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ACTIVATED ANTIROLLING TANKS TESTED ON THE USS PEREGRINE (E-AM373)

by

Francis F. Vane

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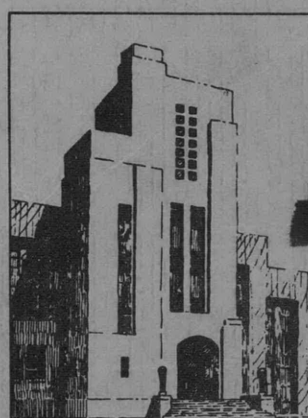

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ACTIVATED ANTIROLLING TANKS TESTED ON THE USS PEREGRINE (E-AM373)

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ABSTRACT

An activated-tank system for ship stabilization, based on Minorsky's principles, was installed on the USS PEREGRINE (E-AM373) and was tested and evaluated. First, salient characteristics of ocean waves and of wave action on ships are described, various types of stabilizers are briefly described and evaluated, and Minorsky's analysis of the activated-tank stabilizer is outlined. Then the activated-tank installation on the USS PEREGRINE (E-AM373), using variable-pitch impellers with amplidyne drive, is described. The specifications and calibrations of the various units of the control system are given. The principles and descriptions of operation of the pumping system including the amplidyne drive are presented, as are a description of the electrical supply and the instruments installed for calibrating and testing.

The installation was tested in dry dock, in calm water, and finally at sea. Considerable reduction in roll was achieved but far less than indicated by theory and design calculations. The insufficient stabilization was judged to be due to: (1) The modifications to the ship which altered its designed stability, (2) the inherent faults of the improvised equipment used, and (3) not being able to operate the system to full capacity. Based on experiences with the PEREGRINE installation, various modifications for simplification, reduction of power and space, and reliability of operation are suggested. It is recommended that a permanent installation of an activated-tank system be made now, and a procedure for this installation is suggested. It is also pointed out that the control system and the amplidyne drive are also applicable to an activated-fin device. A bibliography of stabilization and a list of patents of stabilization devices are presented in appendices.

INTRODUCTION

Since the first shaped vessel, probably a dugout canoe, was built, man has been interested in reducing the oscillations of vessels in seaways. Not only has man worried about the safety of his voyage, but the inner man has been much perturbed by these vessel oscillations. With the widespread commercial introduction of steam propulsion and with increased knowledge of the mechanics of ship motions, ship designers in the late 19th century became encouraged to try to reduce the oscillations of vessels in seaways and have been trying to this day.

The United States Navy has been especially interested in the problem of reducing roll for a number of reasons. Stabilization against rolling would reduce considerably the problems of rocket launching, launching and recovering aircraft by a carrier, cargo handling from vessels such as AKA's in open roadsteads during landing operations, refueling of ships at sea, safety of ships in hurricanes and typhoons, comfort and better surgical conditions on hospital ships, and would ensure greater comfort for personnel, which is especially important prior to landing operations. Reducing roll would simplify gunfiring, particularly of 40-mm and 20-mm antiaircraft mounts; however, for larger guns this problem has been minimized by modern gun computers which automatically correct for the ship's motion. An important reason for stabilization is the increased fuel economy resulting in greater range or smaller fuel load by decreasing that portion of propulsive resistance caused by rolling. Another important advantage of stabilization is the reduction of stresses imposed on ship structure by heavy seaways.* Recent developments of stratagems and tactics involving new types of vessels participating in fleet actions have now reinvigorated the interest of the Navy in reducing ship oscillations.

In the mercantile passenger service, greater comfort will have to be provided on ocean liners to meet the challenge of transoceanic airlines. Smaller rolling oscillations will aid in rescue and salvage operations at sea, will aid in construction of quays and other structures on exposed shores, and ease the ruggedness and assure more continuous service on weather and lightships. Drilling of underwater oil wells could be facilitated by the use of stabilized supply and work ships.

Efforts have been concentrated on rolling—which is of greater magnitude, and at the same time is easier to counteract, than pitching. Many systems have been devised and tried out, most of which have proved impractical because of operating difficulties, cost, power consumption or space requirements.

Antirolling devices which have actually been tried on ships may be classified in two ways, one, as either activated or passive systems, two, as operable under all conditions or only while a ship is in motion. Activated systems—those where a signal proportional to the rolling motion of the ship is imparted to the stabilizing mechanism—are the stabilizing fins, activated antirolling tanks, gyroscopic stabilizers, and moving weights. Passive systems—those where complete dependence is placed on the inherent design of the device and/or on the speed of the ship—are bilge keels and passive Froude and Frahm tanks. Devices effective whether the vessel is stationary or under way

*It is believed by some that the impact of waves against a ship's unyielding sides stresses the ship more than if the ship had been unstabilized. This belief is held only for breaking waves but doesn't hold even under this condition. A nonstabilized ship does not always yield to waves; there are times when the ship's motion is out of phase with the waves. Then the ship rolls toward the waves with appreciable velocity, and therefore the blow is greater than if the ship were stabilized. A completely stabilized ship is subjected to stresses on the slope of a wave because buoyancy is not the same in the two sides. In a nonstabilized ship, stresses, for this reason, may be greater than in a stabilized ship. In addition the nonstabilized ship is subjected to inertia forces during rolling. For example, on a ship rolling $\pm 30°$, at 50-ft above the center of rolling, accelerations of 1/3 g will be encountered at the ends of the swings.

are passive and activated antirolling tanks, gyroscopic stabilizers, and moving weights. Bilge keels and stabilizing fins are effective only when the vessel is under way, and their effectiveness depends in part on the speed of the ship.

The activated-tank system depended upon the development of the theory of control and on enough technological advance to be able to devise a reliable, sensitive, control system. The system is one in which water or other fluid is transferred back and forth across a ship applying restoring forces to the ship in proportion to a signal. The rate of water transfer is dependent upon the pitch of the impellers, and the pitch is controlled by the signal. The activated-tank system has certain attractive advantages in that (1) the effectiveness of the system is not dependent upon ship speed, (2) in case of breakdown there are no dangerous mechanical or stability features and there is still some possibility of stabilization under certain operating conditions as a passive tank system, (3) the weight of the system can be refined to as low as 1 percent of the ship's weight when the water or oil in the stabilizing system is considered to be reserve supply, (4) the power requirement can be made as low as 1 to 2 percent of ship power, and (5) the space requirement can be made low, again considering water or oil as reserve supply.

Minorsky developed a theory of activated tanks, made laboratory tests, and had an activated tank system installed on the USS HAMILTON (DD141) for full-scale tests with temporary inboard tanks. The control system was designed by the Sperry Gyroscope Company, the pumping system with hydraulic pitch mechanism was designed and built by the I.P. Morris Division of the Baldwin-Southwark Company, and an angular accelerometer was used as the initiating signal device, designed by the U.S. Experimental Model Basin and modified for this application. The HAMILTON was rolled in still water by means of the activated tank system as much as 18° from the vertical. Minorsky's theory and model tests indicated that the installation could stabilize the HAMILTON in a seaway almost completely up to rolls of 30°. However, there were difficulties while operating at sea due to the creation of self-excited oscillations in the control system, which manifested themselves in violent fluctuations of the hydraulically operated pitch mechanism. Upon the loan of 50 destroyers to Great Britain and the immediate assignment of the HAMILTON to coastwise patrol in 1939, the work on the activated tanks was terminated abruptly, precluding the opportunity to analyze fully and to correct the difficulties that arose.

In 1948 the interest of the Bureau of Ships in stabilizing ships was augmented by the manifold new problems in ship operation based on evaluation of the requirements for postwar use of Naval vessels. On recommendations made by the David Taylor Model Basin[1] to the Bureau of Ships at the latter's request, the Bureau of Ships directed[2] the Taylor Model Basin to proceed with the problem of ship stabilization by means of activated tanks.

Since it had been decided to use the impellers or hydraulic oscillators saved from the HAMILTON test and since the capacity of these impellers was suitable for a vessel having a product of displacement times metacentric height (W·GM) of slightly less than 3000 ton-feet, the AM371 Class of minesweepers was selected for testing. This class was the only group of vessels that had all of these advantages: Average displacement (light service condition) 1100 tons; average metacentric height, GM, of about 2.68 ft; rolling period of about 9 seconds; and availability of both a-c and d-c power.

The Bureau of Ships therefore asked the Chief of Naval Operations to make available a 220-ft minesweeper, similar to one of the AM371 Class, for continuance and completion of the ship stabilization project.[3] The Bureau of Ships directed the Norfolk Naval Shipyard to design and install certain ship stabilization equipment in collaboration with Model Basin personnel.[4] The Chief of Naval Operations assigned Project Bu/S146/S29, which required assistance to the Bureau of Ships on ship stabilization tests, to Commander, Operational Development Force, authorizing direct communication with the Bureau of Ships, and recommended that the USS PEREGRINE (E-AM373) be made available for the project.[5] The Commander, Operational Development Force directed the Commanding Officer, USS PEREGRINE (E-AM373) to provide services on this project[6]. The stabilization project, requiring the cooperative services of a number of groups, was then ready to roll.

The program in the design of the PEREGRINE installation was to utilize the original impellers or hydraulic oscillators, to select or design appropriate accelerometers, to redesign completely the control system in conformance with Minorsky's principles of control, to adapt amplidyne drive units to the pitch shafts, and to design the tanks and ducts to suit the ship. It was hoped that this plan would help to overcome the difficulties encountered on the HAMILTON installation.

The installation, test, and performance of the PEREGRINE installation constitute the principal topic of this report. Enough theoretical discussion is presented to serve as a basis for understanding the operation of the activated-tank and similar stabilizers. The components of the PEREGRINE installation are described with respect to design data, specifications, and performance The tests of the entire installation—drydock, calm water, and sea tests—are described, and the results are presented and evaluated. Specifications are made for the future installation of either activated tanks or

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[1]References are listed on page 153.

activated fins. Fuel economy, power consumption, weight, and space requirements are considered.

It was thought desirable, however, to make this report serve also the purpose of an introduction to the problem of stabilization in general. Hence, material is first presented concerning ocean waves and their action on ships. Various types of stabilizers are described briefly and evaluated. The discussion of the PEREGRINE installation, forming the body of the report, is followed by a bibliography of stabilization and a list of patents on stabilizing devices.

WAVE ACTION ON SHIPS

The nature of ocean waves, their action on ships, the response of ships to them, and the stability characteristics of ships will be described. These give the forces acting on a rolling ship which have to be counteracted by a stabilizing system in order to diminish appreciably the motion of a ship in a seaway.

NATURE OF OCEAN WAVES

The questions that arise concerning ocean waves are their mechanism, how they are generated, their commonly encountered magnitudes, and their velocities. Much has been written about waves[7,8,9,10] and observational data have been obtained, so that now there is reconciliation between many of the hypotheses and the observed phenomena; however, there are still unexplained aspects of their behavior. These, fortunately, are not pertinent to the ship-stabilization problem. The trochoidal theory and more recently the hydrodynamic theory of wave mechanism and generation have been described in the literature. However, the parameters—height, length, and velocity—are described and their commonly encountered magnitudes are presented since they are of importance in designing a ship-stabilization installation.

Height

The height of waves depends on the strength of the wind and the duration of time the wind acts on the waves. There is a maximum height to which a wind of given velocity may lift waves, no matter how long the wind blows or the length of the fetch (the distance the waves have traveled under the influence of the wind). The height of fully developed waves, in feet, based on many observations, is 0.5 to 0.7 of the strength of the wind in statute miles per hour. Thus a 40-mph gale would produce waves 20 ft or more in height. The upper limit of wave height based on many observations is believed to be 48 to 50 ft unless two waves are superimposed. In Table 1 where wind velocity is compared with height of waves, it is assumed that the wind blew long enough and that there was space enough for the waves to develop. Waves, in general, are highest not when the winds are strongest but after the wind begins to die down or the wave leaves the region of the wind. Under the action of strong winds, the tops of the waves are blown off and tend to fill the troughs.

TABLE 1

Height of Waves for Various Wind Velocities*

Wind Velocity statute mph	Height of Waves feet, double amplitude	Wind Velocity statute mph	Height of Waves feet, double amplitude
10	3	40	26
15	6	45	32
20	10	50	35
25	14	55	38
30	17	60	42
35	21		

*Information given in Tables 1 through 6 was taken from References 7, 8, and 9.

Stormy latitudes experience storms of about equal severity; therefore, storm waves are about the same. This is true of the North Atlantic, South Atlantic, and North Pacific where the fetch is about the same. Here storm waves average 18 to 20 ft high with individual waves over 30 ft and with rare waves over 50 ft due to superposition. The year-round occurrence of various wave heights between Newfoundland and England in the North Atlantic are listed in Table 2.

TABLE 2

Occurrence of Heights of Waves—The Year Round—Between Newfoundland and England in the North Atlantic

Wave Height feet, double amplitude	Occurrence percent
0 - 3	20
3 - 4	20
4 - 7	20
7 - 12	16
12 - 20	10
20 +	13

Strong winds can build up a high sea in an hour or two. With a wind velocity of 36 mph, 20-ft waves can be built up in 40 hours. For this wind velocity, the maximum wave height possible is 26 to 27 ft. However, waves cannot build up to maximum height unless the distance or fetch across which the wave travels and the wind blows be long enough, therefore large seas can develop only in large bodies of water. The heights of waves are proportional to the square-root of fetch until they approach a maximum. For the same wind velocity the heights of waves for various fetches are listed in Table 3. Hurricanes do not produce the largest seas because of variation of the direction of the wind.

TABLE 3

Heights of Waves at the Same Wind Velocity for Various Fetches

Fetch miles	Wave Height feet, double amplitude
10	5
20	7
50	11
200	21
300	26

Length

As the seas increase, the lengths of the waves increase more rapidly than the heights. Low waves either may be very young waves of short period or very old waves of long period. The ratio of length to height is usually less than 30 to 1. For strength calculations of ship hulls the standard trochoidal wave has been assumed to have a ratio of 20 to 1 and a length equal to that of a ship. The ratios determined for various wave heights are listed in Table 4. The average lengths of waves compared with their average height are listed in Table 5 for different wind velocities. The maximum observed length was 2719 ft.

Velocity

The velocity of waves is directly dependent on wind when waves gain energy from it and their height is increasing. During this interval the velocity of waves is about 0.8 that of the wind. Waves may travel faster than the wind and still gain energy and swells have traveled as much as 40 mph in

TABLE 4

Ratios of Lengths to Heights for Various Wave Heights

Wave Height feet, double amplitude	Ratios: L (length) to H (height)			Number of Cases
	Maximum	Minimum	Average	
6 - 9	24	13	17	23
10 - 14	24	10	18.6	12
20 - 29	23	13	15.5	5
30 - 39	18	11	14	11
40 - 50	16	14	15	2

TABLE 5

Average Lengths and Average Heights of Waves
for Various Wind Velocities

Beaufort Scale	Description	Wind mph	Average Height feet, double amplitude	Average Length feet
2	Light breeze	4	2	52
4	Moderate breeze	14	6.6	124
6	Stiff breeze	20	10	261
8	Moderate gale	30	14.5	383
10	Strong gale	67	31.4	827

a dead calm. Large waves travel faster than small waves and longer waves (generally larger) faster than shorter waves. Free wave velocity is approximately 1.3 $\sqrt{L}$ nautical mph. The relationship of period, length, and velocity is:

$$C = 1.34 \sqrt{L} = 3.03\ T$$

where C is in knots,
L is in feet, and
T is in seconds.

Certain values are tabulated for this relationship in Table 6. The longest period recorded has been 26 seconds. The average state of the trade-wind region of the North Atlantic is one with waves of 215-ft length, 5.8-second period, and advance of form of 22 nautical mph. Large waves encountered in

TABLE 6

Wave Length, Wave Velocity, and Wave Period

Period seconds	Velocity nautical mph	Wave Length feet
2	6.1	20.3
4	12.1	82.0
6	18.2	183.3
8	24.3	328.8
10	30.3	511.7
12	36.3	740.0
14	41.4	1003.6
16	48.5	1298.9
18	54.9	1659.6
20	60.5	2046.7
22	66.6	2499.4
24	72.6	2948.7

storms in this region have periods from 6 to 9 seconds. Waves continue advancing in the direction of propagation and as winds shift other waves are propagated and superimposed upon the original waves.

General

The length-height ratio of waves reaches a lower limit of 7:1 at which point the wave becomes unstable. In high winds the crests are lifted off. Apparently, according to more recent analysis of observations and based largely on stereophotograms obtained by Schumacher and Weinblum,[11,12] 7:1 waves are quite common particularly in wave-generating areas. Wind and current against waves help make them steep. From these stereophotograms many waves were measured up to 60 ft high, with wave slopes of portions of the waves up to 24°, and wave lengths from 200 ft to more than 600 ft. Large sea waves have been found to have periods of 12 to 14 seconds and wave lengths over 600 feet, contrary to former belief. Larger ships therefore encounter waves of length equal to or greater than that of the ship which indicates that possibly the stresses in the hulls of these ships have been underestimated. For short portions of waves, slopes of 23 degrees were frequently measured. For 50- to 65-ft portions of waves mean slopes of 12° were measured. The mean slopes for purposes of ship-stabilization study have been assumed to be of the order of less than 5° although this was for the ship on the supposedly steepest portion of the wave without rolling or pitching. The 12° figure must be modified since it is the angle of the wave rather than the ship. However, this mean slope of 12° indicates that effective wave slope of the ship in waves may be greater on occasion than the somewhat less than 5° assumed previously.

MOTION OF SHIPS IN OCEAN WAVES

In considering the effect of ocean waves upon the motion of ships, the primary effort in study and observation has concerned rolling, although some study has been made of pitching and yawing. The remaining motions are so well hidden in the general confused motion of a ship in ocean waves as to make it difficult if not impossible, to separate and distinguish these small-motion components even with precise instruments for measuring them. It was thought[13] at first that rolling was due to the impact of waves on ship sides. Since rolling occurs, however, in long smooth swells with little or no impact of the waves on the ship, this view was abandoned. Froude established the modern theory of rolling, while Rankine, Robb, and Kriloff contributed additional analyses.[14,15,16,17] According to this current theory, rolling is caused by the resultant of fluid pressure acting on the hull of a ship as a whole. A simplified form of the theory was employed by Minorsky in his designs, and this theory, although admittedly imperfect, was taken as a basis for the design to be described later. A brief sketch of this simplification will now be given.

On the slope of a wave the vessel would have approximately equal pressures on both sides of the hull if the masts were normal to the waves. For an actual ship the masts would not be normal to the surface wave for this condition of equal pressures but to a wave contour below the surface depending on the shape and displacement of the ship. The slope is therefore less because the wave becomes flatter with depth. So far it has been assumed that the ship has no moment of inertia and that a ship is always normal to the wave and rolls at the wave period and changes amplitude with amplitude changes from wave to wave, see Figure 1. With great inertia the ship would ride with masts almost vertical, see Figure 2. But the average ship has only a moderate amount of rolling inertia. The heeling force may be considered due to a series of impulses at the slope midpoints acting alternately in opposite directions, the first impulse starting the rolling oscillations and the remainder of the rolling motion depending on the ratio of the natural period of rolling to the period of the impulses. If these two periods happened to be nearly equal, the angle of roll would build up, the rolling in each direction exceeding that from which the ship recovered in the opposite direction. At resonance, the

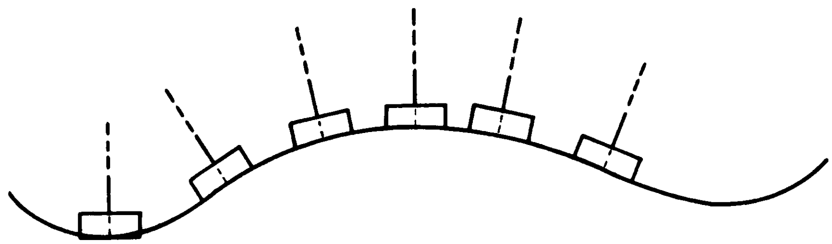

Figure 1 - Successive Positions on a Wave of a Ship Having Zero Moment of Inertia

The dotted lines indicate the normals to the wave.

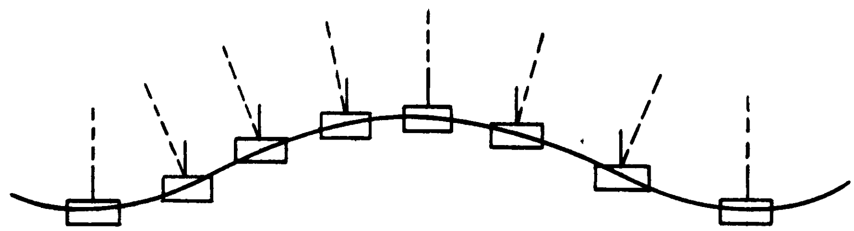

Figure 2 - Successive Positions on a Wave of a Ship Having Extremely Large Moment of Inertia

The dotted lines indicate the normals to the wave.

ship would eventually capsize if it were not for certain restraints. The rolling of a ship is not isochronous, since as the amplitude of roll changes its period changes. Also the waves are not strictly periodic. More important, however, the resistance to rolling motion increases until it balances the rolling forces, and the amplitude of roll no longer increases. The rolling periods of ships in calm water are in the range of 4 to 24 seconds. What happens at sea depends on the periods of ocean waves, which seldom exceed 9 seconds.

This has been a somewhat pictorial exposition of the rolling of a ship. A more detailed and quantitative discussion is necessary for clarification of the forces involved.[10] A basic factor in the rolling characteristics of a ship is the transverse stability. An undisturbed ship in calm water is acted upon by two vertical forces, the downward force of weight and the upward force of buoyancy. If the ship is in equilibrium, at rest, the two forces must be equal, opposite, and vertical. This means the center of buoyancy is on the same vertical line as the center of gravity. The center of gravity is G and the center of buoyancy B, Figure 3a. If the ship is inclined by an external force, a moment formed by the weight and buoyancy tends to restore a stable ship to even keel. As the vessel is inclined more and more, the center of buoyancy shifts along a curve relative to the ship. The center of curvature of this buoyancy curve is the metacenter, M. For small angles of

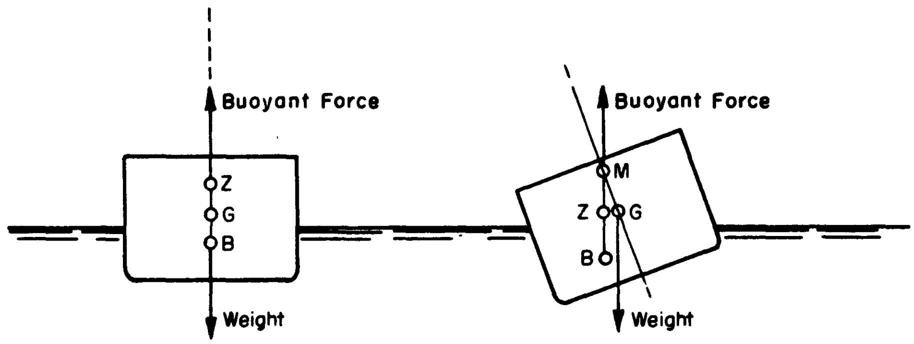

Figure 3a - Ship Vertical

Figure 3b - Ship Inclined

Figure 3 - Forces Acting on a Ship in Calm Water

inclination the metacenter remains at substantially the same point on the ship. The center of buoyancy varies at an especially large rate at 0°. The distance between the metacenter M and the center of gravity G is a measure of the static ship stability and is called the metacentric height, GM. The statical righting moment with the ship inclined is (Figure 3b) $\Delta GZ = \Delta GM \sin\theta$ where Δ is the displacement equal to the weight and equal to the upward buoyant force, GZ is the righting arm, GM is the initial metacentric height, and θ is the angle of inclination. The righting moment varies with the angle of inclination and with the loading of a vessel. Curves may be computed for various angles and loadings.

The rolling motion of a ship can now be determined from basic Newtonian equations by making simplifying assumptions. The ship, in still water, is given an initial inclination by application and removal of an external moment and allowed to roll freely. The equation of motion is

$$I\ddot{\theta} + M = 0$$

where I is the mass moment of inertia about the longitudinal axis through the center of gravity, including a small addition for virtual mass of the water,

M is the righting moment, and

θ is the angle of inclination of the ship from the vertical.

So far only two forces are acting on the ship, the weight of the ship and the buoyant reaction of the water. The restoring moment, M, includes not only the buoyant-force component of the resultant of the water pressure on the ship but also a static component due to the slope of the wave, and another component due to the dynamic pressure caused by relative motions of the water and the ship.

Up to now the ideal fluid was assumed to have no viscosity. Without viscosity, at synchronism, a ship would increase its rolling until it capsizes. Sources of passive resistance to rolling are:

1. Frictional resistance of water on wetted surface.

2. Generation of water waves by the ship's rotation.

3. Similar resistances due to the action of air upon the parts of the ship above water.

The third can be considered negligible compared with the other two forms of resistance. Designating the moment of frictional resistance of rolling as M_{rf}, and the moment of wave generation as M_{rw}, the equation may now be expanded to include these:

$$I\ddot{\theta} + M + (M_{rf} + M_{rw}) = 0$$

where the first term includes the mass and acceleration, the second term the net restoring moment due to weight and waves, and the third term the moments due to frictional resistance and wave generation. The first term, $I\ddot{\theta}$, of course is proportional to the displacement of the ship times the radius of gyration squared. The righting moment of weight and buoyancy, M, is approximately proportional to θ, the angle of inclination. The moment of frictional resistance, M_{rf}, is proportional to $(\dot{\theta})^2$, while the moment of wave generation resistance, M_{rw}, is proportional to $\dot{\theta}$.

From the equation of rolling in terms of ship dimensions and from considerations on which the equation is based, certain properties of ships in seaways which are of interest in connection with the problem of stabilization may be summarized.

1. The period of resisted roll is very little greater than the period of unresisted roll.

2. The amplitude of rolling increases as the ratio of ship-rolling-period to apparent-wave-period approaches unity.

3. Increasing the radius of gyration of a ship causes only a relatively small increase in the period of a ship.

4. An increase in the metacentric height of a ship decreases the period of roll and increases the possibility of synchronous rolling.

5. A ship parallel to waves rolls and pitches; one perpendicular to the waves only pitches.

6. Rolling and pitching create a gyrostatic couple causing yawing.

For a ship of particular size and with minimum requirements of metacentric height very little can be done, in designing the ship, to reduce its amplitudes of rolling; consequently many attempts have been made to design stabilizing devices.

PRINCIPLES AND EFFECTIVENESS OF SYSTEMS TRIED

As can be seen from the equation of rolling stated previously, there are a number of ways in which the rolling amplitude may be diminished or quenched. One is to increase resistance by means such as bilge keels, and another is to apply restoring couples equal and opposite to the couples causing rolling. The stabilizing systems tried so far are described in three groups: (1) Moving weights and gyroscopes, (2) bilge keels, hydrofoil fins, and activated fins, (3) passive and activated antirolling tanks. The first group contains systems that are generally in abeyance at the present time because of unsolved deficiencies and dangers inherent in them—such as danger to a ship in case of failure of the system and unacceptable sacrifice of space, power, and weight in a ship. The second group includes bilge keels, hydrofoil intermittent bilge keels, and activated fins. Bilge keels are extensively used, and there is now active interest in the other two. The third group includes the various inactive and active tank systems tried, culminating in the Minorsky system of activated tanks. The design of the system for the PEREGRINE stabilization was based on the principles of the Minorsky system.

MOVING WEIGHTS AND GYROSCOPES

A saloon hung on gimbals operated by a hydraulic cylinder was tried by Sir Henry Bessemer in 1875 on a cross-channel steamer.[13] Attempts to control the cylinder motion by a gyroscope failed owing to the wrong principle of installation; later, attempts were made to control the cylinder by hand. The system did not work, and the idea was abandoned. Several trial installations were made of moving weights; these also proved impractical.

A quadrant on a shaft, moved from side to side by means of a crank operated by a hydraulic cylinder, was devised by Sir John I. Thornycroft and installed on a ship in 1891.[13] The controlling device was a pendulum, which operated the valves of a hydraulic cylinder through electrical relays. During operation at sea a single amplitude of rolling of 18° was reduced to 9° under certain sea conditions. The method was abandoned, however, because response and accuracy of control was erratic and varied under different sea conditions.

Crémieu devised a moving-weight system where a truck moved on curved tracks in a chamber filled with viscous fluids.[13] The clearance was adjustable so that the damping could be set. The apparatus was essentially a pendulum. Similarly, Norden installed a weight that was moved on rails transversely across a deck, the length of travel being proportional to a signal.

In the activated moving-weight systems such as those of Thornycroft and Norden, difficulty was encountered in making the response of the weight quick enough and accurate enough. A good deal of the trouble lay in the signal device and the control mechanism, but there were also difficulties in the power drive. In both the activated systems such as Norden's and the inactivated systems such as those of Crémieu the main objection has been the hazard and possible damage to a ship in case of failure of the system. For this reason there is at present no enthusiasm for trying to perfect this type of stabilizer.

Schlick first used the moment of a gyroscope in 1907 to stabilize a ship.[18,19,20] The gyroscope produces an external moment proportional to the mass and angular velocity of the heavy rotating element and to the angular velocity of the axis of rotation. When a ship rolls, this axis precesses, and a stabilizing couple is imposed on the ship perpendicular to the plane of precession. Schlick used a steam-driven gyroscope. On one test an average roll of 7° single amplitude with a maximum of 20° was reduced to less than 1/2°. An installation adequate for stabilization proved to be quite heavy, at least 5 percent of the ship's weight. Another objection was the inability to get good stabilization under any but a particular set of wave conditions owing to control of the precessional moment by a fixed weight and to the use of hand-operated brakes to prevent excessive precessional oscillations.

In the Sperry gyroscope stabilizing equipment, these objections were eliminated.[21] The use of electric drive reduced the relative weight of the installation. Rather than the gyroscope itself responding to the rolling of the ship by precessing, a small sensitive gyroscope responding to rolling accelerations was used to cause precession of the main gyroscope by a motor drive. Therefore adequate antirolling moment could be applied as the ship started to roll, thus preventing large angles of rolling from building up. On the SS CONTE DI SAVOIA[21] the installation weight of the Sperry system was 1.72 percent of displacement, and the volume 1.4 percent of gross tonnage. For a period of operation of 18 months the average reduction of roll was 44 percent. This system has been the most satisfactory so far in stabilizing a ship with a practical installation over a long period of time. However, the limit of stabilization seems to be about 50 percent.

Gyroscopes, however, cannot suppress yaw-heel. Also, when gyroscope cages reach their limit stops before the completion of a roll the precessional velocity and therefore the restoring couple become zero. A great hazard is probable mortal injury to the ship if in naval vessels the flywheel is hit and shattered by a projectile, or if, on either naval or mercantile ships, the bearings should seize.

BILGE KEELS, HYDROFOIL FINS, AND ACTIVATED FINS

Various types of protruding longitudinal fins, such as bilge keels, have been tried in order to reduce rolling. These types eventually evolved, about 1870, into the present bilge keels now almost universally installed on ocean-going ships. Bilge keels act to diminish roll by acceleration of water in contact with them and by eddying around their edges. Bilge keels increase the resistance to rolling, R_{rf}, and the virtual mass of the water, R_{rk}, carried with the hull in rolling; both of these are proportional to the square of the velocity of rolling $(\dot{\theta})^2$. A very slight increase of the period of roll results from installation of bilge keels. Since both the frictional and virtual mass effect depend on the velocity of rolling, the bilge keels are effective only at larger angles of roll, and their effectiveness increases, too, with the speed of the ship.[21] Bilge keels have been effective in improving rolling characteristics at a slight sacrifice to propulsive efficiency since the resistance to the forward motion of a ship is slightly increased by their installation. But under rolling conditions this is more than offset by the reduction in resistance to forward motion caused by the diminution of rolling. The application of adequate bilge-keel length and width has been limited by practical considerations of minimizing ship propulsive resistance, by the necessity for strengthening the line of connection to the ship if the bilge keels extend out too far, and by berthing and docking difficulties if they protrude too far. Ships with large midship-section coefficients usually have bilge keels omitted in the midship section because of their ineffectiveness in this region. The effectiveness of bilge keels is greater in ships of low mass moment of inertia, that is, in merchant ships and in naval vessels without armor. The bilge keels have little effect when the ship is at dock or is moving slowly as is necessary in a heavy seaway to prevent excessive working of the hull.

Interrupted bilge keels have been tried,[22] each being of hydrofoil section to give the most lift compatible with minimum ship resistance. However, the total resistance of these small hydrofoil sections has been greater than expected; and each interferes, even with staggered arrangement, with the flow of water behind it, so that successively from fore to aft each section becomes less effective. It has been proposed that several large ones instead of many small ones be installed to reduce the percentage of resistance to ship propulsion and to reduce considerably the interference between hydrofoils. The hydrofoil sections under these conditions are believed to be considerably more effective than bilge keels. Here the question arises whether the difficulties in berthing and docking are great enough to preclude the use of large fins protruding beyond the maximum breadth and depth of the ship.

Another method of stabilizing is that of activated fins. In this type of installation two hydrofoil fins are installed, one on each side of a ship. They are connected by a driving mechanism in such a manner that when one is tilted up the other is tilted down. A velocity signal from a gyroscope actuates the motion of the blades, which have only three positions, namely, zero angle of attack, a position tilted in one direction and another tilted in the opposite direction. As long as a signal in one direction continues, regardless of amplitude, the fins remain in the position for that direction. As the direction of the signal changes, the fins are turned through zero to the position in the other direction. In 1889 Thornycroft patented such a system, but it never was built.[23] Motora independently developed the same system, and it was installed on a number of Japanese ships.[24] On one installation, a 2211-ton ship, the roll was reduced from 20° to 2°; the equipment weighed 15 tons and used 30 horsepower. The system was driven by gears with clutches for retracting the fins.

The Denny-Brown system[25] is similar except that the fins are tilted and retracted by means of hydraulic cylinders. It has been installed on over 100 British ships. The efficiency of the fins varies as the square of the velocity of the ship and therefore is much greater at higher ship speeds and at larger angles of roll. Objections have been raised against the activated fins, and they do have their disadvantages. There are openings in the hull which, if not properly strengthened, are potential sources of flooding. The fins are usually made retractable for berthing and docking purposes since they extend beyond the extreme breadth and depth of a ship, and also in order to reduce propulsive resistance under conditions of little rolling. Providing space for retracting the fins adds considerably to the volume of the installation aboard ship; this space requirement has been a principal objection to the system. It has been proposed recently by the Bureau of Ships that the fins be installed without provision for retraction, the belief being that berthing and docking difficulty will not be too serious. In this way, not only is less space taken up on a ship but the mechanism is simpler and weight is saved. On the other hand, the propulsive resistance would be higher when the ship is progressing in calm water, but it is believed that with proper hydrodynamic design this increase in resistance can be reduced to an acceptably small amount. Possible yawing oscillations because of the activated fins would have to be carefully investigated. Helmsmen on some naval vessels would become frantic if yawing became any greater than it is now. On ships with automatic pilots, refinement of the robot may be necessary.

PASSIVE AND ACTIVATED ANTIROLLING TANKS

Antirolling tank systems apply couples to oppose the forces of rolling. Froude in 1874 used water chambers in the upper part of the ship for stabilizing. The free-surface effect of these water chambers lengthened the period of rolling of a ship and reduced the metacentric height. The water flowing to the low side of the ship counteracted the righting moment, thus reducing rolling. These Froude water chambers were abandoned because of the reduction of righting moment and because when synchronization of the roll of the ship and of the flow of water in the chambers occurred the rolling increased.

Frahm was the first to utilize the U-tube type of antirolling tank.[26] At this time the Frahm system seemed to have two advantages, availability of room for installation above the machinery spaces, and the feature that, with the horizontal leg above the center of gravity, the moment due to horizontal acceleration of the water acts in the same direction as the statical moment of the water in the vertical legs.

It is well at this time to give some details of the design of the Frahm tanks since tank design is a basic need in the activated tank system as well as in the Frahm passive tank system. Essentially, for each installation two tanks, one port and one starboard, are needed plus a crossover duct, of smaller section than the tanks connecting the tank bottoms. In addition, an air duct connecting the tops of the tanks is installed. In this way we have a U-tube with unequal sections of horizontal and vertical portions. For the Frahm tanks to operate properly the water in the tanks must have a period of transfer approximately equal to the period of roll of the ship. The period of transfer is controlled by a valve in the air duct.

Several variations of the Frahm system have been installed. In one variation the tank tops were vented to the atmosphere with no crossover air ducts. In another, no horizontal leg was installed and the bottoms of the tanks were open to the sea. The Frahm passive antirolling tanks have been installed on many ships including freighters, submarines, and the BREMEN and EUROPA. Attempts have been made to activate these tanks so that greater stabilizing action could occur. Variation of compression of air over the water in the tanks was tried. Minorsky suggested another method, that of pumping water in proper phase relative to a correct signal and reducing the energy needed by benefitting from the period of free oscillation of the water equal to that of the period of roll of the ship. The Minorsky system is discussed in the following section.

In many of the systems tried out on shipboard that have just been described, the values cited in the references for the efficacy of the systems were those for optimum conditions and are not to be construed as indices of average performance. Under certain circumstances many of these systems performed poorly. These adverse characteristics are described in the pertinent references.

THE MINORSKY ACTIVATED TANK SYSTEM

The Minorsky system of stabilization with activated tanks was based on Frahm's passive tanks with the addition of control equipment and provision for the pumping of the liquid in the tanks to increase the effectiveness of the moment of the liquid acting in contra-direction to the moment causing rolling. A certain small amount of rolling motion is necessary, since an error signal is needed at the initiation of roll to order the pumping of sufficient liquid in a proper interval of time to prevent any increase in the initial roll from developing. Minorsky developed the principles, ignoring certain minor features for simplicity, and devised a system of control. Then at the Material Laboratory, New York Naval Shipyard, he had built a 1/5-scale tank model and tested it to determine the optimum combination of parameters. Finally a full-scale installation of two tank-systems was tried on the USS HAMILTON (DD141). In this installation variable pitch impellers or oscillators, in effect axial-flow pumps, were installed. The impellers were turned continuously in one direction by geared electric motors with flywheels, and the pitch of the blades was changed by a hydraulic system controlled by an electronic differentiating and phasing system actuated by an accelerometer. During the calm-water self-rolling tests on the HAMILTON, single amplitudes of induced rolling up to 18° were obtained. From the 1/5-scale model tests and by calculation there were indications that if a single amplitude of 20° was obtained on the self-rolling tests, then 30° rolls could be quenched at sea. Difficulties with the hydraulic pitch mechanism and parasitic oscillation in the hydraulic system and in the electronic control system prevented obtaining stabilization data at sea. Unfortunately, there was time neither for analysis nor for remedy of the difficulties, since the HAMILTON was hurriedly assigned to coastwise patrol in view of the world situation in 1939. The work of Dr. Minorsky and his associates is described principally in References 27 through 40.

The following approximate analysis is essentially that of Minorsky and contains the ideas that guided the design to be described. Shortage of time prevented a re-examination of the analysis, a study of which is being made elsewhere.* The equation of motion of the ship, including the effect of the stabilizer, is for simple harmonic excitation

$$I\ddot{\theta} + k_1\dot{\theta} + S + Wh\theta + Wh\Theta \sin \omega t = 0$$

where θ is the absolute angle of rolling,

- W is the ship displacement,
- h is the metacentric height,
- I is the moment of inertia of the ship about a longitudinal axis through the center of gravity,
- S is the stabilizing moment,
- Θ is the angle of the maximum wave slope,
- k_1 is a constant representing the natural resistance to rolling due to skin friction, wave generation, etc., and
- ω is the apparent circular frequency of the waves.

In deriving this equation from the one previously written, the moment M is taken to consist of the usual restoring moment due to buoyancy $Wh\theta$ and a moment $Wh\Theta \sin \omega t$ due to the waves. The latter moment is based on Froude's assumption that the moment due to a given wave slope Θ is the same as the moment due to buoyancy arising from an inclination of the ship through the angle Θ. It is recognized that this assumption may involve a considerable error. The terms $M_{rf} + M_{rw}$ were written by Minorsky in the form $k_1\dot{\theta} + k_2\dot{\theta}^2$. The presence of the quadratic term in this expression, however, requires numerical solution of the equation. In practice the term $k_2\dot{\theta}^2$ is relatively small, at least when the ship is stabilized; hence this term was omitted, leaving only the term $k_1\dot{\theta}$ which represents linear damping.

The constant k_1 is inherent in the ship to be stabilized. Its value can be inferred from the curve of declining angles. If the ship is started rolling in still water and then left to itself, its motion is represented by a solution of the rolling equation taken with $S = 0$ and $\Theta = 0$. This solution, representing the curve of declining angles, is

$$\theta = \theta_0 \, e^{-k_1 t/2I} \cos 2\pi t/T$$

where T represents the period of roll or

$$T = 2\pi\sqrt{\frac{I}{Wh}}\left(1 - \frac{k_1^2}{4I\,Wh}\right)^{-1/2}$$

*Stanford University under an ONR contract is now developing a new analysis of ship stabilization based on modern servomechanism theory.

In practical cases it is sufficiently accurate to write

$$T = 2\pi\sqrt{\frac{I}{Wh}} \qquad \frac{2\pi k}{\sqrt{gh}} \qquad \text{where } I = Wk^2$$

If r represents the ratio of two successive amplitudes of roll to the same side, defined so as to be greater than unity, then

$$r = e^{k_1 T/2I}$$

whence k_1 can be calculated as

$$k_1 = \frac{2I \ln r}{T}$$

where ln stands for the natural logarithm.

After the stabilizer has been installed, the value of k_1 can also be found by using the stabilizer itself to roll the ship in calm water.

The stabilizing moment S can now be made equivalent to an additional linear damping. A control system can be designed that will cause S to vary in proportion to $\dot{\theta}$, at least within limits, so that

$$S = k_s \dot{\theta}$$

where k_s is a constant depending upon the characteristics of the pumping system and on the sensitiveness of the control system. Then the equation of stabilized rolling is obtained in the form

$$I\ddot{\theta} + (k_1 + k_s)\dot{\theta} + Wh\,\theta + Wh\,\Theta \sin\omega t = 0$$

The steady-state solution of this equation is

$$\theta = -\theta_a \sin(\omega t - \psi)$$

where θ_a is the amplitude of residual roll and ψ is the phase angle between the waves and the roll of the ship, or

$$\theta_a = \frac{Wh\,\Theta}{\sqrt{(Wh - I\omega^2)^2 + (k_1 + k_s)^2\omega^2}}$$

$$\psi = \tan^{-1}\frac{(k_1 + k_s)\omega}{Wh - I\omega^2}$$

This solution represents a residual roll of a certain amplitude. The general solution of the differential equation would also contain a transient term, but this term represents a damped roll that will soon die out.

