The Effect of Fender Energy Dissipation on Moored Ship Wave-Induced Motions

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Abstract
This paper reports on a study into fender energy dissipation and its effect on moored ship motions in waves. Firstly, we analyze fender cyclic load tests, to find the amount of energy dissipated in each compression-decompression cycle. We then develop a method to model fender load through any compression-decompression cycle, using a velocity factor based on compression rate. The method is implemented in MoorMotions, a nonlinear time-domain solver for moored ship motions and loads. The results are applied to the GNSS-measured test case of a bulk carrier moored at Berth 5, Geraldton, in a large-swell event. It is found that the inclusion of fender energy dissipation tends to decrease horizontal ship motions by 3-5% and decrease fender compressions by 5-10%.

Keywords: moored ship motions, cyclic loading of fenders.

Nomenclature

CoG	Centre of gravity
DoF	Degrees of freedom
GNSS	Global navigation satellite system
UHDPE	Ultra-high-density polyethylene

1. Introduction

Specifications for fenders at port berths are chosen primarily using the fenders' energy absorption and rated reaction force. The fenders' energy absorption must be able to withstand a ship's arrival berthing energy. The fenders' rated reaction must be able to withstand the compression force produced by changing loads on the ship, such as wind and current loads.

For a berth subject to wave action, a moored ship has cyclic motions. A moored ship's natural motion periods in the horizontal modes (surge, sway and yaw) are typically 50 – 200 seconds, in the "long wave" range of wave periods ([1], pp.21-22). If long waves are present, and especially if the long waves are amplified by an enclosed harbour, resonant ship motions can occur, leading to large ship motions and large mooring loads. In the Port of Geraldton, large vessels are only permitted to moor in long waves up to 0.12 – 0.15 m, when using standard mooring lines.

The amount of damping in the moored ship system is important for minimizing moored ship motions and loads. Mechanical damping (from fenders, mooring lines and any motion damping equipment) may become especially important at long motion periods, where hydrodynamic damping is small.

In this paper, we attempt to quantify the amount of damping through a fender compression-decompression cycle. We then build this damping into a nonlinear dynamic mooring analysis, to assess the importance of fender damping on moored ship motions in waves.

2. Test case: Sea Diamond at Berth 5

In this article, we shall use the full-scale test case of MV Sea Diamond, moored at Geraldton Berth 5, as measured using GNSS equipment on 1st and 2nd October 2015 in large-swell conditions. This test case is described in [2,3].

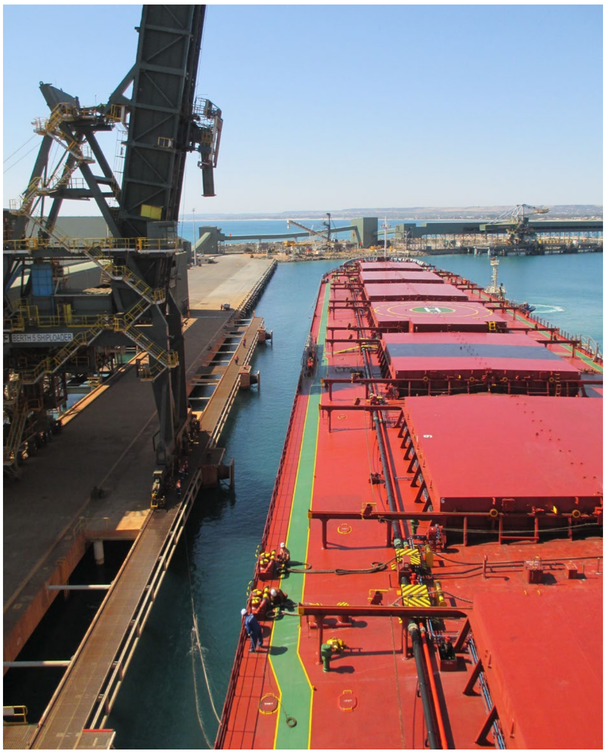

Figure 1 Sea Diamond arriving at Geraldton Berth 5, 1st Oct 2015. Trelleborg Super Cone fenders and low-friction facing are visible.

