4. ROLL STABILISATION SYSTEMS

4.1. Introduction to Roll Stabilisation Systems

<u>The Need for Roll Stabilisation Systems:</u>

A vessel at sea can experience all the degree of freedom – surge, sway, heave, roll, pitch and yaw (see Fig. 4.1). How much the vessel oscillates about the static equilibrium position depends on many factors, important among which are the sea state, the size of the vessel, and the shape of the hull. The naval architect does a much as possible to reconcile all the conflicting requirements in a ship. The ideal would be to have no motions at all except progress at desired speed in the right course relatively to waves.

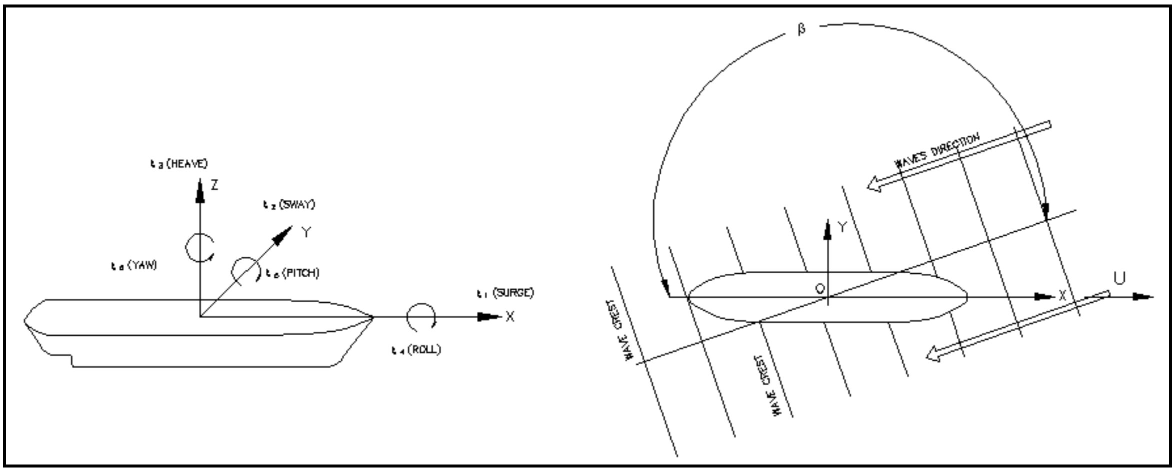


Fig. 4.1 – The right-hand coordinate system and six modes of ship motion, and definition of the ship heading angle.

Generally, by appropriate ship design, the ship's response in all the degrees of freedom under the influence of wind and waves can be limited, leaving direction and yaw control to the rudder and steering gear, and taking care of roll by various stabilisation methods. In fact, roll motion is the only motion which has low inherent damping and relatively small stabilisation moment. In addition to that, the magnification factor of a ship in a irregular seaway can go up to 10.

Pitch and heave inherent damping is very high and the required forces and moments so large that practical stabilisation systems for ships remain an elusive goal. Prior to that, magnification factors of a ship in a irregular seaway are about 1.3 in heave and 1.5 in pitch. Moreover, other modes, such as surge, sway and yaw do not present resonance peaks and therefore their amplitudes never exceed the wave amplitude or wave slope.

Therefore, the operability of a vessel is quite frequently affected by the motions and accelerations due to rolling, which can reduce both crew and propulsion efficiencies. Over the years, many systems have been developed. Some early ones were notably unsuccessful, such as vessels fitted with enormous gyro-scopes to keep them steady, or Bessemer's vessel with its hydraulically operated saloon which was supposed to stay still while the ship rolled around it. Little by little both technical and the control problems were overcome and two main controlled stabilisation methods have emerged and stood the tests of time. These are active fins or rudder(s) stabilisers and controlled passive "U"-type tanks. Bilge keels and passive free surface tanks have been also extensively appended to a ship's hull or supplied. Therefore, a variety of roll stabilisation systems are nowadays available to reduce this degradation to carry out the mission, in some cases with appreciable economic savings. Hence, a problem that should be addressed during the preliminary ship design stage of a vessel is to decide whether bilge keels alone are adequate to keep motions and accelerations within acceptable limits or whether these should be supplemented or replaced by another more cost effective stabilisation system.

4.2 Types of Roll Stabilisation Systems

Roll stabilisation systems can be grouped in two distinct general categories, depending on how they achieve the stabilisation, i.e.:

a) Passive;

b) Active.

Passive stabilisers do not require power or control system to operate. They belong to two separate sub-categories, those that do not have moving parts and those that do, such as:

a.1) Bilge keels;

a.2) Passive anti-roll tanks;

a.3) Moving weight stabilisers (prototype utilised in the past).

Active stabilisers improve the performance of a passive system by adding a sophisticated control system or uses machinery of significant power which modifies this action to make it more effective, such as:

b.1) Controlled or active anti-roll tanks;

b.2) Stabilising fins;

b.3) Combined controlled anti-roll tanks and active fins;

b.4) Stabilising rudder(s);

b.5) Gyro stabiliser (prototype utilised in the past).

a.1) Bilge keels

The most widely used, as well as the simplest kind of roll stabilising devices are the bilge keels, which are fixed fins attached almost perpendicular to the hull near the bilge (see Figure 4.2). The length of bilge keels are normally from 25 to 50% of the ship's length, and the widths vary from 0.3 to 1 meter.

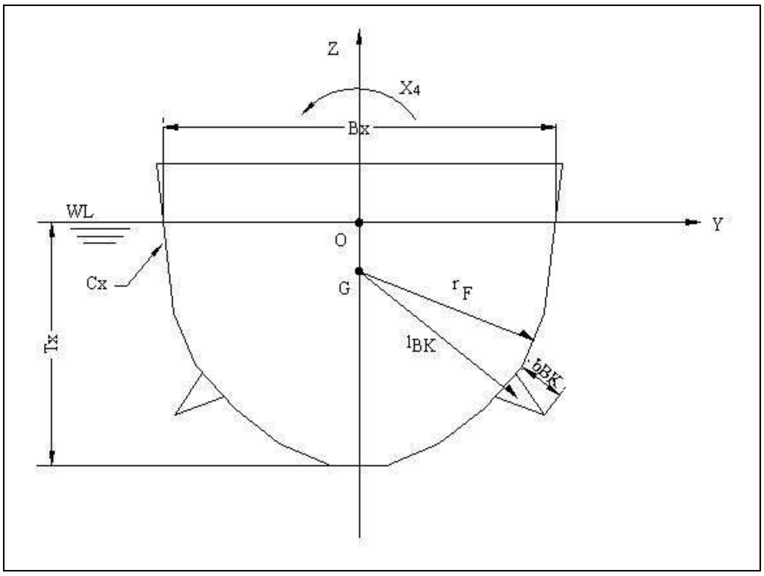


Figure 4.2 – Midship section with bilge keels.

In the case of bilge keels, the damping moment is generated by a component supplied by the pressure resistance of the bilge keels itself and by a component due to the change in the pressure distribution on the hull. For vessels not fitted with bilge keels, roll damping is caused only by the dissipation of energy in surface waves and in viscous-flow around the hull, and by surface tension. The latter component is not important for the full-scale case. Bilge keels increase considerably the energy dissipation due to viscous flow effects.

It has been found by experience that bilge keels are also effective in heavy sea conditions. The only drawback is that hull resistance is increased when the bilge keels are fitted, although this may not be considered a serious problem for high-speed vessels. The damping coefficient can be numerically estimated or by means of model scale rolling experiments. In order to not increase excessively the resistance of the hull during its forward motion, the longitudinal position of the bilge keels should be on the flow lines along the length.

Roll stabilisation by the use of bilge keels is simple and inexpensive but is effective only in the region of resonance.

a.2) Passive anti-roll tanks

This was the first type of roll stabilising tank to be developed and was based on the pioneering work conducted by Sir Philip Watts, he read a paper at a meeting of RINA in 1883 with another paper two years later (Watts, 1883, 1885). He proposed a large, uniform cross section tank partly filled with water, placed athwartships and usually located well above the centre of gravity, see Figure 4.3. The principle was based on the work by Froude (1861) who was the first to frame the effect of waves on the rolling motion of ships.

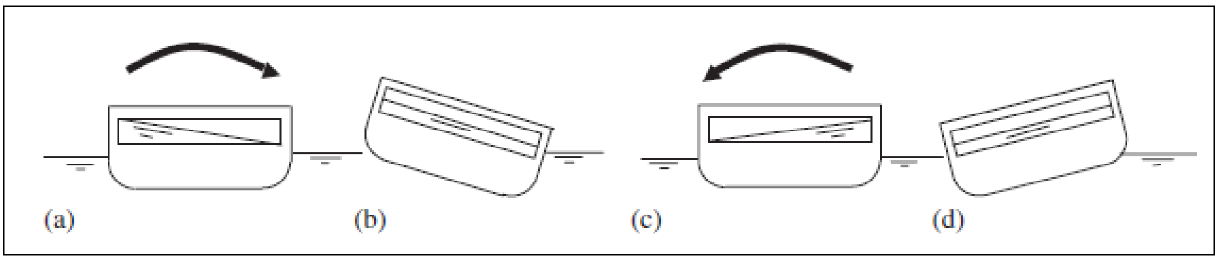


Figure 4.3 – Ideal motion of water in the tank (based on Watts, 1885). (a) t=0, maximum roll rate to starboard; maximum stabilizing moment to port. (b) t=Tn/4, maximum roll to starboard; zero stabilizing moment. (c) t=Tn/2, maximum roll rate to port; maximum stabilizing moment to starboard. (d) t=Tn3/4, maximum roll to port; zero stabilizing moment.

In 1882 Watts and Froude examined this theory with a full-scale trial and excited HMS Inflexible by moving men. They managed to roll the ship by 12º. The motion of the ship was recorded with a variety of water depths. It was observed that the roll damping was most effective when the tank was about half full, which made the natural period of the water the same as the natural period of the ship and reduced the roll by 37.5%. They realized that the water in the tank did not travel athwartships quickly enough as the ship rolled; they suggested that the efficiency of the tank would be improved if changes were made to enable the water to get across the ship more quickly. They also noticed that beyond 4º heel the rolling resistant force of the tank reached its maximum and remained the same, the phenomenon known as 'saturation' in the modern control.

More recently, experimental work investigated the coupling effect on the performance of a rectangular anti-rolling tank. The results of the bench test of an anti-rolling tank including the effect of sway motion as well as roll motion demonstrated that the sway motion reduced the reduction of the roll angle by the anti-rolling tank and lengthened the natural period of the tank. Also sea trial results showed up to 82% roll reduction in light wave conditions. The penalties identified were the volume occupied by the tanks and the reduction of stability due to the free-surface effect, a particular concern when a tank is to be retrofitted to an existing ship.

It was observed that the performance of free-surface tanks is maximized when the natural frequency of the tank is tuned to be close to the roll natural frequency of the ship. This is mainly done by altering the water level inside the tank. This confirmed the findings of Watts and Froude from a century earlier. One disadvantage of rectangular tanks is that it is difficult to control the water, rushing freely from side to side in the tank, threatening the safety of the ship in rough weather. Sometimes a limited control is exerted over the motion of the fluid by installing a restriction or baffle in the centre of the tank. Different responses of tanks can be obtained by changing their shapes, two modifications of such tanks are presented in Figure 4.4.

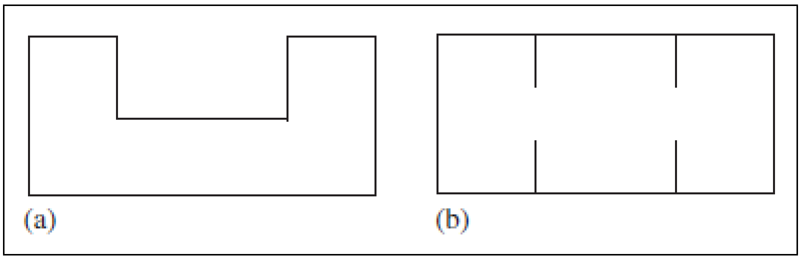


Figure 4.4 – Plan view of modified tanks: (a) C-shape tank and (b) rectangular tank with baffles.

b.1) Controlled or active anti-roll tanks

Controlled tank systems are designed to give 40-50% lower roll on average. Unlike passive tanks, controlled tank stabilisers react immediately and individually to any change in roll motion, providing top performance and flexibility. They can also be effective at low speed or when stationary for wave frequencies below the ship's natural roll frequency.

As illustrated in Figure 4.5, the “U”-shaped tank stabiliser compromises two tanks which are linked by a channel across the ship; one port and another starboard. The system is tuned so that when ship rolls, more water will rise on the “high” side, thus creating a righting moment to reduce the roll. More specifically, the natural oscillation frequency of the water inside the tank should be tuned to the most critical operational condition, which corresponds to the vessel’s natural roll frequency (resonance). At lower wave frequencies, valves block the water flow in cycles by controlling the air pressure above the waterline to create optimal effect (up to 50%). Where tank space is limited, a corrosion-inhibiting heavy fluid can be utilised in the tanks to increase the damping effect for a given size of system.

