MODELING THE EFFECTS OF SHIP APPENDAGES ON THE SIX-DEGREE OF FREEDOM SHIP MOTIONS

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SUMMARY

Ship appendages such as bilge keels, rudders, passive fins, propeller shafts, etc. affect the six-degree of freedom ship motions. This becomes very significant when the incident force frequency is close to the ship's natural frequency. Therefore, it is important to accurately model the effect of these appendages when attempting to predict the six-degree of freedom ship motions.

In this study, we report an optimal approximation method for small scale appendages in potential flow based on the formulation developed by Lin and Kuang (2007b). In this method, we calculate the equivalent dynamic pressure using the effective blocking area of the appendages, instead of using direct surface integration. This method can provide accurate results and is generic for all ship hulls. It is also numerically efficient: only an order of $N\log N$ flops are needed (compared to on the order of N^2 flops for traditional direct integration).

However, this method depends on the time-varying effective blocking areas that can only be accurately evaluated with fully nonlinear six-degree freedom ship motion models, since it is a function of ship motion state, and the underwater ship geometry (both vary in time). Thus, it is implemented in the DiSSEL ship motion model (Lin et al, 2005; Lin and Kuang, 2006 and 2007a). In this study, X-Craft (FSF-1, SEAFIGHTER) is used to demonstrate the "Blocking Theory" and the impacts of the appendages on ship motions. The results show that the relative difference between the numerical calculations for the roll motion without appendages and the experimental data is approximately 35%. However, it is significantly reduced when the ship appendages are included.

The appendage-wave interaction also generates sub-scale flow structures that are beyond numerical model resolution. To account for such effects, we develop a methodology similar to a traditional "wave-breaking mechanism", with one explicit difference, the wave-breaking effect is considered based on the wave number (i.e. the spatial scale), instead of the wave slope. This new approach is tested with a landing craft, AAV7A1, which is a major transportation vessel for Marine Corps in the coastal regions. The numerical simulation shows that the modification to the ship motion due to the damping effects is on average less than 5%. Sometimes it can be substantially higher: e.g. 20% or above. These "wave breaking" effects increase as the water depth decreases. The numerical results from DiSSEL show that numerical simulations with the wave breaking effects agree better with the experimental data than those without.

1. INTRODUCTION

In order to numerically predict the six-degree of freedom ship motions accurately, appendage effects must be included in numerical modeling. In many cases, the appendage effects can be critical. For example, Hayden *et al* (1988, 1989) found in their experiments that the onset of "plow-in" occurred on the landing craft, AAV7A1, with a 4.2-knot speed. Plow-in is the submergence of the bow near hull speed. However, if there is a bow vane, the pitch instability resulting "plow-in" will not occur (Hoyt and Lin, 2007). The bow vane produces mainly a dynamics lift force. But its interaction with surface waves generates sub-scale flow structures which also help eliminate the pitch instability. Not much effort has been put into the study of such instabilities since the roll motion tends to dominate all other types of ship motion (e.g. pitch, yaw motions), and since the pitch instabilities are very difficult to model numerically.

There were several theoretical and experimental attempts to model the bilge keel roll damping effects. Himeno (1981) developed a linearized roll damping model that has been widely used. However, as Himeno himself pointed out, the roll damping is not yet fully understood. In particular, his linearized model cannot be used for large roll motion and fast ship speed. Therefore, until now many ship motion models, such as Large Amplitude Motion Program (LAMP) (Lin et al., 1990, 1994), could only use empirical approaches to determine the roll damping. As Engle and Lin (2007) pointed out, with these traditional methods for predicting roll

response, users must rely on previous experience with similar types of ships to select a proper level of critical damping. If the objective of design is to push the operational envelope, dependencies on empirical methods and/or precedent results can be lacking. Most recently, Lin and Kuang (2007b) developed a nonlinear blocking theory to predict the bilge keel damping effects. This method is generic for arbitrary ship hull and arbitrary environment, and thus robust in addressing the dynamic response of a ship in a seaway (Engle and Lin, 2007). However, it can only be applied for time varying ship motion models, and the accuracy of the roll damping is based on the fidelity of the predicted ship motions.

