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An Overview of Roll Stabilizers and Systems for Their Control

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ABSTRACT: Ship roll motion in waves can be characterized as a strongly non-linear and multivariable dynamic process which is more affected by disturbances, in general, than by the maximal controlling parameter. The article presents methods of roll motion compensation, the majority of which have been used on ships for many years. Although they are not capable of reducing permanently the error to zero, their potential has still not been used to the full. The operational efficiency of roll motion compensators can be improved using control systems. Research activities are in progress to check the applicability of advanced control methods making use of modern computer techniques. Some of them are mentioned in this paper.

1 INTRODUCTION

The first known PID controller, referred to as the three term controller, was used as an autopilot on a ship. The results of the work by Minorski (1922) on the autopilot making use of the three-function mechanism of automatic control were applied in other branches of economy, and at present these controllers contribute in about 90% to a total number of controllers used in the industry. Since 1928 Minorski also worked upon control of active stabilization tanks used for roll motion damping. His works based mostly on intuition. It was Chadwick (1955) who was the first to make use the transfer function in designing control systems for ship stabilization. The state of knowledge on the control theory in the last century's sixties was used by Webster (1967) as the basis for analyzing the problem of active roll motion stabilization.

Nowadays some time delay is observed in introducing new control techniques on ships. New methods are used, in general, only when they have undergone positive verification in inland conditions. This tendency is quite understandable. In the past, the autopilot only replaced a human operator in tiresome work oriented on keeping the course in the environment in which preserving good concentration for a long time was extremely difficult. Possible incorrect operation of the autopilot could be easily detected and corrected by the human wheelman. On the contrary, the new control systems are mainly expected to work reliably, with high efficiency being slightly less important. At present, in case of a failure of the control system, the operation of the majority of its components cannot be substituted by the actions performed by a human being. Moreover, obvious difficulties in contacting the manufacturer's service centers and high cost of servicing is a reason why the solutions most preferred in marine applications are those which have already underwent positive verification for their high reliability and are trusted by the crew. But this does not mean stagnation in the activity of researchers in the field of marine automatics. Like in case of the automatics (now referred to as conventional) which had to find its place next to manual control systems by separating manual and automatic control modes, now the devices improving the operation of the present automatic systems can be disconnected, leaving the control to the devices which have been positively verified and are known to the operator. One of marine automatics segments, although not the most important one is the stabilization of ship roll motion. From among all oscillatory movements which the ship does on a heavy sea, the roll motion reaches the highest amplitudes and can make proper ship's operation much more difficult. Different types of ships require different types of roll stabilization systems. On a cruise ship or ocean liner excessive motions interfere with the recreational activities and comfort of passengers. They can affect the effectiveness of the crew too. On Ro-Ro ships for instance many containers are stowed above deck where they are subjected to large accelerations due to the rolling. In some extreme conditions the lashings can fail and containers may be lost overboard

The paper presents an overview of roll motion stabilizers. The first section presents a mathematical model describing the ship movements, and the simplified model concerning the oscillations done by the ship along its longitudinal axis. Particular types of stabilizers are described in the next sections, with attention paid to control methods which can help to improve their performance characteristics. The final section presents the summary of the overview.

2 THE MATHEMATICAL MODEL

The mathematical model of dynamic processes facilitates their analysis and can be used for preliminary verification of systems designed for their control. At the same time the simplified model is a useful and convenient tool in the synthesis of the control systems.

The motion of a ship on moderate water can be described using six nonlinear differential equations. For the description of this mathematical model three coordinate systems are needed, which are the inertial-, body- and horizontal body coordinate system.

The coefficients in the above equations can be determined analytically and then corrected in model tests. In the process of introduction of ship control systems to marine operation it is important for the calculated results to be verified in full scale sea trials. The below presented model Hamamoto of 6 DOF (2010) was verified in turning circle and zigzag trials, which were done on moderate sea. The roll motion is investigated using simplified models. The 4 DOF surge-sway-yaw-roll model was worked out by Umeda and Hashimoto (2002).

The existing differences are obvious as they result from both model uncertainties and incorrectness, which is usually treated rather lightly – the more so that the ship dynamics depends on the load condition (cargo, fuels and oils, water) and obtaining identical conditions during regular operation is practically impossible. However, the correctness of the mathematical model can be improved after sea trials. The method of correcting the coefficients in the differential equations which describe the motion of the ship on moderate sea in such a way that the results of the sea trials are consistent with the simulations done with the aid of the existing model were proposed by Casado and Ferreiro (2005). The full model of the motion of a ship making use of stabilization fins was developed by Fang and Luo (2007).