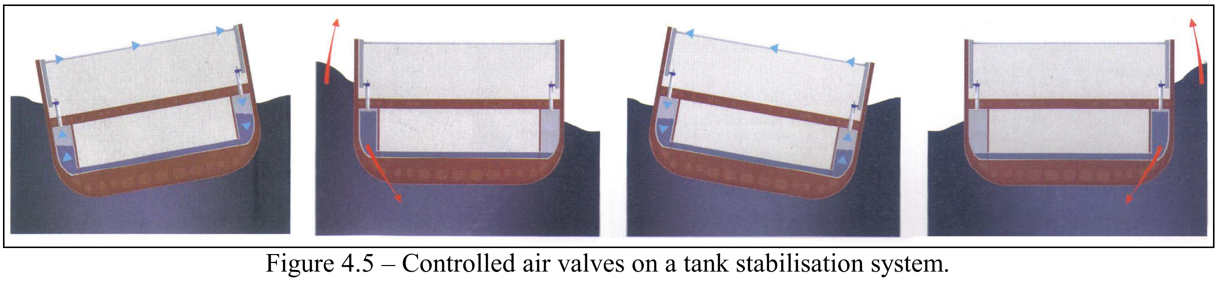


Figure 4.5 – Controlled air valves on a tank stabilisation system.

These tanks usually are small and have a mass of only about 2 to 5% of the total ship’s displacement. The inevitable free surface effect of stabiliser tanks can be compensated if these are included in ship’s design from the outset. As illustrated in Figure 4.6, more than one set of tanks can be fitted when necessary, and these can also be optimised to be effective in different ship’s loading conditions.

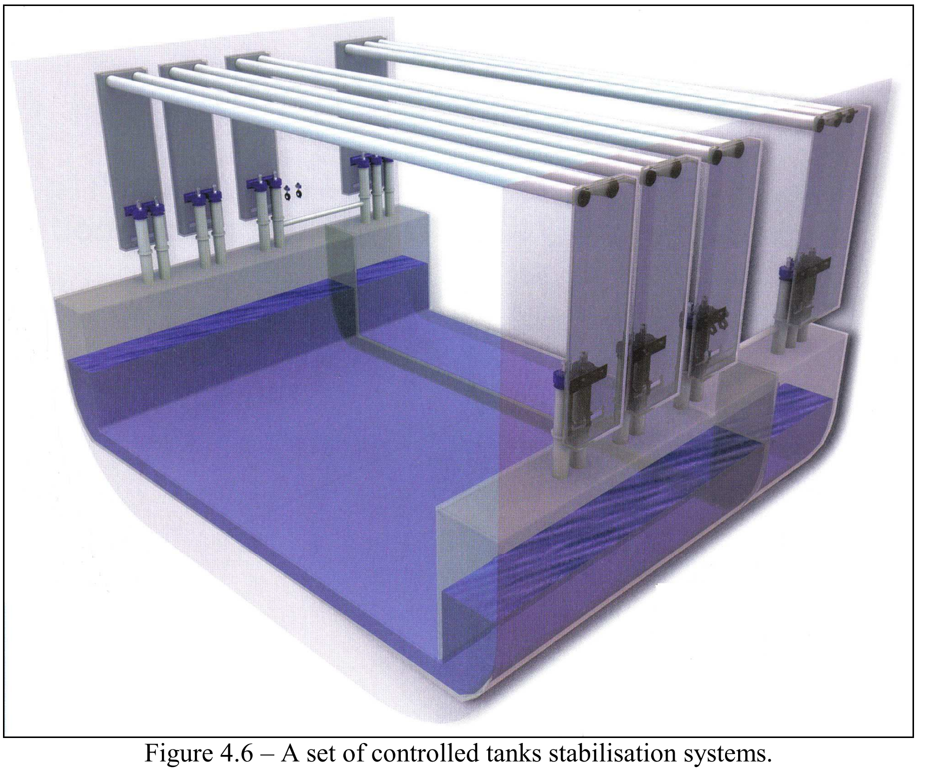


Figure 4.6 – A set of controlled tanks stabilisation systems.

The “U”-shaped tank stabiliser systems use water or heavy fluid (to reduce the space required up to 30%), and can be supplied as stand alone systems, or they can also be combined with an anti-heeling system for increased safety/payload in port (see Figure 4.7).

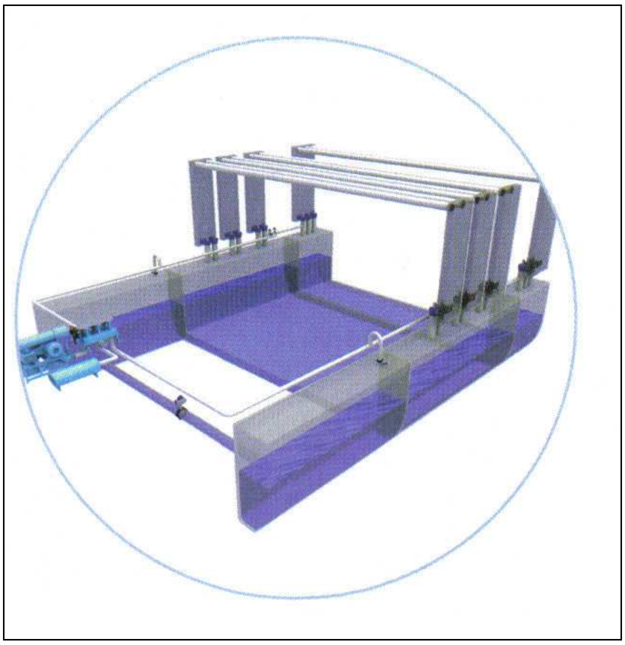


Figure 4.7 – Combination of controlled tank stabilisation with anti-heeling system.

Some vessels, for example, containerships, can be prone to so-called parametric rolling. The stability moment of the ship is the product of the righting lever and the total weight. In head or following seas, the righting lever varies periodically due to changing wave elevation along the ship's hull and the vessel's heave and pitch motions. This, in turn, causes the stability moment to vary, which can trigger rolling. The phenomenon is known as parametric rolling because its source is the time variation of a parameter. This resonance can cause the ship to roll to very large angles in a moderate sea, leading to cargo damage, loss of containers, and in extreme cases structural damage to the ship.

Large containerships are prone to parametric rolling because of the shape of the fore and aft body are different from the box type midship section, leading to a variation of righting levers as wave crests and troughs move alongside the ship's hull. At high speeds the natural dynamic roll damping tends to prevent parametric rolling, and therefore parametric rolling tends to be prominent at low speeds when the wave encounter period is about half the ship's natural roll period. A small initiating force at the right time from rudder or wind gust can trigger the ship rolling to larger angles. Sea conditions likely to produce parametric rolling in large containerships occur about 9% in the North Pacific and 12% in the North Atlantic.

Occurrence of parametric rolling can be mitigated if a system is added to the ship to provide additional damping which counteracts the roll exactly the right time. "U" type tank stabiliser systems are used to prevent parametric rolling. A complete ship system typically includes several pairs of tanks and pneumatically controlled air valves, plus a control unit with pitch roll sensors. The controller detects the starting phase of parametric rolling and tunes the tank water period and its damping characteristics by operating the air valves according to the current ship motions. As illustrated in Figure 4.8, the system is effective in preventing parametric rolling and also reduces the roll motion in general by about 20%. This smaller reduction is because the systems are smaller than conventional tank systems.

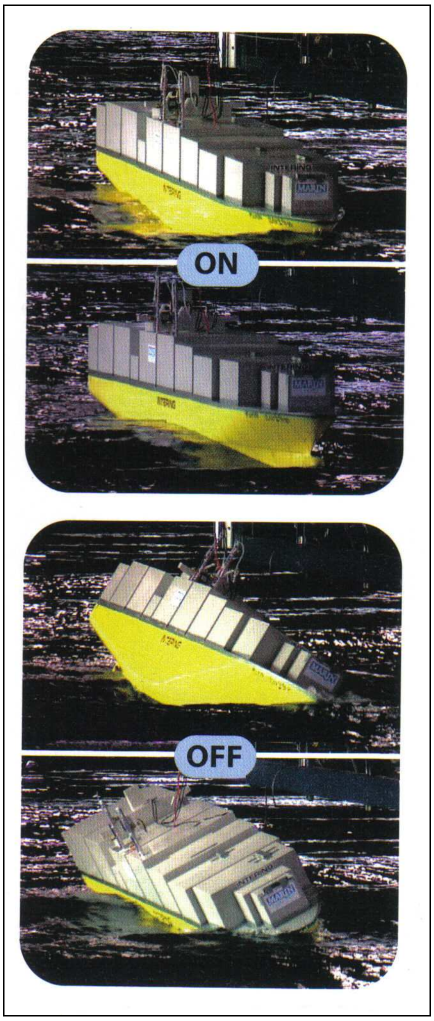


Figure 4.8 – Parametric rolling avoidance using controlled roll tanks stabilisation system.

It is possible to use feedback control systems in anti-roll tanks in a fashion similar to the control systems used in active-fin roll stabilisers. These tanks are invariably of the U-type configuration. The motion of the ship is sensed, this information is processed, and action is taken to change some feature of the tank system accordingly. If the action is such that energy is put into (or extended from) the tank liquid, then it is called a fully-active tank system, usually referred to simply as an active anti-roll tank. Figure 4.9 shows the typical arrangement of this type of roll stabilisation system. Another possible arrangement of an active roll stabilisation system, is made up of a powerful air pump, instead of a water pump, feeding two pipelines and valves connected to each air chamber of the lateral reservoir of the U-type tank. As illustrated, an active anti-roll tank system is basically the same configuration as a U-type tank. Due to complexity of fully-active systems, information about the pump and control system cannot be generalized. However, it is possible to define the required differences in tank geometry. The major differences between an active tank and those designated for pure passive stabilisation is that the natural frequency of the tank with air valve open is 30 to 40% greater than the ship's natural roll frequency and it is required that the tank have an equivalent linear damping ratio somewhat lower than a good passive tank. As a result, care must be taken in order to avoid any superfluous structure within the tank itself. The design of the tank configuration itself, follows along exactly as a U-tube design.

For active tanks, it is necessary to design the control system and to select the control system gains to provide maximum roll reduction and to avoid control system instabilities.

Of the three different types of tank systems discussed so far, the active anti-roll tank offers the best performance in roll motion reduction. However, this system is more costly and requires more complex components. For instance, a typical arrangement includes a controllable-pitch propeller pump connected to a motor. The pitch of the propeller is varied by hydraulic actuators commanded by the automatic control system. In a well designed system, during part of the roll cycle, power is extracted from the tank, and in other parts of the cycle, power is supplied to the tank. In such a well designed system, the average power required is near zero and usually negative (meaning that a net amount of power must be extracted from the tank). However, the instantaneous power required (either into or out of the tank) is usually large. A typical 6000 [ton] ship may require an 1865 [kW] pump for this purpose with an average net power out of the tank of 4%. It is not surprising that the system extracts energy from the tank. It is the energy that the tank has extracted from the ship roll motion which must be dissipated. The pump system provides the means of energy dissipation.

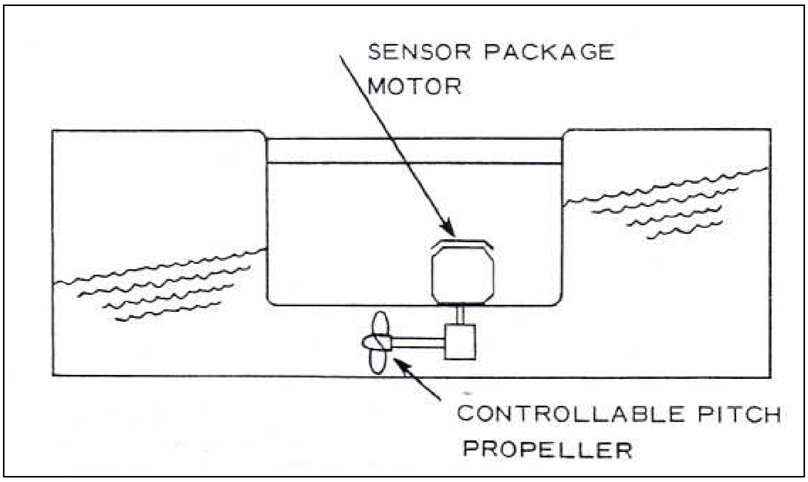


Figure 4.9 – Active anti-roll tank.

b.2) Stabilising fins

In an active-fin roll stabilisation system, flow over the fin caused by the forward speed of the ship generates roll moments which oppose the wave excitation roll moments in response to the command of a control system.

As illustrated in Figure 4.10, active fins stabilisers are fin-type control surfaces, which are usually located just above the turn of the bilge near amidships, port and starboard. The fin system may consist of a simple fin or a fin with a trailing edge flap, or articulated fin which, for a given planform area, can develop greater lift than a simple fin.

As illustrated in Figure 4.11, in a seaway, hydraulic tilting gear continuously varies the angle of attack of the fins, using automatic control system that is sensitive to the roll displacement, velocity, and acceleration of the ship's tendency to roll.

As illustrated in Figure 4.12, the fin is fabricated directly onto the fin shaft to form a single component. A fin unit has the following main components:

Fins

The fin planform is the shape of the fin in plan view and is defined by a span and a mean chord in terms of the geometric aspect ratio, which is equal to span divided by mean chord. The tip of the fin is the outermost end, and the root is that section closest to the hull.

The fin is connected to the shaft (or stock) by means of either a tapered key or a stock nut. The fin is fabricated from steel plate, similar to a rudder. The plating is normally 6 to 9 [mm] thick depending on fin size and plate strength. Plating support is provided by means of webs, intercoastals, and leading and trailing edge castings. The fin is watertight and is normally filled and drained with anti-corrosive coatings prior to installation.

Fin Shafts

Fin shafts or stocks are normally steel forgings or centrifugal castings, usually hollowed centres. Use of high-strength steel is encouraged, since stock diameter can then be kept to a minimum. This, in turn, reduces the size of bearings and castings in addition to the weight, and provides for a narrow root section. The stock diameter and casting thickness at the root of the fin determine the maximum width of the fin at the root. This is usually 15% of the root chord; in fact, from hyrdrodynamic considerations, a 15% ratio is nominally accepted as being satisfactory compromise between delayed cavitation at the leading edge and adequate lift characteristics.

