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Floating Rubber Fender Testing

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This research project evaluated the structural and performance characteristics of two cylindrical rubber fenders under simulated marine vessel loading. The purpose of this research effort is to establish design and performance criteria for vertically floating rubber fenders that can be applied to the specific needs of the Alaska Marine Highway System (AMHS) ferry vessels.

Two foam filled, rubber skinned, cylindrical fenders were fabricated and load tested. The test fenders were 6-feet in diameter by 6-feet long comprised of a 30-inch diameter by 1/2-inch wall steel pipe sleeve wrapped with closed cell polyethylene foam and encapsulated with a 1.25-inch thick skin of rubber reinforced with fiberglass filaments. One test fender contained a foam density of 4 pcf. The other test fender had a foam density of 6 pcf.

Fender testing consisted of the application of a load to the fender body intended to simulate the shape and size of the AMHS vessel sponson that would be in contact with the fender under actual conditions. Load and deflection tests were performed utilizing a calibrated hydraulic press and deflection sensors. Load testing was performed at three locations, 90 degrees apart within the center of fender body. Continuous load, deflection and time data resulting from fender deformation under load was recorded. Cyclic load testing was also conducted on the fender body to determine if multiple load applications would cause material damage.

The load and deflection data was plotted and analyzed to determine load versus deflection curves and the available kinetic energy or load absorption capacity of each test fender. Load versus deflection and a kinetic energy versus deflection curves for each of the fenders are provided in the report. The report further summarizes loading conditions and applicable energy requirements applicable to the use of this type of fender under AMHS service conditions.

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Prescribed by ANSI Std. 239-18 298-102

METRIC (SI*) CONVERSION FACTORS

APPROXIMATE CONVERSIONS TO SI UNITS

Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
in	inches	25.4		mm
ft	feet	0.3048		m
yd	yards	0.914		m
mi	Miles (statute)	1.61		km
		AREA		
in^2	square inches	645.2	millimeters squared	cm^2
ft^2	square feet	0.0929	meters squared	m^2
yd^2	square yards	0.836	meters squared	m^2
mi^2	square miles	2.59	kilometers squared	km^2
ac	acres	0.4046	hectares	ha
		MASS (weight)		
oz	Ounces (avdp)	28.35	grams	g
lb	Pounds (avdp)	0.454	kilograms	kg
T	Short tons (2000 lb)	0.907	megagrams	mg
		VOLUME		
fl oz	fluid ounces (US)	29.57	milliliters	mL
gal	Gallons (liq)	3.785	liters	liters
ft^3	cubic feet	0.0283	meters cubed	m^3
yd^3	cubic yards	0.765	meters cubed	m^3

Note: Volumes greater than 1000 L shall be shown in m^3

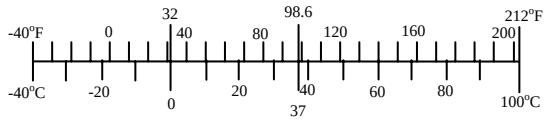
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (°F-32)	Celsius temperature	°C
		ILLUMINATION		
fc	Foot-candles	10.76	lux	lx
fl	foot-lamberts	3.426	candela/m^2	cd/cm^2
		FORCE and PRESSURE or STRESS		
lbf	pound-force	4.45	newtons	N
psi	pound-force per square inch	6.89	kilopascals	kPa

These factors conform to the requirement of FHWA Order 5190.1A *SI is the symbol for the International System of Measurements

APPROXIMATE CONVERSIONS FROM SI UNITS

Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
mm	millimeters	0.039	inches	in
m	meters	3.28	feet	ft
m	meters	1.09	yards	yd
km	kilometers	0.621	Miles (statute)	mi
		AREA		
mm^2	millimeters squared	0.0016	square inches	in^2
m^2	meters squared	10.764	square feet	ft^2
km^2	kilometers squared	0.39	square miles	mi^2
ha	hectares (10,000 m^2)	2.471	acres	ac
		MASS (weight)		
g	grams	0.0353	Ounces (avdp)	oz
kg	kilograms	2.205	Pounds (avdp)	lb
mg	megagrams (1000 kg)	1.103	short tons	T
		VOLUME		
mL	milliliters	0.034	fluid ounces (US)	fl oz
liters	liters	0.264	Gallons (liq)	gal
m^3	meters cubed	35.315	cubic feet	ft^3
m^3	meters cubed	1.308	cubic yards	yd^3

		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 °C+32	Fahrenheit temperature	°F
		ILLUMINATION		
lx	lux	0.0929	foot-candles	fc
cd/cm^2	candela/m^2	0.2919	foot-lamberts	fl
		FORCE and PRESSURE or STRESS		
N	newtons	0.225	pound-force	lbf
kPa	kilopascals	0.145	pound-force per square inch	psi



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Appendix B – Fender Testing Information

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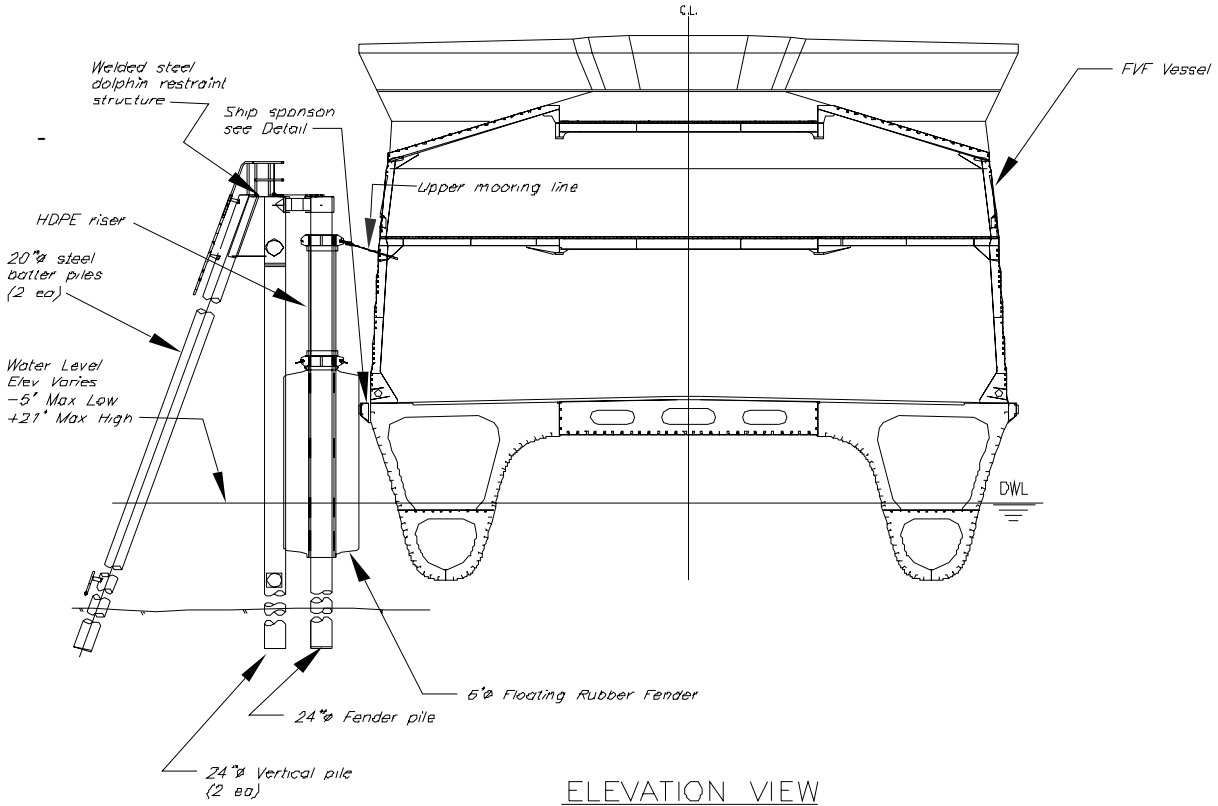
CHAPTER 1 - PROJECT DESCRIPTION AND OBJECTIVES

This research project evaluated the structural and performance characteristics of two cylindrical rubber fenders under simulated marine vessel loading. The purpose of this research effort is to establish design and performance criteria for floating rubber fenders that can be applied to the specific needs of the Alaska Marine Highway System (AMHS) ferry vessels.

Ferry terminal facilities are an important component of the Alaska transportation system. In general, these facilities consist of offshore mooring structures that allow vessels to safely moor while transferring vehicles and pedestrians to shore. Offshore mooring structures have traditionally consisted of fixed fender faces that have limited ability to allow extended periods of unattended berthing due to the large variations in mooring line lengths caused by tidal fluctuations. Over the past few years, DOT&PF has designed and constructed several major ferry terminal improvement projects in support of the new fast ferry vessels and the existing fleet. At several terminal locations, all-tide moorage of the fast ferry vessel is desired without the need for line handling by ship-based personnel. Floating rubber fenders offer a convenient method for allowing the vessel to move with the tidal elevation differences without adjustment of mooring lines. An additional benefit of the rubber fender is the relatively soft berthing surface.

