Lecture 2 - Controlling Ship Dynamics

1. Introduction

A ship must be designed to propel through water with maximum efficiency ideally via use of minimum power in waves (Betram, 2012). Weather phenomena influence ship resistance and associated ship dynamics. The water's resistance to the motion of the ship is referred to as **total hull resistance (R_T)**. This is the resistance force used to calculate a ship's effective horsepower. Total hull resistance increases as speed increases (see Figure 2-1). The resistance curve is not linear. Instead, it increases more steeply at higher speeds. The hump on the resistance curve shown in Figure 2-1 is a phenomenon common to most ship resistance curves.

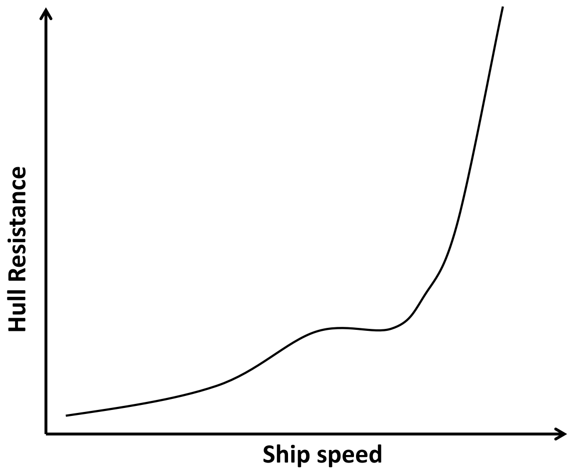


Figure 2-1 *Typical curve of total hull resistance*

Ship **resistance** is a function of many factors, including ship speed, hull form (draft, beam, length, wetted surface area), wave actions and water temperature. In terms of **controlling ship dynamics**, it is key to realize that the power required to propel a ship through the water is the product of total hull resistance and ship speed. Ship power is proportional to *(ship speed)*^{3}. For example, doubling the speed of a ship from 15 to 30 knots requires $2^3 = 8$ times as much power! Thus, for the ship operator planning getting from point A to B in the shortest amount of time requires a lot more power than traveling the same distance at a slower speed. Naturally, this increase in power is felt directly in the amount of fuel burned during the transit. A ship's fuel consumption curve is similar in shape to its horsepower and total resistance curves. Thus, voyage planning requires careful attention to transit speed and fuel consumption rates to ensure that the ship arrives at her destination with an adequate supply of fuel onboard. Whereas these are items that mostly relate with ship efficiency and marine engineering it is key to realize that resistance and propulsion phenomena interplay in terms of realizing ship dynamic phenomena and their impact on ship safety. This is the reason why in naval architecture we tend to link ship safety and performance. This lecture focuses on outlining key definitions related with ship resistance in waves, the dynamics of rudders and ship stabilisation systems. The aim is to outline basic physics and key mathematical principles of broader relevance to ship dynamics and for use in concept ship design development.

2. Key definitions

As a ship moves through the water, the friction of the water acting over the entire wetted surface of the hull causes a net force opposing the ship's motion. This frictional resistance is a function of the hull's wetted surface area, surface roughness, and water viscosity. As water flows along the hull, the laminar flow begins to break down and become chaotic and well mixed. This chaotic behavior is referred to as turbulent flow and the transition from laminar to turbulent flow occurs at the transition point (see Figure 2-2). Turbulent flow is characterized by the development of a layer of water along the hull moving with the ship along its direction of travel. This layer of water is referred to as the *"boundary layer"*. Moving away from the hull, the velocity of water particles in the boundary layer becomes less, until at the outer edge of the boundary layer velocity is nearly that of the surrounding ocean. Formation of the boundary layer begins at the transition point and the thickness of the boundary layer increases along the length of the hull as the flow becomes more turbulent. As ship speed increases, the thickness of the boundary layer will increase, and the transition point between laminar and turbulent flow moves closer to the bow. The latter causes an increase in the friction experienced between the ship and surrounding water.

The main factors affecting total ship resistance are the friction and viscous effects of water acting on the hull, the energy required to create and maintain the ship's characteristic bow and stern waves, and air resistance. Accordingly total resistance (R_T) can be written as

$$R_T = R_V + R_W + R_{AA} \tag{2-1}$$

where: R_T = total hull resistance; R_V = viscous (friction) resistance; R_W = wave making resistance; R_{AA} = air resistance caused by ship moving through calm air. For an illustration of these basic hydrodynamic components of ship resistance see Figure 2-3 and Figure 2-4.

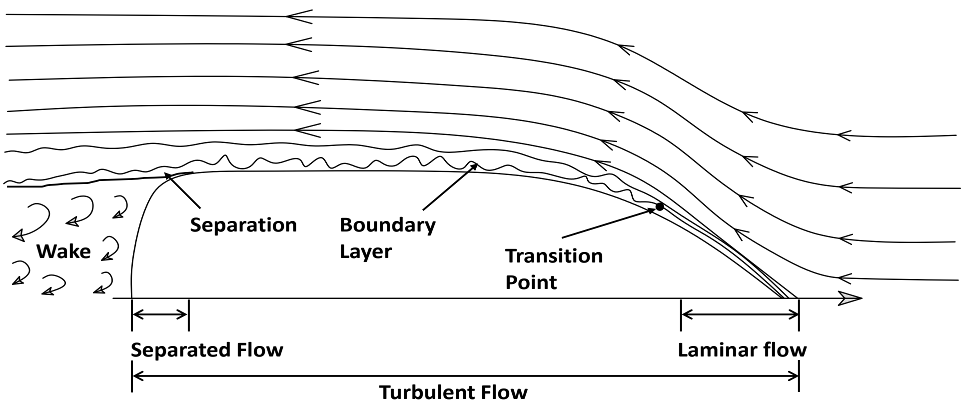


Figure 2-2 Typical water flow pattern around a ship's hull

Viscous resistance (R_V) is essentially the result of water rubbing against the ship side. This rubbing is prone to viscosity; a temperature dependent property of a fluid that describes its resistance to flow. Although water has low viscosity, it produces a significant friction force opposing ship motion. Experimental data have shown that water friction can account for up to 85%of a hull's total resistance at low speed (Froude No., Fn$\leq$ 0.2 or speed-to-length ratio less than 0.4 if the ship speed is expressed in knots), and 40%-50% of resistance for some ships at higher speeds. The influence of **frictional resistance** depends on the thin water viscosity boundary layer generated around the hull surface as a ship progresses in waves. Friction arises from the shear stresses in the fluid and acts tangential to the body. **Viscous pressure resistance** acts normal to the body. In the forward portion of the hull pressure forces act normal to the surface; However, in the aft portion of the hull the boundary layer reduces the forward acting component of pressure. This reduction results in a net resistance force due to pressure acting on the hull. The increase in resistance due to pressure is called "*viscous pressure drag*" or "*form drag*", and is sometimes also referred to as the normal component of viscous resistance. The shape of a ship's hull can influence the magnitude of viscous pressure drag. Ships that are short in length with wide beams (i.e., low length to beam ratio ships) will have greater form drag than those with a larger length to beam ratio. Also, ships that are fuller near the bow (e.g., bulk oil tanker) will have greater form drag than ships with fine bows (e.g., a destroyer or a container ship).

The **residual resistance (R_R)** comprises of a resistance component due to waves formed by ship motions and a component of eddy making resistance formed in way of the stern and projecting parts (e.g., bossing and bilge keels). Wave resistance relates to the fact that as a ship is progressing in waves she generates a typical wave system leading to energy losses. This wave system is decomposed into primary and secondary systems. The former assumes ideal fluid flow (i.e., no viscosity) leading to pressure distributions. Additionally, at the free surface even with the assumption of ideal fluid flow, a typical wave pattern that comprises of transverse and divergent waves is generated (see Figure 2-5). In deep waters, waves form a wedge shape known as ''*Kelvin pattern*''. These waves have a half-angle of 19.5^0 which is independent of the ship actual shape (see Figure 2-5).