Fin Shaft Bearings

These bearings are described as either outboard or inboard bearings. Normally outboard bearings are sleeve stave-type bearings and inboard are anti-friction roller bearings. Stave bearings are preferred for the outboard application because they are easier to maintain, are less susceptible to damage from se water and contaminants, and are more resistant to leakage in conjunction with a conventional stuffing tube. However, stave bearings have higher frictional torque coefficients and this results in higher power requirements than roller bearings. A primary seal assembly is fitted between the outboard bearing and the fin, and a secondary seal is fitted in the inboard bearing.

Hydraulic Power Unit and Actuators

The power unit provides all the hydraulic power required by the fin system. Each power unit assembly is mounted on a fabricated bedplate and is driven by a squirrel cage-marine type motor. Both power unit and hydraulic pump are normally resilient mounted, to decrease structure borne noise and vibration and, in some cases provide resistance to shock loads. The hydraulic pump is normally an axial piston variable-delivery type and is connected to the electric motor by means of flexible coupling. Hydraulic power may be applied to the fin tilting mechanism which applies torque to tiller that is passed through the fin shaft. The tilting mechanism may be either a Rapson slide or double acting cylinders. As illustrated in Figure 4.13, hydraulic fins drives may be of the cylinder type or the rotary vane type, being the latest the most compact and common hydraulic drive system in use nowadays.

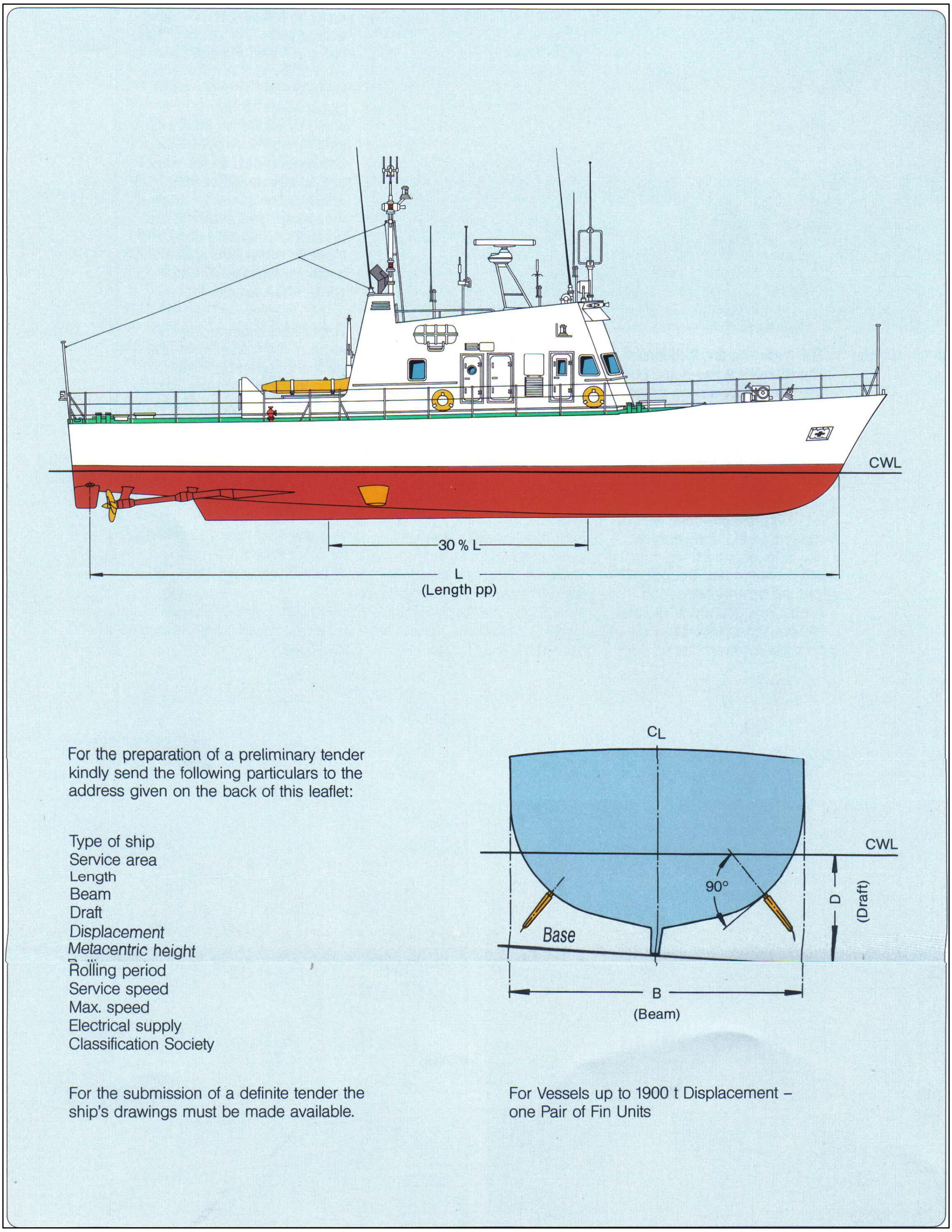


Figure 4.10 – Active-fins stabilisation system.

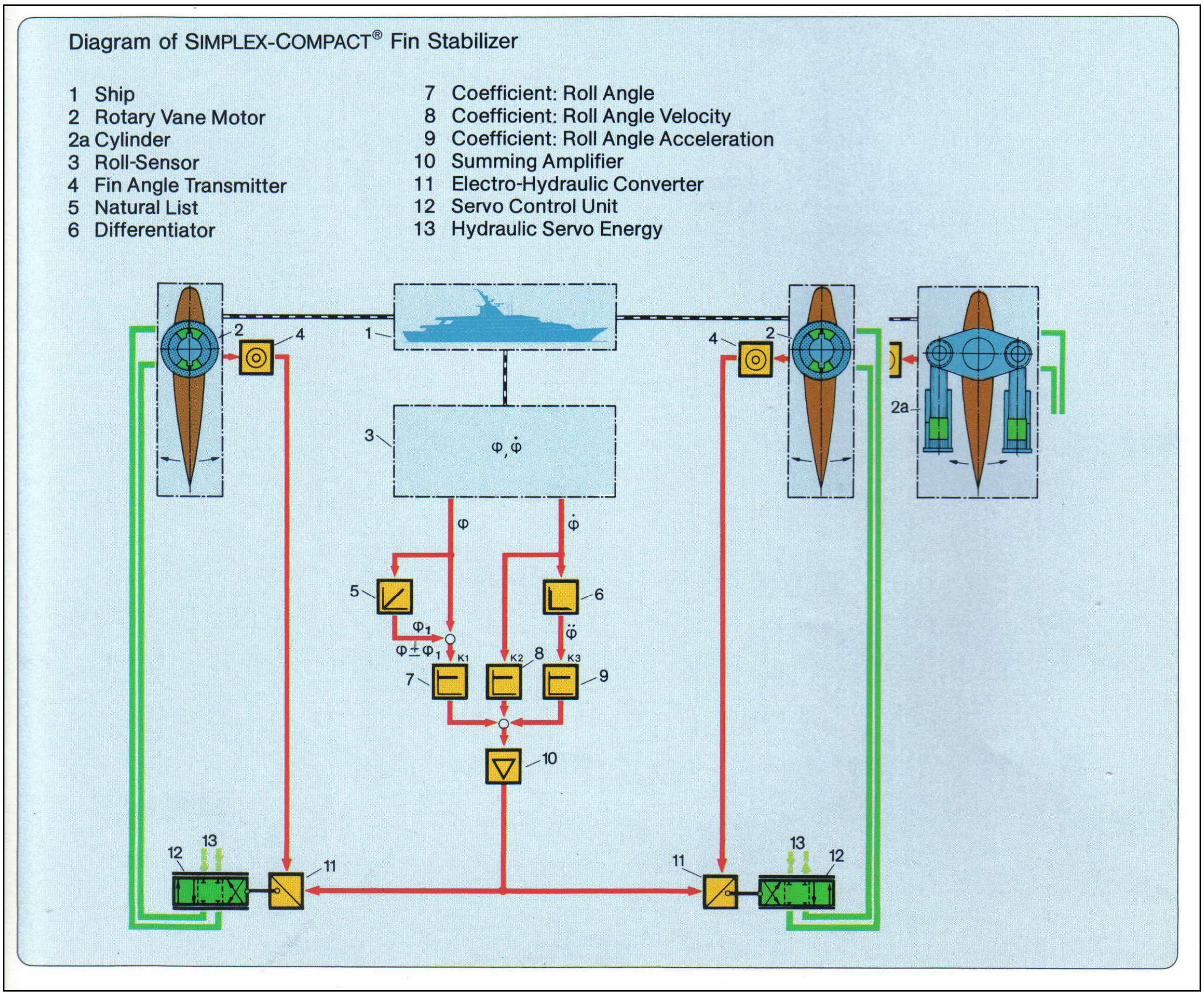


Figure 4.11 – Controller of active-fins stabilisation system.

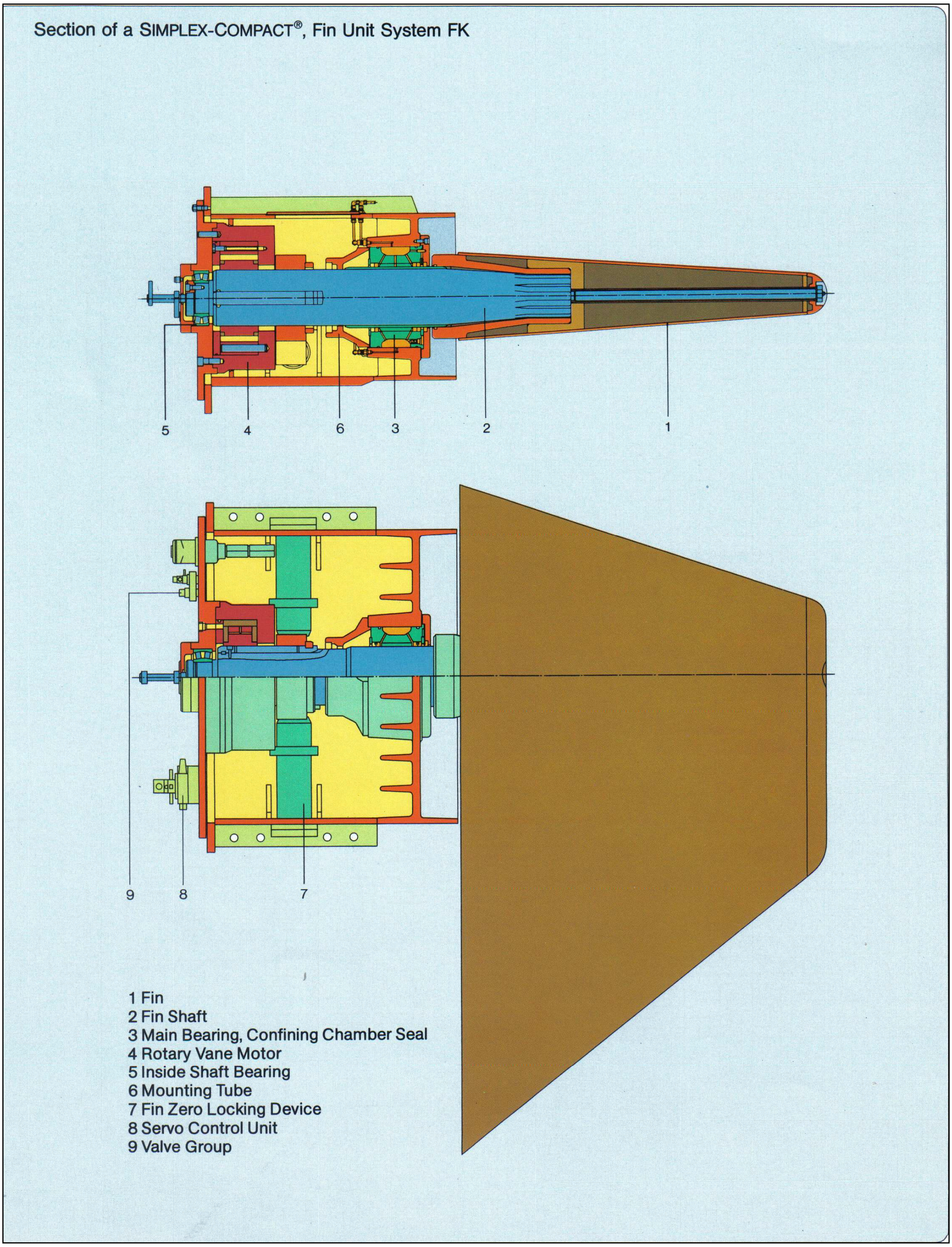


Figure 4.12 –-Fin unit.

