URTO DELLE NAVI

3.2 Design Loads

3.2.1 General

The design load for a limit state is defined as the most unfavourable combination of the characteristic load multiplied by a load coefficient. The limits states are categorized as follows:

- The ultimate limit state (ULS) is related to the risk of failure or large inelastic displacements or strains of a failure character.
- The servicability limit state (SLS) is related to criteria governing normal use or durability.
- The fatigue limit state (FLS) is related to the risk of failure due to the effect of repeated loading.
- The limit state of progressive collapse (PLS) is related to the risk of failure of the structure under the assumption that certain parts of the structure have ceased to perform their load-carrying functions.

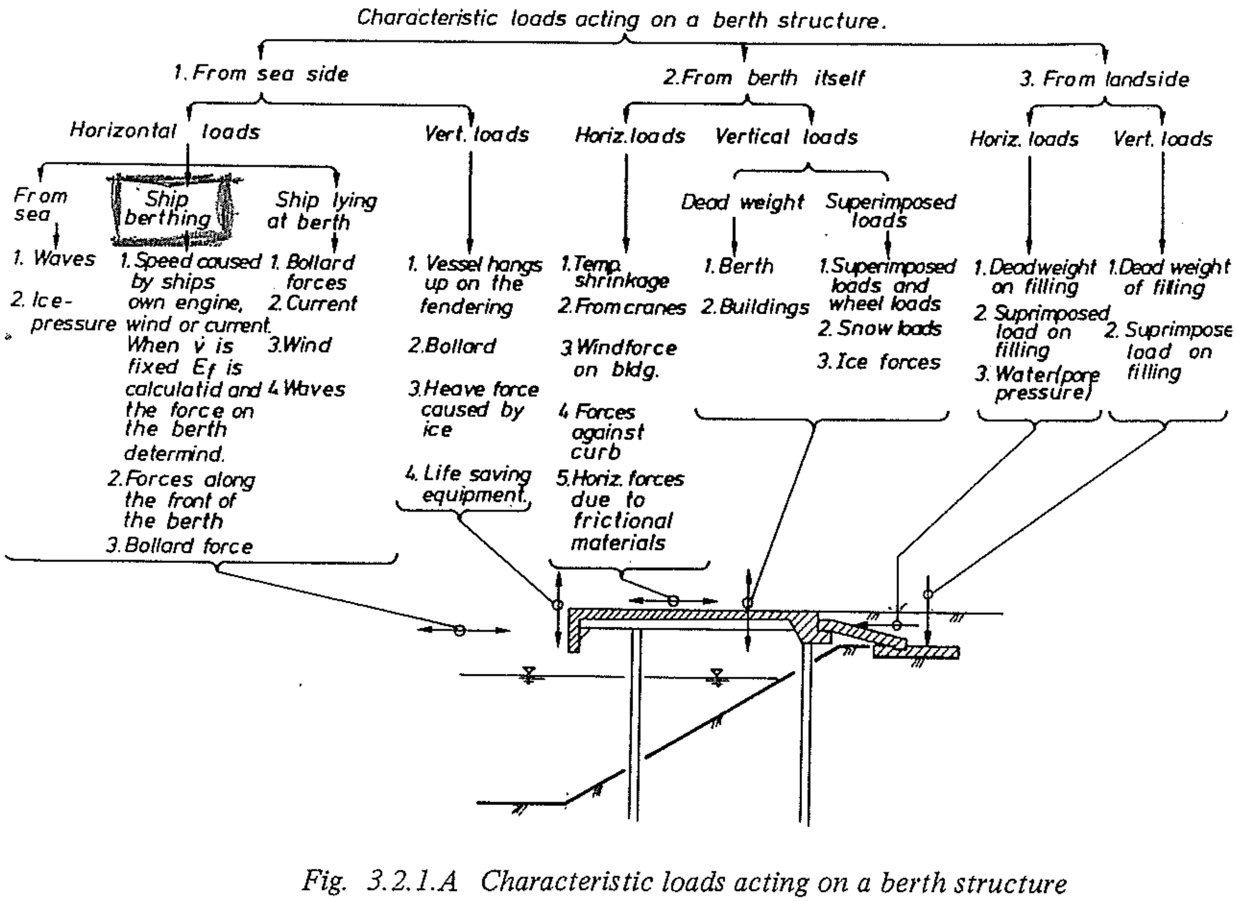


Fig. 3.2.1.A Characteristic loads acting on a berth structure

FASE DI ACCOSTO
DIMENSIONAMENTO DEI FENDER
AZIONE RISULTANTE SULL'OPERA

SHIP BERTHING

A ship coming alongside is usually stopped partly by reversing the engine and partly be retarding by the spring hawser, so that the total design force transmitted to the berth structure through the bollard will at least be equal to the breaking load of the spring hawser. Materials for hawsers are steel wire, manila rope, nylon rope, etc., i.e. different materials implying great variations in the breaking loads and ductility of the various hawsers. Figure 3.2.2.4.A shows a normal mooring arrangement for a ship to a quay via bollards.

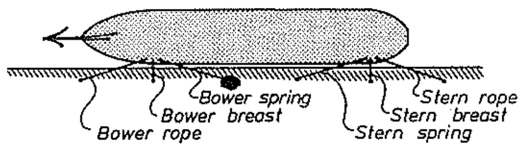


Fig. 3.2.2.4.A Mooring by hawsers and bollards

It has therefore been necessary, as has been done in most port engineering standards, to specify minimum loadings that the bollards shall be able to resist for ships of various tonnages. Thus the bollards, their dimensions and anchoring, and the berth structure itself shall be designed for a certain minimum loading. The idea is that if a ship has a too strong hawser compared to the design load of the bollard, only the latter will break at its footing without the berth structure itself being much affected.

Bollard should be provided at intervals of approximately 5-30 meters depending of the size of the ship along the berthing face, and the bollard load capacity should be as shown in the table below.

The bollard load * and approximate spacing between bollards shall be:
Load Point Load

Ships of displacements in tons up to	Bollard load P, kN (*)	Appr. spacing in m	Bollard load from the berth kN pr. lin m berth (**)	Bollard load along the berth, kN per lin. m berth
2,000	100	5-10	15	10
5,000	200	10-15	15	10
10,000	300	15	20	15
20,000	500	20	25	20
30,000	600	20	30	20
50,000	800	20-25	35	20
100,000	1000	25	40	25
200.000	1500	30	50	30

(*) also Point Load kN per 0,25 m^2
(**) also line Load

For larger ships, specific calculation must be carried out to determine the maximum bollard load, taking into account the type of ship and the environmental loading. Bollard loads are assumed to act in any direction within 180° around the bollard at the sea side, and from horizontally to 60° upwards, as shown in Figure 3.2.2.4.B.

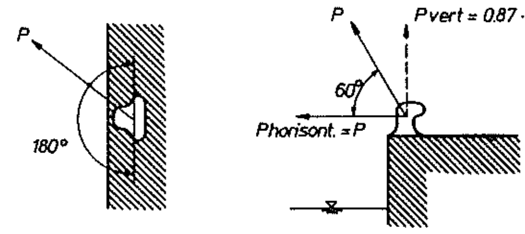


Fig. 3.2.2.4.B Bollard load directions

If the berth structure is much exposed to wind and currents, the above bollard loads should be increased by 25%. When the ship is moored, the bollard will be loaded with a vertical force of 0.87 · P, as shown in Figure 3.2.2.4.B.

FENDER SYSTEM

When the energy to be absorbed by the fender has been established, one should select a fender which will transmit an acceptable horizontal force against the berth structure front. This horizontal force will depend on the characteristics of the fender. It should be taken into account that the ship will also have to resist this force. Generally it is desirable to have these horizontal forces or reaction force and corresponding reaction pressure as low as possible to avoid damage to the side of the ship, and to minimize the construction cost of the berth.

When a ship comes alongside at an angle with the (or not) berth line, (also) longitudinal friction forces parallel to the berthing face will be transmitted via the fenders to the berth structure. When this occurs ship will skid along the structure after the impact while the fenders are still in a compressed state. The front of the berth structure must take up this friction force $F = \mu \cdot P$, where μ is the friction coefficient between ship and fender, and P is the impact force. The friction coefficients are for steel to steel 0,25, for timber to steel 0,4 to 0,6 and for rubber to steel 0,6 to 0,7. The friction force F is usually acting simultaneously with the impactive force perpendicularly to the front of the structure. The longitudinal friction forces can be reduced by provision of low-friction contact surface materials.

To ensure that the front of the berth structure has satisfactory safety under normal calls, the German «Recommendations of the Committee for Waterfront Structures» recommends that the berth front is designed for a horizontal point load equal to the bollard load. This point load shall be allowed to act anywhere at the berth front without the allowable stresses being exceeded, and its contact area shall be limited to 0.25 sq.m. The table ~~below~~ gives point loads and the corresponding increased loadings in kN per lin.m of e.g. quay for various ship displacements.

4.1 General

The marine fenders provide the necessary interface between the berthing ship and the berth structure, and therefore the principal function of the fender is to transform the impact load from the berthing ship into reactions which both the ship and berth structure can safely sustain. A properly designed fender system must therefore be able to gently stop a moving or berthing ship without damaging the ship, the berth structure or the fender. When the ship has berthed and been safely moored, the fender systems should be able and strong enough to protect the ship and the berth structure from the forces and motions caused by wind, waves, current, tidal changes and loading or unloading of cargo. The design of fenders shall also take into account the importance of the consequences suffered by the ship and the berthing structure in case of an eventual accident due to insufficient energy absorption capacity.

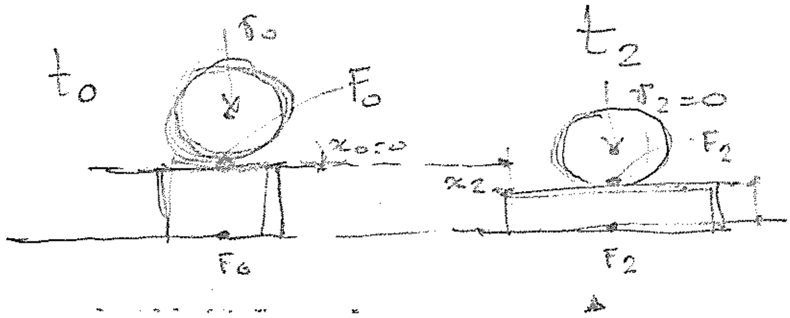
During design of berth and fender constructions in the past and even up to today, one has tended to plan and design the berth structure itself first, and only later the type of fender one hopes will satisfy the requirements as regards berth and ships. This approach to design has resulted in damages occurring quite frequently to berth and fender structures, and to a lesser degree to ships.

