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A SURVEY OF SHIP MOTION REDUCTION DEVICES

by

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INTRODUCTION

The case for reducing ship motions is no longer only one of crew comfort, but one of economics and readiness as well[1], and needs to be examined. Of the six degrees of freedom: surge, sway, heave, roll, pitch, and yaw, only heave, roll, and pitch exhibit resonant behavior. The horizontal motions: surge, sway, and yaw, have no restoring force and therefore, do not display resonant behavior. Furthermore, these motions typically fall in the realm of maneuvering where the design of directionally stable ships is generally understood. So the most likely candidates for motion reduction are the other three: heave, pitch, and roll.

Heave and pitch are difficult to control for conventional ships because they are excited over a wide range of frequencies and involve large forces and moments. With large excitation forces, it is unreasonable to expect small systems to produce appreciable results. One philosophy is to design the hull to minimize heave and pitch, while providing fins and bilge keels to reduce roll[2]. Thus, changes to the hull form or large and powerful systems are required for pitch and heave control. However, when heave and pitch are lightly damped as for SWATHs they become analogous to roll.

Roll is the most easily controlled for two reasons: it is lightly damped and the forces (moments) involved are relatively small. Being lightly damped means the response is narrow banded so the stabilizer can be tuned to one frequency rather than have to deal with a wide range of frequencies. The advantage of having small moments is that the addition of another small moment can produce significant reductions, because the change is large proportional to the existing moments.

According to Wienblum and St. Denis[3], there are three ways to reduce ship motions:

1. *tuning stabilization* – Changing the natural period of the system, separating the transfer function and excitation peaks, so that resonance is less likely to occur.

2. *damping stabilization* – Increase the damping of the system to reduce the resonant peak.

3. *equilibrium stabilization* – Applying a counteracting force to maintain the ship at equilibrium, an equal and opposite force approach.

With proper hull form design and weight placement, ship motions can be reduced through *tuning stabilization*. The natural frequency of the ship is moved away from the excitation peak. The seakeeping optimization program, SKOPT[4], does this with long ships, large waterplanes, and moving most of the buoyancy forward. This makes the ships stiffer in heave and pitch, reducing motions because now the natural frequency is greater than the peak excitation frequency. SWATHs reduce their motions by making the ship softer, i.e., decreasing their natural frequency well below the peak excitation frequency. Bulbous bows have mixed effects on pitch reduction adding strong viscous nonlinearities to the situation[5]. On one hand they provide damping; on the other, they increase length and pitch moment of inertia.

Damping stabilization increases the damping of the system, usually by the addition of appendages, such as bilge keels. Due to their movement through the water, these appendages create viscous drag which acts as a damper to the system. This kind of stabilization has the greatest reduction at resonance, with lesser reductions at other frequencies. Also, their effectiveness increases with increasing relative velocity, i.e. increasing motions.

Equilibrium stabilization is performed by anti-motion systems and relies upon proper phasing of forces and moments to reduce motion. Chadwick[6] developed "elementary stabilization means" as a method of cataloging motion control devices by whether it displaces or accelerates mass inside or outside the ship and whether the mass is solid or fluid. This method determines that only eight possible stabilization means exist and three of them are impossible. By combining the five remaining stabilization means, five fluid mass stabilizer types and three solid mass stabilizer types emerge, see Fig. 1–2. Therefore, while anti-motion devices run the gamut from passive free flooding tanks to fins to gyroscopes, devices with the same stabilization means will behave similarly. The stabilizers can be either active or passive. The active systems measure displacements, velocities, and accelerations and make predictions of future motions and then attempt to counteract the predicted motions. Passive systems are tuned to the correct frequency by restricting the flow in the system, changing the geometry, or mass of the system. Most stabilizer derivations are based upon the mathematically equivalent[7] double pendulum vibration absorber system, even though a more complicated damping may be involved.

BILGE KEELS

Bilge keels are an effective passive roll device, increasing damping at all speeds and sea states. They are simple, inexpensive and should require no more maintenance than the hull. Bilge keels are relatively commonplace and have been the subject of much study. For detailed explanations and derivation of specifics with regard to bilge keels see Himeno[8], Kato[9], Tanaka[10], Martin[11], and Mandel[12].

Bilge keels are between 25% to 50% of the length of the ship and centered near midships. Care should be taken not to extend the bilge keels too far forward or aft, so as to interfere with flow around the hull and cause unwanted coupling with heave and pitch. They are usually attached perpendicularly to the hull and are wide enough to extend beyond the ship's boundary layer, between 0.3-1.2 meters wide. The location girthwise is determined by geometric constraints of trying to maximize the distance between the bilge keel and the roll center, and still not have the bilge keel extend below the baseline or beyond the hull with a 5° heel, see Fig. 3. When aligned along calm water flow lines, the drag penalty is minimal[12, 13]. Multiple sets or discontinuous bilge keels are generally less effective than a continuous one, except at certain speeds where downwash effects improve performance[14].

Bilge keels increase viscous hull damping as first noted by Bryan[15] by changing the roll damping moment by resistance on the bilge keel itself and changing the pressure distribution on the hull. They increase energy dissipation due to viscous flow losses by 50 to 100 percent[16] over bare hull damping. This is a significant improvement over bare hull, but is still less than is possible with other devices. They are most advantageous at low speeds where hull damping is minimal. As speed increases, hull damping increases and viscous effects due to bilge keels become proportionally less of the total damping moment. Their effectiveness also increases with increasing motion, so they are beneficial in large waves.

Various and detailed methods are available for calculating the added damping due to bilge keels. These tend to be semi-empirical due to the complicated form of damping. Kato's[9] method, used in the Standard Ship Motion Program, (SMP84)[17, 18], is a semi-empirical method based on model tests of conventional merchant hulls. An example of a semi-empirical method for the ratio of roll damping with bilge keels to critical damping from Miller et al.[19] is:

$$\left(\frac{C}{C_c}\right)_o = \frac{K[A_{bk}w^{1/2} + 0.0024LBd^{1/2}]d^{5/2}\phi^{1/2}}{\Delta B^2} \tag{1}$$

$(\frac{C}{C_c})_o$ Damping ratio at zero speed.

C_c Critical damping $= 2\sqrt{(I_{xx} + A_{44})(\Delta GM_T)}$.

I_{xx} Roll mass moment of inertia.

A_{44} Roll added mass.

GM_T Metacentric height.

K Empirical constant equal to 0.0153 for metric units and 0.55 for English units.

A_{bk} Total bilge keel area.

d Distance from centerline at load waterline to turn of bilge.

L Waterline length.

w Bilge keel width.

B Waterline beam.

ϕ Single amplitude roll angle, radians.

Δ Displacement.

This equation is valid for fine hulls with relatively round bilges and wide bilge keels (bilge keel length/bilge keel width = 40-50). While the full intricacies of bilge keels are not known, the current methods work well if the assumptions made during the derivation are not violated, e.g. fine hulls and round bilges. This is especially important to note when dealing with oddly shaped hulls that may render the formulae invalid.

The advantages of bilge keels are simplicity and effectiveness. They perform well at all speeds and have increasing damping with increasing roll angle. The disadvantages of increased resistance can be partially overcome by proper alignment with the flow. The important things to remember about bilge keels are:

1. Roll damping increases with an increase of the bilge keel's surface area and/or the distance from rolling axis.

2. Bilge keels contribution to roll damping increases in direct proportion to the roll angle squared.

3. Bilge keels are effective at all ship speeds, particularly zero speed.

4. Bilge keel effects vary inversely with moment of inertia. So one wants a small ship mass moment of inertia about axis, i.e., mass close to centerline.

5. Bilge keels are effective in the region of resonance reducing the resonant peak. A ship's roll period is virtually unaffected.

REMOVABLE STABILIZERS

These stabilizers are not permanent fixtures on the hull, but are devices that can be rigged or deployed as the situation dictates. They tend to be zero to low speed, passive stabilizers for small ships, simply because of the large forces associated with higher speeds and large displacements.

Staysails are effective roll stabilizers for small vessels. The lift on the sail changes during a roll with a phase that removes roll energy from the system.

Weighted drogues or flopper-stoppers hanging from booms rigged over the side also absorb roll energy. One side rises, providing damping and motion reduction, while the other sinks during a roll[16]. McCreight and Jones[20] have developed equations of motion for this device using equivalent linearization to predict the roll damping of the flopper-stoppers. They examined shallow buckets with one way vents in the bottom, although cloth drogues could be used instead. See Fig. 4. Theoretically, the weighted drogues could be put over the side near the bow and stern of the ship to also reduce pitch.

A paravane is another stabilizer that is popular with small fishing craft. These delta shaped foils are towed through the water from booms and are designed in such a way that they pull downward when the boom is rolled upward, see Fig. 5. Paravane roll stabilization at zero speed acts in the same manner as the weighted drogues. This device has been shown to be an effective roll stabilizer. Koelbel et al.[21] present a prediction and design methodology for paravane stabilizers.

Theoretically, the same mechanisms would be effective aboard larger ships. The problem is one of size, as the devices needed to stabilize a large ship would themselves be very large. The rigging of large staysails is reminiscent of the days of tall ships. The sheer number and size of the drogues or paravanes required to increase damping significantly is prohibitive. As a result, these devices remain the prerogative of smaller craft.

TANK STABILIZERS

Stabilizers of this group involve a sloshing liquid to produce damping and restoring forces. The liquid could be fresh water, sea water, fuel, drilling mud, or whatever is convenient, as long as it not too viscous at ambient temperatures. However, corrosion and flammability are concerns if the liquid in tanks is not specially treated fresh water.

The power of a tank stabilizer is generally measured by the static heel angle it is able to induce. This can be found by calculating the heeling moment due to the tank, which is proportional to the weight of the fluid, and consulting a GZ curve. For small angles the following formula is adequate.

$$\phi_s = \frac{53.7Wx}{\Delta GM_T} \tag{2}$$

ϕ_s Static heel angle in radians.

W Weight of fluid transferred in tank.

x Displacement of CG of fluid transferred.

Rorke and Volpich[22] recommend typical values for tank stabilizers of 2-4°, while Bhattacharyya[23] recommends around 4°. General rules of thumb for tank stabilizers are as follows:

1. The price of an effective anti-roll tank is a sacrifice of 20–30% of GM. So a ship must have a large GM to even consider using a tank stabilizer.
2. Natural frequency of tank should be equal or near to ship's natural frequency for passive tanks, higher for active ones.
3. Changes in damping and restoring forces due to tanks are frequency dependent.
4. The amplitude of the tank moment is largest when the tank is well above the roll axis.

FREE SURFACE TANK

First investigated by Froude around 1874, a free surface tank typically extends across the entire ship and the connecting channel between the two tanks has a free surface, see Fig. 6. Its shape, along with internal baffles and obstacles, allow the liquid in the tank to flow from side-to-side with the proper phase to reduce roll motions and generate damping. The free surface effect of the tanks lengthens the roll period of the ship, reducing the chance of encountering a longer wave period and having synchronous rolling. Being a passive device, they are simple to construct, operate, and maintain.

The natural frequency of the tank should be equal or near to the ship's natural frequency. The tank is tuned by adjusting the amount of liquid in the tank. Free surface tanks have the advantage that they can be easily tuned for a range of loading conditions. They are easy to design because they enjoy more internal damping than other tanks which eliminates the need for a precise design to attain the desired amount of damping. The tuning is usually done during ship trials by experimentation using model test results as a starting point[13, 24]. As the restoring moment generated by the tank is proportional to the amount of liquid in the tank, a loss in system performance can occur if this parameter is excessively reduced for tuning purposes. Over filling a tank can cause a loss of performance because the tank becomes saturated at a smaller roll angle. This corresponds directly to operational sea state, with early saturation resulting in poor heavy seas performance. The higher damping of free surface passive tanks "flattens" the response amplitude operators (RAO) curves, making the system kinder to changes in tuning ratios (loading changes)[25]. So some flexibility with loading is permissible without tank loss of efficiency. If the change in loading is large, multi-tank systems are worthwhile. Individual tanks are tuned to different loading conditions such that every condition will have some stabilization dedicated to it.

