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CONCEPTUAL DESIGN OF A NEW PLANAR MOTION MECHANISM FOR INVESTIGATING THE STABILITY AND CONTROL CHARACTERISTICS OF SUBMARINES

by

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ABSTRACT

Model experiments are performed in the straightline basin of the David Taylor Model Basin using the Planar Motion Mechanism (PMM) to investigate the stability and control characteristics of candidate designs of submarines and other submerged vehicles. The apparatus is used to determine the hydrodynamic forces and moments that are developed on the hull, appendages, and propeller. This information is used to determine the stability and control derivatives in both the vertical and horizontal planes of motion, including the static, rotary, and control derivatives. The existing Planar Motion Mechanism System (PMM) has been in use at DTMB since June 1957. There are critical technical limitations associated with the existing apparatus, including the maximum size of the model that can be used for the tests, the maximum angle that the tilt table can be set, the accuracy of setting the tilt table angle, the maximum strut spacing, the range of oscillation frequencies, the inability to remotely set the phase angle between the struts, and the need to electronically separate the in-phase and out-of-phase quadrature. This report presents a conceptual design for a PMM that would eliminate these limitations.

ADMINISTRATIVE INFORMATION

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INTRODUCTION

The stability, control, and maneuvering characteristics of submarines and other submerged vehicles are investigated at the David Taylor Model Basin (DTMB) by performing straightline and rotating arm captive-model experiments, radio-control model experiments, and hydrodynamic analyses. From the results of the captive-model experiments and analyses, the hydrodynamic forces and moments are measured and the appropriate stability and control derivatives and hydrodynamic coefficients are determined. This information, coupled with the motion trajectories from the radio-control model experiments, is used to evaluate the stability and control characteristics of the submarine, to develop equations of motion and a mathematical model of the submarine, and to use the mathematical model to perform computer simulations of the motions of the submarine. A discussion of this process can be found in Reference 1.

Straightline basin captive-model experiments are performed on the Planar Motion

Mechanism (PMM) described in References 2 and 3. The existing Planar Motion Mechanism System (PMM) has been in use at DTMB since June 1957. However, there are critical technical limitations associated with the existing apparatus. This report provides information on how the PMM is used to determine the stability and control characteristics of submarines and other submerged bodies, provides a justification for a new design for a PMM, and provides a conceptual design and specifications for a PMM that would eliminate the aforementioned limitations.

GENERAL CONSIDERATIONS

The PMM is used to determine the stability and control derivatives in both the vertical and horizontal planes of motion, including the static (Z_w', M_w', Y_v', and N_v'), rotary, (Z_q', M_q', Y_r', and N_r'), and control derivatives, and the hydrodynamic force and moment coefficients associated with variations in angle of attack, angle of drift, and over and under propulsion. The nomenclature used for analyzing the stability and control characteristics of submarines and submerged vehicle is provided in Reference 4. If the vehicle is symmetric (for example, a vehicle with a hull that is a body of revolution, fitted with four identical cruciform stern appendages), then only vertical plane experiments need to be performed. A sketch of the existing PMM is provided in Figure 1.

The hydrodynamic forces and moments are measured currently over a range of angles of attack (up to about 18 degrees) and sternplane angles in the vertical plane, and over a range of angles of drift and rudder angles in the horizontal plane. In addition, oscillation experiments are performed in the heaving and pitching mode (swaying and yawing in the horizontal plane) at zero speed and underway. By measuring the in-phase and out-of-phase components of the hydrodynamic force and moment the added mass, added moment of inertia, and rotary (effect of angular velocity) derivatives can be determined.

The standard program of static stability and control experiments is usually conducted at a model speed of about 6.0 knots which corresponds to a Reynolds number, based on a model having a 20-foot overall length, of about 10 to 15 million. Experiments have been performed with various submarine designs to investigate the effect of scaling on the hydrodynamic forces and moments developed on the hull and appendages either at an angle of attack or with the control surfaces deflected to an angle. These experiments have indicated that the hydrodynamic force and moment coefficients vary with Reynolds number. However, there appears to be a Reynolds number above which the hydrodynamic force and moment coefficients no longer significantly change with Reynolds number. Based on comparisons between the results of various captive-model experiments and full-scale trials, if model experiments are performed at Reynolds numbers above 10 to 15 million, then any scale effects between model and full-scale appear to be negligible for the purposes of making stability and control predictions.

JUSTIFICATION FOR A NEW PLANAR MOTION MECHANISM

In the past, most PMM experiments were performed on the U.S. Navy's submarine designs and followed the same standard procedures. The experimental programs were somewhat routine and repetitive. For example, only the hydrodynamic forces and moments developed on the complete model were measured, and these data were used to determine only the basic stability and control derivatives and hydrodynamic force and moment coefficients. Today, new and complex geometries have become commonplace. Many additional measurements are required, including the hydrodynamic forces and moment that are developed on the propeller, on the afterbody, on each control surface, and on certain appendages. There is a better understanding of the hydrodynamics of severe vehicle motions, and this has resulted in more complex equations of motion than the standard equations given in Reference 5 and more complex test programs than the one performed in Reference 6.

For example, measurements are required now during propeller backing conditions, and these measurements are often unsteady. Methods are being developed to collect the large number of channels of data. New data analysis computer programs are required, particularly methods to analyze the frequency content of the unsteady data.