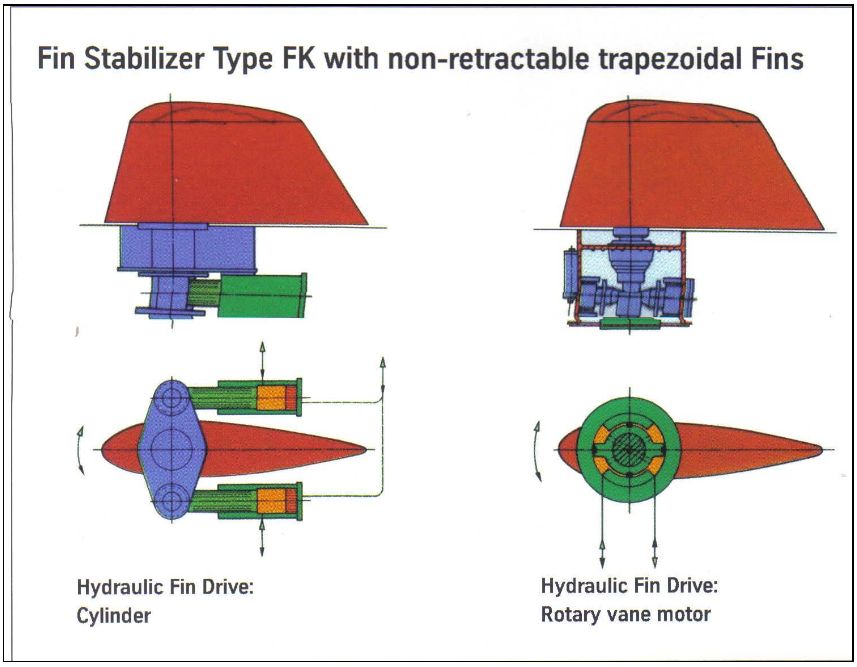


Figure 4.13 – Types of hydraulic drives most utilised in active-fins stabilisation systems.

Active-fin stabilisers require ship forward motion in order to develop lift, and the lift developed increases with the ship speed squared. In practical terms, this speed dependency limits the application of active-fin stabilisers to ships speeds above 10 to 12 knots. Below that speed range the required fin size becomes too large, and other devices (e.g. anti-roll tanks) become more advantageous. As shown in Figures 4.14-16, fin stabilisers may be of the fixed type or retractable depending on hull form and ship's operational profile.

Although, the determination of percentage of stabilisation in a specific case and the design of automatic control entails complex analysis which is briefly covered later on sections 4.3 and 4.5 of these class notes, a simplified calculation of required stabilising moment is presented below.

The required fin size and tilting gear machinery characteristics and location can be determined with sufficient accuracy using a simplified approach. With the simplified approach, it is assumed that a regular beam seaway having small surface wave slope and a wave period approximately equal to ship's natural period can build up large roll angles. By designing fin stabilisers to counteract the wave slope heeling moments, it is possible to reduce the large roll angles. The rolling moment induced in a seaway is expressed as follows:

$$F_{W4} = \Delta.\overline{GM}_t.\sin\theta_w \tag{1}$$

, where:

F_{W4} = roll-induced moment, in [kNm];

Δ = ship displacement, in [ton];

$\overline{GM}_t$ = transverse metacentric height, in [m];

$\theta_w = \sin^{-1}\left(\dfrac{\pi H_w}{\lambda_w}\right)$ = maximum design wave slope (for a wave height, H_w, and a wave length, λ_w), in [deg].

The stabilising moment developed by fin stabiliser is illustrated in Figure 4.23 and is given by equations (9) and $F_{F4} = 2F_L r_F$.

If the stabilising moment computed from equation (1) is equal or greater than the induced rolling moment calculated from equation above, an effective stabilisation system is considered to be achieved. The key to this simplified method is the somewhat arbitrary selection of the seaway wave slope capacity, θ_w, and the associated ship speed, U. Experience has shown that stabilisers should be designed for wave slopes of about 4º to 5º. Lower values are reasonable for very large ships, and higher values may be used for small ships, since small ships are more likely to be subjected to roll excitation in a given seaway.

The lowest ship speed, U, at which significant roll reduction is desired is usually taken to be approximately 50 to 80% of full-power speed. This implicitly recognizes that ship in heavy seaway are generally operated well below full power.

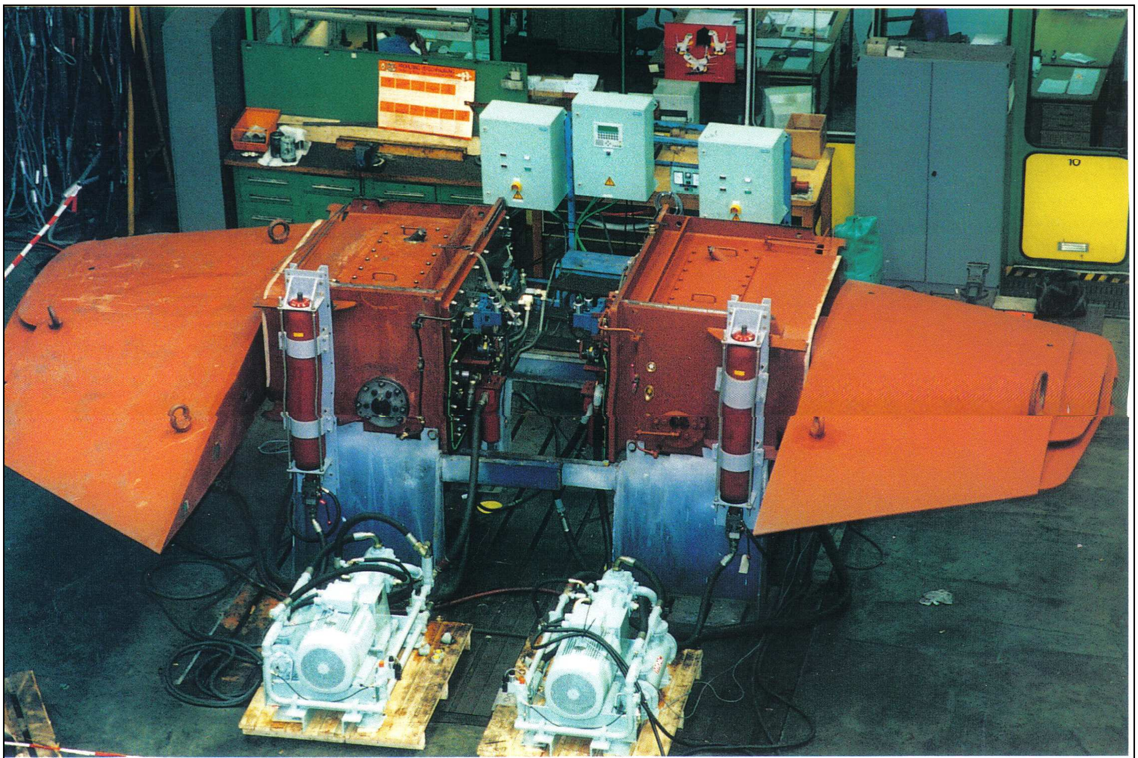


Figure 4.14 – Fixed type fins stabilisation system from SIMPLEX COMPACT series (ThysenKrupp™).

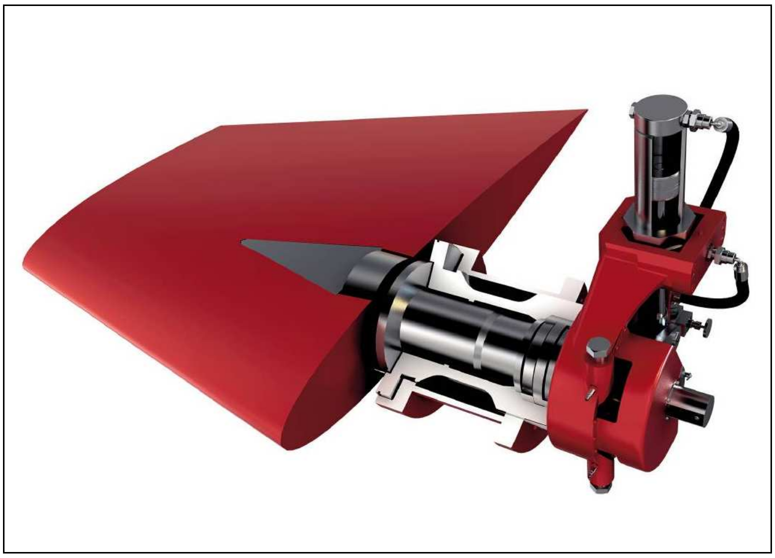


Figure 4.15 – Fixed type fins stabilisation system from GEMINI series (Rolls-Royce™).

Figure 4.16 – Retractable type fins stabilisation system.

b.3) Combined controlled anti-roll tanks and active fins

Fins rely on forward motion to produce the restoring force and hence are ineffective below about 6 knots they also impose a drag penalty which increases with speed. By comparison tanks have been relatively ignored during the last half century because fin stabilisers have been able to provide a good alternative for most applications and tank systems have a reputation for poor performance. Recently there have been a growing number of requirements for stabilisation of vessels at slow speeds, while the advances in control theory and computational power provides tools to combat the poor response times of tank systems making them more attractive at higher speeds too.

Where a vessel's operating profile includes slow and medium speed operation a combination of fins and tanks might be considered. High speed vessels are usually inherently stable in roll at design speed, but at low speeds they can have poor roll characteristics as the hull form has been optimized for speed. A combined controlled active roll tanks and fins system, such as the one shown in Figure 4.17, could be a solution for such vessels which also have to operate at low speeds, for example the new POR Navy patrol boats.

At high speed the tanks are emptied to reduce displacement, while at low speed they can be filled, “U”-type tanks are particularly suited for this. By comparison a fin system which is large enough to provide roll stabilization at slow speed could produce unacceptable drag at high speed.

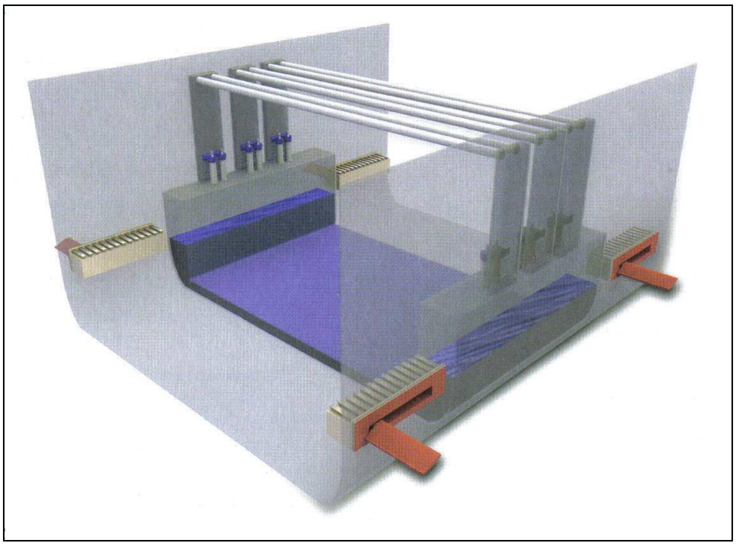


Figure 4.17 – Combined controlled active roll tanks and fins stabilisation system.

b.4) Stabilising rudder(s)

The concept of using the steering gear and rudder for the purpose of stabilising a vessel against rolling was introduced in the 1972, and the first sea trials to prove the concept were performed in 1975 in the UK with simple control systems. Since then, however, the development of rudder roll stabilisation systems has progressed in many countries, including the USA, Sweden, Holland, and Denmark. As illustrated in Figure 4.18, ships ranging in size from small patrol boats and cutters to destroyers and large ferries have subsequently been equipped with rudder roll stabilisation systems. The technique can be considered well proven on vessels that have rudders with the capability of imposing a sufficiently large roll moment on the hull.

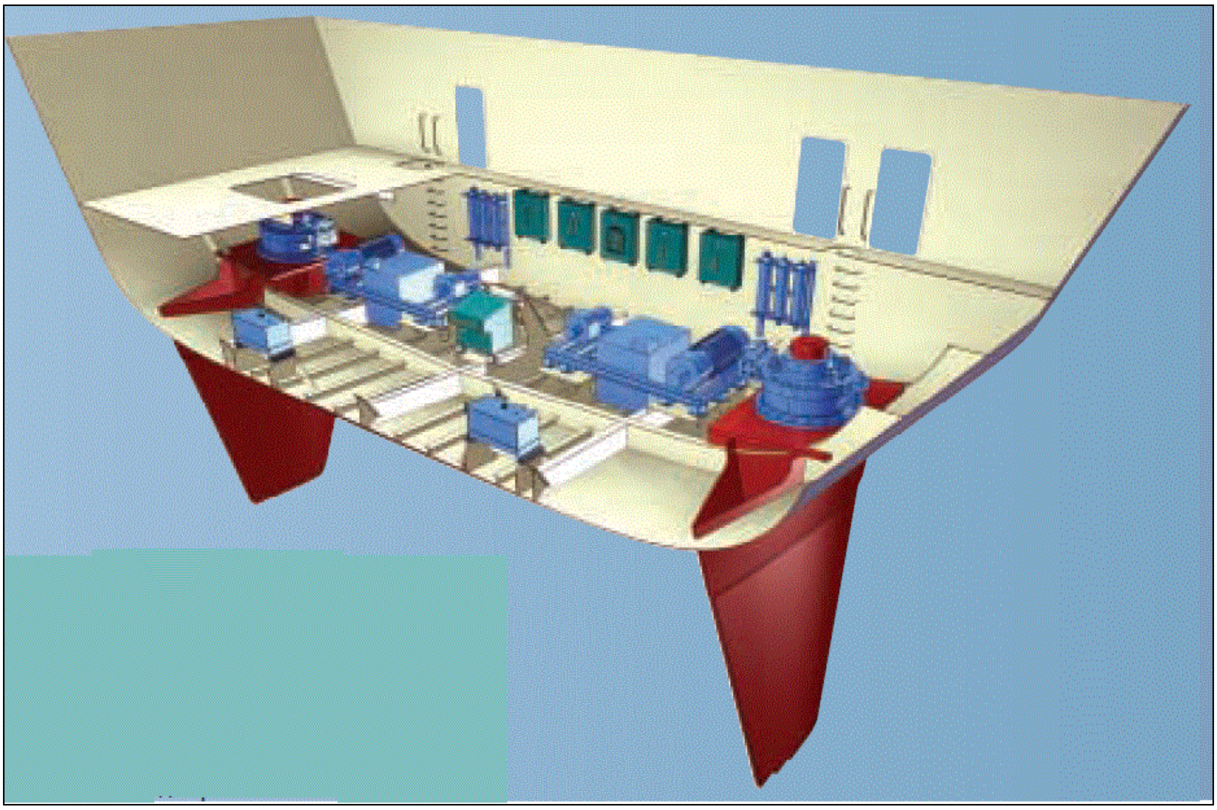


Figure 4.18 – Stabilising rudders arrangement.

