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Guidelines for Design and Testing of Rubber Fender Systems

June 2019

Coastal Development Institute of Technology

Foreword to the English Edition

The fender system plays a key role in the safe berthing of a vessel and stable loading/unloading as auxiliary equipment at the mooring facility. “The Technical Standards and Commentaries for Port and Harbor Facilities in Japan” provide the design method and “the Standard Specifications for Port & Harbor Projects” provide the specifications and testing methods of rubber fender system. Although a change in performance of rubber fender due to differences in berthing speed and environmental conditions was not considered in the design of fender systems according to above, knowledge of the effect of performance changes taking the characteristics of rubber into consideration is increasing in recent years.

Rubber is a material which has viscosity and elasticity and works as a shock absorber for huge, heavy and hard materials like concrete and steel. On the other hands, it is known that the performance of rubber fender is sensitive to using condition and environment.

Accordingly, in order to provide accurate and advanced design and testing methods of rubber fenders system, it was decided to draw up new practical guidelines for more logical design and testing methods of the rubber fenders system indicating the methods of evaluating the effects of performance changes due to berthing speed and environmental conditions. Collaborative research by the Coastal Development Institute of Technology and major five manufacturers of the rubber fender system in Japan started in August 2015. An expert committee was established in fiscal 2017 where discussions had been made on the draft guidelines prepared by collaborative research, which is now finalized and published as the Guidelines for the Design and Testing of Rubber Fender Systems. These guidelines are intended to be specific and easy to understand with many graphical references and design examples of rubber fenders system in the Appendices.

It is my great pleasure to publish the English Edition of this guideline and I hope this will be widely disseminated and contribute the technology of rubber fender in ports around the world.

高橋重雄

Shigeo Takahashi, President
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August, 2019

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Guidelines for Design and Testing of Rubber Fender Systems

CONTENTS

Chapter 1
Preface

A rubber fender is a shock absorber installed on a wharf to absorb the berthing energy of a vessel to relieve impact. In the past, timber and old tires were used before rubber fenders capable of efficiently absorbing the berthing energy were developed. As vessels became larger, the rubber fenders were enhanced in terms of performance and size. A critical factor regarding the selection of rubber fenders is the effective berthing energy of the vessel. With increase in the vessel size, the rubber fenders are required to absorb a large amount of energy, and buckling-type rubber fenders, in which the reaction force does not increase considerably, have been widely used.

In Japan, the design method for rubber fenders is specified by the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [1)], and the specifications and test methods are specified by the Standard Specifications for Ports and Harbour Works [2)]. In contrast, overseas, the British Standard [3)], PIANC (the World Association for Waterborne Transport Infrastructure) Report [4)] and PIANC Guidelines [5)] regulations are widely applied. The PIANC Guidelines [5)], issued in 2002, propose a design method that takes into account the operational conditions and natural environment and their influence on the rubber fender performance in terms of, for example, the berthing velocity and temperature. In Japan, for a long time, only production tolerance has been considered as a performance influence factor; however, overseas, the design and test methods are expected to incorporate the influence of the berthing velocity and temperature. In addition, it is necessary to consider the ageing performance of the rubber fenders to make effective use of existing facilities and prolong the service life.

Considering this background, this guideline is designed in accordance with the PIANC Guidelines [5)] for buckling-type rubber fenders, with the objective of taking into consideration factors such as the ageing of rubber fenders and providing a test method for the design factors. In the design method, in addition to the production tolerance conventionally considered in Japan, the influence factors of rubber fenders, such as the berthing angle, berthing velocity, temperature, repetition fatigue, ageing and the effect of creep characteristics, are considered. Furthermore, critical points regarding the installation distance and height of fenders and design methods for fender panels and chains that are often installed in the case of large rubber fenders are presented. Furthermore, as the test method, we propose a standard method for appropriately evaluating the performance of rubber fenders required for the given design method. The design case study is presented as an appendix.

[References]

1) Ports and Harbours Association of Japan (PHAJ): Technical Standards and Commentaries of Ports and Harbour Facilities in Japan, pp. 1262-1268, 2018
2) Ports and Harbour Bureau, Ministry of Land, Infrastructure, Transport and Tourism (MLIT): Standard Specifications for Ports and Harbor Works, Vol.1, pp.71-72, 2017
3) British Standard: Maritime Structures-Part 4 Code of Practice for Design of Fendering and Mooring Systems, 2014
4) PIANC: Report of the International Commission for Improving the Design of Fender Systems, Bulletin No.45, 1984
5) PIANC: Guidelines for the Design of Fenders System, Report of Working Group 33 of the Maritime Navigation Commission, 2002

Chapter 2
Nomenclature

Symbol		Explanation
A	:	Effective contact area of fender panel
A_{sway}	:	Lateral wind pressure area of vessel
B	:	Beam of vessel
C_a	:	Coefficient of angular berthing
C_{aE}	:	Energy coefficient of angular berthing
C_{agR}	:	Reaction force coefficient of ageing
C_{agE}	:	Energy coefficient of ageing
C_{aR}	:	Reaction force coefficient of angular berthing
$C_{aR(\varepsilon)}$	:	Reaction force coefficient of angular berthing at strain ε
C_b	:	Block coefficient
C_c	:	Berth configuration factor
C_e	:	Eccentricity factor
C_m	:	Virtual mass factor
C_p	:	Coefficient of production error
C_r	:	Coefficient of repetition
C_{re}	:	Coefficient of recovery
C_s	:	Softness factor
C_w	:	Wind drag coefficient of vessel
C_{Φ}	:	Coefficient of angular compression direction
d	:	Draught of vessel
D	:	Allowable deflection of fender
D_S	:	Height of vessel
DT	:	Displacement tonnage of vessel (t)
e	:	Ratio of fender spacing to perpendicular length
E	:	Young’s modulus
E_A	:	Energy absorption (standard value in catalogue)
E_{Aag}	:	Energy coefficient of ageing
E_{AT}	:	Energy absorption at temperature T
$E_{A\theta}$	:	Energy absorption at berthing angle θ
E_{AV}	:	Energy absorption at berthing velocity V
E_b	:	Effective berthing energy
f	:	Freeboard height of vessel
F	:	Berthing force
F_j	:	External force vector of moored vessel
F_m	:	Reaction force of scale model
F_R	:	Reaction force of actual-size fender
F_R	:	Wind drag on vessel
H	:	Fender height
H_F	:	Height of fender panel

Symbol		Explanation
H_{Fe}	:	Effective height of fender panel
H_p	:	Allowable hull pressure
h	:	Residual height of fender after compression
k	:	Parameter for berthing point between fenders
K_{ij}	:	Restoration coefficient matrix of moored vessel
K_r	:	Radius of vessel rotation
L	:	Distance between contact point and fender centre
L_f	:	Length of fender
L_c	:	Length of chain
L_{ij}	:	Delay function of moored vessel
L_{oa}	:	Overall length of vessel
L_{pp}	:	Length between perpendiculars
Ly	:	Contact length of V-type fender
m_{ij}	:	Additional mass matrix of moored vessel
M	:	Mass of vessel
M_f	:	Rotational moment at fender top
M_{ij}	:	Mass matrix of moored vessel
N_{ij}	:	Dumping coefficient matrix of moored vessel
P	:	Average hull pressure
R	:	Reaction force of fender
R_R	:	Standard reaction force of fender (catalogue value)
$R_{F1(\varepsilon)}$	:	Reaction force at strain ε under static compression loading
$R_{F2(\varepsilon)}$	:	Reaction force at strain ε under static unloading
Rr	:	Radius of hull curvature at berthing point
R_S	:	Distance between berthing point and vessel gravity centre parallel to berth line
R_V	:	Reaction force of fender at compression speed V
$R_{V(\varepsilon)}$	:	Reaction force at compression speed V and strain ε
R_T	:	Reaction force of fender at temperature T
$R_{T(\varepsilon)}$	:	Reaction force of fender at temperature T and strain ε
$R(\varepsilon)$	:	Reaction force of fender at strain ε
R_θ	:	Reaction force at berthing angle θ
$R_\theta(\varepsilon)$	:	Reaction force at berthing angle θ and strain ε
S	:	Installation spacing of fenders; scale of modelling
T	:	Tension of chain
TF	:	Temperature factor
TF_R	:	Temperature factor of reaction force
TF_E	:	Temperature factor of energy absorption
$TF_{R(\varepsilon)}$	:	Temperature factor of reaction force at strain ε
$TF_{E(\varepsilon)}$	:	Temperature factor of energy absorption at strain ε
U_a	:	Wind speed
V	:	Compression speed (Strain rate)
V_B	:	Berthing velocity (Speed)
VF	:	Velocity factor

Symbol		Explanation
VF_R	:	Velocity factor of reaction force
VF_E	:	Velocity factor of energy absorption
$VF_{E(\varepsilon)}$	:	Velocity factor of energy absorption at strain ε
$VF_{R(\varepsilon)}$	:	Velocity factor of reaction force at strain ε
V_0	:	Static compression speed (V_0=0.01–0.3%/s)
W_F	:	Width of fender panel
W_{Fe}	:	Effective width of fender panel
W_W	:	Weight of fender panel
x	:	Displacement of vessel in sway
$X_i(t)$	:	Displacement vector of vessel centre at time t
α	:	Ratio of parallel bar length to perpendicular length of vessel
β	:	Angle between chain and fender panel
γ	:	Angle between velocity vector and line connecting berthing point and centre of gravity of vessel
δ	:	Deflection of fender
δ_0	:	Deflection at fender centre when vessel contacts the full surface of fender panel during angular berthing
δ_1	:	Virtual displacement of origin under catalogue value of angular performance
ε	:	Strain (deflection) of object (fender)
η	:	Viscosity coefficient of rubber
θ	:	Berthing angle; angle of berthing point
θ_a	:	Wind incident angle to vessel
θ_W	:	Wave incident angle to vessel
μ	:	Friction coefficient
μ_V	:	Average berthing velocity
ρ	:	Density
σ	:	Standard deviation; stress
τ	:	Stress due to viscosity of rubber
φ	:	Angle of compression direction

Chapter 3
Role of rubber fenders and their types

3.1 General

A rubber fender is a shock absorber installed on a wharf to absorb the berthing energy of a vessel to be berthed and moored. Since various types of rubber fenders are manufactured according to the application, it is necessary to select and design fenders appropriately according to the ship type and berth type. This document focuses on buckling-type rubber fenders because of their large ratio of energy absorption to reaction force for ship berthing. The purpose, type, requirements and buckling-type performance of rubber fenders are described in the following subsections.

The following types of rubber fenders are also used in ports; however, they are not described in this document because they differ from buckling-type rubber fenders.

(1) Rubber fenders with gradually increasing reaction force, such as cylindrical rubber fenders

(2) Pneumatic fenders (for details, readers can refer to ISO 17357-1)

(3) Rotary fenders

(4) Shock absorbers such as square-type, D-type and ladder fenders

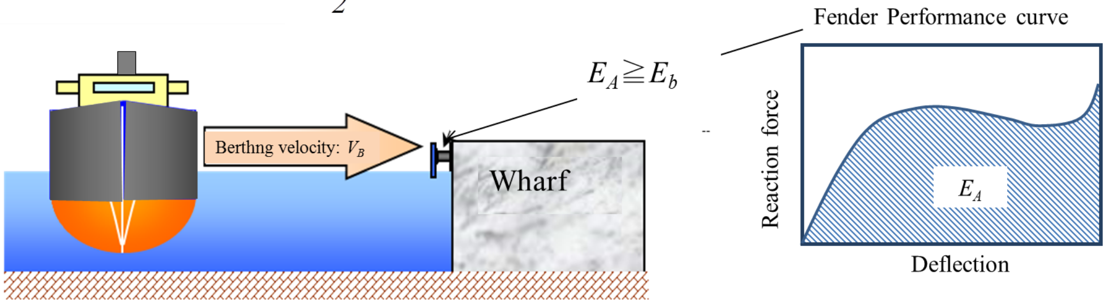
(5) Rubber net and corner protection material

(6) Rubber fenders applied in facilities other than ship mooring facilities, such as bridge piers, canals and locks

3.2 Role of rubber fenders

A rubber fender is a shock absorber installed on a wharf to absorb the berthing energy of a vessel. Timber and old tires were used as shock absorbers until the first rubber fenders were developed in the 1950s. The concept of the role of rubber fenders in the berthing process is illustrated in Figure 3.2.1.

Effective berthing energy: $E_b = \frac{1}{2} \cdot M \cdot V_B{}^2 \cdot C_e \cdot C_m \cdot C_c \cdot C_s$



Here,

E_b : Effective berthing energy (kN·m)

M : Mass of vessel (=Displacement tonnage: DT, t)

V_B : Berthing velocity (m/s)

C_e : Eccentricity factor

C_m : Virtual mass factor

C_c : Berth configuration factor

C_s : Softness factor

Fig. 3.2.1 Concept of berthing

As shown in Fig. 3.2.1, the effective berthing energy E_b is the multiplication product of the normal kinetic energy of vessel: $1/2 \cdot M \cdot V_B{}^2$ and the coefficients (Ce, Cm, Cc, and Cs), as defined in equation (5.3.1) in Chapter 5. Furthermore, as shown in Fig. 3.2.2, the area between the reaction force curve and deflection amount in the x-axis corresponds to the amount of absorbed energy. Thus, for fenders, the absorbed energy E_A must be equal to or more than the effective berthing energy E_b of the vessel.

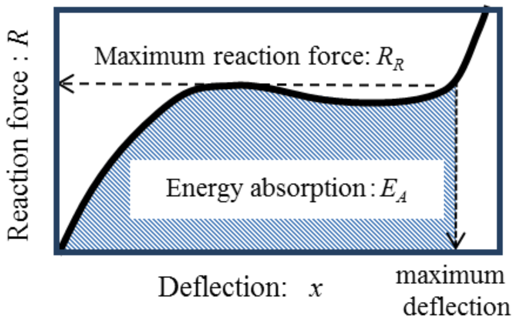


Fig. 3.2.2 Performance curve of buckling-type rubber fender

The reaction force R of a fender acts as a horizontal force on the wharf or on the side hull of the berthing vessel; thus, it is desirable that the reaction force is as small as possible. In contrast, a larger energy absorption E_A is desirable. Therefore, as shown in Fig. 3.2.2, the reaction force curve of the fender exhibits a high spring constant at the initial stage of compression and a constant portion later at higher deflection. Such characteristics are a result of the buckling deformation of the rubber fenders. The initial stage of compression corresponds to a high spring constant; however, once the peak of reaction force is exceeded, the spring constant decreases due to buckling, and as compression progresses, the value of the spring constant increases again, exceeding the value of the previous peak. Assuming that the point at which the reaction force exceeds the peak again is the maximum deflection, and the reaction force R_R at the peak point is the maximum reaction force; if the compression is within the maximum deflection, the reaction force is the maximum horizontal force to the wharf and vessel hull. This mechanism corresponds to the buckling-type performance, and it is currently the mainstream concept for the design of rubber fenders. In the PIANC Guidelines [1)], the coefficient obtained by dividing the energy absorption E_A by the maximum reaction force R_R is known as the fender factor. In terms of fender design, maximizing the fender factor is considered essential to ensure efficient performance. For the same type of rubber fenders, if performance fluctuations exist due to the design condition, the maximum reaction force and the maximum deflection in the design condition are considered to be the design reaction force and design deflection, respectively. This aspect is explained in detail in Chapter 4.

With increase in the size of vessels, the requirements for rubber fenders have diversified; furthermore, it has become necessary to increase the area contacting the vessel hull to reduce the contact pressure, taking into account the hull strength. To this end, a configuration in which a fender panel is fixed to the front of a rubber fender is widely used. In particular, fender panels are attached to most rubber fenders for large vessels. Rubber fenders without fender panels are primarily used for medium- or small-sized vessels for which the hull pressure is not a design issue. Thus, a rubber fender that satisfies the given requirements can be selected. The fender performance can also be adjusted by changing the rubber material, that is, by using materials with a different performance grade or rubber grade.

Fenders also play a crucial role in mooring. A situation in which a ship is moored on a quay is shown in Fig. 3.2.3. In this case, the fender acts when the vessel approaches the quay, whereas the tension of the mooring ropes is the acting force when the vessel moves farther.

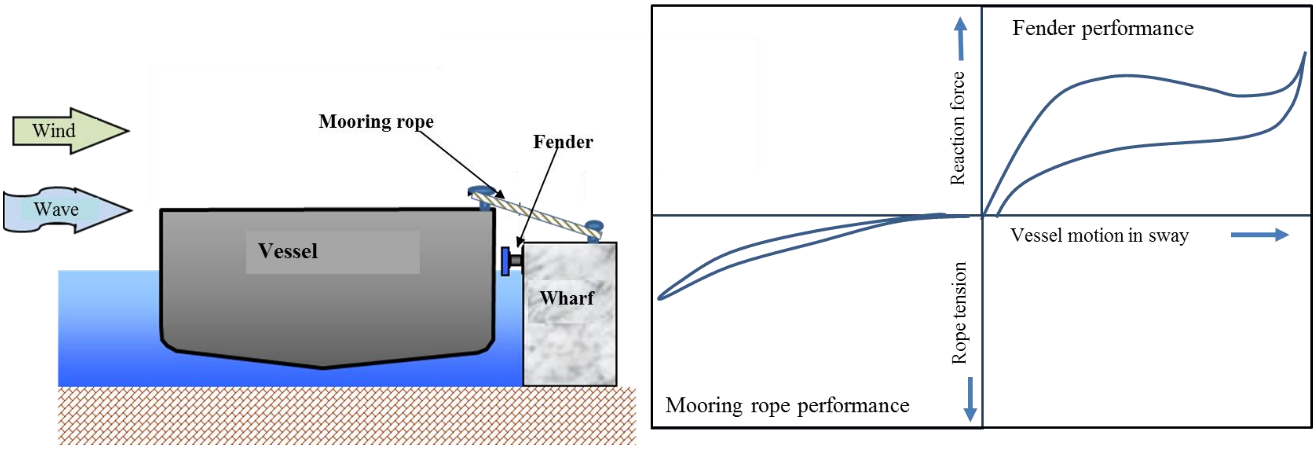


Fig. 3.2.3 Concept of vessel moored using fenders and ropes

3.3 Rubber fenders with fender panels

A fender panel is a frame made primarily of steel, which is fixed to the front of a rubber fender to lower the average face pressure. A rubber fender with a panel has the disadvantages of complex design due to the increase in the number of system parts such as the panel, pads and chains; however, it also has several advantages, as listed. With the increase in the size of vessels, several large fenders have been equipped with fender panels.

(1) The average hull pressure can be adjusted.
(2) The rubber is not in direct contact with the vessel hull.
(3) The system demonstrates satisfactory performance, even in the case of partial contact with projections on the hull.
(4) It is possible to adjust the panel length and number of fenders per panel.

A typical installation of rubber fenders with panels (vertical cylinder type, rectangular type) is shown in Fig. 3.3.1.1 and Fig. 3.3.1.2.

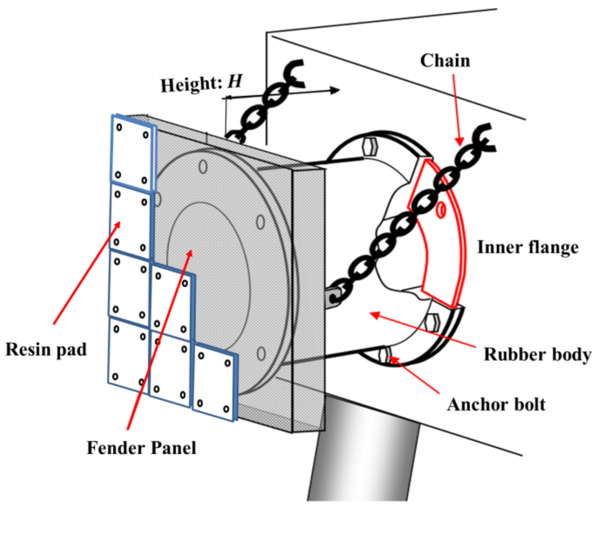


Fig. 3.3.1.1 Rubber fenders with panel
(vertical cylinder type)

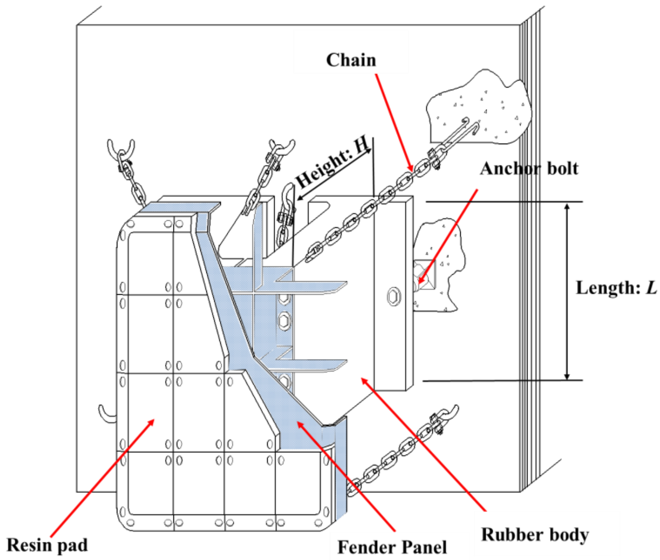


Fig. 3.3.1.2 Rubber fenders with panel
(rectangular type)

The shape of rubber fenders varies depending on the manufacturer. Several examples are shown in Fig. 3.3.2 to Fig. 3.3.6. A special type of V-type fender can have a panel fixed on the top, as shown in Fig. 3.3.6.

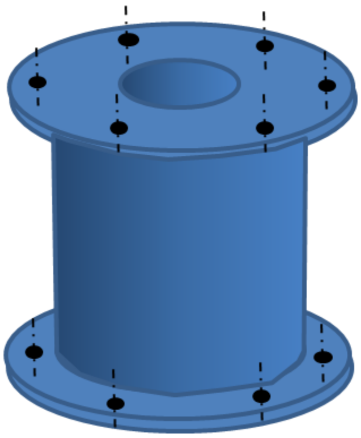


Fig. 3.3.2 Vertical cylinder type

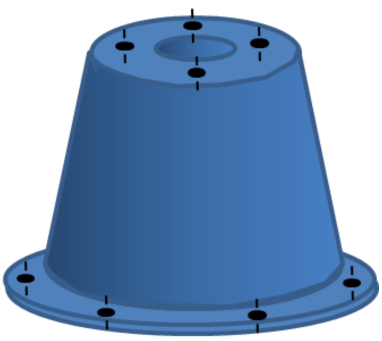


Fig. 3.3.3 Cone type

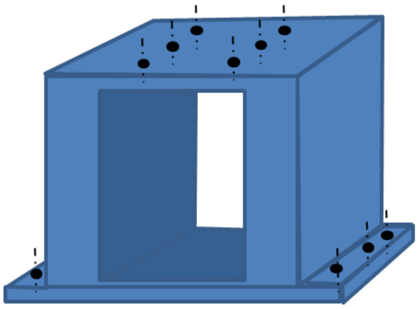


Fig. 3.3.4 Rectangular column
(one-piece)

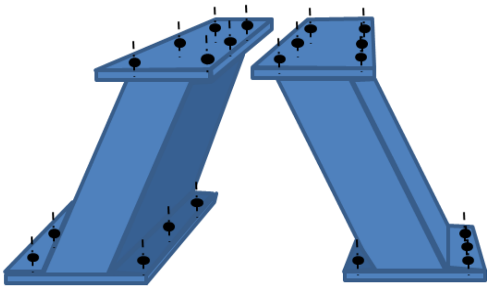


Fig. 3.3.5 Rectangular column
(Separated element)

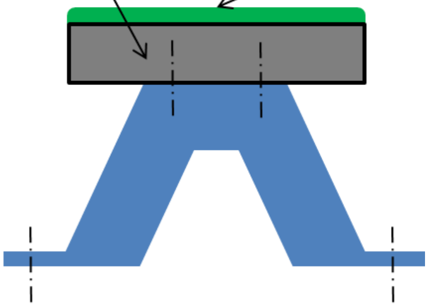


Fig. 3.3.6 V-type fender with panel

There are no specific criteria regarding whether a rubber fender should be fitted with fender panels. Panels are often installed if there is an allowable limit for the hull pressure. The considerations for hull pressure are described in Section 5.7.1 in Chapter 5. Even if there is no restriction regarding the hull pressure, a fender panel may be required when a large contact area is required, for example, in the case of fenders for ferries.

3.4 Rubber fenders without fender panels

In this section, V-type rubber fenders are considered as rubber fenders without fender panels. A V-type rubber fender has a V-shaped cross section with steel plates embedded in the fixing flange portion for installation. The fender has the same cross-sectional shape in the longitudinal direction. Since the rubber comes in direct contact with the vessel hull, partial damage is likely to occur. Energy absorption occurs only due to the contact length, and the contact pressure is high. Although these are notable disadvantages, V-type rubber fenders have the following advantages.

(1) The application range is wide because the length can be adjusted.

(2) Since such fenders exhibit buckling-type performance, the energy absorption is large even for small-sized fenders.

V-type rubber fenders were widely used as the first rubber fenders with buckling-type performance. The typical cross-sectional shapes and installation of V-shaped rubber fenders are shown in Fig. 3.4.1.

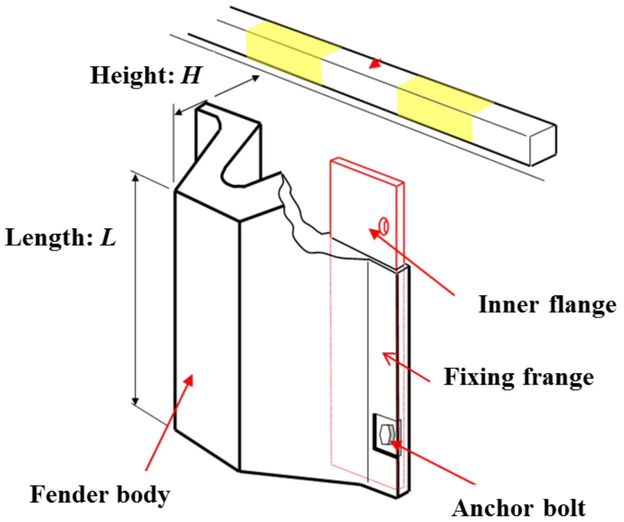


Fig. 3.4.1 V-type fender: Section and installation

This type of rubber fender has had a long development history; first, a flange integrated-type fender was developed, as shown in Fig. 3.4.2. Since then, various shapes have been developed, which have been incorporated in V-type rubber fenders.

Fig. 3.4.2 Flange integrated

Fig. 3.4.3 Centre supported

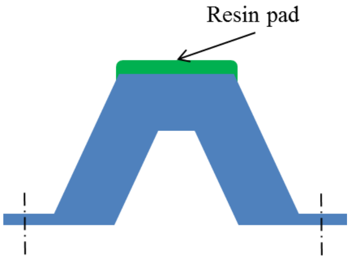


Fig. 3.4.4 With resin pad

V-type rubber fenders can be widely applied according to the conditions of ports and vessels by changing the length and mounting arrangement. Typical mounting arrangements and features are presented in Table 3.4.1. For example, if the ship has a belt on the hull, the effective berthing energy can be absorbed by the horizontal fenders, although the hull belt may ride on the fender and damage it. Furthermore, in the case of vertical installation, if a horizontal hull belt is present, the load is concentrated since contact occurs only at the intersection. In such a case, alternate vertical and horizontal installation can be employed; the effective berthing energy is absorbed by the horizontal fenders, while the vertical fenders prevent the belt from being lifted. Alternatively, the rubber fenders with panels, as described in the previous section, can be applied.

Moreover, as shown in Fig. 3.4.5, a rubber fender shaped such that the V-shaped cross-section is pivoted is also available. The length of such fenders cannot be changed; however, such fenders are nondirectional to angular and shear deformations.

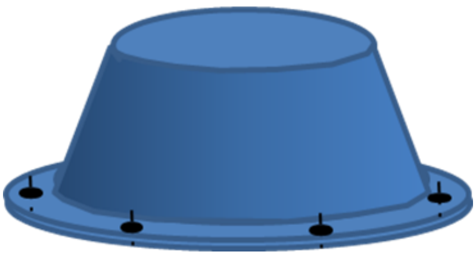
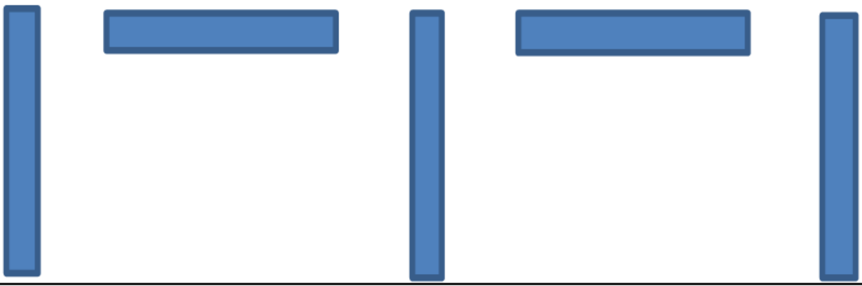
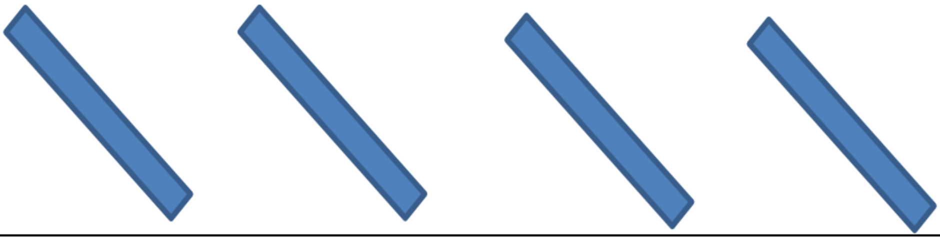


Fig. 3.4.5 Pivoted V-type fender

Table 3.4.1 Typical arrangements and features of V-type rubber fender

Vertical	 The most common installation at fishing ports. It can cope with the up-and-down motion of ship and changes in water levels. In case the fender pitch is large, there is a risk of contact occurring between fenders.
Horrizontal	 Suitable for applications involving a narrow installation space or small vertical fluctuations in the berthing position. Most often used in floating piers. However, they may get caught on horizontal projections such as hull belts.
Alternate	 Combines the advantages of the horizontal and vertical configurations by employing an alternate vertical and horizontal installation approach.
Inclined	 Reduces the number of fenders while exploiting the advantages of the alternate vertical and horizontal installation configuration.

[References]

1) PIANC: Guidelines for the Design of Fenders System, Report of Working Group 33 of the Maritime Navigation Commission, 2002

Chapter 4
Performance of rubber fenders

4.1 General

In the design process, the performance of rubber fenders is considered in terms of the deflection–reaction force characteristics at compression and the energy absorption calculated from these characteristics. Rubber has visco-elastic properties, and its performance is affected by the natural environment and conditions of use. Factors that affect the performance are called influence factors, and they must be taken into consideration when designing rubber fenders. In the PIANC Guidelines [1], factors such as temperature and the influence of berthing speed are specifically defined. The performance under deceleration from the berthing velocity is called the rated performance as it is similar to that during actual berthing. This book defines the standard performance as one that is more manageable and demonstrable while being consistent with the guidelines. In addition, other influence factors introduced in the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [2, 3], which include the angle, repetition fatigue, ageing, and creep, are also considered.

4.2 Influence factors

The Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [2, 3] introduce the following seven factors that affect the performance of rubber fenders. In this book, the standard conditions for each factor are set as follows. The concept of influence coefficients and their effect on the fender design are clarified in Sections 4.2.1 to 4.2.7.

Influence factor	Standard condition
(1) Production tolerance: C_P	1.0 (Standard value)
(2) Angular factor: C_a	Standard angle θ_0 = 0°
(3) Velocity factor: VF	Standard strain rate V_0 = 0.01 to 0.3%/s
(4) Temperature factor: TF	Standard temperature T_0 = 23°C
(5) Ageing: C_{ag}	0 years after production (in principle, within 6 months)
(6) Repetition fatigue: C_r	1.0 (Standard value: the first performance value after fender has been compressed 3 times or more and left for one hour or more)
(7) Creep characteristics	1.0 (No steady load)

A factor influencing the performance can be expressed as in equation (4.2.1) or equation (4.2.2) in terms of the corresponding ratio to the standard performance. The standard performance refers to the basic performance described in manufacturers' catalogues, and it pertains to the performance when all influence factors (1) to (7) correspond to the standard conditions. The performance pertaining to actual design conditions is known as the design performance.

$$\text{Influence factor of reaction force} = \frac{\text{Reaction force under influence factor}}{\text{Reaction force pertaining to standard performance}} \tag{4.2.1}$$

$$\text{Influence factor of energy absorption} = \frac{\text{Energy absorption under influence factor}}{\text{Energy absorption pertaining to standard performance}} \tag{4.2.2}$$

Regarding the velocity factor, the PIANC Guidelines [1] specify the rated performance as corresponding to decelerating compression from the initial speed of 0.15 m/s. However, in this document, apart from the rated performance, the performance at a sufficiently slow speed is specified as the standard performance. This definition is simply a matter of convenience and ability of the test facility. Although the rated performance should be exhibited in actual berthing, the demonstrable performance is regarded as the standard performance. The practical design is expected to correspond to the design performance based on design conditions.

Since the influence coefficients change because of compressive strain, equations (4.2.1) and (4.2.2) can be obtained individually for each strain. The final value obtained at the design strain is termed as the influence factor, and a value at any other strain is known as a strain-specific influence factor. When the performance curve exhibits a shape similar (within 3% of error) to that of the variation of the influence factor, the influence factor is constant regardless of the strain. In such a case, the influence coefficients of the reaction force and energy absorption are also equivalent. When selecting a rubber fender, it is not necessary to determine strain-specific values, and determining only the coefficient at the design strain is sufficient. Thus, the strain-specific influence factors are considered only when required.

4.2.1 Coefficient of production tolerance: C_P

The ratio of static compression test results of an individual fender to the standard performance (catalogue value) is defined as the coefficient of production tolerance, C_p, and it can be defined as in equations (4.2.3) and (4.2.4).

Production tolerance of reaction force : $C_{pR} = R \,/\, R_R$ (4.2.3)

Production tolerance of energy absorption : $C_{pE} = E \,/\, E_A$ (4.2.4)

Here,

C_{pR} : Production tolerance of reaction force

R : Reaction force during compression test under standard condition

R_R : Standard reaction force (catalogue value)

C_{pE} : Production tolerance of energy absorption

E : Energy absorption during compression test under standard condition

E_A : Standard energy absorption (catalogue value)

Rubber fenders are made primarily of natural rubber produced from sap and synthetic rubber made from petroleum; the rubber is mixed with chemicals, such as sulfur and carbon, and subsequently vulcanized by heat and pressure. Therefore, the fender performance is different under various conditions such as seasonal variations and differences in the manufacturing processes. Fig. 4.2.1 shows an example of the frequency distribution of the ratios of the test values to the catalogue values of the energy absorption (C_{pE}) of various rubber fenders pertaining to different manufacturers [4)]. According to Fig. 4.2.1, the production tolerance generally falls within ± 10%. Furthermore, since the change in shapes of performance due to the production tolerance is nearly negligible, the coefficients of reaction force and energy absorption may be equivalent ($C_p = C_{pR} = C_{pE}$).

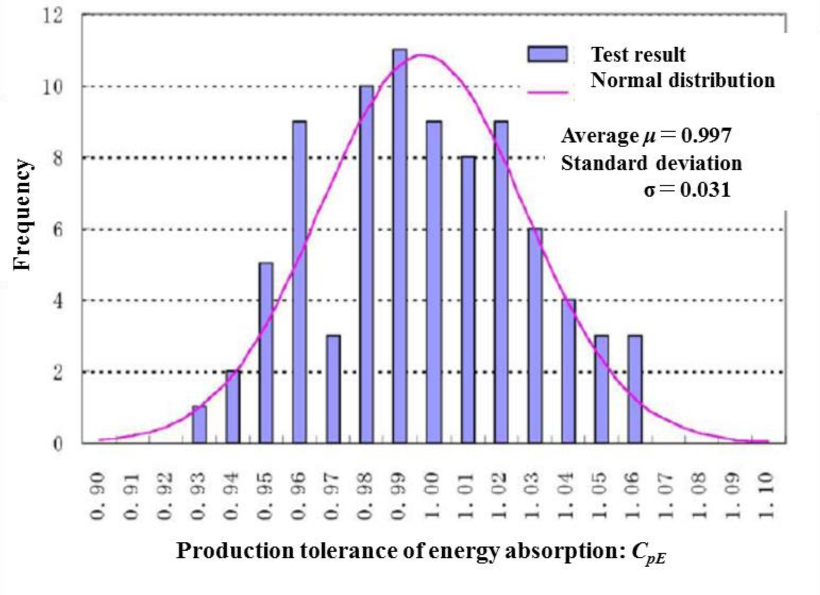


Fig. 4.2.1 Distribution of production tolerance [4)]

4.2.2 Coefficient of angular berthing: C_a

When a rubber fender is angularly compressed, the following two types of inclination angles can be considered. Note that the angular compression described in case (2) is not the same as angular berthing. The inclination angle in case (1) is the angle between the side of the vessel hull and the rubber fender in contact with the vessel, and the inclination angle in case (2) is the angle of the berthing velocity vector with respect to the centreline of the fender (which corresponds to the berthing direction). In general, the berthing angle is often referred to as the ship-side angle in case (1).

(1) Case when the hull of a berthing vessel is inclined to the berth line.

Fig. 4.2.2 shows the case when the vessel side is horizontally inclined and the compression direction is parallel to the fender axis. However, as shown in Fig. 4.2.3, this state also occurs when the flared portion of the hull contacts the fender under a vertical inclination.

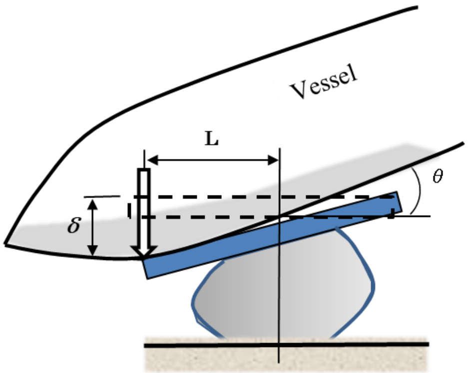


Fig. 4.2.2 Angular berthing (Horizontal)

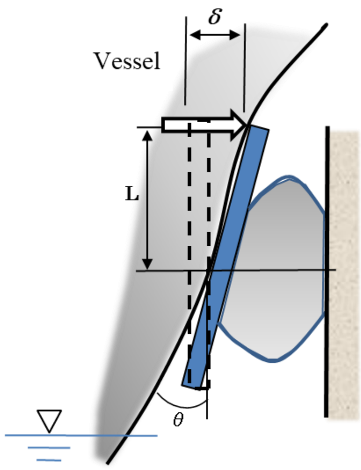


Fig. 4.2.3 Berthing with flare angle (Vertical)

The berthing angle θ represents the angle between the vessel side and fender that is in contact with the vessel. It is considered that the same phenomenon occurs when the hull exhibits a flare angle in the vertical direction, as shown in Fig. 4.2.3. Although the actual hull shape curves in three dimensions, it is assumed that the contact area of the hull is straight with a constant flare angle. The reaction force when a rubber fender is compressed at an angle θ° is slightly lower than that at θ = 0°, as shown in Fig. 4.2.4 for various values of θ.

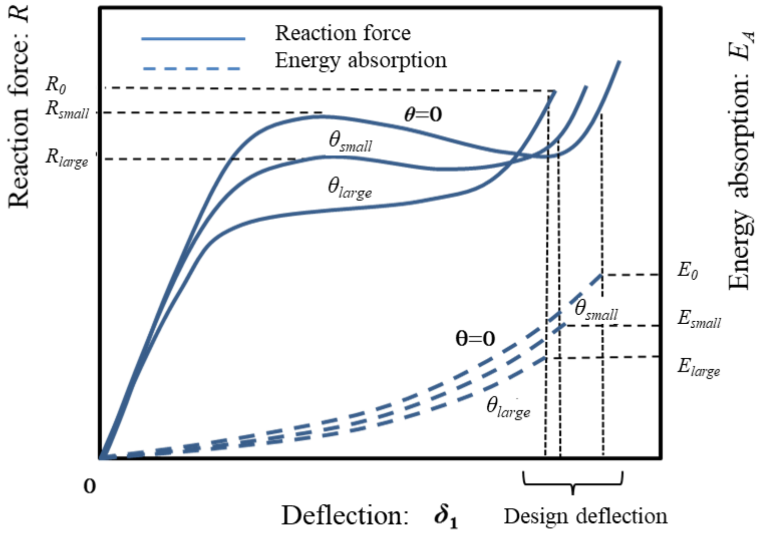


Fig. 4.2.4 Fender performance under angular berthing

The performance under angular compression is subject to the complex effects of the inclination angle and panel size, as described in the following section. If the rubber fender has a panel, as shown in Fig. 4.2.2 or Fig. 4.2.3, the vessel hull will hit the edge of the panel and rotate the panel. When the angle of panel rotation becomes θ, the panel comes into full contact with vessel hull, owing to which the panel and head of the rubber fender stop rotating, and the fender is compressed to the maximum deflection with angle θ. Fig. 4.2.5 and Fig. 4.2.6 show conceptual diagrams of the performance curve.

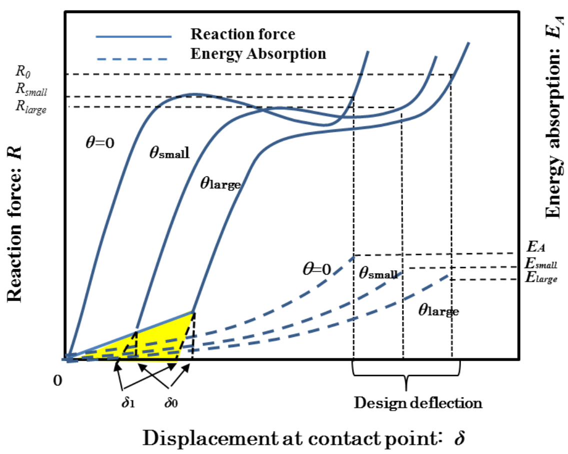


Fig. 4.2.5 Angular performance for various θ
(Constant contact point: L)

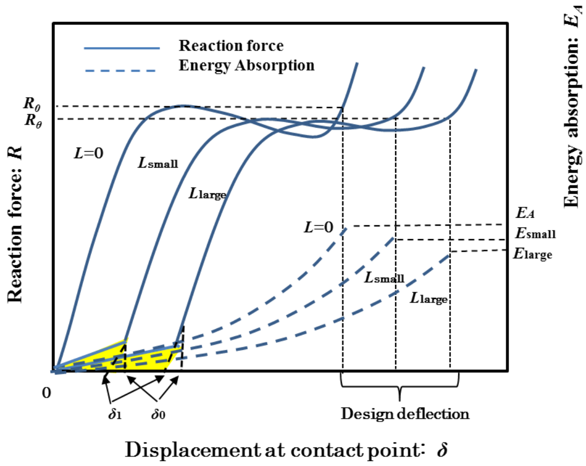


Fig. 4.2.6 Angular performance for various L
(Constant berthing angle: θ)

In Fig. 4.2.5, when the hull contacts the fender at an angle θ, the reaction force increases linearly due to the force that rotates the panel to cause a small displacement; further, the reaction force increases when the entire panel contacts the vessel hull. The yellow triangles in Fig. 4.2.5 and Fig. 4.2.6 are considered to correspond to the energy absorbed by the rotation of the panel. Thus, with increase in the angle θ, the area of the yellow triangle increases. Furthermore, as shown in Fig. 4.2.6, when the angle θ is constant and the size of the panel increases, the contact point moves farther (L_{large}), and the area of the yellow triangle increases. Because this energy absorption is small and depends on the angle and size of the panel, the angular performance defined in the manufactures' catalogues may not consider this energy. It is assumed that the displacement at which the reaction force increases is δ_0 because the rotation angle of the panel and the inclination angle θ on the ship side become equal. This value also differs depending on the angle and size of the panel. The performance curve shown in Fig. 4.2.4 assumes the displacement δ_1 that intersects the horizontal axis of the figure as the origin, disregarding the portion due to the rotation of the panel. However, since the angle does not change in Fig. 4.2.6, the reaction force curve of the angular compression has the same shape if the virtual displacement δ_1 is the origin and the design reaction force R_θ is constant, regardless of the berth position L.

The allowable displacement in angular compression is determined by the manufacturer, taking into account the deformation of the product. For example, at high angles, the outer surface of the fender may be over-compressed before reaching the design-allowable deflection. In such a case, the allowable deflection specified in the catalogue may be smaller than the design deflection determined considering the angular performance.

The performance of angular compression is often presented in manufacturers' catalogues or technical documents. In the case of small rubber fenders such as V-type fenders without panels, it may not be necessary to consider the angular coefficients, as summarized in Table 5.4.4 in Chapter 5 5.4. The performance of V-type rubber fenders without panels also varies depending on the direction of inclination. If the inclination is in the width direction of the fender, the performance must be in accordance with that defined in the manufacturers' catalogues or technical documents. The angular performance in the longitudinal direction can be approximated by assuming that only the contact portion generates a reaction force. The details regarding this aspect are presented in equation 5.6.7, in Section 5.6.2 in Chapter 5.

The influence of the angular compression on the performance is defined as the angular coefficient C_a. In design, the angular coefficient C_a at the angle θ of the design condition is employed with the catalogue value of the standard condition to obtain the design performance.

Angular coefficient of reaction force: $$C_{aR} = R_\theta / R_R \quad (4.2.5)$$

Angular coefficient of energy absorption: $$C_{aE} = E_{A\theta} / E_A \quad (4.2.6)$$

Here,

C_{aR} : Angular coefficient of reaction force

R_θ : Reaction force at angle θ

R_R : Standard reaction force at angle 0°

C_{aE} : Angular coefficient of energy absorption

$E_{A\theta}$: Energy absorption at angle θ

E_A : Standard energy absorption at angle 0°

The fender can be selected by obtaining the reaction force and the energy absorption using equations (4.2.5) and (4.2.6), respectively. The reaction force performances at various angles do not exhibit similar shapes, as shown in Fig. 4.2.4; thus, their values cannot be obtained by simply applying a constant C_a to the reaction force value for each deflection of the standard performance. To obtain the angular reaction force for each deflection, an angular coefficient for each deflection is required, as shown in equation 4.2.7.

Angular coefficient of reaction force at deflection ε (%) : $$C_{aR}(\varepsilon) = R_\theta(\varepsilon) / R(\varepsilon) \quad (4.2.7)$$

Here,

$C_{aR}(\varepsilon)$: Angular coefficient of reaction force at deflection ε (%)

$R_\theta(\varepsilon)$: Reaction force at deflection ε (%) and angle θ

$R(\varepsilon)$: Standard reaction force at deflection ε (%) and angle 0°

The reaction force corresponding to the deflection is necessary, for example, when considering the performance of rubber fenders in terms of horizontal force characteristics such as in single pile dolphins and flexible piers, or when using fender as a mooring spring for calculating the motion of a moored vessel. To obtain the coefficient $C_{aE(\varepsilon)}$ of energy absorption by deflection, it is possible to calculate the reaction force by deflection by using the angular coefficient of equation (4.2.7) in the process of integrating the reaction force from 0% to deflection ε. The integrated value up to the design deflection is the coefficient of energy absorption C_{aE}.

The procedure of determining the angular coefficient to be used for design from the results of the angular test is explained in Section 6.3.2 in Chapter 6.

(2) Case when the direction of berthing vessel is not parallel to the compression axis of the fender

Fig. 4.2.7 shows the case in which a vessel is moving forward and the berth direction is at an angle to the compression axis of the rubber fender. In this case, the compression angle φ is the angle between the berth direction of the vessel and compression axis of the fender. At this time, friction is generated between the surface of the fender panel and hull of the vessel, and the balance with the shear reaction force of the rubber fender causes shaking due to the phenomena of following and sliding. In normal berthing, vessels are supposed to stop moving forward before they compress the fenders; therefore, angular berthing in the design of rubber fenders corresponds to the case shown in Fig. 4.2.2, and the shear compression is not considered unless certain specific conditions exist. Under special circumstances, for example, when there is no assistance from tugboats and the forward speed cannot be eliminated, or compression with sharp angles is required such as in the case of bridge protection, canals, locks, etc., the manufacturer's technical data or individually scaled model testing may be necessary. When the shear displacement is large, in addition to the frictional force, excessive shear deformation can be suppressed by the tension of the shear chain, as shown in Fig. 4.2.7. Similar to that of the angular compression, the influence of the berthing direction on the performance is defined as the angular coefficient C, which can be defined as in equation (4.2.8) and equation (4.2.9).