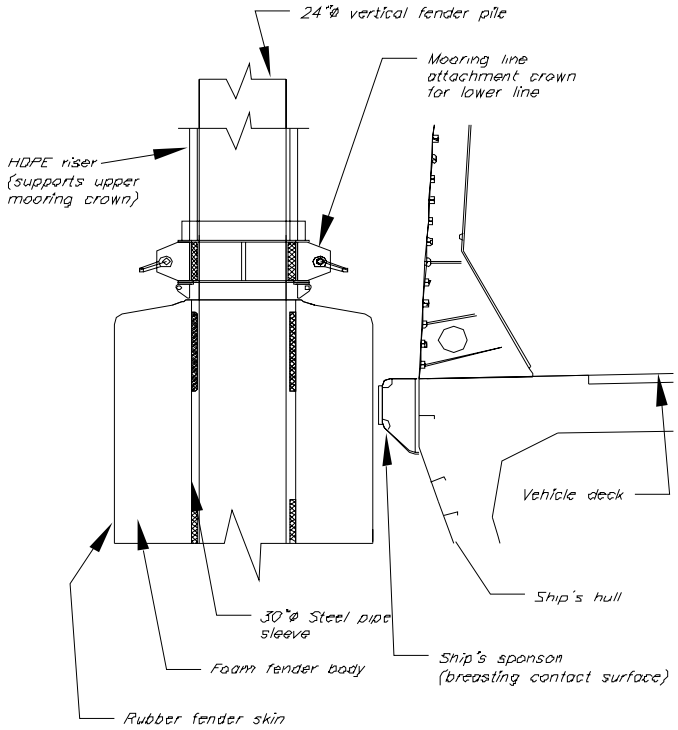
Floating rubber fenders were originally developed as a solid cylindrical shaped unit that was floated in the horizontal position and commonly utilized as a spacer or floating camel between the ship hull and the fender or wharf face. These fenders generally consist of a closed cell, polyethylene foam core covered by a reinforced rubber skin. For our application, the fenders have been modified by adding an inner steel pipe sleeve and floated vertically as indicated on Figure 1. The cylindrical fender is placed over a vertical steel pile and floats up and down with the tide level. This allows the fender to move with the vessel and the mooring lines from the fender to the vessel are maintained at a constant length regardless of tide level.

The introduction of the inner pipe sleeve reduces the overall thickness of the fender section that is available for absorbing berthing energy. The AMHS ferry vessels are also equipped with a narrow rubstrip or sponson that berths against the fender surface. The sponson imparts a load over a small area of the fender. Energy absorption due to ship impact is provided by the compression of the vessel sponson into the fender surface and lateral movement of the fender due to bending of the steel restraint pile. The entire fender surface does not necessarily come in contact with the vessel, nor does the entire fender surface compress during loading. These loading conditions therefore change the performance characteristics that have been previously established by the various fender manufacturers for camel type fenders. Consequently, there is no load or deflection data presently available for the type of loading imposed by the AMHS vessels on a vertical cylindrical shaped floating fender.



ELEVATION VIEW

FLOATING FENDER STRUCTURE/MOORED SHIP



FENDER DETAIL

Figure 1. Drawing. Typical Floating Fender Structure

CHAPTER 2 - FENDER FABRICATION

This research project involved the fabrication of two foam filled, rubber skinned, cylindrical fenders. The test fenders were 6-feet in diameter by 6-feet long comprised of a 30-inch diameter by 1/2-inch wall steel pipe sleeve wrapped with closed cell polyethylene foam and encapsulated with a 1.25-inch thick skin of rubber reinforced with fiberglass filaments. One fender contained a foam density of 4 pcf. The other fender had a foam density of 6 pcf. The size of these test fenders closely approximate the actual diameter utilized in practice; however, the length is much shorter due to the cost and other complexities in manufacturing a longer fender for test purposes. Actual fenders used on several recent installations are on the order of 15-22 feet in length depending on location and vessel use. The fenders were manufactured by Urethane Products Corporation located in Bellflower, CA (Los Angeles area). Fabrication drawings and a summary of pertinent fender specifications and material descriptions that were incorporated into the test fenders are contained in Appendix A.

Fabrication of the test fenders consisted of standard techniques used in the manufacture of full size fenders. The steel pipe sleeve is wrapped with the polyethylene foam that is provided in sheets of about ½-inch in thickness. Each layer of foam is heat bonded to the previous layer using hot-air techniques. After the desired diameter is reached by wrapping successive layers of foam sheets, the exterior surface of the foam was shaped on a mechanical lathe to the specified outside diameter and the ends were formed into tapered shoulders as noted on the fabrication drawing contained in Appendix A. The fender skin was then applied to the exterior of the foam surface utilizing spray-on application methods. The fender is continuously rotated and 1-2 mills of polyurethane is sprayed onto the fender surface with each rotation. As the fender rotates and is being sprayed with rubber, a loom type device spools fiberglass filaments into the rubber coating in successive layers. The filaments are wrapped using a continuous helix pattern. The result is a very durable and flexible reinforced rubber skin. The following photographs illustrate the various phases of the fender fabrication process.

The fabrication methods and materials for the test fenders were observed and inspected by DOT&PF personnel on 1/20/04.

Figure 2. Photo. UPC Production Facility.

Figure 3. Photo. Applying first layer of foam to test fender.

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Figure 4. Photo. Partially wrapped test fender.

Figure 5. Photo. Typical foam heat bonding process.

Figure 6. Photo. Other fenders under production.

Figure 7. Photo. Spray application of fender skin to test fender.

Figure 8. Photo. Fender skin reinforcement filaments.

Figure 9. Photo. Fender skin application.

Figure 10. Photo. Finished test fender.

Figure 11. Photo. Finished test fender.

CHAPTER 3 - FENDER TESTING

Fender testing consisted of the fabrication and placement of structural steel supports and a load application beam that was intended to simulate the AMHS vessel sponson loads. Load testing was performed by Coordinated Equipment Company, Wilmington, CA on 2/25/04 and 2/26/04. The following photographs show the fender testing equipment that was utilized. The fender body was suitably supported through the internal steel pipe sleeve with a 16-inch diameter by 4-inch thick wall steel pipe supported on the ends with concrete blocks. The load application was distributed over a 1-foot wide contact area within the center of the fender body utilizing a wide flange steel beam. The load application and related fender support members closely resemble the loading conditions that the fenders are intended to be subjected to under vessel service conditions. Load and deflection tests were performed utilizing a calibrated hydraulic press and deflection sensors attached to string-pot lines affixed to both ends of the steel beam. Continuous load, deflection and time data resulting from fender deformation under load was recorded on a data logger connected to a lap-top computer.

Figure 12. Photo. Fender setup prior to testing.

Figure 13. Photo. Fender setup prior to testing.

Figure 14. Photo. Hydraulic pump and pressure gage.

Figure 15. Photo. Test on 6 pcf fender in progress.

Figure 16. Photo. Full deflection on 6 pcf fender.

Figure 17. Photo. Full deflection on 4 pcf fender.

Figure 18. Photo. Full deflection on 4 pcf fender.

Figure 19. Photo. Fender rebound directly after test on 4 pcf fender.

Figure 20. Photo. Fender rebound after 12 hours on 6 pcf fender.

Figure 21. Photo. Sample cut from 6 pcf fender following testing.

Testing was generally performed as outlined below:

1. Apply gradually increasing external loads (hydraulic pressure) sufficient to achieve a total deflection of about 60 percent of the thickness of the fender body while acquiring

corresponding deflection and load values. Hydraulic pressure, deflection and time were recorded at one-second intervals. The fender was then rotated 120 degrees and a new load test was performed. A total of three load-deflection tests were performed on each fender. Loads were determined by measuring fluid pressure on the ram of the hydraulic cylinder. The cylinder was previously tested to determine the cross sectional area of the ram. The fluid pressure multiplied by the area of the ram determines the force applied to the fender. A copy of the hydraulic cylinder test data is contained in Appendix B.

2. Apply repetitive loads sufficient to achieve approximately 60 percent deflection of the fender body at the remaining undisturbed fender body location. About 20 load applications were applied and the fender was visually examined for evidence of permanent deformation and/or damage characteristics of rubber skin and foam body as a result of cyclic loads.
3. After testing, cross-sectional samples were cut from the fender surface at the location of the cyclic test load application in order to determine if there was any visible evidence of material deformation or damage.

The testing methods were structured to provide estimates of allowable loads and related absorption energy that the fender may be capable of while in service. All testing on this project was monitored in the presence of DOT&PF personnel. Load and deflection testing will allow designers to estimate allowable berthing energy and associated impact loads for the design of future fender installations. The cyclic load tests provide an indication of suitability of fender materials to withstand damage from repeated vessel contact.

CHAPTER 4 - TEST OBSERVATIONS AND DATA

The testing company had numerous startup problems upon arriving at the test site on February 25, 2004 including set up of the test and getting the computer data logging equipment to function. The contractor was eventually directed to perform one test on the 6 pcf fender and record the data by hand. Hand recorded data methods proved to be cumbersome and it was difficult to record sufficient data points. It was eventually determined that the contractor would continue to work on the automated data recording equipment and the tests would be resumed on the next day. This initial test did prove to be of value in determining how the test should be carried out. This initial test revealed the highly elastic nature of the rubber fender resulting in a decreasing load over time at constant deflection. A load-holding period of 10 minutes at full deflection was established for subsequent tests to determine the load relaxation characteristics over time.

By the morning of February 26, the contractor had fixed the recording system and testing was resumed. Both the 4 pcf and 6 pcf foam fenders were tested on February 26. The fender deformation was also measured at the test location performed the previous day. The fender surface had rebounded to nearly the original diameter.

The testing contractor submitted the test results in an Excel spreadsheet format. The data contains test values for the load in pounds, ram position/fender deflection in inches and time at one-second intervals. Appendix C contains an electronic copy of the test data in Excel spreadsheet format. The test data is labeled as Tests 1.2 and 1.3 for the 6 pcf foam fender and Tests 2.1, 2.2 and 2.3 for the 4 pcf fender. For unknown reasons only two data sets were ultimately submitted for the 6 pcf foam. Subsequent efforts to obtain the data set for Test 1.1 were unsuccessful. It is not known whether the contractor did not properly record this data or otherwise subsequently lost the data.

