Naval Facilities Engineering Command

ENGINEERING SERVICE CENTER
Port Hueneme, California 93043-4370

TECHNICAL REPORT
TR-NAVFAC ESC-CI-1104

FLOATING DOUBLE DECK PIER FENDERS

By

Joseph Abraham
Cameron Stewart
Darren Finklea

July 2011

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REPORT DOCUMENTATION PAGE (SF 298)						*Form Approved* *OMB No. 0704-0811*

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1. REPORT DATE *(DD-MM-YYYY)*	2. REPORT TYPE	3. DATES COVERED *(From – To)*
29-07-2011	Technical Report	10-2009 - 07-2011

4. TITLE AND SUBTITLE	5a. CONTRACT NUMBER
Floating Double Deck Pier Fenders	5b. GRANT NUMBER
	5c. PROGRAM ELEMENT NUMBER
6. AUTHOR(S)	5d. PROJECT NUMBER 22715-01001004
Joseph Abraham Cameron Stewart Darren Finklea	5e. TASK NUMBER
	5f. WORK UNIT NUMBER

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESSES	8. PERFORMING ORGANIZATION REPORT NUMBER
NAVFAC Engineering Service Center 1100 23rd Avenue Port Hueneme, CA 93043	
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)	10. SPONSOR/MONITORS ACRONYM(S)
Naval Facilities Engineering Command	11. SPONSOR/MONITOR’S REPORT NUMBER(S)

12. DISTRIBUTION/AVAILABILITY STATEMENT

13. SUPPLEMENTARY NOTES

14. ABSTRACT

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The modeling proved that the biaxial configuration does provide greater resistance in the direction which was more prone to sideway buckling in the uniaxial configuration. However, the modeling also indicated that the uniaxial fender used in the MHP Test Bed collapsed in both the direction prone to failure and in the direction that was intended to provide greater resistance. Based on these findings, it appears that the friction force between the fender and the mooring shaft may be too large for the proposed biaxial fender configuration despite the addition of two rubber elements. Analysis and testing will need to be carried out to determine if the fender will provide enough resistance to overcome sideway buckling from the frictional forces or alternate mooring concepts may need to be considered.

15. SUBJECT TERMS

16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
a. REPORT	b. ABSTRACT	c. THIS PAGE			
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EXECUTIVE SUMMARY

A long term test program was performed by the University of California San Diego (UCSD) under contract by Berger/ABAM to determine the level of performance of the fender configuration used on the Modular hybrid Pier (MHP) Test Bed. During the test, the fender underwent an undesirable sideway buckling. Berger/ABAM revised the fender design for the Floating Double Deck Pier (FDDP) by redesigning the MHP Test Bed uniaxial fender composed of two rubber elements to a biaxial fender composed of four rubber elements. The Naval Facilities (NAVFAC) Engineering Service Center (ESC) has been tasked to investigate the non-linear behavior of this new rubber fender configuration using finite element modeling to determine if this biaxial configuration is adequate for the FDDP and if additional tests are required.

The modeling proved that the biaxial configuration does provide greater resistance in the direction which was more prone to sideway buckling in the uniaxial configuration. However, the modeling also indicated that the uniaxial fender used in the MHP Test Bed collapsed in both the direction prone to failure and in the direction that was intended to provide greater resistance. Based on these findings, it appears that the friction force between the fender and the mooring shaft may be too large for the proposed biaxial fender configuration despite the addition of two rubber elements. Analysis and testing will need to be carried out to determine if the fender will provide enough resistance to overcome sideway buckling from the frictional forces or alternate mooring concepts may need to be considered.

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ACRONYMS AND ABBREVIATIONS

DMA	Dynamic Mechanical Analysis
ESC	Engineering Service Center
FDDP	Floating Double Deck Pier
FEM	Finite Element Model
MHP	Modular Hybrid Pier
NAVFAC	Naval Facilities
RDT&E	Research, Development, Testing, and Evaluation
UCSD	University of California San Diego
UHMW	Ultra-High Molecular Weight

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1.0 INTRODUCTION

The FDDP is an alternative to traditional pile U.S. Navy berthing piers. The pier is designed and configured to accommodate a variety of Navy vessels to berth. The FDDP floatation enables it to remain at a constant relative position to moored ships as the tides change. Furthermore, the FDDP is designed to provide an economical and functional 100 year service life and may be relocated as U.S. Navy needs change [1].

A preliminary 35 percent design of the FDDP has been created by Berger/ABAM under contract to NAVFAC ESC and is published in reference [1]. To maintain its position in the water, the FDDP is equipped with rubber fenders mounted on the structure. The fenders face the sides of a mooring shaft, which is highlighted by a green circle in Figure 1-1 below. The fenders are expected to resist wind, wave, and tidal forces as well as withstand ship berthing and seismic activity.

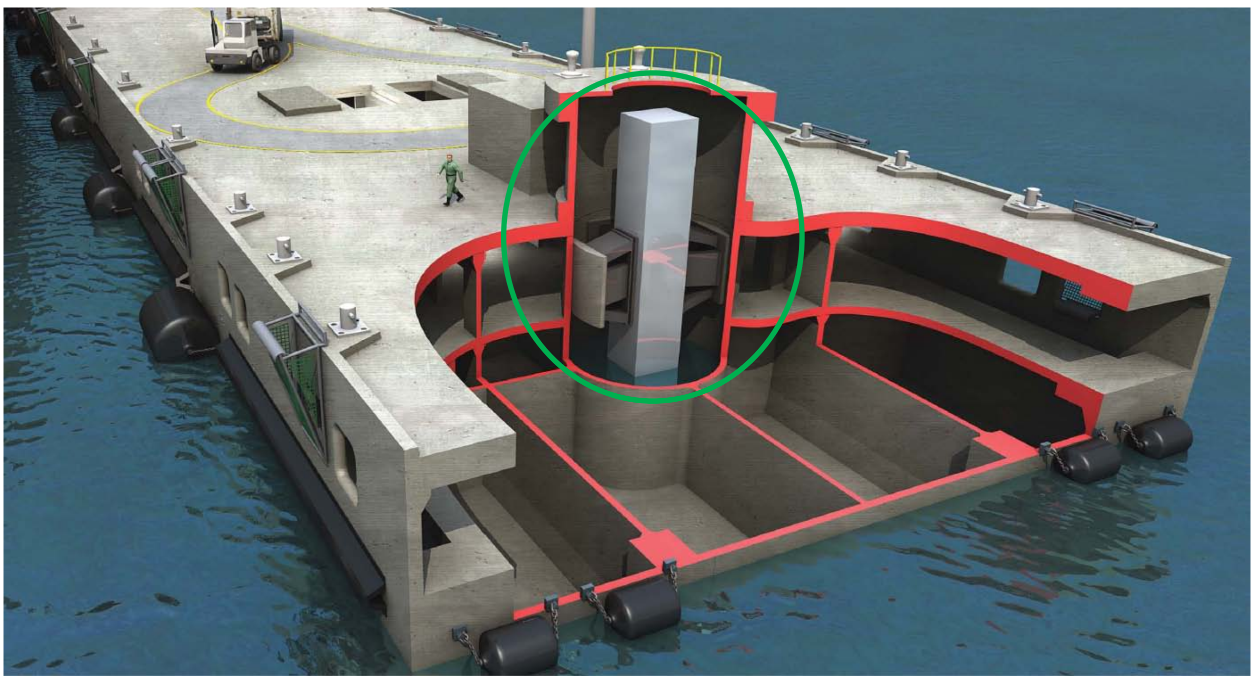

Figure 1-1. FDDP Fenders and Mooring Shaft

The FDDP is based on a Research, Development, Testing, and Evaluation (RDT&E) effort known as the MHP. During the MHP RDT&E effort a Test Bed was constructed and installed at Naval Station San Diego for testing and demonstration purposes. The Test Bed was equipped with four Trellex fenders manufactured by Trelleborg. Each fender is composed of two Trellex MV1000x900B elements. The configuration of the fenders is shown in Figure 1-2. Detailed specifications of the geometry and performance of the Trellex MV1000x900B rubber elements can be found in the Trelleborg product manual in reference [2].