There are critical technical limitations associated with the existing PMM. These limitations include the strength of the apparatus which affects the maximum size of the model that can be used for the tests, the maximum angle that the tilt table can be set, the accuracy of setting the tilt table angle, the maximum strut spacing, the range of oscillation frequencies, the inability to remotely set the phase angle between the struts, the accuracy of setting the phase angle, and the need to electronically separate the in-phase and out-of-phase quadrature to improve the accuracy of the Fourier analysis of the oscillation data.

There is a need to reduce the level of uncertainty in the measurements so that computer simulations which are used to predict the motions of the vehicle can be made with greater confidence. This report provides an uncertainty analysis for data derived from the existing PMM followed by a conceptual design for a new PMM that would improve the uncertainty and eliminate the limitations discussed.

UNCERTAINTY ANALYSIS

INTRODUCTION

As discussed in Reference 7, there are two contributions to the total uncertainty. The first contribution is called bias, and it is defined as any effect which is held constant throughout the experiment and which leads to a constant variation of the results from the true value. The second contribution is defined as the precision error, and it is the random scatter of data which is seen when experiments are repeated under nominally identical conditions. The uncertainty in the measurements of each force and the angle of attack are discussed.

MEASUREMENT OF THE FORCE

Sources of error include the following: (1) the 4-inch block gages (variable reluctance transducers) used to measure the forces, (2) the signal conditioners, (3) the 6-Hz low-pass filters, (4) the 15-bit analog-to-digital converter, (5) the power supply for the signal conditioners, (6) the alignment of the apparatus used to calibrate the 4-inch block gages, (7) the alignment of the gages in the calibration stand, (8) the sensitivity of a gage to forces applied perpendicular to its axis, (9) the errors in the fabrication of the model, (10) the changes in the water temperature which affects the density and viscosity, (11) the currents in the basin, and (12) the errors in ballasting the model for neutral buoyancy and trim, (13) the unanticipated unsteady conditions while data are being collected, and (14) the interpretation of data, fairing of curves through the data, determination of slopes, choice of mathematical fit of data, and choice of data to be fitted. Most of the bias and precision errors are negligible based on observations, tests, and analyses performed over a period of many years.

The calibration precision error of a 4-inch block gage is determined by placing 5 and 10 pounds weights, each having an accuracy of 0.01 percent, to a pan which was attached with a 5 to 1 lever arm to the gage. The maximum load applied to the gage is about 300 pounds, both in the positive and negative directions.

If the calibration were to consist of the infinite number of readings, then the readings would coincide with the Gaussian or normal distribution. The distribution of readings is called the parent population. A sample population is composed of a finite number of readings taken from the parent population. The distribution has both a mean and a standard deviation. As the value of the standard deviation increases, the range of the values of the expected readings also increases. That is, the scatter in the readings is large and thus the precision error is large. The probability of a reading being between a band of plus and minus 1.96 times the standard deviation of the mean is 95 percent. That is, 95 percent of the readings from a Gaussian parent population are within a band of plus and minus 1.96 times the standard deviation of the mean.

The relationship between the sample mean m and the corresponding parent population mean m_* can be determined by making use of the "t" probability distribution. For a given sample size n, the random variable t which has a "t" probability distribution is given by

$$t = (m - m_*)n^{1/2}/s$$

where s is the sample standard deviation. Hence, for a sample of n measurements drawn from a Gaussian distribution a precision limit P can be defined for the mean of the measurements as

$$P = t_1 s/n^{1/2}$$

The calibration of a block gage used to measure the force indicated that the sample mean for the sensitivity was 29.76 millivolts per pound and the sample standard deviation for the sensitivity was 0.12 millivolt per pound for a sample size of 24 different applied loads to the gage. For a sample size of 24, the probability is 0.95 (95 percent confidence) that the random variable t is between $t_1 = -2.069$ and $t_1 = 2.069$. Hence,

$$P = 0.0507 \text{ millivolt per pound}$$

$$P/m = 0.0017$$

The uncertainty U for the 24 calibrations is determined by combining the precision and bias limits by the root-sum-square method. It has not been possible to determine the actual bias limit at this time. However after carefully evaluating the calibration process, it appears that the bias limit would be relatively small compared to the precision limit. Hence, the uncertainty is

$$U/m = P/m = 0.0017$$

or 0.17 percent for a 95 percent confidence. The value of m_* is between 29.71 and 29.81 millivolts per pound.

The relationship between the sample standard deviation s and the parent population standard deviation s_* can be determined from the chi-square probability distribution. For a sample size n, the random variable u having a chi-square probability distribution is given by

$$u = (n - 1)s^2/s_*^2$$

For a sample size of 24 the probability of the random variable u being between $u_1 = 11.688$ and infinity is 0.975 and between $u_1 = 38.076$ and infinity is 0.025. Using the relationship

$$s_*^2 = (n - 1)s^2/u_1$$

the value of s_* is between 0.09 and 0.17.

MEASUREMENT OF GAGE INTERACTIONS

Straightline and rotating arm experiments are performed with the model supported by two vertical struts in tandem, usually spaced 6 to 8 feet apart. The reference point is located midway between the two struts. Three force block gages are located at each strut as an assembly for measuring the longitudinal, lateral, and normal force components with respect to the body axes. The pitching and yawing moments about the reference point are determined from the difference in the measured reaction forces at each strut multiplied by one half the strut spacing. A separate gage to measure the rolling moment is located at either the forward or aft strut.

