Ship Roll Motion Control

Plenary Talk Presented at the 8th IFAC Conference on Control Applications in Marine Systems,
Sept 15th-17th, Rostock, Germany

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Motivation

The effects of roll on ship performance became more noticeable in the mid-19th century when:

- Sails were replaced by engines
- Broad arranges changed to turrets

W. Froude (1819-1879)

Many devices have been proposed leading to interesting control problems

<u>Performance can easily fall short of expectations because of deficiencies in control system design due to</u>

- Fundamental limitations due to system dynamics
- Limited actuator authority
- Disturbances with large changes in spectral characteristics

Outline

- Ship Roll Control Devices
  - Working Principles,
  - Performance,
  - Historical aspects
- Key aspects of ship roll dynamics for control design
  - Models,
  - Wave-induced motion
  - Simulation and control design models
- Control System Design
  - Objectives,
  - Fundamental limitations,
  - Control strategies
- Research Outlook
  - Unsolved problems, new devices

I - Ship Roll Control Devices

Performance,

Working Principles,

and Historical Aspects

Ship Performance and Roll Motion

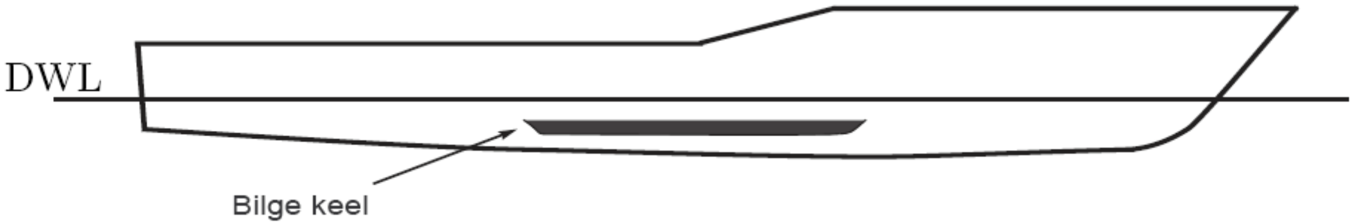
Ship roll motion

- Affects crew performance
- Can damage cargo
- May prevent the use of on-board equipment

From a ship operability point of view is necessary to reduce not only roll angle but also roll accelerations.

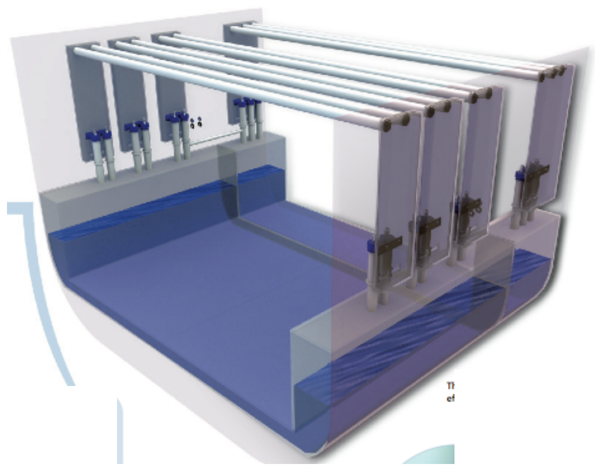
Bilge Keels

- 10to 20% roll reduction (RMS)
- Low maintenance
- No control
- No occupied space
- Low price easy to install
- Increase hull resistance when damping is not needed
- Not every vessel can be fitted with them (ice-breakers)

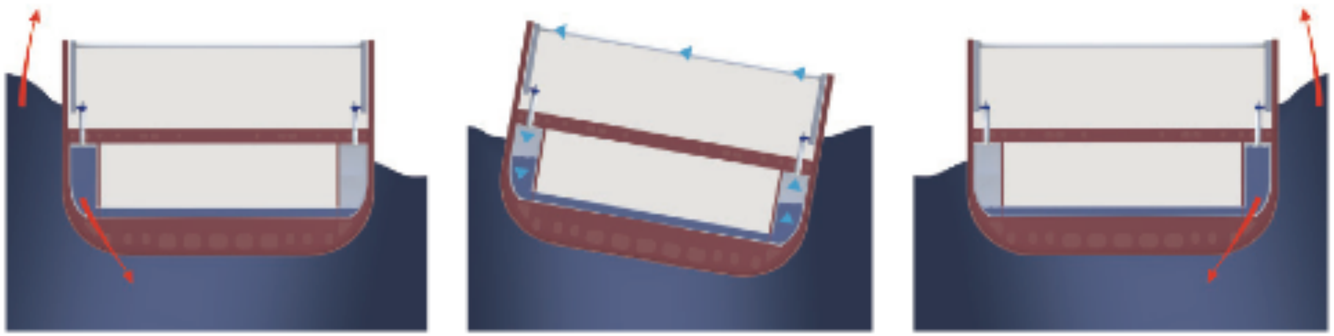


U-tanks

- 40 to 50% roll reduction (RMS)
- Active/Passive
- Independent of the vessel speed
- Anti heeling
- Heavy
- Occupy large spaces
- Affects stability due free-surface



Intering Rolls-Royce



Gyro-stabilisers

- 60% to 90% roll reduction (RMS)
- Performance independent of the vessel speed

1906
Torpedo Boat

1914 to 1925
active control

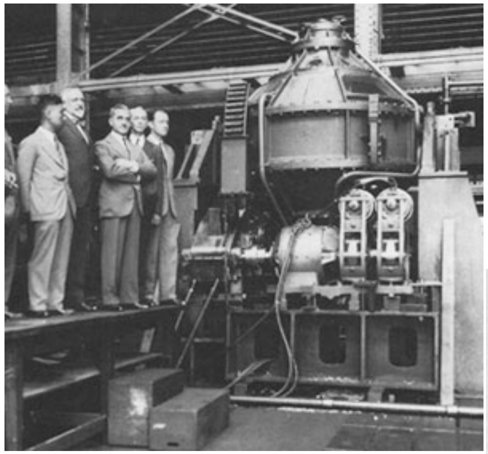
1914,1924
Twin Gyros

Today

Time

Schlick
(Germany)

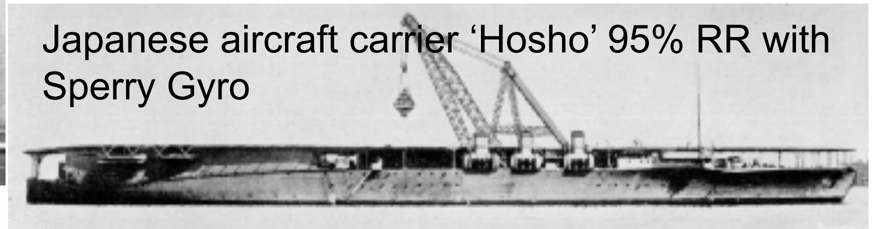
Sperry
(USA)



Conte di Savoia 1932

Fieux
(France)

Halcyon
Ferretti
Sea Gyro
Shipdynamics
Seakeeper

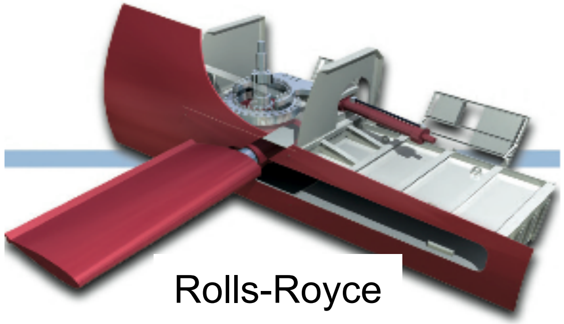


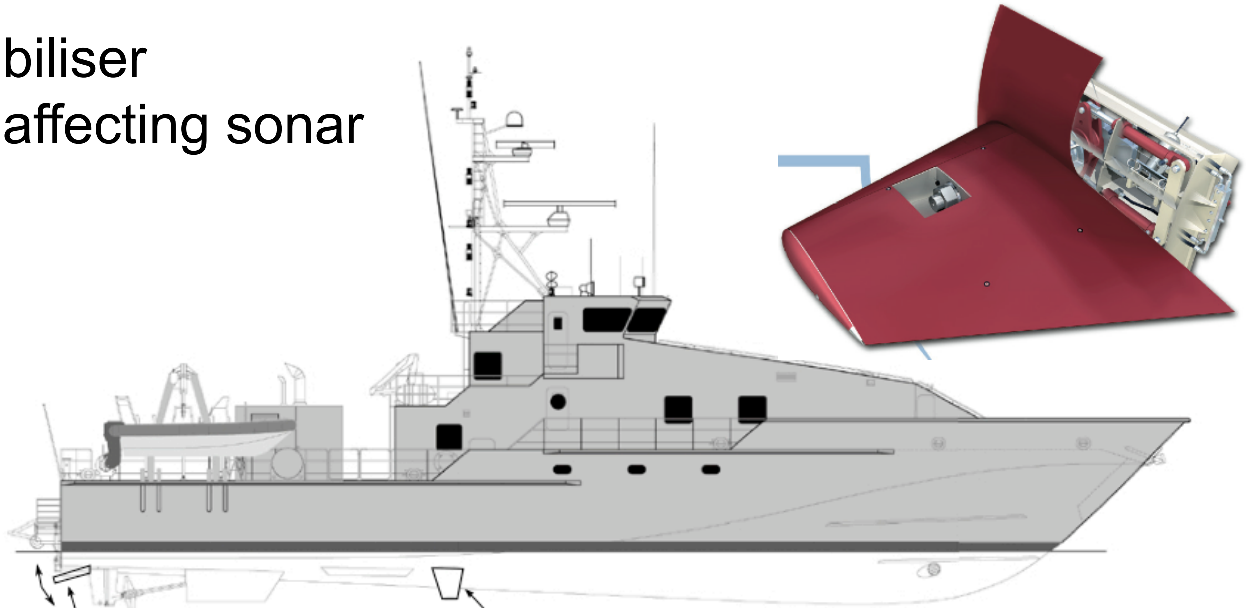
Japanese aircraft carrier ‘Hosho’ 95% RR with Sperry Gyro

Fin-stabilisers

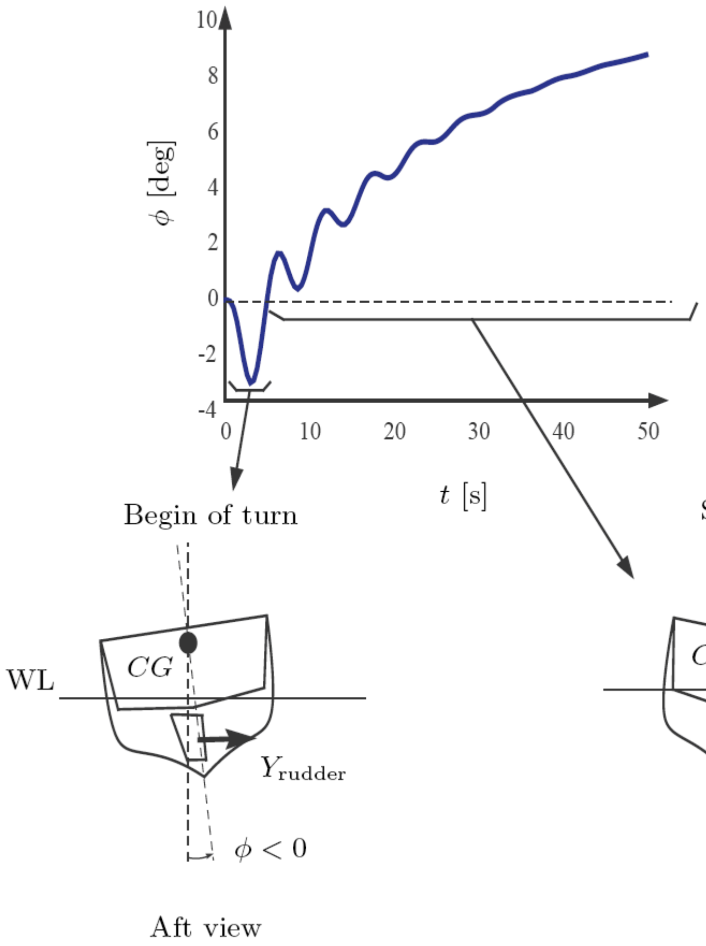
- 60% to 90% roll reduction (RMS)
- Performance depends on speed
- Control is important for performance

- Easy to damage
- Most Expensive stabiliser
- Can produce noise affecting sonar
- Dynamic stall





Rudder Roll Stabilisers (RRS)



- 40% to 70% roll reduction (RMS)
- Performance depends on speed
- Control is important for performance

- Special rudder machinery
- Fundamental limitations in control due to non-minimum phase dynamics (NMP)

Historical Aspects

Year	Device	Ship	Designer	Type
1870	Bilge keels	-	Froude (GBr)	Passive
1880	Tanks	*Inflexible*	Watt and Froude (GBr)	Passive
1891	Weight	*Cecile*	Thornycroft (GBr)	Active
1906	Gyro	*Sea-Bar*	Schlick (Ger)	Passive
1909	Weight	Steamer	Crémieu (Fra)	Passive
1910	U-tank	*Ypiranga*	Frahm (Ger)	Passive
1915	Gyro	*Conte di Savoia*	Sperry Company (USA)	Active
1924	Gyro (double wheel)	Destroyer	Fieux (Fra)	Passive
1924	Fins (variable angle)	*Matsu Maru*	Motora (Jap)	Active
1933	Fins (variable area)	*Aviso Estourdi*	Kefeli (Ita)	Active
1836	Fins (variable angle)	*HMS Bittern*	Denny-Brown (GBr)	Active
1939	U-tank	*Hamilton*	Minorsky (USA)	Active
1972	Rudder	*M.S. Peggy*	van Gunsteren (Ndl)	Active
1974	Rudder	*Manchester Concorde*	Cowley & Lambert (GBr)	Active

T. Perez (2005) *Ship Motion Control*, Springer

II - Key Aspects of Ship Roll Dynamics for Control Design

Models

Ocean Environment

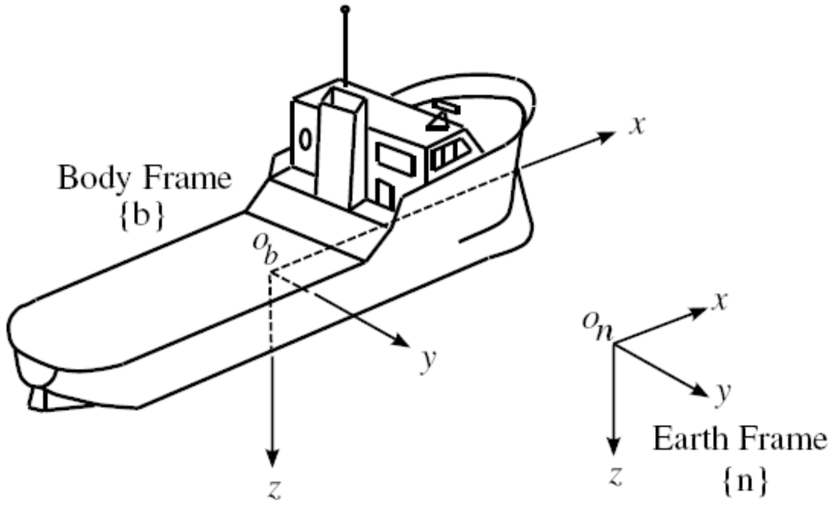
Wave-induced motion

Dynamics of Roll Motion

General model form to describe ship motion:

$$\boldsymbol{\eta} \triangleq \begin{bmatrix} \mathbf{p}_{b/n}^{n} \\ \boldsymbol{\Theta} \end{bmatrix} = [N,\, E,\, D,\, \phi,\, \theta,\, \psi]^T.$$

$$\boldsymbol{\nu} \triangleq \begin{bmatrix} {}^{n}\dot{\mathbf{p}}_{b/n}^{b} \\ \boldsymbol{\omega}_{b/n}^{b} \end{bmatrix} = [u,\, v,\, w,\, p,\, q,\, r]^T.$$

