A simplified model for bottom trawl fishing gears

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10/05/2007

Computational modeling is a valuable complementary tool to assess behavior of bottom trawl fishing gears. A simplified model of the gear that mainly affects to the net is proposed. The model is constrained to steady towing conditions, flat seabed and gear symmetry. Simulations proportionate a number of relevant outcomes like distribution of tensions at the warp, balance of forces at the otterboards or spread under different haul conditions such as depth or towing speed. In this paper we mainly focuss on the description and implementation of the model. Nevertheless, some preliminar comparison with experimental data is also shown.

1 INTRODUCTION

Bottom trawl fishing gears are complex systems in which the different constitutive components (net, sweeps, otterboards and warps) are intimately coupled. Information on gear response has been traditionally inferred from empirical experience, *in situ* data acquisition (Henriques 1992; Sala 2006) and scaled prototypes in flume tank experiments (Fiorentini et al. 2004). In addition to empirical studies, a number of theoretical models of increasing complexity have also emerged in parallel with the development of computational capabilities. However, attempts to model simultaneously all the components of fishing gears are rare (Bessonneau and Marichal 1998). To date, most modeling efforts have been addressed towards the study of net geometries (Bessonneau and Marichal 1998; O'Neill 1999; Wan et al. 2002; Wan et al. 2002; Priour 2003; Suzuki et al. 2003; Shimizu et al. 2004) and net drag evaluations (Reid 1977; Galbraith 1983; Ferro et al. 1996; Hu et al. 2001), normally for the case of pelagic trawls.

In this paper we develop a simplified model for bottom trawl fishing gears which requires the total net drag and the net opening at the wing ends as input parameters. The model predicts the configuration of the gear solving for the equilibrium equations of the otterboards and the system of ordinary differential equations (ODE's) of the warp. Simulations proportionate also the distribution of tensions, otterboards spread and attack angle as well as the balance of forces at the otterboards under different haul conditions like fishing depth or towing velocity.

The manuscript is arranged as follows. Firstly in §2 we discuss the assumptions and limitations of the model. In §3 we derive the governing equations for each component of the gear. In §4 we describe the numerical implementation of the model.

2 MODEL HYPOTHESIS

The behavior of a real gear during a haul is likely to be affected by multiple time-dependent and often unpredictable factors such as, for instance, seabed irregularities, waves or water currents. In order to simplify the problem we constrain to an ideal scenario in which the following hypothesis apply: steady state (i.e. constant towing velocity and negligible effect of waves and currents on the gear), flat seabed and gear symmetry with respect to the vertical plane.

Some additional simplifications for the components of the gear are also contemplated. Rather than the full net geometry we consider only the horizontal net opening and we assume a known total net drag. The sweeps are assumed to behave as a rigid bar. Finally, we consider that the otterboards do neither pitch nor heel.

3 MODEL GOVERNING EQUATIONS

The gear is towed with speed **u** with components: net, sweeps, otterboards and warps. Taking into account the model hypothesis described in section §2 it is only necessary to derive the governing equations of the otterboard and the warp.

3.1 *Otterboard equations*

In this section we derive the governing equations for the otterboards imposing equilibrium of forces and moments. The function of the otterboards is to spread the gear horizontally and, simultaneously, to keep it in contact with the seabed. In general, an otterboard has six degrees of freedom which must be balanced under the steady towing assumption. In order to derive the equilibrium equations let us consider an ortonormal frame of reference $\widehat{\mathbf{x}}_{\mathbf{o}} = <\mathbf{x}_{\mathbf{o}}, \mathbf{y}_{\mathbf{o}}, \mathbf{z}_{\mathbf{o}}>$ attached to the otterboard and having the origin located at the center of pressure. This local basis is defined so that $\mathbf{x}_{\mathbf{o}}$ points backwards (opposite direction of towing), $\mathbf{y}_{\mathbf{o}}$ points outwards, and $\mathbf{z}_{\mathbf{o}}$ points downwards. The system $\widehat{\mathbf{x}}_{\mathbf{o}}$ and the cartesian frame of reference $\widehat{\mathbf{x}}$ are related by $\widehat{\mathbf{x}} = \mathfrak{D} \cdot \widehat{\mathbf{x}}_{\mathbf{o}}$ (or $\widehat{\mathbf{x}}_{\mathbf{o}} = \mathfrak{D}^{-1} \cdot \widehat{\mathbf{x}} = \mathfrak{D}^{T} \cdot \widehat{\mathbf{x}}$) where, in general, $\mathfrak{D}$ results from composing a rotation of heel angle φ along the $\mathbf{x}_{\mathbf{o}}$ axis, a rotation of pitch angle θ along the $\mathbf{y}_{\mathbf{o}}$ axis, and a rotation of yaw angle Ψ (angle of attack) along the $\mathbf{z}_{\mathbf{o}}$ axis. However, since we constrain to the case where the otterboard does neither heel ($\varphi = 0$) nor pitch ($\theta = 0$) the transformation matrix reduces to a single rotation:

$$\mathfrak{D} = \begin{pmatrix} \cos\Psi & -\sin\Psi & 0 \\ \sin\Psi & \cos\Psi & 0 \\ 0 & 0 & 1 \end{pmatrix} \qquad (1)$$

whre Ψ is the angle of attack.

We consider a trawl door with mass M_o, length L_o and height H_o towed at speed **u**.

3.1.1 *Balance of forces*

Forces acting on otterboards include weight, buoyancy, ground contact forces, hydrodynamic forces and tensions exerted by the warp and the backstrop at the attachment points. On a Cartesian frame of reference $\widehat{\mathbf{x}}$ oriented along the tow direction these forces write as follows:

(1) Mass forces due to weight and buoyancy ($\mathbf{F_M}$). Act along the vertical direction:

$$(\mathbf{F_M})_{\widehat{\mathbf{x}}} = \begin{pmatrix} 0 \\ 0 \\ M_o g - \rho_a V_o g \end{pmatrix} \qquad (2)$$

where V_o is the otterboard volume, ρ_a is the sea water density and g stands for the gravity acceleration.

(2) Ground contact forces ($\mathbf{F_G}$). We limite the model to otterboard that touch the seabed slightly.

Therefore, the interaction with the seabed results on a vertical component (the normal or ground reaction force which prevents the otterboard to penetrate the seabed) and on a friction force that acts on the opposite direction of tow and which, as a first approach, can be assumed to be proportional to the normal force:

$$(\mathbf{F_G})_{\widehat{\mathbf{x}}} = \begin{pmatrix} \mu N \\ 0 \\ -N \end{pmatrix} \qquad (3)$$

where μ is the friction coefficient and $N (\geq 0)$ is the normal force. Note that the minus sign has been introduced for coherence with the orientation of the z-axis.

(3) Hydrodynamic forces ($\mathbf{F_H}$). Relative movement among water and otterboards generates spreading (lift) and drag forces on the otterboards. It is well established that these forces depend on the towing speed squared, angle of attack, and otterboard design. In general, hydrodynamic forces are evaluated experimentally in terms of the drag and lift coefficients:

$$(\mathbf{F_H})_{\widehat{\mathbf{x}}} = \frac{1}{2}\rho_a S_o |\mathbf{u}|^2 \begin{pmatrix} C_D(\Psi) \\ C_L(\Psi) \\ 0 \end{pmatrix} \qquad (4)$$

where S_o is the otterboard projected surface, and C_D and C_L are the drag and lift coefficients respectively. According to our sign criteria $\Psi < 0$ for the port door (see Figure 1). The projection of the hydrodynamic forces onto the vertical direction vanishes because we have assumed a zero heel angle. Note also that the spreading and drag coefficients depend on the angle of attack of the otterboard. An efficient design aims to maximize the ratio C_L/C_D while keeping the otterboard stable.

(4) Warp tension at the otterboard attachment ($\mathbf{T_w}$). From Figure 1 it follows that:

$$(\mathbf{T_w})_{\widehat{\mathbf{x}}} = -T_w \begin{pmatrix} \cos\Psi_w \cos\theta_w \\ \sin\Psi_w \cos\theta_w \\ \sin\theta_w \end{pmatrix} \qquad (5)$$

where T_w, Ψ_w and θ_w stand, respectively, for tension, yaw and pitch angles of the warp at the otterboard bracket. For the port otterboard and according to our sign criteria $\Psi_w > 0$ and $\theta_w > 0$.

