The dynamic behavior of wharf with rubber fender subjected to berthing impact

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Abstract

In this study the dynamic behavior for a bridge type wharf system incorporated with typical HP or \( \pi \) type rubber fender system is studied when subjected to berthing impact of large cargo ships. Firstly, the type and size of the fender and the material behavior are taken into examination and comparisons are made between the modeling results and the experimental data. Through a simplified material model for the selected type of fender, the behavior of the material can be predicted during the analysis of the structural movement. The traditional analytical method of energy absorption from the fender to estimate the defection and the reactive force to the wharf structure is modified by applying the momentum effect of the ship berthing. Then the dynamic effect can be evaluated through the duration in addition to the magnitude of the exerting reactions on the wharf. It is found that for the same input energy of ship berthing the momentum plays an important role too. Particularly for the ship with heavier weight but less berthing speed, the larger momentum causes larger vibration amplitude on the wharf. For the berthing motion with same energy and momentum, the reactive capability of the fender may play a more important role and this is widely dependent on the type of fenders.

1 Introduction

To avoid damages of wharf systems induced from the impact of ship berthing
action, many fenders have been designed and applied to the wharf systems [8]. Gravity fender is the one with large mass by converting kinetic energy into potential energy of the fender when the mass was lifted. Buoyancy fender of a hollow body would transfer the kinetic energy into potential energy by submerging to the water. Spring fender is the one that employs elastic compression of a steel spring to convert the kinetic energy of the ship berthing. Pneumatic fenders including air block fenders, floating fenders and pneumatic tire fenders also convert kinetic energy into potential energy by elastic compression. Buckling fenders are mostly made of rubbers, which convert the energy through the combination of bending and compression of the rubber members when subjected to the berthing impact. There are other many types of fenders, which either categorized by its reaction mechanism or by the material element such as the shear fender, hydraulic fender and torsion fender etc. Among those fender systems the buckling fender of rubber material is one of the mostly applied fenders in the harbor.

Even though fender is not a primary member for the wharf structural system it still plays an important role in ensuring the safety of both the wharf structure and the hull of vessels. For the design of the fenders, especially for the buckling type of rubber fender, restrictions are set for the reaction forces and absorption capacity of energy. The limitations are maximum for the reaction and minimum for the energy absorption of the rubber fender. The design process is that according to the estimation for the type, size and berthing speed of the ship, the berthing energy is calculated and set for the fenders and then the maximum reaction force is set for the ship for which the hulls can take under the elastic conditions. However, the process to select fenders that can fit in these two conditions simultaneously will be difficult without sufficient testing data of the fender.

One other concern is that even though the berthing energy is being estimated, with the same energy of ship berthing the wharf can have different dynamic responses. It is found that for the same input energy of ship berthing the momentum plays an important role too. Particularly for the ship with heavier weight but less berthing speed, the larger momentum causes either larger vibration amplitude or longer vibration duration on the wharf. For the berthing motion with same energy and momentum, the reactive capability of the fender is widely dependent on the type of fenders. Therefore, in this study the traditional analytical method of energy absorption capacity from the fender to estimate the deflection and the reactive force for the wharf structure was modified by applying the momentum effect of the ship berthing. Then the dynamic effect can be evaluated through the duration in addition to the magnitude of the exerting reactions on the wharf. Because the larger exerting force induces larger amplitude of vibration the limit set for the reactive capability is not only to prevent the hulls of the ship from permanent deformation but also to prevent the wharf from over-vibrations. Therefore, if the wharf is more sensitive to longer duration of vibration, the momentum induced from berthing activity will be needed to take into accounts for the design of the fender system.
2 Material properties and design of rubber fender

The rubber fender is the most typical one of the buckling type fenders. When the fender subjected to the impact of berthing vessels it will buckle that combines the compression and bending together. However, when testing the characteristics of the fender that was treated as a unit, generally only the compression behavior was taken into consideration. This type of fender can be made out of many shapes and combinations such as the D-shape, O-shape, V-shape, \( \pi \)-type, F-type and etc. They have many advantages such as easy installation and maintenance and good property for energy absorption even though it is sensitive to sunlight and chemical pollution. As we know the natural rubber has nonlinear elastic mechanical characteristics. The elongation of the vulcanized natural rubber can reach up to 500\%, even higher, to 700\% [2], which makes rubber a good material as the dock fender.