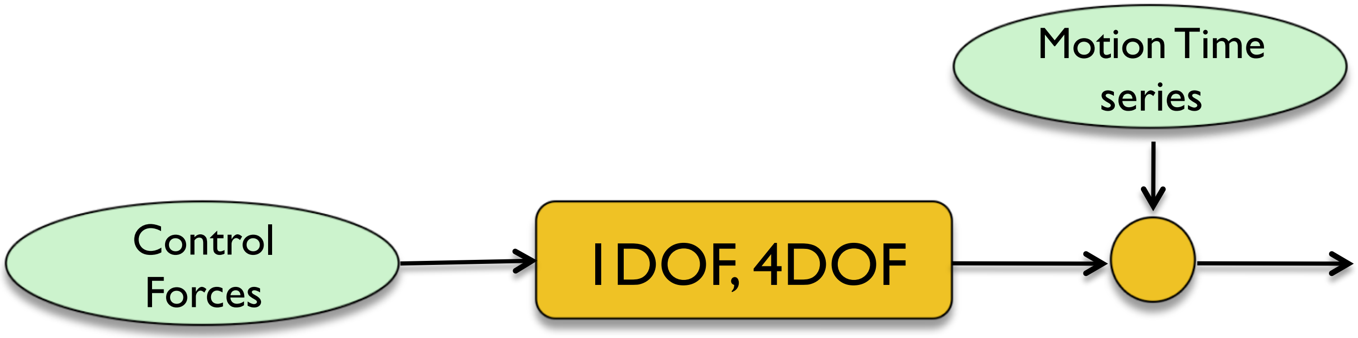


Kinematic model: $\dot{\boldsymbol{\eta}} = \mathbf{J}(\boldsymbol{\eta})\boldsymbol{\nu}$

Kinetic model: $\mathbf{M}\dot{\boldsymbol{\nu}} + \mathbf{C}(\boldsymbol{\nu})\boldsymbol{\nu} + \mathbf{D}(\boldsymbol{\nu})\boldsymbol{\nu} + \mathbf{g}(\boldsymbol{\eta}) = \boldsymbol{\tau}$

Models for Control Design

For control design, output disturbance models are usually adopted:



Force to motion

For roll motion control problems this model is simplified to

- 1DOF: roll
- 4DOF: surge, roll, sway, and yaw

Dynamics of Roll Motion—1DOF Model

Consider only roll motion,

Kinematic model: $\dot{\phi} = p,$

Kinetic model: $I_{xx}\,\dot{p} = K_h + K_c + K_d$

Hydrodynamic moments (K_h), Control moments (K_c), Disturbance moments (K_d)

Hydrodynamic Moments:

$$K_h \approx \underbrace{K_{\dot{p}}\,\dot{p} + K_p p}_{\text{Potential Effects}} + K_{p|p|}\,p|p| + \underbrace{K(\phi)}_{\text{Hydrostatic Effects}}$$

Viscous Effects: $K_p p + K_{p|p|}\,p|p|$

Dynamics of Roll Motion—4DOF

Motion Variables:

$$\boldsymbol{\eta} = [\phi \ \psi]^T,$$

$$\boldsymbol{\nu} = [u \ v \ p \ r]^T,$$

$$\boldsymbol{\tau}_i = [X_i \ Y_i \ K_i \ N_i]^T,$$

Kinematic model:

$$\dot{\phi} = p, \qquad \dot{\psi} = r \cos\phi \approx r$$

Kinetic model:

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

$$\boldsymbol{\tau}_h \approx -\mathbf{M}_A\,\dot{\boldsymbol{\nu}} - \mathbf{C}_A(\boldsymbol{\nu})\boldsymbol{\nu} - \mathbf{D}(\boldsymbol{\nu})\boldsymbol{\nu} - \mathbf{K}(\phi)$$

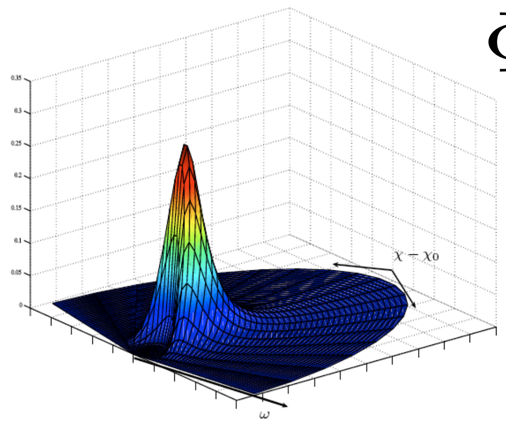
Ocean Environment

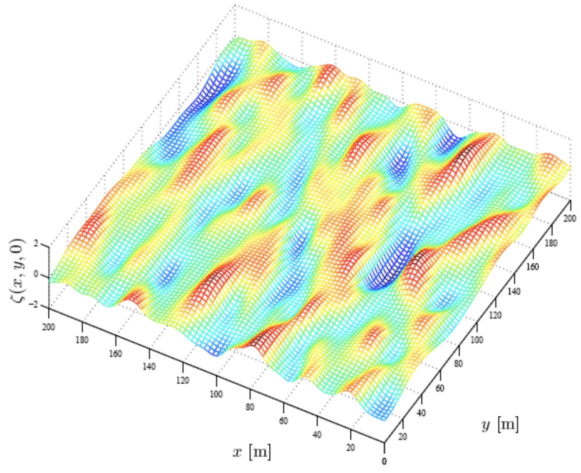
Usual assumptions for sea surface elevation $\zeta(t)$:

- *Zero-mean*
- *Gaussian (depth dependent)*
- *Narrow banded*
- *Stationary (20min to 3 hours)*

All necessary information is then in the wave elevation power spectral density (Sea Spectrum):

$$\Phi_{\zeta\zeta}(\omega, \chi)$$



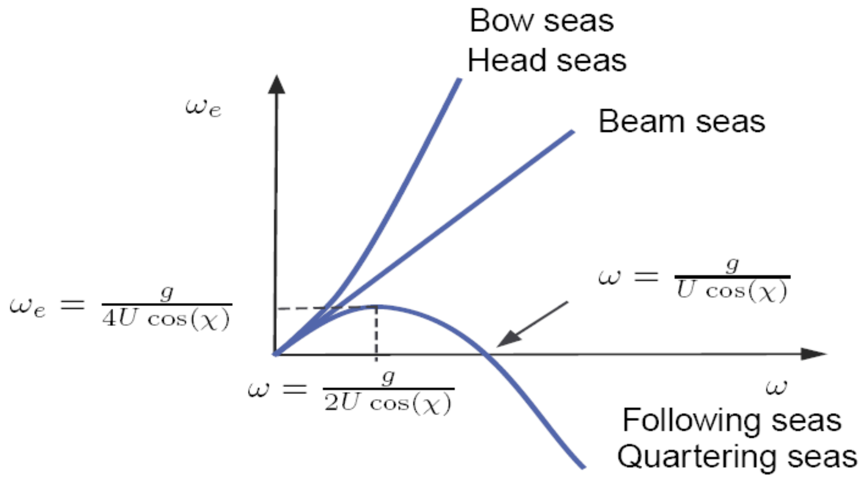


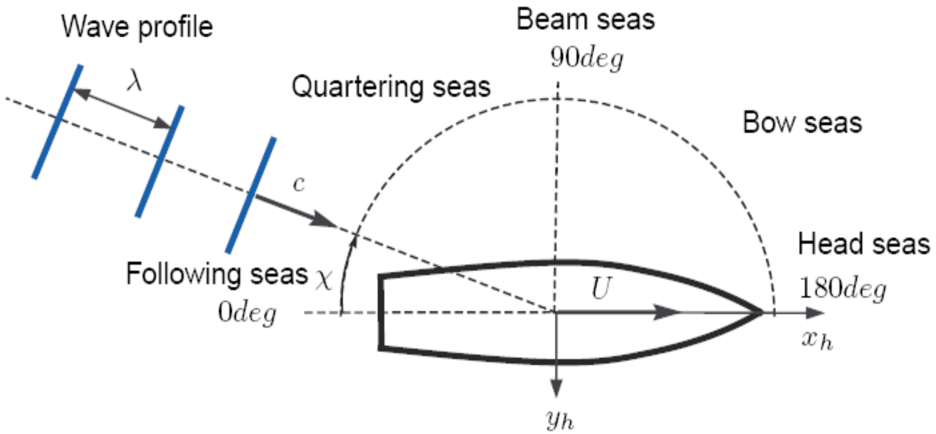
Sailing Condition and Encounter Spectrum

Sailing Condition

- *Speed*
- *Encounter angle*

Encounter Frequency

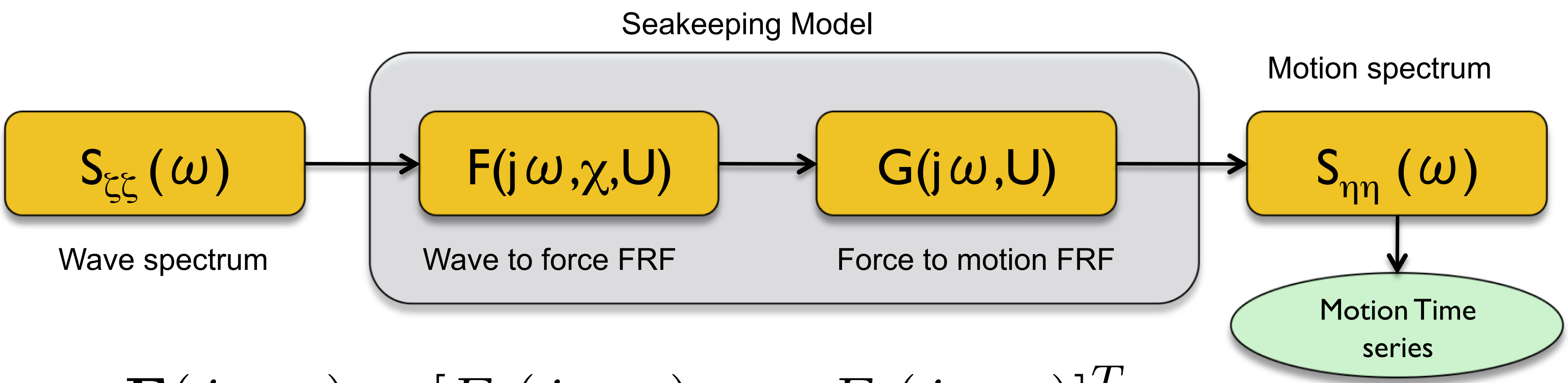




$$\omega_e = \omega - \frac{\omega^2 U}{g}\cos(\chi)$$

$$\Phi_{\zeta\zeta}(\omega_e) = \frac{\Phi_{\zeta\zeta}(\omega)}{\left|1 - \frac{2\omega U}{g}\cos(\chi)\right|}$$

Wave-induced Motion Motion



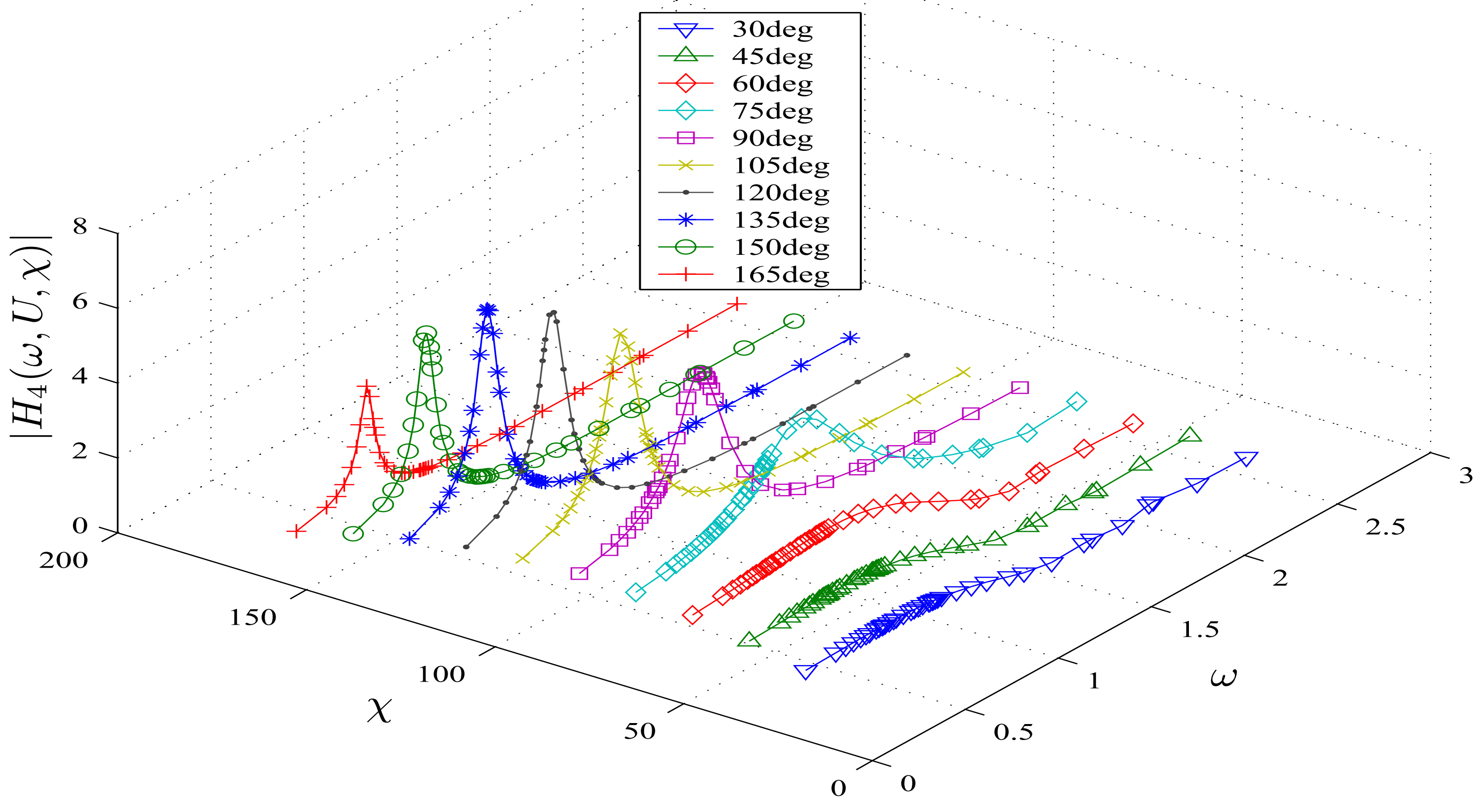
$$\mathbf{F}(j\omega,\chi) = [F_1(j\omega,\chi),\ldots,F_6(j\omega,\chi)]^T$$

$$\mathbf{G}(j\omega,U) = \begin{bmatrix} G_{11}(j\omega,U) & \cdots & G_{16}(j\omega) \\ \vdots & & \vdots \\ G_{61}(j\omega,U) & \cdots & G_{66}(j\omega,U) \end{bmatrix} = \left(-[\mathbf{M}_{RB}+\mathbf{A}(\omega)]\,\omega^2 + j\omega\mathbf{B}(\omega) + \mathbf{G}\right)^{-1}$$