The damping and restoring forces are largely dependent on frequency and amplitude of motion. The motion reduction has been found to be a constant percentage until saturation whereupon the reduction is a constant number of degrees. Their main disadvantage is the loss of stability due to free surface effects, which caused a general lack of interest until the 1950's. Emptying a tank will eliminate the free surface effect and avoid aggravated rolling in long waves. The ability to empty the tanks to improve static stability in case of emergency is an important feature.

Free surface tanks are a good choice for stabilizers if space and weight are not concerns, especially as the tank connection limits fore-aft passageways. If the GM/BEAM ratio is greater than 0.1, the tank should be placed above the roll center for maximum performance[19]. The important aspects of free surface tanks are given in Miller et al.[19] and Barr and Akudinov[26]. Lewison and Williams[24] show the effectiveness of these tanks through case studies of commercial installations.

Free surface tanks are difficult to model because the wavemaking and spray in the tanks is very important to their simulation and are not fully understood. Rigorous derivations of the stabilized equations of motion tend to be non-linear with coefficients that are hard to define. Lewison[27] presents a useful system of non-linear equations and suggests that most of the coefficients be determined by model test. The numerical example given is very helpful and gives means of approximating the coefficients without a model test. The damping coefficient and free surface correction factor used in Goodrich's derivation[28] and mentioned in Vugts' discussion[28] of it, are highly frequency dependent and also need to be determined by model tests. Free surface tanks can also be modeled as special U-tube tanks, as given in Vasta et al.[25].

Advantages of free surface tank:

1. Simple to construct and operate.
2. High internal wavemaking damping eliminates the need for precise design.
3. Possible to tune tank to work for different loading conditions due to flat RAO.
4. Increases a ship's natural period, making synchronous rolling harder.

Disadvantages of free surface tank:

1. Reduces stability with free surface effect.
2. Ineffective in longer waves, where $\omega_e < \omega_\phi$.
3. Ineffective when it saturates in high sea states.
4. Free surface connection limits fore-aft passageways at the location of the tank.
5. Noise produced due to sloshing fluid.

A more efficient variant of this tank type, is the **diversified tank** stabilizer[23]. The diversified tank overcomes the disadvantages of the free surface tank by connecting two or three transverse tanks with flumes, ducts, or nozzles, see Fig. 7. Using 2 or 3 transverse tanks reduces the free surface effect by temporarily restricting flow to the low side and generates damping from flow through the nozzles. Restricting the flow of liquid avoids synchronizing with roll, preserves a 90° phase lag, and at the same time counteracts the tendency to roll past the wave slope. These tanks are tuned by changing the amount of water in the system and using adjustable nozzles. They are very effective, producing about a 75% roll reduction.

The passive Muirhead-Brown tank system is a *passive controlled* diversified tank. It uses valves, controlled by accelerometers, to restrict the flow between tanks. The center tank for this system does not have a free surface. A tank is *passive controlled* when the liquid is moved only in response to roll, i.e. there is no pumping of fluid from one tank to another, and yet that movement is controlled by using valves that respond to ship motion.

U-TUBE TANK

The U-tube tank stabilizer, also known as a Frahm tank, has wing tanks connected by a horizontal passage to complete the "U", see Fig. 8. Oscillating columns of water provide damping and restoring moments, instead of water sloshing from side to side in the free surface tank. The stabilizing moment of the tank is 90° out of phase with the excitation; a phase relationship that reduces motion. By placing the tank above the roll center, the acceleration of the fluid in the crossover duct is also used to reduce roll.

The natural frequency of the U-tube tank should be made equal to that of the ship. The frequency of the tank is given by

$$\omega_t = \sqrt{2g/S}, \tag{3}$$

S The effective length of U-tube, $= \int_0^l \frac{A_o}{A(s)} ds$

A_o Constant cr sectional area of of one wing tank.

$A(s)$ Cross sectional area of U-tube normal to U-tube centerline.

s Girthlike coordinate along centerline of U-tube.

l Total girth length of tank fluid.

Changing the effective length, either by changing th depth of the water or cross section of the tube, changes the frequency of the tank. If the variation in loading is not large, the tank frequency can be controlled by connecting the wing tanks with an air duct instead of venting them. This configuration is also *passive controlled*. The air valve in this duct controls the air flow from tank to tank and hence the rate of water flow. Closing the valve makes the tanks inoperative and eliminates the free surface degradation[26]. Damping of a U-tube tank comes only from frictional losses and is much easier to calculate than the free surface tank. Stiger[29] derived the equations of motion for this system.

The main advantage of the passive U-tube tank is that it has less free surface degradation than free surface tanks. Because it is passive, it is simple to operate and economic to install and retrofit. Emptying the tanks for weight reduction is possible when not stabilizing. The system provides extra storage capacity when filled with fresh water, fuel, etc. There is no substantial difference in performance between the U-tube and free surface tank.

The disadvantages are that the tank frequency cannot easily be tuned to all loading conditions. And being tuned to one frequency, performance in irregular seas is limited as response at other frequencies are not damped as much. The tanks also have an adverse effect on static stability when operational.

Passive-controlled tanks are more effective because the differing air pressure controls high frequency motions. They do require more weight and volume due to a large crossover duct needed to make the tank frequency high[23, 30]. Also mechanical failure of the valves leads to inoperable tanks (valves closed) or inferior (valves open).

Advantages of passive U-tube tank:

1. Simple, its a passive device.
2. Economical to install and retrofit.
3. Fifty percent or more roll reduction.
4. Possibility of weight reduction by emptying tanks when not stabilizing.
5. Provision of extra storage capacity for fresh water, fuel, etc.

Disadvantages of passive U-tube tank:

1. Proper tuning of tanks for all loading variations hard to accomplish.
2. Can only reduce at one frequency, so inadequate for irregular seas.
3. Adverse affect on stability.

ACTIVE ANTI-ROLL TANKS

An active anti-roll tank is invariably a U-tube tank with a pump to move the fluid between tanks and a control system to run the pump. The logical continuation from a passive U-tube tank to an active one by placing a controllable pitch axial flow pump in the cross duct has limited effectiveness for a number of reasons. The first is that when first activated the pump must accelerate a large column of water into the desired tank. So there is a considerable lag time before the restoring moment is achieved after it is ordered. With errors in anticipating the motion due to the controller, this lag time becomes insurmountable. Also the instantaneous pumping power required when switching from side to side can be very high, even as high as 10% installed shaft power[30] of the ship. An installation on a World War II German cruiser used compressed air to move the water quickly from tank to tank[25].

The active Muirhead-Brown system discussed by Bell and Walker[31] is better than a passive U-tube tank and does not require the acceleration of a large column of water, see Fig. 9. With this system a pump runs continuously, pumping water into a central tank or sump. Valves in the water crossover duct direct the fluid from one tank to the other through the central tank. The central tank acts as a buffer to reduce the varying propeller loads and reduce the extreme power demands. This avoids the need for a powerful reversing pump. The hydrostatic head, which is related to tank depth, determines the power demands of the pump. So a large cross sectional area while beneficial from a pump stand point, increases the free surface effect. This trade off is left to the designer.

For active systems the choice of controller is very important. Even the best designed tank can be rendered useless by a poor controller, while a well designed controller can partially make up for tank inefficiency. That the controller is relatively inexpensive when compared with tank construction and yet is so important to performance, makes the choice of controller a high pay off item. Webster[32] deals extensively with the control system of active tanks.

The advantages of active anti-roll tanks over passive tanks are many. Because active tanks rely on a pump to move the fluid rather than sea forces, they are effective over a wide range of frequencies and are not tuned to a specific frequency, making them useful in irregular seas. It is possible to reduce the size of the tanks for space savings and reduction of free surface effect. The pump can be used to either cause the tanks to operate with a constant heel angle or correct a list if the ship is damaged. Along the same lines, it is possible to *induce* motion by moving the tank water in phase with roll. This would be important for ships wishing to free themselves from ice or mud.

The main fear of an active tank is that a mechanical failure would result in a "passive" tank which was worse than no tank at all. Active tanks avoid extraneous structure in the tanks to keep internal damping less than passive tanks and keep response fast for the pumps. This makes the period of an active or controlled-passive tank less than a purely passive tank so synchronous rolling becomes easier. Bell and Walker[31] and Morris and Chadwick[33] provide detailed examinations of active anti-roll tanks.

The physical difference between an active and passive tank is the excitation caused by the pump. With this in mind, the equations of motion for an active tank are the equations of motion for a passive tank with extra excitation terms to account for the pump. The methodology for adding the control law of the pump is similar to that of active fins. The pump power is written in terms of the motion and a set gains which comprise the control law. The difference between active tanks and fins is the phase lag of the pump response to the relatively quick response of the fins. The tank equations of motion are highly non-linear and linearization is necessary.

FREE FLOODING TANKS

Free flooding tanks were also investigated by Frahm[34] at the end of the 19th century. With this stabilizer, the wing tanks are open to the sea and are not connected, except maybe by an air duct, see Fig. 10. This has the advantage of freeing up internal, centrally located space and can take advantage of voids left by blisters or armor. Free flooding tanks derive their ability from oscillating columns of water just like the U-tube stabilizer.

With the tanks open to the sea, venting as well as saturating, becomes a problem with large motions. Both venting and saturation lead to performance losses. That the tanks are open to the sea also leads to problems of corrosion and momentum drag, which increases with the square of the speed. Free flooding tanks have found favor with oil rig platforms were the increased drag at high speeds is inconsequential. External tanks have been considered for pitch reduction, but to be worthwhile they should be at least 20% the ship length and they would incur an enormous drag penalty at higher speeds[23].

Obviously limiting the amount of water in the tanks as a tuning device is impractical with free flooding tanks. The tanks are tuned by shaping the inlets and controlling the air flow between tanks, if they are connected, in a manner similar to the passive controlled U-tube tanks. An application of this concept is patented under the name SLOROL. Dalzell* and Webster et al.[35] have investigated free flooding tanks in conjunction with the *USS Midway*. From this work Dalzell has written a variation of SMP84 that predicts the effects of SLOROL type tanks. The derived equations of motions started from those presented by Blagoveshchensky[36], and were reworked so they could cope with new concepts.

TOROIDAL ABSORBER

This stabilizer is similar to the rectangular free surface tank, but instead of being rectangular it is toroidal, see Fig. 11 This damping device is particularly suited to low frequency planar motion applications[37] found in offshore oil rigs. However, the tank can be tuned to a wide range of frequencies by changing the inner and outer radii, the cross section, and depth of the fluid in the absorber. The frequency of the tank should equal the transverse frequency of oscillation of the structure for maximum damping. Some damping will occur if the tank is tuned to a whole number integer of the transverse frequency. As with other tanks, it can be filled with fuel, fresh water, ballast water, or any other convenient liquid.

*Dalzell in David Taylor Research Center departmental report of limited distribution

The advantages are similar to other passive rectangular tanks. The geometry is relatively simple, so it is easy to build, install, and retrofit. Like other tanks, it does not need contact with the ocean to work. This avoids extra wave excitation. The toroidal geometry has two important advantages over a rectangular tank. The shape allows low frequency fluid oscillation which damps low frequency motion. Secondly, it is omni-directional and performs equally well regardless of direction of the excitation.