In this study, we intend to extend this "Blocking Theory" beyond bilge keels to study the damping effects of all small scale appendages, e.g. T-Foil, etc. We also develop a "Wave Breaking Theory" for modeling sub-scale effects arising from interaction of large scale appendages (e.g. bow vane, track) and surface waves. The "Blocking Theory" and the "Wave Breaking Theory" are described in Section 2. These methods are implemented in the DiSSEL ship motion model (Lin et al., 2005; Lin and Kuang, 2006 and 2007a). The numerical results are benchmarked and validated with experimental data from the David Taylor Model Basin, Seakeeping Division.

2. DAMPING METHODS

We define a "small" appendage if its scale is much smaller than that of ship hulls. An appendage is "large" if its scale is comparable to that of the ship hull. Since the damping effect depends on spatial scales, the damping effects of the appendages need to be analyzed differently. In our approach, we use the "Blocking Theory" and the "Wave-Breaking Theory" approaches for the effects associated with different scales.

2.1 BLOCKING THEORY FOR SMALL SCALE APPENDAGES

Lin and Kuang (2007b) developed a nonlinear damping method (called the "Blocking Theory" in this manuscript) to accurately obtain the roll damping of small scale bilge keels. The "blocking phenomena" is one of the basic physical processes in fluid mechanics and discussed in many textbooks (e.g. Landau and Lifshitz, 1987). This is the effect of a solid object on the passing-by fluid motion. It has been widely discussed in oceanic and atmospheric sciences. Lin and Kuang (2007b) modified this theory to study damping effect of small scale appendages.

We focus on the damping effect of the appendages on the ship rotational motions. The solid body rotation is described by the Louisville equation:

$$\mathbf{I}\cdot\frac{d\mathbf{\Omega}}{dt}+\mathbf{\Omega}\times(\mathbf{I}\cdot\mathbf{\Omega})+D_{rotat}\mathbf{\Omega}=\mathbf{\Gamma}_{rotat}^{I+D+R}+\mathbf{\Gamma}_{rotat}^{restore}, \tag{1}$$

where $\mathbf{I}$ is the moment of inertia, $\mathbf{\Omega}$ is the angular velocity ($\Omega_m=\frac{d\theta_m}{dt}$, θ_m is rotation angle; m=1,2,3, for roll, pitch, and yaw motion respectively), $\mathbf{\Gamma}_{rotat}^{I+D+R}$ is the pressure torque (including the incident waves, diffraction, refraction) on the ship body, and $\mathbf{\Gamma}_{rotat}^{restore}$ is the restoring torque from the buoyancy force. In this study, we focus on evaluating the damping term $D_{rotat}\mathbf{\Omega}$. For the other forcing terms, we refer the reader to Lin and Kuang (2007a).

The rotational damping can be evaluated from the torque arising from the pressure acting on the appendage blocking area A_{block} (Lin and Kuang, 2007b):

$$(\mathbf{\Gamma}_{block})_m=D_{rotat(m)}\frac{d\theta_{(m)}}{dt}=-\rho\int_{A_{block(m)}}(\mathbf{r}\times\mathbf{n})_m P_s ds. \tag{2}$$

Instead of directly calculating the integral (2), we use the following simple, but mathematically consistent approach: an "effective blocking area" A^* is introduced to the model. At every time step (in simulation), it is deducted from the total wetted surface A. The pressure torque $\mathbf{\Gamma}_{rotat}^{I+D+R}$ is then evaluated on the modified wetted surface (A-A^*). By this, the damping effect is included in the model. The effective blocking area A* is defined as

$$A^*=W_{block}*H_{sa}L_{sa}\sin(\alpha), \tag{3}$$

where H_{sa} and L_{sa} are the small appendage width and length, α is the angle between the length of the small scale appendage and the seaway, and W_{block} is a parameter describing the blocking effect and is given by