$$\begin{aligned} m(\dot{u}-v\dot{\psi}) &= (m_y(\omega_e)-X_{v\dot{\psi}})u\dot{\psi}-m_x(\omega_e)\dot{u}-m_z(\omega_e)w\dot{\theta}+ \\ &-m_z(\omega_e)w\dot{\theta}+X_{FK}+X_{RF}+X_{SF}+T(1-t_p)-R \end{aligned} \tag{1}$$

$$\begin{aligned} m(\dot{v}-u\dot{\psi}) &= m_x(\omega_e)u\dot{\psi}-m_y(\omega_e)\dot{v}-Y_v v-Y_{\ddot{\psi}}\ddot{\psi}+ \\ &+Y_{\dot{\psi}}\dot{\psi}+Y_{v|v|}v|v|+Y_{v|\dot{\psi}|}v|\dot{\psi}|+Y_{\psi|\dot{\psi}|}\psi|\dot{\psi}|+ \\ &+Y_{FK}(\omega_e)+Y_{RF}+Y_{SF}+Y_{DF}(\omega_e) \end{aligned} \tag{2}$$

$$\begin{aligned} m\dot{w} &= -m_z(\omega_e)\dot{w}-Z_w(\omega_e)w-Z_{\dot{\theta}}(\omega_e)\theta-Z_{\dot{\theta}}(\omega_e)\dot{\theta}- \\ &-Z_{\ddot{\theta}}(\omega_e)\ddot{\theta}+Z_{FK}(\omega_e)+Z_{DF}(\omega_e)+Z_{SF}+mg \end{aligned} \tag{3}$$

$$\begin{aligned} &(I_{xx}+J_{xx})(\omega_e)\ddot{\phi}-(I_{xx}+J_{xx})(\omega_e)\dot{\theta}\dot{\psi}=-K_{\dot{\phi}}\dot{\phi}+ \\ &+(Y_v v-Y_{\dot{\psi}}\dot{\psi})z_H+K_{FK}(\omega_e)+K_{RF}+K_{SF}+K_{DF}(\omega_e) \end{aligned} \tag{4}$$

$$\begin{aligned} &(I_{yy}+J_{yy})(\omega_e)\ddot{\theta}+(I_{xx}+J_{xx})(\omega_e)\dot{\psi}\dot{\phi}=-M_{\dot{\theta}}(\omega_e)\theta- \\ &-M_{\dot{\theta}}(\omega_e)\dot{\theta}-M_{\dot{w}}(\omega_e)\dot{w}+M_{FK}(\omega_e)+M_{SF}+M_{DF}(\omega_e) \end{aligned} \tag{5}$$

$$\begin{aligned} &(I_{zz}+J_{zz})(\omega_e)\ddot{\psi}-(I_{xx}+J_{xx})(\omega_e)\dot{\theta}\dot{\phi}=N_{\psi|\dot{\psi}|}\psi|\dot{\psi}|-N_{\dot{\psi}}\dot{\psi} \\ &-N_v v+N_{vv\psi}v^2\psi+N_{v\dot{\psi}\dot{\psi}}v\dot{\psi}^2+N_{\dot{\psi}|\phi|}\dot{\psi}|\phi|+N_{DF}(\omega_e) \\ &+N_{\phi}\phi-N_{\dot{v}}\dot{v}-N_{\ddot{\psi}}\ddot{\psi}+N_{v|\phi|}v|\phi|+(Y_{\dot{\psi}}\dot{\psi}+Y_{v|v|}v|v|+ \\ &+Y_{v|\dot{\psi}|}v|\dot{\psi}|-Y_v v)x_H+Y_{\psi|\dot{\psi}|}\psi|\dot{\psi}|+N_{FK}(\omega_e)+N_{RF}+N_{SF} \end{aligned} \tag{6}$$

where; *m*-ship mass, *I*-moment of inertia, *X,Y,Z* external forces; *K,M,N* external moments to surge, sway and heave, respectively; *u,v,w* – surge, sway and heave velocities; φ, θ, ψ – roll, pitch and yaw angles, *R*-ship resistance, t_p - thrust deduction factor.

The subscripts *DF*, *FK*, *RF* and *SF* represent the Froude-Krylov diffraction, the rudder and the stabilizing forces, x_H – coordinate of the midship, z_H – coordinate of the point.

The equation of the main engine:

$$2\pi(I_{pp}+J_{pp})\dot{n}=Q_F-Q_P \tag{7}$$

The energy passed by the waves to the ship is distributed in a number of planes. As can be seen from the through couplings, these movements interact with each other. Their effect on the roll motion mostly depends on the nature of the wave. The roll motion can be described using one differential equation.

2.1 *1-DOF Equation*

The roll motion of the ship can be described (Zborowski & Taylan) by the relation:

$$(I_{xx}+J_{xx})\ddot{\phi}+B_{xx}(\phi;\dot{\phi})+\Delta GZ(\varphi)=K_{ext} \tag{8}$$

where: t- time, φ roll angle, Δ - weight displacement, (I_{xx} + J_{xx}) are the water mass and added mass moments of inertia and B_{xx} is the damping moment, which can be expressed in the linear quadratic form:

$$B_{\phi\phi}(\phi,\dot{\phi})=B_L\phi+B_{NL}\left|\dot{\phi}\right|\dot{\phi}+B_3\phi^3, \tag{9}$$

only in the linear form $B(\varphi)=B_{eq}\dot{\varphi}$, with the equivalent damping coefficient

$$B_{\phi\phi}(\phi,\dot{\phi})=B_L\phi+B_{NL}\left|\dot{\phi}\right|\dot{\phi}+B_3\phi^3, \tag{10}$$

where:

B_f – the skin frictional damping coefficient

B_e – the eddy damping coefficient,

B_L – the lift damping coefficient

B_{BK} – the bilge keel damping coefficient

B_{Fin} – the damping coefficient due to the presence of fins

It can be solved using a formula:

$$B_{eq}=B_L+\frac{8}{3\pi}B_{NL}(\omega\varphi_A)+\frac{3}{4}B_3(\omega\varphi_A)^3 \tag{11}$$

where: ϕ_A is the roll amplitude.

