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Roll Motion of a Ship and the Roll Stabilising Effect of Bilge Keels

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The roll motion of a ship on waves is one of the six degrees of freedom and is the most difficult to predict since it deals with a motion similar to a spring-mass damper system, poorly damped by ship generated waves, in addition to the action of waves from the high seas. This problem increased from the second half of the 19th century when sailing propulsion was replaced by steam machines and iron replaced wood, which led to design modifications in ships affecting transversal stability, with the result of an increment in the ship's roll motion. As a consequence, it has been necessary to develop different systems which, on the one hand, increase the natural period of this motion in order to decrease its amplitude and reduce the risks that the wave encounter frequency would resonate with the natural frequency of the ship's motion; and on the other hand, to generate a damping effect to this motion. Bilge keels, passive stabilisers with no moving parts, form the simplest and cheapest element that may be incorporated on a ship to reduce this motion. This paper studies roll motion in general and then analyses bilge keels from different points of view.

KEY WORDS

1. Six degrees of freedom. 2. Ship's roll. 3. Roll damping. 4. Bilge keel.

1. INTRODUCTION TO ROLL MOTION.

Although a ship's motions on the sea have always been difficult to predict, it is very important to make sure that the ship may safely face the situations encountered in bad weather and at the same time successfully carry out its specific tasks. An understanding of a ship's motions on the sea starts with the study of the nature of waves, through which a ship moves. The response motion of a ship underway in response to the excitation force of waves is a complicated phenomenon where interactions between the ship dynamic and hydrodynamic forces take part. A ship floating on a free surface and navigating among waves[1] will experience some motions which are known as the ship's six degrees of freedom (*DOF's*) (see Figure 1).

Three of these *DOF's* are translation motions (*surge*, *heave* and *sway*) and the other three are rotation motions (*roll*, *pitch* and *yaw*)[2]; they can also be differentiated between motions in the horizontal plane (*surge*, *sway* and *yaw*) and oscillating motions in the vertical plane (*heave*, *pitch* and *roll*). The motions in the horizontal plane are unrestored. Therefore, they do not exhibit resonance and their amplitudes in deep water are never greater than the wave amplitude, or in the case of *yaw*, never greater than the wave slope.

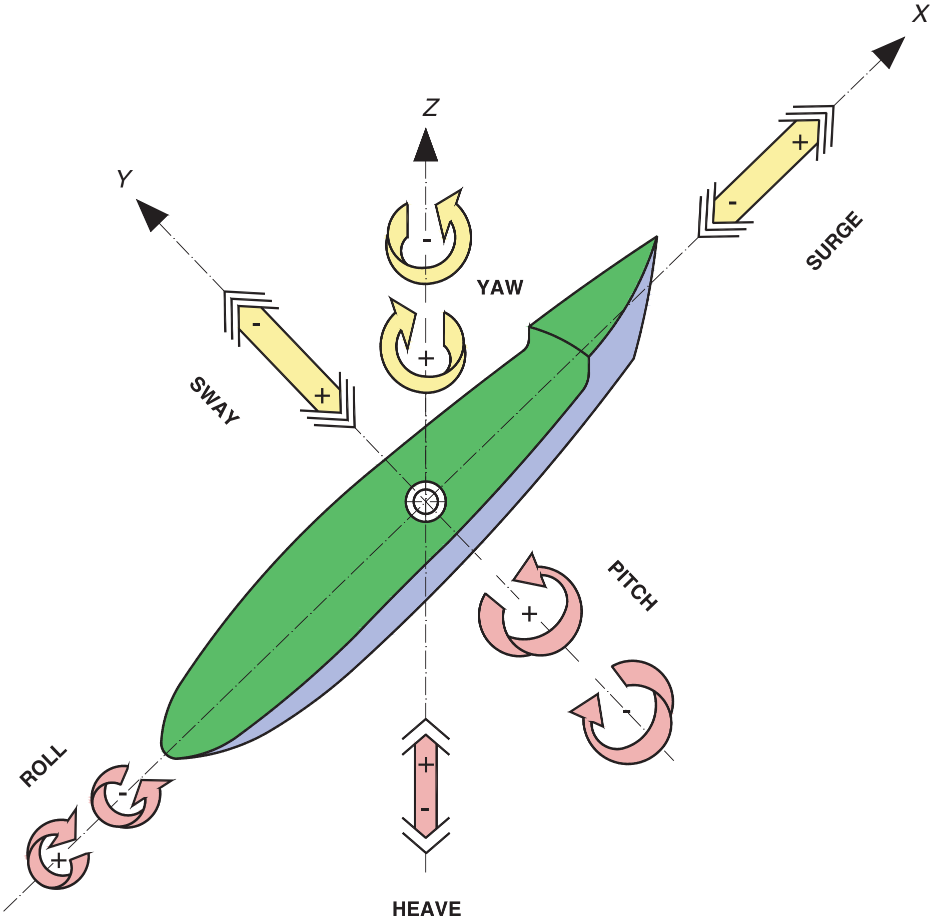

Figure 1. A ship's six DOF's. Drawing: author.

However, when the ship is subjected to motions in the vertical plane, it experiences restoration or restitution forces similar to a damped simple harmonic motion (SHM) and the ship behaves as a spring-mass damper system that can exhibit resonance. Owing to the wave excitation force, a lack of balance is produced derived from the centre of buoyancy $\boldsymbol{B}$ which adopts a new position $\boldsymbol{B}'$ due to the variation in the underwater volume and, as a consequence of that, the ship's weight acting in $\boldsymbol{G}$ and the buoyancy with the centre in $\boldsymbol{B}'$ are out of alignment. As a result, a ship's internal righting moment is generated to re-align $\boldsymbol{B}$ vertically with $\boldsymbol{G}$; however, due to the inertia of the ship's mass, this does not stop in the balanced position, but it exceeds it until a similar symmetric position to the old one is reached[3]. In short, these three oscillating motions in the vertical plane influence the balance of forces between the weight acting in $\boldsymbol{G}$ and the buoyancy acting in $\boldsymbol{B}$.

In Naval Architecture, such movements are studied around three mutually perpendicular axes[4] **x, y, z.** The longitudinal axis of a ship (from bow to stern or from stern to bow) is defined as the x-axis. The rotational motion in the vertical plane around such x-axis is known as roll, with starboard as positive and port as negative

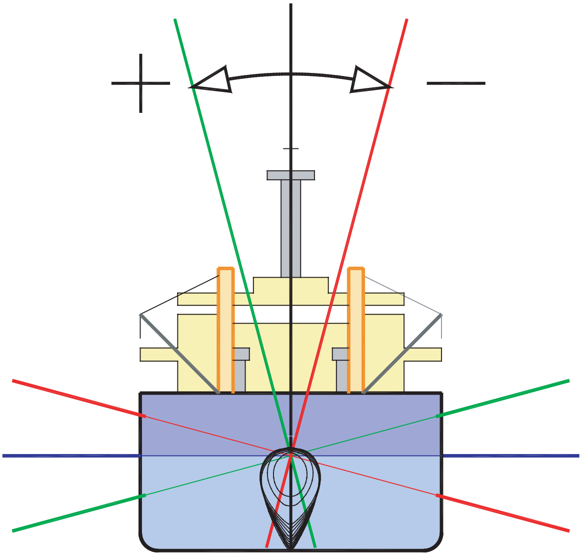

Figure 2. The roll motion around the x-axis. Drawing: author.

(see Figure 2). In addition to the roll, which we have seen is an oscillating motion in the vertical plane, there are two static conditions of inclination around the x-axis: list, originated by an asymmetric distribution of weights and heel, originated by an external force (such as wind or waves) acting on the ship.