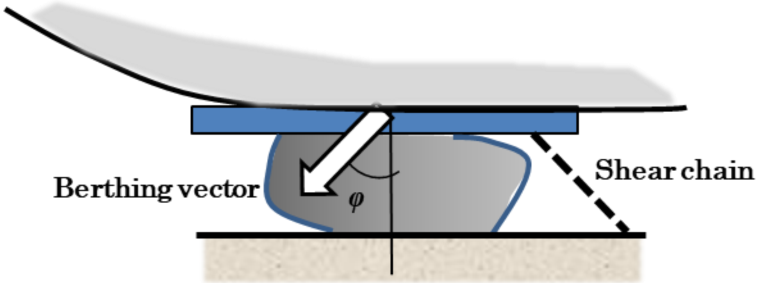


Fig. 4.2.7 Berthing along with angle of direction

Angular coefficient of reaction force : $C_{\varphi R} = R_{\varphi} / R_R$ (4.2.8)

Angular coefficient of energy absorption : $C_{\varphi E} = E_{A\varphi} / E_A$ (4.2.9)

Here,

$C_{\varphi R}$: Angular coefficient of reaction force

R_{φ} : Reaction force at direction angle φ

R_R : Standard reaction force at direction angle of 0°

$C_{\varphi E}$: Angular coefficient of energy absorption

$E_{A\varphi}$: Energy absorption at direction angle φ

E_A : Standard energy absorption at direction angle of 0°

4.2.3 Velocity factor: *VF*

(1) Constant Velocity (CV) and Decreasing Velocity (DV)

Since rubber has visco-elastic properties, the reaction force of a rubber fender changes depending on the compression speed. Generally, a higher deformation speed corresponds to a higher reaction force. However, in actual berthing, compression initiates at the berthing velocity and decelerates as the berthing energy is absorbed, finally coming to a stop. In the PIANC Guidelines [1)], the initial berthing velocity is set as 0.15 m/s, and decelerating compression that nearly stops at the maximum deflection is specified. This performance is termed as the rated performance. Since it is difficult to reproduce such conditions when using a real-sized fender, an alternative method is proposed. First, a scale model is compressed at a constant speed under various strain rates, and the velocity factor for each constant speed is determined; subsequently, the performance for the decelerating compression is estimated via calculations to determine the rated performance. Since the purpose of both the methods is to obtain the rated performance, the method for determining the deceleration (DV) performance from constant speed compression (CV) is introduced.

(2) Constant Velocity (CV)

Fig. 4.2.8 shows the concept of reaction force performance under constant speed compression (CV) when the strain rate is changed. With increase in the strain rate, the reaction force increases. The velocity factor for constant velocity (CV) can be defined as in equation (4.2.10) and equation (4.2.11).

Velocity factor of reaction force: $VF_R = R_V / R_R$ (4.2.10)

Velocity factor of energy absorption: $VF_E = E_{AV} / E_A$ (4.2.11)

Here,

VF_R : Velocity factor of reaction force

R_V : Design reaction force under constant velocity (CV)

R_R : Standard reaction force under standard velocity

VF_E : Velocity factor of energy absorption

E_{AV} : Design energy absorption under constant velocity (CV)

E_A : Standard energy absorption under standard velocity

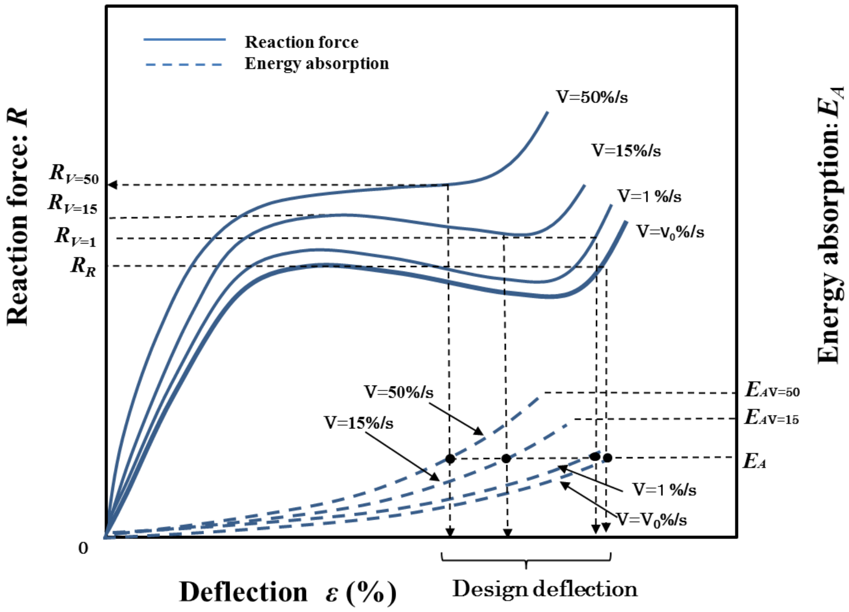


Fig. 4.2.8 Performance under different compression velocities (CV)

In Fig. 4.2.8, at standard performance (V=V_0), the peak reaction force at approximately 25% and the reaction force at the design deflection have the same value, and the design reaction force R_V in this case becomes R_R. At V = 1%/s, since the reaction force at the design deflection is larger than the peak reaction force, the reaction force at the design deflection becomes the design reaction force R_V. Conversely, at V = 15%/s, since the reaction force at the design deflection is smaller than the peak reaction force, the peak reaction force becomes the design reaction force R_V. When the compression speed is extremely large such as V = 50%/s, the reaction force may monotonously increase without exhibiting a local peak. In such a case, the reaction force at a deflection that involves the standard energy absorption is set as the design reaction force R_V. The PIANC Guidelines [1)] prescribe that the reaction force at 35% deflection is used, but depending on the type of fender, the design reaction force may be underestimated, which is a risk condition. The relationship between the strain rate V and the velocity coefficient VF_R of reaction force determined using equation (4.2.10) is shown in Fig. 4.2.9.

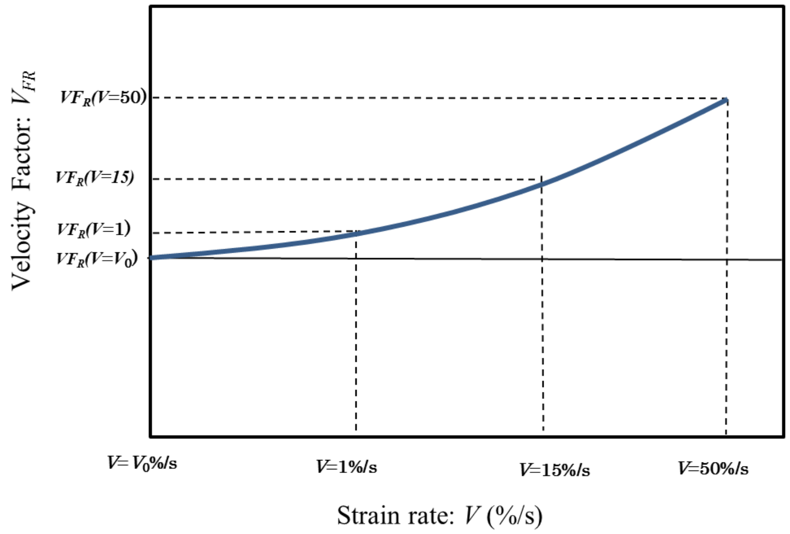


Fig. 4.2.9 Velocity factor and strain rate: Constant velocity (CV)

(3) Decreasing Velocity (DV)

Since the compression speed decreases in actual berthing, the compression type that simulates it is termed deceleration compression (DV). In the PIANC Guidelines [1)], the standard initial velocity is 0.15 m/s, the method of deceleration is approximated via a straight line or cosine waveform, and the final performance at which the effective berthing energy is absorbed and the vessel stops is defined as the rated performance. The reaction force at this instant is termed as the rated reaction force, the deflection amount is the rated deflection, and the energy absorption is the rated energy absorption when the other factors correspond to the standard condition. The method of correcting the fender performance by employing the velocity factor of the berthing velocity for the design condition based on the rated performances has been specified. As explained in Section 6.7 in Chapter 6, the velocity factors of rubber fenders are similar in terms of the strain rate rather than the actual velocity. Therefore, if the horizontal axis in Fig. 4.2.9 is the actual velocity, the velocity factor must be different depending on the size of the rubber fender. However, in this document, the strain rate (V0=0.01 to 0.3%/s) is sufficiently low, as V_0 in Fig. 4.2.9 is defined as the standard velocity and the velocity factor is 1.0. The relationship between the strain rate and velocity factor can be defined in a size-independent manner, making it easy to use for design.

Two methods exist for determining the decelerating compression performance: one in which the decelerating compression is determined by changing the initial speed in a scale model or a small-sized actual fender, and another in which it is calculated using the velocity factor of constant speed compression, as shown in Fig. 4.2.9. Here, an example of the latter calculation method is demonstrated. Fig. 4.2.10 shows an explanatory view of the principle of determining the performance of decelerating compression from the velocity factor of constant speed compression. Consider a rubber fender (1000 H) having a height of 1 m, and assume an initial velocity (berthing velocity) of 0.15 m/s, which corresponds to a strain rate of 15%/s. When the velocity decelerates from 15%/s → 12%/s → 5%/s → 1%/s → 0%/s, as shown in the upper figure in Fig. 4.2.10, the reaction force curve at the speed after deceleration traces the progress of the compression, as indicated by the red line in the lower figure of Fig. 4.2.10, and subsequently, the deceleration performance can be determined.

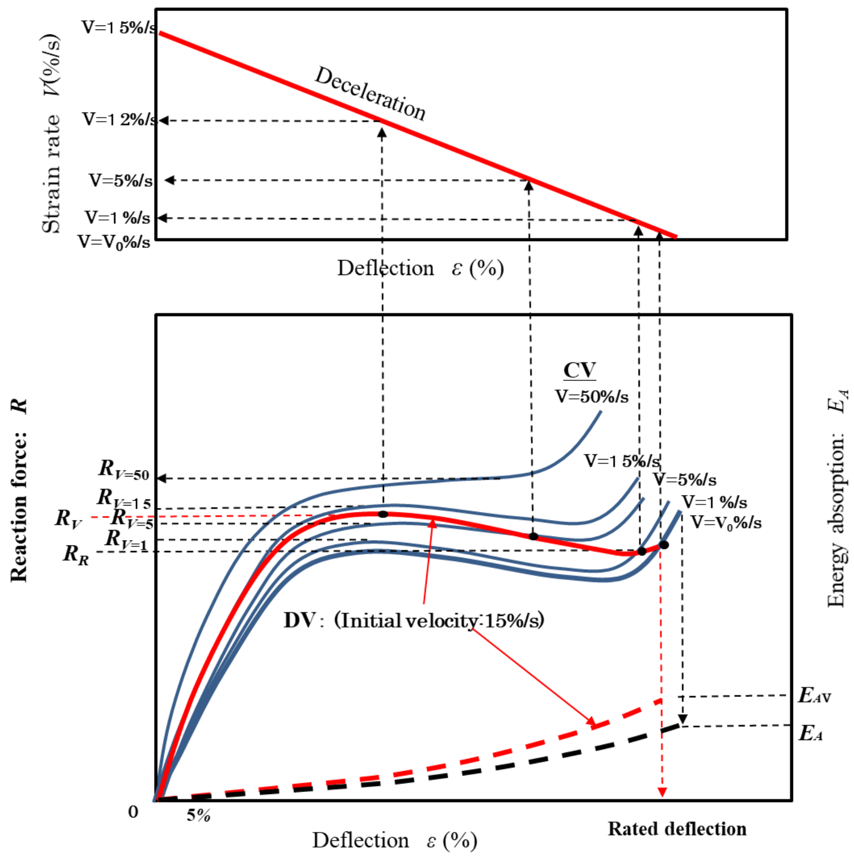


Fig. 4.2.10 Procedure to estimate DV performance from CV performances

If the shape of the reaction force curve changes at a remarkably high compression speed, as in V = 50%/s in Fig. 4.2.10, it is necessary to use a strain-specific velocity factor. However, in the considered example, the buckling-type performance is exhibited at V = 15%/s or less, and thus, the velocity coefficient of Fig. 4.2.9 may be used assuming that the shapes of the curves are similar. The energy absorption can be determined by integrating the reaction force curves with respect to the displacement (deflection); however, if the curves are similar in shape, the velocity factor of energy absorption is equal to the velocity factor of the reaction force.

Since there are several patterns of deceleration, such as linear deceleration, deceleration with a cosine waveform, or deceleration by absorbing effective berthing energy, it is necessary to specify which deceleration type is employed. The test results of deceleration compression (DV) at an initial compression speed of 0.15 m/s and the deceleration performance determine from the CV performance under different velocities are compared in Fig. 6.3.8 in Section 6.3.3 in Chapter 6,.

The comparison of the DV (deceleration) performance conforming to the PIANC Guidelines [1)] under different initial speeds is shown in Fig. 4.2.11. As shown in Fig. 4.2.11, the rated performance is obtained at an initial velocity of 0.15 m/s, while the conditions other than those for speed are standard conditions. In an actual compression test, if the compression speed is zero, the reaction force becomes unstable due to stress relaxation; thus, the PIANC Guidelines [1)] define the final speed as 0.05 m/s or less.

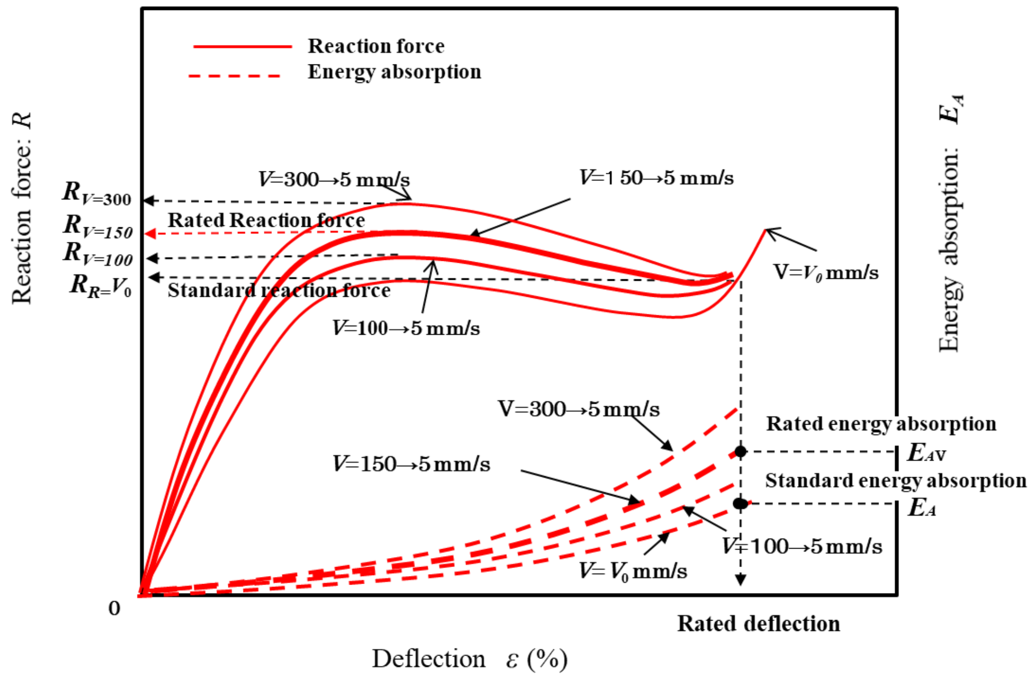


Fig.4.2.11 Fender performances under decreasing strain rates: DV

Fig. 4.2.12 shows the velocity factor of decelerating compression determined using the strain rate as described above and compared with CV in Fig. 4.2.9. This value is smaller than the constant speed performance under a constant initial speed because at the deflection for which the reaction force is the maximum, certain deceleration has already occurred. Because this deceleration performance is used directly in the selection of fenders, it is necessary to mention at least this aspect of deceleration performance in the catalogue. If the difference between the reaction force and energy absorption is small (within 3%), the same numerical value can be used for the velocity factor. The range is also shown in Figure 4.2.12.

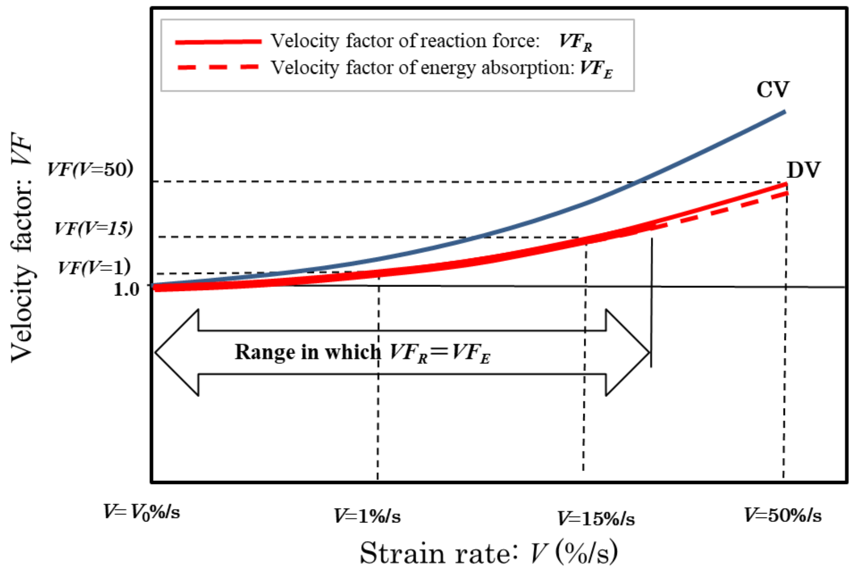


Fig. 4.2.12 Velocity factor and strain rate: decreasing velocity (DV) and constant velocity (CV)

(1) Comparison with PIANC Guidelines [1)]

Table 4.2.1 presents a comparison of the proposed method and that specified by the PIANC Guidelines [1)] regarding the concept of velocity factor. The main difference is that the values used for the denominators of equation (4.2.10) and (4.2.11) in the proposed method are different from those used in the PIANC Guidelines [1]. Even if the coefficients change, the implication of the final result is the same, and thus, the two methods can be considered to be consistent.

Table 4.2.1 Comparison of the velocity factor obtained using the proposed method and that defined by the PIANC Guidelines.

	Method defined in this book	Method in PIANC guideline[1]
Dominant performance for velocity factors	Standard performance: CV performance under constant strain rate 0.01%/s to 0.3%/s	Rated performance: DV performance under initial speed 0.15m/s
Test and specimen for velocity factor	CV(Constant speed) or DV (Decreasng speed) Scale model or real size product	DV (Decreasing speed) test Scale model or real size product
Method to obtain DV performance	Calculate DV performance using CV test results. (Or direct DV test)	Direct DV test
Velocity factor formula	$= \dfrac{\text{Calculated or tested DV performance}}{\text{CV performance under constant strain rate 0.01\%/s to 0.3\%/s}}$	$= \dfrac{\text{DV performance under actual berthing speed}}{\text{DV performance under initial speed 0.15m/s}}$
Design reaction force for fenders without buckling	Reaction force at deflection at which standard energy is absorbed.	Reaction force at 35% deflection
Pros	・Standard performance can be demonstrated by product tests. ・Velocity (Strain rate)does not depend on fender size ・Energy absorption value is slightly lower.	・Can use DV test performance directly for design. ・Energy absorption value will be slightly higher.
Cons	・Calculation required to obtain DV from CV.	・Large test facility is required. ・Several cases of velocity must be considered for each size.

Since a velocity factor for each deflection is required to obtain a detailed reaction force curve, *VF* is considered as a function of deflection ε, as shown in equation (4.2.12). When the energy absorption factor VF_E (ε) for each deflection is required, the reaction force for each deflection obtained using the velocity factor of equation (4.2.12) is integrated from 0% to the design deflection. Details regarding this aspect are presented in Chapter 6, Section 6.3.3.

Velocity factor of the reaction force at deflection ε %: $$VF_R(\varepsilon) = R_V(\varepsilon) \,/\, R(\varepsilon) \qquad (4.2.12)$$

Here,

$VF_R(\varepsilon)$: Velocity factor of reaction force at deflection ε

$R_V(\varepsilon)$: Reaction force at deflection ε with velocity *V*

$R(\varepsilon)$: Standard reaction force at deflection ε

In the selection of ordinary rubber fenders, if a velocity factor for decelerating compression exists, the factors for constant velocity and strain-specific velocity factor are not required; thus, the design can be realized even if the values are not listed in the catalogue. However, since the value is required in the calculation methods of the velocity factor, the motion analysis of moored vessels, and the special design of flexible structures such as single-pile dolphins, it is desirable that the values are presented as technical data.

When the fender height is small, such as in the case of scale model testing, the standard strain rate of 0.01%/s to 0.3%/s may be difficult to realize owing to the limitations of the testing facility. In such a case, a reaction force compressed at an actual speed of 0.01 to 1.3 mm/s may be used.

4.2.4 Temperature factor: *TF*

The performance of rubber fenders also varies with temperature. Normally, the reaction force of rubber fenders increases when the temperature decreases; thus, it is necessary to verify the minimum energy absorption at high temperatures and the maximum reaction force at low temperatures. A conceptual diagram is shown in Fig. 4.2.13. The temperature factor TF can be defined as the ratio of the standard reaction force R_R at the standard temperature of 23°C to a reaction force R_T at the temperature corresponding to design conditions. When determining the temperature factor specific to a deflection, $TF(\varepsilon)$ is defined as a function of deflection ε.

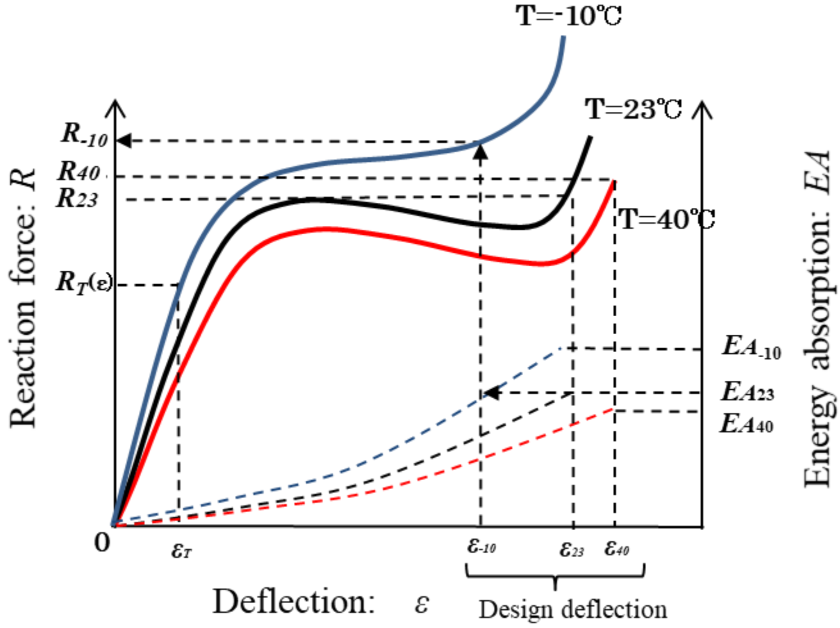


Fig. 4.2.13 Conceptual diagram of temperature dependency of fender performance

The temperature factor TF is defined as in equation (4.2.13) and equation (4.2.14). The conditions for the temperature test and specific calculation examples for the temperature factor are presented in Chapter 6, Section 6.3.4.

Temperature factor of reaction force: $TF_R = R_T / R_R$ (4.2.13)

Temperature factor of energy absorption: $TF_E = E_{AT} / E_A$ (4.2.14)

Here,

T : Temperature (°C)

ε_{-10} : Design deflection (%) at which the standard energy absorption occurs at low temperature of -10°C

ε_{23} : Standard deflection (%) at standard temperature of 23°C

ε_{40} : Design deflection (%) at high temperature of 40°C

ε_T : Arbitrary deflection temperature factor of energy absorption (%) at temperature T°C

TF_R : Temperature factor of reaction force

R_T : Reaction force at temperature T°C

R_{-10} : Design reaction force at temperature of -10°C (at deflection ε_{-10}, as shown in Fig. 4.2.13)

R_R : Standard reaction force (Standard conditions)

TF_E : Temperature factor of energy absorption

E_{AT} : Energy absorption at temperature T°C

E_{A-10} : Design energy absorption at temperature -10°C

E_A : Standard energy absorption (Standard conditions)

E_{A40} : Design energy absorption at temperature 40°C

An example of the temperature factor for the reaction force and energy absorption is shown in Fig. 4.2.14. The same temperature factor may be used if the difference in the values of the reaction force factor and energy absorption factor is within 3%. However, the range within 3% varies depending on the type of rubber fender and rubber grade, and thus, this aspect must be specified in the catalogue.

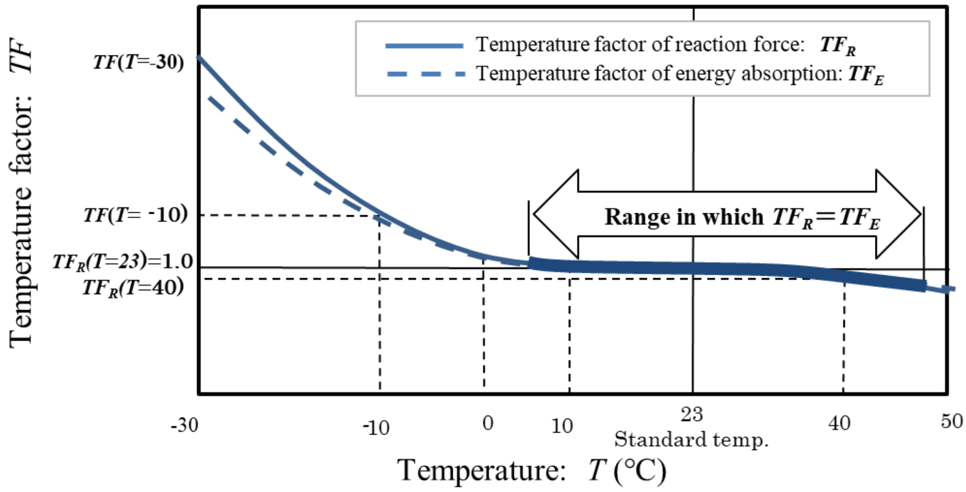


Fig. 4.2.14 Temperature factors of fender performance

The temperature factor for each deflection is required to determine the performance curve. In this case, $TF_R(\varepsilon)$ is considered as a function of deflection ε, as given in equation (4.2.15).

Temperature factor of the reaction force at deflection ε (%): $$TF_R\,(\varepsilon)= R_T\,(\varepsilon)\ /\ R(\varepsilon) \qquad (4.2.15)$$

Here,

$TF_{R\,(\varepsilon)}$: Temperature factor of reaction force at deflection ε (%)

$R_T(\varepsilon)$: Reaction force at temperature T and deflection ε

$R(\varepsilon)$: Standard reaction force at deflection ε at standard temperature

The energy absorption factor for each deflection TF_E (ε) can be calculated by integrating the reaction force for each deflection determined by the temperature factor of equation (4.2.15) from 0% to each deflection. The integrated value up to the design deflection is TF_E. When the temperature becomes extremely low, the reaction force curve continues to increase without exhibiting a peak due to buckling, as shown at T=−10°C in Fig. 4.2.13. In such a case, the maximum reaction force at which standard energy absorption at a standard temperature of 23°C occurs is considered the design reaction force at that temperature. The normal design does not require the use of a temperature factor per deflection; however, at extremely low temperatures, the fenders may not be able to demonstrate a buckling-type performance. Thus, it is desirable to use a deflection-specific temperature factor for evaluating the motion of moored vessels or designing flexible quays such as single-pile dolphins. Conversely, if the temperature is high (approximately 40°C), the standard energy may not be absorbed at the standard temperature. In this case, the size must be reselected in the catalogue.

To set the temperature as a design condition, the statistical records of the maximum (and minimum) daily average temperatures of the JMA (Japan Meteorological Agency), which are near to those at the use site may be considered if there is no record of actual measurements at the use site. However, it is desirable to use these values after correction via certain field observations. Rubber has a low heat transfer coefficient, and it takes approximately 1.2 days to stabilize the temperature at a depth of approximately 15 cm (refer to equation (6.3.1) in Chapter 6, Section 6.3.1). Simply applying the instantaneous maximum and minimum temperatures may result in an overestimation of the temperature dependence; therefore, it is practical to use the abovementioned daily average maximum and minimum temperatures. In addition, the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan -II Marine storage base facility [3] specifies a temperature factor of 0.95 to 1.25.

Furthermore, rubber materials are visco-elastic, and there exists a relationship between the speed dependence and temperature dependence, as defined by the WLF equation [5]. This equation can be used to verify whether the velocity factors and temperature factors are measured suitably (details are provided in Chapter 6).

4.2.5 Ageing factor: C_{ag}

Rubber fenders deteriorate due to the influence of heat, ultraviolet light, oxygen, etc. An oxidation reaction, in which oxygen molecules form radicals (free radicals) may occur, leading to cleaving of the molecules by extracting hydrogen from the molecular chains of rubber; this reaction softens the rubber, and the fender reaction force is reduced. In contrast, sulphur molecules bridge the linear molecules of rubber, although the unbonded sulphur bonds slowly even at room temperature; this reaction cures the rubber, and the fender reaction force increases. Therefore, the ageing reaction of rubber coexists with the softening and hardening reactions. In a survey of the materials used to manufacture rubber fenders, it was noted that the rate of hardening reaction was slightly higher, and the fender reaction force gradually increased with the service life [6]. As shown in equation (4.2.16), the ratio of the standard reaction force at the beginning of service life to that after ageing is defined as the ageing factor C_{agR}, and it is applied to the performance value as an influence factor. Ageing is not considered in the PIANC Guidelines [1], and C_{ag} is considered to be 1.0 in general fender design. However, since the reaction force of rubber fenders tends to increase with the number of service years, it is defined as an influence factor so that it can be evaluated when necessary to consider long-term use and deterioration of a quay.

Ageing factor of reaction force： $C_{agR}=R_{ag}\ ／R_R$ (4.2.16)

The energy absorption is considered to increase with increase in the reaction force; therefore, it is not necessary to consider this aspect in design; however, since there exist cases in which the reaction force decreases with service life, the energy absorption can be considered as defined in equation (4.2.17).

Ageing factor of energy absorption： $C_{agE}\ =E_{Aag}／E_A$ (4.2.17)

Here,

R_{ag} ： Reaction force after ageing
R_R ： Standard reaction force before service
E_{Aag}： Energy absorption after ageing
E_A ： Standard energy absorption before service

The factor of ageing is not usually taken into consideration in the design of fenders for general berths; however, it is necessary to focus on the increase in the reaction force in the case of high temperature environments or when the application involves places or structures in which replacement is difficult. The reaction force of thick-walled rubber fenders gradually increases with age; however, it has been noted that the deterioration of rubber is concentrated near the surface[7]. C_{ag} differs depending on the product, material and environment, and it must be estimated according to the expected service life. In the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan -II Marine storage base facility [3], the increase in reaction force with C_{agR} = 1.0 to 1.05 is considered. The actual ageing coefficient can be obtained using equation (4.2.16) by performing the compression test of an old fender collected from the site of use; however, some studies have reported on estimation performed using scale models by employing the Arrhenius method, among other methods [6, 8].

Fig. 4.2.15 shows an example of the ageing factor and service years of used vertical cylinder (CELL) rubber fenders returned from harbours. The figure also shows an example of the ageing factor of scale models after they underwent heat-accelerated ageing, as determined using the Arrhenius method. However, the Arrhenius method of ageing prediction is likely to cause a large error when considering the activation energy; thus, its validity must be examined by evaluating the products returned from actual sites [8]. The scale model demonstrates good agreement with the old fenders, as shown in Fig. 4.2.15. However, sufficient information regarding the secular change of rubber fenders may not be available, and it is necessary to enhance the investigation of different types and rubber grades.

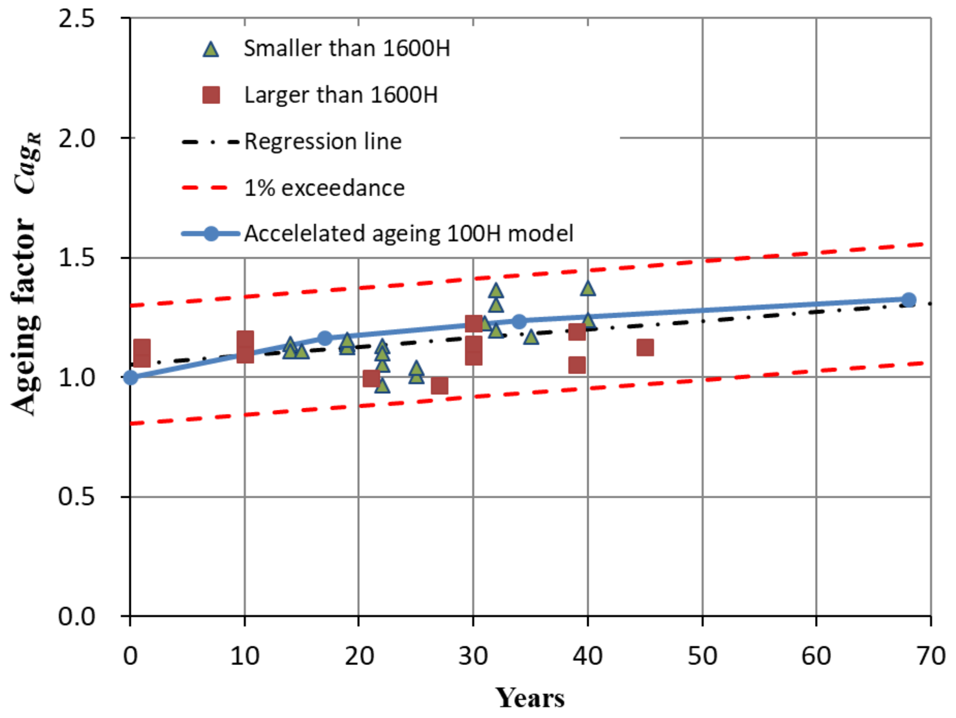


Fig. 4.2.15 Ageing factor of rubber fender and scale model (CELL type)
(Catalogue data used for year 0)

The ageing factor of rubber fenders usually does not take into consideration the reaction force of the first pre-compression; however, if the fender is not compressed for a long period, such as when it is in-stock, re-hardening occurs and ageing is affected. Subsequently, it was found that the fender may be cured to achieve a performance better than the first compression performance after manufacturing [7), 8)]. After this reaction force recovery, if the fender is compressed even once, the reaction force becomes stable and only the influence of ageing remains. However, caution must be exercised because the increase in reaction force affects the strength of the structure, even if it occurs only once. Re-hardening of the rubber material was the focus of a report by Mullins et al. [10)]. Rubber is a visco-elastic material; thus, when rubber is unloaded after being subjected to deformation, the complete recovery of the residual strain does not occur, although the strain remains when the rubber is deformed again, and the resilience of rubber decreases. It is believed that co-existing phenomena occur in which recovery is delayed due to viscosity and molecular breakage occurs due to deformation, which cannot be reversed. The reaction force returns to the original performance with time; however, the recovery due to viscosity is relatively fast and occurs almost completely in a few hours to approximately a day after unloading; the material in this case does not cure to exhibit a performance better than the original one. Furthermore, the recombination of broken molecules takes time, and curing gradually continues on a yearly basis; in some cases, as with ageing, curing may result in the rubber exhibiting more than the original reaction force. In this document, the recovery of the reaction force due to viscosity of the former is termed "recovery", and the recovery of the reaction force due to the recombination of broken molecules is termed "re-hardening".

Fig. 4.2.16 shows an example of the variation in the performance relative to the performance before unloading, with change in the standing time (un-loaded time). The performance was approximated using two types of regression methods before and after 1000 days. When rubber fenders with extremely low frequency of berthing or rubber fenders that have been kept in stock for a long duration are used, it is necessary to consider the re-hardening characteristics. The re-hardening characteristics depend on the shape and material of the rubber fender. Table 4.2.2 presents an example of the re-hardening correction factor C_{re}. For convenience, the coefficients are organized by stages. If the number of days left to stand is large, re-hardening can be taken into account using the ageing correction factor C_{ag}, by considering the re-hardening factor C_{re} presented in Table 4.2.2, as obtained from equation (4.2.18).

Ageing correction factor C_{ag} obtained using re-hardening factor C_{re}

$$C_{ag}\text{ (after correction)} = C_{ag}\text{ (before correction)} \times C_{re}\text{ (re-hardening factor)} \tag{4.2.18}$$

According to Table 4.2.2, the re-hardening factor is not required to be considered if berthing occurs one or more times in 10 days. It is not necessary to consider re-hardening in common mooring facilities, as it is estimated that the berthing interval is often less than a week in normal ports. However, if the frequency of use is extremely low, if the fender is not used for an extended period, or if rubber fenders that have been stocked for a long time are to be reused, the re-hardening factor should be considered in the design reaction force. Alternatively, countermeasures such as limiting the first berthing condition to a smaller value or reapplying a stress relaxation compression may be adopted.

In addition, such re-hardening may occur even in a working fender if the compression deflection due to berthing is always small. Although an example was presented in Table 4.2.2 and Fig. 4.2.16 owing to the importance of this factor, the chemical behaviour of material must be clarified, and the accuracy of the factor must be improved.

Table 4.2.2 Example for re-hardening factor C_{re}

Standing period	Re-hardening factor C_{re}
10 days or less	1.00
10–100 days	1.07
100–1000 days	1.09
1000–10000 days	1.25

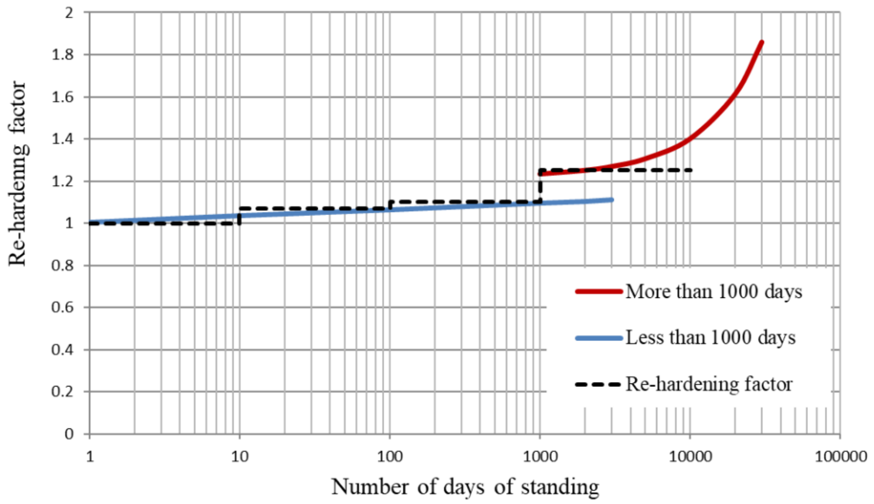


Fig. 4.2.16 Example of re-hardening factor: C_{re}

4.2.6 Repetition factor: C_r

Because a rubber fender is repeatedly compressed, its reaction force decreases due to fatigue. Extreme decrease in the reaction force against the motion of a moored ship or floating structure during stormy weather is a matter of concern. The ratio of the reaction force before repetition fatigue (standard performance) to that after repetition fatigue is defined as the repetition factor C_r. In the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan -II Marine storage base facility [3)], C_r with a value of 0.9 to 1.0 is considered as the repetition factor of the reaction force. In the design of a normal fender for berthing, it is considered that the vessel will be evacuated during a storm, and thus, continuous repetitive compression to the same rubber fender does not occur; in other words, the repetition factor could be C_r = 1.0 for both the reaction force and the absorbed energy.

As mentioned in Section 4.2, the standard repetition factor is C_r=1.0, which corresponds to the performance of the fender after being compressed three times or more and left for one hour or more. This aspect is defined by the quality control test conducted on rubber fenders after production, and it is different from that for repeated fatigue due to normal port operation.

In the same manner as for ageing, sufficient information pertaining to repetition fatigue is not yet available. Fig. 4.2.17 shows the results of a fatigue test conducted by Ueda et al. [11] as an example. As the number of repetitions increases to 100 and subsequently 1000, the reaction force decreases, as does the reduction rate, which appears saturated. If compression continues further, the fender will eventually break and lose its functionality. When a rubber fender is damaged due to fatigue, a sharp decrease in the reaction force occurs, as shown in Fig. 4.2.17; at this instant, another visual symptom is a crack that penetrates the main body, and the number of compressions at which either one is observed is defined as breakage.

According to the PIANC Guidelines [2] and Standard Specifications for Ports and Harbour Works [9], it is required that rubber fenders do not break even if they are subjected to 3000 cycles of continuous compression to the design deflection. This aspect corresponds to a certification test for confirming the durability, considering that rubber fenders are subjected to large compression strains of 50% to 70%. To date, the value of C_r is rarely used in design because the degradation of performance by the fatigue tests are not used in design.

However, to predict the life of rubber fenders, repeated fatigue and ageing, as described in the previous section, are important factors. Akiyama [8] conducted fatigue tests pertaining to the heat-accelerated degradation of scaled models based on Arrhenius's law and reported that the fatigue failure was accelerated remarkably by degradation due to ageing. In the future, it is expected that rational life prediction in design may be realized by additional research.

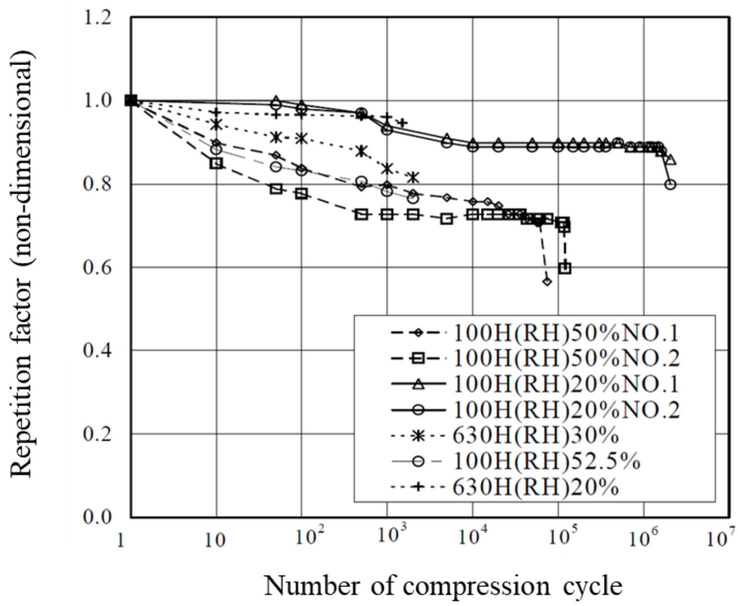


Fig. 4.2.17 Repetition fatigue test of vertical cylindrical fender (CELL-type)[10]
(Initial value: Average of 2nd and 3rd compression)

4.2.7 Creep characteristic

Deflection advances with time if a constant compression load is applied to the rubber fenders; this phenomenon is called creep. The reaction force decreases when a rubber fender buckles, and if the load is continued to be applied, there is a risk that the compression will lead to the designed deflection instantaneously. The results of the constant load test of a vertical cylindrical rubber fender (CELL-type) is shown in Fig. 4.2.18 as an example. Even when the initial deflection is continuously applied for a load of 8% or 10% for 10 h, the deflection saturates and approaches a constant value; however, at a load of 12%, the deflection gradually progresses. Since the deflection corresponding to the peak reaction force of this rubber fender is approximately 25%, 52.5% of the designed deflection is reached instantaneously when the load reaches its peak at 25%; such conditions should be avoided in the long-term mooring of vessels.

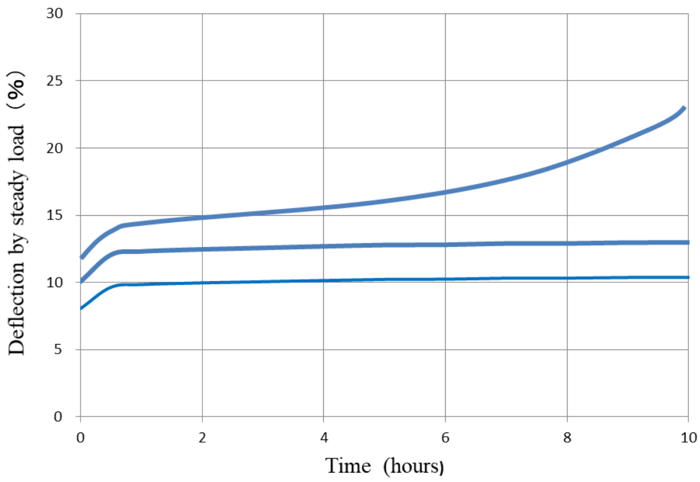


Fig. 4.2.18 Example of creep test

Such a phenomenon is not required to be considered during the design for berthing; however, if there is a possibility that the wind load during mooring may increase, it is necessary to ensure that the rubber fenders do not buckle unexpectedly. In the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan -II Marine storage base facility [3)], the wind pressure is recommended to be within the steady force equivalent to 10% of the deflection of the rubber fender, as shown in equation (4.2.19).

$$\text{Steady external force to rubber fenders} < \text{Reaction force at deflection } \varepsilon = 10\% \qquad (4.2.19)$$

It is also necessary to confirm whether 10% is appropriate as the initial strain for rubber fenders other than that corresponding to Figure 4.2.18. The creep property should be considered as an initial strain due to a steady external force, and it is not used as an influence factor leading to an enhancement or degradation in the performance. The creep is used as a safety standard for fenders that does not cause self-buckling because the displacement progresses temporally to the external force applied constantly. Appendix 4 shows an example of the verification of the wind load in the mooring analysis of a vessel under stormy conditions.

[References]

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Chapter 5
Design

5.1 General

When designing rubber fenders considering the various effects of use conditions and the environment, fenders that can absorb more energy than the effective berthing energy must be selected; furthermore, the maximum reaction force should be safe for mooring facilities. The surface pressure towards the vessel hull should be within the allowable hull pressure of the vessel. In addition, the fender, including the peripheral parts, must be designed to function as a safe system under the considered design conditions.

5.2 Design procedure of rubber fender

Since rubber fenders vary in size and performance in terms of those for leisure boats to those for super-tankers, it is necessary to design fenders that are appropriate for a given condition. In particular, the effective berthing energy is initially calculated, and the type and size of rubber fenders are selected accordingly. Next, the factors affecting the performance are considered. To select the appropriate factor pattern, the design conditions of the mooring facilities and the allowable hull pressure, as a representative of the vessel side strength, are compared with the corresponding performance of the rubber fenders. Rubber fenders are selected considering the influence factors, and a detailed design process that accounts for installation, pitch spacing, fender panel, chain, etc. is performed. Fig. 5.2.1 shows the design process flow of rubber fenders.

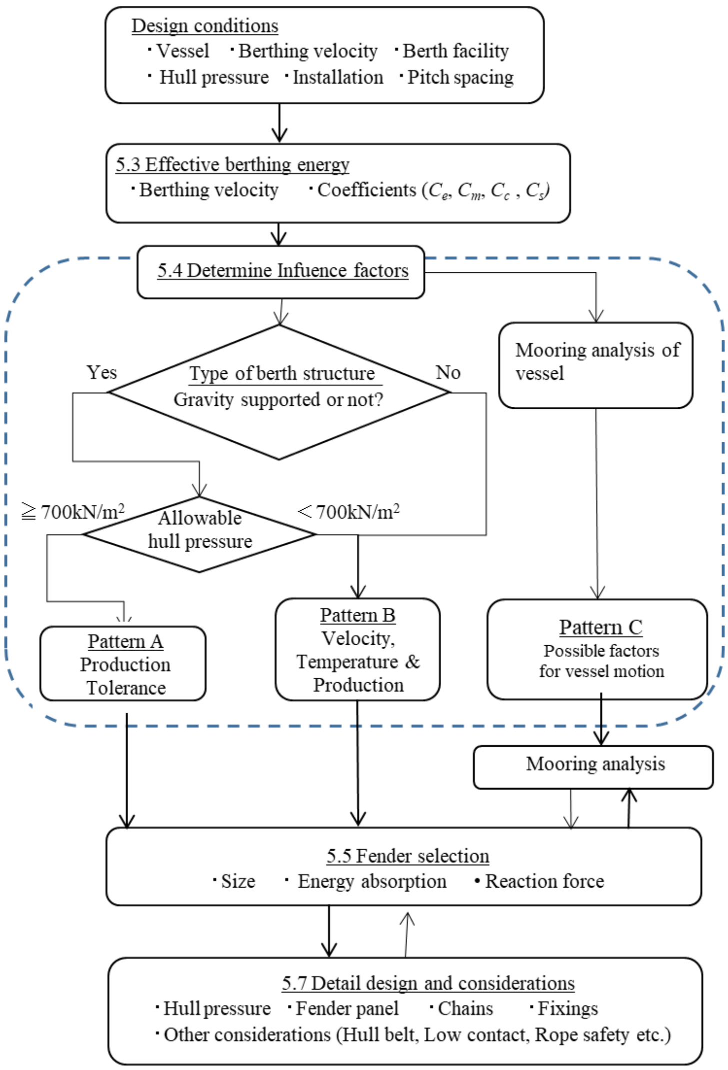


Fig. 5.2.1 Process flow for fender design

The classification of factor patterns is described in Chapter 5, Table 5.4.4, and design examples are presented in the Appendix.

5.3 Effective berthing energy

The effective berthing energy to be absorbed by the rubber fender can be calculated using equation (5.3.1).

$$\text{Effective berthing energy:}\quad E_b = \frac{1}{2}\cdot M \cdot V_B{}^2 \cdot C_e \cdot C_m \cdot C_c \cdot C_s \tag{5.3.1}$$

Here,

E_b : Effective berthing energy (kN·m=kJ)
M : Mass of vessel (＝Displacement tonnage: *DT, t*)
V_B : Berthing velocity (m/s)
C_e : Eccentricity factor
C_m : Virtual mass factor
C_c : Berth configuration factor
C_s : Softness factor

The key variables and coefficients in equation (5.3.1) can be described as follows:

(1) Berthing velocity: V_B

As given in equation (5.3.1), since the berthing velocity is squared when calculating the effective berthing energy, it exerts more influence than other coefficients do. The berthing velocity is affected by the type of ship, loading condition, location and structure of mooring facilities, weather and sea conditions, presence/absence of tug boats, etc. Vessels have evolved over time, and the measured data and standards of berthing velocity have been updated accordingly [1), 2), 3)]. In actual design, it is desirable to set these parameters appropriately based on local measurement data and the latest statistics along with the abovementioned information.

(2) Eccentricity factor: Ce

As shown in Fig. 5.3.1, in most berthing vessels, the hull contacts a fender at berthing angle θ and starts to rotate. As a result, a part of the kinetic energy (berthing energy) is consumed by the rotation. The remaining energy can be determined by performing correction using the eccentricity factor C_e. The eccentricity factor C_e can be expressed using the equation (5.3.2) when the berthing direction is perpendicular to the berth.