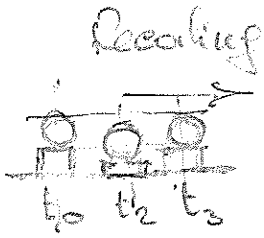
The correct procedure should be to plan and design the fender and berth structures jointly. The choice of fenders shall be dependent on the size of berthing ships and maximum impact energy. After having identified the fender's criteria, one can finalize the design of the berth superstructure. If the following factors are therefore considered in selecting the fender system:

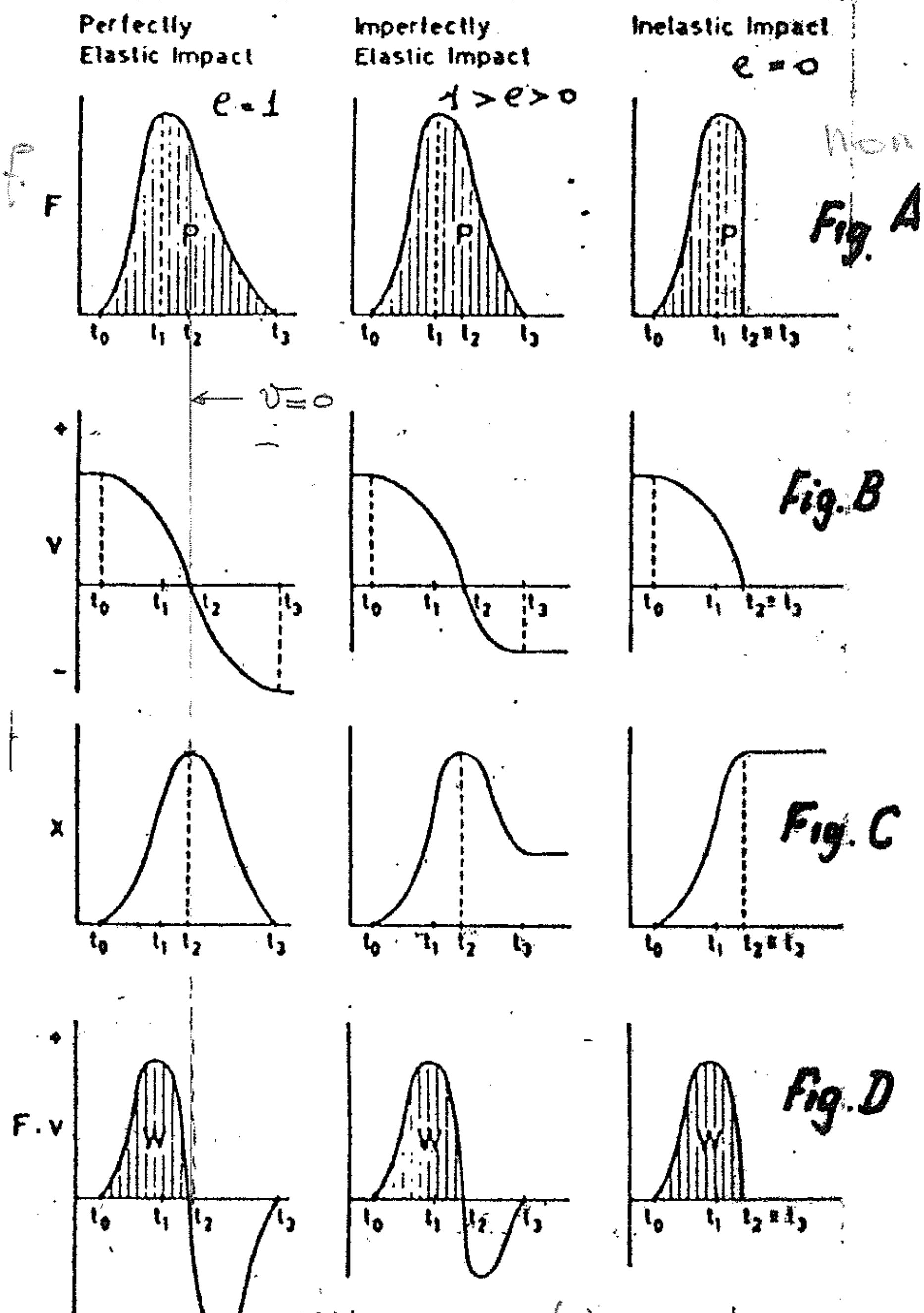
- The fender system must have sufficient energy absorbtion capacity.
- The reaction force from the fender system does not exceed the loading capacity of the berthing system.
- The pressure exerted from the fender system does not exceed the ship hull pressure capacity.
- The capital construction costs and maintenance costs are considered during the design of both the berth structure and fender system.

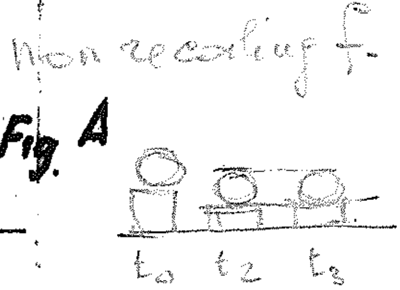
this procedure will lead to:

- Right structural solutions
- Lower construction costs
- Lower annual maintenance costs.

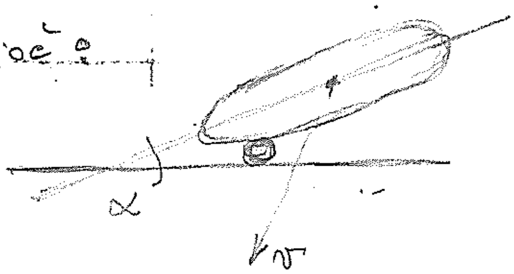








W non eguale. Cioè:



Il lavoro compiuto dalla nave nel sistema urtato nell'intervallo $t_0 \to t_2$, cioè dall'inizio del contatto al tempo per cui $v_2 = 0$, non dipende dall'elasticità dell'impatto e cioè è uguale per recoiling o non recoiling fenders.

3.2.2.3 Impacts from Ships

In the following section the berthing forces that can arise between a berth structure and berthing ship will be discussed. The berthing forces transmitted to the structure will consist of impact loads normal to and frictional loads parallel to the berthing face.

While the vertical loads on a quay from dead weight, live load, crane loads, etc. can be determined very accurately, it is more difficult to evaluate the horizontal loads caused by ships' impact. The size and velocity of ships when berthing, the manoeuvring, direction and strength of current, wind and waves at the quay are factors that often escape an exact quantification and therefore tend to complicate the correct calculation of ships' impacts forces (47) and (50).

Based on normal berthing procedures, the berthing energy and the impact forces from the berthing ship against the berthing structure can be estimated from one of the following theories:

- The theoretical method
- The empirical method
- The statistical method.

The theoretical method is based on the general basic kinetic energy equation (in kNm) due to the impact of a ship on a berthing structure:

$$E = 0{,}5 . M_v \cdot V^2 = 0{,}5(M_d + M_h)V^2$$

where:

M_v = virtual mass (in ton) which equals the displacement of the ship Md plus the hydrodynamic mass moving with the ship M_h.

When the ship is moving through water, there is also the movement of a volume of water around the ship which is entrained with it to be considered. When the ship comes to a stop this additional mass of water will continue to move and press the ship against the berth. This additional mass of water is also known as the added mass or as the hydrodynamic mass.

V = the velocity (in m/s) of the ship normal to the berth line.

For all berth structure design, the displacement for fully loaded ship should be used, except where the berth will be used exclusively for export of cargo, then the desplacement of the ship and the draft may be reduced to the actual value for the ship when berthing, but not less than the ballast displacement.

Out of the total kinetic energy of the ship, the fender system must absorb:

$$E_f = C(0{,}5 . M_d \cdot V^2)$$

where the adjusting factor or berthing coefficient $C = C_H \cdot C_E \cdot C_C \cdot C_S$

where:

C_H = hydrodynamic mass factor =

$$\frac{M_d + M_h \cdot C_{HR}}{M_d} = \frac{M_d + (1/4 \cdot \pi \cdot \rho \cdot D^2 \cdot L) \cdot C_{HR}}{M_d}$$

$$= 1 + \frac{M_h \cdot C_{HR}}{M_d}$$

where:

ρ = specific gravity of sea water (10,3kN per m³)
D = draft of ship
L = length of ship
C_{HR} = reduction factor due to the ship moving at an angle to its longitudinal axis. In principle, the reduction factor for the hydrodynamic mass of a ship moving normal to the berth line in open water will be 1.0, but for a ship moving along its longitudinal axis in open water it can be assumed to be about 0.1.

Professor F. Vasco Costa of Portugal assumes that if the ship moves sideways to e.g. a quay or rotates about its centre of gravity, the value of C_H will be:

$$C_H = 1 + \frac{2D}{B}$$

where D = draft of ship and
B = width of ship

If the ship moves along its longitudinal axis, Professor F. Vasco Costa assumes that the influence of C_H is negligible, i.e. $C_H = 1$.

The exact value of the hydrodynamic mass is very difficult to determine. Investigations and researches have shown that the hydrodynamic mass will vary with the shape of the ship, the underkeel clearance, the ship velocity and the water depth. The hydrodynamic mass usually varies between about 25 and 100% of the displacement of the ship. Generally it is recommended that for water depth of 1.5 times the draft of ship or more, C_H is taken to be 1.5. When the water depth is only 1.1 times the draft of the ship, C_H is taken to be 1.8.

C_E = eccentricity effect =

$$= \frac{i^2 + r^2 \cdot \cos^2 Ø}{i^2 + r^2} =$$ V. foglio esplicativo

where:

i = ship's radius of inertia, generally between 0,2 L and 0,25 L.
r = the distance of point of contact from the centre of mass.

Figure 3.2.2.3.A shows C_E as a function of the angle Ø and the ratio r/2.

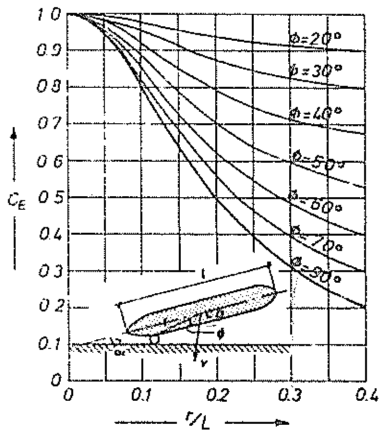


Fig. 3.2.2.3.A Eccentricity effect C_E as function of φ and r/L

If Ø is 90°, the equation will be:

$$C_E = \frac{1}{1 + \frac{r^2}{i^2}}$$

Figure 3.2.2.3.A indicates that the way ships come alongside is a very important matter. This should therefore be studied as closely as possible. The value of Ø = 90° may give too favourable values.

Normally the ship will come alongside either under its own power and with an angle to the berth line, or the ship has nearly stopped outside the berth structure and is carefully manoeuvred towards the structure by help of current, wind and/or waves.

Figure 3.2.2.3.A further indicates that the value of C_E also depends on where on the ship the impact comes. Usually the berthing or angle of approach which is the angle between the berth line and ship, will be about 1 to 5° if the ship is berthing with tugboat assistance. If the berthing manoeuvring is done without tugboat assistance the berthing angle will usually be less than about 10 to 15°. Then the distance between the gravity centre of the ship and the point of the impact, r, is about 0.25-0.35 x L. If now the angle Ø is approaching 90°, there will be a minimum amount of impact energy hitting the berth structure. For a continuous fender system, C_E is generally taken between 0.5 and 0.6 and for berth structures with e.g. individual breasting dolphins, C_E is taken between 0.7 and 0.8.

If the ship comes alongside parallel to the berth front, i.e. $a = 0°$, the ratio r/L also approaches 0, and we get the maximum amount of impact energy. On the other hand, the part or length of the ship hitting the structure is now far greater, implying that the energy to be absorbed per lin. m of berth structure will be less than in the above case.

Therefore, if we assume the most favourable values of, respectively, a, Ø and r/L, we would be able to find theoretically a very moderate impact energy. But in practice, however, manoeuvring will deviate from the ideal assumptions, and it is advisable to choose realistic values.

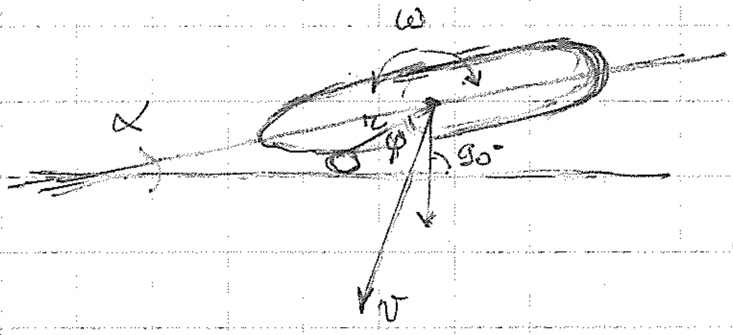
C_C = water cushion effect = 0.8-1.0 at, respectively, solid and open quays. If the bottom is sloping steeply under the quay, the resistance from the water will increase when the ship comes near the quay front. This is particularly true at solid quays where the water between ship and quay has to be squeezed aside before the ship can touch the quay structure.

C_S = softening effect = 0.9-1.0 due to elastic deformations taking place in ship and berth structure.

It appears from the above that sophisticated calculations to establish the magnitude of the adjusting factor or the berthing coefficient C are not justified as long as only approximate values of the velocity v can be found.

To determine the correct value of the approach velocity which is the most significant parameter in the energy equation, is very difficult, but since it appears in the energy formula in the second power one must try to find as accurate a value as possible. This is illustrated with a berthing coefficient C = 1.0 in figure 3.2.2.3.B. Normally, smaller ships have greater velocity when hitting the berth structure than the larger ships. Figure 3.2.2.3.C suggests velocities for different sizes of small and medium ships, berthing without tugboat assistance and related to various weather and manoeuvring conditions.