2. A SHIP'S NATURAL ROLL PERIOD.

The lack of balance created by the wave system around the ship, which creates the roll motion, generates an internal transversal righting moment in order to recover its balanced situation with G and B acting in the same vertical again (see Figure 3). The magnitude of this righting moment in the ship when the roll takes place depends on the righting arm, GZ, and the weight Δ_s of the ship ($righting\ moment = \Delta_s GZ$), with $GZ = GM_T Sin\ \theta$ and for small angles of inclination ($\theta < 10$), as $Sin\ \theta = \theta$, the righting moment[5] is equal to $\Delta_s GM_T \theta$.

As a consequence, we can see that the generation of a rotational moment of restitution is produced in response to the excitation force of the wave system; and so, we are in the presence of a rotational *SHM* similar to the lineal motion of a system formed by a mass linked to a spring by one end and to a damper system by the other. The spring arises from the buoyancy and gravity forces which tend to restore the balance position, i.e. the righting moment $[\Delta_s GZ]$; the external exciting force is the force of the waves coming into contact with the ship; the damping effect of the movement is caused by the waves generated by the ship and the viscous friction between its hull and the seawater; i.e. it arises from the interaction between the hull and the water and finally, the mass is the rotational mass or *moment of inertia* (*MOI*) of the ship (see Figure 4).

Taking into account that the roll is a rotational motion, the *MOI* (I_x) generated when this motion is produced around the x axis, is not a determined quantity as it

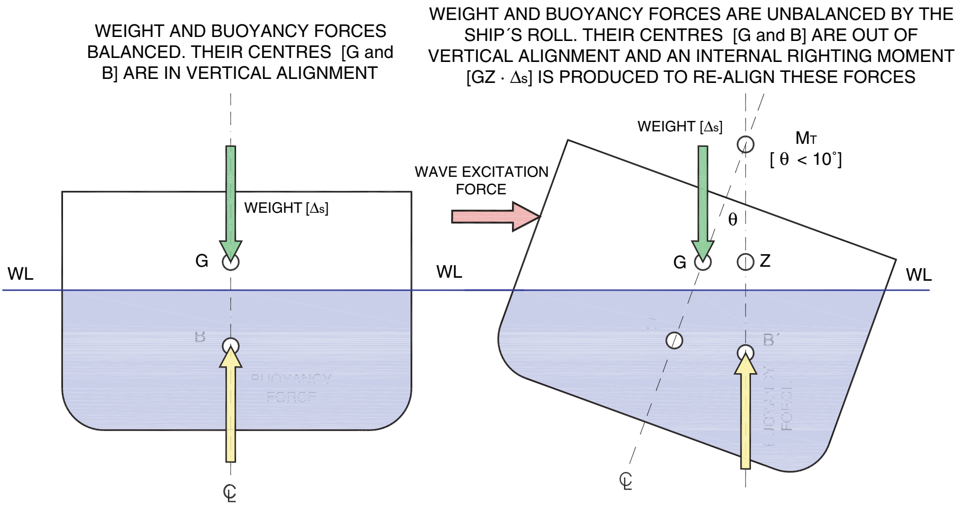

Figure 3. The creation of an internal righting moment by the ship as a response to the wave excitation force which generates a roll motion. Drawing: author.

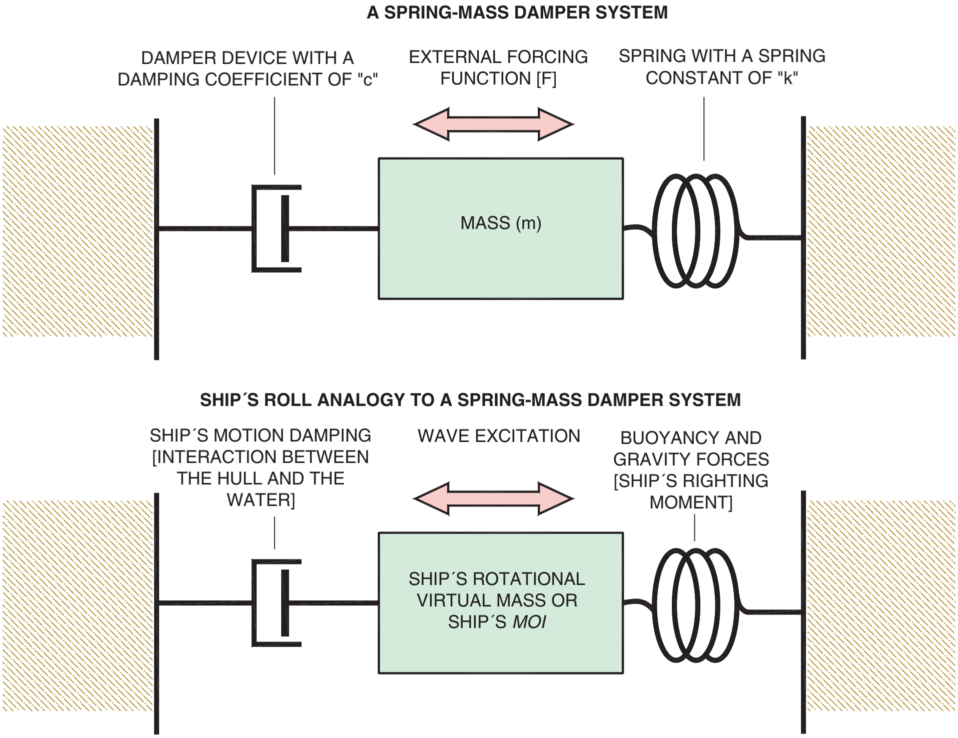

Figure 4. The analogy of the ship's roll motion to a mechanical Spring-Mass Damper System. Drawing: author.

may be the mass or the volume, but its value depends on the position of the rotational axis. The *MOI* is minimum when the rotational axis goes through the centre of the mass. The *MOI* is similar to the inertia[6], except that it is applied to the rotation more than to the linear motion, being able to become a new definition of the mass where the *MOI* is then a rotational mass. Unlike the inertia, the *MOI* also depends on the mass distribution in an object. The further the mass is from the rotational centre, the bigger is the *MOI* in such a way that the *MOI* "I_x" of the mass *m* of an element around a rotational axis *x* with a turning radius[7] *k* is defined by the formula:

$$I_x = k_2\, m. \tag{1}$$

Therefore, Spring constant(k) $\equiv \Delta_s \cdot \mathrm{GM_T}$; with $Mass\ (M) \equiv I_x$

Continuing the analogy with a *SHM* whose theory implies that, knowing the parameters *m* and *k* of the system, the natural frequency itself is produced by the formula $\omega_n = \sqrt{\frac{k}{m}}$, it is possible to predict the natural frequency of the ship's rotational oscillating motion around the *x-axis*, that is to say, the roll:

$$\omega_{roll} = \sqrt{\frac{\Delta_s \cdot GM_T}{I_x}} \tag{2}$$

Considering that for any ship it is difficult and tedious to determine the exact value of radius of gyration of its mass and therefore to calculate the *MOI* value, it is possible to assume that it is a function of the beam[8], in such a way that it varies directly with it. So, knowing the relation between the natural frequency of the roll (ω_{roll}) and the roll period (T_{roll}) $\left[T_{roll} = \frac{2\pi}{\omega_{roll}}\right]$, formula (2) can be experimentally expressed, referred to the roll period, in the following way:

$$T_{roll} = \frac{C \cdot B}{\sqrt{GM_T}}; \tag{3}$$

where: *B* is the maximum ship's beam (in feet); GM_T is the transverse metacentric height (in feet); *C* is an empiric constant whose value for big ships oscillates between 0·35 and 0·55 (depending on the kind of ship and its cargo situation) when GM_T and *M* are measured in feet. In each case, the value *C* depends on the ship's capability to damp the rolls. When this value is unknown, 0·44 is applied by default as it usually produces good results.

The equation shows that the bigger the GM_T of a ship is, the smaller will be the oscillating period, the bigger the internal response force of the ship will be and the bigger the transverse angular acceleration will be. As with any other rigid body motion, the bigger is the motion acceleration, the bigger will be the damage risks to the crew or to the ship's equipment.

