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BAKER MANUFACTURING COMPANY
Evansville, Wisconsin

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ENGINEERING REPORT NO. 248

to the

OFFICE OF NAVAL RESEARCH

on

THE DESIGN OF HYDROFOIL BOATS
WITH PARTICULAR REFERENCE TO
OPTIMUM CONDITIONS FOR
OPERATION IN WAVES

by

J. G. BAKER
Project Director

29 JULY 1960

ON 10-1960

REF ID: A6460

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In the development of *High Pockets* and *High Tail*, the contractor is indebted to many Navy personnel having contact with the projects for helpful advice and criticism. Some of these are: Patrick Leehey, CDR, USN; Professor J. A. Duffie (now Director of Solar Energy Laboratory, University of Wisconsin); J. J. Stilwell, Capt., USN; Eugene P. Clement and R. B. Couch, David Taylor Model Basin; Robert J. Johnston (now President of Miami Shipbuilding Corporation); Robert E. Apple, CDR, USN; J. E. Hammerstone, CDR, USN; and John Bader, Preliminary Design, USN, Bureau of Ships.

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The gathering and sorting of ideas, the analysis, design, fabrication and testing of both *High Pockets* and *High Tail* were guided by engineering conferences among contractor personnel on the project involving most often Maxwell L. Palmer, Project Engineer, and Neil C. Lien, Design Engineer, on engineering questions, and Daryl Hagen, Raymond Custer and the late George Buehler, tool makers, on fabrication problems.

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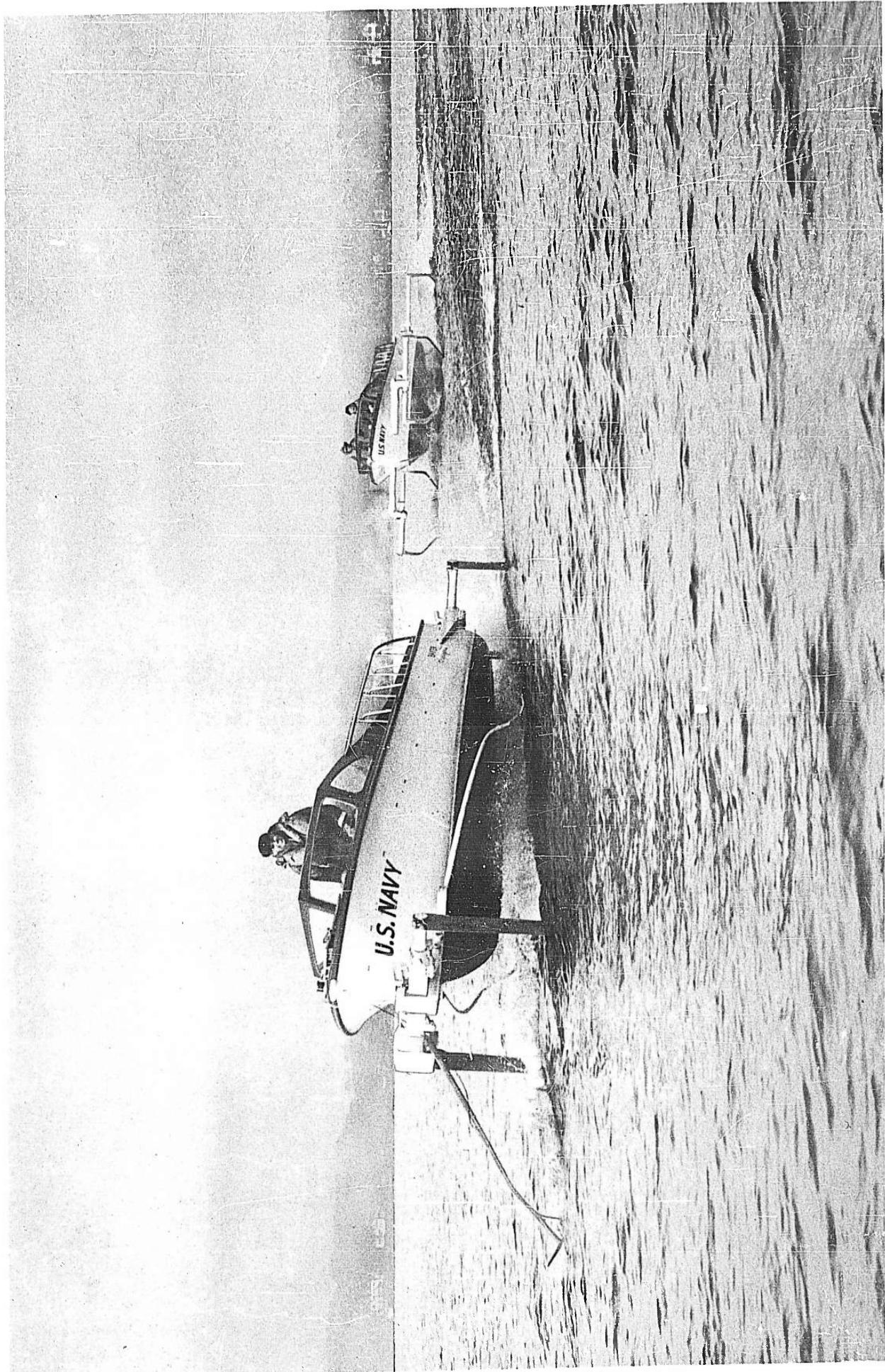


PHOTO 1 BOW VIEW OF HIGH TAIL FOLLOWED BY HIGH POCKET, BOTH FLYING

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SUMMARY

Under the Reference 1 contract, two 24-foot hydrofoil model test boats were designed and constructed on the basis of long term studies by the contractor prior to and under the Reference 1 contract. The first, known as *High Pockets*, with a surface piercing hydrofoil system, reported on in Reference 5, is suitable for Hydrofoil Froude Numbers below about 0.5. The second, known as *High Tail*, with an angle controlled, fully submerged hydrofoil system, is suitable for Hydrofoil Froude Numbers above about 0.5.

The maximum Lift-Drag ratio, with the propeller shaft down in flying position, but with no propeller, measured by towing, is 15 for *High Pockets* and 14.3 for *High Tail*.

High Tail has three hydrofoils, one forward and two aft, three mechanical sensors, one touching the water surface forward of each hydrofoil, and a water propeller driven by an inboard marine engine. The forward hydrofoil and mounting struts are rotated about an upright axis for steering while flying. Hydrofoils, sensors and propeller are all hydraulically retractable.

The contractor suggested both mechanical and electrical hydrofoil angle controls including both mechanical and electronic water surface sensing. The Office of Naval Research specified mechanical control for *High Tail*.

The functional form of the hydrofoil angle control was synthesized mathematically and expressed by Specific Coupling Equations. Important control parameters were selected to minimize the Hydrofoil Draft Increment for a given forward speed and a given limiting vertical acceleration on the basis of a mathematical analysis of *High Tail*'s vertical motions controlled in accordance with the Specific Coupling Equations while flying in waves. Minimum Hydrofoil Draft Increment is important not only to enlarge the areas of practical operation, but also to save weight and drag.

In the design of *High Tail*'s actual control, the coupling relationships expressed by the Specific Coupling Equations, except for the smoothing terms, are produced by a mechanical computer. Three output signals from the computer command three hydraulic servos which introduce smoothing effects and actuate the angular control movements of the hydrofoils. Other output signals control unfolding and folding of the sensors, start and stop the control and indicate the flying elevation at the control station.

The input signals to the computer are:

- 1) the displacements of the three water surface sensors;

- 2) the steering angle of the forward hydrofoil;
- 3) the smoothed hydraulic pressure in the hydrofoil actuating cylinders; and
- 4) the four manual control station adjustments of which three are for pay load, elevation, and roll trimming, and the fourth is for sea condition.

High Tail automatically flies at a constant mean altitude on a straight course and banks and lowers in turns. As *High Tail* accelerates from rest (decelerates from speed) the sensors unfold (fold), the angle control starts (stops), and the hull abruptly climbs to (descends from) the flying elevation all automatically, at predetermined boat speeds.

The performance of *High Tail*'s hydrofoil control during preliminary tests as indicated by movies, Reference 6, and direct observation before installation of recording instruments, appears to be about as predicted in the mathematical analysis.

Tests and test apparatus have been proposed by the contractor for simultaneously recording on board *High Tail* during flight in waves: the vertical accelerations of the boat, the relative position of the boat with respect to the water surface, and the wave surface profile traversed, as determined from sensor movements and vertical boat accelerations, by means of special computers for the purpose.

DEFINITIONS

Bottom Clearance - the amount by which the elevation of the Flying Bottom Plane in flight exceeds the elevation of the water surface at a specific location with respect to the boat and at a specific instant of time.

Bow Computer Section - the portion of *High Tail*'s computer located in the bow compartment forward of the control station.

Cage Speed - the boat speed during deceleration at which the control mechanism is automatically caged and the hydrofoil angle control stops leaving the hydrofoils at predetermined angle settings.

Contour Allowance - the average Bottom Clearance at the hydrofoils allowed in the design to provide adequate Bottom Clearance between the foils when flying over wave crests.

Displacement Ratio - the ratio of the fully loaded displacement of a prototype to that of a model, the model being *High Pockets* or *High Tail*.

Down Speed - the boat speed during deceleration at which the boat abruptly descends from the flying elevation.

Dynamic Pressure - the dynamic pressure of the water flow with respect to the boat.

End Plate Draft Increment - the greatest distance any hydrofoil end plate extends below the Hydrofoil Plane.

Floating Bottom Plane - the plane that is parallel to the load water line and tangent to the hull bottom.

Flying Bottom Plane - the plane that is parallel to the design flying water level and tangent to the hull bottom.

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Flying Draft - the amount by which the elevation of the water surface over a hydrofoil exceeds the elevation of the Hydrofoil Plane at the Hydrofoil.

Flying Draft Variance - the range of variation of the Flying Draft under given operating conditions.

Forward Computer Section - the portion of *High Tail's* computer mounted with the forward hydrofoil.

Full Banking - a roll angle inward on a turn such that the vector sum of the centrifugal force and boat weight is perpendicular to the boat beam.

Function Unit - a control component that produces a signal that is a prescribed nonlinear function of the signal it receives.

Hull Draft - the difference in elevation of the load water line and the Floating Bottom Plane.

Hydrofoil Draft Increment - the distance the Hydrofoil Plane is below the Floating Bottom Plane, for the hydrofoil for which this distance is the greatest.

Hydrofoil Froude Number - the acceleration of gravity multiplied by the fore and aft spacing of the hydrofoils and divided by the square of the forward speed of the boat.

Hydrofoil Plane - the plane that is parallel to the load water line and tangent to the bottom of a hydrofoil when the hydrofoil chord is set parallel to the design flying water level.

Lift Draft - the minimum Flying Draft required to produce the hydrofoil lift necessary to fly.

Lower Range Sea Condition - a sea condition and wave velocity direction that would result in a Flying Draft Variance for *High Tail* of two feet or less.

Planing Point - a point on a sensor ski at the intersection of the planing surface of the ski and the water surface while the boat is flying in smooth water.

Propeller Draft Increment - the amount by which the propeller and its protection, if any, extended ready for flight, extend below the lowest Hydrofoil Plane.

Pump Pressure - the discharge pressure of the pump that delivers fluid to *High Tail's* hydraulic system.

Reciprocal Function Unit - a Function Unit that produces a signal that is the reciprocal of the signal it receives.

Scale Ratio - the ratio of the fore and aft spacing of the hydrofoils of a prototype to that of a model, the model being *High Pockets* or *High Tail*.

Sensor Down Speed - the speed during acceleration at which the sensors are automatically lowered to the water surface.

Sensor Lead - the horizontal distance by which the Planing Point of a sensor ski leads the following hydrofoil.

Sensor Loading Pressure - the hydraulic pressure in any cylinder that applies torque to a sensor mounting shaft to load the sensor ski.

Sensor Radius - the radial distance from a sensor mounting shaft axis to the Planing Point of a sensor ski.

Sensor Up Speed - the speed during deceleration at which the sensors are automatically retracted.

Sequence Function Unit - a Function Unit that receives the smoothed Servo Pressure as a signal and produces pilot pressure changes for automatic control of the boat during acceleration and deceleration.

Servo Pressure - the pressure in the servo cylinders actuating the rotation for control purposes of *High Tail's* forward hydrofoil unless otherwise specified or implied.

Sine Function Unit - a Function Unit that produces a

signal that is the sine of the signal received.

Smoothed Servo Pressure - the output signal of a Smoothing Function Unit that receives the Servo Pressure as an input signal.

Smoothing Function Unit - a Function Unit that produces a signal that is the same as the signal received except for slight delay and reduced perturbations.

Specific Coupling Equations - the Equations (12), (13) and (14) expressing the coupling relations among the signals on which *High Tail's* hydrofoil angle control depends and the angle settings of the hydrofoils.

Steering Function Unit - a Function Unit that receives a signal proportional to the steering angle and produces a signal for automatically lowering the boat or a Function Unit that produces signals for automatically lowering and banking the boat.

Stern Computer Section - the portion of *High Tail's* computer located in the stern compartment just forward of the transom.

Total Draft - the sum of the Hull Draft and the Hydrofoil Draft Increment and either the Propeller Draft Increment or the End Plate Draft Increment whichever is largest.

Tripping - the sudden submergence and downward rotation of a pivoted forward extending sensor about its pivot axis due to the reversal of moment on the sensor because of submergence.

Uncage Speed - the boat speed during acceleration at which the control mechanism is automatically uncaged and the hydrofoil angle control commences operation.

Up Speed - the forward boat speed during acceleration at which the boat abruptly climbs to the flying elevation.

Upper Range Sea Condition - a sea condition and wave velocity direction that would result in a Flying Draft Variance for *High Tail* of more than two feet.

SYMBOLS

$A = \text{control coupling factor relating } \frac{\delta}{1} \text{ to a hydrofoil angle setting}$

$A_a = A$ for an aft hydrofoil or hydrofoils selected on the basis of wave length and the wave train velocity direction with respect to the boat course (Dimensionless)

$A_{an} = \frac{b}{(b+1)} - \frac{4w_f(v+c)^2}{v^2\rho Kg(b-1)} = A$ for an aft hydrofoil for zero Flying Draft Variance in long waves

$A_e = \text{control coupling term which is zero for any boat speed up to Up Speed during acceleration and below Down Speed during deceleration and a constant at higher speeds equal to the change in elevation introduced by the control either at Up Speed during acceleration or at Down Speed during deceleration (ft)}$

A_f	= A for a forward hydrofoil or hydrofoils selected on the basis of wave length and the wave train velocity direction with respect to the boat course (Dimensionless)	d	= Flying Draft Variance = $2(h-y)_0$ (ft)
A_{fn}	= $\frac{b}{(b+1)} + \frac{4w_f(v+c)^2}{v^2 2\rho K g(b+1)}$ = A for a forward hydrofoil for zero Flying Draft Variance in long waves	d_a	= d for aft hydrofoil = $2(h_a-y_a)_0$ (ft)
A_r	= a constant corresponding to the moment or movement tending to restore a displacement in roll (Dimensionless)	d_f	= d for forward hydrofoil = $2(h_f-y_f)_0$ (ft)
a	= width of sensor ski (ft)	d_m	= $d_{rm} + d_n$ (ft)
B_f, B_{ar} and B_{al}	= Control terms in Specific Coupling Equations (12), (13) and (14) representing manual adjustments at the control station for pay load on or trim of the forward, right aft and left aft hydrofoils respectively (Dimensionless)	d_n	= Lift Draft (ft)
b	= longitudinal spacing of hydrofoils (ft)	d_p	= smooth water draft of sensor ski measured from water surface at the ski (ft)
b_w	= lateral spacing of sensors (ft)	d_r	= Flying Draft Variance required to limit the vertical acceleration to a specific tolerable magnitude while flying in a sinusoidal sea of a given wave length with the flight path and wave profile in phase
C	= wave train velocity = $2.265 \sqrt{\lambda}$ (ft/sec)	d_{rm}	= d_r for the wave length λ_m , at which d_r is maximum
C_d	= dimensionless damping parameter = $\frac{C_s v}{K_s b}$	d_s	= $\delta + h - y$ = instantaneous dynamic increase in sensor ski draft over d_p (ft)
	or in the case of a sensor = $\frac{d_p}{a_p b}$	E	= smoothing parameter in the Specific Coupling Equations which depends on the ratio of the rate of fluid flow through the servo valve to the difference between the hydrofoil angle command signal and the hydrofoil angle setting
C_f and C_a	= positive constants of proportionality in Equations (15) and (16) determined by the dimensions of the control mechanism and the ratio Q/q (ft)	E_d	= $\frac{Ev}{b}$ dimensionless smoothing parameter
C_s	= sensor signal filter damping parameter or damping constant for sensor motion at the Planing Point (Diagrams I & II) with respect to water surface, in which case C_s also equals $K_p a_v d_p$ (lb sec.) / ft	e	= distance from aft hydrofoil or hydrofoils to center of gravity (ft)
$C_1, C_2, C_3 \dots$	= dimensionless positive constants of proportionality defined in text	F	= $\frac{g b}{v^2}$ = Hydrofoil Froude Number
c	= wave train velocity component opposite boat velocity as determined along the boat course (ft/sec)	F_a	= upward hydrodynamic force on aft hydrofoil or hydrofoils less force due to weight (lbs)
		F_f	= upward hydrodynamic force on forward hydrofoil or hydrofoils less force due to weight (lbs)
		g	= acceleration of gravity (ft/sec ²)
		h	= distance the instantaneous level of a point fixed to the boat hull is below the mean water level (ft)
		h_1, h_2, h_f, h_a	= h at the forward sensor, h at the aft sensor, h at the forward hydrofoil, h at the aft hydrofoil respectively (ft)

\ddot{h}_t	= tolerable limit on the vertical acceleration (ft/sec ²)	m_d	$= \frac{m_s v^2}{K_s b^2}$ dimensionless mass parameter depending on the inertia element in a sensor signal filter or on the effective mass m_s of the sensor in which case m_d also equals $\frac{m_s}{K_p a_p b^2}$
K	= ratio of an increment in coefficient of lift divided by the corresponding increment in angle of attack of a hydrofoil (Dimensionless)	m_s	$=$ effective mass of sensor at the Planing Point (Diagrams I and II) assuming that the vertical acceleration of the hull is negligible in comparison to the vertical acceleration of the Planing Point, or sensor signal filter inertia parameter $(\frac{\text{lbs sec}^2}{\text{ft}})$
K_a	= constant of proportionality between the square of the steering angle θ and a signal in the Specific Coupling Equations for the aft hydrofoil or hydrofoils which lowers the boat for either direction of turn (ft)	N	= upward hydrodynamic moment on sensor about the sensor shaft (Diagrams I and II) divided by r_s (lbs)
K_f	= constant of proportionality between the square of the steering angle θ and a signal in the Specific Coupling Equation for the forward hydrofoil or hydrofoils which lowers the boat for either direction of turn (ft)	n	= sensor downward loading coefficient equal to sensor loading moment divided by v^2 $(\frac{\text{lbs sec}^2}{\text{ft}})$
K_p	= constant of proportionality defined by Equation (50) $(\frac{\text{lbs t}^2}{\text{ft}^4})$	P	= propeller thrust (lbs)
K_s	= sensor signal filter parameter or stiffness constant for sensor motion at the Planing Point (Diagrams I and II) with respect to the water surface in which case K_s also equals $K_p a v^2 a_p$ (lbs/ft)	p	= differential operator $\frac{d}{dt}$ (1/sec)
K_Θ	= constant of proportionality between the steering angle θ and a roll signal for the rear hydrofoils in the Specific Coupling Equations (12), (13) and (14) (ft)	P_d	= differential operator $\frac{d}{dr} = \frac{b}{v} p$ (Dimensionless)
l	= Sensor Lead, i.e., the horizontal distance by which the Planing Point of a sensor ski leads the following hydrofoil (ft)	Q	= Smoothed Servo Pressure (lbs/ft ²)
l	= One	q	= the Dynamic Pressure, i.e., the dynamic pressure of water flow with respect to boat (lbs/ft ²)
M	= mass of loaded boat $(\frac{\text{lbs sec}^2}{\text{ft}})$	R	= the wave length-amplitude ratio, $\frac{\lambda}{y_0}$ (Dimensionless)
M_a	$= \frac{2M(b-e)}{S_a \rho K b^2} = \frac{2w_a}{g \rho K b^2}$ dimensionless hydrofoil loading parameter for the aft hydrofoil or hydrofoils	r	= radius of gyration of loaded boat (ft)
M_f	$= \frac{2Me}{S_f \rho K b^2} = \frac{2w_v}{g \rho K b^2}$ dimensionless hydrofoil loading parameter for the forward hydrofoil or hydrofoils	r_f, r_a	= push rod pivot radius from hydrofoil pivot axis for forward and aft hydrofoils respectively (ft)
		r_s	= sensor radius, see Diagrams I and II (ft)
		S	= surface area of a hydrofoil (ft ²)

S_f, S_a = surface area of forward hydrofoil or hydrofoils and the surface area of the aft hydrofoil or hydrofoils respectively (ft^2)

T = $\frac{2\pi\lambda}{\cos\beta(v+c)}$ = time required for boat to travel one wave length (sec)

t = time (sec)

v = the horizontal velocity of the boat or hydrofoils (ft/sec)

w = weight carried per unit area of hydrofoil (lbs/ft^2)

w_f, w_a = w for the forward hydrofoil or hydrofoils and w for aft hydrofoil or hydrofoils respectively (lbs/ft^2)

w_{fh}, w_{ah} = w_f, w_a for a pay load one-half full pay load (lbs/ft^2)

x = distance measured in the direction of the boat velocity from a vertical line moving with the wave train (ft)

x_1, x_2, x_f, x_a = the x for the front sensor, the x for the aft sensor, the x for the forward hydrofoil or hydrofoils, the x for the aft hydrofoil or hydrofoils respectively (ft)

$$y_1 = \left[\frac{\lambda(1+C_d p_d) A_f}{l(1+C_d p_d + m_d p_d^2)(1+E_d p_d)} \right] \frac{y_1}{\lambda} + \left[\frac{\lambda c}{b(v+c)} p_d \right] \frac{y_f}{\lambda}$$

$$y_2 = \left[\frac{\lambda(1+C_d p_d) A_a}{l(1+C_d p_d + m_d p_d^2)(1+E_d p_d)} \right] \frac{y_2}{\lambda} + \left[\frac{\lambda c}{b(v+c)} p_d \right] \frac{y_f}{\lambda}$$

y = instantaneous distance to the wave surface measured downward from the mean water level (ft)

y_1, y_2, y_f, y_a = the y for the front sensor, the y for the aft sensor, the y for the forward hydrofoil or hydrofoils, the y for the aft hydrofoil or hydrofoils respectively (ft)

$$Z_3 = \left[\frac{(l+b)(1+C_d p_d) A_f}{l(1+C_d p_d + m_d p_d^2)(1+E_d p_d)} - l+p_d+M_f \left(\frac{e}{b} + \frac{r^2}{be} \right) p_d^2 \right]$$

$$Z_4 = - \left[\frac{(l+C_d p_d) A_f}{(l+C_d p_d + m_d p_d^2)(1+E_d p_d)} - l-M_f \left(\frac{b-e}{b} - \frac{r^2}{be} \right) p_d^2 \right]$$

$$Z_5 = \left[\frac{(l+C_d p_d) A_a}{(l+C_d p_d + m_d p_d^2)(1+E_d p_d)} - l+M_a \left(\frac{e}{b} - \frac{r^2}{b(b-e)} \right) p_d^2 \right]$$

$$Z_6 = - \left[\frac{(l-b)(1+C_d p_d) A_a}{l(l+C_d p_d + m_d p_d^2)(1+E_d p_d)} - l-p_d-M_a \left(\frac{b-e}{b} + \frac{r^2}{b(b-e)} \right) p_d^2 \right]$$

α = angle setting of a hydrofoil with respect to the boat measured from the angle of zero lift in the sense that an increase in α produces an increase in upward force on the hydrofoil or an increase in upward velocity of the hydrofoil

α_a = α for aft hydrofoil or if there are two aft hydrofoils the average α for the two which is equal to:

$$\frac{\alpha_{ar} + \alpha_{al}}{2}$$

α_{al} = α for aft hydrofoil on left

α_{ar} = α for aft hydrofoil on right

α_f = α for forward hydrofoil or if there are two forward hydrofoils the average α for the two which is equal to:

$$\frac{\alpha_{fr} + \alpha_{fl}}{2}$$

α_{fl} = α for forward hydrofoil on left

α_{fr}	= α for forward hydrofoil on right	θ_c	= θ_n when $\theta > \theta_n$
α_p	= trim angle of sensor planing surface with respect to the water surface	θ_c	= $-\theta_n$ when $\theta < -\theta_n$
$\bar{\alpha}_f$, $\bar{\alpha}_a$, etc.	= variation of α , α_f , α_a , etc., respectively from a mean value, i.e., $\bar{\alpha} = \alpha - \frac{w}{Kq}$	$\pm \theta_m$	= specified steering angles beyond which there is no change in any α due to change in θ^2
β	= 180° -- angle between wave velocity and boat velocity	$\pm \theta_n$	= specified steering angles beyond which there is no change in any α due to change in θ
γ	= angle of boat roll with respect to sensing locations on the water surface measured counter-clockwise looking forward	λ	= wave length as determined in the direction of the wave train velocity (ft)
Δ	= Total Displacement of loaded boat (lbs)	λ_f	= the λ in a following sea at which the maximum vertical acceleration with $y = h$ is at the tolerable limit (ft)
δ	= displacement vertically downward of the Planing Point of a sensor ski (ft)	λ_h	= the λ in a head sea at which the maximum vertical acceleration with $y = h$ is at the tolerable limit (ft)
δ_1	= δ for forward sensor or if there are two forward sensors the average of the two or:	λ_m	= λ at which d_r is a maximum = d_{rm} (ft)
	$\frac{\delta_{1r} + \delta_{1l}}{2}$ (ft)	ρ	= mass density of water
δ_{1l}	= δ for forward sensor on left (ft)	$\sigma_{302}, \sigma_{303}, \dots, \sigma_{642}$	= signal displacements in the forward direction of the members designated by the numbers 302, 303 ... 642, respectively (ft)
δ_{1r}	= δ for forward sensor on right (ft)	σ_f, σ_{ar} and σ_{al}	= downward displacement command signals to the forward, right aft and left aft servos respectively
δ_2	= δ for aft sensor or if there are two aft sensors the average of the two or:	σ_h	= output signal of either forward or aft Reciprocal Function Unit which is an increase in length of the Unit (ft)
	$\frac{\delta_{2r} + \delta_{2l}}{2}$ (ft)	t	= tv = dimensionless measure of time
δ_{2l}	= δ for aft sensor on left (ft)	ω	= $\frac{2\pi}{T} = \frac{2\pi \cos\beta(v+c)}{\lambda}$ frequency of wave encounter in radius per second
δ_{2r}	= δ for aft sensor on right (ft)	ω_d	= $\frac{\omega b}{v}$ (Dimensionless)
θ	= steering angle with respect to hull of a forward hydrofoil or hydrofoils measured to the right from center looking forward	A dot over a quantity indicates the derivative of the quantity with respect to time.	
θ_b	= θ when $\theta_m > \theta > -\theta_m$	A subscript zero following a variable or a quantity or added to another subscript indicates the maximum magnitude of the quantity for the wave length and course considered.	
$\dot{\theta}_b$	= $\dot{\theta}_m$ when $\theta > \theta_m$		
$\dot{\theta}_b$	= $-\dot{\theta}_m$ when $\theta < -\theta_m$		
$\dot{\theta}_c$	= $\dot{\theta}$ when $\theta_n > \theta > -\theta_n$		

ASSUMPTIONS

It is assumed:

- a) that all variations are small;
- b) that the horizontal velocity of the boat is constant;
- c) that the horizontal component of the orbital wave velocity is neglectable;
- d) that waves are sinusoidal in shape;
- e) that there are no surface effects on the hydrodynamic characteristics of fully submerged hydrofoils;
- f) that, for purposes of the determination of pitch and heave control signals and coupling parameters for zero Flying Draft Variance in long waves, the Planing Points of all sensor skis follow the water surface;
- g) that there are no interference effects between hydrofoils;
- h) that there is zero time delay between a change in angle of attack of a hydrofoil and the corresponding change in lift; (Reference 15)
- i) that, for the purposes of Appendix C, a forward hydrofoil is located at the center of percussion of the boat, with the center of oscillation at the aft hydrofoils and vice versa;
- j) that, for the purposes of Appendix C, the rate of change of curvature of the wave surface encountered by a hydrofoil boat flying in long waves may be neglected;
- k) that, for the purposes of Appendix D, the stiffness and damping in the connection of any sensor to the hull are both zero;
- l) that the effective trim angle of the sensor ski with respect to the water surface at the ski and the Sensor Lead are constant;
- m) that, for the specific purpose of expressing the vertical force on the sensor in Appendix D, the vertical acceleration of the hull is neglectable compared to that of the sensors;
- n) that the center of mass of the boat is at the same level as the hydrofoils; and,
- o) that the pressure drop, through the valve controlling the hydraulic fluid to the servo cylinders for any hydrofoil, is so large that the rate of flow into or out of the servo cylinders is determined solely by the valve opening; and that the latter is proportional to the difference between the hydrofoil angle command signal and the hydrofoil angle setting.

INTRODUCTION

In the design studies and tests carried out by the contractor other than those reported on herein, it was concluded:

- A. That the drag and vertical acceleration effects for various sizes of hydrofoil boats are best compared quantitatively on the basis of geometrically similar boats and waves with equal Hydrofoil Froude Numbers, provided cavitation at the hydrofoils and hull planing effects are not dominant;
- B. that for low drag, disregarding cavitation effects, the surface piercing type hydrofoil is preferable for Hydrofoil Froude Numbers below about 0.5 and the fully submerged, angle controlled type hydrofoil is preferable for higher Hydrofoil Froude Numbers;
- C. that substantial wake energy can be recovered in a surface piercing hydrofoil system with V-hydrofoils by placing each aft hydrofoil in or nearly in a tracking position with respect to a forward hydrofoil;*
- D. that a dihedral of about 40° is optimum for surface piercing hydrofoils from the standpoints of maneuverability, performance in waves and low drag;
- E. that hydrofoils and struts should be retractable to an above water position when not in use to permit shallow water operation, facilitate docking, increase hydrofoil and strut life, and reduce inspection and maintenance costs;
- F. that large energy absorption in the hydrofoil and propeller mountings is important to reduce the probability of either injury to personnel or damage to the boat, in case a hydrofoil or propeller strikes bottom or an obstacle;
- G. that it is desirable to steer with the bow hydrofoil or hydrofoils to minimize lateral flow at the propeller while in a turn; and to avoid the use of a high speed rudder and the attending drag;
- H. that it is desirable to rotate one or more hydrofoils

*Wake energy recovery on High Pockets is indicated by the fact that the angle setting for the aft hydrofoils is lower than the angle setting for identical design of forward hydrofoils carrying the same load with the same submergence. Wake energy recovery for surface piercing hydrofoils is mentioned in connection with German hydrofoil boats in Reference 12. The theory of Reference 7 indicates that there would not be wake energy recovery with fully submerged hydrofoils with the Hydrofoil Froude Number of High Pockets. Inspection of the wake of a high dihedral, surface piercing hydrofoil indicates that the dominant flow direction is horizontal instead of vertical as in the case of a zero dihedral, high aspect ratio, fully submerged hydrofoil. It is believed that a much shorter wake wave length for horizontal flow as compared to vertical flow accounts for the difference in surface piercing and fully submerged Hydrofoil Froude Numbers required for wake energy recovery.

