ROLL MOTION STABILIZATION FOR SMALL FISHING VESSELS

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BY

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Abstract

A review of various roll stabilizing systems—bilge keels, active fins, passive bilge fins, rudder stabilization systems, passive anti-roll tanks, active anti-roll tanks, and paravanes—is given, along with a more detailed discussion on their suitability for small fishing vessels. Particular attention was paid to the passive flume tanks and paravanes, and an experimental study was carried out on these two kinds of stabilizers.

A flume tank model and a pair of paravanes, as well as an oscillating bench simulating sinusoidal rolling, were designed and then constructed. The tank was tested on the bench for configuring the internal dimensions which give the desired natural frequencies, tuning, and damping. The analysis was carried out by inspecting the phase lags ε_t of the motion of tank water to the roll motion.

Free roll decay tests for a fishing vessel model with the tank and paravanes were carried out for identification of the damping generated by the tank and paravanes. Non-dimensional equivalent linear damping ratios ζ_E were calculated and discussed.

The stabilizers were tested in the wave tank at MUN, both in regular and irregular beam waves, to determine their effectiveness. The study of roll responses, roll energy spectra, response amplitude operators (RAO), and resonant and significant roll amplitude reductions at various wave and stability conditions shows that the tank can provide 50% to 70% roll reductions, which are approximately 30% more than the paravanes can do in the same circumstances. The effects of tuning, liquid level, and internal damping on the efficiency of the tank are discussed. Comparisons in the behaviors of the stabilizers for different wave conditions are also made.

Some design considerations are presented, and further work is recommended to develop a systematic method to design passive stabilizing tanks for small fishing vessels.

Acknowledgements

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List of Symbols and Abbreviations

ϕ	: roll angle
ϕ_A	: roll amplitude
ϕ_m	: mean amplitude of roll angle
$\dot{\phi}$	: roll velocity
$\ddot{\phi}$	: roll acceleration
h_w	: wave height
ω	: frequency of wave exciting moment
ω_0	: peak frequency of irregular waves
ω_ϕ	: roll natural frequency of the ship
ω_t	: natural frequency of the tank moment
ε_t	: phase angle between the tank moment and the roll motion
B, l_t, l_n, b_n	: dimensions of the tank
h	: water level in the tank
w	: weight of tank fluid
M_t	: amplitude of the moment generated by the tank
M_w	: amplitude of the wave exciting moment
B_E	: equivalent linear damping coefficient
ζ_E	: non-dimensional equivalent linear damping ratio
Δ	: mass of the ship model
LWL	: length on waterline of the ship model
GM	: metacentric height of the ship
δGM	: GM loss due to free-surface effect of stabilizer tank

KB	: height of center of buoyancy above keel
BM	: transverse metacentric radius
$S_\zeta(\omega)$	: density function of wave spectrum
$S_\phi(\omega)$	: spectral density function of ship response for rolling
$(h_w)_{1/3}$	: average height of one-third highest waves
$(\phi)_{1/3}$	: average value of one-third highest roll amplitudes
$\lvert H(\omega)\rvert^2$	: response amplitude operator (RAO)

Chapter 1

Introduction

Ship roll can cause various ill effects, such as cargo damage, reduction in crew efficiency, increasing resistance, or even a capsize. The problem of reduction of ship roll has been studied intensively for almost 100 years. A wide variety of stabilization systems have been tried for different types of ships. However, the investigation of stabilizers for small fishing vessels is relatively inadequate. The purpose of this study is to find an appropriate stabilization system for small fishing vessels by experimental methods.

1.1 The Stabilizer Requirements and Selection Criteria for Small Fishing Vessels

Due to their size and potentially rigorous operating environment, small fishing vessels are apt to roll severely. However, there was virtually no significant work on fishing vessel roll reduction until 1965, when Mr. J. J. Van Den Bosch presented a paper entitled "A Free-Surface Tank as an Anti-Rolling Device for Fishing Vessels" [1]. In this paper, he summarized four requirements for a fishing vessel's stabilizer:

- Effective even at low or zero speeds.
- Efficient for many different stability conditions.
- Inexpensive to install and maintain.
- No attention needed in normal use.

For a small fishing vessel, costs should be the first factor to be considered. From the economical point of view, costs of a stabilizer may include [2]:

- total initial costs.
- reduction of cargo carrying capability.
- influence on speed, power and resistance.
- regular maintenance and operation costs.

while benefits may include:

- increased operating efficiency.
- increased crew's comfort.
- savings in fuel costs due to the stabilizer [1].

The benefits should be more than the costs for a stabilizer to be used, although some of the benefits might be difficult to be estimated numerically, and depend on the standpoint of the ship owner.

The selection of the best type(s) of roll stabilization system(s) involves many factors which are unique to the given ship(s), and there is no simplistic method for all questions. Because most small fishing vessels are designed and built without stabilizers, one should select those which are adaptable to the space and equipment the vessels already have.

1.2 Survey and Evaluation of Available Roll Stabilization Systems

Generally, stabilization systems can be classified as three types: fixed, passive and active. Fixed stabilizers are those appurtenances such as bilge keels, bilge fins, paravanes, and gyroscopic systems. The first three simply generate a damping moment due to the roll motion of the ship, while a gyroscopic system counteracts the roll motion by the gyroscopic inertia. Passive stabilizers are those dynamic systems that have their own frequencies nearly equal to that of the ship, and dissipate rolling energy by dynamic coupling. Passive tanks (free surface and U-tube) and moving solid weight systems fall into this type. Active stabilizers include active fins, rudder stabilization, and active anti-roll tanks. They are equipped with feed-back control devices and usually work better than the passive ones but cost more. A brief discussion will be given for each of them except moving solid weight systems and gyroscopic systems, which have been proven unsuccessful due to excessive weight, poor controls, and high costs [3, 4].

1.2.1 Bilge Keels

Bilge keels are probably the earliest form of roll stabilizers. They are flat plates attached almost perpendicularly to the hull along the streamline at the turn of the bilge. They increase the damping moment by increasing eddy making when the ship is rolling. Bilge keels typically influence hull damping at zero speed by factors of 1.5 to 2.0, depending on the area of the bilge keel and distance from the center of roll [2]. They are also found to be effective in heavy sea conditions [3]. The only drawback of bilge keels is that they also increase hull resistance at forward speed. Experiments have shown that during resonant rolling,the increased resistance can be up to 40 percent of that of the hull without bilge keels [2].

Bilge keels are low in cost and weight, effective in all sea states and any speed range, and generally as maintenance-free as the hull. They are probably the most widely used stabilizers. However, there are some shortcomings for bilge keels installed on small fishing vessels. First, fishing involves many over-side operations which may be interfered with by the bilge keels. Second, in cold sea regions such as Newfoundland, protrusive bilge keels are easily damaged by floating ice. Third, many small fishing vessels are built of wood, therefore, reliable installation of bilge keels is relatively difficult and some reinforcement is required, thus resulting in higher costs and larger resistance. For these reasons, many small fishing vessels are built without bilge keels, instead, with sharp chines, bossings and large skegs to increase their hull damping.

Bilge keels give relatively small damping moments, especially for small roll angles. This is because the damping moment generated by bilge keels is approximately dependent on the square of the velocity, and when roll amplitude and frequency decrease, the moment decreases sharply [5]. In many cases, bilge keels do not satisfy requirements, and some other kind of stabilizers have to be added.

1.2.2 Active Fins

The idea of using active fins for roll stabilization appeared long before World War II. But only after World War II, were they fully investigated and adopted for many passenger liners and warships.

The active fin stabilization system consists of at least one pair of airfoil-shaped, wing-like fins—one on the port and the other on the starboard side. By adjusting the attack angles while the fins are moving at a speed, a stabilizing moment can be created by the lift forces produced by the fins. The lift force is proportional to the square of the speed of the ship, to the angle of attack of the fin relative to the flow, and to the lift coefficient of the foil section. A feed-back control system is used to obtain the maximum stabilizing moment. To rotate the fins, a complicated mechanical system is also required.

Active fins are by far the most effective roll stabilizers, as well as the most expensive ones. Even though the effectiveness of active fins can be up to 90 percent reduction of significant roll angles [2], they are obviously far too expensive for small fishing vessels. Other impacts of active fins are: considerable space and weight are required, they are not effective at low or zero speeds, and they are vulnerable. All of these reasons exclude active fins from consideration for small fishing vessels.

1.2.3 Bilge Fins

Bilge fins work on the same principle as active fins. The only thing different from the active ones is that they are fixed instead of moving. This certainly saves money but loses much efficiency. Usually, active fins are much more preferable, and bilge fins are applied only when costs are very critical. Because this is true for small fishing vessels, several pairs of bilge fins have been tested on fishing vessels [6]. Some of them provided useful increases in model roll damping with only minor resistance penalties, some didn't. The tests indicated that passive bilge fins are somewhat attractive for fishing vessels at steaming speeds.

On the other hand, as with bilge keels, bilge fins are not suitable for fishing vessels. Furthermore, bilge fins mainly work by lift force, which means that their efficiency will decrease considerably at low or zero speeds which are the operating speeds of fishing vessels.

In summary, in my view, the use of bilge fins on a small fishing vessel is a structural problem. If the bilge fins can be made in a retractable form and installed reliably with minor cost and resistance penalties, they can be used as part of a combined stabilization system.

1.2.4 Rudder Stabilization

As the position of a ship's rudder is usually below the roll center of the ship, the lift force generated by the rudder causes both roll and yaw moments. If roll moments affect the ship much sooner than yaw moments, rudder-induced roll moments can be used as stabilizing moments without affecting course-keeping. In order to properly phase the rudder-induced roll moment to the wave-excited roll moment, specially designed control systems and steering gear may be required.

As active fins, rudder stabilization systems work better at higher speeds due to higher lift forces. Additionally, they require that the ship responds more quickly to roll moments than to yaw moments, which is not valid for some kind of small boats designed to be easily-turned. At present, only a few naval vessels and high-speed boats have installed rudder roll stabilization systems. This is because the development of this system is immature and its application is rather limited. No small fishing vessel has been found in the literature with a rudder stabilizer.

1.2.5 Passive Anti-roll Tanks

In 1874, W. Froude first considered using a tank partially filled with fluid to reduce ship roll motion. But only after 1910, when Frahm developed his famous U-tube tank, tank stabilizers were installed for actual service. Since World War II, passive anti-roll tanks have become commonplace on all types of vessels.

Common configurations of passive tanks include (Figure 1.1):

- U-shaped tanks with an air connection between the tops of the vertical legs(Frahm tank).
- Side tanks open to the sea at their bottoms,and each vent end at the top to the atmosphere or joined by an air connection.
- Free-surface side tanks connected by a flume with damping introduced by nozzles, plates, or similar-type obstructions (Free-flooding tank).
- Free-surface rectangular tanks with damping introduced by nozzles, plates, or similar-type obstructions.

The basic theory behind passive tank stabilizers is that if the natural period of fluid flow in the tank equals the natural roll period of the ship, then the ship-tank system will be a double resonant system. This means that the moment due to fluid motion in the tank will be 90 degrees out of phase with the roll of the ship at resonance, while the roll of the ship lags wave exciting moment 90 degrees. Thus, the tank moment will lag by 180 degrees the wave exciting moment and counteract roll excitation.

The major advantage of passive tanks is that their operation is independent of ship speed, providing efficient roll reduction for a given range of encounter frequen-

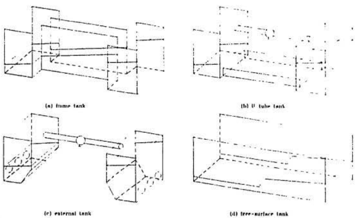


Figure 1.1: Common configurations of passive tanks

cies throughout the entire speed range. Generally, if the tank is designed properly, reduction of significant roll angle near the resonant period can be approximately 50 percent [2]. Since the structure of passive tanks are simple and they don't need any control system, the cost of design and installation are relatively low. They are also maintenance-free when operating although they are required to be properly tuned for different operating conditions.

Although passive tanks require some space and weight(usually around 2 percent of the ship displacement), their locations are versatile, and suitable locations that do not occupy usable cargo volume can generally be found.

