area of cable [m²]
- V = displaced volume of lifted object [m³]

For an axially stiff cable with negligible bending stiffness the offset of a vertical cable with a heavy weight at the end of the cable in an arbitrary current with unidirectional (x-direction) velocity profile is

$$\zeta(z) = \int_z^0 \left[\frac{F_{D0} + \left(\frac{1}{2}\right) \rho \int_{-L}^{z_1} C_{Dn} D_c [U_c(z_2)]^2 dz_2}{W + w(z_1 + L)} \right] dz_1 [m]$$

where

$$F_{D0} = \frac{1}{2} \rho C_{Dx} A_x [U_c(-L)]^2 [N]$$

is the hydrodynamic damping force on the lifted object.

The parameters are defined as

- $\zeta(z)$ = horizontal offset at vertical position z [m]
 ζ_L = horizontal offset at the end of cable $z = -L$ [m]
 L = un-stretched length of cable [m]
 C_{Dn} = damping coefficient for normal flow past cable [-]
 C_{Dx} = damping coefficient for horizontal flow past lifted object [-]
 D_c = cable diameter [m]
 A_x = x-projected area of lifted object [m²]
 $U_c(z)$ = current velocity at depth z [m/s]
 $z1, z2$ = integration variables [m]

The formula can be transformed into

$$\zeta(z) = L \left(\frac{q}{w} K - \lambda \right) \ln \left[\frac{K + \frac{z}{L} + 1}{K + 1} \right] - \frac{q}{w} z$$

where

$$K = \frac{W}{wL} \quad \lambda = \frac{F_{D0}}{wL} \quad q = \frac{1}{2} \rho C_{Dn} D_c U_c^2$$

6.4.2 FEM in 3D Mechanics

In 3D Mechanics, the length and mass of each rigid bar element for a cable length L is

$$\Delta L = \frac{L}{2n}; \quad \Delta m = m \cdot \Delta L$$

The moment of inertia of each rigid bar element is

$$\Delta I_{xx,yy} = \frac{m \cdot \Delta L^3}{12}$$

The axial moment of inertia ΔI_{zz} is negligible.

The spring and damping coefficient of each rigid bar element are

$$\Delta k = EA/2\Delta L$$

$$\Delta c = 2\zeta \sqrt{(\Delta k \cdot m)}$$

6.4.3 Cable Parameters

The tested cable is Lankhorst Ropes' 6x36 WS+IWRC standard wire rope with higher breaking strength.

Table 6.1 6x36 WS+IWRC Parameters

Diameter [mm]	Weight [kg/m]	Elastic Modulus [GPa]	Structural Damping Ratio per meter* [ζ]
50	10.2	211	0.1

*including fluid friction

Table 6.2 Parameters of Cable and Load

C_{Dn}	C_{Dx}	A_x [m ²]	M [kg]	V [m ³]	A^* [m ²]	ρ [kg/m ³]	E [GPa]
1	1.2	3	20000	10	0.001178	1025	211

* effective area ratio = 0.6

6.4.4 Results

- Vertical Offset

Table 6.3 Vertical Offset with Lifted Load [m]

	L=10 m	L=100 m	L=1000 m
DNV	0.00387	0.0403	0.562
n=1	0.00387	0.0403	0.562
n=2	0.00387	0.0403	0.562
n=3	0.00387	0.0403	0.562

- Horizontal Offset

The horizontal assessment was conducted with variables of FEM parts number, current speed and linearity of the shape, i.e. without current force on load the cable will have higher nonlinearity of its shape.