Figures 22 through 32 show plots of the load versus deflection and load versus time data for the various tests. Each test shows that the load gradually increases over time to a maximum value at initial full deflection. The load deflection data, up to the initial full deflection, exhibits a nearly linear, or elastic relationship i.e. $F=kx$ where F is the applied load, k is the slope of the line and x is deflection. The load/deflection relationship for each test was then determined by a line of best fit using linear regression analysis (method of least squares). The line of best fit and resulting equation of the load versus deflection line is indicated on each of the plots using a Y intercept of zero.
Once full deflection was obtained, the load values rapidly decrease and eventually stabilize over time. The load reduction is a result of the deformation within the foam fender body. After a time period of about 5 minutes, the load stabilizes and the decrease in load continues at a much slower rate. All tests were ended after approximately 10 minutes once the load value stabilized.

It should be noted that the data set for Test 1.2 contained information or data points that were not consistent with field observations (reference Figure 22). After reaching full deflection of about 14.5-inches, the data file indicates the load increasing another 30,000 lbs over a time period of about 4-minutes. This was not observed in the record of field notes. As such, the load data beyond the initial 101,414 lbs is attributed to a data collection error and as such was ignored for calculation purposes. The tests for the 6 pcf foam fender (1.2 and 1.3) also do not have as close

of correlation as the tests for the 4 pcf foam fender. The data inconsistencies in test 1.2 and the lack of data for test 1.1 may have something to do with this; however, based on field observations, the data that was obtained for tests 1.2 and 1.3 is considered acceptable for purposes of this project.

After the three load/deflection tests were performed, the fender was rotated 90 degrees to the remaining undisturbed surface and cyclic loads were applied. The cyclic loads consisted of about 20 repetitions of load application to a deflection of 13 to 14-inches over a time span of about 20-25 minutes. The fenders were then visually inspected for wear or other signs of external damage. No damage or other visible evidence of distress were noted following the cyclic tests. A sample of the fender skin and foam core was also cut and examined. The cut samples also exhibited no signs of damage or other material failure. It was very difficult to cut samples from the fender. In general, the fenders appear to be very resilient and hard-wearing.

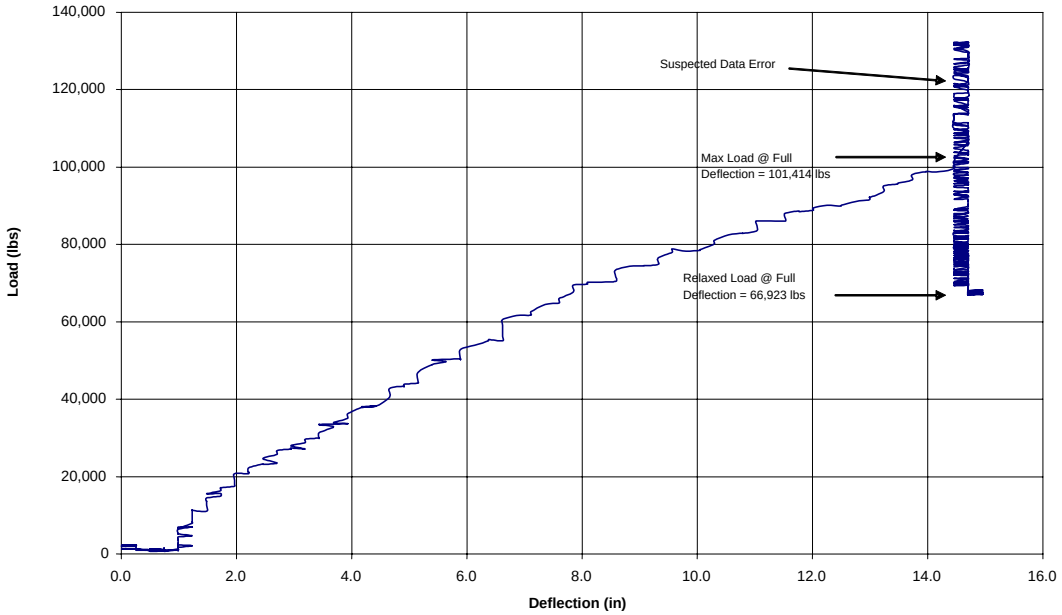


Figure 22.Graph. 6 pcf Foam Fender - Test 1.2 - Load vs Deflection (All Data to Time of Load Removal).

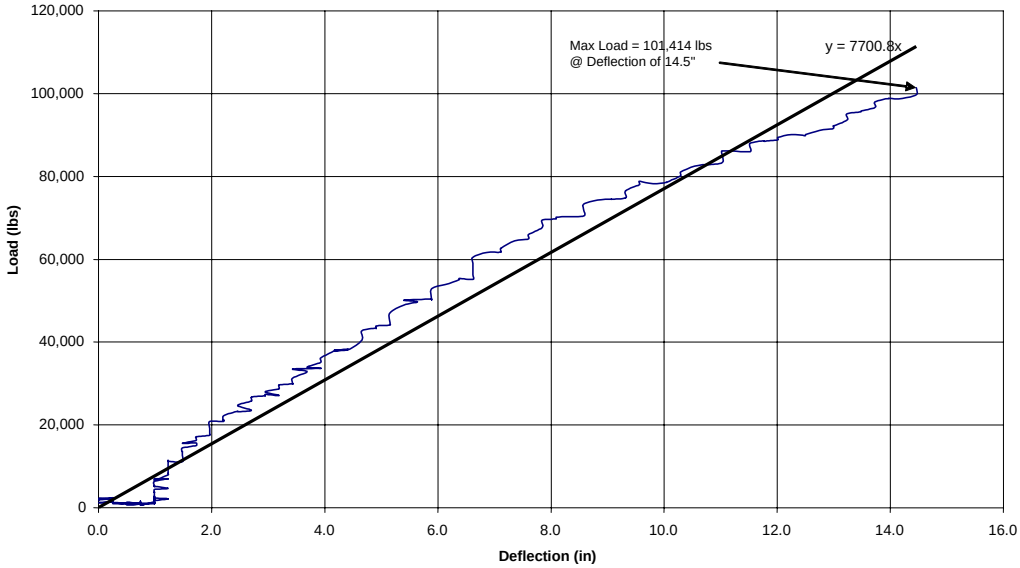


Figure 23. Graph. 6 pcf Foam Fender - Test 1.2 Load vs Deflection - (Data to Initial Max Deflection of 14.5").

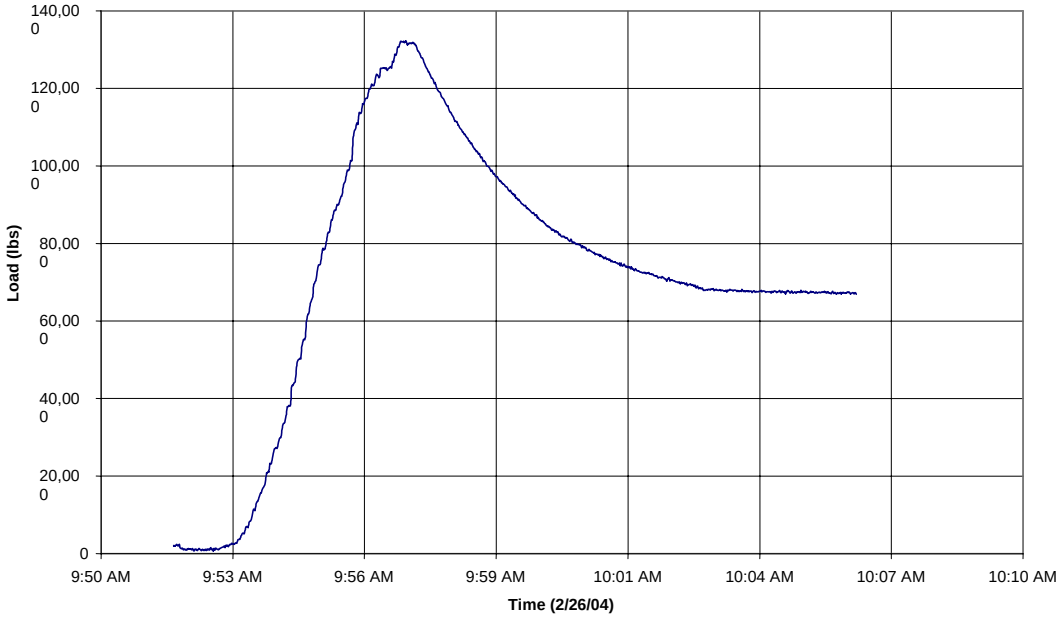


Figure 24. Graph. 6 pcf Foam Fender - Test 1.2 - Load vs Time - (Data to Time of Load Removal).

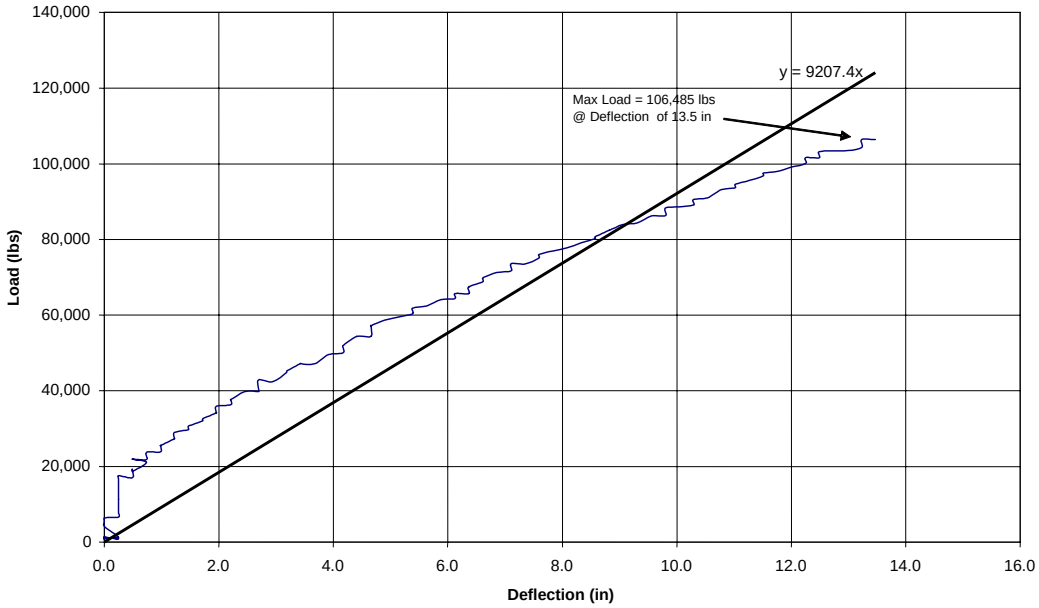


Figure 25. Graph. 6 pcf Foam Fender - Test 1.3 - Load vs Deflection - (Data to Initial Max Deflection of 13.5").