The usual practice is to calibrate each individual gage as discussed previously. It is assumed that when the block gages are assembled into either the forward or aft unit they are properly aligned to measure only the force they are positioned to measure. That is, it is assumed that when a pure normal force is applied to the model, the normal force block gages measure the total normal force and the other force gages measure zero force. However, it has been found that when a large pure normal force is applied to the model, there are small output signals on all of the other gages, particularly the lateral force gage. It is important to be able to quantify these small lateral force output signals when relatively small lateral forces need to be measured.

To determine the interactions among the various block gages, combinations of known loads must be applied to the model. In 1993, a method was developed for calibrating the block gage assemblies by loading a model with a plus or minus lateral force, plus or minus yawing moment, plus normal force, and plus or minus rolling moment. The calibrations were performed in water in the drydock at the end of the towing basin. Another method was developed for calibrating the gages in air by loading the gage channel with all combinations of forces.

A typical calibration sequence was as follows: (1) a zero was taken with all of the weight pans unloaded, (2) calibrated weights are placed on selected pans, and (3) the weights were removed from the pans and another zero was taken. The data that were taken included the outputs from the seven block gages, the weights that were placed on each of the ten pans, and the coordinates of the location at which the loads were applied.

The results of the analysis to determine the interactions among the various block gages indicated, for example, that when a large pure normal force is applied to the model, there are small output signals on all of the other gages, particularly the lateral force gage. Calibrating the individual block gages is an acceptable practice when the forces that are being measured are large. However, it is important to be able to quantify these small lateral force output signals when relatively small lateral forces need to be measured. Although the full linear and nonlinear interaction calibrations do give better matches with the known input forces, the differences overall for the primary forces are not large. The results of the calibrations suggest that systematic cross-channel responses require a full linear or nonlinear interaction matrix. However, the calibrations also indicate that cross-channel block gage outputs have large variations which may mask any systematic trends. For example, there are relatively large uncertainties associated with some of the off-diagonal elements of the interaction matrix. When the variations in the cross-channel block gage outputs are better quantified or eliminated, it will be possible to determine if calibrations to calculate the off-diagonal terms in the matrix are required. Corrections for the deflection of the model marginally improve some of the force predictions, but significantly degrade the rolling moment prediction when the complete linear interaction matrix is used. Based on the variations in the cross-channel outputs and relatively large uncertainties, it appears that any small deflection of the model has only a small effect on accurately resolving the forces and moments.

MEASUREMENT OF THE ANGLE OF ATTACK

Different methods have been used to measure the angle of attack of the model during straightline captive-model experiments performed on the Planar Motion Mechanism. Model support and positioning for this type of experiments is accomplished by an assembly consisting of a tilt table and a pair of twin towing struts. The tilt table is a rectangular frame constructed primarily of 8-inch steel I-beams welded together. A heavy walled steel tubing is inserted transversely through the frame at the longitudinal midpoint and welded to it. The tubing serves as an axle for tilting the table in the pitch plane. The end of the tilt table is moved vertically by a Saginaw ball-bearing screw jack mounted in the support bracket at the carriage end. A system of micro-switches is installed on the support backet with a spacing so that one-degree increments can be set on the tilt table over a range of plus and minus 18 degrees.

To improve the accuracy of the measurement of the angle of attack, a potentiometer and a belt drive was added to the apparatus in 1985. A further improvement was made with the addition of an anti-backlash gear in November 1991. An angle encoder was installed at the same time, but it had to be replaced. The angle encoder is a resolver that provides a digital signal of 4096 bits per revolution (0.088 degree per bit). A gunner's quadrant (a bubble level) was used to provide the reference angle for the tilt table.

A summary of the four types of measurements are as follows:

Type	Description
1.	Micro switch
2.	Potentiometer with belt drive
3.	Potentiometer with anti-backlash gear
4.	Encoder with anti-backlash gear

Calibrations have been performed for each type of measurement and then least-square fitted to the line $y = Ax + B$. The coefficients of the least-square fit

A and B, index of determination ID, and standard error of estimate SEE were determined. A sample standard deviation was derived by taking the difference between the actual reading from the measuring device and the value calculated from the least-square fit and then dividing through by the calculated value.

For example, for the Type 4 measurement, a sample standard deviation was calculated to be 0.0092 degrees for a sample size n of 22 different angles to which the tilt table was set. The probability is 0.95 that the random variable t is between $t_1 = -2.086$ and $t_1 = 2.086$. Hence the precision limit is

$$P/m = 0.0041.$$

As indicated previously, the uncertainty is determined by combining the precision and bias limits by the root-sum-square method. The actual bias limit for the measurement of the angle of attack was not determined. However, it appears that the bias limit for the angle of attack measurement would be relatively small compared to the precision limit.

The coefficient A of the least-square fit can be used to determine the average accuracy of measuring the angle of attack over the range -18 to 18 degrees, whereas the normalized precision limit P/m indicates the uncertainty associated with a calibration consisting of n angles. For example, a comparison between a Type 1 and a Type 4 calibration is as follows:

Type	A	B	ID	SEE	n	Accuracy Percent	Uncert. Percent
1	0.9930 deg/deg	-0.000600 deg	0.999909	0.1006 deg	30	0.70	0.67
4	1.0003 deg/deg	-0.017687 deg	0.999994	0.0255 deg	22	0.03	0.41

As can be seen, there is a significant improvement in the accuracy of measuring the angle of attack by using the encoder (Type 4). The uncertainty would be reduced if the number of angles n used in the calibration were increased.