GZ-righting arm is a nonlinear function of the roll angle and can be expressed as:

$$GZ=C_1\phi+C_3\phi^3+C_5\phi^5 \tag{12}$$

The coefficients C_1, C_3 and C_5 can be calculated from the static stability curve

$$C_1=\frac{d(GZ)}{d\varphi}=GM \quad C_3=\frac{4}{\varphi_\upsilon}(3A_{\varphi_\upsilon}-GM\varphi_\upsilon^2)$$
$$C_5=\frac{-3}{\varphi^6_{\ \upsilon}}(4A_{\varphi_\upsilon}-GM\varphi_\upsilon^2) \tag{13}$$

represents the vanishing angle of stability, $A_{\varphi\upsilon}$ is the area under the GZ curve up to the vanishing stability, K_{ext} - wave exciting moment, GM_0 - metacentric height

In the version simplified only to the Froude'-Krylow term (Cholodin & Shmyrev 1972), the moment generated by the side wave can be given by the relation:

$$K_{ext}(t)=\Delta\cdot GZ\cdot\chi_A(t)\cdot\alpha_m(t) \tag{14}$$

where: χ_A is the reduction coefficient taking into account the effect of the ship dimensions with respect to the wave length; α_m is the wave slope (wave amplitude)

The course of the roll motion is affected by various agents, including the equivalent damping coefficient, the added masses righting arm. Some of them, such as those related to the ship structure for instance, do not change, while the others, like load conditions, mass (including cargo) distribution, free liquid surfaces, and most of all the nature of the external excitations change in time. But of highest importance is generating an additional stabilizing moment.

3 ANTI ROLL TANKS

The first type of roll stabilizing tank was based on the pioneering work of William Froude. In 1874 he installed water chambers for the purpose of achieving stabilization against roll.

In 1877 the stability of a Victorian ironclad battleship HMS Inflexible was questioned due to addition of unconventional armor. In 1880 the ship was equipped with water tanks for damping the roll, which turned out to be ineffective. The next stabilizing tanks were constructed by Watt (1883) as free-surface tanks. The improved version, so-called U-tube, was designed by Frahm (1911).

These tanks belong to the class of stabilizers bearing the name of the moving weights. They played remarkable role in the development of devices used for roll motion damping. Most of all, they presented in a natural way the physics of stabilizing moment generation. Like the see acting on the ship and the tank situated in it, the tank acts on the water inside it.

Generally, these tanks can be divided into passive, controlled–passive, and active tanks. The passive tanks include free-surface tanks and U-tubes.

The U-tube tanks, which take the name after their shape (Fig.1), are situated on both sides of the ship and connected in the water line, while in the air line they are connected or not when the air pressure is compensated by the atmosphere. The U-tube tank has the best performance when its natural frequency is the same as the natural frequency of the ship roll motion. The research activities upon these tanks are oriented on ensuring their good performance within a wide range of excitation frequencies The idea of the use of passive-controlled free surface tanks consist in the use of the relation between the period of water flow in the tank and the degree of its filling. The tank has sensors installed on its bottom to measure the pressure exerted on the bottom. Based on their indications, the phase lag of the stabilizing moment with respect to the heel is calculated, which provides opportunities for tuning the tank in such a way that the maximal stabilizing moment is obtained. Moreover, when the natural frequency of the ship is known, the best damping can be obtained within the range of most vulnerable wave frequencies. Optimal tuning for the tank to generate the stabilizing moment being in counter-phase to the exciting moment can be made possible with the aid of the effective wave slope measured by the wave height sensors.

3.1 *Passive tanks*

A configuration of an U-tube passive tank is shown in Figure 1. The tank consists of two side reservoirs and a connecting duct of constant rectangular cross section.

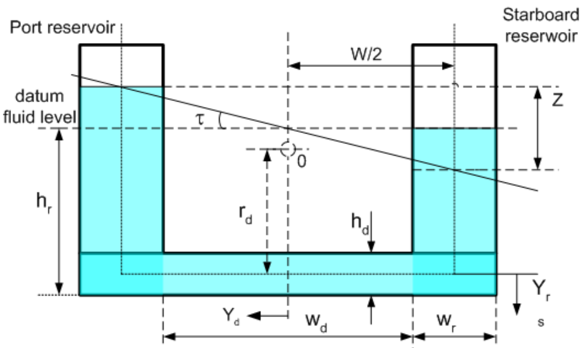

Figure 1. Passive tanks

To description for a purely roll motion of the ship with a passive tank can be the following simplified linear equation used:

$$(I_{xx} + J_{xx})\ddot{\phi} + B_{xx}\dot{\phi} + C_1\phi + [a_{4\tau}\ddot{\tau} + c_{4\tau}\tau] = K_{ext} \qquad (15)$$

where τ is the tank angle defined in Figure 1.

