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The undesirable effects of roll motion of ships (rocking about the longitudinal axis) became noticeable in the mid-19th century when significant changes were introduced to the design of ships as a result of sails being replaced by steam engines and the arrangement being changed from broad to narrow hulls. The combination of these changes led to lower transverse stability (lower restoring moment for a given angle of roll) with the consequence of larger roll motion. The increase in roll motion and its effect on cargo and human performance lead to the development several control devices that aimed at reducing and controlling roll motion. The control devices most commonly used today are fin stabilisers, rudder, anti-roll tanks and gyrostabilisers. The use of different type of actuators for control of ship roll motion has been amply demonstrated for over 100 years. Performance, however, can still fall short of expectations because of difficulties associated with control system design, which have proven to be far from trivial due to fundamental performance limitations and large variations of the spectral characteristics of wave-induced roll motion. This short article provides an overview of the fundamentals of control design for ship roll motion reduction. The overview is limited to the most common control devices. Most of the material is based on Perez (2005), and Perez and Blanke (2012).

1 Ship Roll Motion Control Techniques

One of the most commonly devices used to attenuate ship motion are the fin stabilisers. These are small controllable fins located on the bilge of the hull usually amid ships. These devices attain a performance in the range of 60%-90% of roll reduction (root mean square) (Sellars and Martin, 1992). They require control systems that sense the vessel's roll motion and act by changing the angle of the fins. These devices are expensive, introduce underwater noise that can affect sonar performance, they add to propulsion losses and they can be damaged. Despite this, they are among the most commonly used ship roll motion control device. From a control perspective, highly nonlinear effects (dynamic stall) may appear when operating in severe sea states and heavy rolling conditions (Gaillarde, 2002).

During studies of ship damage stability conducted in late 1800s it was observed that under certain conditions the water inside the vessel moved out of phase with respect to the wave profile; and thus, the weight the water on the vessel counter-acted the increase of pressure on the hull, hence reducing the net roll excitation moment. This led to the development of fluid anti-roll tank stabilisers. The most common type of anti-roll tank is the U-tank, which comprises two reservoirs, located one on port and one on starboard, connected at the bottom by a duct. Anti-roll tanks can be either passive or active. In passive tanks, the fluid flows freely from side to side. According to the density and viscosity of the fluid used, the tank is dimensioned so that the time required for most of the fluid to flow from side to side equals the natural roll period of the ship. Active tanks operate in a similar manner, but they incorporate a control system that modifies the natural period of the tank to match the actual ship roll period. This is normally achieved by controlling the flow of air from the top of one reservoir to the other. Anti-roll tanks attain a medium to high performance in the range of 20%-70% of roll angle reduction (RMS) (Marzouk and Nayfeh, 2009). Anti-roll tanks increase the ship displacement. The can also be used to correct list (steady-state roll angle) and they are the preferred stabiliser for ice-breakers.

Rudder roll stabilisation (RRS) is a technique based on the fact that the rudder is located not only aft, but also below the centre of gravity of the vessel, and thus the rudder imparts not only yaw but also roll moment. The idea of using the rudder for simultaneous course keeping and roll reduction was conceived in the late 1960s by observations of anomalous behaviour of autopilots that did not have appropriate wave filtering—a feature of the autopilot that prevents the rudder from reacting to every single wave; see for example Fossen and Perez (2009) for a discussion on wave filtering. Rudder roll stabilisation has been demonstrated to attain medium to high performance in the range of 50%-75% of roll reduction (RMS) Baitis et al. (1983); Källström et al. (1988); Blanke et al. (1989); van Amerongen et al. (1990); Oda et al. (1992). The upgrade of the rudder machinery is required to be able to attain slew rates in the range 10-20 deg/s for RRS to have sufficient control authority.

A gyrostabiliser uses the gyroscopic effects of large rotating wheels to generate a roll reducing torque. The use of gyroscopic effects was proposed in the early 1900s as a method to eliminate roll, rather than to reduce it. Although the performance of these systems was remarkable, up to 95% roll reduction, their high cost, the increase in weight and large stress produced on the hull masked their benefits and prevented further developments. However, a recent increase in development of gyrostabilisers has been seen in the yacht industry (Perez and Steinmann, 2009).

Fins and rudder give rise to lift forces in proportion to the square of flow velocity past the fin. Hence, roll stabilization by fin or rudder is not possible at low or zero speed. Only U-tanks and gyro devises are able to provide stabilization in these conditions. For further details about the performance of different devices see Sellars and Martin (1992), and for a comprehensive description of the early development of devices see Chalmers (1931).

2 Modelling of Ship Roll Motion for Control Design

The study of roll motion dynamics for control system design is normally done in terms of either one or four degrees of freedom (DOF) models. The choice between models of different complexity depends on the type of motion control system considered.