In general terms, a rudder roll stabilisation system is an adaptive control system that uses the steering gear and rudder. Thus, it is simple to install both on new ships and as a retrofit on existing vessels, and the space requirement is negligible.

A block diagram of a typical system with an integral autopilot is shown in Figure 4.19. The course and speed signals are taken, respectively, from the ship's course gyro and speed log. Both rudder position and the position of the helmsman's wheel are also obtained so that the roll damping function can be provided while the ship is being steered manually. Many of the filters and controllers are normally of the adaptive type in order to accommodate varying sea and wind conditions.

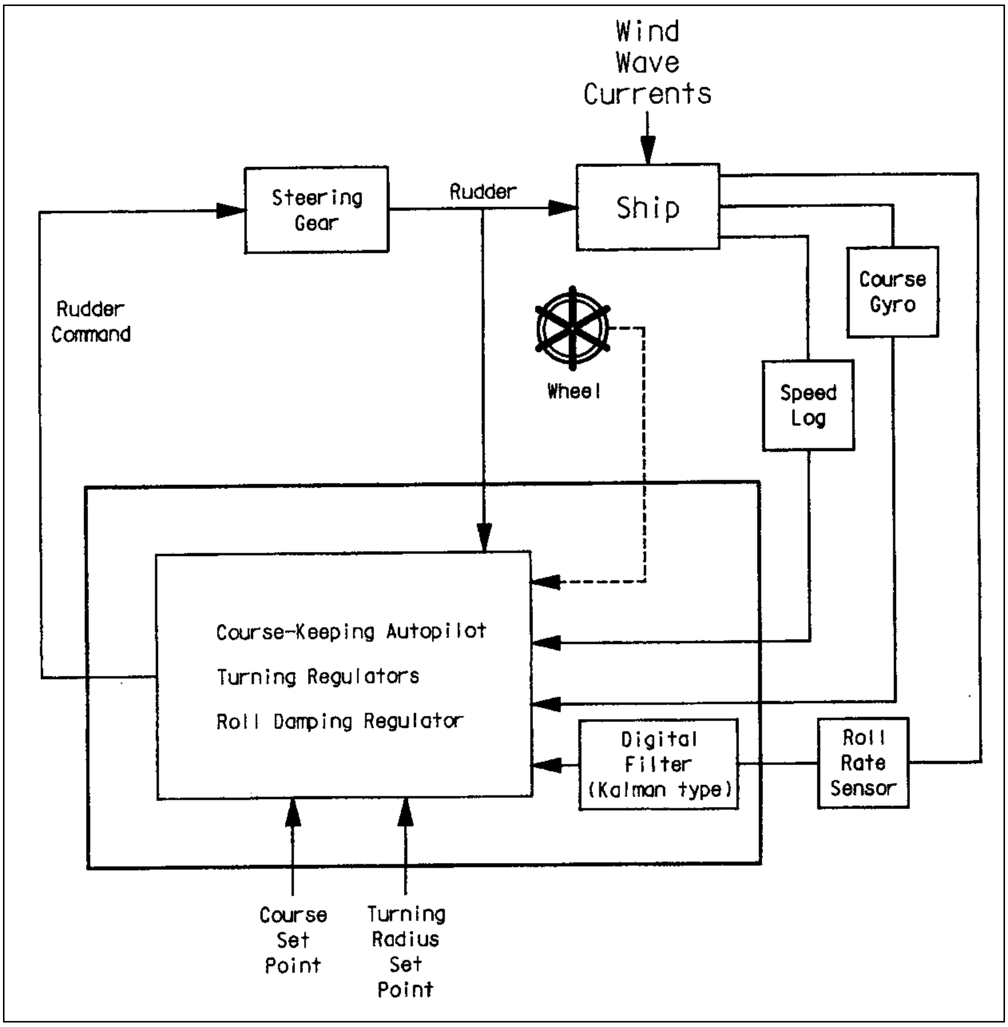


Figure 4.19 – Rudder roll stabilisation system block diagram.

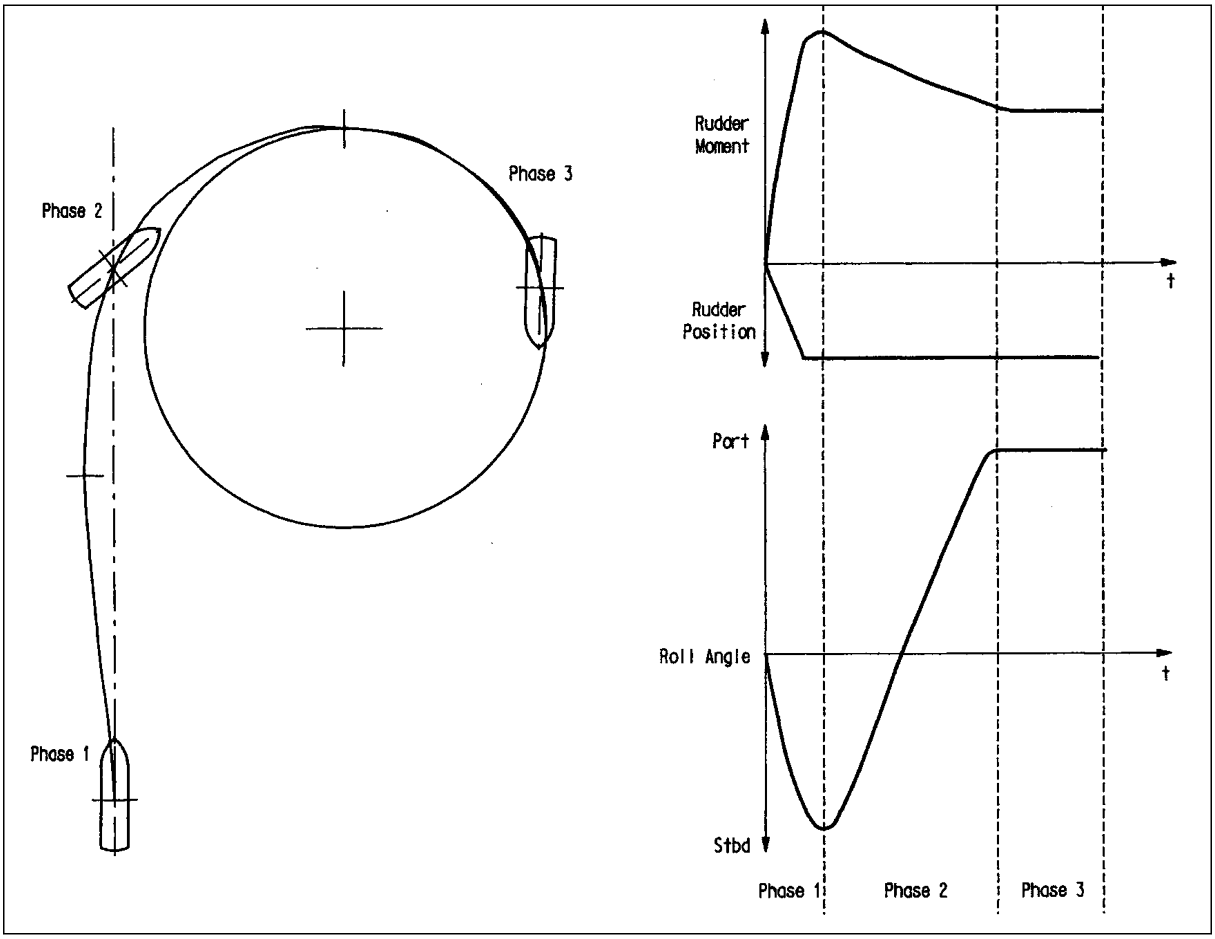


Figure 4.20 – Roll dynamics during a turn.

The operating principle of rudder roll stabilisation is based on the concept of opposing the roll moment created by the waves with an induced roll moment developed by the movement of the rudder, thereby damping the ship's roll motion. Figure 4.20 shows the response of the ship during a turning circle. During phase 1, the rudder is displaced and the ship starts to roll in the direction of the rudder displacement, that is, into the turn; however, during phase 1 the heading of the ship changes very little. During phase 2 of the turning circle, the heading of the ship begins to change significantly and the ships roll to the opposite side, that is, out of the turn. The operating principle of the rudder roll stabilisation system is to use the phase 1 sequence repeatedly as indicated in Figure 4.21. By using the rudder to impose rolling moments on the ship that oppose those created by the waves, the ship's roll motion can be effectively decreased, and the rudder forces used to effect this stabilisation are of such short duration that the ship's heading is not substantially affected.

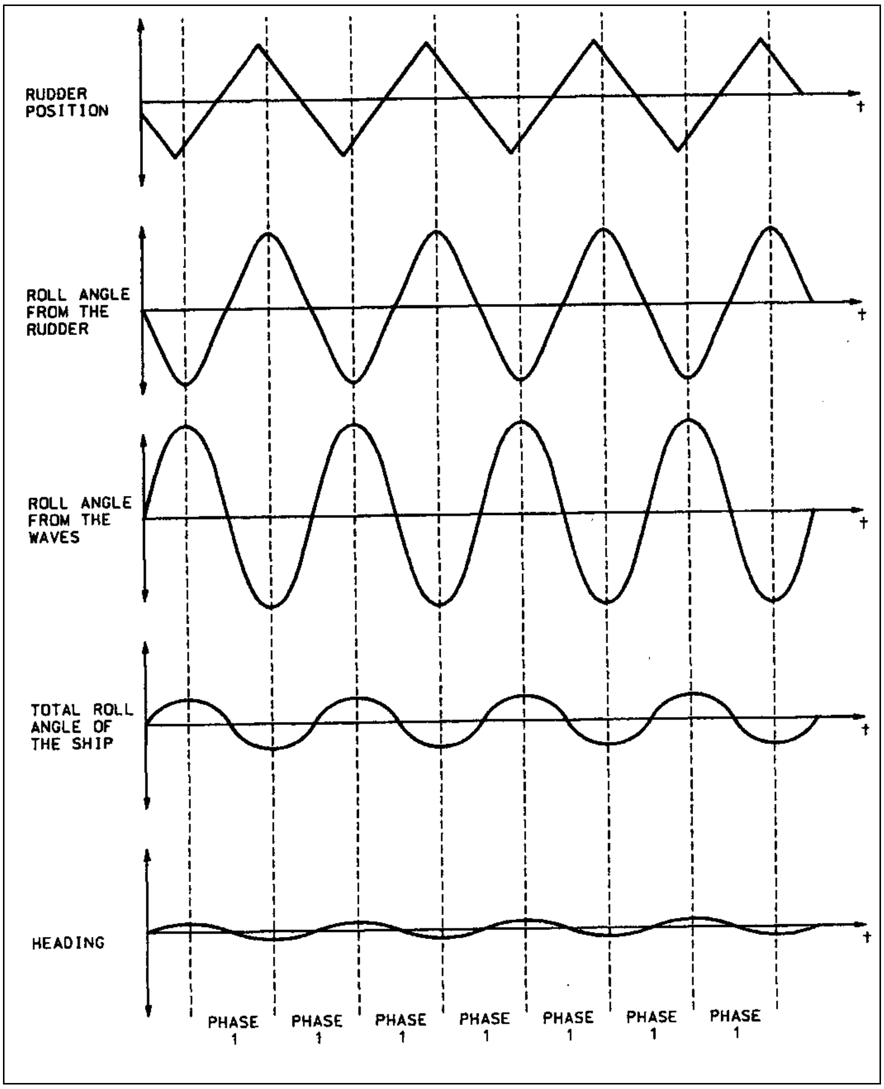


Figure 4.21 – Rudder roll stabilisation system operating principle.

Many factors influence the roll damping efficiency that can be achieved by the rudder roll stabilisation on a specific ship. The more important factors are the ship speed and rudder rate, but the metacentric height, rudder type, rudder area, and rudder position are also influencing factors. Sea trials indicated reductions in rms roll motion of between 36% and 61% at medium speed, and between 21% and 38% at high speed. Up to 40% reductions in the rms roll angles were frequently observed on a destroyer at various ship speeds in beam, quartering, and following seas and in various sea states up to sea state 5.

When using a rudder roll stabilisation system, the loads on steering gear and rudder will increase, especially for systems that require high rudder rates (in the order of 10 to 15 deg/s). Thus, it is important to consider the more rigorous duty cycles. However, steering gears are conveniently of rugged design, and there have been no significant indications of excessive wear or early failures due to rudder roll stabilisation being back-fitted on existing steering gears that were not specifically designed for that purpose.

When roll stabilisation is used, the steering gear and rudder perform two functions, course-keeping and roll damping, and it is important to consider an integrate control system including both autopilot and rudder roll stabilisation. In order to optimize both functions, some rudder roll stabilisation systems have an integrated autopilot function.