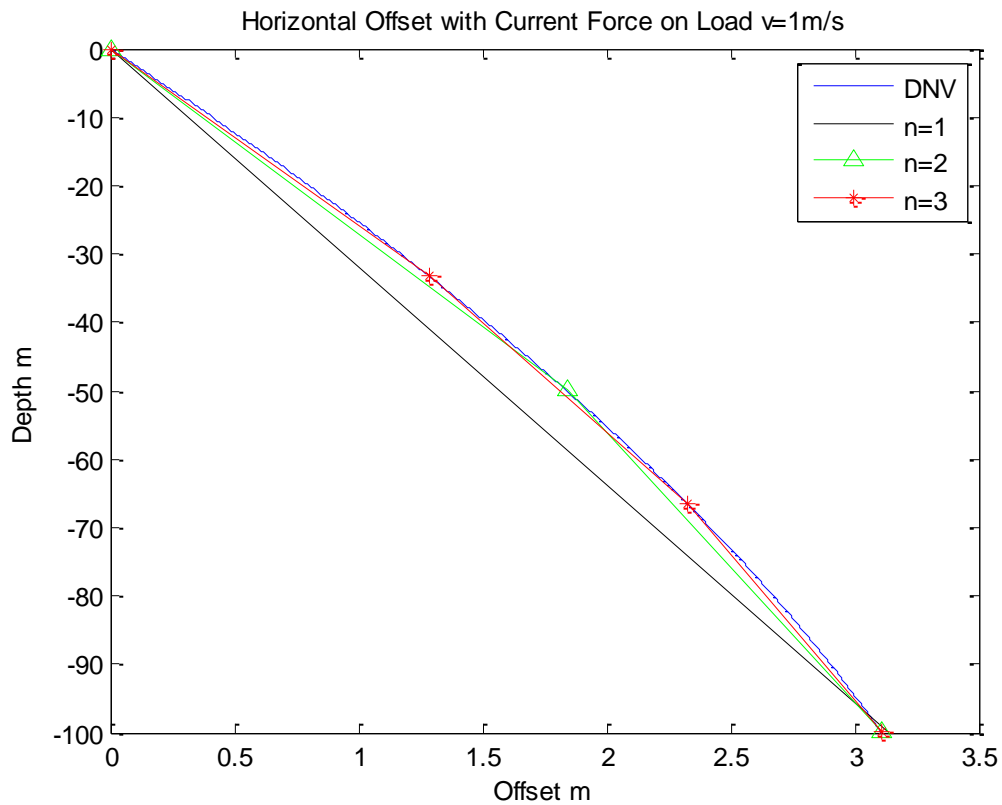


Figure 6.6 Horizontal Offset with Current Force on Load V=1m/s

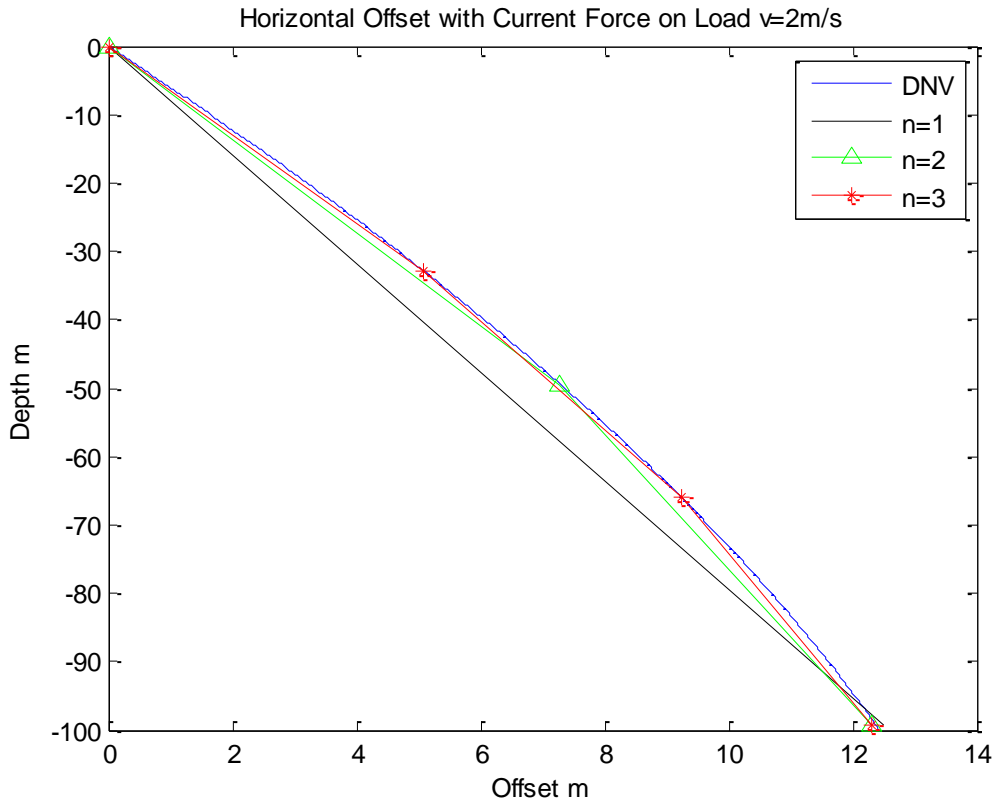


Figure 6.7 Horizontal Offset with Current Force on Load $V=2\text{m/s}$

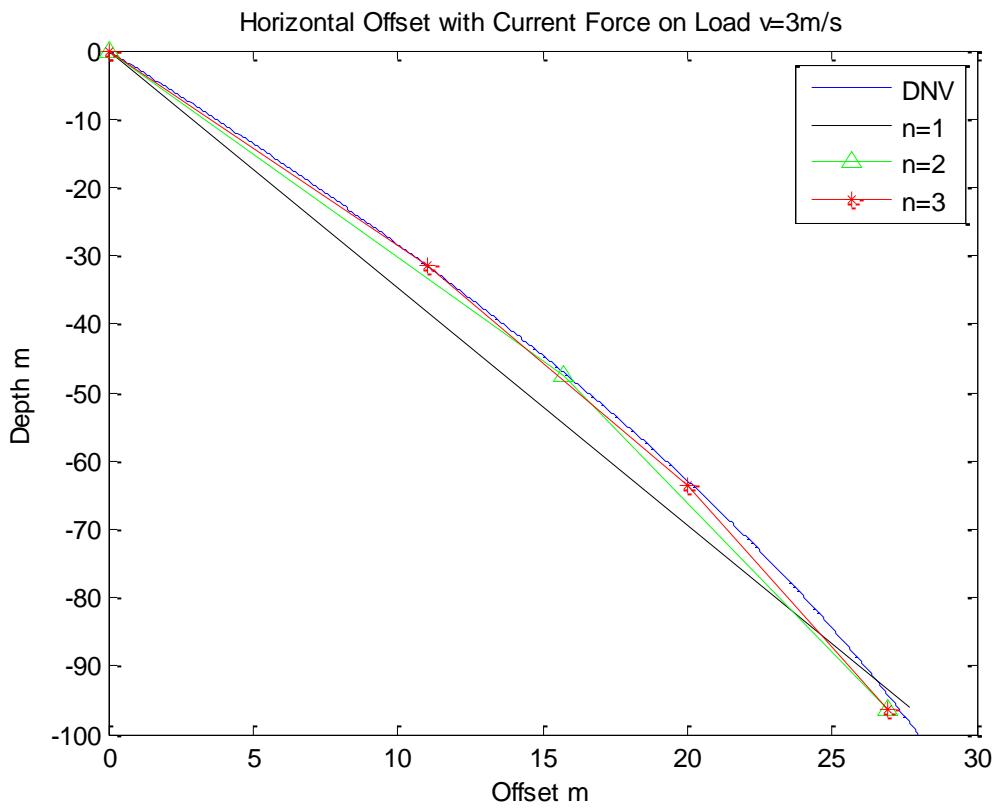


Figure 6.8 Horizontal Offset with Current Force on Load $V=3\text{m/s}$

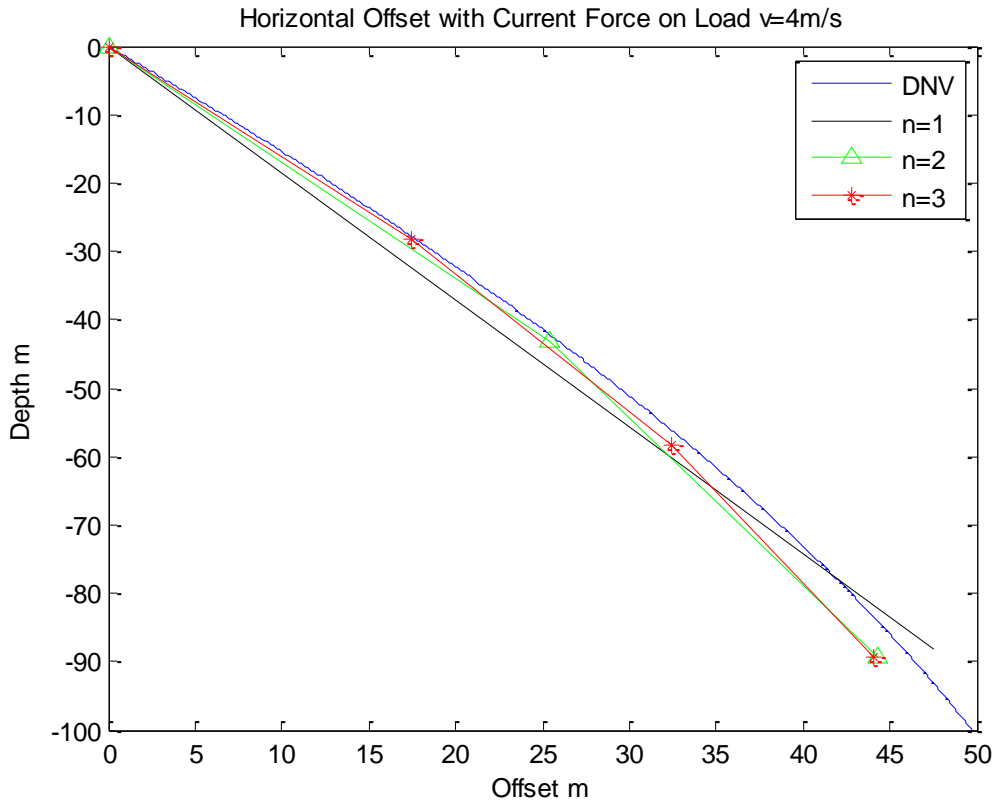


Figure 6.9 Horizontal Offset with Current Force on Load $V=4\text{m/s}$

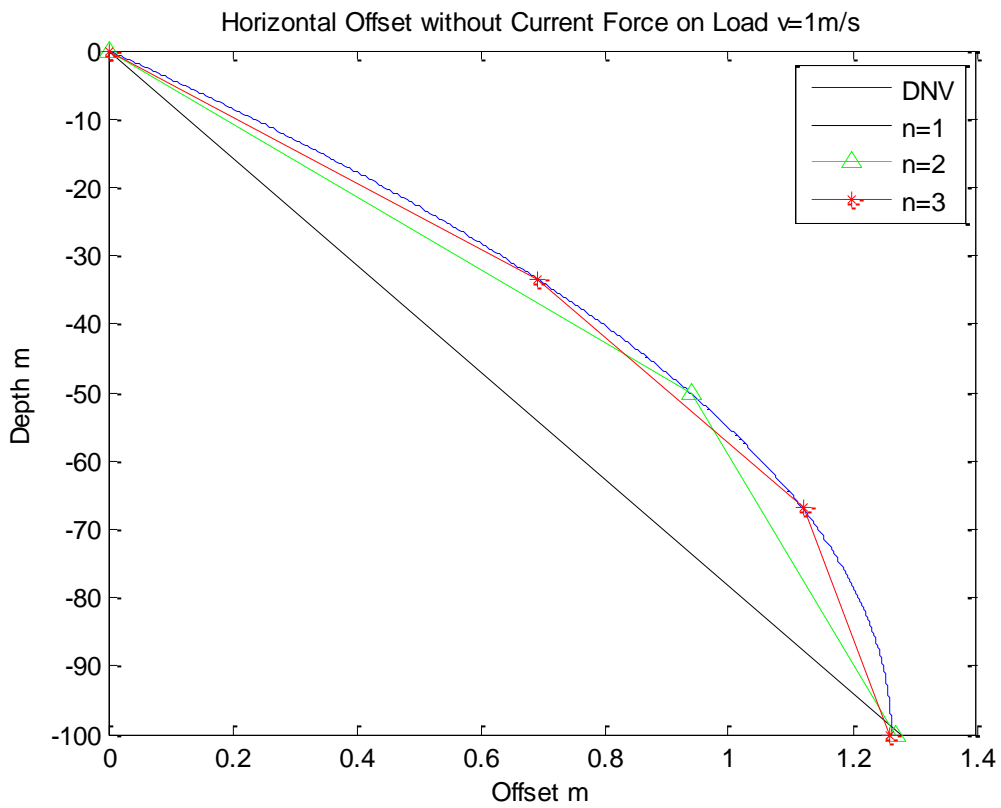


Figure 6.10 Horizontal Offset without Current Force on Load $V=1\text{m/s}$

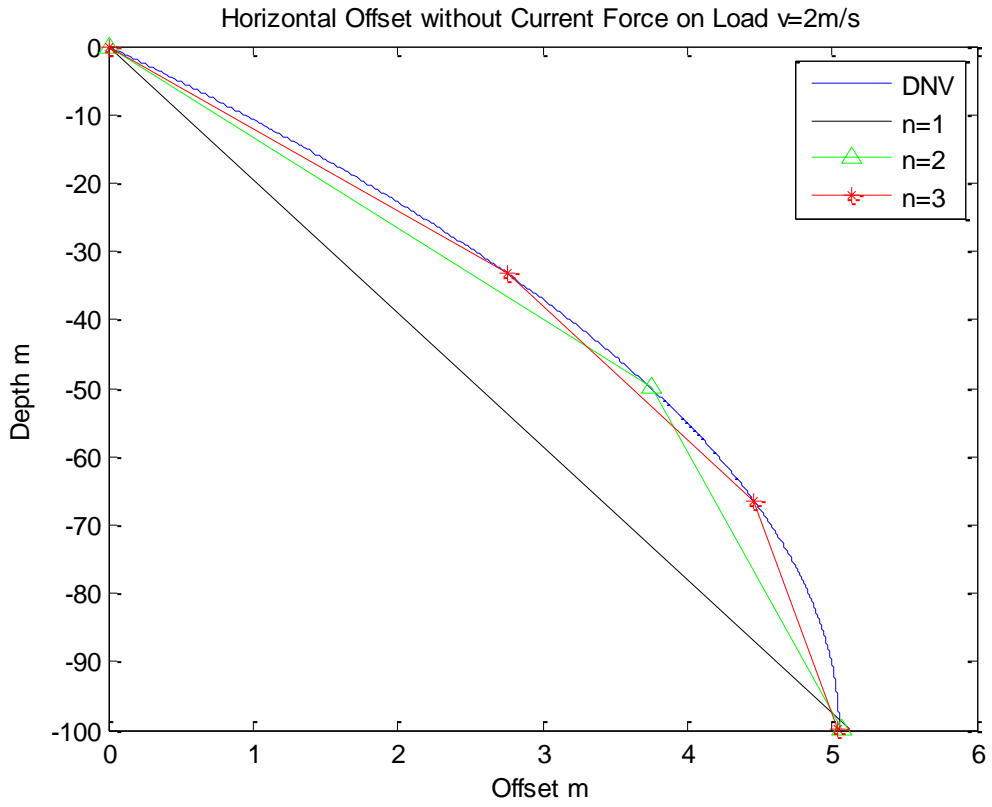


Figure 6.11 Horizontal Offset without Current Force on Load $V=2\text{m/s}$

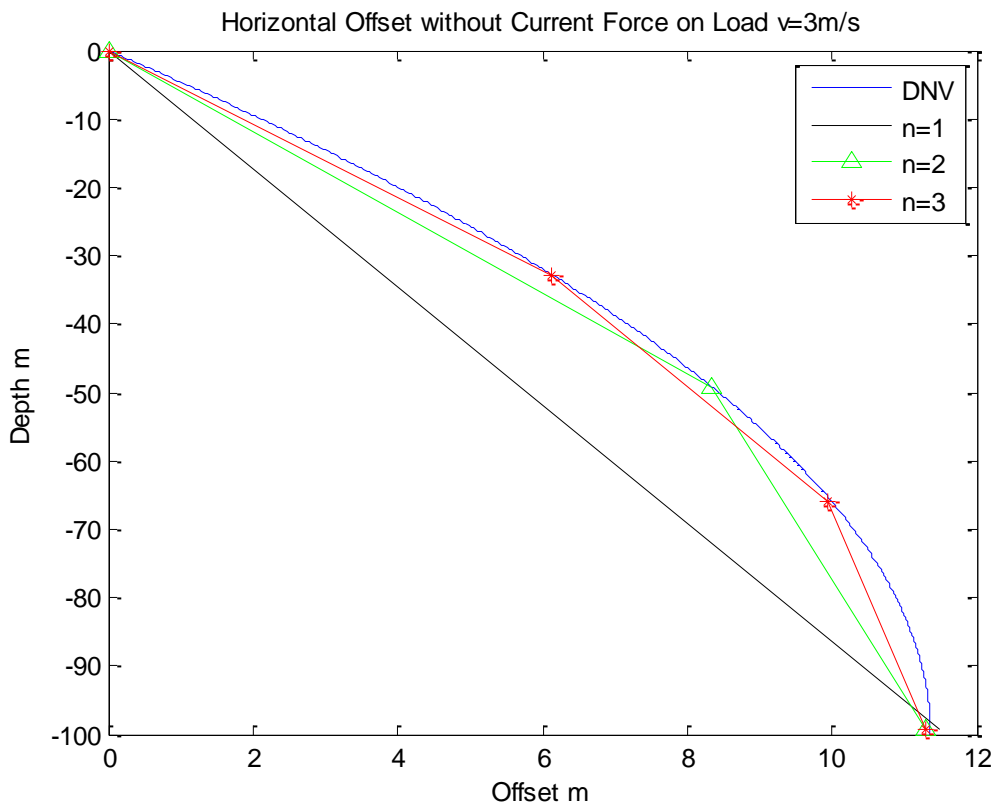


Figure 6.12 Horizontal Offset without Current Force on Load $V=3\text{m/s}$

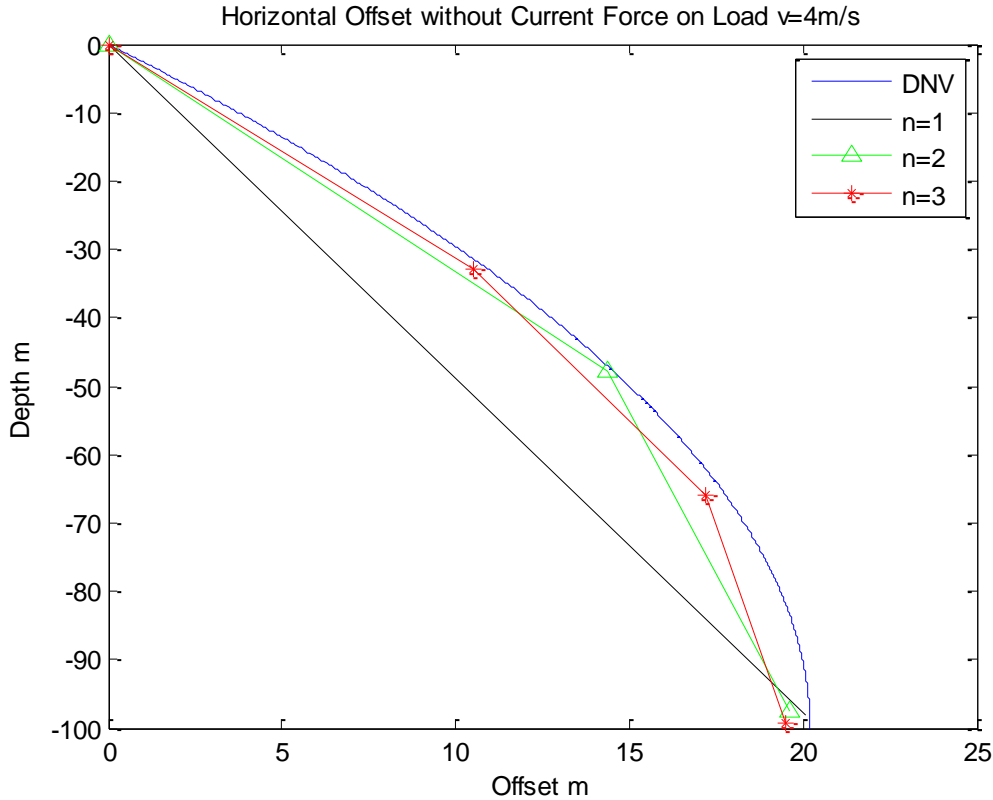


Figure 6.13 Horizontal Offset without Current Force on Load $V=4\text{m/s}$

6.4.5 Discussion

- Vertical Offset

The FEM vertical offset had a perfect compliance with DNV rules no matter how much FEM parts were set because both DNV rules calculation and FEM have analytical results of the problem. Thus for cases where people only interest in vertical offset without current force, a single straight line with one translational connector is sufficient enough.

- Horizontal Offset

The result shows that 1) under small horizontal offset condition ($\zeta/L < 1:5$), the offset at the bottom end of the cable has good compliance with DNV rules calculation result; 2) under large horizontal offset condition ($\zeta/L \geq 1:5$), the bottom end of the cable has significant deviation from DNV rules calculation result; 3) under small horizontal offset condition, nonlinearity shape of the cable trajectory does not affect the accuracy of FEM model.

The reason for significant deviation under large horizontal offset condition is that by the definition of DNV rules:

For an axially stiff cable with negligible bending stiffness the offset of a vertical cable with a heavy weight at the end of the cable in an arbitrary current with unidirectional (x-direction) velocity profile is

$$\zeta(z) = \int_z^0 \left[\frac{F_{D0} + \left(\frac{1}{2}\right) \rho \int_{-L}^{z_1} C_{Dn} D_c [U_c(z_2)]^2 dz_2}{W + w(z_1 + L)} \right] dz_1 \quad [m]$$

the formula is only valid under small-angle approximation where the vertical depth of the cable equals to the axial length of the cable,

