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DEVELOPMENT of ANALYTICAL TECHNIQUES for the ASSESSMENT of ENERGY ABSORPTION MECHANISMS in MARINE FENDER SYSTEMS

March 1983

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Prepared for

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Performance algorithms have been determined relating energy and load/deflection characteristics for various generic types of fendering systems. These systems are typically rubber, pneumatic, foam filled, etc., and have been investigated herein.

The fender type variables have been identified to define the particular performance relationships. This generalized relationship is based on the

(Continued on reverse side)

performance data acquired from numerous manufacture sources for different types of fender systems. The correlation and condensation of particular energy performance curves substantiates the choice of the volumetric interrelation as a significant relationship for the fender types.

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APPENDIXES

I. REVIEW OF TASK OBJECTIVE AND ESSENTIAL TASK ELEMENTS

OBJECTIVE

The proposed two-phased effort is intended to result in an analytical procedure which is capable of predicting the response associated with a fender/vessel interaction. As part of Phase I efforts, performance algorithms would be determined relating energy and load/deflection characteristics for various generic types of fendering systems. These systems are typically rubber, pneumatic, foam-filled, etc. and have been investigated herein. The resulting algorithms would be used to characterize a particular generic type fender which would be represented in the response vessel/fender interaction procedure developed in Phase II.

The response analysis to be developed would be capable of estimating the following:

a. The maximum energy absorbed by a generic type fender represented in the response problem.

b. The maximum reaction load input to the pier and vessel hull during response.

c. The maximum local deflections occurring in the fender and hull during response.

d. The relative amounts of energy stored in fender, local hull, hull mode motions and hydrodynamic dissipation during response, thus defining the energy storage requirements for the fender system.

The benefits of fender/vessel response capability would include:

a. The ability to simply explore a specific vessel/fender response subject to design constraints such as maximum permissible reaction loads and deflections or fender system energy absorption requirements.

b. The ability to plug in and out alternative fender performance characteristics via algorithms and explore overall vessel/fender response.

c. The capability of optimizing a specific vessel/fender response for a given set of problem constraints.

ESSENTIAL TASK ELEMENTS - PHASE I

Task 1 - Acquire, review and assess the performance characteristics of currently used fendering systems in the available literature in order to establish a data base of energy absorption and load deflection data.

Task 2 - Determine algorithms which quantify fender system performance for generic fenders. Identify and rank energy absorption mechanisms for these fender systems.

Task 3 - Based on the literature search performed, identify an approach leading to the development of a rigorous analytical technique for predicting vessel/fender interaction. This technique will be fully developed in Phase II work.

ESSENTIAL TASK ELEMENTS - PHASE II

Task 1 - Formulate the generalized equations of motion for the vessel/fender dynamic interaction problem based on the approach identified in Phase I work. This approach will consider fender performance algorithms, local vessel stiffness, dock mass and stiffness, vessel and berthing characteristics.

Task 2 - Characterize vessel local hull or appendage stiffness.

Task 3 - Characterize dock stiffness and mass characteristics.

Task 4 - Characterize hydrodynamic mass and damping for vessels considered.

Task 5 - Computer code methodology.

Task 6 - Validate results against existing experimental data.

Task 7 - Validate results against proposed test program.

II. DATA BASE ACQUISITION RELATIVE TO FENDER-RELATED INFORMATION

A data base of fender performance data and related information has been established during the initial efforts of this project. Some 110 reports, papers and manufacturers' catalogs have been accumulated and are included as a bibliography with brief abstracts in Appendix A.

The sources of information relative to fender performance data have been designated by single asterisks (*). Similarly, those sources relative to the vessel/fender response problem have been designated with a double asterisk (**).

A search of the available literature listed in Appendix A resulted in the following observations relative to fender performance data.

- The most significant source of energy and load deflection data is contained within the catalogs of individual marine fender manufacturers.
- Fender performance data listed within individual papers or reports generally reflect data extracted from manufacturers' catalogs.
- The most common generic type of fender for which performance data is available is rubber.
- A minimum, and in some cases negligible, amount of performance data is available for wood, gravity, torsion, hydropneumatic, hydraulic and spring fendering systems.
- The second and third most common fendering systems for which performance data is available is pneumatic bag and foam-filled fenders. There appears to be two primary manufacturers of pneumatic fenders, Sumitomo and Yakohama, and two primary manufacturers of foam-filled fenders, Seward and Samson.
- Much of the performance data for large size fenders, especially pneumatic bag types, is extrapolated and not the result of full-scale testing due to the magnitude of the loads required in compression.

III. GENERIC TYPE FENDERS INVESTIGATED

The result of extensive literature search relative to ships fendering systems concluded that fendering systems generally fall into two categories: one considered commonly available, the other highly specialized. Common type fenders are readily obtainable from many marine fender manufacturers who have performed extensive full-scale fender tests relative to energy absorption and load deflection data. These systems are widely used for commercial and naval applications. Other systems considered highly specialized have been determined to have little performance data available and have limited or questionable practical field application. By far the most common fender system, for which extensive performance data was available, was rubber fenders followed by pneumatic and foam-filled fender types. Specialized torsional, hydropneumatic, gravity and hydraulic fenders had little available performance data and relatively limited practical field application. The small quantity of data results available for specialized fender types was concluded so specific and unique to the system investigated that generalized fender performance relationships could not be readily determined. Generalized relationships require performance data for variations of fender system parameters. Thus, the performance data derived from a particular specialized fender test could not be generalized to describe the generic family-type action.

For the commonly available systems consisting of rubber, pneumatic and foam-filled systems, sufficient data was available for variations of system parameters, including basic geometric dimensions, materials, pressures, etc. in addition to full-scale test and extrapolated load and energy deflection performance data. Because of the availability of required information, rubber pneumatic bag and foam-filled fenders were selected for detailed investigation.

Included in this investigation were the following specific type fender systems:

- Rubber
  - Hollow cylinder - transverse loading
  - Hollow cylinder - axial loading
  - Trapezoidal - transverse loading
  - Solid cylinder - shear loading
  - Hollow cubic - transverse loading
  - Hollow cubic - shear loading
  - Rotary donut - transverse loading
- Pneumatic
  - Floating bag - transverse loading
  - Air block fenders - transverse loading
  - Air block cushions - transverse loading
- Foam filled
  - Floating bag - transverse loading

Table 1 indicates the various manufacturers of the fender systems investigated in this task.

Table 1: MANUFACTURER/FENDER AVAILABILITY

FENDER TYPE	MANUFACTURER	BRIDGESTONE	GOODYEAR	VREDESTEIN	JOHNSON	U.S. RUBBER	LORD	REGAL	SUMITOMO	SEIBU	UNIROYAL	FIRESTONE	VIKING	YOKOHAMA	SAMSON	SEAWARD	DUNLOP	MORSE
1. RUBBER																		
HOLLOW CYLINDER (T)		●	●	●	●	●	●		●	●	●		●				●	●
HOLLOW CYLINDER (A)		●	●	●		●	●				●	●						
TRAPEZOIDAL (T)			●					●		●	●							
SOLID CYLINDER (S)				●				●										
HOLLOW CUBIC (T)			●	●	●	●				●								●
HOLLOW CUBIC (S)																		●
BUCKLING COLUMN (A)							●											●
ROTARY DONUT (T)					●				●								●	
2. PNEUMATIC																		
FLOATING BAG (T)									●					●			●	
AIR BLOCK FENDERS (T)														●				
AIR BLOCK CUSHIONS (T)														●				
3. FOAM FILLED																		
FLOATING BAG (T)															●	●		

(T) TRANSVERSE LOAD (A) AXIAL LOAD (S) SHEAR LOAD

IV. DERIVED FENDER PERFORMANCE RELATIONSHIPS

APPROACH

Figure 1 illustrates typical energy and load/deflection performance data for a rubber fender system. In this case the data reflects the performance of a set of hollow cylindrical fenders under transverse loading. This data is typical of data obtained from various manufacturers and test reports. As can be seen, the energy and load relationships vary as functions of geometric and material properties between various manufacturers.

In order to characterize a particular type fender system, it is necessary to determine a relationship between the fender type variables capable of condensing or collapsing the particular performance relationships shown in Figure 1 into more generalized ones which represent a family of curves. An equation for this generalized relationship can then be determined as a function of the variables established. The generalized relationships are based on the performance data acquired from numerous sources and are the basis for characterizing the generalized performance of the fender type.

The resulting generalized relationships derived will have an associated degree of dispersion in the relationship which reflects the choice of variables selected to collapsed performance curves and the accuracy of the test or extrapolated performance data in addition to the effects from manufacturers' material differences. The more accurate the choice of collapsing and nondimensionalizing variables and the more accurate the available test or extrapolated data, the less dispersion will be evident in the relationships determined.

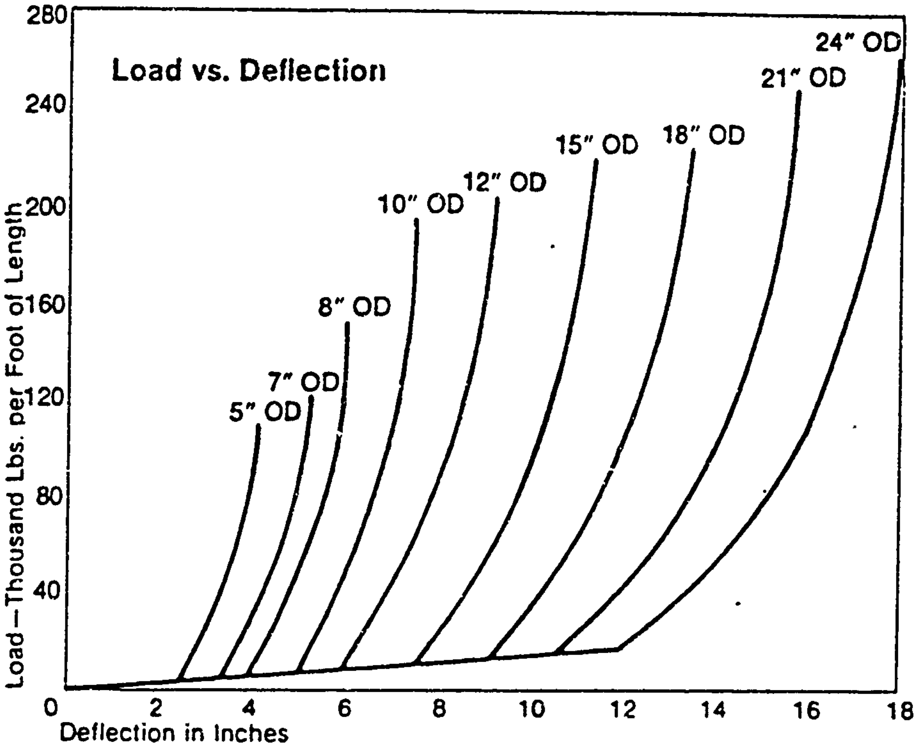
The collapsing variable identification process is illustrated typically in Figure 2 which correlates the energy-absorbing capacity of rubber cylindrical fenders at the rated 50 percent deflection to the volume of material tested.

Figure 1: TYPICAL PERFORMANCE DATA

Load/Deflection/Energy Absorption

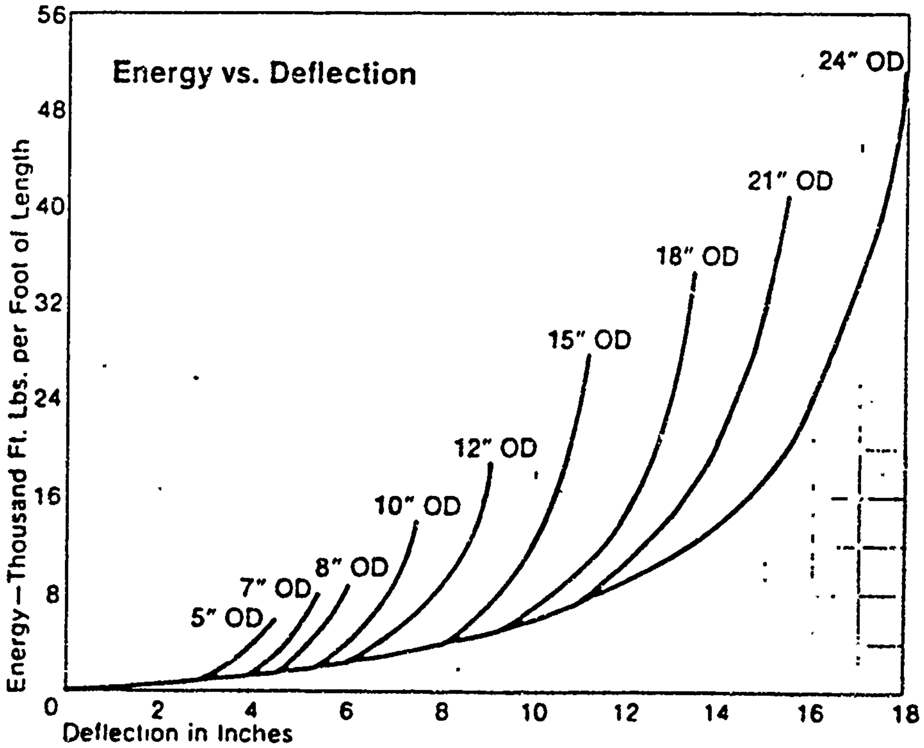
Cylindrical Hanging Fenders 5" to 24" O.D.

Load applied perpendicular to bore. Test length: 1 ft.



Approximate load vs. deflection

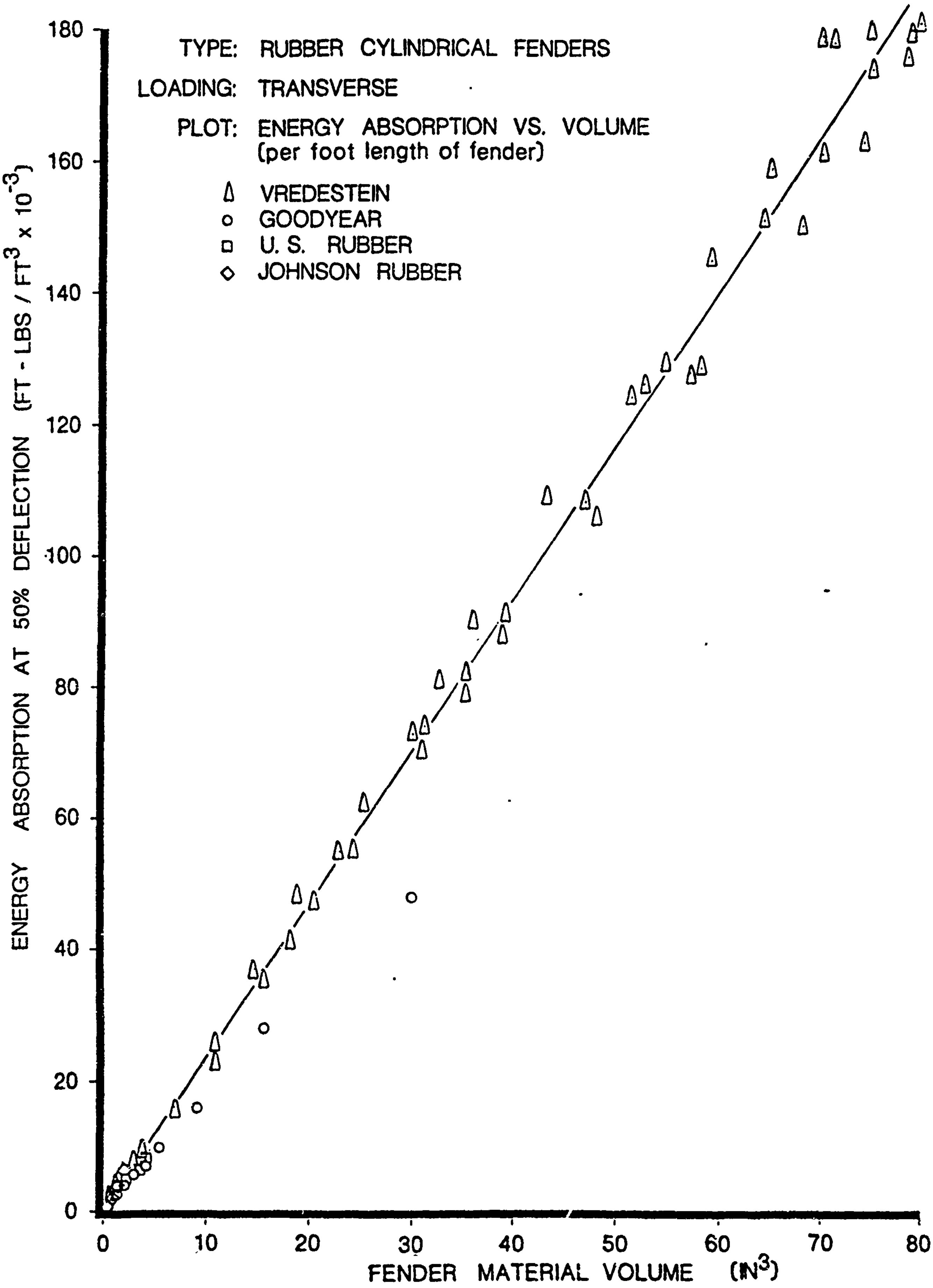
O.D. (in.)	I.D. (in.)	Lbs./ ft. lgth. @ 50%	Lbs./ ft. lgth. @ 67%
5	2½	3,500	25,000
7	3½	4,400	44,000
8	4	5,800	50,000
10	5	7,000	66,000
12	6	8,000	75,000
15	7½	10,000	85,000
18	9	12,000	101,000
21	10½	14,000	106,000
24	12	16,000	110,000



Approximate energy vs. deflection

O.D. (in.)	I.D. (in.)	Ft. lbs./ ft. lgth. @ 50%	Ft. lbs./ ft. lgth. @ 67%
5	2½	365	1,700
7	3½	650	3,000
8	4	970	3,800
10	5	1,460	5,200
12	6	2,000	5,800
15	7½	3,125	11,800
18	9	4,500	15,200
21	10½	6,125	22,800
24	12	8,000	24,000

Figure 2: TYPICAL GENERIC RELATIONSHIP PLOT



Intuitively, it is reasoned that a quantifiable relationship between energy capacity and volume exists but is not evident, since energy storage per unit volume may be significantly affected by manufacturers' brand materials and by the variable ratio of cylinder inner and outer diameters and any other pertinent factors. However, the high degree of linearity illustrated by the resulting relationship from four manufacturers' data sources based on some 70 cylinder sizes indicates that this relationship is well defined, highly linear and has very little associated dispersion, regardless of the material differences and variation of diameter ratio generally existing. This identifies energy storage per unit volume as a strong collapsing relationship for this fender type. (This relationship has prevailed for most types of rubber fenders investigated.) The above correlation and significant condensation of particular energy performance curves which occurs based on this relationship substantiates the choice of the volumetric relationship as a significant relationship for rubber fender types. A relative independence of the effects of geometry and material differences is implied by this relationship for fenders presently available from fender manufacturers. Figures 3 and 4 illustrate the typical effect of a valid and invalid choice of parameter selection. In Figure 3 it is evident that considerable dispersion results from a noneffective choice of condensing parameters in contrast to Figure 4 where the choice has resulted in a more well-defined relationship.

Once a satisfactory set of condensing or collapsing variables has been formulated and determined to result in a minimum of dispersion for the generic relationship when plotted, a polynomial regression analysis is performed to determine the polynomial equation of order (n) which best describes the energy/deflection or load/deflection generalized relationship for the fender type.

Figure 3: NON-EFFECTIVE PERFORMANCE CONDENSATION

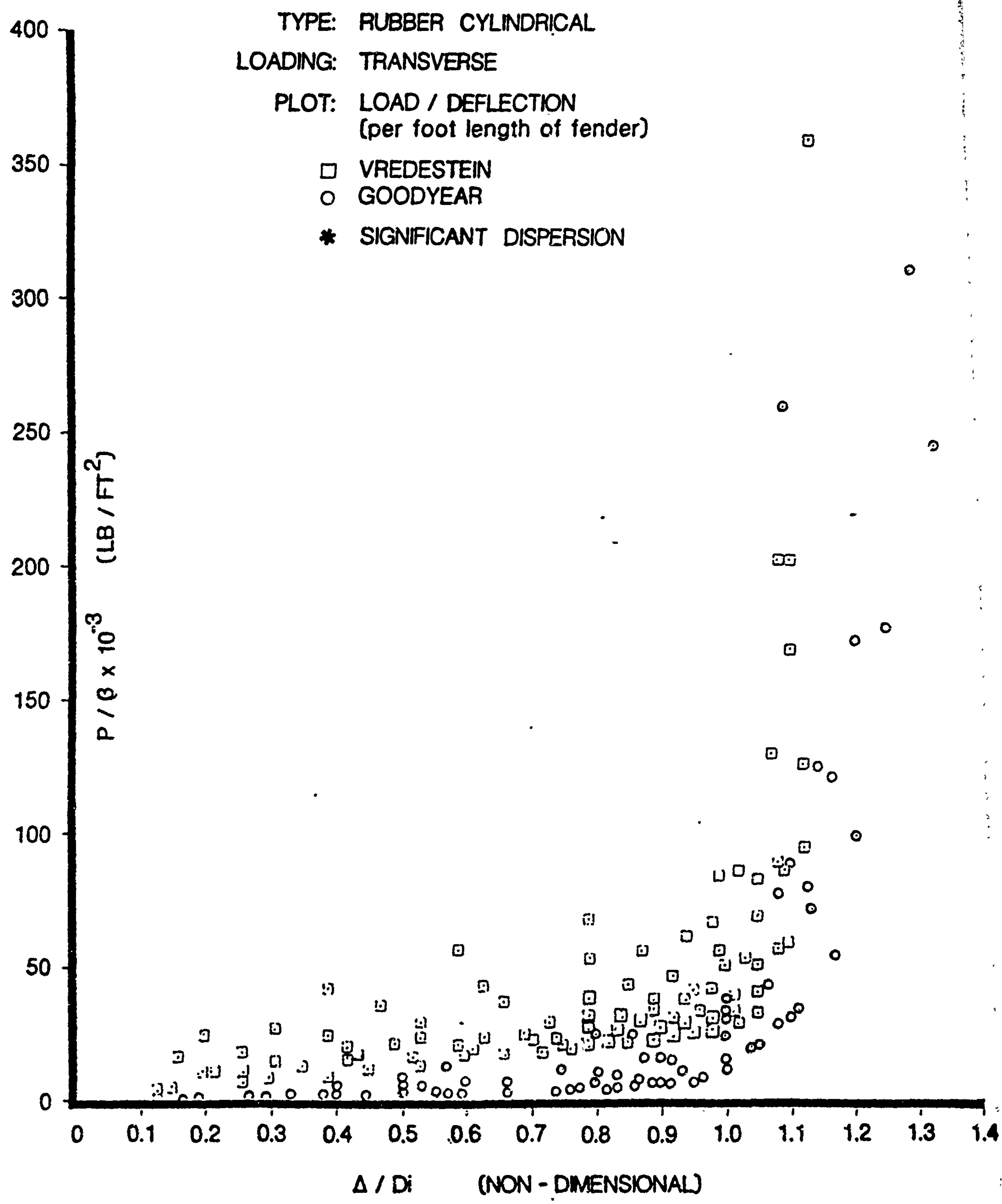
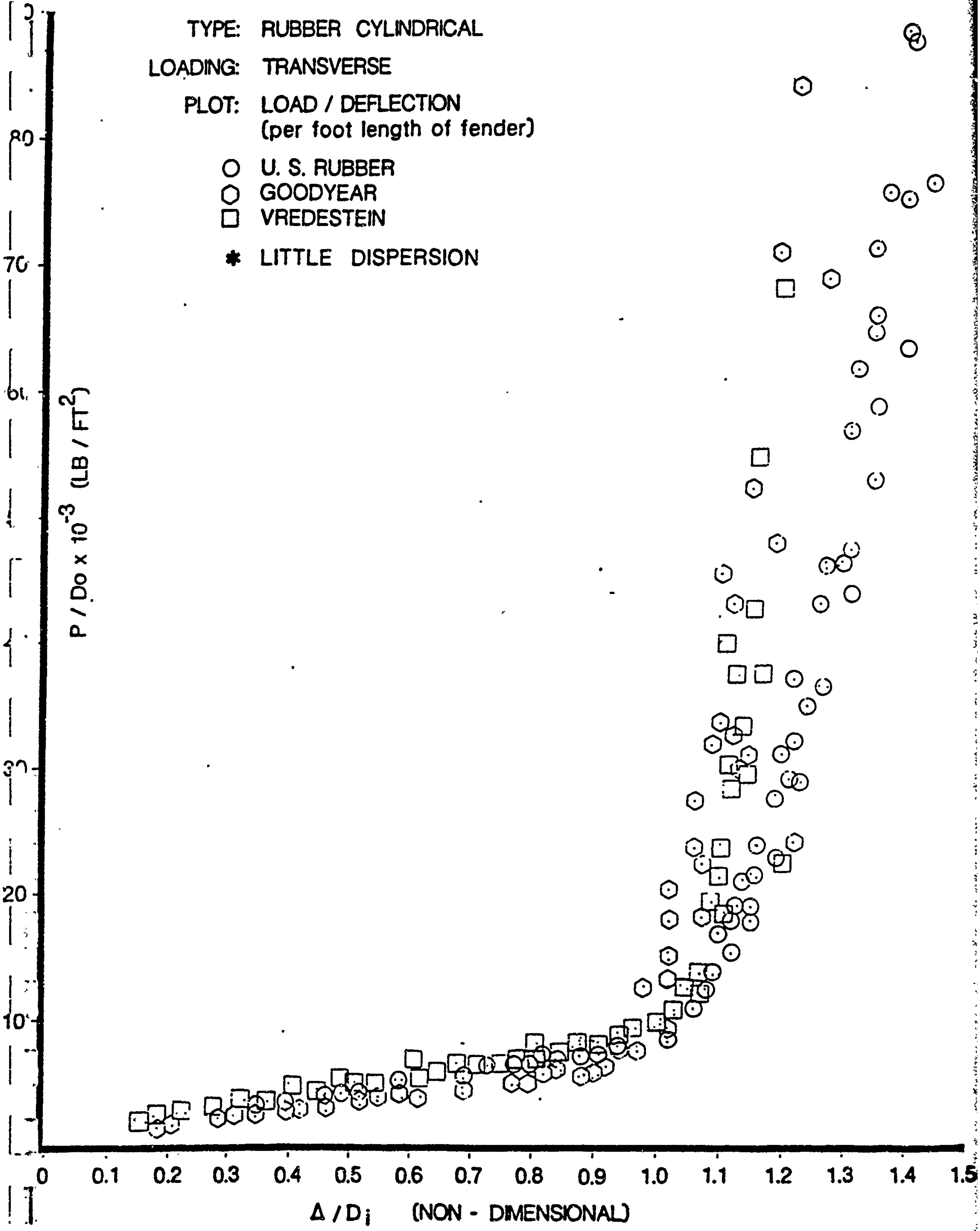


Figure 4: EFFECTIVE PERFORMANCE CONDENSATION



The equations which result are considered characteristic of the performance relationships which exist for the fender type based on available manufacturers' data.

POLYNOMIAL PERFORMANCE EQUATIONS - GENERAL

The polynomial equations derived from regression analysis of manufacturers' data for energy and load/deflection are of the following form:

$$E = \alpha \left\{ \sum_{i=0}^{n+1} \lambda_i X^i \right\} \qquad P = \gamma \left\{ \sum_{i=0}^{n+1} \psi_i Y^i \right\} \qquad (1)$$

where:

E = Fender energy absorption

α = Characteristic fender volume

P = Fender reaction load

γ = Characteristic fender area

X,Y = Nondimensionalized deflection-- $\frac{\Delta}{L}$
(L) = characteristic dimension of fender type

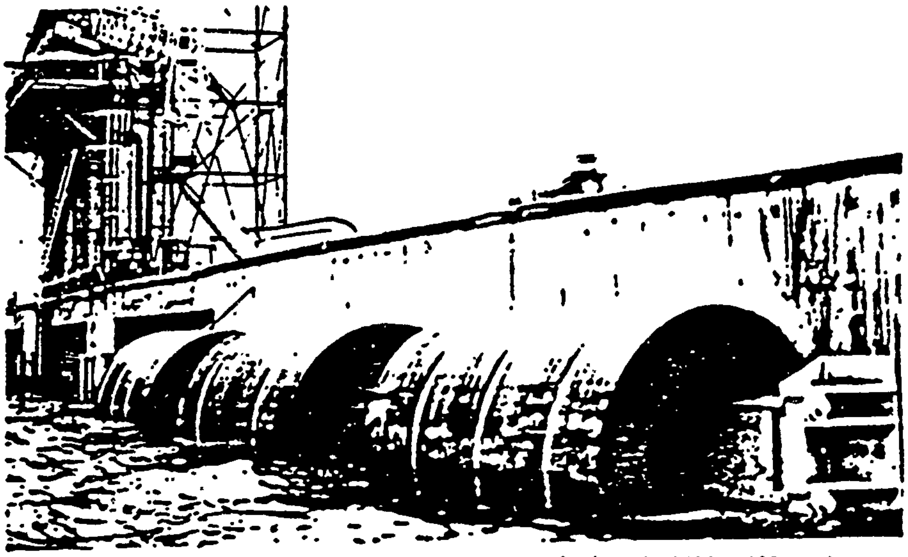
n = Order of polynomial equation used in regression analysis

λ_i, ψ_i = Polynomial coefficients

Tables 2 and 3 located at the end of this section, summarize the polynomial coefficients determined for energy absorption and load deflection described in detail in the following pages.