As we know, the $\mathrm{GM_T}$ can be calculated by measuring the slope of the GZ curve in the origin. For that reason, a ship with a GZ curve with much slope[9], shows that it has a large $\mathrm{GM_T}$, i.e. a *stiff ship* and the ship whose GZ curve has a little slope clearly shows that its $\mathrm{GM_T}$ is small, i.e. a *tender ship* (see Figure 5). Therefore, in the first case, the ship has very violent roll motions, as it is the typical case of ships with very little length to beam ratio, while in the case that the ship has a small $\mathrm{GM_T}$, the roll motions will be much slower. This indicates that the ideal value of a ship's $\mathrm{GM_T}$ constitutes a compromise between a good seakeeping performance (a small $\mathrm{GM_T}$)

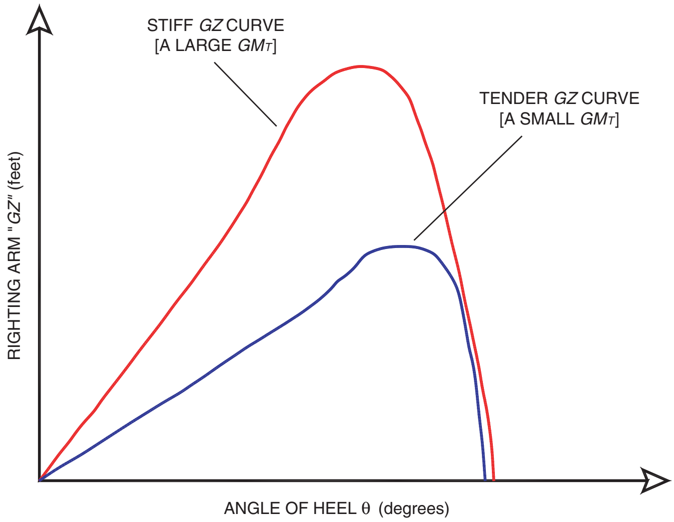

Figure 5. Comparison between two GZ curves of static stability corresponding to two ships with different responses. Drawing: author.

and a good stability (a large GM_T)[10]. Unlike the other two vertical motions *heave* and *pitch*[11], roll comparatively experiences very small damping effects.

3. THE WAVE ENCOUNTER PERIOD.

Having examined the analogies of the roll motion and a spring-mass damper system, we can see that the motion created by the excitation force (in this case, the waves due to the pressure variation on the hull) depends on the magnitude of such force and of its frequency. However, except when the ship is at zero speed, the wave excitation frequency does not just depend on the wave characteristics, amplitude and frequency ω_w, but also on the ship's speed and course or encounter angle. For that reason, the relevant parameter to consider is the wave encounter frequency ω_e, which takes into account the relative speed of the ship and the wave. A complete study of this subject goes further than the content of this article[12] and in this section we are only going to see, with the help of Figure 6, how to calculate the wave encounter period in a regular wave system and which methods the ship's master needs to modify if the resonance phenomenon is produced in the roll motion.

Considering that T_w is the real wave period; λ is the wave length, and; C_w is the wave velocity (more commonly known as *celerity*) which in the figure falls upon the starboard bow with an angle α; from Figure 6 it is clear that:

$$\lambda_e = \frac{\lambda}{Cos\,\alpha};\ V'_w = \frac{C_w}{Cos\,\alpha};\ Relative\ wave\ speed = V'_w + V_s = \frac{C_w}{Cos\,\alpha} + V_s. \tag{4}$$

Since for any wave, its period comes from the formula $T = \frac{\lambda}{C_w}$, the wave encounter period (time that the wave crest n° 2 takes to make the distance A-B) will result from the following equation:

$$T_e = \frac{\lambda}{Cos\,\alpha\left[\frac{C_w}{Cos\,\alpha} + V_s\right]} = \frac{\lambda}{C_w + V_s \cdot Cos\,\alpha} \tag{5}$$

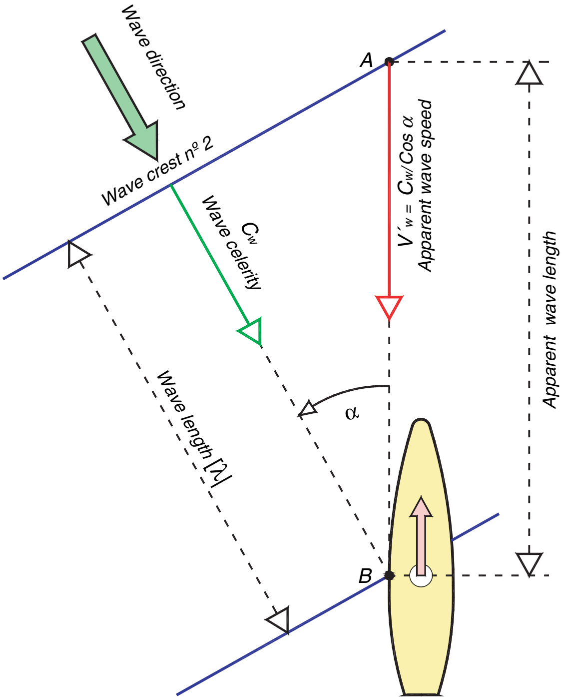

Figure 6. Diagram of vectors for the calculation of the wave encounter period. Drawing: author.

In Figure 6, α shows the direction where the waves come from. When the ship is sailing in bow or head seas, since $\alpha < 90^\circ$, its *Cosine* is positive and therefore in this case, the wave encounter period is smaller than its natural period [$T_e < T_w$]. On the other hand, with ships sailing in following and quartering seas, since $\alpha > 90^\circ$, its *Cosine* is negative and as a consequence the wave encounter period is bigger than its natural period [$T_e > T_w$]. In beam seas, there is no change and both [$T_e = T_w$] are the same. However, the wave encounter length [λ_e] of a wave that obliquely falls upon the ship's centreline is always bigger either on head seas or on following seas. From this formula, we can clearly see that the ship's master may change the course and/or the speed of the ship in order to modify the wave encounter period; and considering that at sea there is usually little scope for altering the ship's natural roll period, the possibility of modifying the wave encounter period becomes a very valuable tool in his hands when the resonance in the roll motion is produced. This will be analysed in section 4.

4. THE PROBLEM OF RESONANCE IN THE ROLL MOTION.

A *SHM* system[13] will experience maximum amplitude oscillations when the excitation force frequency[14] is equal to the natural frequency of the system. This situation is called resonance and it is important to make sure that this condition is not produced with the object of reducing the amplitude of any rigid body motion. This may be applied to a ship whose roll natural frequency ω_{roll}, as we have seen, may be estimated. With the object of reducing the ship's roll, it is important that this

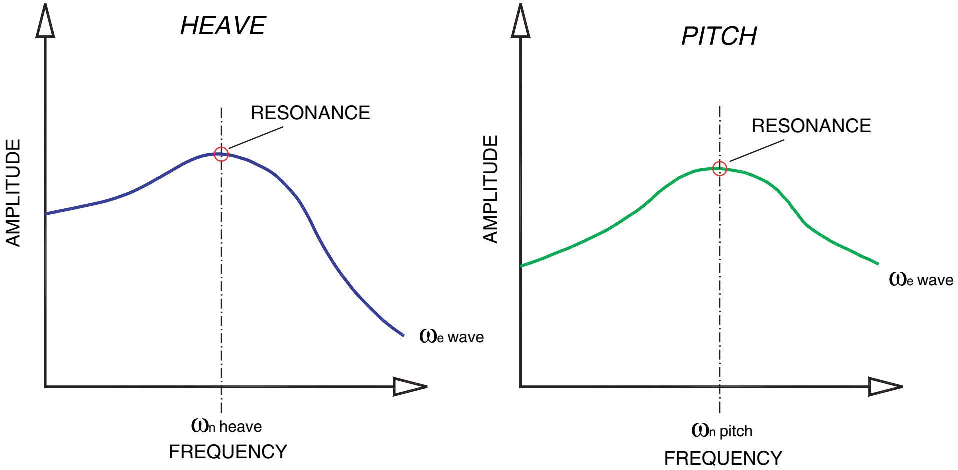

Figure 7. The heave and pitch amplitude of motions versus the respective wave encounter frequency [ω_{heave} and ω_{pitch}] taking into account the natural frequency [ω_{n}] of the ship's motion. In both cases, we can see how their amplitude is not very sensitive to the wave excitation force as they are strongly damped motions. Therefore, the resonance phenomenon becomes less awkward. Drawing: author

natural frequency does not coincide with the wave excitation frequency called wave encounter frequency ω_{e}.