Coefficiente di eccentricità C_E



nave:

i = raggio d'inerzia

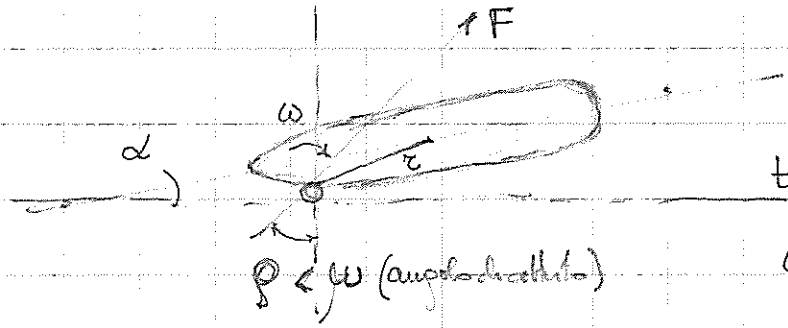
M_v = massa virtuale

$t = t_0$ contatto

$\omega = \omega_0$ (intorno centro di gravità)

$v = v_0$

$$E_0 = 1/2\, M_v v_0^2 + 1/2\, M_v i^2 \omega_0^2$$



$t = t_2$ istante $v = 0$

$\omega = \omega_2$ intorno polo di contatto

$$E_2 = 1/2\, M_v (i^2 + r^2)\,\omega_2^2$$

ω_2? Si calcola con il teorema della conservaz. del mom. della q. di m.

$$M_v v_0 r \sin\varphi + M_v i^2 \omega_0 = M_v (i^2 + r^2)\,\omega_2 \longrightarrow \omega_2$$

$$E_f = E_0 - E_2 = \frac{1}{2} M_v v_0^2 \,\frac{i^2 + r^2\cos^2\varphi}{i^2 + r^2} - M_v v_0 \omega_0 \,\frac{r\, i^2 \sin\varphi}{i^2 + r^2} + \frac{1}{2} M_v \omega_0^2 \,\frac{i^2 r^2}{i^2 + r^2}$$

$$E_f = \frac{1}{2} M_v v_0^2\, C_E \quad \text{con} \quad C_E = \frac{i^2 + r^2\cos^2\varphi}{i^2 + r^2}$$

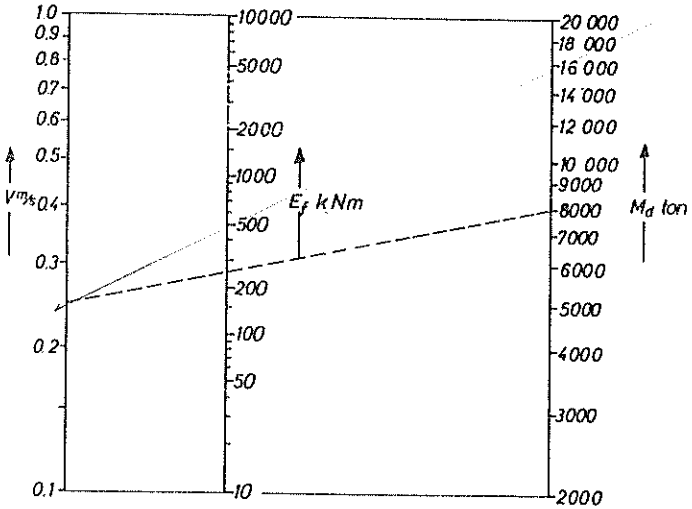


Fig. 3.2.2.3.B The fender energy, E_f, with berthing coefficient $C = 1.0$

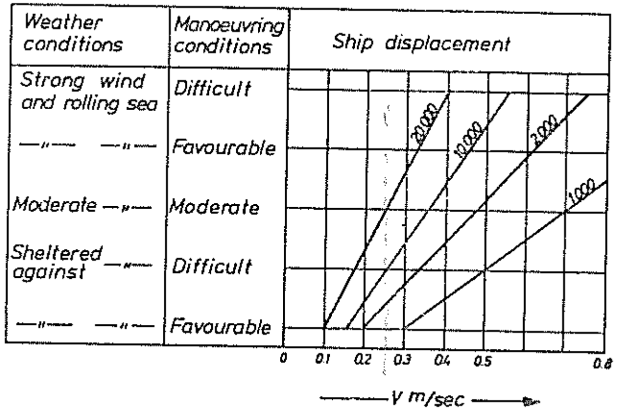
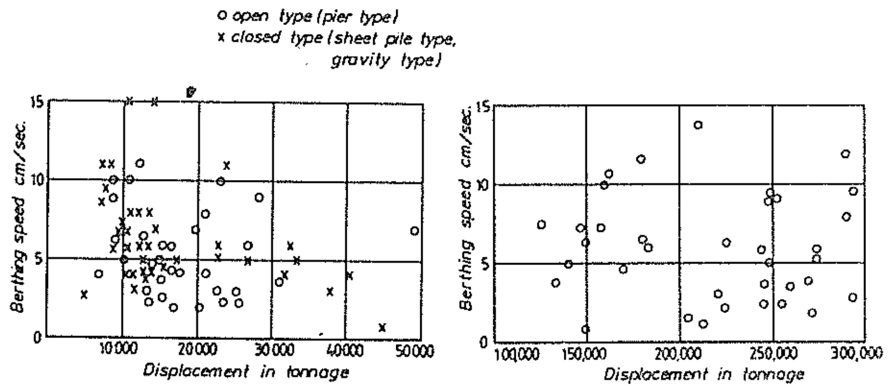


Fig. 3.2.2.3.C Velocity of ship coming alongside

Due to safety reasons and to reduce the probability of damages to the fender systems, PIANC (47) recommend that for the design of fender systems for larger ships, the following berthing velocities with use of tugboat assistance should not be less than:

- Very favourable conditions — 10 cm/sec
- In most cases — 15 cm/sec
- Very unfavourable conditions with cross currents and/or much wind — 25 cm/sec

The Japanese section of PIANC (48) have collected information of the relationship between the ship's displacement and the approaching velocity for large cargo ships and tankers, as shown in figure 3.2.2.3.D. The approaching of the ships were made in such a manner that the ship was stopped parallel with and about 10 to 20 m off the berth structure and then gradually pushed to the berth structure under full control of several tugboats.



For general cargo ships. For tankers.

Fig. 3.2.2.3.D Relation between berthing speed and ship displacement

When designing the fenders for the ramps for ferries and Ro/Ro ships, the berthing bow or stern velocity for ferries and Ro/Ro ships berthing under their own power, will, depending on the stopping distance which the actual ship will use in relation to the length of the berth, vary between about 0.4 to 1.0 m/second (11). Due to the rapid turn-around times and high engine power of most ferries, the berthing velocities are generally higher than for other ships. It is therefor recommended after (50) that the berthing velocity in the direction of approach for the fender design should be:

- For fenders at the corner of the berth structure or the outer or end breasting dolphing: 2,0 to 3,0 m/sec.
- For fenders along the berth structure: 0,5 to 1,0 m/sec.

The empirical method as stated in the British Code of Practice on Maritime Structures (2), the velocity of approach as used in the theoretical method is the most significant and difficult element in the evaluation of the berthing energy imparted to the fender. Therefore the following empirical formula for the maximum impact energy in kN meter to be absorbed by the fender based solely on a ship's displacement may be considered:

$$E_f = \frac{10\,D}{120 + \sqrt{D}} \quad kN \cdot m$$

where D is the displacement tonnage of the berthing ship. A factor of 0.5 may be applied in cases where the impact would either be shared between two fenders or accompanied by rotation of the ship.

The statistical design method is based on measurements of the impact energies actually absorbed by the fenders during berthing. As the method is based on the measurements actually observed at existing berth sites it automatically includes the effect of the berthing velocity, hydrodynamic mass, eccentricity etc.

In figure 3.2.2.3.E the impact energy during berthing operations in normally protected harbours is shown as a function of the displacement of the ship. One curve shows the measurements of the energy in Rotterdam. The two other curves show the impact fender energies recommended by the British Code of Practice and the Norwegian Standard for berth structures. The Norwegian Standard also mentions that for harbours exposed to strong winds and currents or with particularly difficult maneouvring conditions, the impact energy given in figure 3.2.2.3.E shall be increased up to 50%. For structures in the open sea the impact energy shall be increased up to 100%.

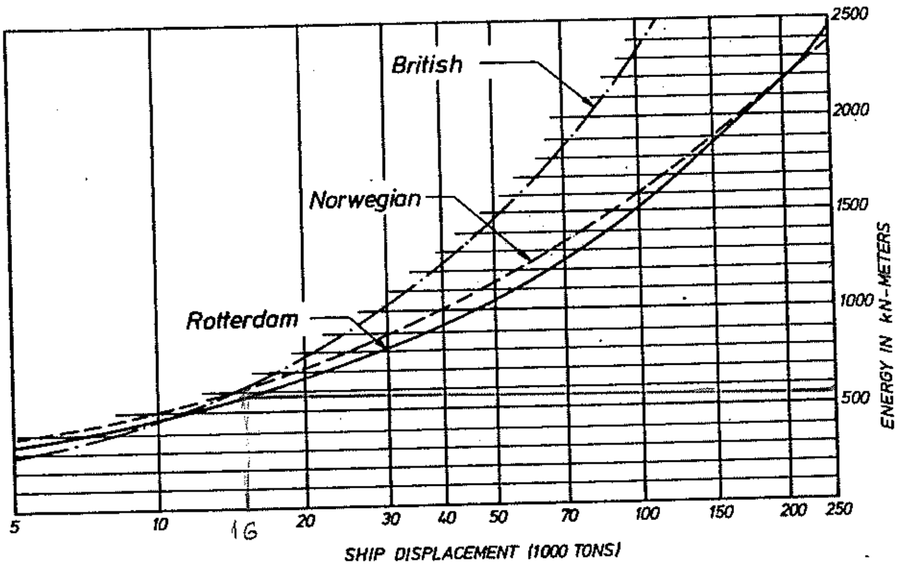
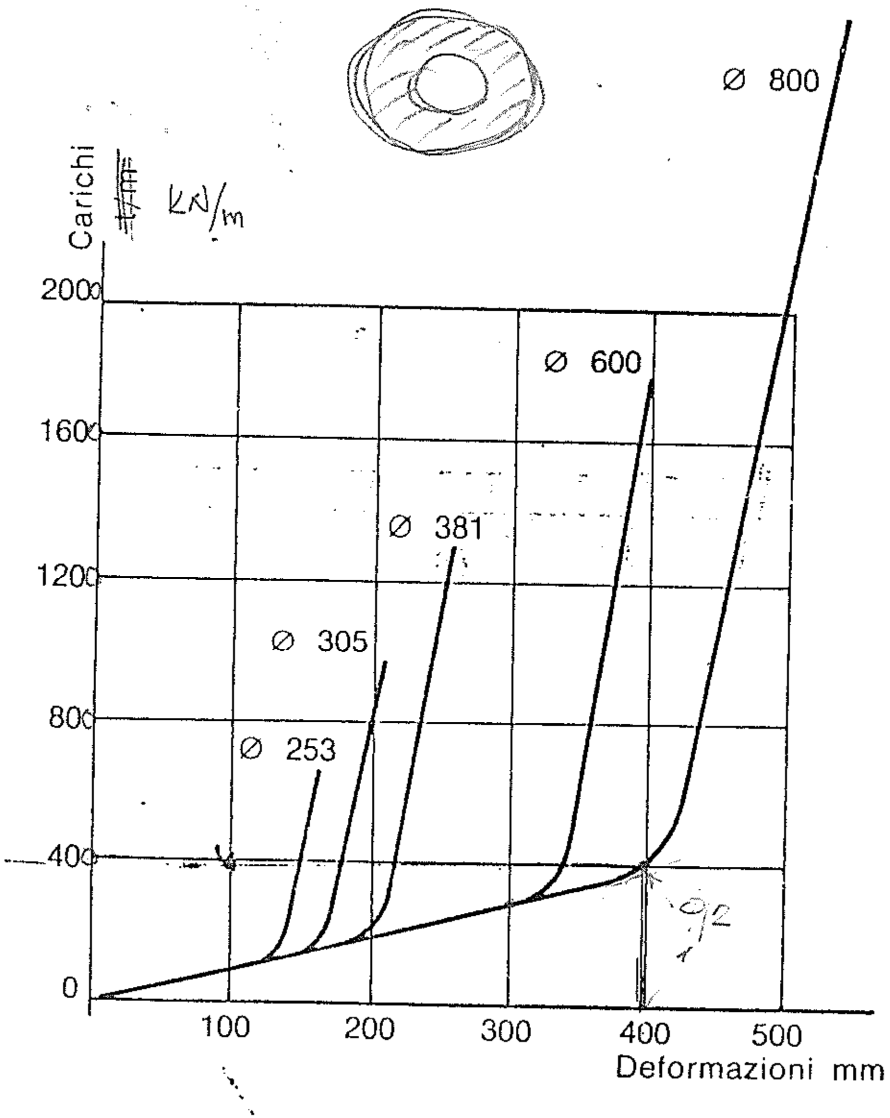


