11th International Conference on Fast Sea Transportation
FAST 2011, Honolulu, Hawaii, USA, September 2011

A Numerical Investigation into the Linear Seakeeping Ability of the T-Craft

John C. Bandas, Jeffrey M. Falzarano

Dept. of Civil Engineering, Ocean and Coastal Division, Texas A&M University, College Station, Texas, USA

ABSTRACT

A computational analysis of the seakeeping of a Surface Effect Ship (SES) known as the T-Craft is presented. The model is simulated using the commercial software WAMIT, using low-order, linear potential theory. The model of the T-Craft consists of catamaran hulls, rigid end skirts, and the interface between the air cushion and the water surface. Beyond the six rigid body modes of the T-Craft, additional modes are added for the motion of the interface panels. These additional modes are similar to standing wave or sloshing modes; i.e., sinusoidal modes are added until the wave motion in the cushion is modelled accurately.

To verify the method used, the model is benchmarked using computational data for a small-scale barge model and experimental data for a T-Craft model. A comparison is performed for the T-Craft with and without its cushion. From the numerical model, the motions are solved in both the frequency and time domain. The results are compared to the experimental data with respect to the response amplitude operators of the rigid body modes. The addition of a cushion improves the heave and pitch response of the craft while amplifying the sway, roll, and yaw.

KEY WORDS

Air Cushion Vehicles, Surface Effect Ships, "T-Craft", Seakeeping, Generalized Modes

1.0 INTRODUCTION

The T-Craft derives its name from its ability to transform from a SES into a hovercraft with side skirts in addition to the end skirts, allowing the vessel to travel onto land. The United States Navy is interested in the T-Craft as a method of offloading supplies and equipment to coastal areas without an accessible port. The T-Craft gets these supplies from a "seabase": an offshore platform or ship. The need for analysing the motions of the T-Craft comes from the possible difference between the motions of the vessel and the seabase. To predict the interactions between the two vessels, it is necessary to accurately model the T-Craft, including the aerodynamic effects of the air cushion.

The book by Yun and Bliault (2000) provides a brief history of the research studying the motions of the SES in waves. The initial motivation for modelling the SES was to increase its maneuverability and reach larger, stable speeds. Linear and nonlinear equations of motion solvable in the time domain were established, which include any combination of the features of air-cushion craft. Some of these features are the rigidity or deformation of the cushion skirts, the diffraction of the incident waves by the cushion pressure, the compressibility of the cushion air, the spatial distribution of the pressure due to nonzero speed, and the damping and added mass caused by the cushion.

One nonlinear analysis of the SES by Zhou et. al. (1980) derived two-dimensional equations of motion for both the transverse and longitudinal motions of a prototype SES. The effects of the air cushion cause the pitch and heave (roll and heave in the transverse case) of the SES to be coupled. The nonlinear effects include buoyancy changes along the rigid hull and the leakage of air from under the cushion seals, both changing due to waves. The cushion pressure was modelled using a linearized adiabatic gas law. The SES is moving with a forward velocity, so the cushion pressure could not be considered uniform: the center of buoyancy was shifted backwards.

The development of seakeeping models for air cushion vehicles first began around the 1970s, to study how cushions could support large offshore structures. These models utilize potential theory and are solved in the frequency domain for the pressures and responses of the structure.

Pinkster et al. (1998) described two numerical approaches which were verified in experiments using a simple rectangular barge supported by a cushion. The motions of the structure, the free-surface, and the cushion pressure were solved either together in one system of equations or separately in multi-body equations. Linear potential theory is used in a panel method with distributed point sources. The air-cushion-water interface is modeled by a series of panels. Each panel is massless but has added mass, damping, and stiffness, and adds one degree of freedom to the system for its vertical motion. Additionally, an aerostatic stiffness term was calculated using the adiabatic gas law. These hydrodynamic and aerostatic coefficients are coupled with adjacent cushion panels.

Lee and Newman (2000) analysed a barge supported by a closed air cushion by adapting their commercial code WAMIT. The equations of motion are created for the six rigid-body motions, with the addition of extra degrees of freedom (modes) for the cushion-water interface. Where Pinkster et al. gave each cushion panel one extra vertical degree of freedom, here all the interface panels are grouped together and given a normal velocity boundary condition that is a Fourier series approximation of the surface waves, with each extra Fourier mode adding one degree of freedom to the system. Additionally, the Helmholtz equation is used to treat the interface as a membrane, which leads to an aerodynamic added mass.

In this paper, the details of these modelling methods are adapted to the T-Craft. WAMIT is used to calculate the hydrodynamic coefficients, and the aerodynamic coefficients are added in and solved for the RAO's. A comparison is presented with available benchmarks (e.g. Pinkster's barge) and with experimental data available on the T-Craft. Comparisons are made of RAO's, hydrodynamic coefficients, and forces. The T-Craft's performance is compared both on and off the cushion. These results are discussed in the context of including the T-Craft model in a multi-body system with the seabase.