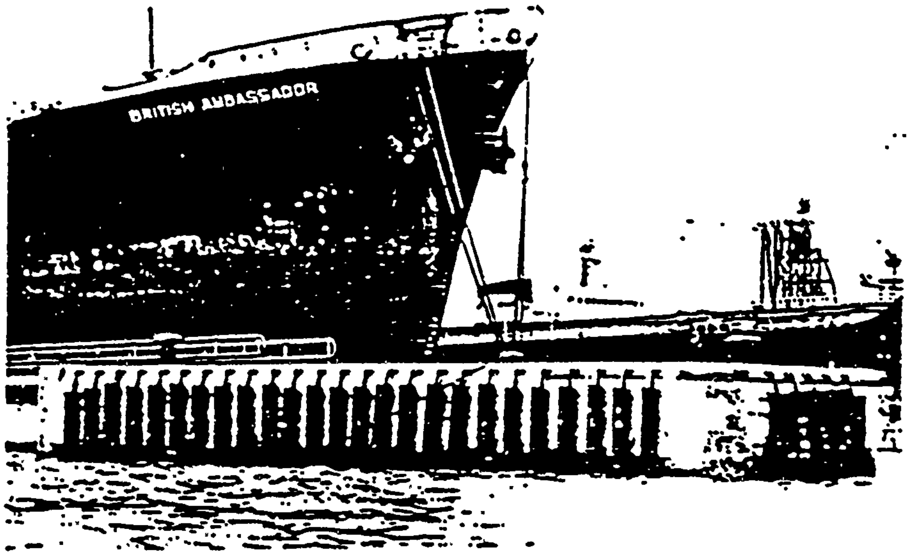
CYLINDRICAL FENDERS - TRANSVERSELY LOADED

Figure 5: CYLINDRICAL FENDER INSTALLATION



Mongstad (near Bergen), Norway
Oiltanker terminal

Jumbo fenders Ø 2700 x 1600 x 4000 mm long.



Antwerp Docks, Belgium

Cylindrical fenders Ø 380 x 190 x 2000 mm long.

CYLINDRICAL FENDERS - TRANSVERSE LOADING

This type fender is by far the most commonly available from various fender manufacturers as indicated in Table 1. The variables determined most effective in condensing the test energy absorption performance data were cylinder volume for energy absorption, and cylinder inside diameter for nondimensionalizing deflection. Figure 6 indicates the generic energy absorption relationship which resulted from the data sources considered. In this case, the length of cylinder considered is on a per foot basis. The relationship between variables which best fits the trend indicated in Figure 7 for energy absorption is:

$$E = \beta L \{0.09X - 5.07X^2 + 9.14X^3\} 10^3 \qquad (2)$$

where:

E = Energy absorption (ft-lb)

D_o = Outside diameter of the cylindrical fender (ft)

D_i = Inside diameter of the cylindrical fender (ft)

L = Length of fender (ft) (plotted in Figure 6 per foot length)

X = Δ/D_i nondimensional

Δ = Deflection under load (ft)

$\beta = \frac{\pi}{4} (D_o^2 - D_i^2)$ (ft^2)

The above equation is representative in the range of $X \leq 1.5$.

The load/deflection relationship indicated in Figure 7 can be characterized by the following equation.

$$P = D_o L \{105.76X - 254.88X^2 + 163.95X^3\} 10^3 \qquad (3)$$

where:

P = Reaction load (lb)

D_o = Cylinder outside diameter (ft)

X = Δ/D_i nondimensional

D_i = Cylinder inside diameter (ft)

Δ = Deflection under load (ft)

L = Length of fender (ft) (plotted in Figure 7 per foot length)

The above equation is representative in the range of $X \leq 1.5$.

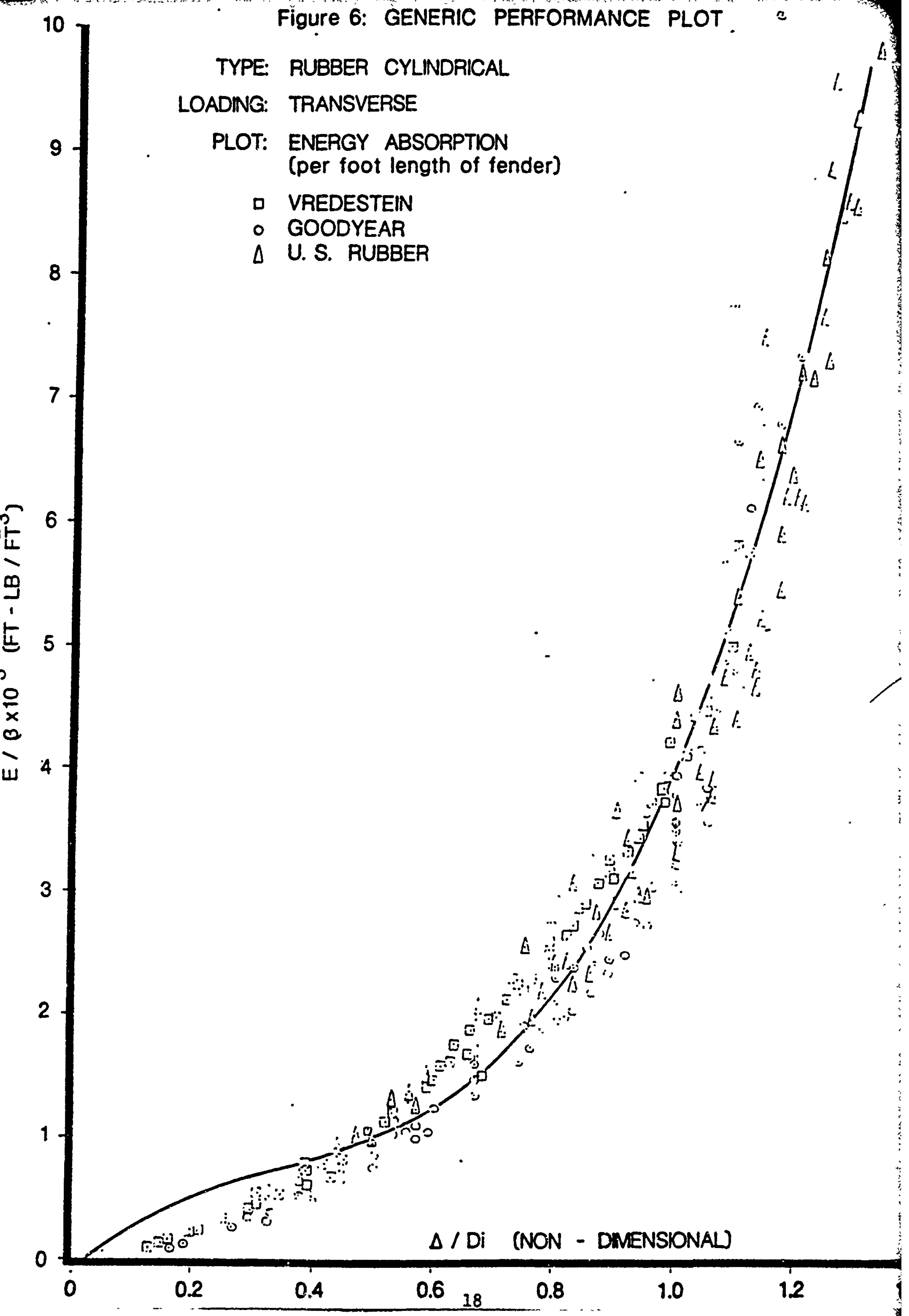
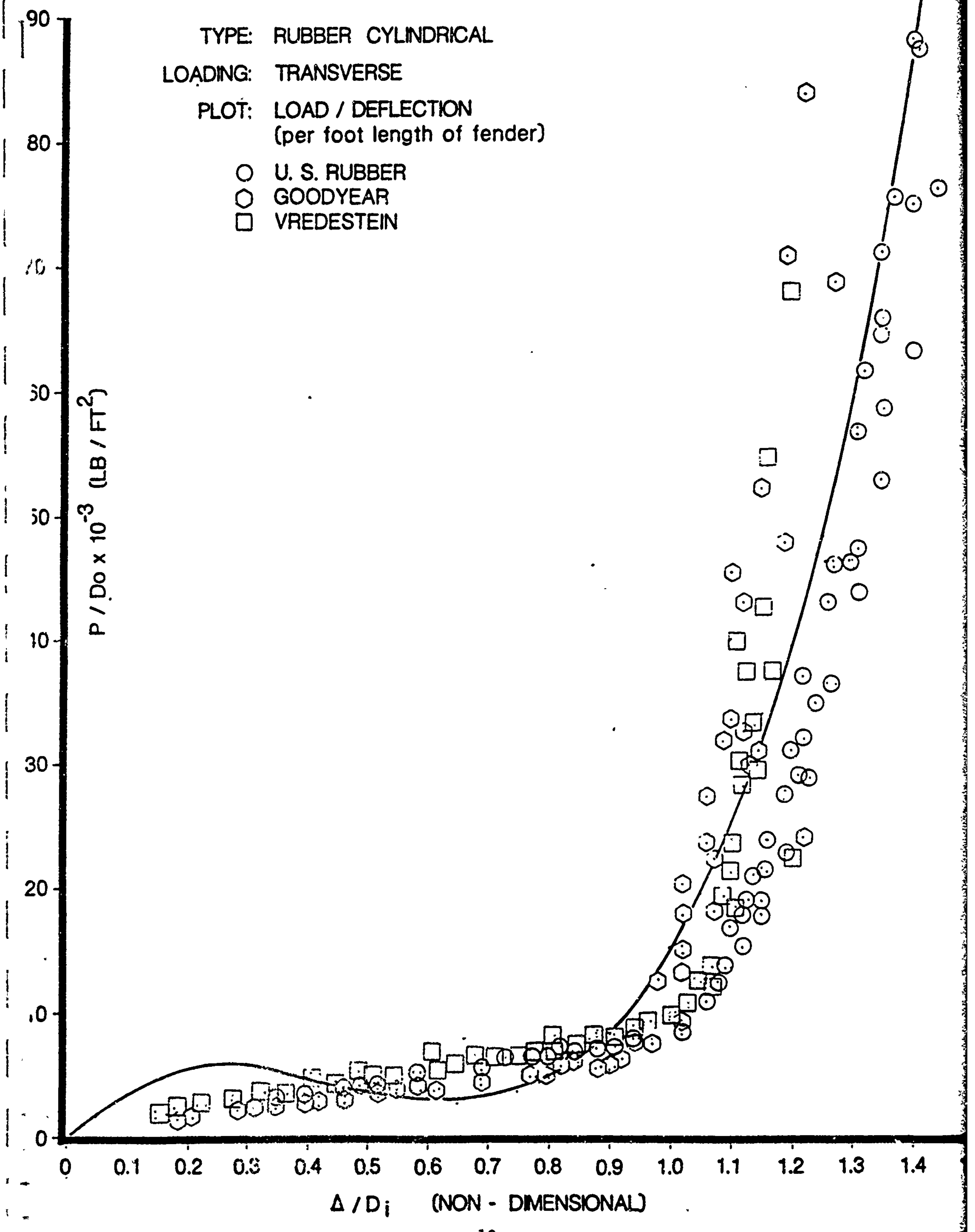


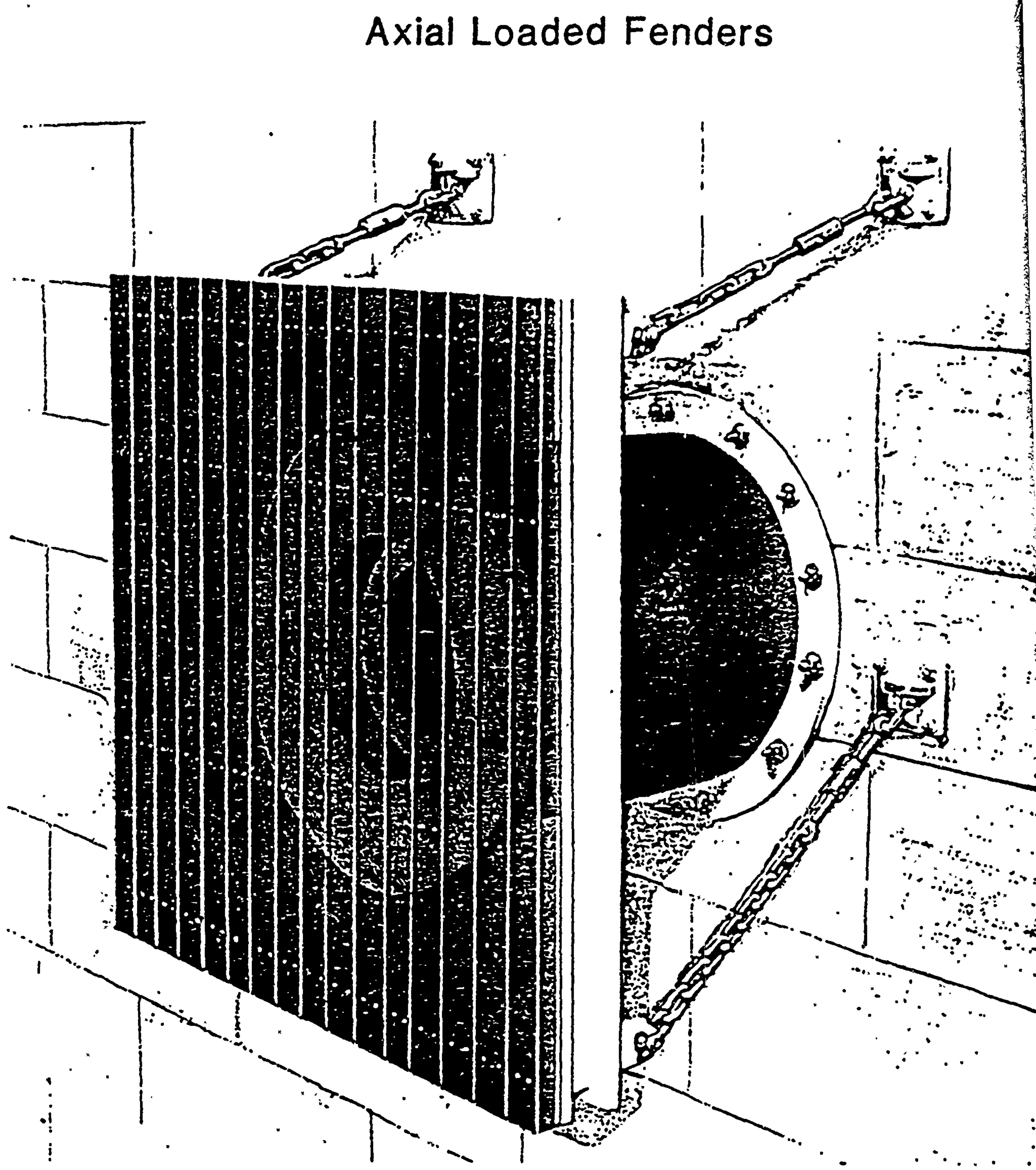
Figure 6: GENERIC PERFORMANCE PLOT
TYPE: RUBBER CYLINDRICAL
LOADING: TRANSVERSE
PLOT: ENERGY ABSORPTION
(per foot length of fender)
□ VREDESTEIN
o GOODYEAR
Δ U. S. RUBBER
E / β x10^-3 (FT - LB / FT^3)
Δ / Di (NON - DIMENSIONAL)
0
1
2
3
4
5
6
7
8
9
10
0
0.2
0.4
0.6
0.8
1.0
1.2

Figure 7: GENERIC PERFORMANCE PLOT



AXIAL LOADED CYLINDRICAL FENDERS

Figure 8: AXIAL LOADED FENDER INSTALLATION



AXIAL LOADED CYLINDRICAL FENDERS

Next to transversely loaded cylinders, the axial loaded fender was the most common type for which performance data was available. These fender systems are annular columns which compress and deflect as buckling columns with added energy capability resulting from its "hoop" effect. Being circular, they have equal shear resistance for all directions of transverse loading.

Figure 9 indicates the generic relationship for energy absorption determined for this type fender system. The energy-volume relationship determined for transversely loaded cylinders was equally effective for axial fender types. In this case, the nondimensionalizing characteristic dimension for deflection was the length of the cylinder. The resulting algorithm for energy absorption derived from Figure 9 was determined to be:

$$E = \beta H \{5.95X + 51.13X^2 + 20.79X^3\} 10^3 \tag{4}$$

The corresponding load/deflection algorithm illustrated in Figure 10 is:

$$P = \beta H \{140.69X + 6.4X^2 - 15.65X^3\} 10^3 \tag{5}$$

where:

P, E, D_o, D_i, β, Δ are previously defined.

$X = \Delta/H$

H = Height of fender (ft)

The applicability of the above equations is $X \leq 0.6$.

Figure 9: GENERIC PERFORMANCE PLOT

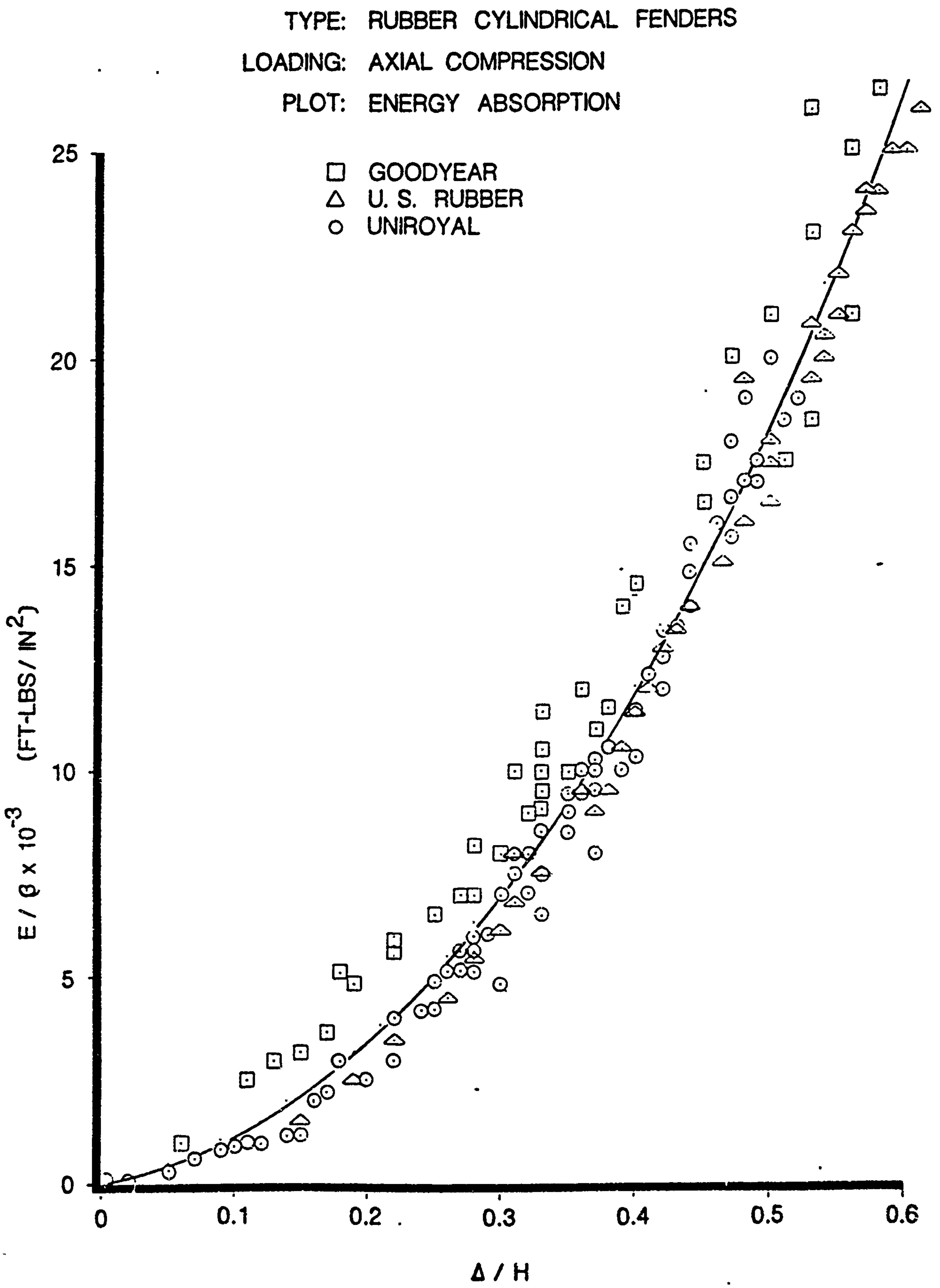
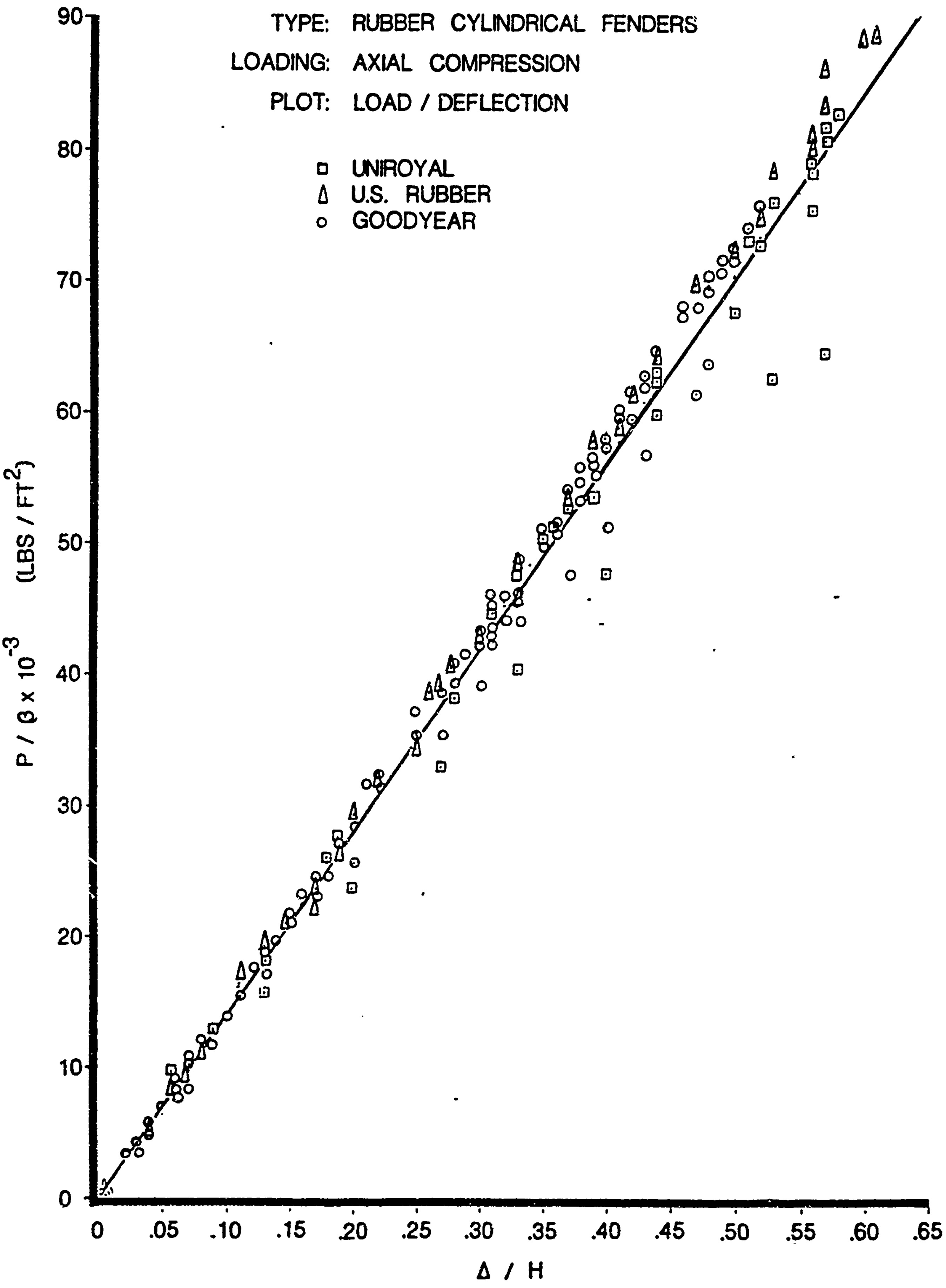
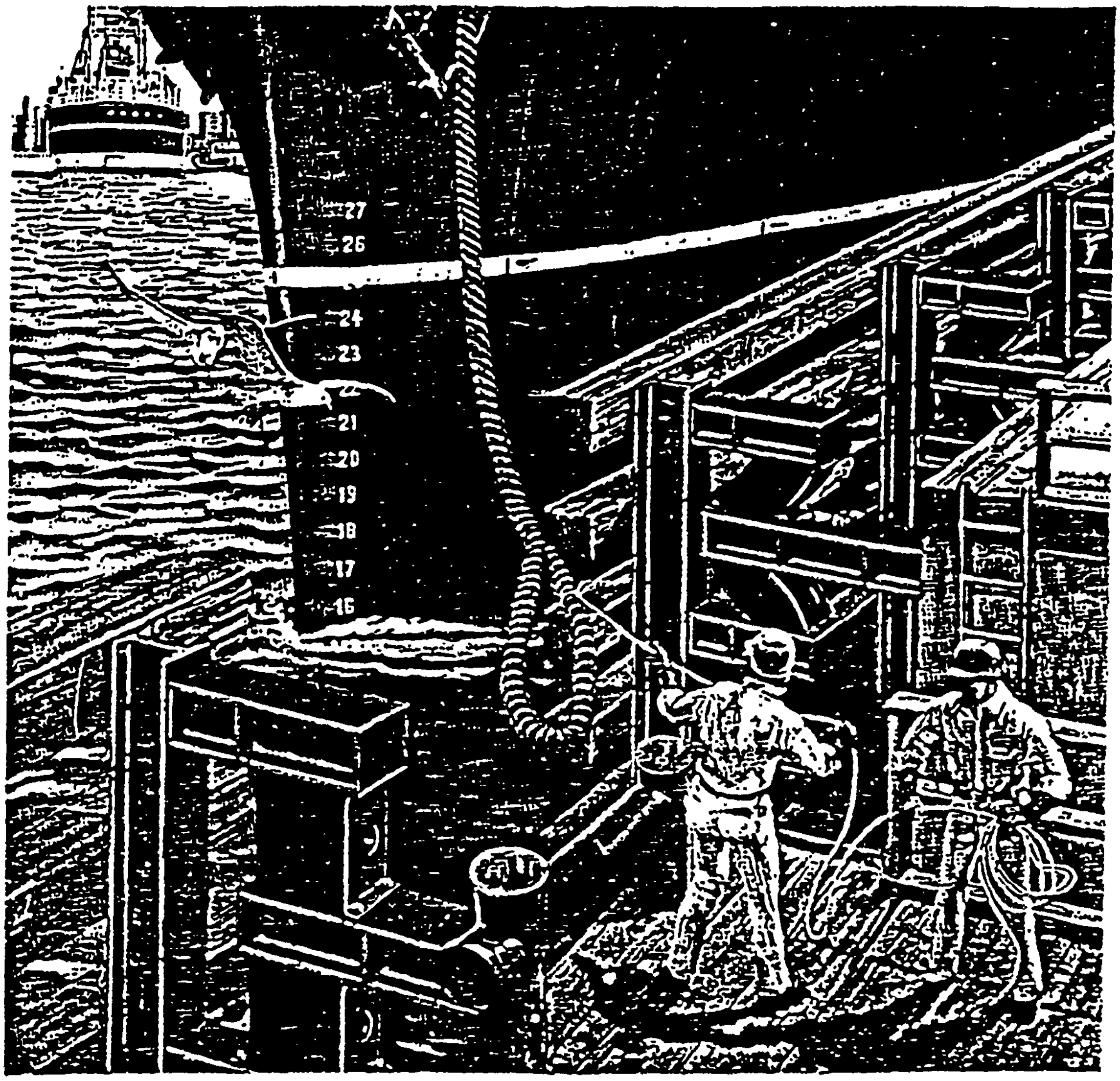


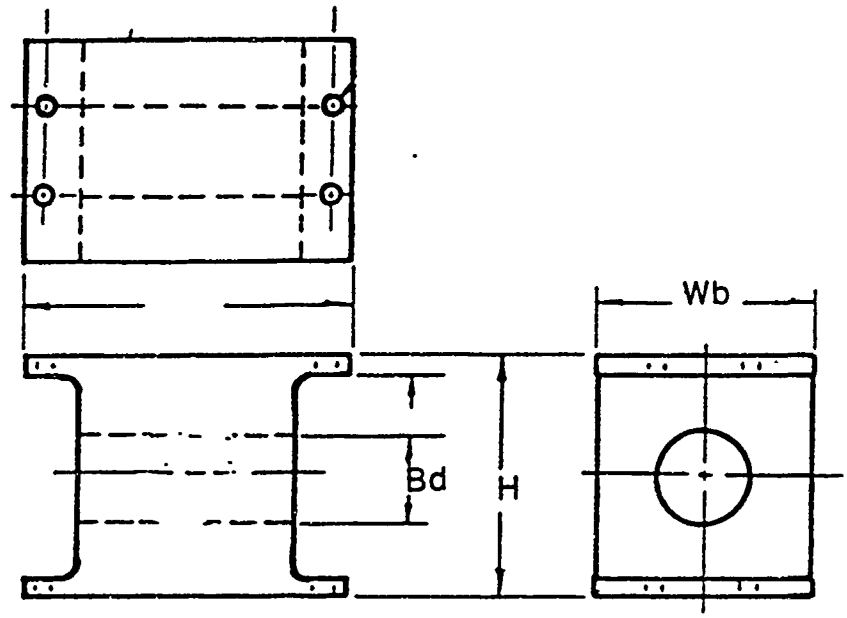
Figure 10: GENERIC PERFORMANCE PLOT



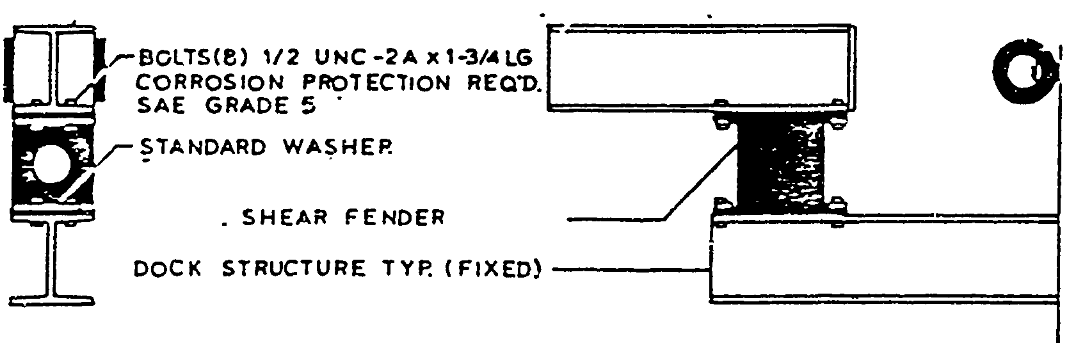
HOLLOW CUBIC RUBBER FENDERS - SHEAR LOADED

Figure 11: CUBIC SHEAR LOADED FENDER INSTALLATION





MOUNTING RECOMMENDATIONS



HOLLOW CUBIC RUBBER FENDERS - SHEAR LOADED

Shear fenders have the ability to stretch in four shear directions in addition to withstanding large compression loads and limited tension loads. This feature allows for wale movement away from, into or tangential to docks as vessels berth. In compression, the fender can be used alone or in tandem, bolted between a wall and whale. Tension and compression loading allow the shear fender to support the wale. This is illustrated in the previous figure (Figure 11). Although this type fender system is simple and effective, it is not commonly available from fender manufacturers as indicated in Table 1.