A band about the least-square fit which has the magnitude of plus and minus twice the standard error of estimate (SEE) will contain approximately 95 percent of the data points if the bias is negligible. A comparison between the values of twice the SEE for Types 1 and 4 calibrations is as follows:

Type	2 x SEE degrees
1	0.201
4	0.051

This comparison also indicates that there is a significant improvement in the accuracy of measuring the angle of attack using the encoder (Type 4).

UNCERTAINTY OF THE CONTROL SURFACE ANGLE

The deflection of the sternplanes, rudders, sailplanes, or bowplanes is usually performed manually by loosening a split clamp that prevents the plane from rotating on the stock. The desired angle of the plane is set using a protractor template. After the angle is set, the split clamp is tightened. A line is inscribed on the hull or the fixed portion of the control surface indicating zero angle. It is estimated that the inscribed line can be as much as 1 degree in error, and this is the called the bias B. The ability to read the protracter while setting an angle on the plane can result in a precision error P of about 0.5 degree. The uncertainty U can be determined by combining the precision and bias limits by the root-sum-square method as follows:

$$U^2 = B^2 + P^2$$

Hence, the uncertainty is 1.12 degrees. Since the control effectiveness derivatives are usually determined by measuring the slope of the curves of nondimensional hydrodynamic force and moment over about 10 degrees of control surface angle, the uncertainty of the control surface angle would be 0.1120.

UNCERTAINTY OF MODEL LENGTH, SPEED, AND DENSITY

The uncertainty in the overall length of the model is estimated to be 1/16 inch in 13.9792 feet or 0.0004. The uncertainty in the carriage speed is estimated to be 0.01 knot in 6.5 knots or 0.0015. The uncertainty in the density is estimated to be 0.0006 $lb\text{-}sec^2/ft^4$ for a change of 3 degrees F in 1.9367 $lb\text{-}sec^2/ft^4$ or 0.0003.

PROPAGATION OF INDIVIDUAL UNCERTAINTIES INTO VARIOUS PARAMETERS

Stability Derivatives

The uncertainties in the individual variables propagate through the data reduction equations into the stability and control derivatives. The uncertainties in the measurement of the force and the measurement of the tilt table angle can be used to determine the uncertainty in the stability derivative Z_w'.

The stability derivative Z_w' can be determined by the following expression:

$$c = Z_w' = kZ/a$$

where Z is the change in the measured force, a is the corresponding change in the measured tilt table angle, and k is the nondimensionalizing constant. The square of the value of the uncertainty in c is given by

$$U_c^2 = (U_Z \, dc/dZ)^2 + (U_a \, dc/da)^2$$

This expression can be written as

$$(U_c/c)^2 = (U_Z/Z)^2 + (U_a/a)^2$$

For example, if the Type 4 measurement of the angle of attack is used for the calculation and if the bias limits are negligible,

$$U_Z/Z = P_Z/m_Z = 0.0028$$

$$U_a/a = P_a/m_a = 0.0041$$

Hence,

$$U_c/c = 0.0050.$$

Control Derivatives

Similarly, the uncertainties in the measurement of the force and the measurement of the control surface angle, denoted by s, can be used to determine the uncertainty in the control effectiveness derivative Z_s' where

$$c = Z_s' = kZ/s$$

The square of the value of the uncertainty in c is given by

$$(U_c/c)^2 = (U_Z/Z)^2 + (U_s/s)^2$$

and $U_c/c = 0.1120$.

REPEATABILITY OF THE STABILITY DERIVATIVES

It is difficult at the present time to quantify all of the individual bias and precision errors. However by using the submarine stability and control data base, the following estimates of repeatability may be assigned to the experimental values of the stability and control derivatives for fully appended submarines: (1) static derivatives Z_w', M_w', Y_v', and N_v' about 5 percent, (2) rotary derivatives Z_q', M_q', Y_r', and N_r' about 10 percent if measured on the rotating arm, (3) control derivatives about 10 percent, and (4) added mass and moment of inertia derivatives $Z_{\dot{w}}'$, $M_{\dot{q}}'$, $Y_{\dot{v}}'$, and $N_{\dot{r}}'$ about 7 percent. The uncertainty error in calculating the nondimensional mass is about 2 percent.

UNCERTAINTY IN DETERMINING THE MARGIN OF STABILITY

The uncertainty errors of the individual stability derivatives propagate into the margin of stability. The margin of stability is a function of four nondimensional stability derivatives, the nondimensional mass of the submarine, and the nondimensional longitudinal location of the center of gravity from the reference point. In the vertical plane

$$G_v = 1 - M_w'(Z_q' + m')/[Z_w'(M_q' - x_G'm')]$$

which has the form

$$G = 1 - e_1(e_2 + e_3)/[e_4(e_5 + e_6e_3)]$$

The square of the uncertainty in G is given by

$$(U_G/G)^2 = (e_1/G)^2(dG/de_1)^2(U_{e1}/e_1)^2$$
$$+ (e_2/G)^2(dG/de_2)^2(U_{e2}/e_2)^2$$
$$+ (e_3/G)^2(dG/de_3)^2(U_{e3}/e_3)^2$$
$$+ (e_4/G)^2(dG/de_4)^2(U_{e4}/e_4)^2$$
$$+ (e_5/G)^2(dG/de_5)^2(U_{e5}/e_5)^2$$
$$+ (e_6/G)^2(dG/de_6)^2(U_{e6}/e_6)^2$$