2.0 AIR CUSHION DYNAMICS

The most basic model of the SES is considering the vessel as a rigid body, including the skirts. In the case of seakeeping, the vessel is has zero forward speed. As long as the vessel is not moving, the cushion pressure is uniform. The water level in the cushion is modelled as an adiabatic piston. There will be added mass and stiffness for the cushion air.

The analysis of an air cushion vehicle using linearized potential theory is done similarly to the analysis of a single-hulled ship. The equations of motion are solved algebraically for the response by calculating the hydrodynamic coefficients and exciting forces.

A major difference between the simulations of a single body and the T-Craft is the addition of a kinematic boundary condition for generalized modes along the water-air cushion interface. . This method is explained in more detail in Lee and Newman (2000). In addition to the six rigid body modes, generalized Fourier modes are added to approximate the elevation of the interface. These Fourier modes are added to the heave elevation and vertical components of the pitch and roll. The Fourier modes are expressed as

$$\varsigma_{mn}(x,y) = \begin{pmatrix} \cos u_m x \\ \sin u_m x \end{pmatrix} \begin{pmatrix} \cos v_n y \\ \sin v_n y \end{pmatrix} \qquad (1)$$

where

$$u_m = \frac{m\pi}{2a}$$

$$v_n = \frac{n\pi}{2b}$$

and ζ is the interface elevation, a is the cushion length, and b is the cushion width. The integers m and n are even or odd, depending on if the mode is proportional to cosine or sine, respectively. Each combination of m and n creates a new generalized mode; the exact combination does not matter as long as the Fourier expansion is complete (converges accurately). This elevation is also the kinematic boundary condition along the interface. By using this boundary condition, the added stiffness of the cushion is embedded in the calculation of the hydrodynamic stiffness.

While the water in the environment is assumed to be incompressible, the air in the cushion is compressible. To account of this, the velocity potential of the air cushion is governed by the Helmholtz equation:

$$\nabla^2\Phi + K^2\Phi = 0 \qquad (2)$$

where Φ is the velocity potential and K is the acoustic wave number. From the solution of this aerodynamic potential, the added-mass of the air cushion is calculated as

$$A_{ij} = \rho_a \int_{S_a} N_i \Phi_j dS \qquad (3)$$

where i and j are mode indices, S_a is the interior air cushion surface (including interface), N is the unit normal vector pointing out of the cushion, and ρ_a is the density of air. It should be noted that in (3), the added-mass will be dependent on the frequency of the incoming waves, and the volume of the air cushion.

From the preceding equations, the equations of motion can be constructed:

$$\sum_{j=1}^{\infty} \xi_j \left[-\omega^2 \left(a_{ij} + A_{ij} + M_{ij} \right) + i \left(\omega b_{ij} + c_{ij} \right) \right] = X_i \qquad (4)$$

The summation represents the inclusion of the Fourier modes created by (1) in addition to the standard six degrees of freedom. The coefficients a_{ij}, b_{ij}, and c_{ij} are the hydrodynamic added mass, damping, and stiffness matrices. X_i is the restoring force, and M_{ij} is the inertia matrix of the ship (note that for j > 6, the inertia matrix is zero). There is no damping caused by the air cushion. Damping would be included by expanding the dissipative effects of the cushion skirts, i.e., allowing air to leak from the cushion and deform the skirts. This would require some finite element modelling of the skirts structurally, which will not be done in this paper.

From (4), one can see that increasing the added mass should decrease the response if the exciting force remains the same. The relationship between the dynamic coefficients and the force will be important in a later section.

3.0 MODELLING CONSIDERATIONS

Three models are used in this paper: a rectangular model barge, the generic T-Craft without the cushion (a catamaran configuration), and the T-Craft on the cushion. The low-order version of WAMIT is used to simulate the models, so the accuracy of the model depends heavily on the number of panels used.

The barge, shown in Fig. 1, has 1127 panels. The dimensions of the barge are 2.5m in length, 0.78m beam, and 0.15m draft. The hull has a thickness of 0.02m on the ends and 0.06m on the sides. The static pressure in the barge is increased above the ambient air such that the water level inside the cushion is 0.05m below the still water line.

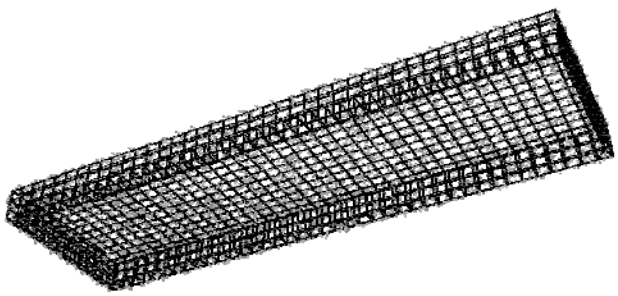

Fig. 1. Model Barge used in Pinkster et al. (1998)

For the barge, waterline panels were created to allow WAMIT to remove irregular frequencies from the simulation results. These panels were not necessary for the T-Craft.