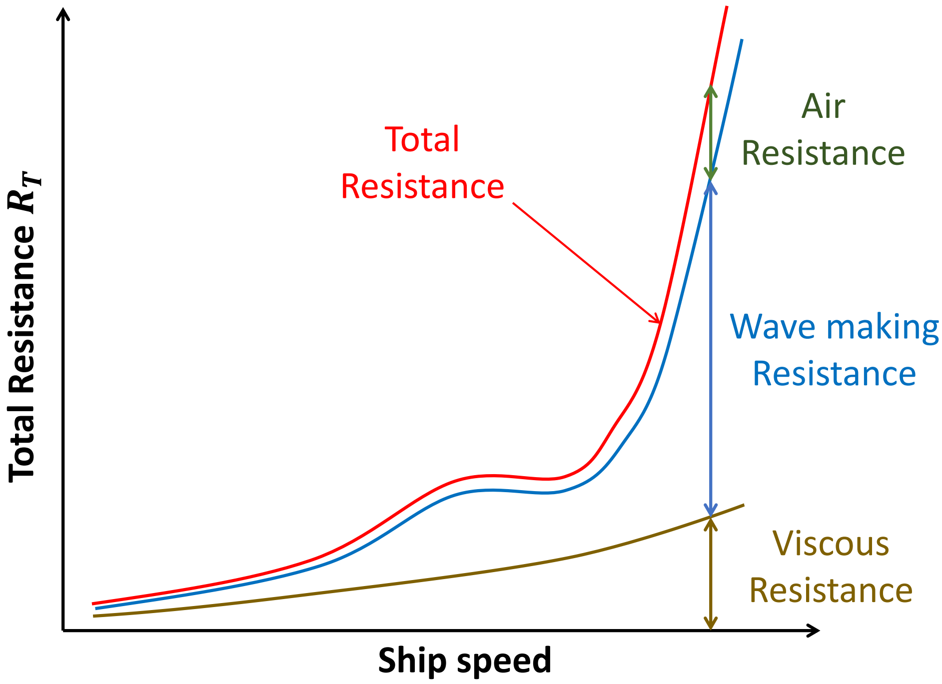


Figure 2-3 Proportional contribution of the components of hull resistance

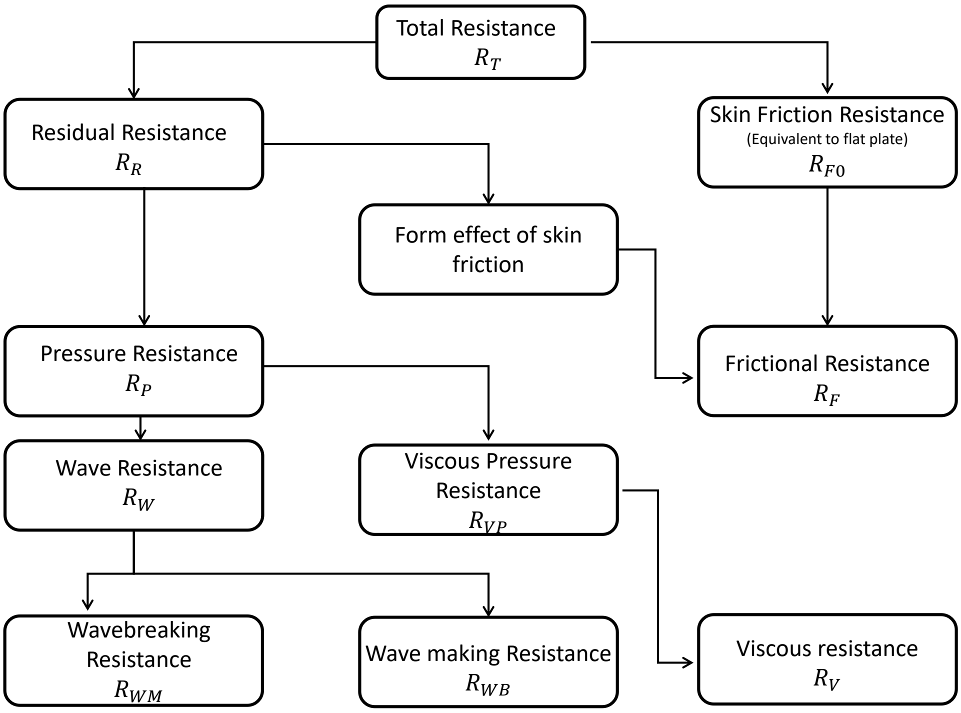
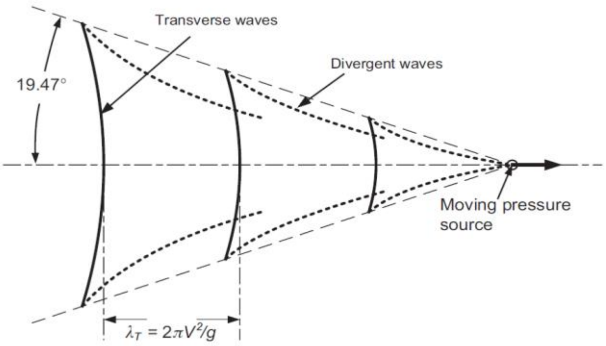
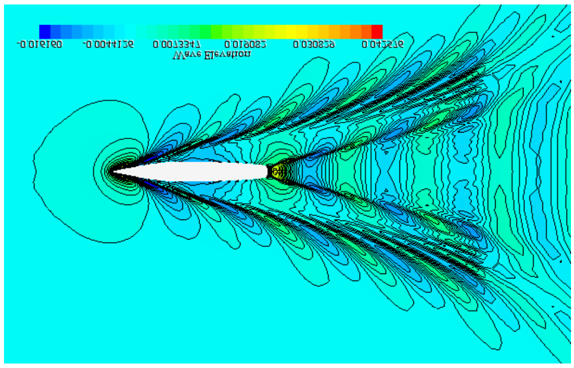


Figure 2-4 Hydrodynamic resistance components

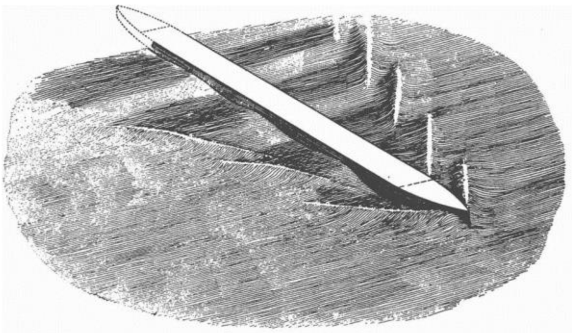
Air resistance (R_A) acts on portions of the ship above the water. It is the resistance caused by the flow of air over the ship with no wind present. This component of resistance is affected by the shape of the ship above the waterline, the area of the ship exposed to the air, and the ship's speed through the water. Ships with low hulls and small "sail area" or projected area above the waterline will naturally have less air resistance than ships with high hulls and large amounts of sail area. Resistance due to air is typically 4% - 8% of the total ship resistance, but may be as much as 10% in high sided ships such as aircraft carriers. Air resistance can be reduced by streamlining hulls and superstructures. **Wind resistance** on a ship is a function of the ship's sail area, wind velocity and direction relative to the ship's direction of travel. For a ship steaming into a 20-knots wind, a ship's resistance may be increased by up to 25% - 30%. Ocean currents can also have a significant impact on a ship's resistance and the power required to maintain a desired speed.



(a) Kelvin wave pattern (Molland, 2017)



(b) CFD simulation of Kelvin wave patterns (Duman and Bal, 2016)



(c) Froude's original sketch of wave patterns (RINA, 2010)

(d) Narrow Kelvin waves by container ship

Figure 2-5 A ship's secondary wave system is idealized by Kelvin wave patterns

Added resistance refers to ocean waves caused by wind and storms and is not to be confused with wave making resistance. Ocean waves cause the ship to expend energy by increasing the wetted surface area of the hull (added viscous resistance), and to expend additional energy by rolling, pitching, and heaving. This component of resistance can be very significant in high sea states. **Resistance in shallow waters** is caused by (a) the speed up of flow of water around the bottom of the hull leading to increased viscous resistance and (b) the consequent decrease of pressure under the hull, causing the ship to *"squat"*. The latter is associated with increased wetted surface area and frictional resistance patterns. The waves produced in shallow waters tend to be larger than waves produced in deep waters at the same speed. The energy required to produce these waves increases; i.e., wave making resistance increases in shallow water and therefore to travel in shallow waters requires more horsepower (and fuel).

3. The dynamics of resistance and thrust

Ship maneuverability related requirements are split into three broad categories namely (a) general maneuverability in open seas; (b) directional stability; (c) low speed maneuverability. These principles are challenged by resistance and thrust dynamics and therefore good design should involve sound understanding of the influence of flow around the hull, the basis of choice of slow and moderate speed steering and maneuvering equipment, and general propulsion dynamics (e.g., propeller - rudder interactions). With the later in mind this section explains principles of ship resistance, thrust, their dynamic interactions.

3.1 Powering calculations

Based on the generic definition $Power = Force \times speed$, the ship effective power (i.e., the power the ship would consume if she sails without power) is defined as

$$P_E = R_T \cdot V_s \tag{2-2}$$

Where $R_T$is the total water resistance excluding appendage resistance and V_s is the ship speed. The fluid flow will vary around the hull.

Thus, the propeller thrust will depend on the speed of advance (flow speed upstream the propeller) rather than the ship speed. The thrust power will then be defined as

$$P_T = T \cdot V_A \tag{2-3}$$

where T is the thrust force generated and V_A is the speed of advance. The thrust power needed during model tests is usually greater than the total resistance R_T. Thus, the propeller induces additional resistance associated with (a) pressure decreases in way of the aft ship body leading to increase of the inviscid resistance; (b) flow velocity increases around the aft - body that may lead to increases in frictional resistance.

To account for the above, we can combine resistance and thrust together using the so-called thrust deduction factor t defined as

$$t = 1 - \frac{R_T}{T} \tag{2-4}$$

The resistance that should be equivalent to the propeller thrust is given by the equation

$$X_{resistance} = -\frac{R_T}{(1-t)} = -\frac{0.5\rho u^2 S C_T}{(1-t)} \tag{2-5}$$

where ρ is the water density, u is the longitudinal-component flow velocity, S is the wetted hull surface, and C_T is the total resistance coefficient and this can be in tabular values against Froude number as a result of empirical analysis or model tests.

The speed of advance is generally slower than ship speed because of the effect of the ship's wake on the propeller inflow velocities. This wake is analyzed into mainly three components namely friction, potential and wave wake. Friction wake relates with flow velocity speed decreases at the aft ship body's boundary layer and leads to flow separation. The potential wake does not account for the effects of viscosity in way of the free surface and assumes there will be lower velocities at the stagnation points (i.e., in way of the bow and stern of the ship). The wave wake assumes that local velocities around the hull may have adverse or beneficial effects depending on the case. The total wake fraction is then defined as

$$w = 1 - \frac{V_A}{V_S} \tag{2-6}$$

Self-propulsion CFD simulations can be conducted to evaluate the thrust deduction factor and the wake fraction. Matusiak (2021) proposed approximate values for wake fraction and thrust deduction factors as shown in Table 2-1.

Table 2-1 Approximate values of the propulsive coefficients (Matusiak, 2021)

Item	**Single screw vessel**	**Multi-screw vessel**
Wake fraction w	0.25	0.05
Thrust deduction factor t	0.25	0.15

For those cases that the total resistance coefficient is not available the propulsive power P_D may be used to evaluate total resistance as follows

$$P_D = \frac{P_E}{\eta_D} = \frac{R_T \cdot V_S}{\eta_D},\ R_T = \frac{P_D \eta_D}{V_s} \tag{2-7}$$

Where η_D is the propulsive efficiency and given by

$$\eta_D = \eta_o \eta_R \frac{(1-t)}{(1-w)} \tag{2-8}$$

where the open water efficiency $\eta_o = 0.65$ and the relative rotative efficiency $\eta_R = 1$.