Fig. 3.2.2.3.E Impact energy during ship berthing to a berth structure

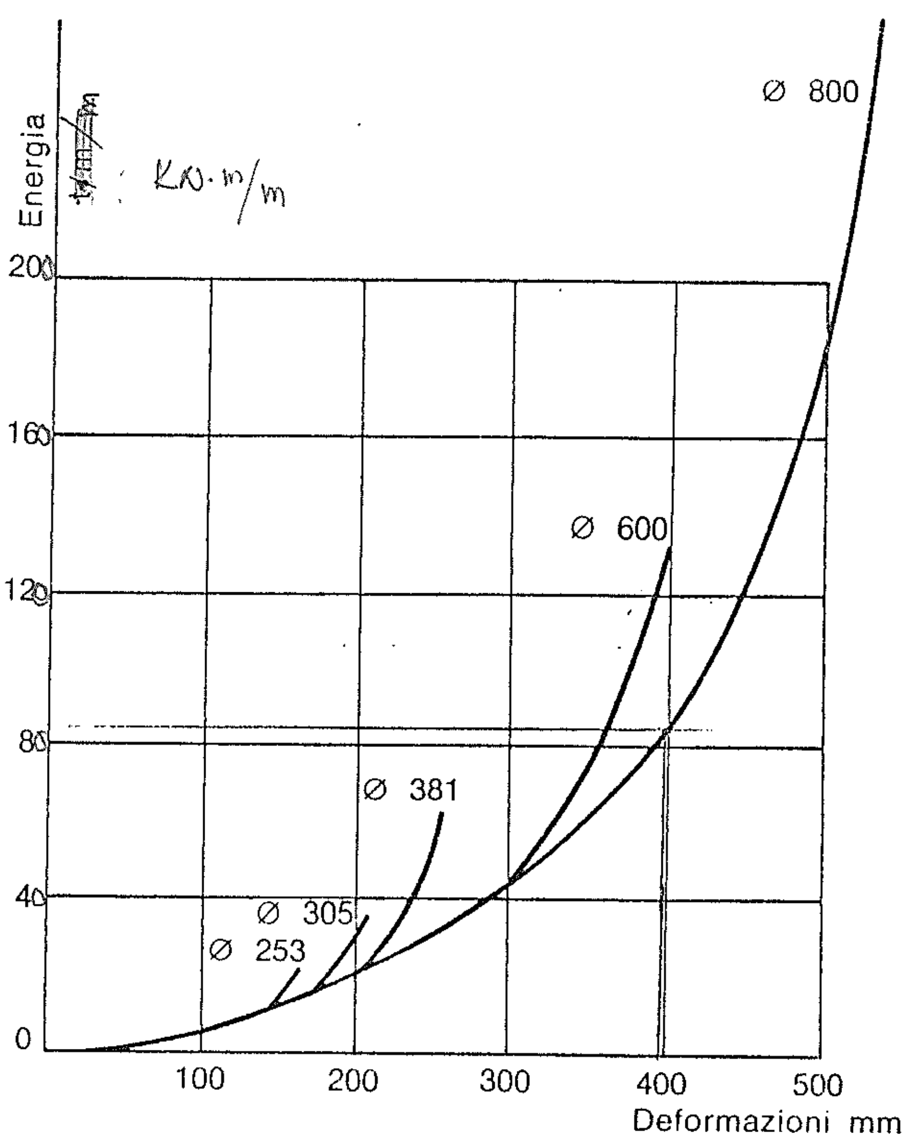
When the energy to be absorbed by the fender has been established, one should select a fender which will transmit an acceptable horizontal force against the berth structure front. This horizontal force will depend on the characteristics of the fender. It should be taken into account that the ship will also have to resist this force. Generally it is desirable to have these horizontal forces or reaction force and corresponding reaction pressure as low as possible to avoid damage to the side of the ship, and to minimize the construction cost.

diagramma carichi-deformazioni
lunghezza 1 m
durezza 76 Shore A - ASTM



$0{,}400 \times 400/2 = 80$

diagramma energia-deformazioni
lunghezza 1 m
durezza 76 Shore A - ASTM



4.4 Different Types of Fenders

In the following the various types of rubber fenders will be discussed, which today are the most used of prefabricated fenders. Since the first rubber fenders were made in the 1930s, they have proved resistant to aggressive and polluted water as well as wear and tear from ships, as long as they have been correctly installed. Their purchase price and maintenance costs are also below those of most other types of fenders. Rubber fenders are produced in many sizes and shapes, depending on their function. One should be aware of the fact that for different manufacturers producing appearantly identical fenders, their fender factors may differ entirely.

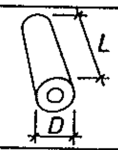
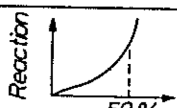
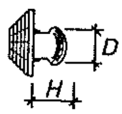
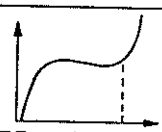
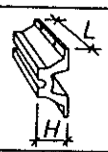
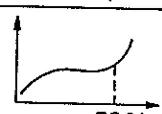
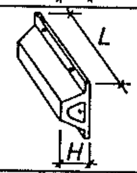
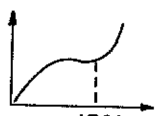
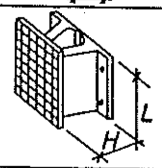
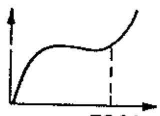
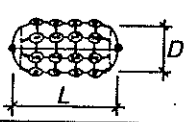
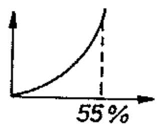
Type	Fendershape	Sizes D/L, H/D, H/L in mm	Reaction kN	Energy kNm	Performance curve
Cylindrical		150/1000 ↓ 2800/5800	80 ↓ 6 600	3 ↓ 5000	 Reaction 50 % Rated compression
Cell		400/550 ↓ 3000/3250	52 ↓ 5 800	8 ↓ 6 700	 47,5 and 52,5 %
V-type		250/1000 ↓ 1000/2000	150 ↓ 2290	15 ↓ 940	 50 %
		200/1000 ↓ 1300/3500	150 ↓ 3400	10 ↓ 1500	 45 %
H-type		400/500 ↓ 2500/4000	140 ↓ 6900	22 ↓ 7000	 50 %
Pneumatic		500/1000 ↓ 4500/12000	50 ↓ 8500	4 ↓ 7000	 55 %

Fig. 4.4.A Different types of rubber fenders

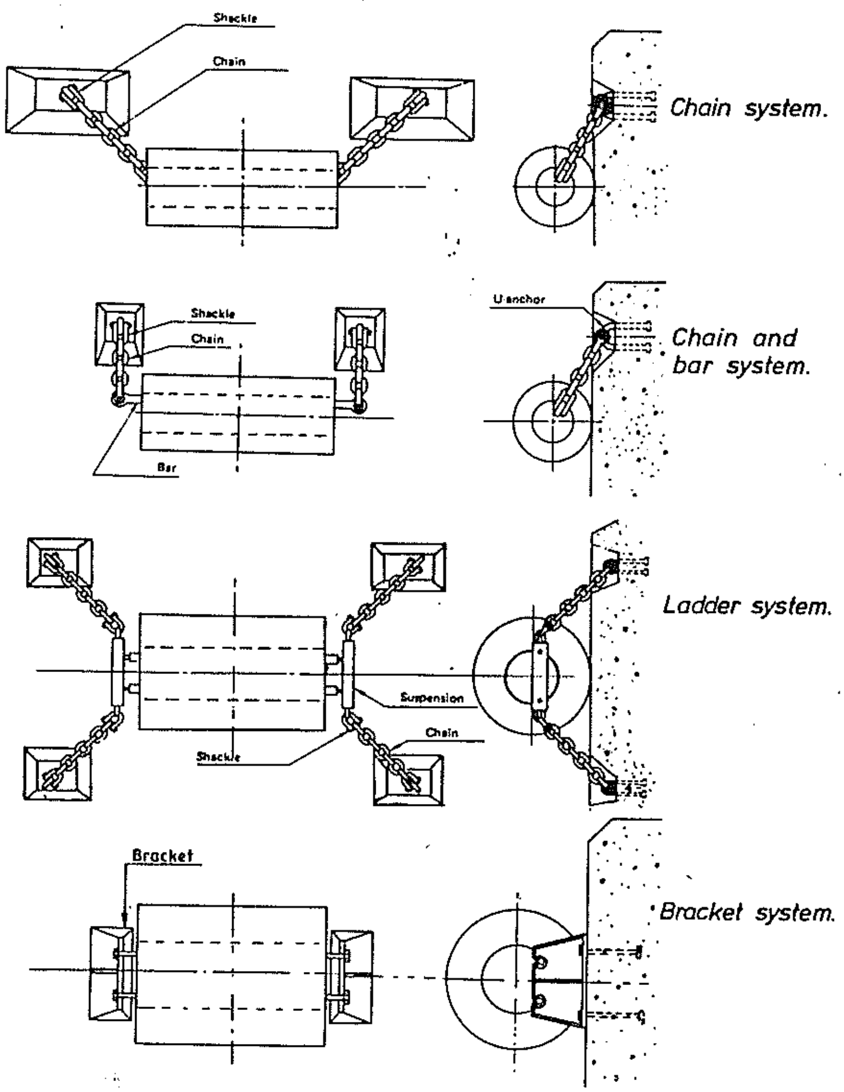


Fig. 4.4.C Different ways of installing cylindrical fenders

Pneumatic balloons are available in sizes ranging from 50 cm outside diameter (OD) and 1.0 m length, to 4.5 m outside diameter and 12 m length. They are well suited as buffers between two tankers or between a tanker and a berth structure.

There is not necessarily any connection between the fender actor P/E_f and the flexibility or the rigidity of a fender. There are fenders with low fender factor (energy-absorbing fender) which are very rigid, and fenders with high fender factor (surface-protecting fenders) which are flexible. For instance, old car tyres used as fenders are very flexible but act as surface-protecting fenders. Even under small loads they are pressed flat and function only as solid fenders.

