THE DEVELOPMENT AND APPLICATION OF UNDERWATER VEHICLE DESIGN TECHNIQUES

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ABSTRACT

The lack of accurate methods for predicting vehicle hydrodynamic characteristics, given the vehicle's external geometric configuration, has been a recurring problem and an uncertain process in submersible vehicle design. Traditionally, extensive testing and modification of models and full-scale vehicles are required before an acceptable configuration is defined - an approach costly in both time and money. The Hydromechanics Division at NCSC began a systematic approach in 1973 for analytically determining the hydrodynamic coefficients of submersibles by developing new semi-empirical methods, consisting of theoretical formulas and data correlations of systematic data bases obtained from wind tunnel tests of a modular model designed to cover submersible configurations. Comparisons of the new methods with experimental data and other available methods are made and their application in the design, fabrication and test of a towed vehicle system is discussed.

INTRODUCTION

The lack of accurate methods for predicting vehicle hydrodynamic characteristics, given the vehicle's external geometric configuration, has been a recurring problem and an uncertain process in submersible vehicle design. Traditionally, extensive testing and modification of models and full-scale vehicles are required before an acceptable configuration is defined - an approach costly in both time and money. This methodology is particularly unacceptable for the nature of programs conducted at NCSC, where funding is normally directed at system components rather than at development of the vehicle itself. A typical program at NCSC generally involves the improvement or development of new sensor technology and is normally considered high risk; consequently, available funds are concentrated on sensor development with little or no funds available to develop an adequate towed or self-propelled vehicle to carry the sensor package. The sensor package, however, may set volume, geometry, or motion limitations on the vehicle, which, if not met, will result in degraded performance or failure of the entire system. Because of the number of programs at NCSC, strict time constraints generate the necessity to

analyze or design about 10 vehicles a year.

To meet the above challenge, the Hydromechanics Division at NCSC began a systematic approach in 1973 for analytically determining the hydrodynamic coefficients of submersibles. The initial phase of this effort involved the adaptation of semi-empirical methods, consisting of theoretical formulas and data correlations of systematic data bases developed in the aerospace community for subsonic aircraft (Ref 1), to underwater vehicles. The heart of the semi-empirical method is the body build-up technique in which hydrodynamic coefficients for isolated components are determined, interference effects between various components are predicted, after which the contributions are totaled to give the hydrodynamic coefficients of the complete vehicle (Fig 1). These coefficients are then utilized in the equations of motion to evaluate vehicle stability and performance as a function of the vehicle mass characteristics and external shape. Because semi-empirical methods are generally algebraic in form, they are rapid, computationally inexpensive, and thus provide a means for analyzing many geometric variations during the design/analysis cycle of an underwater vehicle development.

The accuracy of this methodology depends largely upon the systematic data bases from which it is developed. Methods used for subsonic aircraft analysis were developed from aircraft data bases and experience with these methods has demonstrated various deficiencies in the aircraft technology when applied to submersibles. In particular, the hydrodynamic characteristics of most submersibles are governed by small, low aspect ratio fins mounted on a tapered afterbody where the boundary layer is thick and/or separated as opposed to aircraft where tail fins are essentially in potential flow. Furthermore, no general systematic data base for submersibles is available as the bulk of existing data is for specific vehicle designs and thus of limited value for constructing semi-empirical methods for submersibles.

To remedy this problem, NCSC initiated a comprehensive program in 1976 to (1) develop a series of interchangeable model components to cover a wide range of typical submersible configurations, (2) test sufficient configurations to provide a broad, systematic data base and (3) develop semi-empir-

ical methods from this data base as functions of body, fin, and body-fin geometry. The first effort involved cruciform tail configurations and was performed under contract by Nielsen Engineering and Research, Inc. (Ref 2).

SYSTEMATIC DATA BASE DEVELOPMENT

Modular models were designed to systematically cover those geometry ranges typical of underwater vehicles, viz. (1) bodies (7" diameter) having overall slenderness ratios varying from 4 to 12, (2) ellipsoidal noses having length/diameter ratios of 1/2, 1 and 2, (3) conical afterbodies having length/diameter ratios of 1, 2 and 3 and (4) tail fins with 2 panel aspect ratios of 1/2, 1 and 2, taper ratios of 0, 1/2 and 1 and body diameter/exposed tailspan ratios of 1, 1.4 and 1.8. Special three-component strain gauge balances were built to measure tail fin normal force, pitching moment and root bending moment. These balances were used on a reflection plane to obtain fin-alone data and on the conical afterbodies to obtain fin forces and moments separate from the total configuration forces and moments which were obtained from a six-component balance. The measurement of the tail fin forces and moments with separate balances was vital for determining body-on-tail and tail-on-body interference effects.

Wind tunnel tests were conducted in the 12-foot pressure wind tunnel at NASA/Ames Research Center, Moffett Field, California, an installation chosen for its high Reynolds number capability. A comparison of the Reynolds number range available in the tunnel with the operational Reynolds numbers of typical Navy vehicles is shown in Figure 2. The configurations tested included fin-alone, body-alone and body-plus-tail and, in general, each configuration was tested at free stream Reynolds numbers of 4×10^6 and 8.5×10^6 per foot.

METHODS DEVELOPMENT AND COMPARISON OF TECHNIQUES

Using the data obtained from the wind tunnel tests, semi-empirical methods were developed for predicting the hydrodynamic coefficients of both a complete vehicle configuration and individual vehicle components. For isolated fins, methods were developed to determine variations of fin normal force coefficient and fin chordwise center of pressure location with the angle of attack and Reynolds number, as functions of fin taper ratio and aspect ratio. A comparison of the methods with independent fin data is shown in Figures 3 and 4.

