

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL

**MANEUVERING OF HIGH SPEED DISPLACEMENT VESSELS
IN REGULAR WAVES**



Ph.D. THESIS

Deniz ÖZTÜRK SARIGÜL

Department of Shipbuilding and Ocean Engineering

Shipbuilding and Ocean Engineering Programme

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To my spouse and family,



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Deniz ÖZTÜRK SARIGÜL
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TABLE OF CONTENTS

	<u>Page</u>
FOREWORD	ix
TABLE OF CONTENTS	xi
ABBREVIATIONS	xv
SYMBOLS	xvii
LIST OF TABLES	xxiii
LIST OF FIGURES	xxvii
SUMMARY	xxxv
ÖZET	xxxvii
1. INTRODUCTION	1
1.1 Background and Context	1
1.1.1 Experimental methods	2
1.1.2 Two-time scale models	5
1.1.3 Hybrid approach	6
1.1.4 Computational methods	10
1.1.5 Wave drift loads	11
1.2 Research Gap	12
1.3 Research Questions and Problem Statement	13
1.4 Objectives and Purpose	14
1.5 Significance and Relevance	14
1.6 Scope and Limitations	16
1.7 Methodology	17
1.8 Structure of Thesis	18
2. MEAN WAVE DRIFT FORCE PREDICTION	21
2.1 Comparison of Mean Wave Drift Loads	27
3. EXPERIMENTAL STUDY	33
3.1 Ata Nutku Ship Model Testing Laboratory	33
3.1.1 Towing tank facility	33
3.1.2 Planar motion mechanism	34
3.1.3 Wave generator	36
3.1.4 Experimental coordinate system	41
3.2 ONRT Model	43
3.3 Model Ballasting and Mounting	45
3.4 Test Conditions	48
3.5 Measurement Systems and Calibration Procedures	48
3.5.1 Loadcell	49
3.5.2 Wave height gauge and probes	50
3.6 Data Acquisition Procedures	52
3.6.1 Data acquisition of forces and moment	52
3.6.2 Data acquisition of wave parameters	54
3.7 Data Reduction Procedures	55
3.7.1 Data reduction of forces and moment	56

3.7.2 Data reduction of wave parameters	59
3.8 Experimental Results	60
3.8.1 Mean wave drift loads	60
3.8.2 Wave parameters	65
4. COMPUTATIONAL STUDY	69
4.1 Overview of Equations and Methods	69
4.1.1 Governing equations	69
4.1.2 Turbulence models	70
4.2 Numerical Model	72
4.2.1 Computational domain	73
4.2.2 Mesh resolution	75
4.2.3 Boundary conditions	79
4.2.4 Computational coordinate system	79
4.2.5 Wave generation	81
4.2.6 Kinematics of motion	82
4.2.7 Solution strategy and high-performance computing	83
4.2.8 Determination of the time step	85
4.3 Numerical Results and Discussions	86
4.3.1 Verification	86
4.3.2 Validation	90
4.3.3 Results of mean wave drift loads at different heading angles	91
4.4 Direct Numerical Simulation of Turning Trajectory in Calm Water	93
5. SYSTEM-BASED INVESTIGATION	101
5.1 Basic Assumptions	102
5.2 Coordinate System in System-Based Model	103
5.3 Non-Dimensionalization	104
5.4 Derivation of Equation of Motion with Six Degrees of Freedom	105
5.5 Derivation of Equation of Motion with Four Degrees of Freedom	110
5.6 Explanation of External Forces and Moments	116
5.6.1 Hydrodynamics forces and moments acting on ship hull	117
5.6.2 Forces and moments due to propeller	118
5.6.3 Forces and moments due to rudder	119
5.6.4 Forces and moments due to wave	121
5.7 Turning Circle Simulations	126
5.7.1 Parameters used in the simulations	127
5.7.2 Straight-ahead condition in calm water	132
5.7.3 Turning circle results in calm water	134
5.7.4 Turning circle results in waves	145
6. CONCLUSIONS AND RECOMMENDATIONS	165
6.1 General Remarks	166
6.2 Conclusions	167
6.3 Recommendations	174
REFERENCES	177
APPENDICES	185
APPENDIX A	187
APPENDIX B	193
APPENDIX C	195
APPENDIX D	225
APPENDIX E	229
APPENDIX F	235

APPENDIX G	245
APPENDIX H	255
APPENDIX I.....	263
CURRICULUM VITAE.....	285





ABBREVIATIONS

3D	: Three Dimensional
API	: American Petroleum Institute
Ad	: Advance
AVT	: Applied Vehicle Technology
BNC	: Bayonet Neill–Concelman
CAD	: Computer-Aided Design
CFD	: Computational Fluid Dynamics
CFL	: Courant-Friedrichs-Lewy
CNC	: Computer Numerical Control
DFBI	: Dynamic Fluid Body Interaction
DFT	: Discrete Fourier Transform
DNV GL	: Det Norske Veritas and Germanischer Lloyd
DoF	: Degrees of Freedom
DTC	: Duisburg Test Case
EEDI	: Energy Efficiency Design Index
EFD	: Experimental Fluid Dynamics
FFT	: Fast Fourier Transform
FIR	: Finite Impulse Response
Fr	: Froude number
FVM	: Finite Volume Method
GCI	: Grid Convergence Index
HMI	: Human Machine Interface
HPU	: Hydraulic Power Unit
HRIC	: High-Resolution Interface-Capturing
IIHR	: Iowa Institute of Hydraulic Research
IMO	: International Maritime Organization
ITTC	: International Towing Tank Conference
ITU	: Istanbul Technical University
JBC	: Japan Bulk Carrier
JTTC	: Japanese Towing Tank Conference

KCS	: KRISO Container Ship
MMG	: Maneuvering Modeling Group
MRF	: Moving Reference Frame
N/A	: Not Available
ONRT	: Office of Naval Research Tumblehome
PMM	: Planar Motion Mechanism
RANS	: Reynolds-averaged Navier-Stokes
RBM	: Rigid Body Motion
RE	: Richardson Extrapolation
RMS	: Root Mean Square
RPM	: Revolutions Per Minute
SIMPLE	: Semi-Implicit Method for Pressure-Linked Equations
Stb	: Starboard
STD	: Steady Turning Diameter
STS	: Steady Turning Speed
SYR	: Steady Yaw Rate
TD	: Tactical Diameter
Tr	: Transfer
UPS	: Uninterruptible Power Supply
V&V	: Verification and Validation
VDC	: Volts of Direct Current
VOF	: Volume of Fluid
WL	: Waterline
WS	: Workstation

SYMBOLS

a	: Wave amplitude
a_{24}, a_{26}, a_{46}	: Coupling coefficients
Acc	: Acceleration matrix
a_H	: Rudder force increasing factor
A_R	: Profile area of the rudder
B	: Ship breadth
$\overline{BM}$	: Radius of metacenter
B_{WL}	: Maximum beam of waterline
C	: Coriolis matrix
C_B	: Block coefficient
C_{lin}	: Linear Coriolis matrix
C_{WLft}	: Waterplane area coefficient
c_{YP}	: Hydrodynamic derivative of the lateral force on the propeller
$C_{\varepsilon 1}, C_{\varepsilon 2}$	: Model coefficients at the transport equation for the turbulent dissipation rate
d	: Ship draft
d_i	: Distance between weights in inclining test
d_w	: Water depth
D	: Characteristic dimension
D_P	: Propeller diameter
D_{ref}	: Diameter of the refinement volume
dx	: Distance between two adjacent grid points
E	: Percent error
e_a	: Approximate relative error
E_{abs}	: Absolute error
e_{ext}	: Extrapolated relative error
f	: Wave frequency
F	: Force
f_2	: Damping function at the transport equation for the turbulent dissipation rate

F_N	: Rudder normal force
F_x, F_y	: Longitudinal force and lateral force at midship, respectively
f_α	: Rudder normal force gradient coefficient
g	: Gravitational acceleration
G	: Center of gravity of the ship
$GG1$	: Difference between the experimental $\overline{KG}$ and the original $\overline{KG}$
$\overline{GM}$	: Metacentric height
H	: Wave height
$h, h_1, h_2, h_{1A}, h_{1B}$	: Coefficients to calculate wave drift loads at empirical approach
H_{pp}	: Peak-to-peak value of wave height
H_R	: Rudder span
H_{ref}	: Height of the refinement volume
H_{rms}	: Root mean squared value of the wave height
H_S	: Significant wave height
I	: Moment of inertia
I_{xG}, I_{yG}, I_{zG}	: Components of the mass moment of inertia with respect to the center of gravity
I_{xx}, I_{yy}, I_{zz}	: Diagonal components of moment of inertia
I_{xy}, I_{xz}, I_{yz}	: Cross-inertial terms
J	: Advance ratio
J_{xx}, J_{zz}	: Added moment of inertia for roll and yaw, respectively
k_w	: Wave number
k	: Turbulent kinetic energy
K, M, N	: External angular moments acting on the ship in roll, φ , pitch, θ , and yaw, ψ , directions, respectively
k_0, k_1, k_2	: Coefficients representing the characteristics of K_T
$\overline{KB}$	: Center of buoyancy
$\overline{KG}$	: Vertical center of gravity
K_H, K_R, K_P, K_W	: External roll moments due to hull, rudder, propeller, and wave, respectively
K_T	: Thrust coefficient
K_{xx}, K_{yy}, K_{zz}	: Radius of gyration
L	: Length
l	: Length component at non-dimensionalization

l_R	: Correlation factors for the flow-straightening effect due to yaw rate
LCB	: Longitudinal center of buoyancy
L_{os}	: Longitudinal distance between the fore most and aft most point under water
l_P	: Correction factor lateral velocity due to yaw rate
L_{PP}	: Length between perpendiculars
L_{ref}	: Length of the refinement volume
L_{WL}	: Length on waterline
M	: Moment
m	: Mass
$\tilde{M}$	: Adjugate matrix of matrix M
$ M $	: Determinant of matrix M
M_x, M_z	: Roll and yaw moments at midship, respectively
m_x, m_y	: Added masses for surge and sway, respectively
n	: Propeller rotation rate
N_H, N_R, N_P, N_W	: External yaw moments due to hull, rudder, propeller, and wave, respectively
N_i	: Discretization level
P	: Force and moment matrix
p, q, r	: Angular velocity components of roll, pitch, and yaw motions
$\bar{p}$	: Mean pressure
$\dot{p}, \dot{q}, \dot{r}$	: Acceleration components of the roll, pitch, and yaw, respectively
P_k, P_ϵ	: Production terms at the transport equations
p	: Apparent order
r	: Yaw rate
r_{21}, r_{32}	: Refinement factors
R	: Convergence condition
r_{PMM}	: PMM yaw rate
$\dot{r}_{PMM}$	: PMM yaw acceleration
$rsquare$	: R-squared coefficient of determination
S	: Wetted surface area
S_k, S_ϵ	: User-specified source terms at the transport equations
$ssreg$	: Regression sum of squares
$ssresid$	: Residual sum of squares

T	: Propeller thrust
T_0	: Specific time-scale
T_{90}	: Time when the heading angle is 90°
T_{180}	: Time when the heading angle is 180°
T_P	: Peak period of wave
t_p	: Thrust deduction factor
t_R	: Steering resistance deduction factor
T'_{surge}	: Nondimensional surge parameter
T'_{sway}	: Nondimensional sway parameter
T_Z	: Average zero up-crossing period of the wave
U	: Ship velocity
U_a	: Advance speed
u, v, w	: Velocity components of surge, sway, and heave motions at midship, respectively
$\bar{u}$	: Mean velocity
U_c	: Carriage speed
$\dot{u}, \dot{v}, \dot{w}$	: Acceleration components of the surge, sway, and heave, respectively
u_G, v_G	: Surge and lateral velocity at center of gravity, respectively
$\bar{u}_i, \bar{u}_j$	: Mean velocity vectors in the i^{th} and j^{th} direction of the cartesian coordinate
$u'_i u'_j$	: Reynolds stress tensor
U_R	: Resultant inflow velocity at the rudder
u_R, v_R	: Longitudinal and lateral inflow velocities to the rudder, respectively
V_a	: Actual value
V_c	: Computational value
V_e	: Experimental value
Vel	: Velocity matrix
$\vec{V}_G$	: Relative velocity vector with respect to the center of gravity
v_{PMM}	: PMM sway velocity
$\dot{v}_{PMM}$	: PMM sway acceleration
v_{RP}	: Lateral rudder inflow velocity induced by propeller rotation
W	: Width
w_P	: Wake fraction during maneuvering

w_{p0}	: Effective wake fraction
W_{ref}	: Width of the refinement volume
x, y, z	: Directions of surge, sway, and heave motions, respectively
X, Y, Z	: External forces acting on the ship along the x (surge), y (sway), and z (heave) directions, respectively
x_f	: Forcing length
x_G, y_G, z_G	: Longitudinal, lateral, and vertical positions of the center of gravity relative to the midship, relatively
x_H	: Longitudinal coordinate of additional lateral force
X_H, X_R, X_P, X_W	: External surge forces due to hull, rudder, propeller, and wave, respectively
x_{Los}	: Longitudinal position of $L_{os}/2$ to the point of $L_{pp}/2$
x_P	: Longitudinal coordinate of the propeller disk
x_R	: Longitudinal coordinate of rudder shaft
y^+	: Non-dimensional distance
Y_H, Y_R, Y_P, Y_W	: External sway forces due to hull, rudder, propeller, and wave, respectively
y_{max}	: Sway motion amplitude
y_{PMM}	: PMM-fixed lateral axis
z_H	: z -coordinate of the acting point of the hydrodynamic forces
z_{HR}	: Vertical position of the center of effective rudder force
z_P	: Vertical coordinate of the propeller disk
α_{bow}	: Angle between the vessel x -axis and a line drawn from the foremost point in the water line to the point at $y = B/4$ (ahead of $L_{pp}/2$) on the water line
α_R	: Effective inflow angle to the rudder
α_Z	: Vertical acting point of the lateral added mass component
β	: Drift angle
β_P	: Inflow angle to the propeller
γ_R	: Flow-straightening coefficient in the rudder module
Δ	: Displacement
δ	: Rudder angle
Δt	: Time step
Δw	: Weight to be shifted vertically at model ballasting
$\Delta x, \Delta y, \Delta z$	: Refinement ratios of free surface
ε	: Turbulent dissipation rate

$\varepsilon_{21}, \varepsilon_{32}$	: Discretization absolute error
ε_0	: Ambient turbulence value in the source terms
ε_w	: Ratio of the wake fraction
η	: Ratio of the propeller diameter to the rudder span
$\eta(t)$	: Wave profile
κ	: Experimental constant in the rudder module
λ	: Wavelength
Λ	: Scale
Λ_R	: Rudder aspect ratio
μ	: Dynamic viscosity
ν	: Kinematic viscosity
π	: Pi number
ρ	: Fluid density
$\sigma_k, \sigma_\varepsilon$	: Model coefficients at the transport equation for the kinetic energy
Φ	: Velocity potential
ϕ	: Wave phase
φ, θ, ψ	: Directions of roll, pitch, and yaw motions
φ_{ext}	: Extrapolated values
φ_i	: Integral parameter at verification procedure
φ_k	: Solution scalar on the k^{th} refinement
χ	: Incident wave angle
ψ	: Heading angle (or yaw angle)
ψ_{max}	: Yaw motion amplitude
ω	: Wave frequency
Ω	: Angular velocity vector
ω_{PMM}	: PMM frequency

LIST OF TABLES

	<u>Page</u>
Table 2.1 : Comparison of longitudinal wave drift forces, $\bar{X}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020, \lambda/LWL = 1.00$..	28
Table 2.2 : Comparison of lateral wave drift forces, $\bar{Y}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020, \lambda/LWL = 1.00$..	29
Table 2.3 : Comparison of yaw moments, $\bar{N}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020, \lambda/LWL = 1.00$.	30
Table 3.1 : PMM design parameters (Özden et al., 2021).	35
Table 3.2 : Main particulars of ONRT hull.	44
Table 3.3 : The hydrostatic values of ONRT hull.	45
Table 3.4 : The wave characteristics in static drift tests.	48
Table 3.5 : PMM test conditions.	48
Table 3.6 : Wave drift loads of steady drift tests with various headings at $H/\lambda = 0.020, \lambda/LWL = 1.00$.	63
Table 3.7 : Experimental wave parameters for $H/\lambda = 0.020, \lambda/LWL = 1.00$.	66
Table 3.8 : Experimental wave parameters for $H/\lambda = 0.020, \lambda/LWL = 0.50$.	66
Table 3.9 : Experimental wave parameters for $H/\lambda = 0.025, \lambda/LWL = 1.00$.	67
Table 4.1 : Refinement volumes for applied grid.	75
Table 4.2 : Refinement ratios in Δz of free surface.	76
Table 4.3 : Element counts in the simulation.	79
Table 4.4 : Features of high-performance computing.	83
Table 4.5 : Spatial convergence for surge force obtained from steady drift simulations at $\psi = 0^\circ$.	89
Table 4.6 : Temporal convergence for surge force obtained from steady drift simulations at $\psi = 0^\circ$.	90
Table 4.7 : Results of the validation study.	91
Table 4.8 : Wave drift loads of steady drift simulations with various heading angles ($H/\lambda = 0.020, \lambda/LWL = 1.00$).	93
Table 4.9 : Refinement volumes for the applied grid at direct numerical analysis.	97
Table 4.10 : Element counts at direct numerical simulation.	98
Table 5.1 : Motion, velocity, and force/moment components in the x , y , and z directions.	104
Table 5.2 : Statistics for polynomial fits.	123
Table 5.3 : Turning circle maneuvering simulation matrix.	127
Table 5.4 : Main parameters related to the ONRT model.	128
Table 5.5 : Non-dimensional added mass and inertia terms.	128
Table 5.6 : Dimensional added mass and inertia terms.	129
Table 5.7 : Hydrodynamic coefficients used in the simulations (Guo et al., 2018).	129
Table 5.8 : Parameters in the propeller module.	130
Table 5.9 : Parameters in the rudder module.	131

Table 5.10 : Comparison of trajectory parameters in calm water, $\delta = 35^\circ$ and $\psi = 0^\circ$	139
Table 5.11 : Comparison of trajectory parameters in calm water, $\delta = -35^\circ$ and $\psi = 0^\circ$	139
Table 5.12 : Comparison of trajectory parameters at different heading angles in calm water, $\delta = \pm 15^\circ$	143
Table 5.13 : Comparison of trajectory parameters at different heading angles in calm water, $\delta = \pm 25^\circ$	143
Table 5.14 : Comparison of trajectory parameters at different heading angles in calm water, $\delta = \pm 35^\circ$	143
Table 5.15 : Comparison of trajectory parameters in regular waves, $\delta = 35^\circ$ and $\chi = 180^\circ$	146
Table 5.16 : Comparison of trajectory parameters in regular waves, $\delta = -35^\circ$ and $\chi = 180^\circ$	151
Table A.1 : Comparison of longitudinal wave drift forces, $\bar{X}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$	187
Table A.2 : Comparison of lateral wave drift forces, $\bar{Y}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$	188
Table A.3 : Comparison of yaw moments, $\bar{N}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$	189
Table A.4 : Comparison of longitudinal wave drift forces, $\bar{X}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$	190
Table A.5 : Comparison of lateral wave drift forces, $\bar{Y}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$	191
Table A.6 : Comparison of yaw moments, $\bar{N}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$	192
Table D.1 : Wave drift loads of steady drift tests with various heading angles at $H/\lambda = 0.020, \lambda/LWL = 0.50$	226
Table D.2 : Wave drift loads of steady drift tests with various heading angles at $H/\lambda = 0.025, \lambda/LWL = 1.00$	228
Table G.1 : Comparison of trajectory parameters in calm water, $\delta = \pm 35^\circ$ and $\psi = 90^\circ$	247
Table G.2 : Comparison of trajectory parameters in calm water, $\delta = \pm 35^\circ$ and $\psi = 180^\circ$	250
Table G.3 : Comparison of trajectory parameters in calm water, $\delta = \pm 35^\circ$ and $\psi = -90^\circ$	253
Table I.1 : Comparison of trajectory parameters in regular waves, $\delta = 35^\circ$ and $\chi = 90^\circ$	263
Table I.2 : Comparison of trajectory parameters in regular waves, $\delta = -35^\circ$ and $\chi = 90^\circ$	263
Table I.3 : Comparison of trajectory parameters in regular waves, $\delta = 35^\circ$ and $\chi = 0^\circ$	270
Table I.4 : Comparison of trajectory parameters in regular waves, $\delta = -35^\circ$ and $\chi = 0^\circ$	270
Table I.5 : Comparison of trajectory parameters in regular waves, $\delta = 35^\circ$ and $\chi = -90^\circ$	277

Table I.6 : Comparison of trajectory parameters in regular waves, $\delta = -35^\circ$ and $\chi = -90^\circ$ 277





LIST OF FIGURES

	<u>Page</u>
Figure 2.1 : Comparison of empirical/analytical results of mean longitudinal force with respect to heading angle ($H/\lambda = 0.020, \lambda/LWL = 1.00$).....	28
Figure 2.2 : Comparison of empirical/analytical results of mean lateral force with respect to heading angle ($H/\lambda = 0.020, \lambda/LWL = 1.00$).....	29
Figure 2.3 : Comparison of empirical/analytical results of mean yaw moment with respect to heading angle ($H/\lambda = 0.020, \lambda/LWL = 1.00$).....	30
Figure 2.4 : Comparison of different wave conditions' effects on wave drift loads (DNV AS (2021)): (a) longitudinal wave drift force, (b) lateral wave drift force, (c) yaw moment.....	32
Figure 3.1 : The sketch of the Ata Nutku ship model testing laboratory: (a) towing tank, (b) wave generator, (c) stability tank, (d) cavitation tunnel, (e) circulation channel, (f) storage, (g) paint workshop, (h) CNC workshop.	33
Figure 3.2 : Horizontal planar motion mechanism at Istanbul Technical University.	35
Figure 3.3 : The units of wave generator: (a) electrical control panel, (b) flap unit, (c) hydraulic unit, (d) wave probes and gauges. (1) computer, (2) Akamina – WaveGen Studio, (3) Tiab – Play waveform, (4) UPS, (5) coolant pump, (6) centrifuge pump.	38
Figure 3.4 : Interface of Rexroth – OPT-Computation.	39
Figure 3.5 : Interface of Tiab – Play Wave Form.	40
Figure 3.6 : Interface of Rota Teknik – HPU Monitor.....	40
Figure 3.7 : Interface of AKAMINA – WaveGen Studio.	41
Figure 3.8 : The earth-fixed and ship-fixed coordinate system.	42
Figure 3.9 : Incident wave angle, χ	43
Figure 3.10 : 1/48.9 scaled ONRT model.	44
Figure 3.11 : Inclinator.	47
Figure 3.12 : Connection of loadcell and pantograph.	47
Figure 3.13 : Technical drawing of the load cell.	49
Figure 3.14 : Amplifier and calibration boxes.	50
Figure 3.15 : Wave measurement units: (a) Wave height gauge, (b) Wave height gauge, and a wave probe head.	51
Figure 3.16 : Electrical equipment installation: (a) Calibration boxes, (b) Separation of the force and moment components, (c) Control panel. .	52
Figure 3.17 : (a) Human Machine Interface, (b) The static drift test interface at HMI, (c) The static drift test interface at SIMATIC WinCC Explorer.	53
Figure 3.18 : The steps of obtaining the wave drift load: (a) The force signal, (b) The processed signal using the FFT algorithm, (c) The obtaining wave drift load by using Savitzky-Golay filtering.	58

Figure 3.19 : Comparison of longitudinal wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020, \lambda/LWL = 1.00$	61
Figure 3.20 : Comparison of lateral wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020, \lambda/LWL = 1.00$	62
Figure 3.21 : Comparison of yaw moments of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020, \lambda/LWL = 1.00$	62
Figure 3.22 : Comparison of longitudinal wave drift forces of steady drift tests.	64
Figure 3.23 : Comparison of lateral wave drift forces of steady drift tests.	64
Figure 3.24 : Comparison of yaw moments of steady drift test.	65
Figure 4.1 : The fully appended ONRT geometry. (a) Top view, (b) Side view, (c) Back view, (d) Perspective view.	72
Figure 4.2 : The dimensions of the computational domain from XZ and YZ planes.	73
Figure 4.3 : Refinement volumes and interfaces (a) Perspective view, (b) Side view, (c) Back view.	74
Figure 4.4 : The spatial discretization around the hull (a) Side view, (b) Top view, (c) Front view.	75
Figure 4.5 : Free surface refinement.	76
Figure 4.6 : Overset cell status.	77
Figure 4.7 : Detailed view of the surface mesh over the hull.	77
Figure 4.8 : Boundary layer discretization.	78
Figure 4.9 : Wall y^+ distribution on the hull.	78
Figure 4.10 : Boundary conditions.	80
Figure 4.11 : Earth-fixed and ship-fixed coordinate systems in numerical simulations.	80
Figure 4.12 : Solution procedure.	84
Figure 4.13 : Comparison of longitudinal wave drift forces at $H/\lambda = 0.020, \lambda/LWL = 1.00$	92
Figure 4.14 : Comparison of lateral wave drift forces at $H/\lambda = 0.020, \lambda/LWL = 1.00$	92
Figure 4.15 : Comparison of yaw moment at $H/\lambda = 0.020, \lambda/LWL = 1.00$	93
Figure 4.16 : Flow chart of direct numerical analysis, indicating decision points.	95
Figure 4.17 : Representation of the direct numerical analysis.	96
Figure 4.18 : The dimensions of the computational domain and boundary conditions in direct numerical analysis from XZ and YZ planes.	97
Figure 4.19 : Refinement volumes and interfaces at direct numerical analysis.	98
Figure 5.1 : Polynomial models adapted to wave drift load data for the $H/\lambda = 0.020, \lambda/LWL = 1.00$ condition.	124
Figure 5.2 : Definitions of turning circle (ITTC, 2017a).	126
Figure 5.3 : Trajectory and time-dependent parameters during the straight-ahead condition in 4-DoF S-B simulation.	133
Figure 5.4 : Comparison of turning trajectories and time histories in calm water, $\delta = 35^\circ$ and $\psi = 0^\circ$	140
Figure 5.5 : Comparison of turning trajectories and time histories in calm water, $\delta = -35^\circ$ and $\psi = 0^\circ$	141
Figure 5.6 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = 15^\circ, 25^\circ, 35^\circ$ and $\psi = 0^\circ$	144
Figure 5.7 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 1.00$).	148
Figure 5.8 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 0.50$).	149

Figure 5.9 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.025$, $\lambda/LWL = 1.00$).	150
Figure 5.10 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.020$, $\lambda/LWL = 1.00$).	153
Figure 5.11 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.020$, $\lambda/LWL = 0.50$).	154
Figure 5.12 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.025$, $\lambda/LWL = 1.00$).	155
Figure 5.13 : Comparison of maneuvering parameters in all wave conditions, $\delta = \pm 35^\circ$ and $\chi = 180^\circ$	157
Figure 5.14 : Comparison of maneuvering parameters in all wave conditions, $\delta = \pm 35^\circ$ and $\chi = 90^\circ$	160
Figure 5.15 : Comparison of maneuvering parameters in all wave conditions, $\delta = \pm 35^\circ$ and $\chi = 0^\circ$	161
Figure 5.16 : Comparison of maneuvering parameters in all wave conditions, $\delta = \pm 35^\circ$ and $\chi = -90^\circ$	163
Figure A.1 : Comparison of empirical/analytical results of mean longitudinal force with respect to heading angle ($H/\lambda = 0.020$, $\lambda/LWL = 0.50$).	187
Figure A.2 : Comparison of empirical/analytical results of mean lateral force with respect to heading angle ($H/\lambda = 0.020$, $\lambda/LWL = 0.50$).	188
Figure A.3 : Comparison of empirical/analytical results of mean yaw moment with respect to heading angle ($H/\lambda = 0.020$, $\lambda/LWL = 0.50$).	189
Figure A.4 : Comparison of empirical/analytical results of mean longitudinal force with respect to heading angle ($H/\lambda = 0.025$, $\lambda/LWL = 1.00$).	190
Figure A.5 : Comparison of empirical/analytical results of mean lateral force with respect to heading angle ($H/\lambda = 0.025$, $\lambda/LWL = 1.00$).	191
Figure A.6 : Comparison of empirical/analytical results of mean yaw moment with respect to heading angle ($H/\lambda = 0.025$, $\lambda/LWL = 1.00$).	192
Figure C.1 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$	195
Figure C.2 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$	196
Figure C.3 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$	197
Figure C.4 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$	198
Figure C.5 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$	199
Figure C.6 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$	200

Figure C.7 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$	201
Figure C.8 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$	202
Figure C.9 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$	203
Figure C.10 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$	204
Figure C.12 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$	205
Figure C.13 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$	206
Figure C.14 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$	207
Figure C.15 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$	208
Figure C.16 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$	209
Figure C.17 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$	210
Figure C.18 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$	211
Figure C.19 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$	212
Figure C.20 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$	213
Figure C.21 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 0.50$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$	214
Figure C.23 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$	215
Figure C.24 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$	216
Figure C.25 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$	217

Figure C.26 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$.	218
Figure C.27 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.	219
Figure C.28 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.	220
Figure C.29 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$.	221
Figure C.30 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$.	222
Figure C.31 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$.	223
Figure C.32 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/LWL = 1.00$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$.	224
Figure D.1 : Comparison of longitudinal wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020$, $\lambda/LWL = 0.50$.	225
Figure D.2 : Comparison of lateral wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020$, $\lambda/LWL = 0.50$.	225
Figure D.3 : Comparison of yaw moments of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020$, $\lambda/LWL = 0.50$.	226
Figure D.4 : Comparison of longitudinal wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.025$, $\lambda/LWL = 1.00$.	227
Figure D.5 : Comparison of lateral wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.025$, $\lambda/LWL = 1.00$.	227
Figure D.6 : Comparison of yaw moments of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.025$, $\lambda/LWL = 1.00$.	228
Figure E.1 : Time histories of wave signals at $\psi = 0^\circ$ to 120° ($H/\lambda = 0.020$ and $\lambda/LWL = 1.00$).	229
Figure E.2 : Time histories of wave signals at $\psi = 120^\circ$ to 180° ($H/\lambda = 0.020$ and $\lambda/LWL = 1.00$).	230
Figure E.3 : Time histories of wave signals at $\psi = 0^\circ$ to 120° ($H/\lambda = 0.020$ and $\lambda/LWL = 0.50$).	231
Figure E.4 : Time histories of wave signals at $\psi = 120^\circ$ to 180° ($H/\lambda = 0.020$ and $\lambda/LWL = 0.50$).	232
Figure E.5 : Time histories of wave signals at $\psi = 0^\circ$ to 120° ($H/\lambda = 0.025$ and $\lambda/LWL = 1.00$).	233
Figure E.6 : Time histories of wave signals at $\psi = 120^\circ$ to 180° ($H/\lambda = 0.025$ and $\lambda/LWL = 1.00$).	234
Figure F.1 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$.	235

Figure F.2 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$	236
Figure F.3 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$	237
Figure F.4 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$	238
Figure F.5 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$	239
Figure F.6 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$	240
Figure F.7 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 135^\circ$, (b) $\psi = 150^\circ$	241
Figure F.8 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 135^\circ$, (b) $\psi = 150^\circ$	242
Figure F.9 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 165^\circ$, (b) $\psi = 180^\circ$	243
Figure F.10 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/LWL = 1.00$: (a) $\psi = 165^\circ$, (b) $\psi = 180^\circ$	244
Figure G.1 : Comparison of turning trajectories and time histories in calm water, $\delta = 35^\circ$ and $\psi = 90^\circ$	245
Figure G.2 : Comparison of turning trajectories and time histories in calm water, $\delta = -35^\circ$ and $\psi = 90^\circ$	246
Figure G.3 : Comparison of turning trajectories and time histories in calm water, $\delta = 35^\circ$ and $\psi = 180^\circ$	248
Figure G.4 : Comparison of turning trajectories and time histories in calm water, $\delta = -35^\circ$ and $\psi = 180^\circ$	249
Figure G.5 : Comparison of turning trajectories and time histories in calm water, $\delta = 35^\circ$ and $\psi = -90^\circ$	251
Figure G.6 : Comparison of turning trajectories and time histories in calm water, $\delta = -35^\circ$ and $\psi = -90^\circ$	252
Figure H.1 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = -15^\circ, -25^\circ, -35^\circ$ and $\psi = 0^\circ$	255
Figure H.2 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = 15^\circ, 25^\circ, 35^\circ$ and $\psi = 90^\circ$	256
Figure H.3 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = -15^\circ, -25^\circ, -35^\circ$ and $\psi = 90^\circ$	257
Figure H.4 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = 15^\circ, 25^\circ, 35^\circ$ and $\psi = 180^\circ$	258
Figure H.5 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = -15^\circ, -25^\circ, -35^\circ$ and $\psi = 180^\circ$	259
Figure H.6 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = 15^\circ, 25^\circ, 35^\circ$ and $\psi = -90^\circ$	260

Figure H.7 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = -15^\circ, -25^\circ, -35^\circ$ and $\psi = -90^\circ$...	261
Figure I.1 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 1.00$).....	264
Figure I.2 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 0.50$).....	265
Figure I.3 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.025, \lambda/LWL = 1.00$).....	266
Figure I.4 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 1.00$).....	267
Figure I.5 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 0.50$).....	268
Figure I.6 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.025, \lambda/LWL = 1.00$).....	269
Figure I.7 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 1.00$).....	271
Figure I.8 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 0.50$).....	272
Figure I.9 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.025, \lambda/LWL = 1.00$).....	273
Figure I.10 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 1.00$).....	274
Figure I.11 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 0.50$).....	275
Figure I.12 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.025, \lambda/LWL = 1.00$).....	276
Figure I.13 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 1.00$).....	278
Figure I.14 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 0.50$).....	279
Figure I.15 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.025, \lambda/LWL = 1.00$).....	280
Figure I.16 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 1.00$).....	281
Figure I.17 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.020, \lambda/LWL = 0.50$).....	282
Figure I.18 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.025, \lambda/LWL = 1.00$).....	283



MANEUVERING OF HIGH SPEED DISPLACEMENT VESSELS IN REGULAR WAVES

SUMMARY

This doctoral thesis presents a comprehensive exploration of maneuvering performance in waves, aiming to gain insight into the interactive behavior of waves and vessels and use this interaction to develop a mathematical model of maneuvering in waves. The research focuses on the adaptation of experimental methods, practical system-based methods, and computational analysis techniques to enhance the integration of solution methods.

The central objective of this thesis is to systematically reveal the complicated characteristics of mean wave drift loads. Additionally, the thesis aims to demonstrate the influence of these loads on the assessment of maneuvering performance. The research seeks to bridge the gap between traditional investigation of maneuvering in calm water and growing demand for determining the minimum propulsion power needed to maintain a ship's maneuverability in adverse conditions.

The study employs a systematic approach, combining various research techniques. It utilizes a system-based mathematical method along with laboratory experiments and simulation-based analyses to achieve its objectives. The methodology encompasses theoretical modeling, computational simulations, and practical measurements.

The research reveals that integrating mean wave drift loads, namely surge wave drift force, sway wave drift force, and yaw wave drift moment, obtained with a practical experimental approach or computational methods, can significantly enhance the capacity of system-based models. The data reduction procedure enables to reach of zeroth-frequency values representing the mean wave drift loads.

This thesis contributes a novel approach to research endeavors, experimental initiatives, and software development projects aimed at elucidating the concept of "maintaining maneuverability" in challenging sea conditions. The findings underscore the importance of adapting wave conditions to maneuvering applications developed with new approaches.

The insights from this research hold wide-ranging implications for ocean engineering practitioners and researchers. The presented framework offers a holistic perspective that can guide maneuvering performance and navigational safety. The findings of this thesis will be also presented in the activity named "Assessment of Experiments and Prediction Methods for Naval Ships Maneuvering in Waves", the activity period of which is 2021-2023 and still continues. The activity is carried out within the scope of Applied Vehicle Technology (AVT) and is referred to by the code AVT-348.

In conclusion, this doctoral thesis presents a practical experimental and numerical approach to achieve surge, sway, and yaw wave drift loads in regular waves which are then used to demonstrate the effect of wave-related loads on the maneuvering

performance of surface vessels. The measured wave drift forces and moments are incorporated into the equations of motion for maneuvering to predict the effect of waves on the maneuvering performance of a vessel. A simplified mathematical model to show the behavior of the vessel in waves is represented by four nonlinear equations of motion for surge, sway, yaw, and roll. The turning circle, which is one of the standard maneuvering tests is simulated for ONR Tumblehome (ONRT) at different wave conditions in order to demonstrate the effect of wave drift forces on the trajectories and time histories of maneuvering parameters.



YÜKSEK HIZLI DEPLASMAN GEMİLERİNİN DÜZENLİ DALGALARDA MANEVRALARI

ÖZET

Gemilerin manevra performansının değerlendirilmesi, analizi ve doğrulanması genellikle sakin su koşullarında yürütülmektedir. Gerçek çevre koşullarını tamamen sakin su ile temsil etmek gerçekçi olmamakla birlikte, manevra problemlerinde dalga etkilerini temsil etmek için ampirik düzeltme faktörleri çözüm ortamına dahil edilmektedir. Aslında dalgalarda gemi performansının değerlendirilmesi, geminin farklı deniz koşullarında seyrettiği ve dümen açısının sıfır olduğu varsayımıyla, denizcilik başlığı altında incelenir. Bununla birlikte, geminin farklı deniz koşullarında manevra karakteristiklerinin ortaya çıkarılması için, manevra parametreleri gözeticilerle, dalga kaynaklı kuvvet ve momentler ile gemi arasındaki daha karmaşık olan ilişki incelenmelidir.

Uluslararası Denizcilik Örgütü (International Maritime Organization – IMO) standart manevra testlerini zorunlu hale getirmiş, ticari gemiler için sakin su koşullarında dönme dairesi, zig-zag ve durma performansları için yönergeler oluşturmuştur. IMO'nun Enerji Verimliliği Tasarım İndeksi (Energy Efficiency Design Index – EEDI) ile gemilerde enerji verimliliğinin artırılması ve gemi kaynaklı emisyonunun azaltılması hedeflenmiştir. Söz konusu indeksi sağlamak için gemi hızının düşürülmesi uygulamalı bir çözüm olsa bile, bu çözüm manevra performansını ve seyir emniyetini etkileyebilmektedir. Bu nedenle, IMO olumsuz seyir koşullarında manevra kabiliyetini kaybetmemek için gerekli minimum sevk gücünü hesaplayan bir yönerge yayınlamıştır. Benzer şekilde, güncel NATO manevra standartları dalgalı koşullarda manevra performansı ile ilişkin kriterleri kapsamaktadır. Bu yönerge ve standartların yayınlanmasıyla, farklı deniz koşullarında manevra kabiliyetini araştıran çalışmalara olan ilgi hızla artmıştır.

Bu doktora tezi, dalgalarda manevra performansının kapsamlı bir incelemesini sunarak, dalga ve gemi etkileşimi hakkında fikir edinmeyi ve bu etkileşimi matematiksel bir manevra modeli geliştirmek için kullanmayı amaçlamaktadır. Araştırma, çözüm yöntemlerinin entegrasyonunu geliştirmek için deneysel yöntemlerin ve hesaplamalı analiz tekniklerinin pratik sistem tabanlı yöntemlere uyarlanmasına odaklanmaktadır.

Bu tezin temel amacı, dalga yüklerinin karmaşık özelliklerini sistematik olarak ortaya çıkarmak ve bu yüklerin manevra performansının değerlendirilmesi üzerindeki etkisini göstermektir. Araştırma, sakin suda manevra performansına ilişkin geleneksel inceleme ile olumsuz koşullarda bir geminin manevra kabiliyetini sürdürmek için gereken minimum itme gücünü belirlemeye yönelik artan talep arasındaki boşluğu kapatmayı amaçlamaktadır.

Çalışmada, çeşitli araştırma tekniklerini birleştiren sistematik bir yaklaşım kullanılmıştır. Amaca ulaşmak için sistem tabanlı bir matematiksel yöntemin yanı sıra laboratuvar deneyleri ve simülasyon tabanlı analizler ile çalışılmıştır. Metodoloji, teorik modellemeyi, hesaplamalı simülasyonları ve pratik matematiksel yöntemleri kapsamaktadır.

Literatürde ortalama dalga yüklerini hesaplayan pratik ampirik ifadeler yer almaktadır. Araştırmada ilk olarak ampirik yöntemlerin incelenmesine yer verilmiştir. Daha sonra, deneysel ve sayısal veriler elde edildiğinde, sonuçlar ampirik yöntem sonuçları ile kıyaslanmıştır.

Araştırma, pratik bir deneysel yaklaşım ve hesaplama yöntemleriyle elde edilen ortalama dalga yüklerinin, yani boyuna öteleme dalga kuvveti, yanal öteleme dalga kuvveti ve savrulma dalga momentinin, sistem tabanlı matematiksel modele entegre edilmesinin, sistem tabanlı modellerin kapasitesini önemli ölçüde artırabileceğini ortaya koymaktadır. ONR Tumblehome (ONRT) modelinin manevra kabiliyeti, sıfır hız koşulu altında farklı dalga boyu/gemi boyu oranı ve dalga dikliğine sahip üç düzenli dalga durumu altında deneysel olarak incelenmiştir. Düzenli dalgalar " $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$ ", " $H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$ " ve " $H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$ " özelliklerine sahiptir. Modelin pruva açısı 0° ile 180° arasında 15° 'lik açılarla yatay düzlemsel hareket mekanizması (Planar Motion Mechanism – PMM) kullanılarak değiştirilmiş, üç boyutlu hareketi pantograf ile sağlanmıştır. Düzenli dalgalar halinde modele etki eden kuvvet ve momentler ölçülmüş, zamana bağlı dalga grafikleri oluşturulmuştur.

Hesaplamalı akışkanlar dinamiği analizleri, deneysel koşulları temsil edecek şekilde $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$ dalga durumu için yürütülmüştür. Burada esas amaç, sayısal analizlerin dalgalarda manevra karakteristiklerinin tahmininde deneysel yöntemler yerine kullanılabilirliğini göstermektir. Bunun için, sayısal analiz çıktıları deneysel sonuçlar ile doğrulanmıştır. Ardından sakin suda ONRT modelinin manevra karakteristikleri doğrudan simüle edilerek, dönme dairesi parametreleri sistem tabanlı modelden elde edilen sonuçlarıyla kıyaslanmıştır.

Veri işleme aşamasında temel analiz prosedürleri tercih edilerek, frekans bölgesinde sıfırıncı frekansa karşılık gelen ve ortalama dalga yüklerini temsil eden değerlere ulaşılmıştır. Tespit edilen dalga kuvvetleri ve momentleri, dalgaların bir geminin manevra performansı üzerindeki etkisini tahmin etmek için hareket denklemlerine dahil edilmiştir. Geminin dalgadaki davranışını göstermek için basitleştirilmiş bir matematiksel model, boyuna öteleme, yanal öteleme, yalpa ve savrulma hareketlerini kapsayan doğrusal olmayan dört hareket denklemi ile temsil edilmiştir. Standart manevra testlerinden biri olan dönme dairesi, farklı dalga koşullarında simüle edilmiştir. Dalga yüklerinin ilerleme, taktik çap, transfer ve dönme çapı gibi parametrelere ve zamana bağlı diğer parametrelere olan etkileri literatürde yer alan mevcut sonuçlarla karşılaştırılarak incelenmiştir.

Bu tez ile, zorlu deniz koşullarında "manevra kabiliyetini koruma" kavramını aydınlatmayı amaçlayan araştırmalara, deneysel girişimlere ve yazılım geliştirme projelerine yeni bir yaklaşım katılması hedeflenmiştir. Bulgular, yeni yaklaşımlarla geliştirilen manevra uygulamalarına dalga koşullarının uyarlanmasıyla önemini vurgulamaktadır.

Bu arařtırmadan elde edilen bilgilerin, gemi mhendislięi alanında faaliyet gsteren arařtırmacılar iin geniř kapsamlı ıkarımlara sahip olduęu dřnlmektedir. Sunulan ereve, manevra performansına ve seyir gvenlięine rehberlik edebilecek btncl bir bakıř aısı saęlayabilir. Bu tezin bulguları faaliyet dnemi 2021-2023 yılları olan ve hala devam eden Dalgalarda Manevra Yapan Askeri Gemilere Ynelik Deneylerin ve Tahmin Yntemlerinin Deęerlendirilmesi (Assessment of Experiments and Prediction Methods for Naval Ships Maneuvering in Waves) isimli faaliyette sunulacaktır. Faaliyet, Uygulamalı Ara Teknolojisi (Applied Vehicle Technology-AVT) kapsamında yrtlmekte ve AVT-348 kodu ile anılmaktadır.

Sonuç olarak, bu doktora tezi, daha sonra dalga yklerinin geminin manevra performansı zerindeki etkisini gsteren sistem tabanlı modelde kullanmak zere, dzenli dalgalarda boyuna teleme ve yanal teleme dalga kuvvetleri ile savrulma dalga momentini elde etmek iin pratik bir deneysel ve sayısal yaklařım sunmaktadır.





1. INTRODUCTION

1.1 Background and Context

Traditionally, the assessment, analysis, and validation of the maneuvering capabilities of vessels have been confined to calm water conditions. However, attaining perfectly calm water conditions in the real environment is a near-impossibility, and it leads to the use of empirical correction factors to represent wave conditions. The evaluation of ship performance in waves is the area of interest in seakeeping, and it is assumed that the ship is directly navigating at zero rudder angle. Notwithstanding, the actual maneuvering of a ship in such conditions intricately relies on the interplay between wave-induced forces, moments, and the vessel's response.

The International Maritime Organization (IMO) mandates standard maneuvering trials and provides guidelines for turning, zig-zag and stopping performances for commercial vessels (International Maritime Organization, 2002). These trials are recommended to be directed in calm waters. Similar to this, early naval maneuvering guidelines like NATO ANEP-70 (2003) were developed based on results obtained in calm waters.

It is a method to reduce power and ship speed to reduce ship emissions by complying with the regulations specified in the IMO Energy Efficiency Design Index (EEDI) standard (International Maritime Organization, 2011). Due to possible losses in maneuvering performance and navigational safety, this method has aroused questions. Therefore, the IMO published guidelines for calculating the minimum propulsion power required to keep a ship maneuverable in adverse conditions (International Maritime Organization, 2013). These guidelines caused a large number of studies aimed at examining maintaining maneuverability in a sea environment. Similarly, the most recent NATO maneuvering standards now encompass criteria related to maneuvering performance in wave-influenced circumstances.

Maneuvering a vessel in waves requires understanding the effects of waves on the vessel's performance and how to use the vessel's controls to navigate safely. The

vessel's speed, direction, and heading should be adjusted to account for the type and direction of the waves. Current methodologies to estimate the wave drift forces can be classified as follows (Tello Ruiz et al., 2012):

- Experimental methods (soft mooring, captive model tests, etc.)
- Methods based on two-time scale models,
- Methods based on a hybrid approach (or unified theory),
- Methods using computational fluid dynamics

Before the definition of methods based on a hybrid approach, ITTC proposed the definition of methods based on unified theory (ITTC, 2011). However, since there are studies (Skejic & Faltinsen, 2008) based on a unified method based on a two-time scale, the definition given above has been used.

Experimental methods for maneuvering in waves involve the systematic investigation of how vessels navigate through the water under the influence of wave-induced motions. These studies often employ scaled models to replicate real-world conditions. Two-time scale models separate the equations of motion into two groups as high-frequency wave-induced motion and low-frequency maneuvering motion. The hybrid approach combines both the high-frequency wave-induced motions and the low-frequency maneuvering motion to represent the maneuvering in waves. CFD methods discretize and solve the mathematical equations that govern fluid flow and wave propagation. Virtual environments are created to simulate the behavior of vessels navigating through varying wave scenarios.

1.1.1 Experimental methods

The physical tank model test in waves is an important method used to measure the forces acting on a floating structure due to wave action. Tests to detect wave loads involve allowing a hull to partially drift in a wave field and measuring the forces acting on it. The results of tests are also important for verifying the accuracy of numerical models used to predict the behavior of structures in given wave conditions.

Trajectories for ONRT maneuvering in calm water and waves are investigated by Sanada et al. (2013). In this study, besides the introduction of the Iowa Institute of Hydraulic Research wave basin facility established in those years, the maneuvering characteristics are presented. The focus is on the development of instrumentation of

the model and measurement systems. In 2014, Elshiekh presented the outcomes of a comprehensive series of turning and zig-zag tests involving a freely moving model of the ONR Tumblehome (Elshiekh, 2014). These tests were conducted in regular waves characterized by diverse combinations of wavelength, wave height, and heading, across a spectrum of forward speeds. The study also encompassed uncertainty analyses, and it provided detailed time records of wave-induced motions and the paths followed by the vessel. In this study, the experimental results of this thesis were used in comparative graphs and tables. A part of the results of the thesis are also published in 2019: To assess the maneuvering capabilities of the ONRT model under various conditions, maneuvering tests were conducted in both calm water and regular waves at the IIHR wave basin (Sanada et al., 2019). Standard maneuvering tests, including turning angles of $\delta = \pm 35^\circ$ and zigzag patterns of $10^\circ/10^\circ$ and $20^\circ/20^\circ$, were carried out. Parameters such as advance, tactical diameter, and overshoot angles (1st and 2nd) were evaluated following IMO criteria. Additionally, turning, zigzag, and course-keeping tests were performed in waves with three different wavelengths ($\lambda/L = 0.5, 1.0, \text{ and } 1.2$). The variation of ONRT's advance, tactical diameter, and drift distance with respect to wavelength was smaller compared to other ship hulls like S-175 and VLCC, suggesting ONRT's enhanced maneuverability due to its slenderness and twin-rudder twin-screw configuration. The outcomes of these tests provided insights into drifting distances and directions.

Sprenger et al. (2016, 2017) conducted an experimental work, highlighting the effect of adverse weather conditions. The study exemplifies the scope of the SHOPERA model test program (*Energy Efficient Safe SHip OPERAtion (SHOPERA)*, 2016) with two models namely the KVLCC2 and the DTC. The added resistance tests, propulsion and rudder force tests to obtain the effect of waves on appendages, and free-running tests are performed. The authors noted that progress on maneuvering in waves has been slow. At the same conference, El Moctar et al. (2016) submit a paper that investigates ship maneuvering in waves numerically and experimentally. The focus is on the DTC model only and some of the experimental data presented above research is used. As a result, the drag forces and yaw moments, and added resistance in the waves from numerical and experimental studies are compatible.

Wave drift forces and moments in regular head and beam waves are measured by Yasukawa (2006, 2008). Container ship model SR108 is used in the tests. The turning

trajectories, zig-zag, and stopping maneuvers in the conditions of $\chi = 0^\circ$ and $\chi = 90^\circ$ are compared for different wavelengths. The second report includes the wave loads results for different heading angles. The wave-induced coefficients were obtained by theoretical methods and the authors stated that they would carry out the studies for the development of wave loads estimation approaches in future work. The paper by Yasukawa et al. (2019) begins by presenting novel experimental data regarding wave-induced steady forces (added resistance, wave-induced steady lateral force, and yaw moment) acting on a full hull ship within regular waves. The study then proceeds to validate existing prediction methods for these wave-induced forces through experimental comparisons. Two methods are considered for prediction: the zero-speed 3D panel method and a formula-based approach utilizing the Kochin function denoted as the strip theory-based Kochin-function method. While both methods demonstrated acceptable calculation accuracy for added resistance and steady lateral force, they exhibited insufficiencies in predicting steady yaw moment. Also, The findings revealed the utility of both methods in predicting turning motions accurately within irregular wave conditions.

Milanov et al. (2020) explored the impact of regular waves on the initial parameters of turning maneuvers, along with their implications for a ship's course-keeping capabilities. This investigation was carried out through free-running model tests utilizing the KCS as a benchmark model. Using the data obtained from these tests, a straightforward maneuvering performance criterion was proposed. This criterion involved defining the initial turning time in wave conditions for two different rudder/course ratios. It served as a means to assess adequate maneuverability under challenging sea conditions.

Seo et al. (2021) analyzed the wave drift forces, namely surge wave drift force, sway wave drift force, and yaw drift moment, in oblique waves for ship model KVLCC2. To examine ship motion responses and wave drift forces, several regular wave experiments are carried out under several heading angles. A captive model test is carried out employing a soft spring-type mooring system. A potential-based method is preferred to show the computation capability. The authors stated that friction of the pulley in the soft spring system leads to loss of measurements. Although the measured wave drift loads are not used to reveal the maneuvering characteristics, it can be said that the study forms the basis of maneuvering in waves.

The doctoral thesis by Rabliås (2022) includes both experimental and numerical studies on maneuvering in regular and irregular waves of the DTC container ship. The experimental study is conducted via a free-running model. The two-time scale model is applied and some inconsistencies are observed in the following and oblique waves. That is because of the miscalculation of the slowly varying second-order wave loads. Due to the effect of lateral forces on the DTC hull, turning circles toward the starboard and port side differ from each other. Various publications have been made as outputs of the thesis (Rabliås et al., 2022; Rabliås & Kristiansen, 2022).

1.1.2 Two-time scale models

To solve high-frequency seakeeping problems and low-frequency maneuvering motion two-time scale models are preferred by some researchers. The low-frequency maneuvering motions are related to the wave drift loads. The two-time scale method divides the total ship motion into ship maneuvering and wave-induced motions, which are described by two different sets of motion equations.

Zhang et al. (2017) present a two-time scale method to predict ship maneuvering in regular waves. Maneuvering motion is calculated using an MMG model, while seakeeping is solved using a Rankine panel method. The study validates the method through simulations of turning tests and zig-zag tests of a ship model, comparing numerical results with experimental data. The influence of incident waves on maneuvering is highlighted, particularly with shorter wavelengths. The study suggests the need for further experimental validation and a more complex numerical model to better predict ship behavior, including wave effects on rudder forces.

Xie et al. (2020) introduce the theoretical derivations and the framework of combining seakeeping and maneuvering. The framework focuses on predicting second-order wave loads accurately and employs an innovative perturbation approach to calculate these loads both with and without a vessel's forward speed. The framework utilizes a two-time scale method and Neumann-Kelvin linearization for numerical simulations of maneuvering vessels (KVLCC and KCS models) in waves, yielding turning trajectories and zig-zag test results. The simulations reveal that wave drift loads, along with wave drift damping, cause vessels to deviate from their calm water trajectories, particularly as wavelengths decrease.

Chen et al. (2022) suggest that a two-time scale model based on the Taylor expansion boundary element method can forecast ship turning trajectories in regular waves. 3-DoF MMG model is used to calculate the equation of motion for the subject model KVLCC2. For the comparison, CFD results are used. Numerical simulation conditions are varied under rudder angle $\delta = \pm 35^\circ$, incident wave angle $\chi = 90^\circ, 120^\circ, 180^\circ, 210^\circ$, wave height $H/L_{pp} = 0.01$ to 0.02 and wavelength $\lambda/L_{pp} = 0.5$ to 1.2 . The main conclusion is that incident wave angle is highly affected the turning motion.

Ma, et al. (2022a) compared the two-time scale method and the unified method in regular waves. The newly developed in-house codes are validated by experimental data, displaying acceptable accuracy and calculation efficiency for maneuverability research in regular waves. While some differences between simulated and experimental results exist in short wavelength conditions, accuracy remains mostly adequate, capturing the drift of the turning circle. Notably, roll motions predicted by the two methods occasionally diverge. This is attributed to the unified method's simplified calculation approach for the first-order wave force, which might lead to the underestimation of roll motions.

1.1.3 Hybrid approach

Skejic & Faltinsen (2008) introduced a two-time-scale model that segregated the low-frequency and high-frequency movements for predicting a ship's behavior in regular waves. The analysis of maneuvering utilized a nonlinear slender-body theory, which was adapted to incorporate heel effects. The forces and moments arising from the rudder, propeller, and viscous cross-flow were sourced from state-of-the-art procedures. Linear motions and wave-induced loads were established through a 6-DoF strip theory. To estimate mean second-order wave loads when dealing with oblique sea conditions and incident regular waves, four distinct methods based on potential flow theory were employed. Simulations of turning and zig-zag trials for vessels Mariner and Esso Osaka were conducted and compared against measurements from full-scale trials. Afterward, the study by Skejic & Faltinsen (2013) presents a unified model for predicting the maneuvering behavior of displacement-type ships in a seaway. The model incorporates two distinct time scales for maneuvering and seakeeping, using a modular concept developed by Skejic & Faltinsen (2008). The

unified model is applied to two types of displacement ships: the S-175 container ship and the MARINER ship engaged in turn maneuvers in regular and irregular wave fields. The study emphasizes the need for stochastic approaches in addressing the complex ship maneuvering problem in a seaway.

In 2011, Seo and Kim worked on a unified model for predicting ship maneuvering in wave conditions (Seo & Kim, 2011). Within this framework, second-order wave drift forces were computed using a time-domain 3D Rankine panel method. These force components were then integrated into a 4-DoF modular mathematical model for maneuvering performance prediction. This model facilitated the determination of the vessel's position and velocity, which subsequently aided in calculating average wave drift forces. This iterative procedure was employed to replicate turning and zigzag trials for Series 60 and the S-175 containership models, both in calm waters and wave environments. Various scenarios were considered, including head seas and beam seas, with λ/L ratios of 0.7 and 1.0, and wave H/L ratio of 0.02. The simulation results displayed reasonable agreement with the experimental trajectory data. Seo and colleagues employed a modular 3-DoF maneuvering model to assess ship maneuverability in wave conditions (Seo et al., 2018, 2020). The model utilized a higher-order Rankine panel method to compute wave drift forces across various combinations of wave frequency, wave heading, forward speed, and lateral speed. These computed forces were then stored within a database for utilization in time domain simulations. By conducting turning maneuvers in regular waves, the simulated results were compared with experimental trajectories. Furthermore, the study explored the sensitivity of time signals of wave elevation in irregular wave scenarios, investigating their impact on drifting distance and drifting angle.

Schoop-Zipfel & Abdel-Maksoud (2014) demonstrates the feasibility of simulating ship maneuvers using potential theory. Subsequent free maneuvers are simulated, and the results exhibit good agreement with experimental data. To incorporate wave effects, wave excitation forces are introduced as additional forces acting on the vessel's rigid body, showing good agreement with measured values for restrained ships. However, discrepancies arise in free heave and pitch motion due to omitted frequency dependence in the added mass and damping terms, particularly for high-frequency motion in a seaway. Afterward, the thesis by Schoop-Zipfel (2016) focuses on separately the zero-frequency maneuvering motion and the high-frequency wave-

induced motion. The solution of these motions separately leads to a unified theory. The turning circles of the S-175 container ship and the KVLCC2 tanker in regular waves are simulated. The author states that the determination of the second-wave loads is difficult, so a calculation method is utilized for second-order wave drift forces.

Chilcce & el Moctar (2018) introduced a numerical approach for simulating ship maneuvering within wave conditions, employing the nonlinear 6-DoF equations of motion. The approach operated under the assumption that there's no interaction between calm water maneuvering and wave-induced forces. Hydrodynamic coefficients for calm water forces were determined through PMM tests, utilizing RANS equations. The simulation only accounted for second-order wave-induced forces, which were computed using a 3D Rankine source-based boundary element method. This simulation methodology was applied to the DTC for which both experimental data for maneuvering motions in calm water and wave conditions were available.

Subramanian & Beck (2019) introduces a computational model for analyzing ship maneuvering in a challenging seaway environment. The model addresses interactions between the hull, propeller, and rudder, accounting for different time scales and complex fluid dynamics. A nonlinear time-domain strip theory is employed to solve the wave-body interaction problem efficiently. The model uses a systems-based approach to simulate forces from the hull, rudder, and propeller during maneuvers. Simulations of a containership's turning circle in calm water and waves show good agreement with experiments, though there are some quantitative discrepancies. The role of the propeller in influencing rudder inflow is highlighted, along with potential inadequacies in certain models used.

Lee & Kim (2020) investigated the influence of steady flow coupling on a ship's seakeeping and maneuvering behaviors during steering in waves. Seakeeping analysis was conducted considering forward and oblique ship movements. Numerical simulations were performed for turning tests in regular waves. The study compared maneuvering indices and drifting distances in a turning test. In the portside turning test, the negative yaw moment caused by operational velocity counteracted turning. However, numerical computations using the Rankine panel method showed less rotation delay due to underestimated yaw moments. Consequently, advance and

tactical diameter were underestimated, especially for initial beam waves when the initial speed was relatively fast due to minimal added resistance and speed loss.

Yao et al. (2020) proposed time-averaged equations to address the problem of ship maneuvering in waves, providing a novel approach for predicting such maneuvers. Application and analysis of the method indicate its effectiveness and reasonable accuracy, comparable to existing system-based methods. The potential extension of the method to irregular waves is noted, with a need for an appropriate formulation of wave drift forces, moments, and high-frequency motions. When modeling wave drift forces, moments, and high-frequency motions, a coupling of parameters like rudder angle and maneuvering motions is recommended, rather than relying solely on the wave encounter angle.

Yu et al. (2021) introduced a comprehensive simulation model designed to predict the turning maneuvering of a ship (specifically the S175) in both regular and long-crest irregular waves. The model involves a 6-DoF framework and combines two distinct components: a modular mathematical maneuvering model encompassing 3-DoF (surge, sway, and yaw), and a seakeeping model with 5-DoF (sway, heave, roll, pitch, and yaw). These components are weakly coupled through the unified theory. Notably, the calculation of mean 2nd order wave drift forces employed Maruo's far-field theory, with a focus solely on the vessel's motion-induced component. The researchers performed numerical simulations of the S175's turning behavior within regular and irregular wave conditions, illustrating the model's effectiveness in capturing these dynamic scenarios.

White et al. (2022) evolve an algorithm for coupling maneuvering and seakeeping problems through CFD and boundary element methods. A hybrid method is applied to figure out turning circle trajectories in waves of the DTC model. As a conclusion, the hybrid method is shown to be more computationally efficient.

A comparative study is performed by Zhang et al. (2022) by using a quasi-steady method and a time domain approach. The wave drift loads are calculated via potential flow theory and integrated into MMG equations. The quasi-steady method represents the frequency domain computation assuming the relation of the wave and ship is time-harmonic. In the time domain approach, maneuvering simulation and seakeeping computations are parallel to each other. The authors stated that lateral wave drift force

and yaw moment have less effect on maneuvering in comparison with added resistance.

Gong et al. (2022) present a hybrid approach that combines a 6DOF maneuvering motion model with wave propulsive factors to simulate turn and zigzag maneuvers of a single-screw ship within wave conditions. This approach incorporates the ship's free motion and the influence of waves on the propeller while excluding discrete simulations of the rudder and propeller to enhance computational efficiency. In the context of the turning maneuver, it's observed that compared to beam waves, the ship experiences greater trajectory drift during maneuvers in head and stern waves. Short wavelengths have a notable impact on the ship's turning trajectory, especially when the wavelength is shorter than the ship's length. This effect results in a pronounced increase in trajectory drift during the turn maneuver. Additionally, the initial forward speed's influence on the turn maneuver is as significant as the wavelength. Lower speeds are more susceptible to wave influence, resulting in substantial drift distances during the maneuver.

1.1.4 Computational methods

One area where CFD methods have particularly gained prominence is in seakeeping and maneuvering studies. The preference for CFD can be attributed to several factors.

CFD allows for detailed analysis of fluid flow around a ship, providing accurate predictions of hydrodynamic forces and moments. CFD methods offer flexibility in modeling various complex scenarios, such as regular/irregular wave conditions, different hull forms, appendages, and operating conditions. This allows for a wide range of simulations to be performed, providing insights into different seakeeping and maneuvering scenarios. Conducting physical model tests for seakeeping and maneuvering studies can be expensive and time-consuming. CFD simulations offer a cost-effective alternative, as they require fewer physical resources and can be performed in a shorter time frame. This allows for multiple design iterations and parametric studies to be conducted efficiently.

It's worth noting that while CFD methods have many advantages, the CFD method is less common than numerical simulation based on the two-time scale model and hybrid approach in this field. The two-time scale model and the unified theory are similar in scope and focus but differ in practice.

A stand-alone CFD study is not very common, it has often been seen to be studied in conjunction with experimental or other approaches. Therefore, CFD studies can be found in the studies given in the above sections. Wang & Wan (2018) use an open-source platform to solve the maneuvering in waves for the ONRT model. The time histories of maneuvering characteristics, such as ship speed, heave motion, roll motion, and pitch motion are compared with available experimental data. The authors noted that the computational cost for the turning circle maneuver is too high. Ma et al., (2022) carried out a study involving experimental (Ma et al., 2022) and numerical investigations of PMM tests conducted on the standard ship model KVLCC2 in regular head waves. Wave forces during maneuvering motions are influenced by both high-frequency encounter wave frequency and low-frequency maneuvering motions. Peaks occur around the encounter wave frequency and its multiples, challenging studies using precalculated second-order wave forces and real-time encounter wave frequency.

1.1.5 Wave drift loads

In this chapter, the literature on the investigation of loads is included since the determination of wave drift forces and moments was examined in the first stage of this thesis. Wave drift forces and moments can be calculated using various methods, each with its level of complexity and accuracy. There are two options for approximating the mean wave-drift forces: one involves directly integrating the pressure over the body surface (Chen et al., 2022; Liu & Papanikolaou, 2020), while the other relies on the principle of momentum conservation (Dobrochinski et al., 2023; Maruo, 1960).

There are empirical and analytical approaches that provide simplified methods to calculate mean wave drift forces. Even though they might not fully account for the complexity of the wave-body interaction, they can be helpful for quick calculations or the early phases of design. References to empirical and analytical approaches are included in chapter 2, with details of the methods.

Also, conducting physical model tests in a controlled wave tank environment can provide accurate measurements of the wave drift forces on a scaled-down model of the ship. These tests require a special wave tank; on the other hand, towing tank tests can be extended to include wave-generating mechanisms to measure the forces and moments. For example, the wave drift loads are examined with a series of towing tests

of a bulk cargo ship model by Xu et al. (2011). The most stable data with a length of approximately 5-8 wave periods were selected from the time records. The Fourier analysis is used to analyze the test data collected from the planar motion mechanism. The steady force at zero frequency gives the added resistance. Seo et al. (2021) performed a set of experiments to estimate the wave drift forces. Captive model tests of the KVLCC2 model with a soft spring-type mooring system were carried out under various ship speeds and wave conditions, providing validation data for wave drift forces. The force sensors and tensiometers are used for data collection. A direct method using force sensors was identified as more accurate than an indirect method using tensiometers due to reduced measurement losses.

1.2 Research Gap

While significant advancements have been made in ship maneuvering, there exists a notable research gap in understanding ship maneuvering in waves. Although numerous studies have explored the external forces and moments, a comprehensive investigation into the influence of waves on maneuvering remains largely unexplored. This knowledge gap is particularly significant in the context of system-based methods, where a deeper understanding of wave effects could lead to enhanced approaches. Furthermore, existing research tends to focus on 3-DoF motion, often overlooking the intricate interaction between other motions. The limited studies that do address wave loads frequently lack a holistic approach that integrates both theoretical analyses and experimental approaches.

As stated in (Kinaci & Ozturk, 2022) conventional single-propeller, single rudder ships typically rely on system-based maneuvering mathematical models for maneuvering predictions. These models, in their typical configurations, do not account for the flow imbalance caused by the propeller. To enhance the capacity of the system-based mathematical models, both this previous study and present study focus on ships with two propellers, and two rudders.

The study by Kinaci & Ozturk (2023) focuses on the on the controlled motion of ships at sea, particularly addressing the impact of waves on ship maneuverability. The researchers examined the Japan Bulk Carrier (JBC) in calm water and in the presence of waves, conducting turning circle tests using system-based analysis and direct CFD methods in calm water conditions. In this study, which is described as a preliminary

study of the thesis, wave loads are empirically included in the system-based mathematical model. Although the representation of the waves was done successfully, improvements were needed to reflect the real situation. Therefore, in this thesis study, wave-induced forces and moments were examined experimentally.

Addressing this research gap is critical for advancing ship maneuvering and unlocking its potential for system-based approaches. By comprehensively investigating the impact of waves on ship maneuvering, this doctoral thesis aims to contribute to the broader understanding of ship maneuvering, providing valuable insights for experimental, numerical, and theoretical approaches.

1.3 Research Questions and Problem Statement

In line with the literature presented in the previous sections, the research questions the thesis aims to address are stated as follows:

- What is the effect of wave forces and moments on ship turning circle maneuver performance?
- How is it possible to experimentally measure mean wave loads in a way that can be integrated into the system-based model using the facilities in ITU Ata Nutku Ship Model Laboratory?
- Is it possible to establish a universal mathematical model of maneuvering valid in all cases where wave loads are included?
- Can computational fluid dynamics methods be preferred over experimental fluid dynamics in maneuvering in wave problems?
- Is it possible to develop a practical system-based method based on the knowledge gained from computational fluid dynamics and experimental fluid dynamics methods?

Within the framework of these questions, the problem statement of the thesis was determined as the constitution and implementation of a system-based model, in which experimental data can be processed that will reveal the effect of wave-induced forces and moments on ship turning circle maneuver by comprehensively analyzing the wave effect in different ship headings.

1.4 Objectives and Purpose

In accordance with the problem statement in the previous section, the specific goals and objectives of the thesis are outlined as follows:

- Identifying definitions that affect ship maneuvering performance
- Revealing hull and wave interaction with experimental methods
- Gain insight through a comprehensive study of the interactive behavior of wave and hull and use this interaction to develop a mathematical model of maneuvering in waves
- Investigating reducing dependency on numerical and experimental studies through the effectiveness of system-based methods
- Learning the working principle of the planar motion mechanism by which maneuvering derivatives are obtained

The main purpose of the thesis is to reveal the mean wave drift loads in detail and to show the effects on the estimation of maneuvering performance through various approaches – experimental, numerical, and system-based studies.

1.5 Significance and Relevance

In the present section, the importance of the problem and its contribution to the existing body of knowledge will be explained. It is explained how the study fills the gap in this field. The potential impact of the findings on theory and practice is discussed.

Although there are many system-based, experimental, and numerical studies in the estimation of maneuvering characteristics, there are still controversial issues, primarily about maneuvering in waves. In the light of the assessments presented in the research gap section, the significance and relevance of the proposed thesis can be listed as follows:

- The findings of this thesis will be presented in the activity named "Assessment of Experiments and Prediction Methods for Naval Ships Maneuvering in Waves", the activity period of which is 2021-2023 and continues. The activity is carried out within the scope of Applied Vehicle Technology (AVT) and is referred to by the code AVT-348.

- The effect of the wave on parameters such as turning trajectories, speed, yaw rate, and roll angle of a maneuvering ship is an important research topic. Especially if these parameters are used for hull optimization, they are of greater importance. Within the scope of the thesis, depending on the experimental possibilities, these wave-induced forces and moments will be tried to be determined. It is expected that this study will make an important contribution to current scientific studies.
- Most current mathematical models for ship maneuvering neglect the wave effect. That is, it is assumed that the ship is maneuvering in calm water, and as a result, wave forces and moments are neglected. In other words, the calculations are made as if the current is constant on the hull, even though the ship is exposed to external forces during the maneuvering. The significance of the study would be to include wave-induced external effects in mathematical models. Due to the local flow that differs according to the changing heading angle, the wave creates forces and moments that contribute to the overall loads. It is thought that it will contribute to the literature with the aim of numerically and experimentally examining wave-induced forces and moments and developing existing system-based methods.
- Using CFD-based approaches, it is aimed to provide a clear and detailed understanding of the physical phenomenon of a maneuvering vessel and can make a valuable contribution to further numerical modeling of the maneuvering in waves.
- Detection of wave loads with steady drift tests, that is, in experiments carried out at zero speed and at different heading angles, can assist in developing physically-based maneuvering models and experimental setups in wave basins that do not have the opportunity to do free-model testing.
- Generally, simplified mathematical models are based on solving the 3-DoF motion of the ship. Some researchers have developed systems-based models of 4- and 6-DoF motions to depict a realistic operational scenario. A 4-DoF model will also be used in the thesis, and it is expected to contribute to the transition to the next stage, the 6-DoF model.

- Some empirical or analytical approaches were found to be inappropriate for estimating wave loads: Force and moments are overestimated. Although these results are extremely useful as they are incomparably faster and easier to use than any Navier-Stokes solver, they clearly show the limit of simplified approaches. Within the scope of the thesis, it is planned to benefit from these approaches and show the differences.
- The amount of dataset used to validate simplified maneuvering models and numerical results will be increased for ONR Tumblehome.

1.6 Scope and Limitations

This doctoral thesis focuses on a practical experimental approach to measure the mean wave drift forces and moments. The research investigates the effect of wave loads on ship maneuvering and aims to contribute to the system-based mathematical approaches. The scope includes theoretical modeling, numerical modeling, and experimental approaches to achieve a comprehensive exploration of the research problem.

The study is subject to time limitations, which may impact the depth of numerical simulation. The complexity of directly-solved maneuvering in waves might require further numerical investigation beyond the scope of this study.

The availability of certain resources, for instance, a wave basin that has the opportunity to perform free-model testing, and computer requirements for CFD simulations, may affect the comprehensiveness of the experimental and numerical phase.

The research focuses on heading angles between 0° to 180° , assuming the ONRT model is symmetrical. Extrapolation of results beyond this range, i.e. 180° to 360° , should be approached cautiously and may require additional research.

To streamline the system-based investigation, certain assumptions will be made. These assumptions will be detailed in the following chapters and might errors may arise due to simplification.

The parameters of the system-based model will be found specifically for the ONRT model. While the findings of this study will contribute to the understanding of

maneuvering in waves, their generalizability to other ship models might be influenced by variations of the maneuvering coefficients, rudder and propeller parameters.

1.7 Methodology

An overview of the research methods and techniques used to conduct the study is provided: This study follows a system-based mathematical approach, utilizing both laboratory experimentation and simulation-based analysis to achieve the research objectives. The methodology combines theoretical modeling, computational simulations, and practical measurements.

Firstly, empirical and analytical procedures will be followed to estimate the mean wave drift forces and moments. Any discrepancies will be investigated by comparing them with the experimental and computational results.

A physical experimental setup will be designed and constructed to reach the mean wave drift loads. The setup will include Human Machine Interface for the loads, and a data acquisition program for the waves to ensure accurate data collection. The ship model will be mounted to a pantograph and rotated via a horizontal planar motion mechanism. The time series of forces and moments acting on the hull will be collected using a multi-component load cell. The time series of waves will be captured with wave height gauges and probes.

The research involves the use of computational fluid dynamics software. A detailed model of the steady drift experiments will be replicated within the software, incorporating relevant physical properties, boundary conditions, and system parameters. A series of numerical simulations will be conducted to investigate the effect of wave loads. These simulations involve varying parameters such as the wave height/wavelength (H/λ), the wave/ship length ratios (λ/L_{WL}), and ship heading ψ . The simulations will be verified and validated against the experimental data. Also, a direct computational approach to solve maneuvering motion in calm water will be performed.

Experimental and computational data will be processed to calculate the mean wave drift loads and will undergo analysis using Fast Fourier Transform. The analysis will include Savitzky-Golay filtering to smooth or filter noisy data. Graphical representations and tables will aid in interpreting the results.

A system-based mathematical model for representing the 4-DoF maneuvering of a vessel in calm water will be presented. Then, the measured wave drift loads will be incorporated into the equations of motion for maneuvering to predict the effect of waves on the maneuvering performance of a vessel. Both calm water and three different wave scenarios will be investigated. Comparisons will be made with existing experimental and numerical results in the literature.

1.8 Structure of Thesis

This doctoral thesis is organized into distinct chapters, each contributing to the exploration and understanding of maneuvering in waves. The structure is designed to systematically guide the reader through the process of determining the context and significance of the study, presenting and evaluating the findings. The chapters that make up this thesis are summarized as follows:

This chapter introduces the related literature review, outlines the research problem, and provides the significance of the study. The limitations and methodology are summarized, setting the stage for the subsequent chapters.

Chapter 2 delves into the existing empirical and analytical approaches related to mean wave load calculations. Calculation results are presented for different wave conditions, to be a foundation for the current research.

In Chapter 3, the experimental research methodology is outlined in detail. The chosen experimental method, experimental model, and measurement systems are explained. Quantitative methodologies are presented, providing an understanding of data reduction. The experimental results for both wave drift loads and wave parameters are presented in graphs and tables.

Chapter 4 focuses on the steady drift analysis under the conditions tested in the previous chapter. It presents an overview of equations and methods, numerical models, and simulation scenarios. The findings from the analysis are validated and verified. Experimental results are compared with the numerical outcomes. Detailed analysis and interpretation of results are provided, supported by graphs, and tables. The details of the direct simulation performed in calm water are outlined.

Chapter 5 explores the 4-DoF system-based methods to adapt the mean wave drift loads. It discusses the derivation of 6-DoF and 4-DoF equation of motion and external

forces and moments. Findings from maneuvering in calm water and in waves simulations are discussed.

The thesis is concluded with Chapter 6, which summarized major findings and highlights contributions. Future research and potential extensions of the study are identified.





2. MEAN WAVE DRIFT FORCE PREDICTION

Empirical and analytical approaches to wave drift force prediction are widely used in practice, as they provide a relatively simple and efficient way to estimate wave drift forces without resorting to complex numerical simulations or experimental methods. These approaches are based on formulas or design curves that relate the wave drift force to various parameters such as the wave height and period, as well as the size and shape of the vessel or structure. The following works ensured practical methods for estimating the wave drift force. The underlying assumptions and limitations will be discussed. Also, the effect of wave heading angles at zero speed ONRT model on the steady wave drift forces will be explored for each approach.

Advanced Marine Enterprises Inc. (1996) recommends the following expressions for the calculation of the mean wave drift force and moment components, $\bar{X}_W$, $\bar{Y}_W$ and $\bar{N}_W$:

$$\bar{X}_W = \left[0.0388 \rho g B C_B H_s^2 f(H_s, T) + 0.311 \rho g B^2 \frac{H_s^{2.5}}{L^{1.5}} \right] \cos \chi$$

$$\bar{Y}_W = [0.0388 \rho g L H_s^2 f(H_s, T)] \sin \chi \quad (2.1)$$

$$\bar{N}_W = [-0.125 \rho g L T H_s^2 f(H_s, T)] \cos \chi \sin \chi - 0.03 \bar{Y}_W L$$

where ρ is density, g is gravitational acceleration, B is breadth, C_B is block coefficient, H_s is significant wave height, d is draft, L is ship length, and χ is the incident wave angle. The $f(H_s, d)$ is a function changing with the relation between significant wave height and draft and can be calculated as follows:

$$f(H_s, d) = \sin^2 \left(\frac{d}{2H_s} \right), \text{ if } 0.32d < H_s < d$$

$$f(H_s, d) = \sqrt{\frac{H_s}{d}} \sin^2 \left(\frac{d}{2H_s} \right), \text{ if } H_s > d \quad (2.2)$$

$$f(H_s, d) = 1, \text{ if } H_s < 0.32d$$

It has been stated that the suggested expressions are for full cargo ships. Therefore, it can be said that the expressions are for fuller ships with larger block coefficients, and can only give an idea about the wave drift force and moment components for ONRT, a finer form with a block coefficient of 0.535.

Another difference between the present study and above relation is that the approach is based on the Pierson Moskowitz spectrum which assumes a deep sea and a fully developed sea state. The Pierson-Moskowitz approximation includes parameters such as significant wave height, peak wave period, and spectral shape. On the other hand, the regular wave is a simplified representation of ocean waves and assumes idealized conditions with uniform wave height and period. In the H_s parameter in the above expressions, the peak-to-peak value of the regular waveform is used instead of the significant wave height.

The original version of the formulas also includes the correction coefficients for the shallow water condition. According to the linear wave theory, water depths more than half the wavelength indicate deep water ($d > 0.5\lambda$). Experimental conditions for steady drift tests provide deep water conditions with a water depth of 3.5 m and a maximum wavelength of 3.147 m. Therefore, the shallow water correction coefficients were not considered.

A very simple approach is preferred for the estimation of mean wave drift loads for buoys with a specific diameter in irregular waves by Bergdahl & Kofoed (2015). The assumption in this approach is that the object reflects all waves in the opposite direction to the incoming waves for all component waves. The wave drift force, $\bar{F}_W$, is presented as follows:

$$\bar{F}_W = \frac{1}{2} \rho g \sum_i \frac{1}{2} a_i^2 D = \frac{\rho g H_s^2 D}{32} \quad (2.3)$$

where a_i is the wave amplitude in a Pierson Moskowitz spectrum, equal to $H_s/2$, and D is the characteristic dimension of the buoy. In the case of a surface ship, the characteristic dimension, D , can be taken as the length of the ship, L , for longitudinal wave drift force and as the breadth of the ship, B , for lateral wave drift force. Then the wave drift force expressions become,

$$\begin{aligned}\bar{X}_W &= \frac{1}{32} \rho g L H_s^2 \\ \bar{Y}_W &= \frac{1}{32} \rho g B H_s^2\end{aligned}\tag{2.4}$$

Then the magnitude of the resultant force acting on the hull can be calculated with a very basic way of $\bar{F}_W = \sqrt{\bar{X}_W^2 + \bar{Y}_W^2}$. The effect of changing the heading angle, or incident wave angle in this case, to the wave drift forces can be included by

$$\begin{aligned}\bar{X}_W &= \bar{F}_W \cos \chi \\ \bar{Y}_W &= \bar{F}_W \sin \chi\end{aligned}\tag{2.5}$$

where χ is the incident wave angle.

The authors also calculated the wave drift force resulting from diffraction, from the idea that the case object can not reflect the wave they investigated. The estimated wave drift load due to diffraction is relatively too smaller than the first approximation so the simple approach can be preferred.

For rectangular submerged bodies, a relation for mean wave drift force estimation is also proposed by Gupta (2008). To illustrate how to calculate the mean wave drift loads, consider a rectangular barge with length L , width B , and draft d . The velocity potential of the regular wave acting on this barge from the head and the wave profile can be written as follows:

$$\begin{aligned}\Phi(x, y, z; t) &= \rho g \frac{H}{2\omega} \frac{\cosh k_w(z + d)}{\cosh k_w d} e^{i(k_w x - \omega t)} \\ \eta(t) &= \frac{H}{2} e^{\pm i(k_w \frac{L}{2} - \omega t)}\end{aligned}\tag{2.6}$$

where H is the wave height, ω is the wave frequency, k_w is the wave number ($2\pi/\lambda$), and d is the depth. With the integration of the free surface along the wave, the first component can be found as follows:

$$\bar{F}_1 = \frac{1}{2} \rho g \int_{WL} \eta^2 n dL = \frac{1}{2} \rho g B \left\{ \left[\frac{H}{2} e^{i(k_w \frac{L}{2} - \omega t)} \right]^2 - \left[\frac{H}{2} e^{i(-k_w \frac{L}{2} - \omega t)} \right]^2 \right\}\tag{2.7}$$

$$\bar{F}_1 = \frac{H^2}{8} \rho g B \left\{ \left[e^{i(k\frac{L}{2} - \omega t)} \right]^2 - \left[e^{i(-k\frac{L}{2} - \omega t)} \right]^2 \right\}$$

$$\begin{aligned} \bar{F}_1 = \frac{H^2}{8} \rho g B & \left\{ \cos^2 \left(\frac{kL}{2} - \omega t \right) - \sin^2 \left(\frac{kL}{2} - \omega t \right) \right. \\ & + 2i \sin \left(\frac{kL}{2} - \omega t \right) \cos \left(\frac{kL}{2} - \omega t \right) - \cos^2 \left(\frac{kL}{2} - \omega t \right) \\ & \left. + \sin^2 \left(\frac{kL}{2} - \omega t \right) + 2i \sin \left(\frac{kL}{2} - \omega t \right) \cos \left(\frac{kL}{2} - \omega t \right) \right\} \end{aligned}$$

The above expression can be simplified as follows and equal to zero at which on averaging over one complete cycle.

$$\bar{F}_1 = \frac{H^2}{8} \rho g B [2 \sin(kL) \sin(\omega t)] = 0 \quad (2.8)$$

The quadratic velocity expression in Bernoulli's equation can be written as follows.

$$\begin{aligned} u^2(x, y, z, t) + w^2(x, y, z, t) = & \left(\frac{gkH}{2\omega \cosh kd} \right)^2 \times \\ & \left\{ \cosh^2 k(z + d) \cos^2 \left(\pm \frac{kL}{2} - \omega t \right) + \sinh^2 k(z + d) \sin^2 \left(\pm \frac{kL}{2} - \omega t \right) \right\} \end{aligned} \quad (2.9)$$

By integrating this expression from the bottom of the ship, $-T$, to the waterline level, the following expression is obtained.

$$\bar{F}_2 = \frac{H^2}{4} \frac{\rho g B}{4} \frac{\sinh 2kd - \sinh 2k(d - d)}{\sinh 2kd} \quad (2.10)$$

In deep water, this expression becomes,

$$\bar{F}_2 = \frac{H^2}{16} \rho g B \frac{1 - (\cosh 2kd - \sinh 2kd)}{2} \quad (2.11)$$

If the effect of ship motions on the wave drift force is neglected,

$$\bar{F}_3 = \bar{F}_4 = 0 \quad (2.12)$$

Thus, in regular head waves and deep water, the average longitudinal and lateral wave drift forces acting on a surface ship can be simplified as follows:

$$\begin{aligned}\bar{X}_W &= \frac{H^2}{16} \rho g L \frac{1 - (\cosh 2kd - \sinh 2kd)}{2} \\ \bar{Y}_W &= \frac{H^2}{16} \rho g B \frac{1 - (\cosh 2kd - \sinh 2kd)}{2}\end{aligned}\tag{2.13}$$

American Petroleum Institute (2005) presents design curves of wave drift force and motions. The curves of results are generated for drillships with lengths between 121.92 m and 164.59 m and displacements between 11.176 tons to 21.337 tons. Also, there are some results for four semisubmersible designs. The significant wave height is changed between 1.524 m to 12.192 m. The authors draw attention to that the data can be useful for ship-shape vessels and interpolation methods can be used for different dimensions. The curves for drillships will be used to estimate the mean wave drift force of the ONRT model. The wave drift force and moment estimation for the ONRT model was made using the curve created for the ship with the lowest displacement. The drift forces corresponding to the current wave height are read from the curves and dimensioned according to the drillship characteristics. Then, nondimensionalized longitudinal and lateral forces are dimensionalized according to the ONRT model. Equation 2.5 can be used to compute wave drift forces and moments at different incidence wave angles after determining the resultant force.

DNV AS (2021) recommends the following relations to estimate the mean wave drift forces and moment. The wave drift forces model is based on Pierson Moskowitz wave spectrum.

$$\begin{aligned}\bar{X}_W &= \frac{\rho}{2} g H_s^2 B h f(T'_{surge}) \\ \bar{Y}_W &= \frac{\rho}{2} g H_s^2 L_{os} (0.09 \sin(\psi)) f(T'_{sway}) \\ \bar{N}_W &= \bar{Y}_W \left(x_{Los} \left(0.05 - 0.14 \frac{\text{rad}(\psi)}{\pi} \right) L_{os} \right)\end{aligned}\tag{2.14}$$

where ψ is the heading angle, L_{os} is the longitudinal distance between the foremost and aft-most point underwater, and x_{Los} is the longitudinal position of $L_{os}/2$ to the point of $L_{pp}/2$. The h coefficient can be calculated as follows:

$$h = 0.09h_1h_2 \quad (2.15)$$

The following equations are used to calculate the coefficients h_1 and h_2 .

$$h_1 = h_{1A} + \frac{\text{rad}(\psi)}{\pi}(h_{1B} - h_{1A}) \quad (2.16)$$

$$h_2 = 0.05 + 0.95 \arctan(1.45(\text{rad}(\psi) - 1.75))$$

The coefficients h_{1A} and h_{1B} is found by the following expressions.

$$h_{1A} = 0.8\alpha_{bow}^{0.45} \quad (2.17)$$

$$h_{1B} = 0.7C_{WL\text{aft}}^2 \text{ where } 0.85 < C_{WL\text{aft}} < 1.15$$

where α_{bow} is the angle between the vessel x -axis and a line drawn from the foremost point in the water line to the point at $y = B/4$ (ahead of $L_{pp}/2$) on the water line, and $C_{WL\text{aft}}$ is the water plane area coefficient. Nondimensional T'_{surge} and T'_{sway} parameters can be calculated as follows:

$$T'_{surge} = \frac{T_Z}{0.9L_{pp}^{0.33}} \quad (2.18)$$

$$T'_{sway} = \frac{T_Z}{0.75B^{0.5}}$$

where T_Z is the average zero up-crossing period of the wave. The relations for the function of T' is given in the following expressions.

$$f(T') = 1, \text{ if } T' < 1 \quad (2.19)$$

$$f(T') = T'^{-3}e^{1-T'^{-3}}, \text{ if } T' \geq 1$$

The geometric properties of ONRT are found in the CAD model as $x_{Los} = 0.03 \text{ m}$, $\alpha_{bow} = 0.169 \text{ rad}$, and $C_{WL\text{aft}} = 1.021$. The relation between the zero up-crossing period of the wave, T_Z , and peak period, T_P can be written as $T_P = T_Z\sqrt{2}$. Therefore the zero up-crossing period of the wave for the steady drift test is calculated as 0.950 s.

2.1 Comparison of Mean Wave Drift Loads

The results of wave drift loads and moments can provide insights into the loads exerted on ships, offshore platforms, and other marine structures, helping researchers make informed decisions during the maneuvering phases. Comparing empirical/analytical results of wave drift loads and moments allows us to assess the accuracy and reliability of different models and methodologies. It can be said that the above-mentioned approaches, unlike the ONRT model, are created by starting from rectangular shapes, buoys, or fuller vessels and most of them are valid in irregular waves. Therefore, it is important to recognize the limitations and challenges associated with comparing results. For this reason, experimental studies and computational approaches become essential, as explained in the following sections, differences will be observed when compared to empirical and analytical results.

The results for wave drift forces and moment are plotted with respect to the heading angle and compared in Figure 2.1 to 2.3 for the wave condition of $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$. Also, the data are presented in Table 2.1 to Table 2.3.

Figure 2.1 shows the results of mean longitudinal wave drift forces calculated for the ONRT model with five different approaches. Calculations were performed for the wave steepness value of 0.020 and the wave/ship length ratio of 1.00. As expected, the longitudinal wave drift forces reached their highest values in the head and following seas. In general, it can be seen that the values are symmetrical with respect to $\psi = 90^\circ$. Only in DNV AS (2021) results, the symmetry was disrupted, and the longitudinal wave drift force reached zero at a point where the heading angle was between 90° and 105° .

The graph clearly shows that the results of Bergdahl & Kofoed (2015) are higher than the other four approaches. The authors pointed out that the approach also presented above, which they call simple wave drift estimation, is 40 times the drift force calculated by the diffraction theory. Therefore, it can be considered that this approach presented by the authors for environmental loads on the buoy is not suitable for the ONRT model when compared to other methods.

The results of API (2005) and Gupta (2008) follow a similar trend as in Advanced Marine Enterprises Inc. (1996) and DNV AS (2021). API (2005) presents design curves for estimating mean wave drift forces for drillships and the authors stated that

the design curves are suitable for mobile offshore drilling unit type vessels. In light of this information, it can be expected that the longitudinal wave drift force will be less in military vessels like high-speed corvettes or frigates with low block coefficients.

Table 2.1 : Comparison of longitudinal wave drift forces, $\bar{X}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$.

Heading angle, ψ (deg)	Advanced Marine Enterprises Inc. (1996)	API (2005)	Gupta (2008)	Bergdahl & Kofoed (2015)	DNV AS (2021)
0	-0.267	-1.492	-1.388	-3.849	-0.262
15	-0.258	-1.441	-1.341	-3.718	-0.270
30	-0.231	-1.292	-1.202	-3.333	-0.271
45	-0.189	-1.055	-0.981	-2.722	-0.259
60	-0.133	-0.746	-0.694	-1.924	-0.229
75	-0.069	-0.386	-0.359	-0.996	-0.169
90	0.000	0.000	0.000	0.000	-0.070
105	0.069	0.386	0.359	0.996	0.063
120	0.133	0.746	0.694	1.924	0.200
135	0.189	1.055	0.981	2.722	0.315
150	0.231	1.292	1.202	3.333	0.406
165	0.258	1.441	1.341	3.718	0.480
180	0.267	1.492	1.388	3.849	0.542

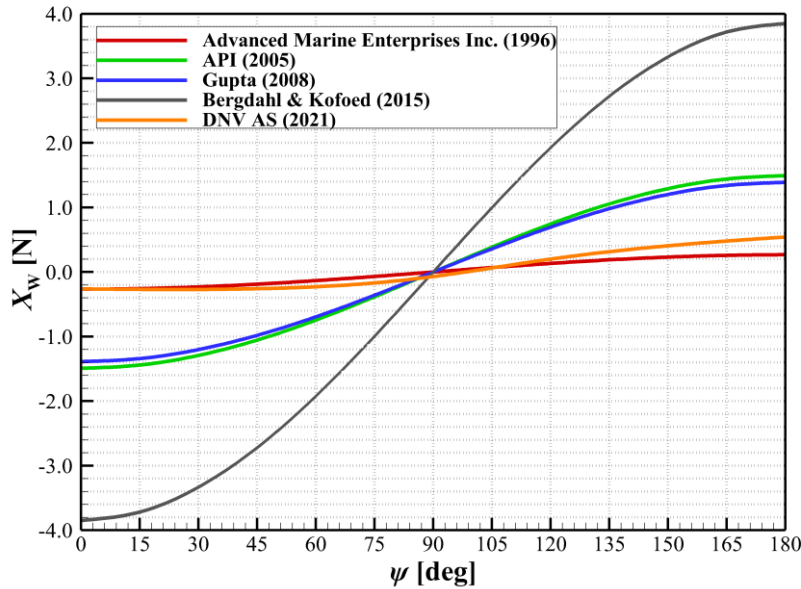


Figure 2.1 : Comparison of empirical/analytical results of mean longitudinal force with respect to heading angle ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

The mean lateral wave drift forces with respect to the heading angle are given in Table 2.2 and Figure 2.2 for the wave condition of $H/\lambda=0.020$ and $\lambda/L_{WL}=1.00$. In beam seas, the mean lateral wave drift forces reach maximum values. As in previous results,

the approximation of Bergdahl & Kofoed (2015) gives higher results. DNV AS (2021), API (2005), and Gupta (2008) give similar results in mean lateral wave drift force.

Table 2.2 : Comparison of lateral wave drift forces, $\bar{Y}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$.