The fluid motion within the toroid is modeled with linear potential flow theory, solving Laplace's equation with appropriate boundary conditions. The fluid motion becomes highly non-linear near resonance and viscous effects may no longer be negligible. Despite these complications, it is possible to size the absorber using linear theory and find the correct fluid depth through trial.

The toroid absorber compares favorably with rectangular tanks and solid weights when seeking maximum damping action for minimum weight.

TRANSLATING SOLID WEIGHT

The translating solid weight reduces motions by the same mechanism as tank stabilizers, except instead of a liquid sloshing back and forth, a solid weight is moved. The weight is kept near the centerline by use of springs, curved tracks, and some damping mechanism. One installation moved the weight through a viscous fluid to create anti-roll damping[25]. Early attempts encountered mechanical difficulties in getting the weight to respond correctly to the controller[38]. Advances in controller technology have helped this concept and Kirkman[39] discusses its recent use for a small Coast Guard patrol boat. As with tanks, the phase lag of the weight to the rolling motion should be 90° to provide a counteracting moment.

The stabilizing power is proportional to the mass of the anti-roll weight and its distance from the roll center. The translating weight stabilizer is primarily useful for smaller vessels, "since the [stabilizing] weight becomes impractically large"[26] with larger displacement. Miller et al.[19] recommend a weight between 0.5 and 2% displacement, depending on GM. Bhattacharyya[23] suggests no more than 5%, and shows that increasing the weight continues to reduce the RAO peak, but does not reduce roll appreciably over a wide range of frequencies.

Moving weight stabilizers also have various levels of activation. A passive weight stabilizer controls the weight's motion between end stop buffers, which keep the weight from rolling off the end of the track. The passive controlled stabilizer adds extra stiffness and dissipative damping to the system. The active system actually generates stabilizing power through the acceleration of the weight. Nonweiler et al.[40] recommend an energy conservative drive, such as a oil hydrostatic type.

The main advantage of the solid weight stabilizer is it has no free surface effect to degrade stability. A well designed system should require less space than an equivalent tank because the weight can have a density greater than any liquid. It has the potential to be the smallest size passive stabilizer[19].

The safety[16] of having a large moving weight in the close quarters of a small ships is a real concern. In the past it has been rarely used because of complications with controlling machinery. Furthermore, Chadwick[6] states that it is almost always possible to find a tank system which is more effective, safer, and less expensive.

The passive translating solid weight can use U-tube stabilizer equations for prediction purposes. The equations are simplified because the damping and restoring forces are known exactly. Nonweiler et al.[40] developed equations for the active translating weight stabilizer system. Lesser degrees of activation are predicted by neglecting the appropriate terms.

ROTATING RING

This is an active stabilizer of a different nature. A continuous chain is supported on sprockets in the plane perpendicular to the fore-aft axis of the ship. The stabilizing torque is the result of accelerating the chain, preferably using an energy conservative drive. The optimum gearing for this device has not been well defined. No end stop buffers are needed for this variation.

This device has great potential for space savings. The chain is distributed along the perimeter of the ship leaving center space unobstructed. Even relatively massive links would not use much space. Proper design can avoid problems of penetrating watertight decks. The larger the "hydraulic radius", or contained area divided by the perimeter, the less power required for a given "wave slope capacity". Also if the ring mass is large, less power needs to be added to the energy conservative system. Equations for a hydraulic pump/motor are given in Nonweiler et al.[40].

GYROSCOPIC STABILIZER

A gyroscopic stabilizer develops a stabilizing moment due to high speed rotation of heavy weights. The restoring moment originates from the conservation of angular momentum and is proportional to the angular velocity about a perpendicular axis. While theoretically sound, early installations had problems with excessive weight and poor controls[41, 42]. It is possible to have a passive gyroscope in that its angular velocity is constant, i.e. independent of roll motion. But the difference in cost, space, and weight between active and passive gyroscopes is minimal, while the performance of an active gyroscope is superior[6]. Therefore, a passive gyroscope should not be chosen over an active one. The gyroscopic restoring moment is given by the following equation:

$$M_z = I_z \omega_r \Omega \tag{4}$$

I_z Mass moment of inertia of spinning element about axis.

ω_r Angular velocity of spinning element.

Ω Angular velocity of motion $= d\phi/dt$.

The stabilizing moment has the desired 90° phase lag from the exciting moment. The maximum stabilizing moment occurs in the nearly upright position. The minimum occurs in the inclined position. This implies that stabilization is very good for small to medium angles and decreases with increasing angle.

The stabilizer controls were improved by Sperry and 3 gyroscopes were installed on the *Conte di Savoia*[43] in 1932. The gyroscopes weighed 1.7 per cent of the displacement and over an 18 month period reduced the average roll angle by 50%. Other later installations, on small ships and submarines, have shown similar reductions, but the cost in weight, space, and power were too high[23, 25, 33]. Also gyroscopes tend to lose stabilization ability as the wave encounter frequency goes to zero.

FINS

Fins are the most powerful and effective motion control devices for high speed applications. Their effectiveness increases with the square of the speed until cavitation or separation occurs on the fin surfaces. At low speeds fins do not generate much lift, but act somewhat as passive viscous dampers. Fins are more useful than other devices for heave and pitch reduction because of the potential to generate large restoring forces and moments. However, fins have found the most use as roll stabilizers because of the very large excitation forces and moments associated with heave and pitch. The static heel angle possible from a set of fins is given as:[†]

$$\phi_s = \frac{\rho U^2 C_L A \hat{y}}{\Delta G M_T} \tag{5}$$

ϕ_s Static heel angle in radians.

ρ Density of fluid.

U Ship design speed.

C_L Lift coefficient.

A Fin planform area.

$\hat{y}$ Moment arm of fin.

The fins can be designed to be retracted into the hull, see Fig. 12, or folded up for non-stabilized operations and port/harbor maneuvering. A reason to not use retractable fins on naval ships is that they are more susceptible to shock damage. Also, most Navy ships cannot afford the space required for retractable fins and simply limit their outreach to minimize possible damage when berthing. Having a limited outreach, means the

[†]The static trim angle is similarly found by calculating the moment longitudinally instead of transversely.

adoption of a trapezoidal planform to avoid the adverse lift/area ratio of low aspect ratio foils[44], see Fig. 13. Merchant ships with retractable fins can use high aspect ratio foils. The fins can either have full flap or trailing edge flap configurations. Passive and active fins are possible, though quick response active fins have a significant performance advantage over passive fins. The main requirements for stabilizer fins are[23]:

1. Symmetry with respect to positive and negative angles of attack.
2. High maximum lift force.
3. Low drag.
4. Low torque.
5. Adequate structural strength.
6. Cavitation resistance.

The main drawback of fin stabilizers is their range of operation. At speeds less than 10 knots they do not produce much lift and at higher speeds cavitation and separation reduce lift. Cavitation resistant foil shapes can be used to improve high speed performance. Circulation control or jet flaps can increase fin lift and delay cavitation without changing its geometric angle of attack. The controller must be properly designed to avoid hard over to hard over operation at low speeds and in heavy seas. The fin should not be driven past its stall angle. Fins also increase resistance and as with all lifting devices, they create an induced drag which adds to the frictional drag.

The predicted effect of fins on ship motions is generally well known. The theory relating to vertical motion can be found in recent SWATH work. Equations from Lee[45] and McCreight[46] contain terms for both potential and viscous fin effects. The effects of active anti-roll fins as used by SMP84 are given by Meyers and Baitis[18]. The active fin terms in SMP84 are purely potential flow. The viscous terms are calculated separately and added to the potential flow terms. Despite the different ways of dealing with potential and viscous terms, fin forces and moments are considered to be well understood.

ANTI-PITCH FINS

If the fins are located near the ends of the ship, they can reduce heave and pitch. Heave is reduced through heave-pitch coupling and phasing of the lift force to counteract heave rather than pitch. Bessho and Kyozuka[47] discribe the methodolgy required to choose fin size and location to reduce both heave and pitch. However, pitch is usally the main concern and heave is rarely targeted for reduction.

The fin location to reduce pitch is a compromise between hydrodynamic efficiency and practicality. Bow fins have higher relative velocities than stern fins[48, 49], which leads to improved performance, but have many ancillary problems. Bow fins experience ventilation and cavitation which leads to excessive lateral vibrations when bubbles collapse on the fin and hull. Slamming re-entry also necessitates the design of structurally sound fins[50]. The activation of bow fins may not be necessary due to the motion at the bow, which, for most ships, already provides sufficient angle of attack. Also the operating machinery for active bow fins may require excessive amounts of weight and space.

Stern fins come under the guise of π- and canted rudders, in addition to fixed fins. Kaplan et al.[51] investigated both these configurations in conjunction with a passive bow fin. The use of high lift foils was considered as a means of improving the performance of stern fins. Stern fins are not as effective as bow fins, even when active, but have less vibration problems. A stern location also has benefits of more room for machinery without sonar dome interference. Ochi[52] reports an increase in resistance of stern fins of two to three times that of bow fins.

General trends for anti-pitch fins from Ochi are:

1. Maximum pitch reduction when fin is located in forward 10% of the ship.

2. More severe vibration is experienced in these forward locations.

3. Planform area and pitch reduction are linearly proportional (for Mariner hull).

4. Vibration increases with fin area and violence of pitching.

5. Fins with properly designed ventilation holes are as effective as solid fins, but have less vibration.

6. Bow fin effectiveness increases with wave height, while stern fins have constant results regardless of wave height.

7. Anti-pitch fin effectiveness from best to worst: fixed bow, fixed stern, active bow/fixed stern combination.

With proper control of the phase between the fins and excitation, it is possible to reduce vertical motion at a specific point on the ship[53]. As a continuation of this, it should be possible to completely eliminate these motions by proper placement of the fins to attain the correct phase. While theoretically possible, the fins must be located well off the ship, making it currently impractical.

ANTI-ROLL FINS

Anti-roll fins are considered the stabilizer of choice for medium to high speed missions, i.e. speeds in excess of 10 knots. They can be either active or passive. As the ship rolls, the fins experience equal but opposite angles of attack. This produces equal but opposite lift forces, resulting in a restoring moment. Activating the fins generates larger lift forces and a larger restoring moment. The maximum fin angle is a function of stall, cavitation, and machinery limits. With multiple fin sets, aft fin sets suffer a lift degradation due to flow disturbances created by the forward fins.

Anti-roll fins are usually located near midships to maximize righting arm and to avoid coupling with other motions. The fin size is determined by the same rules as bilge keels[13] to avoid creating appendages which extend below the baseline or beyond the beam when the ship is heeled 5°.

Anti-roll fins are popular motion control devices and many sources regarding them are available. Lloyd[54] and *Principles of Naval Architecture*[30] are two of many that discuss anti-roll fins in general. Miller et al.[19] identify the positive and negative impacts on the ship associated with fins. Conolly[55] shows the advantages of stabilization by active fins and compares active and passive performance. The advantages and disadvantages of anti-roll fins are those named for fins in general.

The comparison between fin stabilizers and anti-roll tanks can quickly be reduced to one of cost versus space and effectiveness. At high speeds fins are more effective than tanks, but require machinery and controllers which can make the cost as much as 5 times the expense of anti-roll tanks[23]. Anti-roll tank systems, including fluid weight, require two and a half to three times the weight of fins systems and take up valuable space locations high on the ship[25]. Therefore, fins are lighter and more effective than tanks, but are more expensive and require speeds greater than 15 knots.