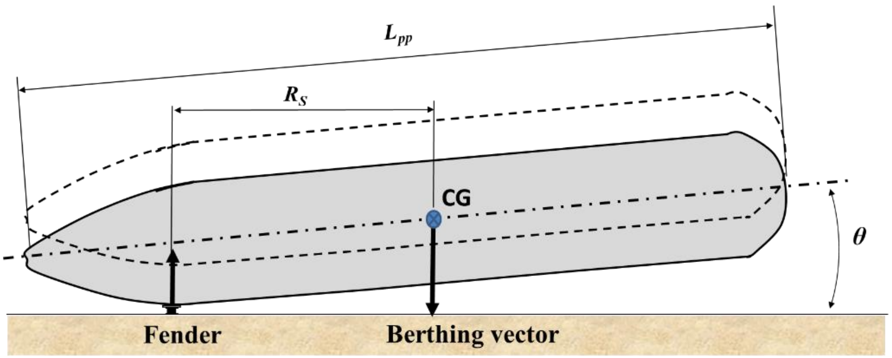


Fig. 5.3.1 Eccentrically berthing vessel

$$Ce = \frac{K_r{}^2}{K_r{}^2 + R_s{}^2} = \frac{1}{1 + \left(\frac{Rs}{K_r}\right)^2} \tag{5.3.2}$$

Here,

K_r : Gyration radius of vessel (m)

$$K_r = (0.19\ C_b + 0.11)\ L_{pp} \tag{5.3.3}$$

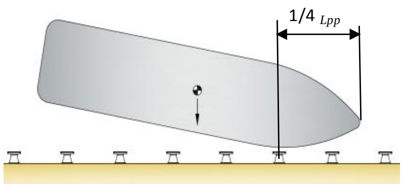
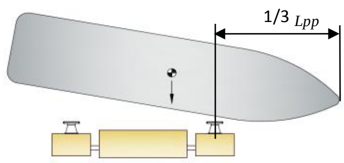
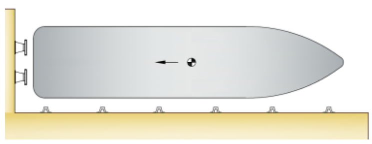
C_b : Block coefficient

$$C_b = \frac{DT}{L_{pp} \cdot B \cdot d \cdot \rho} \tag{5.3.4}$$

DT : Displacement tonnage (t)
L_{pp} : Length between perpendicular (m)
B : Beam of vessel (m)
d : Draught of vessel (m)
ρ : Density of water (Sea water: 1.025 t/m^3)
R_s : Distance between berthing point and vessel gravity centre parallel to berth line (m)

If accurate information is not available, or if only a preliminary review is conducted, the values in Table 5.3.1 may be used.

Table 5.3.1 Preliminary review of eccentric factor using berthing methods

Berthing method	Schematic view	C_e
1/4 point (Continuous berth)		**0.5**
1/3 point (Dolphin)		**0.7**
Stern (Roll on Roll off)		**1**

In Fig. 5.3.1, it is assumed that a rubber fender is present at the contact point of the vessel; however, in reality, rubber fenders are attached at a certain interval S; in addition, as shown in Fig. 5.3.2, there exists a case in which the vessel makes contact between two fenders.
In the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [2)], the eccentricity coefficient C_e at this instant is obtained by determining R_s, as defined in equation (5.3.2). As shown in Fig. 5.3.2, R_s is the distance from the contact point to the centre of gravity of the vessel, and it is measured parallel to the berth line. When the hull of the vessel approaches the berth and contacts two rubber fenders F_1 and F_2, R_s can be determined using equations (5.3.5) and (5.3.6). Here, R_s is R_1 when $k > 0.5$ and R_2 when $k < 0.5$; when $k = 0.5$, R_s is assigned the value of either R_1 and R_2 that corresponds to a larger value of C_e.

$$R_1 = \{0.5\alpha + e\ (1 - k)\}\ L_{pp}\ cos\theta \tag{5.3.5}$$

$$R_2 = \{0.5\alpha - e\,k\}\ L_{pp}\cos\theta \tag{5.3.6}$$

Here,

k : A parameter that represents the closest point of the vessel and berth between fenders F1 and F2. k is $0 < k < 1$; in general, k is approximately 0.5

R_1 : Distance (m) from the berthing point to the centre of gravity (CG) of vessel, measured parallel to the wharf as the vessel contacts fender F1

R_2 : Distance (m) from the berthing point to the centre of gravity (CG) of vessel, measured parallel to the wharf as the vessel contacts fender F2

θ : Berthing angle (°)

e : Ratio of fender spacing S measured in the longitudinal direction of the vessel to the length between perpendiculars L_{pp}

α : Ratio (parallel coefficient) of the length of parallel line at the berthing position to the length between perpendiculars L_{pp}

The eccentricity factor C_e is determined by substituting the R_s obtained using the abovementioned process into equation (5.3.2).

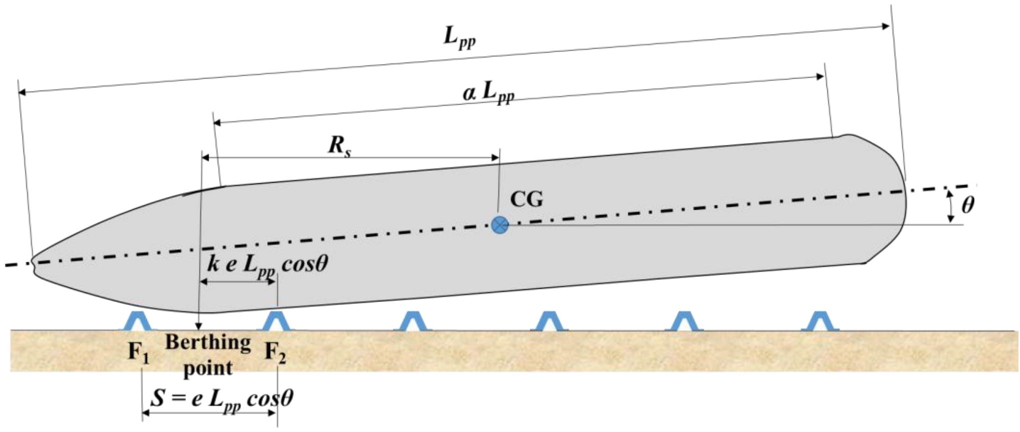


Fig. 5.3.2 Vessel berthing between fenders

(3) Virtual mass factor: C_m

Several formulas have been used to determine the virtual mass (added mass coefficient, hydrodynamic mass), and it has been researched extensively. The following two formulas are recommended in the PIANC Guidelines [4).]

1) Ueda's formula

Ueda's formula was proposed in 1981 and is based on model experiments and field observations. It can be presented as equation (5.3.7).

$$C_m = 1 + \frac{\pi}{2 \cdot C_b} \cdot \frac{d}{B} \quad \text{: Side berthing} \tag{5.3.7}$$

Here,

C_b : Block coefficient (equation (5.3.4))

2) Vasco Costa's formula

In this formula, it is assumed that a certain amount of water mass ($d \cdot d \cdot L_{pp}$) is added at the time of berthing. The total added mass is 2 $d \cdot d \cdot L_{pp}$ because the phenomenon occurs on both sides of vessel. In addition, the mass of the vessel is $L_{pp} \cdot B \cdot d$. Therefore, the virtual mass at the time of berthing can be obtained using equation (5.3.8) by adding the two defined masses.

$$Volume = L_{pp} \cdot B \cdot d + 2 \cdot d^2 \cdot L_{pp} = L_{pp} \cdot B \cdot d \times \left(1 + \frac{2 \cdot d}{B}\right) \quad (5.3.8)$$

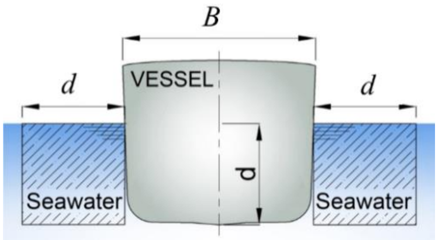


Fig. 5.3.3 Virtual mass of vessel (Vasco Costa)

Subsequently, the virtual mass factor (C_m) can be calculated using equation (5.3.9)

$$C_m = 1 + \frac{2 \cdot d}{B} \quad : \text{ Side berthing} \quad (5.3.9)$$

Here,

L_{pp} : Length between perpendiculars (m)
B : Beam of vessel (m)
d : Draught of vessel (m)

This formula is considered to be effective only under the following conditions; Ueda's equation should be used in all other cases (5.3.7).

- The bottom clearance of the vessel is 0.1×d or more.
- Berthing speed is 0.08 m/s or more.

3) Bow and stern berthing

For bow and stern berthing (roll on roll off), as described in the PIANC Guidelines [4)], a value of 1.1 should be adopted. The British Standard [5)] advocates a value of 1.0. Here, the larger value is selected.

$$C_m = 1.1: \quad \text{Bow and stern berthing} \quad (5.3.10)$$

(4) Berth configuration factor: C_c and softness factor: C_s

The berth configuration factor C_c is thought to be applicable in the case of permeable (such as a pile-supported pier) and non-permeable structures (a quay); when there is no escape for water, the water acts as a cushion and absorbs a certain amount of energy. The softness factor C_s considers energy absorption by elastic deformation of a vessel hull plate. Although both factors represent rational concepts, presently, the value of 1.0 is adopted because no research has yet demonstrated empirical evidence for the determination of numerical values.

5.4 Influence factors

5.4.1 Conditions from design background

The influence factors for rubber fenders must be determined in accordance with the use conditions and purpose. When making a decision, the following aspects must be considered from the viewpoint of strength of the mooring facility and vessel.

(1) Structure of mooring facility

A mooring facility with earth pressure behind a gravity type, sheet pile type or shelf type wall is stable against an external force such as a berthing force or wave force. In a basic design, the inertial forces at the time of an earthquake often dominate in the horizontal direction. Therefore, the reaction force of a fender is relatively small at the time of berthing, and the influence factors are not significant. In these construction types, the strength of the mooring facility is usually sufficient, and the influence factor of rubber fender performance does not need to be considered. However, in the case of piers supported by piles such as single-pile dolphins and jackets, the influence of horizontal force, such as the reaction force of fenders at the time of berthing, cannot be ignored.

(2) Allowable hull pressure of vessel

It is considered that small vessels, such as fishing boats, leisure boats, small barges, small cargo vessels, and floating piers, are less affected by the reaction of fenders against the strength on the hull side. However, with the increase in vessel size, such as in the case of container ships, large tankers, etc., the strength on the side of the vessel becomes relatively important. In such cases, the pressure (face pressure) exerted on the vessel side from the fender is limited. Therefore, when a design limit of allowable hull pressure exists, it is necessary to consider the influence factors of the fender reaction force. In the PIANC Guidelines [4)], the maximum value of the allowable hull pressure is 700 kN/m^2, and thus, this value is used for reference.

(3) Fender design by mooring analysis

In case a vessel is moored in a port facing the open ocean, and in the case of long-term mooring vessels, in which mooring is required even in stormy weather due to the presence of waves and wind, the maximum compression of rubber fenders by the collision force may exceed the design values under the berthing condition. To enable fender design considering such conditions, it is necessary to perform mooring simulations to analyse the motion of the vessels. For general cargo ships and tankers, Ueda and Shiraishi [6)] considered the simulation results of several movements observed according to vessel type, wave direction and wave period and proposed the ratio E/δ pertaining to the ratio of the energy absorption of rubber fenders to the compression deflection δ. E/δ can be used as a reference to evaluate conditions that require the consideration of motion during mooring in the fender design process.

In addition, in the case in which undulating waves enter from the open ocean, the amount of vessel motion increases, and it may not be possible to satisfy the required rate of cargo handling operations. In such cases, it may be necessary to investigate a mooring system consisting of rubber fenders and a mooring line from the viewpoint of securing the necessary cargo handling operations. Ueda et al. [7)] proposed allowable wave heights in various wave directions and wave periods for typical mooring conditions for various vessel types, and these values can be used as a reference when determining conditions that require this type of analysis.

5.4.2 Patterns of influence factors

For the design of rubber fenders, the influence factors are considered according to the conditions of mooring facilities and vessels. The treatment for the design of influence factors described in Chapter 4, Section 4.2 can be classified into the following three patterns.

(1) Pattern A

When the strength of the mooring facility has sufficient tolerance for horizontal force—as in the case of a gravity type wall, sheet pile type wall or shelf type quay—and the vessel has no restrictions pertaining to the hull pressure (or the hull pressure is more than 700 kN/m^2), the effects of natural environment and conditions of use are considered to be small, and the performance of rubber fenders involves the consideration of only the production tolerance. However, even in the case of gravity type quays, the allowable horizontal force may need to be considered in cases involving ageing or conditions in which the influence of temperature is large, such as in a cold region; under such circumstances, the influence of the actual conditions should be considered appropriately according to the evaluation of the designer.

(2) Pattern B

If the horizontal strength of a mooring facility is affected by fender performance, such as in the case of pile piers, dolphins and jackets, or if the vessel is restricted to a hull pressure of less than 700 kN/m^2, the influence of temperature must be considered as a natural condition, and the berthing angle and velocity must be considered as use conditions. However, in the case of a pile-supported jetty for fishing boats and small boats, if the berth force is sufficiently smaller than the horizontal load capacity of the mooring facility, coefficient pattern A may be adopted according to the judgment of the designer.

(3) Pattern C

When vessels are moored on a quay during stormy weather, when moored vessels are affected by swells or long-period waves, or when using rubber fenders for mooring of floating structures, rubber fender performance is different than under berthing conditions; furthermore, the influence of motion on the fender performance is complex. Therefore, it is necessary to carefully consider the influence factors in conditions involving, for example, repetitive fatigue; creep characteristics and hysteresis characteristics as well as other factors must also be considered extensively.

The concept of patterns for considering the influence of various factors is summarized in Table 5.4.4.

5.4.3 Example of managing the influence factors and their patterns

The influence factors for designing rubber fenders are set as follows according to catalogues and technical data. Note that the catalogue display examples are presented in Tables 5.4.1 to 5.4.3; however, these values should not be used in actual design because they are numerical values used only for explanation. Please refer to the manufacturers' catalogues and technical data for the actual design process.

(1) Production tolerance factor: C_p

The production tolerance factor has a maximum value of $C_{pR}{}^{+}$ = 1.1 and minimum value of $C_{p\,\mathrm{E}}{}^{-}$ = 0.9 unless otherwise specified, and these values can be used for cases of both the reaction force and energy absorption.

(2) Angular factor: C_a

The angular factor may be considered in a different manner depending on the manufacturer and type, as described in Chapter 4, Section 4.2.2. Generally, the values for this factor are displayed in the form of a table, as given in Table 5.4.1. At the angle corresponding to the design conditions, the largest reaction force $C_{aR}{}^{+}$ and the smallest energy absorption factor $C_{aE}{}^{-}$ are taken into account in the design. At a small angle, the reaction force may slightly increase, and some reaction force factors may exceed 1.00.

Table 5.4.1 Example of catalogue display of angular factor

Angle (°)	Design deflection (%)	Angular factor of reaction force $C_{a\mathrm{R}}$	Angular factor of energy absorption $C_{a\mathrm{E}}$
0	52.5	1.00	1.00
3	51.9	1.01	1.00
5	51.3	1.01	0.99
8	49.8	1.01	0.95
10	48.8	0.99	0.94
15	45.5	0.98	0.86
20	41.3	0.92	0.71

(3) Velocity factor: *VF*

The velocity factor used for design involves decelerating compression DV with the berthing velocity taken as the initial speed. This factor is represented in the form of a numerical table, such as in the example shown in Fig. 4.2.12 in Chapter 4, Section 4.2.3 or Table 5.4.2. At the strain rate obtained by dividing the design berthing velocity by the fender height, the largest factor of reaction force (V_{FR}^{+}) and smallest factor of energy absorption (V_{FE}^{-}) is considered in the design. In the case of an intermediate strain rate, a linear interpolation can be used. Since nearly equivalent factors (within 3%) may use common values, in this example, if the initial velocity is within 0.1%/s, the influence of the velocity can be neglected and assigned a value of 1.0. In some cases, it may be possible to simplify the process by using a common factor between rubber grades and between the reaction force and energy absorption.

Table 5.4.2 Example of catalogue display of velocity factor (DV)

<table>
<tr><td>Rubber grade</td><td colspan="2">Grade A</td><td colspan="2">Grade B</td><td colspan="2">Grade C</td></tr>
<tr><td>Strain rate (%/S)</td><td>Reaction force VF_R</td><td>Energy absorption VF_E</td><td>Reaction force VF_R</td><td>Energy absorption VF_E</td><td>Reaction force VF_R</td><td>Energy absorption VF_E</td></tr>
<tr><td>30</td><td>1.23</td><td>1.18</td><td>1.21</td><td>1.18</td><td>1.20</td><td>1.17</td></tr>
<tr><td>20</td><td>1.20</td><td>1.17</td><td>1.18</td><td>1.15</td><td>1.17</td><td>1.14</td></tr>
<tr><td>10</td><td>1.17</td><td>1.14</td><td colspan="2">1.15</td><td colspan="2">1.14</td></tr>
<tr><td>5</td><td colspan="2">1.13</td><td colspan="4">1.10</td></tr>
<tr><td>1.00</td><td colspan="2">1.06</td><td colspan="4">1.05</td></tr>
<tr><td>0.50</td><td colspan="6">1.04</td></tr>
<tr><td>0.10</td><td colspan="6" rowspan="3">1.00</td></tr>
<tr><td>0.05</td></tr>
<tr><td>0.01</td></tr>
</table>

(4) Temperature factor: *TF*

The temperature factor is represented by a numerical table, such as in the example shown in Fig. 4.2.14 in Section 4.2.4 in Chapter 4 or Table 5.4.3. In the design, TF_R^{+}, which is the maximum value in the temperature range of the design conditions, is considered for the factor of reaction force, and the factor TF_E^{-}, which is the minimum value at the highest temperature, is considered for energy absorption. Since the values are common in the same manner as the velocity factor, in this example, the temperature effect is considered negligible from 23°C to 30°C.

Table 5.4.3 Example of catalogue display of temperature factor

<table>
<tr><td>Rubber grade</td><td colspan="2">Grade A</td><td colspan="2">Grade B</td><td colspan="2">Grade C</td></tr>
<tr><td>Temperature (℃)</td><td>Reaction force VF_R</td><td>Energy absorption VF_E</td><td>Reaction force VF_R</td><td>Energy absorption VF_E</td><td>Reaction force VF_R</td><td>Energy absorption VF_E</td></tr>
<tr><td>-30</td><td colspan="2">2.28</td><td>1.75</td><td>1.84</td><td>1.69</td><td>1.78</td></tr>
<tr><td>-20</td><td colspan="2">1.77</td><td>1.49</td><td>1.52</td><td>1.44</td><td>1.47</td></tr>
<tr><td>-10</td><td colspan="2">1.39</td><td colspan="2">1.30</td><td colspan="2">1.26</td></tr>
<tr><td>0</td><td colspan="4">1.16</td><td colspan="2">1.13</td></tr>
<tr><td>10</td><td colspan="6">1.06</td></tr>
<tr><td>23</td><td colspan="6">1.00</td></tr>
<tr><td>30</td><td colspan="6">1.00</td></tr>
<tr><td>40</td><td colspan="2">0.94</td><td>0.99</td><td>0.95</td><td rowspan="2">1.00</td><td>0.96</td></tr>
<tr><td>50</td><td colspan="2">0.85</td><td>1.00</td><td>0.90</td><td>0.91</td></tr>
</table>

(5) Overall influence factor

The factors resulting from the consideration of performance influence are not strictly independent but related; for example, it is considered that the velocity factor could be affected by temperature and ageing. It is assumed that the overall influence factor may be predicted by performing superposition by multiplication, as shown in equations (5.4.1) and (5.4.2).

$$\text{Final reaction force} = \text{Standard reaction force (Catalogue)} \times C_{pR} \times C_{aR} \times VF_R \times TF_R \times C_r \times C_{agR} \quad (5.4.1)$$

$$\text{Final energy absorption} = \text{Standard energy absorption (Catalogue)} \times C_{pE} \times C_{aE} \times VF_E \times TF_E \times C_r \times C_{agE} \quad (5.4.2)$$

(6) Influence factor by factor pattern

The influence factors obtained in the manner described above are allocated to the factor patterns classified in Section 5.4.2. Specifically, the categorization is performed as presented in Table 5.4.4.

Table 5.4.4 Factors influencing performance of rubber fender

Influence factors \ Factor pattern		Pattern A Production tolerance	Pattern B Consideration of major factors	Pattern C Mooring analysis
Production tolerance C_p	Reaction force C_{pR}	Maximum C_{pR}^{+}=1.1	Maximum C_{pR}^{+}=1.1	Maximum C_{pR}^{+}=1.1
	Energy absorption C_{pE}	Minimum C_{pE}^{-}=0.9	Minimum C_{pE}^{-}=0.9	Minimum C_{pE}^{-}=0.9
Angular factor C_a	Reaction force C_{pR}	No consideration	Maximum C_{aR}^{+}	No consideration or Maximum $C_{aR(\varepsilon)}^{+}$ Minimum $C_{aR(\varepsilon)}^{-}$ Or by deflection *2
	Energy absorption C_{pE}		Minimum C_{aE}^{-}	
Velocity factor VF	Reaction force C_{pR}	No consideration	Maximum VF_R^{+}	No consideration or Maximum $VF_{R(\varepsilon)}^{+}$, Minimum $VF_{R(\varepsilon)}^{-}$ Or by deflection *2
	Energy absorption C_{pE}		Minimum VF_E^{-} *3	
Temperature factor TF	Reaction force C_{pR}	No consideration	Maximum TF_R^{+}	No consideration or Maximum $TF_{R(\varepsilon)}^{+}$, Minimum $TF_{R(\varepsilon)}^{-}$ Or by deflection *2
	Energy absorption C_{pE}		Minimum TF_E^{-}	
Ageing factor C_{ag}	Reaction force C_{pR}	No consideration	No consideration *4	No consideration or Maximum *4 C_{agR}^{+} or 1.0-1.05
	Energy absorption C_{pE}			
Repetition factor C_r	Reaction force C_{pR}	No consideration	No consideration	No consideration or 0.9-1.0
	Energy absorption C_{pE}			
Creep		No consideration	No consideration	Wind force < Reaction force at 10% *5

*1 Index + indicates the maximum value,- denotes the minimum value

*2 In the consideration of the influence of angle and velocity in the calculation for a mooring vessel with factor pattern C, the angle and speed during mooring fluctuate instantaneously, and it is difficult to perform an accurate simulation unless the influence factor is incorporated into the calculation program. Therefore, the maximum reaction force and amount of maximum motion may be evaluated by analysing two cases involving the maximum and minimum values of factors. (Refer to Section 5.5.3) Additionally, when the reaction force characteristics change in extremely low temperatures, the performance of each deflection must be considered.

*3 Under normal circumstances, the velocity factor VF_E^- of energy absorption can take the value of the deceleration velocity, which is not a standard velocity; in this case, $VF_E = VF_E^- = VF_E^+$.

*4 Although sufficient survey results [8)] pertaining to ageing have not been obtained, they are listed as items because they are expected to be addressed in the future.

*5 The vertical cylindrical rubber fenders (CELL) lead to 10% deflection. The deflections due to other rubber fenders must be individually determined by testing.

5.5 Determination of influence factors

Since the influence factors affect the performance of rubber fenders, the rubber fenders should be selected considering these factors. At the time of selection, the performance value is multiplied by the influence factors to reduce the standard energy absorption E_A in the catalogue to the minimum value E_A^- so that the effective berthing energy E_b satisfies the relationship given in equation (5.5.1).

$$E_A^- \geqq E_b \tag{5.5.1}$$

For the reaction force, the product of the standard reaction force R in the catalogue and the influence factor must be R^+ to satisfy both conditions specified in equation (5.5.2).

$$R^+ \begin{cases} \leqq \text{Design horizontal force to mooring facility} \\ \leqq \text{(Allowable hull pressure} \times \text{Effective area of fender panel)} \end{cases} \tag{5.5.2}$$

The suffix + indicates the maximum value, and −indicates the minimum value.

When it is certain that the ship will contact several rubber fenders, the total energy absorption of the fenders may be considered as E_A. However, it is necessary to determine the minimum value E_A^- of the sum of the respective energy absorptions and the maximum value R^+ of the sum of reaction forces to determine the curvature radius of the vessel hull at the berthing point and the spacing (installation pitch) between fenders.

5.5.1 Calculation of design energy absorption: E_A^-

The minimum value E_A^- of the design energy absorption is determined using equation (5.5.3).

$$E_A^- = E_A \times C_{pE}^- \times C_{aE} \times VF_E \times TF_E^- \times C_{agE}^- \tag{5.5.3}$$

where

E_A^- : Minimum design energy absorption (kN·m)
E_A : Standard energy absorption (catalogue value)
C_{pE}^- : Minimum factor of production tolerance (0.9)
C_{aE} : Angular factor of energy absorption
VF_E : Velocity factor of energy absorption (decreasing velocity from berthing velocity)
TF_E^- : Minimum temperature factor of energy absorption (highest temperature)
C_{agE}^- : Ageing factor of energy absorption

For angular berthing, geometric checks are also required; for instance, the end of the fender panel must not touch the quay during the absorption of berthing energy.

5.5.2 Calculation of design reaction force: R^+

The maximum design reaction force R^+ considering the influence factor of design condition can be obtained using equation (5.5.4).

$$R^+ = R_R \times C_{pR}{}^+ \times C_{aR}{}^+ \times VF_R \times TF_R{}^+ \times C_{agR}{}^+ \quad (5.5.4)$$

Here,

R^+	:	Maximum design reaction force (kN)
R_R	:	Standard reaction force (catalogue)
$C_{pR}{}^+$	:	Maximum factor of production tolerance (1.1)
$C_{aR}{}^+$	:	Angular factor of reaction force or 1.0 (larger value)
VF_R	:	Velocity factor of reaction force (decreasing velocity from berthing velocity)
$TF_R{}^+$	:	Maximum temperature factor of reaction force (lowest temperature)
$C_{agR}{}^+$	:	Ageing factor of reaction force

5.5.3 Calculation of design factors for mooring analysis

In the motion calculation of the moored vessel, it is necessary to express the performance of rubber fenders as accurately as possible. The compression (loading) performance R_{F1} and the return (unloading) performance R_{F2} are different, as shown in Fig. 5.5.1, resulting in energy loss. This loss is known as the hysteresis loss, and a considerable amount of this energy is converted to heat. Assuming that this hysteresis loss is expressed as shown in Fig. 5.5.1, let $R_{(\varepsilon)}$ be an approximate function of the reaction force by the deflection, which is expressed by a regression equation or numerical table. The maximum and minimum factors can be calculated using equations (5.5.5) to (5.5.8).

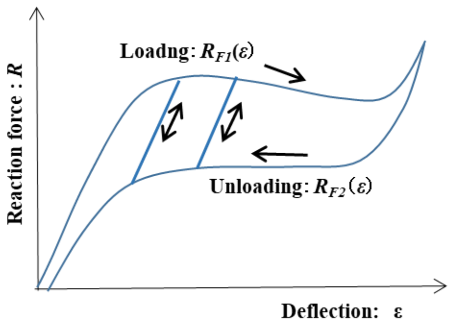


Fig. 5.5.1 Loading and unloading performance of rubber fender

Maximum performance

$$\text{Loading} \quad : R_{F1(\varepsilon)}{}^+ = R_{F1(\varepsilon)} \times C_{pR}{}^+ \times VF_{R(\varepsilon)} \times TF_{R(\varepsilon)}{}^+ \times C_{agR}{}^+ \quad (5.5.5)$$

$$\text{Unloading} : R_{F2(\varepsilon)}{}^+ = R_{F2(\varepsilon)} \times C_{pR}{}^+ \times VF_{R(\varepsilon)} \times TF_{R(\varepsilon)}{}^+ \times C_{agR}{}^+ \quad (5.5.6)$$

Minimum performance

$$\text{Loading} \quad : R_{F1(\varepsilon)}{}^- = R_{F1(\varepsilon)} \times C_{pR}{}^- \times C_{aR(\varepsilon)}{}^- \times TF_{R(\varepsilon)}{}^- \times C_r \times C_{agR}{}^- \quad (5.5.7)$$

$$\text{Unloading} : R_{F2(\varepsilon)}{}^- = R_{F2(\varepsilon)} \times C_{pR}{}^- \times C_{aR(\varepsilon)}{}^- \times TF_{R(\varepsilon)}{}^- \times C_r \times C_{agR}{}^- \quad (5.5.8)$$

Here,

$R_{F1(\varepsilon)}$, $R_{F2(\varepsilon)}$	: Standard reaction force for loading and unloading at deflection ε
$R_{F1(\varepsilon)}{}^+$, $R_{F1(\varepsilon)}{}^-$	: Maximum and minimum reaction force at deflection ε during loading
$R_{F2(\varepsilon)}{}^+$, $R_{F2(\varepsilon)}{}^-$	: Maximum and minimum reaction force at deflection ε during unloading
$C_{aR(\varepsilon)}{}^-$	: Minimum angular factor of reaction force at deflection ε
$C_{pR}{}^+$, $C_{pR}{}^-$	: Maximum and minimum production tolerance factors of reaction force (0.9, 1.1)
$VF_{R(\varepsilon)}$	: Velocity factor of reaction force at deflection ε
$TF_{R(\varepsilon)}{}^+$, $TF_{R(\varepsilon)}{}^-$	: Maximum and minimum temperature factors of reaction force at deflection ε

C_{rR} : Repetition factor of reaction force
$C_{agR}{}^{+}$, $C_{agR}{}^{-}$: Ageing factor of reaction force

The expressions in equations (5.5.5) to (5.5.8) are used as the performance functions of rubber fenders for the mooring simulation of vessels. The angle coefficient C_{aR}, velocity coefficient V_{FR}, and repetition factor C_{rR} are difficult to take into consideration when performing the simulation because the angle and speed change constantly, and repetition fatigue accumulates. As a countermeasure, the coefficients may be substituted by performing the calculation using only the maximum and minimum values of the performance. Furthermore, a complex behaviour is exhibited in which the velocity changes in the middle of compression, and this phenomenon cannot be reproduced without employing a method such as a hybrid simulation [9)]. However, as shown in Fig. 5.5.1, approximations can be made in which the upper and lower connections are established using linear spring constants of the initial deflection. In the design of fenders, the generated reaction force when the maximum factors are adopted is the design load to the structure, and the maximum deflection when the analysis is performed using the lower limit of each factor is the allowable design deflection. In the simulation, the specifications of the rubber fender are determined by trial and error to be within the design limits.

5.5.4 Consideration of number and length of fenders

The performance of a rubber fender with a fender panel can be modified by attaching multiple fenders to one panel. However, as shown in Fig. 5.5.2, if two rubber fenders are attached to one panel, although the energy absorption E_A will be doubled, the reaction force R will also be doubled. Furthermore, the performance of the rubber fender is related to the size; when the size is doubled, the reaction force becomes $2^2 = 4$ times, and the energy absorption becomes $2^3 = 8$ times. In other words, if one rubber fender having a size (height) of approximately 1.26 times is attached, the reaction force can be suppressed to 1.6 times while realizing two times the energy absorption. Therefore, no restrictions exist, such as in terms of the installation space of the quay or maximum height, and the efficiency of energy absorption (E_A/ R) is the maximum when using a single fender. However, when a large panel is required, several fenders may be used to ensure the stability of the system. Considering the length, as shown in Fig. 3.3.4 to Fig. 3.3.6 in Chapter 3, an adjustable type fender can also help modify the performance by increasing the length instead of changing the height. As described above, although various responses are possible depending on the application, when the length is less than the height, the compression mode may change due to the influence of both ends. In addition, it should be noted that the influence of the angle increases with increase in the installation pitch.

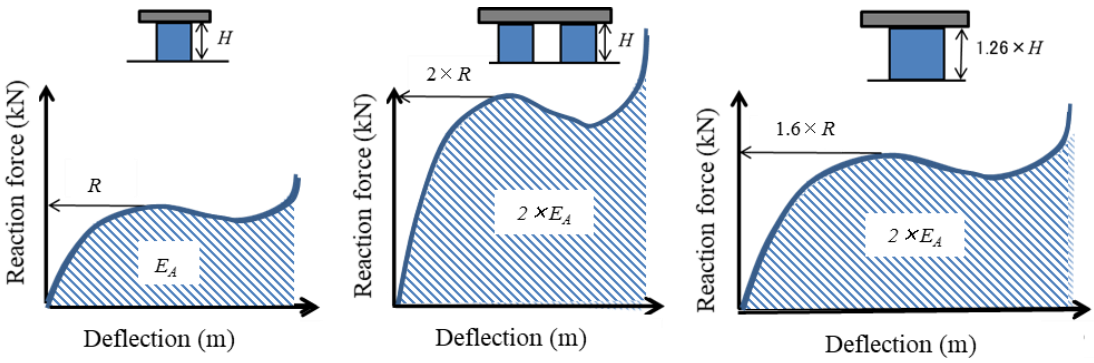


Fig. 5.5.2 Performance comparison of single and multiple fenders for fender panels

When upgrading existing facilities or dealing with larger vessels, the restriction of rubber fender height to the overhang of loading cranes, compatibility of size and pitch of fixing bolts, etc. are also important considerations.

5.6 Arrangement of rubber fenders

In the arrangement of rubber fenders, the effective berthing energy must be able to be absorbed, even in the worst situation, by taking into consideration the installation pitch, shape of the vessel, position of mooring facility, tidal levels, and draft; moreover, crashes should be avoided, and concerns must be clarified and addressed by adopting suitable countermeasures. In certain circumstances, it is difficult to obtain the curvature of the ship hull, which is necessary to determine the installation pitch; thus, this document does not specify the determination method and provides only reference examples.

5.6.1 Installation pitch of fenders

When installing a large number of rubber fenders at a constant pitch on a long quay, the determination of the installation pitch becomes important. As shown in Fig. 5.6.1.1 and Fig. 5.6.1.2, when multiple rubber fenders are installed at a pitch S on a quay, and the pitch S is shorter than the curvature radius of the vessel hull, multiple rubber fenders will be compressed. The installation pitch of rubber fenders needs to be determined so that the part of the hull closest to the quay wall does not make contact when the effective berthing energy is absorbed by the fenders. Several methods of determining the pitch S are proposed as a reference in this subsection. If the installation position is fixed at the existing mooring facilities and it is difficult to shorten the installation pitch, measures such as raising the rubber fender by the base and advancing the berthing line can be adopted.

(1) Design calculation method for example of harbour structure [3]:

Pitch: $S \leqq$ 1/5 to 1/6 times the length perpendicular of vessel ($\alpha \cdot Lpp \cdot cos\theta$) (α: see Fig. 5.3.2)

(2) British Standard [5]: Pitch $S \leqq 0.150.15 \cdot L_{oa}$ （L_{oa}: Overall length of vessel）

(3) Method to determine pitch from the hull radius of vessel: equation (5.6.1)

Assuming that the hull shape at the berthing part is a cylinder with a radius of hull curvature R_r, and assuming that a chord is present between two fenders separated by pitch S, as shown in Fig. 5.6.1.2, the pitch can be calculated according to equation (5.6.1) using the residual height h when the fender is compressed to its design deflection and the estimated values of hull radius R_r.

$$S = 2\sqrt{{R_r}^2 - \ (R_r - h)^2} \qquad (5.6.1)$$

Here,

S : Fender pitch (m)

R_r : Hull radius of vessel at berthing point (m)

H : Residual height when fender is compressed to design deflection (m)

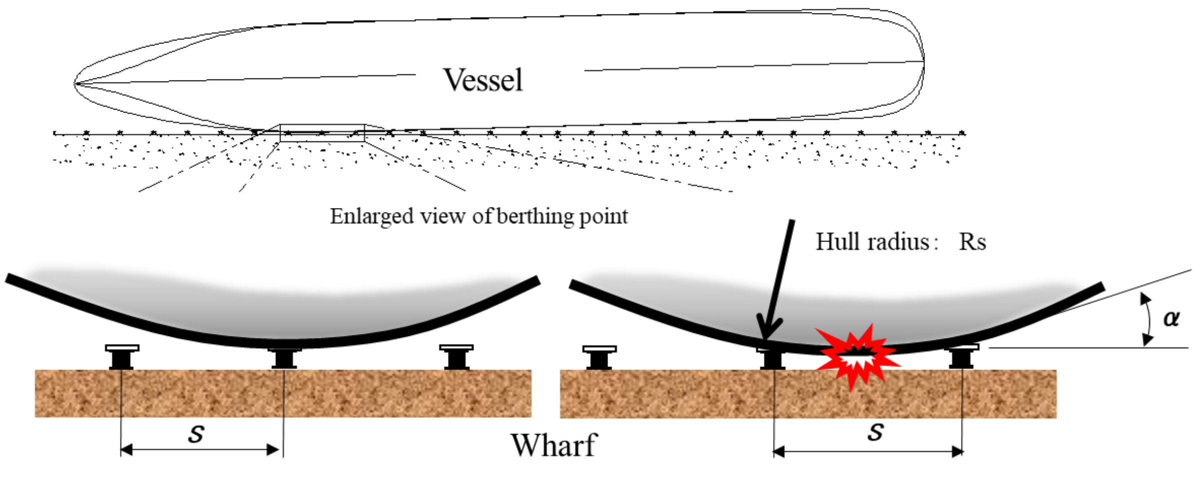


Fig. 5.6.1.1 Berthing at centre of three fenders　　Fig. 5.6.1.2 Berthing between two fenders

Assuming that two rubber fenders are compressed, as shown in Fig. 5.6.1.2, and assuming that the curvature radius of the hull is R_r, the local angle θ on the contact point can be estimated using equation (5.6.2).

$$\theta = sin^{-1}(\frac{S}{2R_r}) \quad (5.6.2)$$

Often, the installation pitch cannot be changed if the quay structure is based on the existing quay wall. In the case of replacement of the existing rubber fenders, it may be difficult to raise the height by a base or spacer.

The curvature radius Rr of the hull is often not available. In such a case, it may be assumed using equation (5.6.3).

$$R_r = \frac{B}{4} + \frac{{L_{pp}}^2}{16B} \quad (5.6.3)$$

Here,

B : Beam of vessel (m)

L_{pp} : Length between perpendiculars (m)

Equation (5.6.3) can be employed under the assumptions of quarter point berthing; furthermore, as shown in Fig. 5.6.2, if a circle passes through the three red points, the relationship presented in equation (5.6.4) holds. If this expression is calculated around R_r, equation (5.6.3) can be obtained.

$$R_r{}^2 = (R_r - B/2)^2 + (L_{pp}/4)^2 \quad (5.6.4)$$

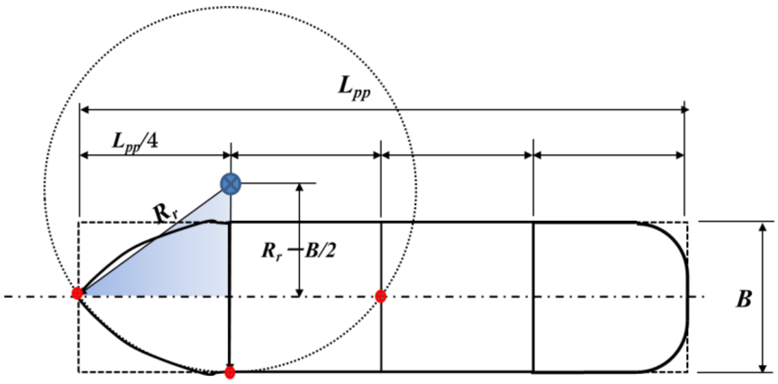


Fig. 5.6.2 Assumption of hull radius at berthing point

When a vessel comes in contact with multiple fenders, it is possible to use the sum of the reaction force and energy absorption of the respective fenders in contact. However, in reality, the shape data of a vessel hull is difficult to obtain, and thus, accurate confirmation is difficult. Therefore, if the radius of curvature cannot be obtained, it can be assumed using equation (5.6.3). In contrast, it is necessary to accurately determine the reaction force when the vessel contacts two fenders, as shown in Fig. 5.6.1.2, and when vessel contacts three fenders in the centre and on both sides, as shown in Figure 5.6.1.1. When the range of the vessel size is large, it is not economical to arrange the rubber fenders for large ships at the pitch calculated using the hull radius R_r of small vessels. In such a case, V-type rubber fenders for small vessels are sometimes placed in the middle of rubber fenders for large vessels. In addition, since the vessel hull above the fender level has a larger flare angle, it is also necessary to consider the contact of the vessel hull with the upper edge of the quay. Depending on the location, the flare angle may exceed the berthing angle by three to four times [10). However, it is difficult to obtain hull shape data for a modern vessel, and thus, the flare angles corresponding to different vessel types and sizes are not available. In principle, as large flares on the hull can cause damage to port facilities, berthing should be controlled to be parallel to the berth line, and large berthing angles should be avoided. In addition, the hull radius is considered to be smaller on the stern side than on the bow side; thus, when considering berthing at the stern side, it is reasonable to assume that a shingle fender must absorb the effective berthing energy.

5.6.2 Vertical installation of fenders

(1) Installation of V-type rubber fender

As shown in Fig. 5.6.3, V-type rubber fenders without fender panels do not contact at the full length for certain vessel positions, and rubber fenders can perform only in the length contacted. It is necessary to estimate the minimum contact length from the minimum deck height, draft, tidal level, etc. and select the length such that the effective berthing energy E_b can be absorbed. This phenomenon is called partial compression, and the performance can be evaluated using equation (5.6.5).

$$\text{(Energy absorption by partial compression)} = \text{(Energy absorption per meter)} \times \text{(Contact length)} \qquad (5.6.5)$$

In the case of rubber fenders with a constant cross-section (for example, V-type rubber fenders) for which the performance can be modified by changing the length, if the length is shorter than the height, the length ratio is affected by both the ends. The length ratio in this case may not correspond to the performance being proportional to the length. The contact length of a rubber fender relates to not only the length of the rubber fender but also the installation position. The installation considerations for V-type rubber fenders are listed as follows, and Fig. 5.6.3 provides an explanatory view of the aspects to be considered.

・ The flare of a large vessel should not contact the edge of the quay, even when the rubber fender is compressed.
・ It is desirable that the top head of fender is always visible and not be submerged in water.
・ The necessary concrete coverage should be secured over the anchor bolts
・ The effective berthing energy must be absorbed by the length in contact with the rubber fenders

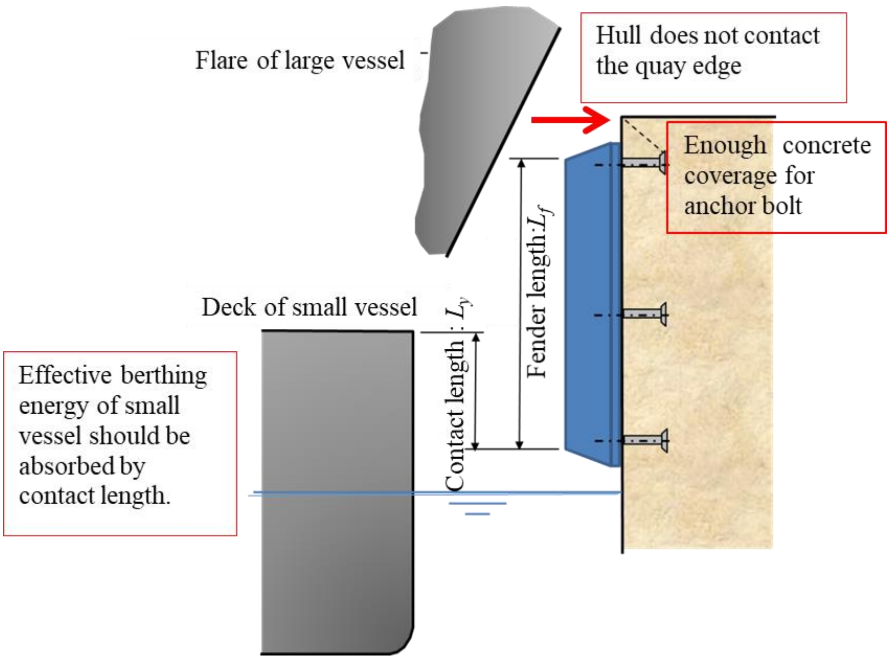


Fig. 5.6.3 Considerations for vertical installation of V-type rubber fender

As shown in Fig. 5.6.3, when the flare of a large vessel comes in contact with a vertically installed V-type rubber fender, the fender is subjected to vertical angular compression. Generally, the angular performance of V-type rubber fenders described in catalogues is the angular performance in the width direction, and the longitudinal angular performance is often not defined because it changes depending on the length of the rubber fender. In such a case, the performance can be determined in the following manner.

The performance curve of the energy absorption E_A of a V-type rubber fender is defined as in equation (5.6.6) as a function of the deflection x by considering a polynomial function or numerical values in a table.

$$E_A \quad = \quad f(x) \qquad \text{Continuous functions such as polynomials} \qquad (5.6.6)$$

If variables are defined as shown in Fig. 5.6.4 for the condition shown in Fig. 5.6.3, the angular reaction force of a rubber fender can be estimated using equation (5.6.7) by integrating the reaction force of the compressed part.

$$E_{A\theta} = \int_{y=0}^{L_y} f(D\text{-}y\tan\theta)\,dy \qquad \text{Continuous function}$$

$$= \sum_{y=0}^{L_y} f(D\text{-}y\tan\theta) \qquad \text{Discrete data} \qquad (5.6.7)$$

Here,

$E_{A\theta}$: Energy absorption at simulated angular compression (kN·m)
Ly : Contact length (m)
D : Design deflection of V-type fender (m)
θ : Flare angle
y : Variable from fender top (m)

The maximum reaction force to the quay is larger when the entire surface is compressed at an angle of 0°; thus, a reaction force with an angle of 0° is used.

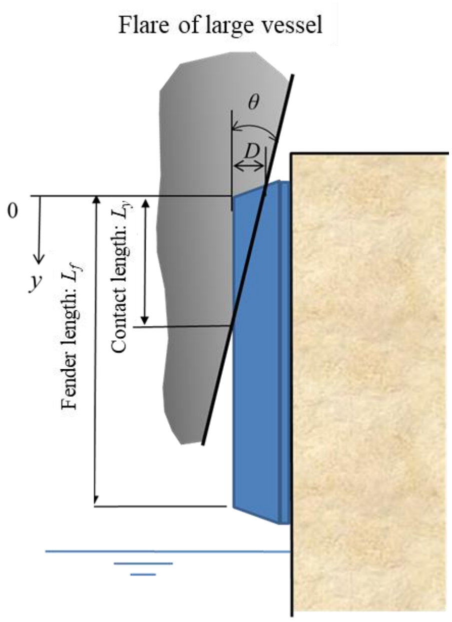


Fig. 5.6.4 Angular compression by hull flare to V-type rubber fender

Since the amount of compression by the hull flare, as calculated using equation (5.6.7), is small, the fender may not be able to absorb the effective berthing energy of a large vessel. In such a case, horizontal installation or alternate vertical and horizontal installation of V-type rubber fenders can be realized. Alternatively, rubber fenders with fender plates may be used.

(2) Installation of rubber fender with panel

Even in the case of rubber fenders with panels, it is necessary to perform a geometric investigation of the minimum vessel freeboard, draft and lowest tidal level. In such cases, the panel, chain, etc. need to be considered in the system, as described in the next section.

5.7 Detailed design and considerations

In the next step for the selection of a rubber fender, the design of parts such as fender panels, chains and fixings is required for the fender system. In particular, in the case of rubber fenders with panels, the design of these parts is a valuable aspect that determines the final size, rubber grade and cost. After the selection, based on the effective berthing energy, if the design of a fender panel or chain for the selected rubber fender is not suitable, the designer may overlook the inherent hazards of the system. It is necessary to perform the design appropriately in accordance with the engineering principle and confirm the results with the purchaser. The actual design calculations may be left to manufacturers because the details of these design procedures vary by type, size, and manufacturer. Therefore, the basic concepts of the fundamental balance of forces and moments are presented here.

5.7.1 Allowable hull pressure

The fender panel should be joinable with the rubber fender body, and its size should be larger than the required contact area for the allowable average hull pressure. The average hull pressure P on the panel is determined by dividing the design reaction force by the effective panel area, as shown in equation (5.7.1), to determine the size of the fender panel.

$$P = \frac{R}{A} \leq H_p \tag{5.7.1}$$

where

P : Average hull pressure (kN/m^2 = kPa)
R : Design reaction force (kN)
A : Effective contact area of fender panel ($A = W_{Fe} \text{ x } H_{Fe}$: m^2)
H_p : Allowable hull pressure (kN/m^2 = kPa)
W_F : Width of fender panel (W_{Fe} = Effective width : m)
H_F : Height of fender panel (H_{Fe} = Effective height : m)

Table 5.7.1 presents the values of the maximum allowable hull pressure recommended for each type of vessel in the PIANC Guidelines [s)] amended with an example of the domestic results. According to the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [2)], the recommended hull pressure is 200 to 400 kN/m^2 on the premise of the frame (ribs) coming into contact on the vessel side. The effective width and effective height of the fender panel may not include the oblique chamfer portion of the end in the area but may include the gaps between the resin pads. This configuration is shown in Fig. 5.7.1.