3.2 Ship thrust

Two models are used to represent the thrust developed by a propeller. **Model 1** is preferred for a fixed pitch propeller and comprises the total thrust evaluated from open water characteristics of the propeller (K_T-J curve) as follows:

$$X_{prop} = Z\rho n^2 D^4 K_T \tag{2-9}$$

where Z is the number of propeller blades, n is the propeller revolutions per second, D is the propeller diameter, and K_T is the thrust coefficient. Propeller revolutions should be adjusted for a required ship speed and to satisfy the condition ($X_{prop} = -X_{Re\,sis\,tan\,ce}$). Then, revolutions are kept constant or to be adjusted to keep the advance coefficient $J = \frac{V_A}{nD}$ constant depending on the type of machinery (see Bertram,2012).

Model 2 is preferrable for controllable pitch propellers (see Figure 2-6). It assumes that the delivered power and the propulsive efficiency are constant. This assumption yields the following relationship between the propeller thrust and the ship speed

$$X_{prop} = \frac{P_D \eta_o \eta_R}{V(1-w)} \tag{2-10}$$

This model allows good control of the propeller pitch, but it is only optimized for certain operational conditions. Thus, efficiency losses in other operating conditions are not considered.

(a) Fixed pitch propeller

(b) Controllable pitch propeller

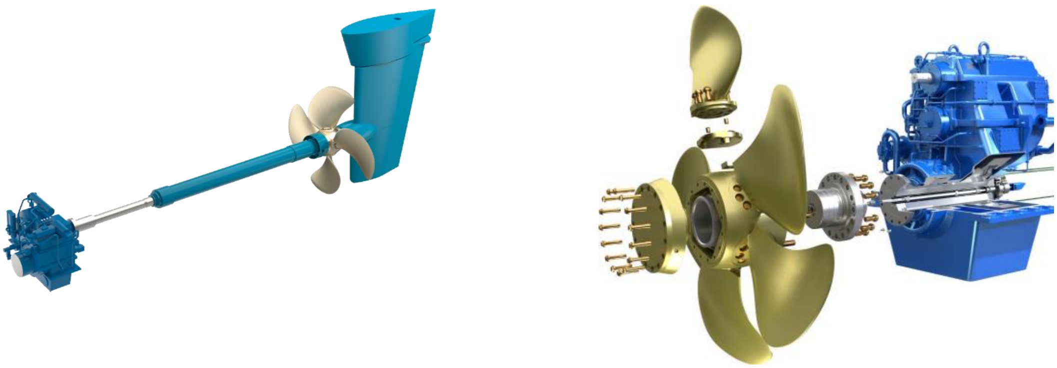


Figure 2-6 controllable and fixed pitch propellers ©Wärtsilä (2020)

4. Ship rudders and their dynamics

Marine rudders are primary control surfaces with streamlined sections used to steer ships through water. They are designed to give good lift and drag ratio and are of double plate construction (see Figure 2-7). They are classified based on how close the center of pressure is to the rudder axis and accordingly known as balanced, semi-balanced and unbalanced rudders.

Whether a rudder is balanced or not is dependent upon the relationship of the center of pressure of the rudder and the position of the rudder stock. For example, rudders that are vertically aligned are fully balanced. On the other hand, when the rudder stock is at the leading edge, it is unbalanced. This is a common arrangement in merchant ships where rudder forces are not excessive. When no lower pintle is used the rudder is termed as **spade**. Spade rudders are commonly used in warships. A **flap rudder** has the feature of a flap at the trailing edge of the rudder. It can improve lift by modifying the airfoil shape and by inducing increase to the turning angle of the rudder. The **flettner rudder** uses two narrow flaps at the trailing edge. As the flaps move to assist the main rudder movement the torque required by the steering gear is reduced. **Active rudders** are spade-type rudders. They include a housing with small electric motor driving small propeller to provide rudder force even when the ship is at rest (**Error! Reference source not found.** and Figure 2-14).

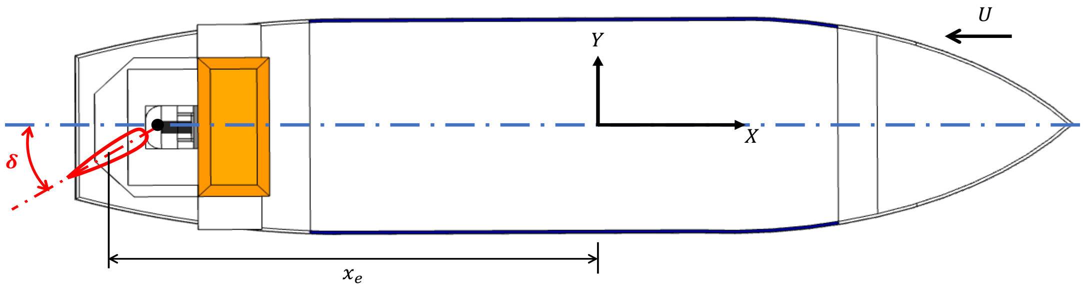


Figure 2-7 Rudder concept in ship steering; water flow in negative X direction, x_e is the distance between the center of the rudder and the center of gravity of the hull, U is the velocity of incoming to the hull waves and δ the abrupt rudder angle (Shijie et al. 2018)

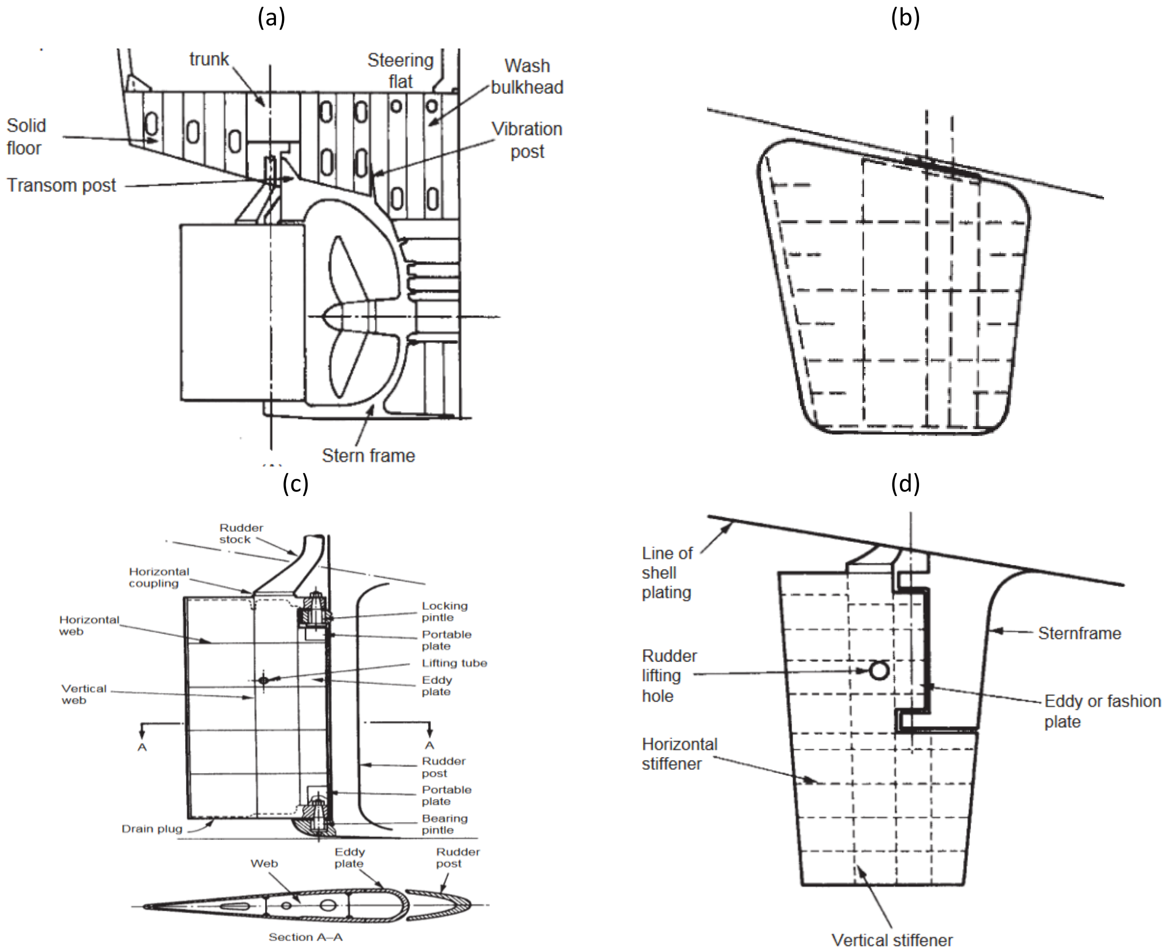
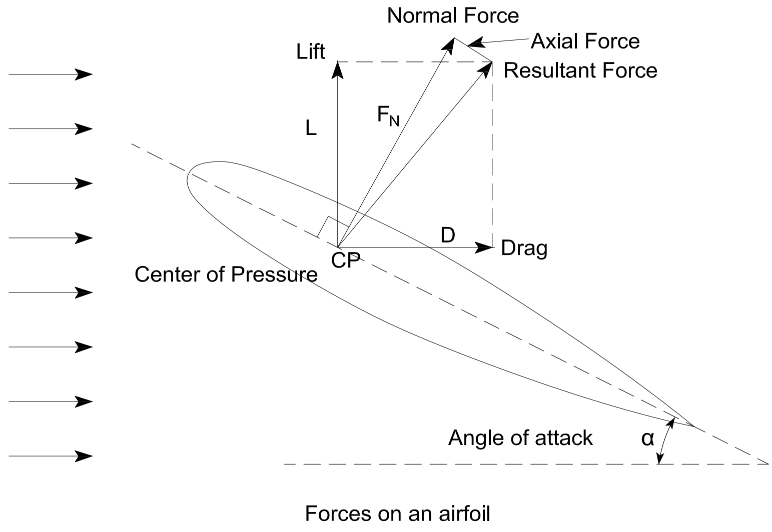