For the dock fenders, usually two restrictions are imposed for the design requirement, namely, the minimum capacity of energy absorption and the maximum reaction force. These reaction forces are limited under conditions that during the normal berthing of the ship no plastic or permanent deformation of ship’s hull should take place. The energy absorption capacity for the fender, which obtained from the integration of the reaction-deflection curve, should be larger than the one caused from the berthing impact. Since there are all different size, shape and even material basis, these mechanical properties must be determined by actual testing of the fender unit.

![Figure 1 Schematic drawing of \( \pi \)-type (or HP-shape) rubber fender](image)

A schematic drawing of the typical \( \pi \)-type (or HP-shape) rubber fender categorized as fender for wharf of large ship is shown in Figure 1, where the rubber is framed in the steel plate and held by two thin rubber legs, which will buckle during the impact of ship’s berthing. The size of each component can be varied corresponding to the merchandise category. A set of typical reaction
force-deflection curves obtained from the monotonic static testing results was shown in Figure 2 (a), (b) and (c) corresponding to each single fender unit testing, namely, type HP-1250, HP-1500 and HP-1750. An additional curve indicating the energy absorption capability was also shown in dot-line in the figure. The nonlinear-elastic behavior can be observed that after reaching the proportional limit of the linear-elasticity, the deflection continues while the loading force remains and even drops a little and after it reaches to certain point the loading will go up again combing with some hardening effect.

In order to incorporate this fender into the wharf system, a simple model of polynomial function of 2.5 degree was employed to simulate the force-deflection curve of monotonic static testing.

\[
\sigma(\varepsilon) = a_1(\varepsilon)^{0.5} + a_2(\varepsilon) + a_3(\varepsilon)^{1.5} + a_4(\varepsilon)^2 + a_5(\varepsilon)^{2.5}
\]

where \(\sigma(\varepsilon)\) is the normal stress, \(\varepsilon\) is the normal strain, \(a_1\) to \(a_5\) are material parameters obtained from the material testing. The relationship between the reaction force \(R\) and the deflection \(\Delta\) can further be obtained from this equation by applying the area of cross section of two legs, \(A\) and the height of the legs, \(h\). For these three types of fender, each of which has different size of the components, we may use the same set of parameters to simulate the mechanical behavior of the fender unit as shown in solid-line in Figure 2 (a), (b) and (c). The comparison between the simulation and the testing results is fitting well each other for these three different sizes of fender unit. The related parameters are also shown in Table 1
Table 1 The parameters for the nonlinear model

<table>
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<th>parameters</th>
<th>$a_1$</th>
<th>$a_2$</th>
<th>$a_3$</th>
<th>$a_4$</th>
<th>$a_5$</th>
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<td>0.00868</td>
<td>-0.0891</td>
<td>0.65689</td>
<td>-1.2996</td>
<td>0.77878</td>
</tr>
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</table>

The absorption capacity of the energy (temporary stored energy) of the fender is further expressed by the integration of the force-deflection curve as

$$E(\Delta) = \int R(\Delta) d\Delta$$  \hspace{1cm} (2)

Once the relationship between the deflection and the energy is known, we may
find the deflection corresponding to the energy required for the wharf design and then further find the corresponding force from Equation (1). Compared to the usual means of design by checking the curves as Figure 2 supplied by the manufacturer, this method will be more specific and ready to apply digitally.

The other advantage is that corresponding to the momentum induced from the same berthing impact the loading duration can be estimated since the testing for the load-deflection curve employed a constant strain rate, which is generally unavailable. With the loading-deflection curve and the loading duration available the complete time history during the berthing process of a ship is obtained and the dynamic response of the wharf can also be found associated with the given structural properties.