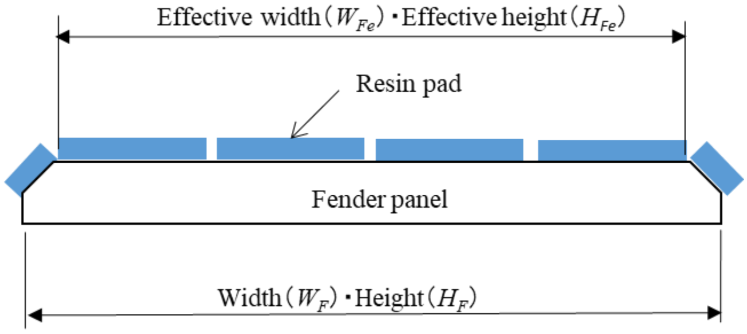


Fig. 5.7.1 Effective size of fender panel

Although the hull pressure is restricted by the strength on the vessel side, the hull pressure is not uniform because the reinforcement on the hull is not uniform. The allowable hull pressure is the average face pressure, used as a standard for determining the size of the fender panel. However, in certain locations, the pressure is higher than the average hull pressure at the reinforcing rib of the vessel side. Furthermore, the contact pressure is zero on the indented area. In addition, in the case of angular berthing, the contact of the panel in the initial stage of berthing is a point or line contact, and the contact area cannot be calculated. Although the hull pressure is usually referred to as the contact pressure, here, it is termed as the average hull pressure to distinguish it from the local contact pressure. Moreover, since this pressure has a large value in a rubber fender without a panel, the panel is often required when the hull pressure is limited.

Table 5.7.1 Guide of allowable hull pressure

Type of vessel	Allowable hull pressure（kN/m^2）
Container vessel	
1st and 2nd generation [*1]	<400
3rd generation (Panamax 1700-2500 TEU)[*1]	<300
4th generation (3600–4800 TEU)[*1]	<250
5th and 6th generation (SuperPanamax, 4900 TEU) [*1]	<200
Recent examples in Japan [*2]	<200–290
General cargo vessel	
≦20,000　*DWT*	<400–700
＞20,000　*DWT*	<400
Oil tanker	
≦60,000　*DWT*	<350
＞60,000　*DWT*	＜300
VLCC (Very large oil tanker)	<150–200
Recent examples in Japan [*2]	<200
Gas carrier　(LNG/LPG)	<200
Recent examples in Japan (LNG)[*2]	<130–134
Recent examples in Japan (LPG)[*2]	<245
Bulk carrier	<200
Recent examples in Japan (Ore carrier)[*2]	<280–320
SWATH (Small-waterplane-area twin hull)	Hull pressure is not considered for vessels with belt protection.
RO-RO (Roll On Roll Off) vessel	
Passenger vessel	

Notes: * 1 The TEU of each generation of container vessels is based on Tanaka et al. [11)].

* 2 Amended PIANC Guidelines [4)] based on examples corresponding to Japan.

5.7.2 Load cases for fender panel

The bending moment applied to the fender panel should be carefully considered according to the manner in which the berth force is applied, as shown in Fig. 5.7.2.

1) Point load: Corner contact at the initial stage of berthing, etc.
2) Line load: Angular berthing, contact of hull flare, etc.
3) Two line loads: Contact of vessel with belted hull
4) Central line load: Contact with the centre of two or more vertical fenders
5) Distributed load: Full contact or partial contact with fender panel

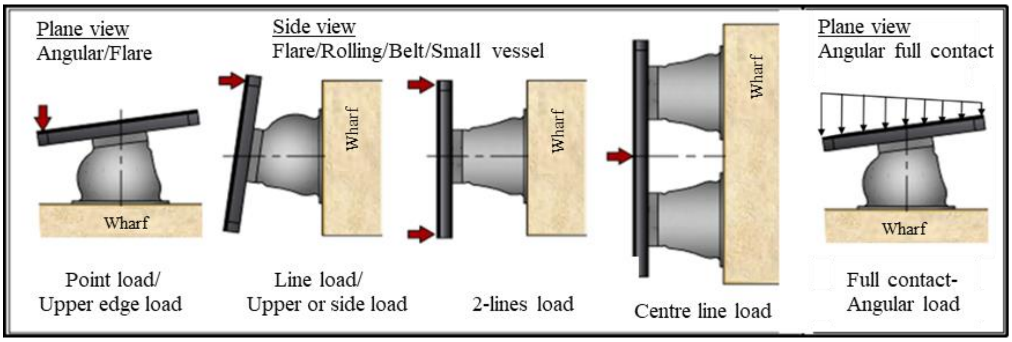


Fig. 5.7.2 Load cases for fender panel

5.7.3 Design Considerations

When designing fender panels and chains, the berthing and cargo handling operations can affect the durability of fenders and the safety of operations. Some representative examples are as follows:

(1) Prevention of breakage due to hooking of mooring ropes

As shown in the left figure in Fig. 5.7.3, the case in which a low freeboard vessel performs berthing lower than a rubber fender is called low contact. In this case, a tension chain is often installed at the upper end of the fender panel; since this chain is connected to the eye plate on the top of panel, as shown in Fig. 5.7.3, the chain can prevent the ropes coming between the quay and fender panel and being hooked when the mooring rope loosens. This consideration is important to ensure that a mooring rope caught on the fender panel does not interfere with rope management or cause an accident. In many cases, rope wear is reduced by covering the chain with a rubber hose or rope guard and processing the corners of the eye plate (chain hanger) to be smooth. Similar to a rope guard, a guard exists that extends the bar from the lower end of the panel to the water to prevent the penetration of the mooring rope underneath the fender. In addition, sharp corners must be removed in locations in which the rope is likely to be caught and to keep the upper end of the panel at a position lower than that of the quay top.

(2) Preventing hooking on the hull belt

As shown in the right figure in Fig. 5.7.3, if the vessel has a belt that extends from the hull, installing a wing to prevent hooking by producing an inclined portion at the upper and lower ends of fender panel is effective. In addition, if such a protrusion is present on the vessel side, the resin pad on the surface of the panel may be damaged rapidly; thus, a resin pad should not be used when constantly receiving loads such as those of a ferry.

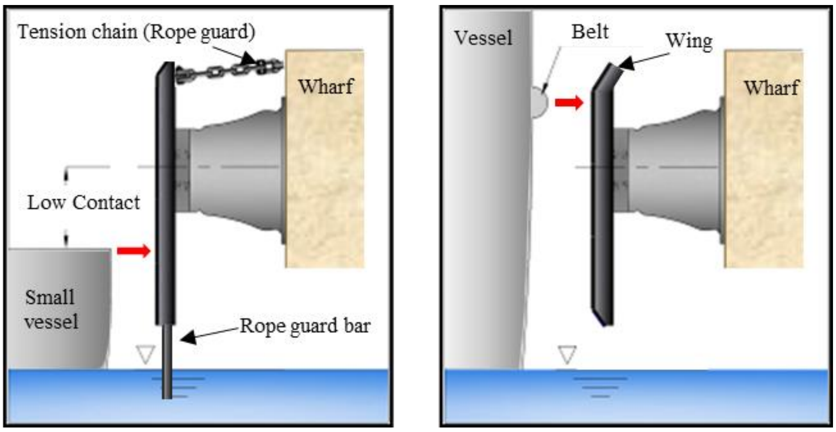


Fig. 5.7.3 Design considerations for areas surrounding fender panel

5.7.4 Load cases for regions surrounding fender panel and chains

The basic concept of each load case occurring in the fender panel is described below. Since these calculations assume rubber fenders acting as spring elements as a function of deflection, the calculation process is often left to the manufacturer. However, it is desirable to be able to verify whether the results are correct based on basic principles.

(1) Line load cases

As shown in Fig. 5.7.4, equation (5.7.2), and equation (5.7.3), when the hull contacts the rubber fender at an angle and is compressed with the line load F, the rubber fender generates a reaction force R by deflection δ_1 and the rotational moment M_f that resists the rotational angle θ. Although shear resistance is also generated, it is ignored because its value is small. The bending moment and chain tension generated in the fender panel can be determined from the balance of forces and rotational moments. When the rotational angle becomes θ, the vessel hull and the fender panel come into full contact, and the pressure distribution is as shown in Fig. 5.7.8. Therefore, Fig. 5.7.4 can be considered to represent the state immediately before full contact.

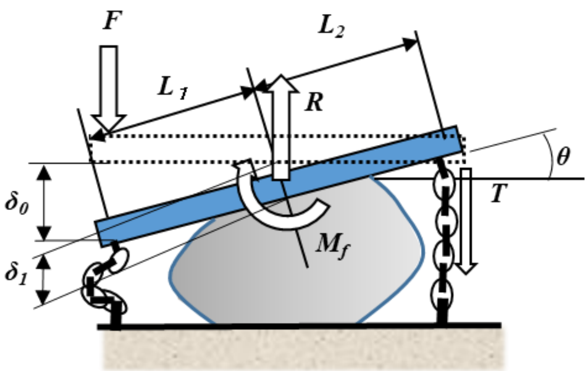


Fig. 5.7.4 Line load by angular berthing: Single fender with chains

In Fig. 5.7.4, the balance between the forces applied to the fender panel and the rotational moments, as well as the geometric relationship between the displacements, are as shown in equation (5.7.2).

$$\left.\begin{aligned} &R = F + T \\ &M_f = T \cdot L_2 - F \cdot L_1 \\ &\delta_1 = \delta_0 - L_1 \cdot \tan\theta \end{aligned}\right\} \quad (5.7.2)$$

Here,

R : Reaction force of fender (kN)
M_f : Rotational moment of fender (kN・m)
F : Berthing force (kN)
T : Chain tension (kN)
δ_0 : Deflection at berthing point (m)
δ_1 : Deflection at fender centre (m)
θ : Berthing angle (°)

If no chain is present, Fig. 5.7.4 changes to Fig. 5.7.5, and the balance between the force and rotational moment can be expressed as in equation (5.7.3).

$$\left.\begin{aligned} &R = F \\ &M_f = F \cdot L_1 \\ &\delta_1 = \delta_0 - L_1 \cdot \tan\theta \end{aligned}\right\} \quad (5.7.3)$$

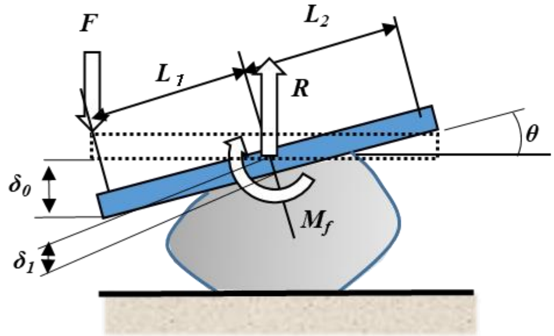


Fig. 5.7.5 Line load by angular berthing: Single fender without chain

If the compression deflection δ_1 (displacement amount / height, %) and the inclination angle θ are determined, the reaction force R and the rotational moment M_f of the rubber fender can be determined using a polynomial function or data table. However, the method of determination differs depending on the type of product and manufacturer, and thus, it is not mentioned in this document. In particular, in the case of multiple fenders, the rotational moment of fenders is often omitted. The reaction force of the fenders and the tension of chains can be obtained from equation (5.7.2) by varying the berthing point deflection δ_0 from 0 in appropriate increments. The strength of the fender panel can be obtained by calculating the generated bending moment and the section modulus of the panel. However, since the structure of the fender panel also differs depending on the manufacturer, this aspect is not considered herein, and only the balance of the loads is demonstrated.

If two rubber fenders are present on a single fender panel, the condition is as shown in Fig. 5.7.6, and the balance between the force and rotational moment can be expressed as in equation (5.7.4).

$$\left.\begin{aligned} & R(1)+R(2) = F + T \\ & M_f(1)+M_f(2) = T\cdot(L_1+L_2+L_3) - R(1)\cdot L_1 - R(2)\cdot(L_1+L_2) \end{aligned}\right\} \qquad (5.7.4)$$

Assuming a chain does not stretch, the geometric relationship of the deflections of the two fenders and the berthing point is as shown in equation (5.7.5).

$$\left.\begin{aligned} & \delta_1=\delta_0 - L_1\cdot tan\theta \\ & \delta_2=\delta_1 - L_2\cdot tan\theta \end{aligned}\right\} \qquad (5.7.5)$$

In addition, in this case, if the deflection δ_0 at the berthing point is changed from 0 to a certain value in appropriate increments, the deflection of each fender can be determined using equation (5.7.5), and the berthing force F and chain tension T can be determined using equation (5.7.4). The energy absorption can be obtained by integrating the berthing force F from the deflection δ=0 to the design deflection δ_{max}, as given in equation (5.7.6).

$$Ea = \int_{\delta=0}^{\delta max} F(\delta)\, d\delta \qquad F(\delta)\text{: Continuous function}$$

$$= \sum_{\delta=0}^{\delta max} F(\delta) \qquad F(\delta)\text{: Discrete data} \qquad (5.7.6)$$

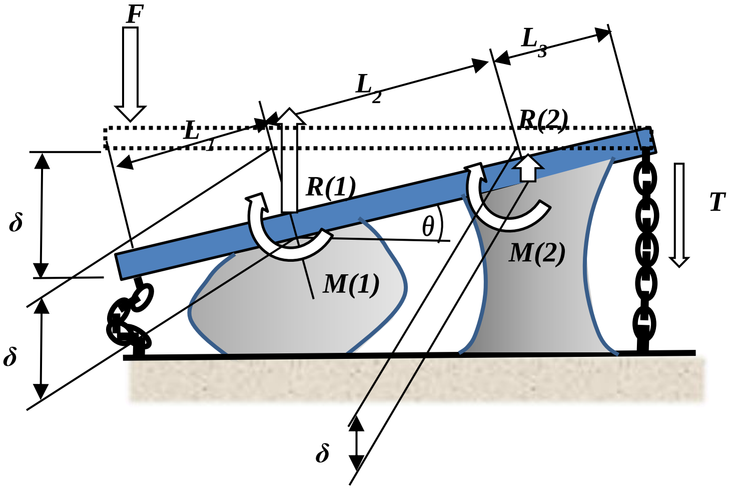


Fig. 5.7.6 Line load by angular berthing: Two fenders with chain

(2) Point load case

When a vessel contacts the fender panel at a three-dimensional angle, it is initially a point load. As the hull approaches further, it becomes a line load and eventually contacts the entire surface of the panel. When several fenders are attached to a single large fender panel or the angle of the vessel hull is large, the point load state continues for a large duration. It should be noted that the maximum bending moment may act on the panel in this state, or maximum chain tension may occur. A point contact with a fender system with an arbitrary number of rubber fenders is illustrated in Fig. 5.7.7, and the state is expressed in equation (5.7.7). The number of equations and number of unknown variables increase; however, if there exists a function of reaction force $R(i,j)$ for the deflection of each fender, the bending moments and chain tension can be determined via numerical calculations. The energy absorption can be obtained using equation (5.7.6). Note that if the fender panel is large and the panel edge contacts the quay before the design deflection of the fender, the fender must be able to absorb the effective berthing energy up to that point. Equation (5.7.7) has several unknown variables, and thus, it needs to be solved using numerical calculations, such as Newton's method. The validity of the result can be confirmed by substituting it into the left side of equation (5.7.7) and ensuring that the right side becomes zero.

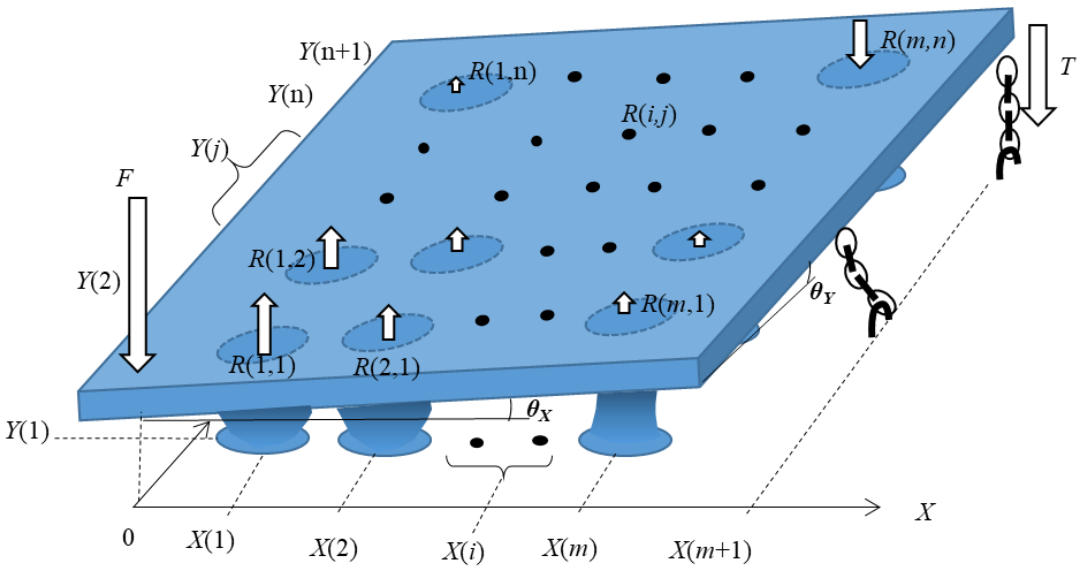


Fig. 5.7.7 Point load by angular berthing: Multiple fenders $n \times m$

$$\left.\begin{array}{l} F+T+\sum_{i=1}^{n}\sum_{j=1}^{m} R(i,j)=0 \\ \sum_{i=1}^{m}\{X(i)\cdot\sum_{j=1}^{n} R(i,j)\}+X(m+1)\cdot T=0 \\ \sum_{j=1}^{n}\{Y(j)\cdot\sum_{i=1}^{m} R(i,j)\}+Y(n+1)\cdot T=0 \end{array}\right\} \quad (5.7.7)$$

Here,

$X(i)$: X coordinate of the i-th rubber fender in X direction (m)
$Y(j)$: Y coordinate of the j-th rubber fender in Y direction (m)
n : Number of fenders in the Y direction
m : Number of fenders in the X direction
$R(i,j)$: Reaction force of fender (i,j) (kN)

Here, the rotational moment of the rubber fender is ignored. Additionally, the geometric relationship of the deflection is as given in equation (5.7.8).

Under such a point load condition, when the deflection of the rubber fender reaches the design deflection, or the fender panel comes into contact with the quay, the chain tension may increase to the design limit. As a result, the link may become too large to be selected. According to the PIANC Guidelines [4), the safety factor of a normal chain is set between 3 and 5, and a safety factor of 2.0 may be adopted when the chain load is increased due to an abnormal berth. As described above, it is acceptable to reduce the safety factor to 2.0 when one chain is subjected to a large tension in a short period.

$$\left.\begin{array}{l} \delta_{(1,1)}=\delta_0 - X(1)\cdot\tan\theta_X - Y(1)\cdot\tan\theta_Y \\ \cdot \\ \delta_{(i,j)}=\delta_0 - X(i)\cdot\tan\theta_X - Y(j)\cdot\tan\theta_Y \\ \cdot \\ \cdot \\ \cdot \\ \delta_{(m,n)}=\delta_0 - X(m)\cdot\tan\theta_X - Y(n)\cdot\tan\theta_Y \end{array}\right\} \quad (5.7.8)$$

(3) Distributed load case

When an angled hull contacts the fender panel, and the entire surface touches the vessel side after the point load and line load, a distributed load is considered to occur, as shown in Fig. 5.7.8.

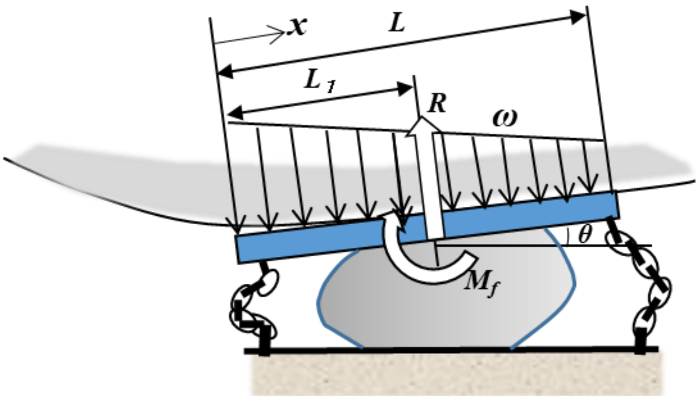


Fig. 5.7.8 Distributed load by angular berthing: Single fender

At this time, the balance between the forces and rotational moment around the fender panel is as given in equation (5.7.9).

$$R = \int_{x=0}^{L} \omega(x)\, dx$$

$$M_f = \int_{x=0}^{L} \omega(x) \cdot x\, dx - R \cdot L_1 \tag{5.7.9}$$

In equation (5.7.9), R is a reaction force for each deflection at the time of angular compression at angle θ. M_f is a rotational moment at the deflection and angle at that instant, which varies depending on the product but is required to determine the slope of the distributed load $\omega(x)$. The distributed load may be an equal distribution load for angle=0, or a slope distribution load for angle=θ, as shown in Fig. 5.7.8. However, in fact, the shape of the distribution depends on the position of the frame (rib) on the vessel hull, and thus, it is difficult to determine the distribution shape. The distributed load in Fig. 5.7.8 is an example for a case in which a linear inclination is assumed. Normally, the strength obtained by the point load or line load is often used, but the distribution load can be used when the angle is unknown, and the state of point load or line load cannot be set.

As described above, the bending moment generated at each position of the fender panel can be calculated from the load balance condition in accordance with the berthing conditions. Subsequently, the maximum bending moment can be determined, and the generated stress can be calculated from the section modulus of the fender panel. This stress must be within the allowable bending stress.

(4) Chain design

Chains are used to restrain undesirable deformations of rubber fenders. Chains may be classified into the following categories according to their functions, and they are designed as needed.

1) Tension chain (Fig. 5.7.9)

- To control the stretching of the rubber fender
- To control the rotation of the fender panel.

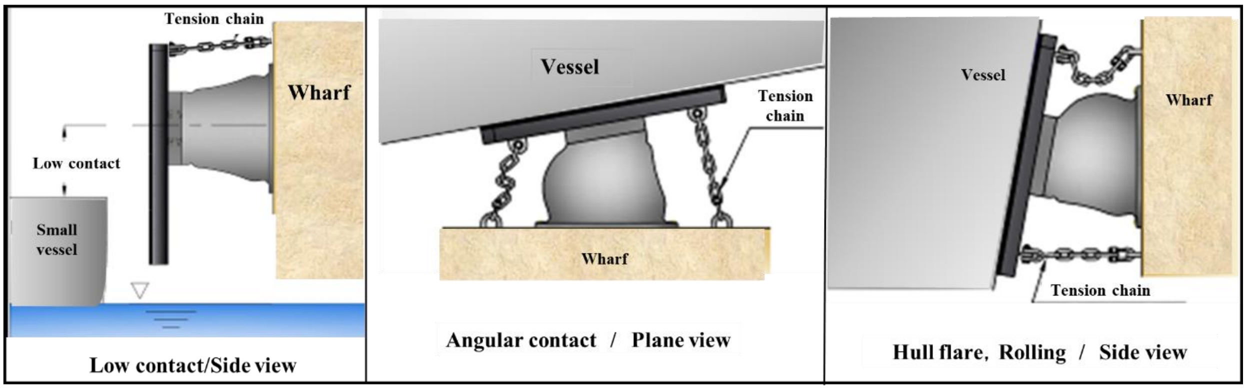


Fig. 5.7.9 Role of tension chain

2) Weight chain (Fig. 5.7.10)

- To control the front drooping of the fender
- A tension chain is needed to control the tipping forward of the fender panel if the weight chain is attached to the lower part of the panel.

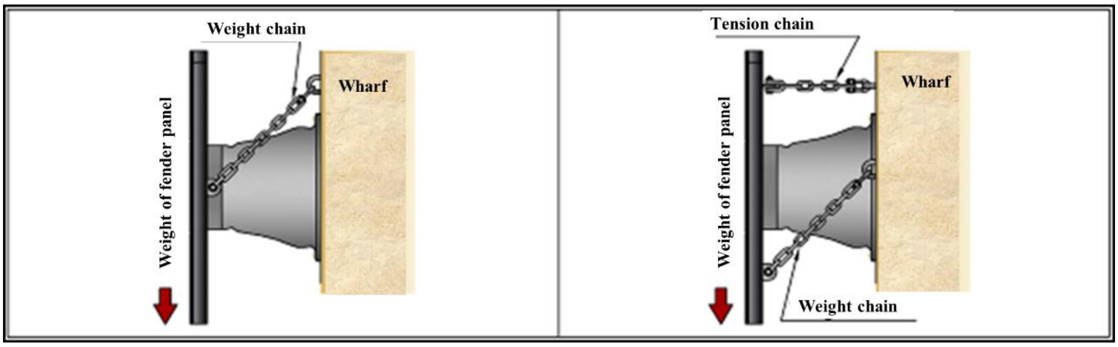


Fig. 5.7.10 Role of weight chain

3) Shear chain (Fig. 5.7.11)

・ To control shearing of the rubber fender

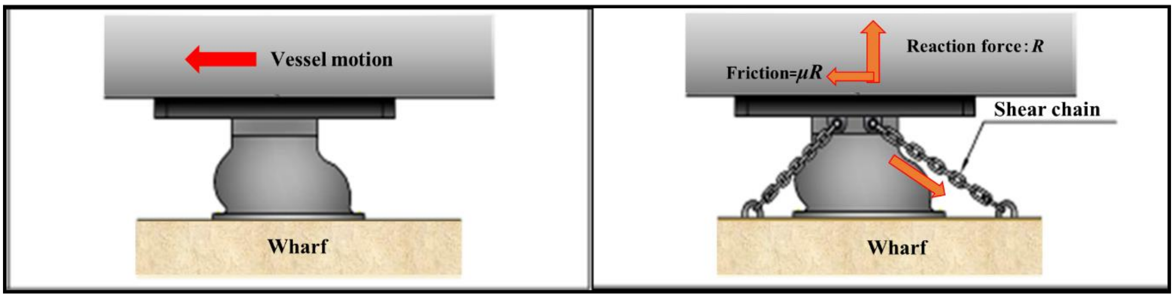


Fig. 5.7.11 Role of shear chain

4) Example of friction coefficient μ

In Fig. 5.7.11, μ is the friction coefficient between the fender panel surface and the vessel hull. The following values are often used as an example of the friction coefficient.

Fender panel (resin pad) and vessel hull (Steel): 0.2 to 0.25

Fender panel (Steel) and vessel hull (Steel): 0.3 to 0.35

Rubber and vessel hull (Steel): 0.3 to 0.4

In most cases, the chain tension is calculated in the process of obtaining the bending moment of the fender panel, as explained in the previous section. The load conditions must be set appropriately, and the maximum tension generated must be determined and ensured to be less than or equal to the design tension of the chain. Not all cases of fenders with panels require chains. The requirement of chains should be carefully considered, as the installation of chain often requires that the fender panel has additional strength.

5.8 Corrosion protection of steel parts

Although it depends on the environment and use conditions, the service life of corroded steel is generally shorter than the life of a rubber body if no measures are adopted. Therefore, when using steel parts, it is necessary to provide appropriate corrosion protection in accordance with the coastal environment. Two primary types of corrosion protection methods exist: cathodic protection and anticorrosion coatings. The Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [2)] recommends that the cathodic protection method should be adopted in cases involving water levels equivalent to or lower than the mean low water level (MLWL), and anticorrosion coatings should be adopted in cases involving water levels higher than 1 m less than the low water level (LWL-1 m). Data regarding corrosion protection and repair of steel structures in ports and harbours are described in detail in, for example, the Manual for Corrosion Protection and Repair of Steel Structures in Ports and Harbours [12)]. In this work, the considerations for coating and plating commonly used for bolts and chains are presented. If part or all of the fender panel falls below the average low water level, coating protection and cathodic protection may be used in combination; for details, the reader may refer to the Manual for Corrosion Protection and Repair of Steel Structures in Ports and Harbors [12].

With steel materials such as fender panels and chains used for rubber fenders, it is necessary to properly perform maintenance and control together with the adoption of anticorrosion measures and partial replacement based on the state of deterioration. The cycle of replacement is considerably affected by maintenance measures such as touch-up painting. To evaluate the degree of deterioration, one may refer to the Guidelines for the Maintenance of Rubber Fender Systems [13].

5.8.1 Painting

Fender panels are mainly protected by painting. An example of the paint specification is described. Because a fender panel can be checked and replaced on a regular basis, in contrast to other steel structures used in mooring facilities, the need for long-term corrosion protection is not as high as for permanent steel structures. However, the environment involves a splash zone with the influence of ultraviolet rays, and thus, a maintenance plan should be carefully considered with equal focus on the economic aspects. According to the Manual for Corrosion Protection and Repair of Steel Structures in Ports and Harbors [12], as given in Table 5.8.1, the useful life is 10 to 15 years for both heavy protection grade-1 and heavy protection grade-2 paints.

The corrosion rate of steel after deterioration of the paint film is defined in Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [2] as follows.

H. W. L or higher: 0.3 mm/y
H. W. L to L. W. L-1 m: 0.1 to 0.3 mm/y

It is necessary to consider the abovementioned corrosion allowance assuming that the paint has deteriorated and lost its effect. As specified in the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [2], the corrosion countermeasure, which is based solely on the additional thickness, should not be applied. Therefore, if painting is performed properly and maintenance is possible, it is not necessary to incorporate the corrosion allowance in the strength design unless extraordinary circumstances exist. However, in a case in which the paint on the front face of the fender panel wears away due to the belt of the vessel, such as in the case of ferries, the corrosion allowance of 0.3 mm/year must be considered in the stress design, and the strength after wear should correspond to the yield point of the steel material.

Table 5.8.1 Corrosion protection for fender plate using paint

Paint spec.	Heavy protection grade-1	Heavy protection grade -2	
Substrate adjustment	Degree of adjustment: Sa 2-1/2 or more	Degree of adjustment: Sa 2-1/2 or more	
Primer	Thin film organic (Epoxy) Zinc rich primer: 20 μm	Thin film organic (Epoxy) Zinc rich primer: 20–30 μm	Thick film organic (Epoxy) Zinc rich primer: 60–70 μm
Top coat	Tar epoxy substitute resin paint 300 μm	Epoxy resin paint 430 μm	Epoxy resin paint 390 μm
Total film thickness	320 μm	450 μm	

5.8.2 Plating

Plating protection is applied at locations involving metal-to-metal contact such as for chains and shackles. The plating specification of a chain generally has a molten zinc adhesion amount of 550 g/m^2 or more or a thickness of 76 μm or more. However, to secure engagement, the threaded portion is not limited to the above values.

Bolts and nuts are often made of stainless steel or plated steel depending on the conditions. Surface treatment according to the material is described in Table 5.8.2.

Table 5.8.2 Surface treatment for bolts and nuts

Material	Surface treatment/Remarks
SS400	Galvanization
SUS304	When the material is subjected to hot forging after solution heat treatment, solution heat treatment is performed again; subsequently, passivation film formation treatment such as acid washing is performed.
SUS316	

5.9 Allowable stress of steel material

The allowable stress of steel materials is presented in Table 5.9.1. Because the allowable stress varies depending on the plate thickness, steel plates of 16 mm or more and stainless-steel materials are considered, for example, as specified in the Design Standard for Steel Structures [14] of the Architectural Institute of Japan.

Table 5.9.1 Allowable stress of steel material (Thickness less than 16 mm)

Allowable stress (N/mm^2)		Technical Standards and Commentaries of Ports and Harbour Facilities in Japan, 1989			
		Tensile strength	Yield stress or 70% of tensile stress	Allowable tensile stress	Allowable shear stress
Structural steel	SS400	400	245	140	80
	SM490A	490	325	190	110

Based on the type and strength of the chain, the following Japanese Industrial Standards can be referred to.

Japanese Industrial Standard JIS F-3303-2010, “Flash butt welded anchor chain":

The breaking load and bearing load capacity of the stud-link and normal link chains are defined. The bearing load is the load at which the material begins to undergo plastic deformation, and its value is approximately 1/1.4 of the breaking load for a chain with a stud and 1/2 for a chain without a stud.

Japanese Industrial Standard JIS F-2106-2013, “Marine General Purpose Chains":

The bearing load and working load are defined for a studless chain; the working load is approximately 1/2.5 of the bearing load.

Assuming that the safety factor is the ratio of the breaking load to the working load, the safety factor is 5.0 for studless chains and 3.5 for stud-link chains. Moreover, in the PIANC Guidelines [4], the safety factor of the chain ranges between 3 and 5, and it is considered to be 2.0 in the case of abnormal berthing. With reference to these values, in the case of a chain for rubber fenders with a fender panel, a safety factor of at least 3.0 should be considered when a load is always applied similar to a weight chain. In addition, as described in Section 5.7.4, when point loads occur in a short period in one chain due to angular berthing, the safety factor may be considered as 2.0.

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14) Architectural Institute of Japan: Design Standard for Steel Structures－Based on Allowable Stress Concept, 2005

Chapter 6
Testing

6.1 General

Rubber is a visco-elastic material that exhibits complicated characteristics. To conduct performance testing of a rubber fender, it is necessary to set appropriate conditions and control them properly to ensure that the test is conducted according to the given conditions. In this book, the tests are classified as follows according to the purpose: a development test, an authentication test, and a quality verification test. Management and operation are performed according to the classified purposes.

6.2 Test classification by purpose

The tests for rubber fenders can be classified into the following three types according to the purpose of the test.

(1) Development test

A series of tests is conducted by the manufacturer for the development of rubber fenders; among these, the tests that are the basis of catalogue values and technical data are called development tests. Such tests are often implemented under various conditions to cover all possible situations. Normally, reproduction is time-consuming and costly and may not be released, as it is a proprietary technology of the manufacturer. Technical documents or catalogues must clarify details regarding the parameters of the tests as much as possible.

(2) Authentication test

A test commissioned by a third party to conduct a type approval or an objective evaluation on a specific item of the fender is known as an authentication test. Several methods of authentication testing exist: for instance, when a sample or an actual product is submitted for testing, and a test is conducted by a third party who issues a certificate; when the test is conducted by the manufacturer in the presence of a third party; or when the test is conducted by the manufacturer and the certificate is issued by a third party upon the submission of the test report.

(3) Quality verification test

A test performed by a manufacturer to confirm the quality of a product before shipping is known as a quality verification test. This test is carried out for all or extracted samples as needed, and the inspection results are reported to the purchaser. The purchaser or agent may also be present during testing. A quality verification test includes the quality control test defined in the Standard Specifications for Ports and Harbor Works, [1)] and the control test is internally conducted by the manufacturer.

Table 6.2.1 summarizes the abovementioned test classifications. The method and purpose differ depending on the test classification; therefore, even the same test should be conducted in an appropriate manner in line with the purpose.

Table 6.2.1 Classification of rubber fender test

	Definition	Test type	Method of disclosure	Practitioner
Development test	Tests conducted by manufacturers during development of rubber fenders based on catalogue data, etc.	Standard compression test; test for influence of angle, velocity and temperature; material test; durability test; repetition test; creep property, etc. as required	Manufacturer’s own disclosures, catalogues, website; however, some information may not be disclosed due to it being proprietary technology, or the tests may take excessive time and cost to reproduce	Manufacturers or their third party agents
Authentication test	Testing mainly entrusted to third parties to obtain type approval and objective evaluation of specific items	Tests that can be performed by third parties (Durability tests can also be performed by manufacturers)	Certification documents designated by third parties	Third party. Durability test can also be performed by the manufacturer
Quality verification test	Test performed by manufacturer to confirm product quality before shipping for all or extracted samples	Standard compression test, material test	Test report by manufacturer or third-party who can perform testing	Manufacturer: (Shipment test) Purchaser: (Acceptance test)

6.3 Development test

The development test is a test carried out by the manufacturer during the development of rubber fenders for the improvement of product quality and the formation of a technical basis including catalogue data, etc. The following is an explanation of the major development tests, in particular, the test methods concerning the influence factors, data processing, catalogue entry items, etc., and their examples. As these example figures are only for the purpose of explanation, they should not be used in actual design, and data from the manufacturers' catalogues, technical documents, etc. must be referred to.

6.3.1 Static compression test (Standard compression test)

The static compression test is an index that represents the basic performance of rubber fenders. In addition to being a development test, it is also performed as an authentication test and a quality verification test, and it is performed widely for actual products to scale models. Although large-scale actual fender tests depend on the scale of the testing facility, manufacturers must be able to carry out this test internally or externally for all types, sizes, and performance grades manufactured in-house. The specific test method can be described as in the following steps according to the PIANC Guidelines [2).

(1) Temperature stabilization of test piece and environment

Since the performance of rubber fenders is affected by the temperature, the test should be conducted in a temperature-controlled environment at the target temperature. The target temperature for measuring the standard performance is 23±5°C (standard temperature), and the static compression performance at this temperature is known as the standard compression performance. It is desirable to conduct the static compression test in an environment having a temperature within the target temperature of ±15°C, even under conditions in which it is difficult to maintain constant temperature. Specifically, the following two methods exist.

1) Temperature control before test

Proper temperature control should be realized by storing rubber fenders in an environment such as that of a temperature-controlled room at a target temperature of ±5°C for a necessary time until stabilization. The number of days required to stabilize the temperature of the entire rubber body can be determined using equation (6.3.1).

$$\text{Days for thermal stabilization} = 20 \cdot (\text{Maximum rubber thickness(m)})^{1.5} \qquad (6.3.1)$$

After the fender is moved out of the temperature-controlled room, if the test environment is out of the range of the target temperature ±5°C, the total test time including the preliminary test should be within 2 h to reduce its influence on performance. Additionally, in case temperature correction is required later, the environmental temperature at which the rubber fender is placed should be recorded throughout the test.

2) Temperature correction as an alternative to temperature control:

In the case of large-sized fenders, it is difficult to maintain the fender at the target temperature ±5°C in advance; furthermore, when the test time including the preliminary compression after taking the sample out of the temperature-controlled room becomes 2 h or more, the temperature is corrected using the average value of the recorded environmental temperature. The correction is performed by multiplying the temperature factor with respect to the time average of the recorded environmental temperature (equation (6.3.1)) with the target temperature. Even in such cases, it is desirable that the test environment is within the range of the target temperature ±15°C. Even when the test room itself is temperature-controlled or when the environmental temperature change is within the target temperature ±5°C, it is recommended that the environmental temperature at which the rubber fender is placed is recorded throughout the test.

(2) Preliminary compression

Compression is performed three or more times depending on the manufacturer's recommendation up to the design deflection or more. Although methods to consider the compression residual strain, compression rate, compression interval, etc. are not specified, it is desirable to unify these concepts. It is recommended that the number of compressions and performance values are recorded.

(3) Main compression (Static)

The sample is maintained at the target temperature ±5°C for 1 h or more after the preliminary compression, and compression is performed one more time to the design deflection or more; the corresponding performance is the static compression performance. To eliminate the residual strain of pre-compression, the initial deflection point at which the reaction force increases is the zero deflection. The compression rate is within the range of standard strain rate (0.01 to 0.3%/s or 0.3 to 1.3 mm/s), and if the compression rate in this range cannot be determined due to size restrictions, etc., the results are corrected considering the velocity factor. In accordance with the capability of the testing facility, it is possible to realize decelerating compression DV from a high initial velocity and determine the performance, including the influence of velocity. The corresponding performance is called the decelerating compression performance, and it is different from the static compression performance. In the case of velocity reduction compression, the initial speed and reduced speeds are also recorded. In addition, when storage cannot be performed at the target temperature ±5°C, the result is temperature-corrected using the average value of the recorded environmental temperatures.

(4) Open leg V-type rubber fenders are fixed to the base plate of the machine so that the legs do not open.

(5) Energy absorption is obtained by integrating the reaction force with respect to the deflection up to the maximum deflection. The integration can be numerical, such as trapezoidal integration, and the step size of the deflection is within 5%.

6.3.2 Angular compression test

For the test of angular dependence, a real-size fender or scale model can be used. As shown in Fig. 6.3.1, the inclination angle θ is the angle between the rubber fender top surface and compression plate of the tester, and the compression direction is the axial direction of the rubber fender.

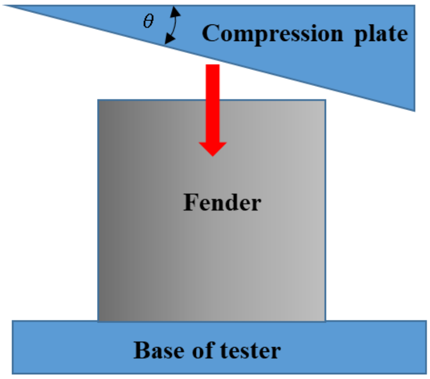


Fig. 6.3.1 Angular compression test

(1) Specimen

When using a scale model, a height of 100 mm or more should be considered.

(2) Temperature stabilization

The specimen is thermally stabilized to a standard temperature of 23±5°C. The stabilizing time pertains to the number of days equal to or greater than that obtained using equation (6.3.1). When the temperature cannot be maintained constant, the temperature history of the stabilization time may be recorded, and the performance may be corrected using the temperature factor.

(3) Preliminary compression

Compression should be performed at least 3 times up to the design deflection or more at an angle of 0± 1°. Methods to consider the residual deflection, compression speed, compression interval, etc. are not specified; however, it is desirable to unify these concepts. It is also recommended that the number of times and reaction force data are recorded.

(4) Thermal stabilization

After preliminary compression, the sample is thermally stabilized at a standard temperature of 23±5°C for 1 h or more.

(5) Main compression

Main compression is performed once to the design deflection or more at an inclination angle θ°± 1°. The compression rate is in the range of strain rate of 0.01 to 0.3%/s or rate of 0.3 to 1.3 mm/s.

(6) In angular compression testing, although the angular test of one specimen for one type of angle is ideal, when using the same specimen repeatedly at different angles, countermeasures to minimize the influence of the history should be used. For example, when the specimen is rotated 180° after the angular test, the reaction force tends to increase due to the history, and when the same specimen is tested at the same angle, the reaction force decreases. To this end, one can use a reference model having the same history as a comparison or eliminate the

influence of the history by averaging and/or maintaining the specimen at the reference temperature of 23±5°C for 2 h or more for recovery. It is desirable to adopt these measures to reduce the effects of history and record the procedures performed.

(7) The ratio of performance at angle 0° to the performance obtained during compression at the set angle is determined. The ratio of the reaction force at the design deflection or the ratio of the maximum reaction force until then is referred to as the angular factor. The ratio of the reaction force to deflection of 5% is referred to as the deflection-specific angular factor.

(8) In angular compression, when the centre of the rubber fender is compressed to the design deflection, the end may exceed the compression limit and part of the body may be damaged. The allowable deflection is set so that the body is not damaged, and the face plates of the tester are not in contact with each other.

(9) V-type rubber fenders with open legs should be fixed so that the legs do not open.

(10) The compression plate of the tester should be made of steel, and water, oil and other substances that change the friction coefficient must be removed from the surface.

(11) The energy absorption is obtained by integrating the reaction force with respect to the deflection up to the maximum deflection. The integration can be numerical, such as trapezoidal integration, and the step size of the deflection is within 5%.

(12) Definition of angular factor

The angular factor is obtained from equation (6.3.2), assuming R_θ (θ: angle) is the reaction force at angle θ.

$$\left.\begin{array}{lll}\text{Angular factor of reaction force} & : & C_{aR} = R_\theta \ / \ R_R \\ \text{Angular factor of energy absorption} & : & C_{aE} = E_{A\theta} \ / \ E_A\end{array}\right\} \qquad (6.3.2)$$

Here,

C_{aR} : Angular factor of reaction force
R_θ : Reaction force at angle θ°
R_R : Standard reaction force (θ=0°)
C_{aE} : Angular factor of energy absorption
$E_{A\theta}$: Energy absorption at angle θ°
E_A : Standard energy absorption (θ=0°)

(13) Example of handling angular test data

The test data processing method for the angular test is described by taking an example of a scale model of a vertical cylindrical rubber fender. Fig. 6.3.2 shows the result data.

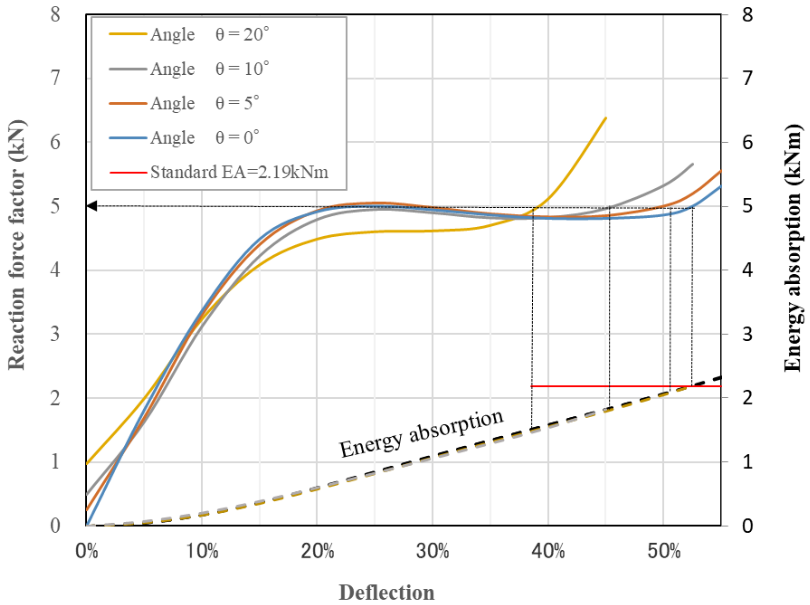


Fig. 6.3.2 Example data of angular test result

As explained in Fig. 4.2.4 in Chapter 4, Section 4.2.2, the example test data corresponds to that after origin adjustment, in which the virtual displacement δ_1 is obtained by neglecting the data before the full contact at low deflection. In the catalogue performance, the reaction force is also zero at a deflection of 0%, but the value is retained as in Fig. 6.3.2. The data processing is summarized in Table 6.3.1.

Table 6.3.1 Example of angular test data processing

	Angle θ=0°			Angle θ=5°				Angle θ=10°				Angle θ=20°			
Deflection ε	Reaction force Data (kN) ③	Reaction force (non-dimension) ④=③／①	Energy absorption factor ⑤	Reaction force Data (kN) ⑥	Angular factor (Reaction force) ⑦=⑥/①	Energy absorption factor ⑧	Angular factor (Energy absorption) ⑨=⑧/②	Reaction force Data (kN) ⑩	Angular factor (Reaction force) ⑪=⑩/①	Energy absorption factor ⑫	Angular factor (Energy absorption) ⑬=⑫/②	Reaction force Data (kN) ⑭	Angular factor (Reaction force) ⑮=⑫/①	Energy absorption factor ⑯	Angular factor (Energy absorption) ⑰=⑮/②
0.0%	0	0	0	0.25	0.05	0	0.00	0.49	0.10	0	0.00	0.97	0.19	0	0.00
5.0%	1.82	0.36	0.01	1.69	0.34	0.01	0.02	1.63	0.33	0.01	0.02	2.00	0.40	0.01	0.03
10.0%	3.37	0.67	0.03	3.30	0.66	0.03	0.08	3.12	0.62	0.03	0.08	3.24	0.65	0.04	0.09
15.0%	4.49	0.90	0.07	4.40	0.88	0.07	0.17	4.23	0.85	0.07	0.16	4.09	0.82	0.08	0.18
20.0%	4.91	0.98	0.12	4.93	0.99	0.12	0.27	4.80	0.96	0.12	0.26	4.49	0.90	0.12	0.27
25.0%	5.00	1.00	0.17	5.05	1.01	0.17	0.39	4.95	0.99	0.16	0.38	4.60	0.92	0.17	0.38
30.0%	4.94	0.99	0.22	4.98	1.00	0.22	0.50	4.90	0.98	0.21	0.49	4.62	0.92	0.21	0.48
35.0%	4.86	0.97	0.27	4.88	0.98	0.27	0.61	4.82	0.96	0.26	0.60	4.70	0.94	0.26	0.59
40.0%	4.81	0.96	0.32	4.83	0.97	0.32	0.72	4.82	0.96	0.31	0.71	5.12	1.02	0.31	0.70
45.0%	4.81	0.96	0.37	4.85	0.97	0.37	0.83	4.97	0.99	0.36	0.82	6.38	1.28	0.37	0.83
50.0%	4.86	0.97	0.41	5.00	1.00	0.42	0.95	5.33	1.07	0.41	0.94				
52.5%	**5.00**	1.00	**0.44**	5.20	1.04	0.44	1.00	5.66	1.13	0.44	1.00				
55.0%	5.32	1.06	0.46	5.56	1.11	0.47	1.07								

Standard reaction force R_R ①=	**5.00 kN**	Deflection at E_A absorbed	50.6%	Angular factor of R Ca_R=	**1.01**	Deflection at E_A absorbed	45.5%	Angular factor of R Ca_R=	**1.00**	Deflection at E_A absorbed	38.6%	Angular factor of R Ca_R=	**1.00**
Standard energy absorption factor E_A②=	**0.44**	Maximum reaction force factor	**1.01**	Angular factor of E Ca_E=	**0.96**	Maximum reaction force factor	**1.00**	Angular factor of E Ca_E=	**0.83**	Maximum reaction force factor	**1.00**	Angular factor of E Ca_E=	**0.67**

As shown in Chapter 4, Section 4.2.2, it is possible to consider the deflection at the fender panel end and create Table 6.3.1 to determine the performance including the initial rotation (yellow part in Fig. 4.2.5 and Fig. 4.2.6). In Table 6.3.1, the standard reaction force at an angle of 0° is 5.00 kN ①. The reaction force data (③, ⑥, ⑩, ⑭) is divided by the standard reaction force ① to obtain a dimensionless reaction force (④, ⑦, ⑪, ⑮), which is integrated to obtain the dimensionless energy absorption (⑤, ⑧, ⑫, ⑯). The same process is performed for angles of 5°, 10° and 20°. At an angle of 5°, the maximum reaction force at a deflection of 25% increases by approximately 0.01. At an angle of 5°, 50.6% of the reaction force at a deflection of 25% becomes the same value in the high deflection region, and it becomes 0.96 when the non-dimensional energy absorption up to this deflection is obtained using linear interpolation. These values become the angular factors C_{aR} and C_{aE} of the reaction force and the energy absorption at an angle of 5°. At an angle of 10°, the reaction force at the deflection of 45.5% at which the standard reaction force is absorbed becomes larger than the maximum reaction force at the strain of 25%; thus, the energy absorption factor (0.83) up to this deflection becomes the angular factor C_{aE}. Similarly, at an angle of 20°, the maximum reaction force (1.00) generates a deflection of 38.6%, and the energy absorption until this point is 0.67. If other angles are similarly processed, the angular factors obtained are as listed in Table 6.3.2.1. If the factor variation within 3% is common, its values can be listed as in Table 6.3.2.2; thus, if the angle is 4° or less, the influence of compression angle can be ignored. It is desirable to present either Table 6.3.2.1 or Table 6.3.2.2 in the catalogue of rubber fenders. However, since the reaction force curve is not similar to that of the angles, the factors of reaction force and energy absorption can be shared only for angles of 4° or less.