Figure 2-8 Typical rudder types (a) simplex balanced (b) spade balanced (c) unbalanced (d) semi-balanced rudder (Tupper, 2013)



$$C_L = \frac{L}{\frac{1}{2}\rho U^2 c}$$

$$C_D = \frac{D}{\frac{1}{2}\rho U^2 c}$$

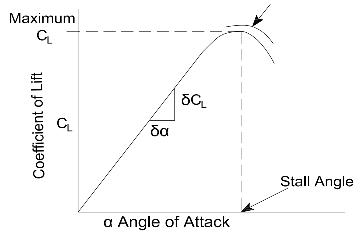
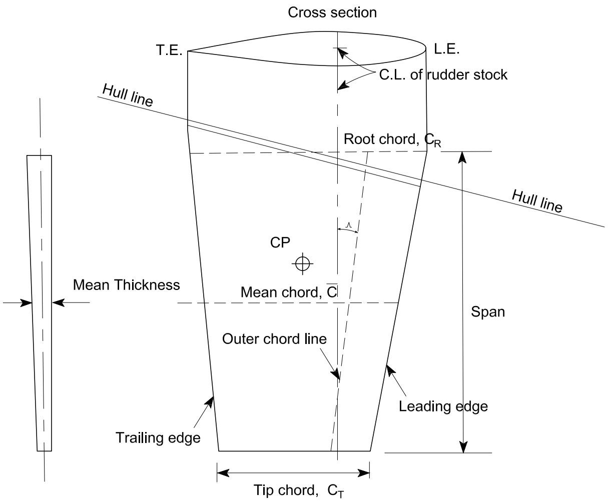


Figure 2-9 Forces on an Airfoil (The lift - L is defined as the component of force perpendicular to the freestream velocity vector. The drag - D is defined as the component of force parallel to the free-stream velocity). The lift increases with increasing angle of attack. Eventually the adverse pressure gradient causes separation over the entire upper surface of the foil, resulting in a loss of lift. The maximum obtainable lift coefficient is called $C_{L,max}$.



- **Mean Span** - average of leading and trailing edge spans
- **Mean Chord** - average of the root and tip chords
- **Profile Area** - product of mean span and mean chord
- **Aspect Ratio** - ratio of mean span to the mean chord
- **Taper Ratio** - ratio of the tip chord to the root chord
- **Sweepback Angle** - angle between $1/4$ chord line and vertical
- **Mean Thickness** - average of the max thickness of the foil at the root and tip

Figure 2-10 Typical rudder design definitions

The rudder profile needs to fit within dimensions dictated by the hull shape. The span is limited by the vessel draft. Therefore, it should not extend below the baseline or penetrate the water surface. The chord is limited by propeller clearance and stern shape. The usual distance between the propeller and the rudder is 0.2 propeller diameter. The rudder should be designed for minimum drag at all speeds. usual section shape is NACA 0015 to 0021 (relatively thick) – see Figure 2-15. These foils have a relatively constant center of pressure and thick sections are better structurally.

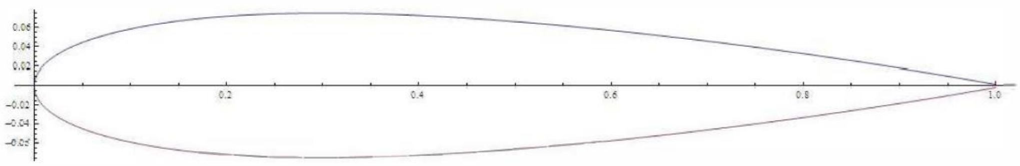


Figure 2-11 Typical Rudder Section

The undesirable effects of the rudder on the ship (e.g. rudder induced vibration) should be kept to a tolerable level. From a hydrodynamic perspective, the basic considerations in rudder design can be summarized as follows:

- Full form ships need larger rudders
- Generally, large rudders provide superior performance
- Rudders should be placed in way of the propeller wake with the aim to improve efficiency
- High aspect ratios improve rudder efficiency. However their practical use may be limited by the hull shape (span by draft; chord by stern shape)
- An Increased rate of swing is good for small ships. However, large ships benefit more from rudder area than from swing rate

According to DNV (2015) a good first estimate of rudder area can be achieved using the following formulae

$$A_R = \frac{T \times L_{BP}}{100}\left[1 + 25\left(\frac{B}{L_{BP}}\right)^2\right] \qquad (2\text{-}11)$$

The above formula applies only for single rudders operating in a propeller wake. For all other arrangements DNV requires a 30% increase in area. The idea behind this choice is that increased area leads to increased flow and better efficiency over the rudder at low speeds. It is useful to compare values from Eq. (2-11) with values used in industry. As shown in Table 2-2 and Table 2-3 the rudder performance is more affected by span length than chord length. An increase in the aspect ratio increases the lift/drag ratio.

Table 2-2 General vessel hull form coefficients

Vessel Type	Typical Form Coefficients and Ratios: C_B	L/B	B/T	Speed V, knots	Froude No. $V/\sqrt{gL}$	Number of Propellers/ Rudders	Rudder Area Ratios[a]	Dynamic Course Stability[b]
Harbor tug	0.50	3.3	2.1	10	0.25	1/1	0.025	S
Tuna seiner	0.50	5.5	2.4	16	0.31	1/1	0.025	S
Car ferry	0.55	5.1	4.5	20	0.34	2/2	0.020	S[e]
Container high speed	0.55	8.3	3.0	28.5	0.53	2/2	0.015	S
Container high speed	0.55	8.3	3.0	28.5	0.53	2/1	0.025	S[c]
Cargo liners	0.58	6.9	2.4	21	0.29	1/1	0.015	S
RO/RO	0.59	6.9	3.0	22	0.26	1/1	0.015	S
Barge carrier	0.64	7.5	2.9	19	0.20	1/1	0.015	S
Container Med. Speed	0.70	7.1	2.8	22	0.25	1/1	0.015	S
Offshore supply	0.71	4.7	2.75	13	0.28	2/2	0.016	S[d,e]
General cargo low speed	0.73	6.7	2.4	15	0.20	1/1	0.015	S
Lumber low speed	0.77	6.7	2.6	15	0.20	1/1	0.025	S
LNG (125 000 m³)	0.78	6.8	3.7	20	0.20	1/1	0.015	U
OBO (Panamax)	0.82	7.5	2.4	16	0.17	1/1	0.018	U
OBO (150 000 dwt)	0.85	6.4	2.4	15	0.15	1/1	0.017	U
OBO (300 000 dwt)	0.84	6.0	2.5	15	0.14	1/1	0.015	U
Tanker (Panamax)	0.83	7.1	2.4	15	0.16	1/1	0.015	U
Tanker 100 000 to 350 000 dwt	0.84	6.2	2.4	16	0.15	1/1	0.015	U
Tanker 350 000 dwt	0.86	5.7	2.8	16	0.13	1/1	0.015	U
U.S. river towboat	0.65	3.5	4.5	10	0.25	2/2	...	U[d,e]

[a] Not for design guidance.
[b] U = unstable course stability; S = stable course stability.
[c] Although the vessel is directionally stable, maneuvering is difficult at low speeds when the propeller wash is not effective over the rudder.
[d] Maneuverability is good owing to installation of Kort nozzles, flanking rudders, and other capabilities.
[e] Twin screw because of restricted draft.

Table 2-3 Rudder area coefficients

Vessel Type	Percent of
Single-screw vessels	$L \times T$
Twin-screw vessels	1.6 to 1.9
Twin-screw vessels with two rudders (total	1.5 to 2.1
area)	2.1
Tanlrers	1.8 to 1.9
large passenger vessels	1.2 to 1.7
Fast passenger vessels for canals	1.8 to 2.0
Coastal vessels	2.3 to 3.3
Vessels with increased maneuverability	2.0 to 4.0
Fishing trawlers and vessels with limited	2.5 to 5.5
saifing area	
Seagoing tags	3.0 to 6.0
Sailing vessels	2.0 to 3.0
Pilot vessels and faxries	2.5 to 4.0
Motorboats	4.0 to 5.0
Keeled launches and yachts	5.0 to 12.0
Centerboard boats	30 or more

The way the rudder affects the forces acting on the flow can be shown using a linear model by setting the rudder to an arbitrary angle δ. This action develops a horizontal force is given as:

$$Y_R = Y_\delta \delta \tag{2-12}$$

where $Y_\delta = \partial Y / \partial \delta$ is a hydrodynamic derivative in way of the axis of reference (Matusiak, 2021). This is followed by a negative turning moment $N_\delta \delta$ which results in a turning motion of the ship to an angle β and amplified by a turning moment $N_v v$ as the ship is positioned in an inclined flow (see

Error! Reference source not found.). There are two main realistic ways to evaluate the rudder forces. The first is known as the **non-linear maneuvering model** that utilizes the hull forces as a function of the rudder angle defined by Eqs. (2-13) to (2-15) below.

$$\begin{aligned} X = {} & X_{\dot{u}}\dot{u} + X_u u + X_{uu}u^2 + X_{uuu}u^3 + X_{vv}v^2 + X_{rr}r^2 + X_{\delta\delta}\delta^2 + X_{vr}vr \\ & + X_{v\delta}v\delta + X_{r\delta}r\delta + X_{vvu}v^2u + X_{rru}r^2u + X_{\delta\delta u}\delta^2u + X_{r\delta u}r\delta u \\ & + X_{rvu}rvu + X_{v\delta u}v\delta u + X_{r\delta v}r\delta v \end{aligned} \tag{2-13}$$

$$\begin{aligned} Y = {} & Y_{uu}u^2 + Y_{\dot{v}}\dot{v} + Y_{\dot{r}}\dot{r} + Y_v v + Y_r r + Y_\delta \delta + Y_{\delta u}\delta u + Y_{vu}vu + Y_{ru}ru \\ & + Y_{vuu}vu^2 + Y_{ruu}ru^2 + Y_{\delta uu}\delta u^2 + Y_{vvv}u^3 + Y_{rrr}r^3 + Y_{\delta\delta\delta}\delta^3 \\ & + Y_{rr\delta}r^2\delta + Y_{vrr}vr^2 + Y_{rvv}rv^2 + Y_{\delta vv}\delta v^2 + Y_{vr\delta}vr\delta \\ & + Y_{\delta\delta r}\delta^2 r + Y_{\delta\delta v}\delta^2 v \end{aligned} \tag{2-14}$$

$$\begin{aligned} N = {} & N_{uu}u^2 + N_{\dot{v}}\dot{v} + N_{\dot{r}}\dot{r} + N_v v + N_r r + N_\delta \delta + N_{\delta u}\delta u + N_{vu}vu + N_{ru}ru \\ & + N_{vuu}vu^2 + N_{ruu}ru^2 + N_{\delta uu}\delta u^2 + N_{vvv}v^3 + N_{rrr}r^3 \\ & + N_{\delta\delta\delta}\delta^3 + N_{rr\delta}r^2\delta + N_{vrr}vr^2 + N_{rvv}rv^2 + N_{\delta vv}\delta v^2 \\ & + N_{rv\delta}vr\delta + N_{\delta\delta r}\delta^2 r + N_{\delta\delta v}\delta^2 v \end{aligned} \tag{2-15}$$

where X and Y are the x- and y-directional force components, N is the fluid moment. X, Y and N are dependent upon the motion variables (u, v and r), their time derivatives and the rudder angle δ. X, Y and N terms marked with the subscripts are called slow motion derivatives; their indices depict the variable they apply to.