The major problem of all anti-roll tanks is saturation, which means the fluid slams against the tank top when very large roll motions are achieved. The tank stabilizing effectiveness will decrease in this case, therefore, a reasonable tank height should be designed according to the design sea states. Other disadvantages of passive tanks are that they decrease initial stability, and they may be noisy in some cases [4].

For the same location, weight, and tank volume, all types of passive tanks will provide approximately the same stabilization moment. The major difference between a U-tube tank and a free-surface tank is that the natural frequency of a U-tube tank is mainly determined by the cross-sectional areas of the wing tanks and water flow crossover duct, while, for the free surface tank, the liquid height in the tank has a large influence on its natural frequency. Changing natural frequency is relatively difficult for a U-tube tank once its configurations are determined, but it can be easily done for a flume tank by adjusting the liquid level in it. This makes it preferable for small fishing vessels whose GM and operating sea state often vary considerably. Furthermore, for small fishing vessels, U-tube passive tanks are complicated, subject to fatigue and would probably be too costly. The first installation of a flume stabilization system on a fishing vessel took place in 1963 [1]. Since then, more and more such systems can be found on fishing vessels due to successful experiences with them.

1.2.6 Active Anti-roll Tanks

To overcome the dependence of the U-tube tank on resonance, a natural improvement is to install a pump(usually a variable-pitch pump) in the crossover duct, thus making it an active system. It is self evident that active tanks can have faster responses than passive tanks to waves in an irregular seaway, as well as a greater damping effect. The greatest advantage of an active tank compared to a passive one is that it takes into account every single wave, not just a train of waves. This makes it much more effective in an irregular seaway. However, the feed-back control system and the motor make it costly, and operating power, especially instantaneous power inputs, must be provided. In general, from the cost-efficiency sense, active anti-roll tanks do not appear attractive compared with active fins. Like active fins, active tanks are too expensive and complicated for small fishing vessels.

1.2.7 Paravanes

Paravane stabilizers, popularly known as "flopper stoppers", were invented by the U.S. west coast salmon fishermen probably more than 25 years ago [7]. Due to their relatively small size and fairly good roll reduction, they are now very popular on small fishing vessels from coast to coast in North America, even though little serious study has been found on their design.

Paravanes are small delta wings with added tail fins suspended by long chains and towed from the ends of booms on each side of the vessel. A typical paravane and towing arrangement is shown in Figure 1.2 and 1.3. The deployed depth of the paravanes needs to be greater than the effective depth of the waves in the resonant range to avoid any loss of the damping forces. When its boom moves downward, the paravane dives sharply keeping tension on the tow wire, and when its boom rolls upward pulling it toward the water surface, it assumes an angle of attack, thereby applying a downward force on the tow wire. In this way the towed paravanes alternately apply moments which resist the rolling of the vessel.

The effectiveness of a paravane is proportional to the wing area of the paravane, the length of the boom, and the towing speed. Koelbel, Fuller, and Hankley [7] developed a design methodology to determine the required wing area based on the amount of damping required to achieve a specified percentage of roll reduction. However, the paravane effectiveness coefficient must be determined from the full

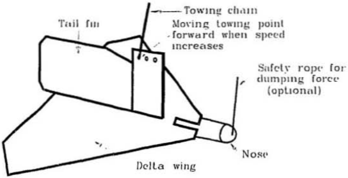


Figure 1.2: The shape of a paravane

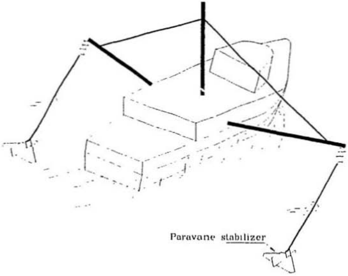


Figure 1.3: Towing arrangement of paravanes

scale tests. Goudey and Venugopal [6] found, from model tests, that paravanes are effective at roll damping, but their resistance penalty at steaming speed is large when compared with bilge fins. Because the angle of attack of a paravane increases as the towing speed increases, it may be desirable to move the towing point on the paravane forward at high speeds to avoid too large drag forces. Tests carried out by Bass and Weng [8] indicate that paravanes contribute quite significantly to the roll damping even at low and zero speeds, which is what a small fishing vessel needs.

Besides increasing resistance, another shortcoming of a paravane stabilizer is that if a paravane on one side is lost, the bias caused by the remaining paravane could lead to a dangerous condition and even a capsize in a heavy seaway [9]. It is believed that a quick break-away mechanism is necessary for both paravanes to cope with this situation. Alternatively, it is suggested in reference [7] that additional safety wires connected to the noses of the paravanes can be used (Figure 1.2). When one towing chain breaks, the operator can pull up the other paravane's nose thus dumping the load, then both paravanes can be saved. When the booms are used to tow nets, paravanes have to be taken back. But in this case, nets contribute some damping moments.

A paravane is a special type of roll stabilizer developed for fishing vessels which have large outrigger booms used to tow nets from both sides. Its wide acceptance in the fishery demonstrates that it is an economical and effective means of controlling roll.

1.3 The Scope of This Study

From the previous discussion, it is concluded that passive flume tanks and paravanes are the most applicable stabilizers for small fishing vessels. Between them, passive tanks are considered superior. Compared to passive tanks, paravanes may interfere with hauling, increase hull resistance, are not very effective at low and zero speeds, and need more maintenance. Particularly in the cold sea regions such as Newfoudland, floating ice may interfere with deploying paravanes and damage them. At present, passive tanks are not very commonly used on small fishing vessels while paravanes are already commonly used on small fishing vessels, therefore, the main impetus for this study is to find a more efficient but still affordable method to stabilize small fishing vessels. Investigations of effectiveness of a passive tank and a pair of paravanes on a fishing boat in various conditions will be carried out in model scale, and comparisons will be made between them.

The major problem in evaluating tank performance is that tank behavior and effectiveness are nonlinear with respect to wave slope because of nonlinear tank moments and saturation effects. Theoretical prediction procedures are considered very inadequate to recognize the nonlinear behavior of a tank, particularly when saturation effects occur. The study will be carried out mostly experimentally.

For paravanes, although they are already proven effective on small fishing vessels, existing studies are not considered exhaustive. There are a number of questions that remain to be answered. For example, are they equally effective at all amplitudes of roll? Are they equally effective in irregular waves as well as in regular waves? These questions should be answered.

Chapter 2

Design and Testing of Tank Model

2.1 Design of Tank Model

For the initial design of a tank model it is necessary to choose the approximate configurations of a tank whose natural frequency is close to the natural frequency of the ship model on which it will be tested. It has been found that for best performance, the natural frequency of a passive tank should be 6 to 10 percent higher than the ship natural frequency [12]. The model tests should be carried out under Froudian similarity law, which means the viscous forces are considered negligible. To avoid scale errors the model should not be too small, usually, at least 2 feet width is desirable [12]. However, the widths of many ship models are relatively small, and there is a danger of scale effects for a tank model installed in a model ship. In this case, it is common to use two tank models, a larger one for roll table tests, and the smaller one for ship model tests. It may be necessary to modify some structural details on the small tank based on the comparison between the tests of the large and small models, so that the same damping characteristics are obtained. Since the main purpose of this study is to provide some useful information on passive tank performance for fishing vessels, and not to design a prototype tank, scale effects were not considered, so, only one model was made.

The flume tank model was constructed in the shape shown in Figure 2.1, where

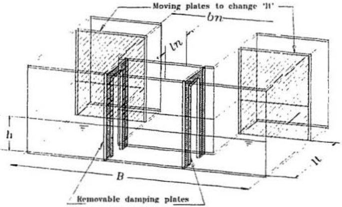


Figure 2.1: Tank Model

l_t is the width of the wing tanks and can be changed by moving the plates in them. b_n and l_n are the length and width of the flume respectively, and B is the overall width of the tank. The damping plates are removeable.

Each of these dimensions, as well as damping, affects in some degree the natural frequency of the tank. However, two factors can be determined at the beginning. First, it is obvious that the moment generated by the tank is proportional to its width. In practice, B is always chosen as large as possible, that is, equal to or close to the beam at midship. Second, the moment is also proportional to the amount of water in the tank Q_t. How much water is suitable depends on the desired roll reduction, the permissible GM losses (if h is fixed to get the right natural frequency, more water implies a greater free surface), and the available loading and space on the ship. Typically, about 2 percent of the ship weight is used.

Because the tank may be tested on several fishing vessel models at MUN, the design of the tank is based on an assumed fishing model which has a a weight of at most 80 kg, a beam at midship of approximate 500 mm, and a typical tested natural roll frequency 3.75 rad/s. So, the tank model will be 500 mm wide on the outside, and have 1-2 kg of water in it.

Three equations have been found to estimate a passive tank's natural frequency. For a simple rectangular tank without restriction, the natural frequency may be taken as [16]:

$$\omega_t = \frac{\pi}{B}\sqrt{gh} \tag{2.1}$$

where B and h are the width and the water level in the tank respectively.

Barr and Ankudinov [12] give another equation for flume tanks:

$$\omega_t = [\frac{\pi g}{B'}\tanh(\frac{\pi h}{B'})]^{\frac{1}{2}} \tag{2.2}$$

where: $B' = B + b_n(l_t - 0.9l_n)/0.9l_n$, and B, b_n, l_t, l_n are defined in Figure 2.1.

Chadwick and Klotter [17] gave an equation for passive U-tube tanks:

$$\omega_t = \sqrt{\frac{2g}{S}} \tag{2.3}$$

where: S = effective length of the U-tube

$= \int_0^l \frac{A_0}{A(s)} ds$

A_0 = area (constant) of the free surface in each wing tank of the U-tube or largest cross section of the U-tube

A = local cross-sectional area of the U-tube normal to the U-tube centerline (variable cross section)

s = girth-like coordinate along the centerline of the tank water

l = total "girth length" of the tank water

This equation is also valid for a flume tank if the wave-making effects are considered relatively small. In this case:

$$S = \int_0^l \frac{A_0}{A(s)} ds = h + \frac{(B - b_n)^2}{4h} + \frac{(B - b_n) l_t b_n}{2 h l_n}$$

For a given tank, equation 2.1 gives the largest value while equation 2.2 gives the smallest. The medium one, equation 2.3, has been used during the design procedure.

Fixing B, there are still four factors l_t, l_n, b_n, and h that need to be determined. Figures 2.2 to 2.5 show the various effects of changing dimensions on the natural frequency when $B = 500$ mm. For a given tank, ω_t increases as h increases (Figure 2.5). When h and the other three dimensions are fixed, Figures 2.2 to 2.4 give the trend of ω_t as only one dimension changes. Figures 2.2 and 2.3 show that the closer l_t and l_n are, that is, the more similar it is to a rectangular tank, the larger ω_t is. This is because, for the natural motion of water in the tank, the rate of transfer of water is larger. As for the effects of b_n on ω_t (shown in Figure 2.4), when $b_n/B < 0.5$, the effects are small, however, when $b_n/B > 0.7$, ω_t goes up sharply as b_n increases. This is because the percentage of water in the flume

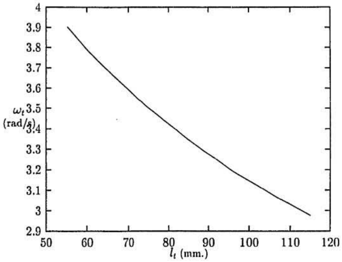


Figure 2.2: ω_t vs. l_t, $l_n = 40$, $b_n = 220$, $h = 50$

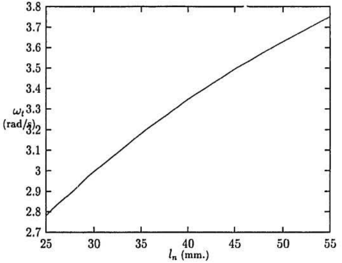


Figure 2.3: ω_t vs. l_n, $l_t = 85$, $b_n = 220$, $h = 50$

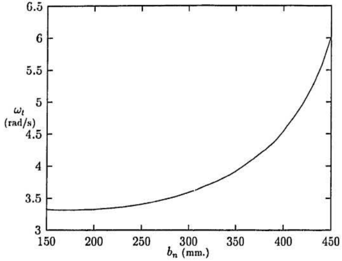


Figure 2.4: ω_t vs. b_n, $l_t = 85$, $l_n = 40$, $h = 50$

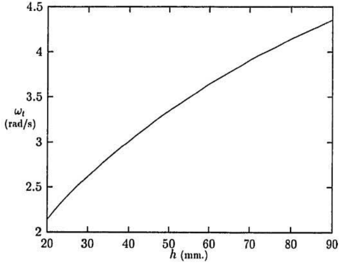


Figure 2.5: ω_t vs. h, $l_t = 85$, $l_n = 40$, $b_n = 220$

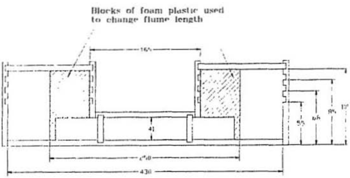
relative to the total amount of water is higher, therefore, the relative transfer of water is greater.