The Dock & Harbour Authority

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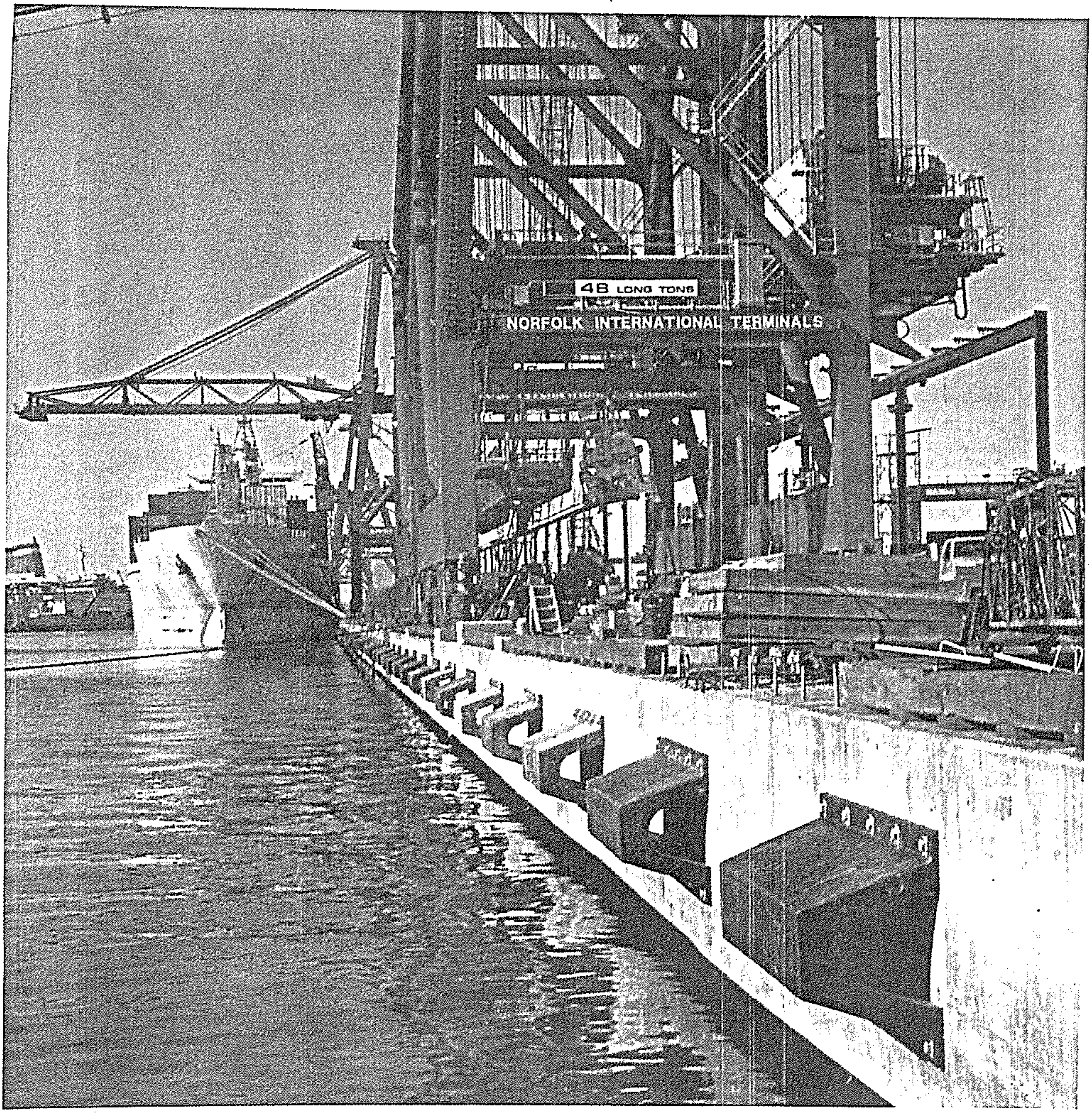
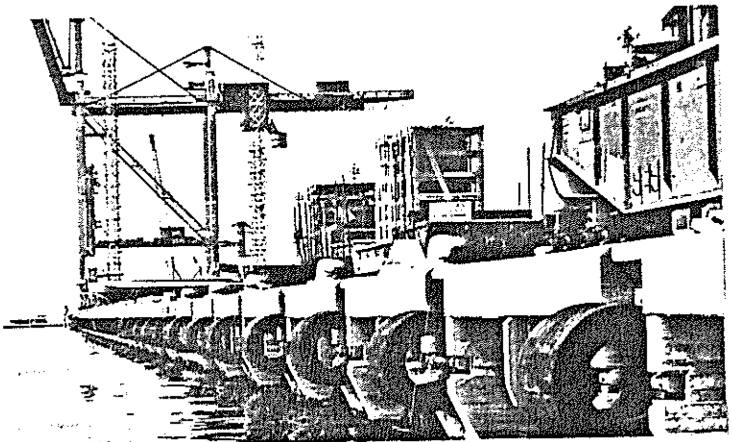
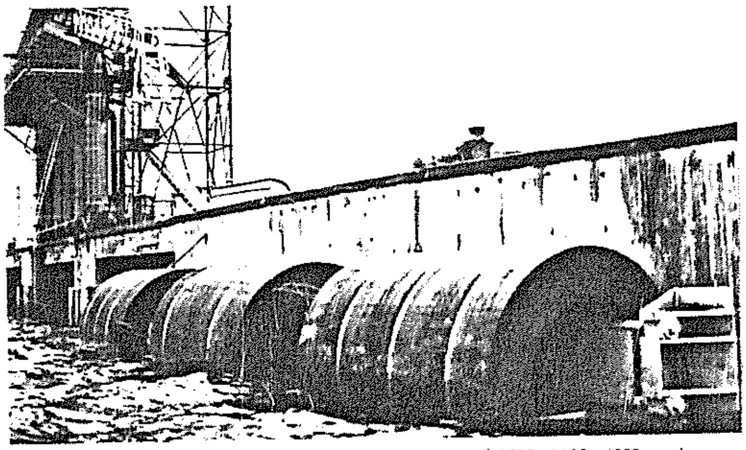


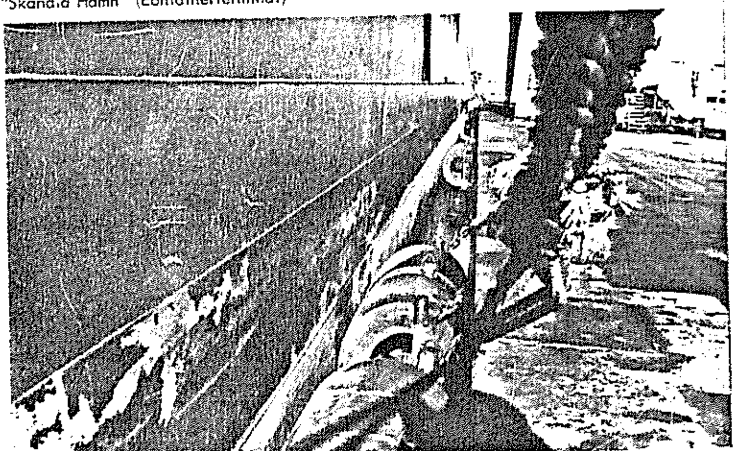
Port Equipment Survey



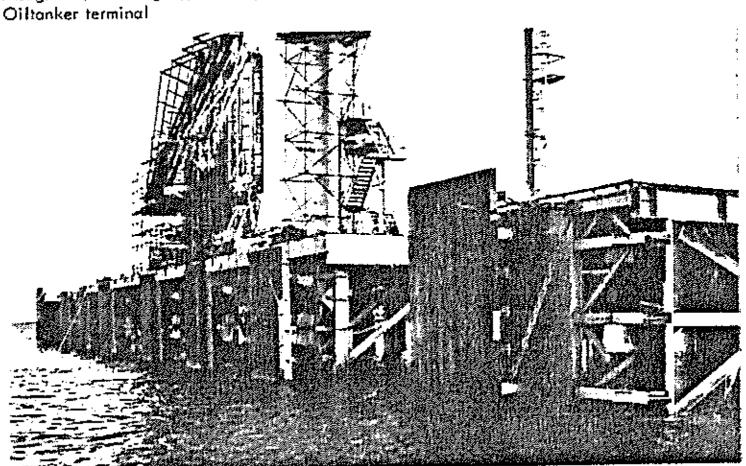
Göteborg, Sweden
"Skandia Hamn" (containerterminal)
Gigant fenders Ø 1200 x 650 x 1500 mm long.



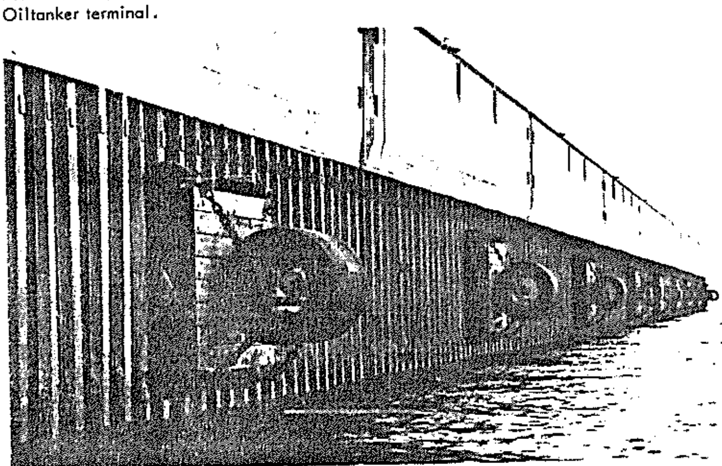
Mongstad (near Bergen), Norway
Oiltanker terminal
Jumbo fenders Ø 2700 x 1600 x 4000 mm long.



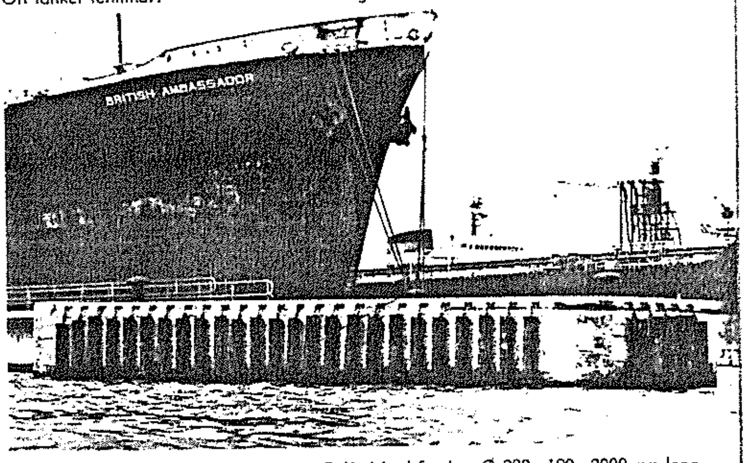
Fos-sur-Mer (Marseille), France
Oiltanker terminal.
Gigant fenders Ø 1400 x 800 x 1500 mm long.



Flotta Island, Orkneys, Great Brittain
Oil tanker terminal.
Jumbo fenders Ø 2450 x 1400 x 5500 mm long.
Gigant fenders Ø 1500 x 800 x 1500 mm long.

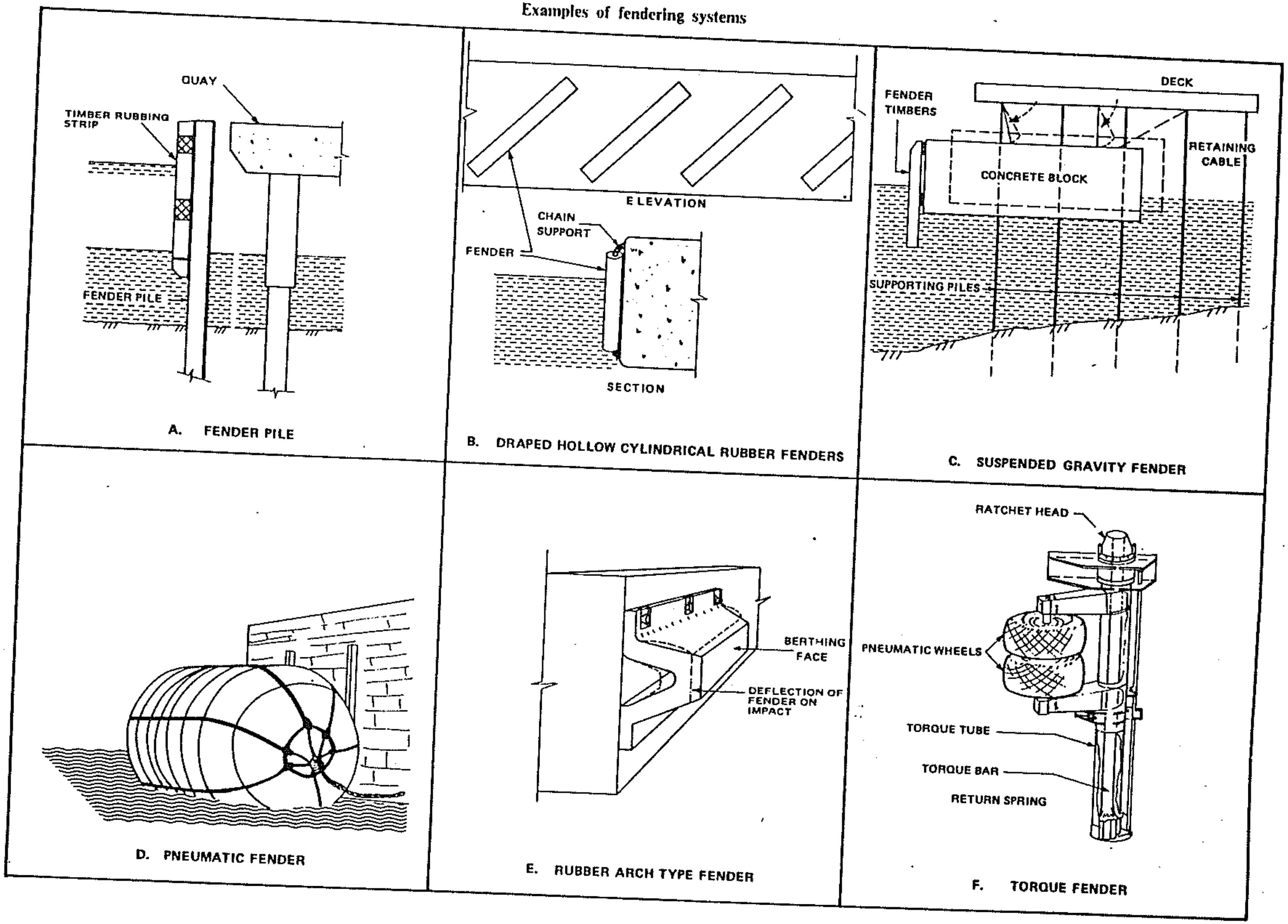


Brunsbüttelkoog, Western-Germany
Bulkcarrier terminal
Gigant fenders Ø 1750 x 1000 x 2000 mm long.



