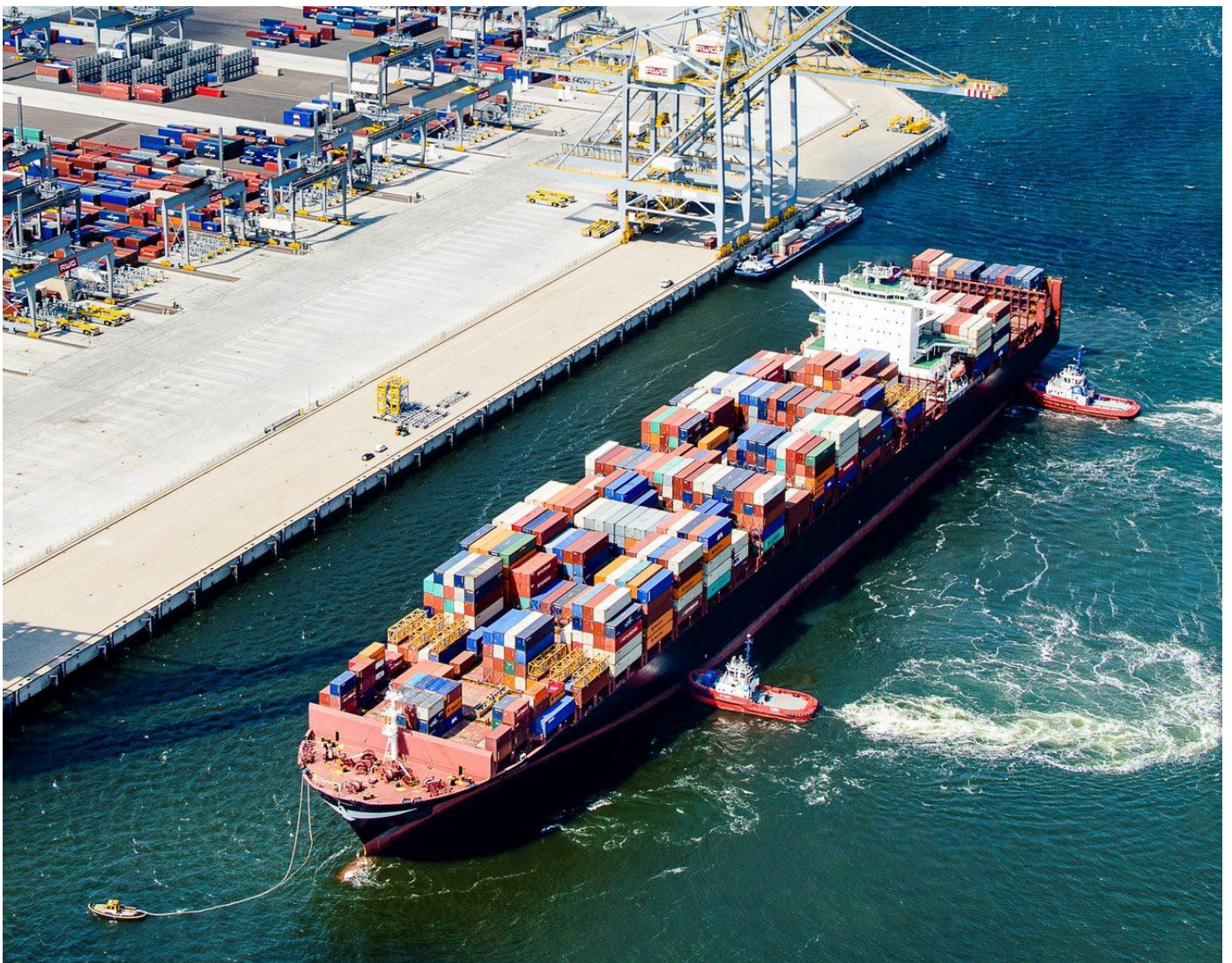




PIANC

The World Association for Waterborne
Transport Infrastructure

PIANC FENDER GUIDELINES 2024



MarCom Working Group Report N° 211 – 2024

PIANC REPORT N° 211

MARITIME NAVIGATION COMMISSION

GUIDELINES FOR THE DESIGN, MANUFACTURING AND TESTING OF FENDER SYSTEMS 2024

March 2024

PIANC has Technical Commissions concerned with inland waterways and ports (InCom), coastal and ocean waterways (including ports and harbours) (MarCom), environmental aspects (EnviCom) and sport and pleasure navigation (RecCom).

This report has been produced by an international Working Group convened by the Maritime Commission (MarCom). Members of the Working Group represent several countries and are acknowledged experts in their profession.

The objective of this report is to provide information and recommendations on good practice. Conformity is not obligatory and engineering judgement should be used in its application, especially in special circumstances. This report should be seen as an expert guidance and state-of-the-art on this particular subject. PIANC disclaims all responsibility in the event that this report should be presented as an official standard.

PIANC HQ
Boulevard du Roi Albert II 20 B. 3
1000 Brussels | Belgium

<http://www.pianc.org>

VAT BE 408-287-945

ISBN 978-2-87223-036-5

© All rights reserved

TABLE OF CONTENTS

TABLE OF CONTENTS.....	4
LIST OF FIGURES.....	9
LIST OF TABLES.....	10
1 INTRODUCTION AND GENERAL ASPECTS	12
1.1 WG 211, A Completely New Guideline.....	12
1.2 Function of Guideline.....	12
1.3 Scope of Guideline	13
1.4 Climate Change	13
1.5 PIANC Certified Fenders and PIANC Type Approval.....	14
1.6 Members of Working Group 211	14
1.7 Meetings.....	15
1.8 Acknowledgements.....	15
1.9 Use of the Guidelines	16
2 INTRODUCTION TO THE PRINCIPLES OF FENDERING	17
2.1 General Design Approach.....	17
2.2 Fender Types and Systems	17
2.3 Fender Efficiency.....	19
2.3.1 Types of Buckling Fenders	19
2.3.2 Types of Sideloaded Fender.....	20
2.4 Fender Selection and Fender System Design.....	20
2.4.1 Buckling Fender Systems	21
2.4.2 Side Loaded Fender Systems	23
2.4.3 Side Loaded Floating Fenders.....	24
2.4.4 Special Applications.....	25
2.5 Ship-to-Ship Fendering	26
2.6 Fenders and Structures	27
3 PARTICULAR ASPECTS REGARDING DESIGN VESSELS	28
3.1 Vessel Characteristics	28
3.1.1 Overview	28
3.1.2 Container Vessels	28
3.1.3 Dry Bulk & Ore Carriers	29
3.1.4 Cruise Vessels	30
3.1.5 RoRo, RoPax & Vehicle (Car) Carriers	30
3.1.6 Tankers	31
3.1.7 Gas Carriers (LPG and LNG)	31
3.1.8 General Cargo, Refrigerated Cargo and Livestock Carriers	31
3.1.9 Passenger Ferries.....	32
3.1.10 Fishing Vessels	32
3.1.11 Offshore Supply Vessels & Harbour Tugs.....	33
3.1.12 Other Vessels.....	33
3.2 Displacement	33
3.3 Influence of Vessel Hull Characteristics on Fender Design	34
4 BASIS OF DESIGN	36
4.1 Functional Requirements.....	37
4.2 Operational Requirements.....	37
4.3 Site Conditions	38
4.4 Design Criteria, Return Period and Design Life.....	38
4.5 Operation and Maintenance	39

4.6	Reliability Requirements.....	40
5	BERTHING ENERGY	42
5.1	Berthing Manoeuvres and Navigation Conditions	42
5.1.1	Alongside Berthing (Parallel or Angular)	42
5.1.2	End Berthing (Longitudinal)	42
5.1.3	Ship-to-Ship Berthing.....	43
5.1.4	Navigation Conditions	43
5.2	Kinetic Energy of a Berthing Vessel.....	44
5.2.1	Fundamentals of Kinetic Energy Method.....	44
5.2.2	Design Berthing Energy	46
5.2.3	Characteristic Berthing Energy.....	46
5.3	Displacement of Vessel	47
5.4	Berthing Velocity	47
5.4.1	Berthing Velocity Based on Local Knowledge, Data and Experience	48
5.4.2	Transverse Velocity	48
5.4.3	Longitudinal Velocity.....	50
5.5	Berthing Angle	50
5.6	Eccentricity Factor	52
5.6.1	Distance r_F and Berthing Impact Point	53
5.6.2	Radius of Gyration	56
5.6.3	Vessel Centre of Mass	56
5.6.4	Eccentricity Factor for End Berthing	57
5.7	Added Mass Factor.....	57
5.7.1	Alongside Berthing	57
5.7.2	End Berthing	58
5.8	Partial Energy Factor.....	58
5.9	Ship-to-Ship Berthing	63
5.9.1	Berthing Manoeuvres	63
5.9.2	Ship-to-Ship Berthing Energy.....	64
5.9.3	Characteristic Berthing Velocity	65
5.9.4	Eccentricity Factor	66
6	FENDER SYSTEM SELECTION	67
6.1	Fender Selection Process and Role of the Designer	67
6.2	Fender Selection Overview	67
6.2.1	Fender Selection Using Project Specific Design Criteria	67
6.2.2	Fender Selection Using Pre-Set Design Criteria.....	70
6.3	Fender System Selection.....	70
6.4	Factors Influencing Fender System Selection	72
6.4.1	Bow Radius	73
6.4.2	Bow Flare	73
6.4.3	Fender System Pitch	75
6.4.4	Single Fender System Contact	77
6.4.5	Multiple Fender System Contact.....	77
6.4.6	Vessel Hull Structure.....	79
6.4.7	Vessel Belting.....	79
6.4.8	Double Hull Contact.....	81
6.4.9	Type of Supporting Structure	82
6.4.10	Flexible Dolphins	82
6.4.11	Fender System Elevation	84

6.4.12	Pneumatic and Foam Fenders.....	84
6.4.13	Number and Size of Fenders	85
6.4.14	Shear of Fenders	85
6.4.15	Submerged Fenders	86
6.4.16	Fender Orientation	86
6.4.17	Ageing Effects on Fenders.....	87
6.4.18	Non-Marking Fenders	88
6.4.19	Mooring Analysis	88
6.4.20	Permanent Mooring	89
6.5	Base Fender Performance.....	90
6.6	Correction Factors.....	91
6.6.1	Application of Correction Factors	91
6.6.2	Velocity Factor.....	92
6.6.3	Temperature Factor	93
6.6.4	Angular Factor	94
6.6.5	Multiple Fender Contact Factor	95
6.7	Determining Fender Design Performance	96
6.7.1	Application of Partial Resistance Factors.....	97
6.7.2	Fender Performance Partial Resistance Factor.....	100
6.7.3	Multiple Fender Contact Partial Resistance Factor	101
6.7.4	Load Partial Resistance Factor.....	102
6.8	Hull Pressure.....	102
6.8.1	Factors Affecting Induced Hull Pressure.....	103
6.8.2	Hull Structure	103
6.8.3	Recent Hull Pressure Research	105
6.8.4	Fender Reaction Force Distribution	107
6.8.5	Hull Pressure for Fender Panels	108
6.8.6	Hull Pressure for Cylindrical Fenders.....	109
6.8.7	Hull Pressure for Foam and Pneumatic Fenders	109
6.8.8	Typical Hull Pressure Capacities	109
6.9	Ship-to-Ship Fendering	111
6.9.1	Ship-to-Ship Fender Selection.....	111
6.9.2	Vessel Stand-off and Number of Fenders	112
6.9.3	Ship-to-Ship Operational Considerations	113
7	FENDER SELECTION UNDER MOORED CONDITIONS.....	114
7.1	Fender Design for Moored Vessels	114
7.2	Characterisation of Vessel and Berth Configuration.....	116
7.3	Mooring Analysis.....	117
7.4	Creep and Fatigue Assessment for Dynamic Mooring Situations	117
7.4.1	Creep and Cyclic Loading Limits for DMA	121
8	FENDER SYSTEM COMPONENT DESIGN.....	122
8.1	Fender Panel Design	122
8.1.1	Load Cases and Structural Analysis	122
8.1.2	Fender Panel Internal Structure.....	123
8.1.3	Edge Chamfers.....	124
8.1.4	Panel Size and Position.....	125
8.2	Chains and Fixings Design	126
8.2.1	Chain Types and Positions	126
8.2.2	Support Chains for Side Loaded Fenders.....	130

8.2.3	Bracket Design	131
8.2.4	Fixing Anchor Design	131
8.3	Low Friction Facing Design	133
8.3.1	Low Friction Resin Facings	134
8.3.2	Steel Facing	134
8.3.3	Fastenings	135
8.4	Parallel Motion Fender Systems	135
8.5	Fender Interfaces with Mooring Lines	136
8.6	Whole-Life Considerations	137
8.7	Corrosion of Fender Components	137
8.8	Marine Growth	138
8.9	Design of Fender Components in Icy Conditions	138
9	MANUFACTURING OF FENDER SYSTEMS	139
9.1	Manufacturer Qualifications	139
9.1.1	Quality Control by Manufacturer	139
9.1.2	Workforce Qualification	139
9.1.3	Storage of Produced Fender Elements	140
9.2	Rubber Fender Compound	140
9.2.1	Natural Rubber	140
9.2.2	Synthetic Rubber	141
9.2.3	Recycled Rubber	141
9.2.4	Fillers	141
9.2.5	Anti-Ageing Agents	142
9.2.6	Oil (Processing Aids)	142
9.2.7	Accelerators and Sulphur	142
9.2.8	Other Ingredients	142
9.3	Manufacturing Process of Rubber Fenders	142
9.3.1	Mixing of the Compound	143
9.3.2	Moulding, Extruding and Wrapping	143
9.3.3	Curing/Vulcanising of the Rubber Element	144
9.4	Fabrication of Steel Fender Panels	144
9.4.1	Panel Internal Structure	145
9.4.2	Material	146
9.4.3	Protection Against Corrosion for Steel Panels	146
9.5	Fabrication of UHMW-PE Low Friction Facing	147
9.6	Fabrication of Accessories	147
9.6.1	Chains	148
9.6.2	Anchors and Accessories	148
9.6.3	Protection Against Corrosion for Accessories	148
9.7	Pneumatic Fenders	149
9.8	Foam Fenders	149
10	TEST PROCEDURES OF MARINE FENDERS	150
10.1	Classification of Rubber Fender Testing	150
10.2	Determination of Fender Base Performance	153
10.2.1	Constant Velocity (CV)	153
10.2.2	Standard compression and Base performance	153
10.2.3	Standard condition	153
10.3	Test Apparatus for Compression Test	153
10.4	Supporting Protocols	154

10.4.1	Break-In Compression Cycle	154
10.4.2	Stabilising Compression Cycles	154
10.4.3	Thermal Stabilisation	154
10.5	Performance Testing Protocol for Standard Compression	155
10.6	Protocol for Durability Test	158
10.7	Material Tests	160
10.7.1	Rubber Compound: Physical Properties	160
10.7.2	TGA (Thermogravimetric Analysis)	163
10.7.3	Test Protocol for Creation of Velocity Factors	163
10.7.4	Test Protocol for Creation of Temperature Factors (C _t)	165
10.7.5	Test Protocol for Angle Factor	166
10.8	Rubber Compounds for Pneumatic Fenders.....	167
10.9	Performance Tests of Pneumatic Fenders	167
10.10	Classification of Foam Fender Testing.....	168
10.11	Material Tests for Foam Fenders.....	169
10.11.1	Foam Core	169
10.11.2	Polyurethane (Solvent Free) Outer Layer Elastomer or Similar Material	169
10.11.3	Reinforcement Layer.....	170
10.12	Performance Tests of Foam Fenders	170
10.12.1	Fundamental and Type Approval Tests for Foam Fenders.....	170
10.12.2	Velocity, Angular and Temperature Factor for Foam Fenders	171
10.12.3	Durability Tests.....	172
10.12.4	Skin Thickness and Foam Density Verification Testing	172
10.12.5	Fender Pull-Through Test.....	173
10.13	Tests for Accessories	174
10.13.1	Fabricated Steel Structures Air-Leakage/Pressure Test Procedure.....	174
10.13.2	NDT (Non- Destructive Testing – All Fabricated Steel Parts)	174
	UHMW-PE Resin and Pads	175
11	INSTALLATION, INSPECTION AND MAINTENANCE	176
11.1	Installation	176
11.1.1	Handling and Installation	176
11.1.2	Safety and Accessibility	176
11.2	Spares and Storage	177
11.2.1	Spares	177
11.2.2	Storage	177
11.3	Inspection and Maintenance	177
11.3.1	Inspection.....	178
11.3.2	Maintenance	179
11.3.3	Emerging Technology	181
12	SUSTAINABILITY OF FENDERS	182
12.1	Circular Economy with Rubber Fenders.....	182
12.2	Carbon Footprint	182
12.3	Rubber Sourcing.....	183
12.4	Fabrication.....	183
12.5	Fender Design and Materials Selection	184
12.6	Recycling	184
12.6.1	Current Practice of Fender Recycling	185
12.6.2	Rubber Recycling	185
12.6.3	Foam Fender Recycling.....	186

12.6.4	Steel Recycling	186
12.6.5	UHMW-PE Recycling	187
12.7	Recommendations for Fender Sustainability	187
13	SPECIFICATION WRITING	188
13.1	Responsibility for Providing Information.....	188
13.2	General	189
13.2.1	Qualification of Supplier	189
13.2.2	Standards and Codes of Practice	189
13.2.3	Quality Control.....	189
13.2.4	Submittal Requirements	189
13.2.5	Records.....	190
13.2.6	Warranty, Product Liability and Compliance	191
13.3	Vessel, Berthing and Quay Structure Considerations	191
13.4	Manufacturing, Testing and Quality Requirements for Fender Units.....	192
13.5	Delivery, Installation and Storage	193
13.6	Sustainability	193
14	REFERENCES.....	194
APPENDIX A: RELIABILITY REQUIREMENTS FOR FENDERS		198
APPENDIX B: THERMAL CONDUCTION OF RUBBER FENDERS.....		203
APPENDIX C MULTI-FENDER CONTACT, GEOMETRY FOR FENDER COMPRESSION CALCULATIONS.....		206
APPENDIX D: ABBREVIATIONS SYMBOLS AND UNITS.....		208
14.1	D.1 ABBREVIATIONS	208
14.2	D.2 SYMBOLS	209
APPENDIX E: TERMS OF REFERENCE WG 211		214

LIST OF FIGURES

Figure 1.1:	Flowchart WG 211 guideline	16
Figure 2.1:	Typical buckling fender deflection curve	20
Figure 2.2:	Sideloaded fender deflection curve.....	20
Figure 3.1:	Parallel mid-body of a typical container vessel hull	29
Figure 3.2:	Fender contact zones on vessel hull at low berthing angles.....	35
Figure 5.1:	Berthing motion and corresponding reference angles.....	42
Figure 5.2:	End berthing mode	43
Figure 5.3:	Degrees of motion of a vessel.....	45
Figure 5.4:	Nomenclature for calculating C_e factor (single fender contact)	52
Figure 5.5:	Nomenclature for calculating C_e factor (multiple fender contact)	53
Figure 5.6:	Vessel centre of mass and under keel clearance	56
Figure 5.7:	Added Mass Factor.....	57
Figure 5.8:	Under keel clearance of a vessel	58
Figure 5.9:	Ship-to-ship berthing for free floating vessels	64
Figure 6.1:	Overview of fender selection process.....	69
Figure 6.2:	Bow radius (RB) and length of bow flare (LBow)	73
Figure 6.3:	Bow flare (β_f) and clearance to supporting structure (c_{eff})	74
Figure 6.4:	Estimate of maximum fender pitch and minimum clearance to the supporting structure	76
Figure 6.5:	Single fender contact	77
Figure 6.6:	Contacted fenders at larger berthing angles (with contact on bow flare)	78
Figure 6.7:	Contacted fenders at smaller berthing angles (almost parallel to the berth)	78
Figure 6.8:	Arch fenders with maximum deflection smaller than belting height	80
Figure 6.9:	Arch fenders with maximum deflection greater than belting height	80

Figure 6.10: Belting causing vessel hull double contact and line loads	81
Figure 6.11: Application of correction factors and partial safety factors	90
Figure 6.12: Calculation of $C_{mult, c}$ for multiple fender contact	96
Figure 6.13: Identification of $R_{f, c}$ and calculation of $R_{f, d}$ for linear force-deflection curves	97
Figure 6.14: Identification of $R_{f, c}$ and calculation of $R_{f, d}$ for non-linear force-deflection curves.....	98
Figure 6.15: Typical vessel hull side structures (TNO, 2019)	104
Figure 6.16: Equivalent hull pressure limits (Berendsen et al., 2024)	106
Figure 6.17: Maximum fender reaction force (Berendsen et al., 2024)	106
Figure 6.18: Contact area of a flat fender panel	108
Figure 7.1: Fender design flow chart for moored conditions.....	115
Figure 7.2: Example of fender deflection time series showing average value and different cycles	120
Figure 7.3: Energy distribution as a function of frequency.....	121
Figure 8.1: Typical design cases of fender contact with vessel hull profile	123
Figure 8.2: Recommended minimum thicknesses for steel in fender panels.....	124
Figure 8.3: Typical fender system chain layout (viewed from rear of fender panel)	127
Figure 8.4: Typical tension chain arrangement	128
Figure 8.5: Recommendations for angles and chain arrangement for non-compressed fenders	130
Figure 8.6: Double and 4-chain arrangements for foam and pneumatic fenders.....	130
Figure 8.7: Typical examples of fender system bracket assemblies	131
Figure 8.8: Typical examples of fender system anchors.....	132
Figure 8.9: Typical facing connection details	135
Figure 9.1: Manufacturing of rubber fender.....	143
Figure 9.2: Typical fender panel cross section samples (U profile on left, T-profiles on right side	145
Figure 9.3: Typical fender frontal, back and internal panel structure views	145
Figure B-1: heat conduction (Nakamura et al., 2020)	203
Figure B-2: relation rubber thickness and temperature adaptation of fender (Nakamura et al., 2020) .	204
Figure B-3: Time history of inner temperature and ambient temperature (Nakamura, Nakamura, Miyata, Yoneyama, & Kashima, 2020)	205
Figure C-1: Fender contact at hull tangent line impacts at fender centreline	206
Figure C-2: Fender contact at hull tangent line centrally between fenders.....	207

LIST OF TABLES

Table 2-1: Typical Cone Fender system (A) and Performance Curves (B).....	21
Table 2-2: Typical Cell Fender system (A) and Performance Curves (B).....	21
Table 2-3: Typical Element/Leg Fender system (A) and Performance Curves (B)	22
Table 2-4: Typical Special Element Fender system (A) and Performance Curves (B).....	22
Table 2-5: Typical Arch/Trapezoidal Fender system (A) and Performance Curves (B).....	23
Table 2-6: Typical Cylindrical Fender system (A) and Performance Curves (B)	23
Table 2-7: Typical Foam Fender system (A) and Performance Curves (B)	24
Table 2-8: Typical Pneumatic Fender system (A) and Performance Curves (B)	25
Table 2-9: Typical Parallel Motion Fender system (A) and Performance Curves (B).....	25
Table 2-10: Typical Pile/Pivot Fender system (A) and Performance Curves (B).....	26
Table 2-11: Typical Rolling Fender system (A) and Performance Curves (B)	26
Table 3-1: Approximate displacement v capacity relationships for various vessel types (WG 235)	34
Table 4-1: Consequence classes and description of failure consequence	41
Table 5-1: Description of navigation conditions at berth	44
Table 5-2: Recommended characteristic values of design variables.....	47
Table 5-3: Characteristic berthing velocity in the absence of site-specific information	49
Table 5-4: Characteristic longitudinal berthing velocity in the absence of site-specific information.....	50

Table 5-5: Berthing angle [degrees] at the moment of contact, no site-specific information available .	51
Table 5-6: Typical C_e factors for different impact point along the vessel	55
Table 5-7: Coefficient of variation in displacement	60
Table 5-8: Reference partial energy factor [γE_{ref}] for 100 berthings per year – single fender contact...	61
Table 5-9: Reference partial energy factor [γE_{ref}] for 100 berthings per year – multiple fender contact	61
Table 5-10: Reference partial energy Factor [γE_{ref}] for 100 berthings per year – for Monitored Berthings	61
Table 5-11: Correction factor for an alternative annual berthing frequency (γn) for $n \leq 100$	62
Table 5-12: Correction factor for an alternative annual berthing frequency (γn) for $n > 100$	62
Table 5-13: Closing velocity for ship-to-ship operations	66
Table 6-1: Application of fender systems for various vessel types	71
Table 6-2: Application of fender systems to various marine applications	72
Table 6-3: Partial resistance factor γf related to the performance of a single fender	101
Table 6-4: Partial resistance factor γ_{mult} related to single and multiple fender contacts	101
Table 6-5: Partial factor γR for reaction forces to be used in fender system design	102
Table 6-6: Typical values of hull pressure capacity	110
Table 6-7: STS recommended stand-off distance, number of fenders and typical sizes for calm conditions	113
Table 7-1: Description of creep damage in fenders depending on their type and material	118
Table 7-2: Description of fatigue damage in fenders depending on their type and material.....	119
Table 8-1: Types of chains	127
Table 8-2: Typical design values of coefficients of friction	133
Table 10-1: Rubber fender testing scheme	152
Table 10-2: Standard conditions where correction factors are 1.0	153
Table 10-3: Additional Information for standard compression test.....	157
Table 10-4: Additional Information for durability test	159
Table 10-5: Physical properties table	162
Table 10-6: Foam fender testing scheme	168
Table 10-7: Foam core physical properties	169
Table 10-8: Polyurethane properties	170
Table 10-9: Nylon reinforcement filaments properties	170
Table 10-10: Verification tests for foam fenders.....	171
Table 10-11: NDT test of all fabricated steel parts	174
Table 10-12: Physical Properties of UHMW-PE resin and pads	175
Table 11-1: Common failure modes for typical fender components.....	179
Table 13-1: Required fender system design information.....	188
Table A-0-1: Probability of failure and associated reliability target for different consequence classes..	199
Table A-0-2: Description of failure consequences in literature	202

1 INTRODUCTION AND GENERAL ASPECTS

1.1 WG 211, A Completely New Guideline

Working Group 211 report began as an update of PIANC MarCom WG 33: 'Guidelines for the Design of Fenders Systems: 2002'. However, this final report is essentially a new guideline rather than an update. This guideline completely supersedes the previous WG 33 guideline (PIANC, 2002) and also all fender design and berthing related guidance in other PIANC MarCom reports referring to WG 33.

Users cannot simply change WG 33 into WG 211 in their own fender specifications. That will lead to significant cost increases due to a different design approach that has been adopted in WG 211. User specifications will require a full update for the following reasons:

- WG 211 describes the physical process of berthing better than WG 33 resulting in higher velocities, lower berthing angles and multiple fender contact.
- Berthing velocity, due to its uncertainty, is the dominant factor over all other factors for fender design.
- WG 211 puts the safety more in the rubber than in the supporting structure. One has to integrally design the total marine structure, not just the fender system.
- WG 211 strongly recommends the use of site-specific information. Only use the velocities in Table 5-3 when no knowledge on, or data of the site is available. Always talk to pilots, harbourmasters and tug masters if present.
- When site-specific information is used, fenders will be slightly smaller than those determined using WG 33 (Roubos, et al., 2023). If local knowledge is ignored, fenders might be over designed.

Since the changes are quite substantial, there is a transition period between WG 33 and WG 211 to reorganise fender suppliers' catalogues according to the new guidelines, which is expected to require a significant amount of type approval testing. This period ends 1-5-2026 (two years after the WG 211 guideline publication). Designers can start using the new guideline but should be aware of what they are specifying for deliveries before this date.

Normal reference to this guideline should be made as: 'PIANC Fender Guidelines 2024', abbreviated as (PIANC) WG 211. The formal reference shall be: 'PIANC MarCom WG 211 report Guidelines for the Design, Manufacturing and Testing of Fender Systems 2024'.

1.2 Function of Guideline

This report provides guidance on designing, manufacturing and testing of fender systems to ensure these fender systems are sufficiently safe for both vessel and berth, reliable, durable and have an optimal total cost of ownership.

In addition, this guideline aims to enhance the knowledge level surrounding fenders (Chapters 2, 4, 8 & 9) and to guide the reader through the steps from initial design (Chapters 4, 5, 6, 7 & 8) up to a tested (Chapter 10) and installed (Chapter 11) fender system. The guideline includes best practice recommendations and takes into account the latest knowledge. However, WG 211 is not intended to be a tender/contractual document.

1.3 Scope of Guideline

This guideline focusses on fender systems for seagoing vessels, primarily tug assisted or equipped with thrusters, often supervised by a pilot and in all cases executing controlled berthing manoeuvres. It is not taking account of collisions, e.g. vessels crashing into structures (this will be covered by MarCom WG 215) and the design of protection structures (e.g. bridge piers), or guidance like on lock chambers (this will be covered by MarCom WG 206 Design of Navigation Locks to be published in 2024).

The guideline covers energy absorbing (mostly) rubber fenders in all shapes and designs, such as buckling and non-buckling, pneumatic and foam fenders. Recommendations primarily come from global experts in the field applying lessons learnt and sharing best practices.

In this guideline the term 'fender' or 'fender unit' is used to describe the energy absorbing element of a fender system. The term 'fender system' is used to describe a single 'fender system', including fender unit, chains, anchors, shackles, panels, etc. Fender system is also used as plural for a group of fender systems.

The guideline can be applied to small seagoing vessels that are not tug assisted or supervised by a pilot. This guideline does not provide design guidance for inland vessels and barges, since these types of vessels often moor against wood or even plain concrete walls (lock chambers) (Broos, van Schaik, & Huitema, 2013). This guideline does not deal with (inland) barges and their operations, however when these vessels berth on energy absorbing fenders, this should be considered. Typically, hull pressures are not an issue for inland barges, however the design should take account of the low freeboard at low tide, together with the typical square bow structure and need for safe access to shore. This guideline should be used with great care if the berth under consideration is only used by inland vessels and small barges. Some of the formulae used in the guideline may not be valid.

The design approach and the associated partial factors of safety proposed in WG 211 are developed for new structures. For assessing existing structures, the partial safety factors in this guideline are generally too conservative. Most codes and standards define three different reliability targets. I) For new structures; II) For upgrades/renovation; and III) For maintaining an asset in service.

The paper of (Roubos, Groenewegen, Ollero, Hein, & Wal, 2018) provides a practical method to adjust the berthing energy for assessing existing structures.

1.4 Climate Change

Climate change is a key issue influencing berthing conditions like wind and waves and subsequently fender system design. Suitable fenders increase the safety of a berth and can improve the resilience of existing port infrastructure. A good knowledge of berthing operations and navigation conditions in a specific port (Chapter 5) may offer the designer the chance to allow bigger vessels to use existing infrastructure (Roubos, et al., 2023), or enhance the operational window, instead of building new infrastructure.

Two of the key challenges with rubber are the carbon footprint and the as yet relatively limited possibility of recycling rubber material (Chapter 12). It is considered of utmost importance to design suitable fender systems with a long lifetime to reduce the carbon footprint of the berth.

1.5 PIANC Certified Fenders and PIANC Type Approval

Type approval is a certificate issued by an independent body that verifies that a specific fender fulfils the recommendations of this guideline. The responsibility for stating this type approval lies completely with this third party. It does not rest with PIANC (Chapter 10).

It is common practice for fender suppliers to claim they have 'PIANC certified' fenders. That cannot be the case since PIANC is not a certifying body. Any PIANC logo or stamp on a certificate is illegal. Fender suppliers can claim that fenders are designed (Chapter 6), produced (Chapter 9) and tested (Chapter 10) according to the recommendations set out in this guideline. However, fenders are never certified by PIANC.

Whether or not a fender system supplier is the manufacturer of the fenders does not matter from the perspective of responsibility. The supplier of the fender (system) is always responsible for all manufacturing quality issues.

1.6 Members of Working Group 211

Name	Representing	Company
Dr. Hitoshi Akiyama	Japan	Coastal Development Institute of Technology
Capt. Dennis Barber	UK	Dennis Barber Marine Consulting
Mr Erik Broos (Chairman)	Platinum Partner	Port of Rotterdam
Mr Marco Gaal	The Netherlands	Trelleborg
Mr Gary Greene	US	Gary Green Engineers
Mr Chistian Hein	Germany	Bremenports GmbH&Co.KG
Mr Rune Iversen	US	Simpson Gumpertz & Heger, Inc.
Mr Mishra Kumar	Platinum Partner	Trelleborg
Mr Patrick Lambiotte	Belgium	Besix
Mr Bob Lamont-Smith	Australia	E & PI Consultants
Mr Luis Lopez Gonzalez	Spain	Sipor21 / Aktys Hydraulics
Mr Masamitsu Ikebe	Japan	Shibata Industrial
Mr Vitomir Mihajlovic	Spain	Prosertek
Mrs Prasanthi Mirihagalla	UK	AECOM
Mr Svein Ove Nyvoll	Norway	Nyvoll Consult as
Mr Jeff Oskamp	US	Moffatt & Nichol
Mr Dominique Polte	Platinum Partner	ShibataFenderTeam
Mr Eduardo Rodero Aristoy	Spain	ShibataFenderTeam
Dr. Alfred Roubos (Secretary)	The Netherlands	Port of Rotterdam/ Delft University of Technology
Mr Sergej Vaughan	UK	Royal HaskoningDHV
Mr Harvinder Singh	Australia	JACOBS
Dr. Shigeru Ueda	Japan	Emeritus Professor Tottori University
Mr Arjan Van der Weck (Mentor)	MarCom	Boskalis
Mr Rob Williams (Editor)	UK	Waves Group
Mr NG Yeow Kiat	OCIMF	ExxonMobil / OCIMF
Dr. Haruo Yoneyama	Japan	Port and Airport Research Institute
Mr Soonhwan Yun	South Korea	Hwaseung Corporation

1.7 Meetings

WG 211 held four physical meetings before the COVID-19 pandemic interrupted the development process. Thereafter, seven online meetings were held, with three further physical/hybrid meetings held to finalise the guideline. All meetings were well attended, the full group has really contributed to the guideline.

1.8 Acknowledgements

The writing of this guideline was not possible without the support of the mother companies of the members, national PIANC committees and input and critical review of external experts amongst them:

DSME, Samsung Heavy Industries, Delft University of Technology, Royal Institution of Naval Architects, TNO, Lloyd's Register, Port of Dover, Port of Melbourne, Port of Rotterdam, Port of Valencia, Port of Wilhelmshaven, Port of Bremerhaven, national subgroups of WG 211 (UK, Japan, US, the Netherlands), Coastal Development Institute of Technology. Knowledge Centre for Manoeuvring in Shallow and Confined Water (MASHCON) – cooperation Ghent University and Flanders Hydraulics, Arcadis, SBE, Witteveen+Bos, BS6349-4 committee.

Jaap Havinga from KIWA, Dr. Alan Muhr of Tun Abdul Razak Research Centre, Ben Bullock and Hari Panchumarthi from Jacobs, Elizabeth Eldridge from AECOM, Perry Groenewegen from Royal HaskoningDHV, Peter Hunter from HR Wallingford. Grard Blankers, Leon Groenewegen, Master graduating students from Delft University: Felix (Orlin), Emma Berendsen, Lucie Roussel, Thomas IJzerman & Iris Heemskerk.

1.9 Use of the Guidelines

The flowchart below presents an overview of how this guideline is structured and can be used.

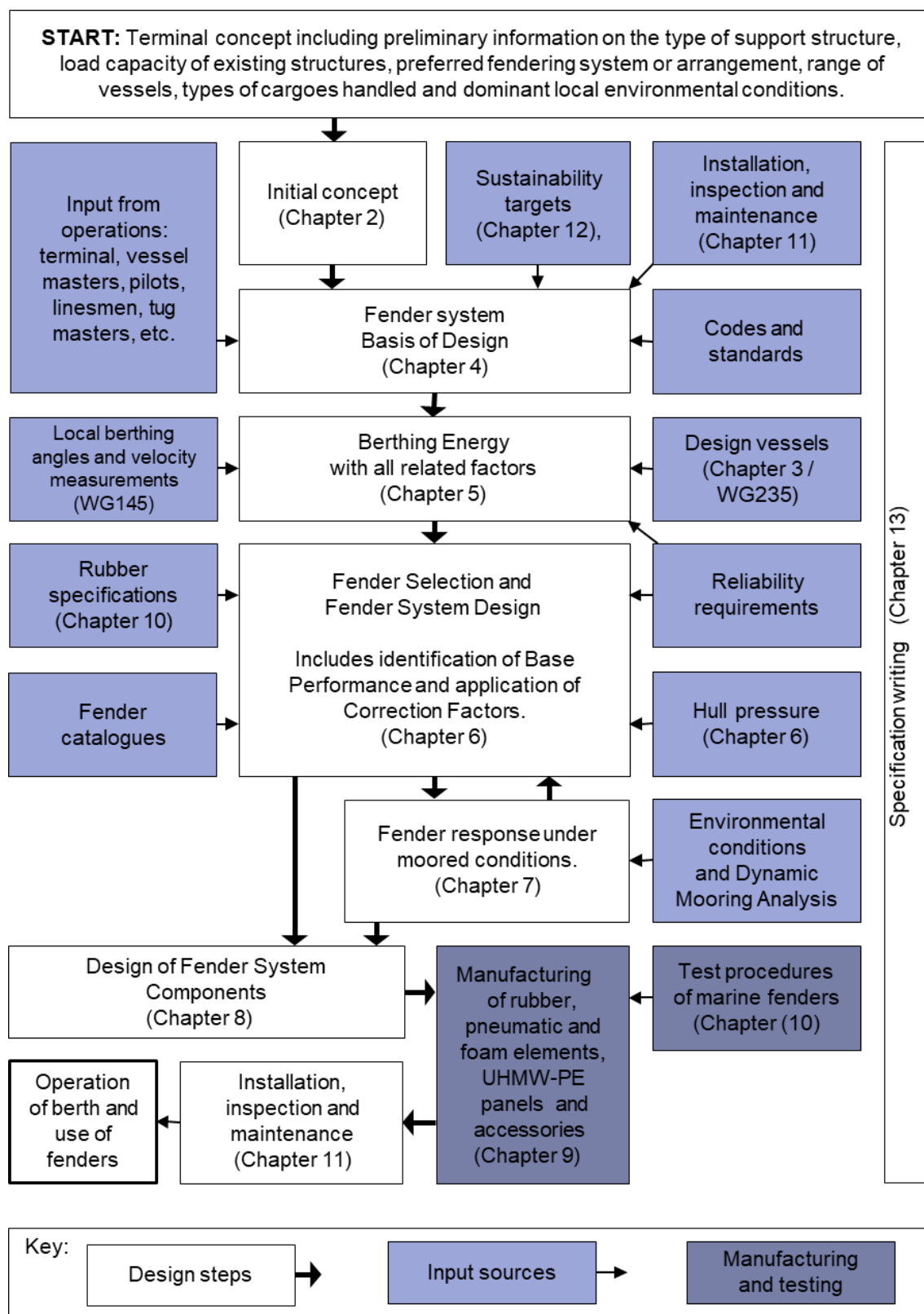


Figure 1.1: Flowchart WG 211 guideline

2 INTRODUCTION TO THE PRINCIPLES OF FENDERING

This chapter provides an overview of the typical characteristics of a range of fenders and fender systems currently in use around the world. It also provides an overview of the general fender selection and design approach.

This overview covers many commonly applied fender types and fender systems in operation at a wide variety of ports and marine facilities however is not considered to be exhaustive.

2.1 General Design Approach

Fender systems in general are part of marine structures, providing an interface between berthing vessels and the supporting structure.

The principal function of a fender system is to absorb the vessel berthing energy and dissipate the associated reaction forces into the supporting structure. A correctly designed fender system will bring the berthing vessel to a complete and safe stop without damaging the vessel hull, the supporting structure or the fender unit. Once the vessel is moored, the fender system should also be able to protect the vessel and the supporting structure from additional loads induced by wind loads and by vessel motions caused by wind, waves, currents, tidal variations, passing vessels and the loading or unloading of cargo. Special consideration is required for the accommodation of cyclic loading (especially on permanently moored vessels) and should be considered separately.

Shear forces resulting from the movement or warping of vessels along a berth, and rotating onto and off the berth, must also be allowed for in the fender system design.

The design of fender systems shall also consider the uncertainties during berthing. Consideration should be given to the consequences of fender failures, frequency of use, type of cargo and a range of other factors as discussed in Chapter 4.

2.2 Fender Types and Systems

Fenders can be categorised according to the mode by which they absorb or dissipate the kinetic energy of berthing and/or moored vessels.

Rubber is used as the fender energy absorbing element in most cases because it is viscoelastic with a loading unloading curve hysteresis and full strain recovery. The area between the loading and unloading curve is the strain energy converted into heat. When fenders are tested the energy absorption is obtained by measuring this area. If a linear elastic material was used as a fender (e.g. steel below its yield stress), the vessel would bounce off the berth due to the vessel's kinetic energy being converted into steel elastic energy and back into vessel's kinetic energy with no losses. If a plastic material was used (e.g. steel past its yield), then the conversion of vessel's kinetic energy into steel plastic strain energy will convert most of the kinetic energy into heat, which is desirable, however there will be incomplete strain recovery, i.e. the only deformation recovered will be the elastic range, which is inconvenient as the fender system will then have a permanent set back. Rubber is not the only viscoelastic material that is used for fendering. Others include polymer foams, bitumen, oil, wood and soil, however rubber is the most efficient. For example, isolated timber fender piles use the energy absorption properties of the soil plus wood. Another example is the shock absorbers in motor vehicles which use the turbulent flow of oil to convert kinetic energy to heat energy.

Fender systems are designed as a combination of different fender elements, steel panels and accessories including chains and anchors. The designer is required to consider a wide range of variables, options and alternatives, operational constraints and design criteria when selecting the appropriate fender and designing the required fender system.

Fender selection depends on a variety of factors and is not limited to fender performance alone. The selection of the correct fender is a significant element of developing the correct fender system design.

Table 2-1 provides an overview of common fender types currently used in the industry. Typical values of the fender size, energy absorption and reaction force are provided for information and relative comparison only and should not be used for fender selection or design purposes. Further details on these fender units and other types are provided in Section 2.4. The various energy absorption and reaction force values and ranges are based on a variety of hardness grades that high performance fenders are available in. Hardness grades will be discussed later in this document, however it should be noted that the stiffer a fender, the higher the energy absorption and reaction force for a given deflection.

The properties listed in Table 2-1 and Section 2.4 are based on data published by fender suppliers relating to berthing impacts perpendicular to the berthing line. This data is current at the time of publication.

Fender systems can also have bespoke, limited applications. These fenders may not have been widely adopted or are customised for one specific project, e.g. fully submerged fenders, temporary fenders or fenders for specific shipyard applications.

There is no single solution to all fendering challenges. Designers should be aware that simplified fender selection tools may lead to generic designs. Combinations of vessels, navigational limits, types of supporting structure, end user preferences and particular berthing conditions present a range of different operational requirements and constraints. Vessels at highly automated container terminals will have very different demands on the fender systems than those at ferry terminals with a high frequency of berthing.

Cone Fenders	Cell Fenders	Element/Leg Fenders
		
Size: h_f 300 - 2500 E _A : ~ 10 - 9500 R _F : ~ 55 - 7200	Size: h_f 400 - 3000 E _A : ~ 10 - 9800 R _F : ~ 50 - 8850	Size: h_f 250 - 2000 E _A : ~ 10 - 1300 R _F : ~ 80 - 1750 <i>Performance values for single unit of 1,000 mm length</i>

Arch Fenders  Size: h_f 150 - 1000 E_A: ~ 5 - 450 R_F: ~ 90 - 1050 <i>Performance values for single unit of 1,000 mm length</i>	Foam Fenders  Size: D 700 - 4200 L 1500 - 8500 E_A: ~ 30 - 9000 R_F: ~ 130 - 8500	Pneumatic Fenders  Size: D 500 - 4500 L 1000 - 12000 E_A: ~ 6 - 9050 R_F: ~ 65 - 10500
<i>All sizes are in mm E_A in kNm R_F in kN Shapes might differ from manufacturer to manufacturer</i>		

Table 2-1: Typical fender types

Fender selection also depends on the properties of the supporting structure. The design of a fendering system has to be integrated with that of the berth structure as not all types of fenders are compatible with all types of structures.

All fender types are characterised by their energy conversion capacities. Fender manufacturers are constantly conducting research and development into improvements to rubber fenders, compounds and components. Designers are therefore recommended to consult fender suppliers to adopt a holistic approach and to consider the entire fender system in the fender selection process.

Designers should also allow for variations in the range of energy absorption and reaction forces of the selected fenders, as performance characteristics for similar fender types and sizes can vary from manufacturer to manufacturer. This is particularly important when considering load-sensitive structures.

2.3 Fender Efficiency

Fenders can be divided into two distinct classes:

Force transmission: Solid fenders such as D-Shape and extruded rubber profiles that primarily transmit forces directly to the supporting structure, with small energy absorption capability. It is acknowledged that other materials, such as tyres and timber, are used as fenders in certain circumstances. In the context of these guidelines, such solutions are considered as force transmission fenders. Consequently, these solutions are generally not covered by this WG 211 guideline. If it is only for very small vessels, D-fenders can be designed as energy absorbing fenders as well. Suppliers provide the necessary data.

Energy absorbing: This class of fenders is covered by this guideline and are sub-divided into two main categories, buckling and side-loaded fenders.

2.3.1 Types of Buckling Fenders

Buckling fenders typically consist of cone, cell, element or leg fenders. The typical indicative load deflection curve of these types of fenders is as indicated in Figure 2.1. The illustrated deflection behaviour provides substantially greater energy absorption capacities, when compared to a sideloaded fender.

A buckling fender typically has an energy capacity in the region of 70 to 80% of the theoretical maximum characteristic energy absorption. The characteristic energy absorption is calculated based on assuming the maximum reaction is maintained throughout the compression. A consequence of this greater overall efficiency is that the maximum reaction force is reached early on in the compression cycle at about half the design deflection, when compared to a side loaded fender.

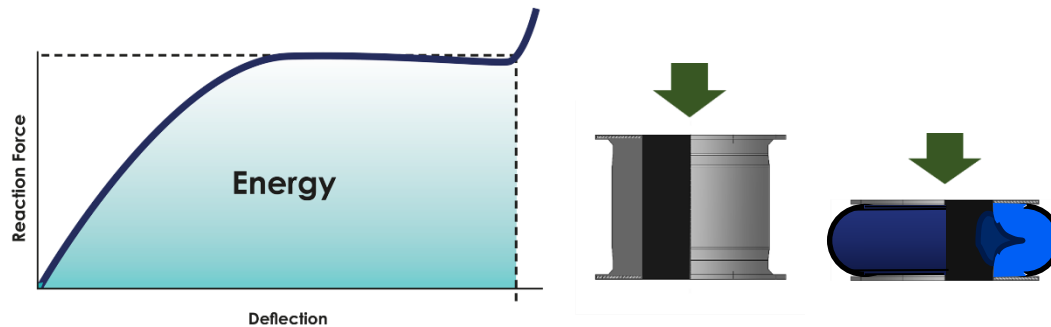


Figure 2.1: Typical buckling fender deflection curve

2.3.2 Types of Sideloaded Fender

Sideloaded fenders include cylindrical, pneumatic and foam fenders. The typical indicative load deflection curve of these types of fenders is shown in Figure 2.2. These fenders are characterised by a constant, almost linear, increase in fender compression and reaction force. This deflection behaviour is often beneficial for accommodating smaller vessels and other specialist applications, due to lower resultant reaction forces for smaller deflections.

The efficiency of these fender types is limited to approximately 50-60 % of the theoretical maximum energy absorption, calculated based on the area under the curve.

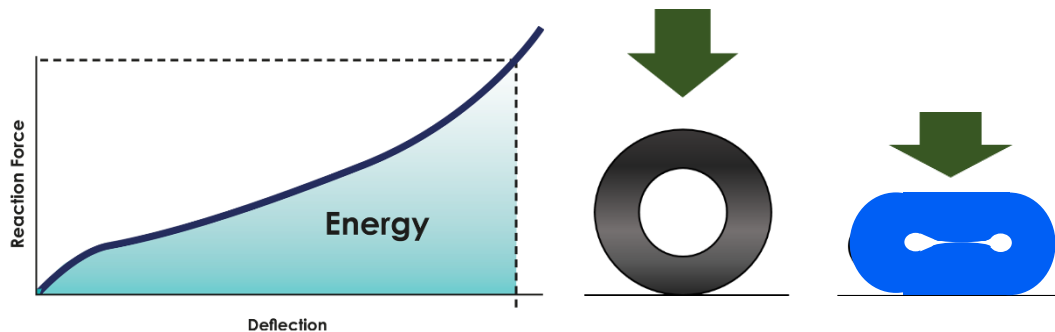


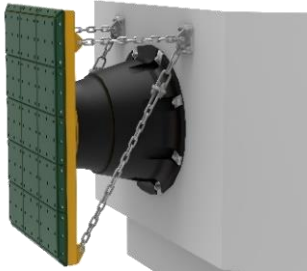
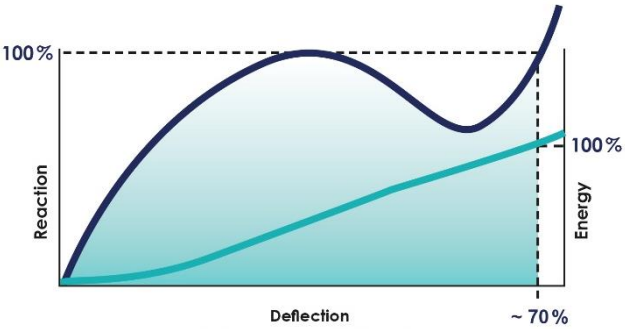
Figure 2.2: Sideloaded fender deflection curve

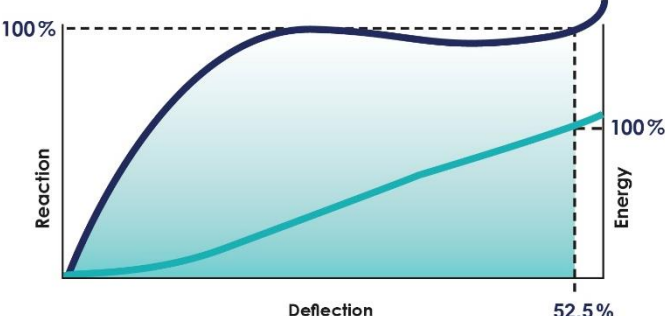
When comparing both types of fenders, designers should be aware of their respective limitations, stand-off requirements and the relative performance of each fender type. The most suitable fender, be it buckling or sideloaded fender, should be selected based on the required project specific design criteria.

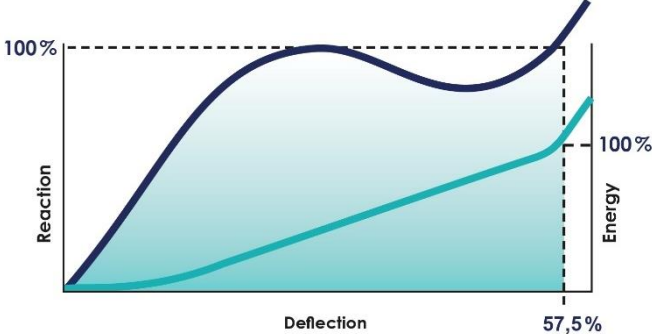
2.4 Fender Selection and Fender System Design

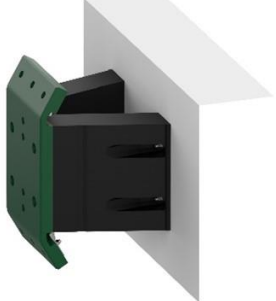
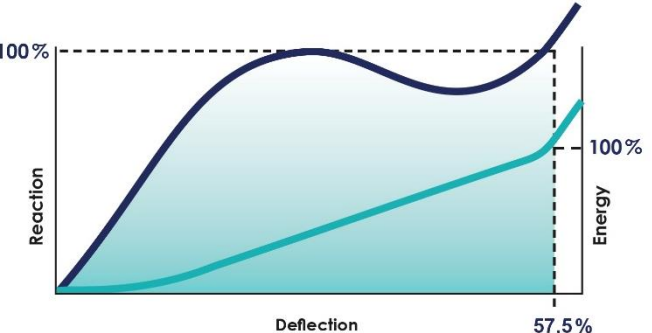
Fender selection and fender system design should be based on the operational and environmental conditions specified for the project site. An overview of the typical characteristics of a range of different fender types is included below.

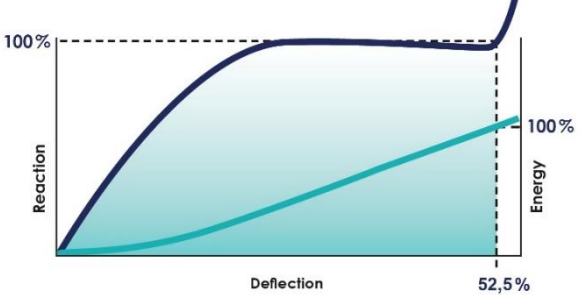
2.4.1 Buckling Fender Systems

A) 	B) 	
Size (H): 250 – 2500 mm	Energy: ~ 10 – 9500 kNm	Reaction: ~ 55 – 7200 kN
The cone fender is an efficient fender that is used in a wide variety of applications. It has a conical body with the larger diameter end of the fender mounted on the supporting berth structure and rubber-encased steel mounting flanges. The typical design deflection is approximately 70 % of the height of the cone fender. The cone fender offers good energy absorption to reaction force ratio (E/R) and high shear stability, with no loss of performance up to a 10° contact angle. Typical applications include container and bulk terminals, oil and gas terminals, general cargo terminals, cruise terminals, ferry and RoRo terminals, and naval berths.		
Table 2-1: Typical Cone Fender system (A) and Performance Curves (B)		

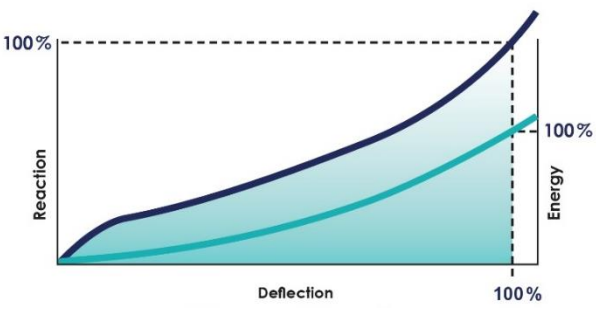
A) 	B) 	
Size (H): 400 – 3000 mm	Energy: ~ 10 – 9800 kNm	Reaction: ~ 50 – 8850 kN
The cell fender is a well-established fender type, with a hollow cylindrical body and rubber-encased steel mounting flanges. The cell fender is designed to deflect in an axial direction up to 52.5 % of its original height. The cell fender is a robust and durable fender which is typically easy to assemble and install and has a good resistance to shearing. Typical applications include container and bulk terminals, oil and gas terminals, general cargo terminals, cruise terminals, ferry and RoRo terminals and naval berths.		
Table 2-2: Typical Cell Fender system (A) and Performance Curves (B)		

A) 	B) 	
Size (H): 250 – 2000 mm	Energy: ~ 10 – 1300* kNm	Reaction: ~ 80 – 1750* kN
Element or leg fenders are modular rubber units with embedded mounting plates combined with steel fender panels. They are based on V-Type Fenders however each leg is separate. The modularity of the fender provides good flexibility and options for the arrangement of the fenders when installed with closed box steel panels or where mounting space is limited. Element fenders can deflect up to 57.5 % in the axial direction. The element or leg fenders are considered to have a good energy absorption to reaction force ratio (E/R) and have a modular and compact design with a small footprint. These fenders can be vertically and/or horizontally mounted as part of a combined fender system. Single element or leg fender units can also be used in combination with fender pile designs, for easy maintenance and replacement. Typical applications include container and bulk terminals, general cargo terminals, RoRo terminals, navy berths and barge berths.		
*Performance values for single unit of 1,000 mm length		
Table 2-3: Typical Element/Leg Fender system (A) and Performance Curves (B)		

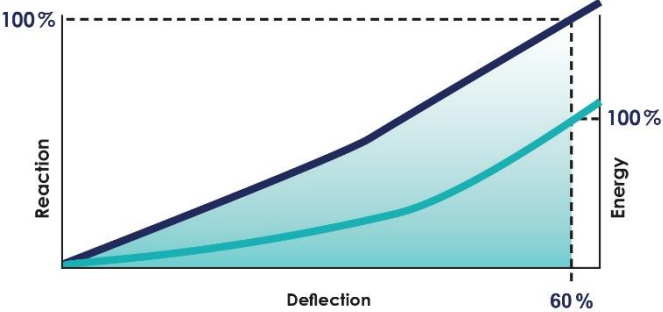
A) 	B) 	
Size (H): 250 – 1250 mm	Energy: ~ 10 – 730* kNm	Reaction: ~80 – 1270* kN
Special element fenders combine rubber Element/Leg Fenders with low friction frontal shields directly fixed to the fenders and belting deflector if it is necessary. They are also designed to deflect about 57.5 % in an axial direction. The special element fenders are available in different sizes / widths and with different colour options of UHMW-PE plates which makes them a flexible system; they offer low friction facing and are considered an alternative to steel panels. Typical applications include container and bulk terminals, jetties and berths for small to medium size vessels, general cargo terminals and naval berths.		
*Performance values for single unit of 1,000 mm length		
Table 2-4: Typical Special Element Fender system (A) and Performance Curves (B)		

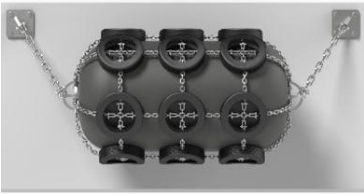
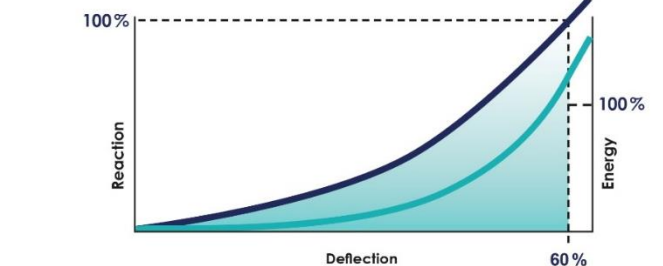
	B) 
Size (H): 150 – 1000 mm	Energy: ~ 5 – 450* kNm Reaction: ~ 90 – 1050* kN
Arch fenders are a rigid, moulded one piece type of fender. They are available in different cross sections and lengths. The main difference in the cross sections is the width of the fender head and the resulting vessel contact surface area. Arch fenders can be equipped with an additional embedded steel plate in the fender head that allows for the installation of low friction frontal plate, a steel fender panel, or the mounting of the fender unit behind a fender pile installation. Arch fenders are designed to deflect about 50 % in an axial direction. Arch fenders are considered to be very robust, durable and require minimal maintenance. These fenders have a high shear resistance in the longitudinal direction and can be installed both vertically and horizontally onto the supporting structure. They are ideally suitable for turning dolphins and pivot points. The main disadvantage is the relatively small contact area, resulting in high hull pressure values. Typical applications include general cargo terminals, multi-purpose-terminals, RoRo ramps, passenger vessel terminals, and berths for barges and tugs.	
*Performance values for single unit of 1,000 mm length	
Table 2-5: Typical Arch/Trapezoidal Fender system (A) and Performance Curves (B)	

2.4.2 Side Loaded Fender Systems

A) 	B) 
Size (H): 100 – 2800 mm	Energy: ~ 1 – 850* kNm Reaction: ~ 45 – 1500* kN
Cylindrical fenders were the first fender type to be produced with a defined performance. Installation is simple using chains, bars, ropes or specially designed ladder brackets, depending on the fender size and supporting structure. As the cylindrical fender is compressed, the energy absorption and associated reaction force increase proportionately up to the characteristic deflection of 100 % of the inner diameter. This can be advantageous in providing lower reaction forces, resulting in a comparatively 'softer' berthing. The cylindrical fender is very robust, simple and easy to install and has a high abrasion resistance. The main disadvantage is the relatively small contact area, resulting in theoretically high hull pressure values. Typical applications include multiple usage at different berths, container and bulk terminals, general cargo terminals, RoRo terminals, and berths for tugs and barges.	
*Performance values for single unit of 1,000 mm length	
Table 2-6: Typical Cylindrical Fender system (A) and Performance Curves (B)	

2.4.3 Side Loaded Floating Fenders

A) 	B) 	
Size (OD): 700 – 3000 mm (L): 1500 – 6500 mm	Energy: ~ 30 – 850 kNm	Reaction: ~ 130 – 3351 kN
Foam fenders are best suited to applications with large variations in water level. They are also considered to be good solutions for cruise terminals, due to the non-marking surface and the ability to adapt to different hull shapes. Foam fenders are fabricated using a resilient, energy absorbing, closed cell foam. A skin covering the foam core is constructed from a continuously wound fabric cord of reinforced polyurethane elastomer. Foam fenders can be fitted with or without a protective chain and tyre net. Typically, these fenders are mounted against a smooth substructure to reduce wear and uneven deflections. Foam fenders are designed to deflect about 60% of their outside diameter. As well as the traditional foam fender, there are other foam fender types, such as 'Donut' fenders, small foam fenders with thru-ropes or plastic pipes to suit a variety of applications. Foam fenders also provide a constant relationship between increasing compression and increasing of reaction force. Foam fenders typically exert low hull pressures and can provide submerged contact area (submarine type foam fenders only) and can be made available in various grades of foam. Typical applications include navy vessel berths, cruise terminals and ship-to-ship operations, locks and dry dock entrance.		
Table 2-7: Typical Foam Fender system (A) and Performance Curves (B)		

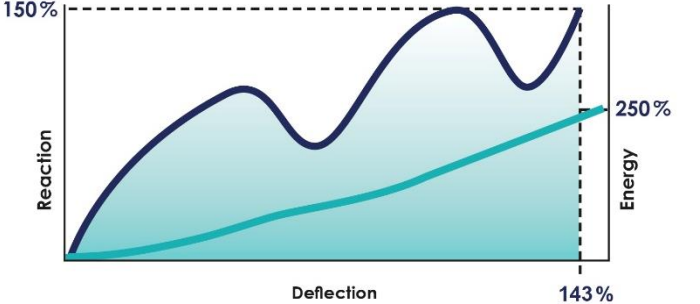
A) 	B) 	
Size (OD): 500 – 4500 mm (L): 1000 – 12000 mm	Energy: ~ 6 – 9050 kNm	Reaction: ~ 65 – 10500 kN
Pneumatic fenders are suited to applications with large variations in water level. They are also considered to be good solutions for ship-to-ship operations. The pneumatic fender has a cylindrical shape with hemispherical ends. The fender body consists of several rubber layers with fabric tyre cord reinforcement layers. The fender is fully vulcanised and filled with air. Pneumatic fenders are available with and without a protective chain and tyre net. For submerged applications, hydro-pneumatic fenders are available, which are partly filled with air and water and positioned with a counterweight. The governing standard for these fenders is ISO17357 and they are typically mounted against a smooth substructure to reduce wear and uneven deflections. Pneumatic fenders are designed to deflect about 60 % of their diameter.		

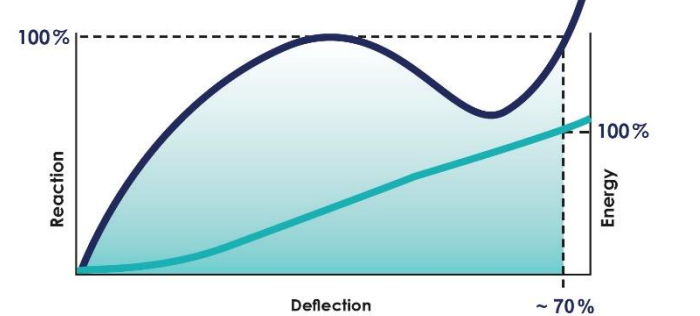
Pneumatic fenders provide a constant relationship between increasing compression and increasing of reaction force. Pneumatic fenders typically exert low hull pressures and can provide submerged contact area (hydro-pneumatic fenders only) and can be provided with different initial air pressures.

Typical application includes ports with extreme tidal variations, ship-to-ship operations in oil and gas (typically FSRU, floating storage and regasification unit), temporary berthing, submarine jetties (hydro-pneumatic fenders only).

Table 2-8: Typical Pneumatic Fender system (A) and Performance Curves (B)

2.4.4 Special Applications

A) 	B) 	
Size: customised	 Energy: customised	 Reaction: customised
Parallel motion fender systems (PMFs) are individually engineered systems and can be designed to utilise several different fenders. Typically, cone, cell or element fenders are used. To maintain a vertical panel face at all levels of compressions, turning lever arms (torsion arms) are mounted between the support structure (concrete or steel) and the frontal steel panel. The arms restrain the panel movement during the entire fender compression, allowing it to move only parallel to the support structure, irrespective of impact level and berthing angle. The main advantages of PMFs are the avoidance of a second contact on the vessel's hull, as well as a substantially increased energy absorbing capacity. For designs that focus on the hull impact only, single fender units could be used. In cases where an increased energy absorption is required, back-to-back cone fender configurations could be used as these substantially increase the energy absorption of the fender system, while keeping the reaction force low. PMFs should only be used for controlled berthing environments as the system is generally more prone to damage if used incorrectly. Typical applications include bulk terminals, oil and gas terminals, ferry and RoRo terminals, mono-piles and load-sensitive structures.		
Table 2-9: Typical Parallel Motion Fender system (A) and Performance Curves (B)		

A) 	B) 	
Size: customised	 Energy: customised	 Reaction: customised

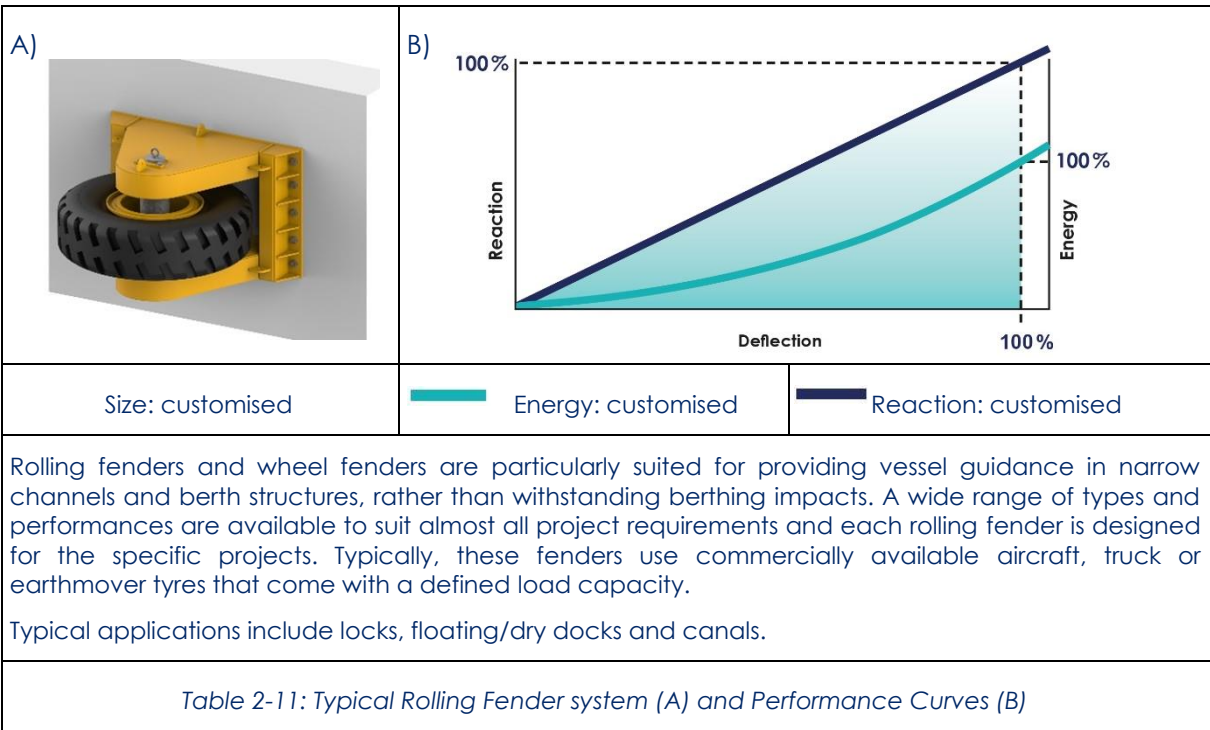
Pile or pivot fender systems are a common alternative to parallel motion fenders and can be designed to utilise a variety of different fenders. Typically, cone, cell or element fenders are used with a pile or pivot fender system.

Pile/pivot fender systems rotate at a level well below the water level where the lower tip of the fender frame is fixed either into a shoe on the seabed or by welding a support bracket to the support structure. Pile/pivot fender systems can provide a single point contact between the vessel and fender system at any water level.

The energy absorbing fender is typically positioned as close as possible to mean sea level to provide the optimum energy absorption at a range of water levels. The panel is typically inclined to the vertical to accommodate vessel belting and differences in vessel draught between unladen and fully laden conditions.

Typical applications include bulk terminals, oil and gas terminals, ferry & RoRo terminals, terminals with sheet pile walls or pile structures with concrete capping beam.

Table 2-10: Typical Pile/Pivot Fender system (A) and Performance Curves (B)



2.5 Ship-to-Ship Fendering

Fendering is also required for operations that are performed directly between vessels or other floating structures. These are known as ship-to-ship (STS) operations and require ship-to-ship fendering. STS operations can include berthed, sailing or anchored vessels to enable operations such as cargo, supplies or personnel transfer. Vessels can be berthed one against another in some special ports or under special conditions (e.g. naval facilities, oil and gas terminals, offshore industry).

There are some special vessels designed to perform STS operations, such as floating storage and regasification/ liquefaction units (FSU, FSRU, FSLU), bunkering vessels and shuttle tankers. These vessels require specialist fender systems to be designed to separate the vessels and prevent hull and vessel contact damage.