To determine l_t, l_n, b_n, and h for certain B, ω_t, and Q_t, two other basic shape factors b_n/B and l_n/l_t, which define the size of the flume relative to overall size of the tank, need to be chosen. There are no definite rules to do this. However, it is clear that one should avoid some extremes. If the flume is too large, h will be too small because Q_t is fixed. On the other hand, too small flume results in a large restriction of water motion, and in order to get the wanted natural frequency, too high a water level is needed. After several trials and endeavouring to keep h moderate, b_n/B is set to 0.4 and l_n/l_t to 0.5 in this design. Then, after solving the set of equations below, all dimensions can be obtained.

$$\begin{cases} \omega_t = \sqrt{\frac{2\times 9.81}{S}} = 3.75 \\ S = h + \frac{(B-b_n)^2}{4h} + \frac{(B-b_n)l_t b_n}{2hl_n} \\ Q_t = h[Bl_t - b_n(l_t - l_n)] = 2\% \times 0.08 \\ B = 0.45 \\ \frac{l_n}{l_t} = 0.5 \\ \frac{b_n}{B} = 0.4 \end{cases} \Longrightarrow \begin{cases} l_t = 0.089\ m. \\ l_n = 0.045\ m. \\ b_n = 0.18\ m. \\ h = 0.050\ m. \end{cases}$$

Due to the enormous number of possible configurations and many other factors involved, this estimation is fairly rough and may not be optimal. During the following experiments, several modifications were tried by changing l_t and putting some solid materials in the tank to increase b_n. Figure 2.6 shows the dimensions of the final tank model made.

The inside dimensions are a little smaller than designed due to the thickness of material. The model is made of plastic glass so that the movement of the inside water can be observed. Two pairs of damping plates had been prepared for tests. One has one 9.5 mm wide slot in each plate, and the other has two 8.8 mm wide



Note Tank height = 150. All dimensions are in millimetres

Figure 2.6: Configurations of the tank model

slots, which cause less damping, in each plate. The damping plates for the modified tank with increased b_n have one 15 mm wide slot in each.

2.2 Oscillating Bench Tests of Tank Model

Before testing a tank model on the ship model, it is common to test it on a rolling bench first, because bench tests can be carried out much cheaper and faster. The bench test results, including the moment generated by the tank versus roll frequency and the phase lag versus roll frequency, are obtained to determine tank configuration, tuning and damping. The amount of damping considered to be satisfactory is not fixed. The determining factors include response of the unstabilized vessel, size of the tank, and the frequency range of operation. Bosch and Vugts [18] developed a design procedure for free-surface tanks using the derived data from a series of bench tests. The state-of-the-art for bench test technology is the roll/sway table of the Anti-roll Tank Facility in David Taylor Naval Ship Research and Development Center [11], which is a simulation facility consisting of a roll/sway table capable of accommodating models up to 4 ft.(1.3 m) in width. An analog computer is used to simulate the dynamic characteristics of the ship in lateral motion and to provide roll and sway signals to the table. The tank-generated roll moment and the sway force and yaw moment are fed back to the analog computer, providing a closed-loop simulation of the ship/tank system. Unfortunately, few laboratories can afford to duplicate such a facility.

Due to financial and time limits, a fairly simply rolling table was constructed, together with a pair of small wave probes used to measure the water elevation in the tank. A dynamometer to measure the moment generated by the tank was not available. It has to be noted here that wave probes only measure the water level at one point in the wing tank, and it might not reflect the instant of maximum water transfer due to the oscillation of the water in the wing tank. This defect is apparent in the later experiments. Therefore, if finances permit, a dynamometer is recommended. In this study, it was assumed that the out-of-phase angle of the tank moment ε_t has approximately the same value as the out-of-phase angle of the water level in the tank. Figure 2.7 shows the principle experimental setup. The motion of the bench driven by the method shown is not exactly sinusoidal, however if the lever is made much longer than the rolling radius, it is very close to a sinusoidal motion. The amplitude of rolling motion was set to be about 10 degrees. For each l_t and h, tests have been done on a range of rolling frequencies, which are obtained by adjusting the supplied voltage to the motor, with different damping conditions (see Table 2.1 and Figure 2.7 for details). The natural frequencies of the tank were obtained from the free decay tests by giving it an impulsive start. Figure 2.8 shows the tank water elevations in a decay test. The curve is analyzed

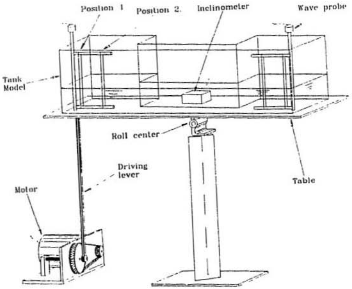


Figure 2.7: Oscillating bench test setup

by a FORTRAN routine giving Fourier fit and frequency.

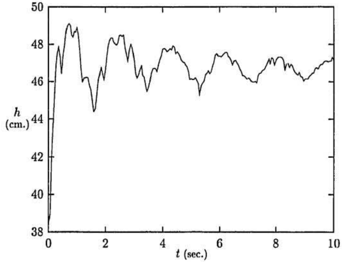


Figure 2.8: A tank decay test for obtaining natural frequency

Table 2.1 shows that the natural frequencies of the tank are about ten percent lower than expected. This is due to several factors, including the inaccuracy of the equation because the formula is based on the assumptions of 'no damping' and 'no free surface' in the tank. Scaling effect can also be considered a factor. Physically, the phenomenon in the tank is dominated by gravity forces and viscous forces have to be neglected when Froudian scaling has to be followed. For large tanks, this could cause little problem due to the very large gravity forces compared to the viscous forces. But in a fairly small tank, viscous forces could be relatively large and cannot be neglected. In this case, viscous forces will cause a longer transfer period than estimated. This is also confirmed by the fact that the higher damping in the flume, which causes larger viscous forces, gives lower frequencies (Table 2.1).

Table 2.1: Tank model natural frequencies

Tank Model		$Q_t/80\,kg.$ (%)	ω_t (rad/s) with low damping	ω_t (rad/s) with high damping	ω_t (rad/s) from eqn. 2.3
l_t (mm.)	h (mm.)				
105	36	1.53	2.735	2.385	3.044
	47	2.0	3.027	2.65	3.458
	58	2.47	3.408	2.925	3.813
85	43	1.55	3.187	2.75	3.583
	55	1.99	3.643	3.125	4.017
	67	2.42	4.012	3.595	4.385
68	49	1.5	3.756	3.23	4.112
	66	2.02	4.277	3.825	4.692
	83	2.54	4.9	4.375	5.152
55	45	1.2	3.881	3.63	4.249
	57	1.5	4.398	4.009	4.704
	76	2.0	5.179	4.655	5.301

Three channels of data from the tests (Figure 2.9) were first analyzed by a program provided by D.W. Bass which gives the best Fourier fit of each curve as well as the frequencies and amplitudes. Then, those fits were calculated by another program giving the phase lag ε_t of the tank-water motion to the roll motion. ε_t was taken from the average value of phase lags obtained from each wave probe relative to the roll motion. Figure 2.10 and equation 2.4 illustrate the strategy. From Figure 2.10 one can easily see that equation 2.4 gives the phase lag in degrees of two sinusoidal motions with the same frequency.

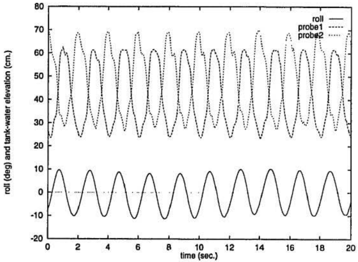


Figure 2.9: Showing tank-water elevations and roll motions

$$\varepsilon_t = \frac{-180 \times (\Delta t_1 + \Delta t_2)\omega}{2\pi} \tag{2.4}$$

Tests were first carried out on four wing-tank widths (l_t) and three waterlevels (h) with the wave probe position at the edge of each wing tank (Figure 2.7, position

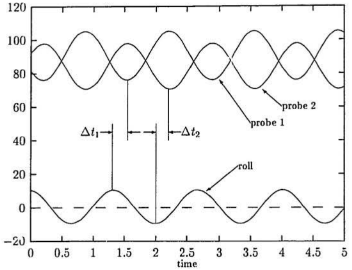


Figure 2.10: Fits of tank-water elevations and roll motions

1). The results of ε_t are plotted in Figures 2.11 to 2.14, versus roll frequency ω. In the plots, high damping means that there is one slot in the damping plates, and low damping means two slots. High, mid, and low water refer to tank water amount are 1.5%, 2.0%, and 2.5% of 80 kg respectively (Table 2.1).

By rights, ε_t should increase as ω_t increases, and be close to $-90°$ as ω approaches ω_t. However, the results shown in Figures 2.11 to 2.14 are 20 to 30 degrees lower than expected. This is because, as mentioned previously, the instant of water level at the probe position reaching a peak is a little bit ahead of the instant that the water stops transfering and begins to flow to the other side. For this reason, another set of probe positions located at the middle of the wing tanks (Figures 2.7, position 2) was tested for the medium water levels. Compared results are plotted against nondimensionalized frequency ω/ω_t in Figures 2.15 and 2.16.

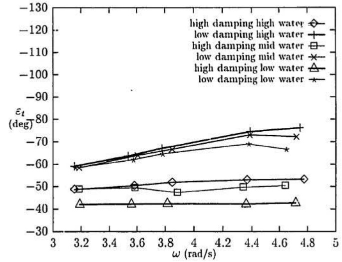


Figure 2.11: ε_t, bench tests for $l_t = 105$ mm.

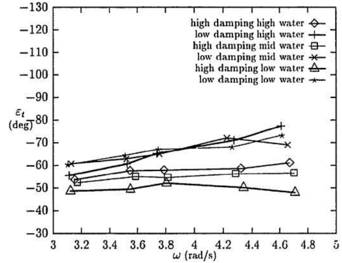


Figure 2.12: ε_t, bench tests for $l_t = 85$ mm.

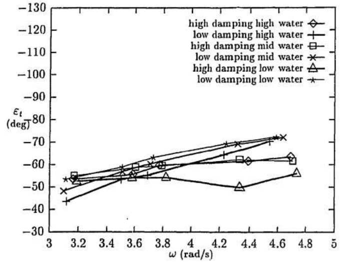


Figure 2.13: ε_t, bench tests for $l_t = 68$ mm.

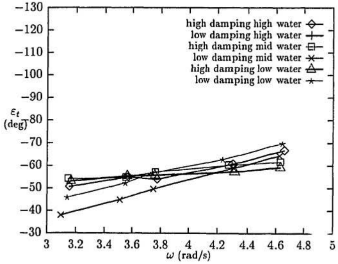


Figure 2.14: ε_t, bench tests for $l_t = 55$ mm.

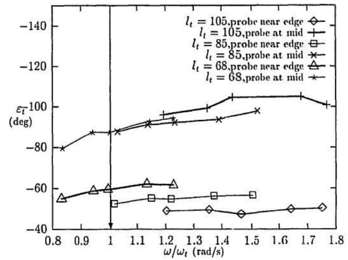


Figure 2.15: Comparison of different probe positions with high damping

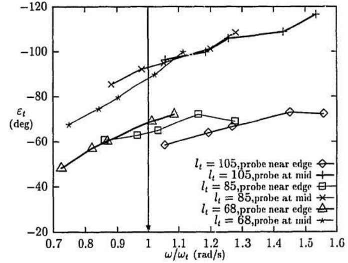


Figure 2.16: Comparison of different probe positions with low damping

It is shown that, with the probes at the middle of the wing tanks, ε_ts are close to $-90°$ when ω/ω_t equals 1. This probe position appears to give more reasonable measurement and is applied in the later tests. It is also noticeable in Figure 2.15 and 2.16 that the change rates of ε_t are broadly the same for two positions. Therefore, the results in Figure 2.11 to 2.14 are still valid providing that only the change rates of ε_t versus ω_t are considered.