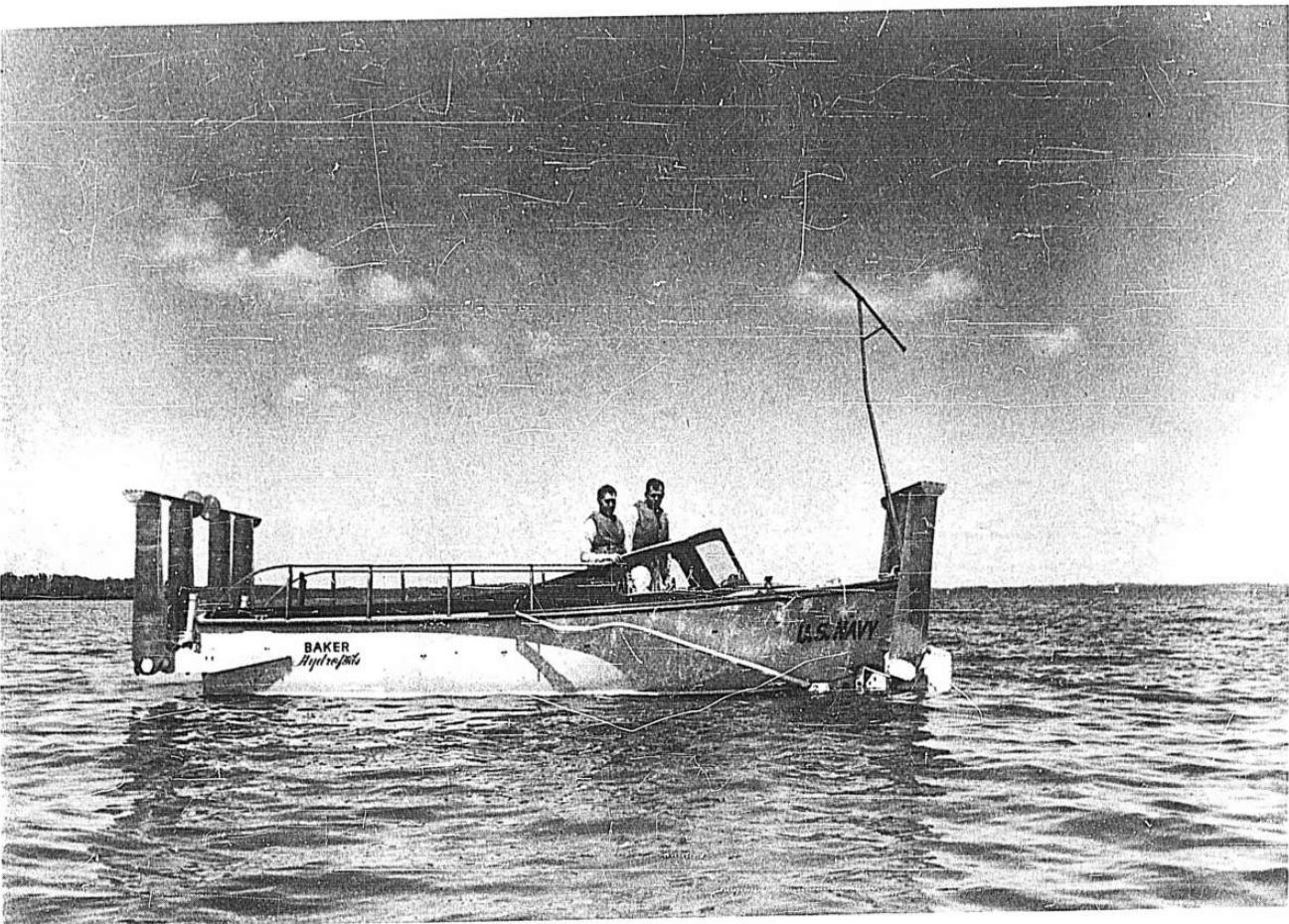


PHOTO 2 STATIONARY - HYDROFOILS, PROPELLER AND SENSORS RETRACTED

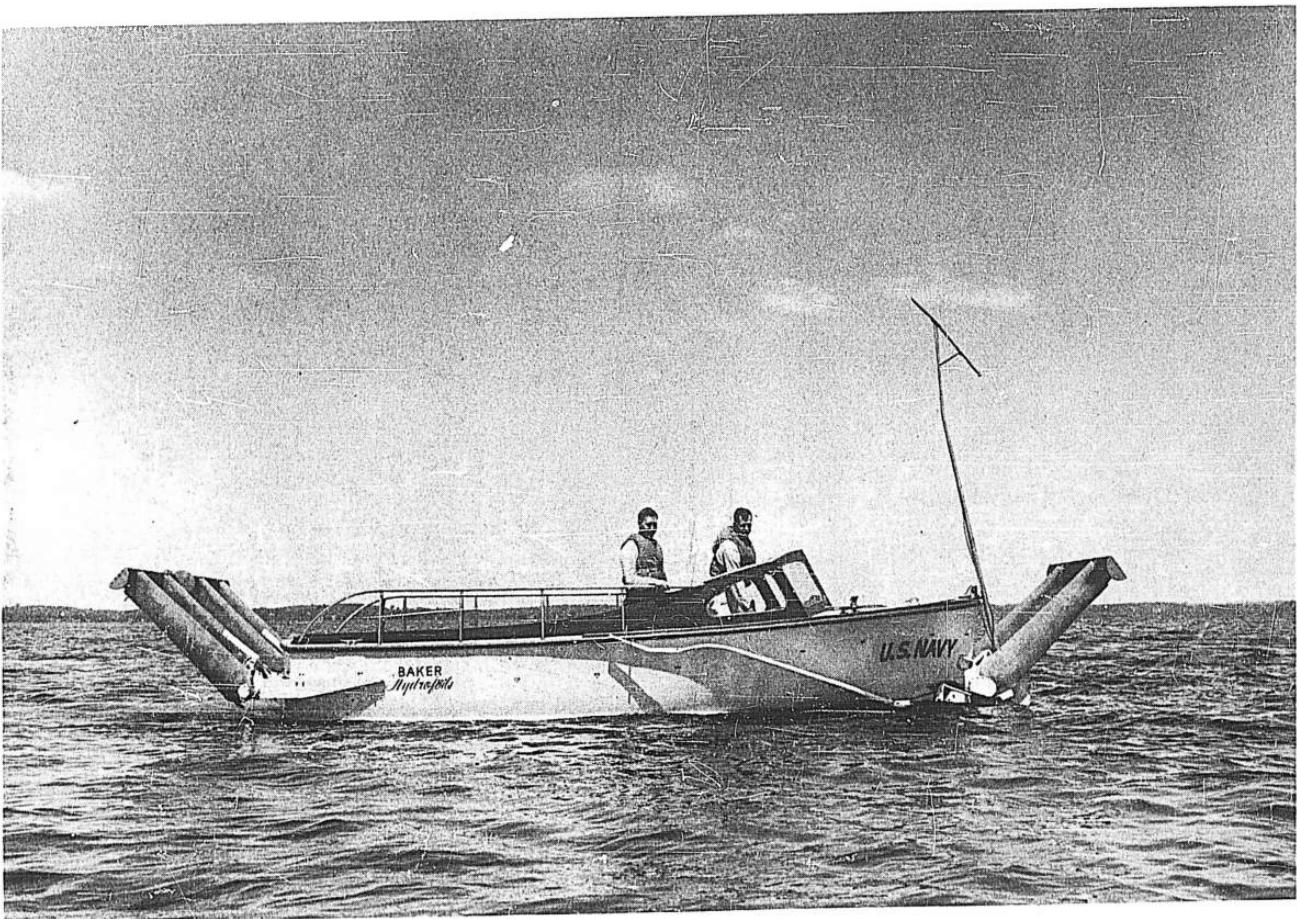


PHOTO 3 STATIONARY - SENSORS RETRACTED, HYDROFOILS PARTIALLY EXTENDED

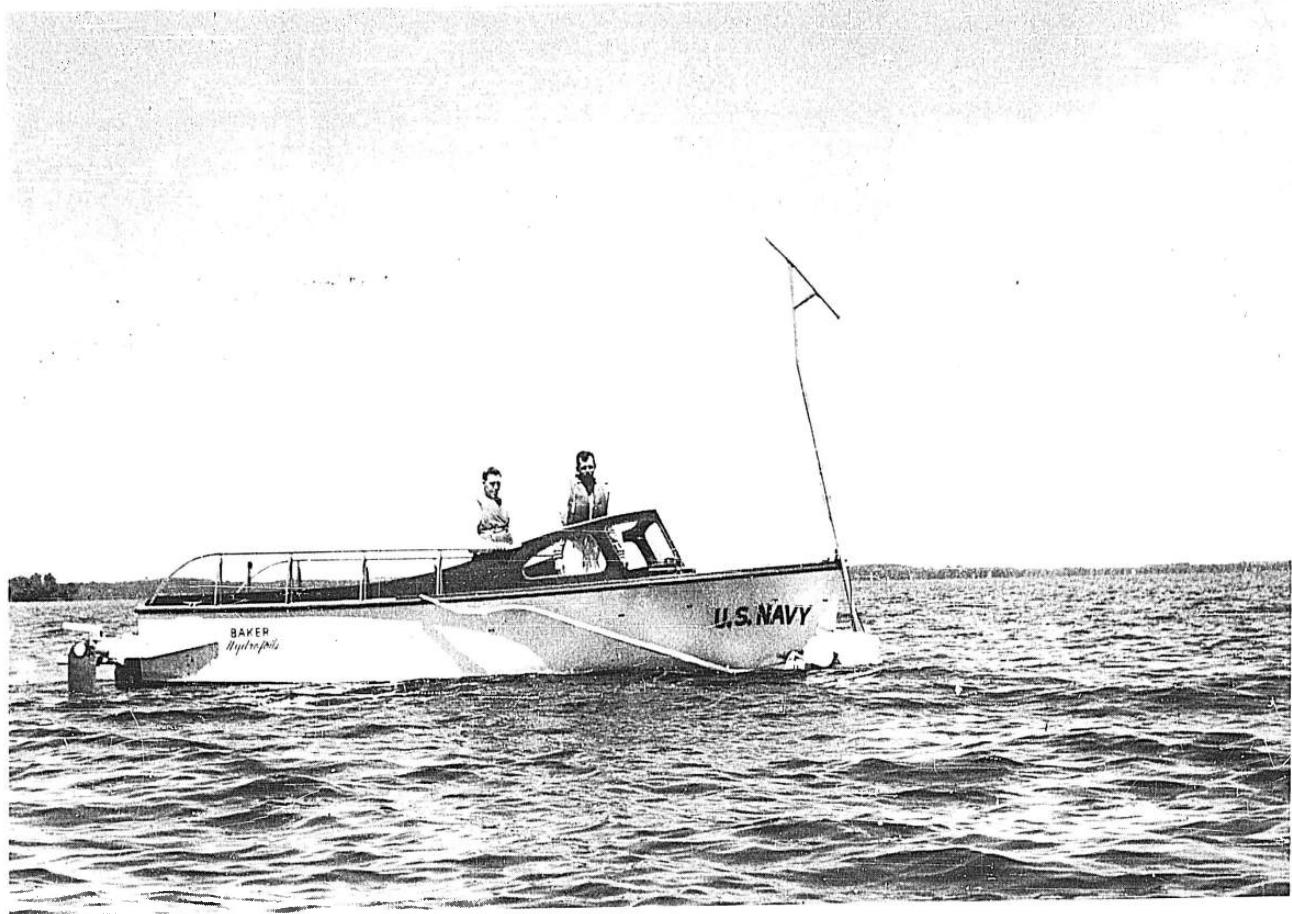


PHOTO 4 STATIONARY - SENSORS RETRACTED, HYDROFOILS EXTENDED

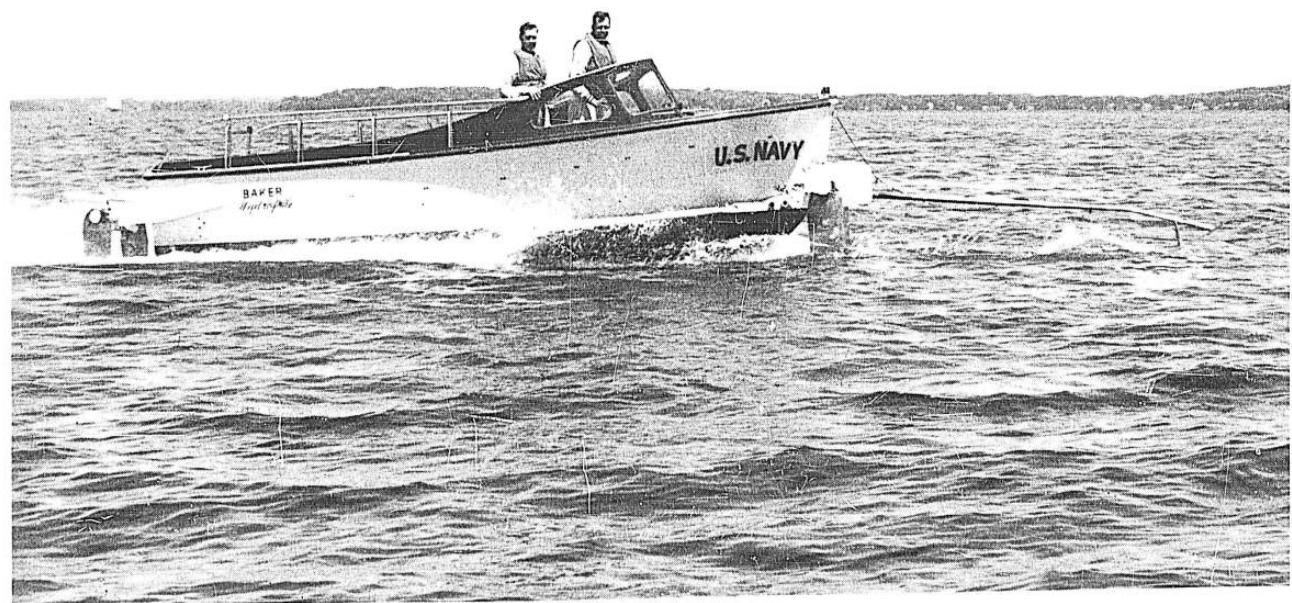


PHOTO 5 LOW SPEED - HYDROFOILS EXTENDED, SENSORS OPERATING

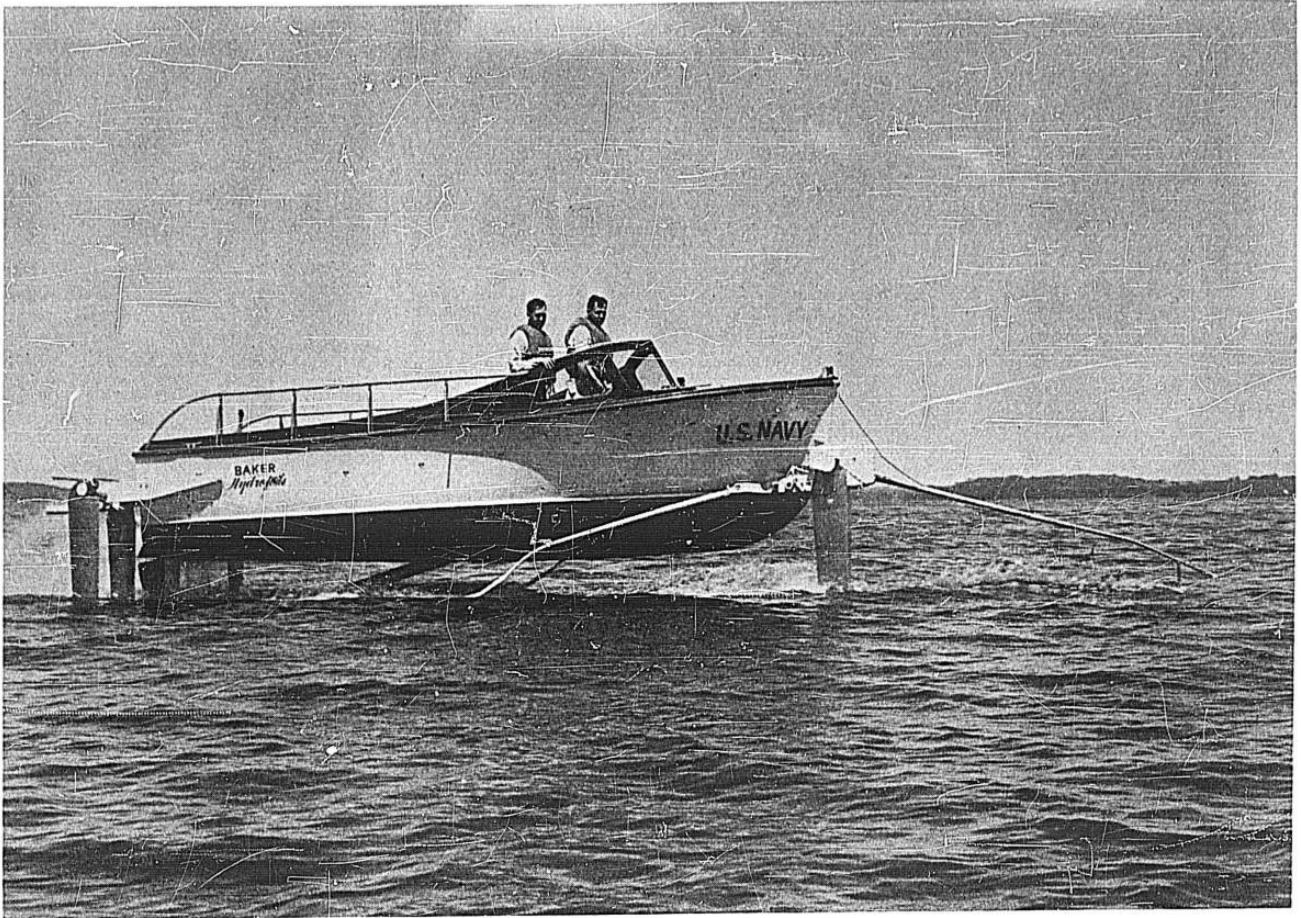


PHOTO 6 FLYING - SIDE VIEW

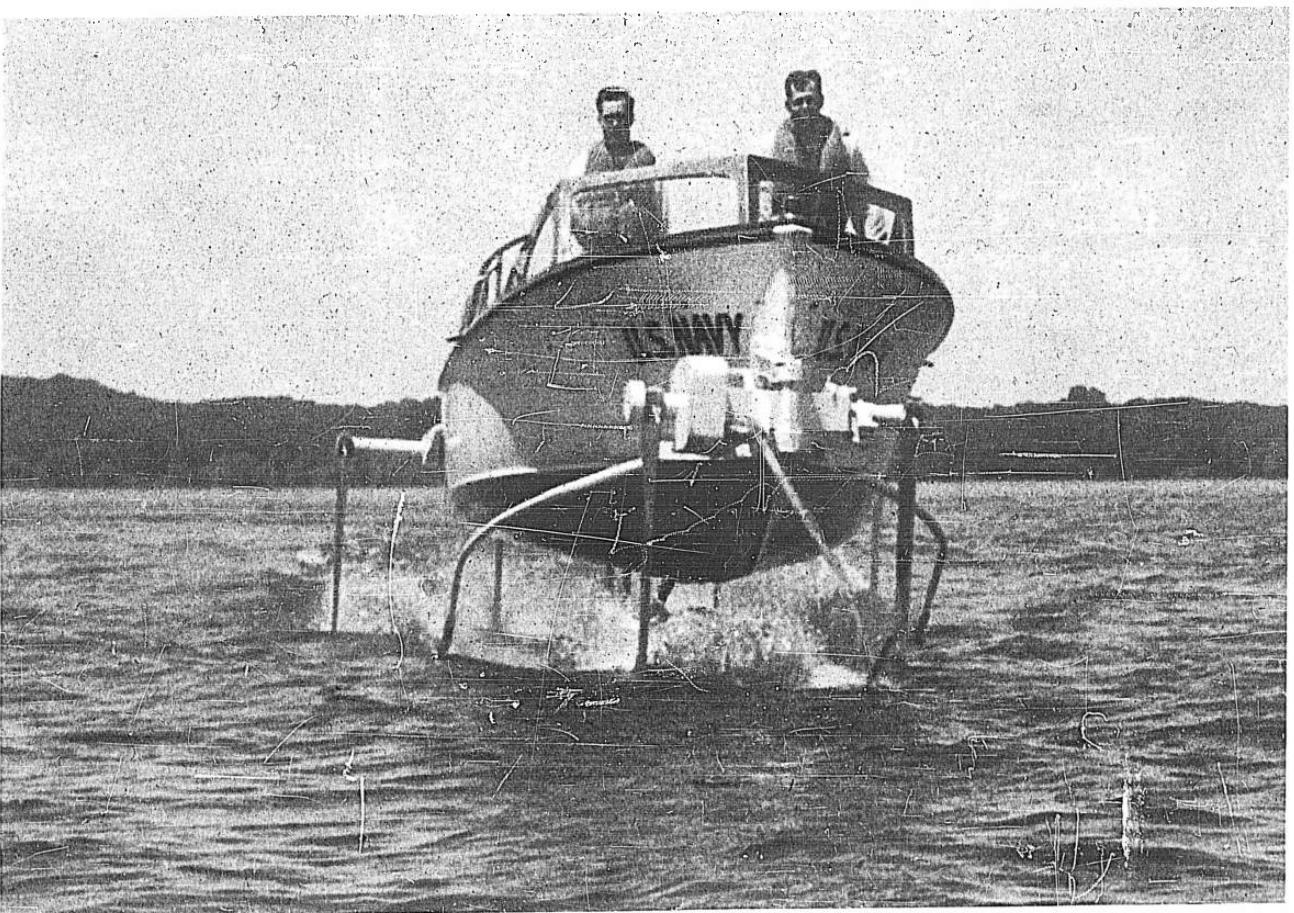


PHOTO 7 FLYING - HEAD ON VIEW

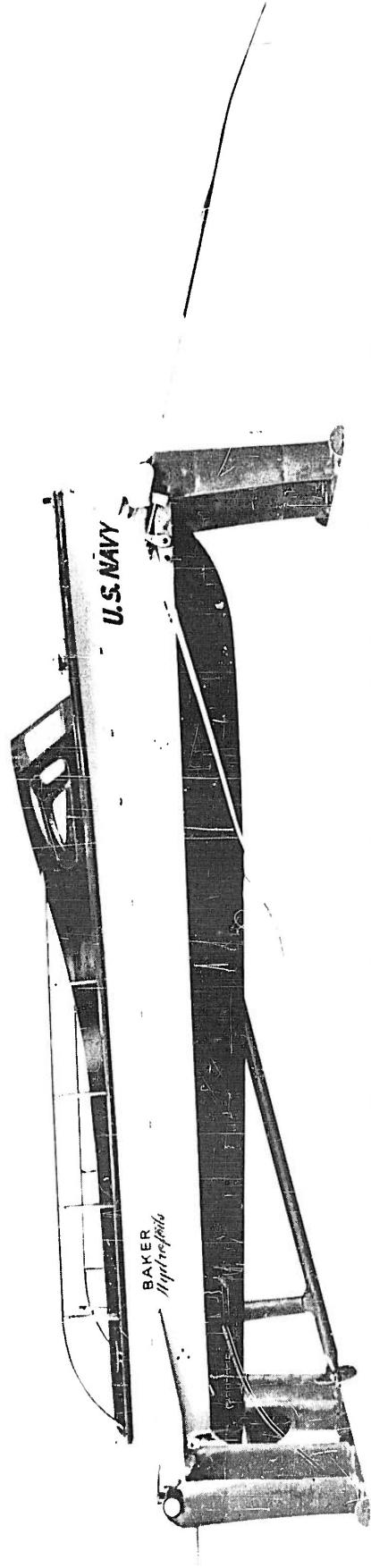


PHOTO 8 SUSPENDED - HYDROFOILS, SENSORS AND PROPELLER EXTENDED

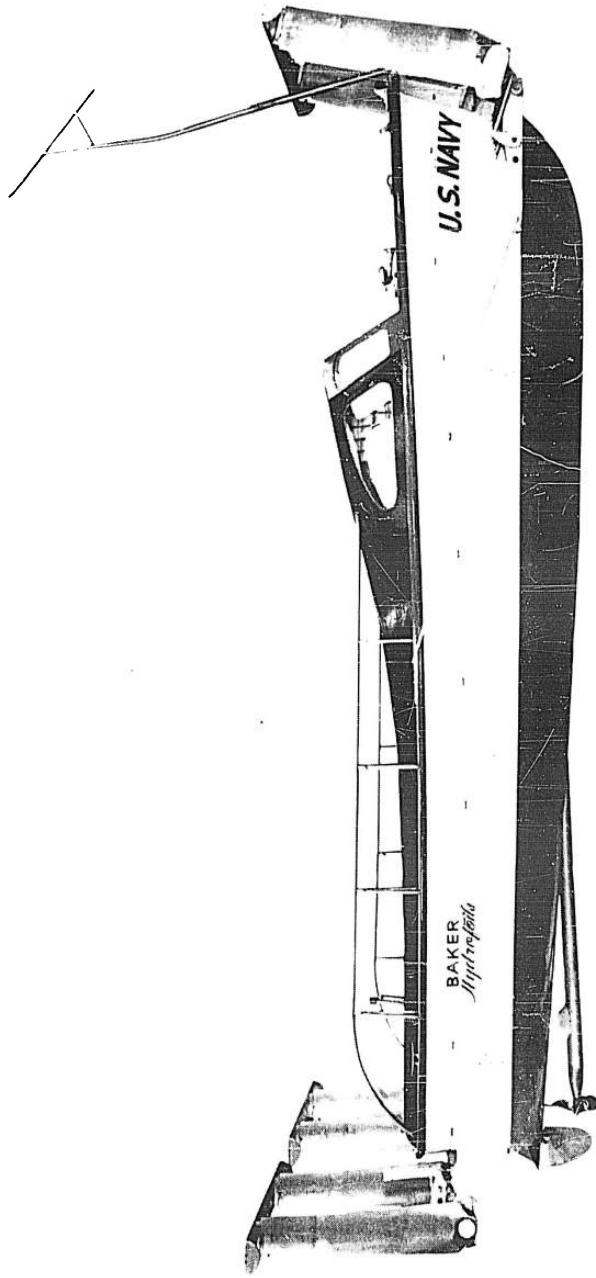


PHOTO 9 SUSPENDED - HYDROFOILS, SENSORS AND PROPELLER RETRACTED

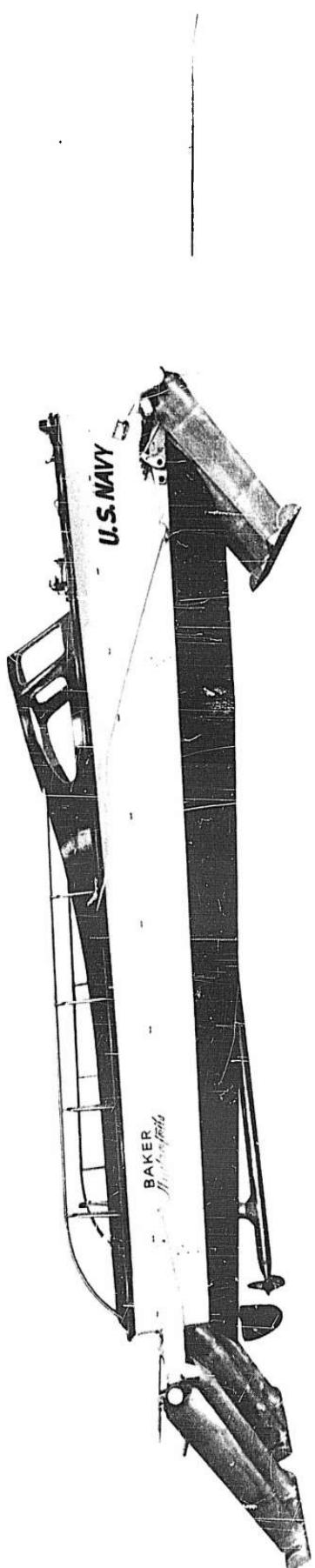


PHOTO 10 SUSPENDED - HYDROFOILS ROTATED REARWARD, PROPELLER MOVED UP SHOWING MOVEMENTS FOR IMPACT ENERGY ABSORPTION

for steering either where surface piercing hydrofoils are used or where high maneuverability in deep water at low speeds is important; and

- I. that the number of manual controls should be minimized.

Under the contract, two 24-foot hydrofoil model test boats were designed and constructed on the basis of the above conclusions. The first, known as *High Pockets*, in its preferred form, has four surface piercing hydrofoils, two forward and two nearly tracking aft. The second, known as *High Tail*, has three fully submerged, controlled angle hydrofoils one forward and two aft. Both boats have substantially the same design of hull, engine and propeller-drive system. Table I gives comparative data on *High Pockets* and *High Tail* and corresponding typical prototypes. The balance of this report is primarily concerned with fully submerged hydrofoils and *High Tail*.

DISCUSSION OF THE DESIGN OF HIGH TAIL

A. HYDROFOIL CONFIGURATION, MOUNTING AND ACTUATION

The *High Tail* arrangement of one hydrofoil at the bow and two at the stern is relatively simple, permits easy hydrofoil retraction and steering, and provides a location for a single propeller where propeller and hydrofoil interferences are small.

Each hydrofoil is pivoted on the lower ends of two hollow struts. Two struts are used instead of one in order to reduce the bending moments in the hydrofoil and to provide the means to sustain a large impact load at one end of the hydrofoil without the excessive pivot loads that would result if a single strut were used.

The hydrofoil is rotated by two hydraulic servo cylinders each connected to the hydrofoil with a push rod. Each push rod extends inside the hollow strut from the servo cylinder at the top of the strut down through guide bearings and a water seal to a spherical rod end connection to the hydrofoil.

A semi-circular end plate extends above each end of each hydrofoil except in the case of the inner ends of the aft hydrofoils to which circular end plates are attached. These end plates reduce the drag at the lower flying speeds and add resistance to lateral motion necessary to prevent yawing instability.

The Hydrofoil Draft Increment which fixes the length of the struts was determined for *High Tail*'s design as follows:

First, on the basis of mathematically predicted Flying Draft Variances given in Table II for the control settings planned,* the entire range of sea conditions assumed

* $A_f = .77$ for all wave lengths in head and following seas,
 $A_a = .99$ for wave lengths greater than 25 feet in following seas, and
 $A_a = 0$ for wave lengths less than 25 feet in following seas and for all wave lengths in head seas.

TABLE I COMPARISON OF HIGH TAIL, HIGH POCKETS AND TYPICAL PROTOTYPES

	<u>High Tail</u>	<u>Typical Prototype</u>	<u>High Pockets</u>	<u>Typical Prototype</u>
Type of Hydrofoil	Horizontal Fully Submerged		V - 42° Dihedral	
Type of Stabilization	Angle Control, Mechanical		Area Variation	
Computers, Hydraulic Servos				
Froude Number	.525	.525	.24	.24
Number of Hydrofoils	3	3	3 or 4	4
Steering	By rotation of front foil unit		By rotation of front foil units	
Hull Length	22'	88'	22'	40'
Overall Length, Foils Retracted	24'	96'	23' 9"	43' 8"
Overall Length, Foils Extended	25' 3"	101'	23'	40' 10"
Overall Width, Foils Retracted	12' 6"	40'	10' 6"	14'
Overall Width, Foils Extended	12' 6"	40'	15' 9"	29' 5"
Overall Width, Hull	7' 6"	20'	7' 6"	10' 4"
Draft Boat Stationary, Foils Retracted, L.W.L.	2' 1"	8' 4"	2' 1"	3'
Draft Boat Stationary, Foils Extended, L.W.L.	3' 5"	13' 8"	3' 5"	6' 8"
Longitudinal Foil Spacing	22'	88'	19'	32'
Transverse Spacing between Center Lines of side-by-side Foils	7'	28'	8' 6"	19' 2"
Height above L.W.L., Foils Retracted	5' 3"	21'	5' 3"	9' 2"
Height above L.W.L., Foils Extended	4' 3"	17'	4' 3"	6' 4"
Foil Chord, Front	9"	20"	7 1/2"	13"
Foil Chord, Aft	7 1/2"	16 5/8"	7 1/2"	13"
Propeller Diameter and Pitch	15" x 16"		15" x 20"	21" x 28"
Full Speed Clearance	1'	4'	1'	1' 8"
Full Speed, Fore Foil Submergence below Mean Sea Level	18"	6'	15"	26"
Full Speed, Aft Foil Submergence below Mean Sea Level	18"	6'	17"	26"
Velocity of Emergence, Knots at $C_L = .6$	17	34	21	27
Cruising Speed in Waves, Knots at $C_L = .4$	22	44	30	39
Approximate Top Speed, Knots	30	60	35	45
Engine	Chrysler M-47S with 1.41 V-Drive, Reduction Gear		Chrysler M-47S with 1.41 V-Drive, Reduction Gear	
Horsepower	135	4,000	135	600
Method of Retraction Foils & Propeller	Hydraulic	Hydraulic	Hand Crank	Hydraulic
Means of Foil-Impact, Load Limitation	Hydraulic	Hydraulic	Friction	Hydraulic
Foil and Strut Material (After conversion)	K-Monel	K-Monel	Alum. Alloy 2024-T4	Alum. Alloy 7075-T6
Type of Foil and Strut Structure	Hollow Weldment		Solid Extrusion	
Fuel Capacity	52 Gal.		42 Gal.	180 Gal.
Hull Construction	Round Bottom Fiberglass		Round Bottom Fiberglass	V Bottom Fiberglass
Turning Circle	9.9 Boat Lengths**		360 Ft. Radius at 32 Knots	
Displacement, Full Load Condition, Lbs.	6,000	118,000	6,000	28,000
Pay Load, Lbs.	915	32,000	950	8,100
Light Condition, Lbs.	4,600	86,000	4,450	18,100
Lift/Drag Ratio*	- 27.5 Knots		16.4	
Lift/Drag Ratio*	- 24.2 Knots		15.0	
Propeller Shaft Up	- 23.8 Knots		17.3	
	- 21.6 Knots	16.0*		
Lift/Drag Ratio*	- 27.5 Knots		13.7	
Lift/Drag Ratio*	- 24.2 Knots		13.0	
Propeller Shaft Down	- 23.8 Knots		15.1	
	- 21.6 Knots	14.4***		

* $\frac{\text{Lift}}{\text{Drag}}$ i.e. $\frac{\text{Weight of Loaded Boat}}{\text{Towing Force without Propeller}}$

** Overall boat length of 25' 3" was used in this determination.

*** Computed by adding *High Tail's* propeller shaft up towing force to the propeller shaft housing and strut drag.

was divided into two parts, the Lower Range including all sea conditions where *High Tail's* Flying Draft Variance is 2 feet or less and the Upper Range including all sea conditions where *High Tail's* Flying Draft Variance is more than 2 feet.

Second, on the basis of experience with *High Pockets*, the Lift Draft was taken as 6 inches.

Third, also on the basis of experience with *High Pockets*, a Contour Allowance of zero was selected for the Lower Range of sea conditions and a Contour Allowance of minus 5 inches was selected for the Upper Range of sea conditions.

Fourth, the Hydrofoil Draft Increment was determined as 2 1/2 feet for both Upper and Lower Ranges of sea conditions by adding the maximum Flying Draft Variance for each range of sea conditions to the Lift Draft and the corresponding Contour Allowance. The change in Contour Allowance is introduced by means of the three control station trimming adjustments.

The control setting for a Contour Allowance of minus 5 inches results in a small added drag in the Upper Range of sea conditions for which it is used. But this extra drag is not serious because of the rare occurrence of sea conditions in the Upper Range.

B. HYDROFOIL AREA

It was planned on the basis of Reference 8 to limit the maximum lift coefficient to 0.8. With the planned limiting acceleration upward due to waves or turning equal to gravity, i.e., 2g total. The 0.8 coefficient of lift in waves corresponds to 0.4 in smooth water. Fortunately the L/D is near maximum at a coefficient of lift of 0.4.

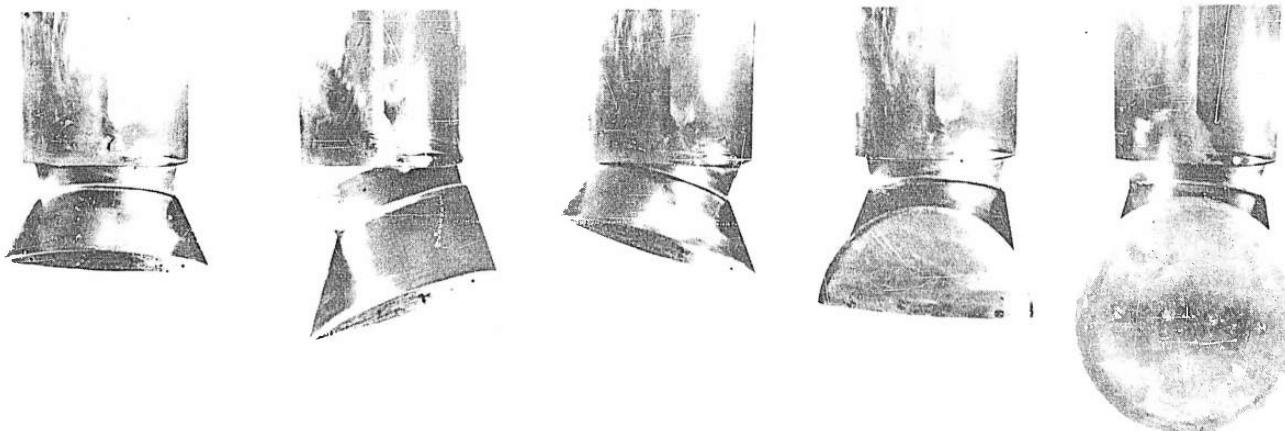
The Up Speed was chosen to place the coefficient of lift in smooth water at 0.6, the difference between 0.6 and 0.8 being again to provide for the smaller vertical acceleration due to waves and turning at Up Speed.

C. THE HYDROFOIL PIVOT AXIS LOCATION AND THE SMOOTHED SERVO PRESSURE

The fact that the moment coefficient for the one-quarter chord point of a hydrofoil is substantially independent of the angle of attack is taken advantage of in the design of *High Tail* (Reference 9). A pivot point on the chord line is considered structurally impractical, but there is a pivot point location sufficiently above the chord line to be structurally suitable and for which the variation in moment coefficient with change in angle of attack is small. The hydrodynamic moment on a hydrofoil with such a pivot location therefore varies only with the Dynamic Pressure.

The reaction of the hydrodynamic moment on a hydrofoil is the sum of the moments due to a) weight of the hydrofoil, b) inertia of the hydrofoil, c) friction resisting rotation of the hydrofoil above its pivot axis, d) drain pressure in the servo cylinders and e) Servo Pressure in servo cylinders, the servo cylinders being the hydraulic cylinders that actuate the rotation of the hydrofoil for control. The moment due to weight is comparatively small and substantially constant; the moment due to inertia is small and alternating; the moment due to friction is 10 to 20 percent of the hydrodynamic moment and alternating; the moment due to drain pressure is small; and the moment due to Servo Pressure is the balance of the reaction of the hydrodynamic moment on the hydrofoil. It follows that the Servo Pressure can be considered as made up of two components 1) a steady component approximately proportional to the Dynamic Pressure and hence varying only with the speed of the hydrofoil with respect to the water and 2) an alternating component proportional to the sum of the inertia and friction moments which have the same frequencies of alternation as that of the water surface passing the hydrofoil. By smoothing, the Servo Pressure alternating component is largely removed so that the Smooth Servo Pressure becomes approximately proportional to the Dynamic Pressure. In other words, the ratio $\frac{Q}{q}$ is approximately constant.

Smoothed Servo Pressures are used on *High Tail* to control the loading on the sensors, to substantially eliminate change in Bottom Clearance with speed at speeds above Up Speed, and to automatically introduce control



(A) CAGED POSITION, (B) MAXIMUM ANGLE, (C) MINIMUM ANGLE, (D) ONE-HALF CIRCLE (E) FULL CIRCLE
END PLATE REMOVED END PLATE REMOVED END PLATE END PLATE END PLATE

PHOTO 11 HYDROFOIL DETAILS

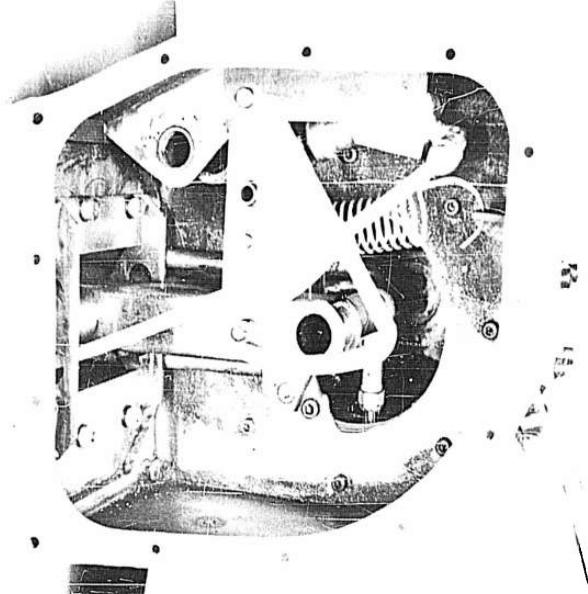


PHOTO 12 RIGHT AFT HYDROFOIL SIGNAL INPUT AND CAGING - COVER REMOVED

operations during acceleration and deceleration of the boat.

D. SERVOS

The flow of high pressure hydrolic fluid to and the drain from the servo cylinder is controled by a valve. A mechanical linkage produces the difference between a hydrofoil angle command signal and the hydrofoil angle. This difference as a displacement operates the hydrolic valve in such a way as to remove the difference. Thus, the hydrofoil angle follows the angle command signal.

The ratio of the rate of fluid flow through the servo valve to the difference between the command signal and the hydrofoil angle setting was selected, in the design, to produce optimum smoothing as determined by the mathematical analysis of Appendix D.

E. STEERING

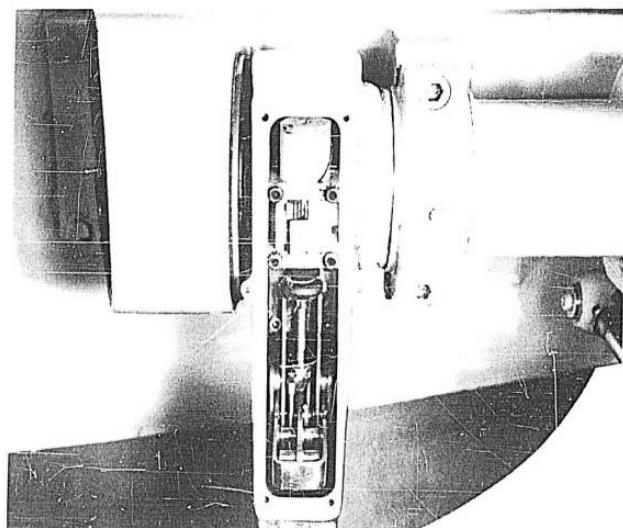


PHOTO 13 RIGHT AFT SERVO - COVER REMOVED

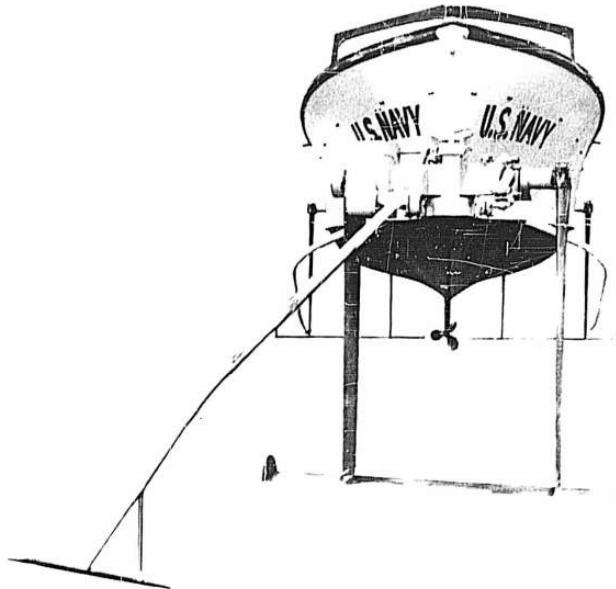


PHOTO 14 SUSPENDED - HEAD ON VIEW, STEERED TO THE RIGHT - SENSORS, HYDROFOILS AND PROPELLER EXTENDED

In the displacement condition with the hydrofoil and the propeller retracted, *High Tail* is steered with a rudder cable connected to a small steering wheel at the control station. This rudder produces very little drag, first, because it is spring centered whenever the small steering wheel is released and, second, in the flying condition most of the rudder surface is above the water.

In the flying condition, the forward hydrofoil unit, including the supporting struts, is rotated about an upright axis for steering. Bow steering, by rotation of the forward hydrofoil unit, was chosen to avoid substantially altering the flow direction of the water approaching the propeller when turning and to improve maneuverability in the floating condition with the hydrofoils extended.