Fortunately, the *SHM*'s of the *pitch* and *heave* motions are very damped. Although the natural frequency of their respective motions comes in resonance with the wave encounter frequency, this does not imply a significant increment of their own amplitude as they do not come into resonance suddenly due to the fact that the energy is quickly dissipated. However, the small damping which the ship experiences when there is a roll makes it very likely that the natural frequency of this motion ω_{roll} coincides with the wave encounter frequency ω_{e}, in which case the amplitude of motions is significantly increased[15] (see Figures 7 and 8 where the different effects of the resonance in the *heave* and *pitch* motions can be compared to the *roll* motion). Resonance may be produced from all motions. However, it is more probable that extreme motions of great amplitude are produced with the *roll* than with the *pitch* and *heave*. As a consequence, due to the noticeable increment of the amplitude of this motion if resonance is produced[16], different anti-rolling devices are used with the aim of reducing the roll (bilge keels, stabilizer fins, anti-roll tanks, etc are typical examples). The following sections study the effect of bilge keels.

5. BILGE KEELS AS ROLL PASSIVE STABILISERS.

5.1. *General considerations.* Roll passive stabilizers do not need energy provision or any control system to operate. There are two categories, dependent on whether they have moving parts or not. The main devices without movable parts are sails and bilge keels. Sailing ships, which frequently sail receiving sea and wind on the beam, have a very effective natural system which dampens roll, due to the sail resistance when following the characteristic ship's roll motion. The change in the lift on the sails

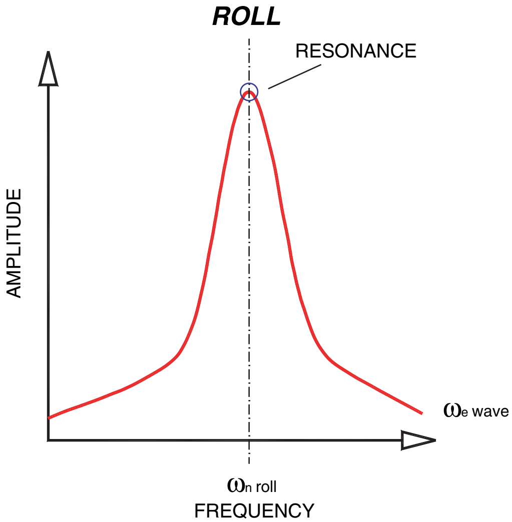

Figure 8. The roll amplitude of motion versus the encounter wave frequency [ω_{roll}] taking into account the natural frequency [ω_{n}] of the ship's motion. If we compare this figure with Figure 7 corresponding to the heave and pitch motions, we can see how, unlike these, its amplitude is very sensitive to the wave excitation force as they are very poorly damped motions. Therefore, the resonance phenomenon becomes really awkward. Drawing: author.

varies as the ship rolls and the phase of these lift forces tend to reduce the energy that the ship's roll generates. For this reason, sails are very effective as stabilizers of roll motion for small ships and they have been used in fishing ships with this aim. The required dimension of the sails together with the necessary tackle to rig them limits their use in large ships with mechanical propulsion.

The roll damping effect, derived from the wave generation of a ship, is almost non-existent with regard to the ship's DOF and on the other hand, the possible water flow in eddy circulations is scattered from the bilge due to the rounded shape that most hulls adopt at this area[17] forming the named *turn of the bilge*. As a consequence, there is only viscous friction to scatter the energy derived from the roll. Therefore, a clean, smooth hull has got a very small roll damping capacity unless some appendages are added in order to increase the generation of water flow eddy making and the viscous friction created by its motion.

The simplest and cheapest system of anti-rolling passive devices in a ship with mechanic propulsion is bilge keels[18], which may reduce roll amplitude by up to 35%. William Froude (1810–1879)[19] can be considered one of the promoters of bilge keels. He had been devoted to the study of ships′ motion although his work was specifically related to the ways of reducing roll motion[20] at sea. But this was before carrying out his notable experiments for the British Admiralty from 1867 using model ships in a test tank which he built at his home, with the aim of determining the ships′ physical laws[21]. Froude had worked with Brunnel[22] from 1837 sharing their interest to discover the secrets to predict the ship's behaviour; and when he was present in the sea trials on board the ship "*Great Eastern*"[23], his suggestion about the incorporation of bilge keels to reduce roll was finally accepted.

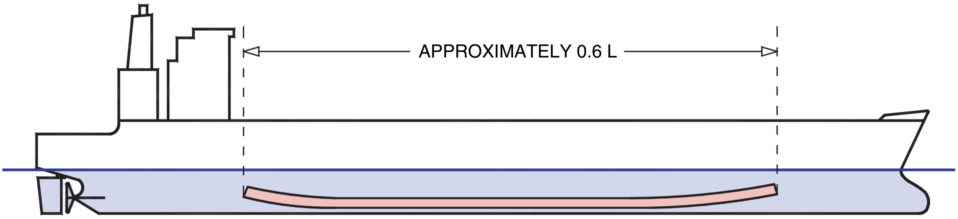

Figure 9. Longitudinal position of the bilge keel. Drawing: author.

These appendages fixed to each side of the hull are essentially small reinforced plates perpendicularly projected outwards in the bilge and approximately at the height of its turn, being longitudinally fixed to the height of its amidships section and generally being a little longer than half its overall length (see Figure 9).

As the ship rolls, the bilge keel quickly takes water that then scatters on its opposite side creating eddies; so there is a very small increment of the added mass[24], but at the same time, there is a significant increment of its damping effect due to the eddies and to the additional friction generated. The design of bilge keels adopts suitable profile and inclination to line with the flow streamlines when the ship is upright and in calm water conditions. Consequently, in this situation, the ship equipped with these appendages has a very small increment of the resistance to forward motion and when rolling, generates a much bigger roll damping effect than its size could suggest, becoming the cheapest anti-rolling device in a ship. Its main function is to be useful to damp the ship's roll motion. Other advantages of these appendages of relative minor importance are serving as a protection to the bilge area in case of grounding, increasing the longitudinal resistance of the hull structure at its level and offering enhanced directional stability.

The main disadvantage is the added resistance to forward motion produced in calm water conditions (when roll reduction is not necessary) and during the ship's navigation at different speeds from that for which the keels were designed since the flow streamlines may no longer be in line with the keels[25]. There are some situations, due to the kind of navigation or to the type of work a ship does, when bilge keels are not fitted. This happens in ships that generally sail in iced regions, where bilge keels are quite vulnerable to damage and, as a consequence, become detached from the hull with the resulting risk of fissures. It also happens in some fishing ships because there is a potential risk that the nets could catch on the keels whilst being deployed or recovered.

Bilge keels have a tendency to have biofouling growth along their outer edge and on the inferior part along the junction of the hull welding whose increase makes the underwater hull rougher, thus damaging the ship's operational efficiency as it diminishes its speed and increases its costs[26]. The cause is thought to be due to eddy currents that can prematurely deplete antifouling paint coatings. As a consequence, and in order to avoid these inconveniences, it is recommended that both the exterior edges and the inferior part along the junction of the hull welding should be additionally protected with antifouling and anticorrosive painting. This would assure the formation of a suitable film with the aim of optimizing its length of time.