The model for the generic T-Craft was provided by the Naval Surface Warfare Center Caderock Division (NSWCCD). This model has been used both in numerical simulations and experiments. The T-Craft has two variants: one has a single cushion, and one has two cushions created by splitting the cushion volume with a flexible plenum skirt. Additionally, the T-Craft can be "off" the cushion, essentially becoming a catamaran. This paper models and compares the T-Craft on a single cushion and off the cushion. The wetted hull of the T-Craft off its cushion is shown in Fig. 2

Table 1. Dimensions of the T-Craft.

Dimension	On Cushion	Off Cushion
Waterline Length (m)	67.52	75.2
Draft (m)	1.33	4.02
Cushion Width (m)	16.5	-
Cushion Length (m)	67.14	-
Cushion Height	1.3	-

The catamaran model consists of 1661 panels. The deck of the T-Craft is very close to the waterline when it is off the cushion. However, it would be very difficult to simulate the model with many panels on the waterline. Therefore, the deck is not included as part of the wetted area of the T-Craft.

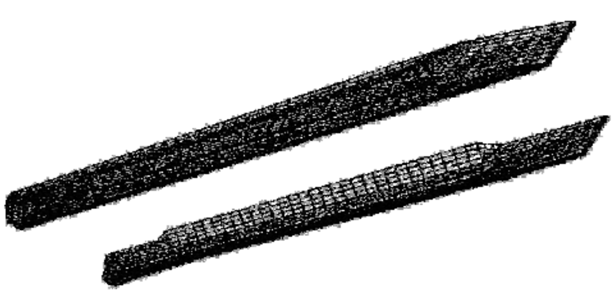

Fig. 2. The generic T-craft in a catamaran configuration.

To model the T-craft on its full-strength cushion, the catamaran is changed as shown in Fig. 3. The draft is decreased, and flat panels are added on the level of the inner air cushion. The skirts are modelled simply as rigid panels at the ends. The intersections between the inner hull and the air cushion create very small, sharp corners, which could cause the simulation to generate errors. Therefore, the mesh density was increased to 2232 panels.

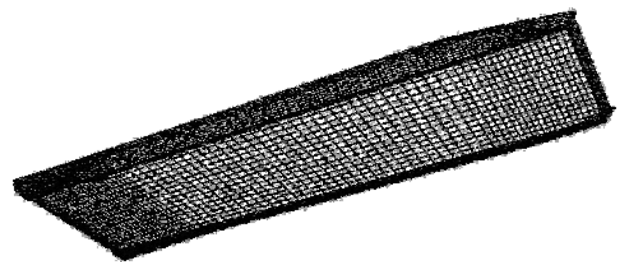

Fig. 3. The generic T-craft on a full cushion.

4.0 SIMULATION BENCHMARKS

To test the results of the simulations, two benchmarks are performed with published data. Pinkster has published forces, hydrodynamic coefficients, and response amplitude operators (RAO) for the barge. This data is generated using Pinkster's panel method, but has also been verified by Lee and Newman with the generalized modes.

Experimental data has been collected for the T-Craft by Bishop et al. (2011). Additionally, Hughes and Silver (2010) of the NSWCCD have benchmarked various seakeeping codes, including WAMIT, against the heave RAO of the experimental data. This benchmark is compared with the simulation of this paper.

4.1 Barge Computational Results

The comparisons with Pinkster and Lee and Newman's results are shown below. In this comparison, the ship is encountering head waves. In Fig. 4, the simulated heave RAO is the solid line, and the published data is the dotted line. The data is done over a relatively large range of frequencies, but the simulation captures most of the interesting phenomena of the system. The most notable features are the humps at 6,8, and 9 rad/s. These features are likely due to resonance with the Fourier modes of the air cushion. The simulation does not capture the feature at 9 rad/s. This could be due to a discrepancy of the number and frequency of the Fourier modes between the model and the benchmark data.

Also of note in the simulation line is a discontinuity at 3 rad/s. This was the location of an irregular frequency. An increase in the number of panels in the barge and the creation of waterline panels resulted in the data shown. There may be some small discrepancies in the irregular frequency removal because of the presence of an inner and outer waterline.

The pitch results are shown in Fig. 5. The amplitude is the same order, but the benchmark data has its peak frequency shifted to a higher frequency. It should be noted that the benchmark was generated using a higher order panel method.