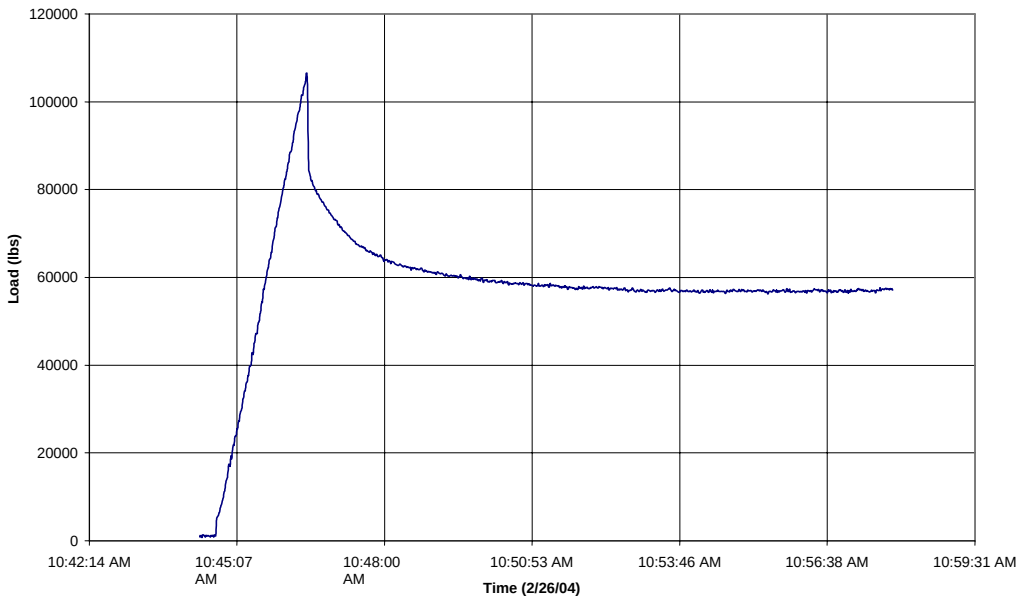


Figure 26.Graph. Test 1.3 - 6 pcf Foam Fender - Load vs Time.

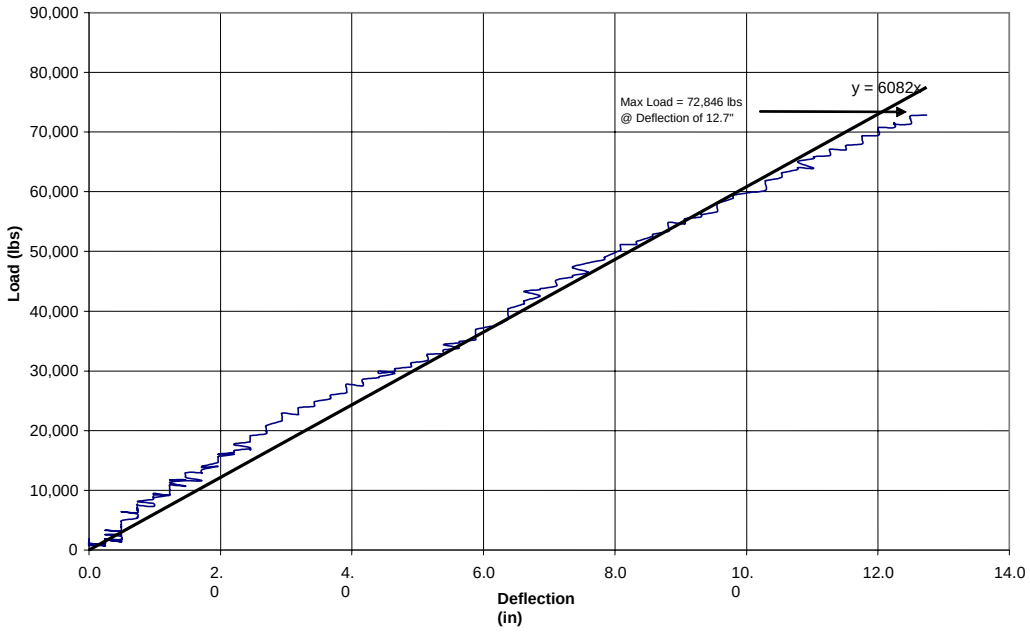


Figure 27.Graph. Test 2.1 - 4 pcf Foam - Load vs Deflection - (Data to Initial Max Deflection of 12.7").

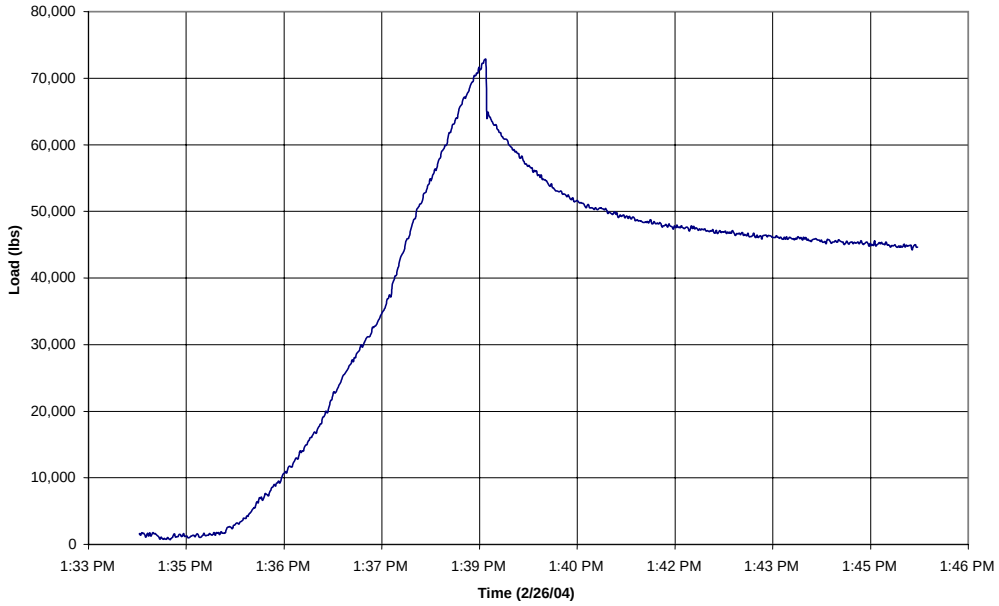


Figure 28. Graph. Test 2.1 - 4 pcf Foam - Load vs Time.

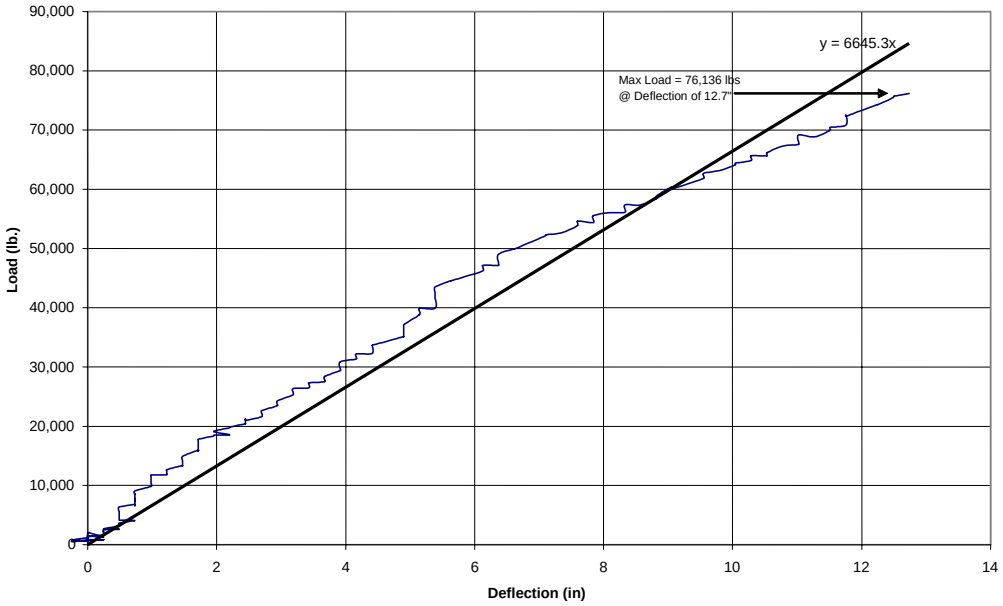


Figure 29.Graph. Test 2.2 - 4 pcf Foam - Load vs Deflection - (Data to Initial Max Deflection of 12.7".