$$\sin \theta \approx \tan \theta \approx \theta \text{ when } \theta \text{ is small angle}$$

yet large horizontal offset cases ($\zeta/L \geq 1:5$) cannot be treated under small-angle approximation anymore, where the resultant cable length under large horizontal offset is significantly longer than the depth. The large horizontal offset condition also contradicts the premise of '*a vertical cable with a heavy weight at the end of the cable*' from the description of the formula.

However, the FEM model in 3D Mechanics is built according to a fixed cable length. Large horizontal offset will cause the end of the cable to be lifted hence a smaller depth. From the small horizontal offset cases the validity of FEM model has been proved. For large horizontal offset condition the FEM model is still applicable for simulation purpose but requires extra evidence to support.

Since under small horizontal offset condition, the FEM model shows good compliance however the parts number is, the choice of the part number only depends on the distribution of current force and nonlinearity of the cable. Higher parts number will have better approximation of the shape, finer distribution of the current force and more accurate dynamic simulation result. But it will also increase the computation amount and time. The decision shall be made under different practical conditions.

7 SIMULATION EXAMPLE

All sub-models introduced in Chapter 3-6 can be assembled into one big model selectively. The interactive physical entities are connected with power port whereas control loop are connected with signal port. Some examples are demonstrated as follows. **Because of the computation limits on PC, hydraulic sub-model was replaced with direct PID control.**

7.1 Vessel + Crane

7.1.1 Model

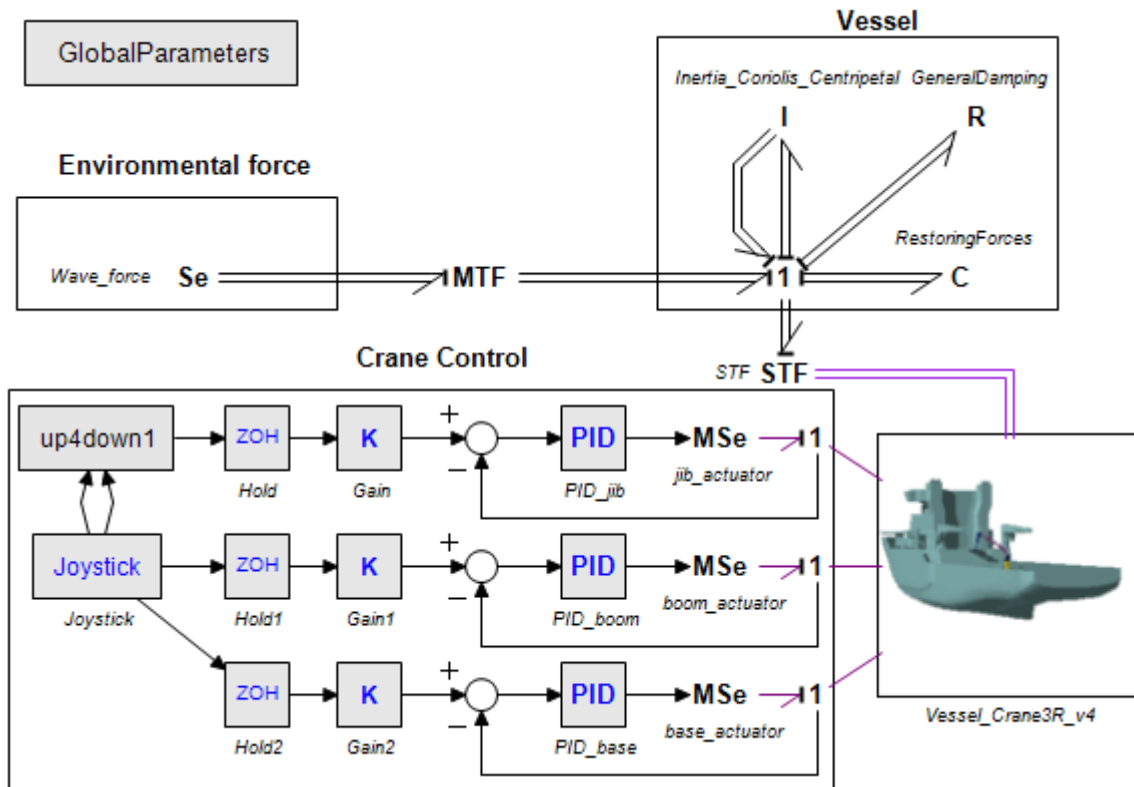


Figure 7.1 Simulation Example: Vessel + Crane

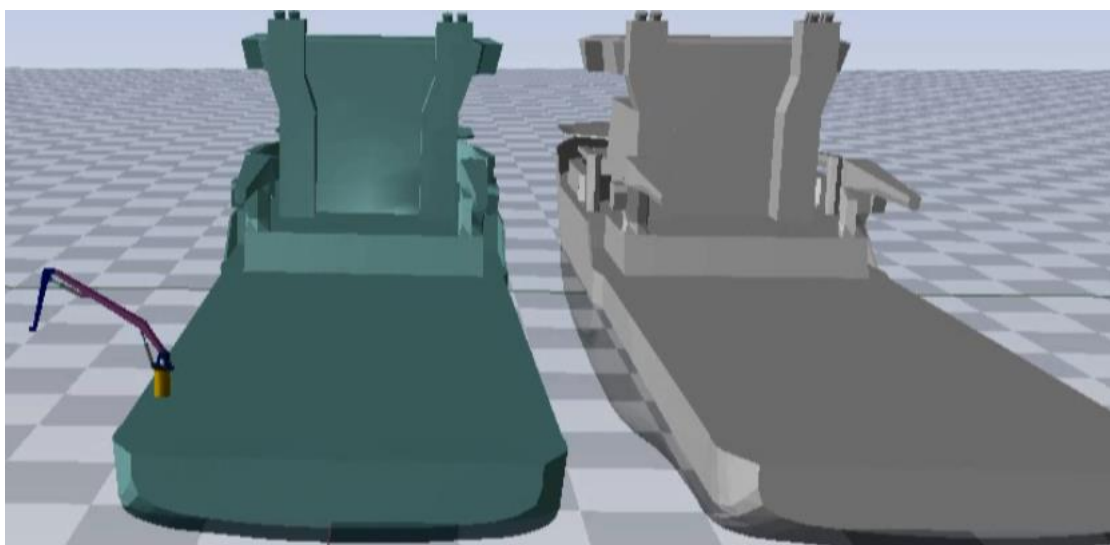


Figure 7.2 Animation: Vessel + Crane

7.1.2 Results

The default ship 's175' from ShipX database is used. The video in the appendix showed the animation of the target vessel (cyan) comparing with another vessel (grey) with SHIPX output movement, under 10 s wave period, 60 deg heading and 1 m wave height. Figure 7.3 shows the 6DOF motion of vessel model in 20-sim during the start of the simulation (transient state).