For additional detail on STS fender selection, refer to Section 6.9.1 in these guidelines.

2.6 Fenders and Structures

Flexible dolphins (typically mono-piles) have very different design requirements compared to closed quay walls or bulkhead wall retaining (semi-closed) structures. These differences in requirements influence the design, selection and type of fenders that are adopted for flexible dolphins. So a flexible dolphin is actually part of the fender system, a quay wall is not.

Fenders mounted on flexible dolphins are designed to absorb a portion of the berthing energy. The remainder of the berthing energy is absorbed by the deflection of the structure. The proportion of energy absorbed by the fender and structure can vary and this is determined as part of the detailed design process and is a function of the operational requirements of the flexible dolphin. For additional detail on fender selection for flexible structures, refer to Section 6.4.12 in these guidelines.

Fenders can also be installed on floating structures ('spacer barges'), between the berthing vessel and the berth structure. These 'spacer barges' or 'camels' might have fenders installed on both sides to allow contact with the hull of the vessel and with the berth structure. For more information refer to UFC 4-152-01 DESIGN: PIERS AND WHARVES (2017).

Special consideration should be given to fenders installed at lock entrances, dry docks and ship lifts. For these applications, a variety of fenders could be used, however typically donut or wheel fenders or guide-walls made of steel fender panels with different rubber units are used. For further information refer to WG 206 – 'Design of Navigation Locks' (in preparation).

3 PARTICULAR ASPECTS REGARDING DESIGN VESSELS

This chapter provides information on vessel types and hull shapes relevant to the design of fender systems and should be read in conjunction with WG 235 – ‘Ship Dimensions and Data for Design of Marine Infrastructure’ (PIANC WG235, 2022).

3.1 Vessel Characteristics

3.1.1 Overview

Vessels are designed for specific purposes and thus their designs vary with their function, cargo, economic market, etc. Key parameters in vessel design can be driven by different objectives such as:

- Maximising the loaded cargo by weight
- Shipping cargo that is defined by volume
- Speed of transit
- Capacity
- Nature of the cargo (value, hazards, parcel size, etc.)
- Fuel efficiency
- Loading & unloading requirements
- Limitations of major canals such as Suez and Panama on beam, draught and length
- Port depth limits
- Port crane or ship (un)loader reach that limit beam
- Safety
- Manoeuvrability

Vessel sizes are not a smooth linear change with increasing size. Sizes distributions are driven by industry demands that are constantly changing. This results in step changes and gaps in the size distribution.

3.1.2 Container Vessels

Container vessels are designed for a high transit speed and rapid loading/unloading. This results in a sleek hull shape with a substantial proportion of the containers stacked above the top deck. Overall vessel dimensions are driven by the standard container dimension and the number of containers to be transported.

Since container vessel beams are a multiple of container widths there is a larger variation in vessel dimensions for the same gross tonnage (GT). At smaller sizes, the variation is larger and significant. Hence the envelope of maximum dimensions can look odd when compared with an individual vessel in that size range. The block coefficient is also variable due to the range of different beams.

The design of fenders for container vessels requires special consideration to be given to the hull flare and the relatively short parallel body length (the flat side) for fenders to support the vessel's hull. The shorter contact length can have consequences on:

- Eccentricity factor C_e that may be higher than other vessels with impact closer to the vessel centre of mass, especially at modest berthing angles.
- Larger fender loads from vessel yawing motion when moored alongside the quay.

- Projection of the ships deck over the quay line due to both horizontal and vertical hull flare when berthing at higher angles. It is recommended this be discussed with pilots and port operators to establish safe set-backs for quay cranes or safe stand-off distances for fenders.

The relatively short parallel mid-body of a typical container vessel is shown in Figure 3.1.



Figure 3.1: Parallel mid-body of a typical container vessel hull

If a vessel has a modest horizontal berthing angle of less than around 5 degrees, its initial fender contact point will be close to the bow tangent line. Depending on the berthing angle, compression and spacing of the fenders and the size of the vessel this may be on the parallel or flared bow side of the tangent line. Contact in the bow area has both a vertical flare angle and a horizontal curvature on the plane of contact point that can lead to a reduced energy absorption capacity of the fender system (refer to Chapter 0).

The elevation of fenders relative to the vessels deck will influence the hull contact point. Consideration needs to be given to quay/ fender levels relative to water level at all tides and the range of different vessel deck heights at arrival.

Some smaller container vessels may have hull belting (possibly non-continuous). This should be considered in selecting and designing the system.

Typically, container berths would be expected to be reasonably well protected because of the requirements to ensure small vessel motions for safe crane operations. In major ports, the number of tugs used would be expected to be sufficient to ensure low berthing angles are achieved. This may not be the case in all terminals and should be checked with the port operator. Additional information on container vessels can be found in WG 235 and WG 186 (PIANC, 2024).

3.1.3 Dry Bulk & Ore Carriers

Dry bulk and ore carrying vessels are designed primarily to maximise the cargo load to reduce the transport cost/tonne and speed is less important. The result is a much more prismatic hull shape with long parallel hull sides and lower cruising speeds of around 14 knots.

Dry Bulk terminals typically service large vessels and thus can have large fenders and larger fender reaction forces. This requires fenders to be fitted with fender facing panels to ensure loads on the hull are sufficiently distributed to meet the hull pressure and line force limits set out in Section 6.8.

There is a large difference between the draught in ballast and laden conditions. Fenders should be able to cater for laden vessels at low tide through to ballast condition at high tide. This difference in draft is much greater for bulk carriers than for other vessels.

3.1.4 Cruise Vessels

Cruise vessel design is driven almost entirely by function however also the need for speed and stability.

Most larger cruise vessels have bow thrusters and some have Azimuth drive propellers that enable great manoeuvrability and control over lateral berthing speed. It is not unusual for these vessels to berth at small ports with one tug or no tugs in good conditions. The draught of cruise vessels does not vary much with loading.

The bows of cruise vessels have large vertical and horizontal flare angles. The parallel hull length for cruise vessels varies with design; typically the bow shape will be similar to container vessels however the stern shape will be less tapered than container vessels.

Cruise vessels can also have hull features hazardous to fenders such as belting (that may be discontinuous and/or may be angled along the vessel hull), shell doors that need to open for provisioning and pilot doors that may be recessed in the hull. These features can hook the edges of fenders causing damage and even destruction of the fenders.

Additional information on cruise vessels and fendering for these vessels can be found in WG 152 (PIANC, 2016).

3.1.5 RoRo, RoPax & Vehicle (Car) Carriers

RoRo and car carrier vessels primarily carry mobile 'drive on and drive off' cargo. This includes cars, trucks, and other motorised vehicles and equipment. The parking decks on these vessels are enclosed and the vessel will typically have its own vehicle ramps at the stern (there are a small number of exceptions).

Some RoRo vessels are a mix of enclosed deck and cargo deck (usually containers) and as a result they have a lower GT. There is also significant variation in beam and length for similar displacement vessels.

Characteristics of these vessels include:

- Significant variation in dimensions. An envelope of maximum lengths, beams or draughts can look odd when compared with an individual vessel in that size range.
- Large range in vessel block coefficient C_b .
- High windage area.
- Large hull flare at the bow then parallel sided to the stern.
- Volume based cargo carriers so rarely operate at the maximum scantling draught.

The values for displacement in the PIANC WG 235 Ship Data Tables are estimated P90 values at the full summer draught. For any given capacity, the values for displacement can vary with vessel design. Lower displacement values may be applicable as typical draughts can be 10 to 15 % lower than the permitted summer draught. It is recommended that data should be obtained from the applicable Port Authority to determine design values for draught and displacement of RoRo and Car Carrier vessels.

Hull belting is present on the majority of RoRo ferries and also some other RoRo vessels. Additional information on belting on RoRo and RoPax vessels and fendering for these vessels can be found in Section 3.1.9 of this guideline and in WG 167 – 'The Design of Terminals for RoRo and RoPax Vessels' (2023).

3.1.6 Tankers

Tanker vessels include crude oil tankers, product tankers, chemical tankers and specialist tankers used for specific products.

Generally, the length and beam of all tanker types for a given deadweight (DWT) does not vary much, however, some special tankers may have different draughts where the product densities vary from the major liquid products.

As noted for bulk carriers, tankers have quite prismatic shapes and their hull capacity should be considered in the design of the fender system. There is similar to bulkers a significant difference between the draught in ballast and laden conditions. Fenders should be able to cater for laden vessels at low tide through to ballast condition at high tide.

3.1.7 Gas Carriers (LPG and LNG)

There are two main types of gas carriers, LNG (Liquified Natural Gas) that carry gas as a cryogenic liquid and LPG (Liquified Petroleum Gas) that may be carried as a cryogenic liquid or pressurised liquid and gas. Gas carrier sizes are based on their cubic metre gas capacity.

LNG Carriers

There are two primary designs for LNG vessels based around the tank design:

- Membrane type with prismatic tanks
- Moss with distinctive spherical tanks

The hulls of these vessels do not form part of their tanks.

LPG Carriers

LPG carriers are often smaller than LNG vessels and look more like oil tankers.

Safety considerations around explosion, fire and spills dictate design of gas terminals that are standalone berths with exclusion zones. The berth arrangement is typically similar to oil terminal arrangements with breasting dolphins protecting the loading/unloading platform and mooring dolphins set back behind the berth line.

The change in the draught of gas carriers between loaded and ballast condition is small because of the low density of the cargo.

Softer fenders are preferable for gas terminals and should be designed to keep contact hull pressures within the guidelines outlined in Section 6.8.

3.1.8 General Cargo, Refrigerated Cargo and Livestock Carriers

General cargo vessels are smaller vessels that carry a mixed variety of cargoes. Some cargoes are containerised and others, not suited to containerisation either because of the nature of the cargo or the volume of the local trade, are loose. In many cases, the vessels are multipurpose cargo carriers and often the vessels are 'geared' self-loading and unloading vessels.

Some smaller general cargo vessels may have side belting on the hull. Where belting is present the design needs to take into consideration the small contact area and the potential line loads, as for some fender types, this will not engage the full capacity of the fender.

Refrigerated cargo carriers and livestock carriers also fit into the size range of general cargo vessels however typically will have a greater wind area.

Many smaller general cargo vessels may include side hull belting and that should be considered in fender design.

3.1.9 Passenger Ferries

Pure passenger ferry designs vary considerably as there are a considerable number of different requirements such as:

- Number of passengers
- Speed
- Route travel distance
- Time of travel and impacts on passenger amenities required (e.g. overnight cabins, restaurants)
- Associated need for cargo and vehicles
- Environment – ocean, lake, river, etc.
- Operating sea state associated with the route, including ice risk

Hull designs include monohulls, catamarans/ SWATHs (Small Waterplane Area Twin Hull) and some trimarans. Large ferries are usually steel monohull designs. Many high-speed ferries are constructed in aluminium and some smaller ferries can be fiberglass (reinforced plastic) or wooden hulls. It is quite common for ferries to have belting on the side of the hull.

Reliable data on the hull configuration of ferries is difficult to obtain so when designing ferry terminals and fenders for such berths, it is essential to obtain specific vessel information from the facility operator.

Many ferries are fitted with belting and the design needs to take the small contact area into consideration as for some fender types this will not engage the full capacity of the fender. Generally, where ferries berth side-on, it is better to arrange fenders in a vertical configuration to cover the potential variation on the impact height due to tide, sea state and variable ferry size/ types. Vertical fender piles can be placed in front of the fender to protect the fenders from belting and manage variations of the impact height.

Parallel motion fenders are often used for high frequency terminals and those with large tidal ranges. Manufacturers of parallel motion fenders provide guidelines for the design, installation and maintenance of these fender types.

Where a ferry terminal is designed for end-on (bow or stern) berthing it is essential to have details of all vessels berthing at the terminal to ensure the arrangement is suitable for all of those vessels.

3.1.10 Fishing Vessels

There are many thousands of fishing vessels of all types working the world's oceans. These also include fish carriers, fish factory vessels, research and patrol vessels and special purpose fishing vessels.

Designers should obtain clear directions from owner/ operators on the range and details of fishing vessels to be accommodated at port fishing facilities.

In designing fenders for fishing vessels consideration should be given to hull belting that is common on some smaller vessels.

3.1.11 Offshore Supply Vessels & Harbour Tugs

Virtually all offshore supply vessels have side belting that may be vertical, inclined or horizontal depending on the location along the hull. They also have a large, flared freeboard forward reducing to a relatively low freeboard at the rear loading deck. Where these vessels have dedicated berths the use of tall fender piles is often the best fendering solution. Some facilities use large tractor tyres hung over fender panels to protect the panels from the belting.

The hulls of harbour tugs typically have a low freeboard and continuous belting along the hull and cushion protection (beard) at the bow for pushing vessels. Rubber rubbing strips are usual either side of the beard. Sides are often fitted with chain hung tyres. Modern harbour tugs (tractor tugs) have azimuth drives and particularly good manoeuvrability. Tugs can berth against port facilities where the port operator is satisfied this can be done safely without damage to the berth fenders.

Many ports have dedicated tug berths or harbours for their tug fleets. These can be floating or fixed structures depending on tide range, cost and local preference. Some tug pens are designed for four point mooring, so the tug does not have, in theory, to come alongside a jetty and is held in the centre of the pen. Nevertheless, some fendering provision should be made. Where tugs are berthing against fendered jetties the principles applied to small ferry fendering are applicable.

3.1.12 Other Vessels

There are numerous other vessel types such as naval vessels, offshore vessels, construction vessels, sea planes and yachts that have specific characteristics and fender requirements.

When dealing with facilities dedicated to special vessels, designers should seek specific vessel data from the owner/ operators of the facility.

3.2 Displacement

The fully laden displacement is listed for various vessel types and sizes in MarCom WG 235 – ‘Ship Dimensions and Data for Design of Marine Infrastructure’ (2022).

The values are notionally intended to be a P90 value however that is a judgement assessment based on the data available which does not permit a precise statistical evaluation.

Designers can also calculate displacement using the formula below or the relationships in Table 3-1.

$$M = L_{BP} \cdot B \cdot D \cdot \rho_w \cdot C_b \quad (3-1)$$

Where:

M	Mass equivalent to the water displacement of berthing vessel [tonnes]
L_{BP}	Length between perpendiculars [m]
B	Beam of vessel [m]
D	Draught of vessel at midships [m]
ρ_w	Density of water (1.025 typical for seawater) [tonne/m ³]
C_b	Block coefficient of vessel

Ballast displacement, where relevant, is also included in the WG 235 tables. It is calculated using the formulae above and the estimated ballast draught at midships.

The relationship between displacement and capacity (DWT or GT) is surprisingly linear, however with some exceptions. A median value for displacement for various ship types can be estimated from the formulae below. The accuracy value represents the difference between the mean value and likely maximum value.

Vessel Type	Mean Displacement Formulae (linear) ^{2,3}	Approx. Accuracy ¹	Comment
General cargo	= 1.35 DWT + 200	12 %	
Container	= 1.29 DWT + 1700	5 %	
Cruise	= 0.50 GT + 1900	7 %	DWT is not related to displacement
Dry bulk	= 1.12 DWT + 3500	5 %	
Crude oil tanker	= 1.13 DWT + 3800	5 %	
Product tanker	= 1.17 DWT + 1300	5 %	
Chemical tanker	= 1.25 DWT + 450	6 %	
RoRo	= 1.50 DWT + 1950	17 %	
Vehicle carrier	= 1.64 DWT + 2900	20 %	
Refrigerated cargo	= 1.50 DWT + 500	17 %	
LPG	= 1.33 DWT + 5400	10 %	
LNG	= 1.33 DWT + 1400	10 %	
Ferry	No reliable formulae		
Fishing vessel	No reliable formulae		
Notes: ¹ Represents likely the maximum deviation, however, there are some outlying values for unusual vessels that may be outside this range. Accuracy typically improves with larger vessels. ² For upper bound values the mean displacement should be adjusted for the variability. ³ Values based on summer seawater density.			

Table 3-1: Approximate displacement v capacity relationships for various vessel types (WG 235)

3.3 Influence of Vessel Hull Characteristics on Fender Design

Vessel hull shape and strength are important considerations when designing fenders and checking clearances between the vessel and shore-based equipment and also the clearance between the vessel and structure itself (e.g. bow flare to quay edge, bulbous bow to underwater quay structure). They vary considerably with the type of vessel, size and individual designs. Ferries, fishing vessels, yachts and other small vessel types may be single or multiple hulls and have a wide range of design features and providing general guidelines is not practical and design data will need to be obtained on a case-by-case basis.

Larger monohull vessels will have similar overall characteristics for each type of vessel however can still have significant variation in hull shape from one design to another. In principle, the key parameters of interest will be:

- Flat (parallel) hull length at the level of the fenders.
- Location of the hull tangent line (TL) between the parallel side hull and the start of the bow curve (or stern curve if stern impact with fenders is a consideration) at the level of the fenders.
- Hull curvature in the horizontal plane at the bow (or stern if near stern impact with fenders is a consideration) at the level of the fenders for various tide levels and vessel draughts.

- Hull flare angle in the vertical plane at the level of the fenders.

Fender design will usually focus on bow impact because:

1. Stern impact is typically avoided because the hull flares rapidly increases near the water line and can cause the vessel to “ride over the fender(s)” or even over the quay if the vessel is in ballast condition and the water level is near the quay deck level.
2. The bow curve to the tangent line is usually longer than the stern curve to the tangent line.

Figure 3.2 defines key zones of a vessel hull with respect to fender contact.

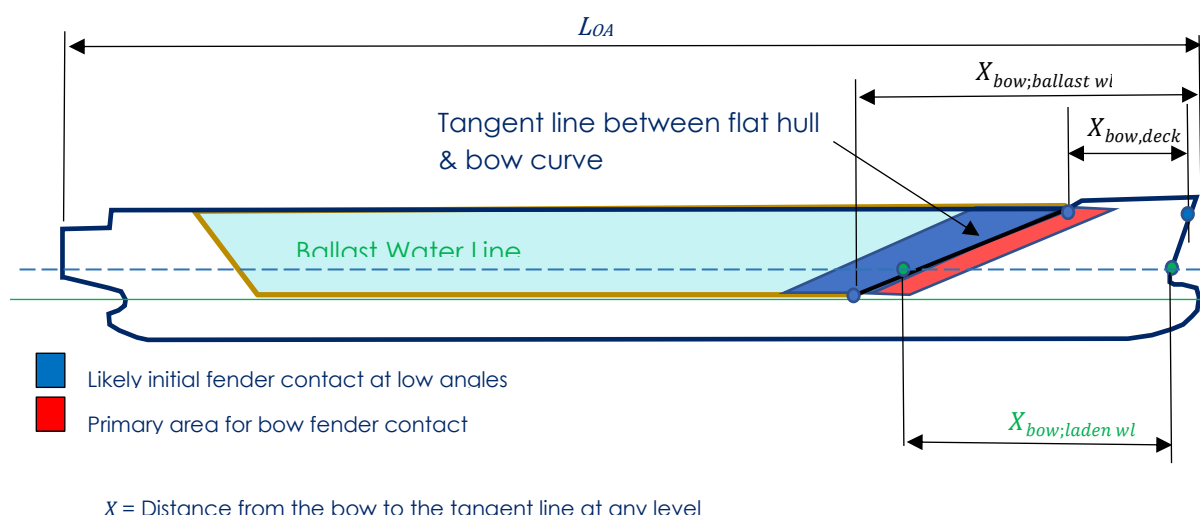


Figure 3.2: Fender contact zones on vessel hull at low berthing angles

For most vessels at very low berthing angles contact will occur with multiple fenders on the parallel part of the hull. With a small increase in the berthing angle further contact can also occur on the bow. As the angle increases the bow fender closest to the tangent line may become the first contact point and contact can occur with other fenders on the bow.

The extent of contact with 'bow fenders' is a function of the berthing angle, vessel length, vessel bow radius, fender pitch and the compression of the first fender contacted.

Vessels with higher sailing service speed will have a sleeker hull shape with less parallel hull length and when berthing at an angle are more likely to impact fenders on curved sections of the hull. Dry and liquid bulk carriers have slower sailing service speed and are designed to maximise the cargo capacity. This results in longer parallel hull length and shorter curved hull at the bow and stern.

Determining the interaction of the curved and flat hull section that will engage with multiple fenders during berthing is a complex geometric exercise that should also take into consideration factors that are independent of the hull shape such as berthing angle, vessel position relative to the fenders, fender pitch, compression of the first fender contacted and height of the fender relative to the vessel deck (varies with draught, tide and fender level). The effect of multiple fender contact on fender design is covered in Chapter 6.

It is recommended that a range of hull drawings should be obtained from the terminal operator for the planning and design of new fenders or replacement of existing fenders, taking also into consideration the possibility of different vessels in the future.

4 BASIS OF DESIGN

This chapter establishes the parameters that should be taken into consideration in the design of a fender system. The chapter can be used as a roadmap for the design process or as a template to develop a basis of design document to record the various parameters that will affect the design. Details of the various steps in the design process can be found in the subsequent sections of this document.

A fender system should be designed to serve the primary functions established in Chapter 2 and to meet both the functional and operational requirements for the marine terminal. The design of a fender system deserves as much attention as the design of any other element of the structure of which it is a part. For a new design, the selection of the type and capacity of the fender system and the berth structure should be interactive, especially with dolphins. For an existing structure the fender system should be selected and designed to fit the type and capacity of the structure. The fender should be designed considering the following requirements:

- Functional requirements including purpose of the berth and what the specific functions of the fender system will be.
- Operational requirements including design vessel(s), design life limitations for operation of the berth, both during berthing and mooring conditions.
- Assessment of the site conditions including bathymetry, wind and current conditions, wave climate, passing vessels and manoeuvrability.
- Assessment of the design criteria which can include national and international standards, local codes, and company requirements and standards to be used for the design.
- Assessment of acceptable reliability levels for the fender system, including considerations regarding potential for loss of life, dangers to the environment and economic impacts to both the terminal and the surrounding area.

These items can be used to produce a basis of design document. This document should include all relevant factors that will feed into the design. Subsequently the design of the fender system is developed, including these steps:

- Calculation of the energy to be absorbed by fender during berthing.
- Calculation of required fender reaction force for the moored condition.
- Select a suitable fender system and type based on these energy and reaction force criteria.
- Determine the maximum/ design fender reaction force and related friction force for further design (e.g. panel, chains, hull pressure)'.- Check reaction force from fender design and compare to supporting structure and vessel capacity.

The above process may have to be repeated several times to select the optimal fender for the specific situation. There are numerous fender brands and each of those brands offers various types of fenders and most often several standard dimensions for each fender type. It is the task of the design engineer to select the fender of which the specified characteristics meet (or come closest to) the design requirements. Where possible the designer should avoid specifications that can be met by only a single supplier. Keep in mind that same energy absorption might result in different reaction force from fenders supplied by different suppliers.

4.1 Functional Requirements

During the service life of the fender system, it should be able to perform its design functions within the limitations stated in the basis of design. Functional requirements describe the specific functions of the fender system, which may include the following:

- Will the fender system be used for energy absorption purposes or just protection to prevent contact between the vessel hull and steel or concrete? An example of fender systems used for protection can be UHMW-PE directly on concrete (Broos, et.al., 2018) . Note that this document is focused on energy absorbing fender systems only.
- Are there special requirements for non-marking of the hull of the vessels that will call at the terminal. This could often be the case for cruise vessels.
- Is the fender system used for regular contact by arriving vessels or is it a safeguard for accidents?
- Will the fender system be expected to absorb energy in compression, shear or both?
- Is the fender system considered sacrificial, so it is expected to fail during a design event? This might be a case for fenders protecting bridge piers.
- Is the fender system expected to absorb all of the energy from the berthing vessel by itself or will the energy absorption be shared between the fenders and the supporting structures?

4.2 Operational Requirements

The operational considerations can have a significant influence on the selection and design of fender systems. The port or facility operations manual or guidance will inform the design together with investigation and assessment by the designer in consultation with the operator. Operational aspects that may need to be considered include:

- The freeboard of the range of vessels.
- How the various vessels will approach the berth and contact the fender system.
- Any special aspects of the vessel design (e.g. flare, belting, ramp or door locations, etc.).
- Survival and operating wind/wave design limit requirements.
- Vessel motions and operating envelopes.
- Any constraints on the stand-off distance from the berth dictated by operations (e.g. plant reach, oil loading arms, vessel gangway length, etc.).
- Operational limits imposed by adverse weather or sea conditions. Some vessels may move off the berth during adverse conditions for safety or comfort. Others may remain at berth throughout adverse conditions meaning that the vessel's motion against the fender system will need to be considered.
- The normal operational conditions at the berth.
- The movement of vessels during loading and offloading or maintenance.
- The design life of the fender system, including the impact of wear and tear caused by operations.
- The risks and consequences of an extreme event in terms of damage to the fender system or the berth structure and impact on operations.
- Other installations on the berth (e.g. ladders, possibly combined with fenders, nearby mooring points and the mooring lines to them, etc.).
- Possible future equipment installations (e.g. shore power).

4.3 Site Conditions

Sufficient data regarding the configuration and location of the berth should be obtained to establish the factors that impact the fender system design. Such factors may impact fender system loading, durability, material selection, etc. The likelihood of several extreme site conditions occurring at the same time shall be considered.

The following site conditions may need to be considered:

- The geographical location of the berth.
- The wind conditions and frequency that various conditions will affect the berth.
- The wave conditions and frequency that various conditions will affect the berth.
- The currents at the berth location.
- The water depth at the berth and in the approach zone and the anticipated effects of climate change.
- The tidal range.
- Water density.
- The temperature range to which the fender unit may be exposed and influence of UV levels.
- Potential for ice build-up or impact during operations.
- The configuration of the berth structure where the fender system will be installed (e.g. could floating fenders roll under or over the quay edge, is there sufficient space for anchor bolts, chains, etc.).
- If the fender system is to be fixed to an existing structure, the load capacity, configuration and remaining service life of the structure.
- Passing vessel forces (both off and along the berth).
- Fenders for permanently moored vessels.

Apart from this data, site conditions can also be described by local experience, for instance by pilots, harbour masters, or local captains.

4.4 Design Criteria, Return Period and Design Life

Following the assessment of the functional and operational requirements and based on the site conditions, the design criteria that will be used in the calculation of the berthing energy and the selection of the fender system can be determined. The design criteria document will include:

- The codes, standards and guidelines to be used.
- The design vessel(s) and their characteristics.
- The approach velocity under normal conditions and extreme conditions.
- Frequency of berthing (fatigue).
- Berthing angle.
- The vessel's allowable hull pressure.
- The desired design life.
- The safety factors to be used.
- Whole life cycle considerations.
- Maintenance periods.
- Site conditions during berthing operations and whilst moored at the berth.

Other criteria to be determined during the design process will include:

- The structural layout of the berth.
- The maximum allowed reaction force (horizontal and vertical).

- The friction coefficient.
- The minimum or maximum fender pitch.

It is common practice around the world to use 50-year return period loads as characteristic, or nominal, loads for design of structures using Load and Resistance Factor Design (LRFD) or Partial Load Factor design. A 50-year return period is therefore used for the characteristic berthing velocities in this guideline, as well as for calculations including berthing energy and energy factor. This choice of load return period is not necessarily tied to the design life of the structure being designed.

BS6349-1-1 defines design life as "assumed period for which a structure or part of it is to be used for its intended purpose with anticipated maintenance but without major repair being necessary".

Service life is the practical result of the design life taking into consideration operation of the fender system within its design limits and maintenance of the fender system according to maintenance manuals.

Fenders, and hardware used for mounting and securing fenders to a berth structure, typically have an expected design life of 20 years. Compared to a structure with a 50-year design life, a fender will therefore appear to have a lower probability of failure, and lower probability of exceeding the design load during its design life. However, a fender itself is not only there to function as an independent structural element, however to function as an interface between berthing vessels and the structure supporting the fender. Since berthing energy ultimately is converted to a reaction force onto the supporting structure, a properly designed and selected fender is important to properly characterise that reaction force. An under-designed fender will influence the magnitude of this reaction force and will have a high probability of being overloaded, reaching a point where it will stop absorbing energy and instead just transfer a reaction force directly to the supporting structure. At this point the fender has stopped performing its primary function. A fender system is therefore considered to fail when it does not fulfil its function (Chapter 2.1), resulting in a high probability of exceeding the design value of the fender reaction force and/or the design value of the associated berthing impact force. Fender system failure can happen even if the fender still appears intact and undamaged for future use. Other causes of failure can be damage due to degradation or mechanical failure of the fender itself.

The structure supporting the fender will typically be designed for a 50-year load, and to get an appropriate characterisation of that load, the fender itself therefore needs to be designed to a load with the same 50-year return period, even if it is expected to be replaced within a 20-year period. This is reflected below and in the following chapters.

4.5 Operation and Maintenance

The design life of a fender system is likely to be less than that of the marine structures it is installed on and interacts with. To ensure the fender system remains functional during the entirety of its service life, a regular maintenance and inspection plan should be implemented. Visual inspections should typically be carried out on an annual basis. General guidance on types and intervals of inspections can be found in WG 103 (PIANC, 2008), ASCE101 (Childs, 2001) and the associated references. These give seven inspection types:

- New construction inspection
- Baseline inspection
- Routine inspection*

- Repair design inspection*
- Special inspection*
- Repair construction inspection*
- Post-event inspection

*These inspections define routine maintenance activities (refer to Chapter 0).

How a fender system can be repaired or replaced needs to be considered during the design stage. Consideration should be given to allow fenders (as much as possible) to be easily maintained/ repaired during all states of the tidal range. This will help to minimise downtime if a repair is required.

The assumptions and design choices made by the designer in the design of the fender system need to be passed onto the operator. This will help in the safe operation of the berth and aid in future fender repair, replacement, upgrades or other changes. This information should be properly documented in the basis of design.

4.6 Reliability Requirements

An important step in the design basis for a fender system is deciding on the appropriate reliability level for the design. Many factors can influence this reliability level, however the most important one is the consequence class for the fender system. This is not explicitly considered in WG 33 (which is superseded by this guideline) although it is mentioned that consequence of fender system failure should be considered in the selection of the abnormal berthing factor.

Understanding the consequences of failure of a fender system is of great importance. In general, when failure consequences are high the required reliability level increases. In some circumstances, failure of a single fender will not result in economic repercussions, whereas in other situations major accidents may occur. When national recommendations regarding reliability targets are lacking, presents typical examples of fender systems for different consequence classes.

The vast majority of the fender systems installed on marine structures correspond to class A or class B. The reader is referred to Appendix A for further background information regarding the consequence classes. The consequence class should be selected by the asset owner (e.g. port authorities or terminals) using appropriate input (refer to Chapter 13). In Section 5.8, the partial energy factor will be determined for all consequence classes, whereas in Section 6.7 the material factors (or partial resistance factors) are presented.

Class	Description of failure consequences	Explanation	Example of fender systems
A	Negligible/ low consequences for risk of loss of human life AND environmental damage AND economic damage.	Failure of a single fender predominantly results in insignificant to no structural damage.	Fenders installed on a marine structure that is part of a terminal or port with functional redundancy ^a and limited number of people at risk. Failure of a single fender is not likely to result in the unavailability of the berth or widespread damage to the marine facility assuming there is sufficient redundancy with additional berths. An example can be a continuous earth retaining quay wall or a dolphin berth with more than two/ redundant berthing (breasting) dolphins or marine facilities with multiple berths having similar capabilities.
B	Some consequences for risk of loss of human life OR environmental damage OR economic damage.	Material damage <u>and</u> functionality losses of significance for owners and operators <u>and</u> low or no social impact.	Fenders installed on a marine structure without functional redundancy ^a . Failure ^b of the fender system is likely to result in the unavailability of the berth with no other alternatives. An example can be a single berth with two berthing (breasting) dolphins.
C	Considerable consequences for risk of loss of human life OR environmental damage OR economic damage.	Material losses <u>and</u> functionality losses of societal significance, causing regional disruptions <u>and</u> delays in important societal services over several weeks.	Fenders installed on marine structures, positioned at locations for which failure of the fender system is likely to put public lives at risk. Fenders installed on a marine structure for which failure ^b of the fender system is likely to close the berth and cause considerable consequential economic loss. Examples can be essential floating powerplants or floating storage regassification units that are prevented from operating after fender failure ^b and sufficient backup measures are available to resume operations.
D	High risk of loss of human life OR environmental damage OR economic damage.	Disastrous events causing severe losses of societal services <u>and</u> disruptions <u>and</u> delays at national scale over periods in the order of months.	Fenders installed on marine structures for which failure ^b of the fender system is likely to lead to significant socio-economic disruptions. Examples are progressive damage or cascading effects of other types of structures, e.g. critical installations such as essential powerplants or floating storage regassification units that are prevented from operating after fender damage with no backup measures available to resume operation.
E	Very high risk for loss of human life OR environmental damage OR economic damage.	Catastrophic events causing losses of societal services <u>and</u> disruptions <u>and</u> delays beyond national scale over periods in the order of years.	Beyond the scope of this guideline. In some cases, owners may choose, for practical reasons, to add an additional berthing criterion to cover 'Extreme Events' where additional energy is absorbed by partial collapse of secondary structural elements to protect critical wharf assets.
a) If a structural component is part of a series system (a configuration such that, if any one of the system components fails, the entire system fails) or if progression of failure is not mitigated, a higher consequence class should be considered. b) A fender system is therefore considered to fail when it does not fulfil its function (Chapter 2.1), resulting in a high probability of exceeding the design value of the fender reaction force and/or the design value of the associated berthing impact force. Fender system failure can happen even if fender units still appear intact and undamaged for future use. Other causes of failure can be damage due to degradation or mechanical failure of the fender system itself.			

Table 4-1: Consequence classes and description of failure consequence

5 BERTHING ENERGY

This chapter provides guidance on calculating the berthing energy of a vessel that the fender system should absorb.

5.1 Berthing Manoeuvres and Navigation Conditions

5.1.1 Alongside Berthing (Parallel or Angular)

For parallel berthing manoeuvres, the vessel is slowed or brought to a stop off the berth and then manoeuvred transversely to the supporting structure (e.g. a jetty, a quay or berthing dolphins) with pilot and tug assistance. The berthing approach angle is typically very low. Illustrations like Figure 5.1 are exaggerated for clarity.

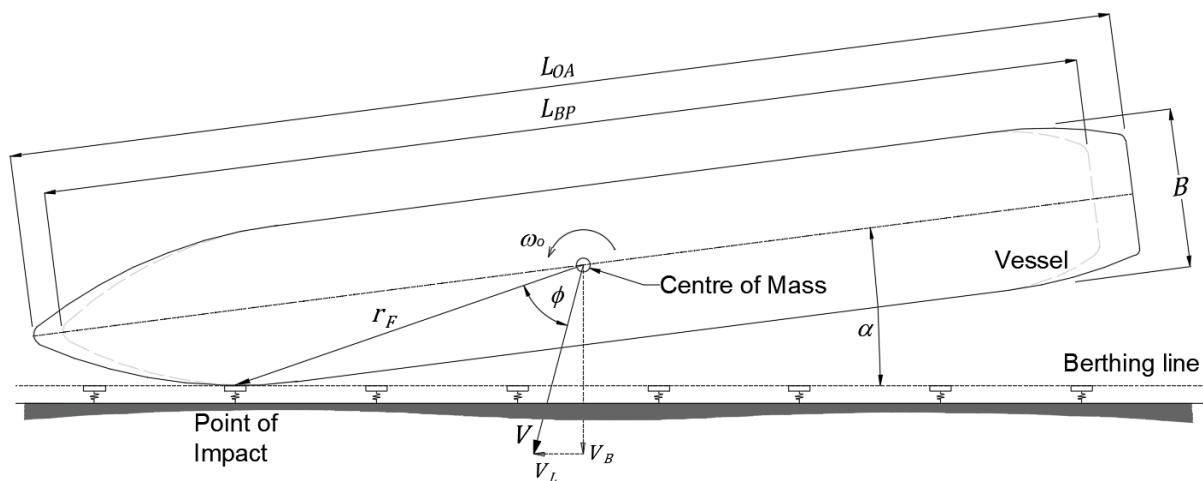


Figure 5.1: Berthing motion and corresponding reference angles

Vessels that may use one tug and have little or no transverse thruster capacity, such as coasters and feeder vessels, typically perform an angular berthing manoeuvre, i.e. landing a spring forward and pushing the stern in with the main engine and rudder, possibly resulting in larger approach angles. High berthing angles can risk impact between an overhanging vessel deck and any quay equipment close to the edge of the shore structure or with the quay edge itself. This must be considered in establishing port berthing procedures.

In some cases, pilots may use an angular approach for vessels with a small under keel clearance or vessels berthing in currents. This berthing manoeuvre is performed to maintain control over the vessel, during which masters and pilots, preferably assisted by tugs, use currents and wind conditions to their advantage during the berthing process.

5.1.2 End Berthing (Longitudinal)

In some situations, with ferries, coastal passenger boats/ catamarans and for some RoRo vessels, end berthing is the most common mode of berthing. The vessel approaches directly towards the end fenders and is generally held in position with forward propulsion after the impact continuously pushing towards the fenders, without mooring the vessel.

RoRo vessels with side ramps typically berth with alongside berthing procedures. RoRo vessels can berth bow or stern first perpendicular to the quay in an end berthing manoeuvre. This is covered in Section 5.4.3.

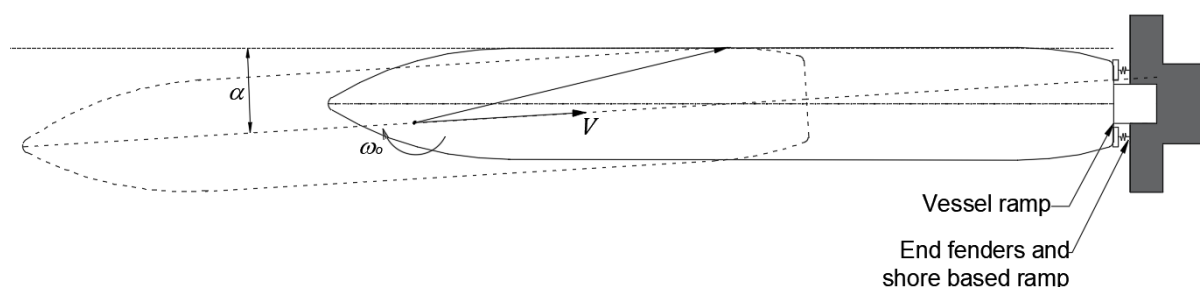


Figure 5.2: End berthing mode

Where the facility has a side quay or dolphin structure the vessels may initially berth alongside this structure while reversing to the unloading berth and stopping short of the unloading berth fenders. In this case, fenders are installed only to take incidental stern impact into account. Common design safety considerations for accidental situations are normally recommended for these situations.

5.1.3 Ship-to-Ship Berthing

A vessel berthing against a vessel (i.e. ship-to-ship berthing), is a special berthing procedure and further guidance is given in Section 5.9.

5.1.4 Navigation Conditions

Guidance on navigation conditions is given in Table 5-1. The navigation conditions are classified favourable when all the descriptions of the corresponding 'favourable' column are true, whereas the conditions will be classified as unfavourable when one of the conditions in the corresponding 'unfavourable' column is true. A navigation condition that is neither favourable nor unfavourable can be considered moderate. Allocating the navigation condition requires active discussion and must be performed and agreed upon with the asset owner prior to commencement of fender design.

If a berthing operation meets two or more of the unfavourable factors, a more detailed study is required to confirm control of the vessel throughout the maneuver and establish environmental thresholds for safe berthing operations. As the unfavourable conditions will also negatively affect other activities such as safe mooring and loading operations.

	Navigation Conditions		
	Favourable	Moderate	Unfavourable
Vessel approach strategy	The vessel can be brought to a controlled stop during the final berthing manoeuvre; AND	Vessels cannot be brought to a controlled stop during the final berthing manoeuvre, e.g. manoeuvring onto the berth by making use of the vessel momentum; OR	The capability to control the vessels approach, even with tug assistance, is significantly affected by the environmental conditions; OR
Resources for Vessel Control (main propeller, rudder, azipods, bow/ stern thrusters, tugs, etc.)	The vessel's movement can be fully controlled using the available resources, with margin. ; AND	The vessel's movement can be controlled using the available resources; however, environmental conditions are elevated and require active/continual use of the available resources to maintain control, margin is reduced. If neither tugs nor thrusters are present, this category may apply for benign environmental conditions.; OR	Environmental forces are significant compared to the available resources for controlling the vessel's movements, approaching limits of control. If neither tugs nor thrusters are present, this category may apply for moderate environmental conditions.; OR
Currents	During the berthing process currents at oblique angles or parallel to the berth having minimal effect on the manoeuvring vessel. Current forces are small and marginally effect the efficiency of the available tug power and/or vessel propulsion; AND	During the berthing process currents are generally parallel to the berth, however, may require continuous use of elevated vessel propulsion and/or tug power to stabilise control of the vessel in its final approach. Oblique current forces are controllable by available tug power and/or vessel propulsion; OR	During the berthing process, strong currents e.g. turbulent currents, at an oblique angle or parallel that require substantial use of propulsion to control the vessel; OR
Waves	During the berthing process wave effects on both the berthing vessel and the assisting tugs are negligible; AND	During the berthing process wave effects on the berthing vessel are small, however, the effectiveness of the tug assistance may be reduced; OR	During the berthing process waves substantially influence both the berthing vessel and the assisting tugs; OR
Wind	During the berthing process wind speeds and/or windage area result in small wind forces that marginally reduce the effectiveness of the available tug power and/or vessel propulsion.	During the berthing process wind speeds and/or windage area result in moderate wind forces that reduce the effectiveness of the available tug power and/or vessel propulsion.	During the berthing process wind speeds and/or windage area result in high wind forces that substantially reduce the effectiveness of the available tug power and/or vessel propulsion.

Table 5-1: Description of navigation conditions at berth

5.2 Kinetic Energy of a Berthing Vessel

5.2.1 Fundamentals of Kinetic Energy Method

The kinetic energy method is based on Newton's second law of motion. The six degrees of vessel motions (Figure 5.3), associated with velocity and rotation in horizontal and vertical

directions, are simplified to sway, surge and yaw. Heaving, rolling and pitching of the vessel are normally ignored in fender design due to their negligible influence on berthing.

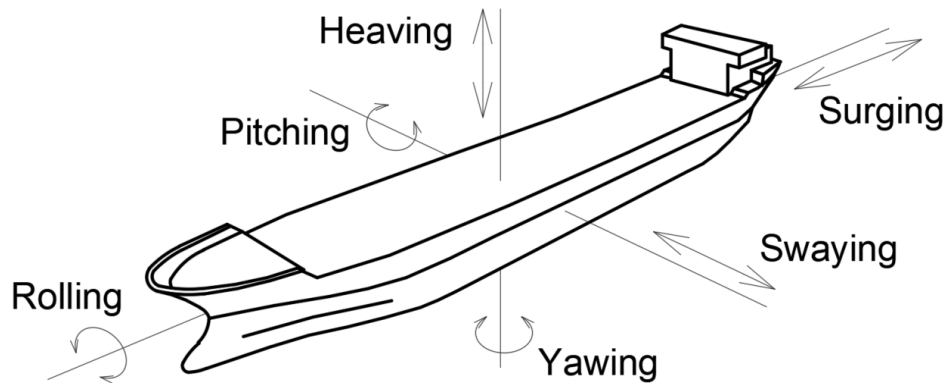


Figure 5.3: Degrees of motion of a vessel

Figure 5.1 illustrates a berthing vessel with approach velocity combined with initial rotation at the time of initial fender impact. The kinetic energy of a vessel in motion may, in berthing situations, be calculated as in Equation (5-1) below. The vessel's initial motion and velocity at the time of first contact with the fender should be the input for berthing energy calculations.

$$E_v = \frac{1}{2}MV^2 + \frac{1}{2}MK^2\omega_0^2 \quad (5-1)$$

Where:

- E_v Kinetic energy of berthing vessel at the time of impact [kJm]
- M Displacement of the berthing vessel, equal to the total mass of the vessel and all its cargo [tonnes]
- V Berthing velocity of vessel at the time of impact [m/s]; see Section 5.4
- K Radius of gyration of vessel [m]; see Section 5.6.2
- ω_0 Initial angular velocity of berthing vessel [radians/s]; see Section 5.3

When the berthing velocity component parallel to the berthing line (V_L) and ω_0 are small the vessel berthing energy can be simplified in calculation to consider only V_B , i.e. the berthing velocity component transverse to the berth (see Figure 5.1). Subsequent to the first impact, the fender will deflect and will continue to absorb energy until the vessel is brought to a stop. Concurrently, some of the translational energy of the vessel will be transformed into rotational energy. The reduction in translational energy due to conversion of part into rotational energy of the vessel is accounted for by an eccentricity factor.

Furthermore, the energy to be absorbed by the fender system is also adjusted to account for the inertia of the mass of water pushing on the vessel at the point of fender contact using the added mass factor. Therefore, the berthing energy that is to be absorbed by the fenders and the supporting structure can be calculated as below:

$$E_k = \left(\frac{1}{2} M V_B^2\right) C_e C_m \quad (5-2)$$

Where

E_k	Kinetic energy to be absorbed by fenders and structure during the impact [kNm]
V_B	Berthing velocity of vessel at the time of impact, perpendicular to berthing line [m/s]; see Section 5.4.1
C_e	Eccentricity factor; see Section 5.6
C_m	Added mass factor; see Section 5.7

Notes:

The berthing energy formula above differs from PIANC WG 33 (and others), for the following reasons:
Softness factor (C_s) has been removed by the Working Group based on (Berendsen E. A., 2022) that shows that the elastic contribution of the vessel hull is below 1 %. When the designer provides justification, e.g. by performing a detailed analysis showing that there is a significant effect of the energy absorption capacity by the vessel, this effect can be taken into consideration in fender design. This requires professional naval architecture review, cannot be carried out by the infrastructure responsible.

Berth configuration factor (C_c) has been removed as this effect is already included in the berthing velocity recordings used to determine the characteristic berthing velocities and partial energy factors.

5.2.2 Design Berthing Energy

The design berthing energy accounts for variations in displacement, effect of vessel berthing frequency and other uncertainties in energy calculation via a partial energy factor. This safety factor covers uncertainties given the characteristic berthing event:

$$E_{k,d} = \gamma_E E_{k,c} \quad (5-3)$$

Where:

$E_{k,d}$	Design energy to be absorbed by fenders in contact during the impact [kNm]
$E_{k,c}$	Characteristic energy to be absorbed by the fenders in contact (and the supporting structure where applicable) during the impact [kNm]
γ_E	Partial energy factor; see Section 5.8

A supporting structure equipped with a fender system must provide sufficient load-transferring resistance to absorb the total energy exerted by the berthing vessel whilst also accounting for uncertainties associated with the fender system. See further guidance in Chapter 0.

5.2.3 Characteristic Berthing Energy

The uncertainty in the berthing velocity significantly influences the uncertainty in the calculated kinetic energy (Ueda, 2010) and therefore berthing velocity is the most dominant design variable when calculating vessel berthing energy. The other two important, but less-dominant variables are the displacement of the vessel (M) and the berthing angle (α).

When the largest displacement, highest berthing velocity and extreme berthing angle are simultaneously considered in the fender selection process, this may lead to a significant overdesign of the fender system. When sufficient data is available, the characteristic values can be derived (Table 5-2) to determine the characteristic berthing energy. When no data or

site-specific information is available the respective sections of this chapter provide design guidance.

Design Variable	Characteristic Value
Berthing velocity (V_c)	0.02 % of probability being exceeded per berthing manoeuvre
Displacement (M_c)	Largest operational displacement of the design vessel resulting in the highest characteristic berthing energy
Berthing angle (α_c)	5 % probability of exceedance per berthing manoeuvre

Table 5-2: Recommended characteristic values of design variables

The characteristic berthing energy is therefore calculated as follows:

$$E_{k,c} = \left(\frac{1}{2} M_c V_c^2 \right) C_{e,c} C_{m,c} \quad (5-4)$$

Where:

- M_c Characteristic displacement of the design vessel [tonnes]
- V_c Characteristic berthing velocity of the vessel [m/s]; see Section 5.4
- $C_{e,c}$ Characteristic eccentricity factor; see Section 5.6
- $C_{m,c}$ Characteristic added mass factor; see Section 5.7

5.3 Displacement of Vessel

The berthing energy calculation takes account of the displacement of a vessel (M). see Chapter 0 for further guidance.

The characteristic displacement (M_c) is defined as the largest operational displacement of the design vessel resulting in the highest characteristic berthing energy. When berths are used by a wide range of berthing vessels the designer needs to evaluate the kinetic energy of each of the design vessels taking into account corresponding differences in berthing velocity and vessel mass (fully laden or ballast conditions).

It is recommended that displacement of berthing vessels is determined based of site specific information. When site-specific information is not available, the recommendations in PIANC WG 235 can be used.

The effect of variation in displacement can be accounted for by applying a partial energy factor when the variation in displacement is moderate or high, see Section 5.8 – Step 3.

5.4 Berthing Velocity

The berthing velocity (V , Figure 5.1) is defined as the vessel's approach velocity at the first moment of contact with the fender system. The berthing velocity should be provided by the asset owner (e.g. port authorities or terminals) using appropriate input (refer to Chapter 13). If site specific information is not available then the recommendations in the following sections of this chapter can be used.

The mean berthing velocity of all berthing manoeuvres at a particular location is generally quite low. However, velocities much greater than the mean value will occur during the design life of a fender and need to be taken into account in the design of a fender system. These higher velocities are considered by distinguishing a 'characteristic' berthing velocity (V_c) which is largely influenced by the local navigation conditions. Recommendation in Table 5-2 is to be considered in berthing energy calculations.

5.4.1 Berthing Velocity Based on Local Knowledge, Data and Experience

It is recommended that site-specific information and experience are used where available when defining the characteristic berthing velocity, e.g. berthing records, field observations, global company records, speed limits, operational limits, past service performance (what worked satisfactory in the past should not necessarily be upgraded), input of pilots and harbour master. When berthing records are available the characteristic velocity can be estimated and should be equal to a 0.02% probability of being exceeded. Characteristic velocities less than 0.10 m/s are not recommended unless it can be demonstrated using site-specific information and experience or when berthing aid systems are installed.

Navigation simulations to evaluate berthing velocities and berthing angles can also be considered. However, a high number of such simulations will be required to arrive at the characteristic design berthing event and typically the actual berthing is not part of those simulations.

Using local knowledge is the essence of this guideline. The velocities presented in Section 5.4.3 and 5.4.3 should only be used if local data, information, and experience is lacking. These velocities are considered to be safe and based on a global dataset therefore have a tendency to be conservative.

5.4.2 Transverse Velocity

For alongside berthing manoeuvres the rotational velocity (ω_0) and the longitudinal berthing velocity of the vessel (i.e. velocity parallel to the berthing line, V_L) at the time of impact (Figure 5.3) are assumed to be insignificant and hence not taken into account in the berthing energy calculation.

For an angular berthing manoeuvre, both the parallel and perpendicular velocity components (transverse velocity, V_B), combined with rotational velocity should be accounted for in the berthing energy calculation. However, typically for larger vessels and to a certain extent smaller vessels, it is common practice to simplify and consider only the transverse velocity when calculating the berthing energy.

When site-specific information is not available for a new design, Table 5-3 can be used to determine characteristic berthing velocities perpendicular to the berthing line at the moment of contact with a fender system.

When port operational limits are present, the characteristic velocities listed in Table 5-3 may be over-conservative. There may also be local restrictions, limitations and specific operational windows which can be used to determine a more appropriate characteristic berthing velocity. For example, for berths with an actively managed berthing velocity monitoring system or when a prescribed berthing velocity limit is embedded in the berthing policy. The velocities in Table 5-3 can therefore be reduced based on a comprehensive review of the port operations and berthing velocity limits.

Any adopted berthing velocity limit should meet the following criteria:

1. A documented formal assessment of the berthing velocity limit prepared by a port or terminal operator who should consult a harbour master, port pilots and a maritime civil engineer familiar with the fendering system and its design.
2. An 'Approach Velocity Measurement System' calibrated to measure velocity to an accuracy of $\pm 5\%$. There should be sufficient points measured to provide a continuous readout of the approach velocity at the centre of mass of the vessel. The velocity data should be displayed in a manner that can be readily read by the pilots while they are engaged in berthing the vessel.
3. Without evaluating the local navigation conditions the berthing velocity limit should not be smaller than 80 % of the characteristic velocity given in Table 5-3. However the adopted berthing velocity limit for new or recently constructed berths is recommended to be not less than 0.10 m/s.

Navigation Condition:	Favourable	Moderate	Unfavourable
Type of Vessel ^a	$V_{B,c}$ (m/s)		
Coaster	0.180 ^f	0.300 ^e	0.400 ^e
Feeder, Handysize	0.150 ^b	0.225 ^c	0.300 ^d
Handymax, Panamax	0.120 ^b	0.200 ^{e,g}	0.275 ^d
Vehicle Carriers	0.120 ^e	0.200 ^e	0.275 ^e
Post Panamax, Capesize (small), Aframax	0.100 ^{b,e}	0.175 ^c	0.275 ^d
New Panamax, Capesize (large), Suezmax, ULCV, VLBC, VLCC, ULCC	0.100 ^{b,e}	0.150 ^{c,f}	0.250 ^d
Cruise & Passenger Vessels	0.100 ^e	0.150 ^{e,f}	0.250 ^e
a. Typical vessel dimensions: Coaster (5,000-15,000 DWT), Feeder, Handysize (15,000-42,000 DWT); Panamax, Handymax (42,000-85,000 DWT); Post Panamax, Capesize, Aframax (85,000-115,000 DWT); New Panamax, Capesize, Suezmax (115,000-170,000 DWT); ULCV, VLBC, VLCC, ULCC (>170,000 DWT). For vessels not listed in the table (e.g. LNG) use equivalent size. Although most gas tanker owners have their own global data set. b. These recommended berthing velocities are largely based on field measurements in Rotterdam and Wilhelmshaven, (PIANC WG145, 2022), (Roubos, Gaal, Hein, Iversen, & Williams, 2022). c. These recommended berthing velocities are largely based on the normal navigation conditions distinguished by PIANC WG 145 (PIANC WG145, 2022) and verified by WG 211 against new data sets from Poland, Korea, US and India. d. These recommended berthing velocities are largely based on the measurements conducted in Bremerhaven, (PIANC WG145, 2022), (Roubos, Gaal, Hein, Iversen, & Williams, 2022). e. These recommended berthing velocities are based on interviews with masters, pilots, harbour masters and experienced port engineers. The values are based on comparison with similar vessel sizes. f. These recommended berthing velocities are based on EAU 2012 and ROM 2.0-11 (2012). g. Some unpublished berthing records, of berths claiming to be moderate, include slightly higher velocities. BS6349 and WG 211 consider this value to be sufficient for the majority of berths.			

Table 5-3: Characteristic berthing velocity in the absence of site-specific information

In circumstances that there are concerns about the validity of the values in Table 5-3, e.g. the berthing velocity is considered to be excessively high it is recommended to rely on local information and other relevant experience to determine an appropriate characteristic berthing velocity instead of using the values from table 5-3.

5.4.3 Longitudinal Velocity

For end berthing, the berthing energy should be calculated the same way as when calculating for parallel or angular berthing approaches. However, the factors applied in Equation (5-2) and in Equation (5-4) should be specific to end berthing where recommendations are given in sections of this chapter. Longitudinal berthing velocity (V_L) will be the governing velocity. Transverse velocity and rotation become insignificant.

The vessel manoeuvring procedure will influence the characteristic value of the longitudinal velocity for use in fender design. A limited number of published data on end berthing velocities are available (BS6349, 2014), (EAU, 2020), (ROM, 2012). It is recommended that characteristic longitudinal velocities are determined using statistical data specific to the berth or factual data obtained from a similar berth. In the absence of specific information, Table 5-4 provides guidance on characteristic velocity ($V_{L,c}$) to be used in fender design, based on design specifications from ferry berths connecting Germany and Sweden.

Navigation Condition:	Favourable	Moderate	Unfavourable
Type of vessel	$V_{B,c}$ (m/s)		
Passenger and vehicle ferries	0.400	0.500	0.600

Table 5-4: Characteristic longitudinal berthing velocity in the absence of site-specific information

For ferry berths increased approach velocities up to 1.0 m/s can occur and are relevant for side fender design as a sliding velocity. Vessels may slide along the side fenders before utilising end fenders. Sliding velocity is different from approach velocity (Chapter 8).

5.5 Berthing Angle

The berthing angle (α) is defined as the angle between the heading of the vessel and the berthing line, measured at the time of its initial point of contact with a fender, as illustrated in Figure 5.2 and defined in Table 5-2. It is not the approach angle, which can be a higher value.

Berthing angle depends on the type of berthing manoeuvre, tug assistance and thruster capacity. When bow flare angles are high and cranes are located close to the edge of a berth, the berthing angle should be small. When site-specific information is available the characteristic berthing angle (α_c) should represent an angle with a probability of 5 % being exceeded per berthing manoeuvre. When site-specific information is not available Table 5-5 can be used to determine the characteristic berthing angle. These values are based on the data of WG 145 and expert judgement by master mariner.

Alongside berthing	Tugs	Thrusters ^a	α_c ^b	α_i ^c	Explanation
Parallel, (Section 5.1.1)	Yes	Yes	2	3	Vessels positioned off the berth and approach parallel. Vessels have sufficient thruster capacity. If under keel clearance is very low and therefore it negatively influences manoeuvrability (local input needed), consider this as a 'no thrusters scenario'.
		No	3	5	Vessels positioned off the berth and approach parallel. Vessels do not have thrusters or very low under keel clearance negatively influences manoeuvrability.
	No	Yes	2	3	Vessels positioned off the berth and approach parallel. Vessels have sufficient thruster/pod capacity on bow and stern (like cruise vessels).
		No	X	X	This manoeuvre can only be done using a current or strong wind. To be discussed with pilots and vessel masters.
Angular (Section 5.1.1)	Yes	Yes	3	5	Vessels have a large angle during the approach. Local current or wind is used to berth the vessel. However, at the moment of fender contact the berthing angle is low. Vessels have sufficient thruster capacity.
		No	4	7	Vessels have a large angle during the approach. Local current or wind is used to berth the vessel. However, at the moment of fender contact the berthing angle is low. Vessels do not have thrusters or under keel clearance is very low and therefore negatively influences manoeuvrability.
	No	Yes	8	15	Smaller coastal vessels perform an angular approach, landing using spring lines and pushing the bow or stern in with engine and rudder. Vessels have some thruster capacity.
		No	10	20	Smaller coastal vessels preform an angular approach, landing using spring lines and pushing the bow or stern in with engine and rudder. Vessels have little or no thruster capacity.
a. Manoeuvrers with thrusters are more controlled and directly in the hand of the master. b. Characteristic berthing angle to be used in the design c. Upper limit of the berthing angle to ensure that the fender pitch is adequate (see Chapter 0)					

Table 5-5: Berthing angle [degrees] at the moment of contact, no site-specific information available

The characteristic berthing angle (α_c) is used both in berthing energy calculations and in fender selection.

The incidental berthing angle (α_i) is generally used to verify whether there is bow (flare) or side contact for the smallest design vessel to determine the minimum fender pitch in order to prevent damage to the supporting structure or berthing facility (Chapter 0). The combination of the incidental angle and a high velocity is not likely to occur simultaneously so that the partial energy factor can be 1.0 when assessing fender spacing.

5.6 Eccentricity Factor

When a vessel berths with the first point of contact with fenders eccentric to the centre of mass of the vessel (e.g. near its bow or stern during side berthing), the reaction to the impact will make the vessel yaw. This will dissipate its berthing energy due to part of its energy transforming into rotational kinetic energy. After initial contact the frictional resistance at the vessel to fender interface will influence the amount of berthing energy that will be absorbed by the fender system. Higher friction between the vessel and the fender results in greater transfer of energy to the fender and less transfer of energy into rotation of the vessel.. The eccentricity factor (C_e and $C_{e,c}$) accounts for the above in the berthing energy calculation.

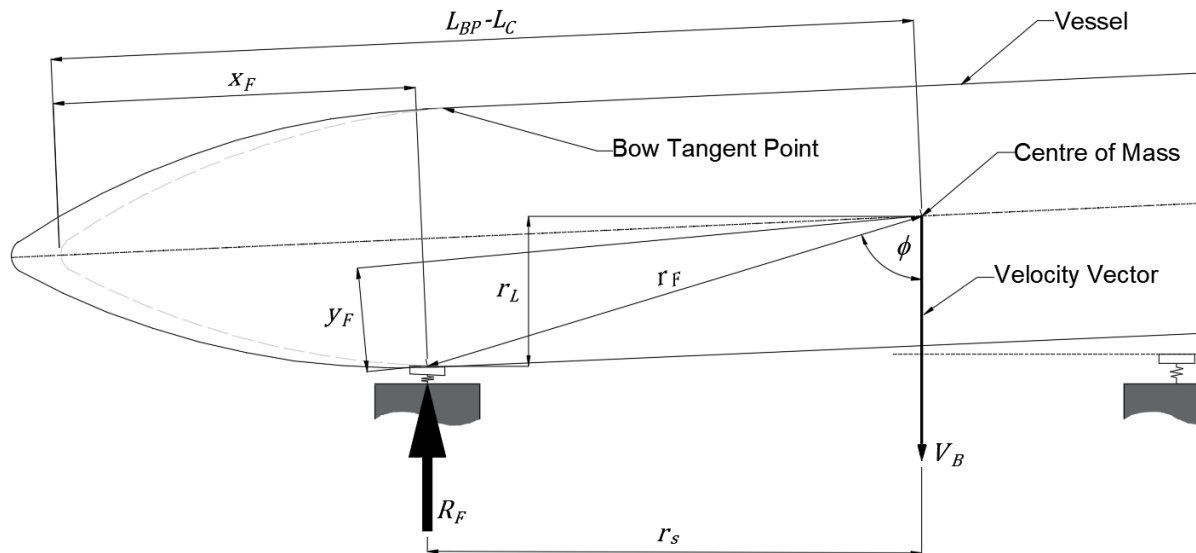


Figure 5.4: Nomenclature for calculating C_e factor (single fender contact)

In instances of insignificant friction or infinite friction (i.e. no sliding) analytical formulae can be used to calculate C_e (and $C_{e,c}$ with corresponding characteristic vessel data) and computer simulations can be used to estimate cases with moderate friction. In general, the differences in the formulae are relatively small, and it is sufficiently accurate for most fender designs to apply the following formulae:

$$C_e = \frac{K^2 + r_F^2 \cos^2(\phi)}{K^2 + r_F^2} \quad (5-5)$$

Where:

r_F Distance from the centre of mass of vessel to the resultant fender reaction force [m]; see Section 5.6.1

ϕ Angle between velocity vector and the line between the hull contact point of the resultant fender reaction force and the centre of mass of vessel [degrees]; see Figure 5.4.

$$\sin(\phi) = \frac{r_s}{r_F} \quad (5-6)$$

r_s Distance from the centre of mass of vessel to the hull contact point of resultant fender reaction force parallel to the berthing line [m]; see Figure 5.4.

The value of C_e normally varies between 0.3 and 1.0 depending on the berthing manoeuvre and berthing angle. When the berthing impact point is close to the centre of mass of the vessel C_e will be closer to 1.0. Such instances are e.g. when multiple fenders are installed at a continuous quay and that the berthing angle is quite small (i.e. less than one degree) or when a vessel is end berthing and in both cases yawing of the vessel becomes insignificant. However, calculating C_e can be complex when yawing is significant and further guidance is given in the subsequent section.

5.6.1 Distance r_F and Berthing Impact Point

For single fender contact the berthing impact point exactly aligns with the first point of contact between the hull of the vessel and the fender system (Figure 5.4). For multiple fender contact and low berthing angles, however, the first point of contact might result in a too optimistic value for C_e , since the resultant reaction force from all fenders in contact with the vessel can be much closer to the centre of mass of the vessel. Consequently, for multiple fender contact the berthing impact point does not necessarily align with the first point of contact with the hull. Then the berthing impact point can be determined by estimating the position of this resultant fender reaction force as shown Figure 5.5, which acts on the vessel hull at a distance r_F from the centre of mass of vessel.

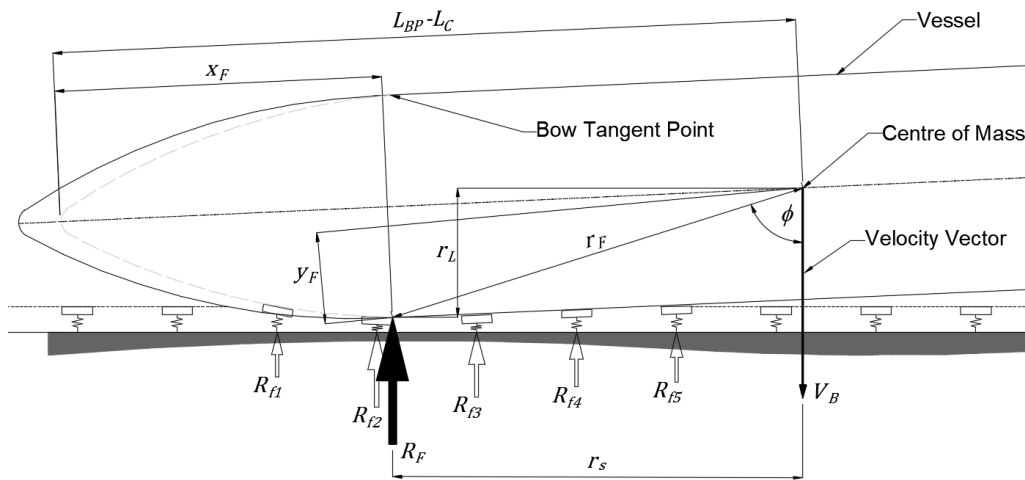


Figure 5.5: Nomenclature for calculating C_e factor (multiple fender contact)

In theory, the type of fender, fender pitch, fender height, berthing angle and geometry of the vessel such as the bow radius and parallel body length can largely influence r_F and hence C_e . To determine r_F , the probable worst outcome for the position of the vessel and the hull contact

point of the resultant fender reaction force should be determined based on the fender system and vessel geometry. It is therefore important to allow an 'out of position' of the vessel along the quay from the planned point of impact. This tolerance can typically range between 2 % to 5 % of the length overall of the vessel. For different vessel hull shapes and varying fender system configurations the value of r_F will differ.

When there is sufficient information about the geometry of the fender system and vessel the following formulae can be used to estimate or verify early design assumptions regarding r_F :

$$r_F = \sqrt{(r_s)^2 + (r_L)^2} \quad (5-7)$$

And,

$$r_s = \frac{\sum_{i=1}^{n_f} R_{fi} r_{si}}{R_F} = \frac{R_{f1} r_{s1} + R_{f2} r_{s2} + \dots + R_{fn} r_{sn}}{R_{f1} + R_{f2} + \dots + R_{fn}} \quad (5-8)$$

$$r_L = \frac{\sum_{i=1}^{n_f} R_{fi} r_{Li}}{R_F} = \frac{R_{f1} r_{L1} + R_{f2} r_{L2} + \dots + R_{fn} r_{Ln}}{R_{f1} + R_{f2} + \dots + R_{fn}} \quad (5-9)$$

Where:

R_F	Resultant reaction force of fenders in contact [kN]; see Figure 5.5
R_{fi}	Reaction force of i^{th} compressed fender [kN]
R_{fn}	Reaction force of n^{th} compressed fender [kN]
r_{si}	Distance from the centre of mass of vessel to compressed i^{th} fender parallel to the berthing line [m]
r_{sn}	Distance from the centre of mass of vessel to compressed n^{th} fender parallel to the berthing line [m]
r_L	Distance from the centre of mass of vessel to the hull contact point of resultant fender reaction force perpendicular to the berthing line [m]; see Figure 5.5
r_{Li}	Distance from the centre of mass of vessel to compressed i^{th} fender perpendicular to the berthing line [m]
r_{Ln}	Distance from the centre of mass of vessel to compressed n^{th} fender perpendicular to the berthing line [m]
n_f	Number of compressed fenders

In the early design stage, however, it is more practical to estimate r_F by assuming a reasonable value for the berthing impact point x_F , since the final dimensions of the fender unit might be unknown. Given a reasonable estimate for x_F , sufficient information about the geometry of the vessels hull and assuming that the centre of mass is close to midship ($\approx 0.5 L_{BP}$) the distance r_F can be calculated as follows:

$$r_F = \sqrt{\left(\frac{L_{BP}}{2} - x_F\right)^2 + (y_F)^2} \quad (5-10)$$

$$\phi = 90^\circ - \alpha - \arccos\left(\frac{\frac{L_{BP}}{2} - x_F}{r_F}\right) \quad (5-11)$$

Where:

L_{BP} Length between perpendiculars of vessel [m]; see Figure 5.1

x_F Distance from hull bow point (corresponding to L_{BP}) to the hull contact point of fender reaction force, R_F at the level of fender contact [m]; see Figure 5.5

y_F Distance from vessel's centre of mass to the hull contact point of fender reaction force, R_F [m]; see Figure 5.5

Note: In Equation 5-11 “ $\arccos$ ” means “ $\cos^{-1}$ ”.