5.2. *Design requirements.* A bilge keels damping action is relatively small but very effective and with no additional cost after the building of the ship. Its exact position in the ship is carefully studied in order to maximize the hydrodynamic roll resistance and to minimize the hydrodynamic forward motion resistance when the ship is sailing. With such an aim, several positions and dimensions are usually tried during the model ship trials to optimize its service and at the same time to calculate the power requests with each option[27]. The bilge keel is usually placed in the amidships section of the ship's hull, often perpendicularly at the turn of the bilge.

There are many different ways to build a bilge keel, some of which are the result of elaborate designs, trying to improve its roll damping effect at the same time as its resistance to forward motion is reduced. Although this appendage would not be considered as a critical structural member of the hull, it should be very carefully designed because the region of the hull it is welded to is fairly highly stressed owing to its distance from the neutral axis of turning of the ship's mass.

Bilge keels are aligned with the flow streamlines around the ship moving at design speed in calm water so that its effect is minimum with regard to the increase of the resistance to forward motion. Studies are carried out with model ships concerning the flow visualization during the resistance to forward motion tests. The flow lines are measured along the bilge by means of dyes or small flags so that the bilge keels are set to coincide with them. Thus, dependent on their extent and depth, the additional resistance may be kept to an additional increment of between one and three per cent of that of the main hull.

5.3. *Construction.* It is very important to build the bilge keels in such a way that the exterior plates to which they are attached stay intact. As a consequence, and in order to avoid damage, their shape as well as their position and their reinforced procedure to the hull are carefully studied. For their protection, the tip of bilge keels should be always arranged so that they lie inside the maximum beam of the ship and above the baseline. If they extended further than these limits, they would be much more exposed to damage risks during docking, dry-docking and in shallow water (see Figure 10).

Unless carefully designed, the bilge keel longitudinal endings tend to create areas of stress concentration that may cause fissures on the bilge plating. To prevent this, the bilge keel surface should be gradually tapered-off at those areas and additionally strengthened with a small double plate welded to the bilge plating at the height of a structural resistant element, for instance, a web frame, with the aim that there is a bigger area of transference of the longitudinal stresses between the bilge keel and the hull[28] (see Figure 11).

Bilge keels are made of steel plates with their free end or tip strengthened. Generally they are not directly welded to the hull, but to a ground bar[29] previously welded to the hull where the bilge keel is welded over it by means of butt weld. That method forms a junction or weak union so that the butt weld yields quite easily and (if hooked on an obstruction, or if there is a collision or grounding) the bilge keel drops with no hull damage to the strengthened plate. Alternately although less usual, the bilge keel is scalloped throughout its length and may be directly welded to the hull by intermittent welds that conform a weak link to the hull. In large ships, the bilge keel usually has a transversal section with a "V" shape and is internally strengthened at regular intervals (see Figure 12).

5.4. *Basis of the bilge keel effectiveness.* The friction and roll damping effect of bilge keels increase with the speed v with which they move from one side to the other of the ship; and it depends not only on the roll angular velocity $\frac{\delta\theta}{\delta t}$, but also on the

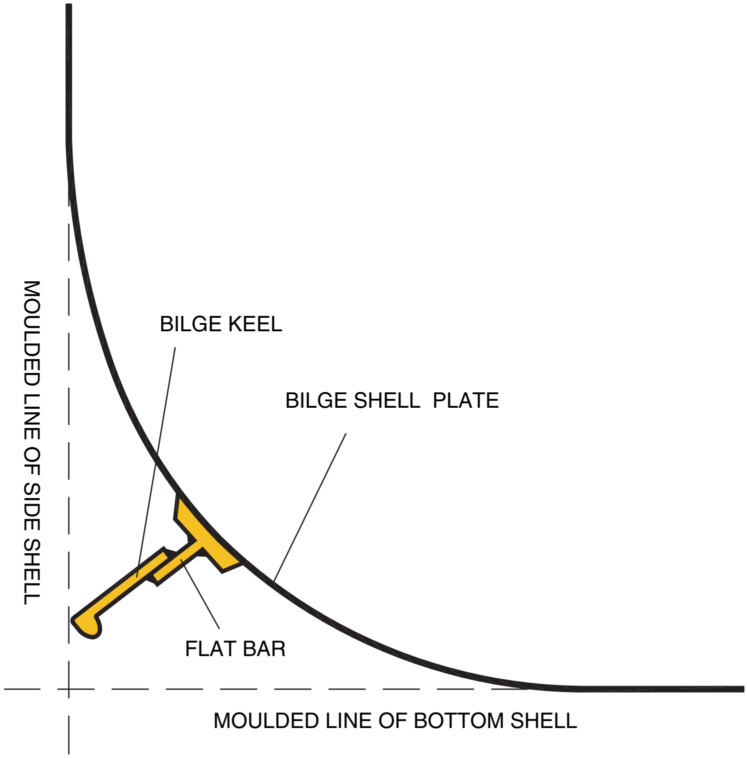

Figure 10. A standard bilge keel. Drawing: author.

radius k of the circular path of the bilge keel (see Figure 13). So, even the relatively small roll period of a large ship may become very effective to damp its roll, as the value of k generates very high lineal velocities of the bilge keel over the water flow $\left[v=k\frac{\delta\theta}{\delta t}\right]$.

In pure simple harmonic motion there is no friction, but naturally when the roll motion takes place the friction, which is always there, increases with the bilge keels. This contributes to damp such motion when dissipating more energy as heat[30]. In this case, the friction has two effects: first, it reduces the angular velocity $\delta\theta/\delta t$ and therefore the roll slows down and its period increases. Secondly, the friction is directly proportional to the angular velocity $\delta\theta/\delta t$ and therefore the higher its velocity is, the bigger will be the dissipated energy.

The roll damping action also increases with the square of ship's forward speed and when this translational motion is combined with the rotational roll motion, the water flow moves over the forward end of the bilge keel with an angle of attack that generates a dynamic lift opposed to the roll. However, this misaligns the bilge keel in relation to the direction of the water flow, and therefore creates an additional resistance to forward motion.

Bilge keels offer a significant improvement in the roll damping compared to a ship without them, but their damping effect is smaller than that obtained by other roll stabilizing devices (i.e., *controllable fins, passive anti-rolling tanks, active anti-rolling tanks, gyroscopic controlled stabilizers*). However, because bilge keels are the only

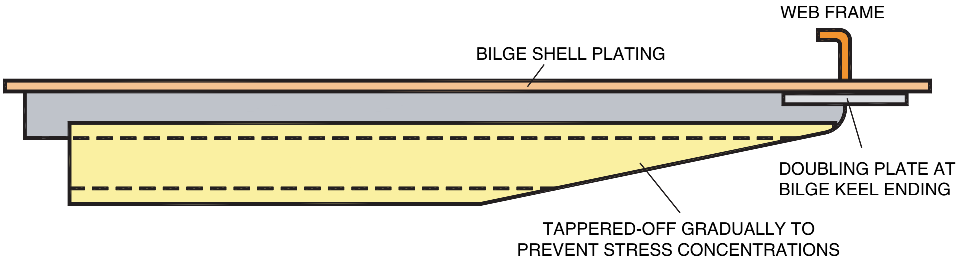

Figure 11. Bilge keel longitudinal ending construction to prevent bilge plating from cracking due to stress concentrations. Drawing: author.

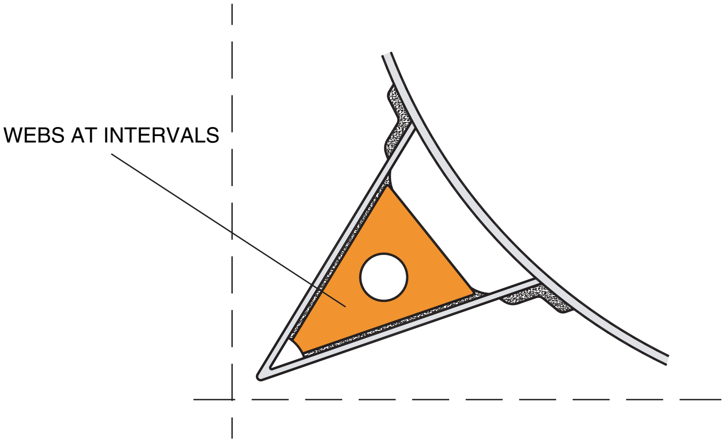

Figure 12. A bilge keel for a large vessel. Drawing: author.

devices which can be used in very rough seas their installation is recommended whenever possible even though other stabilizing devices are set.