Heading angle, ψ (deg)	Advanced Marine Enterprises Inc. (1996)	API (2005)	Gupta (2008)	Bergdahl & Kofoed (2015)	DNV AS (2021)
0	0.000	0.000	0.000	0.000	0.000
15	0.741	0.386	0.359	0.996	0.408
30	1.432	0.746	0.694	1.924	0.788
45	2.024	1.055	0.981	2.722	1.115
60	2.479	1.292	1.202	3.333	1.365
75	2.766	1.441	1.341	3.718	1.523
90	2.863	1.492	1.388	3.849	1.576
105	2.766	1.441	1.341	3.718	1.523
120	2.479	1.292	1.202	3.333	1.365
135	2.024	1.055	0.981	2.722	1.115
150	1.432	0.746	0.694	1.924	0.788
165	0.741	0.386	0.359	0.996	0.408
180	0.000	0.000	0.000	0.000	0.000

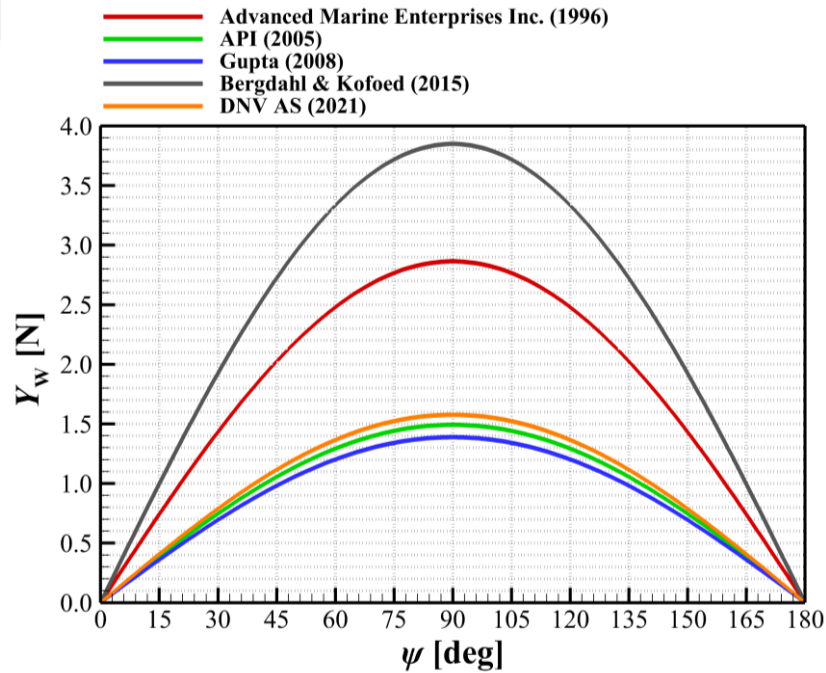


Figure 2.2 : Comparison of empirical/analytical results of mean lateral force with respect to heading angle ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

Table 2.3 and Figure 2.3 presents the results of the yaw moment with respect to the heading angle for the wave condition of $H/\lambda=0.020$ and $\lambda/L_{WL}=1.00$. The symmetrical distribution is not seen in DNV AS (2021) and API (2005). This may be

due to the fact that the center of rotation is in a different position on the hull, both approaches have taken this detail into account in yaw moment formulations.

Table 2.3 : Comparison of yaw moments, $\bar{N}_w$, obtained by empirical/analytical approximations at $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$.

Heading angle, ψ (deg)	Advanced Marine Enterprises Inc. (1996)	API (2005)	DNV AS (2021)
0	0.000	0.000	0.000
15	0.188	0.097	0.062
30	0.312	0.323	0.090
45	0.325	0.527	0.087
60	0.213	0.559	0.055
75	-0.003	0.360	0.005
90	-0.270	0.000	-0.053
105	-0.519	-0.360	-0.108
120	-0.681	-0.559	-0.147
135	-0.708	-0.527	-0.161
150	-0.582	-0.323	-0.143
165	-0.328	-0.097	-0.089
180	0.000	0.000	0.000

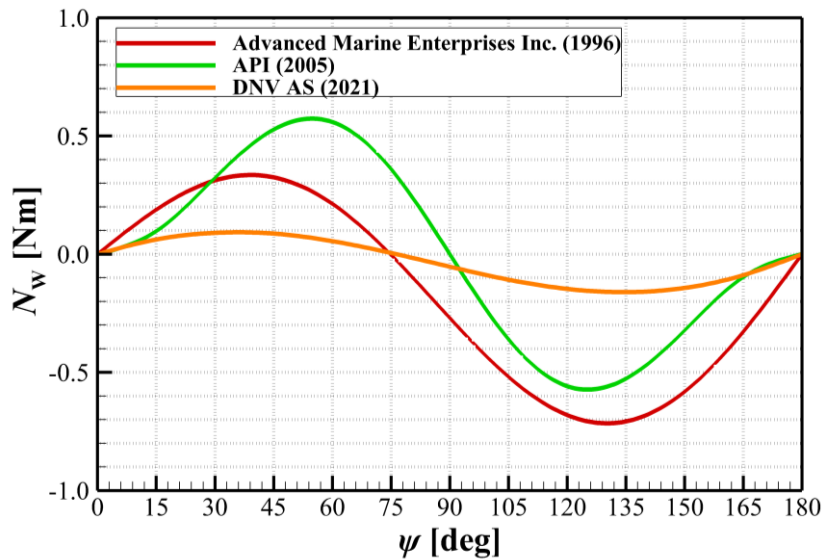


Figure 2.3 : Comparison of empirical/analytical results of mean yaw moment with respect to heading angle ($H/\lambda = 0.020, \lambda/L_{WL} = 1.00$).

The results for the remaining two wave conditions (“ $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ ” and “ $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ ”) are presented in Appendix A. Changing the wave characteristics can have significant effects on wave drift loads, which refer to the forces and moments exerted on the hull due to wave-induced motions. In the second wave condition ($H/\lambda = 0.020, \lambda/L_{WL} = 0.50$), both the wave height and wavelength

are halved compared to the first condition where the wavelength is equal to the ship's length. Reducing wave height and length has decreased the magnitude of wave forces and moment, resulting in lower wave drift loads.

The rate of reduction remained almost constant in both wave drift forces and moments, and the magnitudes decreased by about 60%. In Bergdahl & Kofoed's (2015) approximation, this difference appears to be 75%. As seen in Figure 2.4, a different effect was observed in the DNV AS (2021) results. The reduction in $\bar{X}_W$ was 75%, while the reduction in $\bar{Y}_W$ and $\bar{N}_W$ was 45%.

Considering the situation where they have the same wave height with different wavelength, it can be predicted that the wave drift forces and moment will increase. This is because shorter waves with higher frequencies and shorter periods can lead to rapid and dynamic wave motions. Approximation of DNV AS (2021) also can be examined for better understanding: Zero-up crossing period is used in the calculation of the nondimensional T'_{sway} parameter and the effect of wavelength is also included in the approximation. A decrease in wavelength causes a decrease in the peak period (T_P) and accordingly the zero up-crossing period (T_Z), so the $f(T'_{sway})$ increases in the second case. This normally causes an increase in $\bar{Y}_W$ and $\bar{N}_W$ in case of the same wave height. So, higher frequency waves can induce larger wave drift loads on the hull. However, it can be said that the wave height does not remain the same and has decreased by half, and it is more dominant on the loads than the wavelength. This condition may have caused a 45% reduction in the $\bar{Y}_W$ and $\bar{N}_W$, as seen in Figure 2.4(b) and (c). This effect was not seen in the $\bar{X}_W$, see Figure 2.4 (a), since the $f(T'_{surge})$ does not change with the changing wavelength (due to the condition of $T' \geq 1$).

In the third wave condition ($H/\lambda = 0.025, \lambda/L_{WL} = 1.00$), the increase in wave height was investigated when the wavelength was constant compared to the first condition. A higher wave height leads to larger wave forces and moments acting on the hull. An increase of approximately 60% in wave height was reflected as an increase in forces and moments at similar rates as can be seen in Figure 2.4. The data are presented in detail in Appendix A.

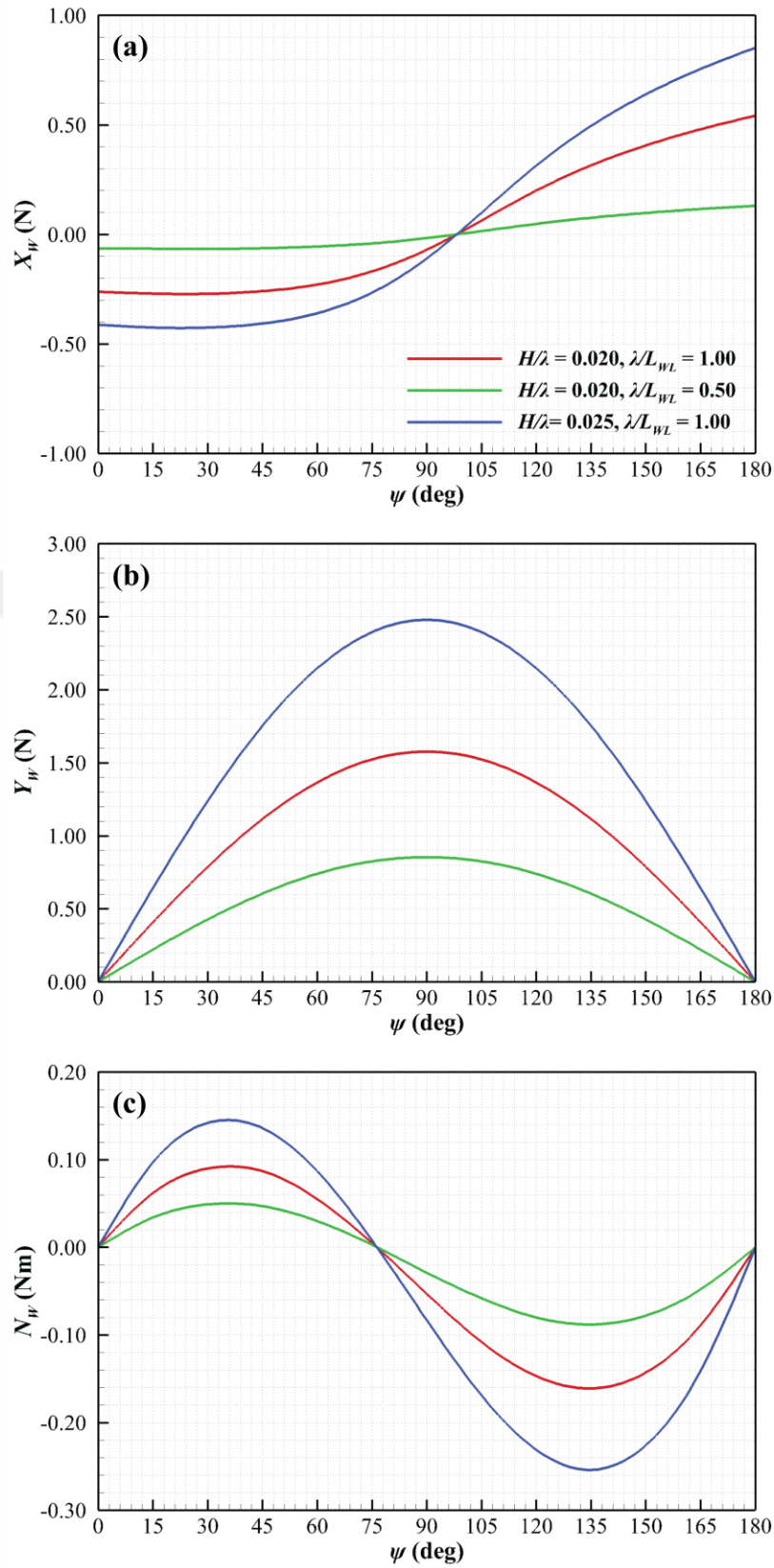


Figure 2.4 : Comparison of different wave conditions' effects on wave drift loads (DNV AS (2021)): (a) longitudinal wave drift force, (b) lateral wave drift force, (c) yaw moment.

3. EXPERIMENTAL STUDY

The benchmark ONRT hull's maneuverability was examined experimentally under a zero-speed condition as well as three different regular wave scenarios with alternative wave/ship length and wave steepness. The ship's three-dimensional (3-DoF) motion was guaranteed by the pantograph during the steady drift tests, which were conducted under various heading angle conditions ranging from 0° to 180° .

3.1 Ata Nutku Ship Model Testing Laboratory

The experimental study is conducted at the Ata Nutku Ship Model Testing Laboratory within the Department of Naval Architecture and Ocean Engineering at Istanbul Technical University. The laboratory includes a towing tank, a circulation channel, a cavitation tunnel, a stability tank, storage, a paint workshop, and a CNC workshop. The sketch of the laboratory is illustrated in Figure 3.1.

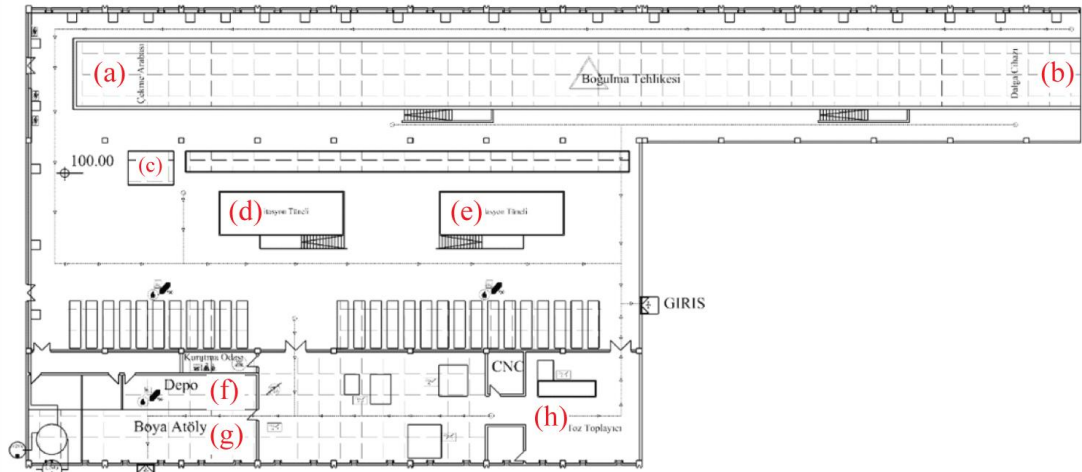


Figure 3.1 : The sketch of the Ata Nutku ship model testing laboratory: (a) towing tank, (b) wave generator, (c) stability tank, (d) cavitation tunnel, (e) circulation channel, (f) storage, (g) paint workshop, (h) CNC workshop.

3.1.1 Towing tank facility

The towing tank has a length of 160 m, a width of 6 m, and a depth of 4.5 m. The maximum speed of the towing carriage is 5.5 m/s. The tank is equipped with a towing

carriage, horizontal planar motion mechanism (PMM), wave generator, and wave-damping beach. Data-acquisition systems and instruments, various dynamometers, and measurement systems to measure forces, moments, ship motions, and carriage speed are located at the towing carriage. There are several types of testing conducted at the laboratory such as ship resistance, self-propulsion tests, seakeeping of ships and floating structures, and maneuvering of ships and submarines.

A 1/48.9 scaled wooden ONRT model is manufactured according to the procedure by ITTC (2017) at the laboratory workshop. The scale ratio for the ONRT hull was intended to equal that of the towing tank employed by the Iowa Institute of Hydraulic Research (IIHR). The model is pulled by towing carriage via PMM pantograph which is used as a attachment point.

3.1.2 Planar motion mechanism

Planar motion mechanisms are used in ship maneuvering experiments to provide accurate and precise control of the ship's motion. This mechanism is composed of a platform, which can be either a circular or a rectangular platform, mounted on the towing carriage. The platform is connected to a motor, which is used to rotate the platform in three directions: pitch, roll, and yaw. The motion of the platform is controlled by a controller, which uses a combination of sensors and software to provide feedback and control the platform's motion. The platform is also equipped with a device to measure the ship's speed and direction. This information is used to help the controller with controlling the platform's motion. By controlling the platform's motion, it is possible to precisely control the ship's motion in the desired direction. This can be used for a variety of experiments, such as ship maneuvering experiments, to study the effects of different variables on the ship's motion.

The ITU-PMM supported by the Presidency of Defence Industries of Turkey was deployed in 2021 at the Ata Nutku Ship Model Testing Laboratory towing tank. The maneuvering tests carried out are static drift, pure sway, pure yaw, and yaw and drift. The length of the test models is constrained to around 3 m as a compromise between these specifications and tank dimensions. The model is coupled to the PMM system via a pantograph, allowing it to work with models of various drafts and to freely 3-DoF motion. The yaw table imposed the yaw motion to the model, and it was made of steel. The synchronous servo motor has 12 Nm nominal torque and 3000 RPM nominal

speed attached to a 1:119 scaled coronex reducer. The yaw table, pantograph, and loadcell can be raised during the assembling process of the model to the carriage. These all parts and the elevator are connected to a steel frame on the towing carriage. There are two linear 40 mm NSK rails across the top and bottom of the 4-meter frame to realize the sway motion. There is a synchronous servo motor to drive the double-side fixed 50 mm NSK ball screw. The synchronous servo motor has 16 Nm nominal torque and a 3000 RPM nominal speed. Table 3.1 lists the specifications of the ITU-PMM and mechanism and its assembled parts are shown in Figure 3.2. The details of the compact motion controller-based PMM installed in ITU for captive maneuvering tests can be found in Özden et al. (2021).

Table 3.1 : PMM design parameters (Özden et al., 2021).

Parameter	Maximum value
Speed	3 m/s
Sway speed	1.2 m/s
Sway acceleration	1 m/s ²
Sway amplitude	±1.25 m
Yaw rate ¹	20 °/s
Yaw angle ¹	20 deg
Static drift angle ¹	35 deg



Figure 3.2 : Horizontal planar motion mechanism at Istanbul Technical University.

There are five quantities describing the horizontal motions as carriage speed, U_c , sway motion amplitude, y_{max} , yaw motion amplitude, ψ_{max} , drift angle, β , and PMM

¹ The values given are valid for cases where the bow of the model is in the $+x$ direction. All angles can be tracked when the model is rotated in z -axis. Therefore, the yaw rate range is $0^\circ/\text{s}$ - $90^\circ/\text{s}$, the yaw angle range is 0° - 360° and the static drift angle range is 0° - 360° .

frequency, ω_{PMM} . The following generic equations of motion may be used to demonstrate how standard static and dynamic PMM tests are performed (Yoon, 2009):

$$\begin{aligned}
y_{PMM} &= -y_{max}\sin\omega_{PMM}t \\
\psi &= -\psi_{max}\cos\omega_{PMM}t + \beta \\
u &= U_c\cos\psi + v_{PMM}\sin\psi \\
\dot{u} &= \dot{v}_{PMM}\sin\psi + r(v_{PMM}\cos\psi - U_c\sin\psi) \\
v &= v_{PMM}\cos\psi + U_c\sin\psi \\
\dot{v} &= \dot{v}_{PMM}\cos\psi - r(U_c\cos\psi + v_{PMM}\sin\psi) \\
r &= r_{PMM} \\
\dot{r} &= \dot{r}_{PMM}
\end{aligned} \tag{3.1}$$

where $v_{PMM} = \dot{y}_{PMM}$, $\dot{v}_{PMM} = \ddot{y}_{PMM}$, $r_{PMM} = \dot{\psi}$, and $\dot{r}_{PMM} = \ddot{\psi}$. The heading angle (or yaw angle) of the model is represented with ψ .

3.1.3 Wave generator

A mono-directional flap-type wave generator aimed at generating both regular and irregular sea conditions. This wave generator was initially produced by Kempf & Remmers in the 1970s. The wave generator was renovated in 2019 and the control system was replaced by a Bosch-Rexroth system. Several wave spectrums are already loaded in the control system and a user-defined wave spectrum can also be generated. In standard seakeeping tests, the following measurements are recorded: significant heave motion amplitude, significant pitch motion amplitude, significant vertical accelerations at several points, mean added wave resistance, and wave height.

The wave generator consists of four main units as shown in Figure 3.3: electrical control panel, hydraulic unit, flap unit, and wave probes. The electrical control panel consists electrical transformer and fuses. It is possible to control the wave generator and the pump that drains the water behind the flap. The device operates when there is energy in the main switchboard. In addition, the control panel includes a computer,

AKAMINA data acquisition system, Tiab hydraulic axis controller, and UPS as can be seen in Figure 3.3(a).

The flap unit is the part of the wave generator that allows waves to form in the tank. The flap is 1350 mm high and is a triangular structure installed on a heel 3150 mm above the bottom in the 4500 mm deep tank. The flap is connected to the piston of the cylinder that gives the motion using a 1.4 m long shaft from the joint. The linear motion of the piston in the cylinder causes the flap to rotate around the joint in the heel and the formation of waves in the tank through this mechanism. The flap unit is shown in Figure 3.3(b).

The hydraulic system indicated in Figure 3.3(c), which is the basic unit of the wave generator, consists of two pumps driven by two 12.5 kW electric motors arranged around an oil tank, a piston giving flap motion, and circuits feeding the piston. One of the two hydraulic circuits is a high-pressure circuit and meets the static pressure resulting from the water level difference on both sides of the flap. This circuit stores the required pressurized oil in an oil tank. The high-pressure oil stored in this oil tank is used to provide static pressure. For this purpose, the pressurized oil circuit coming out of the second oil tank goes to the hydraulic cylinder after it is combined with the pressurized oil circuit coming out of the first tank. The second circuit is the low-pressure circuit and is directly connected to the cylinder that creates the flap motion and operates at high speed.

A total of six wave probes shown in Figure 3.3(d) are used to determine whether the waves are deformed along the width of the tank. The probes are positioned evenly across the width of the tank. The bridge to which the wave probes are located can be carried along the tank.

A wave-damping beach is installed at the beginning of the tank to dampen the transmitted waves. The wave-damping beach is lifted to dampen free-surface disturbances after the carriage run and stays still during the tests. For seakeeping tests, it is expected that the water will settle for about 20 minutes between carriage runs. This time interval is determined sufficient based on visual inspection of the free surface.



Figure 3.3 : The units of wave generator: (a) electrical control panel, (b) flap unit, (c) hydraulic unit, (d) wave probes and gauges. (1) computer, (2) Akamina – WaveGen Studio, (3) Tiab – Play waveform, (4) UPS, (5) coolant pump, (6) centrifuge pump.

The wave generator system utilizes the following software and functions: Rexroth – OPT-Computation, Tiab – Play Wave Form, Rota Teknik – HPU Monitor, and AKAMINA – WaveGen Studio (Akamina Technologies Inc., 2016).

Rexroth – OPT-Computation is the software where the desired wave information is entered and produces the wave function. The user can specify basin properties, and wave details such as spectrum type, period, and height. In the experiments, the regular wave is produced by using this software. In the spectrum tab, the user can select the regular wave and the formula for a regular wave:

$$x(t) = a \cos(2\pi f t + \phi) \quad (3.2)$$

where t is the time, f is the wave frequency, a is the wave amplitude, and ϕ is the phase.

The program requests the wave height and wave period for a regular wave. In addition, the frequency value will be computed but is displayed for information only. Wavelength, based on period, depth, and wave height, is displayed for information. The steepness is computed as wave height/wavelength and is displayed for information. The interface of Rexroth – OPT-Computation is illustrated in Figure 3.4.

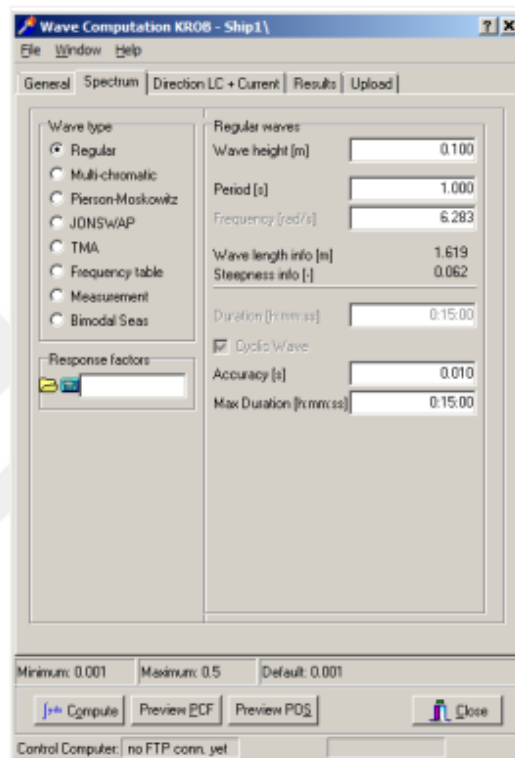


Figure 3.4 : Interface of Rexroth – OPT-Computation.

Tiab – Play Wave Form is the software that transfers the wave function to the flap through the hydraulic system and enables the hydraulic power unit to be operated. The interface of Tiab – Play Wave Form is shown in Figure 3.5. With the *file* button in the upper left part, the created wave file can be selected. With the option below, the number of times the wave will be generated is entered. In the middle graph, the flap angles are shown corresponding to the wave properties. Errors and codes, if any, appear in the box in the lower right corner of the program.

The pumps are started first by pressing the *pump run* button and with the *homing* button, the flap is brought to the 0-degree position. It is possible to operate the flap with the *play* button. If another wave is to be produced, the new file can be imported

after pressing the *pause* button. If the experiment is over, it is important to initialize the flap with the rest position on the right of the screen after pressing *stop*.



Figure 3.5 : Interface of Tiab – Play Wave Form.

Rota Teknik – HPU Monitor shown in Figure 3.6 is the software developed by Rota Teknik Machine Industry and Trade Inc. (Rota Teknik Mak. San. ve Tic. A.Ş.) that regulates the operating parameters of the hydraulic power unit and displays the sensor outputs. While the hydraulic power unit is operating, it enables to monitor the current status, such as oil temperature, oil level, and main pressure.

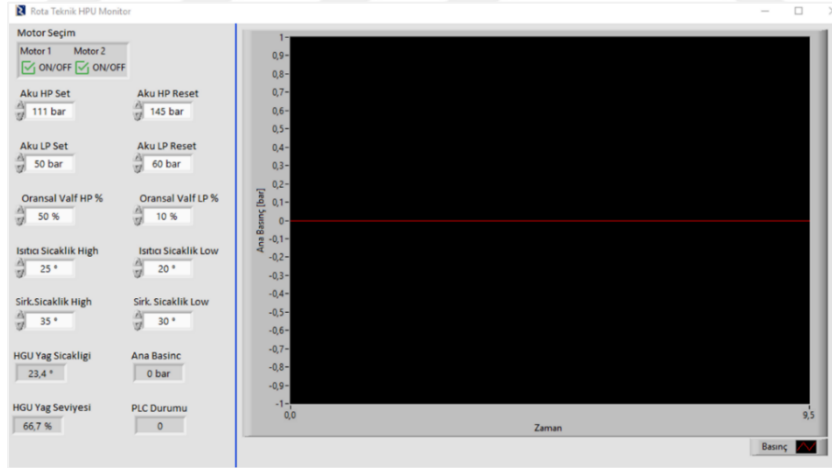


Figure 3.6 : Interface of Rota Teknik – HPU Monitor.

AKAMINA WaveGen – Studio is a system designed for managing physical hydrodynamic studies. It combines the four elements of data acquisition, wave synthesis, wave generation, and wave data analysis in a graphical interface designed to allow users to easily manage the seakeeping experiments. The sensor calibration process is performed through AKAMINA – WaveGen Studio. Calibration is performed in still water conditions just before the experiment starts. Time-dependent

angular moments acting on the ship are K , M , and N in roll, φ , pitch, θ , and yaw, ψ , directions, respectively.

The velocities of the model in the ship-fixed coordinate system's x and y directions are u and v , respectively. The advance speed, U , can be derived from u and v as follows:

$$U = \sqrt{u^2 + v^2} \quad (3.3)$$

The orientation of the model with regard to the advance speed is known as drift angle, β , and can be expressed as

$$\beta = -\arctan(v/u) \quad (3.4)$$

The heading angle of a ship, ψ , refers to the direction in which the ship's bow is pointing relative to a reference direction, x_0 . The lower right drawing represents the incoming wave with the wavelength, λ , and incident wave angle, χ .

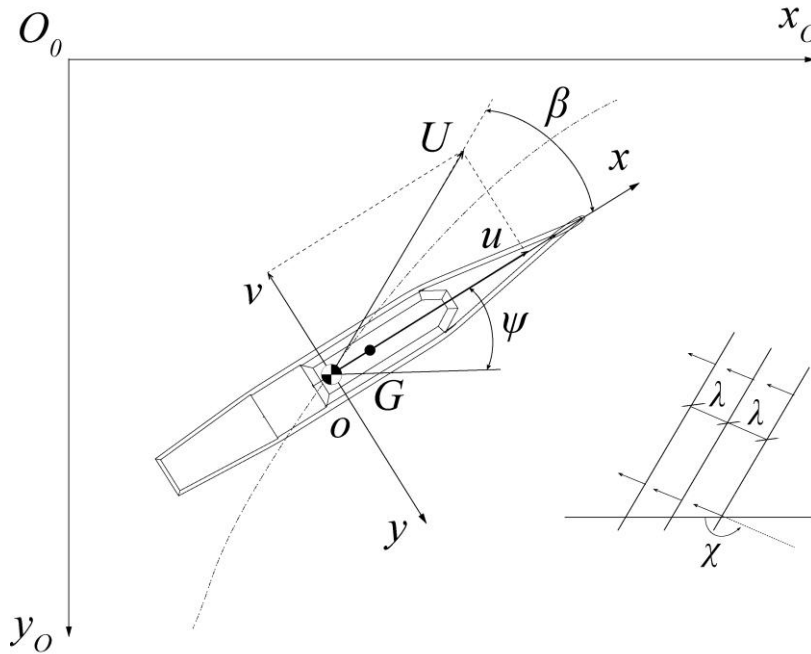


Figure 3.8 : The earth-fixed and ship-fixed coordinate system.

The experiments of static drift, pure sway, pure yaw, yaw, and drift motions can be performed by using ITU-PMM. There are five quantities used in the definition of motions as carriage speed (U_c), sway motion amplitude (y_{max}), yaw motion amplitude (ψ_{max}), heading angle (ψ), and frequency (ω_{PMM}). The only quantity changed in

steady drift tests is the heading angle. The heading angle entered in the PMM mechanism causes the model to rotate clockwise. The definition of the coordinate system including incident wave angle, χ , is illustrated in Figure 3.9. The 180° incident wave angle ($\chi = 180^\circ$) shown in the illustration corresponds to the heading angle of $\psi = 0^\circ$. In general, it can be said that there is a 180° difference between the incident wave angle and the heading angle for steady drift tests. Also, rotating the model more than 90° from the initial position may be hazardous for the system due to the wiring of the load cell. Therefore, for the cases of $90^\circ < \psi < 180^\circ$ the model bow points to the $-x_0$ -direction.

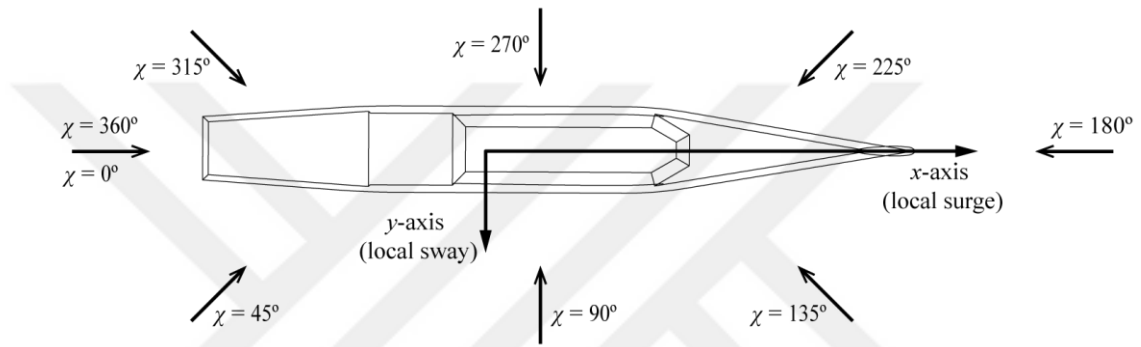


Figure 3.9 : Incident wave angle, χ .

3.2 ONRT Model

The scaled generic model of ONRT (Office of Naval Research Tumblehome) is chosen to perform the experimental studies under prespecified wave conditions and heading angles. The ONR Tumblehome hull 5613 is a preliminary design for the DDG1000 Zumwalt class destroyer used by the United States Navy in the early 2000s. Tumblehome is a type of hull design in which the sides of the ship curve inwards toward the top of the vessel. It is also used to reduce the width of a vessel, making it easier to maneuver through narrow channels or shallow waters. The ONR Tumblehome design features a steeply-angled bow, a sharp entry angle, and a narrower beam. It is a modern warship featuring both wave-breaker form and a 10-degree contraction of the ship's hull over the free surface. This design reduces the size of the ship's profile, making it more difficult to detect and target. The ONR Tumblehome design also improves the ship's hydrodynamic properties, making it more fuel efficient and allowing it to move faster.

Design parameters of the 1/48.9 scaled model of ONRT defined for use as benchmark cases in the Tokyo 2015 CFD workshop (*Tokyo 2015 A Workshop on CFD in Ship Hydrodynamics*, 2015) are presented in Table 3.2. The basic views of the test model are displayed in Figure 3.10. Note that, the superstructure shown in the figures is not get involved in the experiments.

Table 3.2 : Main particulars of ONRT hull.

Parameter	Symbol [units]	Ship	Model
Scale	Λ [–]	1:1	1:48.9
Length on waterline	L_{WL} [m]	154.0	3.147
Maximum beam of waterline	B_{WL} [m]	18.78	0.384
Draft	d [m]	5.494	0.112
Wetted surface area (fully appended)	S [m ²]	N/A	1.345
Displacement	Δ [ton and kg]	8507	72.6
Block coefficient	C_B [–]	0.535	0.535
Longitudinal center of buoyancy (aft of FP)	LCB [m]	N/A	1.625
Vertical center of gravity (from keel)	$\overline{KG}$ [m]	N/A	0.156
Metacentric height	$\overline{GM}$ [m]	N/A	0.042 2
Moment of inertia	K_{xx}/B [–]	0.444	0.444
	K_{yy}/L_{WL}	0.246	0.246
	K_{zz}/L_{WL} [–]		



Figure 3.10 : 1/48.9 scaled ONRT model.

The ship's design includes rudders, shafts and shaft brackets, propellers, skeg, and bilge keels. In the model ship in the present study, there are rudders, shafts and shaft brackets, skeg, and bilge keels. This model without propellers is called fully appended. Turbulence stimulation was obtained by the application of studs. The cylindrical studs with 3 mm height and 2 mm diameter are placed in the 19th section of the model. The stud dimensions and locations conform with the 28th ITTC standards (ITTC, 2017b).

3.3 Model Ballasting and Mounting

A calm water roll decay test can be used to estimate the model's roll gyradius in water as an alternative to measuring it in air. The metacentric height and the roll period of the model, which were discovered during the roll decay test, can be used to directly calculate the roll gyradius. Prior to the roll decay test, an in-water inclining test can be used to determine the metacentric height.

The center of buoyancy ($\overline{KB}$) and the metacentric radius ($\overline{BM}$) were calculated over the CAD program by using the three-dimensional model. The center of gravity ($\overline{KG}$) and metacentric height ($\overline{GM}$) values of the model are already known from the webpage of the international workshop (*Wageningen 2023: A Workshop on CFD in Ship Hydrodynamics*, 2023). The hydrostatic values of the ONRT model are given in Table 3.3.

Table 3.3 : The hydrostatic values of ONRT hull.

$\overline{KB}$ (m)	$\overline{BM}$ (m)	$\overline{GM}$ (m)	$\overline{KG}$ (m)
0.067	0.133	0.0443	0.156

The inclining test in calm water is used to experimentally obtain the $\overline{GM}$ value. The ballasting weights, which are sure to heel the model at small angles, are placed at the longitudinal center of buoyancy (LCB) point at the the port side and starboard side. A total of three weights, one 1 kg, and two 0.5 kg, were placed on the port side of the model ship. Three ballasting weights are equally placed on the starboard side, in line with the weights on the port side. The distance, d , between the centers of the ballasting weights is measured. The inclination angle is measured by transferring the ballasting weights one by one from the port side to the starboard side and vice versa at each step. When the inclining test is completed in both directions, the $\overline{GM}$ is calculated using the following formula:

$$\overline{GM} = \frac{w \times d}{\Delta \tan \theta} \quad (3.5)$$

where w is the displaced weight in kg, d is the distance between weights in m, Δ is displacement in kg, and θ is the inclination angle in degree. The experimental $\overline{KG}$ value is calculated with the following formula by using the $\overline{KB}$ and $\overline{BM}$ values found geometrically and the average $\overline{GM}$ value obtained experimentally:

$$\overline{GM} = \overline{KB} + \overline{BM} - \overline{KG} \quad (3.6)$$

The difference between the experimental $\overline{KG}$ value and the original $\overline{KG}$ value is found. This value is called $GG1$. The amount of weight that needs to be shifted to reach the original $\overline{KG}$ value and the vertical distance are found iteratively with the following formula:

$$GG1 = \frac{\Delta w \times \Delta d}{\Delta} \quad (3.7)$$

where Δw is the weight to be shifted vertically and Δd is the shifting distance. The Δw and Δd values are changed until the $GG1$ value is equal to the $GG1$ value which represents the difference between the $\overline{KG}$ s.

Experimental measurements of the model's inclining motion can be made with a variety of sensors and setup arrangements. The POSITAL three-dimensional inclinometer, see Figure 3.11, available in the test basin is currently used in inclining tests. The AGS030-2-SC1-H0-P8M inclinometer by POSITAL, based on electrolytic fluid technology, is used to measure the deviation angle in the range of $\pm 30^\circ$ around two axes. The main feature of this sensor is high resolution (0.001°) and accuracy (0.01°) of measurements.

In the previous study (Kinaci et al., 2021), the free-running tests of the DTC container ship model were conducted in the towing tank for the self-propulsion case. The guides and a brake used in the experiments did not limit the motion in the surge direction. In the study presented in the thesis, it was necessary to change the mounting system since a study was carried out independently of the forward speed. Therefore, the test model is connected to the load cell via a pantograph as can be shown in Figure 3.12, which allows heave, pitch, and roll motions. The pantograph also provides the opportunity to

adjust the draft of the model. The shafts of this pantograph are made of AISI 316 L grade stainless steel and it is manufactured with precision utilizing Computer Numerical Control (CNC) from aircraft grade 7075 aluminum alloy. When positioning the pantograph in the midship position and attaching the model to the PMM, a laser liner is used to make sure the initial angle of attack is zero.



Figure 3.11 : Inclinometer.



Figure 3.12 : Connection of loadcell and pantograph.

3.4 Test Conditions

Within the scope of the study, maneuverability analysis of the benchmark ONRT hull was experimentally performed in a zero-speed and three different regular wave conditions with various wave/ship length and wave steepness values. While the wave steepness ratios (H/λ) were selected as 0.020 and 0.025, the wave/ship length ratios (λ/L_{WL}) were chosen as 0.50 and 1.00. Under these conditions, three regular waves with different characteristics as shown in Table 3.4 were produced by flap type wave generator. The damping was provided by the presence of a fixed wave-damping beach located at the beginning of the towing tank.

Table 3.4 : The wave characteristics in static drift tests.

Case	Wave steepness	Wave/Ship length	Wave height	Wavelength	Wave period	Wave frequency
-	H/λ	λ/L_{WL}	H	λ	T	f
-	[-]	[-]	[m]	[m]	[s]	[Hz]
1	0.020	1.00	0.063	3.147	1.420	0.704
2	0.020	0.50	0.031	1.574	1.004	0.996
3	0.025	1.00	0.079	3.147	1.420	0.704

The steady drift tests were carried out in different heading angles, ψ , ranging from 0° to 180° , and the 3-DoF motion of the ship was ensured by pantograph. The model is free to heave, pitch, and roll, with zero speed, without a propeller. The test program given in Table 3.5 is intended for comparisons in turning circles.

Table 3.5 : PMM test conditions.

Test	ψ [deg]
Steady drift	0, 15, 30, 45, 60, 75, 90, 105, 120, 135, 150, 165, 180

3.5 Measurement Systems and Calibration Procedures

A measurement system consists of several components, including the measuring instrument, the signal conditioning system, the data acquisition system, and the output display or recording system. Calibration is the process of verifying and adjusting the accuracy and precision of a measuring instrument. The present section describes the systems used to measure the force, moment, and wave parameters. Following this, the calibration procedures of the systems will be mentioned.

3.5.1 Loadcell

In steady drift tests, data acquisition was carried out with a multi-component load cell, which is a part of the PMM. The capacity of the loadcell is 500 N, 500 N, and 500 Nm for longitudinal force, F_x , lateral force, F_y , and yaw moment, M_z , respectively. A six-component Izumi Sokki six-component strain-gage type loadcell with a six-channel Izumi Sokki DA-18K amplifier and a calibration box is used in maneuvering experiments. The technical drawing of the load cell is shown in Figure 3.13.

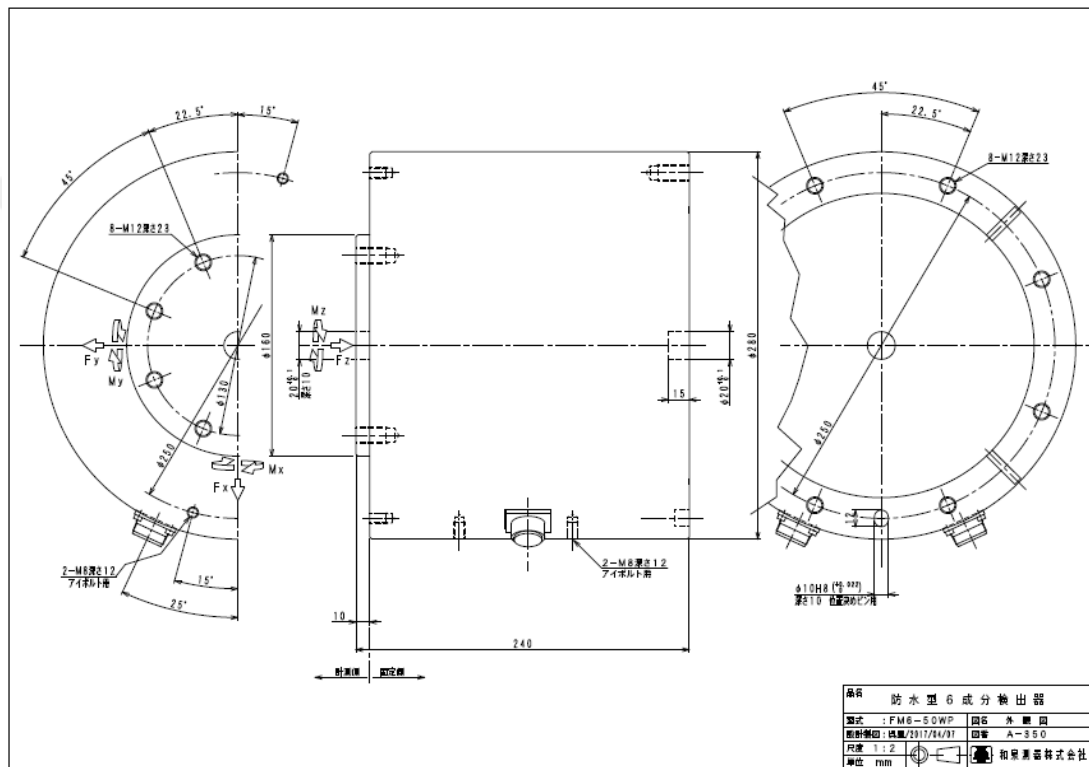


Figure 3.13 : Technical drawing of the load cell.

The calibration procedure for loadcell includes gradual loading and the examination of the output signals at each stage. Also, the weight indicator must revert to zero when the loads are removed to verifying the loadcell system is functioning effectively. A calibration mechanism was established for load cell calibration. With this setup, the load cell's $\pm F_x$, $\pm F_y$ and $\pm F_z$ forces and $\pm M_x$, $\pm M_y$ and $\pm M_z$ moment calibration were performed with the modification of the moment arm. The amplifier shown in Figure 3.14 is connected to calibration boxes for forces and moments, which check the calibration with 250 N force and 250 Nm moment signal.

Calibration boxes are used to check the loadcell calibration before the experiment. After energizing the amplifiers with force and moment components to be used by

pressing the green button, the amplifier is reset with the auto zero button. When the calibration boxes are opened, the force of 250 N and the moment of 250 Nm must be given by the amplifier. If the values in the display are correct, no adjustment is required. The thesis by Özden (2021) can be examined for the details of the calibration procedure.



Figure 3.14 : Amplifier and calibration boxes.

3.5.2 Wave height gauge and probes

A total of six wave height gauges and probes are used to measure the generated wave properties. The probes are positioned evenly across the width of the tank. The AWP-300-3 capacitance-type wave height gauge from Akamina Technologies as can be seen in Figure 3.15(a) is suited for wave height measurements. The digital interface eliminates many issues common in analog data acquisition systems, such as noise and grounding issues over long cables. The interface simplifies the cabling of six wave gauges connected in a daisy-chain fashion. The sensors are highly linear (0.15% over the measurement range) and have a fast response time where the output is updated every 5 ms. The sampling rate of the wave height gauges is 100 Hz.

The AKAMINA wave probe head as can be seen in Figure 3.15(b) is designed to work with the AWP-300 series of wave height gauge electronics. The probe head is part of the wave height measurement system that is partially submerged in water to measure wave heights or water levels. The heads have no electrical interference with other probes or instrumentation. The length of the head is 75 cm. The support rod is manufactured from stainless steel. Also, support and sensor wire clamps are manufactured from hard anodized aluminum.

Calibration is the initial step before proceeding with real measurements. This is required to determine the link between water depth and the AWP-300-3 output

measurement. The instructions for calibration are proposed by AKAMINA Technologies (Akamina Technologies Inc., 2014). The wave probe measurement system is calibrated statically end to end by moving the probe incrementally on a linear, vertical slide and making measurements on a computer at each step. The data are fit using linear regression, the slope of which is used as a scale factor for wave elevation measurements. The form of a linear calibration function should be used in three-point calibration. The calibration procedure should be performed in calm water conditions. Firstly, the water depth is measured at the first calibration point. The other two calibration points are determined by lowering/raising the wave height probe and the same procedure is followed. After the calibration procedure is completed, the wave height gauges are mounted on the bridge from mounting points.

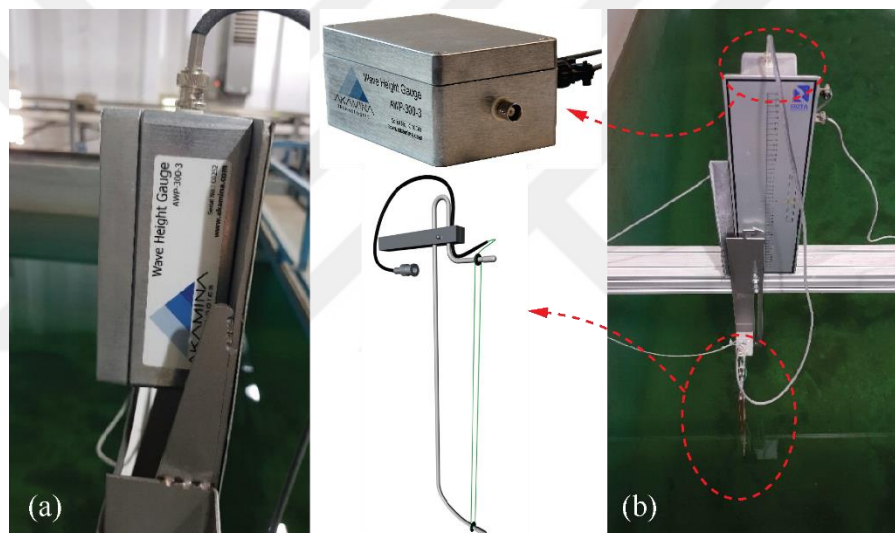


Figure 3.15 : Wave measurement units: (a) Wave height gauge, (b) Wave height gauge, and a wave probe head.

For acquisition, each input channel must be calibrated with a re-zero curve before the experiment begins (Akamina Technologies Inc., 2016). The reference point must be taken for the wave probes at calm water conditions. Any channels in a shortlist that are selected are calibrated at the same time and have the same reference values. Therefore, three channels at a far distance are calibrated together, while three channels at a close distance are calibrated together. When the calibration process is started, the data required for the calibration starts to be collected under calm water conditions. Summary statistics for the obtained data are examined and the calibration is completed by entering the reference value to be associated with the data point.

3.6 Data Acquisition Procedures

Data acquisition and reduction are two key steps in the process of collecting and analyzing data. Data acquisition is the process of collecting, recording and storing data from a variety of sources. This section will describe the data acquisition methodologies for forces, moment, and wave data followed in the experiments.

3.6.1 Data acquisition of forces and moment

Before the model is attached to the pantograph, the in-water inclining test is performed as described in chapter 3.3. The weights are fixed after ensuring the weight distribution to which the real metacentric height is provided. The model is mounted to the pantograph and connected to the load cell so that the F_x , F_y , and M_z values can be collected. The force and moment data transmission cables coming from the load cell are connected to the calibration boxes shown in Figure 3.16(a). The force and moment components separated in the calibration boxes are connected to the amplifier as can be seen in Figure 3.16(b). Components converted to ± 10 Volts of Direct Current (VDC) range in the amplifier are output via Bayonet Neill–Concelman (BNC) cables and the other ends of the cables are connected to the panel displayed in Figure 3.16(c). The grounding output is taken from the amplifier via a cable and the other end of the cable is connected to the grounding point on the panel.



Figure 3.16 : Electrical equipment installation: (a) Calibration boxes, (b) Separation of the force and moment components, (c) Control panel.

The data transfer to the panel is provided via Human Machine Interface (HMI) shown in Figure 3.17(a). The connection of the computer with the panel is provided via an ethernet cable. The model is towed towards the end of the 160-meter-long tank, positioned close to the wave probes, via the towing carriage. The aim here is to stay close to the wave generator and make sure that the generated wave meets the model without absorption.

When the start button on the HMI panel is pressed, preparations for a new experiment are made: The conditions under which the experiment will be performed are entered via the HMI. In this study, the static drift tab is used and the β value is changed in each set. Since the steady drift test is performed, that is, the model velocity is zero, a speed input is not required. The interface of the static drift test in HMI is shown in Figure 3.17(b).

After all connections are made, the project is started by running the SIMATIC WinCC Explorer program on the computer as shown in Figure 3.17(c). When the experiment starts, the data acquisition of force and moment values begins. At the end of the experiment, the collected data is saved. Before starting a new experiment, the model is returned to the zero position and these steps are followed for each set.

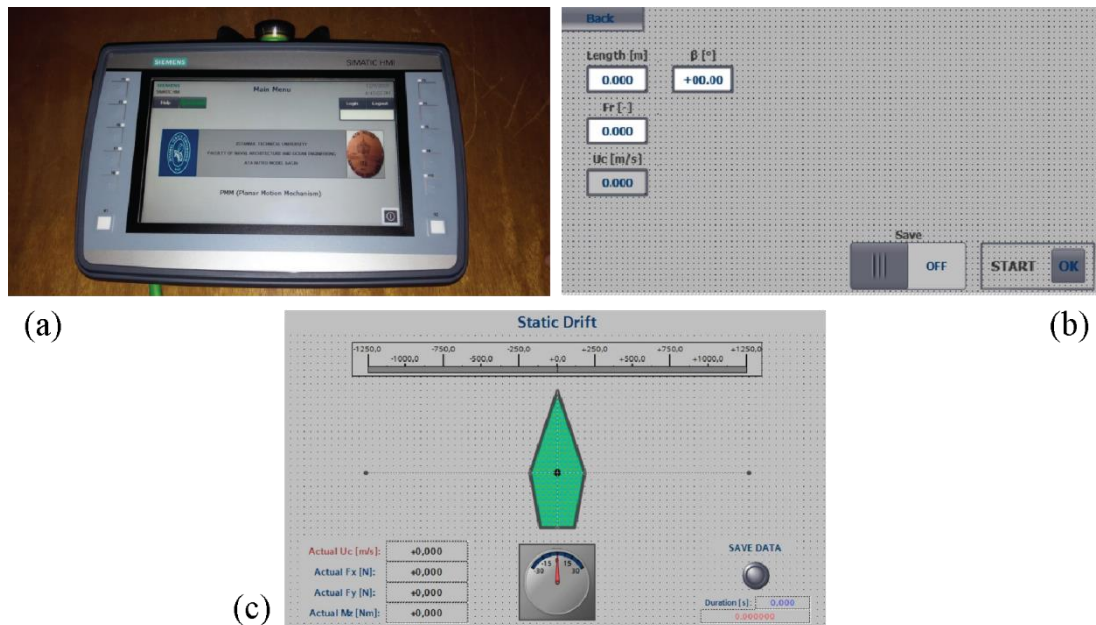


Figure 3.17 : (a) Human Machine Interface, (b) The static drift test interface at HMI, (c) The static drift test interface at SIMATIC WinCC Explorer.

3.6.2 Data acquisition of wave parameters

The power in the main electrical panel of the wave generator is confirmed. The emergency stop button is released by turning it counterclockwise. The emergency stop reset button is pressed for the discharge pump to work. Before the experiment starts, the water behind the flap must be discharged. This is because the water displacement adversely affects the flap's motion during the wave generation.

The computer and the AKAMINA data acquisition system in the electrical control panel are turned on. After the screens in the control cabinet are opened the Tiab – Play Wave Form, the Rexroth – OPT-Computation, the Rota Teknik – HPU Monitor, and the AKAMINA – WaveGen Studio are respectively opened.

While the wave generator was being installed, the current water level was measured and the system inputs were determined according to this value. There is a correction factor to set the decreasing/increasing water level due to water evaporation, leakage, or water refill. The water level is measured before each set of experiments and the difference between the initial water level is taken. The correction factor of ± 0.15 m is entered in the water level section of the wave file created by the Rexroth – OPT-Computation program. Within the scope of the study, only the effect of regular waves was examined: The regular wave type is selected in the spectrum tab. The information on the wave height and the period is entered and the wave file is completed.

With Rota Teknik – HPU Monitor (HPU refers to Hydraulic Power Unit), the data of the hydraulic power unit is displayed and the data is followed throughout the experiment. It is connected to the Tiab – Play Wave Form screen via this interface. By selecting the configuration file path, the generated wave file is imported. Number of replays is entered. The angle of the flap is shown on the screen. After the system is reset, the pump is started, the system is pressurized, and the flap is brought to the 0° position via the homing tab.

After the connection status is active in the data acquisition program named AKAMINA – WaveGen Studio, sensors are calibrated. The water must be calm, so calibration must be done before the towing carriage moves. As described in chapter 3.5.2, a total of six sensors are grouped in threes. During the calibration process, the sensors are selected as a group of three due to differences in location, and the current height of the water in towing tank is defined by the system. Then, a chart is created for each sensor group

via the strip chart tab to instantly monitor the time-dependent change in water level. It is possible to perform the data acquisition process during the experiment with the acquire tab.

After the preparation of the wave generator and the preparations on the towing carriage are completed, the wave generation is started by pressing the play tab in Tiab – Play Wave Form screen. The run tab is clicked on the data acquisition server screen to collect the water level data.

Data collection is terminated when the experiment is complete. On the Tiab – Play Wave Form screen, the flap is brought back to its initial position with the rest position tab. Pumps are stopped. In case of an emergency, the emergency stop button is pressed.

The wave height data is exported in the AKAMINA – WaveGen Studio. To analyze regular waves, the extract tab under the analyze tab is selected. The aim here is to select a suitable interval for analysis by specifying the start time and end time in the time-dependent signal. The selected part of the wave signal is saved and the Root Mean Square (RMS) value of the generated wave is noted by passing to the compute tab. If the RMS value is different from the desired wave height, the experiment is repeated by correcting the water level.

Test outputs are reported in accordance with the data acquisition and reduction methodology and the next test setup is prepared. The measured RMS, minimum, maximum, and mean values appear as a result of the generated regular wave. When the entire test matrix is complete, all the above-mentioned programs are closed. The computer, data acquisition system, and Uninterruptible Power Supply (UPS) on the control panel are turned off. The emergency stop button is pressed therefore hydraulic system and water discharge pump are stopped. The main switch on the control panel is turned from 1 to 0. The control panel and control room are locked.

3.7 Data Reduction Procedures

The procedures for data acquisition and reduction are important steps in the data analysis process, as they can help to improve data accuracy, reduce storage costs, and improve the efficiency of the overall data analysis process. Data reduction is the process of removing or combining data points to make the data set easier to analyze.

In this section, how the force, moment, and wave parameters are analyzed will be explained.

3.7.1 Data reduction of forces and moment

The data reduction of forces and moments (X, Y, N) for steady drift PMM tests are given in the present section. Throughout each carriage runs, all forces and moments are obtained as time histories. It should be noted that, the towing carriage velocity, U_c , and thus model velocity (U) are equal to zero in steady drift experiments.

The time-dependent wave quantities from the time series of measurements are analyzed employing a fast Fourier transform (FFT) algorithm. The FFT algorithm efficiently computes the discrete Fourier transform (DFT) of a signal or data sequence, enabling the decomposition of a signal into its frequency components. By representing a signal in the frequency domain, it becomes easier to analyze and manipulate different frequency components separately. In the scope of this study, the FFT algorithm was used to find the values corresponding to the 0th frequency. These values corresponding to the 0th frequency give the mean wave drift loads of the investigated ship heading and wave condition.

In MATLAB, the *fft* function is used to compute the FFT of a signal or data sequence. The syntax for the *fft* function is as $Y = \text{fft}(X)$ where X represents the input signal, and Y represents the resulting FFT. It is necessary to multiply the resulting signal by two because of the conversion from a two-sided spectrum to a one-sided spectrum. The signal frequency amplitude at a two-sided spectrum's positive and negative frequencies is symmetrical and equal on both sides. However, the first and last values appear once in the spectrum and already have the real amplitude. Therefore, the first value (mean of the signal) corresponding to the 0th frequency and the last value are not doubled.

At first, the tests are conducted over a 20-second time interval. During the FFT analysis in this scenario, oscillations occurred as the frequency approaches zero, which is the point at which the mean wave drift values are determined. To address this issue, the measurement time interval is extended to 40 seconds in the second round of the test series. This extension enhances the quality of FFT results as the frequency approaches zero.

Although regular waves were produced during the experimental study, the wave started to dissipate after 40 seconds due to the limited size of the towing tank. In addition, the wave reflected from the side walls of the towing tank causes the main wave to dissipate. For this reason, it is not possible to create a perfectly regular wave. From a signal processing perspective, the chances of obtaining a waveform with a number of points equal to the 2^n number and ending in an integer number of periods are low. If data longer than a regular wave can be collected and the input signal is optimally acquired, it can be ensured that the FFT analysis provides an accurate representation and a peak at 0th mode (zero Hertz) can be directly used for wave drift loads. In general, a peak at 0th mode represents the DC bias of the signal, in other means the average of the signal. Consequently, an approximation is made for the correct determination of the mean wave drift loads: After computing the FFT, the resulting frequency domain representation is used to perform another signal processing operation, the Savitzky-Golay filtering.

Savitzky-Golay filtering is a digital signal-processing technique used for smoothing or filtering noisy data. This is a linear filter that operates on a moving window of data points. It approximates the data within the window using a polynomial curve and computes the filtered output as the value of the polynomial at the central point of the window. By adjusting the size of the window and the degree of the polynomial, the filter can be adapted to achieve different levels of smoothing or noise reduction.

In MATLAB, the *sgolayfilt* function is used to apply the Savitzky-Golay filter to a signal or data sequence. The syntax for the *sgolayfilt* function is as $y = \text{sgolayfilt}(x, \text{order}, \text{framelen})$ where x represents the input signal, *order* is the degree of the polynomial used for fitting the data, and *framelen* specifies the length of the moving window. The *sgolayfilt* function applies the Savitzky-Golay filter to the input data and returns the filtered output as y .

Before the application of filtering, the single-sided amplitude spectrum is cut off before it reached the peak value. By adjusting the *order* and *framelen* parameters, the trade-off between smoothing and preserving signal features is controlled. A higher *order* value results in a more flexible polynomial fit, which can preserve sharper features but may introduce more noise. A larger *framelen* captures more local information but can lead to over-smoothing. Each FFT result was examined one by one and filtered with appropriate parameters. Figure 3.18 is presented to be an example

of a filtered signal. The sample signal is the time histories of lateral force at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ where $\psi = 90^\circ$. After the time domain representation of the signal is deconstructed into the frequency domain with the FFT algorithm, the range of 0-0.4 Hz is selected to apply Savitzky-Golay filtering. The *sgolayfilt* function is used to apply Savitzky-Golay filtering with a polynomial order of 3 and a window length of 33. The original and filtered signals are plotted to visualize the noise reduction achieved by the filtering process.

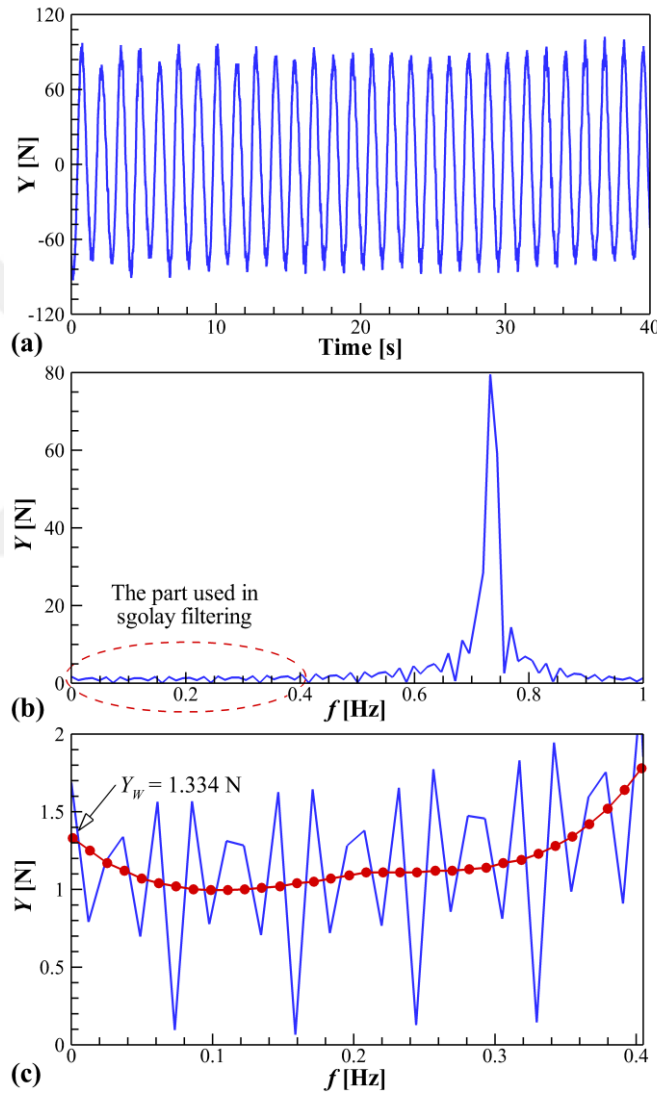


Figure 3.18: The steps of obtaining the wave drift load: (a) The force signal, (b) The processed signal using the FFT algorithm, (c) The obtaining wave drift load by using Savitzky-Golay filtering.

Consequently, the experimental analysis was conducted using the numerical evaluation and the same procedure used in computational results. The steps followed are listed below:

- A certain time is processed by editing the time series data file of X , Y forces, and N moment.
- Depending on the measurement frequency of the load cell, data are sampled at a rate of 200 Hz. In other words, the sampling rate $\Delta t = 0.005$ is fixed in the experimental analysis.
- The sampling period of the signal was calculated using FFT. The period of forces and moment in the measurement recordings, T , is identified from $T = 1/f$.
- The MATLAB function computed the DFT of the processed signal using an FFT algorithm.
- The single-sided spectrum based on the two-sided spectrum and the even-valued signal length was computed.
- The frequency domain, f , in Hz, was defined and the single-sided amplitude spectrum was plotted.
- The MATLAB function applied a Savitzky-Golay finite impulse response (FIR) smoothing filter of polynomial order and frame length to the data in the spectrum.
- The filtered signal was plotted and the values corresponding to the 0th frequency were recorded as the mean wave drift loads of the investigated ship heading and wave condition.

The MATLAB code for FFT analysis and smoothing process is presented in Appendix B.

3.7.2 Data reduction of wave parameters

The wave parameters, wave height, H , wavelength, λ , and wave frequency, f , are derived from the data acquisition. The sampled data is collected and stored during the data acquisition phase and can be exported by the user. It is possible to select a range of data samples from a file based on time which allows the user to remove distortions during the experiment.

The RMS and peak-to-peak values are two different ways of expressing the magnitude of a varying waveform. The RMS value of a waveform is a measure of its average

value. It represents the square root of the average of the squares of all the instantaneous values of the waveform over a given period. On the other hand, the peak-to-peak value of a waveform represents the difference between its maximum and minimum values over a given period. The root mean squared value of the wave height, H_{rms} , is obtained from the collected data in mm. The wave height is calculated by passing from this value to the peak-to-peak value with the following formula:

$$H_{rms} = \frac{1}{2\sqrt{2}} H_{pp} \quad (3.8)$$

The dominant frequency in Hz is read from the FFT result of the time-dependent signal. By using this frequency value, the wave period is passed with $1/f$. Finally, the wavelength is calculated using the formula below:

$$\lambda = \frac{gT^2}{2\pi} \quad (3.9)$$

3.8 Experimental Results

The subsequent chapters present the findings and analyses of the experimental study conducted to investigate the effects of waves on wave drift loads. The implications of these findings will be discussed in light of the comparison with empirical/analytical results.

3.8.1 Mean wave drift loads

Steady drift tests were performed in regular waves at zero Froude number under the conditions of $H/\lambda = 0.020, 0.025$, and $\lambda/L_{WL} = 0.50, 1.00$. The wave scenarios cover head, beam, following, and quartering tests. Time histories of longitudinal force, X , lateral force, Y , and yaw moment, N were collected. These time histories and FFT analyses are given in Appendix C. Although the model is free to heave, pitch, and roll motion, the results of the ship trajectories were not obtained. That is because the only data imported to the maneuvering simulation method is the wave model.

Figure 3.19 illustrates the changes in $\bar{X}_W$ from $\psi = 0^\circ$ to 180° . There is a gradual decrease till $\psi = 0^\circ$ and increase till $\psi = 180^\circ$. At $\psi = 90^\circ$, the force in the x -direction is equal to or close to zero. The longitudinal wave drift forces calculated

according to three approaches (API, 2005; Bergdahl & Kofoed, 2015; Gupta, 2008) are much higher than the experimental results. The lines by two approaches (Advanced Marine Enterprises Inc., 1996; DNV AS, 2021) and experimental results illustrate a parallel pattern over the given heading angle frame.

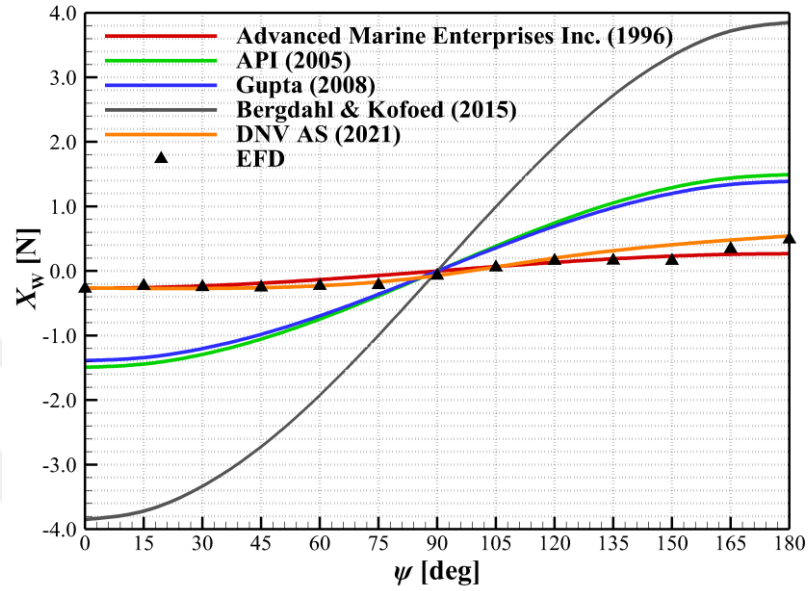


Figure 3.19 : Comparison of longitudinal wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$.

Figure 3.20 depicts the relationship between experimental and empirical/analytical $\bar{Y}_W$. The graph exhibits a peak at $\psi = 90^\circ$ at each approximation. In the following sea and head sea cases, the wave force affected in the y-direction is zero or close to zero. There is a significant difference between the lateral wave drift loads of experimental results and the two results (Advanced Marine Enterprises Inc., 1996; Bergdahl & Kofoed, 2015). The patterns of experimental results and results from remaining approaches overlap (API, 2005; DNV AS, 2021; Gupta, 2008), implying a close resemblance between datasets.

Figure 3.21 shows the yaw moment results obtained with experimental and empirical/analytical approaches. The yaw moment experiences fluctuations with periodic peaks. As in the curve obtained by the approximations of API (2005), the yaw moment is at its highest value somewhere between $\psi = 45^\circ$ and 60° in the experimental data set. The data points in both approximations coincide at various heading angles, showing a high degree of consistency with the experimental results.

Despite the harmony in the forces, the moment values obtained with the DNV approach (DNV AS, 2021) are very small compared to the experimental data set.

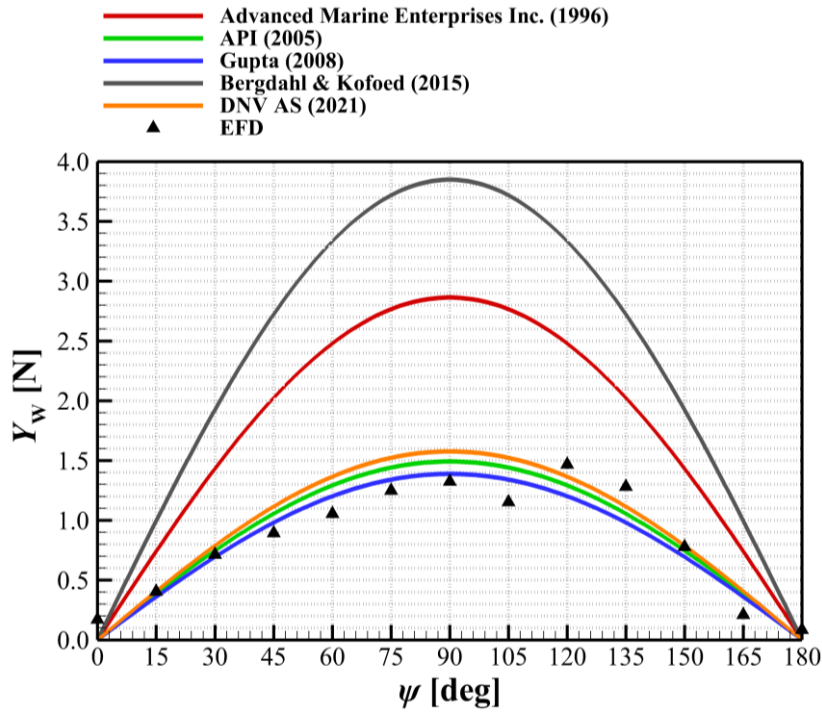


Figure 3.20 : Comparison of lateral wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$.

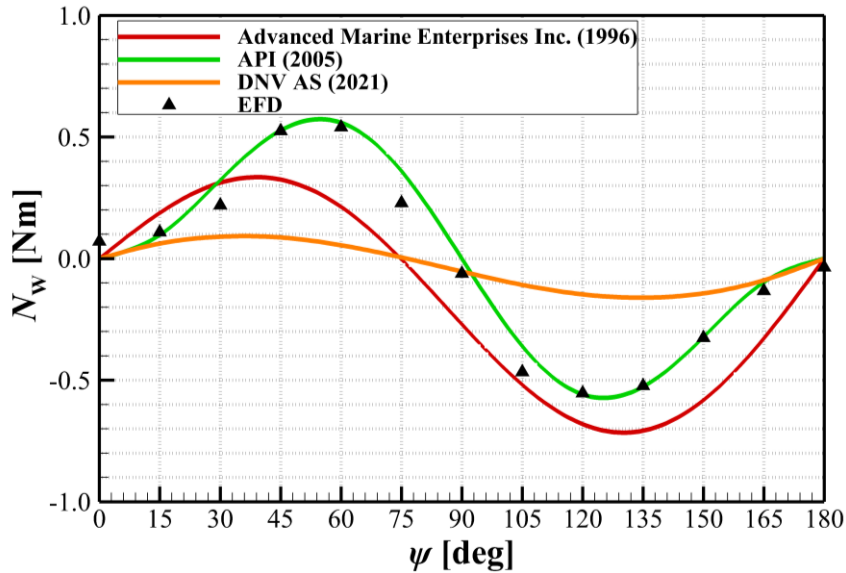


Figure 3.21 : Comparison of yaw moments of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$.

In conclusion, the graphs demonstrate a consistent pattern of experimental data over a specific heading angle. DNV approach (DNV AS, 2021) can be preferred for the calculation of wave forces except moment. The comparative results and tables are

presented in Appendix D for " $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ " and " $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ ". All graphs exhibit a similar trend of wave loads as in the above conditions.

Table 3.6 involves $\bar{X}_W$, $\bar{Y}_W$, and $\bar{N}_W$ values for various heading angles at " $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$ ". The data presented in the table ensures valuable information for maneuvering performance in waves by providing input for the system-based simulation.

Table 3.6 : Wave drift loads of steady drift tests with various headings at $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$.

ψ [deg]	$\bar{X}_W$ [N]	$\bar{Y}_W$ [N]	$\bar{N}_W$ [Nm]
0	-0.275	0.169	0.070
15	-0.231	0.402	0.108
30	-0.250	0.712	0.219
45	-0.256	0.892	0.524
60	-0.231	1.053	0.540
75	-0.219	1.247	0.228
90	-0.074	1.334	-0.062
105	0.053	1.151	-0.467
120	0.157	1.466	-0.554
135	0.158	1.280	-0.524
150	0.156	0.777	-0.326
165	0.338	0.208	-0.133
180	0.489	0.084	-0.036

In the following comparative analysis, three experimental data, namely longitudinal wave drift force, lateral wave drift force, and yaw moment, are detailed for three different experimental conditions: " $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$ ", " $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ ", and " $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ ".

Figure 3.22 depicts the comparison of longitudinal wave drift forces of steady drift tests between the heading angles 0° to 180° . In all cases the condition $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ has the highest values. This is followed by $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$ and $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ has the lowest force values. As the wavelength increased, the longitudinal wave drift force acting on the ship increased. When the wave height and wave length are halved, the longitudinal wave drift force acting on the ship is reduced.

The same pattern is shown in lateral wave drift force: Figure 3.23 shows the comparison of lateral wave drift forces of steady drift tests between the heading angles 0° to 180° . When the wave height and length are reduced by half, a more stable decrease is observed in the $\bar{Y}_W$ force, and the values are reduced nearly by half compared with the wavelength equal to the ship length.

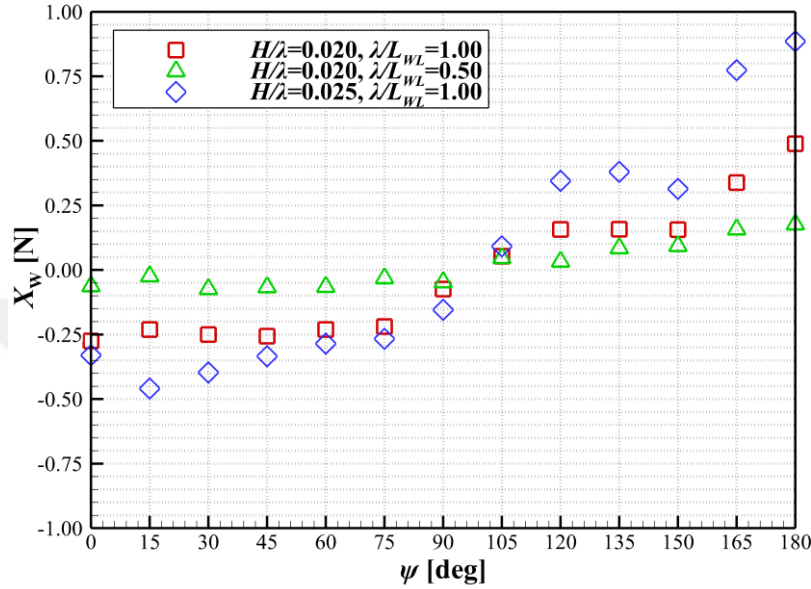


Figure 3.22 : Comparison of longitudinal wave drift forces of steady drift tests.

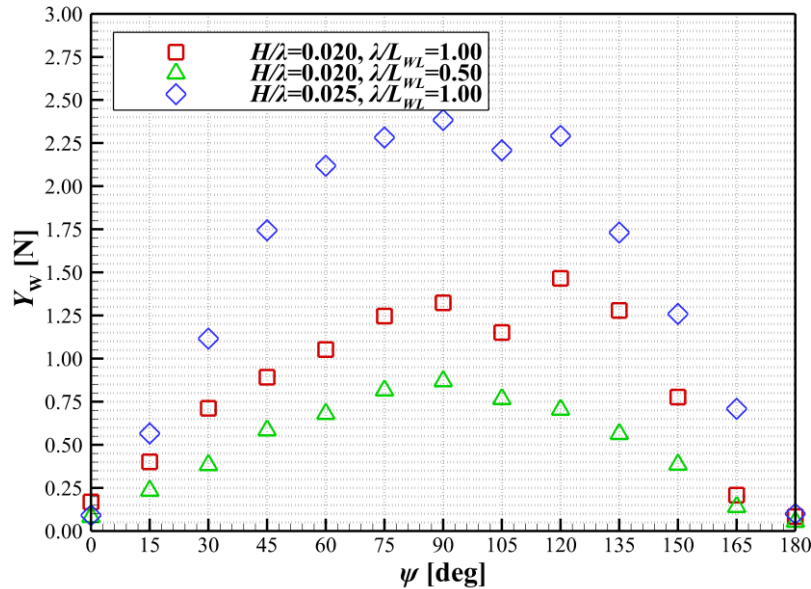


Figure 3.23 : Comparison of lateral wave drift forces of steady drift tests.

Figure 3.24 shows the comparison of yaw moments of steady drift tests between the heading angles 0° to 180° . Again, the results appear in a sensible order, given their wavelengths and heights.

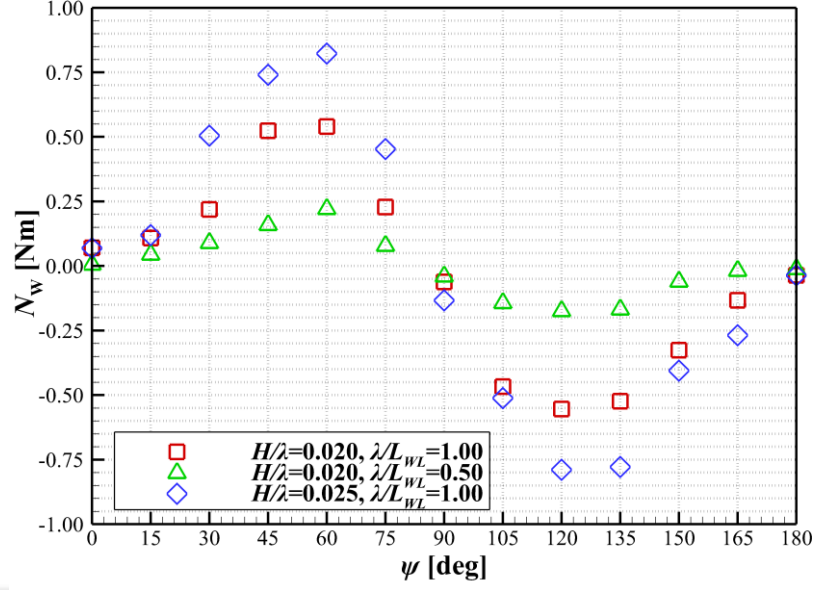


Figure 3.24 : Comparison of yaw moments of steady drift test.

3.8.2 Wave parameters

Wave parameters are fundamental measurements in the experiments conducted in the towing tank to characterize their effects on the subject model. These parameters were obtained through controlled experiments conducted in the towing tank, using specialized instruments and sensors described in chapter 3.5.2 . The time-dependent wave signals were collected during experiments to calculate two key parameters namely wave height and wavelength. Wave height is the vertical distance between the trough and crest of a wave, and wavelength refers to the horizontal distance between consecutive wave crests. The time-dependent wave signals for each test case are presented in Appendix E. Figure E.1 and Figure E.2 illustrate the changes in wave height for " $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$ ". Figure E.3 and Figure E.4 present the wave signals for " $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ ". Lastly, Figure E.5 and Figure E.6 show the wave height over time at " $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ ".

The experimental H/λ and λ/L_{WL} ratios were achieved and the deviation in the test environment was obtained. The absolute error is preferred to quantify how close or how far off the measurement is from the actual value. The absolute error, E_{abs} , is calculated by taking the absolute value of the difference between the actual value, V_a , and the experimental value, V_e :

$$E_{abs} = |V_a - V_e| \quad (3.10)$$

A smaller absolute error indicates a more accurate measurement, while a larger absolute error suggests a less precise measurement or estimation. The absolute errors were calculated in terms of H/λ and λ/L_{WL} for each test case. Experimental wave parameters and absolute errors for " $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$ ", " $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ ", and " $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ " are presented in Table 3.7, Table 3.8, and Table 3.9, respectively. The larger absolute error appears for λ/L_{WL} in $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$. It can be said that the uncertainty increases as the wave height decreases.

Table 3.7 : Experimental wave parameters for $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$.

ψ [deg]	H_{RMS} [m]	H_{PP} [m]	f [Hz]	T [s]	λ [m]	H/λ [-]	E_{abs} [-]	λ/L_{WL} [-]	E_{abs} [-]
0	0.022	0.063	0.750	1.334	2.777	0.023	0.003	0.882	0.118
15	0.023	0.064	0.750	1.334	2.777	0.023	0.003	0.882	0.118
30	0.023	0.064	0.750	1.334	2.777	0.023	0.003	0.882	0.118
45	0.023	0.064	0.750	1.334	2.777	0.023	0.003	0.882	0.118
60	0.022	0.063	0.750	1.334	2.777	0.023	0.003	0.882	0.118
75	0.023	0.064	0.750	1.334	2.777	0.023	0.003	0.882	0.118
90	0.023	0.064	0.750	1.334	2.777	0.023	0.003	0.882	0.118
105	0.022	0.063	0.750	1.334	2.777	0.023	0.003	0.882	0.118
120	0.022	0.061	0.750	1.334	2.777	0.022	0.002	0.882	0.118
135	0.022	0.062	0.750	1.334	2.777	0.022	0.002	0.882	0.118
150	0.022	0.061	0.751	1.331	2.767	0.022	0.002	0.879	0.121
165	0.022	0.061	0.750	1.334	2.777	0.022	0.002	0.882	0.118
180	0.022	0.063	0.750	1.334	2.777	0.023	0.003	0.882	0.118

Table 3.8 : Experimental wave parameters for $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$.

ψ [deg]	H_{RMS} [m]	H_{PP} [m]	f [Hz]	T [s]	λ [m]	H/λ [-]	E_{abs} [-]	λ/L_{WL} [-]	E_{abs} [-]
0	0.012	0.033	1.050	0.953	1.417	0.023	0.003	0.450	0.430
15	0.011	0.032	1.050	0.953	1.417	0.023	0.003	0.450	0.430
30	0.012	0.033	1.050	0.953	1.417	0.023	0.003	0.450	0.430
45	0.011	0.032	1.050	0.953	1.417	0.022	0.002	0.450	0.430
60	0.011	0.031	1.050	0.953	1.417	0.022	0.002	0.450	0.430
75	0.011	0.032	1.050	0.953	1.417	0.023	0.003	0.450	0.430
90	0.011	0.033	1.050	0.953	1.417	0.023	0.003	0.450	0.430
105	0.011	0.032	1.062	0.941	1.384	0.023	0.003	0.440	0.420
120	0.012	0.033	1.050	0.953	1.417	0.023	0.003	0.450	0.430
135	0.012	0.034	1.050	0.953	1.417	0.024	0.004	0.450	0.430
150	0.012	0.033	1.050	0.953	1.417	0.023	0.003	0.450	0.430
165	0.011	0.031	1.050	0.953	1.417	0.022	0.002	0.450	0.430
180	0.011	0.032	1.050	0.953	1.417	0.023	0.003	0.450	0.430

Table 3.9 : Experimental wave parameters for $H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$.

ψ [deg]	H_{RMS} [m]	H_{PP} [m]	f [Hz]	T [s]	λ [m]	H/λ [-]	E_{abs} [-]	λ/L_{WL} [-]	E_{abs} [-]
0	0.029	0.082	0.750	1.334	2.777	0.029	0.004	0.882	0.118
15	0.028	0.080	0.750	1.334	2.777	0.029	0.004	0.882	0.118
30	0.028	0.079	0.750	1.334	2.777	0.029	0.004	0.882	0.118
45	0.028	0.080	0.750	1.334	2.777	0.029	0.004	0.882	0.118
60	0.028	0.079	0.750	1.334	2.777	0.029	0.004	0.882	0.118
75	0.029	0.082	0.750	1.334	2.777	0.029	0.004	0.882	0.118
90	0.027	0.077	0.744	1.344	2.821	0.027	0.002	0.896	0.104
105	0.026	0.074	0.744	1.344	2.821	0.026	0.001	0.896	0.104
120	0.026	0.075	0.750	1.334	2.777	0.027	0.002	0.882	0.118
135	0.027	0.076	0.750	1.334	2.777	0.027	0.002	0.882	0.118
150	0.027	0.076	0.750	1.334	2.777	0.027	0.002	0.882	0.118
165	0.027	0.077	0.750	1.334	2.777	0.028	0.003	0.882	0.118
180	0.027	0.076	0.750	1.334	2.777	0.027	0.002	0.882	0.118



4. COMPUTATIONAL STUDY

This section presents the solution strategy of the flow and some key features of the numerical domain and grid structure that will be used in the subsequent sections. The main purpose of steady drift simulations is to predict the forces and moments as a result of the prescribed heading angles of the model. Afterward, the results are compared with the experimental study. In addition to the simulations of steady drift within the waves, the turning circle in calm water was directly simulated under $Fr = 0.2$ and $\delta = 35^\circ$ conditions.

4.1 Overview of Equations and Methods

The fundamental governing equations of fluid dynamics describe the motion of fluids in response to external forces. The continuity equation, the Navier-Stokes equation, and the energy equation are the three primary equations that characterize fluid dynamics. The solutions of these equations are often obtained using numerical methods, such as finite volume or finite element methods, to provide a detailed understanding of fluid behavior. The following sections describe the governing equations and details of numerical simulation.

4.1.1 Governing equations

The continuity and the Reynolds-averaged Navier-Stokes (RANS) momentum equations for incompressible flow are the governing equations utilized to solve the numerical problem. The following are the averaged continuity and momentum equations:

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0 \quad (4.1)$$

$$\rho \left(\frac{\partial \bar{u}_i}{\partial t} + \bar{u}_j \frac{\partial \bar{u}_i}{\partial x_j} \right) = - \frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j} (\bar{\tau}_{ij} - \rho \overline{u'_i u'_j}) \quad (4.2)$$

where $\bar{u}_i$ and $\bar{u}_j$ are the mean velocity vectors in the i th and j th direction of the Cartesian coordinate, ρ is the density, $\bar{p}$ is the mean pressure and $-\rho\overline{u'_i u'_j}$ is the Reynolds stress tensor.

4.1.2 Turbulence models

Turbulence modeling is a crucial part of engineering analysis, as it is the primary source of randomness and uncertainty in simulations. With the application of mathematical models that describe the physics of turbulent flows, turbulence models offer a technique to anticipate the behavior of turbulent flows. These models make an effort to depict how turbulent motion and energy transfer affect the flow field's characteristics. The turbulent flow around the hull is modeled with the realizable two-layer $k - \varepsilon$ scheme. This model combines the realizable $k - \varepsilon$ model with the two-layer approach. The turbulent eddy viscosity is determined according to the turbulent kinetic energy, k , and the turbulent dissipation rate, ε . Two-layer formulations are valid for both low-Reynolds number-type meshes ($y^+ \sim 1$) and wall-function type meshes ($y^+ > 30$). The model is based on a more accurate representation of the production and dissipation of turbulent kinetic energy, which improves the accuracy of the model in certain flow regimes. Also, the realizable two-layer $k - \varepsilon$ is less sensitive to the boundary conditions than the standard $k - \varepsilon$ model, which can be important for simulations where the boundary conditions are difficult to determine. Therefore, it is suitable for free-surface flows.

The transport equations for the kinetic energy and the turbulent dissipation rate are:

$$\frac{\partial}{\partial t}(\rho k) + \nabla \cdot (\rho k \bar{u}) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right] + P_k - \rho(\varepsilon - \varepsilon_0) + S_k \quad (4.3)$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \nabla \cdot (\rho \varepsilon \bar{u}) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \nabla \varepsilon \right] + \frac{1}{T_e} C_{\varepsilon 1} P_\varepsilon - C_{\varepsilon 2} f_2 \rho \left(\frac{\varepsilon}{T_e} - \frac{\varepsilon_0}{T_0} \right) + S_\varepsilon \quad (4.4)$$

where $\bar{u}$ is the mean velocity, μ is the dynamic viscosity, $\sigma_k, \sigma_\varepsilon, C_{\varepsilon 1}$ and $C_{\varepsilon 2}$ are the model coefficients, P_k and P_ε are production terms, f_2 is a damping function, S_k and S_ε are the user-specified source terms. ε_0 is the ambient turbulence value in the source terms and T_0 is a specific time scale.

The flow field is 3-D, incompressible, unsteady, and fully turbulent. The computations were performed by the commercial CFD code Siemens PLM STAR-CCM+ (v.

16.06.008-R8). The Finite Volume Method (FVM) was utilized for discretization. It involves dividing the domain of the partial differential equations into a finite number of control volumes, and then integrating the governing equations over each of these volumes. To apply the FVM, the domain should be discretized into a finite number of control volumes, as will be explained in chapter 4.2.2.

The velocity-pressure coupling was accomplished using the Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) based pressure correction technique. It is a widely used technique for solving the Navier-Stokes equations, which define the motion of fluid flow. The pressure-velocity coupling is to be solved in a staggered grid arrangement, where the pressure and velocity variables are situated at various grid locations. Firstly, the momentum equations for the velocity field are solved by the finite volume method to reach a local and tentative velocity field. Then, the continuity equation is globally satisfied by solving the pressure correction equation. The pressure correction is solved implicitly in the SIMPLE algorithm. The velocity field is corrected by using the pressure correction and the momentum and continuity equations are satisfied. These steps are repeated until the convergence in velocity and pressure fields is achieved.

A second-order upwind technique was used to discretize convection terms in RANS and turbulence transport equations. The basic idea of the second-order upwind scheme is to use a weighted average of the values of the scalar quantity at neighboring grid points, with a larger weight given to the value on the upstream side of the flow. The weights are chosen to be second-order accurate, which means that the error in the approximation decreases as the grid spacing is reduced.

For an accurate estimation of flow and turbulence characteristics along the wall boundary layer, the two-layer all- y^+ wall treatment was chosen. The approach allows for a relatively coarse mesh in the outer region of the flow domain while still providing accurate predictions of the flow near the wall. Thus, near-wall modeling, which tries to apply high- y^+ wall treatment for coarse meshes and low- y^+ wall treatment for fine meshes was preferred. The details about the boundary layer are examined in chapter 4.2.2.

4.2 Numerical Model

The geometry to be considered is fully appended ONRT with $L_{WL} = 3.147\text{ m}$. The 1:48.9 scaled model was chosen to perform the unsteady simulations under prespecified heading angles. The steady drift results were achieved at zero speed. Numerical studies were carried out on the regular wave for only $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ of the three different wave properties carried out in the experimental study.

The main particulars of the model and the wave parameters are not repeated in this section because were reported in Table 3.2 and Table 3.4, respectively. The basic views of the test model parallel to the global basic planes (XY , XZ , and ZY) and the perspective view are displayed in Figure 4.1.

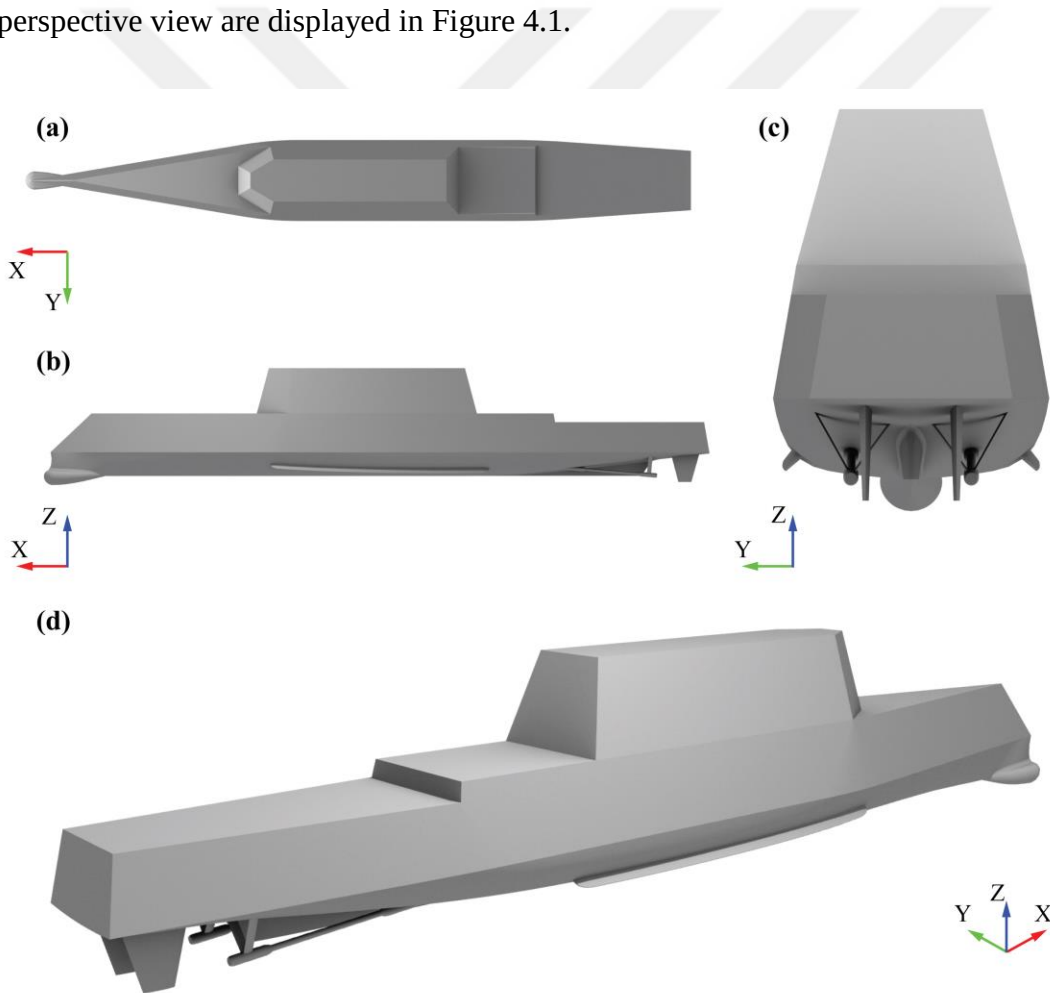


Figure 4.1 : The fully appended ONRT geometry. (a) Top view, (b) Side view, (c) Back view, (d) Perspective view.

4.2.1 Computational domain

In computational fluid dynamics, the computational domain is the physical region of interest for which a CFD solution is sought. It is typically defined by a set of boundary conditions that describe the flow environment and the physical parameters of the problem. The boundary conditions can be specified in terms of the velocity, pressure, temperature, and other properties of the flow. The computational domain is usually divided into a number of grid cells or elements, which are used to solve the governing equations of the problem. The size and shape of the computational domain and the number of grid cells depend on the nature of the problem being solved.