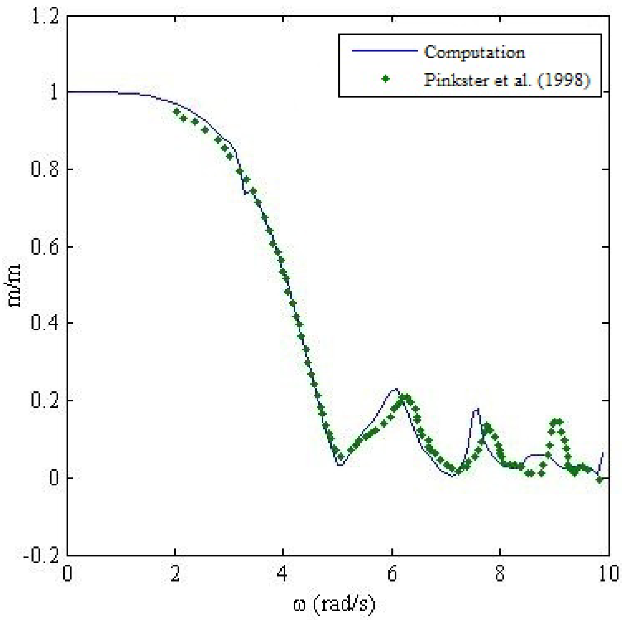

Fig. 4. Heave RAO of the Pinkster barge in head waves.

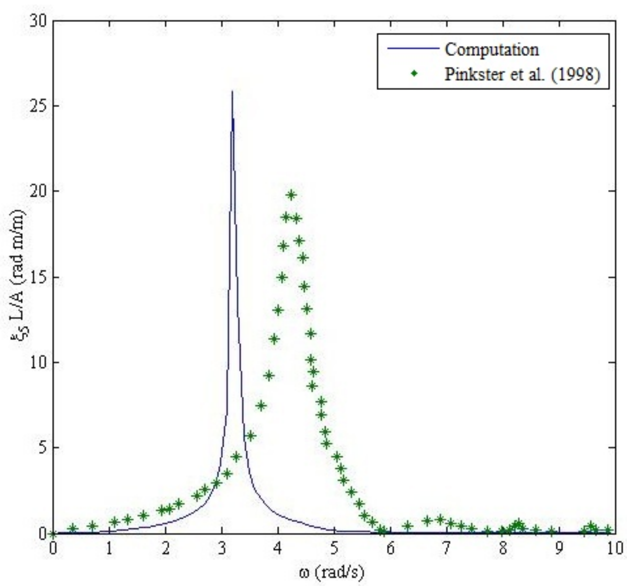

Fig. 5. Pitch RAO of Pinkster barge in head waves.

4.2 NSWCCD Experimental Data

For the T-Craft, Hughes and Silver (2010) used the linear, first order version of WAMIT to generate the RAO of the T-Craft. The simulation and experiment were conducted at relatively low frequencies compared to Pinkster. The wave frequencies are likely more realistic for what the T-Craft will encounter in seakeeping. Higher frequencies do become important in maneuvering.

The comparison of the simulated T-Craft, Hughes simulation, and experimental data in head waves is shown in Fig. 6. This paper's results are the solid line, Hughes data is the dashed line, and the experimental data is the dotted line. It is apparent that linear simulation is missing some behaviour found in the real T-Craft, although the agreement between the two linear computation results is good. A very significant nonlinearity would arise from the dissipative effects of the deforming air cushion skirts. Wave impacts on the skirt, interactions of the cushion air with the fan supply system, and air leakage under the seal are some major features of the T-Craft that are not included in a linear potential source code (see Yun and Bliault, 2000).

Another explanation could be effects from the division of the cushion into two sections by the plenum skirt in the middle. In this case, the generalized modes would be implemented differently for each sub-cushion, leading to an increase in total modes.

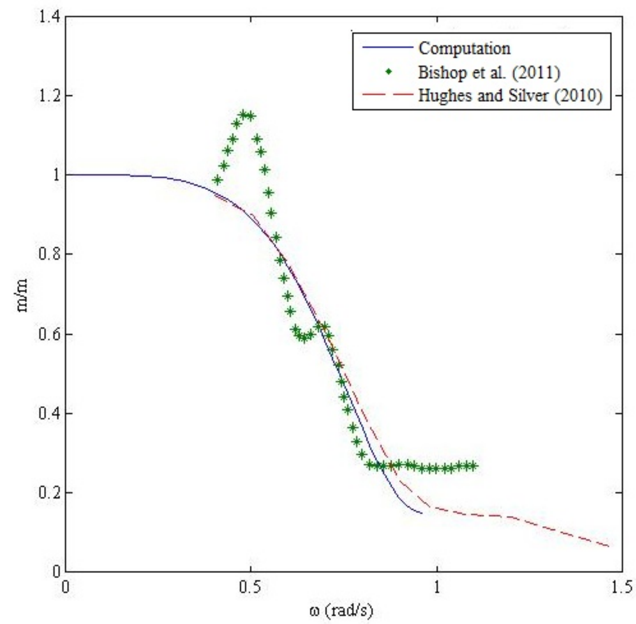

Fig. 6. Heave RAO of T-Craft in head waves.

5.0 COMPARISON OF CUSHION CONDITIONS

To see what advantages the T-Craft has in seakeeping with its extra modes, it is desirable to know the difference gained between the T-craft on and off the cushion. The following section compares the RAO for the two cushion states. The frequency range was restricted from 0 to 2 rad/s because any higher frequency features were insignificantly small.