HIGH LIFT DEVICES

Jet flaps and circulation control are high lift devices that increase the lift of a foil without changing its geometric angle of attack. Water is pumped from a slit parallel to the trailing edge of the fin to delay separation of the flow and the onset of cavitation. On jet flaps, the slit can be rotated to provide a directional jet, see Fig. 14. Goodman and Kaplan[56] have studied the ship application to reduce pitch and cavitation and found it possible to double or triple the no-jet lift without cavitating. The lift coefficient for a jet flapped foil according to Spence[57] is:

$$\bar{C_L} = [\frac{\partial C_L}{\partial \alpha}\alpha + \frac{\partial C_L}{\partial \tau}\tau](1+\delta) + C_j\tau \tag{6}$$

$$\frac{\partial C_L}{\partial \alpha} = 2\pi + 1.152C_j^{1/2} + 1.106C_j + 0.51C_j^{3/2} \tag{7}$$

$$\frac{\partial C_L}{\partial \tau} = 3.54C_j^{1/2} - 0.675C_j + 0.156C_j^{3/2} \tag{8}$$

$$C_j = \dot{m}V_j/\frac{1}{2}\rho c V_o^2 \tag{9}$$

Where

α Geometric angle of attack.

τ Jet angle (maximum of about 60°).

δ Foil thickness ratio.

$\dot{m}$ Mass flow rate of jet per unit length.

V_j Jet velocity.

V_o Ship speed.

ρ Fluid density.

c Foil chord.

The rounded trailing edge circulation control does not have a directional jet and changes the mass flow of the jet to change the lift. The jet adheres to the trailing edge and moves the stagnation point to the foil lower surface, taking advantage of the Coanda effect. The lift coefficient is the same as the jet flap when the jet angle, τ, is fixed. The amount of lift produced is strongly dependent upon the radius of the rounded trailing edge in relation to the rest of the foil. Rounded trailing edge circulation control is relatively independent of ship speed, and so is useful at low speeds.

The main advantage of these high lift devices over conventional fins is their high lift, which makes them useful at low speeds. Up to 50% of the jet can be recovered as thrust[58]. Cavitation and flow separation are delayed and controlled. Drydocking should be eliminated because jet flap fins can be designed so that maintenance can be performed from within the hull. With a fin fixed to the hull, which is a feasible alternative for high lift foils, much higher loads can be designed for than on current active fins.

Their main drawback is the large amounts of instantaneous pumping power required. For ships with little or no existing pumping, the additional pumping power required could be high. Circulation control uses less power than the jet flap, but also produces less lift. The small slot sizes required for high efficiency could create design and operational difficulties.

The addition of jet flap or circulation control to current prediction techniques is simple. It involves substituting the higher lift coefficient or lift curve slope for the lower lift coefficient or lift curve slope. With SMP84, this can easily be done by manually calculating a lift curve slope assuming a constant C_j to represent the average of the expected jet fluctuations and using this as an optional fin input. To avoid optimistic results it is necessary ensure that the pumping power is reasonable and the foil is not undergoing flow breakdown.

CONTROLLED VENTILATION

Controlled ventilation of a foil provides changes in lift force without moving the foil. Ventilating the upper surface will decrease the lift compared with the unventilated foil, while ventilating the lower surface will increase the lift. The fin is ventilated by blowing air through spanwise air holes arranged in the upper and lower surfaces. Which surface to ventilate and the amount of air is determined by the control system. The power required to ventilate the fins is small.

The advantages are simplicity due to a lack of moving parts, low power usage, and the delay of cavitation which extends the operable range of speed and angle of attack. The disadvantages include increased fabrication cost, mechanical complexity, and acoustic signature.

Reference 58 provides a conceptual design outline for installation on a destroyer hull form. The ventilated installation had better performance than the active fin stabilizer. This work resulted in some design guidelines; briefly they are:

1. Foil cross section shape, especially the nose, is important.
2. A spanwise notch on the upper and lower surface helps stabilize the ventilation cavity, and its depth determines separation size at higher speeds.
3. The ventilation holes should be angled toward the trailing edge to avoid leading edge separation.
4. Slight changes in thickness change the location of cavitation inception, not the inception speed.

This concept is also known as forced ventilation.

RUDDER ROLL STABILIZATION

Rudder Roll Stabilization (RRS) is feasible, economical, and essentially free of technological risk.This stabilizer makes use of the existing rudder to control roll motion rather than adding some other stabilizer. Rudder roll stabilization is not possible for every ship because of rudder size, location, and rudder rate requirements. Van der Klugt[60] specifies several necessary ship design requirements to perform RRS and discusses the RRS autopilot used. The practicality of RRS is that rudders can be used to reduce roll without affecting the rudder's primary function, course keeping. As most roll periods are of the 8–12 second range and yaw response around 30–35 seconds this is usually not a problem. It is possible to move the rudder at the roll period to reduce roll without leaving the rudder deflected long enough to create a yaw angle, i.e. the rudder angle changes slow enough to reduce roll and fast enough not change the yaw angle.

In order to be worthwhile, the rudder needs to be able to generate a roll moment that is a significant fraction of the total wave moment and at the roll response frequency. In other words, the rudder needs to be able to excite meaningful roll angles[61]. What this translates to is a relatively large rudder, with a center of pressure well below the roll center, to maximize moment arm. The rudder should also be fast acting to allow timely positioning of the rudder. If the phase between the roll rate and rudder deflection is not 180° the performance will be poor[61]. Therefore, candidates for RRS need large, low, fast acting rudders.

Aside from mere ability to reduce roll motion, there are concerns with maintenance and the rudder controller. The use of the rudder for stabilization should not increase system wear rates or reduce reliability. Lloyd[62] favors a controller law similar to that of fins, one that contains displacement, rate, and acceleration terms. Baitis et al.[63]and Cowley and Lambert[61] recommend only roll rate. They argue that rudders are only really capable of modifying roll rate because they do not generate as big a moment as fins. In that light, the displacement and acceleration terms are ignored. Experimental and trials data have borne this out[63]. The major controller problem is *rate saturation*, the inability of the rudder to follow the control signal. Rudder roll stabilizer controllers must limit maximum rudder excursions adaptively to minimize lag. Carley[64] deals with the intricacies of the control loop and how to maintain stability while including the autopilot. VanAmerongen and VanCapelle[65] developed a mathematical model for rudder roll stabilization.

The advantages of RRS is that very little is added to the ship, hopefully only the controller. When retrofitting this system, much of the existing steering mechanism can be saved, thereby avoiding high capital costs, and space and weight penalties. This method, especially when incorporated in the initial ship design, is extremely cost effective. There is no increase in resistance or degradation of stability as is associated with other devices. Rudder roll stabilization performs well for head to beam seas.

The main problem of RRS is that the stabilizer is impeded by manual steering. This is not viewed as an important short coming, since RRS can be run with an autopilot under normal operating conditions. In close quarters situations such as connected underway replenishment, the RRS system should be deactivated to prevent the helmsman from accidentally increasing roll. In situations of broaching, RRS will actually make matters worse due to erroneous response to the controller[62]. Rudder roll stabilization loses ability as the encounter frequency goes to zero, because the flow past the rudder goes to zero. Some criteria for the use of rudder roll stabilization from Baitis et al.[63] follow:

1. Rudder generated roll moment must be a significant fraction of the total wave moment and be at the frequency of roll response. Roll reduction increase is proportional to increase in rudder moment capacity. (About 15% total wave moment for the HAMILTON class Coast Guard cutter).

2. RRS should not create excessive yaw or increase the helmsman's task.
3. RRS should reduce roll before yaw is changed, i.e., timely application of restoring moment.
4. Steering machinery delays should not interfere with efficient use of rudder moment for stabilization. Rudder rates must allow for timely positioning of rudder by a simple, effective controller law based on roll rate.
5. The use of rudder for stabilization should not increase system wear rates significantly of reduce reliability.

DIRECTED THRUST

These devices include flaps in the wake of ducted propellers[53] and transom flaps[66]. The flap in the wake of a ducted propeller is an attempt to make a stern fin more efficient. This arrangement did show the same trends as other stern fin arrangements. The limiting factor was housing a large flap and its associated hydraulic actuating system in the stern of the ship, see Fig. 15.

To achieve maximum reductions it was necessary to operate the flaps in a hard-over to hard-over fashion. This type of operation is bad for the machinery and risks jamming the flaps. The flaps had little to no effect on heave and added a large disturbance to the wake. The mechanism for the ducted propeller and flap, which did greatly reduce pitch, was considered too complex for installation.

The transom flap controls pitch by creating a variable upward pressure on the stern to counteract pitch. The advantages are that it is contained within the hull and can be retracted to reduce resistance. The transom flap only worked at high speeds and even then did not work that well, requiring a large flap angle for minute pitch reduction.

STABILIZER EFFECTIVENESS

An important, but seldom mentioned aspect of stabilizer selection is the amount of stabilization desired for a particular ship design. If only a small amount of stabilization at low speeds is wanted, it is obvious that large fins would not be the stabilizer of choice. Also if it is possible to have large anti-roll tanks, should all the space and weight be devoted to large tanks? Would more moderately sized tanks produce adequate results and conserve space and weight? To determine answers for such questions, some method of quantifying the stabilizer effectiveness is needed.

Various methods of measuring effectiveness are available. Most involve ratios of some parameters representing the unstabilized and stabilized cases to predict a percentage reduction. The most usual parameters are significant or RMS motions, or damping coefficients.

$$\%reduction = \frac{\sigma_{unstab} - \sigma_{stab}}{\sigma_{unstab}} \times 100 \tag{10}$$

The problem with a percentage reduction method is that it gives the same importance to a reduction of 20° to 10° as reducing 6° to 3°.

Comparing ship motion transfer function peaks suffers from a similar problem. When the response curve is sharply tuned, reduction of the peak is important and the comparison is worthwhile. However, if the ship motion response is broad banded, a reduction of the peak may be meaningless if the ship response at other frequencies is increased at the same time. The percentage reduction methods are easy to apply and understand, if somewhat limited.

The "discriminating measure" described by Chaplin[67], compares the probability of exceeding a given limit with and without stabilization. This avoids the pitfalls of percentage reduction methods, is not overly complicated, and addresses the issue of adequate stabilization.

By assuming a Rayleigh distribution for the motion, we will only be considering the envelope curve and finding the reduction in number of excursions above a certain limit. Examining a Gaussian distribution with the same methodology will determine the amount of time spent above or below a given limit. The probability of exceeding a certain limit is found by integrating the probability density function (PDF) from the limiting value to infinity.

$$P(z \geq z_{lim}) = \int_{z_{lim}}^{\infty} \frac{z}{\sigma^2} e^{\frac{-z^2}{2\sigma^2}} dz = e^{\frac{-z_{lim}^2}{2\sigma^2}} \tag{11}$$

The limiting value, z_{lim}, is the value whose probability of exceedence equals the statistic of the criterion. For example, with a significant value, which is the average of the 1/3 highest values, the limiting value would be that whose probability of exceedence is 1/3, see Fig. 16. This value is found using

$$z_{lim} = \sigma_c \sqrt{2 \ln N} \tag{12}$$

where σ_c is the RMS value of the criterion. As an example, using 8° of roll, significant single amplitude, N = 3 and $\sigma_c = 8/2 = 4$. So the limiting value, z_{lim}, would be $4\sqrt{2 \ln 3} = 5.9°$. The RMS motion value, σ, chosen should be the maximum of the speed-heading combinations examined[‡].