As shown in Figures 2.11 to 2.14, for high damping conditions, ε_t hardly changes in the range of ω, while for low damping conditions, ε_t is a little more sensitive to ω. It can be said that the slope of $\varepsilon_t - \omega$ curve is an indicator of damping level. High damping makes ε_t vary little with ω. In other words, the tank does not need to be re-tuned in a large range of roll frequencies. However, high damping restricts the amount of transfered water, thus reducing the stabilizing moment generated by the tank. Choosing proper damping is critical for optimizing tank performance and is one of the main purposes of a bench test. It is believed that the tank is over damped in the previous tests. Scale effects could be one factor causing this.

Tests were then carried out for the modified tank with increased b_n (Figure 2.6 and Table 2.2). The purpose of trying this is to concentrate more water in the

Table 2.2: Natural frequencies of the modified tank

Tank water level h (mm.)	$Q_t/80\,kg.$ (%)	ω_t (rad/s) with damping plates	ω_t (rad/s) without damping plates	ω_t (rad/s) from eqn. 2.3
49	1.78	3.166	3.32	3.757
60	2.19	3.40	3.70	4.121
70	2.55	3.698	3.97	4.41

wing tanks and thereby get larger stabilizing moment. The first set of tests are for the tank with damping plates that have one 15 mm wide slot in each. In Figure 2.17, when $\varepsilon_t = -90°$, from low water level to high level, ω is covered from 3.1 rad/s to 3.9 rad/s, which means the expected roll frequency range is included in the frequency realm of the tank. In the tested frequency range, ε_t is approximately linearly dependent on ω.

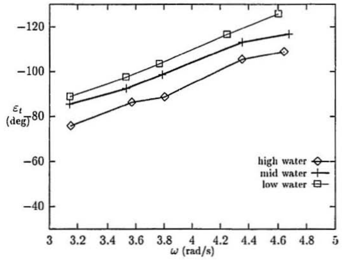


Figure 2.17: ε_t of the modified tank with damping

It should be noticed that when ε_t is plotted versus the nondimensional frequency (Figure 2.18), the effect of waterdepth in the tank becomes insignificant for the relative frequencies. Ideally, if we assume both ω_t and ε_t are linearly dependent on h and ω respectively, the $\varepsilon_t \sim \dfrac{\omega}{\omega_t}$ line should be unique for one tank with certain internal damping level which is reflected by the slope of the line.

Since the internal damping of this tank seems still a little large compared to the tests in references [15] and [19], according to the slope of the $\varepsilon_t - \omega$ curves, it

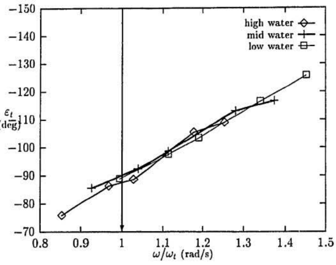


Figure 2.18: ε_t vs. nondimensional frequency, modified tank with damping

was decided to test this modified tank without any damping plates. Much sloshing was experienced and it was necessary to seal the tops of the wing tanks to avoid water splashing out.

As expected, the $\varepsilon_t - \omega$ curves without damping (Figure 2.19) are steep compared to those with damping (Figure 2.17). By observing the water motion in the tank, it is concluded that near the resonant frequency, the moments generated by the tank (both static moment and dynamic moment) increase as the internal damping decrease, because more water moves back and forth with a faster speed. However, because the phase lag changes fast versus roll frequencies when the internal damping is small, the tank could soon be wrongly tuned at non-resonant frequencies and could possibly increase roll angles instead of decreasing them. Without tank moment data, it is hard to say what damping level is the best until

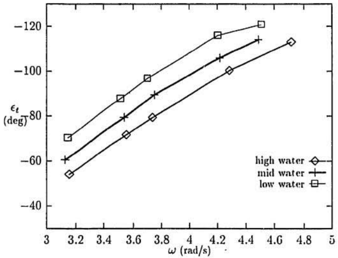


Figure 2.19: ε_t of the modified tank without damping

on-ship tests are carried out.

The modified tank gives better phase angle responses and is chosen to be tested on a ship model.

Chapter 3

Experiments in Calm Water

It was decided to use one of the available fishing boat models which has the least hull damping so that the expected damping from tank or paravanes is more obvious. Model 366 is the one chosen and its body plan and particulars are given in Figure 3.1.

As concluded in the previous chapter, the modified tank model with increased flume length (Figure 2.6) was used in the test.

A pair of small paravanes was also made for the tests. The shape and scaled dimensions of the paravanes are similar to those commonly used in Newfoundland fishing vessels. Figure 3.2 shows the detailed dimensions of the model.

3.1 Experimental Setup

Three tank water levels and two internal damping levels were chosen to be tested in the following tests. According to the results of the rolling bench tests, the tank will be tested with the damping plates, which have one 15 mm wide slot in each of them, and without any damping plates. Taking into consideration that the liquid

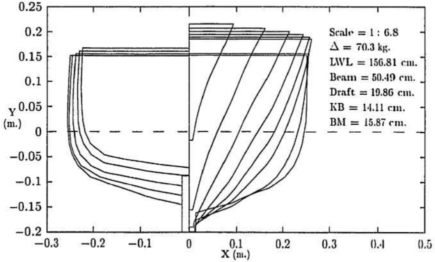
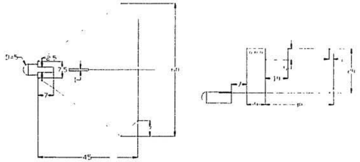


Figure 3.1: Body plan and particulars of Model 366



Note: all dimensions are in millimetres

Figure 3.2: Dimensions of the paravane model

level should not be too high to avoid spilling over the sides, three water levels in the tank model are tested throughout the experiments. They are 30 mm, 47 mm, and 60 mm, which match 1.3, 2.0, and 2.55 percent of the displacement of Model 366 respectively. The natural frequencies of these three levels with two levels of internal damping are given in Table 3.1. As the tank natural frequencies were

Table 3.1: Natural frequencies of the tank model tested on M366

h (mm.)	$Q_t/70$ kg. (%)	ω_t (rad/s) with damping	ω_t (rad/s) without damping	ω_t (rad/s) from eqn. 2.3
30	1.3	2.30	2.55	2.98
47	2.0	2.77	3.19	3.69
60	2.55	3.40	3.70	4.12

lower than expected, M366 is tested at its natural frequency ballasted to about 3.0 rad/s, as well as at 3.75 rad/s. Each natural roll frequency was kept unchanged throughout each set of tests which includes various tank water levels and damping conditions. Due to free surface effects, GM will decrease when the tank is filled with some water. The GM loss due to a flume tank can be calculated as [3]:

$$\delta GM = \frac{i}{\nabla} = [l_t(B^3 - b_n^3) + l_n b_n^3]/12\nabla \tag{3.1}$$

where i is the moment of inertia of the free surface area and ∇ is the volume of displacement of the ship. Equation 3.1 gives the result $\delta GM = 8.9$ mm for the tested model. However, the natural roll frequency of the boat may not consequently decrease because of the dynamic effects of the tank. In fact, it was found that ω_ϕ has a small increase when the tank is filled with some water. Table 3.2 lists out the GM obtained from inclining tests for different tank levels. Note that the center of gravity of the model has been modified for each tank water level so that ω_ϕ is

Table 3.2: GM values of M366 with various tank water levels

h (mm.)	GM (mm.) $\omega_\phi = 3.00$ rad/s	GM (mm.) $\omega_\phi = 3.75$ rad/s
0	37	52
30	27	41
47	29	41
60	29	43

kept constant. In this case, the true GM's obtained from the inclining tests can be expressed as:

$$GM_f = GM_s - \delta GM \pm \delta G$$

where GM_s and GM_f are the GM's before and after the tank is filled with water, and δG is the distance between the modified center of gravity and the original one. Here δG can be regarded as the compensation of the combined influence of dynamic effects of the tank and δGM on natural roll frequency. It can be seen that at the same ω_ϕ, GM of the model with water in the tank is about 20-30% lower than that of the model without water in the tank. Because GM is the handle by which the waves rock the vessel, the stabilizing function of a tank also partially comes from decreasing GM.

Experiments were carried out in the wave tank at Memorial University, which is 57 m. long, 4 m. wide, and 2 m. deep. The tank model was fixed at midships where two light metal booms suspending paravanes were also mounted athwartships. The paravanes are at a distance of 80 cm from the center line of the model. Figure 3.3 shows the experimental setup. The vertically moveable weight was to be used for the adjustment of ω_ϕ.

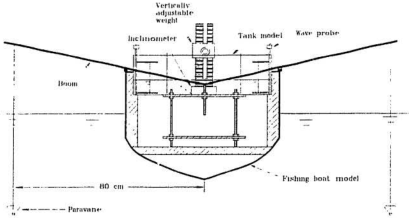


Figure 3.3: Experimental setup

3.2 Free Roll Decay Tests

To identify the damping moments generated by tank and paravanes, free roll decay tests were first performed on M366 with and without tank and paravanes. Experiments were carried out for a series of initial angles for each tank water level and roll frequency. The roll decay curves were analyzed in terms of the non-dimensional equivalent linear damping coefficient ζ_E. The tests with both tank and paravanes were not performed because the roll decays too fast. The values of ζ_E are plotted against mean amplitude of roll angles ϕ_m in Figures 3.4 to 3.8, where the points are experimental data and the lines are linear regressions.

Tests of Tank with Damping Plates

For the values of ζ_E in Figures 3.4 and 3.5, when $h = 0$, the variation of ζ_E with ϕ_m is clearly linear and can be expressed in the form:

$$\zeta_E = m\phi_m + C \tag{3.2}$$

But after the tank is filled with some water, the data points become more scattered, especially in Figure 3.5 where the points are very scattered and it is difficult to

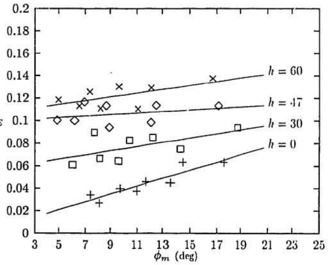


Figure 3.4: ζ_E of M366, tank with damping, $\omega_\phi = 3.75$

conclude any general relationship between ζ_E and ϕ_m. This is because of the different mechanism between the free roll decay of a ship without tank and that of a ship-tank system. In the case of a ship-only decay, the motion equation of a vessel rolling in calm water, in the case of a linearized analysis of roll damping, is given by:

$$I_0\ddot{\phi} + B_E(\phi_m)\dot{\phi} + D(\phi) = 0 \tag{3.3}$$

where B_E is the amplitude dependent equivalent linear damping coefficient, I_0 is the virtual moment of inertia in roll, and $D(\phi)$ is the non-linear restoring moment. When a tank is installed, the equation becomes:

$$I_0\ddot{\phi} + B_E(\phi_m)\dot{\phi} + D(\phi) - M_t(\phi)\sin(\omega_\phi t + \varepsilon_t) = 0 \tag{3.4}$$

Unlike bilge keels or paravanes, whose stabilizing moment can be simplified as $B_S\dot{\phi}$ where B_S is regarded as an increment of B_E, the moment generated by a stabilizing

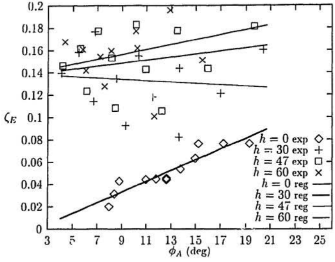


Figure 3.5: ζ_E of M366, tank with damping, $\omega_\phi = 3.00$

tank is dependent on ϕ, and the result of equation 3.4 also depends on the phase lag ε_t.

Nevertheless, assuming the tank moment can be treated as an increment of hull damping moment, the regression lines in Figures 3.4 and 3.5 manifest the average 'damping' increase due to the tank. At both roll natural frequencies, the higher tank water level gives higher 'damping'.