$a_{4\tau}$ - 4th moment due to unit angle acceleration

$c_{4\tau}$ roll moment applied by tank due to unit roll displacement

$$\omega_t = \sqrt{\frac{2gh}{w_r w + 2h_r h_d}} \qquad (16)$$

The passive tanks do not increase hull resistance. Their efficiency does not depend on the ship's speed. After some adaptation they can be used as anti heeling tanks for example for ballasting the train ferries. Their operational costs are low. On the other hand: they require remarkable space, thus reducing the space available for transport of cargo. The free surface of the tank reduces the metacentric height of the ship.

The passive tanks can be tuned automatically. In those cases they are referred to as passive-controlled tanks. This group includes passive-controlled U-tube tanks, tanks with energy In phase 1 the ship reached the maximal angle to recovery and free-surfaces tanks "*Flume*".

The energy for water flow between the tanks is delivered by the waves, while the control system shapes the signal controlling the valves in time. Pairoh and Huang (2007) formulated a series of rules, such as linear-quadratic regulator (LQR), predictive control and the dead beat predictive control, which are valid when controlling these tanks. This last control system is an improved version of the control of activated tanks, developed in the last century's eighties, in which the valves situated in the air line were closed in selected times. The mass difference between two tanks which acted on the arm $h=w/2$ created a stabilizing moment. Opening of the valve provoked fast water flow to the opposite side of the ship, where it was "frozen" again in the next cycle. The phase cycle of such tanks is shown in Figure 2.

In phase 1 the ship reached the maximal angle to port and starts to right to starboard. At this point the water is flowing from the starboard side to the port side due to the effect of gravity. In second phase the water obtained the maximal level in the port side tank, the valves on the port side are closed by the automatic control (point A). The water is kept blocked in the port side tank, due to the low pressure created in the upper part of the tank, from position 2 up to position 4 where the control signal is opening the valves (point B)

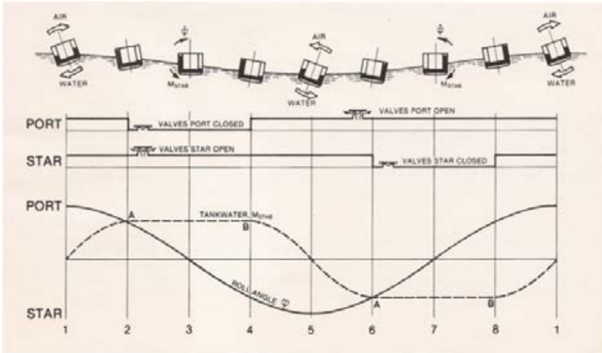

Figure 2. Phase cycle for roll periods longer than natural period of activated tanks INTERING GmbH

3.2 *Active tanks*

Tanks in which the stabilizing moment is generated by forced action of the actuator which presses the water from one ship's side to the other bear the name of active tanks. The performance of the passive tanks is limited by their ability to create the natural water flow from one ship's side to the other. At the same time the active tanks are capable of generating larger stabilizing moments from the same tank volumes. The control process itself requires a large amount of the delivered energy. The ability to pump large volumes of water in a short time requires sufficiently large powers for the actuators.

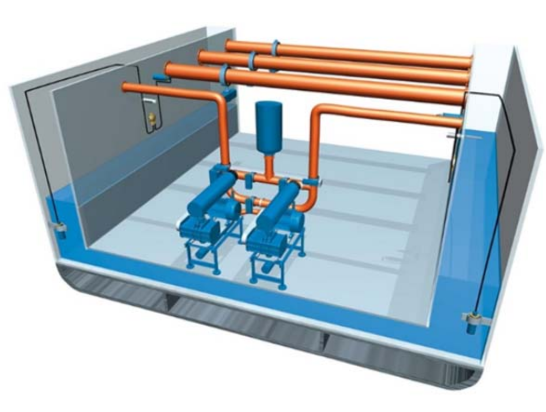

Figure 3. Active tanks system (Source: www.hoppe-marine.com)

The active tanks were mainly used by the Navy, where the economic aspect was of minor importance. Based on the results of laboratory tests (Alarcin 2007a) carried out on small-scale models, a fully active system was installed on the American destroyer USS Hamilton. The system made use of the pump in which the rotor had the blades with the variable attack angle. The measurements performed during the tests on stagnant water have revealed that the use of the active tank makes it possible to incline USS Hamilton by 18º. The tests also revealed that the system has sufficient potential to stabilize the destroyer at open sea after the waves had rocked it off to 30º from the perpendicular. Another application of active tanks was installed on a German cruiser, where a turbine-driven blower provided the air in the ducts obtaining in this way different level in the tanks (Moaleji & Greig 2007). These same authors proposed in their work (2006) regulating the pumps using an adaptive inverse controller.

3.3 *Fin stabilizers*

So-called active fins belong to the group of most popular roll motion stabilizers. They are situated on two sides of the ship and are rotated in opposite direction.