The curve of probability of limit exceedence versus RMS pitch angle is a typical "S" curve, which brings out some important points about stabilizers. Stabilization may not be practical for two cases. The first where the motions are already small and the other where the motions are much larger than the criteria, see Fig. 17. If the motions are already small, there is little room for improvement and it becomes increasingly difficult to make improvements. Stabilizing the ship well below this value results in diminishing returns, because the increase in operability becomes constant. When the motions are large, then the problem is one of stabilizer power. If the stabilizer cannot

[‡] Remember that significant single amplitude values are simply twice the RMS values.

reduce the motions enough to make significant gains in operability, a stabilizer may be unwarranted. Obviously if the motions are in the range where the curve is steep, a stabilizer is appropriate as the increase in operability will be larger (i.e., motion below the criteria) than indicated by the decrease in motion. To achieve maximum benefits, the ship should be stabilized to values just below the RMS value of the criterion. Before deciding about a stabilizer, more than one sea state should be considered to investigate the effects of different wave heights and modal periods.

NUMERICAL EXAMPLE

As an example, add canted rudders to a destroyer to improve pitch performance. It is possible to determine a best rudder area from the operability view and the expected increase of operability due to the new appendages. Consider three different rudder sizes, 108, 214, and 324 ft^2, both active and passive. Pick a desired operating speed, say 20 knots and find the maximum pitch motion regardless of heading for each rudder configuration. Convert the pitch motion to a single amplitude RMS value, for use in Equation 11. Choosing 3° SSA (N=3) as a pitch criterion, yields a $\sigma_c = 1.5°$ and $z_{lim} = 2.22°$. Now calculate the percentage of limit exceedence by multiplying the result of Equation 11 by 100. The percentage motion reduction is calculated in a straight forward manner from Equation 10. The increase in operability for a condition is the difference between the percentage exceedence of the unstabilized and stabilized configurations. The results for this example are given in the following table.

	BASELINE	PASSIVE			ACTIVE		
RUDDER SIZE (ft^2)		108	214	324	108	214	324
PITCH (SSA degrees)	2.01	1.92	1.87	1.83	1.67	1.46	1.31
% EXCEED LIMIT	8.65	6.84	5.92	5.22	2.89	0.97	0.31
% MOTION REDUC		4.48	6.97	8.96	16.92	27.36	34.83
OPER INCREASE (%)		1.81	2.73	3.43	5.76	7.68	8.34

The probabilities for the differently configured canted rudders are plotted on the "S" curve. Because the motions were less than the σ_c, the probability of exceedence is very small, less than 9% for the baseline case. Also the increase in operability and percentage motion reduction are not the same. For the large active canted rudders, the motion is reduced 34.8% from the baseline, while operability only increased 8.34%. Thus, basing decisions on percent motion reduction would be overly optimistic. Furthermore, to use a larger rudder would be unwise as continued motion reduction below 0.6° RMS yields virtually no increase in operability. Therefore, the curve can be used to find to what value to stabilize for maximum use of the stabilizer.

STABILIZER SELECTION

With many stabilizers to choose from, making the optimum stabilizer selection can be difficult. The choice depends on the mission, weight, space, and cost limitations of the ship. Conolly[68] states that stabilizer design should be based on achieving a desired motion level at a given sea state, speed, and heading. However, usually the ship mission determines the stabilizer type, while weight, space, and cost determine stabilizer power. For the actual design of the stabilizer, the cited references should be consulted for a starting point.

Bilge keels should be a first resort to control excessive roll motions. They provide needed damping at low speeds, do not saturate in high sea states, and are relatively inexpensive. Almost all ships have them as a matter of standard practice and most will attest their beneficial effects. Ships that would not be candidates are those that require a bare hull because of resistance concerns; work in restricted waters, icebreakers; or frequent collisions, tugs.

The choice of using a tank, regardless of type (free surface, free flooding, U-tube, active or diversified), to stabilize the ship depends upon the following guidelines[25]:

1. Low speed operations are paramount. They are typically considered either a supplement to or replacement of bilge keels.

2. A hull penetration for retractable fins is not desired, or a desire exists to reduce the appendage drag.

3. "Some" stabilization rather than "a lot" is needed.

4. Cost considerations are more important than the amount of stabilization.

5. Apply to ships and missions where space and weight required for a tank stabilizing system is not a penalty.

The choice of which type of tank stabilizer to use depends on what is available in terms of space and weight on the ship. If stabilizing for a range of loading conditions is desired a free surface tank which can be easily tuned seems appropriate. If fore-aft passageways are required near the location of the tank, some form of U-tube tank is the choice. Free flooding tanks, because of the momentum drag penalty, are most useful for low speed ships. They also appear in ships that have existing voids or spaces that can be used for tanks because of the ease of retrofitting them, e.g. cutting holes in the hull. The choice between passive, passive-controlled, and active tanks is one between the extra cost and the increased performance. So the determination of a specific type of tank depends upon trade offs of the advantages and disadvantages of each tank type.

An active diversified tank holds the most promise in terms of motion reduction. It avoids most of the free surface degradation associated with tank stabilizers and is useful in irregular seas. The choice of controllers is important, because a good controller does not cost much more than a poor one and significantly increases efficiency.

Solid weight installations are few; however, they have uses on smaller ships. They can be used whenever a fluid tank would be used to save space and not degrade stability. Solid weight stabilizers are more expensive than equivalent fluid tanks.

Stabilizing fins can produce large forces and can reduce pitch and heave as well as roll. However, because of the magnitude of forces involved, pitch and heave are not typically stabilized for conventionally monohulls. SWATHs because of their small wave making damping can and do employ anti-pitch fins effectively. Active fins are generally the rule because of the marked improvement upon activation. They should be chosen as stabilizers for the following reasons.

1. Maximum motion reduction required regardless of cost.

2. Medium to high speed mission ($V_k \geq 10$ kn).

3. Want to reduce heave and pitch.

4. Cannot afford the stability degradation resulting from fluid tanks.

The low speed performance of fins can be augmented by the use of jet flaps or circulation control. These high lift devices also delay the inception of cavitation and separation.

Rudder Roll Stabilization is a powerful means of reducing roll. Its main advantage is that it adds very little to the ship in terms of space, weight, and cost and is very effective. If the rudders are canted from vertical, the lift force has a vertical component that can be controlled in a like manner to reduce vertical motions. It is most useful and cost efficient when the following are true:

1. Increasing the drag by adding other appendages would affect mission performance.

2. The existing rudder is large enough to provide significant restoring moment.

3. Rudder is mounted low on the hull to maximize the restoring lever arm, i.e., the distance between the roll center and the rudder's center of pressure.

4. The ship's yaw period is much greater than roll period, to minimize course changes while reducing roll.

5. The existing steering gear can provide the desired rudder rates for RRS.

Often a mission has conflicting stabilizer requirements, i.e. sprint and drift operations, or a ship geometry that does not allow for the size of stabilizer demanded. A combination of stabilizers is needed for such cases. Some common examples for multi-speed mission ships are bilge keel-fin and tank-fin configurations. Multiple tank and fin arrangements are good ways to increase total stabilizing power and at the same time be able to provide various levels of stabilization. These multi-stabilizer systems need to be properly designed to take into account interference effects. And the motion reduction achieved is to a level greater than the individual stabilizers alone, but usually not equal to the sum of the individual performance.

The other stabilizers mentioned, removable stabilizers, gyroscopes, and directed thrust, are poor choices for naval ships when compared with the other devices presented. Removable stabilizers are simply too cumbersome on larger ships. Gyroscope installations tend to use much of the ship's displacement and volume, and are expensive. Directed thrust devices were found to have a small payoff and are presently too complicated for installation.

CONCLUSIONS

Motion stabilization is not only needed, but possible. The choice of stabilizer depends on many ship and mission considerations. The large number of existing stabilizers makes it possible to find a stabilizer for virtually every conceivable mission, be it low speed trawling to high speed pursuit. The question of whether or not to have a stabilizer depends not upon the availability of stabilizers, but rather on whether or not a particular stabilizer will be useful. This can be determined by finding the increase in operability relative to some motion criterion.

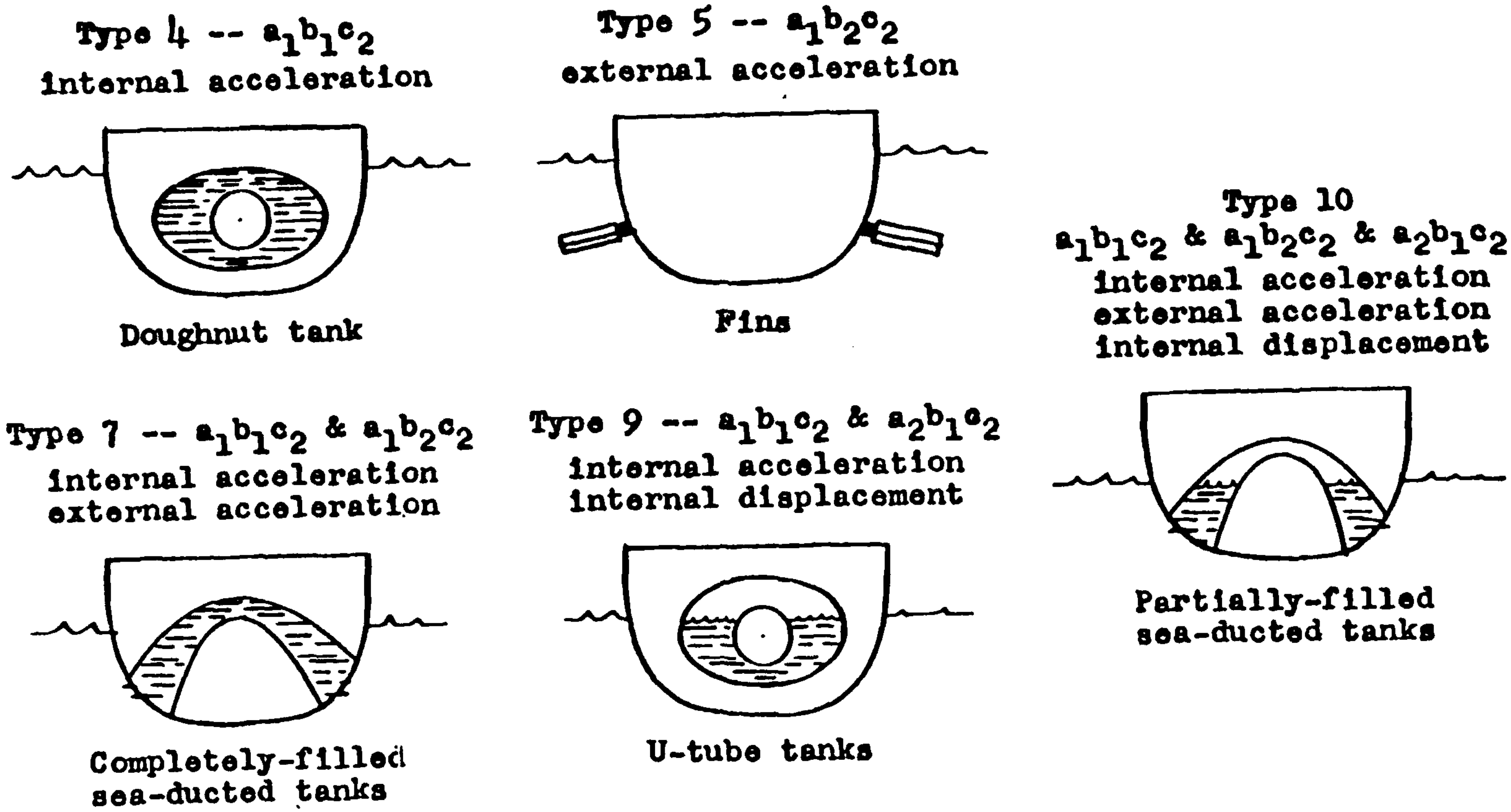

Fig. 1. Stabilizers using fluid mass. Adapted from Reference 6.