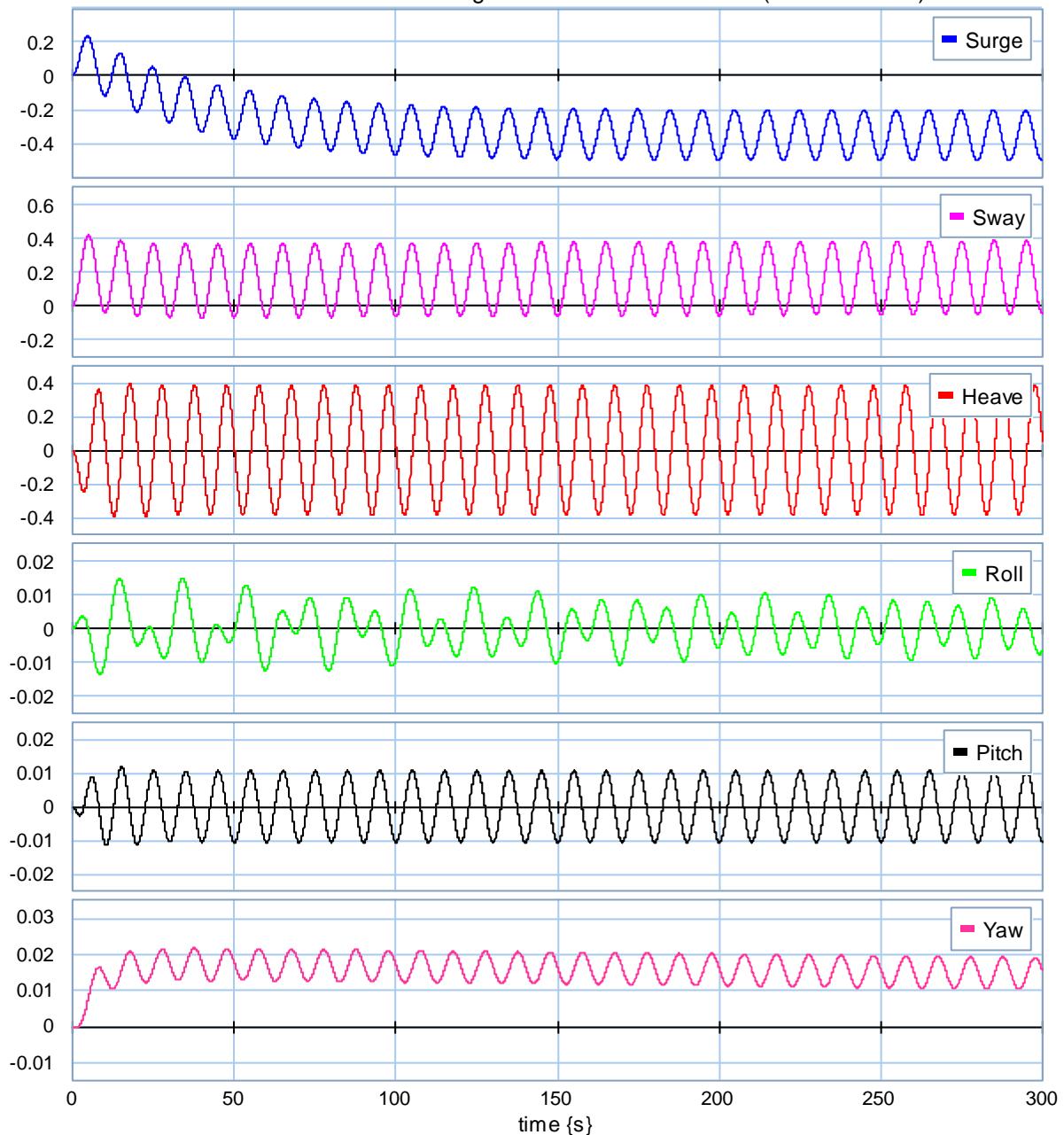


Figure 7.3 Vessel Motion: Vessel + Crane at Transient State

Figure 7.4 shows the roll motion of the vessel model during steady state (1000-1180 sec), which means transient oscillation has died out.

Figure 7.5 shows the 6DOF motion of the vessel according to ShipX RAO. All motions are harmonic signals meaning there is not distinction of transient or steady state.

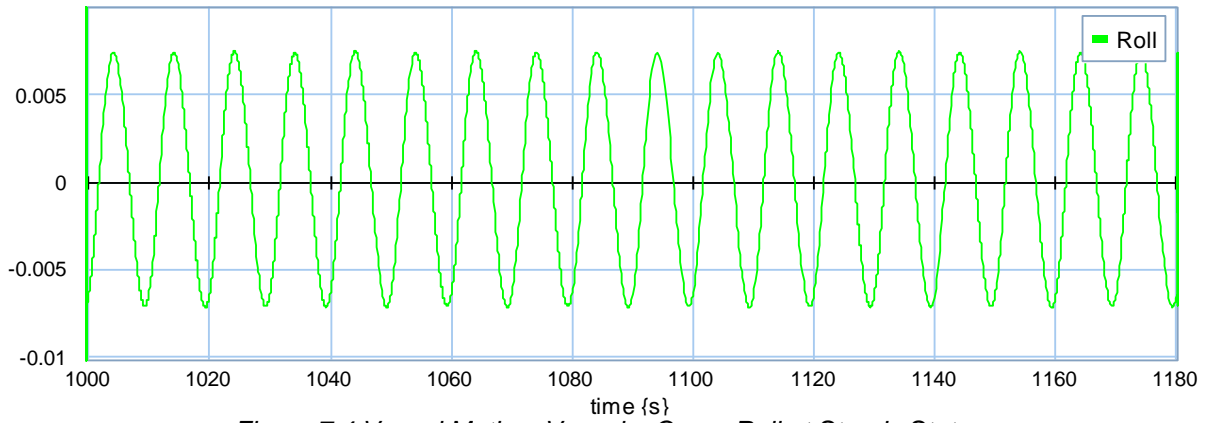


Figure 7.4 Vessel Motion: Vessel + Crane Roll at Steady State

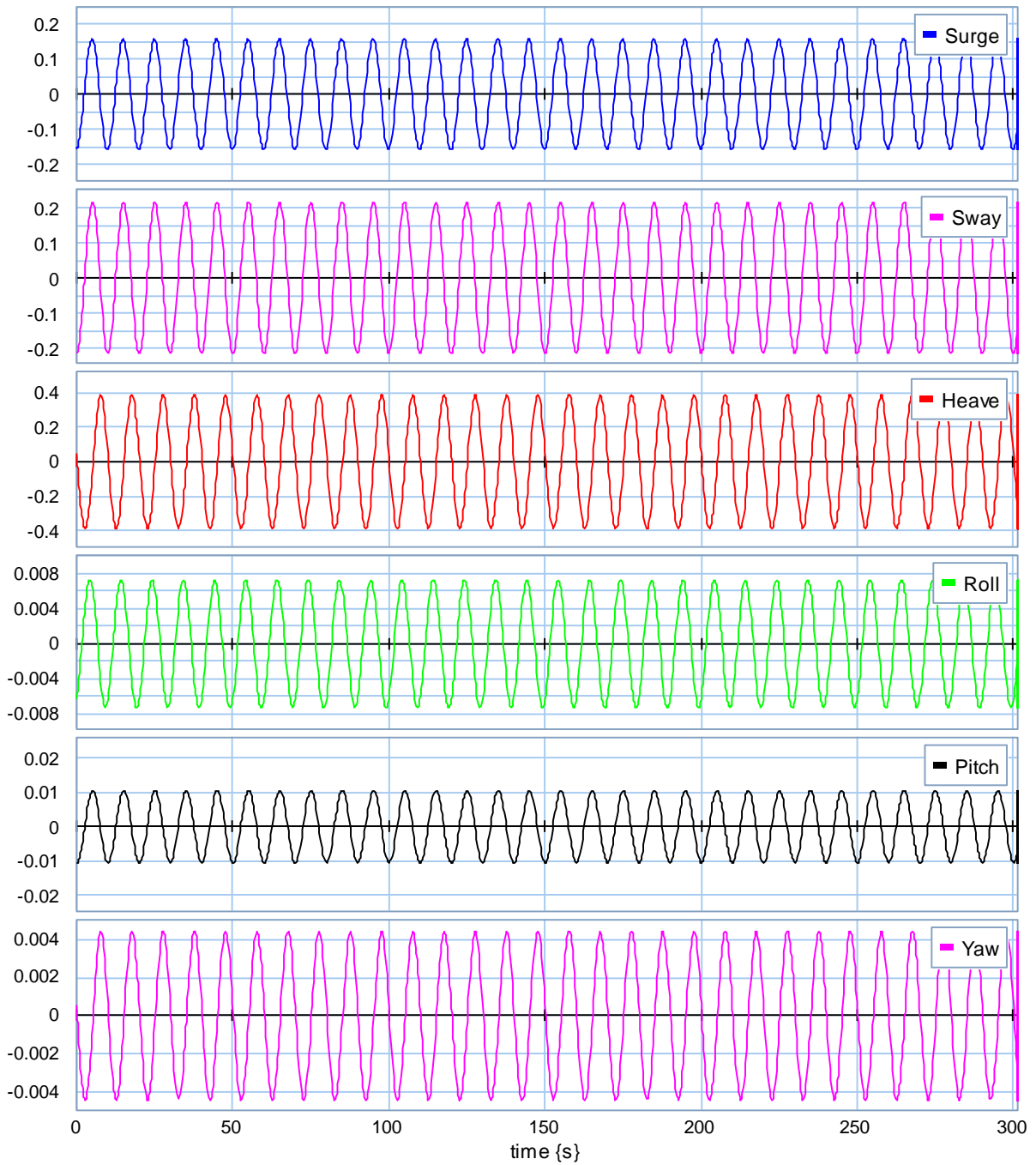


Figure 7.5 Vessel Motion: Vessel + Crane from SHIPX

7.1.3 Discussion

As the plot shows

- 1) There is deviation of 20-sim model from SHIPX RAO at the start of simulation, which is because the model was at transient state, the vessel has both natural frequency oscillation and forced frequency;
- 2) The transient part of roll motion died out slower than the other motions because rolling has the smallest damping of them all;
- 3) The deviation of surge, sway or yaw generated during the transient state will be kept during the steady state because there is no restoring force for those three degrees of freedom.
- 4) After entering their steady state, the two sets of data (20-sim and SHIPX) have the identical amplitude and frequency. For natural resonance cases mentioned in Chapter 3 where 20-sim model has a deviated performance from ShipX RAO, the damping coefficient can be manually tuned accordingly.