In equation 3.2, the magnitude of the constant C is indicative of the wave damping component of the roll and the magnitude of m indicates the significance of the viscous component of the damping [21]. The wave damping is considered linearly dependent on roll velocity while the viscous damping is usually nonlinearly dependent [22]. From Figures 3.4 and 3.5, one can see that the tank basically introduces 'linearity' into the roll damping, in the sense that the damping moment appears to be independent of amplitude. This is reasonable because the action of a passive tank is actually a kind of energy dissipation by wave-making. Unlike bilge keels, a stabilizing tank provides fairly large damping at small roll angles. For this vessel, when $h = 47$ mm and $\phi_m = 7°$, the damping is more than tripled at $\omega_\phi = 3.75$ rad/s and increases by over 400% at $\omega_\phi = 3.00$ rad/s. However, the increment of the damping due to the tank decreases as ϕ_m increases.

Overall, the damping provided by the tank at $\omega_\phi = 3.00$ rad/s is about 20% to 30% higher than at $\omega_\phi = 3.75$ rad/s. This can be explained by the fact that compared to the tank moment, the restoring moment($D(\phi)$ in eqn. 3.4) is relatively small due to the GM reduction at lower frequency, this causes the ship roll to decay faster. Another reason for this could be tuning, that is, the tank natural frequency is better tuned for the roll frequency 3.00 rad/s, which is proved in later tests in beam waves.

Tests of Tank without Damping Plates

The most noticeable phenomena in Figures 3.6 and 3.7 are that the damping that comes from the tank at low roll amplitudes (ϕ_m) is higher than the damping at high roll amplitudes. The main reason for this may be the decreasing of free surface effects with roll amplitude. Saturation could be the other reason. Without damping the amount of transfer water is largely increased, and causes saturation at large roll amplitude.

Compared to the tank with damping, ζ_E is about doubled at $\omega_\phi = 3.75$ rad/s and increased by 40% at $\omega_\phi = 3.00$ rad/s for low roll amplitudes. However ζ_E keeps about the same for both damping conditions at high roll amplitudes. This could be the result of the different tuning (different tank frequencies (ω_t) in Table 3.1), the higher stabilizing moments, and some dynamic effects. The reasoning of these combined effects is complicated and is beyond the scope of this study.

At the high natural roll frequency(Figure 3.6), the high water level($h = 60$ mm) no longer works better than the medium level($h = 47$ mm), and at the low frequency(Figure 3.7), the medium level even does better than the high level. This indicates that when the tank is less damped, in other words, the ship-tank system

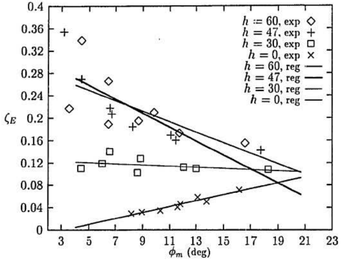


Figure 3.6: ζ_E of M366, tank without damping, $\omega_\phi = 3.75$

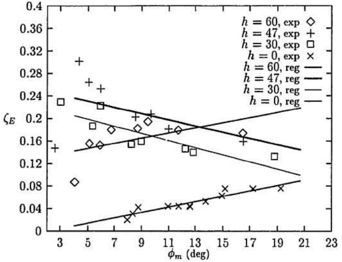


Figure 3.7: ζ_E of M366, tank without damping, $\omega_\phi = 3.00$

is 'softer', the tuning and dynamic effects become more important than the amount of water in it.

Tests of Paravanes

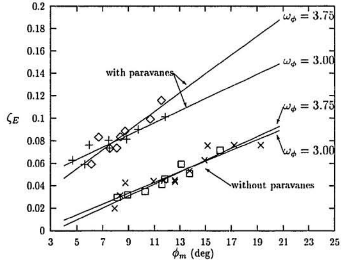


Figure 3.8: ζ_E of M366 with paravanes

Surprisingly, the paravanes contribute quite significantly to the roll damping at zero speed. As shown in Figure 3.8, at $\phi_m = 8°$, the damping is more than doubled. The comparatively small forces generated by paravanes are offset by the long booms. At $\omega_\phi = 3.75$ rad/s, a small increase of the non-linear damping component is introduced. But at $\omega_\phi = 3.00$ rad/s, the slope of the damping curve using paravanes is quite similar to that not using paravanes. This indicates that the damping comes from 'lift' forces rather than 'drag'. The lift forces on the paravane are generated as it moves in a circular path induced by the upward pull of the line at the point of attachment [8].

Compared to the tank, the damping of the paravanes is small at small roll angles but increases faster as roll amplitude increases. In the region of $\phi_m < 10°$, for the medium waterlevel, the damped tank provides 30% to 70% more damping , and the less damped tank provides 100% to 200% more damping than the paravanes (comparing Figures 3.4 - 3.7 to Figure 3.8). The paravanes contribute more damping than the tank with medium and high water levels only after the mean roll amplitude exceeds fairly high values which are usually not often attained when a stabilizer is installed. The high damping from a tank at small roll angles could prevent the initial small roll motion of a ship in waves from developing to a large one.

Chapter 4

Experiments in Regular Beam Waves

Model 366 installed with the tank and paravanes was tested in regular beam waves of two wave heights: 5 cm and 10 cm. The intention of the experiments was to investigate the effectiveness of the tank and paravanes when the boat is free to roll at zero speed in light and moderate waves. As mentioned in chapter one, the efficiency of a tank is independent of the speed of the ship. But for the paravanes it is a different situation. The efficiency of the paravane decreases as the towing speed decreases due to the decrease of lifting forces. The tests for the same paravanes on another ship model [8] show that the non-dimensional equivalent linear damping ratio ζ_E at Froude number 0.2 is about twice as large as ζ_E at zero speed. Because it is not feasible to carry out tests in waves at forward speed in a small towing tank, the tests were only done for zero speed. However one should remember here that the effectiveness of the paravanes is least when it is compared to the effectiveness of the tank. To restrict excessive yaw and drift, the model was lightly tethered at the bow and stern. The tests were carried out for a range of wave frequencies near the two natural roll frequencies, that is, 3.00 rad/s and 3.75 rad/s. When both tank and paravanes are applied, the medium tank water level (47 mm) is used.

4.1 Measures of Effectiveness of a Stabilizer

The obvious measure of the efficiency of a stabilizer is the roll reduction that can be obtained. The roll reduction factor r_f is defined as:

$$r_f = 1 - \frac{\sigma_s}{\sigma_u} \tag{4.1}$$

where σ is the value of roll angle amplitude and subscripts s and u indicate, respectively, stabilized and unstabilized values. In regular beam seas, σ usually refers to the resonant roll angle at wave heights that are within the capacity of the stabilizer [2]. In a seaway, r_f is commonly used as a statistical measure of effectiveness, and σ is the root-mean-square (rms) value of roll angle amplitude and can be replaced by other statistical measures of roll angle amplitude such as mean or significant values [11].

The objection to the use of the roll reduction factor r_f is that it does not discriminate between small and large roll angle amplitudes. A much more pertinent roll reduction factor r_{df} can be defined which focuses on the reduction in the occurrence of roll angle amplitudes greater than a specified limiting value ϕ_a^*. ϕ_a^* is selected on the basis of ship mission and, when the stabilizer system is designed according to a criterion, is the limiting roll angle amplitude value used in the criterion. The discriminating roll reduction factor is defined as [11]:

$$r_{df} = \exp[-\frac{1}{2}(\phi_a^*)^2(\frac{1}{\sigma_s^2} - \frac{1}{\sigma_u^2})] \tag{4.2}$$

Because this is a general study and there is no certain design criterion for the stabilizers, the simple roll reduction factor r_f is used for the evaluation. However, since the roll reduction varies with the ship's stability and speed as well as wave regularity, direction, height and period, one should be careful when using r_f to compare stabilizers [2].

4.2 Tests of Tank with Damping Plates

Figures 4.1 to 4.4 show that the tank and paravanes provide 11% to 71% roll angle reductions at or close to resonant frequencies (also see Table 4.1), while they have little effect on roll at non-resonant frequencies. The fact that the stabilized roll responses have apparently high peaks at resonant frequencies indicates that the tank has too much internal damping. It is also shown in Table 4.1 that, for both natural roll frequencies, the stabilizers work better in light waves. The reason for this could be that the components in the tank moment caused by dynamic effects is not linearly dependent on roll amplitudes. The 'free surface' effect is relatively large at small angles of roll. The total tank moment therefore increases at a slower rate as the roll amplitude increases. This is also true for the paravanes because, as found in the previous decay tests, the forces generated by paravanes are mainly lift forces that are approximately independent of roll amplitudes. For large amplitudes 'stall' type effects may occur and that result in reduced lift forces but increased drag forces, and the total damping forces could be relatively smaller.

For the high natural roll frequency $\omega_\phi = 3.75$ rad/s, the effectiveness of the tank cannot be said to be satisfactory. The best reduction achieved by high tank water level is 25% in moderate waves and 46% in light waves. The peaks of the roll responses do not shift for different tank water levels. Tuning factors seem to have little effect on the roll motion. In Figures 4.5 and 4.6, the phase lag ε_t of tank water motion to roll motion drops down at the resonant frequency for low and medium water levels in moderate waves and for low level in light waves. This could be an explanation for the above phenomenon because phase lags ε_t are closer for different tank levels. The reasons for phase lags ε_t dropping down must be excessive roll angles and heave and sway motions because this did not happen in the bench tests (Figures 2.17). It can be concluded that, when the frequency of the tank is low compared to the natural roll frequency, the motion of tank water is more susceptible to other motions (sway is considered more important than heave).

For the low natural roll frequency, the tank works a little better. Overall, the peak roll angle reductions are about 20% higher than at the high natural roll frequency (Table 4.1). The first reason for this is the smaller wave exciting moment due to the lower GM value, therefore the tank moment is relatively larger. The second reason could be tuning. Figure 3.1 shows that the natural frequencies of the tank are lower than 3.0 rad/s for the low and medium water levels and a little higher than 3.0 rad/s for the high level. As mentioned, the best tuning is achieved when the natural frequency of the tank is a little higher than that of the ship. This is proved in Figures 4.7 and 4.8 showing that phase lags ε_t for the high level is close to -90 degrees when the roll frequency is 3.0 rad/s.

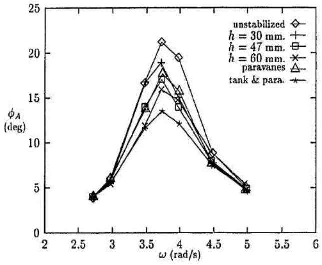


Figure 4.1: Roll response of M366 with damped tank, $\omega_\phi = 3.75$, $h_w = 10$

The experiments also show that the paravanes provide 17% to 37% roll reductions. The effectiveness of the paravanes is not significantly different for different natural roll frequencies, however they are more effective in light waves.