4.3 Numerical Predictions of Ship Motions in Waves:

Predictions of ship motions in waves have been traditionally used by navies for direct engineering design applications, such as calculations of structural responses and speed reduction or added fuel consumption in waves. More recently, ship motion predictions are also being used on many safety and operational effectiveness assessments. This recent trend can be explained a more generalised recognition of the extended influence of ship motions in the effectiveness of the crew and onboard sensor systems. Therefore, validation with experimental data is an essential part of development of any code of ship motion prediction. Both full-scale trials and scaled model tests are recommended for validation of numerical ship motion predictions. Furthermore, full-scale trials are considered essential because they might reveal phenomena that can be absent in model tests, either due to oversight or scaling effects.
Since a ship's bare hull is lightly damped in roll motion, passive and/or active devices are commonly used to increase it. Naval vessels, such as frigates, have made a generalized use of bilge keels and active fin devices appended to a ship's hull. This combination of roll suppression devices has demonstrated enhanced performance at different speeds.

With the essential knowledge of natural frequency and damping coefficients at different speeds, and some additional knowledge of ships particulars, it is possible to predict unstabilised roll responses for any specified seaway. Furthermore, with knowledge of generated lift force of fins and control system gains, stabilized roll motion characteristics can be predicted.

The equations of motion for the vessel in a seaway have been augmented to include the forces generated by the control surfaces and the numerical predictions can then be compared with the results of full scale experiments.

4.3.1 Linear Theory - Equations of Motion and Strip Theory

Newton's law of motion governs the vessel dynamics $[M]\{\ddot{\xi}\}=[F]$, where the excitation forces $[F]$ and the motions $\{\xi\}$ can be conveniently represented on a right handed Cartesian coordinate system, $X=(x,y,z)$, (see Fig. 4.1) fixed with respect to the mean position of the ship and origin in the plane of the undisturbed free surface. The translatory displacements in the x, y, and z directions are respectively surge ξ_1, sway ξ_2, and heave ξ_3, while the rotational displacements about the same axis are respectively roll ξ_4, pitch ξ_5, and yaw ξ_6.
In general, the excitation forces acting on the ship's hull consist of control forces from rudder, and active fins, environmental forces from wind and waves.

In these studies, potential flow in infinite depth is assumed (although viscous effects are may be considered when roll damping is calculated).

Usually, only the forces due to wave excitation and reaction forces due to wave-induced ship motions are taken into account. Wave excitation forces consist of incident wave forces (or Froude-Krylov forces), and diffraction forces, and reaction forces include restoring and radiation forces due to motion. Surge oscillatory motion, is assumed to be negligible.

In an approximate way, radiation and wave excitation forces are then calculated at the equilibrium waterline using a standard strip theory, where the two-dimensional frequency-dependent coefficients of added mass and damping, and the sectional diffraction forces are computed by the Frank´s close fit method.

Under the assumptions presented previously, all hydrodynamic forces are linear and when these are combined with the mass forces, five linear coupled differential equations of motion are obtained, given by:

$$\sum_{j=2}^{6}\left\{\left(M_{kj}+A_{kj}\right)\ddot{\xi}_j+B_{kj}\dot{\xi}_j+C_{kj}\xi_j\right\}=F_k \qquad , \quad k,j=2,...,6 \tag{2}$$

, with the subscripts, k,j indicating forces in the k-direction due to motions in the j-mode. M_{kj} are the components of the mass matrix for the ship, A_{kj} and B_{kj} are the added mass and damping coefficients, C_{kj} are the hydrostatic restoring coefficients and F_k are the complex amplitudes of the exciting forces.

The harmonic j-th response of the vessel, ξ_j, will be proportional to the amplitude of the exciting force, at the same frequency but with phase shift, θ_j, and is then given by:

$$\xi_j(t)=\xi_j^a\cos\left(\omega_e t+\theta_j\right),\ j=1,...,6 \tag{3}$$

If the ship travels at a speed U making an angle β with the direction of incoming waves (see Fig. 4.1), she will encounter regular wave crests with a frequency of encounter, given by:

$$\omega_e=\omega-kU\cos\beta \tag{4}$$

The encountered free surface is given by:

$$\zeta_w=\zeta_w^a\cos k\left[x\cos\beta+y\sin\beta-\left(c-U\cos\beta\right)t\right] \tag{5}$$

In irregular seas the encountered wave profile is given by:

$$\zeta_w=\sum_{n=1}^{N}\zeta_{w_n}^a\cos\left[\begin{array}{l}\dfrac{\omega_n^{\ 2}}{g}\left(x\cos\beta+y\sin\beta\right)\\ -\left(\omega_n-\dfrac{\omega_n^{\ 2}}{g}U\cos\beta\right)t+\varepsilon_n\end{array}\right] \tag{6}$$

, where N is the number of component waves, ω_n the circular frequency, ε_n the random phase angle and $\zeta_{w_n}^a$ the amplitude of the n-th component waves, which are determined from the wave spectrum $S_W(\omega)$.
Because the system is linear, the relationship between the wave spectrum and that of the j-th response and is given by:

$$S_{\xi_j}(\omega)=\left|H_j(\omega)\right|^2\ S_W(\omega) \tag{7}$$

, where $H_j(\omega)$ is the transfer function from wave elevation to the j-th mode.

The variance of a record is given by the zero order moment of each ordinate, as follows:

$$m_{0_j}=\int_0^{\omega}S_{\xi_j}(\omega)\ d\omega_e \tag{8}$$

4.3.2 *Passive roll stabilisation systems of "U" type tanks*

These passive stabilizers are designed to provide good dynamic coupling between the stabiliser and ship, by proper selection of the natural frequency.

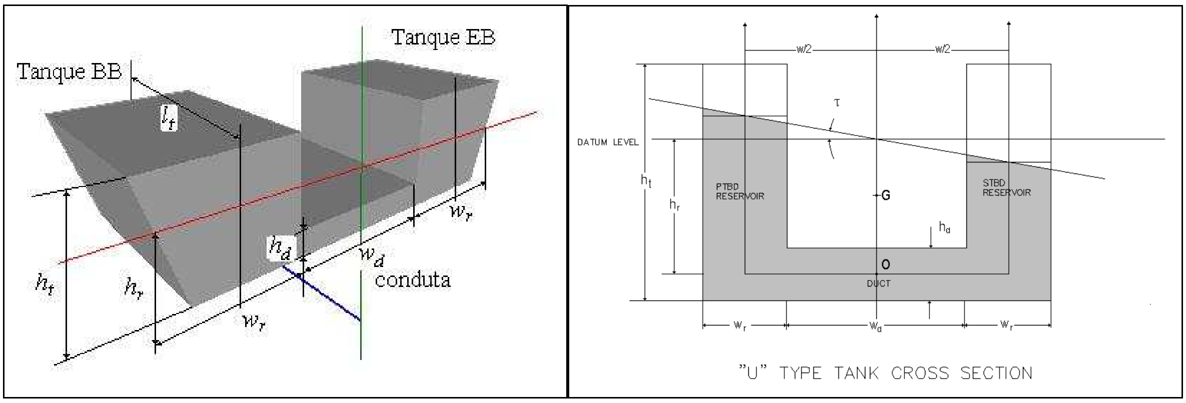


Figure 4.22 - Definition of the passive "U" type tank dimensions.

The tank natural frequency (ω_t) is determined by the values of tank dimensions (shown in Figure 4.21), and the mass of working fluid (m_t), given by:

$$\omega_t = \sqrt{\frac{c_{\tau\tau}}{a_{\tau\tau}}} = \sqrt{\frac{2gh_d}{w_r w + 2h_r h_d}} \qquad (4.9)$$

$$m_t = \rho_t x_t \left(wh_d + 2h_r w_r \right) \qquad (4.10)$$

Where $c_{\tau\tau}$ and $a_{\tau\tau}$ denote the displacement and the acceleration coefficients of the applied roll moment necessary to sustain a steady tank angle (τ) of the fluid.

In the numerical model adopted, the tank angle may be regarded as an additional degree-of-freedom (DOF) in the equations of motion for the ship. Its effects are taken into account by including additional terms into the lateral plane equations of motion.

Introducing the effect of a passive "U" type tank stabilizer into (4.1), noticing that only $k = 2$, 4, 6 directions are affected and that these horizontal plane motions may be decoupled from the vertical plane ones, the following four DOF - sway, roll, yaw and motion of the fluid in the tank – equations are obtained:

$$
\begin{bmatrix} M + A_{22} & A_{24} & A_{26} & a_{\tau 2} \\ A_{42} & M_{44} + A_{44} & A_{46} & -a_{\tau 4} \\ A_{62} & A_{64} & M_{66} + A_{66} & a_{\tau 6} \\ a_{\tau 2} & a_{\tau 4} & a_{\tau 6} & a_{\tau\tau} \end{bmatrix} \begin{bmatrix} \ddot{\xi}_2 \\ \ddot{\xi}_4 \\ \ddot{\xi}_6 \\ \ddot{\tau} \end{bmatrix} +
$$

$$
+ \begin{bmatrix} B_{22} & B_{24} & B_{26} & 0 \\ B_{42} & B_{44} & B_{46} & 0 \\ B_{62} & B_{64} & B_{66} & 0 \\ 0 & 0 & 0 & b_{\tau\tau} \end{bmatrix} \begin{bmatrix} \dot{\xi}_2 \\ \dot{\xi}_4 \\ \dot{\xi}_6 \\ \dot{\tau} \end{bmatrix} + \begin{bmatrix} 0 & 0 & C_{26} & 0 \\ 0 & C_{44} & C_{46} & -c_{\tau 4} \\ 0 & 0 & C_{66} & 0 \\ 0 & 0 & 0 & c_{\tau\tau} \end{bmatrix} \begin{bmatrix} \xi_2 \\ \xi_4 \\ \xi_6 \\ \tau \end{bmatrix} = \quad (4.11)
$$

$$
= \begin{bmatrix} F^a_{W2} \sin(\omega_e t + \theta_2) \\ F^a_{W4} \sin(\omega_e t + \theta_4) \\ F^a_{W6} \sin(\omega_e t + \theta_6) \\ 0 \end{bmatrix}
$$

Notice should be given to the fact that the tank stabilizing moment ($a_{\tau 4}\ddot{\tau} + c_{\tau 4}\tau$) has to be always subtracted to left-hand-side of equation.

4.3.3 Active fins and rudder(s) roll stabilisation systems

Active fin stabilisers are fin-type control surfaces, which are installed onto the ship's hull in a position just above the turn of bilge, near amidships, port (PTBD) and starboard (STBD). 6 shows the forces and moments applied to the ship, by a pair of fins, at an angle of incidence α, to the seawater flow. Each fin develops a lift force given by:

$$
F_L = C_L \frac{1}{2} \rho \mathrm{U}^2 \mathrm{A_F} = \left(\frac{\mathrm{dC_L}}{\mathrm{d}\alpha} \right)^{3D} \alpha \frac{1}{2} \rho_{SW} \mathrm{U}^2 \mathrm{A_F} \quad (9)
$$

, where, according to foils theory, the bi-dimensional lift coefficient at the origin, given by:

$$
\left(\mathrm{C_L} \right)^{2D} = 2\pi \sin\left(\alpha + \frac{2h}{c} \right) \quad (10)
$$

, which should then be corrected to take account of three-dimensional flow effects, by making use of the following equation:

$$
\left(\frac{\mathrm{dC_L}}{\mathrm{d}\alpha} \right)^{3D} = \left(\frac{\mathrm{dC_L}}{\mathrm{d}\alpha} \right)^{2D} \left(\frac{AR}{2 + AR} \right) \quad (11)
$$

, where h/c is the maximum camber, expressed as a fraction of the chord and AR is the dimensionless slenderness of the fin, or the aspect ratio.

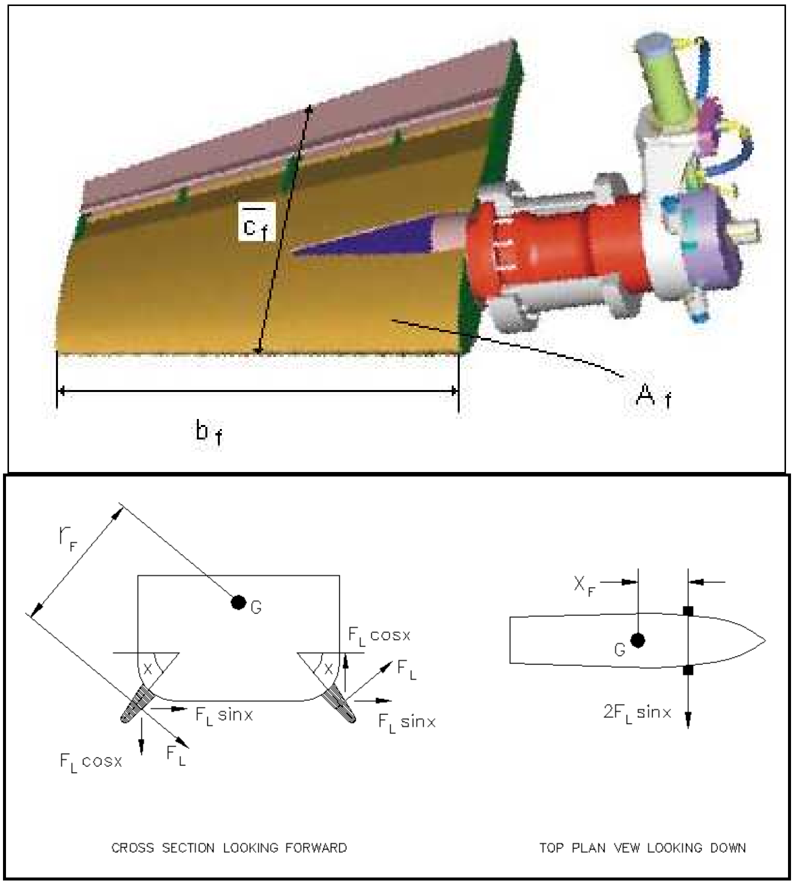


Figure 4.23 – Definition of the fin main dimensions (top) and fins position and orientation angles over the ship's hull (bottom).