$$W_{block(m)}=\sum_{k=0}^{K}\frac{1}{2^{k+1}}(\theta_m\omega)^{2k}, \tag{4}$$

where

$$\omega = \sum_{i}^{N} \frac{A_i}{A} \omega_i \,, \tag{5}$$

is the weighted mean wave frequency of an incident wave with N modes, each mode with the wave amplitude A_i. A in (5) is the sum of the all wave amplitudes. This approach is generic such that it does not depend on particular ship hulls, or on specific incident waves. Therefore it can be applied to arbitrary ship hulls in an arbitrary environment. With this approach, the rotating angles θ_m and the corresponding angular velocity can be determined at each time step in the DiSSEL ship motion model. It should be pointed out that the blocking damping function defined in (4) and (5) includes the amplitudes and the frequencies of incident wave modes. The detail can be found from Lin and Kuang (2007b).

2.2 WAVE BREAKING THEORY FOR LARGE SCALE APPENDAGES

The "Wave Breaking Theory" is used to calculate sub-scale damping effects from the interaction of large appendages and incident waves. Unlike small appendages, the interaction of large appendages and surface waves can be well modeled by the pressure of the potential flow. However, due to numerical resolution, the sub-scale effect (i.e. that from flow not resolvable by numerical models) is not included in the pressure field. Therefore, we introduce an approach similar to the "wave breaking" mechanism to evaluate this sub-scale damping effect. The basic idea of the "wave breaking" theory is that wave breaking reduces the pressure force on the ship hull. Therefore in our approach, the damping coefficient in (1) is described by

$$D_{rotat} = \frac{\mathbf{\Omega}}{|\mathbf{\Omega}|^2} \cdot \iint_s ds(\mathbf{x}_s - \mathbf{x}_c) \times \mathbf{n} P_{wb}(x, y), \tag{6}$$

where $\mathbf{x}_s - \mathbf{x}_c$ is the rotating arm in each wetted surface unit (ds) of the appendages, $\mathbf{n}$ is normal vector. The pressure P_{wb} can be expanded as

$$P_{wb}(x, y) = \sum_{k_m}^{M} \sum_{k_n}^{N} D_{m,n} \varphi_{m,n}(t, z) \exp[i(k_m x + k_y y)] + C.C.\,, \tag{7}$$

where $\varphi_{m,n}(t, z)$ is the spectral coefficients of the velocity potential $\phi(x, y, z, t)$,

$$\varphi_{m,n}(t, z) = \frac{1}{MN} \sum_{m \le M, n \le N} \phi(x, y, z, t) \exp[-i(k_m x + k_n y)]. \tag{8}$$

In the DiSSEL model, they are updated at every simulation time step (Lin and Kuang, 2007a). In our approach, the damping coefficients $D_{m,n}$ are set as following

$$D_{m,n} = \begin{cases} 0, & \text{if} \quad k_m, k_n < k_{cut-off} \\ \varepsilon \sum_{i=1}^{I} d_i \left(k_m - k_{cut-off}\right)^{2i}, & \text{if} \quad k_n < k_{cut-off}, \quad k_m > k_{cut-off} \\ \varepsilon \sum_{i=1}^{I} d_i \left(k_n - k_{cut-off}\right)^{2i}, & \text{if} \quad k_m < k_{cut-off}, \quad k_n > k_{cut-off}, \\ \varepsilon \sum_{i=1}^{I} d_i \left[\left(k_m - k_{cut-off}\right)^{2i} + \left(k_n - k_{cut-off}\right)^{2i}\right] & \text{if} \quad k_m, k_n > k_{cut-off} \end{cases} \tag{9}$$

In our test, we set

$$d_i = \frac{1}{2^i} \quad \text{and} \quad k_{cut-off} = 0.00625 / L^2 * \sqrt{\frac{h}{g}}, \tag{10}$$

where L is the ship length, h is the water depth and g is the gravitational acceleration (g = 9.8m/s). We choose the cut-off wave number (10) because, as observed by Lin and Lin (2005), the wave breaking effect increases as water depth decreases.