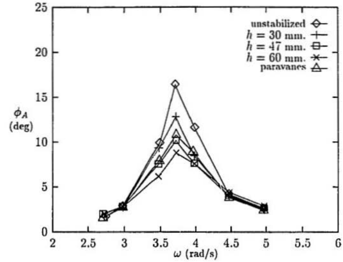


Figure 4.2: Roll response of M366 with damped tank, $\omega_\phi = 3.75$, $h_w = 5$

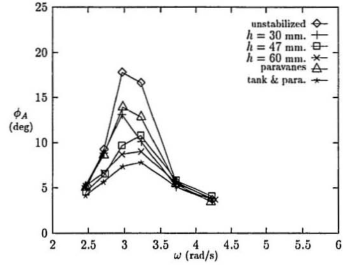


Figure 4.3: Roll response of M366 with damped tank, $\omega_\phi = 3.00$, $h_w = 10$

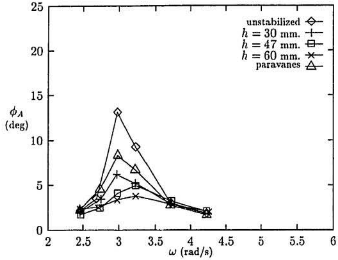


Figure 4.4: Roll response of M366 with damped tank, $\omega_\phi = 3.00$, $h_w = 5$

Table 4.1: Peak roll reductions for damped tank and paravanes

Stabilizer conditions	$\omega_\phi = 3.75$ $h_w = 10$	$\omega_\phi = 3.75$ $h_w = 5$	$\omega_\phi = 3.00$ $h_w = 10$	$\omega_\phi = 3.00$ $h_w = 5$
$h = 30$ mm.	11%	22%	26%	53%
$h = 47$ mm.	19%	38%	39%	62%
$h = 60$ mm.	25%	46%	49%	71%
Paravanes	17%	34%	22%	37%
Tank & Para	36%	-	56%	-

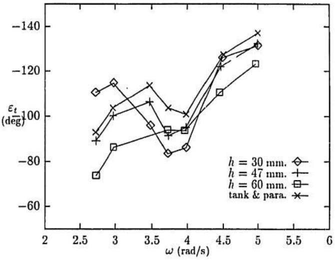


Figure 4.5: Phase lags of damped tank, $\omega_\phi = 3.75$, $h_w = 10$

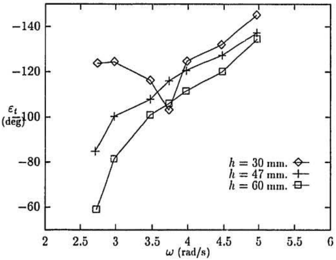


Figure 4.6: Phase lags of damped tank, $\omega_\phi = 3.75$, $h_w = 5$

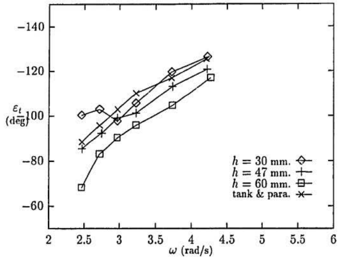


Figure 4.7: Phase lags of damped tank, $\omega_\phi = 3.00$, $h_w = 10$

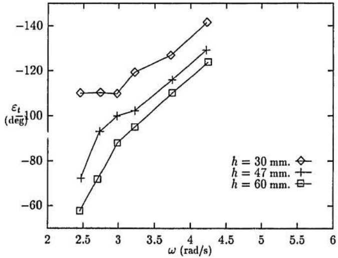


Figure 4.8: Phase lags of damped tank, $\omega_\phi = 3.00$, $h_w = 5$

When both tank and paravanes are used, the total roll reduction at high natural roll frequency is just the sum of reductions of each stabilizer, while at low frequency it is a little smaller than the sum. The latter could be the result of the small decrease of tank moments due to smaller roll angle.

4.3 Tests of Tank without Damping Plates

The previous bench tests and wave tests imply that the tank may have too much internal damping. The natural frequencies of the tank in Table 3.1 also indicate that the tank without damping may have better tuning at the tested frequencies. Therefore, the wave tests were then carried out for the tank with no damping plates.

The following Figures show that the boat gets much better roll reductions after the tank damping is reduced. The peak roll angle reductions for the medium and high tank water levels range from 45% to 70% (Table 4.2), which can be said to be satisfactory according to previous experience [13, 11, 4, 14]. For the high natural roll frequency, the stabilized peak roll angles shift to higher frequencies, peaks (Figures 4.9 and 4.10), which means that the natural frequency of the tank is lower than the resonant frequency, therefore the tank provides more roll reduction at low frequencies. For the low natural roll frequency, which is closer to the tank natural frequency, the fact that the roll responses show two peaks (Figures 4.11 and 4.12) indicates that the tank moments are large enough to counteract the wave exciting moments.

Giving another view of tank's action on the roll motion, Figures 4.13 and 4.14 show the amplification factors ϕ_A/α_0 for the rolls at the low natural frequency. α_0 is the wave slope, and for a sinusoidal wave is given by:

$$\alpha_0 = \frac{\zeta_A \omega^2}{g} \tag{4.3}$$

where ζ_A is the wave amplitude and ω is the wave frequency. It can be conjectured from both figures that the tank eventually moves the maximum values of amplification factor to lower frequencies, although the tested frequencies are not low enough to show them. The benefits of a passive tank can be regarded as compensation of reducing safety in long waves. The amplification factors are small compared to those of unstabilized vessel at the resonant frequency, and waveslopes are also small except in exceptional large waves. Figures 4.13 and 4.14 also show clearly that the tank is more effective in light waves. Using roll response operators obtained from model experiments in regular waves and the results of full-scale trials, Morenshildt [11] indicated that the regular waves used in the model experiments had wave slopes in the range of 4 to 5 degrees, due to nonlinear behavior with wave slope, tank effectiveness will be overestimated for small wave slopes (a common practice) and underestimated for too large a wave slope value. The wave slopes in most of these experiments are fairly small, hence the results probably belongs to the case she referred to. Therefore the effectiveness obtained here may not be directly extrapolated to full scale.

Table 4.2: Peak roll reductions for undamped tank

Stabilizer conditions	$\omega_\phi = 3.75$ $h_w = 10$	$\omega_\phi = 3.75$ $h_w = 5$	$\omega_\phi = 3.00$ $h_w = 10$	$\omega_\phi = 3.00$ $h_w = 5$
$h = 30$ mm.	27%	43%	42%	68%
$h = 47$ mm.	45%	64%	58%	69%
$h = 60$ mm.	44%	65%	59%	70%
Tank & Para	51%	-	62%	-

Compared to the previous tests for the damped tank, the roll responses are more sensitive to water levels. Tuning becomes more important when the tank moment is large. It is noticeable that the medium tank level has the same effectiveness as the high tank level does (Table 4.2). Although the tank moment may be larger because

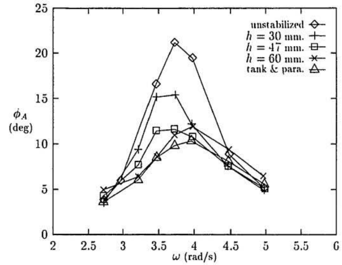


Figure 4.9: Roll response of M366 with undamped tank, $\omega_\phi = 3.75$, $h_w = 10$

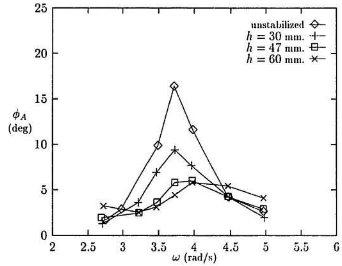


Figure 4.10: Roll response of M366 with undamped tank, $\omega_\phi = 3.75$, $h_w = 5$

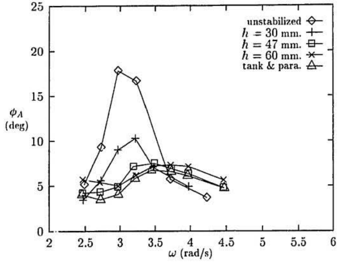


Figure 4.11: Roll response of M366 with undamped tank, $\omega_\phi = 3.00$, $h_w = 10$

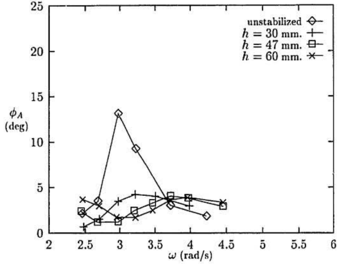


Figure 4.12: Roll response of M366 with undamped tank, $\omega_\phi = 3.00$, $h_w = 5$

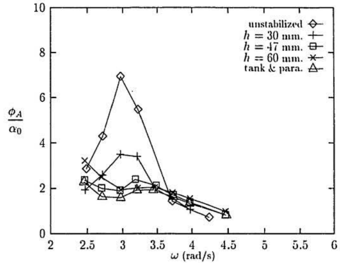


Figure 4.13: Amplification factor at $\omega_\phi = 3.00$, $h_w = 10$

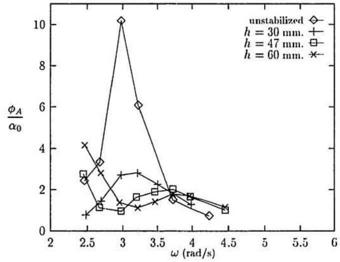


Figure 4.14: Amplification factor at $\omega_\phi = 3.00$, $h_w = 5$

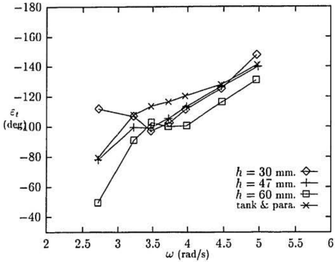


Figure 4.15: Phase lags of undamped tank, $\omega_\phi = 3.75$, $h_w = 10$

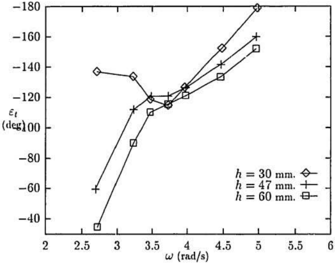


Figure 4.16: Phase lags of undamped tank, $\omega_\phi = 3.75$, $h_w = 5$

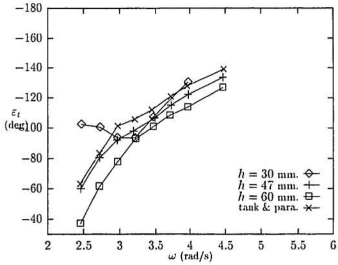


Figure 4.17: Phase lags of undamped tank, $\omega_\phi = 3.00$, $h_w = 10$

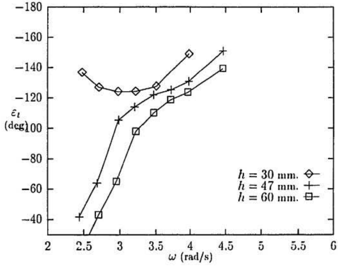


Figure 4.18: Phase lags of undamped tank, $\omega_\phi = 3.00$, $h_w = 5$

of the larger amount of water, it could be offset by the wrong tuning. As observed in the oscillating bench tests, the phase lag of the tank water motion to the roll motion changes more quickly with the roll frequency after the internal damping decreased (comparing Figures 4.15 to 4.18 and Figures 4.5 to 4.7). The fast changing of ε_t causes the tank to increase rather than reduce roll at nonresonant frequencies. However for the tested tank, the increases of roll angles at nonresonant frequencies are small (the maximum is about 2.5 degrees in Figure 4.11), therefore the internal damping of the tank should be close to correct. It may be improved by adding a small amount of damping to the tank to make the roll responses flatter.

The tests also show that the additional roll reductions from the paravanes become much smaller after the tank operates properly. There are only 6% increase of roll redution at $\omega_\phi = 3.75$ rad/s and 3% at $\omega_\phi = 3.00$ rad/s. If this is true for real fishing vessels, it could be a good reason to eliminate paravanes after a stabilizing tank is installed, or not use them unless at steaming speed.

Chapter 5

Experiments in Irregular Beam Waves

The experiments in regular waves are considered inadequate for evaluating a stabilizer because in practical cases ships encounter irregular waves. Since a train of irregular waves contains many waves whose natural frequency may differ from the tank frequency, one may suspect the efficiency of a passive tank, whose working principle is based on harmonic oscillations, in an actual sea state.

5.1 The Method of Analysis

Two methods can be used to investigate irregular waves. In the time domain, an actual sea state can be classified as a stationary stochastic process and all its statistical characteristics can be obtained from a time series. From observation of many wave records, the histograms for wave height (double amplitude) takes the shape of a Rayleigh distribution, which is expressed by the following equation [3]:

$$p(H_i) = \frac{2H_i}{\bar{H}^2} e^{-H_i^2/\bar{H}^2} \tag{5.1}$$

where $p(H_i)$ is the probability density that any particular wave height H_i will be observed, and $\bar{H}^2$ is the average of all the wave heights squared defined by:

$$\bar{H}^2 = \frac{\Sigma[(H_i)^2 \times f(H_i)]}{\Sigma[f(H_i)]}$$

where $f(H_i)$ is the number of occurrences of H_i. The significant wave height (average height of the one-third highest waves $(h_w)_{1/3}$ is given by:

$$(h_w)_{1/3} = 1.41(\bar{H}^2)^{1/2} \tag{5.2}$$

It should be noted that equation 5.2 is valid only if the wave height probability distribution function is described by a Rayleigh distribution. Otherwise $(h_w)_{1/3}$ has to be directly calculated from raw data.