When detailed information about the vessel geometry is not available a conservative estimate would be to assume that y_F equals half the beam of the vessel (i.e. $B/2$). Then distance r_F and r_S can be calculated as follows

$$r_F \approx \sqrt{\left(\frac{L_{BP}}{2} - x_F\right)^2 + \left(\frac{B}{2}\right)^2} \quad (5-12)$$

$$r_S \approx \left(\frac{L_{BP}}{2} - x_F\right) \cos(\alpha) - \frac{1}{2} B \sin(\alpha) \quad (5-13)$$

In the absence of accurate information or where a high level assessment of the berthing energy is sufficient, following values for C_e may be used:

Impact point	$\frac{x_F}{L_{BP}}$	$C_b = 0.55$					$C_b = 0.65$				
		$\alpha=2^\circ$	$\alpha=3^\circ$	$\alpha=4^\circ$	$\alpha=8^\circ$	$\alpha=10^\circ$	$\alpha=2^\circ$	$\alpha=3^\circ$	$\alpha=4^\circ$	$\alpha=8^\circ$	$\alpha=10^\circ$
Fifth point	20%	0.37	0.38	0.39	0.42	0.43	0.41	0.42	0.42	0.45	0.47
Quarter point	25%	0.46	0.47	0.48	0.51	0.52	0.50	0.51	0.51	0.54	0.55
Third point	33%	0.66	0.67	0.67	0.70	0.71	0.69	0.70	0.70	0.73	0.74
Mid ship	50%	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

Impact point	$\frac{x_F}{L_{BP}}$	$C_b = 0.75$					$C_b = 0.85$				
		$\alpha=2^\circ$	$\alpha=3^\circ$	$\alpha=4^\circ$	$\alpha=8^\circ$	$\alpha=10^\circ$	$\alpha=2^\circ$	$\alpha=3^\circ$	$\alpha=4^\circ$	$\alpha=8^\circ$	$\alpha=10^\circ$
5 th point	20%	0.39	0.40	0.41	0.43	0.45	0.43	0.43	0.44	0.46	0.48
4 th point	25%	0.44	0.45	0.46	0.48	0.50	0.48	0.48	0.49	0.51	0.53
3 th point	33%	0.54	0.54	0.55	0.57	0.59	0.57	0.57	0.58	0.60	0.61
Mid ship	50%	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

Table 5-6: Typical C_e factors for different impact point along the vessel

Example: Assume that:

$L_{OA}=229$ m, $L_{BP}=220$ m, $B=32.2$ m, $C_b=0.65$, $\frac{x_F}{L_{BP}}=25\%$, $y_F = \frac{B}{2}$, $\alpha = 2^\circ$ and $\phi = 71.7^\circ$.

Then Equation (5-10) and Equation. (5-14) give $r_F = 57.3$ m and $K = 51.37$ m, respectively. Substituting this in Equation (5-5) returns $C_e=0.50$.

5.6.2 Radius of Gyration

The radius of gyration (K) of a vessel can be calculated using the formulae below (ref: WG 33):

$$K = (0.19C_b + 0.11)L_{BP} \quad (5-14)$$

$$C_b = \frac{M}{L_{BP} \cdot B \cdot D \cdot \rho_w} \quad (5-15)$$

Where:

- C_b Block coefficient of vessel
- B Beam of vessel [m]; see Figure 5.6
- D Draught of vessel [m]; see Figure 5.8

Further information is available in PIANC WG 235. Where a range is given in WG 235 the mean value can be used in the absence of other information.

5.6.3 Vessel Centre of Mass

For most vessels, the draught is approximately constant along the length and the centre of mass can be taken as the midpoint ($\approx 0.5 L_{BP}$). For vessels in ballast (berthing at export facilities), where the draught at the stern is greater than the draught at the bow, the centre of mass will be closer to the stern. The location of the centre of mass from the stern can be estimated as follows:

$$L_c = L_{BP} \frac{\left[D_F + \frac{1}{3}(D_A - D_F) \right]}{[D_A + D_F]} \quad (5-16)$$

Where:

- L_c Distance between centre of mass and the stern [m]; see Figure 5.6
- D_F Forward draught [m]; see Figure 5.6
- D_A Aft draught [m]; see Figure 5.6

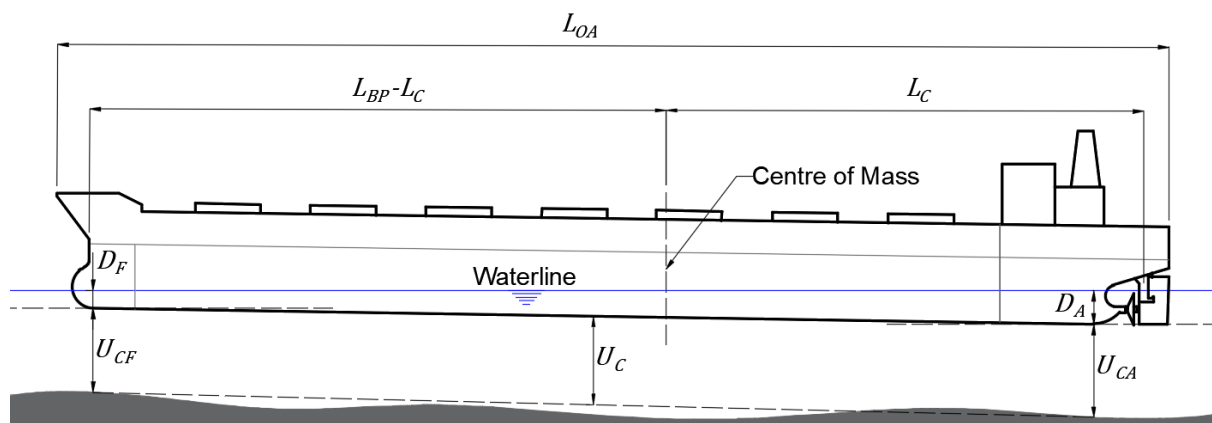


Figure 5.6: Vessel centre of mass and under keel clearance

Some unusual-shaped hulls or small vessels may have larger offsets between the centre of mass and the midpoint. In these circumstances, it is likely that vessel geometry can largely influence the fender design, and hence specific information should be sought from the vessel owners.

5.6.4 Eccentricity Factor for End Berthing

The Eccentricity Factor C_e (and $C_{e,c}$) should be taken as 1.0 for end berthing situations.

5.7 Added Mass Factor

A vessel in motion is subject to a range of hydrodynamic forces, such as drag forces due to viscous effects, which act to resist velocity and slow the vessel down, and inertial forces due to the mass of water surrounding the vessel that 'moves' with the vessel and resists changes in velocity. This inertial force is known as 'added mass' and is accounted for in the berthing energy calculation by multiplying the vessel mass by an Added Mass Factor (C_m). There are numerous references for calculating C_m that have been tested by modelling to various degrees. There is no clear definitive method since the number of variables in the estimation of C_m is considerable. The most important variable, however, is the under keel clearance.

5.7.1 Alongside Berthing

For parallel and angular berthing manoeuvres it is recommended that C_m be estimated from the graph below:

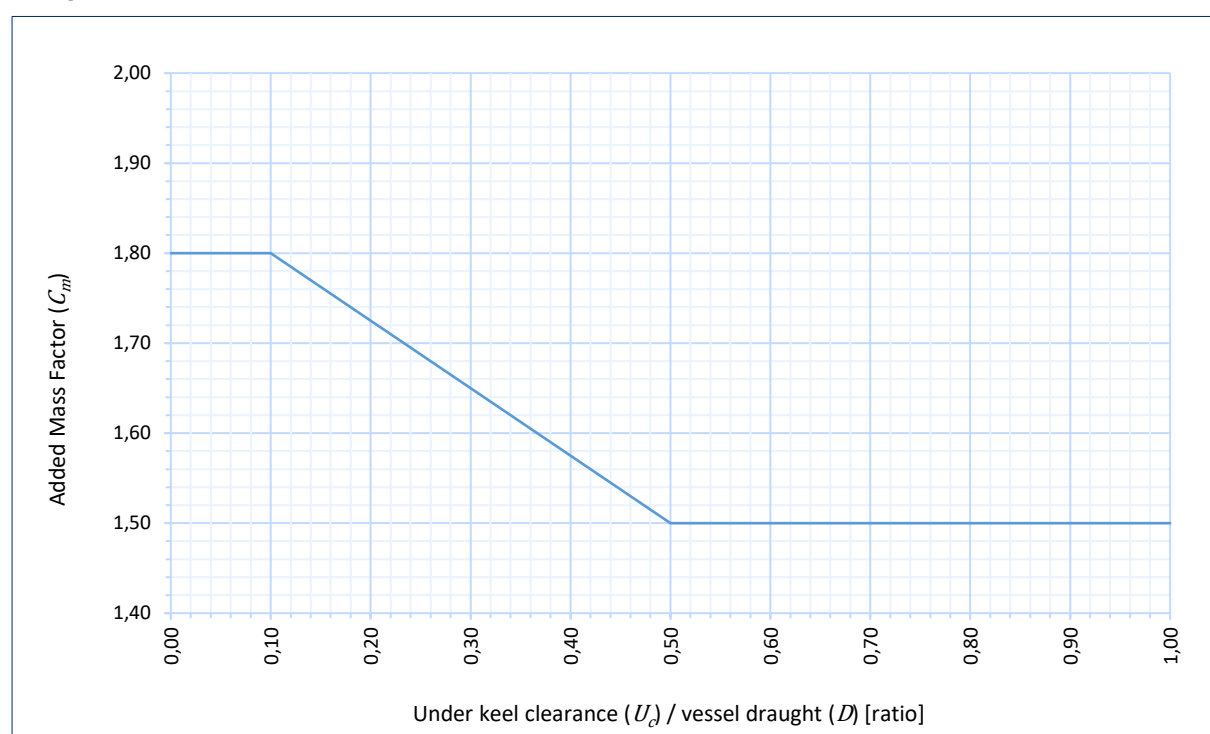


Figure 5.7: Added Mass Factor

The above graph is largely based on the work by PIANC WG 33. In addition at an under keel clearance (U_c) lower than 10 % of the draught of the vessel the C_m ($C_{m,c}$ when characteristic vessel data are used) value is assumed to be constant (i.e. 1.8) based on research carried out

by Delft University (Heemskerk, 2020). This research indicated that vessels with low under keel clearance in free basin conditions stop earlier than same vessels with larger under keel clearance. Therefore, large under keel clearance implies higher berthing speed. These findings have been confirmed during discussions held with experts of Gent University. There is neglected friction from the boundaries i.e. the bottom and if present solid quay wall or slope. The measurement of under keel clearance is shown in Figure 5.6 and Figure 5.8. For U_c/D ratio greater than 1.0, an added mass factor of 1.5 can be used.

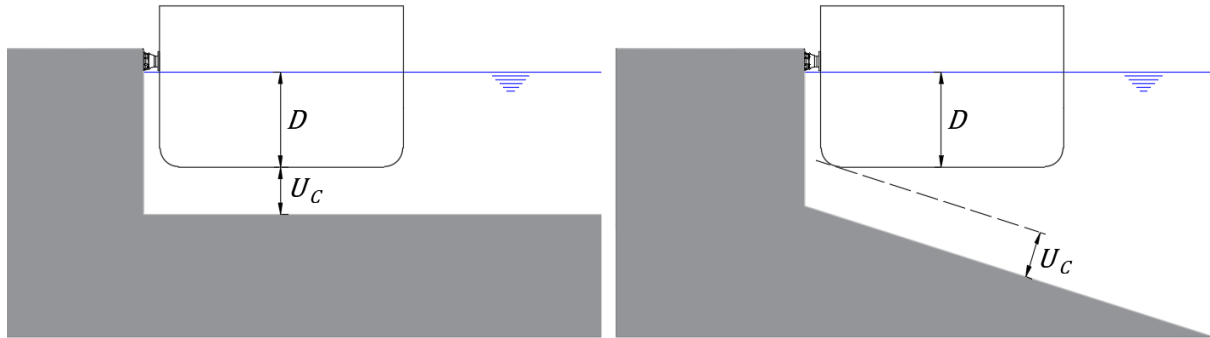


Figure 5.8: Under keel clearance of a vessel

5.7.2 End Berthing

Due to the shape of the hull, the influence of water moving with the vessel when it approaches the berth longitudinally is less significant compared to alongside berthing. For bow-first berthing the additional hydrodynamic mass can be ignored and therefore a C_m of 1.0 can be used in the design. For vessels (e.g. ferries) designed for stern-first berthing a C_m of 1.0 can also be used. For stern-first berthing of vessels not specifically designed for stern-first berthing the C_m can be taken as 1.1.

5.8 Partial Energy Factor

The partial energy factor (γ_E) accounts for the uncertainty in the berthing energy calculation and is applied to the characteristic berthing energy $E_{k,c}$ in order to determine the design value of the berthing energy $E_{k,d}$ (see Equation (5-3)). The partial energy factor is derived as follows:

$$\gamma_E = \gamma_{E,ref} \gamma_n \gamma_p \gamma_c \quad (5-17)$$

Where:

- $\gamma_{E,ref}$ Reference partial energy factor for 100 berthings per year
- γ_n Correction factor for alternative annual berthing frequencies; see Step 6
- γ_p Correction factor for berthings without pilot assistance; see Step 7
- γ_c Correction factor for correlations between design variables; see Step 8

The partial energy factor (γ_E) can be derived using the following steps:

1. Allocate an appropriate consequence class (see Chapter 0 for guidance).
2. Evaluate site specific navigation conditions (see Section 5.1.4 for guidance).
3. Account for variations in water displacement of the berthing vessels.

4. Determine whether a single fender or multiple fenders contribute to berthing energy absorption.
5. Select reference partial energy factor.
6. Adjust reference partial energy factor for alternative annual berthing frequencies.
7. Adjust reference partial energy factor for berthings without pilot assistance.
8. Activate positive effects of correlations between design variables where relevant.

Each step is further described below.

Step 1: Allocate an appropriate consequence class

For the purpose of reliability differentiation, table 4-1 presents consequence classes considering the consequences of failure or malfunction of a fender system that may occur at a specific site. The required level of performance and hence the target reliability level of a fender system vary for each consequence class. Therefore, a higher consequence class would require a higher partial energy factor.

Step 2: Evaluate the local navigation conditions

When calculating the berthing energy, berthing velocity is considered to be the dominant design variable. However, local navigation conditions can significantly influence the variation in berthing velocity and hence influence the probability of failure of a fender system. Therefore, for favourable, moderate and unfavourable navigation conditions (see Table 5-1) different partial energy factors are recommended.

Another method to control uncertainty in berthing energy is to monitor the berthing velocity. When masters and pilots are aware of realistic berthing speed limits AND when berthing aid systems are used, such as portable pilot units or fixed shore-based docking systems, this is defined as monitored berthing and lower partial energy factors can be taken into account. Since the influence of the navigation conditions is already taken into account in the determination of the characteristic berthing velocity (e.g. in the berthing speed limit), the partial energy factor for monitored conditions is not further influenced by the navigation conditions. Refer to Table 5-10 for the reference partial energy factor for monitored conditions.

Step 3: Effects of variations in displacement

In many projects, the characteristic mass equals the maximum displacement of the largest design vessel. For berths facilitating vessels with small variations in displacement this seems a reasonable assumption. However, for berths that accommodate a wider envelope of design vessels the above assumption might be over conservative. In order to prevent overdesigning the fender system, the reference partial energy factor can be determined using the coefficient of variation of the displacement CoV_M of the berthing vessels.

When the characteristic mass does not equal the maximum displacement of the largest design vessel, for instance when a smaller vessel results in the highest characteristic energy, the designer should use the partial energy factor corresponding to low value of CoV_M .

When site-specific information is available the coefficient of variation can be estimated using the following equation:

$$CoV_M = \frac{\sigma_M}{\mu_M} \quad (5-18)$$

Where:

σ_M Standard deviation of the displacement of approaching vessels [tonnes]

μ_M Mean displacement of approaching vessels [tonnes]

A low value of CoV_M means that variations in water displacement of the approaching vessels are fairly small and only marginally influence the probability of failure of the fender system. When no site-specific information is available the qualitative explanation in Table 5-7 can be used for guidance.

Coefficient of variation		Explanation
Low	$CoV_M < 15 \%$	Variations in displacement of the approaching vessels are small and marginally influence the berthing energy. Berths facilitate predominantly the same vessel type and class. Furthermore, variations in draught of the vessels is minimal.
Moderate	$15 \% \geq CoV_M < 50 \%$	Variations in displacement of the approaching vessels are moderate and will influence the berthing energy. Berths facilitate predominantly a small envelope of vessels. Furthermore, variations in draught of the vessels is moderate.
High	$CoV_M \geq 50 \%$	Variations in displacement of the approaching vessels are high and will significantly influence the berthing energy. Berths facilitate a wide envelope of design vessels, e.g. both big and small vessels. Furthermore, variations in draught of the vessels are significant.

Table 5-7: Coefficient of variation in displacement

Step 4: Determine whether a single fender or multiple fenders absorb the berthing energy

This step has a relation with the partial material factor γ_{mult} , which will be further discussed in Chapter 6. When the berthing angle and fender pitch are fairly small, multiple (two or more) fenders may contribute in absorbing the berthing energy. For multiple fender contact the amount of energy that can be absorbed by the fender system is largely influenced by the berthing angle and geometry of the vessel (bow radius and parallel body length), whereas for single fender contact the influence of the berthing angle is much lower. Since the probability that the characteristic berthing velocity and characteristic berthing angle occur simultaneously is quite low, a lower partial energy factor can be taken into account for multiple fender contact (Table 5-9) compared to single fender contact (Table 5-8) in order to prevent overdesigning the fender system.

Step 5: Select the reference partial energy factor

The reference partial energy factor can be selected using Table 5-8 and Table 5-9 that correspond to the relevant consequence class (table 4-1) and the method of fender contact. Reference partial energy factors were derived for a frequency of 100 berthings per year in accordance with (Brolsma, Hirs, & Langeveld, 1977). If berthing will always be supported with a monitoring system, the values in Table 5-10 can be used (see also step 2).

Navigation Condition	CoV _M	Reference partial energy factor for consequence classes [$\gamma_{E_{ref}}$]			
		A	B	C	D
Favourable	High	1.30	1.50	1.60	1.70
	Moderate	1.35	1.55	1.65	1.80
	Low	1.50	1.70	1.80	1.95
Moderate	High	1.35	1.60	1.70	1.85
	Moderate	1.45	1.65	1.75	1.90
	Low	1.60	1.80	1.90	2.10
Unfavourable	High	1.50	1.85	2.00	2.20
	Moderate	1.60	1.95	2.05	2.30
	Low	1.80	2.15	2.30	2.55

Table 5-8: Reference partial energy factor [$\gamma_{E_{ref}}$] for 100 berthings per year – single fender contact

Navigation Condition	CoV _M	Reference partial energy factor for consequence classes [$\gamma_{E_{ref}}$]			
		A	B	C	D
Favourable	High	1.15	1.35	1.40	1.50
	Moderate	1.20	1.40	1.55	1.55
	Low	1.35	1.50	1.60	1.70
Moderate	High	1.20	1.40	1.45	1.60
	Moderate	1.25	1.45	1.55	1.65
	Low	1.40	1.60	1.70	1.80
Unfavourable	High	1.25	1.55	1.65	1.85
	Moderate	1.35	1.60	1.75	1.95
	Low	1.50	1.80	1.95	2.15

Table 5-9: Reference partial energy factor [$\gamma_{E_{ref}}$] for 100 berthings per year – multiple fender contact

Monitored berthing	CoV _M	Reference partial energy factor for consequence classes [$\gamma_{E_{ref}}$]			
		A	B	C	D
Single fender contact	High	1.25	1.40	1.45	1.55
	Moderate	1.30	1.45	1.50	1.60
	Low	1.40	1.55	1.65	1.75
Multiple fender contact	High	1.10	1.25	1.30	1.40
	Moderate	1.15	1.30	1.35	1.45
	Low	1.30	1.45	1.50	1.60

Table 5-10: Reference partial energy Factor [$\gamma_{E_{ref}}$] for 100 berthings per year – for Monitored Berthings

Step 6: Adjust the partial energy factor for alternative berthing frequency

The actual annual berthing frequency can be much higher or lower than the 100 per year assumed in $\gamma_{E_{ref}}$, which can significantly influence the reliability of the fender system. For instance, fender systems that are installed at ferry berths may be subject to a berthing impact every 15 minutes, whereas other berths may have only one berthing per month. The correction factor γ_n adjusts the partial energy factor and can be estimated using the following equation (Roubos, Iversen, & Oskamp, 2024):

$$\gamma_n = a \ln(n) + b \quad (5-19)$$

Where:

a Logarithmic regression coefficient; see Table 5-11 and Table 5-12

b Constant; see Table 5-11 and Table 5-12

n Annual berthing frequency

Class	Annual berthing frequencies, n							Equation (5-19)	
	1	2	5	10	20	50	100	a	b
A	0.68	0.73	0.79	0.84	0.89	0.95	1.00	0.0696	0.6788
B	0.72	0.76	0.82	0.86	0.90	0.96	1.00	0.0605	0.7207
C	0.73	0.77	0.83	0.87	0.91	0.96	1.00	0.0573	0.7356
D	0.76	0.79	0.84	0.88	0.91	0.96	1.00	0.0529	0.7561

Table 5-11: Correction factor for an alternative annual berthing frequency (γ_n) for $n \leq 100$

Class	Annual berthing frequencies, n							Equation (5-19)	
	200	500	1000	2000	5000	10000	25000	a	b
A	1.05	1.13	1.18	1.23	1.30	1.36	1.43	0.0778	0.6416
B	1.05	1.11	1.15	1.20	1.26	1.31	1.37	0.0669	0.6917
C	1.04	1.10	1.14	1.19	1.25	1.29	1.35	0.0630	0.7095
D	1.04	1.09	1.13	1.17	1.22	1.26	1.32	0.0576	0.7340

Table 5-12: Correction factor for an alternative annual berthing frequency (γ_n) for $n > 100$

Step 7: Adjust the partial energy factor for berthings without pilot assistance

Reference partial energy factors, presented in Table 5-8 and Table 5-9, were derived based on field observations where berthing manoeuvres were assisted by pilots and tugs. Pilots are familiar with the local navigation and environmental conditions and therefore no adjustments to the values in Table 5-8 and Table 5-9 are required, i.e. γ_p equals 1.0. For berthing manoeuvres that are assisted by tugs alone, tug skippers are generally guided by the captain or the master who may not be completely familiar with the local navigation conditions. This can lead to higher vessel berthing velocities/ energies and therefore it is recommended that a higher partial energy factor is used. In the absence of field measurements or site-specific information a correction factor γ_p of 1.25 is recommended to be used following the guidelines of the Spanish ROM 2.0-11 (2012).

Step 8: Activate positive effects of correlations between vessel size and berthing velocity, if any

Majority of the vessel berthing records (approach velocity and berthing angle) collected by the PIANC WG 145 do not show a strong relation between vessel size and berthing velocity. Therefore, $\gamma_{E_{ref}}$ was derived assuming that both these design variables are independent. When there is no correlation between vessel size, berthing velocity and berthing angle, γ_c equals 1.0. However, when vessel size, berthing velocity and berthing angle are to some extent dependent, this might lead to over-conservatism in the calculated berthing energy. For instance when a large vessel, having a larger water displacement, has a much lower berthing velocity compared to a small vessel, then a high value of the kinetic energy is less likely to occur during the design life of the fender system. Hence the probability of fender system failure is also low. Where site-specific information is available, for example if a port or terminal has a comprehensive data set, showing that large vessels berth with much lower berthing velocity compared to small vessels, it is recommended to quantify the effect of correlations between design variables. For an example, (Roubos, Iversen, & Oskamp, 2024) found that γ_c is close to 0.6 for a container terminal with favourable navigation conditions in Rotterdam, which indicates that correlations between berthing velocity and vessel size can significantly influence the reliability of a fender system. On the basis of the berthing records that are available, it is not yet possible to provide recommendations for γ_c that are generally applicable. Therefore, in the absence of site-specific information or data it is recommended that γ_c to be taken as 1.0.

5.9 Ship-to-Ship Berthing

This section provides guidelines for designing fenders for in-port or near shore Ship-to-Ship applications not covered by OCIMF STS transfer guidelines (OCIMF, 2013). This section excludes all offshore STS activities.

5.9.1 Berthing Manoeuvres

Ship-to-ship (STS) berthing operations can be divided into three categories:

- A. Double banking transfer – STS operation that is conducted while one ship (usually the larger of the two) is alongside a berth, dolphins or moored to buoys within port limits.
- B. Transfer at anchor – STS operation that is carried out between ships when they are moored alongside each other and where one of the ships is at anchor (or in a flexible mooring system such as MBM).
- C. Underway transfer – STS operation that is conducted between two ships that are underway.

For ship-to-ship transfer operations in category A, the berthing energy calculation should proceed using the equations outlined above in the previous sections of this chapter.

For ship-to-ship transfer operations where both vessels are free-floating (categories B and C), the berthing energy calculation must account for the mass of both vessels. For categories B and C, the berthing energy calculation must be modified to account for two free-floating bodies; this involves similar principles to a vessel berthing against a fixed structure. However, in the case of ship-to-ship operations, both vessels are either in motion prior to berthing or can be set in motion due to the berthing operation. Categories B and C are illustrated in Figure 5.9. For categories B and C the mass (real + added mass) of both vessels is relevant, along with the relative velocity between the vessels prior to fender contact.

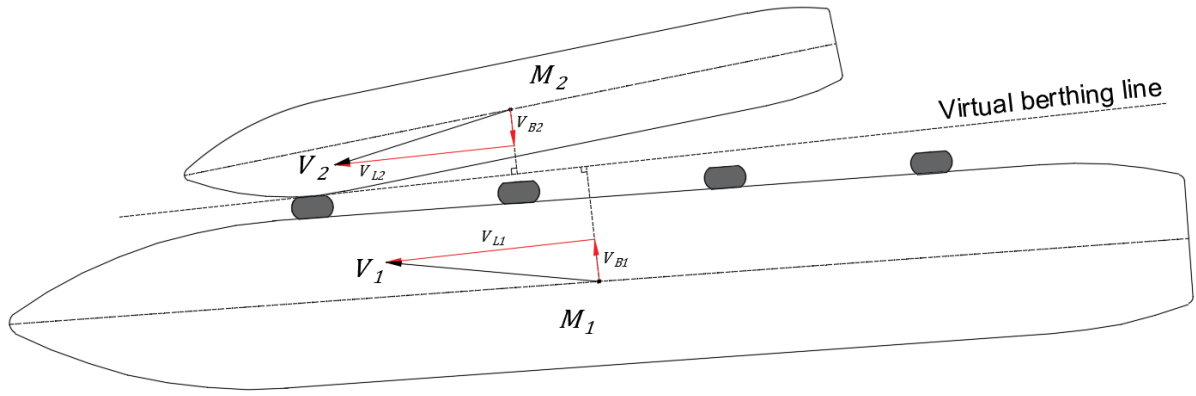


Figure 5.9: Ship-to-ship berthing for free floating vessels

5.9.2 Ship-to-Ship Berthing Energy

The energy to be absorbed by the fenders in a ship-to-ship berthing can be calculated assuming an inelastic collision between the two vessels, which is described by the following equation:

$$E_{k,c} = E_{sts,c} = \left(\frac{1}{2} \cdot M_{sts,c} V_{sts,c}^2 \right) \cdot C_{sts,e} \quad (5-20)$$

Where:

$E_{sts,c}$	Characteristic ship-to-ship berthing energy that needs to be absorbed by the fender system at the time of impact [kNm]
$M_{sts,c}$	Characteristic mass for energy of ship-to-ship berthing, including the effects of the displacement and added mass of both vessels [tonnes]
	$M_{sts,c} = \frac{M_1 C_{m1} M_2 C_{m2}}{M_1 C_{m1} + M_2 C_{m2}} \quad (5-21)$
M_1	Mass equivalent to displacement of Vessel 1 [tonnes]
M_2	Mass equivalent to displacement of Vessel 2 [tonnes]
C_{m1}	Added mass factor of Vessel 1; see Section 5.7.1
C_{m2}	Added mass factor of Vessel 2; see Section 5.7.1
$V_{sts,c}$	Characteristic closing velocity between the vessels [m/s]; see Figure 5.9 for definition of the velocity vectors
	$V_{sts,c} = V_{B1} + V_{B2} \quad (5-22)$
V_{B1}	Berthing velocity of Vessel 1, component perpendicular to berthing line [m/s]; see Section 5.9.3
V_{B2}	Berthing velocity of Vessel 2, component perpendicular to berthing line [m/s]; see Section 5.9.3
$C_{sts,e}$	Eccentricity factor; see Section 5.6

The generally accepted design practice is that each fender in the system should have sufficient energy absorbing capacity to absorb the largest anticipated berthing energy. Each

fender should be capable of absorbing the full impact load since vessels almost always contact a single fender on initial impact. A safety factor (γ_E) should be included to account for conditions (particularly velocities) that exceed the characteristic values. Although no study of partial energy factors was available for STS berthing when preparing this document, a reasonable energy factor can be identified using the methods outlined in Section 5.8.

5.9.3 Characteristic Berthing Velocity

The characteristic berthing velocity ($V_{sts,c}$), also called the 'relative approach velocity' or the 'closing' velocity, is defined relative to the virtual berthing line illustrated in Figure 5.9. This is the final alignment of the two vessels together and (although not known prior to the calculation) represents the reference frame for calculating the closing velocity. In other words, the closing velocity is equal to the sum of the velocities V_{B1} and V_{B2} in Figure 5.9. In practice, the closing velocity can be thought of as the speed at which the two vessel centres of gravity are approaching each other. It is influenced by the wind and sea conditions, skill of the pilot(s), the size and loading condition of the vessels and the type of propulsion. An additional consideration for ship-to-ship applications is whether both vessels are underway or if one is stationary. Because the berthing energy is proportional to the square of the velocity, ' $V_{sts,c}$ ' is the single most important factor in calculating the berthing energy.

The characteristic berthing velocities for ship-to-ship transfers are generally higher than those assumed for vessels berthing at fixed marine structures. Typical characteristic closing velocities for ship-to-ship transfers are summarised in Table 5-13. The full documentation behind how these velocities were developed was not available for review in preparation of this document. However, these velocities are used extensively in the industry and are referenced in peer reviewed publications (e.g. Sakakibara & Yamada (2008)). A few clarifications are helpful for interpreting these velocities:

- In many locations, ship-to-ship berthing may not be considered safe under the conditions listed for the Moderate and Rough (sea state) categories as mentioned in Table 5-13. Many other factors (which are beyond the scope of this guidance) govern the safety of ship-to-ship berthing operations. The velocities reported in this table indicate a reasonable starting point for estimating the closing velocity; however, local data (when available) and the judgement of the designer are paramount for ensuring that the design closing velocity will reasonably reflect the conditions experienced in the field.
 - These velocities are referenced to deadweight tonnage, which may be different for the two vessels mooring together. The deadweight tonnage of the smaller vessel can be used in selecting the berthing velocity.
 - These velocities were developed for tankers, which comprise the majority of ship-to-ship transfer operations, globally; however, these velocities can be used for other types of vessels with similar mass and propulsion characteristics (e.g. bulkers).
- Sea state categories are as defined by the World Meteorological Organisation.

Deadweight Tonnage [tonnes]	Characteristic Berthing Velocity [m/s]		
	Calm Sea State: 0 to 3 Wave Height: < 1.25 m	Moderate Sea State: 4 Wave Height: 1.25 - 2.5 m	Rough Sea State: 5 Wave Height: 2.5 - 4.0 m
< 10,000	0.300	0.400	0.500
10,000 – 50,000	0.250	0.325	0.400
50,000 – 100,000	0.200	0.250	0.300
> 100,000	0.150	0.200	0.250

Table 5-13: Closing velocity for ship-to-ship operations

5.9.4 Eccentricity Factor

The effect of eccentricity is often neglected for ship-to-ship berthing calculations (i.e. $C_{sts,e} = 1.0$), which is conservative. However, the principle of trade-off between translation and rotational energy that leads to the eccentricity coefficient for berthing against fixed fenders can be applicable for ship-to-ship berthing and can be included when appropriate. For ship-to-ship berthings in Category C (both vessels underway), there can be significant hydrodynamic effects that develop between the two vessels that may modify the rotational characteristics of the vessels, making the eccentricity coefficient very difficult to accurately specify. Thus, it is recommended that an eccentricity factor less than 1.0 should only be considered for Category A and Category B ship-to-ship berthings. If an eccentricity coefficient is calculated, it should be calculated using the same approach as for fixed terminal berthing however with the point of contact specified based on the position of the fenders relative to the centre of gravity of each vessel.

6 FENDER SYSTEM SELECTION

6.1 Fender Selection Process and Role of the Designer

This chapter outlines the fender selection process, providing background information on the issues that should be considered when selecting a fender system. The selection of a fender system deserves as much attention as the design of any other element of the structure of which it is a part. The selection of the fender, the type of fender system and the supporting structure are therefore interlinked.

The selection of a fender system should take a common-sense approach and follow the general process as outlined below:

- Determination of the functional requirements (Section 4.1)
- Determination of the operational criteria (Section 4.2)
- Assessment of the site conditions (Section 4.3)
- Establishment of the design criteria (Section 4.4)
- Calculation of the berthing energy to be absorbed by the fender during berthing and/or when moored (refer to Chapter 5)
- Selection of a suitable fender and fender system based on the berthing energy and design criteria (refer to Chapter 0)
- Determination of the fender reaction force and related friction forces (refer to Chapters 0 and 0)
- Confirmation that the supporting structure and vessel hull can accommodate the calculated forces (refer to Chapter 0)

The above process may have to be repeated several times to select the most optimal fender and/or fender system for a specific situation.

There are numerous fender system variations available, and it is the task of the designer to pragmatically select the most suitable and economical fender system which satisfies the design requirements or specified characteristics.

Before designing a fender system, the designer is advised to conduct a comprehensive evaluation of all project specific criteria that the fender system must satisfy. A sound understanding of the principles behind the use of the correction factors is also recommended, to prevent a potentially significant over or under-design of the resulting fender system.

The fender should be selected based on the calculated design berthing energy of the design vessel(s), vessel type and berth use, as outlined in Chapter 5 or based on the permanent mooring / 'lean-on' load(s) as outlined in Chapter 7, whichever is greater.

6.2 Fender Selection Overview

6.2.1 Fender Selection Using Project Specific Design Criteria

Every type and size of fender has different performance properties. Whatever type of fender is selected, the fendering system should have sufficient capacity to absorb the energy of the berthing vessel. When selecting fenders, the designer should consider many factors, including (but not limited to):

- Differences in vessel size and shape
- Approach velocity at moment of fender contact
- Single or multiple fender contact

- Berthing angle at moment of fender contact
- Angular compression of the fender
- Temperature range
- Berthing frequency
- Fender efficiency
- Asset manager / owner preference
- Tidal range

The selection of a suitable fender relies, in part, on carrying out calculations to confirm that the energy absorption capacity of the fender is greater than the berthing energy of the vessel(s). In addition, the associated compressed fender contact pressure should be less than the allowable vessel hull pressure and the capacity of the supporting structure.

Fender selection relies on the experience and judgement of the designer to make reasoned and pragmatic decisions as to which fender and fender systems may function best in the circumstances unique to each project. In conjunction with this experience, the designer is also recommended to seek advice from fender suppliers in identifying suitable fender solutions.

Having determined the characteristic berthing velocity of the governing design vessel(s) and calculated the design berthing energy, as outlined in Chapter 5, the designer should first identify the types of fenders that are most likely to satisfy the primary design criteria for the project. It is possible that several different types of fenders may be considered suitable. The initial fender type selection process is outlined in Figure 6.1. Guidance on the selection and suitability of fenders for certain applications, is included within Section 6.3.

Using the calculated design berthing energy, the designer should then refer to the fender performance data for each type, size and possible rubber grade of the fender, as published in the fender supplier's catalogues, to identify the **Base Performance** (previously known as CV performance in WG 33) properties for each type, size and rubber grade of the fender, (refer to point 2 in Figure 6.1).

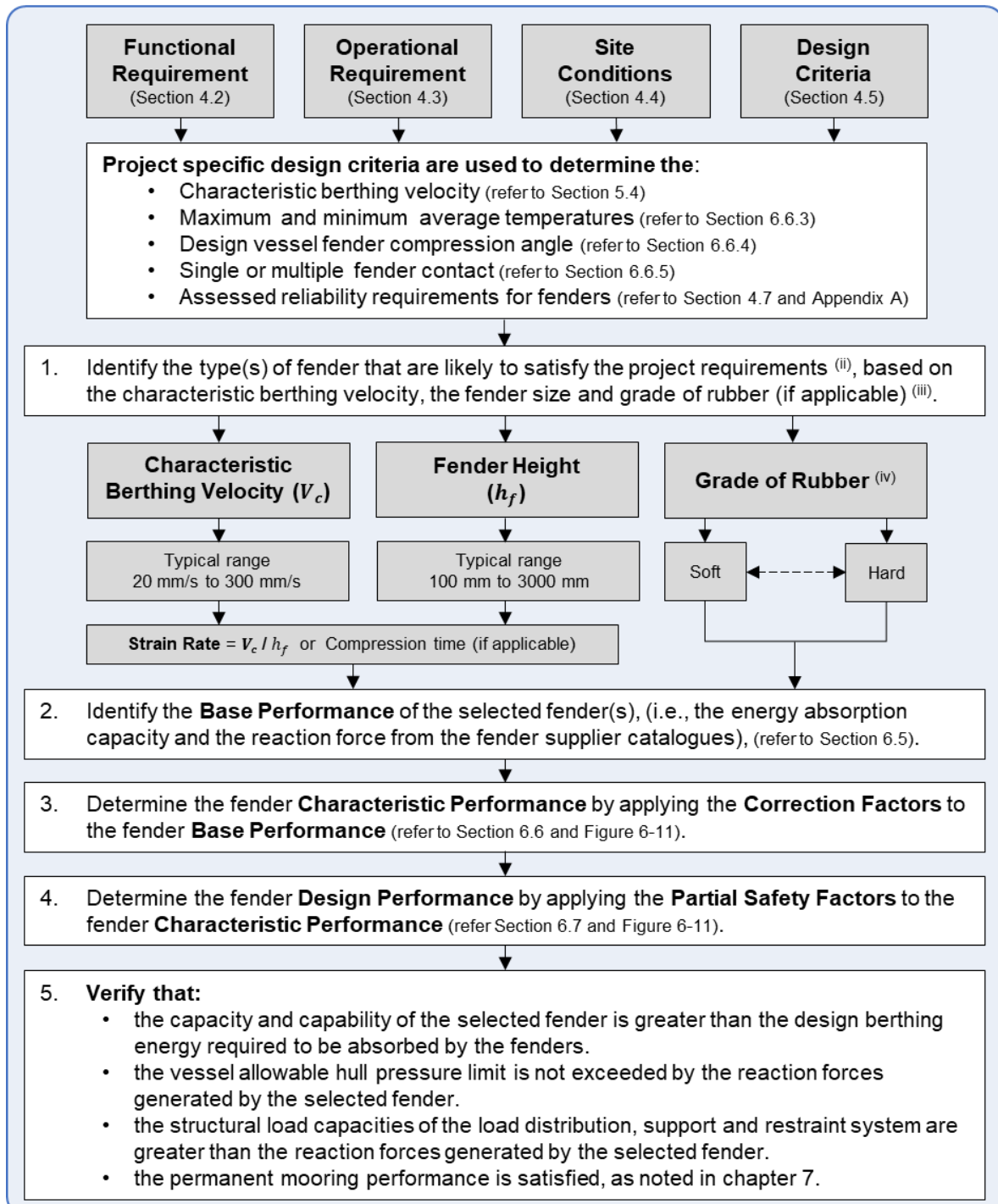
To determine the **Characteristic Performance** of the selected fender, the designer needs to determine the applicable correction factors, using the project specific design criteria. The base performance of the fender is then multiplied by these correction factors, to determine the fender characteristic performance, refer to point 3 in Figure 6.1 and to Section 6.5.

The **Design Performance** of the selected fender requires the designer to determine the applicable partial resistance factors and apply these to the fender characteristic berthing energy and reaction force. Refer to point 4 in Figure 6.1 and to Section 6.7.

At the project outset, specific site-based design criteria may not be available or may need to be estimated. The designer may need to undertake a sensitivity analysis to determine the effect of changes to the assumed design criteria on the required size and performance properties of the selected fender. The designer may also need to iteratively assess a range of different fender types and sizes. The design process is likely to identify several different fender types and sizes that could accommodate the required range of vessels and conditions.

This design process is repeated until the designer has assessed all possible fender solutions and/or identified the optimum solution. It is highlighted that the most effective efficient energy absorbing fender may not be the most suitable fender for the required application.

OVERVIEW OF FENDER SELECTION PROCESS ⁽ⁱ⁾ using project specific design criteria



(i) This flow chart provides an overview of the fender selection process. Many factors (in addition to those presented in this flowchart) influence the fender selection process. Refer to Sections 6.4 to 6.8 for further details on the range of factors to be considered.

(ii) At this stage of the fender selection process, the type of fender system is not expected to have been selected. This flow chart is intended to illustrate the process for selecting the type of fender that will satisfy the energy absorption requirements.

(iii) The type of rubber (natural or synthetic or blend of both) can also be considered in the initial selection of the type(s) of fender to be selected.

(iv) Harder rubber grades may be more economical as it can reduce the required fender height and mounting space. However, the rubber resilience may be sacrificed raising concerns surrounding fatigue resistance.

Figure 6.1: Overview of fender selection process

6.2.2 Fender Selection Using Pre-Set Design Criteria

To quickly assess and compare numerous different types and sizes of fenders, fender suppliers publish fender characteristic performance information ($E_{f;c}$ and $R_{f;c}$) that utilise 'pre-set' design criteria for each type and rubber grade of fender. Using this approach, designers can select fenders 'at a glance', as the appropriate correction factors will have already been applied, such as the following. (Refer to Section 6.6 for more information on correction factors.)

Berthing Velocity (moderate conditions):	175 mm/s
Temperature (Energy Absorption):	25 °C
Temperature (Reaction force):	-10 °C
Fender Compression Angle	5°
Single fender contact	

When using pre-set design criteria, the designer must be aware that this methodology is intended for preliminary fender selection only. The final fender selection may change due to differences between the project design criteria and the pre-set design criteria.

These pre-set design criteria are used to provide the designer with a quick reference comparison, and 'look-up' tool for fender selection and may well differ from the final project specific design criteria. The designer will need to check the suitability of the selected fender prior to finalising the selection.

The partial resistance factors will also need to be applied to complete the fender selection using this pre-set criteria method. If the project design criteria are significantly different to the pre-set criteria, the designer is recommended to revert to the methodology outlined in Figure 6.1.

6.3 Fender System Selection

The selection of a suitable fender system may need to consider a wide range of potentially limiting conditions or restrictions. These may include, but are not limited to, specific types of vessels, matching replacement fender systems to an existing fender system, large vessel stand-off distances, accommodating vessel gangways, vessels with belting, etc.

The range of vessel sizes using the berth will also need to be carefully considered as this may influence the fender type, spacing (pitch) and the size(s) (heights) of the fenders selected.

Sometimes there may be one primary overriding criterion which may govern the whole fender system selection process. Refer to Chapter 2 for a summary of fender types and systems and the preliminary considerations linked to their selection.

As part of the fender selection and design process, the designer is recommended to consider all fender system options that may be considered suitable. Each potential solution should then be assessed against the project specific criteria to determine the most efficient, robust and cost-effective design solution.

Table 6-1 provides an overview of which types of fender system may be suitable for use with a range of different vessel types. Further advice on which fender types may be most suitable to a particular situation, location or vessel, can be obtained from fender manufacturers.

Vessel Types \ Fender System Types	Arch	Cell	Cone	Cylindrical	Donut	Element	Extruded	Foam	Hydropneumatic	Parallel Motion	Pile	Pneumatic	Roller	Wheel
Bulk Carriers	○	●	●	●		○				○	●			
Car Carriers		●	●	○		●				○	○			
Container Vessels	●	●	●	●		○					●			
Cruise Vessels		●	●	○		●		●	●		○	●		
Fast Ferries		●	●			●				○	○			
Fishing Vessels	●			○							●			
Gas Carriers		●	●	○		○		○	○	●	●	○		
General Cargo	●	●	●	●		●		○	○		○	○		
Navy Surface Ships	○	●	●	○		●	○	●	●		○	●		
RoRo Ferries		●	●	○	○	●	○			●	●			
Submarines	○				○				●		○			
Tankers	○	●	●	●		○		○	○	●	●	○		

● Generally considered suitable for this vessel type.
 ○ Suitable for some vessels. Potential limits to be considered.
 Generally not considered suitable for this vessel type except in special applications.

Table 6-1: Application of fender systems for various vessel types

In addition to differing vessel types and as an additional starting point for fender system selection, Table 6-2 provides an indication as to which fender types may be suitable for use in a range of general marine applications.

The designer should not consider Table 6-1 and Table 6-2 to be prescriptive or definitive and the final fender selection should be verified using the calculation methods included in this guidance. The designer is also recommended to consider all permanent and temporary conditions associated with the operation of the berth (e.g. crane offload), to ensure that the most suitable fender type and fender system is selected.

Applications \ Fender System Types	Arch	Cell	Cone	Cylindrical	Donut	Element	Extruded	Foam	Hydropneumatic	Parallel Motion	Pile	Pneumatic	Roller	Wheel
Belted vessels	○	●	●	○	●	●	○	○	○	●	○	○	○	○
Continuous berth	●	●	●	●		●	●	●	○	○	○	●		
Dolphin	○	●	●	○	●	●	○	○	○	●	●	○		
Ice Zones	○	●	●	○		●	○	○			○		○	○
Large bow flares		●	●	○	●	●		●	○			●	○	○
Large tidal zones	○	●	●	○	●	●	○	●	●	●	●	●	○	○
Lay-by berths	●	○	○	●	●	●	○	○	●	○	●	●	○	○
Lead-in structures	●	○	○	○	●	●	○			○	●		○	○
Lock entrances	●	○	○	○	●	●	○				●		●	●
Lock walls	○			●			●						●	●
Low-freeboard vessels	○	●	●	○	●	●	●	○	○	○	●	○		
Monopile	○	●	●		●	●	○			○				
Permanent moorings	●	●	●	●		●	○	○	○	○	○	○		
Ro-Ro ramp	●	●	●	●		●	○			○			○	
Ship carried fenders								○				○		
Ship-to-ship								●	○			●		
Shipyards	●	●	●	●	○	●	●	●	○			●	●	●
Small harbours	●			○							●			
Small tidal zones	●	●	●	●	●	●	●	●	●	●	●	●	●	●
Temporary berths	○			○			○	●	○			●		

● Generally suitable for this application.
 ○ Suitable for some applications. Potential limits to be considered.
 Generally not considered suitable for this marine application.

Table 6-2: Application of fender systems to various marine applications

6.4 Factors Influencing Fender System Selection

Fender systems can be designed to function as single stand-alone fender systems (e.g. berthing dolphins) or they can function as part of a combined multiple fender system arrangement (e.g. on a continuous berth). A wide range of factors and variables should be considered by the designer when selecting a suitable fender system. These factors are outlined in the following sections.

6.4.1 Bow Radius

A vessel's bow radius (R_B) at the level of the contact with the fender system, is an integral part of determining the pitch of the fender system. In the absence of vessel specific information, the bow radius may be estimated using equation (6-1). For more detailed information on hull shape to assist with the estimation of the bow radius refer to Chapter 0.

$$R_{B,F} \approx \left(\frac{B}{4} + \frac{L_{bow,F}^2}{B} \right) \quad (6-1)$$

Where:

- $R_{B,F}$: Bow radius at the elevation of the fender [m]
- B : Beam of the vessel [m]
- L_{BP} : Vessel length between the perpendiculars [m]
- L_{OA} : Vessel length over all [m]
- $L_{bow,F}$: Distance from the bow tangent point to the bow at the elevation of the fender [m]

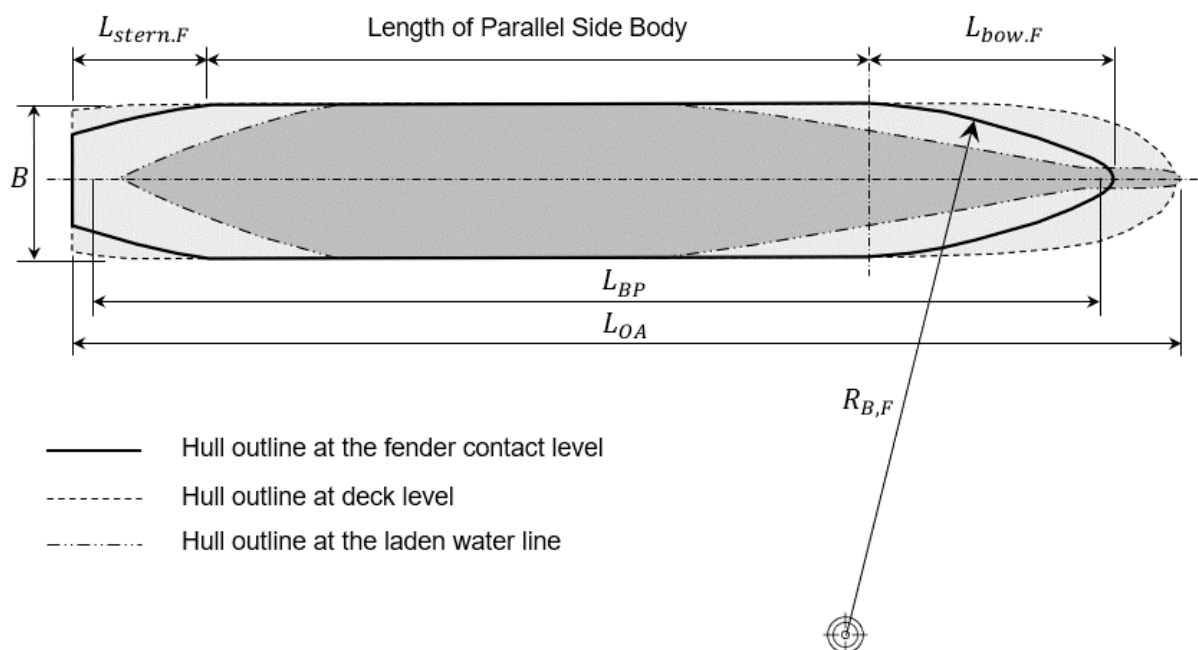


Figure 6.2: Bow radius (R_B) and length of bow flare (L_{Bow})

It should be noted that the accuracy of the formulae for $R_{B,F}$ in Equation (6-1) is less reliable for the calculation of bow radius below the fully laden water line. For part laden vessels and vessels in ballast at a draught of less than 80 % of the fully laden draught, it is recommended that designers should use measured data from the design vessel(s) hull plans. For guidance on $L_{Bow,F}$ (the distance from the bow tangent point to the bow at the elevation of the fender), refer to WG 235 Appendix A.

6.4.2 Bow Flare

The vertical angle of a vessel's bow is known as the bow flare. Many vessels have considerable amounts of topside 'flare' forward and aft, below the main deck level. When a vessel is approaching at an angle to a berth, the fender system will need to accommodate the

differences and changes in this vertical angle of the vessel hull profile at the fender contact locations. High tidal ranges will exacerbate this consideration.

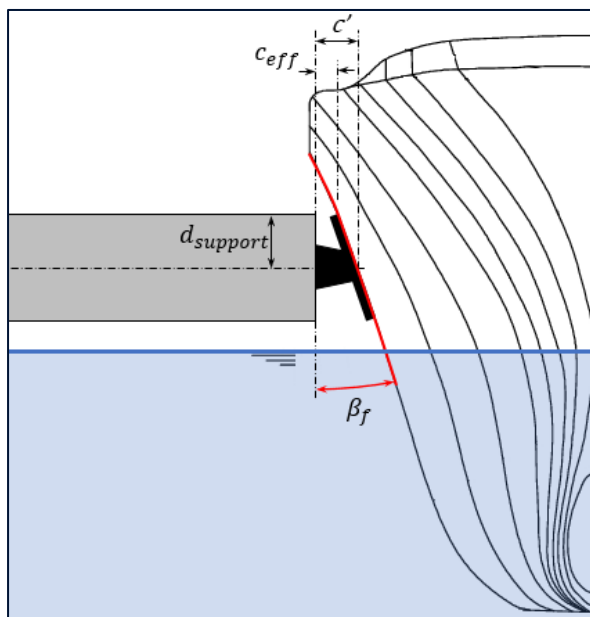


Figure 6.3: Bow flare (β_f) and clearance to supporting structure (c_{eff})

The bow flare angle (β_f) at the contact point with each fender system can vary considerably, depending on the water level in relation to the berth (i.e. tidal water level) and the berthing vessel type, size and draught.

As the bow flare angle is often not known, and information required to calculate the bow flare angle is also limited, the designer may need to consider a range of bow flare angles to assess whether the effective clearance provided between the vessel hull and the supporting structure and quayside equipment (e.g. cranes, ship-loaders, etc.) is sufficient, for a range of vessel approach angles, refer to Section 3.3 .

An estimate of the effective clearance (c_{eff}) to the supporting structure due to the

bow flare angle (β), can be calculated as follows:

$$c_{eff} = c' - d_{support} \times \tan \beta_f \quad (6-2)$$

Where:

- c_{eff} : Effective clearance between supporting structure and vessel hull due to bow flare [m].
- c' : Horizontal distance between the fender centreline and the support structure due to the bow radius and fender compression [m]. Refer to WG 145 2022 Appendix F for more information.
- $d_{support}$: Vertical elevation from fender centreline to the vessel deck or to the top of the supporting structure, whichever is the lower [m]
- β_f : Bow flare angle

The projection of the vessel hull profile over the compressed fender line must be considered in assessing clearances to quayside furniture and equipment. Clearances should account for the minimum separation distances required to the quayside equipment, its placement during vessel berthing, the berthing angle and vessel hull shape geometry.

The designer may elect to complete a geometric assessment of the minimum vessel freeboard, draught and the lowest tidal water level to ensure that the vessel bow flare does not contact the upper surface or edge of the supporting structure during berthing, at the incidental berthing angle (α_i).

In such cases, the effective clearance to all components of the fender system, including the fender panel, brackets, chains, etc. may need to be checked. In extreme cases, the bow or stern flare can overhang the supporting structure.

In addition, geometric checks should be carried out to confirm that the lowest part of the fender system will be positioned at a sufficiently low level to prevent the fender system catching on low freeboard vessels at low tidal states. The level of the upper part of the fender system will also need to be configured to accommodate contact with vessels with considerable amounts of flare angle.

The designer should consider the combined angle of compression of the fender, taking account of the angular compression due to the bow radius and the bow flare, or similar stern geometry if stern impact is dictated by vessel manoeuvres. This combined effect may reduce the overall energy absorption capacity of the selected fender.

6.4.3 Fender System Pitch

The fender system pitch (p_f) refers to the horizontal distance or spacing between each fender system. Various factors should be considered when determining the fender pitch. These include, but are not limited to, the following:

- the range of design vessels. Larger and heavier vessels typically can accommodate a greater fender pitch with the optimum pitch varying with differences in the vessel bow radius.
- different types of fenders have different properties and performance characteristics that affect the required pitch between fenders.
- the primary dimensions of the fenders can affect the required pitch (e.g. the stand-off distance of the berthing line from the supporting structure). Larger fenders typically can accommodate a wider spacing and can also influence the design of a new or replacement supporting structure (i.e. headstock separation distance).
- the absorption of larger berthing energies can require a reduced fender system pitch to ensure sufficient energy absorption capacity.
- specific operational requirements or limitations of the marine facility (e.g. allowable vessel movements or maximum berthing forces) can affect the fender system pitch.
- the existing port infrastructure geometry (e.g. required dolphin separation, quay wall structural arrangement, pile spacing, the level of the fender system above the water line, etc.) can all influence or limit the pitch of a fender system.
- environmental factors (e.g. the requirement to accommodate a certain range of wave heights, currents or water levels) may require a reduced fender pitch.

Determining the optimal pitch is therefore an involved process, typically requiring an iterative approach and the consideration of a wide range of factors.

Often (but not necessarily always), the smallest design vessel determines the maximum fender pitch. However, small(er) vessels result in smaller berthing energy absorption requirements which consequently results in smaller fender compressions. Having considered the maximum pitch for the whole range of design vessels, and provided that it does not govern the selected fender size, the designer can take this possibility of smaller fender compression into account when determining the appropriate pitch.

For continuous berthing structures, an estimate of the maximum pitch should be calculated as part of the fender selection and design process. The maximum pitch ($p_{f,max}$) between two fender systems, for bow contact only, can be calculated as follows:

$$p_{f,max} \leq 2 \sqrt{R_B^2 - (R_B - h_{f,act} + c_{min} + (d_{support} \tan \beta_f))^2} \quad (6-3)$$

Where:

- $p_{f,max}$: Maximum fender system pitch [m]
- R_B : Bow radius [m]
- $h_{f,act}$: Horizontal distance from face of fender system to supporting structure, at actual fender compression, measured on the centreline of the fender for 50 % of the design energy (i.e. fenders each side of the critical location take 50 % of the design energy) [m]
- c_{min} : Minimum clearance to support structure [m]
- $d_{support}$: Vertical distance from fender centreline to the vessel deck or to the top of the supporting structure, whichever is the lower [m]
- β_f : Hull flare angle at the critical location [°]

A minimum clearance (c_{min}), between the vessel hull and the supporting structure should be maintained, refer to Figure 6.4. This minimum clearance distance is recommended to be taken as 10 % of the uncompressed fender height (h_f), with a minimum of 250 mm.

If the berthing angle is such that bow or stern impact is possible and $d_{support}$ is greater than 2 H, particular care should be taken to consider the effect of (β_f), as impact with the support structure can occur in these circumstances. This can be overcome by ensuring the fender level is close to the deck level.

In the absence of a detailed assessment, as is often the case at the concept design stage, an estimate of the maximum pitch can be taken as 0.15 x the smallest design vessel length (LOA_s). Refer to Figure 6.4.

Whilst a large pitch may be preferable, this may allow some vessels, especially smaller ones, to contact the supporting structure in between the fenders. The pitch should be small enough to prevent a vessel hull contacting the supporting structure. The designer may elect to include smaller, intermediate fenders between the larger, primary fender systems to prevent smaller vessels contacting the supporting structure, refer to Section 6.4.5.

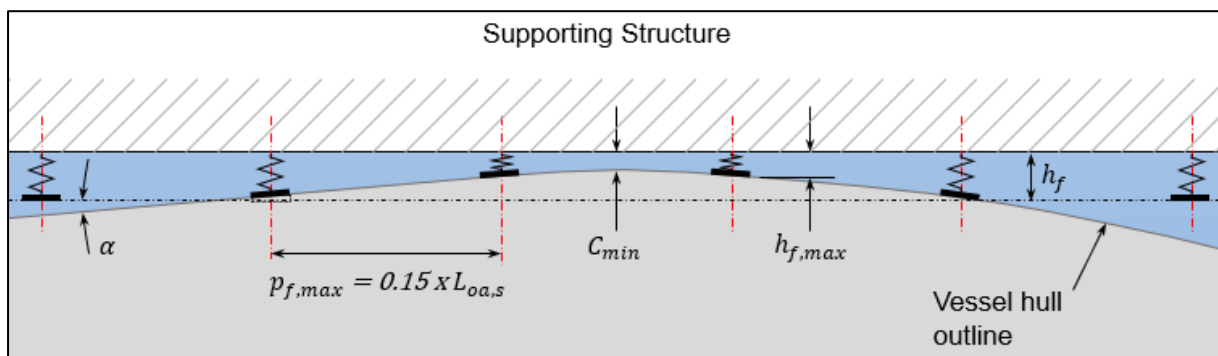


Figure 6.4: Estimate of maximum fender pitch and minimum clearance to the supporting structure

For small vessels (e.g. fishing vessels), the spacing of the fenders will need to be in the order of a few metres to avoid contact with the quay or supporting structure. Small vessels are sometimes berthed stern-on to the quay, also necessitating a close fender pitch.

For vessels approaching at small berthing angles, typically multiple fender systems will be compressed sequentially and by varying amounts. Should the adjustment of the fender pitch

to suit the design vessel geometry not be possible, (e.g. when designing replacement fenders for an existing berth), the designer may need to consider adjustments to the stand-off distance of the berthing line from the supporting structure. This can be done by using larger fenders or mounting fenders on quay line 'out-stands'.

6.4.4 Single Fender System Contact

For a single fender system, (e.g. a berthing dolphin), irrespective of the vessel approach angle and bow radius, the selected fender should be capable of absorbing all the design berthing energy of the approaching vessel, refer to Figure 6.5. Therefore, the worst-case design scenario will be the single fender system contact.

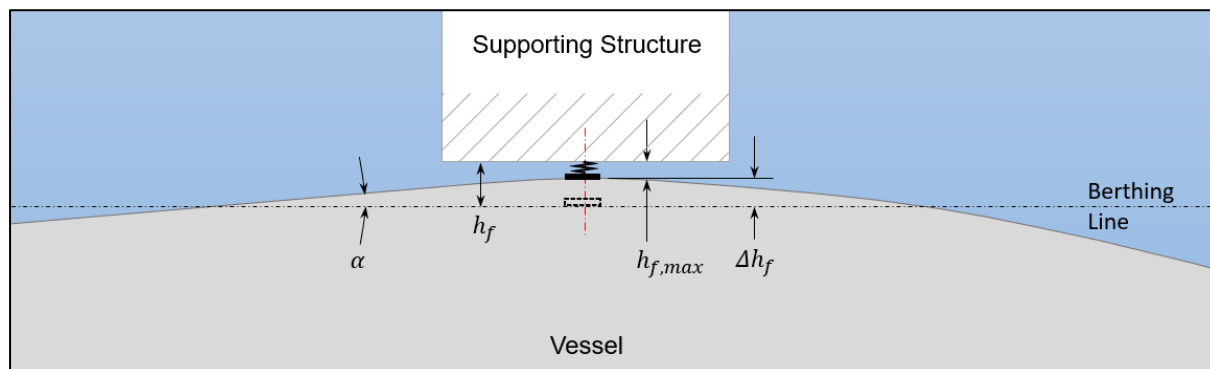


Figure 6.5: Single fender contact

6.4.5 Multiple Fender System Contact

Whilst vessels may attempt to approach parallel to a berth, in practice, given the number of variables influencing the vessel approach and the vessel hull geometry, true parallel berthing is difficult to achieve. Nonetheless, multiple fender system contacts can and does occur at small berthing angles.

For continuous berths, depending on the approach angle of the vessel, the bow radius and the fender system pitch, vessels may contact multiple fender systems sequentially and by varying amounts when berthing.

Multiple fender contacts will provide the greatest berthing energy absorption capacity. This will consequently induce multiple reaction forces on the supporting structure. The supporting structure should therefore be capable of resisting the cumulative sum of these multiple reaction forces from multiple fender system contacts.

The total berthing energy absorption capacity of multiple fender system contacts equates to the sum of the energy absorbed by all the compressed fenders. If one fender system is initially contacted, closely followed by the adjacent fender systems, the energy will be absorbed by all the fenders that are contacted by the vessel. The energy absorbed by each individual fender system will vary depending on the degree to which each fender system is compressed. The fender at the initial point of contact will have the largest deflection and the largest associated reaction force, refer to Figure 6.6.

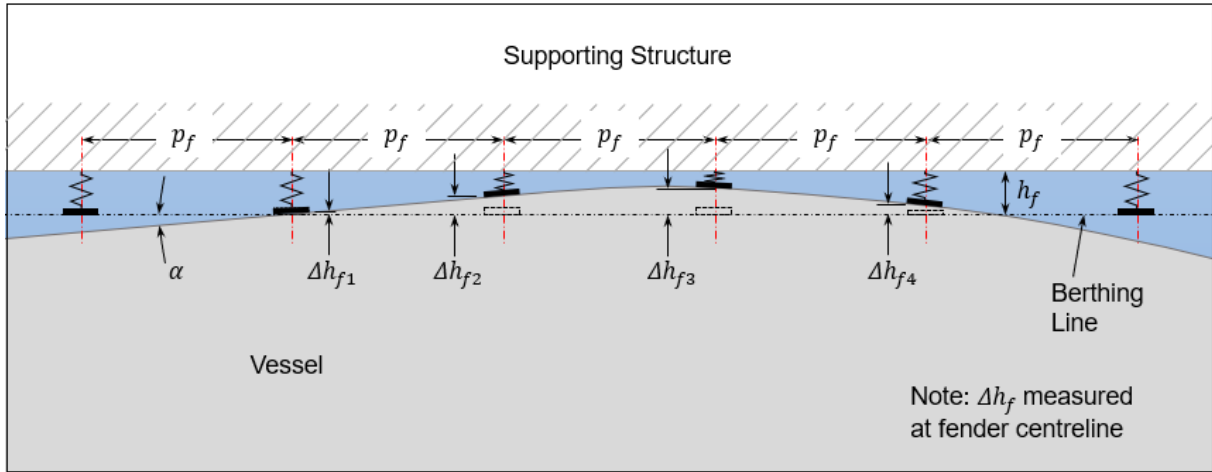


Figure 6.6: Contacted fenders at larger berthing angles (with contact on bow flare)

For vessels with approach angles that are almost parallel to the berth, this can result in many more fenders being contacted, with each contacted fender being compressed by varying amounts, refer to Figure 6.7.

For all cases of multiple fender contact, the cumulative energy absorption capacity of the multiple fenders that are compressed is therefore greater than that for a single fender contact. For multiple fender contacts with buckling fenders, the nonlinear force deflection characteristics of the fender will result in larger overall reaction forces on the structure compared with single fender contact.

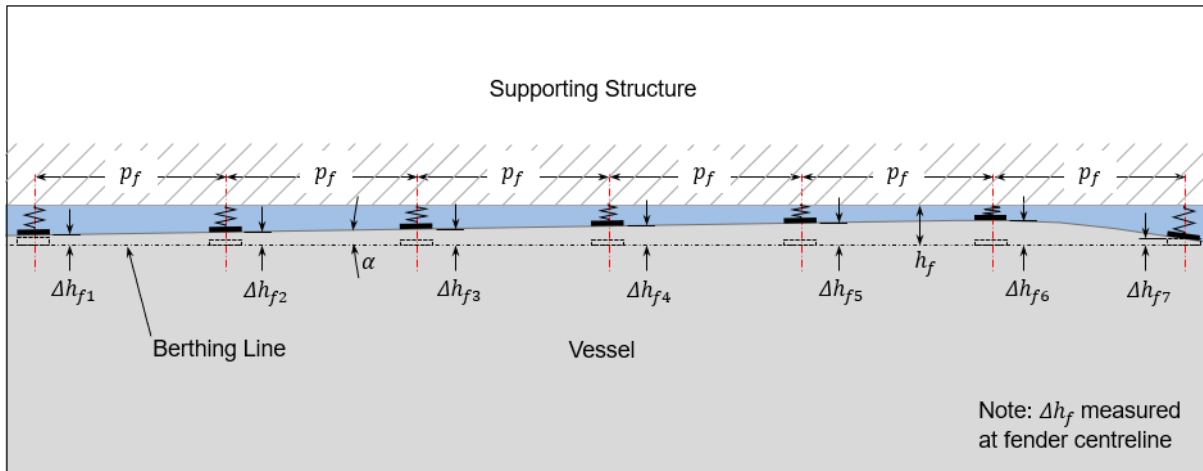


Figure 6.7: Contacted fenders at smaller berthing angles (almost parallel to the berth)

The designer should also consider the range of design berthing velocities associated with the range of design vessels. The associated differences in berthing velocity, considered in combination with the vessel approach angle and vessel geometry, will result in numerous berthing scenarios that may all need to be considered to identify the optimum solution.

Changes in the selected fender pitch, fender type and fender height will also influence the number of fenders that are compressed for a multiple fender contact scenario. The smaller the pitch, the more fenders are likely to be contacted.

If the range of design vessel sizes is large, it is often not economical or practical to arrange the fender systems for the largest vessels, using the pitch calculated for the bow radius of the smallest vessels. In such cases, smaller alternative fenders can be considered between the primary fender systems that are provided to accommodate the largest vessels.

The designer is recommended to consider both single and multiple fender system contact scenarios, for a range of vessel approach angles. Contact with multiple fender systems during berthing increases the potential berthing energy absorption capacity, potentially enabling a smaller fender to be selected or larger vessels to be accommodated on the berth.

Should the initial berthing contact be made by the stern quarter of a vessel and, as the hull radius is typically considerably much smaller at this location, the designer should consider whether single fender contact is also a possibility.

6.4.6 Vessel Hull Structure

The structural capacity of the vessel hull in resisting berthing forces (and thus the allowable hull pressures) needs to be assessed early in the fender system selection process. Guidance on vessel hull structure load capacities is included in Section 6.8.

Limitations on the structural capacity of the vessel hull and belting (if present) may influence the size and number of fender panels required to distribute the fender reaction force into the berthing vessel. The designer should assess all applicable berthing impact loads, the structural capacity of the vessel hull and the belting, and the area over which these loads are to be distributed, to ensure that the selected fenders can absorb the design energy.

6.4.7 Vessel Belting

To accommodate vessels that are equipped with belting, the designer should consider how the belting will interact with the selected fender system. Beltings are often considered to be sacrificial vessel bound protection. A certain amount of damage to beltings is possible and inevitable as this is the primary point of contact between the vessel and the installed fender system. Damage to vessel belting is not expected to occur under normal berthing operations. Vessels that have damaged belting also have the potential to cause damage to the fender during berthing and the designer may wish to consider selecting fenders that are less likely to be affected by damaged belting. Certain fenders can be damaged or perform inefficiently when contacted by vessel belting.

To assist with this process, the designer is recommended to obtain details of the belting profile, structure and load capacity, so that the interactions with the selected fender can be assessed in detail. Friction between the belting and the fender should also be considered, particularly as the belting faces are often heavily worn, with minimal paint system coating as it will have been worn away. Refer to Section 8.3 for more information on applicable coefficients of friction.

Where fender panels have been selected, the potential for double hull contact and high horizontal line loads within the fender panel should be assessed, refer to Section 6.4.8.