Roll RAO: $$H_4(j\omega,\chi,U) = \sum_{k=1}^{6} G_{4k}(j\omega,U)F_k(j\omega,\chi,U)$$

Example Roll RAOs Naval Vessel @15kt

$$H_4(j\omega, \chi, U) = \sum_{k=1}^{6} G_{4k}(j\omega, U) F_k(j\omega, \chi, U)$$



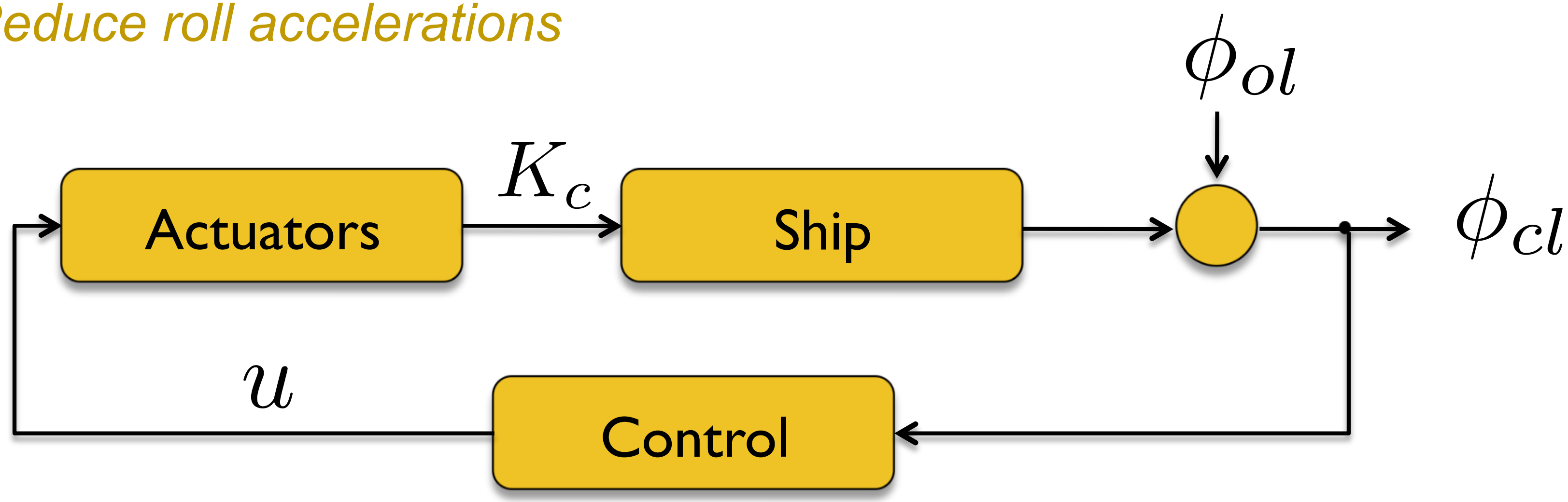
III – Motion Control Design

Objectives,
Performance Limitations,
and Control Strategies

Control Design and Performance Limitations

From the ship performance point of view, the control objectives are

- *Reduce roll angle*
- *Reduce roll accelerations*



Output sensitivity

$$S(s) \triangleq \frac{\phi_{cl}(s)}{\phi_{ol}(s)}$$

Roll spectrum relations

$$\Phi_{cl}(\omega) = |S(j\omega)|^2 \Phi_{ol}(\omega)$$

Control Design and Performance Limitations

If the closed-loop system is stable minimum phase and strictly proper, the following Bode integral constraint applies

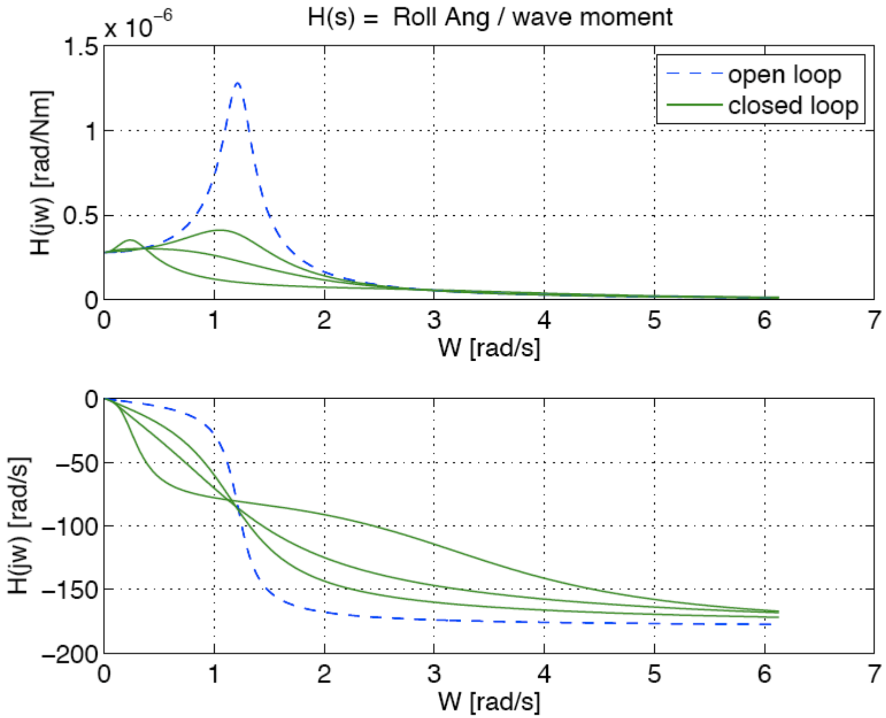
$$\int_0^{\infty} \log |S(j\omega)| \, d\omega = 0$$

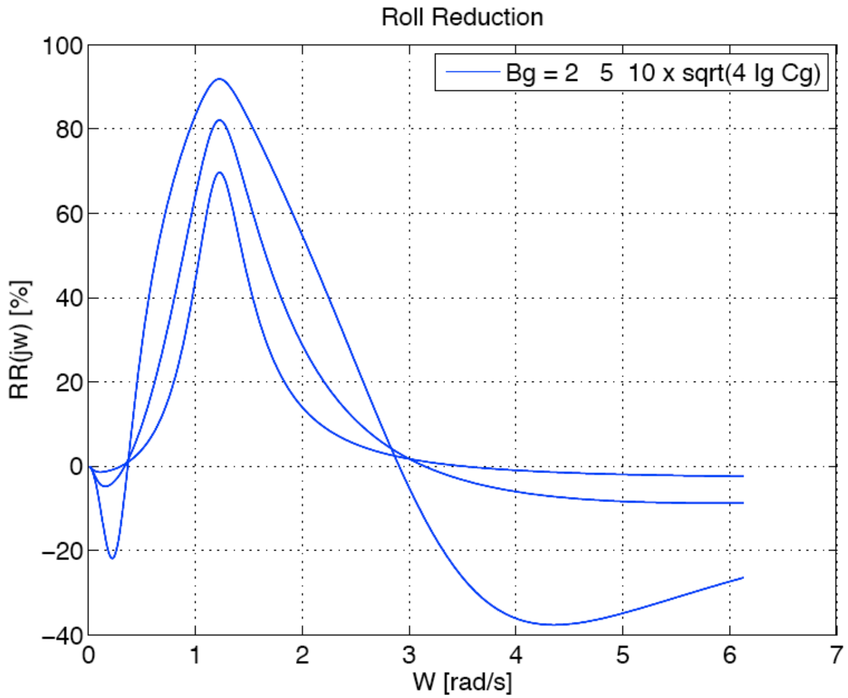
$$\Phi_{cl}(\omega) = |S(j\omega)|^2 \Phi_{ol}(\omega)$$

$$RR(\omega) = 1 - |S(j\omega)| = \frac{|\phi_{ol}(j\omega)| - |\phi_{cl}(j\omega)|}{|\phi_{ol}(j\omega)|}$$

Example Performance Limitations

Gyrostabiliser (Perez & Steinmann, 2009)





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Non-minimum phase Dynamics

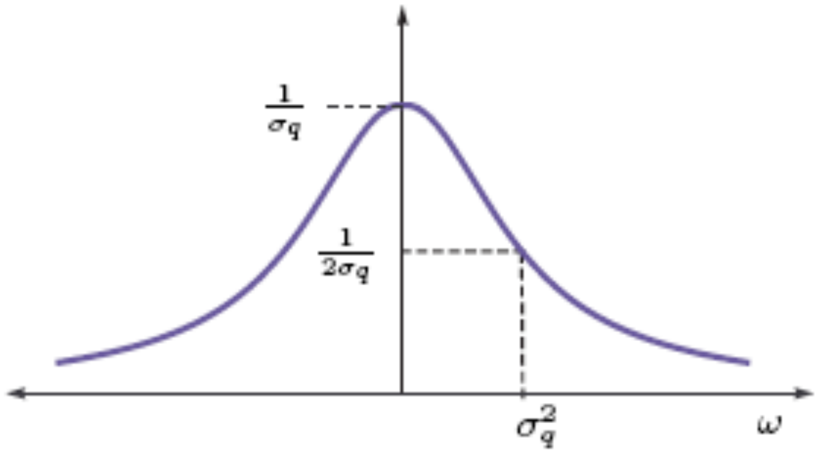
In some cases, the location of the actuators may result in NMP dynamics (with a real zero at s=q.)

Then, we have a Poisson-integral constrain:

$$\int_{-\infty}^{\infty} \log |S(j\omega)| W(q,\omega) d\omega = 0$$

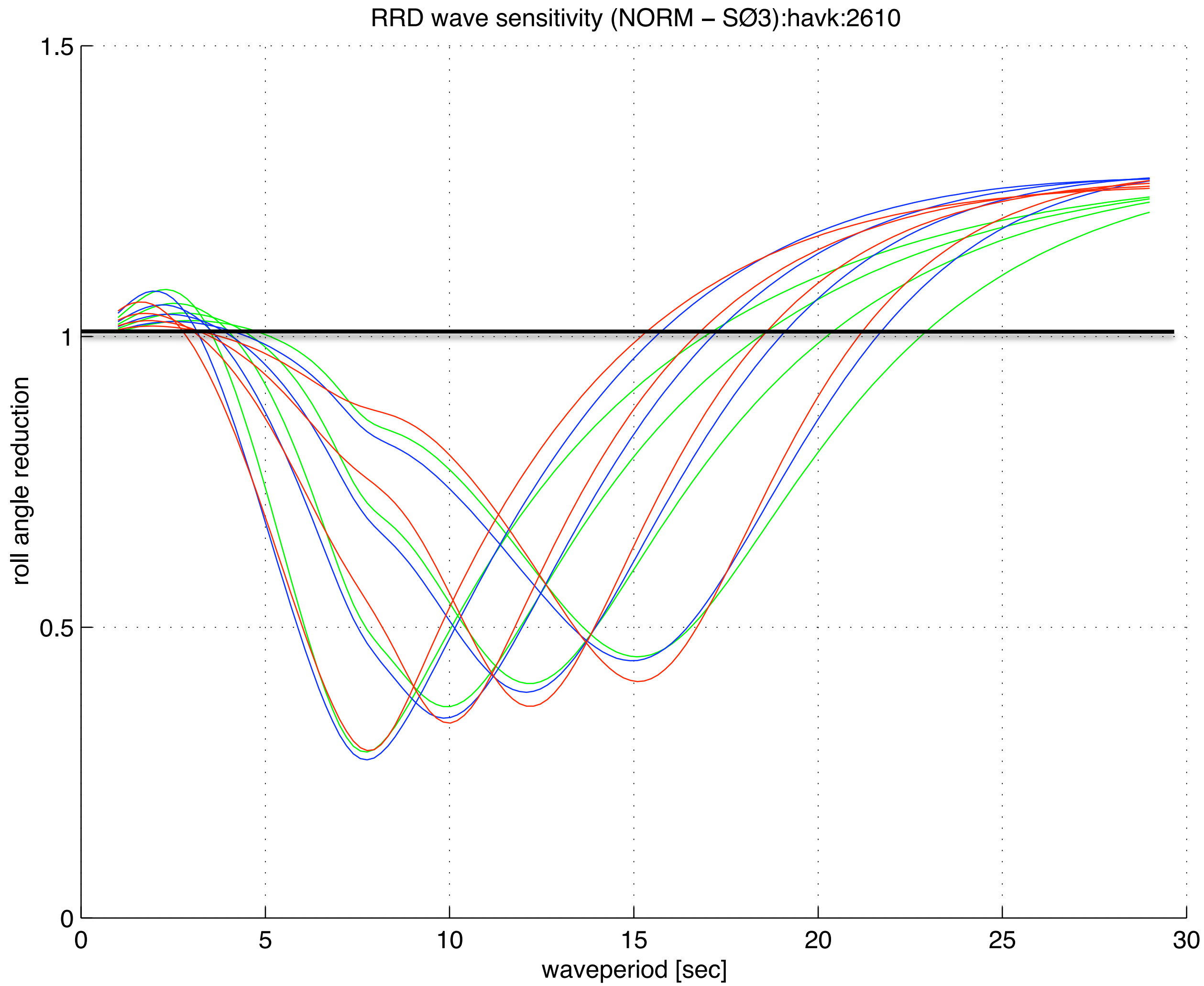
$$q = \sigma_q + j0$$

$$W(q,\omega) = \frac{\sigma_q}{\sigma_q^2 + \omega^2}$$



Non-minimum phase Dynamics

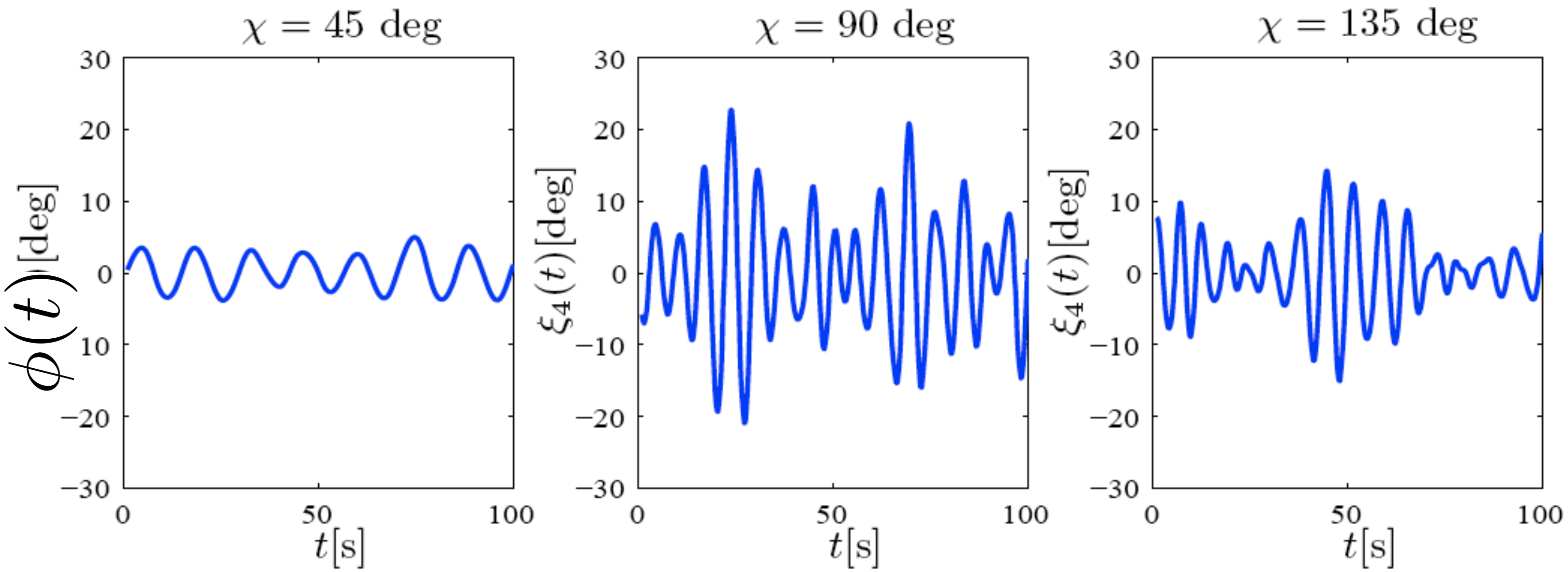
Example Blanke et al (2000)