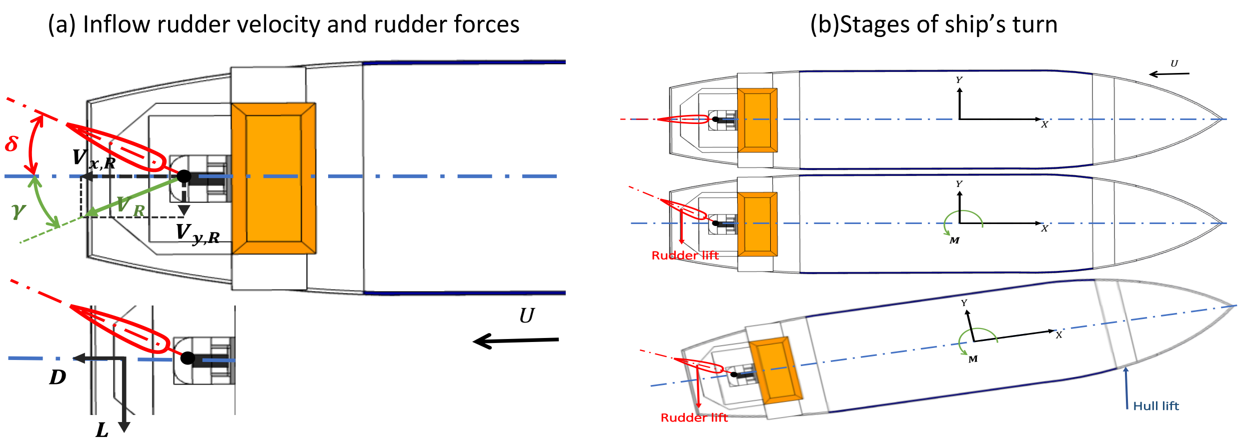


Figure 2-12 The influence of rudder dynamics on a ship's turning ability

Another model to be used is the **modular model** in which the flow in way of the rudder is modelled mathematically. The inflow velocity to the rudder is influenced by yaw and sway motions of the ship. It therefore influences the propeller slipstream flow and the flow velocity. The angle of attack of the flow is not the same as the rudder angle. Consequently, the drag and lift forces depend on the flow velocity in way of the rudder defined as

$$\begin{Bmatrix} V_{x,R} \\ V_{y,R} \\ V_{z,R} \end{Bmatrix} = \begin{Bmatrix} V_x - V_{x,wave} \\ -v + V_{y,wave} \\ -w + V_{z,wave} \end{Bmatrix} - \begin{Bmatrix} -i & j & k \\ p & q & r \\ x_R & y_R & z_R \end{Bmatrix}$$
$$\begin{aligned} V_{x,R} &= V_x - V_{x,wave} - qz_R + ry_R \\ \rightarrow V_{y,R} &= -v + V_{y,wave} - rx_R + pz_R \\ V_{z,R} &= -w + V_{z,wave} - py_R + qx_R \end{aligned} \tag{2-16}$$

where V_x is the longitudinal (x-component) of flow velocity along the propeller slipstream; $p, q\ and\ r$ are the angular velocity components expressed in the moving co-ordinate system x, y, z. The subscript ''*wave*'' is referred to the flow velocities due to the contribution of wave action; (x_R, y_R, z_R) represent the positions of the rudder in the body-fixed coordinate system (see Figure 1-2 and Taimuri et al.,2020)); V_R is the rudder flow velocity vector with angle of attack $\alpha = \delta + \gamma$ where γ is the change of the angle of attack due to the ship motion and the waves (see Figure 2-12)

$$\gamma = \arctan\left(V_{y,R}/V_{x,R}\right) \tag{2-17}$$

A typical ship's rudder is limited to angles in the range of$\pm$ 35^0. This is because at greater angles than these the rudder is likely to stall. Figure 2-13 shows the development of stall as rudder angle increases. By following the basic aerodynamic theory (see Figure 2-9) lift and drag forces are defined as

$$L = \frac{1}{2}\rho C_L A_R V_R^2 \tag{2-18}$$

$$D = \frac{1}{2}\rho C_D A_R V_R^2 \tag{2-19}$$

where $A_R$is the projected area of the side view of the rudder. Consequently, the lift C_L and drag C_D coefficients become:

$$C_L = \frac{2\pi\Lambda(\Lambda+1)}{(\Lambda+2)^2} sin(\delta+\gamma)$$

$$C_D = 1.1\frac{C_L^2}{\pi\Lambda} + C_{D0} \tag{2-20}$$

where $\Lambda = b^2/A_R$ is the aspect ratio between the (rudder length)^2 over the projected area of the side view of the rudder.

The viscous drag coefficient can be calculated according to the ITTC 1957 (Morrall, 1970) as a function of frictional resistance coefficient:

$$C_{D0} = 2.5\, C_F = 2.5\frac{0.075}{(log\, Re-2)^2}$$

$$Re = \frac{V_R c}{v} \tag{2-21}$$

where Re is Reynold's number, c is the mean value of the rudder cord and v is the kinematic viscosity. At small angles, rudder lift is created due to the difference in flow rate across the port and starboard sides of the rudder. However, as the rudder angle increases, the amount of flow separation increases until a full stall occurs in the range of 30^0 to 40^0 (see Figure 2-13). The amount of lift achieved by the rudder reduces significantly after a stall and is matched by a rapid increase in drag. Consequently, rudder angle is limited to values less than the stall angle.

If we assume that the rudder is positioned directly in the propeller slipstream, the rudder forces are higher than the ones generated by a rudder placed outside the propeller. The distance between the propeller and the rudder is small. This results in reduction of the axial velocity of the slipstream V_x. The velocity at the rudder can be then represented as a function of the velocity far downstream the propeller, the radius of the flow far downstream the propeller, and the radius of the slipstream at the rudder

$$\frac{V_x}{V_\infty} = \left(\frac{r_\infty}{r}\right)^2$$

$$\frac{r}{r_0} = \frac{0.14(r_\infty/r_0)^3 + (r_\infty/r_0)(r_\infty/r_0)^{1.5}}{0.14(r_\infty/r_0)^3 + (r_\infty/r_0)^{1.5}} \tag{2-22}$$

where V_∞ is the mean flow velocity far downstream the propeller, r_∞ is the radius of the slipstream far behind the propeller, r is the slipstream radius at the rudder, r_0 is the propeller radius.

Finally, if we assume an ideal propulsor model, the axial mean velocity and the slipstream radius far behind the propeller can be approximated by the formulae

$$V_\infty = V_A\sqrt{1+C_T} \quad \text{for } C_T = \frac{Thrust}{0.5\rho V_A^2 \pi D^2/4} = \frac{8}{\pi}\frac{K_T}{J^2} \quad \text{while} \quad r_\infty = r_0\sqrt{\frac{1}{2}\left(1+\frac{V_A}{V_\infty}\right)} \qquad \text{(2-23)}$$

where C_T is the thrust loading coefficient, and D is the propeller diameter.

(a) Rudder Flow Patterns at Increasing Rudder Angle (b) Lift versus rudder angle

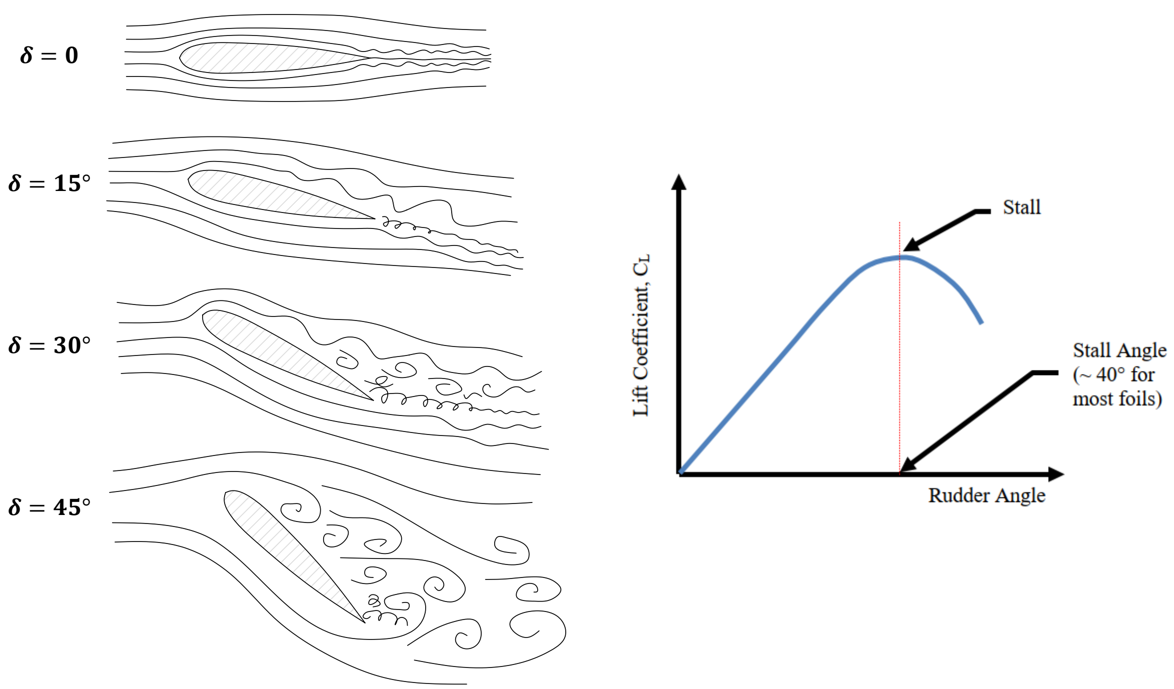


Figure 2-13 Rudder flow patterns at increasing rudder angle

5. The influence of slow speed maneuverability

Safe and sustainable operations require that a ship maintains sufficient and independent directional control in restricted waters such as in ports and channels and in various environmental conditions. Such maneuvers take place at low speeds where rudders are limited in terms of their effectiveness due to lack of flow across their surface. To overcome such problems, over the years, classic propulsion systems improved as follows :