(5) Backstrop tension at the otterboard attachment ($\mathbf{T_b}$). Consider an otterboard rigged with a twin backstrop attachment in which the lengths of both chains are equal and have a pitch angle θ_b. From Figure 1:

$$(\mathbf{T_b})_{\widehat{\mathbf{x}}} = (\mathbf{T_b^u} + \mathbf{T_b^l})_{\widehat{\mathbf{x}}} \qquad (6)$$

2

and

$$
\begin{cases}
\mathbf{T_b^u} = T_b^u \begin{pmatrix} \cos\Psi_b\cos\theta_b \\ \sin\Psi_b\cos\theta_b \\ \sin\theta_b \end{pmatrix} \\
\mathbf{T_b^l} = T_b^l \begin{pmatrix} \cos\Psi_b\cos\theta_b \\ \sin\Psi_b\cos\theta_b \\ -\sin\theta_b \end{pmatrix}
\end{cases}
\tag{7}
$$

In order to close the balance of forces it is necessary to add an additional constrain for T_b^u and T_b^l at the backstrop attachment point:

$$
\begin{cases}
(T_b^u + T_b^l)\cos\theta_b = T_b\cos\epsilon \\
(T_b^u - T_b^l)\sin\theta_b = T_b\sin\epsilon
\end{cases}
\tag{8}
$$

with ϵ defined as in Fig.1.

[Figure 1 about here.]

The balance of forces at the otterboard is obtained by combining equations (2) to (8) and imposing equilibrium. It yields, for each spatial component:

$$
\begin{cases}
\mu N + F_{Hx}(\Psi) + T_b\cos\epsilon\cos\Psi_b = T_w\cos\Psi_w\cos\theta_w \\
F_{Hy}(\Psi) = T_w\sin\Psi_w\cos\theta_w - T_b\cos\epsilon\sin\Psi_b \\
Mg + T_b^u\sin\theta_b = \rho_a V g + N + T_b^l\sin\theta_b + T_w\sin\theta_w
\end{cases}
\tag{9}
$$

The above set of equations is written in a way that highlights the effect of each contribution. Terms in the LHS act in the positive directions whereas terms in the RHS act in the negatives. In the x-direction, friction, hydrodynamic drag and tensions at both backstrop attachments oppose to movement and are balanced just by the component of the warp tension along the tow direction ($x < 0$ direction). In the y-direction hydrodynamic lift tends to spread the otterboards ($y > 0$ direction) whereas warp and backstrop attachment components do the opposite (note that $\Psi_w > 0$ and $\Psi_b < 0$ so that both terms have a positive sign). Finally, in the vertical z-direction, the otterboard weight and the upper backstrop attachment rig pull downwards ($z > 0$ direction) whereas buoyancy, ground reaction, warp tension and lower backstrop attachment rig pull the otterboard upwards.

3.1.2 *Balance of moments*

In addition to the balance of forces, the equilibrium hypothesis requires also a zero net balance of pairs. The balance of moments for the x and z-directions is calculated with respect to the origin of reference of the system $\widehat{\mathbf{x}_o}$ (i.e. the center of pressure). It follows that the pair exerted by the hydrodynamic forces is zero because, by definition, the resultant force acts at the center of pressure. On the other hand, weight and buoyancy forces do not either produce any moment under the hypothesis zero heel. In consequence, we limit to the pairs exerted by the warp and backstrop attachments:

(1) Warp moment ($\mathbf{M_w}$). Let $(\mathbf{r_w})_{\widehat{\mathbf{x}_o}} = (x_w, y_w, z_w)^T$ be the position vector of the warp bracket in the $\widehat{\mathbf{x}_o}$ frame of reference. It follows from (1) that:

$$
(\mathbf{r_w})_{\widehat{\mathbf{x}}} = \mathfrak{D} \cdot (\mathbf{r_w})_{\widehat{\mathbf{x}_o}}
\tag{10}
$$

so that the moment exherted by the warp yields $\mathbf{M_w} = \mathfrak{D} \cdot (\mathbf{r_w})_{\widehat{\mathbf{x}_o}} \times \mathbf{T_w}$ with $\mathbf{T_w}$ given by (5).

(2) Backstrop moment ($\mathbf{M_b}$). Let $(\mathbf{r_b^u})_{\widehat{\mathbf{x}_o}} = (x_b^u, y_b^u, z_b^u)^T$ and $(\mathbf{r_b^l})_{\widehat{\mathbf{x}_o}} = (x_b^l, y_b^l, z_b^l)^T$ be the position vectors of the upper and lower backstrop attachments in the $\widehat{\mathbf{x}_o}$ frame of reference. Then:

$$
(\mathbf{r_b^u})_{\widehat{\mathbf{x}}} = \mathfrak{D} \cdot (\mathbf{r_b^u})_{\widehat{\mathbf{x}_o}}
\tag{11}
$$

and analogously for the lower component $(\mathbf{r_b^l})_{\widehat{\mathbf{x}}}$. The moment exheretd by the two backstrop attachments yields:

$$
\mathbf{M_b} = \mathbf{r_b^u} \times \mathbf{T_b^u} + \mathbf{r_b^l} \times \mathbf{T_b^l}
\tag{12}
$$

with $\mathbf{T_b^u}$ and $\mathbf{T_b^l}$ given by (7). Finally, imposing equilibrium one gets the x and z components:

$$
\begin{cases}
(\mathbf{M_w})_x + (\mathbf{M_b})_x = 0 \\
(\mathbf{M_w})_z + (\mathbf{M_b})_z = 0
\end{cases}
\tag{13}
$$