For a one-degree-of-freedom (1DOF) case, the following model is used

$$\begin{aligned} \dot{\phi} &= p, && (1) \\ I_{xx}\,\dot{p} &= K_h + K_w + K_c, && (2) \end{aligned}$$

where ϕ is roll angle, p is roll rate, I_{xx} is rigid-body moment of inertia about the x-axis of a body-fixed coordinate system, where K_h is hydrostatic and hydrodynamic torques, K_w torque generated by wave forces acting on the hull, and K_c the control torques. The hydrodynamic torque can be approximated by the following parametric model: $K_h \approx K_{\dot{p}}\,\dot{p} + K_p p + K_{p|p|}\,p|p| + K(\phi)$. The first term represents a hydrodynamic torque in roll due to pressure change that is proportional to the roll accelerations, and the coefficient $K_{\dot{p}}$ is called roll added mass (inertia). The second term is a damping term, which captures forces due to wave making and linear skin friction, and the coefficient K_p is a linear damping coefficient. The third term is a nonlinear damping term, which captures forces due to viscous effects. The last term is the restoring torque due to gravity and buoyancy.

For a 4DOF model (surge, sway, roll, and yaw), motion variables considered are $\boldsymbol{\eta} = [\phi\ \psi]^{\mathsf{T}}$, $\boldsymbol{\nu} = [u\ v\ p\ r]^{\mathsf{T}}$, $\boldsymbol{\tau}_i = [X\ Y\ K\ N]^{\mathsf{T}}$, where ψ is the yaw angle, the body-fixed velocities are u-surge, v-sway, and r is the yaw rate. The forces and torques are X-surge, Y-sway, K-roll, and

N-yaw. With these variables, the following mathematical model is usually considered:

$$\dot{\boldsymbol{\eta}} = \mathbf{J}(\boldsymbol{\eta})\,\boldsymbol{\nu}, \tag{3}$$

$$\mathbf{M}_{RB}\,\dot{\boldsymbol{\nu}} + \mathbf{C}_{RB}(\boldsymbol{\nu})\boldsymbol{\nu} = \boldsymbol{\tau}_h + \boldsymbol{\tau}_c + \boldsymbol{\tau}_d, \tag{4}$$

Where, $\mathbf{J}(\boldsymbol{\eta})$ is a kinematic transformation, $\mathbf{M}_{RB}$ is the rigid-body inertia matrix that corresponds to expressing the inertia tensor in body-fixed coordinates, $\mathbf{C}_{RB}(\boldsymbol{\nu})$ is the rigid-body Coriolis and centripetal matrix, and $\boldsymbol{\tau}_h$, $\boldsymbol{\tau}_c$, and $\boldsymbol{\tau}_d$ represent the hydrodynamic, control, and disturbance vector of force components and torques, respectively.

The hydrostatic and hydrodynamic forces are $\boldsymbol{\tau}_h \approx -\mathbf{M}_A\,\dot{\boldsymbol{\nu}} - \mathbf{C}_A(\boldsymbol{\nu})\boldsymbol{\nu} - \mathbf{D}(\boldsymbol{\nu})\boldsymbol{\nu} - \mathbf{K}(\phi)$. The first two terms have origin in the motion of a vessel in an irrotational flow in a non-viscous fluid. The third term corresponds to damping forces due to potential (wave making), skin friction, vortex shedding, and circulation (lift and drag). The hydrodynamic effects involved are quite complex, and different approaches based on superposition of either odd-term Taylor expansions or square modulus ($x|x|$) series expansions are usually considered Abkowitz (1964) and Fedyaevsky and Sobolev (1964). The $\mathbf{K}(\phi)$ term represents the restoring forces in roll due to buoyancy and gravity. The 4DOF model captures parameter dependency on ship speed sa well as the couplings between steering and roll and it is useful for controller design.

2.1 Wave-disturbance Models

The action of the waves creates changes in pressure on the hull of the ship, which translate into forces and moments. It is common to model the ship motion response due to waves within a linear framework, and to obtain two frequency-response functions (FRF): wave-to-excitation $F_i(j\omega, U, \chi)$ and wave-to-motion $H_i(j\omega, U, \chi)$ response functions; where i indicates the degree of freedom. These FRF depend on the wave frequency, the ship speed, and the angle χ at which the waves encounter the ship—this is called the encounter angle.