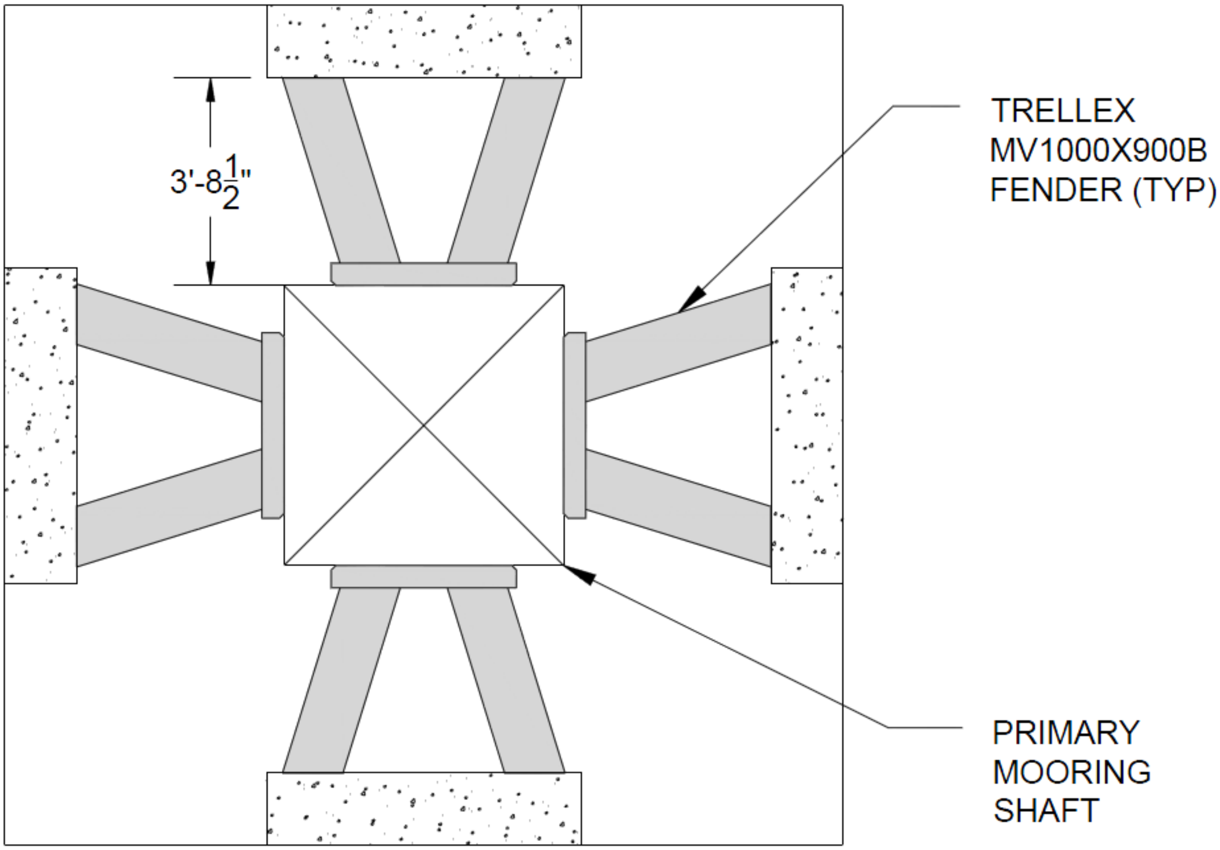

Figure 1-2. Plan View of MHP Fender System

The MV1000x900B elements that comprise the fenders used in the MHP Test Bed are mounted to the structure on one end and to an ultra-high molecular weight (UHMW) polymer face pad on the other end. The UHMW polymer face pad is designed to minimize friction as it interacts with the stainless steel mooring shaft. The fenders are intended to compress in the axial direction normal to the plane of the face pad. If compression occurs in the axial direction, the fender system can resist large forces and can absorb large amounts of energy. Each MV1000x900B element has a rated force capacity of 311kN and can absorb 143 kJ of energy. The axial capacity of a fender is taken as the sum of the rated maximum axial capacity of each MV1000x900B element.

In order to determine the level of performance of the fender configuration used on the MHP Test Bed, UCSD carried out a short term and long term tests under contract by Berger/ABAM. The short term tests were used to explore temporary, high load behavior of the fender system. Long term tests were used to study creep behavior and environmental effects such as wind, tide and temperature. The fender system sustained a load for a prolonged period of time during the long term testing by using two weight blocks and a pulley system [3].

During the long term testing, one fender was tested at 60% of the reaction capacity. The test was scheduled to last four weeks. However, it was stopped two days later due to damage observed in the fender. The fender underwent an undesirable sideway buckling as shown in Figure 1-3 [3].

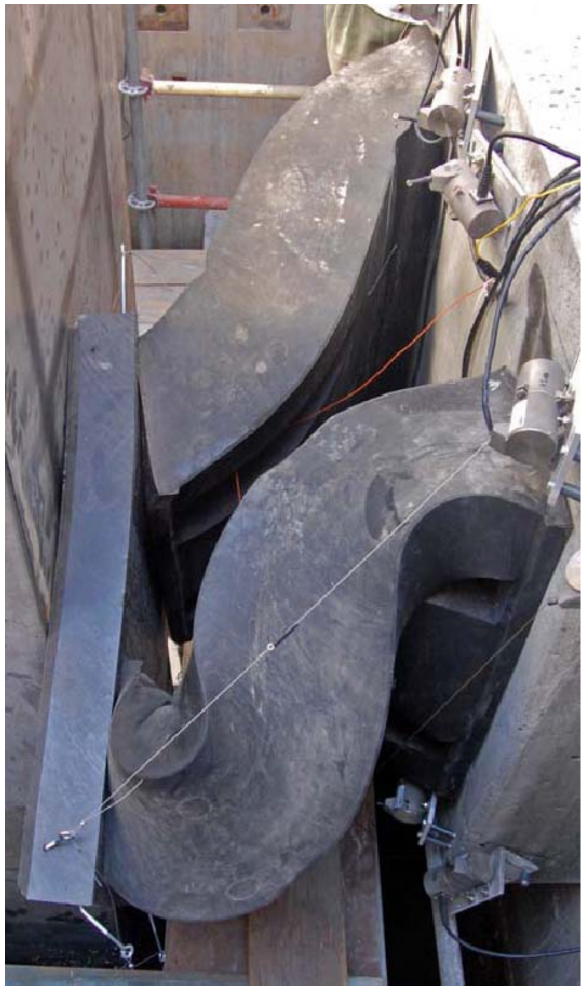

Figure 1-3. MHP Fender Sideway Collapse

The undesirable deformation likely occurred as a result of the friction that developed between the face pad and the mooring shaft due to the load applied with the weight blocks. The sideway buckling behavior was prevented in subsequent tests by installing lateral restraint chains. The lateral restraint chains proved to be effective and prevented the sideway buckling in similar long term tests. However, the use of the lateral chains is not desirable because of the added stiffness and lack of energy absorption.

In the preliminary 35% design prepared by Berger/ABAM for the FDDP [1], the fenders were redesigned from a uniaxial configuration composed of two rubber elements to a biaxial configuration composed of four rubber elements. The uniaxial setup is the configuration used on the MHP Test Bed. The proposed biaxial fenders are composed of UE1200x1200E3.1 elements which are larger than the MV1000x900B elements and constructed with stronger rubber material. Figure 1-4 and 1-5 illustrate the design drawing in the Berger/ABAM preliminary design [1].

