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Semi-Empirical Investigation of the Hydrodynamic drag force over a UUV hull form

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Abstract: This paper investigates the validation of an empirical method that estimates the hydrodynamic coefficients of streamlined underwater vehicles. These coefficients are non-dimensional parameter appearing in the equation of motion of the vehicle. Datcom (U.S, Air Force compilation code) is used to determine these coefficients in aquatic medium. The body configuration in this study includes axisymmetric body without wing tail or body tail. Results obtained with the Datcom are presented here and compared to experimental results taken from a Planar Platform Mechanism mounted over a towing tank. A 6dof balance was used to extract the measurements forces and moments acting on the derived vehicle. These measurements were carried out at typical speeds of autonomous underwater vehicles (0.2-0.6 m / s) by varying the angles of inclination (0-15 degrees). This study was limited to the axial forces which mean the coefficient of drag. Static hydrodynamic characteristics computed by Datcom method are shown to agree closely with experimental results for torpedo slender body of circular cross sectional.

Key words: Underwater Vehicle, Hydrodynamic coefficients, Towing Tank, Datcom.

1. Introduction

In recent years, the focus on unmanned underwater vehicles (UUV's) has increased. A variety of missions includes search and survey, decoy and outboard sensors, Ocean engineering work service, Swimmer Support and Test and Evaluations [1]. As the cost of manned submarines vehicles increases, there are significant advantages to the use of cheaper unmanned vehicles.

However, underwater vehicle dynamics is strongly coupled and highly nonlinear due to added hydrodynamic mass, lift and drag forces acting on the vehicle. Engineering problems associated with the high density, non-uniform and unstructured seawater environment, and the nonlinear response of vehicles make a high degree of autonomy difficult to achieve. Hence six degree of freedom vehicle modeling and simulation are quite important and useful in the development of undersea vehicle control systems [1-3]. The mathematical models of marine vehicles consist of kinematic and dynamic part, where the kinematic model gives the relationship between speeds in a body-fixed frame and derivatives of positions and angles in an Earth-fixed frame. At the present time, there are a number of methods used in the prediction of hydrodynamic efforts. The minimum request in terms of computational efforts comes from the analytical and semi empirical (ASE) approach. This method is used in preliminary design of marine vehicles and employed also for estimating AUV hydrodynamic derivatives [4].The application of analytical and semi-empirical methods to estimate hydrodynamic parameters of the AUV based on vehicle geometry has been described in literature. USAF stability and control datcom presents methods for estimating aerodynamic forces (lift, drag, normal and axial), pitching moment and stability derivatives for various shapes and small and large angle of attack. These methods are based on research by Allen and

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Perkins [8], Hopkins [6], Jorgensen [7], and others that are not mentioned in this paper.

For body lift in the nonlinear angle of attack range DATCOM present three methods. The first method taken from Hopkins [6] for calculating lift coefficient applied only to bodies of revolution and at low angle of attack. The second method applies to bodies of elliptical cross section and bodies of revolution, this method is based on the concept of vortex lift as presented by Polhamus [5]. The third method is in principle the most general in application, it presented by Jorgensen [7], applies to body of arbitrary shape and angle of attack from 0 deg. to 180 deg. in large range of speed.

For estimating body drag due to angle of attack, four method are presented. The first method is taken from Hopkins [6] and applies to a high range of fineness of bodies of revolution. This method assume that the flow is potential over the forward part of the body and has no viscous contribution in this region. The second method taken from Allen and Perkins [8] is not accurate for bodies of low fineness ratio, it assumes that the viscous contribution at each station along the body is equal to the steady-state drag of a section of an infinite cylinder placed normal to the flow with velocity $V\sin(\alpha)$. The third method, taken from Kelly [9] is limited in application to small angles of attack and moderate fineness ratio. The last method, taken from [9] is based on slender-body theory to calculate the axial force coefficient.

Many authors uses *CFD* in submersible vehicle projects and optimizations due to the large amount of information obtained with reduced cost and time compared with experimental tests. Barros *et al.* [15] compared the analytical and semi-empirical results with numerical results for normal force and momentum coefficient of an AUV. It was shown that the *CFD* approach allows for a good prediction of the coefficients and shows qualitative information from flow visualizations; Juong *et al.* [17] used *CFD* to optimize the design of an AUV hull. They used commercial software and they obtained an optimum value of drag force and pressure and velocity fields. It was concluded that the CFD method is well capable of economically evaluating the hydrodynamic derivatives of submersible platforms such as submarines, torpedoes and autonomous underwater vehicles. These methods may be listed as direct numerical simulation (DNS), large eddy simulation (LES) and Reynolds averaged Navier Stokes (RANS) methods, DNS and LES need very high performance computer. RANS models need a large compiling time.