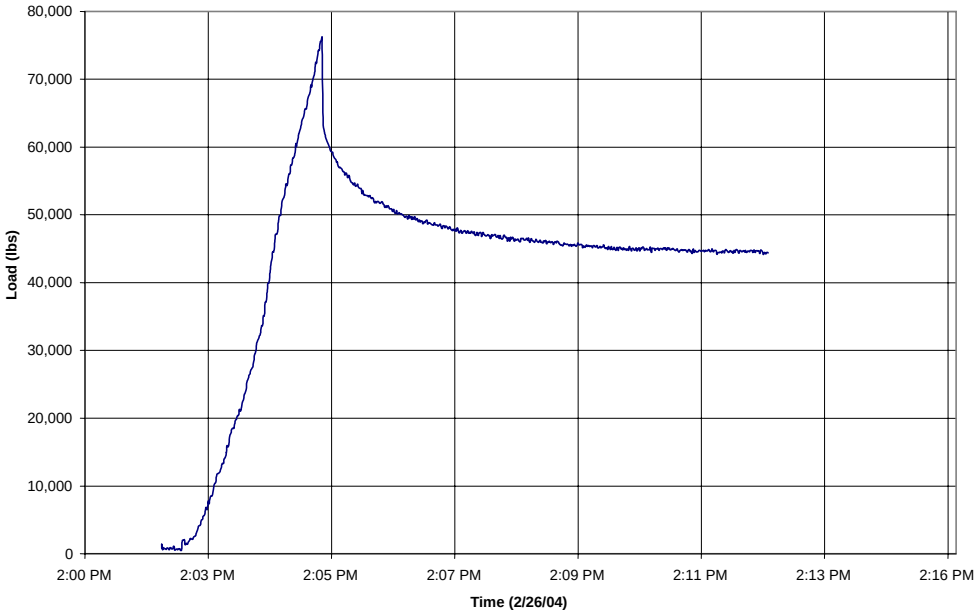


Figure 30.Graph. Test 2.2 - 4 pcf Foam - Load vs Time.

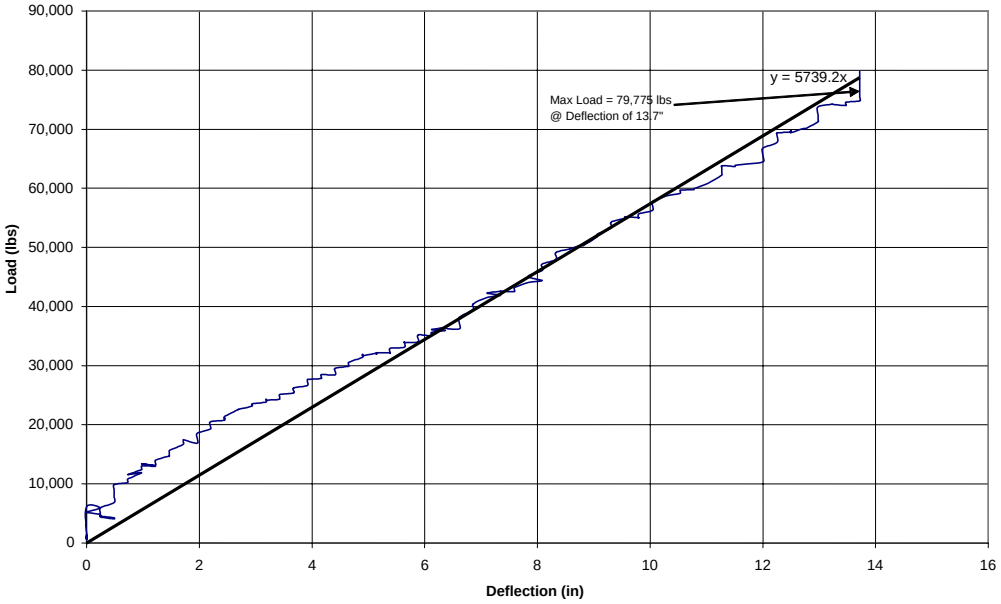


Figure 31. Graph. Test 2.3 - 4 pcf Foam - Load vs Deflection - (Data to Initial Max Deflection of 13.7").

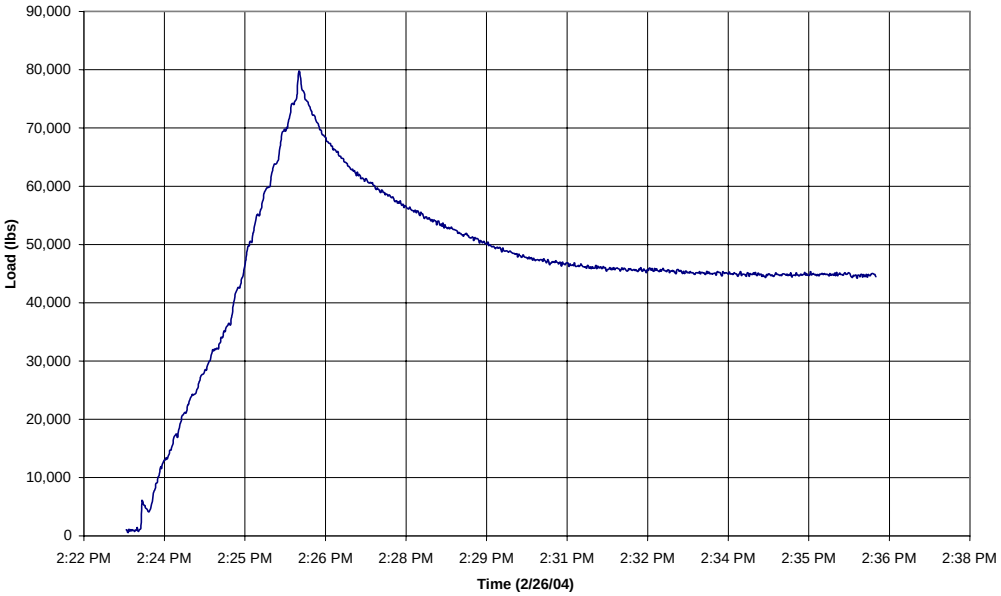


Figure 32. Graph. Test 2.3 - 4 pcf Fender - Load vs Time.

CHAPTER 5 - DETERMINATION OF FENDER CAPACITY

The primary goals of this fender research project were to determine load versus deflection curves and an estimate of the kinetic energy absorption capacity of the test fenders under simulated vessel loads. Fendering systems are essentially designed to withstand, or otherwise absorb the kinetic energy that results from a moving load as a result of ship impact. A short review of fendering principles and determination of kinetic energy design parameters is provided below in order to establish the basis for determining the kinetic energy capability of the test fenders.

The theoretical kinetic energy (KE) of a moving object is represented as:

$$KE = \frac{1}{2} MV^2 = WV_n^2/2g$$

Where W is the loaded displacement weight of the vessel and V_n is the berthing velocity in a direction normal to the structure. V_n is determined by engineering judgement and other factors. It depends on the forward speed of the vessel and the assumed berthing angle to the fender structure at impact. In practice, there are several other factors in the above energy equation and the design vessel berthing energy is often determined by utilizing a berthing coefficient (C_b).

A full explanation of the various berthing coefficients is beyond the scope of this project; however, the energy equation is most commonly expressed as:

$$KE = C_b\, WV_n^2/2g$$

The berthing coefficient takes into account various factors such as the shape and type of vessel, dock structure and fendering system. C_b is generally on the order of 0.5 to 1.0. The general concept for the design of impact forces is that the fender system should absorb the berthing energy of the moving vessel without damage to the vessel and the dock or mooring structure. Fender absorption elements are thus selected to absorb the estimated berthing energy of the design vessel. Another important design parameter is the reaction load imparted to the fender support structure as a result of the energy absorption of the fender.

AMHS vessels vary greatly in length and displacement weight. Table 1 illustrates various design berthing energies that are commonly used on DOT&PF mooring and fender structures. As noted in Table 1, the berthing energy can vary widely depending on the vessel, the selected berthing velocity and whether the ship is impacting along the side of the ship or on the end or stern of the ship.

Table 1. AMHS Vessels and Representative Berthing Energies.

Cb, End Fenders = 0.63

Cb, Side Fenders = 0.90

Vessel	LOA (ft)	Beam (ft)	Ship Weight (Long Tons)	Ship Weight (kips)	End Fenders Berthing Velocity Vn (ft/sec)	Side Fenders Berthing Velocity Vn (ft/sec)	End Fenders Berthing Energy (ft-kips)	Side Fenders Berthing Energy (ft-kips)
MV Kennicott	382	85	7,503	16,507	1.50	1.00	519	161
MV Mat / Mal	408	74	5,553	12,217	1.50	1.00	384	120
MV Columbia	418	85	7,683	16,903	1.50	1.00	531	165
MV Taku	352	74	4,284	9,425	1.50	1.00	296	92
MV Aurora/Leconte	235	57	2,150	4,730	2.00	1.50	264	104
MV Fairweather (FVF)	235	59	779	1,714	2.50	2.00	150	67
MV Lituya	180		713	1,569	2.00	1.50	88	35
MV Prince of Wales (IFA)	200	53	1,934	4,255	2.00	1.50	238	94
MV Oral Freeman	116		318	700	2.50	2.00	61	27

The kinetic energy capacity of a fender absorption element can be found by determining the area under the load versus deflection curve. If the load versus deflection curve is linear, or elastic, the following relationship can be used to determine the available kinetic energy of a fender:

$$KE = \int k(x)dx$$

Integrating from 0 to x, the energy equation can be written as:

$$KE = kx^2/2$$

and noting that:

F = kx (force moved over distance x)
F = load or force imparted to fender
k = spring constant or slope of load versus deflection line
x = deflection of fender

Substituting k=F/x, the kinetic energy can be determined as shown below:

$$KE = Fx/2$$

The above assumptions and derivation assumes that the fender behaves in an elastic manner and has a reaction force of F = kx. The data obtained from the load versus deflection testing performed under this project is essentially linear up to the time of initial maximum deflection as evidenced by the plots presented in Figures 22 through 32. After the initial maximum deflection, the curve becomes non-linear as a result of load relaxation over time. However, the

designer is not particularly concerned with the non-linear portion of the curve since the fender would be designed to absorb the kinetic energy of the vessel up to the time of the maximum allowable deflection or within the linear portion of the load/deflection curve.

