HYDRODYNAMIC MODELLING OF AN AUTONOMOUS SURFACE VESSEL FOR DYNAMIC CONTROLLING

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ABSTRACT

An Autonomous Surface Vessel (ASV) can be defined as a vehicle controlling its own steering and speed for navigation, dynamic positioning, motion stabilization and obstacle detection and avoidance. This novel vehicle mainly consists of two hulls (pontoons), a strut of hydrofoil cross section, a submerged body (Gertler body) connected at the bottom end of the strut and two propellers. This study, kinematics and a complete hydrodynamics analysis of a proposed model of an ASV which travels in the ocean is openly discussed using first principles. The vessel moves in a hydrodynamic environment where many uncertainties, non-linear and non-predictive behaviours always appear. The scope of this research paper is defined by two main objectives; developing complete mathematical model of a surface vessel by analyzing hydrodynamic forces and other main effects arising when manoeuvring in the ocean. It also addresses non-redundant system of speed controlling. Most of the related studies have not addressed angular speed and torque of propellers' actions which power the motion and control the vessel by contributing as force and moment inputs and assumed that the residual resistance is independent of the Reynolds number but which is not accurate and failed in many cases. Practically close model is crucial for researchers who develop controllers for surface vessel, numerical simulations on vessel dynamics, and for further tuning as prior testing of a physical model is quite expensive.

Keywords: *Autonomous surface vessel, Hydrodynamic modelling, Vessel controlling, Vessel dynamics, Vessel kinematics*

INTRODUCTION

Autonomous Surface Vessels (ASVs) are useful, especially for researchers and generally to mankind due to their versatile applications such as in Oceanographic survey (surface, under sea), as a communication interface between underwater vehicles & surface ships, for mine hunting, protection or surveillance, in anti-submarine warfare [1], pollution/chemical detection, bacteriological or radiation detection and so on. Research and development of classical controllers for any dynamical system are initially based on the availability of a mathematical model, which provides close and accurate description of dynamics of the real system. Many previous research publications are available on mathematical modelling, control, navigation, and communication systems for underwater vehicles [2]--[4], but only a limited research has been reported on hydrodynamic modelling of surface vessels, especially for autonomous applications [5], [6] with engine propulsions. A mathematical model for a hovercraft was developed in [7] to introduce stabilization algorithms and such systems are simple when compared to that of a standard surface vessel as it does not contain many of the nonlinear coupled drag terms. In [8] and [9] also, mathematical models were used to test and simulate control algorithms but these rather poorly represent the non-linear dynamics of the vehicle.

This paper shows how the vessel vessel's speeds control and derived with respect to the angular speed and torque of the propellers, how behavior of unsteady forces taking in to account when vessel is accelerating, and solve the issues with residual resistance that is not independent with the Reynolds number (*Rn)* that many researchers inadequately addressed in their publications. These aforementioned issues are very much important to be addressed to realize an accurate model for an ASV because the ocean environment where it operates is characterized by unknowns, undesirables, and perturbations. In this paper, these issues are studied and taken in to consideration when developing the mathematical model and hence computer simulations. Theories described in [10] was used as a main reference and further forces and moments calculated using practical data found in [6],[9],[10],[11] and references there in. Following assumptions are made in the mathematical modeling of the ASV:

1. Surface vessel is rigid.
2. Earth Fixed Frame [EFF] is inertial.
3. Vehicle velocities in vertical, pitch and roll directions are insignificance
4. Gravity and buoyancy forces are acting on *E[Z]* direction only and Body Fixed Frame (BFF), *B[Z]* is always parallel to the *E[Z]* **.**
5. Buoyancy forces are compensated by gravitational forces and then vector of gravitational forces are neglected.
6. Hulls are approximated as flat plates.
7. Fluid interference between the port and starboard hulls is negligible.
8. Thrusts delivered by two propellers are same in magnitudes.

MAIN COMPONENTS & MODEL SPECIFICATION

A Complex 3-DOF dynamic model is derived for the CAD model of the surface vessel shown in Fig.1. Important parameters of the vessel are depicted in Table I. The design of an ASV depends on many factors and conditions. But here, geometry of the vehicle is important only as mathematical relations are obtained using general notations rather than numerical values. Main vessel components are Hulls; two hulls (pontoons) with same shape and dimensions are mounted symmetrically on either side of the body.