The data used to determine the appropriate performance algorithms has been selected from a single source but reflects the eight different size fenders available. These fenders are characterized by a cylindrical bore running lengthwise in the direction of shear loading.

Figure 12 indicates the characteristic energy absorption curve determined by correlating the fender energy volume and the normalized deflection relationship. In this case the characteristic length was the height of the shear fender normal to the direction of shear. The representative equation for energy absorption was determined to be:

$$E = \beta H \{2.63X - 5.39X^2 + 10.62X^3 - 3.44X^4\} 10^3 \qquad (6)$$

The corresponding load/deflection relationship illustrated in Figure 13 is:

$$P = \beta \{21.6X - 21.9X^2 + 19.76X^3 - 5.06X^4\} 10^3 \qquad (7)$$

where:

P = Reaction load (lb)

$\beta = W_b^2 - \frac{\pi}{4} B_d^2 \quad (ft^2)$

W_b = Width of fender base (ft)

B_d = Bore diameter (ft)

H = Height of shear fender (ft)

X = Δ/H (nondimensional)

E = Energy absorption (ft-lb)

The above equations are representative for $X \leq 1.9$.

Figure 12: GENERIC PERFORMANCE PLOT

TYPE: HOLLOW RUBBER CUBIC
LOADING: SHEAR
PLOT: ENERGY / DEFLECTION

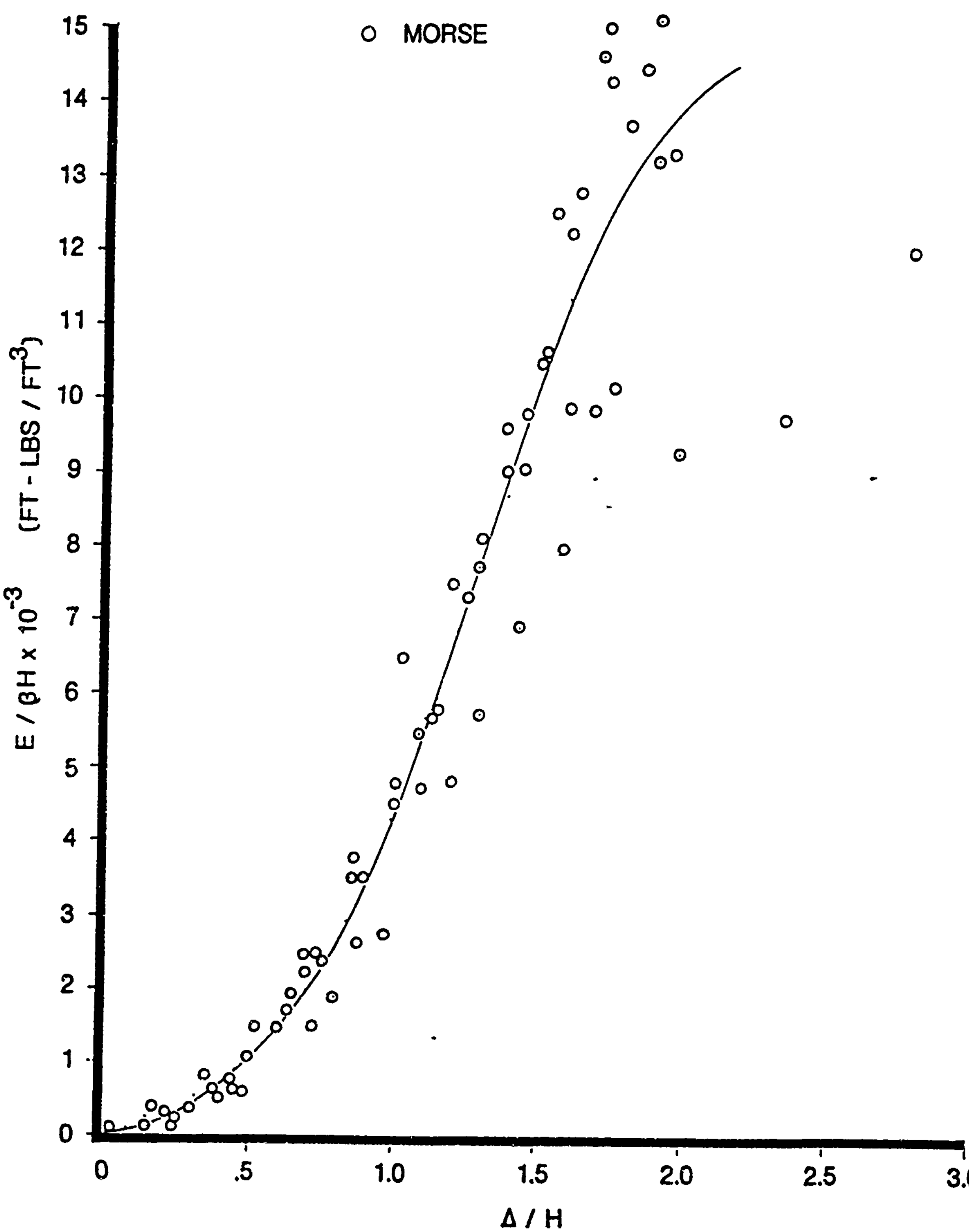
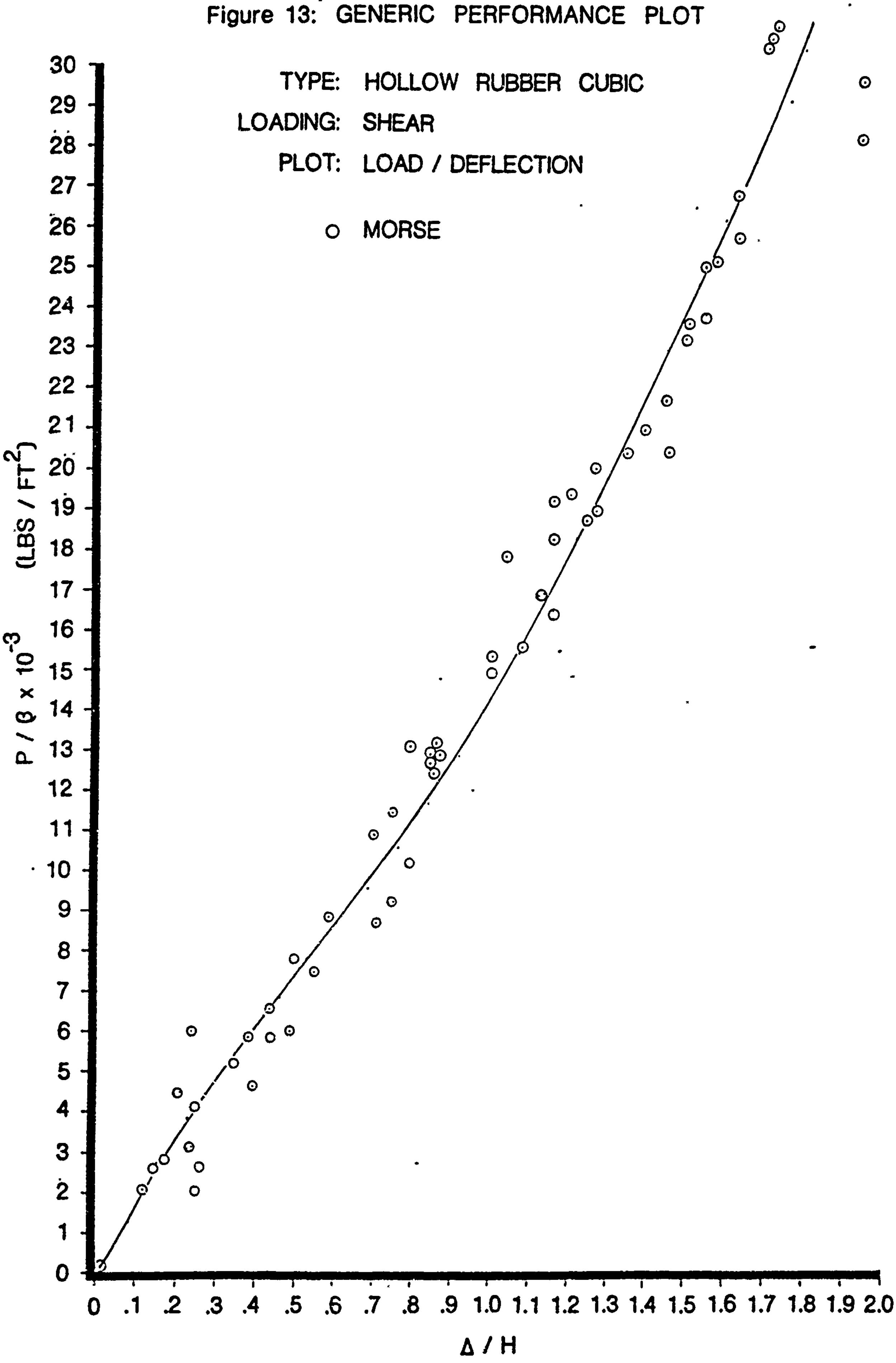
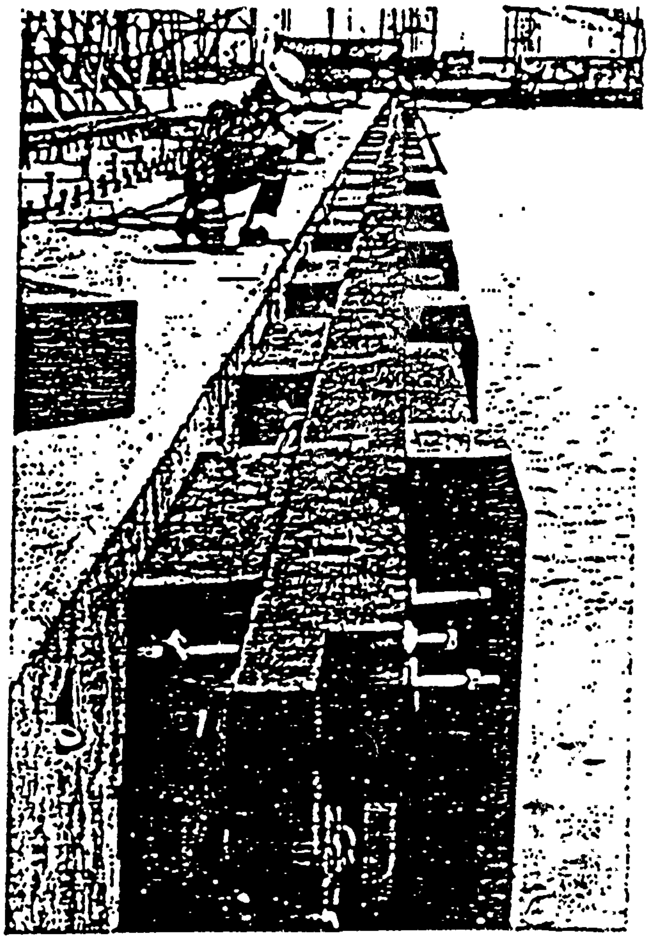


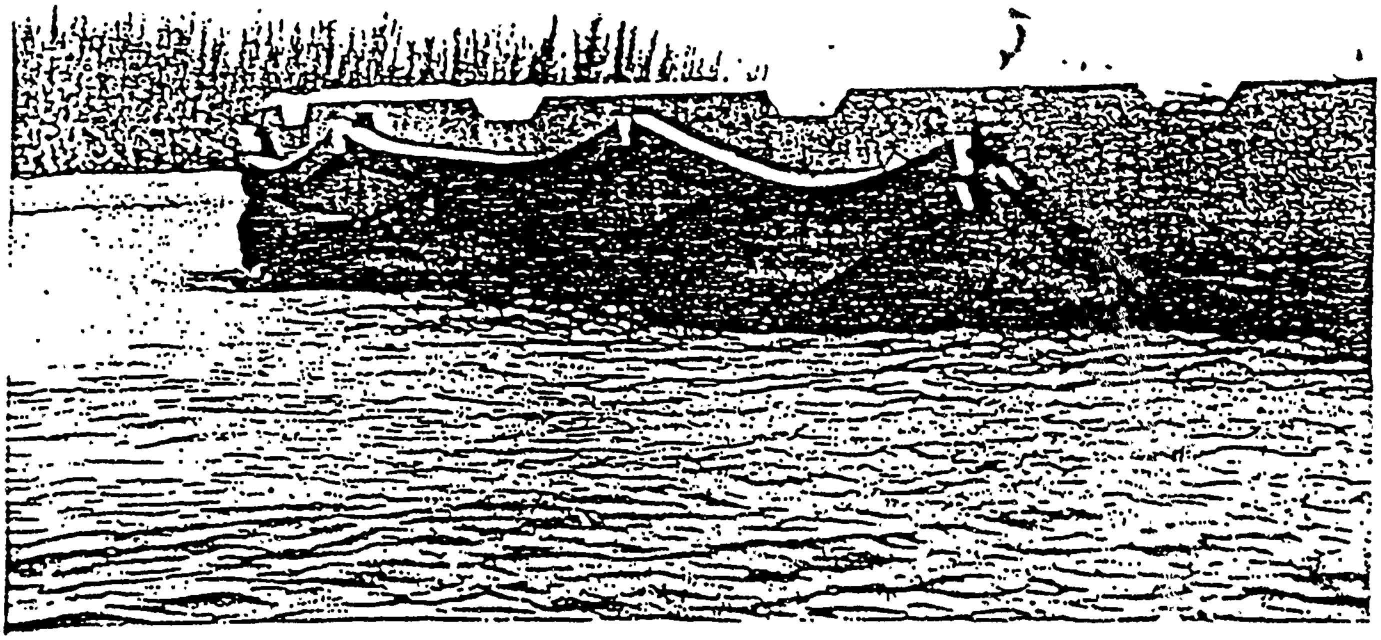
Figure 13: GENERIC PERFORMANCE PLOT



HOLLOW CUBIC RUBBER FENDERS - TRANSVERSELY LOADED

Figure 14: HOLLOW CUBIC COMPRESSION FENDER INSTALLATION





HOLLOW CUBIC RUBBER FENDERS - TRANSVERSELY LOADED

This fendering system generally offers larger energy absorption and larger reaction loads compared to similar sized cylindrical fenders although it does not exhibit the typical buckling phenomenon. Their use is generally between wood walers and concrete piers or draped as indicated in Figure 14.

The parameters determined significant in collapsing the energy and reaction load relationships were determined to be fender volume and fender height in the loaded direction. Fender deflections were normalized by the characteristic fender height.

Figure 15 indicates the resulting generic relationship for energy absorption. This curve can be defined by the following equation.

$$E = HW_bL\,\{21.1X - 74.1X^2 + 208.8X^3\}\,10^3 \qquad (8)$$

The corresponding load/deflection relationship illustrated in Figure 16 is:

$$P = HL\,\{178.7X - 702.8X^2 + 1600.9X^3\}\,10^3 \qquad (9)$$

where:

E = Energy absorption (ft-lb)

P = Reaction load (lb)

H = Height of cubic in direction of loading (ft)

W_b = Base width of cubic (ft)

L = Length of fender (ft)

X = Δ/H (nondimensional), $X \leq 0.65$

Figure 15: GENERIC PERFORMANCE PLOT

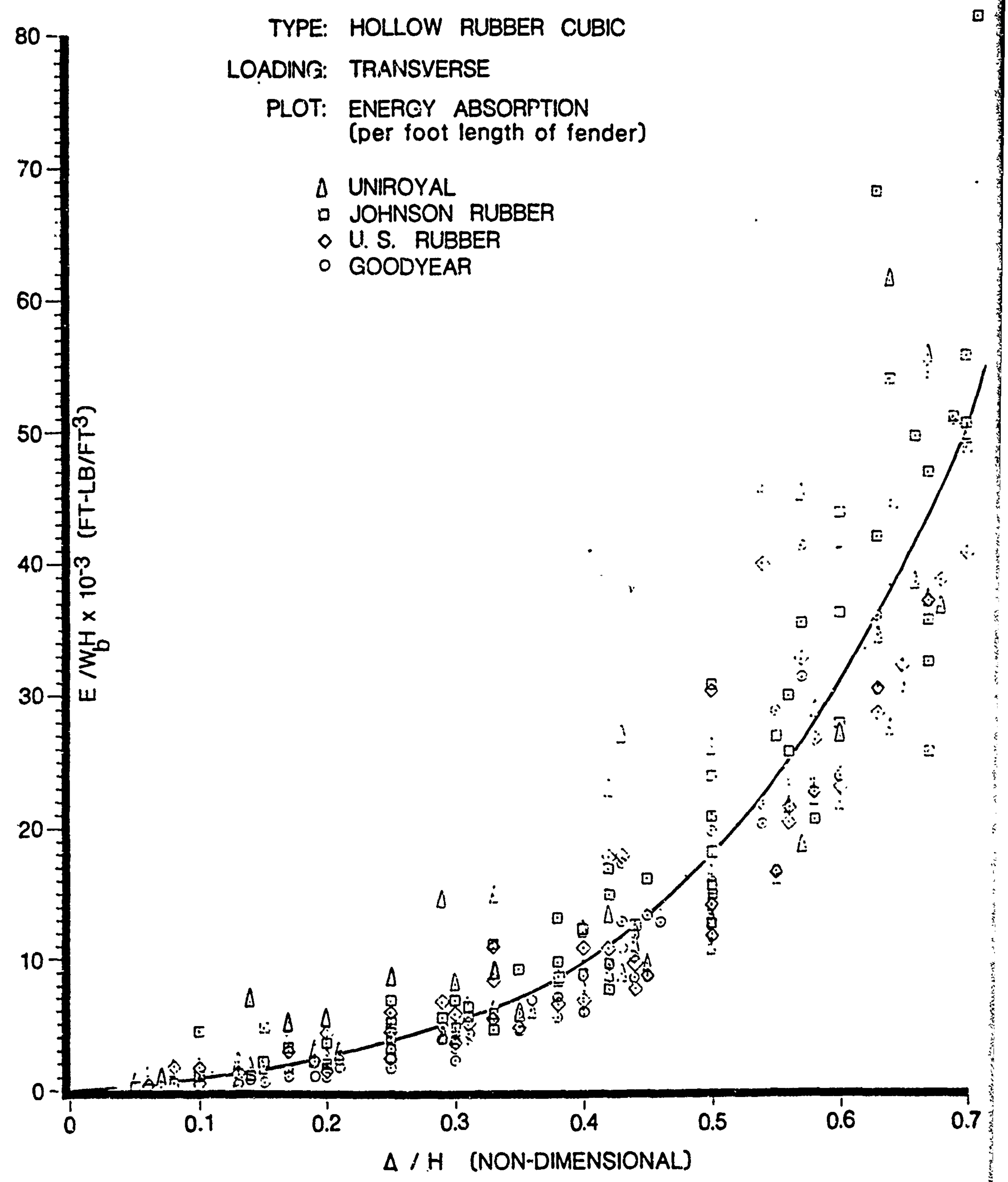
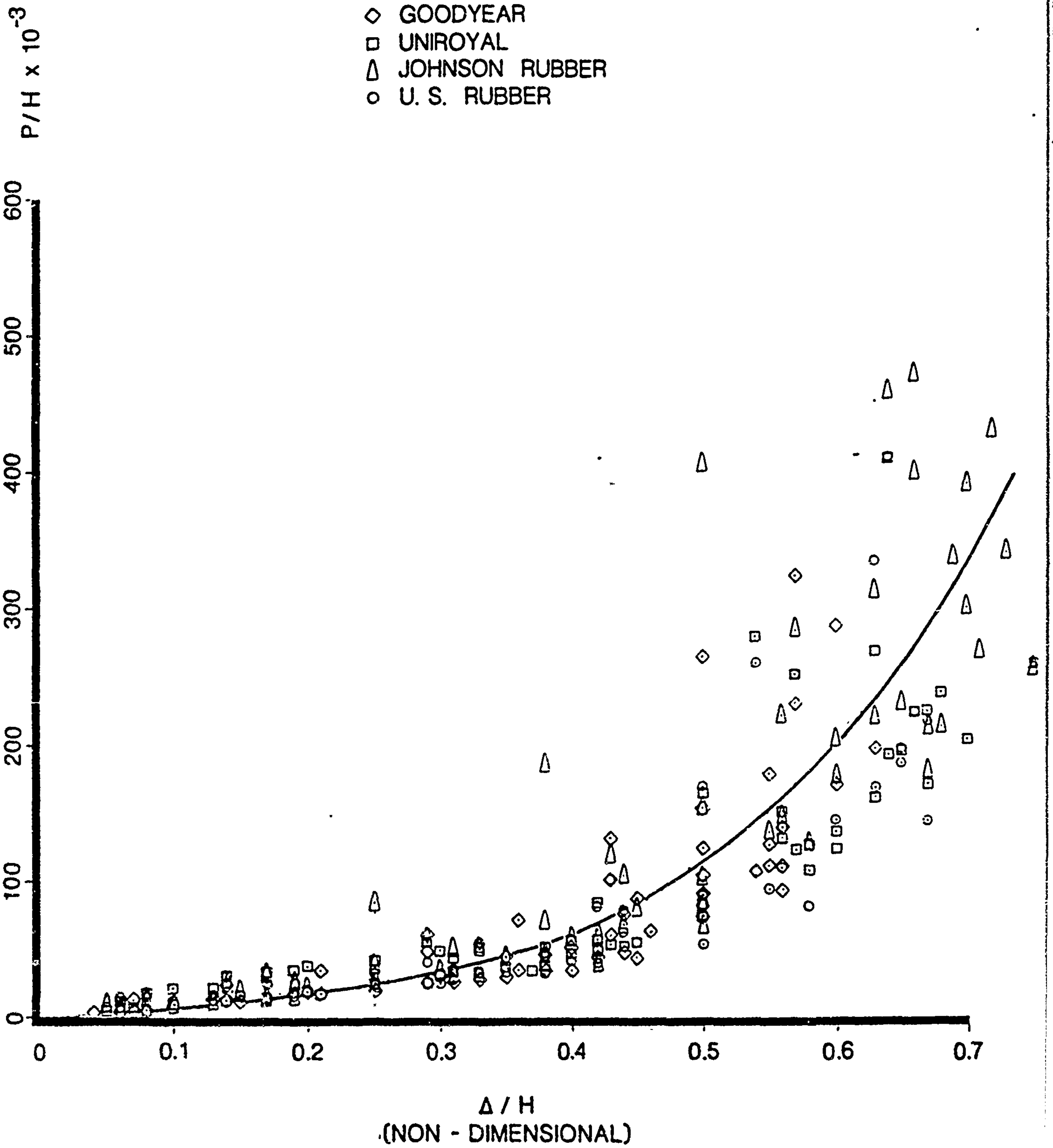


Figure 16: GENERIC PERFORMANCE PLOT

TYPE : HOLLOW RUBBER CUBIC

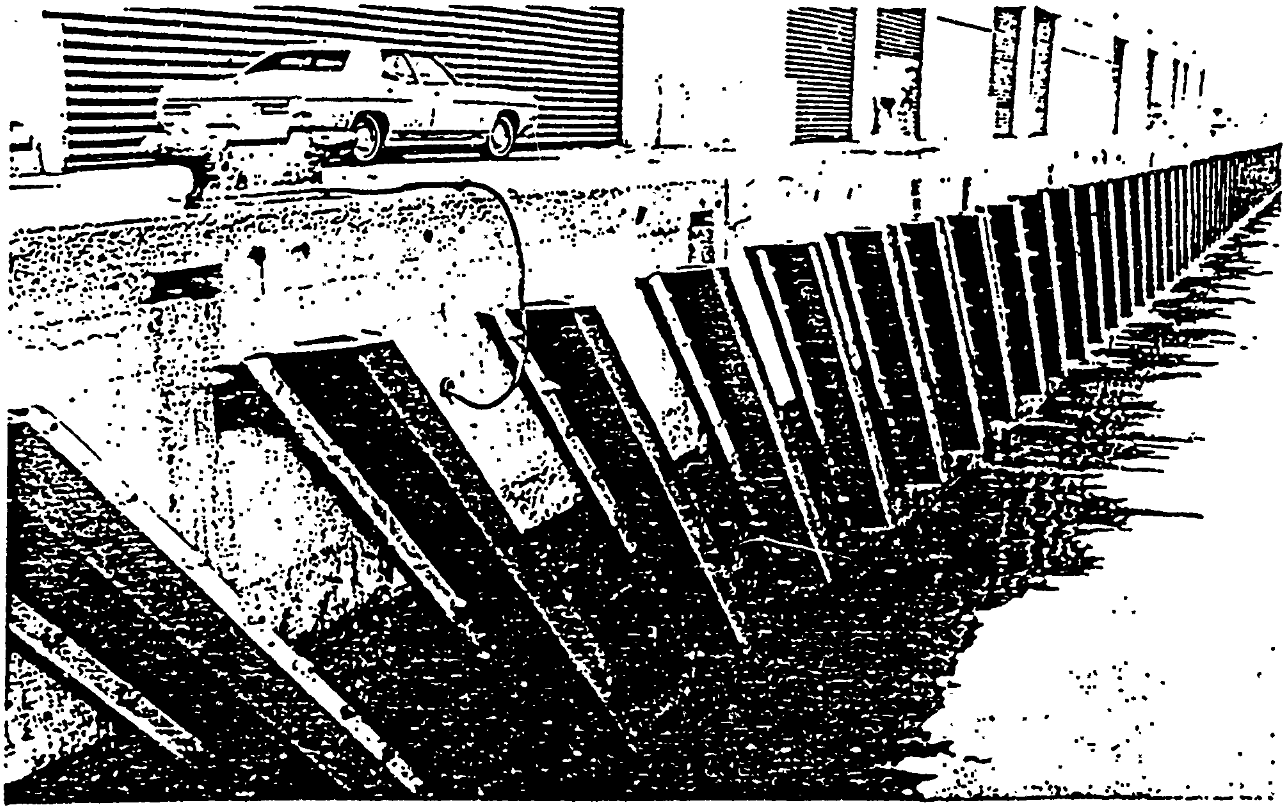
LOADING : TRANSVERSE

PLOT : LOAD / DEFLECTION
(PER FOOT LENGTH OF FENDER)



TRAPEZOIDAL RUBBER FENDERS - TRANSVERSELY LOADED

Figure 17: TRAPEZOIDAL FENDER INSTALLATION



13" high Wing Type Trapezoidal fenders, 10 ft. long mounted diagonally on pier. Northeast Terminal, New York, N.Y.

TRAPEZOIDAL RUBBER FENDERS - TRANSVERSELY LOADED

Trapezoidal rubber fenders employ two mechanisms in the absorption of energy, these are: direct compression and buckling. They are generally mounted directly to open-faced structures,or they can be used in combination with timbering.

Figure 18 indicates the generic load/deflection curve which is characterized by the region of buckling generally occurring at approximately 30 percent deflection.

Figure 19 illustrates the generic energy/deflection curve for this type fender. For this case the characteristic height of the fender in the direction of loading was determined to be the significant condensing parameter for energy absorption and load/deflection.