where dG/de_k are partial derivatives and U_{ek}/e_k are the uncertainties in each e_k. The partial derivatives are given by the following expressions:

$$dG/de_1 = -(e_2 + e_3)/(e_4e_7)$$
$$dG/de_2 = -e_1/(e_4e_7)$$
$$dG/de_3 = -e_1/(e_4e_7) + e_1(e_2 + e_3)e_6/(e_4e_7^2)$$
$$dG/de_4 = e_1(e_2 + e_3)/(e_4^2e_7)$$
$$dG/de_5 = e_1(e_2 + e_3)/(e_4e_7^2)$$
$$dG/de_6 = e_1(e_2 + e_3)e_3/(e_4e_7^2)$$

where $e_7 = e_5 + e_6e_3$.

The uncertainty in G depends on the particular values of the nondimensional stability derivatives, the nondimensional mass, and the nondimensional longitudinal location of the center of gravity from the reference point.

For example, the uncertainties in G for motion in the vertical plane for a typical submarine for various estimated uncertainties in the stability derivatives are as follows:

Z_w' and M_w'	Z_q' and M_q'	G
0.05	0.10	0.10
0.10	0.10	0.12
0.10	0.20	0.19

CONCEPTUAL DESIGN OF A NEW PLANAR MOTION MECHANISM

The functional requirements and design constraints for the new Planar Motion Mechanism include the size of the model, the range of tilt table angles, and the oscillation characteristics. The requirements and constraints are as follows:

Model Size	Length	About 20 feet (The maximum size of a model that can be placed in the dry dock is 23.75 feet.)
	Cross Section	About 4 feet by 4 feet
	Buoyancy	About neutrally buoyant
	Added Mass	The maximum ratio of the added mass of the cross section to the added mass of a circular cross section is about 1.5. The ratio for a square is 1.51 and the ratios for appropriate Lewis contours as they transition from almost a circle to almost a square vary from about 1 to 1.5. (See Reference 8.)
Strut Spacing	Variable	Vary from about 4 to 10 feet.
Phase Angle	Adjustable	Vary from 0 to 360 degrees with a resolution of 1/6 of a degree.
	Control	A computer will control the phase angle remotely. A digital readout will be used.
Frequency	Adjustable	Vary from 0 to 3.333 radian per second or 0.530 Hz. The fluctuation in the frequency must be less than 0.0333 radians per second root-mean-square, that is, there is 68 percent probability of a reading being within 0.0333 radians per second of the set frequency.
Amplitude	Fixed	1 inch with the capability to set crank pins for other amplitudes
Towing Speed	Variable	Up to above 6 knots
Tilt Table	Range	Approximately -22 to +22 degrees
	Accuracy	0.03 degree root-mean-squared
	Readout Resolution	0.01 degree

	Response Time	Start, move 1 degree, and stop within 2.5 seconds.
Drive/Actuator	Oscillator	Electric
	Tilt Table	Electric servo motor equipped with a spring set brake.
Control Electronics	Housing	Standard 19-inch rack mountable units housed in Carriage 2 instrumentation Penthouse.
	Control Modes	Manual control of tilt table and oscillator motors by operator when standing adjacent to to the PMM. Automatic control of tilt table and frequency of oscillator.
	Displays	Frequency, phase angle, and position of forward strut of oscillator and angle of tilt table indicated by readouts in Penthouse. Digital: 16 bit (user selectable units) Accuracy of frequency: +/- 0.01 radian per second root-mean-squared. Accuracy of tilt table angle: +/-0.03 degrees root-mean-squared. Analog: -10 to +10 volts Accuracy of frequency: frequency to voltage conversion +/-0.03 volt root-mean-squared. Accuracy of tilt table angle: 16 bit A/D with +/-0.06 volt root-mean-squared.
	Drive Shaft	Position of oscillator motor read by optical encoder with a minimum of 4096 steps per revolution.
	Status Reports	Top Dead center of drive shaft.
Interfaces	Operator Function	Remote start of tilt table and oscillator with keystrokes from PC in Penthouse.
	Control Software	Interface to FORTRAN program. Control commands data to and from PC in Penthouse via RS 232 or IEEE 485 interface.

Local Control	Control Pendant	Hand held box with switches and readout to control tilt table and read its angle with "home" button to bring respective functions to zero.
Safety	Lockout	Lockout on local control pendant to prevent remote operation while working on model hardware.
	Limit Switches	As needed to prevent damage to any hardware from over travel.
	Hard Stops	As needed to prevent damage to any hardware from over travel.
	Tilt Table Lock	Lock table to A-frame structure at any selected angle of attack.
	Work Platform	Needed on A-frame and table.
Lifting Hardware		Padeyes, slings, and strongbacks required to balance apparatus for ease of carriage installation.

The new PMM will be a twin strut mechanism arranged to conduct experiments on a wide range of submerged vehicles and submerged bodies on Towing Carriage 2. It will be used to tow models at various angles of attack and angles of drift and to oscillate models in pure heaving and pure pitching over a range of towing speeds and oscillation frequencies. The apparatus will be designed to have a fixed amplitude for oscillation. The complexity and cost of the PMM would have increased significantly with no important advantage of having a variable amplitude.