Figures 33 and 34 show combined plots and the associated equations for the line for load and kinetic energy versus deflection. The load versus deflection line is the average of the lines of best fit obtained for the individual tests (the slope of each line for the individual 4 pcf and 6 pcf fender tests were averaged together). The average kinetic energy of the fenders was then determined by the formula derived above, KE = Fx/2. The curves are also extrapolated to a total deflection of 16-inches or approximately 67 percent of the net cylindrical fender thickness. It should be noted that most elastomeric fender materials are designed for a maximum deflection of 50-70 percent of the original thickness.

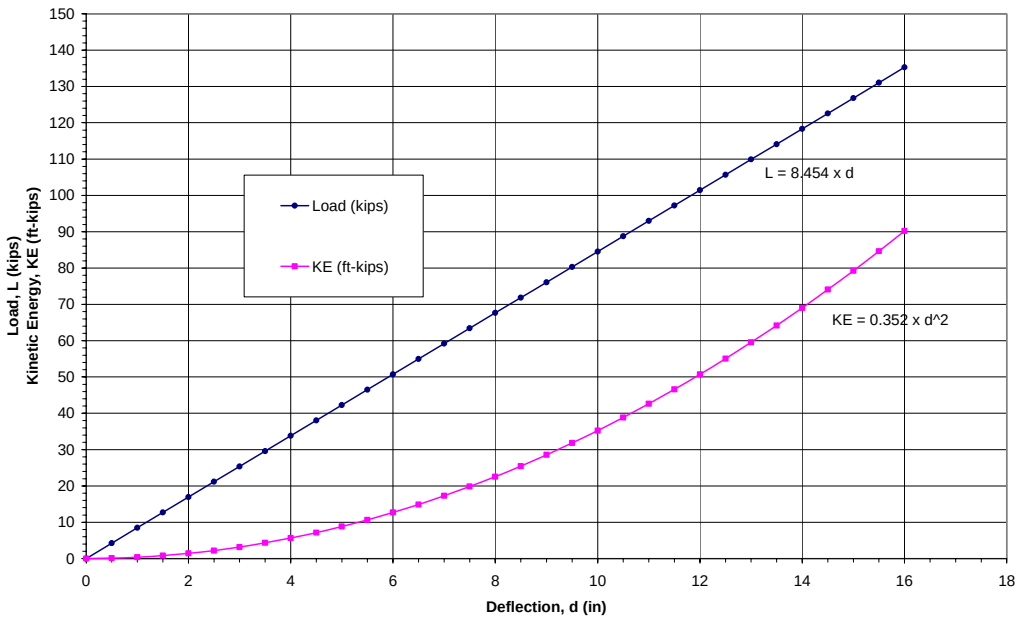


Figure 33. Graph. 6 pcf Foam Fender - Average KE & Load vs Deflection

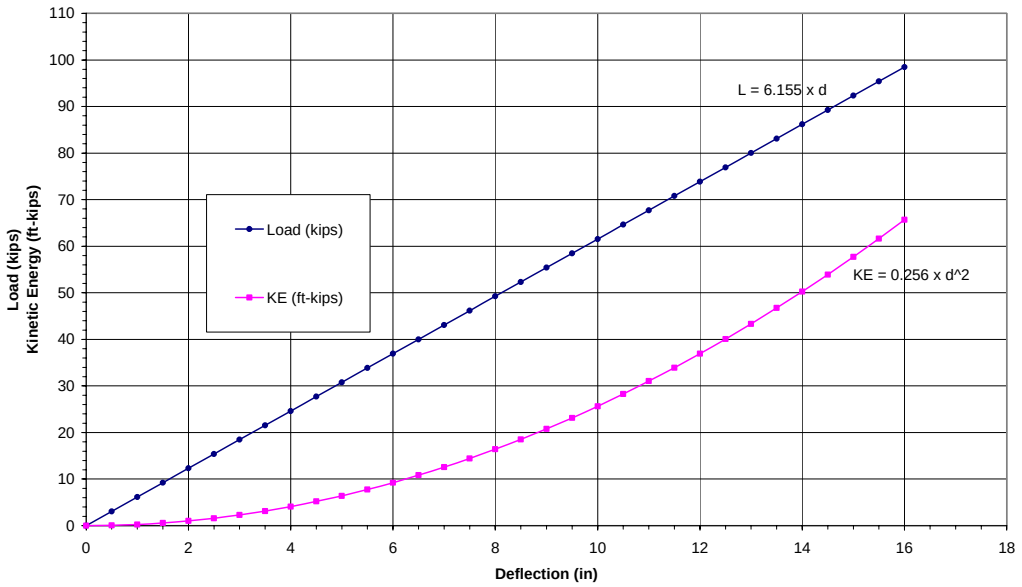


Figure 34. Graph. 4 pcf Foam - Average KE & Load vs Deflection.

CHAPTER 6 - SUMMARY AND CONCLUSIONS

As evidenced by the average load and energy curves shown on Figures 33 and 34, the 6 pcf foam density fender has an estimated maximum kinetic energy of 90 ft-kips and reaction force of 135 kips. The 4 pcf foam density fender has an estimated maximum kinetic energy of about 65 ft-kips and associated reaction force of 98 kips. The averaged curves can be directly used for design purposes. The formulas for each of the curves are shown on Figures 33 and 34 for direct computation purposes. The designer should note that the estimated kinetic energy capacities of these fenders presented on Figures 33 and 34 are based on deflection of the fender body only. The tests were performed with the fender body rigidly supported through the internal steel pipe sleeve. In practice, the floating fenders presently being utilized also have the ability to move by bending and deflection of the internal steel pile thus resulting in additional energy absorption capabilities.

While the individual test values do not closely correlate and the test data is somewhat limited, the averaged test data is considered to provide a good indication of the capacity of these type of fenders under conditions similar to what will be experienced from impact or contact by AMHS vessels. Additional testing would be needed to more closely correlate results or to determine other statistical deviations.

The effect of the density of the foam is not entirely conclusive; however, from examination of the average energy and deflection curves, the 6 pcf foam density fender has roughly a 35 percent greater capacity in both load and kinetic energy than the 4 pcf foam density fender. This gives the designer some indication of the relative stiffness or energy capacity provided by the different foam densities. The foam body materials for these fenders can be provided in densities ranging from about 2 pcf to 10 pcf.

The relative importance or affect of fender diameter and length is also not established as a result of this testing effort. A larger diameter fender would likely provide similar test results up to the deflection distances tested under this project; however, the larger diameter fender would allow further deformation in the elastic range and thus higher energy and reaction loads. It should also be noted that the vessel sponson only extends past the hull of the ship approximately 12-16 inches. Further load compression into the fender beyond the depth of the sponson structure would allow contact of the fender surface with the ship hull. In which case, the capacity of the fender and resulting reaction load would likely be much greater. For fender design purposes, the maximum deflection of this type of fender can be roughly taken as the depth of the sponson. The fender length may also have an effect on the test data. The fenders tested under this project were only 6-feet in length. Actual floating fenders are on the order of 15-22 feet in length. The relatively short fenders tested under this project did exhibit evidence of extensive bulging or pressure at the hemispherical ends as noted in the test photographs. The containment of the energy absorbing foam, provided by the rubber skin at the fender ends, may have resulted in somewhat higher loads and resulting energy capacity than would be provided by a longer fender.

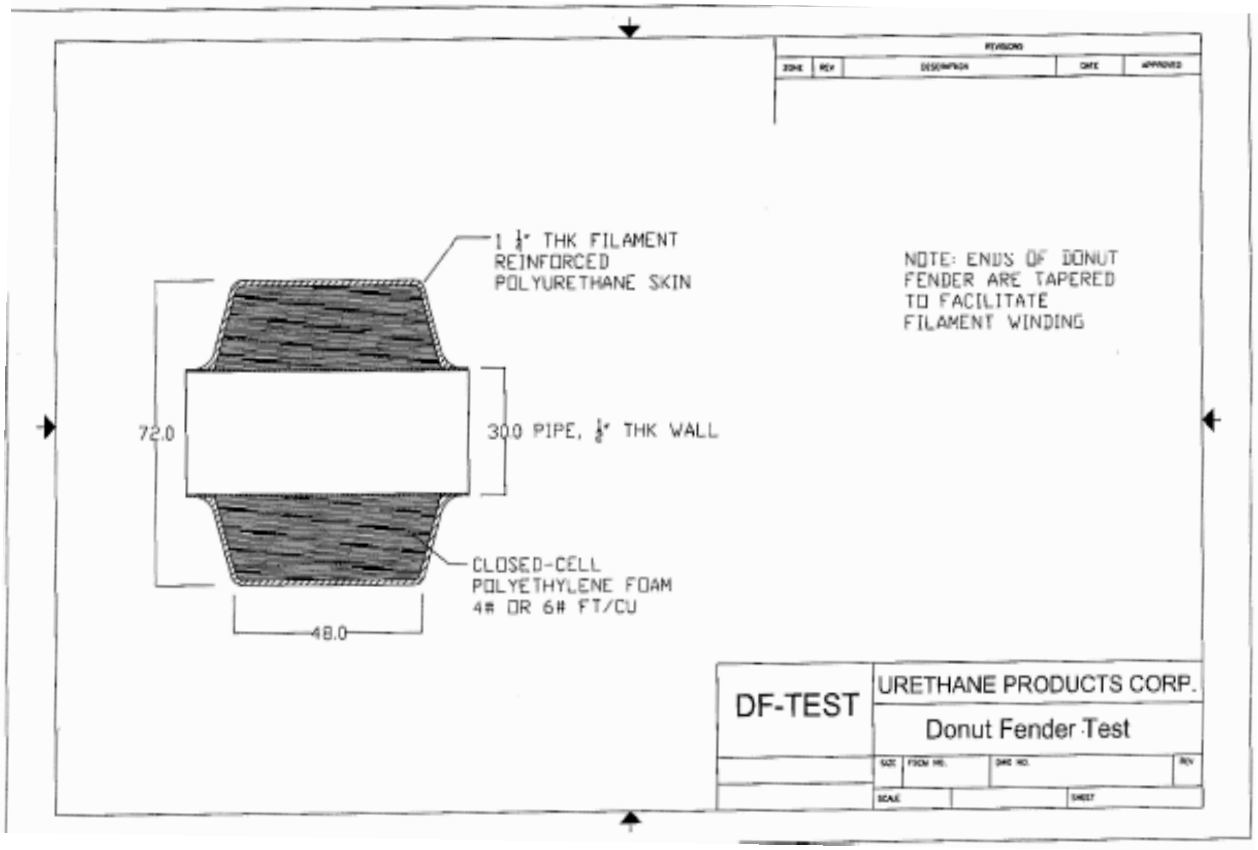
The test data suggests that the individual fenders are limited to relatively light berthing loads as evidenced by the various vessel berthing energies presented in Table 1. However, the fender