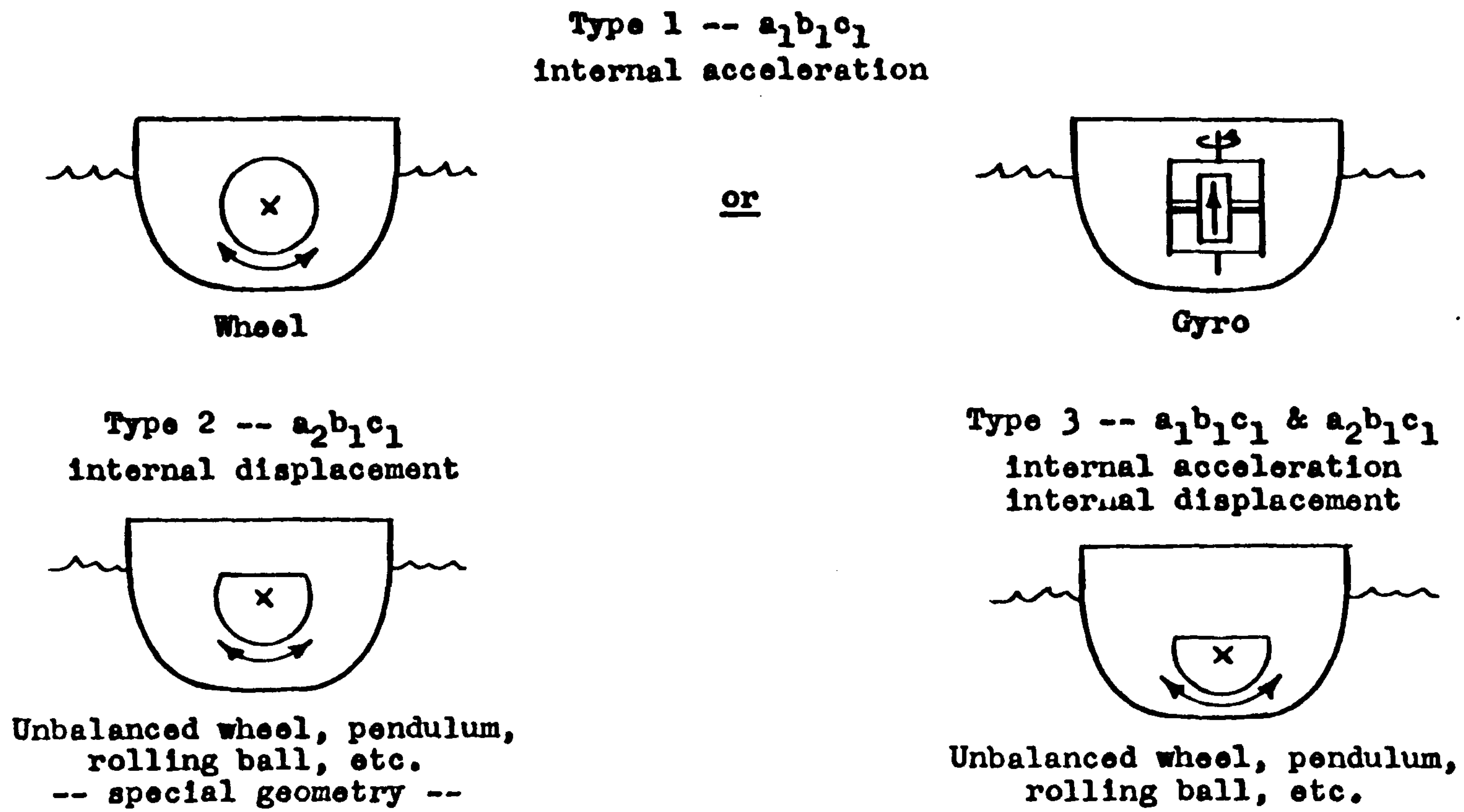

Fig. 2. Stabilizers using solid mass. Adapted from Reference 6.

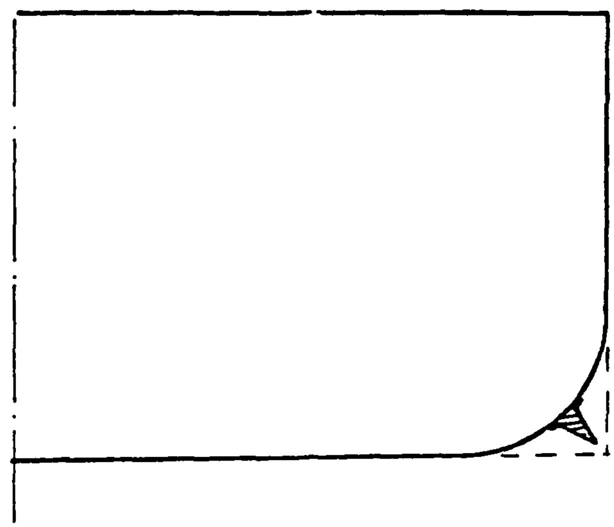

Fig. 3. Bilge keel arrangement. (Adapted from *Principles of Naval Architecture*; Reference 30.)

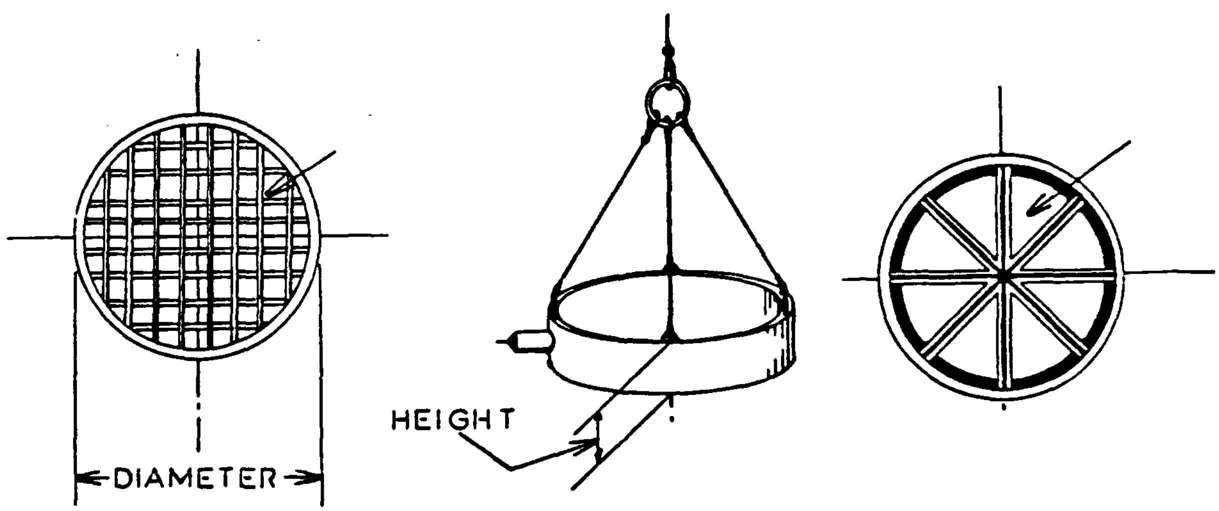

Fig. 4. Flopper-stopper stabilizers design. (Adapted from McCreight and Jones; Reference 20.)

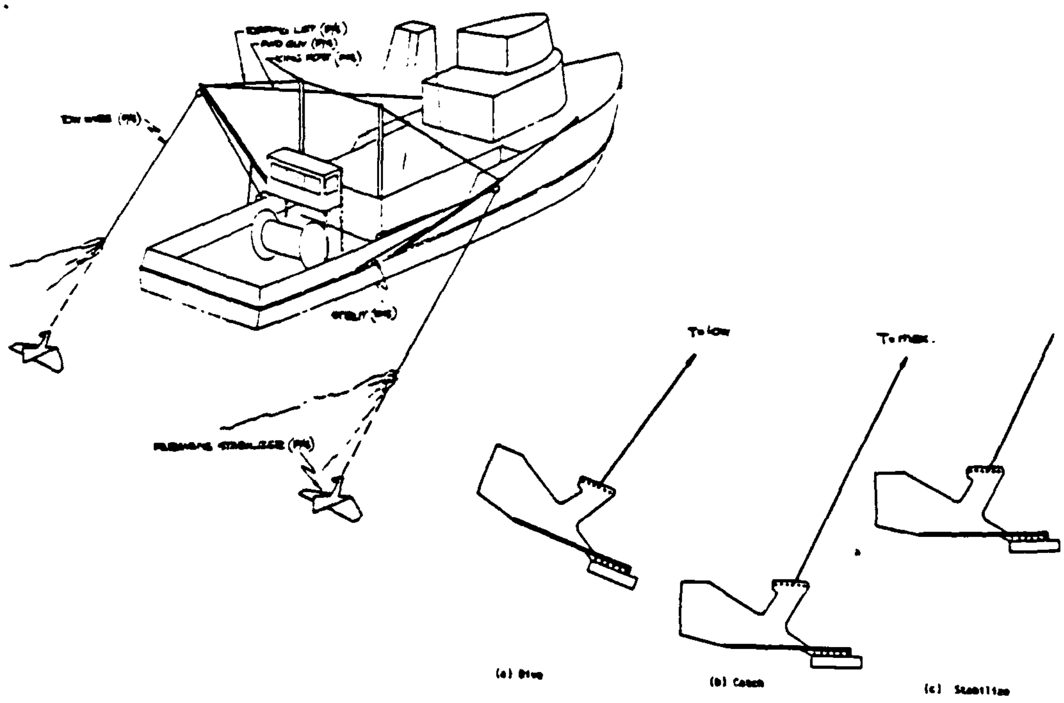

Fig. 5. Paravane stabilizers deployed. (Adapted from Kobel et al.; Reference 21.)

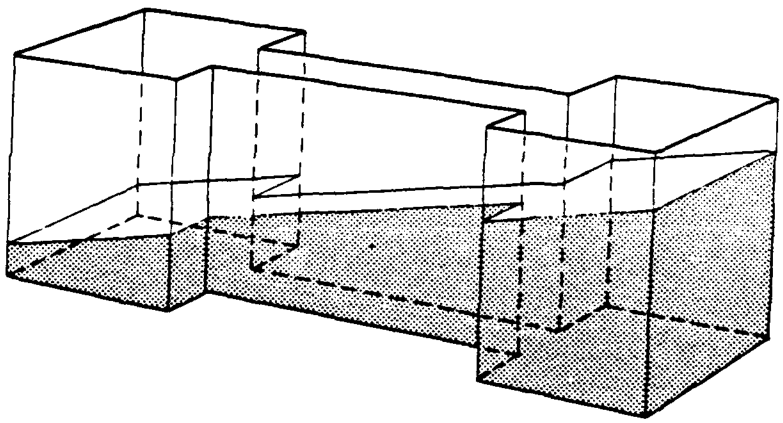

Fig. 6. Passive free surface tank stabilizer. (Adapted from *Principles of Naval Architecture*; Reference 30.)