For all of the following figures, the solid line represents the T-Craft on its cushion and the dotted line is the T-Craft without its cushion. The ships are encountering waves at a 20° heading.

Some interesting behaviour appears when comparing the response of the T-Craft on and off the cushion. The surge RAO is shown in Fig. 7. Surprisingly, the surge for the T-Craft is similar both on and off the cushion. This can be explained by looking at the values of the added masses, stiffnesses, and forces. While the draft has decreased, the frontal area is roughly the same due to the inclusion of the skirts. This area is redistributed to be closer to the free surface, where there are higher pressures acting on the ship. Therefore the surge exciting force on the T-Craft is greater on the cushion compared to off the cushion for high frequencies, and smaller at low frequencies. The surge exciting force is shown in Fig. 8. The fluctuations seen in the curves are similar in phase. Although the force is higher, the some of the hydrodynamic added masses are smaller. The heave-surge added mass is shown in Fig. 9; the decrease in draft has decreased the added masses as well as the forces. It is interesting to note that the added masses are negative across the range of frequencies. The variations in force mitigate the variations of the hydrodynamic coefficients and result in similar responses.

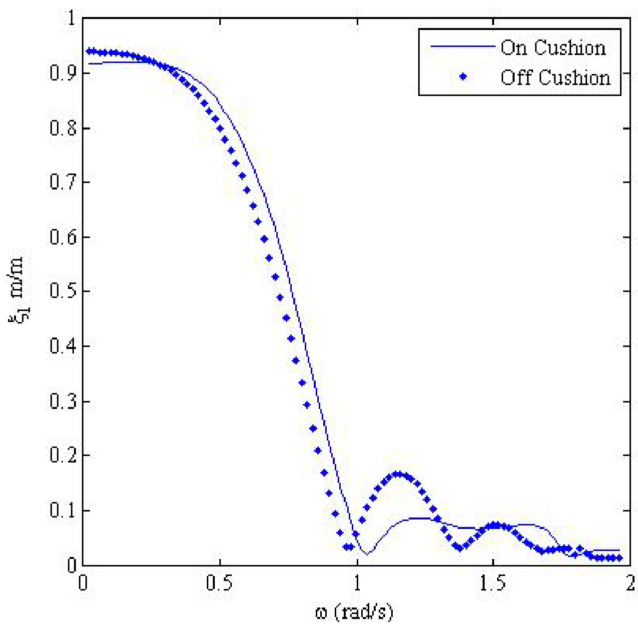

Fig. 7 Surge RAO comparison in 20° bow waves.

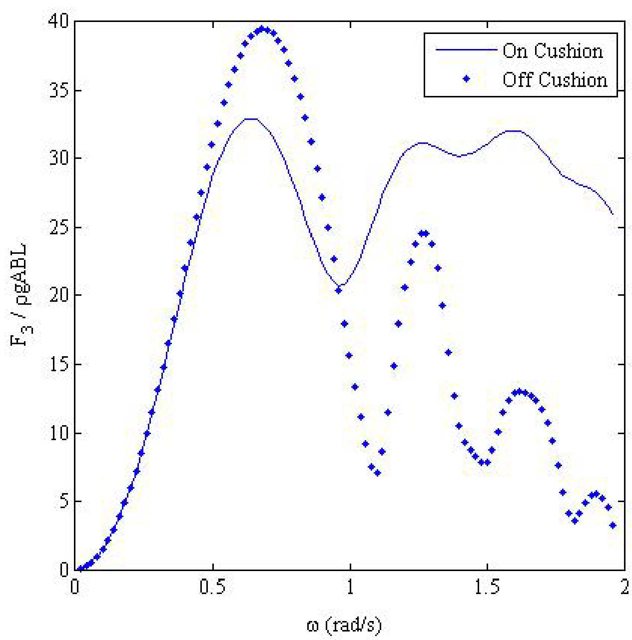

Fig. 8 Surge force comparison in 20° bow waves.

The comparison of sway RAO's, shown in Fig. 10 supports this idea of the sensitivity of the response to change in projected area. The sway of the T-Craft is larger on the cushion. However, as shown in Fig. 11, the exciting force is actually greater when the T-Craft is off of the cushion (note there is some asymptotic behavior at higher frequencies that should be is a discrepancy in the computation). This greater force comes from the increased draft and projected area. The larger force for the catamaran seems to contradict its lesser sway response.