The computational domain is divided into two parts as the background and the overset. The computational domain is a 3-D deep water wave tank and the origin of the coordinate system is located at the center of gravity of the ONRT hull. This point is accepted as the zero point and the tank dimensions are formed as multiples of the ship waterline length or wavelength. While ship length is generally used to determine domain sizes in ship resistance analyses, it may be a good choice to determine the distances of the boundaries, which are perpendicular to the direction of the wave propagation, depending on the wavelength. Therefore, the inlet boundary is located 1.3λ away from the center point, while the outlet boundary is located 2.6λ . The lateral boundaries are $1.3L_{WL}$ away from the centerline of the ONRT model. The distances of the top and bottom boundaries from the free surface are determined as $1.3L_{WL}$ and $2.5L_{WL}$, respectively. The overset domain where the motion is provided has the dimension of $7T$ height, $1.2L_{WL}$ in length and $1.4B$ in width. The dimensions of the computational domain in two perspectives are presented in Figure 4.2.

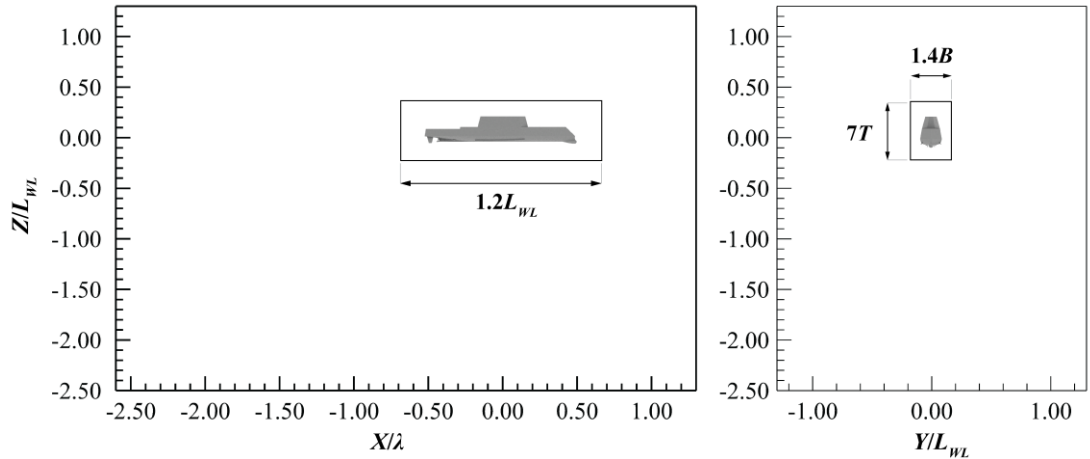


Figure 4.2 : The dimensions of the computational domain from XZ and YZ planes.

The overset interface between the parts must be manually chosen after each part has been allocated to a region. At least one of the boundaries must be set to overset mesh. Then, it is possible to choose and configure how values are interpolated at the interface between various meshes. There are four alternatives available in commercial CFD software: distance weighted, linear, Quasi2D, and least square. The linear overset mesh interpolation scheme is selected to establish the relation between overset and background region.

Steady drift simulations consist of different heading angles. To change the heading angle, the overset domain, local mesh densifications (i.e., overlap geometries), and ONRT model are rotated together by the relevant heading angle. Background domain and wave properties remain the same. Discretization is completed and overset interface is provided again. Then, the cartesian coordinate systems depending on the ship's motion are rotated by the relevant heading angle.

In order to simulate body motion and analyze the flow characteristics correctly, the refinement volumes need to be defined around the moving parts or free surface. Apart from the main background part and overset part, three different refinement volumes as illustrated in Figure 4.3 are imposed in the present study. To elaborate, the three refinement volumes with cross-section types and dimensions are listed in Table 4.1.

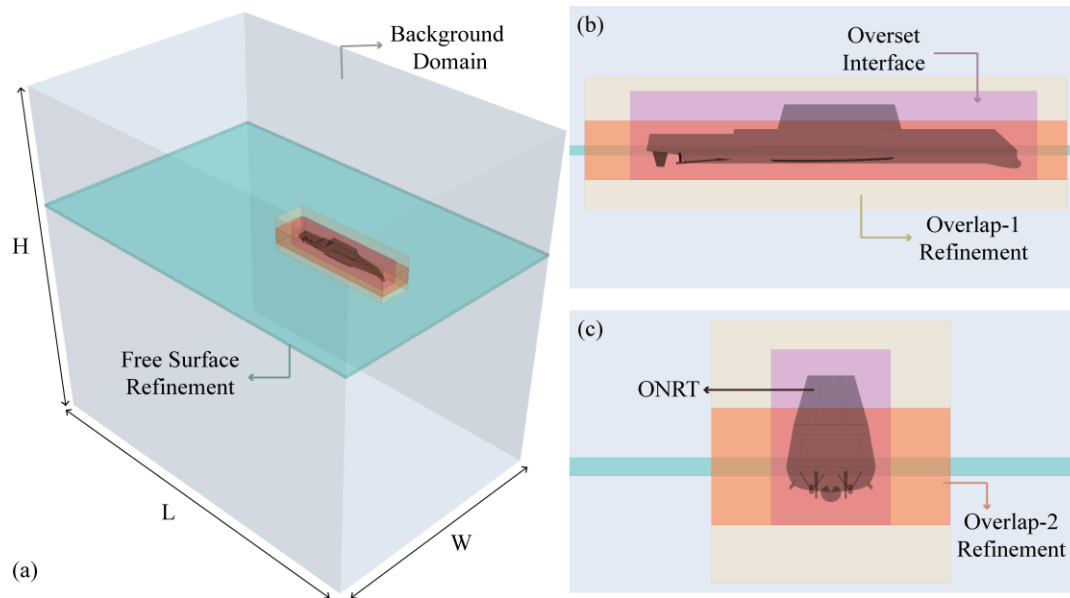


Figure 4.3 : Refinement volumes and interfaces (a) Perspective view, (b) Side view, (c) Back view.

Table 4.1 : Refinement volumes for applied grid.

Refinement	Cross-section type	Dimension ($L_{ref} \times W_{ref} \times H_{ref}$)
Free surface	Rectangular	12.28×8.20×0.08
Overlap-1	Rectangular	4.10×1.02×1.15
Overlap-2	Rectangular	4.10×1.02×0.51

4.2.2 Mesh resolution

Mesh resolution refers to the fineness or coarseness of the computational mesh to discretize the fluid domain. Selecting an appropriate mesh resolution depends on the specific problem being simulated, the available computational resources, and the desired level of accuracy. Extreme grid distortion can result in uneven transition between active and passive cells during the motion of the hull and, ultimately, simulation divergence. To guarantee a smooth grid transition, the grids of the refinement volumes should be gradually raised. Before grid generation, two refinement zones named Overlap-1 and Overlap-2, mentioned in the previous section were defined around the overset region where the motion is critical. In general, Overlap-1 controls the mesh transition between the overset region and the background region, while Overlap-2 regulates the mesh transition between the free surface and the overset region to avoid free surface-induced distortions during motion. The smallest cell in the computational domain is the Δz on the free surface and the refinement ratios are expressed in Δz in Table 4.2. Figure 4.4 depicts the grid arrangement surrounding the ONRT from various perspectives.

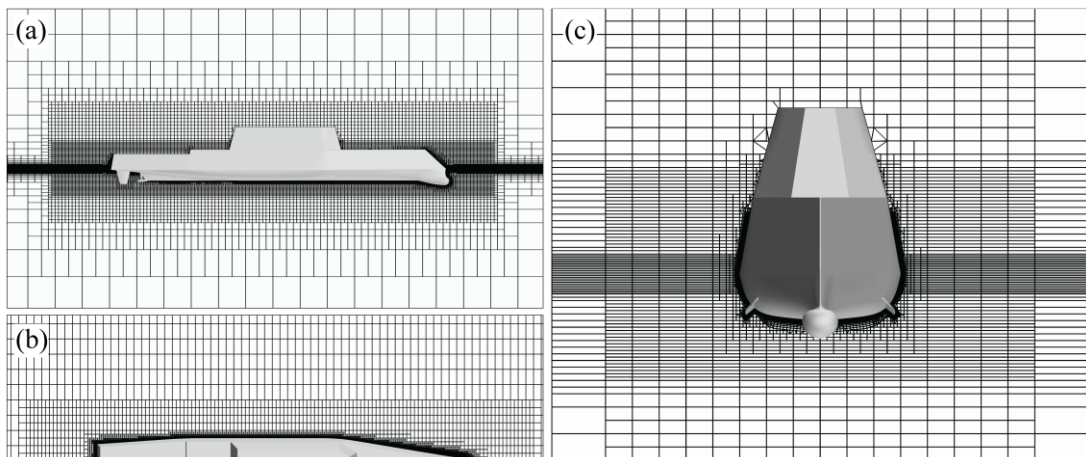
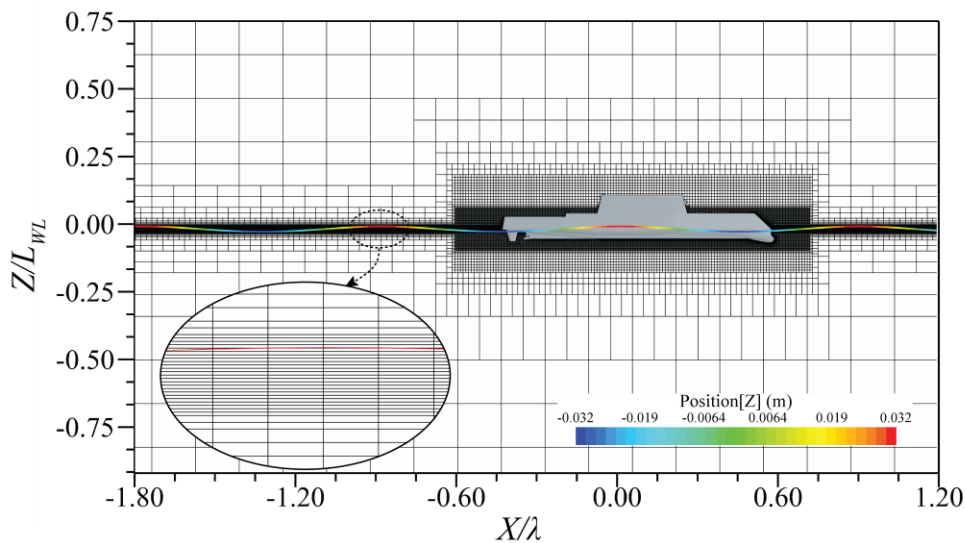


Figure 4.4 : The spatial discretization around the hull (a) Side view, (b) Top view, (c) Front view.

Table 4.2 : Refinement ratios in Δz of free surface.

	Free surface	Overset	Overlap-1	Overlap-2	Background
Δx	16	8	8	8	128
Δy	32	16	16	16	128
Δz	1	8	8	2	128

The development of a qualified phase separation for the free surface problem might be guided by several studies. At least 40 grid points per wavelength are advised by ITTC (2014), and CD-Adapco (2018) suggests utilizing 80-160 grids per wavelength and 16-32 grids per wave height. A minimum of 80 points per wavelength and a minimum of 16 cells were defined in the vertical direction to effectively capture severe free surface deformation. The refinement in the global y -axis direction was chosen to be twice the refinement in the x -axis direction. Additionally, 0.08 m was chosen as the height of the refinement volume for the free surface in z -direction, to cover the wave height of 0.063 m. The resulting fifth-order Stokes wave is completely contained within the appropriate grid refinement, as is seen from the Volume of Fluid (VOF) view on the mesh shown in Figure 4.5.

**Figure 4.5** : Free surface refinement.

To precisely forecast the rotational and translational motions of moving structures, the overset algorithm is a suitable technique. The main benefit of using the overset mesh approach in seakeeping simulations is that it reduces the grid deformations caused by unsteady motions around the geometry. The dimensions of the overset domain and the mesh transition between the overset domain and the background domain are the factors that affect the overset cell status. Therefore, while the domain is being divided into

small subdomains, the overset cell status can be checked to examine whether the transition is well established. As depicted in Figure 4.6, the solution does not include the passive cells (value “1”) inside the overset area. Acceptor cells (value “-2”) separate the active (value “0”) and passive cells. In the later time steps of the simulation, passive cells may become active and active cells may become passive.

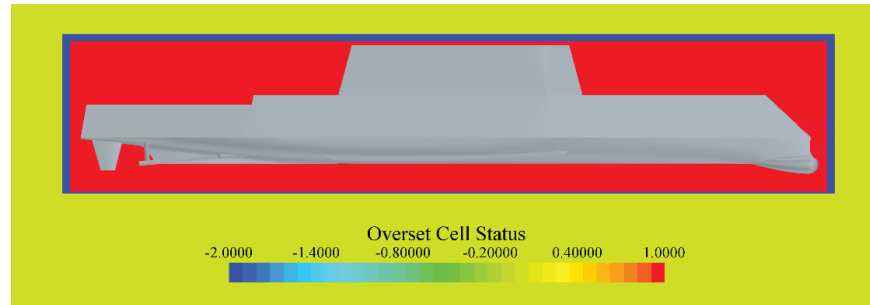


Figure 4.6 : Overset cell status.

After volumetric control, the surface mesh on the body had to be improved. In particular, the surface mesh of the rudders and shafts is one of the factors affecting the flow behavior behind the ship. Therefore, the mesh quality is improved on the surface of the ship below the waterline, especially around the appendages. In the part above the waterline, unnecessary mesh numbers are avoided by using larger cells and disabling the prism layer mesher. A detailed view of the surface mesh over the ONRT hull is given in Figure 4.7.

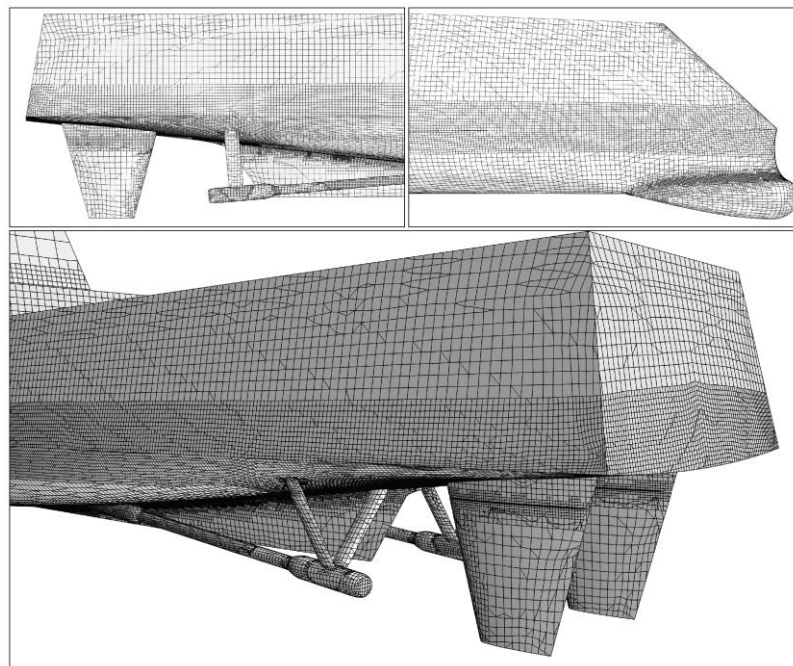


Figure 4.7 : Detailed view of the surface mesh over the hull.

The y^+ value represents the distance from the wall to the first node in the computational mesh, and its calculation includes the friction velocity at the wall. Since the ship velocity is zero in the steady drift problem, wave velocity is used while estimating the prism layer total thickness and number of prism layers. Then the boundary layer is formed along the hull surface as a thin layer of fluid. Figure 4.8 shows the boundary layer grid in detail. For a $k - \varepsilon$ turbulence model, the appropriate value of y^+ depends on the type of wall treatment used in the simulation. There are two main types of wall treatments: wall functions and low-Reynolds number models. If wall functions are used, then a y^+ value between 30 and 300 is typically recommended. If low-Reynolds number models are used, then a y^+ value of less than 1 is recommended. In the present study, a two-layer all- y^+ wall treatment is selected to accurately model the flow near walls with blended wall functions. The low-wall treatment for fine meshes and the high-wall treatment for coarse meshes are implemented. The dimensionless height of the first cell was examined and found to be $y^+ \approx 1$ using the two layer all- y^+ wall treatment. As can be seen in Figure 4.9, the wall y^+ is generally below 1 on the hull.

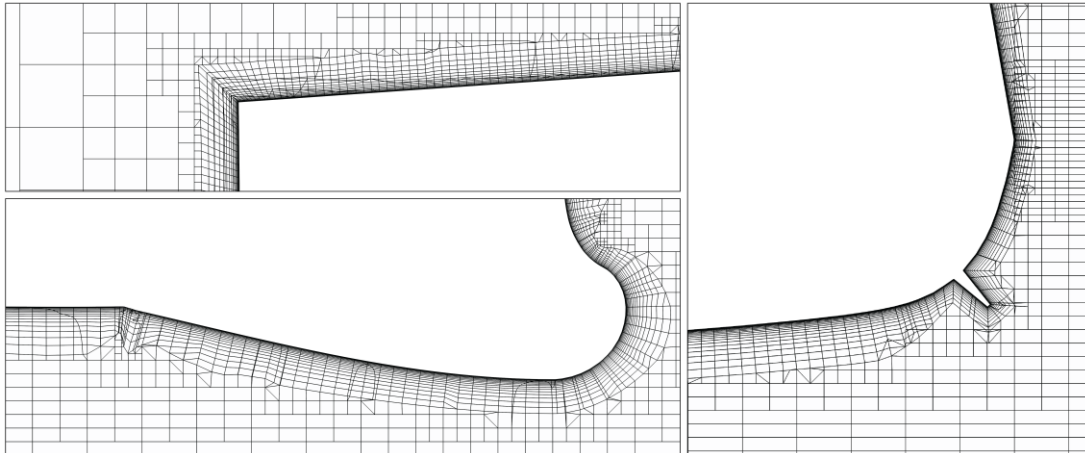


Figure 4.8 : Boundary layer discretization.

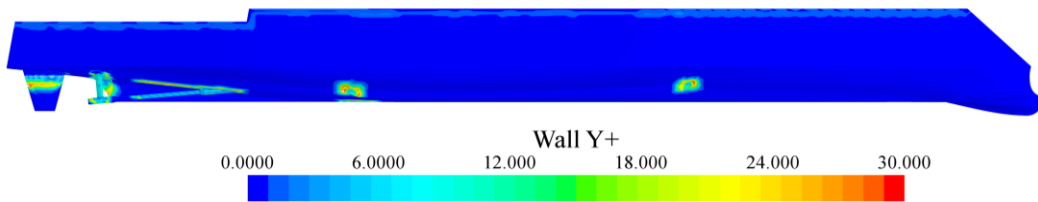


Figure 4.9 : Wall y^+ distribution on the hull.

2.695E+06 cells in total, including 6.140E+05 for the background region and 1.925E+06 million for the overset region, were created to solve the numerical problem. The detail of the grids is listed in Table 4.3.

Table 4.3 : Element counts in the simulation.

Name	Element count	Type
Background Domain	6.140E+05	cells
Bottom	3.840E+02	faces
Inlet	2.944E+03	faces
Outlet	2.944E+03	faces
Side	1.536E+04	faces
Top	3.840E+02	faces
Overset Domain	1.925E+06	cells
Hull	1.298E+05	faces
Rudders	4.210E+03	faces
Total	2.695E+06	element

4.2.3 Boundary conditions

Boundary conditions in computational fluid dynamics are conditions that are applied to the boundaries of a flow domain in order to simulate a realistic flow. Boundary conditions are used to define the state of the fluid at the boundaries and to define how the fluid interacts with the boundaries. Depending on the application, boundary conditions can be specified as fixed values, as user-defined functions, or as input from an external source. Common boundary conditions used in CFD include velocity inlet and pressure outlet, slip walls, no-slip walls, and periodic boundaries.

The downstream boundary in front of the body, side, top and bottom boundaries have the velocity inlet boundary condition. The only pressure-based outlet boundary condition is for the upstream boundary to achieve mass conservation at each time step. The pressure is fixed and the velocity gradients are equal to zero at the upstream boundary. The impermeable hull has the non-slip boundary condition in the calculations. So, the normal velocity at the wall is enforced to zero and the tangential velocity is equal to the wall velocity. The boundary conditions defined in the computational domain are illustrated in Figure 4.10.

4.2.4 Computational coordinate system

The coordinate systems in the numerical simulations consist of a laboratory coordinate system expressing the earth-fixed coordinate system and a local coordinate system following the ship's motion.

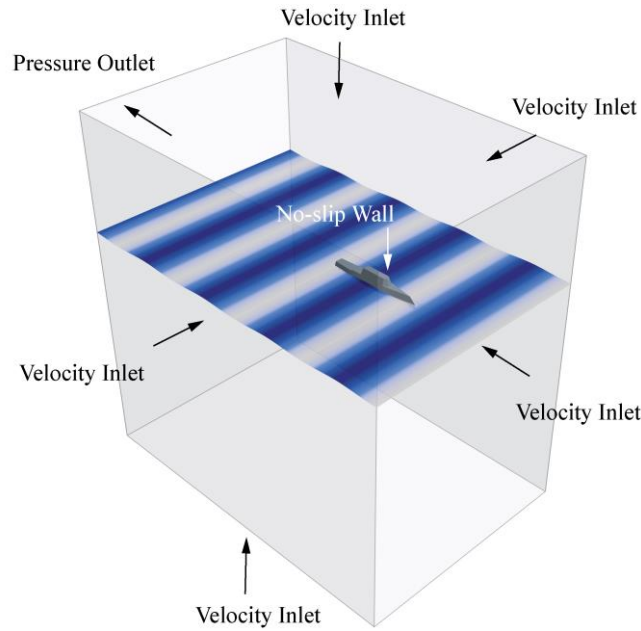


Figure 4.10 : Boundary conditions.

The local coordinate system is defined by three axes: the x -axis, y -axis, and z -axis. The x -axis is aligned with the longitudinal axis of the body, which is typically the axis of symmetry or the direction of motion. The y -axis is perpendicular to the x -axis and is usually aligned with the width or height of the body, depending on the orientation. The z -axis completes the right-handed coordinate system and is perpendicular to both the x -axis and y -axis. The origin of the local coordinate system is located at the center of mass. The orientation of the local coordinate system can be defined to the global coordinate system or itself. At each rotation of the model, i.e. changing the heading angle, the local coordinate system should also be transformed. Figure 4.11 shows the earth-fixed coordinate system and ship-fixed coordinate system.

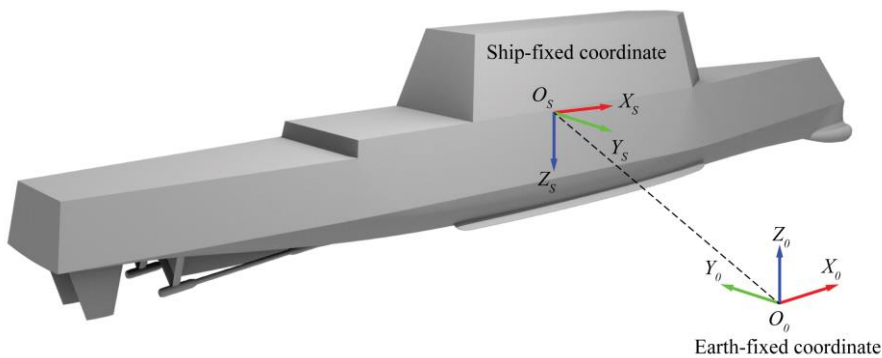


Figure 4.11 : Earth-fixed and ship-fixed coordinate systems in numerical simulations.

4.2.5 Wave generation

A CFD approach called the multiphase model is used to model how fluids interact with solid boundaries. It simulates the flow of fluids around objects and deals with the motion of the fluid over a solid boundary. There are three multiphase models as mixture multiphase, eulerian multiphase, and volume of fluid. The phase separation surface between heavy and light fluids was captured using the VOF method created by Hirt & Nichols (1981) for immiscible fluids. The VOF is well suited for simulating the flows of multiple immiscible fluids on numerical grids that can resolve the interface between the phases of the mixture. This idea is utilized by the VOF approach to monitor the motion and deformation of fluid surfaces as they change over time, enabling the precise simulation of complex fluid flows involving several fluid phases such as two-phase flows and flows involving a liquid and a gas. With this method, each time step's tracking of the gravity-induced surface waves is guaranteed to be accurate and reliable. The volume fraction of a fluid element can range from 0 (representing a light element) to 1 (representing a heavy element). Freshwater was chosen as the heavy fluid, and air was chosen as the light fluid. The freshwater density is $\rho_{water} = 997.561 \text{ kg/m}^3$ and kinematic viscosity is $\nu_{water} = 8.909 \times 10^{-7} \text{ m}^2/\text{s}$. The air density is $\rho_{air} = 1.184 \text{ kg/m}^3$ and kinematic viscosity is $\nu_{air} = 1.567 \times 10^{-5} \text{ m}^2/\text{s}$.

In order to establish a precise fluid interface, the convective term in the transport of volume fraction equation was discretized using the High-Resolution Interface-Capturing (HRIC) approach. The key idea behind HRIC is to use high-resolution discretization methods to accurately capture the position and motion of the fluid interface (Kinaci & Ozturk, 2022). HRIC as a convection scheme under the VOF approach has some benefits for simulating fluid flows with moving interfaces. HRIC, for instance, can manage more complex fluid flows, such as those involving topological changes, and can provide a more realistic fluid interface. It can also deal with issues involving significant interface deformation. The increased processing cost of using HRIC is one of the difficulties though. This is because tracking the location of the interface necessitates high-resolution discretization and the solution of additional equations.

The VOF waves model includes several types of waves such as flat, first order, fifth order, superposition, cnoidal and irregular. The fifth-order VOF wave is selected to produce the regular wave of $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$. It employs a higher-

order construction of the fluid interface as distinct from a first-order wave; therefore, more realistic solutions can be achieved. The fifth-order polynomial are applied to the fluid fraction values at each grid point close to the interface. It leads to improvements in accuracy and reductions in numerical diffusion.

A numerical forcing zone that encircles the solution domain is encountered by the progressive waves. The source term in simulations using a wave treatment can modify a forcing coefficient to fit the transport equations. The choice of damping/forcing length enhances the damping quality and may influence the user to alter the domain sizes. Therefore, by eliminating wave reflections from the boundaries and enabling the solution of the wave in this zone, putting the forcing zone at the boundaries will reduce the computational effort. Wave forcing has a condition as wave forcing length, x_f , to provide the damping quality. Wavelength is used to scale the wave forcing length at different proportions at different boundaries. Consequently, while forcing length is preferred at 0.65λ at the downstream boundary, it is entered as 2λ at the upstream boundary. At the top and bottom boundaries, the forcing length is disabled, $x_f = \lambda$ is selected for side boundaries.

4.2.6 Kinematics of motion

Dynamic fluid body interaction in ship motion is a series of mathematical models and algorithms used to simulate the motion of a ship in a fluid environment. The models capture the dynamics of the ship, such as position, speed, acceleration, and forces acting on the ship, as well as the fluid forces, such as buoyancy, drag, and wave pressure. The 3-DoF ship motion in waves is simulated by using the Dynamic Fluid Body Interaction (DFBI) module. With the 72.6 kg body mass, the body motion option is selected as planar motion carriage. Planar motion carriage gives the user a choice about motions of heave, pitch, and roll. The velocity of the carrier in the x -direction of the laboratory system is left at zero since the model has no forward velocity. The point where the body is initially attached to the planar motion carriage is selected as the center of mass. The diagonal components of the initial moment of inertia are entered as $I_{xx} = 2.11 \text{ kg-m}^2$, $I_{yy} = 43.51 \text{ kg-m}^2$, and $I_{zz} = 43.51 \text{ kg-m}^2$.

The selection of the release and ramp time is another important step in motion analyses. The release time can be defined as a period of time that elapses before the calculations for the body motion begin. The choice of this period depends on how long

the analysis will take in total and the grid structure. In implicit unsteady analyses, it usually varies between 10 and 50 time steps. After the release time expires the motion starts and the body is released gradually until the ramp time. While the forces and moments acting on the body at the moment of release may create negative responses, the ramp time reduces this effect. That is, assuming that the body is suddenly released without an interval such as ramp time, undesirable instabilities may occur, especially in simulations with waves. In the analyses where $\Delta t = 0.0083$ s, the release time was entered as 0.166 s, which is 20 times the time step, while the ramp time was selected as 0.332 s.

4.2.7 Solution strategy and high-performance computing

Figure 4.12 shows the solution strategy that can be followed in case of changing heading angle and wave parameters. Since the steady drift analyses were carried out only for the $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$ case, the steps to be followed in cases where the wave parameters change were omitted.

The number of numerical analyzes carried out within the scope of the thesis has reached 20 with 3 grid independence analyses, 3 time dependency analyses, 13 steady drift analyses in regular sea conditions of $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$, and a direct CFD simulation of turning maneuvering. The pre-processing and post-processing operations were carried out with the author's workstation (WS1). Pre-processing involves the preparation of a computational model before the actual simulation. This stage includes defining the problem's geometry, setting boundary conditions, and meshing. Post-processing involves analyzing the results of the simulation once it is complete. This stage includes visualizing the results, analyzing the data, and drawing conclusions. Calculations were made on other workstations presented in Table 4.4. The steady drift analyses were performed between 15-30 seconds. The cumulative time taken to run the entire solution was approximately 10 to 40 hours.

Table 4.4 : Features of high-performance computing.

Device	Processor	Cores	RAM (GB)
WS1	Intel(R) Xeon(R) CPU E5-2620 v4 @ 2.10GHz	16	45
WS2	Intel(R) Xeon(R) CPU E5-2630 v4 @ 2.20GHz	20	160
WS3	Intel(R) Xeon(R) CPU E5-2630 v4 @ 2.20GHz	20	128
WS4	Intel(R) Xeon(R) CPU E5-2640 v4 @ 2.40GHz	20	160

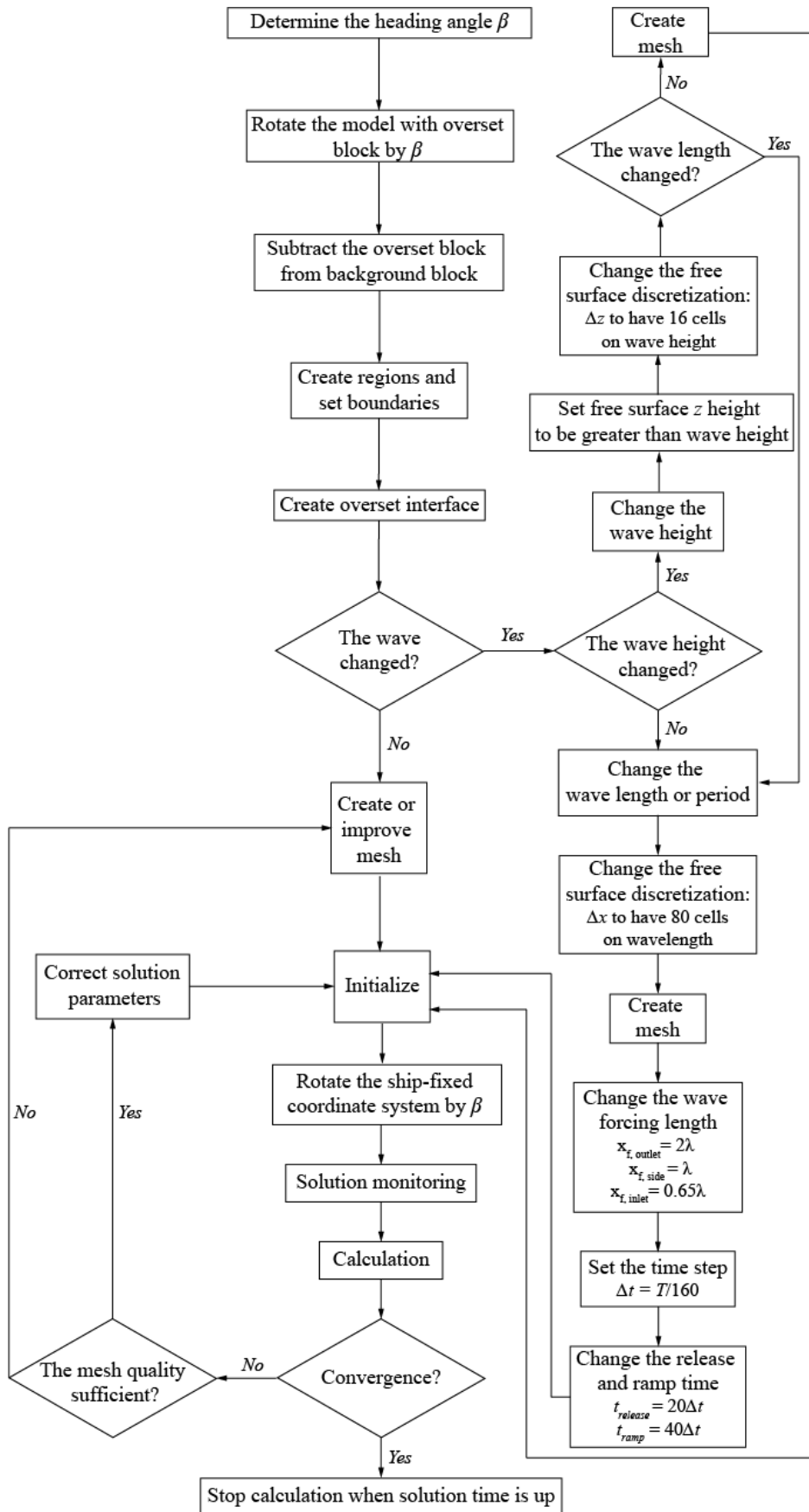


Figure 4.12 : Solution procedure.

4.2.8 Determination of the time step

A time step in CFD is the amount of time that passes between two consecutive iterations of the numerical solver. It is important to choose a time step that is small enough to capture the important physical phenomena in the simulation, but also large enough to keep the computational time reasonable. To determine the time step, the user should consider the physical model, the solution accuracy requirements, the computational resources available, and the size of the domain. There are several recommendations for the determination of the time step. ITTC recommends a procedure for time step selection (ITTC, 2014a). The recommendation is that for periodic phenomena including incoming waves, the time step is 1/100 of the wave period. In the experimental $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$ cases with wave period $T = 1.334$ s, time step was calculated as $\Delta t = 0.0133$ s.

The choice of the time step in a numerical simulation is often determined by the Courant number (ITTC, 2014a). The Courant number, also known as Courant-Friedrichs-Lewy (CFL) number, is a dimensionless parameter used in numerical analysis to assess the stability and accuracy of time-dependent simulations. The Courant number is defined as the ratio of the numerical simulation's time step to the time it takes for a wave to propagate across a grid cell. In other words, it represents the ratio of the distance traveled by a wave to the distance between two adjacent grid points. In numerical analyzes with wave propagation, the safety factor of CFL around the free surface is expected to be between 0.5 and 1.0. Smaller CFL values lead to more stable simulations but also increase the computational cost. CFL value can be calculated by the following expression:

$$CFL = \frac{u\Delta t}{dx} \quad (4.5)$$

where u is the velocity of the wave, Δt is the time step and dx is the distance between two adjacent grid points. In case of $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$, $u = 2.083$ m/s ($u = \lambda f$) and $dx = 0.04$ m; therefore Δt is calculated as 0.0192 s by using equation (4.5 at $CFL = 1$).

There is another guideline to help user select an appropriate time step for wave simulations. CD-Adapco (2018) suggests that the wave period has 160-320 of the time step, Δt , and that brings the CFL number will be less than 0.5 on the free surface.

When this ratio is selected 160, the calculated time step is 0.0083 s. As a result, the simulation is expected to be more accurate and stable as the value of CFL will be smaller with the $\Delta t = 0.0083$ s.

4.3 Numerical Results and Discussions

In this section, verification and validation will be described first and numerical results will be presented. Verification and validation (V&V) in CFD refer to the process of assessing the accuracy and reliability of the simulations. The process of V&V is an important step in ensuring the reliability of CFD simulations, as it allows the user to identify and correct any errors or inconsistencies in the model and assess the level of confidence that can be placed in the simulation results. Following the V&V study, mean wave drift loads simulations with different heading angles will be detailed.

4.3.1 Verification

Verification is the process of determining that a computational model accurately represents the physical system it is meant to simulate. This includes checking that the numerical methods and discretization used in the simulation are appropriate and that the code implementation is correct. Verification can be done by comparing the results of the simulation with analytical solutions or simpler numerical simulations for well-understood problems.

4.3.1.1 Richardson extrapolation

The Richardson extrapolation (RE) method is one of the methods suggested for discretization error estimation. In order to estimate the precise solution to the mathematical model, the discrete solutions of different refined meshes or time steps are combined.

The method is applied to at least three solutions with discretization levels which is called fine (N_1), medium (N_2) and coarse (N_3) mesh. The relation between the discretization levels or time step levels is ensured with a refinement factor ratio, r . While the refinement factor for discretization is recommended as $\sqrt{2}$ by Sakamoto et al. (2012), Roache (1998) stated that this value should be 1.1. The refinement factors are calculated as follows:

$$r_{21} = \sqrt{\frac{N_1}{N_2}}, r_{32} = \sqrt{\frac{N_2}{N_3}} \quad (4.6)$$

In temporal convergence studies, the calculation of the refinement factors needs at least three different time steps:

$$r_{21} = \frac{\Delta t_1}{\Delta t_2}, r_{32} = \frac{\Delta t_2}{\Delta t_3} \quad (4.7)$$

The discretization absolute error, ε ,

$$\varepsilon_{21} = \varphi_1 - \varphi_{exact} = \frac{\varphi_2 - \varphi_1}{r_{12}^p - 1} \quad (4.8)$$

$$\varepsilon_{21} = \varphi_2 - \varphi_1 \text{ and } \varepsilon_{32} = \varphi_3 - \varphi_2$$

where φ_k denotes the solution scalar on the k th refinement. The convergence condition, R , can be found by

$$R = \frac{\varepsilon_{21}}{\varepsilon_{32}} \quad (4.9)$$

There are three convergence conditions as monotonic convergence ($0 < R < 1$), oscillatory convergence ($R < 0$), and divergence ($R > 1$).

Generalized RE is used to estimate the error for monotonic convergence. The apparent order, p , can be calculated with the following expression:

$$p = \frac{1}{\ln(r_{21})} \left| \ln\left(\frac{\varepsilon_{32}}{\varepsilon_{21}}\right) + q(p) \right|$$

$$q(p) = \ln\left(\frac{r_{21}^p - s}{r_{32}^p - s}\right) \quad (4.10)$$

$$s = 1 \cdot \text{sgn}\left(\frac{\varepsilon_{32}}{\varepsilon_{21}}\right)$$

Extrapolated values

$$\varphi_{ext}^{21} = \frac{r_{21}^p \varphi_1 - \varphi_2}{r_{21}^p - 1} \quad (4.11)$$

Approximate relative error

$$e_a^{21} = \left| \frac{\varphi_1 - \varphi_2}{\varphi_1} \right| \quad (4.12)$$

Extrapolated relative error

$$e_{ext}^{21} = \left| \frac{\varphi_{ext}^{21} - \varphi_1}{\varphi_{ext}^{21}} \right| \quad (4.13)$$

The idea of extended RE served as the basis for the Grid Convergence Index (GCI), which is a grid refinement error estimator. This estimator incorporates a safety factor, and GCI is defined for both fine and coarse grids by

$$GCI_{fine}^{21} = \frac{1.25 e_a^{21}}{r_{21}^p - 1} \quad (4.14)$$

4.3.1.2 Spatial and temporal convergence study

The verification procedure for grid refinement and time step was performed via Grid Convergence Index (GCI) using Richardson extrapolation. While the grid size and time step were selected as uncertainty sources, the integral parameter (φ) was the RMS amplitude of the surge force (X) of the steady drift test at $\psi = 0^\circ$. The ratio of wave height to wavelength (H/λ) is equal to 0.020 and the wavelength over ship length (λ/L_{WL}) is equal to 1.00.

According to Richardson extrapolation, systematically refined grids with element counts ranging from 2.14×10^6 to 3.74×10^6 were created with a refinement ratio of $r = \sqrt{2}$. The refinement process was done in overset and background domain, and free surface refinement was not changed. The reason for this is that the grids in the relevant region must have certain ratios for the wave to be formed correctly. Without making any modifications to the simulation setup, the verification study was conducted in a set of setups with three different grid layouts. The fine, medium, and coarse meshes

are respectively represented by the mesh configurations N_1 , N_2 and N_3 . The convergence condition (R) is computed to reflect monotonic convergence which means it is between 0 and 1. Then, the fine-grid convergence index is calculated as 0.696%. Table 4.5 displays the uncertainty components of the spatial convergence study. The calculations for uncertainty demonstrated that a medium grid numerical setup can be employed for further research.

Table 4.5 : Spatial convergence for surge force obtained from steady drift simulations at $\psi = 0^\circ$.

	Fine (N_1)	3.74E+06
Grid number	Medium (N_2)	2.70E+06
	Coarse (N_3)	2.14E+06
Solution scalar	φ_1	6.134
	φ_2	6.261
	φ_3	6.585
Discretization absolute error	ε_{21}	0.127
	ε_{32}	0.324
Convergence ratio	$R (\varepsilon_{21}/\varepsilon_{32})$	0.392
Result	-	MC
Apparent order	p	9.498
Extrapolated value	φ_{ext}^{21}	6.100
Approximate relative error (%)	e_a^{21}	2.070
Extrapolated relative error (%)	e_{ext}^{21}	0.560
Convergence index (%)	GCI^{21}	0.696

For the temporal convergence study, three different time steps Δt_1 , Δt_2 , and Δt_3 were applied to the simulations of the medium grid, N_2 . The refinement ratio was chosen as $r = 2$. Then, the minimum-time step convergence index is calculated as 0.634%. The components related to temporal discretization error were presented in Table 4.6. The temporal convergence study has successfully established an optimal time step size ($\Delta t = 0.0083$ s) that strikes a balance between computational efficiency and solution accuracy, to confidently use CFD simulations for time-dependent problems in waves.

Table 4.6 : Temporal convergence for surge force obtained from steady drift simulations at $\psi = 0^\circ$.

	Δt_1	0.0042
Time Step	Δt_2	0.0083
	Δt_3	0.0166
	φ_1	6.426
Solution scalar	φ_2	6.261
	φ_3	5.222
Discretization absolute error	ε_{21}	-0.165
	ε_{32}	-1.039
Convergence ratio	$R (\varepsilon_{21}/\varepsilon_{32})$	0.159
Result	-	MC
Apparent order	p	-2.646
Extrapolated value	φ_{ext}^{21}	6.459
Approximate relative error (%)	e_a^{21}	2.568
Extrapolated relative error (%)	e_{ext}^{21}	0.505
Convergence index (%)	GCI^{21}	0.634

4.3.2 Validation

Validation is the process of evaluating the performance of the computational model by comparing its predictions with experimental or observational data. This includes evaluating the ability of the simulation to replicate the physics of the system, as well as its ability to predict the correct outcomes. Validation can be done by comparing the simulation results with experimental data, or by using the simulation to make predictions that are later tested experimentally.

To validate the simulation result, the experimental fluid dynamics (EFD) data for the steady drift tests detailed in chapter 33. is used. Among the simulations, medium mesh (N_2) and medium time step (Δt_2) configuration were selected. As in the verification phase, the RMS amplitudes of the time-dependent surge forces were taken.

The percent error (E) can be calculated by the following formulation:

$$E(\%V_e) = \left| \frac{V_e - V_c}{V_e} \right| \times 100 \quad (4.15)$$

where V_c represents the computational value and V_e is the experimental value.

The comparison error between the computational and experimental surge force of the steady drift test at $\psi = 0^\circ$ is given in Table 4.7.

Table 4.7 : Results of the validation study.

Scalar	V_e	V_c	$E(\%V_e)$
RMS amplitude of surge force (N)	6.221	6.261	0.646

4.3.3 Results of mean wave drift loads at different heading angles

Computational steady drift simulations were performed in regular waves at zero Froude number under the one condition of $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$. The wave scenarios cover head, beam, following, and quartering tests. Time histories of longitudinal force, X , lateral force, Y , and yaw moment, N according to the ship-fixed coordinate system were collected. The data reduction procedure presented in chapter 3.7.1 was followed as in the experimental analysis. The time histories of wave drift loads are given in Appendix E.

Computational results are added to the graphs in which the experimental results and the results obtained with the empirical/analytical approaches are presented in chapter 3.8.1. Figure 4.13 shows the longitudinal wave drift loads over the heading angle. CFD results show a consistent downward trend between 0° and 90° when showing an upward trend from 90° to 180° . The data depicted in the graph suggest a high consistency between experimental and computational results.

Figure 4.14 illustrates the changes in longitudinal wave drift forces from $\psi = 0^\circ$ to $\psi = 180^\circ$. There is a peak in the force at 90° , reaching its highest value. Compared to the EFD results, a more consistent data set is observed. The reason for this may be that the uncertainty arising from the experimental conditions in numerical modeling is not reflected in the numerical solution. In addition, in the range where the heading angle is 45° and 105° , the numerical results were found to be about 0.5 N more than experimental $\bar{Y}_W$.

Figure 4.15 depicts the relationship between empirical/analytical approaches, EFD, and CFD for yaw moments. The points in both experimental and computational results run in parallel, indicating a strong consistency between the two datasets. While the overall trends are similar, there are slight deviations at certain points, especially at $\psi = 75^\circ$ and $\psi = 150^\circ$ heading angles.

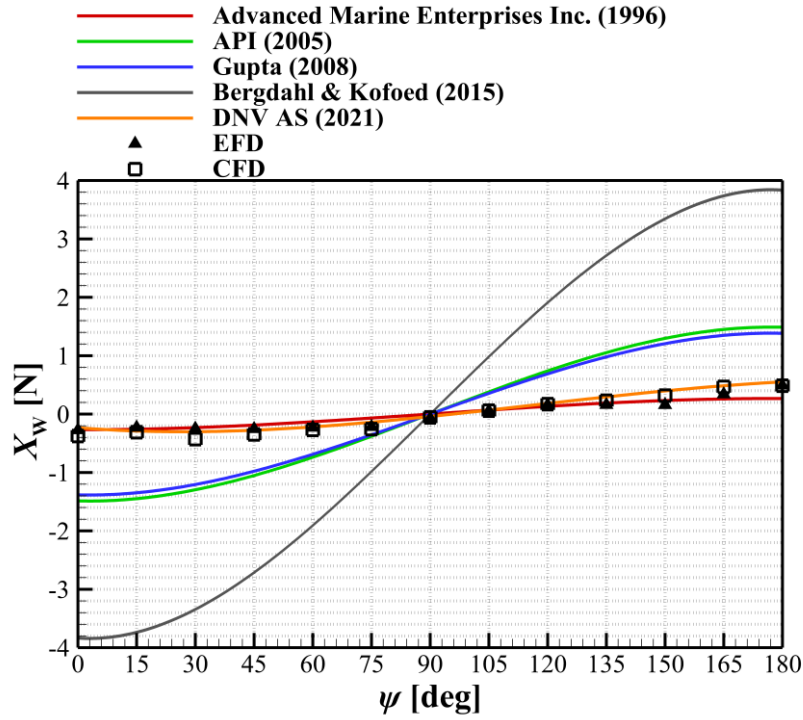


Figure 4.13 : Comparison of longitudinal wave drift forces at $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$.

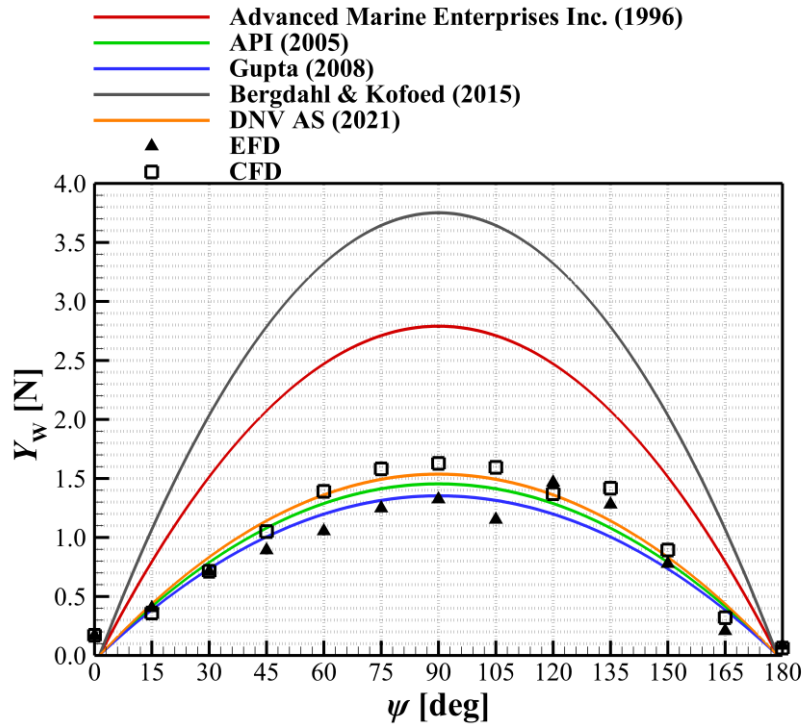


Figure 4.14 : Comparison of lateral wave drift forces at $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$.

Table 4.8 displays the longitudinal, lateral forces, and yaw moment results obtained from computational analyses for $\psi = 0^\circ - 180^\circ$. Eventually, the findings from the computational study show that ship motions in regular waves can be modeled at

different heading angles and used as a stand for particularly long experimental studies, provided that the verification study is carried out.

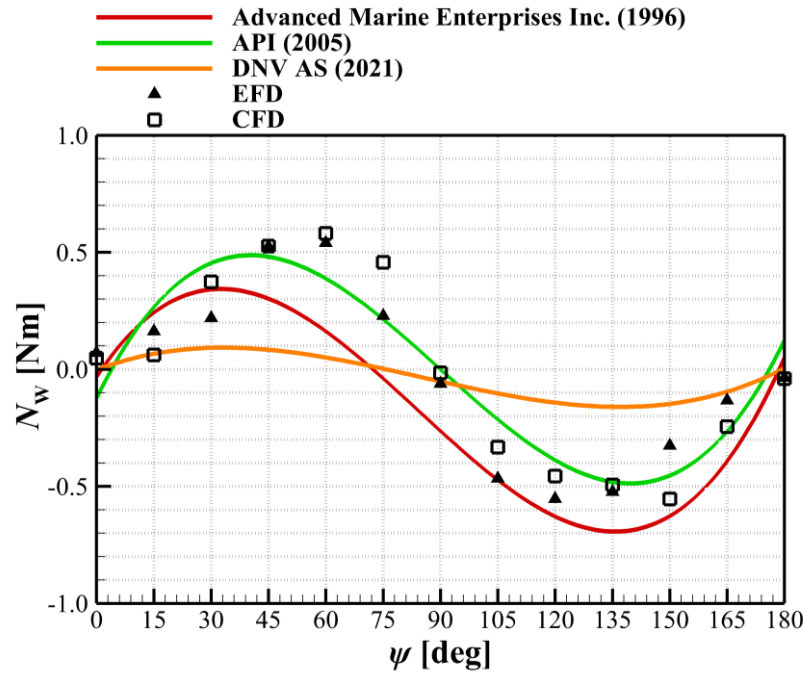


Figure 4.15 : Comparison of yaw moment at $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$.

Table 4.8 : Wave drift loads of steady drift simulations with various heading angles ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

ψ [deg]	$\bar{X}_W$ [N]	$\bar{Y}_W$ [N]	$\bar{N}_W$ [Nm]
0	-0.375	0.171	0.048
15	-0.311	0.360	0.062
30	-0.429	0.713	0.374
45	-0.349	1.050	0.527
60	-0.271	1.391	0.581
75	-0.258	1.581	0.457
90	-0.057	1.628	-0.015
105	0.059	1.594	-0.332
120	0.171	1.371	-0.456
135	0.228	1.417	-0.494
150	0.317	0.894	-0.554
165	0.465	0.320	-0.245
180	0.486	0.063	-0.040

4.4 Direct Numerical Simulation of Turning Trajectory in Calm Water

The standard maneuvering tests are conducted in model scale at an advance velocity ($U_a = 1.111 \text{ m/s}$) corresponding to $Fr = 0.20$. The ONRT model is simulated under steady turns with the rudder deflections of 35° using a constant speed propeller ($n =$

9.0 *rps*). The main purpose of direct computational simulations is to predict the trajectory as a result of the prescribed motion of the control surfaces, i.e. rudders. In the turning circle study, the focus is on the rudder angle of 35° with a constant velocity. Computation is performed to the port side using maximum rudder angles without changing the advance velocity. The general procedure indicating decision points is represented in Figure 4.16 as a flow chart. Generally, the ONRT is brought to a steady course and velocity according to the propeller rotation rate determined through the self-propulsion point. At this stage, the initial velocity of the ship is $U = 0 \text{ m/s}$ and the rudder angle, δ , is in the initial position. The hull is constrained to move only in the surge (x -axis) direction. As soon as the hull velocity is stable at the advance velocity (U_a), the rudder starts to turn, the turning is adjusted to reach the maximum rudder angle in 1 second. At the same time, the hull is allowed additionally to move in all directions. With the rudder reaching the maximum angle, the hull turn continues until a 360° change of heading is completed.

As explained in the flowchart, it can be seen in Figure 4.17 that the ship, which has a speed of 0 m/s at the time $t = 0$, reaches 1.111 m/s at the time t_i . The $0 - t_i$ time interval can be called the steady part. Immediately afterward, in the section stating the unsteady part, it was visualized that the hull started the turning maneuver when the rudder was given an angle of 35 degrees.

The numerical modeling is not detailed, because the setup is largely similar to steady drift simulations. The test conditions are summarized below:

- Two coordinate systems are defined for the turning tests of the ONRT, as in steady drift tests.
- The simulation is performed using the RANS method and by adopting a realizable two-layer $k - \varepsilon$ scheme.
- The free water surface, i.e. flat wave, is modeled with the VOF method.
- The parameters of the boundary layer are set suitable for y^+ to be around 1.
- 6-DoF motion during maneuvering is applied in the DFBI rotation and translation module.
- The initial velocity is selected as $U = 0 \text{ m/s}$.

- The time step and inner iteration are selected as $\Delta t = 0.04 \text{ s}$, and 50, respectively. This almost corresponds to 100 iteration for one propeller rotation at 9 rps rotating rate (Kinaci et al., 2021).

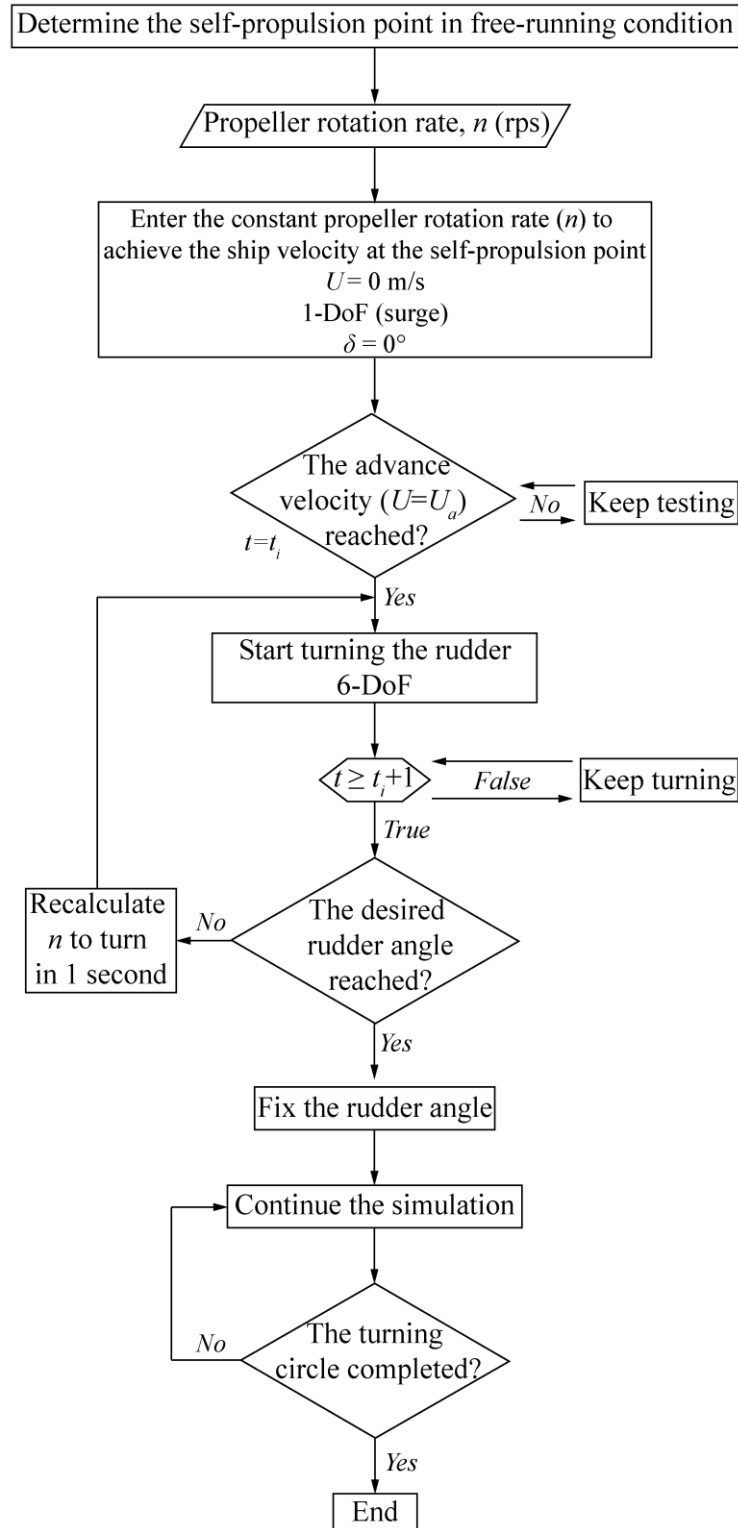


Figure 4.16 : Flow chart of direct numerical analysis, indicating decision points.

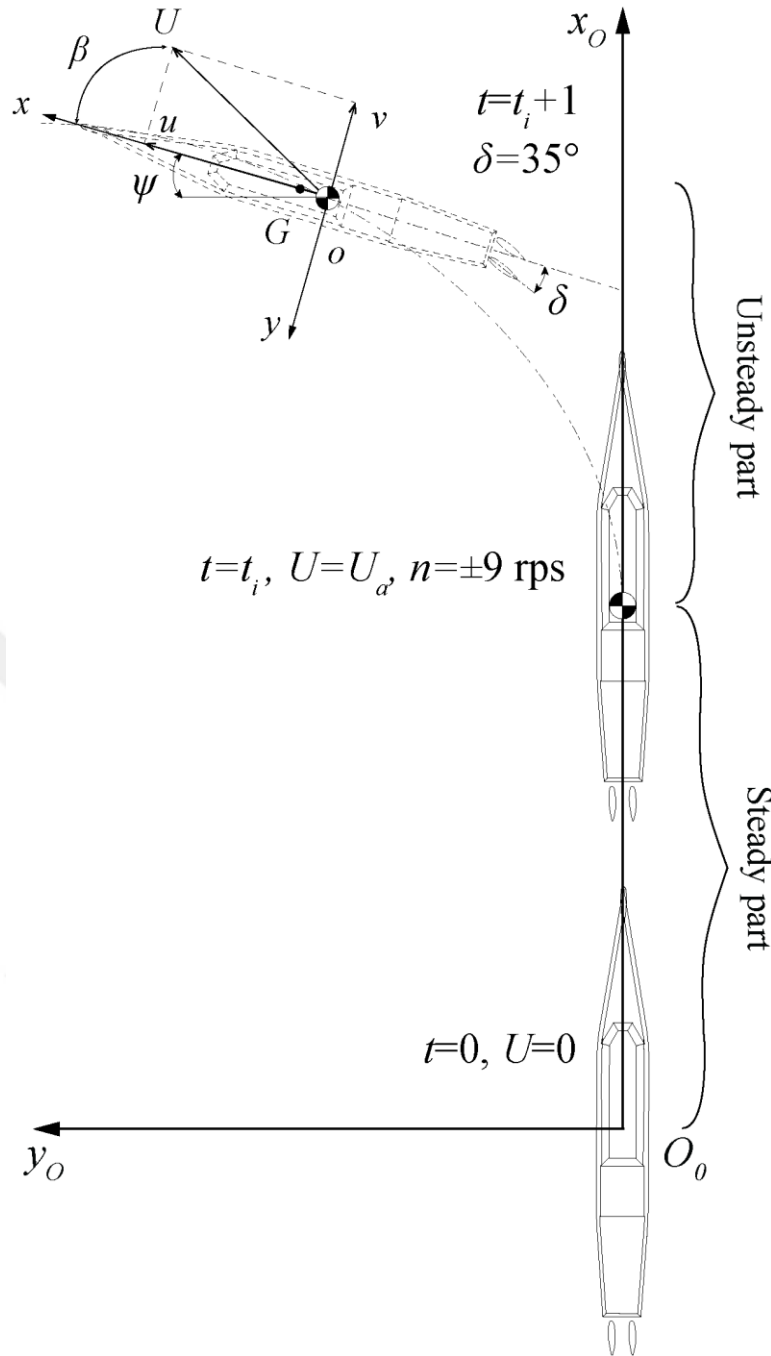


Figure 4.17 : Representation of the direct numerical analysis.

The rectangular computational domain is $4.8L_{WL} \times 3.2L_{WL} \times 1.3L_{WL}$ in the stream, transverse and cross-stream directions, respectively. The center of gravity of the hull is located at a distance of $1.6L_{WL}$ downwind from the inlet boundary and at a distance of $1.1L_{WL}$ from the bottom boundary. The computational domain with boundary conditions and dimensions is depicted in Figure 4.18. The downstream boundary in front of the hull, side, top and bottom boundaries have the velocity inlet boundary condition. The only pressure-based outlet boundary condition is for the upstream

boundary to achieve the mass conversation at each time step. The moving hull with control surfaces and propeller has the non-slip boundary condition in the calculations.

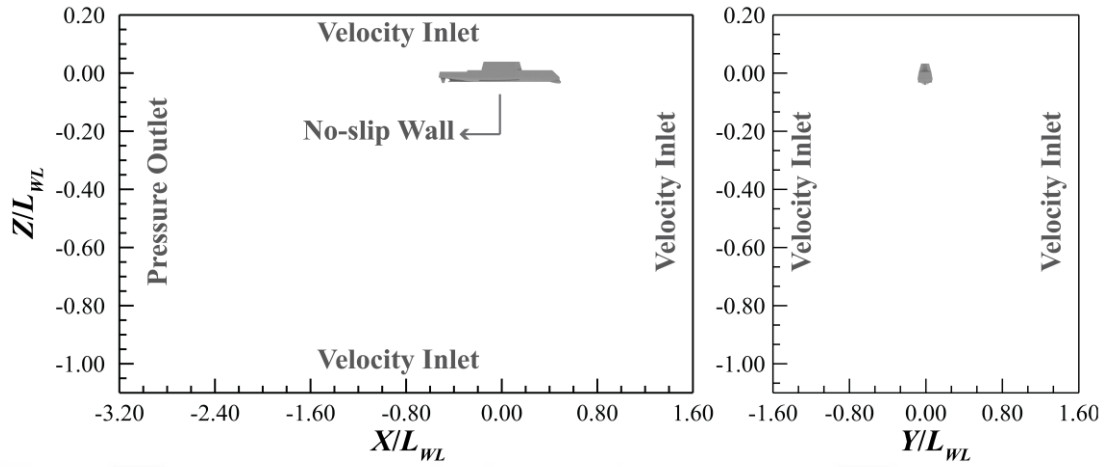


Figure 4.18 : The dimensions of the computational domain and boundary conditions in direct numerical analysis from XZ and YZ planes.

Directly-solved maneuvering simulations inherently involve many moving parts. In order to simulate fluid flows and analyze the flow characteristics correctly, the refinement volumes need to be defined around the moving parts. Apart from the main computational domain, six different refinement volumes are imposed in the present study. The rectangular-shaped one of the four volumes defined as the free surface refinement is arranged to involve the free surface from the profile view. For the flow survey behind the propeller and control surfaces, a rectangular refinement volume is also specified, namely Hull - Stern. Other refinement volumes that are improved with high-quality mesh are placed to surround the propellers, and rudders separately. To elaborate, the six refinement volumes with different cross-sectional shapes and dimensions are listed in Table 4.9.

Table 4.9 : Refinement volumes for the applied grid at direct numerical analysis.

Refinement	Cross-section type	Dimension in m
		$(L_{ref} \times W_{ref} \times H_{ref} \text{ or } H_{ref} \times D_{ref})$
Hull - Stern	Rectangular	0.75×0.3×0.3
Free surface	Rectangular	15×10×0.04
Propeller - Port	Circular	0.05×0.11
Propeller - Stb	Circular	0.05×0.11
Rudder - Port	Rectangular	0.15×0.03×0.15
Rudder - Stb	Rectangular	0.15×0.03×0.15

ONRT turning maneuver simulations established by superposing the rotational motions of the propeller and rudders are solved with the DFBI module integrated into the 1-DoF and 6-DoF motion solver. For the module to be active in rotating elements, interfaces to the moving parts must be defined in a hierarchical order. As can be seen in Figure 4.19, propeller interfaces are defined to provide the rotation of the propellers with MRF, and a total of two rudder interfaces to rotate the rudders at 35 degrees with the MRF.

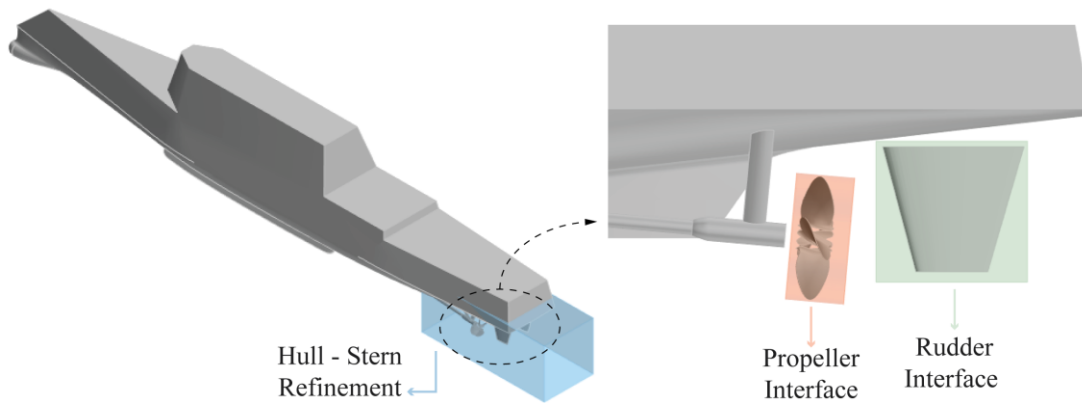


Figure 4.19 : Refinement volumes and interfaces at direct numerical analysis.

The fluid domain was discretized using the hexahedral element meshing approach by using the trimmer and surface remesher in the software. Afterward, an unstructured and predominantly hexahedral mesh was built over non-slip walls by making local improvements. 7.438E+06 elements in total, including 7.033E+06 cells for the background region, were created to solve the numerical problem. The detail of the grids at direct numerical simulation is listed in Table 4.10.

Table 4.10 : Element counts at direct numerical simulation.

Name	Element count	Type
Background Domain	7.033E+06	cells
Bottom	9.880E+02	faces
Inlet	4.302E+04	faces
Outlet	1.075E+04	faces
Top	1.721E+04	faces
Hull	2.503E+05	faces
Rudders Domain	2.280E+04	cells
Rudders	4.980E+03	faces
Propellers Domain	3.540E+04	cells
Propellers	7.880 E+03	faces
Total	7.438E+06	element

One degree of freedom and six degrees of freedom dynamics body motion equations coupled with flow equations were resolved by the DFBI module. Only x -motion of the body in the straight-ahead condition and 6-DoF motion during the turning (x -motion, y -motion, z -motion, x -rotation, y -rotation, and z -rotation) are entered. Apart from the motion of the hull, there are other moving parts, namely the rudders and the propellers, which both follow the hull motion and have certain rotations. In relation to the 6-DoF motion of the body, the predefined rotational motion of the propeller and control surfaces can be superposed. STAR-CCM+ offers several methodologies as a moving reference frame (MRF) approach and rigid body motion (RBM) for the rotation of a body in solid-liquid flows. While each includes a capability for rotating motion, the right choice should be made by focusing on the flow evaluation and calculation intensity. The former effectively imposes the forces that are induced by the rotation, while the second account for slides of the mesh by interfaced region during the motion. MRF method can be preferred to reduce the computational cost, for example, transient simulation is not required for propeller rotation cases because motion is included in the flow equations (Kinaci & Ozturk, 2022). Unlike the MRF method, the RBM is considered to be more robust and gives more substantial outcomes, but the computational cost is higher to a considerable extent. By synthesizing the insights from the study by Kinaci & Ozturk (2023), a more comprehensive understanding of the direct numerical simulation is achieved. In the previous study, the calm water condition was investigated by direct CFD methods, and the rotations of moving parts were successfully adapted via the moving reference frame. Therefore, rudder and propeller motions are resolved by the moving reference frame approach.

The turning trajectory and time-dependent parameters will be discussed in comparison with other solution methods in chapter 5.7.3.



5. SYSTEM-BASED INVESTIGATION

System-based ship maneuvering simulation methods are used to predict the behavior of ships in different operating conditions and to evaluate the effectiveness of ship control systems. The methods typically involve mathematical models that describe the motion of the ship in different degrees of freedom, taking into account the hydrodynamic forces and moments.

A research team named Maneuvering Modeling Group (MMG) presented a method for ship maneuvering simulations at the Japanese Towing Tank Conference (JTTC). The MMG model, which concentrates on the interaction of the hull, propeller, and rudder, is recognized as a representative model to describe ship maneuvers. The 3-DoF MMG standard method, solving the motion equations of the surge, sway, and yaw, composed of four elements is introduced by Yasukawa & Yoshimura (2015). The four elements in the model are the maneuvering simulation model, the procedure of the necessary captive model tests capturing the hydrodynamic forces, the analysis method for figuring out the hydrodynamic force coefficients, and the forecast method for maneuvering motions in full scale. The model was extended to a 4-DoF model in 2019 by adding the roll-coupling effect (Yasukawa et al., 2019). Also, Guo & Zou (2017) preferred a system-based method with the MMG model to examine 4-DoF ship maneuvering motion in calm water for the ONRT model. It is generally known that while maneuvering, especially during hard turning, visible roll motion can be detected for vessels with high service speeds and small metacentric heights such as surface combatants, large container ships, and passenger ships. Since the surface combatant ONR tumblehome is a twin-screw ship, expressions describing the hull-propeller-rudder interaction should also be added to the system-based methods. In accordance with the 3-DoF modeling approach described by Yasukawa & Yoshimura (2015), and the 4-DoF modeling approach depicted by Guo & Zou (2017) and Guo et al. (2018) considered the propeller side force and asymmetric flow-straightening effects for the twin screw ship.

The study by Kinaci & Ozturk (2023) employs the mathematical model neglecting the vertical accelerations due to relatively small heave, roll, and pitch compared to surge, sway, and yaw. Therefore, 6-DoF ship motions were reduced to 3-DoF with this assumption. Paper serves as a foundational work, and the thesis extends the research in this field by exploring the 4-DoF MMG model. The mathematical model in the paper and the thesis are based on the abovementioned literature. In the present study, the system-based method with the 4-DoF MMG model is improved by considering the wave-induced drift forces and moments. The equations obtained from the values of the experimental wave drift loads at different ship heading angles are included in the mathematical model as a wave model. The hydrodynamic derivatives are obtained from different literature that studied the ONRT model and are presented in the following sections. Firstly, the results of the model in calm water were examined and compared with the literature. By adding the wave model, different trajectories and kinematic/dynamic quantities were obtained. The findings at different wave incidence angles are compared with the available experimental and numerical results.

5.1 Basic Assumptions

In the system-based mathematical models, several assumptions are commonly made to simplify the mathematical representation of ship maneuvering motion and to facilitate analysis and control. Some of these assumptions include:

Rigid-body assumption: The ship is thought to behave like a rigid body, hence deformations and flexibility of the hull and other components are ignored.

Quasi-steady flow assumption: The surrounding water flow is assumed to be quasi-steady, which means that it does not change considerably during maneuvering.

Negligible wave effects in calm water: The effects of waves and wave-induced forces are frequently ignored or oversimplified, particularly for short-term maneuvering simulations

Absence of external disturbances: External disturbances of wind, and current are neglected.

The forward velocity: It is assumed that the maneuvering derivatives are independent of the forward velocity of the ship.

The wave radiation: Wave radiation due to the motions in surge, sway and yaw is ignored (assumed to be weak in slender bodies).

Wave excitation: Only mean wave drift forces and moment are considered as wave excitation.

Linear and non-linear terms: It is assumed that the variation in ship motions is small and hence linear and non-linear terms can be separated.

The above-mentioned assumptions facilitate the modeling process and permit the application of standard rigid body dynamics equations.

5.2 Coordinate System in System-Based Model

For the mathematical model of the maneuvering, two right-handed coordinate systems, one earth-fixed coordinate system, and one body-fixed coordinate system are defined. As shown in Figure 3.8, the earth-fixed coordinate system $O_0 - x_0y_0z_0$ that is set to be fixed with the origin and the body-fixed coordinate system $o - xyz$ that is fixed on the hull. There is a difference in the global coordinate system in S-B simulations, and the x and y axes are used interchangeably, unlike the experimental coordinate system. Details about the coordinate system can be found in the study by Sukas et al. (2019).

The origin of the body-fixed coordinate system $o - xyz$ is positioned at the midship, and the x -axis is pointed to the bow. The x and y axes correspond to the longitudinal and lateral directions of the ship, respectively. The vertical axis is perpendicular to both the x and y axes in accordance with the right-hand rule. The center of gravity's trajectory serves as an indication of the ship's course. The drift angle, β , and the heading angle ψ , change by degrees depending on rudder motion. The center of gravity of the ship, G , is located at $(x_G, 0, 0)$ in $o - xyz$ system.

The advance speed, U , of translational motion, has two components as u and v in the directions of the x -axis and y -axis, respectively. The drift angle, β , denotes the direction of the ship's longitudinal axis with respect to the vector of $\vec{U}$. As a result, $u = U\cos\beta$ and $v = -U\sin\beta$ can be used to compute the x - and y -components of $\vec{U}$, respectively. Also, the yaw rate ($r = \dot{\psi}$) is defined as the rate of change of angular heading. The velocity term p denotes the rotational motion about the z -axis and is equal to the derivative of roll motion, $\dot{\phi}$. The dot notation is used for differentiation by

placing a dot over the dependent value. The terminology of motion, velocity, and force/moment components are presented in Table 5.1 to better follow up on the mathematical model.

Table 5.1 : Motion, velocity, and force/moment components in the x , y , and z directions.

Degree of freedom	Motion component	Axis	Force/Moment component	Velocity component
1	Surge	x	X	u
2	Sway	y	Y	v
3	Heave	z	Z	w
4	Roll	φ	K	p
5	Pitch	θ	M	q
6	Yaw	ψ	N	r

5.3 Non-Dimensionalization

Non-dimensionalization is a mathematical technique used to convert a set of equations into a dimensionless form, thereby making them easier to solve. In essence, it is a way of reducing the complexity of a problem by replacing all the variables with a set of dimensionless combinations. It is also used to simplify equations of motion and to analyze the behavior of a system with respect to changes in its parameters. Parameters in the following sections are non-dimensionalized as given by the expressions below. Putting the symbol prime (') means that the value is non-dimensionalized (Kinaci & Ozturk, 2023).

$$F' = F/0.5\rho L_{pp}dU^2$$

$$M' = M/0.5\rho L_{pp}^2dU^2$$

$$m' = m/0.5\rho L_{pp}^2d$$

$$I' = I/0.5\rho L_{pp}^4d \quad (5.1)$$

$$u' = u/U$$

$$l' = l/L_{pp}$$

$$r' = rL_{pp}/U$$

where F is force, M is the moment, m is mass, and I is the moment of inertia. The parameters u , l , and, r denote the velocity component, length component, and rate of turn around an axis, respectively.

5.4 Derivation of Equation of Motion with Six Degrees of Freedom

The equations of motion of a rigid body with respect to its center of gravity can be derived from Newton's equations of motion (Abkowitz, 1964):

$$\vec{F} = m \frac{d\vec{V}_G}{dt} \quad (5.2)$$

$$\vec{M} = I \frac{d\vec{\Omega}}{dt}$$

where m is the mass of the rigid body, $\vec{V}_G$ is the relative velocity vector with respect to the center of gravity, I is the mass moment of inertia, and $\vec{\Omega}$ is the angular velocity vector.

The velocity vector relative to the center of gravity can be written as:

$$\vec{V}_G = \vec{V} + \vec{\Omega} \times \vec{R}_G \quad (5.3)$$

The vectors in terms of their components can be written in ijk notation:

$$\begin{aligned} \vec{V} &= u\vec{i} + v\vec{j} + w\vec{k} \\ \vec{\Omega} &= p\vec{i} + q\vec{j} + r\vec{k} \end{aligned} \quad (5.4)$$

$$\vec{R}_G = x_G\vec{i} + y_G\vec{j} + z_G\vec{k}$$

The velocity components of the surge, sway, and heave are stated as u , v , and w , respectively. The velocity components of the roll, pitch, and yaw are given as p , q , and r , respectively $(\dot{\theta}, \dot{\phi}, \dot{\psi})$. The coordinates of the center of gravity relative to the midship are denoted by x_G , y_G , z_G , respectively.

Substituting equations 5.3 and 5.4 into equation 5.2 gives the following force expression:

$$\vec{F} = m \frac{d\vec{V}_G}{dt} = m \frac{d}{dt}(\vec{V} + \vec{\Omega} \times \vec{R}_G)$$

$$\vec{F} = m \left(\frac{d\vec{V}}{dt} + \vec{\Omega} \times \vec{V} + \frac{d\vec{\Omega}}{dt} \times \vec{R}_G + \vec{\Omega} \times \frac{d\vec{R}_G}{dt} \right)$$

$$\frac{d\vec{V}}{dt} + \vec{\Omega} \times \vec{V} = \dot{u}\vec{i} + \dot{v}\vec{j} + \dot{w}\vec{k} + \begin{vmatrix} \vec{i} & \vec{j} & \vec{k} \\ p & q & r \\ u & v & w \end{vmatrix}$$

$$\frac{d\vec{V}}{dt} + \vec{\Omega} \times \vec{V} = (\dot{u} + wq - vr)\vec{i} +$$

$$(\dot{v} + ur - wp)\vec{j} +$$

$$(\dot{w} + vp - uq)\vec{k}$$

$$\frac{d\vec{\Omega}}{dt} \times \vec{R}_G = \begin{vmatrix} \vec{i} & \vec{j} & \vec{k} \\ \dot{p} & \dot{q} & \dot{r} \\ x_G & y_G & z_G \end{vmatrix} = (z_G\dot{q} - y_G\dot{r})\vec{i} +$$

$$(x_G\dot{r} - z_G\dot{p})\vec{j} +$$

$$(y_G\dot{p} - x_G\dot{q})\vec{k}$$

(5.5)

$$\frac{d\vec{R}_G}{dt} = \vec{\Omega} \times \vec{R}_G = \begin{vmatrix} \vec{i} & \vec{j} & \vec{k} \\ p & q & r \\ x_G & y_G & z_G \end{vmatrix} = (z_Gq - y_Gr)\vec{i} +$$

$$(x_Gr - z_Gp)\vec{j} +$$

$$(y_Gp - x_Gq)\vec{k}$$

$$\vec{\Omega} \times \frac{d\vec{R}_G}{dt} = \begin{vmatrix} \vec{i} & \vec{j} & \vec{k} \\ p & q & r \\ z_Gq - y_Gr & x_Gr - z_Gp & y_Gp - x_Gq \end{vmatrix}$$

$$\vec{\Omega} \times \frac{d\vec{R}_G}{dt} = (y_Gpq - x_Gq^2 - x_Gr^2 + z_Gpr)\vec{i} +$$

$$(z_Gqr - y_Gr^2 - y_Gp^2 + x_Gpq)\vec{j} +$$

$$(x_Gpr - z_Gp^2 - z_Gq^2 + y_Gqr)\vec{k}$$

By substituting the found terms in the expression, the following equations are obtained.

$$\begin{aligned}
\vec{F} = & m(\dot{u} + wq - vr + z_G\dot{q} - y_G\dot{r} + y_Gpq - x_Gq^2 - x_Gr^2 + z_Gpr)\vec{i} + \\
& m(\dot{v} + ur - wp + x_G\dot{r} - z_G\dot{p} + z_Gqr - y_Gr^2 - y_Gp^2 + x_Gpq)\vec{j} + \\
& m(\dot{w} + vp - uq + y_G\dot{p} - x_G\dot{q} + x_Gpr - z_Gp^2 - z_Gq^2 + y_Gqr)\vec{k} \\
\vec{F} = & m[\dot{u} + wq - vr - x_G(q^2 + r^2) + y_G(pq - \dot{r}) + z_G(pr + \dot{q})]\vec{i} + \\
& m[\dot{v} + ur - wp - y_G(p^2 + q^2) + z_G(qr - \dot{p}) + x_G(pq + \dot{r})]\vec{j} + \\
& m[\dot{w} + vp - uq - z_G(p^2 + q^2) + x_G(pr - \dot{q}) + y_G(qr + \dot{p})]\vec{k}
\end{aligned} \tag{5.6}$$

The force expression can be decomposed as follows:

$$\begin{aligned}
X = & m[\dot{u} + wq - vr - x_G(q^2 + r^2) + y_G(pq - \dot{r}) + z_G(pr + \dot{q})] \\
Y = & m[\dot{v} + ur - wp - y_G(p^2 + r^2) + z_G(qr - \dot{p}) + x_G(pq + \dot{r})] \\
Z = & m[\dot{w} + vp - uq - z_G(p^2 + q^2) + x_G(pr - \dot{q}) + y_G(qr + \dot{p})]
\end{aligned} \tag{5.7}$$

The external force components in x , y , and z directions are stated as X , Y , and Z , respectively. The acceleration components of the surge, sway, and heave are given as $\dot{u}$, $\dot{v}$, and $\dot{w}$, respectively. The acceleration components of the roll, pitch, and yaw are given as $\dot{p}$, $\dot{q}$, and $\dot{r}$, respectively ($\ddot{\theta}$, $\ddot{\phi}$, $\ddot{\psi}$).

The moment expression can be extracted using a similar approach as follows:

$$\begin{aligned}
\vec{F} = m \frac{d\vec{V}_G}{dt} = m \frac{d}{dt}(\vec{V} + \vec{\Omega} \times \vec{R}_G) \\
\vec{M}_G = \vec{M} + \vec{R}_G \times \vec{F} = I_G \frac{d\vec{\Omega}}{dt} + \vec{R}_G \times \vec{F} \\
\vec{M}_G = I_{XG}\dot{p}\vec{i} + I_{YG}\dot{q}\vec{j} + I_{ZG}\dot{r}\vec{k} + \begin{vmatrix} \vec{i} & \vec{j} & \vec{k} \\ x_G & y_G & z_G \\ X & Y & Z \end{vmatrix} \\
\vec{M}_G = (I_{XG}\dot{p} + Z y_G - Y z_G)\vec{i} + \\
(I_{YG}\dot{q} + X z_G - Z x_G)\vec{j} + \\
(I_{ZG}\dot{r} + Y x_G - X y_G)\vec{k}
\end{aligned} \tag{5.8}$$

The moment expression can be decomposed as follows:

$$\begin{aligned}
K &= I_{x_G}\dot{p} + Zy_G - Yz_G \\
M &= I_{y_G}\dot{q} + Xz_G - Zx_G \\
N &= I_{z_G}\dot{r} + Yx_G - Xy_G
\end{aligned} \tag{5.9}$$

The external moment components in θ , ϕ , and ψ directions are stated as K , M , and N , respectively. Components of the mass moment of inertia with respect to the center of gravity are denoted by I_{x_G} , I_{y_G} , and I_{z_G} , respectively.

The following expressions can be used for X , Y , and Z :

$$\begin{aligned}
X &= m[\dot{u} + wq - vr - x_G(q^2 + r^2) + y_G(pq - \dot{r}) + z_G(pr + \dot{q})] \\
Y &= m[\dot{v} + ur - wp - y_G(p^2 + r^2) + z_G(qr - \dot{p}) + x_G(pq + \dot{r})] \\
Z &= m[\dot{w} + vp - uq - z_G(p^2 + q^2) + x_G(pr - \dot{q}) + y_G(qr + \dot{p})]
\end{aligned} \tag{5.10}$$

The yaw moment is obtained as follows:

$$\begin{aligned}
K &= I_{x_G}\dot{p} + my_G(\dot{w} + vp - uq) - my_Gz_Gp^2 - my_Gz_Gq^2 + \\
&\quad mx_Gy_Gpr - mx_Gy_G\dot{q} + my_G^2qr + my_G^2\dot{p} - \\
&\quad mz_G(\dot{v} + ur - wp) + my_Gz_Gp^2 + mz_G^2\dot{p} - \\
&\quad my_Gz_Gr^2 - mz_G^2qr + mx_Gz_Gpq - mx_Gz_G\dot{r}
\end{aligned} \tag{5.11}$$

This expression can be edited as follows:

$$\begin{aligned}
K &= [I_{x_G} + m(y_G^2 + z_G^2)]\dot{p} - mx_Gy_G(\dot{q} - pr) - \\
&\quad mx_Gz_G(\dot{r} + pq) - my_Gz_G(q^2 - r^2) - \\
&\quad m(z_G^2 - y_G^2)qr + \\
&\quad m[y_G(\dot{w} + vp - uq) - z_G(\dot{v} + ur - wp)]
\end{aligned} \tag{5.12}$$

According to the parallel axis theorem

$$\begin{aligned}
 I_{xG} &= I_x - m(y_G^2 + z_G^2) \\
 I_{xyG} &= I_{xy} + mx_G y_G = 0 \quad \rightarrow \quad mx_G y_G = -I_{xy} \\
 I_{xzG} &= I_{xz} + mx_G z_G = 0 \quad \rightarrow \quad mx_G z_G = -I_{xz} \\
 I_{yzG} &= I_{yz} + my_G z_G = 0 \quad \rightarrow \quad my_G z_G = -I_{yz}
 \end{aligned} \tag{5.13}$$

Substituting equation 5.13 into equation 5.12 gives the following moment expression:

$$\begin{aligned}
 K &= I_x \dot{p} + I_{xy}(\dot{q} - pr) + I_{xz}(\dot{r} + pq) + I_{yz}(q^2 - r^2) + \\
 &\quad (I_z - I_y)qr + m[y_G(\dot{w} + vp - uq) - z_G(\dot{v} + ur - wp)]
 \end{aligned} \tag{5.14}$$

Similarly, other moment expressions are obtained as follows:

$$\begin{aligned}
 M &= I_y \dot{q} + I_{yz}(\dot{r} - qp) + I_{yx}(\dot{p} + qr) + I_{zx}(r^2 - p^2) + \\
 &\quad (I_x - I_z)pr + m[z_G(\dot{u} + wq - rv) - x_G(\dot{w} + vp - uq)] \\
 N &= I_z \dot{r} + I_{zx}(\dot{p} - qr) + I_{zy}(\dot{q} + pr) + I_{xy}(p^2 - q^2) + \\
 &\quad (I_y - I_x)pq + m[x_G(\dot{v} + ur - pw) - y_G(\dot{u} + wq - vr)]
 \end{aligned} \tag{5.15}$$

Here I_x , I_y , and I_z represent moments of inertia and are written as follows:

$$\begin{aligned}
 I_x &= \sum_{i=1}^N m_i(y_i^2 + z_i^2) \\
 I_y &= \sum_{i=1}^N m_i(x_i^2 + z_i^2) \\
 I_z &= \sum_{i=1}^N m_i(x_i^2 + y_i^2)
 \end{aligned} \tag{5.16}$$

The cross-inertial terms, I_{XY} , I_{YX} , I_{XZ} , I_{ZX} , I_{YZ} , and I_{ZY} are

$$\begin{aligned}
I_{XY} = I_{YX} &= - \sum_{i=1}^N m_i x_i y_i \\
I_{XZ} = I_{ZX} &= - \sum_{i=1}^N m_i x_i z_i \\
I_{YZ} = I_{ZY} &= - \sum_{i=1}^N m_i y_i z_i
\end{aligned} \tag{5.17}$$

5.5 Derivation of Equation of Motion with Four Degrees of Freedom

The modular maneuvering mathematical model MMG with 4-DoF is used in this study to simulate the maneuvering motions of surface combatant ONR Tumblehome in the horizontal plane. A set of equations are described to express the 4-DoF maneuvering motion comprising surge, sway, roll, and yaw. Therefore, surge, sway, roll, and yaw motions can be left out of 6-DoF motion equations. The equations are based on a non-inertial body-fixed coordinate system with the origin positioned at the ship's center of gravity. The object hull is symmetric; so, $y_G = 0$. Under the assumptions, the equations of motion with 4-DoF can be written as follows (Hamamoto & Kim, 1993):

$$\begin{aligned}
F_x &= m(\dot{u} - v_G r) \\
F_y &= m(\dot{v} + u_G r) \\
M_x &= I_{xx} \dot{p} + (I_{zz} - I_{yy}) r^2 \sin \varphi \cos \varphi - I_{xz} \dot{r} \cos \varphi \\
M_z &= (I_{zz} \cos^2 \varphi + I_{yy} \sin^2 \varphi) \dot{r} - (I_{zz} - I_{yy}) r \dot{\varphi} \sin(2\varphi) - \\
&\quad I_{xz} (\dot{p} \cos \varphi - p^2 \sin \varphi)
\end{aligned} \tag{5.18}$$

where u_G , and v_G are the surge and lateral velocity at the center of gravity, respectively. In slender ships, the approximation is made as $I_{zz} \simeq I_{yy}$ and $I_{xz} \simeq 0$. Then, equation 5.18 is simplified as,

$$\begin{aligned}
F_x &= m(\dot{u} - v_G r) \\
F_y &= m(\dot{v} + u_G r) \\
M_x &= I_{xx} \dot{p} \\
M_z &= I_{zz} \dot{r}
\end{aligned} \tag{5.19}$$

Here, F_x , F_y , M_x , and M_z are expressed as follows (Yasukawa et al., 2019):

$$\begin{aligned}
F_x &= -m_x \dot{u} + m_y vr + X \\
F_y &= -m_y \dot{v} - m_x ur - a_{24} \dot{p} - a_{26} \dot{r} + Y \\
M_x &= -(J_{xx} + z_G a_{24}) \dot{p} - (a_{42} + z_G m_y) \dot{v} - (a_{46} + z_G a_{26}) \dot{r} - \\
&\quad m_x z_G ur + K + z_G Y \\
M_z &= -(J_{zz} - x_G a_{26}) \dot{r} - (a_{62} - x_G m_y) \dot{v} - (a_{64} - x_G a_{24}) \dot{p} + \\
&\quad m_x z_G ur + N - x_G Y
\end{aligned} \tag{5.20}$$

Here, the surge force and lateral force are denoted by X and Y , respectively. The added mass components are excluded in X and Y . The roll moment around x -axis and the yaw moment around z -axis are denoted by K and N , respectively. The added moment of inertia components are excluded in K and N . x_G is the longitudinal coordinate of the center of gravity and z_G is the vertical coordinate of the center of gravity. m_x and m_y are the added mass for surge and sway, respectively. J_{xx} and J_{zz} are the added moment of inertia for roll and yaw, respectively. The coupled added mass components for sway, roll, and yaw motions are respectively denoted as a_{24} , a_{26} and a_{46} . The coupled added mass components are taken as follows for simplicity:

$$\begin{aligned}
a_{26} &= a_{62} \simeq 0 \\
a_{24} &= a_{42} \equiv -m_y \alpha_z \\
a_{46} &= a_{64} \simeq 0
\end{aligned} \tag{5.21}$$

Here, α_z is the vertical acting point of the lateral added mass component, m_y .

Assuming that the position of the center of gravity for a ship with starboard-port symmetry, the velocity components at the center of gravity can be written as:

$$\begin{aligned} u_G &= u \\ v_G &= v + x_G r - z_G p \end{aligned} \quad (5.22)$$

where u and v are surge and lateral velocity at midship. Substituting equations 5.20, 5.21, and 5.22 into 5.19 gives the equations of motion in the horizontal plane relative to the axis passing through the midship are obtained as follows:

$$\begin{aligned} (m + m_x)\dot{u} - (m + m_y)vr - x_G mr^2 + z_G mpr &= X \\ (m + m_y)\dot{v} - (m_y \alpha_Z + z_G m)\dot{p} + (m + m_x)ur + x_G m\dot{r} &= Y \\ -(m_y \alpha_Z + z_G m)\dot{v} + (I_x + J_x + z_G^2 m)\dot{p} - x_G z_G m\dot{r} - z_G mur &= K \\ (I_z + x_G^2 m + J_z)\dot{r} - (x_G z_G m)\dot{p} + x_G m(\dot{v} + ur) &= N \end{aligned} \quad (5.23)$$

where m is the mass of the ship, I_x and I_z are the moments of inertia about the longitudinal axis and vertical axis, respectively; m_x and m_y are the added masses, J_x and J_z are the added moments of inertia. u and v are the surge and sway velocities, respectively. p and r are the roll and yaw angular velocities, respectively. $\dot{u}$, $\dot{v}$, $\dot{p}$, and $\dot{r}$ are the acceleration components of surge, sway, roll, and yaw motion, respectively. x_G , and z_G are longitudinal and vertical coordinates of the center of gravity of the ship. α_Z is the vertical acting point of the lateral added mass component, m_y . X , and Y denote the lateral and longitudinal components of external force, while K , and N denote the total roll and yaw moments, respectively.