For isolated bodies, methods were developed to determine variations of the normal force coefficient, the pitching moment coefficient and the axial force coefficient with angle of attack and Reynolds number, as functions of overall body fineness ratio, nose shape, and base taper ratio. Figure 5 compares body-alone normal force coefficient slope $Z'_{w,B}$ as predicted by NCSC's new methods (N, Ref 2) and the U. S. Air Force DATCOM methods (D, Ref 1), the Hydroballistic Handbook methods (H, Ref 3) and early methods by Abkowitz

and Paster (A, Ref 4) for three typical submarines.

For complete configurations, methods were developed for predicting the carryover loading on the body (and its center of pressure) due to the tail fins and the amplification of tail fin normal force (and change in its center of pressure) due to the body. Figure 6 compares body-plus-tail normal force coefficient slope $Z'_{w,BT}$ as predicted by the D, H, A and N methods. Based on extensive comparisons with experimental data, the mean per cent errors for the linear static and rotary coefficients $Z'_{w,B}$, $M'_{w,B}$, $Z'_{q,B}$ and $M'_{q,B}$ are 32%, 10%, 28% and 10% for the D, H, A and N methods, respectively.

Figures 7 and 8 present comparisons of C_N and C_M versus α as predicted by the H and N methods. The N method uses a quadratic term based on crossflow theory to predict nonlinear effects while the H method uses a cubic term obtained from curve fitting a data base. As evidenced in Figure 8, the cubic representation of nonlinear effects can create severe problems if one applies the H method to a configuration that is outside of the original data base.

DESIGN AND ANALYSIS OF A TOWED VEHICLE SYSTEM

The techniques developed at NCSC were applied to the design of a towed environmental sensing system (Ref 5). The major component of the system is a sensor vehicle which samples sea water at a given depth at various tow speeds. The sensor vehicle was analyzed along with the other system components including tow cable, depressor, and tow ship (Fig 9) to meet a specific set of performance requirements. These requirements included operational depths of 0-500 feet, tow speeds from 0 to 20 knots, depth excursions less than ± 3 feet, passive control, designing for multiple sensor vehicles, and insuring that the natural frequency of the sensor vehicle was outside a specified bandwidth.

The analysis of the sensor vehicle began by defining a base case geometry and mass distribution. A base case vehicle was defined which was intended to meet all the design requirements (Fig 10). The longitudinal and lateral hydrodynamic coefficients were then computed as a function of geometry and the equations of motion (Ref 6) were solved in the frequency domain to yield stability and dynamic response data. The base case vehicle was found to be stable both longitudinally and laterally. Even though the vehicle was stable, it had not been determined that this design was the best configuration to meet the design requirements. To evaluate the effectiveness of the base case design, several geometric variations were analyzed and compared. These geometry variations included fin area, center of gravity position, net buoyancy, and cable length from the cable breakout to the sensor vehicle. As an example of the dynamic data produced in this analysis, Figure 11 presents base case vehicle depth response as a function of fre-

quency of oscillation for a ± 1.0 foot heave displacement at the cable breakout for three cable lengths. The response indicates that peak amplitude is increased by 65 percent in changing from a 50-foot cable to a 10-foot cable while the resonant period decreases from 21 seconds to 10 seconds. These data along with similar data from other geometry plots were utilized to determine the best geometry to meet the sensor vehicle performance requirements.

The proper design of the complete towed system required selecting the optimum depressor and cable which would achieve the desired depth/speed performance. A three-dimensional cable catenary code (Ref 7) was used to predict the cable scope and winch tension as a function of downforce. From the cable analysis a dynamic depressor was selected which provided 5000 pounds of downforce at 20 knots. Sectional cable fairing was utilized to reduce hydrodynamic drag. The catenary solution for several speeds is presented in Figure 12. An analytical stability analysis conducted on the depressor revealed that lateral stability was very sensitive to tow point, center of gravity, and center of pressure positions.

In order to estimate the expected sensor vehicle motions, the forcing functions exciting the vehicle must be known. For this towed system, ship stern motion was identified as the primary forcing function. Vertical stern motion response amplitude operators were combined with expected seaway spectra to produce motion estimates at the cable breakout. Using vehicle response data similar to that of Figure 11, maximum vehicle depth and pitch variations were predicted to be ± 2.2 feet and ± 1.5 degrees, respectively, for two foot seas.

Full-scale tests of this system were conducted in late summer 1979 (Ref 8). After several operational problems were solved, the sensor vehicle performance was found to be within the specified requirements as seen by the strip chart data presented in Figure 13.

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FIGURE 1. VEHICLE BUILD-UP PROCEDURES

FIGURE 2. SUBMERSIBLES' REYNOLDS NUMBER RANGE

FIGURE 3. FIN-ALONE NORMAL FORCE COMPARISON

FIGURE 4. FIN-ALONE PRESSURE CENTER COMPARISON

FIGURE 5. BODY-ALONE Z'_W COMPARISON

FIGURE 6. BODY-PLUS-TAIL Z'_W COMPARISON

FIGURE 7. BODY-ALONE C_N , C_M COMPARISON

FIGURE 8. BODY-PLUS-TAIL C_N , C_M COMPARISON

FIGURE 9. SYSTEM COMPONENTS

FIGURE 10. BASE CASE VEHICLE DESIGN

FIGURE 11. VEHICLE RESPONSE TO HEAVING TOW POINT

FIGURE 12. SYSTEM TOW CHARACTERISTICS

FIGURE 13. MEASURED SENSOR VEHICLE RESPONSE