structures that have been presently designed allow further energy absorption through deflection of the driven steel support pile. The fender backup structures also provide additional energy or load reserves since the floating fender can also be compressed onto the steel piles that are placed behind the fender. (Reference Figure 1). A soft fender is actually preferred for berthing of the new fast ferry vessels since these vessels are of aluminum construction. Service observations for recently completed floating fender installations at Ketchikan, Metlakatla, Auke Bay and Cordova, and the test data obtained under this project, suggests that the vertical floating cylindrical fender appears to be well suited for the smaller AMHS vessels and the fast ferries.

In summary, the averaged data presented on Figures 33 and 34 can be used to estimate the initial load and kinetic energy capacity of the fender body for deflection values up to the depth of the vessel sponson. If additional energy capacity is required, the designer must incorporate other measures of load absorption such as allowing the fender body to contact the ship hull, examination of the bending of the internal steel fender restraint pile and other structural factors

APPENDIX A

RUBBER FENDER FABRICATION DRAWINGS & MATERIAL SPECIFICATIONS



Urethane Products Corporation

Standard Specification for *Marine Guard* ™ Floating Foam-Filled Donut Fenders Standard Capacity Model DF 6X16.5

PART 1 - GENERAL

DESCRIPTION: The specification in this section covers the provision of a foam filled donut marine fender, to be used for the protection of vessels and wharf from damage due to vessel maneuvers where vessel-to-vessel or vessel-to-wharf contact may occur.

APPLICABLE DOCUMENTS: The following documents for the issue in effect on the date of this Invitation for Bids or Request for Proposal form a part of this specification to the extent specified herein.

Standards:	FEDERAL	FED-STD-595	-Color
	MILITARY	MIL-STD-129	-Marking for shipment and storage.

American Society for Testing and Materials (ASTM):

A-36	Structural Steel
A-123	Zinc (Hot Galvanized) Coatings on Products fabricated from Rolled, pressed and forged shapes, bar and strips
A-153	Zinc Coating (Hot Dip) on Iron and Steel Hardware
D-412	Rubber Properties in tension, Test Methods for
D-624	Rubber Property Tear Resistance, Test for
D-1052	Measuring Rubber Deterioration, Method for (Cut Growth using Ross Flexing Apparatus)
D-3575	Flexible Cellular Materials Made from Olefin Polymers
D-1630	Rubber Property Abrasion Resistance, Test for (NBS Abrader)
D-2240	Rubber Property Durometer Hardness, Test for

DEFINITIONS: The definitions in this section supplement terms defined in other parts of this specification.

Rated Energy Absorption: The energy absorption capacity of the fender at the rated deflection

Rated Reaction Force: The reaction force of the fender at the rated deflection.

Rated Deflection: The distance that the resilient foam layer is designed to deflect, measured radially inward towards the axis of the fender, on one side of the fender only.

SYSTEM REQUIREMENTS: Dimensions of the foam filled donut marine fender shall be as specified and indicated to provide the required performance and to avoid jamming or excessive cocking on the pile during berthing operations. The fender shall be free floating with a cylindrical body (axis vertical) and shall be free to rotate around the supporting pile, and to move up and down along its for different water levels. The fender consists of the following parts:

INNER BEARING SURFACE: The bearing surfaces between the donut fender and the pile shall permit the fender to rise and fall with the water level, and rotate around the pile without binding. The bearing materials shall be fixed to, but removable from the internal core. When the fender is centered around the pile, the clearance between the bearing surfaces and that pile shall be approximately 1 inch.

INTERNAL CORE: A rigid cylindrical internal core shall be provided, over which the resilient foam layer is affixed. There shall be nothing on the core that would prevent deflection of the resilient foam layer to less than full deflection. The inner surface of the core, to which the bearing surfaces are attached, shall be a steel plate or pipe. All exposed steel surfaces shall be suitably protected against marine environment by hot dipped galvanizing in accordance to ASTM A-123 or ASTM A-153. Lifting eyes shall be provided on both ends of the fender.

RESILIENT FOAM LAYER: An annular layer of resilient closed-cell foam shall be fixed to the internal core. This layer shall be continuous form the internal core surface to the outer skin of the fender.

FENDER SKIN: The exterior surface of the fender (excluding steel parts) shall be a weather-resistant PTMEG, reinforced with continuous nylon filaments, polyurethane elastomer.

PART 2 - CONSTRUCTION

BEARING MATERIAL: The bearing material shall be ultra high molecular weight (UHMW) polyethylene. It shall be resistant to abrasion, oil, seawater and weather. This UHMW shall be at least 1.5 inches thick. It shall have the following nominal properties:

Specific Gravity (ASTM D-792)	: 0.9 – 1.0
Tensile Strength (ASTM D-638)	: 3500 psi (at speed of 2 in/min)
Elongation (ASTM D-638)	: 400% minimum (at speed of 2 in/min)
Friction Coefficient	: 0.2 maximum, measured by sliding of sample piece on slope of steel plate, plated with nickel-chromium.

FOAM CORE: The energy absorbing foam core shall be a resilient closed-cell, cross-linked polyethylene foam with the following properties. The foam core shall be of heat-laminated construction. Use of chipped or granulated particulate foam is not acceptable. The test shown below all form a part of ASTM-D-3575.

Density	: 4.0 lb./ft. to 6.0 lb./ft.
Tensile Strength	: 80-psi minimum
Elongation (Ultimate)	: 170 % minimum
Water Absorption	: less than 0.05 lb./ft² (1.0 kg/m²) of cut surface

FENDER SKIN: The fender skin shall be constructed of elastomer and filament reinforcing as specified. Separate filament reinforcing wraps shall be applied as specified under Filament Wrap. The filament wraps shall be evenly distributed in the inner 75% to 90% of the coating thickness. The outer 10% to 25% of elastomer shall have no filament reinforcement. The elastomer and filaments shall be applied in a continuous manner to assure adhesion between the various layers. There shall be a least one wrap for every 0.116 in (2.9 mm) of skin thickness (rounded to the nearest complete wrap). The fender skin shall extend along the entire longitudinal length of the fender, over the ends of the resilient foam layer, and onto the steel portion of the internal core. The fender skin shall be bonded securely to the steel of the internal core.

ELASTOMER: The elastomer used in the fender skin shall be 2,4-toluene diisocyanate, polytetramethylene ether glycol, and an aromatic amine based urethane elastomer, with the following unreinforced properties:

	Elastomer:
Shore A. Hardness (ASTM D-2240)	: 75 to 95
Tensile Strength (ASTM D-412)	: 2500-psi minimum
Elongation (Ultimate) (ASTM D-412)	: 300 % Minimum
Tear Strength (ASTM D-624)	: 200-lb./in minimum
Flex Life (Ross) (ASTM D-1052)	: 200,000 cycle's minimum
Abrasion Resistance (NBS) (ASTM D-1630)	: 1,500 minimum

FILAMENT WRAP: Construct each filament-reinforcing wrap of continuous filaments applied in a helical pattern, at a helix angle of 45° to 60° to the longitudinal axis of the fender. A wrap shall consist of two such filament helixes of equal but opposing helix angles. The spacing between the filaments in the same helix shall be no more than 0.125 in (3.2 mm), measured in a direction parallel to the longitudinal axis of the fender. Each wrap shall extend along the entire longitudinal axis of the fender and shall also encircle and encase the fender, for the purpose of distributing impact loads over the surface and conducting the forces of impact loads to the pipe core.