The other method, which is more commonly used due to the application of computers, is to analyze the stochastic process in the frequency domain. An irregular wave pattern can be regarded as the combination of a large number of harmonics that have different frequencies and amplitudes with random phases. The total energy per square unit of sea surface is given by:

$$E_T = \frac{1}{2}\rho g(\zeta_{a1}^2 + \zeta_{a2}^2 + \ldots + \zeta_{an}^2) \tag{5.3}$$

where ζ_{a1}, ζ_{a2}, ..., ζ_{an} are the amplitudes of the n dominant wave components. The spectral density of wave energy is given as:

$$S_\zeta(\omega) = \frac{1}{2\delta\omega}\sum^{\delta\omega}\zeta_{ai}^2 \tag{5.4}$$

where $\delta\omega$ is the bandwidth. To find out ζ_a's from a digital signal sequence, a discrete Fourier analysis needs to be performed. There are many available computer packages to perform the task. The function *spectrum* in MATLAB© first divides a sequence of length n into sections of m points each (possibly overlapping), where m must be a power of two, then multiplies the successive sections by a Hanning window, transforms them with an m-point FFT, and finally takes the average of them to get the spectrum.

From a known wave spectrum one can obtain the mean value of the wave elevations squared m_0, which is the area under the $S_\zeta(\omega) - \omega$ curve:

$$m_0 = \int_0^\infty S_\zeta(\omega) d\omega \tag{5.5}$$

From the statistical view, m_0 is the *variance* of the distribution. If the waves follow the Rayleigh distribution, then the significant height can be obtained by

$$(h_w)_{1/3} = 4.0 \times \text{rms} = 4.0\sqrt{m_0} \tag{5.6}$$

For the roll response of a ship in an irregular seaway the same technique can be applieu. However, to predict the roll spectrum from a wave spectrum, it has to be based on two fundamental assumptions [3]:

1. The response of a vessel to any individual regular wave component is a linear function of the amplitude of this component.

2. The response of a vessel to any individual wave component is independent of its response to any other wave component.

With the above assumptions, the spectral density function of ship response for rolling in beam waves and the density function of wave spectrum have the following relationship:

$$S_\phi(\omega) = S_\zeta(\omega) \cdot |H(\omega)|^2 \tag{5.7}$$

where $S_\phi(\omega)$ is the roll spectrum and $|H(\omega)|^2$ is the response amplitude operator (RAO). Knowing wave and roll spectra from experiments, the RAO for roll can be obtained by simply taking the quotient of them. Also based on the above assumptions, if the wave heights follow the Rayleigh distribution, the roll amplitudes will also follow the Rayleigh distribution. Therefore the significant roll amplitude is twice the rms value of amplitudes of the roll motion.

Because in general ship responses in waves are nonlinear, the previous assumption is not valid when large waves are encountered. This may cause some inaccuracies in the estimation of significant roll amplitudes. To check the results from spectra, a FORTRAN program is written to calculate significant roll amplitudes directly from raw data, that is, to pick out the one-third highest roll amplitudes and take the average value of them.

5.2 The Waves

Four spectra of irregular waves comparable to the four tested regular waves were chosen. They have characteristic frequencies at 3.75 and 3.00 rad/s and significant heights of 10 and 5 cm. The wave generator is controlled by a computer program that generates random signals according to a theoretical wave spectrum pattern. Jonswap spectrum, which is a reasonable representation of a North Atlantic wave energy distribution [20], was chosen to be the spectral pattern. The Jonswap spectrum has the following form [20]:

$$S_\zeta(\omega) = \frac{A}{\omega^5} e^{-\frac{B}{\omega^4}} \gamma^a \tag{5.8}$$

where: $a = \exp(-\frac{(\omega - \omega_0)^2}{2\sigma^2\omega_0^2})$. $\sigma = 0.07$ for $\omega \leq \omega_0$

$\sigma = 0.09$ for $\omega > \omega_0$

$A = \frac{5H_s^2\omega_0^4}{16\gamma^{1/3}}$ for $1 < \gamma < 4$

$B = 5\omega^4/4$

and ω_0 is the peak frequency and H_s is the significant wave height. The parameter γ determines the breadth of the spectrum. The larger it is, the more wave energy concentrates on the characteristic frequency. It was chosen to be 3.3 in the experiments. Considering the wave periods, the total sampling time is chosen to be 400 seconds so that enough wave cycles can be acquired. The sampling rate is

20 Hz, therefore the Nyquist frequency is 10 Hz which is high enough to cover all significant wave frequencies. The obtained wave spectra as well as the theoretical Jonswap spectra are plotted in Figures 5.1 to 5.4.

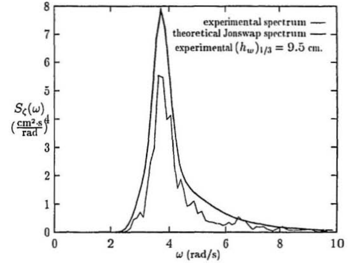


Figure 5.1: Wave spectra for $\omega_0 = 3.75$ and $(h_w)_{1/3} = 10$

The controlling program needs a driver that is generated from previous tests. The more tests done, the better the driver, and the closer the actual spectrum to the theoretical one. Due to time limits, only 5-9 tests were run for each spectrum to prepare the driver. This is why the experimental spectra do not fit very well to the theoretical spectra. The peak frequencies occur at the right places but the significant heights are about 3% to 7% lower than those intended (Figures 5.1 to 5.4). These inaccuracies were tolerated because the spectral pattern was not the main concern in this study.

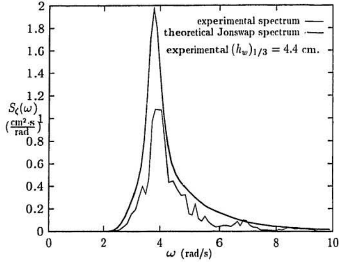


Figure 5.2: Wave spectra for $\omega_0 = 3.75$ and $(h_w)_{1/3} = 5$

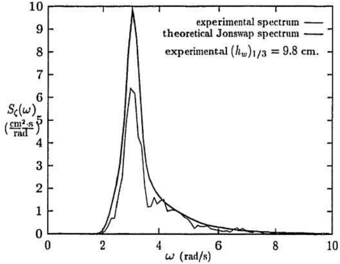


Figure 5.3: Wave spectra for $\omega_0 = 3.00$ and $(h_w)_{1/3} = 10$

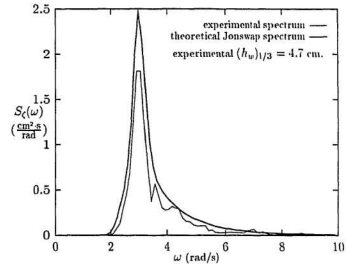


Figure 5.4: Wave spectra for $\omega_0 = 3.00$ and $(h_w)_{1/3} = 5$

5.3 Roll Responses for Damped Tank

For each test, the boat was ballasted with the natural roll frequency at the center frequency of the waves. The tank conditions are the same as in regular wave tests. A pair of roll samples for the unstabilized and stabilized boat are shown in Figure 5.5. Obtained roll spectra and *RAO*s are shown in Figures 5.6 to 5.13.

The energy distributions of the roll motions are rather narrow, mostly concentrated in a range of 1 rad/s around natural frequencies. Generally speaking, as in regular wave tests, the higher tank water level dissipates more energy of the roll motion. The paravanes are about as effective as the low tank level at the high frequency, but inferior to any tank level at the low frequency. The effectiveness of a tank is more frequency dependent than that of paravanes.

The *RAO*s for large roll amplitudes obtained from spectrum analysis are much larger than those derived from regular wave tests, but they do not differ so much

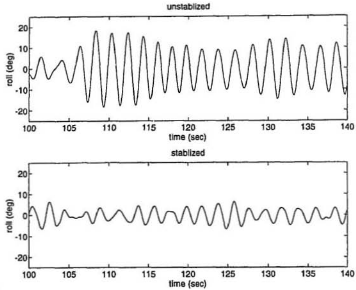


Figure 5.5: Roll samples for irregular wave tests

Table 5.1: $(\phi_A)_{1/3}$ (deg, from spectra) and reductions for damped tank

Stabilizer conditions	$\omega_\phi = 3.75$ $(h_w)_{1/3} = 9.5$		$\omega_\phi = 3.75$ $(h_w)_{1/3} = 4.4$		$\omega_\phi = 3.00$ $(h_w)_{1/3} = 9.8$		$\omega_\phi = 3.00$ $(h_w)_{1/3} = 4.7$	
Unstabilized	18.8		11.5		14.9		10.0	
$h = 30$ mm.	14.6	22%	7.5	35%	9.8	34%	4.0	60%
$h = 47$ mm.	13.1	30%	4.8	58%	8.3	44%	3.8	62%
$h = 60$ mm.	11.1	41%	5.5	52%	7.9	47%	3.1	69%
Paravanes	13.9	26%	7.4	36%	12.0	19%	7.5	25%
Tank & Para	9.2	51%	-		6.0	60%	-	

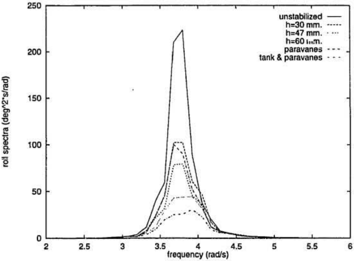


Figure 5.6: Roll spectra for damped tank, $\omega_\phi = 3.75$, $(h_w)_{1/3} = 9.5$

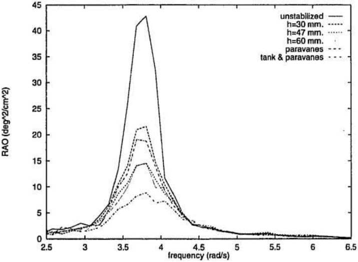


Figure 5.7: RAOs for damped tank, $\omega_\phi = 3.75$, $(h_w)_{1/3} = 9.5$

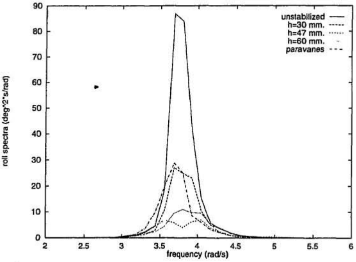


Figure 5.8: Roll spectra for damped tank, $\omega_\phi = 3.75$, $(h_w)_{1/3} = 4.4$

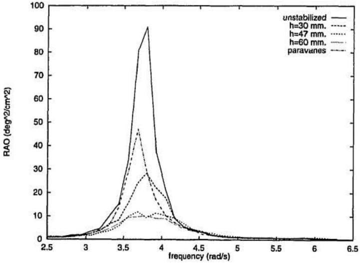


Figure 5.9: RAOs for damped tank, $\omega_\phi = 3.75$, $(h_w)_{1/3} = 4.4$

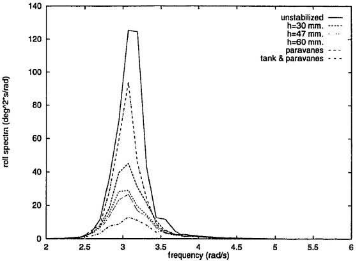


Figure 5.10: Roll spectra for damped tank, $\omega_\phi = 3.00$, $(h_w)_{1/3} = 9.8$

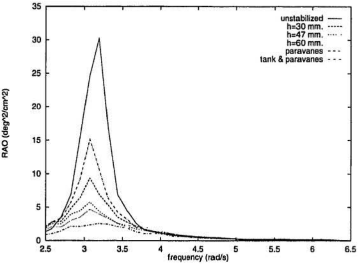


Figure 5.11: RAOs for damped tank, $\omega_\phi = 3.00$, $(h_w)_{1/3} = 9.8$

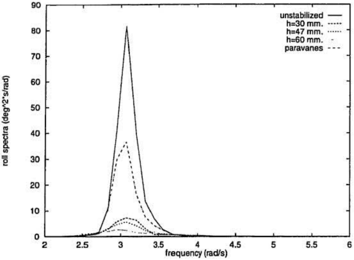


Figure 5.12: Roll spectra for damped tank, $\omega_\phi = 3.00$, $(h_w)_{1/3} = 4.7$

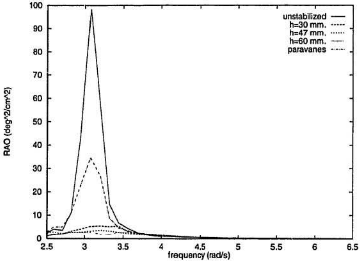


Figure 5.13: RAOs for damped tank, $\omega_\phi = 3.00$, $(h_w)_{1/3} = 4.7$

at small roll amplitudes. For example, in regular waves of 10 cm height when ω_ϕ = 3.75 rad/s (Figure 4.1), the peak *RAO* for the unstabilized roll motion is approximate 18.5 deg^2/cm^2 and for the most stabilized roll motion, 6.8 deg^2/cm^2, while in the corresponding irregular wave tests (Figure 5.7) the values are 43 deg^2/cm^2 and 8 deg^2/cm^2 respectively. The reason for this might be that the roll motions do not satisfy the linear assumptions in page 64 and therefore *RAOs* derived from equation 5.7 are inaccurate.