3 Dynamic response of wharf subjected to berthing impact

The vibration of the wharf induced by the ship berthing leads to the design of the fender, which is based on the input energy of ship berthing. For the berthing maneuver, usually the ship approaches wharf with translation $U_Q$ and rotation $W_Q$. For a ship with radius of gyration $k$ and mass $m$, the kinetic energy due to the ship berthing may be presented as

$$E = \frac{1}{2} m u_0^2 \left( 1 - \frac{a^2}{k^2 + r^2} \right) + m w_0^2 \frac{k^2 a}{k^2 + r^2} + \frac{1}{2} m w_0^2 \frac{k^2 r^2}{k^2 + r^2}$$

where $r$ is the distance from the center of mass to the contact point of the fender while $a$ is the vertical distance of the berthing velocity from the center of mass to the contact point. The first term in Equation (3) represented the kinetic energy from the translation motion while the third term in the equation is from the rotation. If the ship berths by allowing only translation and letting the distance $r$ be as close as possible to $a$, the kinetic energy of ship berthing may be simplified as

$$E = \frac{1}{2} m u_0^2 \left( \frac{k^2}{k^2 + a^2} \right) = \frac{1}{2} m u_0^2 \left( \frac{1}{1 + (a/k)^2} \right)$$

where a coefficient of kinetic energy of berthing is defined as $\frac{1}{1 + (a/k)^2}$. When this coefficient is larger the kinetic energy will be smaller.

The dynamic equation of motion for the engineering structural member with mass $M$, structural damping $C$, and stiffness $K$, subjected to the impact and wave forces propagated normally to the structural member can be written as

$$M \ddot{X}(t) + C \dot{X}(t) + K X(t) = P_w(t) + P_f(t),$$

(5)
where $\ddot{X}(t)$, $\dot{X}(t)$ and $X(t)$ are the acceleration, velocity and displacement of the wharf structural system. Taking into account of the relative motion between the structures and fluids, the wave forces exerted on the body, $P_w(t)$ [3, 5, 9] are

$$P_w(t) = \rho C_m V^* \dot{U}_n(t) - \rho C_a V^* \ddot{X}_n(t) + \frac{1}{2} \rho C_d A^* \left[ U_n(t) - \dot{X}_n(t) \right] \left[ \dot{U}_n(t) - \ddot{X}_n(t) \right]$$  

(6)

where $C_a = C_m - 1$, and $U_n$ and $\dot{U}_n$ are the velocity and acceleration of the fluid normal to the structural member resulted from the horizontal and vertical motion of the fluid, respectively. $\ddot{X}_n$ and $\dot{X}_n$ are the acceleration and the velocity of the structural member in the normal direction. $C_m$ and $C_d$ are coefficients corresponding to inertia and drag effect respectively. $V^*$ and $A^*$ are the displaced volume and the projected front area of the structural member, respectively. The last term in the equation representing the drag force due to the relative velocity of fluid is nonlinear. The nonlinearity of the drag term is modified through the use of the approximate relation derived by Penzien and Tseng [7],

$$P^*_w(t) = \rho C_m V^* \dot{U}_n(t) - \rho C_a V^* \ddot{X}_n(t) + \frac{1}{2} \rho C_d A^* \left[ \dot{U}_n(t) - \ddot{X}_n(t) \right] \left[ \dot{U}_n(t) - \ddot{X}_n(t) \right]$$  

(7)

where $\dot{U}_n$ represents the time average of $|U_n|$.

The impact force $P_I(t)$ is obtained from the loading history of the fender subjected to the ship berthing as

$$P_I(t) = \left[ \beta_1(t)^{0.5} + \beta_2(t) + \beta_3(t)^{1.5} + \beta_4(t)^2 + \beta_5(t)^{2.5} \right] B$$  

(8)

where $\beta_*$'s are parameters obtained through the modification of material parameters $a_1$ to $a_5$ from the material testing. $B$ is the matrix corresponding to the position of the loading and the entire structural system. Assuming that the rubber fender will react similar to the testing behavior, the loading history will similar to the one in the lab test. Thus the loading duration and rising time of the impact are known and thereafter can be applied to the dynamic analysis.