If the stabilizer is turned off, then, in the equation, $k_s = 0$, and the amplitude of residual roll becomes

$$\theta_{a0} = \frac{Wh\,\Theta}{\sqrt{(Wh-I\omega^2)^2+k_1^2\omega^2}}$$

Thus activation of the stabilizer reduces the amplitude of roll in the ratio

$$\frac{\theta_a}{\theta_{a0}} = \left[\frac{(Wh-I\omega^2)^2+k_1^2\omega^2}{(Wh-I\omega^2)^2+(k_1+k_s)^2\omega^2}\right]^{1/2}$$

At resonance, where $Wh = I\omega^2$, this ratio of reduction takes on its maximum value for a given value of ω:

$$\frac{\theta_a}{\theta_{a0}} = \frac{k_1}{k_1 + k_s}$$

It is clear from these formulas that the residual roll can be decreased without limit by increasing the sensitiveness of the control system and thereby increasing the value of k_s. In practice, a residual roll of a certain amplitude is necessary in order to operate the controls. Nevertheless the simple ideal case in which the roll is completely quenched is of use in discussion.

The stabilizing moment required to quench the roll completely, found by putting $\theta = \dot{\theta} = \ddot{\theta} = 0$ in the rolling equation, is

$$S = -Wh\,\Theta \sin \omega t$$

Let S_0 be the maximum moment that the stabilizer can apply to the ship, with the tank on one side full. Then the amplitude of S cannot exceed S_0. Thus the maximum wave slope against which the stabilizer can be completely effective has an amplitude $\Theta = \Theta_0$ where

$$\Theta_0 = \frac{S_0}{Wh}$$

The value of Θ_0 is seen to be equal to the maximum angle of static list that can be produced by means of the stabilizer.

Wave slopes below θ_0 represent the roll-quenching range of the stabilizer. Rolling due to greater amplitudes of wave slope cannot be quenched completely. If it is assumed that in waves of greater amplitude than θ_0 the stabilizer will operate so as to apply a stabilizing moment $S = -S_0 \sin \omega t$, the rolling equation becomes what it would be without stabilizer in a seaway of amplitude $\theta - \theta_0$. Thus any excess of wave amplitude above the limit of the roll-quenching range will generate rolling as it would on an unstabilized ship. In such a seaway the amplitude of roll will merely be decreased by a constant amount equal to the amplitude of roll in a seaway at the limit of the roll-quenching range.

In practice these relations will be somewhat modified because of the residual roll that is necessary to cause the controls to operate.

So far, the seaway has been assumed to be exactly periodic, which it never is. To treat the general case, it would be necessary to replace $\sin \omega t$ by $f(t)$; but the mathematical solution then becomes complicated.

Rolling is usually observed, however, to follow the following pattern. At intervals the amplitude of roll becomes almost zero, then the amplitude increases during a number of rolls to a maximum and then decreases again to almost zero, and such variation is repeated time after time. Rolling of this sort would be caused by the superposition of two trains of waves of slightly different frequency. Let the wave slope be represented by

$$\theta \sin(\omega_0 + \epsilon)t + \theta \sin(\omega_0 - \epsilon)t$$

where ω_0 is the circular frequency of natural roll of the ship or

$$\omega_0 = \sqrt{\frac{Wh}{I}} = \frac{2\pi}{T}$$

where T is the rolling period, and ϵ is a small constant. Then the angle of roll without the stabilizer will be

$$\theta = \theta_{a1} \sin(\omega_0 + \epsilon)t + \theta_{a2} \sin(\omega_0 - \epsilon)t$$

where

$$\theta_{a1} = \frac{Wh\,\theta}{\sqrt{I^2\epsilon^2(2\omega_0 + \epsilon)^2 + k_1^2(\omega_0 + \epsilon)^2}}$$

$$\theta_{a2} = \frac{Wh\,\theta}{\sqrt{I^2\epsilon^2(2\omega_0 - \epsilon)^2 + k_1^2(\omega_0 - \epsilon)^2}}$$

If the ratio ϵ/ω_0 is small, θ_{a1} and θ_{a2} are nearly equal. Then it is clear that the two terms in θ will at certain times almost cancel each other, and at other times will be in the same phase and so will be additive. The interval between times of minimum θ will be of length $t_1 = \pi/2\epsilon$ and it will include about n rolls where

$$n = \frac{\omega_0 t_1}{2\pi} = \frac{\omega_0}{4\epsilon}$$

Thus

$$\frac{\epsilon}{\omega_0} = \frac{1}{4n}$$

The value of n may range roughly from 10 to 20.

If, now, the stabilizer is turned on, k_1 is replaced by $k_1 + k_s$. Thus, dropping ϵ in comparison with ω_0, the stabilizer reduces the amplitude of roll in the approximate ratio

$$\left(\frac{4I^2\epsilon^2 + k_1^2}{4I^2\epsilon^2 + (k_1 + k_s)^2}\right)^{1/2}$$

In terms of n and of r, or the ratio of decrement in the inclining experiment, this ratio of reduction can also be written

$$\frac{\dfrac{\pi^2}{4n^2(\ln r)^2} + 1}{\dfrac{\pi^2}{4n^2(\ln r)^2} + \left(1 + \dfrac{k_s}{k_1}\right)^2}$$

Again the roll can be reduced in any desired ratio by refining the controls so as to increase k_s sufficiently, although the necessary sensitivity will be somewhat greater than in the case of pure synchronous rolling.

It is not intended to imply that intermittent rolling in an actual seaway is caused by two superposed wave trains of constant but slightly different frequencies. The cause in an actual seaway is to be found rather in a variation of the wave amplitude and frequency. The mathematical treatment of the actual case is too complicated to be undertaken here, however, and it is believed that the demonstration of the successful operation of the stabilizer in the similar but simpler case just treated justifies the conclusion that the stabilizer would be equally successful in an actual seaway.

An attempt has been made here to outline and paraphrase Minorsky's analysis of ship stabilization in order to tie in the description of the PEREGRINE installation. Two of Minorsky's papers[29,35] were followed principally in preparing this section. Minorsky treats in addition different methods of solutions of the fundamental equations with fewer simplifications of initial assumptions than shown here. He also treats in detail the hydrodynamic behavior of the impeller and the kinetic reaction of the impellers and considers the analysis of the servomechanism features of the control. However, a completely integrated specific solution of the ship-stabilization problem is needed to give a solution for the entire frequency range of interest. Such a solution would include adequate allowance for the effects of the inertia of the moving water in tanks and ducts.

For the PEREGRINE installation the control system was designed to give water displacement proportional to velocity of rolling as the primary control while the pitch of the impeller blades was the secondary control. Also, float-level control was used to keep the water from drifting to either end of the system. Since retarded action occurred in differentiating circuits and since time was needed to pump water, rolling acceleration was the original signal source and higher derivative terms, $\dddot{\theta}$ and $\overset{iv}{\theta}$, were built into the control system. In this way adjustments could be made to get the same effect as in nonretarded systems; that is, water level proportional to angular velocity of ship rolling. A description of the PEREGRINE installation is presented in the next section.

OPERATION AND CALIBRATION OF THE ACTIVATED-TANK INSTALLATION

A minesweeper, the USS PEREGRINE (E-AM373), was selected for the installation of an antirolling system because its characteristics were similar to those of the USS HAMILTON (DD141) on which the impellers or hydraulic oscillators had been used previously. Four tanks were installed amidships outboard above the main-deck level, two port and two starboard, Figure 4. The two forward tanks port and starboard, were connected together by means of elbows to the shell plating and transverse crossover ducts through the hull, to form one tank system, while the two after tanks, port and starboard, formed another tank system. Above the tanks were installed the impeller and pitch motors and mechanism as well as the amplidyne amplifiers; these were protected by housings constituting the port and starboard motor rooms. The impeller and pitch mechanisms for the forward tank system were contained in the starboard motor room, those for the after tank system in the port motor room. The elbows from the tanks were protected from the sea by sponsons which also served as the supporting cantilever framing for the tanks and motor rooms. A control room was constructed on the boat deck between the two motor rooms and connected to them by skywalks. The control room housed most of the signal-control equipment and all of the testing and recording equipment. Accelerometers, the primary signal source, were installed well forward of the tanks and crossover

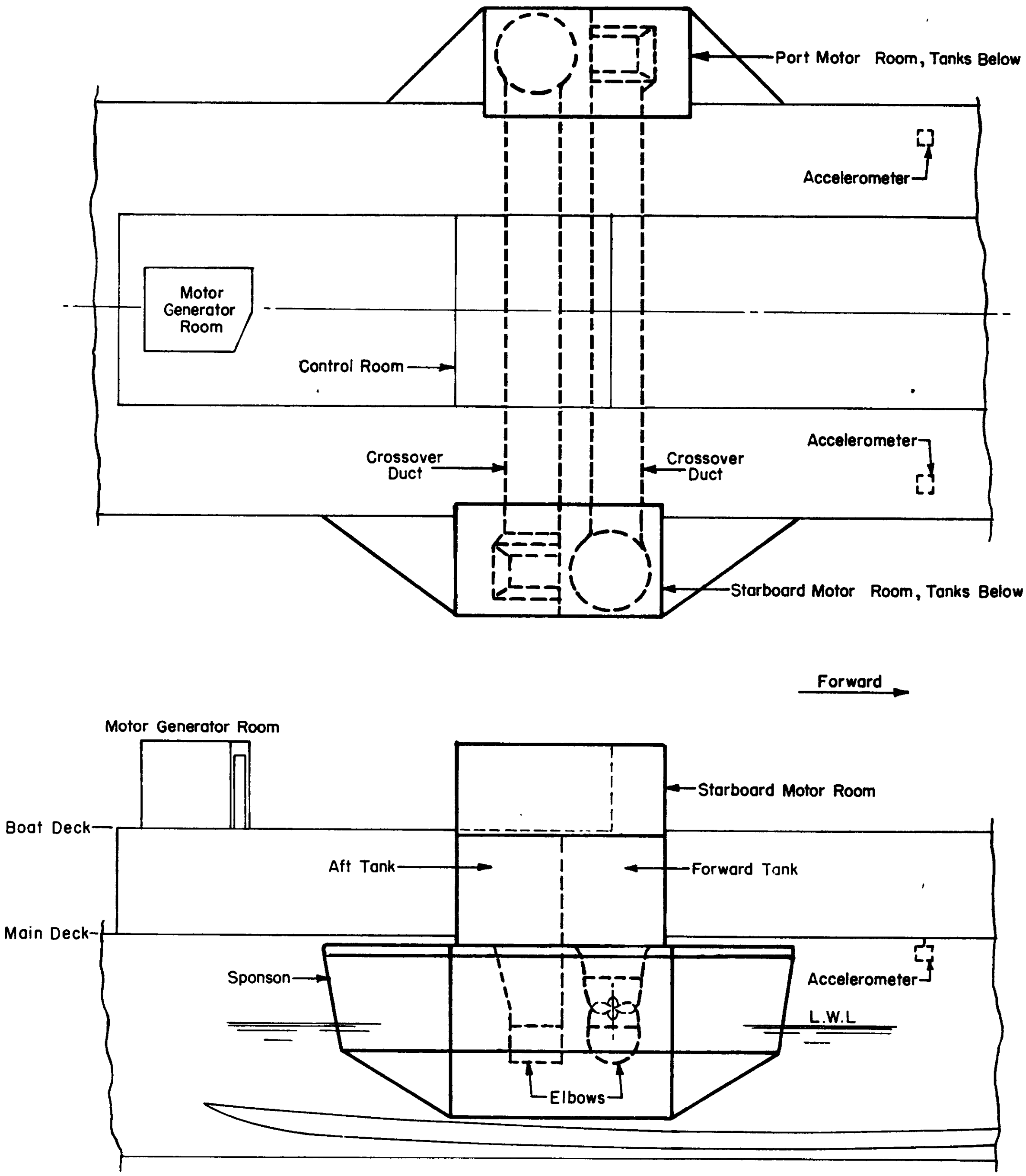

Figure 4 - General Arrangement of the Activated-Tank Installation on the USS PEREGRINE (E-AM373)

ducts on the under side of the main deck in the forward crew's berthing space. A motor-generator room was installed on the boat deck aft of the control room and supplied the power for the d-c impeller motors.

The ship stabilization equipment installed on the USS PEREGRINE (E-AM373) may be considered to be a combination of the following units: The ship, control system, pumping system, and tanks. The interconnection of these various units is indicated in the schematic diagram, Figure 5. As the ship tends to roll in a seaway, the accelerometers generate a signal which is transmitted through the control system. This system filters out undesirable high-frequency signals and provides the desired phase shift and amplification of the signal. The modified signal is then conducted to the pumping system. Here it is fed to the amplidyne amplifiers, port and starboard, which in turn control the amplidyne motor-generator systems, thus determining the direction and amplitude of the pitch of the impeller blades. Meanwhile the impellers are turning continuously and pumping water from side to side, the rate and direction depending on the pitch angle of the impellers with respect to time. The water is pumped from tank to tank in each of the two systems through crossover ducts. Since the impeller and the pitch mechanism for the forward system are on the starboard side and for the aft system on the port side in order to balance the installation weight on the ship, water is transferred from the starboard tanks to the port tanks in the two systems, so to speak, by the starboard impeller pushing water to port and the port impeller pulling water to port. In this manner water is transferred from side to side in proper phase to counteract the tendency of the ship to roll.

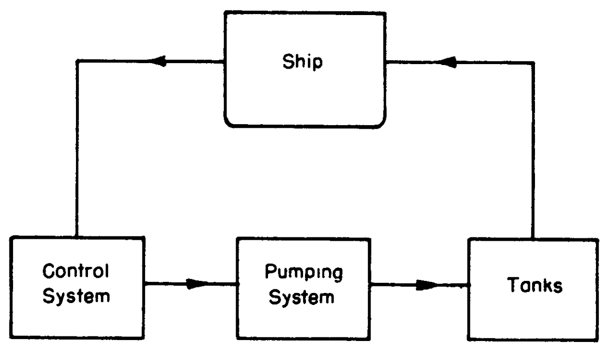

Figure 5 - Schematic Diagram of Main Units in the Activated-Tank Stabilizing System

USS PEREGRINE (E-AM373)

The USS PEREGRINE (E-AM373) was selected for the test of stabilization by means of activated tanks because her displacement, original period of roll, and original metacentric height were very close to that of the USS HAMILTON (DD141). Diagrams of the PEREGRINE showing the general features of the stabilization installation are shown in Figure 4, and photographs in Figure 6. The impellers used on the HAMILTON were used also in the PEREGRINE

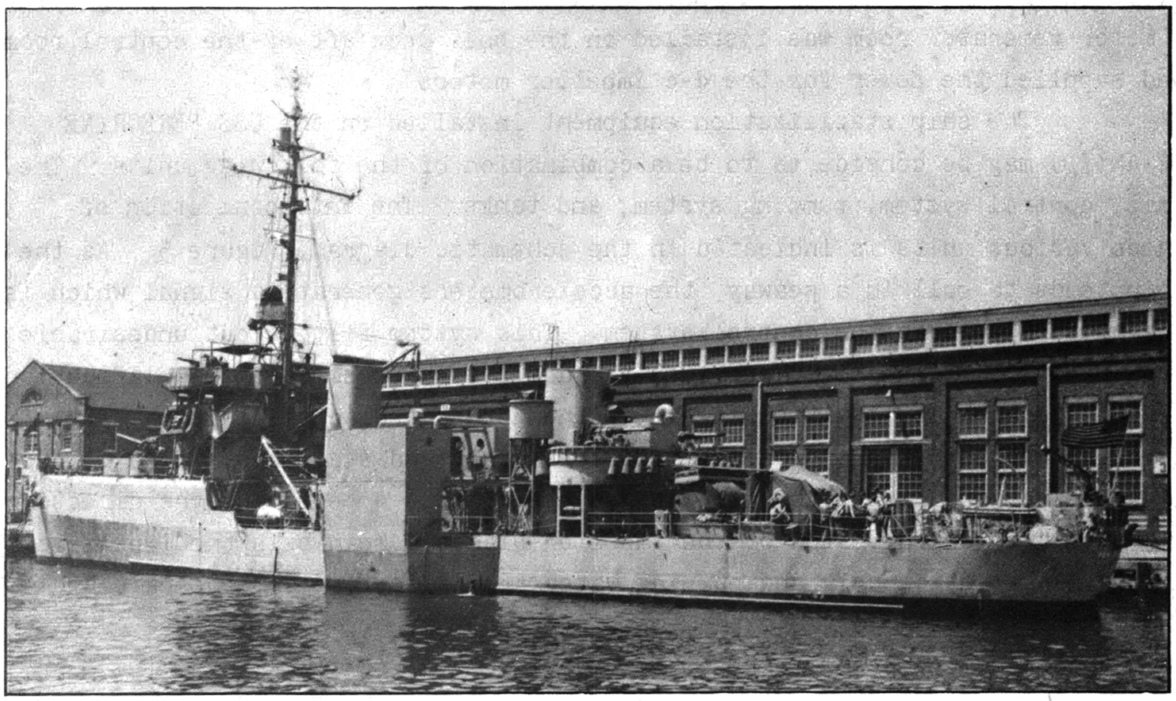

Figure 6a

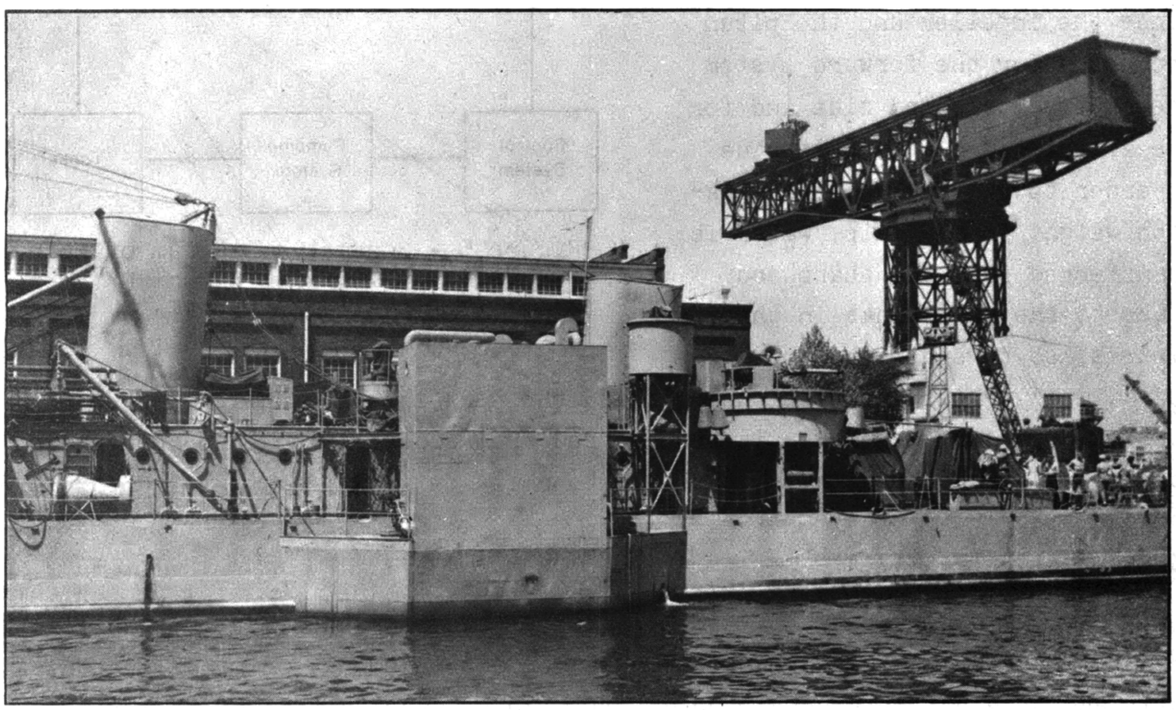

Figure 6b

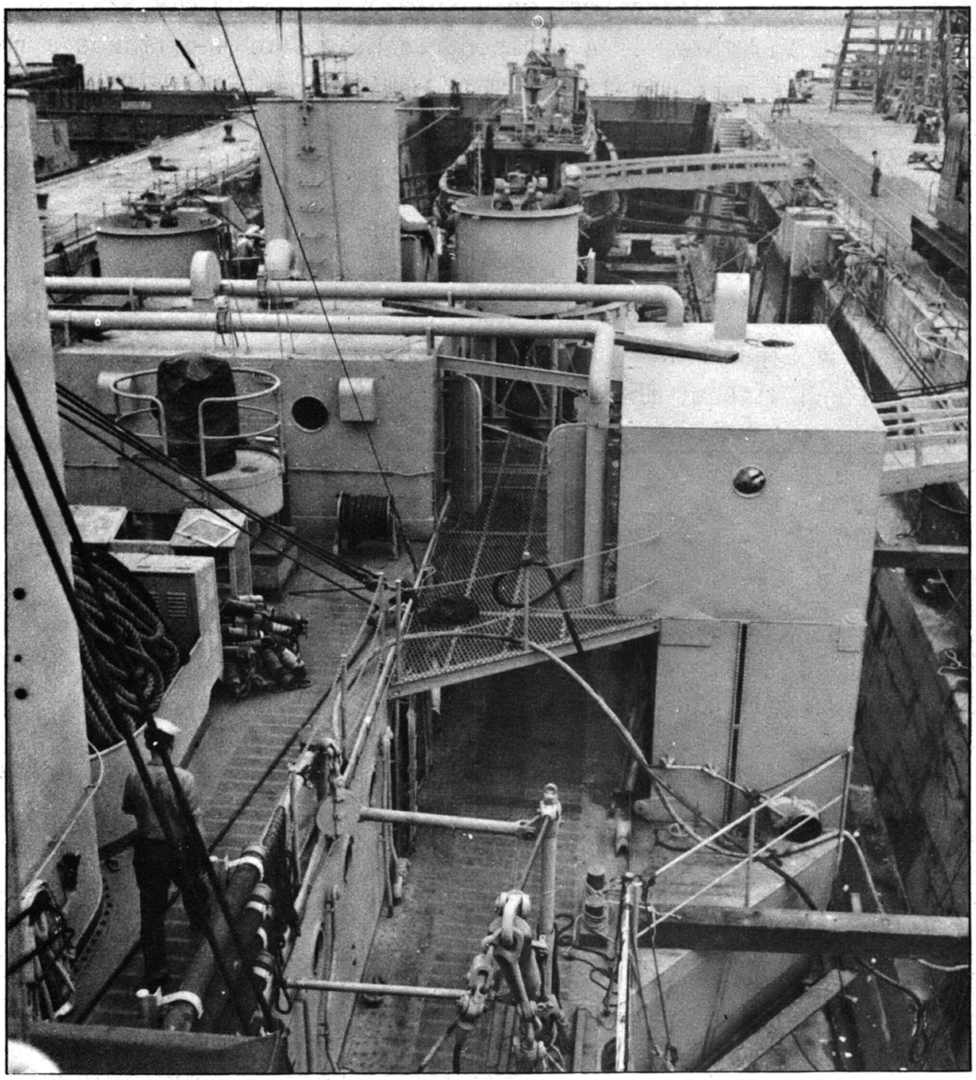

Figure 6c

Figure 6 - Photographs of the USS PEREGRINE (E-AM373) Showing General Features of the Stabilization Installation

installation but the remainder of the necessary equipment was completely redesigned and rebuilt, on the basis of Minorsky's experience with the HAMILTON installation. The PEREGRINE was intended to have a rolling period of 9 1/2 to 12 seconds and a metacentric height of 2 to 2 1/2 ft after installation of the stabilizing equipment.

An inclining test was performed on the PEREGRINE after the test installation was completed, and the metacentric height and the period of roll were found to be quite different from the design requirements. The results of the inclining test with the activated tank stabilizing equipment installed are compared with an inclining test on a ship of the same class without stabilizing equipment in Table 7. A metacentric height of 4.6 ft and a rolling period of 7.9 seconds were obtained. This rolling period was later confirmed by calm-water rolling tests. The increased metacentric height and decreased period were presumably caused by the sponsons that had been added outside of the hull.

TABLE 7

Inclining Experiments on the USS PEREGRINE (E-AM373) Class
With and Without the Activated-Tank Stabilizing System

	Before Installation Inclining Experiment Performed on USS POCHARD (AM375) November 1944			After Installation Inclining Experiment Performed on USS PEREGRINE (E-AM373) May 1949		
Length Over-all	221 ft 2 1/8 in.			220 ft 10 5/8 in.		
Length Between Perpendiculars	215 ft			215 ft		
Molded Breadth	32 ft at 7.9 ft above molded base line			32 ft at 7.3 ft above molded base line		
Molded Depth	17 ft 5/8 in			17 ft		
Extreme Breadth over Sponsons	-			46 ft 5 in		
Period of Roll, sec	8.6			7.9		
	Condition II	Light Service	Maximum Load	Condition A Light	Test 2 Half-Load Liquids	Test 2 Full-Load Liquids
Displacement, tons	797.2	1008.0	1209.0	909	1137	1267
Mean Draft, ft	7 ft 7 1/4 in.	9 ft 5/16 in.	10 ft 5 1/4 in.	8.30	9.75	10.56
CG. above Base Line, ft (No correction for free surface)	13.34	11.98	10.89	14.00	12.67	11.90
Transverse metacenter above Base Line, ft	-	-	-	18.60	17.37	16.96
GM, Metacentric Height (Correction for free surface)	2.76	2.63	3.42	4.60	4.55	4.86
Vertical Lever, ft	-	-	-	14.00	12.69	11.90
Vertical Moment, ton-ft	-	-	-	12,730	14,428	15,078
Moment to Heel 1°, ft-tons Displacement x GM x 0.01746	38.42	46.29	72.19	73	90	107

A list of the dimensions and constants of the USS PEREGRINE (E-AM373) before and after the installation of the stabilizing equipment and the proportions of the equipment are listed in Table 8, page 35, together with those for the USS HAMILTON (DD141)

THE CONTROL SYSTEM

Essentially the function of the control system is to transmit a signal proportional to the angular acceleration of the vessel to the mechanism controlling the pitch of the impellers in the pumping system and at such phase that the water in the tanks is transferred in proportion to angular velocity. It is required that the control system filter out ship vibrations while passing the acceleration signal of rolling within the useful frequency range of periods of 18 to 3 seconds.

The accelerometers, driver unit, mechanical filter, mixer unit, and their associated circuits are the elements of the control system. An accelerometer is used as the transducer because about 140° phase advance of the impeller pitch relative to the velocity of roll is necessary to have the water transferred in phase with the velocity. Therefore the control system is essentially one of velocity control with phase advance to correct for the lags developed in the system. The driver unit amplifies the accelerometer signal and the amplified signal is then fed to the mechanical filter, whose purpose is to eliminate the undesired vibration signals, with an error signal fed back to the driver unit to control the input to the filter. The output of the mechanical filter is supplied to the mixer unit which amplifies and advances phase by means of differentiating circuits, and its output is then supplied to the amplidyne circuit of the pumping system. A schematic diagram of the control system is shown in Figure 7 and photographs in Figure 8.

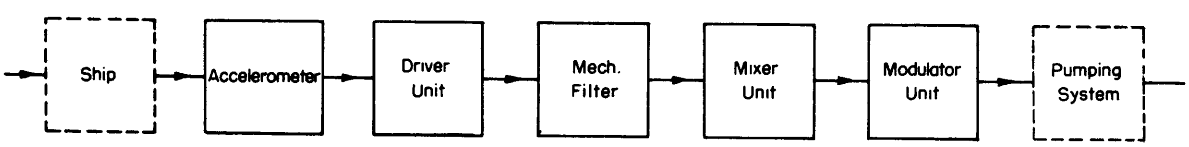

Figure 7 - Schematic Diagram of Control System for Ship Stabilization by Means of Activated Tanks

Accelerometers

Two Schaevitz transformer-type accelerometers were installed on the ship at Frame 37—forward-41 ft, 4 in. from the stabilizing tanks with a transverse distance of 23 ft 2 in. between center lines of the accelerometers—

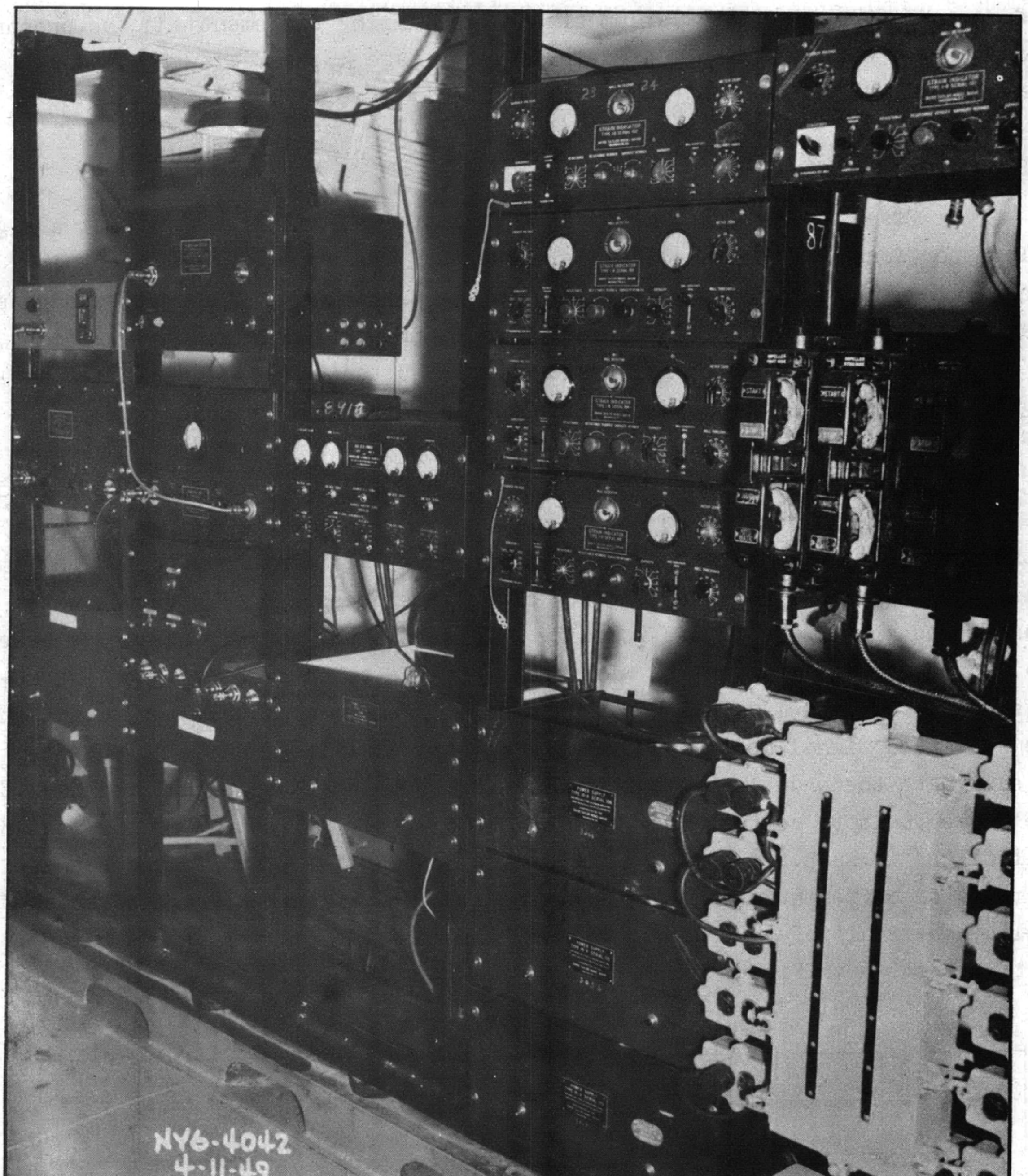

Figure 8a - General View

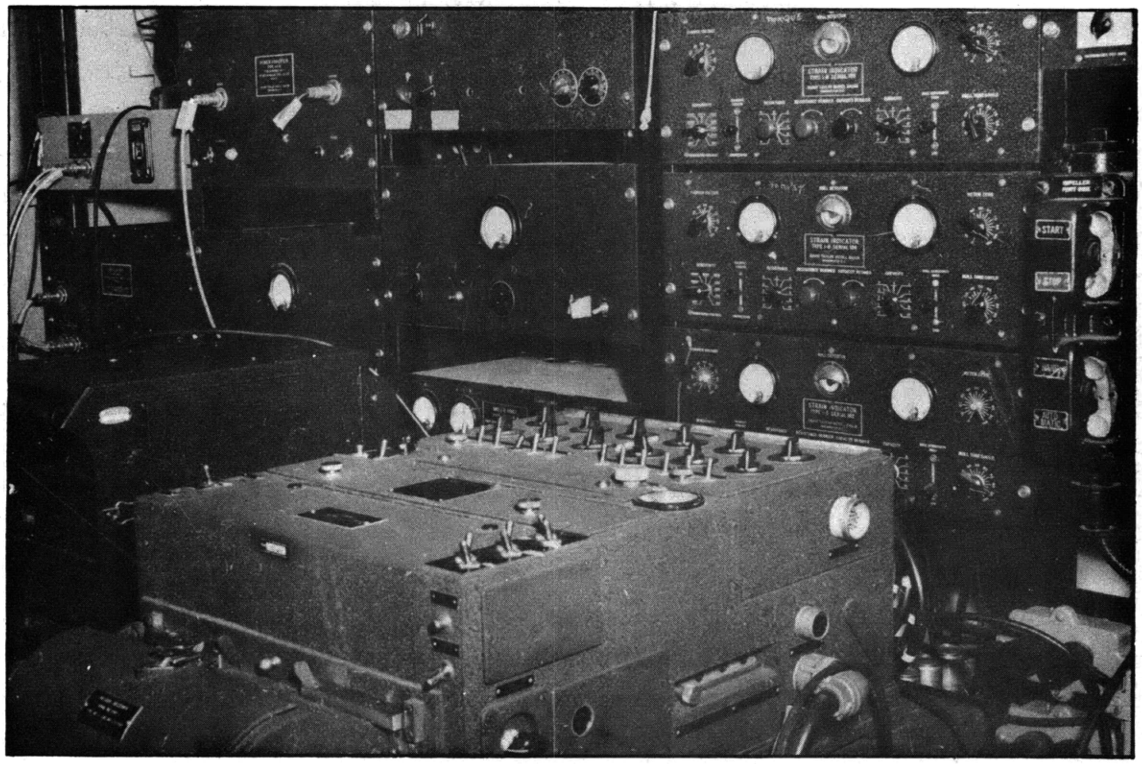

Figure 8b - Control Units

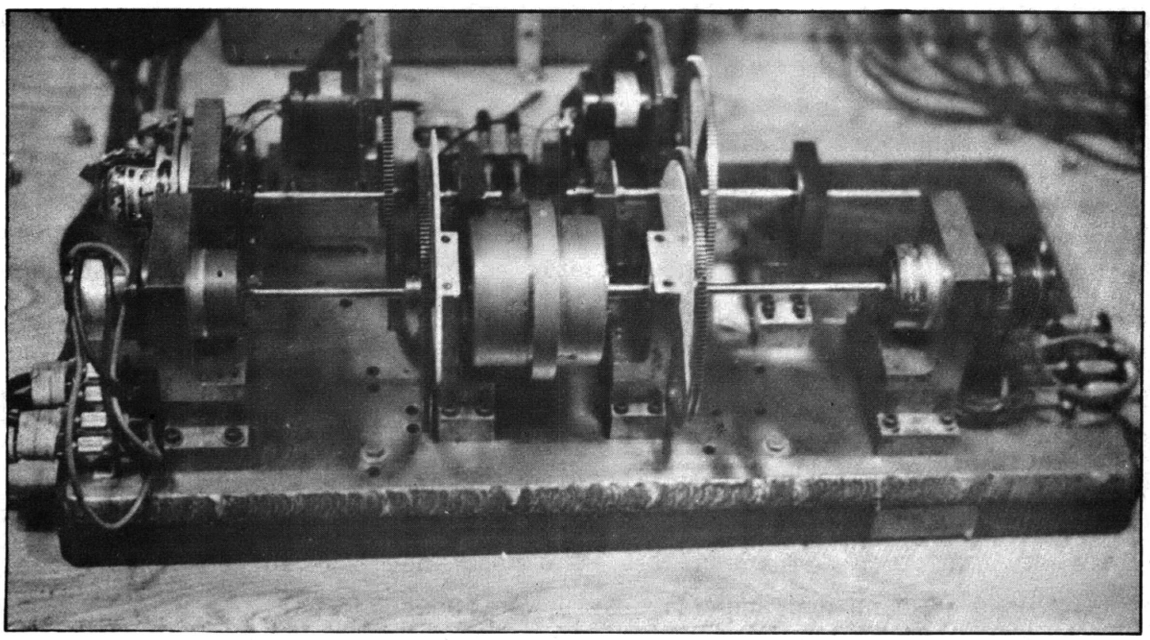

Figure 8c - Mechanical Filter

Figure 8 - Photographs of the Control System for Ship Stabilization on the USS PEREGRINE (E-AM373)

see Figures 9a and 9b. The accelerometers were mounted in felt-lined holders to minimize local vibrations and located at the intersection of longitudinals close to the port and starboard side plating and a heavy transverse girder, all on the under side of the main deck; see Figure 9c. Diagrams indicating the construction of the accelerometers are shown in Figures 9d and 9e. Since there were rather long cables—about 85 ft—between the accelerometers and the remainder of the control system in the control room, balance boxes were necessary to minimize the zero output of the accelerometers and were located close to them. The balancing was done by means of an R, L, and C variable circuit which is shown in Figure 9f. The accelerometers were connected so that the signals of the two gages were additive on rolling and cancelled on heaving or pitching. The remaining three motions did not affect the gages.

Since time was short in preparing for shipboard installation, no time was available for the development of entirely suitable accelerometers; consequently the market was canvassed for available or quickly deliverable units. The Schaevitz accelerometer seemed suitable, and delivery was fast. Schaevitz Type CA with the following nominal characteristics were selected: Nominal natural frequency 50 cps, damping 75 percent of critical, sensitivity with 28-volt (maximum) 400-cps power supply was 0.7 volt per g, with two gages wired additively. Spot calibrations were made of them in order to establish specifications for the remainder of the control system. Meanwhile complete calibrations as well as endurance runs were made on these accelerometers. The gages proved operable for long periods of time, and the results were reproducible.

The Schaevitz linear accelerometer is essentially a linear, variable-differential transformer. Three coils are arranged axially and connected so that the position of the magnetic core is indicated by a voltage on the output leads, see Figures 9d and 9e. The magnetic core is spring-suspended, and the entire unit is mounted in a case filled with organo-silicone damping oil. A flexible diaphragm is built into the case to allow for thermal expansion. A typical calibration curve obtained prior to ship installation is shown in Figure 9g. The upsweep at the low frequency end in this figure is most likely a characteristic of the method of calibration on the ultra-low-frequency calibrator. For the range of interest between 1/18 and 1/3 cps the accelerometer is apparently quite flat in response. The resonance peak for the three accelerometers used (two needed and one spare) was about 38 cps.