The steering rotation is power assisted with a standard automotive power steering unit with a check valve added to prevent sudden rotation of the forward hydrofoil unit as the result of striking an obstacle with the forward hydrofoil.

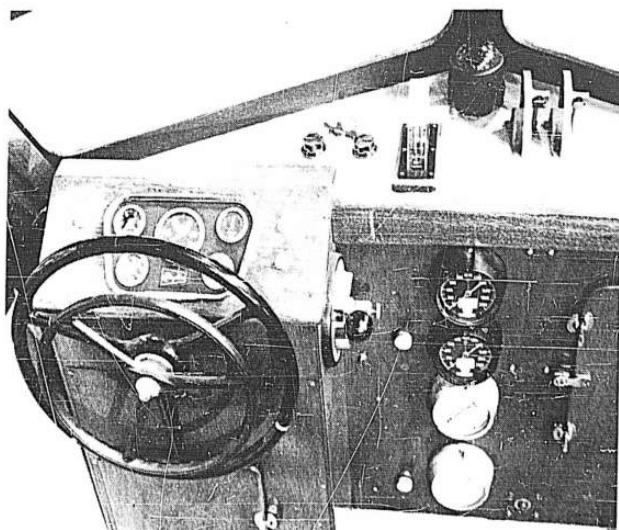


PHOTO 15 CONTROL STATION

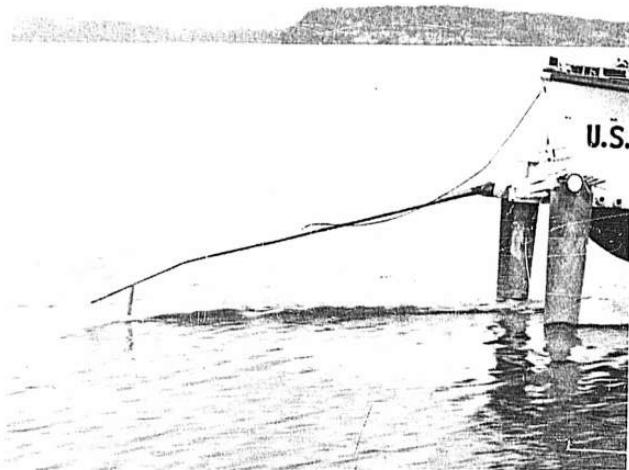


PHOTO 16 FORWARD SENSOR IN ACTION

F. SENSORS

The mathematical analysis reported on herein indicates that the Flying Draft Variance becomes excessive if the Sensor Lead is not near optimum. It does not appear practical to simulate Sensor Lead by the use of a wave slope signal or a time rate of change in sensor signal because of the frequently present short wave length water surface roughness. Actual optimum Sensor Lead therefore appears essential for minimum Flying Draft Variance.

The results of the mathematical analysis shown in Table II indicate that for a boat of *High Tail's* proportions and Hydrofoil Froude Number, the maximum Flying Draft Variance without aft hydrofoil pitch and heave control is more than double that possible with surface sensing pitch and heave control. It follows that the omission of aft hydrofoil pitch and heave control approximately doubles the Hydrofoil Draft Increment required and correspondingly increases the propeller draft needed.

The contractor suggested mechanical sensing and an electronic sensing scheme for *High Tail*. (Reference 30) The choice of mechanical sensing was made for the purposes of the contract by the Office of Naval Research.

High Tail has one forward extending mechanical sensor touching the water surface forward of the forward hydrofoil and two aft extending mechanical sensors one touching the water surface forward of each aft hydrofoil.* All have near optimum Sensor Lead. The forward sensor controls pitch and heave, and the aft sensors control pitch, heave and roll.

If a hydrofoil system were enough narrower than *High Tail's* system to eliminate any need for the boat to follow the water surface in roll as *High Tail* does, a vertical determining device such as a gyro vertical could be used for producing the roll signal and no water sensor

signal would be needed for roll control.

As *High Tail* accelerates from rest (decelerates from speed) the sensors unfold (fold) automatically. Underway, in order to avoid substantial change in sensor penetration of the water surface with change in boat speed, the loading of each sensor is increased automatically with the square of the boat speed. This is done by using the Servo Pressures to control the hydraulic pressures in the cylinders which load the sensors.

Several designs for the part of the sensors in contact with the water were tested seeking to minimize drag, spray and impact. The design that appears best is a narrow, rigid ski with negative dead rise of 45° and with streamlined strut attachments to the sensor arm. This form of sensor penetrates smaller waves reducing the vertical acceleration of the ski without the use of added flexibility in the ski attachment to the sensor arm.

It is natural to be concerned about the possibility of Tripping of a forward extending sensor, i.e., the sudden submergence and downward rotation of the sensor. In the many hours of testing thus far with several sensor designs and a variety of wave conditions, Tripping has never occurred.† Yet since all possible wave conditions have not been encountered the possibility cannot be ruled out. It is certainly important in the design: a) to minimize the probability of Tripping, and b) to avoid damage and delay if Tripping should ever take place.

It is believed that the probability of Tripping on *High Tail* has been minimized by the use of the lowest practical pivot axis for the sensor arm, by streamlining the upper portions of the ski and the strut attachments of the ski to the sensor arm and by the choice of optimum trim angle for the ski. *High Tail* has a cable attachment between the hull and the middle of the sensor arm to minimize damage should Tripping take place. It is doubtful if this arrangement alone would prevent damage to the outer end of the arm.

The avoidance of damage, if Tripping should take place, appears to be possible either by hinging of the sensor arm to permit downward rotation of the ski and outer end of the arm or by rotating the sensor arm with a hydraulic servo responsive to the hydrodynamic force on a ski or other water probe at the outer end of the sensor arm.

For a given Hydrofoil Froude Number, a geometrically similar increase in boat and wave train dimensions decreases the frequency of wave encounter. Unfortunately, the natural frequencies of a sensor arm decrease by a larger ratio. This effect tends to limit the size of a practical sensor and therefore, limits the size of boat, with a given Hydrofoil Froude Number, that can practically use a given type of mechanical sensor.‡ No such limit has been determined by the contractor. However, mechanical sensing within this size limitation has the following advantages:

*W. M. and L. E. Meacham are believed to be the first (1916) to show a forward extending sensor controlling the angle of a forward hydrofoil (Reference 19). Later (1953) C. Hook also shows forward extending sensors using the same principal as the Meachams but different details (Reference 20). See Reference 17 for other information on Hook's scheme and developments therefrom.

†Once when a fracture due to a defective weld occurred in the outer end of the sensor arm removing the ski, the arm, having lost its ski support, rotated downward submerged and was carried against the forward hydrofoil.

‡If the sensor arm is rotated with a hydraulic servo, as mentioned previously, it is believed the size limit is substantially greater than where a ski is depended on for the upward force on the sensor arm.

- 1) Accurate sensing for the control of the rear hydrofoils can be accomplished during take-off despite heavy spray from the hull.
- 2) Optimum Sensor Lead for the forward hydrofoil or hydrofoils is easily obtained.
- 3) Sensing can be very reliable especially for roll control where lack of reliability could cause the boat to capsize.
- 4) Even with very light sensors, adequate energy can be obtained from the sensors to operate the controls except for the actuation of the hydrofoils. Hence no power supply other than hydraulic pressure is required.

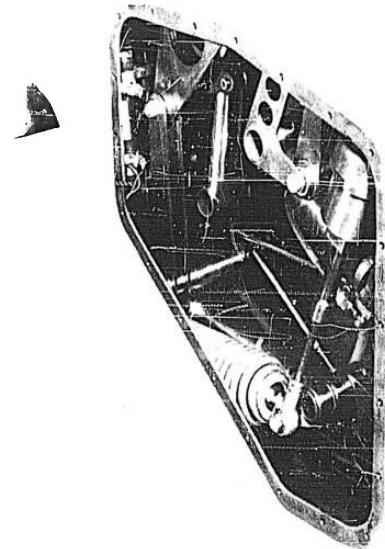


PHOTO 17 FORWARD COMPUTER SECTION AND FORWARD SENSOR CONTROL - COVER OFF - QUARTER - ING VIEW

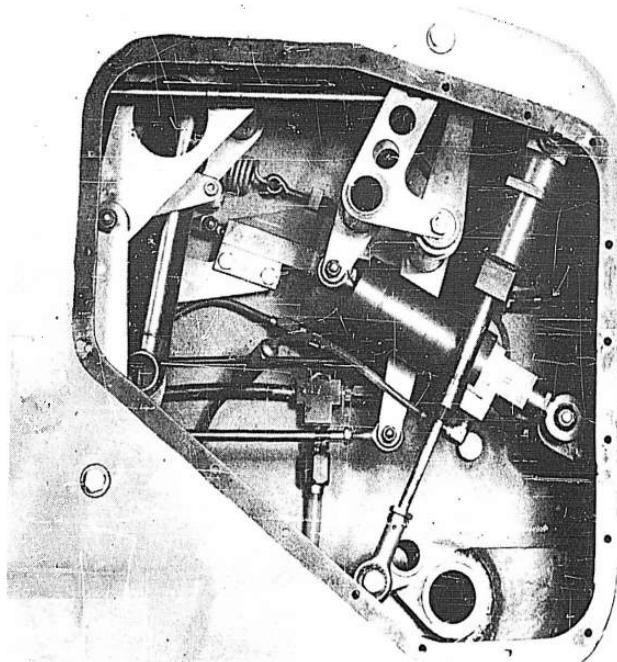


PHOTO 18 FORWARD COMPUTER SECTION AND FORWARD SENSOR CONTROL - COVER OFF - SIDE VIEW

G. COMPUTER

A Mechanical computer is a basic part in *High Tail's* control. The computer is made up of adding and subtracting linkages, multiplying levers and Function Units. The computer puts out three angle command signals, one for each hydrofoil servo and other signals for:

- a) control of the folding and unfolding of the sensors;
- b) starting and stopping the angle control; and
- c) indicating the flying elevation of the boat.

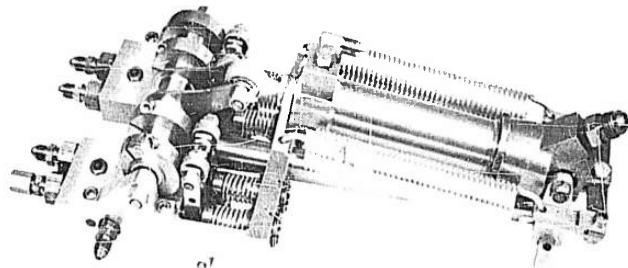


PHOTO 19 SEQUENCE FUNCTION UNIT

The input signals to the computer are:

- 1) the displacements of three water surface sensors;
- 2) the steering angle of the forward hydrofoil;
- 3) the Servo Pressures;
- 4) the four manual control station adjustments of which

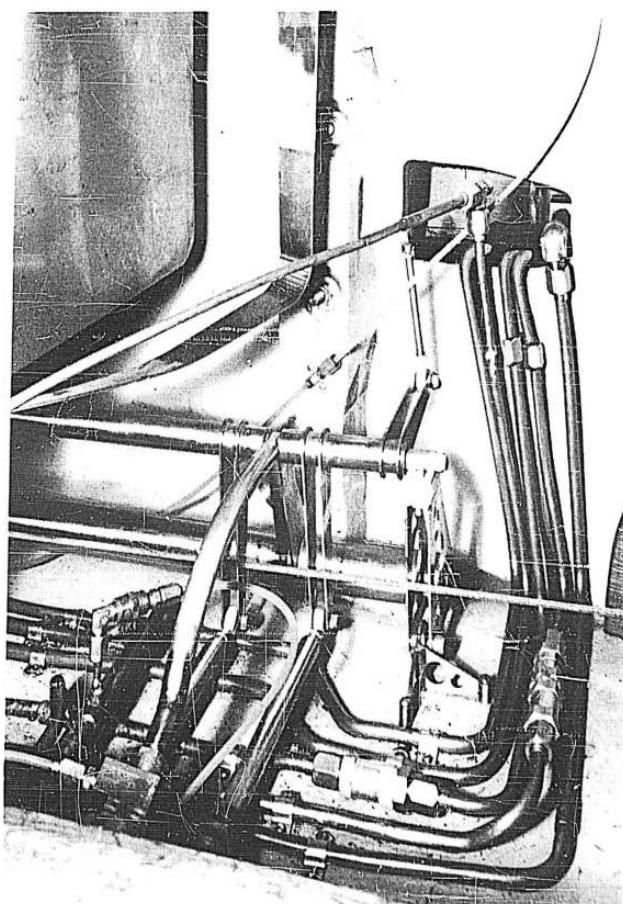


PHOTO 20 STERN COMPUTER SECTION AND RUDDER CONTROL - COVER REMOVED

three are for pay load, elevation, pitch and roll trimming and the fourth is for sea condition.

Because of the computer characteristics, *High Tail* on a straight course automatically flies at a constant mean altitude at all flying speeds and banks and lowers in turns. As *High Tail* accelerates from rest (decelerates from speed) and sensors unfold (fold), the angle control starts (stops), and the hull abruptly climbs to (descends from) the flying elevation, all automatically at predetermined speeds.

The contractor proposed in Reference 30 an alternative electrical computer. The mechanical form was chosen for the purpose of the contract by the Office of Naval Research. Where mechanical sensing is used or where the output signal of the sensor is mechanical as with the sensor proposed in Reference 30, the mechanical form of

the computer appears the best choice because no power supply is involved and the computer is relatively simple, inherently reliable and low in cost and maintenance.

The size of the computer on *High Tail* was determined, to an important degree, by the room required for access to all parts. On this account, the size of a similar computer for a prototype would be much smaller in relation to the size of boat.

If a hydrofoil system were enough narrower than *High Tail's* system to eliminate any need for the boat to follow the water surface in roll, a vertical determining device such as a gyro vertical could be used for producing the roll signal instead of the difference between the aft sensor displacements as in *High Tail's* control. This would, of course, add the vertical determining device but

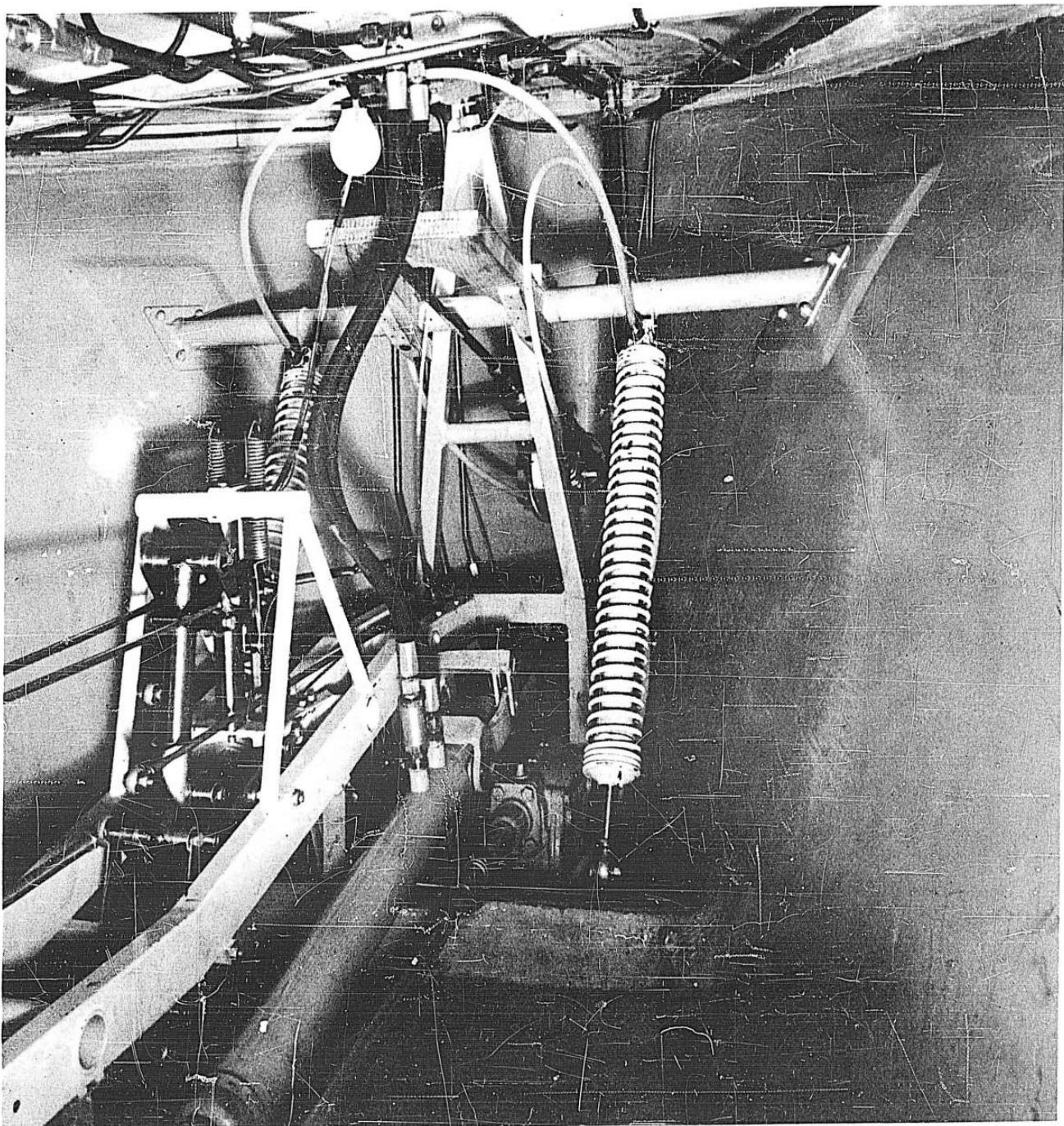


PHOTO 21 BOW COMPUTER SECTION AND REAR SENSOR CONTROL AS VIEWED THROUGH ACCESS DOOR

there would be a substantial simplification in the computer section for the aft hydrofoils.

High Tail's computer section for the forward hydrofoil includes, for test purposes, a multiplier for control station adjustment of the coupling factor A_f during flight. The analysis of Appendix D indicates, and the tests thus far confirm, that no adjustment of A_f is needed on a prototype. On this account, for a prototype, the computer section for the forward hydrofoil could be substantially simplified.

H. HYDROFOIL RETRACTION

High Tail's hydrofoils are retractable to avoid excessive draft in the displacement condition and to aid in maintaining smooth surfaces on the hydrofoils. Each hydrofoil is rotated 180° about a horizontal transverse axis for retraction by means of two manually controlled power operated hydraulic cylinders.

I. PROPULSION AND PROPELLER RETRACTION

The propulsion engine (see Table I for specification) with conventional V reduction gear, drives the propeller shaft through a universal joint. The propeller shaft is supported in a streamlined housing fixed to a streamlined strut which extends upward into and is supported within an upright cylindrical tube rigidly mounted within the hull.

The propeller is moved down and retracted by means of a small hydraulic cylinder the ram of which is coupled to the strut. The corresponding freedom of shaft movement is provided by the universal joint. The propeller is retracted to its uppermost position when the foils are retracted and the boat is operated in the floating condition. The propeller is extended to its lowest position for flying.

Water for cooling the engine and lubricating the propeller shaft bearings is conducted into the hull by means of the propeller shaft housing.

It is of interest to compare the slanting drive shaft system to the right angle bevel gear drive plan which has also been used for hydrofoil boats.* The towing tests of Reference 5 indicate that the added drag of the shaft housing and strut of the slanting drive shaft system is about 10 per cent of the total drag of *High Tail* at 21.6 knots, the design speed. With the right angle bevel gear drive, the submerged nacelle, to house the bevel gears, and the thickened strut, to house the vertical drive shaft, add to the drag. In addition there are the losses in the bevel gears and the reduction in propeller efficiency caused by the interference of the nacelle and vertical strut.[†] All things considered there appears to be no sub-

stantial advantage in the right angle bevel gear drive from the standpoint of efficiency:

The advantages of the slanting shaft plan are:

- 1) no underwater gearing to lubricate and isolate from sea water;
- 2) main engine power available for both floating and flying conditions without added complication; and
- 3) lower first cost and maintenance cost.

The slanting shaft design is believed heavier than the right angle bevel gear design especially if the shaft is solid and the shaft angle is less than 15° . With a hollow shaft and a shaft angle of 15° or greater, it appears that the weight difference is small and justified by the advantages listed above unless the Flying Draft Variance is excessive. It follows that the practical use of the slanting shaft plan may depend on good performance of the hydrofoil angle control.

J. IMPACT ENERGY ABSORPTION

To minimize the shock of striking bottom or an obstacle in the water a high flow velocity relief valve is included within one of the retraction cylinders for each aft hydrofoil to permit rotation of the hydrofoil about the folding axis under high resistance to absorb as much of the impact energy as practical without exceeding the strength of the structure.

In the case of the forward hydrofoil the rotation for retraction is opposite to that needed for shock absorption, so that a separate hydraulic cylinder with external high flow velocity relief valve and a separate hinge axis are provided.

An external high flow velocity relief valve is also used with the propeller retraction cylinder to provide for energy absorption in case of impact on the propeller or propeller shaft housing.

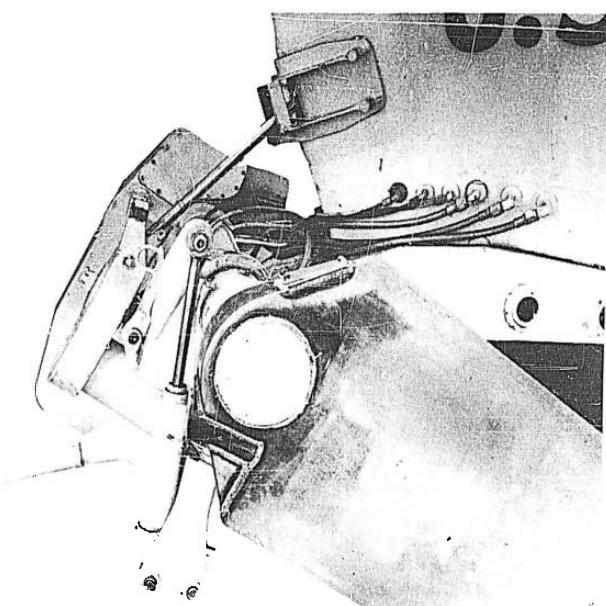


PHOTO 22 SUSPENDED - BOW VIEW, FORWARD HYDROFOIL ROTATED REARWARD SHOWING MOVEMENT AND IMPACT ENERGY ABSORPTION

*A study of a large number of hydrofoil drive systems is reported on in Reference 18.

[†]That the interference with the propeller can be small with the slanting shaft design is indicated by the high propeller efficiency of *High Pockets* as reported in Reference 5.

K. HYDRAULIC SYSTEM

Hydraulics are used in the control for actuation of the controlled hydrofoil rotations, for smoothing, for transmitting manual adjustments from the control station to the computer, for automatic introduction of the control operations required during forward acceleration and deceleration, for eliminating changes in Bottom Clearance with changes in speed and for power assisted steering.

Hydraulics are also used for retraction and extension of hydrofoils, propeller and sensors and for impact energy absorption.

The use of electrical components (precluded by the specification) would somewhat simplify the hydraulic piping.

(1) can be applied to each component with a rough allowance for departures from the assumptions on which (1) is based and the results added to determine the total Flying Draft Variance and acceleration amplitude. Or, more simply, Equation (1) may be applied to the dominant sinusoidal component only with the presence of the components of other wave lengths taken into account in the choice of R. The latter procedure with a slightly different equation was used for purposes of predicting the Flying Draft Variance inherently required for *High Tail*.

The Hydrofoil Draft Increment, needed to provide adequate Bottom Clearance between the hydrofoils for a tandem hydrofoil system, is the sum of the Lift Draft, the Contour Allowance, and the Flying Draft Variance.

SUBMERGED HYDROFOIL CONTROL THEORY AND APPLICATION

A. PITCH AND HEAVE CONTROL REQUIREMENTS FOR MINIMUM DRAFT AND LIMITED ACCELERATION

The Hydrofoil Draft Increment required is of course determined by the maximum Flying Draft Variance to be encountered. As an approach to the approximate determination of the maximum Flying Draft Variance, consider a single hydrofoil flying at a constant horizontal speed v in a sinusoidal wave train of given wave length λ and amplitude y_0 . Assume a sinusoidal hydrofoil flight path of the same phase and wave length as the wave train as determined in the direction of flight. Under these conditions the Flying Draft Variance d_r required at the center of the foil to limit the vertical acceleration of the hydrofoil to a specific tolerable amount, as determined in Appendix B, is given by:

$$d_r = \frac{2\lambda}{R} - \frac{(\ddot{h})_t}{2\pi^2(v \cos\beta \pm 2.265 \sqrt{\lambda})^2} \quad (1)$$

Inspection of Equation (1) indicates that the largest Flying Draft Variance, d_r , for given magnitudes of λ , R, $(\ddot{h})_t$, and v, is with a plus sign in the denominator of the last term and with β equal to zero which is the head sea condition. In the head sea condition with $(\ddot{h})_t = 32.2 \text{ ft/sec}^2$; $v = 36.6 \text{ ft/sec}$; $R = 30$; the maximum $d_r = d_{rm} = 2.08'$ occurs at $\lambda = \lambda_m = 83'$. The relation between d_r and λ is shown in Chart I.

If the flight path and wave profile are not in phase, a larger Flying Draft Variance is required.

An actual wave train is of course not sinusoidal so that for accurate treatment it would be necessary to consider an actual wave train and the corresponding flight path in several component wave trains. For the purpose of estimating draft and acceleration limits it appears only necessary to consider the dominant sinusoidal component and one or two other components. On this basis Equation

B. PITCH AND HEAVE CONTROL SIGNALS AND COUPLING PARAMETERS FOR ZERO FLYING DRAFT VARIANCE IN LONG WAVES

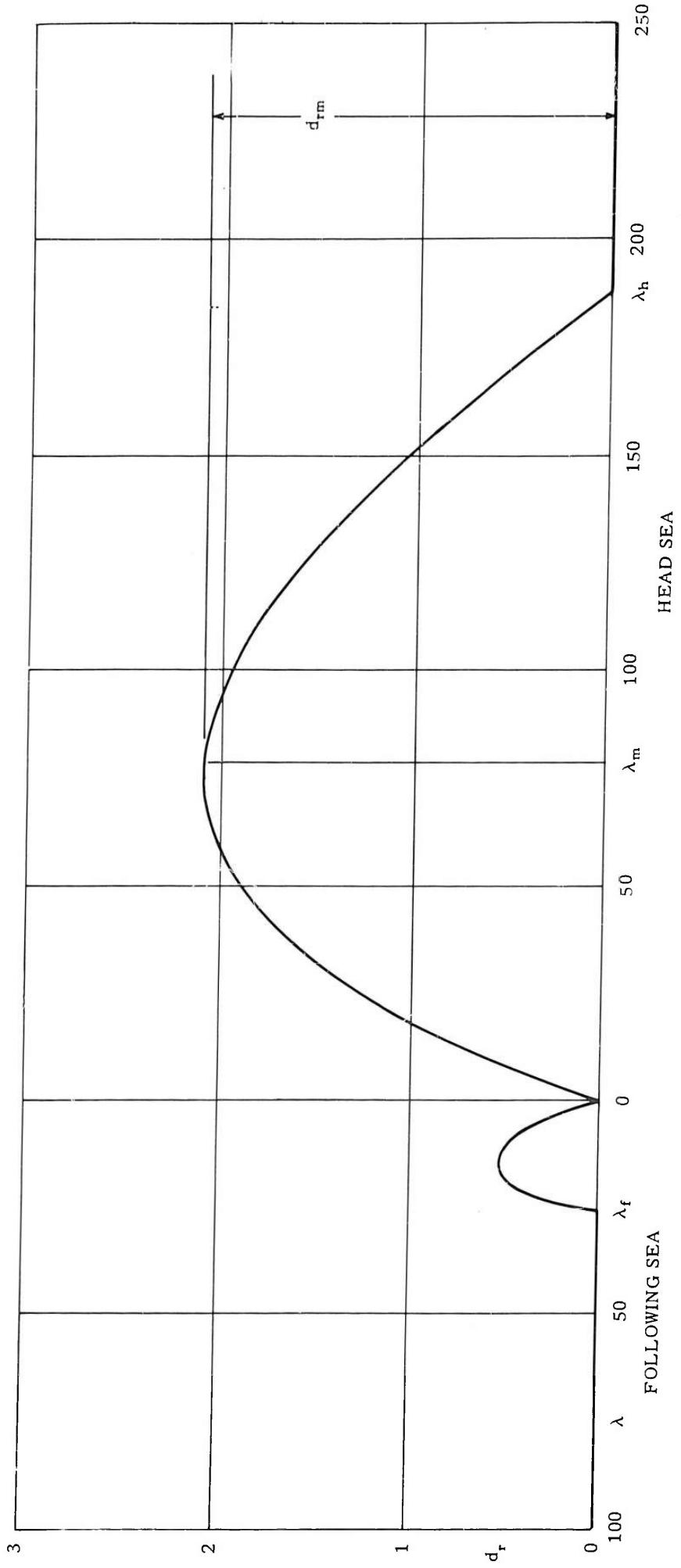
In a tandem, fully submerged hydrofoil system, at least one signal dependent on the elevation of the hull relative to the water surface is necessary for control of the incidence angles of the hydrofoils. For control design purposes it is necessary to select the location or locations of sensing the water surface and to determine the additional signals required.

From Chart I it is apparent that no Flying Draft Variance is required to limit the acceleration to that tolerable for either following sea wave lengths longer than λ_f or head sea wave lengths longer than λ_h . Zero Flying Draft Variance for such wave lengths is desirable in order to permit adequate Flying Draft Variance for shorter waves typically present and to avoid the corresponding vertical acceleration components of objectionable magnitude. Considering only pitch and heave motions and assuming that the wave lengths are long enough to neglect the rate of change in water surface curvature encountered, the condition of zero Flying Draft Variance determines the coupling relationships between the hydrofoil angles and the sensor displacements both with respect to the boat hull. These relationships depend on the fore and aft locations of the sensors and foils, the velocity of the boat, the velocity of the wave train, the course of the boat with respect to the direction of the wave train velocity, and the relative water velocity with respect to the boat.

For the case of a water surface sensor forward of each foil, it is shown in Appendix C that the required coupling relationships between the sensors and the hydrofoils for zero Flying Draft Variance in long Waves is expressed approximately by:

$$\alpha_f = \frac{w_f}{Kq} - \frac{A_{fn}}{1} \delta_1 \quad (2)$$

$$\alpha_a = \frac{w_a}{Kq} - \frac{A_{an}}{1} \delta_2 \quad (3)$$



Flying Draft Variance = d_r

Maximum $d_r = d_{rm} = 2.08$

Limiting Acceleration = $(\ddot{h})_t = 32.2 \text{ ft/sec}^2$

Wave length = λ

Wave length at which d_r is maximum = $\lambda_m = 83$

Wave length - amplitude ratio = $\frac{\lambda}{Y_0} = 30$

Boat Velocity = $v = 36.6 \text{ ft/sec}^2$

CHART I THEORETICAL FLYING DRAFT VARIANCE REQUIRED TO LIMIT VERTICAL ACCELERATION

in which A_{fn} and A_{an} are determined by the wave length and the wave train velocity direction with respect to the boat course.

C. THE SELECTION OF PITCH AND HEAVE CONTROL PARAMETERS FOR MINIMUM DRAFT AND LIMITED ACCELERATION

The Dynamic analysis of Appendix D indicates: a) that the Flying Draft Variance in long waves with the coupling relationships expressed by Equations (2) and (3) is very small, as would be expected; b) that the Flying Draft Variance is excessive for wave lengths about twice the boat length; and c) that the maximum Flying Draft Variance and maximum vertical acceleration for all wave lengths and wave train velocity directions relative to the boat course, can be brought within practical limits by selecting the optimum Sensor Lead, by replacing the coefficients A_{fn} and A_{an} in Equations (2) and (3), with the coefficients A_f and A_a selected to correspond to the wave length and wave train velocity direction, and by introducing first order smoothing. Equations (2) and (3) with these modifications become:

$$\alpha_f + E \dot{\alpha}_f = \frac{w_f}{Kq} - \frac{A_f}{1} \delta_1 \quad (4)$$

$$\alpha_a + E \dot{\alpha}_a = \frac{w_a}{Kq} - \frac{A_a}{1} \delta_2 \quad (5)$$

The results of the analysis indicate that there is latitude in the selection of A_f and A_a . However, selections of particular interest are: $A_f = 0.77$ for all wave lengths in head and following seas, $A_a = 0.99$ for wave lengths greater than 25 feet in following seas, and $A_a = 0$ for wave lengths less than 25 feet for following seas and for all wave lengths in head seas. With these selections, the entire range of head and following sea conditions involves no change in A_f and only two values of A_a ; the computed maximum Flying Draft Variance for the hydrofoils is 2.41 feet; and the computed maximum acceleration at the center of the boat is 35.75 ft/sec².

These figures may be compared to the maximum Flying Draft Variance of 2.08 feet and a maximum acceleration of 32.2 ft/sec² for the idealized single hydrofoil case mentioned in connection with Equation (1). In quartering seas with lower frequencies of wave encounter and higher ratios of wave length to amplitude there is more latitude in the selection of A_f and A_a so that no additional combination of A_f and A_a appears to be needed or desirable.

In view of the definitions of α_a and δ_2 Equation (5) may be expanded to:

$$\alpha_{ar} + \alpha_{a1} + E \dot{\alpha}_{ar} + E \dot{\alpha}_{a1} = \frac{2w_a}{Kq} - \frac{A_a}{1} (\delta_{2r} + \delta_{2l}) \quad (6)$$

D. ROLL STABILIZATION

A larger ratio of error is permissible for roll control than for pitch and heave control. The reason is that the difference in elevation of the outer ends of the hydrofoils with respect to the water surface which the roll control regulates is much smaller in linear measure than the change in elevation of the keel due to pitch and heave motions. Unfortunately where roll is controlled, as in *High Tail*, through the difference in angle settings of two aft hydrofoils, the relation between the sensor movements and the hydrofoil angle settings required for proper roll control is much different than the relation between the sensor movements and the hydrofoil settings required for proper pitch and heave control. In other words, if the sensor forward of each aft hydrofoil were simply coupled directly to that hydrofoil there is no coupling ratio which would be proper for roll control and also proper for pitch and heave control for all wave lengths and wave velocity direction with respect to the boat course.

The angle of roll with respect to the water surface is determined by the difference between right and left sensor movements. Thus in the case of *High Tail* with two hydrofoils and two sensors aft and one hydrofoil and one sensor forward, the angle of roll is given by:

$$\gamma = \frac{\delta_{2r} - \delta_{2l}}{b_s} \quad (7)$$

Since a rolling moment or movement is produced by a difference between right and left aft hydrofoil angles, a restoring effect in roll without smoothing can be expressed by:

$$(\alpha_{ar} - \alpha_{a1}) = \frac{A_r}{b_s} (\delta_{2l} - \delta_{2r}) \quad (8)$$

First order smoothing of the same type as included in Equations (4) and (5) can be added to Equation (8) by replacing α_{ar} and α_{a1} with $(\alpha_{ar} + E \dot{\alpha}_{ar})$ and $(\alpha_{a1} + E \dot{\alpha}_{a1})$ respectively thus:

$$\alpha_{ar} + E \dot{\alpha}_{ar} - (\alpha_{a1} + E \dot{\alpha}_{a1}) = \frac{A_r}{b_s} (\delta_{2l} - \delta_{2r}) \quad (9)$$

Adding Equations (6) and (9) gives:

$$\alpha_{ar} + E \dot{\alpha}_{ar} = + \frac{w_a}{Kq} - \frac{A_a}{2l} (\delta_{2r} + \delta_{2l}) - \frac{A_r}{2b_s} (\delta_{2r} - \delta_{2l}) \quad (10)$$

Subtracting Equation (9) from Equation (6) gives:

$$\alpha_{a1} + E \dot{\alpha}_{a1} = + \frac{w_a}{Kq} - \frac{A_a}{2l} (\delta_{2r} + \delta_{2l}) + \frac{A_r}{2b_s} (\delta_{2r} - \delta_{2l}) \quad (11)$$

		FOLLOWING SEA										HEAD SEA												
λ	Feet	300	200	100	75	50	35	30	25	20	15	10	10	15	20	25	30	35	50	75	100	200	300	
$2y_0$	Feet	.20	13.33	6.67	5.0	3.33	2.33	2.0	1.67	1.33	1.0	.67	.67	1.0	1.33	1.67	2.0	2.33	3.33	5.0	6.67	13.33	20	
$A_t = .77$	$\frac{d_f}{d_a}$.20	.420	1.410	2.02	2.570	2.530	2.430	2.206	1.786	1.468	.784	.632	.828	1.052	1.048	.974	.936	1.030	1.234	1.350	1.320	1.172	
$A_a = 0$	$\frac{h_f}{h_a}$.032	.14	2.34	3.76	3.14	3.03	8.24	15.89	22.13	34.09	21.41	15.78	31.26	30.46	47.18	57.11	59.15	54.87	49.75	45.65	32.81	26.47	
$A_f = .55$	$\frac{d_f}{d_a}$.029	.11	.84	.96	6.29	11.93	11.79	9.02	16.57	28.15	28.36	27.31	29.69	33.84	15.02	9.04	20.99	38.08	41.36	39.41	30.86	25.89	
$A_a = 0$	$\frac{h_f}{h_a}$.029	.12	1.20	1.44	4.0	4.71	2.0	8.29	19.01	9.12	12.74	16.06	3.27	23.67	31.09	25.64	19.11	23.03	33.83	35.74	30.11	25.47	
$A_f = .33$	$\frac{d_f}{d_a}$.031	.13	1.81	2.81	6.71	11.31	11.28	9.80	16.92	26.71	27.30	28.11	30.78	31.55	14.94	12.32	21.68	35.9	38.40	36.80	30.60	26.30	
$A_a = 0$	$\frac{h_f}{h_a}$.031	.13	1.99	2.47	1.52	3.32	2.39	8.34	17.58	7.38	12.98	15.55	1.63	22.69	27.11	20.20	12.84	19.75	31.43	33.95	30.51	26.16	
$A_f = .88$	$\frac{d_f}{d_a}$.035	.16	4.53	9.34	11.24	11.44	10.73	10.28	17.51	25.36	26.31	28.89	31.79	29.48	1.140	1.280	1.382	1.528	2.082	2.842	3.410	4.872	5.288
$A_a = 0$	$\frac{h_f}{h_a}$.036	.17	4.53	8.25	6.45	2.53	2.64	9.04	16.63	6.02	13.3	15.10	.93	21.95	23.54	15.64	7.42	16.55	27.39	29.61	28.05	26.16	
$A_f = .77$	$\frac{d_f}{d_a}$.038	.574	1.144	1.498	2.058	2.330	2.324	2.148	1.756	1.558	.766	.622	.856	1.070	.968	.756	.594	.552	.698	.788	.836	.806	
$A_a = 0$	$\frac{h_f}{h_a}$.030	.12	1.56	2.03	.92	1.52	5.14	13.66	23.14	5.47	16.77	14.17	2.12	32.85	41.98	34.18	21.84	15.12	33.42	36.75	30.32	25.36	
$A_f = .99$	$\frac{d_f}{d_a}$.032	.136	2.24	3.75	3.84	2.97	7.79	14.75	19.50	37.15	16.78	19.75	25.81	26.21	52.67	65.1	66.40	56.69	49.48	45.46	32.63	26.39	
$A_a = 0$	$\frac{h_f}{h_a}$.032	.131	2.01	3.29	4.14	4.29	7.17	14.35	28.5	38.46	30.87	30.44	26.73	43.00	39.43	42.26	48.41	57.5	53.67	47.08	32.04	25.93	
$A_f = .55$	$\frac{d_f}{d_a}$.031	.126	1.71	2.44	1.70	2.32	5.93	13.88	22.82	4.45	17.33	13.58	3.48	33.24	40.80	31.84	18.35	15.45	35.00	38.00	30.75	25.55	

R-30

 $b = 22$ Feet $l = 11$ Feet $v = 22$ Knots $\Delta = 6000$ Lbs.