Table 5.1 shows the reductions in significant roll amplitudes. Compared to the tests in regular waves (Figure 4.1), the tank seems to work a little better at the high natural frequency, while at the low natural frequency the reductions have no significant difference. As indicated in the previous chapter, the tank is wrongly tuned at the high frequency. This defect might be less important in irregular waves than in regular waves because irregular waves contain numerous waves with different frequencies.

Some results given by spectral analysis are doubtful, such as that at ω_ϕ = 3.75 rad/s and $(h_w)_{1/3}$ = 4.4 cm, the reduction (Table 5.1) given by the medium tank level is higher than that given by the high tank level (Note that this does not happen in Table 5.2 where $(h_w)_{1/3}$'s are derived from raw data), and the *RAOs* are unreasonably high. As mentioned above, the spectral analysis is based on linear assumptions and the roll responses must follow a Rayleigh distribution. If these conditions are not satisfied, equations 5.6 and 5.7 are not valid for the calculation of $(h_w)_{1/3}$ and of *RAO*. Significant roll amplitude, as it is defined, can be obtain directly from the original samples by taking the average of one-third highest roll angles (Table 5.2). This gives less information but probably is more reliable.

Generally speaking, there is no significant difference between the values in Table 5.2 and in Table 5.1, except for the value for medium tank level at high frequency and in light waves. Comparing the significant roll amplitudes obtained from raw data, spectral estimates seem to be a little high at large amplitudes (or in moderate

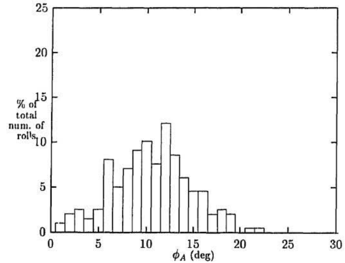


Figure 5.14: Histogram of unstabilized roll

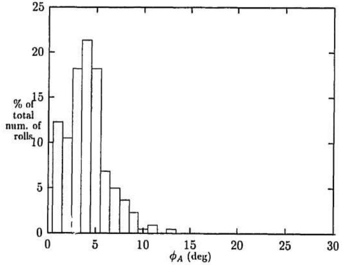


Figure 5.15: Histogram of stabilized roll

Table 5.2: $(\phi_A)_{1/3}$ (deg, from raw data) and reductions for damped tank

Stabilizer conditions	$\omega_\phi = 3.75$ $(h_w)_{1/3} = 9.5$		$\omega_\phi = 3.75$ $(h_w)_{1/3} = 4.4$		$\omega_\phi = 3.00$ $(h_w)_{1/3} = 9.8$		$\omega_\phi = 3.00$ $(h_w)_{1/3} = 4.7$	
Unstabilized	17.1		11.8		15.0		9.9	
$h = 30$ mm.	13.7	20%	7.8	34%	9.6	36%	4.8	52%
$h = 47$ mm.	12.4	27%	6.4	46%	8.1	46%	3.7	63%
$h = 60$ mm.	11.7	32%	5.9	50%	7.7	49%	3.5	65%
Paravanes	12.7	26%	8.2	31%	11.4	24%	7.1	28%
Tank & Para	9.5	44%	-		6.5	57%	-	

waves) and low values at small amplitudes (or in light waves).

Chapter 6

Conclusions and Recommendations

So far in Newfoundland, paravanes are the most commonly used roll motion stabilizers for small fishing vessels. The experiments carried out in this study have shown that a passive tank system could be a more efficient alternative. Without the shortcomings of paravanes and within the financial limitations for small fishing vessels, they can provide approximately 30% more roll reduction than paravanes can.

It is concluded from the tank decay tests that the natural frequency of the tank increases with the rise of tank liquid level. For a flume tank, the internal damping also has a significant effect on the frequency, that means, the damping not only restrains the amount of but also slows down the flow.

The oscillating bench tests were proven to be a useful method to configure the tank conditions so that the expected ship roll frequencies is included in the frequency realm of the tank. The tests are also useful to check the tank damping levels. The effect of tank damping is to change the sensitivity of the phase lag ε_t to the rolling frequency. Because the bench used here cannot provide data of tank moments, the tests may not be a good indicator of the performance of the tank on a vessel. A more sophisticated rolling table built with dynamometer (the best can also simulate sway and heave motions) is needed for this purpose.

For the unrestrained decay tests of M366 with the tank, the variation of equivalent linear damping ratio ζ_E with respect to mean roll amplitude ϕ_m cannot be well represented by a linear regression. However the features of the regression lines indicate that the damping provided by the tank is mainly linear. The decay tests for the paravanes show that the damping forces come from lift forces rather than drag forces at zero speed. The decay tests can provide some comparative information indicating the stabilizers' performance on the vessel. The equivalent linear damping ratio may be applied in a simple mathematical model to predict roll responses in waves, but the accuracy could be fairly limited because the tank-ship system is not a single but a coupled resonant system.

The results of the tests in beam regular waves show that by taking the tank size as 2% of the displacement, the roll reduction can be expected to be as high as 60% to 70%, when the tank is properly tuned and damped. Besides affecting phase lags, tank damping also has significant influence on the tank moments and natural frequencies for a flume tank, and therefore is important for tank efficiency. For both tank and paravanes, the reductions are approximately 10% to 20% higher in light waves than in moderate waves.

In irregular waves, the efficiency of the tank remains approximately the same as in regular waves, and even a little better if the tank is wrongly tuned. The effectiveness of the paravanes has no significant difference from that in regular waves.

In general, the results of this study may not be directly used to develop a full scale tank. For the many possible configurations of a tank, those tested here are rather limited. The degree of internal damping may be extrapolated to full scale if viscous effects are ignored. To design a tank for a ready-made vessel, the procedure probably should begin with the investigation of the ship. The main concerns are such as how much GM loss is permitted, and where and in what shape the tank could possibly be installed and constructed.

The effects of vertical positions on tank efficiency was not investigated in this study. It can be said however, in general the higher the tank is located within the vessel, the more efficient the operation of the stabilizer. This is because the dynamic moment, if located above the center of rotation, can be used to supplement the static couple [1]. On the other hand, too high a location causes too much loss of transverse stability. In practice the tank should be located as high as structural strength and stability permit.

The longitudinal location of the tank could be versatile, provided that the beam in the intended area of installation is nearly equal to the full beam of the vessel. Trim must also be taken into account when determining the location.

After determination of the tank size (usually 1%-2% of displacement), the other two main considerations are tuning and damping. Usually either rolling bench tests or wave-tank tests or both are required to ensure that the ship rolling frequencies are within the stabilizing frequencies (tuning) and the stabilized roll response of the vessel is relatively flat over the entire frequency range (damping). Various internal tank configurations may be tried during the tests.

References

[1] Martin, J.P., "Roll Stabilization of Fishing Vessels", Presented before the Society of Naval Architects and Marine Engineers, San Diego Section, on 19 January 1972.

[2] Sellars, F.H. and Martin, J.P., "Selection and Evaluation of Ship Roll Stabilization Systems", *Marine Technology*, Vol.29, No.2, April 1992.

[3] Bhattacharyya, R., "Dynamics of Marine Vehicles", John Wiley & Sons, New York, 1978.

[4] Vasta, J., Giddings, A.J., Taplin, A., and Stilwell, J.J., "Roll Stabilization by Means of Passive Tanks", Presented at the Annual Meeting, New York, N.Y., November 16-17,1961, of The Society of Naval Architects and Marine Engineers.

[5] Haddara, M.R., and Bennett, P., "A Study of the Angle Dependence of Roll Damping Moment", *Ocean Engineering*, Vol. 16, No. 4, pp. 411-427, 1989.

[6] Goudey, C.A. and Venugopal, M., "Roll Damping on Two New England Trawlers: An Experimental Study", *Marine Technology*, Vol.26, No.2, April 1989.

[7] Koelbel, J.G., Jr., Fuller, N.R., Jr., and Hankley, D.W., "Paravane Roll Stabilization", Presented at the Spring Meeting/STAR Symposium, Houston, Texas, April 25-28, 1979.

[8] Bass, D.W., Weng, C., "The Capsizing of the 'Straits Pride II'; A Study of the Dynamics of Paravanes", STAB Conference, Florida, USA, Nov. 1994.

[9] " Capsize and Sinking of the F.V. 'Straits Pride II' ", *Marine Occurrence Report*, No.M90N5017, The Transportation Safety Board of Canada, 1992.

[10] Martin, J.P., "Roll Stabilization of Small Ships", *Marine Technology*, Vol. 31, No. 4, October 1994, pp. 286-295.

[11] Cox, G.G. and Lloyd, A.R., "Hydrodynamic Design Basis for Navy Ship Roll Motion Stabilization", *Trans. SNAME*, Vol 85, 1977.

[12] Barr, R.A. and Ankudinov, V., "Ship Rolling, Its Prediction and Reduction Using Roll Stabilization", *Marine Technology*, Vol.14, No.1, Jan. 1977.

[13] Lewison, G.R., and Williams, I.M., "An Assessment of NPL Roll Stabilisers in Service", *Trans., RINA*, Vol.114, March 1972.

[14] Plank, W.S., Beardsley, G.F., Jr., and Burt, W.V., "An Experimental Evaluation of A Passive Anti-Roll Tank System", *Ocean Engineering*, Vol.2. Pergamon Press 1972. pp. 131-139.

[15] Bosch, J.J. van dan, and Vugts, Ir.J.H., "Roll Damping by Free Surface Tanks", Netherlands Ship Research Centre, Shipbuilding Department, Report No. 83 S, 1966.

[16] Goodrich, G.J., "Development and Design of Passive Roll Stabilisers", Read in London at a meeting of RINA on March 27, 1968.

[17] Chadwick, J.H., and Klotter,K., "On the Dynamics of Anti-Rolling Tanks", *Schiffstechnik*, 8.Heft, February 1954.

[18] Bosch, J.J. van den, and Zwaan, A.P. de, "Roll Damping by Free Surface Tanks with Partially Raised Bottom", *Technische Hogeschool Delft*, Rapport No. 280, Oktober 1970.

[19] Rakitin, V., Nachev, R., and Tzvetanov, Tz., "Series Stand Tests with Passive Stabilizing Tanks", STAB Conference, Naples, Italy, Sep. 1990. pp. 275-282.

[20] Lever, J.H., and Sen, D., "A Method to Upgrade Iceberg Velocity Statistics to Include Wave-Induced Motion", In the Proceedings of the Fifth International Offshore Mechanics and Arctic Engineering (OMAE) Symposium, April 13-18, 1986, Tokyo, Japan, Volume IV.

[21] Bass, D.W., and Haddara, M.R., "Roll and Sway-Roll Damping for Three Small Fishing Vessels", *Int. Shipbuild. Progress*, 38, No. 413, 1991.

[22] Zou, B., "An Experimental Study of Roll and Roll-Sway Damping for Small Fishing Vessels", M.Eng. Thesis, 1991, Memorial University of Newfoundland.

[23] Blagoveshchensky, S.N., "Theory of Ship Motions", Dover Publications, Inc., New York, 1962.

[24] Price, W.G., Bishop, R.E.D., "Probabilistic Theory of Ship Dynamics", Chapman and Hall, London, 1974.

[25] Korvin-Kroukovsky, B.V., "Theory of Seakeeping", The Society of Naval Architects and Marine Engineers, 74 Trinity Place, New York 6, N.Y., 1961.

[26] "Principles of Naval Architecture", Second Revision, Volume III, SNAME, New York, 1989.