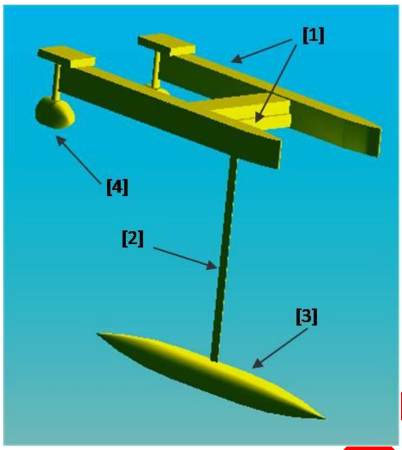

Figure 1. 3D view of preliminary design of the ASV where [1] hulls, [2] vertical strut, [3] Gertler body, and [4] propellers

A. Hulls of the Vessel

Two hulls (pontoons) with same shape and dimensions are mounted symmetrically on either side of the body. Since beam to length ratio (B_h/L_h) is small, hulls are approximated to flat plates for damping calculations.

B. Strut

Strut is a thin column with symmetric airfoil cross section [12] extended from bottom of the body to inside of the sea surface.

C. Gertler Body

Gertler body [13] is a submerged pontoon which is fixed to the bottom end of the strut to stabilize the vessel when travelling on the surface.

D. Propellers

Two propellers are mounted at the back end of the hulls by vertical supports and the thrust inputs are generated by the propulsive action of the power drives. Please refer [11] for different types of propellers and related issues.

Table I. Specifications of the ASV Components

Component	*Parameter*	*Notation*	*Specification*
Hulls	Length	L_h	2.540
	Beam	B_h	0.152
	Draft	T_h	0.127
	Planform area	A_b	0.774
Strut	Length	L_s	1.580
	Beam	b_s	0.025

	Chord	c_s	0.075
	Projected area (- direction)	S_x	0.035
	Projected area (- direction)	S_y	0.119
Submerged body	Length	L_g	1.870
	Greatest diameter	d_g	0.443
Propellers (screw type)	Blade diameter	D_p	0.100
	No of blades	N_p	3
Surface Vessel	Total mass	m	505.120 kg
	Yaw inertia	I_Z	559.225

COORDINATE FRAMES AND KINEMATIC ANALYSIS

The nomenclature defined in [14] is used throughout this paper. Fig. 2 shows two reference frames. The Earth Fixed Frame (EFF) is chosen so that the Centre of Gravity (CG) of the vessel is at the origin at time, . The X_E and Y_E axes are directed toward the North and the East, respectively, while the Z_E axis points downward. The Body Fixed Frame (BFF) is chosen such that its origin is fixed at the CG of the vessel. Three degrees-of-freedom (d.o.f.) system is considered by neglecting vertical (heave), pitch, and roll motions [5], [6], [7], [15]. It is assumed that this popular assumption is valid without appreciable loss in accuracy under typical and slightly severe maneuvers. The configuration vector of the ASV that is the vector of BFF w.r.t. EFF is given as

(1)

where $x = x(t)$, $y = y(t)$ represent the linear displacement and $\varepsilon = \varepsilon(t)$ is the rotation about the vertical E_Z -axis. By defining velocities of surge (X_B-direction), sway (Y_B-direction), and yaw (rotation about Z_B-direction) as $u = u(t)$, $v = v(t)$, and $r = r(t)$, then the velocity vector of the ASV can be written as

(2)

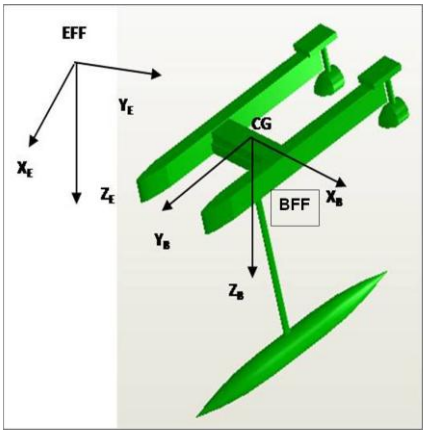

Figure 2. Coordinate frames defined for kinematic analysis

When the angle of trim (θ) and angle of roll (γ) are negligible the rotation matrix from the BFF to the EFF for a 3 d.o.f. system can be derived to be

$$J(\eta) = \begin{bmatrix} \cos\psi & -\sin\psi & 0 \\ \sin\psi & \cos\psi & 0 \\ 0 & 0 & 1 \end{bmatrix}. \quad (3)$$