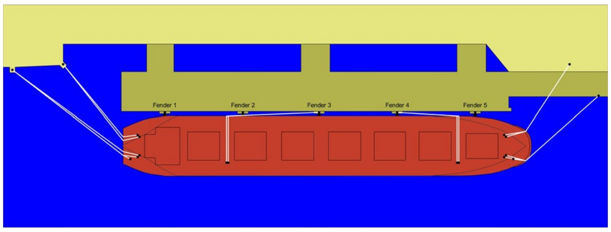

Figure 2 Mooring arrangement, for Sea Diamond at Geraldton Berth 5, 1st – 2nd Oct 2015. Fenders 2, 3 and 4 are in contact with the hull.

We shall use measured ship motion data from the last hour before departure, which is at the swell limit for this berth, as described in [3].

3. Fender compression for moored ship test case

Fender compressions for the Sea Diamond test case are shown in Figure 3. Fender compressions were calculated using MoorMotions software over a 1-hour time interval, and validated against 6-DoF ship motions, as described in [3].

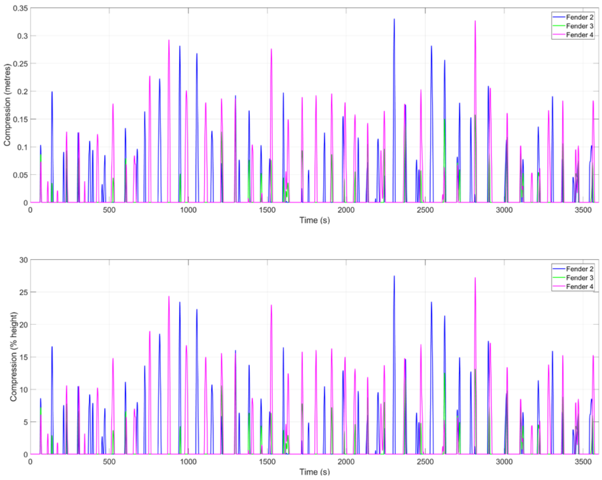

Figure 3 Fender compressions for Sea Diamond test case. Fenders are numbered as shown in Figure 2. (Top) Compression in metres; (Bottom) Compression as percentage of fender height (1.200m).

Fender compressions may also be plotted to include negative compressions, corresponding to ship distance off the fender. These are shown in Figure 4, for the same test case.

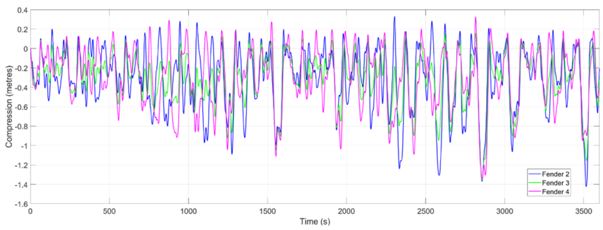

Figure 4 Fender compressions for Sea Diamond test case, including negative compressions (distance off each fender)

Fender compression rates for the same test case are shown in Figure 5.

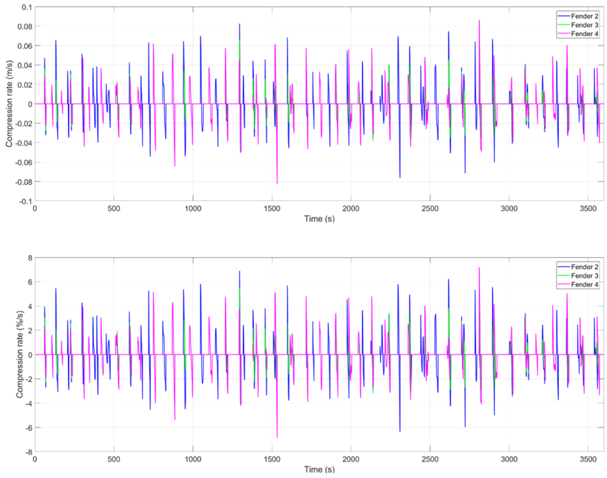

Figure 5 Fender compression rates for Sea Diamond test case. (Top) Compression rate in metres per second; (Bottom) Compression rate as percentage of fender height, per second.

We see that the maximum rate of fender compression is 0.08 m/s. This is around half the berthing impact velocity of 0.15 m/s typically used in fender design for large ships [4].