After operation was started in calm water and at sea, it was found necessary to filter the accelerometer signal ahead of the driver unit because of the rather high natural frequency of the accelerometers, because of high acceleration signals caused by vibration at the mounts, and to ease the burden

TABLE 8

Parameters of the USS PEREGRINE (E-AM373), of the USS HAMILTON (DD141), and of the Activated Tanks Installed on these Vessels

Parameter	USS PEREGRINE (E-AM373)		USS HAMILTON (DD141)	
	Before Installation of Activated Tanks	After Installation of Activated Tanks	Before Installation of Activated Tanks	After Installation of Activated Tanks
Length between Perpendiculars,	215 ft		310 ft	
Length, Over-all	220 ft, 10 5/8 in.			
Breadth - Extreme outside of plating	32 ft, 2 1/2 in.		31 ft	
Designers' Normal Load, W.L.	(At 1131 tons) 9 ft 4 in.			
Displacement as Inclined	1108 tons		1187 lg tons	
Molded Breadth with 7 ft 3 in. W.L.	22 ft			
Extreme Breadth over Sponsons		41 ft 5 in.		
Molded Depth	17 ft		20 ft 9 in.	
Metacentric Height (GM)	2.63 ft	4 ft 6 in. (Light condition draft 8.30 ft)	1.57 ft	1.57 ft
Rolling Period at Sea (Average)	8.6	7.9 sec. 7 sec.		10 sec.
Modulus of Stability	1140 x 2.63 = 2998 ton-ft	1140 x 4.60 = 5244 ton-ft		1187 x 1.57 = 1861 ton-ft
Roll Quenching Limit·Slope $= \frac{50 \times 57.3}{Wh} = \frac{210 \times 57.3}{1187 \times 1.57} = 6.45°$ HAMILTON $= \frac{482 \times 57.3}{1140 \times 4.6} = 5.26°$ PEREGRINE		5.26°		6.45°
Cross-section of Crossover Duct (Each)		10.92 sq ft		9 sq ft
Minimum Area at Upper End of Elbow		8.86		
Area at Crossover Duct Valve		9.42		
Area at Crossover Duct		10.92		
Length of Crossover Duct Along Centerline from Port Tank Top to Starboard Tank Top		64.57		
Length of Crossover Duct Along Centerline to Tank Bottoms		47.73		
Length, One Tank Length, Both Tanks		7.95 ft 15.9 ft		6.75 ft 13.5 ft
Width, One Tank		6.97 ft		7.3 ft
Height, One Tank		8.42 ft		8.5 ft
Horizontal Area, One Tank Horizontal Area, Two Tanks Horizontal Area, One Tank (Corr for stiff) 0.75 percent Horizontal Area, Two Tanks (Corr for stiff) 0.75 percent		55.62 sq ft 111.24 sq ft 55.20 sq ft 110.40 sq ft		49.25 sq ft 98.5 sq ft
Total Volume, One Tank Total Volume, Two Tanks		463.7 cu ft (Corr) 927.4 cu ft (Corr)		418.5 cu ft 837.0 cu ft
Maximum Fresh Water, One Tank Maximum Fresh Water, Two Tanks		12.9 lg tons 25.8 lg tons		11.65 lg tons 23.3 lg tons
Lever Arm from Longitudinal centerline of ship to Longitudinal centerline of One Tank		18.685 ft		9.0 ft
Maximum Static Moment, One Tank-System Maximum Static Moment, Two Tank-System		241 ton-ft 482 ton-ft		105 ton-ft 209.5 ton-ft
Static Moment per foot of Water Diff. One System Static Moment per foot of Water Diff. Two Systems		28.6 57.2		12.35 17.8
Total Volume, One System 2 Tanks and 1 Crossover Duct Total Volume, Two Systems 4 Tanks and 2 Crossover Ducts		1534 cu ft 3068 cu ft		
Total Volume of Water, One System Tanks Half Full Total Volume of Water, Two Systems Tanks Half Full		1070.5 cu ft 2141.0 cu ft		832.5 cu ft 1665 cu ft
Total Weight of Water, One System Tanks Half Full Total Weight of Water, Two Systems Tanks Half Full		29.75 lg tons 59.5 lg tons		23.2 lg tons 46.4 lg tons
Percent of Ship's Displacement of Fresh Water in System		3.91		5.22

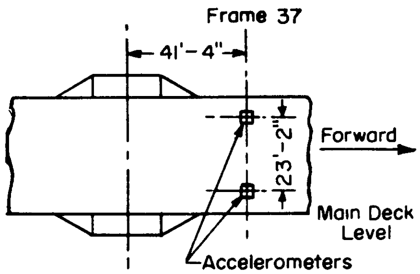

Figure 9a - General Location of Accelerometers

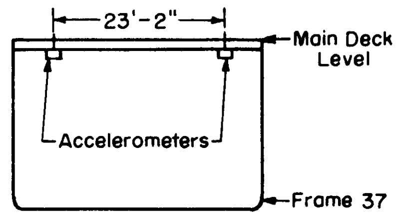

Figure 9b - Transverse Arrangement of Accelerometers

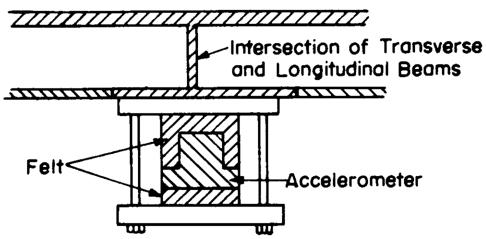

Figure 9c - Accelerometer Mounted in Felt-Lined Support on Overhead at Intersection of Longitudinal and Transverse Beams

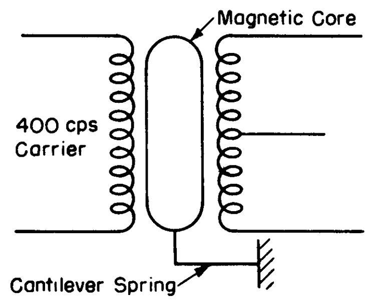

Figure 9d - Schematic Wiring Diagram of Accelerometer

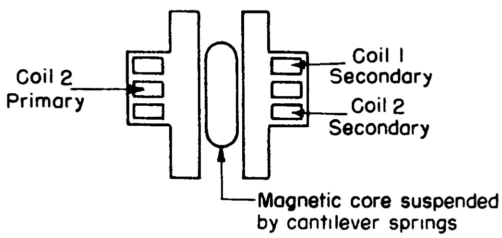

Figure 9e - Schematic Section of Accelerometer

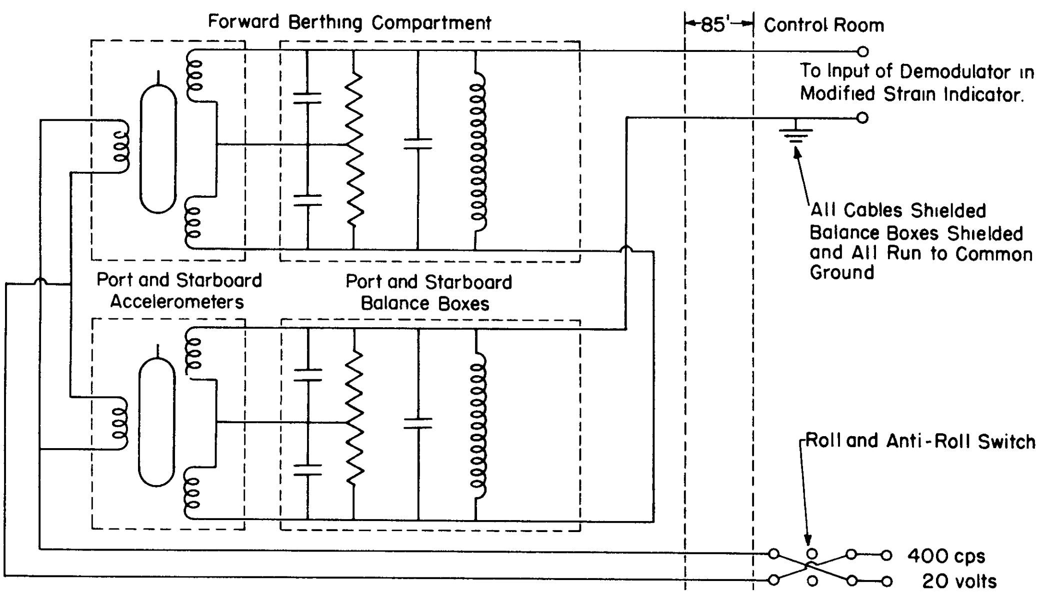

Figure 9f - Accelerometer and Balance-Box Circuits

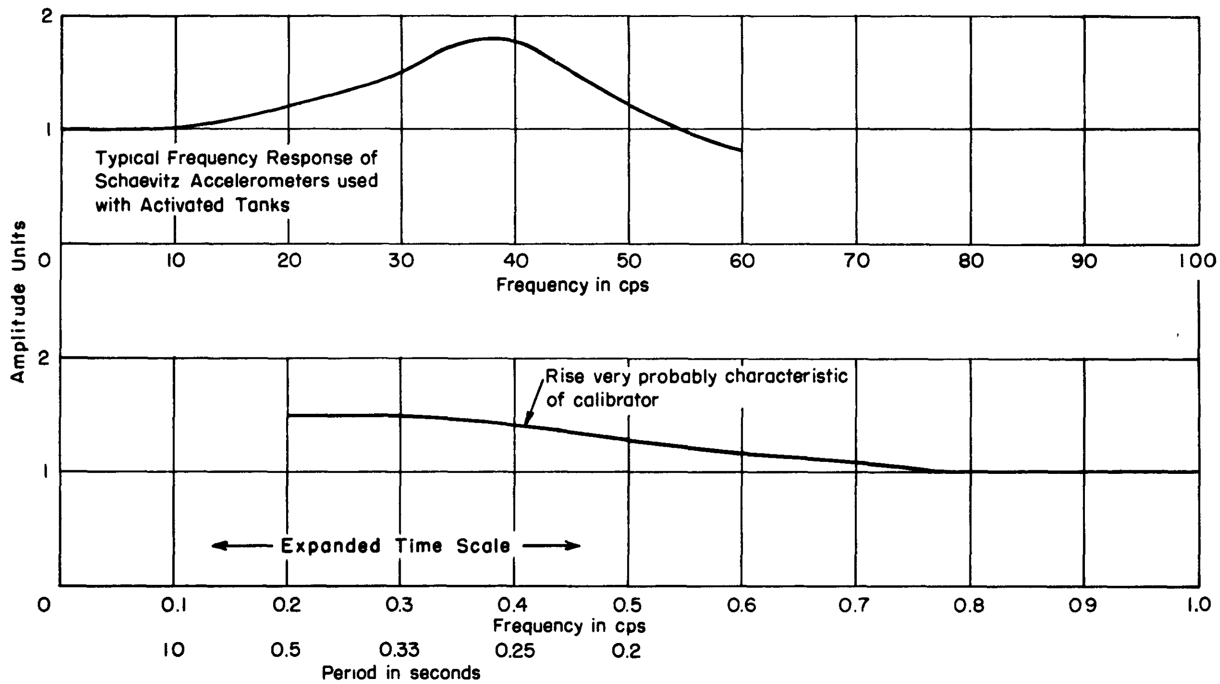

Figure 9g - Typical Frequency Response of Schaevitz Accelerometers Used with Activated Tanks

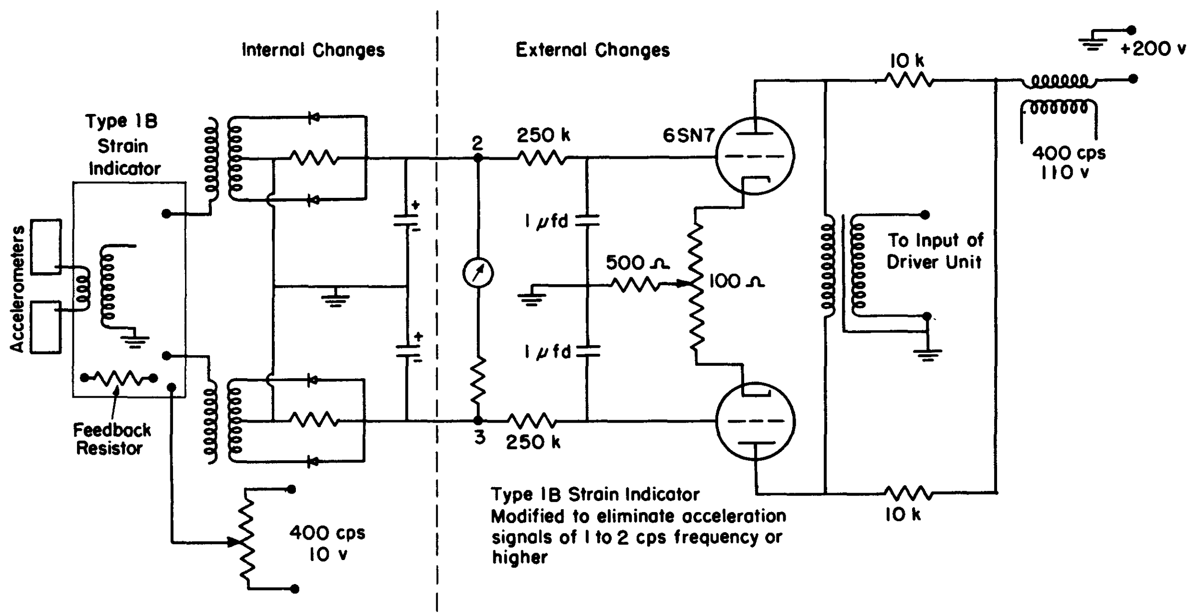

Figure 9h - Filter at Output of Accelerometers and Balance Boxes

Figure 9 - Transformer-Type Accelerometers

on the mechanical filter. A Taylor Model Basin strain indicator was modified for this filtering task. Essentially the filter demodulated the 400-cps carrier accelerometer signal and served as an R-C push-pull filter finally remodulating to 400 cps, Figure 9h.

Driver Unit

Upon determination of the characteristics of the accelerometers, specifications were prepared for the driver unit. This unit was envisioned to be used in two ways, one, directly connected to the mixer unit for calibration of the components of the control system, and two, connected through the mechanical filter to the mixer unit for stabilization.

The driver unit is an audio and power amplifier; the first half, the audio amplifier, is used alone when the driver unit is tied in directly to the mixer unit, and both halves are used when the mechanical filter is in the circuit. A power amplifier was necessary to operate the two-phase low-inertia drive motor of the mechanical filter. To this power amplifier was also fed an error signal from the input synchro of the mechanical filter to maintain the motion of the input end of the mechanical filter in proportion to the accelerometer signal. A schematic diagram showing the tie-in of the driver unit to the accelerometers and to the mechanical filter is shown in Figure 10. The accelerometer signal goes through the audio amplifier after which this

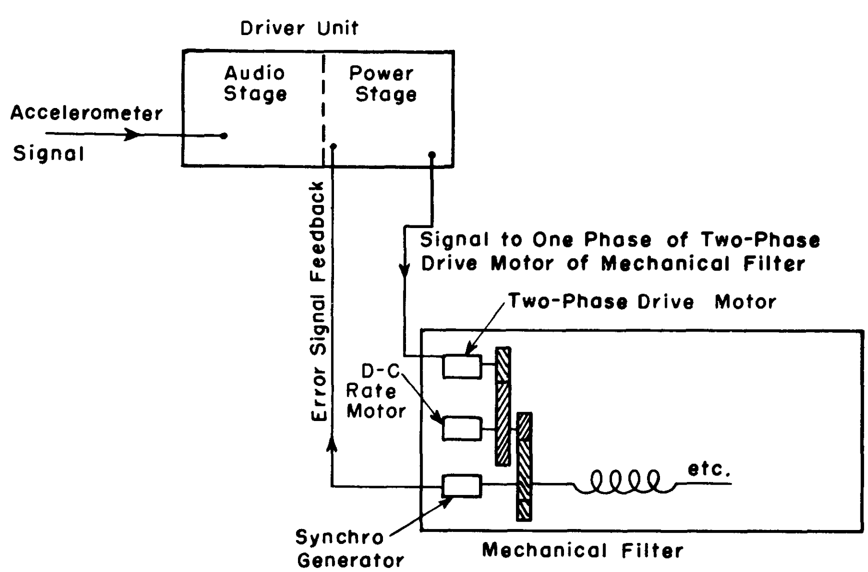

Figure 10 - Schematic Diagram Indicating the Tie-In of the Driver Unit to the Accelerometers and to the Mechanical Filter

signal β is algebraically added to the error signal β - 180° from the follow-up synchro on the input end of the mechanical filter and then goes through the power amplifier to one coil of the two-phase low-inertia drive motor of the mechanical filter at β - 90° phase and with 400 cps carrier. The other coil of the two-phase motor is supplied power at β phase, 400-cps carrier. The variation of voltage on the β - 90° phase creates torque, the amount and direction determining the motion of the input end of the mechanical filter. The 400-cps carrier for both coils originates from the same inverter. A complete wiring diagram together with a parts list of the unit are shown in Figure 11, and a photograph in Figure 12. A standard power supply unit was used in conjunction with the driver unit.

In order to allow plenty of leeway the driver-unit specifications were made as flexible as practicable while still permitting the design and construction of a reliable electronic unit. For test and calibration the unit was provided with cathode-ray oscillograph taps for trouble-shooting, string-oscillograph cathode-follower circuits for recording, and step-gain controls for the audio and power amplifiers. It was well that the requirements were extensive in range; the last few amplitude steps of the audio circuit were nonlinear and nonusable. The remaining steps were very satisfactorily linear. The specifications for the driver unit were:

Input from the Schaevitz accelerometers: The output range of the two push-pull Schaevitz accelerometers is 5 to 180 mv across 20,000 ohms. A 400-cps carrier signal with maximum voltage of 28 v is used as the input to the accelerometers.

Synchro Generator: The synchro generator for error-signal feedback to the driver unit shall have an output range of ±2 v, 400-cps carrier with an output impedance of 300 ohms. This signal is to be fed to the mixer transformer of the driver unit 180° out of phase with the accelerometer amplifier signal (audio stage).

Mixer Circuit: The mixer circuit of the driver unit shall algebraically add the audio-stage signal and the synchro-generator error signal 180° out of phase with the first such that, when their amplitudes are equal, no signal is delivered to the power stage. Otherwise the difference signal controls the input to the power stage.

Power Stage: The power stage shall amplify the difference signal and apply up to 115 v, 20 w, at the terminals of the control phase of the two-phase input motor of the mechanical filter. This amplifier shall have a gain up to 100 in convenient steps.

Two-Phase Motor: The motor shall receive up to 115 v, 20 w, from the driver unit. This voltage shall be 90° out of phase with the 400-cps 115-v power fed into the main phase of the motor. The main phase winding is fed with standard, constant amplitude 400-cps 115 v.

General: The driver unit shall operate at continuous rating. The unit shall be accessible for replacement, repair, and modification. Provide cathode-ray oscillograph and string oscillograph taps for checking and calibrating.

Parts List for Driver Unit

R-101	100 K	
R-102	4.7 K	1 w
R-103	20 K	
R-104	1.2 K	
R-105	50 K	
R-106	30 K	
R-107	10 K	
R-108	5 K	
R-109	3 K	
R-110	1 K	1 w values picked to 2 percent
R-111	500 ohms	
R-112	300 ohms	
R-113	100 ohms	
R-114	100 ohms	
R-115	4.7 K	
R-116	82 K	
R-117	4.5 K	
R-118	300 K	
R-119	1 M	
R-120	4.7 K	
R-121	1 K	1 w
R-122	1 K	
R-123	1 K	
R-124	1 K	
R-125	100 K	
R-126	15 K	
R-127	120 K	
R-128	500 K	
R-129	300 K	
R-130	100 K	
R-131	50 K	1 w values picked to 2 percent
R-132	30 K	
R-133	10 K	
R-134	5 K	
R-135	3 K	
R-136	1 K	
R-137	1 K	
R-138	4.7 K	
R-139	15 K	
R-140	1.5 K	
R-141	100 K	
R-142	100 K	
R-143	100 K	
R-144	1 M	
R-145	4.7 M	1 w
R-146	15 K	
R-147	1.5 K	
R-148	250 K	
R-149	800 ohms	
R-150	100 K	
R-151	100 K	
R-152	100 K	
R-153	1 M	
R-154	10 K	Potentiometer
R-155	25 K	
R-156	200 K	
R-157	300 ohms	
R-158	30 K	1 w
R-159	47 K	
R-160	500 K	
R-161	1 K	
R-162	500 K	Potentiometer
R-163	200 ohms	
R-164	15 K	1 w
R-165	200 ohms	
R-166	15 K	
R-167	50 ohms	10 w
R-168	Value determined during field installation	
C-101	10 µf	450 v Electrolytic
C-102	0.1 µf	Paper
C-103	10 µf	450 v Electrolytic
C-104	0.1 µf	Paper

C-105	10 µf	
C-106	10 µf	450 v Electrolytic
C-107	10 µf	
C-108	Phasing Condenser	
C-109	0.01 µf	Paper
C-110	10 µf	450 v Electrolytic
C-111	10 µf	
C-112	0.06 µf	
C-113	0.1 µf	
C-114	0.1 µf	
C-115	0.002 µf	Paper
C-116	0.002 µf	
C-117	0.002 µf	
C-118	10 µf	450 v Electrolytic
C-119	10 µf	
C-120	0.1 µf	Paper
C-121	10 µf	450 v Electrolytic
C-122	0.01 µf	
C-123	0.01 µf	
C-124	0.01 µf	
C-125	0.005 µf	Paper
C-126	0.03 µf	
C-127	0.01 µf	
C-128	0.21 µf	
C-129	0.5 µf	
C-130	10 µf	450 v Electrolytic
C-131	10 µf	
J-101		
J-102		
J-103		
J-104		
J-105	Chassis Receptacle, Cannon XK-3-14	
J-106		
J-107		
J-108		
J-109		
J-110		
J-111	Chassis Receptacle, Cannon X-3-14	
J-112		
J-113	Chassis Receptacle, Male, Jones P-306-RP	
L-101	Inductor, W.E. Toroid Type X65438, 6.7 H.	
L-102	Inductor, W.E. Toroid Type X65438, 2.5 H.	
S-101	Toggle Switch, D.P.D.T.	
S-102	Rotary Switch, Centralab K-121 with 1 "B" Section	
S-103	Toggle Switch, D.P.D.T.	
S-104	Rotary Switch, Centralab K-121 with 1 "B" Section	
T-101		
T-102	Special Transformer, Audio Development Company Number A5311A	
T-103		
T-104		
T-105	Audio Transformer, Thordarson T-19D03	
T-106	Output Transformer, Thordarson T-15S93	
V-101	6J5	
V-102	6SJ7	
V-103	6J5	
V-104	6SJ7	
V-105	6SN7	
V-106	6N7	
V-107	6SK7	
V-108	6J5	
V-109	6SJ7	
V-110	6SJ7	
V-111	6L6	
V-112	6L6	
BT-101	22 1/2 v Battery	

K corresponds to a multiplying factor of 10^3.

M corresponds to a multiplying factor of 10^6.

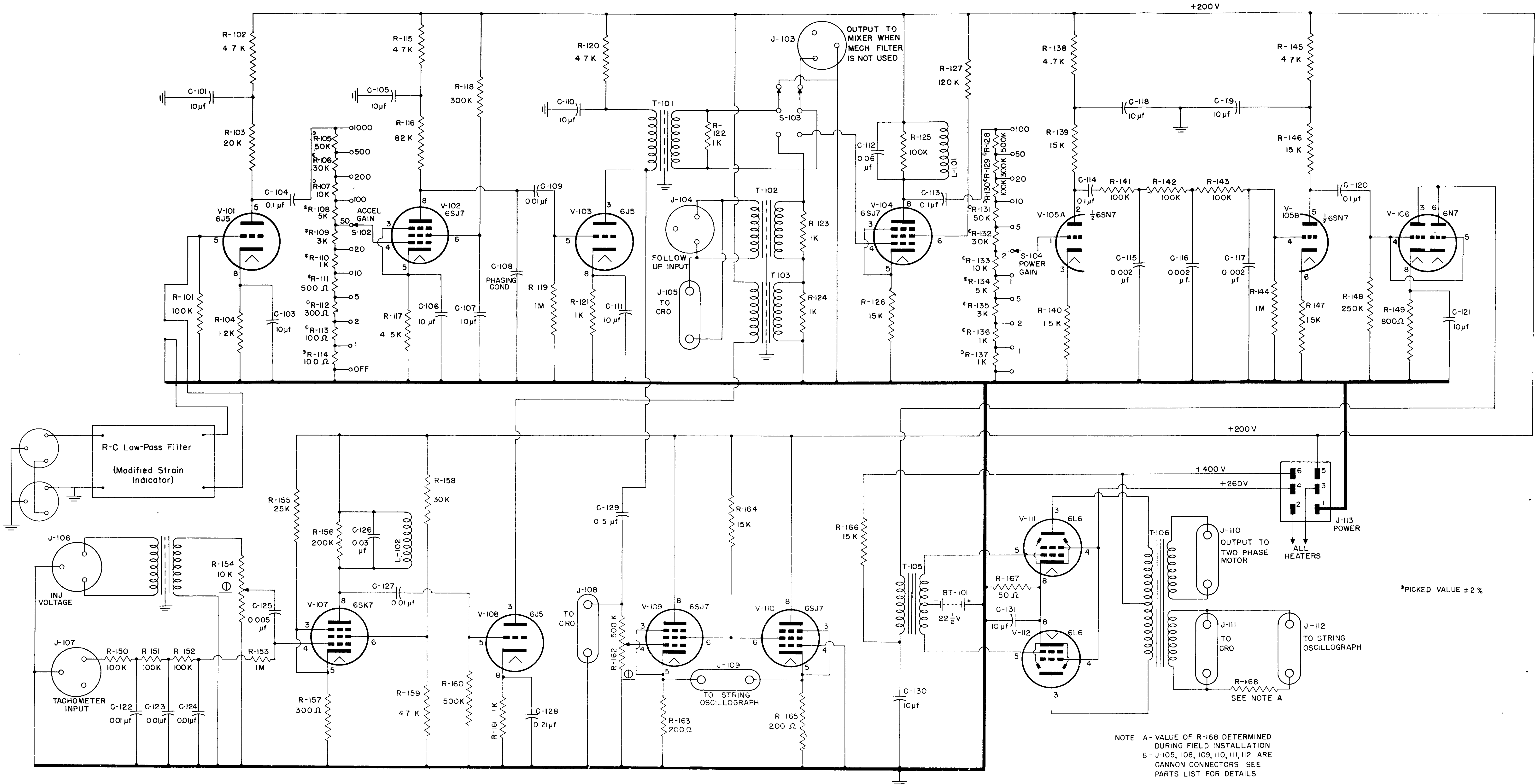

Figure 11 - Wiring Diagram and Parts List of the Driver Unit

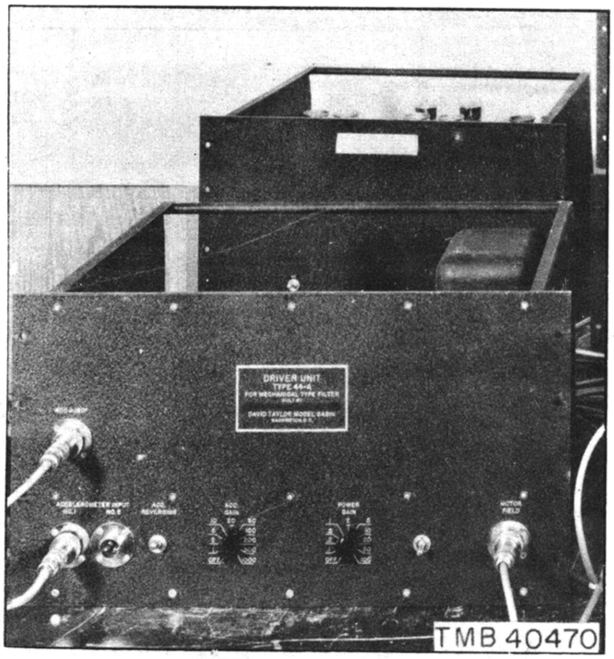

Figure 12 - Photograph of the Driver Unit

Mechanical Filter

The purpose of the mechanical filter in the ship-stabilization control system was to pass the useful frequencies of signal from the accelerometers with little or no phase shift and to attenuate greatly the higher frequencies. The desired frequencies are those in the expected range of the rolling "frequencies" of ships. A range of periods from 18 to 3 seconds more than covers those encountered by all classes of naval vessels. Actually there is no difficulty in operating at periods a little longer than 18 seconds, but eventually friction prohibits obtaining good results at these long periods.

The mechanical filter has an input two-phase motor geared through an intermediate set of gears to a large gear driving the input end of the torsional spring shaft; see Figure 13. On the input end of the spring shaft is a synchro generator which transmits an error signal to the driver unit; the latter has been described in the previous section. A housed two-directional torsional spring connects the input and output portions of the spring shaft. The spring dimensions and the amount of rotational mass of the housing together with the mass of the remainder of the mechanical filter at the output end (two-phase motor rotor and synchro motors, shafts, and gears) resulted in a natural frequency of about 1/3 cps. On the output end of the spring shaft is a synchro generator where output signal (the output signal of the entire mechanical filter) is fed to the mixer unit. Geared to the output end of the

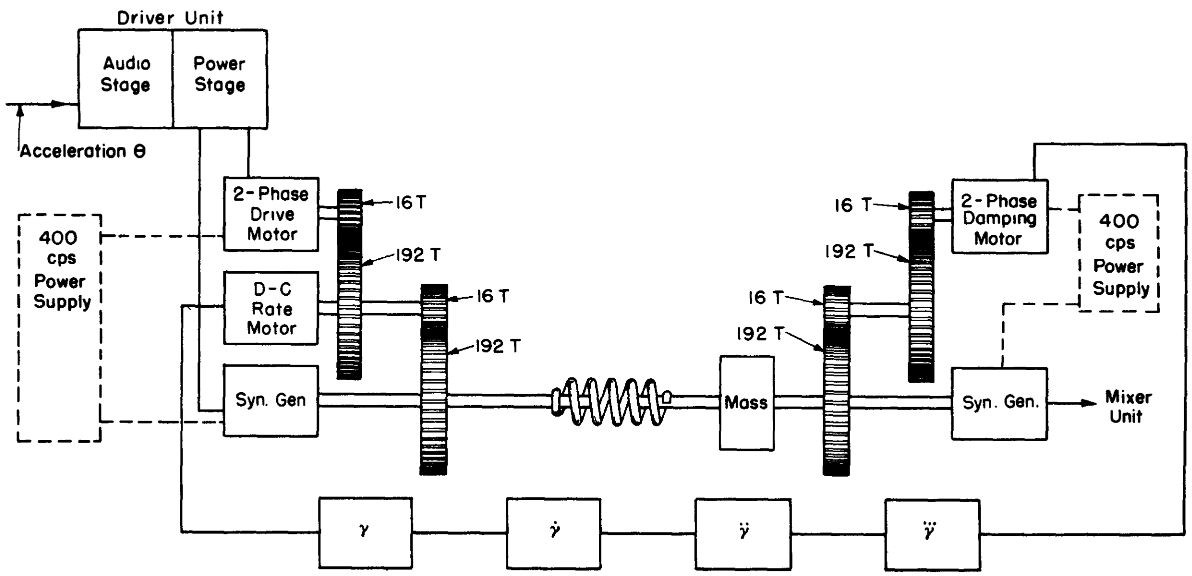

Figure 13 - Arrangement of the Mechanical Filter

spring shaft through intermediate gearing is a two-phase low-inertia motor used as a damping motor. It receives its signal from a d-c rate motor on the input end of the mechanical filter through a damping unit which differentiates the rate signal three times.

For this damping unit a number of requirements were specified:

> A displacement signal is obtained from a d-c rate motor at the input end of the filter and is differentiated three times; considering input displacement α, we get $\dot{\alpha}$, $\ddot{\alpha}$, and finally $\dddot{\alpha}$ as an output to the two-phase low-inertia damping motor. The $\dddot{\alpha}$ is amplified by a power amplifier to give a resultant output signal of 5 to 50 v and 20 w at 300-ohm resistance. The rectification, differentiation, and audio amplification circuits were in one unit, while the power amplification was in another unit.

A wiring diagram of the driver unit has been shown in Figure 11, and a wiring diagram together with a list of parts of the other electronic unit associated with the mechanical filter are shown in Figure 14. In addition to the 110-v d-c plate supply, the same power supply unit serving the driver unit was used for the damping unit. Typical curves showing the performance of the mechanical filter with respect to frequency and with a sinusoidal input are shown in Figure 15. As can be seen from the curves, Figure 15a, where amplitude for a constant input amplitude and phase difference between the input and output are plotted against period as the damping is increased, the resonant peak amplitude is decreased with a slight shift of the peak to lower frequency—contrary to what would be expected with viscous damping. However, as the resonant peak is diminished the phase difference starts to increase at lower frequency. In Figure 15b for different inputs and the same damping the the phase shifts at lower periods for smaller inputs. The mechanical filter

Parts List for Damping Unit

R-201	10 K	
R-202	1 K	
R-203	1 K	1 w
R-204	1 K	
R-205	1 K	
R-206	800 ohms	
R-207	100 ohms	Potentiometer
R-208	200 K	
R-209	200 K	
R-210	1.5 M	
R-211	1.5 K	
R-212	50 K	
R-213	30 K	
R-214	20 K	
R-215	35 K	
R-216	100 K	
R-217	100 K	
R-218	100 K	
R-219	100 K	
R-220	1.5 K	1 w
R-221	50 K	
R-222	30 K	
R-223	20 K	
R-224	35 K	
R-225	100 K	
R-226	100 K	
R-227	100 K	
R-228	100 K	
R-229	1.5 K	
R-230	50 K	
R-231	30 K	
R-232	20 K	
R-233	35 K	
R-234	100 K	Potentiometer
R-235	56 K	1 w
R-236	35 ohms	
R-237		
R-238		
R-239	35 ohms	
R-240	56 K	
R-241	100 K	
R-242	300 K	
R-243	39 K	1 w
R-244	100 K	
R-245	50 K	
R-246	50 K	
R-247	500 K	
R-248	10 K	Potentiometer
R-249	500 K	
R-250	5 K	
R-251	5 K	
R-252	300 K	
R-253	400 K	1 w
R-254	750 ohms	
R-255	680 ohms	
R-256	750 ohms	
R-257	68 K	
R-258	68 K	1 w
R-259	300 K	
R-260	400 K	

C-201	0.02 µf	Paper
C-202	10 µf	450 v Electrolytic
C-203	0.06 µf	
C-204	0.1 µf	
C-205	0.1 µf	
C-206	0.1 µf	
C-207	0.05 µf	
C-208	0.06 µf	
C-209	1 µf	
C-210	0.01 µf	Paper
C-211	0.01 µf	
C-212	0.01 µf	
C-213	1 µf	
C-214	0.01 µf	
C-215	0.01 µf	
C-216	0.01 µf	
C-217	1 µf	
C-218	20 µf	Electrolytic

L-201	
L-202	Choke, U.T.C. VI-C19
L-203	
L-204	Primary of 10-amp filament transformer

V-201	6SN7
V-202	6H6
V-203	6H6
V-204	12SJ7
V-205	12J5
V-206	12SJ7
V-207	12J5
V-208	12SJ7
V-209	12J5
V-210	6AS7
V-211	6AS7
V-212	6SJ7
V-213	6J5
V-214	6SJ7
V-215	6SN7

J-201	
J-202	
J-203	Chassis Receptacle, Cannon XK-3-14
J-204	
J-205	
J-206	Chassis Receptacle, Jones P-306-RP
J-207	General Radio Jack

T-201	
T-202	Audio Development Company Number A-5311-A
T-203	
T-204	
M-201	Meter, Galvanometer 60-0-60 Weston

K corresponds to a multiplying factor of 10^3.
M corresponds to a multiplying factor of 10^6.

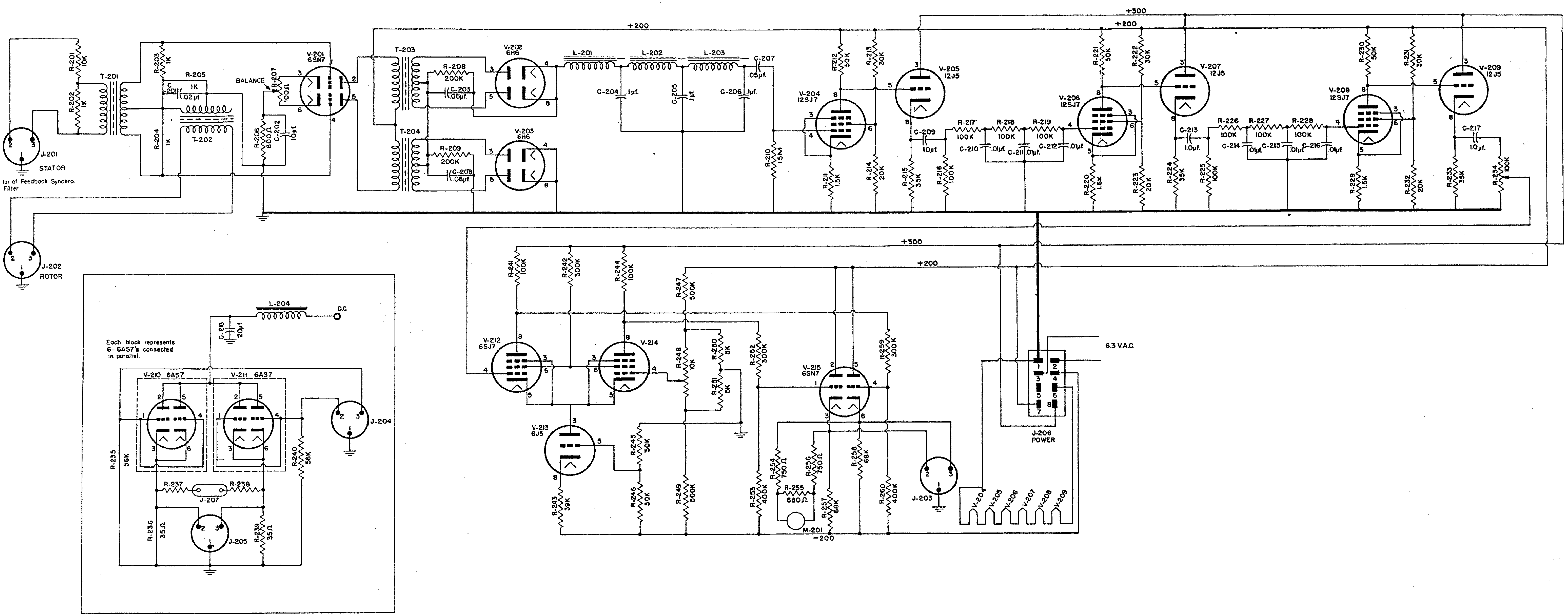

Figure 14 - Wiring Diagram and Parts List of the Damping Unit

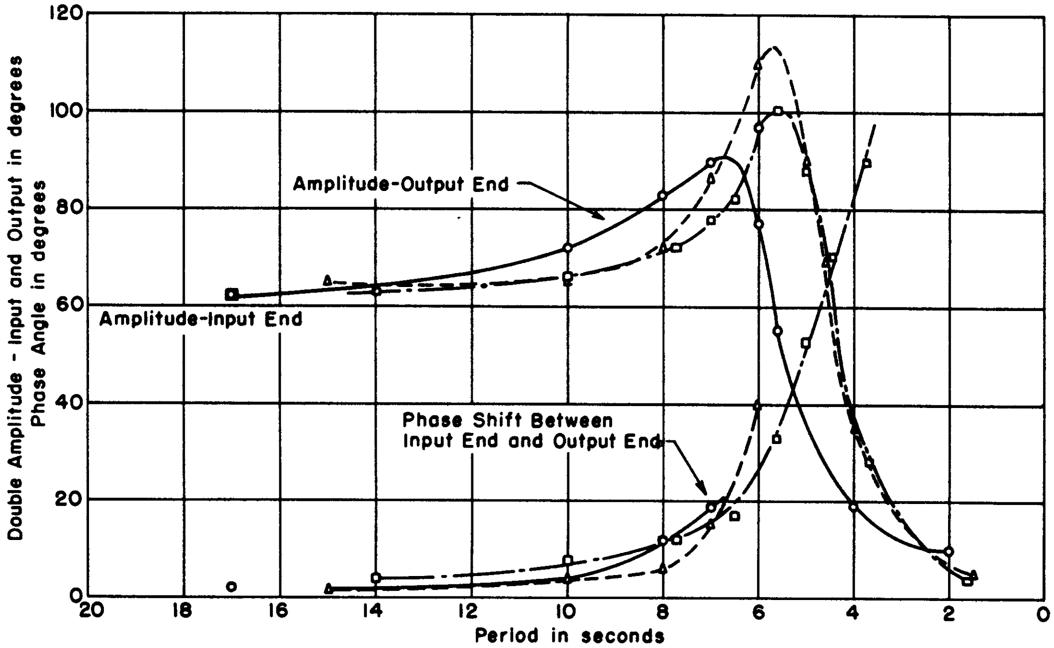

Figure 15a - Amplitudes and Phase Shifts for Constant Input and Different Amounts of Damping Settings

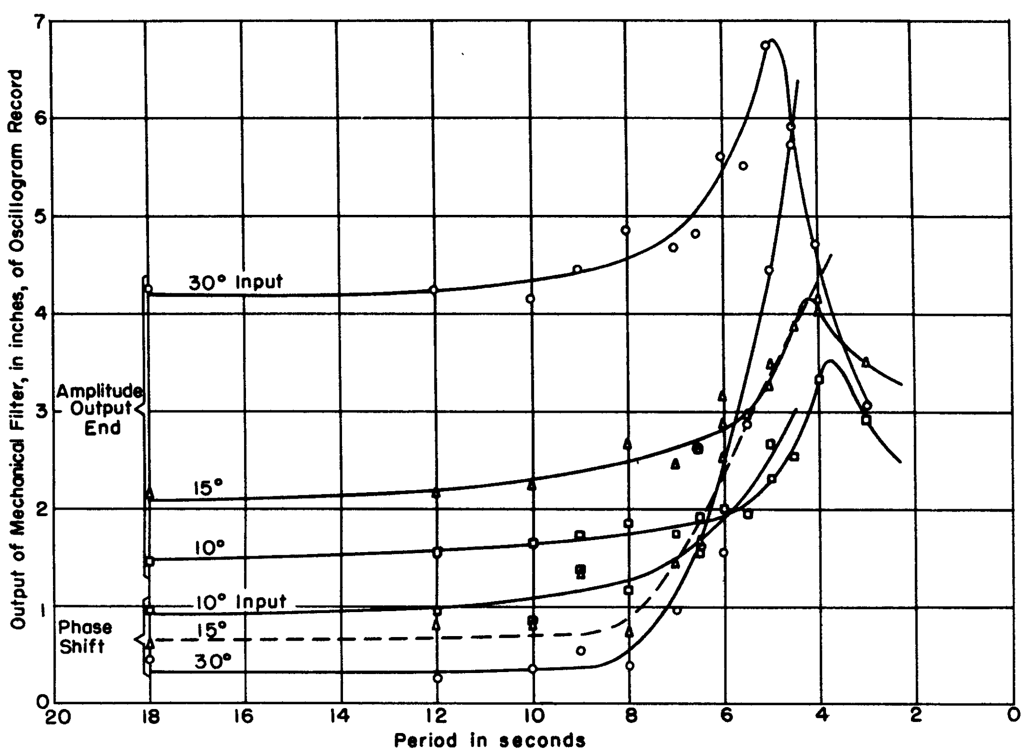

Figure 15b - Amplitudes and Phase Shifts for a Number of Inputs and the Same Damping Setting

Figure 15 - Typical Calibration Curves of the Mechanical Filter with Various Amounts of $\dddot{y}$ Damping Imposed on the Output End

For constant amplitudes at the input end the output amplitudes and the phase shift between the input and output motion are plotted against period.

was designed to give a great diminution in amplitude at about 3 cps with rather small phase difference until the output amplitude was considerably reduced. Photographs of the mechanical filter and of the damping units are shown in Figure 16.

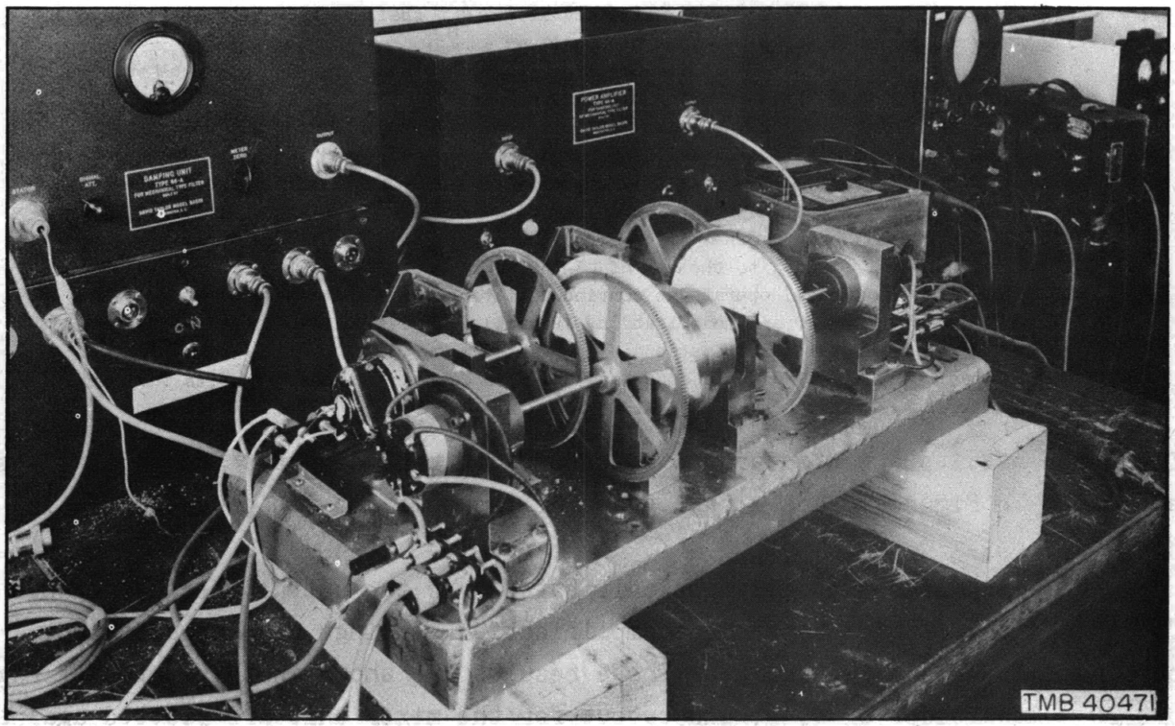

Figure 16 - Photograph of the Mechanical Filter and the Damping Units

Mixer Unit

The accelerometer signal, $\ddot{\theta}$, proportional to roll displacement for a particular frequency, after being filtered to allow through only those signals in the useful frequency range, is transmitted to the mixer unit. The mixer unit served to "anticipate" the roll of the ship in that the signal is differentiated, and the result $\dddot{\theta}$ gives a maximum amplitude when the acceleration signal $\ddot{\theta}$ is zero. The acceleration signal $\ddot{\theta}$ already gives a 90° "lead" with respect to the velocity of roll, $\dot{\theta}$. All this phase "lead" is necessary because there is a little phase lag in the mechanical filter as well as over 90° phase lag in the pumping of water in the tanks.