If equation 5.23 is written in long form,

$$\begin{aligned} (m + m_x)(\dot{u}) + 0(\dot{v}) + 0(\dot{p}) + 0(\dot{r}) + \\ 0(u) - (m + m_y)r(v) + z_G mr(p) - x_G mr(r) &= X \\ 0(\dot{u}) + (m + m_y)\dot{v} - (m_y \alpha_Z + z_G m)(\dot{p}) + x_G m(\dot{r}) + \\ (m + m_x)r(u) + 0(v) + 0(p) + 0(r) &= Y \end{aligned} \quad (5.24)$$

$$\begin{aligned}
0(\ddot{u}) - (m_y \alpha_z + z_G m) \dot{v} + (I_x + J_x + z_G^2 m)(\ddot{p}) - x_G z_G m(\ddot{r}) + \\
0(u) + 0(v) + 0(p) - z_G m u(r) = K
\end{aligned} \tag{5.24a}$$

$$\begin{aligned}
0(\ddot{u}) + x_G m(\ddot{v}) - (x_G z_G m)(\ddot{p}) + (I_z + x_G^2 m + J_z)(\ddot{r}) + \\
0(u) + 0(v) + 0(p) + x_G m u(r) = N
\end{aligned}$$

Assuming $u = U$, $v = 0$, $p = 0$, and $r = 0$ to linearize the equations:

$$\begin{aligned}
(m + m_x)(\ddot{u}) + 0(\ddot{v}) + 0(\ddot{p}) + 0(\ddot{r}) + 0(u) - 0 * v + 0 * p - 0 \\
* r = X
\end{aligned}$$

$$\begin{aligned}
0(\ddot{u}) + (m + m_y) \dot{v} - (m_y \alpha_z + z_G m)(\ddot{p}) + x_G m(\ddot{r}) + \\
0(u) + 0(v) + 0(p) + (m + m_x)U(r) = Y
\end{aligned} \tag{5.25}$$

$$\begin{aligned}
0(\ddot{u}) - (m_y \alpha_z + z_G m) \dot{v} + (I_x + J_x + z_G^2 m)(\ddot{p}) - x_G z_G m(\ddot{r}) + \\
0(u) + 0(v) + 0(p) - z_G m U(r) = K
\end{aligned}$$

$$\begin{aligned}
0(\ddot{u}) + x_G m(\ddot{v}) - (x_G z_G m)(\ddot{p}) + (I_z + x_G^2 m + J_z)(\ddot{r}) + \\
0(u) + 0(v) + 0(p) + x_G m U(r) = N
\end{aligned}$$

Mathematical models can also be expressed in matrix forms:

$$\begin{aligned}
\begin{bmatrix} m + m_x & 0 & 0 & 0 \\ 0 & m + m_y & -(m_y \alpha_z + m z_G) & m x_G \\ 0 & -(m_y \alpha_z + m z_G) & (I_x + J_x + m z_G^2) & -m x_G z_G \\ 0 & m x_G & -m x_G z_G & I_z + J_z + m x_G^2 \end{bmatrix} \begin{bmatrix} \ddot{u} \\ \ddot{v} \\ \ddot{p} \\ \ddot{r} \end{bmatrix} \\
+ \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & (m + m_x)U \\ 0 & 0 & 0 & -z_G m U \\ 0 & 0 & 0 & x_G m U \end{bmatrix} \begin{bmatrix} u \\ v \\ p \\ r \end{bmatrix} = \begin{bmatrix} X \\ Y \\ K \\ N \end{bmatrix}
\end{aligned} \tag{5.26}$$

Rearrange the equation to leave the matrix $[\ddot{u} \quad \ddot{v} \quad \ddot{p} \quad \ddot{r}]^T$ alone on the left-hand side,

$$\begin{bmatrix} \dot{u} \\ \dot{v} \\ \dot{p} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} m + m_x & 0 & 0 & 0 \\ 0 & m + m_y & -(m_y \alpha_z + m z_G) & m x_G \\ 0 & -(m_y \alpha_z + m z_G) & (I_x + J_x + m z_G^2) & -m x_G z_G \\ 0 & m x_G & -m x_G z_G & I_z + J_z + m x_G^2 \end{bmatrix}^{-1} \cdot \left(\begin{bmatrix} X \\ Y \\ K \\ N \end{bmatrix} - \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & (m + m_x)U \\ 0 & 0 & 0 & -z_G m U \\ 0 & 0 & 0 & x_G m U \end{bmatrix} \begin{bmatrix} u \\ v \\ p \\ r \end{bmatrix} \right) \quad (5.27)$$

The matrix is simplified as follows:

$$\begin{bmatrix} \dot{u} \\ \dot{v} \\ \dot{p} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} m + m_x & 0 & 0 & 0 \\ 0 & m + m_y & -(m_y \alpha_z + m z_G) & m x_G \\ 0 & -(m_y \alpha_z + m z_G) & (I_x + J_x + m z_G^2) & -m x_G z_G \\ 0 & m x_G & -m x_G z_G & I_z + J_z + m x_G^2 \end{bmatrix}^{-1} \cdot \begin{bmatrix} X \\ Y - (m + m_x)Ur \\ K + m z_G Ur \\ N - m x_G Ur \end{bmatrix} \quad (5.28)$$

To find the equation for $[\dot{u} \ \dot{v} \ \dot{p} \ \dot{r}]^T$, the inverse of the first term on the right-hand side is taken and the acceleration terms can be written to find the linearized form of the ship motion equations.

Some abbreviations can be made after equation 5.26 to simplify acceleration terms:

$$M = \begin{bmatrix} m + m_x & 0 & 0 & 0 \\ 0 & m + m_y & -(m_y \alpha_z + m z_G) & m x_G \\ 0 & -(m_y \alpha_z + m z_G) & (I_x + J_x + m z_G^2) & -m x_G z_G \\ 0 & m x_G & -m x_G z_G & I_z + J_z + m x_G^2 \end{bmatrix} \quad (5.29)$$

$$= \begin{bmatrix} M_{11} & 0 & 0 & 0 \\ 0 & M_{22} & M_{23} & M_{24} \\ 0 & M_{32} & M_{33} & M_{34} \\ 0 & M_{42} & M_{43} & M_{44} \end{bmatrix}$$

The inverse of the above matrix is taken as follows:

$$M^{-1} = \frac{1}{|M|} \tilde{M} \quad (5.30)$$

where $\tilde{M}$ is the adjugate matrix of M , and $|M|$ is the determinant. The elements of the adjugate matrix of M are

$$\begin{aligned}
\tilde{M}_{11} &= M_{22}M_{33}M_{44} + M_{23}M_{34}M_{42} + M_{24}M_{32}M_{43} - \\
&\quad M_{24}M_{33}M_{42} - M_{23}M_{32}M_{44} - M_{22}M_{34}M_{43} \\
\tilde{M}_{22} &= M_{11}M_{33}M_{44} - M_{11}M_{34}M_{43} \\
\tilde{M}_{23} &= -M_{11}M_{23}M_{44} + M_{11}M_{24}M_{43} \\
\tilde{M}_{24} &= M_{11}M_{23}M_{34} - M_{11}M_{24}M_{33} \\
\tilde{M}_{32} &= -M_{11}M_{32}M_{44} + M_{11}M_{34}M_{42} \\
\tilde{M}_{33} &= M_{11}M_{22}M_{44} - M_{11}M_{24}M_{42} \\
\tilde{M}_{34} &= -M_{11}M_{22}M_{34} + M_{11}M_{24}M_{32} \\
\tilde{M}_{42} &= M_{11}M_{32}M_{43} - M_{11}M_{33}M_{42} \\
\tilde{M}_{43} &= -M_{11}M_{22}M_{43} + M_{11}M_{23}M_{42} \\
\tilde{M}_{44} &= M_{11}M_{22}M_{33} - M_{11}M_{23}M_{32}
\end{aligned} \tag{5.31}$$

The determinant of M is

$$\begin{aligned}
|M| &= M_{11}M_{22}M_{33}M_{44} + M_{11}M_{23}M_{34}M_{42} + \\
&\quad M_{11}M_{24}M_{32}M_{43} - M_{11}M_{24}M_{33}M_{42} - \\
&\quad M_{11}M_{23}M_{32}M_{44} - M_{11}M_{22}M_{34}M_{43}
\end{aligned} \tag{5.32}$$

To simplify the relation, the matrices are redenominated as follows:

$$\begin{aligned}
C_{lin} &= \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & (m + m_x)U \\ 0 & 0 & 0 & -z_G m U \\ 0 & 0 & 0 & x_G m U \end{bmatrix} \\
Vel &= \begin{bmatrix} u \\ v \\ p \\ r \end{bmatrix}, P = \begin{bmatrix} X \\ Y \\ K \\ N \end{bmatrix}, \text{ and } Acc = \begin{bmatrix} \dot{u} \\ \dot{v} \\ \dot{p} \\ \dot{r} \end{bmatrix}
\end{aligned} \tag{5.33}$$

The MMG mathematical model can shortly be demonstrated by,

$$Acc = M^{-1} \cdot (P - C_{lin} \cdot Vel) \tag{5.34}$$

The Coriolis matrix in equation 5.33 is in linearized form. The Coriolis matrix directly taken from equation 5.24 is,

$$C = \begin{bmatrix} 0 & -(m + m_y)r & z_Gmr & -x_Gmr \\ (m + m_x)r & 0 & 0 & 0 \\ 0 & 0 & 0 & -z_Gmu \\ 0 & 0 & 0 & x_Gmu \end{bmatrix} \quad (5.35)$$

Then, the MMG mathematical model will be defined by,

$$Acc = M^{-1} \cdot (P - C \cdot Vel) \quad (5.36)$$

The explicit form of the equation shortly expressed in the above equation is,

$$\begin{bmatrix} \dot{u} \\ \dot{v} \\ \dot{p} \\ \dot{r} \end{bmatrix} = M^{-1} \cdot \begin{bmatrix} X + (m + m_y)vr + mx_Gr^2 - mz_Gpr \\ Y - (m + m_x)ur \\ K + mz_GUr \\ N - mx_GUr \end{bmatrix} \quad (5.37)$$

The first matrix contains the hydrodynamic inertia terms including mass and added mass and moments, and the second matrix contains linearized Coriolis and damping effects proportional to the centripetal force and moments. The forces and moments on the right represent the propeller, rudder, and wave effects.

5.6 Explanation of External Forces and Moments

The unknown terms in equation 5.37 are forces X , Y , and moments K , N . The fundamental components listed below are considered separately due to the modular approach of the MMG model and mathematically given in the subsequent sections.

- A method to calculate the hydrodynamic forces and moments due to hull
- A method to calculate the forces and moments due to propellers
- A method to calculate the forces and moments due to rudders
- A method to impose the wave drift loads based on experimental/numerical results.

The external forces and moments X , Y , K , and N on the right-hand side of equation 5.37, consist of the following components:

$$\begin{aligned}
X &= X_H + X_R + 2X_P + X_W \\
Y &= Y_H + Y_R + 2Y_P + Y_W \\
K &= K_H + K_R + 2K_P - mg\overline{GM}\varphi \\
N &= N_H + N_R + 2N_P + N_W
\end{aligned} \tag{5.38}$$

where g is the gravitational acceleration, $\overline{GM}$ is the metacentric height, φ is the roll angle. The subscript H , R , and P express the hull, rudder, and propeller, respectively.

In the preliminary study, since the JBC model had a single rudder, single propeller configuration, propeller-induced loads and moment were not doubled (Kinaci & Ozturk, 2023). Therefore, the reason for multiplying hydrodynamic forces due to the propeller by two is that the ONRT model has twin propellers. Hydrodynamic forces by steering are not doubled because calculations are made for two rudders in the model. The X_W , Y_W , and N_W denote the external disturbances due to the waves. Another difference of the above model from the conventional 4-DoF maneuvering model is that the propeller-induced lateral forces are also added to the right-hand side of motion equations.

As detailed in previous sections, experimental and numerical studies were carried out in 3-DoF. The current system-based model includes the solution of 4-DoF equations of motion, and the effects of the wave on the roll motion are neglected. Therefore, there is no module containing the wave effect in the roll component, K . In the following subsections, obtaining the components of external forces and moments will be described.

5.6.1 Hydrodynamics forces and moments acting on ship hull

Hydrodynamic forces and moments are the result of the motion of the ship in the water and can be expressed in terms of the relative velocity and acceleration of the ship relative to the water. Hydrodynamic forces X_H , Y_H and moments N_H , K_H acting on ship hull are expressed by,

$$\begin{aligned}
X_H &= 0.5\rho L d U^2 X'_H(u', v', r', \varphi') \\
Y_H &= 0.5\rho L d U^2 Y'_H(v', r', \varphi') \\
K_H &= 0.5\rho L^2 d U^2 K'_H(p', \varphi') \\
N_H &= 0.5\rho L^2 d U^2 N'_H(v', r', \varphi')
\end{aligned} \tag{5.39}$$

where ρ is the density of water in kg/m^3 , L is ship length in m, and d is draft in m. Here, X'_H , Y'_H , K'_H , and N'_H are stated as follows:

$$\begin{aligned}
X'_H &= X'_u u' + X'_{vv} v'^2 + X'_{vr} v' r' + X'_{rr} r'^2 \\
Y'_H &= Y'_v v' + Y'_r r' + Y'_{vvv} v'^3 + Y'_{vvr} v'^2 r' + Y'_{vrr} v' r'^2 + Y'_{rrr} r'^3 + Y'_\varphi \varphi \\
K'_H &= K'_p p' + K'_\varphi \varphi \\
N'_H &= N'_v v' + N'_r r' + N'_{vvv} v'^3 + N'_{vvr} v'^2 r' + N'_{vrr} v' r'^2 + N'_{rrr} r'^3 + N'_\varphi \varphi
\end{aligned} \tag{5.40}$$

5.6.2 Forces and moments due to propeller

The propeller thrust force X_P in the model, which is expressed by the equation,

$$X_P = (1 - t_p)T \tag{5.41}$$

where t_p is thrust deduction factor. This parameter characterizes the influence of the propeller on the total resistance of the ship during self-propulsion and can also be approximated using empirical equations (Ozturk et al., 2022). Propeller thrust T is given by

$$T = K_T \rho n^2 D^4 \tag{5.42}$$

where ρ , n , and D denote the density of water in kg/m^3 , propeller revolution in rps, and propeller diameter in m, respectively. The thrust coefficient, K_T , is expressed as a function of the advance ratio J :

$$K_T = k_0 + k_1 J + k_2 J^2 \tag{5.43}$$

where k_0 , k_1 and k_2 are coefficients obtained from open-water propeller tests. Propeller advance ratio, J , is calculated by,

$$J = \frac{u(1 - w_p)}{nD} \quad (5.44)$$

In this equation, w_p denotes the wake fraction during maneuvering. The wake fraction during maneuvering, w_p , is a form of the effective wake fraction, w_{p0} , and given by

$$w_p = w_{p0}[1 - (1 - \cos^2 \beta_p)(1 - |\beta_p|)] \quad (5.45)$$

The propeller receives an inclined flow with an angle β_p , mathematically expressed by

$$\beta_p = \beta - x'_p r' + z'_p p' \quad (5.46)$$

where β is the drift angle of the ship ($\beta = \arctan(-v/u)$), x_p is the longitudinal coordinate of the propeller position, and z_p is the vertical coordinate of the propeller position. The lateral forces and moments generated by a conventional propeller are not included in the study by Yasukawa et al. (2019), i.e. $Y_p = N_p = 0$. However, lateral forces and moments, Y_p , K_p , and N_p are involved in the present model. Y_p , K_p , and N_p can be calculated according to the study by Guo et al. (2018) as follows:

$$Y_p = c_{YP}(\beta - l'_p r')\rho n^2 D^4$$

$$K_p = z'_p Y_p \quad (5.47)$$

$$N_p = x'_p Y_p$$

where c_{YP} is the hydrodynamic derivative of the lateral force on the propeller, l_p is the correction factor for the lateral velocity due to the yaw rate, and x_p is the longitudinal position of the propeller disk.

5.6.3 Forces and moments due to rudder

The rudder-related forces X_R , Y_R and moments K_R and N_R are calculated by the following equations (Guo et al., 2018; Yasukawa et al., 2019):

$$\begin{aligned}
X_R &= -(1 - t_R)F_N \sin \delta \cos \varphi \\
Y_R &= -(1 + a_H)F_N \cos \delta \cos \varphi \\
K_R &= z_{HR}(1 + a_H)F_N \cos \delta
\end{aligned}
\tag{5.48}$$

$$N_R = -(x_R + a_H x_H)F_N \cos \delta \cos \varphi$$

where t_R is the steering resistance deduction factor, a_H is the rudder force increasing factor, z_{HR} is the vertical position of the center of effective rudder force, x_R is the longitudinal coordinate of the rudder shaft, and x_H is the longitudinal coordinate of additional lateral force.

Guo et al. (2018) investigate the hull-propeller rudder interaction for the twin-screw ship with subject model ONRT. There are differences in the parameters of the portside and starboard side rudder in the calculation of the forces and moments originating from the rudder. Therefore, in the present chapter, the subscripts expressed as ‘ p ’ and ‘ s ’ will represent the portside and starboard side rudder, respectively.

Unlike the preliminary study (Kinaci & Ozturk, 2023), there is a change in the rudder normal force expression to include the loads from the port and starboard propeller. Since the twin screwed ONRT is the subject vessel, the rudder normal force (F_N) is expressed as,

$$F_N = F_{N\{p\}} + F_{N\{s\}} \tag{5.49}$$

Rudder normal force for each rudder is given by,

$$F_{N\{s\}} = 0.5\rho A_R U_{R\{s\}}^2 f_\alpha \sin \alpha_{R\{s\}} \tag{5.50}$$

The profile area of the rudder is expressed by A_R . The rudder normal force gradient coefficient, f_α , is given by Son & Nomoto (1981),

$$f_\alpha = 6.13\Lambda_R/(\Lambda_R + 2.25) \tag{5.51}$$

where Λ_R is the rudder aspect ratio.

The resultant inflow velocity at the rudder, U_R , and the effective inflow angle to the rudder, α_R , are calculated by

$$U_{R\{s\}} = \sqrt{u_{R\{s\}}^2 + v_{R\{s\}}^2} \quad (5.52)$$

$$\alpha_{R\{s\}} = \delta \pm \left[\left[|v_{RP}|/u_R \right]_{\{s\}} - v_{R\{s\}}/u_{R\{s\}} \right] \quad (5.53)$$

The effective inflow angle, α_R , varies according to the direction in which the rudder is rotated and it should be paid attention to ‘ $\pm$ ’. Here, u_R and v_R denote longitudinal and lateral inflow velocities, respectively, and are expressed as follows:

$$u_{R\{s\}} = \varepsilon_w u (1 - w_p) \sqrt{\eta \left\{ 1 + \kappa \left(\sqrt{1 + \frac{8K_T}{\pi J^2}} - 1 \right) \right\}^2 + (1 - \eta)} \quad (5.54)$$

$$v_{R\{s\}} = 0.5U \left\{ \begin{array}{l} \gamma_R^{\pm} [\beta + |\beta| - l_R' (r' + |r'|)] + \\ \gamma_R^{\mp} [\beta - |\beta| - l_R' (r' - |r'|)] \end{array} \right\} \quad (5.55)$$

Here, ε_w is the ratio of the wake fraction, η is the ratio of the propeller diameter to the rudder span defined by D_p/H_R , κ is the experimental constant, and U is the resultant velocity. In the lateral inflow velocity equation, γ_R denotes flow-straightening coefficient and l_R' is the correction factor. γ_R and l_R' get the ‘+’ superscript for windward condition and ‘-’ superscript for leeward condition. To exemplify, when the rudder angle is positive and head sea conditions, the portside rudder is windward and the starboard side rudder is leeward. This is valid for the range of 0-180° heading angle. Between the range of 180° and 360°, leeward parameters for the portside rudder and windward parameters for the starboard side rudder are used.

5.6.4 Forces and moments due to wave

Waves exert two distinct forces on ships: The first type, referred to as first-order oscillatory forces, is influenced by the wave encounter frequency and varies with time. In contrast, the second type, known as second-order drift forces, remains constant. The preliminary study (Kinaci & Ozturk, 2023) exclusively focuses on the analysis of wave drift forces, which have been empirically derived using Ankudinov's relationships

(Ankudinov, 1985). The study adopts an empirical approach, the thesis approaches the same subject from an experimental perspective, incorporating elements of second-order mean wave loads. The wave forces and moments are measured by performing steady drift tests applied on the ship at zero forward speed. Therefore, the effect of forward speed is neglected. The measured forces and moment in the time domain ($X\{t\}$, $Y\{t\}$, and $N\{t\}$) were converted to the frequency domain via Fast Fourier Transform (FFT) as detailed in chapter 3.7.1 and only the value at the 0th frequency was noted. Basically, the forces and moments due to wave can be expressed as,

$$\begin{aligned} X_W &= FFT(X\{t\})_{f_0} \\ Y_W &= FFT(Y\{t\})_{f_0} \\ N_W &= FFT(N\{t\})_{f_0} \end{aligned} \quad (5.56)$$

Forces and moment were measured for each 15° heading angle and plotted accordingly. It was assumed that the curve in the range of 180 – 360° and the curve in the range of 0 – 180° are symmetrical. Subsequently, several polynomials for the range of 0-180° heading angle and a 180 – 360° heading angle were fitted. This step was repeated for each force and moment related to the heading angle.

The fifth-degree, sixth-degree, seventh-degree, and eighth-degree polynomials are fitted to the data of wave drift forces and moments of $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ condition. The statistics for each fit are returned to determine the best fit. Table 5.2 presents the statistics for each fit. In the table, *rsquare* is the R-squared coefficient of determination, *ssreg* is the regression sum of squares, and *ssresid* is the residual sum of squares. The correlation between the response values and the predicted values is expressed as the *rsquare*. An R-square value that is closer to 1 means that the fitting model represents a greater proportion of variance. The overall squared difference between the estimated value and real value is captured by *ssresid*. The regression sum of squares, *ssreg*, is calculated from $ssreg = sstotal - ssresid$. A fit that is better for prediction is shown by the small values of *ssresid*.

Table 5.2 : Statistics for polynomial fits.

Data	Fit (order)	<i>rsquare</i>	<i>ssreg</i>	<i>ssresid</i>
$X_W (0-180)$ and $X_W (180-360)$	5 th	0.9870	0.7810	0.0103
	6 th	0.9872	0.7812	0.0102
	7 th	0.9945	0.7870	0.0043
	8 th	0.9990	0.7905	0.0008
$Y_W (0-180)$	5 th	0.9478	2.6176	0.1441
	6 th	0.9736	2.6889	0.0729
	7 th	0.9796	2.7055	0.0562
	8 th	0.9797	2.7056	0.0561
$Y_W (180-360)$	5 th	0.9586	2.9120	0.1258
	6 th	0.9769	2.9677	0.0701
	7 th	0.9810	2.9799	0.0578
	8 th	0.9811	2.9803	0.0575
$N_W (0-180)$	5 th	0.9817	1.5692	0.0292
	6 th	0.9847	1.5739	0.0245
	7 th	0.9914	1.5847	0.0137
	8 th	0.9945	1.5895	0.0089
$N_W (180-360)$	5 th	0.9797	1.5700	0.0325
	6 th	0.9839	1.5768	0.0258
	7 th	0.9917	1.5892	0.0133
	8 th	0.9928	1.5910	0.0115

The *rsquare* value of 0.99 indicates that the fit accounts for 99% of the total variation in the average-related data. In this study, fitting methods with a minimum R-square of 0.99 were preferred. The *rsquare* did not reach 0.99 for polynomial models fitted to Y_W . Since there is not much difference between the seventh-degree and eighth-degree polynomials, it is also represented by the seventh-degree polynomial. The polynomial fits are plotted in Figure 5.1. Consequently, the plots indicate that the seventh-degree polynomials follow the data well enough. The seventh-degree polynomial models are also fitted to wave drift forces and moments of $H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$ and $H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$ conditions. These equations are then implemented into the mathematical model to represent the wave drift forces and moment in time-based maneuvering simulations.

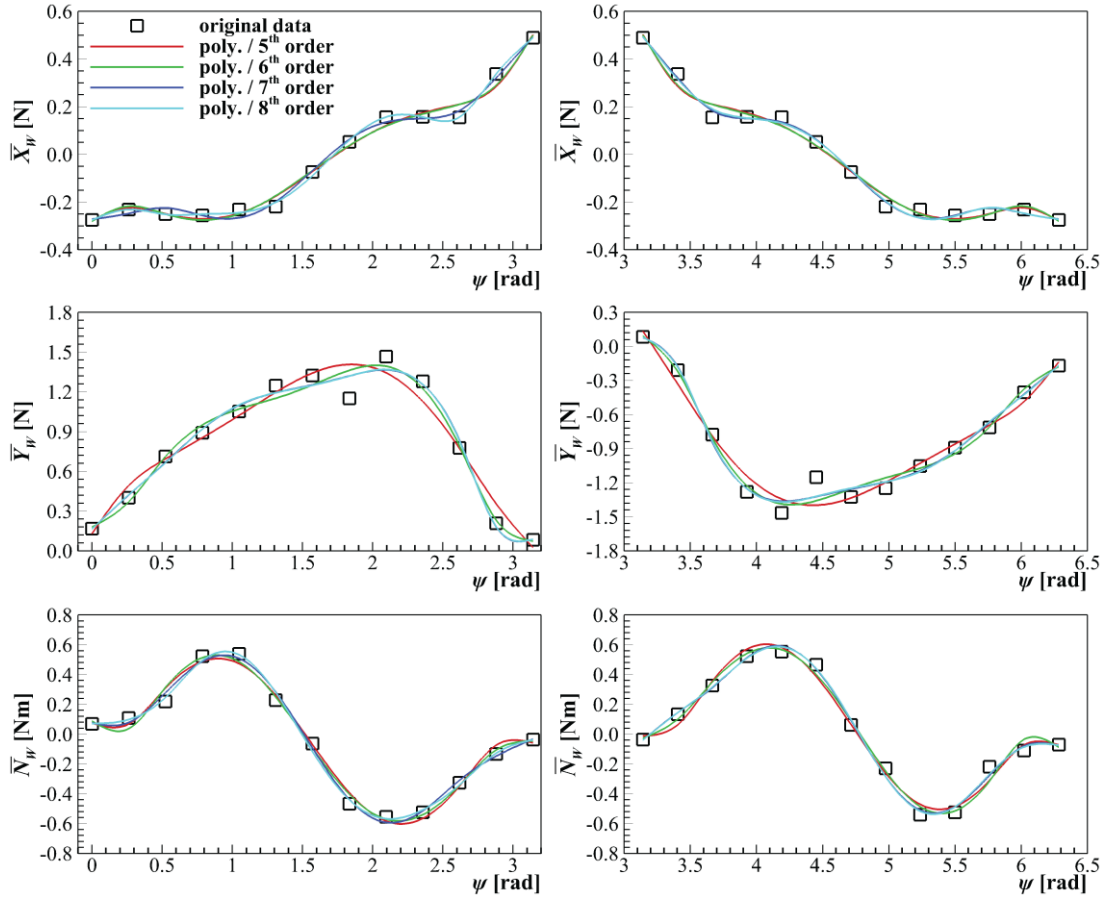


Figure 5.1 : Polynomial models adapted to wave drift load data for the $H/\lambda = 0.020, \lambda/L_{WL} = 1.00$ condition.

In the steady drift tests, the forces and moments were measured according to the coordinate system of the loadcell. These forces and moments are obtained for the cases where the wave is coming from the head and the heading angle of the model changes at intervals of 15 degrees. Modifications were made for each quadrant to adapt the polynomials adapted to forces and moments to the S-B simulations. To adaptation of the wave loads, the following modifications are used.

If $0^\circ < \psi < 90^\circ$;

$$X_W = X_W \cos(\psi) - Y_W \sin(\psi)$$

$$Y_W = X_W \sin(\psi) - Y_W \cos(\psi) \quad (5.57)$$

$$N_W = N_W + X_W x_G \sin(\psi) - Y_W x_G \cos(\psi)$$

If $90^\circ < \psi < 180^\circ$;

$$\begin{aligned}
X_W &= X_W \cos(\pi - \psi) \pm Y_W \sin(\pi - \psi) \\
Y_W &= X_W \sin(\pi - \psi) \pm Y_W \cos(\pi - \psi) \\
N_W &= N_W + X_W x_G \sin(\pi - \psi) \pm Y_W x_G \cos(\pi - \psi)
\end{aligned} \tag{5.58}$$

If $180^\circ < \psi < 270^\circ$;

$$\begin{aligned}
X_W &= X_W \cos(\psi - \pi) - Y_W \sin(\psi - \pi) \\
Y_W &= X_W \sin(\psi - \pi) - Y_W \cos(\psi - \pi) \\
N_W &= N_W + X_W x_G \sin(\psi - \pi) - Y_W x_G \cos(\psi - \pi)
\end{aligned} \tag{5.59}$$

If $270^\circ < \psi < 360^\circ$;

$$\begin{aligned}
X_W &= X_W \cos(2\pi - \psi) \pm Y_W \sin(2\pi - \psi) \\
Y_W &= X_W \sin(2\pi - \psi) \pm Y_W \cos(2\pi - \psi) \\
N_W &= N_W + X_W x_G \sin(2\pi - \psi) \pm Y_W x_G \cos(2\pi - \psi)
\end{aligned} \tag{5.60}$$

At the starboard side turning, the forces and moments from the y-axis of the wave are added in case of $90^\circ < \psi < 180^\circ$ and $270^\circ < \psi < 360^\circ$, while in other cases forces and moments are subtracted. Through these modifications, the transition to the local coordinate system has been achieved, and wave loads acting according to the heading angle of the model at each time step have been added.

The position of the right and left rudders changes according to the direction of the wave and the turning direction of the model. In other words, the rudders stay on the leeward/windward side changes. For this reason, the rudder module should be revised for each incident wave angle case. Flow-straightening coefficients and correlation factors for the flow-straightening effect due to yaw rate, presented in the next section, take different values depending on the position of the rudders. While correction factors with ‘+’ superscript are used for the rudder positioned in the direction of wave, correction factors with ‘-’ superscript are used for the rudder in the opposite side of the wave direction.

5.7 Turning Circle Simulations

Turning circle tests are an essential part of ship maneuvering trials, conducted to assess a vessel's maneuverability and determine its turning characteristics. These tests provide valuable information about a ship's ability to navigate safely in confined spaces, such as ports, narrow channels, or congested waters. The collected data is analyzed to determine the ship's tactical diameter, advance, transfer, loss of speed on the steady turn, drift angle, and other relevant factors. The main parameters of the turning circle are illustrated in Figure 5.2.

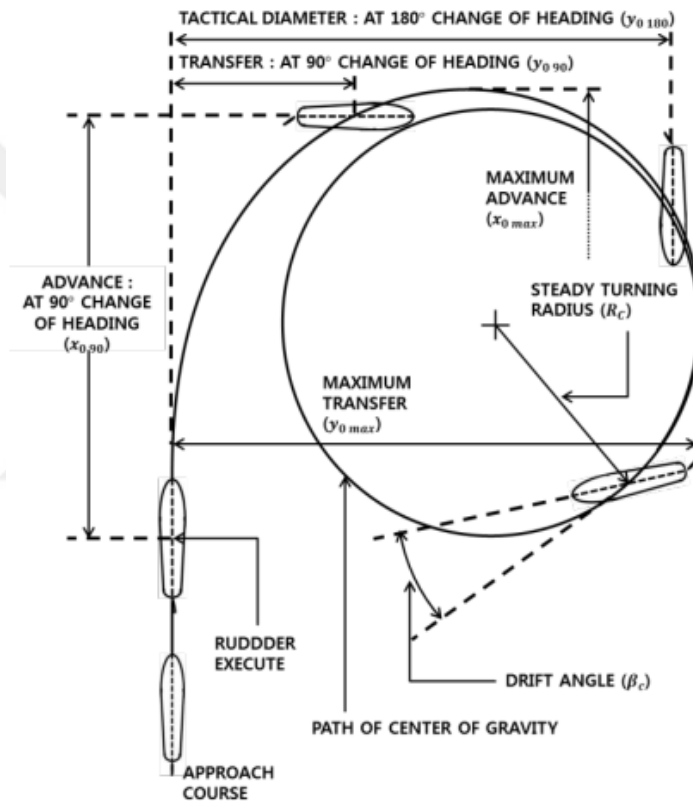


Figure 5.2 : Definitions of turning circle (ITTC, 2017a).

The terms used to describe standard maneuvers are presented as follows:

- .2 Turning circle maneuver is the maneuver to be performed to both starboard and port with a 35° rudder angle or the maximum rudder angle permissible at the test speed, following a steady approach with zero yaw rate.
- .3 Advance is the distance traveled in the direction of the original course by the midship point of a ship from the position at which the rudder order is given to the position at which the heading has changed 90° from the original course.

.4 Tactical diameter is the distance traveled by the midship point of a ship from the position at which the rudder order is given to the position at which the heading has changed 180° from the original course. It is measured in a direction perpendicular to the original heading of the ship. (International Maritime Organization, 2002)

Standard turning circle maneuvering simulations were conducted in calm water and regular wave conditions at $Fr = 0.2$ which corresponds to $U_0 = 1.111$ m/s in the model scale. The wave conditions are the same as experimental study: $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$, $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$, $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$. The turning tests in waves are performed with rudder angles of $\delta = \pm 35^\circ$ and wave incident angles of $\chi = 0^\circ, \pm 90^\circ, 180^\circ$. The simulation matrix is presented in Table 5.3. After performing the turning circle simulations to both port and starboard side, the following main variables were achieved and expressed in the dimensional form: Advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), and steady turning speed (STS).

Table 5.3 : Turning circle maneuvering simulation matrix.

Fr [-]	λ/L_{WL} [-]	H/λ [-]	δ [deg]	ψ [deg]	χ [deg]
0.2	Calm water		$0, \pm 15, \pm 25, \pm 35$	$0, \pm 90, 180$	–
	1.00	0.020	± 35	–	$0, \pm 90, 180$
	0.50	0.020			
	1.00	0.025			

5.7.1 Parameters used in the simulations

The values of maneuvering and coefficients for propellers and rudders used in the 4-DoF system-based model are listed in the present section. The main parameters related to the ONRT model are presented in Table 5.4. The sources of the parameters are also cited in the last column of the table. Some of the parameters are in non-dimensionalized form and can be dimensionalized with expressions given in equation 5.1. The vertical position of the center of gravity from the ship is calculated with $z_G = \overline{GM} - \overline{KM} + d$ where $\overline{KM} = \overline{GM} + \overline{KG}$. Also, non-dimensional α'_z is taken as same as z'_H by Yasukawa et al. (2019).

Table 5.4 : Main parameters related to the ONRT model.

Parameter	Symbol [units]	Value	Source
Length on waterline	L_{WL} [m]	3.147	<i>Wageningen 2023: A Workshop on CFD in Ship Hydrodynamics (2023)</i>
Draft	d [m]	0.112	<i>Wageningen 2023: A Workshop on CFD in Ship Hydrodynamics (2023)</i>
Longitudinal position of the center of gravity from midship	x_G [m]	-0.053	Sadat-Hosseini et al. (2015)
Vertical position of the center of gravity from midship	z_G [m]	-0.044	Yasukawa et al. (2019)
Density of water	ρ [kg/m ³]	1000	—
Metacentric height	$\overline{GM}$ [m]	0.0422	<i>Wageningen 2023: A Workshop on CFD in Ship Hydrodynamics (2023)</i>
z coordinate of the acting point of the hydrodynamic forces	z'_H [—]	0.0344	Guo et al. (2018)
Vertical acting point of the lateral added mass component, m_y	α'_z [—]	0.0344	Yasukawa et al. (2019)
Gravity	g [m/s ²]	9.81	—

Non-dimensional mass, added mass, and inertia terms are presented in Table 5.5.

Table 5.5 : Non-dimensional added mass and inertia terms.

Parameter	Symbol [—]	Value	Source
Ship's mass	m'	0.13070	<i>Wageningen 2023: A Workshop on CFD in Ship Hydrodynamics (2023)</i>
Added masses of x -axis direction	m'_x	0.01307	Araki et al. (2012)
Added masses of y -axis direction	m'_y	0.10901	Araki et al. (2012)
Moment of inertia of ship around x -axis	I'_{xx}	0.00040	Guo et al. (2018)
Added moment of inertia of ship around x -axis	J'_{xx}	0.00004	Araki et al. (2012)
Moment of inertia of ship around z -axis	I'_{zz}	0.00789	Guo et al. (2018)
Added moment of inertia of ship around z -axis	J'_{zz}	0.00789	Araki et al. (2012)

The non-dimensional mass, added mass, and inertia terms are dimensionalized according to equation 5.1 and listed in Table 5.6.

Table 5.6 : Dimensional added mass and inertia terms.

Parameter	Symbol [units]	Value
Ship's mass	m [kg]	72.4865
Added masses of x -axis direction	m_x [kg]	7.2486
Added masses of y -axis direction	m_y [kg]	60.4572
Moment of inertia of ship around x -axis	I_{xx} [kgm ²]	2.1970
Added moment of inertia of ship around x -axis	J_{xx} [kgm ²]	0.2197
Moment of inertia of ship around z -axis	I_{zz} [kgm ²]	43.3363
Added moment of inertia of ship around z -axis	J_{zz} [kgm ²]	43.3363

Hydrodynamic derivatives are essential in comprehending marine vessel motion. Surge, sway, yaw, and roll motions are influenced by their respective derivatives. Surge derivative connects thrust/resistance to fore-and-aft acceleration. Sway derivative involves lateral forces and motion. Yaw derivative dictates rotational motion around the vertical axis. Roll derivative governs angular acceleration in roll motion. The hydrodynamic derivatives derived from the study by Guo et al. (2018) for surge, sway, yaw, and roll motion are listed in Table 5.7. The derivatives include both linear and nonlinear terms.

Table 5.7 : Hydrodynamic coefficients used in the simulations (Guo et al., 2018).

Parameter	Symbol [-]	Value	Parameter	Symbol [-]	Value
Surge maneuvering coefficients	X'_u	-0.0212	Yaw maneuvering coefficients	N'_v	-0.1039
	X'_{vv}	-0.0734		N'_r	-0.0428
	X'_{vr}	-0.05211		N'_{vvv}	-0.5642
	X'_{rr}	-0.013		N'_{vvr}	-0.8704
	Y'_v	-0.2777		N'_{vrr}	-0.2419
Sway maneuvering coefficients	Y'_r	0.02387	Roll maneuvering coefficients	N'_{rrr}	-0.0418
	Y'_{vvv}	-1.5258		Y'_ϕ	-0.0054
	Y'_{vvr}	-0.6218		K'_ϕ	0.0073
	Y'_{vrr}	-0.7367		N'_ϕ	-0.0036
	Y'_{rrr}	-0.0551		K'_p	-0.2429

It is accepted that the most reliable method for calculating the maneuvering derivatives is model experiments. However, since model experiments require a long period of time and finances, simple empirical relations that can be expressed in terms of ship main dimensions, form coefficients, and appendages features were developed by researchers, usually by applying regression analysis to model test results. Apart from

empirical approaches, numerical methods can be preferred such as in the study by Guo et al. (2018). The researchers conducted captive model tests (CMT), static drift tests, and static heel tests by using computational fluid dynamics methods to obtain the derivatives.

The parameters in the propeller module are summarized in Table 5.8. The values k_0 , k_1 , and k_2 that ensured that the vessel in which the rudder angle is zero is heading forwards with a steady speed are used. The coefficients are employed to represent the relation between K_T and J . In the system-based simulations, the details of which will be presented in the next chapter, first of all, the straight-ahead case with $\delta = 0^\circ$ is examined. To detect conditions where the velocity remains constant at 1.111 m/s, k_0 , k_1 , and k_2 are estimated iteratively. Also, the expression giving the longitudinal position of the propeller disk, x'_p , is presented with reference to the forward perpendicular of the ship. In the 4-DoF S-B model, the longitudinal position of the propeller disk to the center of the vessel was calculated.

Table 5.8 : Parameters in the propeller module.

Parameter	Symbol [units]	Value	Source
Propeller diameter	D_p [m]	0.1066	Wageningen 2023: A Workshop on CFD in Ship Hydrodynamics (2023)
Propeller revolution	n [rps]	8.97	Guo et al. (2018)
Thrust deduction factor under straight ahead condition	t_p [–]	0.0854	Guo et al. (2018)
Effective wake fraction under straight ahead condition	w_{p0} [–]	0.1101	Guo et al. (2018)
Coefficients representing the characteristics of K_T	k_0 [–]	0.6300	–
	k_1 [–]	-0.3052	
	k_2 [–]	-0.0671	
Longitudinal position of propeller disk (from FP)	x'_p [–]	-0.9267	Wageningen 2023: A Workshop on CFD in Ship Hydrodynamics (2023)
Vertical position of propeller disk (below WL)	z'_p [–]	- 0.03565	Wageningen 2023: A Workshop on CFD in Ship Hydrodynamics (2023)
Correction factor for the lateral velocity due to yaw rate	l'_p [–]	-0.4804	Guo et al. (2018)
Hydrodynamic derivative of the lateral force on the propeller	c'_{YP} [–]	0.0334	Guo et al. (2018)

The parameters in the rudder module are summarized in Table 5.9. The flow-straightening coefficients and the correlation factors for the flow-straightening effect are normally different for the portside rudder and starboard side rudder. However, as in the study by Guo et al. (2018), the parameters obtained by the CFD computation are preferred. The coefficients were taken the same for both rudders, and only the case of being leeward or windward was considered in the mathematical model.

Table 5.9 : Parameters in the rudder module.

Parameter	Symbol [units]	Value	Source
Rudder span	H_R [m]	0.1599	Wageningen 2023: A Workshop on CFD in Ship Hydrodynamics (2023)
Rudder area	A_R [m ²]	0.0120	
Rudder aspect ratio	Λ_R [–]	1.177	
Steering resistance deduction factor	t_R [–]	0.2011	Guo et al. (2018)
Rudder force increasing factor	a_H [–]	0.1964	
Longitudinal acting point of the additional lateral force component	x'_H [–]	-0.3350	
Longitudinal position of rudder shaft	x'_R [–]	-0.4459	
Vertical position of the center of effective rudder force	z'_{HR} [–]	0.0292	
Ratio of the wake fraction	ε [–]	0.8925	
Experimental constant	κ [–]	0.6911	
The ratio of lateral rudder inflow velocity induced by propeller rotation to the longitudinal component of inflow velocity ahead of the rudders	$ v_{RP} /u_{R\{P\}}$	0.0778	
	$ v_{RP} /u_{R\{S\}}$	0.0730	
Flow-straightening coefficients for windward and leeward	γ_R^+	0.7546	
	γ_R^-	0.4000	
Correlation factors for the flow-straightening effect due to yaw rate	l_R^+	-0.4837	
	l_R^-	-0.8723	

5.7.2 Straight-ahead condition in calm water

The straight-ahead condition refers to the state of a ship when it is moving in a straight line without any lateral deviation. In this state, the ONRT ship's rudder is aligned with the longitudinal axis of the hull, $\delta = 0^\circ$. The straight-ahead condition plays a significant role in determining the course-keeping stability of a ship. Course-keeping stability refers to a ship's ability to maintain a steady course without sideways oscillations or deviations from the intended heading. When a ship is in the straight-ahead condition, if the ship has course-keeping stability, its hydrodynamic forces are typically well-balanced, and there is minimal yawing moment acting on the vessel. Since S-B analyses are carried out as 4-DoF, there is not enough information for the course-keeping ability of the ONRT. Limiting motion causes sway and yaw loads to be neglected. For example, the straight course observed in the 4-DoF straight-ahead condition in the study of Kinaci & Ozturk (2022) is not observed in the 6-DoF condition. Also, the simulation was conducted by Kinaci et al. (2021) at zero rudder angle to investigate course-keeping ability with the mathematical model. Again, model cannot maintain a straight-ahead part due to side forces.

Straight-ahead simulations were mainly used to determine the k_0, k_1 , and k_2 coefficients, as indicated in chapter 5.7.1. The following maneuvering tests are conducted at an advance velocity of $U_a = 1.111 \text{ m/s}$ corresponding to $Fr = 0.20$. In the straight-ahead case, the value of $u = 1.111 \text{ m/s}$ selected as the initial velocity is expected to remain constant after the model starts to move. As a result, a slight change was made in the value of k_0 over the value given in the literature to ensure that the velocity in the progressive time steps is equal to the velocity in the initial time step. Figure 5.3 illustrates the changes in maneuvering characteristics. As can be seen, the trajectory experiences a straight course. The yaw rate, roll angle, and heading angle are always zero over time. There is a slight change in u , accordingly in U ; however, it can be said that the velocity remains constant with the k_0 value given as input. In conclusion, the graph reveals a straight-ahead course indicating the relationship between velocity and propeller coefficients which used in further analysis.

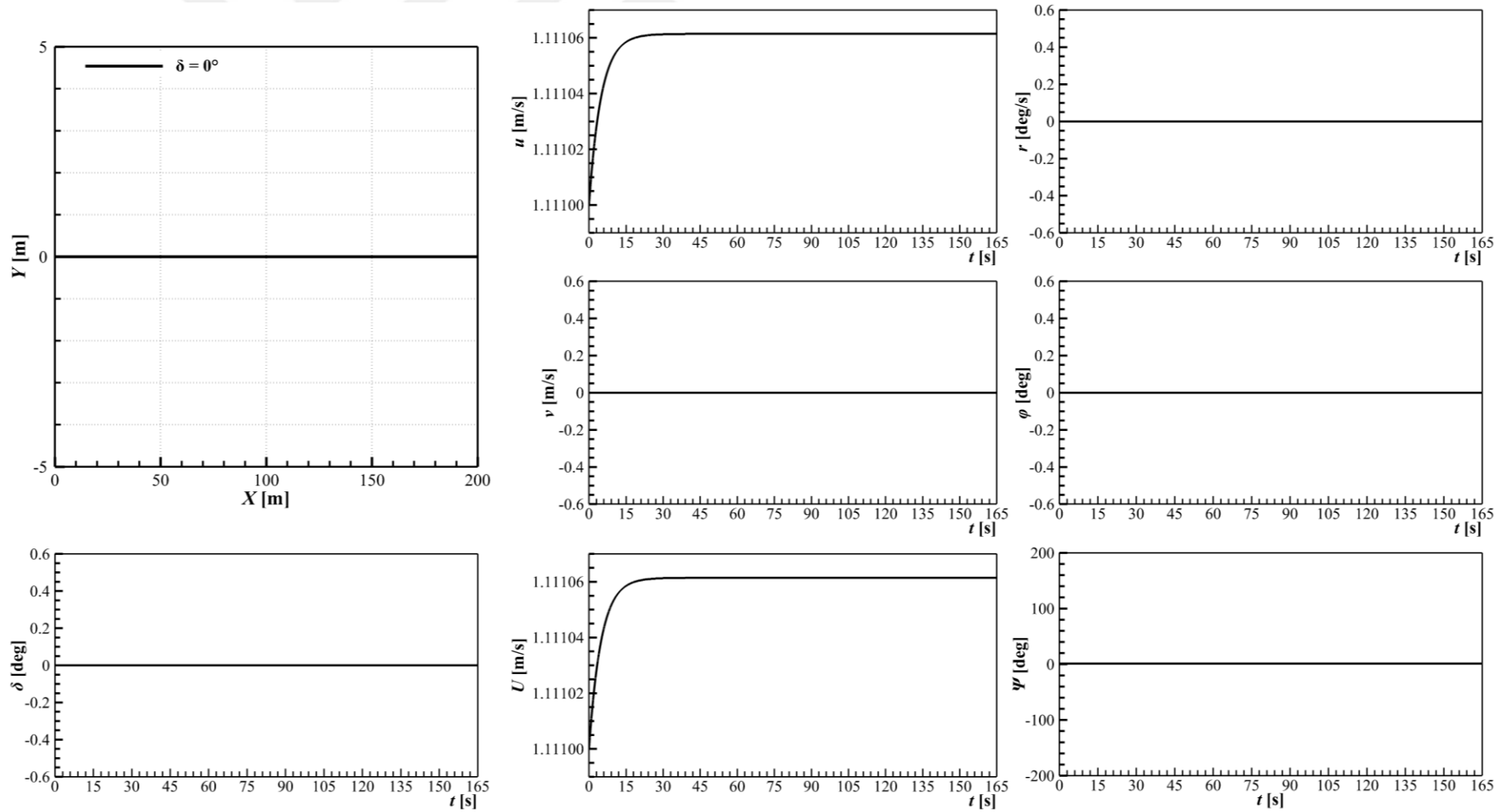


Figure 5.3 : Trajectory and time-dependent parameters during the straight-ahead condition in 4-DoF S-B simulation.

5.7.3 Turning circle results in calm water

Turning circle simulations with $\pm 35^\circ$ rudder angles were first carried out under calm water conditions at $Fr = 0.2$. According to the coordinate system used in the simulations, $\delta = 35^\circ$ represents the portside turning, while $\delta = -35^\circ$ represents the starboard side turning. The heading angles were 0° , $\pm 90^\circ$, and 180° . Simulation results were compared with existing experimental, numerical, and S-B simulations. The compared parameters include advance, transfer, tactical diameter, steady turning diameter, steady yaw rate, steady turning speed, the time when the heading angle is 90° , and the time when the heading angle is 180° . Turning trajectories are visualized and time dependent graphs of rudder angle, surge velocity, sway velocity, resultant velocity, yaw rate, roll angle, and yaw angle are presented.

The S-B turning circle maneuver simulation results for $\delta = 35^\circ$, i.e. portside turning, and $\psi = 0^\circ$ were compared with the experimental results (Elshiekh, 2014), computational results accessed in the literature (Wang et al., 2016), S-B simulation results accessed in the literature (Guo et al., 2018) and present CFD results. The comparative results are presented in Table 5.10 and Figure 5.4. In Table 5.10, the results of the vessel trajectories, advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), steady turning speed (STS), the time when the heading angle is 90° (T_{90}), and the time when the heading angle is 180° (T_{180}) are listed. The error percentages presented in Table 5.10 are found by equation 4.15. The deviation is indicated in percentage between the simulation values and experimental values. Several significant results can be gleaned by comparing the values in the table and the visual results.

Present S-B results are in good agreement with experimental results except for the advance parameter. Compared to the correlation of S-B simulations with Wang's CFD results, the results are in high agreement with Guo et al.'s S-B simulations. The reason for this is obvious, the parameters that Guo et al. (2018) presented were used and the same degree of freedom equations of motion were solved.

While CFD simulations by Wang et al. (2016) and S-B simulations by Guo et al. (2018) have successfully estimated the advance distance, the current S-B calculation has shown less success numerically. However, during his research, the author realized that there was an ambiguity in the experimental results: It is unclear when the rudder

angle was rotated and from what time the turning circle was plotted. What is recommended by ITTC is to start the turning circle plot from 1 second before turning the rudder. Existing S-B and CFD turning trajectories are illustrated in accordance with this recommendation. Clarifying this uncertainty in the experimental results may change interpretations of the advance value comparison. However, it can be seen that the direct CFD result has a high percentage of error. It is thought that direct CFD simulations should be thoroughly re-examined by the author.

It can be seen that the current S-B simulations in the transfer value give one of the most successful results in calm water condition. In fact, it surpassed Wang et al.'s success in advance distance estimation in transfer distance estimation. The time when the heading angle is equal to 90° , T_{90} , is almost the same as the experimental time. Similar to the advance distance, the direct CFD result did not show the expected success in transfer value. This difference also occurs in the analysis of the time when the heading changes 90° from the original course. In direct CFD simulations, the ship has a 90° heading angle in 23.880 seconds, that is, the model reached this angle approximately 9 seconds later than the model in the experimental study. Ultimately, this time difference is related to the fact that the x and y positions are far more distant from the initial position. However, this relation does not fully explain why the values of transfer and advance are high. The S-B simulation demonstrated the estimation success at the y distance by finding the tactical diameter value 0.89% smaller than the experimental result. T_{180} parameter was not given in the literature, while the current S-B simulation showed agreement with a difference of less than 5 percent, so much more successful than Wang's estimate. However, the direct CFD result is far from the findings from both the S-B simulation and available experiments. As with the T_{90} , the difference is substantial in the T_{180} . The model reached a 180° heading angle at 36.240 seconds, almost 10 seconds after the experiment.

One promising observation is that direct CFD simulation calculates a steady turning diameter with less than a 10% difference. CFD by Wang et al. (2016) overestimated the STD value compared to the experiment, while current S-B simulations underestimated it. Also, the error rates of both S-B simulation are almost the same. The error rate of less than 5% demonstrated the success of the S-B simulation.

Additionally, when comparing the values of the parameter steady yaw rate between the solution methods, a notable increase is evident in the S-B simulation. A comment

can be made to explain the difference here: The equations of motion used only in the S-B simulations are 4-DoF. The experimental setup of Elshiekh (2014) was made to allow all motions of the model, and the existing CFD and CFD in the literature were solved for 6-DoF. The difference between CFD results and experimental results is less than the difference between S-B results and experimental results. It may be necessary to examine more data to reach a clear conclusion, but this may indicate that the *SYR* value increases as the degree of freedom of motion decreases.

In all conditions presented in Appendix G, the *SYR* values obtained from the S-B simulations are higher than the experimental results. It can be seen that there is no significant difference between port side turning and starboard side turning. This slight difference between turnings can be thought to be due to the difference in flow around the rudders. Guo et al. (2018) emphasized that there is a serious difference between the lateral forces of portside and starboard side rudders. Correction coefficients are provided for the twin screwed ONRT model, as it has an asymmetric flow-straightening effect. These coefficients ($\gamma_R^{\pm}$ and $l_R^{\pm}$) were taken the same for the starboard side and portside rudder in the S-B model, only the condition of being leeward and windward was considered. However, these coefficients used in the rudder's normal force calculation also vary depending on whether the rudder is on the starboard side or portside. Therefore, this difference in *SYR* value according to the direction of rudder rotation can also be explained by such an approach.

Direct CFD showed the highest consistency in steady turning speed. It can be said that the result of the S-B simulation is also compatible with a difference of 4.63%. The time-dependent variations of the longitudinal velocity and lateral velocity used in the steady turning speed calculation are presented in Figure 5.4. It can be seen from the graphs that the difference in steady turning speed between the S-B simulation and the experimental investigation is due to the sway velocity. While the surge velocity is more in line with the experimental surge velocity, there is some difference with the sway velocity. The sway velocity in S-B simulations was higher than the experimental one.

The values of the roll parameter, illustrated in Figure 5.4, indicate that the S-B results are lower than the experimental results. Immediately after the rudder execution, the ship model achieves an increasing roll motion and then there is another peak in the reverse direction. In the EFD results, the model first turns to the starboard side with a maximum of 2.18° , then it reaches the maximum heeling angle at the portside with the

-3.91°. After a certain period of time, the roll angle becomes stable, with it being negative at the portside turning. In S-B simulations, only the first peak appears at 2°, and then the value decreases and stabilizes.

The results of the starboard side turning simulation with $\delta = -35^\circ$ at $Fr = 0.2$ in calm water condition are also compared with available experimental results. Table 5.11 presents the comparison of the vessel trajectories, advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), steady turning speed (STS), the time when the heading angle is 90° (T_{90}), and the time when the heading angle is 180° (T_{180}). Turning trajectories and time histories are illustrated in Figure 5.5.

The advance parameter is calculated higher compared to the experimental one, as in the case of portside turning. As mentioned before, the conditions used to determine the advance parameter need to be clarified. When both directions of rotation are compared in the case of head sea, it is seen that the advance values are calculated very close to each other. Unlike the advance parameter, the compatibility of the transfer parameter with the experimental result is very good. The transfer value was found to be almost the same in both directions of rotation. The time when the heading angle is equal to 90° , T_{90} , is almost the same as the experimental time.

The tactical diameter parameter was estimated more closely than the port side turning condition. However, since the error rates are below 1%, it can be said that there is no remarkable difference between the two turnings.

The steady turning diameter, estimated with less than 5 percent difference, was obtained smaller than the experimental STD . This slight difference can also be examined from the turning trajectory in Figure 5.5. Compared to T_{90} , the difference in T_{180} is greater. However, this estimate can be considered successful, as there is a difference of less than 5 percent. Additionally, when port side and starboard side turns were compared, it was seen that the STD value did not change. The findings from Kinaci & Ozturk (2022) corroborated that port side turning results in a reduced steady turning diameter. This phenomenon occurs due to the higher steady yaw rate associated with port side turning. In that case, the rotation direction of the propeller caused asymmetry in the flow leading to differences in port and starboard turnings. In

ONRT turning circle maneuvering simulations, the steady yaw rate value remained almost constant, regardless of the turning direction.

A difference of nearly 10% was observed in the *SYR* value. There is this difference in both turnings, as explained in the previous case ($\delta = 35^\circ$ and $\psi = 0^\circ$), so it may be necessary to use separate correction coefficients for the port and starboard rudders. Similar to the port side turning condition, the steady turning speed was estimated with a margin of less than 5 percent higher than the experimental *STS* value. The *STS* value did not change when the turning direction changed.

The turning trajectories and time histories of rudder angle, surge velocity, lateral velocity, resultant velocity, yaw rate, roll angle, and ship heading are also illustrated for other simulation conditions in Figure G.1 through Figure G.6 in Appendix G. Comparison of trajectory parameters are also tabulated in Table G.1 through Table G.3. All S-B simulation results were compared with the experimental study by Elshiekh (2014). In general, the advance parameter has always been overestimated with differences of around 12%. There was no critical change in advance values with the change of turning direction. While the transfer parameter was generally estimated higher with a very small margin, i.e. below 5%, it was smaller than the experimental data at starboard side turning ($\delta = -35^\circ$) only in the $\psi = 90^\circ$ and $\psi = -90^\circ$ condition. However, port side and starboard side results in S-B simulations are almost the same in terms of transfer parameter. The experimental transfer parameter differed according to the turning direction of the ship model. In S-B simulations, the model always reached the 90° heading position in a shorter time. Following this, it usually arrives at the 180° heading position earlier than the experimental one. It can be said that the T_{180} values are almost the same in the case of $\psi = -90^\circ$ and $\delta = 35^\circ$ with only 0.40% difference. Turning diameter was calculated smaller with slight differences compared to the experimental results. Only in the case of $\psi = 180^\circ$ and $\delta = 35^\circ$ was it higher by 0.02%. The steady turning parameter is also in good agreement with the experimental results in general. However, the difference in the $\psi = -90^\circ$ and $\delta = -35^\circ$ condition is higher compared to the others. The differences in the steady yaw rate parameter are higher than the others: higher values were obtained than the experimental results, with differences of around 10 percent. Finally, the steady turning speed parameter was also estimated higher, with differences of around 5% under all conditions.

Table 5.10 : Comparison of trajectory parameters in calm water, $\delta = 35^\circ$ and $\psi = 0^\circ$.

Parameter	EFD (Elshiekh, 2014)	CFD (Wang et al., 2016)	Error [%]	S-B (Guo et al., 2018)	Error [%]	S-B – Present	Error [%]	CFD – Present	Error [%]
Ad [m]	7.4668	7.2854	2.43	7.7590	3.91	8.3475	11.79	10.8187	44.89
Tr [m]	4.0825	4.3255	5.95	4.1056	0.57	4.1245	1.03	6.0603	48.45
T_{90} [s]	15.1400	15.9872	5.60	N/A	N/A	15.0132	0.84	23.880	57.73
TD [m]	10.0453	10.4581	4.11	9.9497	0.95	9.9557	0.89	12.9204	28.62
STD [m]	10.0644	10.5431	4.76	9.6834	3.79	9.7231	3.39	10.8805	8.11
T_{180} [s]	26.0800	27.5894	5.79	N/A	N/A	25.0500	3.95	36.240	38.96
SYR [deg/s]	8.0725	7.6734	4.94	8.8701	9.88	8.8637	9.80	7.5864	6.02
STS [m/s]	0.7188	N/A	N/A	0.7472	3.95	0.7521	4.63	0.7189	0.01

Table 5.11 : Comparison of trajectory parameters in calm water, $\delta = -35^\circ$ and $\psi = 0^\circ$.

Parameter	EFD (Elshiekh, 2014)	S-B – Present	Error [%]
Ad [m]	7.4806	8.3483	11.60
Tr [m]	4.0539	4.1246	1.74
T_{90} [s]	15.1400	15.0160	0.82
TD [m]	10.0068	9.9595	0.47
STD [m]	10.1481	9.7257	4.16
T_{180} [s]	26.0700	25.0699	3.84
SYR [deg/s]	8.0907	8.8626	9.54
STS [m/s]	0.7149	0.7522	5.22

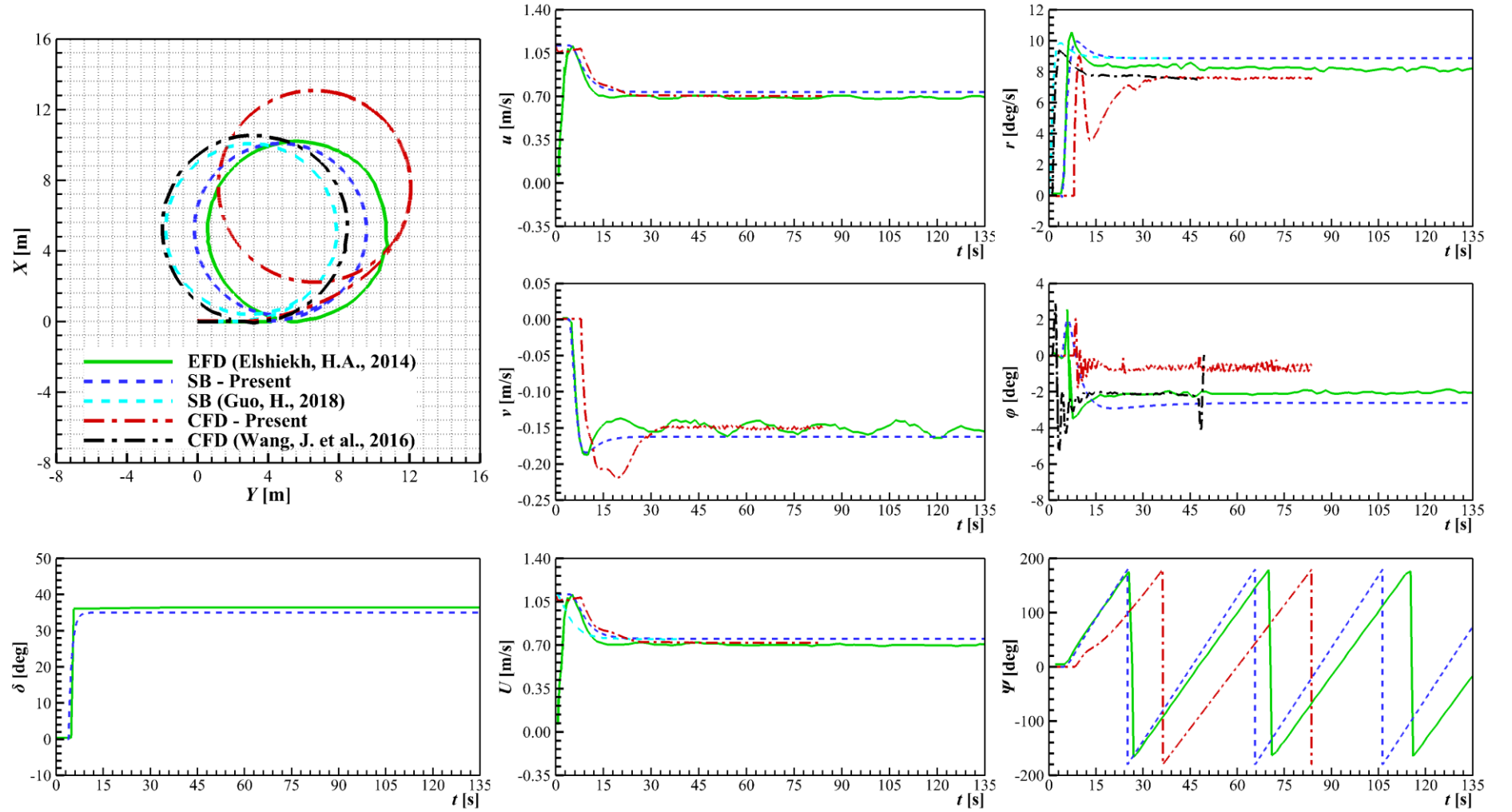


Figure 5.4 : Comparison of turning trajectories and time histories in calm water, $\delta = 35^\circ$ and $\psi = 0^\circ$.

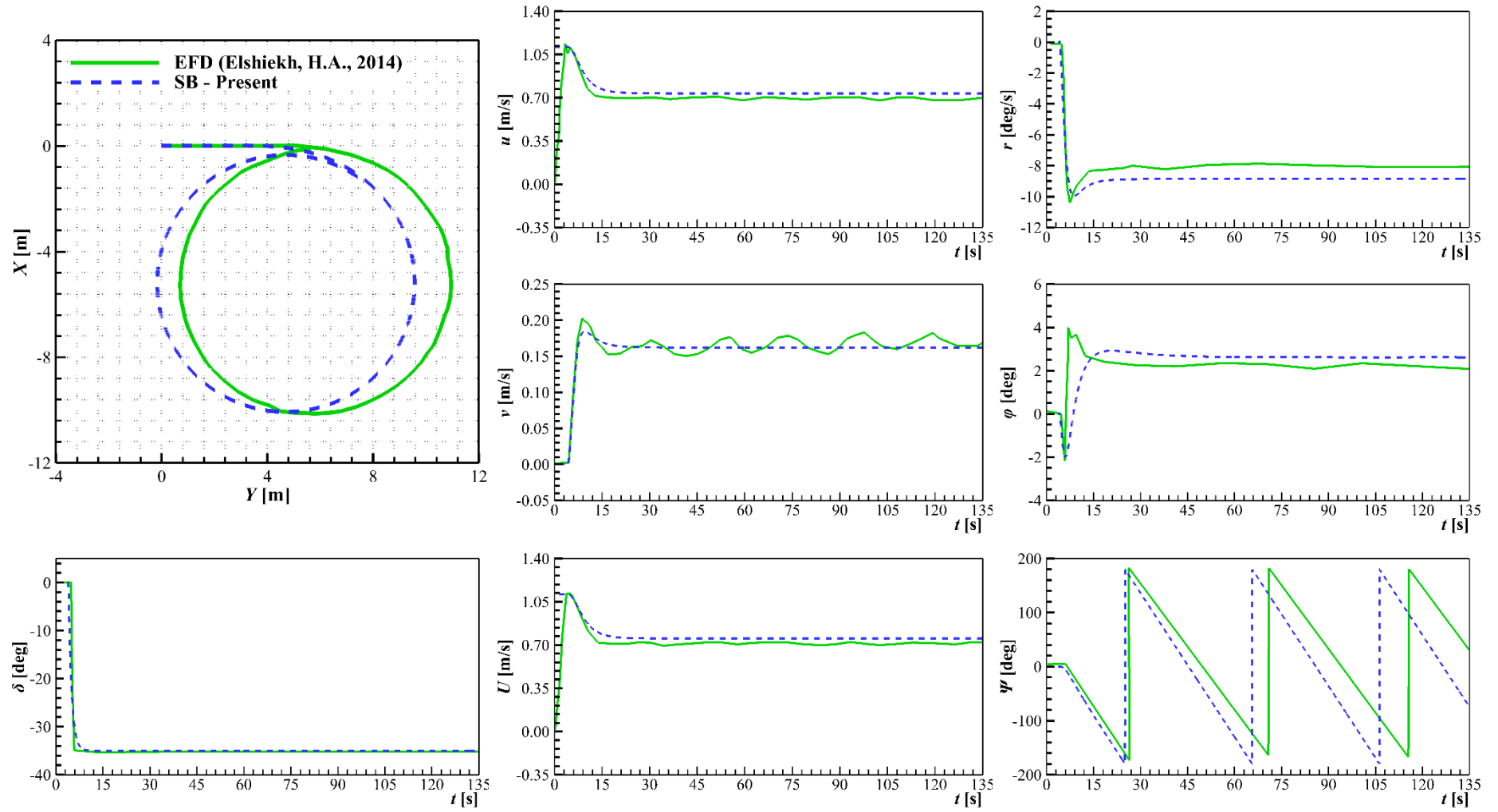


Figure 5.5 : Comparison of turning trajectories and time histories in calm water, $\delta = -35^\circ$ and $\psi = 0^\circ$.

After the examination of $\delta = \pm 35^\circ$ for different heading angles, S-B simulations were conducted at different rudder angles ($\delta = \pm 15^\circ, \pm 25^\circ$). Table 5.12 and Table 5.13 show the results of the vessel trajectories, advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), and steady turning speed (STS) for different heading angles at $\delta = \pm 15^\circ$ and $\delta = \pm 25^\circ$ in tabular format, respectively. The $\delta = \pm 35^\circ$ conditions is also summarized in Table 5.14. In the case of calm water, it is possible to make a general inference from the simulations: Despite the change in the direction angle, it was observed that the maneuver parameters were almost at the same level in calm water conditions. A comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = 15^\circ, 25^\circ, 35^\circ$, and $\psi = 0^\circ$ are illustrated in Figure 5.6. Some basic deductions can be made from this figure:

- Larger rudder angles result in lower advance and transfer, while smaller rudder angles lead to more advance and transfer.
- The rudder angle directly affects the ship's turning diameter. A larger rudder angle will result in a tighter turn with a smaller turning diameter, while a smaller rudder angle will lead to a wider turn with a larger turning diameter.
- As the rudder angle increases, the turning trajectory gets smaller. The steady turning diameter parameter increases as the rudder angle gets smaller.
- As the rudder angle increases, the velocities u , v , and the resultant velocity U decrease.
- A larger rudder angle will cause the ship to change its heading more rapidly, resulting in a higher steady yaw rate. Conversely, a smaller rudder angle will lead to a slower rate of turn.
- As the rudder angle increases, the roll angle decreases and becomes stable faster.
- The time it takes for the ship to complete a turn is affected by the rudder angle. A sharper turn (larger rudder angle) will result in a quicker turning time, while a more gradual turn (smaller rudder angle) will take more time to complete.

The turning trajectories and time histories for other headings are also presented in Figure H.1 through Figure H.7 in Appendix H.

Table 5.12 : Comparison of trajectory parameters at different heading angles in calm water, $\delta = \pm 15^\circ$.

	δ [deg]	15	-15	15	-15	15	-15	15	-15
	ψ [deg]	0	0	90	90	180	180	-90	-90
Fr = 0.2	<i>Ad</i> [m]	16.2874	16.2889	16.2874	16.2889	16.2874	16.2889	16.2874	16.2889
	<i>Tr</i> [m]	10.8743	10.8738	10.8743	10.8738	10.8743	10.8738	10.8743	10.8738
	<i>TD</i> [m]	23.9574	23.9610	23.9574	23.9610	23.9590	23.9610	23.9574	23.9610
	<i>STD</i> [m]	23.8092	23.8134	23.8103	23.8131	23.8134	23.8129	23.8809	23.8145
	<i>SYR</i> [deg/s]	4.9312	4.9304	4.9312	4.9304	4.9312	4.9304	4.9312	4.9304
	<i>STS</i> [m/s]	1.0246	1.0246	1.0246	1.0246	1.0246	1.0246	1.0246	1.0246

Table 5.13 : Comparison of trajectory parameters at different heading angles in calm water, $\delta = \pm 25^\circ$.

	δ [deg]	25	-25	25	-25	25	-25	25	-25
	ψ [deg]	0	0	90	90	180	180	-90	-90
Fr = 0.2	<i>Ad</i> [m]	10.5252	10.5262	10.5252	10.5262	10.5252	10.5262	10.5252	10.5262
	<i>Tr</i> [m]	5.8870	5.8868	5.8870	5.8868	5.8870	5.8868	5.8870	5.8868
	<i>TD</i> [m]	13.7242	13.7282	13.7242	13.7282	13.7242	13.7282	13.7241	13.7282
	<i>STD</i> [m]	13.5517	13.5548	13.5523	13.5547	13.5585	13.5546	13.5515	13.5554
	<i>SYR</i> [deg/s]	7.5066	7.5054	7.5066	7.5054	7.5066	7.5054	7.5066	7.5054
	<i>STS</i> [m/s]	0.8877	0.8878	0.8878	0.8878	0.8877	0.8878	0.8877	0.8878

Table 5.14 : Comparison of trajectory parameters at different heading angles in calm water, $\delta = \pm 35^\circ$.

	δ [deg]	35	-35	35	-35	35	-35	35	-35
	ψ [deg]	0	0	90	90	180	180	-90	-90
Fr = 0.2	<i>Ad</i> [m]	8.3475	8.3483	8.3475	8.3483	8.3475	8.3483	8.3475	8.3483
	<i>Tr</i> [m]	4.1245	4.1246	4.1245	4.1246	4.1245	4.1246	4.1245	4.1246
	<i>TD</i> [m]	9.9557	9.9595	9.9557	9.9595	9.9557	9.9595	9.9557	9.9595
	<i>STD</i> [m]	9.7231	9.7257	9.7244	9.7257	9.7435	9.7257	9.7230	9.7270
	<i>SYR</i> [deg/s]	8.8637	8.8626	8.8637	8.8626	8.8637	8.8626	8.8637	8.8626
	<i>STS</i> [m/s]	0.7521	0.7522	0.7521	0.7522	0.7521	0.7522	0.7521	0.7522

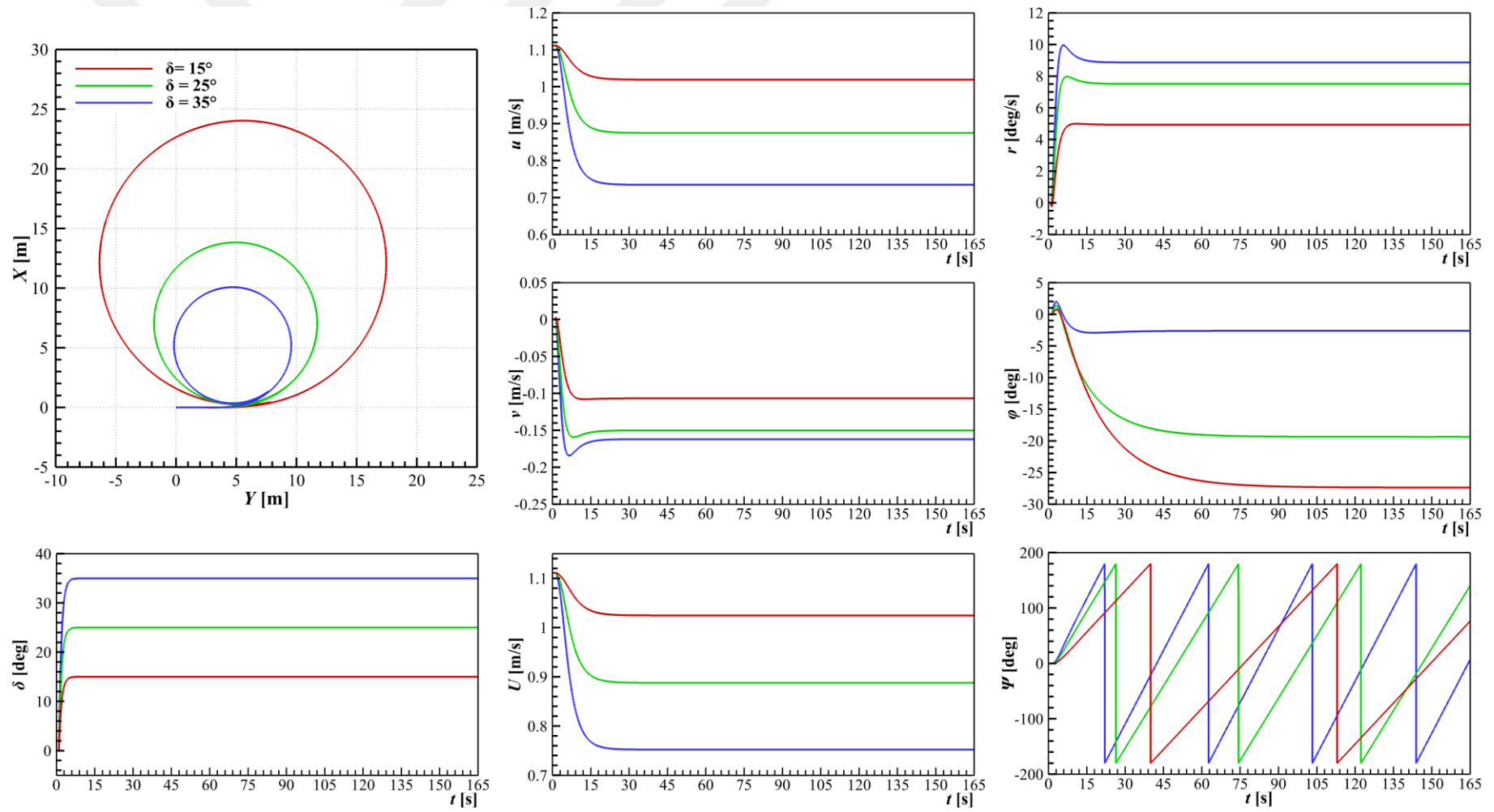


Figure 5.6 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = 15^\circ, 25^\circ, 35^\circ$ and $\psi = 0^\circ$.

5.7.4 Turning circle results in waves

The longitudinal force, lateral force, and yaw moment values acting on the model were measured in the experimental study carried out in different wave conditions at 15-degree intervals in the 0-180 degree heading angle range. The regular wave conditions are listed below:

- $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$
- $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$
- $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$

Wave conditions are listed in the order in which they were applied experimentally. In the first case, the wave height is 0.063 m and the wave length is 3.147 m. In the second case, the wave height and length are halved. In the last case, the case where the wavelength is equal to the ship's length and is 3.147 m and the wave height is 0.079 m has been examined. This set has the feature of having the highest wave height of the experimental set.

Mean wave drift forces and moments corresponding to each heading angle point were determined by FFT analysis. By fitting the polynomial from these points, a curve representing the wave loads corresponding to all heading angles in the range of 0-180 degrees was obtained. The curve representing the wave loads in the 180-360 degree range is taken as the symmetry of the curve in the 0-180 degree heading angle range. This curve fitting process is completed separately for longitudinal wave drift force, lateral wave drift force, and wave drift yaw moment. Mean wave drift forces and moment are added to the X , Y , and N equations in the equations of motion, and the K equation expressing the roll moment is represented only by loads originating from the hull, propeller, and rudder. Eventually, system-based simulation is evaluated to solve the turning circle of ONR Tumblehome with $\pm 35^\circ$ rudder angles under regular wave conditions at $Fr = 0.2$. The incident wave angles, χ , were 0° , $\pm 90^\circ$, and 180° which means the wave conditions cover the following sea, quartering seas, and head sea, respectively.

System-based simulation results of advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), steady turning speed (STS) at $\delta = 35^\circ$ and $\chi = 180^\circ$ are listed in Table 5.15 for all regular wave conditions. A

positive rudder angle causes portside turning and a wave incident angle of 180° corresponds to the head sea condition. All S-B simulation results were compared with the experimental study by Elshiekh (2014). However, the S-B results for $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ are presented without any comparison due to a lack of experimental data.

The turning trajectories and time histories of rudder angle, surge velocity, lateral velocity, resultant velocity, yaw rate, roll angle, and ship heading at $\delta = 35^\circ$ and $\chi = 180^\circ$ are illustrated in Figure 5.7, Figure 5.8, and Figure 5.9 for regular wave conditions of “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ”, “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ”, and “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”, respectively.

Table 5.15 : Comparison of trajectory parameters in regular waves, $\delta = 35^\circ$ and $\chi = 180^\circ$.

	H/λ	λ/L_{WL}	Ad [m]	Tr [m]	TD [m]	STD [m]	SYR [deg/s]	STS [m/s]
EFD	0.020	1.00	6.9965	3.7817	9.4334	N/A	7.9028	0.6850
S-B			8.2505	3.9015	9.8727	9.5862	8.8331	0.7530
Error [%]			17.93	0.39	2.41	N/A	10.30	4.42
EFD	0.020	0.50	7.3010	3.9608	9.7092	N/A	8.0836	0.7085
S-B			8.2809	3.9710	9.8470	9.5755	8.8572	0.7532
Error [%]			13.42	0.26	1.42	N/A	9.57	6.31
S-B	0.025	1.00	8.2178	3.7507	9.7997	9.5203	8.8149	0.7536

In the head wave and portside turning condition, i.e. $\chi = 180^\circ$ and $\delta = 35^\circ$, all parameters were slightly overestimated at different levels with S-B simulations compared to the experimental results. However, present S-B results in all wave conditions are in good agreement with experimental results except for the advance parameter. This difference in advance value was also seen in the calm water results. As described in the previous section, the comparison values are in doubt due to uncertainty about the conditions under which the experimental turning trajectory is presented.

Transfer, Tr , and tactical diameter, TD , display almost identical values to experimental values. These parameters can also be related to the wave parameters: As Tr increased, TD decreased as the wavelength and height decreased by half. While increasing the wave height at a fixed wavelength decreased the Tr value, a slight decrease was

observed in the TD value. In other words, the ship's reach from 0° heading to 90° heading was achieved by traveling a shorter distance in the x - and y -axes.

Experimental STD values are not presented as data in the source. STD values are not achieved from the experimental graphs to avoid an error in the quantitative comparison. A visual comparison for STD values can be made from the turning trajectory graphs that follow. The turning trajectory was computed in similar time periods with the experimental one so that the comparison could be made properly. It can be seen that the STD values are close to each other.

In the S-B results of the first wave condition, see Figure 5.7, it is seen that the turning trajectory has less drift in the $-x$ direction than the experimental one. The mean wave sway forces obtained from the steady drift tests can be evaluated here: If the mean wave sway force is higher, it can be predicted that the results will be closer to the experimental results. The graph that supports this inference can be seen in Figure 5.10. Here, an increase in the mean wave sway force means that the drift of the model in the $-x$ direction will be greater. It can also be seen in Figure 5.8 that the drift in y direction is more pronounced in the second case where the wavelength is shorter and the wavelength is smaller.

A steady yaw rate was achieved by averaging over a given period of time-dependent yaw rate. The difference between the steady yaw rate values offered by the solution methods is around 10% in any case. However, when the waveform changes, the behavior of the SYR parameter also changes in the same way as in the experiments: As the wavelength and height decrease, the SYR value increases for these conditions. In the case where only the wave height was increased, the SYR decreased.

Steady turning speed was obtained by averaging over a given period of time-dependent resultant velocity, U . Considering the simulations performed in three different wave conditions, no significant difference was observed in the STS value. The overestimated values differ by around 5% compared to the experimental results.

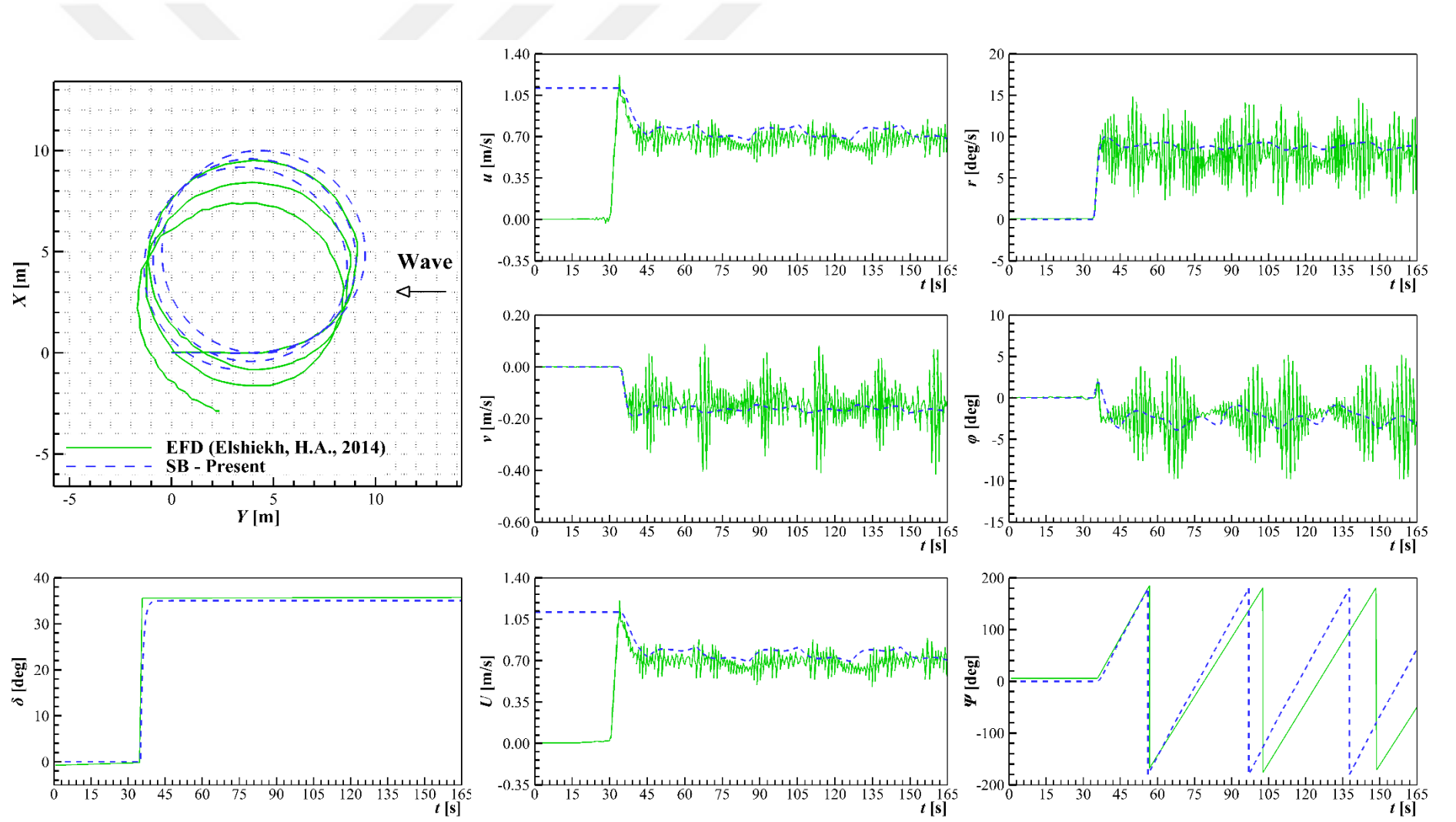


Figure 5.7 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

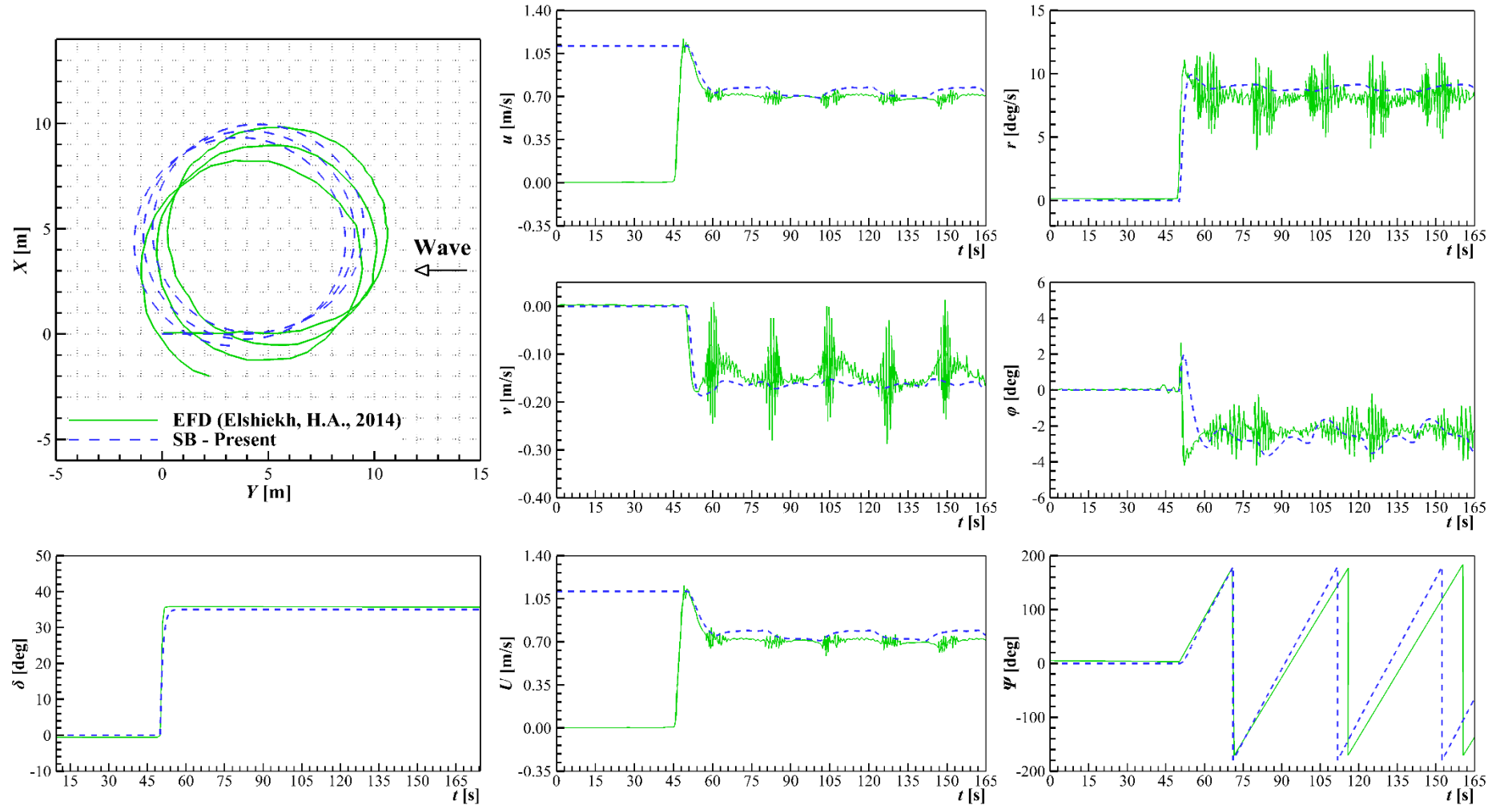


Figure 5.8 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$).

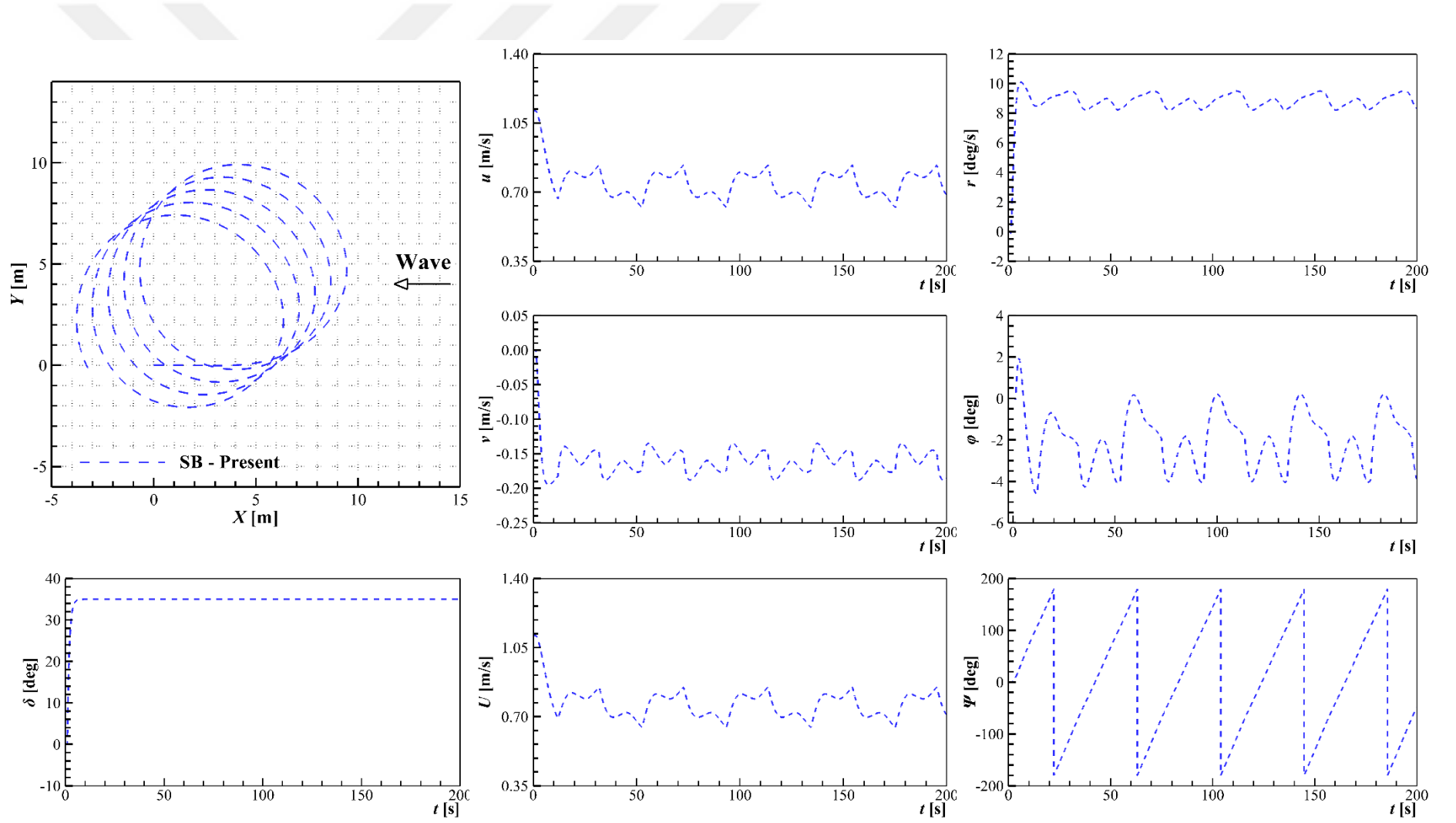


Figure 5.9 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).

System-based simulation results of advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), steady turning speed (STS) at $\delta = -35^\circ$ and $\chi = 180^\circ$ are listed in Table 5.16 for all regular wave conditions. The head wave condition was evaluated under the starboard side turning condition.

The turning trajectories and time histories of rudder angle, surge velocity, lateral velocity, resultant velocity, yaw rate, roll angle, and ship heading at $\delta = -35^\circ$ and $\chi = 180^\circ$ are illustrated in Figure 5.10, Figure 5.11, and Figure 5.12 for all regular wave conditions of “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ”, “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ”, and “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”, respectively. Comparison with experimental data is not possible for the last wave condition and the simulation is set to 200 seconds, unlike the others.

Table 5.16 : Comparison of trajectory parameters in regular waves, $\delta = -35^\circ$ and $\chi = 180^\circ$.

Source	H/λ	λ/L_{WL}	Ad [m]	Tr [m]	TD [m]	STD [m]	SYR [deg/s]	STS [m/s]
EFD	0.020	1.00	6.9978	3.8797	9.5848	N/A	7.8850	0.6948
S-B			8.3170	4.1631	9.6774	9.6045	8.8653	0.7517
Error [%]			18.85	7.30	0.97	N/A	12.43	8.19
EFD	0.020	0.50	7.2614	3.9066	9.6486	N/A	8.0833	0.7062
S-B			8.2829	4.1186	9.8182	9.6038	8.8714	0.7533
Error [%]			14.07	5.43	1.76	N/A	9.75	6.67
S-B	0.025	1.00	8.2916	4.1925	9.4998	9.5674	8.8403	0.7480

In the scenario of encountering head waves and executing a starboard side turning ($\delta = -35^\circ$), the S-B simulations consistently showed a slight overestimation of various parameters when compared to the experimental outcomes. Nonetheless, the current S-B results exhibit satisfactory alignment with experimental findings across all wave conditions, except for the advance parameter. Notably, a similar discrepancy in the advance parameter was observed in portside turning.

The transfer value was estimated with less than 10% difference from the experimental results in each wave condition. The transfer parameter decreased when the wavelength and height were halved in S-B simulations, unlike portside turning. The largest transfer value in starboard side turning was seen in the “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”

condition. In portside turning, the highest value was calculated in the “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ” condition.

The turning diameter value was calculated with less than a 5% difference from the experimental outcomes in each wave condition. The smallest TD value appeared in the “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ” condition as in port side turning; however, there was no significant change in TD when the wave characteristic changed.

The STD value was almost unchanged when the wavelength and height were reduced equally, with the smallest estimated in case “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”. To compare STD values with experimental data, see Figure 5.10, Figure 5.11, and Figure 5.12. In all conditions, the ship performed the turning circle by drifting on a diagonal in the $+x$ and $-y$ directions, which is consistent with the experiments. In case of “ $H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$ ”, the drift in the y direction appears to be less than the experimental one. On the other hand, in case “ $H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$ ”, the ship seems to start turning earlier in the S-B simulations compared to the experimental environment. Since there is no experimental output, a comparison cannot be made for the “ $H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$ ” case.

Considering the differences in the estimation of the SYR value, it is seen that it is not very successful, especially in the first case: Absolute SYR values were estimated larger. Still, the difference between results is around 10 percent. In addition, it can be said that the effect of wave characteristics on SYR parameter is low. As the wavelength and height decrease, the SYR value shows a slight increase. In the case where only the wave height was increased, the SYR decreased.

The time-dependent behaviors of u , v , and U appear to be in agreement with the experimental results. In particular, it can be seen that the surge velocity is higher than the experimental surge velocity and these differences are reflected in the resultant velocity. Again, higher STS values were obtained when compared to the experimental results. As in portside turning, the change in wave conditions affected the STS parameter very little.