The wave elevation in deep water is approximately a stochastic process that is zero-mean, stationary for short periods of time, and Gaussian (Haverre and Moan, 1985). Under these assumptions, the wave elevation ζ is fully descried by a power spectral density $\Phi_{\zeta\zeta}(\omega)$. With a linear response assumption, the power spectral density of wave to excitation force and wave to motion can be expressed as

$$\Phi_{FF,i}(j\omega) = |F_i(j\omega, U, \chi)|^2 \Phi_{\zeta\zeta}(j\omega), \qquad \Phi_{\eta\eta,i}(j\omega) = |H_i(j\omega, U, \chi)|^2 \Phi_{\zeta\zeta}(j\omega).$$

These spectra are models of the wave-induced forces and motions,respectively, from which it its common to generate either time series of wave-excitation forces in terms of the encounter frequency to be used as input disturbances in simulation models or time series of wave-induced motion to be used as output disturbance, see, for example, Perez (2005) and references herein.

3 Roll Motion Control and Performance Limitations

The analysis of performance of ship roll motion control by means of force actuators is usually conducted within a linear framework by linearising the models. For a SISO loop where the wave-induced roll motion is considered an output disturbance, the Bode integral constraint applies. This imposes restrictions on one's freedom to shape the closed-loop transfer function to attenuate the motion due to the wave induced forces in different frequency ranges. These results have important consequences on the design of a roll-motion control system since the frequency of the waves seen from the vessel change significantly with the sea state, the speed of the vessel, and the wave encounter angle. The changing characteristics on open-loop roll motion in conjunction with the Bode integral constraint makes the control design challenging since roll amplification may occur if the control design in not done properly. For some roll motion control problems, like using the rudder for simultaneous roll attenuation and heading control, the system presents non-minimum phase dynamics. In this case, the trade off of reduced sensitivity *vs* amplification of roll motion is dominating at frequencies close to the non-minimum phase zero—a constraint with origin in the Poisson Integral (Hearns and Blanke, 1998); see also Perez (2005).

It should be noted that non-minimum phase dynamics also occurs with fin stabilisers, when the stabilisers are located aft of the centre of gravity. With the fins at this location, they behave like a rudder and introduce non-minimum phase dynamics and heading interference at low wave-excitation frequencies. These aspects of fin location were discussed by Lloyd (1989).

The above discussion highlights general design constraints that apply to roll motion control systems in terms of the dynamics of the vessel and actuator. In addition to these constraints, one needs also to account for limitations in actuator slew rate and angle.

4 Controls Techniques used in Different Roll Control Systems

4.1 Fin stabilisers

In regard to fin stabilisers, the control design is commonly address using the 1DOF model (1)-(2). The main issues associated with control design are the parametric uncertainty in model and the Bode integral constraint. This integral constraint can lead to roll amplification due to changes in the spectrum of the wave-induced roll moment with sea state and sailing conditions (speed and encounter angle). Fin machinery is designed so that the rate of the fin motion is fast enough and actuator rate saturation is not an issue in moderate sea states. The fins could be used to correct heeling angles (steady state roll) when the ship makes speed, but this is avoided due to added resistance. If it is used, integral action needs to include anti-windup. In terms of control strategies, PID, $\mathcal{H}_\infty$, and LQR techniques have been successfully applied in practice. Highly nonlinear effects (dynamic stall) may appear when operating in severe sea states and heavy rolling conditions, and proposals for applications of model predictive control have been put forward to constraint the effective angle of attack of the fins. In addition, if the fins are located too far aft along the ship, the dynamic response from fin angle to roll can exhibit non-minimum phase dynamics, which can limit the performance at low encounter frequencies. A thorough review of the control literature can be found in Perez and Blanke (2012).

4.2 Rudder-roll stabilisation

The problem of rudder-roll stabilisation requires the 4DOF model (3)-(4), which captures the interaction between roll, sway and yaw together with the changes in the hydrodynamic forces due to the forward speed. The response from rudder to roll is non-minimum phase (NMP) and the system is characterised by further constraints due to the single-input-two-output nature of the control problem—attenuate roll without too much interference with the heading. Studies of fundamental limitations due to NMP dynamics have been approached using standard frequency domain tools by Hearns and Blanke (1998) and Perez (2005). A characterisation of the trade-off between roll reduction vs increase of interference was part of the controller design in Stoustrup et al. (1994). Perez (2005) determined the limits obtainable using optimal control with full disturbance information. The latter also incorporated constraints due to the limiting authority of the control action in rate and magnitude of rudder machinery and stall conditions of the rudder. The control design for rudder-roll stabilisation has been addressed in practice using PID, LQG, and $\mathcal{H}_\infty$, and standard frequency-domain linear control designs. The characteristics of limited control authority were solved by van Amerongen et al. (1990) using automatic gain control. In the literature, there have been proposals put forward for the use of model predictive control, QFT, sliding-mode nonlinear control, auto-regressive stochastic control. Combined use of fin and rudder has also be investigated. Grimble et al. (1993) and later Roberts et al. (1997) used $\mathcal{H}_\infty$ control techniques. Thorough comparison of controller performances for warships was published in Crossland (2003). A thorough review of the control literature can be found in Perez and Blanke (2012).