The apparatus will be rigged to the towing carriage by making use of the dry dock located at the east end of the towing basin. It is desirable for the struts to clear the top of the dry dock door by raising the A-frame on the vertical rails located at the front of the towing carriage. The apparatus should be designed to be placed in the existing storage cradle. A sketch of a preliminary design for the new apparatus is shown in Figure 2. A block diagram of the control electronics is shown in Figure 3.

The spatial envelope constraints imposed on the design of the apparatus by the physical arrangement of the towing basin, Towing Carriage 2, and the existing struts are shown in Figure 4. The spatial limits are for a 20-foot by 4-foot by 4-foot model attached to two 10-foot long by 1-foot in chord struts whose longitudinal axes are 10 feet apart. The envelope is shown with the model at zero degrees angle of attack and with an angle of attack of +/-22 degrees.

Calculations indicate that a 12-foot long tilt table appears to accommodate the maximum strut separation of 10 feet. If the pivot axis is located 7.8 feet from the carriage rails, then there would be a clearance of about 0.5 foot between the lower end of the rail and the strut when the model is at an angle of 22 degrees toward the carriage. There would be about 6 feet of vertical clearance above the table when the tilt table is at an angle of 22 degrees away from the carriage. When the tilt table is at zero degrees, it can be raised high enough so that the upper surface of the model is out of the water so that the trim angle can be checked. The tilt table can be raised so that the sternplanes can be adjusted from the towing carriage catwalk.

It is anticipated that the apparatus would be constructed mostly of steel with a yield strength of 36,000 pounds per square inch and an ultimate strength of 58,000 pounds per square inch. The allowable stresses in tension, shear, and bending would be about 0.45, 0.30, and 0.50 times the yield strength, respectively.

A contractor has developed a preliminary detailed design for the apparatus. It is anticipated that the detailed design will be completed by August 1995 and construction will then begin.

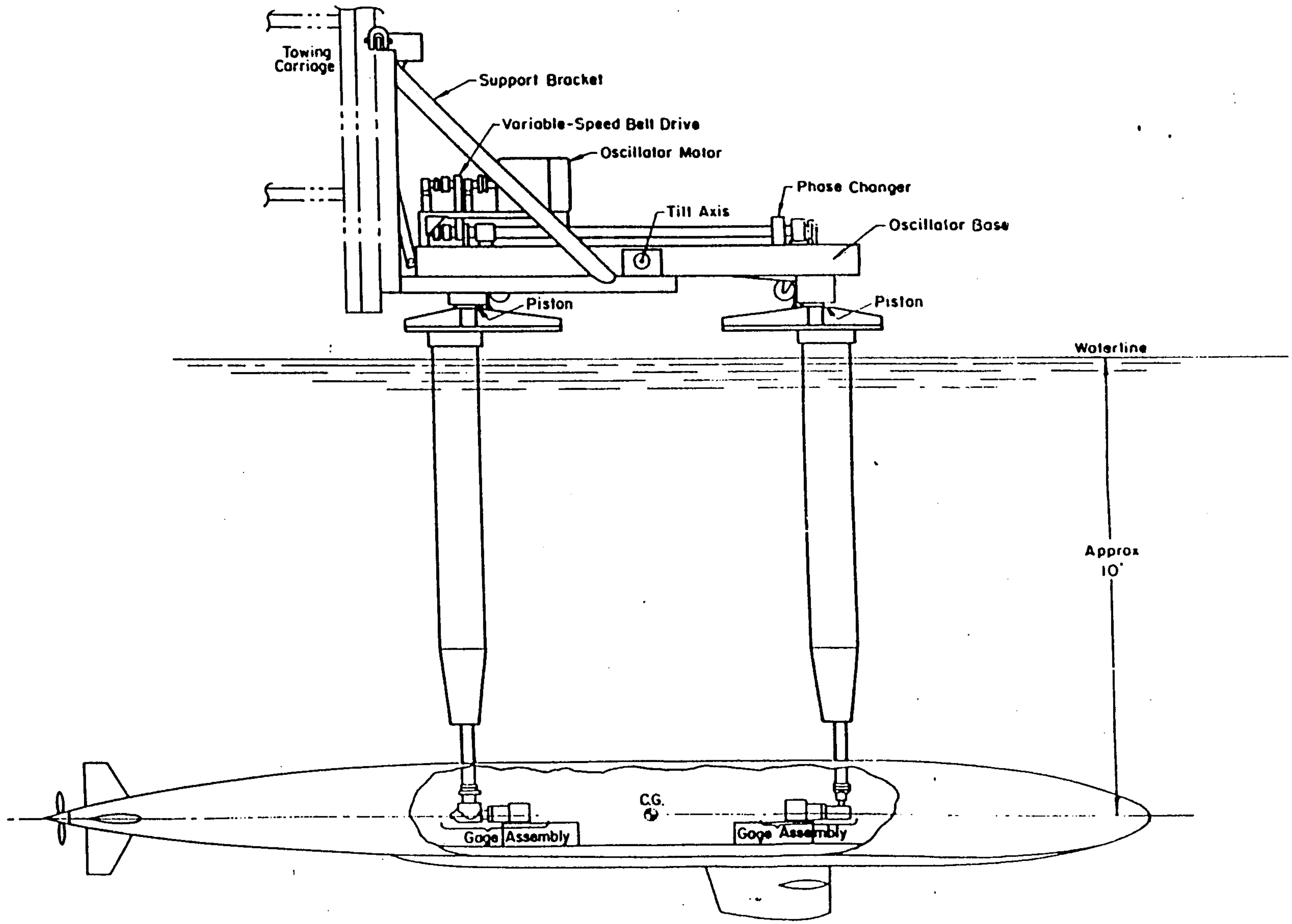

Fig. 1. Sketch of the existing Planar Motion Mechanism with a model attached.