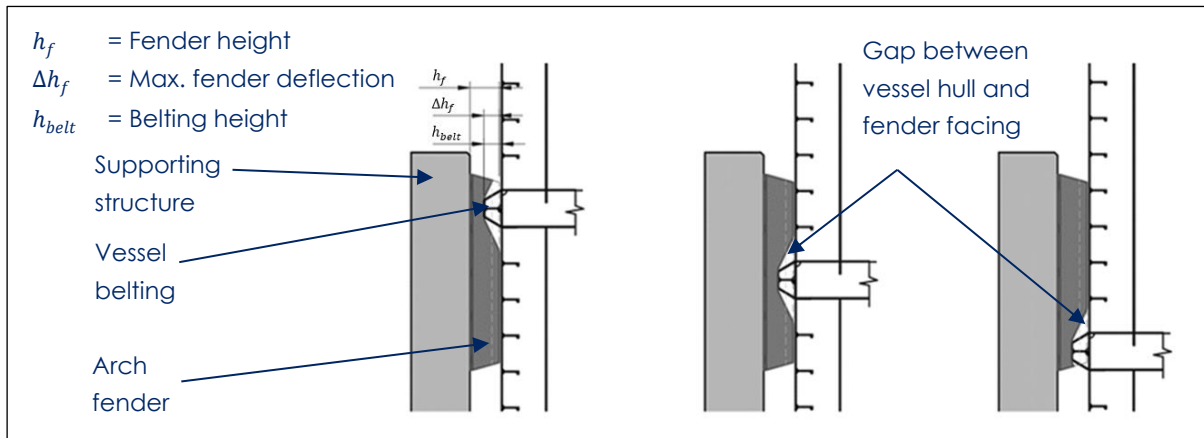


Figure 6.8: Arch fenders with maximum deflection smaller than belting height

Where arch type rubber fenders without fender panels have been selected and are mounted vertically on a supporting structure, the maximum energy absorption capability of the arch fender is limited to the actual contributing length of the fender. The remaining uncompressed lengths of the arch fender will therefore not contribute to the absorption of the berthing energy, refer Figure 6.8.

If the maximum fender compression is greater than the belting height, direct hull contact may also occur on sections of the hull plating adjacent to the belting. This will provide some increase in energy absorption however also result in a significantly larger reaction force on the vessel hull, refer Figure 6.9.

The sizing and selection of the arch fender is dependent on the belting geometry and a larger section arch fender may be required to accommodate the design berthing energy. An incorrectly sized fender could result in damage to the supporting structure or to the vessel hull due to high, locally concentrated reactions forces.

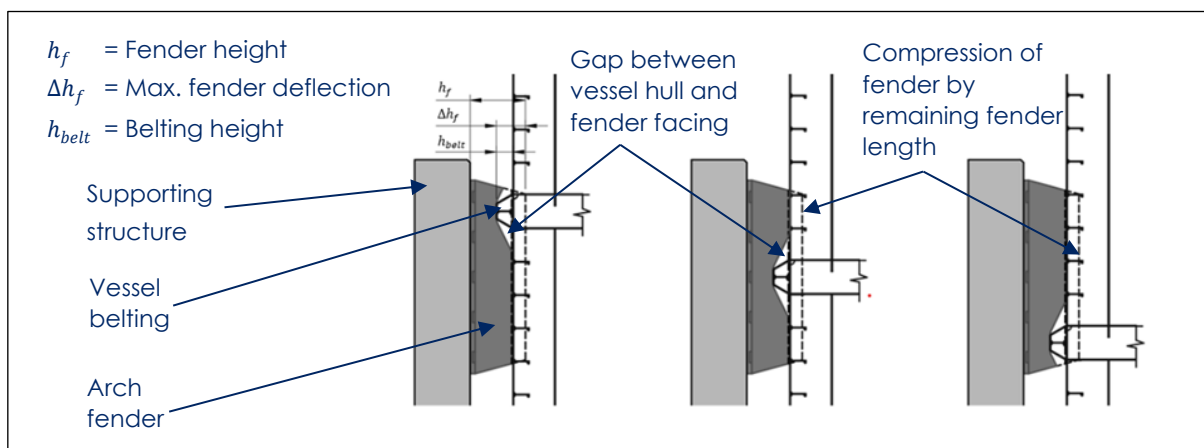


Figure 6.9: Arch fenders with maximum deflection greater than belting height

When considering the selection of other panel-free fender systems, e.g. foam, pneumatic or cylindrical fenders, the designer should consider the interaction of the fender with the vessel belting. This is of particular significance if the fender is mounted horizontally resulting in an increased risk of the fender becoming caught under the vessel belting as it moves up and down against the berth.

The designer should also take account of variations in the shape of the ends of the belting, changes in the belting cross sectional profile and/or discontinuities in the belting line (e.g. pilot door openings). An angular shaped belting end has the potential to cause damage to the fender. Consideration should be given to the use of fender chamfers and sloping fender profiles to prevent the belting becoming 'snagged' on the fender.

Multiple beltings at various heights on the vessel, and at differing positions along the vessel hull, also should be assessed in relation to the position of the vessel when berthing and when moored, potentially requiring the selected fender system to extend above the deck of the supporting structure.

6.4.8 Double Hull Contact

Double hull contact by a fender system typically occurs with vessels that are equipped with belting, refer to Figure 6.10. Double hull contact is of particular significance when a fender panel is included. This issue will be exacerbated where a large tidal range exists requiring longer fender panels.

Double hull contact for fender systems without fender panels, (e.g. foam or cylindrical fenders), is less likely to occur; however the designer should consider how this type of fendering will interact with the vessel belting during vessel berthing.

The designer may consider undertaking a geometric assessment of the vessel and fender system interaction at extreme water levels, to develop the fender and fender panel geometry and to check that no part of the vessel may be damaged during berthing (e.g. low level windows and doors on cruise vessels).

Double hull contact will result in loads being imposed on the vessel belting, as well as on the vessel hull plating both above or below the belting line, as illustrated in Figure 6.10. The type of facing of the fender panel (e.g. steel plating or low friction panels) will also affect how the fender panel interacts with the vessel hull plating, with these concentrated loads potentially cause buckling and deformation of the vessel hull plating.

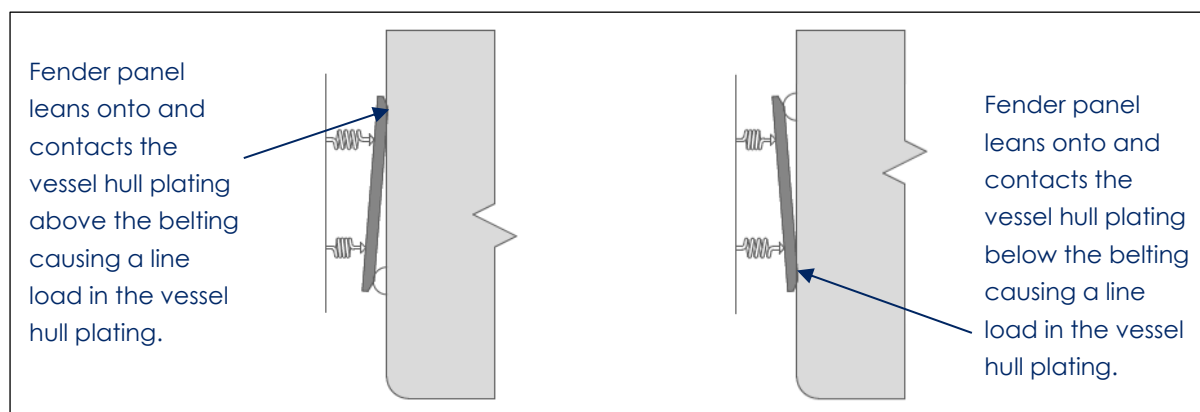


Figure 6.10: Belting causing vessel hull double contact and line loads

For long fender panels, multiple fenders may be required to support the fender panel and efficiently absorb the berthing energy.

6.4.9 Type of Supporting Structure

The type of supporting structure to which the fender system is to be attached is a primary consideration when selecting a suitable fender system. A supporting structure consisting of a gravity or retaining wall is generally stable against external forces such as the vessel berthing or wave forces. The reaction force imposed by the fender system is also relatively small when compared with the earth forces on the retaining wall and the effect of the characteristic correction factors on the berthing energy and reaction forces may not be critical.

However, in the case of jetties, open piled wharf structures, guiding structures and dolphins, the influence of the reaction force from the fenders may be a defining load case for the design of the overall structure. In such cases, all characteristic correction factors are significant and should be assessed in detail.

Based on the design of the proposed or existing supporting structure, limitations associated with the area upon which the fender can be mounted, the required stand-off distance, minimum anchor edge distances, etc., will also need to be considered by the designer.

The age of the supporting structure will also need to be considered in detail, (i.e. 'new-build' design or an existing structure) to ensure that the residual capacity of the existing structure can stand the reaction forces imposed by the selected fender system.

The geometry of an existing structure may preclude the use of certain fender systems that may necessitate significant modifications to the supporting structure. Maintaining a straight berthing line and/or working within a set stand-off distance may also limit the options for fender and fender system selection.

6.4.10 Flexible Dolphins

Flexible dolphins (typically mono-piles) are vertical or near vertical piles cantilevered from the sea, harbour or river bed. These load-sensitive structures absorb the vessel berthing energy by the combined deflection of the mono-pile and the compression of the associated fender system.

They are commonly used at berths where unloading takes place at dedicated facilities, e.g. at liquid bulk (oil, gas) jetties, where the berthing structures can be separated from the operational (loading) platform.

Flexible dolphin piles are generally of circular shape and consist of several sections welded together, each with different wall thicknesses to satisfy the changes in bending moments at various levels. The upper sections of the flexible dolphin mono-pile should be fabricated to be easily connected (i.e. bolting) to facilitate onsite connection of the upper section, deck, fender system mounting or other fixtures and fittings, e.g. access ladder.

The energy absorption capacity of a flexible dolphin is proportional to the square of the steel stress and linear to the selected pile wall thickness. Hence, the use of a higher-grade tensile steel and a large wall thickness is effective for providing high energy absorption characteristics. The energy absorption capacity can be increased by using a larger pile diameter and/or wall thickness.

The designer should consider all relevant design codes while designing large diameter mono-piles and considering the effects of local buckling. The mono-pile can also be filled with a mixture of sand and gravel to assist with preventing local buckling.

Consideration should also be given to the corrosion of the steel pile and the associated welds. If corrosion, combined with fatigue effects, are considered significant, the designer should consider a lower steel grade strength.

However, flexible dolphins are not suited to all geological conditions and their capacity is dependent on the properties of the ground conditions. The ground conditions should be capable of resisting the horizontal loads exerted by the embedded length of the pile caused by the design berthing impact of the vessel. The ground conditions should also be capable of returning the pile to its original position when berthing or other applied forces have ceased to act, including for a nominal allowance for 'post-holing'.

The designer should also consider the design bed level and make an allowance for potential scour and changes in the bed level which may affect the level of fixity of the pile and potentially reducing the energy absorption capacity of the structure.

As a minimum, flexible dolphins should be designed to resist the following:

- Berthing impact forces from the range of characteristic design vessels
- Forces imposed by vessels while moored at the dolphins
- Wind, wave, and current forces acting directly onto the flexible dolphin
- Mooring line forces (if the berthing dolphin is also used for securing mooring lines)

The design energy absorption performance of the selected flexible dolphin pile ($E_{d,d}$) and the selected fender system ($E_{f,d}$), should be greater than the design (berthing) kinetic energy ($E_{k,d}$) of the berthing vessel:

$$E_{f,d} + E_{d,d} \geq E_{k,d} \quad (6-4)$$

The selection of a suitable flexible dolphin and fender system should also consider the following:

- The maximum reaction forces generated by the selected fender system ($R_{f,d}$) must be able to be resisted by the structural capacity of the selected mono-pile.
- The mono-pile should also have sufficient residual structural capacity to absorb the proportion of the vessel berthing energy to be resisted by the mono-pile.
- Mounting of the selected fender system onto the mono-pile may be difficult due to the comparable limited size of the mono-pile.
- The distance between the face of the fender panel (fender line) and the mono-pile structure should be sufficient to ensure that contact between the vessel (e.g. hull plating, heel, belting, etc.) and the mono-pile (or its potentially larger platform on top) cannot occur when the design berthing load is applied.
- The flexible dolphin structure may have potential restricted access that requires the selection of a low maintenance fender system.
- For berths subject to numerous berthing events, it is recommended that the designer considers the impact of fatigue on the design of the flexible dolphin pile, (e.g. a permanent FSRU in exposed conditions).
- The designer should consider appropriate load factors to determine the design berthing energy and to ensure that the permissible elastic deflection of the flexible dolphin is not exceeded for the selected range of berthing energies and associated reactions forces.

6.4.11 Fender System Elevation

The selection of a fender system located in tidal waters should be able to accommodate the full range of design vessels, for the full range of water levels expected to be experienced at the berth.

The selected fender system should also be capable of accommodating the vessel load condition (laden or in ballast) and to accommodate the vessel's response to sea state conditions (e.g. waves) and other design environmental or meteorological effects (e.g. storm surge), whilst the vessel is alongside the berth. Designers should also consider the position of the top of the fender system in relation to the top of the supporting structure as mooring ropes, cables, or loading platforms from vessels using the berth could become snagged if parts of the fender system protrude above the top of the supporting structure.

6.4.12 Pneumatic and Foam Fenders

The selection of pneumatic and foam fenders requires the consideration of several additional factors. From a design perspective, both these types of fenders should generally be fully supported by a smooth surface (e.g. concrete, steel, etc.) or backing structure. Full flat contact and support between the footprint of the fender and the supporting structure is required to support the fully compressed surface area of the fender and to provide the full design performance of the fender at 60 % compression.

For pneumatic fenders, the height of the compressed fender at 60 % compression is approximately 1.4 times the diameter of the fender. The corresponding flat contact footprint height with the supporting structure is approximately 0.95 times the fender diameter. For foam fenders, the vertical height of the compressed fender at 60 % compression is comparable to the uncompressed diameter of the fender. The corresponding flat contact footprint height with the supporting structure is approximately 1.1 times the fender diameter.

The elevation of the backing structure and the compression of the fender relative to the waterline should also consider the potential for vertical movement of the fender caused by variations in water level due to tides and waves and variations in the vessel's draught as it is loaded or unloaded. This may require the vertical height of the backing structure to be increased to accommodate potential vertical movement of the fender.

When the backing structure surface edges are close to the edges of the compressed fender footprint, the backing surface should have a smooth radius or chamfer to prevent potential incidental damage to the fender skin.

In circumstances where full backing to the foam fender cannot be provided, the performance capabilities of the fender will be reduced as the compression of the fenders will be limited by the extent of the backing surface. For reduced backing structure surface areas, the diameter of the fender should be reduced to accommodate the available extent of contact surface area. Alternatively, the selected fender should be designed for the required performance at a partial compression of the fender to match with the available contact footprint related to the backing structure surface height.

A full backing structure is not always necessary for pneumatic fenders as the energy absorption mechanism differs for pneumatic fenders compared to that of foam fenders. Pneumatic fenders can remain fully functional even at a compressed angles of up to 20 degrees. The provision of a backing structure for pneumatic fenders is primarily required to enhance the

overall durability of the fender as compression against uneven surfaces will induce additional stresses on the internal fender body.

Changes to the fender selection may be accomplished by adopting the same diameter with a higher performance capacity, or with a reduced diameter and increased length of fender. The fender should not be compressed beyond the available backing surface limits.

Pneumatic and foam fenders also have the capability to accommodate many different belting profiles whilst maintaining the stated design performance. Specific belting profiles should be evaluated based on the relative fender diameter and performance. In such cases, the maximum design compression distance is measured from the face of the belting (not the vessel hull) to the backing structure and the maximum energy absorption capacity will be reduced accordingly.

When considering the selection of pneumatic or foam fenders for a particular berth, designers should be aware of the potential for damage to occur to vessel beltings. This damage could rip or puncture pneumatic or foam filled fenders.. Consideration should also be given to the potential for fenders to get caught under larger vessel beltings, forcing the fender downwards on a falling tide, creating abnormal loading to the support restraints and the ends of the foam fenders.

For foam fenders, horizontally or vertically mounted strips of low friction facing material (e.g. UHMW-PE) can be installed on the face of the backing structure or concrete surface. These low friction strips can help reduce the wear to the skin of the foam fender caused by the constant rubbing against the backing structure. The low friction strips also provide a locking function, helping to keep the fender in place when compressed and subjected to lateral vessel movements. The potential for localised permanent compression and creep of the fender associated with longer term or permanent mooring conditions should be considered. Further information on the use of pneumatic and foam fenders for permanent moorings is provided in Chapter 0.

6.4.13 Number and Size of Fenders

The performance of a fender system can be modified by using multiple fenders attached to one fender panel. Whilst doubling the number of fenders above or next to each other, will in theory double the energy absorption capacity, the reaction force will also double, potentially having a significant effect on the supporting structure.

If a larger but softer fender is selected, the energy absorption capacity will be increased. For the selected design vessel, the associated reaction force may therefore be lower. However, the opportunity to utilise a larger fender might be limited by the required stand-off distance of the vessel from the supporting structure or certain cargo handling requirements.

In addition, if the vessel berthing velocity increases or a vessel larger than the design vessel calls at the berth, the designer should be aware that the potential maximum reaction force could be significantly greater and may detrimentally affect the supporting structure.

6.4.14 Shear of Fenders

When a fender is subjected to a lateral force, (e.g. a vessel movement parallel to the berthing line), shear forces can be induced within the fender system. These shear forces can cause

deformation and stress concentrations within the fender and in the worst case, result in the failure of the fender.

Excessive deformation of a fender through shear can impair the fender's ability to absorb vessel impacts effectively and limit the fender's ability to protect the vessel and the supporting structure. Shear forces can also create areas of localised stress concentration within the fender material, potentially weakening the fender and making it more susceptible to damage or failure. The designer should therefore be aware of the effect that shear has on a fender and on the energy absorption capacity.

Specific design information related to the compression characteristics and shear capacity of different fender types is typically not included within fender suppliers published catalogues. It is recommended that designers seek further information from the fender suppliers on the maximum shear force that the selected fender can withstand and the maximum displacement of the fender when subject to the design shear force.

When a fender system incorporates a fender panel, to resist excessive shear forces being imposed on a fender, it is common for shear restraint chains to be provided. Refer to Section 8.2 for further information on the design of shear chains. However, the designer should be aware of the importance of correctly positioning the shear chains and the angle of the chains to the berthing line to ensure that these chains function correctly and as intended.

6.4.15 Submerged Fenders

When fenders are submerged, the voids in some fenders can fill with water. This issue typically occurs with cone and cell type fenders. When a fender of this type is compressed during a vessel berthing, the water may be unable to escape quickly enough, resulting in significantly increased reaction forces.

To cater for this, adequate venting should be provided within the fender system to allow the water to escape within the time that the fender is compressed. Designers are recommended to request additional fender performance data from the fender suppliers for fully submerged fenders.

The possibility of marine growth obstructing any vent holes should also be considered when establishing the size of the vents and future maintenance provisions.

6.4.16 Fender Orientation

For some types of fenders (i.e. cone and element fenders), the fender orientation needs to be determined by the designer. The standard orientation of these fenders is for the wider footprint to be connected to the supporting structure and the narrower footprint to be connected to the back of the fender panel. For some berths, the designer may consider it beneficial to install the fender in a non-standard arrangement (i.e. 'inverted', with the smaller footprint attached to the supporting structure). This orientation requires less space on the supporting structure for the mounting bolts and thus could reduce/optimize the capping beam (or dolphin cap block) height and therewith result in concrete savings.

The energy absorption capacity of the fender is unchanged in the non-standard orientation, therefore in principle it is possible to use an inverted fender. However, inverting a fender can result in a different distribution of stresses, with potential additional stresses induced in certain parts of the fender.

Regardless of the fender orientation, care should be taken when designing the chain support system to ensure that the weight of the fender panel and shear forces from the vessel contact do not exceed the permissible stresses of the fender. As the geometry and wall thickness of fenders is not the same across all suppliers, the designer should be aware of the potential differences with this arrangement if considering the inverted orientation of a fender and is recommended to consult with the fender supplier in such cases.

To accommodate for the vertical movement of vessels on a berth, e.g. due to a large tidal range, the designer may consider selecting certain fender types that can be mounted in a vertical orientation, e.g. arch fenders. Alternatively, multiple fenders supporting a fender panel can be mounted one above the other, enabling taller fender panels to be utilised.

6.4.17 Ageing Effects on Fenders

The impact of ageing on the long-term performance of a fender within the design life of the selected fender can be considered by the designer. Ageing effects appear in the later stages of fender life, typically after more than 25 years of operation and use. As a result of the ageing effects on rubber, an increase in the fender reaction force can occur. The ageing effects are influenced by a range of factors, including:

- Environmental temperature/humidity/ozone concentration
- Rubber compound composition
- Amount of recycled rubber/white fillers
- Fender rubber thickness
- Curing system used in the fender manufacturing process

The effects of ageing on fenders are difficult to measure. The effect of sunlight, oxygen and ozone will largely occur within the surface zone of the fender leading to a hardening of the fender surface. However, the ageing effects of heat will penetrate deeper into the fender body and could cause further fender degradation over time.

Advances in technology have enabled sensor-based mechanisms to be installed on some newly manufactured fenders to measure the deflection of a fender over time and to provide a comparison of fender deflection for similar vessels and similar berthing velocities. Eventually, fender hardening can lead to increases in the fender reaction force generated by the fender for the same or reduced fender compression. There are (2024) no results currently available of these sensor-based technologies.

In addition, the ageing of rubber fenders has been studied for cell fenders which have a long operational history, (Akiyama, Shiomi, Omura, Yamamoto, & Ueda, 2017). This research has indicated that the associated fender reaction forces can be increased by up to 20 %. Currently, there remains insufficient information on the ageing effects for cone fenders.

As part of the studies of cell fenders, the increase in reaction force due to ageing was found to be smaller for used fenders returned from the operational site situations than for fenders artificially aged in laboratory conditions.

This research also proves that if fatigue from repeated compression or deep surface cracks occurs due to the ageing and degradation of the fender, the loss of performance and reaction force can be far greater. The designer should therefore be aware of the implications of the potential for changing fender reaction forces in the later stages of the service life of a fender.

Whilst the reaction force of fenders used for long periods of time increases, there are also other factors that reduce the reaction force, such as fatigue from repeated compression and surface cracking. Accounting for only ageing effects within the fender selection process, without considering these additional factors, can lead to uneconomical and conservative fenders system designs.

Table 6-5 provides recommended Partial Load Resistance factors (γ_R) of 1.30 to 1.50 to account for potential increases in reaction force e.g. by ageing which is considered sufficient for the purposes of design, so a dedicated ageing factor is not introduced by WG 211.

The designer should also provide recommendations for periodic inspection and a maximum fender operational lifetime.

6.4.18 Non-Marking Fenders

In certain circumstances, vessel owners and operators may request cylindrical or floating fenders, to be 'non-marking', (e.g. for cruise vessels). Non-marking fenders are designed to leave no visible residue on the vessel hull surface following contact, berthing and whilst moored.

Rubber materials, both natural and synthetic, and the majority of the chemicals and ingredients used in the manufacture of fenders are typically white, light in colour or colourless. These substances, once mixed and formed into a fender, would not leave any marks on the hull of the vessel when contacted. The non-marking properties of a fender are therefore not necessarily determined by the colour (e.g. white or grey). The addition of carbon black, providing a fender with its characteristic black colour, serves to enhance the physical properties of the rubber. When a fender is rubbed against the hull of the vessel, this black rubber material is rubbed off the fender due to friction and has the potential to leave black markings.

Grey or white fenders are potentially more prone to cracking and discoloration, primarily due to the comparatively weaker reinforcing fillers. In addition, the UV resistance of grey or white fenders is not as effective as that of black fenders, as the latter's UV resistance properties are provided primarily by the presence of carbon black. Further information is provided in Chapter 0.

6.4.19 Mooring Analysis

An effective mooring analysis, conducted as part of the fender system selection process, can assist in identifying optimised fender system solutions. The location and orientation of certain berths may require the designer to consider the effect of vessel motions whilst moored at the berth, and thus the so-called 'lean-on' loads during mooring. Typical examples of such conditions are:

- Berths exposed to long fetches of open water
- Long-term mooring of vessels
- Potential for strong and gusting winds
- Mooring required in a range of adverse weather conditions
- Where non-forecastable, sudden extreme weather events (e.g. tsunamis, squalls, etc.) can occur;
- Restricted operational limitations on the amount of vessel movement at the berth
- Moored vessels subjected to demanding met-ocean conditions at the berth, such as infragravity waves, preventing operational limitations being achieved
- Other specific limitations to the performance of the vessel at berth

In such cases, the design fender energy absorption capacity ($E_{f,d}$), and design reaction force ($R_{f,d}$) may, on occasion, be exceeded due to mooring conditions and the resulting lean-on loads at the berth.

The designer should be aware of the potential effects of vessel movements whilst moored at the berth, and the potential demands that this could place on the requirements of the selected fender system. The forces associated with the warping of vessels along a berth when 'leaning on' can drag against and laterally load fenders significantly. The loads imposed by a moored vessel may therefore, in certain circumstances (e.g. permanently moored and exposed FSRUs), could be more critical for fender selection and design than the berthing energy.

To account for these potentially adverse conditions, it is recommended that the designer considers undertaking mooring simulations to assess the moored vessel motions and resulting (fender) loads on the berth. Refer to section 7.3 for more information on mooring analyses.

6.4.20 Permanent Mooring

In a permanently moored vessel situation, the fender deflection and magnitude of the cyclic loading are the most important considerations when assessing the fender durability and long-term performance. In these situations, rubber fenders are subject to visco-elastic characteristics such as hysteresis loss and creep that are difficult to represent in numerical simulations.

The selection of a suitable fender depends on the mooring arrangement, location and the predominant environmental conditions (e.g. wind speed and direction, wave height and period). To adequately accommodate cyclic loading, fenders should be selected to ensure that the maximum deflection is limited so that the fenders do not buckle and/or that the fenders are not subjected to repetitive motions of the vessel at deflections which exceed the peak of the reaction force. The fender selection should also account for the potential for creep in the fender following prolonged static loads by continuous wind or currents. Refer to Chapter 0 for more information on permanent mooring of vessels.

The fatigue life of the fender should also be assessed with reference to the expected number of compression cycles and the fender durability tests. This assessment will help determine the fender design life and the recommended maintenance and inspection requirements throughout the operational life of the fender.

Pre-tensioned mooring lines and sustained wind loading (constant loads) can induce permanent deflections in a fender, which can increase over time. This type of fender deflection is known as 'creep'. Fenders should be carefully selected so that they do not buckle due to creep after prolonged static loads by continuous wind or currents. Further information on the recommended deflection limits is outlined in Chapter 7 and advice on the selection of suitable fenders for specific situations can be obtained from fender suppliers.

When selecting fenders for permanent mooring situations, consideration should be given to alternative fender rubber grades and fender heights which may be more efficient or suitable. A larger fender compression may be considered more appropriate instead of a harder grade of rubber. Pneumatic or foam fenders often provide different characteristics which may be operationally preferable.

6.5 Base Fender Performance

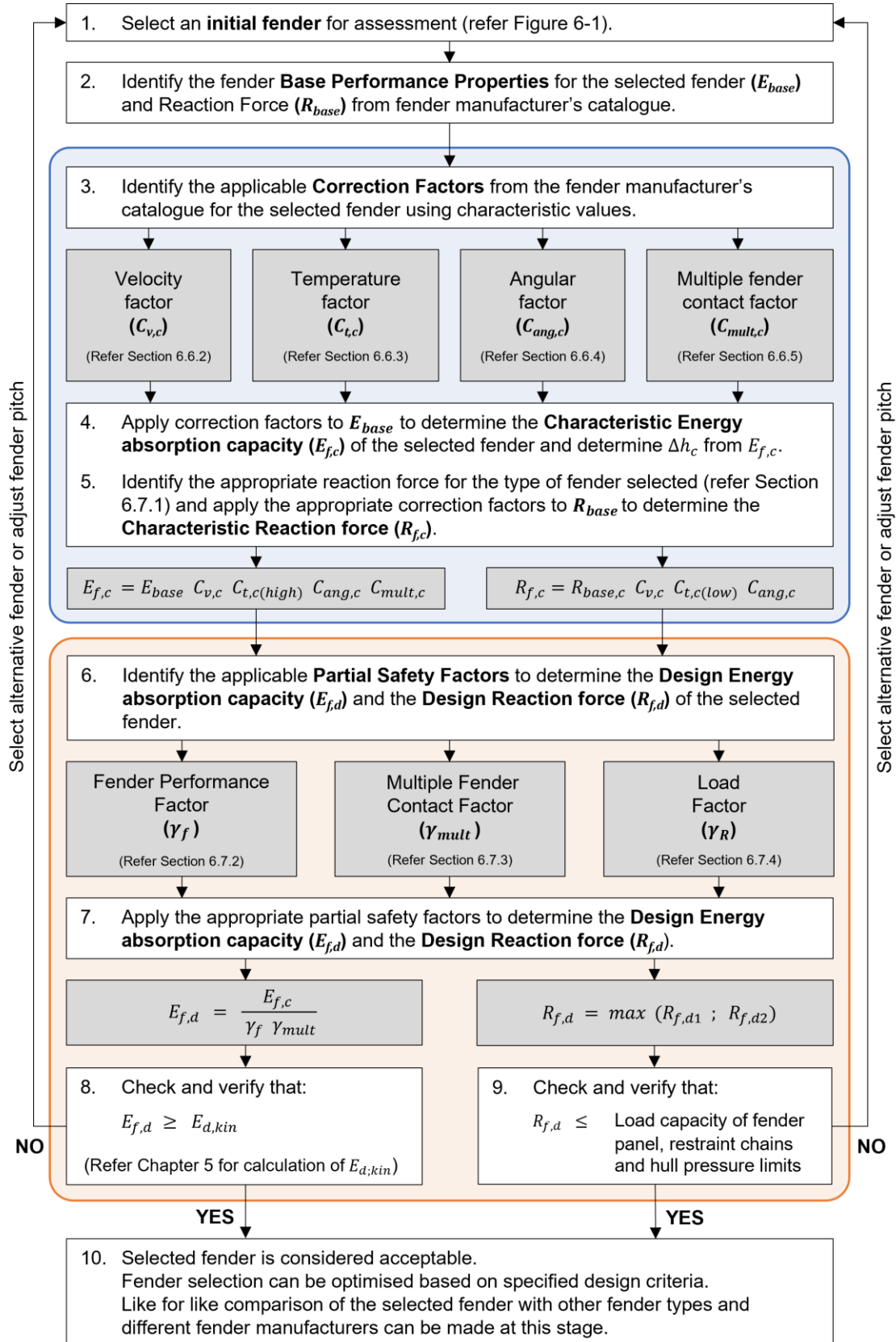


Figure 6.11: Application of correction factors and partial safety factors

To determine whether a selected fender is suitable, the designer should first determine the base fender performance for the selected fenders from the fender supplier's catalogues. Refer to Figure 6.11 for further information on how to identify the fender base performance properties, the application of the correction factors and the partial safety factors as part of the fender selection process.

The performance of a fender depends predominantly on its type and size and the material grade. Several other factors, including the amount of angular compression, ambient temperature, speed of compression and the number of fenders contacted during berthing, also influence the performance of the fender.

The base fender performance represents the mean value of fender performance and is used for testing of the fender.

To determine the base fender performance properties, fender suppliers undertake slow Constant Velocity (CV) compression testing for each fender type and rubber grade of fender. These CV fender compression tests typically determine the base energy absorption (E_{base}) and base reaction force (R_{base}) properties. Further details of CV testing for all fender types is outlined in Chapter 0.

The E_{base} and R_{base} performance properties are published by fender suppliers for use in the first stage of the fender selection process.

6.6 Correction Factors

Fender selection and the design of buckling rubber type fender systems requires the E_{base} and R_{base} to be adjusted by correction factors that are directly applicable and relate to the project specific conditions and design criteria. These correction factors consist of the following:

- Velocity factor (C_v), refer to Section 6.6.2
- Temperature factor (C_t), refer to Section 6.6.3
- Angular factor (C_{ang}), refer to Section 6.6.4
- Multiple fender contact factor (C_{mult}), refer to Section 6.6.5

Correction factors are not applicable to pneumatic fenders. Designers should seek further clarification from fender suppliers regarding the application of correction factors for cylindrical and foam fenders. If applicable, these correction factors will be provided by the fender suppliers.

The characteristic values of the vessel berthing velocity, ambient temperatures at the project site, and vessel approach angle are used to determine the correction factors. The significance of the dominant and non-dominant design variables is outlined in Chapter 5.

6.6.1 Application of Correction Factors

The correction factors are applied to the E_{base} and the R_{base} of the selected fender to determine the characteristic fender energy absorption capacity ($E_{f;c}$) and characteristic fender reaction force ($R_{f;c}$), refer also to Figure 6.11.

The following formulae are used to calculate the characteristic fender energy absorption capacity ($E_{f;c}$) and characteristic fender reaction force ($R_{f;c}$).

$$E_{f,c} = E_{base} C_{v,c} C_{t,c} C_{ang,c} C_{mult,c} \quad (6-5)$$

$$R_{f,c} = R_{base} C_{v,c} C_{t,c} C_{ang,c} \quad (6-6)$$

The methodology for the selection of R_{base} , depending on the selected fender type, is outlined in Section 6.2. It should be noted that different characteristic values for temperature are used to calculate $E_{f,c}$ and $R_{f,c}$; refer to Section 6.6.3.

6.6.2 Velocity Factor

Typically, the initial berthing velocity of vessels ranges from between 20 mm/s to 300 mm/s and decreases as the energy is absorbed by the fendering system as it deflects. Ideally, fender suppliers would test fenders starting at the initial design berthing velocity to determine the required performance of the fenders and to enable fender selection. However, in practice this is very difficult to achieve, given the size and complexity of the testing equipment required and the range of fenders to be tested. Hence in practice fender suppliers either test the fenders with a slower velocity using either a Constant Velocity (CV) or Decreasing Velocity (DV) profile.

For CV testing, fender suppliers can determine the base fender performance properties using slow compression tests on full sized fenders or on scale models at velocities of between 0.33 to 1.33 mm/s. Refer to Section 10.7.3 for further information on the test protocols for velocity factors.

To account for the difference in the velocity between the base performance tests and the in-service design berthing velocity profile, fender suppliers determine, and should publish in their catalogue, velocity factors (C_v) for a range of compression velocities using either the CV or DV method. If the supplier wishes to test the fenders using a constant velocity profile, then CV/DV factor(s) should be established to convert the results to an in-service condition. This should be published by the supplier in their catalogue.

Note, in the WG33 Report, the DV performance for a berthing velocity of 15 cm/s was known as the RPD fender performance, which is no longer applicable as the berthing velocity is now selected on a case-by-case basis.

The C_v factors depend on the type and rubber grade of the fender and are determined from the strain rate or the compression time. The designer should use the velocity factors provided in the supplier's catalogue to correct the fender's performance. Failure to apply C_v factors could lead to the wrong choice of fender and an underestimation of the forces acting on the fender system and supporting structure.

To determine the characteristic value for the velocity factor ($C_{v,c}$) the designer should first determine either the strain rate (which is a non-dimensional compression velocity) or the fender compression time (by dividing the design deflection by the average compression velocity), as identified in Chapter 5 to calculate $E_{f,c}$.

The strain rate or average compression time is then identified in the supplier's published data and used to identify the associated value of $C_{v,c}$. Refer to Section 10.7.3 for further information.

Fender suppliers may also publish $C_{v,c}$ factors for a range of different rubber compounds. A natural rubber fender is less prone to high velocity effects when compared to a synthetic rubber fender, when tested at the same compression velocity. For more details on rubber blends and compounds, refer to Chapter 0.

6.6.3 Temperature Factor

Elastic materials, including rubber, typically get softer when warm and stiffer when cold. A fender's minimum energy absorption will occur at the highest operating temperature, whereas the maximum reaction force will occur at the lowest operating temperature.

The ambient temperature to which a fender is exposed to during its design life influences the overall rigidity of the compound. This effect should therefore be taken into consideration when undertaking calculations to determine the characteristic berthing energy and characteristic reaction forces of a fender. The effect of partial or permanent submerged fenders should also be considered. Failure to do so could lead to an underestimation of the forces acting on the fender system and supporting structure.

To enable the selection of a fender that is suitable for the design temperature range, fender suppliers typically provide temperature factors (C_t), for a range of fender types and grade. The datum for C_t , (where C_t is equal to 1), is defined as 23°C. For more details on the effects of temperature on fender performance, fender testing and fender stabilisation, refer to Chapter 0.

The C_t factor is dependent upon the type and blend of rubber used in the manufacture of the fender. Therefore, C_t varies with the type and grade of rubber used in each fender and between different fender suppliers. Typically, this factor is close to 1, except for very cold regions.

Ideally, the temperature considered in the design should be as recorded from the project site where the fender is to be installed. Practically, and in the absence of such records, meteorological records in the general regional vicinity of the project site can also be used.

To determine the Characteristic Fender Energy absorption ($E_{f,c}$), designers should determine the high characteristic temperature ($T_{c,high}$) from the average monthly ambient air temperature of the hottest month ($T_{mean,hot}$) for the project site. The average monthly temperature is recommended as a characteristic value, as temperature is not the 'dominant' design variable when evaluating the energy absorption capacity of a fender system and the effect on fender energy absorption in milder climates is negligible. The designer should then select the relevant characteristic temperature factor $C_{t,c}$ from the suppliers published data.

In contrast, to determine the Characteristic Fender Reaction force ($R_{f,c}$), designers should determine the average ambient air temperature ($T_{c,low}$) of the coldest day of the year with a return period of five years, for the project site. In the absence of site-specific data, $T_{c,low}$ can be estimated using the formula given in Equation (6-7), where the average daily temperature of the coldest month ($T_{mean,cold}$) and the average daily minimum temperature of the coldest month ($T_{min,cold}$) are established from publicly available climate normal weather records:

$$T_{c,low} = T_{min,cold} - 1.65 (T_{mean,cold} - T_{min,cold}) \quad (6-7)$$

The designer should then select the relevant $C_{t,c}$ using the suppliers published data. The average daily temperature is used to determine the characteristic low temperature (compared with the monthly average used to determine the characteristic high temperature), as the effect of variations in temperature in colder climates can significantly influence the fender reaction force.

It is recommended that the fender suppliers are consulted for fenders located where extreme high and low temperatures are expected to be encountered.

6.6.4 Angular Factor

The fender compression angle is defined as the angle that the vessel's hull creates against the berthing structure, at the point of contact. This angle is associated with the horizontal angle of approach of the vessel onto the berth and a combination of both the horizontal and vertical angles of the hull profile at the bow or stern of the vessel.

Larger vessels generally approach a berth at a small angle, typically between 0 and 3 degrees to the berthing line. Ultra-large vessels tend to approach parallel to the berth, as far as practically possible. Guidance on vessel approach angles is provided in Section 5.5. If the berthing angle and fender spacing are low, such that contact with the fender is on the flat side of the hull, the hull shape can be ignored.

Where it is applicable, the hull shape should be determined from the vessel type and size. Information on typical hull flare angles and bow radius may be obtained from the port operator or vessel plans, where available. Some guidance on bow radius is also provided in WG 235 Report. Designers should be careful to adopt reasonable geometric parameters to avoid applying a low probability, design event coupling maximum velocity with an unlikely/rare hull impact geometry.

In the case of dolphins and mono-pile berths for large vessels, the effect of angular compression has paramount importance. Geometric checks are required to verify that the fender panel or the vessel does not contact the support structure during compression of the fender.

The angular factor (C_{ang}) accounts for the loss of energy absorption capability of a fender, due to the non-uniform deflection of the fender. Failure to account for this reduction in energy absorption capability could lead to an underestimation of the forces acting on the fender system and supporting structure. Once the compression angle is determined from the berthing geometry, C_{ang} can be obtained from the supplier's published data.

The angle of compression of the fender can particularly affect some fender systems due to the varying degrees of compression of different parts of the fender. Fender suppliers typically publish angular factors to allow for this effect on the fender performance. It should be noted that angular factors differ between fender suppliers even for similar types of fenders. Designers should carefully consider this when selecting and comparing fenders from different suppliers.

The energy absorption of a fender reduces with larger compression angles. Therefore C_{ang} reduces as the compression angle increases. Thus, the fender should be selected based on the angular compression case. However, the largest fender reaction forces will occur at the smallest compression angles, when the energy absorption is at a maximum and the angular factor is at or close to 1. The datum for the C_{ang} (where the horizontal and vertical C_{ang} factor is equal to 1.0), is defined as 0°, parallel to the berthing line.

Designers can use the characteristic berthing angle (as identified in Chapter 5) combined with the horizontal/vertical, bow or stern flare angles, to determine $C_{ang,c}$ from the supplier's published data along with other potential (geometric) aspects.

Whilst the angular factor varies for each fender type and compression angle, the $C_{ang,c}$ for calculating $E_{f,c}$ is not taken larger than 1.0 and is typically 0.7 to 0.8 for compression angles of up to 20°. Cone fenders can have a C_{ang} factor greater than 1.0 for compression angles between 0° and 10°, however this is often ignored for safety/conservative fender selection reasons. Some fender types have a $C_{ang,c}$ of as low as 0.5 for compression angles of 20°.

For calculating $R_{f,c}$, the angular factor is typically taken as 1.0. However, a $C_{ang,c}$ of 0.8 or 0.9 can be applied for certain fender types for compression angles of up to 20°.

The designer selects the compression angle from the characteristic berthing angle or hull profile angle for the calculation of $C_{ang,c}$ for axis-symmetrical fenders. Non-axis-symmetrical fenders and their proposed orientation will require the calculation of $C_{ang,c}$ in both orientations.

Designers should also consider the influence of fender panels and restraint chains on the design of the fender system as they will perform differently to a single fender, e.g. tension loads during angular compression.

6.6.5 Multiple Fender Contact Factor

A berthing approach that is almost parallel to the berthing line, (i.e. a characteristic berthing angle of 2° or less), will typically result in the vessel contacting multiple fenders. Refer to Chapter 5 for further information related to the characteristic berthing angle of the vessel. Refer also to Appendix C for geometric formulae for the compression of multiple fenders.

The characteristic berthing angle of the vessel determines the number of fender systems that contribute to absorbing the berthing energy. The proportion of the energy absorbed by each contacted fender depends on the amount that each fender deflects. The cumulative energy absorption capacity of the multiple contacted fender systems is therefore greater than that for a single fender system contact. Contact with multiple fender systems at smaller berthing angles is also influenced by the fender pitch, refer also to Section 6.4.

For larger berthing angles (i.e. with a characteristic berthing angle of greater than 2°), the number of contacted fenders is primarily influenced by the geometry of the vessel's bow, refer to Section 6.4 for further information.

To account for the influence of contacting several fenders in the design and selection of the fender, the multiple fender contact correction factor (C_{mult}) has been introduced.

Depending on the length of the design vessel, and the number of fenders contacted (based on the characteristic berthing angle), the characteristic value of $C_{mult,c}$ can be in the region of 1.5 to 3.0. This corresponding increase in the total energy absorption capacity enables the designer to potentially select a smaller fender, generating lower reactions forces to be accommodated by the support structure.

Consequently, multiple compressed fenders can also simultaneously create a much greater total reaction force on the structure. Whether this is a critical load case should also be considered by the designer.

An iterative approach is always required when considering multiple fender contacts to identify the optimum fender solution. Variations in fender pitch and the fender size will change the overall energy absorption characteristics of the berth. The parameters of several different design vessels may also require a comparative geometrical assessment to determine the critical design case.

The calculation of $C_{mult,c}$ can also be undertaken using simulations to replicate the berthing approach. This enables the efficient assessment of numerous variations in vessel dimensions, berthing approaches and berth configurations.

The characteristic multiple fender contact correction factor ($C_{mult,c}$) can be calculated by the designer, using the following methodology.

- Using the dimensions of the design vessel and the proposed fender arrangement, carry out a geometric analysis of the vessel alongside the berth, at the characteristic berthing angle; refer to Figure 6.12.

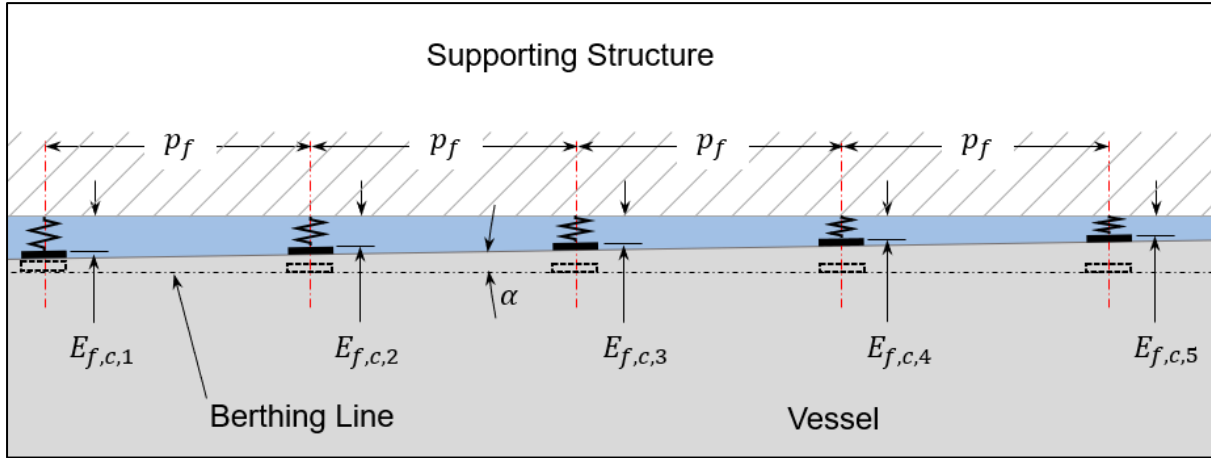


Figure 6.12: Calculation of $C_{mult,c}$ for multiple fender contact

- The geometric analysis should identify the number of contacted fenders based on the maximum design deflection of the first fender at the initial point of contact, assuming this first fender takes its maximum permissible load (i.e. single fender contact).
- For each contacted fender, calculate the fender deflection using the fender pitch and characteristic berthing angle.
- Using the base energy absorption of a single fender (E_{base}) and the generic performance curves for the selected fender type (as published by the fender suppliers), identify the energy absorbed for the calculated deflection of each contacted fender.
- Calculate $C_{mult,c}$ using the following formula:

$$C_{mult,c} = \frac{E_{f,c,1} + E_{f,c,2} + E_{f,c,3} + \dots + E_{f,c,n}}{E_{base}} \quad (6-8)$$

Where:

$E_{f,c,n}$ Characteristic berthing energy absorbed by n^{th} fender of the partially compressed fenders [kJm]

6.7 Determining Fender Design Performance

The fender design performance is derived by applying partial resistance factors of safety to the characteristic performance values, determined in Section 6.6. The design fender performance largely depends on the project reliability requirements for the fender system.

To determine the fender design performance, partial resistance factors related to the energy absorption of a single fender (γ_f) (see Section 6.7.2) (which includes the variable effects of the manufacturing process), the effect of a single or multiple fender contact (γ_{mult}) (see Section 6.7.3) and a partial load factor (γ_R) (see Section 6.7.4

and their application to the characteristic energy absorption ($E_{f,c}$) and characteristic reaction force ($R_{f,c}$) is outlined in the following sections and illustrated in Figure 6.11.

6.7.1 Application of Partial Resistance Factors

The design energy absorption capacity ($E_{f,d}$) is determined by dividing the characteristic energy absorption capacity ($E_{f,c}$) by the partial material factor (γ_m). γ_m can be calculated from the partial resistance single fender performance factor (γ_f) and the partial resistance multiple fender contact factor (γ_{mult}), refer to Equation (6-9). The product of these partial factors is applied to the $E_{f,c}$ to determine $E_{f,d}$, as presented in Equation (6-10). Recommended factors for γ_f and γ_{mult} are included in Sections 6.7.2 and 0 respectively.

$$\gamma_m = \gamma_f \gamma_{mult} \quad (6-9)$$

$$E_{f,d} = \frac{E_{f,c}}{\gamma_m} = \frac{E_{f,c}}{\gamma_f \gamma_{mult}} \quad (6-10)$$

The design reaction force ($R_{f,d}$) is used to design the other components of the fender system and to conduct the hull pressure check.

To determine $R_{f,d}$, the performance curves for the selected fender type(s) are used to identify the applicable values of the reaction force ($R_{base,c}$ or $R_{base,max}$), as the value of $R_{f,d}$ may be different for linear and non-linear fender compression. Refer to Figure 6.13 and Figure 6.14, where $R_{f,d1}$ applies to R_{base} and $R_{f,d2}$ applies to $R_{base,max}$. The maximum value of either $R_{f,d1}$ and $R_{f,d2}$ is used as the design reaction force, $R_{f,d}$.

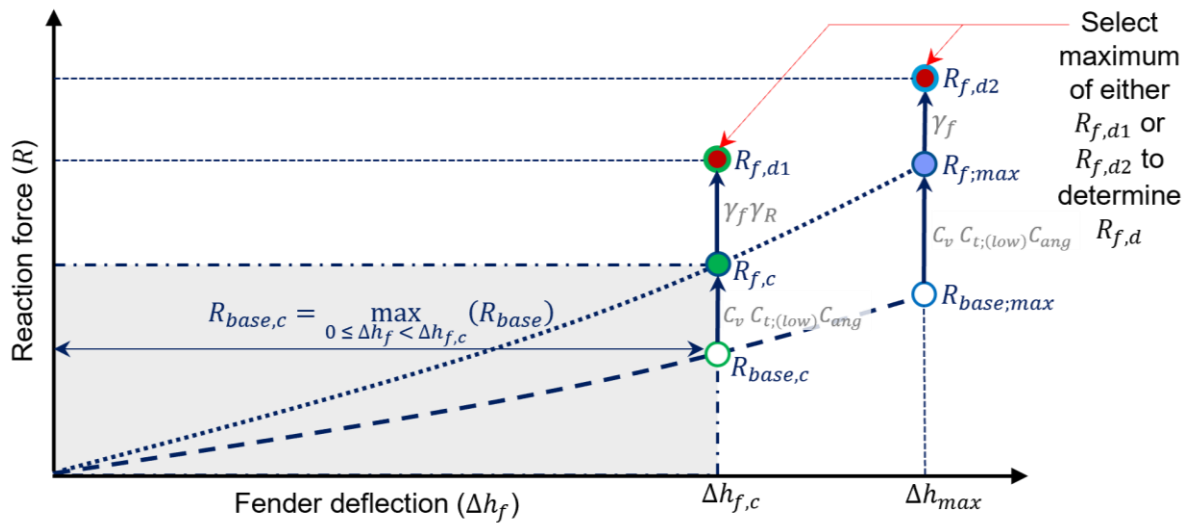


Figure 6.13: Identification of $R_{f,c}$ and calculation of $R_{f,d}$ for linear force-deflection curves

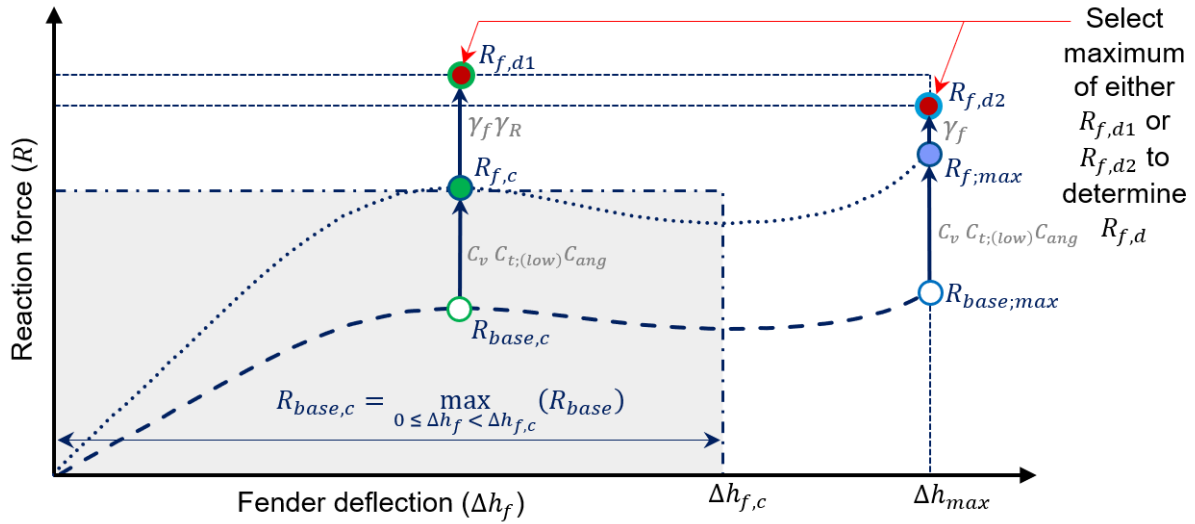


Figure 6.14: Identification of $R_{f,c}$ and calculation of $R_{f,d}$ for non-linear force-deflection curves

To determine $R_{f,d1}$, the deflection of the selected fender ($\Delta h_{f,c}$) at the characteristic energy absorption capacity ($E_{f,c}$) must first be determined. The maximum value of $R_{base,c}$ is then identified within the characteristic fender deflection range, refer to Equation (6-11), Figure 6.13 and Figure 6.14.

$$R_{base,c} = \max_{0 \leq \Delta h_f < \Delta h_{f,c}} (R_{base}) \quad (6-11)$$

For linear fenders, $R_{base,max}$ is defined as the maximum reaction force generated by the fender at the fender's maximum compression.

The applicable values of $R_{base,c}$ and $R_{base,max}$ should then be multiplied by the appropriate correction factor values to determine $R_{f,c}$ and $R_{f,max}$, refer to Equations (6-12) and (6-13).

$$R_{f,c} = R_{base,c} C_v C_{t,low} C_{ang} \quad (6-12)$$

$$R_{f,max} = R_{base,max} C_v C_{t,low} C_{ang} \quad (6-13)$$

Subsequently, to determine $R_{f,d1}$, $R_{f,c}$ is multiplied by γ_f and γ_R , refer to Equation (6-14). To determine $R_{f,d2}$, $R_{f,max}$ is multiplied by only γ_f , refer to Equation (6-15). Recommended partial factors for γ_f and γ_R are included in Sections 6.7.2 and 0.

$$R_{f,d1} = R_{f,c} \gamma_f \gamma_R \quad (6-14)$$

$$R_{f,d2} = R_{f,max} \gamma_f \quad (6-15)$$

For berthing events that involve multiple fender contacts, the cumulative sum of the fender reaction forces acting on the supporting structure may be greater than the design reaction force $R_{f,d}$ for one or two fender systems. This should be carefully considered in the design of a fender supporting structure with multiple fender systems.

As the characteristic energy absorption and reaction force of a fender represents a conservative estimate of the mean fender performance properties, the partial resistance factors are applied to determine the design fender performance. These partial resistance

factors account for the rarest, extreme design events that may occur based on the assessed consequence class, as outlined in Chapter 0.

Uncontrolled, accidental berthing events or collisions are not accounted for within the fender design performance properties, as these are extreme design conditions and are defined as accidental limit state events. The requirement to design for accidental berthing events is a unique client requirement and should be based on a residual assessment of the capacity of the selected fender system, with all partial factors of safety set to 1.0.

6.7.1.1 Berthing Energy Fender Verification

To verify that the selected fender is suitable, in conjunction with the associated fender selection considerations, the designer should check that the fender has a design energy absorption capacity ($E_{f,d}$) greater than or equal to the vessel's Design Kinetic Energy ($E_{k,d}$). Refer also step 8 of Figure 6.11.

$$E_{f,d} \geq E_{k,d} \quad (6-16)$$

6.7.1.2 Fender Component Design

The design reaction force ($R_{f,d}$) of the selected fender, (as determined in Equation (6-14) and Equation (6-15)) is used to determine the structural design of the fender panel, chains, and components of the fender system.

$$R_{f,d} \leq \text{Structural capacity of the fender panel, chains, and components.} \quad (6-17)$$

6.7.1.3 Hull Structure Limit Check

The design reaction force ($R_{f,d}$) of the selected fender and the fender panel contact area is also used to determine the applicable load acting on the hull structure of the berthing vessel and to verify that the allowable hull pressure limit is not exceeded.

$$R_{f,d} \leq \text{Load capacity of vessel hull structure, e.g. the hull plating, stiffeners, and web frames (refer to section 6.8).} \quad (6-18)$$

6.7.1.4 Supporting Structure Design

The design of the supporting structure, to which the fender system(s) are to be attached, should consider the characteristic berthing impact force (F_c), which is equal to the characteristic reaction force ($R_{f,c}$).

As the fender induced berthing impact force is typically considered to be a variable load, national standards usually require the application of a partial load factor of between 1.3 and 1.7 to the characteristic berthing impact force (F_c). F_c should be modified by the applicable partial load factor from the relevant national code or annex to determine the design berthing impact load (F_d). This is the design load to be resisted by the supporting structure, refer to Equation (6-19).

F_c is generally defined as a live load having a return period of 50 years. For further information on return periods, refer to Chapter 0. The characteristic reaction force ($R_{f,c}$), provides a conservative estimate of F_c and can be used in the design of the supporting structure to determine the associated design berthing impact force (F_d).

$$F_d = F_c \gamma_Q \varphi_Q = R_{f,c} \gamma_Q \varphi_Q \leq \text{Structural capacity of support structure.} \quad (6-19)$$

Where:

γ_Q : Partial load factor in accordance with international or national code, standard or annex

φ_Q : Combination factor in accordance with international or national code, standard or annex

On some occasions, the calculation of F_d , may be considered conservative (i.e. if an oversized fender has been selected). As many national codes consider $R_{f,c}$ to represent a load with a return period of 50 years, the designer may decide to conduct a statistical analysis, accounting for all the applicable variables, to determine the maximum characteristic berthing impact force.

The designer should be aware that in certain circumstances, the fender reaction force from the berthing contact is the governing load for the design of the supporting structure, e.g. a berthing dolphin equipped with a buckling-type fender system.

For a supporting structure such as a suspended deck or quay wall equipped with a fender system consisting of multiple fenders, the fender reaction forces are generally not considered to be the dominant design loads. A load combination factor, determined by national design codes, can often be adopted.

The type of fender selected influences the frequency of the occurrence of the maximum reaction force and also influences the distribution of the berthing impact force into the supporting structure. For berthing events that involve multiple fender contacts simultaneously, the sum of the fender reaction forces acting on the supporting structure needs to be taken into consideration in the design.

6.7.2 Fender Performance Partial Resistance Factor

Differences in the variety of the components and processes that are part of the manufacturing of fenders can lead to variations in the properties of a fender. Such components include:

- Raw materials and rubber compounds
- Manufacturing process of the rubber or fender element
- Moulds
- Curing process
- Storage temperature and humidity
- Testing equipment

Partial resistance factors (γ_f), related to the fender performance are included in Table 6-3. This factor replaces the manufacturing tolerance factor in PIANC WG 33 (2002) that was applied to take account of variances in fender production. The γ_f factors are based on the mean performance of the fender, as typically published by the fender supplier.

If the minimum or lower bound performance parameters of a fender are provided by the supplier, and are proposed for use by the designer to determine the fender design performance γ_f , independent third-party testing of the fenders would need to demonstrate that the actual achieved manufacturing tolerances are consistently lower than those tolerances presented in Table 6-3.

It should be noted that the factors presented in Table 6-3 are only valid when the fender production tolerances are in line with the recommended values stated by the fender supplier. Also, this partial factor does not prevent fender failure caused by 'low-quality' fenders or other production issues. Such failures should be prevented by performing adequate quality control procedures and fender testing. Refer to fender testing in Chapter 0.

Performance tolerance	γ_f factor for all consequence classes	Typical example types of fender system
+/- 10%	1.10	Cone, cell, arch, element and cylindrical (wrapped) fenders
+/- 15%	1.15	Foam fenders
+/- 20%	1.20	Cylindrical (extruded), extruded, composite and shear fenders, wheel and roller fenders
Reference ISO17357-1	1.00	Pneumatic fenders ^{a)}
^{a)} $\gamma_f = 1.0$ for the calculation of the Design Berthing Energy according to ISO 17357-1. Pneumatic fender performance is given as a guaranteed value, hence no material factor needed.		

Table 6-3: Partial resistance factor γ_f related to the performance of a single fender

6.7.3 Multiple Fender Contact Partial Resistance Factor

For scenarios involving multiple fender system contacts, the energy absorption capacity is affected by variations in the berthing angle of the approaching vessel, as noted in Section 6.6.5. A particularly low characteristic berthing angle (α_c of 2° or less) may result in a high value of $C_{mult,c}$, increasing the overall berthing energy absorption capacity of the berth.

To manage uncertainties arising from small berthing angles, based on local information, coupled with multiple fender system contacts, a partial resistance factor for multiple fender contacts (γ_{mult}) is introduced to ensure compliance with reliability requirements and to prevent the over-estimation of the capacity of the selected fender system. The recommended γ_{mult} factors for single and multiple fender contact are presented in Table 6-4. The principles of multiple fender contact are also outlined in Section 6.4.5.

Fender contact	Berthing angle ^{a)}	Partial Safety Factor γ_{mult} for each Consequence Class			
		A	B	C	D
Single	All.	1.0			
Multiple	$\alpha_c \geq 2^\circ$	1.0			
Multiple	$\alpha_c < 2^\circ$	1.00 ^{b)}	1.10 ^{b)}	1.15 ^{b)}	1.20 ^{b)}
^{a)} Due to a relatively large characteristic berthing angle, the bow radius dominates the respective deflection of the fenders. For a relatively low characteristic berthing angle, the parallel side body length of the vessel predominantly influences the number of fenders that contribute to absorbing the kinetic energy of the berthing vessel.					
^{b)} These factors are derived based on the characteristic berthing velocity having a probability of exceedance of 0.02 % and a berthing angle having a probability of exceedance of 5 % (Refer to Chapter 5).					

Table 6-4: Partial resistance factor γ_{mult} related to single and multiple fender contacts

6.7.4 Load Partial Resistance Factor

For some types of fender, a comparatively small fender deflection results in the maximum reaction force of the fender (e.g. cell or cone fender). For other fender types, this occurs only at the maximum deflection of the fender (e.g. cylindrical, foam or pneumatic fenders).

When a fender exceeds its maximum design deflection, the associated reaction force typically increases rapidly. In addition, some fender types demonstrate linear hardening effects over time, whereas others indicate nonlinear softening. This can result in significantly greater forces being imparted onto the supporting structure and being transferred to the vessel hull structure.

The type of fender selected therefore influences the frequency of the occurrence of the maximum reaction force. The designer is recommended to investigate the potential frequency of the occurrence of the characteristic fender reaction force ($R_{f,c}$), to ensure that the appropriate reaction force is applied to the design of the fender system, refer to Chapter 2 for indicative performance graphs for different types of fenders.

The use of a Partial Load Resistance factor (γ_R) should not be considered when assessing fender induced berthing impact loads acting on the support structures, since recommendations for load factors and combination factors are generally prescribed within national codes and standards; refer to section 6.7.1.4 for further information.

γ_R is applied to the characteristic reaction force ($R_{f,c}$) to determine the design reaction force ($R_{f,d}$) for component design. The recommended partial load factors γ_R are presented in Table 6-5.

Partial Load factor	Factor γ_R for each Consequence Class			
	A	B	C	D
γ_R	1.30	1.40	1.45	1.50

Table 6-5: Partial factor γ_R for reaction forces to be used in fender system design

The $R_{f,d}$ of the selected fender is used to design all the components of the fender systems (e.g. the fender panel, restraint chains, anchor bolts, etc.). Refer to Section 8.2 for recommendations on the applicable factors of safety for the design of the restraint chains and determining the loads on the brackets. $R_{f,d}$ is also used to assess the load distribution on the vessel hull structure, refer to Section 6.8. $R_{f,d}$ is not to be used for the design of the support structure. F_d is to be used for the design of the support structure, see Equation ((6-19).

6.8 Hull Pressure

While absorbing the berthing energy of a vessel, the fender will exert a reaction force on both the vessel and supporting structure. Under normal berthing conditions, no plastic deformation of the vessel's hull structure is accepted.

Coordination between fender designers and naval architects remains limited. Forces induced by wave impacts during vessel voyages continue to be the governing criteria for hull design. The vessel Class provision for minimum hull plate thickness in the fender contact zone is also limited. Typically, vessel designers and builders will only consider additional reinforcement to the fender contact area if required by the vessel owners and/or to fulfil specific port requirements.

With vessel sizes continuing to increase, hull plate thicknesses are relatively reduced. The spacing between web frames is also increasing, driven primarily by the requirement to reduce

the costs of vessel construction. Consequently, it appears that in some cases the permissible loads on the vessel hull structure stated by vessel owners are decreasing, although factual evidence to confirm this is unavailable.

Where possible, and depending on the range of applicable design vessels (i.e. a purpose built new build vessel), designers are recommended to seek further advice, information and clarification on the maximum permissible distributed and point loads from the vessel owners, designers and/or builders in relation to the capacity of the vessel hull structure.

6.8.1 Factors Affecting Induced Hull Pressure

Several factors influence the magnitude of the hull pressure induced by fenders on a vessel hull. These are outlined as follows and the designer should be aware of the significance of each item:

1. Vessel approach velocity
2. Vessel approach angle
3. Vessel hull profile (e.g. bow flare)
4. Fender pitch
5. Fender contact area
6. Multiple simultaneous fender contact
7. Fender panel facing shape and orientation

Arguably, the most significant of these is the vessel approach velocity. There may also be other port or vessel specific influencing factors and the designer should consider their significance when establishing the fender design criteria and assessing the operational parameters of the port.

6.8.2 Hull Structure

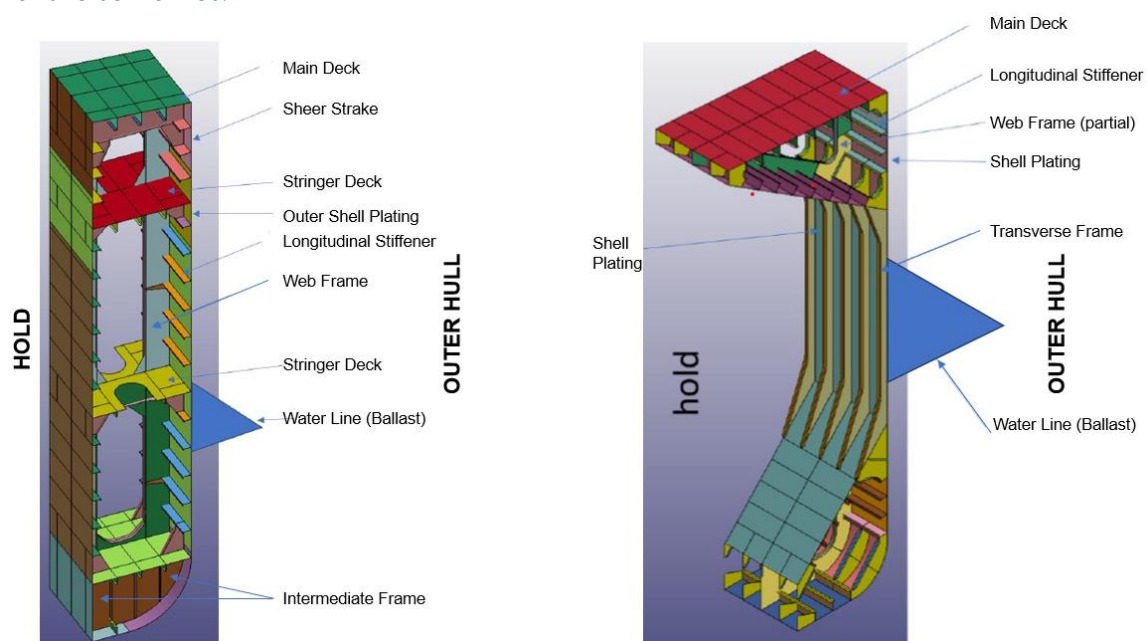
Vessel hull structures can typically be constructed from a wide variety of materials, including steel, aluminium, wood and composite materials. Most vessels are constructed using steel and this document provides guidance on fenders only for steel hulls. Typical examples of vessel hull structures are outlined in Figure 6.15.

The vessel's hull structure is generally comprised of four components. A typical steel hull consists of steel plating welded to longitudinal (horizontal) stiffeners that are welded to the web frames connected by one or more decks. These stiffeners are designed to provide the necessary hull strength to resist the loads, forces, bending moments and stresses imposed by cargo and environmental loads.

Certain parts of the vessel hull may be locally strengthened to accommodate the additional forces of tugs pushing against the side-shell plating. Alternatively, as is more common, the positions along the hull in line with the transverse bulkheads (which are more resistant to transverse loading in a small area) may be marked for tug contact points.

The stiffeners are generally of sufficient size and strength to accommodate the loads from fenders and span between the transverse frames. By implication of the tug push markers, the shell plating between them is likely to tolerate less concentrated pressures than can be applied from a tug. On vessels with large displacements this could occur on berthing impact by contacting a single or small number of fenders. General descriptions and typical sizes of vessel hull components are summarised below.

1. **Side-shell plating.** Typical thickness ranges between 15 mm and 20 mm. The thickness is very dependent on the overall size of the vessel. Smaller general cargo vessels can have side-shell plating of about 8 mm to 10 mm. On container vessels, the sheer strake (uppermost section of side-shell plating) is typically significantly thicker. For very large vessels, side-shell plating thickness can be greater than 15mm. Most ocean-going vessels are double skinned (incorporating an inner hull) with the exception of the majority of larger bulk carriers, which are generally single skinned in way of the transversely framed hold sides.
2. **Longitudinal stiffeners.** Typically spaced at between 0.8 m and 0.9 m and dependent on the overall size of the vessel. Smaller vessels will have a smaller spacing, within the limits of what can be practically welded and vice versa. Where the vessel is longitudinally stiffened, web frames are typically spaced approximately 1.8 m to 4.0 m apart. This geometry is also dependent on overall vessel size, however the very large container vessels can have even larger spacings. For container vessels, the frame spacing can also vary along the length of the vessel.
3. **Decks.** Besides the main deck, one or more decks can be found in the double hull. The most well-known is the engineers passageway as found on all container vessels. These decks provide, together with the frames, a strong boundary of the grillage of stiffeners that supports the hull plates. Decks normally also support the belting of a vessel if present.
4. **Transverse frames.** Typically spaced at between 0.6 m to 1.0 m for the sides of holds in bulk carriers. However, these vessels are typically of a composite construction with longitudinal framing in the topside and double bottom structures where the convention for longitudinally framed vessels is followed, with web frames positioned every three or four transverse frames.