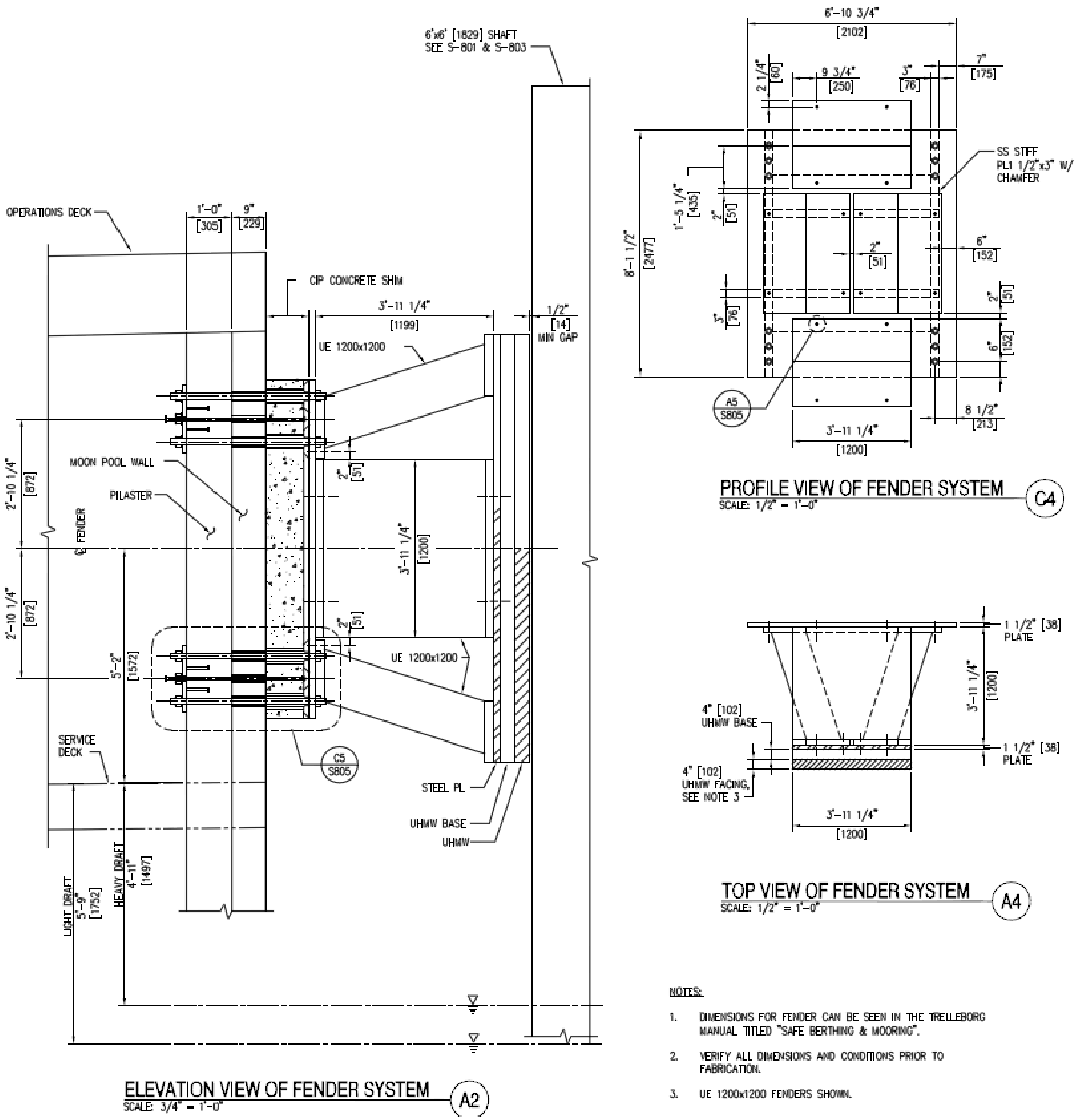

Figure 1-4. Preliminary Design of FDDP Fender

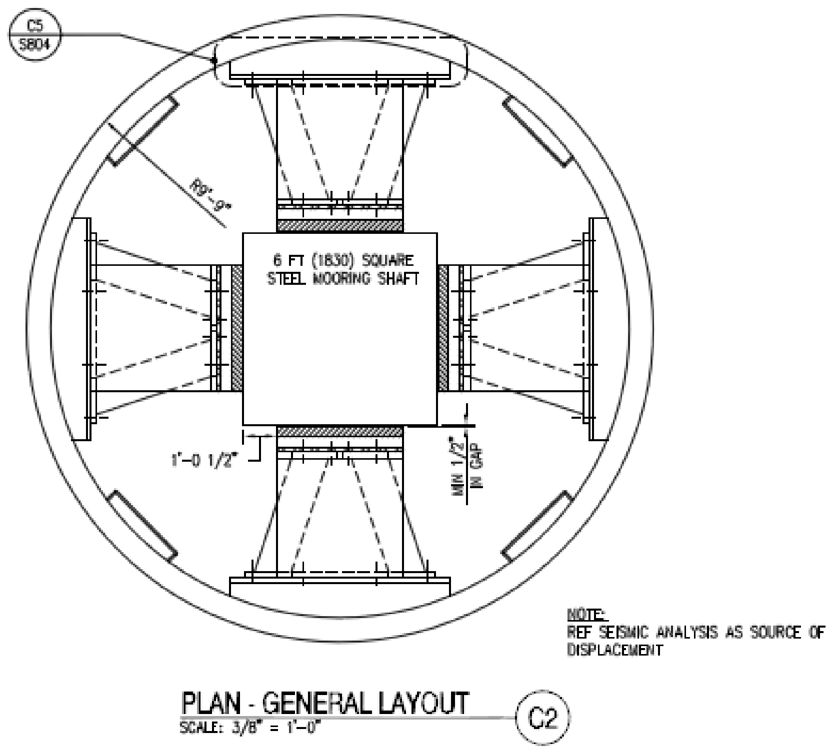

Figure 1-5. FDDP Fender – General Layout

The biaxial fender configuration is expected to provide more resistance to the sideway buckling behavior compared to the uniaxial configuration. The expectation of improved performance is due to the addition of two elements oriented to prevent sideway buckling.

NAVFAC ESC has been tasked to investigate the non-linear behavior of this new rubber fender configuration using a finite element model (FEM) to determine if this biaxial configuration is adequate for the FDDP and recommend if additional tests are required.

2.0 FINITE ELEMENT MODEL

A three dimensional explicit finite element code was determined to be the best approach to model the large fender deformations and non-linear behavior. LS-Dyna is a general purpose explicit dynamic finite element program capable of simulating the complexity of the rubber fender and has been used to model the behavior of the MV1000x900B and UE1200x1200E3.1 elements. Figures 2-1 and 2-2 illustrate the FEM developed for the uniaxial and biaxial fender configurations.

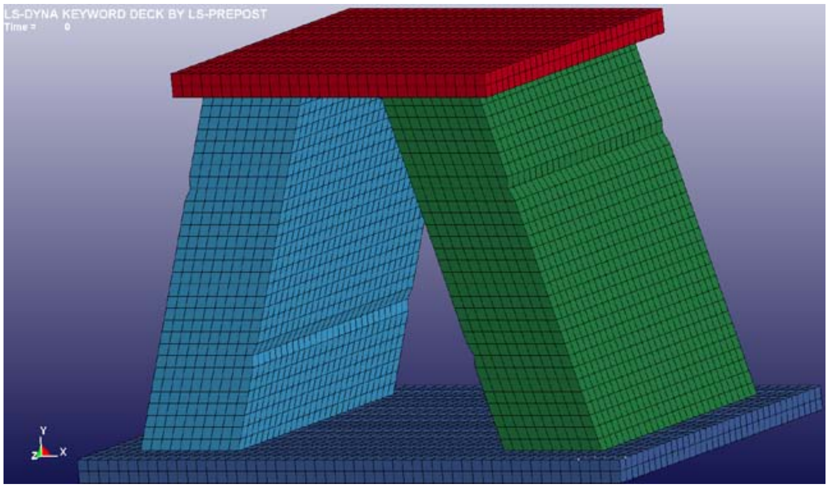

Figure 2-1. Uniaxial Fender Composed of MV1000x900B Elements

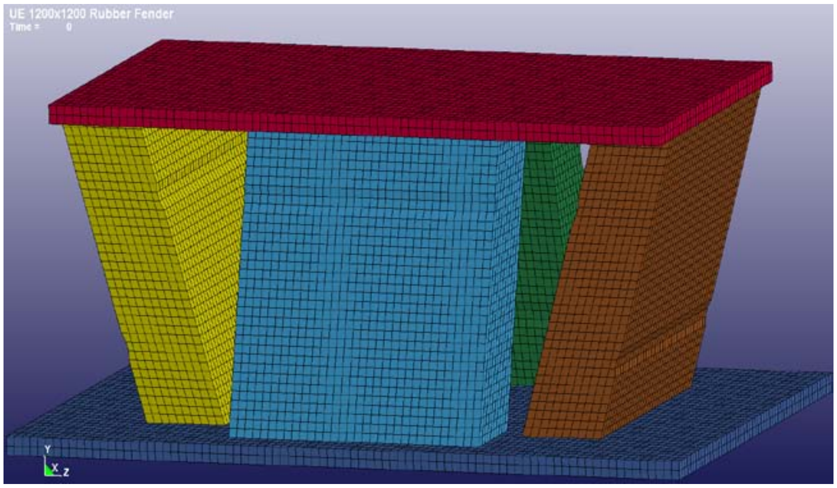

Figure 2-2. Biaxial Fender – Composed of UE1200x1200E3.1 Elements

2.1 Element Formulation

The FEMs illustrated in Figures 2-1 and 2-2 are composed of 8-node hexahedron solid elements. The integration rule used to solve the strain-displacement of the solid elements is a 1-point integration that assumes constant stress in the element [4]. An explicit solver was used because I allowed the use of contact surface and the loading was monotonic. This type of integration was selected for the rubber fender model because it is computationally efficient. Furthermore, it was necessary to use the 1-point integration instead of an 8-point fully integrated element because elements may lock up for Poisson's ratios that approach 0.5, which is the case for rubber. The constant stress 1-point integration precludes locking but introduces undesirable hourglass modes that must be resisted using viscous damping or with a small elastic stiffness capable of stopping the formation of the anomalous modes. The mechanism used for hourglass control must have negligible effect on the stable global modes [4]. The stiffness formulation of the hourglass control was used in the model because the response of the fender is quasi-static. Viscous hourglass control is more suitable for dynamic simulations at higher loading rates.