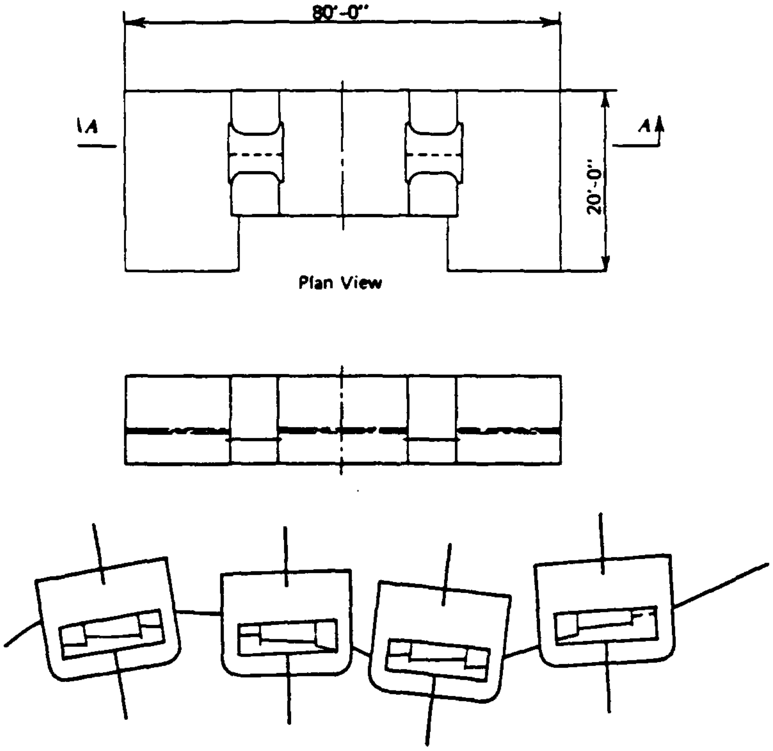

Fig. 7. Passive diversified tank stabilizer. (Adapted from Bhattacharyya; Reference 23.)

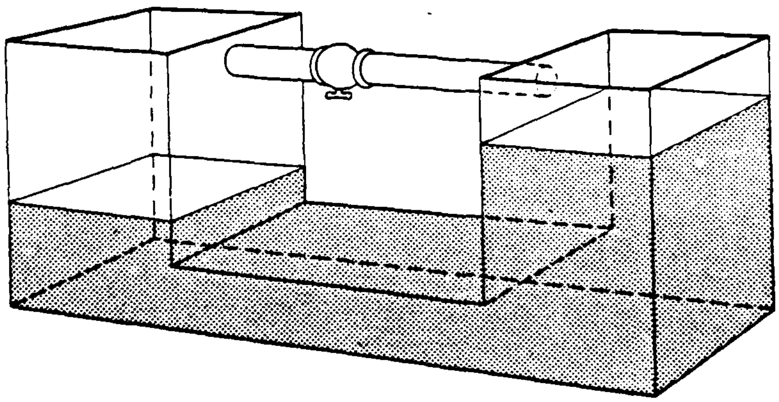

Fig. 8. Passive U-tube tank stabilizer. (Adapted from *Principles of Naval Architecture*; Reference 30.)

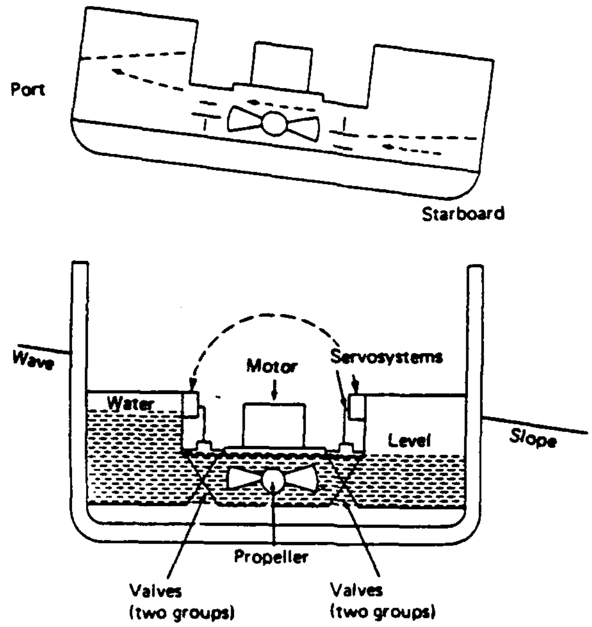

Fig. 9. Active Muirhead-Brown tank stabilizer. (Adapted from Bhattacharyya; Reference 23.)

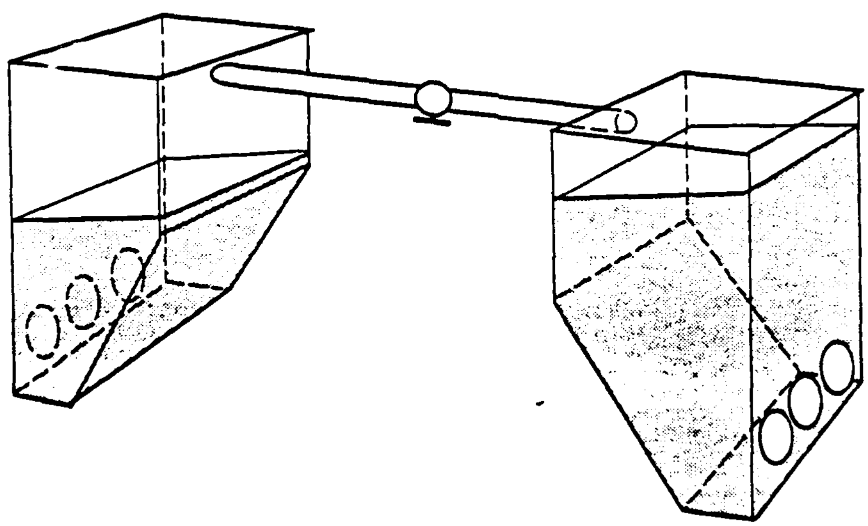

Fig. 10. Passive free flooding tanks. (Adapted from *Principles of Naval Architecture*; Reference 30.)

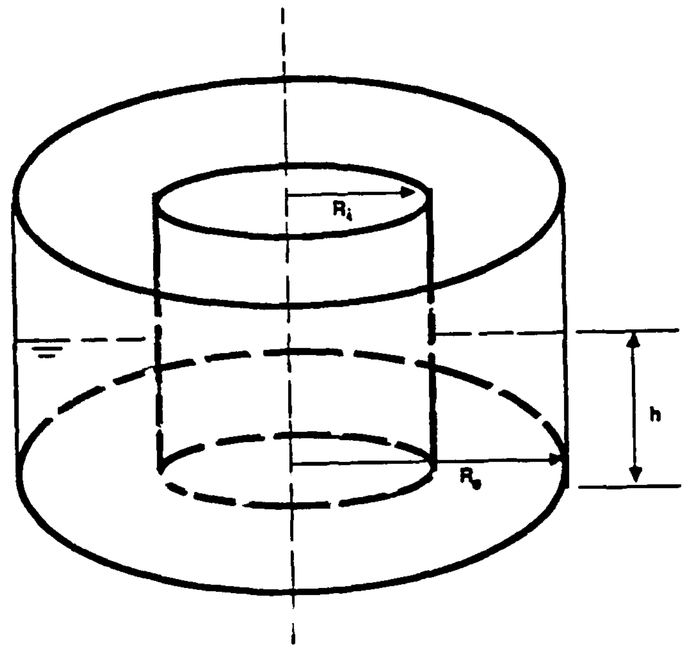

Fig. 11. Passive toroidal absorber. (Adapted from Venugopal; Reference 37.)

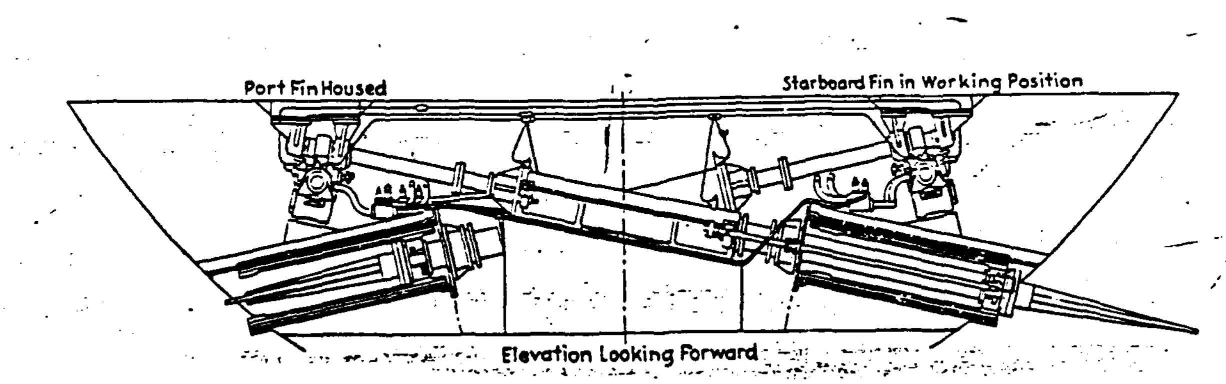

Fig. 12. Retractable high aspect ratio fin installation. (Adapted from Vasta et al.; Reference 25.)

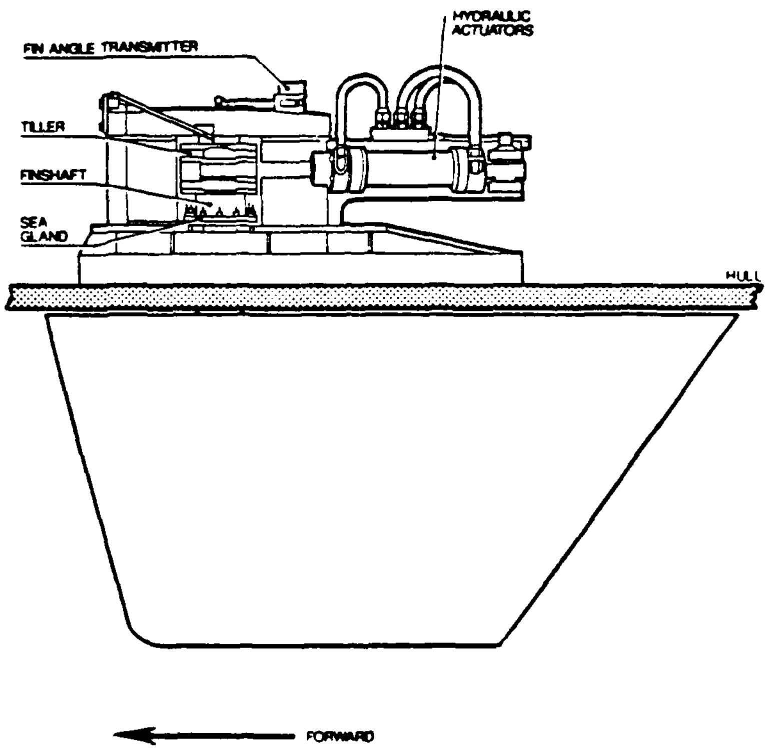

Fig. 13. Non-retractable low aspect ratio fin installation. (Adapted from Stafford; Reference 44.)

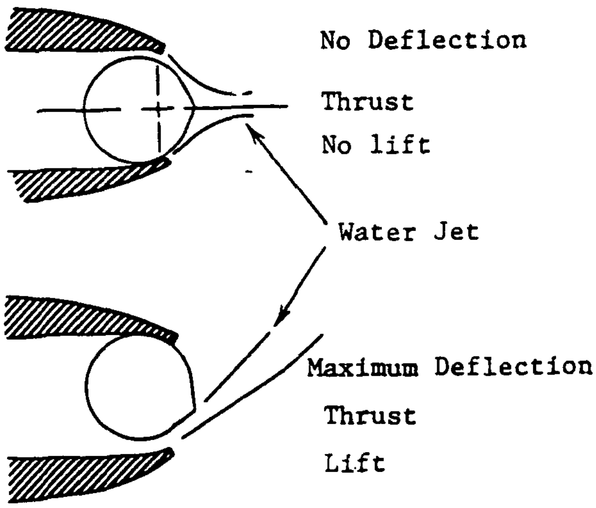

Fig. 14. Jet flap foil cross section. (Adapted from Bhattacharyya; Reference 23.)