Energy of Wave Excitation

The energy of the wave excitation changes with the sea-state and sailing conditions (speed and heading)

For example a change in encounter angle can shift the energy significantly:



Fin Stabilisers (minimum phase case)

IDOF model including fin forces:

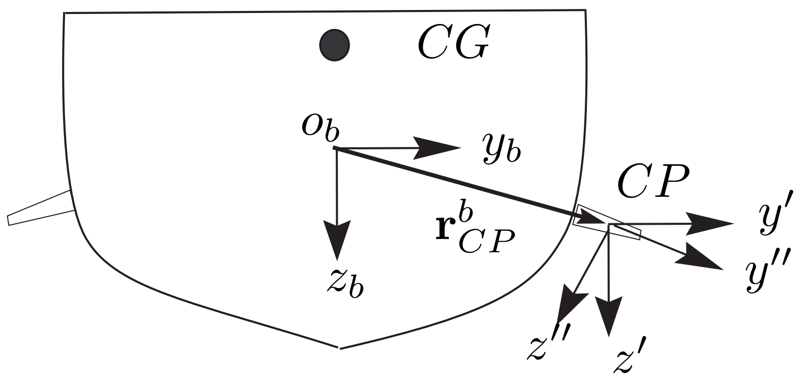
$$\dot{\phi} = p,$$
$$[I_{xx} + K_{\dot{p}}]\,\dot{p} + (K_p + 2\,r_f K_\alpha U)\,p + K_\phi\,\phi = K_w - 2U^2 K_\alpha\,\alpha$$

Control issues:

- Parametric uncertainty
- Sensitivity-integral constraints

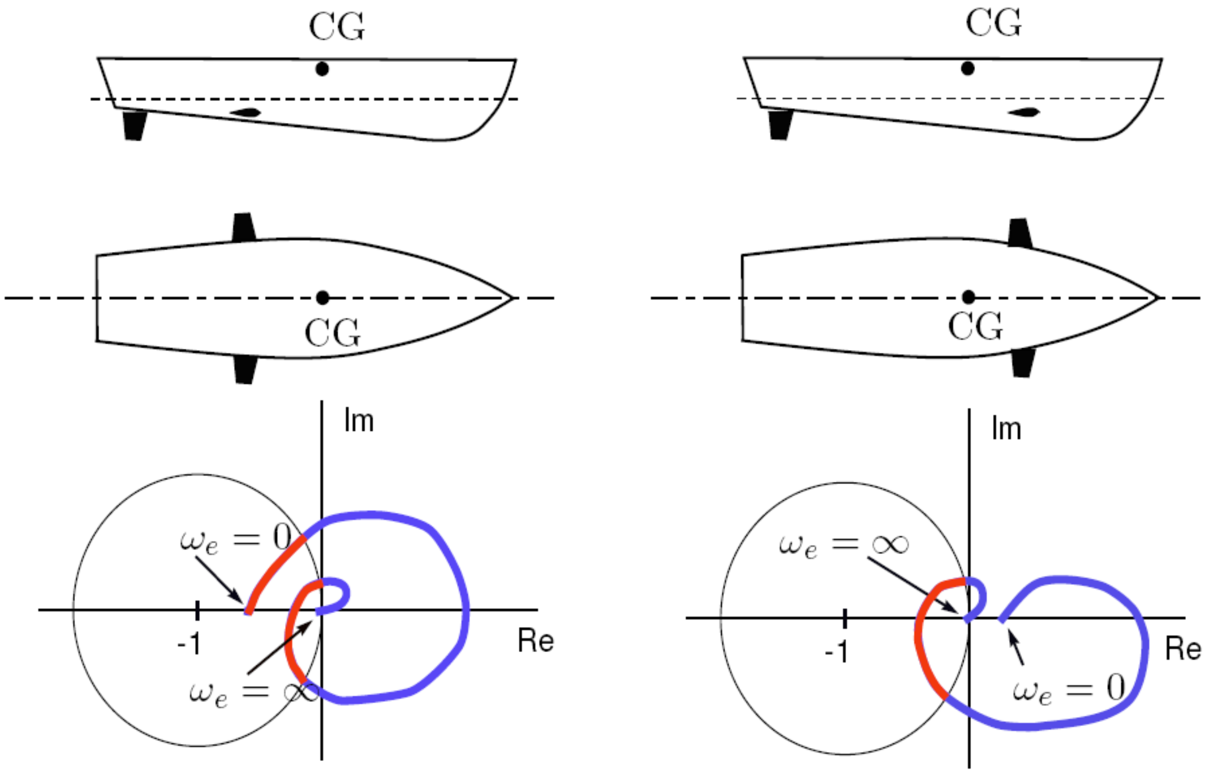
Control strategies:

- PID, Hinf



Fin Stabilisers and NMP Dynamics

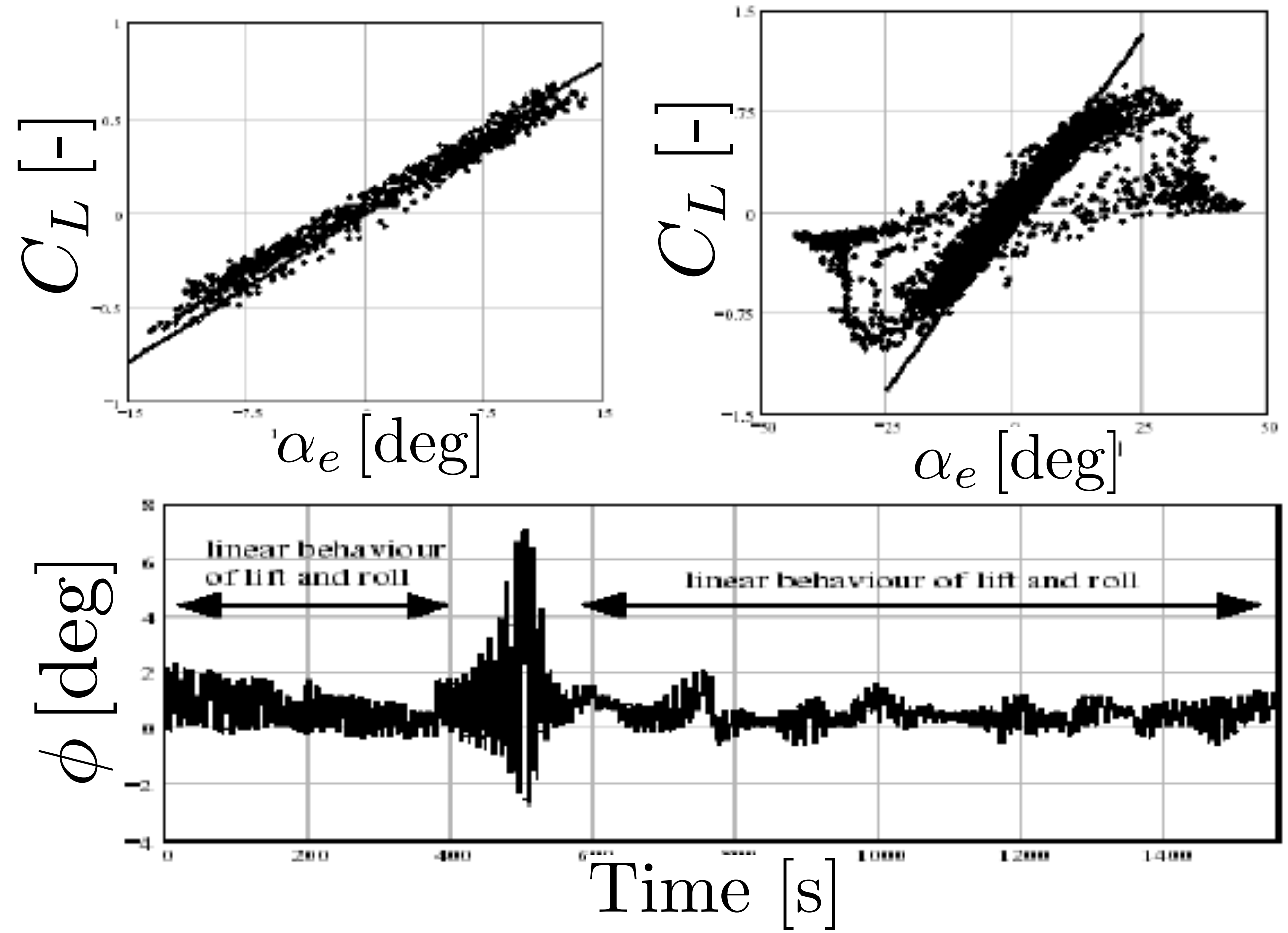
If the response from the fins is NMP, then the IDOF cannot be used, in this case roll-sway-yaw interactions need to be considered to avoid large roll amplifications at low frequencies.



Observations about fin NMP dynamics were made by Lloyd (1989).

Fin Stabilisers – Dynamic Stall

Experimental results of Galliarde (2002) (MARIN, Ned)



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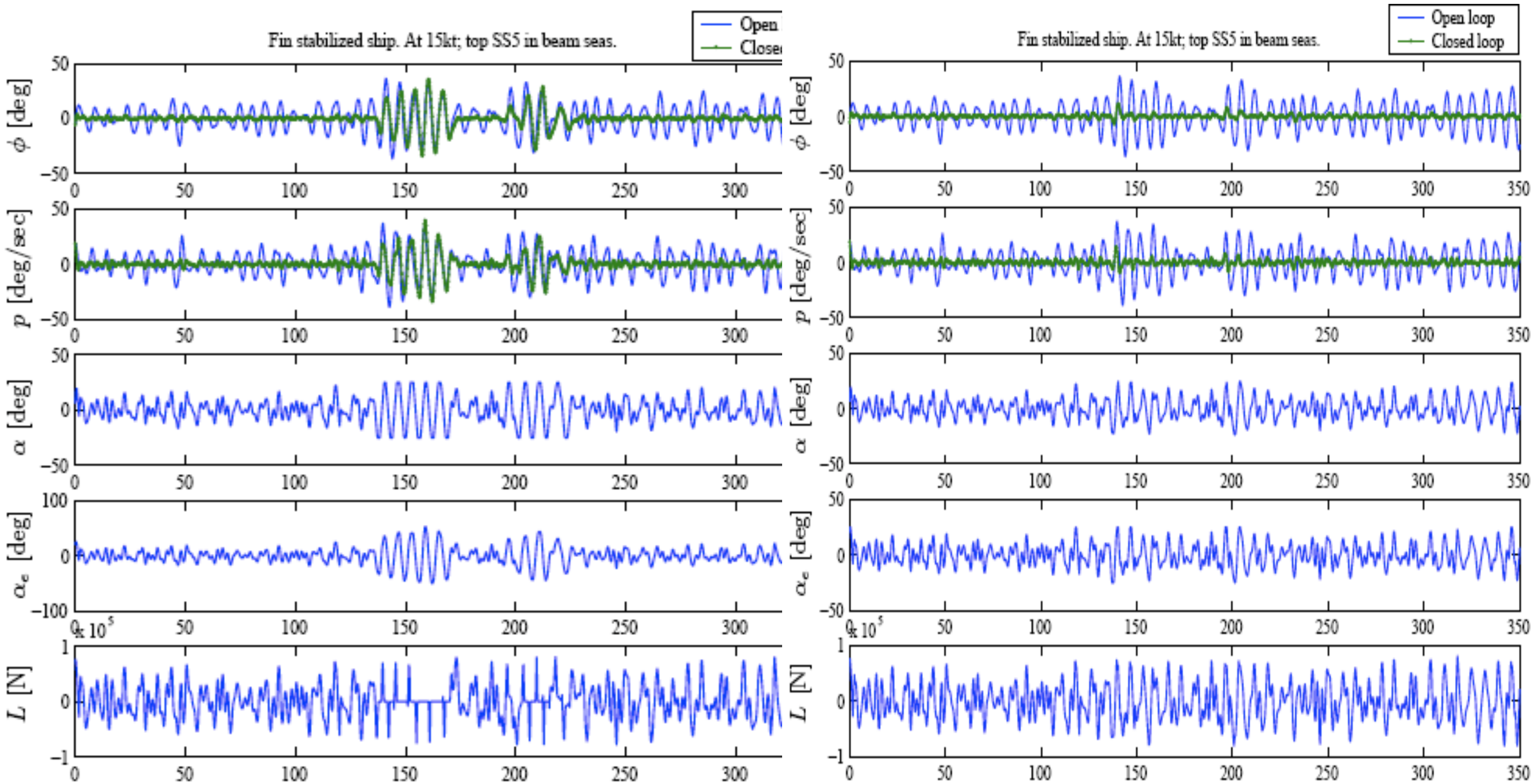
Fin Stabilisers

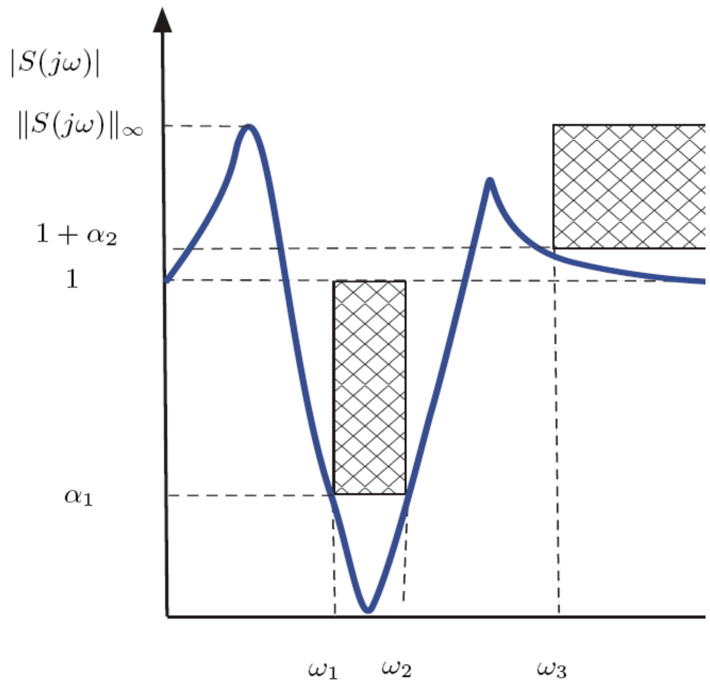
Perez & Goodwin (2003): MPC constraints effective angle of attack.