The additional damping generated by bilge keels is proportional to their area and it is clear from the results of different tests carried out by different researchers that when the area is the same, in the case of *slender ships*, the bigger their aspect ratio is, the more effective bilge keels are. However, in *full ships* with a big block coefficient C_b, the smaller their aspect ratio[31] is, the more effective they are.

Independently of the specific structural shape they have, bilge keels work in accordance with a very simple, effective theory. To see the effect of bilge keels with regard to the decrease of the natural period of this motion[32], we start from the formula of the ship's roll natural frequency we saw in section 2, $\left[\omega_{roll} = \sqrt{\frac{\Delta_s\, GM_T}{I_x}}\right]$ and from there we get the formula of the ship's roll natural period (provided that the angle of heel induced in the roll is less than 10°) depending on its mass radius of gyration K. We should take into account that $T_{roll} = \frac{2\pi}{\omega_{roll}}$; $\Delta_s = M \cdot g$; $I_x = K^2 \cdot M$ and substituting these values in the formula, we get the following expression[33]:

$$T_{roll} = 2\pi \sqrt{\frac{I_x}{\Delta_s \cdot GM_T}} = 2\pi \sqrt{\frac{M \cdot K^2}{M \cdot g \cdot GM_T}} = 2\pi \frac{K}{\sqrt{g \cdot GM_T}} \tag{6}$$

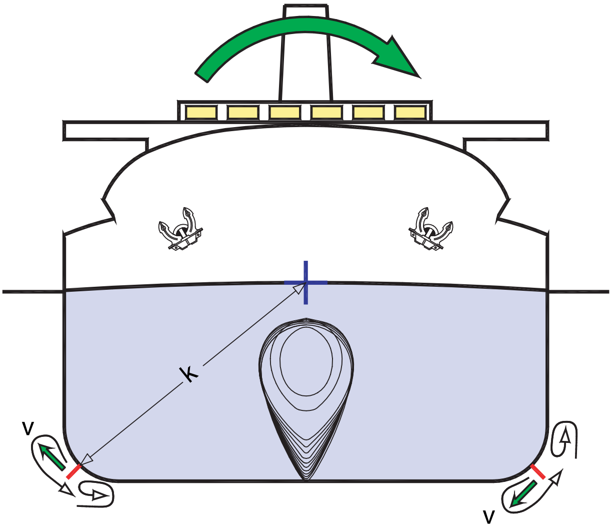

Figure 13. Importance of the radius of gyration k of the added water mass which bilge keels drag in the damping effect of the roll motion. Drawing: author.

Each element of the ship individually considered has its *MOI* which is equal to mk^2. Therefore, the ship's *MOI* as a whole is defined by the expression $I_x = \Sigma(m_1k_1^2 + m_2k_2^2 + ... + m_nk_n^2)$; i.e. the summation of products of elementary masses and the squares of their radius of gyration. As a consequence, the radius of gyration K of the ship's virtual mass when rolling will be $K = \sqrt{\frac{\Sigma(m_1k_1^2 + m_2k_2^2 + ... + m_nk_n^2)}{M}}$.

With the bilge keels projecting themselves from the ship's sides at the height of the turn of the bilge, an increment of the added mass of water m is generated as it moves with the ship's roll with a large radius of gyration k of that water mass, which makes the K value from the equation increase. Consequently, if we take into account the equation of the ship's roll period depending on the radius of gyration of its mass, it is clear that one of the effects of bilge keels is to increase such a period. However, the main effect of bilge keels is due to the increase of viscosity produced as a consequence of the eddies which indirectly generates an increase in the roll resistance. According both to theory and to observation, we can see that bilge keels are more effective in a ship when it moves forward in waves than when it is stopped and not making way[34]. Such observations suggest that there is a lift at the forward sections of bilge keels. This lift is opposed to the roll motion and it additionally contributes to the ship's course stability. Regarding this, bilge keels are essentially long fins designed in order to give a reaction force when they are moving in relation to the water flow surrounding them (i.e. acting as hydrofoils) and they represent a special application of fixed stabilising fins. Bilge keels should be considered as a means of roll reduction, not designed to achieve its complete elimination as there must be some roll for them to become effective.

5.5. *Stresses generated by the roll motion.*

5.5.1. *The racking stresses.* The restoring forces generated by the ship as an answer to the wave exciting forces provoking the roll motion initially act over the underwater volume of the ship. The ship's structure must be able to transmit these forces to all the parts over the waterline. These have the tendency to continue their rotation further than the amplitude of the roll itself until the restoring forces are effectively transmitted. This transverse distortion produces stresses in the ship which, known as racking stresses, tend to deform the box section of the hull as a hull girder (see Figure 14).

The tendency to deformation that the hull as a whole suffers (owing to the unequal pressure distribution around the ship that tends to bend side plating and transverse frames) is bigger in the main deck margins and in the turn of the bilges (see Figure 15). At design level and in order to resist the racking stresses in these parts of the ship, reinforcements such as knees, web framing between frames and corner brackets are used.

5.5.2. *Torsional stresses due to modification in the distribution of the buoyancy as a consequence of the roll*. As a ship heels over, the hull rotates about the waterplane axis, in such a way that the waterplane area becomes more and more asymmetric about the ship's centreline and an excess of buoyancy is produced due to the flares at the fore and aft ends. This generates bending stresses in these areas which cause the hull to bodily rise while the rolling axis shifts away from the centreline towards the excess of buoyancy. However, another effect of these characteristic shapes of a ship is that, as it reaches the maximum roll to both sides, the restoring forces increase more quickly at bow and stern than amidships; thus they expose the ship to twisting or torsional stresses that tend to deform the hull longitudinal structure which consequently must be strong enough as a whole so as to resist these stresses and transmit the excessive righting forces of its ends towards the amidships area to minimize the twisting. The object of this is to minimize the torsional forces produced when the ship reaches the maximum roll. That is why it is necessary to incorporate reinforcement all along the length of the ship so that its structure does not get damaged.

5.5.3. *Bilge keels as elements of structural reinforcement*. As we have seen, bilge keels are placed in an area where big racking stresses are generated when rolling. Therefore, the whole area of the bilge needs to be specially strengthened and the bilge keels suitably built, will help to reinforce structurally such areas. On the other hand, the torsional stresses produced with the roll justify (as we have pointed out in 5.3) the necessity that the width at their ends decreases progressively in order to get a suitable transmission of the bilge keel's efforts to the hull to minimize the risks of possible cracks.

6. CONCLUSIONS.

Of the three ship's *DOF's* at sea (*heave*, *pitch* and *roll*), which have some analogies with a spring-mass damper system, the roll motion is the one which presents less damping and therefore, it is the most probable one to enter resonance thus increasing drastically the amplitude of motion. So, particularly since the introduction of mechanical propulsion in ships during the second half of the nineteenth century, it has been necessary to incorporate anti-roll systems; bilge keels are a passive system with no movable parts and are incorporated in almost all ships.