2.2 Geometry and Boundary Conditions

The models were constructed using the element dimensions specified in the Trelleborg product manual [2] and were placed relative to each other in accordance with the drawings in [1]. The bottom blue plate shown in Figures 2-1 and 2-2 was fixed in all simulations and represents the structure of the FDDP to which the fender is bolted. The red plate in the figures represents the face pad, which moves as a rigid object in the FEM according to a set of prescribed displacements in the x, y and z directions. Two indentations exist on each fender element that enables the element to collapse into an “S” shape when it is compressed. The location of the indentations was not specified in [2] and had to be approximated.

The coincident nodes of the solid FEM elements between the rubber components and the top and bottom plate are merged. In this manner, the boundaries of the elements will be subject to the same constraints and prescribed motions as the plates. Furthermore, the rubber nodes that are merged with the plate will not be able to expand or deform during compression. This is found to be a reasonable representation of the actual fenders because a thin steel plate is present inside the rubber elements as shown in Figure 2-3 and through which the rubber fender is tightly bolted to the face pad and to the FDDP. The steel plate prevents the rubber from expanding in the vicinity of the FDDP and the face pad, which is also the case in the FEM.

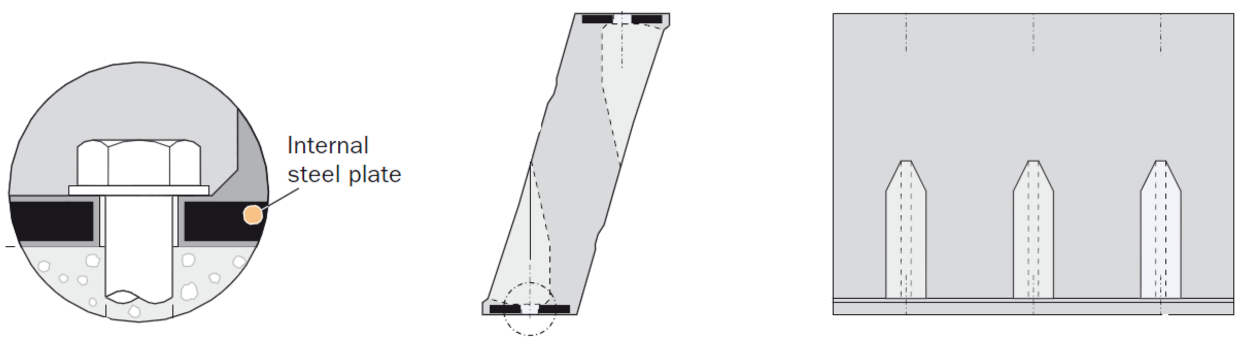

Figure 2-3. Illustration of Steel Plate Inside Fender Element

2.3 Rubber Material Model

LS-Dyna contains a library of material and equations of state models that can be implemented in a FEM by invoking the name of material in the input card. Material model *MAT_SIMPLIFIED_RUBBER_WITH_DAMAGE [5] was selected and used to simulate the fender rubber. This material model provides an incompressible rubber model defined by a single uniaxial load curve for loading and a single uniaxial load curve for unloading. The model was developed by DaimlerChrysler, Sindelfingen, Paul Du Bois, Livermore Software Technology Corp and Professor Dave J. Benson from UCSD [5].

In addition to the loading and unloading uniaxial load curves needed to use the model, the mass density and linear bulk modulus of the rubber needed to be known. Trelleborg, the manufacturer of the MV and UE fender elements, allowed NAVFAC ESC to purchase rubber samples in order to carry out material tests and determine the rubber properties required for the model to work properly. This model was selected based on ease of implementation and accuracy in representing non-linear behavior of rubber. Several different models require extensive material testing to determine several input parameters.

3.0 MATERIAL TESTING

The Trelleborg product manuals list approximately 25 rubber compounds of varying hardness that can be used to manufacture the UE and MV elements. Trelleborg allowed NAVFAC ESC to purchase samples of only three rubber compounds. Five rubber pads of dimensions 15.5 x 19.5 x 0.2 cm were provided for each compound to NAVFAC ESC. Unfortunately, Trelleborg did not identify the corresponding rubber compounds in the product manual for any of the three compound samples. The three compound samples were labeled by their hardness as: Shore A60, A65, or A70. The compound used for the UE1200x1200E3.1 fender elements is the hardest of all the 25 compounds. On the other hand, the hardness of the compound used for the MV1000x900B fender elements is unknown relative to the other 25 compounds.

3.1 Mass Density

The density of each rubber compound was determined using two methods. The first method involved measuring the length, width, and thickness of small rectangular specimens that had been cut from the rubber pads provided by Trelleborg and weighting each sample. The density was subsequently computed by dividing the mass by the volume. The second method used a small graduated cylinder filled with water in which each rubber sample was placed in the water and the water displacement was measured. The water displacement was subsequently used to determine the volume of each sample and subsequently the density was determined by dividing the mass by the volume. For each compound, the density was measured with both methods using four rubber specimens and the results were averaged. The averages from all the measured densities for each rubber compound are summarized in Table 3-1.

Elastomer	Density (kg/m^3)
A60	1110
A65	1133
A70	1141

Table 3-1. Density of Rubber Compounds

3.2 Loading/Unloading Curves

The uniaxial loading and unloading curve data for the model must be obtained from tensile and compressive testing recorded as engineering stress versus engineering strain or force versus actual change in gauge length [6]. The load and unload curves should also cover the complete range of expected loading, i.e. the smallest stretch ratio to the largest [5]. The thickness of the rubber pad provided by Trelleborg was too thin to enable compression testing. On the other hand, it was possible to use a standard ASTM die to cut dumbbell shaped specimens for tensile testing. Cutting Die D from ASTM D412-06a was used to cut the dumbbell specimens [7]. Symmetry about the line $y = x$ was used to extrapolate the loading and unloading curves for compression. This rough assumption for the compression should be fairly reasonable for a material such as rubber.

For each rubber compound, five specimens were tested using an Instron machine and the measured loading and unloading curves have been averaged for each compound. The gauge length, width and depth of the specimens were also recorded since they are required input for the material model. The tensile test was carried out on the dumbbell specimens by stretching the rubber at a rate of 5 inches per minute.

Figure 3-1 illustrates a dumbbell rubber specimen being tested in the Instron machine. The curves that were obtained from the testing and were used in the LS-Dyna FEM for the three rubber compounds are shown in Figures 3-2 to 3-4.