The mixer unit performs a number of operations, shown schematically in Figure 17. The various circuits are an input amplifier, a demodulator, two differentiating circuits, a ($\ddot{\theta}$, $\dddot{\theta}$) mixer, a ($\ddot{\theta}$, $\dddot{\theta}$, $\overset{iv}{\theta}$) mixer, and an output amplifier. The specifications perhaps will indicate the functions:

A signal is injected to the input amplifier from the synchro generator at the output end of the mechanical filter or from the first half of the driver unit. The synchro output is up to ±2 v with a threshold sensitivity of ±0.001 v at 400 cps with an impedance of 300 ohms. The input amplifier shall have amplification up to 100 times with step variation. The output of this stage is $\ddot{\theta}_M$.

The detector shall eliminate the 400-cps carrier, the signal becomes $\ddot{\theta}_R$, and the detector shall amplify if necessary prior to injection into the first differentiating stage.

The output of the first differentiating stage, $\dddot{\theta}_R$, shall be such that the voltage amplitude shall be of the same order of magnitude as $\ddot{\theta}_R$.

The output of the second differentiating stage, $\overset{IV}{\theta}_R$, shall be such that the voltage amplitude shall be of the same order of magnitude as $\dddot{\theta}_R$.

The mixer unit shall have a mixer circuit controlling simultaneously and adding $\ddot{\theta}_R$ and $\dddot{\theta}_R$ so that when $\ddot{\theta}_R$ is increased sinusoidally, $\dddot{\theta}_R$ is decreased sinusoidally—in other words added vectorially. The result is to be indicated by the control knob in terms of degrees phase shift from $\ddot{\theta}_R$.

The unit shall have another mixer circuit where varying amounts of $\overset{IV}{\theta}_R$ up to 10 percent of ($\ddot{\theta}_R$, $\dddot{\theta}_R$) are added to the sum of ($\ddot{\theta}_R$, $\dddot{\theta}_R$).

The mixer unit shall operate at continuous rating. The unit shall be accessible for replacement, repair, and modification. Possible radical modification is expected at the input end and at the output end.

Provide cathode-ray oscilloscope taps and also cathode-follower circuits for connection to string galvanometers at $\ddot{\theta}_R$, $\dddot{\theta}_R$, and $\overset{IV}{\theta}_R$.

The mixer has its own standard power supply. Wiring diagrams and parts lists are shown in Figure 18 and a photograph of the unit and its power supply in Figure 19.

The mixer unit had been calibrated over the useful frequency range for ship stabilization at various amplification settings and for different combinations of $\ddot{\theta}_R$, $\dddot{\theta}_R$, and $\overset{IV}{\theta}_R$. These results are shown in the curves of Figure 20. Amplitudes of $\ddot{\theta}$, $\dddot{\theta}$, and output for the mixer unit are plotted against period for different settings of the $\ddot{\theta} + \dddot{\theta}$ attenuator, Figure 20a. As the

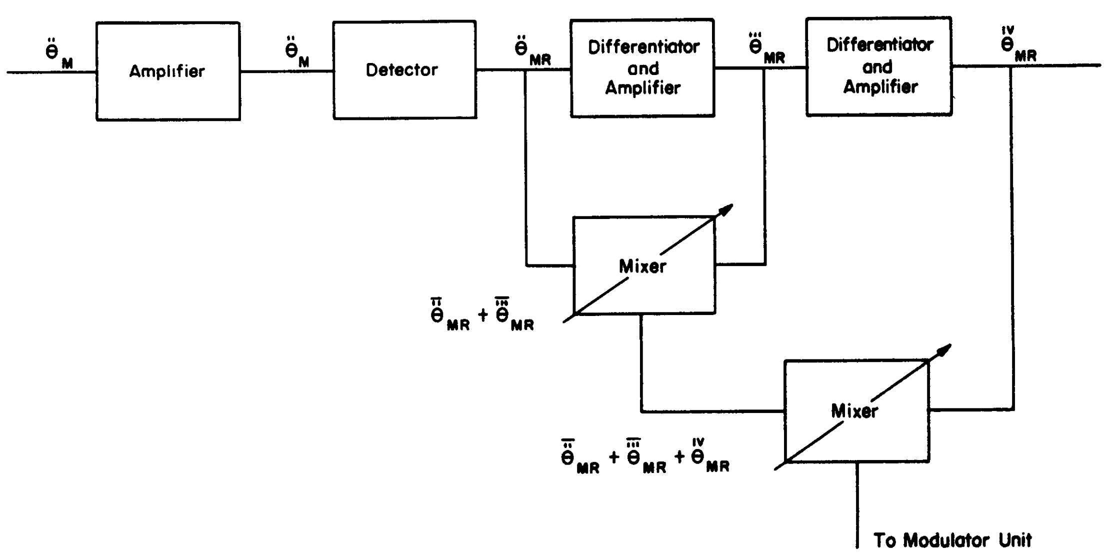

Figure 17 - Schematic Diagram of the Function of the Mixer Unit

$\ddot{\theta} + \dddot{\theta}$ setting is increased starting at 0, the output is wholly $\ddot{\theta}$ and at 90° wholly $\dddot{\theta}$, with $\ddot{\theta}$ sinusoidally decreasing and $\dddot{\theta}$ sinusoidally increasing with the output amplitude substantially constant for the range of settings. The $\ddot{\theta}$ curves can be compared in amplitude with each other only, and not with $\dddot{\theta}$ and output curves, since $\ddot{\theta}$, $\dddot{\theta}$ and output each has its separate cathode-follower circuit for recording on a string oscillograph. The $\ddot{\theta}$ curves are flat for the frequency range in which the calibration was made. In the range of periods of 5 to 2 seconds, however, an increase in amplitude occurs in $\dddot{\theta}$, and this becomes greater with the increase of the $\ddot{\theta} + \dddot{\theta}$ setting. The output of course follows $\dddot{\theta}$ closer and closer in this range as the $\ddot{\theta} + \dddot{\theta}$ setting increases since the output signal becomes almost wholly $\dddot{\theta}$. It is to be expected that as the frequency increases the amplitude would increase since the circuit differentiates. However, the differentiating circuits were designed to give constant amplitude for the useful frequency range. The phase difference between the input and the output of the mixer unit is plotted against period in Figure 20b for different $\ddot{\theta} + \dddot{\theta}$ settings. The phase difference results in leading phase for the output of the mixer unit. For a particular $\ddot{\theta} + \dddot{\theta}$ setting the phase at first increases gradually and then increases more rapidly, reaching a peak at the high end of the frequency range. With an increase of the $\ddot{\theta} + \dddot{\theta}$ setting the gradual increase of the phase difference becomes steeper and the peak difference occurs at lower frequencies.

Modulator Unit

Error in operating the pumps through the previously described control system can cause the water in the stabilizing tanks to pile up one way or the other and either swing about the continually changing zero position or be piled up so high that full transfer of water consonant with the signal received cannot be achieved. The purpose of the modulator unit is to tend to keep the zero position of the water at the mid-height of the tanks in both systems.

The modulator unit, Figure 21, receives the signal from the mixer unit, modulates it with a 60 cps carrier, then, independently for the two tank systems, adds the signal from their respective float-level synchros. The voltage to both float-level synchros is adjusted by a common variac while the signal from each synchro can be controlled by its own variac. This permits control of the amount of float-level signal to be added to the mixer output signal. Thus, for a particular condition, the minimum amount of float-level signal necessary to keep the water in the two systems at about mid-height can be added to the mixer output signal independently for each of the two systems. Finally, the last operation is the demodulation of the 60-cps carrier. The

Parts List for the Mixer Unit

R-301	5 K	
R-302	3 K	2 percent
R-303	1 K	
R-304	500 ohms	
R-305	300 ohms	2 percent
R-306	100 ohms	
R-307	50 ohms	
R-308	30 ohms	
R-309	10 ohms	
R-310	10 ohms	
R-311	20 K	
R-312	1.5 K	
R-313	1 K	
R-314	500 K	1 w
R-315	1 K	
R-316	1 K	
R-317	1 K	
R-318	100 ohms	Potentiometer
R-319	800 ohms	
R-320	200 K	1 w
R-321	200 K	
R-322	200 K	Potentiometer
R-323	1520 ohms	
R-324	4510 ohms	
R-325	7370 ohms	
R-326	10 K	
R-327	12,320 ohms	2 percent
R-328	14,280 ohms	
R-329	15,800 ohms	
R-330	16,840 ohms	
R-331	17,360 ohms	
R-332	20 K	1 w
R-333	800 ohms	
R-334	10 K	Potentiometer
R-335	50 K	
R-336	60 K	1 w
R-337	100 K	
R-338	68 K	
R-339	1520 ohms	
R-340	4510 ohms	
R-341	7370 ohms	
R-342	10 K	
R-343	12,320 ohms	2 percent
R-344	14,280 ohms	
R-345	15,800 ohms	
R-346	16,840 ohms	
R-347	17,360 ohms	
R-348	200 K	
R-349	10 K	Potentiometer
R-350	100 K	
R-351	20 K	
R-352	20 K	
R-353	39 K	
R-354	40 K	
R-355	210 K	
R-356	56 K	
R-357	160 K	1 w
R-358	40 K	
R-359	750 ohms	
R-360	70 K	
R-361	20 K	
R-362	750 ohms	
R-363	100 K	
R-364	50 K	Potentiometer
R-365	180 K	
R-366	100 K	1 w
R-367	100 K	
R-368	100 K	
R-369	25 K	Potentiometer
R-370	80 K	
R-371	100 K	
R-372	20 K	1 w
R-373	750 ohms	
R-374	20 K	
R-375	750 ohms	1 w
R-376	100 K	Potentiometer
R-377	68 K	
R-378	50 K	
R-379	60 K	1 w
R-380	100 K	
R-381	500 K	
R-382	10 K	Potentiometer
R-383	20 K	
R-384	800 ohms	
R-385	400 ohms	
R-386	400 ohms	
R-387	200 K	
R-388	75 K	1 w
R-389	120 K	
R-390	47 K	
R-391	280 K	
R-392	12 K	10 w
R-393	500 ohms	
R-394	120 K	
R-395	47 K	
R-396	280 K	1 w
R-397	200 K	
R-398	75 K	
R-399	10 K	
R-3001	400 ohms	Potentiometer
R-3002	20 K	10 w
R-3003	30 K	1 w
R-3004	500 K	

J-301	
J-302	Chassis Receptacle, Cannon XK-3-14
J-303	
J-304	Chassis Receptacle, AN-3102-16S-1P
J-305	Chassis Receptacle, Cannon XK-3-14
J-306	
S-301	Switch Centralab K-121 with 1 "B" Section
S-302	Switch Centralab K-122 with 4 "J" Sections
S-303	Switch Toggle, Spring return 8831 K-2

C-301	10 µf	Electrolytic
C-302	1 µf	Paper
C-303	0.015 µf	
C-304	0.02 µf	
C-305	10 µf	Electrolytic
C-306	0.06 µf	
C-307	0.06 µf	
C-308	0.1 µf	
C-309	0.3 µf	
C-310	1.0 µf	Paper
C-311	0.002 µf	
C-312	1.0 µf	
C-313	0.05 µf	
C-314	1.0 µf	Paper
C-315	1.0 µf	Paper
C-316	0 002 µf	

V-301	6SN7	V-308	6SN7
V-302	6SN7	V-309	6SN7
V-303	6H6	V-310	6SN7
V-304	6H6	V-311	6SJ7
V-305	6SN7	V-312	6SN7
V-306	6SJ7	V-313	6L6
V-307	6SN7	V-314	6L6
		V-315	6AC7

L-301, L-302	Choke, U.T.C. VI-C19
M-301, M-302	Meter, milliammeter D.C. 5-0-5 Weston
T-301, T-302, T-303, T-304, T-305	Audio Development Company Number A-5311-A

K corresponds to a multiplying factor of 10^3.

M corresponds to a multiplying factor of 10^6.

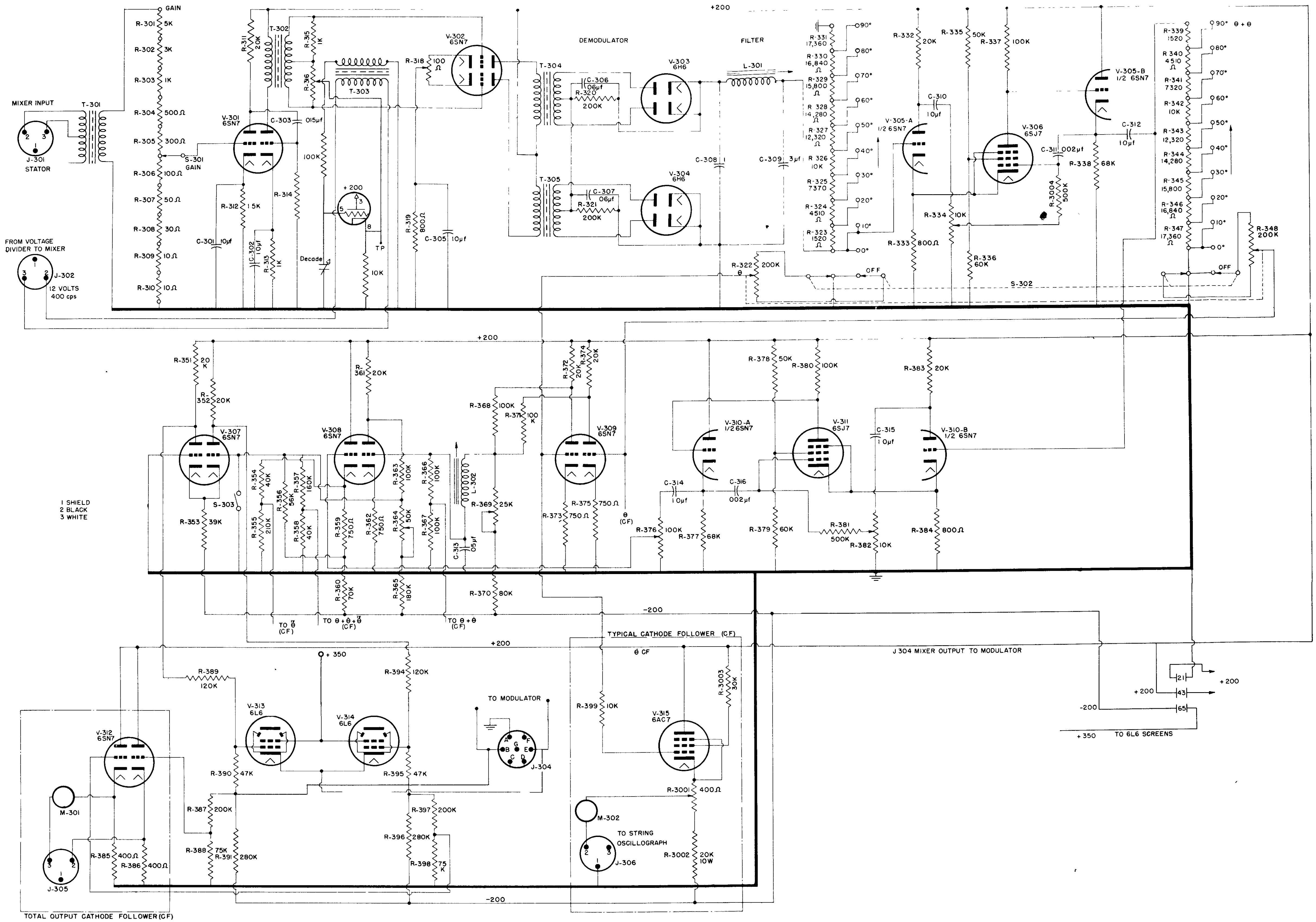

Figure 18 - Wiring Diagram and Parts List of the Mixer Unit

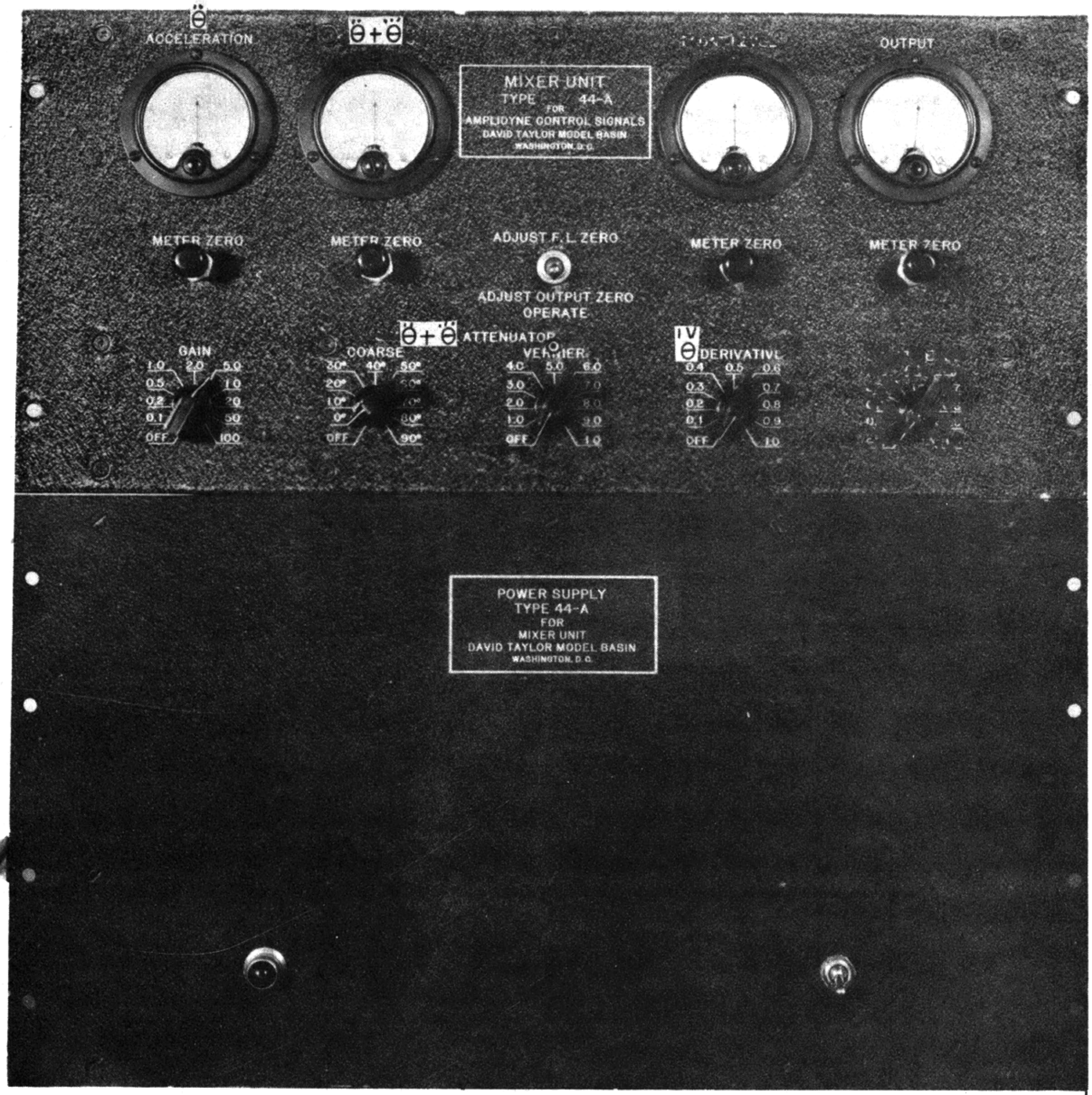

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Figure 19 - Photograph of the Mixer Unit and Its Power Supply

two signals are then transmitted to their respective pumping systems. A diagram of the circuits for this modulator unit for water-level control is shown in Figure 22.

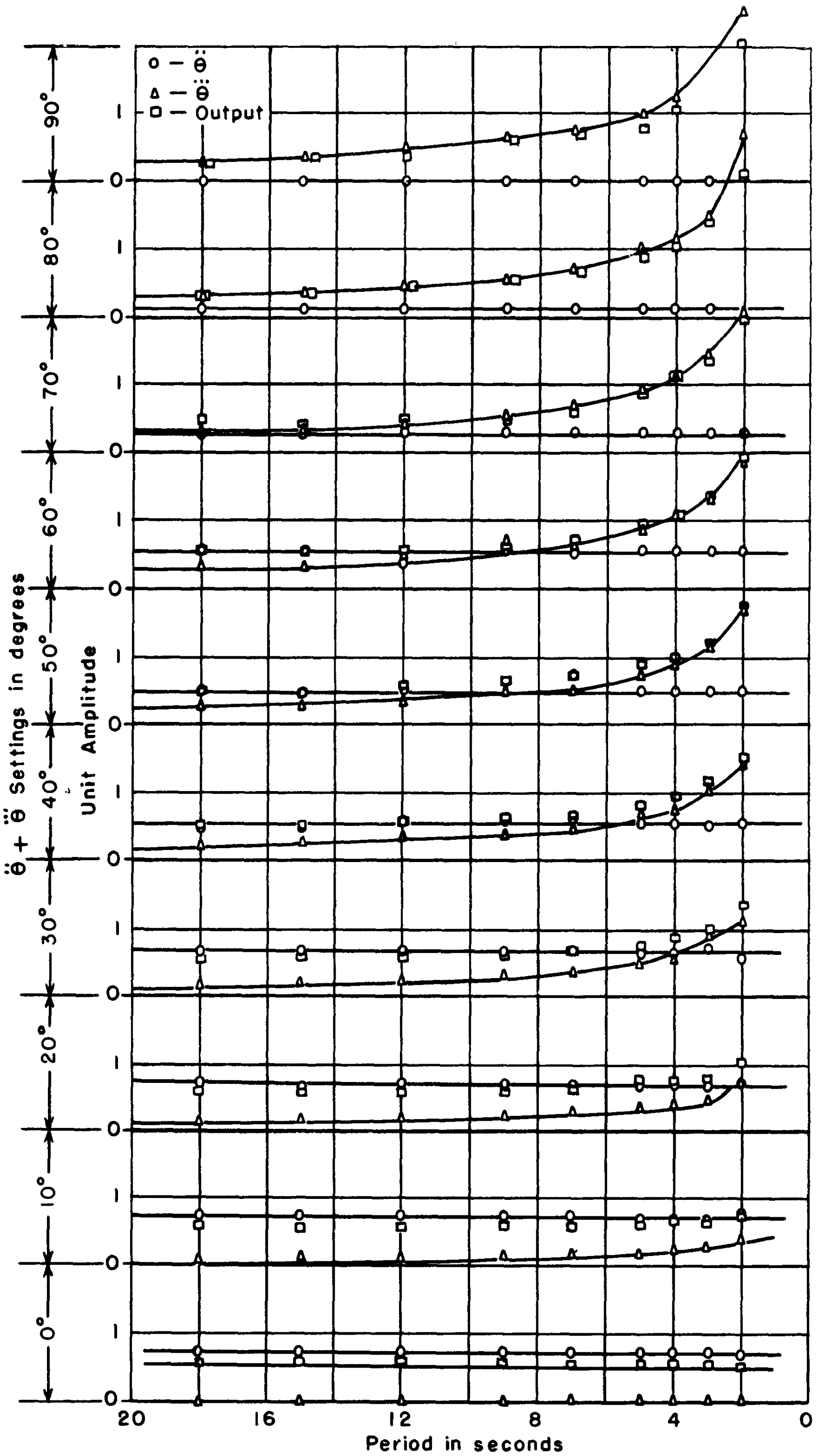

Figure 20a - Amplitude of $\ddot{\theta}$, $\dddot{\theta}$, and Output for Various $\ddot{\theta}$ and $\dddot{\theta}$ Settings Plotted against Period

Figure 20 - Calibration Curves of the Mixer Unit

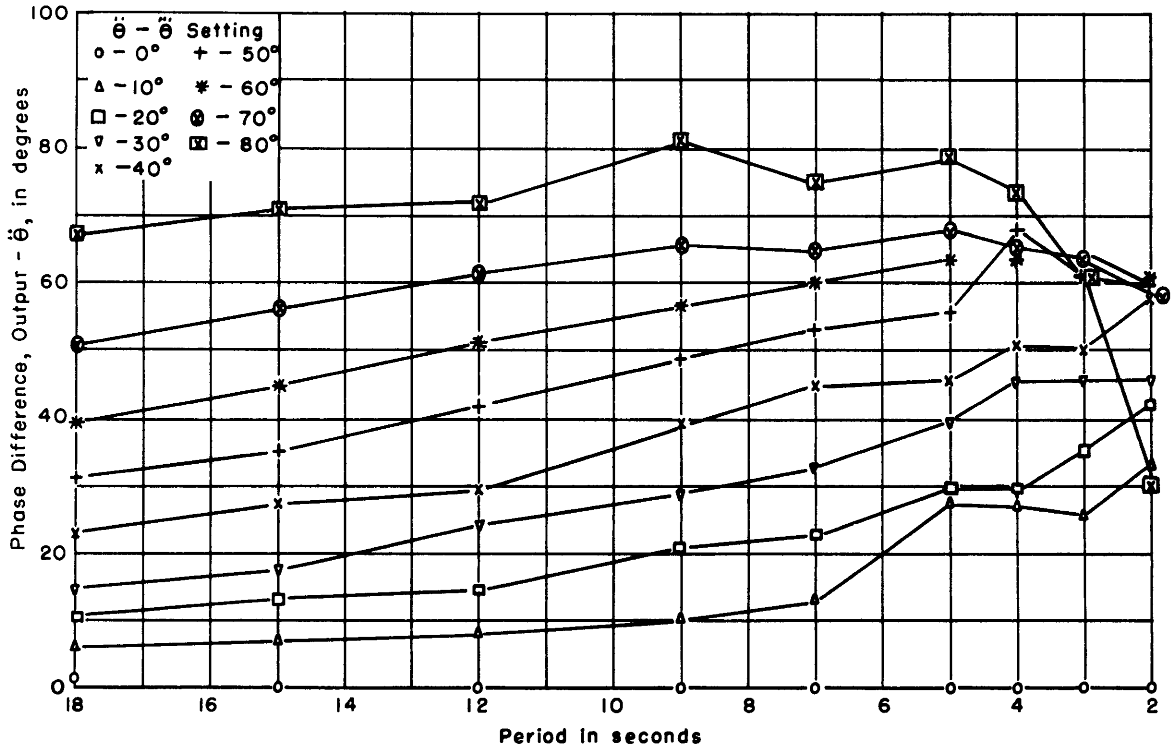

Figure 20b - Phase Difference between Output and $\ddot{\theta}$ Plotted against Period

Figure 20 - Calibration Curves of the Mixer Unit

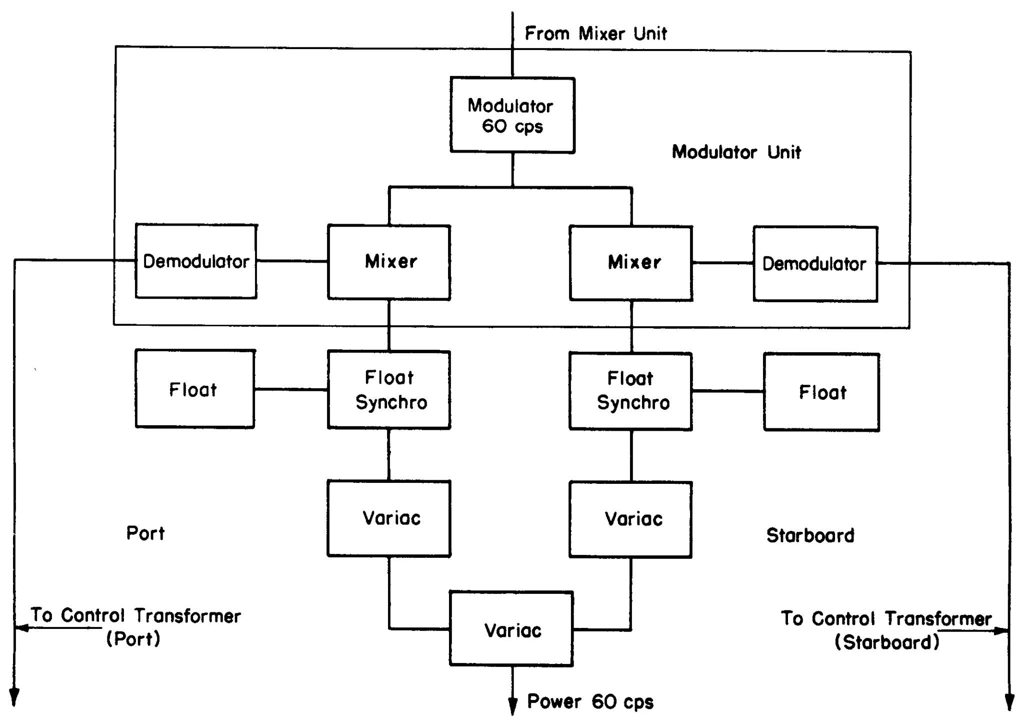

Figure 21 - Schematic Diagram of the Modulator Unit

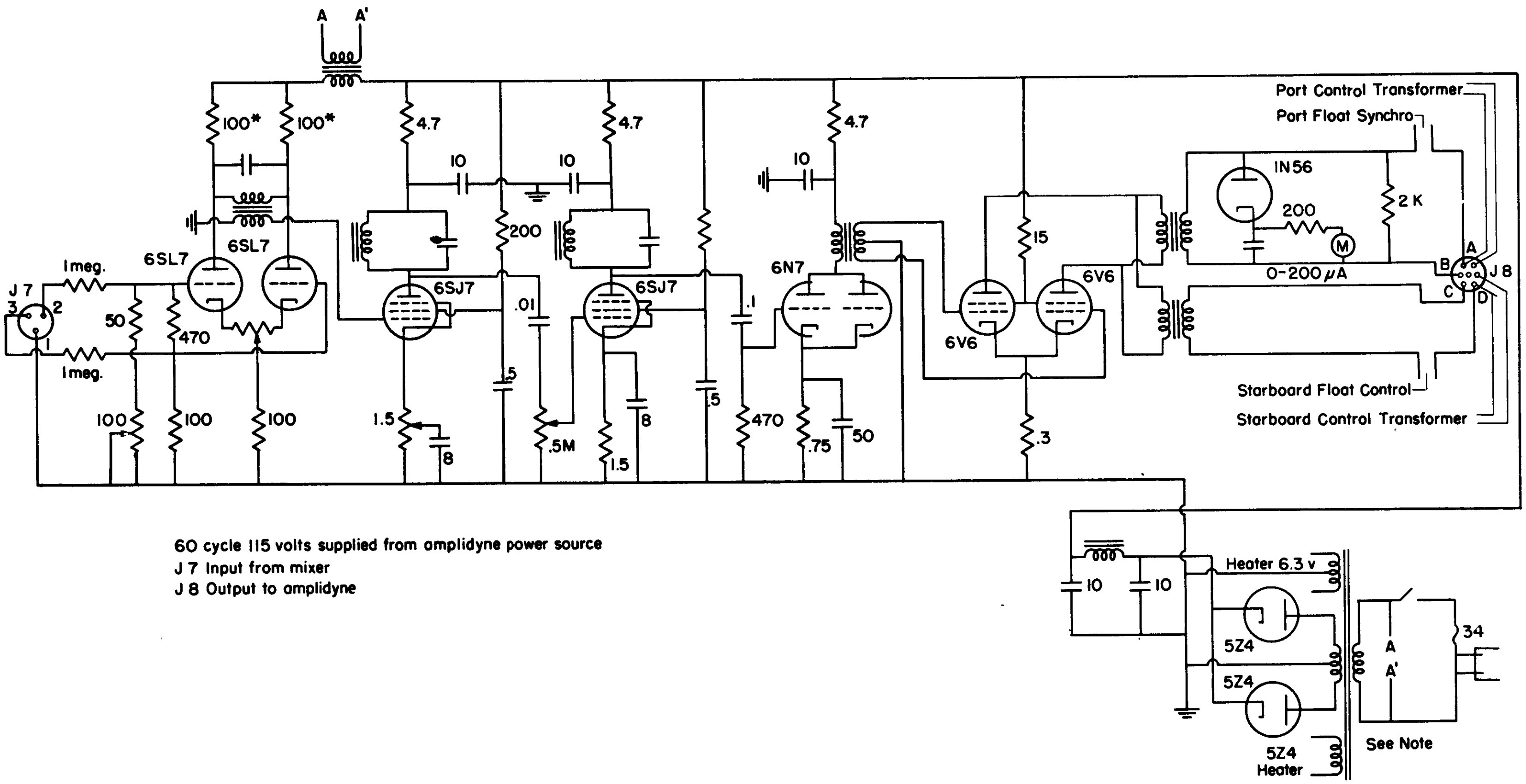

Figure 22 - Wiring Diagram of the Modulator Unit

Over-all Calibration of the Control System

In addition to the calibration of the individual units of the control system (the driver unit, the mechanical filter, and the mixer unit) over-all calibrations were made of the driver unit and mixer unit operating together, and of the driver unit, mechanical filter, and mixer unit together. The driver unit and mixer unit were calibrated operating together in the hope that the mechanical filter would not be needed for operating the stabilizing system. A calibrator built for the project was used to obtain the control-system data. It is a system of variable-diameter pulleys and belts driving a synchro which generated the 400-cps signal, amplitude modulated sinusoidally, for simulating the accelerometers. The calibrator had a speed range such that periods of from 18 to 1/10 seconds could be produced. Data were obtained on the string oscillographs at the points desired in the control system through cathode-follower and resistance circuits.

In calibrating the driver and mixer units together, curves almost exactly the same as those in Figures 20a and 20b were obtained where amplitude and phase were plotted against period. This was to be expected since the driver unit was very linear at its useful settings. However, the calibration proved there were no coupling effects between the two units.

A proof of the effectiveness of the individual control units is an over-all calibration of the entire control system. In Figure 23 amplitudes and phase differences are plotted against period for two extreme conditions, one with the mixer $\ddot{\theta} + \dddot{\theta}$ setting at 0°, that is, with no differentiation of the mixer input, and the other with the setting at 90° under which condition the entire mixer input is differentiated and is the only output. In Figure 23a amplitudes for a mixer $\ddot{\theta} + \dddot{\theta}$ setting of 0° are plotted against period for mechanical filter inputs of 40°, 20°, and 10° double amplitude. Here, as in the individual calibration of the mechanical filter, the output is fairly uniform in amplitude except at the shorter periods where resonant peak amplitudes are obtained. The peaks seem to shift to lower periods with a decrease in the mechanical-filter inputs, the damping being less for the smaller inputs. The resonant peaks, as in the mechanical-filter calibration, Figure 15, are double-peaked in the resonant region—indicating a coupling effect in this frequency range. There no doubt are two degrees of freedom owing to mechanical or electrical components or both. Fortunately the resonant peaks are in the region of frequencies which would not be encountered in the rolling of ships, although higher frequencies do have some effect. In Figure 23b the amplitudes are again plotted against period but for a mixer $\ddot{\theta} + \dddot{\theta}$ setting of 90°. The curves are characterized by the same flatness in the range of 18 to 7 seconds, but the peaks are greater since the mixer input signal is differentiated. In both Figures 23a and 23b the resonant region is characterized by the irregularity of the data. This can be caused by a number of factors such as instantaneous variation of the calibrator speed, variation of gear friction and damping within a cycle in the mechanical filter, variation in supply voltage, and small errors in the analysis of the many records necessary to obtain these curves. On the resonant portions of the curve a small change in frequency can cause a big change in amplitude.

Phase differences between the output of the mixer unit and the input of the driver unit are plotted against period in Figure 23c for a mixer $\ddot{\theta} + \ddddot{\theta}$ setting of 0°. Here the phase difference remains on the order of 10° to 20° from 18 to 6 seconds and then rapidly increases to about 145° from 6 seconds to 2 seconds. The phase difference is a lag of the mixer output with respect to the driver input. In Figure 23d differentiation at a mixer $\ddot{\theta} + \ddddot{\theta}$ setting of 90° causes a lead of about 70° in the mixer output and, as the period decreases from 6 seconds to 2 seconds, the lag of the mechanical filter is superimposed on the lead until at 2 seconds period there is a net lag of about 90°.

In this series of calibrations the damping on the mechanical filter was kept at the same setting, and the calibration was completed without turning off the equipment. Widely scattered data were checked by the reanalysis of records and by repeated runs.

Typical oscillograms of the control-system calibration are not presented in this report for various reasons. All traces of those calibrations obtained without the mechanical filter in the system were sinusoidal for $\ddot{\theta}$, $\dddot{\theta}$, $\ddot{\theta}$ and $\ddddot{\theta}$, and for the output of the mixer unit. The difference between the traces was in the phase shift, and all values have been presented previously in graphs. Only at high calibrator speeds at a period of about 3 seconds was slight distortion noticed. With the mechanical filter in the control system the oscillograms indicated badly distorted wave shapes on many runs. These traces were used for the calibration curves, however, even though later correction of difficulties (such as bad tubes, condensers, and an improperly isolated connection in the driver unit) eliminated much of the distortion. However, there was no opportunity later to repeat this set of calibrations,

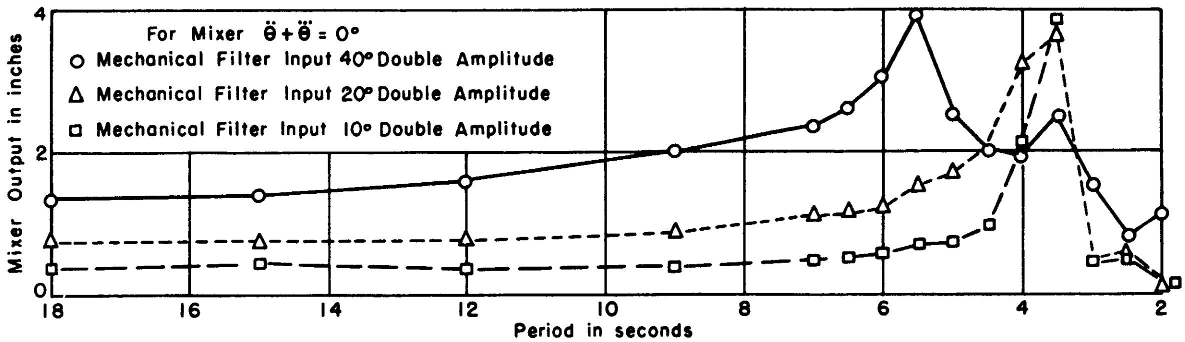

Figure 23a - Amplitude of the Output of the Mixer Unit Plotted against Period for a Mixer $\ddot{\theta}+\dddot{\theta}$ Setting of 0 Degrees

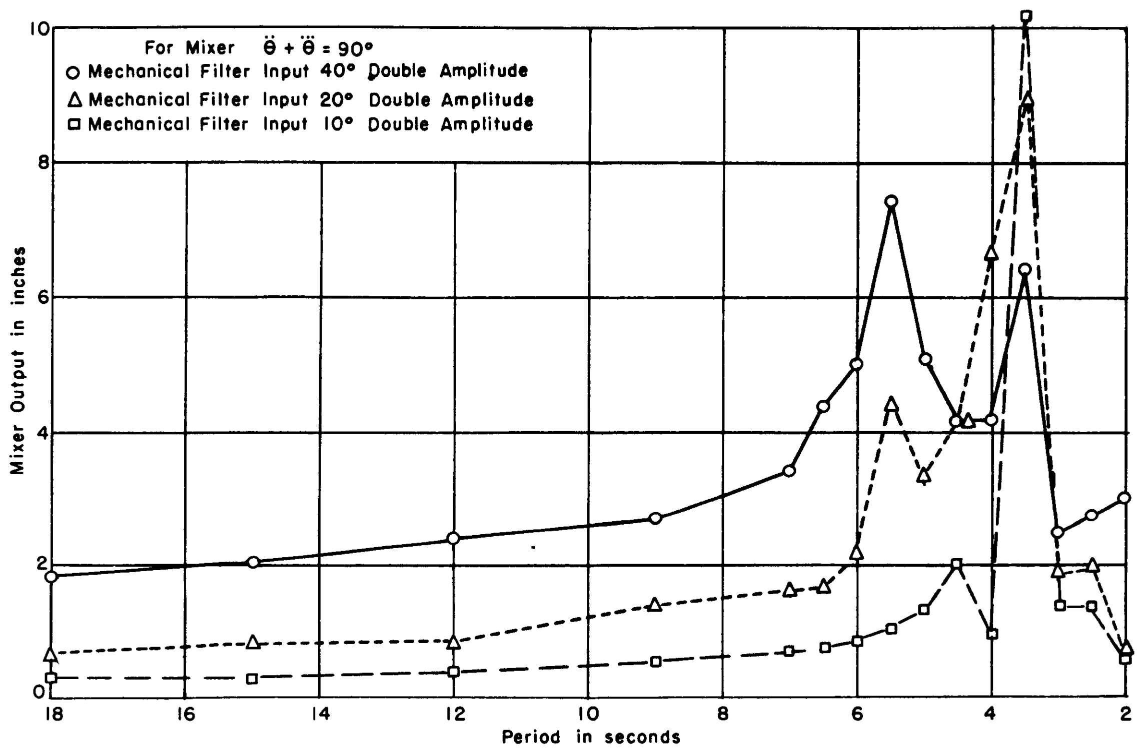

Figure 23b - Amplitude of the Output of the Mixer Unit Plotted against Period for a Mixer $\ddot{\theta}+\dddot{\theta}$ Setting of 90 Degrees

Figure 23 - Effects of Amplitudes and Phase with Respect to Period of the Output of the Mixer Unit upon Over-All Calibration of the Control System

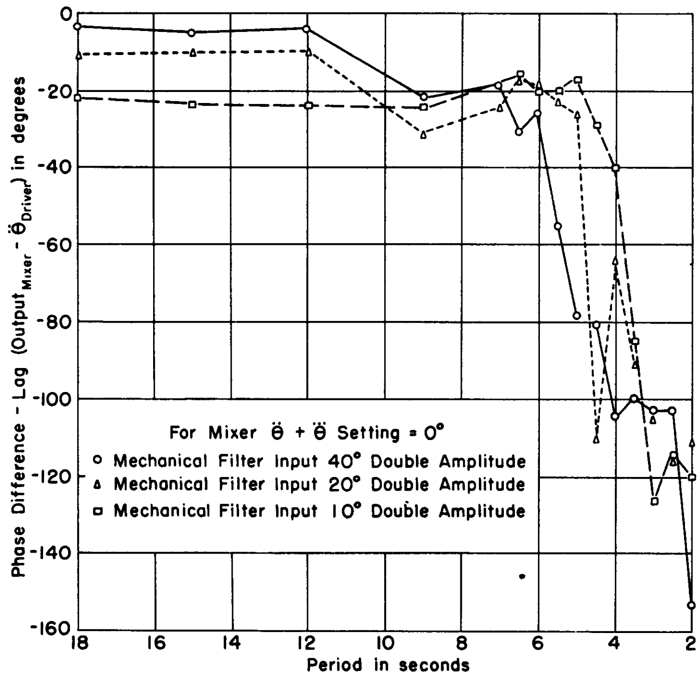

Figure 23c - Phase Difference Between the Mixer Output and Driver Input Plotted against Period for a Mixer $\dot{\theta}$ + $\ddot{\theta}$ Setting of 0 Degrees

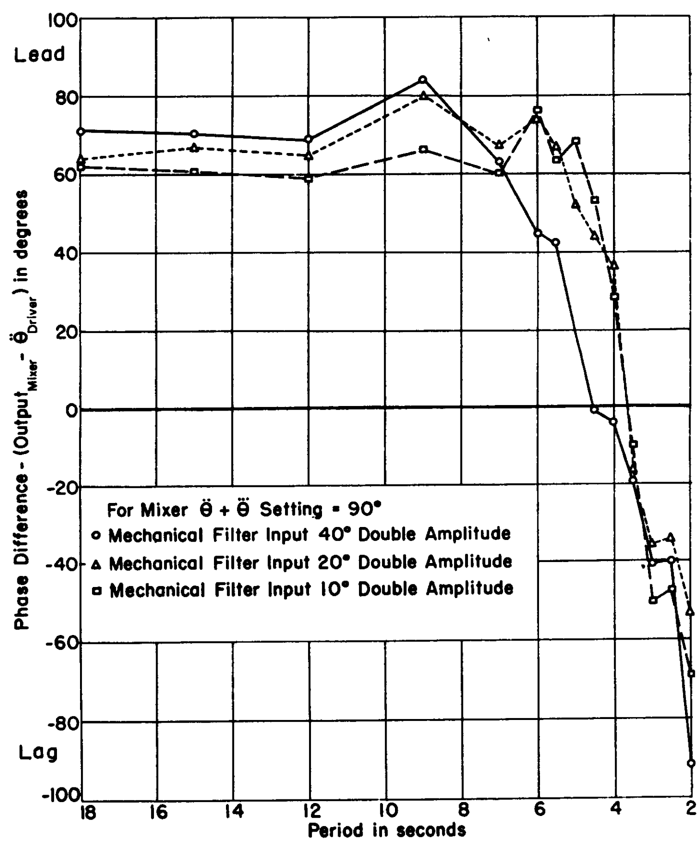

Figure 23d - Phase Difference Between the Mixer Output and Driver Output Plotted against Period for a Mixer $\dot{\theta}$ + $\ddot{\theta}$ Setting of 90 Degrees although visual and spot calibration indicated great improvement in the wave shape.