In this paper, we are interested to investigate the hydrodynamic drag force over a UUV hall form with the Datcom method.

2. Experimental description

2.1 General

Measurement of hydrodynamic forces acting on an underwater vehicle is important when developing the dynamic model, improving accurate motion control, and achieving accurate tracking of desired trajectories in aquatic environments [10-13]. Although measurements provide valuable data, most experimental research on axisymmetric bodies were conducted in a water tank made a critical comparison between the drag characteristics based on volume, surface area, and frontal area for different axisymmetric bodies. Fig.1 shows the most common shape adopted in AUVs such as the MAYA by Barros [15] and REMUS by Prestero [11].

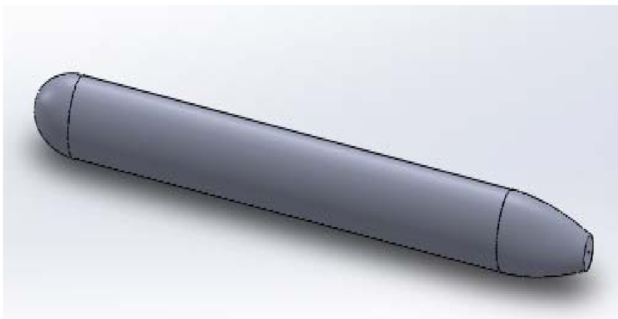

Fig. 1 Over hull UUV torpedo shape

In order to calculate the vehicle coefficients, first we must establish the vehicle profile, define its mass, buoyancy, and finally inertia moments. The bare hull of the AUV has a torpedo shape composed by a nose, a middle body, and a tail section. The nose and the tail shape are based on Myring [16] profile equations, which describe a body contour with minimal drag coefficient for a given fineness ratio (body length/maximum diameter, For reference, Myring [16] assumes a total body length of 100 units, and classifies body types by a code of the form *a/b/n/θ/½*d, where θ is given in radians. The curve shape of the nose and tail sections are determined from (1) and (2).

$$r(x) = \frac{1}{2} d \left[1 - \left(\frac{x + a_{offset} - a}{a}\right)^2\right]^{\frac{1}{n}} \quad (1)$$

$$r(x) = \frac{1}{2} d - \left[\frac{3d}{2c^2} - \frac{\tan(\theta)}{c}\right] (x - (a+b))^2 + \left[\frac{d}{c^3} - \frac{\tan(\theta)}{c}\right] (x - (a+b))^3 \quad (2)$$

Table 1 gives the dimensions of the shape parameters and the designed shape of the AUV hull based on the these equations is shown in Fig. 1.

2.2 Hydrodynamic 6Dof balance

The 6Dof PMM is designed and build to measure axial, normal, and side forces and moments in pitching and yawing simultaneously. This balance contains six load cells connected through six small rods with flexures at each end which decouple the hull forces in six components acting along the rods. The model is mounted via a strut to the 6Dof balance, and is covered with foil to reduce the vortexes developed by the end of the strut as presented in fig. 2.

Fig. 2 6 Dof balance

2.3 Balance calibration

The load cells of the 6Dof balance are calibrated to a specific force before it will be installed. The force measured by the sensors can be given by the following relation:

$$Force(N) = v_i \times c \times 9.81 \quad (3)$$

Table 1 Myring parameters of the hull AUV

Parameter	Value	Units	Description
a	0.100	m	Nose Length
a_{offset}	0	m	Nose Offset
b	0.966	m	Midbody Length
c	0.250	m	Tail Length
c_{offset}	0.066	m	Tail Offset
n	2	n/a	Exponential Coefficient
θ	23	radians	Included Tail Angle
d	0.125	m	Maximum Hull Diameter
l_f	1.250	m	Vehicle Forward Length
l	1.316	m	Vehicle Total Length

Where $v_i(volts)$ is the typical output corresponding to the "i" load cell from the "i" direction, c in $(kg/volts)$ is the calibration tensor. In reality this equation was affected with other sensors in several directions. A test was done to develop the correction factor for each load cell for the multiplication with the force value obtained during the test. A calibration matrix was developed by elements of matrix by values from corresponding calibration curves which is obtained as

$$[f] = [C] \times [F] \tag{4}$$

Where $[F]$ is the resultant applied forces acting in each direction on the AUV hull, $[C]$ is the calibration matrix, and $[f]$ is the load on each sensor correspond to the output voltage.