Like for the rudder blade, when the water flows round the fins the zones of high and low pressure are created on their surfaces, thus generating a force perpendicular to the fin surface *S*. This force can be divided into two components: the lift *L* directed perpendicular to the horizon line

$$L = C_L(\alpha) \cdot S \frac{\rho u^2}{2} \tag{17}$$

and the drag

$$D = C_D(\alpha) \cdot S \frac{\rho u^2}{2} \tag{18}$$

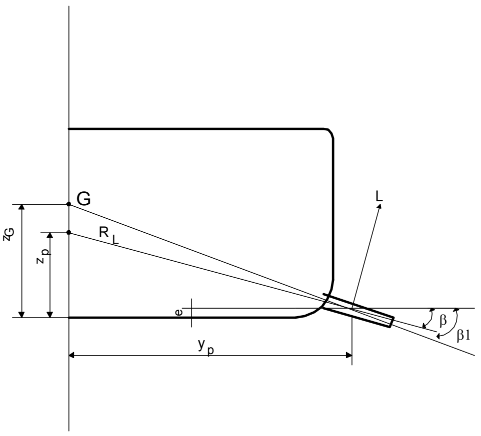

Figure 4. Localising active fins on ship hull

The increase of the lift is proportional to the fin surface. However, the fin surface area is limited not only for constructional reasons as it should not be protruding from the ship contour, but also because of the limitation of maximum stabilizing moment, which can not cause excessive heel angle of the ship more than 5 degrees.

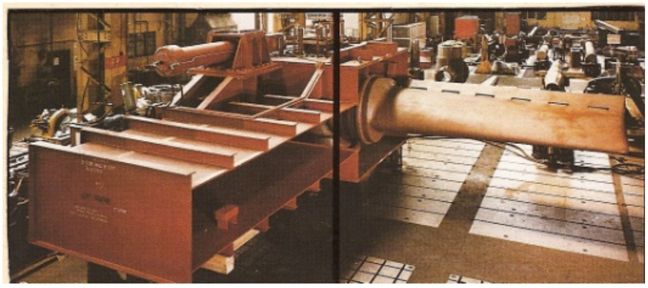

Figure 5. Fin with gear

At high ship speeds and larger attack angles the water flow separates from the fin surface. To avoid this unfavourable phenomenon, the fins are complemented by an additional flap which rotates around a pin situated at the rear fin edge (Fig.6). Due to this flap the maximum fin deflection can be increased by some degrees, which provides opportunities for obtaining a larger buoyancy force from the same fin surface.

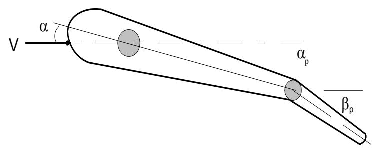

Figure 6. Sectional fin

The flap rotation mechanism is coupled in a non-inertial way via a gear with the fin shank drive. This way the flap inclination angle is proportional to the main fin inclination angle.

$$\beta_p = K_{prz} \cdot \alpha_p \tag{19}$$

where: β_p – flap inclination angle, α_p – main fin inclination angle K_{prz} – fin rotation angle gear

The presence of the flap increases the efficiency of the use of the fin surface, which leads to the increase of the lift force.

$$C_{Lk} = k2 \cdot C_L \tag{20}$$

*k*2 – coefficient calculated from the Karafoli formula (Cholodin & Shmyrev 1972)

$$k2 = 1 + \frac{\beta_p}{\alpha_p}\sqrt{\frac{b_k}{b}} \tag{21}$$

where: b_k – flap chord, *b* - chord of the entire fin

The ratio β_p/α_p is usually equal to 1.5, while the flap is approximately equal to one a quarter of the main fin width. The attack angle of the water flowing round the fin depends on the fin rotation angle α_p and the water flow direction, which is the resultant of the ship speed, the rolling speed, and the heaving velocity *u*. Finally the formula for the hydrodynamic inclination angle of the fin takes the form:

$$\alpha = \alpha_p + arctg\frac{\dot{\phi}}{u} \pm arctg\frac{v}{u} \quad (22)$$

In the fin stabilization systems, the gyroscope sensor measures the roll velocity signal which is then passed to the PID controller. If we take into account that the control input signal is heel angle then we became a PDD² control law given by

$$u(t) = K_p\phi(t) + K_D\dot{\phi}(t) + K_{D^2}\ddot{\phi}(t) \quad (23)$$

It is important to limit the fin deflection angle in proportion to the ship speed, to avoid the phenomenon of cavitation. The permissible angle of the fins depends on so-called fin ratio λ_F which reflects the relationship between the length and its width. The maximum fin deflection should not exceed 30 degrees.

Since the fin stabilizer is adopted for the roll reduction, the stabilizer fin-induced force and moment must be derived and the related formulas are summarized as below (Fang et al. 2010)

$$X_{FF} = -F_{RD} - F_{LD} \quad (24)$$

$$Y_{FF} = F_{RL}\cos\phi_F - F_{LL}\cos\phi_F \quad (25)$$

$$Z_{FF} = -F_{RL}\sin\varphi_F - F_{LL}\sin\varphi_F \quad (26)$$

$$\begin{aligned} K_{FF} &= -z_F \cdot [F_{RL}\cos\phi_F - F_{Ll}\cos\phi_F) + \\ &\quad + y_F \cdot (F_{RL}\sin\phi_F - F_{LL}\sin\phi_F)] \end{aligned} \quad (27)$$

$$M_{FF} = z_F \cdot X_{FF} \quad (28)$$

$$N_{FF} = x_F \cdot Y_{FF} \quad (29)$$

where the subscripts *RL*, *RD*, *LL* and *LD* represent the fin-induced lift force (*L*) and drag force (*D*) on right-hand side (*R*) and left-hand side (*L*), respectively. x_F, y_F, and z_F, are the coordinates of the action center of the fin force, φ_F the angle between fin and the plumb line.