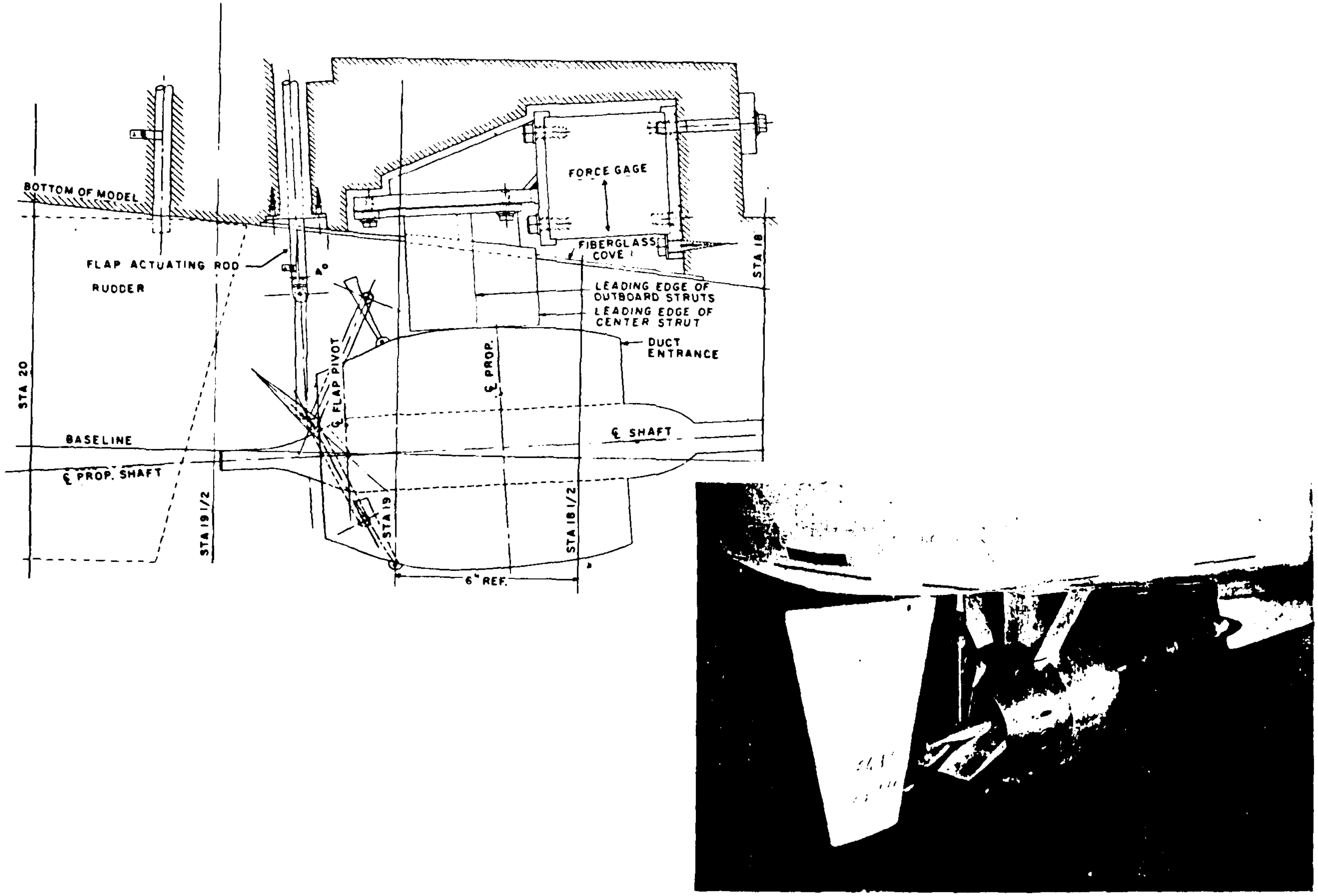

Fig. 15. Active flap in ducted propeller as a directed thrust device.
(Adapted from Gersten; Reference 53.)

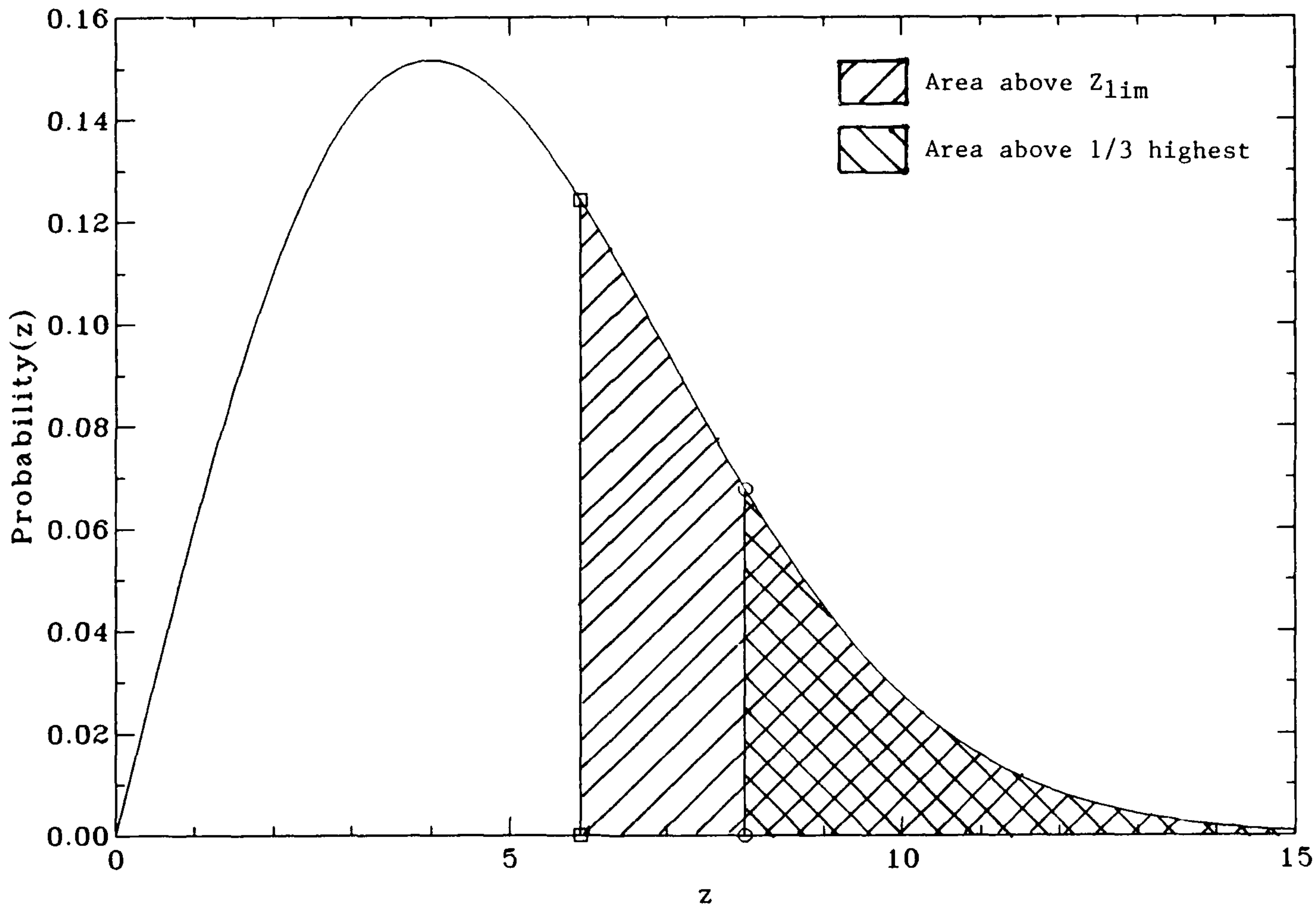

Fig. 16. Typical Rayleigh distribution with Z_{lim} and relationship to criterion.

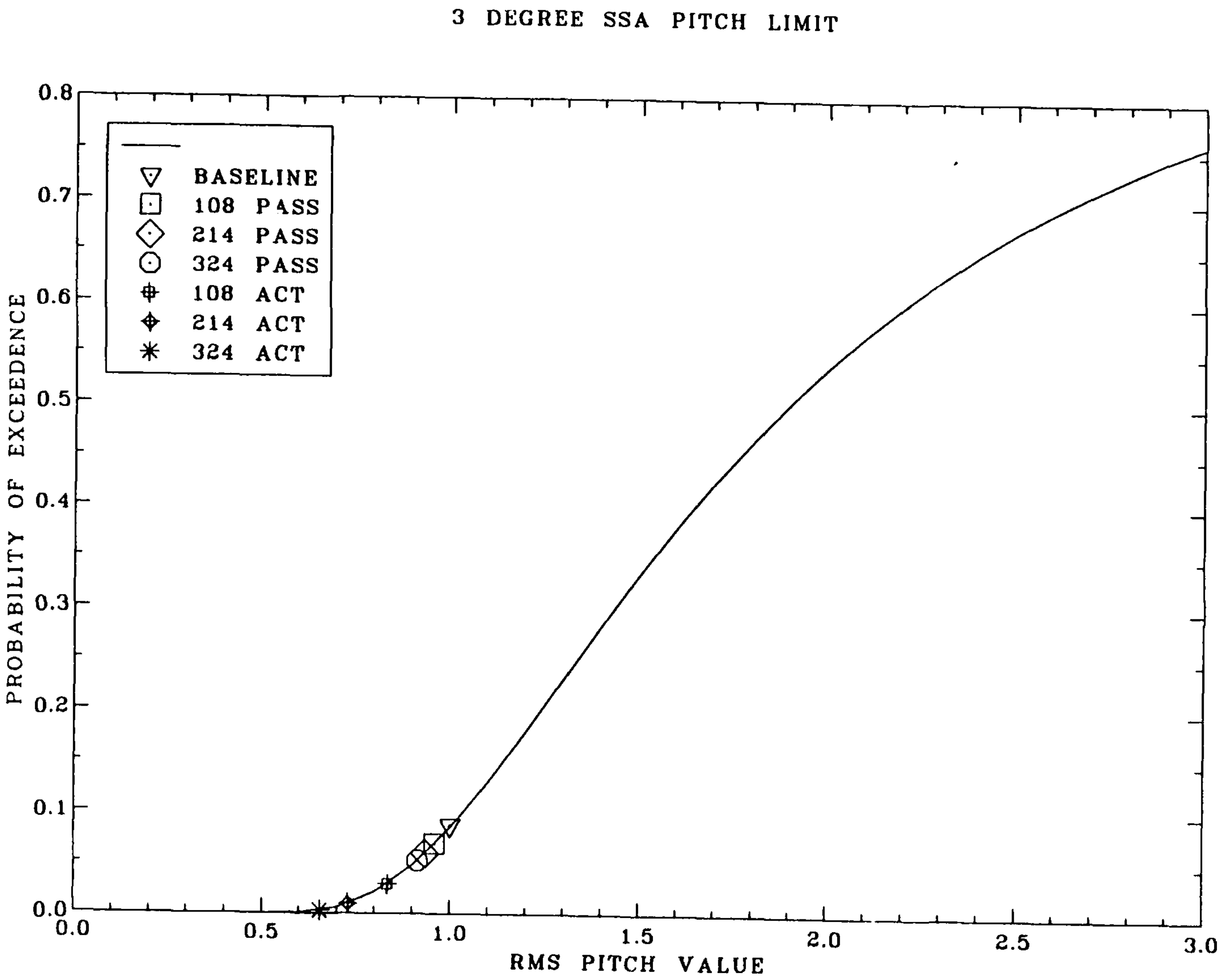

Fig. 17. Probability of exceedence versus RMS of variable resulting in a typical S-curve.

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