Then, the relation between the $\eta(t)$ and $V(t)$ can be expressed as

$$\frac{d(\eta(t))}{dt} = \dot{\eta}(t) = J(\eta)V(t). \quad (4)$$

Newton-Euler equations governing the motion of the ASV under the influence of external forces and torques are written referring to Fig. 2. The resulting general equations of motion of the ASV along the directions of BFF are

$$\begin{aligned} &\text{Surge:}\ \ m(\dot{u} - vr + y_G \dot{r} - x_G r^2) = X \\ &\text{Sway:}\ \ m(\dot{v} - ur + x_G \dot{r} - y_G r^2) = Y \\ &\text{Yaw:}\ \ I_Z \dot{r} + m[x_G(\dot{v} + ur) - y_G(\dot{u} - vr)] = N \end{aligned} \quad (5)$$

where X , Y , and N are the external forces and moment acting on the vehicle. x_G and y_G are the distances to the CG of the ASV from the origin of the BFF. Here, by placing origin of the BFF at the CG, x_G, $y_G \to 0$, and the above equations are simplified. The above equations of motion, (5), are rewritten as

$$M\dot{V}(t) + [C(v) + D(v)]V(t) + g[\eta(t)] = T_R(t); \quad (6)$$
$$t \geq 0$$

where M is the positive definite mass-inertia matrix, $C[v(t)] \in \Re^{3\times3}$ is the total matrix of *Coriolis* and centripetal terms, $D[v(t)] \in \Re^3$ is the damping vector, $g[\eta(t)] \in \Re^3$ is the vector of gravitational forces and moments, and $T_R(t) = [X(t), Y(t), N(t)]^T \in \Re^3$ is the input vector that represents the external forces and moments that will be described shortly.

Further, M and $C[v(t)]$ consist of sum of two components representing rigid body (RB) and added mass (AM) matrices, M_{AM} and C_{AM} that are formed due to the forces of the surrounding fluid are described later section of this paper. The gravitational forces and moments include the forces and moments due to the weight (W) of the vessel and the buoyancy (b) force;

$$g[\eta(t)] = g_W(t) + g_b(t) \tag{7}$$

where $g_W = mg$ and $g_b = -\rho(volume)g$. As vertical or the heave motion of the SV is approximated to zero, term $g[\eta(t)]$ is diminishing.

HYDRODYNAMICS MODELLING

In this section, all hydrodynamics damping forces applied on the vessel, added mass coefficient matrices and propellers' outputs/thrusts (forces and moments) will be explained. As the ASV dynamics depend on external hydrodynamic damping forces, it becomes essential to analyze all resistant forces; drag, viscous friction, residual or wave friction, and other ways of friction forces. Afterwards, a complete damping matrix will be derived and hence the mathematical model. Fluid properties; density (as 1025 and viscosity(υ) as $1.4\times10^{-6}\ m^s/s$ at temperature 10^0C and 3.3% salinity [10] are assumed. The *planform* area depends heavily on fluid density. The fluid properties are particularly required in friction calculations described later on this section.

A. Analysis of Hydrodynamics Damping Forces and Moments

Dimensional analysis is particularly useful in studying the forces exerted by the fluid on a moving body. Results of model tests, close approximation to simple geometric shapes or objects (eg: surface hulls as flat plates as beam to length ratio is small) and empirical equations proved by practical data contribute mainly for formulating forces and moments. Velocity of any point of the vehicle can be represented using linear and angular velocity couplings derived from velocities at CG and distances, d_x, d_y (in X_B and Y_B-directions), from the CG to the point considered. To this end, the velocity of any point can be written as

$$\begin{aligned} u_{pt} &= u(t) - d_y r(t) \\ v_{pt} &= v(t) + d_x r(t) \\ r_{pt} &= r(t). \end{aligned} \tag{8}$$