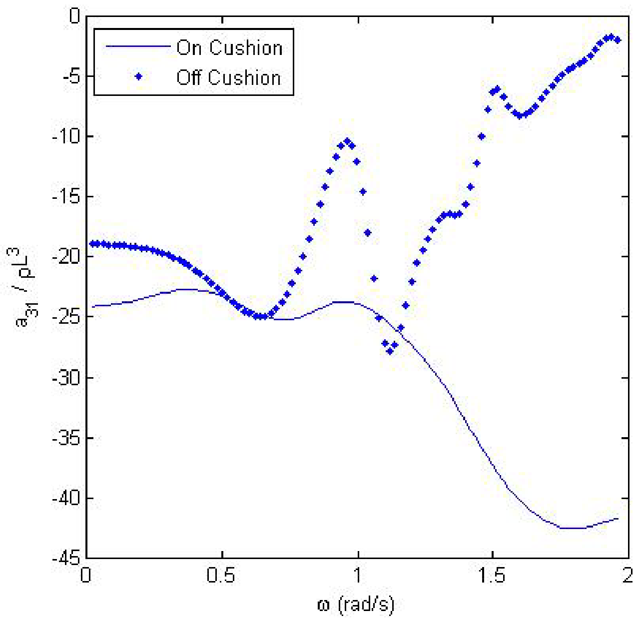

Fig. 9 Heave-surge added mass comparison in 20° bow waves.

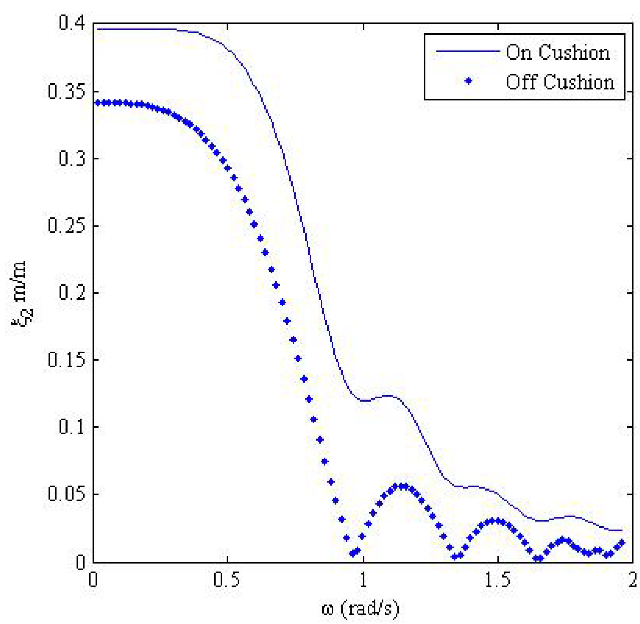

Fig. 10. Sway RAO comparison in 20° bow waves.

However, a change in the added mass matrices again affects the response of the ship. Smaller added masses will increase the RAO calculated as well as larger forces. Some of the sway added masses are very small or even negative for the T-Craft on its cushion. One of these coefficients, the sway-roll added mass, is compared in Fig. 12. When on the cushion, an added mass is generated that is smaller over the entire range of frequencies. The difference comes from the integration of (3), since the T-Craft has the added boundary conditions of (1). When rolling, instead of just accelerating the water displaced around a rigid body, the water volume is acting with the cushion. As a result, the contributions of the smaller added masses with the higher force creates a RAO that is higher for the T-Craft on its cushion.

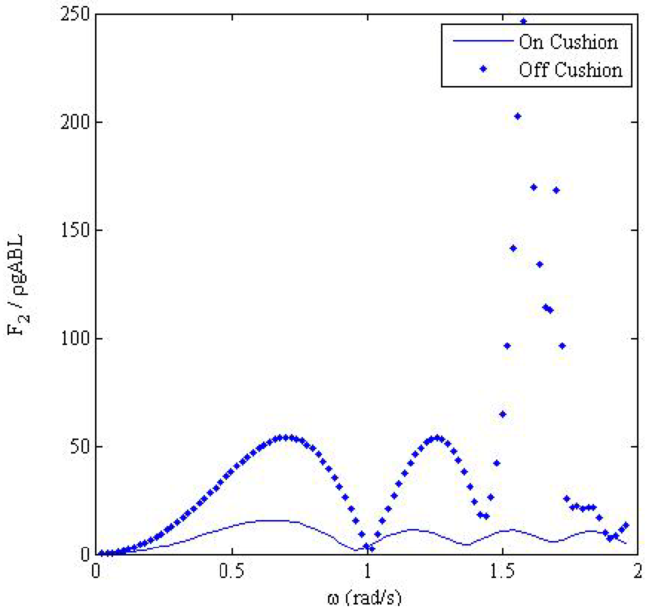

Fig. 11 Sway force comparison in 20° bow waves.

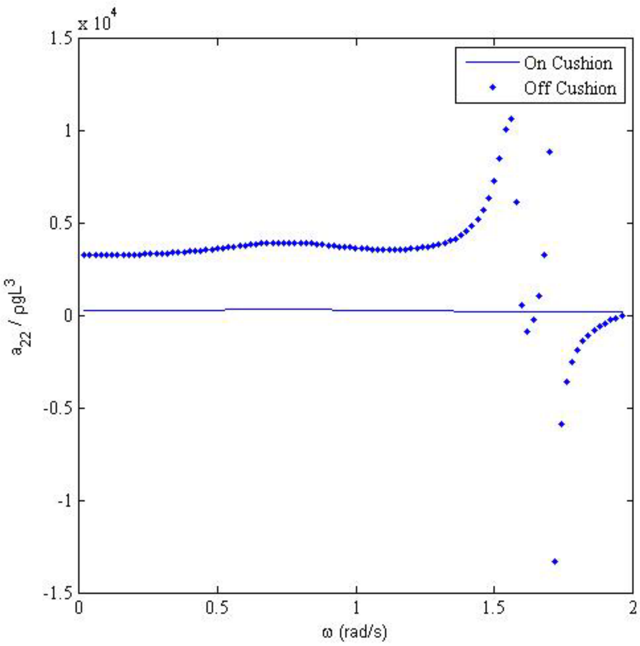

Fig. 12. Sway-sway added mass comparison in 20° bow waves.