We can develop the equation as

$$\begin{bmatrix} n_1 \\ n_2 \\ n_3 \\ s_1 \\ s_2 \\ a_x \end{bmatrix} = \begin{bmatrix} 0.344 & 0.002 & -0.005 & 0.004 & 0.009 & 0.006 \\ 0.002 & 0.121 & 0.118 & 0.003 & 0.005 & 0.008 \\ 0.001 & 0.004 & -0.005 & 0.245 & 0.244 & 0.289 \\ 0.003 & 0.004 & 0.002 & 0.214 & -0.209 & 0.006 \\ 0.005 & -0.003 & 0.001 & -0.198 & -0.189 & 0.199 \\ -0.001 & 0.156 & -0.158 & 0.003 & 0.009 & 0.007 \end{bmatrix} \times \begin{bmatrix} N_1 \\ N_2 \\ N_3 \\ S_1 \\ S_2 \\ A_x \end{bmatrix} \tag{5}$$

Where N_1, N_2 and N_3 are the normal forces, S_1 and S_2 are the slide forces, A_x is the axial force acting on the AUV., and $n_1, n_2, n_3, s_1, s_2, and\ a_x$ are the components of $[f]$ on each sensor. By inverting this matrix and multiplying by the measurement forces related to the output voltages, we obtain the full resultant forces.

$$[F] = [C]^{-1} \times [f] \tag{6}$$

In this application, $[C]^{-1}$ is defined by:

$$[C]^{-1} = \begin{bmatrix} 2.9068 & 0.0366 & -0.0749 & 0.0388 & 0.0022 & -0.0687 \\ -0.0881 & 4.2966 & -0.1891 & 0.0629 & -0.0549 & 3.6808 \\ -0.0148 & 4.2456 & -0.0166 & -0.0276 & -0.0349 & -3.2924 \\ -0.0041 & -0.067 & 1.011 & 2.3888 & -1.2797 & -0.0688 \\ 0.0066 & 0.0566 & 1.0889 & -2.8928 & -1.2877 & -0.0889 \\ -0.0489 & 0.0287 & 2.0892 & 0.0492 & 2.5766 & -0.0267 \end{bmatrix} \tag{7}$$

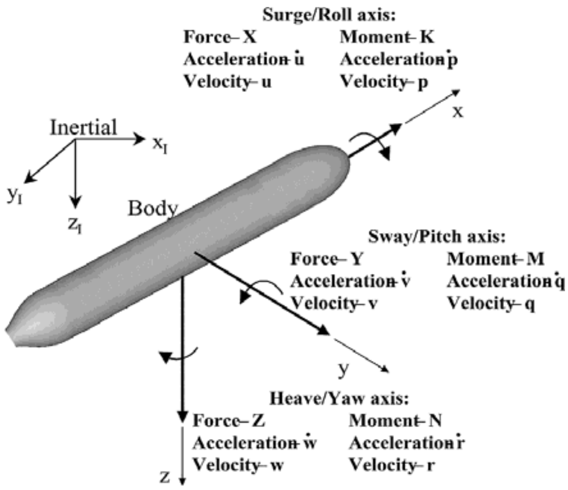

Fig. 3 Reference frame and degrees of freedom

To extract the axial, the normal and the side forces, and the pitching, the rolling, and the yawing moments acting on the hull body related to the body coordinate system, we refer to these equations as

Axial force

$$F_a = A_x \tag{8}$$

Normal force

$$F_n = N_1 + N_2 + N_3 \tag{9}$$

Side force

$$F_s = S_1 + S_2 \tag{10}$$

Pitching moment

$$M_p = (N_1 + N_2)l_1 - N_3 l_3 \tag{11}$$

Rolling moment

$$M_r = N_1 b_1 - N_3 b_2 \tag{12}$$

Yawing moment

$$M_y = S_1 c_1 - S_2 c_2 \tag{13}$$

Where l_i,b_i and c_i present the strain gauge location from the nose of the model respectively in the x, y, and z axes. Fig. 3 shows the lift and drag forces in the longitudinal plane (the x-z plane refers to the flow direction) acting on the body, and the resolution of these forces into components along the x and z axes. So, we can express theses forces as following:

$$D = F_a cos\alpha - F_n sin\alpha \tag{14}$$

$$L = -(F_a sin\alpha + F_n cos\alpha) \tag{15}$$

Where D and L are the lift and drag forces respectively, and α is the angle of attack.
The drag, lift, and the moment coefficients can be expressed as follows:

$$C_D = D/(1/2\rho V^2 A_{ref}) \tag{16}$$

$$C_L = L/(1/2\rho V^2 A_{ref}) \tag{17}$$

$$C_M = M_p/(1/2\rho V^2 A_{ref} L \tag{18}$$

Where C_D is drag coefficient, C_L is lift coefficient, C_M is the pitching moment coefficient, ρ is the density of water, V is the speed of the AUV, A_{ref} is the reference, and L is the length of the AUV. area In this paper, we use cross-sectional area as the reference area.