Antwerp Docks, Belgium
Cylindrical fenders Ø 380 x 190 x 2000 mm long.

FIGURE 33
Examples of fendering systems



PARABORDI A GRAVITA'

Principio: l'energia cinetica della nave è convertita in energia potenziale del parabordo mediante il sollevamento di una massa (contrappeso);

Pro: può essere adattato ad un ampio gamma di condizioni di accosto

Contro: elevati costi di installazione e di manutenzione,
richiede una struttura del muro di banchina molto resistente per "sostenere" il contrappeso;
è necessario prevedere una protezione addizionale per la carena della nave

PARABORDI CILINDRICI CARICATI RADIALMENTE (CILINDRI TIPO PIRELLI)

Principio: l'energia cinetica della nave è convertita in energia potenziale del parabordo mediante una combinazione di deformazioni elastiche dell'elemento di gomma indotte da sforzi di compressione e di flessione. La reazione del parabordo è una funzione esponenziale della deformazione

Pro: facilità di installazione;
discreta varietà di offerta sul mercato in termini di forme, dimensioni e produttori;

Contro: valori elevati del coefficiente di attrito parabordo/carena che possono provocare rotture dei parabordi
area di contatto parabordo/carena modesta e quindi elevate pressioni esercitate sulle carene.

PARABORDI CON CORPO CILINDRICO O TRAPEZOIDALE CON ASSE PERPENDICOLARE AL FRONTE DI ACCOSTO DOTATI LATO MARE DI UN PANNELLO DI RIPARTIZIONE

Principio: l'energia cinetica della nave è convertita in energia potenziale del parabordo e calore mediante deformazioni assiali elastiche dell'elemento di gomma indotte da fenomeni di instabilità flessionale (carico di punta) ed isteresi.

Pro: facilità di installazione;
il pannello di ripartizione riduce i valori della pressione esercitata sulla carena della nave
rispetto agli altri tipi di parabordi a parità di reazione massima assorbe una maggiore energia

Contro: valori elevati della reazione anche in occasione dell'accosto di navi di dimensioni inferiori a quella di progetto

PARABORDI GALLEGGIANTI PNEUMATICI

Principio: l'energia cinetica della nave è convertita in energia potenziale del parabordo mediante una compressione elastica dell'aria contenuta al suo interno.

Pro: facilità di installazione;
elevati valori dell'energia assorbita associati a reazioni relativamente modeste
pressione esercitata sulla carena della nave uniforme e di valore modesto
le valvole di sovrappressione prevengono i danni dovuti ad impatti anomali
si "adatta" alle diverse forme delle carene delle navi senza indurre concentrazioni di carico
adatto in località con elevate escursioni di marea

Contro: elevati costi di manutenzione e gestione (periodico controllo pressione interna, danni per forature etc.);
dimensioni elevate e quindi elevate distanze della nave dal limite del fronte di accosto;
la nave durante le fasi di scarico tende a "salire" sul parabordo
costo elevato

PARABORDI GALLEGGIANTI DI POLIETILENE ESPANSO

Principio: l'energia cinetica della nave è convertita in energia potenziale del parabordo mediante una compressione elastica del materiale (polietilene espanso) contenuto al suo interno.

Pro: facilità di installazione;
elevati valori dell'energia assorbita associati a reazioni relativamente modeste
non subisce collassi a causa di forature e/o tagli
adatto in località con elevate escursioni di marea

Contro: pressioni esercitate sulle carene delle navi non sono uniformi e quindi può indurre concentrazioni di carico
dimensioni elevate e quindi elevate distanze della nave dal limite del fronte di accosto;
la nave durante le fasi di scarico tende a "salire" sul parabordo
costo elevato

PARABORDI TRAPEZOIDALI

Principio: l'energia cinetica della nave è convertita in energia potenziale del parabordo e calore mediante deformazioni assiali elastiche dell'elemento di gomma indotte da fenomeni di instabilità flessionale (carico di punta) ed isteresi.

Pro: facilità di installazione;
rispetto agli altri tipi di parabordi a parità di reazione massima assorbe una maggiore energia

Contro: valori elevati della reazione anche in occasione dell'accosto di navi di dimensioni inferiori a quella di progetto
ridotte superfici di contatto parabordo/carena e quindi elevati valori della pressione esercitata sulla carena

BRICCOLE SU PALI (DUCA D'ALBA)

Principio: l'energia cinetica della nave è convertita in energia potenziale del parabordo mediante la deformazione elastica (spostamento) di uno o più pali di acciaio. C'è una relazione lineare tra la reazione e la deformazione della struttura. A volte sulla sommità vengono installati parabordi di gomma con pannello di ripartizione per incrementare la capacità di assorbimento di energia del sistema e ridurre la pressione esercitata sulla carena della nave.

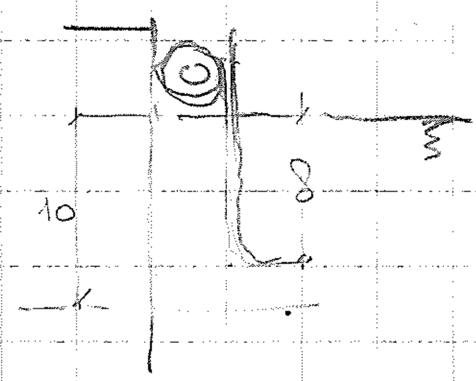
Pro: il sistema combina insieme le due funzioni di parabordo e di struttura di accosto;
la flessibilità e l'assorbimento di energia del sistema si riduce con la lunghezza libera di inflessione del palo

Contro: in caso di carico eccentrico con rotazione del gruppo di pali il comportamento del sistema è inadeguato

Portacontainer 13000 dwt

$M_d = 16\,000\ t = 16 \cdot 10^6\ Kg$

$L = 160\ m \quad B = 25\ m \quad$ pesc. $8\ m$



$r/L = 0.3 \quad r = 48\ m$

$i/L = 0.2 \quad i = 32\ m$

$h = 10\ m \qquad h/pesc = 1.25 \quad C_H = \begin{cases} 1.5 & h/pesc = 1.5 \\ 1.8 & h/pesc = 1.1 \end{cases} \longrightarrow 1.6$

Vasco Costa $C_H = 1 + \dfrac{2\,pesc}{B} = 1.64$ OK

Curve 3.2.2.3C $\quad v_0 = 0.27\ m/s$ (cattive condizioni)

$\varphi = 50^\circ \qquad C_E = \dfrac{i^2 + r^2 \cos^2\varphi}{i^2 + r^2} = 0.6 \qquad C_c \text{ e } C_S = 1$

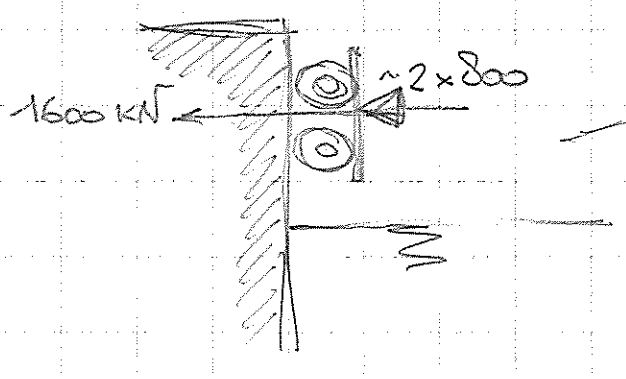
$E_f = \dfrac{1}{2}\, 16 \cdot 10^6 \cdot 0.27^2 \cdot 1{,}6 \cdot 0{,}6 \cdot 1 \cdot 1 = \underline{560\ kN \cdot m}$

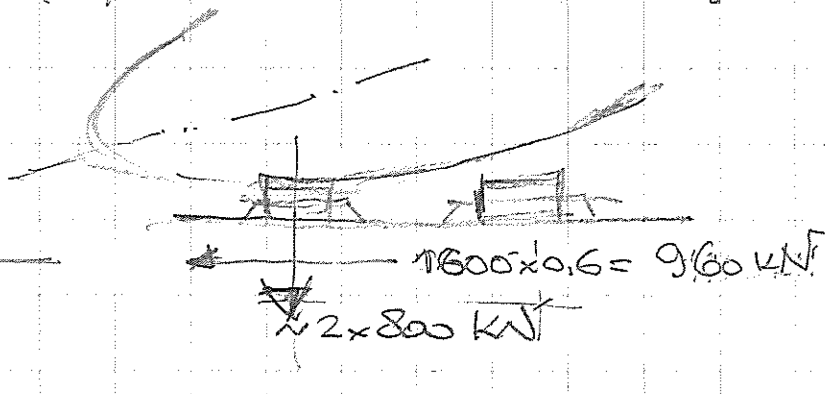
curve statistiche $\quad E \simeq 550\ kN \cdot m \quad$ OK

Volendo usare il fender a manicotto del grafico successivo:
Cilindro 1500/800 $l = 1500\ mm$ (v. grafico) si ha

$\rightarrow E = 290\ kN \cdot m$ per $\delta = 0.71\ m \quad P/E = 2.75 \quad P \simeq 800\ kN$

cioè ce ne vogliono 2 $\quad (290 \times 2 = 580\ kN \cdot m \simeq 560)$





4.5 Effects of Fender Compressions

After having calculated the probable impact energy a ship will have when berthing, one can deduce from the manufacturer's catalogues the compression of the various fenders and the thrust the latter will transmit to the structure. Manufacturers always provide two diagrams for fenders, one showing the relationship between energy and compression and the other the impact force/compression relationship.