One of the key issues in control of stabilizers is the adaptation to the changes in the environmental conditions. A classic control system of fin stabilizers using a PDD² controller is able to provide the correction of the phase shift between the wave and stabilizing moment.

The control signal components of proportional, derivative and double derivative part depend on the wave frequency what allows them to change the phase lag in a limited interval according to the varying wave frequency.

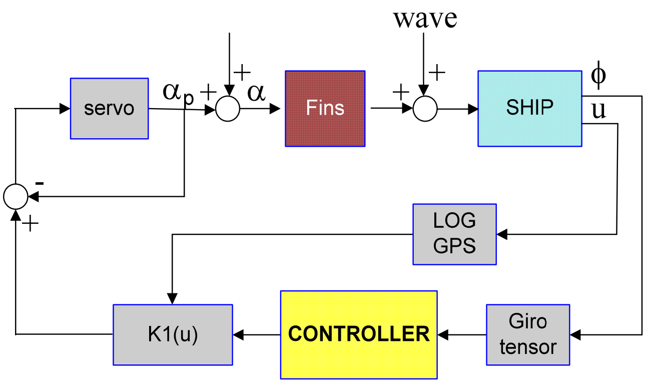

Figure 7. Schema of the control system for fin stabilizers.

Classic control system of fin stabilizers used a PDD² controller that was able to provide optimal phase shift since the stabilization, so that it was in opposite until you require.

Alarçin (2007) designed for fin stabilization an internal model control system. He used three-layer neural network with back propagation to build an inverse model of the roll dynamic. The idea of IMC implementation in fin stabilization realized also Tzeng and Wu (2000). They introduced additionally to the internal model the inverse of nonlinearities such as saturation of fin angle. Generally the lift generates when the flow passes the fin but Zhou et al.(2008), that a different mode of fin should be applied to generate on the fin enough lift to reduce ship roll at anchor .

The fin stabilizers can reach over 90 % roll damping in regular waves. In irregular waves, it is lower but it can be increased by using a cascade control structure which task is to adjust the fin deflection to achieve the desired effective angle of attack of the water flow on the fin. (Kula 2014).

Jin et al. (2008) describe the property of fins at zero speed to generate the lift which is normal to the surface and is in direct proportion to tonnage per unit time. The fin stabilizer is considered as a plane for analyzing this problem easily. Such use of the fins is defined as so-called fin stabilizer system at anchor.

3.4 *Rudder roll stabilization*

The effect of rudder inclination on the generation of heel has been recognized relatively late, when the crew of the vessel SS American Resolute started complaining about discomfort during ship steering.

When the rudder was steered, the ship heeled in inward direction due to the rolling moment acting on the rudder.

This phenomenon is illustrated in Figure 8

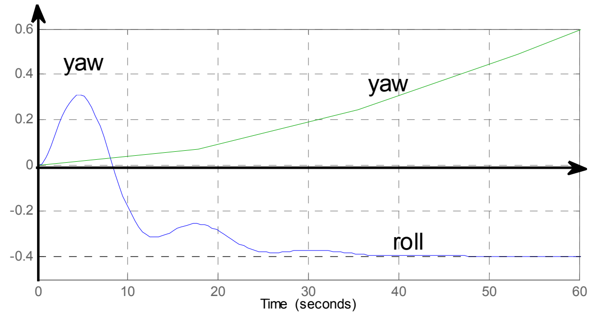

Figure 8. Yaw and roll responses in reaction of rapid rudder angle change (U=18 kn)

In order to find the reason for this effect, the above case was examined in detail. The examination resulted in developing the RRS system which extended the operation of the autopilot by introducing the function which compensated the effect of waves on ship roll motion. The rudder angle δ required for the both functions can be expressed as

$$\delta = \delta_{\psi} + \delta_{\phi} \qquad (30)$$

where δ_{ψ}, δ_{ϕ}, are applied for course stabilization and roll damping , respectively

The use of one control signal for the purpose of two outputs is possible due to the difference in scale between the dominating time constants, while reaching the expected result required increasing the speed of the steering gear action. The standard speed of rudder blade rotation was from 3°/s on merchant ships, up to 5-7°/s on Navy vessels. The requirements formulated for the actuators in this control system amount to 12°/s, the minimum (Roberts 2008). The fins are used only for roll stabilisation, and should interfere very little with the heading.On the contrary, rudders have a great influence on roll motions, but are primary used to control the yaw,

A design of this system was presented by Cowley and Lambert (1972). It's positive that between the heel caused by rudder and yaw there is the vast separation of frequencies. Fast and a short movement of the rudder in order to compensate heel has a negligible impact on the change of course (Roberts et al. 1997).