4. Cyclic load testing

As can be seen from Figure 3, the wave-induced motions of the Sea Diamond test case produce around 40 fender compressions per hour, with amplitude up to 30% of the fender height. This cyclic fender loading has fatigue implications for the fenders. For example, if the berth has 50% occupancy over the year, 40 compressions per hour translates to 175,000 compressions per year.

For ports which are affected more by short-period waves, the number of fender compressions per hour can be much higher than at Geraldton. Fenders may experience millions of compressions over their lifetimes, almost all of which will be at low deflection (<35%). Therefore, durability of the fender is a primary concern. The fender designer should ensure the quality and longevity of fenders, by designing bespoke cyclic load testing programs, in consultation with the manufacturer.

5. Cyclic load testing for 400 mm cell fender

Trelleborg undertook cyclic load testing on a 400 mm cell fender, for 25,000 cycles at 50% compression and 30% shear, with cycle time 10 seconds [5]. An example measured compression-decompression curve is shown in Figure 6.

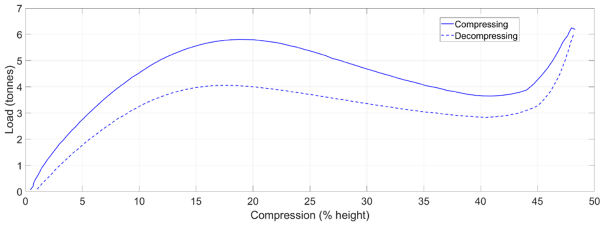

Figure 6 Compression-decompression curve after 1778 cycles, for Trelleborg SCK400 cell fender. Data from [Trelleborg14, Fig. 5].

We see that the fender is not perfectly elastic; the load when decompressing is less than when compressing. Work is done on the fender through a compression-decompression cycle. The excess energy is primarily converted to heat in the rubber. This effect is described in ([6], p.358).

The "loss factor", or "tangent delta", is a measure of the amount of energy lost per cycle during deformation of an elastomer. It is the ratio between energy lost through a compression-decompression cycle (loss modulus), to energy absorbed through the compression cycle (storage modulus).

The loss factor is strongly influenced by the choice of polymer. The addition of carbon black significantly increases the loss factor in rubber compounds. Plasticizers may slightly increase the loss factor, but in some cases, they can significantly reduce it. In natural rubber compounds, the choice of accelerator and the amount of sulphur in the cure system did not have much impact on the loss factor.

The measured loss factor for the SCK400 fender is shown in Table 1.

Table 1 Loss factor for SCK400 fender

Fender tested	*Loss factor*
Trelleborg SCK400 cell fender [Trelleborg14]	0.281

6. Cyclic load testing for 55.5 mm cell fender

Testing was done on small rubber cell fenders with 55.5 mm height and 80 mm diameter [7]. The test fenders were a compound of natural rubber and styrene butadiene rubber, with similar properties to modern cell fenders and cone fenders. The fenders were subjected to cyclic loading and the load-compression curves measured. Since fender damping is primarily a property of the material and strain rate, measured damping should be applicable to larger fenders. Results are shown in Figure 7.

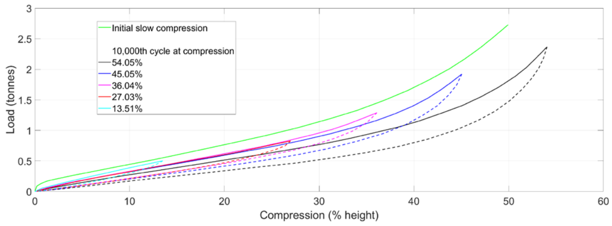

Figure 7 Compression-decompression curves for 55.5 mm cell fender. Solid line = compression. Dashed line = decompression. Data from ([7], Fig. 2 and Fig. 7).

We see that cyclic compressions soften the fender over time, with the large-amplitude compressions having more softening effect.

The loss factor is shown in Table 2.

Table 2 Loss factor for 55.5 mm cell fender

Fender tested	*Loss factor*
Chin-Cheng 55.5 mm cell fender [7]	0.217 – 0.265

7. The Velocity Factor

Most fender manufacturers publish load-compression curves for their fenders. These are typically measured under very slow-speed compression. An example for Trelleborg cell fenders is shown in Figure 8.