Similarly, the damping effect to translational motion can be evaluated via

$$D_{trans}\mathbf{v} = \mathbf{v}\left[\frac{\mathbf{v}}{|\mathbf{v}|^2}\cdot\iint_s ds\mathbf{n}P_{wb}\right], \tag{11}$$

where **v** is the translational velocity vector**.**

3. NUMERICAL RESULTS

3.1 "BLOCKING THEORY" RESULTS

The X-craft catamaran (FSF-1, SEAFIGHTER) with T-Foil, shown in Figure 1, is used for testing our blocking damping method. The ship's maximum length and beam are 74.4 meters, and 21.06 meters, respectively. The displacement is approximately 1300 metric tons. The mean draft is 3.26 meters, and the mean freeboard is 4.4 meters. The roll, pitch and yaw gyradii are 7.9, 24.94 and 24.94 meters, respectively.

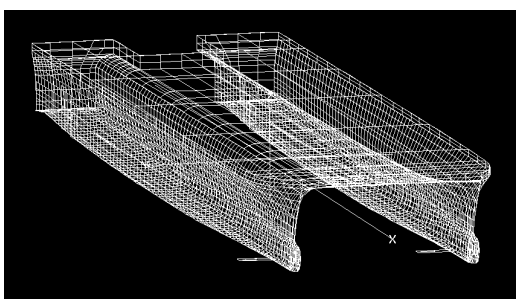

Figure 1: The grid of X-Craft with T-Foil (from David Taylor Model Basin, Allen Engle)

The incident waves in the irregular beam sea are used in our test. Their wave amplitudes with respect to the frequencies are shown in Figure 2. The numerical results without appendage damping effect (red curve) for the roll motion of the X-Craft at Froude Number = 0.38 are shown in Figure 3a, together with the experimental data (blue curve). The numerical results with the appendage damping are shown in Figure 3b. As one can observe from the figures, the numerical roll motions without the damping (Figure 3a) are nearly 25% greater than those measured. However, the numerical results with the damping agree well with the experimental data (see Figure 3b).

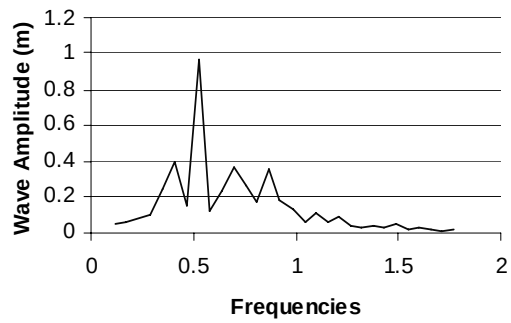

Figure 2: Distribution of the wave amplitudes with the frequencies of the incident waves used in the X-craft test.

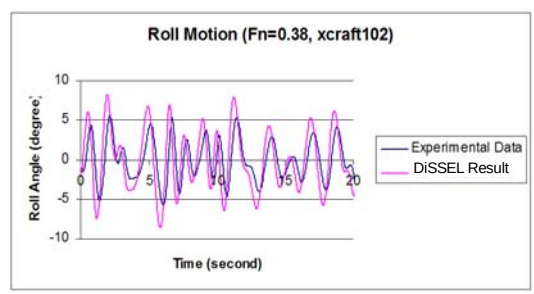

Figure 3a: The roll motion of X-craft with Froude number = 0.38. Red curve is the DiSSEL numerical simulation results without T-Foil; the blue curve is the experiment data.

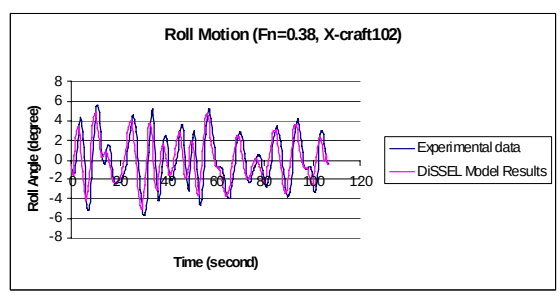

Figure 3b: Similar to Figure 3a, but the numerical results are with the T-Foil (with blocking effect).