(a) Typical container, tanker and general cargo vessel side structures. Note some older and smaller general cargo vessels have only single skin side structures.

b) Typical dry bulk cargo vessel (Handymax/Panamax sizes) side structure also applies to large cape size bulk carriers.

Figure 6.15: Typical vessel hull side structures (TNO, 2019)

The dimensions of all four components vary with the size, type and age of vessel, classification society rules, designer and the vessel builder.

The names of typical structural members used within vessel hull design are included in Figure 6.15 which shows transverse slices of a typical vessel side structure. These are typically repeated longitudinally along the vessel, usually with a fixed web frame spacing. A web frame spacing is subdivided into intermediate frame spacings as indicated.

Except for bulk cargo vessels (Handymax/ Panamax and Capesize), a vessel's side structure general arrangement looks very similar, albeit that the structural dimensions (scantlings) vary significantly.

With advances in vessel design, the following tendencies have been identified:

1. The use of steel with higher yield strength. Whilst this increases the hull plating's resistance to fracture, the thickness tends to be reduced and the plating is more susceptible to deformation. On larger hulls, such as Panamax and particularly Capesize bulk carriers, deformation due to impacts can require Class supervised repair necessitating an extended port stay.
2. Increasing the distance between transverse frames, (e.g. 3 m for early generation container vessels increasing to up to 6 m for later generation vessels).
3. Berthing energies are increasing (due to increases in vessel size).

Car carriers typically have single skin side shell structures and vehicle decks that span the full breadth and length of the vessel. There may be some ballast tanks at the outboard side of the lower decks which have similar looking construction to a typical container/ tanker, however they do not extend all the way to the upper deck.

Similarly, RoRo vessels tend to have a single skin side shell above the main cargo/ vehicle deck. This will typically be longitudinally stiffened, with transverse web frames every three to four frame spaces, although it is possible that they can be transversely stiffened, similar to the side shell of a bulk carrier. Depending on the arrangement and if the RoRo vessels have a lower deck below the main deck, then a similar side tank arrangement, as indicated in Figure 6.15, may be seen up to the level of the main deck level.

Very Large Ore Carriers (VLOC) also differ as they have very large ballast tanks situated either side of the central holds. Side-shell plating to accommodate fender contacts will be constructed similarly to tankers with longitudinal stiffening and web frames.

6.8.3 Recent Hull Pressure Research

Historically, fenders were dimensioned using a maximum allowable hull pressure to spread the design reaction force in a safe way over the hull of the vessel. Theoretically this means that cylindrical fenders cannot be used as these typically generate a high hull pressure. However, based on recent observations and experience it is acknowledged that vessels of all sizes can and do berth safely on cylindrical fenders. The Port of Rotterdam has undertaken research that demonstrates the safe use of cylindrical fenders for container vessels (Lloyd's, 1989), oil and LNG vessels (Broos, Rhijnsburger, Vredeveltdt, & Hoebee, 2018) and on all general classes of vessels (TNO, 2019).

As part of the work undertaken by WG 211, additional research was undertaken (Berendsen E. A., 2022). The FEM analyses of the fender (allowing for buckling with various panel sizes) and hull structure interaction indicated that when using large panels or contact areas, the load capacity of the vessel hull can be exceeded (refer to Figure 6.16). In addition, very small fenders (e.g. cylindrical and arch fenders) have hull pressures significantly greater than the recommended hull pressures stated in WG 33.

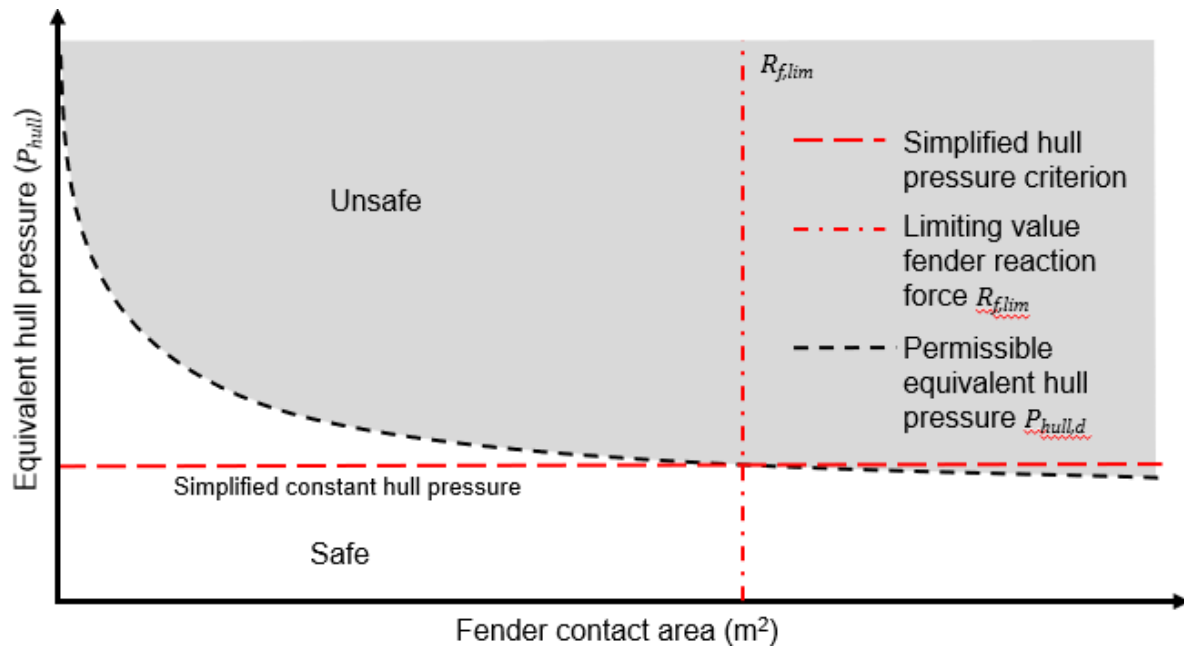


Figure 6.16: Equivalent hull pressure limits (Berendsen et al., 2024)

For fenders that utilise fender panels, increasing the size of the fender panel makes it possible to increase the reaction force from the fender onto the vessel hull. However, it was identified that there is a maximum point that represents the upper limit of the capacity of the hull. In such cases, the stress concentration in either the web frame or deck becomes critical. Therefore, simply increasing the contact area of the fender panel does not necessarily ensure that these larger reaction forces can be accommodated by the vessel hull, as can be seen in

Figure 6.17.

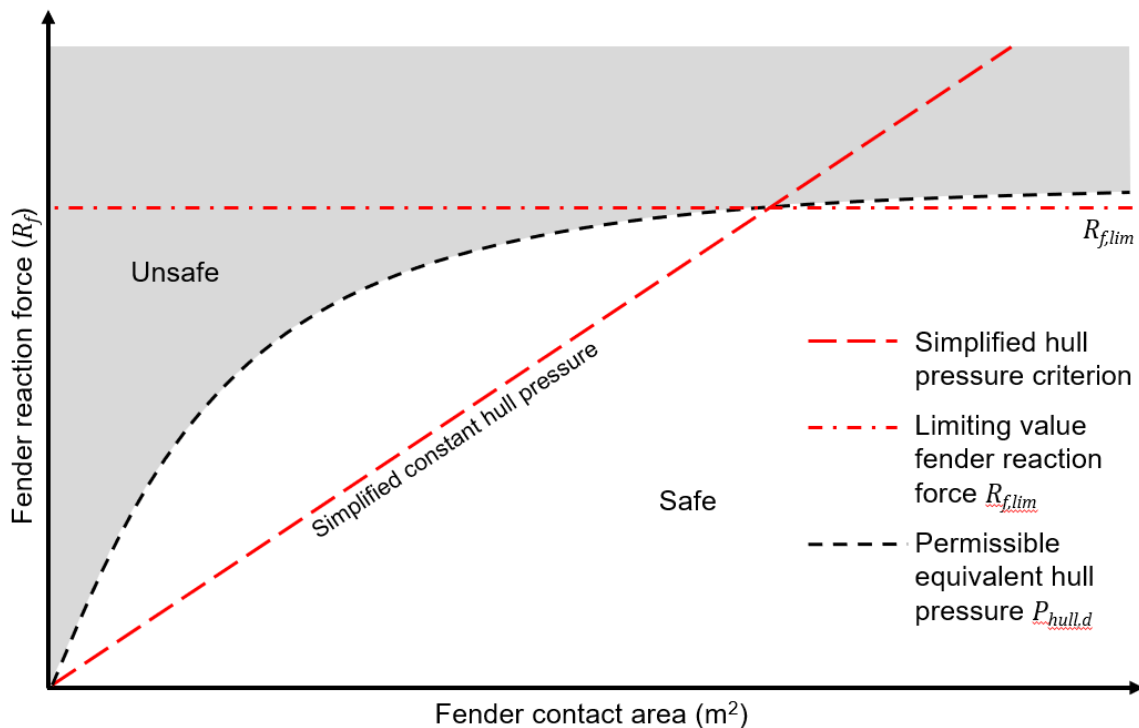


Figure 6.17: Maximum fender reaction force (Berendsen et al., 2024)

It was concluded by (Berendsen, Roubos, Williams, Walters, & Broos, 2024) that:

- The structural layout of the vessel, dimensions of the fender panel and location of the berthing impact largely influence the stress distribution of fender impact in the vessel's hull. The fender impact resulting in the critical stress is defined as the onset of plasticity in the structural members, such as the hull plates, web frames and stiffeners.
- When a fender panel subject to a berthing impact load activates a web frame or deck structure, the critical stress is reached by applying larger fender impact loads.
- For larger vessels, typically wide fender panels are considered to be more efficient when compared to taller fender panels. However, in tidal ports fender panels are likely to already be comparatively tall (to accommodate the variance in water levels) and it may therefore be more efficient to engage with a vessel's deck structure.
- Design codes used to assess the pressure on the parallel hull of the vessel assume a constant hull pressure criterion. The results of this study showed that constant hull pressure can over-estimate the structural capacities of the vessel hull. Consequently, it is recommended that a limiting value for the total fender-induced reaction force is included for the design of fender systems.

The berthing impact force concentrates in large structural members, i.e. web frames, in contact between large fender panels and vessels during berthing. Increasing the fender panel size beyond these structural elements does not generate additional capacity to withstand berthing loads. This results in the limiting value fender reaction force $R_{f,lim}$.

The prevailing failure mode largely depends on the dimensions of the fender panel. For example, relatively wide fender panels that activate a web frame, or high panels that activate a deck, results in critical stress concentrations within the web frames. For small and narrow panels, the plate and stiffener induced failure appears to be the governing failure mode, whereas high and slim panels result in the buckling of the vessel hull stiffeners.

6.8.4 Fender Reaction Force Distribution

Vessels can resist forces acting upon on their hull up to a defined limit. It is therefore important to define the correct fender panel size or fender contact area to ensure that the imposed force and resulting hull pressure is less than the permissible limits.

For buckling fenders, typically fitted with fender panels, the fender panel size is defined according to the vessel hull maximum structural capacity, the selected fender reaction force, the vessel freeboard and tidal range. These criteria are used to ensure that the contact area of the panel, excluding chamfers, distributes the hull pressure below the allowable limits.

Where fenders are located adjacent to each other (e.g. a vertical or horizontal separation that causes each fender to act in combination with the adjacent fender) and are considered to share the absorption of the vessel berthing energy, the designer may need to consider the forces acting on the hull from these fenders as a group to ensure that the maximum equivalent permissible hull pressure or maximum allowable fender reaction force capacity is not exceeded.

With reference to recent investigations (Berendsen, Roubos, Williams, Walters, & Broos, 2024), contact pressures between the fender panel and vessel are not uniformly distributed. Elastic deformation of the vessel hull will result in comparatively small, direct contact areas close to the edges of the fender panel with the vessel hull, often referred to as 'line loads'. During the initial contact with the fender panel, the hull pressure may be close to a uniform distribution. However, when the fender is fully compressed, the contact pressure at the edges of the fender panel is much higher than that at the middle of the fender panel if the panel is sufficiently stiff.

When the fender panel contact area is small, the generated hull pressures can result in the maximum stress in the hull plate being reached. However, as the vessel hull plate redistributes the fender panel forces into the stiffeners and web frames, these loads are generally not critical for small fender contact areas. For larger fender panel contact areas, the fender forces are distributed over a great number of stiffeners and/or web frames with the vessel hull structure.

For non-buckling fenders (e.g. pneumatic, foam or cylindrical fenders), the contact area of the fender on the vessel hull increases as the fender is compressed and does not initiate local force concentrations on the hull. For each stated energy absorption capacity, the hull pressure of the selected fender is provided by the fender supplier and should be checked to ensure that the hull pressure is below the allowable limits.

6.8.5 Hull Pressure for Fender Panels

The average hull pressure ($P_{hull,av}$) for a fender panel is calculated as follows:

$$P_{hull,av} = \frac{R_{f,d}}{A_c} \quad (6-20)$$

Where:

- $R_{f,d}$: Design fender reaction force (kN). Refer to Section 6.7.1.3.
- A_c : Area of contact, excluding chamfers (m²)

The Contact Area (A_c) of the flat panel which is in contact with vessel hull is calculated as the product of the width of the panel, excluding lateral chamfers (w_{pan}) and height of the panel (h_{pan}), excluding top and bottom chamfers. Refer to Figure 6.18 and Equation (6-21).

The calculation of $P_{hull,av}$ assumes that the forces in the vessel hull are fully distributed and is based on a non-linear empirical analysis.

As it is rare for the whole surface of the fender panel to be in uniform contact with the surface of the vessel hull, (as there is not always a flat hull surface for the fender panels contact), the calculation of $P_{hull,av}$ is limited to certain situations. Therefore, the calculation of $P_{hull,av}$ is an estimate of the hull pressure based on the true distribution and the developed contact area, which is not linear.

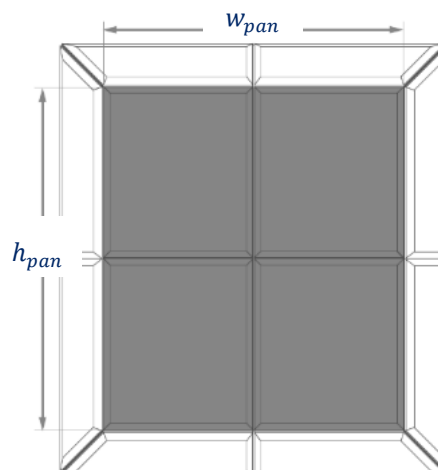


Figure 6.18: Contact area of a flat fender panel

$$A_c = w_{pan} \times h_{pan} \quad (6-21)$$

Calculated hull pressure values should be based on the worst case of full or partial contact with the fender face allowing for tides, loading conditions and vessel freeboard when berthing.

6.8.6 Hull Pressure for Cylindrical Fenders

Cylindrical fenders rarely induce their maximum reaction force ((Lloyd's, 1989), (Lloyd's, 1993), (Sijberden, Smedes, & de Gijt , 1996), (TNO, 2019); (Berendsen E. A., 2022) due to the linear force-deflection curve. Refer to Chapter 2 for more information.

In general, the maximum allowable pressure generated by cylindrical fenders should be limited to 500 kN/m². The maximum outer diameter is recommended to be 1,400 mm with a maximum length of 2,000 mm. The outer/inner diameter ratio is generally equal to 1.75. Multiple fenders (mounted in the horizontal and/or vertical plane) may be required to absorb the berthing energy of the design vessel and the cumulative reaction forces considered.

Larger or stiffer cylindrical fenders can also be used, however specialist advice from a naval architect regarding the vessel hull capacity is recommended.

6.8.7 Hull Pressure for Foam and Pneumatic Fenders

Foam and pneumatic fenders tend to have much larger diameters and lengths compared to rubber cylindrical fenders. Foam fenders can be assessed in a similar way to cylindrical fenders, however there are important differences to be considered due to the larger length in contact with the vessel hull structure. It is therefore important to limit the maximum reaction force on the vessel structure.

The general recommendation is to limit the hull pressure imposed by foam and pneumatic fenders to follow the maximum forces recommended in Table 6-6, however typical reaction pressures for foam and pneumatic fenders are in the order of 170 kN/m².

Some vessels may require the use of very low hull pressure fenders. To solve this, some manufacturers offer extra-soft foam fenders that are characterised by a low hull pressure limit, as low as 110 kN/m².

For some vessels (e.g. naval ships or cruise ships, which also often require non marking fenders), hull pressures limits may be even lower than the values given in Table 6-6. For specific applications such as this, it is recommended that fender manufacturers are consulted.

6.8.8 Typical Hull Pressure Capacities

Ideally, designers of a fender system would be able to obtain, or be provided with, the permissible hull pressure limits of the design vessel(s) that the fender system is required to accommodate. In practice, permissible hull pressure limits are difficult to obtain and as such, the designer should make some reasonable assumptions to size the fender panel or contact area correctly. There is a trend in the industry to lower allowable hull pressures. If a value is provided the fender designer should always ask for the substantiation of the value as it is sometimes anecdotal and the corresponding maximum allowable fender reaction force.

It is acknowledged that most berths are required to be designed to accommodate a wide range of vessels and to provide the operator of the facility with the greatest degree of operational flexibility. Calculation of the maximum equivalent permissible hull pressure for every vessel type is not practical or cost-effective and the use of general guidance is considered to be sufficient.

Table 6-6 can be used as a general guide for a range of applicable hull pressures in the absence of vessel specific information. These values include the factors of safety normally used by Classification Societies. The design reaction force ($R_{f,d}$) of the selected fender and the

fender panel contact area is used to determine the applicable load acting on the hull structure of the berthing vessel and to verify that the ultimate hull pressure limit is not exceeded. Refer to Section 6.7.1.3.

Type of vessel	Maximum ultimate hull pressure ($P_{hull,d}$) ⁽¹⁾ (kN/m ²)	Maximum ultimate fender reaction force ($R_{f,lim}$) (kN)
General Cargo		
≤ 20,000 DWT	500	NK ⁽⁶⁾
> 20,000 DWT	400	NK ⁽⁶⁾
Bulk Carriers		
≤ 60,000 DWT	200	2,200 ⁽⁷⁾
> 60,000 DWT	320	3,800 ⁽⁷⁾
Container		
Panamax and smaller	400	1,500 ⁽⁷⁾
Neo/post Panamax and larger	200	5,600 ⁽⁷⁾
Tankers (see WG 153)		
≤ 60,000 DWT	300	1,800 ⁽⁷⁾
> 60,000 DWT	200	NK ⁽⁶⁾
Gas carriers (LPG & LNG)	200	NK ⁽⁶⁾
Cruise		
≤ 20,000 DWT	400	NK ⁽⁶⁾
< 60,000 DWT	300	NK ⁽⁶⁾
100,000 DWT	200	NK ⁽⁶⁾
Passenger Ferries and RoRo		
RoRo (belting)	Refer Notes 3, 4 and 5 below	NK ⁽⁶⁾
RoRo (no belting)	Refer to equivalent size of Cruise Vessel	NK ⁽⁶⁾
Passenger (belting)	Refer Notes 3, 4 and 5 below	NK ⁽⁶⁾
Passenger (no belting)	Refer to equivalent size of Cruise Vessel	NK ⁽⁶⁾
SWATH (double hull vessels)	Refer Note 5 below	NK ⁽⁶⁾
NOTES:		
1. The maximum hull pressure values above include for the deduction of factors often used by Classification Societies and can be considered as calculated pressures for semi-probabilistic analysis, associated with non-failure conditions. These values can be considered to act independently to each contact area, with the exceptions as noted in points 2, 3 and 4 below.		
2. Where fenders are located adjacent to each other (e.g. a vertical or horizontal separation that causes each fender to act in combination with the adjacent fender) and are considered to share the absorption of the vessel berthing energy, the designer may need to consider the forces acting on the hull from these fenders as a group to ensure that the maximum ultimate hull pressure or maximum ultimate fender reaction force capacity is not exceeded.		
3. Car carriers, RoRo, ferries, cruise, fishing boats, barges and some auxiliary vessels with small displacements, often include one or more belting lines located at different levels on the vessel hull. Refer to section 6.4.8 for more information. These beltings are typically rectangular, trapezoidal or circular in section, protruding approximately 20 to 40 cm from the vessel hull. The designer should check that the magnitude, position and direction of all loads generated by the presence of the belting can be accommodated by the vessel hull structure/plating/belting and the fender system itself. The geometric profiles of the fender panel may need to be refined to accommodate and prevent double hull contact at various states of the tide.		
4. The typical maximum permitted loads on beltings for vessels with steel hulls is usually between 2,500 and 5,000 kN/m. However, it is recommended that specific information on the structural capacity of the belting is obtained to verify that the forces generated by the fender system can be accommodated. For line loads not on beltings 1000-1500 kN/m is often used. These are empirical numbers that have not been verified.		
5. Some fast ferries, as well as catamarans, especially if they have aluminium hulls, are not capable of accommodating any kind of direct impact load onto the bottom sections of the hull. In such cases, unless they have special designed belting, the designer will need to ensure that the berthing load impact area is located at a strengthened area of the vessel, independent of the water level.		
6. 'NK' (not known) means that these values have not been calculated by (Berendsen E. A., 2022) or WG 211.		
7. The numbers are based on (Berendsen E. A., 2022) and are derived from specific vessels. The designer should verify if design vessel and calculated vessel indeed match.		

Table 6-6: Typical values of hull pressure capacity

The equivalent maximum ultimate hull pressure ($P_{hull,d}$) should therefore be greater than the average hull pressure ($P_{hull,av}$) or the maximum ultimate hull pressure ($P_{hull,max}$).

$$P_{hull,d} \geq \max (P_{hull,av} ; P_{hull,max}) \quad (6-22)$$

Since the equivalent maximum ultimate hull pressure $P_{hull,d}$ is non-linear, the values listed in Table 6-6 can only be used when $R_{f,d} \leq R_{f,lim}$.

Where hull pressures may be critical to the design of the fender system (i.e. naval/military vessels), a naval architect or vessel owner should be consulted for the provision of specific requirements.

Certain operators may require a berth and associated fender system to accommodate a specific, unique vessel type. In such cases, if details of the vessel side plating, longitudinal stiffeners and side transverse frame separation information is provided, the ultimate hull pressure can be calculated, providing the designer with greater confidence in the fender system design.

6.9 Ship-to-Ship Fendering

The requirements for the provision of offshore fendering for ship-to-ship transfers are covered in detail in the MEG4 (OCIMF, 2018), the OCIMF STS Guideline (OCIMF, 2013), (IMO, 2011) and the particular Class requirements for offshore cargo transfers.

The design considerations included within WG 211 are related to nearshore transfers, the use of Floating Storage Regasification Units (FSRUs), etc. where the preferred fenders for ship-to-ship berthing are somewhat different to those for vessel-to-dock berthing. For ship-to-ship situations, the most important parameters to consider are stand-off distance, relative approach velocity in different weather conditions and fender type.

The fender reaction against the vessel hull structure is also an important design consideration. However, most fenders suitable for the use as ship-to-ship fenders are designed to exert reaction forces substantially lower than the ultimate vessel hull structure limits.

It is rare that vessels carrying out ship-to-ship berthing manoeuvres make a parallel approach. They almost always contact only one fender at the point of the initial impact. Each individual fender should therefore have sufficient energy absorption capacity to accommodate the largest anticipated individual design berthing load. A proportion of the berthing energy is absorbed by the vessel pivoting about the fender. In the event of parallel contact, the berthing energy will be distributed across all contacted fenders.

6.9.1 Ship-to-Ship Fender Selection

To select a suitable fender for ship-to-ship berthing, the designer should calculate the ship-to-ship berthing energy ($E_{k,d}$) at the point of contact. The berthing energy calculation for ship-to-ship berthing is outlined in Chapter 5.

The relative approach velocity of the berthing vessel can have a significant effect on the berthing energy absorption requirements of the fenders. The selected relative approach velocity should consider the effects of local weather, sea and swell conditions, vessel tug or thruster capability and the overall size of the vessels involved. Smaller vessels tend to have higher approach velocities. Recommended characteristic approach velocities for a range of conditions, are also included in Chapter 5.

The selected fender should have a Design Energy absorption performance ($E_{f,d}$) greater than the ship-to-ship berthing energy ($E_{k,d}$).

$$E_{f,d} \geq E_{k,d} \quad (6-23)$$

The selected fender is also assessed to ensure that it provides the required stand-off distance between the vessels and that an adequate number of fenders is provided.

Many ship-to-ship operations use large foam-filled or pneumatic fenders. The fenders used in ship-to-ship transfer operations offshore are typically divided into two categories.

- Primary fenders, which are positioned along the flat sided parallel body of the vessel to afford the maximum possible protection while alongside.
- Secondary fenders may be used to protect bow and stern plating from inadvertent contact during berthing and unberthing.

6.9.2 Vessel Stand-off and Number of Fenders

The required stand-off distance for ship-to-ship operations is often the primary consideration in the selection of the appropriate fender diameter or type.

The roll angle of the vessel (which is a function of sea conditions, vessel dynamics, vessel forward speed and orientation, relative to the prevailing wind and sea conditions) needs to be assessed to determine the minimum required stand-off distance. The stand-off distance should be large enough to keep the vessel hulls and superstructures from contacting as the vessels move independently of one and other.

Product transfer equipment (i.e. hoses, manifolds, booms, etc.) may also need to be considered in the required stand-off distance. The stand-off distance should also include a sufficient margin of safety.

The approximate stand-off distances, the recommended minimum number of fenders and the typical fender sizes are provided in Table 6-7, for high pressure pneumatic fenders in calm weather conditions. These parameters are related to the displacements of both vessels involved in the ship-to-ship berthing, defined by the Combined Vessel Mass Coefficient (M_{cv}). The method for calculating the M_{cv} is calculated as follows.

$$M_{cv} = \frac{2 M_1 M_2}{M_1 + M_2} \quad (6-24)$$

Where,

M_{cv} Combined vessel mass coefficient (tonnes) (without added mass factor)

M_i Displacement of vessel (tonnes)

The final required stand-off distance is likely to be greater if more onerous sea and weather conditions are to be accommodated (e.g. large wave heights, strong winds, etc.) or special operational limitations on vessel movement are required.

It is recommended that fender suppliers and operators experienced with ship-to-ship berthing are consulted prior to confirming the selected fender size and required number of fenders.

Combined Vessel Mass Coefficient (M_{cv}) (tonnes)	Minimum Stand-Off Distance (metres)	Minimum Number of Fenders ^(a)	Typical High Pressure Pneumatic Fender (50 kPa) size ^(a) (metres)
6,000	1.2 - 2.0	3	2.5 x 5.5
10,000	1.5 - 2.2	3	2.5 x 5.5
25,000	1.5 - 2.2	4	3.3 x 5.5
50,000	1.8 - 2.5	4	3.3 x 6.5
100,000	2.4 - 3.3	4	3.3 x 6.5
150,000	2.4 - 3.7	5	3.3 x 6.5
200,000	2.4 - 3.7	5	3.3 x 6.5
330,000	3.3 - 4.0	4	4.5 x 9.0
470,000	4.0 - 4.5	4	4.5 x 9.0
790,000	4.2 - 4.5	4	4.5 x 12.0
Note: If the characteristic design vessel displacement is between two of the values stated, then use the larger vessel displacement for fender stand-off distance selection. ^(a) (OCIMF, Ship to Ship Transfer Guide for Petroleum, Chemicals and Liquefied Gases, Section 9.1.2 and Table 9.1., 2013)			

Table 6-7: STS recommended stand-off distance, number of fenders and typical sizes for calm conditions

6.9.3 Ship-to-Ship Operational Considerations

Ship-to-ship operational considerations are highly dependent upon the specific type of the operation, including any operational limitations on permissible sea states. It is recommended that the vessel operators are consulted before the final fender system is selected and that the mooring arrangement for the ship-to-ship berthing should be considered by the designer. The mooring arrangement should be compatible with quick departure manoeuvres under emergency conditions.

For light/ballasted or other similar cargo transfer operations, one vessel is initially fully laden and the other light/ballasted. The berthing energy for a given approach velocity would therefore be less than two fully laden vessels. Allowance for this can be made by adjusting both the displacement tonnage and the added mass of the light vessel.

As a light vessel is likely to have a greater windage area, the berthing approach manoeuvre may be difficult. The effect of waves may also be greater on the light vessel. Both factors may contribute to higher approach velocities than normally anticipated.

After the two vessels are located alongside one another, operational considerations will dictate whether the actual transfer operation can occur while the vessels are underway and maintaining steerage or whether the vessels must stay in one position. For a moored vessel with a second vessel alongside, the upper limit of sea conditions under which the operations can continue must be considered.

7 FENDER SELECTION UNDER MOORED CONDITIONS

This chapter outlines the design considerations that are applicable for fender systems that are required to accommodate moored vessels. The moored condition can be the governing condition for either the fender, the supporting structure or both.

For fender systems designed specifically for berthing conditions, it is necessary to check the suitability of the selected fender system for moored vessels, in particular the 'lean-on' mooring loads (pushing the moored vessel onto the fenders). Mooring conditions are important when considering the dynamic external actions acting on the vessel. These actions include, but are not limited to, waves, wind, current and passing vessel loads. The potential for extreme events such as hurricanes, squalls, etc. should also be investigated and included within the assessment if the vessel stays moored (mentioned in the basis of design). Whilst berthing manoeuvres can be controlled, the mooring conditions may transmit higher loads into the fender system.

Where berths are located close to navigation channels and other vessel manoeuvring areas, the effect of the passing vessel close to the moored vessel at the location of the fenders should be considered in a fender selection process. Passing vessel suction forces and wake wash can lead to a significant response on the moored vessel and, consequently, on the fenders.

Pre-tensioned mooring lines, e.g. with constant-tension winches, and sustained wind loading (constant loads) can induce permanent deflections in a fender, which can increase over time.

In all cases, mooring conditions and loads should be assessed.

A wide variety of conditions can affect the dynamic response of a vessel. These conditions can induce the movement of a moored vessel, indicated by the cyclic movements in the six degrees of freedom. These movements can subsequently induce cyclic loading within the different elements of the mooring and fender system. As mooring arrangements are dynamic, a careful assessment should be undertaken to establish the effects on the moored vessel.

7.1 Fender Design for Moored Vessels

The typical fender system design approach is to calculate the vessel berthing energy and then select an appropriate fender system, refer to chapters 5 and 6. However, berthing energy is not always the only design condition to be considered for the selection of a fender system. The fender system should also be capable of accommodating the range of conditions that will be experienced by the vessel while moored for prolonged periods of time. For example, certain environmental conditions that may occur while the vessel is moored may result in higher loads on the fender system (both normal reaction and friction forces) than would be typically experienced during berthing.

Figure 7.1 illustrates how a mooring analysis often fits with the fender system design process. A mooring analysis can be performed at the early stage of a design project to assist with optimising the fender configuration and design. Initiating the mooring analysis early is particularly important if the designer expects that the moored conditions may be the limiting factor in the fender system design or that the fenders may be the limiting factor in determining the operational conditions for the berth. The design of the fender system may therefore have a direct impact on the berth operation and downtime performance.

The designer should define or select a set of limiting scenarios for mooring analysis, including the design vessels or fleet, mooring arrangement of the vessel, mooring equipment on the berth, weather conditions (wind, waves, current, water level, etc.) and other excitation forces, depending on the project.

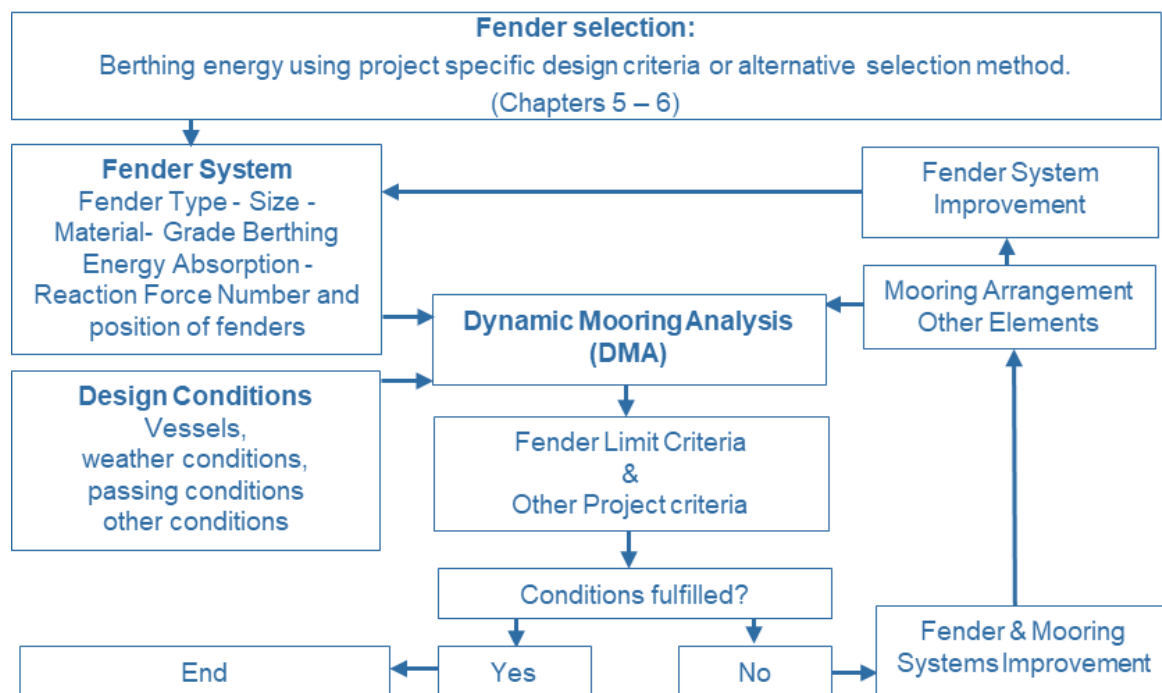


Figure 7.1: Fender design flow chart for moored conditions

Once the mooring analysis has been carried out, the results are compared with the project design applied to both the fender system and to the mooring system. If the criteria are satisfied, the mooring analysis is completed. If it is not satisfied, it is necessary to improve the fender or mooring systems and repeat the analysis. In some cases, the solution may consist of setting up a set of operational measures.

The differences in fender performance between different types, sizes, and grades of fenders may lead to significant differences in the dynamic response of the moored vessel. If the design of the fender system is controlled by the outputs of the mooring analysis, it is recommended that a sensitivity analysis is undertaken to compare the different fender performance characteristics to determine whether a particular fender is more suitable. This is particularly beneficial at the project tender stage as the fender supplier is not usually known.

In some design situations, the fenders are not the only flexible part of the structure. When the fender is fixed to a flexible structure (e.g. flexible dolphins), the response characteristics of the support structure should also be considered as to accurately simulate the dynamic response of the mooring system.

The vessel hull response is not included in a mooring analysis due to the high stiffness of vessel hulls compared to the fender stiffness. Hull pressure, resulting from the peak fender reaction, should be checked against the hull pressure criteria as noted in Section 6.8.

Although the mooring analysis is often performed using the average performance curves for the selected fender, designers should also be aware of the non-elastic characteristics of the fender performance. In certain mooring conditions, the non-linear characteristics of the fender element can be important.

Safety factors and manufacturing tolerances are not usually included in the fender characteristics for mooring analysis, these factors and tolerances are applied, as defined by the manufacturers, in the postprocessing to check the limits. The average reaction-deflection performance curve is the most common choice to describe the moored vessel behaviour.

Depending on the objective of the study, performance curves different from the average may be selected.

The non-elastic effects occur in the form of hysteresis in the fender performance curves. The rate at which the fender rebounds is not simply a function of the instantaneous position on the performance curve, however is also a function of how the fender has been worked up to that point in time. Usually, this information is not available and cannot be included in the analysis.

In a permanent mooring situation and considering cyclic loading, the fender element can dissipate a significant amount of energy (in the form of heat), which affects the fender performance.

Under high constant loading of the fender, creep in the fender element can lead to deformation or buckling in a rubber fender at a load significantly less than the design reaction force. A mooring analysis allows a check on the fender deflections (or compressions) under constant load conditions to be conducted, using the creep criteria; refer to Section 7.4.7.

7.2 Characterisation of Vessel and Berth Configuration

Evaluating a fender's performance with a vessel moored alongside requires an understanding of the relevant vessel and the characteristics of the berth, as well as an understanding of the likely mooring configurations. Chapter 4 includes information that can be used to characterise vessels, when data is not available for a project.

The parameters for fender contact with a moored vessel can vary based on the specific project. The main parameters to be considered include the size/position of the vessel parallel mid-body, the presence (or absence) of vessel belting, and the strength/structure of the hull. The location of the vessel fairleads is also often relevant, both in support of the mooring analysis and for checking for geometric interferences between mooring lines and the fenders.

An accurate layout of the terminal geometry, including the number, size and location of fender systems, is important for evaluating the fender-vessel interface under moored conditions. The designer should ensure that adequate fender-vessel contact is maintained throughout the range of anticipated water levels and vessel loading conditions.

All possible mooring positions and configurations should be considered by the designer to ensure that the fender system can support vessels for all service conditions. For example, an oil tanker berth may always moor a vessel in the same orientation and approximately the same location (+/- several metres). However, a bulk terminal may require the vessel to moor at various positions and both orientations to fill all holds of the vessel systematically.

Mooring configurations can vary greatly from jetty or quay wall moorings to unique layouts involving combinations of fixed structures with mooring buoys and/or use of the vessel's anchor. A mooring assessment is also required for ship-to-ship moorings to assess the mooring arrangement and the behaviour of the moored vessel including the relative motions between both vessels.

All elements of the interface between the fender system, the vessel and the berth should be checked for interferences that are likely to occur during vessel loading/unloading operations. All vessel positions should consider, as a minimum, the following interferences:

Interferences between the fender systems, mooring lines and mooring points on the front side of the berth.

Interferences between the fenders and ladders, gangways, etc. and access to the vessel. Interferences between the fenders with loading arms or flexible hoses, cranes or another loading system.

If possible, interferences must be avoided or minimized during design process, if not feasible, suitable protection is required for the elements involved (mooring lines, fenders, fender panels, ladders, other equipment onboard).

7.3 Mooring Analysis

A mooring analysis is the design process assessing the response of the moored vessel to imposed environmental loads such as winds and waves, when moored at a berth. A defined mooring layout and fender system is considered and assessed against the effect of a set of defined met-ocean conditions or other external forces. Refer to PIANC WG 186 for guidance on how to select the analysis method, how to perform a mooring analysis, for determining what conditions should be simulated and what design criteria should be applied.

A mooring assessment can be a particularly important part of the fender design process for marine terminals subject to cyclical loading from swell waves, infra-gravity waves, high forces from passing vessels, extreme high wind events and/or tsunamis.

Mooring analysis results can provide an estimate of the maximum fender loads under the design or operational environmental conditions, as well as the distribution of mooring loads between multiple fenders. Additionally, a mooring analysis can illustrate how the vessel will move while at berth and assist with optimising the fender placement to ensure contact of the fenders with the vessel's hull.

The decision to improve the fender system is usually based on the performance of the fenders under the design conditions (maximum forces or deflections). However, different criteria (operational and safe mooring limits, downtime, fatigue, etc.) may also indicate that changes or refinements are required to the fender system design. In the same way, the results of the fender system assessment may suggest the improvement of the mooring layout or changes in the berth configuration or mooring equipment in the terminal.

Static Mooring Analysis (SMA) can be used in the first assessment of a project. The dynamic effects can induce higher forces than the typical static forces plus safety margin in SMA. The designer should assess the necessity of performing a dynamic mooring analysis (DMA) for each project, as SMA may lead to underestimated maximum forces on the fenders.

7.4 Creep and Fatigue Assessment for Dynamic Mooring Situations

Fender fatigue analysis is typically not required in protected ports. However, in exposed locations, the effects of constant swells or frequent and strong gusty wind should be carefully considered. In the same way, extreme conditions may reduce the resisting capacity of the fenders. Since fenders can perform substantially differently than the published catalogue data under cyclic load conditions, the fender supplier should be consulted for design assistance in these situations.

Both high constant fender loads (e.g. high pretension or sustained winds) and cyclic fender loads (e.g. under swell waves) are substantially different from the testing conditions represented by catalogue fender performance data.

The effect of fatigue and creep on fenders should be assessed. When identified that the local conditions at the berth may be of relevance for fatigue fenders, an assessment should be performed.

Fatigue in fenders is a process of progressive (internal) damage occurring in the fender body material that is subjected to cyclic loading and which may lead to the failure of the fender in satisfying the purpose it was designed for.

Fatigue life is defined as the maximum number of loading cycles that an element can sustain before failure occurs. The fatigue life depends on the nature of the fender and of the nature of the loading cycles, being the shape of the loading cycles, for which the most important parameters are the shape and the period.

All fender types suffer fatigue of a certain nature and there are important differences depending on:

- Material, additives, grade and manufacturing process.
- Fender type, considering the shape of the performance curve.
- Ageing.
- Service temperature.
- The magnitude of the compression loads itself.

Fatigue effects can affect all the installed fenders, however it is especially important and should be assessed in the design process for:

- Permanent mooring solutions.
- 'Important' cyclic loading due to local conditions.

Besides the rubber element of a fender system, the steel structures should also be checked for fatigue loading e.g. torsion arms and chain fixings.

Creep is the permanent deflection of fenders after loading (CDIT; Design, 2019). Creep in rubber and foam filled fenders is caused by different phenomena. Foam filled fenders require a longer time to recover after each load cycle as some inner foam cells release the entrapped air, and a reduced number of these cells do not recover the original status, leading to creep.

For all type of fenders, there is an important relationship between creep and fatigue damage as creep increases the effect of cyclic loading.

Fender Type	Creep Damage Description
Rubber fender	Constant loading does not create important permanent deflection on the rubber fenders but affects the way cyclic load damage the fender.
Foam filled fender	Under long term static loading from a moored vessel, e.g. with pre-tensioned mooring lines, the deflection can be permanent (creep) Constant loading induces important permanent deflection on the foam fenders, this effect is additional to the permanent deflection caused by cyclic loading.
Pneumatic fender	This fender type does not typically show creep in their behaviour and permanent deflection is not found.

Table 7-1: Description of creep damage in fenders depending on their type and material

The effect of **fatigue** depends on the fender type and construction material. For rubber fenders, the heat generated inside the fender material due to the load cycles increases the working temperature of the fender. This occurs when the periods are small (especially

compared to the berthing period). The shorter the time available to a fender to cool down, the higher the temperature rise will be. In the same way, the higher the load in the load cycle, the greater the amount of energy to be dissipated, and the effect on fatigue will be greater. The main measure to prevent fatigue damage is selection of the right fender type and rubber compound to prevent the fender from continuous buckling and rubber compound with as less heat build-up as possible.

Fender Type	Fatigue Damage Description	
Rubber fender	Buckling <i>Including cone, cell, leg and arch types</i>	Buckling fenders suffer a definitive change in shape after the start of buckling. Before buckling, the buckling fenders show an almost linear behaviour with the fender load. This is important as dynamic load buckling of the fender has an important role on fatigue damage. Therefore buckling should be prevented in fatigue loading situations.
	Non-Buckling <i>Including cylindrical and solid types</i>	This occurs when there is no buckling effect and no sudden change in the behaviour of the loaded fender. There is always a slight change in fender shape, loading the fender over this point increases fatigue damage. See Table 2-6 and Table 2-7.
Foam filled fender <i>Floating fender</i>	Fatigue affects foam fenders in a different way, as every time a foam fender is compressed, entrapped air in the inner cells of the foam is released and a reduced number of these cells do not recover the original status. The recovery time of a foam fender is most often longer than the frequency of the fatigue load meaning that a permanent deflection of the fender (creep) occurs. The magnitude of this creep is highly dependent on the local conditions and mooring layout of the vessel at berth. Creep reduces the future energy absorption capacity of the fender so the design of foam filled fender system for permanent moorings should consider this effect.	
Pneumatic fender <i>Floating fender</i>	Pneumatic fenders do not suffer significantly from fatigue, as the main elastic element is the air inside. Fatigue on the skin (rubber) is very limited and is considered not relevant. However there are two important factors to be considered by the designer: • Fatigue on valves and elements of the pressure control system. • Risk of losing the fender due to damage. In that case, the fender is destroyed and loses all functionality. The possibility of such damage should be taken into account by the designers, especially in permanent moorings.	

Table 7-2: Description of fatigue damage in fenders depending on their type and material

Within current guidelines, there is insufficient published data to provide sound limits for fatigue and creep. Specialists' advice from manufacturers for fatigue and creep assessment is required, for those cases, where permanent deflections or cyclic loading is expected.

Operational conditions should be described in the specification and during the purchasing process in order to assess the nature and characteristics of the cyclic loading.

Designers are responsible for considering these effects and looking for manufacturer's advice when assessing fatigue. Manufacturers are responsible for defining the fatigue limits when there is no data published in support of these limits.

If fatigue may be an issue due to the berth conditions, fatigue testing results should be performed and compared with the expected number of cycles during the lifetime of the fender from dynamic mooring analysis (DMA).

Different limits may be required for creep and fatigue when assessing fatigue. When accurate data for fatigue fender design limits is available, only fatigue should be assessed, however due to a lack of data, both creep and fatigue limits should be applied.

Due to the potential permanent deflection that may be found within foam filled fenders, the selection of this type of fender should be carefully assessed for locations where permanent loads or significant cyclic loads are expected.

In order to assess the constant fender loads, the constant deflection of the fenders should be considered. It is important to note that static or quasi-static calculations are not a valid method to assess creep except in fully static situations where loads are due only to pre-tensioned mooring lines with no significant cyclic loading.

Constant deflection can be estimated from a dynamic mooring analysis as the average of the deflection time series during three hours under the design conditions. The design conditions are taken as extreme conditions for permanent mooring and maximum safe mooring conditions for temporary stays of the vessel at the berth. The longer the fender is exposed to this load, higher the risk of suffering from creep.

The following picture shows the time series of fender deflection for an extreme gusty wind, pushing the vessel against the fenders. The average of the time series is the parameter to assess creep.

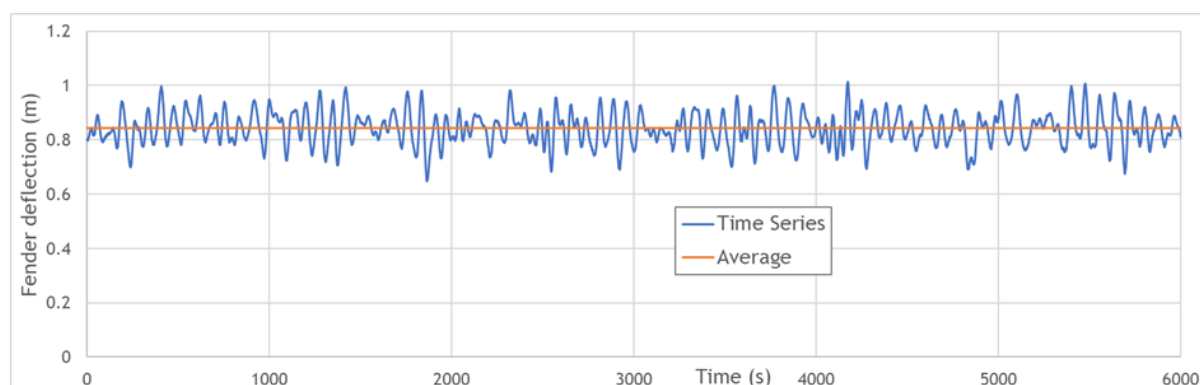


Figure 7.2: Example of fender deflection time series showing average value and different cycles

Durability test data for fenders (refer to Section 10.6) typically considers repeated compressions with a cycle duration of 150 s. However, environmental loading can cause compressions at a much shorter cycle period, resulting in a much higher rate of heat transfer in the fender. This can lead to softening of the rubber and larger deflections, limiting the ability of the fender to rebound between cycles.

Loading with a longer cycle period results in slower heat transfer to the fender elements and allows more time for the fender to rebound between compressions.

Fender fatigue analysis is typically not required in protected ports. However, in exposed locations, the effects of constant swells or frequent and strong gusty wind should be carefully considered. In the same way, extreme conditions may reduce the capacity of the fenders. Since fenders can perform substantially differently from the published catalogue data under cyclic load conditions, the fender manufacturer should be consulted for design assistance in these situations. The analysis should consider not only the maximum load or deflection, but the distribution of loads as a function of frequency.

When specific data from the manufacturer is not available and cyclical loads are expected, engineering judgement is required in determining a reasonable limiting cyclic load condition for the fenders.

The following picture shows the energy power spectrum for a single fender under certain conditions. The peaks of the spectrum show the frequencies where energy is concentrated. In that case two peaks can be clearly identified. The effect of these two peaks of energy on the fender is different, depending on the capacity of the fender to dissipate the heat generated by the deflection and on the frequency (higher frequency, less time to dissipate the heat).

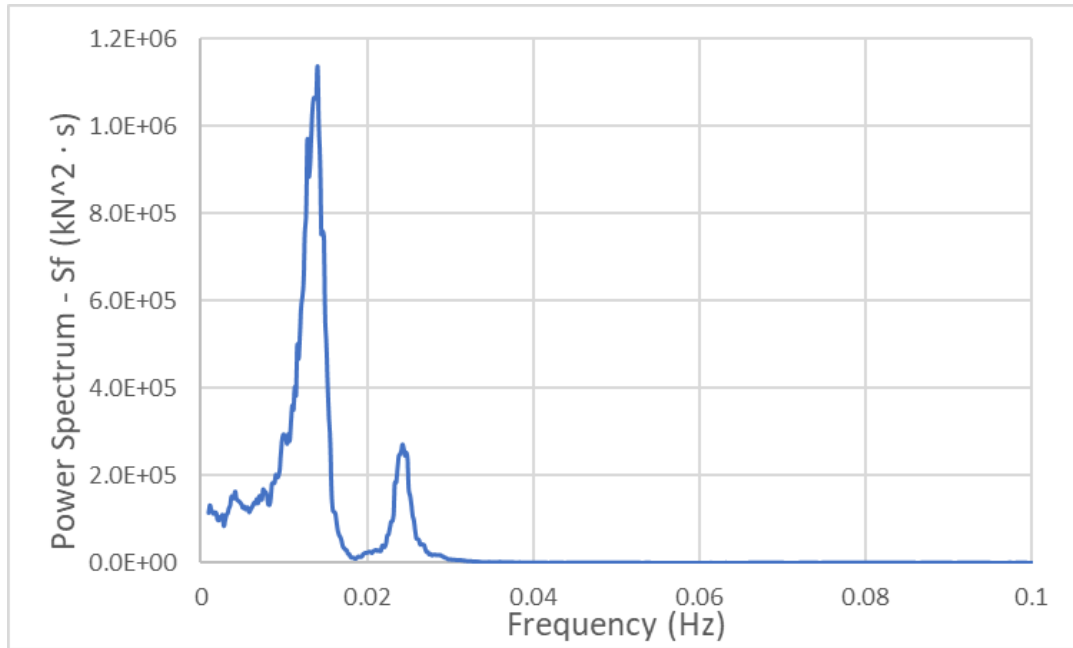


Figure 7.3: Energy distribution as a function of frequency

7.4.1 Creep and Cyclic Loading Limits for DMA

The reference from Japan (CDIT; Design, 2019) provides references for creep and fatigue limits that can be extended to buckling fenders. In absence of project specific limits provided by the manufacturers or other reliable source, the following limits can be used for buckling fenders (cone, cell, leg and arch fenders):

- Buckling fenders should be carefully selected so that they will not buckle due to creep after prolonged static loads by continuous wind or currents, or not to be subjected to repetitive motions of vessel at the deflections which exceed the buckling peak of the reaction force.
- Creep limit: Continuous load under design conditions will not exceed 40% to 50% $R_{f,c}$, typically equivalent to 5% - 10% deflection.
- Fatigue limit: Maximum cyclic deflections under design conditions will not exceed the buckling limit (peak of the performance curve)

At present, there is not enough data to provide general creep and cyclic loading limits. We include some general guidelines for all type of fenders that may help designers:

- Fenders should be carefully selected so that they will not creep under prolonged static loads by continuous wind or currents and not be subjected to repetitive motions of the vessel that lead to a fast deterioration of the fender.
- Using design deflections as the limit for fenders under significant cyclic loading is not a conservative approach and may lead to non-conservative results when assessing mooring conditions.
- Some studies suggest for some specific fenders the change in the behaviour of the fender (i.e. start of buckling, change in fender shape,) may be considered a limit for cyclic loading. Unfortunately, these results cannot be generally extended to all fender types.
- Foam fenders, due to creep, might be more sensitive to effects of cyclic loading or prolonged static loading than other fender types.

The application of these limits for buckling fenders and comments does not discharge the designer from the responsibility of researching for specific data and to consult with the suppliers.

8 FENDER SYSTEM COMPONENT DESIGN

Careful consideration should be given to the design of the fender panel, chains, UHMW-PE pads and fixing details to ensure an efficient, yet robust design of fender systems, and should be part of the integral berth design, since it is necessary that berth structure configuration can accommodate the required fender system and all its components (chains, anchors, etc.). It should be noted that there is a serious risk of failure or reduction of durability of the fender systems if the design of the components is not correct or if the berth design does not match the fender system requirements for the structure.

This chapter outlines the design process for fender system components, providing background information on the issues that should be considered when designing these key elements of the fender system.

8.1 Fender Panel Design

The fender panels are an integral part of several fender systems (refer to Chapter 2). The fender panel should provide a suitable contact surface positioned at a suitable level. The designer should consider variations in tide, weather, operations, etc. Variations in vessel hull profile and structure should be accounted for in the panel design to distribute the fender reaction forces into the vessel's hull structure. Fender panels may be subject to a combination of uniformly distributed loads, line loads and point loads according to the hull profiles of the vessels using the berth.

During berthing and mooring, vessels contact the fender panels, inducing bending moments and shear forces into the panel structure which must be accounted for in the panel design. The designer should also be aware of the effects of asymmetric load cases and consider where stress concentrations may exist, such as at bolted fender system connections and chain brackets connections.

Fender panels should be designed to give a minimum service as defined in Chapter 0, assuming adequate planned maintenance during the serviceable life, according to suppliers maintenance manual.

Fender panel design should determine the overall panel dimensions, edge chamfers and pressure distributions within the panel. The design should also determine the internal structure of the panel, including the location and thickness of all plates, stiffeners and the associated weld sizes. Limit state design codes should be used to determine the construction of the steel panels and frames. Input loads from the fender reactions, the vessel hull profile and chain connections should all be considered.

The designer is recommended to consider corrosion protection of the panel to optimise durability.

8.1.1 Load Cases and Structural Analysis

Vessels typically contact fender panels either as a flat hull contact or, if beltings are present, as several line loads or point loads. The exact geometry of the vessel contact is dependent on the type of vessel and the angle of approach of the vessel to the berth.

The designer should consider each design case as being unique and should carry out a detailed assessment of all fender design criteria to determine the specific panel design load

cases. Typical vessel contacts with fender panels and example load cases are indicated in Figure 8.1.

The range of different load cases identified will create varying demands on the fender panel structure. A durable fender panel design is highly dependent on correctly identifying and defining all applicable load cases that can reasonably be expected to occur during the life cycle of the fender system.

Design calculations should consider bending, shear, local buckling and crushing effects in the steel panels and fender frames. Local buckling should be checked as inadequately supported webs in the panel grillage may be prone to collapse under line load or point load conditions.

Vessels with belting can be particularly damaging to panel structures and facings, leading to line or point load cases. A careful determination of load cases and structural analysis can assist in providing sufficiently robust structures. Consideration as to whether vessels with belting are berthed frequently or if the belting on the vessels is poorly maintained can also be of benefit in the design process.

Berthing frequency should be considered in the design of the fender system if exposed to fatigue conditions (see chapter 7), fatigue analysis should be undertaken as part of the design process. Special care is needed where stresses are concentrated, such as chains, brackets and bolted connections.

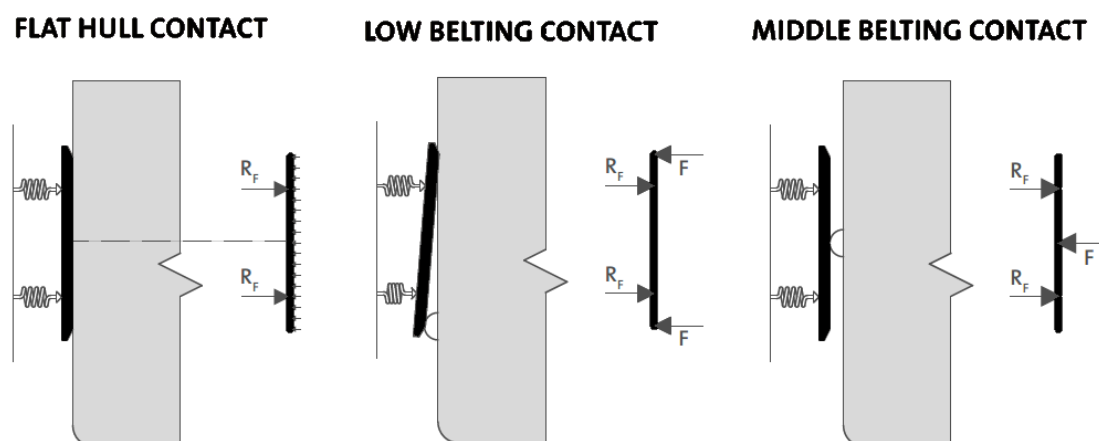


Figure 8.1: Typical design cases of fender contact with vessel hull profile

Fender panel design calculations should, as a minimum, consider static principles to determine shear forces and moments. More detailed analyses, i.e. finite element analysis methods, should be considered to study the more complicated effects of asymmetric load cases and the distribution of stresses induced within the steel structure. Partial load factors according to Section 6.7.4 unless local codes should be applied for the design of the fender panels when considering the characteristic and design impact cases. With reference to the application of partial safety factors related to the applied impact loads, these should be considered as non-permanent loads.

8.1.2 Fender Panel Internal Structure

The overall weight of a fender panel provides a preliminary estimate and general 'rule of thumb' of the panel capacity to support particular load combinations. A typical range of panel weights and intended uses is as follows (these values are only indicative):

- Standard duty panels: approximately 250 to 300 kg/m². Normally designed for flat hull contact.
- Heavy duty panels: approximately 300 to 400 kg/m². Normally designed for belting and line loads.
- Extreme heavy-duty panels: over 400 kg/m². Normally designed for point loads or special cases.

The typical panel weight can be defined in the project design criteria and can be considered when comparing different alternatives to validate the likely robustness of the steel fender panel design.

Generally, two types of fender panel structural forms of construction are available, the 'closed' and 'open' box. The 'closed box' form of fender panel is considered to have a higher strength-to-weight ratio and creates a simple exterior shape which is easier to paint and maintain. The inside of the 'box panel' is protected against corrosion as it should be fully sealed, and pressure tested. It is therefore recommended that quality control of the internal panel construction is verified before the box is closed and sealed.

Recommended minimum thicknesses for the steel sections in fender panels are as follows:

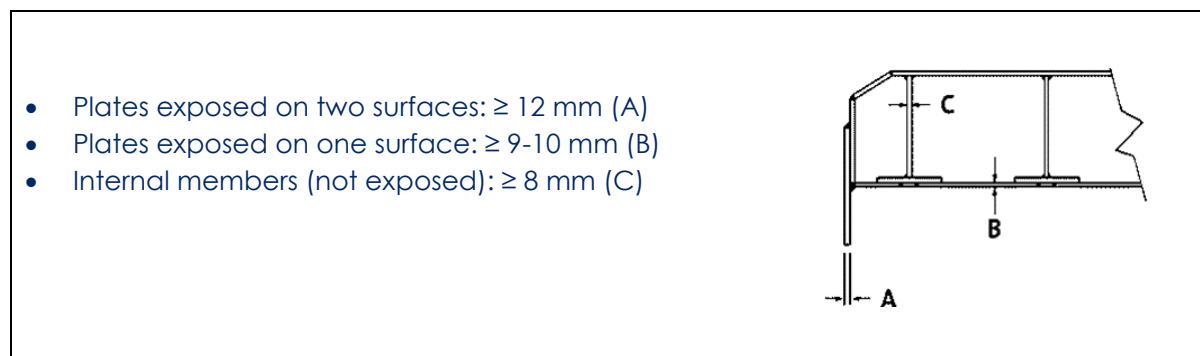


Figure 8.2: Recommended minimum thicknesses for steel in fender panels

A panel is a construction of welded steel plates forming one single piece. Construction of a single fender panel should not be made one or more pieces bolted to each other. It is possible to link together two separate fender panels by panel hinges built in the side of the panels.

Chains are connected to brackets welded to the internal structure of the panel; therefore, chain position and loads are considered as part of the panel structure design.

The grade of steel used for the fabrication of the fender panel should be clearly specified considering the design temperature range (as low temperatures require special steel grades which do not become brittle), fender panel loading and the availability of certain steel grades.

8.1.3 Edge Chamfers

The fender panel design should consider the potential for the direct application of vertical or horizontal loads to the perimeter of the panel. Such forces could result from:

- vessel belting contacting the top of the panel during vessel movement whilst at berth.
- vessel belting contacting the top of the panel on a falling tide and/or when being loaded.
- a low freeboard vessel or vessel belting becoming hooked underneath the fender panel.
- vessels with discontinuous and multi-level beltings.
- direct horizontal loads to the fender panel side due to small manoeuvring mistakes.

These types of impacts can seriously affect the durability of the fender system, introducing potentially destructive shear forces into the rubber unit, which it is not designed to resist. Fender panel design should consider options to minimise potential damage to the fender system by including edge chamfers or bevels around the edges of panels.

The minimum recommended width and angle of a chamfer is dependent on the size of beltings.

8.1.4 Panel Size and Position

The fender panel size, position and fender unit orientation, in relation to the berth structure, should be carefully considered by the designer. Failure to undertake this part of the design process may result in reduced energy absorption capabilities of the fender and a reduced operational life of the fender system.

8.1.4.1 Panel Width

The width of the panel should be determined based on the following:

- Sufficient panel width to provide the minimum panel contact area, required to keep the hull contact pressure below the maximum limit permitted by the design vessels.
- Sufficient panel width to accommodate the connection detail of the selected rubber unit and restraint chains arrangement. The arrangement of the chain connection eye pad can sometimes become a dominant factor in determining the overall panel width.
- As far as practical, consider the maximum width readily transported from the manufacturing facility to site.
- Compliance with any project specific requirements for minimum panel width conditions.

8.1.4.2 Panel Height

The height of the panel should be determined based on a combination of the following:

- In combination with the panel width, sufficient height is needed to provide sufficient contact area to keep the hull contact pressure below the maximum limit permitted for the design vessel.
- Consider the full range of vessels likely to use the berth and account for any unique vessel design features.
- Cater for the full tidal range.
- Provide an allowance for variations in water level.
- Accommodate variations in vessel operational draught and trim during (un)loading of cargo.

Based on the above considerations, the level of the top of the fender panel will largely be determined by the minimum draught of an unladen vessel at the mean high water spring (MHWS) , with reference to the top level of the berthing structure. To define the level of the bottom of the fender panel, the minimum freeboard of the design laden vessel, required panel contact area and mean low water spring (MLWS) will apply.

Fender panels can extend above the supporting structure deck level; however, the position of the mooring lines should be checked to ensure that the vessel can be correctly moored and that the mooring lines do not clash with the fender panels and/or do not become snagged. The extension of fender panels above the deck of the berth structure can also cause problems

where vessels with side or quarter ramps are to be accommodated and where the berth is used for other types of cargo vessels.

To accommodate vessels with belting, the fender panel should be high enough to accommodate the full range of the moored vessel movement at the berth. The panel height should be sufficient to minimise the risk of the vessel belting riding over the top of, or being caught underneath, the fender panel whilst the vessel is at berth. An allowance may also need to be given for the effects of wave height at the berth during adverse weather conditions.

The fender panel maximum height should be proportional to the size of the fender rubber unit height and elevation on the supporting structure. For cases that require long fender panels in combination with relatively small sized rubber unit fenders, consideration should be given to installing double or even triple rubber unit fenders arranged one above the other, to provide adequate support and load transfer from the fender panels to the support structure. This means the supporting structure will need to be high enough to allow for placing multiple rubber unit fenders above each other. There is a certain amount of engineering judgement required for a proportional design.

The panel height should be designed considering the relationship between the fender centreline level and its location relative to the panel to keep peak hull pressures below the required limits.

8.1.4.3 Panel Thickness

Fender panel thickness is closely related to the panel internal structure. The designer should consider the stand-off distance between the structure cope line and berthing line when determining the fender panel thickness.

Cone fender systems require a spacer (or mount/ flange) located between the top of the fender (small base) and the rear of the panel or the supporting structure to enable additional fender compression. Sometimes the spacer is also needed for construction (tightening bolts). This will increase the stand-off distance of the fender system.

8.1.4.4 Position of Fender on the Panel

For cone , cell, element or equivalent fender systems, the centre of the fender should be positioned on or above the horizontal centreline of the fender panel and below the upper third of the panel height. Locating the fender below the horizontal centreline of the fender panel should be avoided. The optimum fender panel design is likely to require additional restraint and support chains.

Locating the fender unit too close to the panel edges can lead to rotation of the panel without compressing the fender when loaded. This action induces moments and forces in the fender that may cause damage and reduce the energy absorption capability. Eccentric positioning of the fender relative to the panel centreline will also lead to increases in induced loads on hull structures.

8.2 Chains and Fixings Design

8.2.1 Chain Types and Positions

Chains are used to control the position of the fender panel and have the following functions:

- Prevent excessive panel horizontal and/or vertical movements and/or rotations during berthing
- Assist in supporting the weight of large panels, to prevent dropping/sagging
- Increase fender compression and energy absorption capacity in low-level impact cases
- Counteract horizontal (shear) loads e.g. due to vessel warping

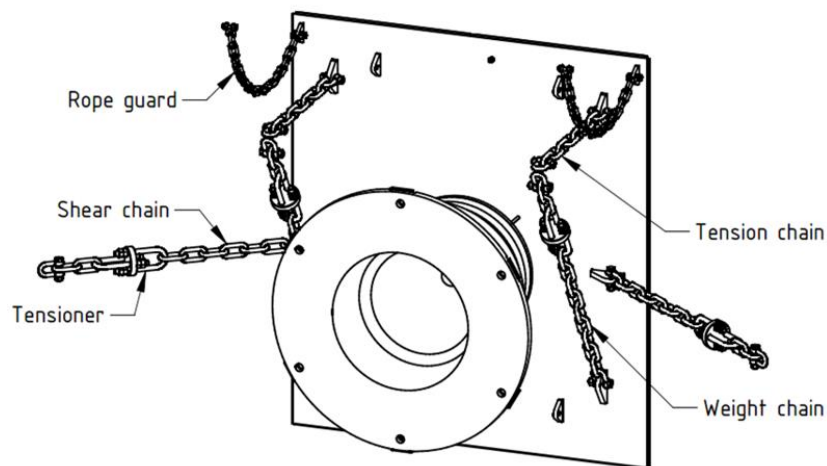


Figure 8.3: Typical fender system chain layout (viewed from rear of fender panel)

The following types of chains are generally used within fender systems, the positions of which are indicated in Figure 8.3.

Weight chains	support the fender system and prevent excessive drooping of the system caused by self-weight forces. They may also resist vertical downwards shear forces caused by vessel movements or changing draught.
Tension chains	Restrict tension forces in on the fender rubber. Correct location can optimise the deflection geometry.
Shear chains	Resist horizontal forces caused during longitudinal approaches or warping operations.
Uplift chains (not indicated in Figure 8-3)	Prevent vertical shear uplift forces in conjunction with the weight chains. These are often specified for exposed offshore berths with large wave induced vessel movements. This type of chain is however typically used in special cases and is not very common. The use of this type of chain should be verified with the fender supplier. If uplift is expected, this chain can be used on side loaded fenders as well to limit lift movement.
Rope guard chains	Prevent mooring lines from getting caught behind fender panels, particularly on panels with no top tension chains.
Keep chains	Used to moor floating or to prevent loss of fixed fenders in the event of accidents.
Supporting chains	Floating fenders (i.e. foam, pneumatic) and cylindrical fenders require supporting chains, see below Section 8.2.2.

Table 8-1: Types of chains

The inclusion of weight and tension chains is always recommended for all fender system installations to support the fender panel and ensure the longevity of the system. The governing criteria for the design of tension chains is often the panel rotation angle resulting from low level vessel impacts or vessel rolling. In the case of larger berthing angles, rotation of the panel about the vertical axis can provide the governing load case on the tension chains.

Typically, shear chains are required on berths where substantial longitudinal shear forces from vessel movements are anticipated. This includes berths where vessels slide along the berthing face during vessel manoeuvres (e.g. ferry or RoRo berths) or where vessels are warped along the berth during loading or discharge operations. The correct position for shear chains is as shown on Figure 8.5 below, attached to the side of the panel. Therefore, when this type of chain is required the design of the concrete structure need to consider required space for shear chain at the required position and angle on the sides of the fender system.

Certain fender system designs and installation locations cannot accommodate shear chains and the berth structure should be sized accordingly to provide connections at the required locations. The designer is recommended to consider how the vessel is expected to interact with the fender system by determining the movement of the panel and chains as a whole system to establish when the chains are in tension, to select the most appropriate combination of restraint chains and if shear chains are required the structure design should provide sufficient space to allow its installation.

The crossing of shear chains should also be carefully considered to ensure that the chains are installed with the required angle, are separated and prevented from rubbing. If crossed shear chains are used, both upper and lower shear chain sets may be required to prevent the rotation of the fender panel.

The provision of uplift, rope guard and keep chains is also closely linked to the design requirements and geometry of the fender system. The requirements for each chain type should therefore take account of specific design considerations.

Chain systems work in combination with the shackles, chain tensioners, connection brackets and anchors, the general position of which is indicated in Figure 8.4.