After the substitution of Equations (7) and (8) into Equation (5), and letting $K^* = K$, the dynamic equation of motion takes the form [4]as

$$M^* \ddot{X}(t) + C^* \dot{X}(t) + K^* X(t) = C_m \dot{U}(t) + C_d U(t) + P_I(t)$$  

(9)

where with the transformation matrix $B_1$,

$$M^* = M + \rho C_a V^* B_1$$  

(10)

$$C^* = C + \rho C_d A^* \dot{U}_n B_1$$  

(11)
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\[ C_m^e = \rho C_m V^e B_1, \]  
\[ C_d^e = \frac{1}{2} \rho C_d A^e |U_n| B_1, \]

For the bridge type wharf with HP-type rubber fenders, the dynamic analysis in the time domain was then carried out by using the step-by-step method [1, 6] and numerical examples were presented.

4 Numerical results and discussion

In the numerical analysis, a typical bridge type wharf structure of 2-bay located at a slope was adopted and redesigned as shown in Figure 3. The outer diameter and the thickness of the steel pile is 914 mm and 6.4 mm respectively. The piles were assumed clamped on the sea floor at the apparent fixity level. The length of the pile from the deck to the fixity level was assumed to be \( L_f = L + 6D \) with \( L \) and \( D \) as the original pile length and pile diameter respectively as customary application. When the data of the soil is absent a nominal apparent fixity depth is taken as \( 6D \). The density of the structural material and water are \( 7.8 \times 10^3 \text{ kg/m}^3 \) and \( 1.0 \times 10^3 \text{ kg/m}^3 \) respectively. A 2 ton/m² living load and 1.7 ton/m² dead load were assumed to be uniformly distributed on the deck.

Two analyses were carried out for the fender performance. Firstly, two types of fender were applied for the same ship berthing impact and comparison was made for the responses of the wharf. The next analysis is to examine the influence of the momentum effect of berthing impact on the same type of fender when the momentum is varied with respect to a constant kinetic energy.

In the first numerical analysis the loading was induced by a 20,000 D/W cargo berthing in a 10 cm/sec. speed exerted on the fender. As shown in Figure 4

![Figure 3. Schematic drawing of the bridge type wharf structure](image-url)
is the comparison for the displacement response of the wharf when the fender system is varied from HP-1250 to HP-1750. It shows that wharf with type HP-1750 fender system has smaller amplitude and shorter duration of vibration.

![Figure 4](image1.png)

**Figure 4** The displacement comparison for various fender installation

In the second analysis, corresponding to a constant kinetic energy induced by a 20,000 D/W cargo ship berthing in a speed of 15 cm/sec. (loading 2), a 40,000 D/W cargo ship will berth in 10.6 cm/sec. (loading 3) and 10,000 D/W in 21.2 cm/sec. (loading 1). It is noted that the momentum varied in a ratio of square root of the mass ratio. For the same kinetic energy, larger cargo ship has larger momentum effect, which is not considered generally in the fender design. Figure 5 shows the comparison of the response for the wharf of which the fender system subjected to the berthing impact of the ship is all the same as HP-1750. It is

![Figure 5](image2.png)

**Figure 5** The displacement comparison for various berthing momentum

noted that when the momentum is larger the vibration becomes more significant and lasts longer. If the ultimate response is already exceeded then the influence of this momentum induced duration effect will be minor. Nevertheless, if the ultimate response is not reached for smaller momentum berthing, then it can be reached for a larger momentum berthing.
5 Conclusions

According to the numerical results it was found that for the same input berthing energy the response of the wharf during the impact is varied corresponding to various fender system. For the wharf installed with same rubber fender the response of the wharf is varied corresponding to the same input energy but various momentum. Therefore, it is concluded that for a cautious design, particularly for the wharf subjected to larger and heavier ship berthing, a check on the momentum effect will be recommended. It is also suggested that when choosing the rubber fender system, the dynamic analysis for the largest momentum berthing may be essential for further protection of the wharf structures.

References