Table 6.3.2.1 Example of angular factor

Angles (°)	Angular factor of reaction force Ca_R	Angular factor of energy absorption Ca_E
0	1.00	1.00
3	1.02	1.01
4	1.02	0.99
5	1.01	0.96
6	1.01	0.95
10	1.00	0.83
15	1.00	0.74
20	1.00	0.67

Table 6.3.2.2 Example of angular factor (Simplified)

<table>
<tr><th>Angles (°)</th><th>Angular factor of reaction force Ca_R</th><th>Angular factor of energy absorption Ca_E</th></tr>
<tr><td>0</td><td colspan="2" rowspan="3">1.00</td></tr>
<tr><td>3</td></tr>
<tr><td>4</td></tr>
<tr><td>5</td><td rowspan="5">1.00</td><td>0.96</td></tr>
<tr><td>6</td><td>0.95</td></tr>
<tr><td>10</td><td>0.83</td></tr>
<tr><td>15</td><td>0.74</td></tr>
<tr><td>20</td><td>0.67</td></tr>
</table>

(14) Shear compression

When performing shear compression, as described in Chapter 4, Section 4.2.2, a tester capable of measuring the displacement and load in two axial directions as shown in Fig. 6.3.3.1 is required; however, as shown in Fig. 6.3.3.2, compression is possible using only a single axial tester if an angular jig is used.

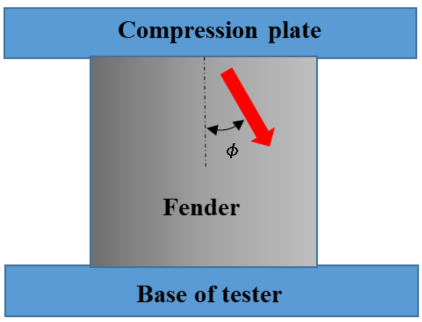


Fig. 6.3.3.1 Shear compression test (2-axis)

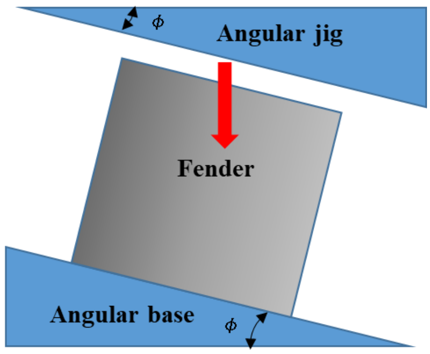


Fig. 6.3.3.2 Shear compression test (1-axis with jig)

(15) Angular performance of V-type rubber fender

V-type rubber fenders can also be subjected to the angular compression test by employing the abovementioned method; however, two angular directions are present: in the width and longitudinal directions. The value for the longitudinal direction can be obtained by performing calculations using equation (5.6.7) in Chapter 5, Section 5.6.2.

6.3.3 Velocity dependency test

Two types of velocity dependent tests exist: CV constant velocity compression that involves compression at a constant velocity, and DV deceleration compression, at which the velocity becomes zero (5 mm/s at PIANC Guidelines

[2] at the design deflection. Although it is ideal to perform DV deceleration compression directly, the relationship between the actual berthing velocity and fender height is diverse; therefore, after performing CV tests under various strain rates, the DV deceleration performance can be calculated based on the CV results. This section describes the CV test method, data processing, and calculation procedure to obtain the DV performance.

(1) Specimen

When using a scale model, a height of 100 mm or more should be employed.

(2) Temperature stabilization

The specimen is thermally stabilized to a standard temperature of 23±5°C. The stabilizing time corresponds to the number of days equal to or greater than that specified in equation (6.3.1).

(3) Preliminary compression

Compression should be performed at least 3 times up to the design deflection or more. The consideration of residual deflection, compression speed, compression interval, etc. is not specified; however, it is desirable to unify these concepts. It is also recommended that the number of times and reaction force data is recorded.

(4) Thermal stabilization

After preliminary compression, the specimen is thermally stabilized at a standard temperature of 23±5°C for 1 h or more.

(5) Main compression

Main compression is performed once to the design deflection or more. The strain rate is in the range of 0.01 to 50%/s for at least three different strain rates.

(6) When the same specimen is used repeatedly, it is necessary to adopt measures to minimize the influence of the compression history. For example, the specimen can be maintained at a standard temperature of 23 ± 5°C. for a time of 2 h or more. Alternatively, the same historical reference model can be used as a comparison reference. As an alternative, the influence of history can be minimized by averaging. In all these cases, it is advisable to record the procedures employed.

(7) The ratio between the performance at standard (static) compression speed to that obtained during compression at the target speed is determined. The ratio of the design reaction force or the maximum reaction force up to that deflection is used as a velocity factor, and the ratio of the reaction force for every 5% of the deflection is known as the deflection-specific velocity factor.

(8) The energy absorption is obtained by integrating the reaction force with respect to the deflection up to the design deflection. The integration can be numerical such as trapezoidal integration, and the step size of deflection is within 5%. An example of the test conditions is presented in Table 6.3.3.

Table 6.3.3 Example of velocity test conditions

	Items	Conditions	Remarks
Specimen	Type of fender	V	
	Performance grade（Rubber）	3 grades	Highest, Middle and Lowest
	Number of specimens per grade	More than one/Grade	Specimen:1 and reference:1 piece
	Size (height ×length)	Larger than 100mm × 100mm	Consider ends effect
Preliminary compression	Number of compressions	More than 3 times	
	Final deflection（Rated deflection+α）	More than rated deflection	
	Stabilizing temperature before test	23℃	
	Tolerance of stabilizing temperature before test	±5℃	
	Stabilizing temperature during test	23℃	
	Tolerance of stabilizing temperature during test	±5℃	
Main compression	Number of compressions	1 or more	
	Temperature during compression	23±5℃	
	Final deflection（Rated deflection+α）	More than rated deflection	
	Stabilizing temperature before test	23℃	
	Tolerance of stabilizing temperature before test	±5℃	
	Tolerance of ambient temperature during test	±5℃	
	Interval of compression	Constant interval	
	Temperature of storage after test	23±5℃	
Measures to negate history effect	When one specimen is used in multiple velocity conditions, use reference model of same history of standard velocity or averaged results of different test order to minimize history influence.		

(9) Example of handling of constant velocity (CV) test results

The velocity factor in the CV test is defined as in equation (6.3.3).

$$\left.\begin{array}{ll}\text{Velocity factor of reaction force} & : \ VF_R = R_V / R_R \\ \text{Velocity factor of energy absorption} & : \ VF_E = E_{AV} / E_A\end{array}\right\} \quad (6.3.3)$$

Here,

VF_R : Velocity factor of reaction force

R_V : Design reaction force at constant velocity V (CV)

R_R : Standard reaction force at standard (static) velocity

VF_E : Velocity factor of energy absorption

E_{AV} : Design energy absorption at constant velocity V (CV)

E_A : Standard energy absorption at standard (static) velocity

The results of constant velocity (CV) test of the scale model of a vertical cylinder rubber fender are presented as follows as an example of the data processing method. Fig. 6.3.4 shows an example of the result data.

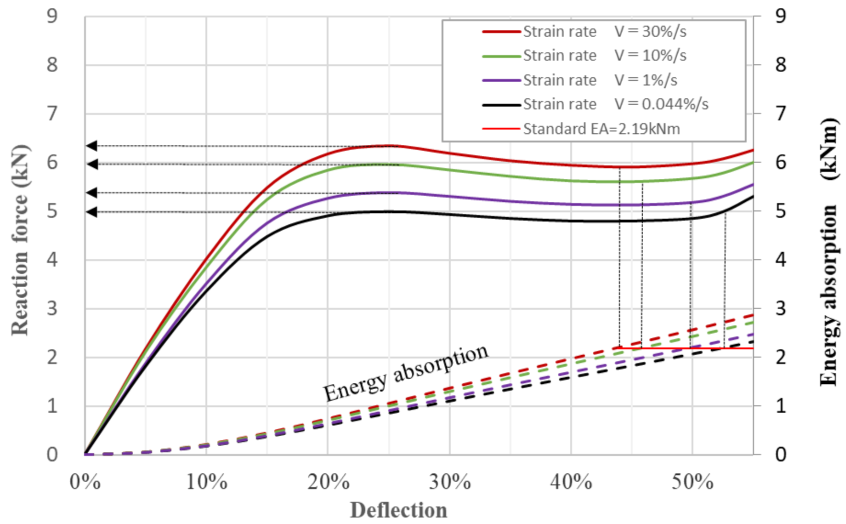


Fig. 6.3.4 Example data of constant velocity (CV) test result

The test results and calculation of each velocity factor is explained in Table 6.3.4.

Table 6.3.4 Example of constant velocity (CV) test data processing

	Strain rate V=0.044%/s			Strain rate V=30%/s					Strain rate V=10%/s					Strain rate V=1%/s				
Deflection ε	Reaction force Data (kN) ③	Reaction force (non-dimension) ④=③／①	Energy absorption factor ⑤	Reaction force Data (kN) ⑥	Velocity factor (Reaction force) ⑦=⑥/①	Energy absorption factor ⑧	Velocity factor of R at each deflection ⑨=⑦/④	Velocity factor of E_A at each deflection ⑩=⑧/⑤	Reaction force Data (kN) ⑪	Velocity factor (Reaction force) ⑫=⑪/①	Energy absorption factor ⑬	Velocity factor of R at each deflection ⑭=⑫/④	Velocity factor of E_A at each deflection ⑮=⑬/⑤	Reaction force Data (kN) ⑯	Velocity factor (Reaction force) ⑰=⑯/①	Energy absorption factor ⑱	Velocity factor of R at each deflection ⑲=⑰/④	Velocity factor of E_A at each deflection ⑳=⑱/⑤
0.0%	0	0	0	0.00	0	0	1.00	1.00	0.00	0.00	0	1.00	1.00	0.00	0.00	0	1.00	1.00
5.0%	1.82	0.36	0.01	2.17	0.43	0.01	1.19	1.19	2.11	0.42	0.01	1.16	1.16	1.87	0.37	0.01	1.03	1.03
10.0%	3.37	0.67	0.03	4.02	0.80	0.04	1.19	1.19	3.85	0.77	0.04	1.14	1.15	3.52	0.70	0.04	1.05	1.04
15.0%	4.49	0.90	0.07	5.48	1.10	0.09	1.22	1.20	5.24	1.05	0.09	1.17	1.15	4.76	0.95	0.08	1.06	1.05
20.0%	4.91	0.98	0.12	6.17	1.23	0.15	1.26	1.22	5.84	1.17	0.14	1.19	1.16	5.27	1.05	0.13	1.07	1.05
25.0%	5.00	1.00	0.17	6.33	**1.27**	0.21	1.27	1.23	5.96	**1.19**	0.20	1.19	1.17	5.39	**1.08**	0.18	1.08	1.06
30.0%	4.94	0.99	0.22	6.18	1.24	0.27	1.25	1.24	5.85	1.17	0.26	1.18	1.17	5.31	1.06	0.23	1.08	1.06
35.0%	4.86	0.97	0.27	6.03	1.21	0.33	1.24	1.24	5.71	1.14	0.32	1.18	1.18	5.21	1.04	0.29	1.07	1.07
40.0%	4.81	0.96	0.32	5.94	1.19	0.39	1.24	1.24	5.62	1.12	0.37	1.17	1.18	5.15	1.03	0.34	1.07	1.07
45.0%	4.81	0.96	0.37	5.90	1.18	0.45	1.23	1.24	5.60	1.12	0.43	1.17	1.17	5.14	1.03	0.39	1.07	1.07
50.0%	4.86	0.97	0.41	5.97	1.19	0.51	1.23	1.24	5.67	1.13	0.49	1.17	1.17	5.19	1.04	0.44	1.07	1.07
52.5%	**5.00**	1.00	**0.44**	6.07	1.21	0.54	1.21	**1.23**	5.78	1.16	0.51	1.16	**1.17**	5.32	1.06	0.47	1.06	**1.07**
55.0%	5.32	1.06	0.46	6.25	1.25	0.57	1.18	1.23	6.00	1.20	0.54	1.13	1.17	5.56	1.11	0.50	1.05	1.07

Standard reaction force R_R ①=	**5.00 kN**	Deflection at E_A absorbed	43.85%	Velocity factor of R VF_R=	**1.27**	Deflection at E_A absorbed	45.82%	Velocity factor of R VF_R=	**1.19**	Deflection at E_A absorbed	49.68%	Velocity factor of R VF_R=	**1.08**
Standard energy absorption factor E_A②=	**0.44**	Maximum reaction force factor at E_A absorbed	**1.27**	Velocity factor of E_A VF_E=	**1.23**	Maximum reaction force factor at E_A absorbed	**1.19**	Velocity factor of E_A VF_E=	**1.17**	Maximum reaction force factor at E_A absorbed	**1.08**	Velocity factor of E_A VF_E=	**1.07**

In the data presented in Table 6.3.4, the standard reaction force at standard speed (0.044%/s) is 5.00 kN (①). The reaction force data is divided by the standard reaction force to obtain a dimensionless reaction force (④), which is integrated to obtain a dimensionless standard energy absorption value (②). Table 6.3.4 presents the same processing at compression rates of 30%/s, 10%/s and 1%/s. At a compression rate of 30%/s, a non-dimensional standard energy absorption of 0.44 kN·m can be realized at a deflection of 43.85%. The maximum reaction force until then is the peak reaction force at a deflection of 25%. For this fender, the maximum reaction force is a peak value at a deflection of 25% at any speed, and the buckling type characteristics are maintained. Since the standard reaction force R_R is non-dimensional, the maximum reaction force at each velocity becomes the velocity factor VF_R, and the energy absorption up to the deflection at which the same reaction force as the peak value is generated in the high deflection region is VF_E. In Fig. 6.3.5, the velocity factor of the reaction force and energy absorption are plotted in relation to the strain rate. Note that the strain rate is logarithmic.

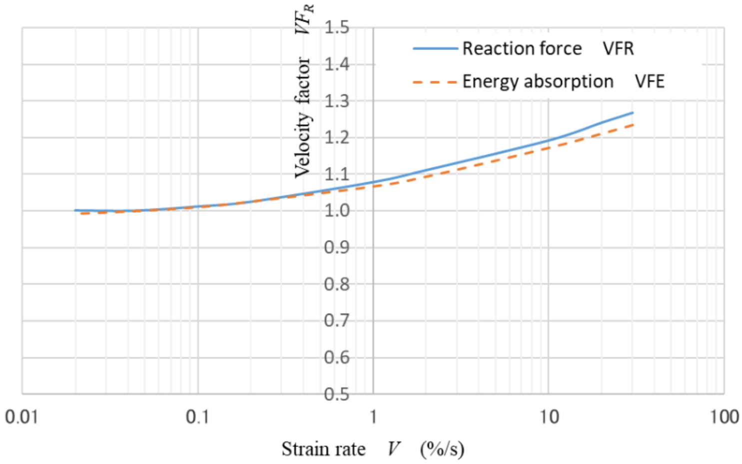


Fig. 6.3.5 Example of velocity factor and strain rate (CV)

(10) Example of processing constant velocity (CV) results into decelerating velocity (DV) factor

Since the velocity factor directly used for design is a factor corresponding to the decelerating velocity (DV) performance, it is calculated as follows using the factors of the CV test described above. The conceptual diagram is shown in Chapter 4, Fig. 4.2.10, and a concrete calculation example is demonstrated here.

In deceleration compression (DV), as the deflection progresses from the initial velocity, the velocity decelerates and becomes close to zero at the design deflection. The PIANC Guidelines [2)] proposes linear deceleration, cosine waveform, etc. as the modes of deceleration. Deceleration by a cosine waveform is a mode in which 0% of the deflection is considered as 1.0 in equation (6.3.4), and a coefficient of cosine that becomes $\pi/2$ in the design deflection is calculated under the initial velocity.

$$V = V_B \cdot \cos(\pi\delta/2D) \tag{6.3.4}$$

Here,

V : Strain rate (%/s)

V_B : Initial strain rate (＝Berthing velocity / Fender height) (%/s)

δ : Deflection (%)

D : Design deflection of fender (%)

As an example, the decelerating performance for each deflection of the rubber fender with a standard reaction force R_R = 2327 (kN) at a height of 2 m (2000 H) is presented in Table 6.3.5. Table 6.3.5 indicates the performance at cosine deceleration. The velocity factor VF_R used in this case is a coefficient of the CV performance demonstrated in Fig. 6.3.5. From Table 6.3.5, the velocity factors of the deceleration performance can be calculated as follows.

Initial berthing velocity	:	V_B ＝7.5 (%/s)＝0.15 (m/s)
Velocity factor of reaction force at decelerating performance (DV)	:	VF_R＝2670／2327＝1.15
Velocity factor of energy absorption at decelerating performance (DV)	:	VF_E＝2315／2043＝1.13

Since the initial velocity is 0.15 m/s, the design reaction force is 2670 kN, and the design energy absorption is 2315 kN·m. In the above example, the difference between the velocity factor of the reaction force and absorbed energy is approximately 2% because, as seen in Fig. 6.3.4, the buckling type characteristic is maintained with respect to the change in velocity, and the reaction force curve has a substantially similar shape.

Table 6.3.5 Example of processing CV results into DV performance

Deflection δ (%)	Displacement ε(m)	Velocity (m/s)	Strain rate V (%/s)	Velocity factor of CV VF_R	Performance by deflection			
					Standard reaction force (kN)	Standard energy absorbtion (kN×m)	Reaction force of DV $R_V(\varepsilon)=$	Energy absorption of DV (kN×m)
	$\delta \times H$	equation(6.3.4)		Fig.6.3.5	$R_R(\varepsilon)$	E_A	$R_R(\varepsilon)\times VF_R$	$=\int R_V(\varepsilon)d\varepsilon$
0	0	0.150	7.50	1.17	0	0	0	0
5	0.1	0.148	7.42	1.17	845	42	985	49
10	0.2	0.144	7.20	1.16	1566	163	1822	190
15	0.3	0.136	6.82	1.16	2088	346	2421	402
20	0.4	0.126	6.31	1.15	2287	564	2639	655
25	0.5	0.113	5.67	1.15	**2327**	795	**2670**	920
30	0.6	0.098	4.91	1.14	2300	1026	2621	1185
35	0.7	0.081	4.05	1.13	2262	1254	2559	1444
40	0.8	0.062	3.12	1.12	2237	1479	2510	1697
45	0.9	0.042	2.11	1.11	2237	1703	2487	1947
50	1	0.021	1.07	1.08	2262	1928	2443	2194
52.5	1.05	0.011	0.54	1.05	**2327**	**2043**	2435	**2315**
55	1.1	0.000	0.00	1.00	2473	2163	2473	2438

A size of 2000H was assumed in the above example; however, since the velocity factors are based on the strain rate, the values do not depend on the size of fenders. By changing the initial strain rate and performing the same processes with other performance grades, the velocity factor table is similar to Table 6.3.6.

Table 6.3.6 Example of DV velocity factor for cosine deceleration
(catalogue display, before simplification)

Performance grade	Grade A		Grade B		Grade C	
Strain rate (%/S)	Reaction force VF_R	Energy absorption VF_E	Reaction force VF_R	Energy absorption VF_E	Reaction force VF_R	Energy absorption VF_E
30	1.23	1.18	1.21	1.18	1.20	1.17
20	1.20	1.17	1.18	1.15	1.17	1.14
10	1.17	1.14	1.15	1.13	1.14	1.12
5	1.13	1.11	1.12	1.10	1.11	1.09
1.00	1.06	1.06	1.06	1.05	1.05	1.05
0.50	1.04	1.05	1.04	1.04	1.04	1.04
0.10	1.01	1.01	1.01	1.01	1.00	1.00
0.05	1.00	1.00	1.00	1.00	1.00	1.00
0.01	1.00	1.00	1.00	1.00	1.00	1.00

In Table 6.3.7, differences within 3% are standardized. It is desirable that the catalogue displays the data as given in Table 6.3.6 or Table 6.3.7 or as a graph shown in Fig. 6.3.6. In the case of only a graph display, it is necessary to display the approximate coefficients of the formula to ensure that the numerical values used for the design can be determined.

Table 6.3.7 Example of DV velocity factor for cosine deceleration
(catalogue display, after simplification)

Performance grade	Grade A		Grade B		Grade C	
Strain rate (%/S)	Reaction force VF_R	Energy absorption VF_E	Reaction force VF_R	Energy absorption VF_E	Reaction force VF_R	Energy absorption VF_E
30	1.23	1.18	1.21	1.18	1.20	1.17
20	1.20	1.17	1.18	1.15	1.17	1.14
10	1.17	1.14	1.15		1.14	
5	1.13		1.10			
1.00	1.06		1.05			
0.50	1.04					
0.10	1.00					
0.05						
0.01						

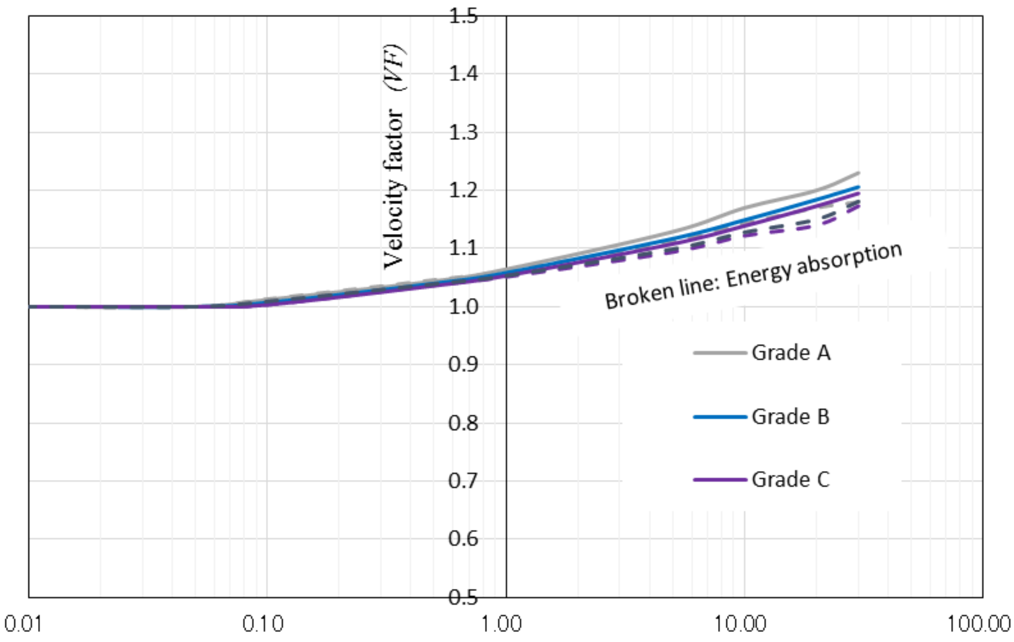


Fig. 6.3.6 Sample graph of DV velocity factor for cosine deceleration

(11) Mode of velocity reduction

Although the above description is based on deceleration corresponding to a cosine curve, the deceleration mode is not particularly defined and may be linear. The deceleration mode in actual berthing can be estimated as follows by obtaining the remaining amount of effective berthing energy. As another example, let the fender have a height of 1000 mm. We attempt to determine the mode of deceleration at the initial velocity of 0.15 m/s, as used in the PIANC Guidelines [2)]. Since the initial berthing velocity corresponds to 15%/s, the initial velocity factor is VF_R (V = 15%/s), and the initial deflection of the fender, for example, the reaction force at the standard compression velocity at a deflection of 5% is R_5 ($V = V_0$); the reaction force R_5 (V = 15) at a velocity of 0.15 m/s is as shown in equation (6.3.5).

$$R_5(V{=}15) = R_5(V{=}V_0) \times VF_R\,(V{=}15) \qquad (6.3.5)$$

The energy ΔE absorbed up to a strain of 5% is the area of a triangle surrounded by the reaction force of a strain rate of 0% and 5% and the horizontal axis, and it can be expressed as in equation (6.3.6).

$$\Delta E = R_5(V{=}15)\times\varepsilon(=0.05\ \text{m}) / 2 \qquad (6.3.6)$$

The effective berthing energy E_b can be calculated using equation (5.3.1) in Chapter 5, that is

$$E_b = \frac{M}{2}\cdot V_B{}^2 \cdot C_e \cdot C_m \cdot C_c \cdot C_s \qquad (5.3.1)$$

Here,

E_b : Effective berthing energy （kN·m=kJ）
M ： Mass of vessel (＝Displacement tonnage: *DT*, *t*)
V_B ： Berthing velocity (（m/s)
C_e ： Eccentricity factor
C_m ： Virtual mass factor
C_c ： Berth configuration factor
C_s ： Softness factor

Next, we summarize the coefficients, and define C_{total} as follows.

$$C_{total} \;=\; C_e \cdot C_m \cdot C_c \cdot C_s \qquad (6.3.7)$$

The effective berthing energy E_b can be determined using equation (6.3.8).

$$E_b = \frac{M}{2}\cdot V_B{}^2 \cdot C_{total} \qquad (6.3.8)$$

The compression speed is reduced from V_0 to V_5, as shown in equation (6.3.9) since ΔE is absorbed at a deflection of 5%.

$$\frac{V_5}{V_0} = \sqrt{\frac{E_b - \Delta E}{E_b}} \qquad (6.3.9)$$

If the velocity factor VF_R at the compression velocity V_{10} is determined and the same calculations are repeated with increased deflection, the deceleration performance (DV) by energy absorption can be obtained. It is also possible to compress the scale model fender by assuming the mode of deceleration to be a cosine waveform or a straight line to obtain the deceleration performance; however, it is expected that in actual berthing, the mode of deceleration is as shown in equation (6.3.9). Fig. 6.3.7 compares the linear deceleration, cosine waveform deceleration, and energy absorption deceleration according to equation (6.3.9). The decelerating velocity due to energy absorption is similar to that for the cosine waveform at low deflection; however, it increases at high deflection. Additionally, if the velocity factor VF_R is expressed in terms of the deflection and the factor at each velocity and each deflection is used, it is possible to obtain the DV performance in each deceleration mode. In this manner, the decelerating performance calculated using the CV performance and the test results of actual compression under a cosine waveform from the initial speed of 0.15 m/s can be determined, as shown in Fig. 6.3.8. In all cases, the maximum value of reaction force increases by approximately 20% at an initial velocity of 0.15 m/s (here, 15%/s) compared with the static performance. The difference among the deceleration methods and that between the test value and the calculated value appear in the high deflection area; however, the differences are small. The reason the test value is higher in the high deflection region is the deference of stress relaxation of the rubber due to the difference between the compression time at a constant velocity and reduced time owing to deceleration from a large initial velocity. Although the energy absorption is slightly larger at the test value than at the calculated value, the calculation method involving the CV performance is considered to ensure safety in fender design. To summarize, the influence of velocity on the reaction force is large (approximately 20%) and needs to be considered in fender design; however, since the influence of the mode of deceleration is small, it does not lead to a considerable difference in the design.

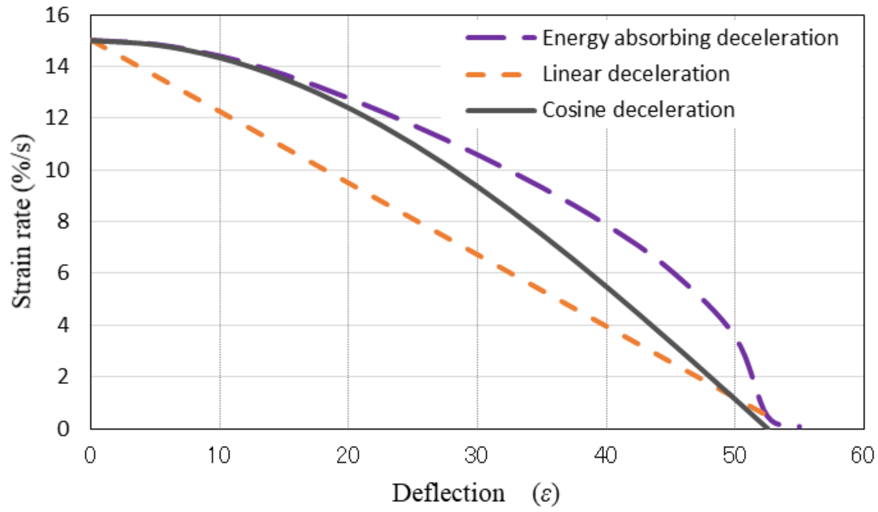


Fig. 6.3.7 Deceleration modes of decreasing velocity (DV)

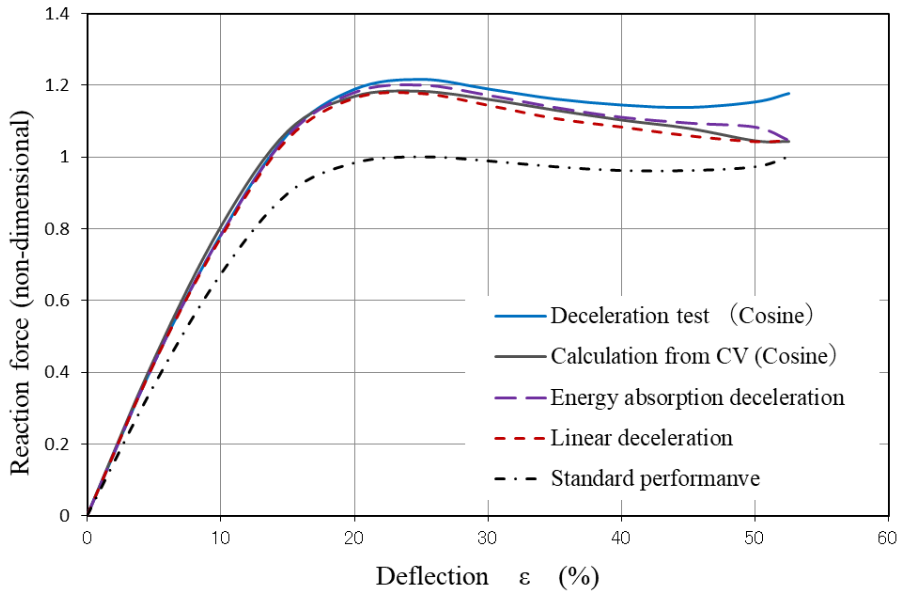


Fig. 6.3.8 Performances of decelerating test and calculation (DV)

6.3.4 Temperature dependency test

The test of temperature dependency involves measuring the temperature dependency of the elastic modulus of the material and directly measuring the temperature dependency of rubber fenders (real product or scale model). Because testing rubber fenders is time consuming and costly, it is acceptable to perform testing at only representative temperatures (e.g., -30°C, 0°C, 23°C, 50°C) and estimate the factors at other intermediate temperature by interpolating the material test results, as described later.

(1) Temperature dependency test of rubber fenders

1) Specimen

When using a scale model, a height of 100 mm or more should be employed.

2) Temperature stabilization

The specimen is thermally stabilized to a target temperature ±5°C. The stabilizing time corresponds to the number of days equal to or greater than that specified in equation (6.3.1).

3) The temperature range is measured every 10°C from -30°C to 50°C, and a standard temperature of 23°C is used instead of 20°C.

4) Preliminary compression

Compression is performed at least 3 times up to the design deflection or more. The consideration of residual deflection, compression speed, compression interval, etc. is not specified; however, it is desirable to unify the concepts. It is also recommended that the number of times and reaction force data are recorded.

5) Thermal stabilization

After preliminary compression, the specimen is thermally stabilized at a target temperature ±5°C for 1 h or more.

6) Main compression

Main compression is performed once to the design deflection or more. The compression rate should be set in the range of strain rate of 0.01 to 0.3%/s or speed of 0.3 to 1.3 mm/s.

7) The reaction force at the design deflection at the target temperature or its maximum value (design reaction force) up to that point is divided by the reaction force at the standard temperature of 23°C, and the resulting value is known as the temperature factor. The ratio between the reaction forces for every 5% deflection to that at the standard temperature is called the deflection-specific temperature factor. When the reaction force increases monotonously and does not have a peak point, the reaction force at the time of standard energy absorption at the standard temperature of 23°C is adopted as the design reaction force.

8) If the same specimen is to be used at different temperatures, measures must be adopted to minimize the effects of the compression history. To this end, the specimen can be placed in a constant temperature room for the temperature stabilization time plus 2 hours or more; alternatively, a specimen having the same compression history can be used as a reference, or the results can be averaged to negate the influence of history.

9) The energy absorption is obtained by integrating the reaction force with respect to the deflection up to the design deflection. The integration can be numerical such as trapezoidal integration, and the step size of deflection is within 5%.

An example of the test conditions is presented in Table 6.3.8

Table 6.3.8 Example of temperature test conditions

	Items	Conditions	Remarks
Specimen	Type of fender	V	
	Performance grade (Rubber)	3 grades	Highest, Middle and Lowest
	Number of specimens per grade	More than one/Grade	Specimen:1 and reference:1 piece
	Size (height ×length)	Larger than 100mm × 100mm	Consider ends effect
Preliminary compression	Number of compressions	More than 3 times	
	Final deflection (Rated deflection+α)	More than rated deflection	
	Stabilizing temperature before test	23℃	
	Tolerance of stabilizing temperature before test	±5℃	
	Stabilizing temperature during test	23℃	
	Tolerance of stabilizing temperature during test	±5℃	
Main compression	Number of compressions	1 or more	
	Compression speed	0.01 to 0.3%/s or 0.3 to 1.3mm/s.	Standerd compression speed
	Final deflection (Rated deflection+α)	More than rated deflection	
	Temperature stabilizing time : T1	20×(Thickness)$^{1.5}$/24 or longer	equation (6.3.1)
	Recovery time after compression : T2	2 hours or longer	
	Compression interval : T=T1+T2	T=T1+T2 or longer	
	Stabilizing temperature before test	Target temperature	
	Tolerance of stabilizing temperature before test	±5℃	
	Tolerance of ambient temperature during test	±5℃	
	Temperature of storage after test	23℃±5℃	
	Tolerance of heat chamber	±3℃	
Measures to negate history effect	When one specimen is used in multiple velocity conditions, use reference model of same history of standard velocity or averaged results of different test order to minimize history influence.		

10) Temperature factor: *TF*

The temperature factor *TF* is defined as in equations (6.3.10) and (6.3.11).

Temperature factor of reaction force : $TF_R = RT／R_R$ (6.3.10)

Temperature factor of energy absorption : $TF_E = E_{AT}／E_A$ (6.3.11)

Here,

TF_R : Temperature factor of reaction force

R_T : Design reaction force at design temperature: T°C

R_R : Standard reaction force at standard temperature: 23°C

TF_E : Temperature factor of energy absorption

E_{AT} : Design energy absorption at design temperature: T°C

E_A : Standard energy absorption at standard temperature: 23°C

When TF_R (ε) is determined separately for deflection, TF_R is considered as a function of deflection ε, as in equation (6.3.12).

Temperature factor of reaction force at deflection ε% : $TF_R(\varepsilon) = R_T(\varepsilon) / R_R(\varepsilon)$ (6.3.12)

Here,

$TF_R(\varepsilon)$: Temperature factor of reaction force at deflection ε%

$R_T(\varepsilon)$: Design reaction force at design temperature at deflection ε%

$R_R(\varepsilon)$: Standard reaction force at standard temperature at deflection ε%

11) Handling example of temperature test data

The test data processing method of the temperature test is described by taking an example of a scale model of vertical cylinder rubber fenders. Fig. 6.3.9 shows the result data.

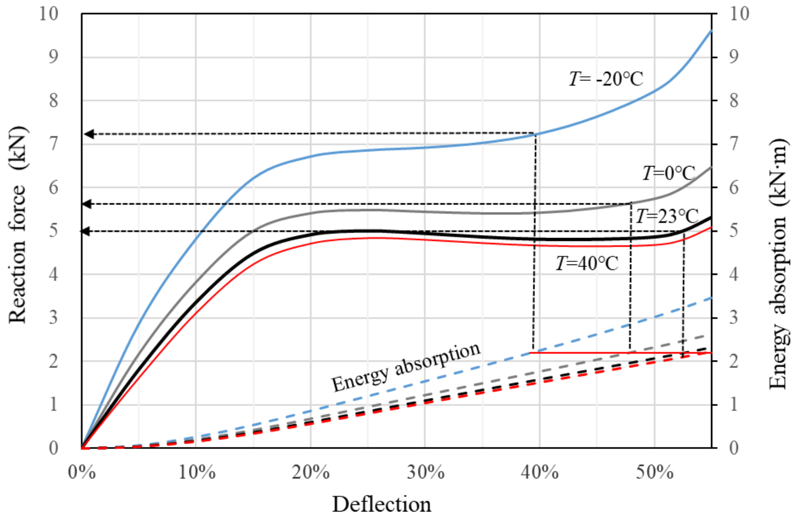


Fig. 6.3.9 Example data of temperature dependency test

The results are presented in Table 6.3.9.

Table 6.3.9 Example of processing of temperature dependency test data

	Standard temperature T=23℃			Cold temperature T= -20℃					Cool temperature T= 0℃					Hot temperature T= 40℃				
Deflection ε	Reaction force Data (kN) ③	Reaction force (non-dimension) ④=③/①	Energy absorption factor ⑤	Reaction force Data (kN) ⑥	Temperature factor (Reaction force) ⑦=⑥/①	Energy absorption factor ⑧	Temperature factor of R at each deflection ⑨=⑦/④	Temperature factor of E_A at each deflection ⑩=⑧/⑤	Reaction force Data (kN) ⑪	Temperature factor (Reaction force) ⑫=⑪/①	Energy absorption factor ⑬	Temperature factor of R at each deflection ⑭=⑫/④	Temperature factor of E_A at each deflection ⑮=⑬/⑤	Reaction force Data (kN) ⑯	Velocity factor (Reaction force) ⑰=⑯/①	Energy absorption factor ⑱	Velocity factor of R at each deflection ⑲=⑰/④	Velocity factor of E_A at each deflection ⑳=⑱/⑤
0.0%	0	0	0	0.00	0.00	0.00	1.00	1.00	0.00	0.00	0.00	1.00	1.00	0.00	0.00	0.00	1.00	1.00
5.0%	1.82	0.36	0.01	2.86	0.57	0.01	1.57	1.57	2.17	0.43	0.01	1.19	1.19	1.62	0.32	0.01	0.89	0.89
10.0%	3.37	0.67	0.03	4.82	0.96	0.05	1.43	1.51	3.83	0.77	0.04	1.14	1.17	3.12	0.62	0.03	0.93	0.91
15.0%	4.49	0.90	0.07	6.22	1.24	0.11	1.39	1.45	5.00	1.00	0.08	1.11	1.14	4.24	0.85	0.07	0.95	0.92
20.0%	4.91	0.98	0.12	6.72	1.34	0.17	1.37	1.42	5.41	1.08	0.14	1.10	1.13	4.71	0.94	0.11	0.96	0.94
25.0%	5.00	1.00	0.17	6.87	1.37	0.24	1.37	1.41	5.48	1.10	0.19	1.10	1.12	4.84	0.97	0.16	0.97	0.94
30.0%	4.94	0.99	0.22	6.93	1.39	0.31	1.40	1.40	5.45	1.09	0.25	1.10	1.12	4.80	0.96	0.21	0.97	0.95
35.0%	4.86	0.97	0.27	7.04	1.41	0.38	1.45	1.41	5.41	1.08	0.30	1.11	1.11	4.72	0.94	0.26	0.97	0.95
40.0%	4.81	0.96	0.32	7.25	1.45	0.45	1.51	1.42	5.42	1.08	0.35	1.13	1.12	4.67	0.93	0.30	0.97	0.96
45.0%	4.81	0.96	0.37	7.64	1.53	0.53	1.59	1.44	5.53	1.11	0.41	1.15	1.12	4.65	0.93	0.35	0.97	0.96
50.0%	4.86	0.97	0.41	8.22	1.64	0.60	1.69	1.46	5.74	1.15	0.47	1.18	1.12	4.68	0.94	0.40	0.96	0.96
52.5%	**5.00**	1.00	**0.44**	8.76	1.75	0.65	1.75	**1.47**	6.00	1.20	0.50	1.20	**1.13**	4.80	0.96	0.42	0.96	**0.96**
55.0%	5.32	1.06	0.46	9.62	1.92	0.69	1.81	1.49	6.47	1.29	0.53	1.22	1.13	5.09	1.02	0.45	0.96	1.02

Standard reaction force R_R ①=	**5.00 kN**	Deflection at E_A absorbed	39.17%	Temperature factor of R TF_R=	**1.44**	Deflection at E_A absorbed	48.00%	Temperature factor of R TF_R=	**1.13**	Deflection at E_A absorbed	54.33%	Temperature factor of R TF_R=	**1.00**
Standard energy absorption factor E_A②=	**0.44**	Maximum reaction force factor at E_A absorbed	**1.44**	Temperature factor of E_A TF_E=	**1.47** (=1.00 at E_A is absorbed)	Maximum reaction force factor at E_A absorbed	**1.13**	Temperature factor of E_A TF_E=	**1.13** (=1.00 at E_A is absorbed)	Maximum reaction force factor at E_A absorbed	**1.00**	Temperature factor of E_A TF_E=	**0.96**

In Table 6.3.9, similar to the test results of other influencing factors, a dimensionless performance is calculated by dividing the reaction force data by the standard reaction force of 5.00 kN (①) at a deflection of 25% under the standard temperature of 23°C. The value is integrated with respect to the deflection to determine the non-dimensional energy absorption (②). The same treatment is performed at temperatures of -20°C, 0°C and 40°C. However, at a temperature of −20°C, the reaction force does not have a local peak value and increases monotonically; therefore, the reaction force at the deflection at which standard energy absorption occurs (②) is determined via linear interpolation (TF_R =1.44). Note that the energy absorption has a design deflection value of 52.5% (TF_E =1.47), which means that it is 47% larger than the standard energy absorption, while the reaction force is 75% larger than the standard value. Therefore, at extremely low temperatures, although the fender performance is higher than that in the design condition, it is not used due to it exceeding the required value. Furthermore, at a temperature of 40°C, the energy absorption at the design deflection decreases to TF_E = 0.96. Therefore, one should be careful as the temperature affects the size selection of the fender.

If the other temperatures are similarly organized, they can be presented as in Table 6.3.10.1. As with other factors, the factors that can be shared within 3% are as given in Table 6.3.10.2. In the range of 10°C to 30°C, the difference due to the performance grade is negligible; however, since the performance curve is not similar in shape at low temperatures, the range in which the value can be shared is limited. It is desirable to use deflection-specific performance curves, especially when performing mooring analysis using Grade A rubber fenders in cold climates.

Table 6.3.10.1 Example of temperature factors (catalogue display, before simplification)

Performance grade	Grade A		Grade B		Grade C	
Strain rate (%/S)	Reaction force TF_R	Energy absorption TF_E	Reaction force TF_R	Energy absorption TF_E	Reaction force TF_R	Energy absorption TF_E
-30	2.28	2.29	1.75	1.84	1.69	1.78
-20	1.77	1.76	1.49	1.52	1.44	1.47
-10	1.39	1.40	1.30	1.30	1.26	1.26
0	1.16	1.18	1.16	1.16	1.13	1.13
10	1.04	1.07	1.06	1.06	1.06	1.06
23	1.00	1.00	1.00	1.00	1.00	1.00
30	0.99	0.98	1.00	1.00	1.00	1.00
40	0.97	0.94	0.99	0.95	1.00	0.96
50	0.87	0.85	1.00	0.90	1.00	0.91

Table 6.3.10.1 Example of temperature factor (catalogue display, after simplification)

Performance grade	Grade A		Grade B		Grade C	
Strain rate (%/S)	Reaction force TF_R	Energy absorption TF_E	Reaction force TF_R	Energy absorption TF_E	Reaction force TF_R	Energy absorption TF_E
-30	2.28		1.75	1.84	1.69	1.78
-20	1.77		1.49	1.52	1.44	1.47
-10	1.39		1.30		1.26	
0	1.16				1.13	
10	1.06					
23	1.00					
30	1.00					
40	0.97	0.94	0.99	0.95	1.00	0.96
50	0.87	0.85	1.00	0.90		0.91

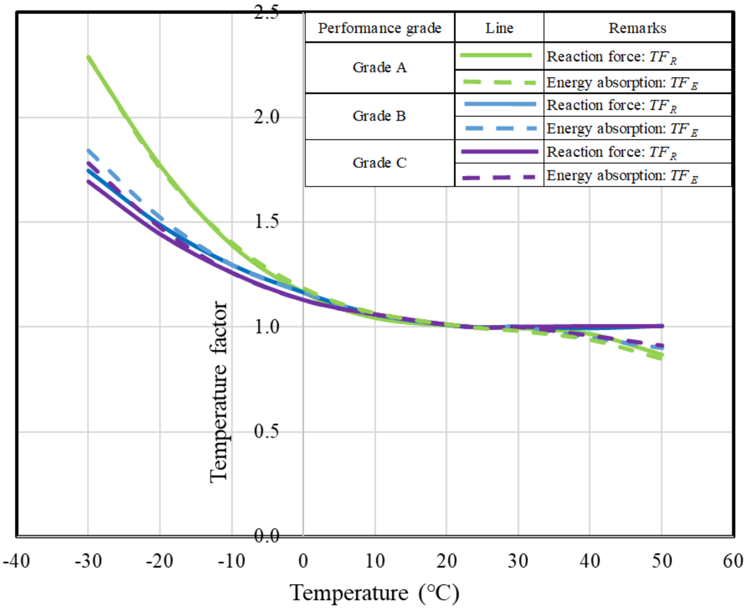


Fig. 6.3.10 Example of temperature factor (catalogue display, graph)

These tendencies are shown in Fig. 6.3.10. The rubber fender catalogue should preferably present either Table 6.3.10.1, Table 6.3.10.2 or Fig. 6.3.10. When only a graph is shown, it is desirable to present an approximate expression and its coefficients so that it can be used for design.

(1) Temperature dependency test of rubber

The temperature dependency of fenders is due to the temperature dependency of rubber. Therefore, testing the temperature dependency of rubber is useful in complementing the temperature dependency of the fender.

1) Specimen

A dumbbell or ring must be used, as specified in JIS K6250.

2) Temperature stabilization

The specimen is thermally stabilized in a thermostatic chamber. The temperature stabilizing time is 1 h or more at the target temperature ± 3°C.

3) Tests

The stress, tensile strength, and elongation in increments of 10% tensile strain (10% to 100%) are measured, and the hardness is measured separately.

4) Test conditions

The temperature range is measured every 10°C for -30°C to 50°C, and a standard temperature of 23°C is employed instead of 20°C.

5) Representative value

A representative value for determining the temperature factor is employed. Here, we present an example in which the tensile stress (Md100) at a strain of 100% is used, and the ratio of Md100 (T=23°C) to the corresponding value at the target temperature is considered. The representative value does not have to be Md100, but it is desirable to use a representative value that reflects the changes in the reaction force of the rubber fender.

The temperature dependence of a sample material is as shown in Fig. 6.3.11.

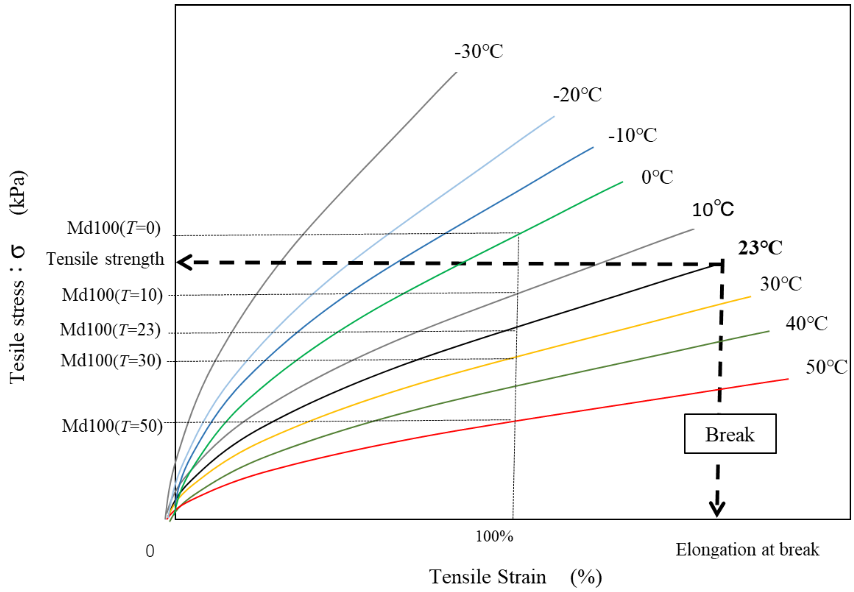


Fig. 6.3.11 Concept of temperature test results for rubber

In Figure 6.3.11

Md100 (T=23): Tensile stress (kPa) at 100% strain at target temperature 23°C

Md100 (T= 0) : Tensile stress at 100% strain at a temperature of 0°C (kPa)

Md100 (T=10): Tensile stress at 100% strain at a temperature of 10°C (kPa)

Md100 (T=30): Tensile stress at 100% strain at a temperature of 30°C (kPa)
Md100 (T=50): Tensile stress at 100% strain at a temperature of 50°C (kPa)

Md100 is selected for the temperature closest to that corresponding to the temperature factor of the rubber fender (real product or scale model), and the temperature factor of a rubber fender is estimated by interpolation, taking the ratio of Md100 between previous and next temperature. For example, assuming that temperature factors of -30°C, 0°C, 23°C and 50°C are available, the temperature factor *TF* (T=10) and *TF* (T=30) of the rubber fender at temperatures of 10°C and 30°C can be obtained by linear interpolation using equation (6.3.13). The method of interpolation can be selected appropriately according to the tendency of the data.

$$\left.\begin{aligned} TF(T=10) &= TF(T=23) + \frac{(TF(T=0)-TF(T=23))\cdot(\mathrm{Md100}(T=10)-\mathrm{Md100}(T=23))}{(\mathrm{Md100}(T=0)-\mathrm{Md100}(T=23))} \\ TF(T=30) &= TF(T=50) + \frac{(TF(T=23)-TF(T=50))\cdot(\mathrm{Md100}(T=30)-\mathrm{Md100}(T=50))}{(\mathrm{Md100}(T=23)-\mathrm{Md1{+}00}(T=50))} \end{aligned}\right\} \quad (6.3.13)$$

6.3.5 Relationship between temperature factor and speed factor

Rubber is a visco-elastic material, and a relationship between speed dependency and temperature dependency exists. This relation was reported by Williams, Landel & Ferry (WLF) [3] and can be used to verify whether the data are accurately measured. In the case of shortage of data, a reasonable estimation can be made based on the nature of the rubber, and this method is better than general interpolation or extrapolation. The formula of WLF is as shown in equation (6.3.14).

$$log_{10}a_T = -\frac{c_1(T-T_s)}{c_2+(T-T_s)} \quad (6.3.14)$$

α_T : Shift factor
T : Temperature (°C)
T_S : Standard temperature $T_S \doteqdot T_g + 50$°C
T_g : Glass transition point of rubber
c_1 : Coefficient: 8.86
c_2 : Coefficient: 101.6

The verification of correlation of temperature factor and speed factor can be performed as follows.