Figure 3-1. Tensile Testing of Rubber Specimens

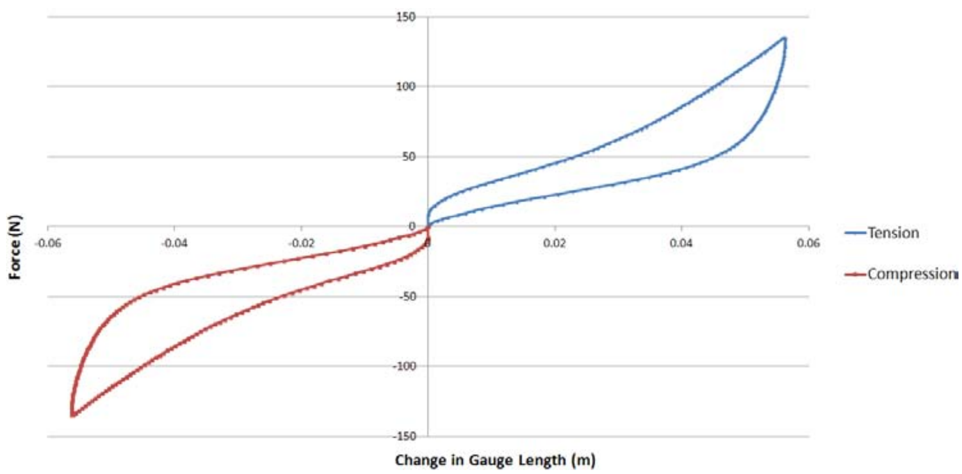

Figure 3-2. Loading/Unloading Curve for A60 Compound

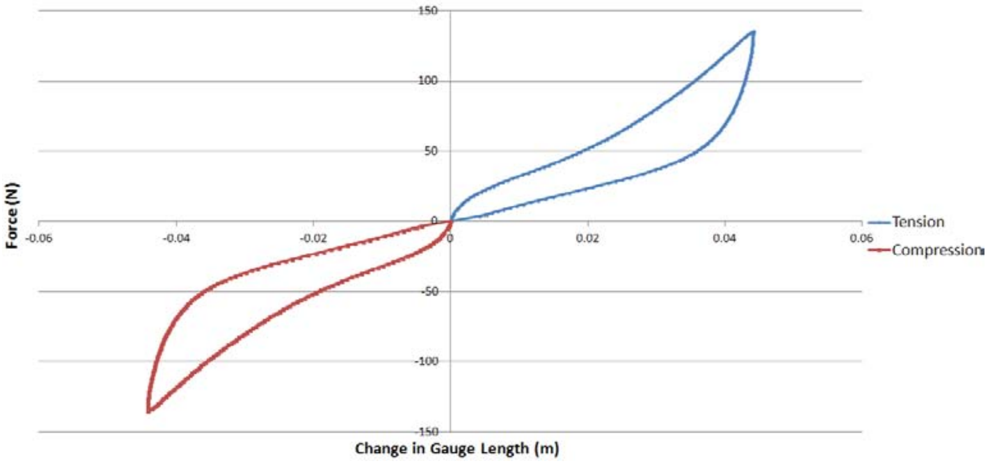

Figure 3-3. Loading/Unloading Curve for A65 Compound

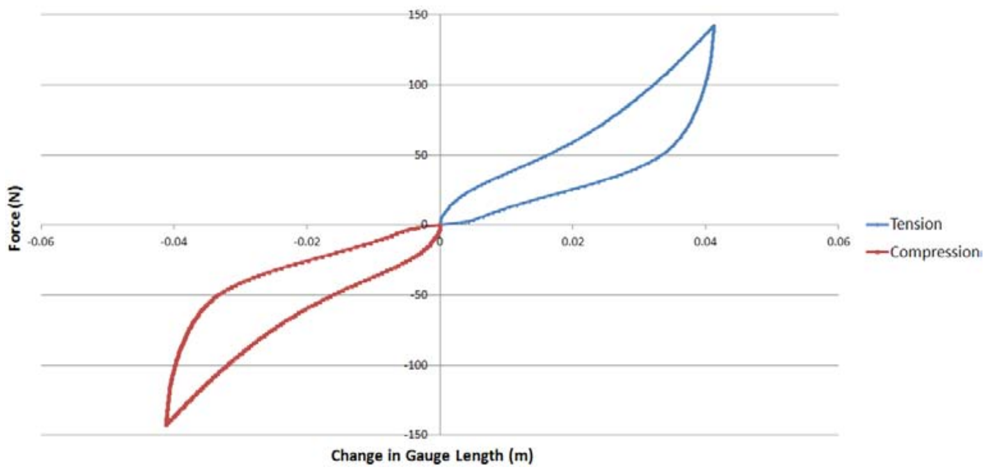

Figure 3-4. Loading/Unloading Curve for A70 Compound

The dimensions for each rubber specimen were measured and averaged for each compound. The measurements of the specimens are reported in Table 3-2.

Elastomer	Specimen Gauge Length (cm)	Specimen Width (cm)	Specimen Depth (cm)
A60	2.54	1.36	0.19
A65	2.54	1.34	0.20
A70	2.54	1.34	0.21

Table 3-2. Specimen Dimensions

3.3 Linear Bulk Modulus

The linear bulk modulus is the last parameter needed in order to use the *MAT_SIMPLIFIED_RUBBER_WITH_DAMAGE and it can be derived algebraically from the elastic storage modulus. The elastic storage modulus, E, can be obtained from a Dynamic Mechanical Analysis (DMA) machine [8]. The elastic storage modulus can be related to the linear bulk modulus, K, using the following relationship:

$$K = \frac{E}{3(1-2\nu)} \ ,$$

where ν is Poisson's ratio. Poisson's ratio was measured during the tensile test described in section 3.2 above by measuring the transverse and axial strains of the rubber specimens. The elastic storage modulus was measured with the DMA machine using specimens of the following dimensions 35.5x12.7x2mm. Figure 3-5 illustrates the DMA machine with a specimen mounted into the cantilever frame. The specimens were tested at a frequency of 10 Hz and strain amplitude of 90 μm. Table 3-3 summarizes Poisson's ratio, the elastic storage modulus and the linear bulk modulus values obtained for each rubber compound from the testing.

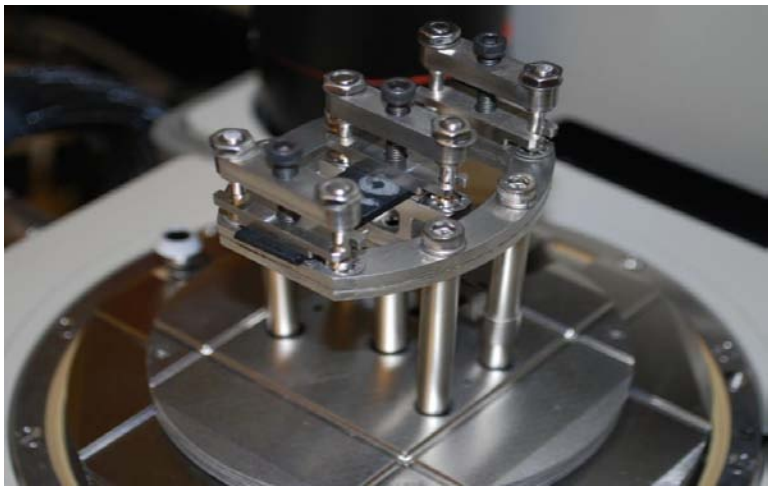

Figure 3-5. DMA Machine With a Test Specimen

Elastomer	Poisson's Ratio	Elastic Storage Modulus (MPa)	Linear Bulk Modulus (MPa)
A60	0.427	21.26	48.74
A65	0.446	40.34	124.08
A70	0.467	52.25	262.75

Table 3-3. Input Values for Rubber Material Model

4.0 MODEL VALIDATION

Figure 4-1 is from the Trelleborg product manual [2] and illustrates the deformation of a fender element in a test facility that is deflected in the axial direction normal to the face pad.

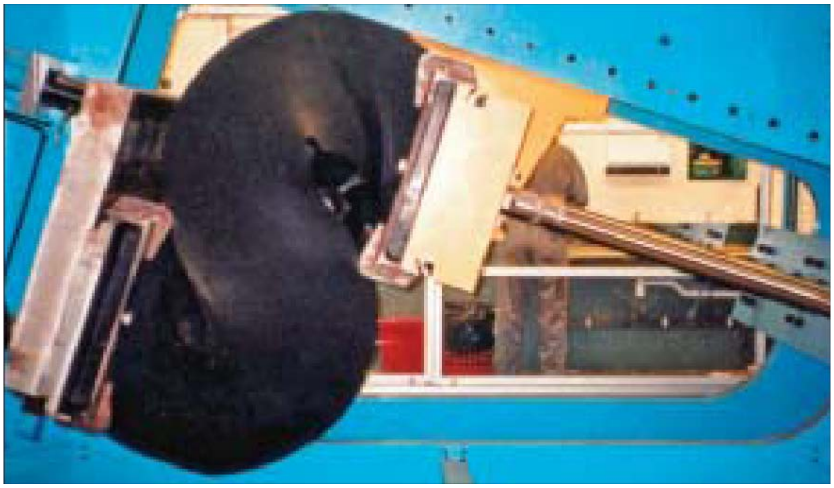

Figure 4-1. Single Element Deflection in a Test Structure

Figure 4-2 illustrates a generic performance plot for the reaction force and energy absorbed. A table in the Trelleborg product manual provides the rated performance data for every fender size and compound. The rated reaction force in the performance data table corresponds to the 100% value in Figure 4-2. For example, if the rated capacity of a fender is rated as 100 kN, then the 100% value on the y-axis is equivalent to 100 kN and the 40% value corresponds to 40 kN.