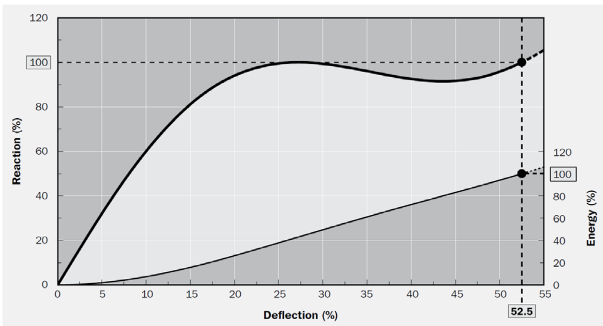

Figure 8 Load-compression curve for Trelleborg cell fenders, from ([8], p.21).

For faster compression, the reaction force is higher; a Velocity Factor is applied to the rated load-compression curve. For Trelleborg cell fenders, the Velocity Factor is shown in Table 3.

Table 3 Velocity Factor for Trelleborg cell fenders ([8], p.22)

Compression time (s)	*Compression rate (%/s)*	*Velocity Factor*
1	52.5	1.20
2	26.3	1.16
3	17.5	1.14
4	13.1	1.13
5	10.5	1.11
6	8.8	1.10
7	7.5	1.09
8	6.6	1.09
9	5.8	1.08
10	5.3	1.07
11	4.8	1.07
12	4.4	1.06
13	4.0	1.06
14	3.8	1.05
15	3.5	1.05
16	3.3	1.05
17	3.1	1.04
18	2.9	1.04
19	2.8	1.04
20	2.6	1.03

The Velocity Factor is given in terms of the "compression time" from initial impact to the rated deflection (52.5% for Trelleborg cell fenders). We have also shown the "compression rate" as an additional column in Table 3. This is calculated as the *average* compression rate (rated deflection divided by compression time). However, we shall also use it as the *instantaneous* compression rate, to specify the fender load in terms of compression and compression rate, for dynamic mooring analysis. That is, we calculate the "slow-speed" fender load from the published load-compression curve, then multiply this by the appropriate Velocity Factor, based on compression rate.

For the cyclic load tests shown in Figure 6, the compression rate is sinusoidal in time. Using the measured compression rate, the corresponding Velocity Factor (from Table 3) is shown in Figure 9.

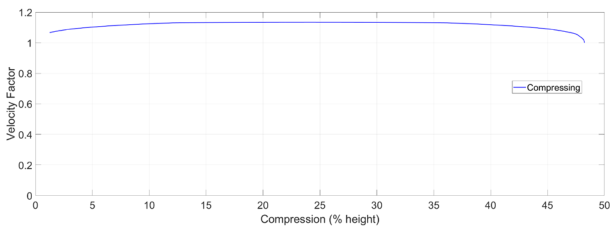

Figure 9 Calculated Velocity Factor for SCK400 fender during compression phase of cyclic load testing

We can divide by the Velocity Factor to calculate the equivalent slow-speed compressing curve for the SCK400 measurements, as shown in Figure 10.

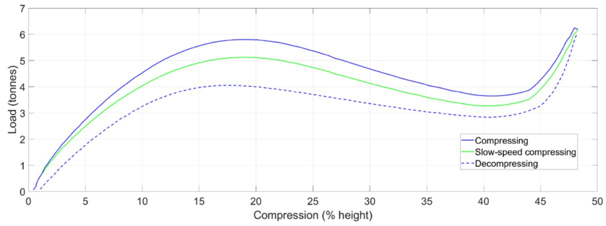

Figure 10 Compression-decompression curve for Trelleborg SCK400 cell fender during cyclic load testing, together with equivalent slow-speed compressing curve

Velocity Factors are generally not published for negative compression rates (while the fender is decompressing). Here we shall use Figure 10 to calculate Velocity Factors for the SCK400 fender while decompressing. The corresponding Velocity Factor is shown in Figure 11.