STEP-1: First, plot the temperature factor and speed factor, as shown in Fig. 6.3.12.1 and Fig. 6.3.12.2. It is not easy to change the compression speed when using large test equipment, and the temperature is easier to change than the compression speed. Here, it is assumed that 9 temperature factors and 4 speed factors have been measured.

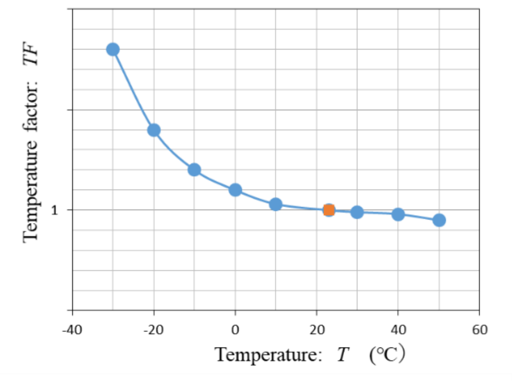


Fig. 6.3.12.1 Example of temperature factor

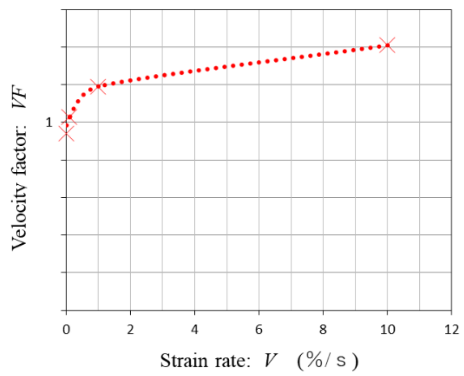


Fig. 6.3.12.2 Example of velocity factor

STEP-2: Next, the horizontal axis of each graph is displayed logarithmically. At this time, the X axis of temperature T substitutes the test temperature in equation (6.3.14) and becomes α_T. The initial value of Ts is 50°C, as shown in Fig. 6.3.13. The speed factor is shown in Fig. 6.3.14 with the logarithm of the strain rate on the X axis.

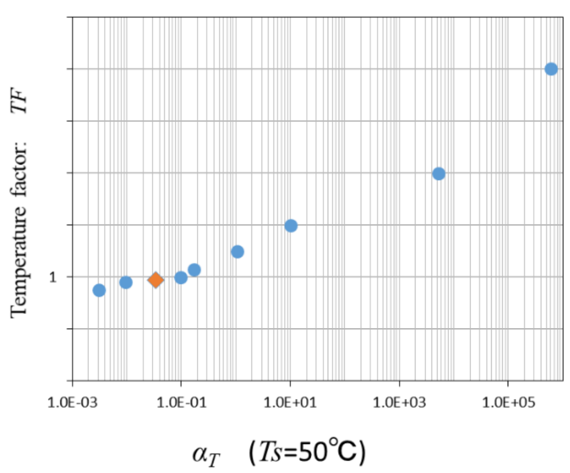


Fig. 6.3.13 Temperature factor (log)

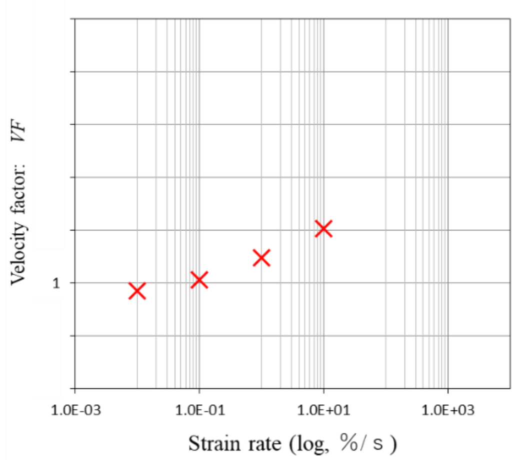


Fig. 6.3.14 Velocity factor (log)

STEP-3: Overlay both graphs. The velocity factor and temperature factor are plotted on the same Y-axis; α_T and log velocity are plotted on the same X-axis. Since changing the value of Ts changes the value of α_T, the value of Ts is adjusted so that the plots of the velocity factor and the temperature factor overlap. The temperature factor may be approximated and the value of Ts may be determined using an Excel Solver function or similar approaches to minimize the residual sum of squares to the velocity factor.

Both the graphs should have curves of similar shapes, and if the curves are not similar, the test results may have errors. The state of superposition is shown in Fig. 6.3.15. The temperature factor (blue circle) and the velocity factor (red X point) are almost identical. The orange square point denotes the standard conditions (V_0=0.01 to 0.3%/s at temperature 23°C) and its value is 1.0. In this manner, while interpolating between test data, factors at very low and high velocities can be rationally estimated by superposing the temperature factors.

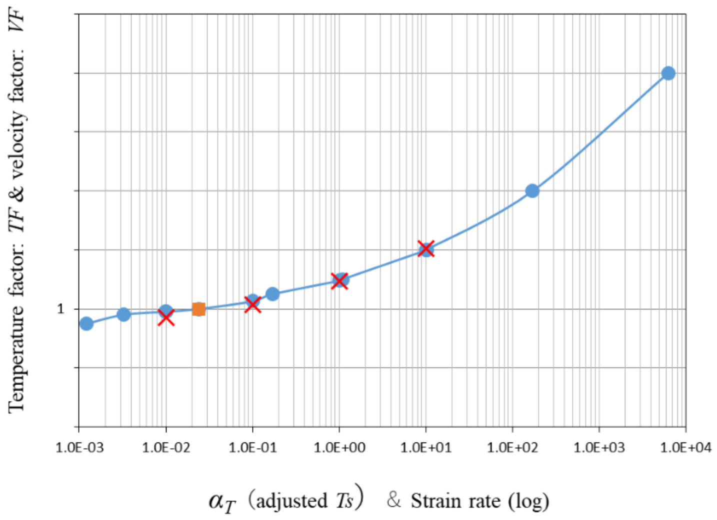


Fig. 6.3.15 Overlapping of temperature factor and velocity factor

6.3.6 Material tests

The quality of rubber materials is important because rubber fenders are often subjected to high strain. Although material testing may be carried out not only in development tests but also in authentication tests and quality verification tests, it is desirable that all tests are conducted following the same procedure.

For rubber materials, (1) to (4) must be satisfied as basic requirements.

(1) A vulcanized product must be obtained by mixing natural or synthetic rubber containing carbon black or white carbon.
(2) The material must have ageing resistance, seawater resistance, ozone resistance, abrasion resistance, etc.
(3) The material must be homogeneous and free from foreign matter, bubbles, scratches, cracks and other harmful defects.
(4) Embedded steel plate for installation flange must be covered and firmly cured by vulcanization; furthermore, it should not be exposed.

In addition, in the Standard Specifications for Ports and Harbor Works [1)], the standard physical property values of rubber used for rubber fenders are defined, as given in Table 6.3.11.

Table 6.3.11 Physical properties of rubber for fenders

Material test		Test standard	Requirement
Accelerated ageing	Ageing method	Heat ageing JIS K 6257 Accelerated ageing (AA-2), 70℃±1℃ × 96 $^{+0/-2}$ hours	
	Tensile strength	Tensile test JIS K 6251	Not less than 80% of original value
	Elongation at break		Not less than 80% of original value
	Hardness	Durometer hardness (Type-A) JIS K 6253-3 (ISO 7619-1)	Must not exceed +8 point of original value
Ozone resistance	Static ozone degradation	JIS K 6259 (ISO 1431-1) 50±5pphm, 20±2% elongation, 40±2℃ × 72hours	No visible cracks after 72 hours

6.4 Authentication test

The authentication test is a test entrusted to a third party to obtain an objective evaluation, and the following durability test is currently prescribed by the Standard Specifications for Ports and Harbor Works [1]. The specific procedure is described in detail, for example, in the Standard Procedure to Durability Test for Rubber Fender Units [4].

6.4.1 Durability test

Fatigue deterioration in a durability test is not similar to that of a scale model; therefore, commercially available products are used as test specimens. The available size is selected by a manufacturer according to the capacity of the test facility. Because of the difference in heat generation and heat dissipation, larger sized fenders exhibit a larger increase in rubber temperature due to continuous compression. However, since actual fenders in ports are not compressed continuously, time for heat dissipation and recovery is available. Although recognizing the influence of such aspects, a durability test is the most direct test method to evaluate the basic durability of rubber fenders, and this test must be performed as an authentication test to validate the durability.

(1) Temperature stabilization

The specimen is thermally stabilized to a standard temperature of 23±5°C. The stabilizing time corresponds to the number of days equal to or greater than that obtained using equation (6.3.1).

(2) Preliminary compression

Compression is performed at least 3 times up to the design deflection or more. The consideration of residual deflection, compression speed, compression interval, etc. is not specified, but it is desirable to unify these concepts and record the number of times and reaction force data.

(3) Static compression for pre-test performance

After preliminary compression, the specimen is thermally stabilized at a standard temperature of 23±5°C for 1 h or more. Next, the specimen is compressed again to its design deflection or more at a strain rate ranging from 0.01 to 0.3%/s.

(4) Durability test

The durability test consists of 3,000 compression cycles to a design deflection or more at a constant speed (CV) or deceleration (DV) with the period of one compression cycle not exceeding 150 s. The waveform of deceleration (DV) is not specified, but the compression and unloading start positions should be fixed.

(5) Visual check

Visually confirm that no cracks or defects are visible to the naked eye after the durability test.

(6) Static compression for post-test performance

Within 24 hours after the durability test, the specimen is compressed again until the design deflection or more, and its performance is considered as the static compression performance after the durability test. At the time of testing, it is desirable to record the temperature of the environment in which the rubber fender is placed.

Examples of durability test results are given in Table 6.4.1.

Table 6.4.1 Example of durability test result

Rubber fender	Sample: 600H	No. Sam001		
Pre-test performance	Test date	Dec., 1st, 2016		
	Energy absorption	196.7 kN·m		
	Reaction force	429.5 kN		
Post-test performance	Test date	Feb.,6th, 2016	Durability test end date	Feb.,5th, 2016 18:15
	Energy absorption	176.2kN·m	Post-test compression start date	Feb.,6th, 2016 06:15
	Reaction force	392.3 kN	Recovery time	12 hours
Visual check	Defects	None	Judgement	Satisfactory

The durability test is also called the repetition fatigue test when compression is continued until failure, and an example is shown in Fig. 4.2.17 in Chapter 4. It is desirable that the fatigue characteristics are presented in the technical data as needed, even if they are not presented in the catalogue.

6.4.2 Other authentication tests

Static compression tests ((3) and (6) in 6.4.1) before and after the durability test and material tests are also required, as specified in the Standard Procedure to Durability Test for Rubber Fender Units [4]. These tests are equivalent to the static compression test described in Section 6.3.1 and the material test described in Section 6.3.6.

6.5 Quality verification test

The quality verification test is used by manufacturers to ensure product quality. The test is conducted on the premise that products to be delivered are individually implemented, and the results are presented to the purchaser. Specifically, the test should be conducted based on the agreement between the purchaser and manufacturer. Generally, static compression tests and material tests are required to be performed. The test methods are the same as those for the development test; however, they are described in detail in this section because the test results are accompanied by a pass/fail judgment of products before delivery to customer.

6.5.1 Static compression test

The static compression test leads to an index that represents the basic performance of rubber fenders. Furthermore, the test is performed as a development test and an authentication test in addition to the quality verification test, and it is performed widely for specimens ranging from actual products to scale models. Although large-scale actual fender tests depend on the scale of the testing facility, manufacturers must be able to carry out this test internally or externally for all types, sizes, and performance grades manufactured in-house. The specific test method is carried out in the following steps according to the PIANC Guidelines [2].

(1) Temperature stabilization of test piece and environment

Since the performance of rubber fenders is affected by the temperature, the test should be conducted in a temperature-controlled environment at the target temperature. The target temperature for measuring the standard performance is 23±5℃ (standard temperature), and it is desirable to conduct the static compression test in an environment within the target temperature ±15°C even in an environment where it is difficult to maintain constant temperature. Specifically, the following two methods can be employed.

1) Temperature control before test

Proper temperature control should be performed by storing rubber fenders in an environment such as that of a temperature-controlled room at a target temperature of ±5°C for a necessary period until the fenders are stabilized. The number of days required to stabilize the temperature of entire rubber body is determined using equation (6.3.1).

$$\text{Days for thermal stabilization} = 20\cdot (\text{Maximum rubber thickness(m)})^{1.5} \qquad (6.3.1)$$

After taking the fender out of the temperature-controlled room, if the test environment is out of the range of the target temperature ±5°C, the total test time including the preliminary test should be within 2 hours to reduce performance influence. Additionally, for the case temperature correction is needed later, the environmental temperature at which rubber fender is placed should be recorded throughout the test.

2) Temperature correction as alternative of temperature control:

As in the case of large sized fenders, when it is difficult to maintain the fender at the target temperature ±5°C in advance, or when the test time including preliminary compression after the specimen has been taken out of the temperature-controlled room is more than 2 h, the temperature is corrected using the average value of the recorded environmental temperature. The correction is performed by multiplying the temperature factor with respect to the time average of the recorded environmental temperature (equation (6.3.1)) with the target temperature. Even in such a case, it is desirable that the test environment is within the temperature range of target temperature ±15°C. Even when the test room itself is temperature-controlled or when the ambient temperature change is within the target temperature ±5°C, it is recommended that the ambient temperature at which the rubber fender is placed throughout the test is recorded.

(2) Preliminary compression

Compression must be performed three or more times upon the manufacturer's recommendation up to the design deflection or more. The consideration of compression residual strain, compression rate, compression interval, etc. is not specified; however, it is desirable to unify these concepts. It is recommended that the number of compressions and performance values are recorded.

(3) Main compression (Static)

Leave the specimen at the target temperature ±5°C for 1 h or more after preliminary compression and compress the specimen again to the design deflection or more; the corresponding performance is the static compression performance. To eliminate the residual strain of preliminary compression, the initial deflection point at which the reaction force starts to increases is the origin of the deflection axis. The compression rate is within the range of the standard strain rate (0.01 to 0.3%/s or 0.3 to 1.3 mm/s), and if the compression rate in this range is difficult to be realized due to size restrictions, etc., correct the result using the velocity factor. In accordance with the capability of the testing facility, it is possible to execute decelerating compression DV from a high initial velocity to measure the performance including the influence of velocity. This performance is known as the decelerating compression performance, and it is different from the static compression performance. In the case of velocity reduction compression, the initial speed and reduced speeds are also recorded. In addition, when storage cannot be performed at the target temperature ±5°C, the result is temperature-corrected using the average value of the recorded ambient temperatures.

(4) Number of samplings for quality confirmation test

The quality confirmation test includes a sampling test and a 100% test. The frequency of the sampling test is set at a rate of 1 unit if the number of fenders is 10 or less, and 1 unit per 10 units if more than 10 units are used; the fraction value is rounded up.

(5) Treatment at time of failure in quality confirmation test

If a failure occurs in the 100% test, it is rejected. If a rejection occurs in a sampling test, the rejected sample is removed and the number of samplings for the remaining products is doubled (1 unit for 5 units, 2 units for group of less than 5 units) for the additional tests. If failure still occurs, 100% inspection is conducted and the failed products are rejected.

(6) Break-in compression

The rubber fenders for which the effect of reaction force on the safety of a structure cannot be ignored, in other words, for which the design condition is other than the influence factor pattern A specified in Table 5.4.4 in Chapter 5, Section 5.4, are compressed to the design deflection or more at least once after being manufactured. This compression is expected to remove the first high reaction force of virgin products and is known as break-in compression. This process must be performed even for the products that are not selected for quality confirmation tests.

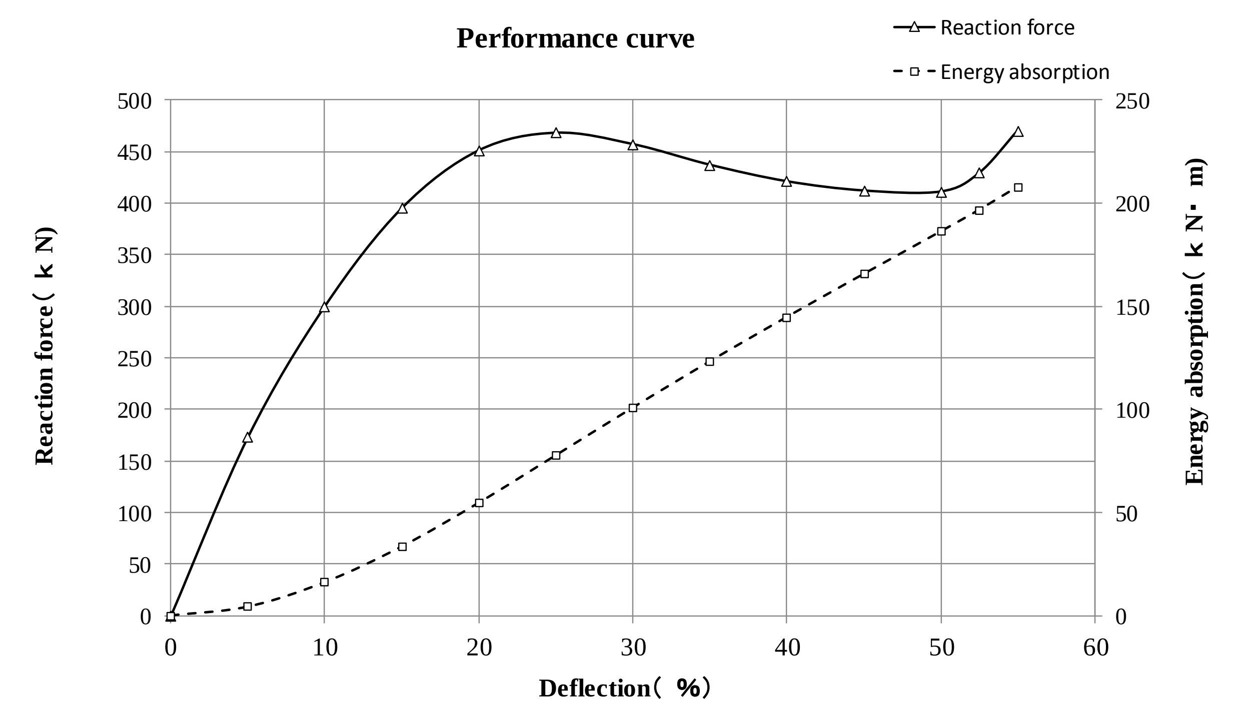
(7) V-type rubber fenders of open leg type are fixed to the base plate of the tester to ensure that the legs do not open.

(8) Energy absorption is obtained by integrating the reaction force with respect to the deflection up to the maximum deflection. The integration can be numerical such as trapezoidal integration, and the step size of deflection is within 5%.

An example of a static compression test for quality verification is presented in Table 6.5.1.

Table 6.5.1 Example of static compression test for quality verification

Compression performance (Test result)							
Product	Sample 1000H				Rubber grade	Test date	Feb., 6th, 2017
Product No.	Sample1	Temperature	Start: 24℃ Finish:24℃		Grade A	Main compression	11：30,Feb., 7th, 2017
Deflection		Reaction force (kN)					Energy absorption
		Preliminary compression			End time	Main compression	
(mm)	(%)	(1)	(2)	(3)	13:00	(4)	(kN·m)
0	0	0.0	0.0	0.0		**0.0**	**0.0**
50	5	201.0	168.0	165.0		**173.0**	**4.3**
100	10	336.0	291.0	286.0		**300.0**	**16.2**
150	15	452.0	384.0	376.0		**395.0**	**33.5**
200	20	538.0	440.0	428.0		**451.0**	**54.7**
250	25	578.0	458.0	442.0		**468.0**	**77.7**
300	30	579.0	448.0	432.0		**457.0**	**100.8**
350	35	549.0	429.0	413.0		**437.0**	**123.2**
400	40	508.0	413.0	398.0		**421.0**	**144.7**
450	45	474.0	404.0	390.0		**412.0**	**165.6**
500	50	450.0	402.0	390.0		**411.0**	**186.2**
525	52.5	443.0	420.0	406.0		**429.0**	**196.7**
550	55	454.0	456.0	448.0		**470.0**	**207.9**
Reaction force at 52.5% (kN)		Standard	Result	Deviation	Energy absorption (kN)	Standard	Result	Deviation
		437+10%	429.5	-1.72%		192-10%	196.7	2.45%



6.5.2 Material tests

The quality of rubber materials is important because rubber fenders are often subjected to high strain. Although material testing may be carried out not only in quality verification tests but also in authentication tests and development tests, it is desirable that all tests are conducted following the same procedure.

For rubber materials, (1) to (4) must be satisfied as basic requirements.

(1) A vulcanized product must be obtained by mixing natural or synthetic rubber containing carbon black or white carbon.

(2) The material must have ageing resistance, seawater resistance, ozone resistance, abrasion resistance, etc.
(3) The material must be homogeneous and free from foreign matter, bubbles, scratches, cracks and other harmful defects.
(4) Embedded steel plate for installation flange must be covered and firmly cured by vulcanization; furthermore, it should not be exposed.

In addition, in the Standard Specifications for Ports and Harbor Works [1)], the standard physical property values of rubber used for rubber fenders are defined as given in Table 6.5.2. Rubber exerts its physical properties by means of a vulcanization reaction caused by heat and pressure. The method of applying pressure and temperature differs depending on the product because thick materials such as rubber fenders require a long time for the internal part to be affected. The process is a unique expertise of each manufacturer. Even if the material test sheet is selected from the same mixing lot as the product, the time to reach the heat of vulcanization is different depending on the depth from the surface of product; thus, it is impossible to match the history of the temperature for any part of product. In the product, any physical property is distributed naturally. Therefore, it must be noted that the material test described here is the confirmation of the standard value necessary to reliably exhibit the required performance when the material constitutes a rubber fender, and it is not always necessary to ensure the same physical properties everywhere.

Table 6.5.2 Physical properties of rubber for fender (Same as Table 6.3.11)

Material test		Test standard	Requirement
Accelerated ageing	Ageing method	Heat ageing JIS K 6257 Accelerated ageing (AA-2), 70℃±1℃ × 96 +0/-2 hours	
	Tensile strength	Tensile test JIS K 6251	Not less than 80% of original value
	Elongation at break		Not less than 80% of original value
	Hardness	Durometer hardness（Type-A） JIS K 6253-3 （ISO 7619-1)	Must not exceed +8 point of original value
Ozone resistance	Static ozone degradation	JIS K 6259 (ISO 1431-1) 50±5pphm, 20±2% elongation, 40±2℃ × 72hours	No visible cracks after 72 hours

(1) Batch and lot in quality verification test of material

In principle, the test pieces to be subjected to material testing involve one set of samples taken from the same lot as the product. The rubber material is prepared as raw rubber before vulcanization, which is used for the product in the kneading step. One batch of the material obtained in one kneading step, and the total of each batch continuously produced with the same composition is called a lot. For large products, multiple batches may be used for one product, and depending on the amount of rubber, multiple lots may be used. The same lot to be subjected to the material test refers to the material extracted from any batch in the kneading lot of the material used for the product. In addition, when the material is used for multiple different products from one lot, the material extracted from the batch is used for the target product.

(2) Response to failure in quality verification test

If the quality verification test does not meet the requirement values specified in Table 6.5.2, retesting is performed using two more sets of samples from the same kneading lot. If all the samples pass, and if the products that use the same lot pass the other quality verification tests, it is acceptable to assume that the product can pass the quality verification test.

(3) Submission of used material sample (for ingredient confirmation)

Although the material sample to be subjected to the test must be from the same mixing lot, it may be difficult to prove this aspect to a third party due to the nature of the production process. The purchaser can request the manufacturer to submit the material sample used in the quality verification test. The purchaser can retrieve the sample and store them, and he/she can independently confirm the identity of the basic component with the delivered product at any time and method using a third-party organization.

An example of the material test result for the quality verification test is given in Table 6.5.3.

Table 6.5.3 Example of material test result

Physical property test report

(1) Hardness
Method: JIS K 6253-3
Accelerated Aging: JIS K 6257
Condition: 70℃±1℃ × 96 +0/-2 hours

Before ageng	After ageing	Difference	Requirement	Judgement
56	62	+6	Not more than +8	Satisfactory

(2) Tensile test
Method: JIS K 6251
Accelerated Aging: JIS K 6257
Condition: 70℃±1℃ × 96 +0/-2 hours

Tensile strength (kN)

Before ageng	After ageing	Ratio	Requirement	Judgement
16.7	15.8	95%	Not less than 80% of before aging	Satisfactory

Elongation at break (%)

Before ageng	After ageing	Ratio	Requirement	Judgement
540	480	89%	Not less than 80% of before aging	Satisfactory

(3) Ozone resistance
Method: JIS K 6259-1
Condition: 50±5pphm ozone, 20±2% elongation, 40±2℃x72hours

Results	Requirement	Judgement
No cracks	No visible cracks after 72 hours	Satisfactory

6.6 Influence factor and classification of tests

The various tests described in the previous section differ in terms of the classification of tests to be performed considering the performance influencing factors defined in Chapter 5, Section 5.4. The test classification is presented in Table 6.6.1.

Table 6.6.1 Performance influencing factors and types of tests

Influence factor	Production tolerance only (Pattern A)	Major influence factors (Pattern B)	Mooring analysis (Pattern C)
Static compression (Delivery test)	Quality verification test (Sampling)	Quality verification test (Sampling)	Quality verification test (100%)
Material test	Quality verification test	Quality verification test	Quality verification test
Durability test (3000 cycle)	Authentication test	Authentication test	Authentication test
Angular compression	N/A	Development test	Development test
Velocity dependency	N/A	Development test	Development test
Temperature dependency	N/A	Development test	Development test
Aging	N/A	N/A	Development test /reseach
Repetition fatigue	N/A	N/A	Development test
Creep characteristic	N/A	N/A	Development test

(1) The quality verification tests (static compression test and material test) and authentication tests (durability test), which are listed in Table 6.6.1 are defined in the Standard Specifications for Ports and Harbor Works [1).

(2) The durability test is also a development test; however, since it has already been established as an authentication test and is defined in the Standard Specifications for Ports and Harbor Works [1), the 3000-cycle durability test was classified as an authentication test.

(3) In several cases, it is difficult to reproduce a development test as an authentication test under the same conditions at a third party premises, to ensure the results in the presence of a third party such as for a quality verification test or to submit the used sample after testing. Therefore, manufacturers should explain the test contents using technical documents.

(4) The research for ageing is still under development, and presently, its testing requires a large duration. However, certain information in terms of test data for used fenders [6) and accelerated ageing testing of scale models [7) is available; thus, one may refer to the technical data of the manufacturer, if necessary.

(5) Repetition fatigue and creep characteristics tests are conducted using the manufacturer's technical data, if necessary [7).

6.7 Similarity rules in scale model testing

Rubber fenders are large in size, and it is sometimes difficult to conduct various tests on an actual product. In such cases, a scaled model is used. The model test of rubber fenders employs a scale model, which is smaller than the product, and has the same material as that of an actual fender.

The size of an actual fender is indexed by *R* and that of the model is represented by *m*; the scale *S* is defined using equation (6.7.1).

$$S = H_R/H_m \tag{6.7.1}$$

Here,

S : Scale

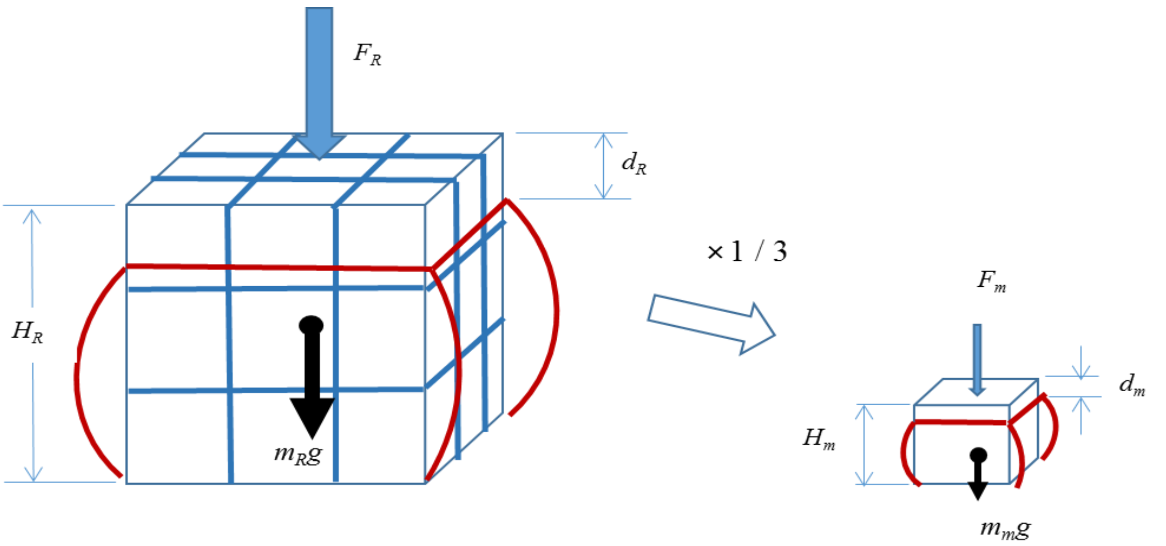
H_R : Height of fender

H_m : Height of scale model

Since the exact match between a scale model and actual product is reflected only in a full-scale test, the physical quantity of interest must ideally be similar. For example, in a hydraulic experiment, both the product and model must use water, and the gravitational acceleration must be the same as well. Therefore, the material property and time must be adjusted to match the Froude number to ensure that the flow field is similar to the free surface. Furthermore, in experiments involving pipe flows around obstacles, the physical property values and time must be adjusted to match the Reynolds numbers.

The characteristics of deflection and reaction force of rubber fenders do not approximate such motions; however, the stress and strain generated in rubber must be similar. To make the concept easy to understand, assume a cubic rubber block, as shown in Fig. 6.7.1. Furthermore, assume that the size of the model for the test is one-third that of the product, and both the model and product are made of the same material. It can be seen that the physical quantities in Fig. 6.7.1 are scaled down, as defined in equation (6.7.2).

$$\left.\begin{array}{rl} \text{Height} & : H_R/H_m = S = 3 \\ \text{Deflection} & : d_R/d_m = S = 3 \\ \text{Area} & : A_R/A_m = S^2 = 9 \\ \text{Volume} & : V_R/V_m = S^3 = 27 \\ \text{Weight} & : W_R/W_m = S^3 = 27 \\ \text{Strain } \varepsilon(\text{-}) & : (d_R/H_R) \,/\, (d_m/H_m) = (d_R/d_m) \,/ (H_m/H_R) = S \times 1/S = 1 \\ \text{Reaction force at strain } \varepsilon & : F_R/F_m = S^2 = 9 \end{array}\right\} \qquad (6.7.2)$$



Area: A_R
Volume: V_R
Weight: W_R

Area: A_R
Volume: V_R
Weight: W_R

Fig. 6.7.1 Cubic fender and 1/3 scale model

If it takes t_R seconds to compress the real cubic fender by d_R, equation (6.7.3) can be used to obtain the time t_m required for compression of the model to the same deflection.

$$(H_R/d_R) / t_R = (H_m/d_m) / t_m \qquad (6.7.3)$$

Considering $H_R/d_R = H_m/d_m$,

$$t_R = t_m \qquad (6.7.4)$$

The following five mechanical quantities are considered to discuss the validity of the abovementioned similarity rules.

1) Elastic force: Linear force proportional to strain (Hooke's law)
2) Viscous force: Force proportional to deformation speed (Newton's viscosity)
3) Friction force: Force proportional to and perpendicular to vertical load (Coulomb friction)
4) Weight: Downward force by gravity
5) Inertial force: Product of mass and acceleration (Newton's law of inertia)

(1) Elastic force

Assuming that the stress is σ, strain is ε, and Young's modulus is E, equation (6.7.5) holds.

$$\sigma = \varepsilon \cdot E \tag{6.7.5}$$

Considering a rubber block having a spring constant K, according to Hooke's law, if the compression displacement is δ and elastic force is F, equation (6.7.6) holds.

$$F = K \cdot \delta \tag{6.7.6}$$

Assuming that the representative area (cross-sectional area) of this rubber block is A, and the representative length (height) is H, the spring constant K can be expressed using equation (6.7.7).

$$K = E \cdot A / H \tag{6.7.7}$$

As shown in Fig. 6.7.1, the real product is represented by $_R$, and the model is represented by the subscript $_m$. According to equations (6.7.6) and (6.7.7),

$$\begin{aligned} &\text{Real product} \quad : \quad F_R = K_R \cdot \delta_R = (E_R \cdot A_R / H_R) \cdot \delta_R \\ &\text{Scale model} \quad : \quad F_m = K_m \cdot \delta_m = (E_m \cdot A_m / H_m) \cdot \delta_m \end{aligned} \tag{6.7.8}$$

Assuming that the product and model are made of the same material, $E_R = E_m$. The elastic force ratio F_R / F_m is

$$F_R / F_m = (E_R \cdot A_R / H_R) \cdot \delta_m / \{(E_m \cdot A_m / H_m) \cdot \delta_R\} = (A_R / A_m) \cdot (H_m / H_R) \cdot (\delta_R / \delta_m) \tag{6.7.9}$$

Substituting $A_R / A_m = S^2$, $H_m / H_R = 1/S$, and $\delta_R / \delta_m = S$ from equation (6.7.2) in equation (6.7.9),

$$F_R / F_m = S^2 \tag{6.7.10}$$

Therefore, the elastic force is considered to be proportional to the square of the scale as well as the area.

(2) Viscous force

The performance of rubber fenders is affected by the compression speed, due to the viscosity of rubber. The stress τ due to viscosity is proportional to the strain rate $d\varepsilon / dt$, and equation (6.7.11) holds if the proportional constant η is considered as the viscosity coefficient.

$$\tau = \eta \cdot (d\varepsilon / dt) \tag{6.7.11}$$

The viscosity coefficient η is a physical property value of the material, and it is not influenced by the dimensions of the object. When the area is multiplied by equation (6.7.11), the viscous force F is obtained, and when the real product is represented by $_R$ and the model is represented by the subscript $_m$, the viscous force can be obtained using equation (6.7.12).

$$\begin{aligned} &\text{Real product} : F_R = \eta_R \cdot A_R \cdot (d\varepsilon_R / dt_R) \\ &\text{Scale model} : F_m = \eta_m \cdot A_m \cdot (d\varepsilon_m / dt_m) \end{aligned} \quad (6.7.12)$$

The strain rate is dimensionless, and it can be considered that $\varepsilon_R = \varepsilon_m$. Therefore, the ratio of viscous force F due to viscosity can be determined using equation (6.7.12).

$$F_R / F_m = (\eta_R \cdot A_R \cdot t_m) / (\eta_m \cdot A_m \cdot t_R) \quad (6.7.13)$$

From equation (6.7.2), $A_R / A_m = S^2$

$$F_R / F_m = S^2 \cdot (\eta_R / \eta_m) \cdot (t_m / t_R) \quad (6.7.14)$$

If both the real product and model are made of the same material, $\eta_R = \eta_m$; thus, equation (6.7.14) becomes

$$F_R / F_m = S^2 \cdot (t_m / t_R) \quad (6.7.15)$$

Therefore, from equation (6.7.4), if $t_m = t_R$, the viscous force is considered to be proportional to the square of the scale, similar to the elastic force.

(3) Friction force

The frictional force F is proportional to the vertical load R, and when the proportional constant is μ (friction coefficient), the frictional force can be expressed using equation (6.7.16).

$$F = \mu \cdot R \quad (6.7.16)$$

The suffix $_R$ corresponds to the real product, and $_m$ corresponds to the scale model. The ratio of friction forces is as given in equation (6.7.17).

$$F_R / F_m = (\mu_R \cdot R_R) / (\mu_m \cdot R_m) \quad (6.7.17)$$

If the same material is used for the product and model, the friction coefficients are equal. Furthermore, if $\mu_m = \mu_R$, the ratio of friction forces becomes the same as the ratio of the reaction forces, and equation (6.7.18) holds.

$$F_R / F_m = R_R / R_m = S^2 \quad (6.7.18)$$

Therefore, it is considered that the friction force is also proportional to the square of the scale, similar to the force due to elasticity and viscosity.

(4) Weight

Although it is considered that the elastic force, viscous force, and friction force are all proportional to the square of the scale, this similarity rule does not hold for the weight even though the dead weight is a downward force. Assuming that the weight of a rubber fender is W, the density is ρ, and the volume is V, the weight can be calculated using equation (6.7.19).

$$W = \rho \cdot V \quad (6.7.19)$$

The real product is represented by $_R$, and the model is represented by $_m$. The ratio of the self weights of the product and model is as shown in equation (6.7.20).

$$W_R \;/\; W_m = (\rho_R \cdot V_R) / (\rho_m \cdot V_m) \qquad (6.7.20)$$

Since V_R / V_m = S^3 from equation (6.7.2), it is necessary to satisfy ρ_R / ρ_m = 1 / S to ensure that the weight W is proportional to the square of the scale along with the elastic force, viscous force and friction force. To this end, it is necessary to use a material having a density S times more than that of the model, which is impossible when using the same material. Table 6.7.1 presents an example of the self-weight and reaction force of fenders as reference for comparing the error.

Table 6.7.1 Self weight and reaction force of fenders

Fender size: H	Self-weight: W (kg)	Reaction force: R (kN)	W / R (%)
100H (Model)	0.79	57	0.014
1000H	700	568	1.21
3000H	16600	5670	2.87

Since rubber fenders are attached horizontally, the reaction force and the weight are orthogonal; although the weight does not necessarily act as an error of the reaction force directly, the error is larger for larger sizes, as shown in Table 6.7.1. It can be seen that even the largest rubber fenders have a weight that is less than 3% of the reaction force. A larger sized fender is more likely to be deformed by its own weight (front drooping) and needs to be supported by the weight chain.

(5) Inertial force

The inertial force F is expressed as a product of the mass and acceleration, as given in equation (6.7.21), where M is the mass and x is the displacement.

$$F = M \cdot (d^2x / dt^2) \qquad (6.7.21)$$

Assuming that the acceleration is an average value, the real product is represented by $_R$, and the model is represented by a subscript $_m$, the ratio of inertial forces is defined as in equation (6.7.22).

$$F_R / F_m = (M_R \cdot x_R / t_R^2) / (M_m \cdot x_m / t_m^2) = (M_R / M_m) \cdot (x_R / x_m) \cdot (t_m / t_R)^2 \qquad (6.7.22)$$

Since the mass is proportional to volume, equation (6.7.23) is obtained from equation (6.7.21) if M_R / M_m = S^3 and x_R / x_m = S.

$$F_R \;/\; F_m = S^4 \cdot (t_m / t_R)^2 \qquad (6.7.23)$$

Therefore, for the ratio F_R / F_m of the inertial forces to be a square of the scale along with the elastic force, viscous force, and friction force, it is necessary to satisfy equation (6.7.24). To this end, the model must deform S times faster.

$$t_R / t_m = S \qquad (6.7.24)$$

This aspect contradicts the expression of the viscous force with t_R / t_m = 1; for S=1, i.e., at full size, the viscous force and inertial force are similar simultaneously.

As a reference, let us estimate the inertia force and reaction force at the time of berthing of a rubber fender. Assuming that the compression is performed at an initial velocity V and stopped at 50% deflection, the time t at this point can be expressed using equation (6.7.25).

$$t = 0.5H / V \tag{6.7.25}$$

Assuming that the average acceleration is $V / t = V^2 / (0.5H)$ and the mass moved by compression is one half the weight of the rubber fender, the inertial force F can be expressed using equation (6.7.26).

$$F = W/(2g) \cdot V^2 / (0.5H) \tag{6.7.26}$$

The estimated results are as presented in Table 6.7.2. It can be seen that the influence of the inertial force is negligibly small compared to that of the self-weight of the fender.

Table 6.7.2 Inertial force and reaction force of rubber fender at berthing

Fender size: H	Velocity: V (m/s)	Self-weight: W (kg)	Inertial force F (kN)	Reaction force: R (kN)	F / R (%)
100H (Model)	0.01	0.79	8.05E-05	57	1.41E-04
1000H	0.1	700	0.71	568	0.13
3000H	0.3	16600	50.76	5670	0.90

The above aspects can be summarized as follows. It should be noted that similarity rules do not hold for these physical quantities.

(1) Deflection by self weight

The deflection of a fender by its self-weight does not become similar in scale (S) because its own weight is (S^3) and reaction force is (S^2). No major impact on performance is noted as the direction is different with respect to the berthing.

(2) Inertial force due to fender deformation

The inertial force generated by the deformation of rubber fenders is not similar because it is related to the weight of the rubber; however, its influence during actual berthing is small.

(3) Temperature

The heat generation is proportional to the volume (S^3), heat conduction is proportional to the distance (S), and heat dissipation is proportional to the surface area (S^2); therefore, the similarity does not hold. It is thus necessary to focus on repetition fatigue tests involving heat generation.

(4) Error in production

In addition to the difference in the accuracy between the dimensions of the actual product and scale model, heat transfer between the actual product and scale model cannot be made similar; therefore, the curing conditions are not exactly the same, and the thickness distribution of rubber properties is slightly different even when the same materials are used. Although the model vulcanization conditions are devised to obtain a performance close to that of the actual product, the methods often correspond to a manufacturer's own technology.

To minimize the effects of these errors, it is desirable that the test results of the scaled model use coefficients with the test data at the standard conditions of the model rather than scaling up the model data. Such an additional form is called a reference model.

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Appendices

A.1 General

The impact of the proposed rubber fender design on the conventional design specifications is as follows.

(1) When considering only manufacturing tolerance: Pattern A
No impact, as there is no change in the design method employed in Japan.

(2) When considering the major influence factors: Pattern B
This pattern is equivalent to the PIANC Guidelines [1]. The variations in the performance, particularly when the reaction force is redesigned at the berthing velocity (deceleration) and/or low temperature, may meet the requirements of energy absorption even with slightly smaller sizes or performance grades, while the reaction force and hull pressure may increase slightly.

(3) When designing fenders through mooring analysis involving numerical calculation: Pattern C
No influence, as the design method is different depending on the case, and it has been studied independently.

The following sections present examples of design considering each factor pattern proposed in this document. The performance data and influence factors used here are samples for explaining the design process, and thus, they are different from actual data. When designing, please refer to the manufacturers' catalogues and technical data and use data that matches the product and performance grade.

A.2 Pattern A: Manufacturing tolerance

A.2.1 General cargo vessel of 5,000 *DWT*: Without fender panel

We consider the trial design of fenders expected to receive 5,000 *DWT* cargo ships on a gravity quay. Since no restriction of allowable surface pressure exists and the quay is gravity type with earth pressure behind, the design example presented is of a case in which the allowable horizontal load is sufficiently large with respect to the reaction force of the rubber fender.

(1) Design conditions
The vessel specifications are listed in Table A.2.1. When the target vessel is known, the vessel statement and the latest statistics [2] can be used. In this case, it is assumed that the data in Table A.2.1 is available.

Table A.2.1 Vessel data

Items	Condition	Unit
Type of vessel	General cargo	
Dead weight tonnage : *DWT*	5,000	t
Gross tonnage : *GT*	2,645	t
Length between perpendiculars : L_{pp}	99.46	m
Beam of vessel : *B*	16.96	m
Full draught of vessel : *d*	6.44	m
Berthing velocity : V_B	0.2	m/s

(2) Effective berthing energy

The effective berthing energy is calculated as presented in Table A.2.2.

Table A.2.2 Effective Berthing energy
(Equation (5.3.1), Section 5.3, Chapter 5.3)

Displacement tonnage	DT	5,870 t
Block coefficient	C_b	0.53
Virtual mass factor	C_m	2.13
Eccentricity factor	C_e	0.50
Softness factor	C_s	1.00
Berth configuration factor	C_c	1.00
Effective berthing energy	E_b	125 kN·m(＝kJ)

(3) Selection of rubber fenders

Since the mooring facilities are gravity type, they correspond to factor pattern A specified in Table 5.4.4 in Chapter 5, and it is sufficient to consider only the manufacturing tolerance as the influence factor for the standard performance. Therefore, from the manufacturer's catalogue, a size corresponding to an energy absorption of 139 kN· m, as obtained by increasing the effective berthing energy 125 kN· m by 10% is selected. Rubber fenders are usually selected with alternative choices of size and performance grade. For example, considering V-type fenders, candidates that satisfy the abovementioned energy absorption for 500H, 600H, and 800H can be selected as shown in Tables A.2.3, A.2.4, and A.2.5, respectively.

Table A.2.3 Catalogue example: V-type fender 500H

Performance Grades	Grade A		Grade B		Grade C	
Design deflection	45%		45%		45%	
Length \ Performance	Reaction force (kN)	Energy absorption (kN･m)	Reaction force (kN)	Energy absorption (kN･m)	Reaction force (kN)	Energy absorption (kN･m)
1000mm	460	81	383	67	306	54
1500mm	689	$\boxed{121}$	575	101	460	81
2000mm	919	$\boxed{161}$	766	$\boxed{135}$	613	108
2500mm	1149	202	958	$\boxed{168}$	766	$\boxed{135}$
3000mm	1379	242	1149	202	919	$\boxed{161}$

Table A.2.4 Catalogue example: V-type fender 600H

Performance Grades	Grade A		Grade B		Grade C	
Design deflection	45%		45%		45%	
Length \ Performance	Reaction force (kN)	Energy absorption (kN･m)	Reaction force (kN)	Energy absorption (kN･m)	Reaction force (kN)	Energy absorption (kN･m)
1000mm	552	$\boxed{117}$	460	$\boxed{97}$	368	78
1500mm	828	$\boxed{175}$	690	$\boxed{146}$	552	$\boxed{117}$
2000mm	1104	234	920	195	736	$\boxed{156}$
2500mm	1380	292	1150	243	920	195
3000mm	1656	350	1380	292	1104	234

Table A.2.5 Catalogue example: V-type fender 800H

Performance Grades	Grade A		Grade B		Grade C	
Design deflection	45%		45%		45%	
Length \ Performance	Reaction force (kN)	Energy absorption (kN･m)	Reaction force (kN)	Energy absorption (kN･m)	Reaction force (kN)	Energy absorption (kN･m)
1000mm	707	$\boxed{189}$	589	$\boxed{157}$	471	$\boxed{126}$
1500mm	1061	283	884	236	707	$\boxed{189}$
2000mm	1414	377	1179	314	943	251
2500mm	1768	471	1473	393	1179	314
3000mm	2121	566	1768	471	1414	377

In Tables A.2.3, A.2.4, and A.2.5, the red boxes are candidates for the effective berthing energy. If the fender length can be adjusted every 100 mm, it is possible to determine the optimum length by proportioning the energy absorption surrounded by two red frames to the length. The sizes and performance grades presented in Table A.2.6 can be selected.

Table A.2.6 Selected V-type fender for each performance grade

Performance Grades	Grade A			Grade B			Grade C		
Performance / Size	Length (mm)	Reaction force (kN)	Energy absorption (kN・m)	Length (mm)	Reaction force (kN)	Energy absorption (kN・m)	Length (mm)	Reaction force (kN)	Energy absorption (kN・m)
500H	1800	827	145	2100	804	141	2600	797	140
600H	1200	662	140	1500	690	146	1800	662	140
800H	800	566	151	900	530	141	1200	566	151

The choice of size from Table A.2.6 is determined by the installation and other considerations, as described in Chapter 5, Section 5.6. For example, when the difference in water levels or the difference in the vertical position of the vessel berthing points is large, the size can be small, for example, 500H×2600 mm for a low performance grade fender with long length. However, the fender may slightly long with respect to the height. Conversely, when the height must be increased due to the restrictions of installation pitch, etc., 800H×800 mm can be selected. However, if the length is less than the height, a certain minimum length is necessary because the deformation mode may change due to the influence of both ends of the rubber fender. For lengths not specified in the catalogue, it is necessary to ask the manufacturer for the possibility of adjusting the length. In this case, V-type 600H×1.5 m (grade B) is selected from the catalogue as a rubber fender capable of absorbing the effective berthing energy of the vessel.