A large number of transient calibrations were made of the control system with two arrangements: One with the driver and mixer units and the other with these units together with the mechanical filter. A variety of transient signals were imposed at the input end of these arrangements: Opening, closing, and reversing to impose step pulses, rectangular pulses, impulses of less than 1/30-second duration, and sinusoidal signals of varying frequencies.

No attempt was made at the time to analyze these calibrations completely since the analysis more properly belongs with the problem of the redesign of control circuits. Some salient characteristics are described, however. In the driver-mixer setup with step and rectangular pulses there was almost perfect response. Recovery occurred in less than 1/10 second. Impulses had negligible effect. Interrupted circuits, which would not be encountered in normal operation, of course, resulted in larger recovery times, of the order of 1/2 second, necessitated by the time necessary to recharge condensers to zero signal level. With the sinusoidal signal a resultant resonant period of about 0.2 second was obtained. With the mechanical filter in the circuit, the disturbances were much greater. For constant amplitude input, as the duration of the rectangular pulses and their spacing decreased, the disturbance was less. The transient disturbance always resulted in the free oscillation of the output end of the mechanical filter at the 3-second natural period. This resulted too from the varying-frequency, sinusoidal, input signal. Impulses had negligible effect.

THE PUMPING SYSTEM

The pumping system may be considered that portion of the stabilization installation starting at the output of the modulator unit of the control system and ending with the pumping of water in the stabilizing tanks. Its principal components are the amplidyne amplifiers and their connection to the control system, the amplidyne power supply, the pitch motor, the pitch mechanism, the pumping mechanism, and the pumps or impellers.

A signal from the modulator unit is fed to the port amplidyne amplifier for the forward tank system and another signal is fed to the starboard amplidyne amplifier for the aft tank system through control transformer circuits. The amplidyne amplifiers control the amount and the polarity of the d-c power furnished to the pitch motors; consequently they control the amount and direction of the motion of the d-c pitch motors in proportion to the accelerometer signal modified by the control system. The pitch motors control the amount and direction of the pitch of the impeller blades through gearing, a pitch shaft inside the hollow impeller shaft, and linkage in the hollow impeller hub. Meanwhile the impellers are turned continually in one direction—driven by d-c motors through herringbone gears at constant speed; the amount and direction of the pumping of the water depends on the pitch. A general diagram of the pumping machinery is shown, together with photographs, in Figure 24.

Amplidyne Units

The signal from the modulator is injected into the amplidyne amplifier to the grids of V161 and V162 through a circuit which included a zeroing synchro and a control transformer on the pitch mechanism. Figure 25 is a diagram showing this arrangement. The purpose of the zeroing synchro is to position initially the impeller blades after adjustment of the control circuits. The amplidyne amplifier, in turn, controls the amount and polarity of the power supplied to the drive motor of the pitch mechanism, thus determining the pitch-time relationship of the impellers.

Rather than try to design a drive for the pitch mechanism in the brief time available for design and construction of the stabilization installation, and, since an amplidyne-type drive seemed very desirable for the ship stabilizing system anyway, it was decided to adapt appropriate amplidyne gun-mount drives for the purpose of changing pitch of the impeller blades. Based on deductions from the power requirements of the hydraulic pitch mechanism of the HAMILTON installation, a 7-hp motor was deemed more than sufficient to change the pitch under the conditions of operation for ship stabilization. Therefore the Five-Inch Elevation Power Drive Mark 14 Mod 0 was selected for adaptation to the pitch mechanism.

Upon installation and trial operation, the drive motor was found to be inadequate for changing pitch more than ±5° with the impellers rotating. To find out why, the entire pitch mechanism was disconnected from the pitch shaft; then, with the impeller motor still connected to the impeller shaft, holes were burned in the overheads of the motor rooms and a dynamometer test was performed on the pitch shafts with the impellers running. Over 7000 lb were required to change pitch at large angles, 20° to 30°, under these conditions. The data from this test are summarized in Table 9. This particular pitch installation will not be described since it is similar to the Five-Inch Train Power Drive Mark 14 Mod 0 finally installed. This was selected as a substitute since the drive motor had a rating of 40 hp and was more than adequate for the duty requirements now definitely known. Furthermore, it could

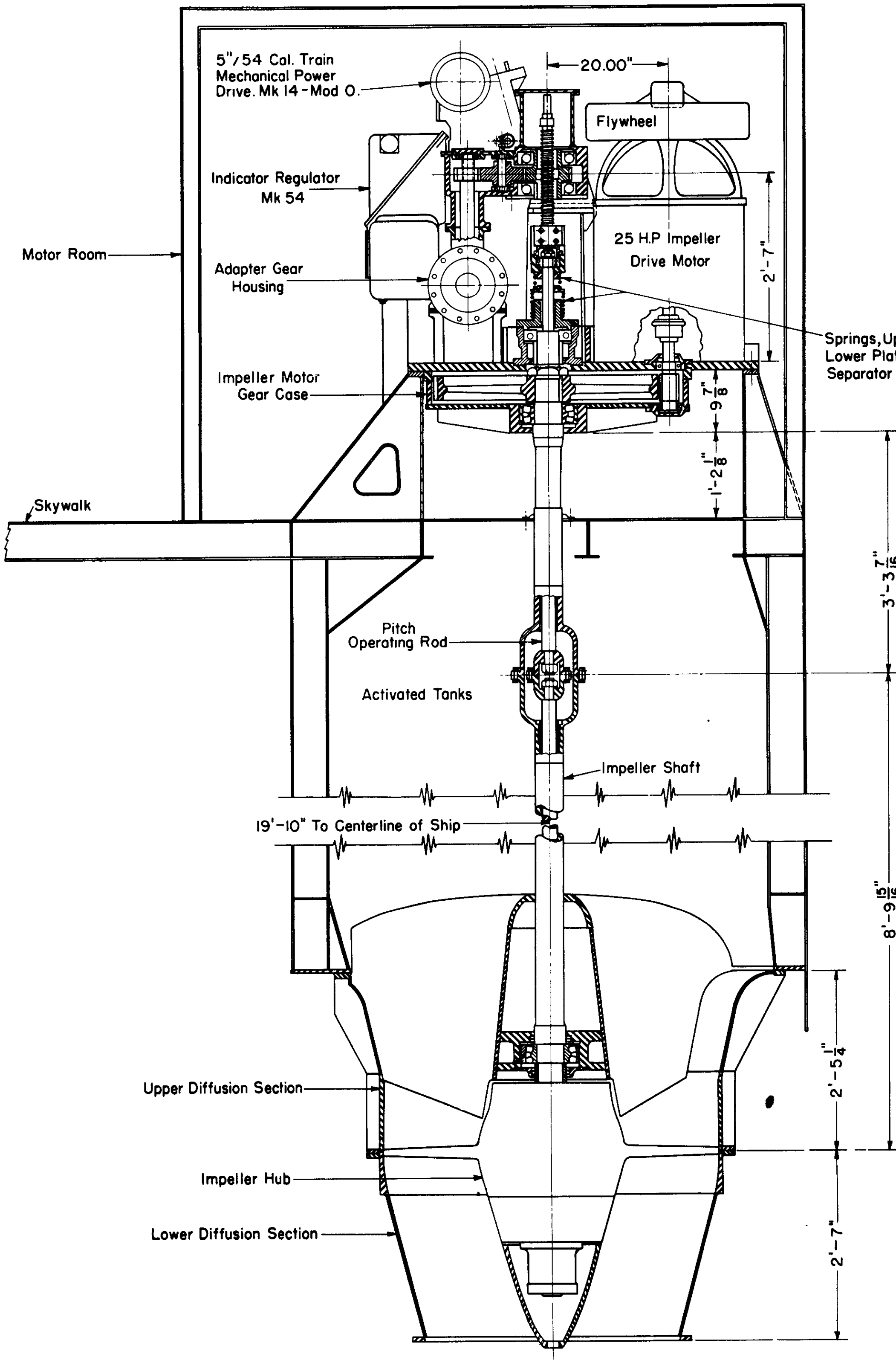

Figure 24a - Sectional Diagram of the Tank Arrangement,
Pumping and Pitch Mechanism, and the Impellers

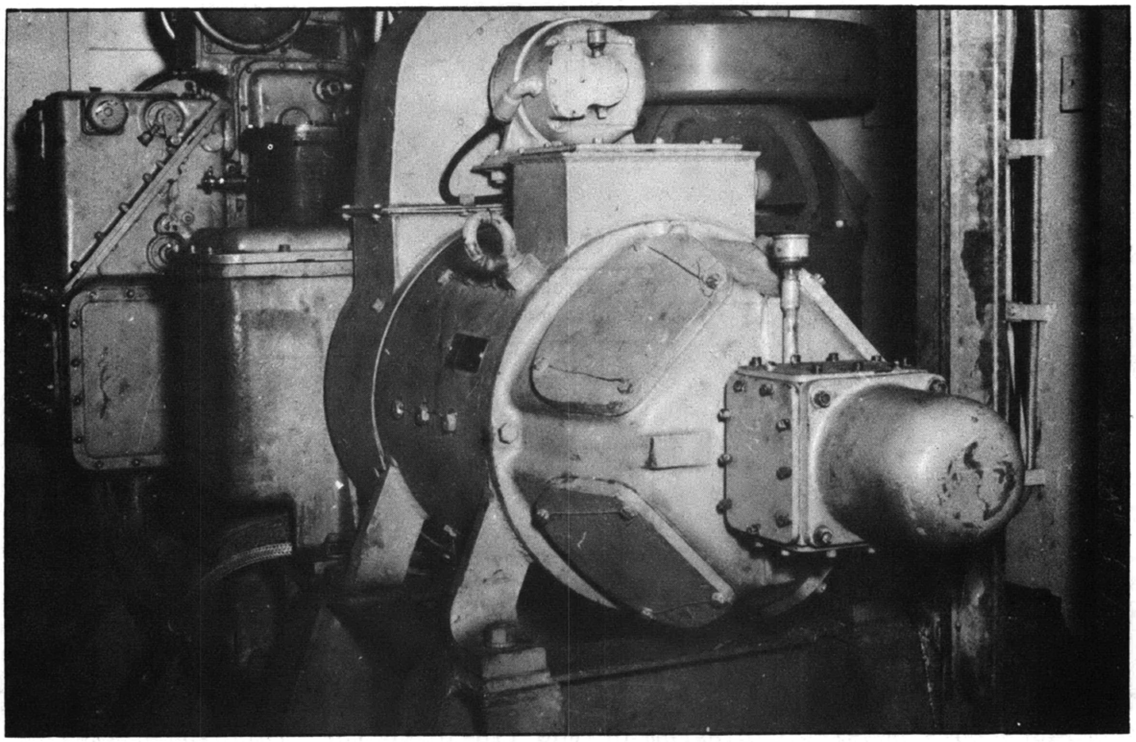

Figure 24b - Photograph of the Pumping Mechanism and the Pitch and Impeller Motors

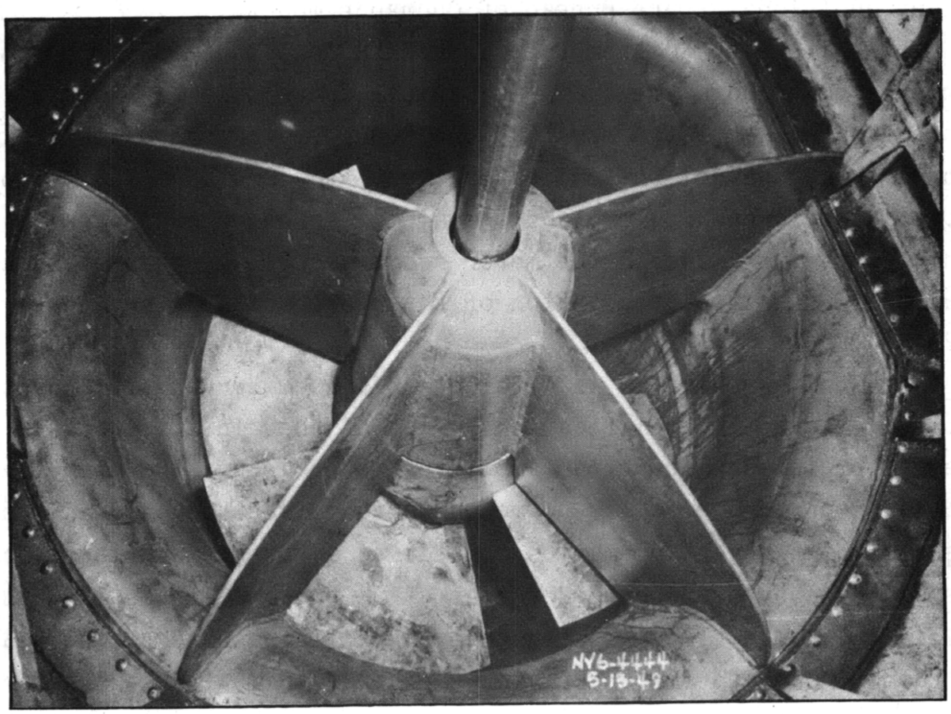

Figure 24c - Photograph of the Impeller

Figure 24 - Diagrams and Photographs of the Pumping Machinery

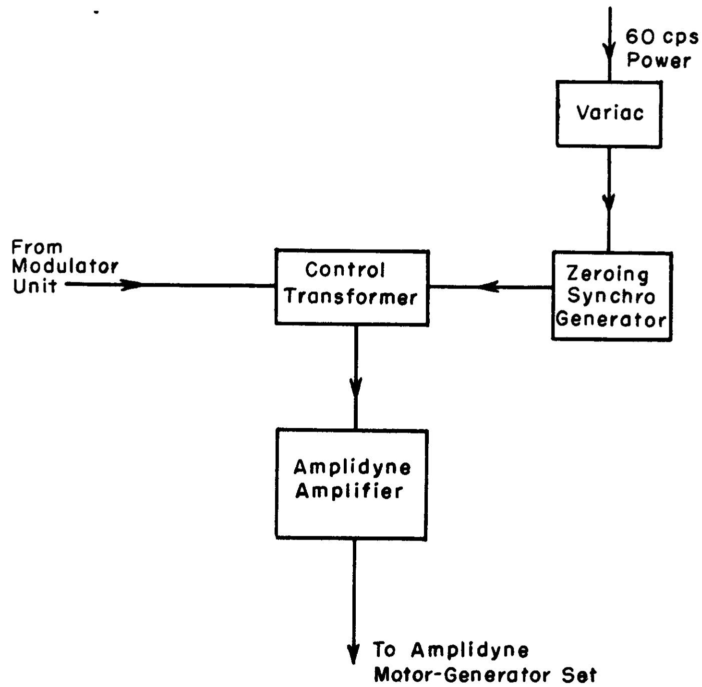

Figure 25 - Schematic Diagram of the Tie-In to the Amplidyne Units

be removed from the same 5-inch gun mounts as the elevation drives without pirating any additional spare Bureau of Ordnance mounts.

The units removed from the gun mounts for use in changing the pitch of the impellers included the train mechanical drive unit together with the Gun Train Indicator Regulator Mark 54, the 40-hp train motor, the brake unit, the train selector switch, the indicator light, the train amplidyne motor generator, the magnetic controller, and the amplifier unit. Photographs of these units, which were installed in the motor rooms and on the boat deck, are shown in Figure 26. A detailed description of their operation is presented in Reference 42.

The amplidyne system is a follow-up system serving two purposes: Remote control and power amplification. Most of the amplification in the amplidyne system is supplied by an amplidyne motor generator which furnishes the direct current to the follow-up motor which drives the load. Even an ordinary direct-current generator may be considered a power amplifier since only a small exciting current is needed to magnetize the pole pieces and the output is generally about 100 times greater than the exciting power. The additional power is furnished by the motor driving the generator. In the ordinary d-c generator the amplification is inadequate, and the response is too sluggish for use with follow-up systems. The amplidyne generator is one specially

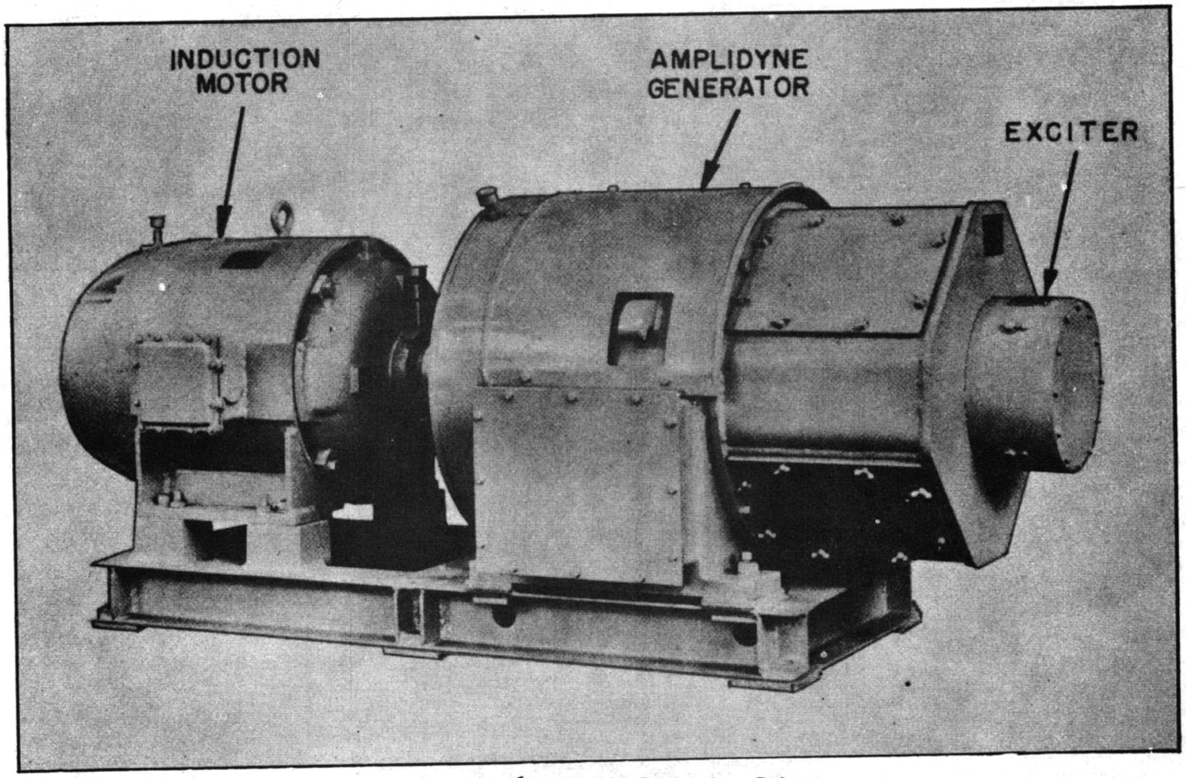

Figure 26a - Motor-Generator Set

designed to give high amplification and rapid response. It can supply power to the follow-up motor 10,000 times the excitation power in its control windings. The control current to the exciter windings is supplied by a vacuum-tube amplifier, thus furnishing additional power amplification at this point.

The load, which in this instance is the pitch mechanism, is driven by the follow-up motor which gets its power from the amplidyne d-c generator. The rotor of a synchro control transformer is connected mechanically to the pitch mechanism, the housing remaining fixed. This transformer receives an order signal from the modulator unit of the control system. The unit operates so as to indicate electrically what the position of the pitch mechanism should be. The synchro compares the ordered position and, if there is not agreement, it generates an a-c signal transmitted to the amplifier. The angular difference between the two positions is called the error, and the signal to the amplifier is an error signal and indicates the size and direction of the error. A schematic diagram showing the relationships is given in Figure 27, the tie-in to the modulator unit in Figure 25, and a more detailed diagram of the amplidyne installation for stabilization in Figure 28.

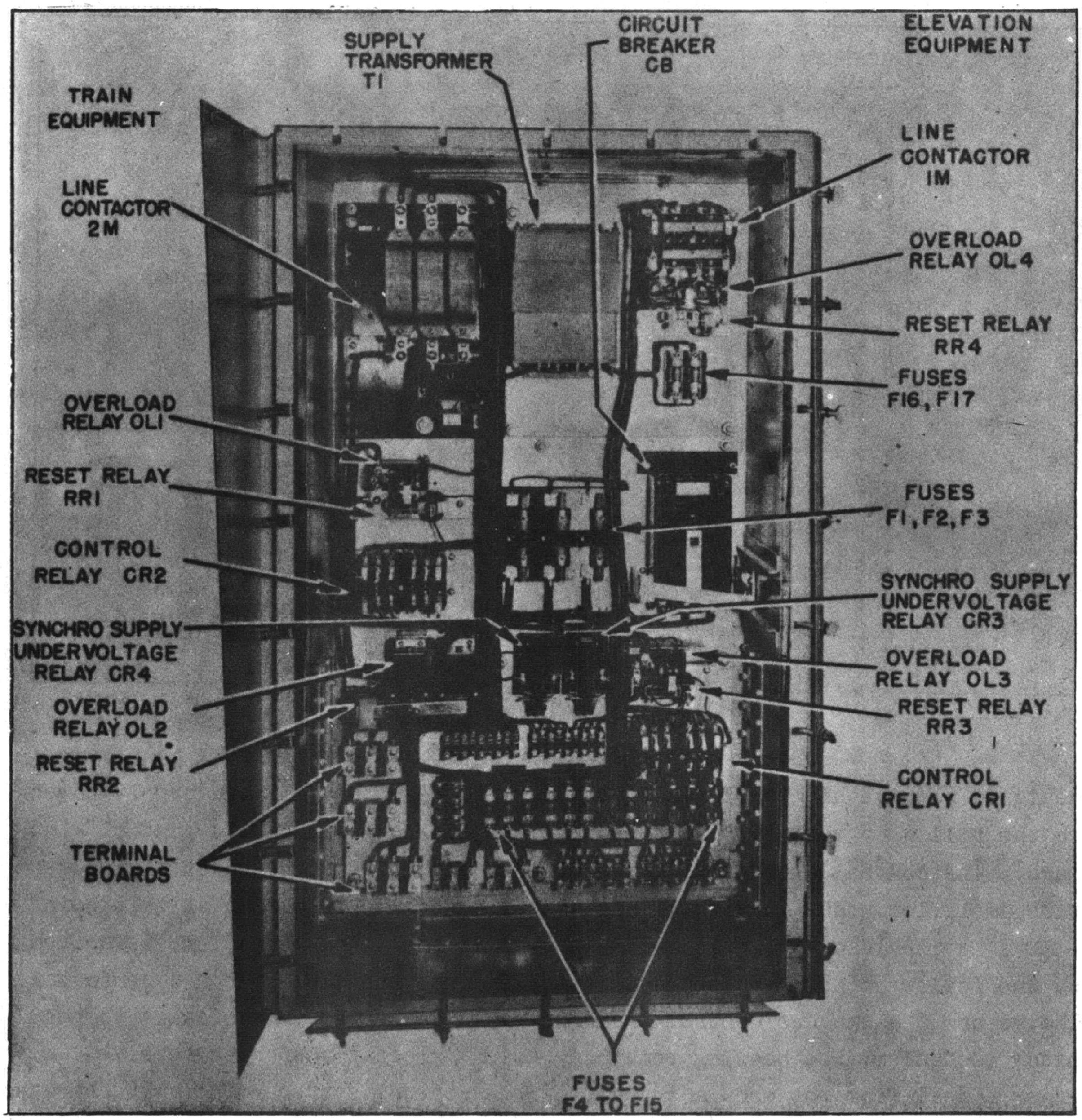

Figure 26b - Magnetic Controller

Since the synchro-control transformer and the amplidyne generator are important elements in the stabilization system, more needs to be said of them. The operation of the synchro-control transformer is compared with that of the synchro generator; both are shown schematically in Figure 29. The stator of the control transformer is similar to that of the synchro, having three S windings but with more turns. The rotor differs however in having a number of coils set in slots around the rotor and connected in series. The

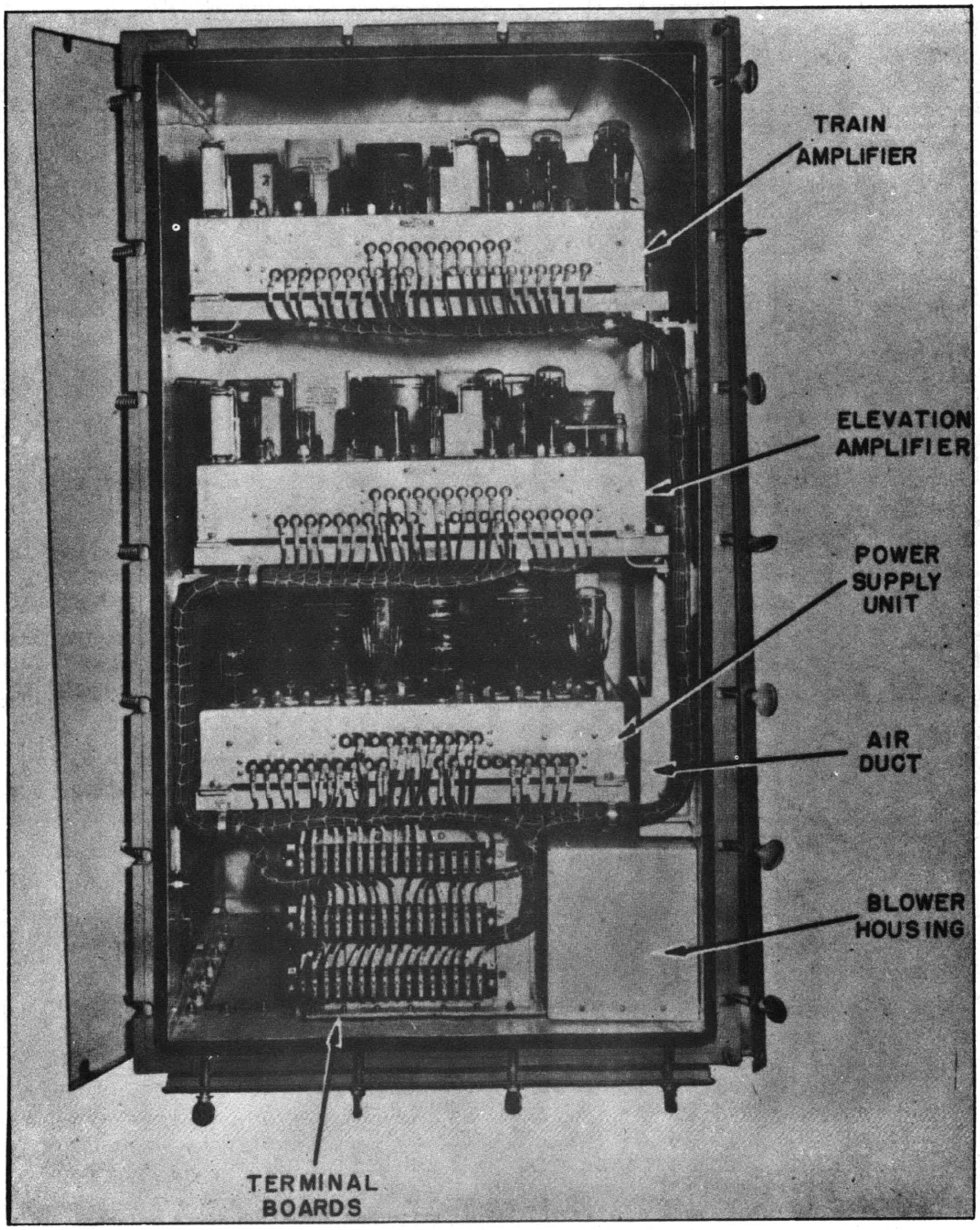

Figure 26c - Amplifier Unit

two ends of the winding are connected to slip rings, R_1 and R_2. When the rotor of the transformer is at electrical zero, it is turned 90° to the S_2 winding. When the stator coils of a control transformer are energized by a synchro generator, the rotor does not turn but a voltage is induced in it. The rotor of the transformer is turned mechanically; the rotor windings are arranged so that induced currents in the rotor do not affect the stator currents. In effect, the turning of the control transformer changes the field. If the rotor by being mechanically turned is changing its field

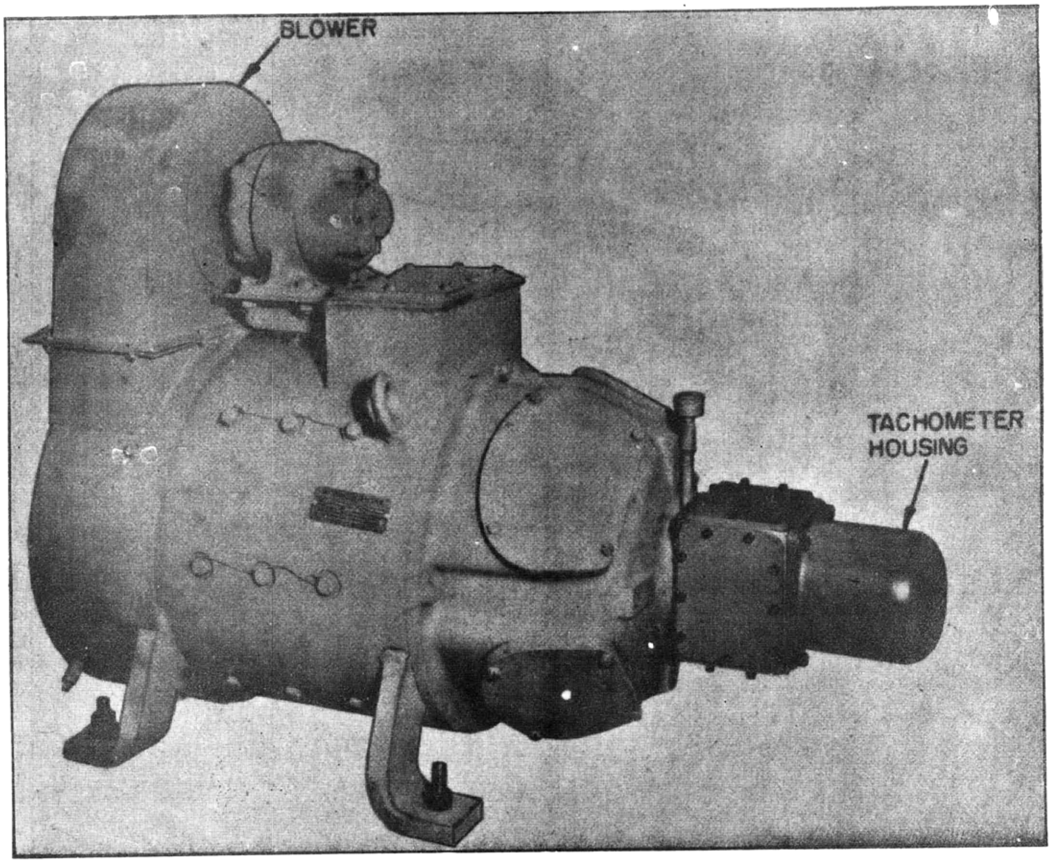

Figure 26d - Train Motor

at the same rate that the stator field of the transformer is changing, no current will flow at R_1 and R_2, and therefore no error signal is induced. The signal from the modulator is supplied to the stator of the control system as shown in the diagram of the stabilizing signal tie-in, Figure 25.

The amplidyne generator is essentially an ordinary d-c generator with certain modifications which permit great power amplification, accuracy of control, and rapidity of response. A coil of wire rotated in a magnetic field induces voltages in a coil and, if the ends of the coil are connected, current will flow. An ordinary generator is made up of a stator and a rotating armature. A coil of wire is wound about the stator and supplied with a small exciting current which serves to magnetize the iron in the stator and the rotor to provide the necessary magnetic field for the armature coils to cut. These coils are connected to armature bars which rotate with the armature. When a commutator is connected to outside circuits, current will flow. A simplified diagram showing the stages in the transformation of an ordinary d-c generator to an amplidyne d-c generator is shown in Figure 30. In Figure 30a is shown the ordinary d-c generator where 100 watts excitation on the field results in 100 volts, 100 amperes, and 10 kilowatts. Other conditions being equal, the

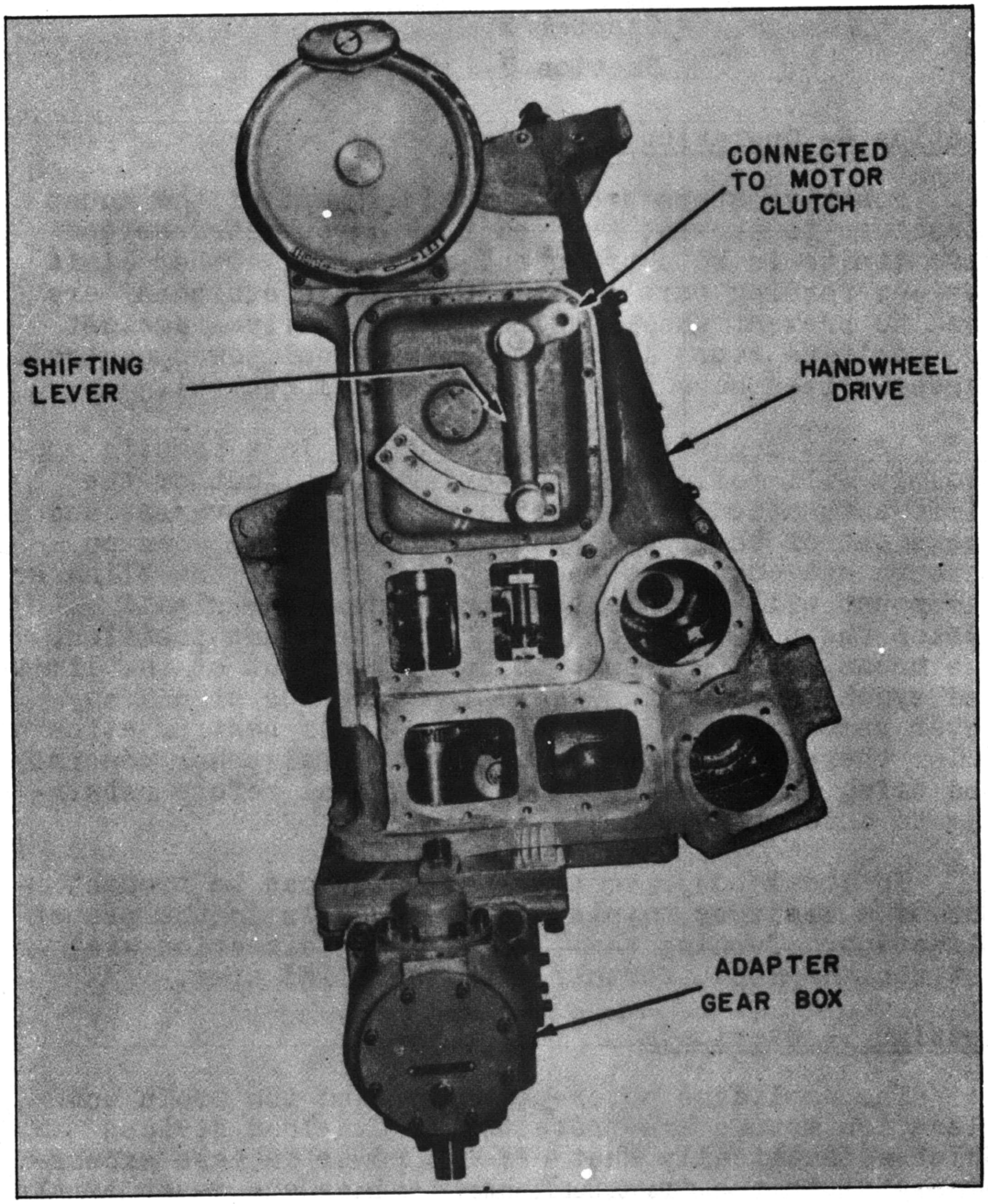

Figure 26e - Handwheel Driver and Adapter Gear Box

Figure 26 - Photographs of the Units Taken from the
5-Inch Train Power Drive Mark 14 Mod 0

power output of the generator is proportional to the power input to the excitation winding under normal operation. With 100 w excitation and 10 kw output the amplification is 100 to 1. The excitation current produces a magnetic field FC. This magnetic field induces 100 v across the brushes. At the same

TABLE 9

Dynamometer Tests on the Pitch Shafts to Determine the Axial Force Needed to Change Pitch*

Unit	Angle of Impeller Blade	Force pounds	Impellers Running or Off	Remarks
Starboard	About 15°	6500	Running	Impeller speed down some owing to loading
	25°	6500	Running	Impeller speed down 2/3
	About 30°	5800	Running	Impeller motor out but impeller still turning
	20°	6200	Running	Impeller speed down
Port	10°	2400	Off	To pull back to 0°
	10°	6200	Running	To pull back to 0°
	10°	2600	Off	To pull back to 0°
	10°	6000	Running	To pull back to 0°
	(Down to 10° from 0° and back quickly)	2400	Running	To pull back to 0°
	10°	6400	Running	To pull back to 0°
	5°	2400	Off	To pull back to 0°
	5°	6400	Running	To pull back to 0°
	5°	2600	Off	To pull back to 0°
	5°	6500	Running	To pull back to 0°

* Values could not be obtained at angles greater than 10° single amplitude except by change because the impeller-motor controllers kept kicking out.

time 100 amp flowing in the armature coils creates another magnetic field FS at right angle to FC. This field—armature reaction—does no useful work in the ordinary generator and is troublesome. It has about the same strength as the field FC. If the brushes are short-circuited, as shown in Figure 30b, an extremely large current flows. The excitation may now be considerably reduced, say to 1 w, and still produce the normal full current of 100 amp through the short-circuited path. The same armature reaction FS is produced as before. The armature reaction FS induces a current in the armature the same as flux FC, but the voltage on the commutator is 90° from the voltage induced by FC and is 100 v. Now if new brushes are added 90° from the original brushes and if the 1-ohm load is connected to them, Figure 30c, the 100 v formerly at these new points is reduced almost to zero. The current flowing

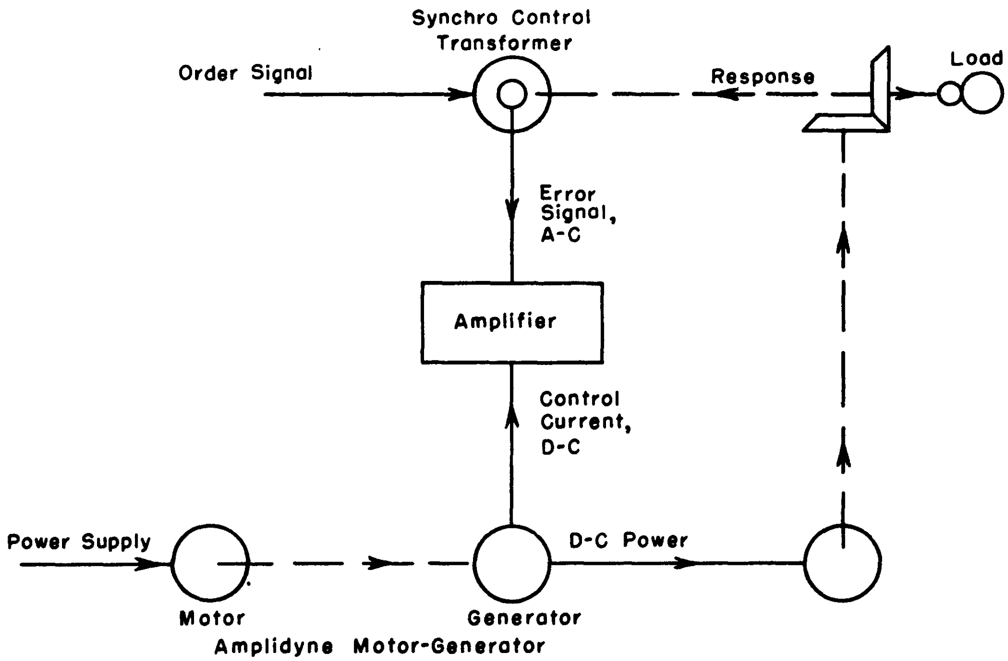

Figure 27 - Schematic Diagram of the Amplidyne Units
Used for Changing Pitch

in the coils between the two new brushes creates a second armature reaction FA opposing the exciting field FC and reduces its effect by reducing FS and consequently reducing the voltage across the new brushes. The last necessary modification is shown in Figure 30d. The armature current is sent through a compensating field winding and creates a magnetic field FB opposing FA and may be adjusted to balance out FA restoring the full effect of the exciting field FC. Now the full-load current may be drawn from the new brushes and, since both FA and FB depend on armature current, they will always balance and the output voltage is nearly independent of the armature current. Full-load output is now obtained with only 1-watt excitation and the amplification is now 10,000 to 1. This is the basic form of all amplidyne generators although refinements are necessary to produce fast, stable operation necessary in follow-up systems. In the equipment adapted for the ship stabilization test, the difference between the two excitation-control currents determines the amount of power supplied to the motor. By balancing the control currents the amplidyne output becomes zero and the follow-up motor is stationary. The direction of the magnetic field FC and the polarity of the output of the generator depends on the winding receiving the stronger current.