Finally, looking at the roll angle graphs, it can be said that the roll angle exhibits similar behavior with the experimental values.

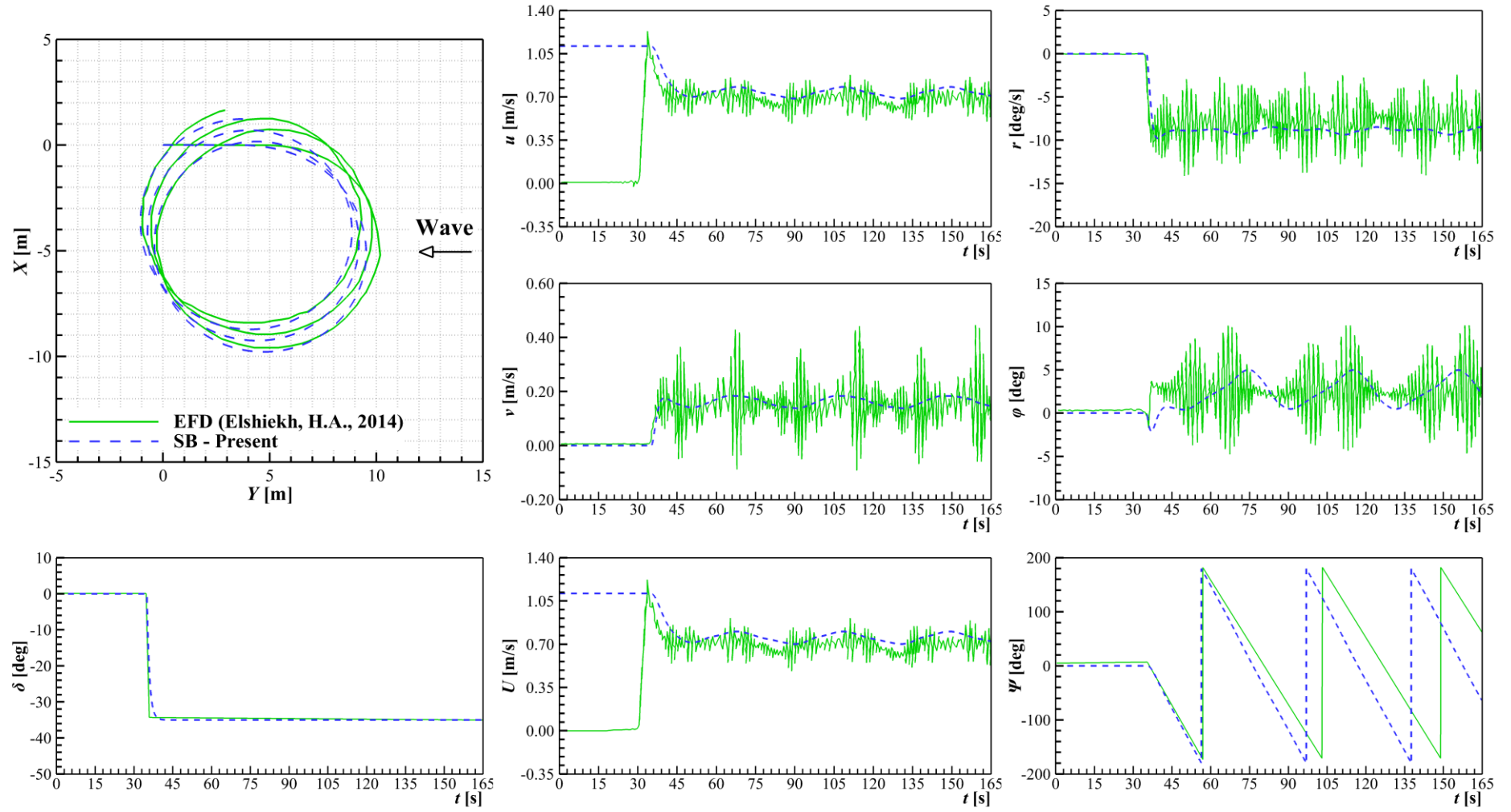


Figure 5.10 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

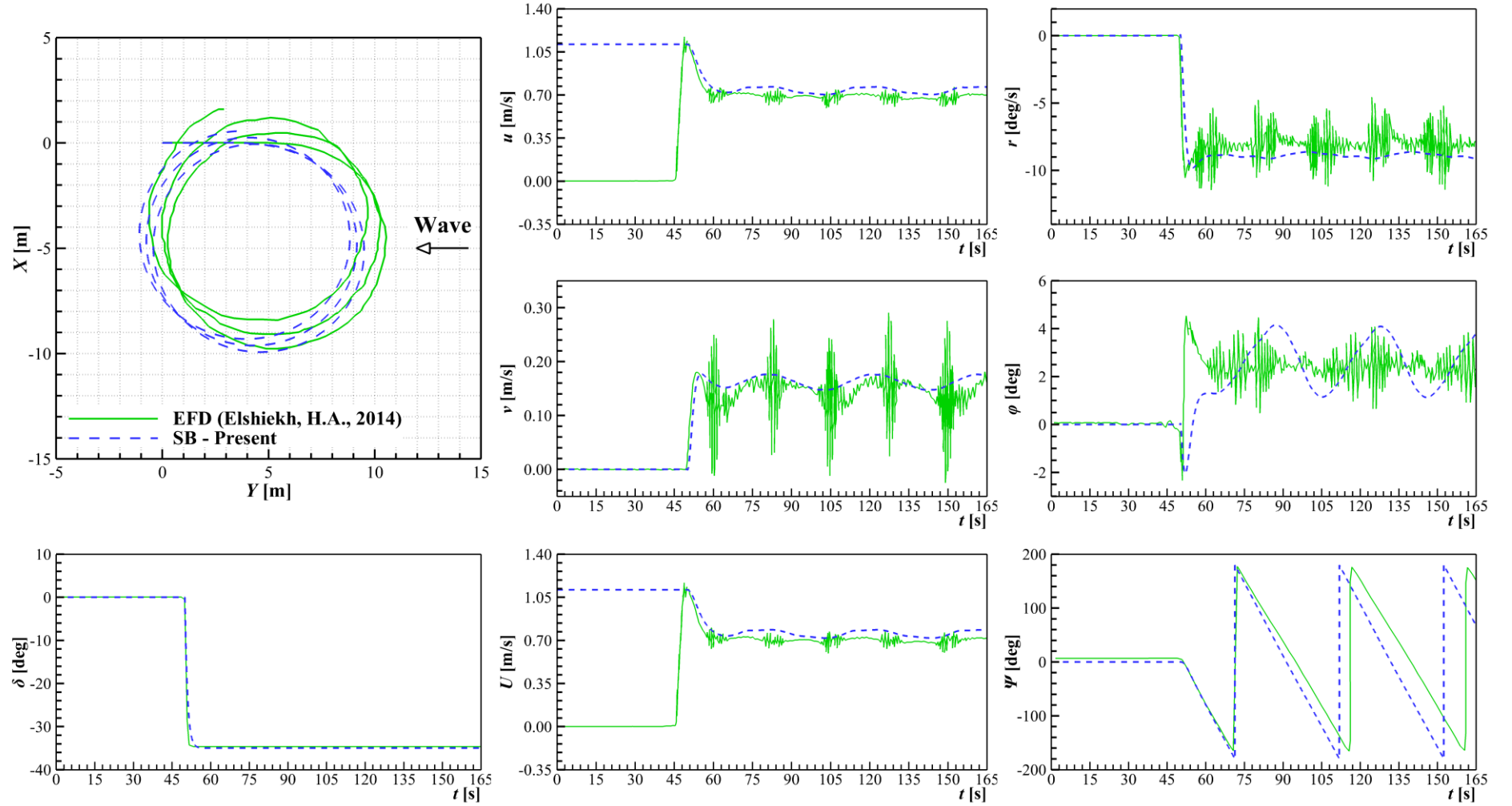


Figure 5.11 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$).

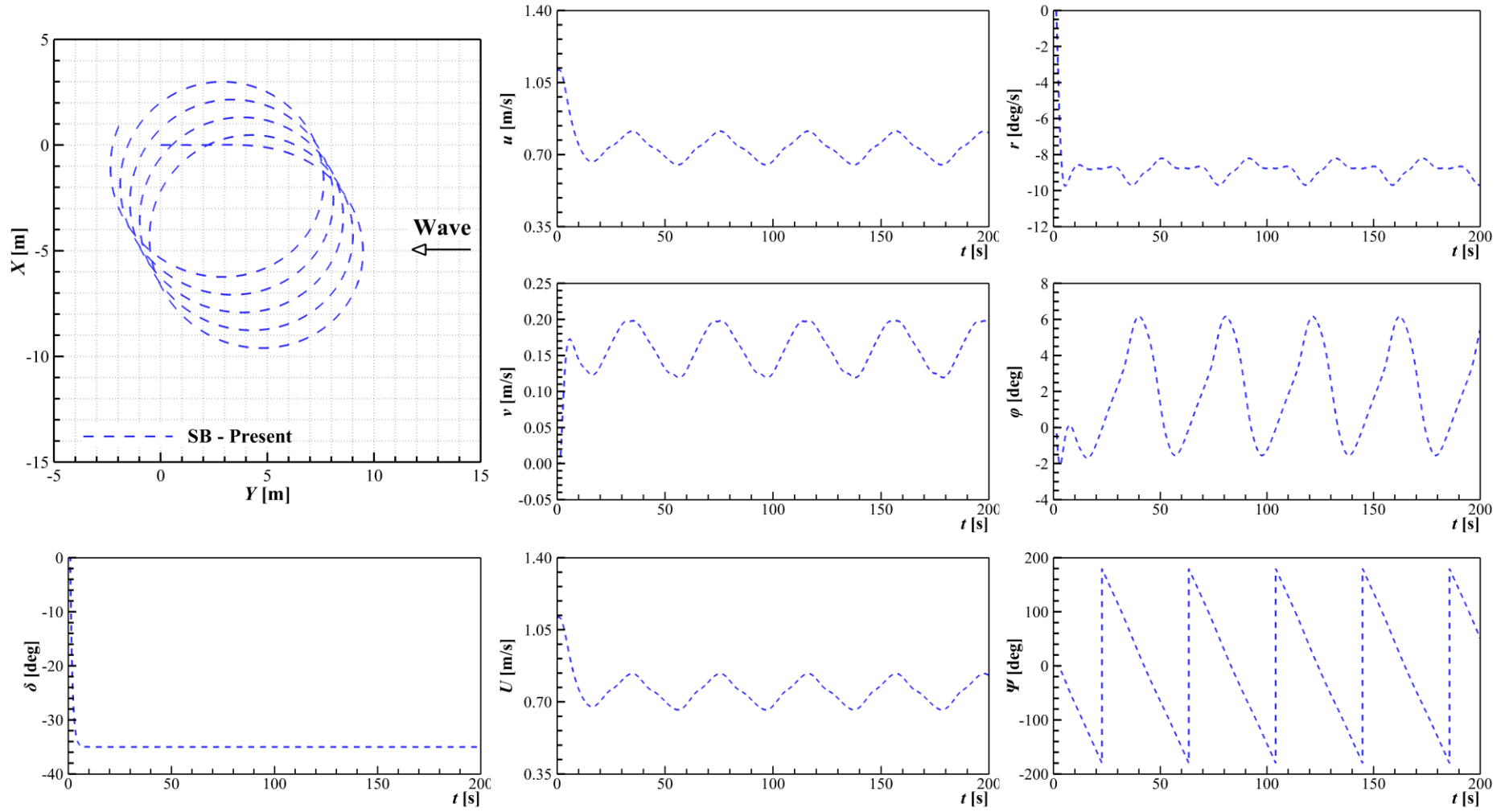


Figure 5.12 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 180^\circ$ ($H/\lambda = 0.025$, $\lambda/L_{WL} 1.00$).

System-based simulation results of advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), steady turning speed (STS) at $\delta = 35^\circ$ and $\chi = 90^\circ$ are listed in Table I.1 for all regular wave conditions. The comparative results followed by results for $\delta = -35^\circ$ and $\chi = 90^\circ$ in Table I.2.

The turning trajectories and time histories of rudder angle, surge velocity, lateral velocity, resultant velocity, yaw rate, roll angle, and ship heading at $\delta = 35^\circ$ and $\chi = 90^\circ$ are illustrated in Figure I.1, Figure I.2, and Figure I.3 for wave conditions of “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ”, “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ”, and “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”, respectively. The comparative figures followed by results for $\delta = -35^\circ$ and $\chi = 90^\circ$ in Figure I.4, Figure I.5, and Figure I.6.

System-based simulation results of advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), steady turning speed (STS) at $\delta = 35^\circ$ and $\chi = 0^\circ$ are listed in Table I.3 for all regular wave conditions. The comparative results followed by results for $\delta = -35^\circ$ and $\chi = 0^\circ$ in Table I.4.

The turning trajectories and time histories of rudder angle, surge velocity, lateral velocity, resultant velocity, yaw rate, roll angle, and ship heading at $\delta = 35^\circ$ and $\chi = 0^\circ$ are illustrated in Figure I.7, Figure I.8, and Figure I.9 for wave conditions of “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ”, “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ”, and “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”, respectively. The comparative figures followed by results for $\delta = -35^\circ$ and $\chi = 0^\circ$ in Figure I.10, Figure I.11, and Figure I.12.

System-based simulation results of advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), steady turning speed (STS) at $\delta = 35^\circ$ and $\chi = -90^\circ$ are listed in Table I.5 for all regular wave conditions. The comparative results are followed by results for $\delta = -35^\circ$ and $\chi = -90^\circ$ in Table I.6.

The turning trajectories and time histories of rudder angle, surge velocity, lateral velocity, resultant velocity, yaw rate, roll angle, and ship heading at $\delta = 35^\circ$ and $\chi = -90^\circ$ are illustrated in Figure I.13, Figure I.14, and Figure I.15 for wave conditions “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ”, “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ”, and “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”, respectively. The comparative figures followed by results for $\delta = -35^\circ$ and $\chi = -90^\circ$ in Figure I.16, Figure I.17, and Figure I.18.

The bar charts clearly illustrate the maneuvering parameters for each rudder angle and wave incident angle in terms of wave characteristics. Bar charts were created in order to more clearly express the relationship between wave height and length characteristics and maneuver parameters. In each bar chart, there are the data of the portside and starboard side turn in the relevant heading angle. However, it should be noted that the difference between the values is very small when making the evaluation. There are wave conditions on the x -axis of the bar charts. Wave conditions are ordered from largest to smallest according to the values obtained from mean wave drift forces and moments analysis. On the y -axis, the corresponding value of the maneuver parameter, i.e. advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), and steady turning speed (STS), in each wave conditions is located. In the case of head sea the following outcomes have been observed (Figure 5.13):

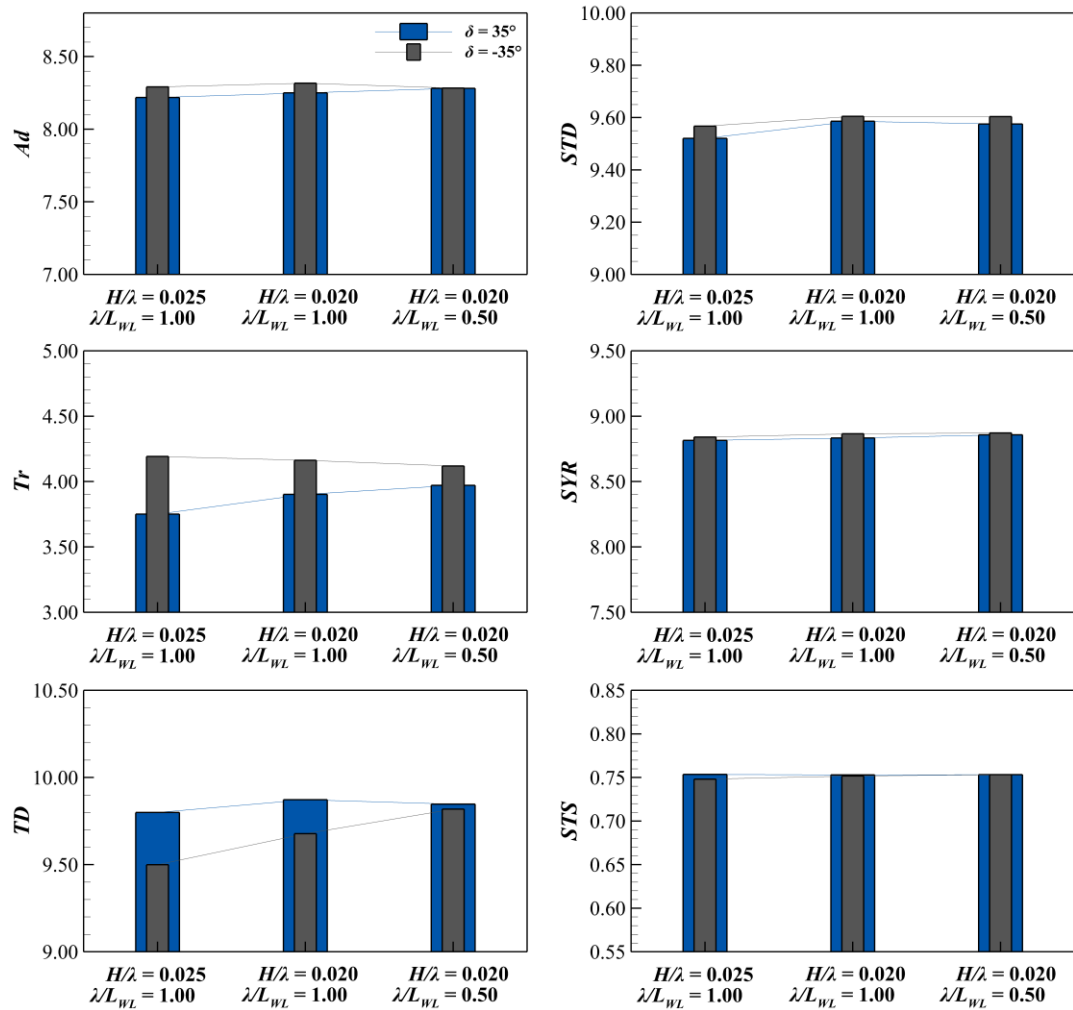


Figure 5.13 : Comparison of maneuvering parameters in all wave conditions, $\delta = \pm 35^\circ$ and $\chi = 180^\circ$.

- The case where the Ad value is maximum is “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ” and the case where it is minimum “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ” for $\delta = 35^\circ$. By a slight margin the “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ” is highest for $\delta = -35^\circ$. Increasing the wavelength at the fixed wave height caused a decrease in the Ad value. On the other hand, decreasing the wavelength and height at the same rate caused an increase in the case of $\delta = -35^\circ$. When the rudder angle was -35° , i.e. starboard side turning, the Ad value is higher in every wave condition compared to $\delta = 35^\circ$. The values are almost the same in both turnings at “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ”.
- The case where the value of Tr is maximum is “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ” and the case where it is minimum “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ” in starboard side turning. Conversely, the case where the value of Tr is maximum is “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ” and the case where it is minimum “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ” in portside turning. Graphs have reverse behavior in portside and starboard side turns. In the case where the rudder angle is 35° , i.e. port side turning, the Tr value decreased in each wave condition compared to $\delta = -35^\circ$.
- The case where the TD value is maximum is “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ” and the case where it is minimum is “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ” in portside turning. While starboard side turning takes the minimum value in the same wave condition, the maximum value is seen differently. Increasing the wave height at constant wavelength decreased the TD value in both turning. Graphs have almost similar behavior in portside and starboard side turns. In the case where the rudder angle is -35° , i.e. starboard side turning, the TD value decreased in each wave condition compared to $\delta = 35^\circ$.
- The case where the STD value is maximum is “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ” and the case where it is minimum is “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”. Increasing the wave height at a constant wavelength decreased the STD value. Also, decreasing the wavelength and height at the same rate resulted in a very small decrease in the STD value. Graphs have similar behavior in portside and starboard side turns. Turning direction and wave characteristics did not have a serious effect on STD .

- It was observed that the changes in wave characteristics and turning direction did not have a serious effect on the *SYR* parameter.
- The *STS* value remained almost the same as the wave characteristics and turning direction changed.

Comparison of maneuvering parameters in all wave conditions for $\delta = \pm 35^\circ$ and $\psi = 90^\circ$ are illustrated in Figure 5.14. In the case of $\chi = 90^\circ$ the following outcomes have been observed.

- As the mean wave loads decreased, the advance parameter gradually decreased. This output is valid for both turnings. When the rudder angle was 35° , i.e. port side turning, the *Ad* value is higher in every wave condition compared to $\delta = -35^\circ$.
- As the mean wave loads decreased, the transfer parameter gradually decreased in both turnings. The minimum *Tr* value is seen at “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ”, while the maximum value is at “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”. In any case, starboard side turning outcomes are lower than portside turning outcomes.
- In all wave conditions, the *TD* value was significantly higher in the portside turning condition. As the mean wave loads decreased, the *TD* value gradually decreased in both turnings.
- The *STD* value is unaffected by the change of wave characteristics and its general behavior is the same in both turning conditions. The maximum value of *STD* is observed at “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ”.
- The *SYR* value is slightly smaller for portside turning than starboard turning in all wave conditions. In general, *SYR* slightly increased when wave loads decreased.
- The *STS* value is unaffected by the change of wave characteristics and turning directions.

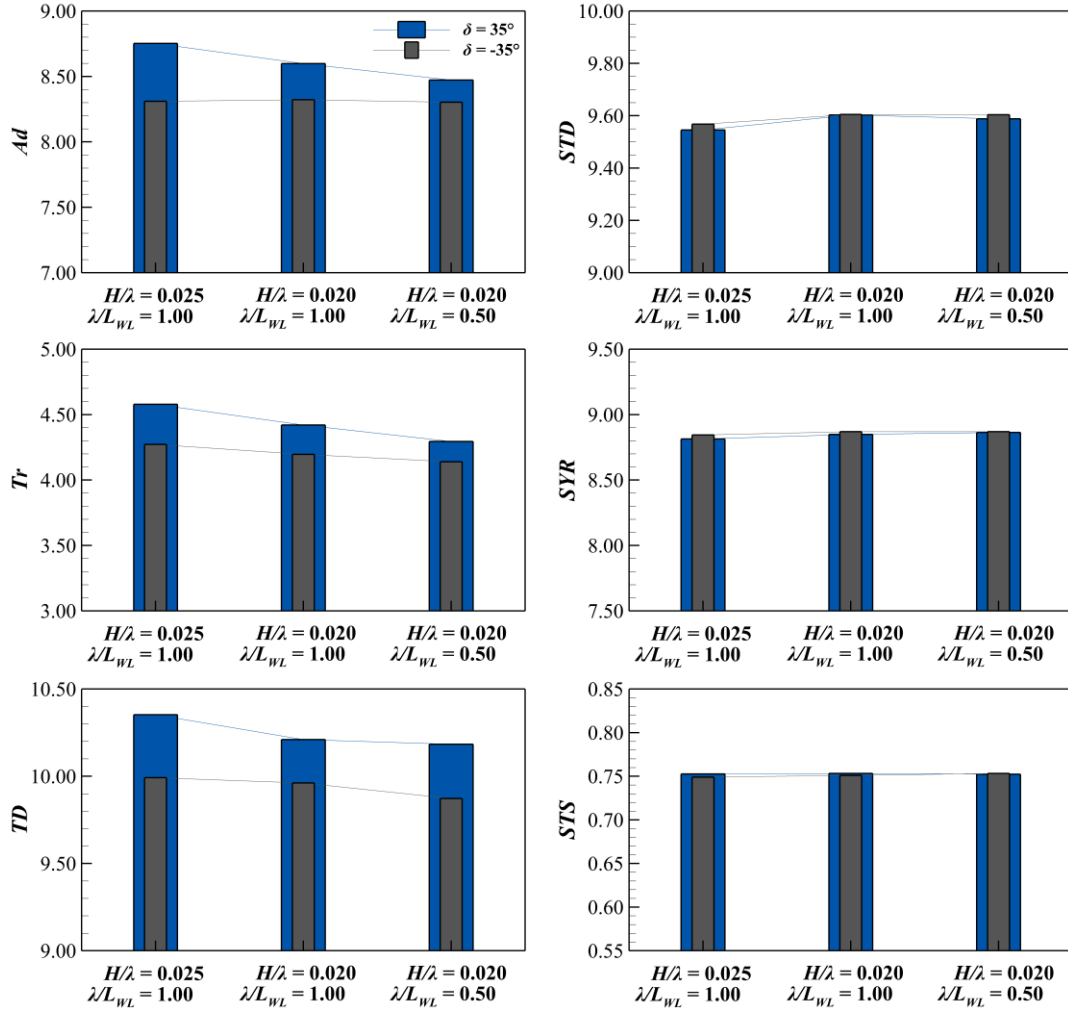


Figure 5.14 : Comparison of maneuvering parameters in all wave conditions, $\delta = \pm 35^\circ$ and $\chi = 90^\circ$.

Figure 5.15 illustrates the comparative results for the following waves and the findings can be listed below:

- The advance value showed increasing behavior as the wave load decreased in both portside turning and starboard side turning. The values of the $\delta = 35^\circ$ case are higher than the values of the $\delta = -35^\circ$ case.
- The Tr value remained almost constant at $\delta = -35^\circ$. At $\delta = 35^\circ$, however, it exhibited a decreasing behavior as the wave load decreased. The values of the $\delta = 35^\circ$ case are higher than the values of the $\delta = -35^\circ$ case.
- Tactical diameter exhibited opposite behavior in both turning conditions. While the highest value was seen at “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ”, the lowest value was seen at “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ” at portside

turning. Contrarily, the highest value of TD is calculated where the mean wave loads are minimum.

- The STD value is nearly unaffected by the change of wave characteristics and its general behavior is the same in both turning conditions. The maximum value of STD is observed at “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ”. However, the change in STD is very low despite changing the wave properties.
- Steady yaw rate exhibited similar behavior in both turning conditions. The values of SYR are almost identical for both turnings and wave conditions.
- The values of STS are almost identical for both turnings and wave characteristics.

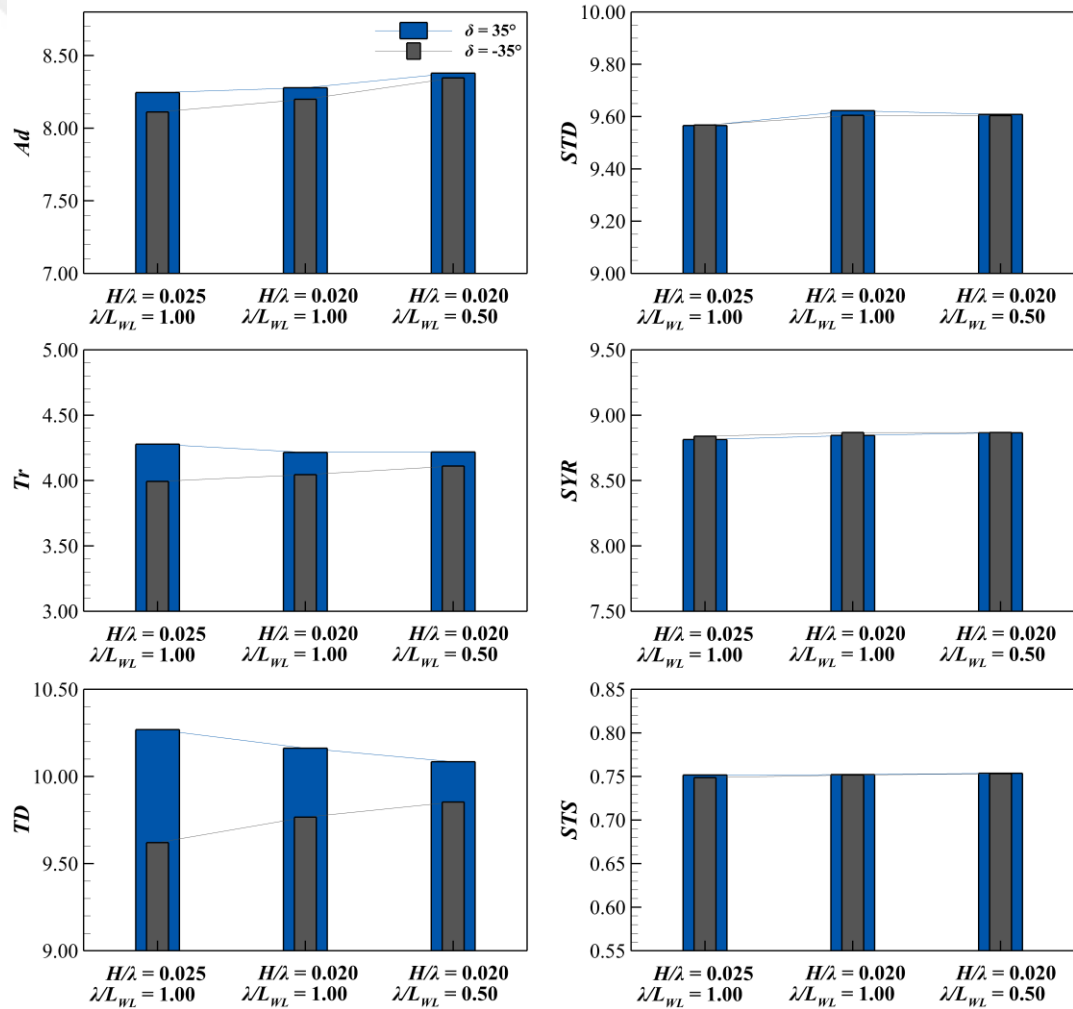


Figure 5.15 : Comparison of maneuvering parameters in all wave conditions, $\delta = \pm 35^\circ$ and $\chi = 0^\circ$.

Figure 5.16 depicts the comparative results for $\chi = -90^\circ$ and the outcomes can be summarized as follows:

- The advance value showed increasing behavior as the wave load decreased in both portside turning and starboard side turning. It has been observed that the changes in the advance parameter with the change of turning direction are very small.
- When the rudder angle was -35° , i.e. starboard side turning, the Tr value is higher in every wave condition compared to $\delta = 35^\circ$. Although the Tr parameter in starboard side turning did not change much as the wave properties changed, it increased as mean wave loads decreased in portside turning.
- The most significant difference between portside and starboard side turning was seen in the TD parameter of the $\chi = -90^\circ$ case. When the rudder angle was -35° , i.e. starboard side turning, the TD value is higher in every wave condition compared to $\delta = 35^\circ$. The case where the TD value is maximum is “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ” and the case where it is minimum is “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ” for portside turning condition. This order is reversed in the starboard side turning condition.
- The maximum value of STD is observed at “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ”. However, the change in STD is very low despite changing the wave properties.
- Steady yaw rate exhibited similar behavior in both turning conditions. The values of SYR are almost identical for both turnings and all wave conditions.
- The change in STS is very low despite changing the wave properties and turning direction.

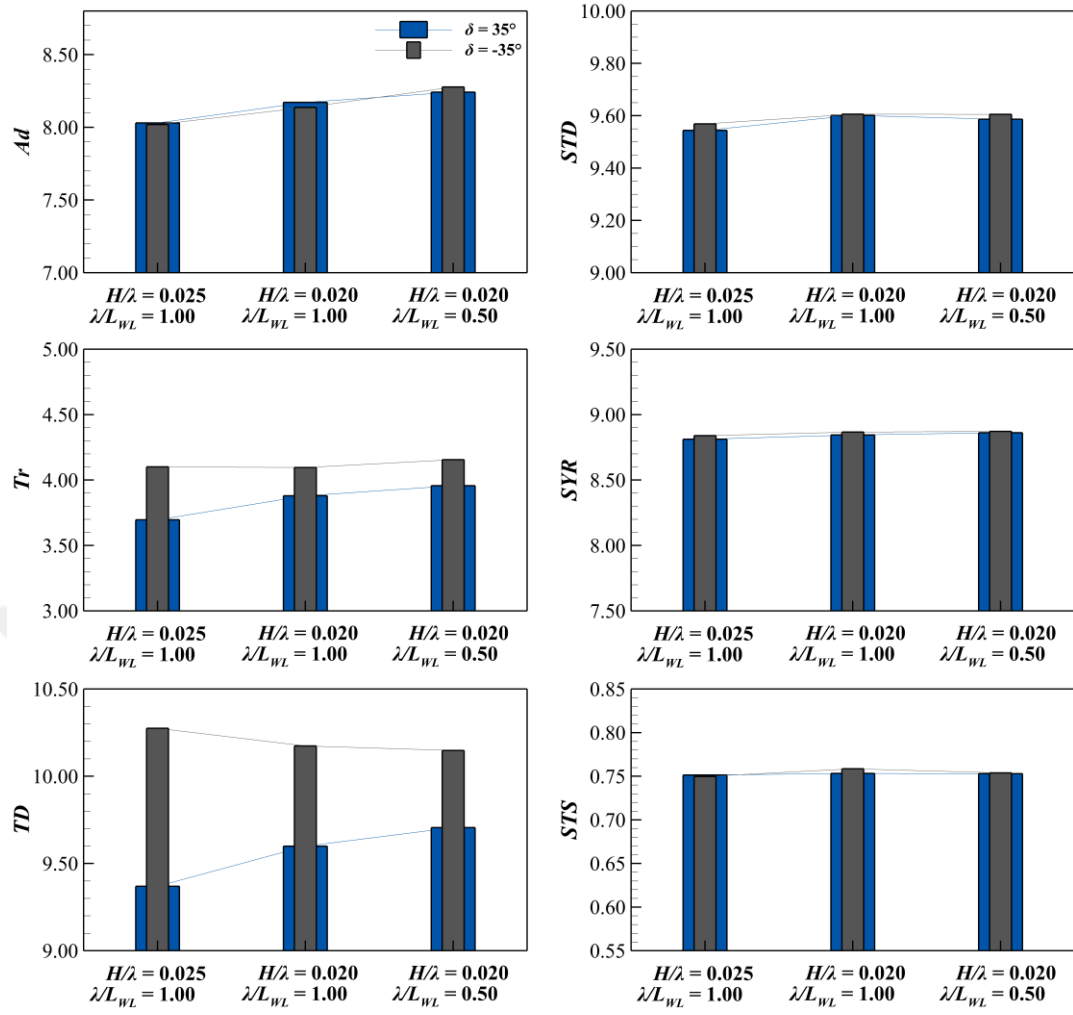


Figure 5.16 : Comparison of maneuvering parameters in all wave conditions, $\delta = \pm 35^\circ$ and $\chi = -90^\circ$.



6. CONCLUSIONS AND RECOMMENDATIONS

Increasing concern has arisen regarding the maneuvering performance of commercial and naval ships operating in realistic environments influenced by wind, currents, and waves. The motivation behind this thesis stems from limited attention to maneuvering in waves. The awareness of critical issues, such as the limited number of experiments on this subject and the lack of systematic studies on wavelength and steepness, coupled with a solid commitment to finding innovative solutions, served as a starting point for this research.

This thesis, titled “Maneuvering of High Speed Displacement Vessels in Regular Waves”, represents a comprehensive exploration of experiments and prediction methods for maneuvering in waves within the context of AVT-348 activity. The research begins with an introduction that highlights the significance of the study and outlines the research question and objectives. A theoretical foundation was established by thoroughly reviewing the literature to set the stage for the subsequent investigation by finding significant gaps and knowledge deficits. The research methodology section summarizes the systematic approach employed in this study, including research methods and techniques, data collection methods, and data analysis techniques. A meticulous presentation and analysis of the calculated or collected data provide valuable insights into maneuvering in waves, offering a nuanced understanding of the phenomenon under investigation.

In this doctoral thesis, the effect of wave-related forces and moments on ship maneuvering performance is investigated by modeling the turning circle maneuver. The research aimed to address the growing demand for system-based models by integrating the wave loads. Through a combination of experimental study, simulation, and theoretical modeling, research made significant contributions to the assessment of experiments and prediction methods for naval ships maneuvering in waves.

The concluding chapter of turning circle simulations distills the key findings and reiterates the contributions made to the field. It emphasizes the practical implications of the research.

6.1 General Remarks

The main objectives of the study were to reveal hull and wave interaction with experimental and numerical methods, gain insight through a comprehensive study of the interactive behavior of wave and hull, and use this interaction to develop a mathematical model of maneuvering in waves. The methodology involved experimental investigation researching the naval ship model in different wave conditions at zero speed with different heading angles. Empirical and analytical approaches to estimating wave drift loads were studied in depth and applied to the conditions of the present problem. Numerical simulations replicating experimental conditions were followed by direct CFD solutions. Findings from experimental and numerical studies were incorporated into system-based modeling.

The primary objectives set forth in the inception of this research have been met with resounding success. The definitions that affect ship maneuvering performance are identified. The subject focused on these external effects is waves, and hull-wave interaction has been revealed by using experimental methods in line with the existing experimental possibilities. Through this study, the working principle of the horizontal planar motion mechanism in ITU Ata Nutku Ship Model Testing Laboratory was understood with the aim of evaluating this knowledge in future studies. It has been conclusively shown that experimentally obtained wave loads can be incorporated into system-based methods, and as a result, the integration of wave forces and moments has significant effects on the ship turning circle. With the mathematical model developed, a step has been taken towards reducing the dependence on numerical and experimental studies. In addition, research into CFD simulations paved the way for more practical and faster solutions.

While this research represents a significant step forward, it is crucial to acknowledge its limitations. The investigation was confined to specific wave steepness and wave/ship length configurations, and further studies should explore a broader spectrum of scenarios. Time constraints affected the study's ability to conduct a thorough numerical simulation. Given the inconsistency in the results, it can be said that the focus on directly-solved numerical simulations needs to be increased. In addition, the experimental research focuses on heading angles in the range of 0° to 180° because the subject model is symmetrical. Since the effect of wave loads will change in non-

symmetrical models, the range of 0-360 degrees should be examined. Additionally, while applying the developed system-based model to other ship models, difficulties may be encountered depending on the maneuvering parameters and geometric characteristics of the hull. As such, future research endeavors should focus on bridging the gap between other ship models and variations of the maneuvering coefficients, rudder, and propeller parameters.

In conclusion, this thesis serves as a progress to the maneuverability in adverse weather conditions by examining the complex challenges associated with hull and wave interaction. For this purpose, a practical experimental approach to measure the mean wave drift forces and moments in a towing tank is presented. Then, the measured wave drift loads are incorporated into the equations of motion. The standard maneuver of turning circle is simulated for a model of ONR Tumblehome for different wave conditions in order to demonstrate the effect of wave drift loads on the trajectories of the vessel. The contributions of this research lie in the development of accurate system-based models that contribute to the advancement of ship maneuvering. The outcomes have practical implications for engineers, academicians, and stakeholders involved in the field of maneuvering.

6.2 Conclusions

By addressing the fundamental research questions, it was demonstrated that it is possible to develop a 4-DoF wave model integrated system-based model that accurately captures the turning circle maneuver parameters. The wave forces and moments were successfully measured by experimental methods and computational methods are validated. The conclusion and main findings are presented in this section for all chapters, respectively.

Chapter 2 presented the empirical and analytical prediction methods to calculate the wave drift loads. These approaches have played a pivotal role in shaping the landscape of research within the field of maneuvering in waves. This thesis has harnessed the empirical/analytical approaches to investigate, compare, and expand upon existing theories. The aim is to gain a clearer vision of the possibilities and challenges that lie ahead. Therefore, prior to experimental and numerical studies, current empirical and analytical approaches have been focused. For this, five different approaches are examined and their formulations are presented step by step. By using the features of

the ONR Tumblehome model, mean longitudinal, and lateral wave drift forces and mean yaw moments were obtained. The ship heading angle was studied in the range of 0-180 degrees, and the effects of three waves with different wave steepness and wave/ship length ratios were examined. The main findings obtained from this part of the study can be listed as follows.

- An important insight has been obtained about the changes in wave loads according to the heading angle. There is a noticeable downward trend between 0° and 90° degrees and an upward trend between 90° and 180° in the mean longitudinal wave drift force. The mean lateral wave drift force reached a peak at 90° and fell to its lowest points at 0° and 180°. Lastly, an oscillation is seen in the mean yaw moment graph.
- Changing the wave characteristics has significant effects on wave drift loads. Reducing wave height and length has decreased the magnitude of wave forces and moments, resulting in lower wave drift loads. A higher wave height leads to larger wave forces and moments acting on the hull.
- The approach by Bergdahl & Kofoed (2015) is not suitable for the ONRT model because presented for the calculation of environmental loads on the buoy.
- The approach by API (2005) is not proper for the ONRT model because the design curves are suitable for mobile offshore drilling unit type vessels.

The experimental evaluation of the 1/48.9 scaled ONRT hull's maneuverability, presented in Chapter 3, involved testing it at zero speed and in three distinct regular wave scenarios. The experiments were performed at the Ata Nutku Ship Model Testing Laboratory within the Department of Naval Architecture and Ocean Engineering at Istanbul Technical University. To ensure the ship's motion in three dimensions, a pantograph system was used during steady drift tests. Before the steady drift tests in-water inclining tests were conducted to determine the metacentric height. The steady drift tests were carried out at different heading angles, spanning from 0° to 180°. A mono-directional flap-type wave generator is used to generate regular waves. These wave scenarios had varying combinations of wave/ship length, and wave steepness: “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ” where $H = 0.063\text{ m}$ and $\lambda = 3.147\text{ m}$, “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$ ” where

$H = 0.031\text{ m}$ and $\lambda = 1.574\text{ m}$, and “ $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$ ” where $H = 0.079\text{ m}$ and $\lambda = 3.147\text{ m}$. The longitudinal force, lateral force, and yaw moment were collected via a load cell to get second-order mean wave drift loads. FFT algorithm is applied during the data reduction procedures and Savitzky-Golay filtering is used to filter the force and moment data. This section of the research yields the following primary findings.

- It has been revealed that maneuvering in waves examination can be performed by using the horizontal planar motion mechanism as an alternative in cases where there is no maneuvering tank and therefore free-running tests cannot be performed.
- While the study by Kinaci & Ozturk (2023) takes an empirical approach, the thesis approaches the same subject from an experimental standpoint. Mean second-order wave drift forces and moments have been successfully determined by steady drift tests carried out at zero speed and different heading angles.
- The appropriate procedure should be applied to obtain mean second-order wave drift loads from time-dependent force and moment signals obtained from experiments conducted in waves. Here, although the FFT algorithm is suitable for switching to the frequency domain, it has been revealed that filtering is needed to accurately predict the value corresponding to the zeroth frequency. Savitzky-Golay filtering, which is applied to a certain part of the frequency domain before reaching the peak, has been successful.
- The parameters used in the application of the Savitzky-Golay filter could be different according to the frequency signals.
- The experimental longitudinal wave drift forces and two approaches (Advanced Marine Enterprises Inc., 1996; DNV AS, 2021) illustrated a parallel pattern over the given heading angle interval.
- The patterns of experimental results of lateral wave drift loads and results from three approaches overlap (API, 2005; DNV AS, 2021; Gupta, 2008), implying a close resemblance between datasets.

- A comparison of yaw moments showed that the DNV approach (DNV AS, 2021) can be preferred for the calculation of wave forces except moment. For certain, this inference is valid for models like ONR Tumblehome.
- As the wavelength increased, the longitudinal wave drift force acting on the ship increased. When the wave height and wave length are halved, the longitudinal wave drift force acting on the ship is reduced.
- When the wave height and length are reduced by half, a more stable decrease is observed in the $\bar{Y}_W$ force and the values are reduced nearly by half compared with the wavelength equal to the ship length.
- The results appeared in a sensible order in yaw moment comparison for different wave scenarios.
- Regular waves were successfully produced using the wave generator in the laboratory. However, the uncertainty increases as the wave height decreases.

One of the steady drift scenarios was simulated by using CFD tools to predict the forces and moments as a result of the prescribed heading angles of the model. The solution strategy of the flow and details about the numerical domain were presented in Chapter 4. To investigate the efficiency of CFD tools, the “ $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$ ” case was simulated by using a three-dimensional RANSE-based CFD approach adopting the realizable two-layer $k - \varepsilon$ scheme. The overset interface between the parts in the computational domain was defined. In total, 2.695E+06 cells were created to solve the numerical problem. The fifth-order VOF wave is selected to produce the regular wave of $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$. To assess the accuracy and reliability of the simulations, verification and validation procedures were followed. This section of the investigation reveals the following main findings:

- During the performing different heading angles, it is important to rotate the overset domain, local mesh densifications (i.e., overlap geometries), and ONRT model together.
- Through the elimination of wave reflections at the boundaries and facilitating the wave solution, placing the forcing zone at the boundaries resulted in a reduction in computational workload.

- Although there is more than one approach to determining the time step, it was found appropriate to use the time step obtained from the guideline by CD-Adapco (2018).
- The calculations for uncertainty demonstrated that a medium grid numerical setup can be employed for the research.
- The temporal convergence study has successfully established an optimal time step size.
- During the validation phase, the computational and experimental surge force of the steady drift test at $\psi = 0^\circ$ were found to be quite compatible.
- Computational and experimental wave loads were compared: The longitudinal and lateral wave drift forces and yaw moments were found to be compatible. As a result, it has been shown that the experimental setup can also be designed computationally.

Furthermore, a direct simulation of the turning circle in calm water was conducted under 6-DoF, $Fr = 0.2$, and $\delta = 35^\circ$ conditions, in addition to the simulations of steady drift within the waves. In total, $7.438E+06$ cells were created to solve the numerical problem. Computation is performed to the port side using maximum rudder angles without changing the advance velocity. As soon as the hull velocity is stable at the advance velocity, the rudder starts to turn, and the turning is adjusted to reach the maximum rudder angle in 1 second. The turning circle is plotted when a 360° change of heading is completed. The main conclusions of this part will be given in the following paragraphs.

The system-based ship maneuvering simulation integrating the wave model is detailed in Chapter 5. The interaction of the hull, propeller, and rudder was ensured by a 4-DoF MMG model with a roll-coupling effect. After the given basic assumptions, the derivations of the equation of motion with 6-DoF and 4-DoF were given respectively. The unknown terms in the equation of motion were forces X , Y , and moments K , N . To calculate these unknown terms, MMG's modular approach feature was used. The method to calculate the hydrodynamic forces and moments due to hull, the method to calculate the forces and moments due to propellers, the method to calculate the forces and moments due to rudders, and the method to impose the wave drift loads based on experimental results were presented separately. The non-dimensionalization procedure

used in the calculations was given. The parameters of the surface combatant ONR Tumblehome were reached by several sources and the effects due to the twin screw of the ONRT model are included in the model.

Standard turning circle maneuvering simulations were conducted in calm water and regular wave conditions at $Fr = 0.2$ which corresponds to $U_0 = 1.111$ m/s in the model scale. After performing the turning circle simulations to both port and starboard, the following main variables were achieved and expressed in the dimensional form: Advance (Ad), transfer (Tr), tactical diameter (TD), steady turning diameter (STD), steady yaw rate (SYR), and steady turning speed (STS). The key discoveries in the calm water condition are as follows.

- Straight-ahead simulations were successfully performed to determine the k_0, k_1 , and k_2 coefficients. The initial velocity remained constant for some time after the forward motion.
- Turning circle simulations with $\pm 35^\circ$ rudder angles under calm water conditions at $Fr = 0.2$ were compared with existing experimental, numerical, and S-B simulations. The comparative investigations show that present S-B results are in good agreement with experimental results except for the advance parameter. This remarkable consistency between S-B simulations and experiments was always valid at different heading angles of $0^\circ, \pm 90^\circ$, and 180° , and turning directions ($\delta = \pm 35^\circ$).
- The inconsistency in advance parameters both in calm water and in wave conditions is remarkable. However, it is not possible to make a definite judgment about S-B simulations from this relation. Because in the experimental trajectory, it is not clear when the rudder begins to turn, nor how long before the rudder begins to turn is reflected in the trajectory figure. Existing S-B and CFD turning trajectories captured the motion state of the hull one second before the rudder started turning.
- The best agreement with the existing S-B simulations in calm water was obtained with the results of the S-B simulations in the literature (Guo et al., 2018). The geometric and maneuver-related parameters of the ONRT model were obtained from this source, and as a result of the comparison, it can be thought that the current model was correctly established.

- When comparing the values of the parameter "steady yaw rate" of the calm water cases among the different solution methods, a significant rise can be observed in the S-B simulations. A comment can be provided to clarify this distinction: The equations of motion utilized exclusively in the S-B simulations are limited to 4 degrees of freedom. While further data analysis may be required for a definitive conclusion, this observation could suggest that the increase in *SYR* value is linked to a reduction in the degree of freedom of motion.
- In the case of calm water, it was seen that the maneuver parameters values were very close to each other in the comparison of the portside and starboard side turning directions.
- The direct CFD simulation outcome did not align with the anticipated results. The simulation results have presented an opportunity for further analysis and refinement.
- Three different rudder angles were examined to observe the effect of rudder angle on maneuver parameters in portside and starboard side turnings. A larger rudder angle resulted in lower advance and transfer, a smaller turning diameter, a turning trajectory, velocity, and roll angle. On the contrary, a larger rudder angle caused a higher steady yaw rate and a sharper turn

It was assessed that the use of system-based simulation to address the ONR Tumblehome's turning circle problem, specifically focusing on $\pm 35^\circ$ rudder angles in regular wave conditions at $Fr = 0.2$. Polynomials are fitted to the second-order mean wave loads obtained from the experimentally measured forces and moments in the three-wave scenario. These obtained equations are integrated into the S-B simulations as a wave model. The incident wave angles, denoted as χ , encompass 0° (representing following seas), $\pm 90^\circ$ (indicating quartering seas), and 180° (corresponding to head seas), thereby encompassing a range of wave conditions. The central results derived from S-B turning circle simulations in regular waves can be outlined as follows.

- All S-B simulation results were compared with the experimental study by Elshiekh (2014). Parameters other than the advance parameter, as in calm water cases, are consistent with the experimental results.

- The modification due to the differences of experimental and S-B coordinate systems, was important to apply. If this modification is not made, the forces and moments are not adapted correctly and the drift of the model cannot be predicted correctly.
- In the head wave cases, it is seen that the turning trajectory has less drift in the $-x$ direction. It may be caused by the magnitude of the mean wave sway force. Because when the Y_w force was increased randomly in simulations, it was seen that this amount of drift increased and became more compatible with the experimental trajectory.
- The leeward and windward position of the rudders during the turning maneuver may be important for the final trajectory.
- The relation between the wave parameters and turning circle parameters was laid out in detail. The bar charts clearly illustrate the maneuvering parameters for each rudder angle and wave incident angle in terms of wave characteristics.

6.3 Recommendations

Based on the findings and analysis presented in this study, several key recommendations emerge for further research and practical applications:

- Wave spectra variation: Extend the simulation capabilities to account for a broader range of wave spectra and wave conditions, including irregular waves, in order to provide a more comprehensive assessment of the ship's maneuvering performance.
- Incorporate forward speed: Modify the simulation software to include the influence of forward speed on the vessel's maneuvering behavior. This can be achieved by integrating the relevant hydrodynamic equations and parameters that consider the vessel's speed through the water.
- Rudder coefficients: Flow-straightening coefficients and correlation factors for the flow-straightening effect due to yaw rate take different values depending on the position of the rudders. In the S-B simulations, the difference is taken into account according to the position, but the same values were used for the

right and left rudders. These coefficients should vary with changes in the rudder.

- Experimental capability: Enable the experimental setup to 4-DoF or 6-DoF which enhances the vessel's response.
- Direct CFD simulations: Reconfigure solver settings carefully, including time step size, convergence criteria, and numerical schemes.

These recommendations aim to advance the capabilities of maneuvering simulation for the ONR Tumblehome, ensuring that it serves as a powerful tool for research, training, and design optimization while accounting for the complexities of wave-ship interactions.





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APPENDICES

APPENDIX A: Empirical/analytical results of wave drift loads for " $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ " and " $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ "

APPENDIX B: Matlab code for Fast Fourier Transform (FFT) and Savitzky-Golay smoothing filter

APPENDIX C: Time histories and FFT analyses of the experimental study

APPENDIX D: Experimental results of wave drift loads for " $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ " and " $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ "

APPENDIX E: Time histories of wave signals

APPENDIX F: Time histories and FFT analyses of the computational study

APPENDIX G: Turning trajectories and time histories in calm water

APPENDIX H: Turning trajectories and time histories in calm water for varying rudder angles

APPENDIX I: Turning trajectories and time histories in waves



APPENDIX A Empirical/analytical results of wave drift loads for " $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ " and " $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ "

Table A.1 : Comparison of longitudinal wave drift forces, $\bar{X}_w$, obtained by empirical/analytical approximations at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$.

Heading angle, ψ (deg)	Advanced Marine Enterprises Inc. (1996)	API (2005)	Gupta (2008)	Bergdahl & Kofoed (2015)	DNV AS (2021)
0	-0.089	-0.590	-0.552	-0.934	-0.063
15	-0.086	-0.570	-0.533	-0.902	-0.065
30	-0.077	-0.511	-0.478	-0.809	-0.065
45	-0.063	-0.417	-0.390	-0.660	-0.063
60	-0.044	-0.295	-0.276	-0.467	-0.055
75	-0.023	-0.153	-0.143	-0.242	-0.041
90	0.000	0.000	0.000	0.000	-0.017
105	0.023	0.153	0.143	0.242	0.015
120	0.044	0.295	0.276	0.467	0.048
135	0.063	0.417	0.390	0.660	0.076
150	0.077	0.511	0.478	0.809	0.098
165	0.086	0.570	0.533	0.902	0.116
180	0.089	0.590	0.552	0.934	0.131

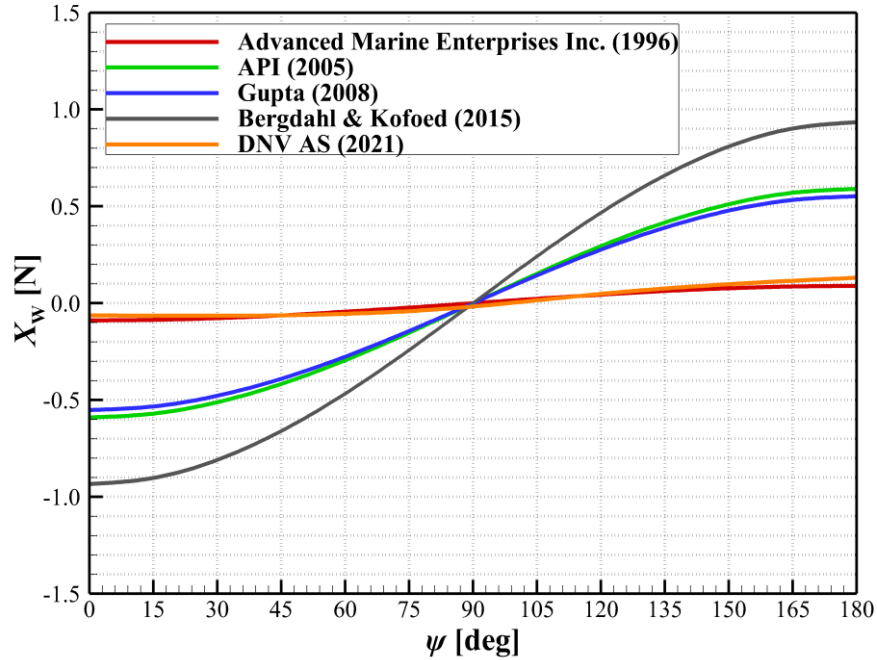


Figure A.1 : Comparison of empirical/analytical results of mean longitudinal force with respect to heading angle ($H/\lambda = 0.020, \lambda/L_{WL} = 0.50$).

Table A.2 : Comparison of lateral wave drift forces, $\bar{Y}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$.

Heading angle, ψ (deg)	Advanced Marine Enterprises Inc. (1996)	API (2005)	Gupta (2008)	Bergdahl & Kofoed (2015)	DNV AS (2021)
0	0.000	0.000	0.000	0.000	0.000
15	0.298	0.153	0.143	0.242	0.221
30	0.575	0.295	0.276	0.467	0.427
45	0.814	0.417	0.390	0.660	0.604
60	0.997	0.511	0.478	0.809	0.740
75	1.112	0.570	0.533	0.902	0.825
90	1.151	0.590	0.552	0.934	0.854
105	1.112	0.570	0.533	0.902	0.825
120	0.997	0.511	0.478	0.809	0.740
135	0.814	0.417	0.390	0.660	0.604
150	0.575	0.295	0.276	0.467	0.427
165	0.298	0.153	0.143	0.242	0.221
180	0.000	0.000	0.000	0.000	0.000

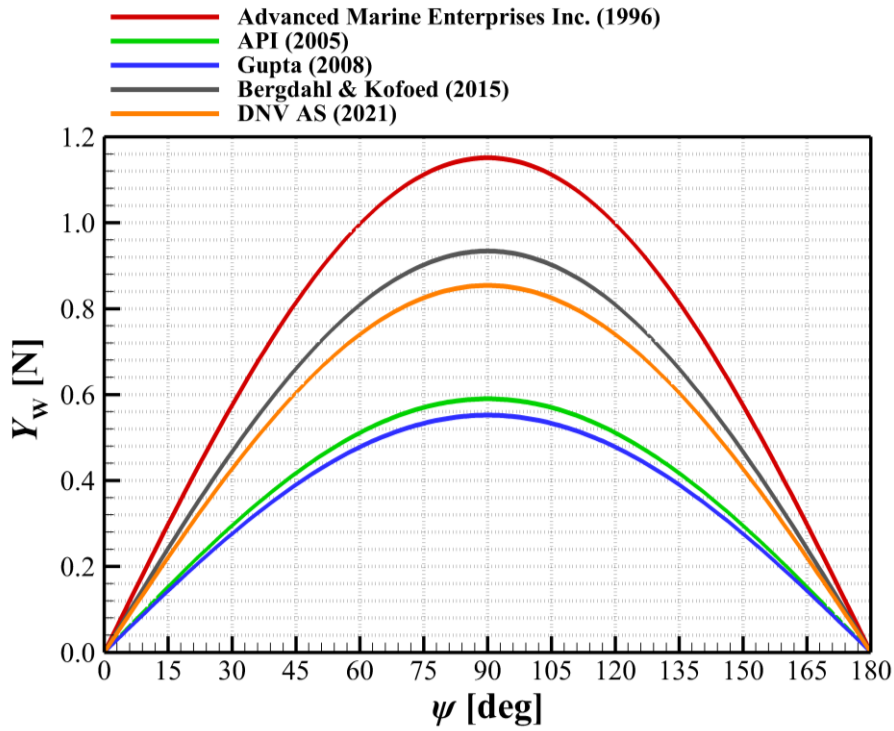


Figure A.2 : Comparison of empirical/analytical results of mean lateral force with respect to heading angle ($H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$).

Table A.3 : Comparison of yaw moments, $\bar{N}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$.

Heading angle, ψ (deg)	Advanced Marine Enterprises Inc. (1996)	API (2005)	DNV AS (2021)
0	0.000	0.000	0.000
15	0.076	0.038	0.034
30	0.125	0.128	0.049
45	0.131	0.209	0.047
60	0.086	0.221	0.030
75	-0.001	0.143	0.003
90	-0.109	0.000	-0.029
105	-0.209	-0.143	-0.058
120	-0.274	-0.221	-0.080
135	-0.284	-0.209	-0.088
150	-0.234	-0.128	-0.078
165	-0.132	-0.038	-0.048
180	0.000	0.000	0.000

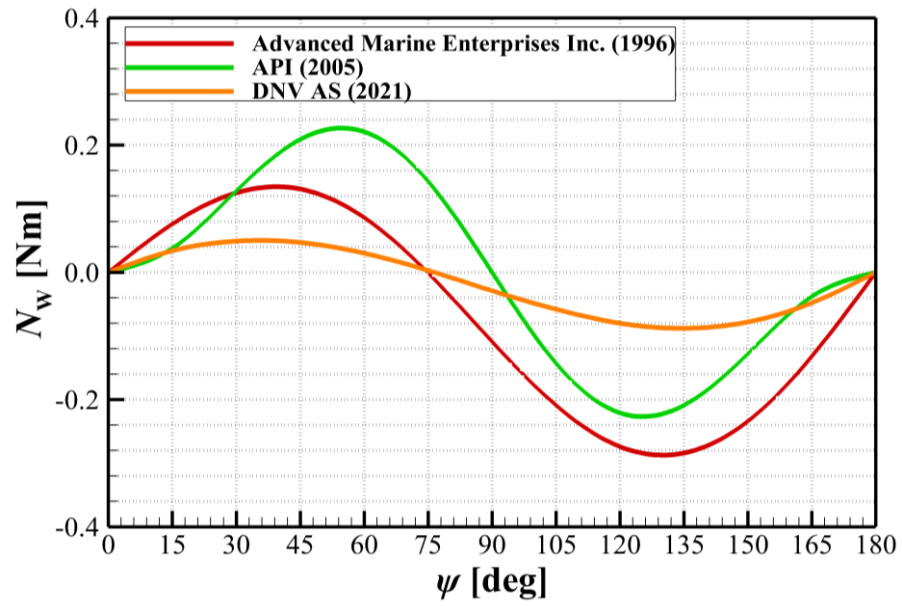


Figure A.3 : Comparison of empirical/analytical results of mean yaw moment with respect to heading angle ($H/\lambda = 0.020, \lambda/L_{WL} = 0.50$).

Table A.4 : Comparison of longitudinal wave drift forces, $\bar{X}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$.

Heading angle, ψ (deg)	Advanced Marine Enterprises Inc. (1996)	API (2005)	Gupta (2008)	Bergdahl & Kofoed (2015)	DNV AS (2021)
0	-0.348	-1.858	-2.187	-6.064	-0.412
15	-0.336	-1.795	-2.112	-5.857	-0.425
30	-0.301	-1.609	-1.894	-5.251	-0.425
45	-0.246	-1.314	-1.546	-4.288	-0.407
60	-0.174	-0.929	-1.093	-3.032	-0.360
75	-0.090	-0.481	-0.566	-1.569	-0.266
90	0.000	0.000	0.000	0.000	-0.110
105	0.090	0.481	0.566	1.569	0.099
120	0.174	0.929	1.093	3.032	0.314
135	0.246	1.314	1.546	4.288	0.495
150	0.301	1.609	1.894	5.251	0.639
165	0.336	1.795	2.112	5.857	0.755
180	0.348	1.858	2.187	6.064	0.852

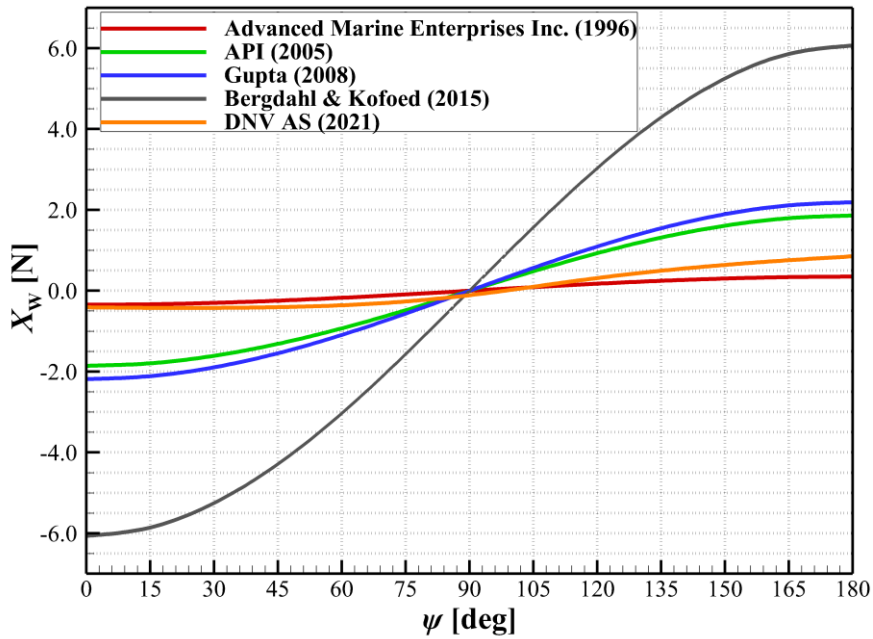


Figure A.4 : Comparison of empirical/analytical results of mean longitudinal force with respect to heading angle ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).

Table A.5 : Comparison of lateral wave drift forces, $\bar{Y}_W$, obtained by empirical/analytical approximations at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$.

Heading angle, ψ (deg)	Advanced Marine Enterprises Inc. (1996)	API (2005)	Gupta (2008)	Bergdahl & Kofoed (2015)	DNV AS (2021)
0	0.000	0.000	0.000	0.000	0.000
15	0.820	0.481	0.566	1.569	0.642
30	1.583	0.929	1.093	3.032	1.239
45	2.239	1.314	1.546	4.288	1.753
60	2.743	1.609	1.894	5.251	2.147
75	3.059	1.795	2.112	5.857	2.394
90	3.167	1.858	2.187	6.064	2.479
105	3.059	1.795	2.112	5.857	2.394
120	2.743	1.609	1.894	5.251	2.147
135	2.239	1.314	1.546	4.288	1.753
150	1.583	0.929	1.093	3.032	1.239
165	0.820	0.481	0.566	1.569	0.642
180	0.000	0.000	0.000	0.000	0.000

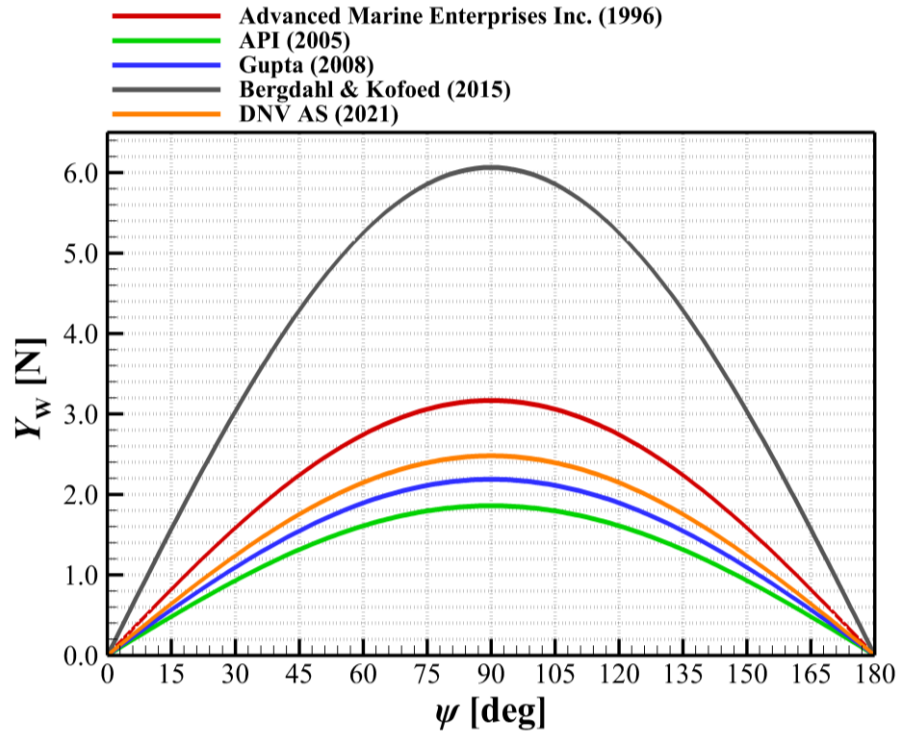


Figure A.5 : Comparison of empirical/analytical results of mean lateral force with respect to heading angle ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).

Table A.6 : Comparison of yaw moments, $\bar{N}_w$, obtained by empirical/analytical approximations at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$.

Heading angle, ψ (deg)	Advanced Marine Enterprises Inc. (1996)	API (2005)	DNV AS (2021)
0	0.000	0.000	0.000
15	0.208	0.120	0.097
30	0.345	0.402	0.142
45	0.360	0.657	0.136
60	0.236	0.697	0.087
75	-0.003	0.449	0.008
90	-0.299	0.000	-0.083
105	-0.574	-0.449	-0.169
120	-0.754	-0.697	-0.231
135	-0.783	-0.657	-0.254
150	-0.644	-0.402	-0.225
165	-0.363	-0.120	-0.141
180	0.000	0.000	0.000

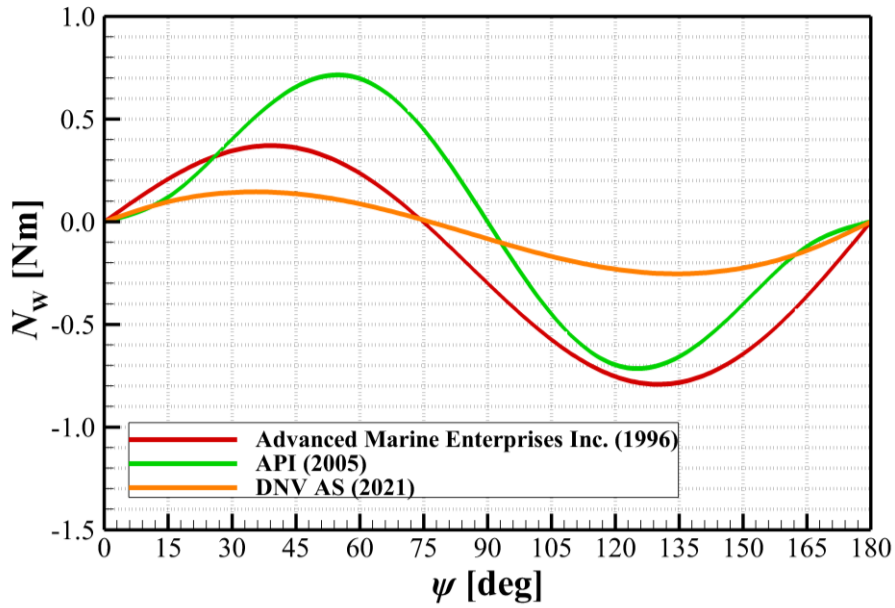


Figure A.6 : Comparison of empirical/analytical results of mean yaw moment with respect to heading angle ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).

APPENDIX B Matlab code for Fast Fourier Transform (FFT) and Savitzky-Golay smoothing filter

```
clear; clc; close all;

X=dlmread('0X.dat'); % The sample signal

Fs = 1/0.005; % Sampling frequency
T = 1/Fs; % Sampling period
L = length(X); % Length of signal
t = (0:L-1)*T; % Time vector

% Identify an input length that is the next power of 2 from the
original signal length
n = 2^nextpow2(L);

% Perform FFT analysis
Y = fft(X,n);

% Compute the frequency axis
f = Fs*(0:(n/2))/n; % Frequency axis

% Compute the two-sided spectrum P2
P2 = abs(Y)/(L);

% Compute the single-sided spectrum P1
P1 = P2(1:n/2+1);
P1(2:end-1) = 2*P1(2:end-1);

% Apply Savitzky-Golay filtering
a = 30;
P0 = P1(1:a);

order = 3; % Polynomial order
framelen = 11; % Frame length

sgf = sgolayfilt(P0,order,framelen);

% Plot the magnitude spectrum
figure(1)
plot(f,P1)
title("Single-Sided Amplitude Spectrum of S(t)")
xlabel("f (Hz)")
ylabel("|P1(f)|")
set(gca, 'Xlim', [0,1.1])

% Plot the original and filtered signals
figure(2)
plot(P0, ':')
hold on
plot(sgf, '-.')
legend('signal', 'sgolay')
```



APPENDIX C Time histories and FFT analyses of the experimental study

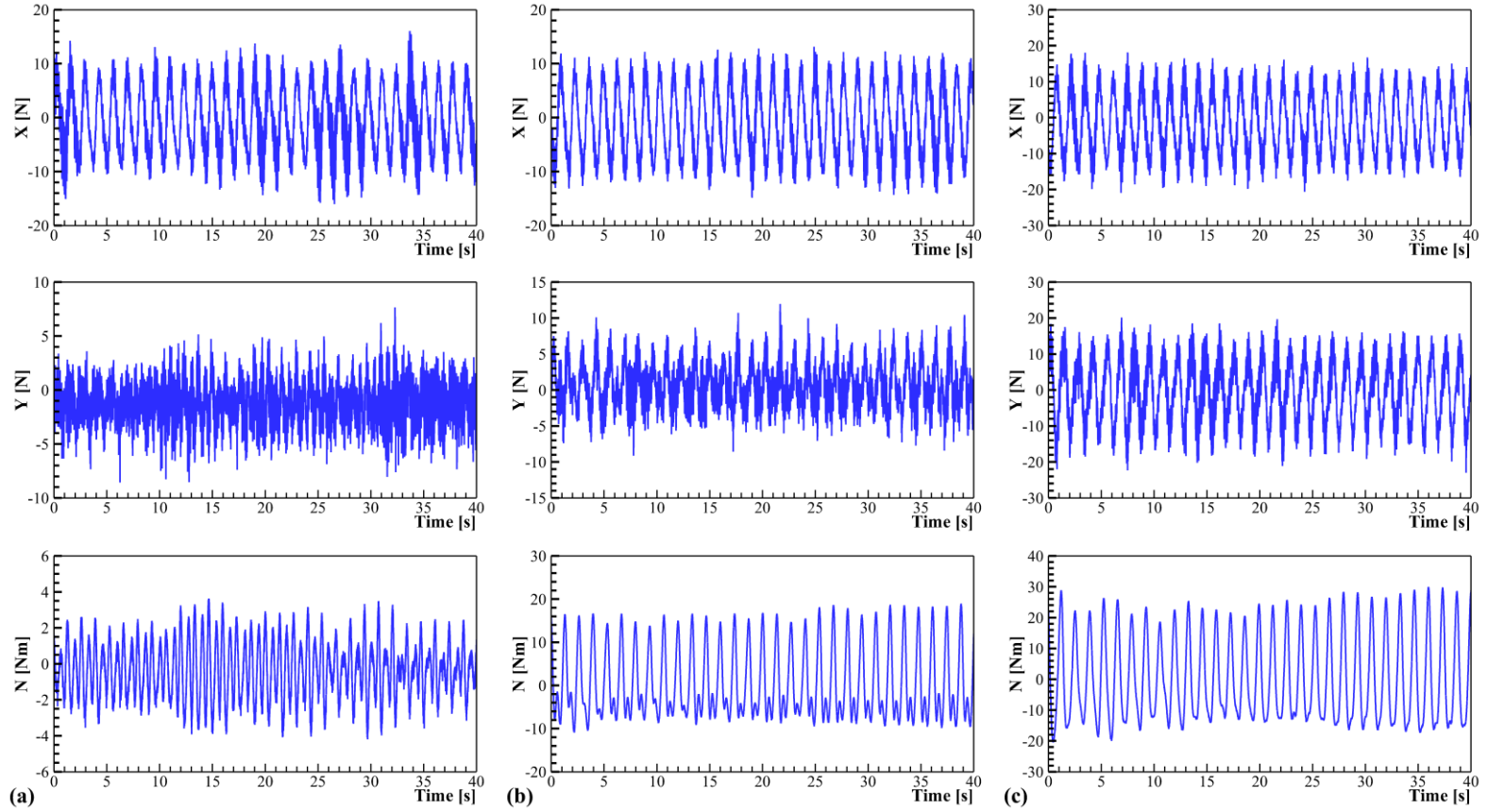


Figure C.1 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$.

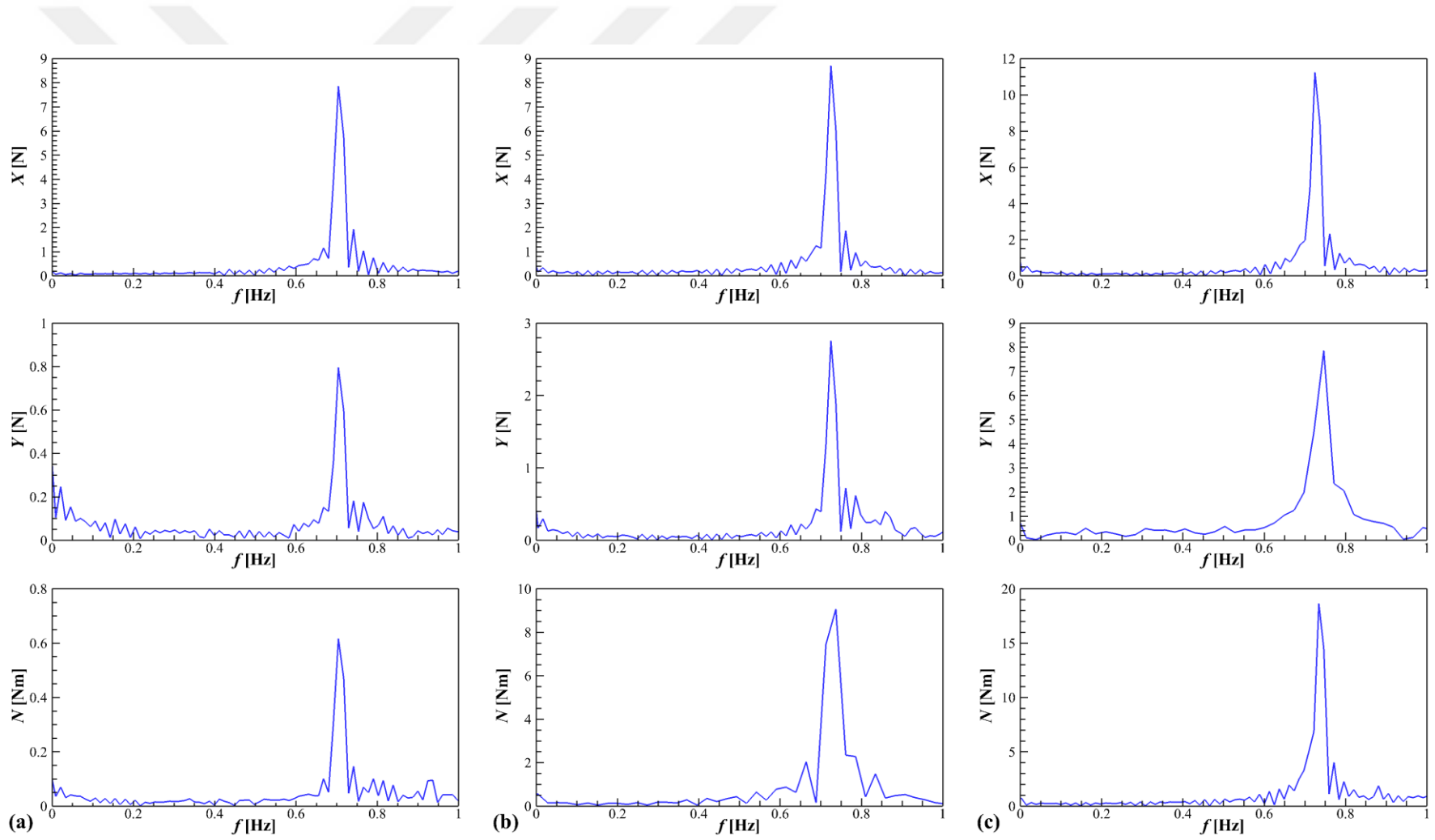


Figure C.2 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$.

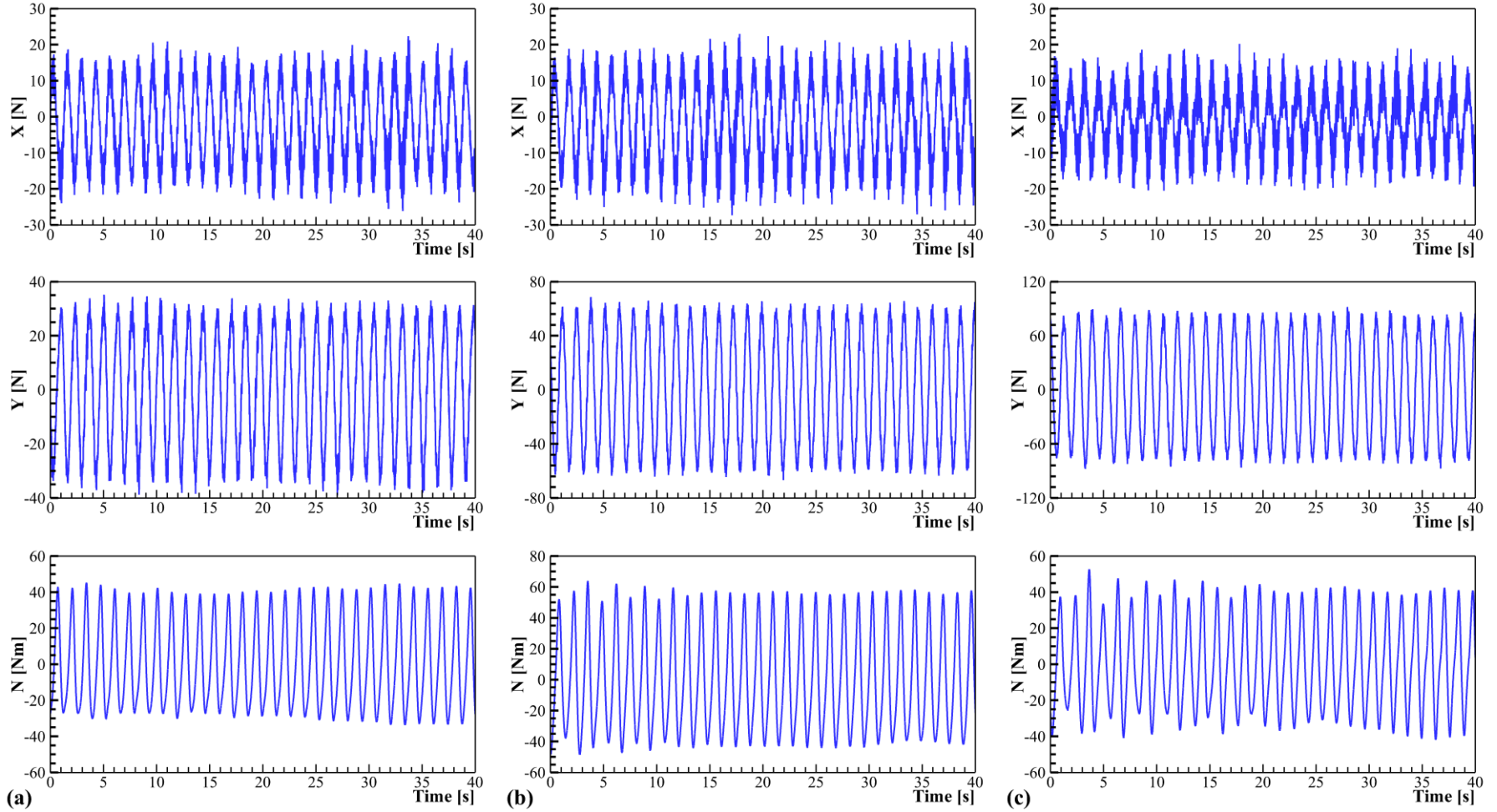


Figure C.3 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$.

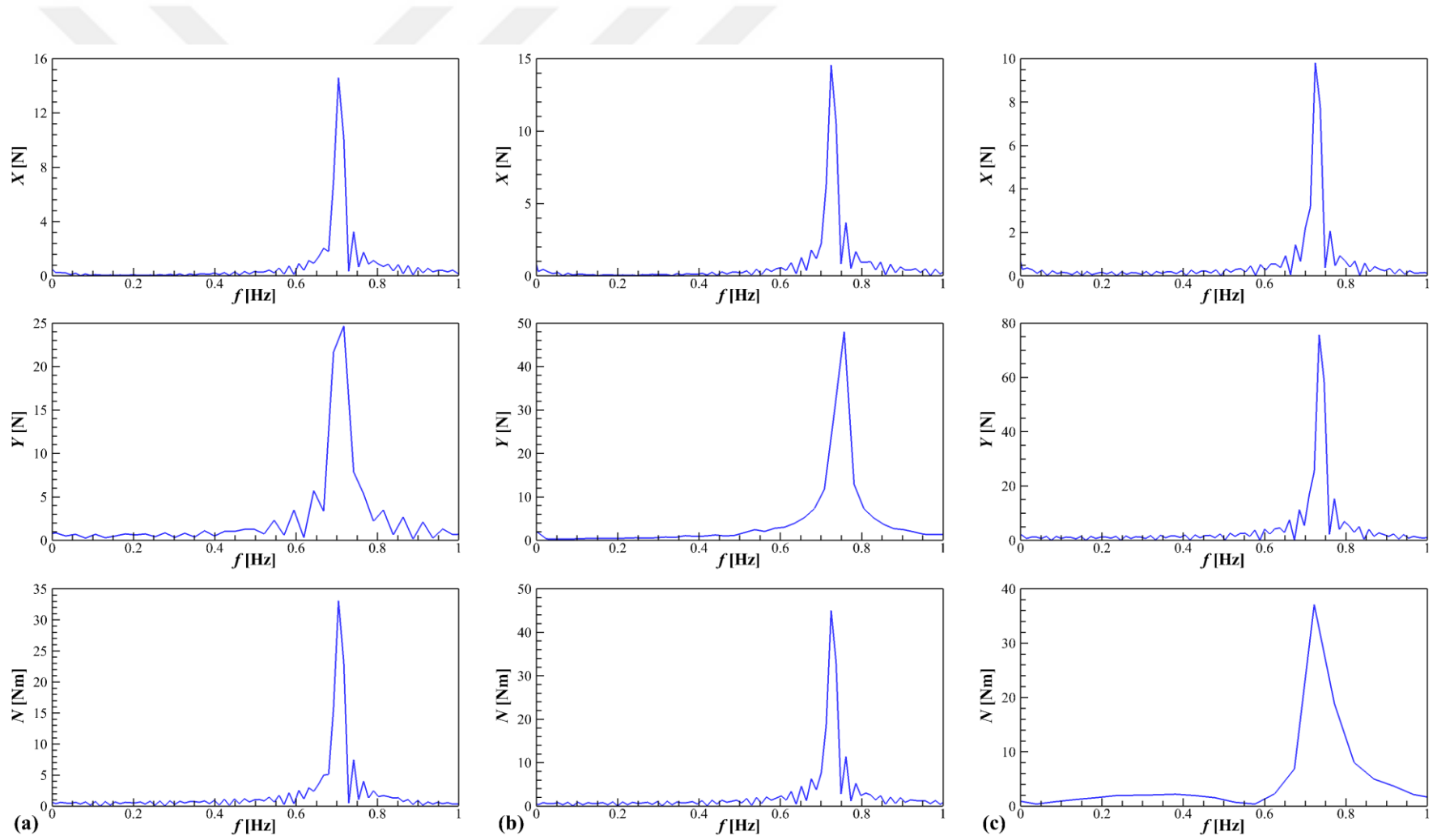


Figure C.4 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$.

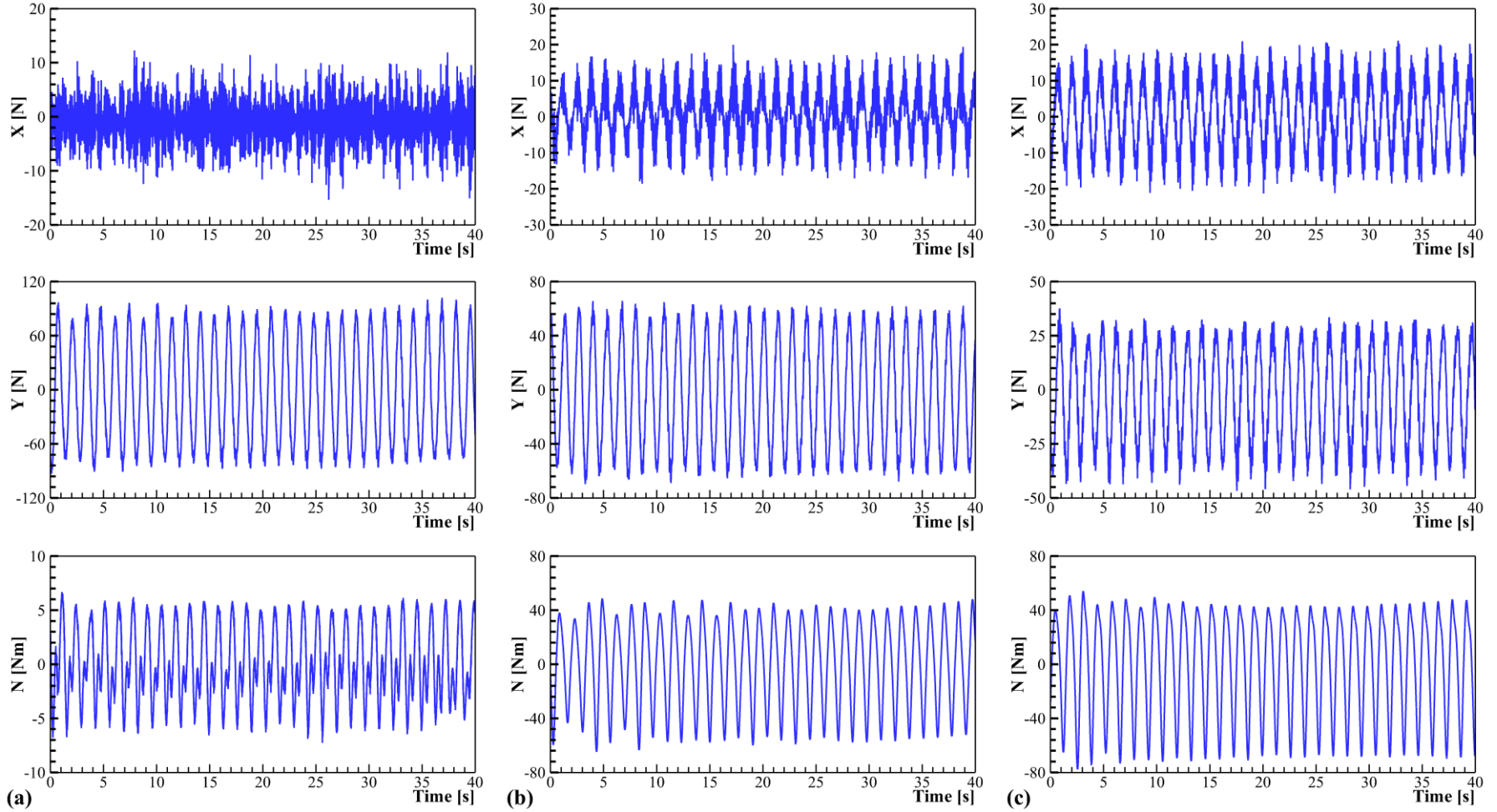


Figure C.5 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.

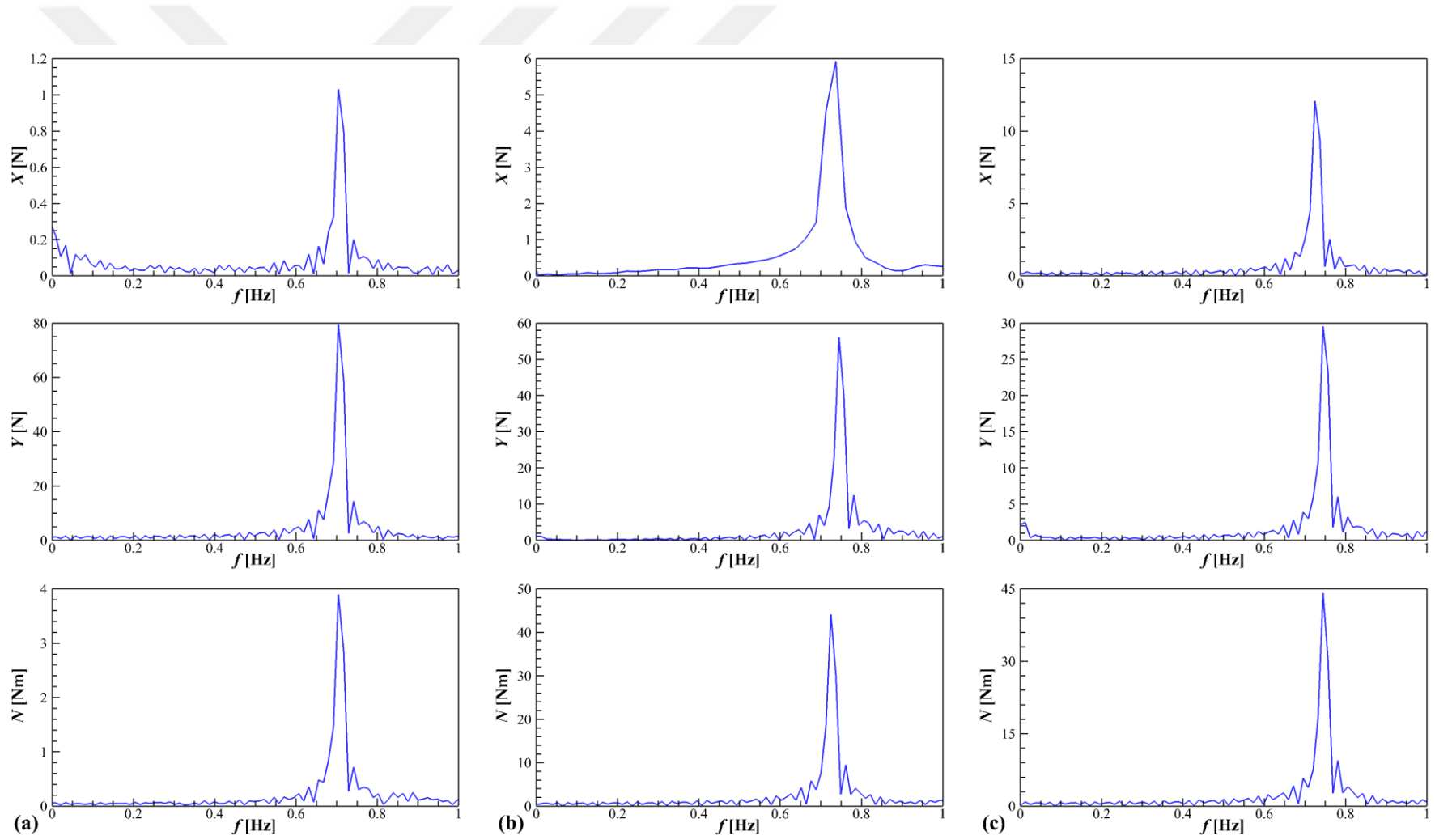


Figure C.6 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.