TABLE II PREDICTED FLYING DRAFT VARIANCES AND VERTICAL ACCELERATIONS OF HIGH TAIL IN WAVES

Equations (4), (10) and (11) express coupling relationships for control of pitch, heave and roll on a straight course with first order smoothing.

E. ROLL AND ELEVATION CONTROL IN TURNS

In order to reduce the side loads on both the struts and occupants of the boat it is desirable to use some banking of the boat in a turn. Full Banking for a boat of *High Tail's* beamwise dimensions is not compatible with minimum Hydrofoil Draft Increment for sharp turns because the hydrofoil on the outside of the turn would broach the surface. On *High Tail* about one half Full Banking is produced automatically by using the steering rotation of the forward hydrofoil as an added signal in the roll control for the rear foils. This scheme can be expressed by introducing the terms $-K_e \theta_c$ and $+K_e \theta_c$ into Equations (10) and (11) respectively as shown in Equations (13) and (14).

Even with one half Full Banking, lowering of the boat in a turn is used on *High Tail* to avoid adding to the Hydrofoil Draft Increment. This lowering can be expressed by introducing the term $-K_f \theta_b^2$ into Equation (4) and the term $-K_a \theta_b^2$ into each of the Equations (10) and (11), as shown in Equations (12), (13) and (14) respectively.

F. AUTOMATIC CLIMB AND DESCENT DURING ACCELERATION AND DECELERATION

As *High Tail* reaches Up Speed (Down Speed) the hull abruptly climbs to (descends from) the flying elevation automatically. This behavior can be expressed by subtracting the term A_e from each of the sensor displacements, δ_1 , δ_{2r} and δ_{2l} in the coupling Equations (4), (10) and (11), as shown in Equations (12), (13) and (14) respectively.

G. MANUAL ADJUSTMENTS FOR PAY LOAD AND TRIMMING

The term $\frac{w_f}{Kq}$ in Equation (4) may be approximated

by the quantity $\left[\frac{w_{fh}}{Kq} + B_f \right]$. The term $\frac{w_a}{Kq}$ in Equation (10) may be approximated by the quantity $\left[\frac{w_{ah}}{Kq} + B_{ar} \right]$.

The term $\frac{w_a}{Kq}$ in Equation (11) may be approximated by the

quantity $\left[\frac{w_{ah}}{Kq} + B_{al} \right]$. The first term in each of the

bracketed quantities above varies only with q . The last term represents manual adjustment for pay load. However, the last term can also represent adjustment for trimming the elevation, pitch or roll of the boat.

Trimming adjustments for such purposes would be more ideal if they were combined with the term A_e . But this would complicate the computer in a way that seems unnecessary unless it is important to have the sea condition adjustment independent of the trimming adjustment for any adjustment of elevation.

H. SPECIFIC COUPLING EQUATIONS AND HIGH TAIL'S CONTROL

The Equations (4), (10) and (11) modified as described in the preceding sections become the Specific Coupling Equations:

$$a_f + E \dot{a}_f = \frac{w_{fh}}{Kq} + B_f - \frac{A_f}{1} (\delta_1 - A_e + K_f \theta_b^2) \quad (12)$$

$$\begin{aligned} a_{ar} + E \dot{a}_{ar} &= \frac{w_{ah}}{Kq} + B_{ar} - \frac{A_a}{2l} (\delta_{2r} + \delta_{2l} - 2A_e \\ &\quad + 2K_a \theta_b^2) - \frac{A_r}{2b_w} (\delta_{2r} - \delta_{2l} + 2K_e \theta_c) \end{aligned} \quad (13)$$

$$\begin{aligned} a_{al} + E \dot{a}_{al} &= \frac{w_{ah}}{Kq} + B_{al} - \frac{A_a}{2l} (\delta_{2r} + \delta_{2l} - 2A_e \\ &\quad + 2K_a \theta_b^2) + \frac{A_r}{2b_w} (\delta_{2r} - \delta_{2l} + 2K_e \theta_c) \end{aligned} \quad (14)$$

High Tail's hydrofoil angle control continuously produces the angle settings a_f , a_{ar} and a_{al} , expressed by the Equations (12), (13) and (14). The sensors, computer, servos and their connections are all involved. The signals functionally related to the sensor displacements δ_1 , δ_{2r} and δ_{2l} , the steering angle θ , and the Dynamic Pressure q , are transmitted to the computer. The computer is adjusted manually for the sea condition and for pay load and trim. The sea condition adjustment is functionally related to the parameter A_a , and the pay load and trim adjustments are proportional to the parameters B_f , B_{ar} and B_{al} .

The computer produces the signals σ_f , σ_{ar} and σ_{al} which are proportional to the right hand members of the Specific Coupling Equations (12), (13) and (14) respectively. The signals σ_f , σ_{ar} and σ_{al} are transmitted as commands to the servos. The servos are designed to produce from these commands the hydrofoil angle settings a_1 , a_{ar} and a_{al} that satisfy the Specific Coupling Equations (12), (13) and (14).

I. FUNCTION UNITS

The term Function Unit as used herein designates a control component which produces a signal that is a prescribed function, other than a simple linear relationship, of the signal received. *High Tail* has five kinds of Function Units, sixteen units in all. The three servos are Smoothing Function Units. In the computer there are three more Smoothing Function Units, five Sine Function Units, two Steering Function Units, two Reciprocal Function Units and one Sequence Function Unit. Each of the five signals transmitted to the computer is received by one or more Function units.

The three sensor displacements are transmitted to the computer as angular displacements of the three sensor mounting shafts. Each of these angular displacements is approximately the arc sine of δ , δ being the downward vertical displacement of the sensor contact with the water surface. For each sensor shaft a Sine Function Unit is used to produce a signal proportional to δ from the angular displacement of the shaft. Each Sine Function Unit consists of a crank on the sensor shaft and a connecting rod transmitting the signal, with the angle between the crank and the connecting rod equal to the angle between the Sensor Radius and a line perpendicular to the Flying Bottom Plane.

A signal, proportional to the steering angle θ , is received by two Steering Function Units, one in the control for the forward hydrofoil and one in the control for the aft hydrofoils. Both Steering Function Units produce a signal proportional to the function θ_b^2 appearing in the Specific Coupling Equations (12), (13) and (14). In each unit the square function is produced with a crank and connecting rod on dead center at $\theta = 0$, and the discontinuities in the functional relation between θ and θ_b are produced with lost motion in the connection rod and with a related stop and spring. In addition, the Steering Function Unit for the aft hydrofoils produces a signal proportional to θ_c by means of an overload release equipped connecting rod. This connecting rod has a displacement lengthwise at one end proportional to the steering angle θ and a displacement lengthwise at the other end limited by stops corresponding to the limits on θ_c . The change in length of the connecting rod is accommodated by its overload release.

The characteristics of the servos as Smoothing Function Units is included in Appendix D.

The Servo Pressure*, which is the signal specifically related to the Dynamic Pressure q , is received by the three other Smoothing Function Units each of which consists of an accumulator connected to the Servo Pressure with a passage restricted by an orifice. The pressure in the accumulator is the Smoothed Servo Pressure. Because the smoothing effects of the Smoothing Function Units are not identical, there is a Smoothed Servo Pressure for each of the three Smoothing Function Units.

One such smoothed Servo Pressure is the signal to a Sequence Function Unit which is a battery of four pilot valves (only three are used on *High Tail*). These valves are closed (opened) in sequence as the Smoothed Servo Pressure is increased (decreased) by the increase (decrease) in forward speed of the boat. The corresponding

sequence of abrupt increases (decreases) in the three pilot pressures automatically extends (folds) the sensors at Sensor Down (Up) Speed, starts (stops) the hydrofoil angle control at Uncage (Cage) Speed, and introduces the abrupt climb to (descent from) the flying elevation at Up (Down) Speed during acceleration (deceleration). The climb (descent) operation is specifically related to the term A_e in the Specific Coupling Equations (12), (13) and (14).

The second such Smoothed Servo Pressure is transmitted to two Reciprocal Function Units, one in the control for the forward hydrofoil and one in the control for the aft hydrofoils. These Reciprocal Function Units produce identical signals functionally related to the terms $\frac{w_{fh}}{Kq}$ and $\frac{w_{ah}}{Kq}$ in the Specific Coupling Equations (12), (13)

and (14), the Smoothed Servo Pressure Q being the signal for the Dynamic Pressure q as explained in Section C of the Discussion of the Design of *High Tail* herein. These signals substantially eliminate change in Bottom Clearance with change in boat speed if the boat speed is above Up Speed.

Each Reciprocal Function Unit is a variable length connecting link in the computer mechanism which includes a nonlinear spring member loaded with a small low friction hydraulic cylinder to which the Smoothed Servo Pressure is connected. The arrangement is such that the Smoothed Servo Pressure determines the deflection of the nonlinear spring member and the latter determines the increase in length of the Reciprocal Function Unit as a connecting link which is the signal σ_h for both Reciprocal Function Units.

There is considerable freedom in the choice of the functional relation between load and deflection in the design of a nonlinear spring member. But there is a stability problem if the deflection must decrease with increase in load on the spring member. Such a relationship would be involved if the signal σ_h were to equal either the $C_f \frac{w_{fh}}{Kq}$ or $C_a \frac{w_{ah}}{Kq}$. To avoid the stability problem consider that the signal σ_h is given either by:

$$\sigma_h \equiv - C_f \left[\frac{w_{fh}}{Kq} \right] \quad (15)$$

or by:

$$\sigma_h \equiv - C_a \left[\frac{w_{ah}}{Kq} \right] \quad (16)$$

where C_f and C_a are positive constants determined by the dimensions of the control mechanism and the approximately constant ratio $\frac{Q}{q}$. With the relationships (15) and (16)

an increase in length of a Reciprocal Function Unit results from an increase in the Smoothed Servo Pressure so that there is no stability problem.

*The Servo Pressure has nothing to do with the servos as Smoothing Function Units.

TEST RESULTS AND PROPOSALS

A. FLYING DRAG

High Pockets was used to tow *High Tail* in the flying condition with the propeller of *High Tail* retracted above the mean water level. Because of the very small wake of *High Pockets*, this plan gives a direct accurate measure of drag. Severe winter weather set in before the towing tests could be finished; but a lift-drag ratio of 16.0 in very small waves was measured at 21.6 Knots, the model design speed. The maximum lift-drag ratio measured by towing *High Pockets* with the propeller shaft retracted was 17.3.

The power required to furnish high pressure hydraulic fluid to the control system at the designed speed in smooth water is estimated to be 0.35 horsepower. The brake horsepower corresponding to the drag measured at design speed is 40.2 horsepower as determined from *High Pockets*' data. Therefore, the control in smooth water at the design speed requires 0.87 per cent of the propulsion brake horsepower. For the most severe sea condition of Table II it is estimated that 3 horsepower would be required to furnish the high pressure hydraulic fluid to the control system. In the design of a prototype, relative power requirement could be substantially reduced.

The propeller shaft housing and strut add about 42.8 pounds to the drag of *High Tail* at 21.6 knots.* With the propeller shaft down in flying position and with .87% added to the drag as the drag equivalent of the control power requirement in smooth water, the corresponding Lift-Drag ratio for *High Tail* at 21.6 knots is 14.3.

B. FLYING PERFORMANCE IN WAVES

High Tail was operated in waves visually estimated 60 to 90 feet in wave length with maximum wave heights of 4 to 6 feet respectively. With computer settings of $A_f = .88$ and $A_a = 0$ in a head sea and $A_f = .88$ and $A_a = .88$ in a following sea, the performance as judged visually without recording instruments was about as predicted in Table II. With computer settings of $A_f = .88$ and $A_a = 0$ in a following sea, increased Flying Draft Variance caused brief racing of the engine due to broaching of the propeller as each wave was traversed. No such difficulty was observed with $A_f = .88$ and $A_a = .88$ in a following sea.

It was not possible in the test area to get a run of sufficient length for a conclusive test in any other heading. But full turns of about 14 boat lengths diameter were made in these same waves without difficulty. Turns of about 10 boat lengths diameter have been made on other occasions without difficulty in waves of about one foot height.

*This was computed by dividing the towing test drag observations on *High Pockets* at 24.2 and 27.5 knots by the square of these respective speeds, multiplying by 21.6 squared and taking the average of the two results.

C. PROPOSED METHOD OF SIMULTANEOUSLY RECORDING VERTICAL DISPLACEMENT, VERTICAL ACCELERATION, FLYING DRAFT AND WAVE PROFILE TRAVERSED

It is proposed:

- 1) to use accelerometers to produce signals for recording vertical accelerations of the boat;
- 2) to continuously integrate these accelerometer signals with a network to obtain the vertical displacements of the boat;
- 3) to use the sensor displacement signals as the Flying Draft signals; and
- 4) to subtract the sensor signals from the corresponding vertical displacement signals to obtain the wave profile signals.

An on-board oscilloscope would be used to record the above measurements and such additional measurements as: engine r.p.m., forward speed, relative speed with respect to the water, the angular positions of the hydrofoils and sensors, etc.

FABRICATION PROBLEMS

The hydrofoils, struts and mountings of *High Tail* were fabricated as hollow weldments from 17-7 PH stainless steel recommended and supplied for the application by the American Rolling Mill Company.

Considerable development was required in order to obtain the accurate shapes necessary for the hydrofoils and struts and on account of the peculiarities of the material.

After the boat was under tests for about two weeks the tests were interrupted to repair a fracture due to a defective weld in a sensor and to introduce automatic acceleration control. During this interruption the boat was stored about 500 feet from shore.

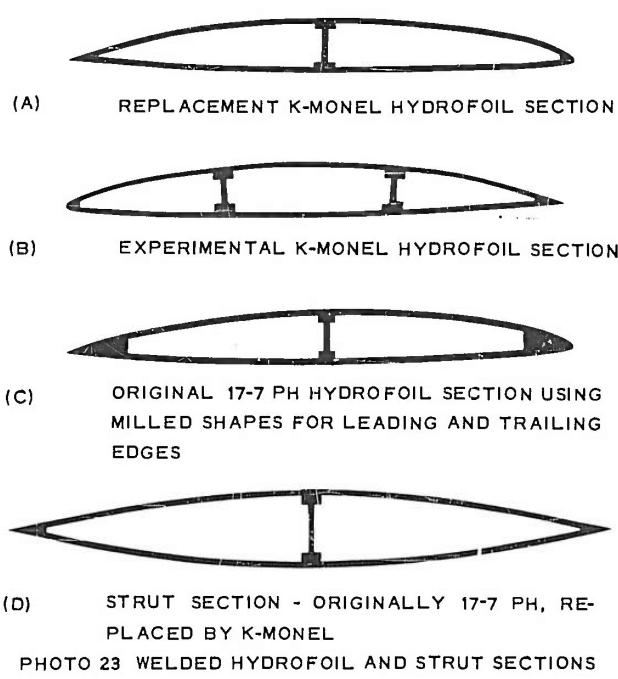


PHOTO 23 WELDED HYDROFOIL AND STRUT SECTIONS

Eight months later when the new control components were being installed, through cracks were noticed in the struts and hydrofoils. Some of the cracks were in the welds or extended into the welds, others were remote from any weld. The supplier of the material admitted that such cracks were to be expected under the circumstances despite recommendation of the material by the supplier.

The cracks were repaired and tests were resumed in fresh water. The struts and hydrofoils are now considered unsafe for extended operation in salt water.

Replacement components are under construction from K-Monel. Unfortunately the fabrication characteristics of K-Monel are not only different but in some respects unknown so that further development of fabrication techniques and changes in design detail have become necessary. Now, except for some mortality of components during fabrication due to occasional unexplained cracking in the K-Monel, no new fabrication problem is anticipated.

CONCLUSIONS

The conclusions A through I set forth in the Introduction of this report are based on design studies and tests carried out by the contractor other than those reported on herein. From the studies herein reported it is further concluded:

- J. that for given fore and aft spacing of hydrofoils, and surface sensing angle control, if any, and with a given wave train, the vertical accelerations, flight path and hydrofoil angle settings remain substantially unchanged with changes in total weight or in the division of weight between forward and aft hydrofoils so long as the hydrofoil area is also changed to hold unaltered the load per unit area of hydrofoil for any given elevation of the hull above the water surface; (Appendix D)
- K. that the dynamic behavior of a hydrofoil boat in waves should be such as to minimize Flying Draft Variance to reduce the Hydrofoil Draft Increment and thereby reduce drag, increase the pay load and enlarge areas of flying operation;
- L. that to fly in waves of any single wave length, with any single wave train velocity direction with respect to the boat course, and with an ideal control, requires a Hydrofoil Draft Increment that increases:
 - a) with decrease in the limiting tolerable vertical acceleration,
 - b) with decrease in the wave length-amplitude ratio, and
 - c) with increase in forward speed of the boat;
- M. that at a critical head sea wave length the maximum Flying Draft Variance occurs which added to the Lift Draft determines the smallest possible Hydrofoil Draft Increment;
- N. that the Flying Draft Variance required is a minimum if the control places the flight path in phase with the wave profile at the critical wave length even though the flight path is not in phase with wave profile at other wave lengths so long as the tolerable vertical acceleration is not exceeded and the Flying Draft Variance does not exceed that for the critical wave length;
- O. that the maximum Flying Draft Variance without aft hydrofoil pitch and heave control may be more than double that possible with aft hydrofoil surface sensing pitch and heave control and that therefore the omission of aft hydrofoil pitch and heave control may double the Hydrofoil Draft Increment required and correspondingly increase the propeller draft needed; (Appendix D)
- P. that to achieve minimum Flying Draft Variance and therefore minimum Hydrofoil Draft Increment the sensor lead must be near optimum; (Appendix D)
- Q. that Optimum Sensor Lead can apparently be practically obtained for a forward hydrofoil with a forward extending mechanical sensor and obtained for aft hydrofoils with rearward extending mechanical sensors provided the boat is not so large as to preclude adequately high natural frequencies of the sensor arms;
- R. that first order smoothing of the sensor signal is desirable to reduce high frequency accelerations; (Appendix D)
- S. that neither second order filtering nor large sensor movement relative to the water surface is desirable; (Appendix D) and
- T. that the slanting propeller shaft enclosed in streamlined housing is preferable to a right angle bevel gear propeller drive primarily because the main engine power is usable with the former drive in both floating and flying conditions without extra complication.

APPENDIX

A. APPROXIMATION FOR WAVE TRAIN VELOCITY

From Reference 14 the velocity c of a wave train is given by:

$$C^2 = \frac{g\lambda}{2\pi} \left(1 + \frac{4\pi^2 y_0^2}{\lambda^2} \right) \quad (20)$$

Under Assumption a, listed herein, $\frac{4\pi^2 y_0^2}{\lambda^2}$ may be neglected and (20) becomes the approximation:

$$C = \sqrt{\frac{g\lambda}{2\pi}} = 2.265 \sqrt{\lambda} \quad (21)$$

The wave train velocity along the boat course is given by:

$$c = \frac{C}{\cos\beta} \quad (22)$$

B. PITCH AND HEAVE CONTROL REQUIREMENTS FOR MINIMUM DRAFT AND LIMITED ACCELERATION

Consider a single hydrofoil flying through a sinusoidal wave train with a sinusoidal flight path of the same wave length and phase as the wave train. The frequency of wave encounter is given by:

$$\omega = \frac{2\pi(v \cos\beta + C)}{\lambda} = \frac{2\pi(v \cos\beta \pm 2.265 \sqrt{\lambda})}{\lambda} \quad (23)$$

With the amplitude of vertical hydrofoil displacement and acceleration expressed as h_0 and $(\ddot{h})_0$ respectively, the simple harmonic vertical motion of the hydrofoil requires that:

$$(\ddot{h})_0 = h_0 \omega^2 \quad (24)$$

The combination of Equations (21), (23) and (24) gives:

$$h_0 = \frac{(\ddot{h})_0 \lambda^2}{4\pi^2(v \cos\beta \pm 2.265 \sqrt{\lambda})^2} \quad (25)$$

The Flying Draft Variance d is of course twice the difference between the wave amplitude y_0 and flight path

amplitude h_0 or:

$$d = 2(y_0 - h_0) \quad (26)$$

Combining (25) and (26) and substituting $\frac{\lambda}{R}$ for y_0 gives:

$$d = \frac{2\lambda}{R} - \frac{(\ddot{h})_0 \lambda^2}{2\pi^2(v \cos\beta \pm 2.265 \sqrt{\lambda})^2} \quad (27)$$

Consider further that $(\ddot{h})_0$ is taken as the tolerable limit on the amplitude of acceleration which is $(\ddot{h})_t$. Then (27) becomes:

$$d_r = \frac{2\lambda}{R} - \frac{(\ddot{h})_t \lambda^2}{2\pi^2(v \cos\beta \pm 2.265 \sqrt{\lambda})^2} \quad (1)$$

which is the equation plotted in Chart I.

C. PITCH AND HEAVE CONTROL SIGNALS AND COUPLING PARAMETERS FOR ZERO FLYING DRAFT VARIANCE IN LONG WAVES

Referring to Diagram I, consider an angle controlled tandem hydrofoil boat flying in a wave train moving to the left. The xy rectangular coordinate system shown moves with the wave train. The boat velocity with respect to the moving coordinates is $v + c$.

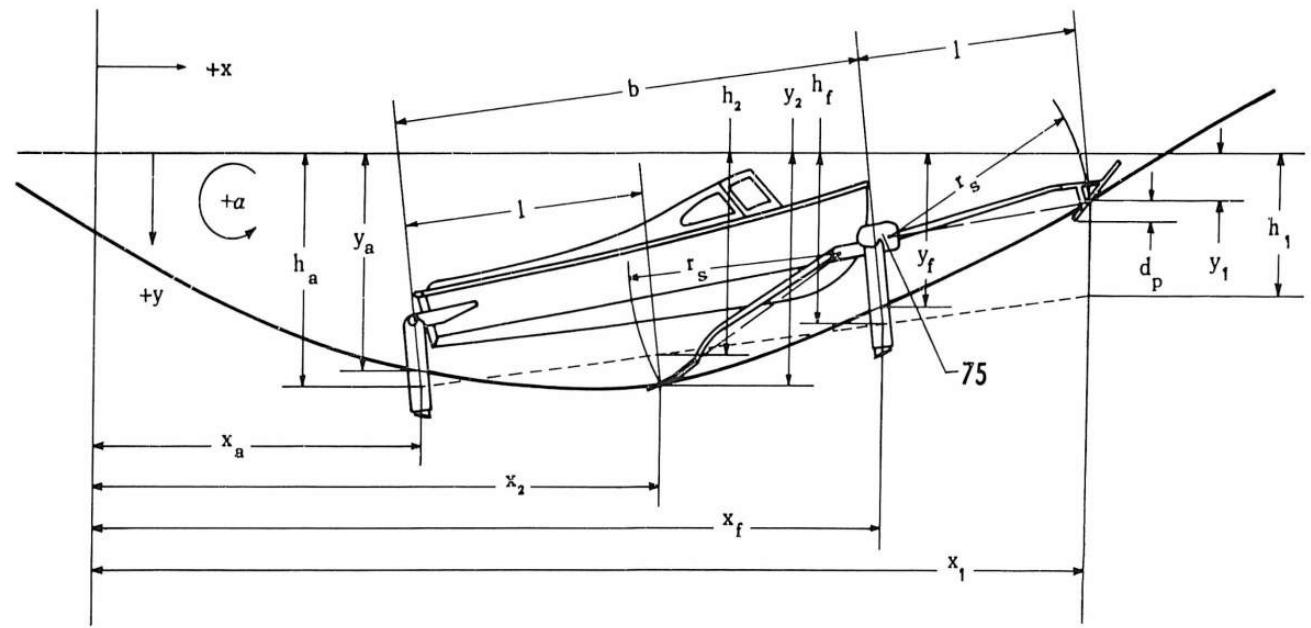
For the purposes of this section it is assumed that each hydrofoil is located at the center of percussion of the boat with respect to a center of oscillation at the other hydrofoil. Under this Assumption the boat may be considered as two concentrated masses with one mass at each hydrofoil

Since the velocity with respect to the moving coordinate system of a water particle near the water surface must be parallel to the water surface, the forward hydrofoil angle setting for zero Flying Draft Variance measured with respect to the setting for zero lift in smooth water is:

$$a_f = \frac{h_f - h_a}{b} - \frac{\dot{h}_f}{v+c} - \frac{w_f \ddot{h}_f}{gKq} + \frac{w_f}{Kq} \quad (30)$$

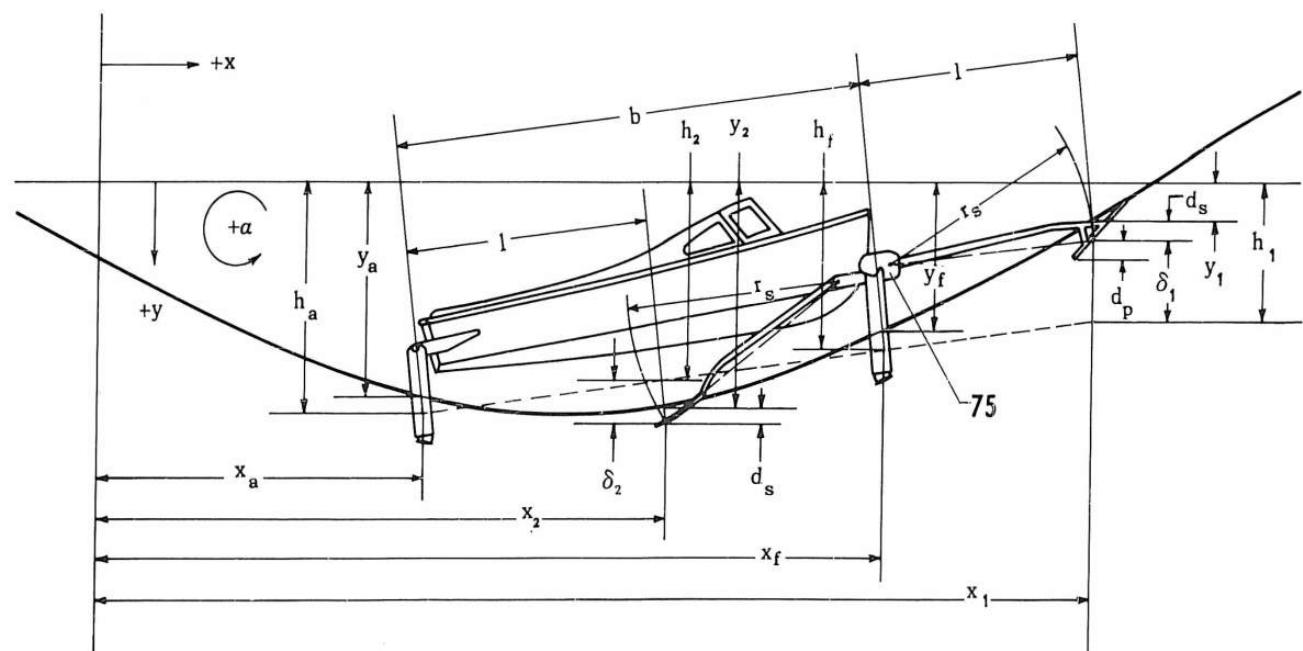
The hydrofoil angle setting of an aft hydrofoil for zero Flying Draft Variance is:

$$a_a = \frac{h_f - h_a}{b} - \frac{\dot{h}_a}{v+c} - \frac{w_a \ddot{h}_a}{gKq} + \frac{w_a}{Kq} \quad (31)$$



SYMBOL ILLUSTRATIONS WITH SENSOR PLANING POINT AT WATER SURFACE

DIAGRAM I



SYMBOL ILLUSTRATIONS WITH SENSOR PLANING POINT DEFLECTED FROM WATER SURFACE

DIAGRAM II

in view of the Assumption j listed herein, the wave surface depth at the points x_f and x_a can be expressed as a three term Taylor's expansion thus:

$$y_1 = y_f + \frac{dy_f}{dx_f} + \frac{l^2}{2} \frac{d^2y_f}{dx_f^2} \quad (32)$$

$$y_a = y_f - \frac{dy_f}{dx_f} + \frac{b^2}{2} \frac{d^2y_f}{dx_f^2} \quad (33)$$

$$y_2 = y_f - (b-1) \frac{dy_f}{dx_f} + \frac{(b-1)^2}{2} \frac{d^2y_f}{dx_f^2} \quad (34)$$

With zero Flying Draft Variance any h equals the y with the same subscript. Converting the derivations with respect to time to derivations with respect to x in (30) by using the boat velocity ($v+c$) with respect to the coordinate system and then combining the Equations (30) and (33):

$$a_f = \frac{w_f}{Kq} - \left[\frac{b}{2} + \frac{w_f(v+c)^2}{gKq} \right] \frac{d^2y_f}{dx_f^2} \quad (35)$$

Similarly the combination of Equations (31) and (33) gives:

$$a_a = \frac{w_a}{Kq} + \frac{dy_f}{dx_f} - \frac{b}{2} \frac{d^2y_f}{dx_f^2} - \frac{dy_a}{dx_a} - \frac{w_a(v+c)^2}{gKq} \frac{d^2y_a}{dx_a^2} \quad (36)$$

With $\frac{d^2y}{dx^2}$ constant in accordance with Assumption j listed herein:

$$\frac{d^2y_f}{dx_f^2} = \frac{d^2y_a}{dx_a^2} \quad (37)$$

and

$$\frac{dy_f}{dx_f} - \frac{dy_a}{dx_a} = b \frac{d^2y_f}{dx_f^2} \quad (38)$$

then (36) becomes:

$$a_a = \frac{w_a}{Kq} + \left[\frac{b}{2} - \frac{w_a(v+c)^2}{gKq} \right] \frac{d^2y_a}{dx_a^2} \quad (39)$$

With the sensors following the water surface in accordance with Assumption f, listed herein, the signal deflections of the sensors are:

$$\delta_1 = y_1 - h_1 \quad (40)$$

$$\delta_2 = y_2 - h_2 \quad (41)$$

From the geometry of the Diagram I,

$$h_1 = h_f \left(1 + \frac{1}{b} \right) - \frac{lh_a}{b} \quad (42)$$

$$h_2 = \frac{1}{b} h_f + \left(1 - \frac{1}{b} \right) h_a \quad (43)$$

Then using the conditions of zero Flying Draft Variance, Equations (32), (33), (40) and (42) combine to form:

$$\delta_1 = \frac{1}{2} (b+1) \frac{d^2y_f}{dx_f^2} \quad (44)$$

Similarly Equations (33), (34), (41) and (43) combine to form:

$$\delta_2 = \frac{1}{2} (1-b) \frac{d^2y_f}{dx_f^2} \quad (45)$$

Combining (35) and (44) to eliminate $\frac{d^2y_f}{dx_f^2}$ gives:

$$a_f = \frac{w_f}{Kq} - \left[\frac{b}{l(b+1)} + \frac{4w_f(v+c)^2}{v^2 \rho l K g (b+1)} \right] \delta_1 \quad (46)$$

Combining (39) and (45) to eliminate $\frac{d^2y_f}{dx_f^2}$ gives:

$$a_a = \frac{w_a}{Kq} - \left[\frac{b}{l(b-1)} - \frac{4w_a(v+c)^2}{v^2 \rho l K g (b-1)} \right] \delta_2 \quad (47)$$

From the definitions of A_{fn} and A_{an} Equations (46) and (47) become:

$$a_f = \frac{w_f}{Kq} - \frac{A_{fn}}{1} \delta_1 \quad (2)$$

$$a_a = \frac{w_a}{Kq} - \frac{A_{an}}{1} \delta_2 \quad (3)$$

D. ANALYSIS OF PITCH AND HEAVE MOTIONS IN WAVES

Referring to the Diagram II, the forward sensor is mounted on the sensor shaft 75. In smooth water, the Planing Point of the sensor ski rides at the water surface and the draft of the sensor ski is d_p . In waves, d_s is the instantaneous submergence of the Planing Point.

The upward hydrodynamic force perpendicular to the sensor radius r_s can be approximately expressed by:

$$N = K_p a v^2 (d_p + d_s) \left(a_p + \frac{\dot{d}_s}{v} \right) \quad (50)$$

Performing the indicated multiplications, neglecting $\frac{d_s \dot{d}_s}{v}$ on the basis of Assumption a, listed herein, Equation (50) becomes:

$$N = K_p a v^2 d_p a_p + K_p a v d_p \dot{d}_s + K_p a v^2 a_p d_s \quad (51)$$

Dynamic equilibrium in view of Assumption m, listed herein is expressed by:

$$N - n v^2 = - m_s \ddot{\delta}_1 \quad (52)$$

Eliminating N from Equations (51) and (52) gives:

$$\begin{aligned} K_p a v^2 d_p a_p - n v^2 + K_p a v d_p \dot{d}_s + \\ K_p a v^2 a_p d_s + m_s \ddot{\delta}_1 = 0 \quad (53) \end{aligned}$$

Taking a_p as constant in accordance with Assumption l,

listed herein, the first two terms of (53) represent equal and opposite forces which cancel. Replacing d_s with $(\delta_1 + h_1 - y_1)$ on the basis of Diagram II, $K_p a v d_p$ with C_s , and $K_p a v^2 a_p$ with K_s , Equation (53) becomes:

$$K_s (\delta_1 + h_1 - y_1) + C_s (\dot{\delta}_1 + \dot{h}_1 - \dot{y}_1) + m_s \ddot{\delta}_1 = 0 \quad (54)$$

A similar development applies to each aft sensor thus:

$$K_s (\delta_2 + h_2 - y_2) + C_s (\dot{\delta}_2 + \dot{h}_2 - \dot{y}_2) + m_s \ddot{\delta}_2 = 0 \quad (55)$$

If the sensors are designed so as to make the error in following the water surface neglectable, then Equations (54) and (55) may be considered filter Equations where, in either Equation, the sensor signal to the filter is $(y-h)$ and the filter output signal is δ .

From the geometry of the boat base line with respect to the mean water level:

$$h_1 = h_a + (h_f - h_a) \left(\frac{b+1}{b} \right) \quad (56)$$

$$h_2 = h_a + (h_f - h_a) \frac{1}{b} \quad (57)$$

The coupling Equations (2) and (3) synthesized in Appendix C, with the specific coefficients A_{fn} and A_{an} replaced with as yet undetermined coefficients A_f and A_a and with first order smoothing added, become:

$$a_f + E \dot{a}_f = \frac{w_f}{Kq} - \frac{A_f}{1} \delta_1 \quad (4)$$

$$a_a + E \dot{a}_a = \frac{w_a}{Kq} - \frac{A_a}{1} \delta_2 \quad (5)$$

First order smoothing, represented in Equations (4) and (5) by the terms with the coefficient E, is introduced in the control by the servo linkage and valve action in accordance with Assumption o, listed herein.