Parts List for Train Amplifier

R-101	39,000 ohms	1 w
R-102	5,000 ohms	
R-103	5,000 ohms	25 w
R-104	500 ohms	
R-105	27,000 ohms	
R-106	27,000 ohms	1/2 w
R-107	3,900 ohms	
R-108	3,900 ohms	
R-109	10,000 ohms	
R-110	10,000 ohms	1 w
R-111	200,000 ohms	
R-112	200,000 ohms	
R-113	50,000 onms	1/2 w
R-114	50,000 ohms	
R-115	200,000 ohms	2 w
R-116	470,000 ohms	
R-117	470,000 ohms	1/2 w
R-118	50,000 ohms	
R-119	50,000 ohms	
R-120	1,000 ohms 1,000 ohms	2 w
R-121	25,000 ohms	1/2 w
R-122	25,000 ohms	
R-123A	10,000 ohms	
R-123B	10,000 ohms	2 w
R-124	2,500 ohms	
R-125	20,000 ohms	
R-126	20,000 ohms	
R-127	100,000 ohms	
R-128	100,000 ohms	
R-129	5,000 ohms	
R-130	1,000 ohms	
R-131	1,000 ohms	1/2 w
R-132	100,000 ohms	
R-133	100,000 ohms	
R-134	200,000 ohms	
R-135	200,000 ohms	
R-136	100,000 ohms	
R-137	100,000 ohms	
R-138	250 ohms	10 w
R-139 R-140	G.E. Thyrite K-8355083G-1	
R-141	390,000 ohms	1 w
R-142	10,000 ohms	1/2 w
R-143	25,000 ohms	2 w
R-144	25,000 ohms	
R-145	15,000 ohms	1/2 w
R-146A	100,000 ohms	2 w
R-146B	130,000 ohms	
R-147A	160,000 ohms	
R-147B	160,000 ohms	
R-148A	160,000 ohms	1 w
R-148B	160,000 ohms	
R-149	100,000 ohms	
R-150	100,000 ohms	1/2 w
R-151	10,000 ohms	
R-152	25,000 ohms 25,000 ohms	2 w
R-153	15 ohms	1/2 w
C-101	0.1-0.1 µf	
C-103	5.0 µf	
C-104	5.0 µf	
C-105	0.5 µf	
C-106	0.5 µf	600 v
C-107	0.5 µf	
C-108	0.5 µf	
C-109	2.0 µf	
C-110	1.0 µf	400 v
C-111	0.05-0.05 µf	
C-113	0.1-0.1 µf	600 v
C-115	0.1-0.1 µf	
C-117	0.01 µf	
C-118	0.01 µf	300 v
C-119	5.0 µf	600 v
C-120	0.05 µf	

T-101	Filament transformer, Catalog number 80G142
T-102	Plate transformer, No. 80G172
T-103	Catalog No. 80G145
T-104	Catalog No. 80G145
T-105	Input transformer, 80G144
T-106	Plate transformer, 68G80
T-107	Reactor, 68G82
V-101	6X5GT/G
V-102	6X5GT/G
V-103	6SN7GT
V-104	6X5GT/G
V-105	6N7GT/G
V-106	6N7GT/G
V-107	6SN7GT
V-108	6SL7GT
V-109	6SL7GT
V-110	6L6GA
V-111	6L6GA
V-112	6L6GA
V-113	6L6GA
V-114	2B23
V-115	2B23
K-101	IN.C. -4 N.O. Catalog Number A-18391
K-102	IN.C. -2 N.O. Catalog Number A-19269

Parts List for Power Supply Unit

R-161	680,000 ohms	1/2 w
R-162	500,000 ohms	
R-163	4,700 ohms	2 w
R-164	39,000 ohms	
R-165	15,000 ohms	2 w*
R-166	1 megohm	
R-167	1 megohm	1/2 w
R-168	150,000 ohms	
R-169	150,000 ohms	
R-170	500 ohms	2 w*
R-171	50,000 ohms	2 w*
	50,000 ohms	
R-173	27,000 ohms	
R-174	27,000 ohms	1/2 w
R-175	68,000 ohms	
R-176A	39,000 ohms	
R-176B	39,000 ohms	2 w
R-177	1,200 ohms	
R-178	50,000 ohms	2 w*
R-179	47,000 ohms	1 w
R-180	10,000 ohms	1/2 w
R-184	6.8 megohms	2 w
R-185	6.8 megohms	
R-261	680,000 ohms	1/2 w
R-262	500,000 ohms	
R-263	4,700 ohms	2 w
R-264	39,000 ohms	
R-265	15,000 ohms	2 w*
R-266	1 megohm	
R-267	1 megohm	1/2 w
R-268	150,000 ohms	
R-269	150,000 ohms	
R-270	500 ohms	
R-271	50,000 ohms	2 w*
	50,000 ohms	
R-273	20,000 ohms	
R-274	20,000 ohms	1/2 w
R-275	50,000 ohms	
R-276	75,000 ohms	2 w
R-277	6,800 ohms	
R-278	50,000 ohms	2 w*
R-279	47,000 ohms	1 w
R-280	10,000 ohms	1/2 w
R-281	1,000 ohms	1 w
R-282A	8,200 ohms	
R-282B	8,200 ohms	2 w
R-282C	8,200 ohms	
R-283	2,500 ohms	1/2 w

*Potentiometer

C-161	0.05 µf	600 v
C-162	1.0 µf	400 v
C-163	0.25 µf	
C-164	0.25 µf	
C-165	0.1-0.1 µf	600 v
C-167	0.05 µf	
C-168	10.0 µf	
C-169	1.0 µf	400 v
C-170	0.01 µf	300 v
C-261	0.05 µf	600 v
C-262	1.0 µf	400 v
C-263	0.25 µf	
C-264	0.25 µf	
C-265	0.1-0.1 µf	600 v
C-267	0.05 µf	
C-268	10.0 µf	
C-269	1.0 µf	400 v
C-270	.01 µf	300 v

T-161	Filament, Catalog Number 80G141
T-162	Grid, 80G146
T-163	Plate, 68G80
T-164	Plate, 80G173
T-165	Reactor, 68G818
T-261	Filament, 80G141
T-262	Grid, 80G146
T-263	Plate, 68G80
T-264	Plate, 80G140
T-265	Reactor, 68G82

V-161	6SN7GT
V-162	6N7GT/G
V-163	6N7GT/G
V-164	5U4G
V-261	6SN7GT
V-262	6N7GT/G
V-263	6N7GT/G
V-264	5U4G

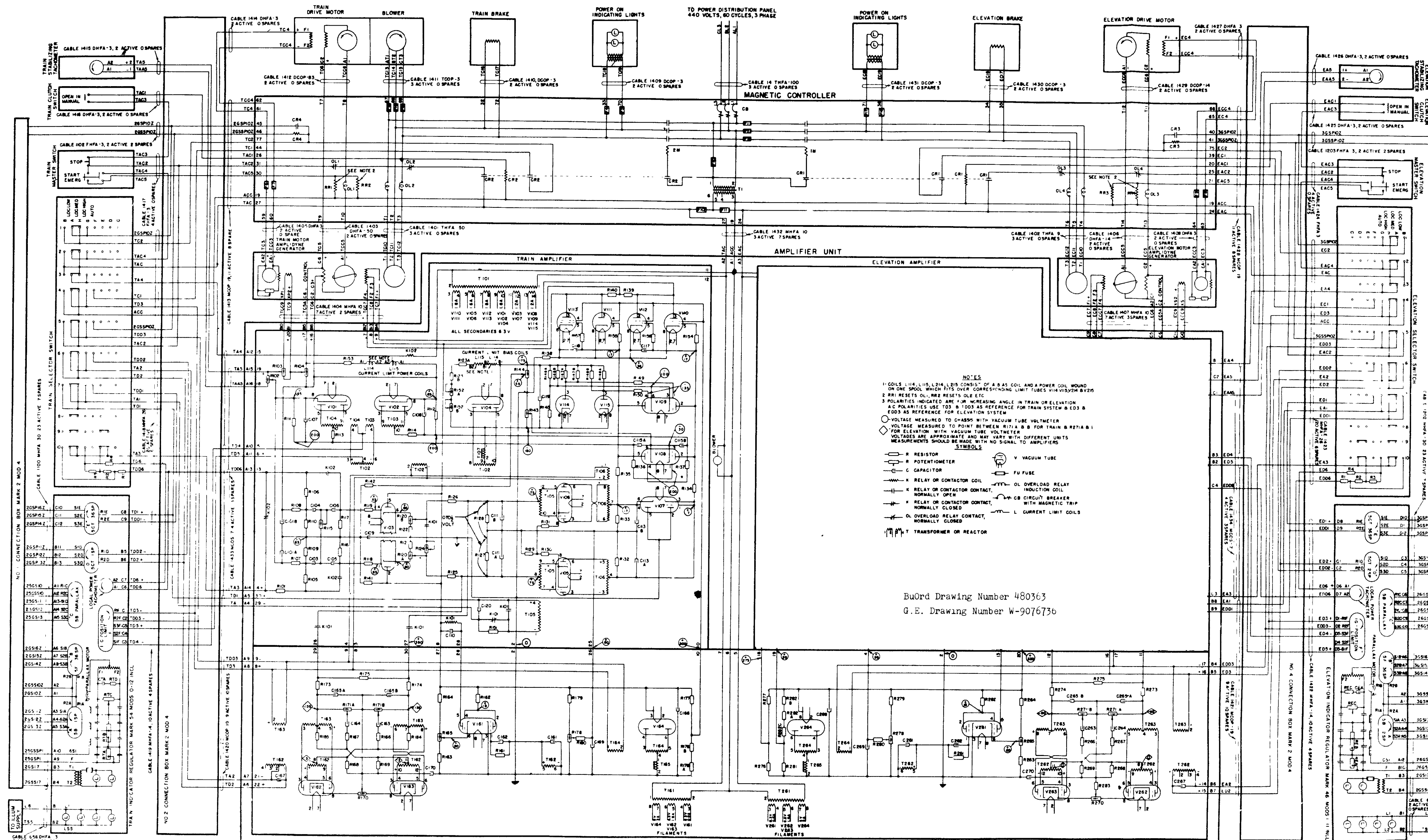

Figure 28 - Detailed Wiring Diagram of the Amplidyne Units

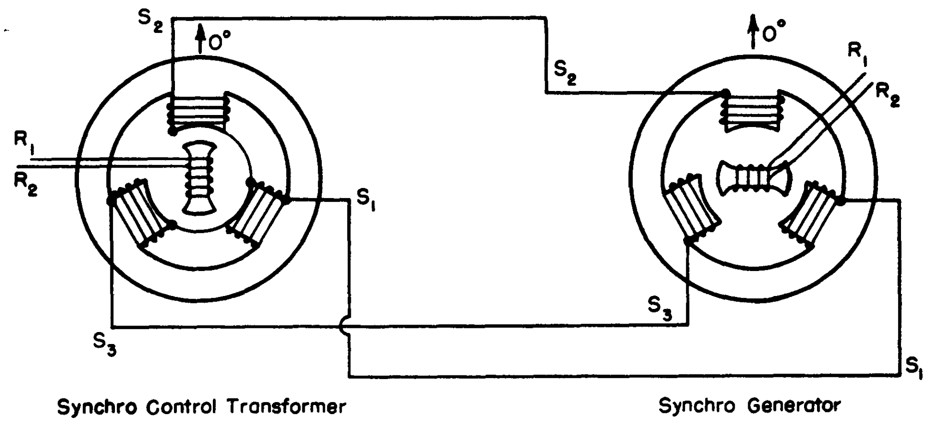

Figure 29 - Schematic Diagram of a Synchro Control Transformer and a Synchro Generator

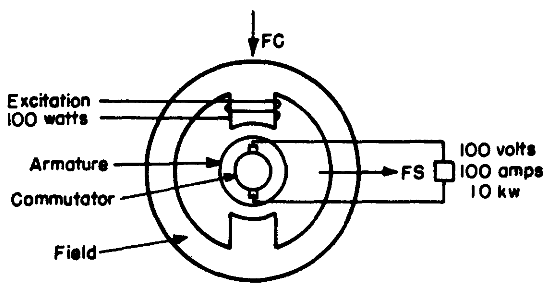

Figure 30a - Ordinary D-C Generator

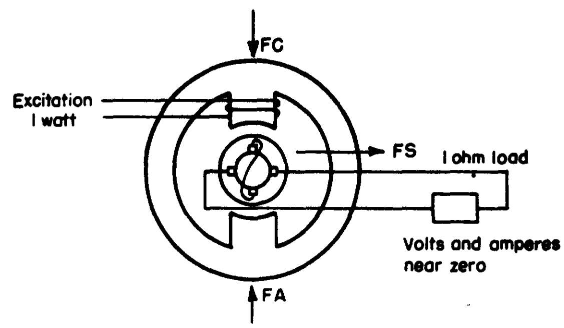

Figure 30b - Brushes Short-Circuited and Excitation Reduced

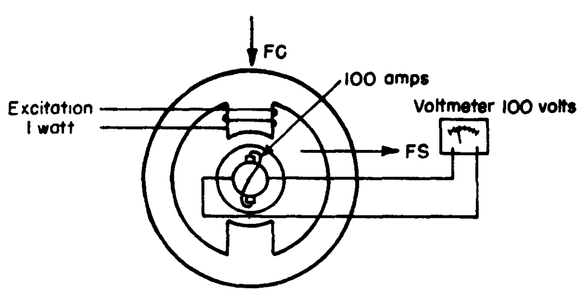

Figure 30c - Load Connected to New Brushes

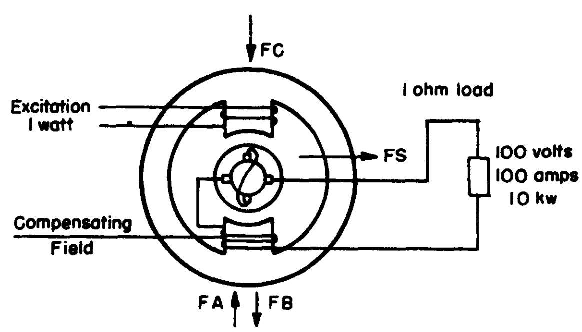

Figure 30d - Compensating Field Added

Figure 30 - Transformation of an Ordinary D-C Generator into an Amplidyne Generator

Impellers and Mechanisms

The two impellers or pumping units, the d-c impeller motors, reduction-gear boxes, impeller and pitch shafts, pitch linkage, and impeller casings or diffuser sections were the same units used previously in the HAMILTON installation. Certain alterations had to be made: The shafts had to be cut to a length to suit the PEREGRINE installation, the parts and pistons used for hydraulic operation of blade pitch on the HAMILTON installation were removed, and the end of the pitch shaft was fitted with a worm gear to be driven by the remainder of the mechanism necessary to change pitch.

The entire pitch and impeller drive has been shown generally in the over-all diagram of the pumping machinery, Figure 24. A diagram of the pitch-control adapter assembly from the power drive to the pitch shaft is shown in section in Figure 31a, and in the photographs of Figure 31b. The arrangement of the clutch-shifting mechanism between the d-c motor clutch and the adapter assembly is shown in Figures 31c and 31d. The pitch mechanism has been designed so that, for appreciable revolutions of the d-c drive motor with power supply from the amplidyne motor-generator set, rapid changes of pitch could occur from 0° to 30° in 3/4 second. The mechanism essentially is a speed reducer and changes rotational motion to reciprocating motion. The clutch mechanism was designed so that mechanically a shift could be made from the d-c motor drive to manual drive. This permitted checking of the mechanism and helped to simplify certain static calibrations of devices for indicating and recording before all the quirks were eliminated on the over-all operations of the stabilizing equipment. These devices also helped in finding these quirks.

The d-c impeller motor runs at from 1900 to 1600 rpm depending on percentage load and voltage variation. It is a 25-hp motor fitted with a cast-steel flywheel weighing 795 pounds (Figure 32). The purpose of the flywheel is to conserve the size of the unit necessary since the load on the impeller will vary more or less with pitch angle, and the pitch will change in a sinusoidal manner. The impeller units were designed and built by the I.P. Morris Division of the Baldwin-Southwark Corporation, which has built many variable-pitch Kaplan turbines for moderate and low-head hydroelectric installations. Power for the port and starboard d-c impeller motors was furnisned by the motor-generator set installed in the motor-generator room on the boat deck aft of the control room. Herringbone gears with the pinion driven by the motor shaft are used to reduce the rpm to that desired for pumping water (Figure 33). The pinion has 14 teeth and the bull gear 190 teeth—giving a speed reduction of 190:14. The large herringbone gear rotates the outer hollow impeller shaft which turns the blades continuously. The pitch shaft is driven by the follow-up motor through the pitch mechanism. The shaft motion is reciprocal; the

(Text continued on page 83.)

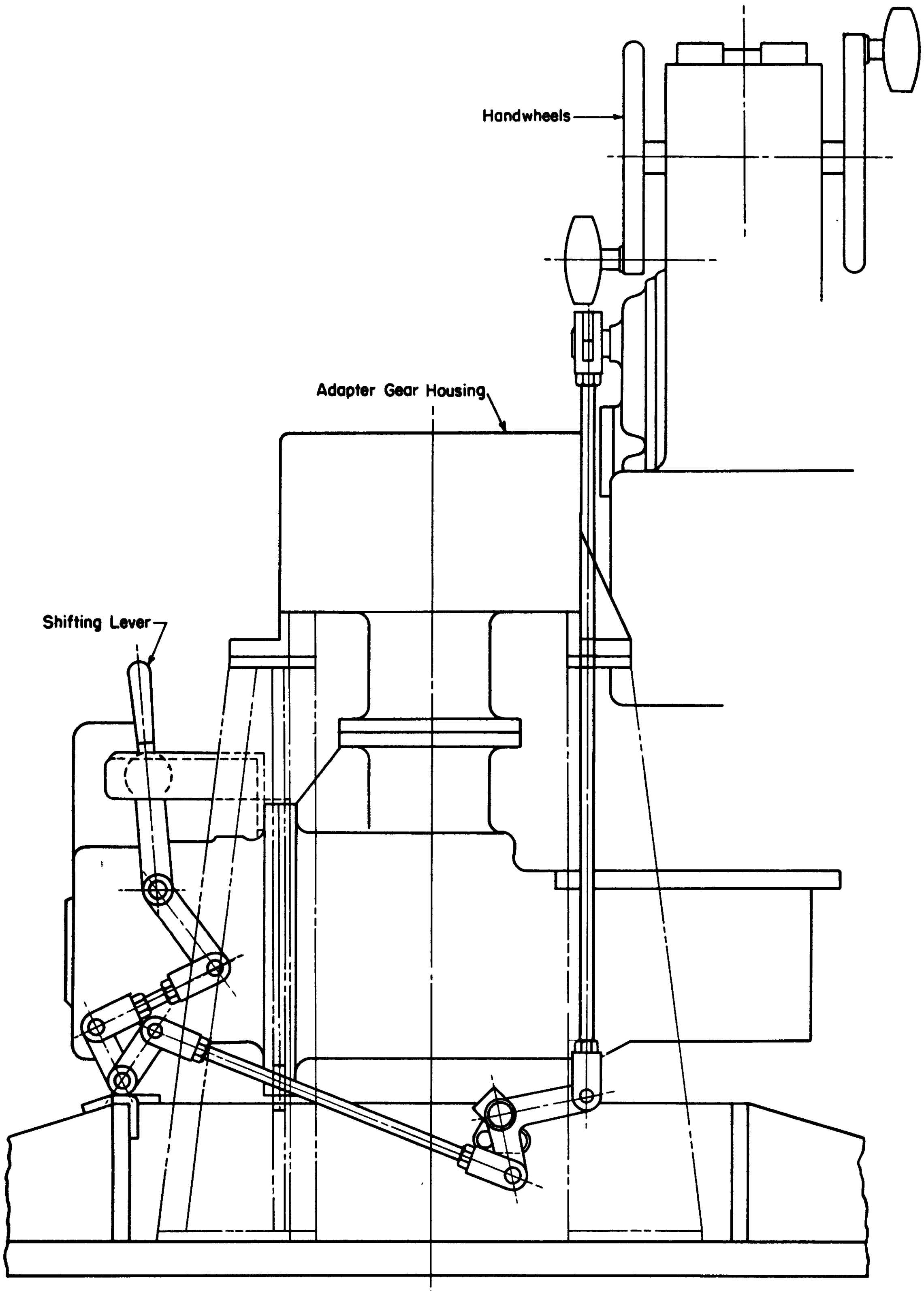

Figure 31a - Diagram of the Pitch Control Adapter Assembly between the Power Drive and the Pitch Shaft

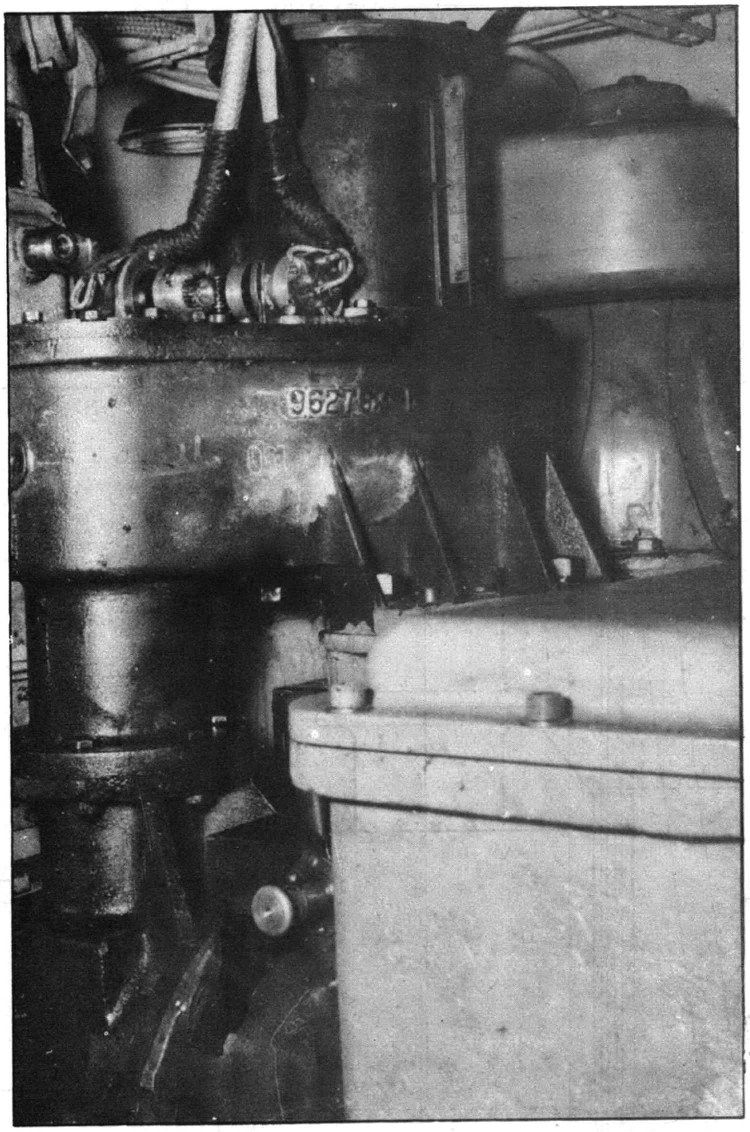

Figure 31b - Photograph of the Pitch Control Adapter Assembly

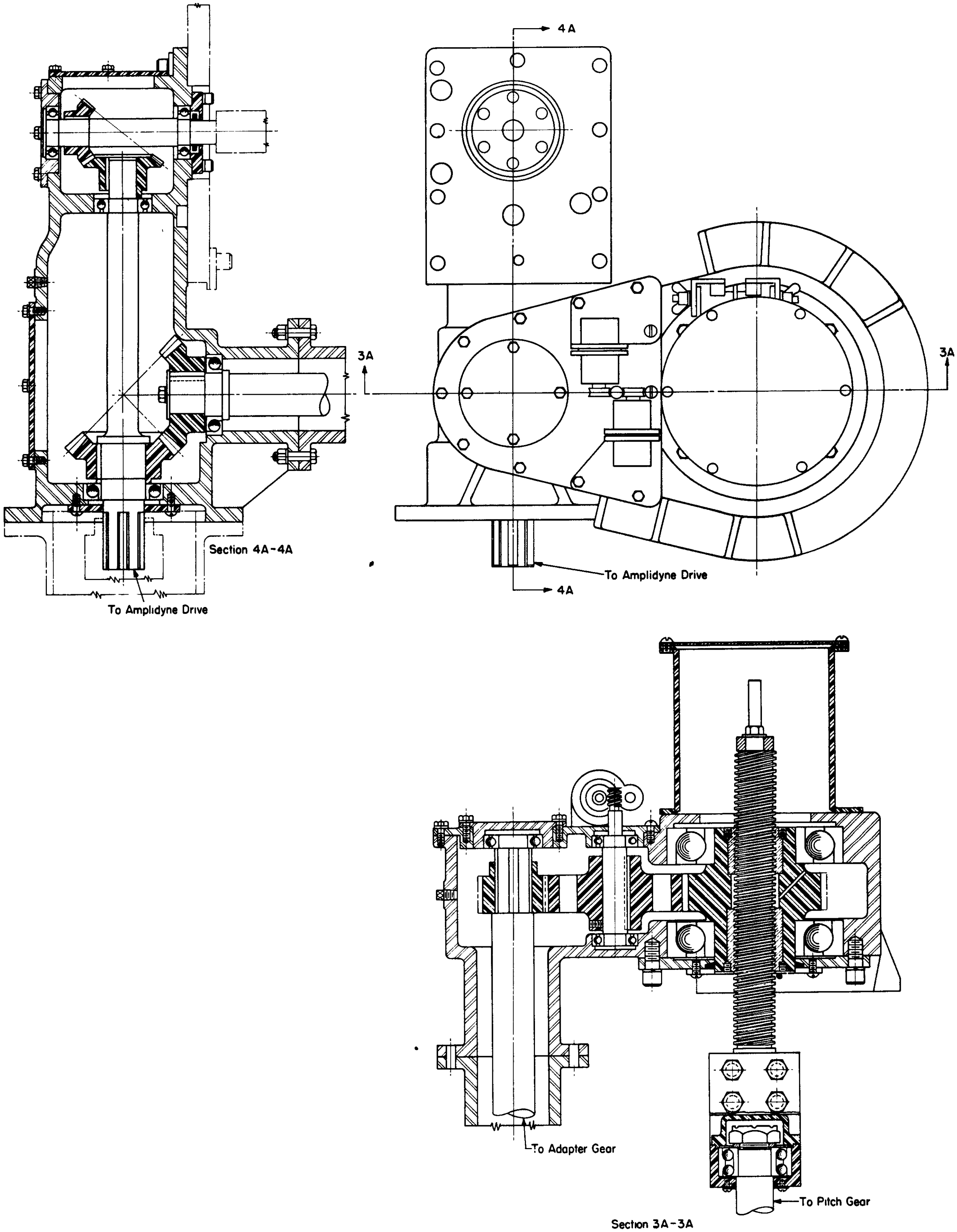

Figure 31c - Diagram of the Clutch Shifting Arrangement between the Amplidyne D-C Motor Clutch and the Pitch Mechanism

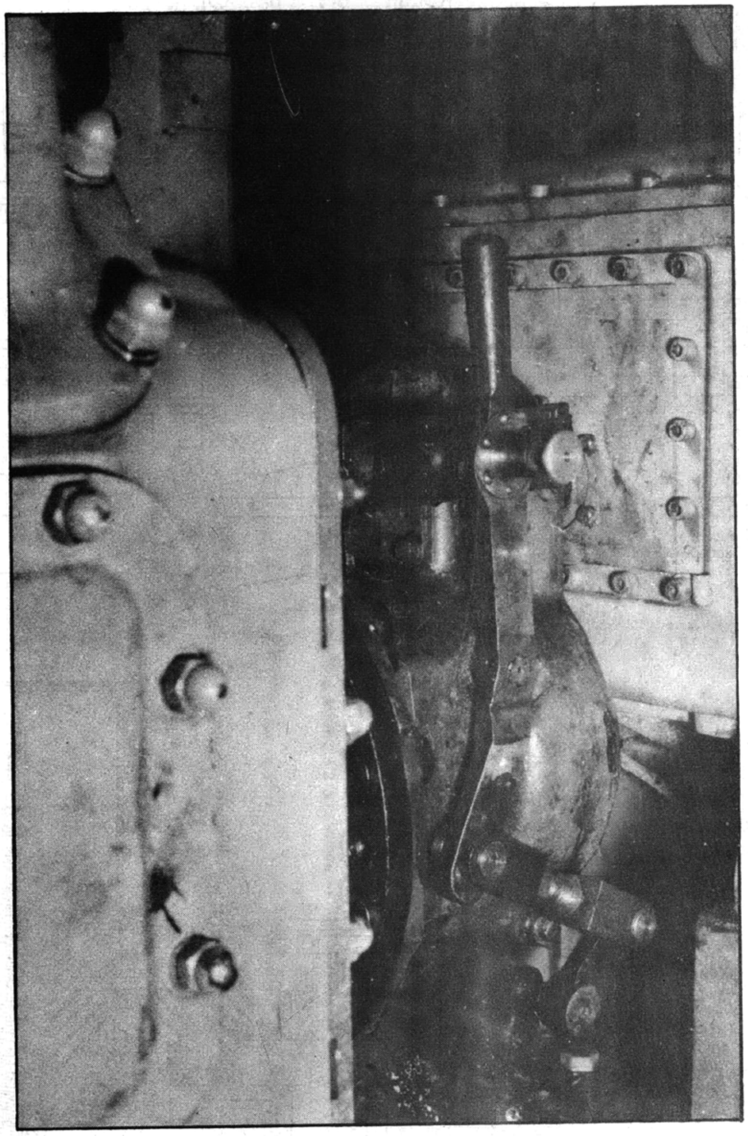

Figure 31d - Photograph of the Clutch Shifting Arrangement

Figure 31 - Diagrams and Photographs of the Pitch Mechanisms

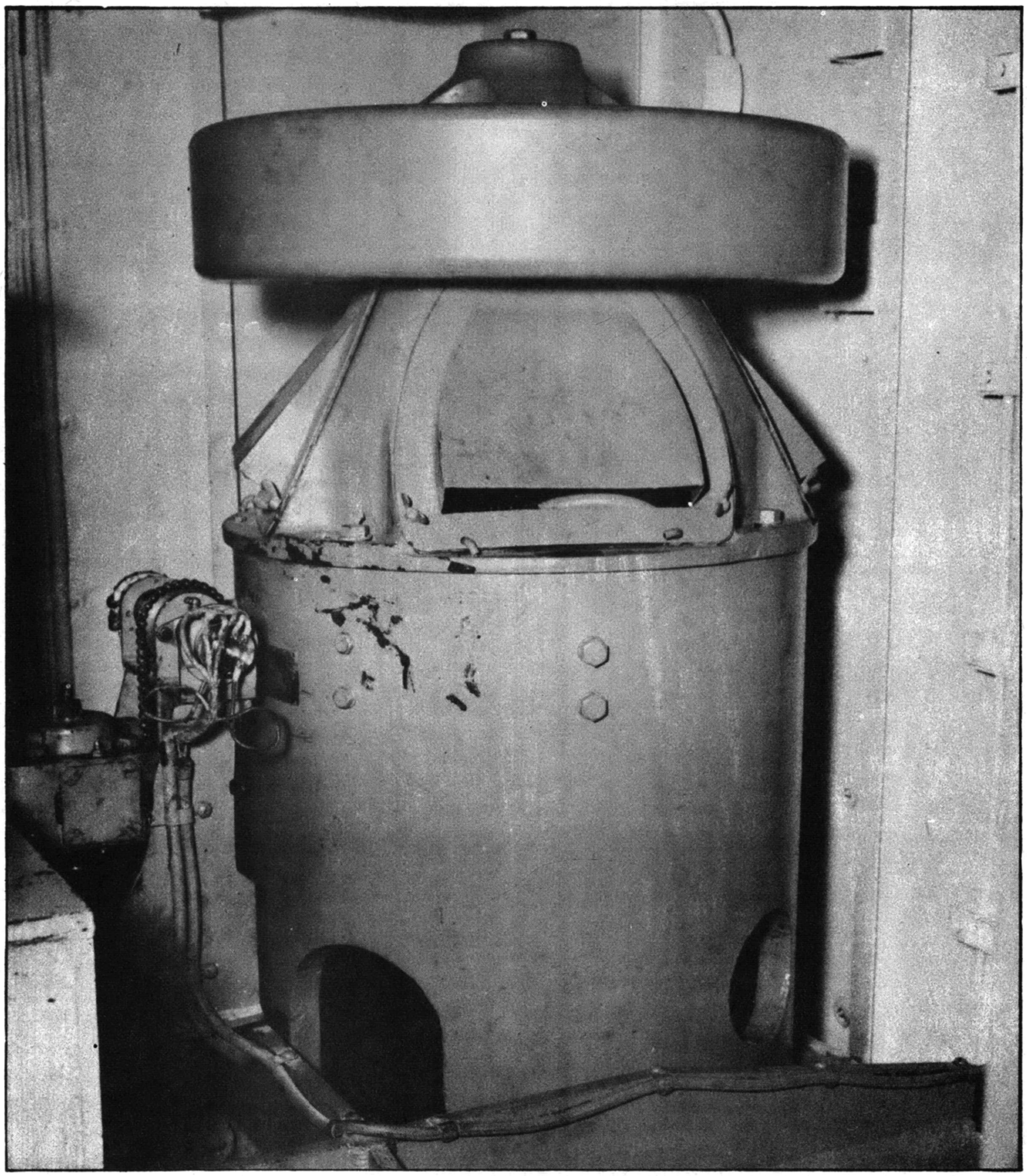

Figure 32 - Photograph of Impeller Motor and Flywheel

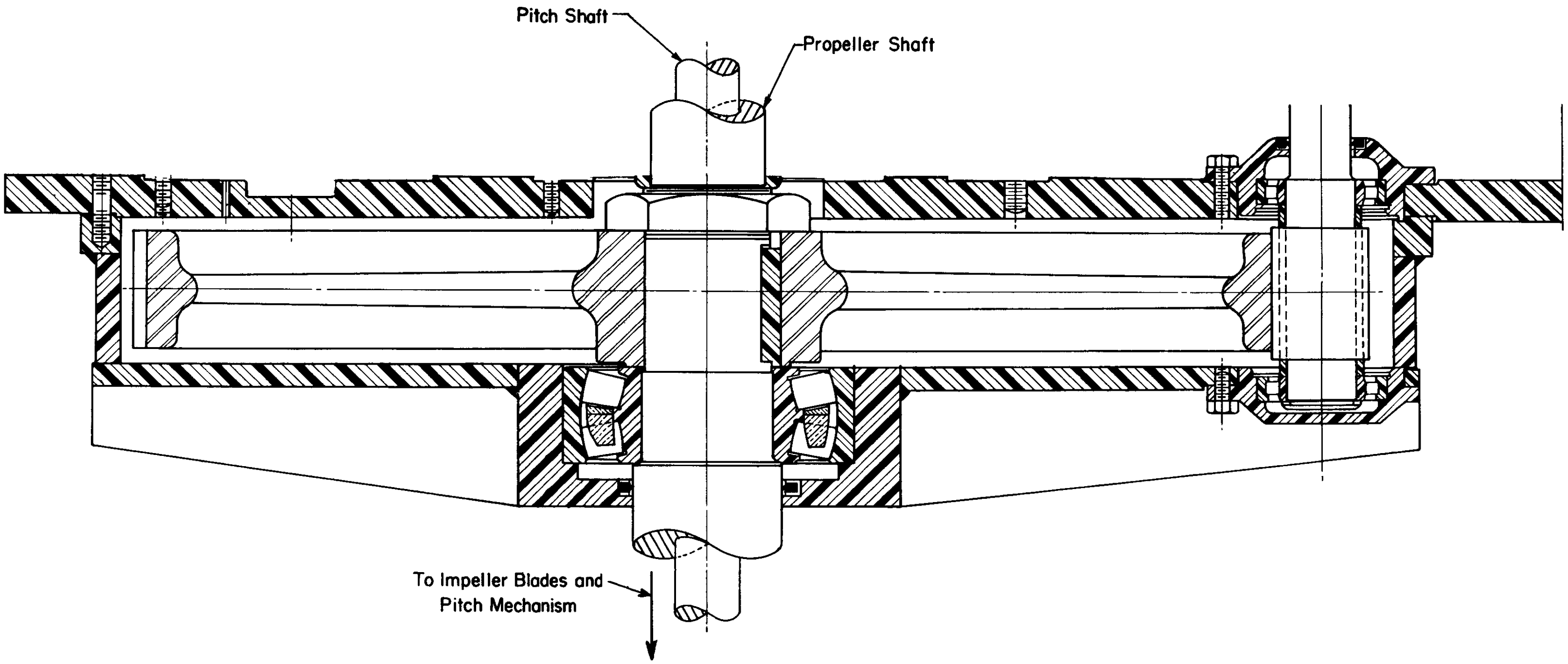

Figure 33 - Reduction Gearing of the Impellers

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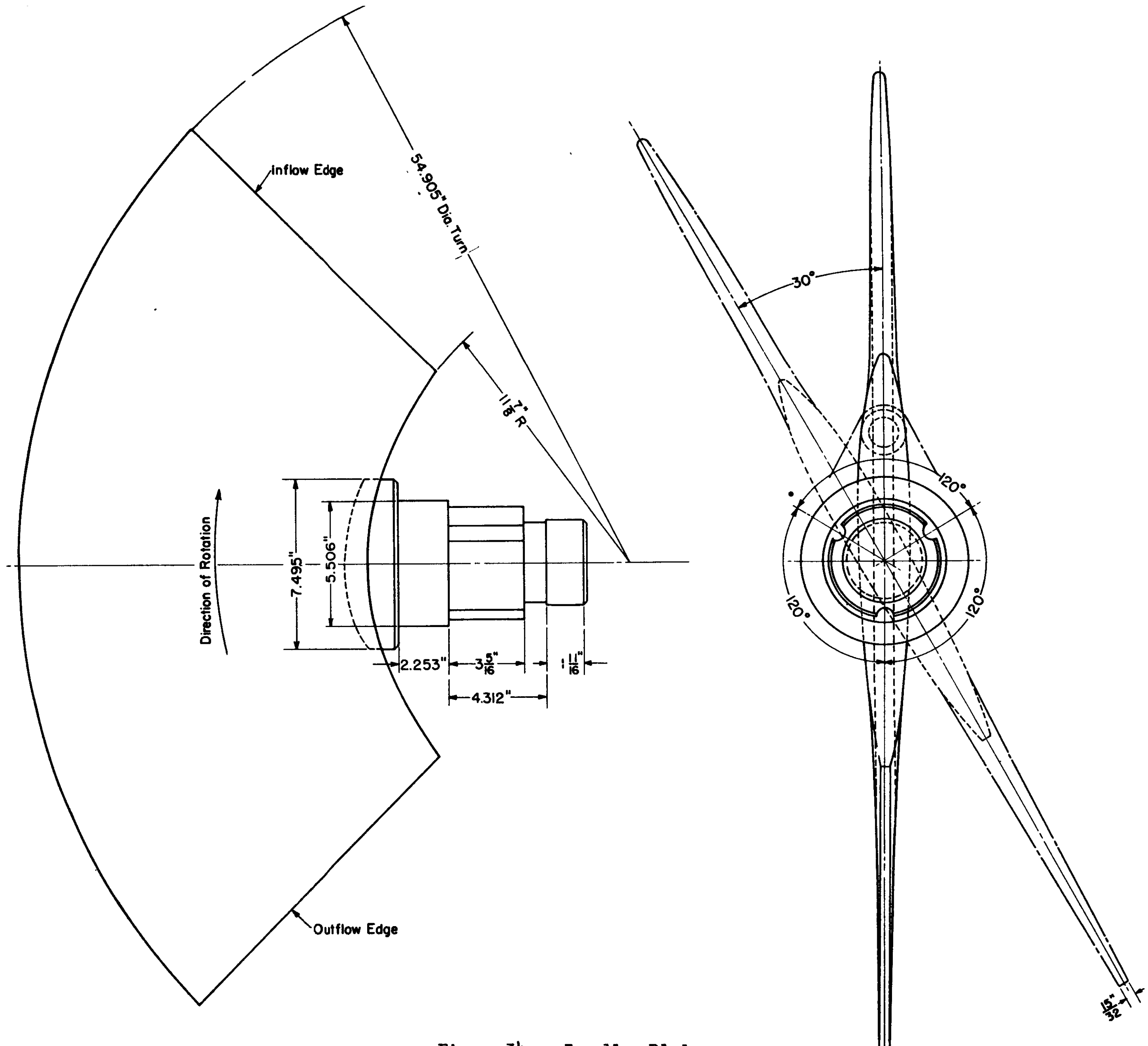

Figure 34a - Impeller Blades

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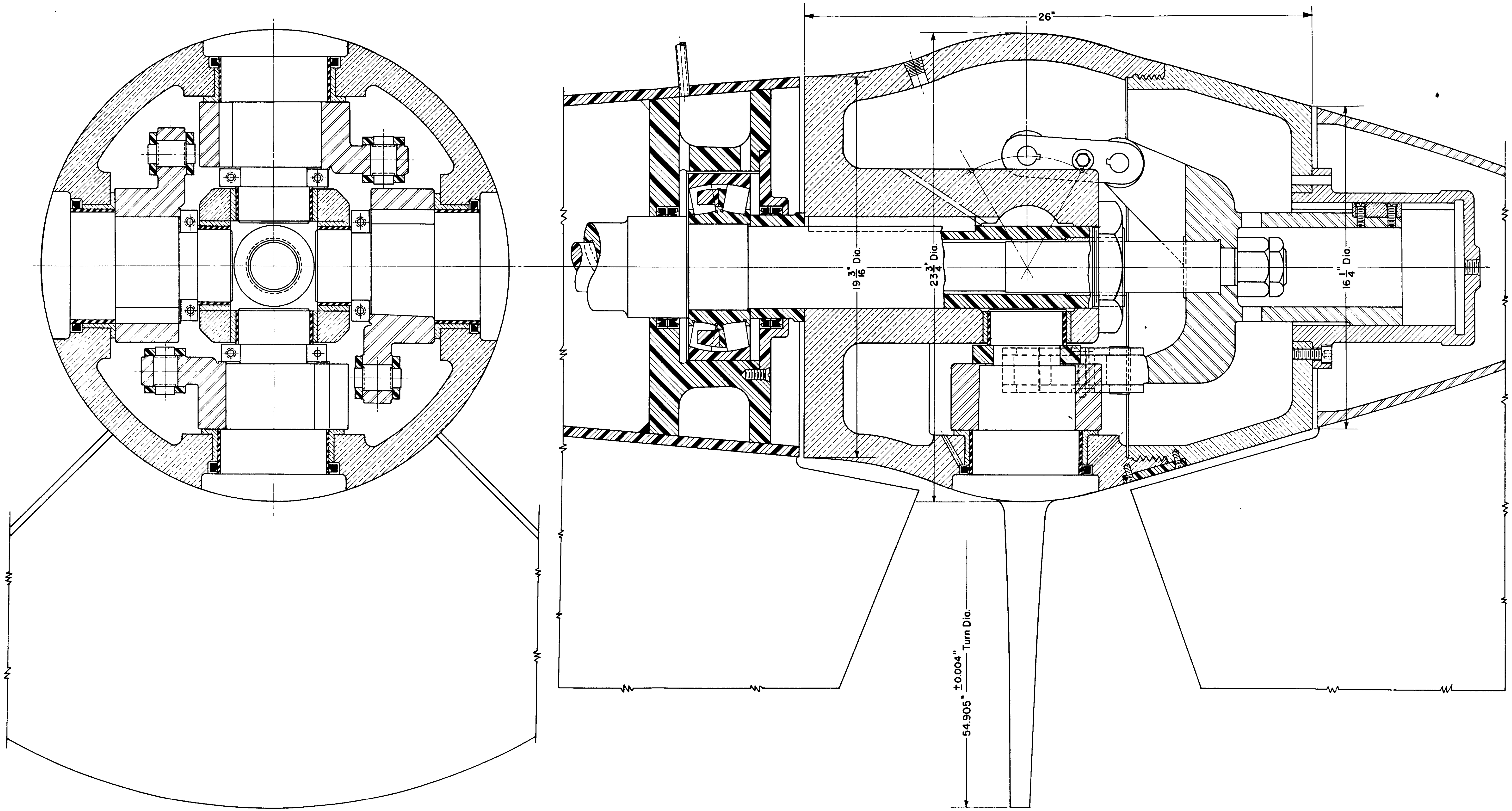


Figure 34b - Arrangement of Hub Mechanism and Lower Bearing

Figure 34 - Connection of the Pitch Mechanism to the Impeller Blades

lower end of the pitch shaft in the impeller hub is connected by a yoke and linkages, Figure 34b, to the bosses at the roots of the four blades and by means of the linkages changes the pitch of the blades from 0° to ±30°. The impeller hub is packed with grease for lubrication. The blade including its hub, one of four in each impeller, is shown in Figure 34a.