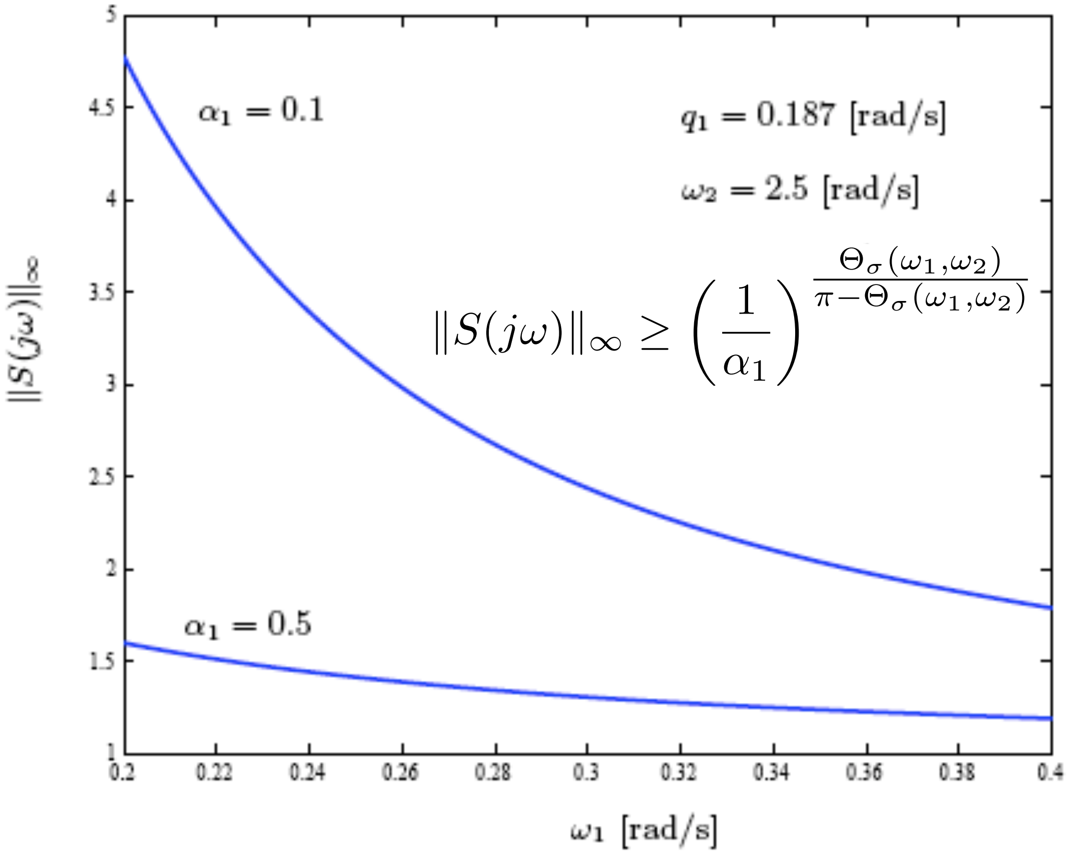


Rudder Roll Damping

- Potential discovered from autopilot without wave filter (Taggart 1970)
- Results reported by van Gunsteren in 1972 (the Netherlands)
- Cowley & Lambert (1972) used roll fbk, reported yaw interference.
- Carley & Duberley (1972) integrated rudder-fin control
- Carley (1975) & Lloyd (1975) recognise limitations of NMP dynamics
- Baitis et al.(1983) highlighted need of adaptation
- Advent of Computers in 1980 resulted in several successful results
  - Netherlands, Denmark, Sweden, United Kingdom
- Hearns & Blanke (1998) limitations using the Poisson Integral
- Perez (2003) analysed min variance and RR vs yaw interference

Performance Limitations (Hearns & Blanke, 1998)





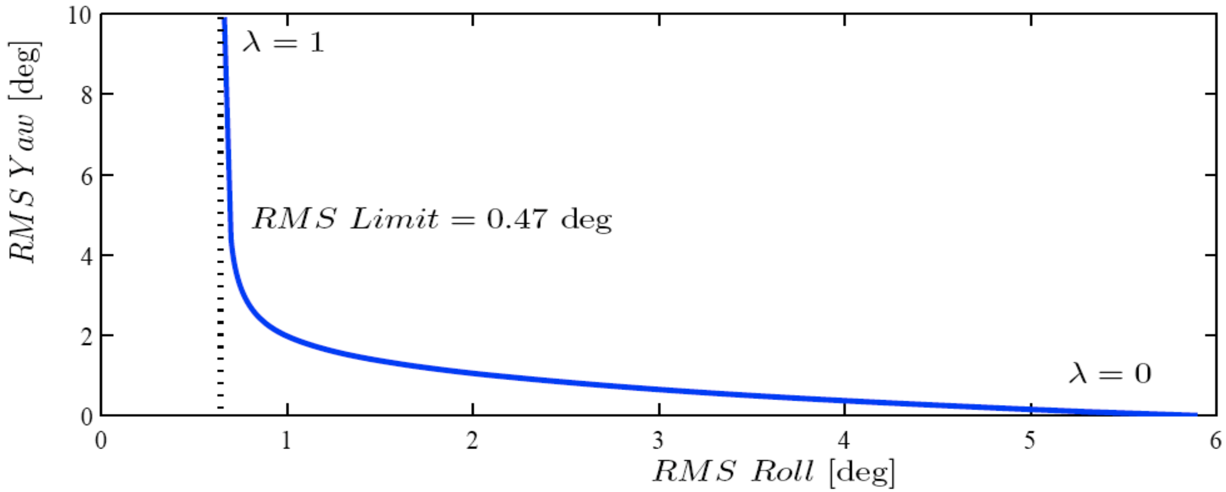
If the controller does not adapt, we can easily have roll amplification

Performance Limitations – Perez et al. (2003)

Limiting Optimal Control with full knowledge of wave spectrum:

$$J = \mathrm{E}[\lambda\phi^2 + (1-\lambda)(\psi - \psi_d)^2]$$

Minimum variance case (λ=1): $\mathrm{E}[\phi^2] \geq 2\, q\, \Phi_{\phi\phi}(q)$



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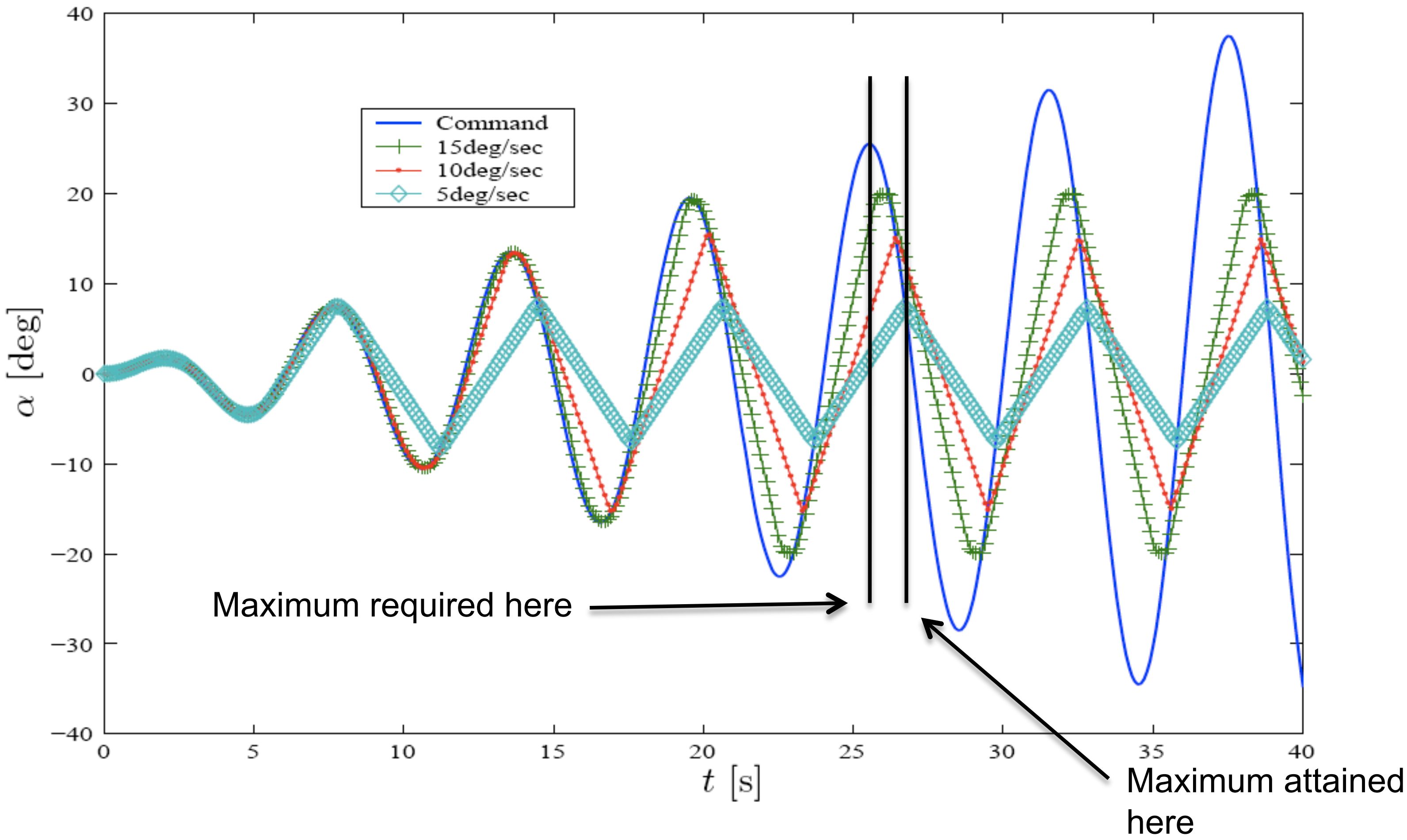
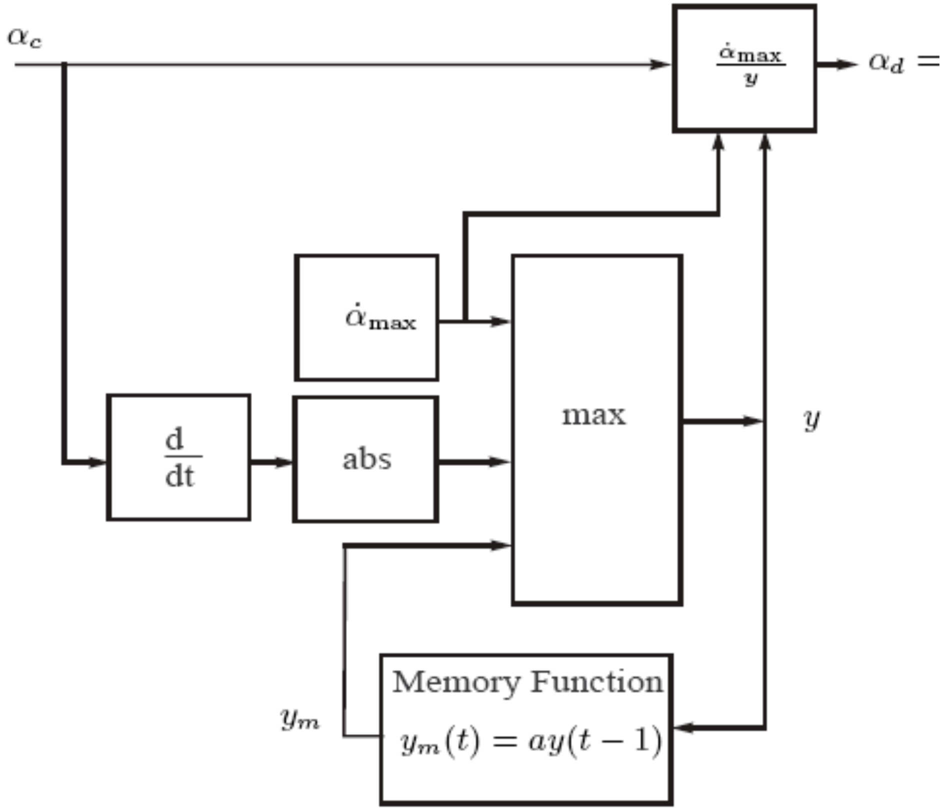
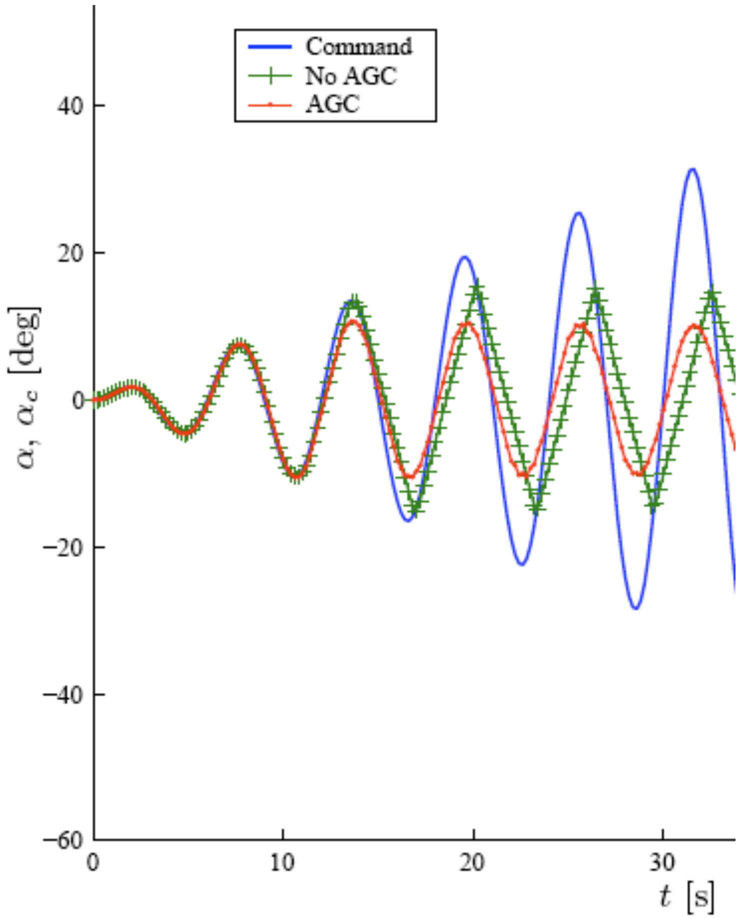
Actuator limitations on RRD

- Actuator rate limits may affect performance



AGC – Van Amerongen et al (1982)