2.4 Experimental procedure

Methods for measuring hydrodynamic forces vary greatly and many distinct approaches exist in the literature such as Planar Motion Mechanisms (PMMs). Due to complexity, no load cell was installed inside the body like some previous experiment, [14,15]. In order to simulate the hull vehicle, a 9 meters towing tank was used with carriage system pulled by 2 Hp motor. The motor is controlled by a speed driver system with operating frequencies between 0 and 50 Hz. A gear box increases the power delivered and reduces the rpm to the sixth. The carriage was positioned on eight ball bearing to equal weight distribution on each bearing along the rails allowing the carriage to slide with minimal friction. In order to measure the forces and torques, a 6Dof dynamometer was used.
Throughout the static hydrodynamic tests, the hull vehicle, the model is towed at a constant depth, range of speed from 0.2 to 0.6 m/s and a range of angle of attack between 0 and 15 degrees. These series of experiments were conducted for nominal free stream Reynolds numbers from 2 10^5 to 7.5 10^5.

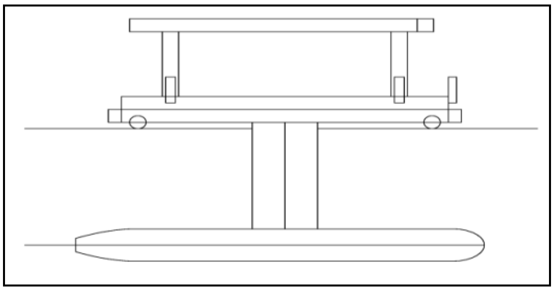

Fig. 4 Test section of a wind tunnel

The experimental setup is shown by a schematic representation in Fig. 4. A data acquisition system was used to collect signals provided from the six load cells, each signal, and the resultant forces [F] were obtained by multiplied the calibration factors with the recorded force at each direction.

3. Results

In this section, the hydrodynamic characteristics of a standard AUV motion in water are studied. The effect of the change of speed on the hydrodynamic drag coefficient derived from ASE and experimental was studied and compared.

3.1. Influence of strut on results

The model was mounted via a strut to the 6 DOF balance. Although the strut was covered with foil shape to reduce the resistance with the flow water. The influence of strut on hydrodynamic force coefficients is assessed by ASE estimation and experimental tests. Results highlights that experimental drag coefficient is almost closed to the Datcom estimation. The shape of the curve of the drag of the body with and without strut are in a good agreement. Figure 5 shows the difference of the drag between the two body conditions obtained, which is equal to 36% at Reynolds number 2.5 10^5 and to 35% at Reynolds number 7.5 10^5.

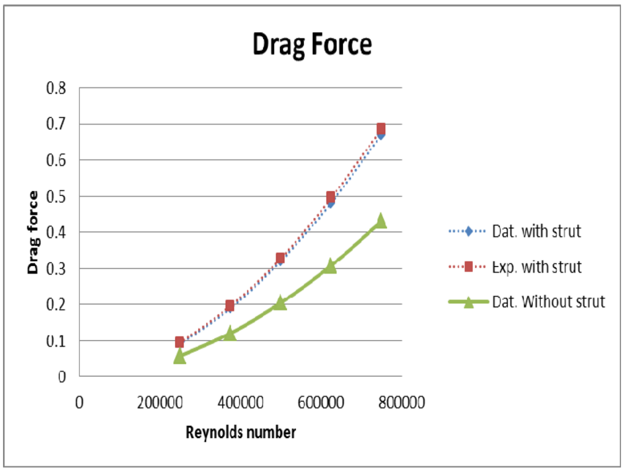

Fig. 5 Drag and Lift forces vs Reynolds Number

3.2. Influence of Reynolds number on total drag force and coefficient

In this section, the effect of Reynolds number on the drag force and drag coefficient was investigated. The drag in newton with varying Reynolds number, in another mean speed, was illustrated in fig. 5. According to these results, it has been observed a comparison between data provide from ASE method (Datcom) and towing tank experiments. The relative speed flow is exposed in Reynolds number ranging from 2.5 10^5 to 7.5 10^5 from 0.2 m/s to 0.6 m/s. An accurate correlation was found between the drag loads determinate from the different methods, for both with and without strut.