The following equations are representative of tradezoidal fender performance:

$$E = HLW_b \{0.57X + 36.55X^2 - 56.55X^3 + 40.37X^4\} 10^3 \quad (10)$$

The corresponding load/deflection relationship is:

$$P = HL \{105.82X - 207.06X^2 - 48.24X^3 + 423.72X^4\} 10^3 \quad (11)$$

where:

P = Reaction load (lb)

W_b = Fender base width (ft)

H = Fender height in direction of load (ft)

L = Fender length (ft)

E = Energy absorption (lb-ft)

X = Δ/H (nondimensional, $X \leq 0.53$)

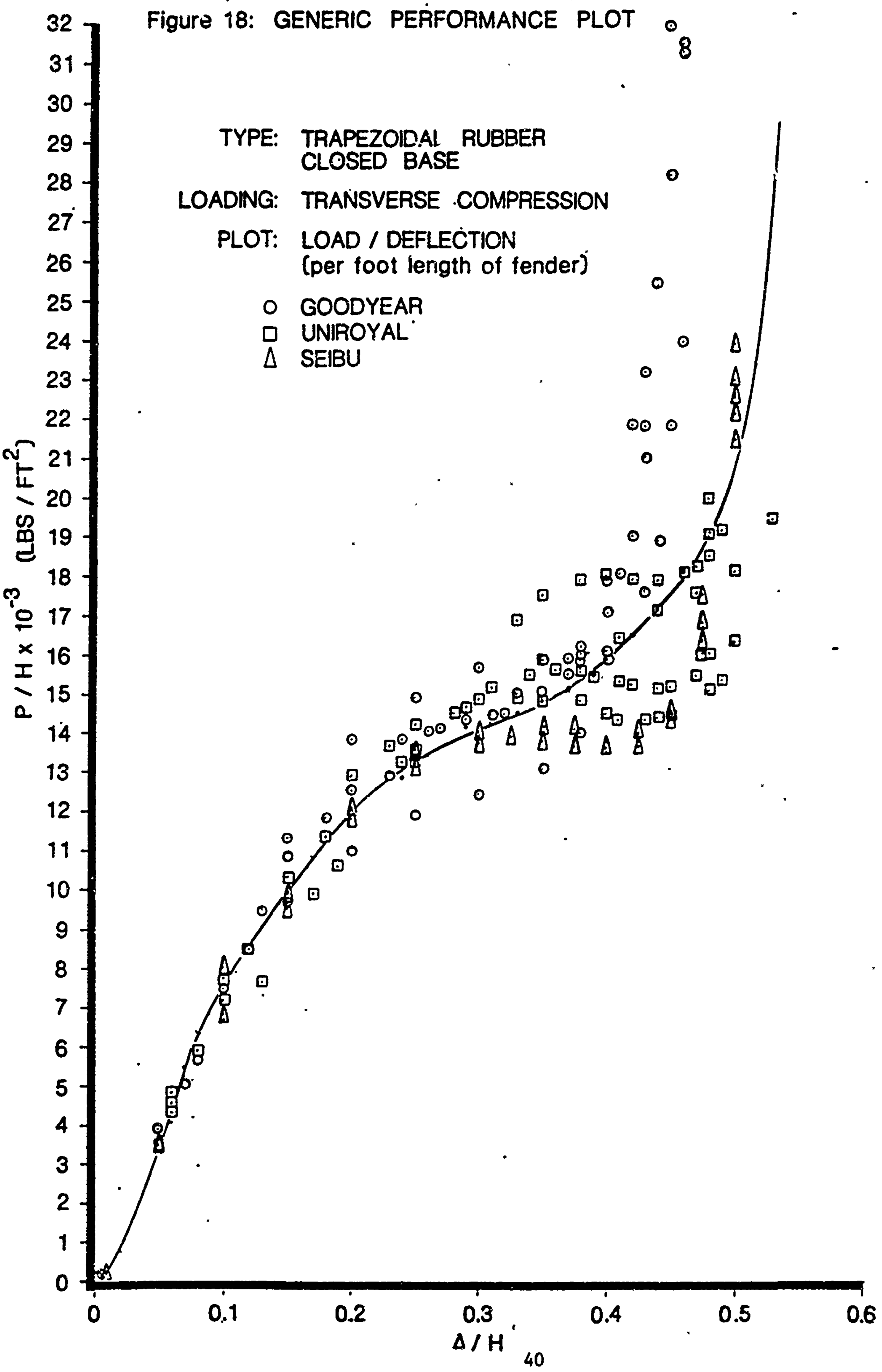
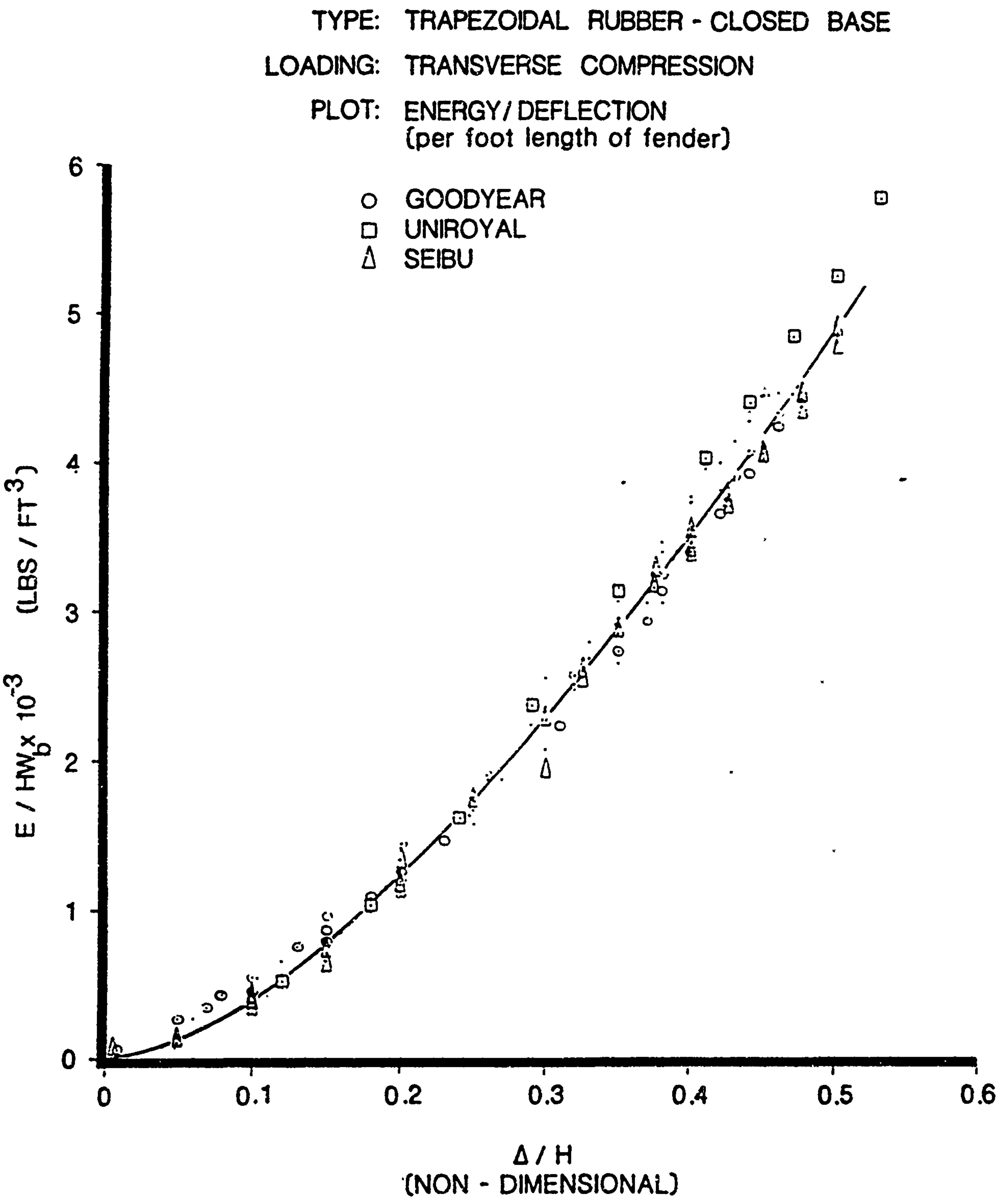


Figure 18: GENERIC PERFORMANCE PLOT
TYPE: TRAPEZOIDAL RUBBER CLOSED BASE
LOADING: TRANSVERSE COMPRESSION
PLOT: LOAD / DEFLECTION (per foot length of fender)
GOODYEAR
UNIROYAL
SEIBU
P / H x 10^-3 (LBS / FT^2)
Δ / H
0
0.1
0.2
0.3
0.4
0.5
0.6
32
31
30
29
28
27
26
25
24
23
22
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0

Figure 19: GENERIC PERFORMANCE PLOT



SOLID CYLINDRICAL SHEAR FENDERS

Figure 20: SOLID CYLINDRICAL SHEAR FENDER INSTALLATION

SOLID CYLINDRICAL SHEAR FENDERS

In shear fender installations, as in Figure 20, wooden fendering is fitted over by means of supports on metal plates. The fenders are then mounted on brackets secured to the quay. When the wood fendering is compressed, the shear fenders are loaded into shear. Since the shear modulus of rubber is only a third of its modulus of elasticity, reaction forces are kept low for this type configuration.

Figure 21 indicates the generic energy absorption relationship which characterizes this type fender. In the figure fender energy absorption curves have been condensed by the volume of the fender. The deflection under load has been nondimensionalized by the fender height normal to the loading.

The energy equation which characterizes this relationship is:

$$E = \beta H \{0.54X + 8.79X^2\} 10^3 \qquad (12)$$

The corresponding load/deflection relationship illustrated in Figure 22 is:

$$P = \beta \{22.77X + 1.14X^2 - 1.43X^3\} 10^3 \qquad (13)$$

where:

P = Reaction load (lb)

$\beta = \dfrac{\pi D_o^{\,2}}{4}$ (ft^2)

$X = \Delta/H$, $X \leq 1.0$

D_o = Diameter of cylinder (ft)

H = Height of fender (ft)

Δ = Fender deflection under load

E = Energy absorption (ft-lb)

Figure 21: GENERIC PERFORMANCE PLOT

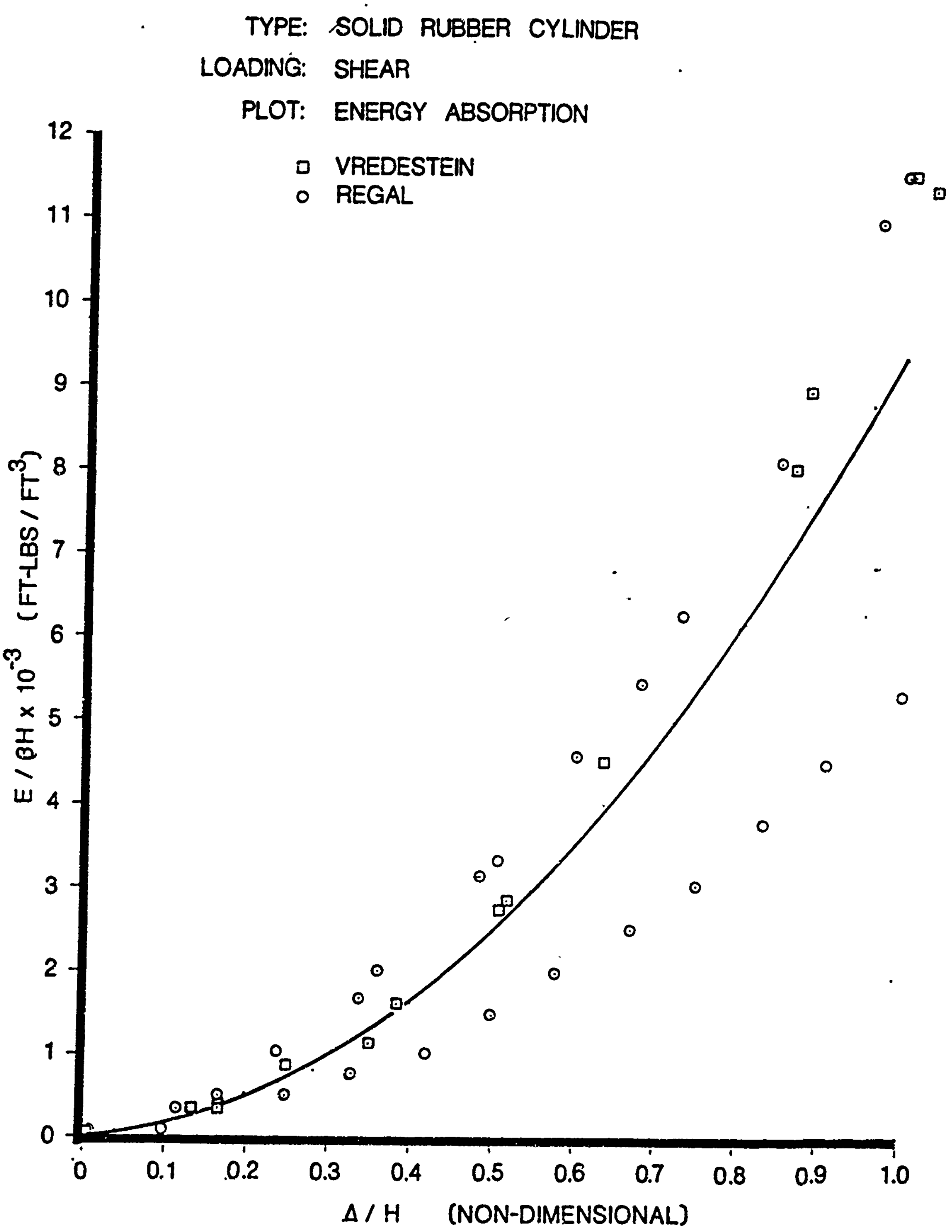
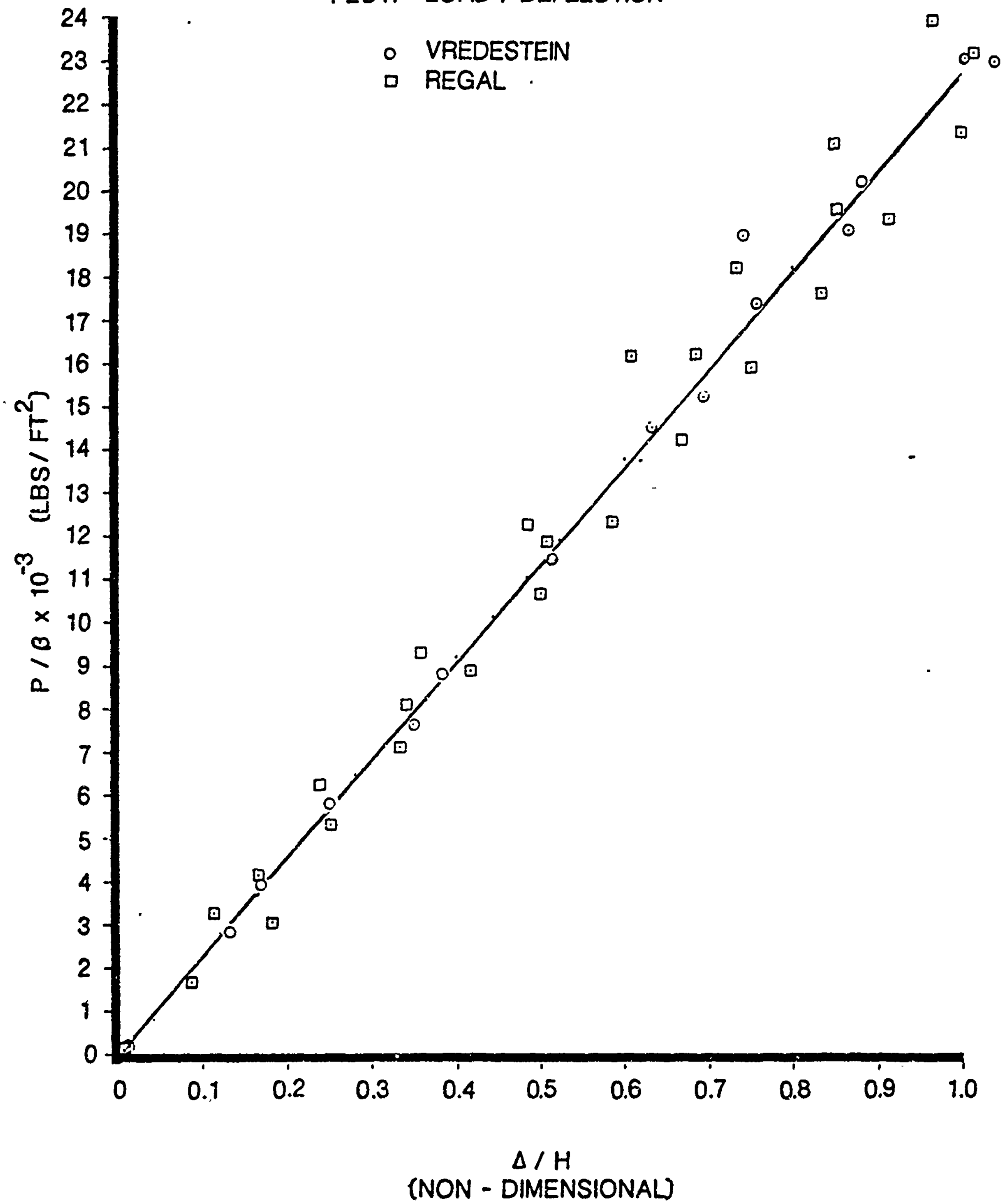


Figure 22: GENERIC PERFORMANCE PLOT

TYPE: SOLID RUBBER CYLINDER

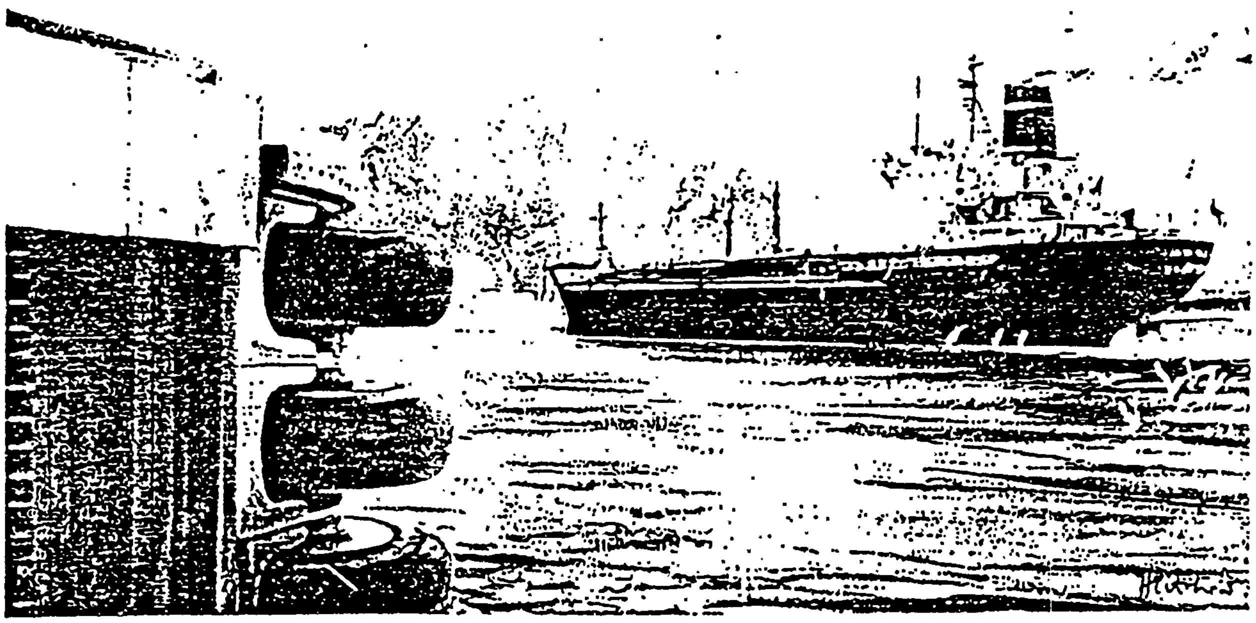
LOADING: SHEAR

PLOT: LOAD / DEFLECTION



ROTARY DONUT FENDERS - TRANSVERSE COMPRESSION

Figure 23: ROTARY DONUT FENDER INSTALLATION



ROTARY DONUT FENDERS - TRANSVERSE COMPRESSION

Rotary fenders consist of hollow section rubber wheels which are mounted on a control axis that allows them to rotate freely when ships horizontal shear forces are applied. This type fender is available in multiple-wheeled configurations in a variety of wheel size diameters. The hollow wheeled section essentially absorbs its energy in material compression and exhibits significant absorption compared to the reaction loads developed.

Figures 24 and 25 illustrate the energy absorption and reaction load deflection curves derived for single-, double- and triple-wheeled fender configurations. In these figures the energy absorption relationships have been modified by characteristic dimensions of fender inner and outer diameters, the number of donuts per axial and the width of the donut base. For the reaction load relationship: the number of donuts, outer diameter and base width were significant. For both relationships deflection was normalized by the characteristic depth of the donut tire.

The generic relationships derived for energy absorption and load/deflection and indicated in Figures 24 and 25 are:

$$E = N\beta \{5.51X - 21.31X^2 + 28.05X^3\} 10^3 \quad (14)$$

The corresponding load deflection curve determined was:

$$P = ND_oW_b \{-0.45X + 67.32X^2 - 189.6X^3 + 188.46X^4\} 10^3 \quad (15)$$

where:

N = Number of donuts per axial

$$\beta = D_oW_b\left(\frac{D_o - D_i}{2}\right)$$

D_o = Outer donut diameter (ft)

W_b = Base width of donut (ft)

D_i = Inner donut diameter (ft)

E = Energy absorption (lb-ft)

P = Reaction load (lb)

$X = \dfrac{\Delta}{\frac{D_o - D_i}{2}} \quad X \leq 0.68$

Δ = Fender deflection (ft)

Figure 24: GENERIC PERFORMANCE PLOT

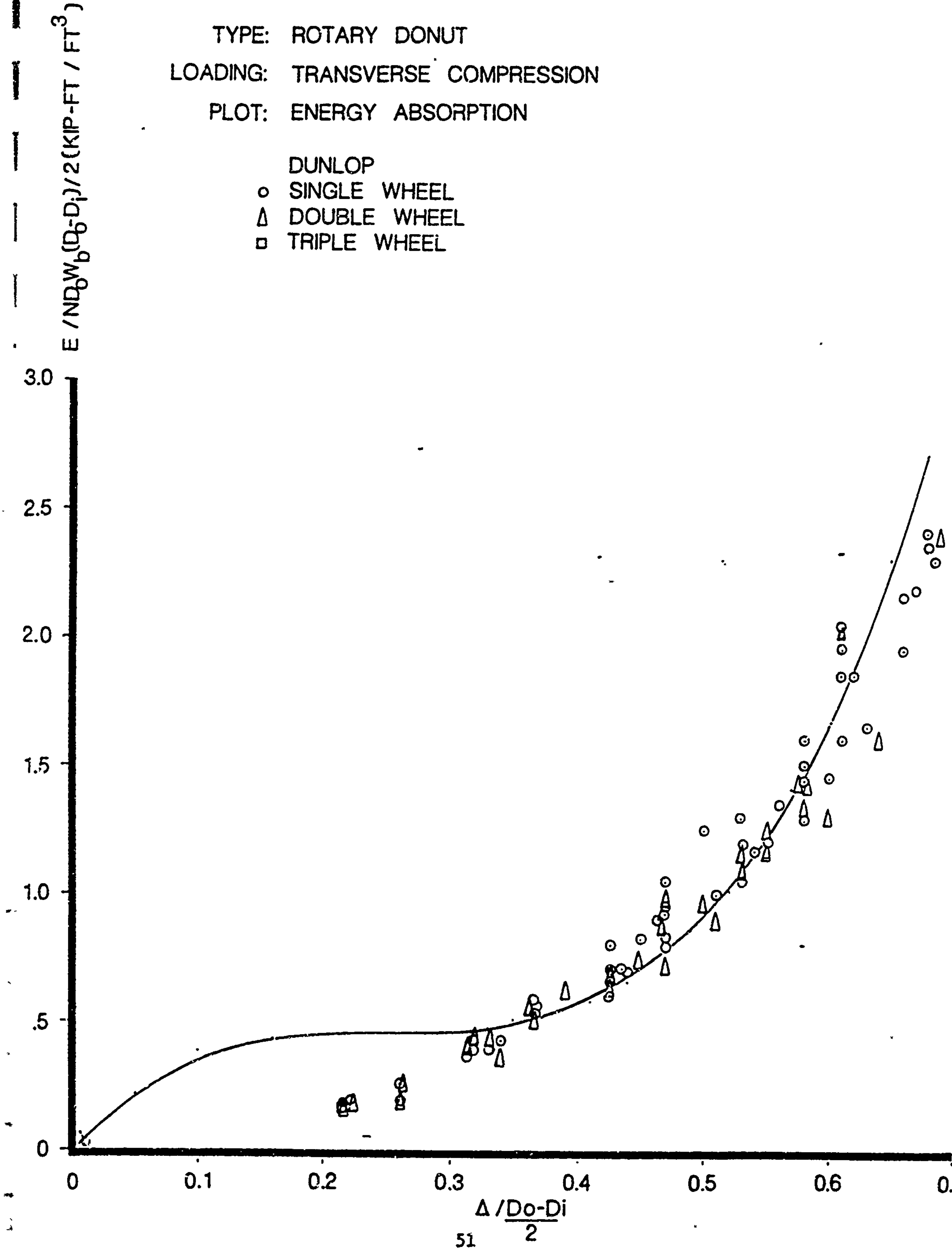
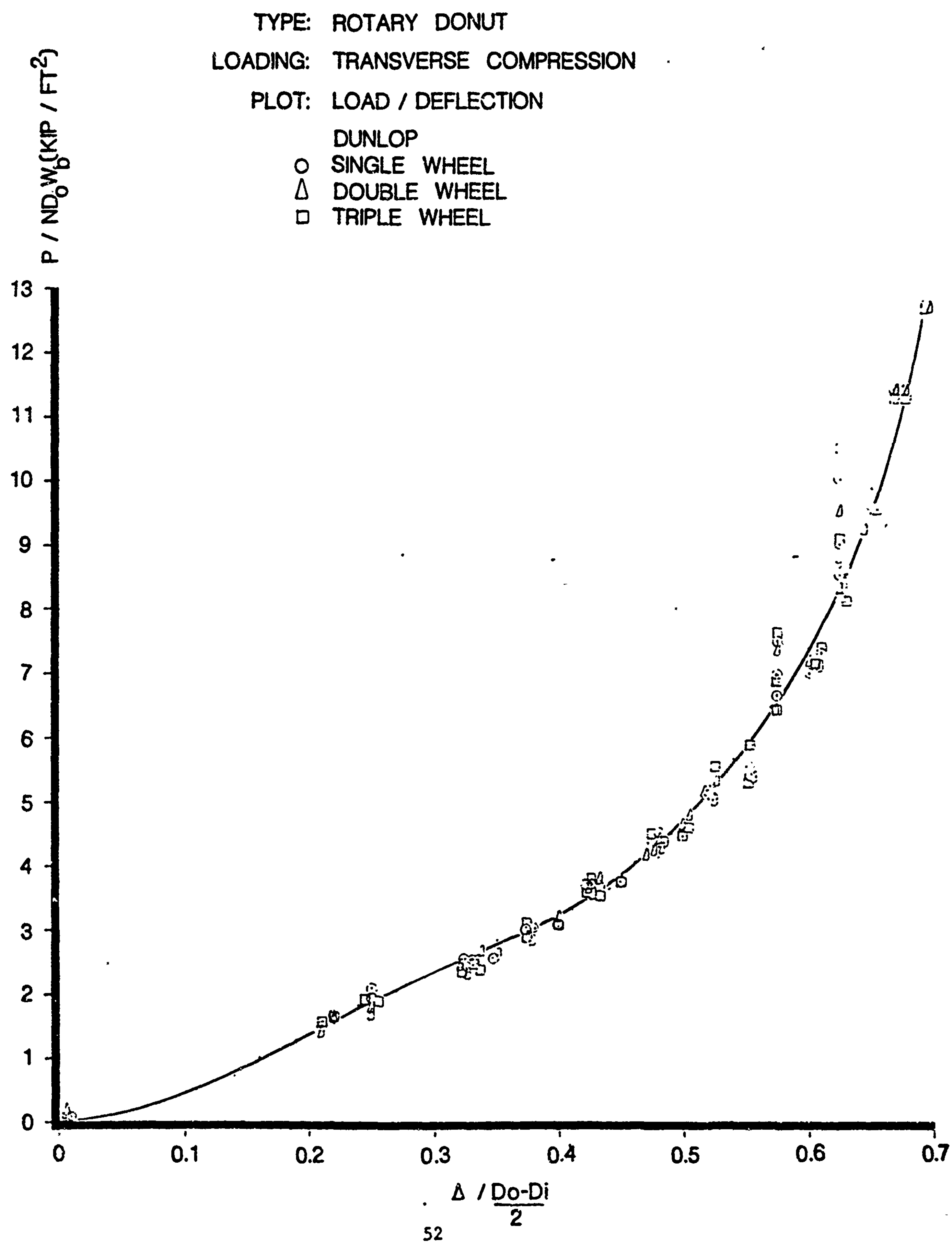
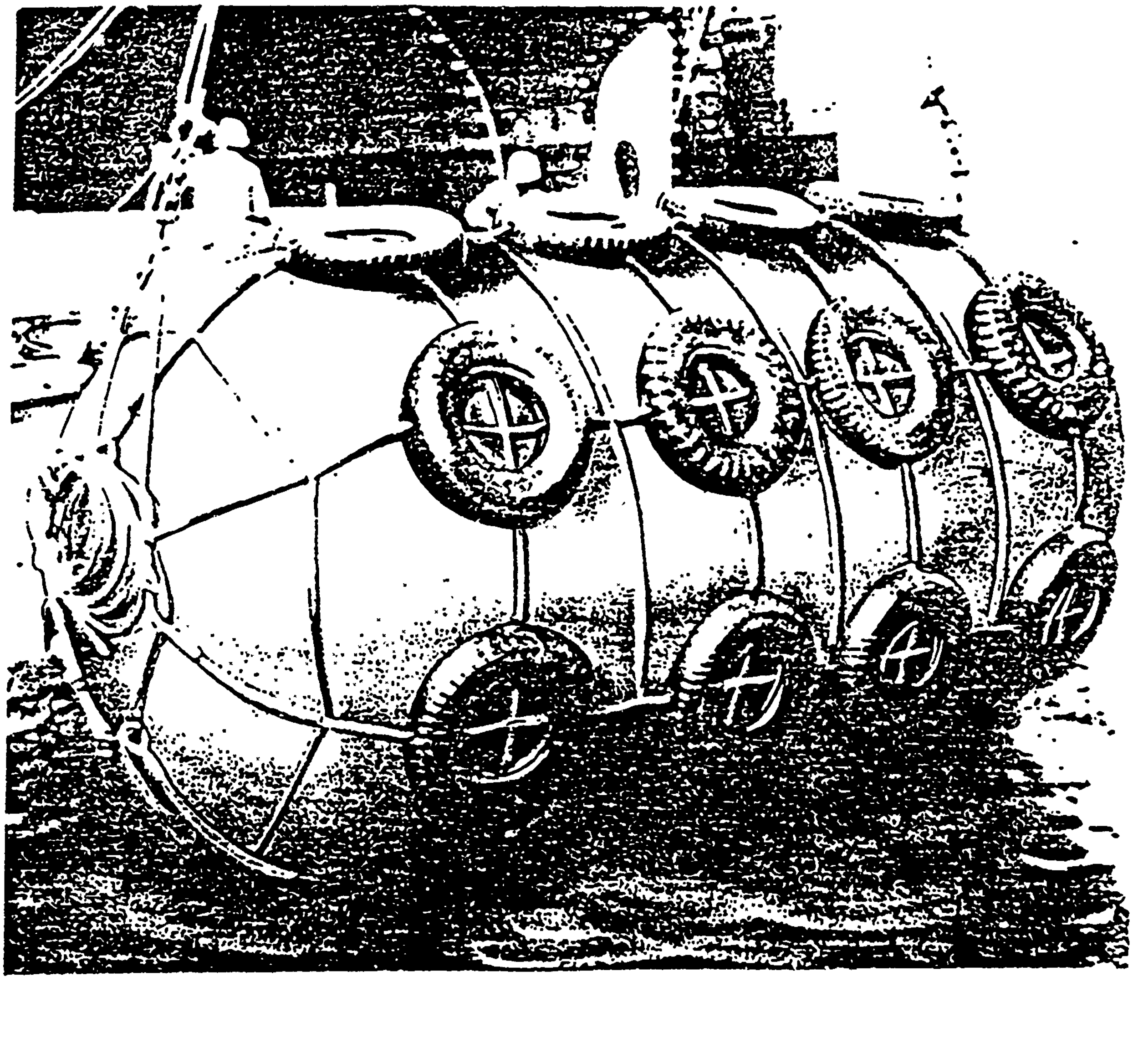


Figure 25: GENERIC PERFORMANCE PLOT



PNEUMATIC RUBBER FENDERS - FLOATING TYPE

Figure 26: PNEUMATIC FLOATING FENDER



PNEUMATIC RUBBER FENDERS - FLOATING TYPE

Floating pneumatic fenders utilize the compressive elasticity of air to support loads. For this reason performance deterioration due to fatigue is absent. For realistic oblique ship loading, pneumatic type fenders do not suffer significant loss of energy absorbing capacity as do solid rubber fenders. For rough weather mooring, this type fendering system exhibits much less damage due to the fact that maximum reaction forces under combined shear and compression increase slowly and sustain large allowable deflections. Under excessive loads these fenders do not result in excessive reaction loads as do solid or bottomed out rubber fenders.