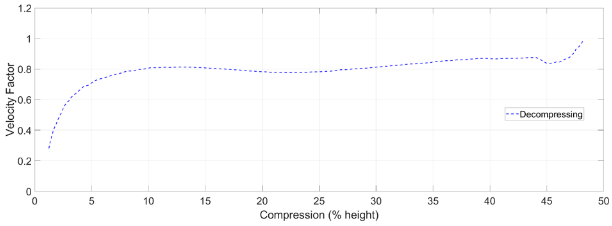

Figure 11 Calculated Velocity Factor for SCK400 fender, during decompression phase of cyclic load testing

We see that the Velocity Factor while decompressing is approximately 0.8, over most of the decompression cycle. Combining Figure 9 and Figure 11 gives the Velocity Factor over the whole compression-decompression cycle, as shown in Figure 12.

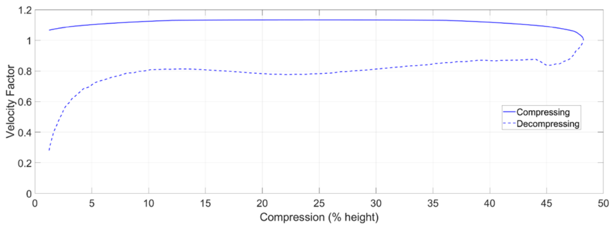

Figure 12 Velocity Factor for SCK400 fender, over complete compression-decompression cycle

In order to use the Velocity Factor in a dynamic mooring analysis, we aim to specify it in terms of compression rate, as shown in Table 3 for positive compression rates. For decompression, plotting the Velocity Factor against compression rate gives the results shown in Figure 13. The lower 25% of fender compressions are not shown, as these produce very small loads.

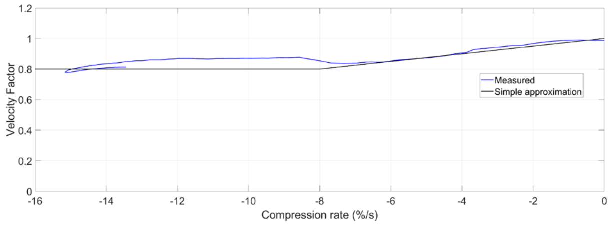

Figure 13 Calculated Velocity Factor for SCK400 fender while decompressing, plotted against compression rate. A simple approximation to the measured curve is also shown.

The Velocity Factor can now be specified in terms of positive compression rates (using Table 3) and negative compression rates (using Figure 13). The result is shown in Figure 14. It is important that the Velocity Factor is a continuous function of compression rate, to ensure smooth behaviour of time-domain dynamic mooring analysis.

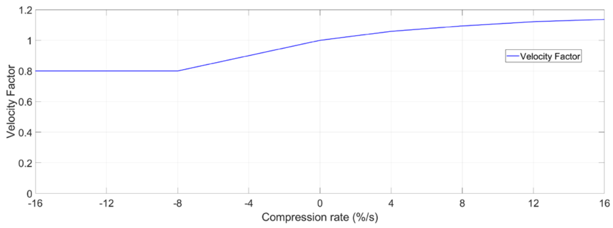

Figure 14 Velocity Factor for SCK400 fender, as a function of compression rate

8. Application to fenders at Geraldton Berth 5

Geraldton Berth 5 uses Trelleborg Super Cone SCN1200 E1.1 fenders, which have the same characteristics as the F1.1 fenders described in ([8], p.9). The fenders are fitted with low-friction UHDPE facing panels. The slow-compression load-compression curve for these fenders is shown in Figure 15.

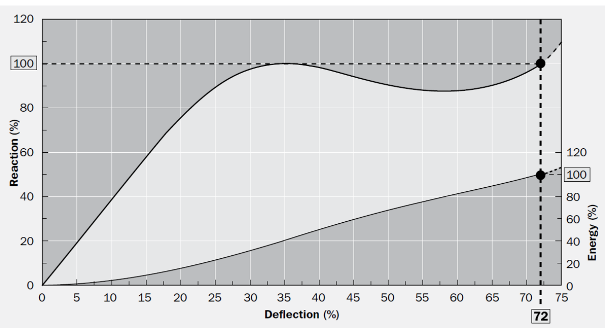

Figure 15 Load-compression curve for Trelleborg Super Cone fenders, as used at Geraldton Berth 5

The Velocity Factor for positive compression rates is shown in Table 4.