This article, which introduces roll motion study whose considerations help as an initial basis to understand any anti-rolling system, has been specifically oriented to the study of bilge keels from the constructive and operative point of view, trying to

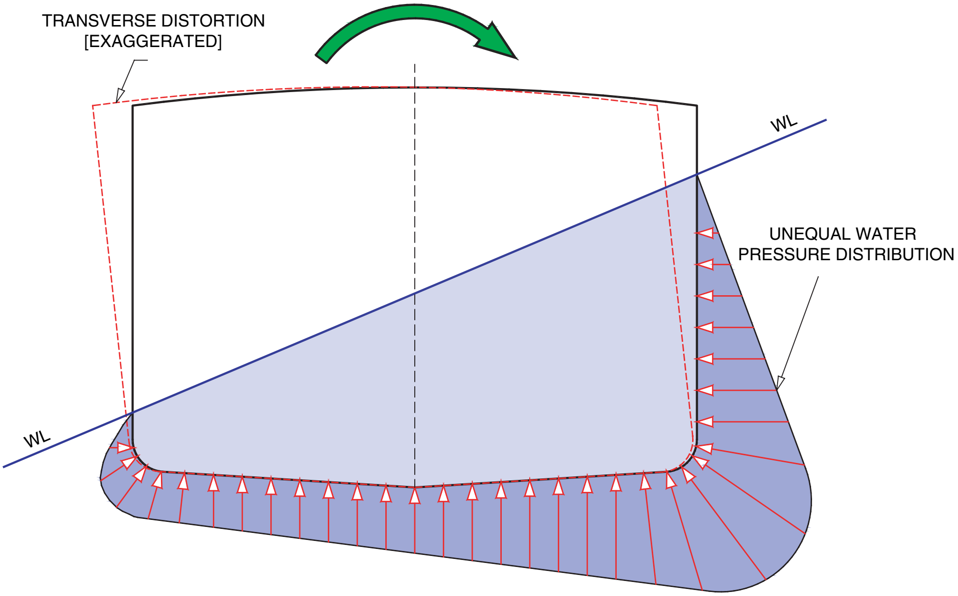

Figure 14. Racking stresses. Drawing: author.

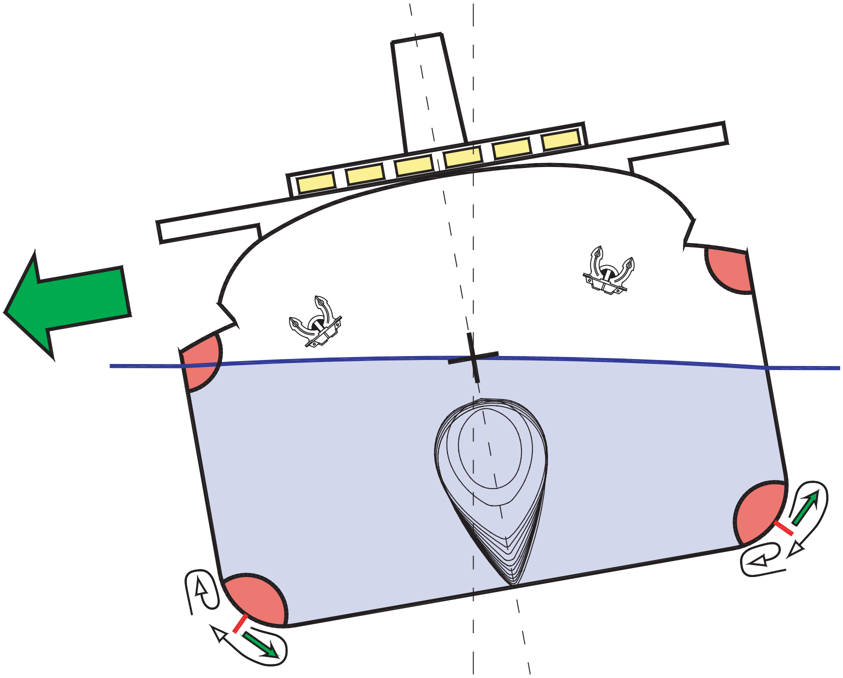

Figure 15. Areas of the hull under greatest stress due to racking when heavy rolling. Drawing: author.

show the effects produced in a ship, particularly emphasizing effectiveness. In short, the bilge keels effect is produced by:

- The increase of the ship's roll natural period and;
- The generation of a damping moment resulting from the viscous eddy flows around their surface, the pressure resistance around it and the hydrodynamic lift in the forward sections of bilge keels.

Due to the different origin of the aspects which determine their functionality, there has not been unanimity among the authors when defining it in a general sense[35], using terms such as *stabilising*, *damping* or *reduction effect*, although it seems that "*stabilising*" is the most accepted term.

Starting from the analogy of the roll motion with a spring-mass damper system, the formula of a ship's roll natural period has been deduced and it has been shown why bilge keels contribute to increase the radius of gyration of a ship's mass and therefore its roll natural period. Taking into account that at sea, in general, the ship's master cannot modify the ship's roll natural period, supposing this motion comes in resonance with the wave encounter period, the only good seamanship at his/her hands is to modify the wave encounter period by changing ship's course and/or the speed.

The effect of bilge keels on the ship's roll damping is quite complex and depends on a lot of factors such as the keel's area, its aspect ratio, the kind of ship and its speed and depth, the radius of gyration of the ship's mass, the GM_T and the bilge shape. With regard to the configuration, efficiency and building of bilge keels, many theoretical as well as experimental results have been published. However, in spite of the progress achieved in its design, there is a current need to continue investigating more sophisticated methods to determine the best parameters of bilge keels for each ship.

END NOTES

[1] It constitutes a part of the ship's dynamics study known as *seakeeping* – the other one is *manoeuvring*, which deals with the ship's motion without the wave excitation force, i.e. with calm sea and where the motion is generated by the action of control surfaces and propulsion units and it is associated with dynamic stability, course stability, path motion stability, etc. Seakeeping is associated with the ship motion due to the wave excitation force while the ship keeps its course and speed constant, which defines a balanced state of motion, and the wave action makes the ship oscillate with regard this balance.

[2] Translation refers to motion in a straight line and rotation refers to rotating or spinning around an axis.

[3] As we know, the ship's weight acts vertically downwards in **G** and only changes in the weight distribution affect its position. Therefore, if there are no weight changes, no shifts in **G** happen.

[4] The hydrodynamic theory usually puts the origin of axes at the ship's centre of gravity **G**, which not necessarily lies on the waterplane.

[5] The metacentre in this case stays more or less static and consequently the GM_T value for these small angles of inclination can be considered constant on the one side and on the other, the roll period will be independent of its amplitude.

[6] As we know, inertia is the tendency of an object to stay at rest or go on moving straight at the same speed.

[7] The distance from the rotation axis "x" to the point where the whole mass of the object can be concentrated without changing its MOI.

[8] Taking into account the *real mass* of the ship, the approximate value of its radius of gyration in terms of its beam is from $0 \cdot 30 \cdot B$ to $0 \cdot 34 \cdot B$. However, when there is roll motion, the *added mass* of inertia generated with it and whose value (as a general approximation, the radius of gyration of the mass can be

considered to increase 25%) increases as the relation between beam and draught is bigger or smaller than 2 (see a complete study about this in Clark, 2005, pp. 217–220).

[9] It must be taken into consideration that the fact a ship has a GM_T bigger than another one does not mean it is safer and more stable, but in this sense the most desirable characteristics of ships are those combining a suitable maximum righting moment and a suitable angle of inclination with a considerable margin of positive stability. Thus, liners have a GM_T relatively small compared to other ships in consideration to the passengers′comfort while its margin of positive stability is generally big mainly due to its big freeboard (see in this sense Gillmer, et al., 1982, p. 146).

[10] In order to get a suitable compromise between these two alternatives, naval architects generally try to maintain the value of the GM_T of the ships they design between 5 and 8% of the beam.

[11] Both the *heave* and *pitch* motions generate wave systems that dissipate the energy of the motion quickly. Although its study goes further than the content of this article, suffice it to say that in the case of the *heave*, the internal restitution force of the ship depends on the TPI (**T**ons **p**er **I**nch immersion) and in the case of the *pitch* it depends on the MT1 (**M**oment to **T**rim **1** inch).