Thus, the fins exert a roll moment $F_{F4} = 2F_L r_F$ about the centre of gravity, where the fin lever r_F is measured from the axis, through the centre of gravity, to the lift vector (assumed to be acting at the centre of pressure of each fin, taken to be placed at a distance of one third of the span of the fin measured from its root).
The vertical components of the two lift forces cancel-out each other. However, the horizontal components add and yield a sway force given by:

$$F_{F2} = -2F_L \sin\chi \tag{12}$$

If the fins are mounted x_F meters forward of the centre of gravity, this horizontal component will exert a yaw moment given by:

$$F_{F6} = -2F_L x_F \sin\chi \tag{13}$$

Introducing the effect of stabilisers into equation (4.1), noticing that only $k = 2, 4, 6$ directions are affected and that these horizontal plane motions may be decoupled from the vertical plane ones, we obtain the following three Degrees-Of-Freedom (DOF) equations:

$$\begin{bmatrix} M + A_{22} & A_{24} & A_{26} \\ A_{42} & M_{44} + A_{44} & A_{46} \\ A_{62} & A_{64} & M_{66} + A_{66} \end{bmatrix} \begin{bmatrix} \ddot{\xi}_2 \\ \ddot{\xi}_4 \\ \ddot{\xi}_6 \end{bmatrix} +$$

$$+ \begin{bmatrix} B_{22} & B_{24} & B_{26} \\ B_{42} & B_{44} & B_{46} \\ B_{62} & B_{64} & B_{66} \end{bmatrix} \begin{bmatrix} \dot{\xi}_2 \\ \dot{\xi}_4 \\ \dot{\xi}_6 \end{bmatrix} = \qquad (14)$$

$$= \begin{bmatrix} F_{W2}^{a} \sin(\omega_e t + \theta_2) - \sum EF_L \sin\chi \\ F_{W4}^{a} \sin(\omega_e t + \theta_4) + \sum EF_L r_F \\ F_{W6}^{a} \sin(\omega_e t + \theta_6) - \sum EF_L x_F \sin\chi \end{bmatrix}$$

, where the summations refer to the number of fins fitted to the ship and E is an effectiveness factor, usually assuming a value less than *1.0* because of various hydrodynamic losses, and defined as the relation between the effective lift of fin and the nominal lift of fin.
If a simple 1 DOF uncoupled rolling model is adopted for controller design, then the stabilised roll equation of motion is modelled as a linear second-order differential equation, given by:

$$(M_{44} + A_{44})\ddot{\xi}_4 + B_{44}\dot{\xi}_4 + C_{44}\xi_4 = F_{W4} + F_{F4} \qquad (15)$$

, and with a Proportional, Integral and Derivative (PID) controller, the stabilizer displacement is proportional to the sum of the roll, displacement, velocity and acceleration, given by:

$$\alpha_D = K_1\xi_4 + K_2\xi_4 + K_3\xi_4 \qquad (16)$$

, where: $K_1 =$ roll angle sensitivity, $K_2 =$ roll velocity sensitivity, and $K_3 =$ roll acceleration sensitivity.
Finally, in respect to rudder(s) roll stabilisation systems the same equations can be applied providing geometrical differences between position of rudder(s) and fins are taken into account.

4.4 Case Study of a Roll Stabilisation System Selection

a) Main Characteristics

The patrol vessel is a slender twin- controllable-pitch propeller hull with deep-V forms in the fore body, and a deadrise at the aft body. The vessel has a main section with rising floor and tumbled sides. Figure 4.24 presents the bodylines and a 3D view of the ship's hull, while Table 4.1 presents the ship's main particulars.

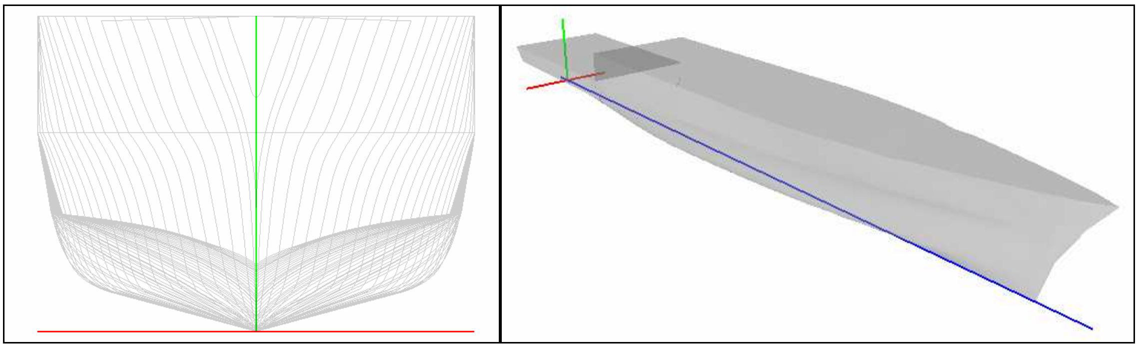


Figure 4.24 - Body plan and 3D view of the hull of the patrol vessel.

Table 4.1 Main particulars of the patrol vessel.

Length betw. perp., L_{pp} (m)	57.5
Beam at waterline, B (m)	8.5
Draught, T (m)	2.70
Displacement, Δ (ton)	674
Long. pos. of CG (m)	-1.58
Vert. pos. of CG (m)	1.44
Block coefficient, C_b	0.50
Roll gyr. Radius (K_{xx}/B)	0.36
Metacentric height (m)	0.59
Natural roll period (sec)	9.6
Stabiliser design speed (kts)	15.4

The following solutions have been considered in the assessment:

- A pair of triangular shape bilge keels, 0.5 metres wide with its trailing edge located 25.11 metres in front of the aft perpendicular and 0.73 metres above the baseline. The length of the bilge keels is approximately 9 metres;

- A passive "U" type tank stabiliser has been also selected for these vessels due to low speed requirements for a large number of launch-and-recovery operations; and the chance of swing utilisation between the roll stabilisation system and the stern ballast system. After review of available space and deadweight, a "U" type tank with 3 metres in length has been selected to be installed around the stern dock, located aft of the aft machinery space. As shown in table 2, to achieve a natural frequency of the passive tank slightly higher than vessel's natural roll frequency of 0.65 (rad/sec), a connecting duct between the two reservoirs with 0.4 metres height has been selected.

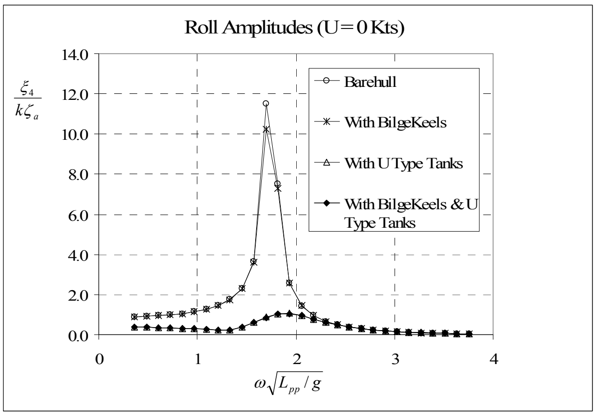
Width of each reservoir (w_r)	*2.54 [m]*
Width of the connecting duct (w_d)	*3.5 [m]*
Height of reservoirs (h_r)	*1.5 [m]*
Height of the connecting duct (h_d)	*0.4 [m]*
Height of the tank (h_t)	*3.2 [m]*
Length of the tank (x_t)	*3.0 [m]*
Working fluid mass (m_t)	*28 [ton]*
Tank natural frequency (ω_t)	*0.68 [rad/sec]*

Table 4.2 Characteristics of the passive "U" type tank.

Notice should be given to the fact that to allow swing of utilisation and to prevent saturation of these passive “U” type tanks, at high frequencies of encounter with waves, the 28 tonnes of working fluid (saltwater) can easily be pumped outboard by two ballast pumps of 300 (m^3/hr), each;

b) Seakeeping Performance Calculations

The methodology presented in section 4.3 is applied to calculate the response amplitude operators shown below.



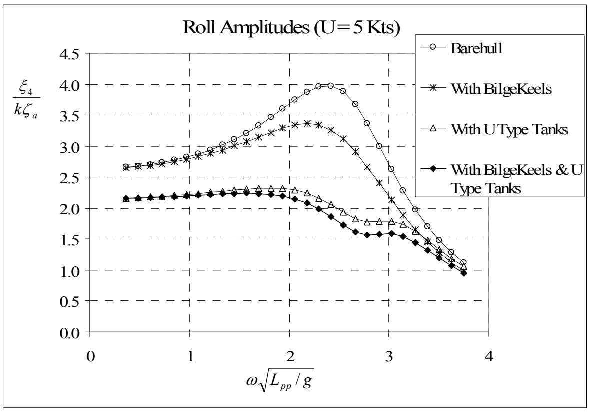


Figure 4.25 - Transfer function amplitudes of unstabilised and stabilised roll responses in beam waves for the patrol vessel at speeds of 0 and 5 knots.

Based on vessel’s operating profile and the transfer function showed above, indexes of operability per year on the Atlantic coast off the Portugal (design point) were determined. Next, based on acquisition and operational direct costs, an economical analysis of each configuration was conducted, and it was concluded that a combination of fins and tanks should be adopted.

4.5 Description of Ships and Roll Stabilisation Sea Trials

a) Ship’s Main Characteristics

General characteristics of the “Vasco da Gama” vessel, during sea trials, are shown in Table 4.3 and a general profile view of a vessel of this class is shown in Figure 4.26.

Figure 4.26 – General view of the frigates “Vasco da Gama” class.

To ensure roll stabilisation in the slow speed range, the ship's hull is also fitted with bilge keels. These hull appendages have a span b_{BK} of *1000 [mm]* and a length L_{BK} of *14.4 [m]*, mounted in the turn of bilge in STBD and PTBD sides, just before the anti-roll fins, to prevent the fins' effectiveness degradation. If a bilge keel were mounted a short distance abaft a stabiliser fin, the former would experience a downwash and develop lift which opposes the fin lift but this is not the present case.

Table 4.3. General characteristics of the "Vasco da Gama" vessel during sea trials.

Length Between Perpendiculars (L_{pp})	*109.0 [m]*
Displacement at Current Water Line (Δ)	*3186.7[ton]*
Beam, Maximum at CWL (B)	*13.8 [m]*
Draft, amidships at CWL (T)	*4.1 [m]*
Transverse Metacentric Height ($\overline{GM}_t$)	*1.07 [m]*
Stabiliser Design Speed (U)	*18 [Kts]*
Natural Roll Period (T_n)	*12 [sec]*

Table 4.4. Characteristics of the active anti-roll fins, type Simplex Compact FK51.

Profile Area ($A_F = \int cdb = \bar{c} \times b_F$)	*5.58 [m2]*
Geometric Aspect Ratio ($AR = \dfrac{b_F}{\bar{c}}$)	*0.60*
Mean Chord ($\bar{c} = \dfrac{c_R + c_T}{2}$)	*3.05 [m]*
Fin Span (b_F)	*1.83 [m]*
Fin Lever (r_F)	*8.86 [m]*
3D Lift Coefficient Slope $\left(\dfrac{\mathrm{dC_L}}{\mathrm{d}\alpha}\right)^{3D}$	*0.036 [1/deg]*
Section Shape	*NACA 0015*
Overall Effectiveness of Stabiliser (E)	*0.96*

b) Active Fins Characteristics

The active anti-roll fins of the "Vasco da Gama" class of the type Simplex Compact FK51 have no camber, a trapezoidal planform area and the characteristics shown in Table 4.4.

The active fin stabilization system incorporates two fins, mounted into the ship's hull, at a longitudinal distance from the forward perpendicular to the fin stock X_{FP} of *58.35* meters. As shown in Figure 4.27, these fins which look so small in comparison to the rest of the hull, when viewed in dry dock, can have a significant effect, when deliberately controlled to induce roll in a calm sea or to force the vessel to return to a stabilized datum in waves.

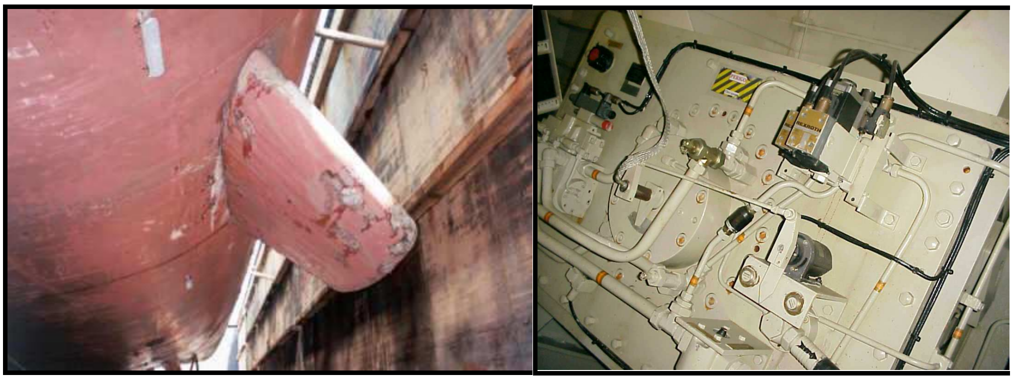


Figure 4.27 - Left: PTBD fin as fitted into ship's hull (viewed in dry dock); Right: Details of the mechanical connection between fin stock top-end and fin-angle transmitter, and fin stock servomechanism valves.