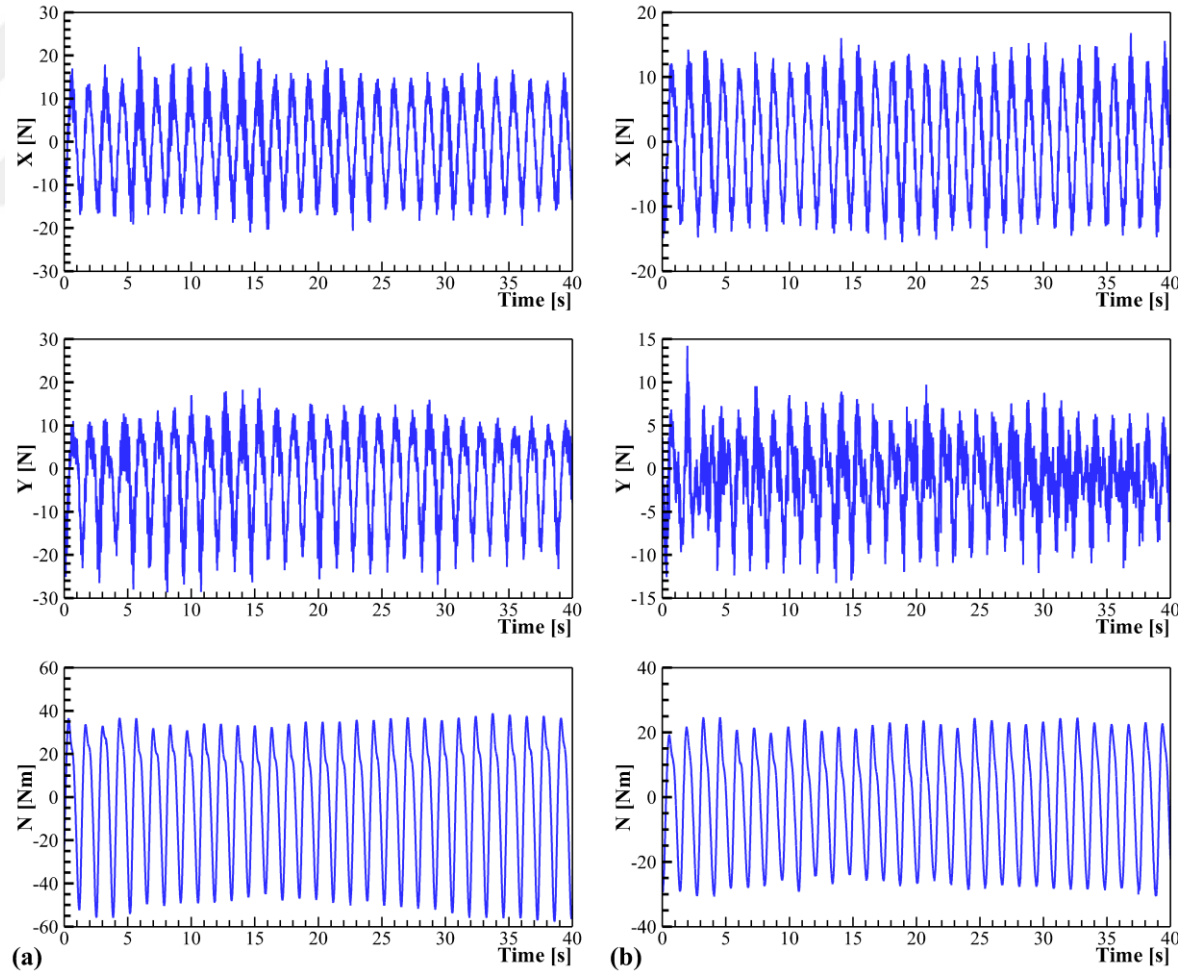


Figure C.7 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$.

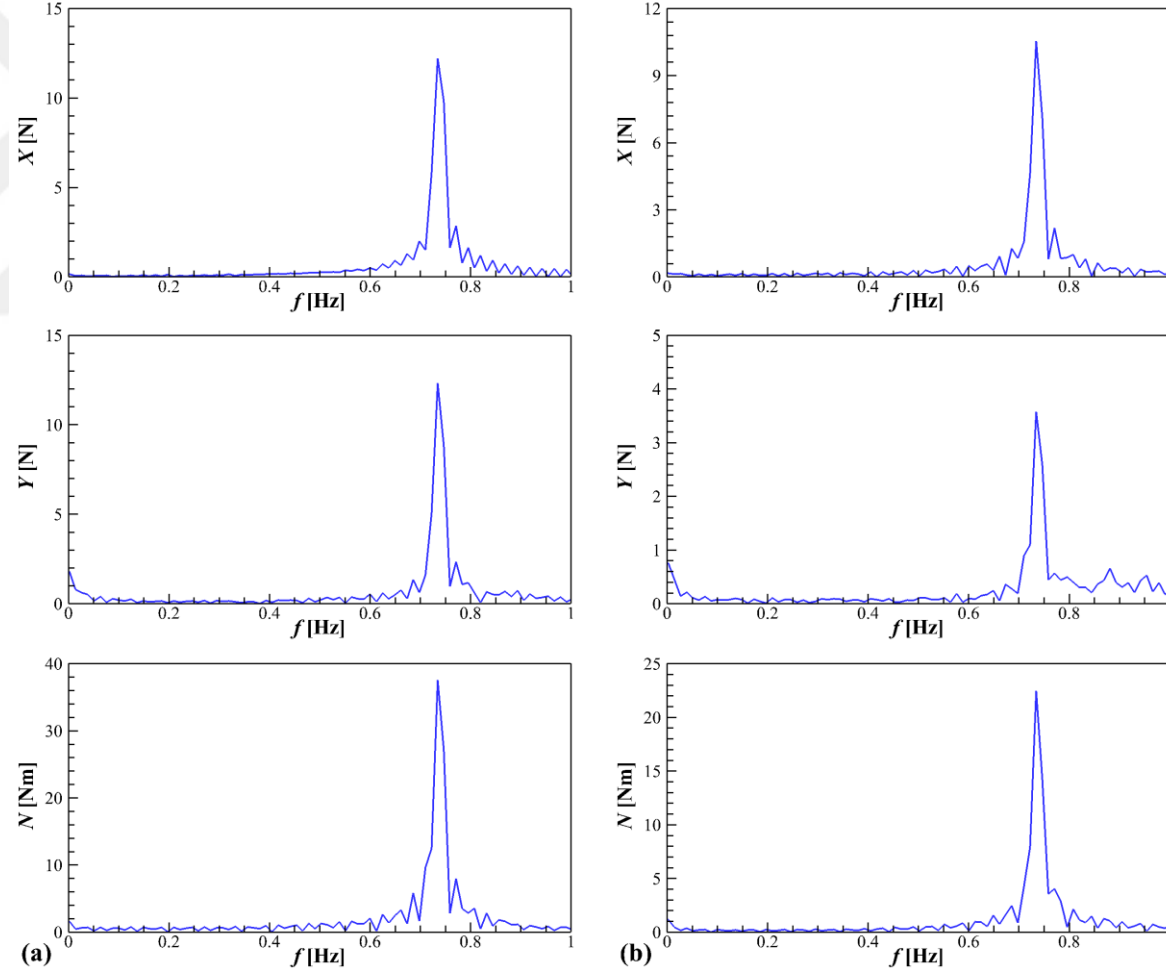


Figure C.8 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$.

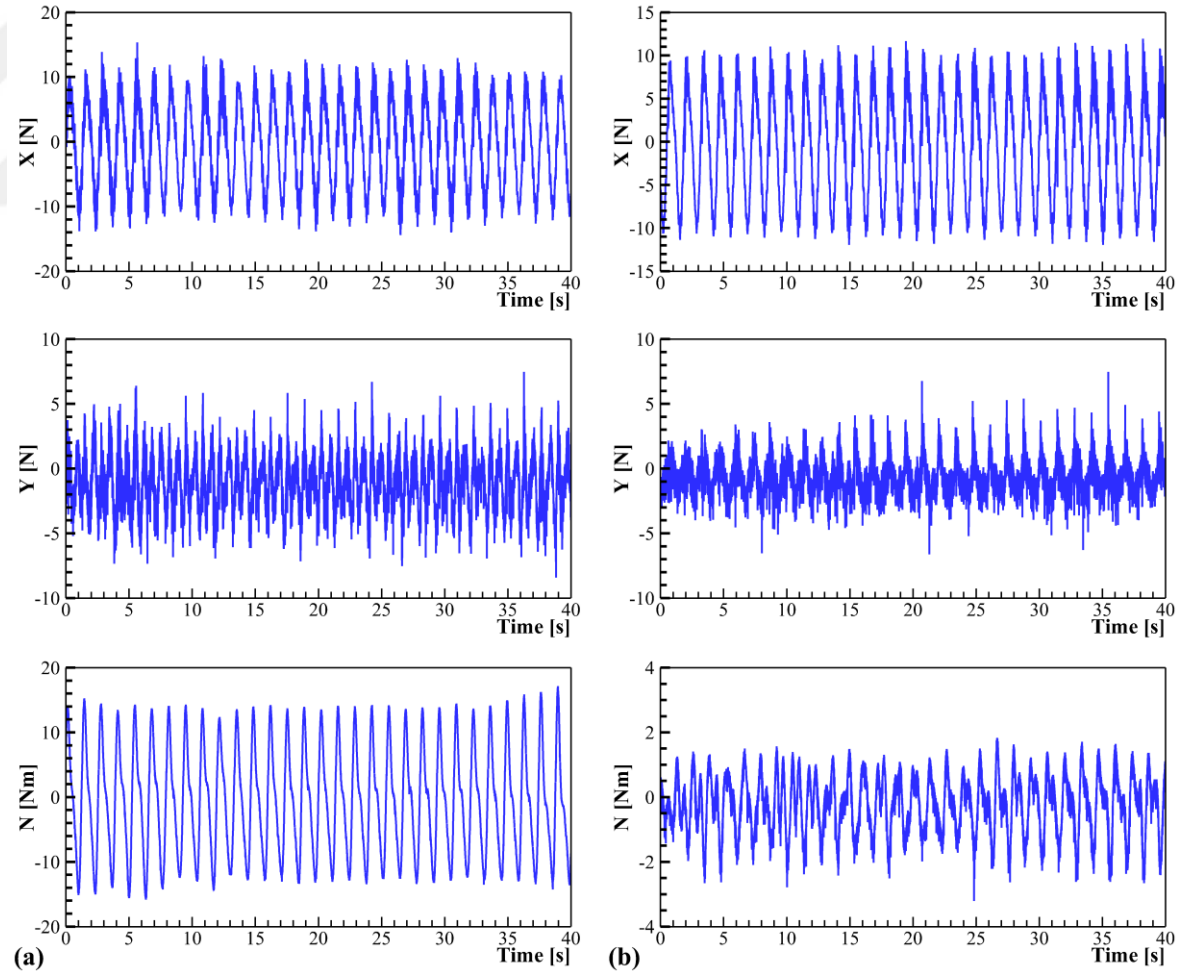


Figure C.9 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$.

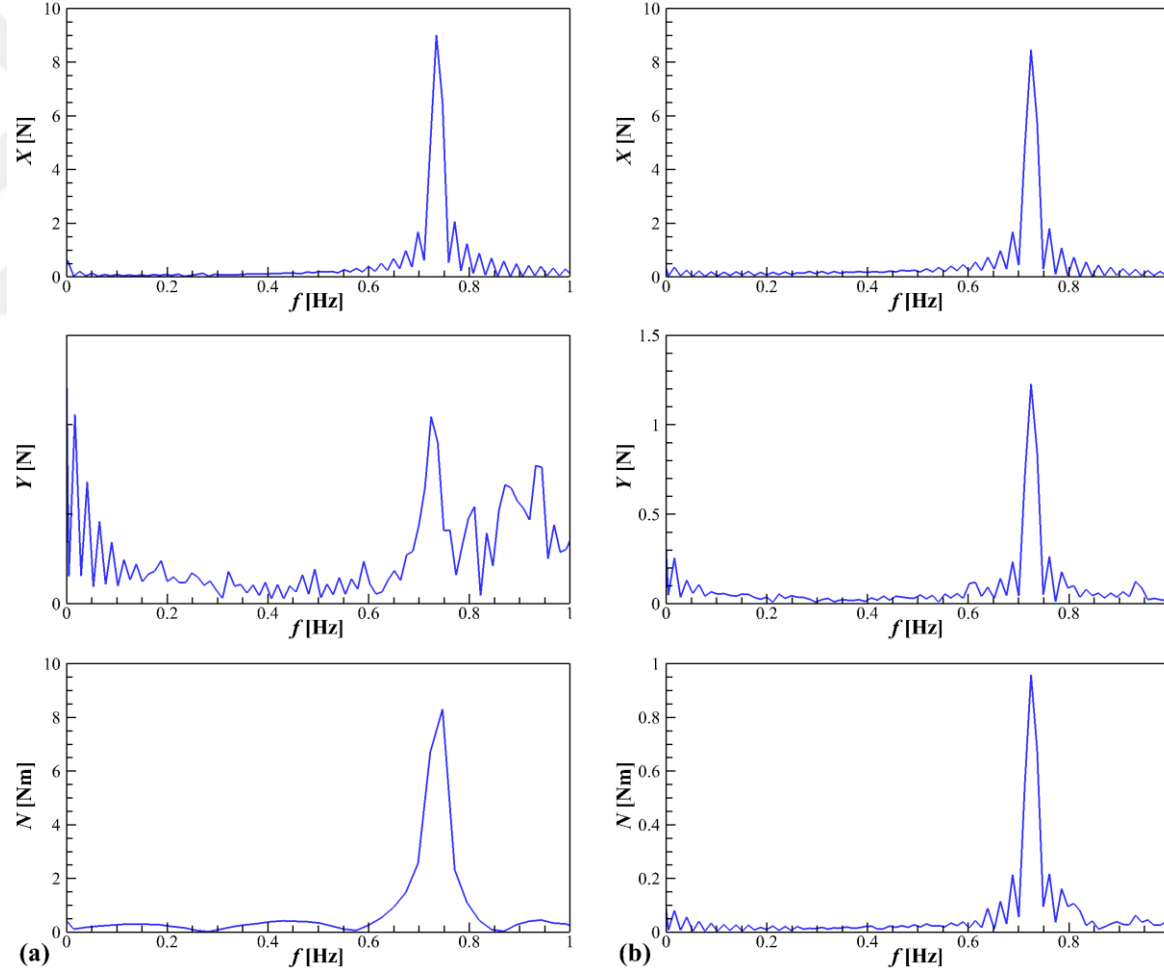


Figure C.10 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$.

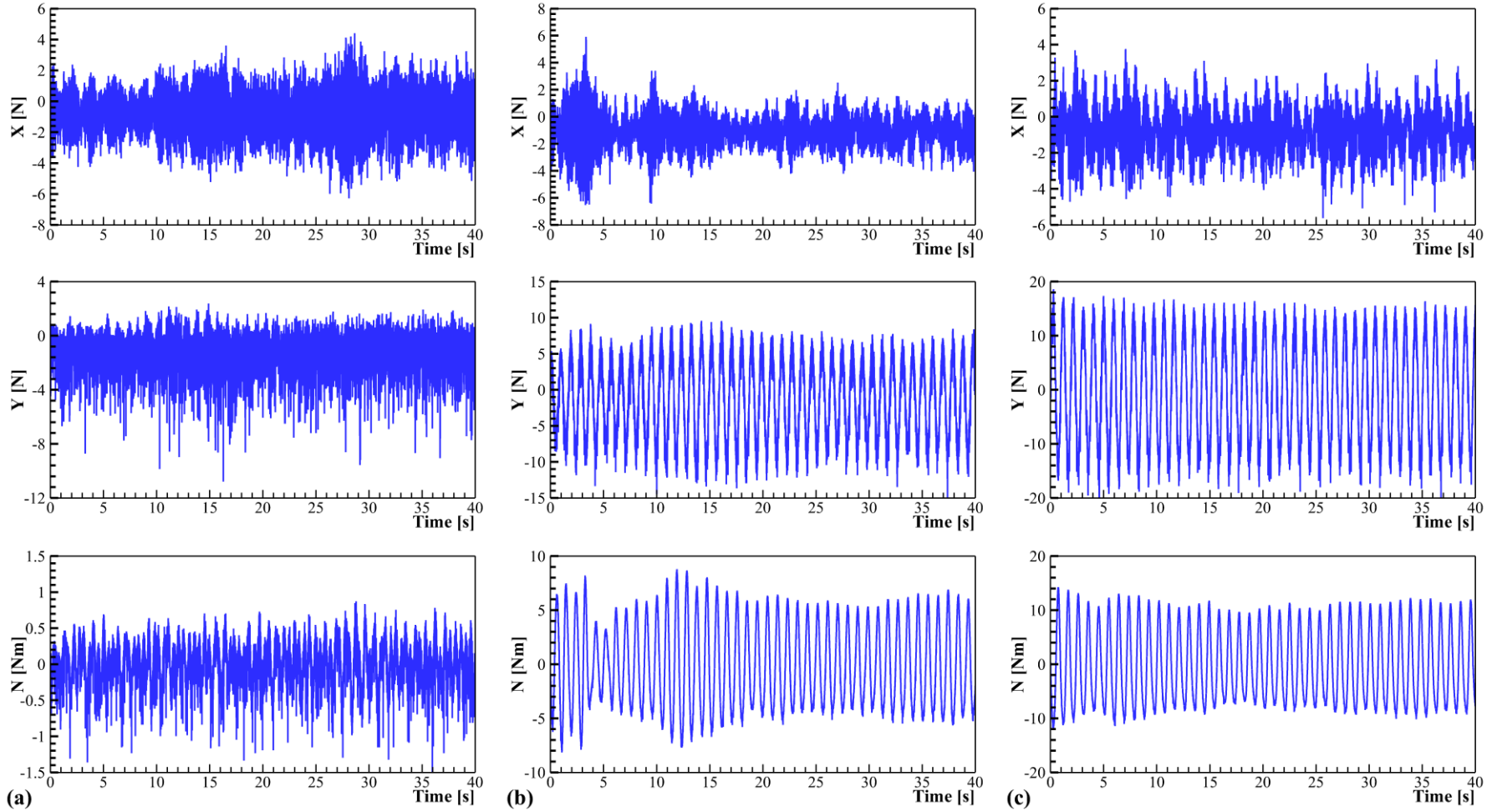


Figure C.11 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$.

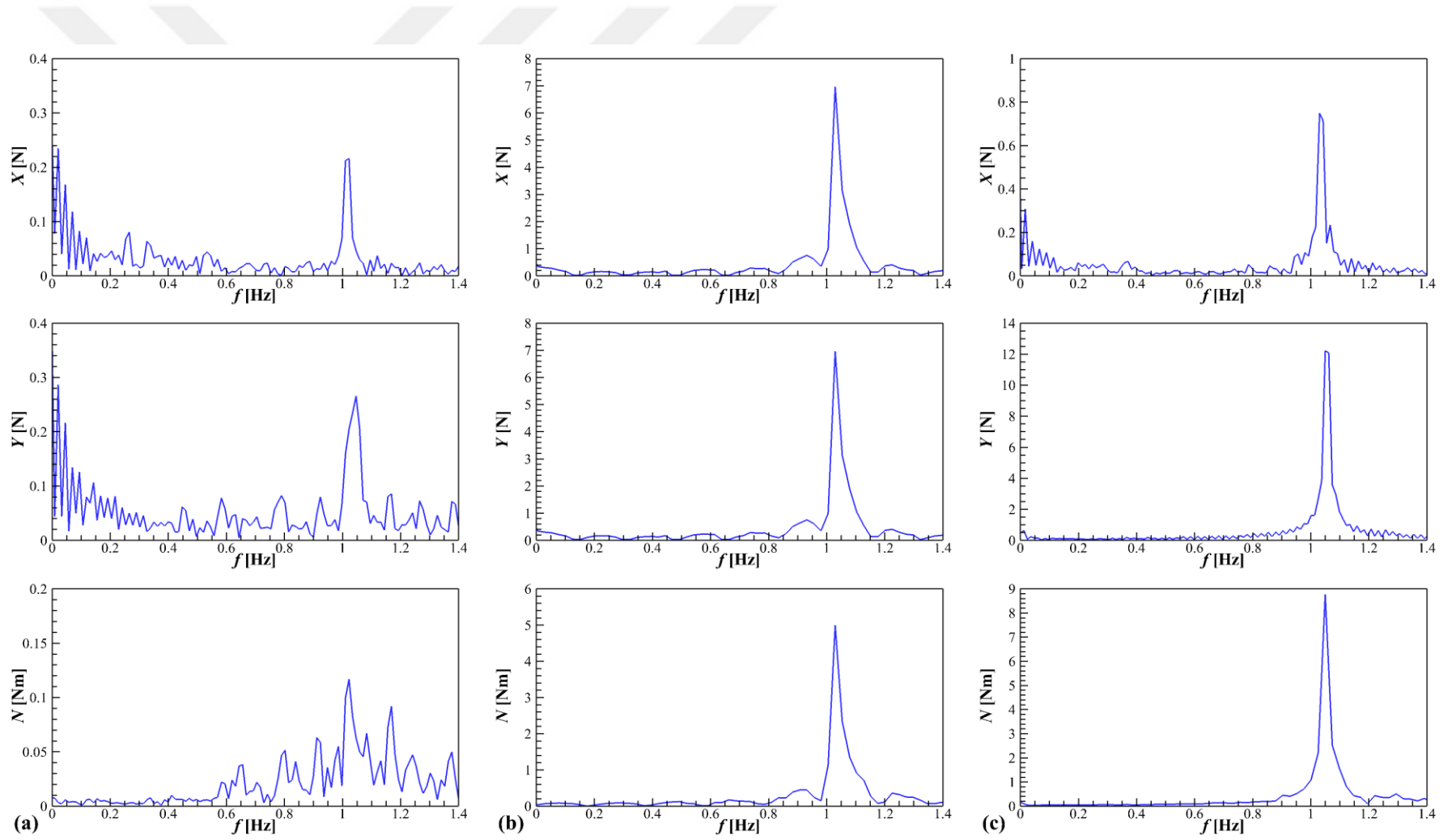


Figure C.12 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$.

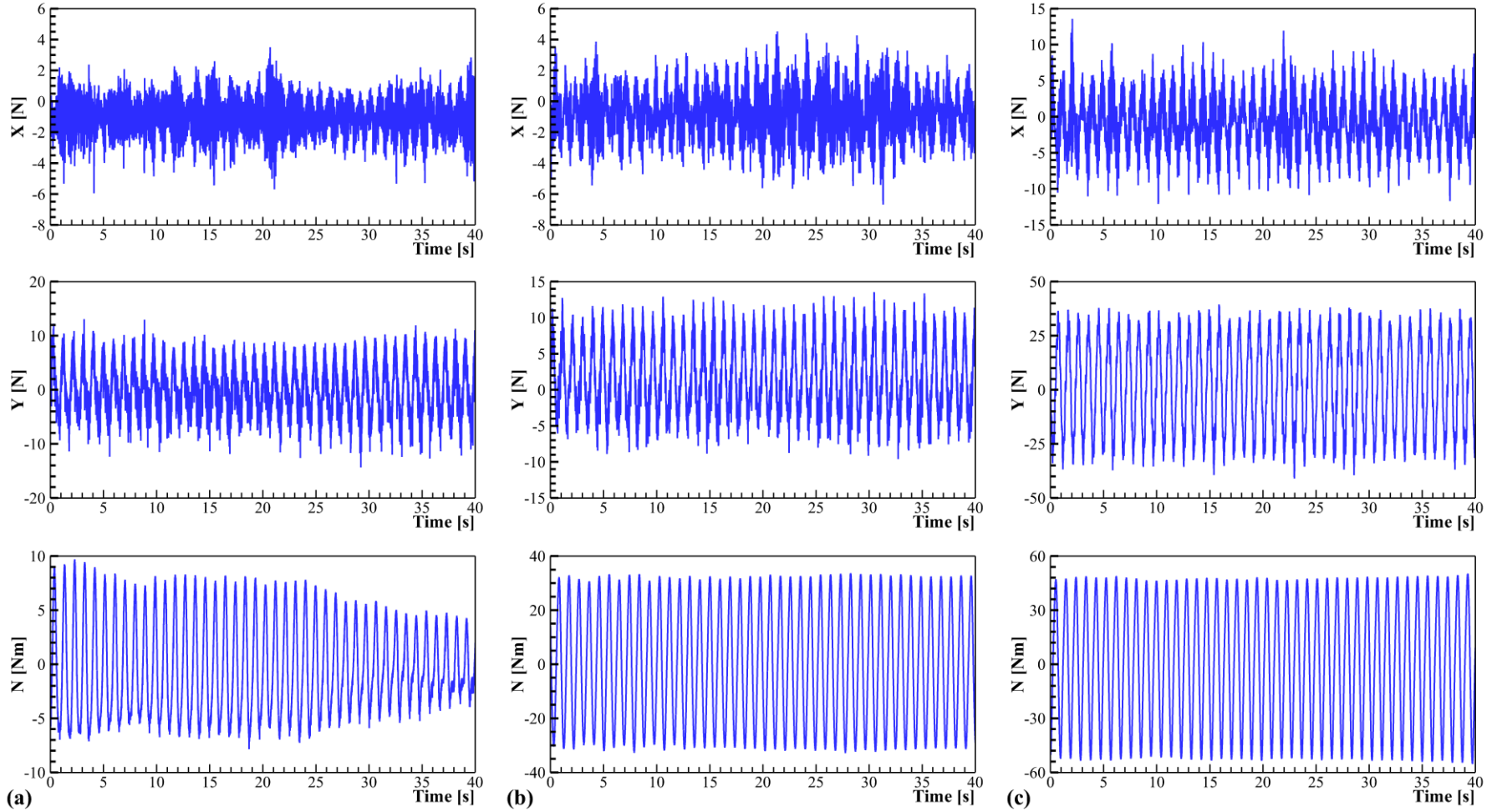


Figure C.13 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$.

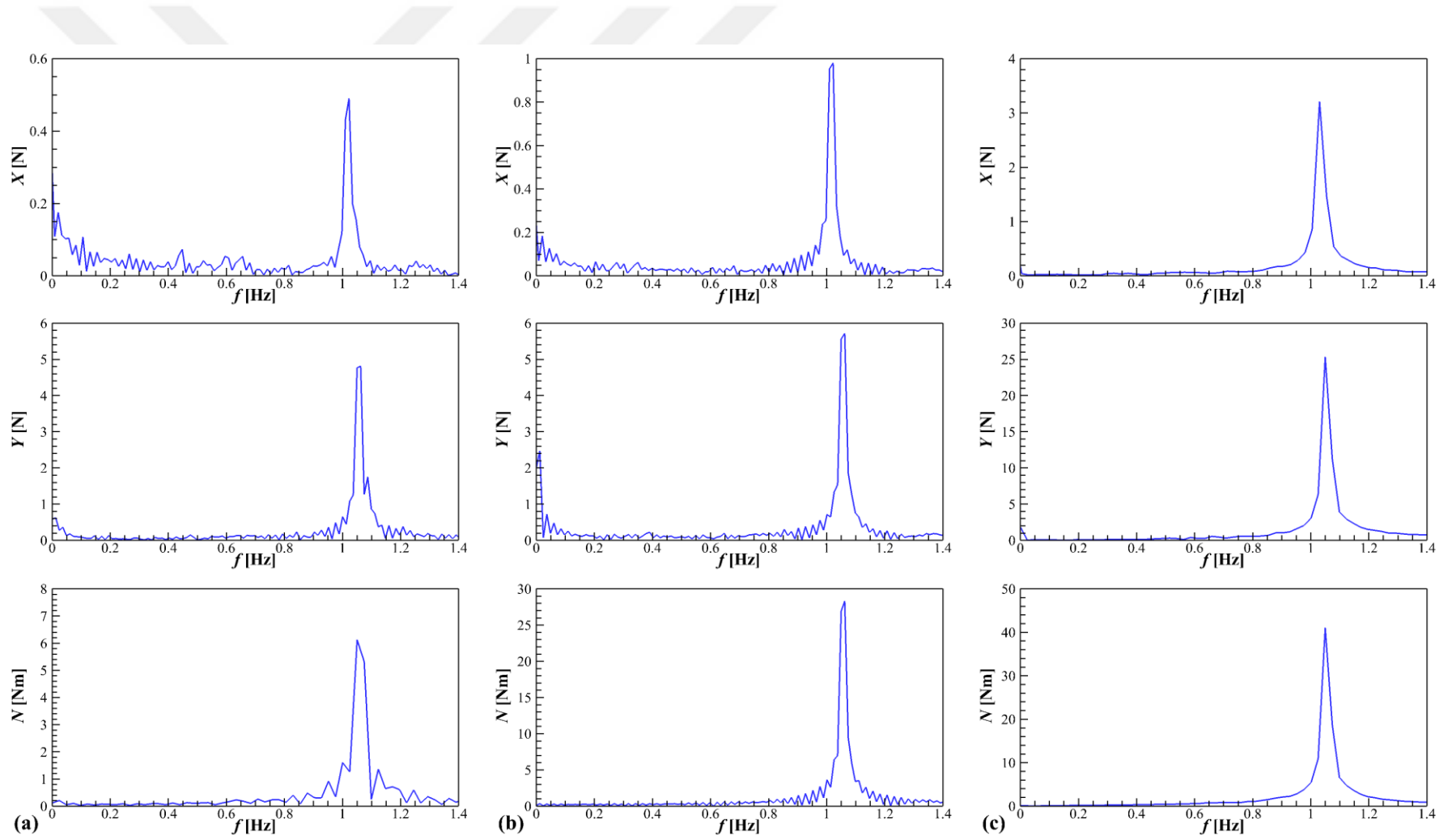


Figure C.14 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$.

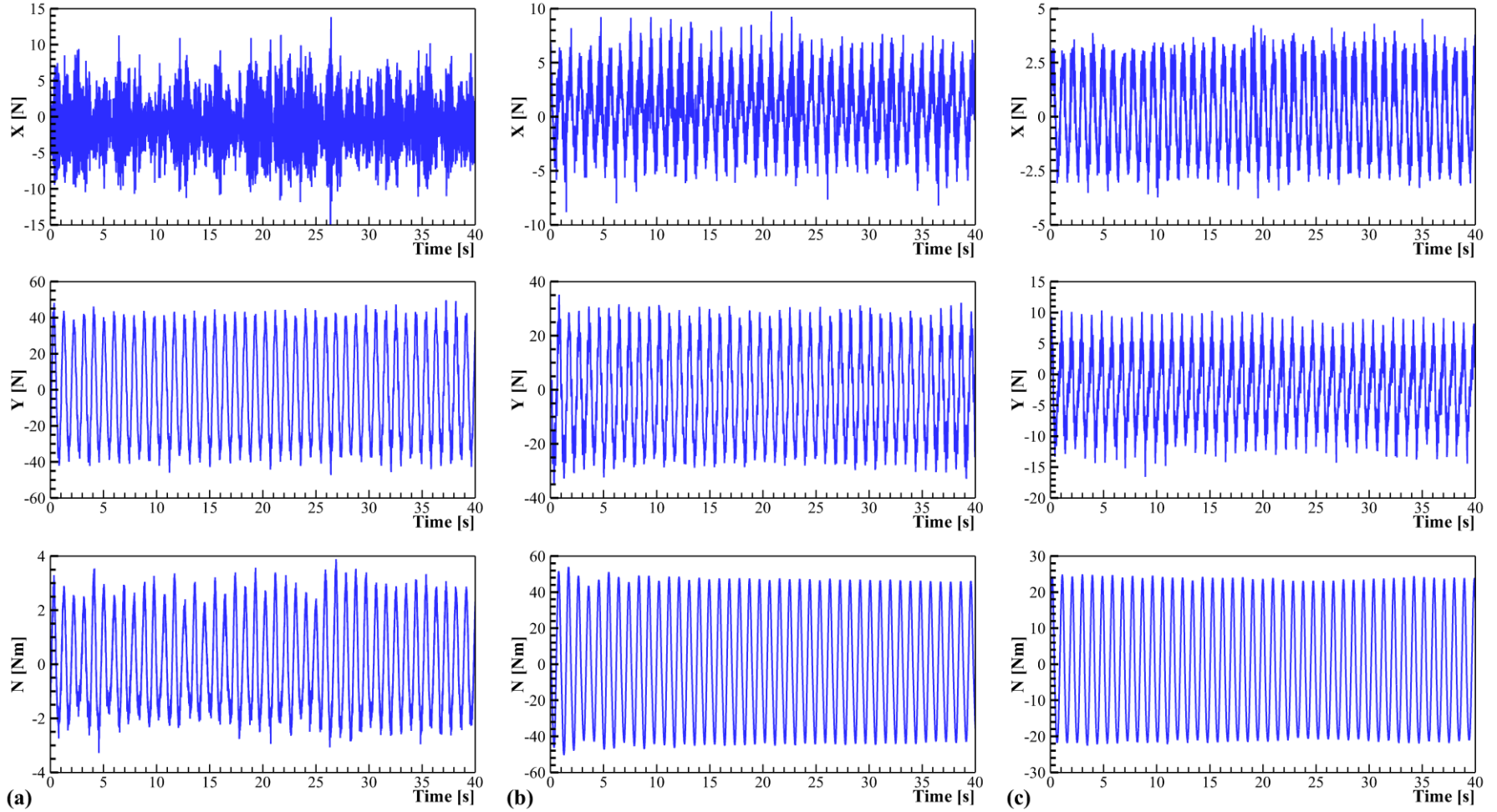


Figure C.15 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.

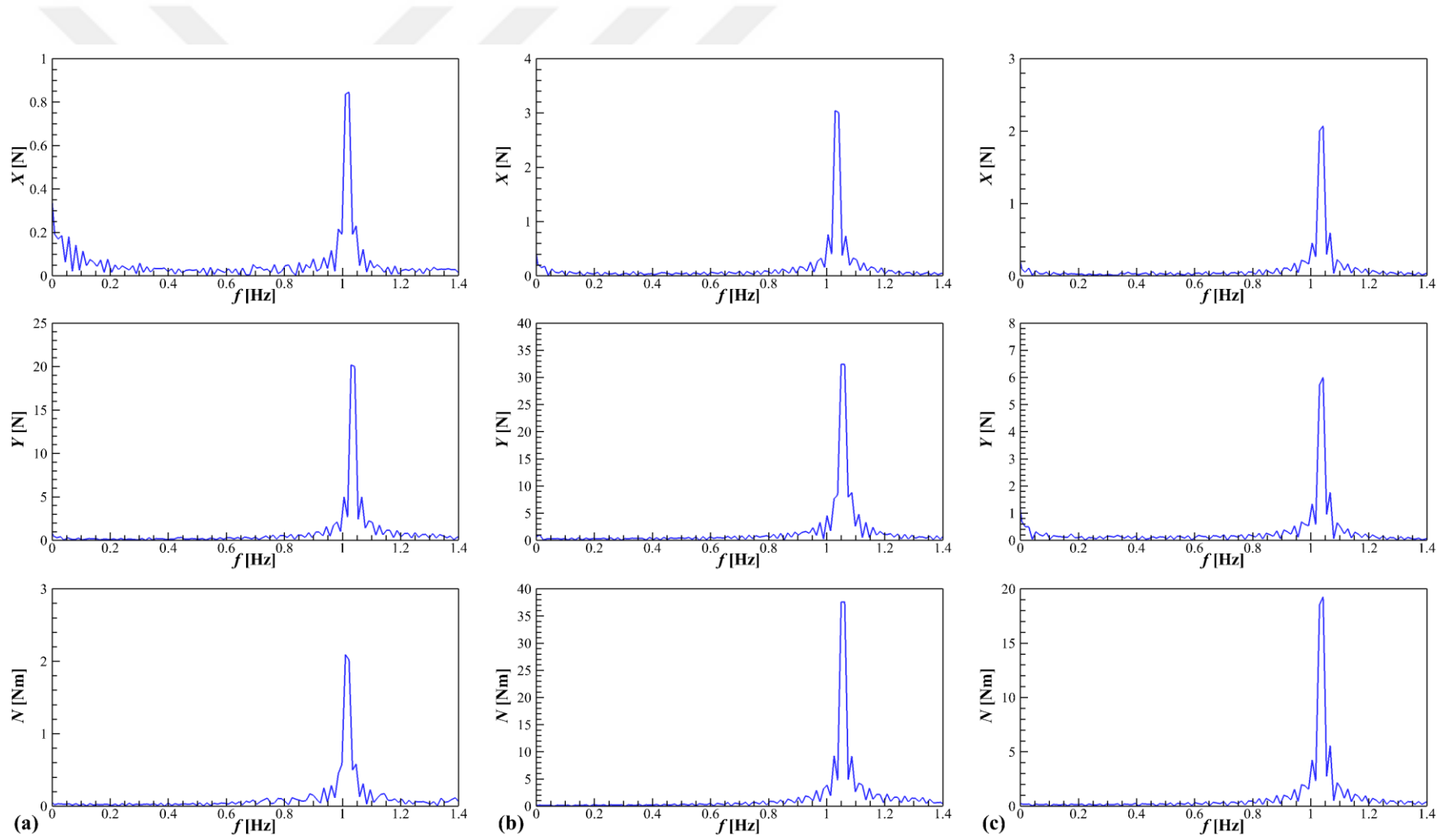


Figure C.16 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.

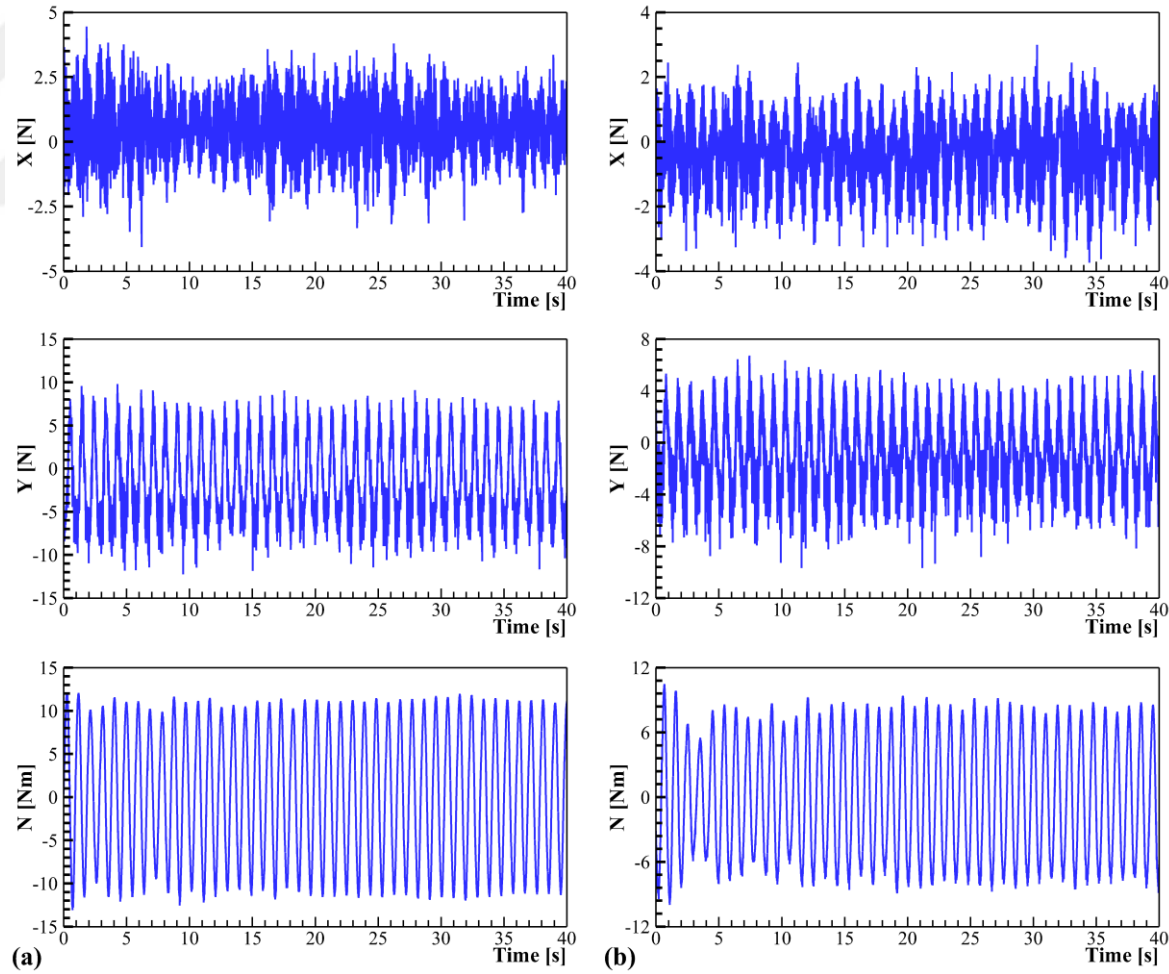


Figure C.17 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$.

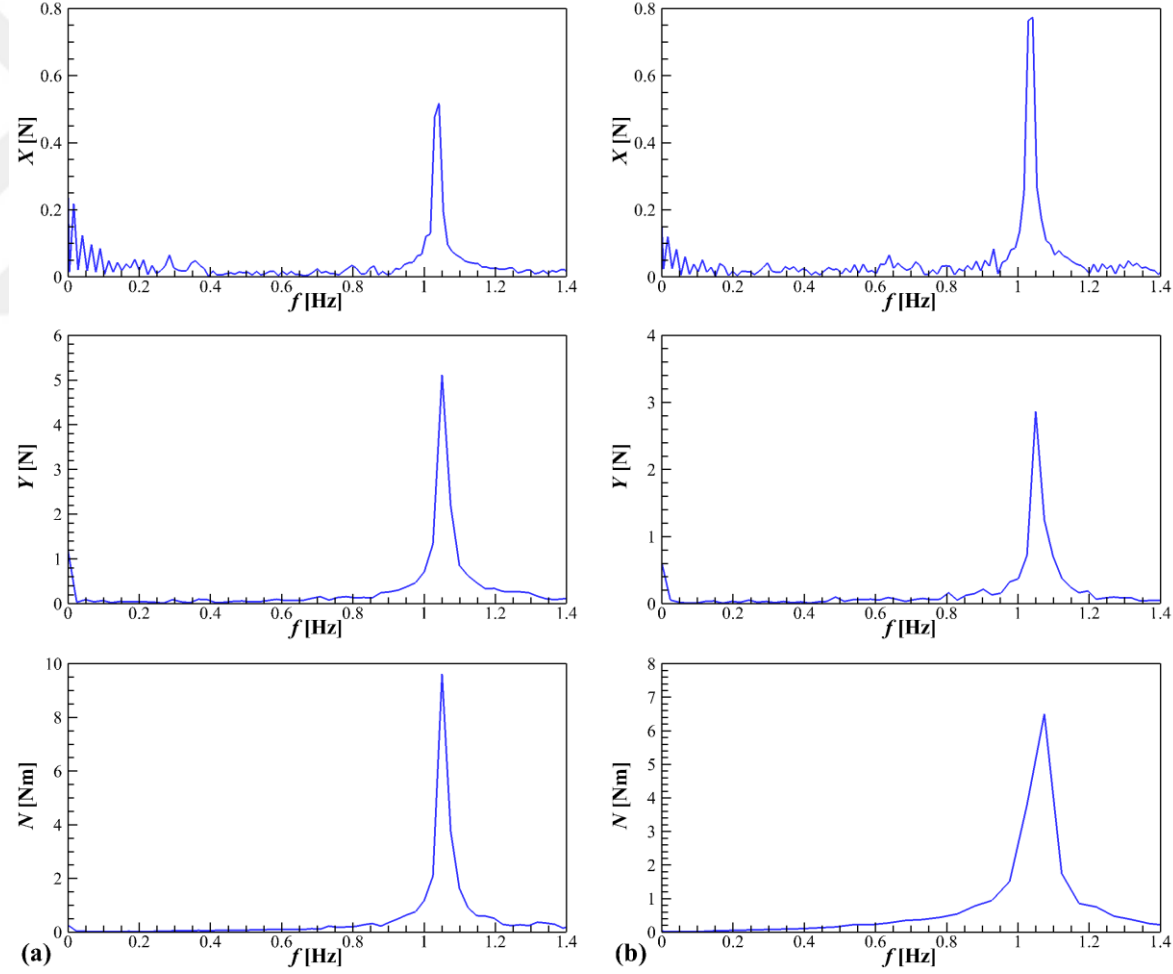


Figure C.18 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$.

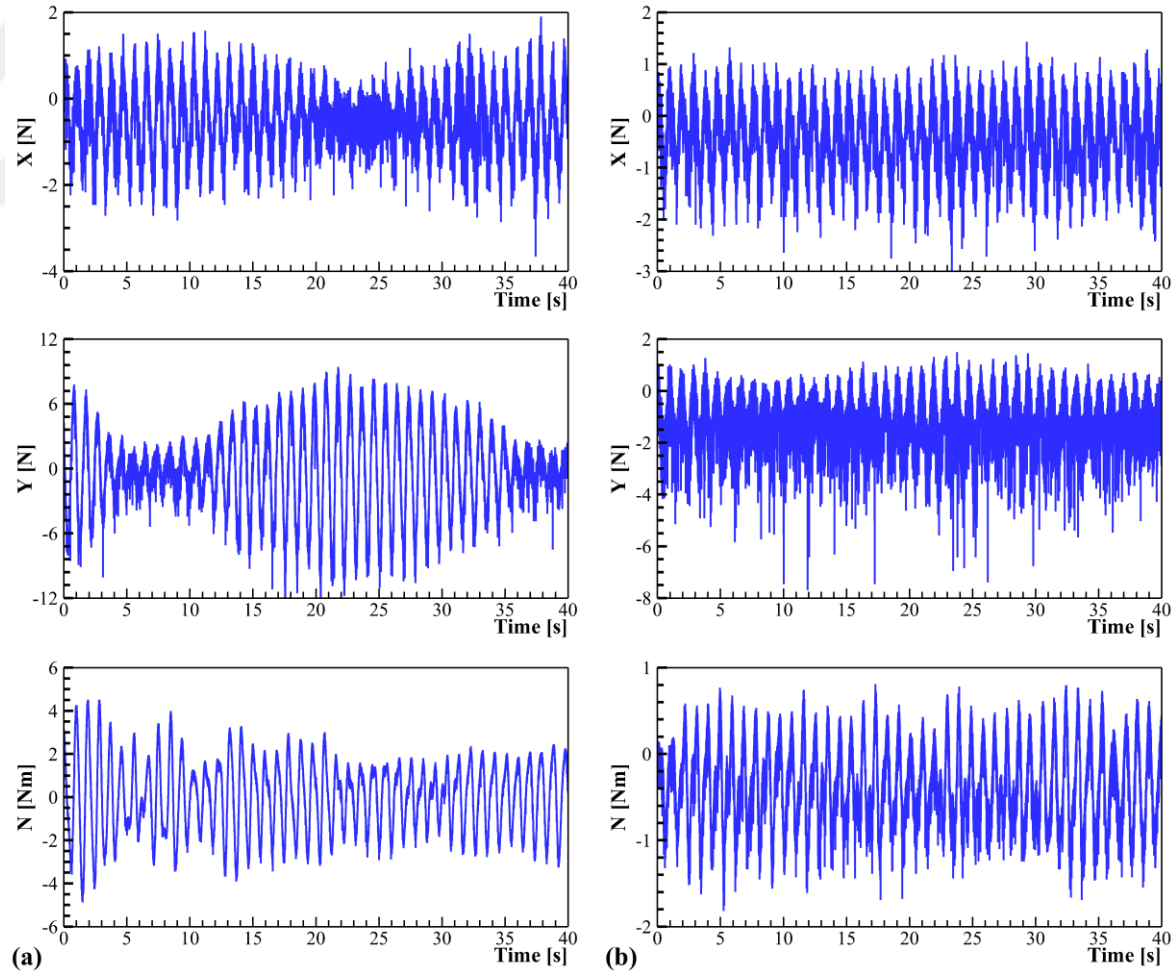


Figure C.19 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$.

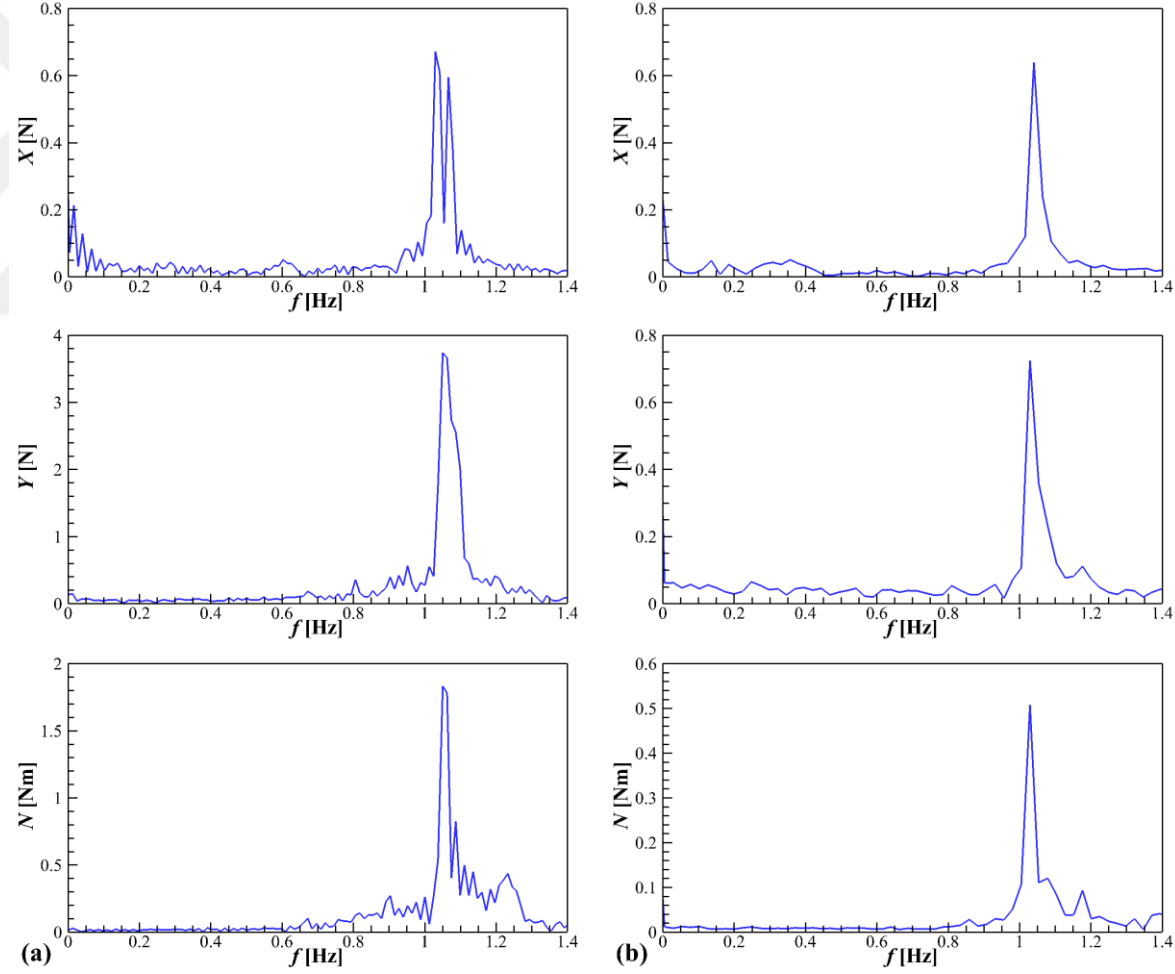


Figure C.20 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$.

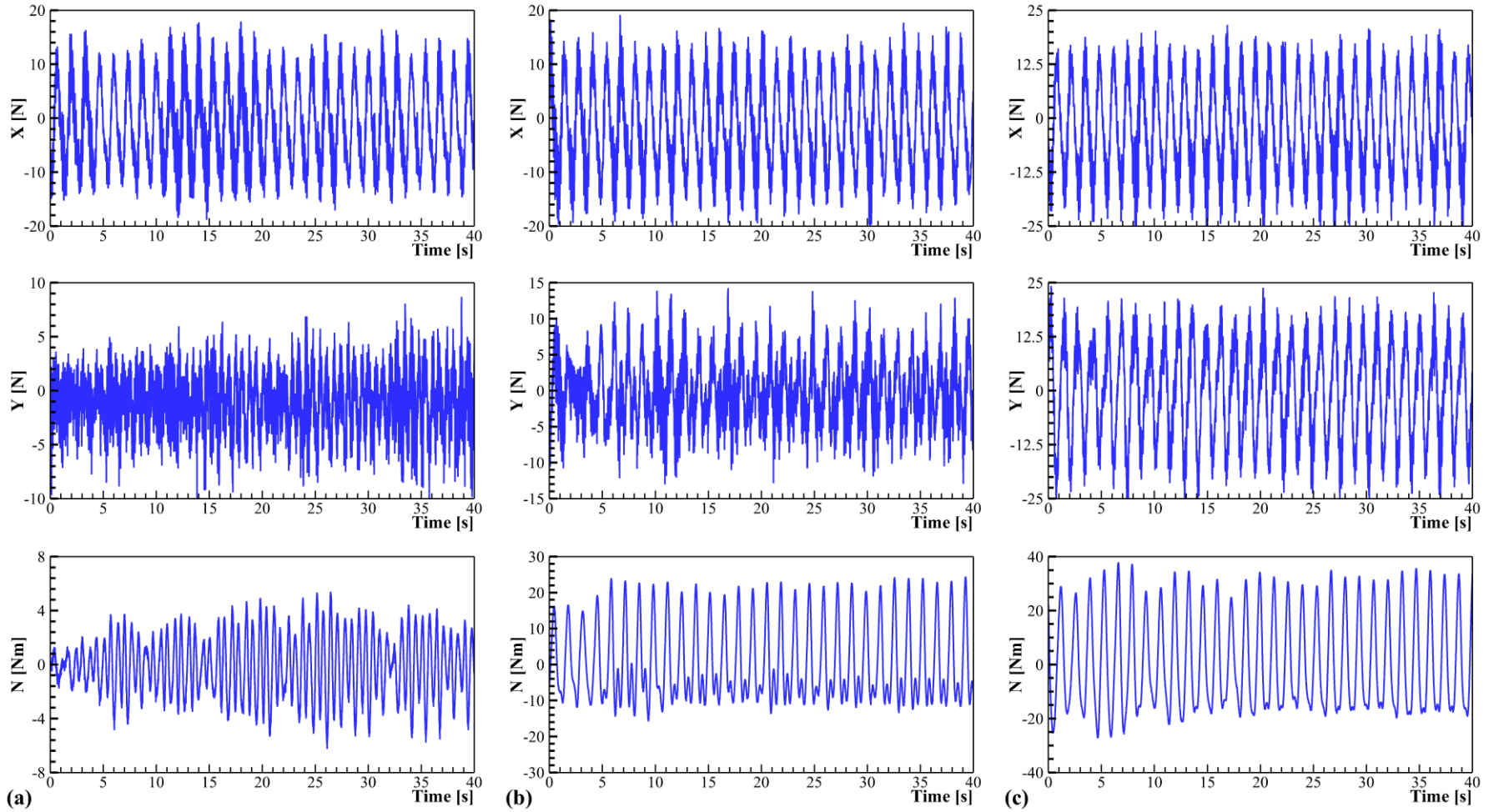


Figure C.21 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$.

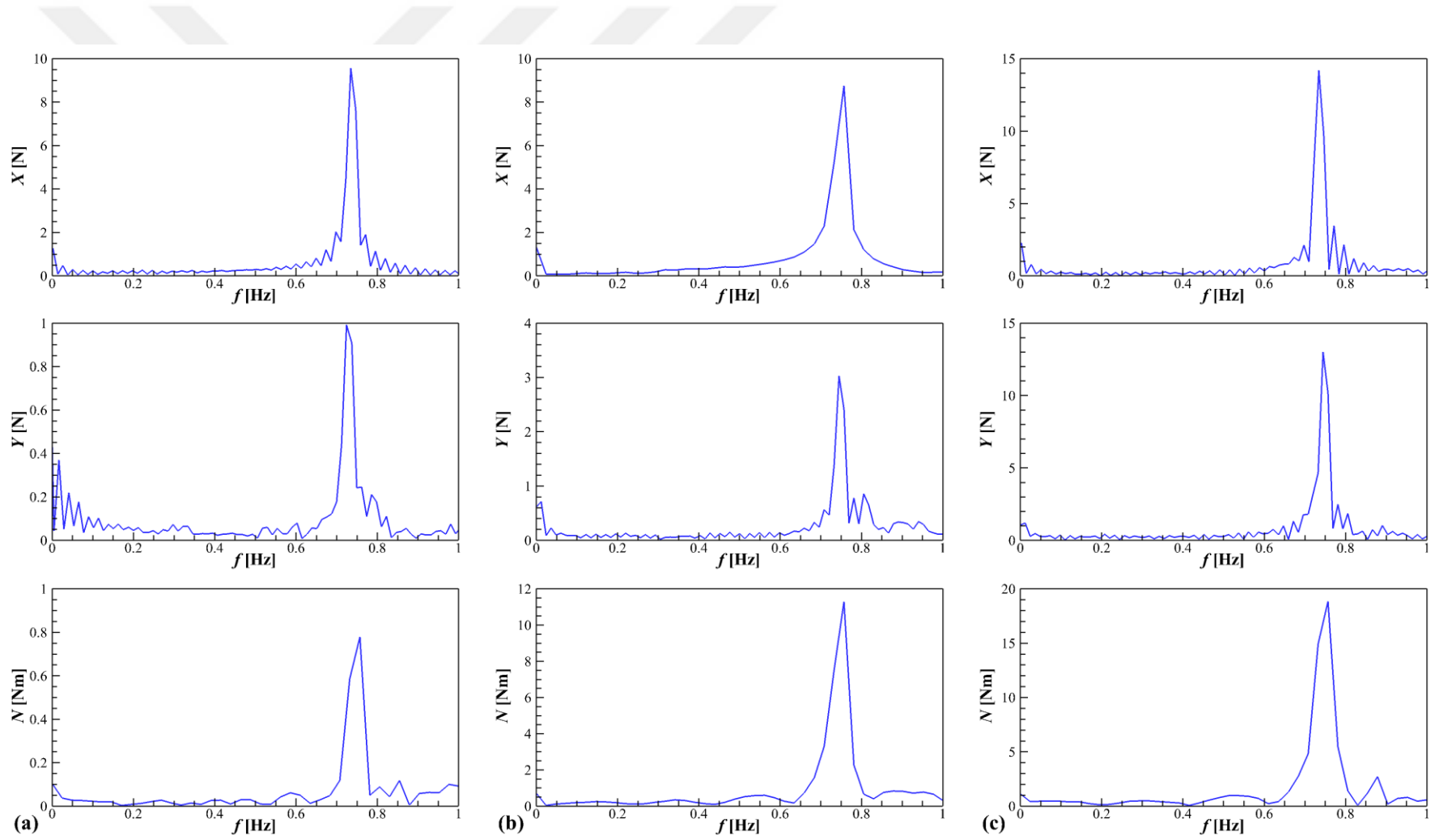


Figure C.22 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$.

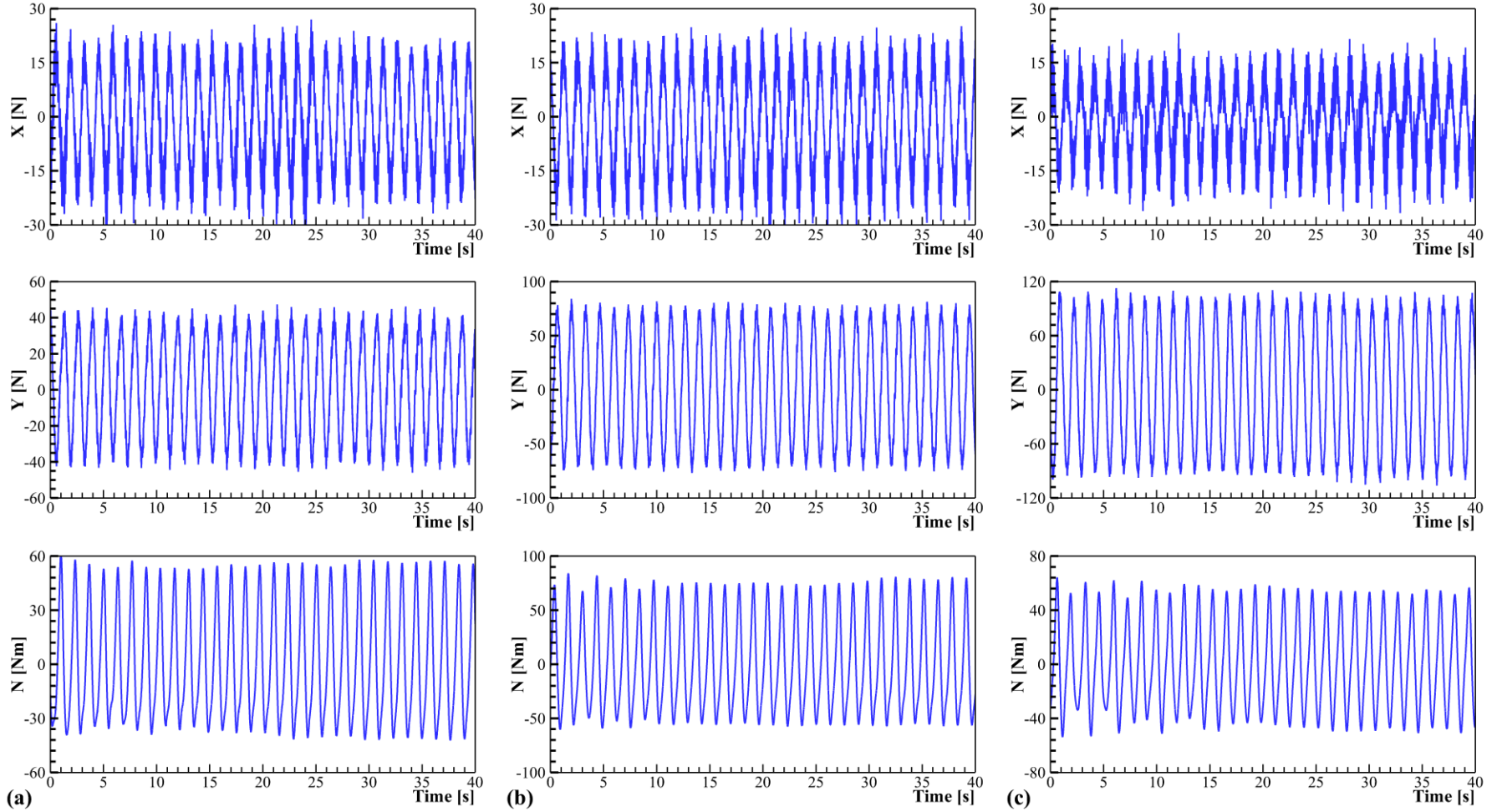


Figure C.23 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$.

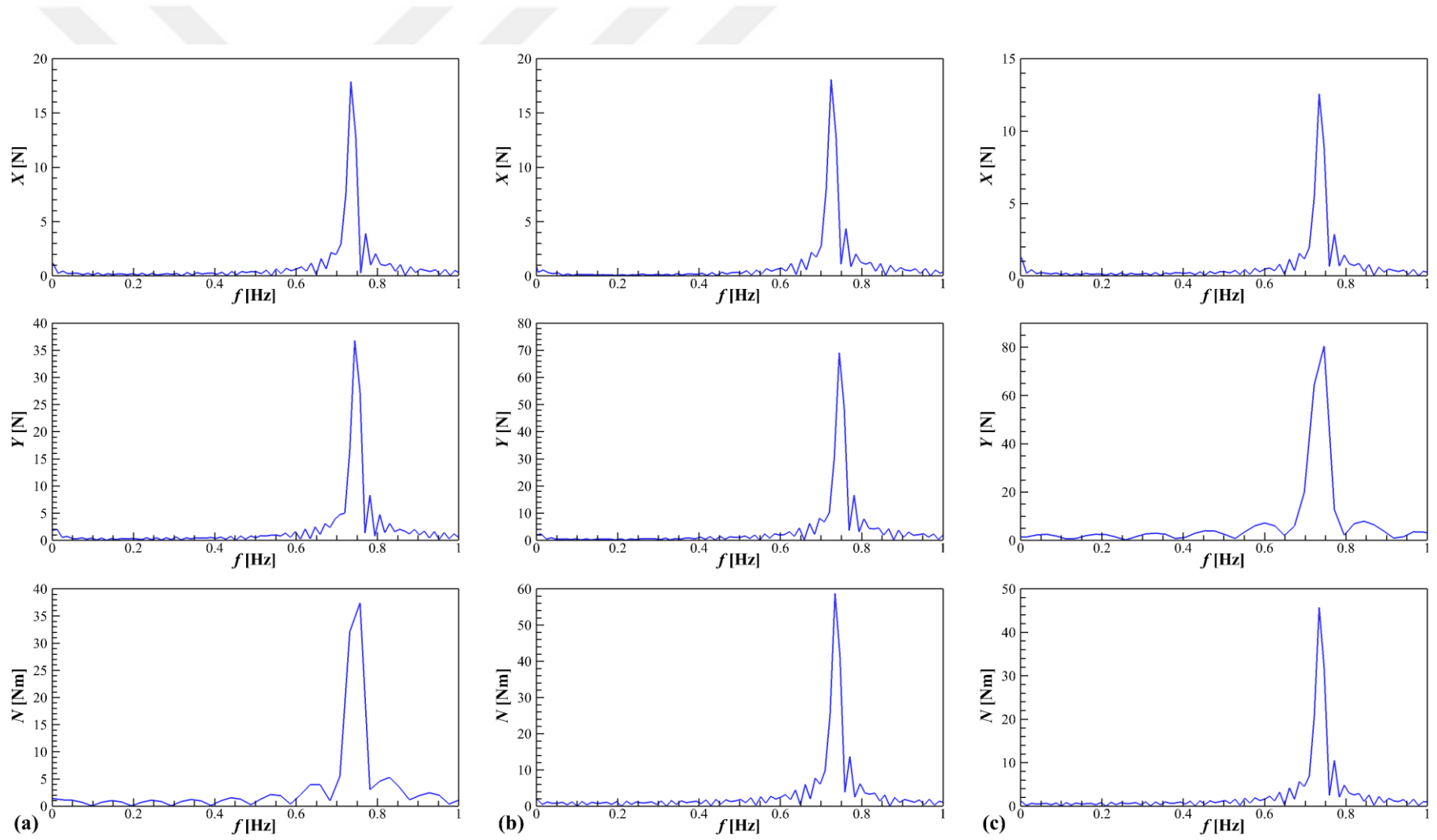


Figure C.24 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$.

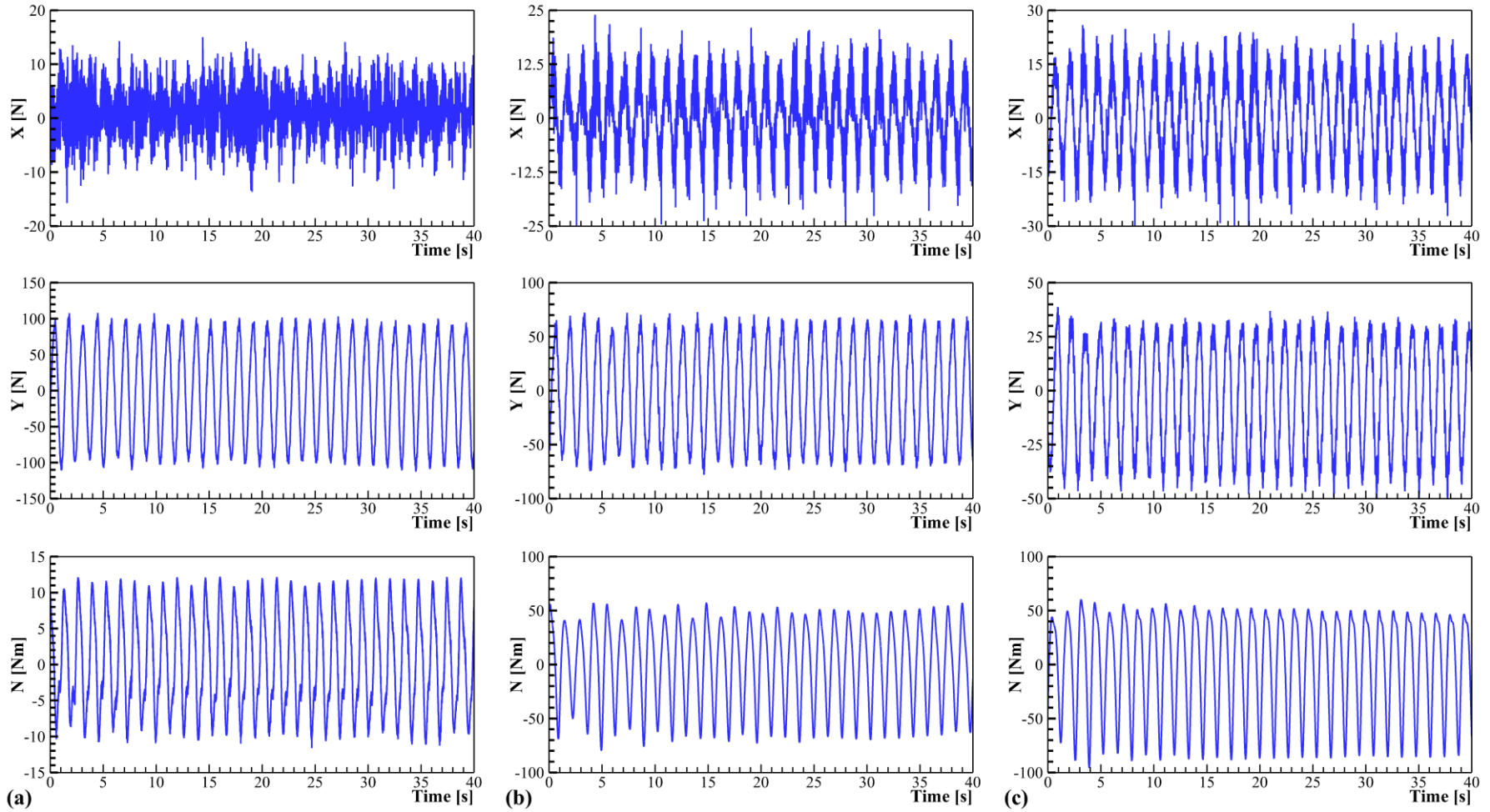


Figure C.25 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.

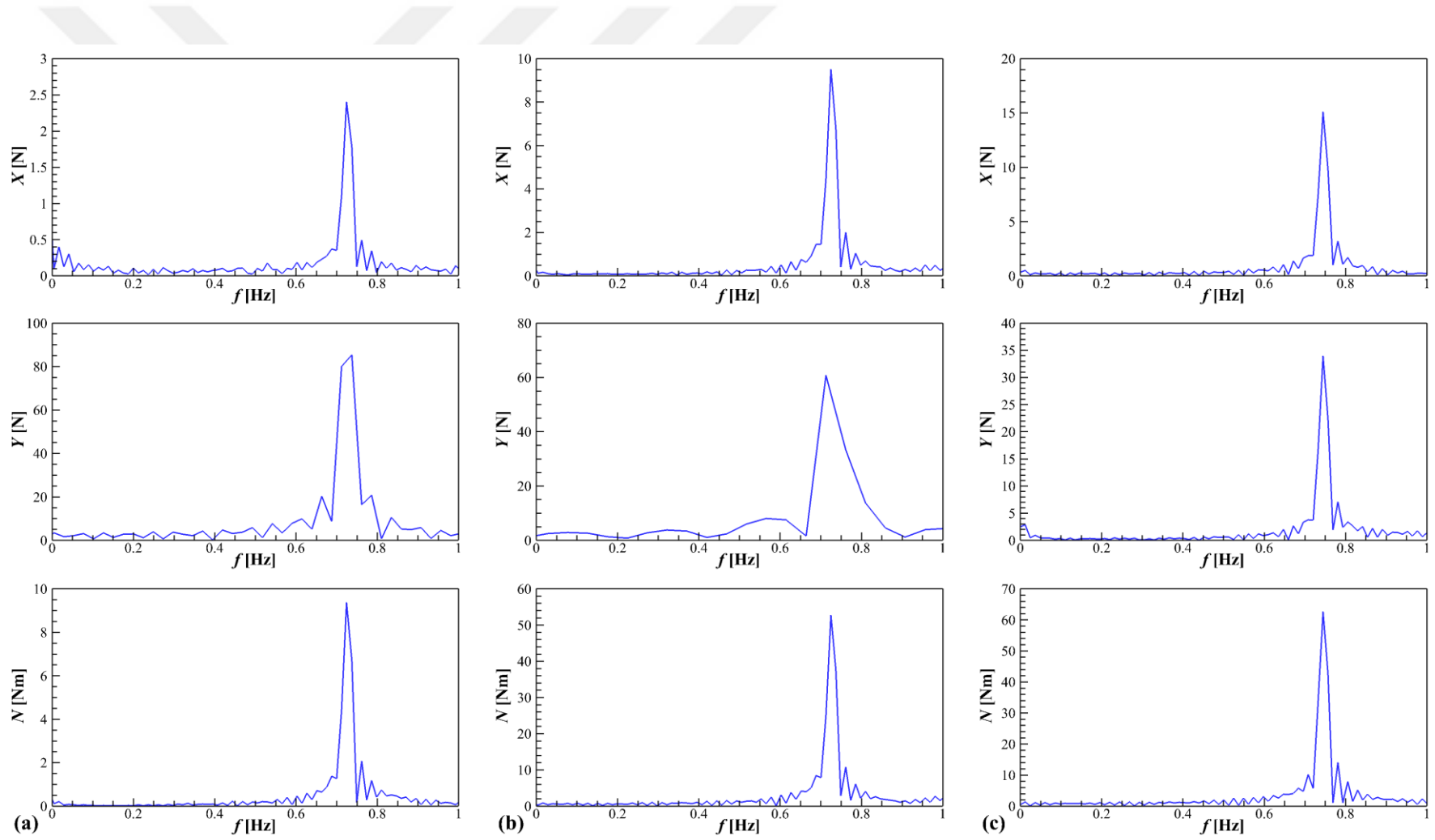


Figure C.26 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.

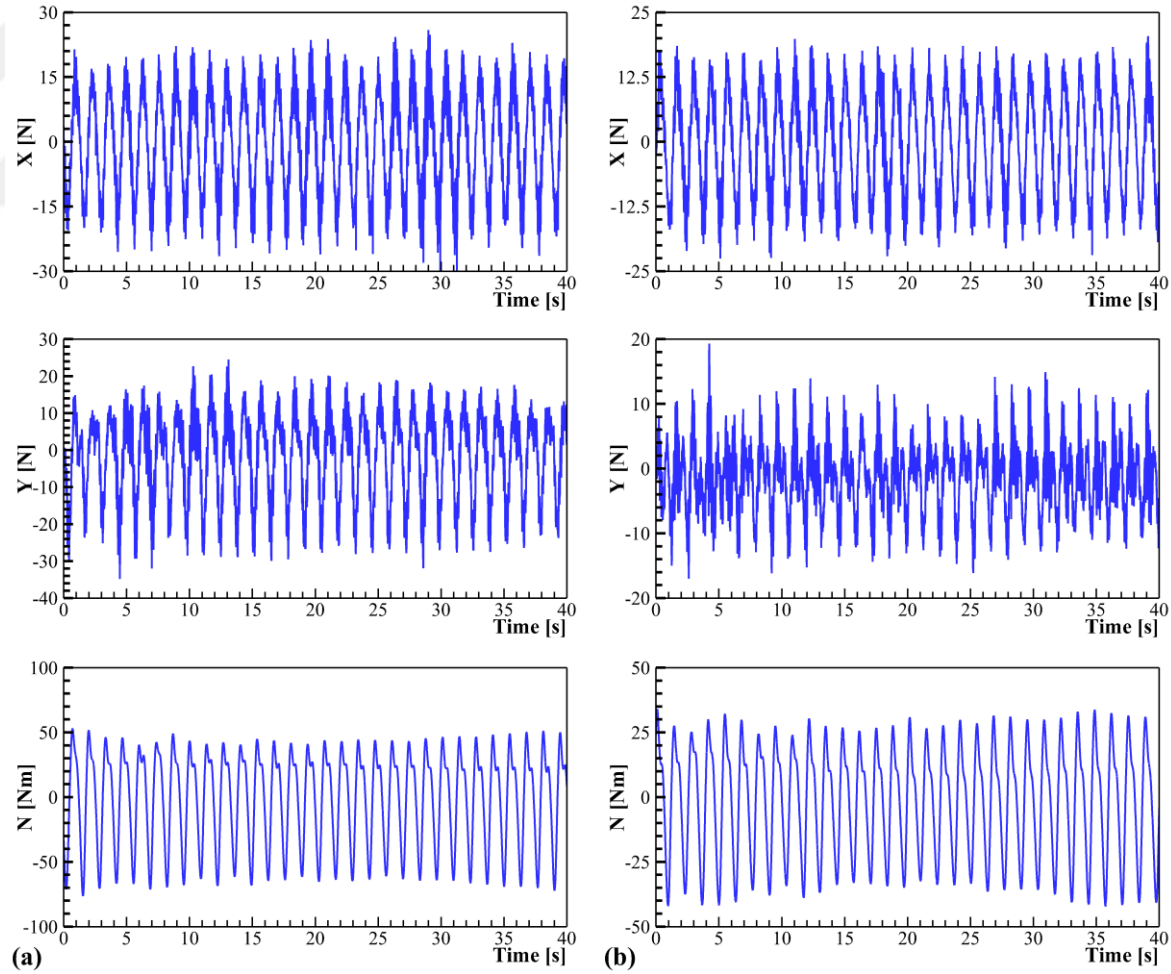


Figure C.27 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$.

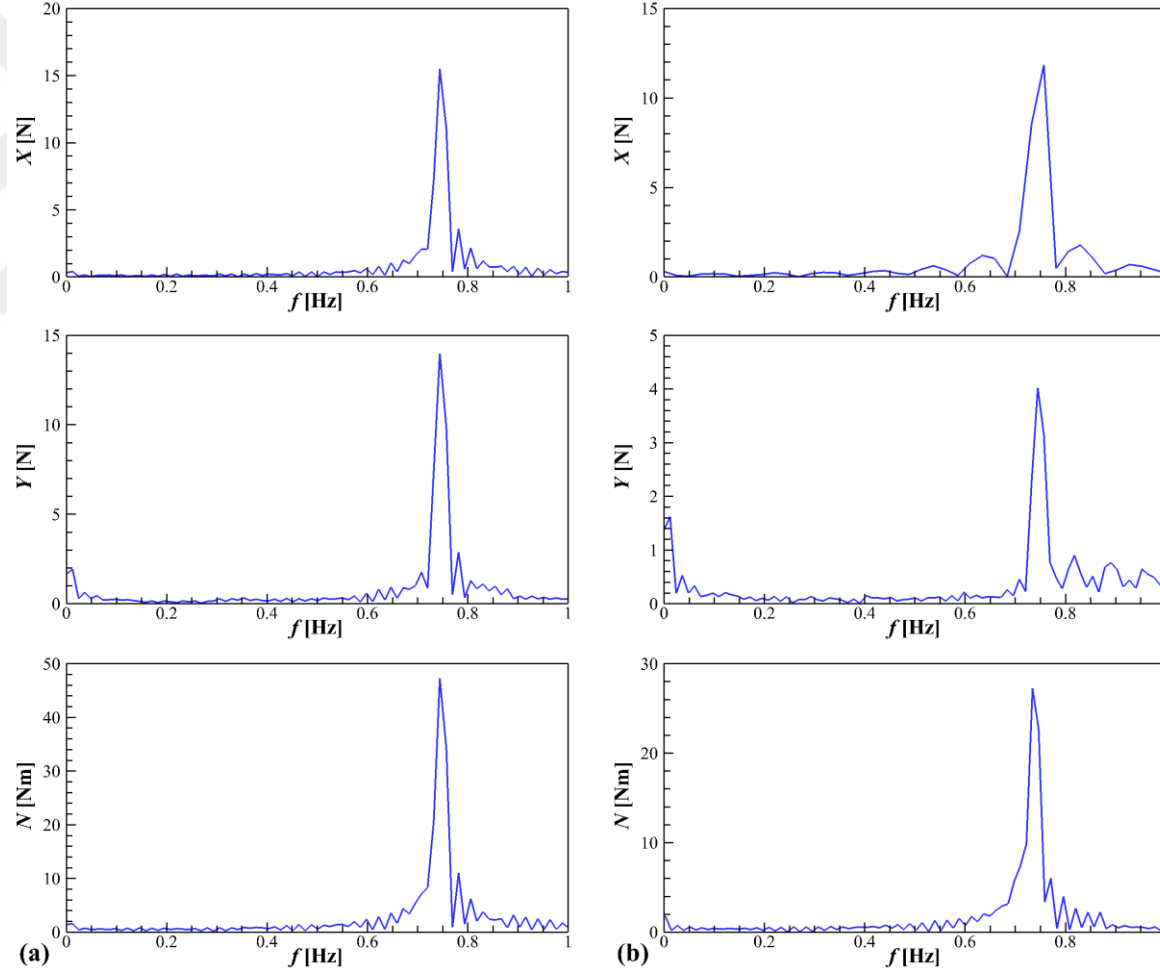


Figure C.28 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 135^\circ$, and (b) $\psi = 150^\circ$.

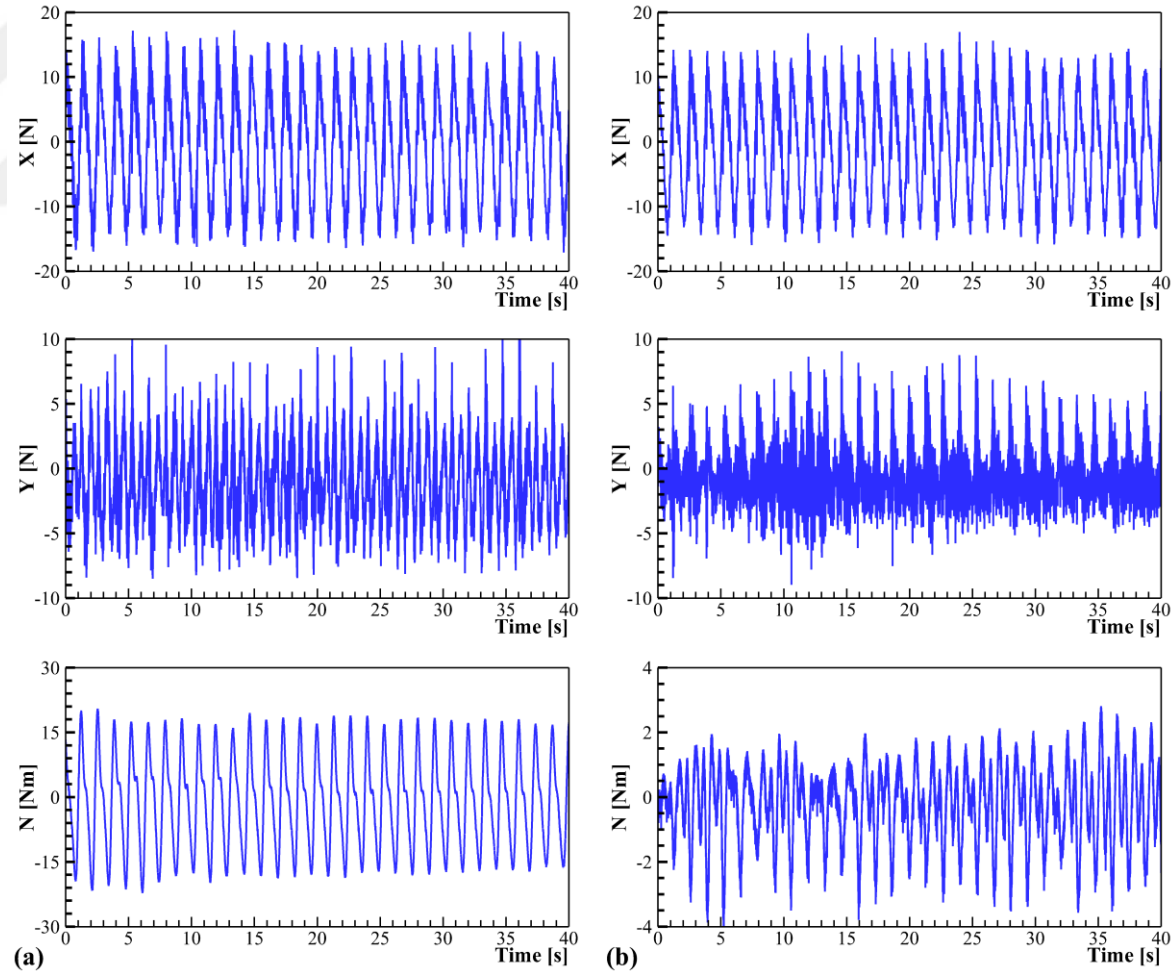


Figure C.29 : Time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$.

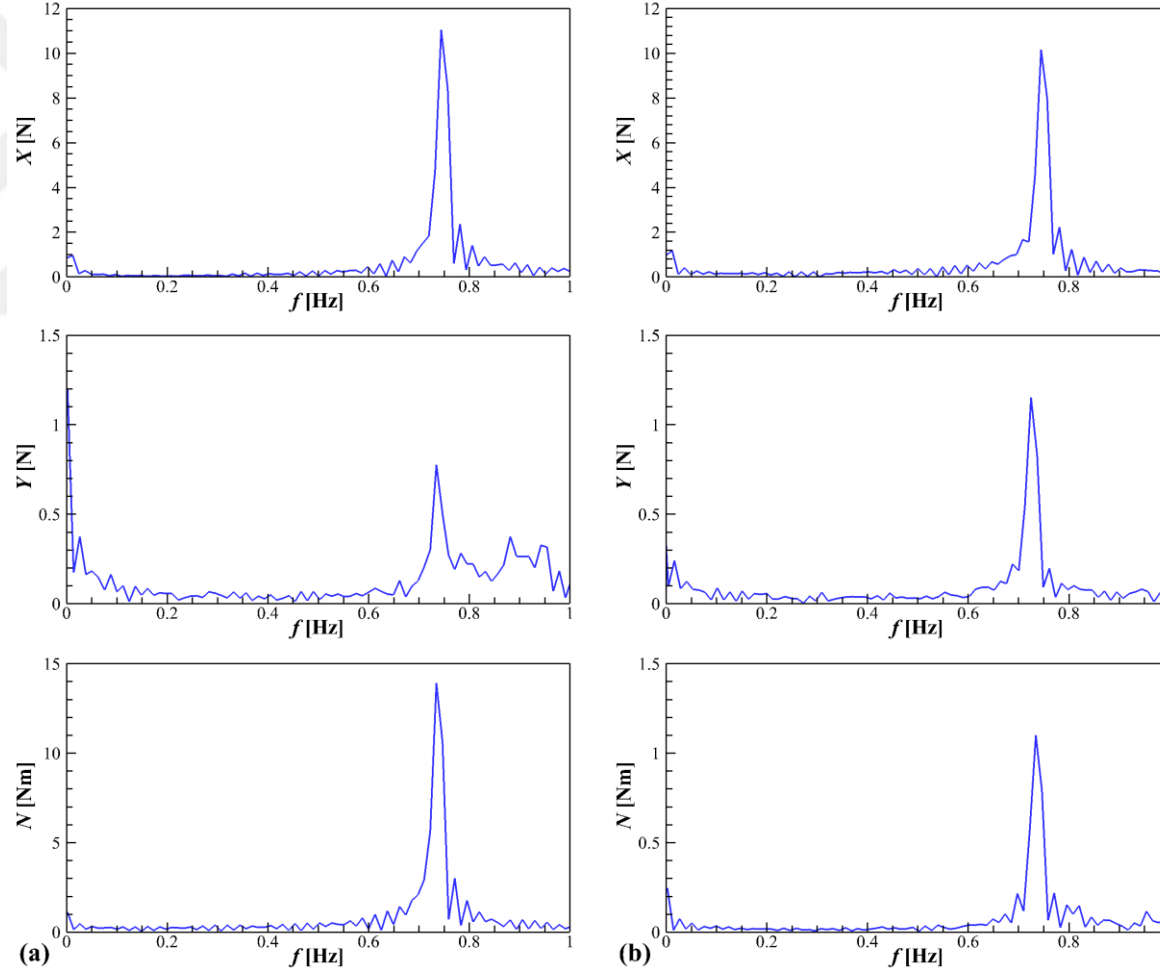


Figure C.30 : FFT analysis of time histories of experimental X , Y forces and N moment at $H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 165^\circ$, and (b) $\psi = 180^\circ$.

APPENDIX D Experimental results of wave drift loads for " $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$ " and " $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$ "

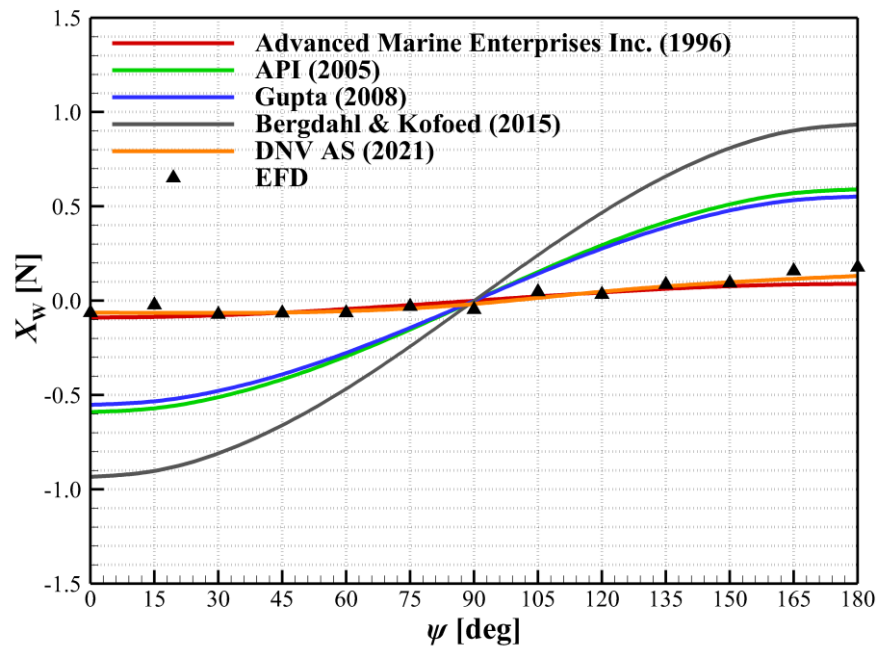


Figure D.1 : Comparison of longitudinal wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$.

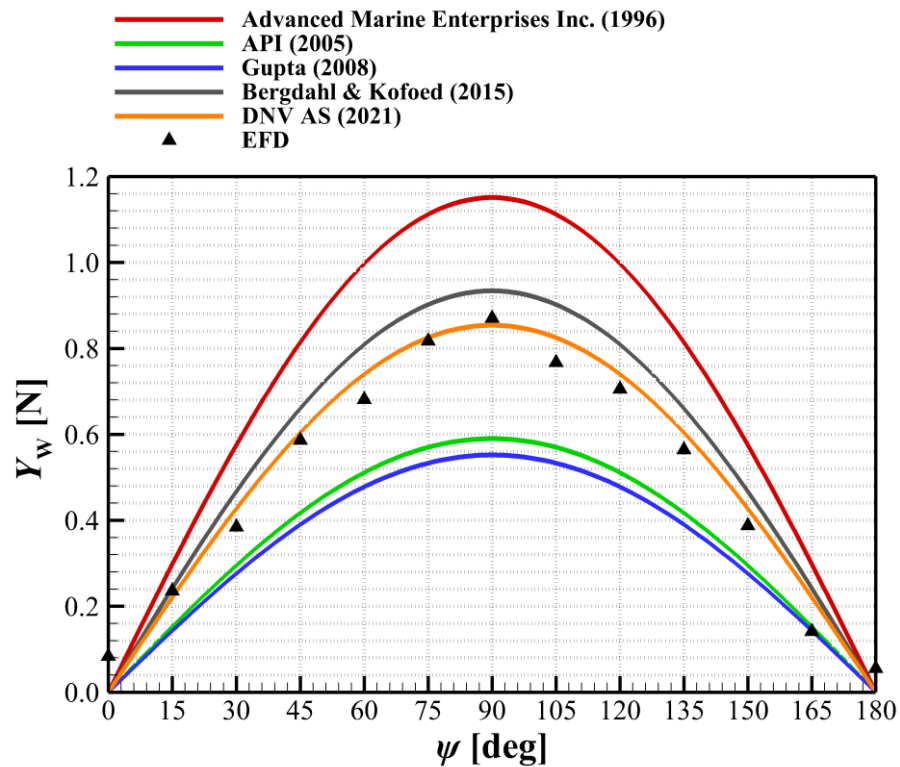


Figure D.2 : Comparison of lateral wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$.

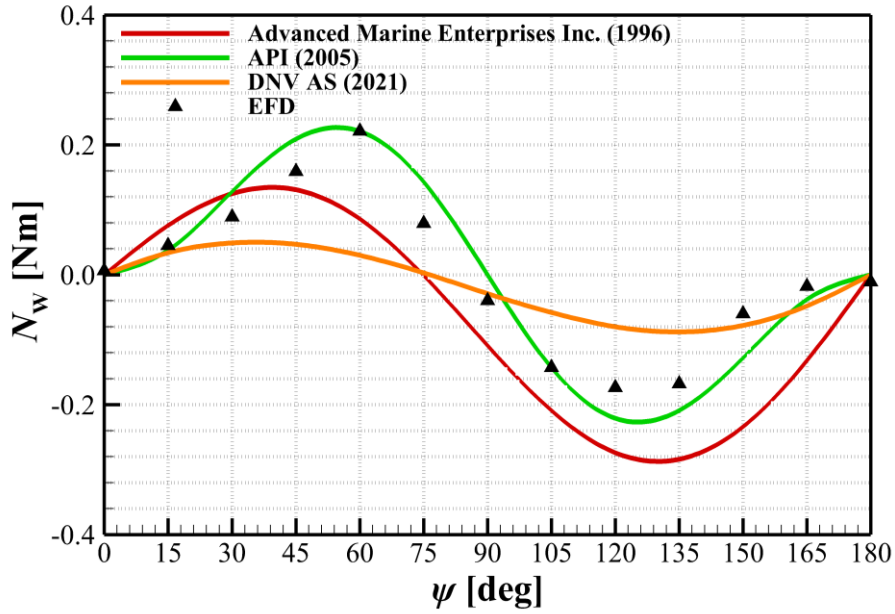


Figure D.3 : Comparison of yaw moments of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$.

Table D.1 : Wave drift loads of steady drift tests with various heading angles at $H/\lambda = 0.020, \lambda/L_{WL} = 0.50$.

ψ [deg]	$\bar{X}_W$ [N]	$\bar{Y}_W$ [N]	$\bar{N}_W$ [Nm]
0	-0.063	0.083	0.006
15	-0.023	0.235	0.045
30	-0.073	0.384	0.089
45	-0.066	0.586	0.159
60	-0.065	0.681	0.221
75	-0.031	0.817	0.079
90	-0.047	0.870	-0.040
105	0.047	0.767	-0.143
120	0.033	0.705	-0.174
135	0.085	0.564	-0.168
150	0.093	0.387	-0.060
165	0.158	0.141	-0.018
180	0.177	0.055	-0.011

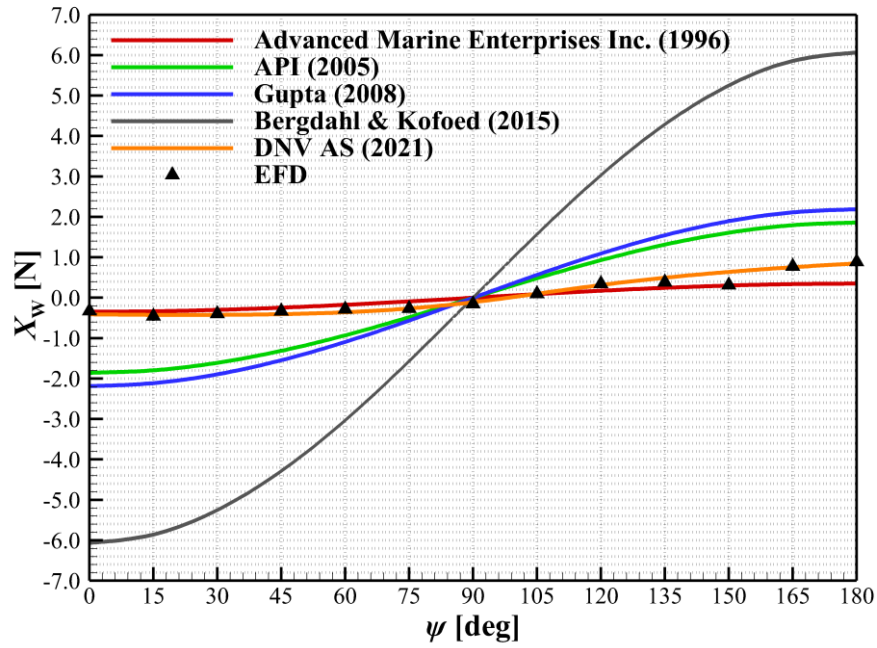


Figure D.4 : Comparison of longitudinal wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$.