Furnished with the impellers were upper and lower diffusion sections with five splitters for the four-bladed impellers in order to smooth out the water flowing either way. The impellers were located somewhat below the tanks, with the top of the upper diffuser section flush with the tank bottom. The impeller for the forward pumping system was located on the starboard side and that for the aft system was located on the port side.

Tanks and Ducts

Two systems were installed on board the PEREGRINE, one immediately aft of the other so that the tanks on each side had a common transverse bulkhead; see Figure 35. Each system includes two tanks, one on the port side and one on the starboard side. The impeller was installed beneath one tank of the system with the top of its upper diffusion section, Figure 37a, flush with the bottom of the tank. An elbow of circular cross section connected the lower diffuser section, Figure 37b, of the impeller with a transitional portion to the rectangular section of the crossover duct at the hull. This elbow was supplied with splitters, see Figure 38a, to distribute the load as the direction of flow of water was changed by the elbow. The rectangular crossover duct, which ran horizontally from port to starboard hull surface, was 2.75 ft by 4 ft with a cross-sectional area of 11 square feet except where a butterfly valve was installed; there the cross-sectional area was 9.4 square feet. The duct was smoothly lined and was 32 ft long from hull surface to hull surface. The butterfly valve was installed to cut down the free surface effect when the system was not operating. The other end of the crossover duct was connected to the bottom of the other tank by an elbow of rectangular cross section, also divided into three parts by splitters as shown in Figure 38b. The two tanks were also connected by an 8-in. air line running across the ship over the top of the control room. Its purpose was to prevent compression of air as water rose in a tank, since compression of the air would prevent the water from rising to its proper height thus reducing the effective stabilization. A schematic diagram of the tank and duct arrangement is shown in Figure 36, together with pertinent dimensions and characteristics.

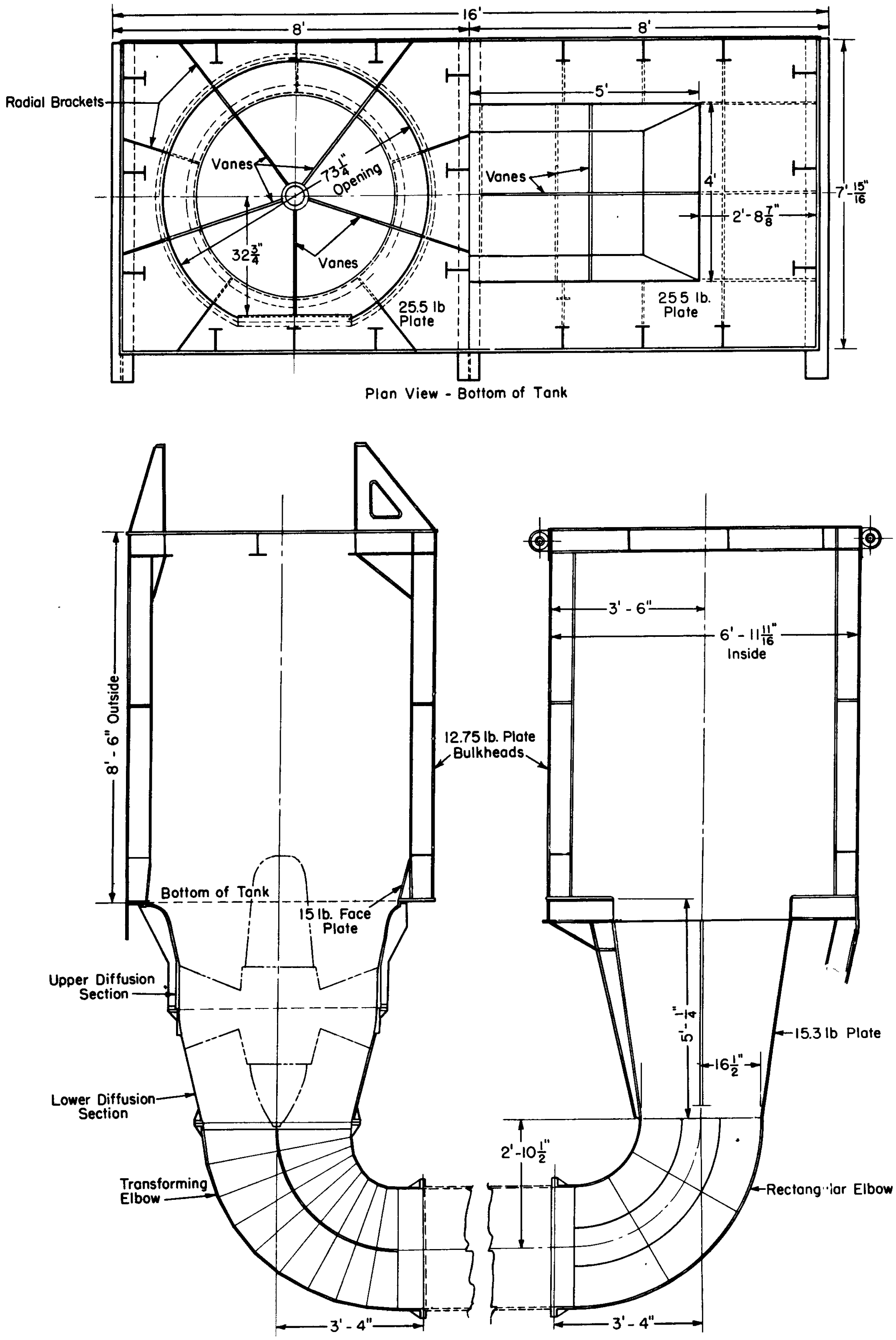


Figure 35 - Tank Arrangement and Construction

One tank of each system was provided with a float guided by vertical T-section rails; see the arrangement in Figure 39. At first the float was connected to a potentiometer, but later this was discarded owing to poor, jerky, lagging operation; the reduction-gear unit and synchro which were substituted operated satisfactorily. The signal from the synchro served two purposes: One, as a signal to a string oscillograph for continuous recording of water level and two, as a feedback signal to the modulator unit in order to tend to keep the water oscillating about the midheight of the tank.

The tanks, elbows, and motor rooms over the tanks, were supported on brackets secured to the hull transverse members. The brackets had to carry the dead load of all the structure, as well as the dynamic reactions of the pumping equipment (pitch and impeller mechanism) and the reactions of the elbows. The tanks had very heavy vertical stiffeners in order to carry the

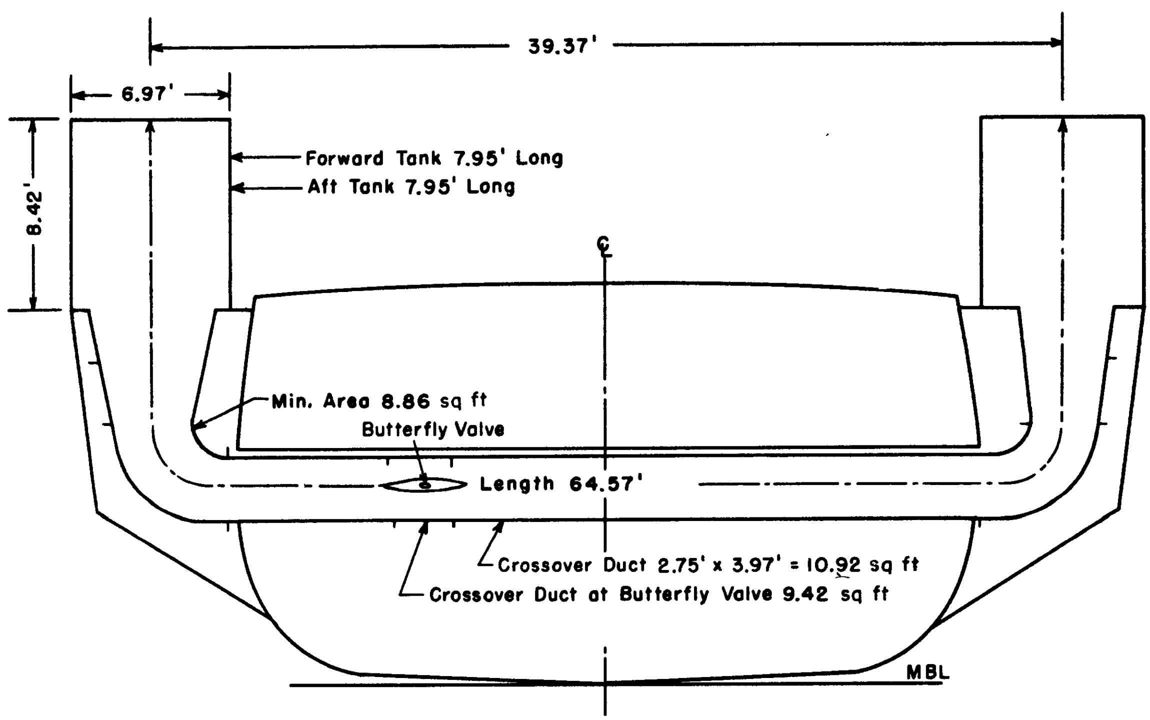


Figure 36 - Schematic Arrangement of Tank and Ducts

Total volume of water in forward and aft systems all tanks half full	2,138.8 cu. ft.
Total volume of forward system	1,534.2 cu. ft.
Combined area of forward port and starboard tanks (0.75 percent correction for stiffeners)	110.4 sq. ft.
Total weight of water	59.5 tons
$\frac{59.5}{1164 \text{ (normal } \Delta)}$	5.11 percent
Maximum static moment	509 ft. tons

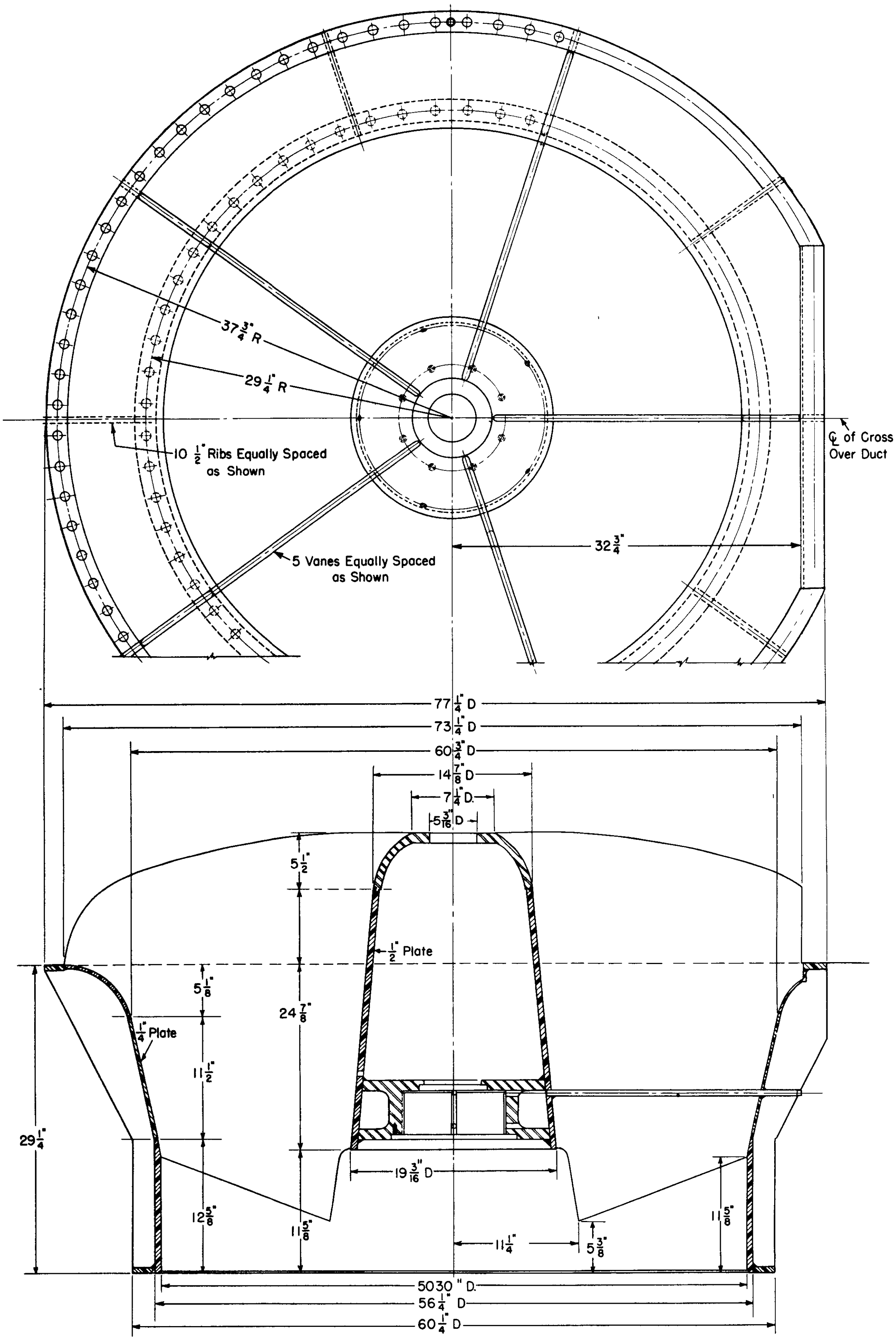


Figure 37a - Upper Diffuser Section

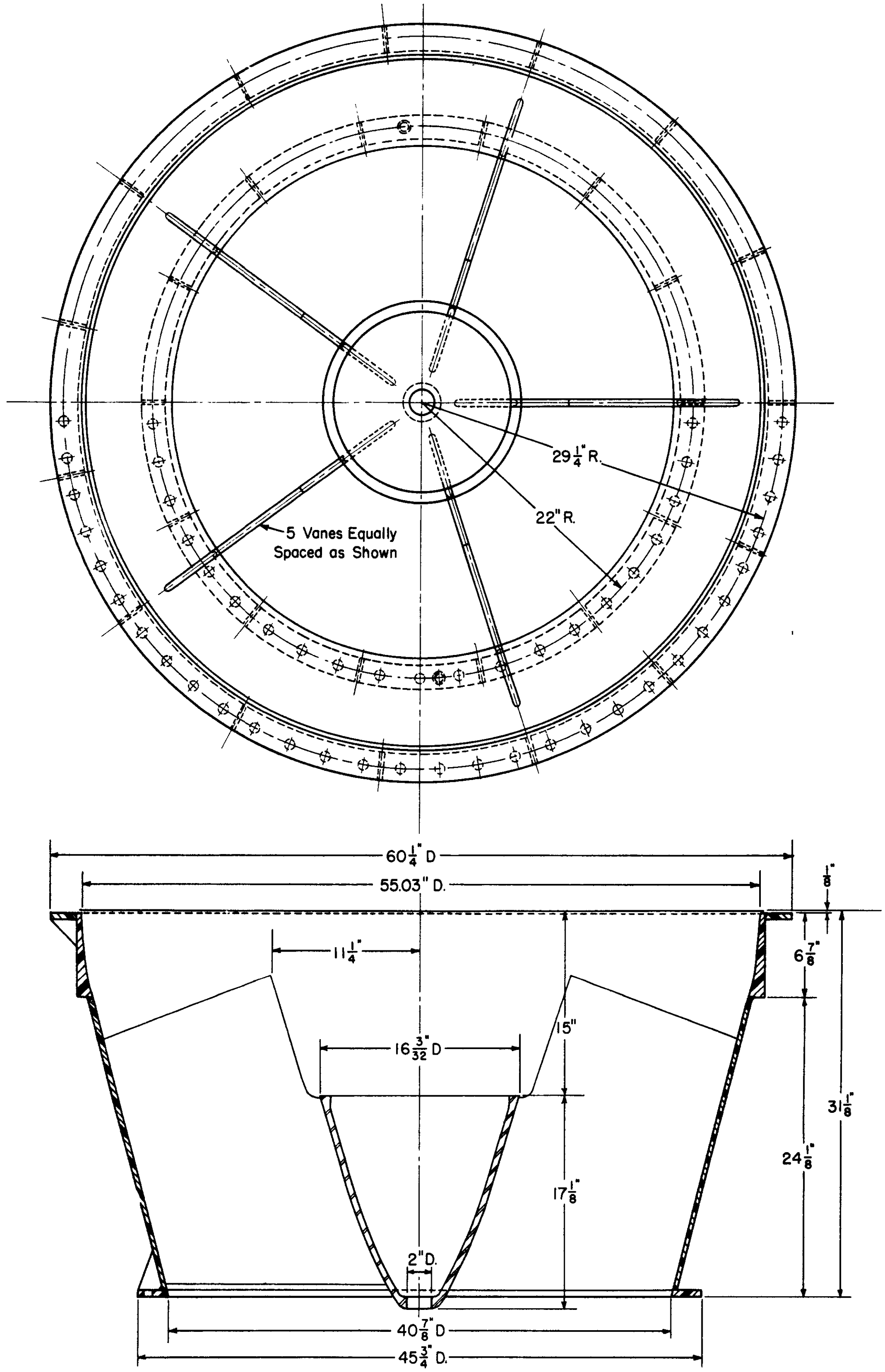


Figure 37b - Lower Diffuser Section

Figure 37 - Impeller Diffuser Sections

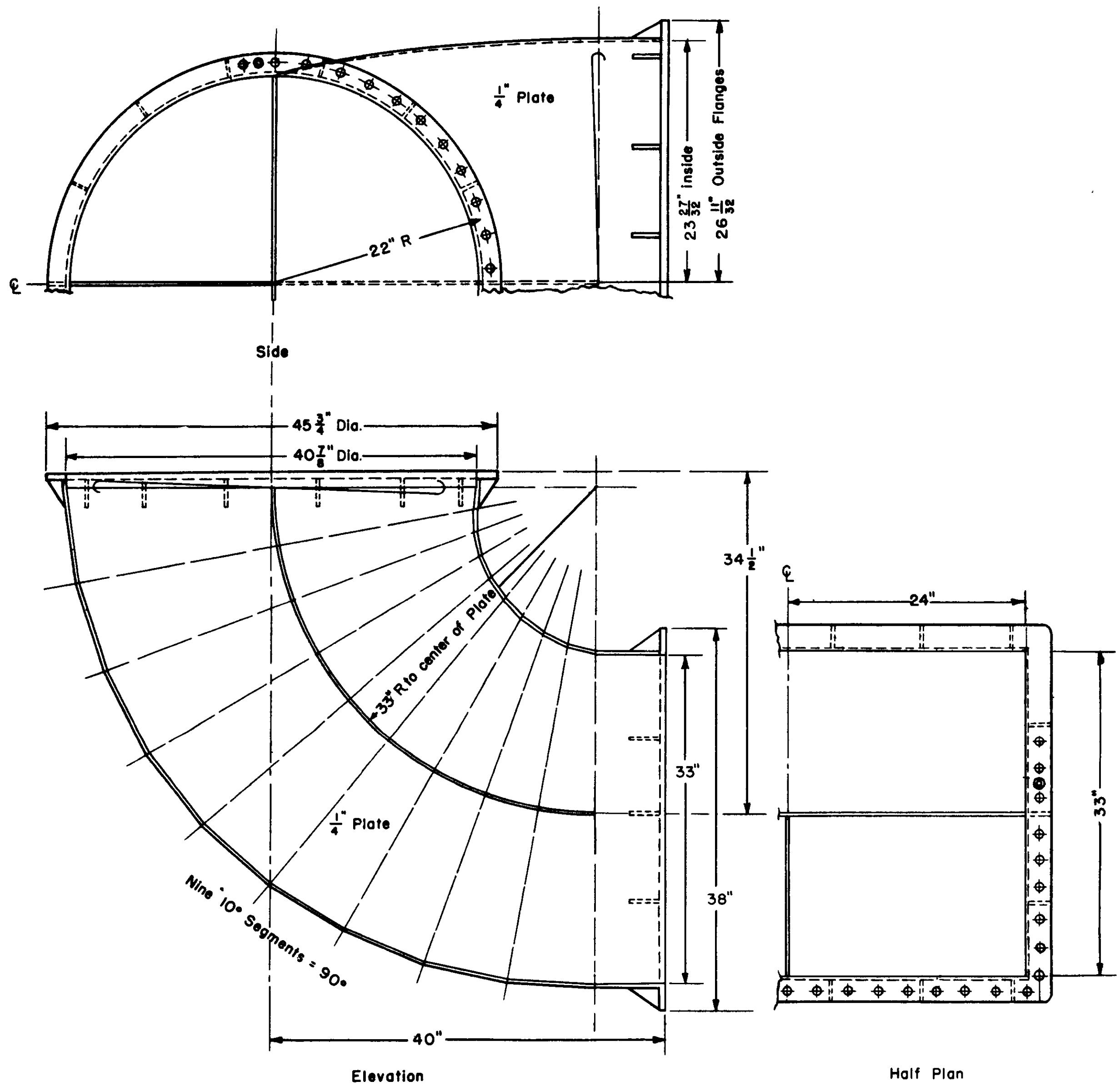


Figure 38a - Elbow of Circular Section

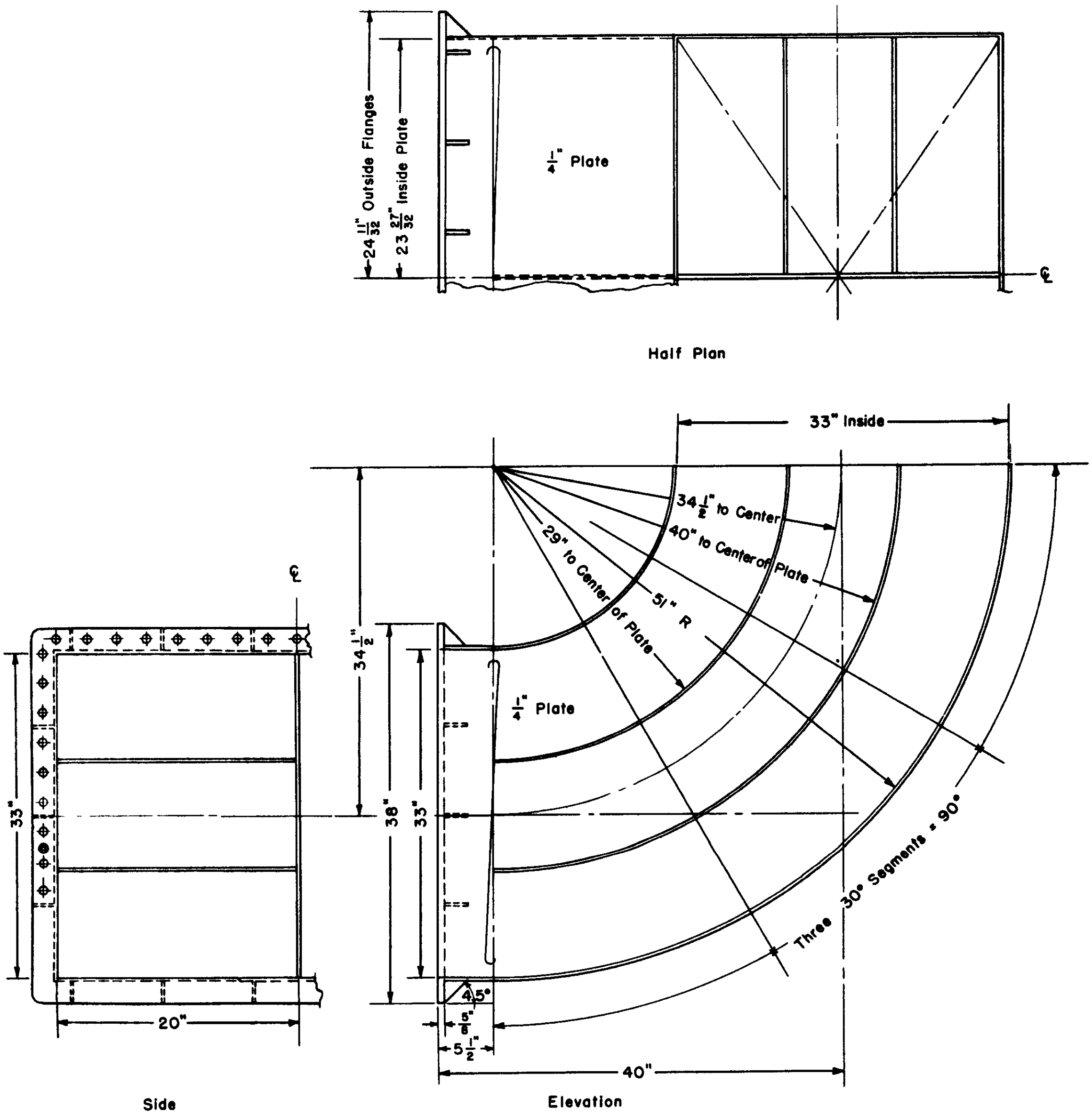


Figure 38b - Elbow of Rectangular Section

Figure 38 - Elbows and Splitters

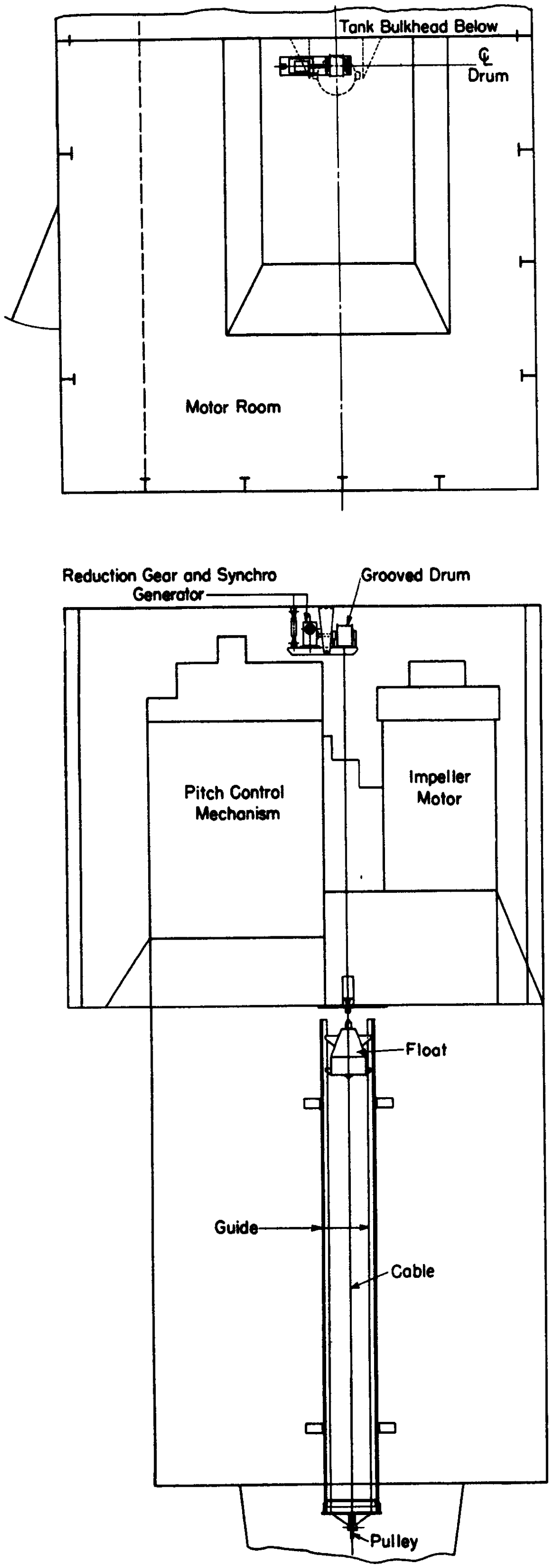


Figure 39 - Float Arrangement

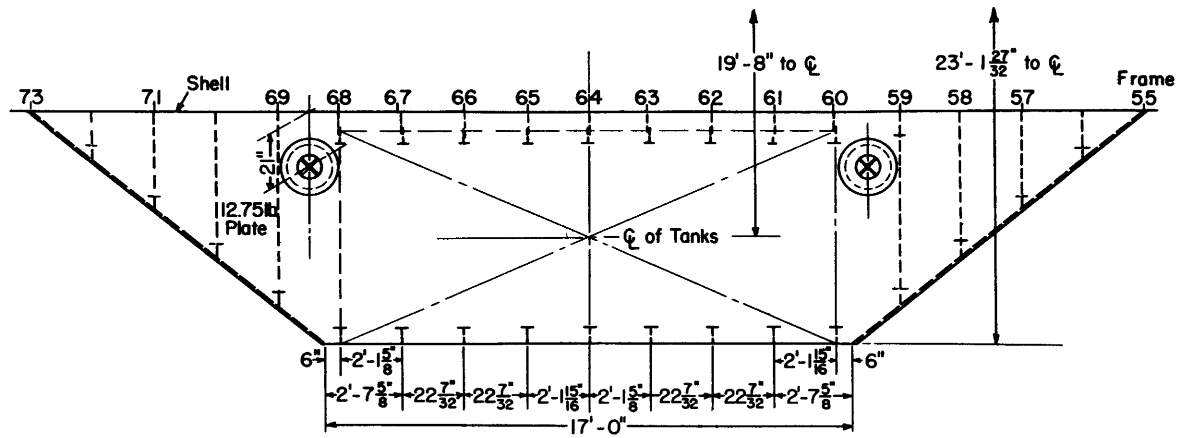
motor-room loads down to the brackets. The elbows were covered with sponsons tapered downward and fore-and-aft to reduce somewhat the resistance of the sponsons when the ship was under way, Figure 40.

While the stabilizing system was operating, when everything was performing properly, there was no noticeable vibration in way of the tanks or ducts in spite of rather rapid changes of impeller pitch at times. There was only one point where noise (no vibration) occurred, and it was localized to an area of a few square feet. This was caused by the fairing in one duct being improperly secured at one point so that, when water was being transferred back and forth, the fairing plate panted and banged against a stiffener.

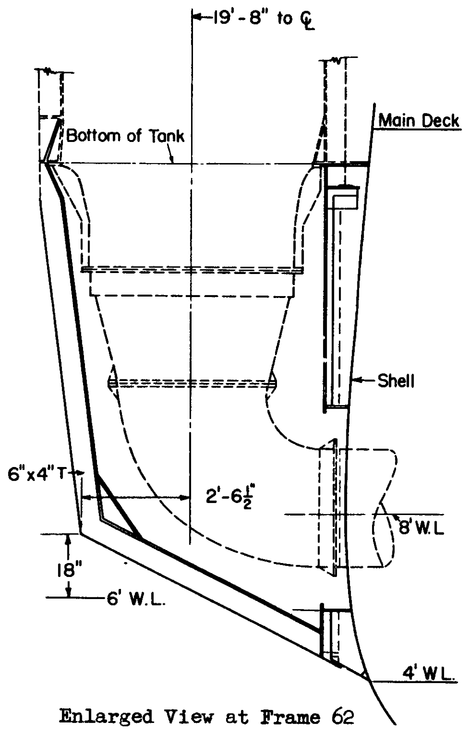
ELECTRICAL SUPPLY

A variety of electric-power installations were needed for the operation of instruments used in testing, calibrating, and obtaining data during stabilizing. For operation of the control equipment well-regulated 115-v d-c, 115-v a-c, and 400-cps a-c supplies were needed. For the pumping machinery 440-v, 3-phase a-c power and 115-v d-c power were required, and for instruments 115-v a-c power was needed. Power had to be distributed to the starboard and port motor rooms, the motor-generator room, and the control

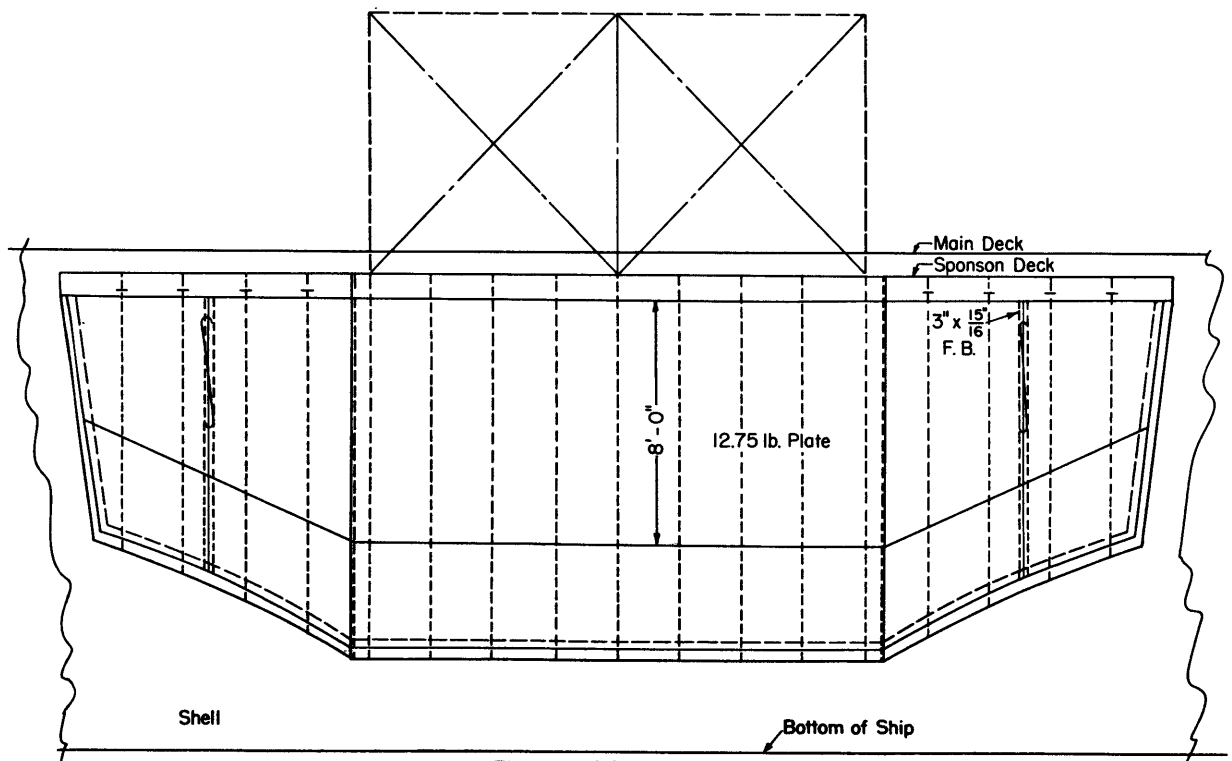
Figure 40 - Sponsons



Plan of Sponson



Enlarged View at Frame 62



Elevation of Sponson

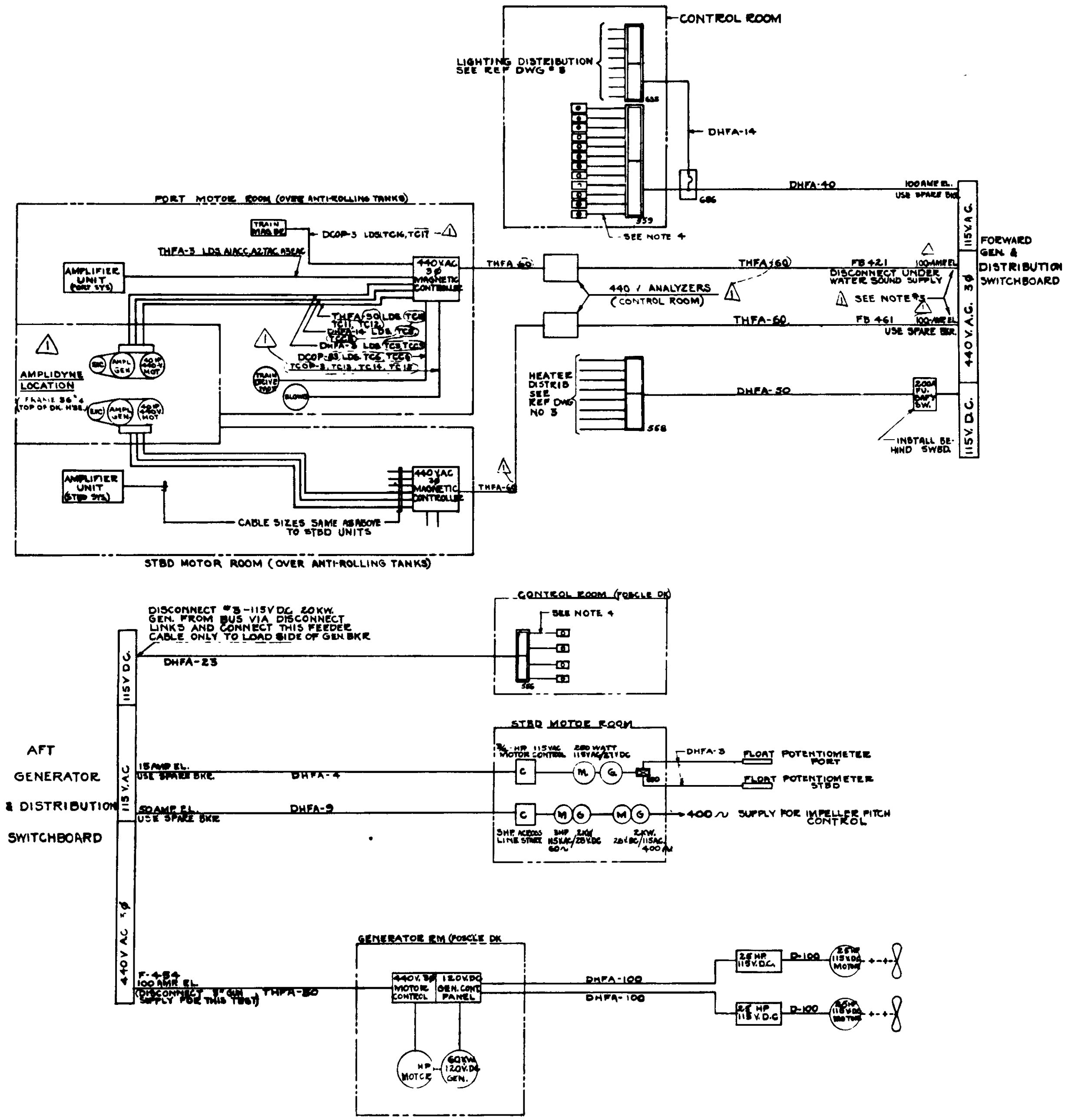


Figure 41 - Schematic Diagram of Power Distribution for the Ship-Stabilizing System

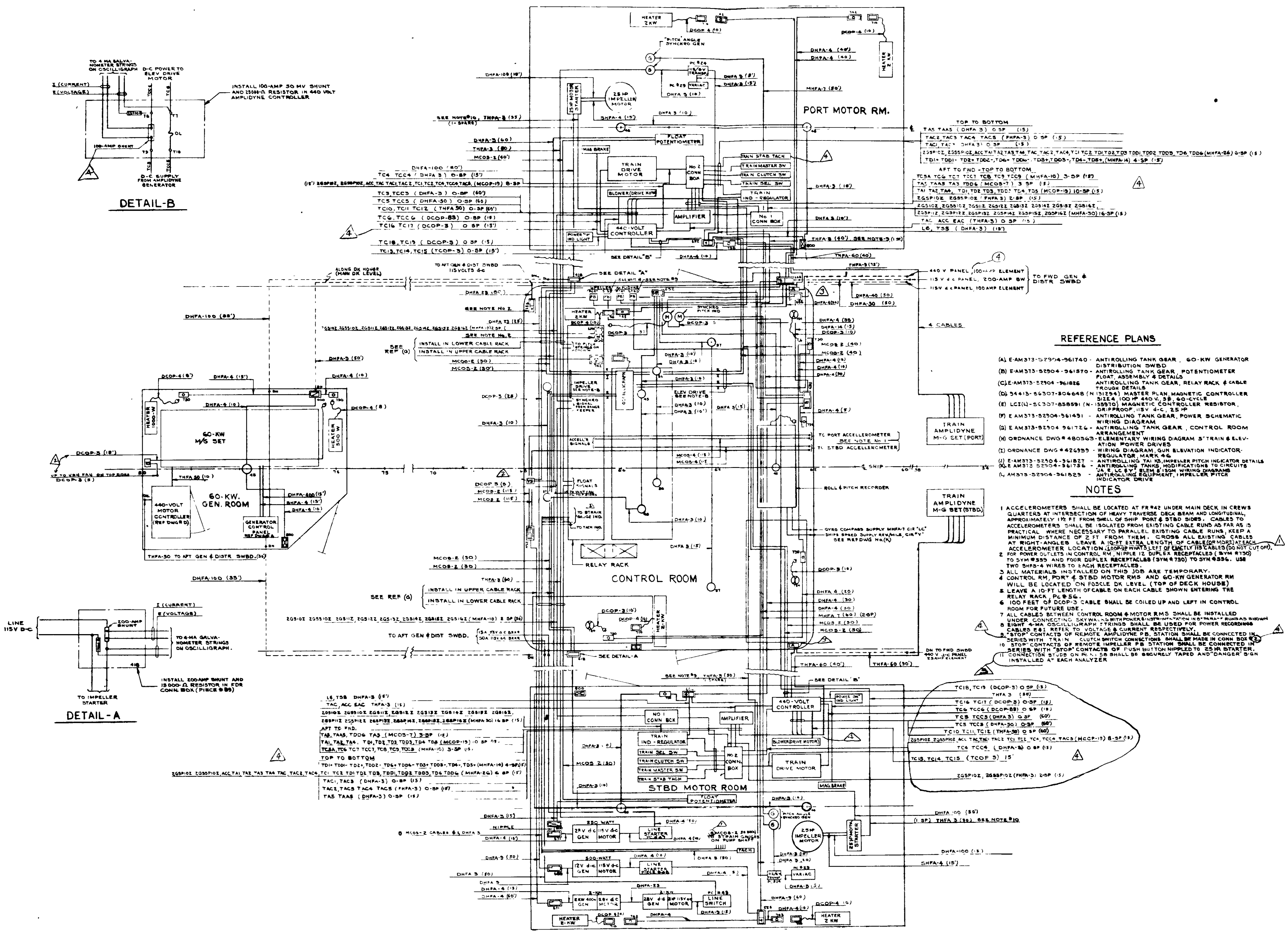


Figure 42 - Electric Cable Layout for Power, Lighting and Instrumentation Used with the Ship-Stabilization System

room. Power was also needed in these compartments for heaters, blowers, and lights.