Given the values of tension and yaw angle at the backstrop attachment point (T_b, Ψ_b) and the pitch angle of the warp at the warp-otterboard bracket (θ_w), equations (9) and (13) together with the constrains (8) constitute a system of 7 non-linear equations for the 7 unknowns: otterboard angle of attack Ψ, ground reaction N, warp tension and yaw angle at the otterboard attachment (T_w and Ψ_w), tensions at the upper and lower chain backstrop attachments (T_b^u and T_b^l), and ϵ angle. In fact, the non-linear system can reduce to 5 equations with unknowns: Ψ, T_w, Ψ_w, T_b^u and ϵ.

3.2 *Warp equations*

Consider a warp of length L and diameter D towed from a vessel at speed $\mathbf{u}$. The warp has neither stretch nor torsion. The goal is to determine the tension T and the coordinates of each point of the warp on the Cartesian basis $\widehat{\mathbf{x}} = < \mathbf{x}, \mathbf{y}, \mathbf{z} >$. An ortonormal set of unit vectors $\widehat{\mathbf{x}_w} = < \mathbf{t}, \mathbf{n}, \mathbf{b} >$ is defined at each point of the warp with $\mathbf{t}$ tangential to the warp, $\mathbf{n}$ normal to the warp and laying in the vertical plane, and $\mathbf{b}$ orthogonal to the formers ($\mathbf{b} = \mathbf{t} \times \mathbf{n}$). The orientation of this local basis at a certain length or parameter arc $s \in [0, L]$ is given by the Euler angles $\Psi(s)$ and $\theta(s)$, that is, the system $\widehat{\mathbf{x}_w}$ and the cartesian frame of reference $\widehat{\mathbf{x}}$ are related by $\widehat{\mathbf{x}_w} = \mathfrak{R} \cdot \widehat{\mathbf{x}}$, where:

$$
\mathfrak{R} = \begin{pmatrix}
\cos\Psi\cos\theta & \sin\Psi\cos\theta & \sin\theta \\
-\cos\Psi\sin\theta & -\sin\Psi\sin\theta & \cos\theta \\
\sin\Psi & -\cos\Psi & 0
\end{pmatrix}
\tag{14}
$$

and $\mathfrak{R}^{-1} = \mathfrak{R}^{\mathbf{T}}$ ($\mathfrak{R}$ is an orthogonal matrix). In addition to tension, forces per unit of length acting on the warp include:

(1) Mass forces due to weight and buoyancy ($\mathbf{F_M}$):

$$(\mathbf{F_M})_{\widehat{\mathbf{x}}} = \begin{pmatrix} F_{Mx} \\ F_{My} \\ F_{Mz} \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ w \end{pmatrix} \qquad (15)$$

or, using (14):

$$(\mathbf{F_M})_{\widehat{\mathbf{x}_w}} = \begin{pmatrix} F_{Mt} \\ F_{Mn} \\ F_{Mb} \end{pmatrix} = w \begin{pmatrix} \sin\theta \\ \cos\theta \\ 0 \end{pmatrix} \qquad (16)$$

where ρ_w is the warp density and $w = (\rho_w - \rho_a)\pi D^2 g/4$.

(2) Hydrodynamic forces ($\mathbf{F_H}$) which, in general, can be inertial and non-inertial. Under steady towing conditions the inertial forces vanish and, in consequence, one has to consider only the non-inertial forces along the tangential (F_{Ht}) and normal directions (F_{Hn} and F_{Hb}). Hydrodynamic resistance of a warp is well described by the Morison's equation (Faltinsen 1990):

$$(\mathbf{F_H})_{\widehat{\mathbf{x}}} = \begin{pmatrix} F_{Ht} \\ F_{Hn} \\ F_{Hb} \end{pmatrix} = \frac{1}{2}\rho_a D \begin{pmatrix} C_t \pi |u_t| u_t \\ C_n \sqrt{u_n^2 + u_b^2}\, u_n \\ C_n \sqrt{u_n^2 + u_b^2}\, u_b \end{pmatrix} \qquad (17)$$