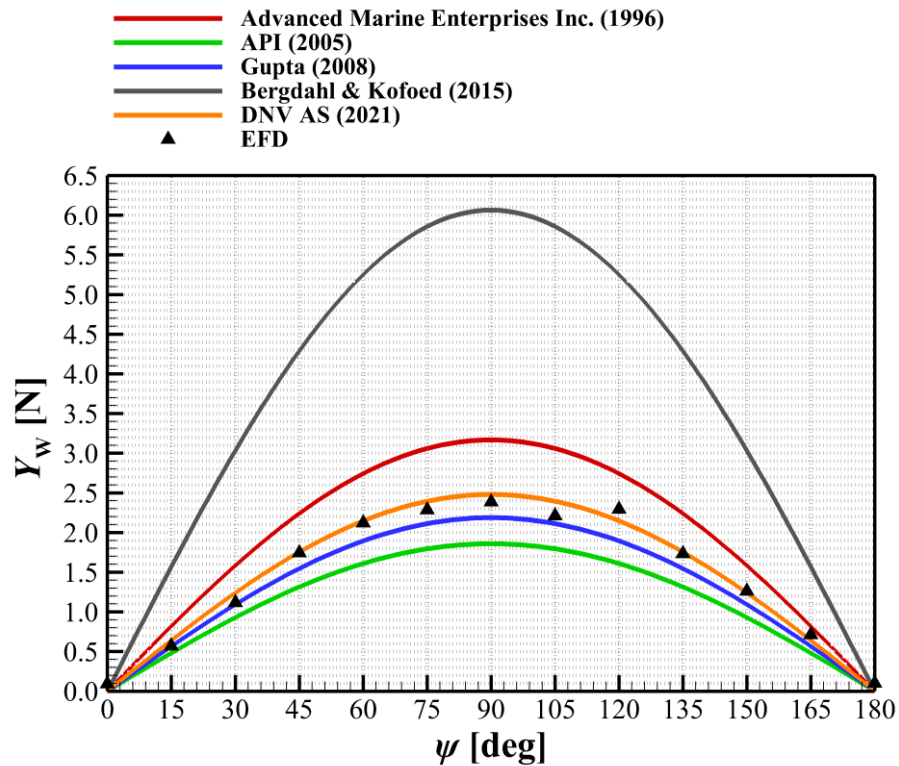


Figure D.5 : Comparison of lateral wave drift forces of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$.

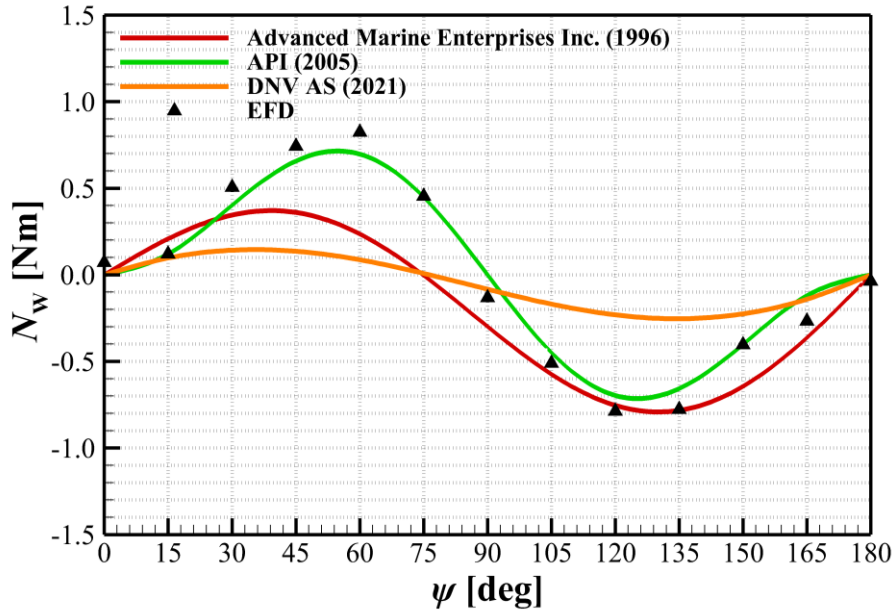


Figure D.6 : Comparison of yaw moments of steady drift tests and empirical/analytical approaches at $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$.

Table D.2 : Wave drift loads of steady drift tests with various heading angles at $H/\lambda = 0.025, \lambda/L_{WL} = 1.00$.

ψ [deg]	$\bar{X}_W$ [N]	$\bar{Y}_W$ [N]	$\bar{N}_W$ [Nm]
0	-0.330	0.092	0.070
15	-0.459	0.567	0.120
30	-0.397	1.116	0.505
45	-0.334	1.744	0.741
60	-0.285	2.119	0.823
75	-0.267	2.283	0.453
90	-0.154	2.384	-0.133
105	0.092	2.209	-0.511
120	0.346	2.293	-0.789
135	0.381	1.732	-0.778
150	0.314	1.260	-0.405
165	0.774	0.710	-0.268
180	0.886	0.100	-0.037

APPENDIX E Time histories of wave signals

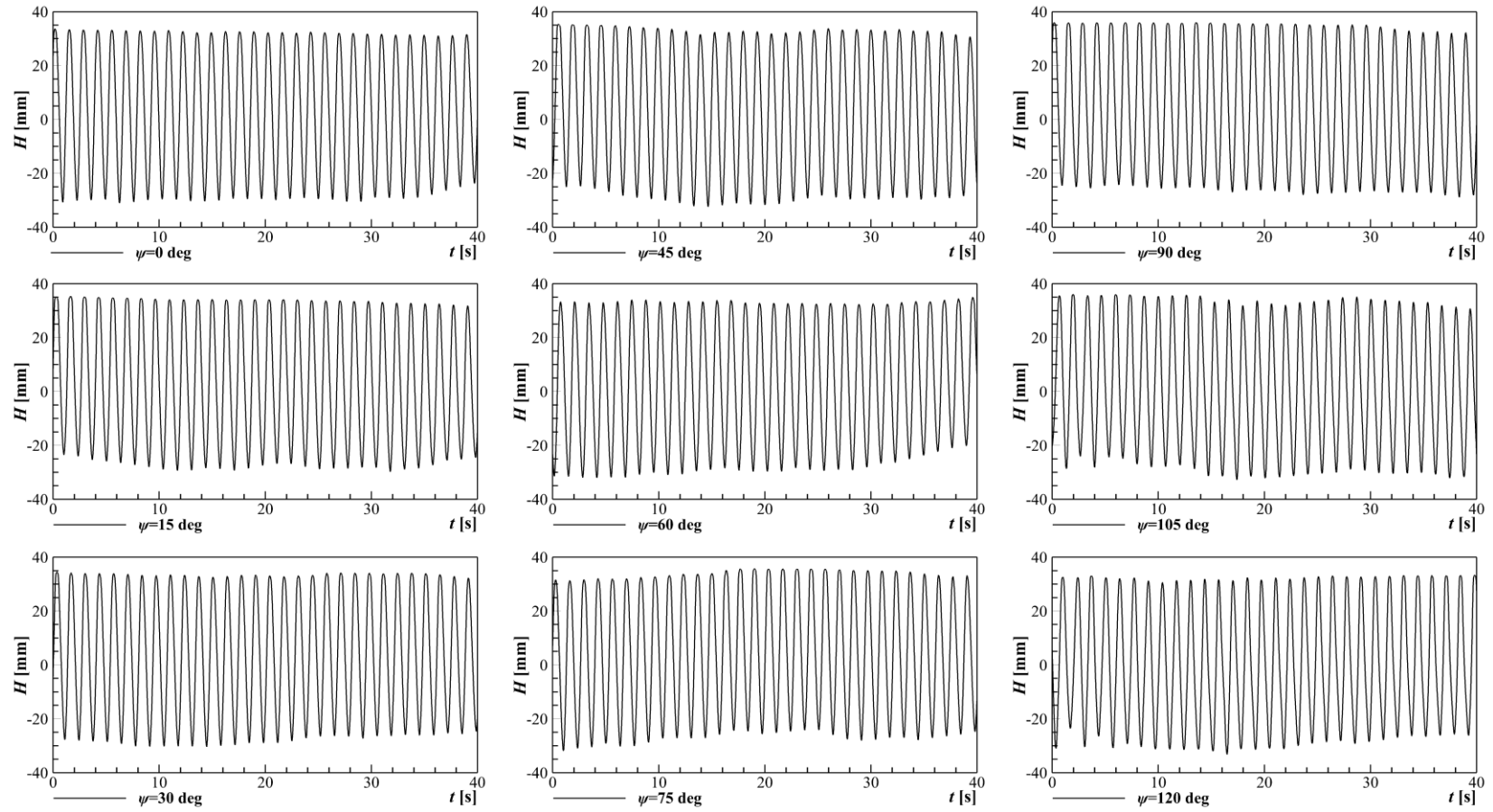


Figure E.1 : Time histories of wave signals at $\psi = 0^\circ$ to 120° ($H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$).

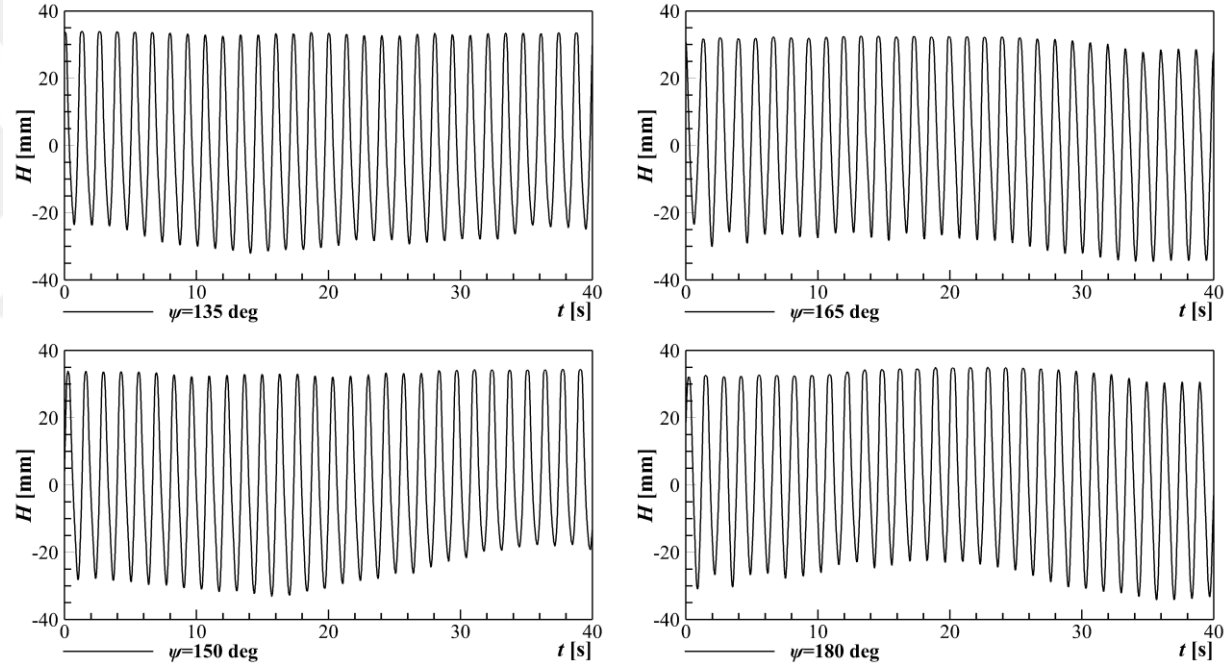


Figure E.2 : Time histories of wave signals at $\psi = 120^\circ$ to 180° ($H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$).

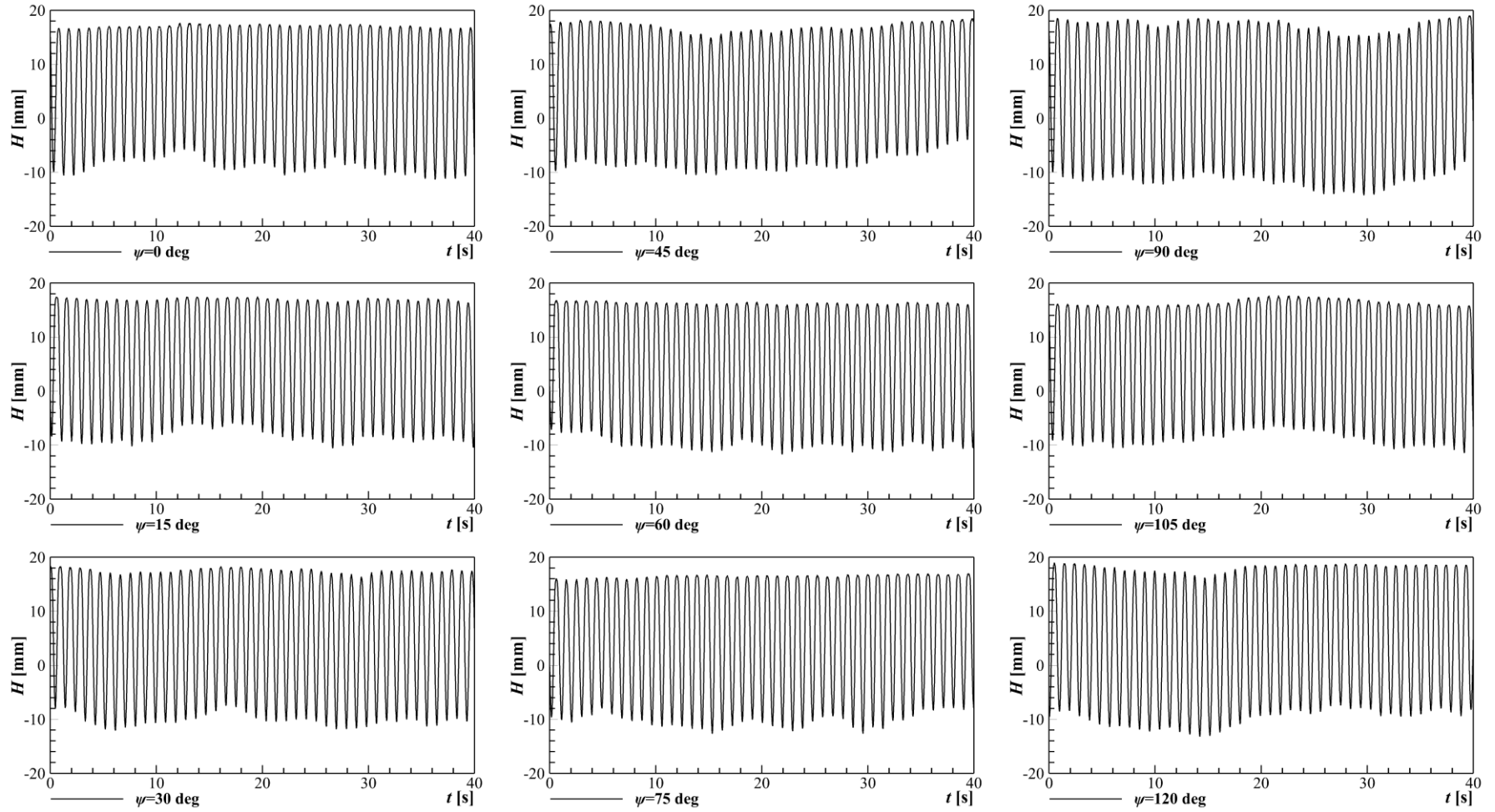


Figure E.3 : Time histories of wave signals at $\psi = 0^\circ$ to 120° ($H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$).

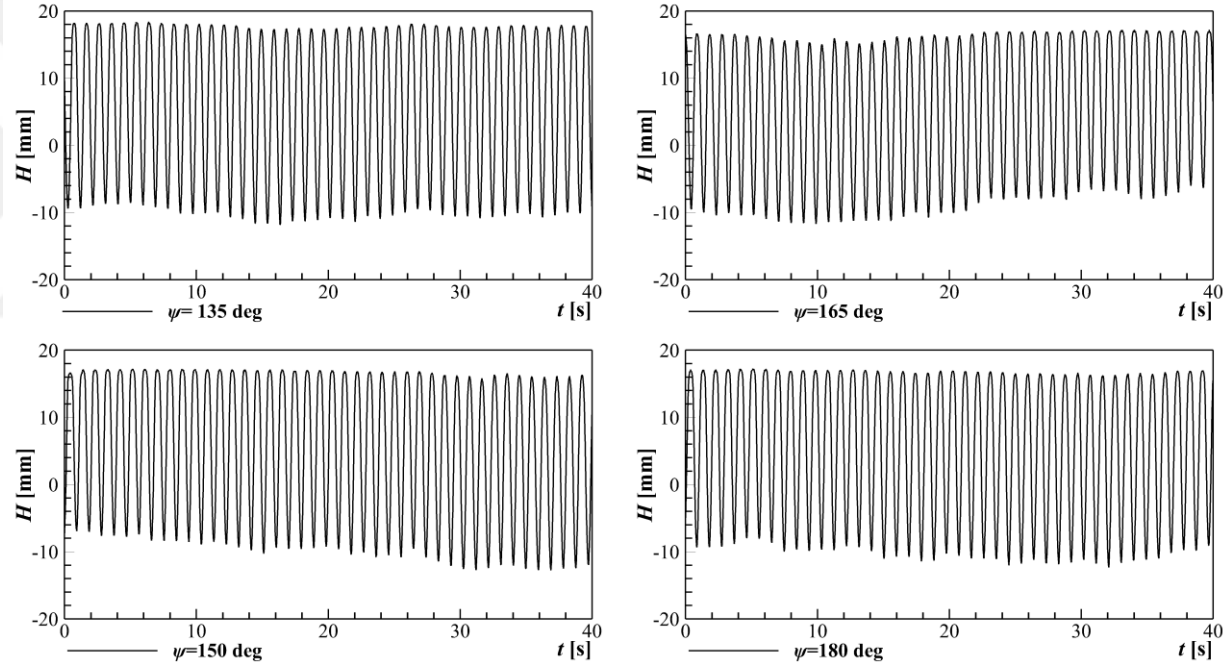


Figure E.4 : Time histories of wave signals at $\psi = 120^\circ$ to 180° ($H/\lambda = 0.020$ and $\lambda/L_{WL} = 0.50$).

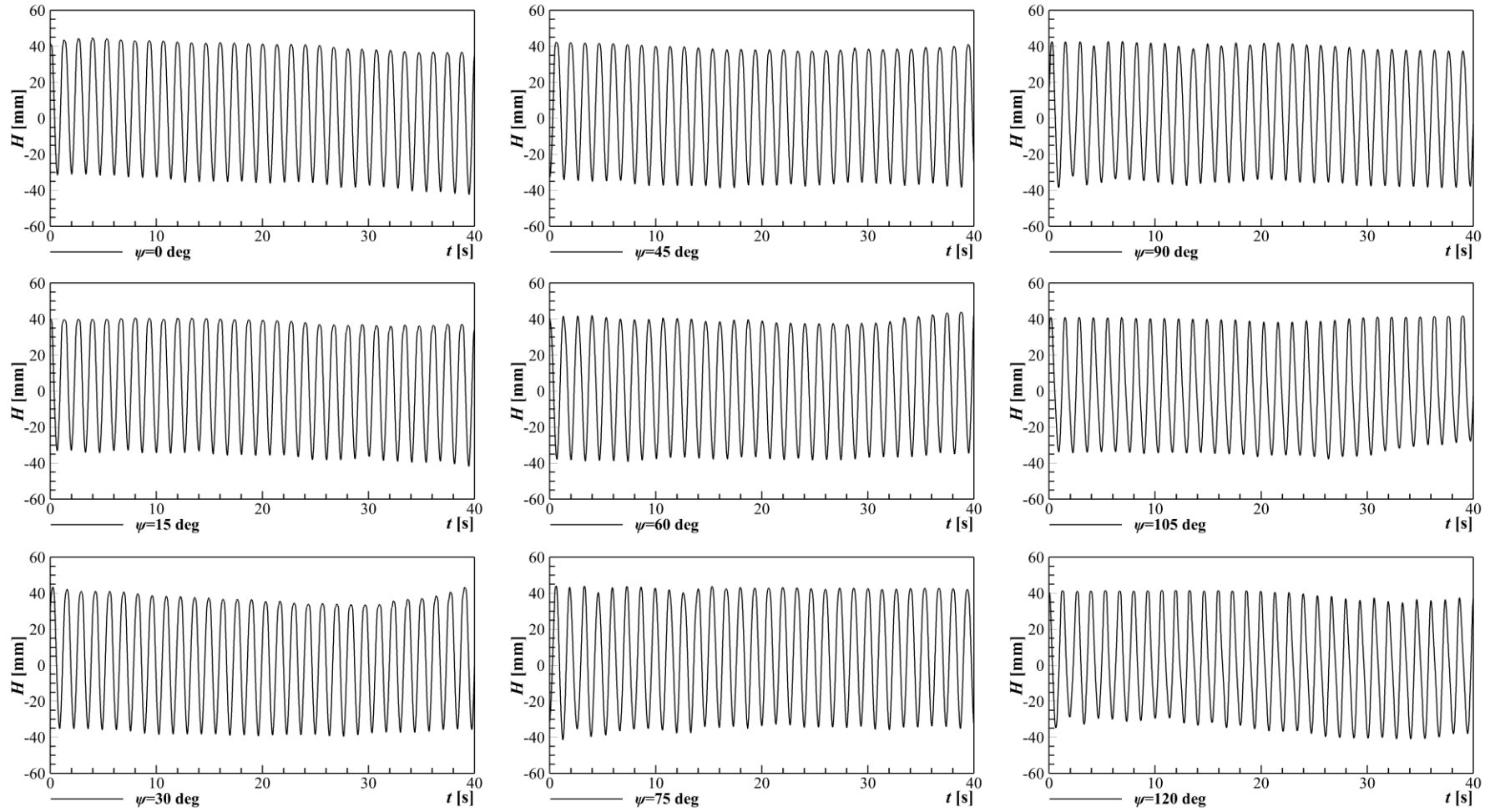


Figure E.5 : Time histories of wave signals at $\psi = 0^\circ$ to 120° ($H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$).

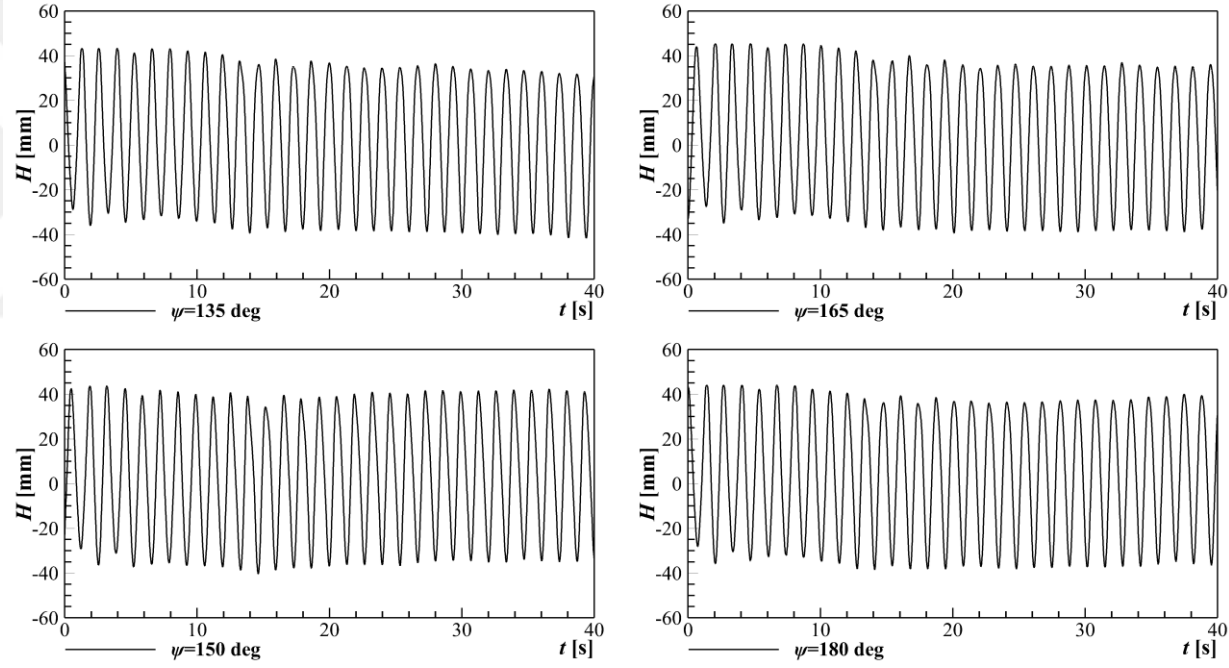


Figure E.6 : Time histories of wave signals at $\psi = 120^\circ$ to 180° ($H/\lambda = 0.025$ and $\lambda/L_{WL} = 1.00$).

APPENDIX F Time histories and FFT analyses of the computational study

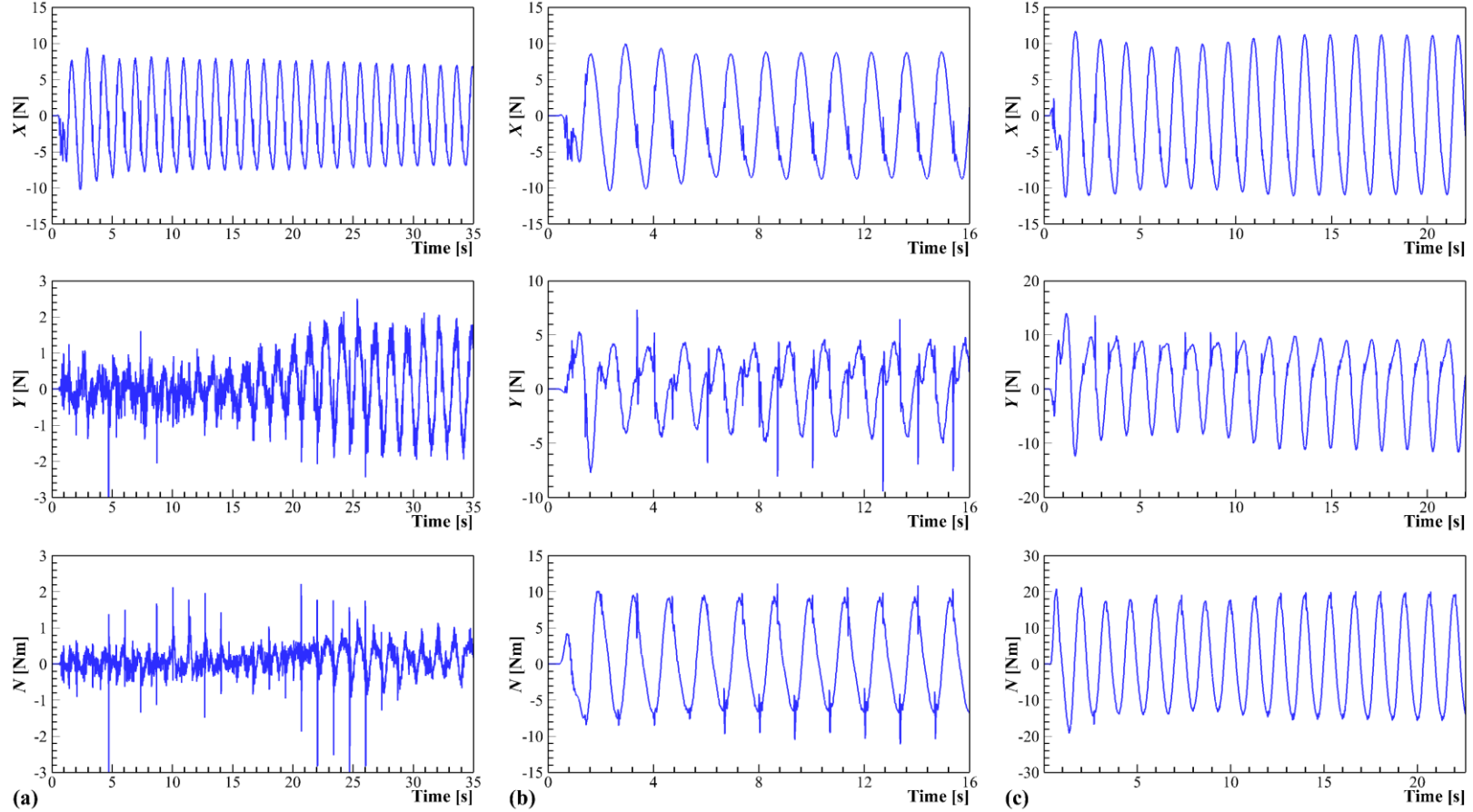


Figure F.1 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$.

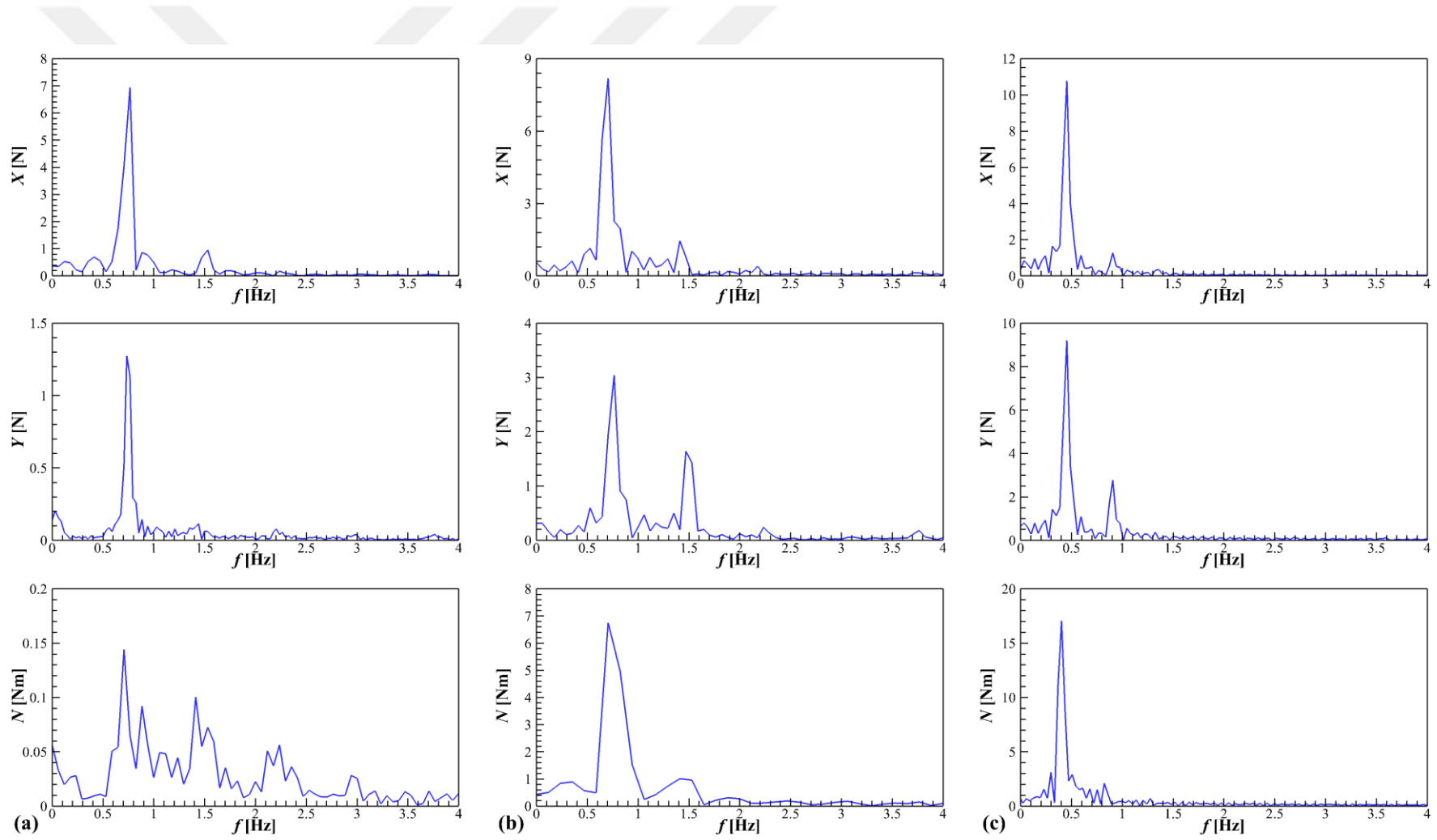


Figure F.2 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 0^\circ$, (b) $\psi = 15^\circ$, and (c) $\psi = 30^\circ$.

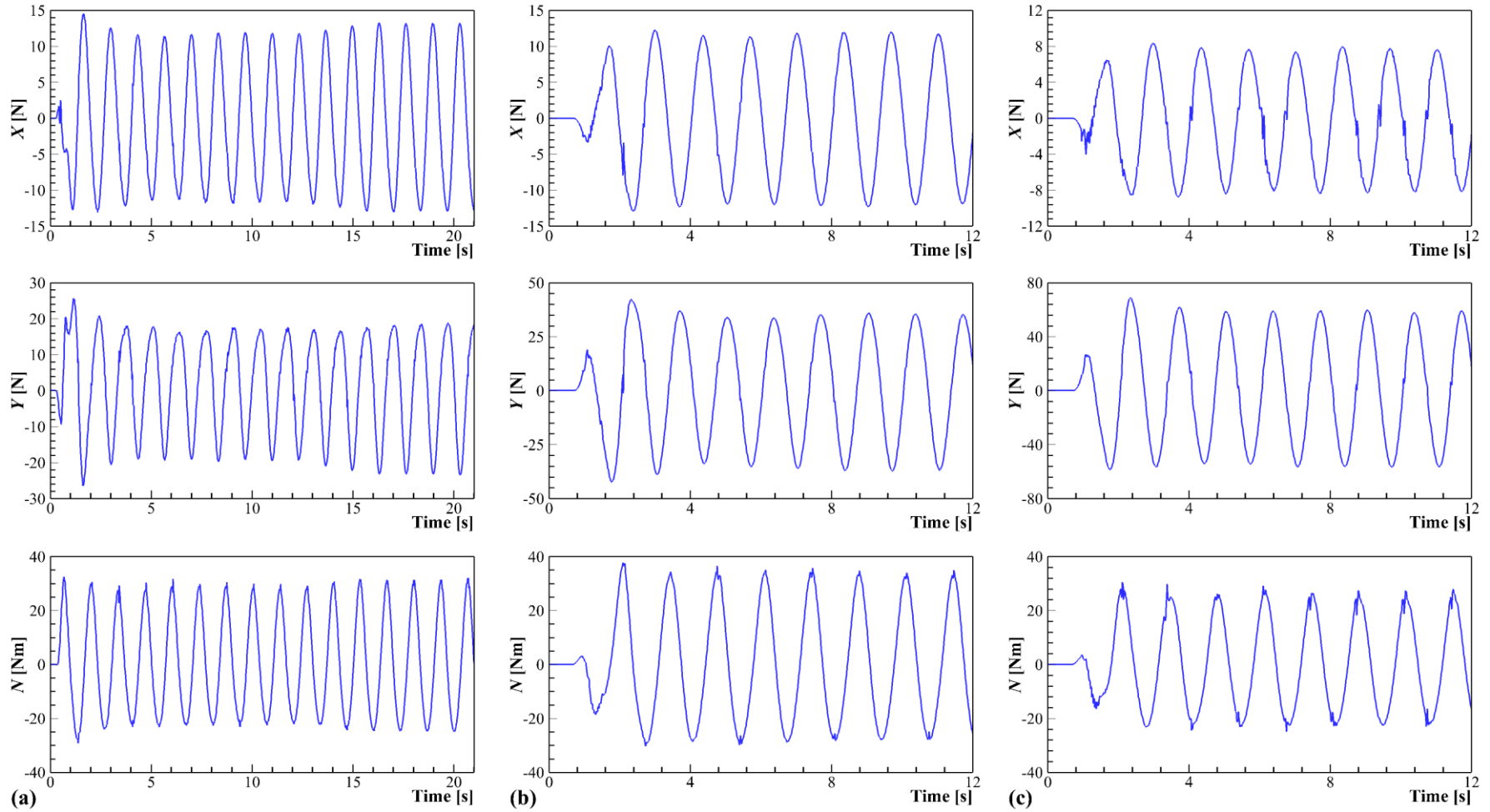


Figure F.3 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$.

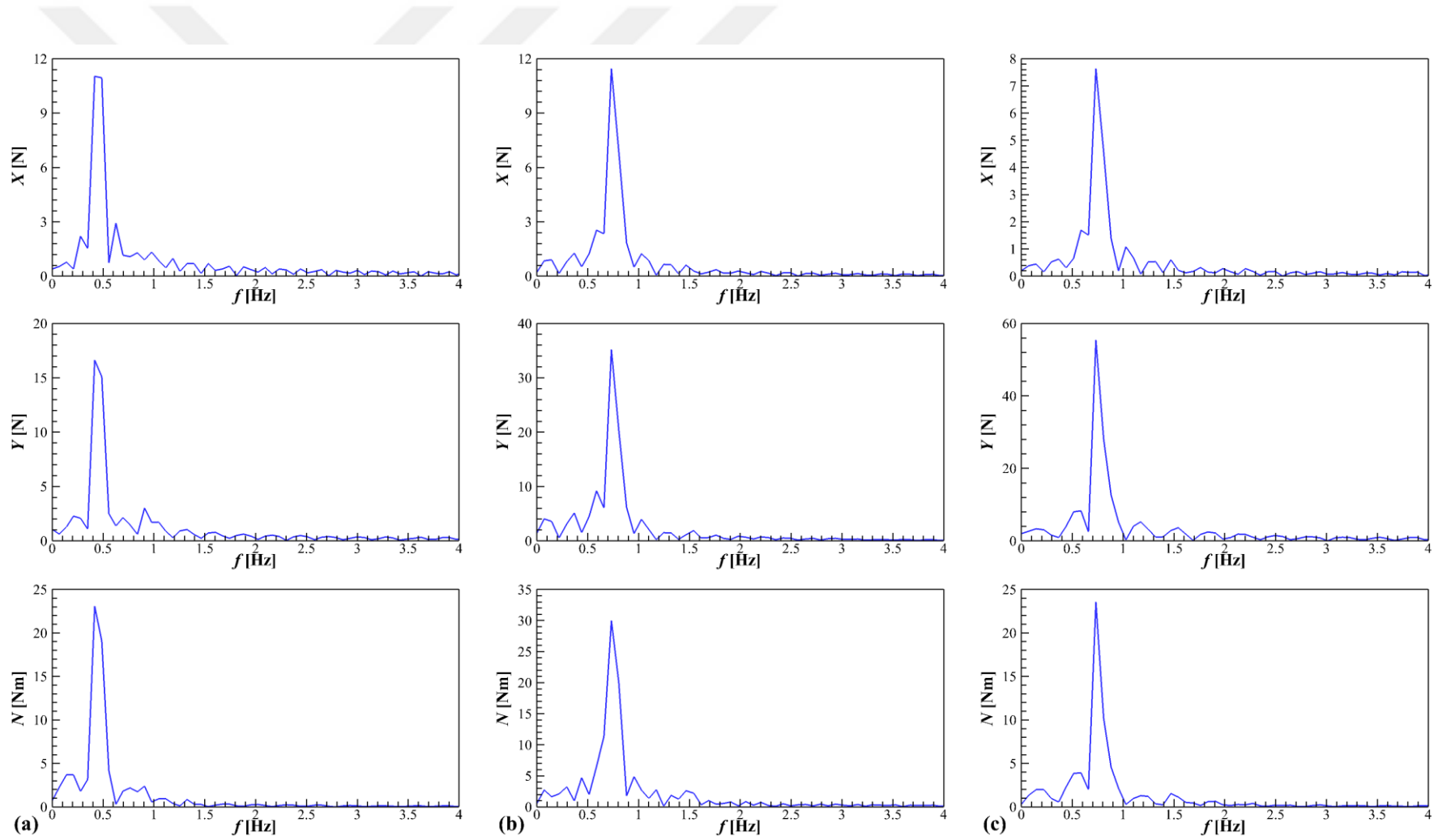


Figure F.4 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 45^\circ$, (b) $\psi = 60^\circ$, and (c) $\psi = 75^\circ$.

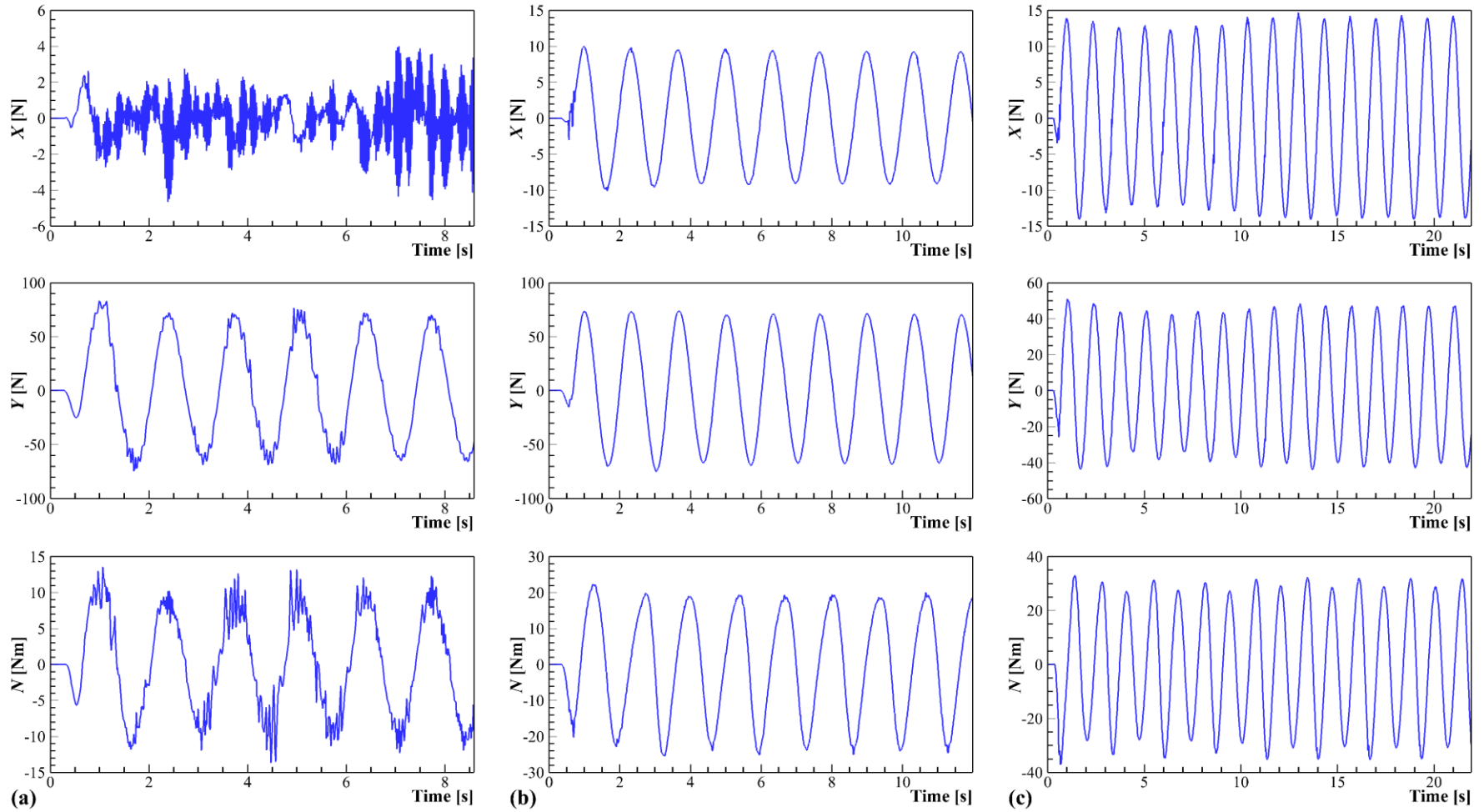


Figure F.5 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.

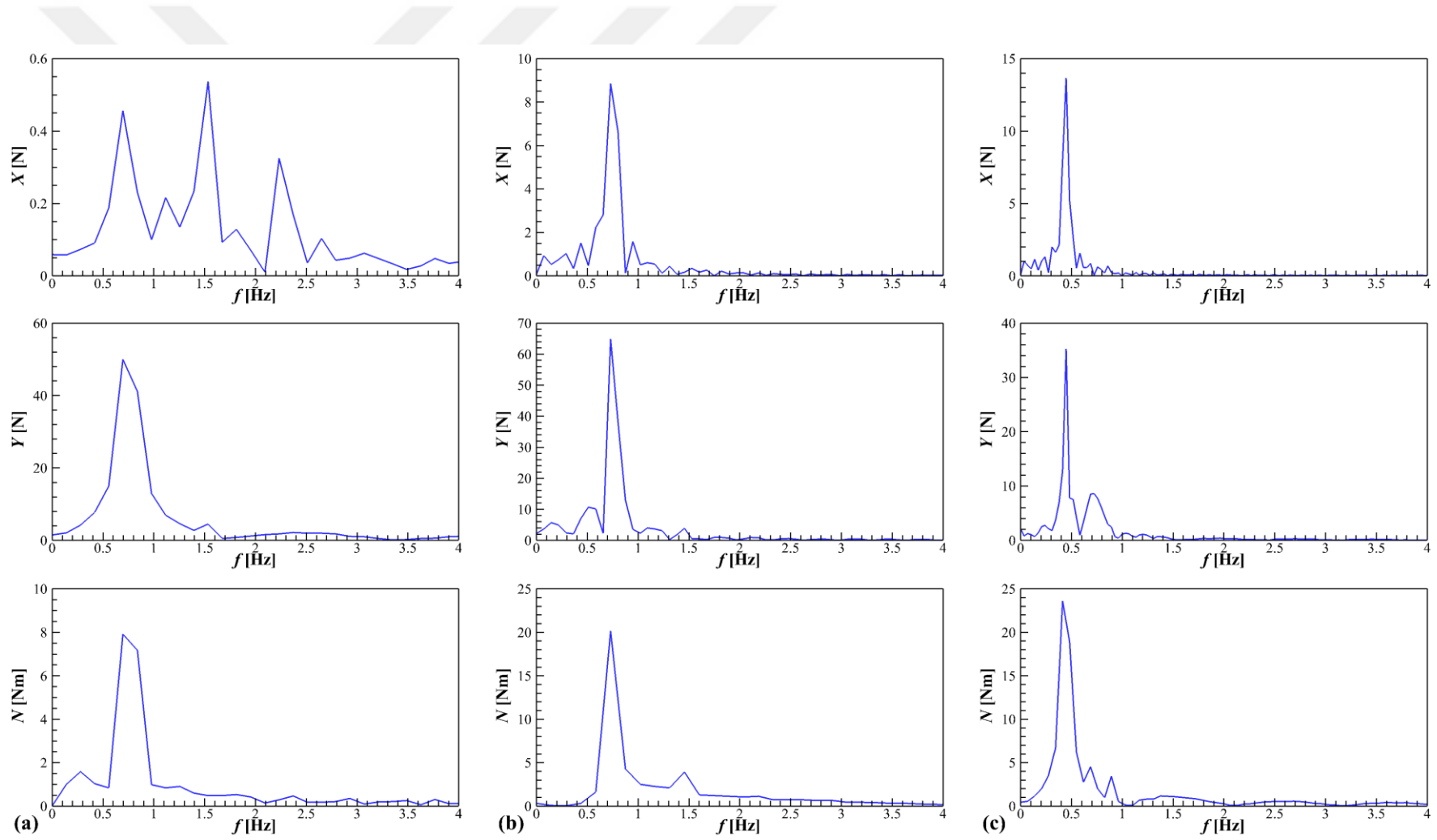


Figure F.6 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 90^\circ$, (b) $\psi = 105^\circ$, and (c) $\psi = 120^\circ$.

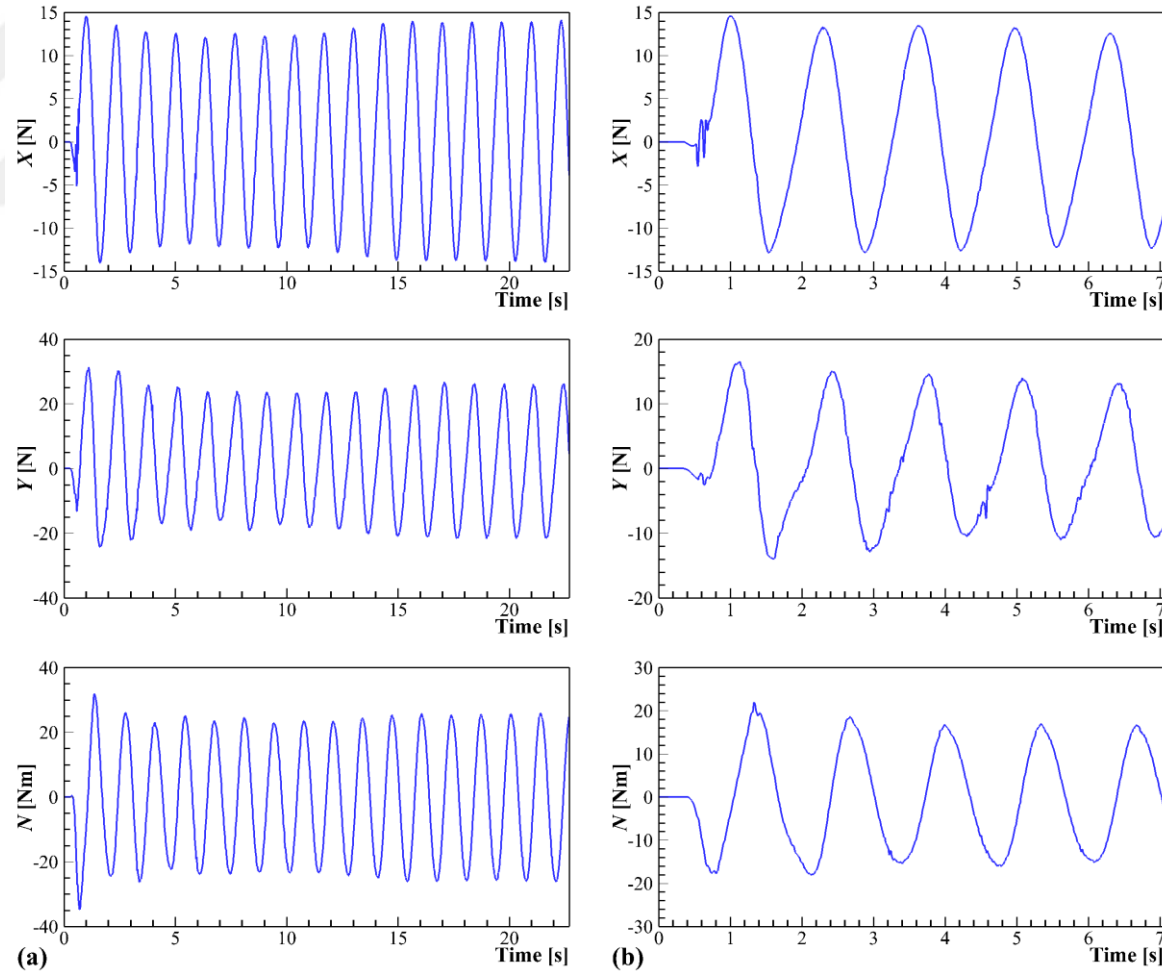


Figure F.7 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 135^\circ$, (b) $\psi = 150^\circ$.

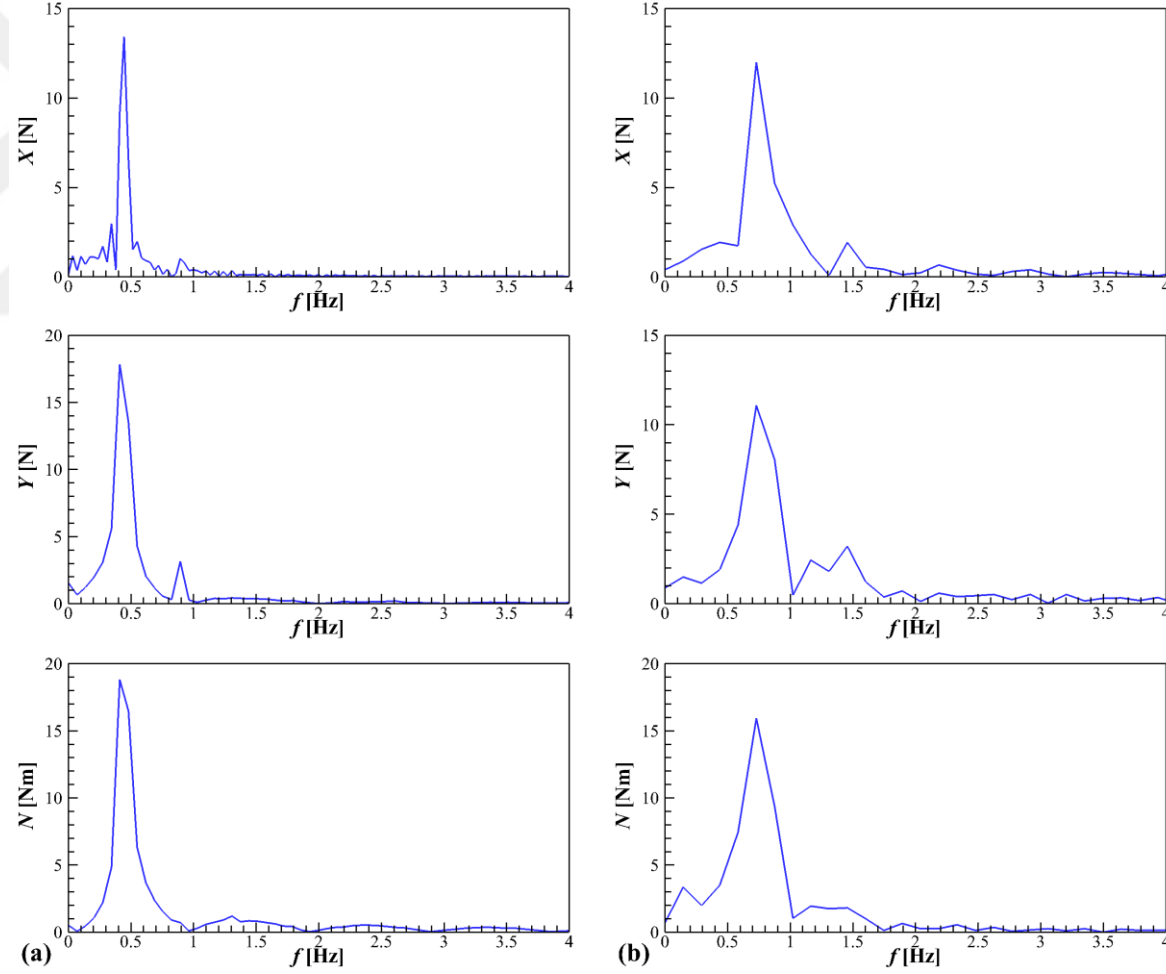


Figure F.8 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 135^\circ$, (b) $\psi = 150^\circ$.

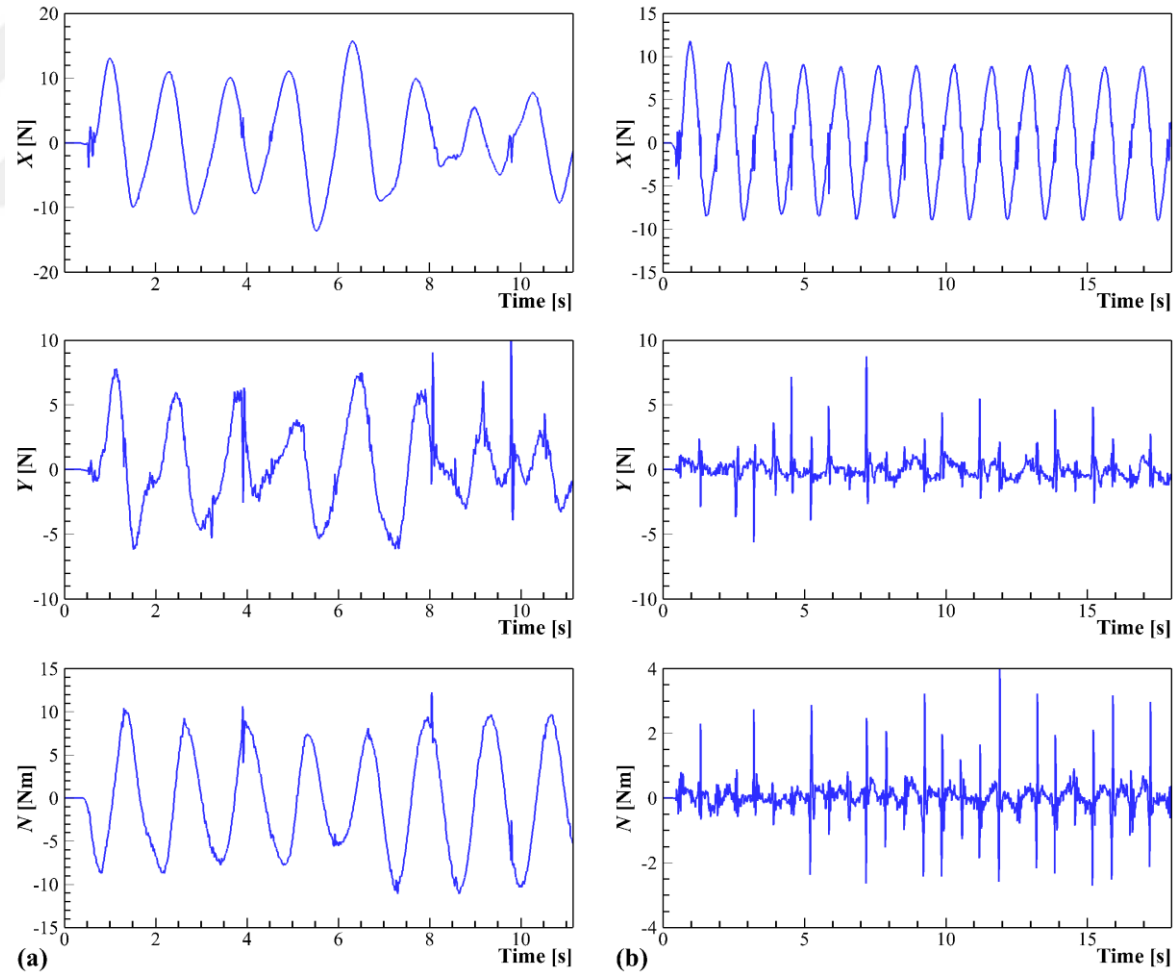


Figure F.9 : Time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 165^\circ$, (b) $\psi = 180^\circ$.

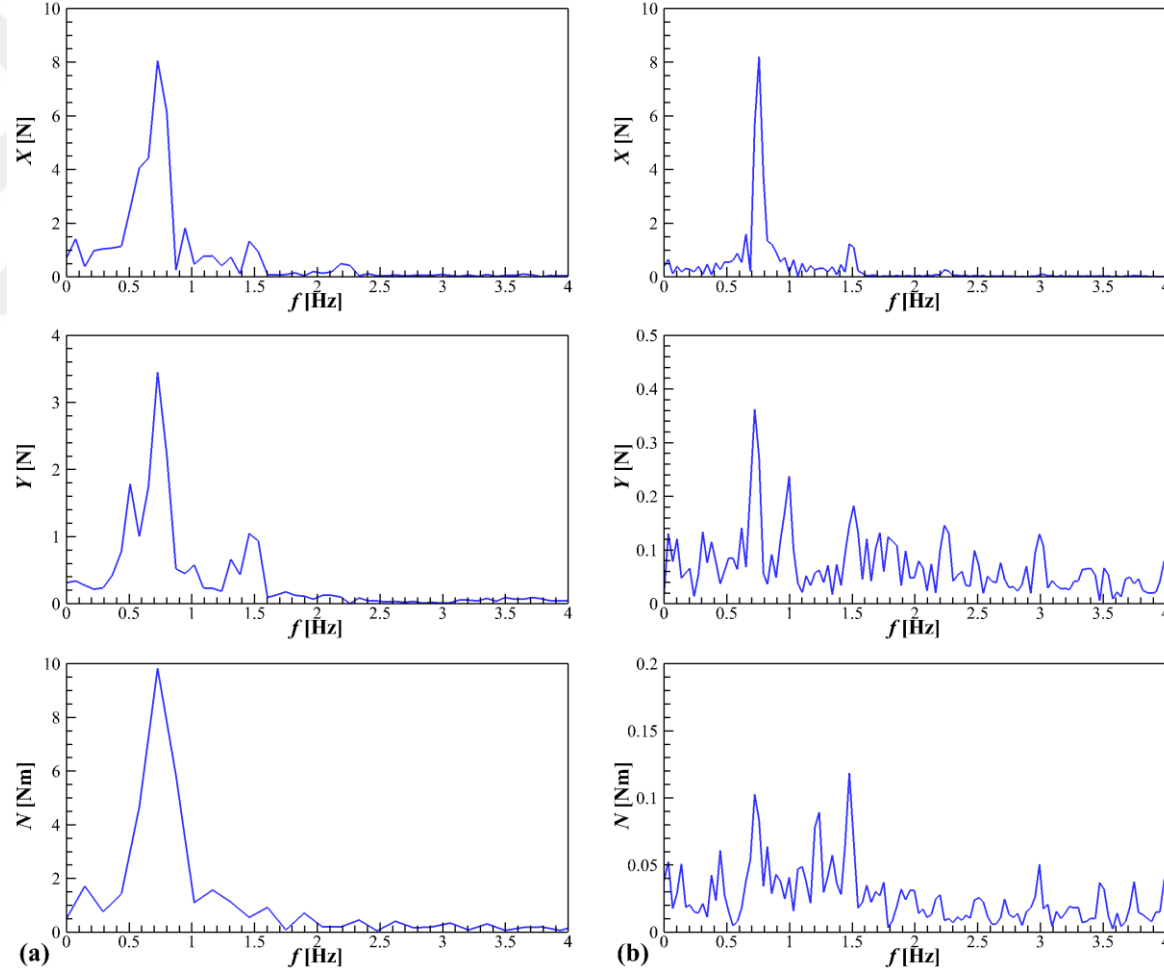


Figure F.10 : FFT analysis of time histories of computational X , Y forces and N moment at $H/\lambda = 0.020$ and $\lambda/L_{WL} = 1.00$: (a) $\psi = 165^\circ$, (b) $\psi = 180^\circ$.

APPENDIX G Turning trajectories and time histories in calm water

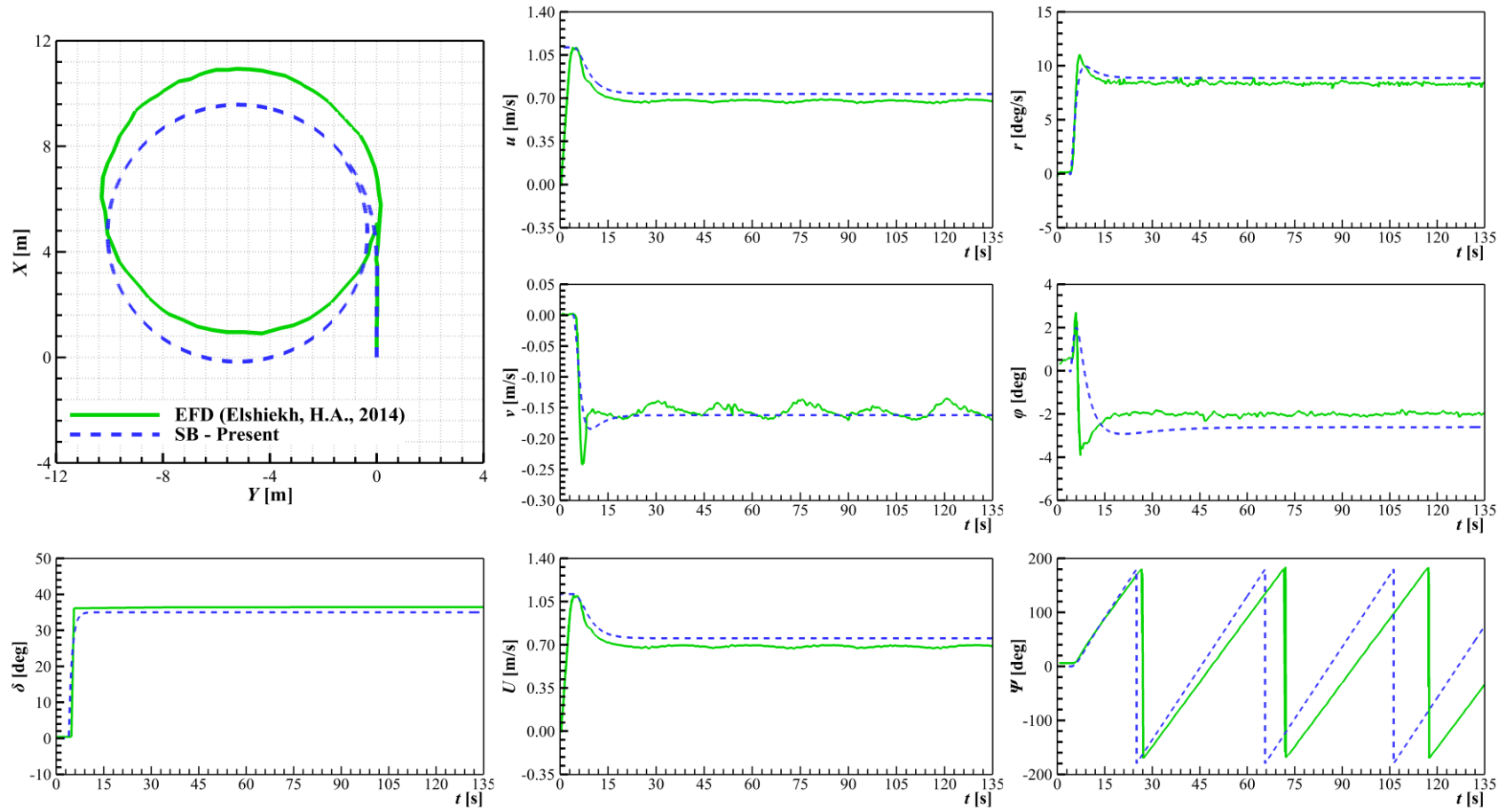


Figure G.1 : Comparison of turning trajectories and time histories in calm water, $\delta = 35^\circ$ and $\psi = 90^\circ$.

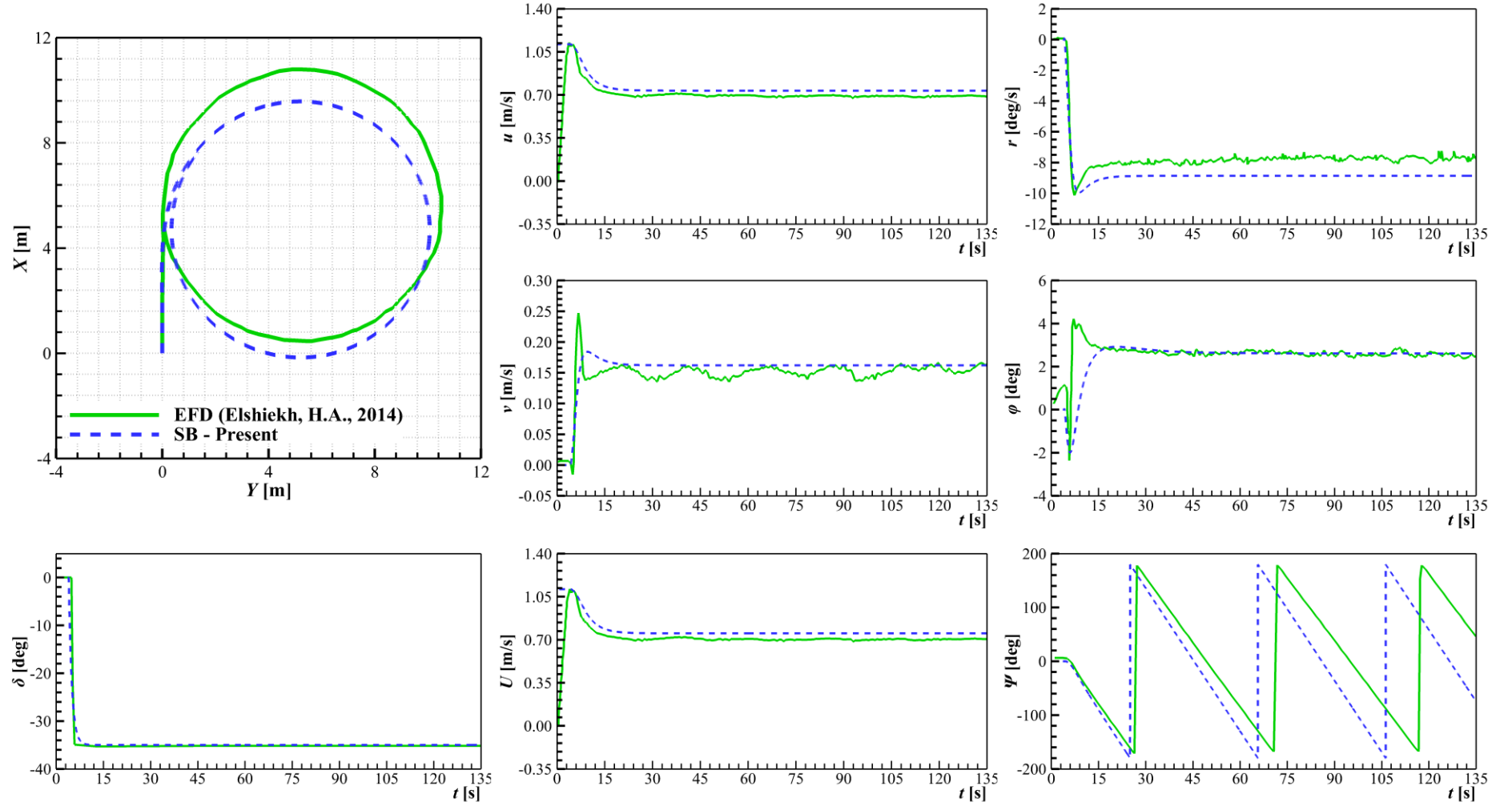


Figure G.2 : Comparison of turning trajectories and time histories in calm water, $\delta = -35^\circ$ and $\psi = 90^\circ$.

Table G.1 : Comparison of trajectory parameters in calm water, $\delta = \pm 35^\circ$ and $\psi = 90^\circ$.

Parameter	EFD (Elshiekh, 2014)	S-B – Present	Error [%]	EFD (Elshiekh, 2014)	S-B – Present	Error [%]
δ [deg]		+35			-35	
ψ [deg]		90			90	
Ad [m]	7.4301	8.3475	12.35	7.3969	8.3483	12.86
Tr [m]	4.0007	4.1245	3.09	4.2110	4.1246	2.05
T_{90}	15.3333	15.0166	2.07	15.2167	15.0138	1.33
TD [m]	10.1117	9.9557	1.54	10.2815	9.9595	3.13
STD [m]	10.0727	9.7244	3.46	10.3043	9.7257	5.62
T_{180}	26.4833	25.050	5.41	26.2333	25.0643	4.46
SYR [deg/s]	7.8986	8.8637	12.22	7.9466	8.8626	11.53
STS [m/s]	0.7081	0.7521	6.21	0.7203	0.7522	4.43

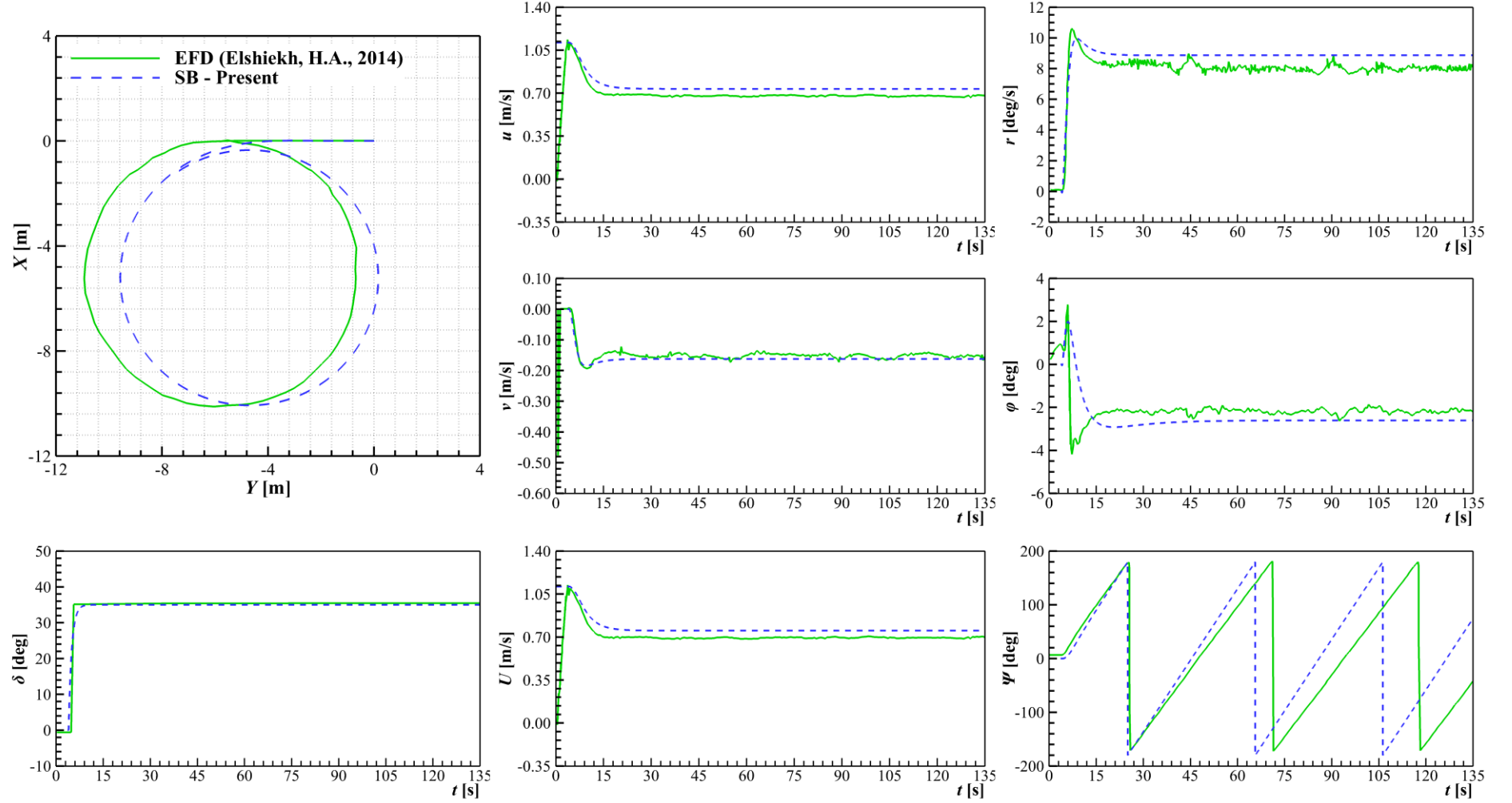


Figure G.3 : Comparison of turning trajectories and time histories in calm water, $\delta = 35^\circ$ and $\psi = 180^\circ$.

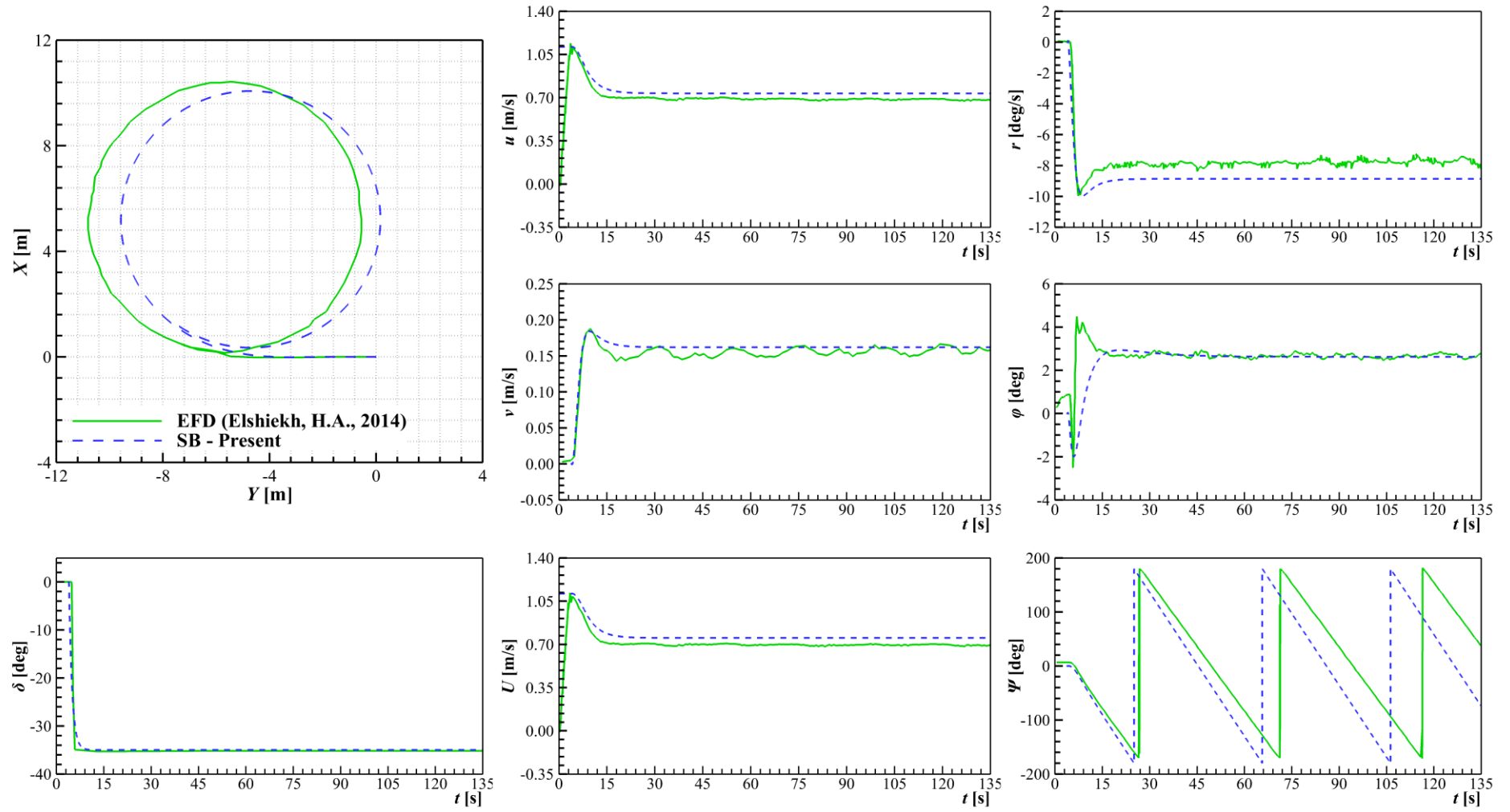


Figure G.4 : Comparison of turning trajectories and time histories in calm water, $\delta = -35^\circ$ and $\psi = 180^\circ$.

Table G.2 : Comparison of trajectory parameters in calm water, $\delta = \pm 35^\circ$ and $\psi = 180^\circ$.

Parameter	EFD (Elshiekh, 2014)	S-B – Present	Error [%]	EFD (Elshiekh, 2014)	S-B – Present	Error [%]
δ [deg]		+35			-35	
ψ [deg]		180			180	
Ad [m]	7.4301	8.3475	12.35	7.4723	8.3483	11.72
Tr [m]	3.9527	4.1245	4.35	4.1152	4.1246	0.23
T_{90}	15.2000	15.0127	1.23	15.2333	15.0219	1.39
TD [m]	9.9533	9.9557	0.02	10.0979	9.9595	1.37
STD [m]	10.1499	9.7435	4.00	10.1113	9.7257	3.81
T_{180}	26.3167	25.0600	4.78	26.3000	25.0360	4.81
SYR [deg/s]	7.9034	8.8637	12.15	8.0088	8.8626	10.66
STS [m/s]	0.7055	0.7521	6.61	0.7159	0.7522	5.07

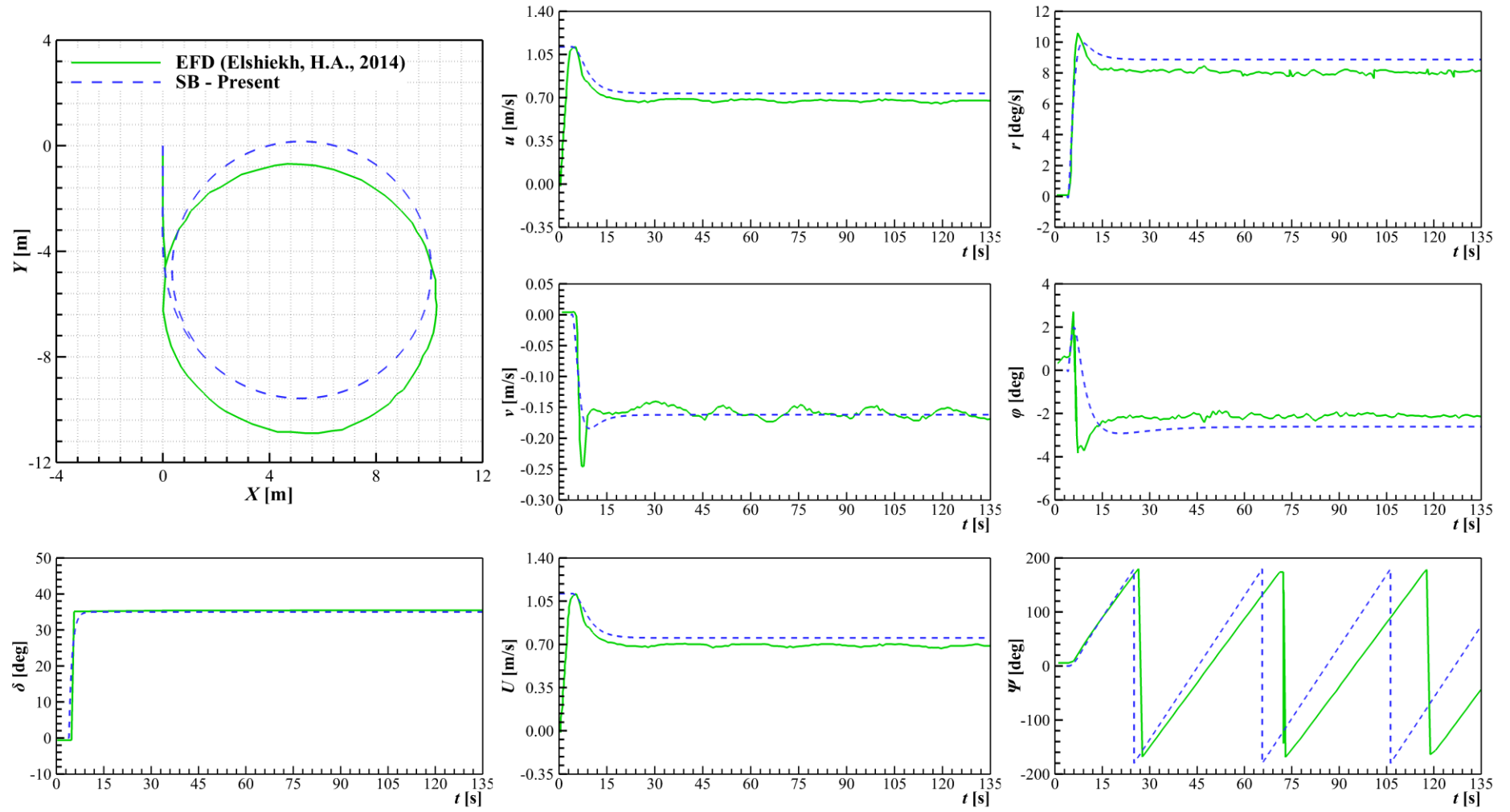


Figure G.5 : Comparison of turning trajectories and time histories in calm water, $\delta = 35^\circ$ and $\psi = -90^\circ$.

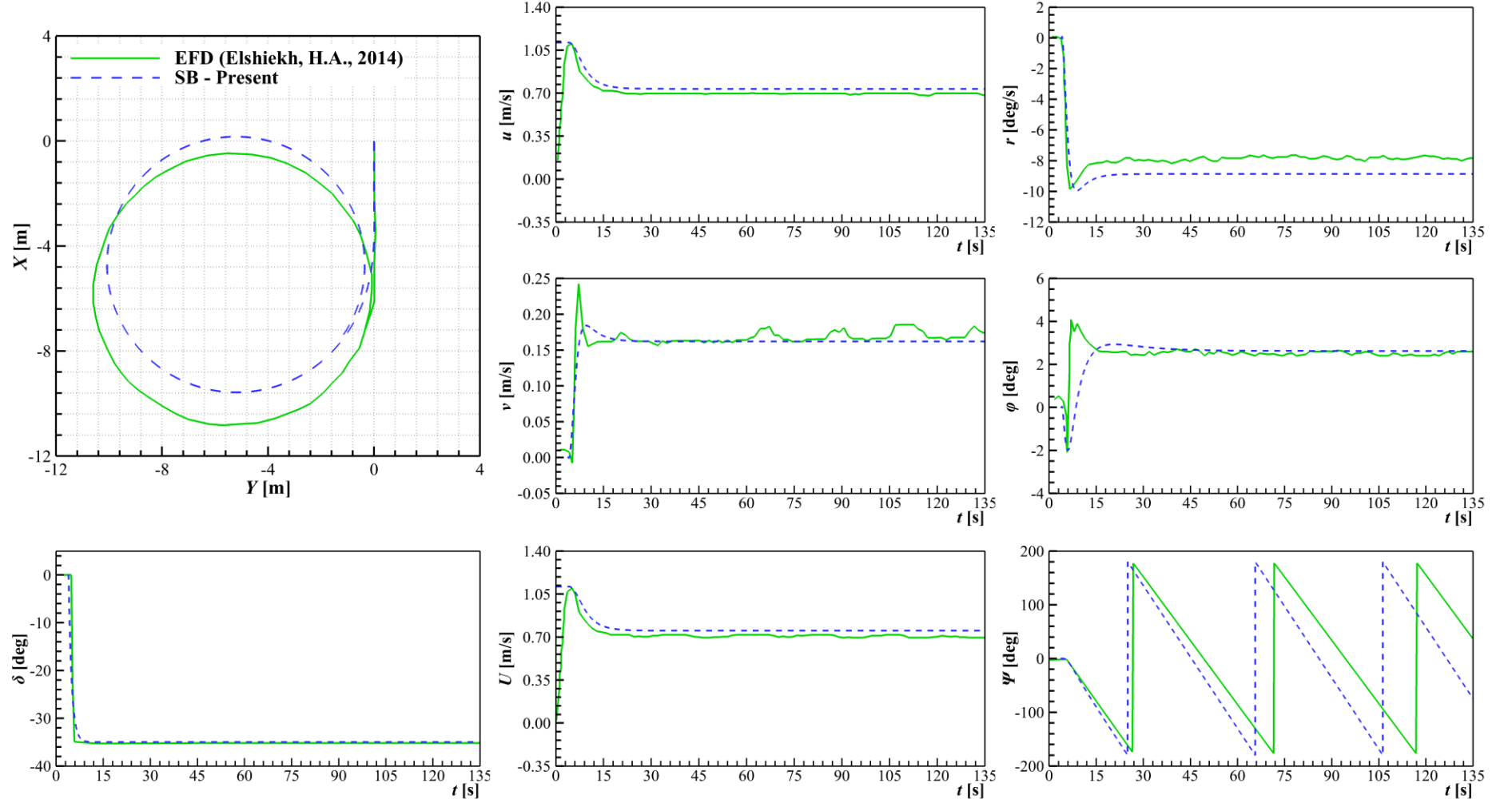


Figure G.6 : Comparison of turning trajectories and time histories in calm water, $\delta = -35^\circ$ and $\psi = -90^\circ$.

Table G.3 : Comparison of trajectory parameters in calm water, $\delta = \pm 35^\circ$ and $\psi = -90^\circ$.

Parameter	EFD (Elshiekh, 2014)	S-B – Present	Error [%]	EFD (Elshiekh, 2014)	S-B – Present	Error [%]
δ [deg]		+35			-35	
ψ [deg]		-90			-90	
Ad [m]	7.4293	8.3475	12.36	7.4806	8.3483	11.60
Tr [m]	4.0822	4.1245	1.04	4.1486	4.1246	0.58
T_{90}	15.3000	15.0122	1.88	15.3167	15.0158	1.96
TD [m]	10.1721	9.9557	2.13	10.3365	9.9595	3.65
STD [m]	10.1885	9.7230	4.57	10.7288	9.7270	9.34
T_{180}	26.4333	26.5402	0.40	26.4000	25.050	5.11
SYR [deg/s]	7.8986	8.8637	12.22	7.9779	8.8626	11.09
STS [m/s]	0.7081	0.7521	6.21	0.7264	0.7522	3.55



APPENDIX H Turning trajectories and time histories in calm water for varying rudder angles

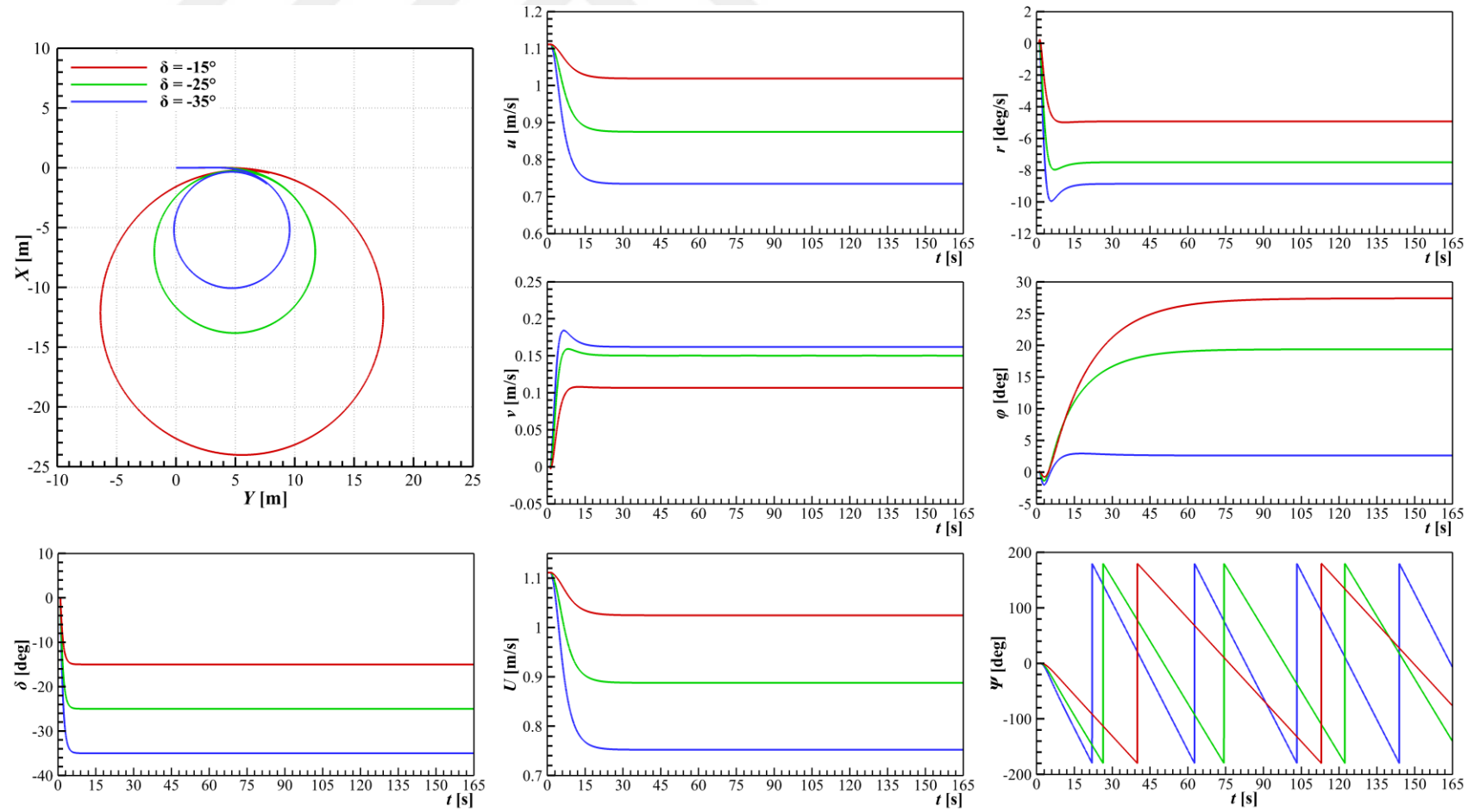


Figure H.1 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = -15^\circ, -25^\circ, -35^\circ$ and $\psi = 0^\circ$.

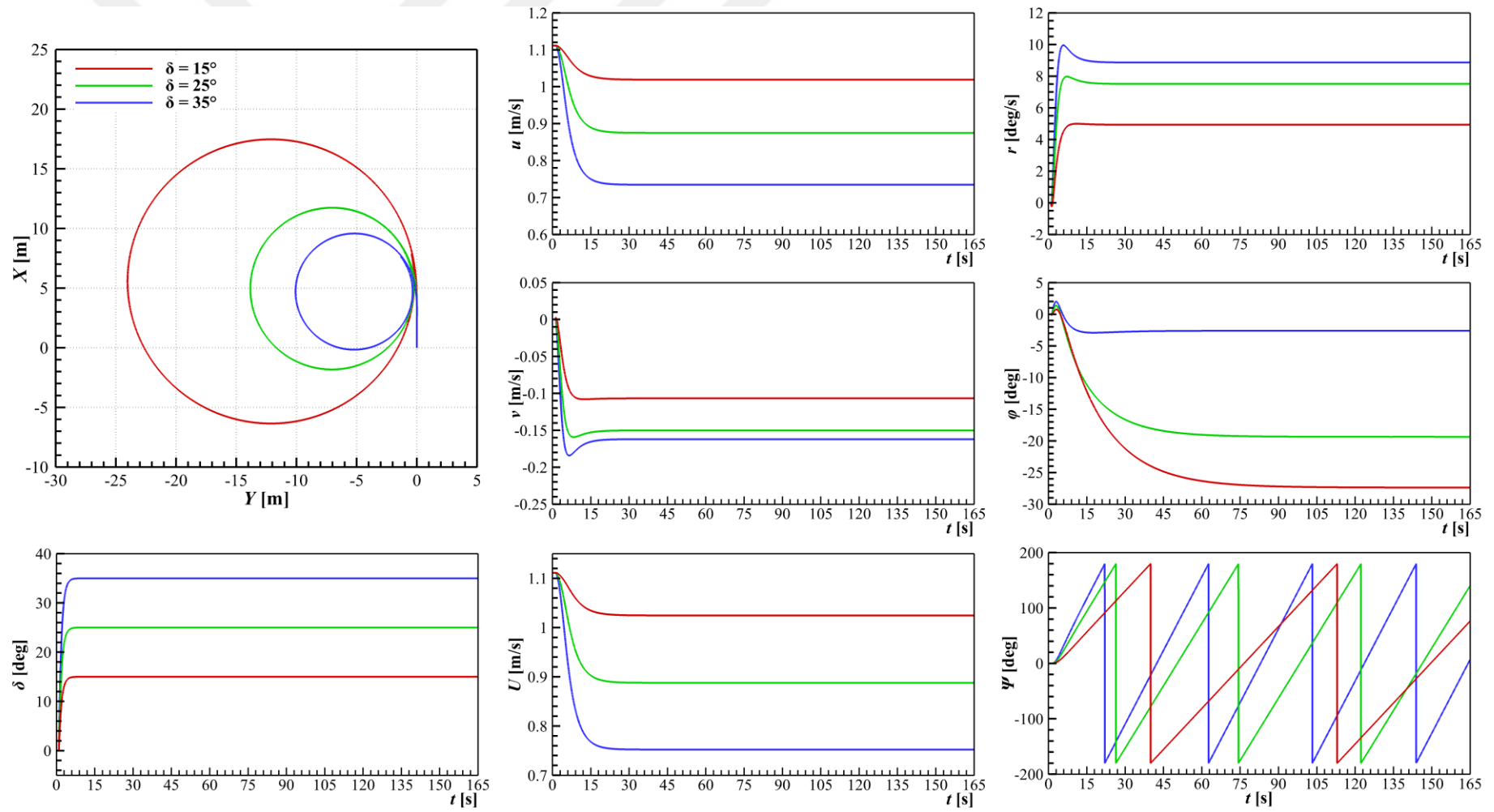


Figure H.2 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = 15^\circ, 25^\circ, 35^\circ$ and $\psi = 90^\circ$.