Also, total resistance on the ASV can be split up into [11]; frictional resistance or viscous friction (R_{fric}), form resistance (R_{form}) and wave (residual) resistance (R_{res}) where later two resistances constitute *Froude's* residual resistance. The total resistant coefficient can then be written as

$$C_t = (1+k).C_f + C_w + C_a \tag{9}$$

where C_f - frictional resistance coefficient, C_w - wave resistance coefficient, C_a - additional resistance coefficient and $(1+k)$ - form factor where k is a constant. $(1+k)$ is known as form factor which varies for different block coefficients (C_B) defined as the ratio between the ship's displacement volume and that of a rectangular box in which the ship's underwater volume "just fit".

From *Froude's* approach [11];

$$C_f = \frac{R_{fric}}{\frac{1}{2}\rho V^2 S} \tag{10}$$

where ρ – density of the sea water in kg/m^3, V – model speed and S – wetted surface area of the object/hull in m^2. *Renold* number, R_n, defined by $R_n = \frac{Ul}{\upsilon}$ is used to estimate resistance coefficients [9] with correct substitution for characteristics velocity, U, which is defined same as u_{pt} in (11) and characteristics length, l. Relationships given in (13) and (14) are applied for our model to calculate C_f and C_w [11] respectively

$$C_f = \frac{0.075}{[\log(R_n) - 2]^2}. \tag{11}$$

$$C_w = C_t - (1+k)C_f \tag{12}$$

C_a is estimated based on using water plane length as in [11].

Then, the total resistance force (in magnitude and direction) can be generalized as

$$F_t = \frac{1}{2}\rho AU|U|C_t \tag{13}$$

where A is projected/effective area and U is the velocity at CG of the particular object or element calculated same as u_{pt} in (8).

B. *Damping Forces on Hulls*

Resultant force in X_B -direction is contributed by viscous/skin drag (VD) and residual resistance /wave drag (WD) for both hulls (port side hull and starboard side hull) are

$$F_{h,X_B}^{VD}(u,r) = C_f \rho A_h U|U| \tag{14}$$

$$F_{h,X_B}^{WD}(u,r) = C_W \rho A_h U|U| \tag{15}$$

where $U = u - d_{h,y}r$ and $d_{h,y}$ is the distance from the CG to the port side hull (same for starboard side hull as well) in Y_B-direction and, A_h is the wetted *planform* area of the either hull [10].

Damping force in Y_B-direction can be calculated by integrating elemental damping forces for both hulls as

$$F^D_{h,Y_B}(v,r) = \int_{aft}^{forward} dFy = \int_{-dax}^{dfx} \frac{1}{2} C_{Dh} T_h \rho v(x)|v(x)| dx \qquad (16)$$

where

$$v(x) = v + \frac{v_{bow} - v_{stern}}{L_h}$$

with v_{bow} and v_{stern} being the velocities at bow (front end) and stern (back end) of the hulls, and $C_{Dh} = 2$ for cross flow over a flat plate [5]. Total damping moment in ψ_B direction is calculated by summing the total moments generated by the forces in X_B and Y_B- directions as

$$m_h(u,v,r) = m_{h,x}(u,r) + m_{hp,y}(v,r) + m_{hs,y}(v,r) \qquad (17)$$

where $m_{h,x}(u,r)$ the moments exerted by the forces in the X_B-direction and other terms are defined as

$$m_{hp,y}(v,r) = m_{hs,y}(v,r) = \int_{aft}^{forward} x.dF_y \ .$$

C. *Strut Damping Forces*

Damping forces on the strut consist of both viscous drag and wave drag and the prior one is derived based on lift (*L*) and drag (*D*) forces exerted on the hydrofoil section. In this work, we adopt the hydrofoil section of the type NACA 0012 described in [12]. The angle of attack, α, defined as the angle between “nose-tail line” of the hydrofoil and free-stream direction is used to find the drag and lift coefficients (C_D and C_L) [6].