<Standard performance>
Size: V type 600H × 1500L (performance grade B)
Design deflection: Within 45.0 (%)
Reaction force: Standard R_R=690, Max. R^+=760 (+ 10%) (kN)
Energy absorption: Standard E_A=146, Min. E_A^-=132 (-10%) > 125 (kN· m)

(4) Arrangement of rubber fenders (reference)

The rubber fenders can be considered to be installed vertically, horizontally, in an alternating manner (in the vertical and horizontal directions), etc. In this case, the most commonly used configuration, that is, vertical installation is chosen, and the recommended installation pitch S is calculated. First, the curvature radius R_r of the hull of the 5,000 DWT cargo vessel is obtained as follows using equation (5.6.3) of Section 5.6.1, Chapter 5.

$$R_r = B/4 + L_{pp}^2 / (16B) = 40.7 \text{ (m)}$$

Therefore, the installation pitch S can be determined as follows using equation (5.6.1) of Section 5.6.1, Chapter 5.

$$S = 2 \times \sqrt{40.7^2 - (40.7 - 0.6 \times 0.55)^2} = 10.3 \quad \text{(m)}$$

A.2.2 Allowable berthing velocity of oversized vessel for existing fender

Assuming that a full-load 10,000 DWT passenger vessel berths to the quay of the 5,000 DWT cargo vessel designed in A.2.1, The allowable conditions for berthing are as follows.

(1) Design conditions

Table A.2.7 presents the specifications of the vessels expected to berth on this quay. In this case, the following restrictions other than the effective berthing energy are not considered; however, that these aspects need to be checked and considered in actual cases.

1) Limitations of water depth and loaded draft (Bottom clearance)
2) Restriction of protrusion of fender line
3) Presence or absence of hull fenders (Belted hull)
4) Considerations stated in Chapter 5 (e.g., Fig. 5.6.3)

Table A.2.7 Vessel data

Item	Condition	Unit
Type of vessel	Passenger	
Dead weight tonnage : DWT	10,000	t
Gross tonnage : GT	89,390	t
Length between perpendiculars : L_{pp}	271	m
Beam of vessel : B	32.4	m
Full draught of vessel : d	8.11	m
Displacement tonnage : DT	51,220	t
Block coefficient : C_b	0.53	
Virtual mass factor : C_m	1.787	
Eccentricity factor : C_e	0.5	
Softness factor : C_s	1.0	
Berth configuration factor : C_c	1.0	

(2) Calculation of limit value of the berthing speed

From the previous section, the details of existing rubber fenders are as follows, and the minimum value of energy absorption E_A^- is 132 kN· m.

<Standard performance>
Size: V type 600H × 1500L (performance grade B)
Design deflection: Within 45.0 (%)
Reaction force: Standard R_R=690, Max. R^+=760 (+ 10%) (kN)
Energy absorption: Standard E_A=146, Min. E_A^-=132 (-10%) > 125 (kN· m)

The berthing velocity V_B at which the effective berthing energy of the passenger vessel in Table A.2.7 is 132 kN· m is as follows when the factors are summarized, and C_{total} can be defined as follows from equation (5.3.1) in Chapter 5.

$$C_{total} = C_e \cdot C_m \cdot C_c \cdot C_s = 0.5 \times 1.787 \times 1.0 \times 1.0 = 0.8936$$

The allowable berthing velocity can be calculated as follows:

$$V_B = \sqrt{\frac{2 \cdot E_A}{M \cdot C_{total}}} = \sqrt{\frac{2 \times 132}{51220 \times 0.8936}} = 0.076 \qquad \text{(m/s)}$$

Therefore, for a 10,000 DWT passenger vessel, it is necessary to carefully control the berthing velocity to be less than 0.076 m/s.

A.3 Pattern B: Considering major influence factors

A.3.1 General cargo vessel of 30,000 *DWT*: Without fender panel

In this case, a 30,000 DWT cargo vessel berths to a quay that is a pile type pier, and the reaction force of the fender is considered as a horizontal load to the quay. The rubber fender design corresponds to Pattern B, in which the influence factors must be considered.

(1) Design conditions

It is assumed that the specifications of the vessel are as listed in Table A.3.1.1. If these data are unknown, the Technical Note of National Institute for Land and Infrastructure Management [3)] may be helpful.

Table A.3.1.1 Vessel data

Item	Condition	Unit
Type of vessel	General cargo	
Gross tonnage：GT	15,870	t
Dead weight tonnage：DWT	30,000	t
Displacement tonnage：DT	35,220	t
Length between perpendiculars：L_{pp}	171.00	m
Beam of vessel：B	28.30	m
Full draught of vessel：d	16.22	m

1) Berthing conditions

Table A.3.1.2 Berthing conditions

Berthing velocity：V_B	0.10	m/s
Berthing angle：θ	5.00	°
Installation pitch：S	12.0	m

2) Quay structure: Pile supported

3) Allowable hull pressure: Not specified

4) Temperature: 4℃-34℃ (Lowest and highest of daily average)

(2) Effective berthing energy

Table A.3.1.3 Effective berthing energy
(Equation (5.3.1), Section 5.3, Chapter 5.3)

Block coefficient	C_b	0.673
Virtual mass factor	C_m	1.870
Eccentricity factor	C_e	0.553
Effective berthing energy	E_b	182 kN·m(＝kJ)

(3) Selection of rubber fender

An example of a catalogue for a rubber fender that can absorb the effective berthing energy of vessel is given in Table A.3.1.4.

Table A.3.1.4 Catalogue example: V-type fender 800H

Performance Grades	Grade A		Grade B		Grade C	
Design deflection	45%		45%		45%	
Performance / Length	Reaction force (kN)	Energy absorption (kN･m)	Reaction force (kN)	Energy absorption (kN･m)	Reaction force (kN)	Energy absorption (kN･m)
1000mm	707	189	589	157	471	126
1500mm	1061	283	884	236	707	189
2000mm	1414	377	1179	314	943	251
2500mm	1768	471	1473	393	1179	314
3000mm	2121	566	1768	471	1414	377

From Table A.3.1.4, V-type 800H×1400L (Grade B) is selected. The selected size is determined in proportion to the numbers in the red frame for a length of 1400 mm.

<Standard performance>
Deflection: Within 45.0 (%)
Reaction Force: R= 825 (kN) (less than + 10%)
Energy absorption: E_A= 220 (kN·m) (-10% or more)

Because the quay is a pile pier, it is necessary to consider the influence factors of fender performance; thus, factor pattern B is focused on. Each factor of pattern B is determined as follows using the manufacturer's catalogue or technical data.

1) Manufacturing tolerance: C_p

Manufacturing tolerance is usually described along with the performance in the catalogue.

Lower limit: $C_p^-=0.90$, Upper limit: $C_p^+=1.10$

2) Angular factor: C_a

The angular factor is given, for example, as shown in Table A.3.1.5. The angular factors for a berthing angle of 5° are $C_{aR}=1.01$ for the reaction force and $C_{aE}=0.99$ for energy absorption.

Table A.3.1.5 Example of angular factor

Angle (°)	Angular factor of reaction force C_{aR}	Angular factor of energy absorption C_{aE}
0	1.00	1.00
3	1.01	1.00
5	1.01	0.99
8	1.01	0.95
10	0.99	0.94
15	0.98	0.86
20	0.92	0.71

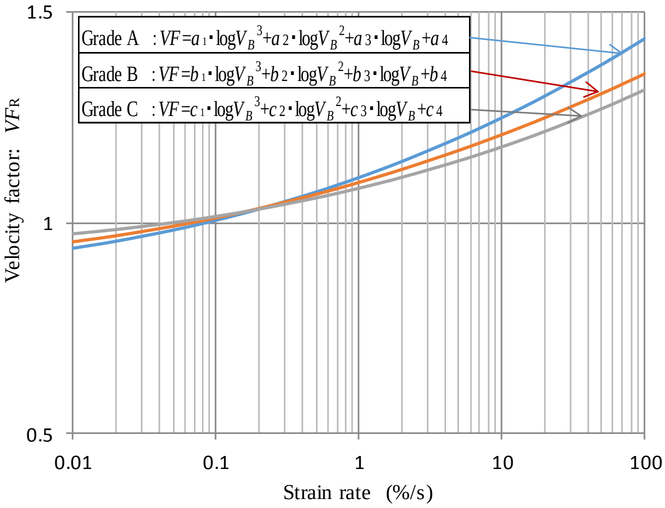
3) Velocity factor: VF

The velocity factor VF_R of the reaction force varies depending on the shape and material of the rubber fender, and is displayed as shown in, for example, Fig. A.3.1.1. If the graph and the polynomial are displayed as shown in Fig. A.3.1.1, the velocity factor can be calculated if the coefficients of the polynomial are given. If the velocity is displayed in the number table, one can read from the number table and prorate as needed. Although the display mode (graph or table or figures) may differ depending on the manufacturer, the factor must be catalogued in a manner that allows data to be extracted for design.

Since the size of the rubber fender is 800H, the berthing velocity of 10 cm/s has a strain rate of 12.5%

/s. From the table on the right of Fig. A. 3.1.1, VF_R can be prorated as in equation (A.3.1.1).

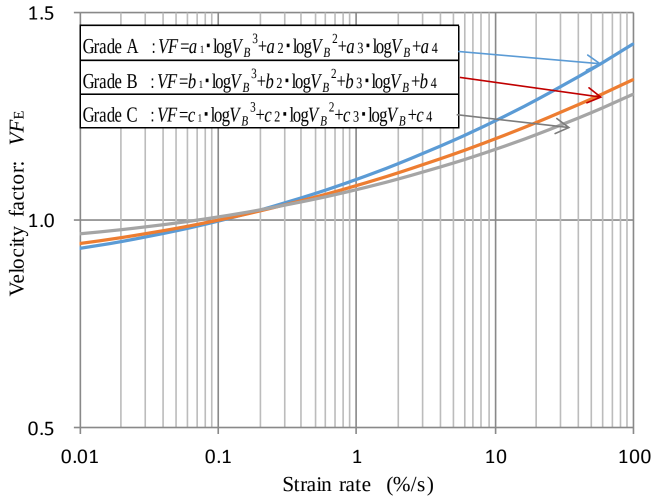
At strain rate 12.5%/s: $VF_R = (12.5-10) / (20-10) \times (1.25-1.21) + 1.21 = 1.22$ (A.3.1.1)



Strain rate (%/s)	Performance grade		
	Grade A	Grade B	Grade C
0.01	0.94	0.95	0.97
0.02	0.96	0.97	0.98
0.05	0.98	0.99	1.00
0.1	1.01	1.01	1.02
0.2	1.03	1.03	1.03
0.5	1.07	1.07	1.06
1	1.11	1.09	1.08
2	1.15	1.13	1.11
5	1.20	1.17	1.15
10	1.25	1.21	1.18
20	1.30	1.25	1.22
50	1.38	1.31	1.27
100	1.44	1.35	1.31

Fig. A.3.1.1 Example of velocity factor (decreasing velocity) of reaction force

The velocity factor VF_E of energy absorption is displayed, for example, as shown in Fig. A.3.1.2. The VF_E of deceleration compression at a strain rate of 12.5% /s, by interpolation, becomes VF_E = 1.21, similar to that obtained using equation (A.3.1.1).



Strain rate (%/s)	Performance grade		
	Grade A	Grade B	Grade C
0.01	0.93	0.94	0.96
0.02	0.95	0.96	0.97
0.05	0.97	0.98	0.99
0.1	1.00	1.00	1.00
0.2	1.02	1.02	1.02
0.5	1.06	1.06	1.05
1	1.10	1.08	1.07
2	1.13	1.11	1.10
5	1.19	1.16	1.13
10	1.24	1.20	1.17
20	1.29	1.24	1.20
50	1.36	1.29	1.26
100	1.42	1.34	1.30

Fig. A.3.1.2 Example of velocity factor (decreasing velocity) of energy absorption

As mentioned above, the velocity factors of reaction force and energy absorption often change by only approximately 1%; therefore, some manufacturers may use a common factor; however, the original data are used in this case.

4) Temperature factor: *TF*

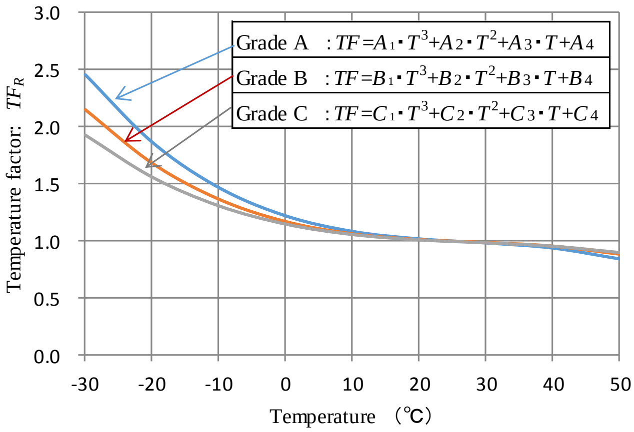
The temperature factor varies depending on the shape and material of the rubber fender. The reaction force and energy absorption are displayed, for example, as shown in Fig. A.3.1.3 and Fig. A. 3.1.4, respectively. Thus, if a graph and a polynomial are displayed, the temperature factor can be calculated if the coefficients of the polynomial are given. If the factor is displayed in a number table, one may read the value from the table in the same manner as the velocity factor and prorate as needed.

From Fig. A.3.1.3 and Fig. A.3.1.4, the temperature factor *TF* is as follows.

Temperature factor of reaction force TF_R : 1.12 (4°C)–0.97 (34°C)
Temperature factor of energy absorption TF_E : 1.11 (4°C)–0.96 (34°C)

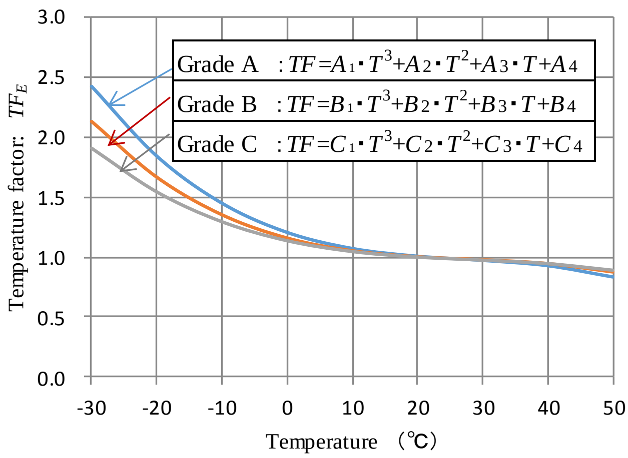
Therefore, the maximum value TF_R^{+} = 1.12 is adopted for the reaction force, and the minimum value TF_E^{-} = 0.96 is adopted for the energy absorption.

Similar to the velocity factor, the temperature factor for the reaction force and energy absorption changes by only approximately 1%, and simplification can be performed by using a common temperature factor; however, the original data is used in this example.



Temperature (℃)	Performance grade		
	Grade A	Grade B	Grade C
-30	2.45	2.16	1.93
-20	1.86	1.68	1.56
-10	1.46	1.36	1.31
0	1.21	1.17	1.15
10	1.08	1.06	1.06
20	1.01	1.01	1.01
30	0.98	0.98	0.98
40	0.93	0.95	0.95
50	0.84	0.88	0.90

Fig. A.3.1.3 Example for temperature factor of reaction force



Temperature (℃)	Performance grade		
	Grade A	Grade B	Grade C
-30	2.43	2.13	1.91
-20	1.84	1.67	1.54
-10	1.45	1.35	1.29
0	1.20	1.16	1.14
10	1.07	1.05	1.05
20	1.00	1.00	1.00
30	0.97	0.97	0.97
40	0.92	0.94	0.94
50	0.83	0.87	0.89

Fig. A.3.1.4 Example for temperature factor of energy absorption

From the above examples, the design performance of the selected rubber fender V-type 800H × 1400L (grade B) is as follows.

5) Design reaction force

$$R^{+} = R_R \times C_p^{+} \times C_{aR} \times VF_R \times TF_R^{+}$$
$$= 825 \times 1.1 \times 1.01 \times 1.22 \times 1.12 = 1252 \text{ (kN)}$$

R^{+} : Design reaction force after considering influence factors (kN)
R_R : Standard reaction force (kN)
C_p^{+}: Production tolerance (1.1)
C_{aR} : Angular factor of reaction force (C_{aR}=1.01 : Angle 5° from Table A.3.1.5)
VF_R : Velocity factor of reaction force (VF_R=1.22, DV of 12.5%/s, from Fig. A.3.1.1)
TF_R^{+}: Maximum temperature factor (TF^{+}=1.12 at 4℃ from Fig. A.3.1.3)

6) Design energy absorption

$$E_A^{-} = E_A \times C_p^{-} \times C_{aE} \times VF_E \times TF_E^{-}$$
$$= 220 \times 0.9 \times 0.99 \times 1.21 \times 0.96 = 227.7(\text{kN}\cdot\text{m}) \geqq 182(\text{kN}\cdot\text{m})$$

E_A^{-} : Design energy absorption after considering influence factors (kN・m)
E_A : Standard energy absorption (kN・m)
C_p^{-} : Production tolerance (0.9)
C_{aE} : Angular factor of energy absorption (C_{Ae}=0.99 at angle 5°, from Table A.3.1.5)
VF_E : Velocity factor of energy absorption (VF_E=1.21, DV of 12.5%/s from Table A.3.1.5)
TF_E^{-} : Minimum temperature factor (TF_E^{-}=0.96 at 34°C, from Fig. A.3.1.4)

The effective berthing energy of the vessel can be absorbed even when considering the influence of the temperature and berthing angle. The lower limit value of the velocity factor VF_E of the abovementioned energy absorption adopts the value of the deceleration (DV) performance at V = 12.5%/s. For example, as discussed in section A.2.2, when large vessels are expected to berth at a slow berthing velocity, it is better to adopt the static performance instead of the DV deceleration performance (VF_E^{-} = 1.0). Even in this case, the energy absorption E_A is 188 kN·m, and the required performance can be satisfied.

As described above, it is possible to design a V-type rubber fender having no fender panel for a 30,000 DWT class vessel; however, it is premised that no limitation on the hull pressure exists. When a V-type rubber fender is selected, the maximum average hull pressure is assumed if the contact width of the fender head is considered to be 0.6 m.

$$1252 \text{ kN} / (0.6 \text{ m} \times 1.4 \text{ m}) = 1491 (\text{kN/m}^2)$$

This value is considerably large, and in case such a value cannot be managed, one must change the value to select the fender panel. In addition, a vessel having no restriction on the allowable hull pressure may have a belt (continuous fender protruding from hull), and the contact surface may be partially lost by the belt impacting the rubber. Thus, a fender panel with wings may also be considered, as shown in Fig. 5.7.3 in Chapter 5, Section 5.7.3.

A.3.2 Tanker of 100,000 *DWT*: With fender panel

Rubber fenders for tankers are often installed on pile-type dolphins, and a hull pressure restriction exists on the vessel side. Therefore, it is necessary to carefully estimate the maximum reaction force. This section presents an example of the design of a rubber fender with a panel in which the influence factor of pattern B is considered assuming the following case.

(1) Design conditions

No hull fenders exist for a tanker, and the vessel often makes full contact with the fender panel. In addition, although the berthing velocity is well-managed because tugboat assistance can be realized, the safety considerations are important because the load involves hazardous material.

1) It is assumed that the following constraints exist on the size of the rubber fenders.
Allowable average hull pressure: 200 (kN/m²)

Horizontal load capacity of quay (dolphin): 3000 (kN) (per rubber fender)
Installation space in front of the dolphin: Horizontal 6 m × vertical 5 m
Fender height: Within 2.5 m from quay line including the fender panel

2) The vessel specifications are presented in Table A.3.2.1.

Table A.3.2.1 Vessel data

Item	Condition	Unit
Type of vessel	Tanker	
Dead weight tonnage：*DWT*	100,000	t
Gross tonnage：*GT*	53,500	t
Length between perpendiculars：L_{pp}	238.35	m
Beam of vessel：*B*	42.72	m
Full draught of vessel：*d*	14.75	m
Berthing angle： θ	5.00	°
Berthing velocity： V_B	0.15	m/s

3) Temperature: 5°C-31°C (Lowest and highest of daily average from field measurement)

(2) Effective berthing energy

The calculation results for the effective berthing energy are presented in Table A.3.2.2, and the berthing condition is shown in Fig. A.3.2.1.

Table A.3.2.2 Berthing data
(from equation (5.3.1) of Section 5.3 in Chapter 5)

Displacement tonnage	*DT*	123,500 t
Block coefficient	C_b	0.80
Virtual mass factor	C_m	1.68
Eccentricity factor	C_e	0.61
Softness factor	C_s	1.00
Berth configuration factor	C_c	1.00
Effective berthing energy	E_b	1421 kN·m(＝kJ)

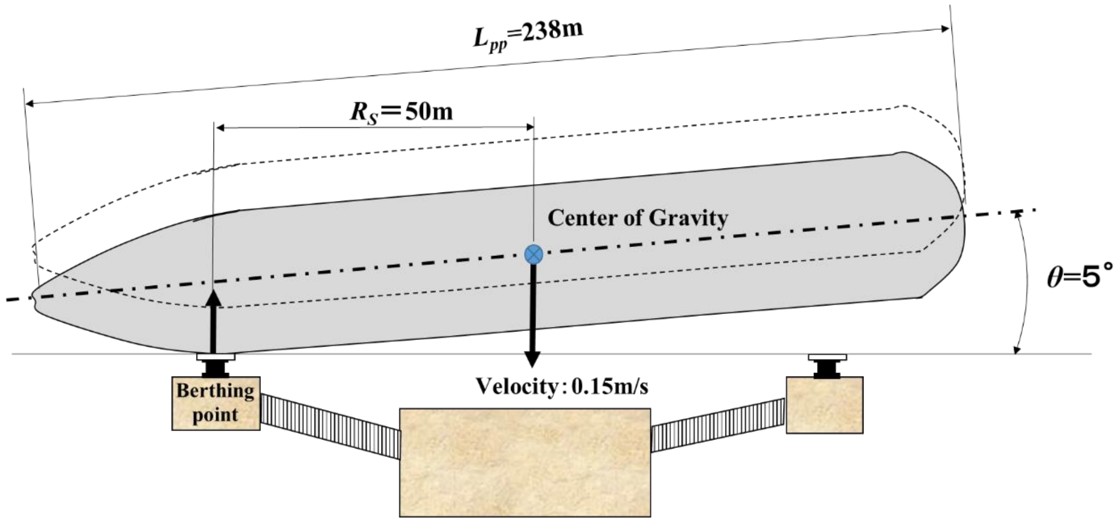


Fig. A.3.2.1 Berthing of tanker to dolphin

(3) Selection of rubber fender

A rubber fender of the vertical cylinder type with the configuration of 2000H (grade B) can absorb the effective berthing energy of the vessel. The standard performance from the catalogue is as follows.

<Standard performance>

Size: 2000H × 1 × 1 (Grade B)
Reaction force R_R: 1750 (kN)
Energy absorption E_A: 1540 (kN·m)

Considering the factor of influence for pattern B, the values for each factor are as follows, as obtained from the manufacturers' catalogues or technical data.

Manufacturing tolerance : $C_p^-=0.9$、$C_p^+=1.1$
Angular factor : $C_{aR}=1.00$、$C_{aE}=0.96$ (from Table 6.3.2.2 in Chapter 6; for angle of 5°)
Velocity factor : $VF_R=VF_E=1.125$ (from Table 6.3.7 in Chapter 6; 0.15 m/s=7.5%/s)
Temperature factor: $TF_E^-=1.00$, $TF_R^+=1.11$ (from Table 6.3.10.2 in Chapter 6; 31°C-5°C)

In this case, the influence factors of the reaction force and energy absorption are common.

1) Design reaction force

$$R^+=R_R\times C_p^+\times C_{aR}\times VF_R\times TF_R^+$$
$$=1750\times1.1\times1.00\times1.125\times1.11=2404\ \text{(kN)}$$

R^+ : Design reaction force after considering influence factors (kN)
R_R : Standard reaction force (kN)
C_p^+: Production tolerance (1.1)
C_{aR} : Angular factor of reaction force
VF_R : Velocity factor of reaction force
TF_R^+: Maximum temperature factor

2) Design energy absorption

$$E_A^{-} = E_A \times C_p^{-} \times C_{aE} \times VF_E \times TF_E^{-}$$
$$= 1540 \times 0.9 \times 0.96 \times 1.125 \times 1.00 = 1497 (\mathrm{kN \cdot m}) \geqq 1421 (\mathrm{kN \cdot m})$$

E_A^{-} : Design energy absorption after considering influence factors (kN・m)
E_A : Standard energy absorption (kN・m)
C_p^{-} : Production tolerance (0.9)
C_{aE} : Angular factor of energy absorption
VF_E : Velocity factor of energy absorption (DV, Common with VF_R)
TF_E^{-} : Minimum temperature factor

(4) Detail design

1) Required area of fender panel

As explained in Chapter 5, Section 5.7.1, the required effective area A of the allowable surface pressure H_P to the fender panel is as follows.

$$A = R^{+} / H_P = 2404 / 200 = 12.02\ (\mathrm{m}^2)$$

The size of the fender panel is determined by the number and spacing of the resin pads placed on the contact surface. Although these values differ depending on the manufacturer, the following fender panel is considered as an example.

Horizontal direction: 3560 (mm) (including both side chamfers of 100 mm)
Vertical direction: 3940 (mm) (including upper chamfer of 100 mm)
Thickness: 330 (mm) (including resin pad of 30 t)
Weight: 3698 (kg)
Average hull pressure: 186 (kN/m^2) (except peripheral chamfers of 100 mm)

2) Strength of fender panel

For the bending moment generated in the horizontal direction of the fender panel, consider the distributed load to be as shown in Fig. 5.7.8 in Chapter 5. The data generated by fender compression to the design deflection at an angle of 5° are as follows. Please note that since this calculation was performed using a software owned by a manufacturer, the detailed calculation steps are omitted.

Berthing point displacement δ_0: 1.15 (m)
Fender centre displacement δ: 1.02 (m) (Central deflection 51%,
At this time, deflection of the end becomes 55% the limit)
Design reaction force of fender R^+: 2404 (kN)
Rotational moment of fender M_f: -13.84 (kN·m)
Distance from fender centre to berthing point L_1: 1.68 (m) (except end chamfer of 100 mm)
Distributed load on the fender panel: $\omega(x) = -4.378x + 722.8$ (kN/m)
Distance from the end of panel x: See Fig. 5.7.8 in Chapter 5 (m)
Horizontal bending moment generated in panel: 1023.5 (kN·m)
Horizontal section modulus of panel: 10.821 (m^3)
Horizontal bending stress: 94.58 (kN/m^2)

In the case of a tanker, since the vertical angle is due to the rolling of the vessel, it is often sufficiently safe if the vertical strength is the same as the horizontal strength. However, in the case of berthing at the flare part of the hull or in the case of a vertically long panel, the vertical strength is dominant.

3) Weight chain

To support the weight of the fender panel, two chains are installed: one each on the left and right sides of the fender body, as shown on the right in Fig. 5.7.10 in Chapter 5. The tension T applied to the chain can be obtained by equation (A.3.2.1), assuming that the angle of elevation of the chain is β.

$$T = (\mu \cdot R^{+} + W_W) / \cos(\beta) = 619 \ (\text{kN}) \ (309 \ \text{kN/chain}) \tag{A.3.2.1}$$

Here,

μ : Friction coefficient between fender panel (resin pad) and vessel hull (=0.2)

β : Angle between chain and fender panel $\left(=\sin^{-1}\{(H-\delta)/L_c\}\right)$

R^{+} : Design reaction force (2404 kN)

H : Fender height (2000 mm)

δ : Deflection of fender (500 mm at maximum reaction force)

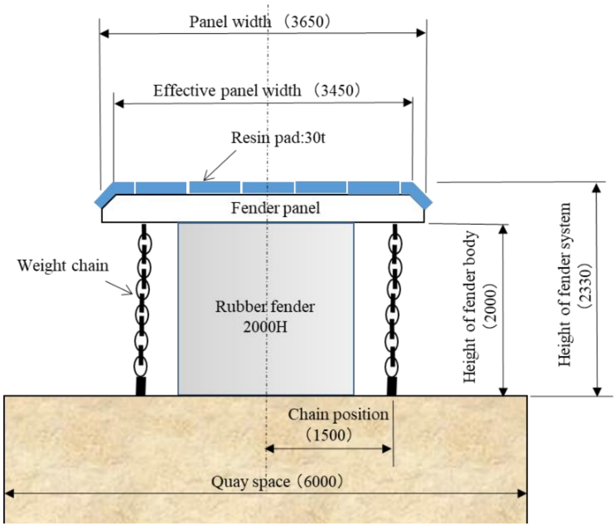
L_c : Length of chain (2730 mm; calculated length +1 to +3%)

W_W : Weight of fender panel (3698 kg)

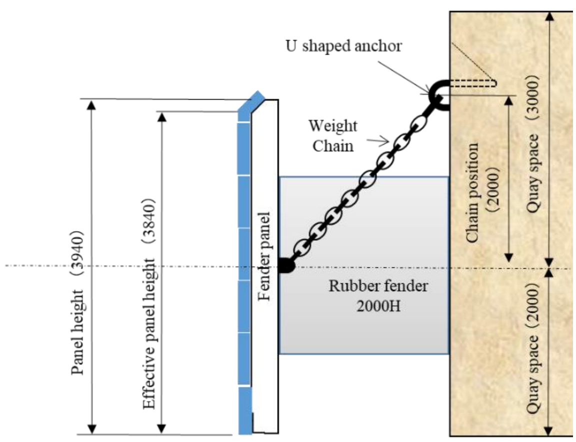
4) Space for rubber fender installation

The condition in which the rubber fender is attached to the quay is shown in Fig. A.3.2.2. Usually, the design of the dolphins is performed prior to the selection of rubber fenders, or often, the dolphin has already been constructed. In Fig. A.3.2.2, both the rubber fenders and chains are contained in the quay mounting space. However, if the top of the quay is low, it will not be possible to secure a concrete thickness commensurate with the embedded depth of the U-shaped anchor. Generally, it is recommended that the concrete is secured with the same thickness as the embedding depth assuming a 45° cone failure; however, this aspect is difficult to be realized when the quay top is excessively low. Fig. A.3.2.3 shows an example of the countermeasure when the concrete cover cannot be secured. A corresponds to the method of bevelling the space for the U-shaped anchor at the corner of the quay to ensure oblique embedding. B corresponds to the method of lowering the position of the weight chain; in this case, the fender panel may incline forward, and thus, the upper end needs to be tensioned using a tension chain (see Chapter 5, Fig. 5.7.10). C shows the method of building a post on the quay top to ensure attachment for the weight chain when other methods are not sufficient. As described above, it is desirable to consider not only fixing bolts for installation but also employing various chain arrangement in advance in the space.

The specifications of the weight chain are determined by the fender size and weight of the fender panel. It is necessary to consider this aspect, as judgment criteria exist for each rubber fender and manufacturer.

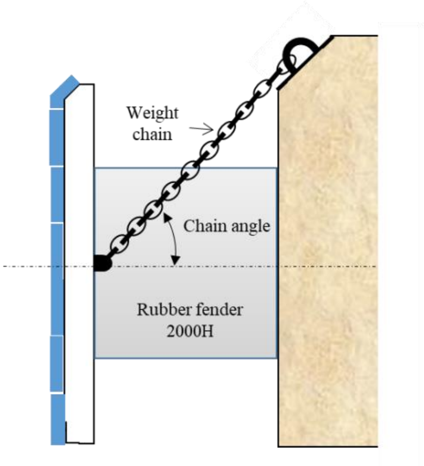


Plain view

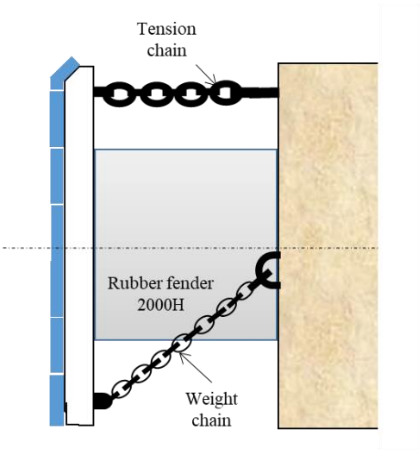


Side view

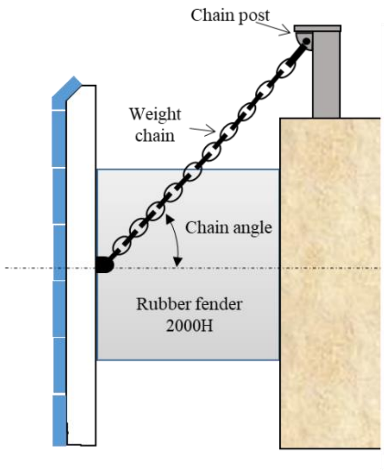
Fig. A.3.2.2 Installation of rubber fender to dolphin (2000H×1×1)



A: Chamfered quay top



B: Lowering weight chain



C: Chain post on the top

Fig. A.3.2.3 Possible countermeasures for weight chain arrangement

A.3.3 Long distance ferry of 10,000 *DWT*: With large fender panel

Rubber fenders for ferries are often attached to a large-sized panel and installed on a parapet on the quay. The quay is expected to possess a sufficiently large horizontal strength against an earthquake if it is a gravity type; however, it is necessary to carefully estimate the maximum reaction force when it is necessary to install the chain on the front of a parapet.

This section presents a design example in which the influence factor pattern B is considered assuming the abovementioned case.

(1) Design conditions

1) The vessel data is presented in Table A.3.3.1.

Table A.3.3.1 Vessel data

Item	Condition	Unit
Type of vessel	Long distance ferry	
Dead weight tonnage : *DWT*	10,000	t
Gross tonnage : *GT*	23,520	t
Length between perpendiculars : L_{pp}	182.70	m
Beam of vessel : *B*	28.20	m
Full draught of vessel : *d*	2.27	m
Berthing angle : θ	6.00	°
Berthing velocity : V_B	0.15	m/s

2) It is assumed that the following constraints exist on the size of the rubber fender.

Allowable average hull pressure: No limit
Horizontal load capacity of quay (Parapet): 2000 (kN) (per fender system)
Minimum size of fender panel: 4 m × 4 m or more
Maximum height: Within 1.2 m including fender panel

In the case of a ferry, hull belts exist, and the vessel hull rarely makes full contact with the fender panel. However, the tugboat assistance is often not available, a wide fender panel is preferred as the target, and berthing velocity can also be high. Furthermore, in several cases, height restrictions exist to ensure boarding and deboarding of passengers.

3) Temperature: 5°C–30°C

(2) Effective berthing energy

The calculation results for the effective berthing energy are presented in Table 3.3.2.

Table 3.3.2 Berthing data
(from equation (5.3.1) of Section 5.3 in Chapter 5)

Displacement tonnage	DT	29,165 t
Block coefficient	C_b	0.50
Virtual mass factor	C_m	1.25
Eccentricity factor	C_e	0.50
Softness factor	C_s	1.00
Berth configuration factor	C_c	1.00
Effective berthing energy	E_b	206 kN·m（=kJ）

(3) Selection of rubber fenders

A vertical cylinder type rubber fender with a size of 1000H (performance grade: standard) is capable of absorbing the effective berthing energy of vessels. The standard performance from the catalogue is as follows.

<Standard performance>
Size : 1000H × 1 × 1 (single unit)
Reaction force : 625 (kN) (+10%)
Energy absorption : 224 (kN・m) (−10%)

However, using only one 1000H unit on a 4 m × 4 m fender panel leads to poor balance. Therefore, if two units are arranged vertically and horizontally, the required amount of absorbed energy per unit becomes 206 / 4 = 51.5 (kN・m), and the rubber fender is downsized to 630H. The standard performance of the 630H fender is as follows.

<Standard performance>
Size : 630H × 2 × 2 (4 unit)
Reaction force : 249×4＝996 (kN) (+10%)
Energy absorption : 56.3×4＝225.2 (kN・m) (−10%)

However, the above selection is based on the premise that the vessel hull is in full contact with the fender panel at an angle of 0° in the abovementioned cases. In fact, when the vessel is berthing at an angle of 6°, the time of point contact at the corner may be more than that of the full contact. It is thus necessary to select a fender considering this aspect as well, as the fender will absorb less energy than a standard fender.

If corner point contact is assumed, the balance between force and bending moment must be considered, as described in Fig. 5.7.7 in Chapter 5, Section 5.7.4. The correct size is required to be determined via trial and error; however, assuming that the rubber fender has a size of 800H, the arrangement is as shown in Fig. A.3.3.1.

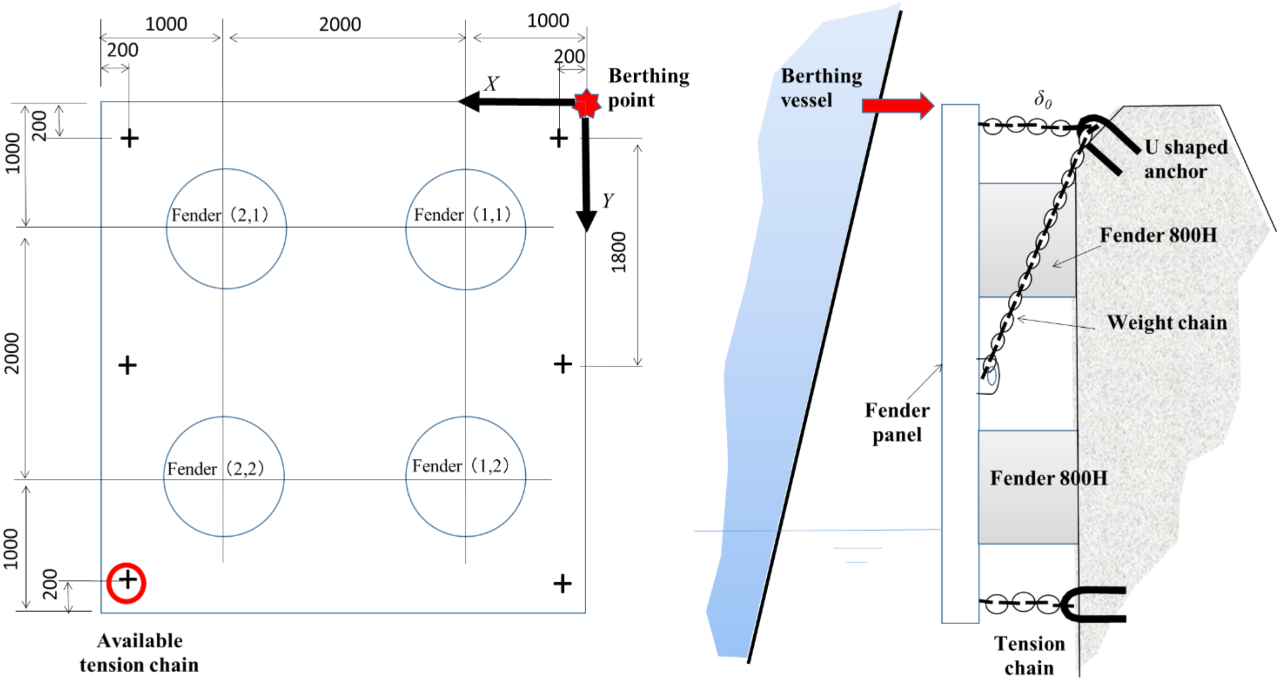


Fig. A.3.3.1 Fender system concept (800H×2×2 for ferry)

The equations of balance between the forces and bending moments, from equation (5.7.7) in Chapter 5, Section 5.7.4, are as follows. Please note that the rotational moments of the rubber fenders are ignored.

$$F + T + \sum_{i=1}^{2} \sum_{j=1}^{2} R(i,j) = 0 \tag{A.3.3.1}$$

$$\sum_{i=1}^{2} \{X(i) \cdot \sum_{j=1}^{2} R(i,j)\} + X(3) \cdot T = 0 \tag{A.3.3.2}$$

$$\sum_{j=1}^{2} \{Y(j) \cdot \sum_{i=1}^{2} R(i,j)\} + Y(3) \cdot T = 0 \tag{A.3.3.3}$$

Here,

X(i) : *X* coordinate of *i*-th rubber fender in *X* direction (*i* = 3 is chain position)
Y(j) : *Y* coordinate of *j*-th rubber fender in *Y* direction (*j* = 3 is chain position)
R(i,j) : Reaction force of rubber fender (*i*,*j*) (kN)

It is desirable that the reaction force *R* (*i*, *j*) of the rubber fender for any displacement δ is represented as a polynomial in the manufacturer's catalogue or technical data. Additionally, the geometric relationships of the displacement amounts are as follows.

$$\begin{aligned}
\delta_{(1,1)} &= \delta_0 - X(1) \cdot tan\theta_X - Y(1) \cdot tan\theta_Y \\
\delta_{(1,2)} &= \delta_0 - X(1) \cdot tan\theta_X - Y(2) \cdot tan\theta_Y \\
\delta_{(2,1)} &= \delta_0 - X(2) \cdot tan\theta_X - Y(1) \cdot tan\theta_Y \\
\delta_{(2,2)} &= \delta_0 - X(2) \cdot tan\theta_X - Y(2) \cdot tan\theta_Y
\end{aligned} \tag{A.3.3.4}$$

θ_X and θ_Y are angles of the horizontal and vertical directions of the fender panel, respectively.
The standard performance of the vertical cylinder fender 800H is as follows.

<Standard performance>
Size : 800H × 2 × 2 (4 units, Grade A)
Design deflection : Within 55%

Reaction force : $280 \times 4 = 1120$ (kN)
Energy absorption : $104 \times 4 = 416$ (kN·m)

Considering the influence factor in Pattern B, the velocity factor and temperature factor are obtained from the manufacturer's catalogue or technical data as follows.

Manufacturing tolerance: $C_p^- = 0.9$, $C_p^+ = 1.1$
Angular factor: 1.00 (Berthing angle 6°; however, it is not considered as the inclination to 6° is gradual)
Velocity factor: $VF_R = 1.20$, $VF_E = 1.17$
(DV deceleration performance with V = 18.75%/s; refer to Chapter 6, Table 6.3.7)
Temperature factor: $TF_R = TF_E = 1.00$-1.11 (30℃-5℃; see Chapter 6, Table 6.3.10.2)

Therefore, the maximum reaction force acting on the parapet is 1120 × 1.1 × 1.20 × 1.11 = 1641 kN (410 kN per unit). The angular factor is assumed to 1.0 because the angle changes according to the process of compression. The performance curve of the fender unit is as shown in Fig. A.3.3.2. Since the compression velocity varies depending on the position, the minimum value of velocity factor (1.0) is assumed. Thus, the minimum value of the design reaction force is 1120 × 0.9 × 1.0 × 1.0 = 1008 kN (252 kN per unit).

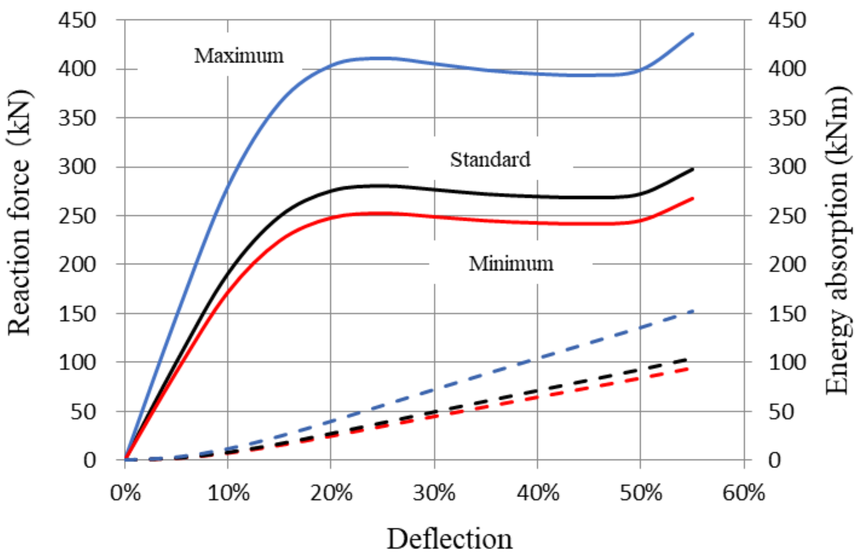


Fig. A.3.3.2 Performance of vertical cylinder 800H (Single unit)

If the reaction force of the rubber fender in Fig. A.3.3.2 is substituted into a mathematical expression as $R(i,j)$ in equation (A.3.2.1), the berthing point displacement δ_0, berthing force F, and chain tension T can be calculated numerically. The solution of equations (A.3.3.1) to (A.3.3.4) for the system in Fig. A.3.3.1 is presented below. Table A.3.3.3 and Table A.3.3.4 respectively present the results for the maximum and minimum values of the reaction force.

Table A.3.3.3 Performance of each fender and total system (maximum influence factors)

Total system						Fender (1, 1)			Fender (1, 2)			Fender (2, 1)			Fender (2, 2)		
Displacement δ_0	Horrizontal angle of panel	Vertical angle of panel	Chain tension T	Berthing force F	Energy absorption E	Deflection δ	Reaction force R	Energy absorption E	Deflection δ	Reaction force R	Energy absorption E	Deflection δ	Reaction force R	Energy absorption E	Deflection δ	Reaction force R	Energy absorption E
mm	deg	deg	kN	kN	kN×m	%	kN	kN×m	%	kN	kN×m	%	kN	kN×m	%	kN	kN×m
0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
59	-0.44	-0.44	178.42	217.47	6.29	5.43	155.94	3.33	3.49	98.90	1.35	3.49	98.90	1.35	1.55	42.16	0.26
118	-0.89	-0.89	355.37	421.65	25.37	10.87	291.01	13.22	6.99	199.31	5.54	6.99	199.31	5.54	3.11	87.38	1.06
177	-1.33	-1.33	500.59	568.72	54.90	16.30	369.72	27.80	10.48	283.12	12.33	10.48	283.12	12.33	4.66	133.35	2.43
236	-1.78	-1.78	606.50	656.06	91.29	21.74	398.40	44.64	13.97	343.09	21.14	13.97	343.09	21.14	6.21	177.99	4.37
295	-2.22	-2.22	677.89	699.70	131.45	27.17	398.97	62.03	17.47	379.42	31.29	17.47	379.42	31.29	7.76	219.77	6.84
354	-2.67	-2.67	723.91	718.46	173.37	32.61	391.56	79.22	20.96	396.57	42.17	20.96	396.57	42.17	9.32	257.66	9.81
413	-2.75	-3.47	752.49	724.85	216.00	38.04	386.22	96.11	22.87	400.06	48.26	26.04	400.06	58.41	10.87	291.01	13.22
472	-3.55	-3.55	772.26	727.23	258.88	43.47	383.90	112.84	27.95	398.04	64.51	27.95	398.04	64.51	12.42	319.50	17.02
531	-4.00	-4.00	786.44	729.55	301.82	48.91	386.61	129.55	31.44	393.14	75.56	31.44	393.14	75.56	13.97	343.09	21.14
590	-4.44	-4.44	**804.70**	**751.61**	**345.33**	54.34	416.71	146.83	34.93	388.82	86.49	34.93	388.82	86.49	15.53	361.96	25.52

Table A.3.3.4 Performance of each fender and total system (minimum influence factors)

Total system						Fender (1, 1)			Fender (1, 2)			Fender (2, 1)			Fender (2, 2)		
Displacement δ_0	Horrizontal angle of panel	Vertical angle of panel	Chain tension T	Berthing force F	Energy absorption E	Deflection δ	Reaction force R	Energy absorption E	Deflection δ	Reaction force R	Energy absorption E	Deflection δ	Reaction force R	Energy absorption E	Deflection δ	Reaction force R	Energy absorption E
mm	deg	deg	kN	kN	kN×m	%	kN	kN×m	%	kN	kN×m	%	kN	kN×m	%	kN	kN×m
0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
59	−0.44	−0.44	112.41	137.01	3.96	5.43	98.24	2.10	3.49	62.31	0.85	3.49	62.31	0.85	1.55	26.56	0.16
118	−0.89	−0.89	223.88	265.64	15.98	10.87	183.34	8.33	6.99	125.57	3.49	6.99	125.57	3.49	3.11	55.05	0.67
177	−1.33	−1.33	315.37	358.29	34.58	16.30	232.92	17.51	10.48	178.37	7.77	10.48	178.37	7.77	4.66	84.01	1.53
236	−1.78	−1.78	382.10	413.32	57.51	21.74	250.99	28.12	13.97	216.15	13.32	13.97	216.15	13.32	6.21	112.13	2.75
295	−2.22	−2.22	427.07	440.81	82.81	27.17	251.35	39.08	17.47	239.03	19.71	17.47	239.03	19.71	7.76	138.46	4.31
354	−2.67	−2.67	456.06	452.63	109.22	32.61	246.68	49.91	20.96	249.84	26.57	20.96	249.84	26.57	9.32	162.33	6.18
413	−2.75	−3.47	474.07	456.65	136.08	38.04	243.32	60.55	22.87	252.04	30.40	26.04	252.04	36.80	10.87	183.34	8.33
472	−3.55	−3.55	486.52	458.16	163.09	43.47	241.86	71.09	27.95	250.77	40.64	27.95	250.77	40.64	12.42	201.28	10.72
531	−4.00	−4.00	495.46	459.62	190.15	48.91	243.56	81.62	31.44	247.68	47.60	31.44	247.68	47.60	13.97	216.15	13.32
590	−4.44	−4.44	**506.96**	**473.51**	**217.56**	54.34	262.53	92.50	34.93	244.96	54.49	34.93	244.96	54.49	15.53	228.03	16.08

The final performance at the corner point berthing is as follows.

<Corner point berthing performance>

Size: 800H × 2 × 2

Final displacement at berthing point: 590 (mm)

Reaction force at berthing point: 752 (kN) from Table A.3.3

Total energy absorption: 218 (kN・m) ≧206 from Table A.3.3.4

Tension of chain: 805 (kN) from Table A.3.3.3

Final fender panel angle: 4.4° (common to vertical and horizontal) <design condition 6°

The performance curve for the entire system is shown in Fig. A.3.3.3.