3.2 “WAVE BREAKING THEORY” RESULTS

A different vessel, the landing craft, AAV7A1, is used for this part of study since there are available experimental results from David Taylor Model Basin. The ship hull is shown in Figure 4. Its length and beam are 8 meters and 3.27 meters, respectively. Its displacement is 23.9 metric tons with an LCG of 3.7 meters forward of the transom.

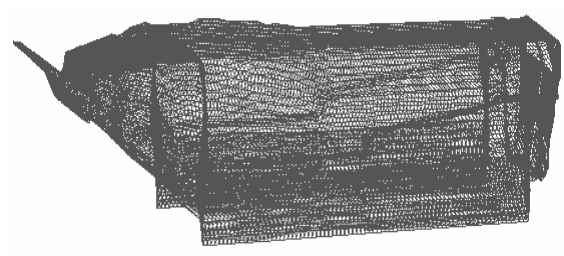

Figure 4: The landing craft with track and bow vane.

Hoyt and Lin (2007) showed in their work that bow vane is critical in avoiding the “plow-in” pitch instability. Their numerical results show that, if there is no bow vane, the instability occurs when the land craft forward speed is between 4.0 knots < Vs < 5.0 knots. However, such instability disappears if the bow vane is included in the DiSSEL model, regardless the presence of any incident waves. Their results agree well with the experimental data by Hayden, et al, (1988 and 1989) and Hoyt III, J.G, et al. (1994). In their results, the wave-breaking effect of the bow vane is included. However, they did not demonstrate in detail how important is the wave-breaking mechanism in the “plow-in” pitch instability.

From basic fluid dynamics, one can conjecture that first and foremost, the bow vane produces a strong lifting force when entering water, thus reducing the “plow in” angle. But, when it interacts with the surface waves, additional damping, i.e. the wave-breaking effect, also appears to modify ship motion. How strong is this effect? To understand this in more detail, we carried out two numerical experiments, one with the wave-breaking effect (8), (9) and (10); and the other without the effect. Two sets of simulation results of the pitch motion of AAV in irregular head sea are shown in Figure 5. In the two simulations, we set the significant wave height (SWH) =3ft for two forward speeds: (a) Vs=4.5knots (Fr=0.2624) (in the instability region) and (b) Vs=6.0knots (Fr=0.35) (outside the instability region). In both cases, the relative difference between the two sets of the solutions is, on average, less than 5%, though in isolated periods it can reach 20% or above. For example, at t = 53.25 seconds in the first case (Figure 5a), and t = 46 seconds in the second case (Figure 5b), the difference is significant. It should be point out that, as shown in Figure 5b, DiSSEL results with the wave breaking effect are closer to the experimental data.

4. CONCLUSIONS

In this study, we extend Lin and Kuang’s “Blocking Theory” to study the damping effects of small appendages on ship hull. The small appendage damping effects are very important in accurately simulating ship solid body rotation, as demonstrated by Lin and Kuang (2007b) in their studies of the bilge keels on ship roll motion. Our study on another small appendage, the T-Foil, shows further the importance of small appendages in modeling the vessel’s roll motion. Our results also demonstrate that the formulations of the blocking effect, Eq. (3) through (5), are generic, and can be used for arbitrary ship hulls and arbitrary appendages.

Interactions of appendages with surface waves generate small scale flow that can be difficult or impossible for numerical simulation. To effectively include these sub-scale phenomena in numerical model, we developed a method, defined in Eq. (7) through (10), similar to that in traditional wave-breaking analysis. Application of our “Wave Breaking Theory” to the bow vane and the pitch instability studies demonstrated that this sub-scale effect is noticeable. And it should be included for better agreement between numerical simulation results and experimental data.