- *The rudder is often positioned directly upon the control surface.* A skilled helmsman can then combine the throttle control and rudder angle to vector thrust laterally and create a larger turning moment with minimal advance and transfer;
- *Two propellers can be set to work in unison (twin propulsion concept).* Such system offers the opportunity to apply a *"twist maneuver"* ; i.e., turn one propeller in reverse direction and the

other in forward direction and therefore create large turning moments with hardly any forward motion;

- *Lateral thrust at the bow and the stern is possible when bow thrusters are enclosed in transverse tubes/tunnels (see Figure 2-14) known as suction tunnels*. In typical thruster models the suction tunnel length is usually two to three times the diameter of the unit. The average bow thruster power in ferries is 0.54 KW/m^2 (total bow thruster power/projected windage area), varying up to 0.96 kW/m^2. Nowadays, the tendency seems to be towards 0.6 – 0.8 kW/m^2. Stern thrusters are dimensioned at 0.2 – 0.25 kW/m^2 (Wärtsilä, 2020).
- *Rotational thrusters (also known as azimuth propulsion systems) that rotate up to 360^0 (see* Figure 2-15*) can be used to improve efficiency in oblique flow conditions.* These are shaft-less electric propulsion systems prone to minimal noise and vibration. A simple model of the forces acting on the azimuth propellers is outlined on Figure 2-15 (b). In this model that the rate of change of the pod is small. Thus, we deal with it in a quasi-stationary manner wile assuming incompressible flow. Note that we also do not deal with the forces acting on the pods but the propeller disk (forces acting on the pod and the strut are disregarded).

(a) (b)

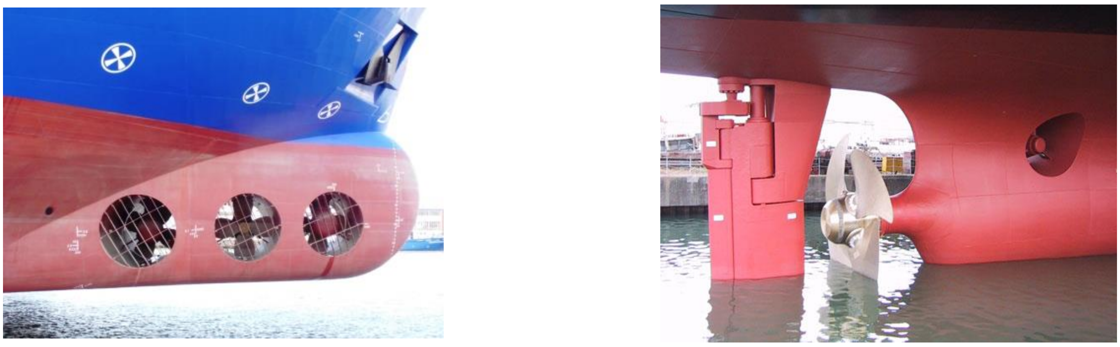
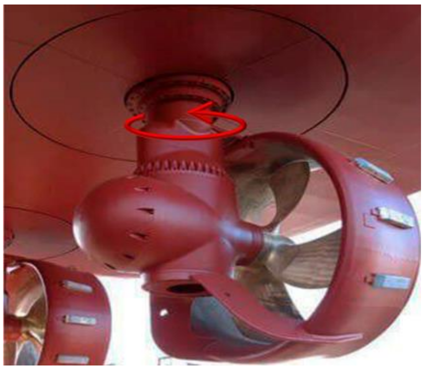


Figure 2-14 (a) Bow thrusters (b) stern thruster © kamome-propeller

(a) Typical rotational thruster design ©Kvaerner Masa, Finland

(b)Oblique flow kinematics for a thruster unit (for further details on symbols and equations see Matusiak, 2021)



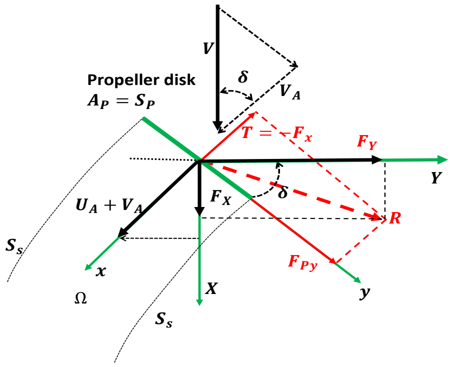


Figure 2-15 Rotational thruster dynamics

The forces acting on a propeller in oblique flow are changing **axially and radially**. Accordingly, an in-plane force will be developed in addition to the axial component and the magnitudes of corresponding unsteady forces and moments amplify. The governing equation used to derive these forces is the momentum equation. This equation in its integral form states that : *''the rate of change of momentum in an arbitrary domain is equal to the body forces and the surface forces acting on the domain boundaries''*. Mathematically this is expressed as follows

$$\frac{D}{Dt}\int_{\Omega}(\rho v_i)d\Omega = \frac{\partial}{\partial t}\int_{\Omega}\rho v_i d\Omega + \int_{S}\rho v_i(\vec{v}\cdot\vec{n})dS = \int_{S} q_i dS + \int_{\Omega} F_i d\Omega \qquad (2\text{-}24)$$

where Ω denotes the arbitrary control volume; $\frac{\partial}{\partial t}\int_{\Omega}\rho v_i d\Omega$ is the rate of change of momentum in the control volume; $\int_S \rho v_i(\vec{v}\cdot\vec{n})dS$ represents the total momentum flux outflow the domain/control volume; $\int_S q_i dS$ are the surface forces acting on the boundaries of the domain and $\int_{\Omega} F_i d\Omega$ are the body forces in the domain which may be assumed to be small (negligible). The surface forces at any boundary are disregarded unless for the surface of the propeller disk S_P (see Figure 2-15 b). The momentum equation is accordingly reduced to:

$$\int_{A_P} q_i dS = -\int_{S_p}\rho v_i(\vec{v}\cdot\vec{n})dS = \vec{R} \qquad (2\text{-}25)$$

where A_P is the propeller disk area. Equation (2-25) demonstrates that the total momentum flux acting on the propeller disk is equal to the total surface forces $\vec{R}$ acting on it. The in-plane component of the surface forces acting on the propeller disk is given by:

$$F_{Py} = \rho A_P (V_A + U_A) V_S \sin\delta \tag{2-26}$$

where U_A is the propeller induced velocity in the propeller plane and given as:

$$U_A = \frac{1}{2} V_A (-1 + \sqrt{1 + C_T}) \tag{2-27}$$

We have two coordinate systems namely (a) the ship fixed coordinate system $(X - Y)$ and (b) the one that moves with the pod $(x - y)$.The X-component force (F_x) drives the ship in ahead/astern direction. The Y-component force (F_y) acts instead of a rudder. These components can be defined in terms of the propeller thrust $T$and the in-plane force F_{Py}as:

$$F_x = T \cos\delta - F_{py} \sin\delta \tag{2-28}$$

$$F_y = T \sin\delta + F_{py} \cos\delta \tag{2-29}$$

Another difference between conventional propellers and azimuthing ones is the difference in the stopping action. The later have quite faster stopping than the former.

6. Stabilisation systems

Some conventional ship stabilisation systems that may be used to control roll motions are shown in Figure 2-16**:**

Bilge keels are the most conventional passive systems for roll motion reduction. They are plates projecting out from the turn of bilge. They extend over half to 2/3 of the ship's length. The projection out from the bilge of the keel should not be too large to avoid damage or collision especially in restricted waters. However, they may be extended enough to penetrate the boundary layer. The movement of a body of water with the ship due to the existence of the bilge keel may lead to reduction in the amplitude of roll and increase in the roll period. They work by generating drag forces which oppose the rolling motion of the ship. Thus, their effects are prominent at higher speeds. The advantage of bilge keels is that they are simple, inexpensive and can be maintained as part of the hull maintenance procedure. Their disadvantage is that they increase the resistance of the ship.

Passive tanks are stabilizers that involve a sloshing liquid used to produce damping and restoring forces. They work by shifting weight of the liquid so that it exerts a roll moment on the ship and (by suitable design) this can be arranged to damp roll motion. Accordingly, the natural frequency of the tank should be equal or near the ship's natural frequency and to achieve this the tank is tuned by adjusting the amount of liquid in the tank or by the design of a baffle. A passive tank system is good choice if space and weight requirements are not of major concern. There are no moving parts and it requires little maintenance. U shaped passive tanks include water that moves and helps the ship stabilize as she inclines. A drawback for such system is that the tank can be adjusted to only one frequency corresponding to the roll natural period at which large motion amplitudes occur. The water free surface in the tank may affect stability and must be allowed for.