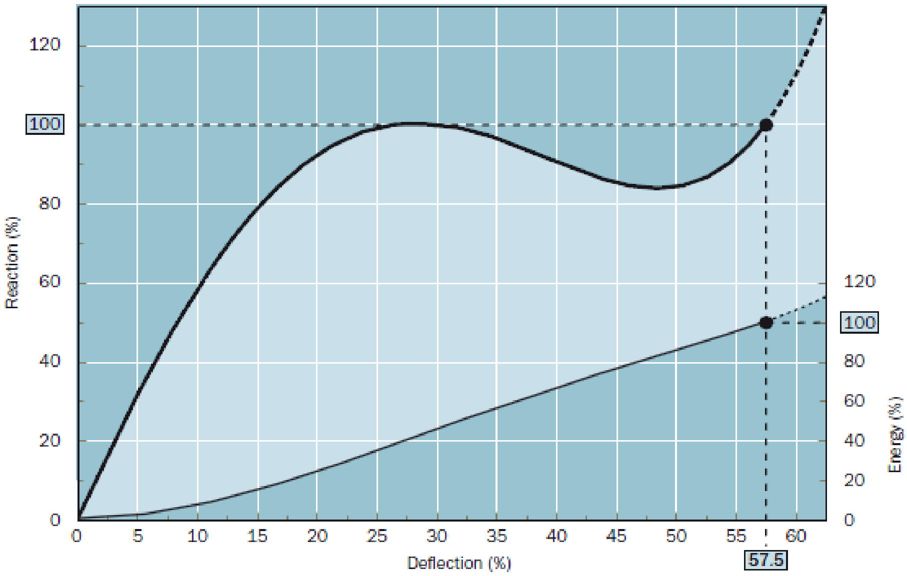

Figure 4-2. Rated Performance of Trelleborg Fender Elements

4.1 Uniaxial MV1000x900B Rated Performance

The uniaxial fender used in the MHP Test Bed, has been modeled with the FEM according to the product data for the MV1000x900B elements described in reference [2]. The rated reaction force of the MV1000x900B elements is 311 kN. Figure 4-3 illustrates the deformation of the uniaxial fender when subjected to the same deflection as the Trelleborg test facility. Comparison of Figures 4-1 and 4-3 demonstrates that the FEM accurately reproduced the deformation of the fender.

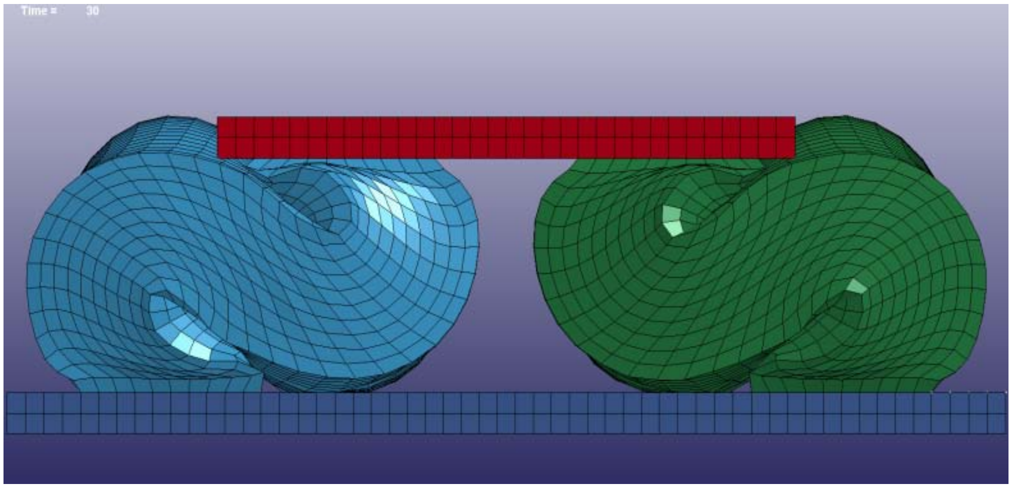

Figure 4-3. Uniaxial MV1000x900B Fender Deflection

The material model data from the A60, A65 and A70 rubber samples was used to model the deflection of the MV fender. The predicted element reaction from a simulation with the A60 rubber was substantially under-predicted in comparison with the MV1000x900B rated capacity indicating that the A60 rubber is softer than the MV1000x900B rubber. On the other hand, the predicted element reaction from a simulation with the A70 rubber was substantially over-predicted in comparison with the MV1000x900B rated capacity indicating that the A70 rubber is harder than the MV1000x900B rubber. The A65 compound properties provided the best approximation for the rated reaction of the MV fender. The FEM model approximation of the reaction force is plotted in Figure 4-4. The reaction force plotted in Figure 4-4 for each MV element has a peak reaction force of 270 kN. The force plot exhibits a similar nonlinear behavior as that provided in the product manual and shown in Figure 4-2. The reason the peak reaction force from the A65 FEM model is different from the rated performance may be because the hardness of the A65 rubber is not the same as the MV element rubber. Furthermore, the reaction force of the rubber element is dependent on loading rate and temperature, which have not been considered in this model.

The similarity between the deformation of the fender in the FEM to the picture provided by Trelleborg and the similarity between the non-linear behaviors captured in the force versus deflection plots provide enough confidence to use the FEM to make predictions. While the exact magnitudes may not be the same the general response of the fender can be predicted so that an assessment of the uniaxial configuration and sideway buckle can be made.

Comparison between the Trelleborg data and the FEM for the energy absorbed by the fender during the deformation indicates that the values are reasonably close. The results of this comparison are not in the report but since energy can be approximated as the area below the force versus deflection plot, it can be seen from Figures 4-2 and 4-4 that the computational and manufacturer data are comparable in magnitude.

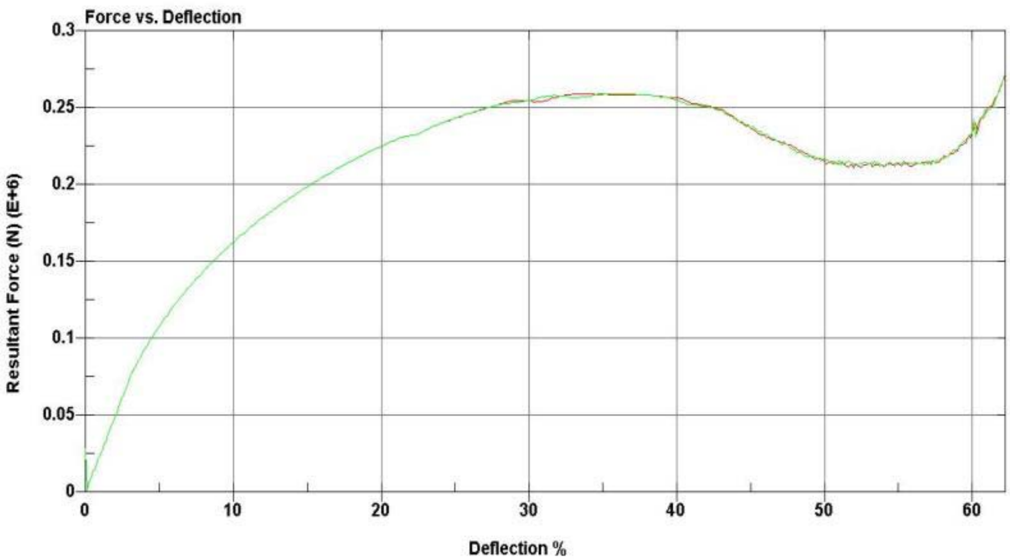

Figure 4-4. FEM Performance of MV1000x900B Elements

4.2 Biaxial UE1200x1200E3.1 Rated Performance

The biaxial fender used in the FDDP design, has also been modeled with the FEM according to the product data for the UE1200x1200E3.1 elements described in reference [2]. The rated reaction force of the entire fender with four UE elements is 3388 kN. Figure 4-5 illustrates the deformation of the biaxial fender. The material model data from the A60, A65 and A70 rubber samples was used to model the deflection of the UE fender. The predicted element reaction from simulations with the A60 and A65 rubber was substantially under-predicted in comparison with the UE fender rated capacity indicating that the compounds are softer than the UE rubber. The A70 compound properties provided the best approximation for the rated reaction of the UE fender. The FEM model approximation of the reaction force for the A70 compound is plotted in Figure 4-6. The reaction force plotted is for the combined reaction of the four UE elements. The peak computed reaction force is about 2450 kN. The reaction force is under-predicted but this result was expected since the E3.1 compound used on the UE fender is the hardest compound available from Trelleborg.

Similar to the previous section, the comparison between the computed and rated performance of the fender is close but not exact. However, the overall predicted behavior by the FEM model is accurate and the non-linear performance of the rubber matches that data provided by the manufacturer. Furthermore, the rubber compound samples used to create the material model, do not match the actual rubber compounds for which the rated performance is provided.

The current configuration of the biaxial fender may allow interference between the UE elements if the face pad is deflected as shown in Figure 4-5. The interference observed in the FEM between the elements was minor but the tolerances between the elements should be revisited in future designs of the biaxial fender.