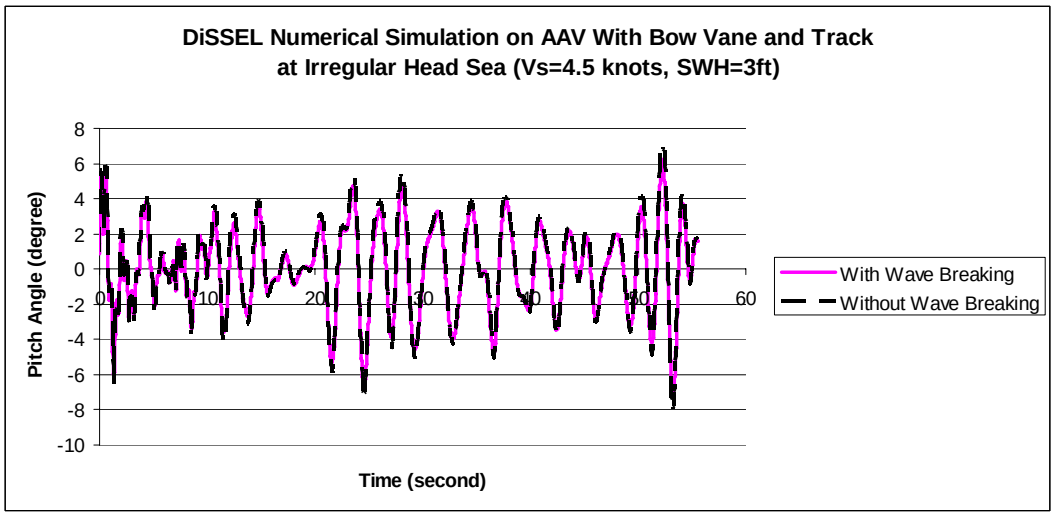

Figure 5a: The pitch motion of AAV with bow vane and track in irregular head seas, Vs=4.5knots, SWH=3ft. The solid red curve is the DiSSEL numerical results with the wave breaking effect; the blue dashed curve are the results without the wave breaking effect.

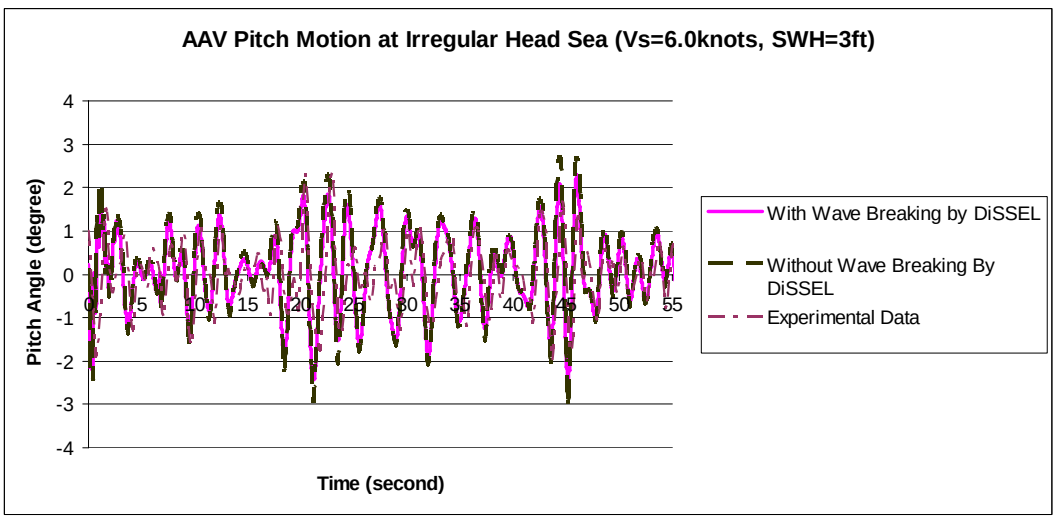

Figure 5b: The pitch motion of AAV with bow vane and track in irregular head seas for Vs=6.0knots, SWH=3ft. The red solid curve are the numerical results with the wave breaking effect, the blue dashed curve are the numerical results without the wave breaking; and the brown dashed-doc curve are the experimental data.

5. ACKNOWLEDGEMENT

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