The upward force on each hydrofoil is determined by multiplying the angle of attack of the hydrofoil by $S_f q k$, thus:

$$F_f = \left[\frac{(h_a - h_f)}{b} + \bar{a}_f + \frac{\dot{h}_f}{v} - \frac{c}{v} \frac{dy_f}{dx_f} \right] S_f q k \quad (58)$$

$$F_a = \left[\frac{(h_a - h_f)}{b} + \bar{a}_a + \frac{\dot{h}_a}{v} - \frac{c}{v} \frac{dy_a}{dx_a} \right] S_a q k \quad (59)$$

Dynamic equilibrium of the boat mass in the vertical direction requires that:

$$F_f + F_a = -M \left[\ddot{h}_a + \frac{e}{b} (\ddot{h}_f - \ddot{h}_a) \right] \quad (60)$$

Dynamic equilibrium of the boat in rotation about a horizontal transverse axis through the center of gravity requires:

$$F_f (b-e) - F_a e = \frac{Mr^2}{b} (\ddot{h}_a - \ddot{h}_f) \quad (61)$$

Solving (60) and (61) for F_f and F_a gives:

$$F_f = -\frac{Me}{b} \left[\ddot{h}_a + \frac{e}{b} (\ddot{h}_f - \ddot{h}_a) \right] + \frac{Mr^2}{b^2} (\ddot{h}_a - \ddot{h}_f) \quad (62)$$

$$F_a = -\frac{M(b-e)}{b} \left[\ddot{h}_a + \frac{e}{b} (\ddot{h}_f - \ddot{h}_a) \right] - \frac{Mr}{b^2} (\ddot{h}_a - \ddot{h}_f) \quad (63)$$

Eliminating F_f from (58) and (62), F_a from (59) and (63), converting the derivatives with respect to x to derivatives with respect to time by using the boat velocity, ($v+c$), with respect to the moving coordinate system gives:

$$S_f q k \left[\frac{(h_a - h_f)}{b} + \bar{a}_f + \frac{\dot{h}_f}{v} - \frac{c}{v} \frac{\dot{y}_f}{(v+c)} \right] = -\frac{Me}{b} \left[\ddot{h}_a + \frac{e}{b} (\ddot{h}_f - \ddot{h}_a) \right] + \frac{Mr^2}{b^2} (\ddot{h}_a - \ddot{h}_f) \quad (64)$$

$$S_a q k \left[\frac{(h_a - h_f)}{b} + \bar{a}_a + \frac{\dot{h}_a}{v} - \frac{c}{v} \frac{\dot{y}_a}{(v+c)} \right] = -\frac{M(b-e)}{b} \left[\ddot{h}_a + \frac{e}{b} (\ddot{h}_f - \ddot{h}_a) \right] - \frac{Mr^2}{b^2} (\ddot{h}_a - \ddot{h}_f) \quad (65)$$

Using Equations (56), (57), (4) and (5) to eliminate h_1 , h_2 , δ_1 and δ_2 from (54) and (55) gives:

$$K_s \left[-\frac{(\bar{a}_f + E\dot{a}_f) l}{A_f} + h_a + (h_f - h_a) \frac{(b+1)}{b} - y_1 \right] +$$

$$C_s \left[-\frac{(\dot{a}_f + E\ddot{a}_f) l}{A_f} + \dot{h}_a + (h_f - h_a) \frac{(b+1)}{b} - \dot{y}_1 \right] -$$

$$\frac{m_s (\ddot{a}_f + E\ddot{a}_f) l}{A_f} = 0 \quad (66)$$

$$K_s \left[-\frac{(\bar{a}_a + E\dot{a}_a) l}{A_a} + h_a + (h_f - h_a) \frac{l}{b} - y_2 \right] +$$

$$C_s \left[-\frac{(\dot{a}_a + E\ddot{a}_a) l}{A_a} + \dot{h}_a + (h_f - h_a) \frac{l}{b} - \dot{y}_2 \right] -$$

$$\frac{m_s (\ddot{a}_a + E\ddot{a}_a) l}{A_a} = 0 \quad (67)$$

Eliminating \bar{a}_f from Equations (64) and (66) and \bar{a}_a from Equations (65) and (67), replacing t with the dimensionless r , using the dimensionless operator p_d , and otherwise converting to dimensionless form, gives:

$$\left[\frac{(b+1)(1+C_d p_d) A_f}{1(1+C_d p_d + m_d p_d^2)(1+E_d p_d)} - 1 + p_d + M_f \left(\frac{e}{b} + \frac{r^2}{be} \right) p_d^2 \right] \frac{h_f}{b} -$$

$$\left[\frac{(1+C_d p_d) A_f}{(1+C_d p_d + m_d p_d^2)(1+E_d p_d)} - 1 - M_f \left(\frac{b-e}{b} - \frac{r^2}{be} \right) p_d^2 \right] \frac{h_a}{b} =$$

$$\left[\frac{\lambda(1+C_d p_d) A_f}{1(1+C_d p_d + m_d p_d^2)(1+E_d p_d)} \right] \frac{y_1}{\lambda} + \left[\frac{\lambda c}{b(v+c)} p_d \right] \frac{y_f}{\lambda} \quad (68)$$

$$\left[\frac{(1+C_d p_d) A_a}{(1+C_d p_d + m_d p_d^2)(1+E_d p_d)} - 1 + \right]$$

$$\begin{aligned}
& \left[M_a \left(\frac{e}{b} - \frac{r^2}{b(b-e)} \right) p_d^2 \right] \frac{h_f}{b} - \\
& \left[\frac{(1-b)(1+C_d p_d) A_a}{1(1+C_d p_d + m_d p_d^2)(1+E_d p_d)} - 1 - p_d - \right. \\
& \quad \left. M_a \left(\frac{b-e}{b} + \frac{r^2}{b(b-e)} \right) p_d^2 \right] \frac{h_a}{b} = \\
& \left[\frac{\lambda(1+C_d p_d) A_a}{1(1+C_d p_d + m_d p_d^2)(1+E_d p_d)} \right] \frac{y_2}{\lambda} + \\
& \quad \left[\frac{\lambda c}{b(v+c)} p_d \right] \frac{y_a}{\lambda} \quad (69)
\end{aligned}$$

Setting y_1 , y_2 , y_f and y_a all equal to zero in Equations (68) and (69) and applying the criteria of Routh (Reference 13) indicates that the system represented is stable for the dimensions of *High Tail* and the ranges of other parameters of interest.

Equations (68) and (69) may be abbreviated as:

$$Z_3 \frac{h_f}{b} + Z_4 \frac{h_a}{b} = Y_1 \quad (70)$$

$$Z_5 \frac{h_f}{b} + Z_6 \frac{h_a}{b} = Y_2 \quad (71)$$

Solving Equations (70) and (71) for the boat motions gives:

$$\frac{h_f}{b} = \frac{Y_1 Z_6 - Y_2 Z_4}{Z_3 Z_6 - Z_5 Z_4} \quad (72)$$

$$\frac{h_a}{b} = \frac{Y_2 Z_3 - Y_1 Z_5}{Z_3 Z_6 - Z_5 Z_4} \quad (73)$$

With the wave length λ , the functionally related wave velocity c , the control parameters C_d , m_d , E_d , A_f and A_a , the mass M , the radius of gyration of the boat r , and the dimensions of the boat all given, the Flying Draft Variances d_f and d_a and the other variables are determined by solving (72) and (73) with the well known method of substituting $\omega_n \sqrt{-1}$ for p_d and dropping the imaginary component of the result.

Equations (72) and (73) thus become the basis of finding optimums, considering all wave lengths λ and corresponding wave velocities c , for the sensor lead 1, the control coupling factors A_f and A_a , the damping and inertia parameters C_d and m_d , and the smoothing parameter E .

To examine the effect of changes in the total weight and changes in the division of weight between forward and aft hydrofoils consider the case in which the mass and mass distribution of the boat are approximately represented by two masses, one at each hydrofoil. In this case both of the coefficients $\left(\frac{b-e}{b} - \frac{r^2}{be} \right)$ and $\left(\frac{e}{b} - \frac{r^2}{b(b-e)} \right)$ in Equations (68) and (69) become zero and each of the coefficients $\left(\frac{e}{b} + \frac{r^2}{be} \right)$ and $\left(\frac{b-e}{b} + \frac{r^2}{b(b-e)} \right)$ in the same equations become one. Thus Equations (68) and (69) become independent of the total mass or the division of mass between forward and aft hydrofoils so long as the loadings per square foot on the forward and aft hydrofoils, represented by M_f and M_a respectively, are unchanged.

Study of the effect of introducing a signal filter on the basis of Equations (72) and (73) and the choice of the parameters C_d and m_d indicates that an improvement in control operation is possible for a given wave length. But such improvement does not appear practical because of the poor operation indicated at other wave lengths, which might be simultaneously present. This same mathematical result indicates that large errors in sensor following are a disadvantage. It is, therefore, concluded that C_s and m_s should be small. Setting C_s and m_s equal to zero, and introducing the approximation mentioned above of representing the boat mass as two concentrated masses, one at each hydrofoil, in Equations (68) and (69), gives:

$$\left[\frac{(b+1) A_f}{1(1+E_d p_d)} - 1 + p_d + M_f p_d^2 \right] \frac{h_f}{b} -$$

$$\left[\frac{A_f}{(1+E_d p_d)} - 1 \right] \frac{h_b}{b} =$$

$$\frac{\lambda}{1(1+E_d p_d)} \frac{A_f}{\lambda} \frac{y_1}{\lambda} + \frac{\lambda}{b(v+c)} p_d \frac{y_f}{\lambda} \quad (74)$$

$$\left[\frac{A_a}{(1 + E_d p_d)} - 1 \right] \frac{h_f}{b} - \left[\frac{(1-b) A_a}{1(1 + E_d p_d)} - 1 - p_d - M_a p_d^2 \right] \frac{h_a}{b} = \frac{\lambda}{1(1 + E_d p)} \frac{A_a}{\lambda} \frac{y_2}{\lambda} + \frac{\lambda}{b(v+c)} p_d \frac{y_a}{\lambda}$$

the solutions of which are obtained with the same method
as used for Equations (68) and (69).

1

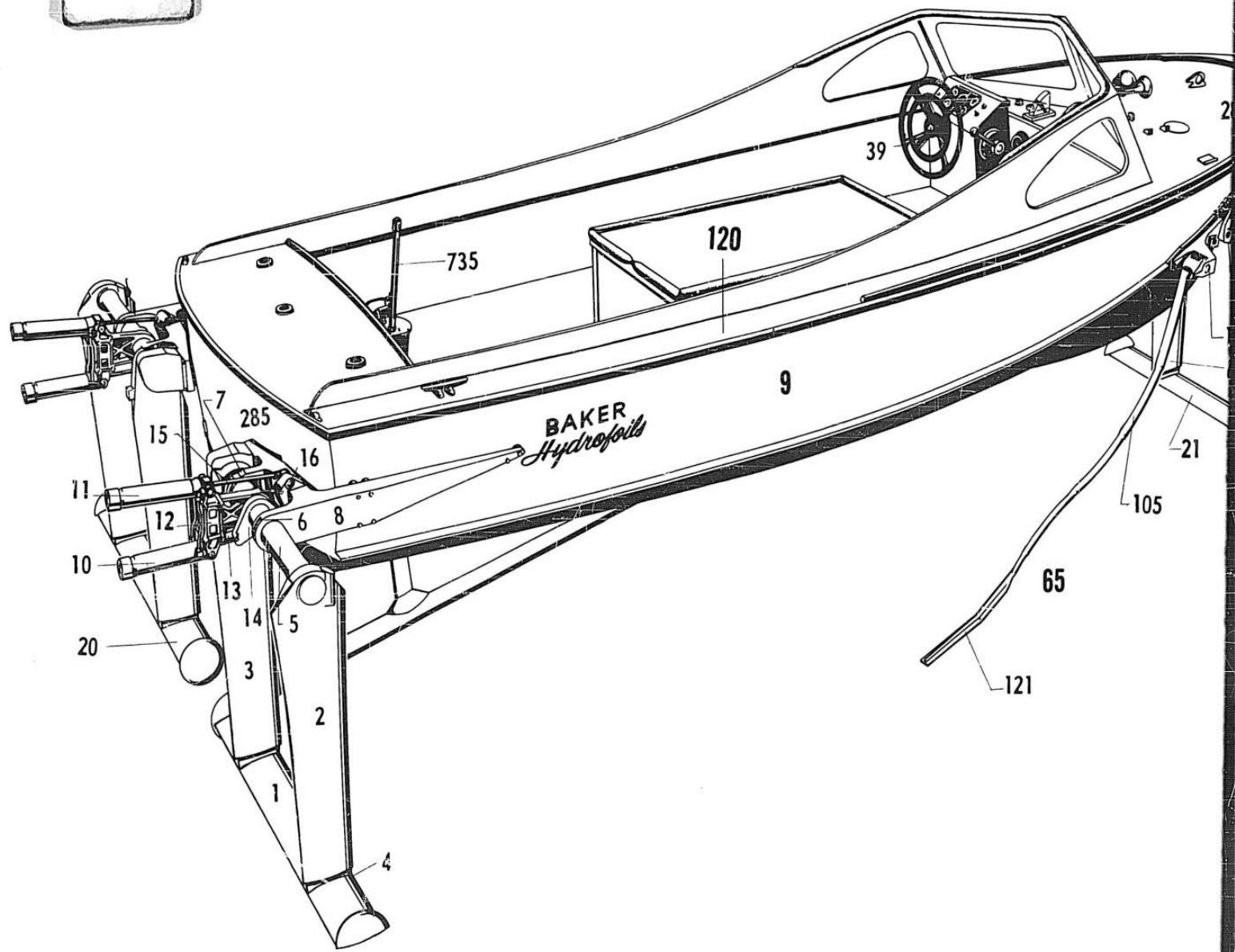
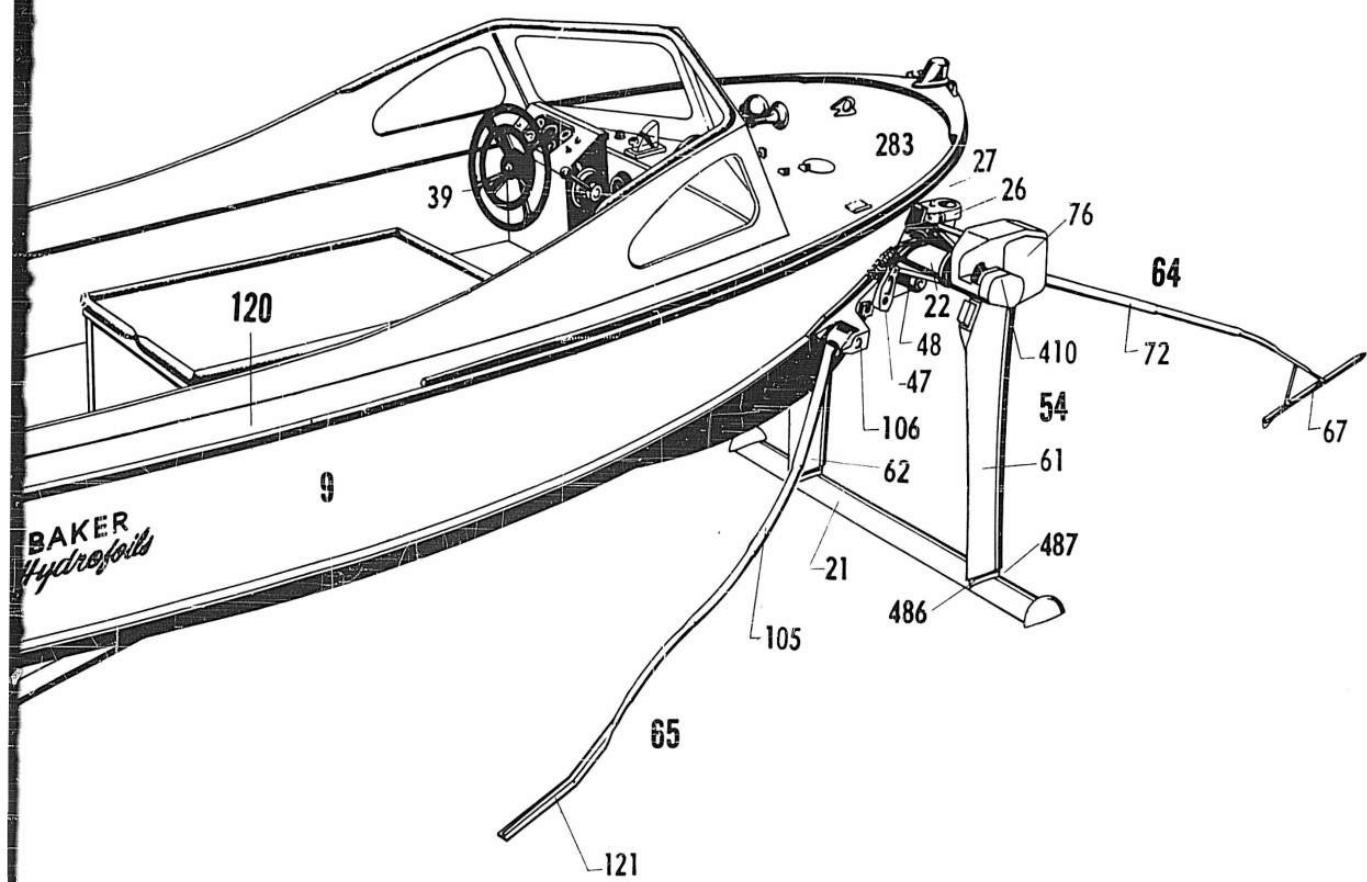


FIG. 1 ENTIRE BOAT SUSPENDED WITH HYDROFOILS, SENSORS AND PROPELLER EXTENDED, PERSPECTIVE DRAWING, .REA



2

HYDROFOILS, SENSORS AND PROPELLER EXTENDED, PERSPECTIVE DRAWING, REAR QUARTER VIEW

1

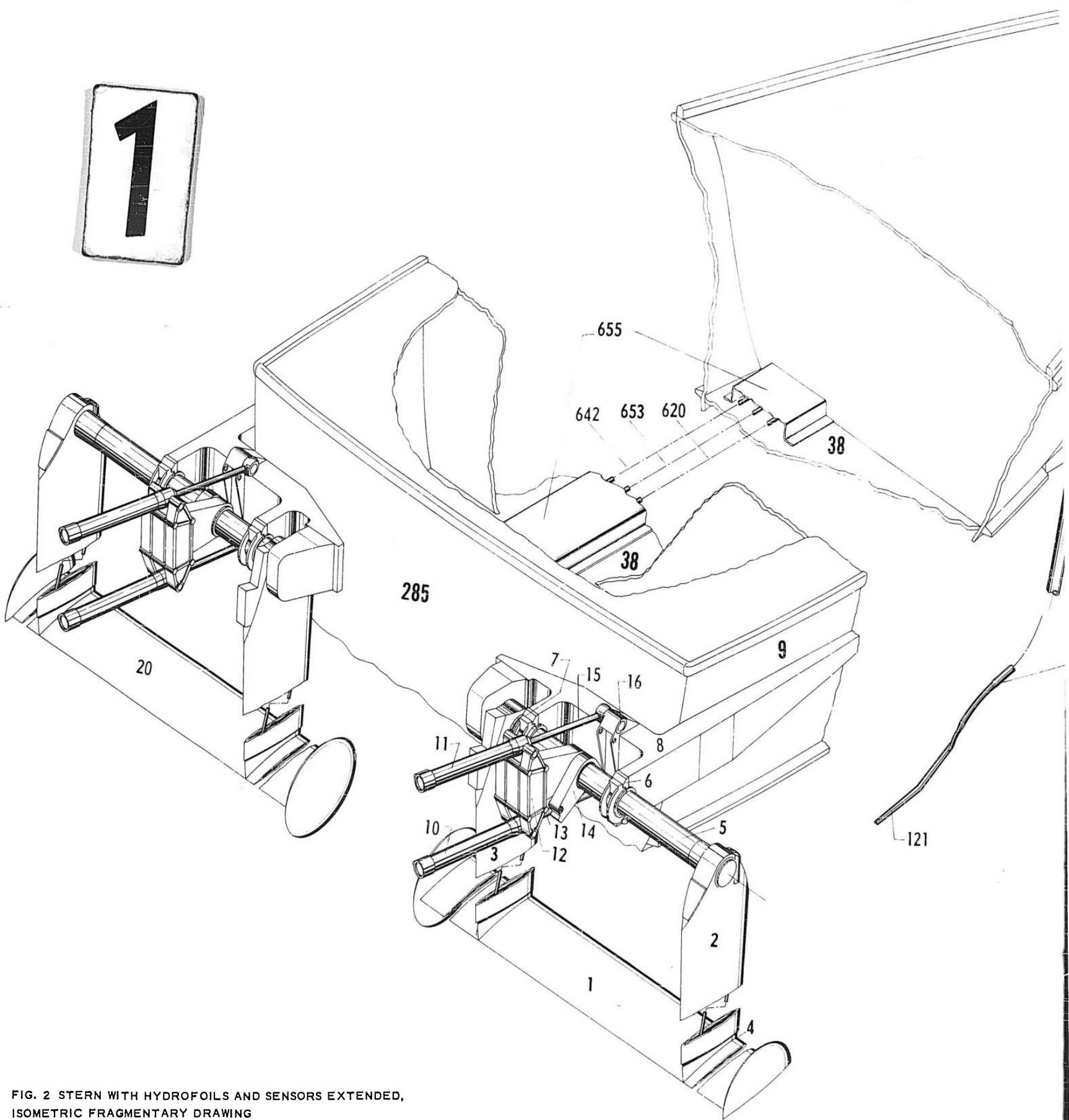


FIG. 2 STERN WITH HYDROFOILS AND SENSORS EXTENDED,
ISOMETRIC FRAGMENTARY DRAWING

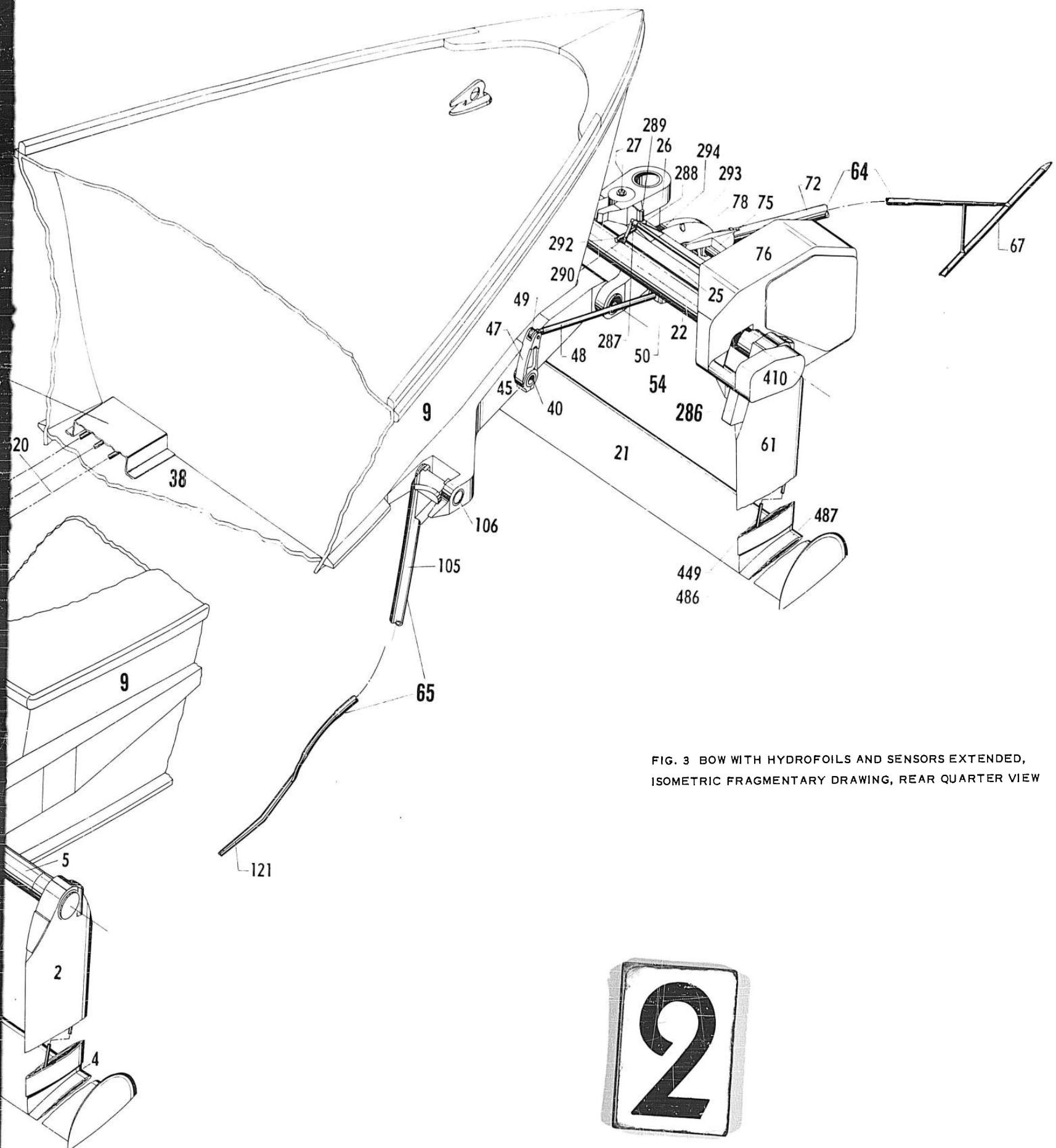
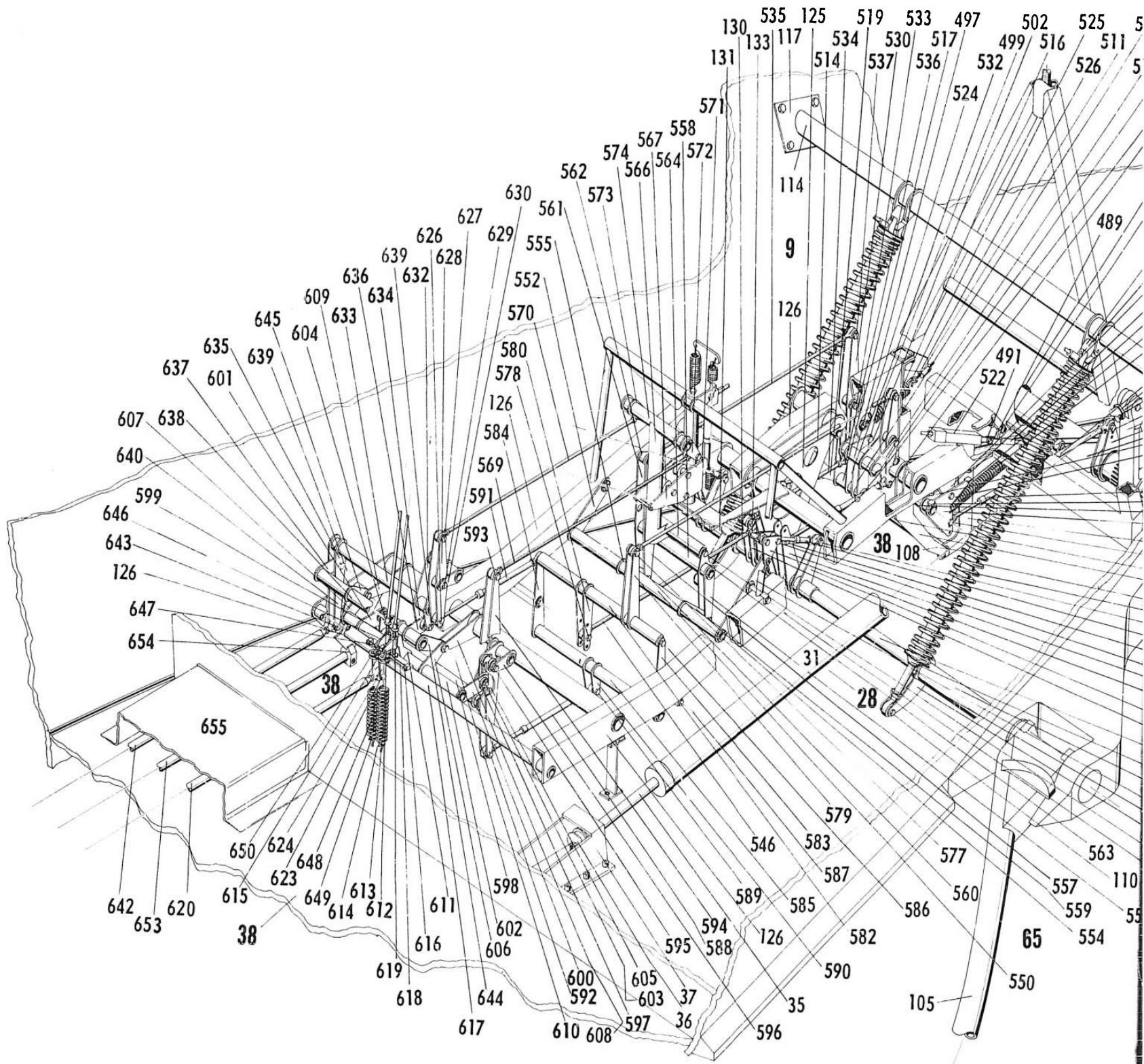
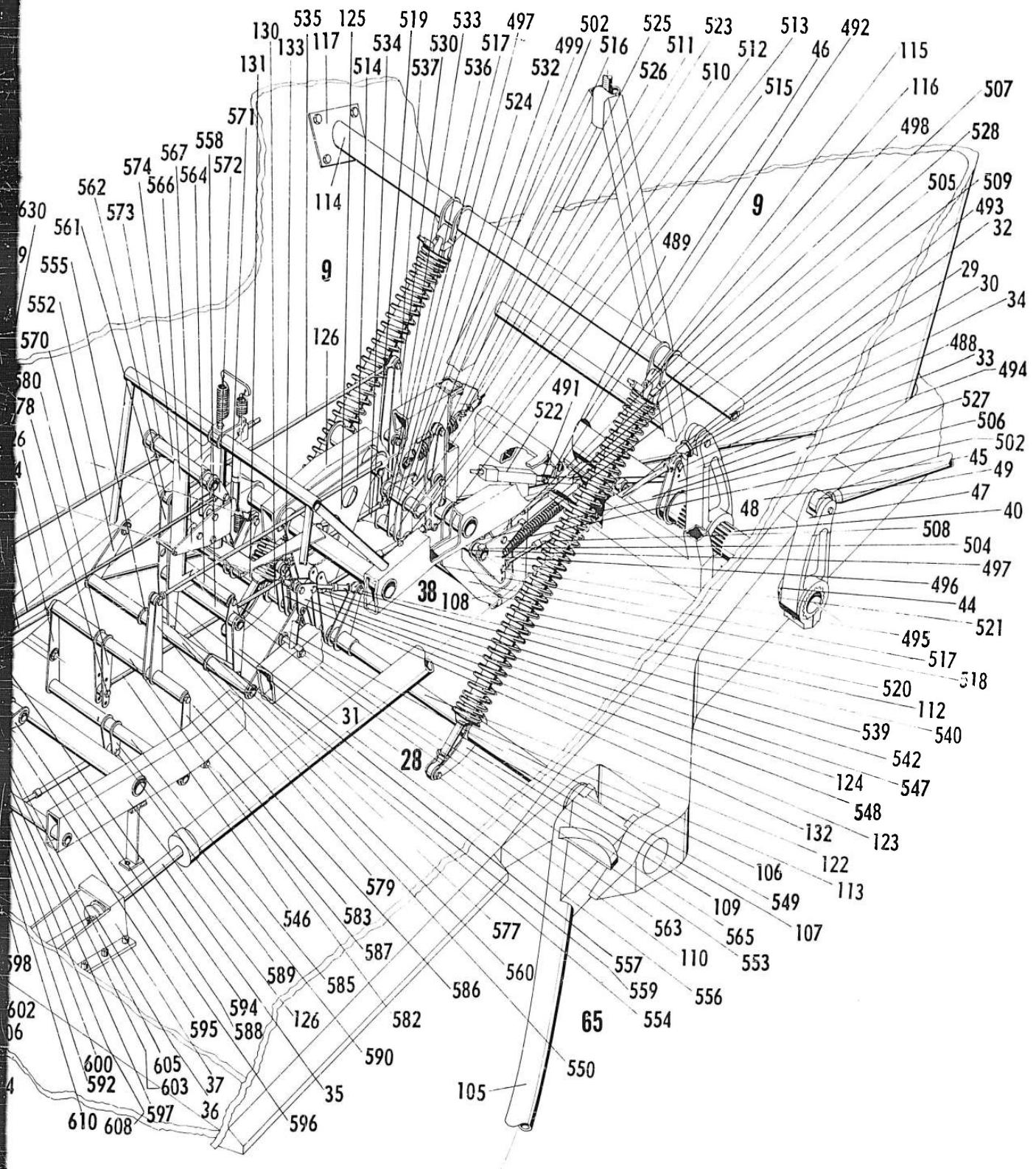


FIG. 3 BOW WITH HYDROFOILS AND SENSORS EXTENDED,
ISOMETRIC FRAGMENTARY DRAWING, REAR QUARTER VIEW



1

FIG. 4 BOW COMPARTMENT ISOMETRIC FRAGMENTARY DRAWING WITH DISTORTION



2

COMPARTMENT ISOMETRIC FRAGMENTARY DRAWING WITH DISTORTION

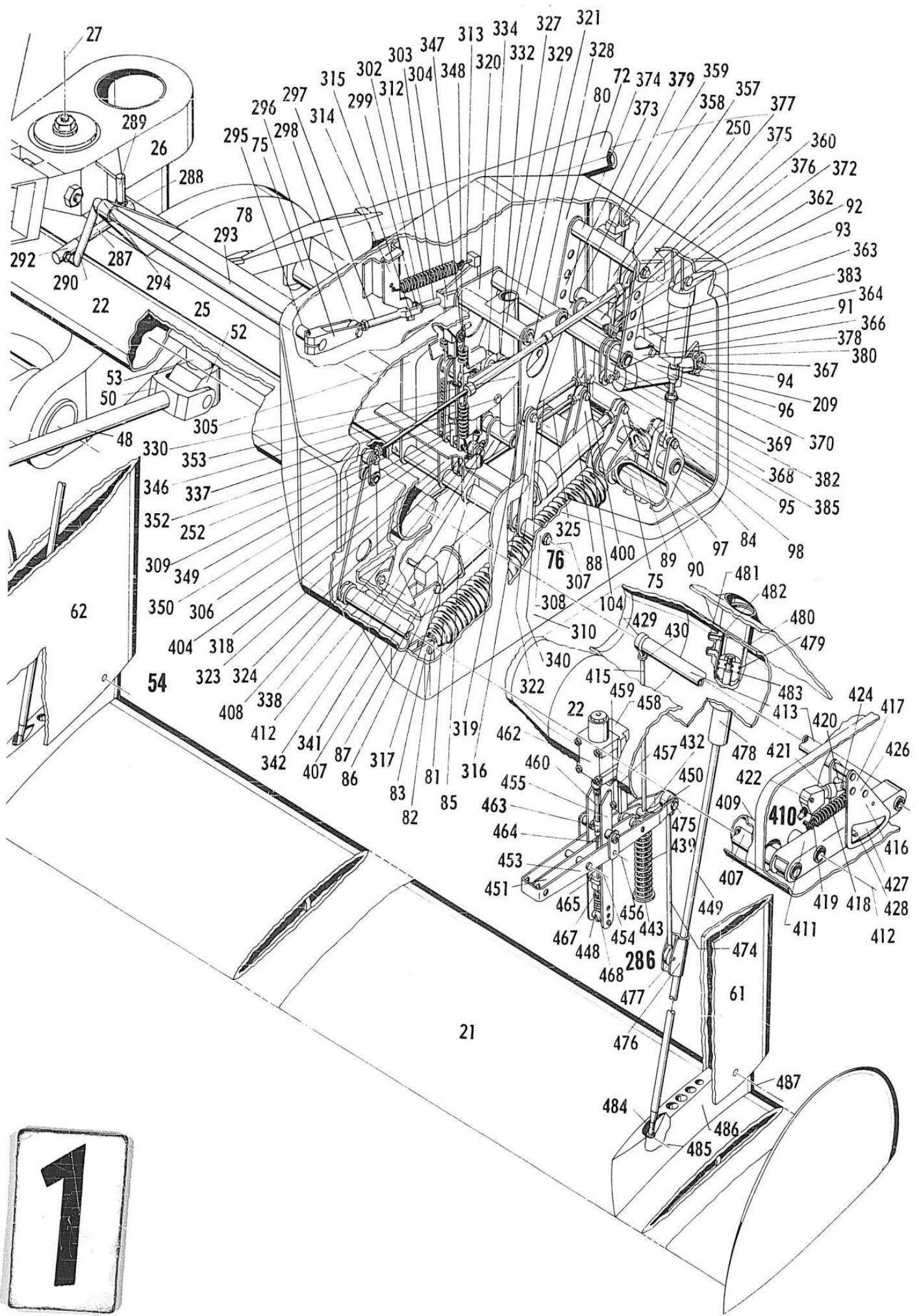


FIG. 5 STEERING, FOR
ISOMETRIC FRA

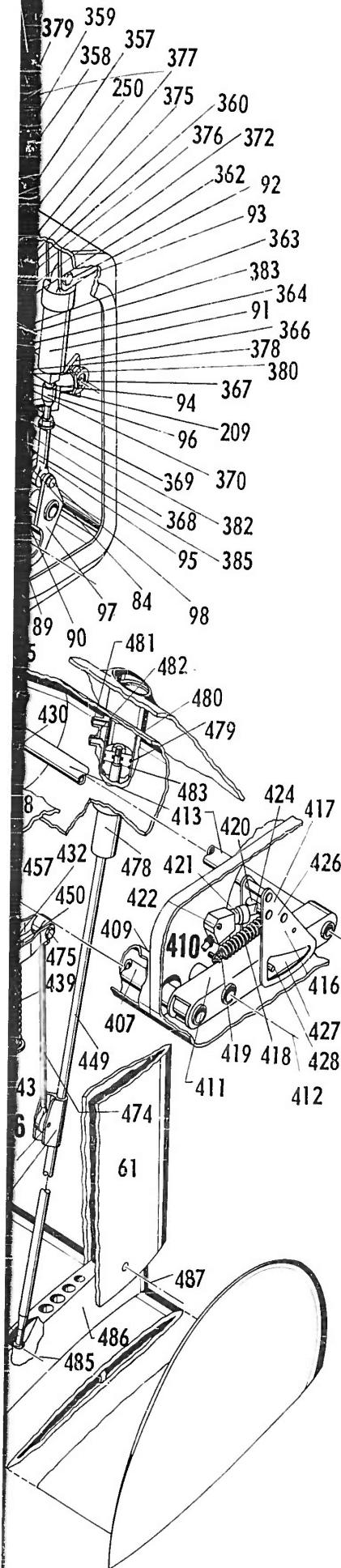


FIG. 5 STEERING, FORWARD SENSOR CONTROL, FORWARD COMPUTER SECTION, FORWARD CAGING AND SERVO,
ISOMETRIC FRAGMENTARY DRAWING WITH DISTORTION