Figures 27 and 28 indicate the results for 32 different size pneumatic fenders investigated. These fenders ranged in pressures from 4.3 to 11.4 psi internally. Figure 27 illustrates the characteristic energy absorbing relationship resulting from condensing the plot of energy absorption by the relationship $p^{1/1.4}\,LD_o^{\,2}$, the pressure, length and diameter characteristic of the fender. This quantity is plotted against the deflection normalized by the diameter of the cylinder bag.

The following relationship was determined representative of energy absorption for pneumatic fenders.

$$E = \beta D_o \{0.82X - 2.54X^2 + 17.94X^3\}\, 10^3 \qquad (16)$$

The corresponding load/deflection relationship illustrated in Figure 28 is:

$$P = \beta \{5.19X + 39.95X^2 - 77.02X^3 + 149.09X^4\}\, 10^3 \qquad (17)$$

where:

$$\beta = p^{1/1.4}\, LD_o$$

p = Internal pressure (psf)

L = Length of fender (ft)

D_o = Fender diameter (ft)

X = Δ/D_o, $X \leq 0.55$

Δ = Fender deflection (ft)

E = Energy absorption (lb-ft)

P = Reaction load (lb)

Figure 27: GENERIC PERFORMANCE PLOT

TYPE: RUBBER PNEUMATIC - FLOATING BAG
LOADING: TRANSVERSE COMPRESSION
PLOT: ENERGY ABSORPTION

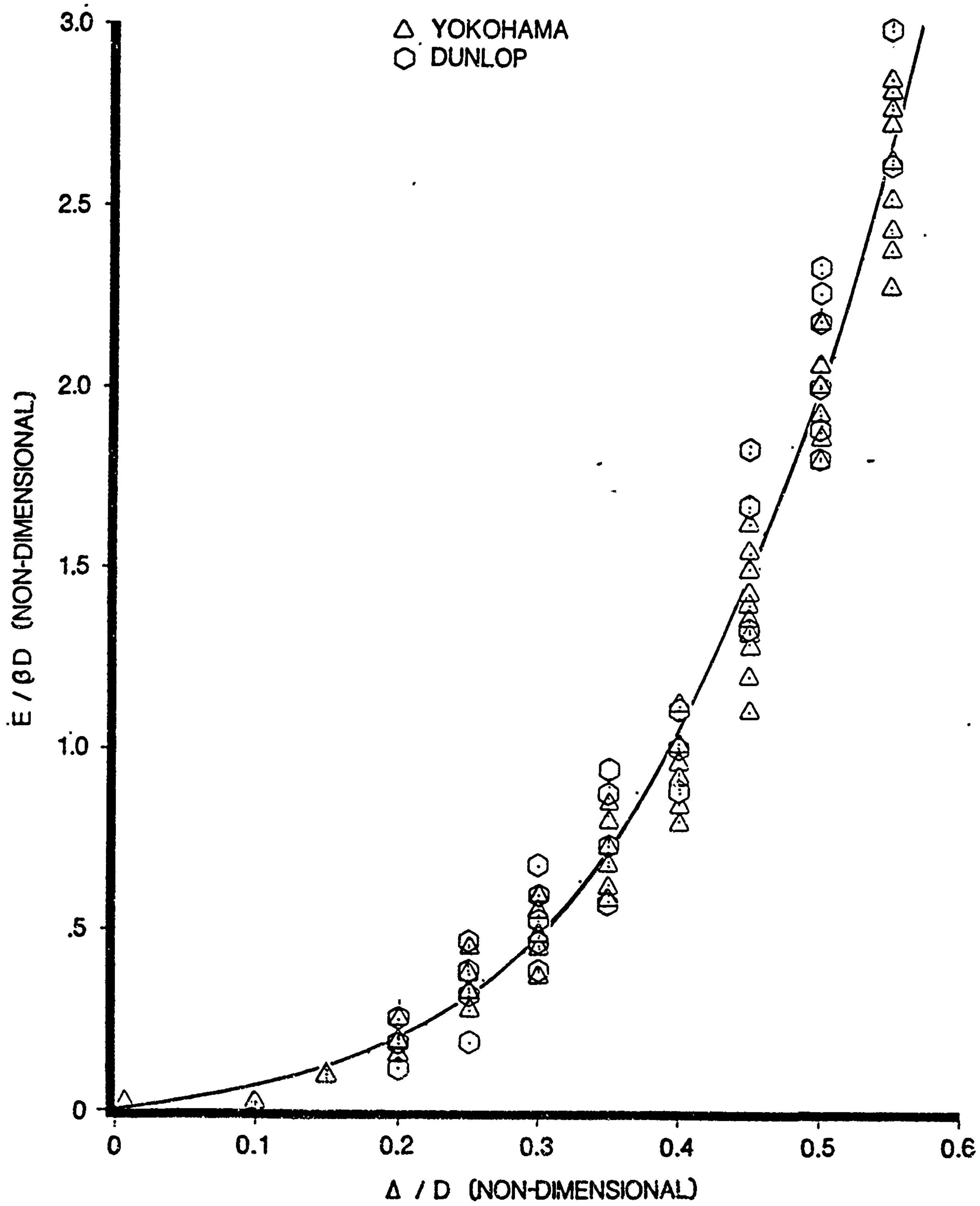
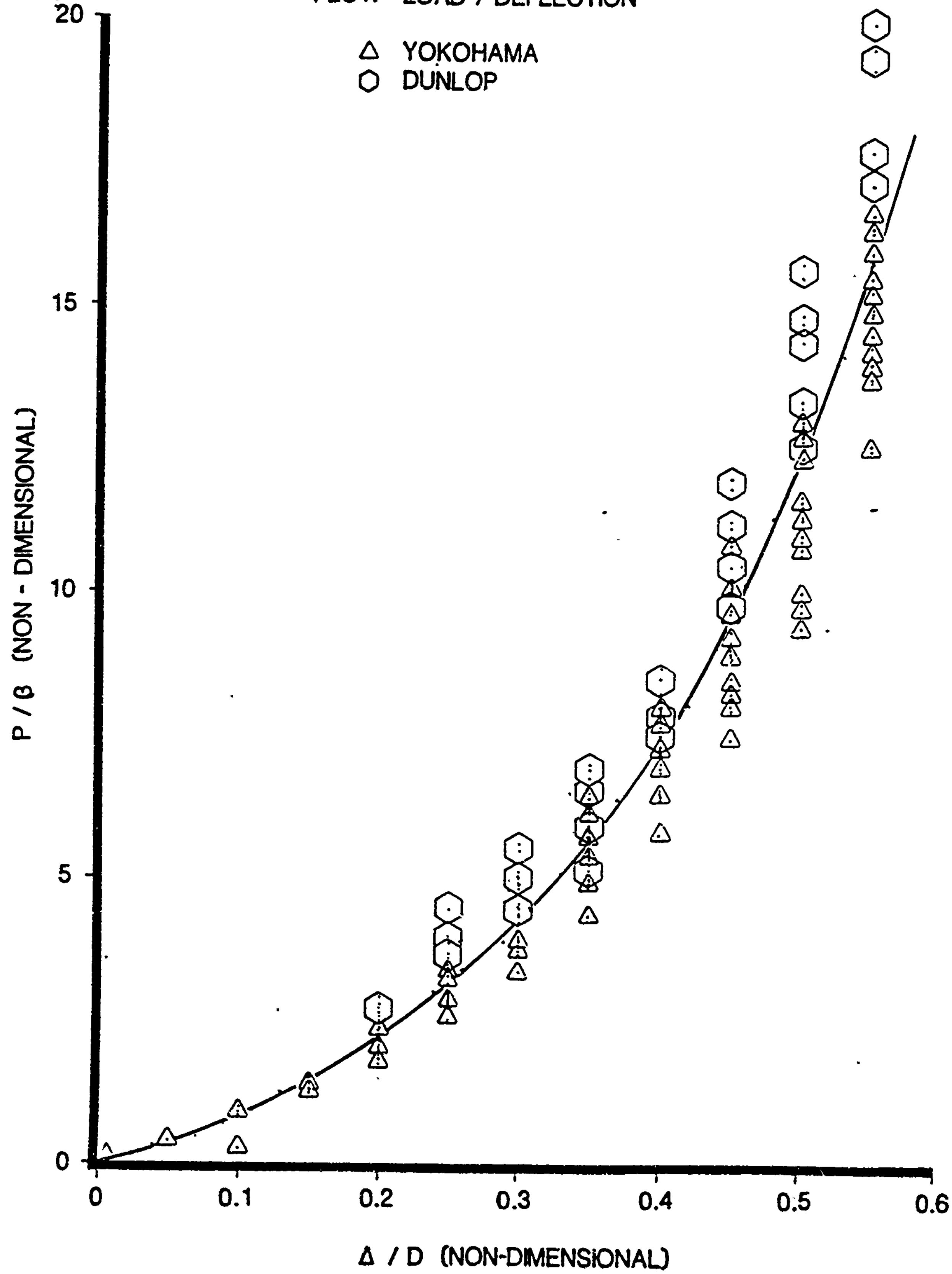


Figure 28: GENERIC PERFORMANCE PLOT

TYPE: RUBBER PNEUMATIC - FLOATING TYPE

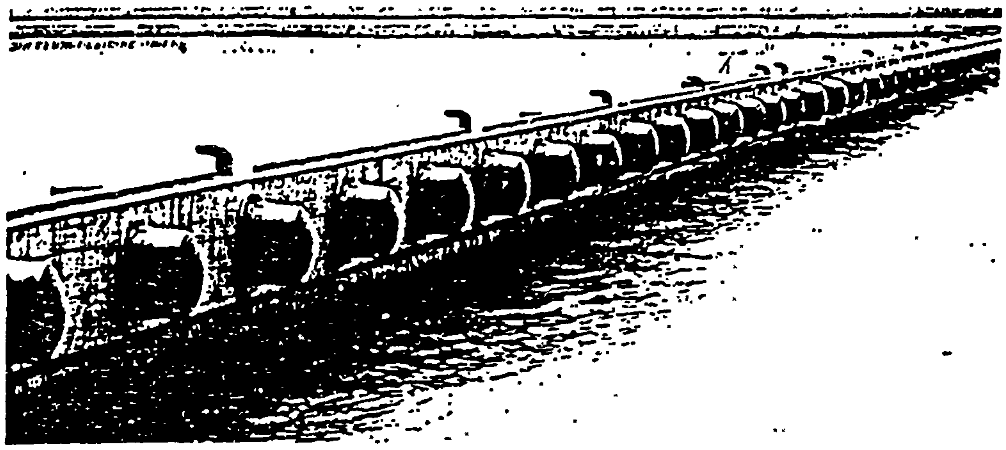
LOADING: TRANSVERSE COMPRESSION

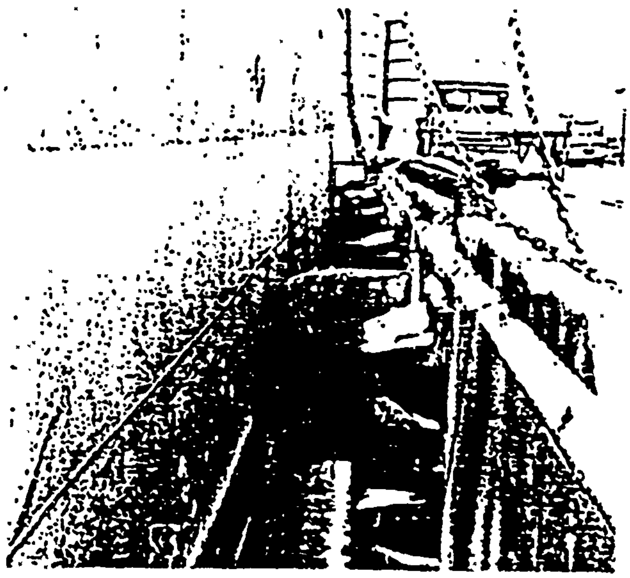
PLOT: LOAD / DEFLECTION



RUBBER PNEUMATIC - AIR BLOCK FENDER

Figure 29: PNEUMATIC AIR BLOCK FENDER INSTALLATION





RUBBER PNEUMATIC - AIR BLOCK FENDER

Air block fenders are pneumatic, axially loaded fenders which can be bolted to docks and applicable when floating bag types cannot be used. They offer all the performance advantages that pneumatic bag types generally exhibit.

The characteristic performance curves illustrated in Figures 30 and 31 were determined by nondimensionalizing the energy absorption and load/deflection curves by P, H, and D, the characteristic pressure, height and diameter of the block fender. The energy absorption and load/defelction relationships were then plotted against nondimensional deflection Δ/H, the percentage fender height. The resulting relationships are based on the investigations of 13 fender sizes at 14.2 psi. Since this type fender was available in only one pressure size, the pressure variable was considered similar to the relationship determined for floating bag types.

The following relationship is representative of the energy absorption of air block fenders illustrated in Figure 30:

$$E = \beta D_o \{2.58X + 9.73X^2 - 13.40X^3 + 40.09X^4\} 10^3 \quad (18)$$

The corresponding load/deflection relationship illustrated in Figure 31 is represented by:

$$P = \beta \{43.96X - 8.77X^2 - 62.48X^3 + 256.23X^4\} 10^3 \quad (19)$$

where:

E = Energy absorption

P = Reaction load

$\beta = p^{1/1.4} HD_o$

H = Fender height

D_o = Fender diameter

p = Internal pressure

X = Δ/H (nondimensional) $X \leq 0.6$

Δ = Fender deflection

The above relationships are valid for any set of consistent units.

Figure 30: GENERIC PERFORMANCE PLOT

TYPE: RUBBER PNEUMATIC - AIR BLOCK FENDERS
LOADING: AXIAL COMPRESSION
PLOT: ENERGY ABSORPTION

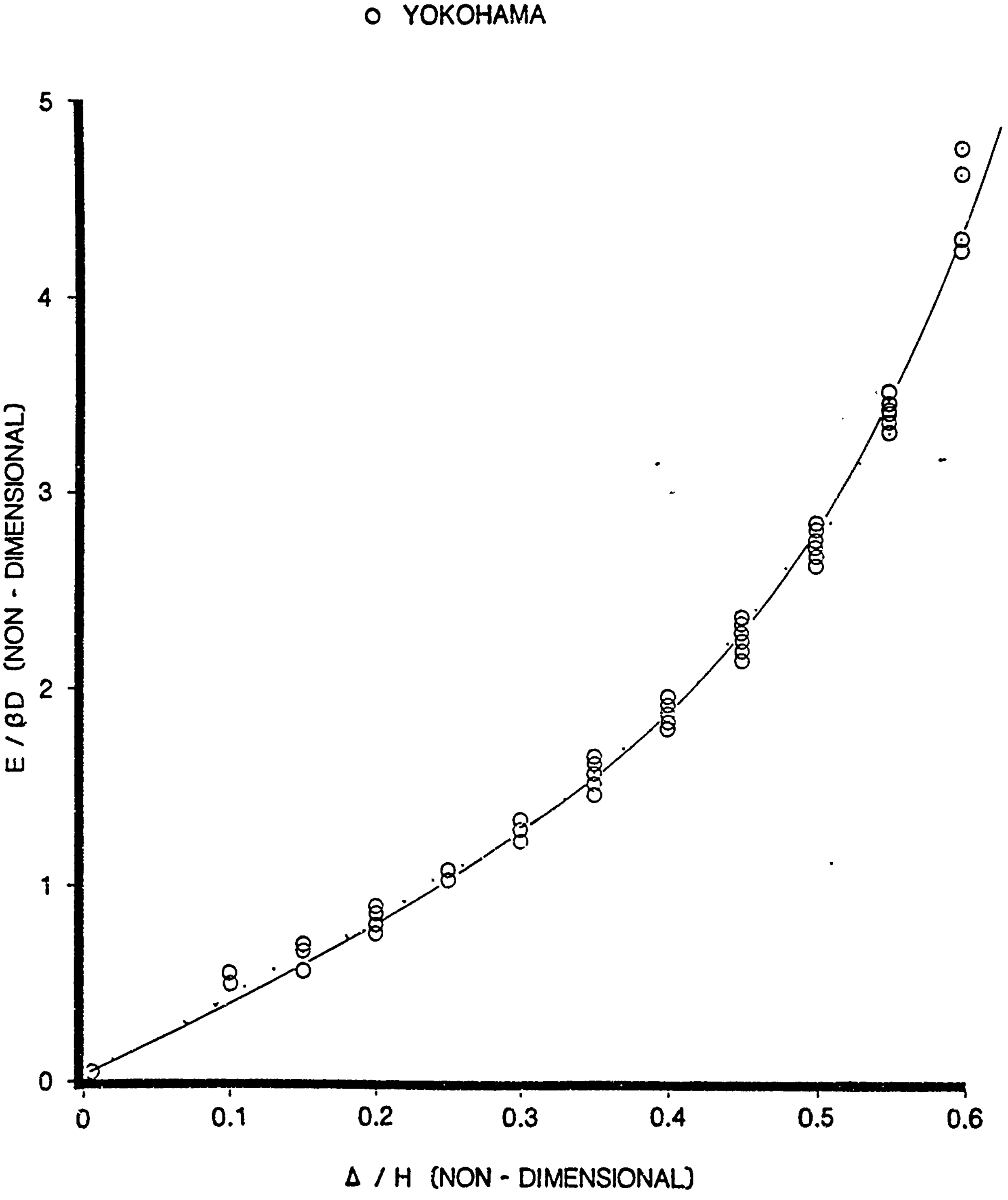
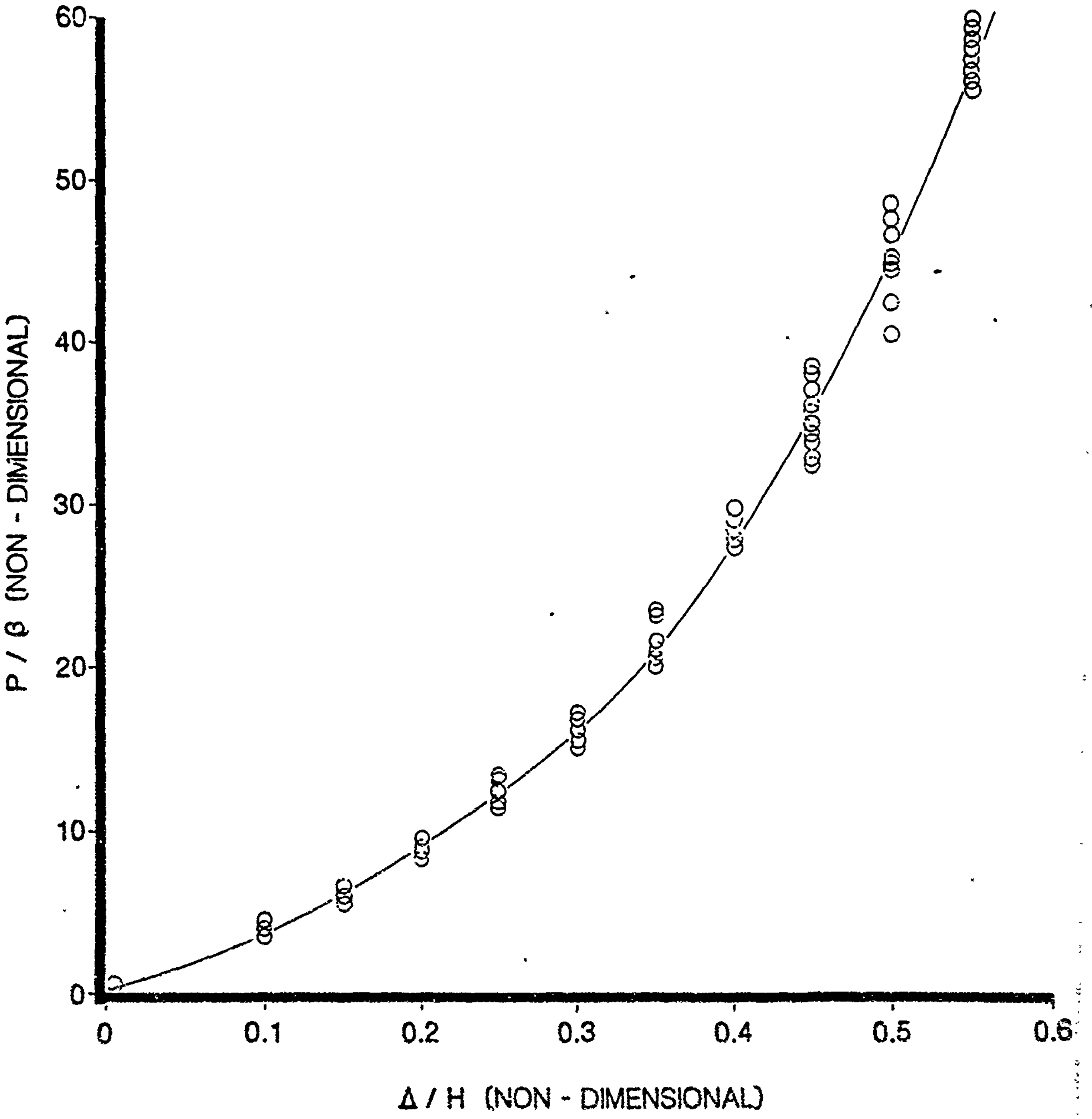


Figure 31: GENERIC PERFORMANCE PLOT

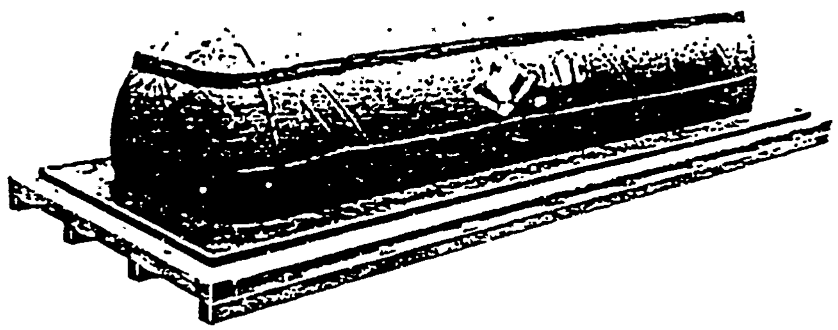
TYPE: RUBBER PNEUMATIC - AIR BLOCK FENDERS
LOADING: AXIAL COMPRESSION
PLOT: LOAD / DEFLECTION

o YOKOHAMA



RUBBER PNEUMATIC - AIR BLOCK CUSHIONS

Figure 32: AIR BLOCK CUSHION



RUBBER PNEUMATIC - AIR BLOCK CUSHIONS

This system is similar to the air block fenders except that it is rectangular in shape. It is mounted on a steel backing plate which can be bolted to docks or semisubmersible drill rig legs. Although this fendering system appeared to have a significant number of merits typically associated with pneumatic systems, they were only available from one fendering manufacturer.

The data represented in Figures 33 and 34 are for only two fender lengths at the same internal pressure. In these figures the energy and load curves have been normalized by the characteristic pressure, length and base width dimensions, while the deflection has been normalized by the cushion height in the direction of loading.

The derived relationship which best fits the condensed data for energy was determined to be:

$$E = \beta H \{-0.12X + 7.46X^2 - 12.71X^3 + 14.77X^4\} 10^3 \tag{20}$$

The corresponding load deflection relationship is:

$$P = \beta \{9.22X - 4.16X^2 + 5.10X^3 + 29.55X^4\} 10^3 \tag{21}$$

where:

E = Energy absorption

P = Reaction load

β = $p^{1/1.4} W_b L$

p = Internal pressure

W_b = Base width of cushion

L = Cushion length

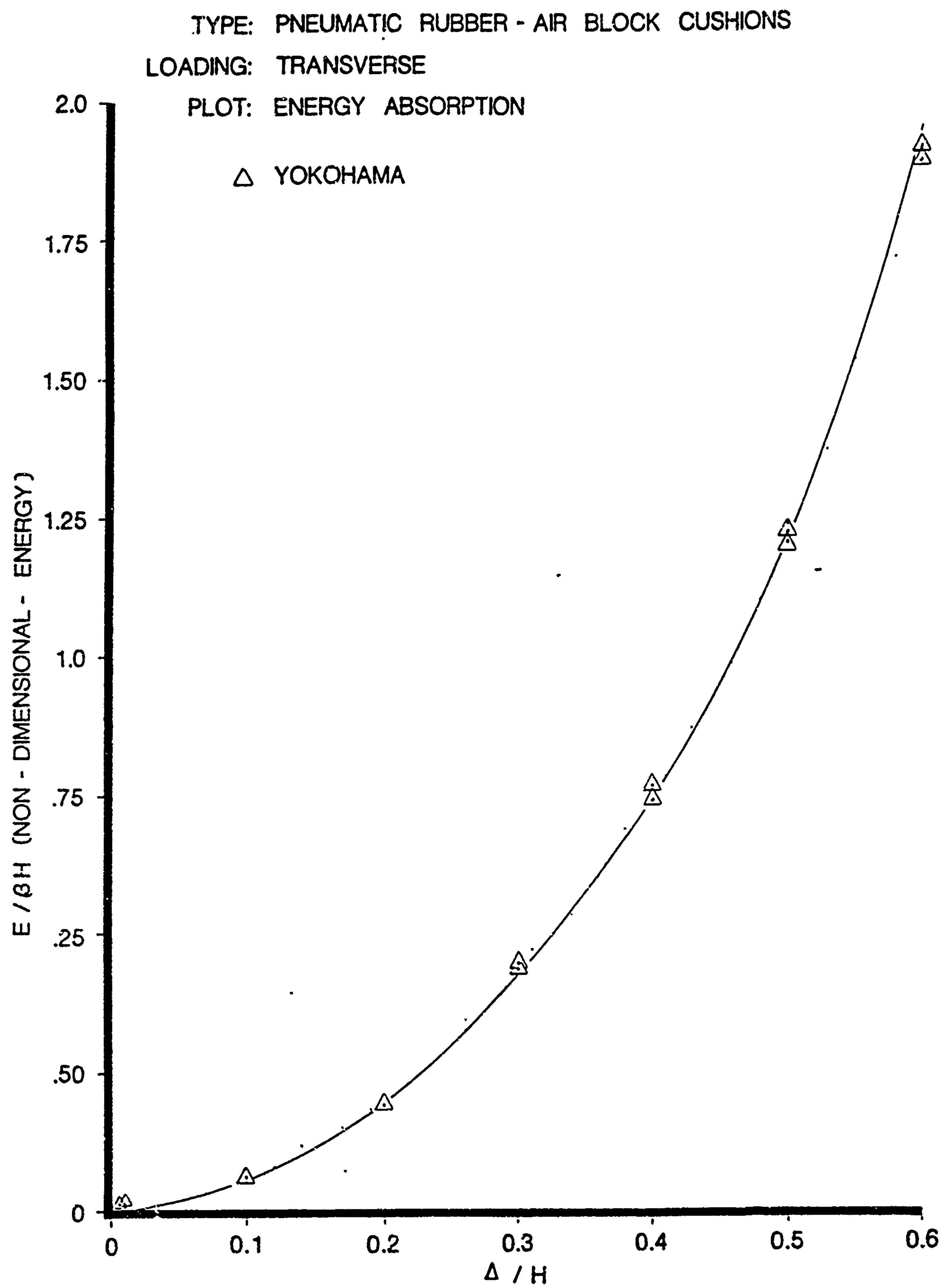
H = Cushion height in direction of load

X = Δ/H, $X \leq 0.6$

Δ = Fender deflection

The above relationships are valid for any set of consistent units.