To that end,

$$L = -\frac{1}{2}\rho U|U| S_y C_L \qquad (18)$$

$$D = \frac{1}{2}\rho U|U| S_x C_D \qquad (19)$$

where S_x and S_y are projected area of the strut in X_B and Y_B-directions respectively. Therefore, the total force in the X_B-direction is;

$$F^{VD}_{S,X_B}(u,r) = L\sin\alpha + D\cos\alpha \qquad (20)$$

$$F^{WD}_{S,X_B}(u,r) = \frac{1}{2}\rho V_{S,X_B}|V_{S,X_B}| A_x C_{rx} \qquad (21)$$

and that in the Y_B-direction is

$$F^{VD}_{S,Y_B}(v,r) = L\cos\alpha - D\sin\alpha \tag{22}$$

$$F^{WD}_{S,Y_B}(u,r) = \frac{1}{2}\rho V_{S,Y\,B}\left|V_{S,Y_B}\right| A_y C_{ry} \tag{23}$$

where A_x and A_y are projected areas, V_{S,X_B} and V_{S,Y_B} are the velocities of the strut of the format as defined in (8). C_{rx} and C_{ry} are residual friction coefficients**,** respectively, in X_B and Y_B -directions, estimated based on Froude number, $F_n = U/\sqrt{gl}$, and empirical data in [9].

Total damping moment, $m_S(u,v,r)$, for the strut can then be derived by multiplying total forces by respective distances.

D. Gertler Pontoon Damping Forces

Gertler body force can be calculated as normal damping force in (13) using resistant coefficient (C_g) values interpolated for different R_ns as follows;

$$F^{VD}_{G,X_B}(u) = \frac{1}{2}\rho u|u| A_{g,x} C_g \tag{24}$$

Total drag force, F^{VD}_{G,Y_B} in Y_B-direction is calculated by summing up elemental drag forces (F^{VD}_{G,Y_Bi}) as in (25). Pontoon is divided into 10 (i=1, …, 10) elements in Y_B-direction as cross section is different in each.

$$F^{VD}_{G,Y_Bi} = \frac{1}{2}\rho A_{yi} v_{gi}\left|v_{gi}\right| C_{gi} \tag{25}$$

then

$$F^{VD}_{G,Y_B} = \sum_{i=1}^{10} F^{VD}_{G,Y_Bi} \tag{26}$$

Damping moment in ψ_B-direction;

$$m_G(u,v,r) = \sum F^{VD}_{G,Y_Bi}(v,r) d_{gi} \tag{27}$$

where $v_{gi} = v + d_{gi} r$, and d_{gi} is the distance from the CG of the pontoon to element CG in Y_B-direction. Moment derived from the force in (24) is cancelled as the vessel CG vertical plane is aligned with submerged body mid-vertical plane.

E. Thrust and Propulsion

A propulsion system is needed to overcome the resistance and drive the vessel in desired trajectories by desired velocities and accelerations. Propulsion is an energy transfer generating when a propulsor delivers thrust and translation for torque inputs. Fixed pitch open screw propellers, controllable pitch propellers, ducted propellers, thrusters, *cycloidal* or *voiht-schneider* propellers, water jets are names for such propulsors mainly used in present systems. Different propulsion configurations, specific applications and advantages can be found in [11]. The efficiency of propulsors varies widely depending on propulsion method used (type) and among above types, screw propellers has not yet been equaled in most cases for other types (greater efficiency) and is therefore the most commonly used type in ships. Screw type propellers consist of several hydrofoil-type lifting surfaces, arranged in a helicoidally fashion to produce thrust when rotated about the axis [2]. Two screw propellers selected for modelling are shown in Fig. 3. In the sequel, we use the following notations; T – Axial thrust force, Q – Shaft torque, U_a – Axial velocity, n - Angular velocity of blades, d - Diameter of the propeller. As in (28) and (29), thrust, T, and torque, Q, can be derived using the thrust coefficient, K_T, and torque coefficient, K_Q, and their values are selected based on advance ratio referring [10] and [11].

$$T = K_T \rho d^4 n^2 \tag{28}$$

$$Q = K_Q \rho d^5 n^2 . \tag{29}$$

Resultant thrust vector (input force and moment) is given by;

$$T_R = \begin{bmatrix} X & Y & N \end{bmatrix} . \tag{37}$$

When T_P and T_S are thrusts delivered by port side and starboard side propellers, respectively, α_P and α_S are inclinations to the X_B-axis, and d_{P,X_B} and d_{P,Y_B} respectively are distances in X_B and Y_B-directions.