where $(u_t, u_n, u_b)^T$ are the components of the water speed (as seen by an observer attached to the warp), and C_t and C_n are the tangential and normal warp coefficients. It is obvious that the water speed has the opposite sense of the vessel towing speed (of the towing velocity $\mathbf{u}$), that is:

$$\begin{pmatrix} u_x \\ u_y \\ u_z \end{pmatrix} = |\mathbf{u}| \begin{pmatrix} 1 \\ 0 \\ 0 \end{pmatrix} \qquad (18)$$

and therefore

$$\begin{pmatrix} u_t \\ u_n \\ u_b \end{pmatrix} = |\mathbf{u}| \begin{pmatrix} \cos\Psi\cos\theta \\ -\cos\Psi\sin\theta \\ \sin\Psi \end{pmatrix} \qquad (19)$$

The governing equations for the warp result finally from imposing equilibrium of forces (Chin et al.

2000):

$$\begin{cases} \dfrac{dT}{ds} = -F_{Mt}(\theta) - F_{Ht}(\Psi,\theta) \\[4pt] \dfrac{d\theta}{ds} = -\dfrac{1}{T}(F_{Mn}(\theta) + F_{Hn}(\Psi,\theta)) \\[4pt] \dfrac{d\Psi}{ds} = \dfrac{1}{T\cos\theta}(F_{Mb}(\theta) + F_{Hb}(\Psi,\theta)) \\[4pt] \dfrac{dx}{ds} = \cos\Psi\cos\theta \\[4pt] \dfrac{dy}{ds} = \sin\Psi\cos\theta \\[4pt] \dfrac{dz}{ds} = \sin\theta \end{cases} \qquad (20)$$

The above constitutes a system of ODE's. Given the coordinates (x, y, z), the Euler angles (θ, Ψ), and the tension (T) of the warp at the bracket, the system is numerically integrated backwards along the parameter arc $s \in [0, L]$ from the door ($s = L$) to the vessel ($s = 0$).

4 MODEL IMPLEMENTATION

Model inputs are the total net drag (T_n) and the horizontal net opening (HNO) at the wing ends together with other general parameters of the gear and the haul. We present the model implementation step by step:

Step 1. Solve the equilibrium equations of the otterboards (9) and (13) given values for the yaw angle Ψ_b and tension T_b at the backstrop attachment point and for the pitch angle at the warp bracket θ_w. Note that Ψ_b is also the yaw angle of the sweep and that T_b is one half of the total net drag $T_n/2$ because a zero tension drop is assumed to occur along the sweeps. The outcomes are the attack angle Ψ and the tension T_w and the yaw angle ψ_w at the warp attachment point.

Step 2. Solve the warp ODE backwards in the interval $s \in [0, L]$, with initial values: $T(L) = T_w, \Psi(L) = \Psi_w, \theta(L) = \theta_w$. The final calculated values for (y, z) are written as $y_{ship} = y(0)$ and $z_{ship} = z(0)$.

Step 3. The global function $F(\Psi_b, \theta_w) = (y_{ship}, z_{ship})$ is defined following steps 1 and 2.

Step 4. Find (Ψ_b, θ_w) such that z_{ship} equals the fishing depth H and y_{ship} equals half of the warp separation at the vessel stern, Y (i.e. ensure that the vessel lays in the vertical plane of symmetry):

$$F(\Psi_b, \theta_w) = (Y, H) \qquad (21)$$

Numerical algorithms to solve steps 1 and 2 have been implemented using the open software libraries MINPACK and ODEPACK, respectively. The problem (21) is equivalent to find a zero of a system of nonlinear functions which is solved using MINPACK library, the jacobian is calculated by a forward-difference approximation. The initial condition is $\theta_w = \arcsin(H/L)$ (straight warp) and Ψ_b varying from $-\pi/4$ to 0 until convergence is achieved.

5 SUMMARY

We have developed a simplified model for bottom trawl fishing gears. The numerical implementation allows for an efficient and consistent coupling among gear components. A relevant feature of the model is that it skips a detailed simulation of the net and hence proportionates approximate results at negligible computational cost.

The model proportionates a detailed analysis of the otterboards including the horizontal opening, the angle of attack, the tensions at the backstrop and warp attachments, and the balance of forces and moments. In addition, the tensions and the resulting geometry of the warp are also calculated.