Figure 33: GENERIC PERFORMANCE PLOT



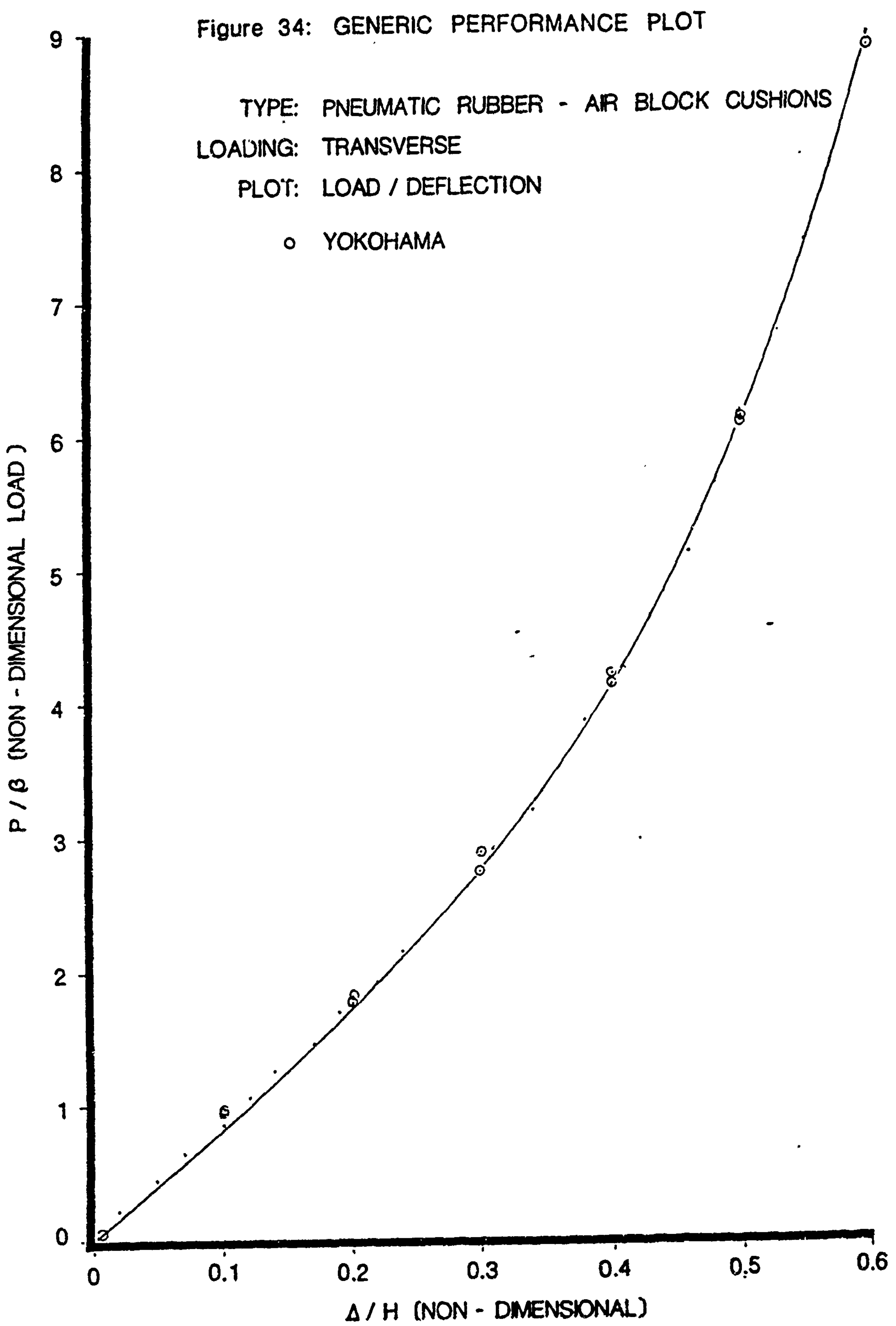
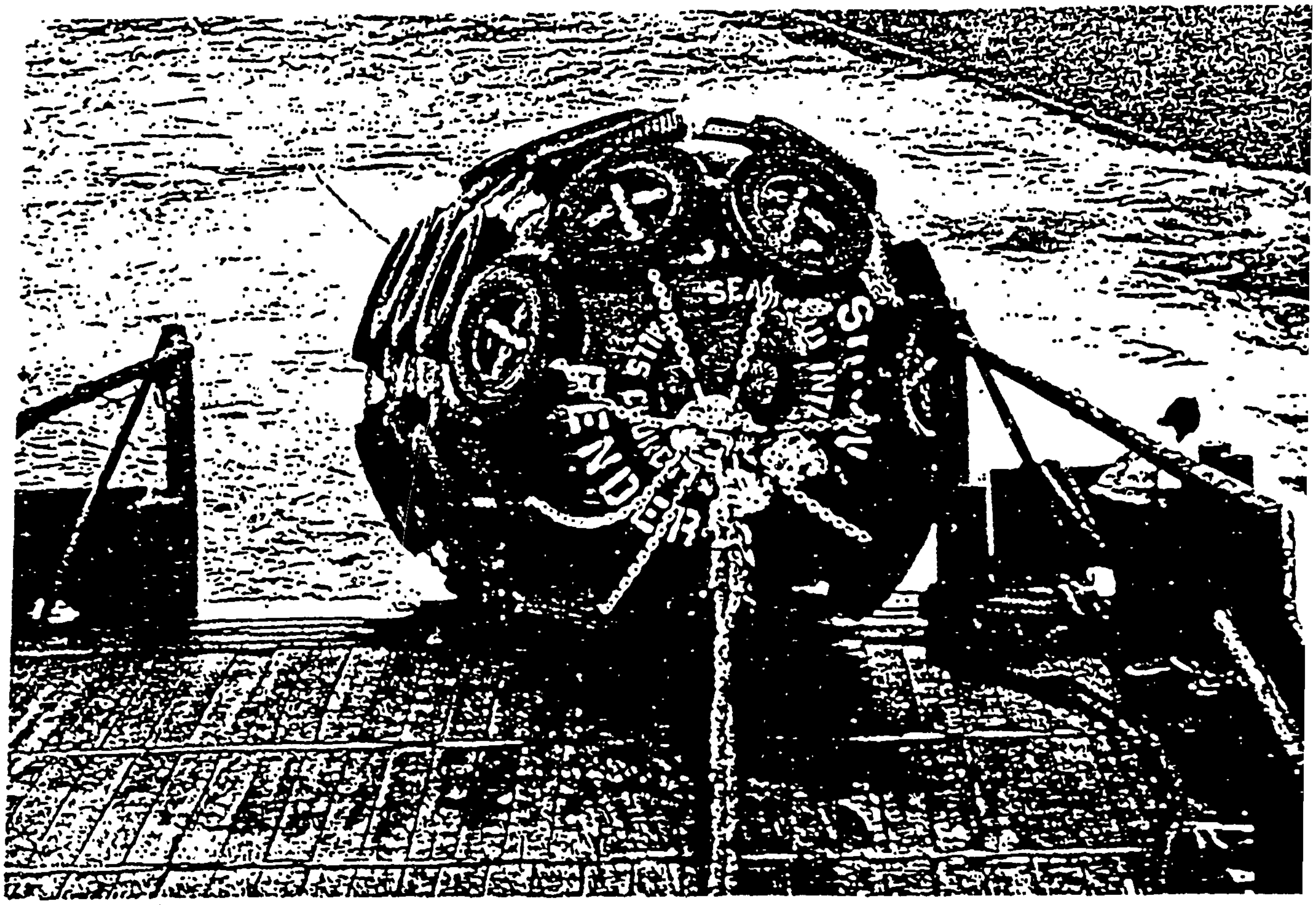


Figure 34: GENERIC PERFORMANCE PLOT
TYPE: PNEUMATIC RUBBER - AIR BLOCK CUSHIONS
LOADING: TRANSVERSE
PLOT: LOAD / DEFLECTION
o YOKOHAMA
P / β (NON - DIMENSIONAL LOAD)
Δ / H (NON - DIMENSIONAL)
0
1
2
3
4
5
6
7
8
9
0.1
0.2
0.3
0.4
0.5
0.6

FOAM-FILLED RUBBER FENDERS

Figure 35: FOAM-FILLED FENDER



FOAM-FILLED RUBBER FENDERS

This system resembles the pneumatic bag type fenders except internally they are completely filled with resilient, closed-cell foam. They typically have greater energy absorption and less reaction force than pneumatic fenders of equal size. They are generally lighter than pneumatic fenders of equal capacity and cannot explode or sink if punctured.

Figures 36 and 37 indicate the results of investigating two primary sources of performance data for this type fender system. For these curves the parameters found to condense the performance relationship were fender length and diameter.

The energy absorption relationship found characteristic of Figure 36 was determined to be:

$$E = \beta D_o \{0.27X - 1.03X^2 + 6.43X^3 - 4.69X^4\}\ 10^3 \qquad (22)$$

The corresponding load deflection relationship illustrated in Figure 37 is:

$$P = \beta \{1.77X + 6.25X^2 - 13.81X^3 + 16.32X^4\}\ 10^3 \qquad (23)$$

where:

E = Energy absorption (ft-lb)

P = Reaction load (lb)

β = $D_o L$

D_o = Fender diameter (ft)

L = Fender length (ft)

X = Δ/D_o (nondimensional)

Δ = Fender deflection (ft)

Figure 36: GENERIC PERFORMANCE PLOT

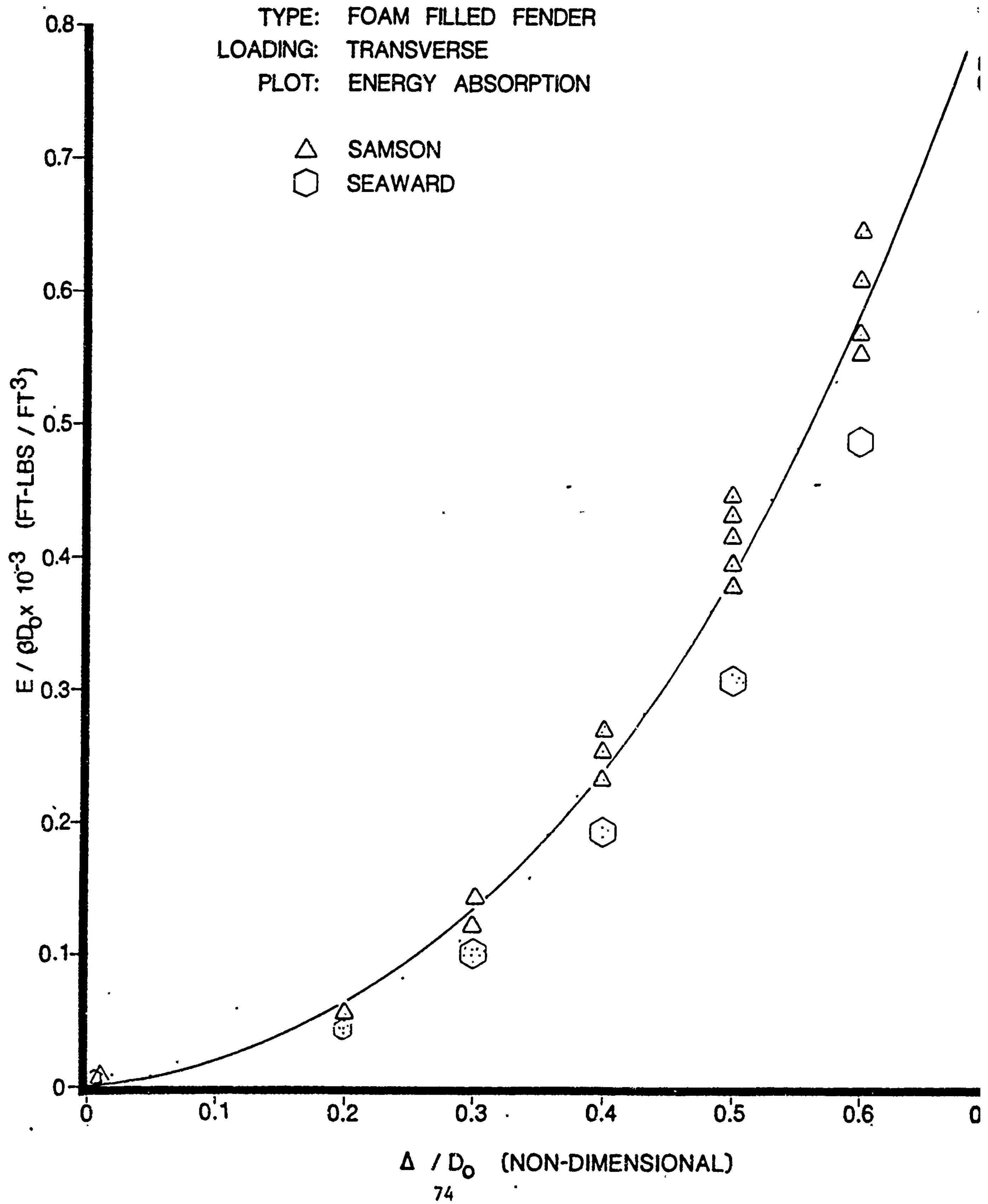
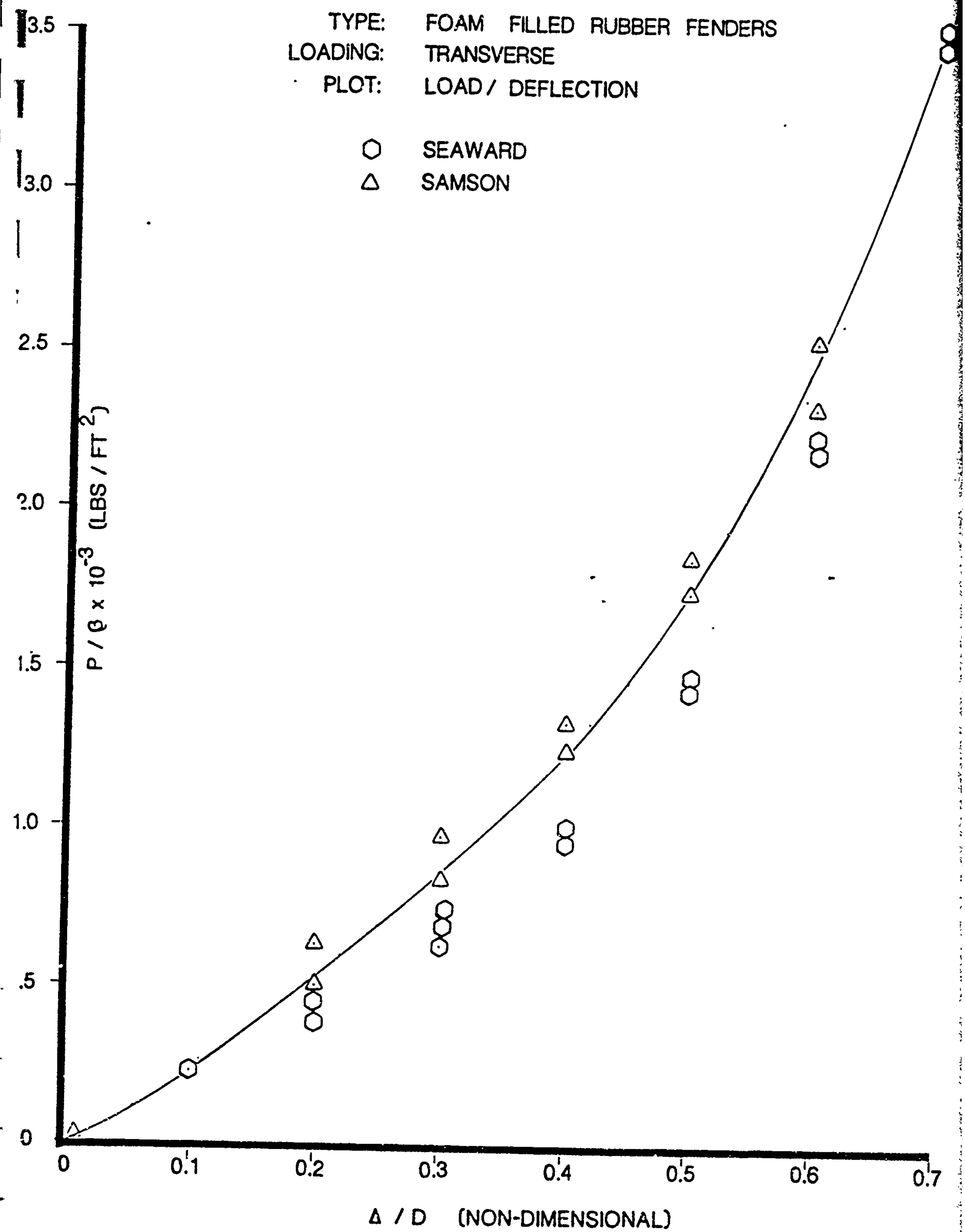


Figure 37: GENERIC PERFORMANCE PLOT



SUMMARY TABLES FOR GENERIC FENDER ALGORITHMS

FENDER TYPE & LOADING	GENERIC ALGORITHMS FOR FENDER ENERGY ABSORPTION								
	DEFLECTION (X)	FORM of the EQUATION	EQUATION CONSTANTS				STANDARD DEVIATION	EQUATION RANGE	VARIABLE DEFINITIONS
			B1	B2	B3	B4			
RUBBER									
HOLLOW CYLINDER (T)	Δ / D_i	$E/\frac{L\pi(D_o^2-D_i^2)}{4}=B_1X+B_2X^2+B_3X^3$	0.09	-5.07	9.14		3.02	$X \leq 1.5$	Δ DEFLECTION
HOLLOW CYLINDER (A)	Δ / H	$E/\frac{H\pi(D_o^2-D_i^2)}{4}=B_1X+B_2X^2+B_3X^3$	5.95	51.13	20.79		1.57	$X \leq 0.6$	L LENGTH
TRAPEZOIDAL (T)	Δ / H	$E/HLW_b=B_1X_1+B_2X^2+B_3X^3+B_4X^4$	0.57	36.55	-56.55	40.37	0.12	$X \leq 0.53$	H HEIGHT
SOLID CYLINDER (S)	Δ / H	$E/4D_o^2H\pi=B_1X+B_2X^2$	0.54	8.79			1.31	$X \leq 1.04$	
HOLLOW CUBIC (T)	Δ / H	$E/HLW_b=B_1X+B_2X^2+B_3X^3$	21.10	-74.10	208.80		5.06	$X \leq 0.67$	W_b BASEWIDTH
HOLLOW CUBIC (S)	Δ / H	$E/H(W_b^2-\pi B_d^2/4)=B_1X+B_2X^2+B_3X^3+B_4X^4$	2.63	-5.39	10.62	-3.44	1.14	$X \leq 1.97$	B_d BORE DIA
ROTARY DONUT (T)	$\Delta / \frac{(D_o-D_i)}{2}$	$E/ND_oW_b(D_o/2-D_i)=B_1X+B_2X^2+B_3X^3$	5.51	-21.3	28.05		0.75	$X \leq 0.68$	
PNEUMATIC									N # OF WHEELS D_o OUTSIDE DIA
FLOATING BAG (T)	Δ / D_o	$E/LD_o^2P^{1/1.4}=B_1X+B_2X^2+B_3X^3$	0.82	-2.54	17.94		0.13	$X \leq 0.55$	D_i INSIDE DIA
AIR BLOCK FENDER (A)	Δ / H	$E/HD_o^2P^{1/1.4}=B_1X+B_2X^2+B_3X^3+B_4X^4$	2.58	9.73	-13.40	40.05	0.28	$X \leq 0.60$	P INITIAL PRESS
AIR BLOCK CUSHION (T)	Δ / H	$E/LHW_bP^{1/1.15}=B_1X+B_2X^2+B_3X^3+B_4X^4$	-0.12	7.46	-12.17	14.77	0.01	$X \leq 0.60$	
FOAM FILLED									
FLOATING BAG (T)	Δ / D_o	$E/LD_o^2=B_1X+B_2X^2+B_3X^3+B_4X^4$	0.27	-1.03	6.43	-4.69	0.04	$X \leq 0.70$	

(T) TRANSVERSE (A) AXIAL (S) SHEAR

FENDER TYPE & LOADING	DEFLECTION (X)	FORM of the EQUATION	GENERIC ALGORITHMS FOR FENDER LOAD DEFLECTION — EQUATION CONSTANTS B1	B2	B3	B4	STANDARD DEVIATION	EQUATION RANGE	VARIABLE DEFINITIONS
RUBBER									
HOLLOW CYLINDER (T)	Δ / Di	$P/D_oL = B_1X + B_2X^2 + B_3X^3$	105.76	-254.88	163.95		16.98	$X \leq 1.50$	Δ DEFLECTION
HOLLOW CYLINDER (A)	Δ / H	$\frac{P}{\pi/4(D_o^2 - D_i^2)} = B_1X + B_2X^2 + B_3X^3$	140.69	6.40	-15.65		4.27	$X \leq 0.60$	L LENGTH
TRAPEZOIDAL (T)	Δ / H	$P/HL = B_1X + B_2X^2 + B_3X^3 + B_4X^4$	105.82	-207.06	-48.24	423.72	2.51	$X \leq 0.53$	H HEIGHT
SOLID CYLINDER (S)	Δ / H	$\frac{P}{\pi/D_o^2/4} = B_1X + B_2X^2 + B_3X^3$	22.77	1.14	-1.43		0.88	$X \leq 1.04$	
HOLLOW CUBIC (T)	Δ / H	$P/HL = B_1X + B_2X^2 + B_3X^3$	178.70	-702.80	1600.90		63.50	$X \leq 0.65$	W_b BASEWIDTH
HOLLOW CUBIC (S)	Δ / H	$\frac{P/}{W_b^2 - \pi B_d^2/4} = B_1X + B_2X^2 + B_3X^3 + B_4X^4$	21.63	-21.92	19.76	-5.06	1.25	$X \leq 1.90$	B_d BORE DIA
ROTARY DONUT (T)	$\Delta / \frac{(D_o - D_i)}{2}$	$\frac{D/}{ND_oW_b} = B_1X + B_2X^2 + B_3X^3 + B_4X^4$	-0.45	67.32	-189.60	188.46	0.37	$X \leq 0.68$	
PNEUMATIC									N # OF WHEELS
FLOATING BAG (T)	Δ / D_o	$\frac{P}{LD_oP1/1.4} = B_1X + B_2X^2 + B_3X^3 + B_4X^4$	5.19	39.95	-77.02	149.09	1.04	$X \leq 0.55$	D_o OUTSIDE DIA; DI INSIDE DIA
AIR BLOCK FENDER (A)	Δ / H	$\frac{P}{HD_oP1/1.4} = B_1X + B_2X^2 + B_3X^3 + B_4X^4$	43.96	-8.77	-62.48	256.23	1.01	$X \leq 0.60$	P INITIAL PRESS
AIR BLOCK CUSHION (T)	Δ / H	$\frac{P}{P1/1.15WL} = B_1X + B_2X^2 + B_3X^3 + B_4X^4$	9.22	-4.16	5.1[illegible]	29.55	0.06	$X \leq 0.60$	
FOAM FILLED									
FLOATING BAG (T)	Δ / D_o	$P/DL = B_1X + B_2X^2 + B_3X^3 + B_4X^4$	1.77	6.25	-13.81	16.32	0.17	$X \leq 0.70$	

(T) TRANSVERSE (A) AXIAL (S) SHEAR

V. RANKING OF FENDER SYSTEM MECHANISMS

Commonly available fenders from manufacturers of fender systems operate on the basis of one or more mechanisms which determine the way in which the fender stores energy and deflects under loading. These mechanisms are commonly:

a. Axial compression
b. Transverse compression
c. Transverse shear
d. Pneumatic bag compression
e. Foam-filled bag compression

These different mechanisms result in considerable differences in the basic performance characteristics of the individual fender types. The ranking of energy-absorbing mechanisms takes on a significantly different importance depending on the measure of merit or goal which is established for the ranking process. Since a designer is concerned with many variables such as fender energy absorption, reaction load, deflection, relative system costs, system durability, etc., the ranking of fender mechanisms will vary in accordance with his selected criteria. For purposes of this discussion, only two measures of merit are considered: energy absorption capability and reaction load as a function of deflection. These measures of merit are generally diametrically opposed. From a design point of view, one would like maximum energy absorption with minimum reaction load generation for a given deflection. In ranking the candidate mechanisms, the first approach considers which mechanisms absorbed the most energy for a given deflection with reaction load not a factor. The second viewpoint considers which mechanisms resulted in the least reaction load for a given deflection not considering energy absorption.

VI. PRELIMINARY FORMULATION OF THE FENDER/VESSEL INTERACTION RESPONSE PROBLEM

The references included in Appendix A relative to the ship/fender response problem have been designated by (**). These references approach the dynamic response problem in various ways and to various depths. Of the references cited in Appendix A, the "Dynamic Response of the Ship and the Berthing Fender System after Impact," (37) included as Appendix B for ready reference, was considered the most appropriate for further development.

The response problem formulation appears generalized enough to be adapted to include the generic fender algorithms preliminarily developed in Phase I work and hull, dock and berthing characterizations.

The essential task steps envisioned for Phase II efforts would include:

- Formulate the generalized equations of motion for the vessel/fender dynamic interaction problem based on the approach identified in Phase I work. This approach will consider fender performance algorithms, local vessel stiffness, dock mass and stiffness, vessel and berthing characteristics.
- Characterize vessel local hull or appendage stiffness.
- Characterize dock stiffness and mass characteristics.
- Characterize hydrodynamic mass and damping for vessels considered.
- Computer code methodology.
- Validate results against existing experimental data.
- Validate results against proposed test program.

It is envisioned that the first two task elements above would be based on Phase I results, references (37, (32) and (33), the basic methodology for the dynamic problem and studies related to local hull stiffnesses. Task 3 will be approached through a representative dock characterization for the ship selected initially as part of the response problem. The hydrodynamic mass and damping characteristics for this vessel would be investigated using references (37), (20) and (75) in addition to other relevant sources of mass and damping information.

It is assumed that computer coding of the dynamic equations of motion and their solution will require the significant Phase II effort. An initial validation effort will include correlation between program results and any known test results for which comparisons can be made. These will consider the results in references (8), (62), (21), (26) and (3) but not be limited to those references.

Actual test programs to be developed as part of task 7 would consider validation of fender algorithms for large size generic fenders via static or model testing since most data issued by manufacturers is based on extrapolation of small scale test data. This would be further developed as part of Phase II efforts. In addition, test programs could include validation of response program results through small scale model testing. This also would be developed further into Phase II efforts.

Appendix A

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*82. "Engineering Data on Duramax Commercial Fenders, Johnson Marine Fender Catalog." Johnson Rubber Company, January 1982. Included in this package is a description of Johnson's commercial bumpers and marine products in general. Detailed information on Johnson's rubber fender systems is provided, including energy absorption, deflection and reaction data. This booklet is a valuable source of engineering data.

83. "Fender Manufacturers - Expanding Range and Capacity." Cargo Systems International, July 1980. The results of a Cargo Systems survey of the worldwide fender market are presented. Only fourteen fender manufacturers responded; their products and capabilities are briefly summarized.

84. "Fender Selection Guide." Intertrade Industries, Ltd. Intertrade produces the Ship Guardian foam-filled fender. The company's brochure emphasizes the importance of guidelines for selecting fender systems. Energy absorption, reaction force, and standoff are discussed.

*85. "Fender System - Laid on the Line." Cargo Systems International, February 1980. A brief article describing Bridgestone's latest fender type - the Cell Shock Absorber; performance curves are included.

86. "Fenders for Marine Structures and Ships." International Dredging and Port Construction, January 1981.

*87. "General Tire Marine Fender Guide." General Tire & Rubber Company. This pamphlet describes General Tire's marine fender systems. Two tables depicting energy deflection curve and load deflection curve are included.

88. "Heavy Duty Fendering." Tanker & Bulk Carrier, February 1973.

89. "Lord Kinematics Vibration/Shock/Noise Control Products." Lord Kinematics, Lord Corporation, 1978.

90. "Marine Fenders - Becoming an Essential Insurance Against Damage." Cargo Systems International, March 1979. A brief article examining the design and selection of marine fender systems, with emphasis on the changing requirements for fenders. Some of the fender types discussed include pneumatic, rubber, foam, gravity and pile systems.

*91. "Marine Fenders Engineering Data." Goodyear Tire & Rubber Company. This fender manual is very detailed and informative. It is divided into chapters dealing with descriptions of Goodyear's product line, applications, engineering calculations and performance data, rubber compounds and the marine hardware used in installation. An extremely useful research tool.

*92. "Morse Fendering Products." Morse Chain, Borg Warner, 1982.

93. "Nordell Hydrocarbon Rubber, Engineering Properties and Applications." DuPont Company, Elastomer Chemicals Department.

94. "Offshore Mooring Fenders for ULCC's." Marine Engineers Review, October 1973.

95. "Pier Designs for the Fleets of the 1990's and Beyond. Proceedings of a Workshop Sponsored by CEL." CEL, February 1981. The workshop, sponsored by NAVFAC/CEL, was held in order to obtain a cross-section of fleet requirements, problem areas, priorities, and ideas to aid in the development and evaluation of improved pier design concepts. The Proceedings include the findings and long and short range recommendations of the workshop.

*96. "Raykin Marine Dock Fendering." General Tire & Rubber Company, Engineered Construction Products.

*97. "Regal Marine Products." Regal Tool & Rubber Co., 1979. Regal's marine products include boat bumpers, sub strips and shock cells, among others. This package provides basic information on the products with details on performance data.

98. "Rubber Fendering Reduces Damage to Vessels and Piers." Maritime Reporter, July 1981. A brief article describing the advantages of rubber fender systems, their energy absorption capabilities and reaction loads.

99. "Samson Ocean Systems Marine Products." Samson Ocean Systems, Inc., November 1976. Short brochures and a descriptive paper on non-pneumatic fenders and flotation devices.

100. "Seaward Marine Fenders." Seaward International, Inc., May 1977.

101. "Seibu Dock Fenders." Seibu Polymer Chemical Co., Ltd., 1981. Seibu manufactures rubber fenders for a wide variety of marine applications. Seibu's manual is a comprehensive guide to the company's product line. It includes detailed performance data, photographs and diagrams.

*102. "Sumitomo Rubber Fenders." Sumitomo Rubber Industries, Ltd. Brief descriptions of Sumitomo's full line of fenders, including pneumatic, rotary and rubber V-type and D-type fenders.

103. "Survey of Naval Port Fender Systems." VSE Corporation/CEL, January 1980. A survey of Navy pier fender systems was conducted to evaluate the need for an RDT&E program leading to improved fender systems. Eighteen major activities were surveyed by mail and on-site visits were made to activities in San Diego and Norfolk. Among other problems, the survey revealed poor to fair conditions for pier fender systems, increasing costs and declining quality of timber materials.

104. "Survey of Naval Port Fender Systems. Berthing of Submarines, Aircraft Carriers, Hydrofoils and Surface Effect Ships." VSE Corporation/CEL, June 1980. This survey is a follow-on to the overall survey of naval port fender systems conducted from May 1979 to January 1980. This present survey provides information, data, conclusions and recommendations concerning fender systems and camels used for submarine berthing, aircraft carrier berthing, and berthing of hydrofoils and surface effect ships.

*105. "Technical Manuals for Oreco Protective Systems." Oreco III, Inc., January 1982. Oreco III manufactures boat and offshore platform bumpers of the donut type. Their brochure gives a brief description of their product.

*106. "The Irving Marine Fender, Non-recoiling." Marine Aluminum Aanensen and Co., A/S Hawgesund, Norway.

*107. "Uniroyal Marine Fendering Systems." Uniroyal, Inc. Uniroyal produces primarily rubber fenders, small line of pneumatic. This comprehensive fendering manual includes detailed descriptions of and extensive performance data for Uniroyal's line of rubber fenders. Also included are results of full-scale testing, rubber compound specifications and descriptions of marine hardware for installation.

*108. "Vredestein Industrial Products in Building, Dredging, Marine and Offshore." Vredestein Building/Dredging/Marine, 1981. Vredestein's marine manual reflects the varied character of the company. It includes detailed descriptions of and performance data for Vredestein's rubber dock fenders as well as for dredging and sealing equipment.

*109. "Yokohama Pneumatic Rubber Fenders." Yokohama Rubber Co. Ltd., 1980. A comprehensive manual describing Yokohama's line of pneumatic rubber fenders. The manual contains photographs and diagrams of the fenders plus numerous tables and figures depicting performance data.