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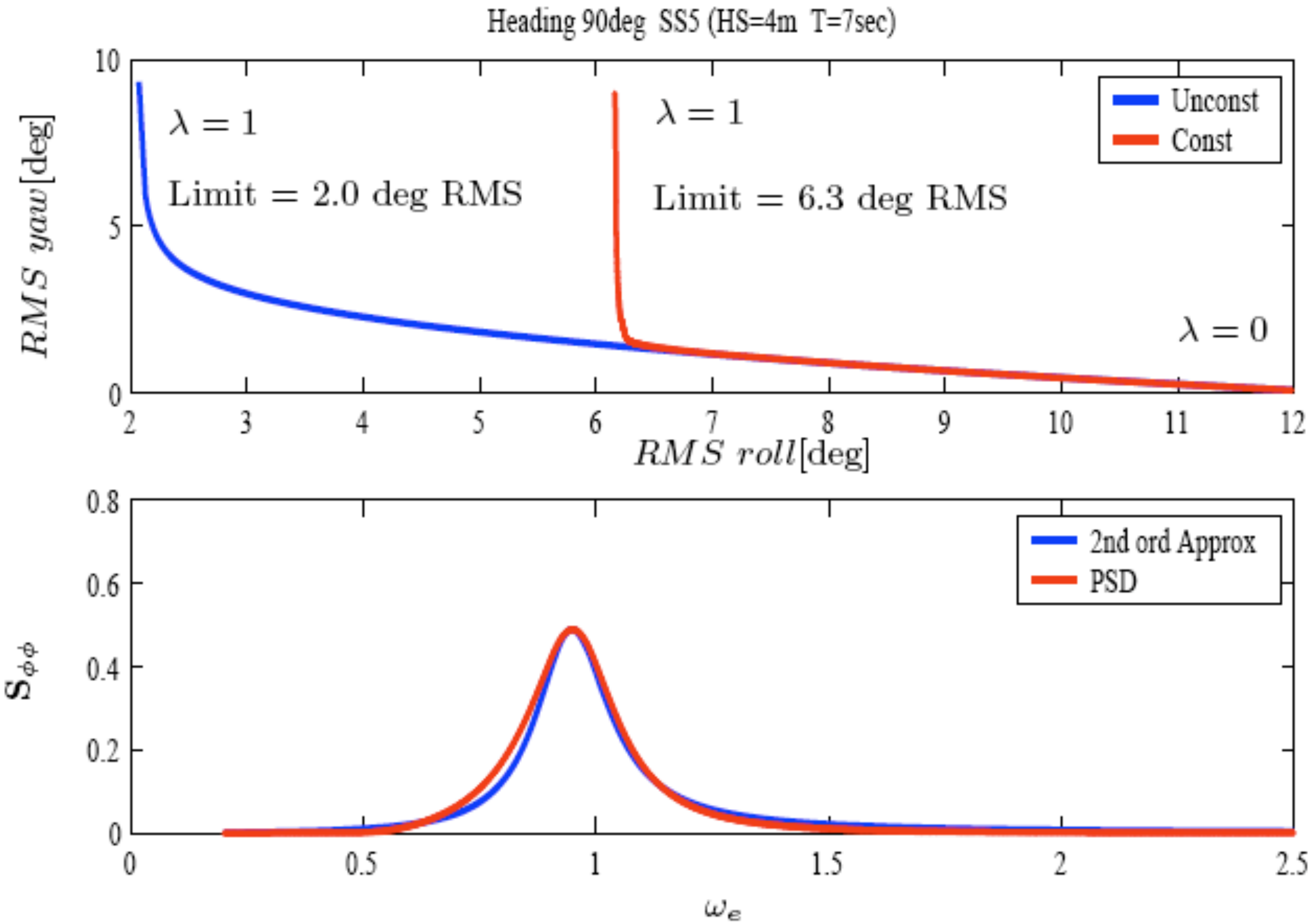
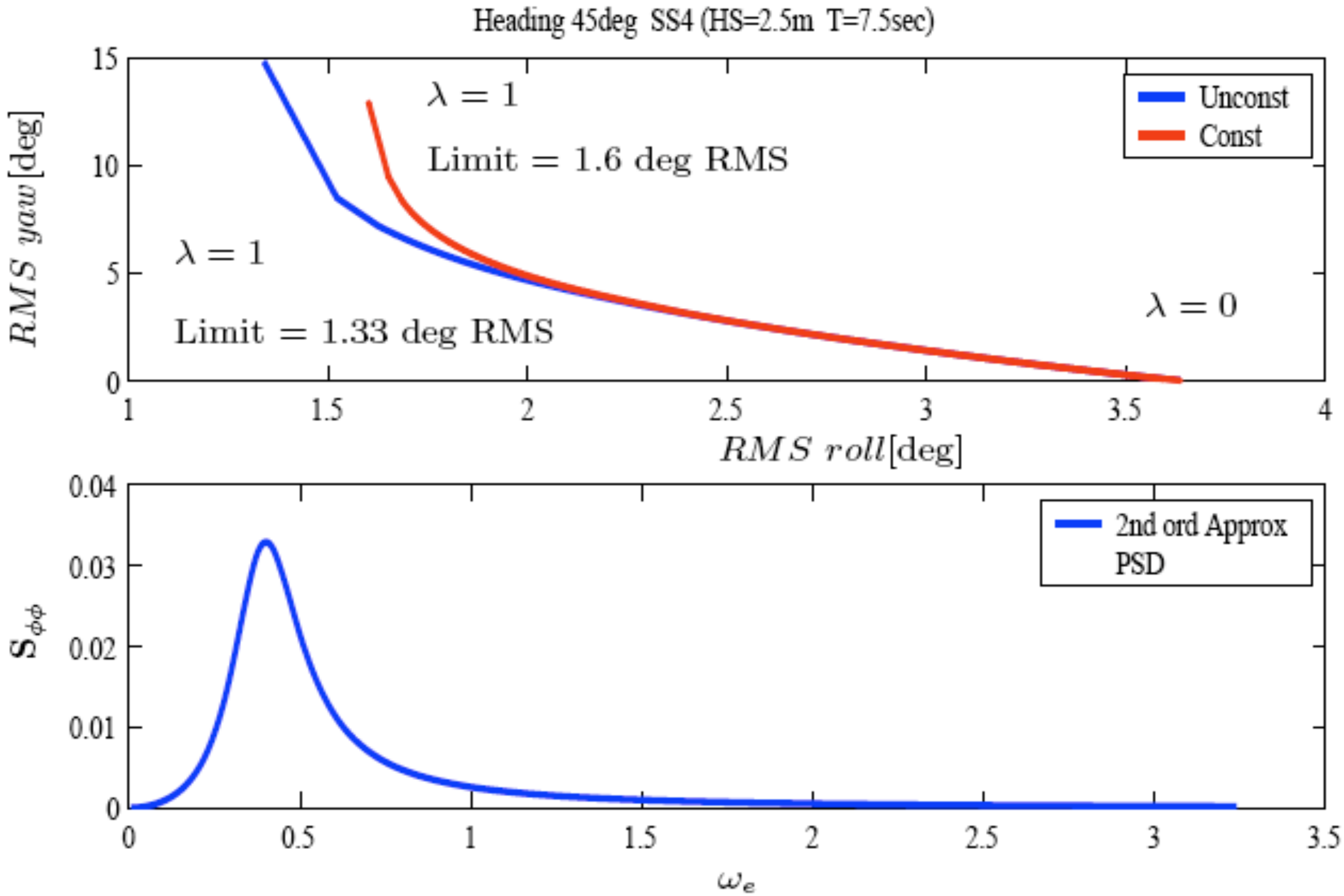
Performance Limitations – Perez et al. (2003)

IVC OCP: Rudder angle RMS limited to 15deg



Performance Limitations – Perez et al (2003)

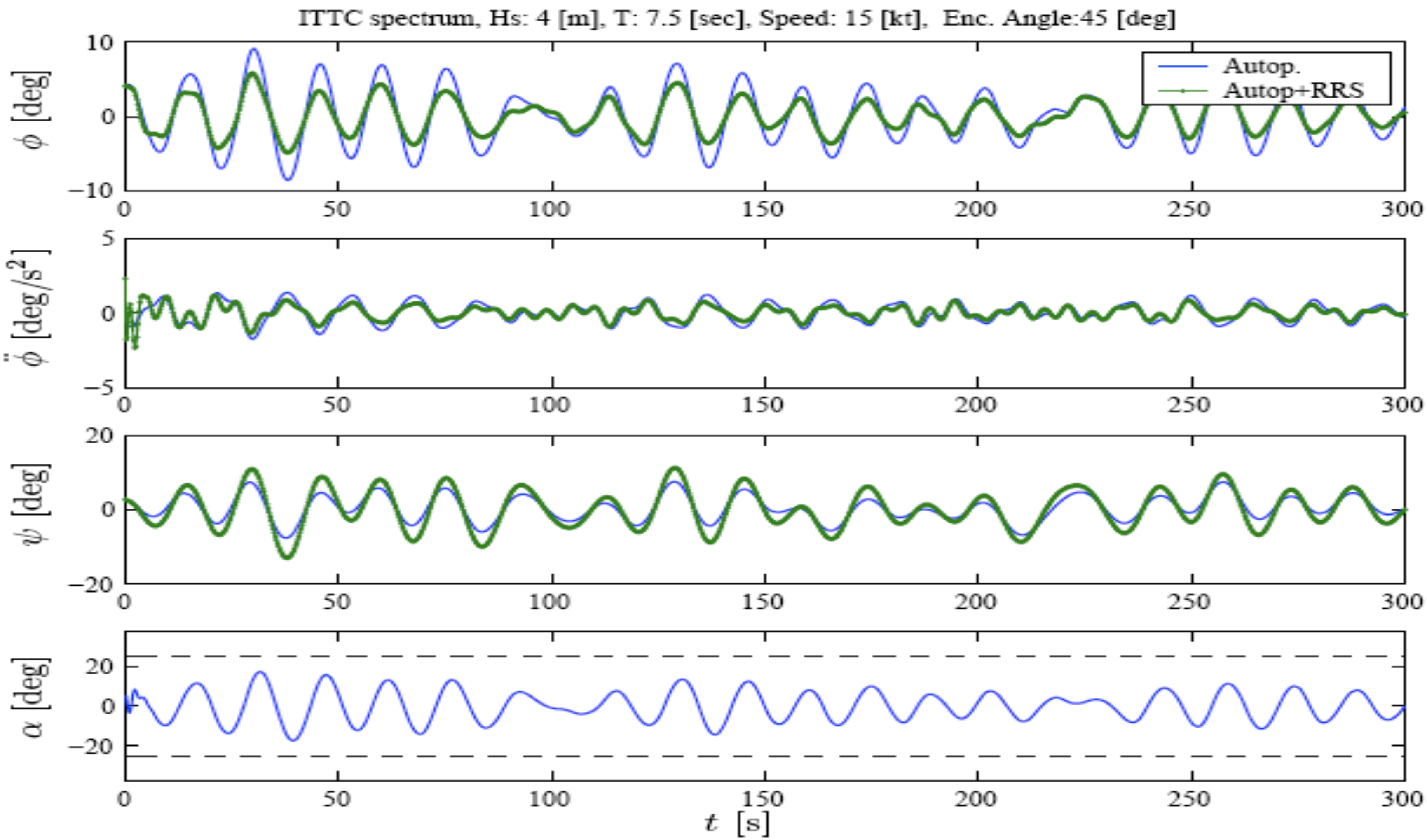
IVC OCP: Rudder angle RMS limited to 15deg



Performance Limitations

- At low encounter frequencies,
  - RR vs yaw interference can be large
  - MNP dynamics limits RR achievable performance.
- At high encounter frequencies,
  - Limitations of rudder machinery usually dominate RR achievable performance.

Performance Limitations (quatering seas RR 40%)



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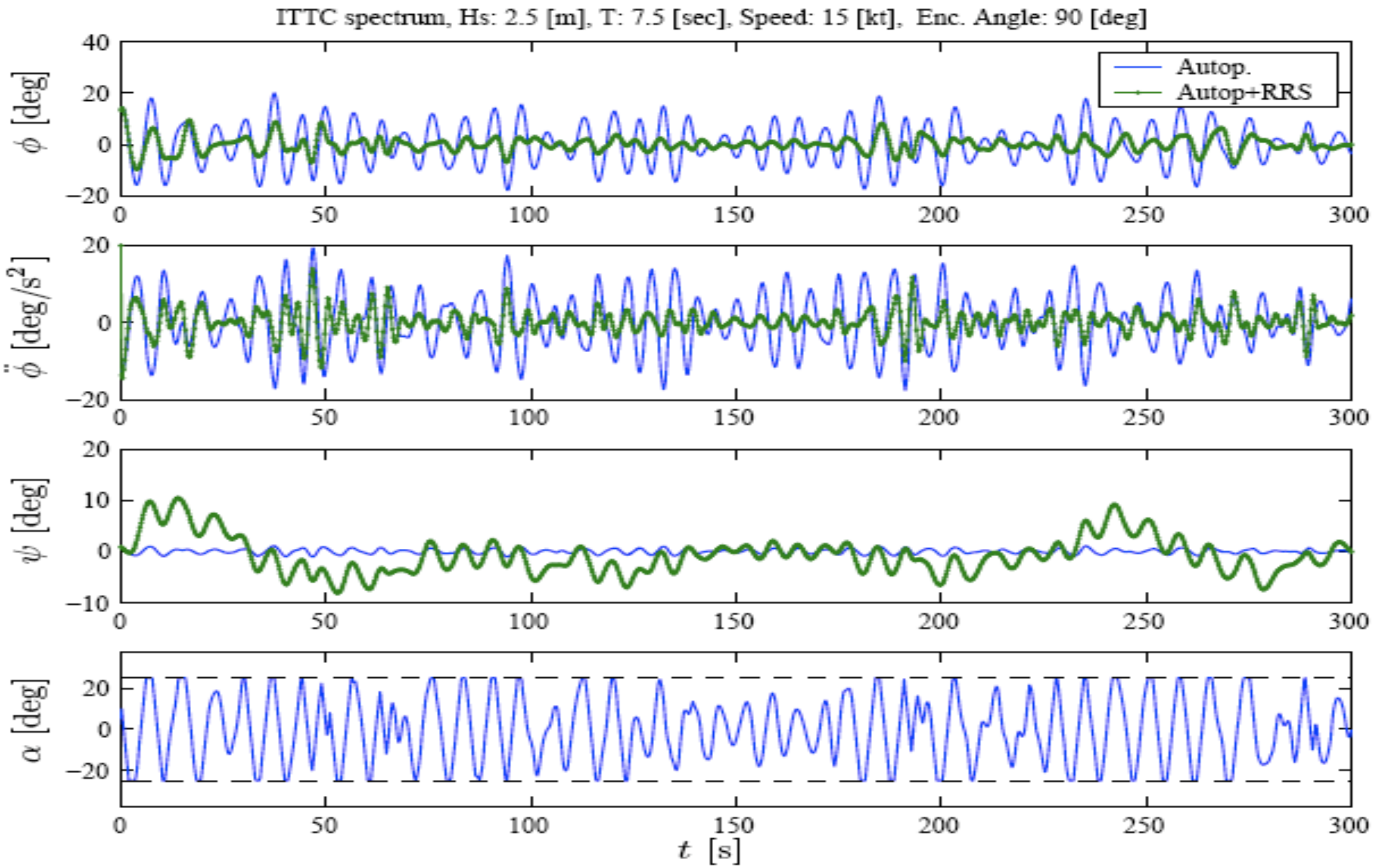
Performance Limitations (beam seas RR 64%)