FILAMENT REINFORCEMENT: The reinforcing filaments in the outer skin shall be nylon tire cord of 2,520 denier (0.280 g/m) weight with the following properties:

Breaking Strength	: 52-lb.
Elongation (Ultimate)	: 16%

REINFORCED SKIN PROPERTIES: The reinforced skin, when constructed in accordance with this specification, shall exhibit the following properties when tested in accordance with the guidelines below:

Tensile Strength: When a full-thickness specimen of the skin is tested in tension, with the longitudinal direction of pull aligned with the filament direction (in one helix direction), the specimen shall have the following breaking strength

: 10,000-psi minimum

Elongation (Ultimate): In the test for Tensile Strength, the specimen shall have the following Ultimate elongation at break:

18% minimum

Tear Strength: When a full-thickness specimen of the skin is tested in tear, with the specimen Shaped like Die C in ASTM D-624, and with the direction of pull aligned with the filament Direction (in one helix direction), the specimen shall have the following tear strength

: 1323 lb./in

PERFORMANCE: The fender shall be designed so that when the resilient foam layer is compressed to its rated deflection by a flat plate extending the full length and width of the fender (on one side of the fender only), the fender shall absorb the rated energy absorption (+/- 15%) with a corresponding reaction force (+/- 15%). Performance shall be based on the following characteristics:

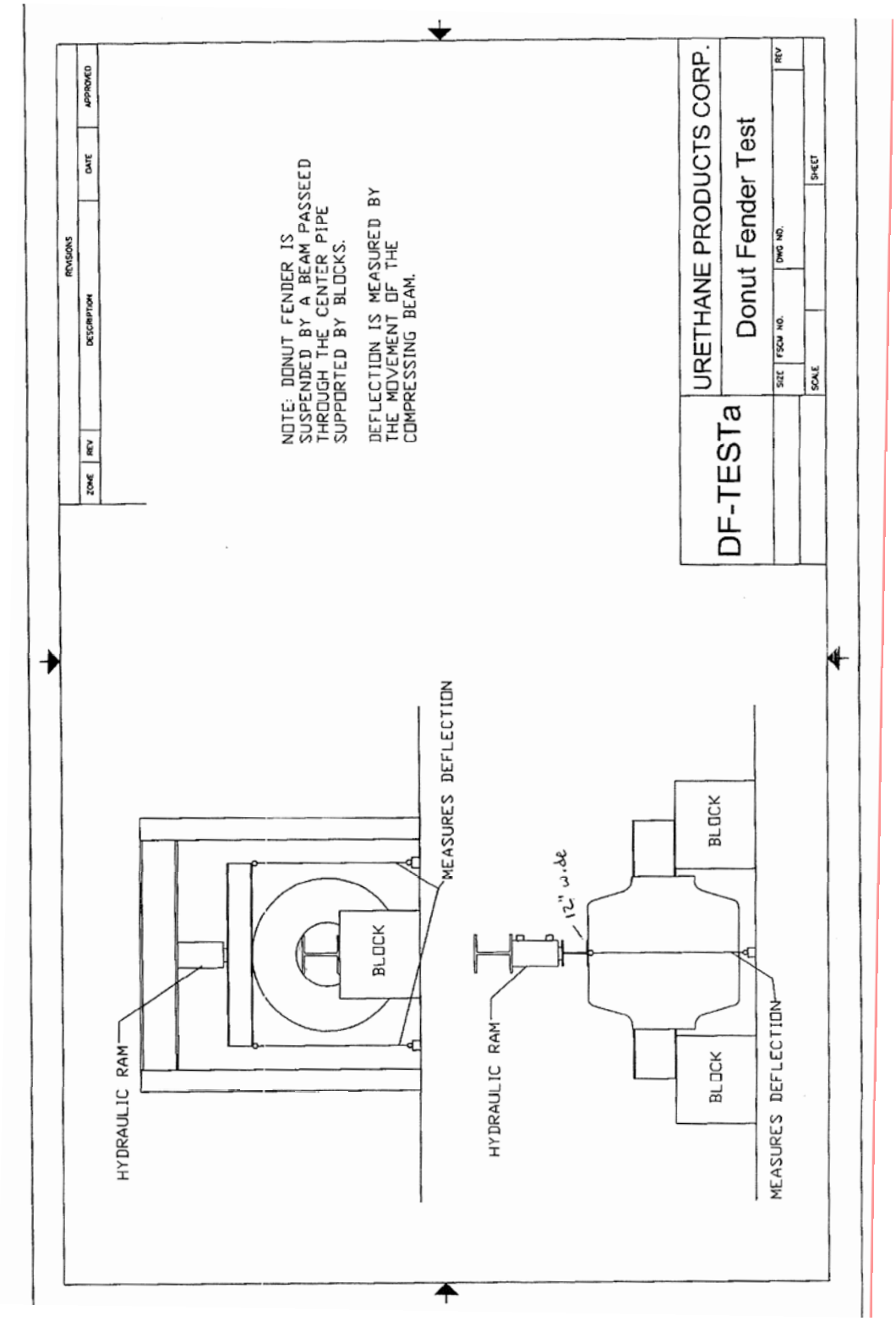
Pile Diameter:	2.0 feet
Fender Diameter:	6.0 feet
Flat Height:	4 ~~14.5~~ feet
Skin Thickness:	1.25 minimum
Energy Absorption (60% compression):	~~57 ft-kips~~
Reaction Force (60% compression):	~~100 kips~~

* Actual values may vary plus or minus 15% from stated values due to variations in material properties dimensional tolerances, temperatures and speed of compression.

Source: An acceptable product meeting the requirements of this specification is manufactured by Urethane Products Corporation, Bellflower, California, U.S.A. Telephone: 1 (800) 913-0062, 1 (562) 630-4982, Fax: 1 (562) 630-6974.

APPENDIX B

FENDER TESTING INFORMATION



BALANCE INDUSTRIAL SCALE COMPANY

P.O. BOX 5478
LOS ALAMITOS, CA. 90720
(562) 799-0299

CALIBRATION TEST REPORT

CUSTOMER: COORDINATED EQUIPMENT CO. (WILMINGTON, CA.) CERT. #1751-AP

SYSTEM DISCRIPTION: HYDRAULIC CYLINDER WITH ASHCROFT PRESSURE GAGE

HYDRAULIC CYLINDER **S/N:** CEC1025 **CAPACITY:** 0-400,000 LBS.

INFO/ID: N/A

PRESSURE GAGE **S/N:** A4A-127811 **STANDARD:** COMPRESSION LOADCELL

ID: 73033-2

TRUE LOAD LBS.	IND. PSI RUN#1	IND. PSI RUN#2
0	0	0
40,000	500	500
80,000	1010	1010
120,000	1515	1515
160,000	2020	2020
200,000	2525	2525
240,000	3030	3030
280,000	3535	3535
320,000	4040	4040
360,000	4540	4545
400,000	5045	5045

COMMENTS: CALIBRATION DUE DATE: 04-03-04

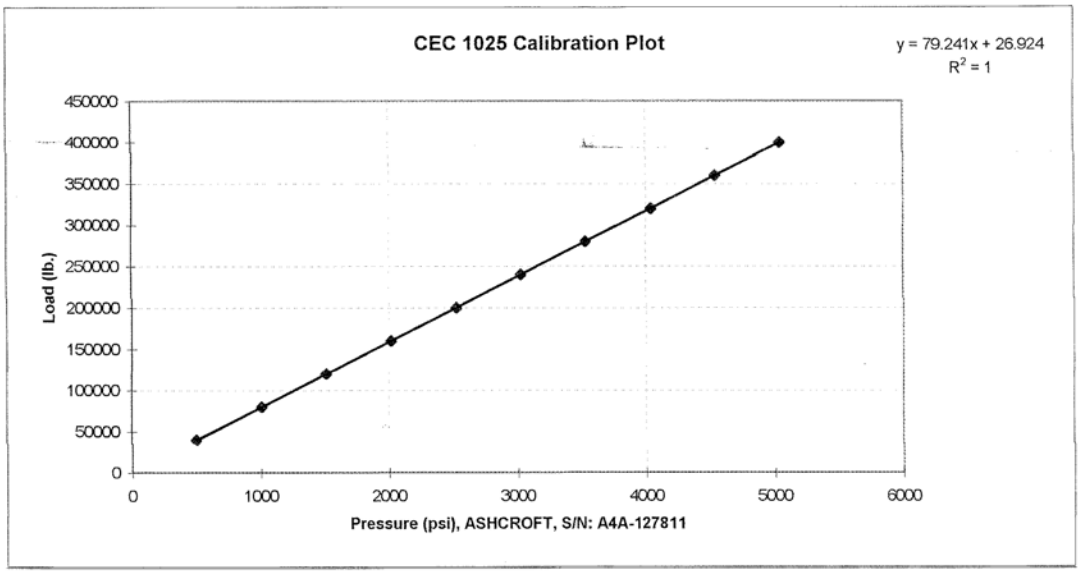
The Calibrated accuracy is traceable to the National Institute of Standards and Technology (NIST) through certified standards maintained in the calibration laboratory of Balance Industrial Scale Company. Calibration systems and procedures established and maintained per ANSI/NCSL Z540-1. This certificate shall not be reproduced except in full without the written approval of Balance Industrial Scale Company.

DATE APRIL 04, 2003 **TECHNICIAN** ANDREW PRITCHARD **STATE REG. #** 4069 **-2370**

TITLE Q.A./TECH **SIGNED** Andrew Pritchard

Calibration Record Hydraulic Cylinder, S/N: CEC1025

Gauge No:	ASHCROFT, A4A-127811					
Load (lbs)	Test Run No.1	Test Run No.2	Ave. Pressure (psi), (X)	Load (lbs), (Y)	X * Y	X^2
0	0	0	0	0	0	0
40,000	500	500	500	40000	20000000.0	250000.00
80,000	1010	1010	1010	80000	80800000.0	1020100.00
120000	1515	1515	1515	120000	181800000.0	2295225.00
160000	2020	2020	2020	160000	323200000.0	4080400.00
200000	2525	2525	2525	200000	505000000.0	6375625.00
240000	3030	3030	3030	240000	727200000.0	9180900.00
280000	3535	3535	3535	280000	989800000.0	12496225.00
320000	4040	4040	4040	320000	1292800000.0	16321600.00
360000	4540	4540	4540	360000	1634400000.0	20611600.00
400000	5045	5045	5045	400000	2018000000.0	25452025.00
			27760	2,200,000	7773000000.0	98083700.00
			2776	220000		
Slope of Straight Line:		79.24				



Calibration Date: 4/3/03
Calibration Due Date: 4/3/04

APPENDIX C

ELECTRONIC FILE DATA OF TEST RESULTS

(See Attached Disk)