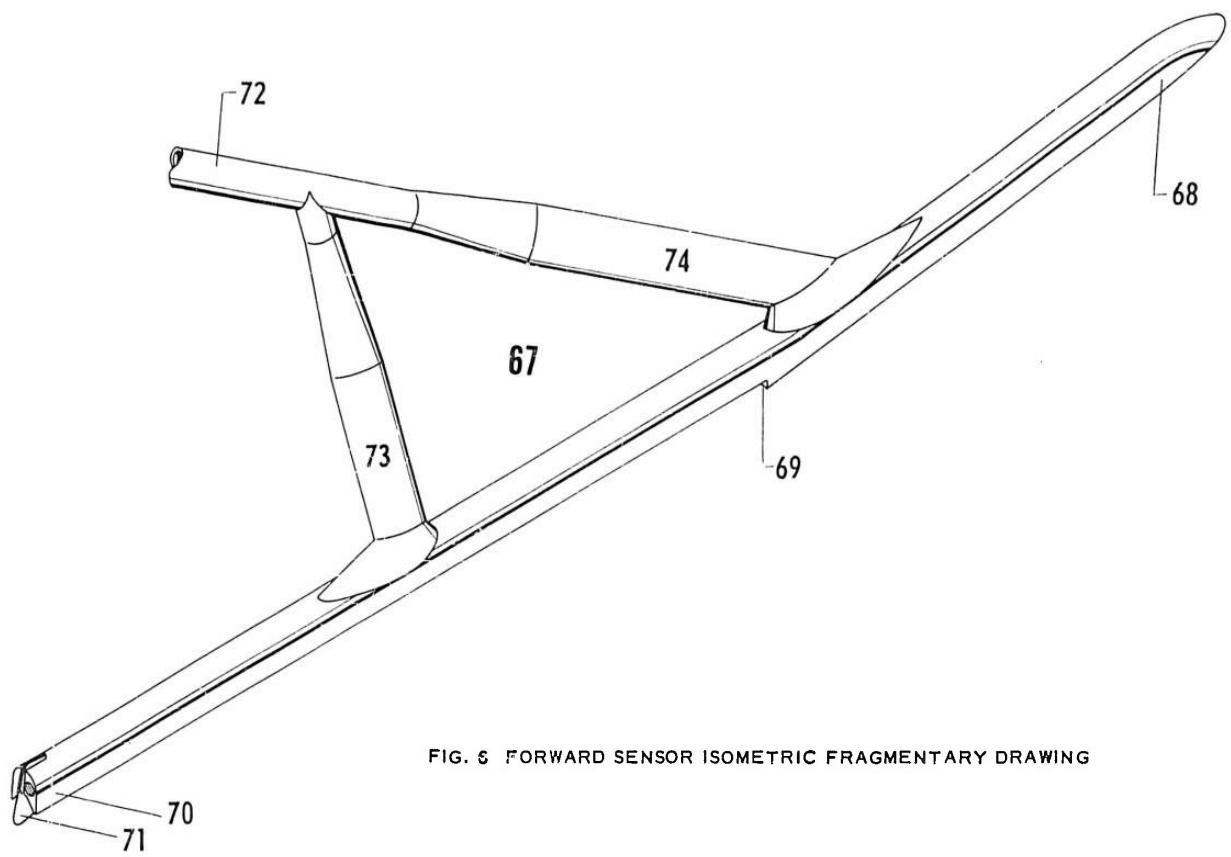


FIG. 6 FORWARD SENSOR ISOMETRIC FRAGMENTARY DRAWING



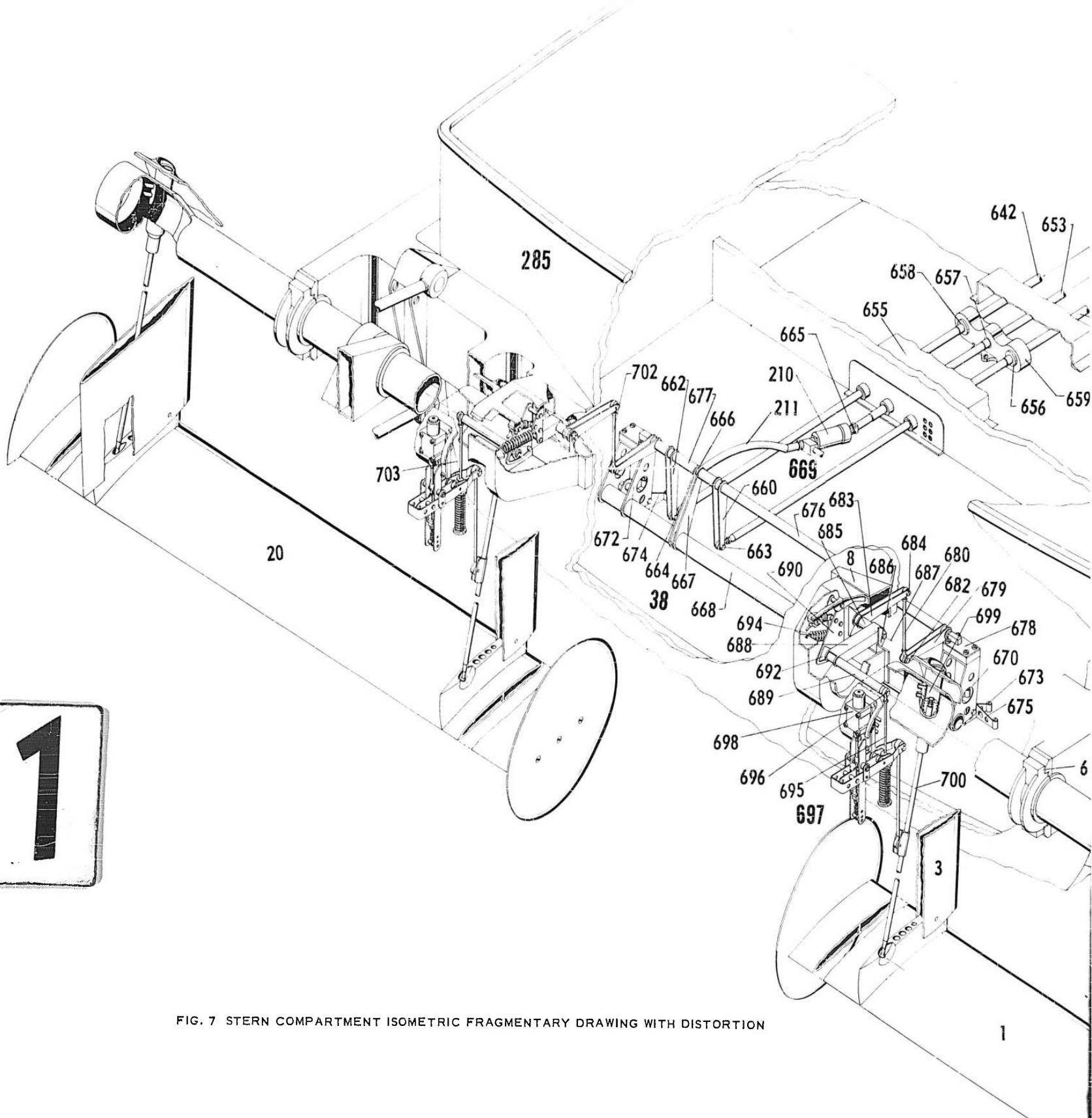
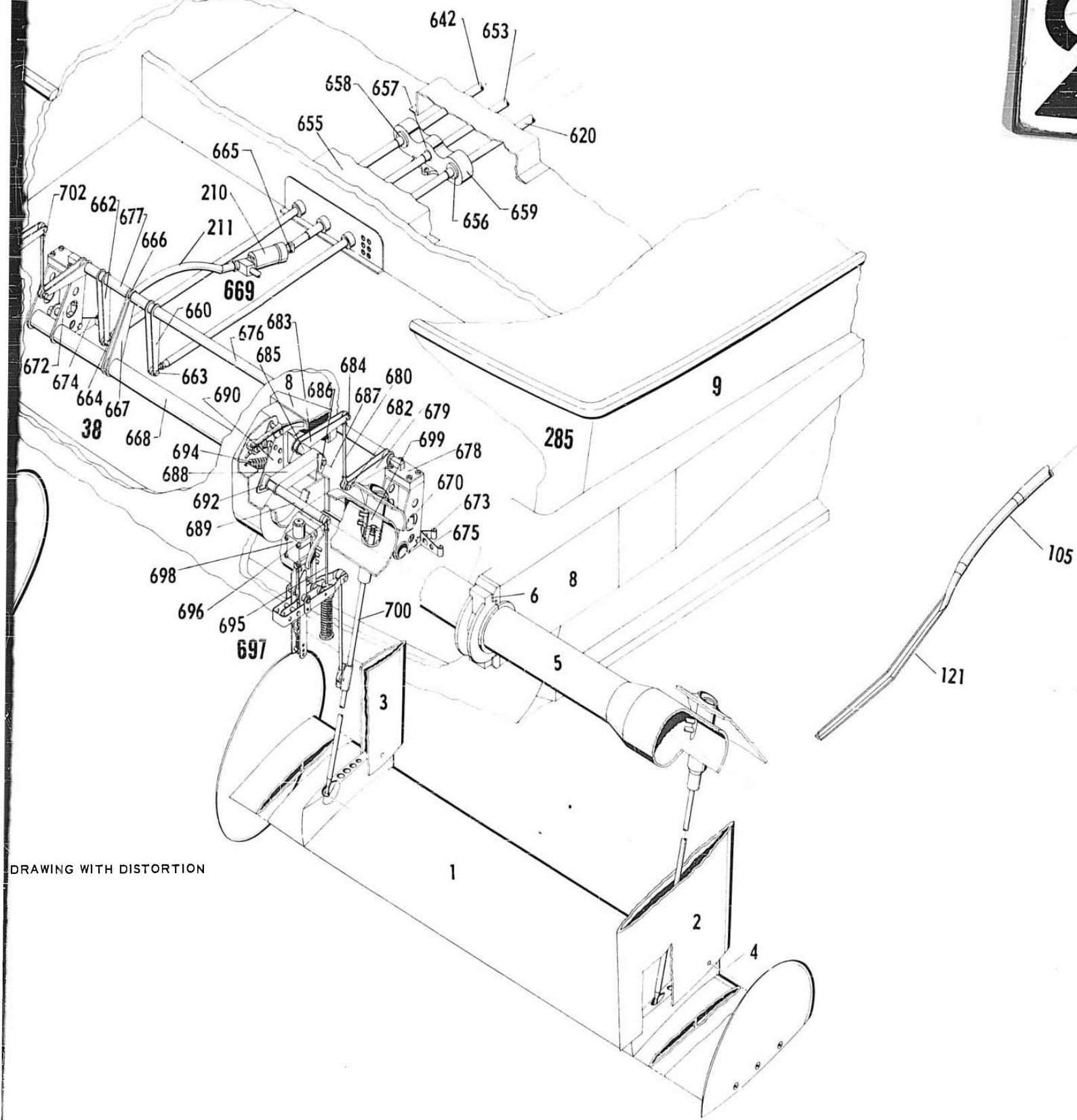


FIG. 7 STERN COMPARTMENT ISOMETRIC FRAGMENTARY DRAWING WITH DISTORTION



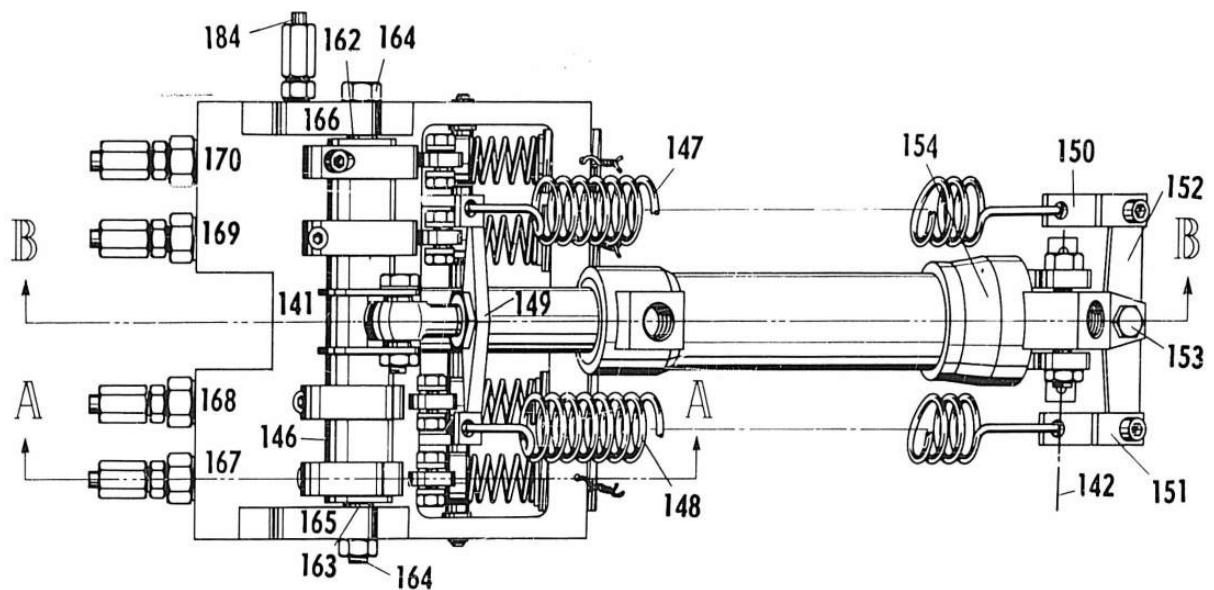


FIG. 8 SEQUENCE FUNCTION UNIT - FRAGMENTARY DRAWING

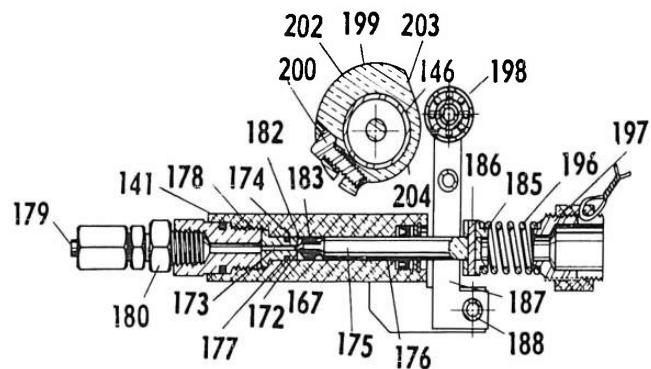


FIG. 9 SEQUENCE FUNCTION UNIT - DRAWING PARTIAL SECTION ALONG THE LINE AA OF FIGURE 8

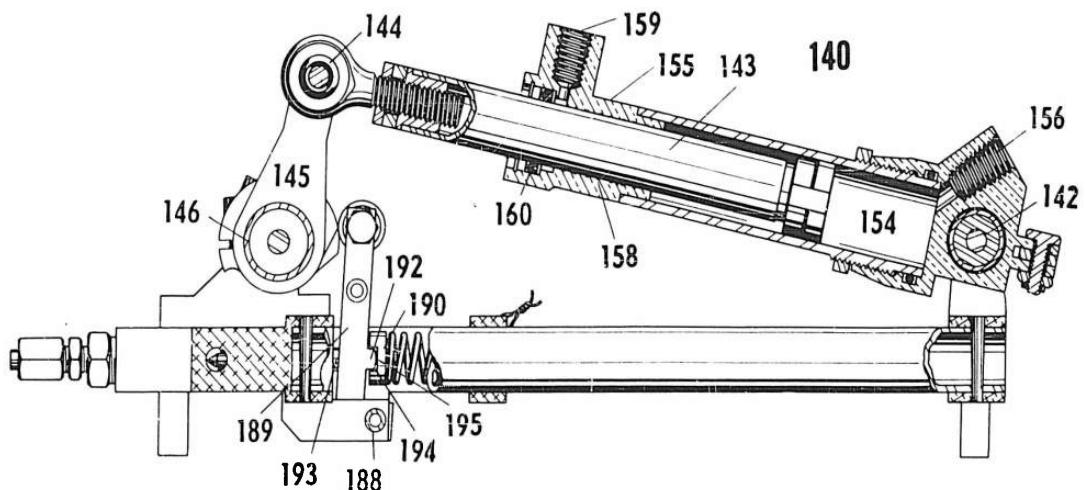


FIG. 10 SEQUENCE FUNCTION UNIT - DRAWING PARTIAL SECTION ALONG THE LINE BB OF FIGURE 8

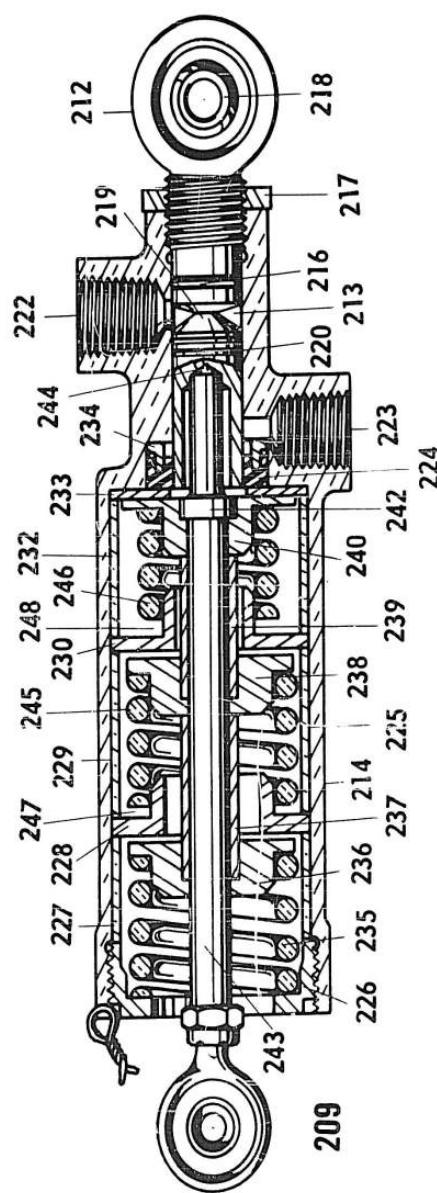


FIG. 11 RECIPROCAL FUNCTION UNIT - DRAWING PARTIAL SECTION ALONG AXIS

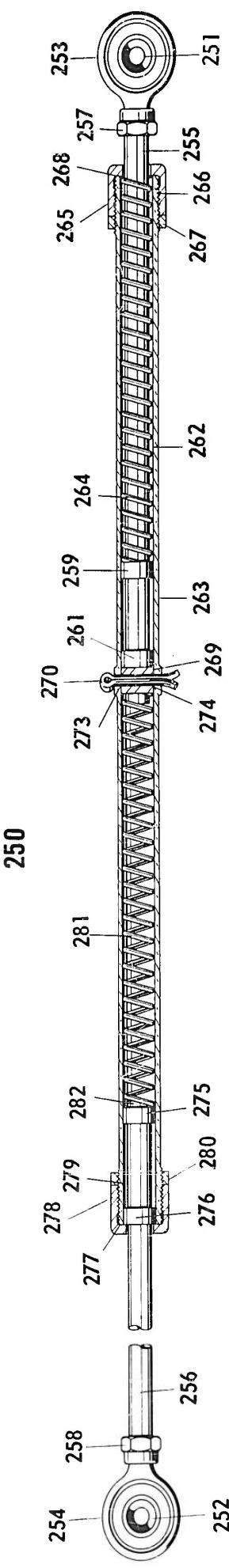


FIG. 12 OVERLOAD RELEASE - DRAWING PARTIAL SECTION ALONG AXIS

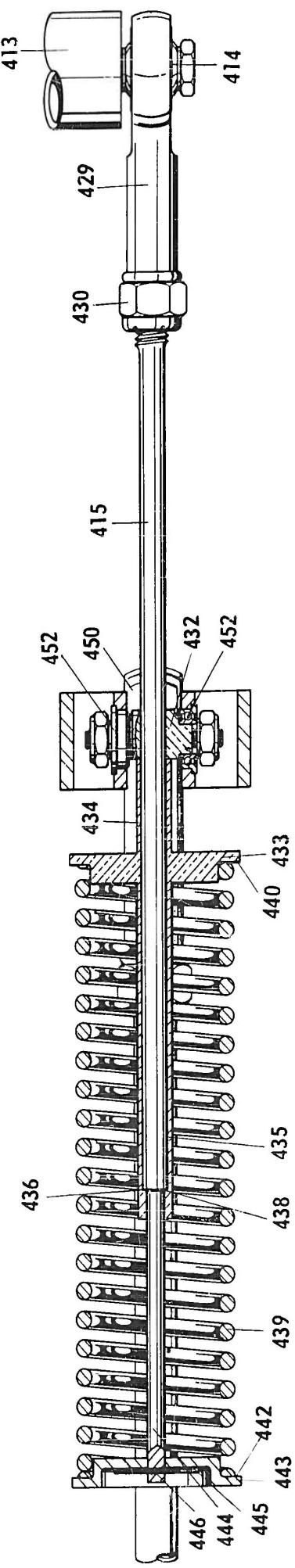


FIG. 13 SIGNAL INPUT FORWARD HYDROFOIL SERVO - DRAWING PARTIAL SECTION ALONG AXIS

DESCRIPTIVE APPENDIX

DETAIL DESCRIPTION OF HIGH TAIL

A. HYDROFOIL MOUNTING, RETRACTION, IMPACT ABSORPTION AND STEERING

Referring to Figures 1, 2 and 15, the aft hydrofoil 1 is pivoted on the struts 2 and 3 about the axis 4. The struts 2 and 3 supporting the right rear hydrofoil 1 are welded to the hollow shaft 5 which is journalled in the bearings 6 and 7. These bearings are carried on the bracket 8 attached to the hull of the boat 9. The cylinders 10 and 11 are trunnion mounted on the bracket 12 which is journalled on the shaft 5. The ram 13 of the cylinder 10 is pivoted on the forward leg (not visible) of the crank 14 attached to the shaft 5. The ram 15 of the cylinder 11 is pivoted on the pin 16 projecting from the bracket 8.

When Pump Pressure is applied to the ram end of the cylinders 10 and 11, the rams 13 and 15 are drawn in causing clockwise rotation of the shaft 5 and the attached struts and hydrofoil through an angle of 180° for retraction. When Pump Pressure is applied to the aft end of the cylinders 10 and 11, the rams 13 and 15 extend causing the counter-clockwise rotation of the shaft 5 until the hydrofoil 1 reaches its operating position, where it is held not only by the Pump Pressure but also by a check valve 17 which forces any oil leaving the cylinders through a high pressure relief valve 18. There is also a high pressure, high flow velocity relief valve 19 resisting the flow of oil through a hole through the plunger of the cylinder 10. With this arrangement, should the hydrofoil 1 strike an obstacle while the boat is underway, the hydrofoil 1 can swing rearward under heavy resistance to avoid impact damage. The arrangements are similar for the left aft hydrofoil 20.

The mechanism for the forward hydrofoil 21 differs in three respects from that for each aft hydrofoil. First, the retraction rotation is opposite to the impact rotation, second, the energy absorbing cylinder is separate from the retraction cylinders and third, steering is provided by rotation of struts and hydrofoil about an upright axis.

Referring to Figure 3, the hollow shaft 22 is journalled near its ends in the bearings 23 and 24 (not visible). These bearings are carried by the steering support member 25 which is pivoted for steering, on knuckle 26, about the axis 27.

Referring to Figure 4, the hydraulic steering unit shown partially at 28 is a power steering aid commonly used on automotive trucks.

The rod end 29 is screwed into the forward extension 30 of the hydraulic cylinder body 31, of the hydraulic steering unit 28, and locked with the nut 32. The rod end 29 is pinned to the crank 33 with the pin 34. The ram 35 of the cylinder 31 is attached with the rubber coupling 36 to the bracket 37 which is securely bolted to the deck 38. The extension and withdrawal of the ram 35, which rotates the crank 33 for steering, is controlled by the steering wheel 39 at the control station shown in Figure 16 in a manner similar to that well known in the automotive industry.

The crank 33 is splined on the shaft 40. The shaft

40 passes first through an aperture 42 (not visible) in the right side of the hull 9 and then through an aperture 44 in the mounting bracket 45 within which the shaft 40 is journalled and sealed to prevent water entering the hull. The shaft 40 also passes through an aperture 46 in the left side of the hull and is journalled in the left side of the bracket 45.

Referring to Figures 3 and 5, the crank 47 is splined and clamped to the shaft 40 and coupled to the connecting rod 48 with the ball joint 49. The connecting rod 48 ends at its forward end in a crotch 50 which is journalled to the block 52. The latter is journalled on a pin 53 projecting downward from the steering support member 25. With this arrangement, fore and aft movements of the cylinder 31, as commanded by the corresponding rotations of the steering wheel 39, rotate the forward hydrofoil assembly 54 about the upright axis 27 for steering. Referring to Figure 1, the forward hydrofoil assembly 54 includes the struts 61 and 62, the hydrofoil 21 and the hollow shaft 22 to which the upper ends of the struts 61 and 62 are attached.

A check valve 63 in the line supplying hydraulic fluid to the cylinder 28 eliminates back flow from the cylinder and thus prevents rotation of the forward hydrofoil assembly 54 about the steering axis 27, as a result of impact on one end of the hydrofoil 21, even if the hydraulic pressure in the cylinder 28 far exceeds the Pump Pressure.

B. SENSOR DESIGN, ACTUATION AND SIGNAL SOURCE

Referring to Figures 1 and 6, each of the three sensors, 64, 65 and 66 (the latter not visible) consists of a sensor arm and a ski rigidly connected together. On the basis of tests thus far, the form preferred for the forward ski 67 of *High Tail* has three planing surfaces in tandem all of which are 31/32-inch wide. The forward planing surface is 2½ inches long and extends from the forward tip of the ski to the step 68. The forward planing surface is flat with a smooth water trim angle of 26°. The intermediate planing surface is 13 inches long and extends from the step 68 to the step 69. The intermediate planing surface has a negative dead rise angle of 45° and a smooth water trim angle of 21°. The chine of the intermediate planing surface is a continuation of the chine of the forward planing surface. The aft planing surface is 22 inches long and extends from the step 69 to the aft end 70 of the ski. The aft planing surface has a 45° negative dead rise and a smooth water trim angle of 15°. A small vertical center fin 71 is attached at the aft end of the ski 67 to damp lateral vibrations of the ski.

The ski 67 is mounted on the forward end of the sensor arm 72 by means of the struts 73 and 74. The upper portions of the struts are circular tubes and the lower portions are streamlined tubes with fairings at the intersections with the ski. In the form preferred, the ski and struts are constructed as thin walled tubes of high strength corrosion resistant material.

For either of the aft sensors 65 and 66, the sensor arm is attached to the forward end of the ski so that no struts are required. Tubular construction with two planing surfaces and dead rise angles of 45° as described for

the forward sensor ski 67 are preferred, but T-cross sections and flat planing surfaces as shown in Figures 1 and 2 are adequate.

In Figures 1, 2, 5 and 6, the forward sensor 64 is shown in its operating position. The sensor arm 72 is clamped and keyed to the shaft 75 which extends into the computer case 76. The computer case 76 is rigidly attached to the steering support member 25. The shaft 75 is journaled in two bearings one of which, 77, (not shown) is supported on the bracket 78 rigidly attached to the steering support member 25. The other bearing 79 (not shown) is supported in the left wall 80 of the computer case 76.

The aft end of the tension spring 81 is pivoted at the pin 82 to the bracket 83 fixed to the computer case 76. The tension spring 81 exerts a counter-clockwise moment on the crank 84 greater than the clockwise moment of the weight of the sensor 64 for all angular positions of the sensor 64.

With the boat floating at rest or underway at a speed below the Sensor Down Speed there is no other comparable moment on the sensor 64 or the shaft 75 so that the sensor 64 retracts to an upright position if it is not already in this position.

The aft end of the hydraulic cylinder 85 is pivoted at the pin 86 to the bracket 87 fixed to the computer case 76. The ram 88 of the hydraulic cylinder 85 is pivoted on the crank 89 at the pin 90. The crank 89 is rigidly attached to the shaft 75. The hydraulic cylinder 91 is pivoted at the pin 92 on the bracket 93 fixed to the case 76. The ram 94 of the cylinder 91 is connected to the rod end 95 through the coupling 96. The coupling 96 puts a limit on the length of the connection between the ram 94 and the rod end 95 but telescopes to permit reduction of this length without restraint. The rod end 95 is pivoted on the crank 97 at the pin 98.

With the floating boat starting forward from rest, the sensor 64 remains retracted in an upright position until the boat reaches Sensor Down Speed when the Sequence Function Unit automatically places the Sensor Loading Pressure on the hydraulic cylinder 85 producing compression in the cylinder 85 and a counter-clockwise moment on the shaft 75. At the same time the Sequence Function Unit automatically places Pump Pressure on the cylinder 91 producing tension in the cylinder 91 and a clockwise moment on the shaft 75. The latter moment, being dominant when the sensor 64 is upright, starts the rotation of the sensor 64 clockwise. When the crank 89 moves over center, the moment produced by the cylinder 85 also becomes clockwise and the sensor 64 accelerates under the torque of both cylinders. Well before the crank 97 reaches dead center, the ram 94 reaches the end of its travel upward. But the telescoping of the coupling 96 permits the continuation of the rotation of the sensor 64 through its full operating range without restraint from the cylinder 91. Beyond the angle at which the ram 94 reaches the end of its travel, the moment of the cylinder 85 continues the rotation of the sensor 64 until the sensor ski 67 strikes the water surface. Then the moment of the cylinder 85, in combination with the weight and spring moments also present, maintains the proper planing load on the ski 67 for all operating speeds above Sensor Down Speed during acceleration or above Sensor Up Speed during deceleration.

At Sensor Up Speed during deceleration of the boat, the cylinders 85 and 91 are automatically drained caus-

ing the sensor 64 to retract to an upright position under the action of the tension spring 81.

The crank 104 attached to the shaft 75 introduces the signal motion of the sensor 64 into the Forward Computer Section.

Referring to Figure 1, the right aft sensor 65 is shown in its operating position. Referring also to Figures 3 and 4, the sensor arm 105 for the right aft sensor 65 is clamped and keyed to the hollow shaft 106 which extends into the bow compartment of the boat where the Bow Computer Section is located. The ram 107 of the hydraulic cylinder 108 is pivoted on the crank 109 at the pin 110. The crank 109 is rigidly attached to the shaft 106. The tension spring 112 is also attached at its lower end to the ram 107 by means of the threaded adapter 113. The upper end of the cylinder 108 is attached to the cross member 114 by means of the universal joint 115. The upper end of the tension spring 112 is attached to the upper end of the cylinder 108 by the threaded cap 116 of the cylinder 108. The cross member 114 is attached to the skin of the hull 9 at its left end by the flange mounting 117. A similar attachment is used for the right end of the cross member 114 (not shown).

The tension spring 112 exerts a greater clockwise moment on the shaft 106, through the crank 109, than the counter-clockwise moment of the weight of the sensor 65 for all angular positions of the sensor 65. With the boat floating at rest or underway at a speed below a predetermined speed there is no other comparable moment on the sensor 65 or the shaft 106, so that the sensor 65 retracts to a position in which the ski is just under the level of the deck 120 if it is not already in this position.

With the floating boat starting forward from rest, the sensor 65 remains retracted until the boat reaches Sensor Down Speed when the Sensor Loading Pressure is automatically conducted to the cylinder 108 producing a compression in the cylinder 108 and a dominant counter-clockwise moment on the shaft 106 causing the sensor 65 to rotate counter-clockwise downward until the sensor ski 121 strikes the water surface. Then the moment of the cylinder 108, in combination with the spring and weight moments also present, maintains the proper planing load on the sensor ski 121. The rotation of the sensor 65 is transmitted by the shafts 106 and 122 to the cranks 123 and 124 which introduce the sensor signal motion into two parts of the Bow Computer Section. To minimize distortions of the signal motions caused by deflections of the shaft 106 perpendicular to its axis, the shafts 106 and 122 are coupled together by means of a universal joint (not shown) within the hollow shaft 106, and the shaft 122 is journaled on the extension 125 of the frame 126. At Sensor Up Speed during deceleration of the boat, the cylinder 108 is automatically drained causing the retraction of the sensor 65 under the action of the spring 112.

The mounting and actuation of the left aft sensor 66 is similar to that described for the right aft sensor 65. The rotation of the sensor 66 is transmitted by the shafts 130 and 131 to the cranks 132 and 133 which introduce the sensor signal motions into two parts of the Bow Computer Section. The arrangement to minimize distortion of the signal motions, caused by deflections of the shaft 130 perpendicular to its axis, is similar to that described for the sensor 65.

C. THE SEQUENCE FUNCTION UNIT

Referring to Figures 8, 9 and 10, the single acting hydraulic cylinder 140 is hinged on the frame 141 at the axis 142. The ram 143 of the cylinder 140 is pivoted at the ball joint 144 to the crank 145, attached rigidly to the hollow cam shaft 146. The tension springs 147 and 148 extend from the cross member 149 mounted on the ram 143, to the links 150 and 151 which connect the springs 147 and 148 to the evener bar 152, pivoted at the pin 153 on the hydraulic cylinder head 154. The evener bar 152 equalizes the tensions in the springs 147 and 148 and, thus, eliminates bending moment on the ram 143 in order to minimize sliding friction between the ram 143 and the cylinder body 155.

The Smoothed Servo Cylinder Pressure is connected to the hydraulic cylinder 140 through the port 156 in the cylinder head 154. The ram 143 of the hydraulic cylinder 140 is sealed with a labyrinth seal 158 to minimize seal friction. The leakage through the seal 158 is conducted out of the port 159 to drain. The chevron seal 160 prevents loss of the leakage oil, but, not being under appreciable pressure, contributes only slightly to the friction.

The cam shaft 146 is journalled on the ball bearings 162 and 163 which are supported by the bolt 164 extending through the two brackets 165 and 166. The latter brackets are bolted and doweled to the frame 141.

There are four needle valves generally located at 167, 168, 169 and 170 within the frame 141. Figure 9 shows the needle valve located at 167 which is typical. The needle valve cone 172 is adapted to close the passage 173 of the valve seat member 174. The valve stem 175 integral with the needle valve cone 172 is a close free fit in the cylindrical hole 176 within the frame 141. The valve seat member 174 is also sealed within the hole 176 by means of the ring seal 177. The valve seat member 174 is held in place by the fitting 178. The pilot pressure tube 179 is connected by means of the tube fitting 180.

The tube 179 is connected directly to the hydraulic component being controlled such as a valve or cylinder and also connected through a restrictor to the Pump Pressure. The needle valve passage 182 is connected by a hole 183 drilled in the frame 141 to the drain connection 184. The other three needle valve passages are also drained by the hole 183 and the connection 184. The gimbal ring 185 is attached to the valve stem 175 by means of the pin 186. The rocker arm 187 is journalled on the frame 141 on the axis 188. The rocker and gimbal construction is further shown for the rocker arm 189 and the gimbal ring 190 in Figure 10. The two projections 192 on the rocker arm 189 straddle the valve stem 193 and protrude into two notches 194 in the gimbal ring 190 to prevent the valve stem 193 from rotating and to provide contacts 195 between the rocker arm 189 and the gimbal ring 190 along a line parallel to the axis 188 and the intersecting axis of the valve stem 193. Referring back to Figure 9, the compression spring 196 seats on the gimbal ring 185 and on the spring support screw 197 threaded into the frame 141. The roller 198 is a ball bearing pinned to the rocker arm 187. All rockers, rollers and needle valve assemblies are of identical design. The cam 199 is securely clamped to the hollow cam shaft 146 by means of the cap screw 200. The cam 199 has a large constant radius portion 202, a transition portion

203 and a clearance portion 204. All four cams are identical except for the angular position of the transition portion of the cam with respect to the crank 145.

In operation, the Smoothed Servo Cylinder Pressure within the cylinder 140 pushes the ram 143 out extending the springs 147 and 148 to a length corresponding to the Smoothed Servo Cylinder Pressure and therefore corresponding to the forward speed of the boat. Each length of the springs 147 and 148 corresponds to an angular position of the cam shaft 146. Starting with the boat at rest the cam shaft 146, as viewed in Figure 10, is rotated to its extreme clockwise position. In this position, all cam rollers are lifted by being in contact with the large constant radius portion of the respective cams, all needle valves are open, and all pilot pressures are at drain pressure. As the speed of the boat increases, the cam shaft 146 rotates counter-clockwise as viewed in Figure 10. As the transition portion of each cam rolls under the cam roller, the latter is lowered and the spring 196, as viewed in Figure 9, closes the needle valve 175 blocking the drain 183 on the pilot line 172 and thus raising the pilot pressure which initiates the corresponding control operation.

D. THE RECIPROCAL FUNCTION UNITS

The two Reciprocal Function Units used on *High Tail* are identical except for different length adaptations. Each Reciprocal Function Unit produces a signal σ_h from the Smoothed Servo Pressure Q as the signal for the Dynamic Pressure q in accordance with Equations (15) and (16) and corresponding to the terms $\frac{w_{fh}}{Kq}$ and $\frac{w_{ah}}{Kq}$ in the Specific Coupling Equations (12), (13) and (14). Such use of Reciprocal Function Units substantially eliminates change in Bottom Clearance with change in forward speed at speeds above Up Speed.

Figure 11 is a cross sectional view of the Reciprocal Function Unit 209 used in the control of the forward hydrofoil 21. The Reciprocal Function Unit 210 for the two aft hydrofoils differs only in that its plug and length adapter 211 is much longer than the plug and length adapter 212 for the Reciprocal Function Unit 209.

Referring to Figure 11, the bore 213 of the cylinder 214 is closed with the plug and length adapter 212 which is screwed into the body 214 of the Reciprocal Function Unit 209, sealed with the ring seal 216, and locked in place with the lock nut 217. The ball joint 218 is formed integral with the plug and length adapter 212. The ram 219 is a close free fit in the bore 213 and has several circumferential sealing grooves 220 to reduce leakage. The Smoothed Servo Pressure is conducted through the port 222. Leakage passing the fit between the ram 219 and the bore 213 is drained by a drain line attached to the drain port 223. The seal 224, not being under pressure, adds very little to the friction but prevents leakage other than through the drain port 223.

225 is a circular cylindrical bore in the body 214 coaxial with the bore 213. The hollow nut 226 is threaded into the body 214, coaxial with the bores 213 and 225, and holds the tube 227, washer 228, the tube 229, the washer 230, the tube 232, the washer 233, the seal 224, and the ring 234 tightly in place. The spring 235 is centered at its left end by the hollow nut 226 and at its right

end by the washer 236. Initial compression in the spring 235 holds the washer 236, the tube 237, the washer 238, the tube 239, and the washer 240 either against the washer 233, if there is no Smoothed Servo Pressure, or against the shoulder 242 of the spindle 243 or against both the washer 233 and the shoulder 242 if there is Smoothed Servo Pressure. The spindle 243 fits into the socket 244 of the ram 219 and takes the reaction of the latter.