In figure 4.5.A two such diagrams have been combined to illustrate what happens when a ship is berthing. The fender with an 1500 mm $_f$ and 800 mm ID and a 1500 mm length will with 50% compression absorb an impact energy of 330 kNm. The resulting force to be resisted by the berth structure will be 900 kN with a fender factor P/E_f 900/330 = 2.7 kN/kNm. What is interesting about these large fenders which are designed for bigger ships, is that they have

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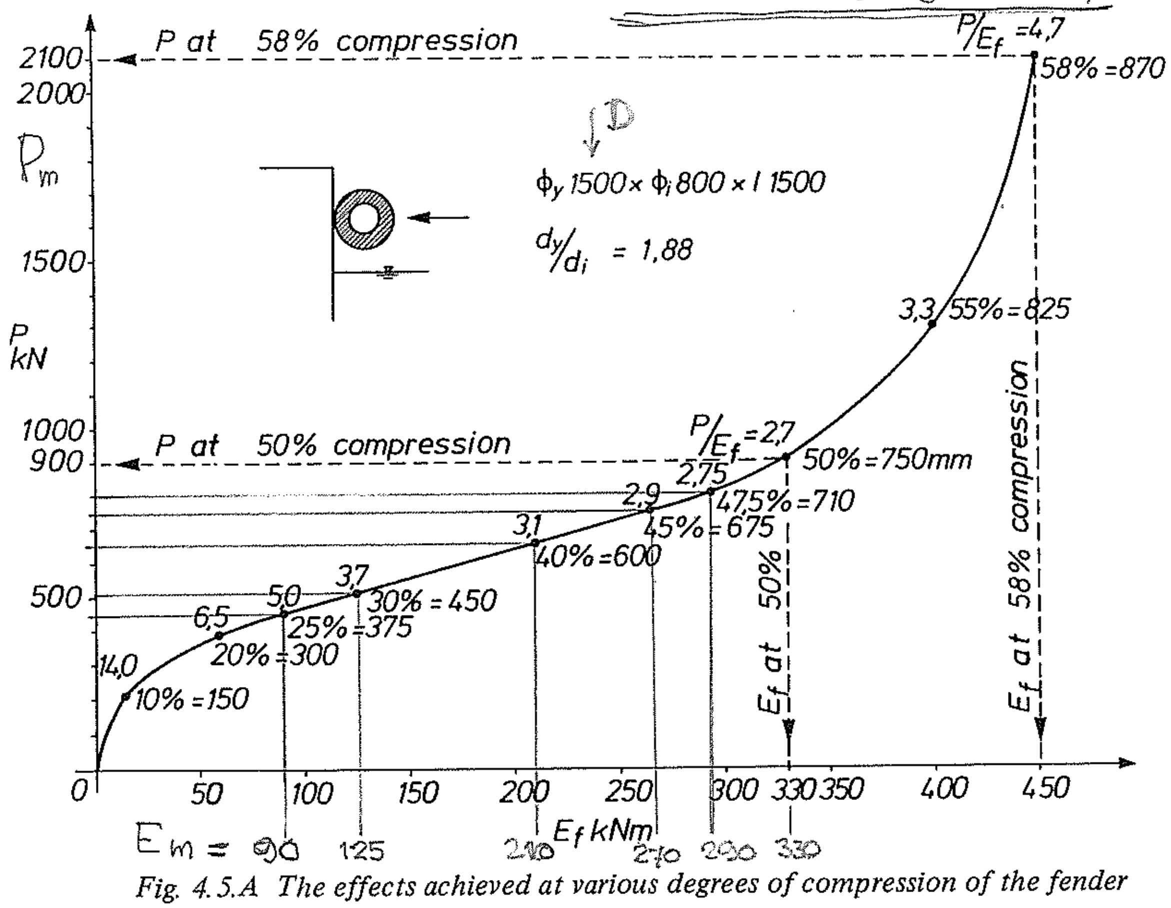


Fig. 4.5.A The effects achieved at various degrees of compression of the fender

a high fender factor with low compression (at 10% compression P/E_f = 14.0 kN/kNm. Where smaller ships are concerned, they will have little energy-absorbing effect but function more as surface-protecting fenders. The curve shows that the fender factor decreases with increasing compression, as far as 50% when it is 2.7 kN/kNm. Beyond this the factor increases with increasing compression.

Invece di adottare due manicotti sovrapposti in casi particolari si può ricorrere al DOUBLE F. S.

The double-fender system shown in figure 4.7.A-D, is often called an ideal fender system because of its energy absorption and reaction force characteristics. When a cell fender and a cylindrical fender are combined in a double-fender system as illustrated in figure 4.7.B, the cylindrical fender will «soften» the reaction/compression characteristics of the double-fender unit. This will make the double-fender more useful and it will act as an energy-absorbing fender also for smaller ships (see chapter 4.5). The cylindrical fender must be so large that the reaction force when the cylindrical fender is closed (compression equal to about 50% of the outside diametre), is equal to the reaction force needed to compress the cell fender. In order to prevent the cylindrical fender to be compressed more than about 50%, a compression stopper or protector can be mounted as illustrated in figure 4.7.B.

DOUBLE FENDER

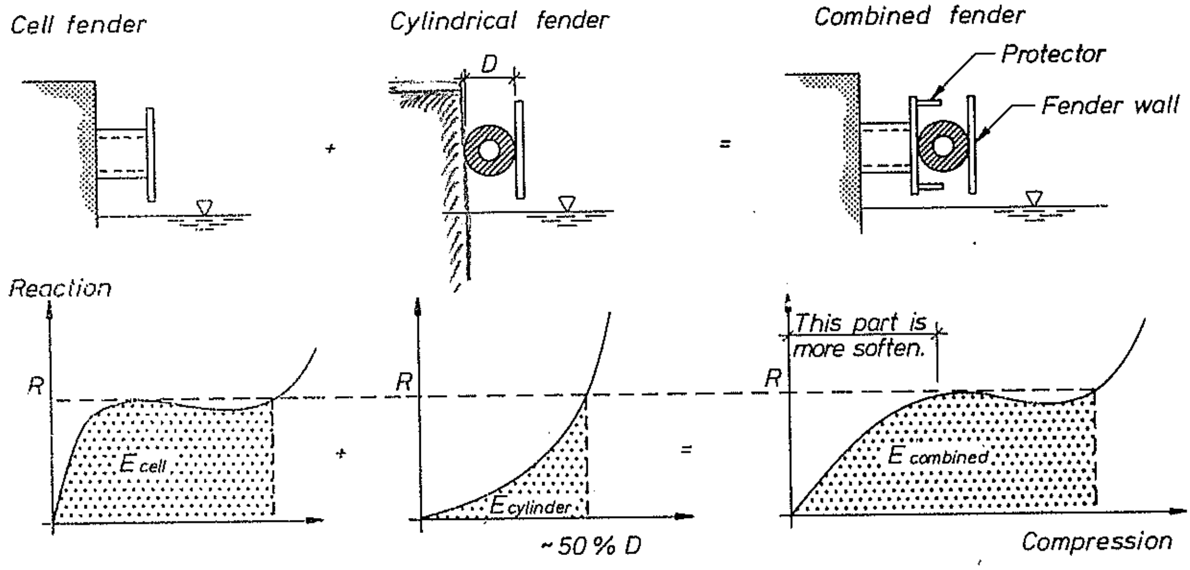


Fig. 4.7.B Reaction/compression characteristics of double fender

The design of a double-fender system should be done according to the trail-and-error method, and the procedure will be as follows:

- Calculate the ship's impact energy
- Choose an impact force P equal to the horizontal force the berth structure or ship's hull can resist divided by a safety factor.
- Check that the total fender energy absorbed is at least equivalent to the ship's impact energy.
- Check the fender factor.

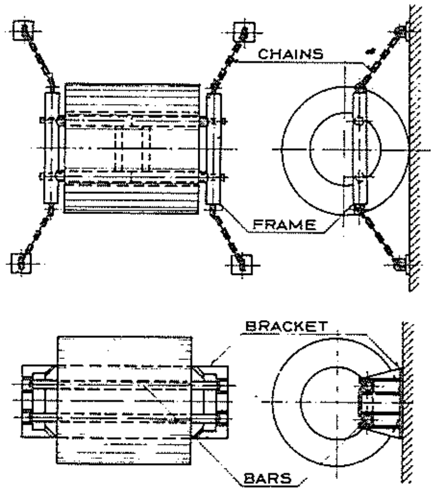


Fig. 8.7 (c) Installation Arrangement for Hollow Cylindrical Fenders

the same energy but the reaction for 5 is less than for 3 and 4. In addition, the contact pressure is smaller and fender 5 weighs about 5 percent less than the other two. It is therefore difficult to see how fenders 3 and 4 would be preferred to 5.

The influence on fender characteristic by the rubber compound is indicated by comparing fenders 6, 7 and 8. With a softer compound, the rated energy capacity and the reaction force decreases together with the contact pressure, all of which are important parameters.

Fenders Nos. 1, 2 and 9 to 11 have been included merely to indicate the range of possibilities. The load/energy ratio (P/E) decreases from 5.5 to approximately 1.5 with increase in fender size.

It is common to install hollow cylindrical fenders on the quay wall supported by brackets and bars or by frames suspended in chains and let the vessels moor directly against them, see Fig. 8.7(c). If water level variations are so that small vessels will be trapped under the fenders or if the fenders serve vessels with horizontal fenders themselves, e.g. ferries, it might be expedient to place pile supported pannels in front of the rubber fenders as indicated in Fig. 8.7(d). Flexible piles fixed in the sea bed will assist in absorbing energy.

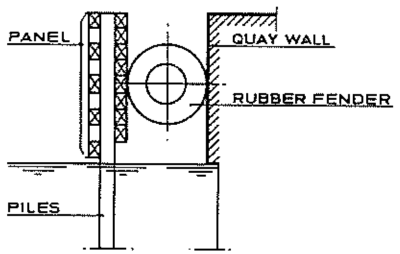


Fig. 8.7 (d) Hollow Cylindrical Fenders with Panel for Smaller Vessels

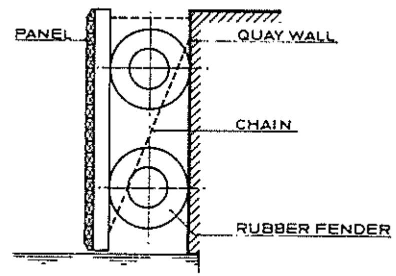


Fig. 8.7 (e) Hollow Cylindrical Fenders with Panel for Increase of Contact Area

Where the contact pressure between fender and vessel is not acceptable, softer hollow cylindrical rubber fenders may be used or panels of steel and/or hardwoods may be used to increase the contact area. Such panels, which will also reduce the friction force between fender and vessel could be supported on piles or suspended in chains as indicated in Fig. 8.7(e).

Axial loading of hollow, cylindrical rubber fenders is possible, but only if they are covered by a front panel. Characteristics of such fenders are of course, completely

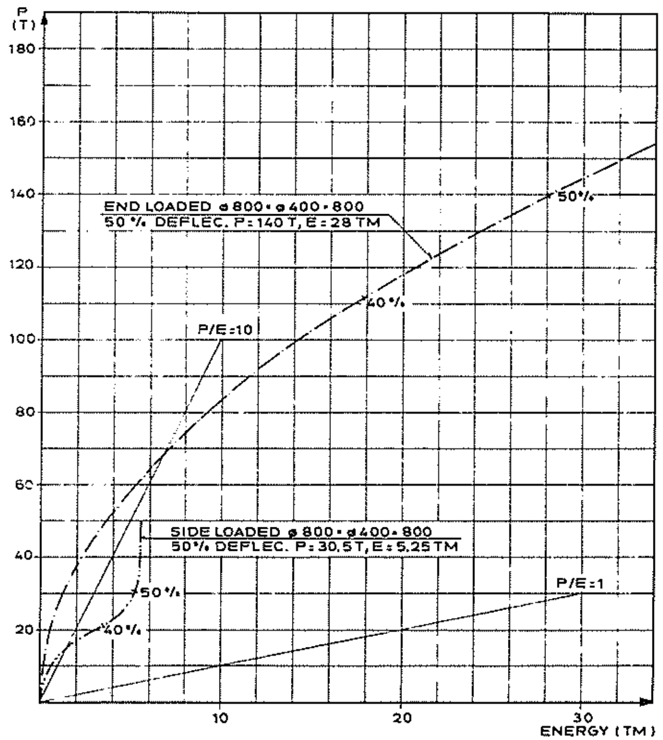
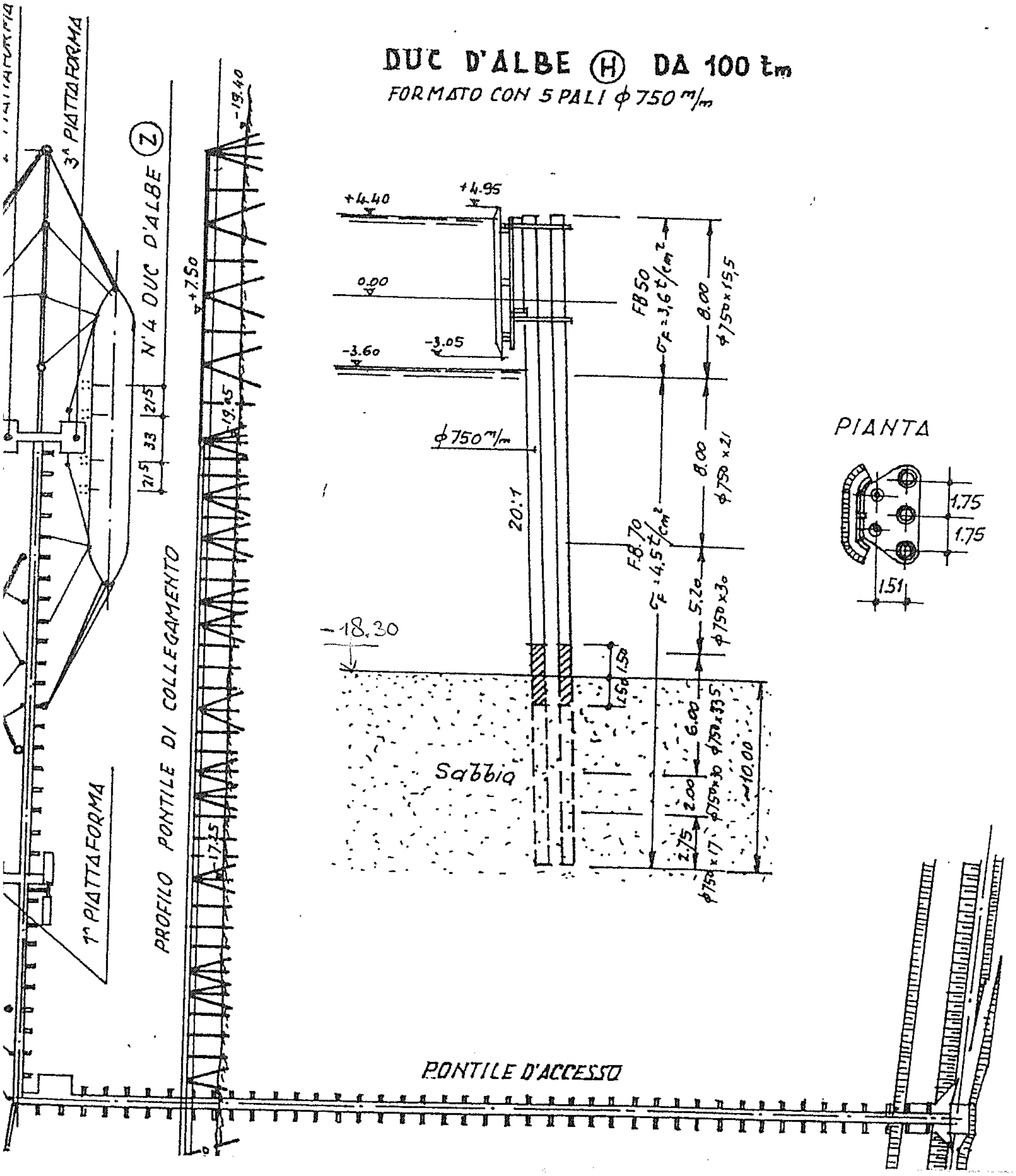