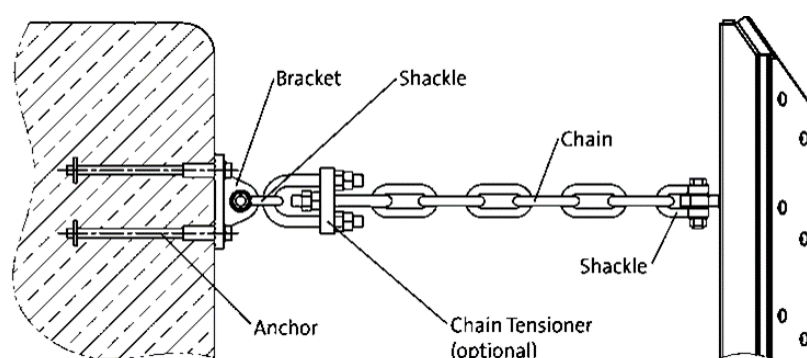


Figure 8.4: Typical tension chain arrangement

Adjustments in chain length can be accommodated with the inclusion of chain tensioners. For tension, weight and shear chains a tensioner can be added to periodically re-tension the chain as needed. It is sometimes not possible to add a tensioner to a tension chain due to the overall chain length being too short. In the case that a tensioner is not possible to be included (e.g. for relatively short tension chains), advise should be taken from the fender supplier on the

tension chain design considering the maximum allowable sag of the tension chain. . Typically, all other chains are long enough to include a chain tensioner, if required.

A 'weak link' in the chain system, typically one shackle, is desirable to prevent damage to the other more costly components. Chains should be hot dip galvanised to protect against corrosion.

On one end chains are attached to the berth structure by brackets or U-anchor connections. On the other end the chains are connected to brackets welded to the internal structure of the panel.

The chain size will be fabricated according to the required minimum breaking load (MBL) of the chain. MBL determines the different dimensions of the chain: diameter, inside link length, inside link width and outside link width, as well as chain link length (typically 4D or 5D will be used) and chain capacity.

The total number of links determines the overall chain length and that is determined by the fender system design and dimensions, including rubber unit size, panel geometry, chain anchor position.

Chains typically work in combination with shackles, which are typically of two types: D-type or Bow-type.

Fender system accessories (chains, anchors, brackets, etc.) should be designed by a suitably qualified design engineer familiar with the design of fender accessories and customised for each specific project application. They should also be designed to be compliant with the specific requirements of the project.

The static angle and the overall length of the chain are used when determining the required chain design load capacity, refer to Figure 8.5. It is important however that the designer assesses the chain tension, and hence load on the fixings, at all stages of the fender system compression, as well as for the static case.

The designer should be aware that the largest chain loads will not necessarily occur when the maximum fender reaction force occurs. The point at which the largest chain loads occur must be determined by the designer as part of the geometric assessment of the fender system compression cycle. The total weight of the fender panel is not applicable when considering the tensile capacity of the shear chains.

The recommended minimum factor of safety applied to the characteristic reaction force ($R_{f,c}$) in chain design should be taken according to local code. If no code exists, a minimum of 3.0 is recommended. So the $WLL \geq R_{f,c}$ and $WLL \leq 33\% \text{ MBL}$ (OCIMF, 2018). These recommendations do also apply for floating fenders (i.e. foam, pneumatic) in which chains are under permanent movement and wear. Such chains can rapidly lose cross sectional profile, regular inspections are recommended to identify any potential areas of excessive wear enabling replacement at an early stage, selecting a higher initial factor of safety can be a choice to reduce the frequency of chain replacement.

As the shackles, chains and tensioners are suffering constant and substantial wear due to the movement and corrosion of the fender system due to the direct exposure to marine environment, the factors of safety applied to shackles, chains and tensioners are different and independent from the safety factor applied to brackets and anchors, which are based on the general structural design codes and standards.

Refer to Chapter 0 for details on chain fabrication. Recommendations on maximum chain angles are shown in Figure 8.5, however should always be checked per individual project, as the chain should be tensioned before the shear capacity of the fender element is reached.

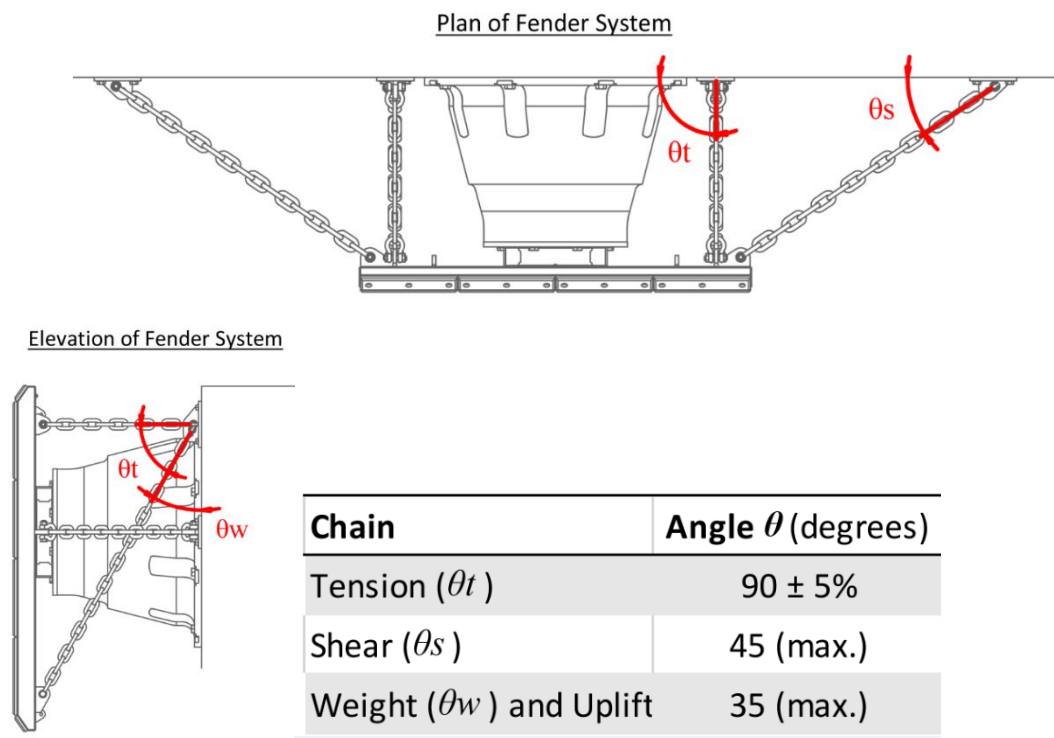


Figure 8.5: Recommendations for angles and chain arrangement for non-compressed fenders

8.2.2 Support Chains for Side Loaded Fenders

Cylindrical, foam and pneumatic fenders use support chains attached to the supporting structure. One end of each chain is attached to the supporting structure, typically via a shackle and bracket. The other end is connected to a swivel connection to allow the fender to rotate, or bar for cylindrical. Different chain layouts can be used depending on whether the fender is fully supported or floating. A 4-chain arrangement can be used to prevent changes in the level of the fender.

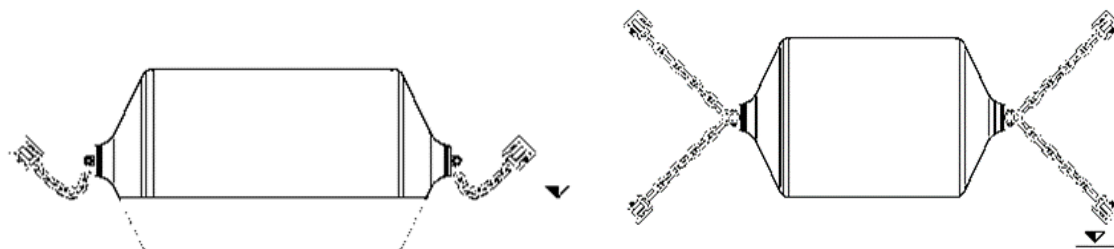


Figure 8.6: Double and 4-chain arrangements for foam and pneumatic fenders

The supporting chains should minimally be designed to accommodate the frictional forces between the vessel and the fender and between the fender and the supporting structure, the potential uplift forces, the self-weight of the fender and to fit within the general arrangement of the supporting structure.

8.2.3 Bracket Design

Brackets are typically required to connect the chain assembly to the support structure and can be designed to suit new or existing structures. The design should allow the chain to freely rotate through its full arc and the position and geometry should not interfere with other brackets, the fender panel or the rubber fender body during compression.

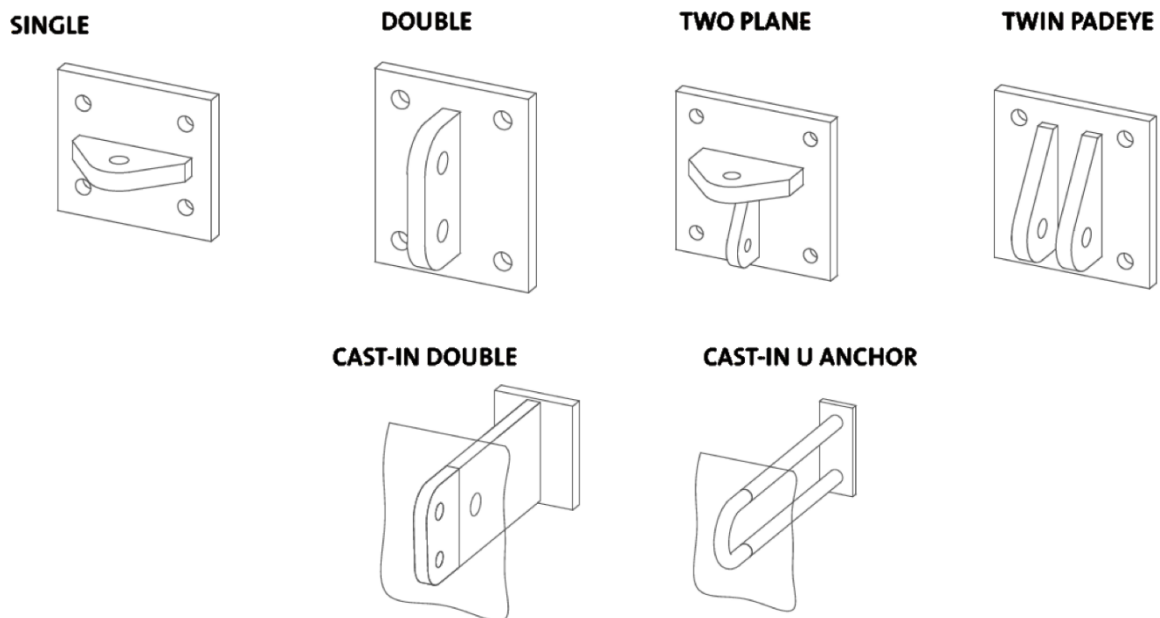


Figure 8.7: Typical examples of fender system bracket assemblies

The brackets typically experience large, concentrated forces transferred from the connected chain assemblies. The brackets should be sufficiently thick or include spacer plates to support the correct size and type of shackle. Induced localised stresses in and around the brackets should be carefully checked and stiffeners provided to resist these stresses.

The weld size holding the bracket to the base plate is critical and should be considered carefully in conjunction with the manufacture of the brackets. The design of the brackets should therefore comply with the relevant codes and standards for steel fabrication and weld design.

The position and orientation of the brackets on the fender panel and support structure should be assessed to ensure that the chains are always straight and that the force induced in the bracket acts along the long axis of the bracket.

8.2.4 Fixing Anchor Design

Fixing anchors are required to attach fender systems, including the bracket and chain assemblies to the supporting structure. The design of the fixing anchors should determine the anchor diameter, required embedment or bond length, type and material.

Fender anchors are subject to both shear and to a lesser extent, tension forces. Bracket and chain assembly anchors are subject to both tension and shear forces. The proportion of each of these forces is dependent on the angle of the chain and the geometry of the bracket. These forces should be determined based on the characteristic load of the selected chain assembly multiplied by the appropriate factor of safety and by assessing each specific load case and design load chain arrangement. Cast-in U-anchors should consider an additional thickness allowance to accommodate for wear due to chain shackle friction.

Guidance on the required diameter and minimum embedment of the anchors for each fender type are often provided by the fender supplier. However, the design of the supporting structure is the responsibility of the structural designer, and they should check that the proposed anchors comply with the appropriate standards. As such, the designer of the supporting structure is responsible for the design of the anchors.

Anchor capacity is dependent on the concrete strength, the bonded length of the anchor, the size of the bracket bearing plate and the concrete reinforcement details. For all anchor applications into concrete (individual and groups of anchors), the designer should consider the potential modes of anchor failure, including pull-out, concrete splitting, crushing and cone breakout, all of which can affect the anchor design and performance.

The minimum edge distances should be assessed which should also be incorporated within the design of the support structure. In the absence of specific design information, the designer should consider a minimum edge distance of 5 times the anchor diameter (a value of 8 to 10 times the diameter is preferable) or the overall length of the anchor, whichever is the longer. However, the required minimum edge distance could be reduced due to the presence of reinforcement within the concrete. The design and integration of anchors into the concrete supporting structure is recommended to be carried out in conjunction with the design of the supporting structure.

Cast-in anchors sockets (into which anchor bolts can be secured) are recommended to be specified for all new concrete structures. Drilling and chemical resin anchor fixing should only be considered for existing concrete structures. In such circumstances, the designer may need to assess the residual concrete strength and integrity to determine the required anchor diameter, drill hole size and bonded length. The use of threaded bars with chemical anchors should be used with caution if used for anchoring fenders to support structures as (opposed to brackets) the protruding threaded bar can impact upon the body of the fender when fully compressed, potentially causing damage to the fender.

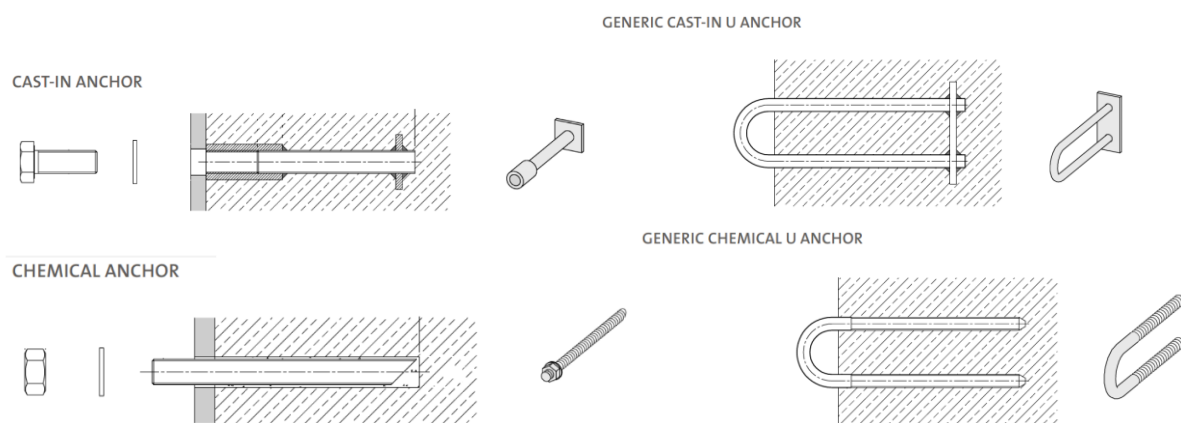


Figure 8.8: Typical examples of fender system anchors

The factors of safety applied to brackets and anchors are based on the general structural design codes and standards used for the supporting structure which typically are 1.5 (according to Eurocode) or 1.67 (according to ASTM) over the maximum chain force characteristic value.

All fixings should be protected against corrosion using stainless steel. If galvanised anchors are utilised, designers should be aware of the potential for the galvanizing to wear off over time, e.g. U bolt anchors, potentially necessitating the need for replacement.

8.3 Low Friction Facing Design

Friction has a large influence on the design of a fender system, particularly for the restraint chains. Low friction facing materials are often used on fender panels to reduce friction. Vessels moving longitudinally or vertically on a berth induce friction forces at the contact surface between the fender system and the vessel hull.

These forces induce shear deformations in the fender and should be kept within the fender design limits. Large shear deflections should be limited by chains connecting appropriate parts of the fender to the supporting structure. Shear forces should be calculated using the relevant coefficient of friction, multiplied by the design reaction force at the fender face.

Facings can be manufactured from several different materials including Ultra-High Molecular Weight Polyethylene (UHMW-PE), steel, timber and rubber. The use of UHMW-PE on fender panels is common due to its inherently low coefficient of friction. Other materials, such as polyurethanes that are typically used for the skin of foam fenders, also have lower friction coefficients than rubber against steel or concrete.

The designer should select the correct material considering the vessel hull material (e.g. steel), the vessel geometry, potential berthing impact and wear allowance, vessel movement on the berth and frequency of use. The low friction facing should be durable, replaceable and be designed to prevent damage to the vessel hull and paintwork.

The coefficient of friction for each facing material varies considerably. In the absence of specific design information, the coefficients of friction provided in Table 8-2 may be used. A higher design value is recommended to be adopted in the design of other fender components and the supporting structure to account for factors such as surface roughness, temperature, wet or dry conditions and the contact pressure which can affect the friction coefficient. A lower friction coefficient can also be considered in the case of dynamic mooring analyses, where less fender friction can result in the larger moored vessel motions and mooring line loads.

Materials (fender system facing against vessel hull)		Friction Coefficient	
		Typical Range	Design
Ultra-high molecular weight polyethylene (UHMW-PE) (wet)	Steel (wet)	0.1 - 0.15	≥ 0.3
Ultra-high molecular weight polyethylene (UHMW-PE) (dry)	Steel (dry)	0.15 - 0.2	≥ 0.3
High Density Polyethylene (HDPE)	Steel	0.2 - 0.25	≥ 0.3
Polyurethane	Steel	0.3 - 0.4	≥ 0.4
Steel	Steel	0.3 - 0.5	≥ 0.5
Timber	Steel	0.3 - 0.6	≥ 0.6
Concrete	Steel	0.4 - 0.65	≥ 0.7
Rubber	Steel	0.6 - 0.7	≥ 0.7

Table 8-2: Typical design values of coefficients of friction

8.3.1 Low Friction Resin Facings

One of the more commonly adopted facing systems is low friction facing panels, manufactured using UHMW-PE. Large UHMW-PE sheets are cut to size, drilled and chamfered to create individual sections which are fixed to the fender panel. These can be attached to steel fender panels using welded studs, bolts or low-profile fixings.

The design of the low friction facing should take into consideration the type and size of fixing and include a wear allowance. The following should be considered for low friction facing design:

- The minimum UHMW-PE thickness should consider the wear allowance, the nut thickness and the countersunk UHMW-PE residual thickness to stop the bolt from pulling through the UHMW-PE sheet. The thickness requirements should be assessed on a case-by-case basis accounting for the level of expected use.
- Low friction facings should be designed to facilitate easy replacement.
- If access to enable facing replacement is difficult, then an extra wear allowance may be useful to reduce maintenance and replacement intervals.
- Vessels with belting can be particularly damaging to low friction facing panels and may not be the optimum solution in such cases.
- Typically, the coefficient of friction between a vessel and the low friction facing is between 0.15 and 0.2. However, for the purposes of design of other fender components and the supporting structure, it is not recommended to use a coefficient of friction less than 0.3 to account for contact pressure, facing deterioration and temperature variations and to allow for wear during the design life.

Typically UHMW-PE- facings are supplied as virgin and regenerated material. Virgin is made from new resin material and has the best durability properties (higher wear resistance and better charpy impact strength). Regenerated material is typically made from re-used production material or can be made from recycled material and is a more cost efficient and potentially a more sustainable solution (see Section 12.6.5). For fenders where durability is a key consideration (i.e. frequently used ferry berths, belted vessels, remote locations, etc.) virgin material is best considered. For fenders under more mild operational conditions regenerated material can be considered as alternative to virgin. See section 0 for suggested material properties for both virgin and regenerated.

8.3.2 Steel Facing

Where a high frequency of berthing takes place, such as RoRo ferry ports, steel can be adopted as a robust and durable fender facing. Higher friction forces can occur between the vessel's hull and the fender face, which can be beneficial when vessels are berthing in reducing vessel approach velocity. However, these forces should be accommodated within the fender system design and supporting structure(s).

Consideration should be given to the potential effects of bimetallic corrosion when using a steel facing in combination with the fender system. Consideration should also be given to the possibility of sparks arising from steel-to-steel contact when designing crude oil, chemical products and gas terminals with steel facings.

8.3.3 Fastenings

Low friction facing pads are attached in various ways according to the type and structure of the fender panel. Studs or bolts with blind nuts are commonly used for closed box panels. Standard nuts are used for open panels and structures. Flange nut fixings can provide a greater wear allowance.

Larger washers, typically 2.2 and 2.5 times the bolt diameter, are required to spread the loads due to friction and prevent pull through. The minimum thickness of the low friction facing under the head of the washer is recommended to be 30% of the overall low friction pad thickness.

To provide greater wear allowance, the heads of the fastenings should be set as low as possible from the surface of the facing. The top face of the studs or bolts should be minimum 10 mm to 20 mm below the surface of the facing. Typical fixing arrangement is shown in Figure 8.8.

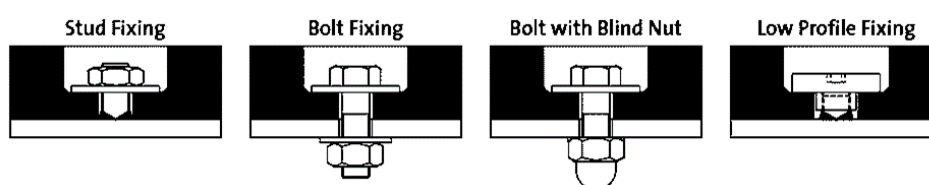


Figure 8.9: Typical facing connection details

When fender panels are fabricated as closed box structures, the bolts should be secured with a watertight threaded blind boss. If the threaded stainless steel bolt shaft to the blind boss fixture is used, anti-galling compounds should be used. To prevent contact with dissimilar metals, proprietary PTFE washers can also be considered.

Extreme shear forces or protrusions in the vessel hull have the potential to displace the UHMW-PE pads. For those cases, adding a shear key in the panel design will help to prevent this problem.

8.4 Parallel Motion Fender Systems

A parallel motion fender system (as illustrated in Section 2.4.4) is a specialist fender system. The selection and design of such fender systems should be carried out in consultation with fender manufacturers experienced in their design and manufacture and managed by an experienced port engineer.

Parallel motion fender systems are typically considered for selection when the supporting structure is load sensitive. They can reduce berthing reaction forces by up to 60 %, when compared with conventional buckling fender system designs. However, a wide range of factors must also be considered in the fender selection process; refer to Section 6.

The parallel motion fender system geometry and function ensures that the panel always remains vertical, enabling a range of different vessels and water levels to be accommodated. A vertical berthing face also avoids the potential risk of double contact of the fender panel with the vessel hull. Parallel motion fender systems can also accommodate large berthing approach angles, with minimal loss in the energy absorption capacity.

The energy absorption capability of a parallel motion fender system can be increased by utilising conical back-to-back fenders. This results in an increase in the available compression

distance and energy absorption, whilst also reducing the corresponding reaction forces. A reduction in the imposed reaction forces can result in lower forces needing to be accommodated by the supporting structure.

The design of a parallel motion fender system should consider the effects of fatigue on the various structural components resulting from the continuous and repetitive movement of the bespoke system. Depending on the frequency of vessel berthing, fatigue can be a particular issue for parallel motion fender systems and a detailed assessment should be undertaken as part of the design process.

As parallel motion fenders can be primarily pre-assembled at a fabrication facility, installation on site can be simple and quick. However, the designer will need to consider provisions for the installation requirements and the future maintenance of the often heavy parallel motion system, i.e. crane access for installation and removal of components for inspection.

A parallel motion fender system typically consists of the following components; however, this may vary depending on the project requirements.

- Rubber fender units – can also be mounted in pairs in a back-to-back configuration.
- Closed box panel (frame) – typically fully sealed and pressure tested and including chamfered or bevelled edges to prevent snagging of mooring lines or vessel protrusions.
- Torsion tube and arm assembly – also typically consisting of a closed-box construction. The tube and arms keep the panel vertical at whatever level the berthing load is applied.
- Hinge units – these typically consist of stainless-steel pins and bearings allowing free rotation to accommodate berthing angles and allowing compression of the fender system.
- Low friction facing – typically provided to reduce large friction forces being induced into the system.

The design may also need to consider the provision of 'check chains' to act as rope deflectors to prevent ropes from becoming snagged around the parallel motion system.

8.5 Fender Interfaces with Mooring Lines

When not considered, mooring lines can damage fender systems in various ways and vice versa. The following principles should be considered by the designer when designing fender systems:

1. A line should never get stuck under a fender. If that risk is unavoidable during berthing, a quick release hook should be used instead of a bollard. The hook can then be opened to prevent damage to the fender. This especially can occur with sinking steel wires or large tidal ranges.
2. During vessel departure, lines that are released should not be caught or snagged by the fender panel or restraint chains. There is the increased risk of damage occurring to the fender system if this happens.
3. Mooring lines should not be able to get stuck between the fender panel and the vessel hull. This contact will damage the mooring line and can lead to mooring line failure.

The designer should carry out geometric investigative studies to model the position of mooring lines in relation to fender positions at all tidal heights and vessel ballast conditions.

8.6 Whole-Life Considerations

Whole-life consideration in the design of fender systems is very important. Designers and operators should consider the whole life of a fender system prior to committing to the chosen system.

By its very nature, any fender system will sustain impact from vessels, and thus the general philosophy of fender system design is to ensure that the fender system is good for purpose, robust and can be repaired or replaced easily.

A reasonable life expectancy of a fender system is defined in Section 4.4 if planned maintenance, in accordance with the supplier's requirements, is undertaken. For more information about maintenance refer to Section 11.3.

Fender system components design should consider, where possible, advances in vessel design. In recent years, advances in naval architecture have produced various hull shapes that differ from the classic vessel shape.

The provision of spares for fender systems should be considered when procuring new or replacement fender systems, to enable quick replacement in the event of damage or deterioration. For more information about spares for fender systems, refer to Section 0.

8.7 Corrosion of Fender Components

All metals suffer from some form of corrosion in a marine environment. This is mainly due to the following:

- Formation of galvanic cells
- Microbial action inducing galvanic cells
- Erosion of products of corrosion
- Inadequate preventative maintenance.

Consideration should be given to the influence of corrosion on the design of fenders and their accessories. Unprotected steel will begin to reduce in thickness immediately after it is installed. The onset of corrosion may be delayed by an appropriate paint system, suitably maintained. A sacrificial steel thickness can also be included within the design to accommodate expected rate of corrosion.

Corrosion rates will vary according to local conditions and the position of the fender relative to the inter-tidal zone. Corrosion rates may also be significantly higher in hotter climates.

The effect of corrosion on fender integrity and the inclusion of safety factors will also depend upon whether the steel is exposed on both faces or just one face.

In the case of restraint chains and bolts, loss of diameter affects the sectional area with very rapid loss of strength capacity occurring once corrosion begins. This type of corrosion often requires periodic inspection and replacement, which is often considered more economical, rather than using excessively large chain link sizes to maintain minimum safety factors throughout the full fender system life.

Corrosion protection methods are different for each fender component, the options are as follows:

- Steel fender panel - coating system
- Chains - Hot dip galvanization
- Anchors and hardware (bolts, washers, nuts) - Stainless steel or hot dip galvanization

8.8 Marine Growth

Marine growth can hide defects or even cause maintenance issues. In areas prone to heavy marine growth and strong currents or tides, marine growth can increase drag forces or substantially increase the weight and or prevent rotation of the fender system. Inspection and removal of marine growth should be part of a maintenance program.

Special attention should be given to reducing the number of bolts below water as far as possible since when covered by marine growth they could be difficult to find and replaced if necessary.

8.9 Design of Fender Components in Icy Conditions

The potential for ice accumulation and the effects of ice on the berthing and mooring of vessels should be considered when designing new or replacement fender systems.

In regions prone to the formation of ice, there is the potential for fenders to become frozen within the ice, and for any water trapped in the rubber fender to become frozen. Subsequent changes in water level can add additional loads to the fender system, including the supporting chains and anchors. Special consideration should be given to floating fenders which have a fixed lower position in relation to the tide.

The accumulation of ice on and around the fender system can also increase the 'self-weight' of the fender, leading to additional vertical forces that must be accommodated by the fender and support chains.

Strong currents and/or wind combined with ice formation presents a significant issue for fender design in the form of drifting ice. The fender panel can be exposed to excessive wear and tear. The low friction facing panels may need more frequent replacement to maintain the design coefficient of friction.

The longitudinal forces from the ice can be significant and greater than the potential longitudinal friction forces from a berthing vessel. These forces need to be considered in the design of the fender system and subsequently the supporting structure.

Drifting ice flows can also induce large forces on the fender system fixings, which can result in the breaking of chains and, in worst cases, the removal of the fender from the support structure. The risk of damage caused by drifting ice is increased with the use of larger fender panels.

In areas subject to drifting ice, a closed box panel is recommended to reduce the possibility of ice accumulation. Consideration should be given to the paint system applied to the steel panel as it should be suitable for use in ice conditions.

9 MANUFACTURING OF FENDER SYSTEMS

Fender systems typically include the following components:

- Rubber (or foam) element to absorb energy.
- Steel frontal panel to distribute loads and hull pressure (if applicable).
- Low friction facing pads to reduce friction and abrasion effects.
- Accessories like chains, required to improve the performance of the system working under forces coming from different sources and directions; and
- Anchors, to fix the fenders and the chain brackets and transfer the loads into the structure.

This chapter covers the general manufacturing process of a fender system as a whole which will give designers basic understanding of each element.

9.1 Manufacturer Qualifications

Manufacturers shall demonstrate an acceptable level of experience with materials in the manufacture of a marine fenders and associated accessories of a fendering system. Manufacturers shall possess adequate test data of all materials when requested and to reliably be able to predict and verify product performance within the allowable tolerances.

Published performance in manufacturers catalogues shall include capabilities for fender based on a proven track record of performance testing that they have carried out on their fenders. This shall include, but not limited to: energy absorption, reaction force, Hull pressure, material properties, durability and longevity of materials

9.1.1 Quality Control by Manufacturer

Manufacturer shall have an Inspection Test Plan (ITP) in place to ensure that fenders are produced within reasonable, and industry accepted limits.

The following verification shall be included as part of the ITP and must be carried out on every project to make sure they are manufactured with an acceptable quality. Within the ITP the following items shall be addressed as a minimum requirement:

- Materials
- Manufacturing
- Process control
- Dimensional control
- Finishing
- Final inspection
- Testing

Manufacturers must provide test results or certificates, or any additional information related to the fender production when requested (Refer to Chapter 10: Testing of fenders).

9.1.2 Workforce Qualification

The fabrication unit must be ISO 9001 certified. The workmanship must be of the highest quality during all phases of the work, skilled workforce shall be deployed and must be qualified enough to tackle relevant standards and type of work. The welding procedures must be conducted as per relevant standards and only qualified welders must be deployed on work.

The fabricator should adopt good practices and procedures to minimise the repair works and proper standard operating procedures (SOPs) should be followed. Only calibrated measuring instruments must be used, and certificates must be readily available for verification purposes.

9.1.3 Storage of Produced Fender Elements

The final product should be stored in a shaded area and must be covered using packing material to avoid any sort of damage. The storage area must be free of hazards and unwanted vehicle movements. All threaded parts should be protected in order to avoid thread damage while handling.

9.2 Rubber Fender Compound

Rubber fender compounds consist of a set of ingredients which form the compound composition of the material. In this section, effects and influence of each key mixture which forms the compound composition is provided. Manufacturers may choose to adjust their mixture as they see fit and add any other necessary ingredients which may enhance the fender performance and/or durability.

The addition of various ingredients to the polymer (natural or/and synthetic rubber) in order to impart desirable properties is known as rubber compounding. Typical ingredients include reinforcing fillers, antioxidants and antiozonants, process aids, vulcanising agents and many special additives in a small quantity. The ratio of these ingredients varies between fender manufacturers, and it depends on the fender size, shape and required performance. Therefore, it is important for a designer to understand the basics of fender manufacturing and use of materials for manufacturing.

The energy absorption and reaction force characteristics of a marine rubber fender is governed by the geometry, size and hardness grade.

Elastic polymers (natural and synthetic rubber), to which various ingredients are added to create a mixture known as a rubber compound, they set the basic boundaries for properties of the rubber fenders.

After vulcanisation, rubber compounds become elastic, they also dissipate energy because of their viscoelastic nature. Their strength is high even when the fender is under shear and compressive deformations.

Manufacturers are continuously innovating their compounds by introducing new materials in their compounds for better performance, processability and durability.

The following sub-sections describes key ingredients of rubber compound used in the manufacturing of rubber fenders.

9.2.1 Natural Rubber

Raw natural rubber (NR) is found in the extract of many plants (shrubs, vines and trees), the principal of which is the *Hevea Brasiliensis* tree, native to Brazil. Typically, natural rubber is cultivated in an area of 15° North and South of the equator, with Southeast Asia being the main producer worldwide. After the latex is processed, natural rubber becomes an elastomer with excellent mechanical properties. The weak points are weather resistance and ageing compared to some synthetic rubbers. NR builds up heat slower than synthetic rubber.

Compounds with natural rubber have a typical service temperature range between -55°C and 70°C.

9.2.2 Synthetic Rubber

Synthetic rubber is any artificial elastomer which are polymers synthesised from petroleum by products. Synthetic rubber commonly used by manufactures are styrene-butadiene rubber and butadiene rubber.

Synthetic rubber is frequently used for rubber compounds. Styrene-butadiene rubber (SBR) is a type of synthetic rubber commonly used in the fender industry. SBR has an ageing resistance similar to NR and tear resistance (little less than NR) however, general mechanical properties are not as good as natural rubber. SBR has a typical service temperature range between -45°C and 100°C. Abrasion resistance is much higher than NR.

Manufacturers can also use Ethylene Propylene Diene Monomer (EPDM), an alternative type of synthetic rubber. This elastomer is characterised by a wide range of high-temperature applications. EPDM rubber has much better resistance to ageing, weathering, ozone, oxygen and many chemicals. The weak points are poor adhesiveness and tear resistance.

9.2.3 Recycled Rubber

Recycled rubber used in the fender industry consists mainly of crumbed and reclaimed (devulcanized) rubber. Crumb rubber is produced by grinding waste rubber using standard grinding, cryogenic grinding (typically used) and wet grinding processes.

The manufacturing of reclaimed rubber is an aggressive, energy-intensive process in which rubber powder is cooked under pressure with certain corrosive chemicals. This process breaks the long rubber molecules into shorter lengths resulting in a soft, rubber-like material.

Designers should understand that recycled materials do not have the same mechanical properties as virgin rubber. Ground rubber cannot be considered more than a filler. Reclaimed rubber may potentially be used to replace virgin rubber material in the future. The use of recycled/reclaimed material at present (2023) is possible however requires specialist knowledge not yet present everywhere in the fender industry. Unskilled manufacturing with recycled rubber will result in the production of low-quality fenders.

Recycled rubber, incorrectly blended with virgin rubber, has various performance disadvantages. These disadvantages can include higher hysteresis and heat build-up, poorer flex and weather resistance, greater risk of cracking and a poor compression set as well as a reduced durability and life cycle of the fender.

There is no straightforward way to identify the quantity and quality of the recycled rubber used in a rubber compound. Recycled rubber in a rubber compound usually reduces the physical properties refer to (Akiyama H. S., 2017), (Akiyama H. , Study on aging of circular hollow rubber fender in long term usage. Doctoral Thesis (in Japanese), 2018), (Shimizu, Yasui, & Yamase, 2015) especially after ageing.

9.2.4 Fillers

Fillers are used as reinforcement to improve properties of all rubbers. They are used not only to enhance the physical properties of rubber compounds, but to improve the processing properties and to impart UV resistance. The level of reinforcement of fillers is defined by particle

size, structure and surface activity. The smaller the particle size, the higher surface area is achieved which potentially makes the fender much harder.

The most common filler in the fender industry is carbon black, which strengthens the compound (needed for SBR) and makes it UV resistant, however there are also other active and non-active mineral fillers, that are used to produce marine rubber fenders. The activity grade of mineral fillers is determined by the surface area (the higher, the more active the filler).

For coloured compounds, white active fillers like silica are used in the mix for white or navy grey compounds. These fenders shall comply with the same rigorous testing standards as black fenders.

9.2.5 Anti-Ageing Agents

Antioxidants help to protect the compound from thermal ageing while in use and while the compound is being mixed and processed. These ingredients absorb free radicals that can break the polymers' bonds and reduce the service life of a fender.

The most common antiozonant is wax, used to bloom out on the surface after moulding to protect the rubber from ozone attack.

9.2.6 Oil (Processing Aids)

Oil is included in a formula to help incorporate all the dry ingredients with the rubber blend while mixing in a rubber mixer. They also help to reduce the viscosity of the overall compound to help with the moulding process.

It is important to understand that fillers and oil ratio controls the stiffness and hardness of the rubber compounds which influences the different energy absorption grades of rubber fenders.

9.2.7 Accelerators and Sulphur

An accelerator in the rubber industry is a substance that causes vulcanisation of rubber to occur more rapidly at lower temperatures. Many classes of compounds act as accelerators, the most important being organic materials containing sulphur and nitrogen.

Sulphur-based chemicals are used for rubber vulcanisation. Mineral sulphur is a widely used ingredient to form cross-links between the rubber chains in the vulcanisation process.

Vulcanisation of rubber involves heating it in the presence of sulphur. This results in cross-linking of the chains of rubber and sulphur to form a stable polymer which is stronger and more stable.

9.2.8 Other Ingredients

Many other ingredients including plasticisers are used in a small quantity to impart different properties, colour and facilitate the mixing and manufacturing processes. These ingredients often influence the rubber fender properties and sometimes are used to drive the cost of manufacturing down.

9.3 Manufacturing Process or Rubber Fenders

The manufacturing process of a marine fender starts with a rubber formulation, in which ingredients suitable for the intended application and its percentage expressed in PHR (parts per hundred parts of rubber) are selected.

The final product is produced and supplied following the various steps described below which follow after the compounding process (mixing of ingredients with polymer):

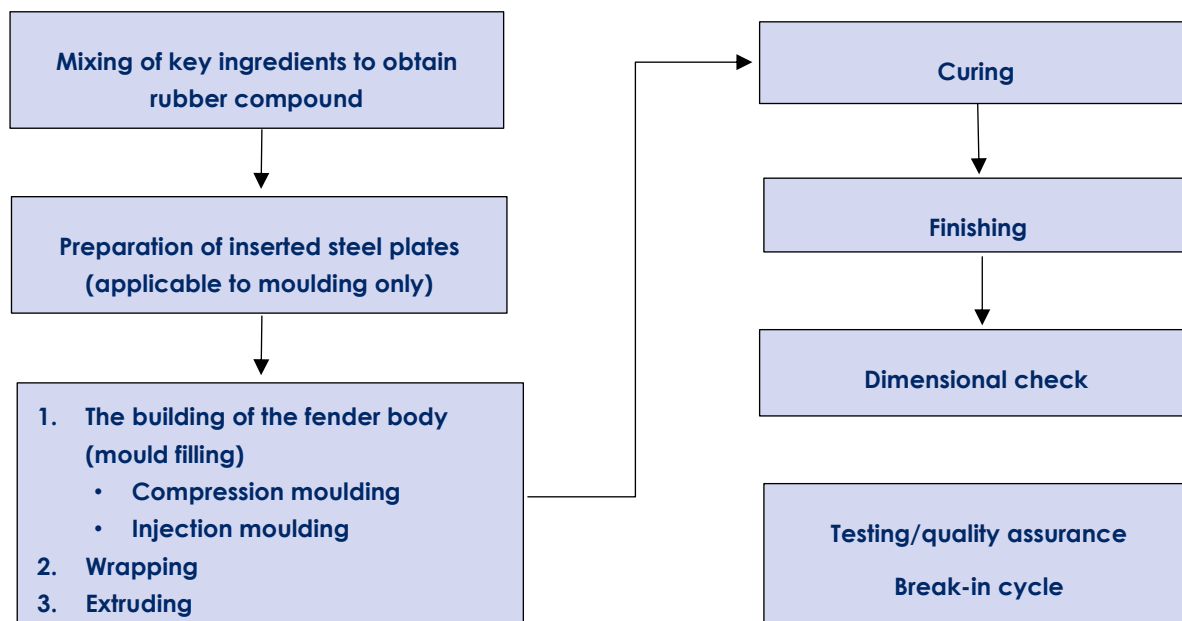


Figure 9.1: Manufacturing of rubber fender

9.3.1 Mixing of the Compound

The mixing of ingredients is done in several stages in internal mixers and or mills. Special attention must be given to the sequence of ingredient addition and the subsequent mixing times.

In a first stage, masticated natural rubber (NR) is mixed with synthetic rubbers to create a uniform rubber blend. In a second stage, reinforcing fillers (carbon black and others) and other ingredients are added to the rubber blend for the next mixing sequence. Filler dispersion commonly measured by a carbon black dispersion in the final compound has a large impact on the quality and fender performance. A high dispersion rating is preferred for a good rubber compound. Poor dispersion can result in reduction of fatigue life, poor performance and product appearance. On the other hand, over-mixing at high temperature degrades the rubber quality by destroying polymer chains. Manufacturers sometimes combine several ingredients to create optimum performance.

Compounds with different elastic modulus or stiffness values (slope of stress vs. strain curve) are used to manufacture fenders with different energy absorption grades. Soft grade fenders need compounds with lower modulus compounds, whereas hard grade fenders need high modulus compounds.

9.3.2 Moulding, Extruding and Wrapping

Rubber fenders are produced by a process called vulcanisation using heat and pressure. The main difference in this production process is the method of building of fender body.

Traditionally, steel moulds are used and filled manually with un-vulcanised rubber sheets or blocks. This is still practiced today by some fender and manufacturers. The advantage of this method is that the different materials can be included into the mould, such as locating the anti-ageing material at the surface of product and high modulus material at the centre of

body. The disadvantage is that some manufacturers could end up using a high-quality rubber on the fender surface just to demonstrate compliance with their TGA (thermogravimetric analysis, refer to Section 10.6.2) testing and various other tests however could potentially end up with lower quality rubber on the inside with the intention to reduce cost. After filling the moulds, the final shape is completed using a compression press at a high temperature or by putting the moulds into autoclave (heat oven with pressure) for vulcanisation.

The other method is high pressure injection moulding, where soft rubber is pumped into the mould. The advantage of this method is that the rubber is already heated when it is injected in moulds therefore the curing time can be significantly reduced, however manufacturers cannot utilise different quality rubber compound inside the fender due to the manufacturing method. This method however requires high skill labour as there are stringent control in temperature requirements, pressure and proper pumping speed to ensure a good quality product is produced. The mould filled with injected rubber is compressed by a heating press or by putting into an autoclave or heated by itself with hot steam passage or electric heater equipped located inside the mould casing itself which are called jacket moulds.

Fenders such as D-fenders and other small longitudinal fenders, are typically produced by extrusion. Rubber compound is fed into an extruder, after heating up and being softened, then pushed by an internal screw through the mouthpiece of the extruder, the die, which forms the section profile in the corresponding shape. This is followed by the vulcanising process in an autoclave. Extruding is an effective production method which does not require a steel mould. Generally extruded products do not have steel plates embedded in their flanges.

Straight axisymmetric shapes can be made by a wrapping method. Mandrel wrapping is the process by which uncured rubber sheets are wrapped on a pipe mandrel until the required outer diameter and desired shape is reached. Cylindrical fenders and the main body of cell fender are often produced using the wrapping process.

The combination of above methods is also possible. For instance, wrapping the hot extruded rubber around mandrel is a good improvement to reduce vulcanisation time. Thus, note that manufacturers are always trying to improve the quality and productivity, so manufacturing method is changing over time.

9.3.3 Curing/Vulcanising of the Rubber Element

Curing/vulcanisation is a process that is virtually connected to the forming process as a final step of rubber fender manufacturing and influences the properties of the rubber fender. For vulcanisation, sulphur is the most common curative for rubber compounds. Rubber polymers are just entangled hydrocarbon chains that will not hold shape and external forces. Rubber polymer chains are bonded together side by side by heating rubber with sulphur under pressure, called vulcanisation, to form sulphur bridges between molecules. Typically, autoclave or moulds are used for curing purposes.

The rate of reaction between rubber and sulphur is a slow process and commercially unviable. Therefore, accelerators are used in a formulation to increase the speed of the curing reaction.

9.4 Fabrication of Steel Fender Panels

Steel Panels are critical to the performance of most fender systems. The following key points summarise the purpose of the steel panel, and should be considered when designing, specifying, and manufacturing steel fender panels:

- Withstand the load combination transmitted by the vessels during the berthing approach and mooring;
- Provide a suitable supporting surface, located at the correct levels, considering tide variations, weather variations, operational variations, vessel dimensions, load conditions and fender level, able to efficiently transmit the loads into the deflecting rubber unit, allowing vessel energy absorption;
- Distribute the reaction force into the hull of the vessel, below the maximum allowed pressure limit;
- Protect the rubber fender from damage due to abrasion or tearing
- Give a service life of a minimum of 20 years assuming adequate and planned maintenance, provided there is no damage caused by accidental situations.

9.4.1 Panel Internal Structure

A typical fender panel cross-section includes several vertical and horizontal stiffeners, usually U-profiles or T-profiles fabricated from steel plates. The external plate thickness, size and type of stiffeners will depend on many factors.

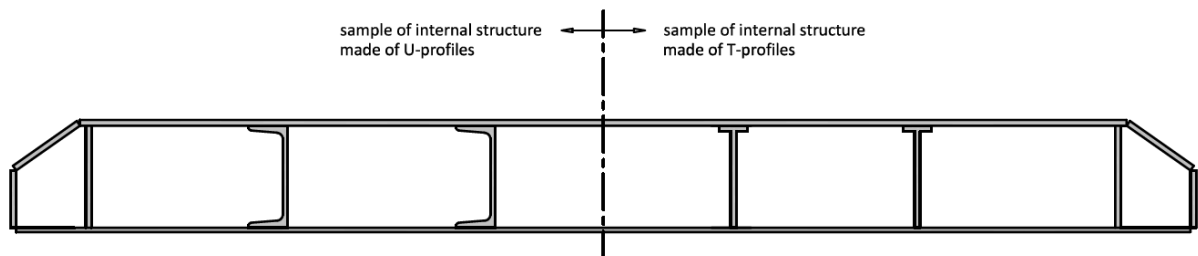


Figure 9.2: Typical fender panel cross section samples (U profile on left, T-profiles on right side)

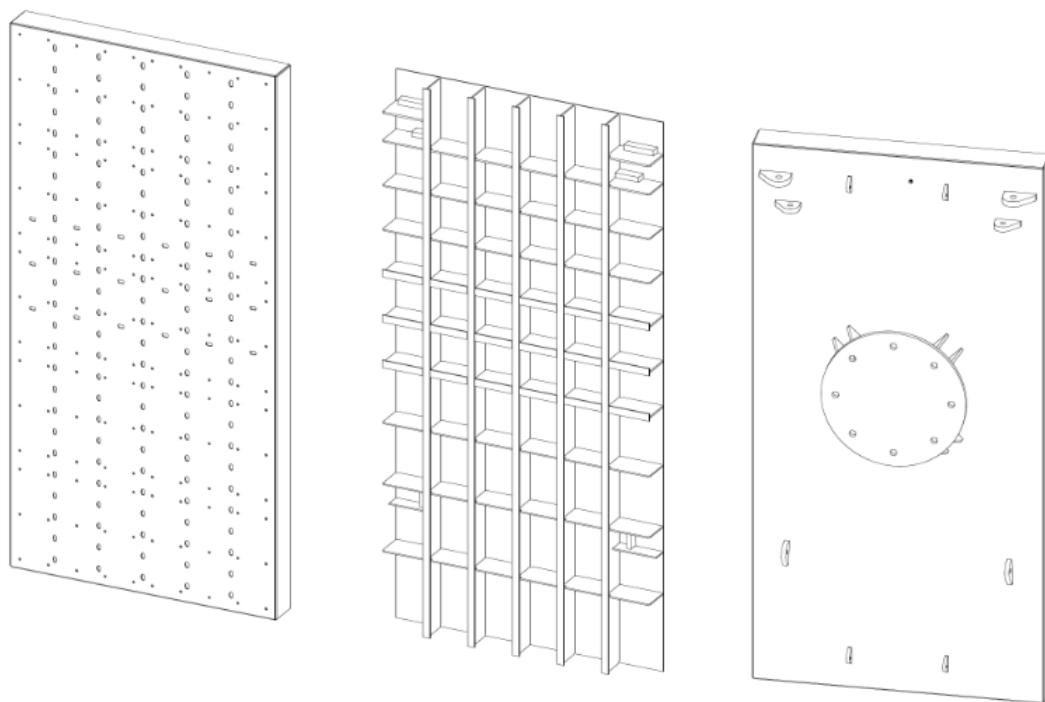


Figure 9.3: Typical fender frontal, back and internal panel structure views

There are many demands on the fender panel which causes bending, shear, torsion, crushing and fatigue. Section 8.1 of these guidelines provides a good insight into panel structural design. Panel structural design should be carried out by manufacturer/suppliers' team of experts in advanced structural analysis in compliance with international, national and local design codes, in order to justify the design of the external plate number, thicknesses, size, type of stiffeners and welding connections.

In general steel panel structures are closed-box type construction, which has a high strength to weight ratio and creates a simple exterior shape that is easier to paint and maintain. The inside of the panel is protected against corrosion since it is fully sealed, and pressure tested.

9.4.2 Material

Fender panels are made from weldable structural steel. The grade should be selected depending on design, local conditions, and availability.

The steel used shall be structural and in accordance with the relevant standards and shall be free of scale, blisters, laminations, or any sort of defects.

All the welding work must be in accordance with the approved specifications, drawings and standards.

9.4.3 Protection Against Corrosion for Steel Panels

All metals suffer more from corrosion in a marine environment than onshore. This is mainly due to the formation of galvanic cells within the metals of the structures acting as anodes and cathodes and the solution of salts in seawater acting as the electrolyte or microbial action inducing galvanic cells.

Corrosion rates will vary according to local conditions and the position of the fender in the intertidal zone and may be significantly higher in hotter climates.

9.4.3.1 Paint Coatings

Coating systems are the main solutions for corrosion protection of steel panels. Typically, ISO 12944 should be followed as a minimum standard for the paint system selection and generally system Im2 as per table 2 of the ISO 12944 -2 applies for fenders since this system is recommended for immersed structures without cathodic protection. However, on some occasions other systems might be more applicable as per table 1 of the ISO 12944 or Im4 in case of cathodic protection. It is recommended for the designer to carefully consider the ISO 12944 in those cases.

For the longest service life in seawater, splash zones and inter-tidal locations durability class 'vh' ('expected service life' of 20 years, vh = very high durability according to ISO 12944-1) is recommended.

The surface preparation and paint application are as critical as the paint selection to ensure proper cure, adhesion and Dry Film Thickness (DFT) of the paint system. To ensure maximum performance of a paint system, the majority of the coats of the system or, if possible, the complete system, should be applied in the workshop/factory. The surface preparation is recommended to be as per ISO 12944-4 and the paint application as per ISO 12944-7.

After completion of installation on site, any damage shall be repaired. Proper inspection and preventive maintenance are essential to achieve the design life. It is essential that the paint system is suitably maintained according to paint manufacturers recommendations.

9.4.3.2 Sacrificial Anodes

Waterproof closed-box type panels coated with high durability paints designed for very high corrosion environments have proved to be a very efficient way to protect against corrosion over the years in many marine installations and can be considered enough protection in the majority of the cases and in the majority of cases sacrificial anodes are not required. For some special cases which should be determined by the designers, where the fenders are located in places under very severe and extreme corrosion conditions the addition of zinc anodes to the external plates of the steel panels could be a good option to reduce the corrosion deterioration when the steel parts are submerged in water, as they provide a large zinc reservoir so they can protect steel. It is important that the anode is permanently immersed to avoid build-up of an oxide surface layer which prevents the anode from working.

9.4.3.3 Impressed Current

Corrosion protection of some steel berthing structures is achieved by Impressed Current Cathodic Protection (ICCP), a system that prevents corrosion by applying an electric current to the steel structures, converting these from active to passive sites. It is possible to include the fender panels into the system, by providing electrical bonding between the panels and the ICCP system.

In such a case, all fender design parameters must be communicated to the ICCP designer ahead of the full system development.

9.5 Fabrication of UHMW-PE Low Friction Facing

UHMW-PE face pads can be installed to cover the entire face of the panel (including the face of all the chamfers). They are intended to create a smooth sacrificial rubbing surface designed to limit frictional forces on the fender and the panel, when it comes into contact with the vessel.

Ultra-high-molecular-weight polyethylene (UHMW-PE) is synthesised from its monomer ethylene. UHMW-PE is an engineering thermoplastic with a molecular weight of greater than 3 million g/mol. The high molecular weight enhances a number of important physical properties, including abrasion resistance, high impact strength and a low coefficient of friction.

Specifications and/or drawings should define the low friction facing colour. The colourants are blended into the raw resin material, as to colour the entire pad thickness.

UHMW-PE low friction facing pads are manufactured by compression moulding. Heat and hydraulic pressure are used to fuse the polymer powder inside a mould between large platens. Then the UHMW-PE is machined in order to adjust shape, edges and holes to the dimensional specifications.

9.6 Fabrication of Accessories

High-quality chains, accessories, compatible fastenings, brackets, anchors, etc. must ensure the correct support and load distribution on the fender system. There is a wide range of materials suitable for fender applications and these must be tested and certified.

The correct fabrication of fender accessories is critically important for the performance and durability of the fender unit therefore adequate details should be included in the specification and drawings.

9.6.1 Chains

Chains are manufactured according to international and national standards.

9.6.2 Anchors and Accessories

Fender elements and chain brackets are fixed on concrete structures by means of steel anchors.

These can be either cast-in or resin anchors; whenever possible, preference should be given to cast-in anchors as this minimises the risk of installation-related quality issues and saves human efforts and time.

The anchors should be manufactured as per the prevailing project standards and requirements. When these are not specified, internationally recognised standards can be applied.

Anchors are to be manufactured based on the factored design load.

The anchor's mechanical properties are directly impacted by the material durability.

Although stainless steel fixings may be recommended/specified in corrosive environments, there is no need for full cast-in assemblies to be made of the same steel grade. Stainless steel sockets may, for example, be used in combination with carbon steel anchors. Although no galvanic corrosion would occur in the absence of an electrolyte.

9.6.3 Protection Against Corrosion for Accessories

9.6.3.1 Galvanising

Chains assemblies' components, anchors and chain brackets are usually galvanised. This method provides good corrosion protection for chains and other steel accessories.

There are various processes to apply the zinc layer, depending on the element thickness and the coating thickness to be achieved. The method used for fender components is typically hot-dip galvanising, where the elements to be galvanised are immersed in a bath of molten zinc, allowing for a nominal zinc layer of about 80 μm (for element thicknesses > 6 mm). ISO 1461, ASTM A153, ASTM A123 are the commonly followed international standards for hot-dip galvanisation. Hot-dip galvanisation is not an applicable option for steel panel structures since technically it is not possible to manufacture hot-dip galvanised closed box steel panels.

Paint can be applied to hot-dip galvanised surfaces. ISO EN 12944-5: shall be followed for DFT (dry film thickness) recommendations.

9.6.3.2 Use of Stainless-Steel Components

In highly corrosive environments, some fender elements such as anchors or fixings may be specified to be stainless steel to ISO 3506 or equivalent. The designer shall decide the appropriate stainless-steel grade applicable to local design requirements. Since stainless steel load capacity is limited compared with other steel types, it is not recommended the use of stainless steel for chains or U-anchors.

Designers should be aware that galvanic corrosion occurs when stainless-steel material is in direct contact with galvanised material.

9.7 Pneumatic Fenders

Pneumatic fenders consist of a cylindrical air bag with hemispherical heads at both ends, which shall be filled with compressed air that absorbs the energy of the berthing vessel.

The basic body construction of this fender consists of an outer rubber layer, synthetic-tyre-cord layer for reinforcement layer and an inner rubber layer.

The outer rubber protects the cord layers and inner rubber from abrasion and other external forces. This rubber compounds shall have enough tensile and tear strength and the fenders should be built with rubbers and metal components to provide a long lifespan. The reinforcement synthetic-tyre-cord layers needs to be strong enough to hold the internal pressure in both compressed and non-compressed situations.

These layers are vulcanised together to ensure bonding between layers of dissimilar characteristics. For more details on minimum material requirements for pneumatic fenders, refer to the relevant ISO 17357-1-2014 standard.

9.8 Foam Fenders

There are generally three primary configurations or types of foam fenders:

- Cylindrical mid-body and conical shaped ends ending in a swivel end fitting on the cylinder/conical centreline at each end.
- Cylindrical mid body and hemispherical shaped ends and typically fitted with chain/tyre nets as a protection/wear element and to facilitate attachment via flange plates (with clevis pins which are part of the net).
- Circular or donut type fender with bearing pads, with a composite reinforced elastomer skin which houses an energy absorbing foam annulus.

Foam fenders are constructed with a resilient, energy absorbing closed cell foam core. The core can be constructed of laminated layers to create a virtually solid core. This core is covered by a composite skin of elastomer and flexible reinforcement. The purpose of the skin is to contain and to protect the energy absorbing material. This surface should allow for the anticipated level of wear during the service life of the fender.

Skin thickness, reinforcement layer and the density of the foam determine the performance and the durability of foam fenders. The skin is necessarily reinforced to ensure this function is performed reliably.

Depending on the foam filled fender design, features such as integral end swivels, attachment points, chain/tyre nets bearing materials may be included.

10 TEST PROCEDURES OF MARINE FENDERS

This chapter provides an overview of the recommended testing protocols for evaluating the performance, ensuring material quality, and reporting the performance of wrapped, moulded, and extruded rubber, pneumatic, and foam-filled marine fenders (except extruded profiles e.g., DD fenders, Composite and fenders used for mounting on tugs).

These testing procedures are specifically designed to ensure that the engineering data presented in manufacturers' catalogues, as well as the verification of fender performance and durability, are based on standardized testing methodologies widely recognized and practiced within the industry. By adhering to these common testing procedures, manufacturers can provide reliable and comparable data, enabling customers to make informed decisions when selecting marine fenders for their specific applications.

10.1 Classification of Rubber Fender Testing

The testing of rubber is classified into three types as can found in Table 10-1.

Type of testing	Definition	Types of testing	Extent of disclosure	Testing conducted by
Fundamental testing	Testing of fenders and material to create data for publication in the catalogue for either a new product launch or enhancement of features and /or improvement of performance of the existing products.	- Base performance (see Section 10.1) - Creation of velocity factor (C_v), temperature factor (C_t), angle factor (C_{ang}), (see Section 10.7.3-10.7.5) - Durability test (see Section 10.6) - Chemical composition or Thermal Gravimetric Analysis (TGA, refer 10.7.2) of rubber compound of the grades published in the catalogue - Physical properties of vulcanised rubber compound (see section 10.7.1) - TGA of sample fenders tally with rubber compounds, samples for physical properties (see Section 10.7.2) and values published in the catalogue.	By the manufacturer in the manuals/brochures/ websites	Manufacturers or third-party agents engaged by the manufacturer. All tests are mandatory
Type approval testing	Fundamental testing witnessed and verified by third-party for obtaining Type Approval certificate	- Base performance (see Section 10.1) - Creation of velocity factor (C_v), temperature factor (C_t), angle factor (C_{ang}), (see Section 10.7.3-10.7.5) - Durability test (minimum 3000 cycles) (see Section 10.6) - Chemical composition or Thermal Gravimetric Analysis (TGA, refer 10.7.2) of rubber compound of the grades published in the catalogue - Physical properties of rubber compound (see Section 10.7.1) - TGA of sample fenders tally with rubber compound, samples for physical properties (see Section 10.7.2) and values published in the catalogue.	Testing data or sources of testing data for factors to be disclosed upon request.	Testing conducted by manufacturer and verified by third-party agents or testing conducted in a third-party testing facility. All tests are mandatory
Verification testing	The testing of commercial fenders to ascertain their adherence to the published performance standards or customer-specified energy and reaction requirements.	Mandatory tests: - Verification of base performance (see Section 10.1) - Physical properties of rubber compound (see Section 10.7.1) Highly recommended tests: - Performance and physical properties testing witnessed by a third-party, or using a third-party testing jig or in a third-party testing facility. - Chemical composition or Thermal Gravimetric Analysis (TGA, refer 10.7.2) of rubber compound used for production - TGA analysis of samples from rubber fenders Optional tests: - Verification of factors (see Section 10.7.3-10.7.5)	To the project	Mandatory tests are conducted by manufacturer. Highly recommended tests are witnessed/ verified by third-party agent or tests are conducted in a third-party testing facility based on the agreement between manufacturer and

Type of testing	Definition	Types of testing	Extent of disclosure	Testing conducted by
		- Verification of durability test (see Section 10.6) - Shear- compression test (see Section 10.6) - Fatigue testing (see Section 10.6) Note: - The selection of appropriate tests for a project is at the discretion of the designers or end users, taking into consideration factors such as the level of severity and risk associated with the site, as well as the type and quantity of fenders employed. - The project or purchase specifications should clearly outline the specific tests required for evaluation. - In certain cases, deviations from mandatory testing requirements may be permitted based on project specific requirements or specifications. Flexibility may be allowed to accommodate variations from the standard testing protocols, provided it aligns with the project's specific needs and complies with the specified criteria.		designers/project specification.
Notes on third-party testing providers: • The owner or end users/ consultants of a project may seek assistance from a manufacturer in selecting a third party for verification testing, with the final decision resting in their hands. • The chosen third party should possess the qualities of being an independent and accredited service provider or an independent testing facility, with a proven track record of testing knowledge and industry experience. • To meet the recommended standards set by ISO 17025, the testing facility must be adequately equipped, certified, and possess the necessary capacity for conducting verification testing. Additionally, the calibration of the test apparatus should be annually checked by a qualified external organisation using instrumentation that is traceable to a certified national standard. • Before each verification test at the designated location, third-party testing equipment should undergo calibration and verification using a dummy or known sample with a similar load or reaction. Calibration certificates issued by external agencies should be provided to the end user as part of their quality assurance requirements, and these certificates must be brought to the testing facility. It is essential to perform calibration for both single load cells and any group of load cells. The equipment setup needs to be agreed upon by the manufacturer and the third party, as it can significantly impact the test results. • Differences in testing facilities may have an impact on the test results. Therefore, the third party should minimize such discrepancies by employing calibrated measuring equipment. In the event of inconsistencies between the test results of the manufacturer and the third party, further investigations should be initiated to address and mitigate any differences. • The format of all test result reports should be in line with the manufacturer's format to facilitate seamless comparison.				

Table 10-1: Rubber fender testing scheme

10.2 Determination of Fender Base Performance

10.2.1 Constant Velocity (CV)

The compression of a fender at slow constant speed and determining performance is called the Constant Velocity (CV) performance. The slow constant speed is speed of 0.33-1.33 mm/s (strain rate $\dot{\epsilon}_0 = 0.01$ to 0.3 %/s).

10.2.2 Standard compression and Base performance

The performance Base reaction force (R_{base}) and Base energy absorption (E_{base}) tested in a standard condition shall be called as the Base fender performance. This is the basic data published in manufacturer's catalogue along with design factors.

10.2.3 Standard condition

The basic condition for the fender testing. At the following standard conditions, the characteristic correction factors are each 1.0.

Characteristic correction factors	Values at Standard condition
Angular factor: C_{ang}	Standard angle $\theta_0 = 0^\circ$
Velocity factor: C_v	Compression speed = 0.33-1.33 mm/s (2-8 cm/min) or compression strain rate = 0.01 to 0.3 %/s
Temperature factor: C_t	Standard temperature $T_0 = 23 \pm 5^\circ\text{C}$

Table 10-2: Standard conditions where correction factors are 1.0

10.3 Test Apparatus for Compression Test

- The test apparatus shall be equipped with a calibrated (not more than 12 months at the time of testing) load measuring device such as load cell(s) or pressure transducers and linear transducer(s) or calibrated laser displacement sensors for measuring displacements, all capable of providing continuous monitoring of fender performance.
- The test apparatus shall be capable of recording and storing load-cell and transducer data at intervals of $0.05 h_f$ or smaller, where h_f is a fender's normal height. The following information shall be included, as a minimum:
 - Serial number and description of test item
 - Date, time at the start and at the end of the test
 - Location of the test facility and the test apparatus ID
 - Stabilisation temperature of the test specimen
 - Test ambient temperature
- For fender tests, all equipment used to measure and record force and deflection shall be calibrated and certified accurate to within ± 1 (one) percent in accordance with ISO or equivalent requirements. Calibration shall be performed within one year of the use of the equipment, or less if the normal calibration interval is shorter than one year. Calibration of test apparatus shall be checked annually by a qualified external organisation, using instrumentation which is traceable to a certified national standard.

10.4 Supporting Protocols

10.4.1 Break-In Compression Cycle

The first compression cycle of the fender exhibits unpredictable reaction force and deviates from the values provided in the manufacturer's catalogue. However, subsequent compressions gradually bring the fender's performance closer to the published values. This first compression, referred to as the 'break-in' compression, occurs when the fender is compressed to its designed deflection shortly after manufacturing and is considered to be part of the manufacturing process for big fenders:

- "Break-in" compression is mandatory for buckling type rubber elements used for fender systems with a catalogue reaction force more than 1000 kN (100 tonnes) or those intended for installation on load-sensitive structures such as dolphins.
- For other types of fenders, the decision and specification regarding 'break-in' cycles, if necessary, should be determined by the end users or their designers. Upon request, the manufacturer should provide evidence for all 'break-in' compressions. It is not mandatory to subject such fenders to temperature stabilisation.

In the case of fenders used in low-temperature environments (-10°C and below) or stored for an extended period of time, it is advisable to perform the 'break-in' process before delivery.

10.4.2 Stabilising Compression Cycles

Stabilising compressions refer to the initial compressions conducted before the fender performance tests in order to minimise the impact on the fender's performance of residual stresses generated during the manufacturing process. These compressions aim to stabilise the performance variance. The testing conditions for the stabilising compressions should adhere to standard conditions.

Typically, at least two stabilising cycles are performed to ensure that the difference in the first peak values of the reaction force between two consecutive compressions falls within the range of 0-5 %. Once the difference between two consecutive compressions is less than 5 %, further compressions are not permitted as the desired level of stability has been achieved.

10.4.3 Thermal Stabilisation

During and after the vulcanization process, residual heat is retained within the fender body. The performance of the fenders can be influenced by factors such as residual heat and environmental temperature. Therefore, it is crucial to stabilise the fender's temperature before conducting performance tests.

Temperature stabilisation should be conducted in a conditioning room set at the standard temperature of $23 \pm 5^{\circ}\text{C}$ for a duration determined using Equation (10-1). Alternatively, the number of days required for temperature stabilisation can be calculated by measuring the centre temperature of the fender body using methods such as thermocouples, heat transfer analysis, or other equivalent approaches. (See Appendix B).

$$t(\text{days}) = (12.675 \ln(\Delta T) - 2.0352) (\text{maximum rubber thickness})^2 \quad (10-1)$$

Where,

When the initial temperature higher than 28°C, $\Delta T = \text{Initial temperature} - 28$

When the initial temperature lower than 18°C, $\Delta T = 18 - \text{Initial temperature}$

- The duration of thermal stabilisation after curing process shall be calculated using the 'curing temperature' as an 'initial temperature'.
- If the fender has been stored at ambient temperature for an extended period, the temperature of the fender should be checked and recorded for the past week. The highest or lowest temperature for which ΔT is highest, shall be chosen as an initial temperature.

During the conditioning of the fenders in the conditioning room, the manufacturer should monitor the process using suitable methods such as video streaming or thermal data loggers to ensure that the fenders are stored within the designated room. Temperature logging should be performed at a frequency of no less than three times per day, and the duration should align with the thermal stabilisation days calculated using Equation (10-1).

The body surface temperature of the fenders just before testing should be measured using appropriate equipment such as thermocouples or handheld non-contact thermometers. The measurement should be conducted at a distance of no more than 1 metre from the rubber surface, with a frequency of no less than three times per day at a minimum of three different spots. The measurement duration should align with the thermal stabilisation days calculated using Equation (10-1).

It is important to note that temperature stabilisation is mandatory for all fundamental, type approval, and verification testing processes.

10.5 Performance Testing Protocol for Standard Compression

The performance test shall be conducted under standard condition using the CV testing method.

- Thermal stabilisation is mandatory for all compression cycles, prior to conducting the performance test. Additional 'break-in' compression is not needed if the fenders selected for verification testing are randomly chosen from the pool of already broken-in fenders.
- If applicable, perform a complete break-in compression cycle by deflecting the specimen once to its designated deflection point.
- Conduct stabilising compression cycles on the break-in specimen by deflecting it a minimum of two times, with a minimum 5-minute interval between each compression (See 10.4.2).
- Allow the specimen to undergo a recovery period of at least one hour maintaining the ambient temperature of $23 \pm 5^\circ \text{C}$.
- Before proceeding with the final compression, verify the temperature of the fender at various locations to ensure it remains within the specified limits.
- Perform another deflection of the specimen under standard conditions. This cycle will be regarded as the 'standard compression' cycle, and the performance values obtained during this cycle will serve as the base performance.

It is important to complete the entire test process, from the break-in compression cycle to the standard compression, within a maximum period of 24 hours.

For fenders stored beyond the maximum recovery time of 24 hours, it is expected that the reaction force will increase (above the fourth compression value but always below the first compression value) depending on factors such as the grade and type of rubber, as well as the time and temperature of the storage area. If performance of such fenders needs to be determined, additional stabilising compression cycle should be conducted followed by the standard compression.