REFERENCES

Bessonneau, J. and D. Marichal (1998). Study of the dynamics of submerged supple nets (applications to trawls). *Ocean Engineering 7*, 563–583.

Chin, C., R. May, and H. Connell (2000). A numerical model of a towed cable-body system. *ANZIAM J. 42 (E)*, C362–C384.

Faltinsen, O. (1990). *Sea loads on ships and offshore structures*. Cambridge University Press.

Ferro, R., B. vanMarlen, and K. Hansen (1996). An empirical velocity scale relation for modelling a design of large mesh pelagic trawl. *Fisheries Research 28*, 197–230.

Fiorentini, L., A. Sala, K. Hansen, G. Cosimi, and V. Palumbo (2004). Comparison between model testing and full-scale trials of new trawl design for italian bottom fisheries. *Fisheries Science 70*, 349–359.

Galbraith, R. (1983). The marine laboratory four-panel trawl. *Scottish Fisheries Research Report 8*, 1–21.

Henriques, V. (1992). *Relatários Técnicos e Científicos. No. 47 Projecto e teste de aparelhos de arrasto*. INIP, Lisboa.

Hu, F., K. Matuda, and T. Tokai (2001). Effects of drag coefficient of netting for dynamic similarity on model testing of trawl nets. *Fisheries Science 67*, 84–89.

O'Neill, F. (1999). Axisymmetric trawl cod-ends made from netting of a generalized mesh shape. *IMA Journal of Applied Mathematics 62*, 245–262.

Priour, D. (2003). Analysis of nets with hexagonal mesh using triangular elements. *International Journal Numerical Methods in engineering 56*, 1721–1733.

Reid, A. (1977). A net drag formula for pelagic nets. *Scotish Fisheries 7*.

Sala, A. (2006). PREMECS-II, 2006. development of predictive model of cod-end selectivity. Final report to the european commission of the RTD project Q5RS-2002-01328. pp. 1–265.

Shimizu, T., T. Takagi, K. Suzuki, T. Hiraishi, and K. Yamamoto (2004). Refined calculation model for nala, a fishing shape simulator, applicable to gill nets. *Fisheries Science 70*, 401–411.

Suzuki, K., T. Takagi, T. Shimizu, T. Hiraishi, K. Yamamoto, and K. Nashimoto (2003). Validity and visualization of a numerical model used to determine dynamic configurations of fishing nets. *Fisheries Science 69*, 695–705.

Wan, R., F. Hu, and T. Tokai (2002). A static analysis of the tension and configuration of submerged plane nets. *Fisheries Science 68*, 815–823.

Wan, R., F. Hu, T. Tokai, and K. Matuda (2002). A method for analyzing the static response of submerged rope systems based on a finite element method. *Fisheries Science 68*, 65–70.
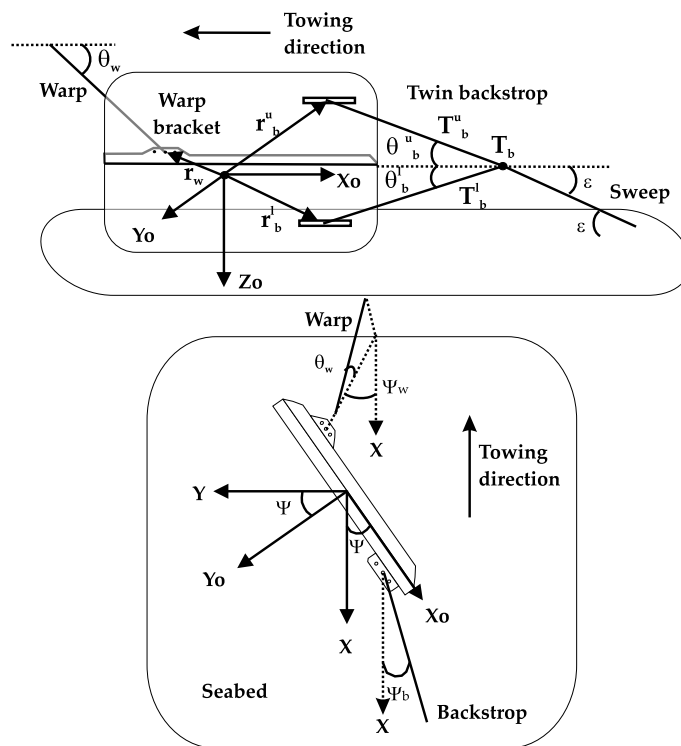
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Figure 1: Port otterboard with a twin backstrop adjustment. Top: Lateral view. Bottom: top view.