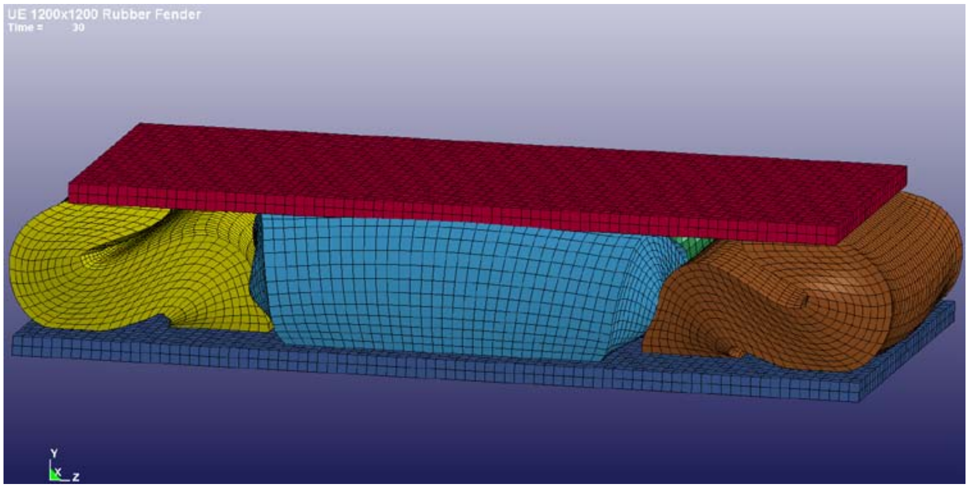

Figure 4-5. Biaxial UE1200x1200E3.1 Fender Deflection

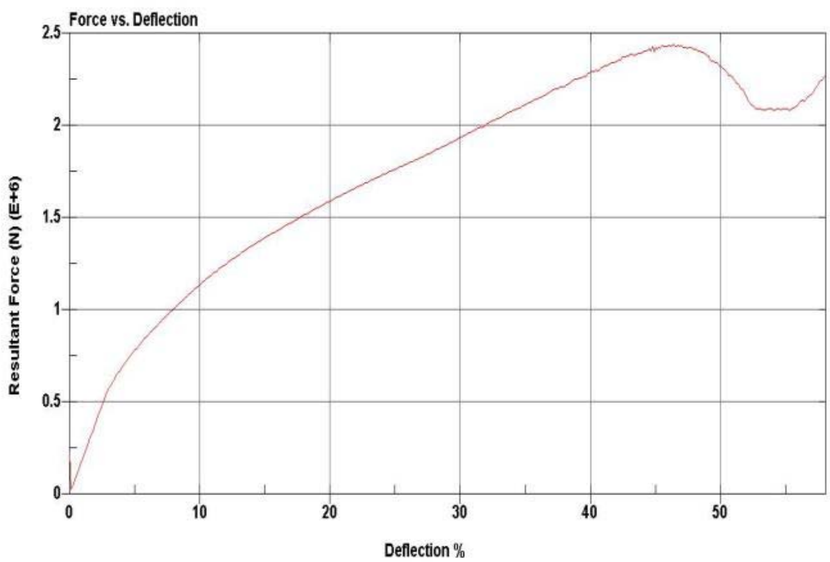

Figure 4-6. FEM Performance of Biaxial UE Fender

5.0 COMPUTATIONAL RESULTS

The FEMs described in the previous sections for the uniaxial and biaxial fenders have been used to analyze the sideway deformation observed in the long term tests on the MHP Test Bed. During the long term testing, the MHP fender was instrumented with twelve displacement transducers and cameras. The displacement transducers were installed to capture the deflection of the face pad and the cameras were installed to capture images of the fender throughout the test. The large deformation that occurred during the long term testing was captured by the cameras and is illustrated in Figure 5-1. Unfortunately, during this event, the deformed rubber damaged the displacement transducers and no data was recorded to indicate the deflection of the face pad [3].

Figure 5-1. MHP Fender Sideway Collapse

In this section, the sideway collapse of the MHP fender is reproduced using the validated model for uniaxial fender with the MV1000x900B elements and the deflections of the face pad are determined from the calculations. The FDDP biaxial fender model with the UE1200x1200E3.1 fenders are subsequently subjected to the same deflections and the reaction forces are captured. A uniaxial fender FEM was also created using the UE1200x1200E3.1 elements. Comparing a uniaxial and biaxial fender composed of the same rubber element is the best way to gage the direct benefit of the adding the two elements. Comparing the MV1000x900B uniaxial fender with UE1200x1200E3.1 fender is not ideal because they are composed of different size and different rubber compound elements.

5.1 Sideway Failure of MV1000x900B Uniaxial Fender

To reproduce the sideway collapse in Figure 5-1 with the FEM, the face pad was deflected and rotated in a variety of directions. A coordinate system is shown in the bottom left corner of Figure 5-2 which will be used in this section to indicate the direction of the pad deflection. The sideway deformation shown in Figure 5-2 was produced with the FEM and characterizes the rubber deformation in Figure 5-1. The sideway deformation was produced in the FEM by subjecting the face pad to the following deflections: -0.4 m in the y-direction which is the normal axial direction to the pad, -0.55 m in the x-direction and -0.4 m in the z-direction.

The FEM indicates that substantial deflection in the z-direction is required in order to reproduce the sideway collapse that occurred during the long term testing. Prior to this modeling, it was assumed based on the picture in Figure 5-1, that the sideway collapse was the result of deformation only in the x-direction and that the fender resisted forces in the z-direction.

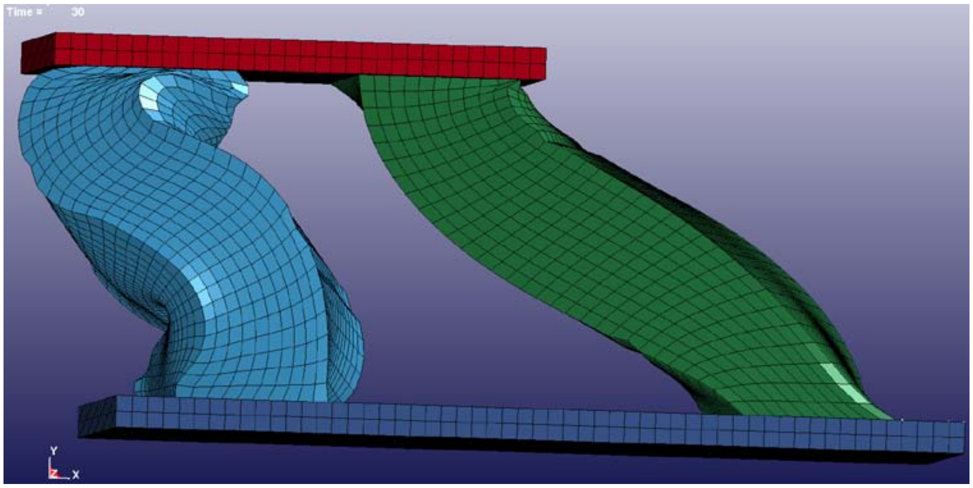

Figure 5-2. FEM Uniaxial Fender Sideway Collapse

5.2 Sideway Failure of UE1200x1200E3.1 Uniaxial Fender

The uniaxial FEM model with UE12000x1200E3.1 elements was subjected to the same deflections described in Section 5.1: -0.4 m in the y-direction which is the normal axial direction to the pad, -0.55 m in the x-direction and -0.4 m in the z-direction. The fender reaction forces in the x and z directions were computed with the FEM and plotted in Figure 5-3. The reaction forces in Figure 5-3 are the combined reaction of both rubber elements. The A curve represents the reaction force in the x-direction which is smaller in magnitude than the reaction force in the z-direction. This result is expected because the geometry of the fender allows it to resist higher forces in the z-direction versus the x-direction. The deformation of this FEM model is nearly identical to that shown in Figure 5-2 for the MV1000x900B elements.