$$X = T_P \cos\alpha_P + T_S \cos\alpha_S \tag{38}$$

$$Y = T_P \sin\alpha_P - T_S \sin\alpha_S \tag{39}$$

$$N = X \times d_{p,Y_B} + Y \times d_{P,X_B} . \tag{40}$$

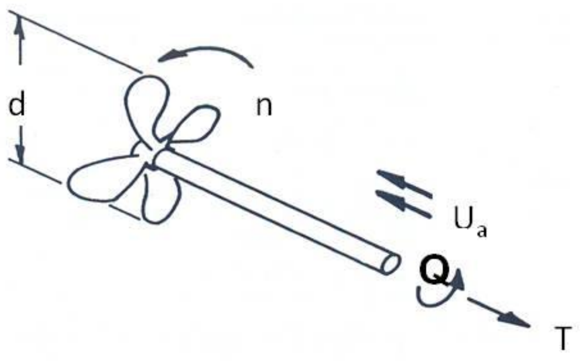

Figure 3. Seleted screw propellor and repersentation of forces and moments

F. Propeller-Hull Interaction

The propellers operate in the wake of the hull and the velocity is generally retarded to a degree depending on the fullness of the hull and the position of the propeller. One of major interaction of propeller results when the suction effect of the propeller modifies the flow past the hull and the resulting drag on the hull.

The forward velocity of hull i.e. the effective velocity of advance experienced by the propeller in the presence of the hull through the undisturbed water is given by

$$U_A = (1-w)U = ndJ \tag{41}$$

Wake fraction, w, was taken as 0.1 $(0 < w < 0.4)$. Since the propellers are mounted at the back-end of the hull, the pressure at the stern is reduced and hence drag force (D_P) is increased. Thrust is related to D_P by thrust- deduction coefficient $(t_P, < 0.2)$ as in (42).

$$D_P = (1-t_P)T \tag{42}$$

In this work, t_P is chosen as -0.315 for 1.5 hull efficiency [10].

G. Unsteady Forces

All friction forces are calculated in above sections by assuming steady-state sea environment. But this is not satisfied in most of the time as acceleration of the vessel can be changed. Therefore, viscous friction coefficient should be altered according to those variations. The non-dimensional term, Ut/l, i.e., the number of body lengths traveled in a time, *t*, defined in [10] is used to differentiate this effect as in (43) and (44).

$$C_F(R_n, Ut/l) \approx C_D(R) \quad ; Ut/l \leq 1 \tag{43}$$

$$C_F(R_n, Ut/l) \approx C_D(\alpha, Ut/l) \; ; Ut/l > 1. \tag{44}$$

Therefore, this effect that is often ignored in literatures for example as described in [6], [7], and [15], are considered for our hydrodynamic model.

H. Added Mass Coefficients

Values of added mass coefficients represent the amount of fluid accelerated with the body in physical standpoint. However, the added-mass coefficients for translation generally differ depending on the direction of the body motion, as opposed to *Newton's* equation of motion. Referring [10], [16]--[18] and 3D model developed using open source FreeCAD parametric modelling software [19] (Fig.1) are employed to calculate added mass coefficients as in (46) through (48) and then added masses of *Coriolis* and centripetal matrices.

Added mass matrix of the hulls ($M_{A,H}$), strut ($M_{A,S}$) and the Gertler body ($M_{A,G}$) are as follows;

$$M_{A,H} = \begin{bmatrix} -0.0007 & 0 & 0 \\ 0 & 7.5673 & 0.5060 \\ -0.0003 & -2.7924 & 2.6779 \end{bmatrix} \quad M_{A,S} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 2768 & 106.4 \\ 0 & 1835.1 & 631 \end{bmatrix}$$

$$M_{A,G} = \begin{bmatrix} 13.8159 & 0 & 0 \\ 0 & 149.53 & 7.8324 \\ 0 & 45.2148 & 18.7314 \end{bmatrix}$$