Appendix B

"DYNAMIC RESPONSE OF THE SHIP AND THE BERTHING FENDER SYSTEM AFTER IMPACT"

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DYNAMIC RESPONSE OF THE SHIP AND THE BERTHING FENDER SYSTEM AFTER IMPACT

By *Sadao* KOMATSU* *and Abdel Hamid* SALMAN**

ABSTRACT

This paper describes a mothod of analysis for evaluating the portion of ship kinetic energy and impact force transmitted to a berthing structure provided with fenders which have linear or non-linear spring constants. In the analysis, presented herein, the dynamic responses of the ship, fender and berthing structure, after impact, are considered, and derived equations for the selection of different parameters needed for the solutions of the dynamic equations are included. These are comprised of the virtual mass of the ship, in both translational and rotational motion, in addition to the time interval required for the solutions of the motion equations by numerical integration methods.

1. INTRODUCTION

Since the size of ships, particularly tankers, has increased in recent years, the design of off-shore berthing structures has become more important. One of the prime difficulties facing designers is the evaluation of the portion of ship kinetic energy and the impact force transmitted to each of the fenders and the berthing structure, especially structures provided with rubber-like fenders.

(1) The Kinetic Energy of the Berthing Ship

When a ship is approaching the berth with both translational and rotational motion, its kinetic energy is given by the following equation;

$$E_0=\frac{1}{2}M_sV_0^2+\frac{1}{2}I_G\omega_0^2 \qquad \cdots\cdots(1)$$

where M_s=virtual mass of the ship,
I_G =virtual moment of inertia about the vertical axis through the ship's center of gravity,
V_0 =velocity of translation,
ω_0 =angular velocity.

(2) The Effective Energy for Fender System Design

During berthing the kinetic energy of the ship may be dissipated in several ways, among which are the following:

i) Elastic deformation of the structure and fender.
ii) Swinging of the ship due to yawing motion.
iii) Heeling of the ship due to rolling motion.
iv) Elastic deformation of the ship's hull.
v) Piling of the water trapped between the ship's hull and the face of the berthing structure. (This occurs in the case of a long closed structure.)

Designers who are involved with marine structures design are interested in the portion of energy indicated by i) which is called the effective energy (E_e). The problem of determining the effective energy has been treated analytically by several investigators. Michalos[1,2] treated the problem as one which had a single-degree-of-freedom dynamic motion, and considered the theory of elastic impact in his analysis. The judgement of others[3,4], including the authors, is that the ship's dynamic impact on the structure can be considered as a plastic impact, where, upon impact, both the ship and the fender system move together as one combined mass. Vasco Costa[5] has derived a dynamic equation for estimating the effective energy in which only the portion of the energy dissipated by the yawing motion of the ship was considered. The rolling motion and the influence of the fender system dynamic response were ignored. Hayashi and Shirai[6] have dealt with the problem as one which

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has three-degrees-of-freedom dynamic motion; swaying, yawing, and rolling motion. The equations are valid i) for structures which are provided with linear spring constants, such as steel spring-like fenders. ii) for only one case of approaching mode of berthing, in which the vector of the approaching velocity is perpendicular to the arm connecting the ship's center gravity with the point of contact.

Besides, the dynamic response of the fender system was ignored. The empirical equation for determining the effective energy (E_e) is also used for design purposes and is of the form

$$E_e = CE_0 \qquad \cdots\cdots(2)$$

where E_0 represents the approaching ship's kinetic energy and C is the reduction or dispersion factor. Pages[10] suggests the following equation for determining for C;

$$C = 1/(1+16a^2) \qquad \cdots\cdots(3)$$

where $a = d/L$. L represents the ship's total length and d represents the distance between the ship's center of gravity and the point of contact, measured parallel to the berthing face. Other designers have selected a value of C which varies from 0.2 to 1.0 depending on several factors, such as the mode of berthing operation, local hull deformation, structure type, etc.[3),6),7),11)]

From published information it became clear that the portion of the energy transmitted separately to the berthing structure and fender is still a problem, especially in the case of fenders with non-linear spring constants, and rubber-like fenders, which are in this class, are being used extensively due to their large energy-absorbing characteristics.

The authors have presented a method of analysis based on the dynamic behavior of the system, after collision, to evaluate the impact load and the portion of the ship's kinetic energy transmitted to the berthing structure and fenders which includes fenders that have both linear and non-linear spring constants. Also, to evaluate the portion of the energy dissipated in the swinging and rolling of the ship after impact.

2. DYNAMIC RESPONSE OF THE SHIP AND THE BERTHING STRUCTURE AFTER THE FIRST IMPACT

(1) General Mode of Berthing

When the ship is approaching the berth under its own power, it is angled in to make the first contact with the fender system at a point near its bow or stern. This point of contact is always located in a horizontal plane higher than that passing through the ship's center of gravity. During this mode of berthing, the ship will undergo dynamic motion which has three-degree-of-freedom, namely; swaying, yawing, and rolling. The other motions, heaving, pitching and

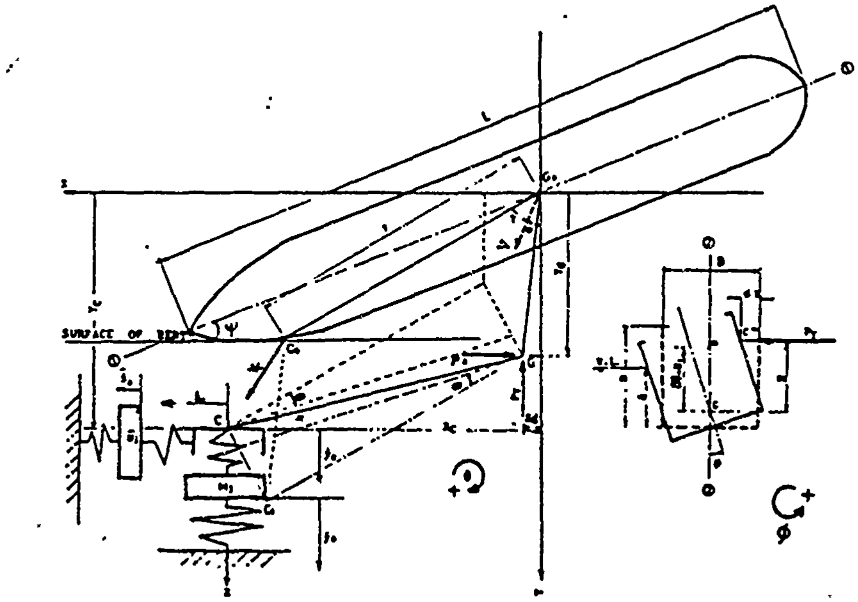


Fig. 1 Behaviour of ship after contact.

surging are of little consequence in energy dissipation and may be neglected.

(2) Equations of Motion

In the following analysis it is assumed that no sliding contact is made along the fender's surface.

Consider the motion of the ship, as a free dody under the action of the load P acting at the point of contact C, we have;

At any time t, after the ship came in contact with the fender system, Fig. 1, its center of gravity will sway in the direction of the acting load, to the position G. Then the ship will swing about its vertical axis passing through G, an angle θ due to the yawing motion and finally it will roll about its horizontal longitudinal axis an angle ϕ. Denote the coordinates of the final positions of G and C, with respect to the axes X and Y, by (X_G, Y_G) and (X_C, Y_C) respectively. The X and Y axes are taken parallel and normal to the surface of berthing respectively. From the figure, the relation connecting the fender system motion at C with the motion of G is;

$$\left.\begin{aligned} \ddot{X}_C &= \ddot{X}_G+(r\ddot{\theta}+H\ddot{\phi})\cos(\gamma+\alpha) \\ \ddot{Y}_C &= \ddot{Y}_G-(r\ddot{\theta}+H\ddot{\phi})\sin(\gamma+\alpha) \end{aligned}\right\} \quad \cdots\cdots(4)$$

the second order terms in the above equation, $\dot{\theta}^2$, $\dot{\phi}^2$ are neglected.

Consider the dynamic equilibrium of G, the following equations of motion will hold;

$$\left.\begin{aligned} &\text{i) SWAY} \\ &\quad M_2\ddot{X}_G=-P_X-Rw_X \\ &\quad M_2\ddot{Y}_G=-P_Y-Rw_Y \\ &\text{ii) YAWING} \\ &\quad I_{3-3}\ddot{\theta}=P_Y\cdot r\cdot\sin(\gamma+\alpha) \\ &\qquad\qquad -P_X\cdot r\cdot\cos(\gamma+\alpha)-N_1 \\ &\text{iii) ROLLING} \\ &\quad I_{1-1}\ddot{\phi}=(P_Y\cdot\cos\psi-P_X\cdot\sin\psi)H \\ &\qquad\qquad -W\cdot H_1\cdot\phi-N_2 \end{aligned}\right\} \quad \cdots\cdots(5)$$

P_X, P_Y and Rw_X, Rw_Y are the components of the fender reaction and the water resistance, after the ship came in contact with the fender system, respectively. N_1 and N_2 denote the water resistance to yawing and rolling motions respectively. However, as the time of contact is very small, water resistance, is safely allowed. Water resistance is effective in the time between the first and the second impact, this will be discussed in details in next paper.

As for the berthing structure response, the following eq. will hold;

$$\left.\begin{aligned} M_1\ddot{X}_s &= P_X-K_X\cdot X_s-\mu_X\cdot\dot{X}_s \\ M_1\ddot{Y}_s &= P_Y-K_Y\cdot Y_s-\mu_Y\cdot\dot{Y}_s \end{aligned}\right\} \quad \cdots\cdots(6)$$

In the previous equations the value of P is evaluated from the given load—deflection relation (P-D) of the fender in question. The load is considered to be applied in small increments associated with the time interval. This P-D relation is determined from statical analysis or tests. At any time t, the deflection D will be calculated from the displacement of the point of contact C and the deflection of the structure, which will equal to;

$$\left.\begin{aligned} D_X &= X_C-X_s \\ D_Y &= Y_C-Y_s \end{aligned}\right\} \text{ and } D=\sqrt{D_X^2+D_Y^2} \quad \cdots\cdots(7)$$

In considering the second impact, the velocity of the point of the first contact at separation, magnitude and direction is needed, which will equal to;

$$\left.\begin{aligned} V_C &= \sqrt{\dot{X}_C^2+\dot{Y}_C^2} \\ \alpha_t &= \tan^{-1}(\dot{X}_C^2/\dot{Y}_C^2)_t \end{aligned}\right\} \quad \cdots\cdots(8)$$

At time of contact $\alpha_{t_0}=\alpha$, the angle that the approaching velocity makes with Y-axis. α is considered positive when the velocity vector of V_0 at G points towards the point of contact C.

If the value of the spring constant of the ship hull at point of contact is available, the elastic deformation of the ship's hull can be evaluated. Let K_h, S_h, and μ_h define the spring constant, deflection, and the damping coefficient of the hull at point of contact respectively, then the equation of deflection of the ship's hull, in P-direction, will be;

$$\ddot{S}_h=(P-K_h\cdot S_h-\mu\cdot\dot{S}_h) \quad \cdots\cdots(9)$$

In this case P will be function of $(X_C, Y_C, X_s, Y_s, S_h)_t$. The initial conditions of motion, at time of contact, are;

$$\left.\begin{aligned} &X_G=Y_G=\theta=\phi=X_C=Y_C \\ &\qquad =X_s=Y_s=S_h=0.0 \\ &\dot{X}_G=\dot{X}_C=V_0\sin\alpha \\ &\qquad \dot{Y}_G=\dot{Y}_C=V_0\cos\alpha \\ &\dot{\theta}=\dot{\phi}=\dot{X}_s=\dot{Y}_s=\dot{S}_h=0.0 \end{aligned}\right\} \quad \cdots\cdots(10)$$

The solution of dynamic equations is carried out by numerical integration methods with the help of the digital computer.

(3) Energy Equations

The developed previous equations are valid as long as the ship being in contact with the fender. During this time, the following energy equations are valid;

I. Part of the ship's kinetic energy transmitted to;

i) Fender $V_f=\int^t P\cdot dD_t$

ii) Structure $V_s=\int^t K_s\cdot dS_t$

iii) Ship's Hull $V_h=\int^t K_h\cdot dS_{h_t}$

$=$effective energy E_e

II. Work done by the ship in rotational motions;

i) Swinging $W_s=\int^t P\cdot r\cdot d\theta_t$

ii) Heeling $W_h=\int^t P\cdot H\cdot d\phi_t$

III. Part of energy induced in the system vibration;

i) Ship $E_{sh.}=\frac{1}{2}M_2(\dot{X}_G^2+\dot{Y}_G^2)$

$+\frac{1}{2}I_{2-2}\dot{\theta}^2+\frac{1}{2}I_{1-1}\dot{\phi}^2$

ii) Structure $E_s=\frac{1}{2}M_1(\dot{X}_s^2+\dot{Y}_s^2)$

......(11)

The above equations should satisfy the conservation of energy during berthing *i.e.*

$$E_0=\frac{1}{2}M_2V_0^2=E_e+E_{sh.}+E_s \qquad \text{......(12)}$$

(4) Broadside Berthing

If the motion of the ship during the berthing operation is mainly governed by tugboats, as is always the case with the large ships, the ship can make contact with the berthing structure entirely broadside. In this case the ship will undergo dynamic motion which has two-degree of-freedom; swaying and rolling, and terms containing θ in equations (5) and (11) with vanish.

If the energy dissipated by rolling motion is neglected for safety, then the fender system will be designed to absorb all of the kinetic energy of the ship, which is the case when $C=1.0$ in equation (2). This mode of berthing is considered ideal as the ship impact load will be uniformly distributed on the structure[10].

If we assume that the berthing structure is provided with a fender which has a linear spring constant K_f the equations of motion will be;

i) Structure

$\ddot{Y}_s=(P-K_s\cdot Y_s-\mu\cdot\dot{Y}_s)/M_1$ ......(13)

ii) Ship (+fender)

$\ddot{Y}_c=-P/M_2$

in which $P=K_f(Y_c-Y_s)$ and Y_s, Y_c are the displacements of the structure and the point of contact respectively in Y-direction Fig. 1.

If, at the time of contact, the following conditions exist;

$$Y_s=Y_c=0.0 \quad \dot{Y}_s=0.0 \quad \dot{Y}_c=V_0 \qquad \text{......(14)}$$

and assuming $\mu=0.0$, the analytical solution of these eq. (13) is given by;

$$\left.\begin{aligned}Y_s&=A_1\sin p_1t+A_2\sin p_2t\\ Y_c&=A_1B_1\sin p_1t+A_2B_2\sin p_2t\end{aligned}\right\} \text{......(15)}$$

The equations of energy become:

$$\left.\begin{aligned}V_s&=\int^{Y_s}K_sY_sdY_s=\int^tK_sY_s\dot{Y}_sdt\\ &=1/2K_s(A_1\sin p_1t+A_2\sin p_2t)^2\\ V_f&=\int^tK_f(Y_c-Y_s)(\dot{Y}_c-\dot{Y}_s)dt\\ &=1/2K_f[A_1(B_1-1)\sin p_1t\\ &\quad+A_2(B_2-1)\sin p_2t]^2\end{aligned}\right\}\text{......(16)}$$

In the case of a very rigid berth, which offers a large resistance, the deflection Y_s will be very small and, consequently, its ability for energy absorption will be very poor, and can, therefore, be neglected. In this case all the portion of energy consumed in the swaying motion of the ship should be absorbed by the fenders.

Notations

I_{1-1}=polar moment of inertia about the longitudinal axis (1−1) passing through the C.G.

I_{2-2}=polar moment of inertia about the vertical axis (2−2) passing through the C.G.

M_1=the effective mass of the structure.

M_2=the virtual mass of the ship while swaying.

P=the ship acting load.

H, H_1=the vertical distances between the C.G. and the point of contact and the meta-center respectively.

r=the distance from the C.G. to the point of contact.

w=the ship's displacement weight.

γ=the angle that the velocity vector makes with the arm r at time of the first contact. ......(17)

K_x, K_y = structure stiffness in X and Y direction.

μ_x, μ_y = structure damping coefficients in X and Y direction.

X_s, X_y = the displacement of the structure in X and Y direction.

$$A_1 = V_0/\rho_1(B_1 - B_2)$$
$$A_2 = -V_0/\rho_2(B_1 - B_2)$$
$$\rho_{1,2}^2 = (a+b)/2 \mp \sqrt{[(a+b)/2]^2 + bc}$$
$$B_1 = b/(\rho_1^2 - a)$$
$$B_2 = b/(\rho_2^2 - a)$$
$$a = (K_s + K_f)/M_1$$
$$b = K_f/M_1$$
$$c = K_f/M_2$$

3. SELECTION OF THE PARAMETERS FOR SOLUTIONS OF THE DYNAMIC EQUATIONS

The dynamic equations of motion presented in section 2 contain the following parameters;

for the ship, V_0, M_2, I_{1-1}, I_{2-2}, H_1,

for the structure, M_1, K_s,

for the fender, $(P\text{-}D)$

and in practical design, the ship's displacement weight (W) is, generally, the only value given with the information for the marine structure to be constructed.

This section includes equations, tables, and graphs to help in selecting or computing the unknown parameters for solutions of the dynamic equations.

(1) Approach Velocity, V_0

From eq. (1) the impact energy is proportional to the square of the approach velocity. Thus, the energy level will increase considerably if the velocity is only slightly increased. In the selection of this velocity for design, many factors should be considered, such as:

1) Method of docking

A ship approaching the berth under the control of tugboats usually berths with less velocity than one approaching under its own power.

2) Berthing conditions

In the case where the berth is exposed to wind, waves, currents, etc., it is more difficult to control the ship velocity than in a sheltered berth and the velocity may become large. However, for extremely high wind velocities in the order of 100 to 120 mph that occur during short periods, it is advisable to require ships to temporarily anchor away from the berth in order to avoid being damaged[11].

3) Ship size

Larger ships are always berthed with great care and with the assistance of tugboats. It is generally assumed that the larger the ship, the smaller will be the velocity with which the ship will contact the fender system[8].

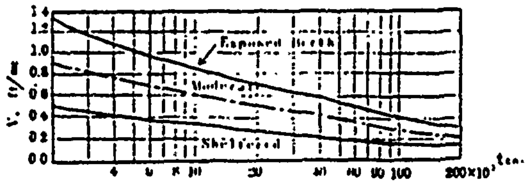


Fig. 2 Berthing velocity normal to dock vs. Ship-displacement weight (after Lee).

T. Lee presented the curves shown in Fig. 2 from which the berthing velocity may be selected for design. Under various conditions of berthing, Vasco Costa[3] recommended Table 1 as a guide for the selection of velocity.

Before proceeding to the selection of other parameters, some relations concerning the ship characteristics will be discussed. For example, a tanker of length L, draft d and beam breadth B, Fig. 1, will have the following empirical relations:

$$W = c_B \rho L B d$$
$$c_B = 0.75 \text{ to } 0.80 \doteqdot 0.78$$

Table 1 Approach velocity of berthing ships

(ft/sec)

Wind and swell	Approach conditions	Displacement of the ship		
		Up to 3,000 ton	Up to 10,000 ton	Over 30,000 ton
Strong wind and swell	Difficult	2.5	2.0	1.5
Strong wind and swell	Favourable	2.0	1.5	1.0
Moderate wind and swell	Moderate	1.5	1.0	0.8
Protected	Difficult	1.0	0.8	0.6
Protected	Favourable	0.8	0.6	0.4

(after V. Costa)

$$\left.\begin{array}{l} D/d=\text{total depth/draft}=1.33^{[13]} \\ L/D=13.75 \\ L/d=18.25 \\ \overline{OG}=d-0.52D=0.31d \\ \quad k=\text{radius of gyration about vertical} \\ \quad\quad \text{axis through } G=0.2L^{[8]} \\ \quad k_x=\text{radius of gyration about longi-} \\ \quad\quad \text{tudinal axis through } G=n'B \\ \quad n'=0.37 \text{ to } 0.47\cong 0.42 \end{array}\right\}\cdots\cdots(18)$$

(2) Derivation of Added Mass Equations for Berthing in Shallow Water

1) Added mass in horizontal motion, M_1

It is well known that when a ship moves from deep to shallow water, as in the case when berthing, the added virtual mass is increased due to the presence of restricting boundaries. Koch[13] investigated the effects of shallow water on added virtual weight for both vertical and horizontal vibration. The measurements were made for a block having a half-beam b and a draft d. The results of the experiments are shown in Fig. 3. To apply the Koch results on hull shape sections, it is suggested that the added weight to any section of the ship, calculated for deep water for this particular section shape, should be increased by the ratio of the added weights in shallow and deep water for a rectangular section of the correct beam-draft ratio and depth of water.

Furthermore, on the basis that many modern ·nkers have very similar hull shapes, sections

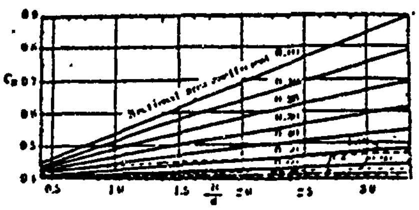


Fig. 5 Values of added mass coefficient C_H for horizontal motion in deep water.

for typical tankers, Fig. 4[13], are considered in the analysis hereafter.

With the help of the curves of Fig. 5 and these sections, the distribution of the added mass along the ship length may be obtained (step 6 in Table 2, and curve a Fig. 7). Considering the Koch results for rectangular sections, and the procedure above, the added mass of actual hull shape sections, where the ratio of water depth T to draft d is 2.0, can be evaluated (steps 7 to 10, Table 2). The results are shown by curve b of Fig. 7).

However, the water depth considered by Koch was deeper than that required for berthing. The experimental results obtained by Marwood and Johnson, Fig. 6 are of considerable help in this field[13]. This is bearing on the fact that the percentage increase in C_H in shallow water, where $T/d=2.0$, than that in deep water for the ship mid-sections (4-7), computed on the basis

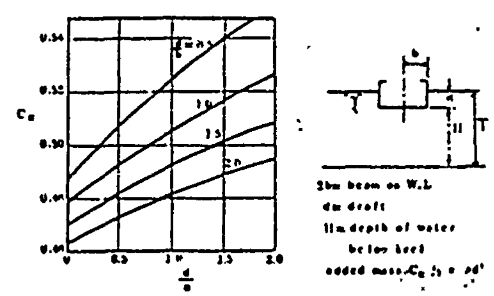


Fig. 3 Effect of shallow water on added mass in horizontal vibration, Curves of C_H for rectangular section (after Koch).

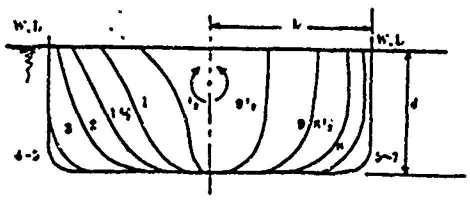


Fig. 4 Actual sections of ship (after Kumai).

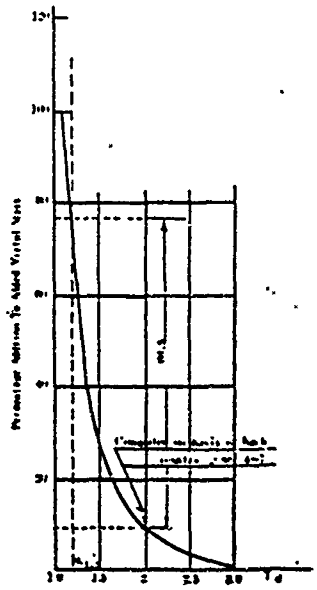


Fig. 6 Effect of shallow water on added mass —(Marwood & Johnson).

Table 2 Evaluation of added mass in shallow water

	Section No. →	0	$\frac{1}{2}$	1	$1\frac{1}{2}$	2	3	4	5	6	7	8	$8\frac{1}{2}$	9	$9\frac{1}{2}$	10
1	b=1/2 beam on W.L.	0.0	12.0	18.5	23.0	26.0	27.5	27.5	27.5	27.5	27.5	26.5	24.5	19.0	10.0	0.0
2	$A=b\times d$ (d=21)	0.0	441	388.5	483	546	577.5	577.5	577.5	577.5	577.5	556.5	514.5	399	210	0.0
3	S=Actual area	0.0	111	250.3	339.5	441.5	558.5	574.4	574.4	573.8	573.8	536	573.13	365.5	195	0.0
4	σ=Area. coeff. S/A	1.0	0.252	0.644	0.703	0.809	0.967	0.995	0.995	0.994	0.994	0.963	0.929	0.916	0.929	1.0
5	$2b/d$	0.0	1.143	1.762	2.190	2.476	2.619	2.619	2.619	2.619	2.619	2.524	2.333	1.810	0.952	0.0
6	C_{H_0} (Deep water) Fig. 5	0.0	0.49	0.42	0.42	0.405	0.44	0.46	0.46	0.46	0.46	0.44	0.42	0.415	0.41	0.0
7	$C_{H'}$ (Deep water) σ=1	0.0	0.445	0.45	0.46	0.465	0.465	0.465	0.465	0.465	0.465	0.465	0.46	0.45	0.44	0.0
8	d/b	0.0	1.75	1.135	0.913	0.808	0.764	0.764	0.764	0.764	0.764	0.792	0.857	1.105	2.10	0.0
9	$C_{H''}$ (Shallow W.) T/d=2.0 σ=1 Fig. 3	0.0	0.488	0.501	0.510	0.512	0.516	0.516	0.516	0.516	0.516	0.513	0.511	0.502	0.48	0.0
10	C_H (Shallow W.) $C_H=C_{H_0}\frac{C_{H''}}{C_{H'}}$	0.0	0.537	0.468	0.466	0.446	0.488	0.510	0.510	0.510	0.510	0.485	0.467	0.463	0.477	0.0
11	C_H for shallow W. T/d=1.2	0.0	0.886	0.772	0.769	0.736	0.805	0.840	0.840	0.840	0.840	0.80	0.771	0.764	0.738	0.0

of the Koch results, showed a close agreement with Marwood and Johnston's experimental results, as indicated in Fig. 6.

Taking into account the results shown in Fig. 6, the derived values of the added mass coefficients in step 10 are re-calculated for a water depth and draft ratio of 1.2 (step 11, Table 2 and curve c, Fig. 7).

By integrating along the length and substituting for $d=B/2.61$ from Table 2, the following equation, for the added mass in horizontal vibration, was derived;

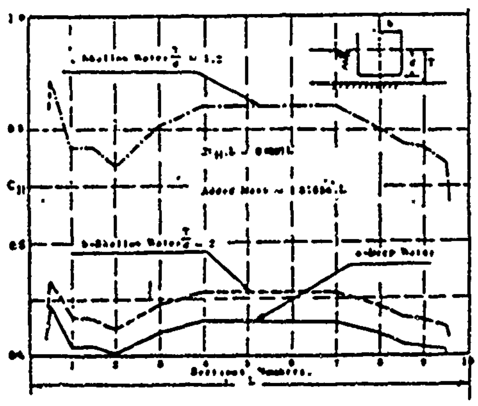


Fig. 7 Distribution of added mass along length.

$$m'=\frac{1}{2}\rho dBL \qquad \cdots\cdots(19)$$

The virtual mass becomes

$$M_1=m+m' \qquad \cdots\cdots(20)$$

and the virtual moment of inertia I_{1-1}, in yawing motion will be[1,3]

$$I_{1-1}=M_1k^2 \qquad \cdots\cdots(21)$$

2) Checking the derived formula

Taking into account both model and prototype experiments, Vasco Costa[8] presented the following equation for estimating the virtual mass of a berthing ship;

$$M_v=M+m'=m(1+2d/B) \qquad \cdots\cdots(22)$$

Shu T'ien Li[11] also presented the following equation;

$$M_v=m(1+\pi B/(16D)) \qquad \cdots\cdots(23)$$

The virtual mass for different tankers, using the three equations, was calculated, Table 3. It can be seen that the derived equations yield results which are about 5% more than the Shu T'ien Li equation and about 5% less than the Vasco Costa equation.

3) Added mass of inertia in rolling motion

The added mass moment of inertia coefficients depend upon the sectional shapes and the ratio of beam to draft; the same parameters which

Table 3

D/W	Displ. W.	$\frac{L}{m}$	B	d	I	II	III	$\frac{III}{I}$
Tankers	ton	m	m	m	ton	ton	ton	%
10 000	13 300	140	17.2	7.9	25 500	19 000	24 700	97.0
20 000	26 700	178	22.4	9.5	48 000	39 000	47 750	99.9
30 000	40 000	200	25.8	10.3	72 000	59 600	68 200	95.0
50 000	66 600	230	32	11.4	114 000	103 000	105 700	92.5
85 000	113 000	260	38.1	14.0	196 000	173 000	180 000	92.0
100 000	133 000	285	41.2	14.6	223 000	206 700	213 000	95.0

I=Vasc Costa Formula
II=Shu T'ien Li Formula
III=The New Formula

average: 95%

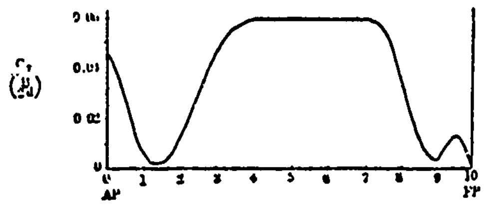


Fig. 8 Distribution of added virtual weight along length in torsional vibration (after Kumai).

effect C_H. Also, these coefficients depend on the location of the center of rotation. Model experiments were carried out by Kumai on prismatic models having sections corresponding to those of a tanker[11]. Applying the experimental results on the actual sections of a tanker ship, Fig. 4, Kumai obtained the distribution of the added mass moment of inertia along a typical tanker hull in the loaded condition, Fig. 8. By integrating along the length, Kumai derived the following expression;

$$\Delta I_t = 0.00531(1+0.365d/d_f)B^4L \quad \text{(ton-m}^2\text{)} \qquad \cdots\cdots(24)$$

For a fully loaded tanker ($d=d_f$), the above expression becomes;

$$\Delta I_t = 0.00725B^4L \qquad \text{(ton-m}^2\text{)} \qquad \cdots\cdots(25)$$

Eq. (24) represents the added mass moment of inertia in deep water, but, as was discussed previously, berthing always takes place in shallow waters. Therefore, the effect of shallow water on the inertial moment has to be considered.

Matsuura and Kawakami[12] performed numerical computations, applying the finite element method, on the effect of the restricted water on the inertial coefficients, C_T. Rectangular sections having a ratio of half-beam to draft of 1.0 and two locations of center of rotation, $y_0/d=0.0$ and 1.5, were investigated, Fig. 9.