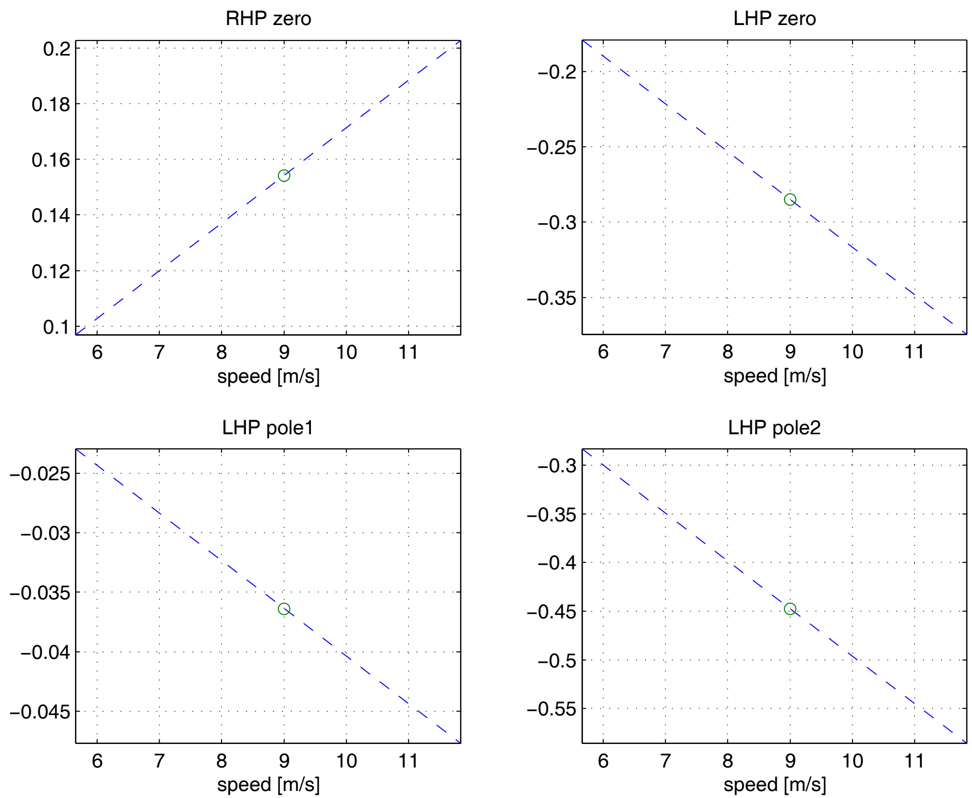
Control Strategies

- PID,
- Hinf
- Loop shaping
- Adaptive LQG
- Model Predictive Control
- Switched control
- Nonlinear control

Changes with Speed (Blanke & Christiansen 1993)

- Rudder to Roll TF: $G_{\phi\delta}(s) = \dfrac{c_{\phi\delta}(1+s\tau_{z1})(1-\frac{s}{q})}{(1+s\tau_{p1})(1+s\tau_{p2})(\frac{s^2}{\omega_p^2}+2\zeta_p\frac{s}{\omega_p}+1)}$

Changes in dynamic response due to speed coupled with changes in disturbance spectrum results in a strong need for adaptation.



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H2 – Optimal Design

$$\Phi_{\phi\phi}(\omega) = H_d^*(j\omega)H_d(j\omega)$$

$$E(\phi^2) = \|H_d - QV\|_2^2 \qquad V = H_d G_{\delta\phi}$$

$$C_{\delta\phi}(s) = (G_{\delta\phi}^-)^{-1} \frac{H_d^{-1} H_d^-}{1 - \frac{s-q}{s+q} H_d^{-1} H_d^-}$$

$$G_{\phi\delta}^-(s) = G_{\phi\delta}(s)\frac{s+q}{s-q} \qquad H_d(s)\frac{s+q}{s-q} = H_d^-(s) + H_d^+(s)$$

This shows the strong dependency on sea state, sailing conditions and changes in the vessel model --- adaptation is required.

Direct Sensitivity Specification (Blanke et al. 2000)

Targeted (Desired) Sensitivity:

$$S_d(s) = \frac{\frac{s^2}{\omega_d^2} + 2\zeta_d \frac{s}{\omega_d} + 1}{(1 + \frac{s}{\beta\omega_d})(1 + \frac{\beta s}{\omega_d})} \qquad S(s) = (1 + C_{\delta\phi}(s)G_{\phi\delta}(s))^{-1}$$

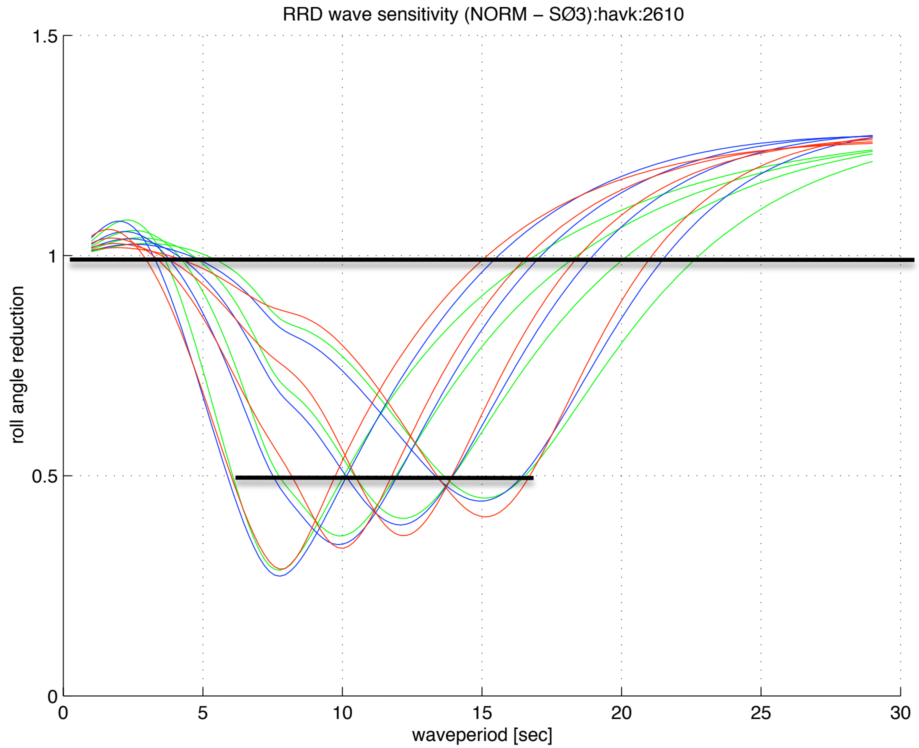
$$\Downarrow$$

$$\boxed{C^s_{\delta\phi}(s) = (S_d(s)^{-1} - 1)G^{-1}_{\phi\delta(s)} \frac{1 - \frac{s}{q}}{1 + \frac{s}{q}} P(s)^{-1}}$$

Direct Sensitivity Specification (Blanke et al. 2000)

$$C^s_{\delta\phi}(s) = \frac{k_1\, s}{c_{\phi\delta}\omega_d}\,\frac{\frac{s^2}{\omega_p^2} + 2\zeta_p \frac{s}{\omega_p} + 1}{\frac{s^2}{\omega_d^2} + 2\zeta_d \frac{s}{\omega_d} + 1}\,\frac{(1+s\tau_{p1})(1+s\tau_{p2})}{(1+s\tau_{z1})(1+\frac{s}{q})}$$

Successful performance was demonstrated for the SF300 patrol boat of the Danish Navy



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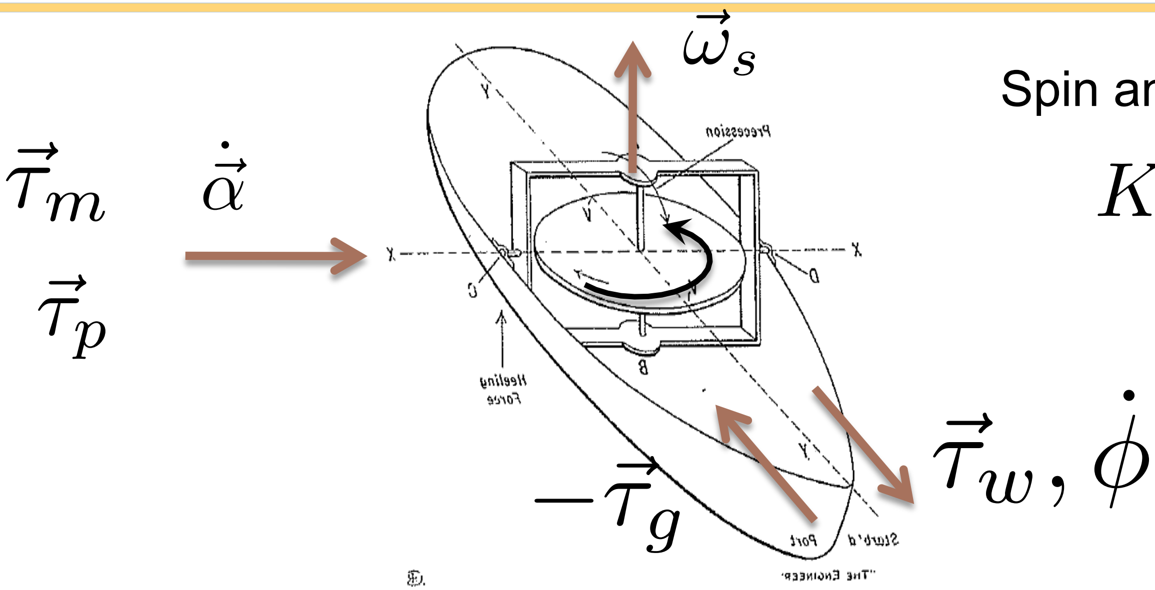
Gyro Stabilisation



Spin angular momentum

$$K_g = I_s\,\omega_s$$

$$I_{44}\ddot{\phi} + B^l_{44}\dot{\phi} + B^n_{44}|\dot{\phi}|\dot{\phi} + C_{44}\phi = \tau_w - \underbrace{K_g\dot{\alpha}\cos\alpha}_{\tau_g},$$

$$I_g\ddot{\alpha} + B_g\dot{\alpha} + C_g\sin\alpha = \underbrace{K_g\dot{\phi}\cos\alpha}_{\tau_m} + \tau_p$$

Gyro as a Damper

$$\tau_p : \dot{\alpha} \approx q\,\dot{\phi}$$

$$I_{44}\ddot{\phi} + (B^e_{44} + \underbrace{nK_g q}_{})\dot{\phi} + C_{44}\phi = \tau_w$$

Increase of roll damping due to the gyrostabiliser

n- number of gyros

q-constant

Gyro-control Design

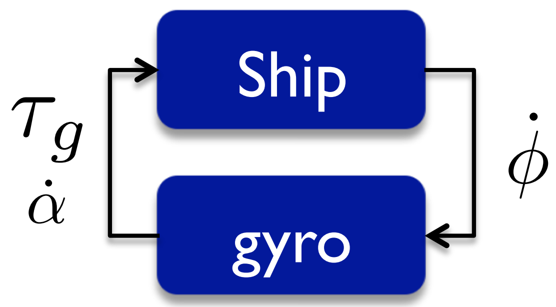
Gyro full state control: $\tau_p = -K_a \alpha - K_r \dot{\alpha}$

Roll to precession Transfer function:

$$G(s) = \frac{\alpha(s)}{\phi(s)} = \frac{\dot{\alpha}(s)}{\dot{\phi}(s)} = \frac{K_g s}{I_g s^2 + B'_g s + C'_g}, \qquad \begin{aligned} B'_g &= B_g + K_r, \\ C'_g &= C_g + K_a \end{aligned}$$

The control design objective becomes:

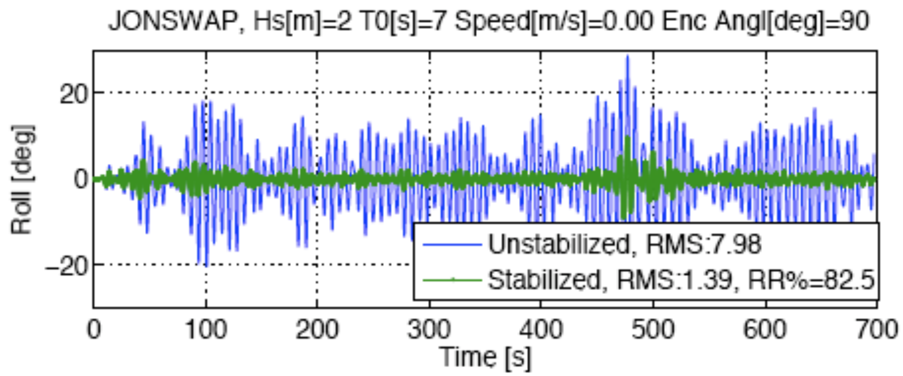
$$\tau_p : \dot{\alpha} \approx q\,\dot{\phi} \rightarrow \quad |G(\jmath\omega)| \approx q, \qquad \forall \omega \in \Omega \quad \arg G(\jmath\omega) \approx 0$$

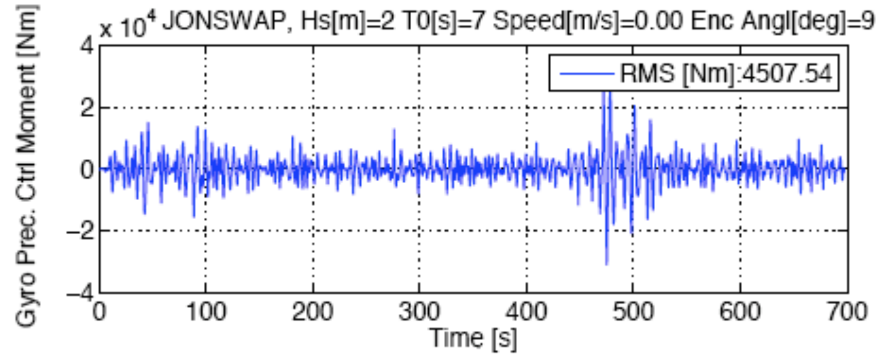


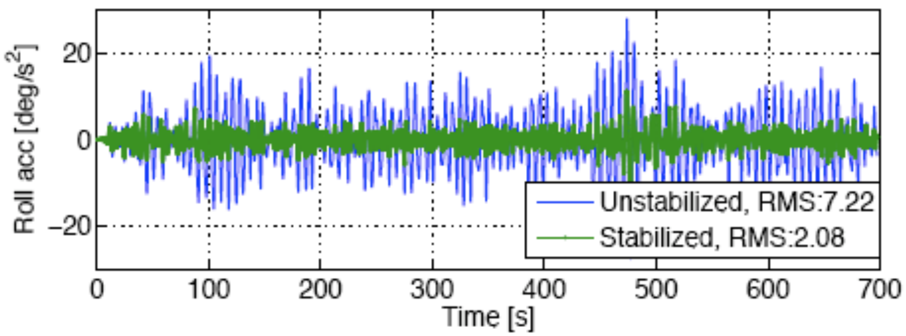
This can be achieved by forcing two real poles on G(s) (Perez Steinman, 2009)