The heave RAO shown in Fig. 13 is very similar both on and off the cushion. The very low frequencies dominated by hydrostatic effects are virtually unchanged, and at the higher frequencies the T-Craft cushion has a response that fluctuates less. This is accounted for by the pressure of the air cushion replacing lost buoyancy due to decreased draft, and the very small change in the projected area of the T-Craft from the bottom.

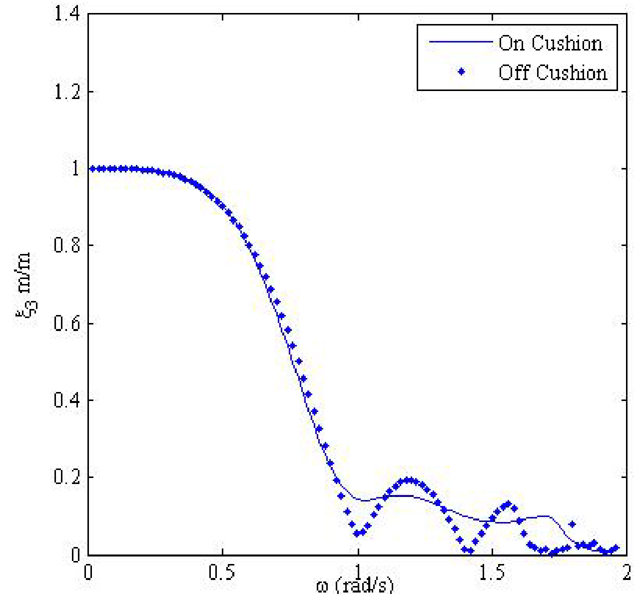

Fig. 13. Heave RAO comparison in 20° bow waves.

The roll RAO, shown in Fig. 14, is relatively small for both cases. However, there is an increase in the roll behaviour when the T-Craft is on the cushion. The decreased draft and the addition of air pressure decreases the righting of the T-Craft.

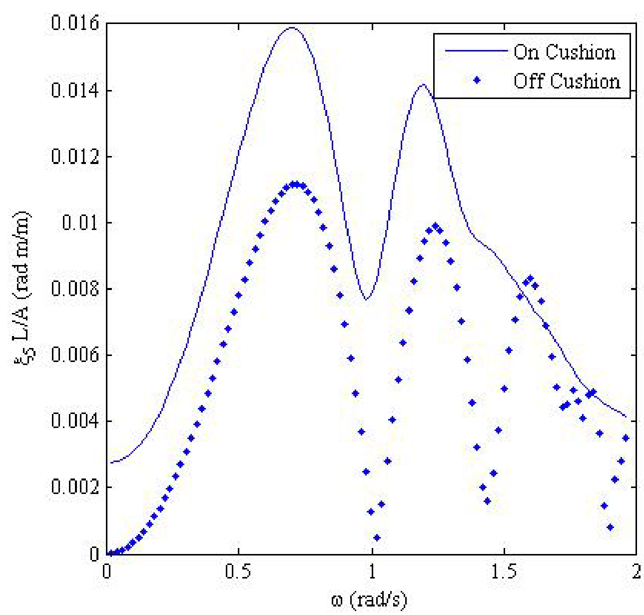

Fig. 14. Roll RAO comparison in 20° bow waves.

The pitch RAO is also relatively small. In Fig. 15, the response on the cushion is lower. As with the surge, this similarity could be due to the presence of the rigid skirts, whose faces are at an angle from the vertical plane. The forces on these skirt panels cause some additional moment in the longitudinal plane. The T-Craft is very long compared to its width, which is why the difference in pitch is much smaller than the difference in roll.

The yaw RAO is shown in Fig. 16. The responses are once again small, although the yaw of the T-Craft on its cushion is significantly larger. An extra moment is exerted by the skirts.