Table 4 Velocity Factor for Trelleborg Super Cone fenders ([8], p.12).

Compression time (s)	*Compression rate (%/s)*	*Velocity Factor*
1	72.0	1.20
2	36.0	1.16
3	24.0	1.14
4	18.0	1.13
5	14.4	1.11
6	12.0	1.10
7	10.3	1.09
8	9.0	1.09
9	8.0	1.08
10	7.2	1.07
11	6.5	1.07
12	6.0	1.06
13	5.5	1.06
14	5.1	1.05
15	4.8	1.05
16	4.5	1.05
17	4.2	1.04
18	4.0	1.04
19	3.8	1.04
20	3.6	1.03

Again, we have included an extra column of “compression rate”, which is the average compression rate (rated deflection divided by compression time).

For negative compression rates, we do not have Velocity Factor data for the Super Cone fenders. However, noting that Super Cone fenders use the same blend of natural and synthetic rubber as the cell fenders analyzed in Section 7, and that Velocity Factor should be primarily a function of the material and strain rate, we shall use the same Velocity Factor as developed in Figure 13.

The resulting Velocity Factor, over the complete range of compression rates, is shown in Table 5.

Table 5 Modelled Velocity Factor for SCN1200 fenders, as used in dynamic mooring analysis for Geraldton Berth 5

Compression rate (%/s)	*Velocity Factor*
-16	0.80
-12	0.80
-8	0.80
-4	0.90
0	1.00
4	1.04
8	1.08
12	1.10
16	1.12

9. Time-domain moored ship motions and loads

The nonlinear time-domain solver MoorMotions (www.moormotions.com) was used to study the effect of fender damping for the Sea Diamond test case. MoorMotions uses the fourth-order Runge-Kutta time-stepping method ([9], p.710) to solve the equation of motion ([10], eq.4.23):

$$\sum_{j=1}^{6}\left[M_{ij}+A_{ij}(\infty)\right]\ddot{x}_j = X_i^{(1)}+X_i^{(2)}+F_i^{(\text{lines})}+F_i^{(\text{fenders})} - \sum_{j=1}^{6} C_{ij}x_j + B_i^{(\text{viscous})} - \int_0^\infty \sum_{j=1}^{6} L_{ij}(\tau)\ddot{x}_j(t-\tau)d\tau$$

The symbols are defined as follows:
x_j=motion in each degree of freedom, $j=1,\ldots,6$
M_{ij} =mass matrix
$A_{ij}(\infty)$ =added mass at infinite frequency
$X_i^{(1)}$ =first-order wave load
$X_i^{(2)}$ =second-order wave load
$F_i^{(\text{lines})}$ =net force produced by mooring line tension at each instant in time
$F_i^{(\text{fenders})}$ =net force produced by fenders at each instant in time
C_{ij} =linear restoring coefficients
$B_i^{(\text{viscous})}$=additional viscous damping (e.g. roll)
$L_{ij}(\tau)$ =hydrodynamic impulse response functions

The coordinate system used is:
x_1= "surge" (fore-aft CoG motion, positive forward)
x_2 = "sway" (transverse CoG motion, positive port)
x_3 = "heave" (vertical CoG motion, positive up)
x_4 = "roll" (angle, positive to starboard)
x_5 = "pitch" (angle, positive bow-down)
x_6 = "yaw" (angle, positive bow-to-port).

Further details, together with basic MoorMotions settings for the Sea Diamond test case, are described in [3].

10. Calculations for Sea Diamond test case

Here we calculate moored ship motions and loads for the Sea Diamond test case, with and without fender energy dissipation. All settings are as used in [3], except as modified here. A line pre-tension of 0.5 tonnes is used in all lines.

We firstly do "free decay" calculations, without and with fender damping. Results are shown in Figure 16 and Figure 17.