[12] The outstanding characteristic of the open ocean waves is its irregularity, which gives rise to sudden an unpredictable growth of the roll amplitude. The irregular waves can be described by statistical Mathematics on the basis of the assumption that a large number of regular waves having different lengths, directions and amplitudes are linearly superimposed (for more information about this subject, see Tupper, 2004 pp. 89–101).

[13] When there are forces which are opposed to motion in a system, the energy is not constant and it is called damped motion. However, if there is an external excitation force, as in the case of a wave system, this force gives energy to the system and the motion is again similar to SHM. But its amplitude depends on the frequency of the external excitation force, so that when this frequency is the same as the natural frequency of the rigid body over which it acts, the amplitude is maximum and the system enters resonance.

[14] In the case of a ship, the force generated by the waves. This external force, which gives energy to the system so that the motion is kept, is generally called "*the external force function*" in English.

[15] The smaller the damp with a SHM system is, the more sensitive to the frequency of the excitation force will be. Therefore, the resonance phenomenon is more critical in the roll (less damped) than in the *heave* and *pitch* (more damped).

[16] The kind of ship, its dimensions and the state of the cargo, its course and relative speed in relation to the wave direction as well as to the sea state are elements which determine the possibility that this motion can come in resonance; therefore, it should be studied in each particular case (see comments in *Principles of Naval Architecture*. *Vol III* 1989, pp. 79–80).

[17] In this sense, (following Graham, 1989, pp. 79–80) we should take into account that with regard ship design, there will be a bigger damp if the hull is designed with a small *turn of the bilge* and with the appendices (in this case bilge keels) as far as possible from the centre of gravity.

[18] Generally in English it is known as "*bilge keels*" whose literal translation into Spanish would be "*quilla de pantoque*". Consequently, we could say that if we abide its denomination, its function is the relevance in Spanish while in English the relevance is its location, although sometimes the name "*antirrolling keels*" is also used to refer to these appendices. In this case, the name makes reference to its function instead of its location (Gillmer, et al., 1982, p. 269).

[19] "*The papers of William Froude*" M. A. LL. D. F. R. S. 1810–1879: WESCOSTT, A., chapter A Memoir, pp. xi–xiii; GAWN, R. W. L., chapter Evaluation of the Work of William Froude, pp. xv–xxii and FROUDE, W., chapter On the Practical Limits of The Rolling Of Ships In a Sea-Way, pp.101–110, The Institution of Naval Architects, London 1955.

[20] See Clark, 2005, p. 36 and Gillmer, et al., 1982, p. 204.

[21] The well-known n° *Froude* "*Fn*" (dimensionless), expressed as the relation between the ship speed (in m/s) and the square root of the product of the length at the waterline-LWL- (in m) by the gravity acceleration (in m/s^2), is still used nowadays by naval architects to predict the ship's behaviour on the basis of scale models.

[22] A famous English civil engineer (1806–1859) convinced of the economies of the scale provided by large ships in special with regard the fuel consumption. In this sense, Brunnel stated in 1836: "*It is well known that the proportional consumption of fuel decreases as the dimensions and power of engines are increased and consequently a large engine can be much more economical than a small one. The resistance of vessels on the water does not increase in direct proportion to the tonnage …*" (CORLET, E. D. *The Iron Ship*. Arco Publishing, New York, 1974, p.10).

[23] A ship 693 feet length and 27 000 tons dwt (including the coal used as fuel) designed and built by Brunnel after the "*Great Western*" and the "*Great Britain*" to carry out even longer trips than the previous two ships (to Australia and South-East Asia) although this project turned out to be a disaster eventually from the financial point of view even before it started working.

[24] The *virtual mass* of a ship is a concept used by naval architects to quantify the necessary force to vary the speed of a ship a fixed quantity in different situations, i.e. to generate a particular positive or negative acceleration with or without course change. Newton's second law of motion sets that an object's acceleration is directly proportional to the force applied and conversely proportional to its mass ($F = m \cdot a$). The increment of the apparent inertia of a ship due to the energy absorbed by the acceleration of water particles produced by the ship's motion in water can be seen as an increment of its apparent mass known as "*added mass*". This increased mass of the ship as a whole is called "*virtual mass*", which is the sum of its "*real mass*" and its "*added mass*". The added mass in deep waters of a well designed ship is generally between 5 and 10% of its real mass although it increases noticeably as the ship sails in shallow waters. Any positive appendix of the ship (as there are what we could call negative appendices such as bow thrusters tunnels which diminish the underwater volume of the ship) increases its *added mass* to a greater or lesser degree and that is why we talk about the additional increase of the added mass produced when setting bilge keels to a ship. This added mass is a dynamic effect which disappears if there is no motion, being this the reason why it is sometimes called "*hydrodynamic mass*" (see Clark, 2005, p. 198).

[25] See Sellars, pp. 84–101.

[26] See Taylor, A. H. & Rigby, G. (2002) "*The Identification and Management of Vessel Biofouling Areas as Pathways for the Introduction of Unwanted Marine Organisms*". Report prepared for Australian Agriculture Fisheries and Forestry [http://www.asa.com.au/CommercialVesselsBiofoulingProjectFinalReport.pdf].

[27] For more details, see Ikeda, 2004, pp. 89–93.

[28] See in this sense "*Crack in shell plate at bilge keel termination*" news item number 1, 2003 published on 04.03.03 by *DNV* Classification Society, where after analysing the extension of damage due to a crack in the shell plate at bilge keel ending of a tanker of 140000 grt, it concludes that one of the factors which contributed to such crack was that the bilge keel ending suddenly reduced from its normal width to its ending (ratio 1:3) and that the angle should have been smaller [http://www.dnv.com/publications/classification_news].

[29] It is generally recommended that such plate is as thick as the hull plating at the bilge height or at least of 14 mm in the case that its thickness is bigger and that both plates have the same quality with regard steel.

[30] Regarding the viscous forces, the kinetic energy of the hull is transferred to the fluid due to viscous effects (skin friction, flow separation and eddy making). These forces depend on the relative velocities between the hull and the fluid.

[31] In this sense, it can be established that when some authors (see "*Damping effect of bilge keels with different aspect ratios*" http://virtual.vtt.fi/maritimeinstitute/pdf/mrn1_2006.pdf) state that at equal areas, the bigger their aspect ratio is, the more effective bilge keels are, it must be interpreted that they are only referring to ships with a high length/beam ratio and high flares at bow and stern. Supporting this statement, see Ikeda 2004 where the results of the tests carried out in a full ship with small flares with a C_b equal to 0.8 and a slender ship with high flares and therefore with a smaller C_b are stated (in the first case, a bilge keel with a small aspect ratio turned out more effective, while in the second case it was with a big aspect ratio).

[32] The reduction of the roll natural period contributes to the reduction of this motion as Froude proved [Froude, W. "*On the rolling of ships*" Trans. of the Institution of Naval Architects, 2, pp. 180–227, 1861] when he stated that it is the waves slope and not their height what excites the roll motion, and taking into account that small wave-length waves have more slope than the ones with longer wave-length, we can consider that the bigger the natural roll motion period is, the less possibilities there are that resonance is produced with the wave encounter period.

[33] When the GM_T is expressed in feet, lots of authors [see for instance Gillmer et al, 1982] use this simplified formula $T_{roll} = \frac{1 \cdot 108 \cdot K}{\sqrt{GM_T}}$ which comes from substituting the expression $\frac{2\pi}{\sqrt{g}}$ by its value $\frac{2\pi}{\sqrt{g}} = \frac{2\pi}{\sqrt{9 \cdot 81 \frac{m}{s^2} \cdot \frac{3 \cdot 28084 \frac{feet}{s^2}}{1 \frac{m}{s^2}}}} = 1 \cdot 108.$

[34] Although much less, in this last case they also provide some reduction to the roll motion [see Taylor & Tang, 2006 p. 171].

[35] See in this sense, Tristan Perez, 2005, p. 114.

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