7.2 Vessel + Crane + Cable + Load

7.2.1 Model

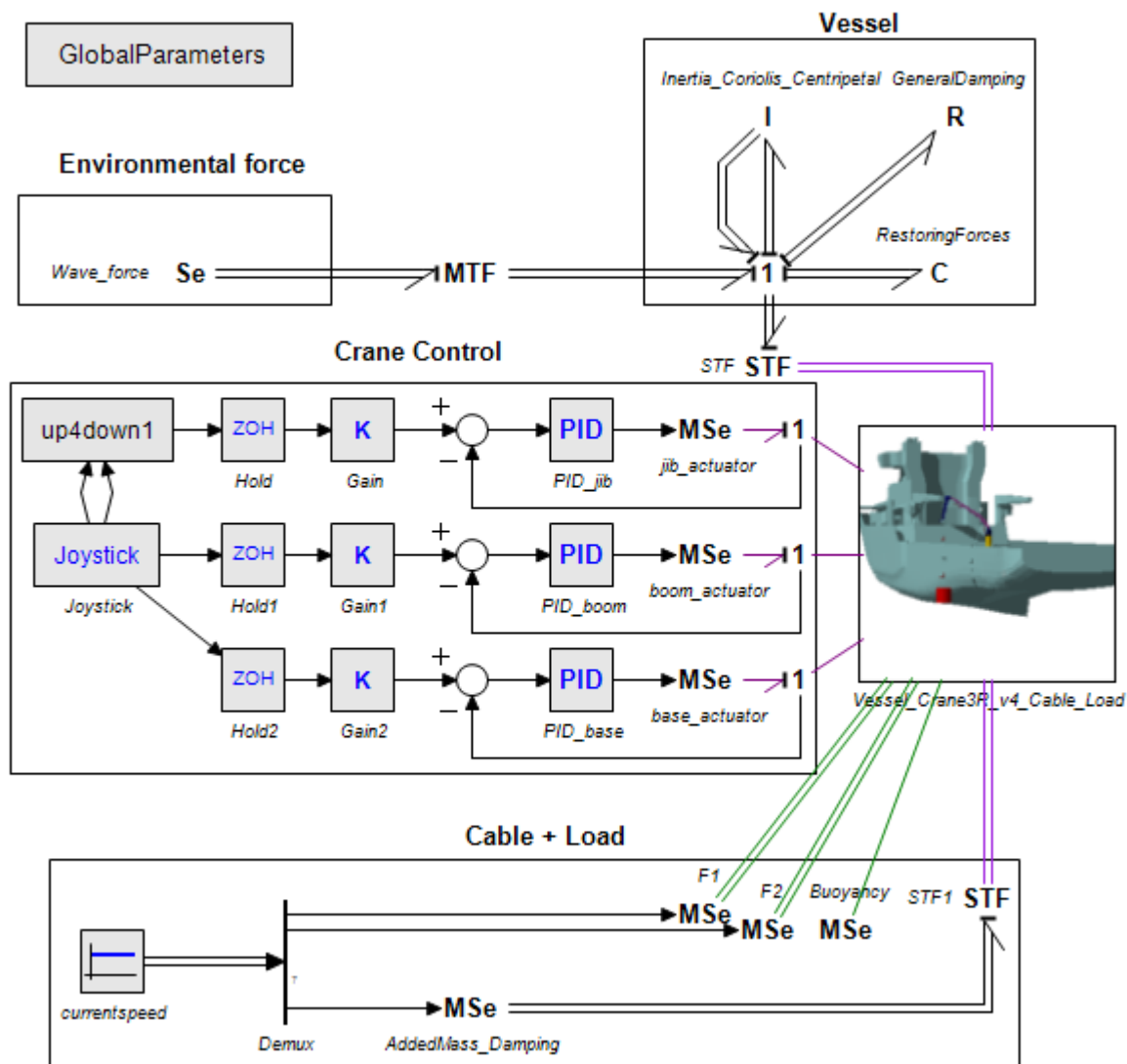


Figure 7.6 Simulation Example: Vessel + Crane + Cable + Load

With addition of cable and load model, the interaction between the crane and vessel can be seen in the simulation. Apart from vessel + crane animation, the video in the appendix also showed how vessel is leaning to the crane side due to the effect of the heavy load (50 t & 100 t) . As for now, added mass and damping of load model is set as constant. ($C_a = 1$; $C_d = 1.2$)

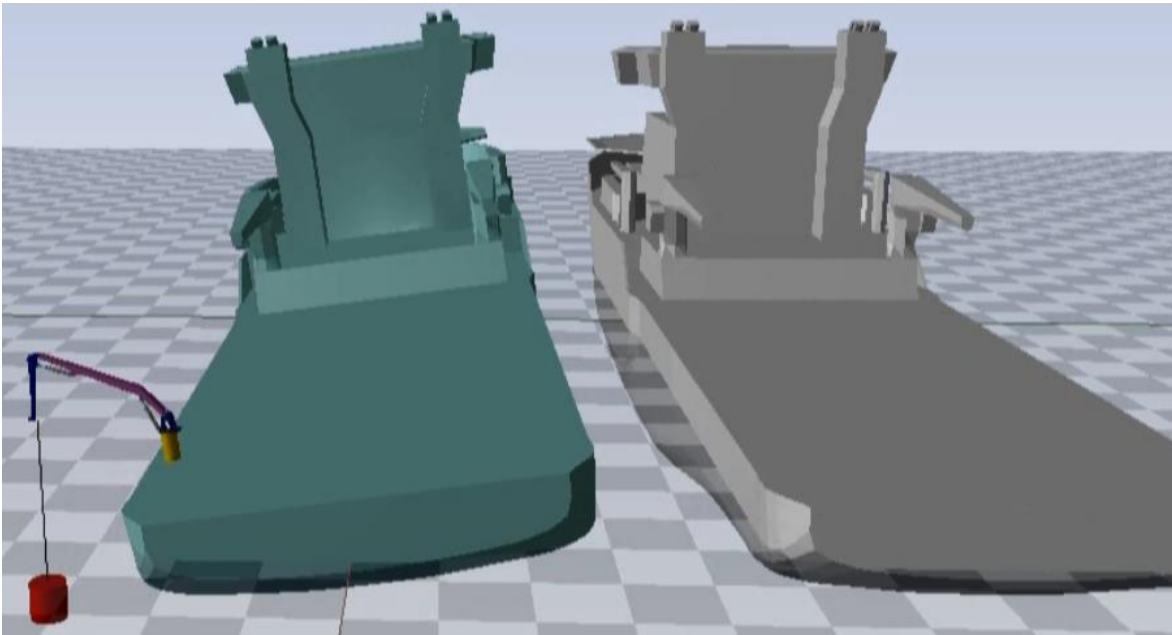


Figure 7.7 Animation: Vessel + Crane + Cable + Load (100 t)

The dynamic behaviour of the cable and load is shown in another video because it is hard to identify the effect in the entire model. Moreover, the cable model will significantly increase the complexity of the model hence the computation amount. As stated in Chapter 6, for most cases, the cable can be represented as an straight bar.

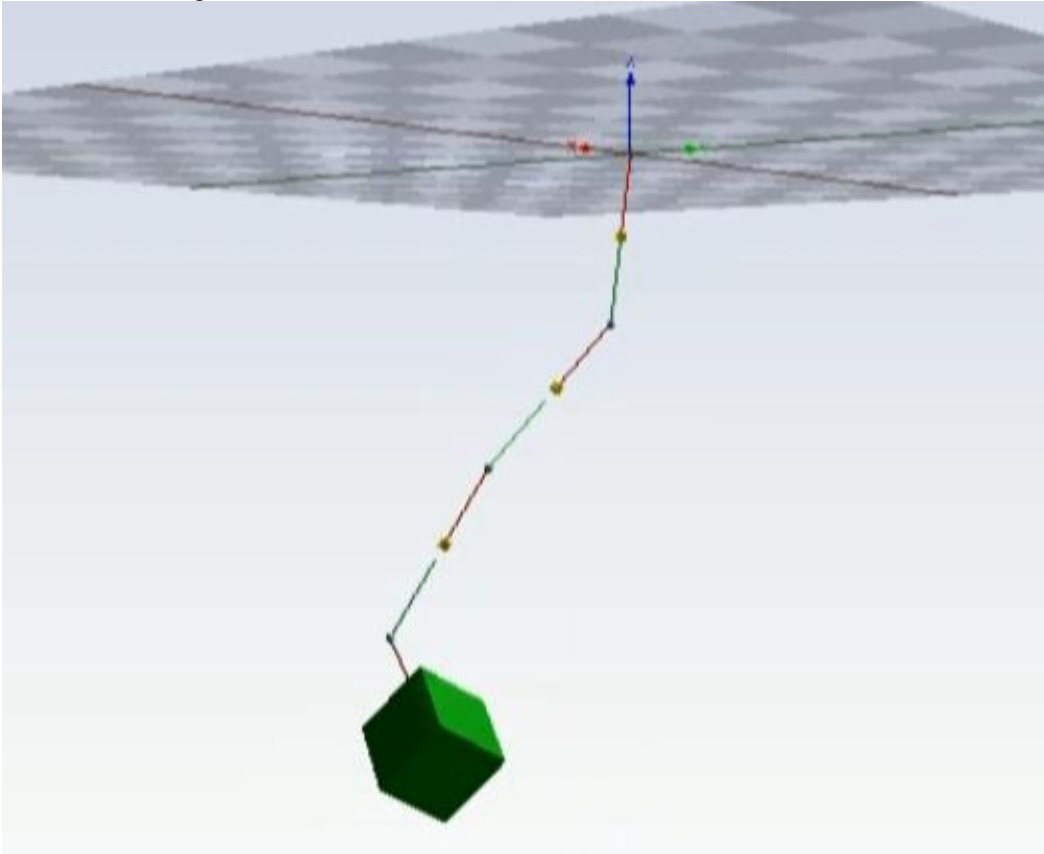


Figure 7.8 Animation: Dynamic Behaviour of Cable + Load

7.2.2 Results

Figure 7.9 shows the 6DOF motion of vessel model with cable and 50t of load in 20-sim during the start of the simulation (transient state).