The difference in data was found to range from 2% to 8%, with the results increasing with increasing speed. This difference provide from the limitation of the towing tank in high speed.

Figure 6 shows that the value of drag coefficient decrease by 7.2% and 3.1% in the rand of Reynolds number of 2.5 10^5 to 7.5 10^5. Highlight that as the speed increase, the drag coefficients decrease. This fact is attributed in to the increase of the speed with the decrease of the pressure coefficient decrease without any change in friction coefficient. While at a specific Reynolds number, the drag coefficient becomes almost constant. This can be deduced from the fact that the total drag coefficient does not experience significant changes from Re > 6 10^5. Above this Reynolds number, the effect of the flow speed is very limited.

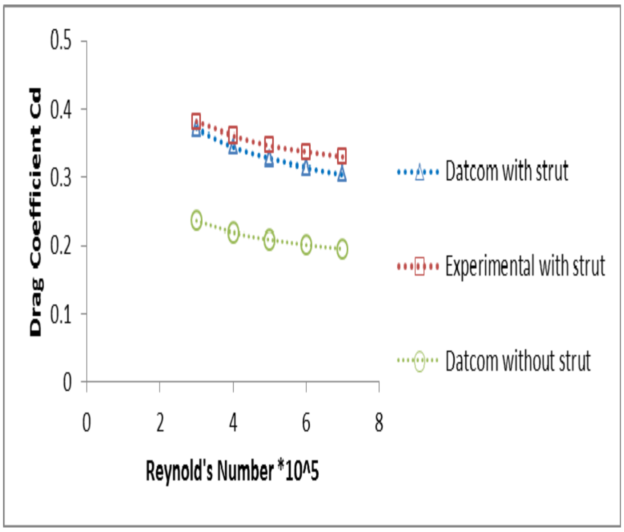

Fig. 6 C_D vs Reynolds Number

3.3. Influence of angle of attack

Figure 7 presents the variation of the measured drag force coefficient for different Reynolds number at various angles of attack. In this experimental work, the used static AOA are at 0°, 5°, 10° and 15° at different speed of 0.2 m/s, 0.3 m/s, 0.4 m/s, 0.5 m/s and 0.6 m/s. according to these results, it has been observed that the coefficient of drag value increases with the increase of the AOA in the range of Reynolds Number 2.5 10^5 to 7.5 10^5. Also, it has been noted that the value of the drag coefficient increases by 3.5% and 4.6%, 16.7% and 19.7%, and 71.4% and 87.1% as AOA increases from 0° to 5°, 10°, and 15° in the range of Reynolds number considered with reference to AOA equal to 0°. It can be seen an increase of the drag coefficient above 10° of the AOA.

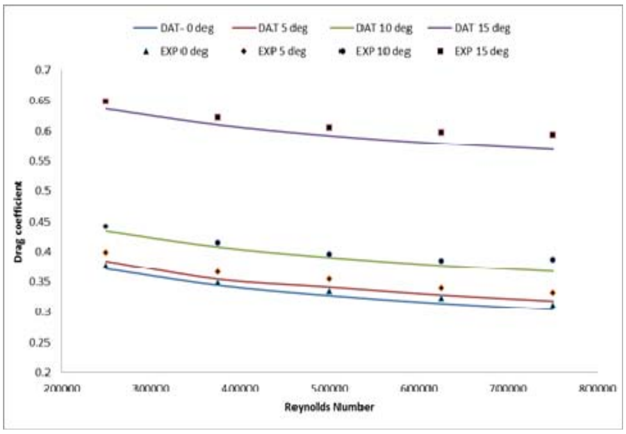

Fig. 7 C_D vs Reynolds Number at various angles of attack

4. Conclusions

Datcom method was used for predicting the hydrodynamic characteristics of torpedo shape underwater vehicle in pitching attitudes and a wide range of speed. This model was then tested in a water tank to obtain systematic data over relevant ranges of attitudes and flow conditions. A six Dof hydrodynamic balance was used for this reason. To validate this method, the experimental data were then used and coupled with theoretical Datcom results. This study was limited to the axial forces which mean the coefficient of drag. Static hydrodynamic characteristics computed by Datcom method are shown to agree closely with experimental results for torpedo slender body of circular cross sectional. However, because the experimental results are limited to angle of attack of less than 15° and velocities only less than 2 Km/h (0.6 m/s), further comparison of the Datcom method with more data is needed.

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