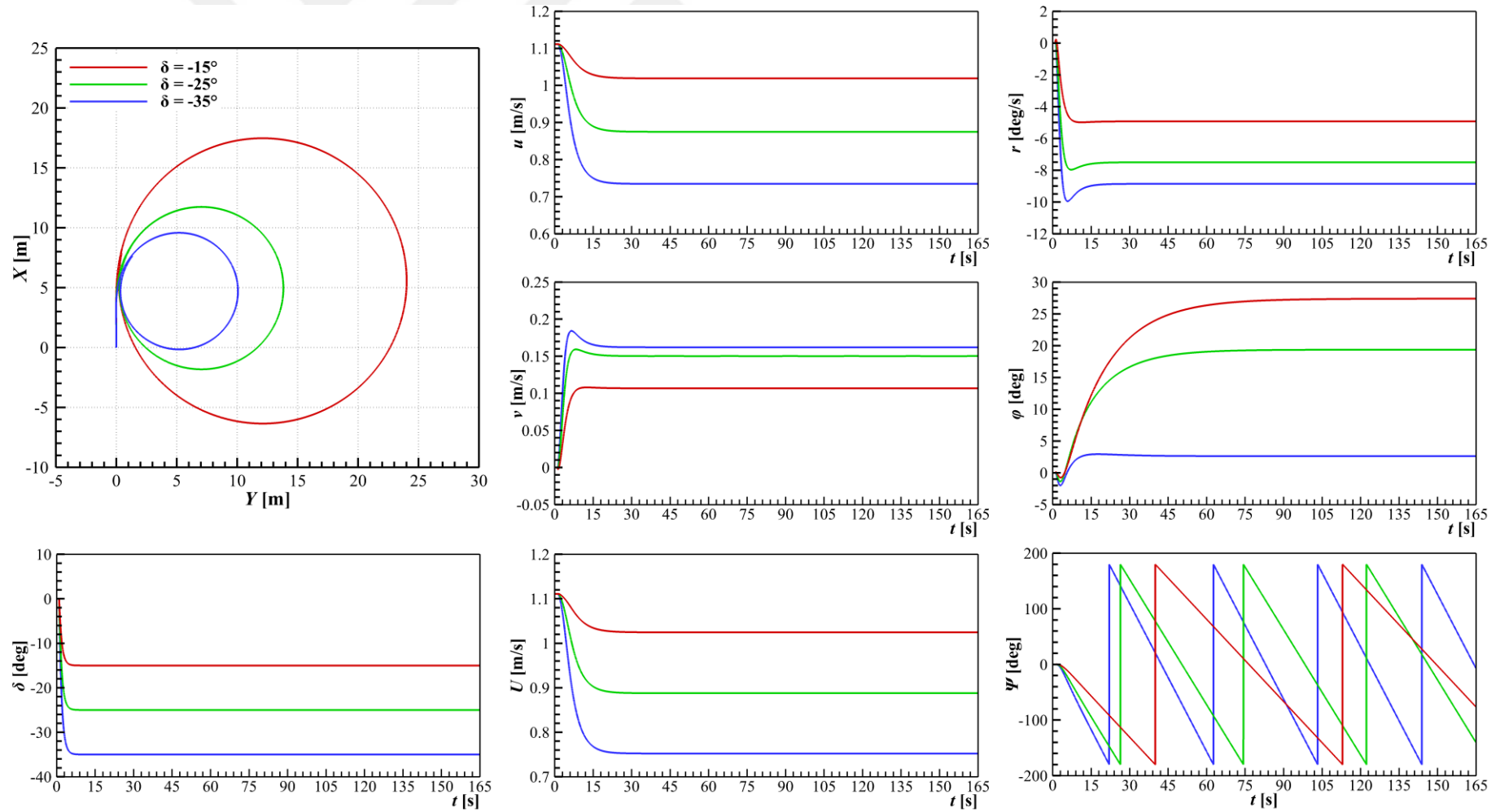


Figure H.3 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = -15^\circ, -25^\circ, -35^\circ$ and $\psi = 90^\circ$.

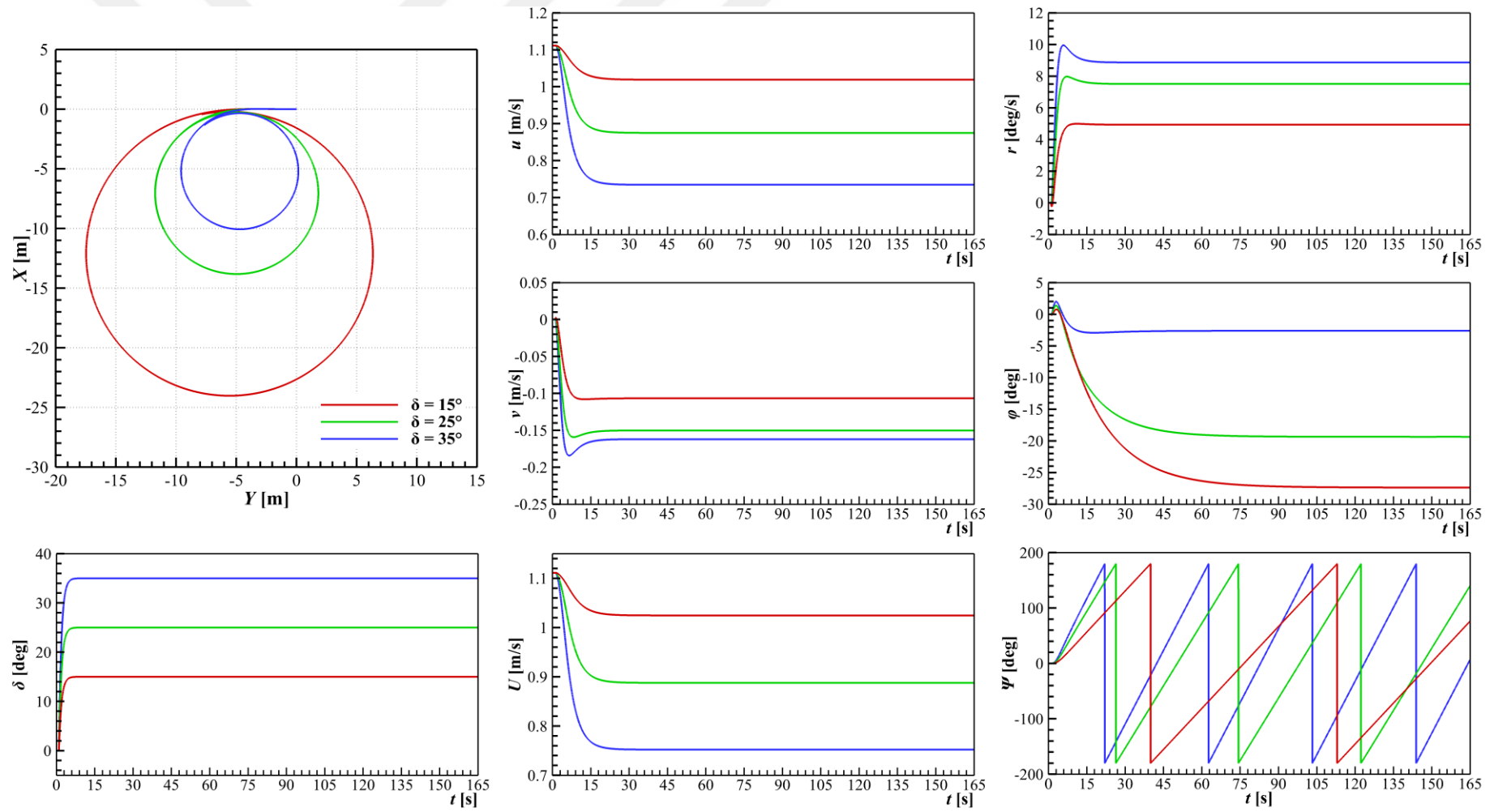


Figure H.4 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = 15^\circ, 25^\circ, 35^\circ$ and $\psi = 180^\circ$.

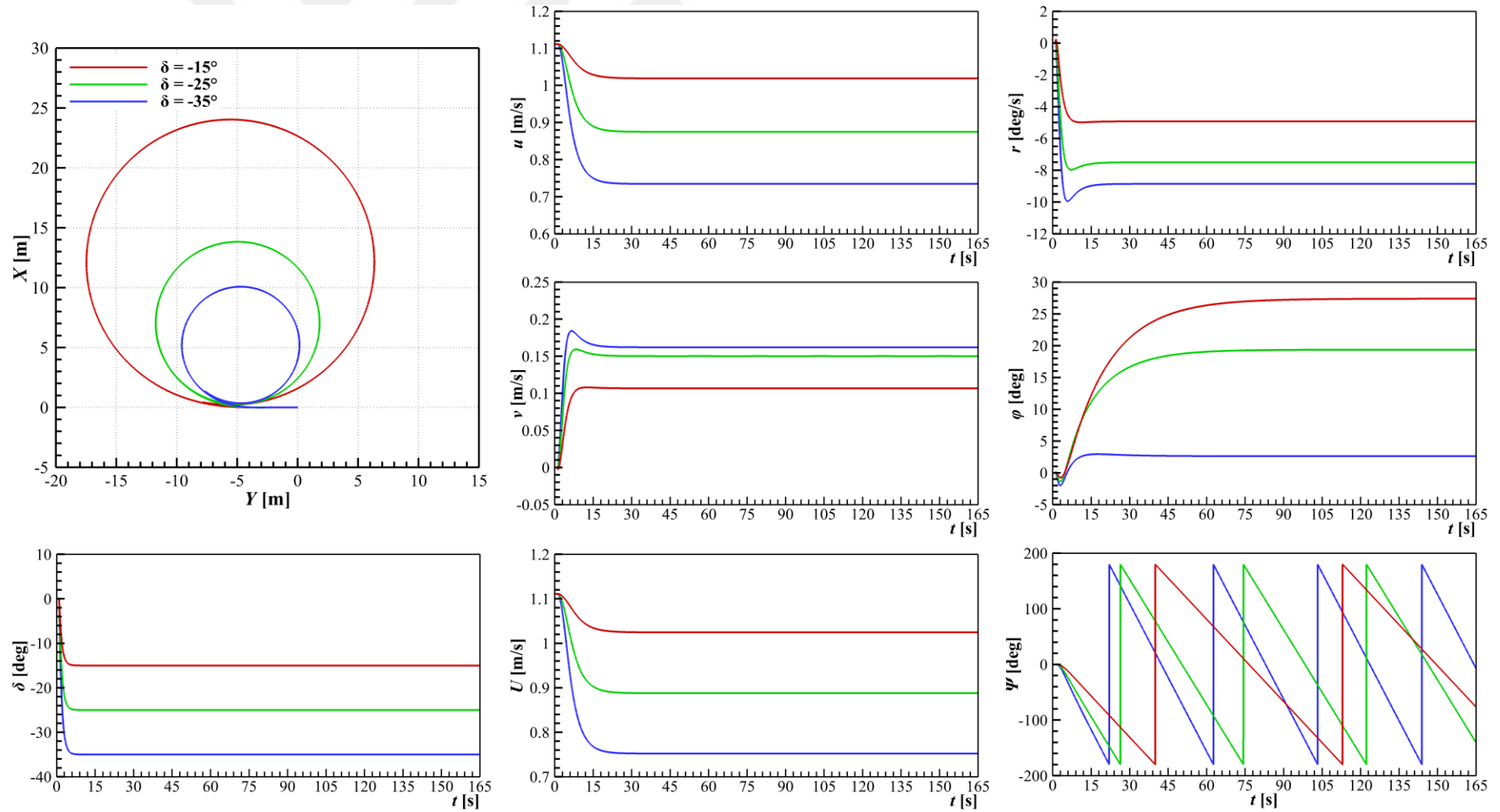


Figure H.5 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = -15^\circ, -25^\circ, -35^\circ$ and $\psi = 180^\circ$.

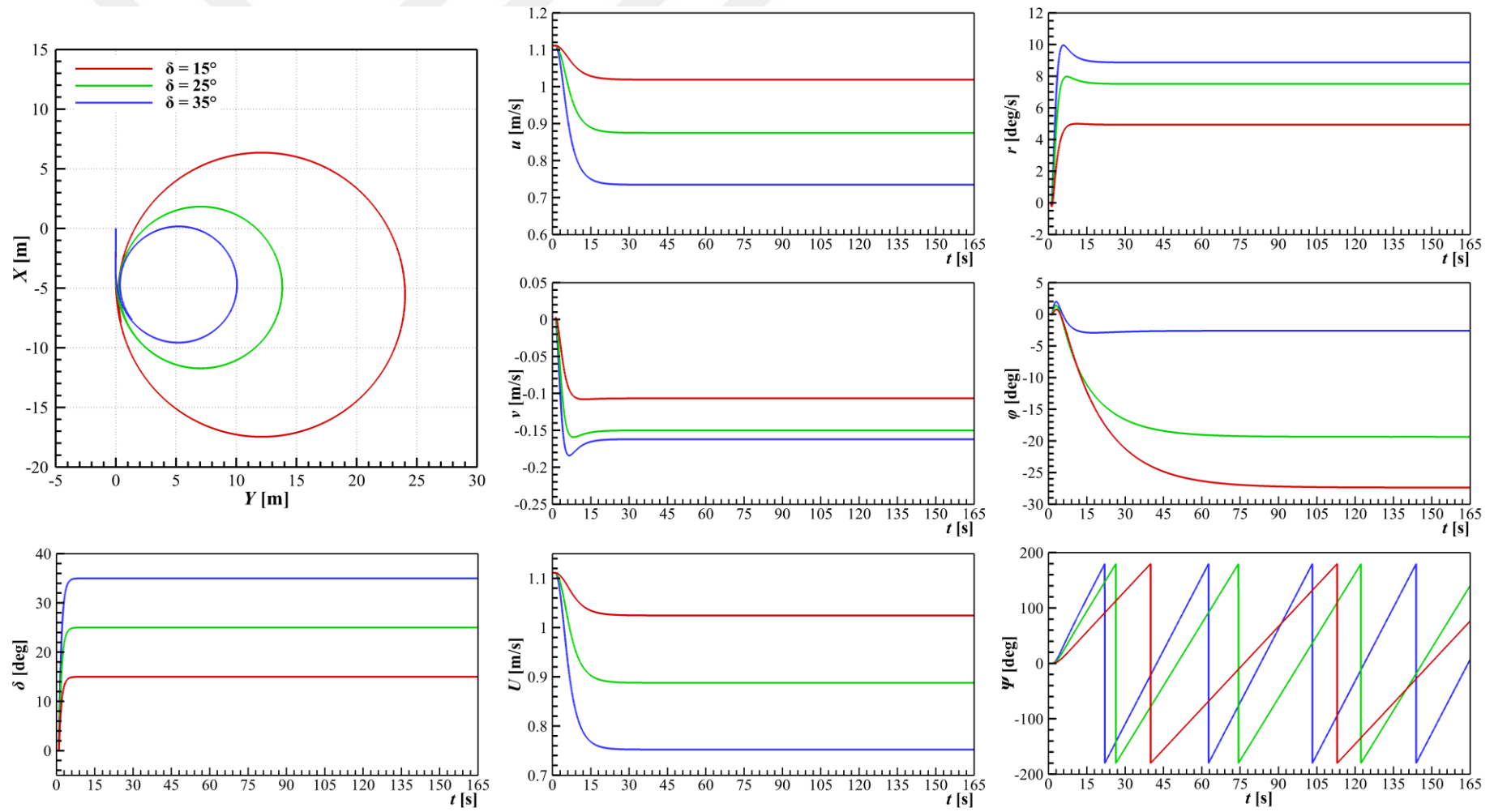


Figure H.6 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = 15^\circ, 25^\circ, 35^\circ$ and $\psi = -90^\circ$.

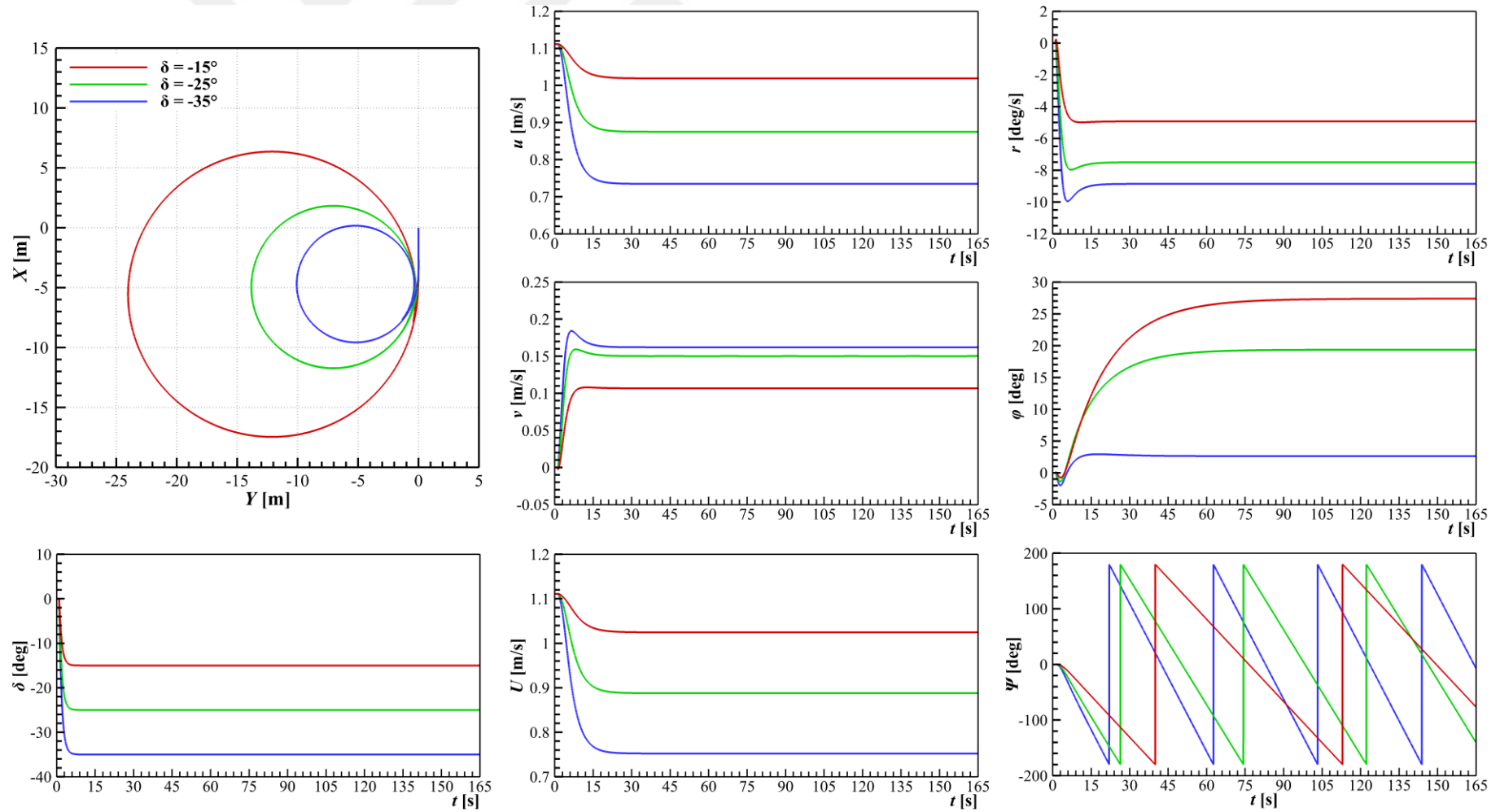


Figure H.7 : Comparison of turning trajectories and time histories in calm water for varying rudder angles, $\delta = -15^\circ, -25^\circ, -35^\circ$ and $\psi = -90^\circ$.



APPENDIX I Turning trajectories and time histories in waves

Table I.1 : Comparison of trajectory parameters in regular waves, $\delta = 35^\circ$ and $\chi = 90^\circ$.

	H/λ	λ/L_{WL}	Ad [m]	Tr [m]	TD [m]	STD [m]	SYR [deg/s]	STS [m/s]
EFD (Elshiekh, 2014)			7.7655	4.3372	10.1347	N/A	7.7334	0.6895
S-B – Present	0.020	1.00	8.5982	4.4203	10.2101	9.6029	8.8481	0.7532
Error [%]			-10.72	-1.92	-0.74	N/A	-14.41	-9.24
EFD (Elshiekh, 2014)			7.5370	4.1282	9.9090	N/A	7.9471	0.7072
S-B – Present	0.020	0.50	8.4724	4.2945	10.1838	9.5881	8.8631	0.7524
Error [%]			-12.41	-4.03	-2.77	N/A	-11.53	-6.39
S-B – Present	0.025	1.00	8.7523	4.5783	10.3523	9.5455	8.8134	0.7527

Table I.2 : Comparison of trajectory parameters in regular waves, $\delta = -35^\circ$ and $\chi = 90^\circ$.

	H/λ	λ/L_{WL}	Ad [m]	Tr [m]	TD [m]	STD [m]	SYR [deg/s]	STS [m/s]
EFD (Elshiekh, 2014)			7.1621	4.0334	10.1544	N/A	7.6718	0.6732
S-B – Present	0.020	1.00	8.3220	4.1965	9.9614	9.6045	8.8701	0.7510
Error [%]			-16.19	-4.04	1.90	N/A	-15.62	-11.56
EFD (Elshiekh, 2014)			7.3350	4.1250	10.3559	N/A	7.8660	0.7017
S-B – Present	0.020	0.50	8.3032	4.1394	9.8725	9.6037	8.8697	0.7533
Error [%]			-13.20	-0.35	4.67	N/A	-12.76	-7.35
S-B – Present	0.025	1.00	8.3095	4.2723	9.9919	9.5675	8.8439	0.7490

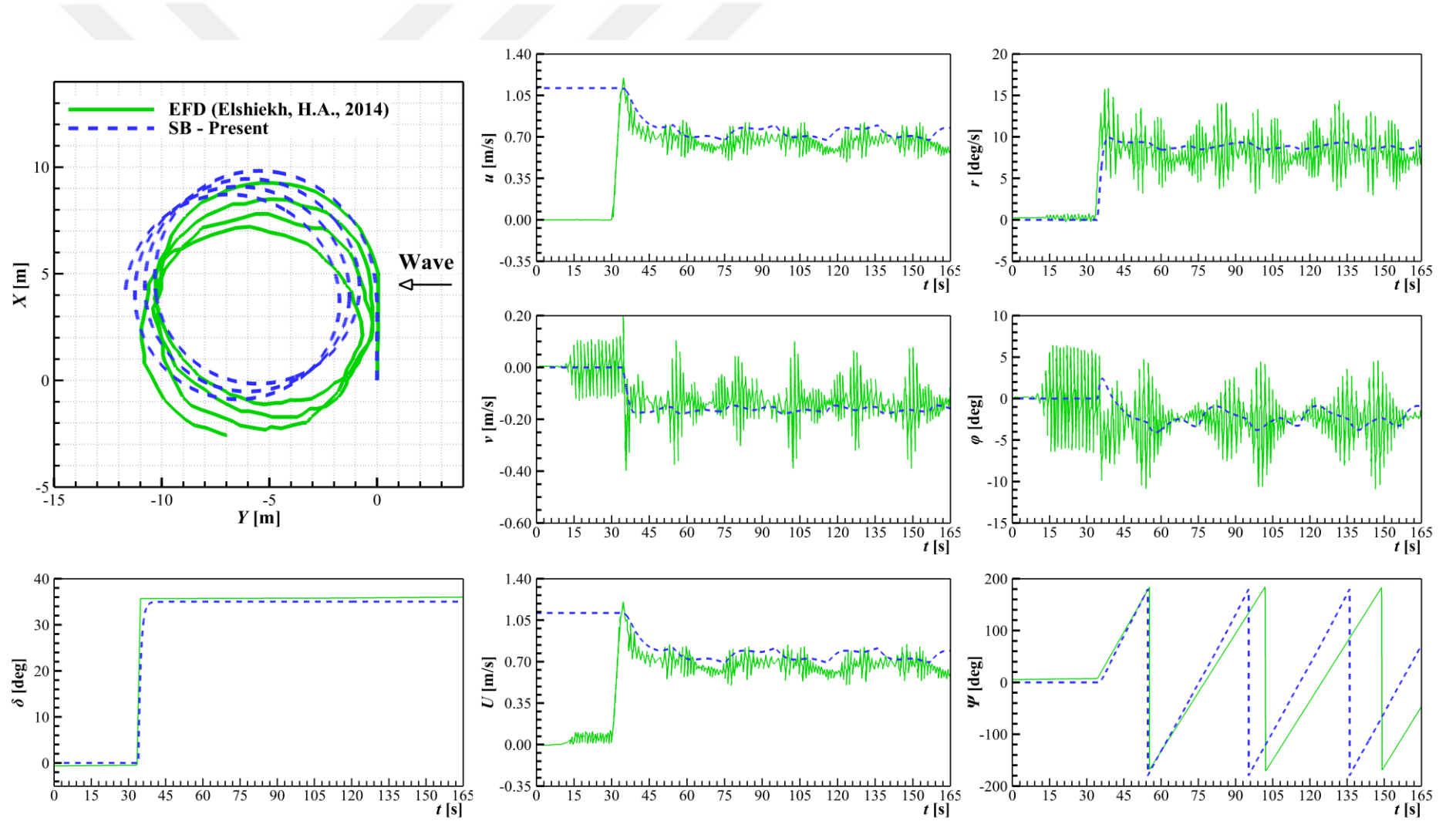


Figure I.1 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

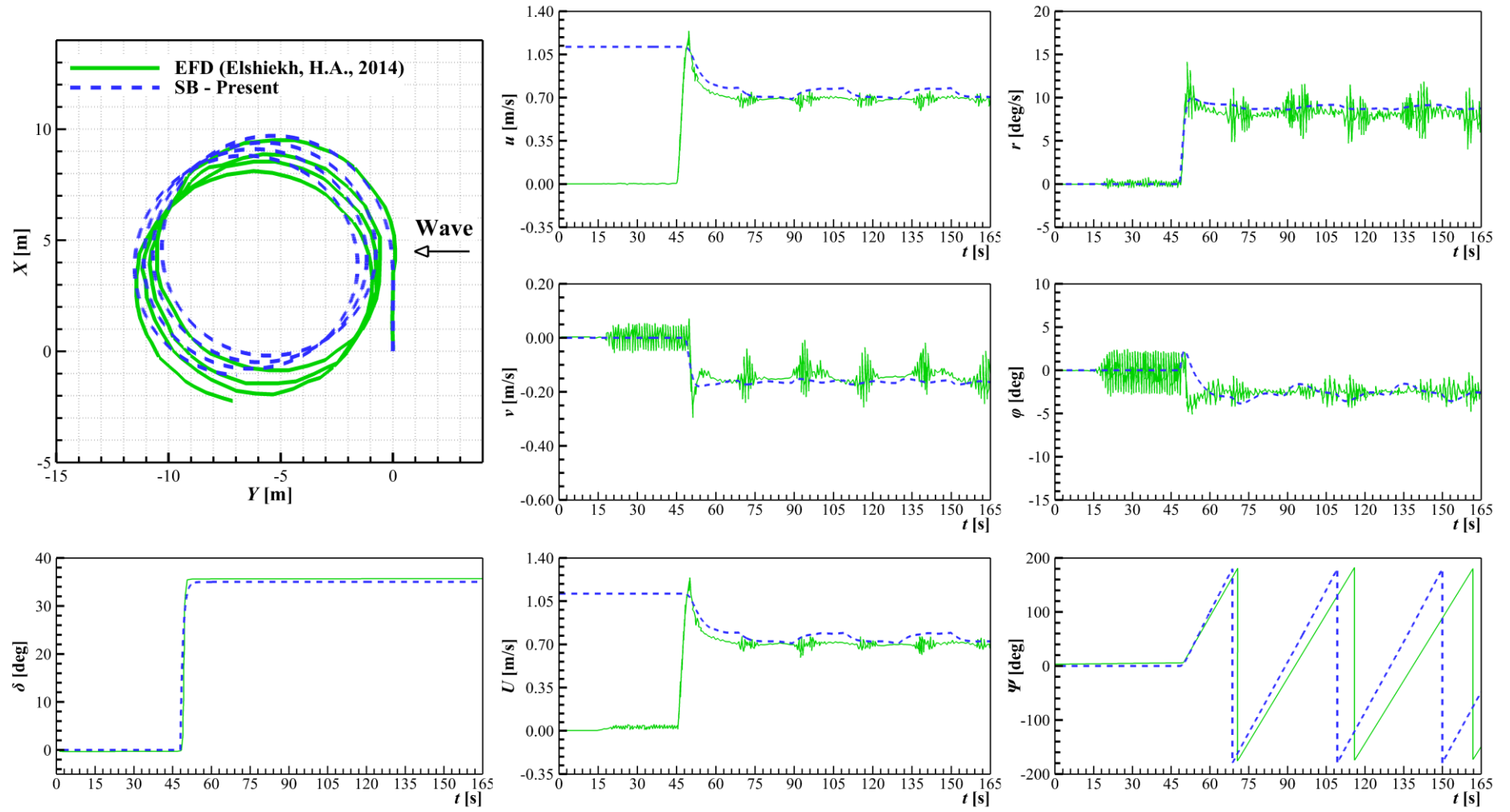


Figure I.2 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$).

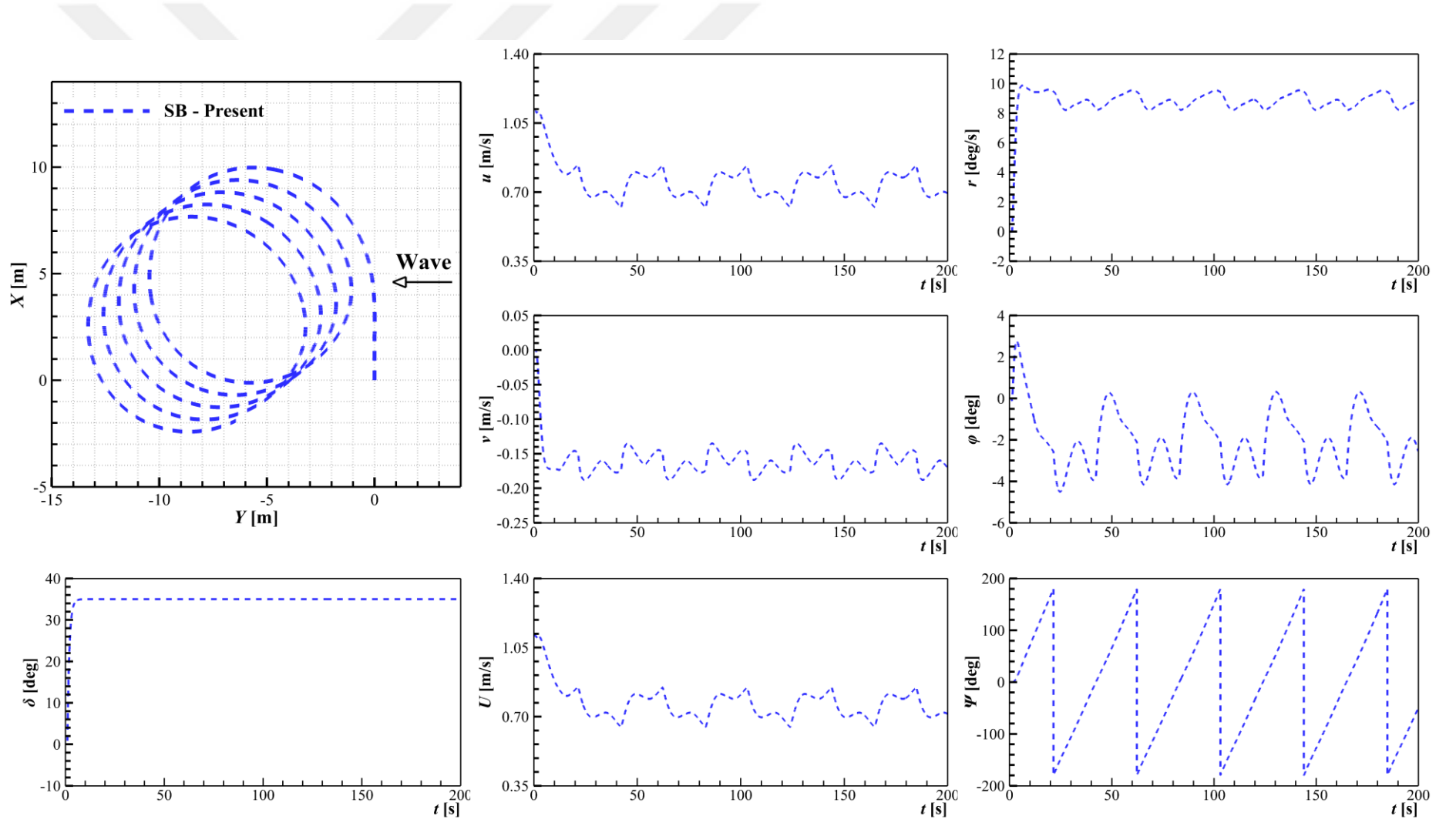


Figure I.3 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).

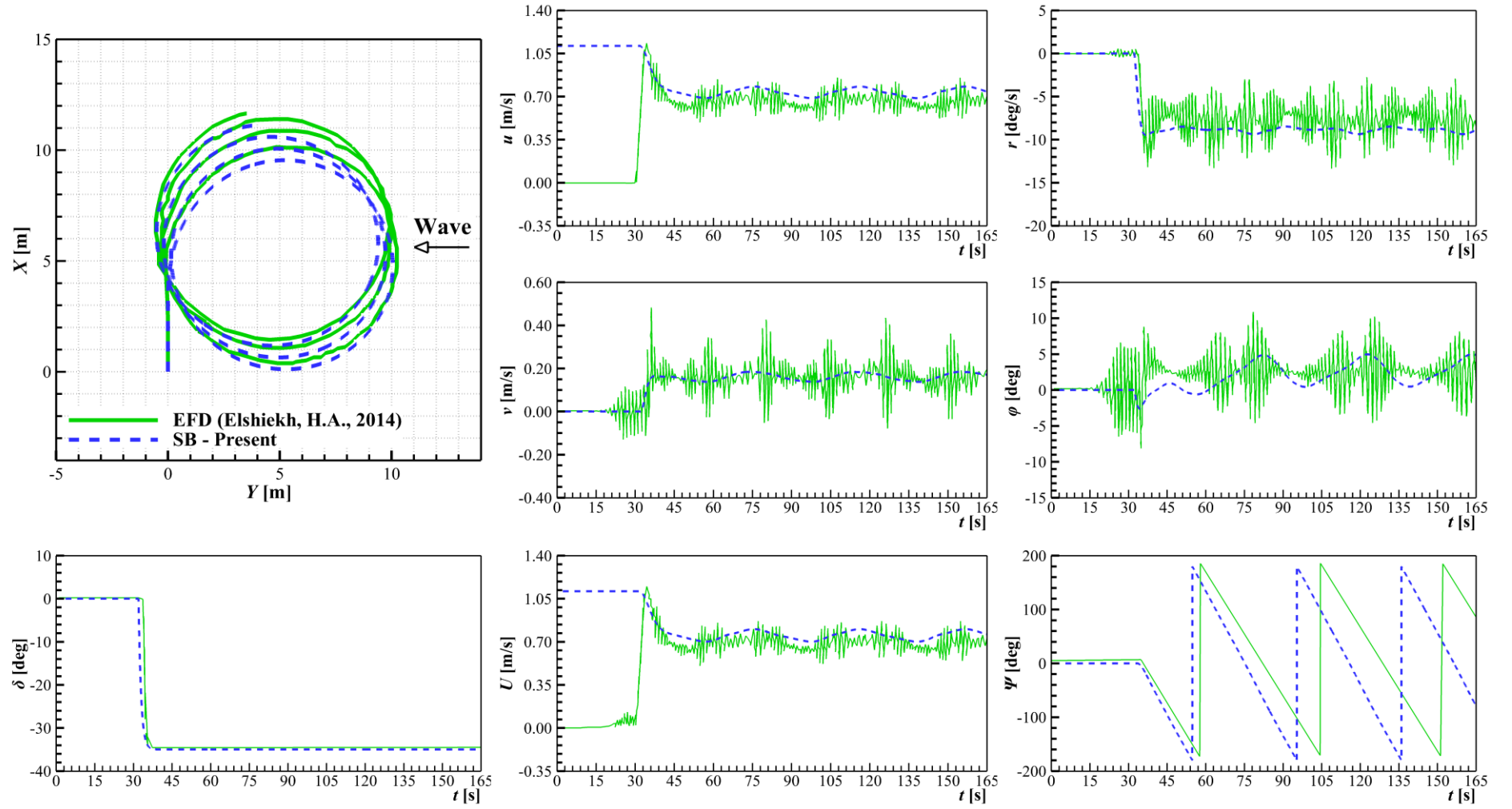


Figure I.4 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

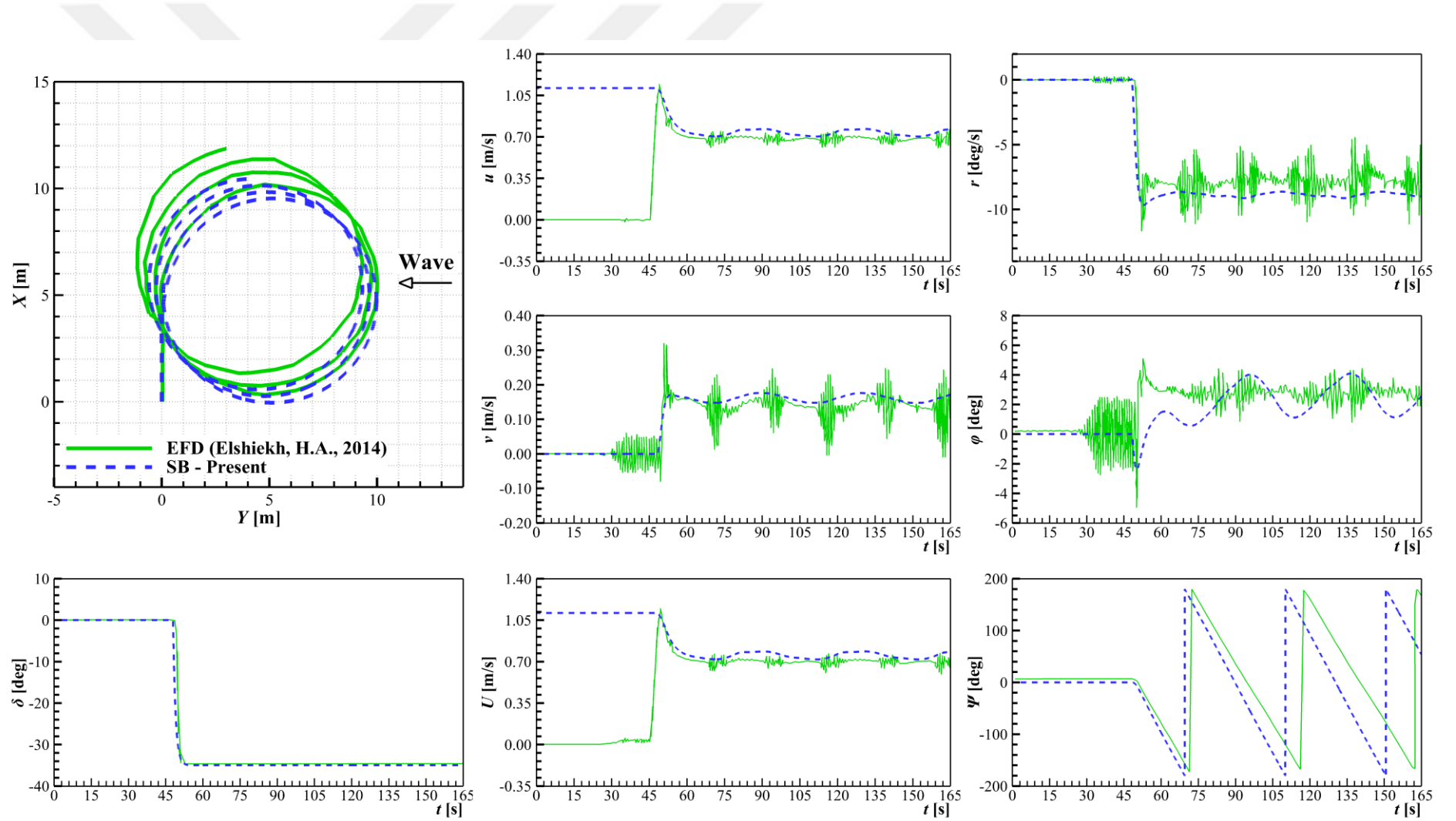


Figure I.5 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$).

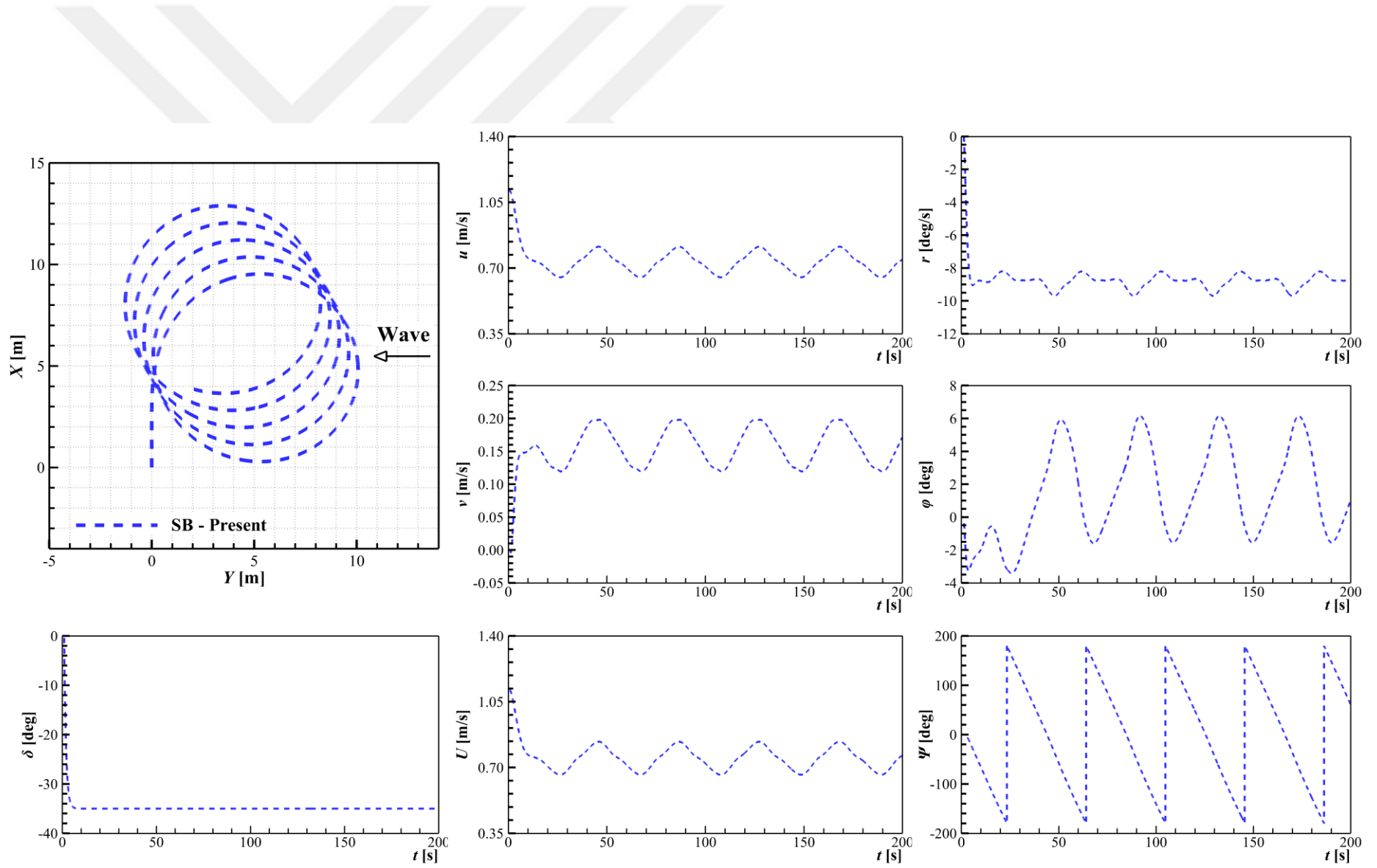


Figure I.6 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 90^\circ$ ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).

Table I.3 : Comparison of trajectory parameters in regular waves, $\delta = 35^\circ$ and $\chi = 0^\circ$.

	H/λ	λ/L_{WL}	Ad [m]	Tr [m]	TD [m]	STD [m]	SYR [deg/s]	STS [m/s]
EFD (Elshiekh, 2014)			7.6891	4.0705	10.2595	N/A	7.7248	0.6781
S-B – Present	0.020	1.00	8.2784	4.2158	10.1623	9.6226	8.8448	0.7522
Error [%]			-7.66	-3.57	0.95	N/A	-14.50	-10.93
EFD (Elshiekh, 2014)			7.6003	4.1330	10.3164	N/A	7.9927	0.7006
S-B – Present	0.020	0.50	8.3791	4.2186	10.0848	9.6083	8.8652	0.7537
Error [%]			-10.25	-2.07	2.24	N/A	-10.92	-7.58
S-B – Present	0.025	1.00	8.2456	4.2786	10.2687	9.5656	8.8142	0.7517

Table I.4 : Comparison of trajectory parameters in regular waves, $\delta = -35^\circ$ and $\chi = 0^\circ$.

	H/λ	λ/L_{WL}	Ad [m]	Tr [m]	TD [m]	STD [m]	SYR [deg/s]	STS [m/s]
EFD (Elshiekh, 2014)			7.7526	4.1252	10.3656	N/A	7.7586	0.6841
S-B – Present	0.020	1.00	8.1994	4.0445	9.7666	9.6045	8.8673	0.7517
Error [%]			-5.76	1.96	5.78	N/A	-14.29	-9.88
EFD (Elshiekh, 2014)			7.5949	4.2698	10.5633	N/A	7.9783	0.7072
S-B – Present	0.020	0.50	8.3459	4.1112	9.8537	9.6037	8.8674	0.7532
Error [%]			-9.89	3.71	6.72	N/A	-11.14	-6.50
S-B – Present	0.025	1.00	8.1115	3.9928	9.6207	9.5675	8.8389	0.7487

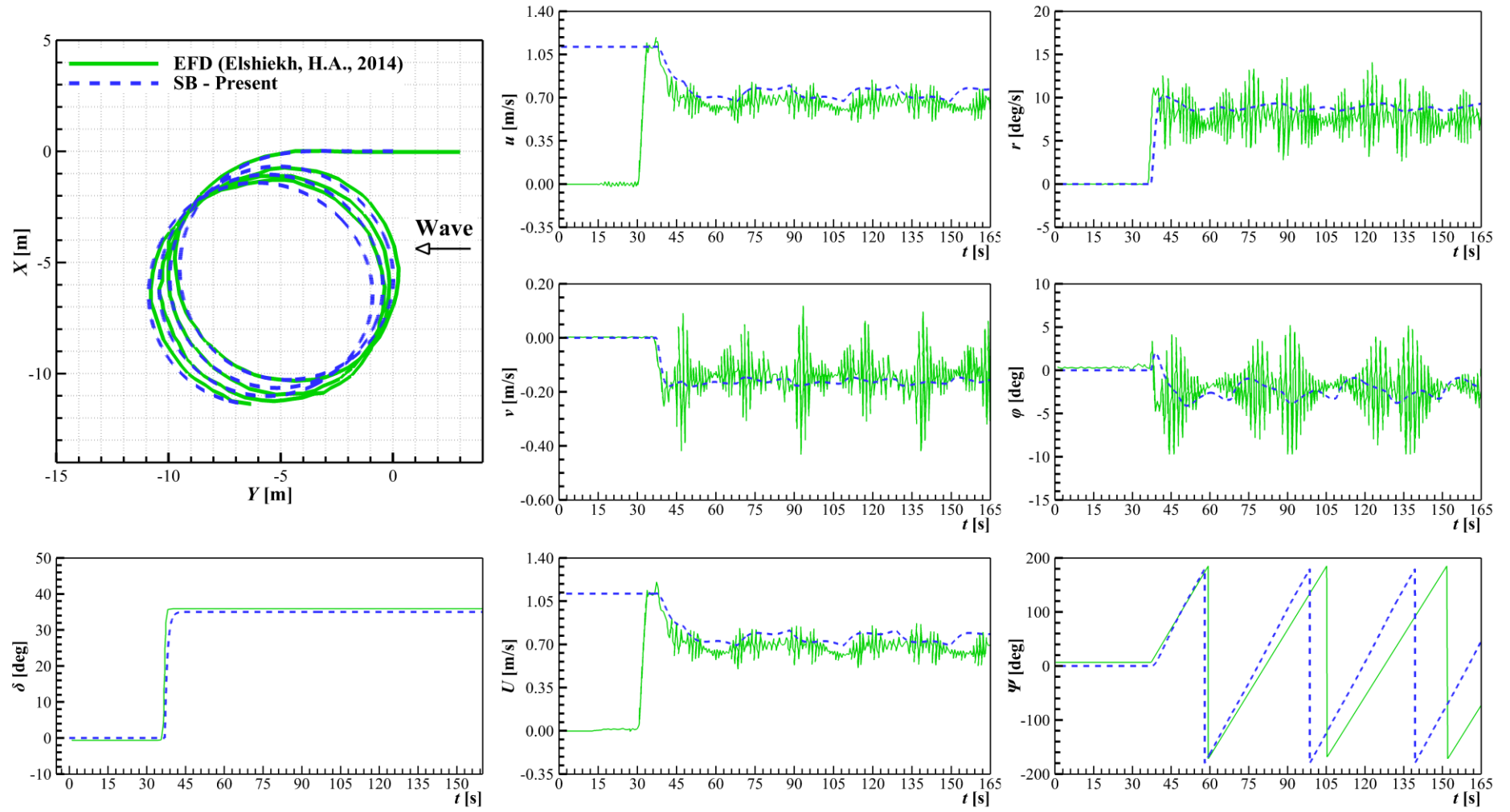


Figure I.7 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

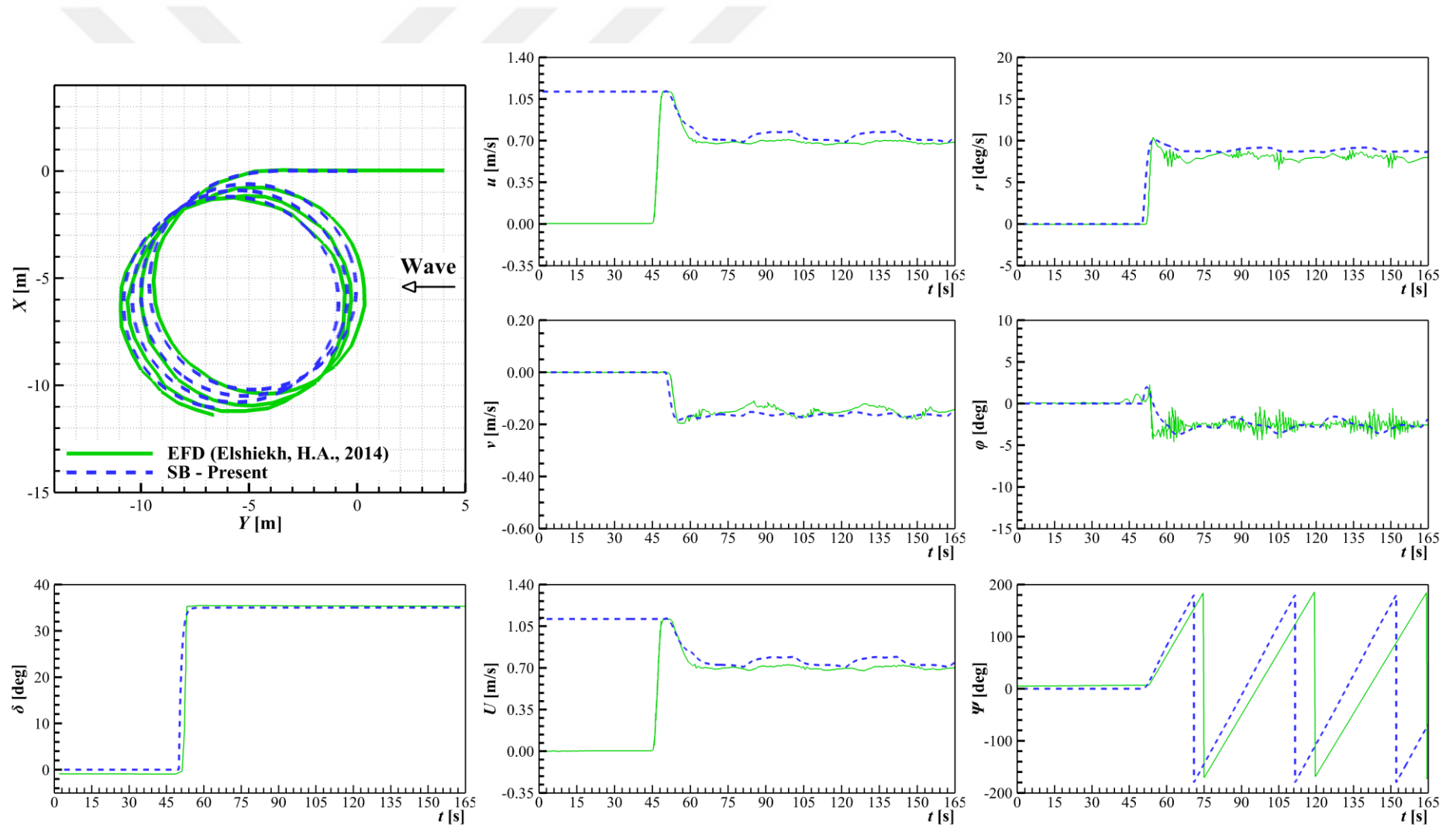


Figure I.8 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$).

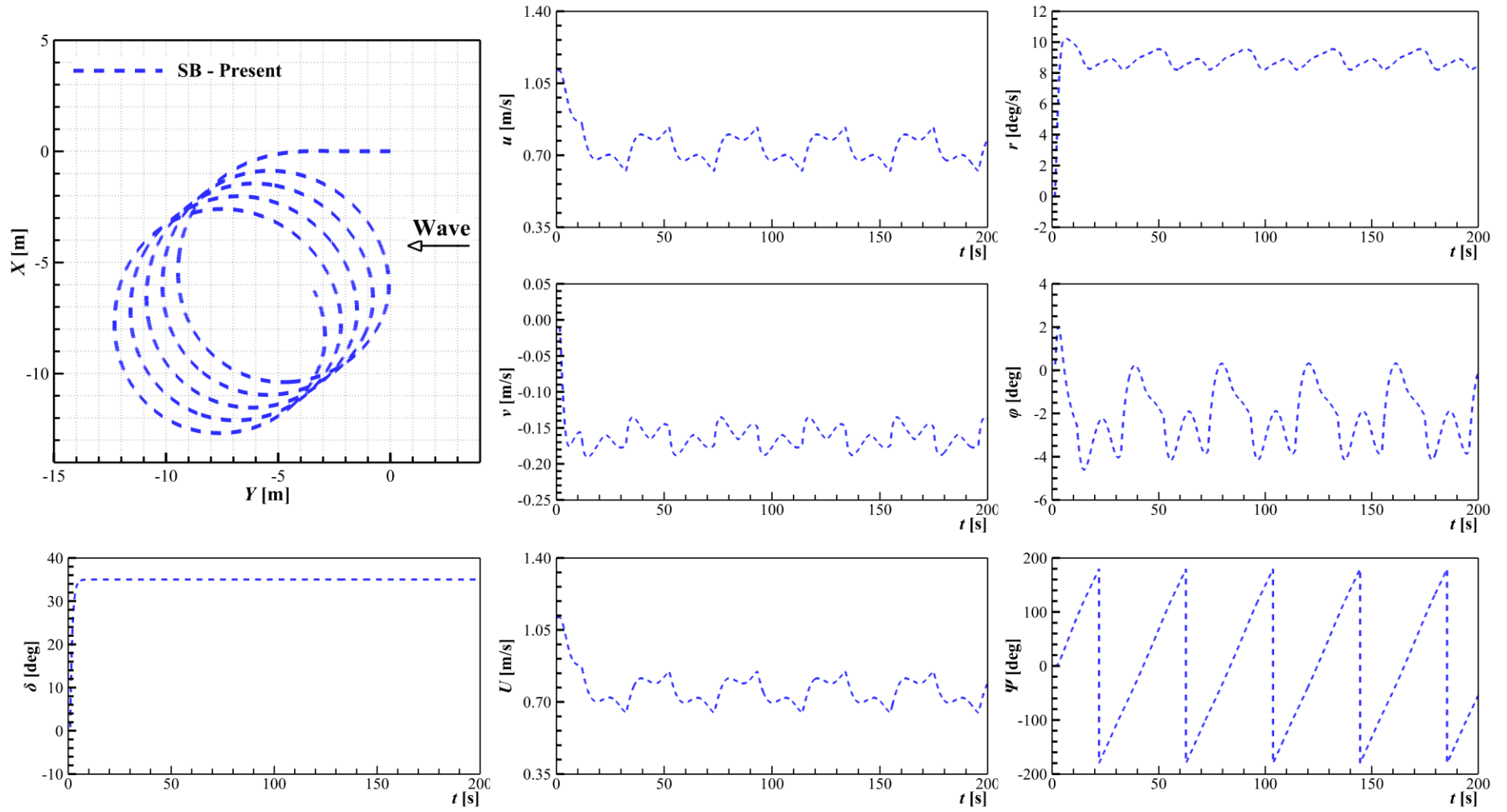


Figure I.9 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).

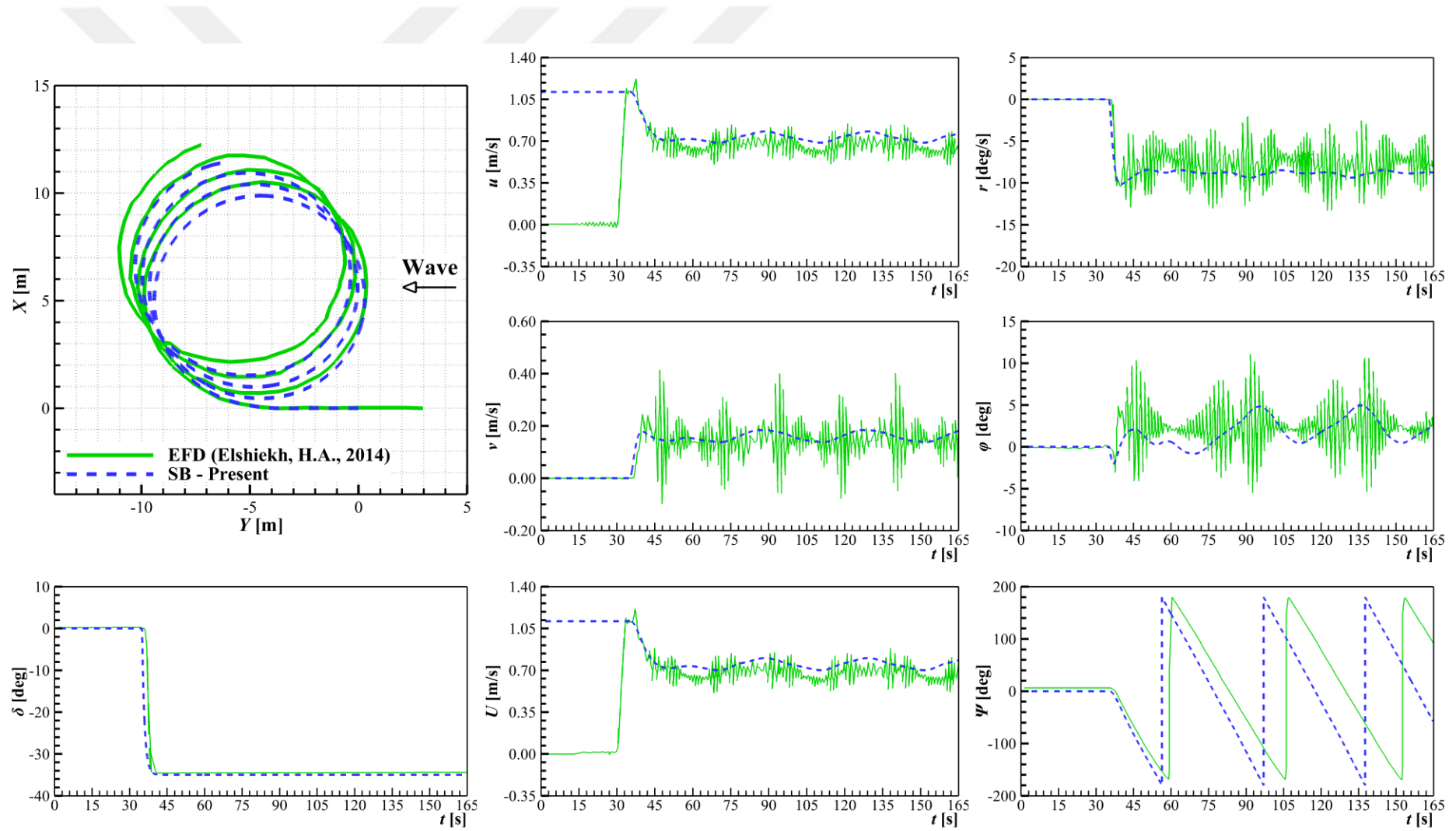


Figure I.10 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

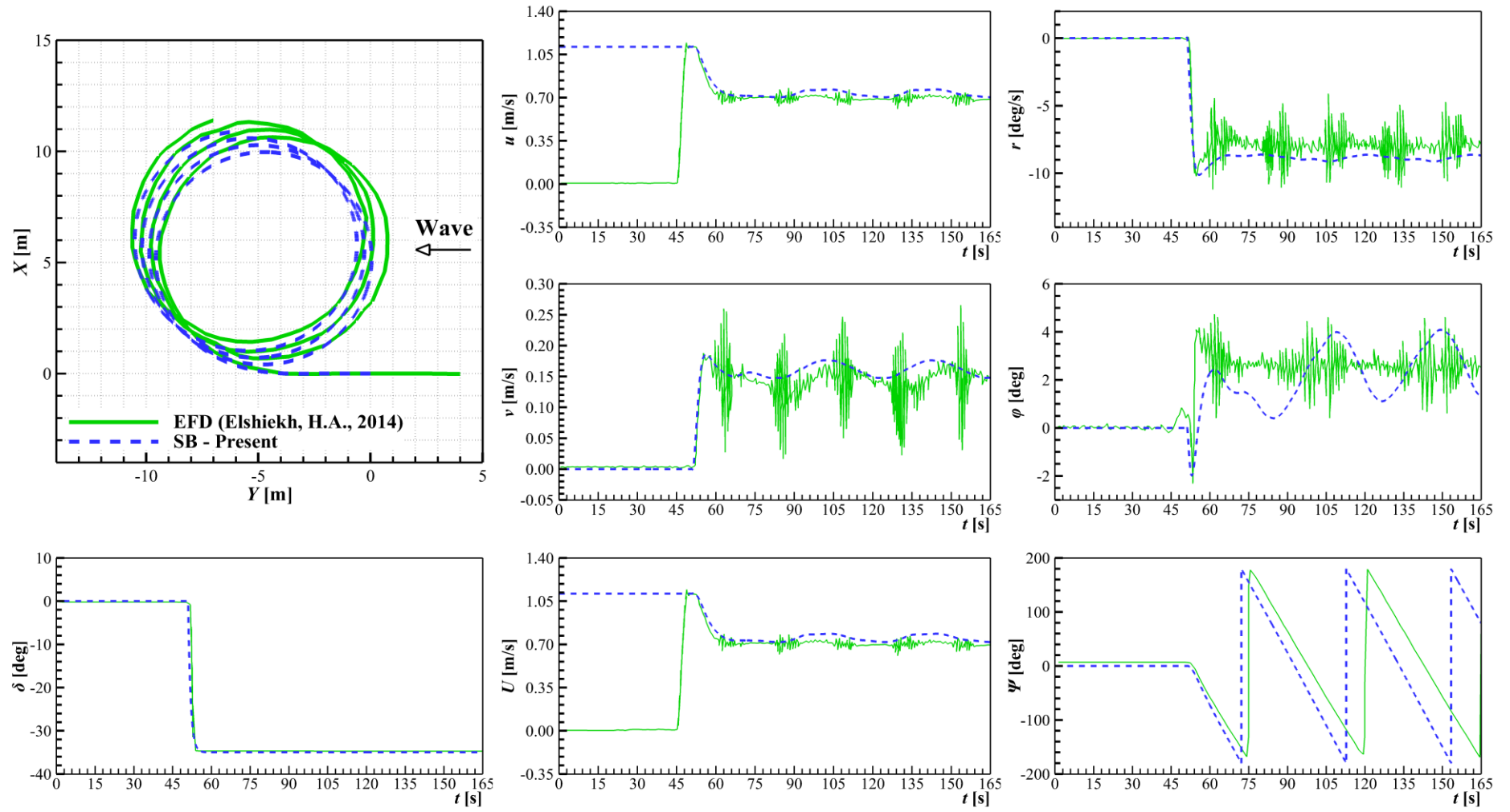


Figure I.11 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$).

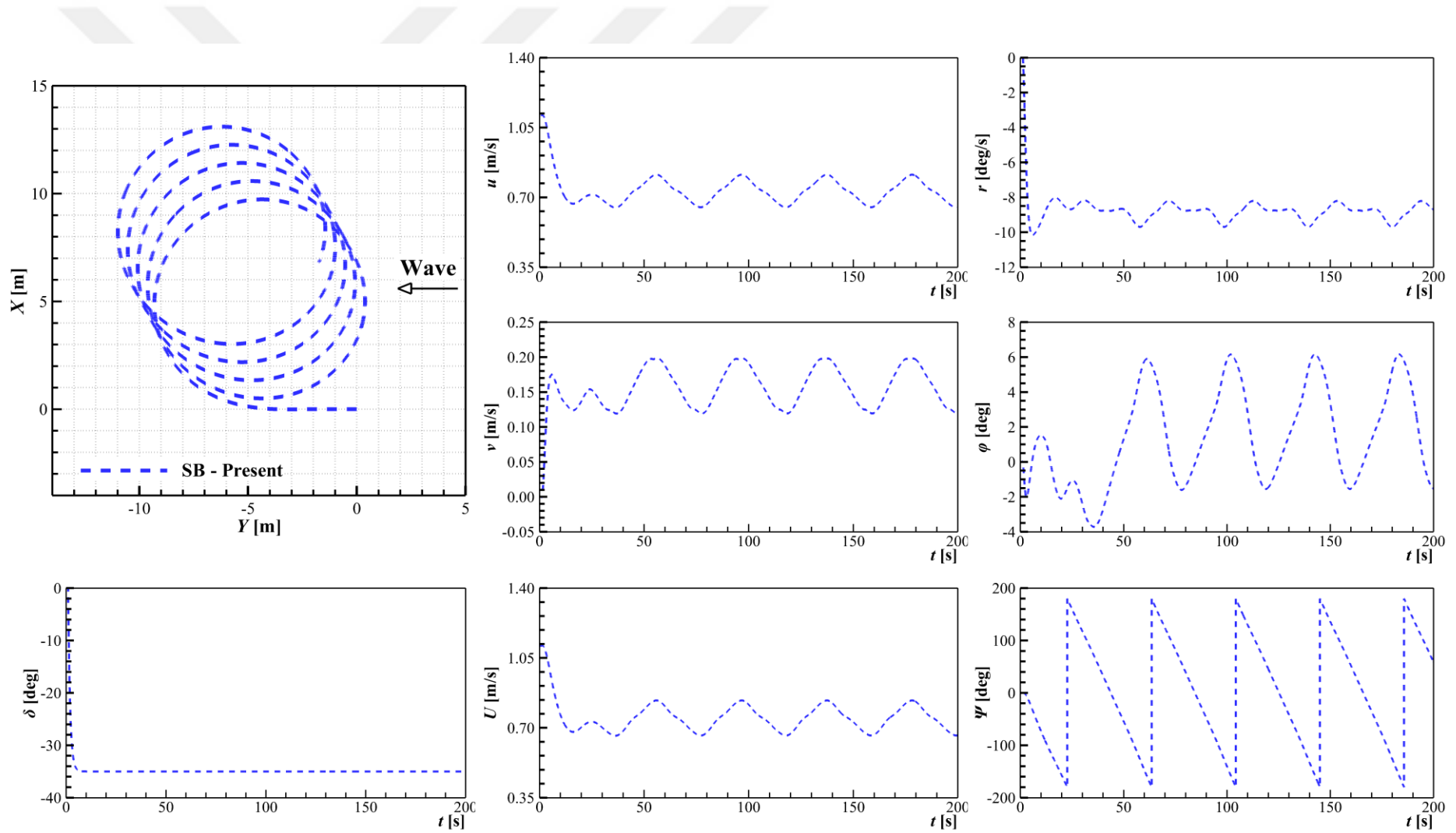


Figure I.12 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = 0^\circ$ ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).

Table I.5 : Comparison of trajectory parameters in regular waves, $\delta = 35^\circ$ and $\chi = -90^\circ$.

	H/λ	λ/L_{WL}	Ad [m]	Tr [m]	TD [m]	STD [m]	SYR [deg/s]	STS [m/s]
EFD (Elshiekh, 2014)			7.1905	4.0929	10.2928	N/A	7.6392	0.6874
S-B – Present	0.020	1.00	8.1715	3.8790	9.5992	9.6014	8.8443	0.7532
Error [%]			-13.64	5.23	6.74	N/A	-15.78	-9.57
EFD (Elshiekh, 2014)			7.1864	4.1493	10.1896	N/A	8.0278	0.7002
S-B – Present	0.020	0.50	8.2421	3.9561	9.7061	9.5866	8.8618	0.7530
Error [%]			-14.69	4.66	4.75	N/A	-10.39	-7.54
S-B – Present	0.025	1.00	8.0303	3.6960	9.3692	9.5437	8.8120	0.7514

Table I.6 : Comparison of trajectory parameters in regular waves, $\delta = -35^\circ$ and $\chi = -90^\circ$.

	H/λ	λ/L_{WL}	Ad [m]	Tr [m]	TD [m]	STD [m]	SYR [deg/s]	STS [m/s]
EFD (Elshiekh, 2014)			7.8430	4.3103	10.0725	N/A	7.7624	0.6783
S-B – Present	0.020	1.00	8.1371	4.0955	10.1740	9.6056	8.8665	0.7586
Error [%]			-3.75	4.98	-1.01	N/A	-14.22	-11.84
EFD (Elshiekh, 2014)			7.5393	4.0272	9.8652	N/A	7.9648	0.6980
S-B – Present	0.020	0.50	8.2779	4.1551	10.1476	9.6051	8.8717	0.7538
Error [%]			-9.80	-3.18	-2.86	N/A	-11.39	-7.99
S-B – Present	0.025	1.00	8.0195	4.1010	10.2747	9.5685	8.8394	0.7498

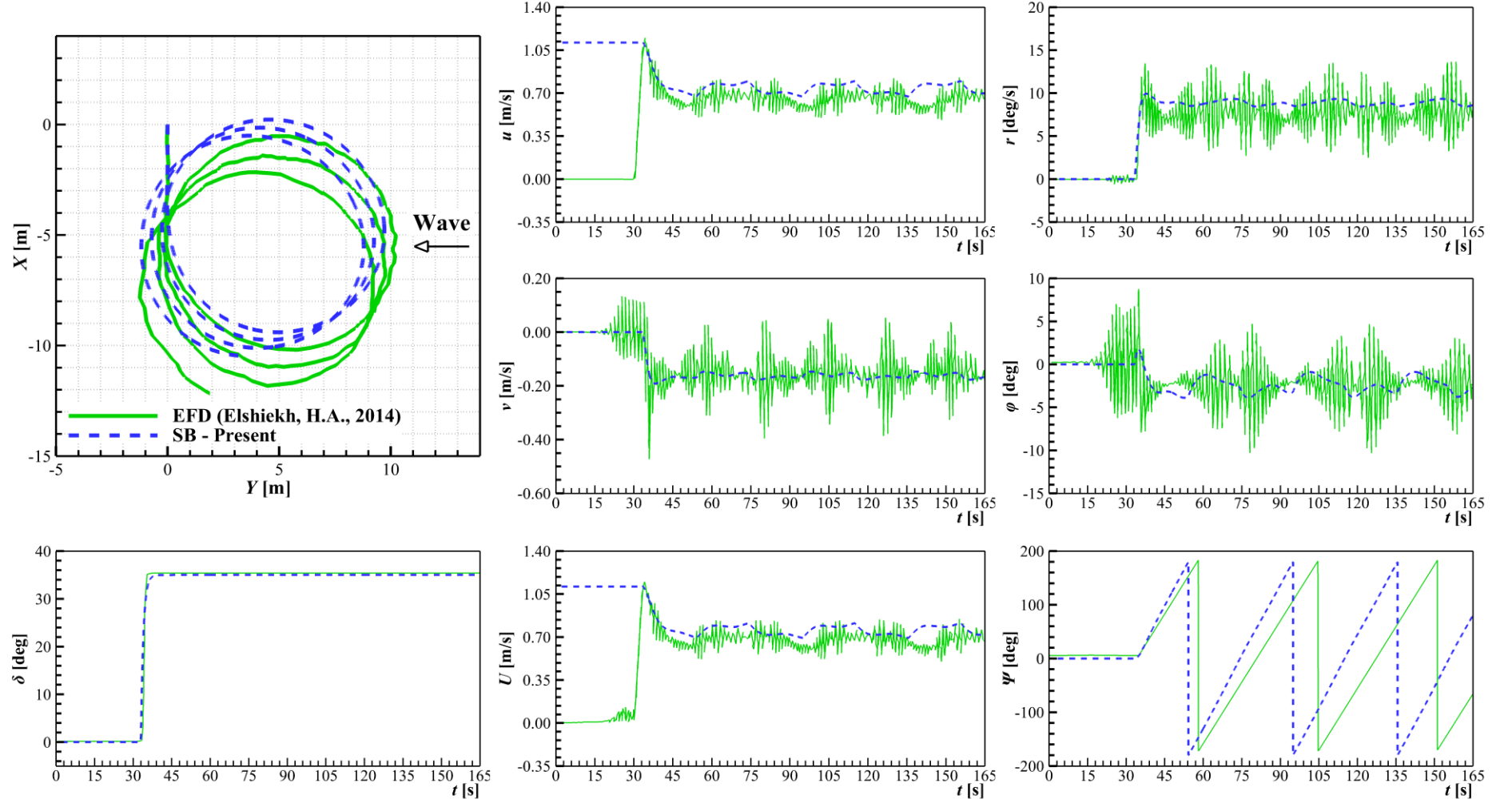


Figure I.13 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

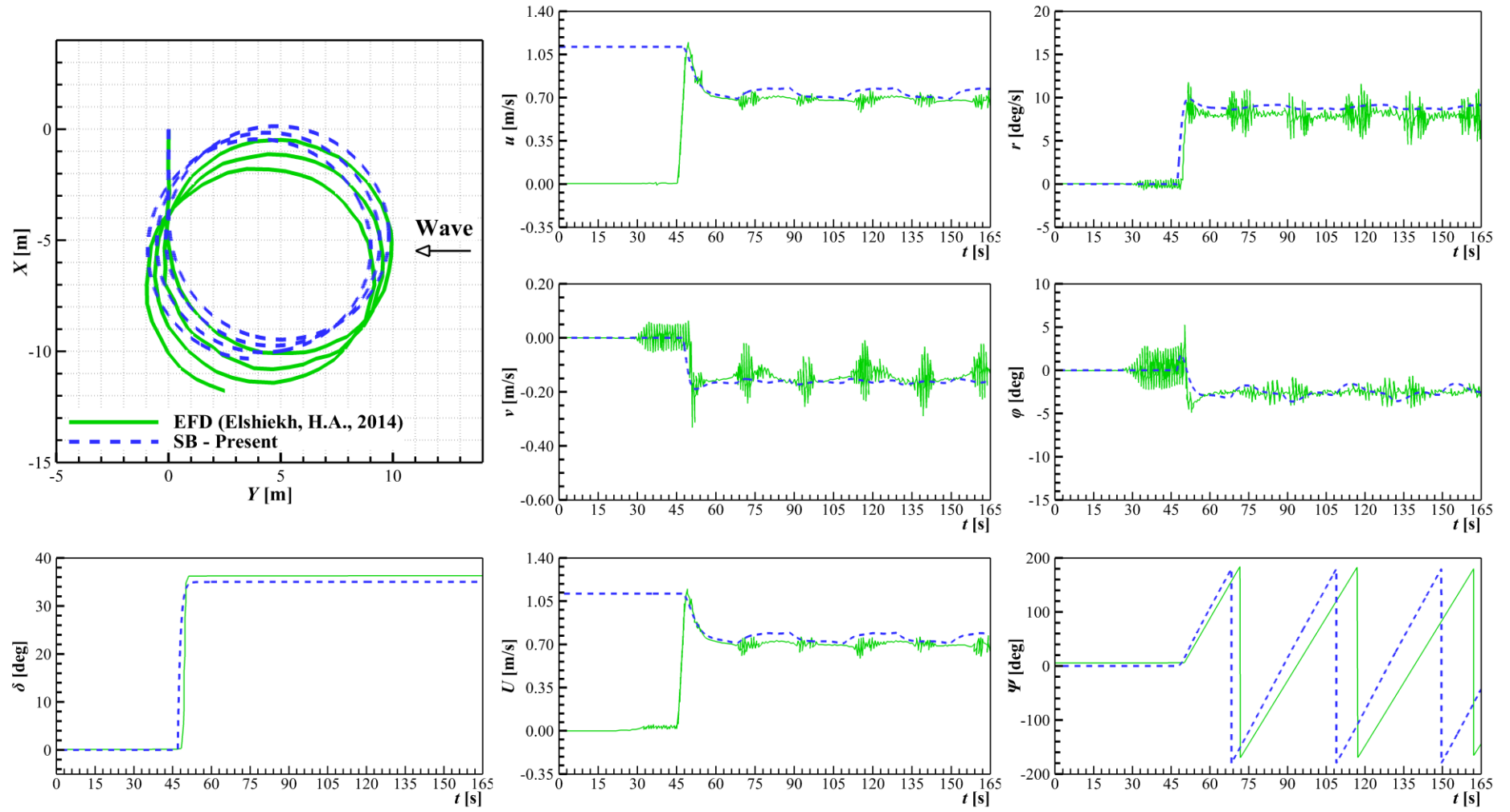


Figure I.14 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$).

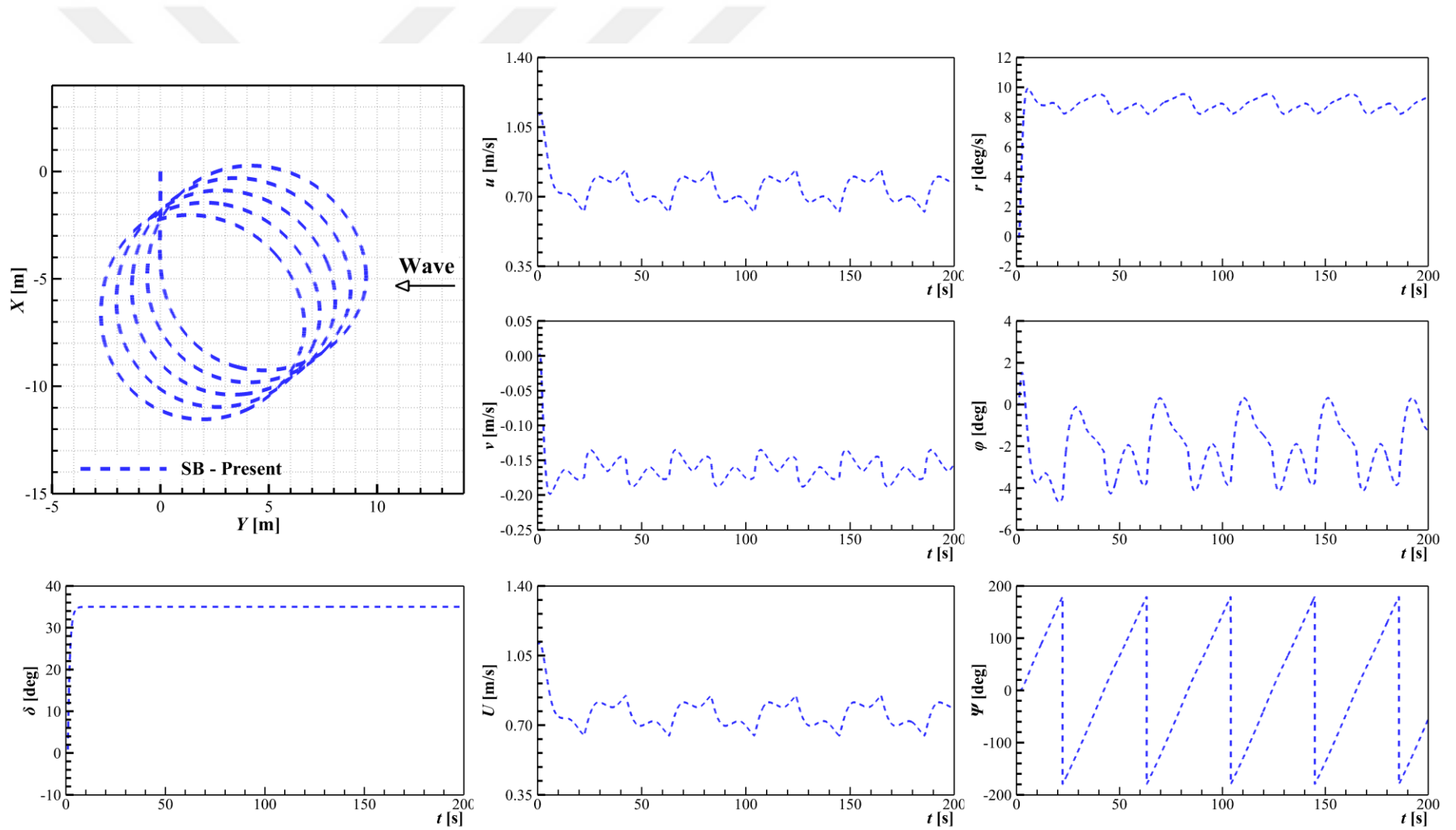


Figure I.15 : Comparison of turning trajectories and time histories in waves, $\delta = 35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).

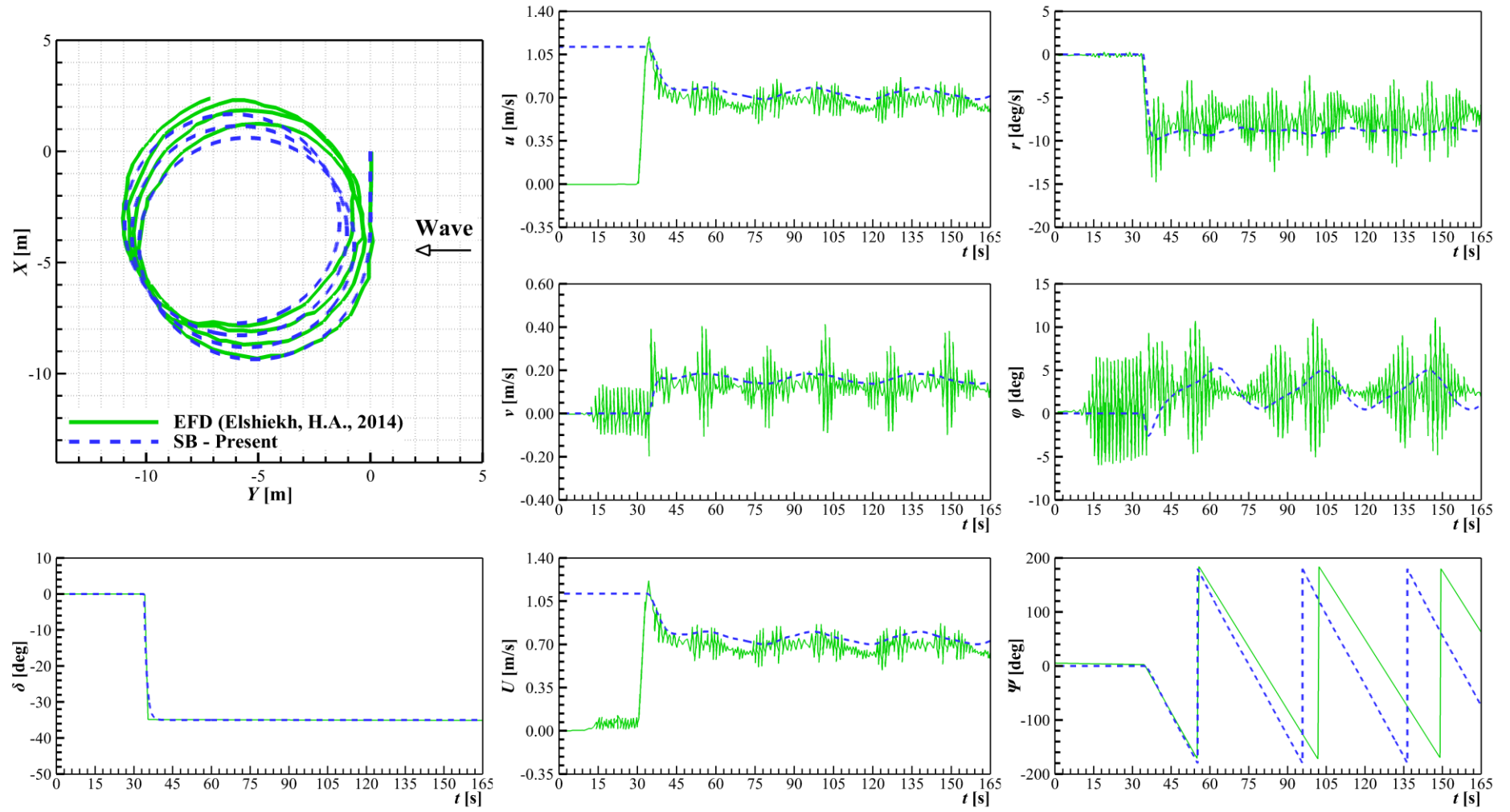


Figure I.16 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 1.00$).

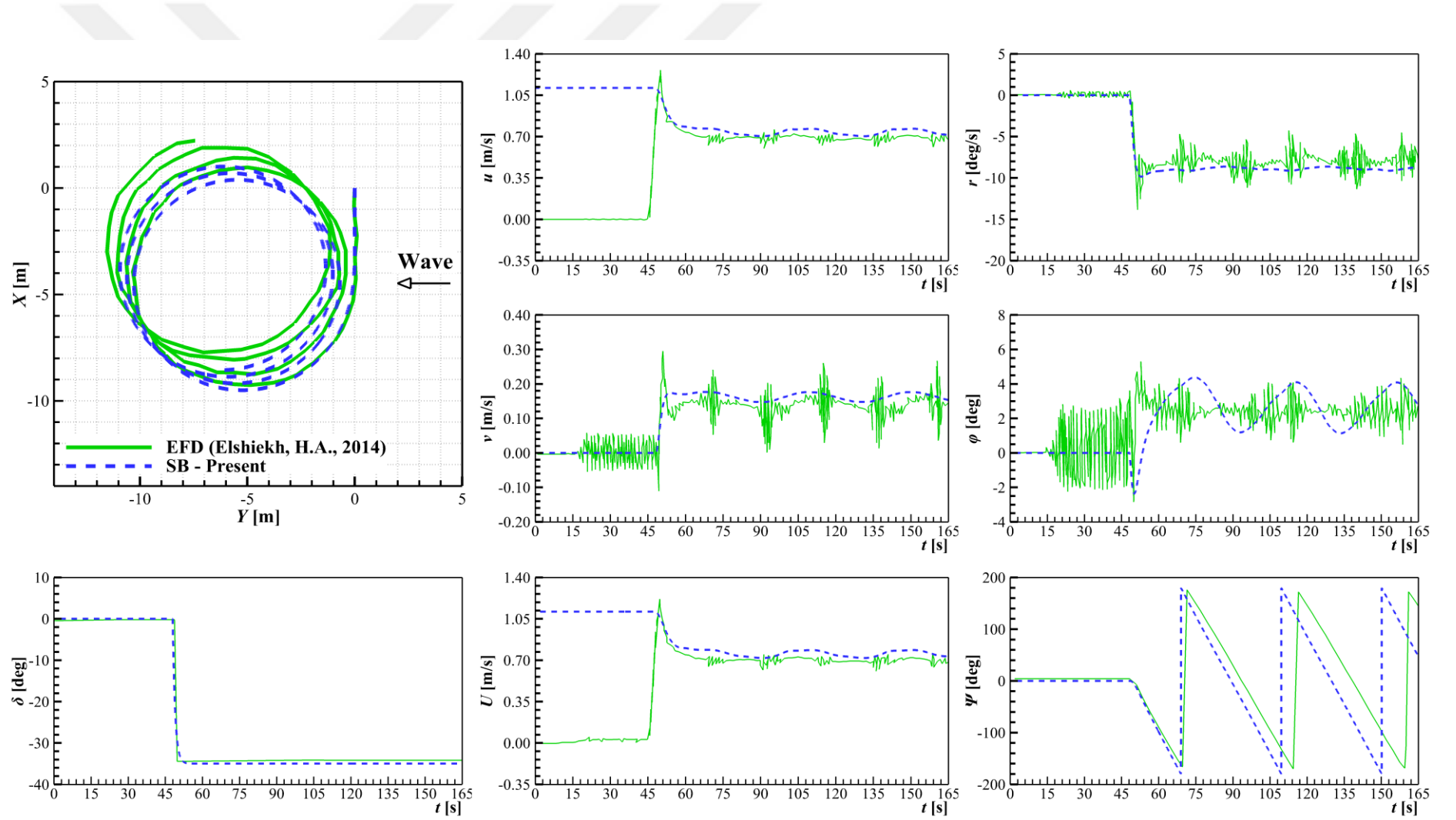


Figure I.17 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.020$, $\lambda/L_{WL} = 0.50$).

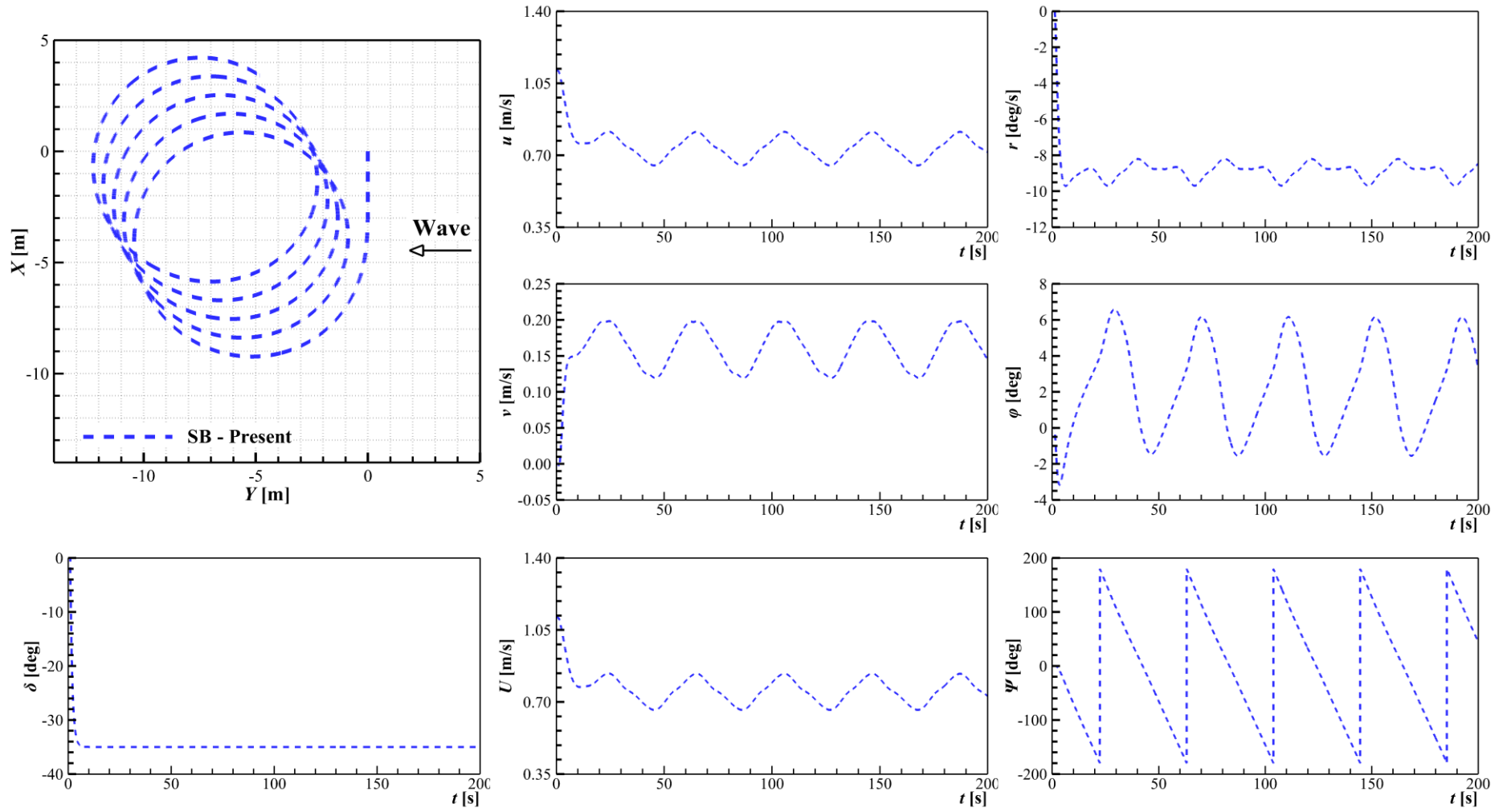


Figure I.18 : Comparison of turning trajectories and time histories in waves, $\delta = -35^\circ$ and $\chi = -90^\circ$ ($H/\lambda = 0.025$, $\lambda/L_{WL} = 1.00$).



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Ozturk, D., Delen, C., Mancini, S., Serifoglu, M. O., & Hizarci, T. (2021). Full-scale cfd analysis of double-m craft seakeeping performance in regular head waves. *Journal of Marine Science and Engineering*, 9(5). <https://doi.org/10.3390/jmse9050504>

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Ozturk, D., Giray, İ., Gul, M., Mentes, A., & Akyildiz, H., (2018). *Assessment of Barriers to Short Sea Shipping in Turkey*. 3rd International Naval Architecture and Maritime Symposium, INT-NAM (pp.483-494). Istanbul, Turkey

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