The RRS system was the subject of numerous publications. Moreover, this process, very interesting from the point of view of the theory of control, was complemented by designs of control systems making use of H_∞ norm, as well as the Model Reference Adaptive Control and the Model Predictive Control. The discrete model which makes it possible to predict course changes and/or roll motion can be presented using the equation:

$$x(n) = \sum_{m=1}^{M} a(m)x(n-m) + \sum_{m=1}^{M} b(m)\ Y(n-m) + u(n) \qquad (31)$$

where x(n) –denotes a 2-dimensional state vector whose components are yaw and roll motion, Y(n) – rudder motion and u(n) denotes a white noise vector, M-the order of the model /obtained by the procedure (Akaike & Nakagawa, 1994)/, a(m), b(m) - coefficients

Part of these systems underwent sea trials and were successfully implemented on ships. The success of the rudder roll stabilization was the motivation for making attempt to integrate the roll motion stabilization systems which make use of the main rudder and side fins. This system is referred to as the Integrated Fin and Rudder Roll Stabilization (INFRRS). Its advantage is that the requirements formulated for the speed of operation of the steering gear are not as restrictive as in the previous case, as even at lower rudder revolutions the RRS system can improve the effect of operation of the stabilizing fins. The results of this cooperation, supported by sea trials, were presented by Roberts et al. (1997)

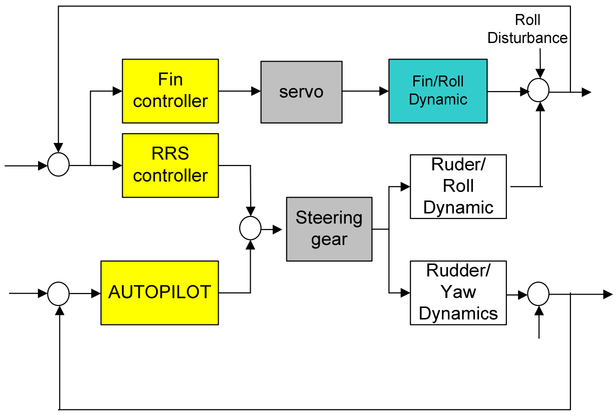

Figure 9. The integrated fin and rudder roll stabilization control system

The block scheme of this system is shown in Figure 9. Perez (2005) presented a concept of the integrated fin steering with rudder assistance which made use of the model predictive controller. Law et al. (2005) have made a comparison several combinations of controls to the Sliding Mode Control (SMC), PID, dual Loop Transfer Recovery (LTR) controllers working in this system. It was possible using the LTR controller to reduce the ship roll in 30 %. Crossland (2002) shows on example an ASW frigate that an IFRRS system indicates an 3.8 % improvement rather than a standard fin arrangement. Agarwal (1997) proposed a control design for this system using H∞ approach. Oda et al. (1996) discussed a possible compromise when realizing two RRS goals in the statistical approach, which was keeping the ship's course as the main goal, and additionally reducing the occurring roll movements. To smooth the rudder movements, the multivariable auto-regressive rudder roll control system MARCS takes into account the operation of the steering gear in the steering quality criterion. The quality criterion J_p (Oda et al., 1996) formulated in the above way aims at limiting three undesired quantities: the first is the deflection of the roll motion and ship's course from the set values, the second is the amount of rudder motion, and the third the rate of change of the steering gear.

$$J_p = \left[\sum_{n=1}^{p} \left\{ \begin{array}{l} X(n)^t QX(n) + Y(n-1)^t RY(n-1) + \\ \left(Y(n-1) - Y(n-2)^t\ T(Y(n-1) - Y(n-2)\right) \end{array} \right\} \right] \qquad (32))$$

where T- weighting matrix

In this formulation the third term is the penalization of inclination rudder angle changes. In this case the control rule is obtained from the relation:

$$Y(n) = GZ(n) + FY(n-1) \tag{33}$$

where: *G* and *F* are the optimal gain and smoothing coefficients.

If the weighting matrices *T* take the zero value, then the criterion function is reduced to the quadratic criterion. To make use of the structure of the primary autopilot installed on the ship, the MARCS was installed in the emergency circuit of the autopilot, as shown in Figure 10.

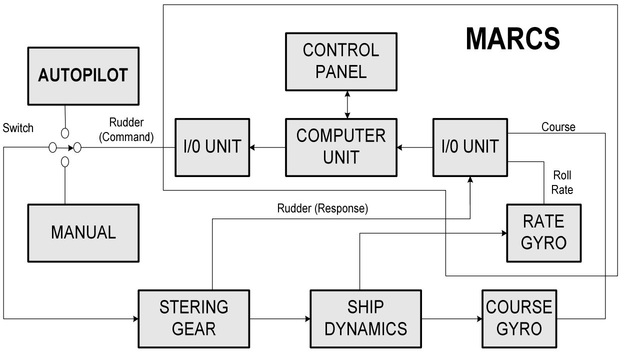

Figure 10. Block scheme of the rudder/roll control system.