Description of Forced Rolling Sea Trials:
The forced and transient rolling trials onboard the Frigate F330 - "Vasco da Gama", were carried out on June 22, 2001, with the ship sailing on the Atlantic coast, off the Lisbon.
The procedures of the trials were determined, taking into consideration the recommendations given in literature.
For the entire duration of the trials, the ship was run in calm water so that roll moment from waves was practically negligible. In order to minimize the effect of small waves present on the seaway, the ship was run in head seas. Rudder motions also have influence on the rolling motions of the ship, therefore, the autopilot was switched off and the helmsman kept the wheel amidships during all the runs.

The stabilizer controller was isolated from the system by breaking the circuit at point D in Figure 4.28.

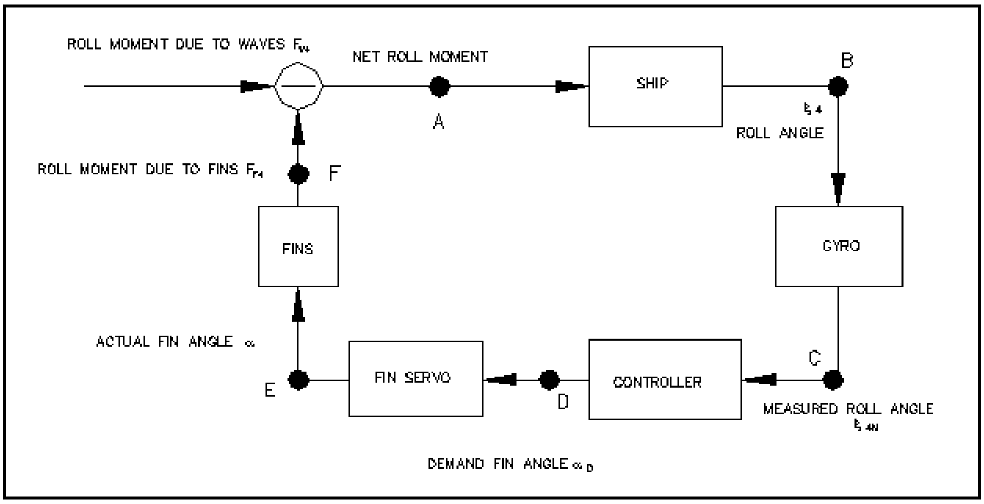


Figure 4.28 - Block diagram of the ship fitted with active roll stabiliser fins (simplified).

The fin servomechanisms were, instead, driven by a sinusoidal demand signal equivalent to *21 [deg]* fin amplitude, at several different selected frequencies. The performed trial runs are presented in *Table 4*.5.

Table 4.5. Sea trials runs onboard F330

	Swing Period [sec]				
Speed [kts]	*1.3*	*2.6*	*5.3*	*10.6*	*21.2*
10			✓		
15			✓		
18	✓	✓	✓	✓	✓
30			✓		

In order to obtain a free-decay roll curve (in the transient regime), at the end of each individual trial, the controller was rapidly switched off and the acquisition data sequence was terminated, only when the ship assumed her mean upright position.

<u>Ship Heave, Roll And Pitch Transfer Function of the Unstabilised Ship Motions:</u>

Heave, roll and pitch RAOs have been computed, using the linear theory presented in section 4.3.1, both numerically and from experiments.
In Figures 4.29 to 4.31, the theoretical RAOs are compared with those estimated from the experiments at the advance speed of *18* knots and for five headings relative to the predominant wave direction. The results are presented in non dimensional form as function of non-dimensional wave frequency, given by: $\hat{\omega} = \omega_0 \sqrt{L_{PP}/g}$. Translational motions are non-dimensionalised by dividing displacement amplitudes, by the wave amplitude and rotational motions displacement amplitudes are divided by the wave slope $k\zeta_a$.

Figure 4.29 shows the good agreement between theoretical and experimental heave responses, in head, quartering and beam seas, at the high frequencies.

On Figure 4.30, a dominant roll resonant frequency can be easily detected in the beam seas spectrum, around *0.8 [rad/sec]* and the peak amplitude value compares well with theoretical predictions when taking into account a calibrated passive roll damping coefficient obtained from the full scale free-decay curve. However, considering sea trials have been conducted on sister ships with identical loading conditions, the experimental value is below the previously identified natural roll period, expected to be in the range *10* to *12* seconds. Also, it appears from this figure that trial results are more accurate when confined to frequencies higher than the natural roll frequency. This deviation might be explained by the effect of the spreading of wave energy at lower wave frequencies since unidirectional waves never occur at sea. For frequencies higher than the natural rolling frequency, the wave system has little influence on the rolling motion, and therefore it is possible to analyze the ship trials' results in this frequency range, as though the waves were unidirectional.

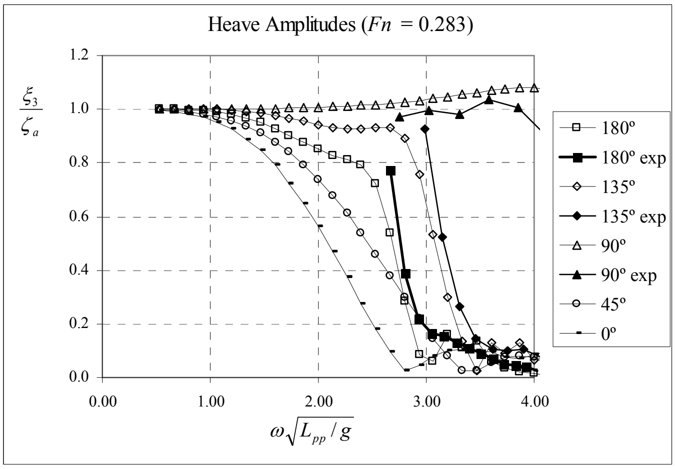


Fig. 4.28 - Comparison of experimental and theoretical heave response amplitudes.

In Figure 4.31, it can be seen that theoretical pitch RAO compares well with trial results in quartering and following seas, at the high frequencies. Although there are no trial results to compare with theory at lower frequencies, it is thought that these would compare well, since linear strip theory gives accurate predictions for pitch motion, in this frequency range.

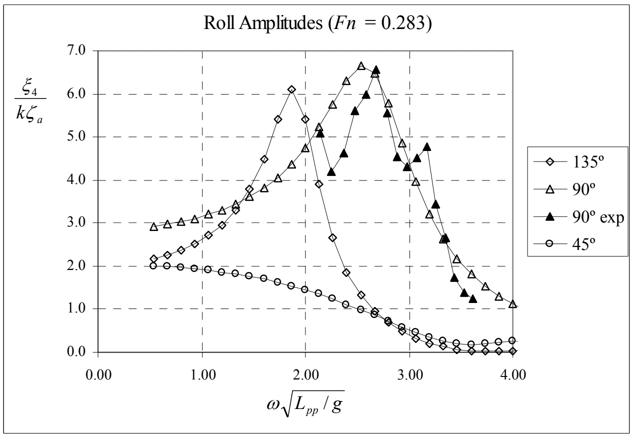


Fig. 4.29. Comparison of experimental and theoretical roll response amplitudes.

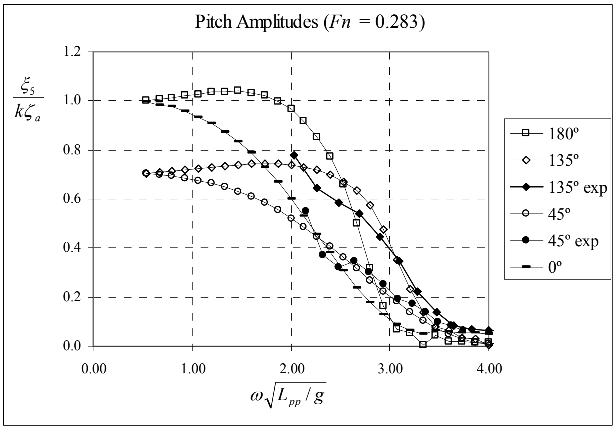


Fig. 4.30. Comparison of experimental and theoretical pitch response amplitudes.

It is considered that theory gives a satisfactory representation of the analyzed motions in moderate sea states. These results, taken in conjunction with forced rolling simulations, give a reasonable confidence in the use of linear theory for the prediction of ship motions where passive roll damping can be easily estimated by a calibrated Ikeda's method.

Stabilised Ship Motions:

From forced oscillation and inclining tests it was found that the coefficients in the roll equation of motion are:

$$\begin{aligned} M_{44} + A_{44} &= 1.18 \times 10^8 \left[Kg.m^2\right] \\ B_{44} &= 2.27 \times 10^7 \left[N.m.s\right] \\ C_{44} &= 3.35 \times 10^7 \left[N.m\right] \end{aligned} \quad (17)$$

In order to have a complete description of the stabilized roll dynamics model, it is necessary to know the generated lift force and moment, on each fin. For reasons associated with coupling to roll-sway modes, it is not advisable to calculate the total fin induced roll moment F_{F4} from a fin static roll angle obtained from sea trials. Hence, the method adopted before consisted in choosing one forced rolling trial run, where both variables, i.e. roll amplitude response and total fin induced roll moment, behaved as pure sinusoidal signals having the same oscillation frequency. Feeding the measured (steady-state) roll amplitude response, and the above calculated damping B_{44} and restoring C_{44} coefficients into equation (15), a total fin roll moment developed at service speed $F_{F4} = 2F_L r_F$ of about *3150 [KN.m]* was developed.

As before, due to operational commitments, stabilised roll trials at a speed of *18* knots and different headings relative to the waves, were not performed. So, comparisons of numerical predictions of ship responses with trial measurements cannot be provided. Nevertheless, for results shown in Figure 4.32, the formulation has been extended to include the effects of an active roll stabilisation system.

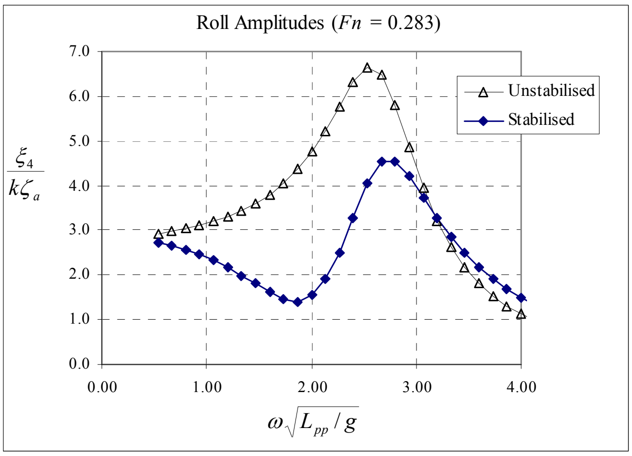


Figure 4.32 - Comparison between stabilised and unstabilised theoretical predictions of roll response amplitudes; active fins controlled by PID.

The comparison is drawn between unstabilised (ξ_{4U}) and stabilized (ξ_{4S}) roll responses, at *18* knots in regular beam seas (assuming an wave slope of 40), for the controller sensitivities presented. These sensitivities allow the fin controller to provide a phase advance which exactly compensates at the natural roll frequency for the phase lags introduced by the other individual components, and their values are $K_1 = 0.0$, $K_2 = 8.75$, and $K_3 = 5.0$.

Figure 4.33 presents the roll spectra calculated from the stabilised and unstabilised transfer functions with the ship advancing with 18 knots in beam seas with a significant wave height of 1.2 metres and a peak period of 7.6 seconds (the same as measured during the trials). Though this is not an optimal solution, one observes a significant reduction of the spectral amplitudes around the roll resonance frequency when the stabilising system is used.

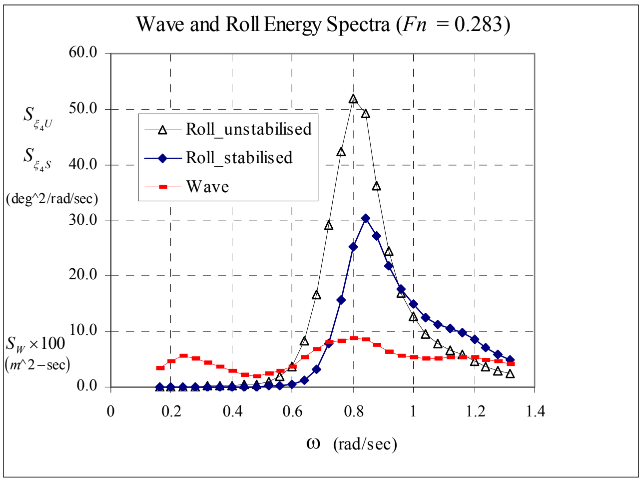


Fig. 4.3. Energy spectra of roll in irregular beam seas at 18 knots.

<u>Conclusions:</u>

Linear theory, based on strip theory method, has been compared with the results of trial measurements on POR Navy frigates of “Vasco da Gama” Class. The comparison shows that the unstabilised behaviour of these ships in moderate sea states, could be adequately represented by theory. These numerical predictions give generally good agreement, because of the slender hull form of the naval frigates. As expected, agreement is better for heave and pitch than for roll.

Finally, the classical theory has been extended to include the effects of an active roll stabilisation system, and a comparison is made between unstabilised and stabilised roll responses, in irregular beam seas, for a basic controller that responds in a linear fashion, to the rolling motion of the ship. As recommendation, it is essential that stabilised roll motion predictions, presented herein, should be validated against experimental data, in the future.