Fundamental testing	Type approval testing	Verification testing
1. The test specimens utilised shall consist of either full-size fenders or scale-down models. 2. For fenders or specimens, it is crucial to subject them to compression at the maximum achievable deflection. The deflection at which the efficiency, represented by the E/R ratio (energy absorption to reaction force ratio), reaches its peak is defined as the 'Design deflection' of the fender. At this deflection point, the base reaction force and base energy absorption of the fenders shall be determined.	1. The test specimens shall consist of full-size fenders or scale-down models. 2. During the testing procedure, the fenders or specimens should be compressed to meet and slightly exceed the designated design deflection. 3. The determination of the number of fenders for testing purposes is the responsibility of the manufacturer. It is recommended to select a minimum of three rubber grades and three heights that represent the corresponding catalogue grades and heights for each fender type and energy-absorbing material (rubber grades). 4. In cases where intermediate grades are involved, interpolation techniques may be applied to derive results. 5. Change in geometry needs a new set of Type approval testing	1. The specimens used for the tests shall consist of full-size fenders of the specific project. 2. For certain types of fenders, such as cone-shaped fenders, the use of a steel spacer between the fender and the plates may be necessary. 3. The number of samples selected for testing should be determined based on an agreed sample scheme between the customer and the fender manufacturer. If a specific sampling scheme has not been established, a minimum of 10 % of the fender order should be randomly selected and tested to ensure compliance with performance requirements. 4. Fender compression should be stopped at the design deflection point. In some cases, it may be beneficial to exceed the design deflection by up to 1 % in order to analyse the trend of the increase or decrease in reaction force. 5. A fender is deemed to meet the required performance criteria for berthing if it achieves the base (CV) energy value (within production tolerance) without exceeding the base (CV) reaction value (within production tolerance) at any deflection point. However, the compression needs to be conducted up to the maximum design deflection. 6. In situations where the primary function of a fender is to absorb minimum berthing energy without exceeding maximum reaction force, and the subsequent reaction force is not of critical importance, it is possible for the reaction force to exceed above production tolerances even before reaching the design deflection. However, in projects where precise reaction force values are crucial for facility design (e.g. mooring simulation), it is essential to establish an agreed-upon acceptance policy with the customer. 7. If any randomly selected fender, representing 10 % of the order quantity, fails to meet the performance criteria, testing should be extended to include 20 % of randomly selected fenders. 8. If any additional fender from the 20 % lot fails the performance test or if the fenders are intended for use in load-sensitive structures, 100 % of the fenders need to be tested. 9. Chemical composition or TGA (See 10.7.2) testing of the rubber compound obtained from the body of the tested fenders should tally with values published, rubber compound used in the production floor and the samples used for physical property testing (see 10.7.2). 10. A witness should be allowed to observe the compression procedure near the testing press, ensuring safety protocols are followed. It is not recommended to view the testing procedure via off-site remote streaming devices unless there has been prior discussion and mutual agreement between the manufacturer and the client regarding such arrangements. 11. If applicable, the location of the load cells on the third-party independent testing jig should be discussed and finalised between the manufacturer and the third party before conducting the tests. 12. Folding marks appear on the fender inner or outer surface of the fender during compression are acceptable unless proven to be crack marks. 13. While asymmetric folding of fenders during compression testing is not expected, it can be accepted as long as it does not impact performance values and durability.

Table 10-3: Additional Information for standard compression test

10.6 Protocol for Durability Test

The objective of this test is to simulate and assess the long-term fatigue performance of a fender within a condensed timeframe, providing insights into its longevity. The durability test encompasses three main phases: pre-repetitive compression, repetitive compressions, and post-repetitive compression. The procedure for conducting this test is outlined as follows:

- a. Conduct thermal stabilisation of the specimen, ensuring it is maintained within the temperature range of $23\pm5^{\circ}\text{C}$.

Pre-repetitive compression:

- b. Conduct the break-in compression by deflecting the specimen once to its designed deflection see Table 10-3.
- c. Stabilise the specimen by deflecting it twice to its designed deflection under standard conditions. Record the reaction force of the last compression as R_{s2} and the corresponding energy absorption as E_{s2} .
- d. Transfer the specimen to a conditioning room controlled at $23\pm5^{\circ}\text{C}$ and allow it to 'recover' for a minimum of one hour and a maximum of 24 hours.
- e. Measure the height of the specimen (h_{s2}) and verify the temperature on its surface to ensure it remains within the specified range of $23\pm5^{\circ}\text{C}$.

Repetitive compression:

- f. Apply repetitive deflections to the specimen at its designed deflection. The compression speed and strain rate are not specifically defined, however the duration of one complete cycle should not exceed 150 seconds. Note that the specimen may generate heat during repetitive deflection and should not be artificially cooled.

Combined shear- compression test:

Conduct repetitive deflections of the specimen to 50 % of the fender height (less for certain fender types, like V-fenders), while simultaneously applying shear forces causing deflections of 20-30 % of the fender height (as specified by the project specifications). These deflection percentages are recommended as guidelines, but the designer may select values based on the project requirements.

- g. Return the specimen to the conditioning room, ensuring the temperature is within the specified range, and allow it to "recover" for a minimum of one hour and a maximum of 24 hours

Post-repetitive compression:

- h. Measure the height of the fender (h_{fsc}) and perform a single deflection of the specimen at the standard condition. Record the reaction force of this compression as R_{fsc} and the corresponding energy absorption as E_{sc} .

Fundamental testing	Type approval testing	Verification testing
1. The testing process shall include the evaluation of at least three distinct rubber grades, representing the softest and hardest variations within each fender type. 2. The specimen used for testing purposes shall be either a real size fender or a scaled-down version. In the case of leg fenders, the L/H ratio of the specimen must not fall below the minimum ratio found in the catalogue of fenders with the same basic design. 3. The minimum number of compression cycles to be conducted during testing is set at 3000, unless otherwise determined by the manufacturer based on specific considerations.	1. It is recommended that a minimum of two rubber grades are included in the testing process, specifically the hardest and softest variants. 2. The specimen used for testing purposes shall be either a real size fender or a scaled-down version. In the case of leg fenders, the L/H ratio of the specimen must not fall below the minimum ratio found in the catalogue of fenders with the same basic design. 3. Prior to and following the completion of the test, a TGA (Thermogravimetric Analysis) examination of the test specimen should be conducted by an accredited third-party laboratory. 4. The testing protocol requires a minimum of 3000 compression cycles to be performed. 5. Acceptable requirements for the durability test are as follows: - No visible defects such as cracks. - The remaining value of reaction force $R_{f,SC}/R_{f,S2}$ shall be above 80 % - The remaining value of energy absorption E_{SC}/E_{S2} shall be above 80 % - The residual strain $(h_{f,S2} - h_{f,SC})/h_{f,S2}$ shall be less than 5 %.	1. For durability test: - Select a randomly chosen commercial fender, specifically manufactured for a project, as the specimen. If a full-size fender is not feasible, use a scaled-down model that adequately represents the commercial fender and fits within the available test press at the third-party or manufacturer's facility. - Ensure that the scaled-down model possesses the same fundamental design and compound as the commercial fender. - For leg fenders, the specimen's L/h ratio should not be lower than the minimum ratio of any catalogue models with the same basic design. - The project specification, agreed upon by the manufacturer and designer/end user, should indicate the size and type of samples required. - For Shear-compression test: The specimen shall be a scaled down model, or a commercial fender manufactured for a project. 2. The samples for chemical composition or Thermal Gravimetric Analysis (TGA, refer 10.7.2) of the specimen's body, both before and upon completion of the durability test, must be collected by a third-party witness. The results should tally with the TGA of the fender body manufactured for the project. 3. Acceptable requirements for the durability test are as follows: - No visible defects such as cracks. - The remaining value of reaction force $R_{f,SC}/R_{f,S2}$ shall be above 80 % - The remaining value of energy absorption E_{SC}/E_{S2} shall be above 80 % - The residual strain $(h_{f,S2} - h_{f,SC})/h_{f,S2}$ shall be less than 5 %. For shear-compression test: - Test sample shall be free from any indications of crack or permanent deformation in shape. 4. The number of cycles for the durability test should be a minimum of 3,000 or as agreed upon between the manufacturer and purchaser, or as specified in the project specification.

Table 10-4: Additional Information for durability test

Fatigue testing:

- Fatigue testing is highly recommended for fenders installed at exposed berths, high-frequency berths, or between two permanently moored vessels. It becomes particularly crucial when a buckling fender element is permitted to deflect beyond the linear elastic range of the performance curve. (For detailed information, refer to Chapter 7.)

- Fatigue testing entails subjecting the fender to compression cycles exceeding 3,000 cycles. The specific number of cycles and compression percentage should be determined based on dynamic mooring analysis or the project specification.
- The test samples should be scaled down models that closely resemble the shape, basic design, and compound of the actual fenders.
- The standard approach for fatigue testing involves matching the strain rate between the actual fender used in projects and the model fender. This method should be employed unless an alternative approach is justified with technical reasoning. In the case of a buckling fender that experiences repeated deflections beyond the linear elastic range of the performance curve, alternative scaling methods or testing techniques may be necessary to accurately simulate the accumulation of heat in the fender element.

10.7 Material Tests

10.7.1 Rubber Compound: Physical Properties

In the rubber production line, the raw materials, including rubbers and chemicals, undergo a physical mixing process. In this context, a 'batch' refers to the mixing capacity of a mixer used for compound mixing. It is common for multiple batches to be employed in the production of a single fender. On the other hand, a 'lot' is defined as the quantity of materials used for a specific project, encompassing each fender type and grade. To assess the quality and characteristics of the rubber compounds utilised in the products, it is highly recommended to conduct physical property tests. These tests serve as a means of evaluating the suitability of the rubber compounds for various stages, including Fundamental testing, Type Approval testing, and Verification testing.

- Sample type: The sample used for testing should consist of un-vulcanised rubber compounds obtained from the production floor and subsequently vulcanise them in the laboratory following the standards for sample preparation.
- The responsibility for preparing the test specimens from the rubber compound lies with either the designated third-party laboratory responsible for conducting the tests or the manufacturer's laboratory, as specified in the project specification agreed upon by the manufacturer, designer, or end user.
- Sample collection method: The collection of samples should be carried out by obtaining the required number of samples from each type, grade, and lot of compounds used in the production for the project. Alternatively, a sampling scheme agreed upon between the customer and the fender manufacturer can be followed to determine the sample collection approach.
- Testing Different Compounds: In cases where different compounds are utilised in a fender, it is recommended that samples are obtained from all rubber compounds involved to conduct physical properties tests.
- Recommendation for Physical Property Tests: *Table 10-5* provides a set of recommended physical property tests for rubber compounds used in rubber fenders (except extruded profiles, e.g. DD fenders, composite fenders and fenders used for mounting on tugboats). These recommendations serve as a guide and are not limited to the listed tests.
- Verification Testing: For verification purposes, the chemical composition or Thermo-Gravimetric Analysis (TGA, see Section 10.7.2) test results of the rubber specimens used for physical properties testing should tally with the values obtained from samples collected from a commercially manufactured fender specifically produced for the project.

Property	Testing standard	Condition	Requirement	Applicability
Tensile strength	ISO 37	Before Ageing	$\geq 16 \text{ MPa}$	All
	ISO 188	Aged for 96 hours at 70°C	$\geq 12.8 \text{ MPa}$	All
		Aged for 168 hours at 70°C	$\geq 12.8 \text{ MPa}$	Applications above 40 degrees Celsius site temperature
Elongation at break	ISO 37	Before ageing	$\geq 350 \%$	All
	ISO 188	Aged for 96 hours at 70°C Aged for 168 hours at 70°C	$\geq 280 \%$ $\geq 260 \%$	All Applications above 40 degrees Celsius site temperature
Tensile product (Tensile strength X Elongation @break)		Before ageing	$\geq 6000 \text{ MPa. \%}$	All
		After ageing	$\geq 4800 \text{ MPa. \%}$	All
Compression set	ISO 815-1	22 hours at 70°C	$\leq 30 \%$	All
Tear resistance	ISO 34-1, method C procedure b	Original	$\geq 70 \text{ kN/m}$	All
Static Ozone resistance	ISO 1431-1	50pphm, 100 hours, 20 % strain, 40°C	No cracks	All
Dynamic Fatigue	ISO 132	15000 cycles	Grade 0-2	All
Bond strength	ISO 813	Rubber to steel	$\geq 7 \text{ N/mm}$ and failure in the rubber.	Optional to metal inserted in a fender body
Sea water resistance	ISO 1817	28 days at 85°C, refer DIN 50905-4 for the preparation and composition of sea water	Volume change: $\leq +10/-5 \%$	Optional to fenders 100 % submersed in sea water all the time
Abrasion loss	ISO 4649	Original	$\leq 150 \text{ mm}^3$	Optional to direct contact of vessels and fenders

Property	Testing standard	Condition	Requirement	Applicability
Heavy Oil resistance	ISO 1817	72h at 23°C in IRM oil #901	Volume change: $\leq \pm 10 \%$	Optional to fenders exposed to heavy oil (e.g., under marine loading arm)
Hardness	ISO 48-4	Original Value (Shore A)	≤ 78 ≤ 75	standard Applications above 40 degrees Celsius site temperature
		Aged for 96 hours at 70°C	Increase from the original value: ≤ 8 points	
Low-temperature brittleness	ISO 812: procedure c	Rubber specimens to be exposed to -40°C for 3±0.5 minutes.	No cracks, fissures or holes visible to the naked eye and no complete separation into two or more pieces.	Optional to fenders used for very low temperature applications
Notes: - The rubber formulation employed in the extrusion process differs from that of the moulding process, particularly in terms of the carbon black variant used. Consequently, the physical property measurements for extruded profiles vary from those of moulded products. It is recommended to consult the manufacturer for precise specifications regarding the physical properties of extruded profiles and composite fenders. - The test results obtained from samples prepared from the fender body may differ from those obtained from laboratory sheets. This disparity can be attributed to variations in the sample preparation process and the varying degrees of vulcanisation. Therefore, it is essential for the specification to clearly specify which aspects are to be tested using laboratory samples and which are to be tested using the ready fender. - Maturation time of the freshly prepared samples from the uncured rubber compound shall be minimum 16 hours. Tensile strength, elongation at break, tear strength shall be carried out after maturation time. After ageing properties, compression set, ozone tests shall be carried out after stipulated time provided in the respective ISO standards. - Verification Test: It is strongly recommended that two sheets of uncured rubber compound are provided for testing when conducting tests in a third-party laboratory. In the event that the specimens prepared from the first piece of rubber compound do not meet the specified values, but the fender performance test adheres to the given specifications, it is advisable to carry out a retest of the failed properties in a third-party laboratory. The second piece of uncured compound should be utilised for specimen preparation in this retesting process. Furthermore, the results obtained from the Thermogravimetric Analysis (TGA) performed on the fresh samples should tally with the TGA results obtained from the fenders, (see Section 10.7.2). If the tests conducted at the third-party laboratory yield unsatisfactory results, the designer or end user retains the right to reject the entire batch of fenders manufactured using the unsuccessful rubber compound.				

Table 10-5: Physical properties table

10.7.2 TGA (Thermogravimetric Analysis)

The rubber chemical composition test involves employing TGA (Thermogravimetric Analysis) equipment and adhering to ISO 9924 or an equivalent rubber compound testing standard. ISO 9924 outlines a thermogravimetric method for assessing 'Total organic content', carbon black content, and ash in vulcanisates and uncured compounds, with the mass loss at 300°C approximately indicating low volatile-materials content. 'Total organic content' encompasses polymers and low volatile materials within the compound.

To quantify specific quantities of low volatile materials like waxy hydrocarbons, mineral oils, and sulphur, an acetone or solvent extraction, following ISO 1407 or an equivalent standard, should precede TGA testing. TGA can be applied to un-vulcanised rubber compounds, vulcanized rubber samples for physical properties testing, or samples from final products. Sample quantity determination may be recommended by the testing laboratory or agreed upon between the customer and the fender manufacturer.

For verification testing, at least 10 % of the fenders should be randomly selected, with sampling locations diversified on a fender to account for compound variations. The manufacturer should identify the sampling locations where the effect of scraping on a fender body has the minimum influence on the fender quality. Any remaining samples post-TGA should be retained for customer retrieval upon request.

Maintaining consistent TGA values is vital in fender production across multiple segments, including the uncured rubber compound, cured rubber compounds for physical testing, and fender body samples. All TGA values should align with published values, although some variations are anticipated when comparing results from different sample sources.

Part of type approval is TGA. This results in a bandwidth per content component for each rubber grade, to be shared by the supplier on request. TGA of production fenders should stay within this bandwidth. Differences between laboratories should be cleared up via the sheets tested per project (Section 10.7.1).

Reliable results necessitate conducting a minimum of three TGA samples from various fender areas, specimens, and rubber compound batches, with an average providing a representative measurement. To comprehensively assess fender quality, TGA testing should complement performance and durability assessments due to the sensitivity of rubber compounds to their manufacturing and curing history. It should be noted, TGA testing serves for determination of quantities in the compound only. Consistent TGA results indicate a continuous quality level of the compound mixing. Significant deviations in TGA results should trigger additional property testing (Section 10.7.1).

To accommodate the anticipated variation, a permissible tolerance of ± 1 % is recommended to each content component below 10 % weight of the compound, ± 2 % for content components between 10 %-30 % and ± 3 % for content components larger than 30 %. For example, if the target total organic content is 50 %, the acceptable, limits of tolerance range from 47 % to 53 %. Before TGA is executed final margins should be agreed upon.

10.7.3 Test Protocol for Creation of Velocity Factors

The compression test protocol is intended to create a velocity factor (C_v) as a part of the fundamental testing in widely divergent compression speeds.

- a) Prior to conducting velocity factor tests, it is essential to ensure the thermal stabilisation of the specimens at a temperature of $23 \pm 5^\circ\text{C}$, see Section 10.4.3.
- b) To complete the break-in compression cycle, the specimen must be deflected once to its design deflection, see Section 10.4.1.
- c) The break-in specimen must undergo complete stabilising compression cycles by deflecting it twice or more at the standard condition. The compression cycle should be stopped when the reduction of the reaction force (first peak) for two consecutive compressions falls within the range of 0-5 %. A 5-minute interval should be maintained between two compressions. The reaction force of the final compression is denoted as R_{s2} , and the energy absorption is denoted as E_{s2} .
- d) Transfer the specimen to the conditioning room controlled at the standard temperature with an accuracy of $\pm 1^\circ\text{C}$ and allow it to 'recover' for a minimum of one in the conditioning room.
- e) The specimen should be deflected to its designed deflection using one recommended compression speed or strain rate; see note below for the specific details. The compression speed can be Constant Velocity (V_{CV}) or Decreasing Velocity (V_{DV}), starting at the initial design speed and gradually decreasing with deflection.
The reaction force of this cycle is referred to as R_V , and the energy absorption is denoted as E_V . If the fender loses its typical buckling characteristics due to excessive speed, the design reaction force R_{fv} is determined at the deflection where the standard energy is absorbed. In such cases, $C_{v,E}$ equals to 1.0.
If the CV method is adopted for creating the velocity factor, a conversion factor or calculation is required to convert the CV testing speed to DV testing speed. However, the DV method does not necessitate any conversion factor.
- f) After the deflection cycle, the specimen should be allowed to 'recover' for a minimum of one hour and a maximum of two hours in the conditioning room.
- g) Deflect the specimen again at the standard condition at standard speed. This cycle shall be the same as a 'Standard compression' cycle and the reaction force is termed as R_{sc} and the energy absorption is E_{sc} .
- h) The velocity factor C_v , is determined by Eq. (10-2) and (10-3) for each recommended compression speed specified.

$$\text{For reaction force} \quad C_{v,R} = 2 \times R_{fv} / (R_{fs2} + R_{fsc}) \quad (10-2)$$

$$\text{For energy absorption} \quad C_{v,E} = 2 \times E_V / (E_{s2} + E_{sc}) \quad (10-3)$$

- i) Repeat steps from d) to h) for each recommended compression speed.
- j) Compute the velocity factor table against strain rate or compression time & publish in the catalogue.

Notes:

- It is required to complete a comprehensive test, starting from the break-in phase up to the final compression, for a single velocity condition within a maximum timeframe of 24 hours.
- The recommended options for strain rates are 0.5, 1, 10, 20, 30, 50, 75, and 100 % per second, or alternatively, compression speeds of 0.05, 0.10, 0.15, 0.20, 0.25, and 0.30 metres per second. These options apply to both Constant Velocity (CV) and the initial speeds of Decreasing Velocity (DV).
- The calculation for strain rate is as follows:

$$\text{Strain rate (\%/s): } V_0 \cdot 100/h_f \quad (10-4)$$

Where,

V_0 Compression speed (mm/s)

h_f Height of a fender (mm)

Compression time (second): designed deflection (mm)/ average decelerating compression speed (mm/s).

- Decreasing Velocity (DV) tests involve starting at various recommended compression speeds at 0 % deflection and gradually reducing to a rate slower than 5 mm/s at the designed deflection. The method employed for decreasing velocity can be linear, sinusoidal, or simulating a berthing scenario. The specific form of decreasing velocity should be noted and documented.
- Manufacturers have the option to choose either CV or DV testing methods for creation of velocity factors. In the event that the CV method is selected, it becomes necessary to provide a CV to DV speed correction factor in order to accurately calculate the compression speed of a fender. Conversely, if the DV test method is chosen, no correction factor is required for the calculation of the fender's compression speed.
- The test specimen may consist of a scale model that shares the same material and geometry as the fender depicted in the manufacturer's catalogue. The minimum size of the scale model should correspond to the smallest size provided in the catalogue or as determined by the manufacturer. It is necessary to conduct tests on at least three different rubber grades, encompassing the softest, medium, and hardest variations for each type of rubber.
- Equations (10-2) and (10-3) are recommended to nullify the compression history of a specimen if a single specimen is used for different speeds. While alternative methods for cancelling the compression history may be acceptable, a logical explanation must be provided. It is advisable to employ a new specimen for each test speed.

10.7.4 Test Protocol for Creation of Temperature Factors (C_t)

This compression test protocol is intended to create temperature factors as a part of the fundamental testing in a wide range of temperatures.

- Prior to conducting temperature factor tests, it is essential to ensure the thermal stabilisation of the specimens at a temperature of $23 \pm 5^\circ\text{C}$. See Section 10.4.3.
- The break-in compression cycle must be completed by deflecting the specimen once to its designed deflection, see Section 10.4.1.
- The break-in specimen should undergo complete stabilizing compression cycles by deflecting it twice or more at the standard condition. The compression cycle should be stopped when the reduction of the reaction force (first peak) for two consecutive compressions falls within the range of 0-5 %. A 5-minute interval should be maintained between two compressions. The reaction force of the final compression is termed as R_{s2} , and the energy absorption is termed as E_{s2} .
- Place the specimen in a controlled conditioning room set at the recommended testing temperature with an accuracy of $\pm 1^\circ\text{C}$. Allow the specimen to 'recover' and reach thermal stabilisation for the duration specified in Eq. (10-1).
Deflect the specimen once to its designed deflection. The reaction force for this cycle is referred to as R_v , and the energy absorption is denoted as E_v . In the event that the fender loses its buckling due to extremely low temperatures, the reaction force $R_{f,v}$ should be measured at the deflection where the standard energy is absorbed. In this case $C_{t,E}$ can be considered 1.0.
- Transfer the specimen to the conditioning room maintained at the standard temperature with an accuracy of $\pm 1^\circ\text{C}$. Allow the specimen to 'recover' and reach thermal stabilisation for the duration specified in Eq. (10-1).
- Deflect the specimen again under standard conditions and temperature. This cycle should be considered as a 'Standard compression' cycle. The reaction force for this cycle is denoted as $R_{f,SC}$, and the energy absorption is denoted as E_{SC} .
- The temperature factor C_t is determined by Eq. (10-5) and (10-6).

$$\text{For reaction force} \quad C_{t,R} = 2 \times R_{f,v} / (R_{f,s2} + R_{f,SC}) \quad (10-5)$$

$$\text{For energy absorption} \quad C_{t,E} = 2 \times E_v / (E_{s2} + E_{SC}) \quad (10-6)$$

- For each specimen and for each recommended temperature, repeat the steps from d) to h), (see note below).

i. Compute & publish the factors against temperatures in the catalogue.

Notes:

- The recommended temperatures for testing include: +50°C, +40°C, +30°C, +10°C, 0°C, -10°C, -20°C -30°C, and -40°C.
- It is required to complete a comprehensive test, starting from the break-in phase up to the final compression, for a single temperature within a maximum timeframe of 24 hours.
- The test specimen can consist of a scaled model that possesses the same material and geometry as the actual-sized fender depicted in the manufacturer's catalogue. The minimum size of the scale model should align with the smallest size published in the catalogue or as determined by the manufacturer. It is essential to conduct tests on at least two different rubber grades, encompassing the softest and hardest variations for each type of rubber.
- Equations (10-5) and (10-6) are recommended for cancelling the compression history if a single specimen is utilized for testing at different temperatures. While various methods for cancelling the compression history may be acceptable, it is imperative to provide a reasonable explanation. It is advisable to employ a new specimen for each test temperature.
- The compression velocity and angle should remain consistent and at standard conditions for all tests.

10.7.5 Test Protocol for Angle Factor

This compression test protocol is intended to create angle factors (C_{ang}) as a part of the fundamental testing in a wide range of angles. In this context, the term 'angle' refers to the inclination of the vessel's hull resulting from berthing and/or hull flare. Consequently, it is crucial that the direction of angular compression in the test is aligned parallel to the compression axis of the fender.

- a) Prior to conducting the angle factor tests, it is essential to perform thermal stabilisation of the specimens at a temperature of $23 \pm 5^\circ \text{C}$, see Section 10.4.3.
- b) The break-in compression cycle should be completed by deflecting the specimen once to its design deflection see Section 10.4.1.
- c) The break-in specimen should undergo complete stabilising compression cycles by deflecting it twice or more at the standard condition. The compression cycle should be stopped when the reduction of the reaction force (first peak) for two consecutive compressions falls within the range of 0-5 %. A 5-minute interval should be maintained between two compressions. The reaction force of the final compression is termed as R_{s2} , and the energy absorption is termed as E_{s2} .
- d) Transfer the specimen to the conditioning room controlled at the standard temperature with an accuracy of $\pm 1^\circ \text{C}$ and allow it to 'recover' for a minimum of one in the conditioning room.
- e) Install a properly designed angular jig in the compression testing press.
- f) Deflect the specimen once to its maximum deflection for a recommended design angle (refer to the accompanying notes below), under standardised conditions of compression speed and temperature. The reaction force for this cycle is termed as R_v , and the energy absorption is denoted as E_v .
- g) Return the specimen to the conditioning room controlled at the standard temperature with an accuracy of $\pm 1^\circ \text{C}$, allowing it to 'recover' for a minimum of one hour.
- h) Remove the angular jig from the compression testing press.
- i) Deflect the specimen under standard conditions, which shall be considered as a 'Standard compression' cycle.
- j) The reaction force is called R_{sc} and the energy absorption is called E_{sc} .
- k) The Angular Factor (C_{ang}) of this condition is determined by Eq. (10-7) and (10-8).

$$\text{For reaction force} \quad C_{ang,R} = 2 \times R_{fv} / (R_{fs2} + R_{fsc}) \quad (10-7)$$

$$\text{For energy absorption} \quad C_{ang,E} = 2 \times E_v / (E_{s2} + E_{sc}) \quad (10-8)$$

Notes:

- The recommended design angles for testing are 5, 10, 15, and 20 degrees.
- It is required to complete a comprehensive test, starting from the break-in phase up to the final compression, for a single angle within a maximum timeframe of 24 hours.
- The test specimen may consist of a scaled model that shares the same material and geometry as the actual-sized fender described in the manufacturer's catalogue. The minimum size of the scale model should correspond to the smallest size published in the catalogue or be determined by the manufacturer. It is crucial to test at least three different rubber grades, including the softest, medium, and hardest variants of each type of fender.
- Equations (10-7) and (10-8) are recommended for cancelling the compression history when a single specimen is used for testing at various angles. Acceptable alternative methods for cancelling the compression history may be employed, provided a reasonable explanation is provided. It is advisable to use a new specimen for each rubber grade and each angle.
- If the fender exhibits non-axisymmetric characteristics, testing is necessary in two orthogonal directions. Additionally, the centre position for deflection must be clearly defined.
- Typically, the Angle Factor at any angle does not exceed 1.0 during zero-degree compression. As a result, $C_{ang,E}$ holds greater importance in fender design than $C_{ang,R}$.

10.8 Rubber Compounds for Pneumatic Fenders

Sample type: un-vulcanized rubber compound obtained directly from the production floor and vulcanized in the laboratory following established standards for sample preparation.

The recommended tests for evaluating these samples can be found in Table 3 of the latest ISO 17357-1 standard.

10.9 Performance Tests of Pneumatic Fenders

The performance of a pneumatic fender exhibits stability when the initial internal pressure is precisely set to the specified pressure level. However, it is important to note that the performance may vary in response to changes in temperature.

To validate the performance of prototype fenders, it is recommended to conduct tests in accordance with Section 8 of the latest ISO 17357-1 standard.

10.10 Classification of Foam Fender Testing

The testing of foam is classified into three types:

Type of testing	Definition	Types of testing	Extent of disclosure	Testing conducted by
Fundamental testing	Testing of fenders and material to create data for publication in the catalogue for either a new product launch or enhancement of features and/ or improvement of performance of the existing products.	- Base performance (see Section 10.12.1) - Creation of Velocity Factor (C_v), Temperature factor (C_t), Angle factor (C_{ang}) (see Section 0) - Durability test (see Section 10.12.3) - Physical properties of foam core, polyurethane skin layer, reinforcement layer (see Section 10.11)	By the manufacturer on the manuals/brochures/websites	Manufacturers or third-party agents engaged by the manufacturer. All tests are mandatory.
Type approval testing	Fundamental testing witnessed and verified by third-party party for obtaining Type Approval certificate	- Base performance (see Section 10.12.1) - Creation of Velocity Factor (C_v), Temperature factor (C_t), Angle factor (C_{ang}) (see Section 0) - Durability test (see Section 10.12.3) - Physical properties of foam core, polyurethane skin layer, reinforcement layer (see Section 10.11)	Testing data or sources of testing data for factors to be disclosed upon request.	Testing conducted by manufacturer and verified by third-party agents or testing conducted in a third-party testing facility. All tests are mandatory.
Verification testing	The testing of commercial fenders utilised in job sites to ascertain their adherence to the published performance standards or customer-specified energy and reaction requirements	Highly recommended tests: - Performance and physical properties testing witnessed by a third-party, or using a 3rd party testing jig or in a 3rd party testing facility - Skin thickness and foam density verification testing (see Section 10.12.4/10.12.4) - Fender pull-through test (10.12.5) Note: - The selection of appropriate tests for a project is at the discretion of the designers or end users, taking into consideration factors such as the level of severity and risk associated with the site, as well as the type and quantity of fenders employed. - The project or purchase specifications should clearly outline the specific tests required for evaluation. - In certain cases, deviations from highly recommended testing requirements may be permitted based on project specific requirements or specifications. Flexibility may be allowed to accommodate variations from the standard testing protocols, provided it aligns with the project's specific needs and complies with the specified criteria.	To the project	Highly recommended tests are witnessed/verified by third-party agent or tests are conducted in a third-party testing facility based on the agreement between manufacturer and designers/ project specification.

Table 10-6: Foam fender testing scheme

10.11 Material Tests for Foam Fenders

10.11.1 Foam Core

The manufacturer receives and evaluates resilient closed cell foam sheets for utilisation in the fender production process. Acceptance of the quality (referring to properties *Table 10-7*) dependent upon the examination of test certificates containing specified values provided by the vendors.

Additionally, it is recommended that the manufacturer conduct periodic testing of foam material samples in a third-party laboratory. This testing should involve obtaining random samples from the production floor at least once per year for each foam density. The test records generated by the third-party testing serve as supporting evidence of foam quality.

Alternatively, a tailored testing scheme may be established between the designer and the manufacturer to ensure the foam quality meets the requirements of a specific project, e.g. the recommended tests and requirements are as listed *Table 10-7*.

Property	Testing standard	Condition	Requirement
Tensile Strength	ISO 1798	Unaged	As specified in manufacturer's catalogue
Elongation at break	ISO 1798	Unaged	As specified
Compression set	ISO 7214	Unaged	As specified
Compressive strength	ISO 7214	Unaged	As specified
Density	ISO 2781	Unaged	As specified
Water absorption	ISO 2896	Unaged	Max. 1.0 kg/m ² of cut surface

Table 10-7 Foam core physical properties

10.11.2 Polyurethane (Solvent Free) Outer Layer Elastomer or Similar Material

The manufacturer receives two components of polyurethane materials intended for use in the fender production process. Acceptance of the quality (referring to properties *Table 10-8*) is determined by conducting in-house or third-party tests on Tensile strength, Elongation-at-break, and hardness.

Furthermore, it is recommended that the manufacturer periodically assesses the quality by conducting tests at a third-party laboratory. Random samples should be obtained from the production floor at least once per year for this purpose.

The test records generated from the third-party testing serve as evidence of the polyurethane material quality. Alternatively, a customised testing scheme can be established between the designer and the manufacturer to ensure the materials meet the quality requirements of a specific project.

The recommended tests and requirements are as listed in *Table 10-8*.

Property	Testing standard	Condition	Requirement
Tensile Strength	ISO 37	Unaged	Min. 13.8 MPa
Elongation at break	ISO 37	Unaged	Min. 300 %
Hardness	ISO 48-4	Unaged	As specified
Tear Strength	ISO 34-1	Unaged	Min. 32.4 kN/m
Flex Fatigue life (Ross)	ASTM D-1052	Unaged	Min. 250,000 cycle
Abrasion resistance (NBS)	ASTM D 1630	Unaged	Min. 100

Table 10-8: Polyurethane properties

10.11.3 Reinforcement Layer

For the outer skin, it is recommended that nylon tyre cord of 2520 or an equivalent material is used with physical properties as listed in Table 10-8 as the reinforcing filaments. The manufacturer receives raw filaments for the production process. Acceptance of the quality (referring to Table 10-9) is dependent upon conducting in-house or third-party tests on Breaking strength and Elongation-at-break.

In order to ensure quality, it is advised that the manufacturer conducts annual testing of the filaments in a third-party laboratory. Random samples should be obtained from the production floor for this purpose. The test records generated from the third-party testing serve as evidence of the quality of the reinforcing filaments. Alternatively, a mutually agreed upon testing scheme can be established between the designer and the manufacturer to assess the quality of the filaments for a specific project. The recommended tests and requirements are as listed in Table 10-9.

Property	Testing standard	Condition	Requirement
Breaking Strength	ISO 2062	Unaged	Min. 231N
Elongation at break	ISO 2062	Unaged	Min. 16 %

Table 10-9 Nylon reinforcement filaments properties

10.12 Performance Tests of Foam Fenders

10.12.1 Fundamental and Type Approval Tests for Foam Fenders

To determine foam fender base performance and include it in the catalogue the following procedure is recommended:

1. Stabilize fender temperature for 48 hours at $23 \pm 5^\circ\text{C}$ before testing.
2. Perform a break-in with three compression cycles, slowly reaching the design deflection at a constant and slow velocity (CV) ranging from 0.33 to 1.33 mm/s.

3. Do not rotate the test fender after break-in cycles.
4. Foam fenders exhibit significantly slower compression recovery rates. Hence sufficient recovery time should be allowed. Allow sufficient recovery time (max 24 hours) until the fender regains over 95 % of its original height. Then, deflect it again at 0.33 to 1.33 mm/s. The test should be terminated once the deflection reaches the design deflection point or slightly beyond.

Fundamental Testing	Verification testing
a) It is recommended to conduct tests for each combination of fender configuration (length/diameter) and energy absorbing material (fender grades/densities). b) The specimens utilised for testing purposes should either be full-size fenders or scaled-down models, with the same length-to-diameter ratio.	a) To guarantee adherence to published performance criteria and customer-specific energy and reaction specifications, it is essential to carry out verification tests. b) It is crucial to establish a mutually agreed testing scheme between the manufacturer and the designer, clearly documented within the specification. The test protocol outlined in section 10.11.1 serves as a sound guideline for conducting verification testing. c) The specimens utilised for testing purposes should either be full-size fenders or scaled-down models, with the same length-to-diameter ratio. (see note above) d) The testing process should be conducted either in a third-party testing facility or in the manufacturer's facility with third-party witness or by employing an independent testing jig, as agreed upon by both the designer and manufacturer. e) Prior to testing, fenders must be free from any surface damage, cuts, or cracks. However, folding marks that may develop during testing are deemed acceptable. To meet the required performance standards, a fender must achieve at least catalogue specified energy (CV value) – manufacturing tolerance without exceeding the designated reaction force (CV value) + manufacturing tolerance at any point during the fourth compression. f) The number of samples selected for testing should be determined based on an agreed-upon sampling scheme between the customer and the fender manufacturer. In the absence of a specific scheme, a minimum of 10 % of the fender order should be randomly selected for testing, ensuring compliance with energy and reaction requirements. g) Inspectors should be granted access to witness the compression procedure in proximity to the testing press, while maintaining safety standards. h) Fenders should be tested under standard conditions, without utilising Velocity Factor, Temperature Factor, or Angle Factor adjustments to manipulate the performance results.
Notes: • Test Apparatus: The test apparatus should conform to the specifications outlined in section 10.4. • Sample size/type: For fenders with a diameter equal or smaller than 1,800 mm, actual fender elements manufactured for the specific project locations should be used. • For fenders larger than 1,800 mm diameter by 3600 mm long, the tests of a 1,200 mm diameter by 2,000 mm long or larger fender may be selected to determine the energy and reaction forces of the fender. The actual fender for a project and the test fender should both be constructed with the same materials, have the same general configurations of the ends and have the same skin thickness-to-diameter ratio. Scaling shall be conducted per the following equations: ○ Energy = test fender energy x diameter ratio squared x overall length ratio. ○ Reaction Force = test fender reaction x diameter ratio x overall length ratio.	

Table 10-10: Verification tests for foam fenders

10.12.2 Velocity, Angular and Temperature Factor for Foam Fenders

Velocity and temperature factors are often neglected for foam fenders; however, these effects can be significant in some cases. For higher capacity and smaller size foam fenders in

particular, these fenders can exhibit significant velocity factors. Temperature effects can be important for extreme temperature environments. Manufacturers who report correction factors for temperature and velocity effects on foam fenders can follow similar test procedures as for rubber fenders, with the following clarifications.

Refer to Section 10.7.3 to 10.7.5 for the application of the aforementioned principle in generating the velocity and temperature correction factors. Some modifications are required in the procedures to account for the differences between foam and rubber. The procedure outlined in section 10.12.1 serves as a sound guideline for conducting these tests.

Note that:

- Length of the specimens used for angular correction factors should be determined based on an L/D ratio of 1.5 to 2.0.
- For L/D ratios larger than this range, the manufacturer should determine the angle factors based on the specific requirements of the project.

Angular effects are not typically determined using dedicated testing, however can be calculated based on the relative deflection of each portion of the fender.

10.12.3 Durability Tests

Foam fenders, unlike their rubber counterparts, exhibit significantly slower compression recovery rates. This characteristic is attributed to the unique structural composition and material properties of foam. When subjected to compression, foam fenders take a longer duration to regain their original shape and height compared to rubber fenders. Consequently, conventional durability testing methods that rely on constant compression without allowing sufficient time for recovery fail to provide reliable insights into the built quality of foam fenders.

Recognizing the limitations of durability testing, use of the conventional durability tests is not recommended for assessing foam fenders quality. Instead, it is suggested that, in projects requiring durability verification, a customised test protocol should be collaboratively established between the fender manufacturer and the designer/owner.

10.12.4 Skin Thickness and Foam Density Verification Testing

The energy absorption capacity and reaction force of foam fenders are significantly influenced by the density of the foam core and the thickness of the Polyurethane skin. Moreover, the appropriate skin thickness also contributes to the abrasion resistance of the fender.

To ensure the quality of the supplied fenders, it is recommended to conduct performance verification tests in conjunction with tests verifying the skin thickness and foam density.

In certain projects where conducting the performance test is impractical, the foam density verification test can serve as a suitable substitute for performance verification.

The following protocol shall be followed:

1. The project specification should include a specific requirement for the skin thickness. This measurement should be calculated based on the reaction force, friction coefficient of the Polyurethane skin, and an additional allowance for normal wear. The necessary data for these calculations should be obtained from the manufacturer.
2. A representative sample of 10 % of the fenders from each production lot, or an agreed-upon number of samples between the manufacturer and the client, should be selected for testing purposes.

3. To assess and examine the thickness of the elastomer and the placement of the reinforcement layer, a core drill with a diameter ranging from 6 mm (minimum) to 51 mm (maximum) should be used to extract a core sample from the fender skin.
4. In addition to assessing the skin thickness, the drill may be extended up to 40 % of the fender diameter to collect foam samples for verifying foam density. Care should be taken during this drilling process to minimise heat generation, which can potentially alter the foam density.
5. Skin testing should be conducted at two specific locations on each tested fender: the mid-body and one conical end.
6. The skin thickness should be measured and recorded on the extracted core samples, while paying attention to the placement of the reinforcement layer.
7. The measured skin thickness should fall within a range of specified thickness with a tolerance of -10 %.
8. Fenders that do not meet the specified skin thickness measurement, falling below the -10 % threshold, should be rejected. In such cases, the manufacturer should perform thickness tests on an additional 10 % of the production lot's fenders.
9. Rejected fenders should be replaced with fenders that meet the specifications, and skin thickness testing should be conducted on the replacement units as well.
10. The testing process should be witnessed either by a certified, independent inspection agency or as agreed upon by the manufacturer and the client. The manufacturer should provide a minimum 10 working days' notice before conducting the skin thickness tests.
11. After completing the skin thickness testing, the core holes should be patched using elastomer of the same composition and thickness as the specified elastomer skin. Nylon reinforcing is not necessary for these core patches.
12. Prior to delivering the fenders to the job site, it is mandatory to conduct a minimum of two skin thickness tests for each test fender.

10.12.5 Fender Pull-Through Test

This optional test aims to evaluate the ability of the end fittings and central chain member to withstand longitudinal pull forces along the axis of the fender. The test is considered a failure if the central chain member, end fittings, or reinforced skin are unable to resist the fender's design safe working end pull load.

Test procedure:

1. Compress the fender to 40 % of its original diameter (60 % compression). If a scaled-down model, is used for testing, also use it for this test.
2. Attach a hydraulic cylinder to the fender end fitting bracket, incorporating a load cell in series to accurately measure the applied load.
3. Gradually increase the load on the hydraulic cylinder until the required design safe working end pull load is attained. Maintain this load for approximately 5 minutes.
4. Gradually reduce the load to rest, and carefully inspect the fender for any signs of damage.
5. The test will be deemed unsuccessful if there is visible evidence of permanent damage, cracking, or tearing in the fender body or steel end fittings.

10.13 Tests for Accessories

10.13.1 Fabricated Steel Structures Air-Leakage/Pressure Test Procedure

This test should be conducted as part of the panel structure manufacturing process.

1. Drill hole/s of an appropriate diameter and install M16 or suitable connections on the rear side of the panel.
2. Install pipe nipple-valves and connect the air line.
3. Apply air pressure to the panel, maintaining a pressure of 1.0 kg/sq.cm (100 kPa) (Higher or lower air pressure values may be agreed upon between the customer and the manufacturer).
4. Spray a soapy water solution onto all external welds.
5. Maintain the pressure for a duration of 15 minutes (Higher or lower time may be agreed upon between the customer and the manufacturer).
6. Inspect the panel for any indications of leaks and subsequently depressurise the panel.
7. For each identified leak point, grind out at least half of the existing weld depth and perform a reweld.
8. Repeat the aforementioned steps until no leaks are observed.
9. Remove the pipe nipple and seal the valve using a gasket and plug bolt.

10.13.2 NDT (Non- Destructive Testing – All Fabricated Steel Parts)

Description	Acceptance Criteria	Type of Record
Die Penetration (DP) or Magnetic particle Inspection (MPI) test	As per relevant standard and approved procedure	DP Report or MPI report
Ultra-Sonic (UT) Testing	As per relevant standard and approved procedure	UT Report
Note: Designer should decide and specify the type of test applicable for the steel components.		

Table 10-11: NDT test of all fabricated steel parts

UHMW-PE Resin and Pads

No.	Property	Test method	Unit	Requirement (virgin)	Requirement (regenerated)	Comments
1	Density	ISO 1183-1	g/cm ³	0.920-0.945	0.930-0.955	
2	Tensile strength at yield	ISO 527-2	MPa	17 - 21	Min. 17	
3	Elongation at break	ISO 527-2	%	Min. 150	Min. 150	
4	Hardness	ISO 868	HDD	60-65	60-70	Shore D
5	Double notch Charpy Impact Strength	ISO 21304-2	KJ/m ²	Min. 120	Min. 70	Average of four samples
6	Abrasion resistance (sand slurry test)	ISO 15527	ml/g	Max. 110	Max. 150	Average of min. two samples. Sample preparation: ISO 11542
7	Mass Melt-flow rate (MFI)	ISO 1133 (@190°C, 21.6 Kg)	g/10 min	0 to 0.1	0 to 0.3	Samples from the pads can be used for testing.
8	Friction coefficient (Static, Dry)	ISO 8295 (with Steel)	—	Min. 0.2	Min. 0.2	
9	Molecular weight	ISO 1628-3	g/mol	~ 5.0 x 10 ⁶	~ 2.0 x 10 ⁶	For selection of material, testing is not needed.

Notes:

1. Pads are produced through the compression moulding process, using high temperature and pressure, with granular UHMW-PE resins.
2. The supplier is required to furnish a material testing certificate, encompassing properties such as density, tensile strength, elongation, abrasion resistance, hardness, and Charpy impact strength. The material certificate should comply with the specifications of a 3.1 certificate as per EN 10204 or an equivalent standard.
3. To ensure quality assurance, the end user has the option to validate the quality by obtaining samples from randomly selected pads and conducting tests for density, impact strength, abrasion resistance, and MFI (melt flow index) at an independent third-party laboratory.
4. The UHMW-PE moulder should provide the specifications and quantities for the moulded products. The frequency of testing should involve conducting one test per production lot or adhering to an agreed sampling scheme established between the customer and the manufacturer.
5. Regenerated UHMW-PE presents a potential sustainable alternative to virgin material. However, for heavy-duty applications such as belted vessels, the use of virgin material is recommended to ensure optimal performance and durability.

Table 10-12: Physical Properties of UHMW-PE resin and pads

11 INSTALLATION, INSPECTION AND MAINTENANCE

The performance of a fender system throughout its service life is dependent on the correct installation and maintenance procedures specified by the designer and suppliers. It is essential that the end users, under guidance of suppliers, ensure that these are implemented.

Considering the loads and the demanding operating environment of the various components of the fender systems, these cannot be maintenance-free.

A lack of fender system maintenance is commonplace on numerous facilities. It is important that due consideration is given at the design stage to this key topic and a low-maintenance design should be favoured whenever economically possible.

11.1 Installation

11.1.1 Handling and Installation

It is essential that fender systems are always handled correctly to avoid any damage that could affect performance. In advance of the delivery of the fender systems, the fender supplier is expected to provide detailed handling and installation guidelines, to ensure that the installation contractor may develop the relevant detailed method statements accordingly.

Depending on the fender system complexity and/or the teams experience, the installation contractor may also consider training of their staff by the fender supplier for the assembly and installation of the first fender systems.

Checks shall always be performed upon receipt of the fender systems on site to confirm that no damage occurred to the fenders and their accessories during shipping.

For large fender systems that require crane lifting, it is essential that the suppliers include in their design the necessary lifting lugs for safe installation.

Upon completion of the installation, and depending on the project size or complexity, the end user may invite the fender supplier to visually inspect the works and issue an acceptance report confirming that the fender systems have been adequately installed and free of damage, ahead of the issue of any warranty or insurance certification.

11.1.2 Safety and Accessibility

Particular attention should be given at the design stage of the berthing structure to access each fender system location safely, practically and economically. This should consider installation, inspection, maintenance, and replacement activities and should include consideration for suitable sized cranes.

For fenders mounted directly onto the berthing structure (such as cone or arch fenders), installation contractors typically use suspended temporary structures for safe access to the fender rubber and chain fixing locations. End users and port facility operators may consider retaining these work platforms after installation, to benefit from this safe system of access for inspection or maintenance activities of the fender systems in the future.

11.2 Spares and Storage

11.2.1 Spares

Due consideration should be given for the provision of spare parts for the fender system. This may typically consist of items subject to wear and tear or accidental damage, such as:

- Facing pads
- Facing pads fixings
- Chains
- Chain weak link (when installed)

It is recommended that the end user, with the support of the fender systems designer and the supplier, conducts a risk assessment to identify the recommended list of spare parts. These spare parts can then be included within the purchase order for the complete fender system.

For facing pads, end users may prefer to order large sheets to cut out the replacement facing pads that are needed, as fender panels are typically fitted with various pads sizes depending on their position.

For critical facilities with a limited number of units installed, end users may consider ordering a spare fender system; the same would need to be adequately stored, and the maintenance team attention drawn on the durability aspect.

However, for rubber fenders, considering the ageing process, it is not recommended to order spare units; on facilities with a large number of fenders systems installed, such as continuous quays, damaged units could be replaced with end-units, with limited operational impact during the replacement units procurement process.

11.2.2 Storage

Rubber elements can be stored for a period that depends on their composition and the storage conditions, which are of the utmost importance for fender systems. It is therefore essential that fender manufacturers provide detailed packing and storage guidelines for the products supplied in order to minimise unwanted changes in properties or surface deterioration. This guidance (including the recommended storage durations) should be developed according to ISO 2230 "Rubber products - Guidelines for storage" or an equivalent standard.

In the guidelines provided by the fender supplier, the end user's attention should be drawn to the natural hardening process of the fender rubber during delayed usage, which will often result in a temporary hardening; while the resulting higher reaction forces would not exceed hard berthing conditions, it is essential that the first berthings against stored units are carefully controlled to remain within the overall safety margins.

The same requirement applies to unused installed units or units where installation has been delayed at construction stage.

11.3 Inspection and Maintenance

Any fender system will deteriorate with time due to ageing, environmental conditions, operational loads and/or unforeseen events such as hard or accidental berthing events. These could affect the performance of the fender system and increase the risk of failure. They may also lead to additional incidents and/or disruption to operations of the berth. As such, the

routine inspection and maintenance of fender systems is essential to ensure their durability and performance throughout their service life and potentially reducing life cycle costs.

An effective inspection and maintenance regime should consider as a minimum:

- 1 The age of the equipment.
- 2 The criticality and occupancy of the berth.
- 3 The availability and accessibility of the spare parts.
- 4 The environmental conditions and the operating philosophy.

Fenders suppliers should provide, along with their products, a manual with typical maintenance recommendations as well as detailed guidelines for parts replacement.

Given these parameters vary from site to site, definitive inspection and maintenance intervals can be difficult to recommend. Due consideration should be given to all these factors while developing the inspection and maintenance programmes, in addition to any specific local regulations or port authority requirements.

This section therefore provides general guidance only. Operators may refer to more detailed documents to develop their specific inspection and maintenance plans, such as:

(CDIT; maintenance, 2019): "Japanese Guidelines for the Maintenance of Rubber Fenders Systems".

PIANC report 103 (PIANC, 2008): – "Life Cycle Management of Port Structures".

The upcoming PIANC MarCom WG 233 guidelines: "Inspection, Maintenance & Repair of Waterfront Facilities".

(ASCE, 2015): "The American Manual on Waterfront Facilities Inspection and Assessment".

11.3.1 Inspection

The prime causes of fenders system deterioration are harsh environment, excessive berthing loads and degradation of facing panels and chain systems; deterioration processes accelerate when these are neglected. Inspection regimes typically vary between ports; the frequency of the inspection will depend on the port use and port exposure as well as on the port's inspection and maintenance philosophy. As a minimum, an annual inspection of all fender systems should be carried out, irrespective of port operations.

Regular inspection of fender systems provides a mechanism for early detection of fender system deterioration. This enables planned maintenance to be carried out to ensure safe operation and minimise berth downtime. As such, inspections should be conducted at recurring intervals and before/after certain activities, considering the potential failure modes of each of the components; these would include the fender and all its accessories and are not limited to the following:

- Cracks
- Signs of over-compression
- Unit droop (sagging) due to loose support connections
- Non-vertical panels
- Loose chains
- Broken chains / fuses
- Missing fixings / fittings

For systems with a low elevation, these inspections are best undertaken at low tides; if the tidal range is large and mobile suspended equipment is not available, boat access will need to be considered.

Table 11-1 lists the common failure modes for typical fender components, affected by environmental conditions such as splashing water, UV exposure, extreme temperatures, marine growth, etc., that cause deterioration of fender components over time. Operators should follow the fender supplier's recommended inspection regimes, adjusting the same to the specific site conditions to arrive at a cost-effective inspection regime.

Component	Deterioration / failure modes
Rubber body	Tear/chipping Cracking/splitting Ozone cracking Burn Sagging Steel inserts separation Permanent distortion/deformation Hardening
Steel frames and structural members	Atmospheric/splash zone corrosion Weld crack Dent/impact damage General deformation Changed panel alignment Loss of air tightness
Panel facing (low friction)	Wear/cuts/cracks Burn Missing pad Loose/missing/protruding fasteners
Panel facing (timber)	Decay/rot Marine borer attack Splitting/cracking Chipping/uneven facing Loose/missing/protruding fasteners
Pneumatic fenders	Puncture/deflation Cracking Missing/damaged tyres Excessive wear of outer skin Loose end fittings
Foam fenders	Puncture Permanent deformation Loose end fittings Excessive wear of outer skin
Fender chains and fittings	Severe corrosion Localised wear (diameter reduction) Missing/damaged fittings Broken fuse (weak link) Damage to the bar (cylindrical)
Fixings – general	Corroded bolts/nuts Loose/cut Missing bolts/nuts

Table 11-1: Common failure modes for typical fender components

Hard berthing, extreme storms, tsunamis or earthquakes may damage rubber fender components and this potential damage may lead to unexpected safety and environmental issues as well as financial consequences. It is therefore recommended to carry out a thorough inspection of the fender system before resuming berth operation after abnormal events.

11.3.2 Maintenance

The results of each inspection should be evaluated to determine the next recommended course of action. In addition to the supplier's recommendations, which should be adhered to so as to maintain any warranty validity, there are numerous guidelines available to assist in the

evaluation of the deterioration of various components. Recommended remedial actions are also provided as documented (CDIT; maintenance, 2019). Operators should however also refer to any specific local regulations or Port Authority requirements.

As a general approach, irrespective of the fender type or specific findings, attention should be given to the following elements:

- Structural steel members (such as panels) are generally protected from corrosion by coating systems, in addition to sacrificial material thickness and/or cathodic protection. It is recommended that any coating damage is promptly repaired to maintain the overall structure durability as exposed steel can corrode rapidly in an exposed marine environment, especially within the splash zone.
- In case of significant corrosion and/or a major overhaul event, closed box panels should be pressure-tested to verify their integrity and prevent the risk of internal corrosion.
- Low friction facing pads are subject to wear and tear. Regular maintenance and replacement should be considered, as surface degradation increases friction forces with vessels over time. Attention should also be given to the minimum pad thickness, below which the fixings are exposed, resulting in possible hull damage and/or risks of sparks. This is of utmost importance at berths where petroleum products or hazardous cargoes are handled. Partial replacement of low friction pads shall take into consideration the adjacent pads thickness to avoid localised high stresses and /or scraping the paint of vessel hull by the exposed edges of the new pads.
- Chain corrosion impacts the chain link diameter and weakens the chain capacity. Some cost-effective designs may consider chain replacement during the fender system service life. In such cases, particular attention should be brought to the chains integrity, especially in the absence of any safety features (such as a weak link).
- Under certain conditions (e.g. wave or vessel engine induced vibrations), fixings and fittings could become loose and may need to be tighten up. If not identified and dealt with at design and fabrication stage, loosening of components can be addressed using non-permanent thread-locker.
- In areas with a risk of severe ice formation, ice should be regularly cleared from fenders systems as it adds weight that could affect their overall performance and prevent the fenders from being compressed. While doing so, care should be taken in order to prevent damage the fender components (Refer to Chapter 8.9).

For pneumatic fenders particular points of attention are as follows:

- Check the air pressure on regular basis and adjust as required (after identification and repairs of the possible leak source).
- Loose or worn rubber sleeves can damage the net of the fender; in such cases, fibre rope or fibre jacket can be used temporarily until the rubber sleeves are replaced.
- Apply grease when necessary to ensure the proper rotation of the swivels.
- If present, inspect and clean the safety valve on annual basis (as a minimum).

For foam fenders, the following recommendations would apply:

- Inspect for damage to the fender skin; cuts, gouges or cracks should be repaired as soon as possible; repair kits are typically available from suppliers for minor damage.
- Cuts that go through the skin to reveal the foam core need immediate repair.
- The shape of foam fenders, especially when used on (semi-)permanent moorings.
- Inspect for marine growth that developed on the fender body; it should be removed when appearing to be detrimentally affecting the flotation, stability or rotation of the fender;

pressure washing/ scraping shall however be used cautiously to avoid damage to the fender skin.

Subject to design berthing loads not being exceeded and adequate maintenance being performed, a fender's service life should meet the minimum design life requirements. Should the same be exceeded, and although rubber fenders may appear to be in sound condition, particular care should always be taken in case of extended service life, as rubber hardens over time and increases the fender reaction forces. This may lead to damage to both the berth structure and the vessel hull structure. Hardness tests using a 'durometer' can be performed to compare the current fender hardness values against the original fender hardness, to detect potential ageing and the potential need for replacement. Alternatively, some fender units can be removed for load testing, however this is likely to incur significant additional costs.

11.3.3 Emerging Technology

Inspections are traditionally performed by port and terminal maintenance staff by visually inspecting the installed fender systems. This method can be time-consuming and can be subject to reading/ recording errors or omissions.

Operators may therefore consider some emerging technologies that now have the potential to capture real-time data, provide instant system status, analyse the operational performance, deliver historical records in order to help improving berth utilisation, which can extend the fender system life and reduce the overall life cycle costs.

These systems typically consist of sensors that can be easily installed on or near the fender system, possibly without the need for power and data cabling. These sensors can measure numerous parameters such as vessel berthing speed, fender deflection, compression duration, berthing angle, fender shear movement, panels air tightness, etc. The data can be either stored locally or transferred in real-time to the cloud for storing and remote processing. The analysis of this data can create greater insights into the fender system performance over time for operators, maintenance teams and future design considerations.

12 SUSTAINABILITY OF FENDERS

In 2015, the United Nations set out the Sustainable Development Goals. The Sustainable Development Goals are the blueprint to achieve a better and more sustainable future for all. In total there are 17 Sustainable Development Goals and they address the global challenges we face, including for fenders relevant goals like **'Industry, Innovation and Infrastructure'**, **'Reducing Inequality'** and **'Responsible Consumption and Production'** (UN, 2022).

Sustainability in general has seen a growing focus in the last decade and has a priority within PIANC. Reference is made to the 'PIANC Declaration on Climate Change', in which PIANC recognises the importance of the climate change challenge and said it will actively pursue the sustainable future of the waterborne industry.

PIANC Report EnviCom WG 150 (PIANC, 2014) creates awareness of the advantages of implementing a green port philosophy and what this philosophy means at present for ports and port authorities around the world and community support for port growth. This will be achieved by supplying tools and guidance that show how proactive environmental measures and strategies can contribute to obtaining consent for future operations and developments, how opportunities can be created through own initiatives (thereby remaining ahead of legislation) and how green growth can be realised.

A growing number of ports have a sustainability programme and require their investments and new projects to be sustainable and meet their internal and national minimum requirements. When designing or acquiring fender systems the full life cycle and recyclability of fender systems to work towards sustainability development goals should be considered.

12.1 Circular Economy with Rubber Fenders

Many countries are taking steps towards a full circular economy. This means that a product should be 100 % recycled and reused in the same or similar applications. With the current state of technology achieving 100 % recycling is a significant challenge for rubber products in general and thus fenders. However, given the public importance of the circular economy and the responsibility that manufacturers have, this is something that the fender (rubber) industry should strive for. Furthermore, end users have the responsibility to require from manufacturers a certain level of recyclability of the fender system, because manufacturers generally follow the market demand.

When the sustainability of a fender system is considered all the relevant UN Sustainable Development Goals should be taken into account. The full life cycle of the fender system needs to be looked at, from cradle to grave which includes material sourcing, manufacturing and recycling all fender components after its service life.

12.2 Carbon Footprint

Greenhouse gas (GHG) emissions are the main contributors to global warming and climate change. Anthropogenic carbon dioxide (CO₂) emissions are a leading cause of climate change. A carbon footprint is the total **greenhouse gas (GHG) emissions** caused by an individual, event, organisation, service, place or product, and it can be expressed as **carbon dioxide equivalent** (Carbontrust, 2009), abbreviated as CO₂e. The CO₂e of rubber production depends on various factors such as the type of rubber, the production method, and the location of the production.

Reduction of the carbon footprint is essential for climate actions. Fenders being one of the essential components of infrastructure for waterborne transport, the fender industry should develop programmes with clear goals to reduce their carbon footprint and users can play a catalytic role by selecting least carbon intensive products.

The carbon footprint of rubber fluctuates depending on the sourcing and production method (refer to Section 12.3 and 12.4) and carbon footprint reduction starts with understanding the current footprint level. Fender manufactures can consider to perform a Life Cycle Analysis (LCA) and obtain a 3rd party certified Environment Performance Declaration (EPD) for its products. For instance, the average CO₂e of natural rubber production in Thailand was found to be around 6.05 kg CO₂e/kg of rubber, while the CO₂e of synthetic rubber production ranges from 3.85 to 15.63 kg CO₂e/kg of rubber, depending on the specific type of synthetic rubber and the production method (Chaikoon & Faungnawakij, 2016). The carbon footprints are only including emissions during production of rubber.

Reducing carbon footprint should consider the full life cycle: Raw materials, production, logistics and recycling. Sometimes CO₂ emissions are unavoidable; in such cases manufacturers could consider compensation for residual CO₂ in carbon dioxide balancing projects.

12.3 Rubber Sourcing

Sourcing sustainable raw material is a general challenge for the rubber industry. The main two types of virgin rubber that is used for fender production are natural rubber (NR) and synthetic rubber (SBR) with each their own challenges:

- i. Natural rubber may come with irresponsible farming methods, deforestation and human right abuses. Global platforms for sustainable natural rubber fenders (GPSNR, 2022) are being developed and this is something PIANC encourages to be considered when specifying/ purchasing fenders.
- ii. SBR is petroleum-based and therefore not considered to be sustainable. No good alternatives are available since butadiene is a key intermediate in the production of synthetic rubber. Rather than sourcing it from petroleum, innovative alternatives are being developed however are not currently available (ACS, 2022).

Generally, a blend of NR + SBR is being used, however not all applications require this blend and a NR only compound from a sustainable source can be considered.

Besides from the most common source 'rubber tree' (Hevea Natural Rubber) NR can also come from alternative plants such as Russian Dandelion and Guayule providing a sustainable alternative to synthetic rubbers. These alternatives have been used i.e., in car tyres however the use is limited due to different material properties and limited availability (considering the global demand) (Sarkar & Bhowmick, 2018).

Besides rubber, a fender also consists of other ingredients (refer to Section 9.2). More sustainable alternatives are being developed and used, such as recycled carbon black and sustainable oils. Today the application of these alternatives is limited due their impact on the final rubber product properties (Sarkar & Bhowmick, 2018).

12.4 Fabrication

ISO14001 sets out the criteria for an environmental management system that manufacturers can implement and be certified to. It maps out a framework that a company or organisation

can follow to set up an effective environmental management system (ISO14001, 2015). It is recommended that this is considered as a requirement for manufacturers of fenders and all components of a fender system.

The impact of the ISO14001 is limited and the ISO14000 series include more ground breaking compliance standards however compliance to these standards take time and the industry in general have yet to adopt these.

12.5 Fender Design and Materials Selection

The end user, designer and fender manufacturers can make a positive impact on sustainability through appropriate and long-lasting design and fender selection criteria. When selecting and designing fender systems the following aspects are worth considering:

- Minimising materials by avoiding over-designing fender systems; reference is made to Chapter 5 'Berthing Energy' & Chapter 6 – 'Fender Design'.
- Considering in the fender design process the GHG emission budgets of steel and rubber, and particularly over the life cycle of the infrastructure which might include several fender upgrades. The rubber carbon dioxide equivalent is comparable to the carbon footprint of crude steel production at around 1.8 kg CO_{2e}/kg (Chaikoon & Faungnawakij, 2016).
- The future design requirements: it is more sustainable to buy appropriately sized fenders once than upgrading every few years.
- Measuring berthing speeds to avoid unneeded fender replacement or over/under-designing. Also installing berthing aid systems to influence the design berthing speed such that fenders rarely need replacement.
- An efficient fender (e.g. cone fender system) that will absorb more energy with less rubber than a less efficient fender (e.g. a cylindrical fender system).
- Materials that can be easily re-used or recycled (e.g. the steel fender panel).
- Optimise fender rubber grade. A smaller but stiffer fender will result in less rubber, but to meet the hull pressure requirements it will result in a larger steel panel.
- Various fender solutions such as foam fender, pneumatic fender, rubber fender at its full life cycle including recycling or re-use.
- Increase design life of the fenders by specifying a high-quality rubber fender as well as other high-quality materials that help to extend the life of the fender system.
- When selecting materials, the wear and tear and impact on the environment should be considered. A high-quality UHMW-PE will wear less than a grade with a low wear resistance that needs to be replaced earlier resulting in more microplastics finding their way into the environment.
- Avoid harmful materials wherever possible, i.e., avoid coal-tar epoxy-based paint systems (which are already forbidden in most countries) that can be easily replaced with other (epoxy) paint systems.

12.6 Recycling

Fenders are found in almost every port and are commonly used for several years. Due to changed operational requirements, changed design criteria or wear and tear of fenders, berths are upgraded and fenders are replaced on a regular basis.

A sustainable way to deal with old fenders is to reuse them on a new berth. This is something that can be considered for fenders that are not yet at the end of their service life, i.e. in cases where fenders need to be upgraded to accommodate a different range of vessels.

After an initial visual examination of the fenders to understand their condition, a more in-depth examination can be made by checking the original test records, understanding the frequency of use, and checking surface conditions for cracks, ozone cracks and damage. For berths in a more demanding situation a performance test is recommended. Refer to section 6.4.17 on ageing of fenders and performance loss of a fender during its life and to Chapter 10 on testing of fenders.

In most cases a used fender is found unfit for re-use and needs to be recycled. A fender system consists of various components made from various materials. Commonly used materials in fender systems are rubber, foam, PU, steel and UHMW-PE. These materials all have very different possibilities in recyclability.

12.6.1 Current Practice of Fender Recycling

Ports struggle with the disposal of the old fenders in a sustainable way. The current infrastructure for recycling rubber products is focused on thin rubber products such as car and truck tyres and conveyor belts. Due to the size of the fenders and the large steel inserts, it is difficult to process old fenders and grind them into small useable particles.

In practice, ports either store their old fenders or they are used for landfill or taken by waste disposal companies that burn them as a fuel (also called Tyre-Derived Fuel (TDF). This burning is typically done (or should be done if this form of disposal is chosen) at extreme high temperature to limit the environmental impact and maximise energy gain from burning. Due to the high level of oil in rubber and other additives that burn easily burning rubber generates a large amount of energy. Nevertheless, burning fenders is not considered as a sustainable solution and is certainly not circular.

12.6.2 Rubber Recycling

There are various solutions for processing used rubber to a re-usable product. The costs and economics, use of the end product and demand vs availability of end product decide the preference of recycling process.

Typically, the following types of recycled rubbers are available:

- Ground rubber crumbs. Reclaimed mechanically or via waterjet. Rubber crumbs are larger in particle size than powder which means that they are technically less usable or the use is limited to less demanding applications. This process for reclaiming is more cost-efficient compared to the other processes.
- Ground rubber. This is achieved cryogenically, or via waterjet or a mechanical process and results in powder or small granules. Typically, a rubber powder with a small particle size is preferred as this increases the usability by adding a small percentage in new applications or a higher percentage in less demanding applications. This process is expensive and requires more energy.
- Reclaim by mechanical & chemical process. This results in uncured rubber slabs, well usable in all sorts of applications however the process is expensive and requires more energy and chemicals.

The use of recycled rubber in general is limited despite the large availability of recycled rubber. Recycled rubber has a great impact on the rubber properties of the final product (reference is made to Section 9.2.3). The adverse effects on properties in reality mean that the life expectancy with increased recycled content is greatly reduced. The reduced life expectancy makes the use of recycled rubber less sustainable.

This means that a recycled rubber compound can only be used in small percentage in new products, in a higher percentage in low-quality applications or as additives in various other application areas, for example as a filler of concrete and asphalt.

Due to this, rubber is currently not truly recyclable, although it is re-usable in different forms and applications. This results in that reclaimed rubber is not really being recycled, as intended with a fully circular economy, however it can be used for low duty applications or as filler/ additive. Examples of low duty applications are rubber playground mats, sports surfaces, flooring, etc. Reclaimed rubber is used as filler/ additive in cement and asphalt.

Pneumatic fenders have an additional challenge where the pneumatic fender body is reinforced with nylon fabric. It is hard to impossible to separate the nylon fabric from the rubber meaning that the reclaimed rubber crumbs or granules will be polluted with nylon. This strongly limits the further use of this type of fender.

Another form of converting rubber waste is Pyrolysis. Pyrolysis is the thermal degradation of the organic components of the rubber in a reactor, at typical pyrolysis temperatures of about 500°C. With Pyrolysis a large amount of waste rubber can be disposed of while at the same time obtaining high-value products, such as oil, chemicals, carbon black, and fuel gas with high calorific value; and in addition steel can be recovered (Han, Han, & Chen, 2023) (Akbas & Yuhana, 2021).

12.6.3 Foam Fender Recycling

A closed cell foam fender is a fender that can be refurbished. The skin can be repaired or an entire new skin can be applied over the old foam. The closed cell foam has a very high durability although some performance loss after refurbishment should be considered. By adding new hardware, the fender is like in new condition. However, logistics sometimes prevent this to be an economical solution.

Biodegradable foams are being developed and offered however have not found their way yet to fenders due to the degrading nature of the material and additional costs.