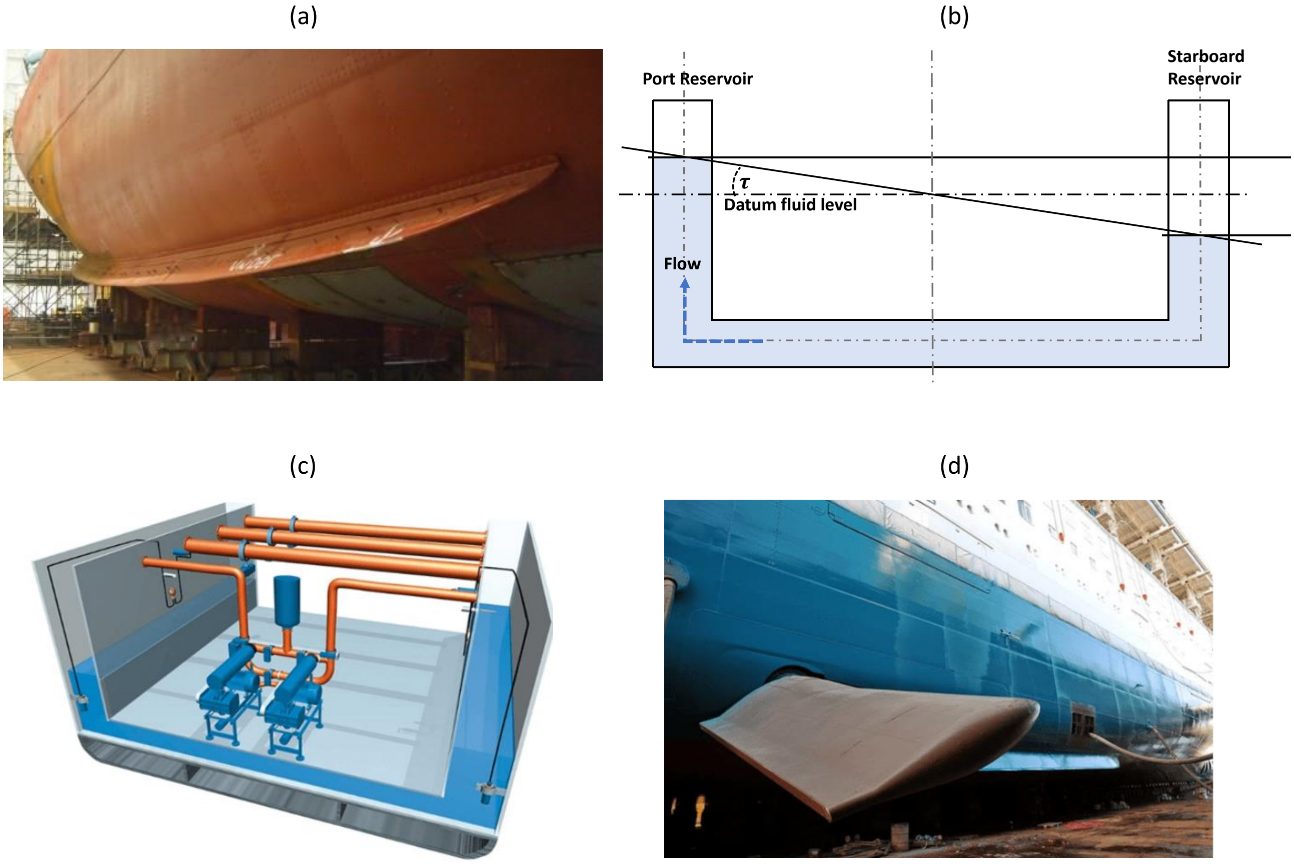


Figure 2-16 (a) Bilge keel (b) Passive rolling tank (c) active rolling tank (d) active fins ©Marine insight

Active tanks are similar in principle to passive tanks, but the movement of water is controlled by pumps or by air pressure above the water surface instead of the passive action of water. They can be two separate tanks on both sides of the ship or connected with lower limb. The system can deal with more than one frequency and in similar fashion to the passive system it can stabilize at zero ship speed. There are no projections outside the hull.

Active Fins are active systems including one or more pairs of fins fitted on both sides of the ship. They move with the aid of an actuator in response to signals based on gyroscopic measurements of roll motions. The fin may move totally or partly. The fins may permanently protrude from the bilge or may, at the expense of some complication, be retractable. The lift force on the fin is proportional to the square of the ship's speed. At low speed they will have little effect although the control system can adjust the amplitude of the fin movement to take account of speed, using larger fin angles at low speed.

Active T-foils are the same on shape and principle to active fins. However, they have active flaps that can be adjusted to reduce roll, pitch and heave motions. They are commonly noticed in catamaran ships as they are relatively shorter than displacement ships and motion reduction in this type of ships is demanding.

7. Control Systems

A mathematical description of ship motions is important to facilitate and verify the design of motion control systems. Several mathematical models can be used for this purpose. For example a 6 - DOF model in moderate waters is formulated by six nonlinear differential equations in three-coordinate systems namely inertial-, body- and horizontal body - coordinate systems. The equations are expressed as follows

$$\begin{gathered} m(\dot{u}-v\dot{\psi})=\left(m_y(\omega_e)-X_{v\dot{\psi}}\right)u\dot{\psi}-m_x(\omega_e)\dot{u}-m_z(\omega_e)\dot{\theta}+ \\ -m_z(\omega_e)w\dot{\theta}+X_{FK}+X_{RF}+X_{SF}+T\left(1-t_p\right)-R \end{gathered} \tag{2-30}$$

$$\begin{aligned} m(\dot{v}-u\dot{\psi})=m_x(\omega_e)u\dot{\psi}-m_y(\omega_e)\dot{v}-Y_v v-Y_{\dot{\psi}}\ddot{\psi}+ \\ +Y_{\dot{\psi}}\dot{\psi}+Y_{v|v|}v|v|+Y_{v|\dot{\psi}|}v|\dot{\psi}|+Y_{\psi|\dot{\psi}|}\psi|\dot{\psi}|+ \\ +Y_{FK}(\omega_e)+Y_{RF}+Y_{SF}+Y_{DF}(\omega_e) \end{aligned} \tag{2-31}$$

$$\begin{gathered} m\dot{w}=-m_z(\omega_e)\dot{w}-Z_w(\omega_e)w-Z_{\dot{\theta}}(\omega_e)\theta-Z_{\dot{\theta}}(\omega_e)\dot{\theta}- \\ -Z_{\ddot{\theta}}(\omega_e)\ddot{\theta}+Z_{FK}(\omega_e)+Z_{DF}(\omega_e)+Z_{SF}+mg \end{gathered} \tag{2-32}$$

$$\begin{gathered} (I_{xx}+J_{xx})(\omega_e)\ddot{\phi}-(I_{xx}+J_{xx})(\omega_e)\dot{\theta}\dot{\psi}=-K_{\dot{\phi}}\dot{\phi}+ \\ +\left(Y_v v-Y_{\dot{\psi}}\dot{\psi}\right)z_H+K_{FK}(\omega_e)+K_{RF}+K_{SF}+K_{DF}(\omega_e) \end{gathered} \tag{2-33}$$

$$\left(I_{yy}+J_{yy}\right)(\omega_e)\ddot{\theta}+(I_{xx}+J_{xx})(\omega_e)\dot{\psi}\dot{\phi}=-M_{\dot{\theta}}(\omega_e)\theta- \\ -M_{\dot{\theta}}(\omega_e)\dot{\theta}-M_{\dot{w}}(\omega_e)\dot{w}+M_{FK}(\omega_e)+M_{SF}+M_{DF}(\omega_e) \tag{2-34}$$

$$(I_{zz}+J_{zz})(\omega_e)\ddot{\psi}-(I_{xx}+J_{xx})(\omega_e)\dot{\theta}\dot{\phi}=N_{\psi|\dot{\psi}|}\psi|\dot{\psi}|-N_{\dot{\psi}}\dot{\psi} \\ -N_v v+N_{vv\dot{\psi}}v^2\psi+N_{v\dot{\psi}\dot{\psi}}v\dot{\psi}^2+N_{\dot{\psi}|\phi\phi}\dot{\psi}|\phi|+N_{DF}(\omega_e) \\ +N_\phi\phi-N_{\dot{v}}\dot{v}-N_{\dot{\psi}}\dot{\psi}+N_{v|\phi|}v|\phi|+\big(Y_{\dot{\psi}}\dot{\psi}+Y_{v|v|}v|v|+ \\ +Y_{v|\psi|}v|\dot{\psi}|-Y_v v\big)x_H+Y_{\psi|\psi|}\psi|\dot{\psi}|+N_{FK}(\omega_e)+N_{RF}+N_{SF} \tag{2-35}$$

where m refers to ship mass, I is the moment of inertia, $X, Y\ and\ Z$ are the external forces; $K, M\ and\ N$ are the external moments to surge, sway and heave, respectively; $u, v\ and\ w$ are surge, sway and heave velocities; $\phi, \theta\ and\ \psi$ are roll, pitch and yaw angles; ω_e is the encounter wave frequency; subscripts DF, FK, RF and SF represent the Froude-Krylov diffraction, the rudder and the stabilizing forces; x_H is coordinate of the midship; Z_H is the coordinate of the z point; R is the ship resistance and t_p represents the thrust deduction factor.

Each of the coefficients in the above equations can be defined analytically and then can be verified by model tests and/or full-scale measurements. It can be clearly observed that the energy from waves is distributed between the different motions. These motions interact with each other in the 6-DOF model. However, the roll motion, which is the most critical, can be simplified by 1-DOF equation as follows :

$$(I_{xx}+J_{xx})\ddot{\phi}+B_{xx}(\phi;\dot{\phi})+\Delta GZ(\phi)=K_{ext} \tag{2-36}$$

Where Δ is ship displacement; $I_{xx} and\ J_{xx}$ are water mass and added mass moments of inertia; ϕ is roll angle and B_{xx} is the dumping moment due to the skin frictional, eddy formation, bilge keel and fin. The righting arm GZ can be expressed as a nonlinear function of the roll angle and K_{ext} and is practically a moment generated by the side wave:

$$K_{ext}(t)=\Delta\cdot GZ\cdot\chi_A(t)\cdot\alpha_m(t) \tag{2-37}$$

where χ_A is a reduction coefficient depends on ship dimensions and wave length and α_m is the wave slope.