To apply the results shown in Fig. 9 to the case of a fully loaded ship, the following corrections are necessary; a) first, since we are interested in the moment of inertia (I_{1-1}) about the center of gravity of the ship where $y_0/d=0.31$, eq. (18), the effect of shallow water on the cen-

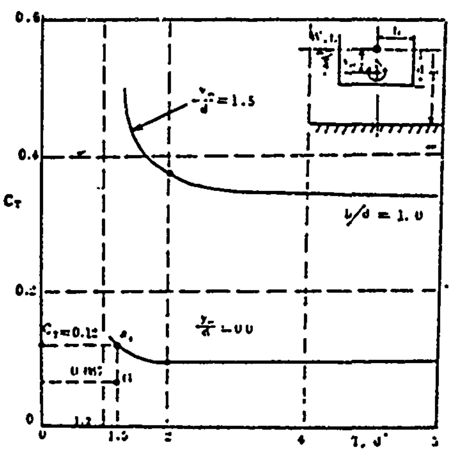


Fig. 9 Effect of shallow water on added mass moment of inertia in rolling (after Matsuura & Kawakami).

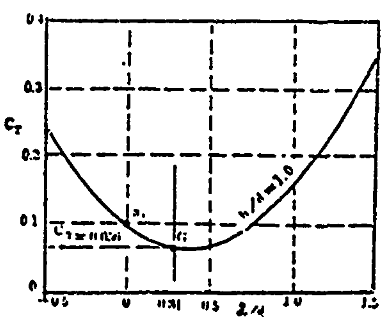


Fig. 10 Variation of C_T with respect to center of rotation for $b/d=1.0$ (after Matsuura & Kawakami).

ter of rotation becomes important. This is obtained by comparing the inertial coefficients, C_T, at point a_1 where $T/d=1.2$, Fig. 9, with point a_2 in Fig. 10. That is,

$$C_{Ta_1}/C_{Ta_2}=0.12/0.1=1.2$$

From which the value of C_T at $y_0/d=0.31$ and $T/d=1.2$, namely C_{TG}, will approximately be equal to

$$C_{TG}=1.2(0.056)=0.067 \qquad \cdots\cdots(26)$$

b) The second correction is obtained from the consideration of three dimensional motion. The correction factors from 2 to 3 dimensional motion can be obtained from Fig. 11[14]. If $L/d=18.25$, eq. (18), the correction factor corresponding to pure rolling, $(n=0)$, is equal to 0.96. Multiplying eq. (26) by this value yields;

$$C_{TG}=0.067(0.96)=0.064 \qquad \cdots\cdots(27)$$

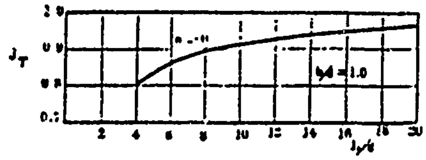


Fig. 11 Taylor correction factor from 2 to 3 dimensional motion.

c) The third correction is obtained from the effect of actual ship hull sections. A comparison was made with sections 4 to 7, Fig. 8, having C_T in deep water equal to 0.06, and the rectangular-shaped sections where C_T in shallow water was derived, eq. (27). From Table 2 the area coefficient (σ) of these sections is equal to 0.996 or about 1.0, giving a very slight effect due to the round edges of these particular sections. The other sections have already been considered by the use of the equation derived by Kumai, eq. (24). Hence, for deduction, the C_T values of ship sections in shallow water (values included in Fig. 8) can be multiplied by the ratio 0.064/0.06, which denotes the C_T value in shallow water as compared to deep water for sections 4 to 7. This will lead, finally, to multiplying eq. (25) with the above ratio for obtaining the added mass moment of inertia, as follows, in shallow water;

$$\Delta I_t=0.00774B^4L \quad (\text{ton}\cdot\text{m}^2) \qquad \cdots\cdots(28)$$

The polar moment of inertia, I_t, about a longitudinal axis passing through G is equal to

$$I_t=m(k_x)^2$$

and substituting from eq. (18) results

$$I_t=0.78\rho LBd(0.42B)^2$$

Letting $d=B/2.62$ from Table 2, step 5, $\rho=1.03$ ton/m³

$$I_t=0.0341B^4L \quad (\text{ton}\cdot\text{m}^2)$$

$$\Delta I_t/(I_t)=(0.00774B^4L)/(0.0341B^4L)=0.143 \qquad \cdots\cdots(29)$$

This ratio is in close agreement with the ratio of 0.15 given by Prof. Hayashi[4]. The virtual moment of inertia I_{1-1} becomes

$$\left.\begin{aligned} I_{1-1}&=1.143m(k_x)^2=0.202B^3m \quad (\text{ton}\cdot\text{m}^2)\\ \text{or}\quad I_{1-1}&=I_t+\Delta I_t=0.0618B^4L \quad (\text{ton}\cdot\text{m}^2)\end{aligned}\right\} \qquad \cdots\cdots(30)$$

4) H_1 or $\overline{GM}$, which denotes the vertical distance between the ship's center of gravity, G, and its metacentre, M, Fig. 1, can be calculated from the following equation[15];

$$B=L/C_1+\overline{GM}(d/D)C_2 \qquad \cdots\cdots(31)$$

where C_1 and C_2 are constants having the following values for oil tankers (where $\overline{KG}/D=0.52$);

$C_1=12.5$ (U-shape)~13.2 (V-shape)

$C_2=5.7$

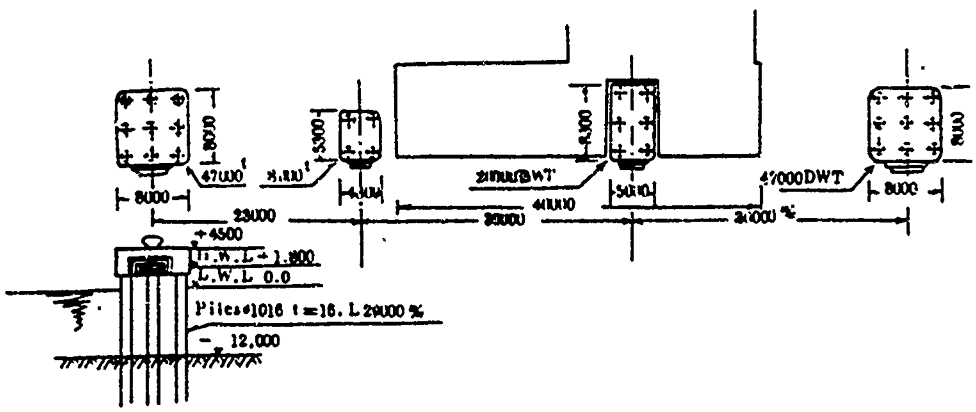


Fig. 12 Mitsubishi Shoji Oil Berth.

5) Effective mass, M_1, of the structure

For determining the value of M_1 that should be substituted in the motion equations, the authors investigated some existing berths in Kobe harbor.

One is an oil berth belonging to the Mitsubishi Shoji Co. which consists of four dolphins of different sizes and a platform, Fig. 12, located back from the berth line. The berth is provided with V-type rubber fenders. The berth was designed to accomodate tankers varying in size (8 000, 20 000 and 47 000 DWT) under the control of tugboats.

a) The 47 000 DWT berths against two dolphins which have an effective weight equal to

cap+effective piles weight

$$=2(160+180)=680 \text{ tons.}$$

b) The 20 000 DWT should use two dolphins which have an effective weight equal to

$$2(100+120)=440 \text{ tons.}$$

c) The 8 000 DWT should berth against two dolphins which have an effective weight equal to

$$2(60+80)=280 \text{ tons.}$$

The displacement weights of the above ships are approximately (1.3 W) or 61 000, 26 000 and 10 400 tons, respectively. The virtual masses (M_2) will be equal to 104 000, 48 000 and 25 000 tons, respectively, Table 1 and the ratio M_1/M_2 will be equal to 680/104 000=0.007, 440/48 000=0.009, and 280/25 000=0.011.

$$M_1/M_2=0.01 \qquad \cdots\cdots(32)$$

Thus for the first design approximation, we can assume that M_1 will be as much as 1% of the virtual mass, M_2, of the approaching ship. If, at the end of the calculations, the difference between the derived value of M_1 and the assumed value is great, the calculations can be repeated using the derived value for M_1.

6) Structure and fender resistance

As discussed previously for the design of marine structures, the function of the structure should be known in advance in addition to the ship displacement weight. In the selection of the resistance (K_S) of the structure, the structure functions, whether rigid or flexible, should be considered. If the berth carries heavy loads (heavy, delicate equipment carried on the deck, cranes, power station, etc.), there is no choice; deflection must be limited. The construction should be rigid and provided with elastic fenders to absorb the ship's kinetic energy. On the other hand, if deflection is allowed, the berth can be flexible. A pile dolphin is a good example for this type.

4. SOLUTIONS OF THE DYNAMIC EQUATIONS BY NUMERICAL INTEGRATION

The differential equations of motion of the ship and the berthing structure are in the form of:

$$\left.\begin{array}{ll} \text{Structure} & \ddot{X}_s=f(X_s, P(X_s, X_c), \dot{X}_s) \\ \text{Ship} & \\ \quad \text{i) roll} & \ddot{\phi}=G(\phi, P(X_s, X_c), \dot{\phi}_1) \\ \quad \text{ii) yaw} & \ddot{\theta}=G_1(\theta, P(X_s, X_c), \dot{\theta}) \\ \quad \text{iii) sway} & \ddot{X}_c=G_2(X_c, \ddot{\theta}, \ddot{\phi}, P(X_s, X_c), \\ & \qquad (X_c-X_s)) \end{array}\right\} \qquad \cdots\cdots(33)$$

(These notations are explained in section 2.)

In eq. (33) the first differential terms represent the damping effect which is generally neglected. Solutions of eq. (33) are quite difficult to be carried out analytically, especially in the case where fenders (such as rubber) that have non-linear spring constants are used.

A numerical solution implies the determination of the displacement and velocity of a system as a function of time. These displacements and velocities are obtained in a step-by-step integration procedure, starting with given initial conditions. There are many different methods of numerical integration from which two methods, the Newmark β method[16] and the Runge-Kutta-Gill method[17], will be explained.

(1) Time Interval Effect on the Two Numerical Integration Methods

Tests have been conducted to study the effect of the time interval on the accuracy of the two methods. In these tests, fenders with linear spring constants were used, for which the exact solution of the equation of motion is obtained by using eq. (15). Through the comparison of results which are included in Table 4 and Fig. 13, the following conclusions could be made:

1) The error involved in fender absorbed energy, V_F, is relatively small compared to the energy V_s absorbed by the structure.
2) The percentage of error involved in the V_s values is nearly twice that of the structure maximum deflection, X_1, *i.e.* $\Delta V_s\%=2\Delta X_1\%$.
3) For the same time interval the error incurred using the Runge-Kutta-Gill method is greater than that involved in the Newmark β method. The difference also increases as the time interval is increased.

Table 4 Convergence of error with respect to time interval variation

Berthing data K_1 ton·cm	K_2 ton·cm	M_1 ton·sec²·cm	M_2 ton·sec²·cm	V cm·sec	Max value of	Exact solution	Method name	ΔT=0.01 Absolute	ERROR %	0.0075 Absolute	ERROR %	0.005 Absolute	ERROR %	0.0025 Absolute	ERROR %
800	60	0.3	30	20	X_f cm	1.049	Newmark	1.299	23.8	1.190	13.4	1.67	10.6	1.091	4.0
							Rung-Kutta	••	••	6.710	539	2.167	106	1.198	14.2
					V_s ton·cm	440	Newmark	674	53	569	29	538	22	476	8.2
							Rung-Kutta	••	••	18 009	~	1 879	327	575	30.7
					V_f ton·cm	5 568	Newmark	5 823	4.5	5 751	3.2	5 672	1.8	5 604	0.6
							Rung-Kutta	••	••	10 778	••	6 335	14	5 670	1.8
200	30	0.3	30	20	X_f cm	2.867	Newmark	3.076	7.3	3.027	5.3	2.959	3.41	2.912	1.6
							Rung-Kutta	3.964	38.3	3.410	18.9	3.111	8.5	2.952	3.0
					V_s ton·cm	821	Newmark	946	15.2	916	11.6	875	6.5	848	3.3
							Rung-Kutta	1 571	91	1 163	41.6	968	17.9	871	6.8
					V_f ton·cm	5 261	Newmark	5 397	2.5	5 364	1.9	5 321	1.14	5 290	0.55
							Rung-Kutta	6 043	14.8	5 630	7.0	5 420	3.0	5 315	1.0
800	60	4.5	90	15	X_f cm	1.376	Newmark	1.419	3.1	1.406	2.2	1.395	1.4	1.385	0.6
							Rung-Kutta	1.462	6.3	1.434	4.2	1.410	2.5	1.392	1.16
					V_s ton·cm	757	Newmark	805	6.3	791	4.5	778	2.7	767	1.3
							Rung-Kutta	855	12.9	823	8.7	795	5.02	775	2.37
					V_f ton·cm	9 479	Newmark	9 551	0.78	9 531	0.5	9 512	0.3	9 495	0.15
							Rung-Kutta	9 589	1.16	9 552	0.77	9 522	0.45	9 498	0.2

•• derived results are meaningless.

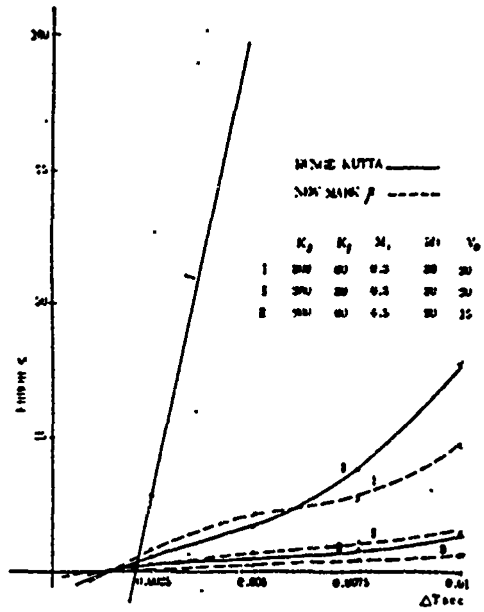


Fig. 13 Convergence of error with respect to time interval.

(2) Selection of a Suitable Time Interval

Through many investigations carried out by the authors, Fig. 14, a formula linking the time interval variation, the berthing data, and the error involved in the maximum deflection of the berthing structure was developed.

The curves shown in Fig. 15 were plotted from Fig. 14 for an error equal to 2% of the structure maximum deflection. This gives 2% error in the structure maximum reaction (K_sX_s) and nearly 4% in the structure's stored energy. In this figure the x-axis represents the berthing data, which is the factor N, and the y-axis represents the time interval Δt. The factor N is a function of the ship's virtual mass, the structure's effective mass, the structure and fender spring constants, and the ship's approaching velocity according to the following equation;

$$N=(M_1/M_sA_1)(100)\ \text{(cm)} \qquad \cdots\cdots(34)$$

where A_1 is obtained from eq. (17).

From Fig. 15 the relation between N and the time interval Δt for an error of 2% in ΔX_1 is given by:

i) The Newmark β method ($\beta=1/4$)

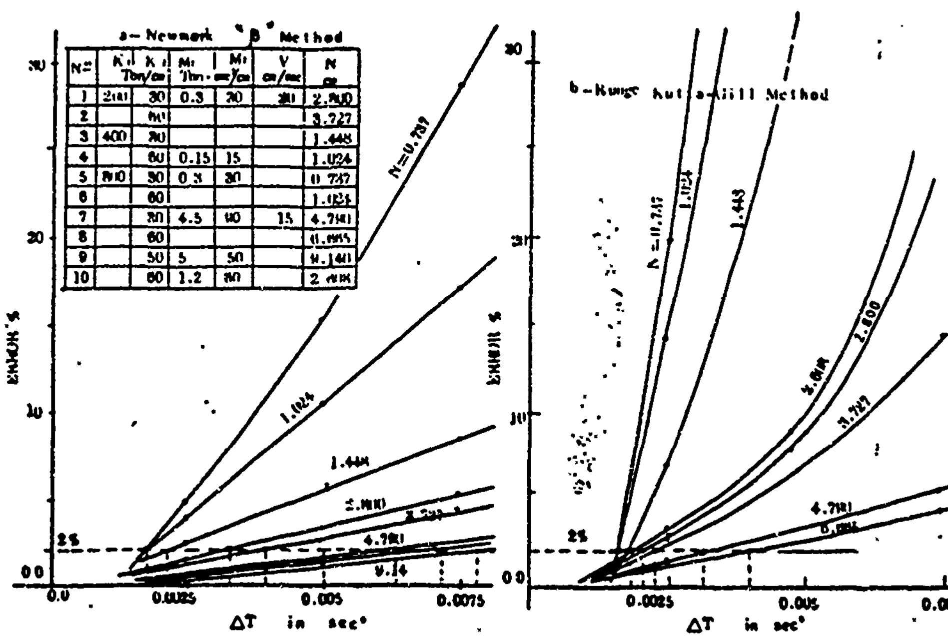
$$\Delta t=0.00067N+0.0012\ \text{(sec)}. \qquad \cdots\cdots(35)$$

$$N\leq 3.73$$

$$\Delta t=0.00015N^3-0.0029N^2+0.018N-0.0313\ \text{(sec)} \qquad \cdots\cdots(36)$$

$$N\geq 3.73$$

ii) The Runge-Kutta-Gill method



N=	K_f Ton/cm	K_s Ton/cm	M_1 Ton·sec²/cm	M_s	V cm/sec	N cm
1	200	30	0.3	30	[illegible]	2.900
2		60				3.727
3	400	30				1.448
4		60	0.15	15		1.024
5	800	30	0.3	30		0.737
6		60				1.024
7		30	4.5	90	15	4.781
8		60				[illegible]
9		50	5	50		9.140
10		60	1.2	30		[illegible]

Fig. 14 Effect of time interval on the berth structure max deflection for different cases of berthing.

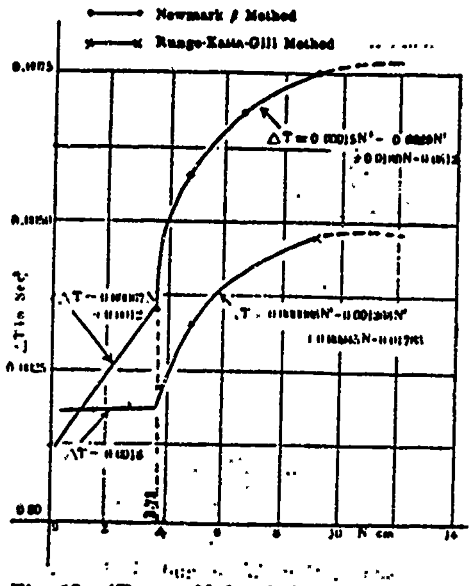


Fig. 15 ΔT, vs. N for 2% error in berth structure max deflection.

$$\Delta t = 0.0018 \quad (\text{sec}) \qquad \cdots\cdots(37)$$
$$N \leq 3.73$$
$$\Delta t = 0.000066N^3 - 0.00137N^2 + 0.0095N - 0.0178 \quad (\text{sec}) \qquad \cdots\cdots(38)$$
$$N \geq 3.73$$

(3) Application to the Non-linear Spring Constant Fenders

Both rubber and retractable fenders possess non-linear relations between the load and the displacement. To apply the preceding equations for selecting the time interval for structures provided with these type of fenders, the load is considered to be applied in small increments associated with the time interval. The procedure for calculation is as follows:

1) The spring constant (K_S) of the fender corresponding to zero displacement is taken from the given load-displacement relationship.
2) The factor N can be calculated and, consequently, the time interval from eqs. (35), (36) or eqs. (37), (38).
3) Substituting in the equations of motion, the displacement of the fender at the end of the interval can be obtained.
4) The spring constant corresponding to this displacement can be calculated as indicated in step 1).
5) Repeating steps 2), 3), and 4) until the allowable fender displacement is reached.

5. APPLICATION

(1) A Case of General Berthing and Fenders of Non-linear Spring Constant

Solved Example

a) Data Given;

A tanker ship of displacement weight $W = 40\,000$ ton

Approaching velocity $V_0 = 15$ cm/sec

Ship characteristics are;

Length$=200$ m Breadth$=25.8$ m

Draft$=10.8$ m

Berthing data;

$r=80$ m $H=3.0$ m $\gamma=50°$ $\alpha=20°$

Assuming, for the first trial, the fender system data as;

$K_S=15$ ton/cm Fender$=2$ pieces of type I, Fig. 17.

b) By applying the formulas included in section 3, the following data was computed;

$M_1=70.69$ ton·sec²/cm

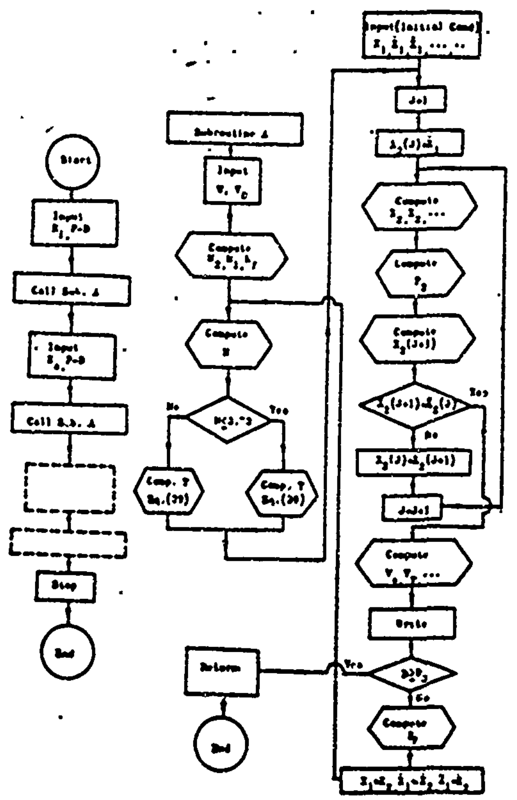


Fig. 16 Flow chart for design procedure by using newmark "β" method.

Table 5

Case No.	Mode of berthing	γ*	r m	H m	K_s ton/cm	Fender Fig. 17	Effective energy E_e	$C=\frac{E_e}{E_0}$ %	$\dot{\theta}$ sec⁻¹	$\dot{\phi}$ sec⁻¹
1	General berthing	50	80	3	15	Type I **	41.79	52.6	0.1112×10^{-2}	0.151×10^{-2}
2	No-rolling	〃	〃	0	15	〃	42.09	53.44		
3	Fender of linear spring constant ***	90	60	0	25	6.25	36.70	56.5		
					$K_T=500$ ton/m $=C_T$ ****		37.60	57.7	Tables 3 and 4 refer. 4)	

* $K_\gamma = K_y$
** The fender stiffness used in these cases is chosen by trials to give $V_c=0.0$ at max. sway.
*** $M_2=130$ ton·sec²/cm, $V_0=10$ cm/sec
**** K_T=the compound stiffness of the fender system$=K_sK_f/(K_s+K_f)$

$M_1=0.7$ ton·sec²/cm
$H_1=\overline{GM}=2.27$ m
$I_{1-1}=54\times10^6$ ton·sec²·cm²
$I_{2-2}=113\times10^7$ ton·sec²·cm²

c) For numerical integration, Newmark β method with $\beta=1/4$ was used. Besides, equations (35) and (36) were applied for selecting the time interval. Calculations were carried out by the digital computer, the flow of computations is shown by the block diagram Fig. 16. Results is included in Table 5.

(2) Besides, for comparing the presented method with other investigators methods, two other examples were tried, Table 5.

6. COMMENTS ON THE RESULTS

(1) Verification of the Developed Method of Analysis

To verify the assumptions presented in establishing the dynamic equations included in section 2, conservation of energy before and after collision is checked from the deduced results.

1) *Conservation of the kinetic energy*

From Table 5 we have at $V_c=0.09\doteqdot0.0$

i) Part of the ship's kinetic energy transmitted to:
Fender system=Effective energy
$E_e=41.79$ (ton/m)

ii) Part of energy induced by the system vibration:
Ship

$$E_{sh}=\frac{1}{2}M_2(\dot{X}_G^2+\dot{Y}_G^2)+\frac{1}{2}I_{2-2}\dot{\theta}^2+\frac{1}{2}I_{1-1}\dot{\phi}^2$$

at $V_0=0.0$ this eq. yields to;

$$E_{sh}=\frac{1}{2}M_2(r\cdot\dot{\theta}+H\cdot\dot{\phi})^2+\frac{1}{2}I_{2-2}\dot{\theta}^2+\frac{1}{2}I_{1-1}\dot{\phi}^2$$

substituting for $\dot{\theta}$ and $\dot{\phi}$ their values, 0.11124×10^{-2} and 0.151×10^{-2} sec⁻¹ respectively, we obtain;

$$E_{sh}=29.96+6.69+0.616=37.26 \quad \text{(ton·m)}$$

Structure

$$E_s=\frac{1}{2}M_1\cdot\dot{S}^2$$
$$=\frac{1}{2}\times0.7\times(-4.659)^2/100=0.077 \quad \text{(ton·m)}$$

$$E_{sh}+E_s=37.26+0.077\doteqdot37.3 \quad \text{(ton·m)}$$

Total energy at $V_c=0.0$ will be;

$$E_e+E_{sh}+E_s=79.09 \quad \text{(ton·m)} \quad \cdots\cdots(39)$$

But the ship's approaching energy is;

$$E_0=\frac{1}{2}\times70.69\times15^2/100=79.520 \quad \text{(ton·m)} \quad \cdots\cdots(40)$$

Eq. (39)$\doteqdot$Eq. (40) which satisfied the conservation of kinetic energy before and collision.

2) *Conservation of the moment of momentum*

i) At time when the ship made the first contact, the moment of momentum is given by;

$$MM_1=M_2\cdot V_0\cdot r\sin\gamma$$
$$=70.69\times15\times8\,000\times0.766/100$$
$$=649.8\times10^3 \quad \text{(ton·sec·m)} \quad \cdots\cdots(41)$$

ii) At time when the ship started to rebound, the moment of momentum is equal to;

$$MM_2=I_{2-2}\dot{\theta}+I_{1-1}\dot{\phi}+M_2\sqrt{\dot{X}_G^2+\dot{Y}_G^2}\cdot r$$
$$=126.10\times10^3+8.15\times10^3$$

$$+527.80\times10^3$$
$$=662.05\times10^3 \quad (\text{ton}\cdot\text{sec}\cdot\text{m}) \qquad \cdots\cdots(42)$$

Eq. (41)=Eq. (42) which satisfied the moment of momentum principles.

(2) Checking of Time Interval Selection Procedure

The accuracy of the deduced results, as seen in the previous paragraph, verified, on one hand, the right procedure of developing the equations of motion, and on the other hand, it supported the author's recommendations for the evaluation of the time interval ΔT presented in section 4. Fig. 18 shows the variation of ΔT through the numerical integration process according to the variation of the fender's stiffness shown by Fig. 17. Any mis-choice of the time interval will lead to large errors, and sometime, leads to unreasonable results as included in Table 4.

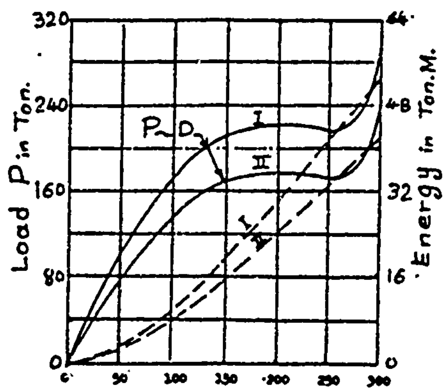


Fig. 17 The load and energy vs. deflection for V600H rubber fender (Tokyo Rubber Dock Fenders).

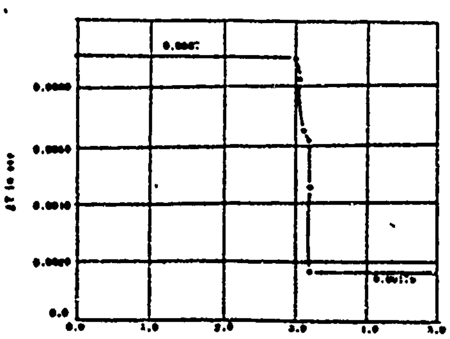


Fig. 18 Variation of time interval ΔT vs. time of berthing.

7. COMPARING THE METHOD WITH OTHER INVESTIGATORS METHOD

i) Case 2, Table 5, in which $H=0.0$, is similar to the case treated by Vasco Costa, *i.e.* swaying and yawing motions are only considered. From the table we have; at $V_C=0.69\div0.0$ cm/sec.

$$E_e=(V_{sx}+V_{sy}+V_f)=(18.40+3.42+20.27)$$
$$=42.09 \quad (\text{ton/m}) \qquad \cdots\cdots(43)$$

Substituting with the given data in the equation given by Vasco Costa, we obtain;

$$E_e=(W)=\frac{1}{2}M_sV_s\frac{k_s^2+r^2\cos\gamma^2}{k^2+r^2}$$
$$=42.15 \quad (\text{ton/m}) \qquad \cdots\cdots(44)$$

ii) Case 3 is similar to the case treated by Hayashi & Shirai, *i.e.* fender of linear spring constant and γ, Fig. 1, equals to 90°.

From i) and ii), if we considered some error due to numerical integration, the author's method will be in agreement with the two special cases treated in references 4) and 5).

8. SUMMARY AND CONCLUSIONS

The foregoing study describes an analytical treatment of the ship berthing problem, based on the dynamic response of the ship and the fennder system during berthing.

The presented analysis covers almost the main factors involving in berthing operations. These are comprised of;

i) The approaching mode of the ship with reference to the face of the berth, designed by α and γ.
ii) The location of the point of contact on the ship's hull, denoted by r and H.
iii) The structure stiffness and the fender stiffness, whether the latter of a linear or non-linear spring constant, beside to the hull stiffness at the point of contact, if available.
iv) The mechanical behaviour of the fender system with respect to the variation of the acting load direction. This is very important in case of rubber-like fenders, as their energy absorbing capacity is a function of the load direction.
v) Consideration of shallow water effect in swaying, yawing and rolling.

For solving the developed equations of motions, recommendations and formulas for estimation and selection of the different parameters, particularly the time interval, are presented.

The solution of these equations will lead to the evaluation of the data required for designing the fender system;

i) Evaluation of the energy absorbed by the fender.

ii) Evaluation of the energy absorbed by the fender structure and its dynamic reaction, magnitude and direction.

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