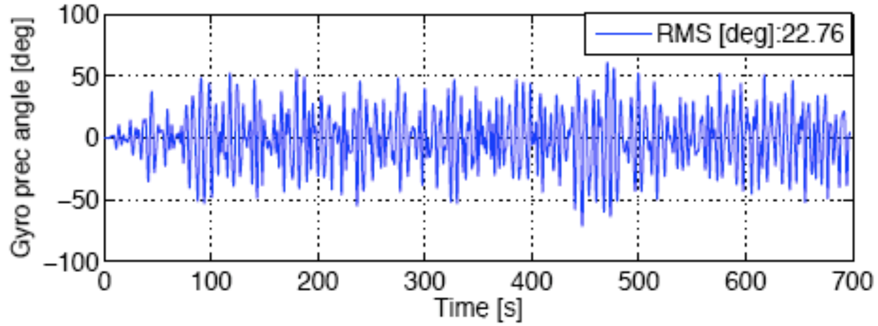
Irregular Sea Performance (Perez & Steinamnn 2009)

Roll Reduction in RMS: 82%
Vessel displacement: 360 ton
Gyro unit weight: 13 ton









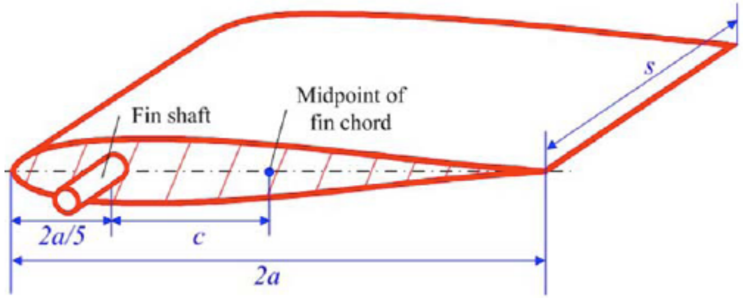
IV - Research Outlook

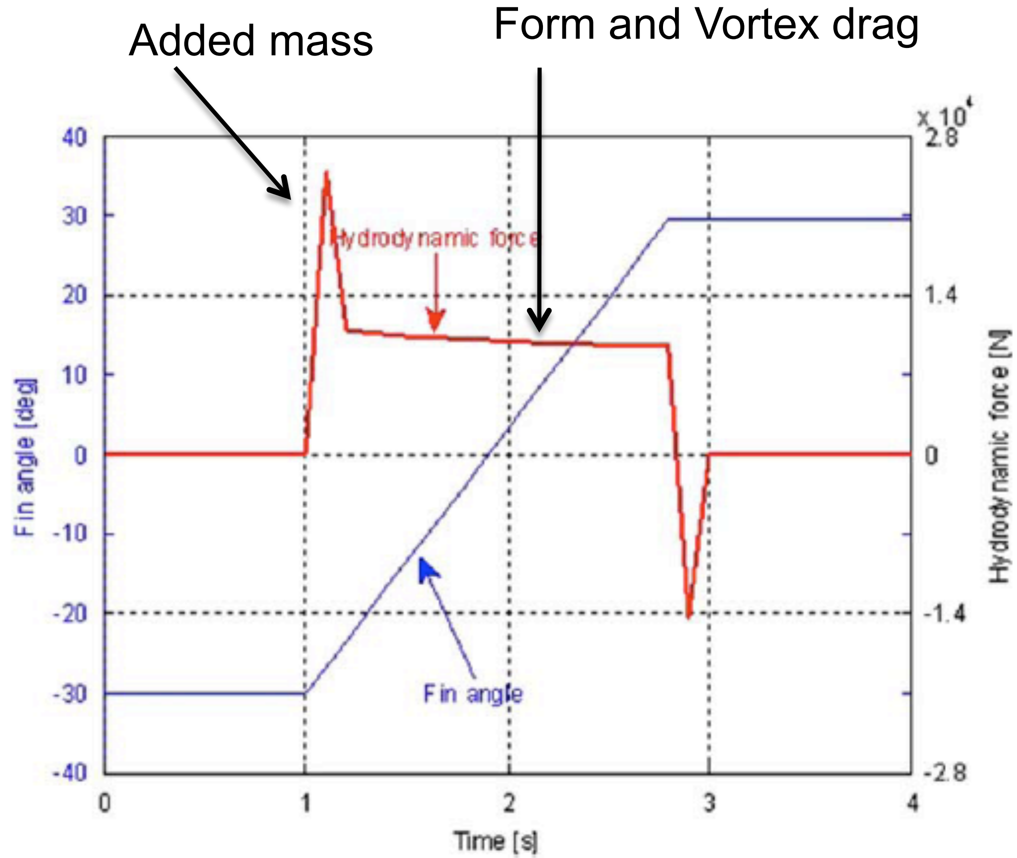
New Devices

Unsolved Problems

New Devices

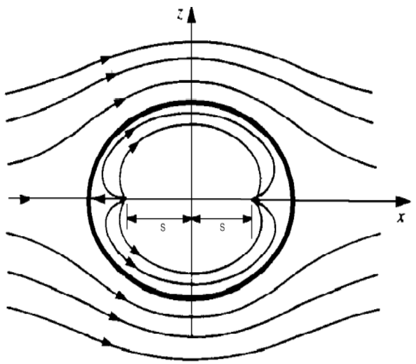
Flapping fins (Fang et al. 2009 Ocean Engineering 36)

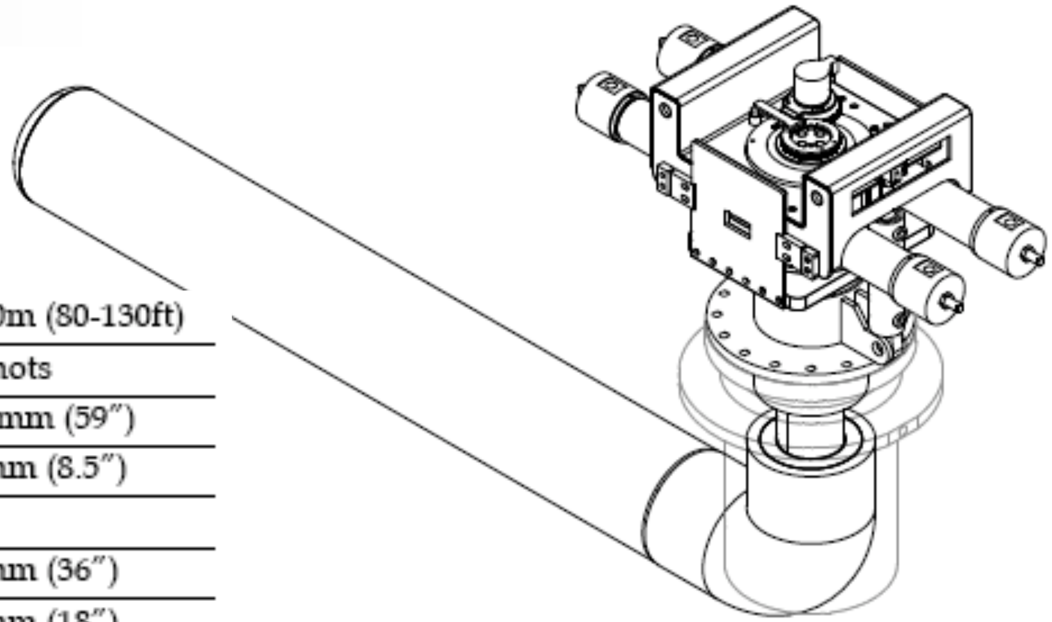




New Devices

Rotor Stabiliser (Quantumhydraulic.com)

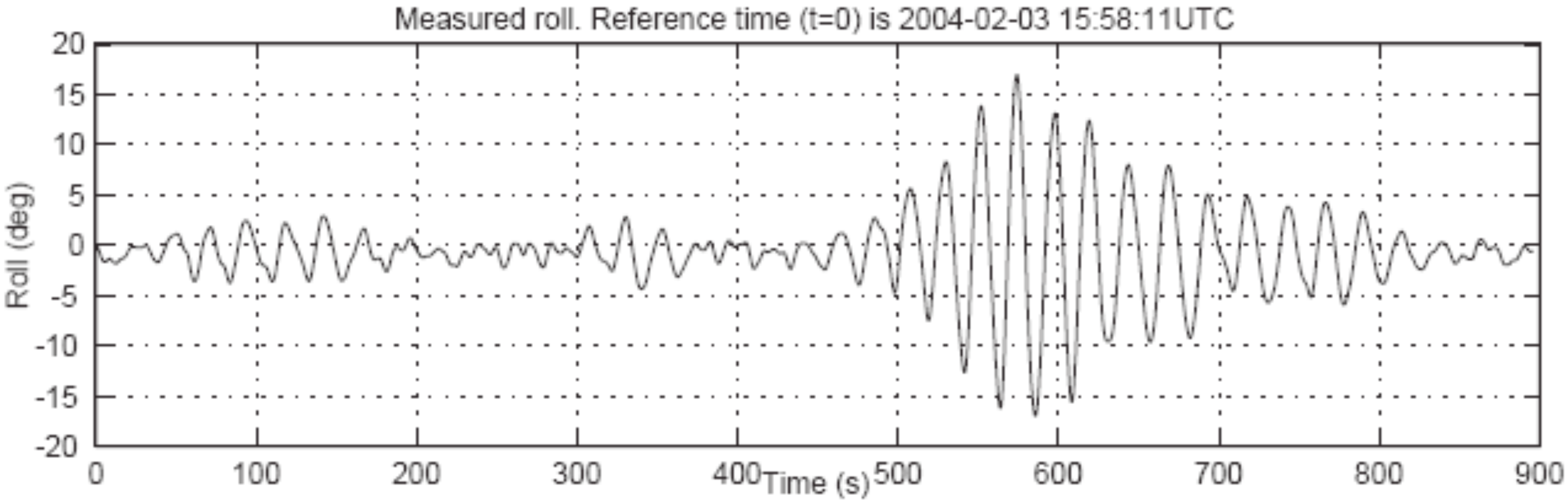




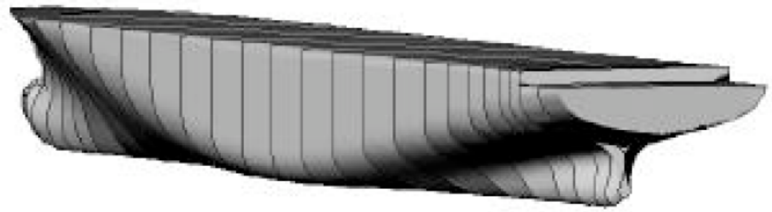
Typical Vessel Length*	25-40m (80-130ft)
Maximum Underway Operating Speed	14 knots
Rotor Length	1500mm (59″)
Rotor Diameter	217mm (8.5″)
Angular Travel *(total mechanical)*	150°
Length *(inside vessel after installation)***	920mm (36″)
Width**	450mm (18″)
Height *(overall)***	1110mm (44″)
Height *(inside vessel, retracted)***	890mm (35″)

Parametric roll

Parametric roll is an auto-parametric resonance phenomenon whose onset causes a sudden rise in roll oscillations.



Conditions for PR



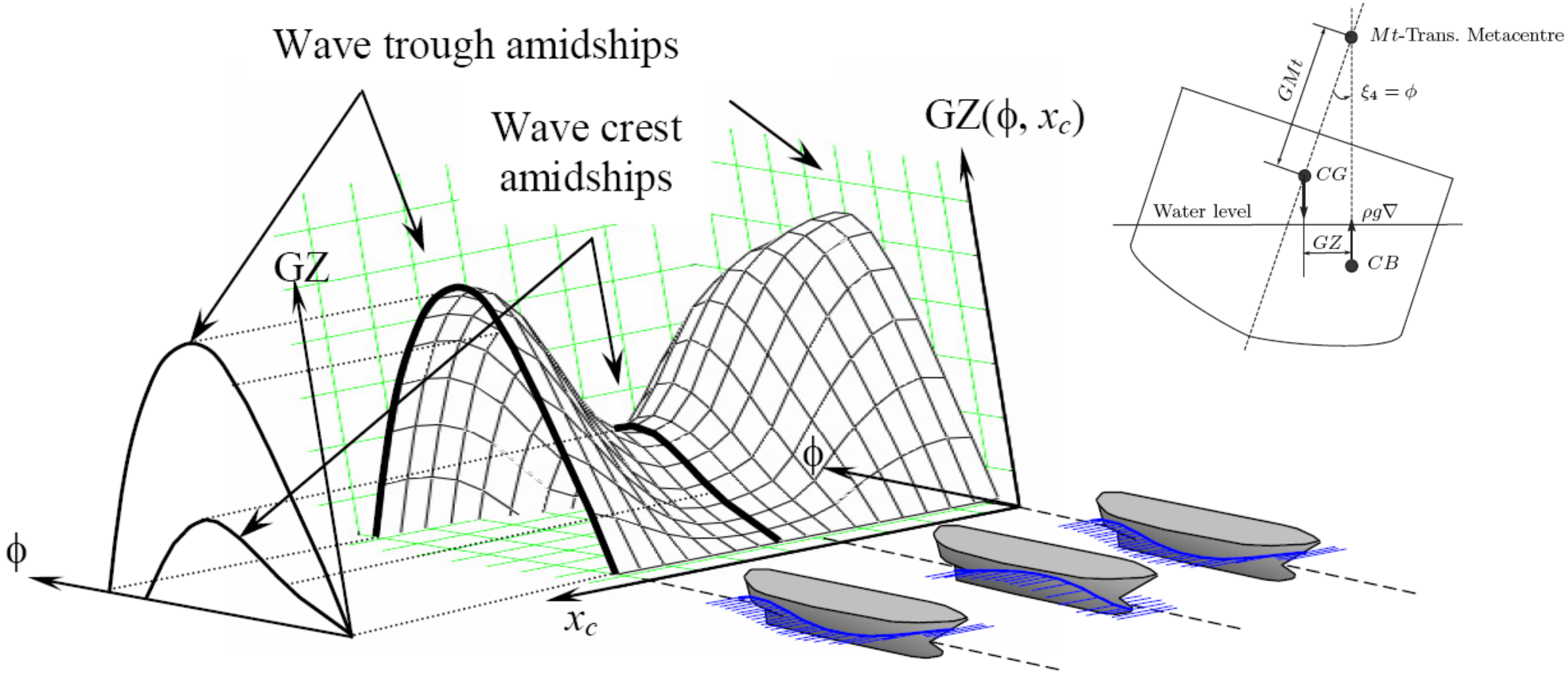
The following conditions can trigger the effect:

- Hull designs with significant bow flare and hanged stern
- Sailing in longitudinal waves
- Wave length close to the length of the vessel
- Encounter frequency is twice the roll natural frequency
- Low roll damping

Parametric roll is particular of modern container ships, cruise ships, car ferries, and fishing vessels.

Restoring in Waves – Dynamic Stability

$$K(\phi) = \rho\, g\, \nabla\, GZ(\phi)$$



Videos

Conclusion

- For over 100 years different devices have been proposed to control roll motion (reduction)
- Most of these devices require control systems to work
- Control design is not trivial due to
  - Fundamental limitations
  - Widely-varying disturbance characteristics
  - Actuator limited authority

Current State and Research Outlook

- New developments are being put forward by yacht industry
  - Gyros, Flap fins, rotor stabilisers (need for zero speed performance)
- Navies are still considering RRD
- Submarines need roll control at low speeds when in the surface

Research outlook

- Adaptation to sea state and sailing conditions still remains an issue
- New devices with interesting hydrodynamics
- Integrated vessel and roll control design (performance prediction)
- Parametric roll

Thank You

Death Roll of Sailing Vessels