Fig. 8.7 (f) Load-Energy Curves for Hollow Cylindrical Fender. End and Side Loading

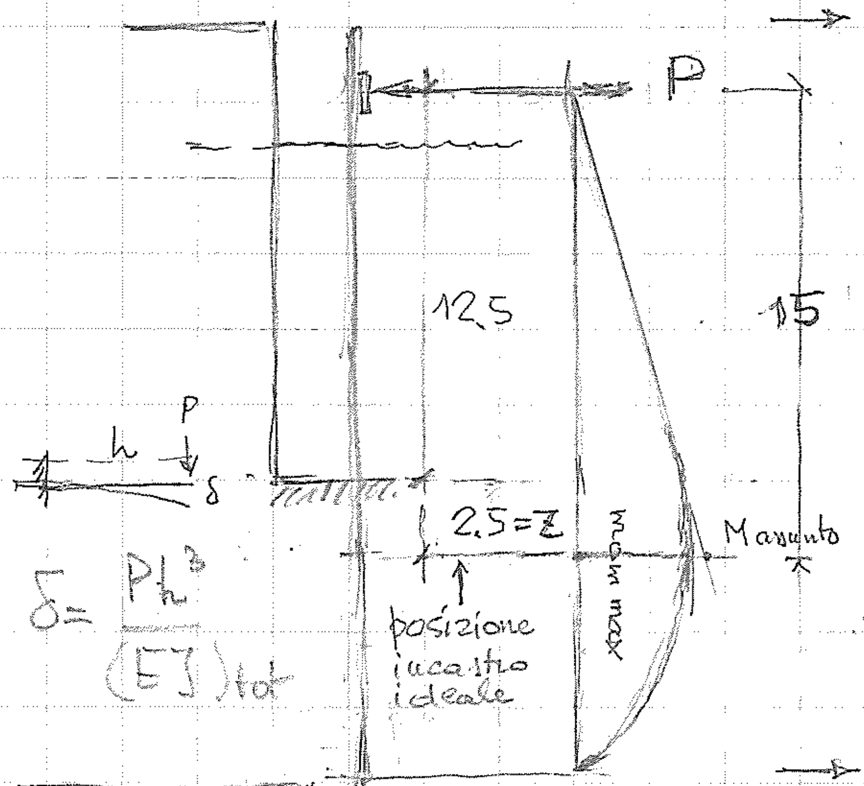
ANTO DI SCARICO DI WILHELMSHAVEN (1958)

RMINALE DELL'OLEODOTTO WILHELMSHAVEN (COLONIA)



Dimensioniamo con $E_f = 560$ kN·m un dolphin elastico ad es di 6 pali Diam 0,70 m spessore 0,015 m

→ P in kN è l'incognita



$\delta = \frac{Ph^3}{(EJ)_{tot}}$

$E = 200\cdot10^6$ kN/m² mod. di elast.

$J = \pi s R^3 = 2\cdot10^{-3}$ m⁴ (1 palo)

$EJ = 400\cdot10^3$ kN·m² (1 palo)

$$\delta = \frac{P\cdot15^3}{(6\cdot400\cdot10^3)} = P\cdot1,4\cdot10^{-3}\ \text{m}$$

EJ_{tot}

$$\frac{P\cdot\delta}{2} = 560 \ \text{kN·m}$$

↑ Dato

$$\frac{P\cdot(P\cdot1,4\cdot10^{-3})}{2} = 560 \ \text{kN·m}$$

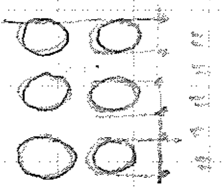
→ $P = 894$ kN su 6 pali

$\delta = 2\cdot560/894 = 1,25$ m!

troppo flessibile.

NB il calcolo è per tentativi perché z è funzione di P (livello M max)

$$\frac{z^3}{6} + \frac{z^2 b}{2} - \frac{P}{\gamma' K_p} = 0 \quad \text{(Blum 1932, EAU 1980)}$$



Ho assunto $z = 2,5$ ora controllo

Pongo $b = 5\times0.7 = 3.5$ m

$\gamma' = 11$ kN/m³ (Sabbia)

$P = 894$ kN (calcolo)

($\varphi = 30°$; $\delta_p = 20°$) $K_p = 6.1$ (v. dolphin)

ottengo $z = 2,48$ m OK

se ammessi $\delta_p = 0°$ $K_p = 3$: $z = 3.42$ m

Proverei anche con $h = 12,5 + 3.42 \cong 16$ m

$M = 894/6 \times 15 = 2235$ kN·m

mom. flett. per ogni palo (leggermente maggiorato v. fig.)

$$\sigma = \frac{MD/2}{J} = 391125 \ \text{kN/m}^2$$

391 MPa

$(391125\times10^3)\cdot10^{-1}\cdot10^{-4} = 3911$ Kgf/cm²

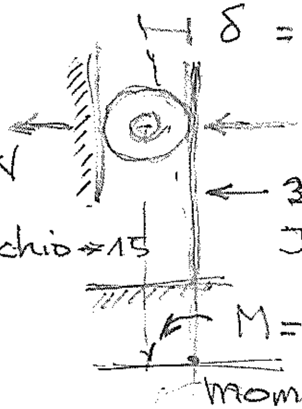
eccessivo: bisogna provvedere e fare i pali di maggior spessore

$D = 1{,}5\ m$ (manicotto)

v. CURVA "COMBINATA"

Sommo le E a parità di δ

δ/D	$(P/E)_m$	E_m	P_m	δ	P_d	E_d	E_{tot}
–	m^{-1}	kN·m	kN	m	kN	kN·m	kN·m
0.25	5.0	90	450	0.375	268	50	140
0.30	3.7	125	463	0.450	321	72	197
0.40	3.1	210	651	0.600	429	128	338
0.45	2.9	270	783	0.675	482	162	432
0.475	2.75	290	797	0.710	507	180	470
0.50	2.7	330	891	0.750	536	201	531 $\simeq$ 560 kN·m



$\delta = 0{,}75\ m$

891 kN

$891 + 536 = 1427$ kN

3 pali $\phi 700$ $s = 0{,}03\ m$

J 1 palo $= 4 \times 10^{-3}\ m^4$

a occhio ≈ 15

$$P_d = \delta / 1{,}4 \times 10^{-3}\ N \left(\delta = \frac{P_d \cdot 15^3}{\underbrace{200 \cdot 10^6}_{E} \cdot \underbrace{3 \cdot 4 \times 10^{-3}}_{J}} \right)$$

$$E_d = P_d \cdot \delta / 2\ N \cdot m$$

$M = 536 \times 15 = 8040$ kN·m (2680 kN·m per ciascuno dei 3 pali)

momento massimo = incastro

$\sigma \cong 235$ MPa

48 - LAVORI MARITTIMI IN AMBITO PORTUALE

48.10 - FORNITURA E POSA DI PARABORDO

			Euro	% Man
48.10.010	Provvista e posa in opera di parabordo formato da un manicotto cilindrico in gomma, colore nero, completo di complesso metallico di appensione e fissaggio, delle dimensioni di:			
48.10.010.010	381x191x1000 mm	cad	773,96	10,3
48.10.010.020	600x300x1000 mm	cad	1.898,74	7,7
48.10.010.030	1000x500x1000 mm	cad	4.297,90	3,4
48.10.010.040	1500x750x1500 mm	cad	10.322,84	1,4
48.10.010.050	1500x750x2000 mm	cad	14.845,39	1,4
48.10.020	Provvista e posa in opera di parabordi trapezoidali di sola gomma nera, completi dei relativi tirafondi e bullonerie, delle dimensioni di:			
48.10.020.005	300x1000 mm	cad	1.163,97	6,8
48.10.020.010	300x1500 mm	cad	1.635,97	4,9
48.10.020.015	300x2000 mm	cad	2.111,48	3,8
48.10.020.020	300x2500 mm	cad	2.610,91	3,1
48.10.020.025	400x1000 mm	cad	1.713,60	7,0
48.10.020.030	400x1500 mm	cad	2.395,75	5,0
48.10.020.035	400x2000 mm	cad	3.077,90	3,9
48.10.020.040	400x2500 mm	cad	3.789,94	3,2
48.10.020.045	500x1000 mm	cad	2.528,26	4,7
48.10.020.050	500x1500 mm	cad	3.633,55	3,3
48.10.020.055	500x2000 mm	cad	4.721,99	2,5
48.10.020.060	600x1000 mm	cad	3.349,59	3,6
48.10.020.065	600x1500 mm	cad	4.923,38	2,4
48.10.020.070	600x2000 mm	cad	6.461,30	1,9
48.10.030	Provvista e posa in opera di parabordi cilindrici di polietilene espanso rivestito con poliuretano, corredati di tubi centrali con piastre di sospensione, compresi golfari e catenacci di sospensione, delle dimensioni di:			
48.10.030.005	0,90x1,50 m	cad	5.069,69	6,4
48.10.030.010	1,22x2,44 m	cad	8.186,01	4,0
48.10.030.015	1,22x3,05 m	cad	9.871,57	3,3
48.10.030.020	1,52x2,44 m	cad	10.936,48	3,0
48.10.030.025	1,52x3,05 m	cad	13.595,49	2,4
48.10.030.030	1,83x3,66 m	cad	19.279,35	1,7
48.10.030.035	2,44x3,66 m	cad	32.039,62	1,1
48.10.030.040	2,80x4,00 m	cad	41.385,14	0,8

48.11 - FORNITURA E POSA DI PALANCOLE

			Euro	% Man
48.11.010	Provvista e posa in opera mediante infissione di palancole del tipo Larseen di qualita' ST 37 ST 45 nei profili normali, compreso l'eventuale taglio e regolarizzazione delle teste, la movimentazione del macchinario e dei mezzi necessari :			
48.11.010.010	per esecuzione via mare.	Kg	2,59	18,4
48.11.010.020	per esecuzione via terra.	Kg	1,89	11,5