Controlling rolling motions is affected by constant parameters related to the ship structure and variable parameters, related to the external excitation in different loading conditions. After defining the mathematical model of the roll motion, the stabilizing moment should be evaluated for the design of the roll control system. For example, the roll motion of ships with a **passive tank** can be described using the following equation:

$$(I_{xx} + J_{xx})\ddot{\phi} + B_{xx}\dot{\phi} + C_1\phi + [a_{4\tau}\ddot{\tau} + c_{4\tau}\tau] = K_{ext} \quad (2\text{-}38)$$

where τ is the tank angle, defined in Figure 2-16 (b); $a_{4\tau}$ is the 4th moment due to unit angle acceleration and $c_{4\tau}$ is the roll moment applied by the tank due to unit roll displacement. The energy of water flow between side tanks is delivered from waves, while the flow rate is adjusted via a control system and valves.

In fin stabilization systems, the water flow around the fins generates lift L and drag D forces. Similarly to rudders, these forces are a function of the fin surface S, the angle of attack α and the flow speed u:

$$L = C_L(\alpha) \cdot S\frac{\rho u^2}{2} \quad (2\text{-}39)$$

$$D = C_D(\alpha) \cdot S\frac{\rho u^2}{2} \quad (2\text{-}40)$$

$$\alpha = \alpha_p + \operatorname{arctg}\frac{\dot{\phi}}{u} \pm \operatorname{arctg}\frac{v}{u} \quad (2\text{-}41)$$

where α_p is the fin inclination angle and v is the heaving velocity.

The controlling system of the fin angle diplayed in Figure 2-17 measures the roll velocity via a gyroscope sensor. Consequently, the controller analyzes the signal and sends the stabilizing command to the servo-equipment. The latter adjusts the fin angle α_p to generate the required lift forces that dampen roll motions.

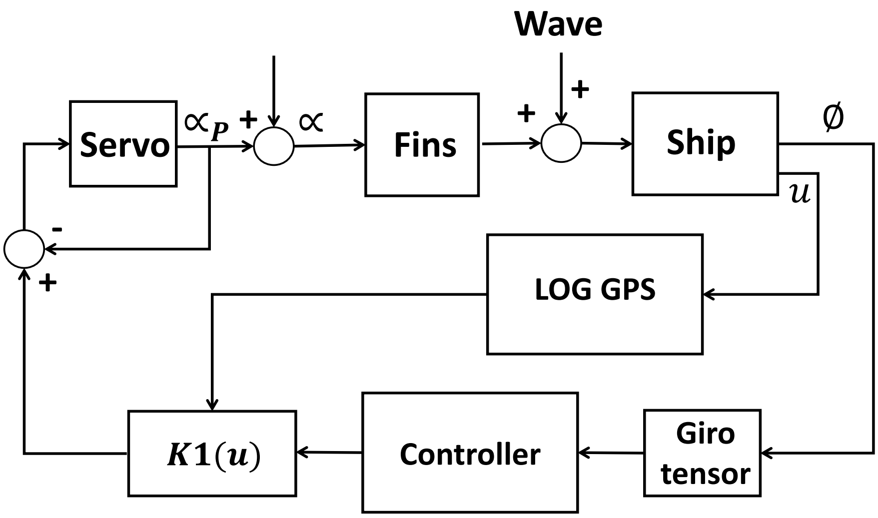


Figure 2-17 Schematic of the control system for fin stabilizers (Kula, 2015)

8. Autopilot steering

The autopilot is an automatic control system used for automatic navigation. The system can steer the vessel and keep it on the given course without any intervention from the captain, helmsman and the crew. The autopilot system has full control of the rudder and the steering devices based on predefined limits and coordination. Hence, it will keep the vessel on the correct heading when no one is present in the wheelhouse. This system exists in all seagoing and oceangoing commercial vessels. However, over-reliance on it without adequate comprehension of its limitations and efficiency may cause accidents. Accordingly the operator shuld be in position to understand how to set the optimal limits for :

- the rudder angle and rate of turn (values should not exceed the maximum rudder turning angle);
- the running steering gear pumps (hydraulic rudders should be turned on only in high traffic areas);
- the off course alarm onlyin those cases that the ship considerably deviates from her course;
- the manual or automatic modes depending on traffic density;
- speed (the autopilot should work efficiently in all speed ranges including high speed);
- weather conditions (autonomous and manual system function should be avaliable for use in severe sea states).

The system typically consists of the following devices :

- **A control head** thatdisplays status and heading information from the processor and allows the operator to input steering commands and operating parameters;
- **A compass** that reads the vessel's actual heading and sends it to the processor SPU;
- **Navigation devices** that communicate navigation information to the processor;
- **A Processor SPU** that calculates the rudder position to steer the vessel on the desired heading and controls the steering system accordingly;
- **A steering system (actuator)** comprising of a hydraulic ram or an electric motor which is mechanically connected to the rudder to move the rudder in response to the control signals from SPU ;
- **A rRudder Follower Unit (RFU)** that is mechanically connected to the rudder to measure the rudder position and send it back to the SPU

In a simple PD-controller based autopilot the control function of the rudder angle to maintain the ship's heading ψ_0 is :

$$\delta_T = C_1(\psi - \psi_0) + C_2\dot{\psi} \tag{2-42}$$

where C_1 and C_2 are the gain factors of the autopilot. The actual rudder angle can be evaluated by a simple linear differential equation

$$\dot{\delta} = \mathrm{sgn}\,(\delta_T - \delta)\omega_\delta \tag{2-43}$$

where ω_δ is the prescribed rate of turn of the rudder. In an efficient and adaptive autopilot operation, the fuel consumption can be decreased by keeping the angle of the rudder movement low and thus by allowing small deviations to the course-line, see Figure 2-18.

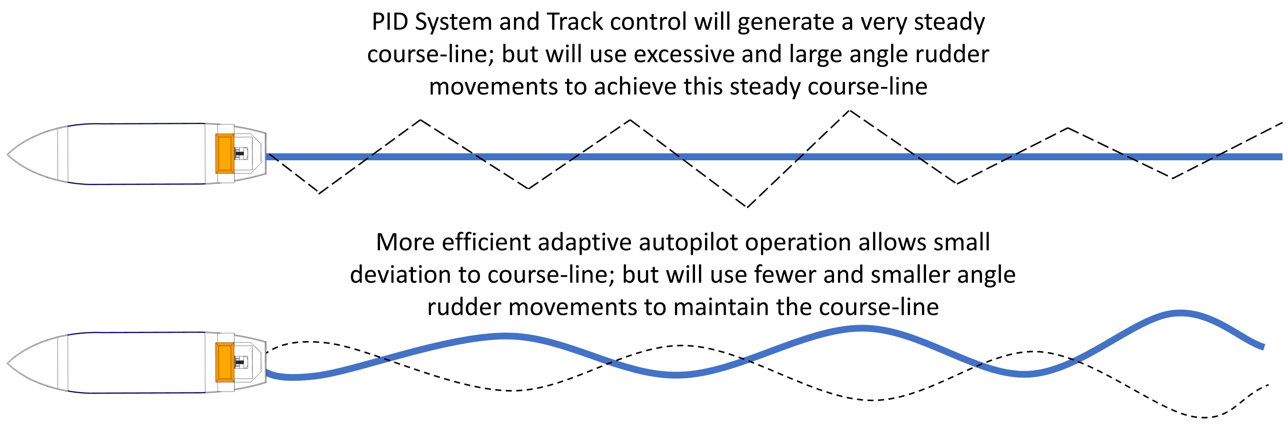


Figure 2-18 PID system and adaptive autopilot operation ©DNV GL

9. Questions

1. Name the components of the ship's total hull resistance in calm water. Which components dominate slow and high speeds?
2. Why does it take more power to achieve the same speed in shallow rather than in deep waters? What dangers are associated with operating with high speed in shallow waters?
3. Explain the different types of rudders and their pros and/or cons. Use sketches to elaborate your answers.
   Sketch the relationship of (a) the total resistance components in still water versus Froude number and (b) rudder lift coefficient vs rudder angle
4. For a rudder what we define as a stall angle and what is the range of this angle? How does it affect the maximum effectiveness of the rudder?
5. Name the two methods we can use to model the rudder action. Select one model and describe it briefly.
6. Which are the propulsors that can be used in different ship types ? Discuss which one is commonly used in passenger vessels.
7. Compare between constant and controllable pitch propeller characteristics and their adequacy.
8. Explain the in-plane force acting on the azimuth propeller in oblique flow. How does it affect the forces that drive the ship?
9. Discuss the functions, advantages and limitations of active and passive stabilization systems.
10. Discuss the influence of speed on ship maneuverability.

10. References

Bertram, V. (2012). *Practical ship hydrodynamics*, 2nd Edition, Elsevier, ISBN: 9781483299716.

DNV (2015). Rules on Hull Rudders and Steering, Part 3, Ch. 14.

Matusiak, J. (2021). The *dynamics of a Rigid Ship - with applications*. Aalto University publication series SCIENCE + TECHNOLOGY, 4/2021, ISBN : 978-952-64-0398-4 (printed).

Morrall, A. (1970). *The 1957 ITTC model ship correlation line values of frictional resistance coefficient*. National Physical Laboratory (NPL) Ship Division*, Ship Report 142*.

RINA (2010). *Proceedings of the Royal Institution of Naval Architects - William Froude Conference: Advances in Theoretical and Applied Hydrodynamics - Past and Future* 24 and 25 November 2010, Portsmouth, UK – ISBN : 978-1-905040-77-3.

Taimuri, G., Matusiak, J., Mikkola, T., Kujala, P., Hirdaris, S. (2020). A 6-DoF maneuvering model for the rapid estimation of hydrodynamic actions in deep and shallow waters, *Ocean Engineering*, 218:108103.

Tupper,E.C. (2013). *Introduction to Naval Architecture*, Butterworth-Heinemann, Oxford, ISBN-13: 978-0080982373.

Wätsilä (2020). E*ncyclopedia of Marine Technology* (Ed. Jan Babicz), ISBN : 978-952-93-5536-5.

Kula, K. S. (2015). An overview of roll stabilizers and systems for their control, *Trans Nav.International Journal on Marine Navigation and Safety of Sea Transportation* , 9.3.