As the Smoothed Servo Pressure is increased, the ram 219, together with the spindle 243 and the washers and tubes strung thereupon, moves to the left, compressing the spring 235 and moving the springs 245 and 246 to the left until the gap 247 between the spring 245 and the washer 228 is closed, after which the movement to the left of the spindle 243 for a given increase in the Smoothed Servo Pressure is decreased due to the stiffness of the spring 245. With still further increase in the Smoothed Servo Pressure, the gap 248 between the spring 246 and the washer 230 closes further decreasing the movement to the left of the spindle 243 for a given increase in the Smoothed Servo Pressure. The required nonlinear relation between the Smoothed Servo Pressure and the increase in length σ_h of the Reciprocal Function Unit 209 is thus approximated.

E. OVERLOAD RELEASES

There are twelve overload release devices in *High Tail's* control. One of these used with the forward servo is shown in Figure 13 and described in Section I of this Descriptive Appendix. Two others of the same design are used with the aft servos.

Figure 12 shows a partial section view of the overload release 250 used in the Forward Computer Section. The overload release 250 is a link in the computer mechanism pivoted at the pins 251 and 252. The rod ends 253 and 254 are screwed into the rods 255 and 256 and locked in place with the nuts 257 and 258 respectively.

The rod 255 has two enlargements 259 and 261 which fit freely and align the rod 255 within the bore 262 of the tube 263. The compression spring 264 slides freely over the rod 255 and within the bore 262. The spring 264 is retained by the threaded cap 265, which is screwed on to the threaded end 266 of the tube 263 and locked with the lock nut 267.

The compression load in the spring 264 is exerted against the inward extending flange 268 of the cap 265 and against the plug 269 fitting the bore 262 of the tube 263 and located by the cotter pin 270. The cotter pin 270 is fitted into the matching holes 273 and 274 in the tube 263 and the plug 269 respectively. The compression load in the spring 264 is equal to the limiting tension in the overload release 250 beyond which the withdrawal of the rod 255 from the tube 263 lengthens the overload release 250.

The rod 256 has two enlargements 275 and 276 which fit freely and align the rod 256 within the bore 262 of the tube 263. The enlargement 276 is retained by the inward extending shoulder 277 of the threaded cap 278 which is screwed on the threaded end 279 of the tube 263 and locked with the lock nut 280. The compression spring 281 slides freely within the bore 262 of the tube

263. The compression load in the spring 281 is exerted on the plug 269 and the inner end 282 of the rod 256. This compression load in the spring 281 is the limiting compression in the overload release 250 beyond which the rod 256 moves into the tube 263 shortening the overload release 250.

The other overload releases in the computer are variations of the overload release 250 and that shown in Figure 13.

F. LOCATIONS OF THE COMPUTER SECTIONS IN THE BOAT

The computer is physically divided into four parts, the Forward Computer Section, the Bow Computer Section, the Sequence Function Unit and the Stern Computer Section. The Forward Computer Section is contained within the case 76 and is part of the forward hydrofoil control. The Bow Computer Section is located within the bow enclosure 283 and is part of the control for all three hydrofoils. The Sequence Function Unit is mounted on the forward wall of the bulkhead at the control station shown in Figure 16. The Stern Computer Section is located within the stern compartment just forward of the transom 285 and is part of the control for the aft hydrofoils.

G. THE FORWARD COMPUTER SECTION

Referring to Figure 5, the Forward Computer Section located within the case 76 is part of the hydrofoil angle control for the forward hydrofoil. Steering, elevation, Smoothed Servo Pressure and sensor signals, as well as manual adjustments for pay load, trim and multiplier settings are transmitted to the case 76.

All important axes of the Forward Computer Section are parallel to the axis of the tubular shaft 22. Lost motion in the Forward Computer Section is prevented by a load throughout the mechanism introduced by the forward hydrofoil servo shown generally at 286 and described in Section I of this Descriptive Appendix. The forward hydrofoil 21, the struts 61 and 62, the shaft 22, the sensor 64, the steering support member 25 and the computer case 76 all rotate together about the upright axis 27 when the boat is steered.

The signal for the steering angle θ is transmitted from the knuckle 26 by means of the connecting rod 287 the forward end of which is pivoted at the ball joint 288 on the extension 289 of the knuckle 26. The aft end of the connecting rod 287 is pivoted on the crank 290 at the ball joint 292. The crank 290 is part of the hollow shaft 293 which is supported by the bearing 294 attached to the steering support member 25 and the bearing 295 in the left skin 80 of the computer case 76 where the shaft 293 enters the computer case 76.

The crank, connecting rod, stop and spring, now to be described, comprise the Steering Function Unit of the Forward Computer Section. The near dead-center crank 296 is keyed and clamped to the shaft 293 just inside the computer case 76. The connecting rod 297 is attached at its aft end to the crank 296 at the ball joint 298. The

connecting rod 297 telescopes so that it may be increased in length but not shortened below a fixed minimum length. The forward end of the connecting rod 297 is the crotch 299 which is pivoted at the pin 302 to the ram 303 of the hydraulic cylinder 304 mounted on the crank 305 of the multi-crank member 306, the latter being journalled about the axis 307 in bearings 308 and 309 supported in the right and left skins 310 and 80 of the case 76. The tension spring 312, attached to the crank 305 with the eye bolt 313 and to the case mounted bracket 314, pulls rearward on the crank 305 putting compression in the ram 303 with a reaction either through the pin 302 and the connecting rod 297 to the ball joint 298 or to the stop 315.

From the geometric relations between the crank 296 and the connecting rod 297 the rearward movement of the pin 302, before the aft end of the ram 303 strikes the stop 315, is approximately proportional to the square of the angular displacement of the crank 296 away from the dead center, which is also proportional to the steering angle θ . When θ reaches either $+\theta_m$ or $-\theta_m$ the aft end of the ram 303 strikes the stop 315 preventing further rearward displacement of the pin 302. The telescopic extension of the connecting rod 297 permits the further unrestrained rotation of the crank 296. With this mechanism the forward displacement σ_{302} of the pin 302 may be expressed:

$$C_1 \sigma_{302} = -K_f \theta_b^2 \quad (80)$$

At Up Speed during acceleration, the Sequence Function Unit subjects the hydraulic cylinder 304 to Pump Pressure, which causes the ram 303 of the hydraulic cylinder 304 to move rearward against a stop within the hydraulic cylinder 304. This is the position of the ram 303 shown in Figure 5. At Down Speed during deceleration, the Sequence Function Unit drains the hydraulic cylinder 304 causing the tension spring 312 to withdraw the ram 303 forward against another stop within the cylinder 304.

The forward displacement σ_{304} of the cylinder 304 equals the forward displacement σ_{302} of the pin 302 minus the forward displacement σ_{303} of the ram 303 with respect to the cylinder 304 thus:

$$\sigma_{304} = \sigma_{302} - \sigma_{303} \quad (81)$$

Choosing σ_{303} equal to $-\frac{A_e}{C_1}$ and combining Equations (80) and (81) gives:

$$C_1 \sigma_{304} = + A_e - K_f \theta_b^2 \quad (82)$$

The two-layer crank 316 and the cranks 317 and 318, all of which are appendages of the multi-crank member 306, form a hinge with the two-axis cranks 319 and 320 of the multi-crank member 321 about the hinge axis 322. The two-layer crank 323, which is the rigid extension of the two-axis crank 320 of the multi-crank member 321, is

pivoted to the aft end 324 of the connecting rod 325 at the pin 326 (not shown).

The multiplier-shaft 327 is hinged on the two-axis cranks 319 and 320 of the multi-crank member 321 about the hinge axis 328. The multiplier-beam 329 is rigidly attached to the multiplier-shaft 327. The carriage 330 and the attached parts provide a vertically movable fulcrum for the multiplier-beam 329. The multiplier-beam 329 is maintained in contact with the roller 332 by a load from the forward hydrofoil servo as will be explained below. The roller 332 is pivoted on the carriage 330 with the pin 333 (not visible). Two other rollers, 334 and 335 (the latter not shown), pivoted on the carriage 330 on the pins 336 and 337 (the former not shown), guide the carriage 330 on the vertical cylindrical post 338 fixed to the case 76. The load introduced by the forward servo, as described in Section I of this Descriptive Appendix, maintains contact between the rollers 334 and 335 and the cylindrical post 338. Under these conditions the multiplier-beam 329 is constrained to rotate about the axis 333, hence the forward displacement σ_{340} of the connecting rod 340, which is pivoted to the two-layer extension 341 of the multiplier-beam 329 at the pin 342 is given by:

$$\sigma_{340} = -C_2 A_f \sigma_{328} \quad (83)$$

where A_f is a coefficient in the Specific Coupling Equation (12) and $-C_2 A_f$ is the multiplication ratio of the multiplier-beam 329 determined by the elevation of the carriage 330 with respect to the hinge axis 328 which is adjusted at the control station with the handle 343 shown in Figure 16. The handle 343 is connected to a master cylinder 344, shown in Figure 15, located below the deck 120. A single hydraulic line (not shown) connects the master cylinder 344 to the slave cylinder 346, Figure 5. The slave cylinder 346 is connected to the bracket 347 with the pin 348. The bracket 347 is attached to the upper skin of the case 76. The ram 349 of the slave cylinder 346 is connected to the carriage 330 with the pin 350. The two tension springs 352 and 353 are hooked around the pin 350 at their lower ends and around the pin 348 at their upper ends. The springs 352 and 353 act to return the ram 349 when the ram of the master cylinder 344 is withdrawn by the control handle 343 at the control station. Fluid leakage is replaced at the end of the stroke of the master cylinder 344 in a way well known for such hydraulic repeater systems, so that the extension of the ram 349 of the slave cylinder 346 is determined by the position of the handle 343 at the control station. The scale 354 at the control station which indicates the position of the handle 343 is calibrated to read the coefficient A_f .

The forward displacement σ_{304} of the cylinder 304 is proportional to the forward displacement σ_{322} of the hinge axis 322 because both are attached to the multi-crank member 306, thus:

$$C_3 \sigma_{304} = \sigma_{322} \quad (84)$$

The relations among the forward displacements σ_{328} , σ_{322} and σ_{325} of the axis 328, the axis 322 and the connecting rod 325 are determined by their relative elevations to be:

$$C_4 \sigma_{328} = \sigma_{322} - C_5 \sigma_{325} \quad (85)$$

Combining the Equations (82), (83), (84) and (85) gives:

$$\sigma_{340} = \frac{C_2 C_3}{C_4 C_1} A_f \left[\frac{C_1 C_5 \sigma_{325}}{C_3} - A_e + K_f \theta_b^2 \right] \quad (86)$$

The manual pay load and trimming adjustment at the control station for the forward hydrofoil, corresponding to the term B_f in the Specific Coupling Equation (12), is made with and indicated by the crank and screw assembly 355 as shown in Figure 16. The crank and screw assembly 355 moves the ram of a master cylinder 356, shown in Figure 15, located under the deck 120. The master cylinder 356 is connected with a single hydraulic line (not shown) to the slave cylinder 357, Figure 5. The slave cylinder 357 is hinged with a pin 358 on the bracket 359 attached to the top of the case 76. The ram 360 of the slave cylinder 357 is pivoted on the two-layer crank 362 by means of the pin 363. The two-layer crank 362 is rigidly attached to the shaft 364. The tension spring 365 (not shown) extending from the pin 363 to the pin 358 acts to return the ram 360 when the ram of the master cylinder 356 is withdrawn. Fluid leakage is replaced at the end of the stroke of the master cylinder 356 in a way well known for such hydraulic repeater systems, so that the extension of the ram 360 of the slave cylinder 357 corresponds to the setting of the crank and screw assembly 355 at the control station.

The shaft 364 is journalled in the left side 80 of the case 76 and in the bracket 366, attached to the forward side of the case 76. The shaft 364 has the eccentric 367 integral therewith.

The Reciprocal Function Unit 209 as a link extends from the eccentric 367 to the pin 368 which pivots the Reciprocal Function Unit 209 to the two-layer extension 369 of the member 370. The Reciprocal Function Unit 209 is shown in cross section in Figure 11 and described in Section D of this Descriptive Appendix. The Reciprocal Function Unit 209 is subject to the Smoothed Servo Pressure Q. The increase in length, σ_h , of the Recipro-

cal Function Unit 209, corresponding to the term $\frac{w_{fh}}{Kq}$ in

the Specific Coupling Equation (12), is expressed by Equation (15).

The member 370, which includes the rigidly attached bars 372 and 373 and the tube 374, is hinged on the bracket 375 mounted on the upper skin of the case 76 and on the left skin 80 of the case 76 about the axis 376. The double crank member 377 is hinged on the extensions 378 and 379 of the member 370 about the axis 380. The two-layer arms 382 and 383 are the cranks of the double crank member 377. The connecting rod 340 is hinged on

the two-layer crank 382 at the pin 385. The connecting rod 250 is pivoted on the two-layer crank 383 at the pin 251 (not shown).

With this arrangement, the counter-clockwise angular displacement of the member 370 about the axis 376 and therefore the forward displacement σ_{380} of the axis 380 are proportional to the pay load and trimming adjustment term B_f in the Specific Coupling Equation (12), minus a quantity proportional to the increase in the length σ_h of the Reciprocal Function Unit 209 as given by Equation (15), thus:

$$C_6 \sigma_{380} \equiv B_f r_f + C_7 C_f \left[\frac{w_{fh}}{Kq} \right] \quad (87)$$

With the components designed to make $C_7 C_f = r_f$, Equation (87) becomes:

$$\frac{C_6 \sigma_{380}}{r_f} = \frac{w_{fh}}{Kq} + B_f \quad (88)$$

The relations between the forward displacements σ_{340} , σ_{250} and σ_{380} of the connecting rods 340 and 250 and the axis 380 are determined by their relative elevations to be:

$$C_8 \sigma_{340} + C_9 \sigma_{250} \equiv \sigma_{380} \quad (89)$$

It follows from Equations (88) and (89) that:

$$C_9 \sigma_{250} = - C_8 \sigma_{340} + \frac{r_f}{C_6} \left[\frac{w_{fh}}{Kq} + B_f \right] \quad (90)$$

The crank 104, attached to the shaft 75 and pivoted to the connecting rod 325 at the pin 400, introduces the sensor displacement signal into the Forward Computer Section. The combination of the crank 104 and the connecting rod 325, with the angle between chosen equal to the corresponding angle between the Sensor Radius and a perpendicular to the Flying Bottom Plane, is a Sine Function Unit so that the connecting rod 325 transmits a forward displacement signal σ_{325} to the pin 326 that is proportional to δ_1 thus:

$$C_{10} \sigma_{325} = \delta_1 \quad (91)$$

The connecting rod 250 extends from the pin 251 to the crank 404 to which it is pivoted at the pin 252. Included in the connecting rod 250 is the overload release shown in Figure 12 and described in Section E of this

Descriptive Appendix. The crank 404 is rigidly attached to the tubular shaft 407 at the jaw coupling 408. 407 extends to the right through a tubular extension 409 of the case 76 and into another case 410 which is attached to the extension 409 and sealed to the inner diameter of the outer end of the tubular shaft 22, with a rotary ring seal (not shown) coaxial with the tubular shaft 22. Within the case 410, the shaft 407 is rigidly attached to the crank 411. The crank 411 is pivoted on the case 410 and the crank 404 is pivoted on the case 76, both about the axis 412. The tubular extension 413 is rigidly mounted to the crank 411 and carries the pin 414, as shown in Figure 13, at its left end on which is pivoted the vertical connecting rod 415 which connects to the forward hydrofoil servo below.

The downward displacement σ_f of the connecting rod 415 is proportional to the forward displacement σ_{250} thus:

$$C_{11} \sigma_{250} = \sigma_f \quad (92)$$

Combining Equations (86), (90), (91) and (92) gives:

$$\frac{C_6 C_9}{r_f C_{11}} \sigma_f = \frac{w_{fh}}{Kq} + B_f - \frac{1C_8 C_2 C_3 C_6}{r_f C_4 C_1} \frac{A_f}{I} \left[\frac{C_1 C_5}{C_3 C_{10}} \delta_1 - A_e + K_f \theta_b^2 \right] \quad (93)$$

Choosing the mechanism dimensions to satisfy the conditions:

$$\frac{1C_8 C_2 C_3 C_6}{r_f C_4 C_1} = 1 \quad (94)$$

and

$$\frac{C_1 C_5}{C_3 C_{10}} = 1 \quad (95)$$

Equation (93) becomes:

$$\frac{C_6 C_9}{r_f C_{11}} \sigma_f = \frac{w_{fh}}{Kq} + B_f - \frac{A_f}{I} \left[\delta_1 - A_e + K_f \theta_b^2 \right] \quad (96)$$

The downward displacement σ_f of the connecting rod 415 is the output signal of the Forward Computer Section and the command signal to the forward servo,

shown generally at 286, which must produce the α_f that satisfies the Specific Coupling Equation (12). Since this equation must be satisfied for all conditions, any condition may be used to determine the constant of proportionality relating σ_f to α_f . Take the condition $\dot{\alpha}_f=0$. The right hand member of Equation (96) is identical to the right hand member of the Specific Coupling Equation (12). Equating the left hand members and using the condition $\dot{\alpha}_f=0$ gives:

$$\frac{C_6 C_9}{r_f C_{11}} \sigma_f = \alpha_f \quad (97)$$

The mechanism dimensions are selected such that the ratio $\frac{C_6 C_9}{r_f C_{11}}$ is suitable for the servo.

H. CAGING

Caging to force the output signal of the Forward Computer Section to a predetermined displacement, suitable for operation of the boat at speeds below Uncage Speed, is accomplished by the caging unit located within the case 410 shown in Figure 5. The cam rocker 416 is pivoted on the case 410 about the axis 417. The tension spring 418 extends from the pin 419, projecting from the wall of the case 410, to the pin 420 attached to the cam rocker 416. The tension spring 418 tends to rotate the cam rocker 416 clockwise. The single acting hydraulic cylinder 421 is pivoted on the case 410 at the pin 422. The ram 424 of the cylinder 421 is hinged on the cam rocker 416 about the axis 426.

Hydraulic pressure to the cylinder 421 is controlled by the Sequence Function Unit. With the boat at rest or under way at speeds below the Uncaged Speed during acceleration or below the Caged Speed during deceleration, the cylinder 421 is drained so that the cam rocker 416 rotates clockwise under the pull of the spring 418. With this rotation, the sides of the cam opening 427 of the cam rocker 416 cage the crank 411 by forcing the roller 428 pivoted to the crank 411 into the forward narrow end of the opening 427.

The caged position of the crank 411 is chosen to produce the most favorable trim of the boat at forward speeds below the Caged Speed during acceleration. Caging of the crank 411 is possible despite input signals to the Forward Computer Section corresponding to σ_f other than the caged position because of the overload releases in the Forward Computer Section which tolerate the displacement incompatibilities involved.

When the boat accelerates to Uncaged Speed, the Sequence Function Unit admits Pump Pressure to the cylinder 421 rotating the cam rocker 416 counter-clockwise under the dominant torque of the cylinder 421, and the roller 428 is released from the sides of the cam opening 427, uncaging the crank 411 and allowing the signal σ_f to go through to the forward hydrofoil servo.

Separate caging of the computer output signals σ_{ar} and σ_{al} is accomplished in the same way for the right aft and left aft hydrofoils.

I. SERVOS FOR HYDROFOIL CONTROL

Each of the three hydrofoils is rotated for control purposes by a hydraulic servo. The design of all three servos is the same except some of the dimensions are larger for the forward servo. Only the forward servo shown generally at 286 in Figure 5 will be described in detail.

Referring to Figures 5 and 13, the rod end 429 is screwed on to the upper end of the rod 415 and locked in place with the lock nut 430. The rod end 429 is pivoted on the arm 413 at the pin 414. The rod 415 extends downward through the trunnion guide 432 and the spring support 433 with freely slidable fits in both. The spring support 433 has upper and lower tubular extensions 434 and 435 respectively. The rod 415 has a shoulder 436 below which the rod diameter is reduced. The lower tubular extension 435 also has a shoulder 438 below which its inside diameter is reduced. The compression spring 439 extends between an upper seat 440 on the spring support 433 to a lower seat 442 on the support washer 443. The latter surrounds the lower end 444 of the rod 415 and is held in place by the pin 445 inserted in the hole 446 in the lower end of the rod 415.

During ordinary operating conditions where there is no need for relative motion between the rod 415 and the trunnion guide 432, the compression load in the spring 439 holds the shoulder 438 of the spring support 433 against the shoulder 436 of the rod 415, and the tension in the spring 448 holds the lower side of the trunnion guide 432 against the upper end of the upper tubular extension 434. On the other hand, if the absence of Pump Pressure or some other condition prevents movement of the rod 449, and thereby limits movement of the trunnion guide 432, the rod 415 can move through the trunnion guide 432 without damaging the mechanism. When this movement is downward the assembly below the trunnion guide 432 moves downward with the rod 415 as the latter slides unrestrained through the trunnion guide 432. When this movement is upward the spring support 433 is prevented from moving upward by the contact between the bottom of the trunnion guide 432 and the upper tubular extension 434 so that reduction in the length of the compression spring 439 is necessary. The compression load reached in the spring 439 by this reduction in length is not large enough to damage the mechanism and yet the initial load in the spring is adequate to prevent the ordinary control forces from moving the rod 415 through the trunnion guide 432.

The trunnion guide 432 is journalled in the two-layer extension 450 of the beam 451 by means of the ball bearings 452. The beam 451 is pivoted on the two-layer beam 453 about the axis 454. The two-layer beam 453 is journalled on the four-layer bracket 455 about the axis 456. The four-layer bracket 455 includes the flange 457, which is fastened to the valve plate 458 by means of four screws, one of which is 459.

The valve stem 460 which controls the valve 462 is rigidly attached to the cross bar 463. Two rods, sym-

metrically disposed with respect to the valve stem 460, one of which is 464, rigidly join the crossbar 463 to the cross member 465. The latter has two strips (not shown) extending upwardly one on the right and one on the left of the beam 451, to which the beam 451 is pivoted about the axis 466 (not shown) coincident with or slightly above or below and parallel to the axis 456.

The upper end of the tension spring 448 is hooked into the downward extending eye 467 which is part of the cross member 465. The lower end of the tension spring 448 is hooked around the pin 468 which extends through holes in the lower end of two layers of the four-layer bracket 455. When the control is operating, tension in the spring 448 puts a downward load on the beam 451 which, through the trunnion guide 432 and the ball bearings 452, puts a tension in the rod 415 and corresponding loads throughout the entire mechanism. These loads eliminate lost motion. One such load holds the multiplier beam 329 against its roller fulcrum as explained in Section G of this Descriptive Appendix.

When the control is not operating and the motion of the push rod 449 is blocked, only a small downward displacement of the trunnion guide 432 is possible before the valve stem 460 reaches a stop (not shown) which sustains the tension in the spring 448 and thereby removes the corresponding loads throughout the mechanism.

The two-layer beam 453 is pivoted on the connecting rod 474 at the pin 475. The connecting rod 474 is also pivoted on the U-clamp 476 at the pin 477. The U-clamp 476 is rigidly clamped to the push rod 449, which extends upward through the seal 478 and through the piston 479 to which it is rigidly attached and sealed. The piston 479 is sealed within the bore of the closed top cylinder 480.

The hydraulic valve 462 is mounted and sealed to the valve plate 458. The hydraulic tubes (not shown), within the hollow shaft 22, conducting oil to and from the valve 462 are joined and sealed to the valve plate 458. Hydraulic fluid controlled by the valve 462 is conducted to and drained from the hydraulic cylinder 480 through the tube connection 481. The space within the cylinder 480 below the piston is connected to the tube 482 by the annular passage 483.

The push rod 449 is fitted with a rod end 484 which is spherically journalled about the axis 485 on the hinge block 486 of the hydrofoil 21. The hinge block 486 is spherically journalled on the strut 61 about the axis 487.

With the arrangement described, for any downward displacement of the push rod 449 and therefore any angular position of the hydrofoil 21, there is a downward displacement σ_f of the axis of the ball bearings 452 which places the axis 466 in coincidence with the axis 456. This σ_f will be referred to as the Coincident σ_f . For the Coincident σ_f , the stem 460 of the valve 462 is in its center position blocking the oil flow through the tube 481 to or from the top of the cylinder 480. On the other hand, if the downward displacement σ_f is greater than the Coincident σ_f , the valve stem 460 is displaced downward conducting hydraulic fluid to the cylinder 480 through the tube 481. If σ_f is less than the Coincident σ_f , the valve stem 460 is raised and the valve 462 drains the cylinder 480 through the tube 481. The amount of valve opening being in either case determined by the difference between the actual σ_f and the Coincident σ_f .

J. BOW COMPUTER SECTION

Referring to Figures 3 and 4, the Bow Computer Section is part of the control for the aft hydrofoils. The Bow Computer Section receives aft sensor signals, as described in Section B of this Descriptive Appendix, steering signals, signals from the Sequence Function Unit, and manual sea condition, pay load and trimming adjustments.

All important axes of the Bow Computer Section are beamwise. Lost motion in most of the Bow Computer mechanism is prevented by leads introduced by the aft hydrofoil servos in the same manner as explained in Section I of this Descriptive Appendix.

The Steering Function Unit for the Bow Computer Section will now be described. Referring to Figure 4, the tubular splined shaft 40 is rotated for steering. Two steering signals are obtained from this rotation. The first is the forward displacement σ_{488} of the pin 488 given by:

$$C_{14} \sigma_{488} = -\theta \quad (100)$$

and the second is the forward displacement σ_{489} of the pin 489 when the ram 491 is not in contact with the stop 492, given by:

$$C_{15} \sigma_{489} = \theta^2 \quad (101)$$

The connecting rod 493 is pivoted on the two-layer crank 494 attached to the shaft 40 at the pin 488 and transmits σ_{488} from the pin 488 to the pin 495 which pivots the connecting rod 493 on the two-layer crank 496 of the multi-crank member 497. The latter is hinged on the brackets 498 and 499 of the frame 126 about the hinge axis 502. The frame 126 is rigidly fastened to the hull framing and to the deck 38. The contact points 504 and 505 attached to the crank 496 and the mating contact points 506 and 507 forming part of the frame 126 impose limits on the angular rotation of the crank 496 and therefore on the multi-crank member 497 at the steering angles θ_n and $-\theta_n$. Thus the forward displacement σ_{495} is given by:

$$C_{14} \sigma_{495} = -\theta_c \quad (102)$$

Forward travel of the pin 488 in excess of that corresponding to the stop limits is permitted by the overload release 508 forming part of the connecting rod 493, working in conjunction with the tension spring 509 extending between the crank 496 and an eyebolt (not shown) attached to the frame 126.

The two-layer arm 510 is also part of the multi-crank member 497. The two-layer lever 511 is pivoted on the two-layer arm 510 at the pin 512. The horizontal connecting rods 513 and 514 are pivoted on the two-layer lever 511 at the pins 515 and 516 respectively, equidistant from the pin 512. The forward displacement σ_{512} of the pin 512 is proportional to the rearward displacement of the pin 495. Hence, from Equation (102) the

forward displacement σ_{514} of the connecting rod 514 is given by:

$$\sigma_{514} = -\sigma_{513} + C_{15} \theta_c \quad (103)$$

The multi-crank member 517 is pivoted on the brackets 518 and 519, which are part of the frame 126, about the axis 520. The arm 521 of the multi-crank member 517 carries the hydraulic cylinder 522. The two-layer crank 523, of the multi-crank member 517, carries the pin 524 around which the aft end of the tension spring 525 is pivoted. The forward end of the spring 525 is hooked into the eyebolt 526 attached to the frame 126.

The combination of the two-layer crank 527, the connecting rod 528, the stop 492, the hydraulic cylinder 522, the ram 491 of the latter, the spring 525, and the arm 521 of the multi-crank member 517 for producing a θ_b^2 signal is the same scheme as the combination of the crank 296, the connecting rod 297, the stop 315, the hydraulic cylinder 404, the ram 303 of the latter, the spring 312 and the arm 305 of the multi-crank member 306 described in Section G of this Descriptive Appendix for the Forward Computer Section. The forward displacement σ_{522} of the hydraulic cylinder 522 is therefore given by:

$$C_{16} \sigma_{522} = -2A_e + 2K_a \theta_b^2 \quad (104)$$

The signs of the right hand member of Equation (104) are opposite to those in the right hand member of Equation (82) because the arrangement of the mechanism is opposite to that in the Forward Computer Section.

The two-layer lever 530 is pivoted on the two-layer crank 532, which is part of the multi-crank member 517, at the pin 533. The horizontal connecting rods 534 and 535 are pivoted on the two-layer lever 530, at the pins 536 and 537, equidistant from the pin 533. The forward displacement of σ_{533} of the pin 533 is proportional to σ_{522} expressed in Equation (104). Hence, the forward displacement σ_{535} of the connecting rod 535 is given by:

$$\sigma_{535} = -\sigma_{534} + C_{17} \left[-2A_e + 2K_a \theta_b^2 \right] \quad (105)$$

The signal from each aft sensor 65 and 66 is introduced twice in the Bow Computer Section by means of two crank-connecting rod combinations. Each of such combinations is a Sine Function Unit as explained in Section I of the Submerged Hydrofoil Control Theory and Application herein.

The signal from right aft sensor 65 is introduced into the Bow Computer Section by the crank 123 and the connecting rod 539, pivoted to the crank 123 at the pin 540, and by the crank 124 and the connecting rod 513 pivoted to the crank 124 at the pin 542. The connecting rods 539 and 513 include overload releases. The rearward displacement of connecting rods 539 and 513 are proportional to δ_{2r} thus:

$$-\sigma_{539} = C_{18} \delta_{2r} \quad (106)$$

$$-\sigma_{513} = C_{19} \delta_{2r} \quad (107)$$

Similarly the signal from the left aft sensor 66 is introduced into the Bow Computer Section by the crank 132 in combination with the connecting rod 534 and the crank 133 in combination with the connecting rod 546. The rearward displacements of the connecting rods 534 and 546 are given by:

$$-\sigma_{534} = C_{18} \delta_{2l} \quad (108)$$

$$-\sigma_{546} = C_{19} \delta_{2l} \quad (109)$$

Next, the paths of the variables will be traced through the Bow Computer Section. The connecting rod 539 is pivoted at the pin 547 on the extension 548 of the arm 549 which is part of the multi-arm member 550. The arms 549 and 552 of the multi-arm member 550 form a hinge with the frame 126 about the axis 553. The arms 554 and 555, of the multi-arm member 550, form a hinge with the two-layer arms 556 and 557 of the multi-arm member 558 about the hinge axis determined by the pins 559 and 560. The geometry of the multi-arm member 550 and the connecting rod 539 is such that the forward displacement σ_{559} of the hinge axis determined by the pins 559 and 560 is equal to the aft displacement $-\sigma_{539}$ of the connecting rod 539 as given by Equation (106). Thus:

$$\sigma_{559} = C_{18} \delta_{2r} \quad (110)$$

The multi-arm member 558 has the two-layer arm 561 to which is pivoted the connecting rod 535 at its aft end 562. The righthand layer of the two-layer arm 561 and the single layer arm 563 of the multi-arm member 558 form a hinge with the multiplier shaft 564 about the hinge axis 565. The axis determined by the pins 559 and 560 and the aft end 562 of the connecting rod 535 are equidistant from the axis 565 so that the forward displacement σ_{565} of the axis 565 is given by:

$$2\sigma_{565} = \sigma_{559} + \sigma_{535} \quad (111)$$

From Equations (105), (108), (110) and (111):

$$2 \frac{\sigma_{565}}{C_{17}} = \frac{C_{18}}{C_{17}} \delta_{2r} + \frac{C_{18}}{C_{17}} \delta_{2l} - 2A_e + 2K_a \theta_b^2 \quad (112)$$

With the dimensions chosen such that:

$$\frac{C_{18}}{C_{17}} = 1 \quad (113)$$

Equation (112) becomes:

$$\sigma_{565} = \frac{C_{17}}{2} \left(\delta_{2r} + \delta_{2l} - 2A_e + 2K_a \theta_b^2 \right) \quad (114)$$

The multiplier beam 566, rigidly attached to the multiplier-shaft 564, the carriage 567 carrying the fulcrum for the beam 566, and the hydraulic repeater for adjusting the elevation of the carriage 567 with the handle 568 at the control station shown in Figure 16, are similar to the multiplier beam 329, rigidly attached to the multiplier-shaft 327, the carriage 330 carrying the fulcrum for the beam 329 and the hydraulic repeater for adjusting the elevation of the carriage 330 with the handle 343 at the control station shown in Figures 5 and 16 described in Section G of this Descriptive Appendix.

The connecting rods 569 and 570 are pivoted about the axis 571 on the two-layer extensions 572 and 573 of the crossbar 574 rigidly attached to the multiplier beam 566. From the Equation (114), the equal forward displacements σ_{569} and σ_{570} of the connecting rods 569 and 570 are given by:

$$\sigma_{569} = \sigma_{570} \equiv$$

$$- \frac{C_{17}C_{20}}{2} A_a \left(\delta_{2r} + \delta_{2l} - 2A_e + 2K_a \theta_b^2 \right) \quad (115)$$

in which A_a is a coefficient in the Specific Coupling Equations (13) and (14) and $-C_{20}A_a$ is the multiplication ratio of the multiplier beam 566 determined by the elevation of the carriage 567 which is adjusted for sea condition with the handle 568 at the control station, shown in Figure 16.

The connecting rods 514 and 546 are pivoted at the pins 577 and 578 on the cranks 579 and 580 respectively of the double crank member 582. The latter is hinged on the arms 583 and 584 of the multi-arm member 585 about the axis 586. The pins 577 and 578 are equidistant from the axis 586 so that the forward displacement σ_{586} of the axis 586 is given by:

$$2\sigma_{586} = \sigma_{514} + \sigma_{546} \quad (116)$$

The arms 583 and 584 also form a hinge for the multi-arm member 585 on the frame 126 with a hinge axis 587.

Combining Equations (103), (107), (109) and (116) gives:

$$\frac{2}{C_{19}} \sigma_{586} = \delta_{2r} - \delta_{2l} + \frac{C_{15}}{C_{19}} \theta_c \quad (117)$$

With the mechanism dimensions chosen such that:

$$\frac{C_{15}}{C_{19}} = 2K_\Theta \quad (118)$$

and Equation (117) becomes:

$$\sigma_{586} = \frac{C_{19}}{2} \left(\delta_{2r} - \delta_{2l} + 2K_\Theta \theta_c \right) \quad (119)$$

The connecting rod 588 is pivoted at the pin 589 on the two-layer arm 590 which is part of the multi-arm member 585. The connecting rod 591 is pivoted on the arm 584 about the axis 586. Each of the connecting rods 588 and 591 has an overload release. The pin 589 and the axis 586 are equidistant from the axis 587 so that the forward displacements of the connecting rods 588 and 591 are given by:

$$\sigma_{591} = \sigma_{586} = -\sigma_{588} \quad (120)$$

The connecting rods 588 and 569 are pivoted, at the pins 592 and 593 respectively on the two-layer lever 594. The connecting rod 588 has an overload release. The two-layer lever 594 is pivoted at the pin 595, on the two-layer extension 596, of the two-layer arm 597 of the multi-arm member 598. The pins 592 and 593 are equidistant from the pin 595 so that the forward displacement σ_{595} of the pin 595 is given by:

$$2\sigma_{597} = \sigma_{588} + \sigma_{569} \quad (121)$$

The two-layer arms 597 and 599, of the multi-arm member 598, with the arms 600 and 601 of the multi-arm member 602 with the pins 603 and 604 form a hinge with the hinge axis 605.

The arms 606 and 607 of the multi-arm member 602 with the two-layer extensions 608 and 609 of the frame 126 form a hinge with the hinge axis 610. The arm 611 of the multi-arm member 602 has the hole 612 into which is hooked the upper end of the tension spring 613 which is hooked at its lower end into the hook 614 screwed into the deck 38 below. The arm 611 is threaded to receive the lefthand screw 615. The tension in the spring 613 forces the lower end of the lefthand screw 615 to bear against the plate 616 mounted on the cross member 617 of the frame 126. The detent 618 together with the star wheel 619, which is part of the lefthand screw 615, prevent unintentional rotation of the lefthand screw 615. The guided connecting tube 620 is pivoted at the pin 623

to the two-layer arm 624 of the multi-arm member 602.

Rotation clockwise of the lefthand screw 615 with the knob 625 at the control station, as shown in Figure 16, rotates the multi-arm member 602 counter-clockwise about the axis 610 producing a forward displacement σ_{605} of the hinge axis 605. This displacement is proportional to the negative of the pay load and trimming adjustment term B_{ar} , in the Specific Coupling Equation (13), for the right aft hydrofoil thus:

$$\sigma_{605} = -C_{21} r_a B_{ar} \quad (122)$$

The forward displacement σ_{620} of the guided connecting tube 620 is given by:

$$\sigma_{620} = -C_{22} \sigma_{605} + C_{23} \sigma_{595} \quad (123)$$

with the constants C_{22} and C_{23} being determined by the dimensions of the multi-arm member 598.

Combining Equations (115), (119), (120), (121), (122) and (123) gives:

$$\frac{-\sigma_{620}}{C_{22} C_{21} r_a} = B_{ar} -$$

$$\frac{C_{23} C_{17} C_{20} A_a}{4 r_a C_{22} C_{21}} \left(\delta_{2r} + \delta_{2l} 2A_e + 2K_\Theta \theta_c \right) -$$

$$\frac{C_{23} C_{19}}{4 C_{22} C_{21} r_a} \left(\delta_{2r} - \delta_{2l} + 2K_\Theta \theta_c \right) \quad (124)$$

Choosing the mechanism dimensions such that:

$$\frac{C_{23} C_{17} C_{20}}{2 C_{22} C_{21} r_a} = \frac{1}{1} \quad (125)$$

and

$$\frac{C_{23} C_{19}}{2 C_{22} C_{21} r_a} = \frac{A_r}{b_w} \quad (126)$$

Equation (124) becomes:

$$\frac{-\sigma_{620}}{C_{22}C_{21}r_a} = B_{ar} -$$

$$\frac{A_r}{21} \left(\delta_{2r} + \delta_{2l} - 2A_e + 2K_a \theta_b^2 \right) -$$

$$\frac{A_r}{2b_w} \left(\delta_{2r} - \delta_{2l} + K_\Theta \theta_c \right) \quad (127)$$

The connecting rods 591 and 570 are pivoted at the pins 626 and 627 respectively on the two-layer lever 628. The connecting rod 591 has an overload release. The two-layer lever 628 is pivoted at the pin 629 on the two-layer extension 630 of the two-layer arm 632 of the multi-arm member 633. The pins 626 and 627 are equidistant from the pin 629 so that the forward displacement σ_{629} of the pin 629 is given by:

$$2\sigma_{629} = \sigma_{591} + \sigma_{570} \quad (128)$$

The two-layer arms 634 and 635 of the multi-arm member 633 with the arms 636 and 637 of the multi-arm member 638 form a hinge with a hinge axis 639. The two-layer arm 640 is a downward extension of the two-layer arm 635. The guided connecting tube 642 is pivoted on the two-layer arm 640 at the pin 643. The arms 644 and 645 of the multi-arm member 638, with the two-layer extensions 608 and 609 of the frame 126, form a hinge with a hinge axis 610. The arm 646 of the multi-arm member 638 has a hole 647 into which is hooked the upper end of the tension spring 648 which is hooked at its lower end into the hook 649 screwed into the deck 38 below. The arm 646 is tapped to receive the right-hand screw 650. The tension in the spring 648 forces the lower end of the right-hand screw 650 to bear against the plate 616 already mentioned. The star wheel and detent arrangement with the screw 650 is similar to that already described for the screw 615.