- i. Foam
Typically, EPE (expanded polyethylene) type of foams are being used for foam fenders. EPE foam can be recycled by hot melting it to granulate PE pellets which can be used in various applications (Greenmax, 2022).
- ii. PU Skin
PU in general is recyclable (i.e., mattresses) (Americanchemistry, 2022). However, the PU skin of a foam fender has similar recycling challenges than rubber with a limited infrastructure to recycle this kind of PU waste. The nylon reinforcement in the PU skin results in an additional challenge. When the skin is being scrapped the nylon reinforcement pollutes the scrapped material limiting a wider use of it. Currently this is being disposed via channels similar to rubber recycling. Since the foam fender skin is relatively thin compared to a solid fixed rubber fender the amount of disposal material is less and easier to process.

12.6.4 Steel Recycling

Fender panels as well as various other fender components are typically made from steel. Steel in general is the most recycled material, even more than all other material combined. Approximately 86 % of steel in general is currently being recycled (Steel, 2022).

Steel is 100 % recyclable and generally new steel contains up to 25 % recycled steel (Recycle, 2022). The use of scrap steel saves up to 74 % of the energy needed to make steel from virgin materials (Recycle, 2022).

12.6.5 UHMW-PE Recycling

UHMW-PE becomes available for recycling at the end of its lifetime which can be at the end of the lifetime of the fender systems or, since UHMW-PE is a wear item, at some point during the lifetime of a fender system.

UHMW-PE can be easily reprocessed into a re-usable material. Pure UHMW-PE can be reprocessed to a regenerated grade of UHMW-PE that can be used for fender application again. The material properties generally are less (including wear resistance) compared to virgin UHMW-PE. A positive exception is the coefficient of friction that is the same for virgin as for regenerated UHMW-PE. Generally regenerated UHMW-PE is broadly accepted for fender applications.

Generally regenerated UHMW is produced from recycled material that comes from production scrap at the manufacturing ensuring them full control of the quality of the recycled raw material.

Reusing recycled material that comes from disposed UHMW-PE that has reached the end of its life cycle is generally not used for regenerated UHMW-PE. Since in most cases the original quality of the disposed UHMW-PE is unknown, manufacturers are not keen on producing regenerated UHMW-PE with this recycled material since they do not know the origin and quality of the material. Hence, they cannot be sure that the quality of the regenerated UHMW-PE manufactured with this disposed UHMW-PE will meet their standards.

Still UHMW-PE fender pads can be recycled and re-used in the plastic industry however then often is re-used for low-spec applications (outdoor plastic furniture, traffic poles) or used in other industries like in the paper industry.

12.7 Recommendations for Fender Sustainability

The increasing global focus on sustainability and the need for real change also makes its way to ports and the manufacturing industry. Rubber manufacturing including fender manufacturing still have various challenges to face, such as recycling of their products and mapping their carbon footprint.

The fender industry together with the end users as well as designers and other stakeholders should take the lead and challenge each other and co-operate to come to the most sustainable solution for their projects and develop ways to develop, manufacture and dispose of fenders in the most sustainable way possible.

When the sustainability of fender systems is being considered for a project than all parties involved in the process of selecting, designing, purchasing and manufacturing the fender systems should make their contribution to design and manufacture the most sustainable fender the industry can do at the moment.

Although there are limitations, as it currently stands there are things that already can be done by all stakeholders in the process of fender system selection, designing and manufacturing. By being smart with materials and engineering, the sustainability of the fender system can be improved and the impact on our environment can be reduced.

13 SPECIFICATION WRITING

13.1 Responsibility for Providing Information

Accurate and complete specifications are important to achieve an economical and durable fender system complying with the required performance.

Fender specifications should contain sufficient information to carry out a design, manufacturing and testing of the fender system. As a minimum, a fender system specification should contain the critical information required by fender manufacturers, designers and end users. It is recommended that the required information is provided as given in Table 13-1:

Fender System Information	Purchaser (designer, contractor, end user, port authority)	Supplier (contractor, manufacturer)	Chapter
General port and quay information	X		-
Water levels, depth and met-ocean conditions	X		-
Fender system selection	X	X	2, 0
Design vessel information such as displacement, draught, length, beam, hull characteristics, etc.	X		0
Consequence class and design life	X		4.6
Vessel berthing and navigation consideration, berthing velocity and angle, tug assisted or not, etc.	X		5
Manufacturing, durability and quality requirements	X		0
Testing requirements	X		0
Additional design specification notes (optional)	X		-
Preliminary design (optional)	X		-
Final design including rubber fender properties		X	-
Installation, maintenance and storage manuals		X	0
Recycling, rubber sourcing, carbon footprint, etc. note	X	X	0
Approval of final design criteria	X		-

Table 13-1: Required fender system design information

It is recommended a Basis of Design document in line with Chapter 4 is produced and agreed with all parties before the fender system design commences.

13.2 General

13.2.1 Qualification of Supplier

The tender specification should clearly indicate the required qualifications of the whole supplier chain. These may include the following:

- Be able to predict and verify product performance.
- Track record with clients' reference letters and details of production facilities.
- Valid product liability insurance.
- Records of any claims, if requested.

Further information on the qualification can be found in Section 9.1.

Supplier chains with no track record (new suppliers/manufacturers) should provide additional documentation prior to qualification, such as general company information, production line process, quality control, source of materials, prior testing/rubber verification, etc.

13.2.2 Standards and Codes of Practice

All relevant national or international standards, codes of practice, guidelines and other specified standards should be referenced and ranged clearly.

13.2.3 Quality Control

The tender systems manufacturer should apply a certified system of Quality Management which conforms to ISO 9000/9001 or a recognised equivalent.

If requested, the supplier should present an inspection test plan, refer to Chapter 9.1.

13.2.4 Submittal Requirements

Specification should clearly outline required documentation to be submitted at each stage in the supply process, i.e. tender submission, prior to fabrication, post-fabrication and post-delivery.

As a minimum, it is recommended that the tender documents should require the following information:

- Experience - experience of executing similar projects, past performance with similar projects.
- Design capability - proven design experience with projects of comparable size.
- Manufacturing methodologies - design solution and compliance with the project requirements and specifications.
- Manufacturing facility - indicate where tender system will be manufactured.
- Programme and testing regime - realistic planning and testing sequencing.
- Source of rubber - natural rubber, preferably from a sustainable source. Proposed rubber compound, i.e. natural, synthetic, mix, etc.
- Type approval testing - manufacturers catalogue.

Further submittal requirements (after contract award) should be detailed in the specification. These submittals may include:

Fender unit

- Quality assurance certification and all documents in English language (or local language if accepted by the purchaser).
- Actual location of manufacturing.
- Method and date of manufacturing.
- Size and rubber grade of fender unit.
- Date of delivery.
- Verification test results.
 - Physical property test results (hardness, tensile strength, elongation at break, etc.).
 - Chemical composition for rubber traceability (TGA – polymer, carbon black, ash as a minimum). The values are manufacturer specific and generally should not be posted in the specification.
 - Load deflection graphs
 - Durability test results (if applicable)
- Confirmation that each fender unit has been subject to a break-in cycle, if required (see Chapters 10.4 and 11.3.2).
- Foam density and skin thickness (nylon reinforcement check) if applicable.

Fender panel and accessories

- Calculations showing the requirements and designs of the fender panel. This should include sketches of various anticipated loading conditions, formulae, and design principles.
- Confirmation and results of the fender panel pressure test.
- Coating materials and application methods, dry film thickness (DFT) certifications.
- Calculations and drawings showing the requirements and designs of accessories such as chains, anchors, panels, etc.
- Internal and/or third-party records of low friction UHMW-PE facing properties including the friction coefficient.

Other

- Design computations that indicate the fender system meets the project specified criteria.
- Calculations or diagrams showing the fenders and mooring lines interaction at all states of the tide in loaded and ballasted condition.
- Installation, maintenance, and removal manuals.
- Internal and/or third-party records of coatings DFT for all painted elements or galvanisation certificates.
- Any other relevant information agreed upon between the purchaser and supplier.

All input should be provided and reviewed prior to manufacturing and both design and manufacturing must be carefully planned to avoid delay caused by factors such as additional testing or manufacturing acceptance.

13.2.5 Records

Project record requirements should be clearly given in the specification, which in most cases means a full record of manufacturing and installation to be kept and provided to the user upon request.

Fender rubber units should be permanently marked so that they can be individually identified. The marking should include the following:

- i. Rubber unit name and size.
- ii. Manufacturing date (month and year).

- iii. Serial number (unique to the individual rubber unit).
- iv. Rubber grade.
- v. Name of fender manufacturer (and if different, the supplier). Showing two identification marks on opposite sides of the rubber unit is recommended.

13.2.6 Warranty, Product Liability and Compliance

Specification should clearly state the required warranty period and minimum product liability insurance for the project. Appropriate warranty type and period may vary depending on the vessel type, number of berthing operations per year and the design vessel characteristics (i.e. with belting or not), port conditions and operations.

It is recommended that provisions are included regarding countermeasures for the product and compensation for damage incurred in case testing or quality fraud is discovered.

13.3 Vessel, Berthing and Quay Structure Considerations

Pertinent information about the vessels, berthing operations and terminal structures should be included in the specification and responsible for berthing energy calculation should be clearly stated.

Many fender types change performance under certain loading conditions such as fatigue loading, partially compressed fenders, etc. so loading conditions need to be set out very clearly.

Pertinent information to be included in the specification may include:

- Operational requirements and design life limitations for operation of the berth, both during berthing and mooring conditions.
- Assessment of acceptable reliability levels for the fendering system.
- Design vessel specifications and hull shape characteristics, together with dimensions to be selected from the shipping table (refer to Chapter 3).
- Special features such as bow flare angles, beltings, other protrusions or special shapes, allowable mean hull pressures or belting loads, etc. (refer to Chapter 6).
- Met-ocean conditions. Exposure to wind, waves, currents, tides, etc. Likely effects of ice flows or tropical climates, passing vessel influence, etc. (refer to Chapter 7).
- Tide levels, surge, sea level rise, location of fixings, etc.
- Range of ambient temperature.
- Navigation conditions (refer to Section 5.1.4).
- Berthing angles and berthing velocity (refer to Sections 5.5 and 5.4).
- Frequency of use of the berth by different classes of vessel, draught, and freeboard for largest and smallest vessel.
- Whether the berth is for loading or discharge or both.
- Tug assisted or not.
- The maximum and minimum allowable fender projection considering; the reach of crane, loading arm, collision caused by flare angles etc.
- Mooring lines interference, i.e. size of fender panel, fender elevation, etc. (refer to Chapter 8).
- Type of quay structure, i.e. open, closed, dolphin, etc. (refer to Section 6.4.8).
- Type of support structure(s) the fender system would be fitted to, i.e. new or existing structure, steel or concrete, including check space requirement for installation (chains, connections).
- Seabed level near the berth, berth cope level.

- The maximum allowable (design) reaction force on existing quay structure, if any.

The designer should give special consideration to any permanently moored vessels. In those circumstances, issues with the fender system fatigue, rubber longevity, inspections and maintenance need to be studied and allowed for in the design.

13.4 Manufacturing, Testing and Quality Requirements for Fender Units

The purchaser should carefully consider including the following requirements in the specification, as appropriate:

- Design life should be considered prior to committing and environmental standards capturing a whole life cycle of the fender system should be followed as given in Chapter 12.
- Materials and workmanship should conform to current standards and good practice and should be deemed suitable for use in aggressive marine/ tidal environments.
- Testing reports with fender performance curves, physical properties of rubber certificates, mill certificates for steel panels, low friction facing, chains and bolts should be supplied for each different fender type/ size at the time of delivery (refer to Chapter 10).
- Fender system including accessories should be fabricated, assembled, installed and tested in accordance with Manufacturer's instructions and PIANC recommendations (refer to Chapter 11).
- The production facility of materials should not be changed from the first technical submission or during a project, unless the Manufacturer has demonstrated that the materials from the new source can at least equally meet the requirements.
- Manufacturers should provide comprehensive documentation on the fender products where, as a minimum, the base performance properties are clearly specified (refer to Chapter 10).
- Purchasers should be able to witness (or measure independently) quality control, testing of both the fender and physical property/ TGA tests for the materials either themselves or using independent, experienced, third parties using their measuring equipment (refer to Chapter 10).
- Purchasers that hire a third party to witness and verify fender testing and production are recommended to select a reputable and qualified company.
- Calibration of the testing equipment.
- Manufacturers should provide information on the tolerances of their product for both energy absorption and reaction force and also indicate how the fender unit performs under other conditions such as angular compression, under water and overload (maximum deflection limits) (refer to Chapter 10).

If rubber fender performance needs to be verified by the third party selected by the purchaser, the supplier must allow the third party an access to the factory and fully cooperate with the testing. It is recommended that following information from the third party is specified by the purchaser:

- Proven independence.
- Knowledge and experience with the testing set-up.
- ISO 17025 certified with the corresponding credentials.
- Equipment used and methodology.
- Quality and accuracy.
- Third party scope – fundamental, durability, type approval, verification testing.
- Verification scope

- Verify that fenders meet the project specifications, i.e. material and performance testing, dimensions, visual check, etc.
- Ensure third party equipment is working properly and identify the standard deviation between third party and manufacturers equipment.
- A clear and agreed test set-up is crucial to obtain comparable values.
- A clear and unambiguous acceptance procedure should be agreed in terms of test results between third party and manufacturer. Results between test facilities can differ for both performance testing and material testing. Acceptance criteria should be based on the manufacturer's catalogue values plus tolerance values. Tolerance values should be agreed upon prior to the verification testing.

13.5 Delivery, Installation and Storage

Supplier documentation should detail the requirements regarding delivery, installation and storage. The following requirements should be considered:

- Handling, storing and installation should be in accordance with the manufacturer's instructions and recommendations in order to prevent them from being distorted, overstressed or damaged in any way.
- All slings, ropes or chains for handling fenders should be rubber or nylon sheathed.
- Great care should be taken to prevent cutting or tearing of the rubber, particularly in the area of embedded plates, around bolt holes and washer recesses in the fender base, flanges or fins.

Fender systems should normally be stored in the transport packings (refer to Section 9.1.4 and 11.3).

As part of the tender submission, the fender manufacturer should provide instructions on what maintenance and inspections are required for the fender system.

13.6 Sustainability

At the end of the service life of the fender system or components that need to be changed, should be easily replaced, and as much as practical, all components of the fender system should be recycled. Further guidance is available in Chapter 12.

14 REFERENCES

- ACS. (2022). ACS Green Chemistry Institute. Retrieved from <https://www.acs.org/greenchemistry.html>.
- Akbas, A., & Yuhana, N. (2021). Recycling of Rubber Wasters as Fuel and Its Additives. *Recycling*, Issue 6, 78.
- Akiyama, H. (2018). Study on aging of circular hollow rubber fender in long term usage. Doctoral Thesis (in Japanese). Osaka, Japan: Osaka University.
- Akiyama, H. S. (2017, Vol. 5). Investigation on service years of large size rubber marine fenders. *Journal of JSCE*, pp. 392-401.
- Akiyama, H., Shiomi, T., Omura, A., Yamamoto, S., & Ueda, S. (2017). Method to Estimate the Aging of Large Rubber Marine Fender. *International Offshore and Polar Engineering Conference* (pp. 1459-1466). San Francisco, California, USA: ISOPE.
- Americanchemistry. (2022). Retrieved from <https://www.americanchemistry.com/industry-groups/center-for-the-polyurethanes-industry-cpi>.
- ASCE. (2015). *Waterfront Facilities Inspection and Assessment*. ISBN (print): 9780784413579. Reston, Virginia,: American Society of Civil Engineers.
- ASCE 7. (2022). *ASCE 7. Minimum design loads for buildings and other structures*. American Society of Civil Engineers. Reston, USA. ISBN: 978-0-7844-1085-1.
- Berendsen, E. A. (2022). Impact resistance of ship hull to berthing loads: quantifying critical fender impact, Delft University of Technology, November 1, 2022.
- Berendsen, E. A., Roubos, A. A., Williams, R., Walters, C. L., & Broos, E. J. (2024). Structural capacities of ships parallel hull subject to fender-induced berthing impact loads. *PIANC World Congress*. Capetown: PIANC.
- Brolsma, J. U., Hirs, J. A., & Langeveld, J. M. (1977). On Fender Design and Berthing Velocities. *24th International Navigation Congress* (pp. Sect II, Subject 4, pp.87-100). Leningrad: PIANC.
- Broos, E. J., Rhijnsburger, M. P., Vredeveltdt, A. W., & Hoebee, W. (2018). The safe use of cylindrical fenders on LNG, Oil and Container Terminals. *PIANC-World Congress Panama City*. Panama: PIANC.
- Broos, E. J., van Schaik, C., & Huitema, S. (2013). Innovative Rigid Concrete Fender System Reduces Life Cycle Costs., *ASCE COPRI Ports '13 conference* Seattle.
- BS6349. (2014). *BS 6349-4:2014, Maritime works – Part 4: Code of practice for design of fendering and mooring systems*. London: The British Standards Institution .
- Carbontrust. (2009). https://web.archive.org/web/20090511102744/http://www.carbontrust.co.uk/solutions/CarbonFootprinting/what_is_a_carbon_footprint.htm.
- CDIT; Design. (2019). *Guidelines for Design and Testing of Rubber Fender Systems*, No.51E. Coastal Development Institute of Technology (CDIT), Japan. Tokyo: Coastal Development Institute of Technology (CDIT).

- CDIT; maintenance. (2019). Guidelines for the Maintenance of Rubber Fender Systems (2nd Edition). Japan Tokyo: Coastal Development Institute of Technology (CDIT).
- Chaikoon, T., & Faungnawakij, S. (2016). Carbon footprint of natural rubber production in Thailand. *Journal of Cleaner Production*, Volume 135, pages 1306-1314.
- Childs, K. M. (2001). ASCE manual of practice no. 101, Underwater Investigations: Standard Practice Manual.
- DNV. (1992). Structural reliability analysis of marine structures. Classification notes NO. 30.6. Hovik, Norway: Det Norske Veritas.
- EAU. (2020). Empfehlungen des Arbeitsausschusses "Ufereinfassungen" Häfen und Wasserstraßen EAU 2020,. Ernst & Sohn ISBN: 978-3-433-03316-6.
- EN 1990. (2011). NEN-EN 1990. Eurocode – Basis of structural design. European Committee for standardization. Brussels, Belgium.
- GPSNR. (2022). Global Platform for Sustainable Natural Rubber (GPSNR). Retrieved from <https://sustainablenaturalrubber.org/>.
- Greenmax. (2022). Retrieved from <https://www.greenmax-machine.com/what-kind-of-polyethylene-material-can-be-recycled.html>.
- Han, W., Han, D., & Chen, H. (2023). Pyrolysis of Waste Tires: A Review. *Polymers*, Issue 15, 1604.
- Heemskerk, I. (2020). Determining the mechanisms causing the hydraulic damping during ship berthing, Delft University of Technology, May 28, 2020.
- IMO. (2011). IMO Manual on Oil Pollution, Section I, Prevention. London: IMO.
- ISO14001. (2015). Environmental management systems. Technical Committee ISO/TC 207, Environmental management, Subcommittee SC 1, Environmental management systems.
- ISO2394. (2015). General principles on reliability for structures. International Organization for Standardization. Geneva, Switzerland.
- JCSS. (2001). Probabilistic model code. Part 1. www.jcss.byg.dtu.dk: Joint Committee on Structural Safety.
- Ligtoet, A., & Lei van der, T. (2012). Analyse bedrijfswaarden HbR. Delft, the Netherlands: Technische Universiteit Delft.
- Lloyd's. (1989). Strength of side shell structure to resist quayside fender loads, Class Computational Services Group, Report no. CSD 89/24, SEPT 1989 . London: Lloyd's Register of Shipping, CONFIDENTIAL.
- Lloyd's. (1993). A Container Ship's Ability To Resist Quayside Fender Loads, Report No. CSD 92/38. London: Lloyd's Register Of Shipping.
- Nakamura, T., Nakamura, T., Miyata, M., Yoneyama, H., & Kashima, H. (2020, March). Study for revision of a design method that considers the usage environment of fenders. Technical Note of National Institute for Land and Infrastructure Management, No.1101 (in Japanese).
- OCIDI. (2009). Technical Standards and Commentaries for Port and Harbour Facilities in Japan. Tokyo, Japan. The Overseas Coastal Area Development Institute. Tokyo, Japan: OCIDI.

- OCIMF. (2013). Ship to Ship Transfer Guide for Petroleum, Chemicals and Liquefied Gases, Section 9.1.2 and Table 9.1. London: The Oil Companies International Marine Forum (OCIMF).
- OCIMF. (2018). Mooring Equipment Guidelines 4th edition (MEG4). London: The Oil Companies International Marine Forum (OCIMF).
- Orlin, F. (2020). Reliability-based Assessment for Fender Systems. Delft: Delft University of Technology.
- PIANC. (2002). Guidelines for the Design of Fenders Systems, report of PIANC MarCom WG33. Brussels: PIANC.
- PIANC. (2008). Life Cycle Management of Port Structures, Recommended Practice for Implementation, report of PIANC MarCom WG103 . Brussels: PIANC.
- PIANC. (2014). Sustainable Ports - A Guide for Port Authorities (2014), report from PIANC EnviCom WG150. Brussels: PIANC.
- PIANC. (2016). Guidelines for Cruise Terminals, report of PIANC MarCom WG 152:.
- PIANC. (2019). Design Principles of Dry Bulk Marine Terminals, report of PIANC MarCom WG184. Brussels: PIANC.
- PIANC. (2024). Mooring of Large Ships at Quays (2024), report of PIANC MarCom WG186. Brussels: PIANC.
- PIANC WG145. (2022). Berthing velocity analysis of seagoing vessels over 30,000 dwt (2022), report of PIANC MarCom WG 145.
- PIANC WG235. (2022). Ship Dimensions and Data for Design of Marine Infrastructure (2022), report of PIANC MarCom WG235. Brussels: PIANC.
- Rackwitz, R. (2000). Optimization – the basis of code making and reliability verification. Structural Safety. 22. pp. 27–60.
- Recycle. (2022). Recycle-more.co.uk . Retrieved from Recycle-more.co.uk .
- ROM. (2002). ROM 0.0 2002 General procedure and requirements in the design of harbor and maritime structures. Part I. Spain: Puertos del Estado, ISBN: 84-88975-30-9.
- ROM. (2008). ROM 0.5-05, (2008), Geotechnical Recommendations for the Design of Maritime and Harbour Works. Madrid, Spain: ROM. ISBN 978-8488975622. Madrid.
- ROM. (2012). ROM 2.0-11; Recomendaciones para el proyecto y ejecucion en Obras de Atraque y Amarre. Puertos del Estado (2012): . Puertos del Estado.
- Roubos, A. A., Gaal, M., Hein, C., Iversen, R., & Williams, R. (2022). Recommendations for berthing velocity in PIANC WG 211. ASCE COPRI Ports 2022. Honolulu: ASCE .
- Roubos, A. A., Groenewegen, L., Ollero, J., Hein, C., & Wal, E. v. (2018). Design values for berthing velocity of large Seagoing vessels, PIANC World Congress Panama 2018.
- Roubos, A. A., Mirihagalla, P., Gaal, M., Blankers, G., Groenewegen, P., Eldridge, E., & Roussel, L. (2023). Effect of PIANC WG211 design method on fender dimensions. PIANC America 2023 Conference. Fort Lauderdale: PIANC USA.
- Roubos, A., Iversen, R., & Oskamp, J. (2024). Backgrounds Partial Energy Factor For Fender Desing Proposed by PIANC WG211. PIANC World Conference South Africa 2024. Cape Town: PIANC.

- Sarkar, P., & Bhowmick, A. K. (2018). Sustainable rubber and rubber additives. *Journal of applied polymer science*, 135(24), 45701. Wiley Online Library.
- Shimizu, K., Yasui, M., & Yamase, S. (2015). A study of durability for marine fenders with reclaimed rubber and calcium carbonate,. *Proceeding of 25th ISOPE*, (pp. 1506-1512).
- Sijberden, H. L., Smedes, R. H., & de Gijt , J. G. (1996). Fender design a new concept. *Harbour congres*. Antwerpen.
- Steel. (2022). Steel Recycling Institute. Retrieved from <https://www.steel.org/>.
- TNO. (2019). Ship side - cylindrical fender contact characteristics, a systematic investigation, TNO 2018 R10427 | Final report. Delft: TNO.
- Ueda, S. Y. (2010). Reliability Design of Fender System for Berthing ships. *PIANC MMX Congress Liverpool UK*.
- UN. (2022). <https://www.un.org/sustainabledevelopment/sustainable-development-goals/>.

APPENDIX A: RELIABILITY REQUIREMENTS FOR FENDERS

Load and Resistance Factor Design Approach

While the load and resistance factor design approach is widely adopted in the design of marine structures, the work of PIANC WG 33 (2002) incorporated safety using a global safety factor. It was unclear, however, whether this design approach results in an appropriate reliability level. Several aspects that significantly influence the reliability level, such as berthing frequency, correlations between design variables and multiple fender contact, were not embedded in the design approach of PIANC WG 33.

In accordance with other design codes and standards, PIANC WG 211 considers a fender system to be reliable when the design value of the energy absorption capacity ($E_{f,d}$) is greater than the design value for the kinetic energy ($E_{k,d}$).

$$E_{f,d} \geq E_{k,d} \quad 1.$$

Where:

$E_{f,d}$ = Design value for the capacity of the fender system [kNm].

$E_{k,d}$ = Design value for kinetic energy exerted by the berthing vessel [kNm].

Since uncertainties are present in the energy absorption capacity of a fender system as well as in the kinetic energy exerted by the approaching vessel acting on the fender system, PIANC WG 211 introduced the partial material factors and a partial energy factors. These partial factors can be applied to the characteristic values of fender capacity $E_{f,c}$ (Chapter 6) and kinetic energy $E_{k,c}$ (Chapter 5) of the approaching vessel in order to determine the associated design values $E_{f,d}$ and $E_{k,d}$.

$$E_{k,d} = \gamma_E E_{k,c} \quad 2.$$

$$E_{f,d} = \frac{E_{f,c}}{\gamma_m} \quad 3.$$

Where:

$E_{k,c}$ = Characteristic value of kinetic energy of the approaching vessel [kNm]

$E_{f,c}$ = Characteristic energy absorption capacity of fender system [kNm]

γ_E = Partial energy factor [-]

γ_m = Partial material factor related to the resistance of the fender system [-]

Reliability and Failure Consequences

The allocation of a reliability target is crucial when designing or assessing a fender. When the failure consequences of a fender are high, higher factors of safety representing a higher reliability level need to be taken into consideration. In general, the acceptable probability of failure is expressed by a specific target reliability level. Deriving a project specific reliability target is fairly complex, since multiple aspects need to be taken into account. Consequently, design codes and standards incorporated reliability and/or consequences classes. This principle is also considered here, since the target reliability of a fender system is largely influenced by the consequences of fender (system) failure, which can significantly differ per berth or type of support structure.

Table A-1 presents target reliability indices for different consequences classes, e.g. class A, B, C, D and E. In this table, the reliability level of a fender is related to the probability of failure during a certain reference period, for instance the design life of 20 years. Since uncertainty in berthing velocity generally dominates the uncertainty in fender reliability individual failure events are largely independent (Ueda, 2010). Hence, the probably of failure and the associated reliability target can be determined using the following relations.

$$P_{f;t_{ref}} = 1 - (1 - P_{f;t_1})^{t_{ref}} = 1 - (1 - P_{f;i})^{n_{ref}} \quad 4.$$

$$P_{f;t_{ref}} = \Phi(-\beta_{t;t_{ref}}) \quad 5.$$

Where:

$P_{f;t_{ref}}$ = Probability of failure during a certain refence period [-]

$P_{f;t_1}$ = Annual probability of failure [-]

$P_{f;i}$ = Probability of failure single berthing event i [-]

t_{ref} = Reference period [years]

n_{ref} = Number of berthings during a certain reference period [years]

$\beta_{t;t_{ref}}$ = Reliability target related to a certain reference period [-]

$\Phi(..)$ = Standard normal cumulative distribution function [-]

An important step in the design basis for a fender system is deciding on the appropriate reliability level for the design. Many factors can influence this reliability level, however the most important one is the consequence class for the fender system. Table A-1 presents an overview of the reliability targets for different consequence classes and reference periods and can be used in combination with national codes and standards, such as the ASCE, BS, EAU, Eurocode, ROM or OCDI, which often provide recommendations for the reliability targets or acceptable probability of failure.

Name	t_{ref}	Class A	Class B	Class C	Class D	Class E
		Negligible/ low	Some	Conside- rable	High	Very high
Probability of failure P_f						
$t_{ref} = 1$ year	1	1.08E-04	1.33E-05	5.41E-06	1.30E-06	9.96E-08
$t_{ref} = 25$ years	25	2.69E-03	3.34E-04	1.35E-04	3.25E-05	2.49E-06
$t_{ref} = 50$ years	50	5.38E-03	6.67E-04	2.71E-04	6.50E-05	4.98E-06
Target reliability index β_t						
$t_{ref} = 1$ year	1	3.70 ²	4.20 ¹	4.40 ¹	4.70 ¹	5.20
$t_{ref} = 25$ years	25	2.78	3.40	3.64	3.99	4.57
$t_{ref} = 50$ years	50	2.55 ⁴	3.21 ⁴	3.46 ⁴	3.83 ⁴	4.42 ⁴
¹⁾ Reliability indices are based on (ISO2394, 2015). ²⁾ Reliability index is based on (Rackwitz, 2000). ³⁾ Reliability indices is based on (EN 1990, 2011) ⁴⁾ Values are in the range of the values suggested in several national and international codes and standards all over the world, such as (ASCE 7, 2022), (OCDI, 2009), (EN 1990, 2011) and (ROM, 2008). ⁴⁾ Based on a berthing frequency of 100 berthings per year.						

Table A-0-1: Probability of failure and associated reliability target for different consequence classes

Understanding the position and failure consequences of a fender system is of great importance. In general, when failure consequences are high the required reliability level increases. In some circumstances, failure of a single fender will not result in economic repercussions, whereas in other situations major accidents may occur. When national recommendations regarding reliability targets are lacking, Chapter 4 presents typical examples of fender systems for different consequence classes. Most of the fender systems installed on marine structures correspond to class A or class B. Table A-2 lists typical descriptions of failure consequences in literature, which can serve as background information when selecting a consequence class.

Description	Consequence class				
	A	B	C	D	E
Qualitative	Negligible/low	Some	Considerable	High	Very high
Human safety					
- Number of fatalities (ISO2394, 2015)	$N \leq 1$	$N \leq 5$	$N \leq 50$	$N \leq 500$	$N > 500$
- Number of people at risk (ASCE 7, 2022)	$N_{PAR} < 5$	$N_{PAR} < 50$	$N_{PAR} < 500$	$N_{PAR} < 1500$	$N_{PAR} > 1500$
- Degree of warning (ASCE 7, 2022) (DNV, 1992)	Progression of failure is not possible and people at risk are able to escape in time.	Redundant structural response and progression of failure is mitigated and failure is not sudden providing adequate warning signals.	Progression of failure is mitigated, but failure is sudden without providing warning signals.	Widespread progression of damage is likely to occur and failure is sudden without providing warning signals.	Widespread progression, induced by unexpected and sudden environmental disasters, is possible.
- Social and environmental repercussion index (ROM, 2002)	$SERI \leq 5$	$SERI \leq 15$	$SERI \leq 25$	$SERI \leq 30$	$SERI > 30$
Economic					
- Description (ISO2394, 2015)	Predominantly insignificant material damage.	Material damage <u>and</u> functionality losses of significance for owners and operators <u>and</u> low or no social impact.	Material losses <u>and</u> functionality losses of societal significance, causing regional disruptions <u>and</u> delays in important societal services over several weeks.	Disastrous events causing severe losses of societal services <u>and</u> disruptions <u>and</u> delays at national scale over periods in the order of months.	Catastrophic events causing losses of societal services <u>and</u> disruptions <u>and</u> delays beyond national scale over periods in the order of years.
- Accessibility (Ligtvoet & Lei van der, 2012)	Very little hindrance to shipping, railway transport, pipeline systems (Very	Small consequences for availability of navigation channels, railways, roads or pipeline corridors.	Short period of barricade with regard to navigation channels, railways, roads or pipeline corridors. (The	Damage to navigation channels, railways, roads or pipeline corridors. (The	Loss of main navigation channels, railways, roads or pipeline corridors. (Main transport routes are

Description	Consequence class				
	A	B	C	D	E
	short period, less than one day).	(Barricade measures for a period of one day).	availability is lower for a period of one week)	availability is lower for a period of weeks)	unavailable for a period of months)
-Ratio between direct failure costs and costs of safety investments $\rho = C_{f,direct}/C_{Investments}$ (JCSS, 2001)	$\rho \leq 1$	$\rho \leq 2$	$\rho \leq 5$	$\rho \leq 10$	$\rho > 10$
Environmental (ISO2394, 2015)	Damage to the qualities of the environment of an order that can be restored completely in a matter of days.	Damage to the qualities of the environment of an order that can be restored completely in a matter of weeks.	Damage to the qualities of the environment limited to the surroundings of the failure event and that can be restored in a matter of weeks.	Significant damage to the qualities of the environment contained at national scale but spreading significantly beyond the surroundings of the failure event and that can only be partly restored in a matter of months.	Significant damage to the qualities of the environment spreading significantly beyond the national scale and that can only be partly restored in a matter of years to decades.
Reputation (Ligtvoet & Lei van der, 2012)	No negative attention in media and no damage to the image of the port.	Very short period of negative attention in local, regional and national media (>1 day). Serious concerns among people living in the vicinity, local government, national government or external clients. Damage to image of a few stakeholders.	Short and limited period of negative attention in local, regional and national media (> 2 days). Serious concerns among people living in the vicinity, local government, national government or external clients. Damage to image of the port for some time.	Period of negative attention in local, regional and national media (>week). Serious concerns among people living in the vicinity, local government, national government or external clients. Damage to image of the port for some time.	Long period of negative attention in local, regional and national media (> month). Very serious concerns among people living in the vicinity, local government, national government or external clients. Permanent damage to image of the port.

Table A-0-2: Description of failure consequences in literature

APPENDIX B: THERMAL CONDUCTION OF RUBBER FENDERS

The symbols in this appendix deviate from the main report as these come from dedicated reference (Nakamura, Nakamura, Miyata, Yoneyama, & Kashima, 2020).

Rubber is a material with low thermal conductivity. It takes a long time for heat to penetrate inside fender body. During compression tests, measuring surface temperature is not enough and the thermal stabilisation time is important to estimate. The following theory is used to estimate time for thermal stabilisation.

1 . Theory of Heat Conduction:

One dimensional heat conduction theory is used to predict heat transfer inside a rubber body. Fig. B.1 shows the calculation model of fender which is infinite rubber plate of thickness: $2L$.

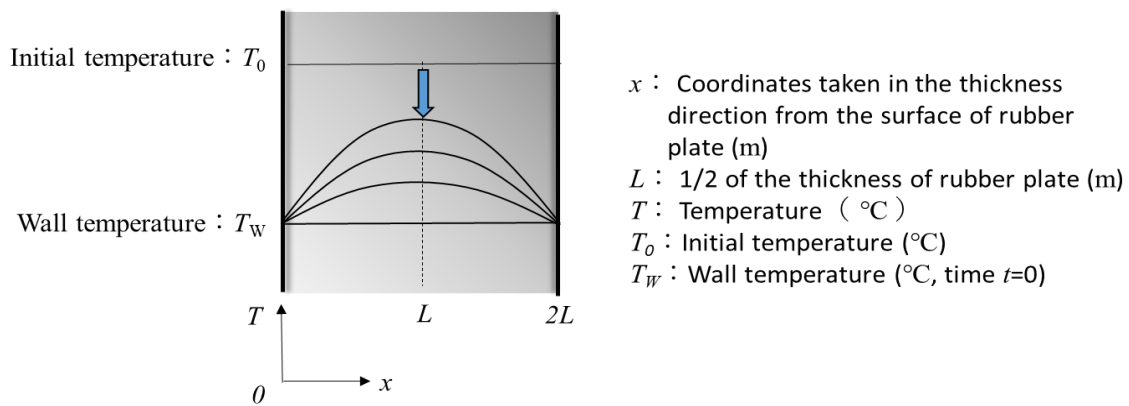


Figure B-1: heat conduction (Nakamura et al., 2020)

The equation of heat conduction will be as follows:

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \quad \alpha = \frac{k}{\rho C_p}$$

α : Thermal diffusivity (m^2/s)
 k : Thermal conductivity ($=0.15 \text{ W/m}\cdot\text{K}$)
 ρ : Density ($=1200 \text{ kg/m}^3$)
 C_p : Constant pressure specific heat ($=1900 \text{ J/kg}\cdot\text{K}$)

Assuming that the temperature difference is θ ($\theta = T_0 - T_W$), the equation will be:

$$\frac{\partial^2 \theta}{\partial x^2} = \frac{1}{\alpha} \frac{\partial \theta}{\partial t}$$

Initial conditions and boundary conditions are:

- Initial condition at $t=0$ and $0 \leq x < 2L$, $\theta = \theta_0 = T_0 - T_W$
- Boundary condition at $x=0$ and $t > 0$, $\theta = 0$
- Boundary condition at $x=2L$ and $t > 0$, $\theta = 0$

The temperature difference at the centre of rubber body at $t=t$ will be:

$$\frac{\theta}{\theta_i} = \frac{T-T_i}{T_i-T_1} = \frac{4}{\pi} \sum_{n=1}^{\infty} \frac{1}{n} e^{-(\frac{n\pi}{2L})^2 \alpha t} \sin \frac{n\pi x}{2L} \quad (n=1, 3, 5 \dots)$$

(B.1)

2 . Calculation of Thermal Stabilisation Time, Background of Equation (B.1)

The time until the temperature at the centre of rubber plate approaches 23°C when the specimen is moved from the initial temperature (ambient temperature, vulcanisation temperature, etc.) to the standard temperature (23°C), can be estimated by using equation B.1 above. The calculation conditions of rubber material were set as follows. These values may be changed due to the actual materials.

- Thermal conductivity : $k = 0.15 \text{ W/m} \cdot \text{K}$
- Density : $\rho = 1200 \text{ kg/m}^3$ (see table 10-1)
- Constant pressure specific heat : $C_p = 1900 \text{ J/kg} \cdot \text{K}$

The result is shown in Fig. B.2. The number of required days is close or shorter when the initial temperature is daily atmosphere (-10 to 40°C), however longer when the initial temperature is as high as vulcanisation temperature especially for large size fenders.

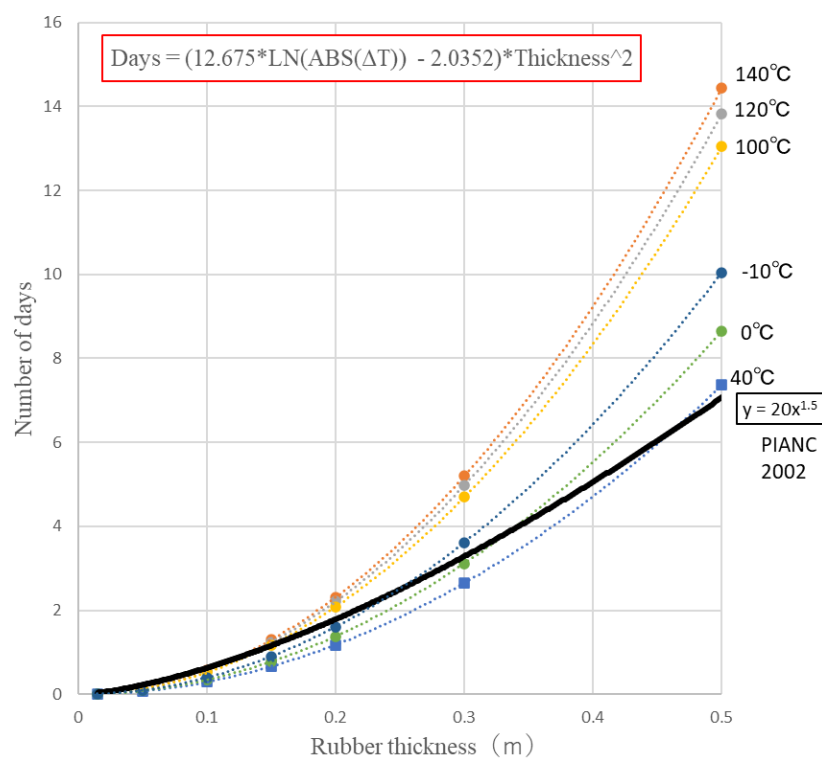


Figure B-2: relation rubber thickness and temperature adaptation of fender (Nakamura et al., 2020)

3 . Determining Design Temperature for Rubber Fenders

The performance of rubber fenders is affected by ambient temperature. So, the design temperatures have a large influence on the choice of fender system. The ideal information should be the temperature data recorded at the site where the fender is located. In practice, the meteorological records in websites are very useful, however the designer should be careful on the selection of appropriate temperature data for the fender design.

The above heat conduction theory is used to determine the design temperature from statistical weather records. The record of air temperature in February 2018 at the Kushiro port in Northern Japan is used as an example.

The lowest was -16.5°C and the highest is 4.1°C . The temperature was recorded with one-hour increments and displayed in Fig. B.3 with orange dots and line. The red curve is the time history of rubber temperature at the depth of 10 cm and the yellow curve is at 20 cm deep. The black line is the mean daily temperature.

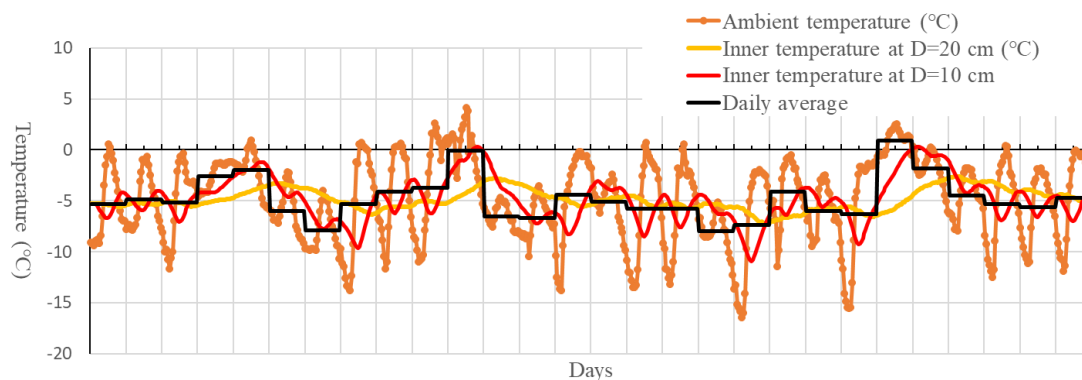


Figure B-3: Time history of inner temperature and ambient temperature (Nakamura, Nakamura, Miyata, Yoneyama, & Kashima, 2020)

The response of temperature depends on the thickness of rubber however according to Fig. A.3, the highest and lowest daily average look appropriate for the highest and lowest design temperature for rubber fender. However some extreme low temperatures are lower than the lowest daily average. So, the fenders for the very cold weather, e.g. below -10°C and the port is in operation, actual measurement at site is recommended.

APPENDIX C MULTI-FENDER CONTACT, GEOMETRY FOR FENDER COMPRESSION CALCULATIONS

Where multi-fender contact occurs the berthing energy is distributed between the contacted fenders. This is described in Section 6.4.5. To calculate the contribution of multiple fenders acting during berthing, the vessel and fender geometry are required. Figures C.1 and C.2 demonstrate the geometry of multi-fender contact and the impact of fender geometry.

Because the location of the tangent point on the hull varies with the elevation, designers should consider the effects of the vessel draught, tide and elevation of the fenders. At a lower elevation the fender contact on the hull will be further away from the bow and thus will result in a higher eccentricity factor (C_e) in the berthing energy calculation (refer to Chapter 5).

Case 1: Multi Fender Contact at Low Approach Angle α – Contact at Hull Tangent Point

$$\Delta_{H2} = \Delta_{H1} - S \tan \alpha$$

$$\Delta_{H3} = \Delta_{H1} - 2S \tan \alpha$$

$$\Delta_{H4} = \Delta_{H1} - 3S \tan \alpha$$

Etc

$$\Delta_{B1} = \Delta_{H1} + S \tan \alpha - F / \cos \alpha$$

Δ_{B2} no contact assumed ≥ 0

Multi-Fender Contact

$$\sin \theta = (S / \cos \alpha) / R_B \approx S / R_B \text{ for small } \alpha$$

$$F = \text{Bow hull offset at B1 from line of flat hull}$$

$$F = 2R_B \sin(\frac{1}{2}\theta) \cos(90 - \frac{1}{2}\theta) = 2R_B \sin^2(\frac{1}{2}\theta)$$

$$\text{Alternatively, } F = R_B(1 - \cos \theta)$$

α = Approach angle and H = Fender height

R_B = Bow Radius at the level of the fender

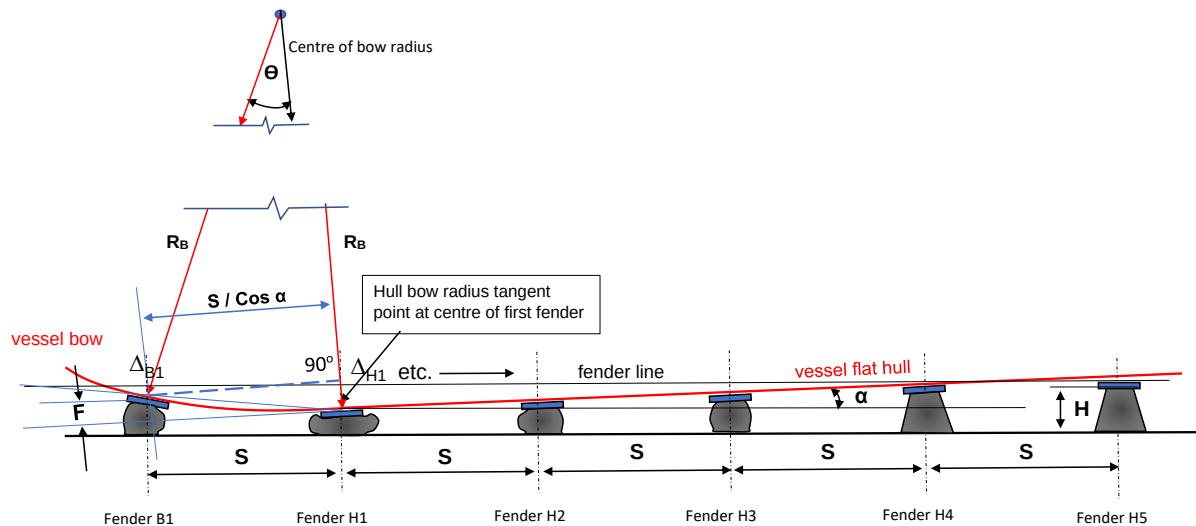


Figure C-1: Fender contact at hull tangent line impacts at fender centreline

Case 2: Multi Fender Contact at Low Approach Angle α – Hull Tangent Point between fenders

$$\Delta_{H2} = \Delta_{H1} - S \tan \alpha$$

$$\Delta_{H3} = \Delta_{H1} - 2S \tan \alpha$$

$$\Delta_{H4} = \Delta_{H1} - 3S \tan \alpha$$

Etc.

$$\Delta_{B1} = \Delta_{H1} + S \tan \alpha - F / \cos \alpha$$

$$\Delta_{B2} \text{ no contact assumed } \geq 0$$

Multi-Fender Contact

$$\sin \theta = (\frac{1}{2}S / \cos \alpha) / R_B \approx S / 2R_B \text{ for small } \alpha$$

F = Bow hull offset at B1 from line of flat hull

$$F = 2R_B \sin(\frac{1}{2}\theta) \cos(90 - \frac{1}{2}\theta) = 2R_B \sin^2(\frac{1}{2}\theta)$$

α = Approach angle, H = Fender height

$$\text{Alternatively, } F = R_B(1 - \cos \theta)$$

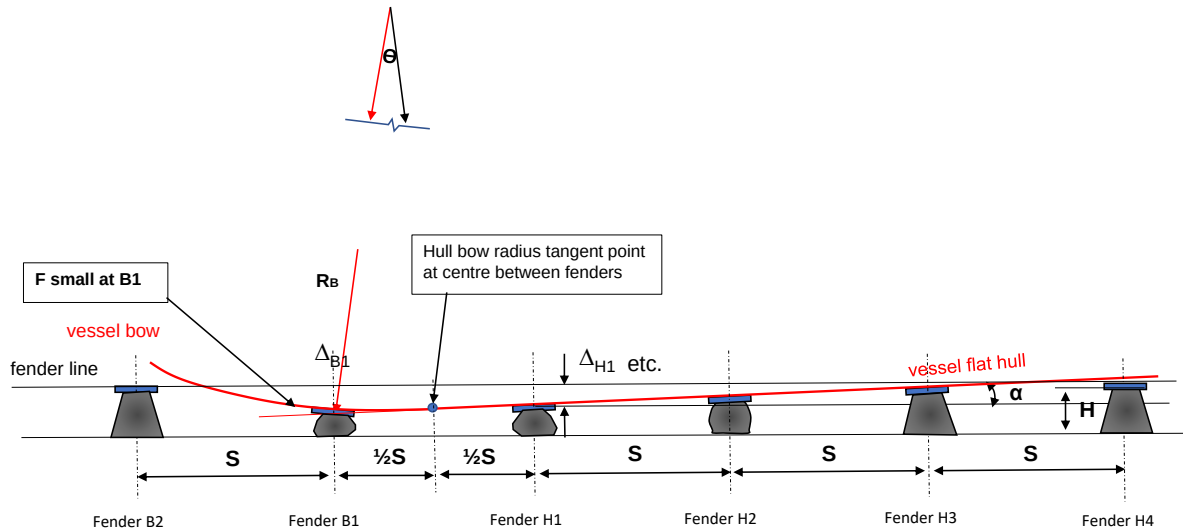


Figure C-2: Fender contact at hull tangent line centrally between fenders

It should be noted that the eccentricity factor (C_e) will change as more fenders are contacted as the distance from the vessel centre of mass to the centroid of the fender reactions will change. Refer to Section 5.6 for further explanation.

APPENDIX D: ABBREVIATIONS SYMBOLS AND UNITS

14.1 D.1 ABBREVIATIONS

BS	British Standard
DWT	Deadweight Tonnage (metric tonnes)
EAU	Empfehlungen des Arbeitsausschusses "Ufereinfassungen", German standards
GT	Gross Tonnage
IMO	International Maritime Organisation
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
MarCom	Maritime Navigation Commission (PIANC)
MBL	Minimum Breaking Load (OCIMF, 2018)
MBM	Multi Buoy Mooring (also known as Conventional Buoy Mooring)
OGV	Ocean Going Vessel
PIANC	World Association for Waterborne Transport Infrastructure (previously Permanent International Association of Navigational Congresses)
PPU	Portable Pilot Unit
ROM	Recomendaciones de Obras Marítimas, Spanish Standards
RoRo	Roll-on/roll-off
SPM	Single Point Mooring
STS	Ship-to-Ship
TEU	Twenty foot equivalent Unit (container unit)
ULCC	Ultra Large Crude Carrier
ULCV	Ultra Large Container Vessel
UV	Ultraviolet
VLBC	Very Large Bulk Carrier
VLCC	Very Large Crude Carrier
WG	Working Group (PIANC)
WLL	Working Load Limit (OCIMF, 2018)

14.2 D.2 SYMBOLS

Symbols cover all Chapters and Appendices except for Appendix B and C.

Symbol	Unit	Definition
A_c	(m ²)	Contact area of fender panel with vessel hull, excluding chamfers
a	(-)	Logarithmic regression coefficient correction factor for alternative annual berthing frequencies
a_w	(m)	Wave amplitude
B	(m)	Beam (breadth moulded) of vessel
b	(-)	Constant correction factor for alternative annual berthing frequencies
C_{ang}	(-)	Angular factor
$C_{ang,c}$	(-)	Characteristic angular factor
$C_{ang,E}$		Angular factor for energy absorption
$C_{ang,R}$		Angular factor for reaction force
C_b	(-)	Block coefficient of vessel
C_c	(-)	Berth configuration factor
C_e	(-)	Eccentricity factor
$C_{e,c}$	(-)	Characteristic eccentricity factor
c_{eff}	(m)	Effective clearance between supporting structure and vessel hull due to bow flare.
C_m	(-)	Added mass factor
$C_{m,c}$		Characteristic added mass factor
C_{m1}	(-)	Added mass factor of the Vessel 1
C_{m2}	(-)	Added mass factor of the Vessel 2
c_{min}	(m)	Minimum clearance to support structure
C_{mult}	(-)	Multiple fender contact factor
$C_{mult,c}$	(-)	Characteristic multiple fender contact factor
$C_{sts,e}$	(-)	Eccentricity factor (ship-to-ship berthing)
C_t	(-)	Temperature factor
$C_{t,c}$	(-)	Characteristic temperature factor
$C_{t,E}$	(-)	Temperature factor for energy absorption
$C_{t,low}$	(-)	Characteristic temperature factor low
$C_{t,high}$	(-)	Characteristic temperature factor high
$C_{t,R}$	(-)	Temperature factor for reaction force
C_v	(-)	Velocity factor
$C_{v,c}$	(-)	Characteristic velocity factor
$C_{v,E}$	(-)	Velocity factor for energy absorption
$C_{v,R}$	(-)	Velocity factor for reaction force
c'	(m)	Horizontal distance between the fender centreline and the support structure due to the bow radius and fender compression

Symbol	Unit	Definition
CoV_M	(-)	Coefficient of variation of displacement of the vessel
D	(m)	Draught of vessel
D_F	(m)	Forward draught
D_A	(m)	Aft draught
$d_{support}$	(m)	Vertical elevation from fender centreline to the vessel deck or to the top of the supporting structure, whichever is the lower
E_{base}	(kNm)	Base fender energy absorption
$E_{f,c}$	(kNm)	Characteristic fender energy absorption
$E_{f,d}$	(kNm)	Design fender energy absorption
$E_{d,d}$	(kNm)	Design energy absorption for flexible dolphin structure
E_k	(kNm)	Kinetic energy to be absorbed by the fenders and structure during the impact
$E_{k,c}$	(kNm)	Characteristic energy to be absorbed by fenders in contact (and the supporting structure where applicable) during the impact
$E_{k,d}$	(kNm)	Design energy to be absorbed by fenders in contact during the impact
E_{SC}	(kNm)	Energy absorbed after single deflection at the standard condition
$E_{sts,c}$	(kNm)	Characteristic ship-to-ship berthing energy that needs to be absorbed by fender system at the time of impact
E_v	(kNm)	Kinetic energy of berthing vessel at the time of impact
F_c	(kN)	Characteristic berthing impact load acting on support structure
F_d	(kN)	Design berthing impact load acting on support structure
$F_{d;ND}$	(kN)	Non-Dominant Berthing Impact load acting on support structure
G	(kg)	Total weight of fender panel
h_{belt}	(m)	Belting height
h_f	(m)	Fender height
$h_{f,act}$	(m)	Horizontal distance from face of fender system to supporting structure, at actual fender compression, measured on the centreline of the fender for 50 % of the design energy
$h_{f,c}$	(m)	Horizontal distance from face of fender system to supporting structure, at the characteristic energy absorption, measured on the centreline of the fender
$h_{f,max}$	(m)	Horizontal distance from face of fender system to supporting structure, at maximum fender compression, measured on the centreline of the fender
$h_{f,SC}$	(m)	Fender height at the standard condition
h_{pan}	(m)	Fender panel height
K	(m)	Radius of gyration of vessel
L	(m)	Length of chain, pin-to-pin
L_{BP}	(m)	Length between perpendiculars of vessel
$L_{bow,F}$	(m)	Distance from the bow tangent point to the bow at the elevation of the fender
L_c	(m)	Distance to centre of mass of vessel from the stern
L_{OA}	(m)	Length overall of vessel
LOA_s		Length overall of vessel of smallest design vessel
M	(tonnes)	Displacement of the berthing vessel, equal to the total mass of the vessel and all its cargo
M_1	(tonnes)	Mass equivalent to displacement of the Vessel 1

Symbol	Unit	Definition
M_2	(tonnes)	Mass equivalent to displacement of the Vessel 2
$MBL_{d;ch}$	(kN)	Design minimum breaking load of one chain assembly
M_c	(tonnes)	Characteristic mass equivalent to water displacement of the berthing vessel at the time of impact [m/s]
M_{cv}	(tonnes)	Combined vessel mass coefficient
M_m	(tonnes)	Mean displacement of approaching vessel
$M_{sts,c}$	(tonnes)	Characteristic mass for energy of ship-to-ship berthing, including the effects of the displacement and added mass of both vessels
n	(-)	Annual berthing frequency
n_{chain}	(-)	Number of chains acting together
n_f	(-)	Number of compressed fenders
n_{ref}	(-)	Number of berthings during a certain reference period
P_{hull}	(kN/m ²)	Hull pressure
$P_{hull,av}$	(kN/m ²)	Average hull pressure
$P_{hulldmax}$	(kN/m ²)	Maximum hull pressure
$P_{hull,d}$	(kN/m ²)	Maximum equivalent permissible hull pressure
$P_{f,i}$	(-)	Probability of failure single berthing event i
$P_{f;t_{ref}}$	(-)	Probability of failure during a certain refence period t_{ref}
$P_{f;t_1}$	(-)	Annual Probability of failure
p_f	(m)	Fender system pitch
$p_{f,max}$	(m)	Maximum fender pitch [m]
$p_{f;max}$	(m)	Maximum fender system pitch
R_B	(m)	Bow radius
$R_{B,F}$	(m)	Bow radius at the elevation of the fender
R_{base}	(kN)	Base fender reaction force (mean)
$R_{base,c}$	(kN)	Characteristic base fender reaction force (mean)
$R_{base,max}$	(kN)	Maximum base fender reaction force (upper bound)
R_F	(kN)	Resultant reaction force of the fenders in contact
$R_{f,c}$	(kN)	Characteristic fender reaction force
$R_{f,d}$	(kN)	Design fender reaction force
R_{fi}	(kN)	Reaction force of the i^{th} compressed fender
$R_{f,lim}$	(kN)	Limiting value fender reaction force
R_{fn}	(kN)	Reaction force of the n^{th} compressed fender
$R_{f,sc}$	(kNm)	Reaction force after single deflection at the standard condition
r_F	(m)	Distance from the centre of mass of vessel to the resultant fender reaction force
r_L	(m)	Distance from the centre of mass of vessel to the hull contact point of resultant fender reaction force parallel to the berthing line
r_{Li}	(m)	Distance from the centre of mass of the vessel to compressed i^{th} fender perpendicular to the berthing line
r_{Ln}	(m)	Distance from the centre of mass of the vessel to compressed n^{th} fender perpendicular to the berthing line

Symbol	Unit	Definition
r_s	(m)	Distance from the centre of mass of vessel to the hull contact point of resultant fender reaction force parallel to the berthing line
r_{si}	(m)	Distance from the centre of mass of vessel to compressed i^{th} fender parallel to the berthing line
r_{sn}	(m)	Distance from the centre of mass of vessel to compressed n^{th} fender parallel to the berthing line
T_0	(°C)	Standard temperature
$T_{c,high}$	(°C)	Average monthly ambient air temperature of the hottest month
$T_{c,low}$	(°C)	Average ambient air temperature of the coldest day of the year with a return period of five years
$T_{max,cold}$	(°C)	Average daily maximum ambient temperature of the coldest month
$T_{mean,cold}$	(°C)	Average daily ambient temperature of the coldest month
$T_{min,cold}$	(°C)	Average daily minimum ambient temperature of the coldest month
$T_{max,hot}$	(°C)	Average monthly maximum ambient temperature of the hottest month
$T_{mean,hot}$	(°C)	Average monthly ambient air temperature of the hottest month
$T_{min,hot}$	(°C)	Average monthly minimum ambient temperature of the hottest month
t_{ref}	(year)	Reference period
U_C	(m)	Vessel under keel clearance
U_{CF}	(m)	Vessel under keel clearance (Forward)
U_{CA}	(m)	Vessel under keel clearance (Aft)
V	(m/s)	Berthing velocity of vessel at the time of impact
V_0	(mm/s)	Compression speed
V_B	(m/s)	Berthing velocity of vessel at the time of impact, perpendicular to the berthing line
$V_{B,c}$	(m/s)	Characteristic berthing velocity of vessel at the time of impact, perpendicular to the berthing line
V_{B1}	(m/s)	Berthing velocity of the Vessel 1
V_{B2}	(m/s)	Berthing velocity of the Vessel 2
V_c	(m/s)	Characteristic berthing velocity at the time of impact
V_{CV}	(m/s)	Fender compression speed for the Constant Velocity
V_{DV}	(m/s)	Fender compression speed for the Decreasing Velocity
V_L	(m/s)	Berthing velocity of vessel at the time of impact, parallel to the berthing line
V_{L1}	(m/s)	Berthing velocity of the Vessel 1 at the time of impact, parallel to the virtual berthing line (ship-to-ship berthing)
V_{L2}	(m/s)	Berthing velocity of vessel at the time of impact, parallel to the virtual berthing line (ship-to-ship berthing)
$V_{L,c}$	(m/s)	Characteristic berthing velocity of vessel at the time of impact, parallel to the berthing line
$V_{sts,c}$	(m/s)	Characteristic closing berthing velocity between vessels
w_{pan}	(m)	Fender panel width
$X_{bow;ballast\ wl}$	(m)	Distance from the bow to the tangent line at ballast water level
$X_{bow;deck}$	(m)	Distance from the bow to the tangent line at deck level
$X_{bow;laden\ wl}$	(m)	Distance from the bow to the hull/bow tangent point at water level for a laden vessel
x_B	(m)	Distance from bow at the elevation of the fender to centroid of fender reaction force

Symbol	Unit	Definition
x_{chain}	(m)	Panel movement due to chain arc
x_F	(m)	Distance from hull bow point (corresponding to L_{BP}) to the hull contact point of fender reaction force (R_F) at the level of fender contact
y_F	(m)	Distance from vessels centre of mass to the hull contact point of fender reaction force (R_F)
α	(degrees)	Berthing angle
α_c	(degrees)	Characteristic berthing angle
α_i	(degrees)	Incidental berthing angle
β_t	(-)	Target reliability index
$\beta_{t;t_{ref}}$	(-)	Reliability target related to a certain reference period t_{ref}
β_f	(degrees)	Bow or hull flare angle
Δh_f	(m)	Fender compression/ deflection
$\Delta h_{f,c}$	(m)	Characteristic fender compression
$\Delta h_{f,i}$	(m)	Compression of fender i
Δh_{max}	(m)	Maximum fender compression
ΔT	(°C)	Temperature difference
γ_c	(-)	Correction factor for correlations between design variables
$\gamma_{E,ref}$	(-)	Reference partial energy factor for 100 berthings per year
γ_f	(-)	Partial resistance single fender energy absorption factor
γ_n	(-)	Correction factor for alternative annual berthing frequencies
γ_m	(-)	Partial resistance single fender energy material factor
γ_{mult}	(-)	Partial resistance multiple fender contact factor
γ_p	(-)	Correction factor for berthings without pilot assistance
γ_R	(-)	Partial load factor applicable to the characteristic reaction force
γ_Q	(-)	Partial load factor from international code, standard or national annex
θ	(degrees)	Angle of chain(s)
θ_0	(degrees)	Static angle of chain(s), undeflected fender
θ_1	(degrees)	Dynamic angle of chain(s), deflected fender
θ_t	(degrees)	Angle of tension chain
θ_s	(degrees)	Angle of shear chain
θ_w	(degrees)	Angle of weight chain
μ_f	(-)	Coefficient of friction
μ_M	(tonnes)	Mean displacement of the approaching vessels
ρ_w	(tonne/m ³)	Density of water
σ_M	(tonnes)	Standard deviation of the displacement of the approaching vessels
$\emptyset$	(degrees)	Angle between velocity vector and the line between the hull contact point of the resultant fender reaction force and the centre of mass of vessel
Φ	(-)	Standard normal cumulative distribution function
φ_Q	(-)	Combination factor from international code, standard or national annex
ω_0	(radians/s)	Initial angular velocity of the berthing vessel

APPENDIX E: TERMS OF REFERENCE WG 211

Terms of Reference Updating of WG 33 – ‘Guidelines For The Design Of Fender Systems’

1. Background

Marine fenders are a critical component in maritime infrastructure, absorbing the energy of vessels during berthing and while moored, and protecting both vessels and berth structures from damage.

In 1984 PIANC published a Supplement to Bulletin No. 45, containing improved design methods for fender systems.

This guideline was followed in 2002 by PIANC WG 33, which included more advanced fender design and testing methods, taking into account performance modification factors for manufacturing tolerances, temperature and velocity.

Since the WG 33 guideline was published:

- there have been further advances in design methods for fender systems
- vessel dimensions and hull shapes have further evolved
- WG 145 (being finalised) has collected and analysed berthing velocities and angles
- WG 145 has also addressed reliability design, vessel dimensions and container vessel flare angles
- WG 186 (in progress) is considering mooring requirements for large vessels at quay walls
- an update of WG 24 is proposed ('Criteria for Movement of Moored Ships in Harbours')
- manufacturers have undertaken further research into fender materials, performance, durability and impacts of aging
- improvements to WG 33 guidelines have been suggested by users, including improved fender testing and verification procedures, performance requirements for fender system elements and the addition of maintenance and repair guidelines
- automated mooring systems are increasingly being used in conjunction with conventional fender systems.

Under these circumstances it is proposed to update WG 33.

2. Objectives and Scope

Prepare a general update of WG 33, including:

- update the basis of design and design guidance in relation to:
 - alignment with current practice including reliability design methods
 - latest simulation software for evaluation of fender/vessel interaction
 - recent vessel trends including increases in vessel sizes, developments in hull shapes and the implications for fender engagement
 - vertical and horizontal forces on fenders
 - hull pressures
 - outline consideration of special issues applying to wheel fenders and foam filled fenders
- consider and incorporate relevant outcomes of other working groups including:
 - WG 145 in relation to analysis of berthing velocities and angles
 - WG 145 in relation to reliability design
 - WG 145 in relation to the problem of hull flare angles on large container vessels
 - WG 186 in relation to mooring requirements of large vessels at quay walls
 - WG 24 (update proposed) in relation to acceptable movements of vessels at berth

- review recent research by fender manufacturers and update guidance in relation to durability and performance, including:
 - fender materials composition and influence on performance factors
 - fender durability and causes of failure
 - impact of aging on fender performance and reactions
- review the WG 33 guidance in relation to testing procedures for fender materials and fender performance, with specific reference to:
 - representative scope of testing
 - consistency of test procedures and compliance in testing facilities
 - representative quality and quantity of samples to be tested
 - independence of testing and verification of fender materials and performance
- clarify that PIANC is not a fender certifying body and cannot endorse certification of compliance by third parties using the guidelines
- provide guidance on design of other fender system components including facing panels, chains and fixings
- provide guidance on durability, maintenance and repair of fender system components
- provide guidance on the implications of automated mooring systems used in conjunction with fender systems.

3. Documents to be Reviewed

Documents to be reviewed include, but are not limited to, the following:

- Supplement to PIANC Bulletin No. 45 (1984): "Report of the International Commission for Improving the Design of Fender Systems".
- PIANC MarCom Working Group 33 (2002): "Guidelines for the Design of Fender Systems".
- EAU (2004): "Recommendations of the Committee for Waterfront Structures, Harbours and Waterways".
- "Technical Standards and Commentaries for Port and Harbour Facilities in Japan", 2009.
- Spanish ROM 0.2-11 (2011): "Recommendations for the Planning and Execution of Berthing and Mooring Works".
- BS6349 (2014): "Code of Practice for Design of Fendering and Mooring Systems", Maritime Structures Part 4.
- PIANC MarCom Working Group 145 (2020): "Berthing Velocity Analysis of Seagoing Vessels over 30,000 dwt".
- PIANC MarCom Working Group 24 (1995) (update proposed): "Criteria for Movement of Moored Ships in Harbours".
- PIANC MarCom Working Group 186 (in preparation): "Mooring of Large Ships at Quay Walls".
- "Maintenance Guideline for Rubber Marine Fenders", Coastal Development Institute of Technology Library, No.40, 2013.
- Recent technical papers and presentations including the following presented at the PIANC Australia Fender Workshop and PIANC/IEAust Ports and Coasts Conference (2017):
 - "Approaches to Life Evaluation of Rubber Marine Fenders", by H. Akiyama, T. Shiomi, K. Shimizu, S. Yamamoto, S. Ueda and T. Kamada.
 - "Life Evaluation of Fenders Including Effect of Compounding Fillers", presentation by H. Akiyama, Bridgestone Corporation.
 - "Large Vessel Profile and Fender System Design Study", presentation by H. Akiyama, Bridgestone Corporation.
 - "Ensuring Fender Performance through Compression and Material Testing", presentation by M. Kumar, Trelleborg Marine Systems.

- “Fender Compression and Durability Testing – First Hand Experience and Room for Improvement”, presentation by H. Singh, CH2M.

6. Intended Product

- The intended product is an update of WG 33 providing concise and clear design guidance for fender systems, in alignment with current practice, together with updated guidance in relation to fender materials, performance, testing, durability and maintenance.

5. Working Group Membership

Working Group members should be drawn from:

- Marine terminal owners/operators
- Marine infrastructure designers
- Marine fender and mooring system manufacturers
- Relevant research organisations.

6. Relevance to Countries in Transition

The updated guideline will be of value to countries in transition by providing practical guidance on fender system selection, procurement and maintenance, and on the impact of aging on fender performance.

7. Relevance of Climate Change

- The activities of this Working Group will not be related directly to climate change.



PIANC HQ: Boulevard du Roi Albert II 20 B. 3 | 1000 Brussels | Belgium

<http://www.pianc.org> | VAT BE 408-287-945 | ISBN 978-2-87223-036-5

© All rights reserved