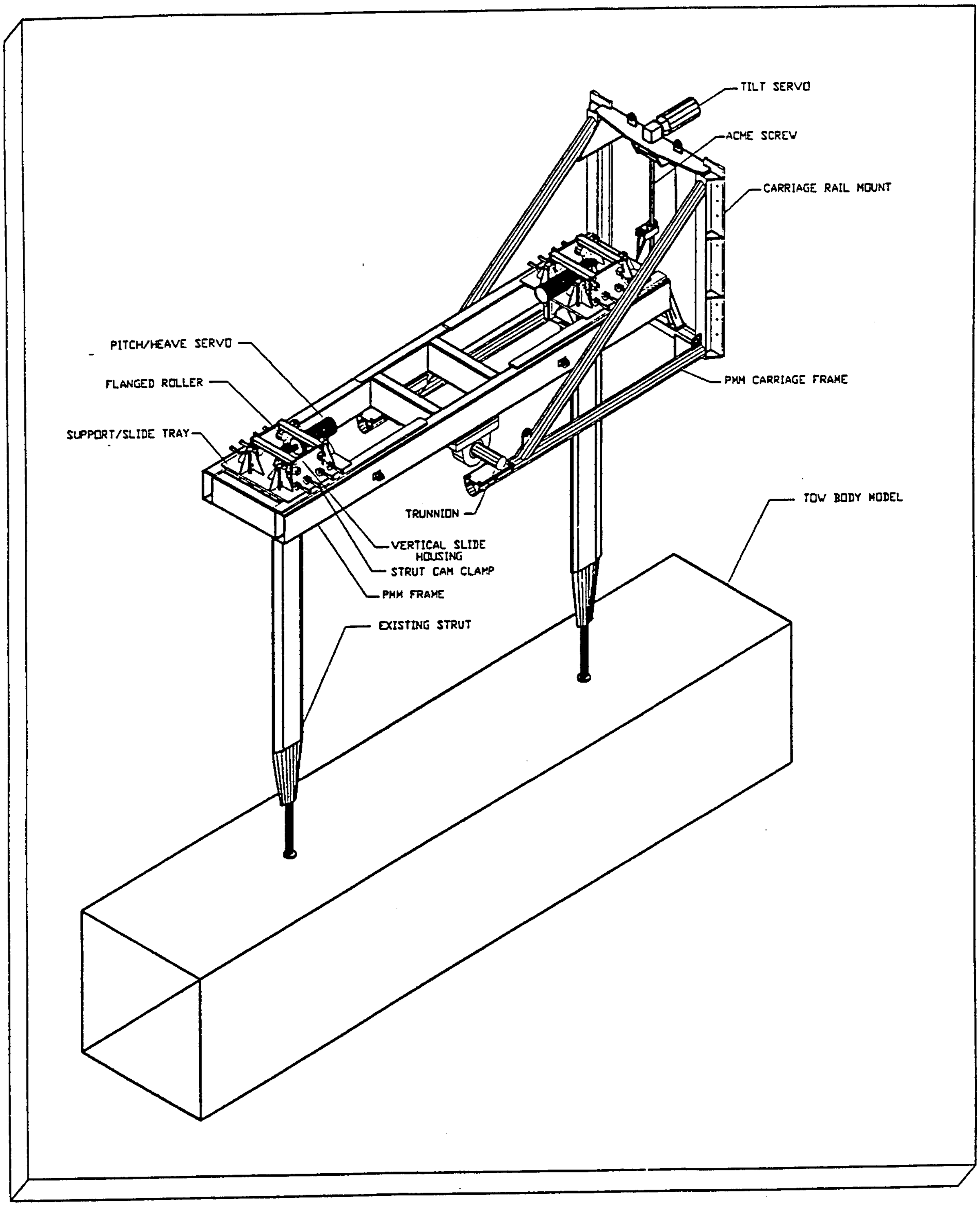

Fig. 2. Sketch of a preliminary design of a new Planar Motion Mechanism.