Rotation clockwise of the right-hand screw 650, with the knob 652 at the control station, as shown in Figure 16, rotates the multi-crank member 638 clockwise about the axis 610 producing a forward displacement σ_{639} of the hinge axis 639. This displacement is proportional to the negative of the pay load and trimming adjustment term B_a in the Specific Coupling Equation (14) for the left aft hydrofoil thus:

$$\sigma_{639} \equiv - C_{21}r_a B_{al} \quad (129)$$

The forward displacement σ_{642} of the guided connecting tube 642 is given by:

$$\sigma_{642} \equiv C_{22} \sigma_{639} - C_{23} \sigma_{629} \quad (130)$$

The constants C_{22} and C_{23} appearing in Equation (123) are repeated in Equation (130) because pertinent dimensions of the multi-arm member 633 are chosen similar to the corresponding dimensions of the multi-arm member 598.

Combining Equations (115), (119), (120), (128), (125) and (126) gives:

$$\frac{-\sigma_{642}}{C_{22}C_{21}r_a} = B_{al} -$$

$$\frac{A_r}{21} \left(\delta_{2r} + \delta_{2l} - 2A_e + 2K_a \theta_b^2 \right) +$$

$$\frac{A_r}{2b_w} \left(\delta_{2r} - \delta_{2l} + 2K_\Theta \theta_c \right) \quad (131)$$

The guided connecting tubes 620 and 642 transmit the forward displacements σ_{620} and σ_{642} respectively to the Stern Computer Section. The guided connecting tube 653 attached with the clamp 654 to the frame 126 transmits the position of the frame 126 to the Stern Computer Section and compensates for strain and temperature changes as will be explained below. The three guided connecting tubes 620, 642 and 653 pass through the tunnel 655 attached to the deck 38. Within the tunnel 655 the tubes 620, 642 and 653 are supported by guide bearings, such as 656, 657 and 658 which are carried by the mounting bracket 659, as shown in Figure 7.

K. STERN COMPUTER SECTION

Referring to Figure 7, the aft ends of the guided connecting tubes 620 and 642 are pivoted on the cranks 660 and 662 at the pins 663 and 664. The aft end of the guided connecting tube 653 is pivoted on the Reciprocal Function Unit 210 at the pin 665. The rearward extension of the plug and length adapter 211, of the Reciprocal Function Unit 210, is pivoted at the pin 666 on the two-layer arm 667 which is part of the multi-arm member 668. The combination of the guided connecting tube 653, the Reciprocal Function Unit 210 and the plug and length adapter 211 will be referred to herein as the connecting combination 669. The two-layer arms 670 and 672 of the multi-arm member 668 form a hinge with the brackets 673 and 674 with a hinge axis 675. The brackets 673 and 674 are rigidly fastened to the deck 38 below.

The arms 670, 672 and 667 of the multi-arm member 668 form hinges with the double crank members 676 and 677 with the common hinge axis 678.

On the right side, the rearward extending crank 679 of the double crank member 676 is pivoted on the lower end of the vertical connecting rod 680 at the pin 682. The connecting rod 680 is pivoted at its upper end on the two-layer crank 683 at the pin 684. The crank 683 is part of the double crank member 685 which is hinged on

the bracket 686 about the horizontal hinge axis 687. The bracket 686 is rigidly attached to the bracket 8 which is fastened to the transom of the boat 285. The double crank member 685 extends through both the bracket 8 and the transom of the boat 285.

The tubular crank arm 688 and the extension 689 therefrom are parts of the double crank member 685. The cam rocker 690, the cam opening 692, the cam roller 693 (not shown) and the spring 694 are similar in design and operation to the cam rocker 416, the cam opening 427, the cam roller 428 and the spring 418 described in Section H of this Descriptive Appendix.

The vertical connecting rod 695 is pivoted on the extension 689 at the pin 696. The connecting rod 695 is similar to the connecting rod 415 described for the forward servo in Section I of this Descriptive Appendix. The servo mechanism shown generally at 697, the valve 698, the servo cylinder 699 and the push rod 700 are all similar in design to corresponding components described for the forward servo.

On the left side, the design and operation of the signal transmission, caging and servo are the same as described for the right side except that the double crank member 677 is shorter than the double crank member 676.

Because the connecting rods 680, 702, 695 and 703 are vertical, horizontal deflections of the transom 285, equal changes in length of the connecting combination 669 and the guided connecting tubes 620 and 642, and changes in length of the deck 38 have no significant effect on the signal displacements σ_{ar} and σ_{al} . On this account, errors due to temperature changes of the guided connecting tubes 620 and 642 and the hull are practically eliminated by placing the connecting combination 669 in the same enclosure as the guided connecting tubes 620 and 642 to equalize temperatures and by constructing the connecting combination 669 of the same material as the guided connecting tubes 620 and 642.

The downward displacements σ_{ar} and σ_{al} of the connecting rods 695 and 703 respectively are proportional to the amount by which the aft displacements $-\sigma_{620}$ and $-\sigma_{642}$ exceed the increase in length σ_h of the Reciprocal Function Unit 210, thus:

$$C_{24} \sigma_{ar} = -\sigma_{620} - \sigma_h \quad (132)$$

$$C_{24} \sigma_{al} = -\sigma_{642} - \sigma_h \quad (133)$$

From Equations (16), (124) and (131), Equations (132) and (133) become respectively:

$$C_{24} \sigma_{ar} = C_a \left[\frac{w_{ah}}{Kq} \right] + C_{22} C_{21} r_a \left[B_{ar} - \frac{A_a}{2l} (\delta_{2r} + \delta_{2l} - 2A_e + 2K_a \theta_b^2) - \right]$$

$$\frac{A_r}{2l} (\delta_{ar} - \delta_{2l} + 2K_\Theta \theta_c) \Big] \quad (134)$$

$$C_{24} \sigma_{al} = C_a \left[\frac{w_{ah}}{Kq} \right] + C_{22} C_{21} r_a \left[B_{al} - \frac{A_a}{2l} (\delta_{2r} + \delta_{2l} - 2A_e + 2K_a \theta_b^2) + \frac{A_r}{2b_w} (\delta_{2r} - \delta_{2l} + 2K_\Theta \theta_c) \right] \quad (135)$$

Choosing the mechanism dimensions such that:

$$\frac{C_{22} C_{21} r_a}{C_a} = 1 \quad (136)$$

Equations (135) and (136) become:

$$\frac{C_{24} \sigma_{ar}}{C_a} = \frac{w_{ah}}{Kq} + B_{ar} - \frac{A_a}{2l} (\delta_{2r} + \delta_{2l} - 2A_e + 2K_a \theta_b^2) - \frac{A_r}{2l} (\delta_{2r} - \delta_{2l} + 2K_\Theta \theta_c) \quad (137)$$

$$\frac{C_{24} \sigma_{al}}{C_a} = \frac{w_{ah}}{Kq} + B_{al} - \frac{A_a}{2l} (\delta_{2r} + \delta_{2l} - 2A_e + 2K_a \theta_b^2) + \frac{A_r}{2l} (\delta_{2r} - \delta_{2l} + 2K_\Theta \theta_c) \quad (138)$$

The downward displacements σ_{ar} and σ_{al} of the connecting rods 695 and 703 are the output signals of the Stern Computer Section and the command signals to the aft servos which must produce the a_{ar} and the a_{al} that satisfy the Specific Coupling Equations (13) and (14). Since these equations must be satisfied for all conditions, any condition may be used to determine the constant of proportionality relating σ_{ar} to a_{ar} and σ_{al} to a_{al} . Take the conditions $\dot{\sigma}_{ar} = \dot{\sigma}_{al} = 0$. The right-hand members of Equations (137) and (138) are identical to the right-hand members of the Specific Coupling Equations (13) and (14). Equating the corresponding left-hand members and using the condition $\dot{\sigma}_{ar} = \dot{\sigma}_{al} = 0$ gives:

$$\frac{C_{24}}{C_a} \sigma_{af} = a_{ar} \quad (139)$$

$$\frac{C_{24}}{C_a} \sigma_{al} = a_{al} \quad (140)$$

The mechanism dimensions must be selected such that the ratio $\frac{C_{24}}{C_a}$ is suitable for the servos.

L. PROPULSION AND PROPELLER RETRACTION

The propulsion system is shown in Figure 14. Propulsion is by a standard marine propeller 710 driven by the propeller shaft 711 (not visible) supported within the streamlined housing 712 on rubber water-lubricated bearings 713 (not visible) spaced at intervals along the propeller shaft 711 and fastened within the streamlined housing 712. The forward end of the propeller shaft 711 and the aft end of the stub shaft 714 are connected with a standard universal joint 715 (not visible) within the hollow spherical joint 716. The stub shaft 714 extends through and is coaxial with a quill output shaft 717 (not visible) and the flange coupling 718 of the V-drive 719. The forward ends of the stub shaft 714 and the quill output shaft 717 are coupled with the flange coupling 718. The streamlined housing 712 is flange coupled at its forward end to the aft extension 720 of the inner spherical part 721 of the spherical joint 716. The forward extension 722 of the outer spherical part 723 of the spherical joint 716 is flange mounted to the V-drive 719. The centers of the spherical joint 716 and the universal joint 715 substantially coincide.

The streamlined strut 724 is welded to the streamlined housing 712 and extends upward within the upright accurately bored tube 725 where it is flange coupled to the center support 726. The accurately bored tube 725 is sealed watertight and clamped to the bottom of the hull and securely bolted to the hull framing. The center support 726 is sandwiched between and guided by the castings 728 and 729 (the latter not shown). The castings 728 and 729 with flanges machined to fit closely within the bore of the tube 725 support the center support 726 and the strut 724 laterally. Two rectangular blocks 730 and 731 (the latter not visible) one on the right side and one on the left of the center support 726, are pivoted to the center support 726 on the pin 732. These blocks fit into fore and aft rectangular grooves 733 and 734 (not visible) machined into the castings 728 and 729 and support the propeller assembly consisting of center support 726, strut 724, propeller 710, streamlined housing 712 and inner spherical part 721.

The hydraulic cylinder 735 has the ram 736 which is connected at its lower end to the casting 728. The propeller assembly is moved and positioned vertically by the hydraulic cylinder 735. The fore and aft location of the strut 724 is determined by the spherical joint 716.

Fore and aft motion of the propeller assembly with respect to the castings 728 and 729 is permitted by sliding of the rectangular blocks 730 and 731 in the grooves 733 and 734. The propeller 710 is positioned by the hydraulic cylinder 735, either *down* as shown when operating on hydrofoils, or *up* as shown dotted, when operating in the displacement condition with the hydrofoils retracted.

When the propeller 710 is operating in the down position and either the propeller 710 or the streamlined housing 712 strikes an object with sufficient force, a relief valve 737 in the hydraulic system allows the propeller 710 and the streamlined housing 712 to move upward under heavy resistance to reduce the impact load.

The bellows 738 is clamped and sealed to the vertical surface 739 of the hull 9 and clamped around the aft extension 720 of the inner spherical part 721. The bellows 738 allows vertical motion of the propeller assembly and seals the hull against water entry.

Sea water is taken in through the openings 740 in the streamlined housing 712, conducted forward in the streamlined housing 712 and through the aft extension 720 into the tube connection 741. The flexible hose 742 clamped to the tube connection 741 conducts the water to the engine cooling system 743. Cooling water from the engine is discharged into the exhaust system 744 in a well-known manner. The rubber propeller shaft bearings 713 within the streamlined housing 712 are lubricated by the sea water passing through the housing 712.

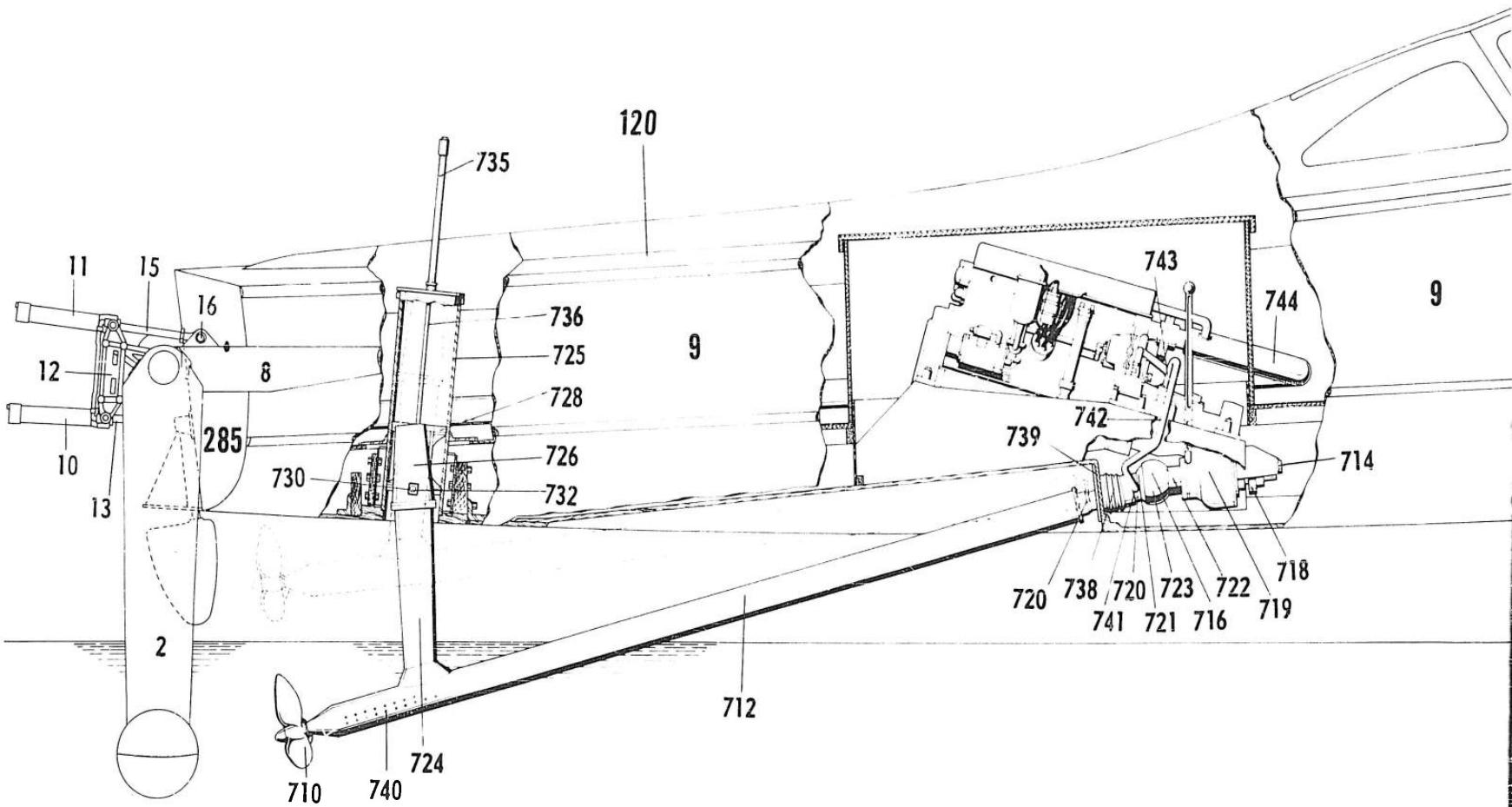
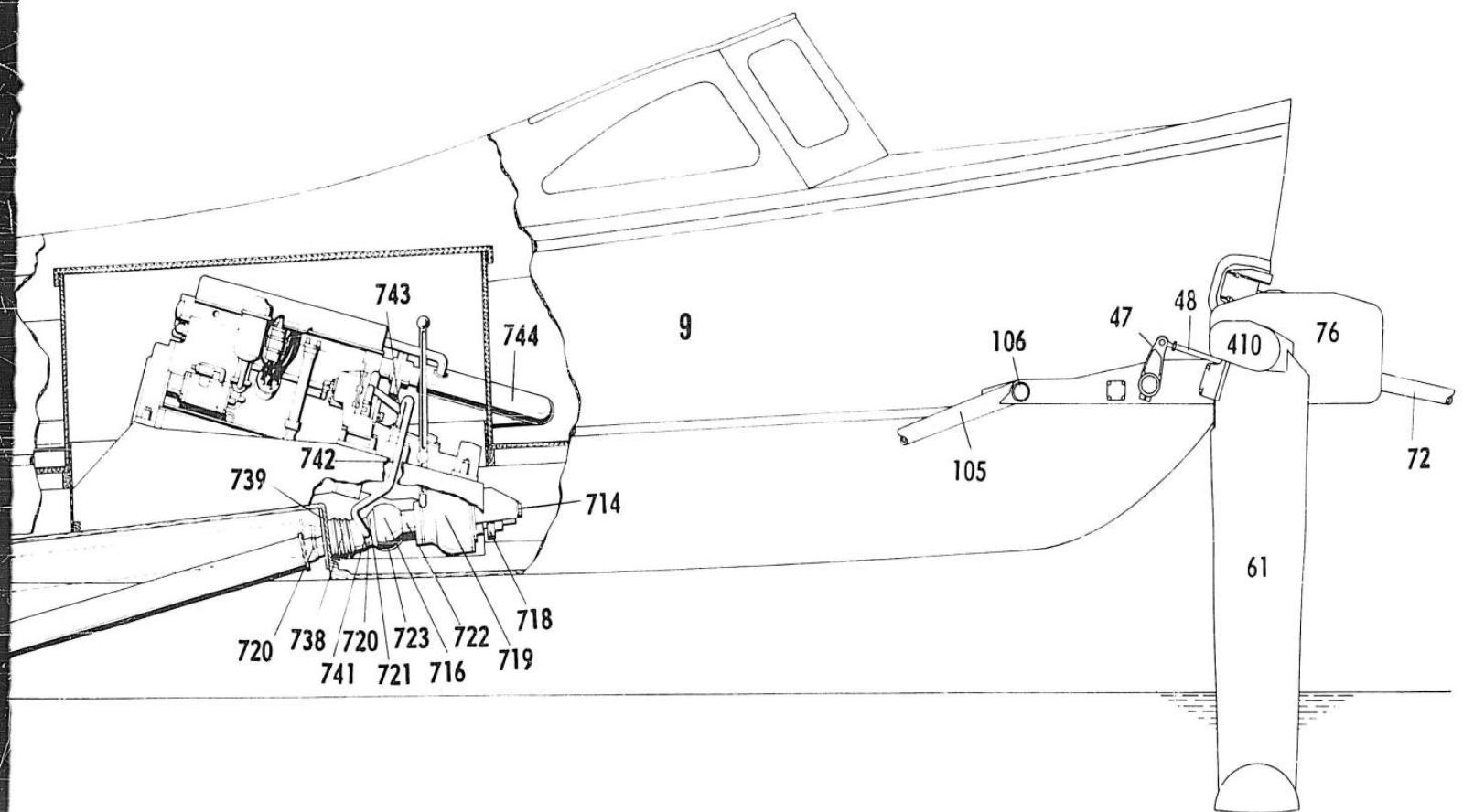


FIG. 14 PROPULSION AND PROPELLER RETRACTION - DRAWING PARTIAL SECTION ALONG VERTICAL PLANE CONTAININ

1



RACTION - DRAWING PARTIAL SECTION ALONG VERTICAL PLANE CONTAINING PROPELLER SHAFT AXIS



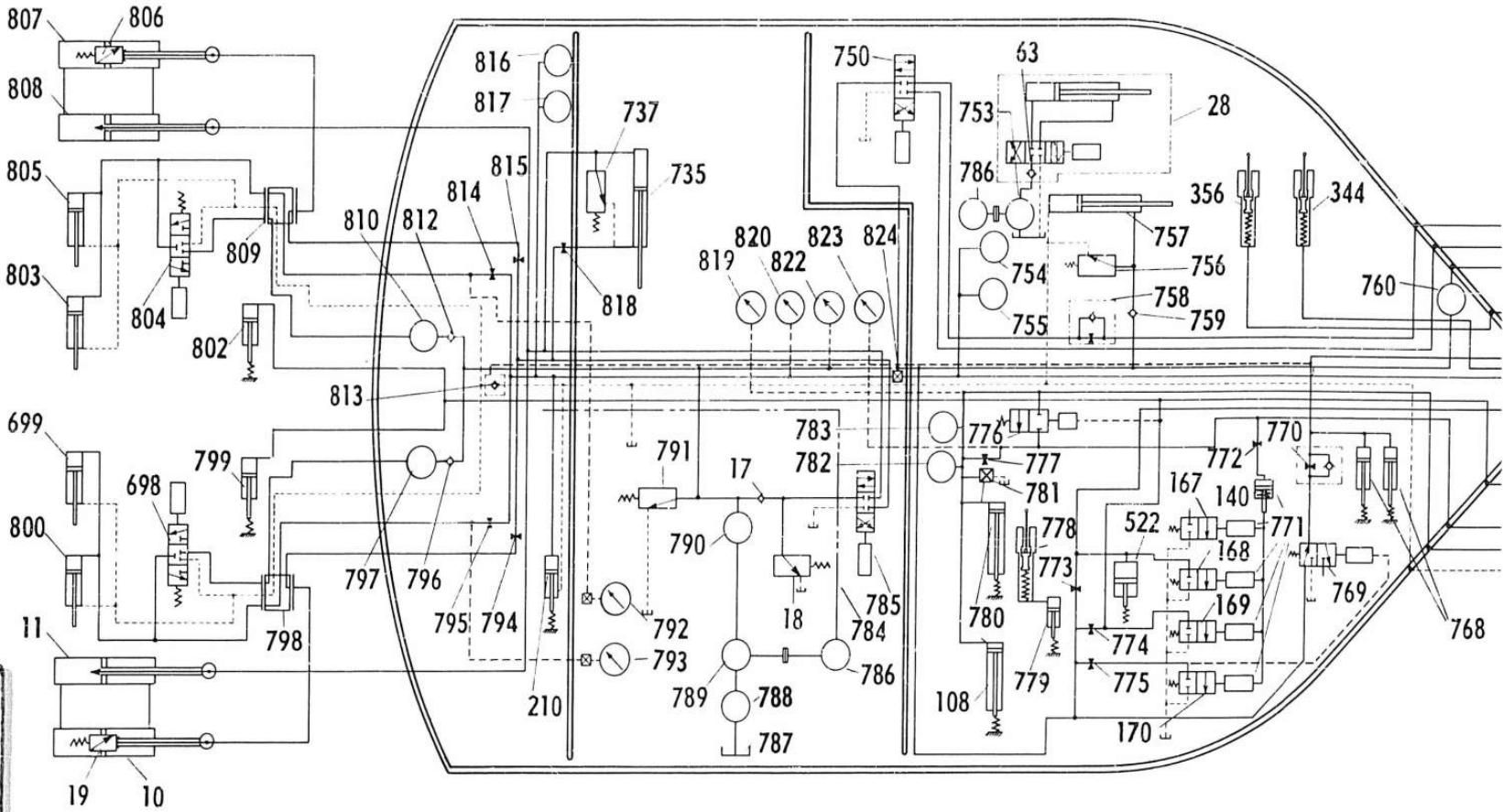
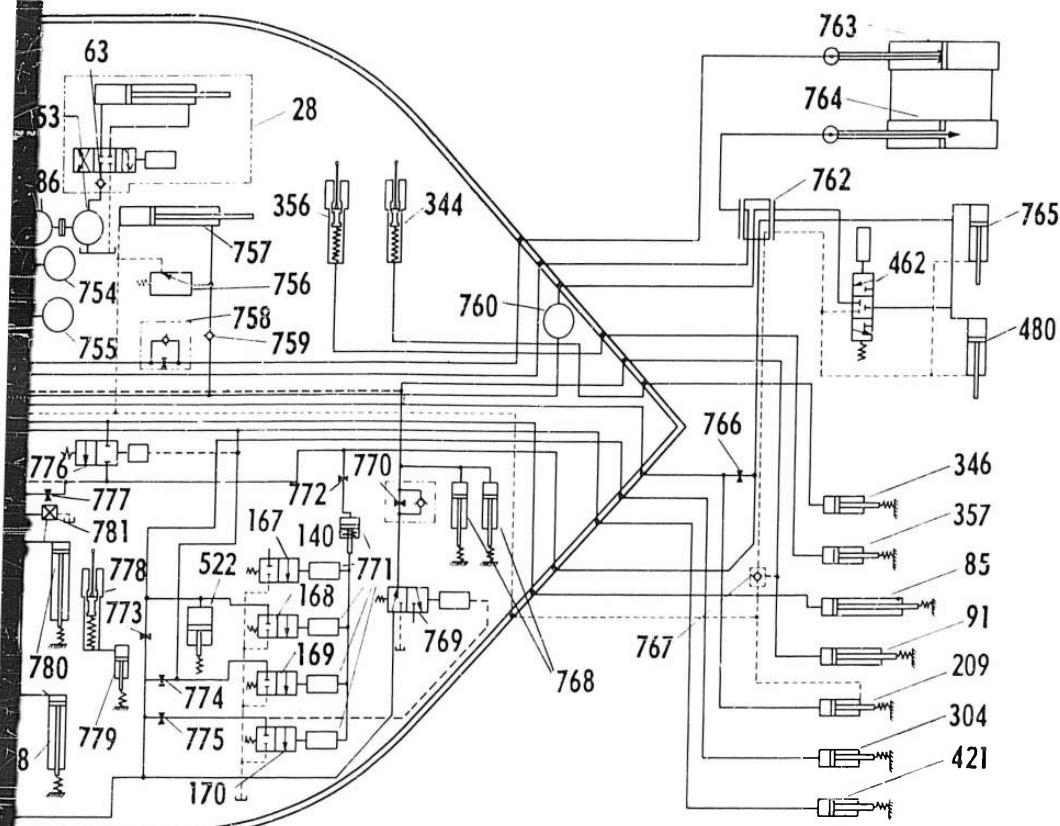


FIG. 15 HYDRAULIC CIRCUIT SCHEMATIC DRAWING

M. HYDRAULIC SYSTEM KEY

- 750 Hydrofoil Retraction Control Valve - forward - manually actuated
- 753 Power Steering Pump
- 63 Check Valve - steering impact
- 754 Accumulator - 100 psi precharge - forward Reciprocal Function Unit
- 755 Accumulator - 300 psi precharge - forward Reciprocal Function Unit
- 756 Relief Valve - impact energy absorption
- 757 Forward Impact Energy Absorption Cylinder
- 758 Restrictor - one-way forward foil counterbalance
- 759 Check Valve - impact energy absorption
- 28 Power Steering Unit
- 356 Forward Pay Load and Elevation Trim Master Cylinder
- 344 Front Multiplier Master Cylinder
- 760 Hydraulic Fluid Filter
- 762 Rotary Ring Seal Unit - forward
- 763 Forward Hydrofoil Retraction Cylinder
- 764 Forward Hydrofoil Retraction Cylinder
- 765 Left Forward Hydrofoil Servo Cylinder
- 462 Forward Hydrofoil Servo Valve - mechanically actuated
- 480 Right Forward Hydrofoil Servo Cylinder
- 766 Restrictor - to smooth Servo Pressure forward to Reciprocal Function Unit

- 767 Pilot-Operated Check Valve - proposed to restrain hydrofoil rotation during very low speed in waves and during backing
- 346 Front Multiplier Slave Cylinder
- 357 Forward Pay Load and Elevation Trim Slave Cylinder
- 85 Forward Sensor Loading Cylinder
- 91 Forward Sensor Unfolding Cylinder
- 209 Forward Reciprocal Function Unit
- 304 Forward Climb Cylinder
- 421 Forward Caging Cylinder
- 768 Rear Sensor Unfolding Cylinder
- 769 Valve - sensor unfolding - hydraulically actuated
- 770 Restrictor - one-way - sensor unfolding
- 771 Sequence Function Unit - hydraulically actuated
- 140 Sequence Function Unit Actuation Cylinder
- 772 Restrictor - remove surge pressure to Sequence Function Unit Actuation Cylinder
- 167 Cam Actuated Valve - spare
- 168 Cam Actuated Valve - climb
- 773 Restrictor - pump pressure to climb actuation
- 169 Cam Actuated Valve - uncaging
- 774 Restrictor - pump pressure to uncaging actuation
- 170 Cam Actuated Valve - sensor unfolding
- 775 Restrictor - pump pressure to sensor unfolding actuation
- 522 Rear Climb Cylinder
- 776 Valve - servo loading cylinder
- 777 Restrictor - pressure to servos
- 778 Rear Multiplier
- 779 Rear Multiplier
- 108 Right Rear Sensor
- 780 Left Rear Sensor
- 781 Valve - shut-off drain
- 782 Accumulator - pressure
- 783 Accumulator - pressure
- 784 Engine Fuel Line
- 785 Hydrofoil Reverser - manually actuated
- 786 Engine
- 18 Relief Valve
- 17 Check Valve
- 787 Hydraulic Fluid
- 788 Strainer
- 789 Pump - variable
- 790 Hydraulic Fluid
- 791 System Relief
- 792 Pressure Gauge
- 793 Pressure Gauge
- 210 Rear Reciprocal
- 794 Restrictor - rear



EMATIC DRAWING

Valve - proposed to
tion during very low
ing backing
inder
elevation Trim Slave Cy-

Cylinder
Cylinder
tion Unit

Cylinder
- hydraulically actuated
isor unfolding
it - hydraulically actu-

Actuation Cylinder
ge pressure to Sequence

Cylinder

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ensor unfolding

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- 776 Valve - servo cylinder pressure to sensor loading cylinder - hydraulically actuated
- 777 Restrictor - to prevent servo cylinder surge pressure to sensor loading pressure
- 778 Rear Multiplier Master Cylinder
- 779 Rear Multiplier Slave Cylinder
- 108 Right Rear Sensor Loading Cylinder
- 780 Left Rear Sensor Loading Cylinder
- 781 Valve - shut-off - sensor loading pressure to drain
- 782 Accumulator - 100 psi precharge - sensor loading pressure
- 783 Accumulator - 300 psi precharge - sensor loading pressure
- 784 Engine Fuel Line
- 785 Hydrofoil Retraction Control Valve - rear - manually actuated
- 786 Engine
- 18 Relief Valve - rear hydrofoil retraction to drain
- 17 Check Valve - rear hydrofoil impact retraction
- 787 Hydraulic Fluid Reservoir
- 788 Strainer
- 789 Pump - variable delivery
- 790 Hydraulic Fluid Filter
- 791 System Relief Valve
- 792 Pressure Gauge - left rear servo cylinder
- 793 Pressure Gauge - right rear servo cylinder
- 210 Rear Reciprocal Function Unit
- 794 Restrictor - right rear hydrofoil retraction

- 795 Restrictor - to smooth Servo Pressure right rear to Reciprocal Function Unit
- 796 Check Valve - proposed to inhibit roll in case of accidental loss of hydraulic pressure
- 797 Hydraulic Fluid Filter
- 798 Rotary Ring Seal Unit - right rear
- 799 Right Rear Caging Cylinder
- 698 Right Rear Hydrofoil Servo Valve - mechanically actuated
- 699 Right Rear Left Hydrofoil Servo Cylinder
- 800 Right Rear Right Hydrofoil Servo Cylinder
- 19 Piston Relief Valve - impact energy absorption
- 10 Cylinder - right rear hydrofoil retraction - impact energy absorption
- 11 Cylinder - right rear hydrofoil retraction
- 802 Left Rear Caging Cylinder
- 803 Left Rear Right Hydrofoil Servo Cylinder
- 804 Left Rear Hydrofoil Servo Valve - mechanically actuated
- 805 Left Rear Left Hydrofoil Servo Cylinder
- 806 Piston Relief Valve - impact energy absorption
- 807 Cylinder - left rear hydrofoil retraction - impact energy absorption
- 808 Cylinder - left rear hydrofoil retraction
- 809 Rotary Ring Seal Unit - left rear
- 810 Hydraulic Fluid Filter
- 812 Check Valve - proposed to inhibit roll in case of accidental loss of hydraulic fluid
- 813 Pilot-Operated Check Valve - proposed to restrain hydrofoil rotation during very low speed in waves and during backing
- 814 Restrictor - to smooth Servo Pressure left rear to Reciprocal Function Unit
- 815 Restrictor - left rear hydrofoil retraction
- 816 Accumulator - 100 psi precharge - rear Reciprocal Function Unit
- 817 Accumulator - 300 psi precharge - rear Reciprocal Function Unit
- 818 Restrictor - propeller counterbalance
- 737 Relief Valve - impact energy absorption - propeller retraction
- 735 Cylinder - propeller retraction
- 819 Pressure Gauge - smoothed servo pressure to sensor loading cylinders
- 820 Pressure Gauge - smoothed servo cylinder pressure to Reciprocal Function Unit
- 822 Pressure Gauge - system pressure
- 823 Pressure Gauge - front servo cylinder pressure
- 824 Shut-Off Valve

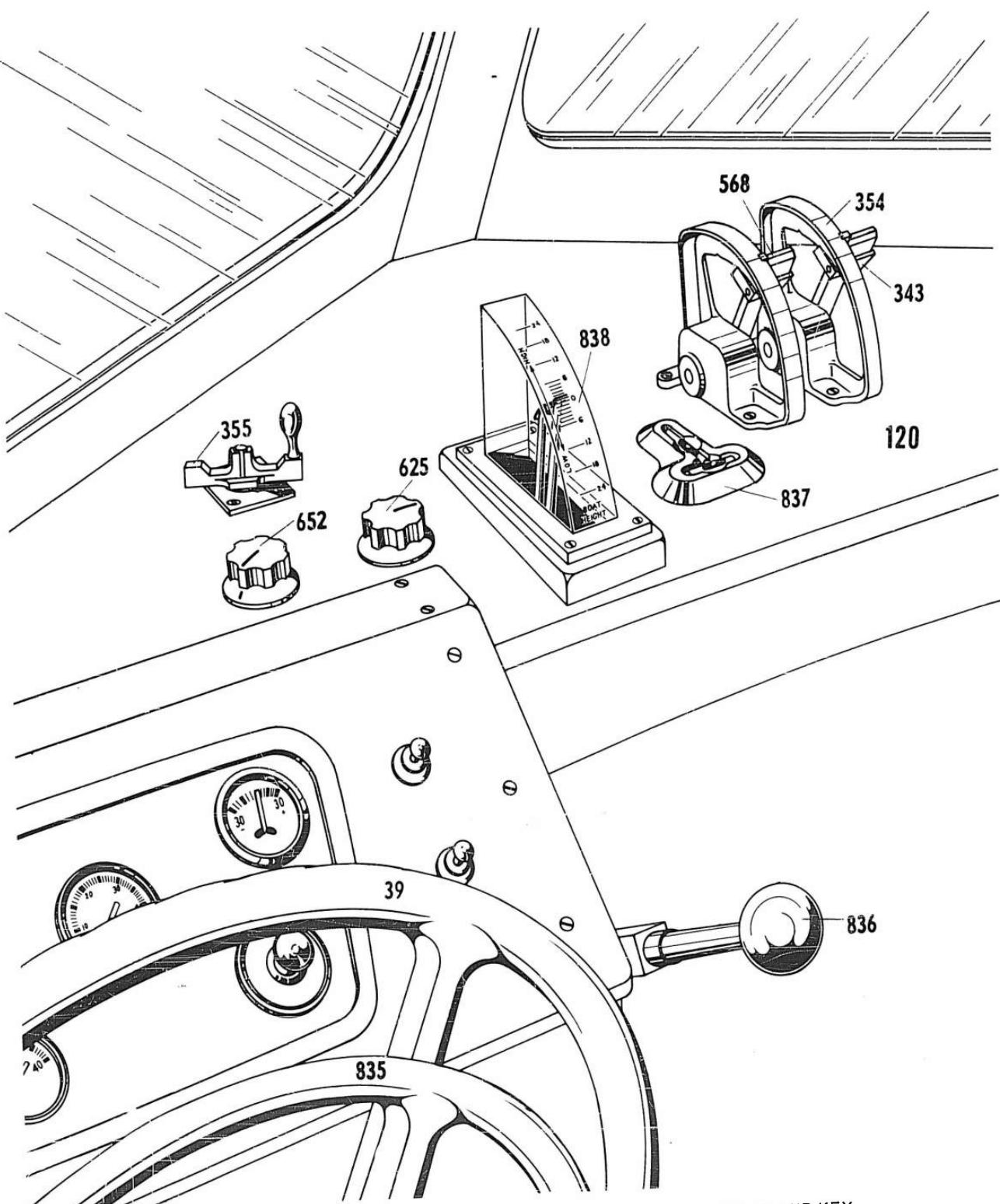


FIG. 16 CONTROL STATION - FRAGMENTARY PERSPECTIVE DRAWING AND KEY

N. CONTROL STATION KEY

- 39 Large steering wheel for steering with hydrofoils extended
- 835 Small steering wheel for steering with hydrofoils retracted. Wheel automatically centers when released to avoid heavy drag of rudder at take-off if not centered.
- 836 Standard single control handle - forward, reverse, engine clutch and throttle control
- 837 Pitch and roll spirit indicators

- 838 Indicators of flying elevation of hull at the aft sensor locations. Normal height: zero
- 355 Pay load and trim adjustment for forward hydrofoil
- 625 Pay load and trim adjustment for right aft hydrofoil
- 652 Pay load and trim adjustment for left aft hydrofoil
- 568 Sea condition adjustment. Only two settings required - one for head sea and one for following sea
- 343 Sensing ratio adjustment for forward hydrofoil
(Believed not required on a prototype)

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