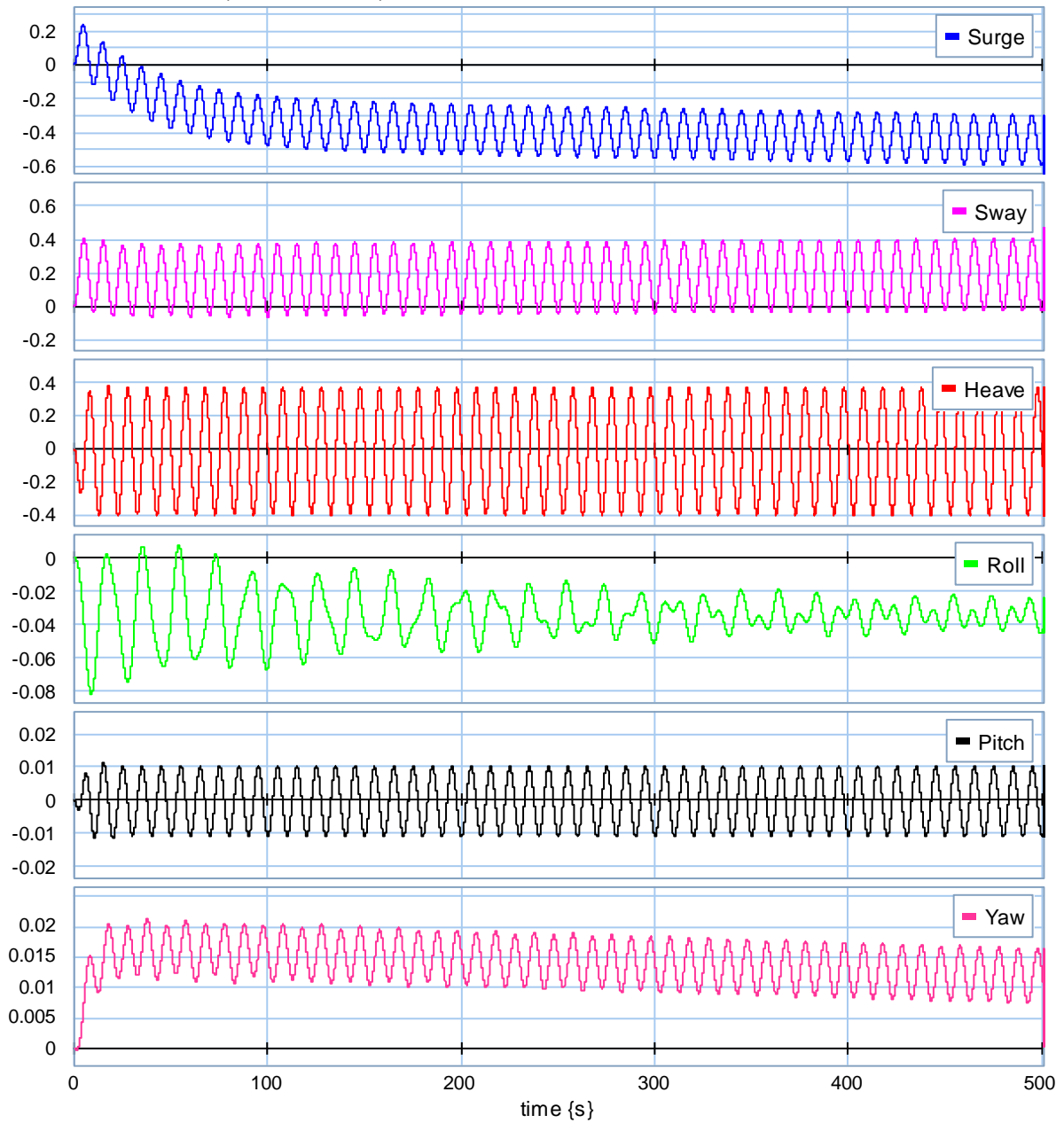


Figure 7.9 Vessel Motion: Vessel + Crane + Cable + Load (50 t) at Transient State

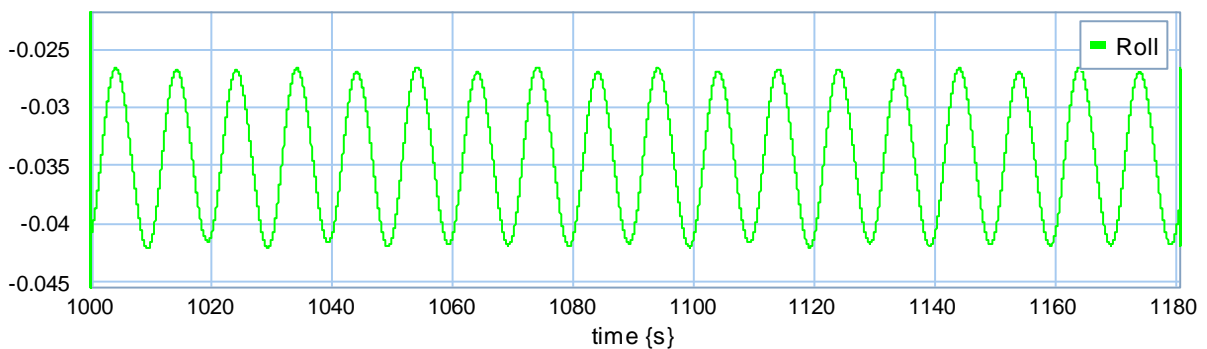


Figure 7.10 Vessel Motion: Vessel + Crane + Cable + Load (50 t) Roll at Steady State

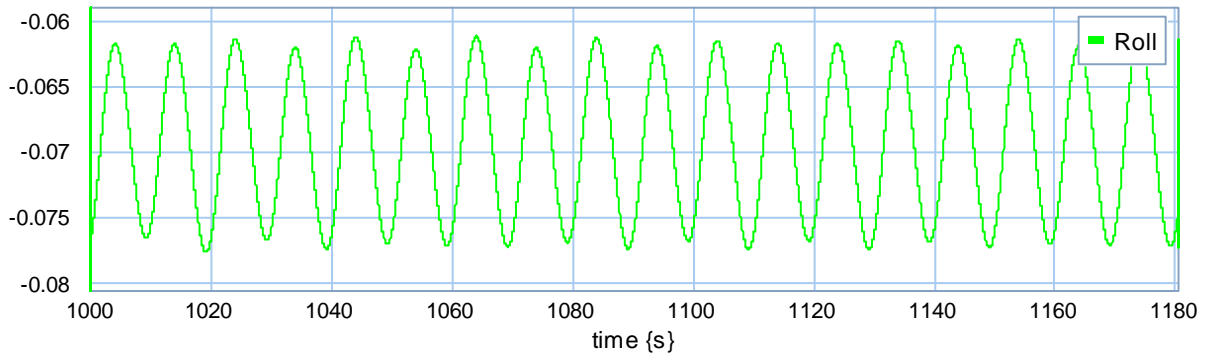


Figure 7.11 Vessel Motion: Vessel + Crane + Cable + Load (100 t) Roll at Steady State

Figure 7.10 and 7.11 show the roll motion of vessel model during transient state with the load weight of 50t and 100t respectively.

7.2.3 Discussion

As the plot shows

- 1) After putting load and cable into the model, the roll motion of the vessel has higher irregularity at the transient state and leans to the load side due to the pulling force.
- 2) The roll motions of the vessel with load of 50t or 100t have different steady angles (0.035/0.07 rad) yet identical amplitude at steady state. This is because although the load is heavy enough to cause dramatic leaning of the vessel, it is still not heavy enough to significantly change the natural frequency of roll motion, and the damping coefficient of the vessel is constant regardless of the leaning angle. Therefore the amplitude of the forced oscillation of the vessel will have no observable changes.
- 3) Motions without restoring force (surge, sway and yaw) continue to deviate in the steady state as a result of the coupling effect between different degrees of freedom. Counter-measures including potential dynamic positioning method is to be expected. The deviation is also due to the limited integration accuracy during the simulation.