Following that total added mass matrices are

$$M_{AM} = M_{A,H} + M_{A,S} + M_{A,G} \tag{45}$$

$$C_{AM} = C_{A,H} + C_{A,S} + C_{A,G}\,. \tag{46}$$

I. State-space Representation

Here, we develop the state-space representation of the plant that is important in state-space methods of controller design and analysis. Equation (6) can be rewritten as

$$\dot{V}(t) = -M^{-1}[C(v) + D(v)]V(t) + M^{-1}T_R\,. \tag{47}$$

Now, we can define the state vector as

$$X = \begin{bmatrix} \eta^T & V^T \end{bmatrix}^T. \tag{48}$$

From (4) and (51), the state space model of the ASV can be obtained as

$$\dot{X} = f(X) + BT_R \tag{49}$$

where $B = \begin{bmatrix} 0 \\ M^{-1} \end{bmatrix}$ and $f(X) = \begin{bmatrix} J(\eta)\,V(t) \\ -M^{-1}[C(v)+D(v)]V(t) \end{bmatrix}$

CONTROLLER DESIGN

To demonstrate practical use of the mathematical model, a CTC: computed torque-like controller [21] for tracking of predefined trajectories relative to EFF coordinate system is designed. This illustrates the accuracy and efficiency of the models derived above.

The CTC control law is

$$T_R(t) = M\hat{\dot{V}}(t) + [C(V)+D(V)]V(t) \tag{50}$$

$$\hat{\dot{V}}(t) = [J(\eta)]^{-1}\{\ddot{\eta}_d(t) + K_d\dot{e}(t) + K_p e(t) - \dot{J}(\eta)V(t)\} \tag{51}$$

where $\eta_d(t)$ represents the desired trajectory and $e(t) = \eta_d(t) - \eta(t)$ the tracking error. The controller gains, K_d and K_p, are diagonal positive definite matrices. The system (6) with the controller (51) result in the error dynamics as

(52)

Recalling the fact that the gravity term, $g[\eta(t)]$, is ignored, exponential stability of $e(t)$ can be guaranteed by suitably selecting control parameters; K_d and K_p related to three directions; surge, sway and yaw.

NUMERICAL SIMULATIONS

For numerical simulation, a circular path tracking and maneuver was tested with the use of CTC controller. Results include the desired and actual trajectories and orientation so that the behavior of the vessel is completely shown. For simulation purposes the dynamics explained by (49) is used with the controller inputs given by (50) for desired trajectory η_d described as

$$\eta_d = [x_d \quad y_d \quad \psi_d]^T \tag{53}$$

where

$$x_d = A\sin(\omega t + \phi)$$
$$y_d = A\cos(\omega t + \phi)$$
$$\psi_d = -(\omega t + \phi)$$

and the phase shift, $\phi = 45^0$ is selected for this simulation. The radius of the trajectory is $A = 10m$ and angular velocity $\omega = 1.rpm\ (= \pi / 30\ rad / s)$ rad/s. The behavior of the vehicle is simulated for 60 s, or one complete cycle. Controller gains are chosen as

$$K_d = diag\,(10,10,10) \text{ and } K_p = diag\,(10,10,10).$$

The desired path and corresponding actual path, and surge, sway and yaw rates are shown in Fig. 4. The vehicle converges quickly to the desired trajectory and follows the same for rest of the time. Surge velocity rises up to $6m/s$ and achieves constant velocity about $1\ m/s$ within $4s$ time. Sway velocity settles down to zero velocity and yaw rate remains $-0.1\ rad/s$.