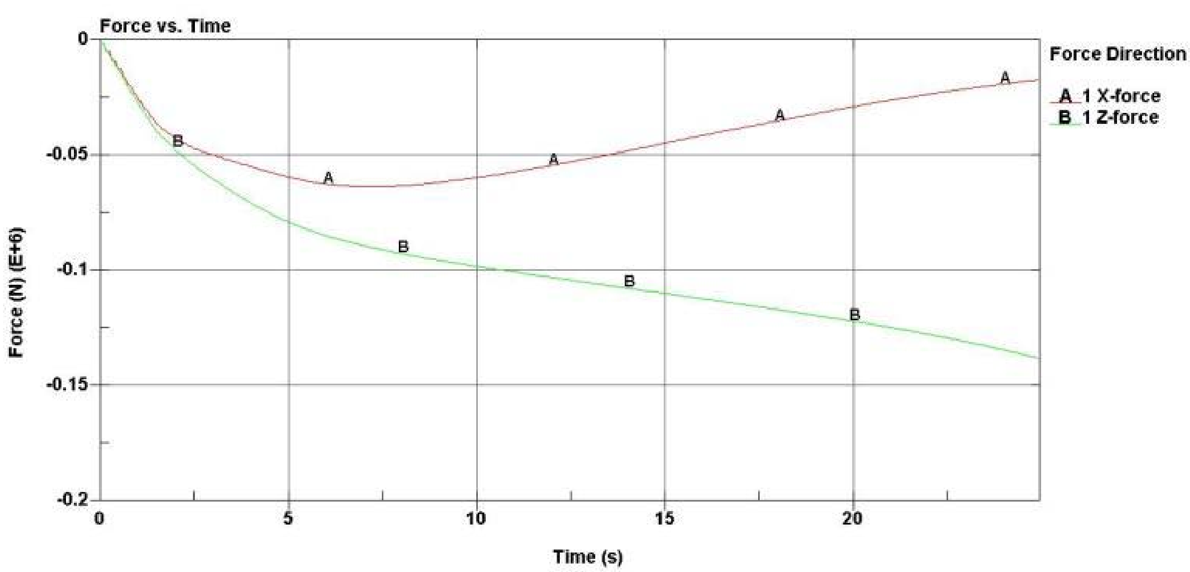

Figure 5-3. Uniaxial Fender Reaction Forces in x- and z-directions

5.3 Sideway Failure of UE1200x1200 Biaxial Fender

The biaxial fender configuration was designed to prevent the sideway buckling behavior by using four rubber elements. The performance was expected to improve due to the addition of two elements oriented to prevent sideways buckling. The FEM for the biaxial UE1200x1200E3.1 elements was subjected to the same deflections described in Section 5.1: -0.4 m in the y-direction which is the normal axial direction to the pad, -0.55 m in the x-direction and -0.4 m in the z-direction. The sideway collapse is shown in Figure 5-4. The combined fender elements reaction forces in the x and z directions were computed with the FEM and plotted in Figure 5-5. As expected, the reaction forces have increased, especially in the x-direction. The reaction force in the x-direction increased by a factor of 3 compared to the uniaxial fender presented in Section 5.1. The reaction force also increased in the z-direction even though the orientation of the two additional fenders does not provide significant resistance.

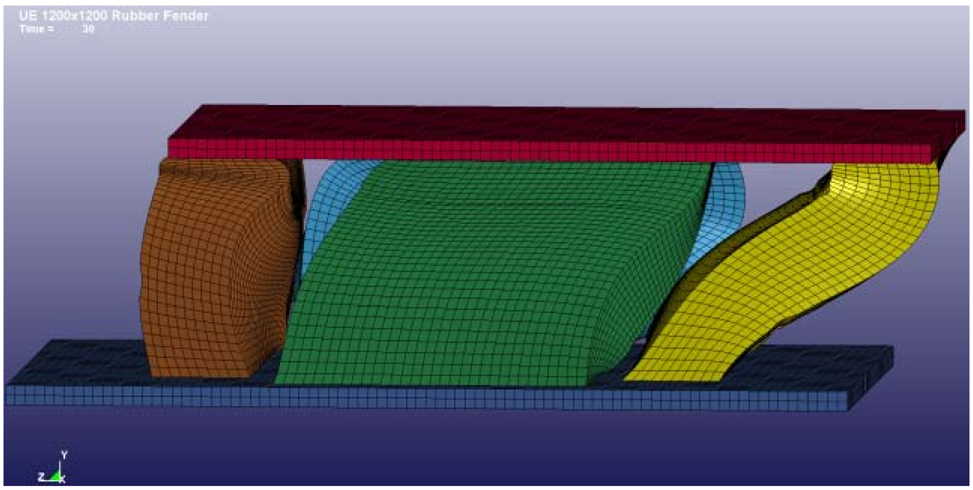

Figure 5-4. FEM Biaxial Fender Sideway Collapse

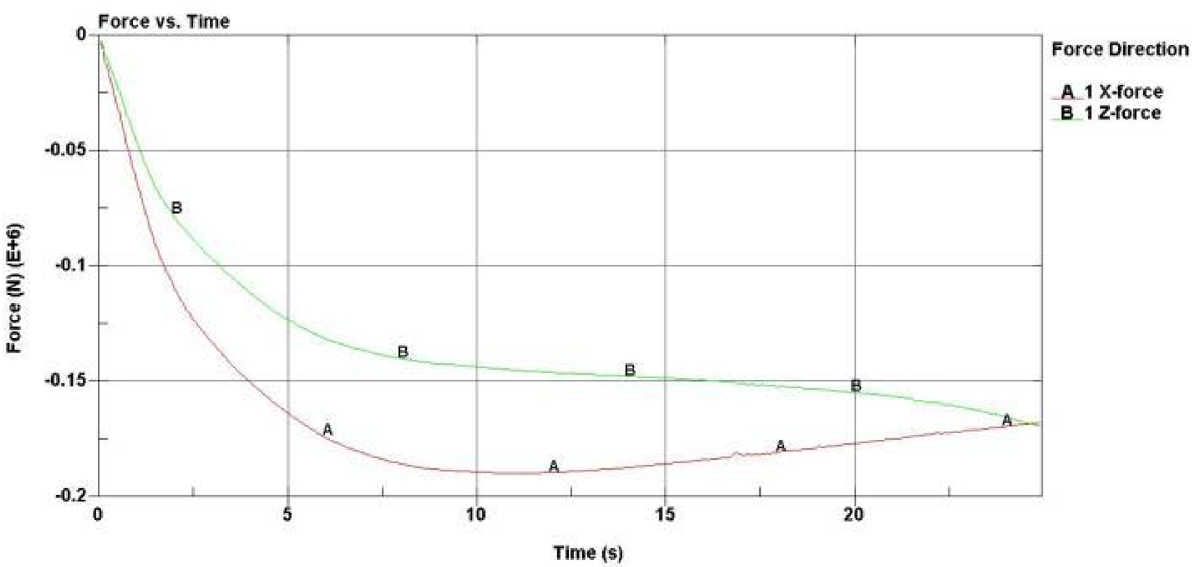

Figure 5-5. Biaxial Fender Reaction Forces in x- and z-directions

6.0 CONCLUSION/RECOMMENDATIONS

The biaxial fender configuration was designed to prevent the sideway buckling behavior by using four rubber elements. An improvement in performance was expected because two elements were added to provide greater resistance to prevent sideway buckling. The FEM for the uniaxial MV1000x900B indicates that substantial deflection in the direction less prone to buckle is required in order to reproduce the sideway collapse that occurred during the MHP long term testing. Prior to the modeling presented in this report, it was assumed based on the picture from the MHP long term testing, that the sideway collapse was the result of deformation only in the direction more prone to buckle.

The modeling reported in this report proves that the biaxial configuration does provide greater resistance in the direction which was more prone to buckle in the uniaxial configuration. However, the modeling also indicated that the uniaxial fender used in the MHP Test Bed collapsed in both the direction prone to failure and in the direction that provides greater resistance. While the UHMW pad is designed to minimize friction between the face pad and the stainless steel mooring shaft, the friction forces may still be too large for the proposed fender configuration. Careful analysis and testing will need to be carried out to determine the friction forces acting on the face pad and if the fender will have enough resistance to resist the forces. Alternative mooring concepts may also be considered if the biaxial fender configuration is determined to be not viable or impractical.

Finally, the tolerances between the fender elements in the biaxial configuration may have to be revisited. Small interferences between the elements could occur if the fender is fully compressed.

ACKNOWLEDGEMENT

Dr. Robert Zueck has provided advice on how to model the rubber material.

Dr. Ray Chiou has provided suggestions on how to develop and validate the FEM.

Catherine Kerrigan has provided help and advice during the material testing of the rubber samples.

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REFERENCES

7.0 BIBLIOGRAPHY

[1]. **BergerABAM.** *Preliminary Design Floating Double Deck Pier (FDDP), Volumes 1,2,3.* Contract N62583-09-C-0026, 2010.

[2]. **Trelleborg.** *Modular Fenders Section 1 and 2.* M1100-S01-V1.2-EN, M1100-S02-V1.2-EN.

[3]. **Jerry Lee, Elizabeth Schroth-Nichols, James D. Newall, Michael Gebman, Chia-Ming Uang.** *Modular Hybrid Pier Structural Capacity Tests: Long Term Fender Tests.* 2008.

[4]. **Hallquist, John O.** *LS-DYNA Theory Manual.* 2006.

[5]. **Livermore Software Technology Corporation.** *LS-DYNA Keyword User's Manual, Volume I.* Livermore : LSTC, 2007. Version 971.

[6]. *A tabulated formulation of hyperelasticity with rate effects and damage.* **Killing, S., Bois, P.A. Du and Benson, D.J.** 40:885-899, Computational Mechanics, 2007.

[7]. **ASTM International.** *Standard Test Methods for Vulcanized Rubber and Thermoplastic Elastomers-Tension.* 2008. Designation: D 412-06a..

[8]. **TA Instruments.** *Dynamic Mechanical Analyzer Operator's Manual.* 2000. DMA 2980.

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