7.3 Vessel + Crane + Cable + Load + AHC

7.3.1 Model

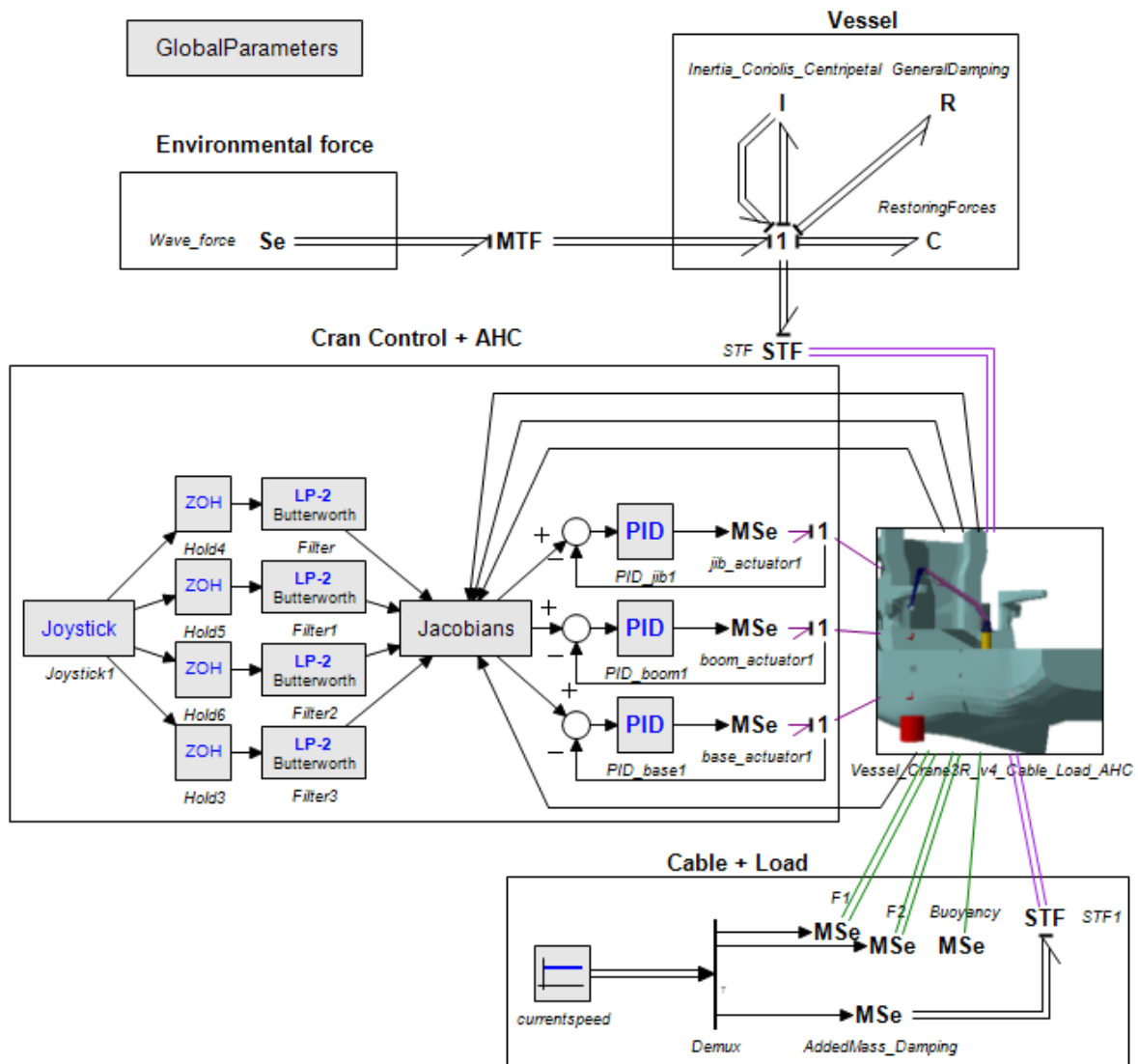


Figure 7.12 Simulation Example: Vessel + Crane + Cable + Load + AHC

With the help of AHC system and possibly other control scheme, the crane can not only be manually but also automatically controlled. The video in the appendix also showed how the load is being controlled by AHC system.

7.3.2 Results

Figure 7.13 is the heave motion of the crane pedestal and the load without AHC system, from transient state at the beginning to the steady state at the second half.

Figure 7.14 is the heave motion of the crane pedestal and the load with AHC system, from transient state at the beginning to the steady state at the second half.

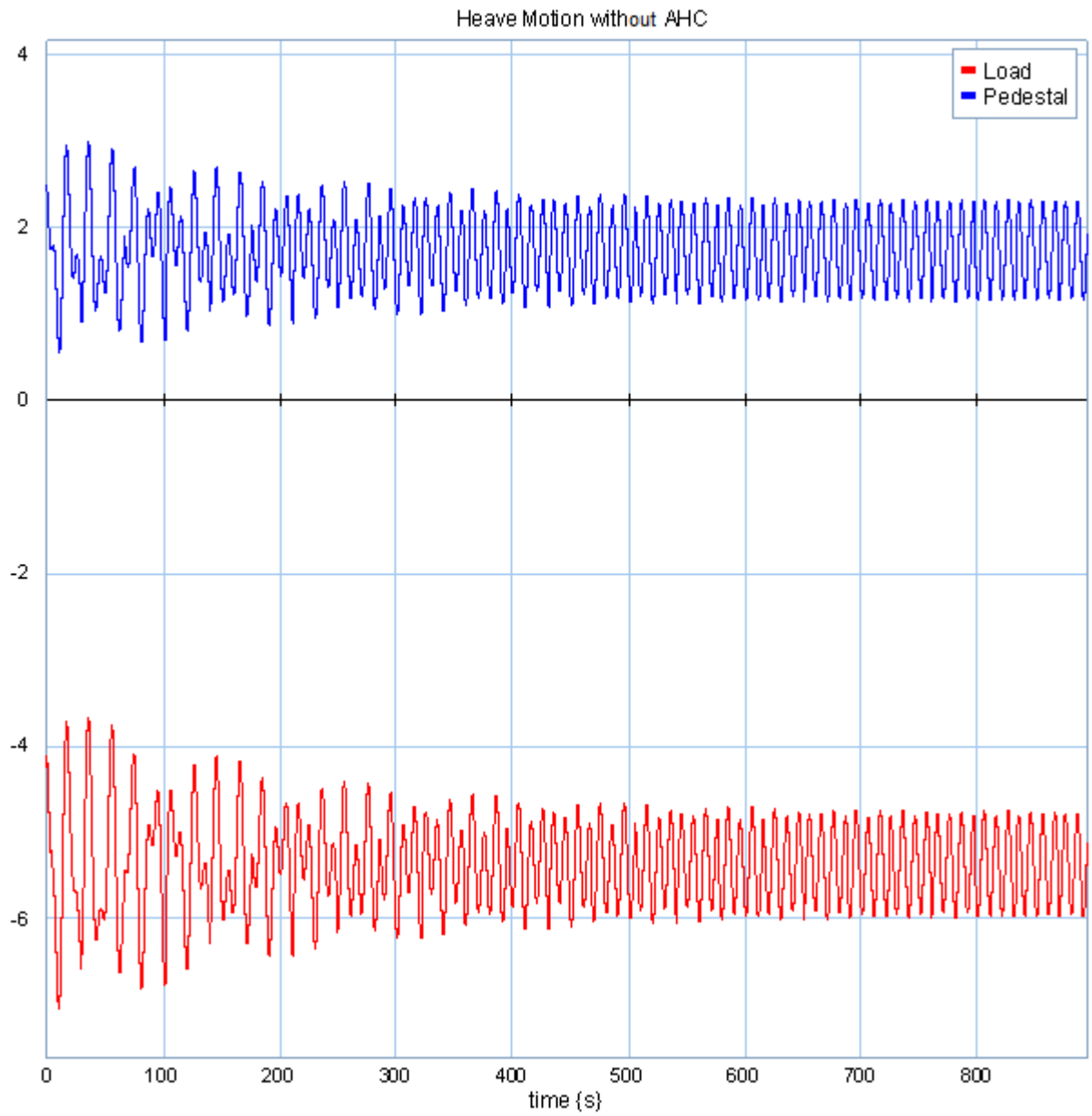


Figure 7.13 Load & Pedestal Heave: Vessel + Crane + Cable + Load (100 t)

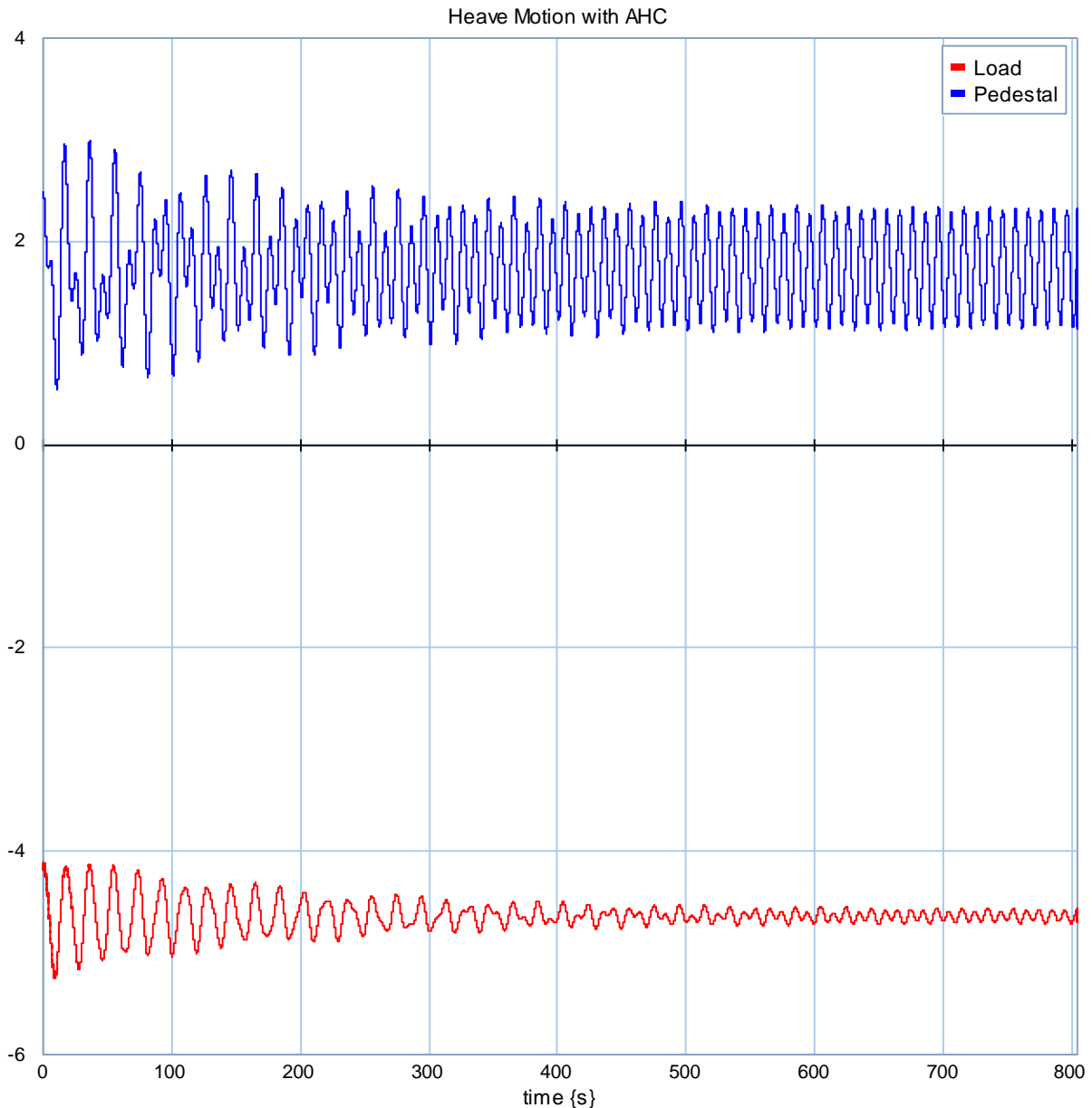


Figure 7.14 Load & Pedestal Heave: Vessel + Crane + Cable + Load + AHC (100 t)

7.3.3 Discussion

As the plot shows

- 1) Without AHC system, the heave motions of the load and crane pedestal are identical, because the crane does not have relative motion between them and the small deviation caused by rotational motion can be ignored.
- 2) With the effect of AHC system, the load does not have good compensated motion at the beginning of transient state, but the amplitude gradually decreases while entering to the steady state. This is because the AHC system used in the model so far is based on crane kinematics, which has limitations and response time. It has better performance towards regular and small motions rather than irregular and drastic motions. Winch based AHC system may have different performance in this case, which can be investigated in the future.