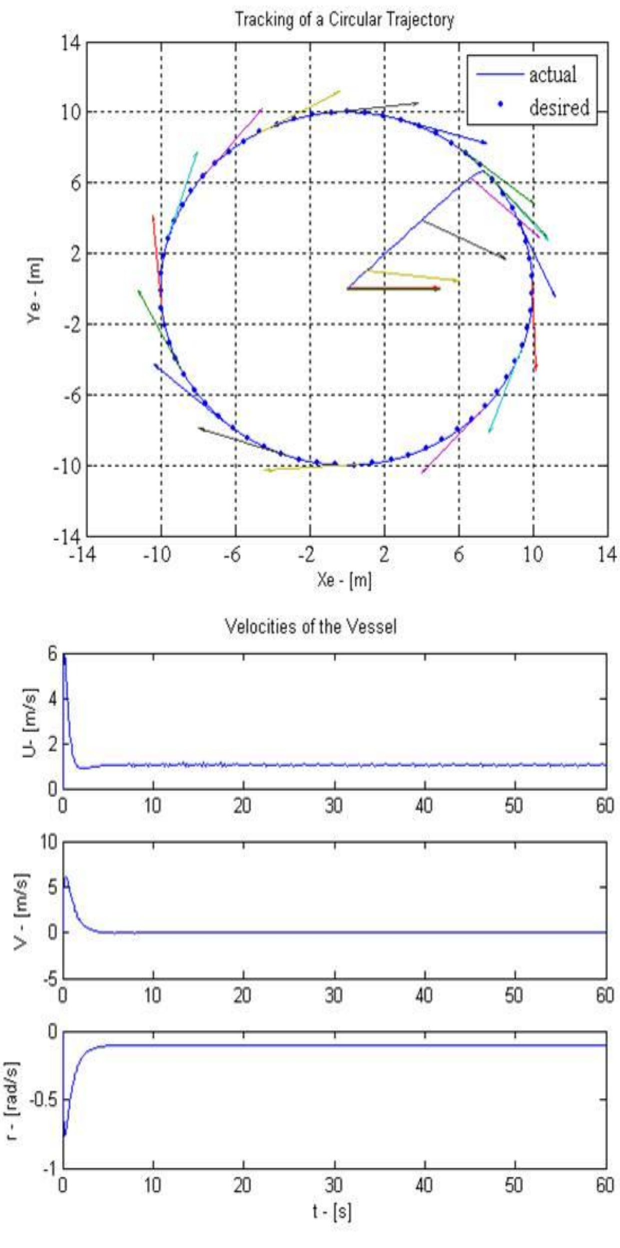

Figure 4. Tracking of circular trajectory (arrows show orientation of the ASV)

DISCUSSION

The dynamical behaviours of a surface vessel can only be partially depicted using current modelling techniques in the literatures. Further hydrodynamics of SVs are complicated when the vessel's motion is in transient. When considering a tracking problem where the vessel may be required to follow search patterns or its desired path is constantly being modified with the presence of large, unknown perturbations in ocean environment. For high-performance control of a mechanical system in terms of accuracy, stability, and robustness, it is of paramount importance to consider its mechanical structures, torques and forces acting upon it. Therefore, it is necessary to have accurate dynamic models of the controlled system, and controllers should be synthesized on the basis of modelled dynamics. On the other hand, the dynamics of surface vessels are apparently of highly complex nature and are difficult to be modelled accurately. Hence, the controllers should also be able to online adapt to varying conditions and environments, and to efficiently compensate for modelled and unmodelled uncertainties, strong nonlinearities and coupling effects. All *Coriolis* and centripetal terms, damping forces acting on various components of vessel, their interaction as a whole body and added masses were analyzed through this work by filling the current gap of the field.

It is important to note that in practice there are only two control inputs (thrust and angle of propellers) control three d.o.f. (swage, sway, and yaw) when controlling two propellers simultaneously. Because of the lack of a third independent control input, the yaw motion control is generally ignored in many research. The ignored yaw motion works as the internal dynamics whose stability is not proved. Importance of yaw motion control is particularly high in automated docking of surface vehicles. CTC is based on the requirement that exact knowledge of the vehicle dynamics is available. Obviously, this is a rather strong requirement that generally difficult to achieved in practice. These issues arise the need of a control system that is capable of self-tuning or self-adapting.

CONCLUSION

In this paper, a complete analysis of dynamics of an autonomous surface vessel is presented. An accurate and reliable mathematical model is developed by analyzing kinematics, vehicle body properties, damping forces and marine-hydrodynamics behavior of the different components of the surface vessel. Screw type propellers are proposed to power the vessel due to their high efficiency and availability. A computed toque-like controller is designed and the model is simulated numerically to track a circular trajectory. Results show the linear stabilization of velocities and nearly close follow of the actual trajectory with constant yaw rate though it moves in sway direction from center to perimeter initially. Friction forces induced on the surface vessel in the sea is unpredictable and non-linear in nature. To track a given path or destinations are highly surface vessel should equipped with self-adapting controller. Authors are intending to use this model for their future researches on development of a fully autonomous controller for path tracking and stable speed maneuvering prior to develop physical model of the surface vessel designed and described through this work.

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