The processor unit comprises the computer, the interface, and the roll motion speed sensors. The operating unit provides opportunities for selecting one out of three control gains, and setting the selected course. This way, by switching on the mode switch in the operating unit we can easily choose one of two available control methods: the primary autopilot or MARCS. Moreover, in cases when abnormal conditions occur in the MARCS system, we can easily switch it off and continue the steering with the aid of the autopilot or manually. The roll motion can be reduced in this system by 30-50%. Linear-quadratic regulator LQR it is the quadratic criterion and robust control were proposed (Sharif et al. 1995) to provide roll stabilization with 30 deg/s fins and a 6 deg/s rudder. The model predictive control with the effective attack angle constraint was used by Perez and Goodwin (2008) to prevent dynamic stall of fin stabilizers. Several automated gain tuning algorithms were suggested in order to improve the performance of rudder roll stabilization controllers in saturation, including the automatic gain controller (AGC) (v.d. Klugt 1987),(v.d. Amerongen & v.d. Klugt 1990) and the time-varying gain reduction (TGR) algorithm. The model predictive control was applied to the rudder roll problem by Perez (2005). The internal model control (IMC) making use of neural networks was investigated by Alarcin (2007) who used fin stabilizers to obtain 94% roll angle reduction. A third-order controller for a fin stabilizer roll reduction system was reported by Tzeng and Wu (2000) as applying the maximum of 38 dB feedback.

Figure 11. Azimuthing propulsion.

In the case of the track keeping and roll damping Fang & Luo (2006) composed their control system of sliding mode controller. Controlling the motion of the steering blade with the aid of the robust adaptive fuzzy control (RAFC) and its application to ship roll stabilization were presented by Yang, Zhou & Jia (2002).Similar cooperation of two propellers to that observed in INFRRS was proposed by Lee et al (2011). The use of INFRRS improves the damping if the ship sails at a moderate speed, as the minimum. At low maneuvering speeds their efficiency is low. The situation is different in case of a pod propulsion system which can support the action of stabilization fins in the range when their efficiency is already very low. A mathematical model of the force due to the pod propeller build in (Stettler & Hover 2004). The force components: the normal force F^N and the thrust force F^T can be expressed as a function of the propeller RPS, propeller diameter, thrust and torque coefficients:

$$F^N = \rho n^2 D^4 K_N(\alpha_{Pod}, J_{Pod}) \tag{34}$$

$$F^T = \rho n^2 D^4 K_T(\alpha_{Pod}, J_{Pod}) \tag{35}$$

where: n - resolution per second, D -propeller diameter K_T K_N –non-dimensional coefficients

The surge and sway force, roll moment and yaw moment of the pod thruster should be added according to the equations 1, 2, 4, 6 respectively.

$$X_P = F^T(\cos\delta_1 + \cos\delta_2) - F^N(\sin\delta_1 + \sin\delta_2) \tag{36}$$

$$Y_P = F^T(\sin\delta_1 + \sin\delta_2) + F^N(\cos\delta_1 + \cos\delta_2) \tag{37}$$

$$K_P = -Y_P \times r_P \tag{38}$$

$$N_P = -Y_P \times x_P \tag{39}$$

where:

δ_1, δ_2 – the angles of the propeller

r_p , x_p – the vertical and longitudinal distance between the center of gravity of the ship and the center of the propeller

Steering actions the azimuthing thrusters and fin unit requires a two-dimensional control system (TITO). The nominal plant and the frequency-weighted linear-quadratic regulator LQR are applied to reduce the roll motion in irregular waves. The roll motion of ships is effectively reduced when the fin and pod propeller are used as the control actuators at low speeds.

In regular waves by u=7 kn the fins compensated the rolling motion to 25% , pod propellers to 38% and both they reached 52 % of the amplitude of the non stabilized ship

4 SUMMARY

The choice of stabilizer depends on many ship and mission considerations. The large number of existing stabilizers makes it possible to find a stabilizer for virtually every conceivable mission, be it low speed trawling to high speed pursuit. The question of whether or not to have a stabilizer depends not upon the availability of stabilizers, but rather on whether or not a particular stabilizer will be useful. This can be determined by finding the increase in operability relative to some motion criterionIt would seem that in a changing environment in which is carried out the roll stabilization finds wide application the adaptive control. However, the phenomenon of resonance requires that the classic linear controller according to Minorsky's theory is tuned to a frequency close to the natural frequency of the ship.

A potential research activity has a control system design of free surface tanks, which could realize the adaptation to the changes in the sea environment but the effectiveness of these stabilizers independently of the control is relatively low.

Through adaptation of PI/PID controller settings it may be possible that the current stabilizing moment by this wave frequency counteracts the excitation moment.

For some vessels of varying over a wide range dynamic, it may be desirable to adapt the controller to the new natural frequency of the ship. This requires the identification, which under the influence of disturbances can cause significant number of difficulties.

Researchers in the works devoted to the synthesis of stochastic stabilization systems despite the characters of the dominant disturbances rarely come across a probabilistic approach control. It appears that the use for example of minimum variance strategy, which the objective is to minimize the steady-state output variances would be justified.

The adaptive minimum variance control can be used in predictive form too.

To control a ship motion for certain operating conditions, a particular controller may yield a most suitable performance. Therefore, a set of different types of controllers should be designed depending on various speeds, environmental and sea conditions so that appropriate controllers are selected in correspondence to these conditions. If it is difficult to obtain satisfactory results using one controller, it can turn to switched control techniques what is implemented for instance in some devices of an integrated fin and rudder roll stabilization

As shown by test results presented in the cited papers in the control of stabilizers systems may also be successfully used various model-based controllers for instance the model predictive controllers, fuzzy logic and artificial neural networks controllers. In recent years were tested Linear Quadratic Gaussian (LQG), as well Loop Transfer Recovery (LTR) procedure.

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