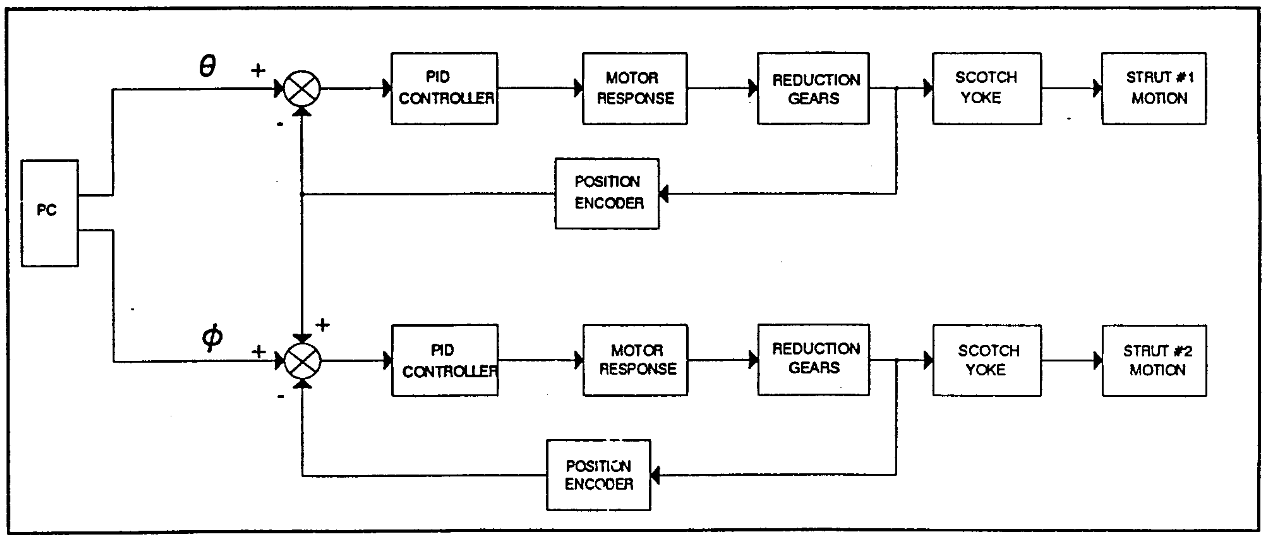

Fig. 3. Control block diagram for the new Planar Motion Mechanism.

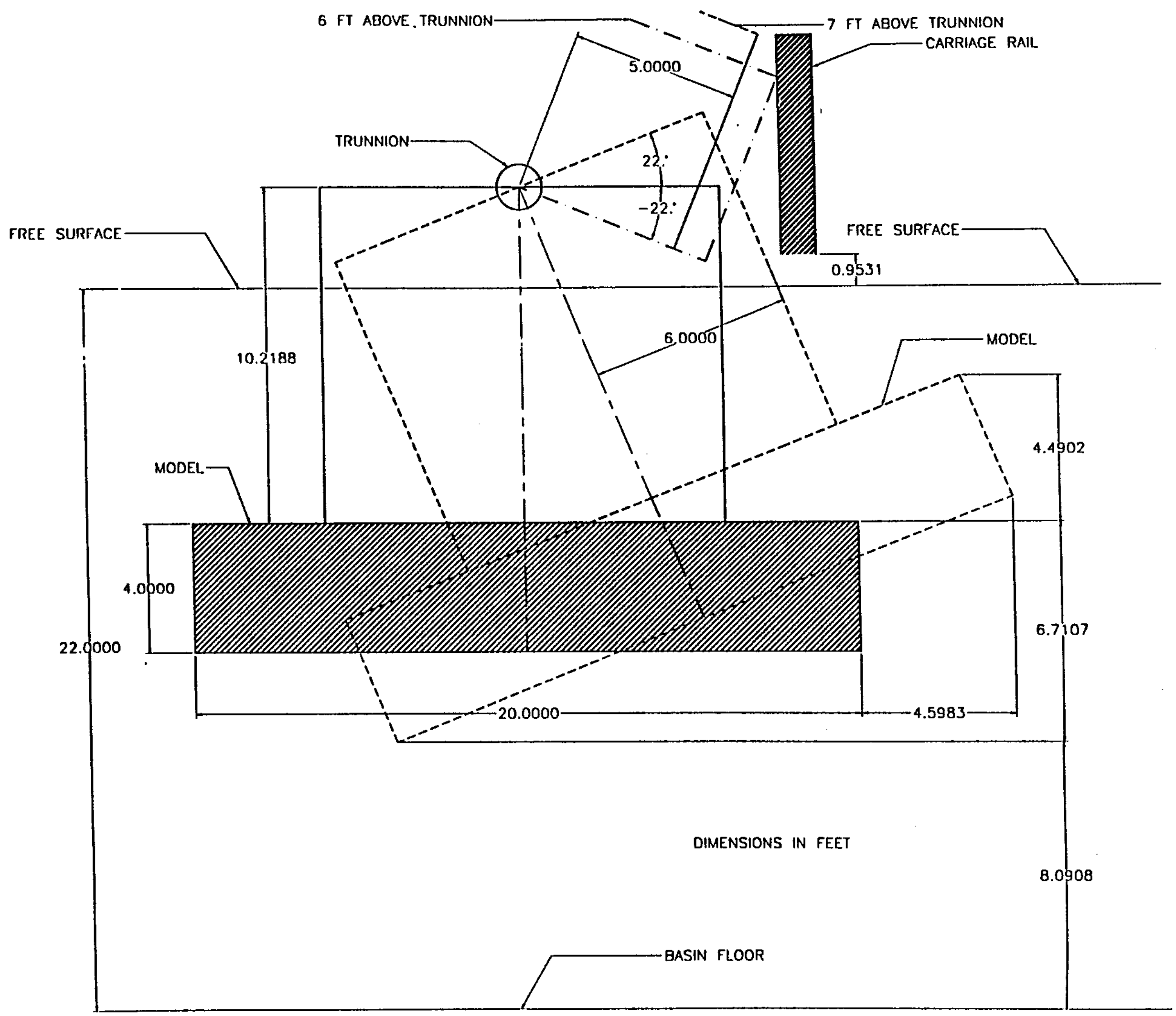

Fig. 4. Space envelope with a 22-degree tilt table angle for the new Planar Motion Mechanism.

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