8 CONCLUSION & FUTURE WORK

8.1 Conclusion

This thesis demonstrated a generic modelling approach for marine operation. The aim is to propose a standard application of modelling each sub-system in the marine operation however the design is. The model consists of several modelling methods from other researchers yet with necessary modification to make a compatible system. The general second order differential equation of vessel hydrodynamics is from (Fossen and Fjellstad 2011). MATLAB files for ShipX data transformation is from (Fossen and Perez 2014). Bond graph modelling technique is from (Pedersen 2008). Crane hydraulics and control model is from (Chu 2013). Cable model inspired by (Johansson 1976). Load model inspired by (Halse, et al. 2014). Combined the knowledge of previous researches, the modelling method proposed in this thesis enables user to assemble all sub-models into a complete operation model. The complete model can then be served for virtual dynamic analysis or training purpose. In the complete model, most of the 3D mechanics modelling are done inside the 3D Mechanics, a toolbox of 20-sim, but with connectors, actuators and sensors, the 3D Mechanics model receives information and interacts with bond graph and control scheme outside. The bond graph and control scheme expand the function of 3D Mechanics model and give user the freedom to modify the design. The realistic physical entities interacts as power flow and information interacts as signal flow, which gives a clear idea to separate the different level of modelling. Besides crane, a winch, a propeller or a drilling tower can also be modelled by following the same idea.

8.2 Future Work

As a part of the ongoing research, there are many things yet to be done. So far the modelling method is still based on 20-sim, which has many restrictions, e.g. only diagonal matrix of inertia tensor is allowed inside 3D Mechanics; only linear spring/damping is allowed inside 3D Mechanics; the function of real-time interaction with Matlab has not been realised. In the future development, a better platform or direct programming of physical engine may be required to expand the function of the model. Also, in 3D Mechanics, the model is eventually being solved as a big matrix yet only occupying single CPU thread, which is time consuming and inefficient. Multithreads solution using GPU and parallel computation should be applied in the future. The model also requires collision property and event detector to control the different state of marine operation, e.g. crossing splash zone will have a different model which requires splash model and detectors to detect the start and the end of the phase.

The ideal model shall have the function of 'Plug and Play' which minimize the modelling procedure and maximize the flexibility for different models. Thus, a standard interface between sub-models is also required. Different parties shall have the same UI to develop their own sub-model, which in 3D gaming industry often called as MOD (modification). Many sub-models can then be added into a product library let users and customers to choose and test. Remote control and local PC experience is also to be expected in the final package. There are still so much to be done.

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APPENDIX

Appendix A **User Manual**

The manual explains the steps from ShipX calculation to *.mat file transfer and eventually 20-sim simulation.

1. Install MSS toolbox into your 32-bit Matlab. (see <http://www.marinecontrol.org/download.html>)
2. Run SHIPX Veres, get .re1, .re2, .re7, .re8, .hyd data files (see 2008_06_19_MSS_vessel_models.pdf, page.18-20). If you do not have .re2 file, comment on line 131 and line 136 in veres2vessel.m. the following calculation options and settings must be chosen to comply with the MSS data structure.
 - i. Ordinary strip theory (recommended but other methods can be used)
 - ii. Added resistance – Gerritsma & Beukelman
 - iii. Generate hydrodynamic coefficient files (*.re7 and *.re8)
 - iv. Calculation options: choose z-coordinates from CO (**O_b**).
 - v. Vessel velocities must always include the zero velocity: it is optional to add more velocities that are needed for manoeuvring.
 - vi. Wave periods: it is recommended to use values in the range 2.0s to 60.0s.
 - vii. The wave heading must be chosen every 10 deg starting from 0 deg.
3. Put SHIPX Veres data files, vereas2vessel.m, into the same folder and set it as the working folder in Matlab.
4. Type veres2vessel('input') in Matlab, generate *.mat file (see 2008_06_19_MSS_vessel_models.pdf, page.20-21).
5. Add 20SIM script toolbox 'library' and its subfolders into your Matlab toolbox path.
6. Run simulation.m and follow the commands.
7. Modify the 'parameters' block in 20SIM model to modify the rest parameters to run different simulations.
8. With 3D animation plot, you can directly compare the difference between results from SHIX (grey) and 20SIM (cyan).

Appendix B *Matlab scripts*

● 20-sim running

```
clear;
%% Connect to 20 SIM
xxsimConnect();

% Open the model Simple vessel.emx
fprintf('Please select a 20SIM model,press [Enter] to continue\n');
fprintf('\n');
pause;
[modelname,modelpath] = uigetfile({'*.emx','20SIM-Model(*.emx)'},'Select
20SIM Model');

% Check if the user pressed cancel on the dialog.
if isequal(modelname,0) || isequal(modelpath,0)
    disp('Program cancelled');
    return
else
    disp(['User selected:', fullfile(modelpath, modelname)]);
end
xxsimOpenModel(fullfile(modelpath,modelname));

% process the model
xxsimProcessModel();
% Load *.mat data file
fprintf('Please select a SHIPX data, press [Enter] to continue\n');
fprintf('\n');
pause;
[dataname,datapath] = uigetfile({'*.mat','Matlab-data(*.mat)'},'Select
SHIPX data');

% Check if the user pressed cancel on the dialog.
if isequal(dataname,0) || isequal(datapath,0)
    disp('Program cancelled');
    return
else
    disp(['User selected:', fullfile(datapath, dataname)]);
end
load(fullfile(datapath,dataname));
%% Select condition
% Define target vessel speed
knot = vessel.velocities/1.852*3.6;
fprintf('\nThe vessel speeds in knots are [');
fprintf('%g ',knot);
fprintf('\b).\n');
vel = str2num(input('Choose target vessel speed in knots:', 's'));
for i=1:length(vessel.velocities)
    if abs(vel-knot(i))<0.1
        velno=i;
        break
    end
end
% Define target wave period
wp = 2*pi./vessel.freqs;
fprintf('\nThe wave periods are [');
fprintf('%g ',wp);
fprintf('\b).\n');
period = str2num(input('Choose wave period:', 's'));
for j=1:length(vessel.freqs)
    if abs(period-wp(j))<0.1
```

```

        freqno=j;
    break
end
end
% Define target wave heading
hd = vessel.headings*180/pi;
fprintf('\nThe wave headings are [ ');
fprintf('%g ',hd);
fprintf('\b].\n');
head = str2num(input('Choose wave heading:', 's'));
for k=1:length(vessel.headings)
    if abs(head-hd(k))<0.1
        headno=k;
        break
    end
end

% ForceRAO
forceRAOamp = zeros(6,1);
forceRAOphase = zeros(6,1);
for i=1:6
    forceRAOamp(i,1) = vessel.forceRAO.amp{i}(freqno,headno,velno);
    forceRAOphase(i,1) = vessel.forceRAO.phase{i}(freqno,headno,velno);
end

% MotionRAO
motionRAOamp = zeros(6,1);
motionRAOphase = zeros(6,1);
for i=1:6
    motionRAOamp(i,1) = vessel.motionRAO.amp{i}(freqno,headno,velno);
    motionRAOphase(i,1) = vessel.motionRAO.phase{i}(freqno,headno,velno);
end

%% Curve for Ca and Cd of the object
% Za=rand(7,8); % CFD matrix for Ca, function of period and amplitude
% Zd=rand(7,8); % CFD matrix for Cd, function of period and amplitude
% p=3;
% q=3;
% aaCa=fitting(p,q,Za); % Curve fitting of Ca
% aaCd=fitting(p,q,Zd); % Curve fitting of Cd
Ca=1; % Added mass coefficient
Cd=1.2; % Damping coefficient
%% Set parameters
% Transfer nested variables
xxsimSetParameters({'g','Lpp','T',...
    'rho','m','GML','GMT','Cb','Ca','Cd','MRB','DsubL',...
    'Ma','C','forceRAOamp','forceRAOphase',...
    'forceRAOw','motionRAOamp','motionRAOphase',...
    'motionRAOw'},{vessel.main.g,vessel.main.Lpp,vessel.main.T,...
    vessel.main.rho,vessel.main.m,vessel.main.GM_L,...
    vessel.main.GM_T,vessel.main.C_B,Ca,Cd,vessel.MRB,...
    vessel.B(:, :, freqno, velno), vessel.A(:, :, freqno, velno), ...
    vessel.C(:, :, freqno, velno), forceRAOamp, forceRAOphase, ...
    vessel.forceRAO.w(1, freqno), motionRAOamp, motionRAOphase, ...
    vessel.motionRAO.w(1, freqno)});
%% Run the model
xxsimRun();

```

● DNV checking for cables

```
% Matlab script for calc. cable responses for crane operations
% Input variables
g = 9.81; % Acceleration of gravity (m/s^2)
rho = 1025; % density of water, (kg/m^3)
L = 100; % Length of cable (m)
D = 0.05; % Diameter of cable (m)
E = 2.1E11; % Youngs modulus (N/m^2)
A_c = pi*(D^2)/4; % Cross-section area (m^2)
A_eff = 0.6*A_c; % Effective area
%
% Mass of cable
m = 10.2; % Mass of cable (kg/m)
w = m*g-rho*g*A_eff; % weight of cable (in water) (N/m)
Cd_c = 1.0; % Drag coeff for cable
%
% Mass of load
M = 20000; % Mass of load (kg)
V=10;
W = M*g-rho*V*g; % Weight of load (N)
A_W =0; % Projected area of load (m^2)
Cd_W = 1.2; % Drag coeff for load
% Current velocity
U = 4; % Current velocity (m/s)
%
% -----
% Drag force on lifted object
Fd_load = 0.5*rho*Cd_W*A_W*U^2;
%
% Drag force per unit length of cable
q = 0.5* rho*Cd_c*D*U^2;
%
% Weight of Load/weight of cable
K = W / (w*L);
%
% Drag of Load/weight of cable
lambda = Fd_load / (w*L);
% -----
% Determine the shape of the cable
dz = 0.2; % Element length
z=0:dz:L;
% displ - function to calculate
% static displacement due to current
zeta = displ(-z, q, w, K, lambda, L);
```

Appendix C **20-sim models (digital)**

Appendix D **Video footage of simulation (digital)**

Appendix E **20-sim scripting library**