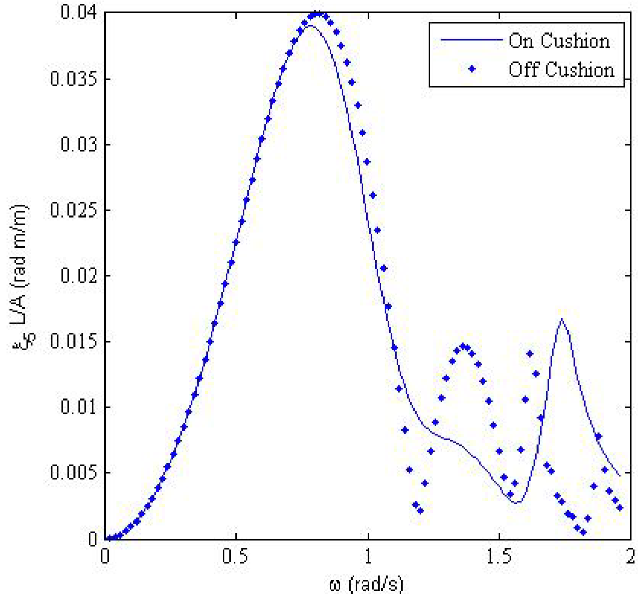

Fig. 15. Pitch RAO comparison in 20° bow waves.

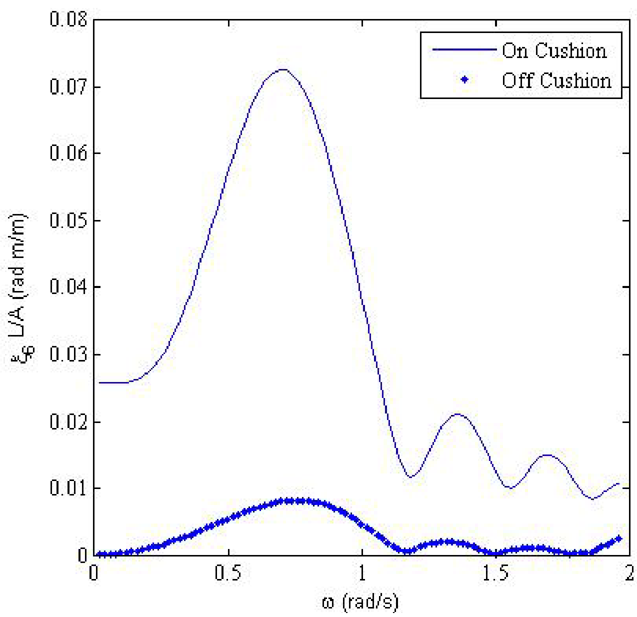

Fig. 16. Yaw RAO comparison in 20° bow waves.

It is apparent that the skirts and cushion have an important effect on the RAOs of the T-Craft. The assumption of rigid skirts is necessary for the linear panel method, but creates large flat areas for the exciting force to act on. Allowing the cushion skirts to deform would add damping to the system. It could be possible to add some empirical damping from the skirts, but there is no data readily available concerning this. In addition to the deformation of the skirts, the air cushion supply has some energy loss. The equations of the fan performance could be coupled to the equations of motion to account for this.

6.0 CONCLUSION

The method of Lee and Newman (2000) has been applied to the case of the T-Craft using the low-order panel module of WAMIT. The method involves the addition of air cushion boundary conditions based on Fourier modes, designed to approximate the elevation of the water level in the air cushion.

The low-order results are benchmarked against the compuational barge results of Pinkster et al. (1998), and the experimental results of Bishop et al. (2010).

Reviewing the comparisons between the data of the T-Craft on its cushion and off its cushion, it is unsurprising that there are significant differences in the seakeeping performance. Some degrees of freedom are amplified while on the cushion, such as the roll, sway, and yaw. These motions will affect the interaction between the ship and the seabase ramp structures.

REFERENCES

Bishop, R. C., Silver, A. L., Tahmasian, D., Lee, S. S., .Park, J. T, Snyder, L. A., and Kim, J. (2011). "TCraft Seakeeping Model Test Data Report-Update." NSWCCD-50-TR–2010-062, Hydro-mechanics Department Report.

Hughes, M. J. and Silver, A.L. (2010). "Method of Evaluation of Multi-Vessel Surface Effect Ship Motion Prediction Codes." Hydro-mechanics Department Report.

Lee, C.H., and Newman, J.N. (2000). "Wave Effects on Large Floating Structures with Air Cushions." Marine Structures **12**, pp. 315-30.

Pinkster, J.A., Fauzi, A., Inoue, Y., Tabeta, S. (1998). "The Behaviour of Large Air Cushion Supported Structures in Waves." Proceedings of the Second International Conference on Hydroelasticity in Marine Technology, Fukuoka, Japan.

Yun, L. and Bliault, A. (2000). Theory and Design of Air Cushion Craft. Arnold, London.

Zhou, W.L., Hua, Y., Yun, L. (1980). "Nonlinear Equations for Coupled Heave and Pitch Motions of Surface Effect Ships in Regular Waves." CACTS International Conference, Ottawa, Canada.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the support of the Office of Naval Research T-Craft Tools development program ONR Grant N00014-07-1-1067 with program manager Kelly Cooper. The authors would like to thank Mike Hughes and Andy Silver of NSWCCD for providing mesh models of the T-Craft and releasing experimental data for comparison. Additionally, thanks to C.H. Lee for his technical advice on the generlaized modes code in WAMIT.