4.3 Gyro stabilisers

Using a single gimbal suspension gyro stabiliser for roll damping control, the coupled vessel-roll-gyro model can be modelled as follows:

$$\dot{\phi} = p, \tag{5}$$

$$K_{\dot{p}}\,\dot{p} + K_p\,p + K_\phi\,\phi = K_w - K_g\dot{\alpha}\cos\alpha \tag{6}$$

$$I_p\ddot{\alpha} + B_p\dot{\alpha} + C_p\sin\alpha = K_g\,p\cos\alpha + T_p, \tag{7}$$

where (6) represents the 1DOF roll dynamics, and (7) represents the dynamics of the gyrostabiliser about the axis of the gimbal suspension, where α is the gimbal angle, equivalent to the the precession angle for a single gimbal suspension, I_p is gimbal and wheel inertia about the gimbal axis, B_p is the damping, and C_p is a restoring term of the gyro about the precession axis due to location of the gyro centre of mass relative to the precession axis (Arnold and Maunder, 1961). T_p is the control torque applied to the gimbal. The use of twin counter spinning wheels prevents gyroscopic coupling with other degrees of freedom. Hence, the control design for gyrostabilisers can be based on a linear single degree of freedom model for roll.

The wave-induced roll moment K_w excites the roll motion. As the roll motion develops, the roll rate p induces a torque along the precession axis of the gyrostabiliser. As the precession angle α develops, there is reaction torque done on the vessel that opposes the wave-induced moment. The later is the roll stabilising torque, $X_g \triangleq -K_g\dot{\alpha}\cos\alpha \approx -K_g\dot{\alpha}$. This roll torque can only be controlled indirectly through the precession dynamics in (7) via T_p. In the model above, the spin angular velocity ω_{spin} is controlled to be constant, hence the wheels' angular momentum $K_g = I_{spin}\,\omega_{spin}$ is constant.

The precession control torque T_p is used to control the gyro. As observed by Sperry (Chalmers, 1931), the intrinsic behaviour of the gyrostabiliser is to use roll rate to generate a roll torque. Hence, one could design a precession torque controller such that from the point of view of the vessel, the gyro behaves as damper. Depending on how precession torque is delivered, it may be necessary to constraint precession angle and rate. This problem has been recently considered in Donaire and Perez (2013) using passivity-based control.

4.4 U-tanks

U-tanks can be passive or active. Roll reduction is achieved by attempting to transfer energy from the roll motion to motion of liquid within the tank and using the weight of the liquid to counteract the wave excitation moment. A key aspect of the design is the dimension and geometry of the tank to ensure that there is enough weight due to the displaced liquid in the tank and that the oscillation of the fluid in the tank matches the vessel natural frequency in roll, see Holden and Fossen (2012) and references herein. The design of the U-tank can ensure a single frequency matching, at which the performance is optimised, and for this frequency the roll natural frequency is used. As the frequency of roll motion departs from this, a degradation of roll reduction occurs. Active U-tanks use valves to control the flow of air from the top of the reservoirs to extend the frequency matching in sailing conditions in which the roll dominant frequency is lower than the roll natural frequency – the flow of air is used to delay the motion of the liquid from one reservoir to the other. This control is achieved by detecting the dominant roll frequency and using this information to control the air flow from one reservoir to the other. If the roll dominant frequency is higher than the roll natural frequency, the U-tank is used in passive mode, and the standard roll reduction degradation occurs.

5 Summary and Outlook

This article provided a brief summary of control aspects for the most common ship roll-motion control devices. These aspects include the type of mathematical models used to design and analyse the control problem, the inherent fundamental limitations and constraints that some of the designs are subjected to, and the performance that can be expected from the different devices. As an outlook, one of the key issues in roll motion control is the model uncertainty and the adaptation to the changes in the environmental conditions. As the vessel changes speed and heading, or as the seas builds up or abate, the dominant frequency range of the wave-induced forces change significantly. Due to the fundamental limitations discussed, a non-adaptive controller may produce roll amplification rather than roll reduction. This topic has received some attention in the literature via multi-mode control switching, but further work in this area could be beneficial. In the recent years, new devices have appeared for stabilisation at zero speed, like flapping fins and rotating cylinders. Also the industry's interest in roll gyrostabilisers have been re-ignited. The investigation of control designs for these devices has not yet received much attention within the control community. Hence, it is expected that this will create a potential for research activity in the future.

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