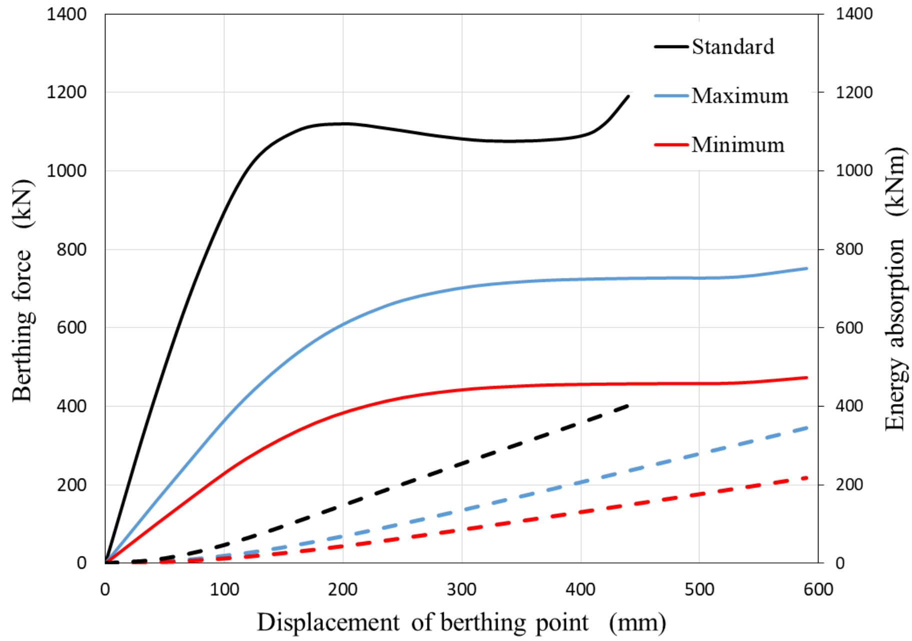


Fig. A.3.3.3 Performance of fender system: vertical cylinder 800H × 2 × 2

Since the fender panel angle (4.44°) is less than the berthing angle (6°), the berthing energy is absorbed during point contact at the corner of the fender panel. Although this system is not an efficient one from the viewpoint of energy absorption, it can be a rational system in view of the requirements of fenders for ferries. As seen in Fig. A.3.3.3, the curve of berthing force considering the angle differs from the standard performance. For example, when the fender is in contact with a location at which the vessel has a flare angle, it is more accurate to use the reaction force by deflection, as shown in Fig. A.3.3, for a mooring simulation. Such consideration is possible when the flare angle of the contact point is known.

(4) Weight chain

The tension of the weight chain is calculated as follows. Assuming that the friction coefficient μ between the hull and fender panel is 0.3 when the fenders are compressed uniformly and the hull moves downward, the tension T applied to weight chain is as shown in Fig. A.3.3.4. The tension T is obtained using equation (A3.3.5).

$$T = \left[\mu\left\{\sum_{i=1}^{2}\sum_{j=1}^{2} R(i,j)\right\} + W_W\right] \Big/ cos(\beta) = 544 \text{ (kN) (272 kN per chain)} \qquad \text{(A.3.3.5)}$$

Here,

μ : Friction coefficient μ between hull and fender panel (=0.3)

Because ferries often have a hull belt, which comes into contact with the steel plate without resin pads on the fender panel, μ was set to 0.3.

β : Angle between chain and fender panel $\left(=\sin^{-1}\left\{(H-\delta)/L_c\right\}\right)$

H : Height of fender (800 mm)

Strictly, the height of the fender depends on the position of the eye plate and U-shaped anchor; thus, this height is the centre-to-centre distance of the shackle pin.

δ : Design deflection of fender (440 mm)

L_c : Length of chain (1200 mm)

W_W : Weight of fender panel (4000 kg)

$\sum_{i=1}^{2}\sum_{j=1}^{2} R(i,j)$: Total reaction force of fenders (1600 kN)

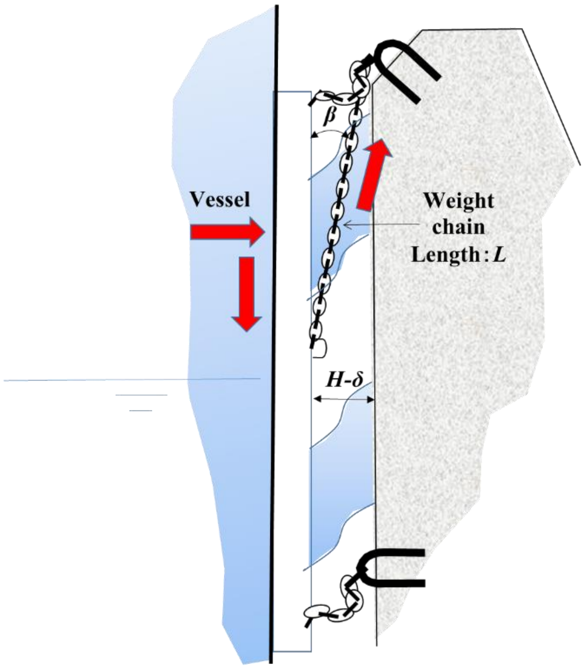


Fig. A.3.3.4 Weight chain design

As a result, the maximum tension occurred at 55% of the fender height with a value of 272 kN per chain. In such a system, the tension is high. In particular, since a tension chain is subjected to a tension of 805 kN, it is possible that the number of links cannot be secured. In this case, it is necessary to review the entire system. The allowable tension of the tension chain can be considered as a short-term load, as described in Chapter 5.9, and the safety factor can be set as 2. In weight chain design, the long-term load (safety factor 3.0 or more) considering the panel weight and the short-term load during shear compression (safety factor 2.0 or more) calculated using equation (A.3.3.5) are compared, and design is performed considering severe conditions.

A.4 Pattern C: Fender design by mooring analysis

When using factor pattern C, in certain cases, a floating structure such as a floating pier, floating oil storage, or floating bridge is permanently moored, or a vessel is moored to a quay during a storm or long-period wave. In this work, as an example of rubber fender design considering the vessel motion, a numerical simulation is performed to estimate whether the vessel can be safely moored under rough weather conditions by rubber fenders and mooring ropes. Normally, during stormy weather such as a typhoon, ships evacuate offshore; however, depending on the situation, the ships may remain moored to the wharf to allow the alleviation of congestion in the water area. The process flow of this investigation is shown in Fig. A.4.1.

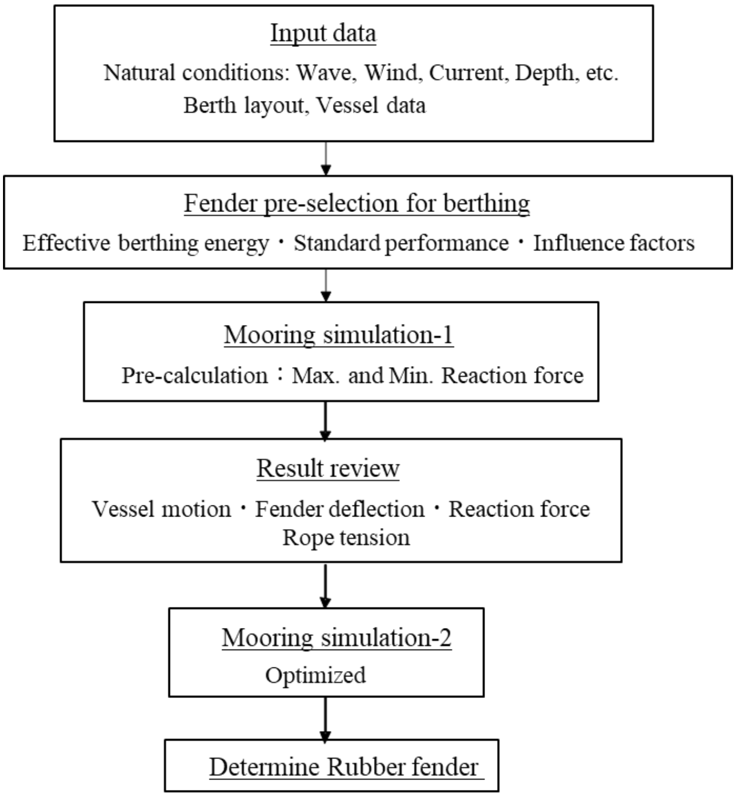


Fig. A.4.1 Flow of fender design by mooring simulation

The example vessel is assumed to be a medium-sized container vessel of approximately 10,000 *DWT*, as shown in Fig. A.4.2. It is assumed that the rubber fenders selected considering the effective berthing energy are already installed in the same manner as described in this book, and whether the fender will be compressed beyond the design deflection in rough weather has been confirmed via mooring simulation.

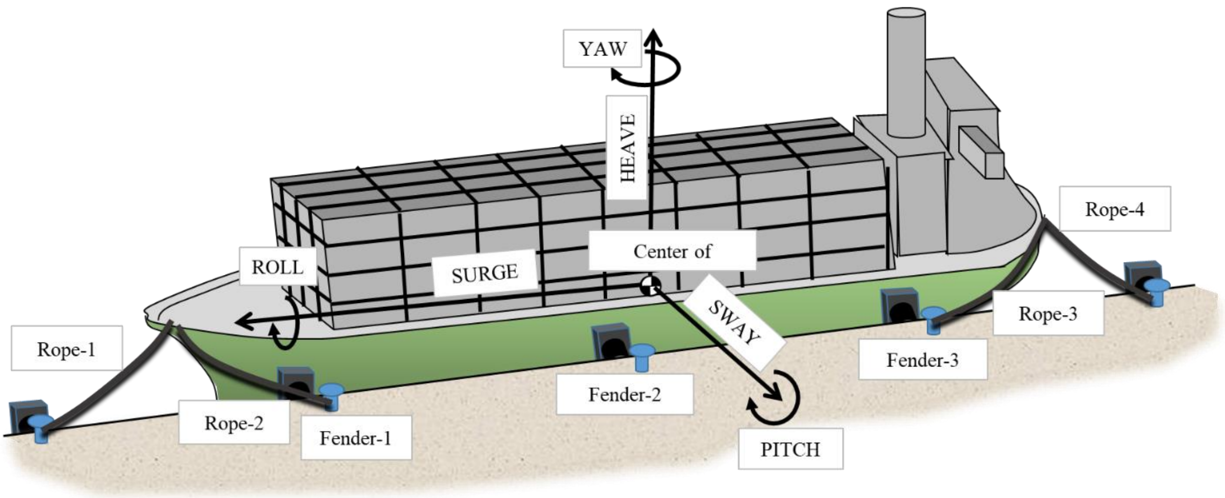


Fig. A.4.2 Birds-eye view of moored medium-sized container vessel

(1) Motion equation of moored vessel

The equation of motion for the moored vessel as shown in Fig. A.4.2 can be written as equation (A.4.1) [1].

$$\sum_{\mathrm{i}=1}^{6}\{M_{ij}+m_{ij}(\infty)\}\ddot{X}_i(t)+\sum_{\mathrm{i}=1}^{6}\left\{\int_{-\infty}^{t}L_{ij}\,(\mathrm{t}-\tau)\dot{X}_i(\tau)\mathrm{d}\tau+N_{ij}(\mathrm{t})X_i\,(t)\right\}+\sum_{i=1}^{6}(K_{ij}+G_{ij})X_i(t)=F_j(t)$$

(A.4.1)

Here,

M_{ij} : Mass matrix of moored vessel

$m_{ij}(\infty)$: Additional mass matrix of moored vessel

$X_i(t)$: Displacement vector of vessel centre at time t

$L_{ij}(t)$: Delay function of moored vessel at time t

$N_{ij}(t)$: Dumping coefficient matrix of moored vessel at time t

K_{ij} : Restoration coefficient matrix of moored vessel

G_{ij} : Mooring force matrix

$F_j(t)$: External force vector

i と j : Mode of vessel motion (i,j=1-6)

The delay function and additional mass can be expressed as follows:

$$L_{ij} = \frac{2}{\sigma}\int_0^{\infty} B_{ij}\,(\sigma)\cos\sigma t d\sigma \qquad \text{(A.4.2)}$$

$$m_{ij}(\infty) = A_{ij}(\sigma) + \frac{1}{\sigma}\int_0^{\infty} L_{ij}\,(t)\sin\sigma t dt \qquad \text{(A.4.3)}$$

Here

$A_{ij}(\sigma)$: Additional mass at σ

$B_{ij}(\sigma)$: Dumping coefficient at σ

σ : Angular frequency

The solution of equation (A.4.1) is complicated, and it is more convenient to use commercially available software than a newly developed one. Several programs are available; however, it is desirable that programs verified using hydraulic experiments or field measurement are used, if possible.

(2) Input data

When designing rubber fenders by using mooring simulations, the most crucial step is the setting of the conditions and calculation cases for the simulation. Hereinafter, calculation examples are described along with the procedure.

1) Natural conditions

In principle, natural conditions that are severe and occur frequently for rubber fenders should be selected based on local survey results. In this work, two types of conditions are assumed: one involving strong wind as a condition of stormy weather (denoted as strong wind), and another involving long period swells, in which the wave height and period are set to be large and long, respectively. To set severe conditions for rubber fenders, the wind is considered to be offshore wind blowing perpendicular to the long axis of the vessel, and the waves are at 45° from the short axis of the vessel to the stern, considering the compression of the rubber fender by vessel yawing. It is assumed that this condition corresponds to that shown in Fig. A.4.3. It was assumed that no current was present.

A list of natural conditions is presented in Table A.4.1.

Table A.4.1 Natural conditions

Significant wave height	0.5 m	0.8 m
Wave period	6 s	10 s
Wave spectrum	Bretschneider–Mitsuyasu	
Wave direction：θw	45°	45°
Average wind speed (10 min)	25 m/s	15 m/s
Wind spectrum	Davenport	
Wind friction coefficient	0.003	
Wind drag coefficient	0.898	
Wind direction: θa	0°	0°

2) Berth layout

Fig. A.4.3 shows the berth layout in plain view. In the two cases—10,000 *DWT* medium-sized container vessel and 5,000 *DWT* small-sized container vessel—full loading and empty loading are considered. It is easy to believe that the smaller ship is safer; however, it is necessary to confirm the more severe condition because only 2 units are in contact with the 5,000 *DWT*, while 3 rubber fenders come in contact with the 10,000 *DWT*. The cross-section is as shown in Fig. A.4.4.

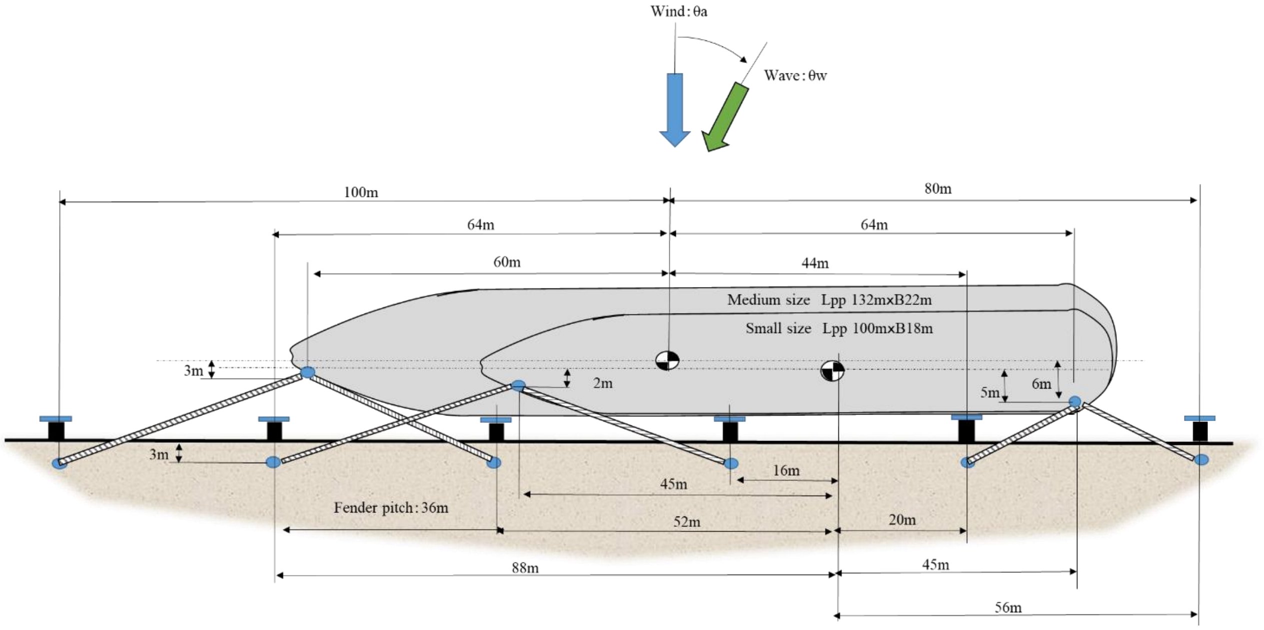
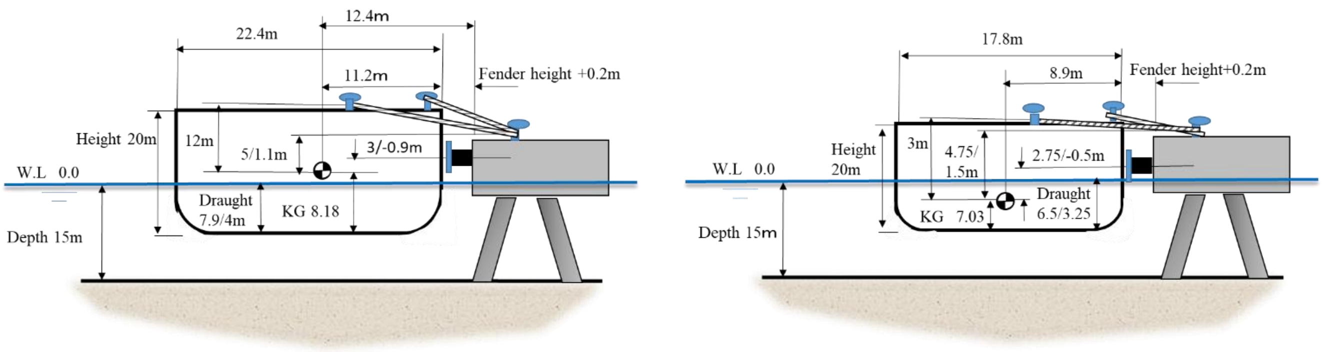


Fig. A.4.3 Berth layout: Plain view



Medium container vessel 10,000 *DWT*

Small container vessel 5,000 *DWT*

Fig. A.4.4 Berth layout: Cross-section

3) Vessel data

Table A.4.2 presents the vessel data of the medium and small-sized container vessels. The vessel data are derived primarily from the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [4] and the latest statistics [3]; however, some data must be investigated or assumed. In Table A.4.2, the yellow highlighted red numbers within the bold frame are less common and must be researched. The overall height *D* (type depth) is also often omitted in recent statistics.

In berthing design, it is often not necessary to consider the ballast condition because the mass of an empty vessel is considerably less. However, when mooring, the ballast is susceptible to the wind when the vessel is empty and has a wider area exposed to the wind. This situation is difficult for the fenders. The deflection of rubber fenders at the ballast condition is usually large and severe in the medium-sized vessel; thus, calculations for the small vessel were performed considering only the ballast condition.

Table A.4.2 Vessel data for container

	Vessel size-Draught		
Data	Medium-Full	Medium-Ballast	Small-Ballast
Effective berthing energy(kN·m)	162.3	162.3	84.6
Dead Weight tonnage : *DWT*	10,000	10,000	5,000
Displacement tonnage: *DT*	15,017	6,184	3,217
Gross tonnage : *GT*	8,820	8,820	4,410
Length between perpendiculars: *Lpp* (m)	131.83	131.83	99.8
Beam : *B* (m)	22.4	22.4	18.37
Height : D_s (m)	20	20	9
Draught : *d* (m)	8.02	4	3.25
Freeboard : *f* (m)	12	16	5.75
Water depth (m)	15	15	15
KG (m)	8.18	8.18	7.03
GMT (m)	1.73	3.63	2.92
GML (m)	172.5	172.5	120.82
Radius of Gyration-Roll (m)	7.84	7.84	6.49
Radius of Gyration-Pitch (m)	32.96	32.96	24.96
Radius of Gyration-Yaw (m)	32.96	32.96	324.96
Center of floatation*1 (m)	0.53	0.528	0.459
Wind pressure area-Surge (m^2)	409.9	499.6	329
Wind pressure area-Sway (m^2)	1882.8	2410.9	1480.8
Natural period-Roll (s)	11.95	8.25	7.62

*1 Distance from center of floatation to center of gravity in surge direction

(3) Selection of rubber fenders and determination of influence factors

The influence factors of rubber fender performance described in Chapter 4 were selected as follows, and the maximum and minimum values of the influence factors were estimated.

Manufacturing tolerance : $C_p^-=0.9$、$C_p^+=1.1$

Angular factor: Maximum angle: 10°($C_{aR}^-=0.98$), Minimum angle: 0°($C_{aR}^+=1.0$)

Velocity factor: Velocity changes during mooring. Here, it is assumed that one deflection cycle is 55%×4 and takes 6 s (Wave period), and the average strain rate is estimated as 30%/s. The velocity factors are $VF_R^-=1.0$, $VF_R^+=1.236$.

Temperature factor: Temperature ranges from 0℃ to 30℃; thus, the temperature factors are $TF_R^-=0.984$、$TF_R^+=1.116$

Ageing factor: From the Technical Standards and Commentaries of Ports and Harbour Facilities in

Japan-II Floating Oil Storage Facility [5], the ageing factors are $C_{ag}^{-}=1.0$、$C_{ag}^{+}=1.05$

Repetition fatigue: From the Technical Standards and Commentaries of Ports and Harbour Facilities in Japan-II Floating Oil Storage Facility [5], the repetition factors are $C_r^{-}=0.9$、 $C_r^{+}=1.0$

Creep characteristic: It was confirmed that the average value of wind pressure was within the reaction force of 10% deflection as follows.

Wind pressure:

$$F_w=0.5\times\rho\times U_a^2\times C_w\times A_{sway}= 277\ \text{(kN) (Medium-Ballast, 3 fenders)}$$
$$< 299\ \text{(kN) (1000H at 10\%)}, 323\ \text{(kN) (1250H at 10\%)} \qquad \text{(A.4.4)}$$

Here,

F_w: Wind pressure (kN)

ρ : Air density (=0.00123 t/m^3)

U_a: Wind speed (=25 m/s)

C_w: Wind drag coefficient of sway (from Table A.4.1, C_w =0.898)

A_{sway}: Wind pressure area of sway (from Table A.4.1, Medium-Ballast: A_{sway} =2411 m^2)

The performance curves of the rubber fender assumed in this case is shown in Fig. A.4.5, and the specifications are listed in Table A.4.3. The minimum value of energy absorption and the maximum reaction force are considered to realize the absorption of the effective berthing energy. When designed at berthing with an angle of 10°, the berthing energy can be sufficiently absorbed if the fenders have a size of 1000H; however, calculation for the vessel motion at mooring indicates that larger fenders (1250H) are necessary. Since the mooring simulation is a type of numerical experiment, the appropriate size and performance cannot be obtained unless trial and error is performed. In this case, two types of sizes are examined assuming that the rubber fender is required to be eventually increased in size to 1250H. As shown in Fig. A. 4.5, the reaction force of over-compression is assumed to exhibit a linear increase to ensure the accuracy above the allowable deflection of rubber fender is low.

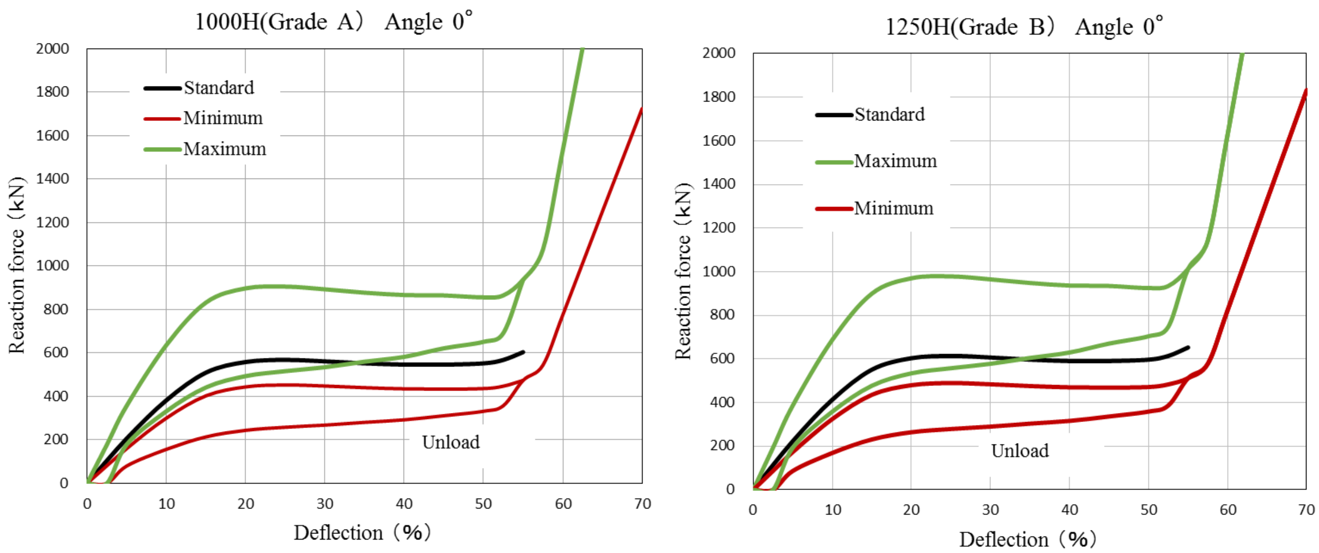


Fig. A.4.5 Performance curve of rubber fenders

Table A.4.3 Design performance of fenders and influence factors

Fender size	1000H	1250H	Remarks
Performance grade	Grade A	Grade B	Allowable deflection: 55%
Unit	3	3	Allowable deflection: 55%
0° Normal performance			
Standard reaction force	568kN	614kN	Influence factor
Minimum reaction force	453kN	489kN	$C_p{}^{-}(0.9)\times C_{aR}{}^{-}(0.98)\times VF_R{}^{-}(1.0)\times$ $TF_R{}^{-}(0.984)\times C_r(0.9)\times C_{ag}{}^{-}(1.0)$
Maximum reaction force	905kN	979kN	$C_p{}^{+}(1.1)\times C_{aR}{}^{+}(1.0)\times VF_R{}^{+}(1.236)\times$ $TF_R{}^{+}(1.116)\times C_r(1.0)\times C_{ag}(1.05)$
10°Angular performance			
Standard reaction force	643kN	589kN	Influence factor
Minimum reaction force	506kN	547kN	$C_p{}^{-}(0.9)\times C_{aR}{}^{-}(0.98)\times VF_R{}^{-}(1.0)\times$ $TF_R{}^{-}(0.984)\times C_r(0.9)\times C_{ag}{}^{-}(1.0)$
Maximum reaction force	927kN	1003kN	$C_p{}^{+}(1.1)\times C_{aR}{}^{+}(1.0)\times VF_R{}^{+}(1.236)\times$ $TF_R{}^{+}(1.116)\times C_R(1.0)\times C_{ag}(1.05)$
Minimum energy absorption	182kNm	246kNm	$C_p{}^{-}(0.9)\times C_{aE}{}^{-}(0.98)\times VF_E{}^{-}(1.0)\times$ $TF_E{}^{-}(0.984)\times C_r(0.9)\times C_{ag}{}^{-}(1.0)$

(4) Mooring rope

Along with rubber fenders, mooring ropes also have a substantial impact on vessel motion. However, the tension generated in the mooring rope is small because the wind and waves are directed from the offshore to evaluate the safety of fenders in this case. Therefore, the mooring ropes were of the following types with a total of four ropes: one bowline, one stern line, and two spring lines were deployed, as shown in Fig. A.4.3. The specifications of the ropes are as follows:

Rope specification: 2 strands Nylon ϕ 60
Breaking load: 920 (kN)

(5) Result of mooring simulation

The simulation results are presented in Table A.4.4. The calculation cases are as follows: 2 cases for different natural conditions (strong wind, long wave), 3 cases for different ship specifications (medium size-full draught, medium size ballast and small size-ballast), 2 cases for different sizes of rubber fenders (1000H and 1250H), and in total, 24 preliminary calculation cases with maximum and minimum reaction forces. First, two cases with 1000H rubber fenders were calculated, and in Case 2 (Ballast), fender No. 1 was overcompressed, exceeding the allowable deflection. Therefore, the size of rubber fenders was changed to 1250H, and subsequently, the calculation was performed for seven optimized cases listed in Table A.4.4. Each calculation was performed for 100 waves (600 s, 1000 s) in increments of 0.1 s. The Technical Standards and Commentaries of Ports and Harbour Facilities in Japan [4] recommends a calculation time corresponding to more than 100 waves. In addition, the standard deviations were obtained from the calculation results of the fender deflection and reaction force, and the expected value for 1000 waves was also calculated.

First, no problems were encountered because the maximum deflection of the rubber fender is 26% and the expected value of the 1000 wave simulation is 43% when calculated considering the medium-sized vessel full draught and strong wind conditions in Case 1. However, when the ballast is loaded as in Case 2, the corresponding value is 63% and the fender is overcompressed. Although the size of 1000H (height of 1 m) was sufficient for berthing, it is insufficient for mooring in stormy weather. Therefore, after Case 3, the size of fender was increased to 1250 H (1.25 m in height) and the rubber was downgraded (Grade A to Grade B); moreover, the fender reaction force adopted the minimum value of the influence factors. The result of Case 3 corresponds to less than the allowable deflection; however, in case 4, which involves ballast loading, the value for fender 1 is 51%, which is close to the limit (55%), and the 1000 wave expected value is 56%, which is slightly more than 55%. In the ballast case of a small vessel of Case 5, the maximum value is 51%, and the expected value of 1000 waves is 53%; both these values are within the limit of allowable deflection. The maximum deflection for the case of a medium-sized vessel-full draught with a wave having a height and period of 0.8 m and 10 s in Case 6 is 36%; however, the 1000 wave expected value is 57%, which slightly exceeds the allowable deflection. It is necessary to ensure that severe conditions are not overlooked, since the motion of the moored vessel changes in a complex manner with change in the condition. It is up to the designers to make the final decision. The maximum reaction force generated is in case 4 in which a medium-sized vessel is ballast loaded under strong wind conditions. Case 7 corresponded to the same conditions except that the fenders had a high reaction force. As a result, the maximum value was 979 kN, and the expected value of 1000 waves was 1804 kN. If this value exceeds the allowable horizontal force of the quay structure, it is necessary to recalculate using a different performance grade of a larger sized fender with a lower reaction force.

Table A.4.4 Final result for mooring simulation

Case	Natural condition	Vessel			Rubber fender		Fender Deflection (%)		Reaction force (kN)		Mooring rope		
		Vessel condition	Direction	Maximum motion	Fender specification	Fender No.	Maximum	1/1000 expected value	Maximum	1/1000 expected value	Specification	Rope No.	Maximum tension
Case 1	Strong wind	Medium Full draught	SURGE	0.16m	1000H-Minimum	1	26%	43%	453kN	860kN	2 strands Nylon ϕ 60	1	10kN
			SWAY	0.24m								2	14kN
			HEAVE	-0.03m	1000H-Minimum	2	22%	32%	448kN	788kN	2 strands Nylon ϕ 60	3	18kN
			ROLL	1.21°								4	13kN
			PITCH	0.18°	1000H-Minimum	3	22%	26%	448kN	707kN	2 strands Nylon ϕ 60	Not tensed	
			YAW	0.20°									
Case 2	Strong wind	Medium Ballast	SURGE	0.20m	1000H-Minimum	1	**63%**	**107%**	1048kN	687kN	2 strands Nylon ϕ 60	1	11kN
			SWAY	0.43m								2	15kN
			HEAVE	-0.05m	1000H-Minimum	2	43%	64%	453kN	453kN	2 strands Nylon ϕ 60	3	29kN
			ROLL	-3.24°								4	22kN
			PITCH	0.23°	1000H-Minimum	3	37%	42%	453kN	451kN	2 strands Nylon ϕ 60	Not tensed	
			YAW	0.66°									
Case 3	Strong wind	Medium Full draught	SURGE	0.16m	1250H-Minimum	1	21%	36%	482kN	895kN	2 strands Nylon ϕ 60	1	10kN
			SWAY	0.25m								2	14kN
			HEAVE	-0.03m	1250H-Minimum	2	18%	27%	465kN	771kN	2 strands Nylon ϕ 60	3	18kN
			ROLL	1.29°								4	13kN
			PITCH	0.18°	1250H-Minimum	3	18%	22%	458kN	685kN	2 strands Nylon ϕ 60	Not tensed	
			YAW	0.21°									
Case 4	Strong wind	Medium Ballast	SURGE	0.18m	1250H-Minimum	1	51%	**56%**	489kN	547kN	2 strands Nylon ϕ 60	1	11kN
			SWAY	0.40m								2	15kN
			HEAVE	-0.05m	1250H-Minimum	2	30%	43%	489kN	489kN	2 strands Nylon ϕ 60	3	30kN
			ROLL	-3.43°								4	24kN
			PITCH	0.23°	1250H-Minimum	3	31%	43%	489kN	489kN	2 strands Nylon ϕ 60	Not tensed	
			YAW	0.71°									
Case 5	Strong wind	Small Ballast	SURGE	0.16m	1250H-Minimum	1	51%	53%	489kN	934kN	2 strands Nylon ϕ 60	1	10kN
			SWAY	0.52m								2	15kN
			HEAVE	-0.09m							2 strands Nylon ϕ 60	3	17kN
			ROLL	-4.98°	1250H-Minimum	2	37%	48%	489kN	937kN		4	33kN
			PITCH	0.47°							2 strands Nylon ϕ 60	Not tensed	
			YAW	0.70°									
Case 6	Long wave	Medium Full draught	SURGE	0.38m	1250H-Minimum	1	26%	40%	489kN	489kN	2 strands Nylon ϕ 60	1	20kN
			SWAY	-0.34m								2	23kN
			HEAVE	-0.31m	1250H-Minimum	2	28%	31%	489kN	489kN	2 strands Nylon ϕ 60	3	46kN
			ROLL	-4.53°								4	35kN
			PITCH	-0.91°	1250H-Minimum	3	36%	**57%**	489kN	556kN	2 strands Nylon ϕ 60	Not tensed	
			YAW	0.65°									
Case 7	Strong wind	Medium Ballast	SURGE	0.16m	1250H-Maximum	1	34%	36%	**979kN**	**1804kN**	2 strands Nylon ϕ 60	1	20kN
			SWAY	-0.18m								2	23kN
			HEAVE	-0.31m	1250H-Maximum	2	13%	16%	813kN	1149kN	2 strands Nylon ϕ 60	3	46kN
			ROLL	-0.05°								4	35kN
			PITCH	0.24°	1250H-Maximum	3	21%	28%	972kN	1647kN	2 strands Nylon ϕ 60	Not tensed	
			YAW	0.60°									

Note: Fender deflection is close to allowable limit (50 - 55%).

Red Fender deflection is over allowable limit (more than 55%).

As described above, it is important to set suitable calculation cases without overlooking the key conditions that are critical to the design. While trying to cover all possible conditions consumes many person-hours and time, excessive risk of overlooking critical phenomena exists if this process is omitted. It is recommended that the initial estimation is performed under typical conditions, and the cases are gradually modified during consideration.

A.5 Calculation example: Effective berthing energy

The effective berthing energy needs to be calculated each time according to the type and size of vessel, method of berthing, berthing velocity, etc., as shown in equation 5.3.1 in Chapter 5. When studying the outline of rubber fenders, it is helpful a rough indication depending on target vessel is present. The following example corresponds to the calculation of the effective berthing energy after setting the typical conditions.

The conditions for calculation can be set as follows.

(1) Vessel data

1) Displacement tonnage: DT

Estimated dead weight tonnage (DWT) and gross tonnage (GT) of full draught condition with reference to the Technical Note of The Port and Airport Research Institute [2], based on values with 75% coverage.

2) Eccentricity factor: C_e, virtual mass factor: C_m

From the Technical Note of National Institute for Land and Infrastructure Management [3], the survey results (cover rate 75%) for the length between perpendiculars (L_{pp}), width (B) and full draught (d) were used. The eccentricity factor (C_e), gyration radius (K_r), block coefficient (C_b) and virtual mass factor (C_m) are calculated using equations (5.3.2), (5.3.3), (5.3.4), and (5.3.7), respectively.

(2) Berthing method: Assuming 1⁄4 point berthing: $R_S = 1/4 \times L_{pp}$, as shown in Fig. 5.3.1

(3) Berthing velocity: Three typical values are used: V_B = 0.1 m/s、0.15 m/s、0.20 m/s.

Tables A.5.1 to A.5.10 present the calculation results for the effective berthing energy for different types of vessel.

Tables A.5.1 Effective berthing energy: General cargo vessel

$DT(75\%)=$ 2.92 $\times DWT$ ^ 0.924

Dead weight tonnage DWT (t)	Displacement tonnage DT (t)	Length between perpendiculars Lpp (m)	Beam B (m)	Full draught d (m)	Eccentricity factor Ce	Virtual mass factor Cm	Effective berthing energy (kN·m)		
							Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s
700	1,242	53	9.6	3.3	0.49	1.75	5.37	12.08	21.5
1,000	1,727	57	10.4	3.7	0.51	1.73	7.6	17.2	30.5
2,000	3,277	71	12.8	4.6	0.51	1.74	14.5	32.7	58.2
3,000	4,767	81	14.3	5.3	0.51	1.77	21.4	48.2	85.6
5,000	7,642	95	16.6	6.2	0.51	1.77	34.5	77.5	138
10,000	14,500	118	20.3	7.7	0.51	1.78	65.9	148	263
12,000	17,161	125	21.4	8.1	0.51	1.77	78.0	175	312
18,000	24,961	141	24.0	9.2	0.52	1.77	114.2	257	457
30,000	40,017	166	27.9	10.8	0.52	1.78	184	414	735
40,000	52,202	181	30.3	11.8	0.52	1.78	241	541	962
55,000	70,061	200	32.3	13.0	0.53	1.78	329	740	1316
70,000	87,549	216	32.3	14.0	0.55	1.78	428	963	1712
90,000	110,434	234	38.2	15.1	0.52	1.78	513	1155	2053
120,000	144,061	256	41.5	16.6	0.52	1.79	673	1514	2691
150,000	177,048	274	44.3	17.7	0.52	1.78	827	1861	3309
200,000	230,959	300	48.1	19.4	0.53	1.79	1084	2439	4335
250,000	283,844	319	56.2	20.8	0.50	1.78	1271	2859	5083
300,000	335,925	324	57.3	22.0	0.52	1.75	1543	3471	6170
400,000	438,214	353	65.0	23.1	0.53	1.69	1949	4386	7798

Tables A.5.2 Effective berthing energy: Container vessel

$DT(75\%)=$ 1.634 $\times DWT$ ^ 0.986

Dead weight tonnage DWT (t)	Displacement tonnage DT (t)	Length between perpendiculars Lpp (m)	Beam B (m)	Full draught d (m)	Eccentricity factor Ce	Virtual mass factor Cm	Effective berthing energy (kN·m)			Remarks TEU (Coverage: 25～75%)
							Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s	
5,000	7,252	101	18.3	6.0	0.46	1.81	30.2	68.0	121	400～500
10,000	14,363	130	22.2	7.9	0.45	1.91	61.9	139	248	200～1000
20,000	28,449	165	27.0	10.2	0.45	1.97	126	284	505	1100～1800
30,000	42,432	190	30.3	11.9	0.45	2.02	192	431	767	1900～2700
40,000	56,349	215	31.8	11.9	0.48	1.87	251	565	1004	2800～3500
50,000	70,216	255	32.3	12.8	0.47	1.96	320	720	1281	3600～4400
60,000	84,044	272	35.5	13.5	0.46	1.95	375	843	1499	4500～5300
100,000	139,076	322	45.3	14.6	0.46	1.79	575	1294	2300	7900～8700
140,000	193,791	353	48.5	15.8	0.49	1.73	815	1833	3259	11400～12100
165,000	227,872	360	52.0	16.2	0.50	1.67	947	2131	3789	13500～14300
185,000	255,084	382	59.4	16.2	0.48	1.63	993	2234	3971	16300～18200
200,000	275,466	382	59.4	16.2	0.50	1.59	1087	2447	4350	17800～19700

Tables A.5.3 Effective berthing energy: Tanker

$DT(75\%) = 1.688 \times DWT^{0.976}$

Dead weight tonnage DWT (t)	Displacement tonnage DT (t)	Length between perpendiculars Lpp (m)	Beam B (m)	Full draught d (m)	Eccentricity factor Ce	Virtual mass factor Cm	Effective berthing energy (kN·m) Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s
1,000	1,430	57	10.2	4.1	0.44	2.08	6.53	14.7	26.1
2,000	2,813	72	12.4	5.0	0.45	2.03	12.9	29.0	51.6
3,000	4,179	84	13.9	5.6	0.46	2.01	19.2	43.1	76.6
5,000	6,880	100	16.1	6.4	0.47	1.96	31.4	70.7	126
10,000	13,532	128	19.7	7.8	0.47	1.93	61.8	139	247
15,000	20,102	148	22.1	8.8	0.48	1.92	92.2	208	369
20,000	26,618	164	24.0	9.5	0.48	1.90	122	274	488
30,000	39,541	168	26.9	10.6	0.53	1.77	184	413	735
50,000	65,098	193	32.9	12.3	0.53	1.72	296	666	1184
70,000	90,404	213	32.9	13.5	0.57	1.69	435	978	1739
90,000	115,535	228	43.5	14.5	0.52	1.67	499	1122	1995
100,000	128,048	235	43.5	14.9	0.53	1.66	563	1266	2251
150,000	190,212	263	48.9	16.7	0.55	1.62	842	1894	3367
300,000	374,147	322	60.2	22.1	0.54	1.68	1700	3824	6799

Tables A.5.4 Effective berthing energy: Roll On Roll Off vessel

$DT(75\%) = 8.728 \times DWT^{0.79}$

Gross tonnage GT (t)	Displacement tonnage DT (t)	Length between perpendiculars Lpp (m)	Beam B (m)	Full draught d (m)	Eccentricity factor Ce	Virtual mass factor Cm	Effective berthing energy (kN·m) Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s	Remarks
3,000	4,873	110	19.0	5.6	0.36	2.14	18.7	42.1	74.9	Japanese gross tonnage (Japanese ship statement)
5,000	7,296	131	22.2	6.2	0.35	2.11	27.3	61.3	109.0	
10,000	12,616	161	27.4	7.0	0.36	2.01	45.0	101.3	180	
15,000	17,379	161	30.3	7.6	0.38	1.86	61.9	139	248	
20,000	21,814	181	27.3	7.9	0.42	1.83	84.4	190	338	International gross tonnage (Lloyd's data)
40,000	37,717	191	31.5	9.1	0.47	1.68	150	337	600	
60,000	51,958	191	34.2	9.9	0.52	1.58	212	478	850	

Tables A.5.5 Effective berthing energy: Car carrier (PCC) vessel

$DT(75\%) = 1.946 \times DWT^{0.898}$

Gross tonnage GT (t)	Displacement tonnage DT (t)	Length between perpendiculars Lpp (m)	Beam B (m)	Full draught d (m)	Eccentricity factor Ce	Virtual mass factor Cm	Effective berthing energy (kN·m) Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s	Remarks
3,000	2,580	106	17.3	5.0	0.30	2.65	10.14	22.8	40.5	Japanese gross tonnage (Japanese ship
5,000	4,081	130	21.2	6.1	0.28	2.91	16.5	37.1	65.9	
40,000	26,411	192	33.1	10.2	0.36	2.22	104	234	416	
12,000	8,959	136	24.0	6.5	0.36	2.03	33.0	74.1	132	International gross tonnage (Lloyd's data)
20,000	14,173	151	26.3	7.0	0.40	1.84	52.3	118	209	
30,000	20,398	164	28.3	7.5	0.43	1.73	76.4	172	306	
40,000	26,411	174	31.4	9.2	0.41	1.90	102	230	409	
60,000	38,012	192	33.3	10.2	0.43	1.85	152	341	606	
70,000	43,656	220	33.3	10.9	0.42	1.96	179	402	715	

Tables A.5.6 Effective berthing energy: LPG carrier

DT(75%)= 4.268 ×*DWT*＾ 0.914

Gross tonnage *GT* (t)	Displacement tonnage *DT* (t)	Length between perpendiculars *Lpp* (m)	Beam *B* (m)	Full draught *d* (m)	Eccentricity factor *Ce*	Virtual mass factor *Cm*	Effective berthing energy (kN·m)			Remarks
							Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s	
3,000	6,432	92	16.2	6.0	0.49	1.83	28.6	64.4	114.5	International gross tonnage (Lloyd's data)
5,000	10,258	106	18.5	7.0	0.50	1.82	46.3	104.1	185	
10,000	19,330	130	22.3	8.6	0.51	1.80	88.3	199	353	
20,000	36,422	159	26.7	10.5	0.52	1.77	169	380	675	
40,000	68,629	219	37.3	12.2	0.47	1.76	287	647	1150	
50,000	84,156	219	37.3	12.2	0.53	1.62	363	818	1454	

Tables A.5.7 Effective berthing energy: LNG carrier

DT(75%)= 1.601 ×*DWT*＾ 0.97

Gross tonnage *GT* (t)	Displacement tonnage *DT* (t)	Length between perpendiculars *Lpp* (m)	Beam *B* (m)	Full draught *d* (m)	Eccentricity factor *Ce*	Virtual mass factor *Cm*	Effective berthing energy (kN·m)			Remarks
							Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s	
20,000	23,790	159	26.8	8.0	0.48	1.69	96.1	216	384	International gross tonnage (Lloyd's data)
30,000	35,253	183	30.6	8.9	0.48	1.66	141	318	565	
50,000	57,862	217	36.0	10.1	0.49	1.62	230	517	920	
80,000	91,283	255	41.9	11.5	0.50	1.59	361	811	1442	
100,000	113,342	275	45.0	12.2	0.50	1.58	447	1005	1786	
130,000	146,190	301	48.9	13.1	0.50	1.57	575	1293	2299	
160,000	178,808	333	54.6	13.8	0.48	1.57	680	1529	2719	

Tables A.5.8 Effective berthing energy: Passenger carrier

DT(75%)= 2.73 ×*DWT*＾ 0.871

Gross tonnage *GT* (t)	Displacement tonnage *DT* (t)	Length between perpendiculars *Lpp* (m)	Beam *B* (m)	Full draught *d* (m)	Eccentricity factor *Ce*	Virtual mass factor *Cm*	Effective berthing energy (kN·m)			Remarks
							Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s	
3,000	2,916	81	16.5	4.2	0.41	1.79	10.56	23.8	42.3	International gross tonnage (Lloyd's data)
5,000	4,550	96	18.5	4.8	0.41	1.78	16.7	37.5	66.7	
10,000	8,321	122	21.8	5.7	0.42	1.77	30.7	69.1	122.8	
20,000	15,218	155	25.5	6.4	0.44	1.67	55.9	125.9	224	
30,000	21,664	178	28.0	6.9	0.45	1.63	79.7	179	319	
50,000	33,804	213	32.3	7.6	0.46	1.59	123	276	491	
70,000	45,315	239	32.3	8.0	0.49	1.54	172	387	688	
100,000	61,825	270	35.6	8.4	0.50	1.50	233	524	932	
130,000	77,698	297	38.5	8.8	0.51	1.48	290	653	1162	
160,000	93,100	311	41.0	9.1	0.52	1.45	348	783	1392	

Tables A.5.9 Effective berthing energy: Middle distance (less than 300 km) ferry

DT(75%)= 4.98 ×*DWT*＾ 0.855

Gross tonnage *GT* (t)	Displacement tonnage *DT* (t)	Length between perpendiculars *Lpp* (m)	Beam *B* (m)	Full draught *d* (m)	Eccentricity factor *Ce*	Virtual mass factor *Cm*	Effective berthing energy (kN·m)			Remarks
							Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s	
400	836	46	11.6	2.8	0.42	1.69	2.99	6.73	11.96	Japanese gross tonnage (Japanese ship statement))
700	1,348	58	13.2	3.3	0.41	1.75	4.86	10.94	19.5	
1,000	1,829	67	14.4	3.6	0.41	1.76	6.59	14.8	26.3	
3,000	4,679	104	18.6	4.7	0.40	1.79	16.9	38.0	67.5	
7,000	9,656	145	22.6	5.8	0.40	1.81	35.0	78.8	140	
10,000	13,099	167	24.6	6.4	0.40	1.84	47.7	107	191	
13,000	16,393	186	26.1	6.8	0.40	1.84	59.7	134	239	

Tables A.5.10 Effective berthing energy: Long distance (more than 300 km) ferry

DT(75%)= 15.409 ×DWT ^ 0.735

Gross tonnage GT (t)	Displacement tonnage DT (t)	Length between perpendiculars Lpp (m)	Beam B (m)	Full draught d (m)	Eccentricity factor Ce	Virtual mass factor Cm	Effective berthing energy (kN·m) Velocity V_B 0.10m/s	Velocity V_B 0.15m/s	Velocity V_B 0.20m/s	Remarks
6000	9,220	136	22.5	6.2	0.39	1.91	34.4	77.5	138	Japanese gross tonnage (Japanese ship statement))
10000	13,421	162	25.9	6.2	0.40	1.75	47.3	106.4	189	
15000	18,080	187	27.6	6.9	0.40	1.79	64.8	146	259	
20000	22,337	207	27.6	7.4	0.41	1.82	83.0	187	332	

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Guidelines for Design and Testing of Rubber Fender systems

June 2019

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