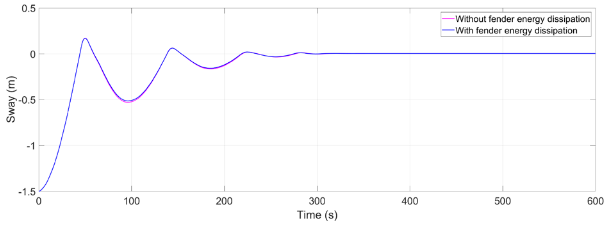

Figure 16 Sway free decay test, without and with fender energy dissipation

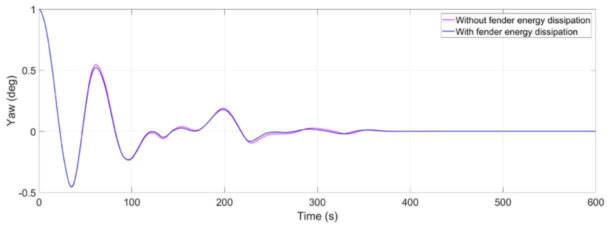

Figure 17 Yaw free decay test, without and with fender energy dissipation

We see that fender energy dissipation has a small effect on the free-decay motions, reducing the sway amplitude after one cycle by 3% and reducing the yaw amplitude after one cycle by 4%. Clearly, other damping mechanisms (principally hydrodynamic damping) have a dominant effect on the sway and yaw motions.

We now run the same test case as described in [3], for Sea Diamond moored at Berth 5 in large-swell conditions. Results showing the effect of fender energy dissipation are shown below. Results are averaged over 10 runs with different input wave phasing.

Table 6 Comparative moored ship motions for Sea Diamond test case. Results are maximum peak-to-peak values in a one-hour period.

	Without fender energy dissipation	*With fender energy dissipation*	*% change*
Surge	2.776 m	2.692 m	-3%
Sway	1.557 m	1.473 m	-5%
Heave	0.182 m	0.182 m	0%
Roll	0.413°	0.411°	0%
Pitch	0.142°	0.144°	+1%
Yaw	1.360°	1.318°	-3%

Table 7 Comparative moored ship line loads for Sea Diamond test case. Results are maximum values in a one-hour period.

	Without fender energy dissip-ation	*With fender energy dissip-ation*	*% change*
Stern lines	5.30 t	5.38 t	+1%
Aft breast	12.97 t	12.68 t	-2%
Aft spring	17.91 t	16.90 t	-6%
Fwd spring	9.21 t	9.15 t	-1%
Fwd breast	14.15 t	13.05 t	-8%
Head lines	6.41 t	6.35 t	-1%

Table 8 Comparative moored ship fender compressions for Sea Diamond test case. Results are maximum values in a one-hour period.

	Without fender energy dissip-ation	*With fender energy dissip-ation*	*% change*
Fender 2	0.348 m	0.332 m	-5%
Fender 3	0.199 m	0.188 m	-6%
Fender 4	0.375 m	0.340 m	-9%

11. Conclusions

The conclusions from this study are as follows:

- Cyclic loading and fatigue are important issues for fenders in wave-exposed ports, as the fenders can have hundreds of thousands of compressions per year.
- Conventionally, a fender system selection and design process considers fender compression for a small number of cycles (3000 as per PIANC guidelines). Designers must take into consideration the huge number of low-compression cycles a fender system has to go though in permanent mooring situations. Low compression with high frequencies generates heat in the fender body which has impacts on the ageing process of a rubber fender. Hence, selection of proper height, compound and grade of fenders becomes crucial.
- Fender impact velocities, while a ship is moored in waves, are comparable to initial berthing velocities.
- Fenders are not perfectly elastic, but dissipate some energy as heat, through each compression-decompression cycle.
- The fender damping effect can be included in a dynamic mooring analysis, by using a Velocity Factor, together with the published load-compression curve.
- The Velocity Factor needs to be a continuous function of compression rate. It is greater than 1 for positive compression rate, and less than 1 for negative compression rate.
- The Velocity Factor may be found from cyclic load testing of fenders. For Trelleborg rubber fenders, an appropriate Velocity Factor has been developed in this report and applied to a full-scale test case in Geraldton.
- Calculations show that for the Geraldton test case, fender energy dissipation decreases the horizontal ship motions by 3-5% and decreases the fender compressions by 5-10%.
- Unused, freshly cured fenders were used for the experimental data in this report. Some of the above conclusions may differ for used fenders. Similar studies should be conducted using naturally or artificially aged fenders, which was beyond the scope of this paper

12. References

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