The power supplied for the stabilization equipment had to be obtained from the ship in such a way that the a-c and d-c power for the control system and for the instruments was well regulated to eliminate drifting caused by voltage fluctuation. Otherwise calibrations and diagnosis of troubles would have been well-nigh impossible. The power had to be furnished by a number of sources of ship supply so that there would be no overloads, whether in starting or in protracted operation.

Power was obtained through the ship's forward generator and distribution board in the forward engine room through one 100-amp breaker on the 115-v a-c panel, two 25-amp breakers in the 440-v a-c panel, and one 200-amp switch on the 115-v d-c panel. The 115-v a-c lighting power, and instrument power (mostly to double-plug outlets) was supplied from the secondary side of the closed-delta transformer bank through the distribution boxes on the bulkhead of the control room. The 440-v 3-phase a-c power was supplied through junction boxes on the control-room bulkhead to the motor-generator sets of the amplidyne system forward of the control room. The 115-v d-c power to the miscellaneous heaters and blowers in the control, P-S motor, and the motor-generator rooms was supplied through a temporary bulldog switch behind the switchboard. This was the source, too, for the 110-v d-c to 12-v d-c motor generator installed in the starboard motor room as a power source for the Consolidated oscillograph.

Power was obtained through the ship's after generator and distribution board in the after engine room through the No. 3 generator circuit breaker on the 115-v d-c panel, through the 115-v 15-amp, and 50-amp a-c breakers, and through Circuit Number 454 100-amp breaker on the 440-v a-c panel. From here the direct current was supplied for the control equipment and instruments in the control room and for the controller in the motor-generator room. A 3-hp 115-v a-c to 2-kw 28-v d-c motor generator supplied power to the 2 kw 28-v d-c to 115-v 400-cps dynamotor. Both of these were installed in the starboard motor room. The 115-v 400-cps supply was distributed at various voltages through a panel in the control room to the accelerometers, to the 2-phase motors and synchros of the mechanical filter, to the calibrator synchro, to the driver unit for phasing, and to the mixer unit as a reference voltage. The 440-v 3-phase power was supplied to the 3-phase 440-v to 115-v d-c motor generator with starter and controller in the motor-generator room. This unit was the source of power for the port and starboard d-c impeller motors.

A schematic diagram showing the sources of power and the distribution points for the ship stabilizing system is shown in Figure 41. The cable layout

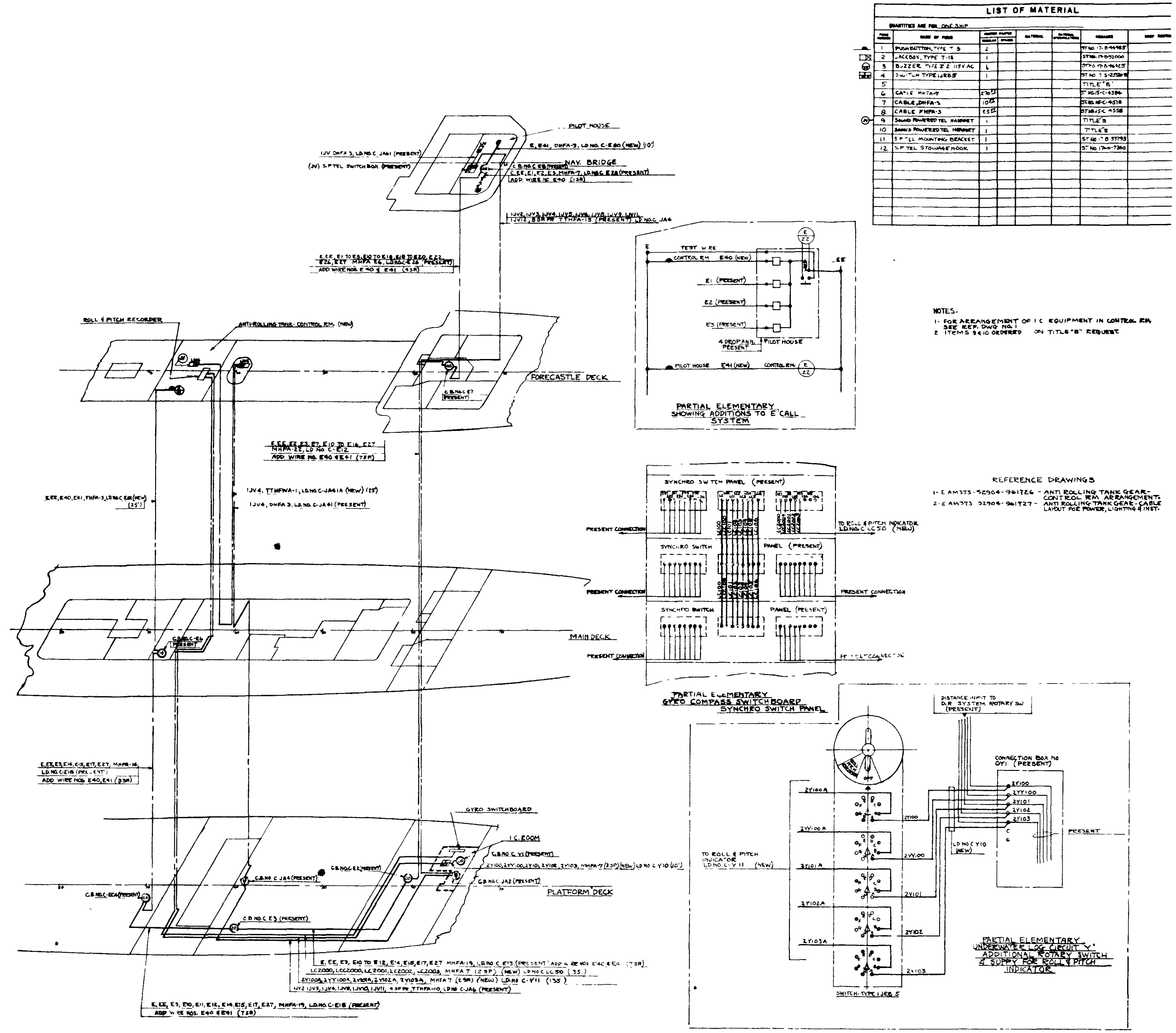


Figure 43 - Modifications to Ship Circuits for the Roll-and-Pitch Recorder

o the various rooms, together with connection boxes and units to be supplied, s shown in Figure 42. The roll-and-pitch recorder in the control room was perated by 115-v a-c power and was connected to the ship's gyroscope and pit og through a multi-conductor cable tied in to the ship's "Y" circuit (Figure 3).

NSTRUMENTATION

In order to check the performance of the various units of the con- rol system, the operation of the pumping system, and the efficacy and perform- nce characteristics of the process of stabilization with the activated tanks, he measurements to be made had to be planned before the equipment was built.

In the control system—driver, damping, and mixer units as well as he mechanical filter—oscilloscope taps were installed in the circuits at oints where it was thought checking might be necessary or trouble occur. For onnection to string galvanometers in a string oscillograph, cathode-follower ircuits and taps were installed at the output of the voltage amplifier and t the output of the current amplifier of the driver unit. Similarly a cathode- ollower circuit was installed at the output of the damping unit prior to the -c damping motor. Also for connection to string galvanometers, cathode- ollower circuits were installed at the input of the mixer unit after demodu- ation $\ddot{\theta}$, after differentiating once $\dddot{\theta}$, after differentiating twice $\overset{iv}{\theta}$, and fter mixing, $\ddot{\theta}$ and $\dddot{\theta}$, $\ddot{\theta}$ and $\dddot{\theta}$ and $\overset{iv}{\theta}$, and output. Resistance circuits for ie-in with the string galvanometers were installed at the mechanical filter o the outputs of the error and output synchros and to the input of the d-c amping motor.

All these galvanometers were installed in an 18-channel Consolidated scillograph. Additional circuits were used to record water level (port and tarboard) by means of float synchro outputs and to record blade pitch by port nd starboard synchro outputs with the synchros geared to the pitch shafts. he mechanical-filter and driver-unit measuring circuits had a 400-cps carrier, hose of the mixer unit had no carrier, while those for pitch and water level ad 60-cps carrier. The oscillograph was used with 1/10-second timing marks cross the photographic record paper as an aid to analysis. In addition a ecord synchronous timer actuated one string galvanometer in this oscillograph s an assurance of accurate timing. The data obtained from the Consolidated ecords permitted the determination not only of relative amplitudes of the river and mixer units and absolute amplitudes of the mechanical filter, blade itch, and water level, but also phase relationships which are such an impor- ant consideration in evaluating the performance of the control equipment and f the stabilization equipment as a whole.

Calibration of the individual control units—driver, mechanical filter, and mixer—was obtained by means of the string oscillograph and the variable-speed calibrator. Representative calibrations of these units have been presented in earlier sections of this report. The tank system was calibrated in drydock (both statically and dynamically), in calm water, and in rough water by means of the oscillograph records obtained; these calibrations are described in subsequent sections. The galvanometer strings which indicated pitch angle and water level for the forward and aft tank systems were calibrated by observation of the pitch angle on the pitch mechanism and observation of the water position in the tanks while the outputs of the pitch and water-level synchro generators were recorded on the oscillograph. The angular position of the rotor of the pitch synchro was changed by gearing in proportion to the pitch of the impellers. Consequently the electrical signal was proportional, since the electrical output of the synchro was linear for the small angular rotations $\pm 30°$.

In the case of water-level indication, here again a synchro generator was used within a range of rotations of the rotor of $\pm 30°$, and the rotor was connected to the float in the tank through reduction gearing and grooved pulleys to the float which was guided by vertical rails. The pitch calibration was quite linear. A certain amount of backlash occurred upon reversal of the pitch mechanism owing to gear and linkage clearances and was a little greater for one pitch mechanism than for the other but in no case exceeded 2° and that for only a portion of the range. The float followed the water surface quite well. With the water in the tank being oscillated by means of the impellers with sinusoidally altered pitch, the float followed the water surface exactly except at the point of reversal of the water. For example as the water came to its high point and started down, the line of submergence would decrease; as water came to its low point and started up, the line of submergence increased. For periods of 7 seconds and greater the line of submergence shifted at most 2 in., which was at the low and high points of the water, and was less when the water transfer was smaller. For shorter periods it might have been as much as 3 in. for the maximum water transfer possible at those periods. On some of the early tests a potentiometer was connected to the float. It required considerable attention since the friction of the potentiometer slide varied, and there may have been errors of as much as 1/2 ft before the difficulty was noticed. To the pitch synchro generators were connected synchro motors with scales and pointers in the control room to indicate pitch angle of both units. The water-level synchro generators were connected to meters in the control room—also to indicate water level in the two systems.

For some of the runs made in drydock, in calm water, and at sea, power and strain measurements were obtained on a Hathaway oscillograph. The d-c volts and amperes used during pumping by the drive motor and the impeller motor were recorded so that the average power as well as the maximum power needed to operate the impellers and pitch mechanism on both tank units could be determined. Strain measurements were made on the starboard impeller shaft—to measure torque and thrust under various conditions of loadings—by means of SR-4 wire-resistance strain gages and TMB Type-1B and Type-1D strain indicators with recording on the Hathaway string oscillograph. The strain signal was conducted from the gages to the strain indicators through slip rings. A diagram showing the arrangement of the strain gages and the slip-ring assembly is shown in Figure 44. It was unfortunate that the difficulty of installing strain gages on the pitch shaft precluded making these measurements, but it was felt that some measure of shaft loadings could be obtained from data on the impeller shaft even though it was the stronger of the two.

The instrument used for evaluating the effectiveness of the ship stabilizing system was the Ship's-Motion Recorder Mark 1 Model 0 (Figure 45), designed and built for the Bureau of Ships in the Material Laboratory, New York Naval Shipyard.[43] The ship's-motion recorder is a means of recording angles of roll and pitch, ship's course, and ship speed. The recorder has its own power supply operated by 115-v a-c power and a master clock for timing. The time is stamped on the records for every 3 in. of record-paper feed. The recorder has its own gyroscope stable element from which, by means of servomechanisms, the recording pens for roll and pitch are actuated. Similar mechanisms are used for recording ship's course by tying in to the ship's gyroscope and for recording ship speed by tying in to the ship's pit log. Only the roll and pitch elements were used during the stabilization tests. Many records were obtained with the ship unstabilized as well as with the stabilizing equipment in operation. The recorder was operated continuously when the PEREGRINE was at sea.

The purpose of the instrumentation, the two string oscillographs and the ship's-motion recorder, was to obtain data from which the performance of the activated-tank equipment as a whole and the performance of the individual units could be analyzed and to evaluate certain design features in order to write specifications for permanent installations.

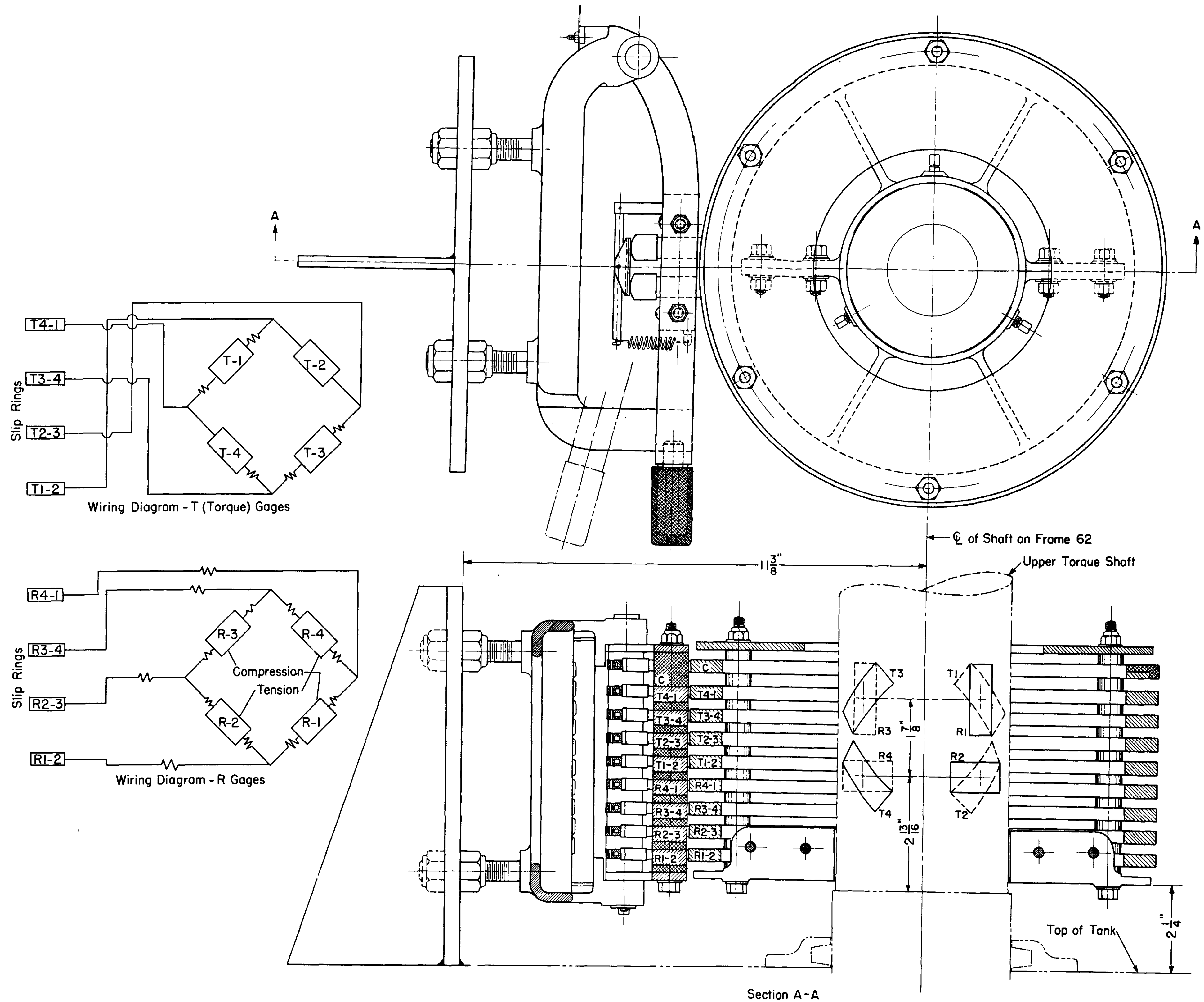


Figure 44 - Strain-Gage Arrangement and Slip-Ring Assembly for Measuring Torque and Thrust on an Impeller Shaft

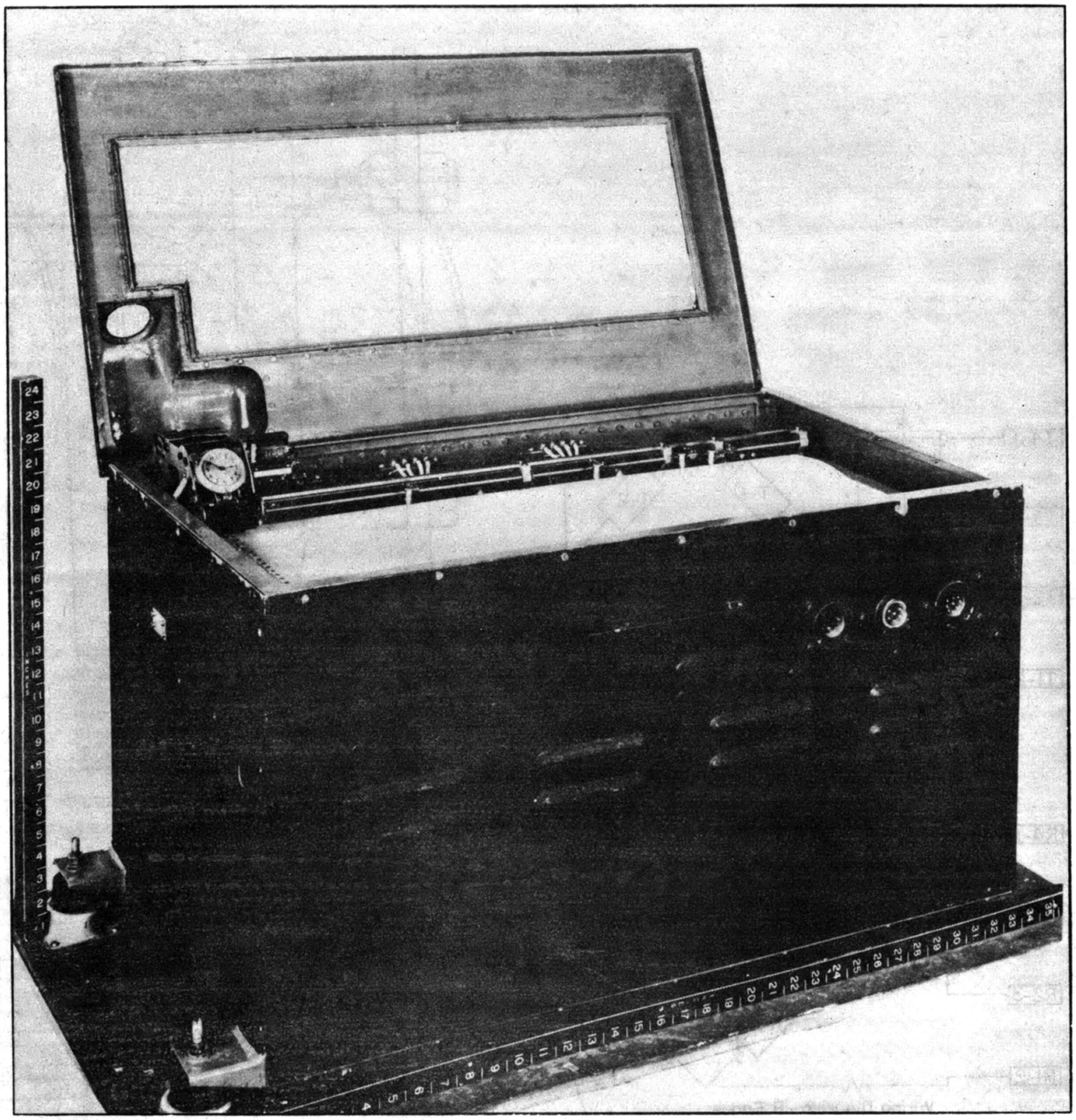


Figure 45a - Over-All View

DRYDOCK, CALM-WATER, AND ROUGH-WATER TESTS

The main objectives of the drydock, calm-water, and rough-water tests was first to evaluate the Minorsky principle of ship stabilization by means of activated tanks and second to evaluate the particular application of this principle in the stabilization of the USS PEREGRINE (E-AM373).

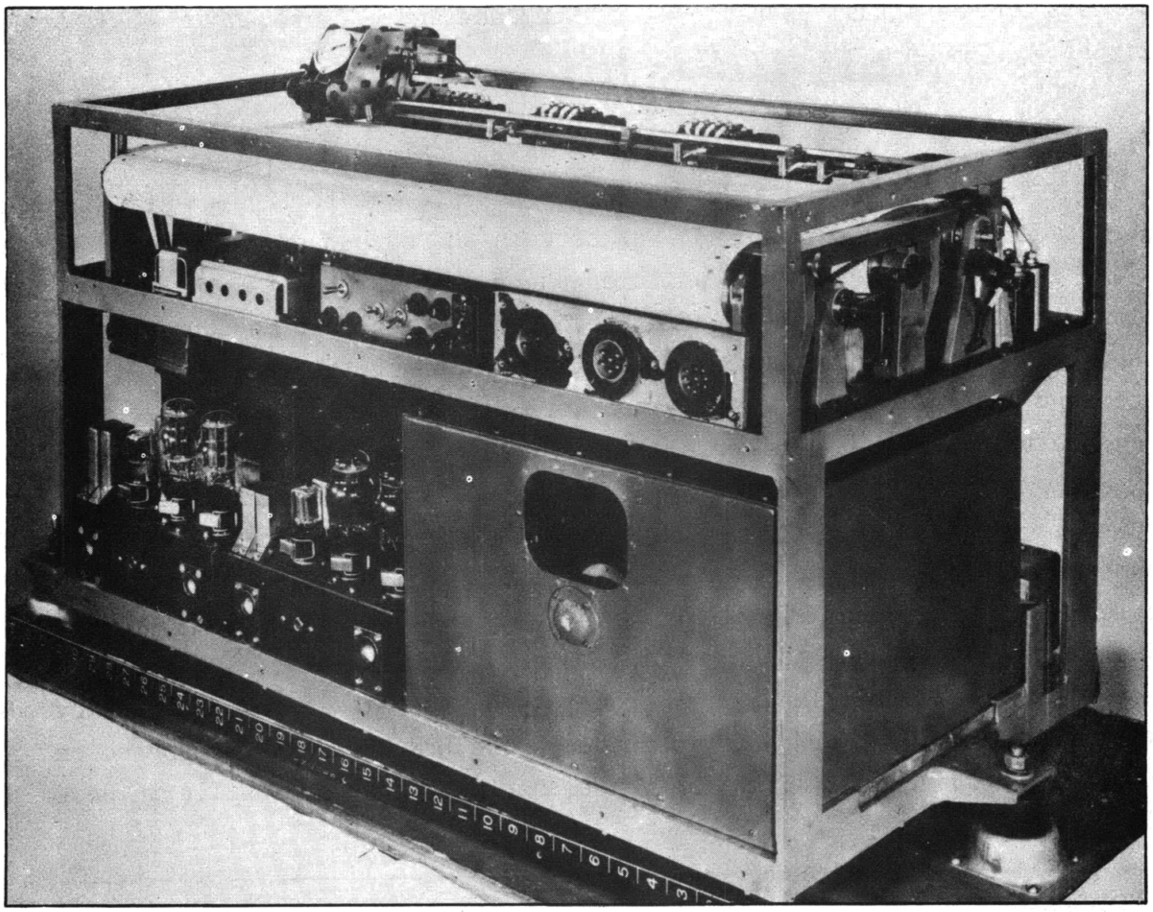


Figure 45b - View with Case Removed

Figure 45 - Ships' Motion Recorder Mark 1 Model 0

In order to achieve these two objectives, a series of tests, in drydock, in calm water, and in rough water, was planned so that the equipment could be adjusted and modified where necessary and so that the entire installation could be calibrated and tested. The drydock tests permitted check of the performance of the control equipment (except the accelerometers and the mechanical filter) together with the pumping equipment and permitted check and adjustment of the amplidyne systems. The drydock tests permitted accurate calibrations of devices for recording pitch and float synchros and volt- and ampere-measuring circuits. In the drydock tests, too, with the ship firmly supported, static and dynamic water-transfer data of the tank systems were obtained. The calm-water tests served to show how much could be accomplished by rolling the ship and then stabilizing it; the question of correct wiring of accelerometers for the maximum output, reduction of noise level, and correct wiring for stabilization were also settled. The sea or rough-water tests gave data on the actual operational performance and effectiveness of the stabilizing system in a variety of seaways.

DRYDOCK TESTS

The primary purpose of the drydock tests was to try out the pumping systems and to tie in the connections between the output of the control system and the input to the amplidyne units of the pumping system. For checking and calibrating the ship had to be held steady so that no ship motion would be caused by the transfer of water and would thus affect the results. Static tests were made to determine the water level in the tanks at various steady-pitch positions of the impeller blades. Dynamic tests were made to determine the peak water levels at various frequencies and amplitudes of pitch oscillations. These tests—static and dynamic—were made in September 1949. After a number of calm-water and rough-water tests, a portion of each impeller at the leading edge was cut off to shift the center of pressure on the blades close to the pivotal axes and to utilize more fully the natural oscillation of water in the ducts. Then, in February 1950, the static and dynamic drydock tests were repeated. For the drydock tests the PEREGRINE was supported on blocks in a graving dock with transverse shoring to the dock walls in order to

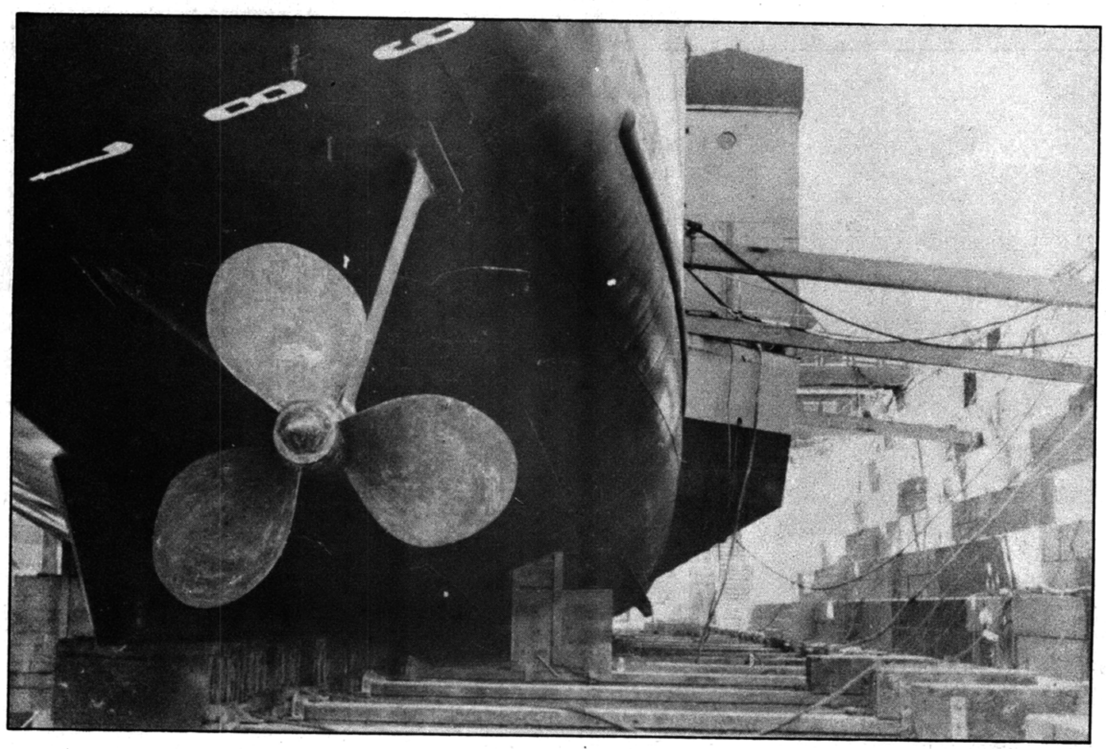


Figure 46 - The USS PEREGRINE (E-AM373) in a Graving Dock Supported by Blocks and Transverse Shoring for Drydock Tests

prevent any rocking and consequent damage to the ship's bottom, Figure 46. In this manner the ship was rigidly held in the vertical position during the drydock tests.

Static Tests

There were two test periods, Drydock Test 1 in September 1949, and Drydock Test 2 in February 1950—the latter after the impeller blades had been cut down. In these static tests, the pitch was adjusted by means of the manual-control handwheel mechanism to various settings, starting from 0° and changed at first in 1/2° steps and then in 1° steps until the d-c impeller motor kicked out. The pitch angle was held at each setting with the impeller running until the water in the particular tank system reached a constant level. Observations of water level were made at the plastic sight slits in the tank opposite to the one in which the impeller was located. The two tank systems were filled so that water was exactly halfway up in both tanks of each pair when no pumping was being done.

Curves of water level plotted against pitch angle of the impellers—obtained on these static tank-system calibrations—are shown in Figure 47. The pitch of the impeller blades was changed from zero, at which setting no water was pumped, to the maximum plus-or-minus pitch setting possible before the d-c impeller motor kicked out. An indication of the relative heights of water level in the port and starboard tanks for plus and minus pitch angles is shown in the sketches in Figure 47 for the port impeller and its after tank system and for the starboard impeller and its forward tank system. Also shown in this figure is the configuration of the impeller blades before being cut down and again after one-sixth of the blade area was removed by a radial cut at the leading edge of the blade. After being cut, the leading edge was rounded and faired into the blade surface.

The curves of water level plotted against pitch angle for the port impeller and its after tank system are plotted in Figure 47a. The solid-line curve is for Drydock Test 1 before the cutting of the blades. Here the maximum change in water level, +3.7 ft, occurred at a pitch of -5°, and a change of -3.68 ft at a pitch of +5°. The impeller motor with negative pitch angle kicked out at -12°, and with positive pitch angle at +13°. The dotted-line curve is for Drydock Test 2 after the cutting of the blades. Here the maximum change in water level, +3.0 ft, occurred at a pitch of -11.5°, and a change of -2.73 ft at a pitch of +13°. The impeller motor with negative pitch angle kicked out at -18°, and with positive pitch angle at +16°.

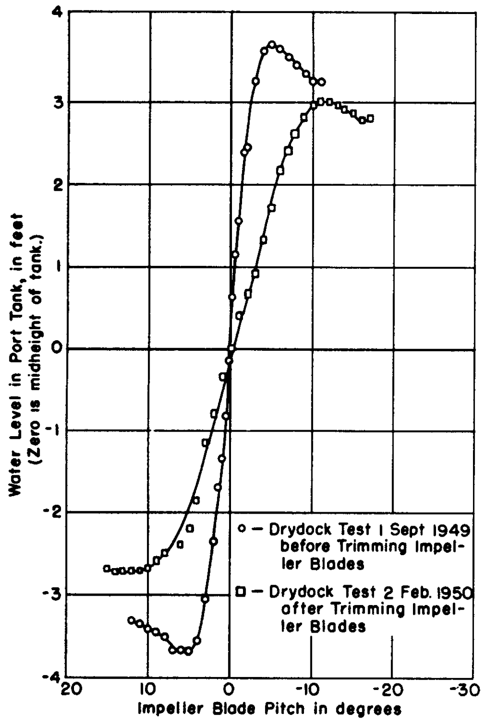
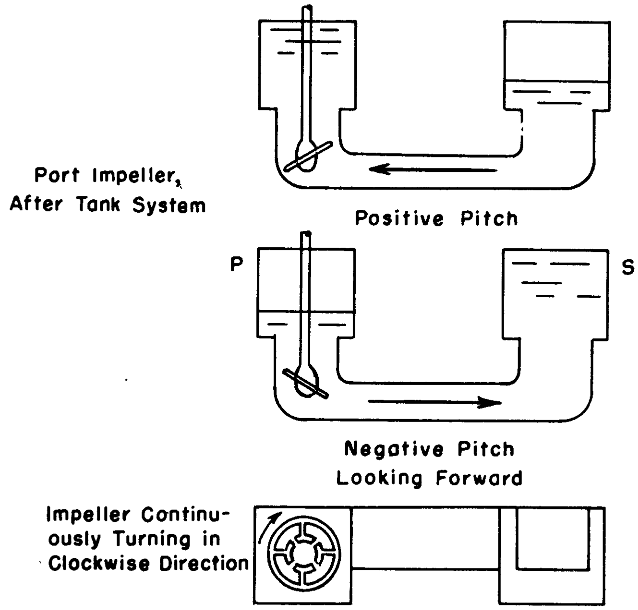
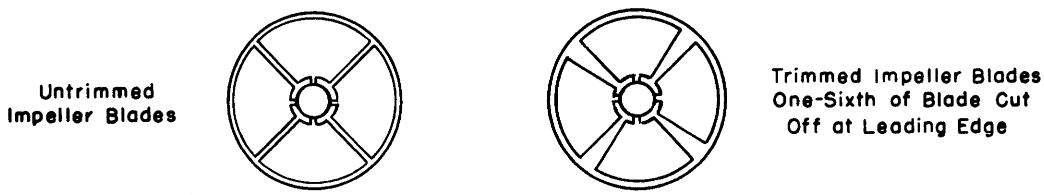


Figure 47a - Water Level Plotted against Pitch for the Port Impeller and After Tank System with the Impeller Blades Untrimmed and Trimmed





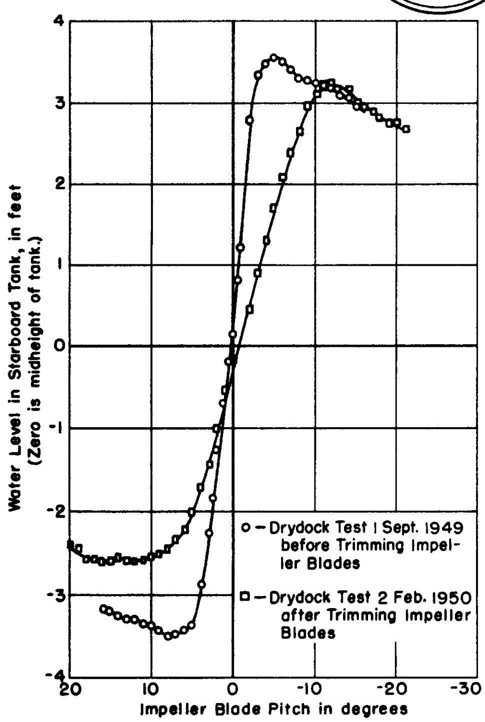


Figure 47b - Water Level Plotted against Pitch for the Starboard Impeller and Forward Tank System with the Impeller Blades Untrimmed and Trimmed

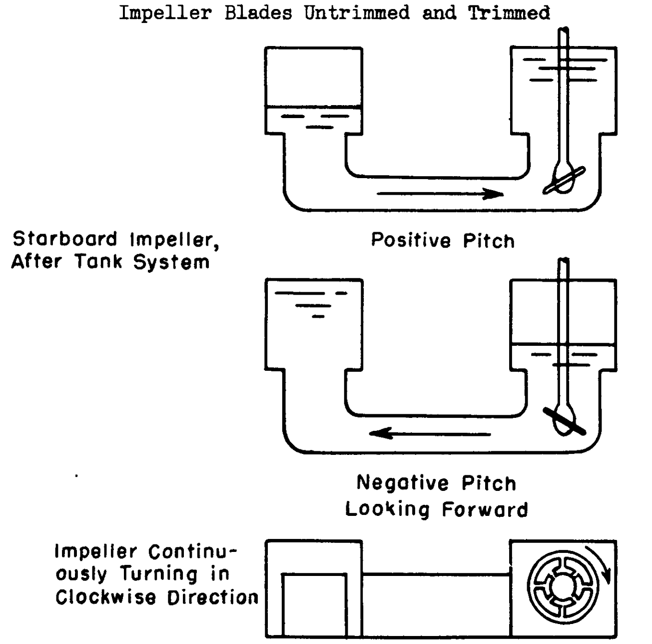


Figure 47 - Static Water-Transfer Calibrations, Water Level for Different Constant Impeller Pitches of the After and Forward Tank Systems

The curves of water level plotted against pitch angle for the starboard impeller and its forward tank system are plotted in Figure 47b. The solid-line curve is for Drydock Test 1 before the cutting of the blades. Here the maximum change in water level, +3.55 ft, occurred at a pitch of -5°, and a change of -3.5 ft at a pitch of +7.5°. The impeller motor with negative pitch angle kicked out at -15°, and with positive pitch angle at +17°. The dotted-line curve is for Drydock Test 2 after the cutting of the blades. Here the maximum change in water level, +3.16 ft, occurred at a pitch of -11°, and a change of -2.6 ft occurred at a pitch of +14°. The impeller motor with negative pitch kicked out at -22°, and with positive pitch angle at +21°.

Certain results are evident from these data. The maximum head due to impellers runnnng at various constant blade pitches occurred at smaller angles before the blades were cut than after the blades were cut but the value was greater before cutting than after. On the port impeller a total head of 7.38 ft was obtained for a pitch change from -5° to +5° before cutting the blades, and a total head of 5.73 ft for a pitch change from -11.5° to 13° after cutting the blades—a reduction in head of 22.3 percent. On the starboard impeller a total head of 7.05 ft was obtained for a pitch change from -5° to +7.5° before cutting the blades, and a total head of 5.76 ft for a pitch change from -11° to +18° after cutting the blades—a reduction in head of 18.3 percent.

For the uncut blades, Drydock Test 1, an over-all head of 7.38 ft was obtained for the port impeller and a head of 7.05 ft for the starboard impeller. The somewhat smaller head, by 0.33 ft, for the starboard impeller can be accounted for by the poorer fit in the herringbone reduction gears between the impeller motor and the impeller shaft as manifested by greater noise and higher temperatures in the gear box. For the cut blades, Drydock Test 2, an over-all head of 5.73 ft was obtained for the port impeller, and a head of 5.76 ft for the starboard impeller—substantially equal for both impellers. This may be explained as being caused by the realignment of gears and replacement of bearings in the starboard unit during the overhaul period when the impeller blades were cut. The smaller ratio of water level to pitch after cutting the blades was to be expected since the blade area was reduced. The dips after the peaks in the curves are indications of heavy loading on the d-c impeller motors. At this stage the slower speed begins to have more effect than the increase in pitch; therefore the head decreases for a time with increase in pitch until the impeller motor kicks out owing to the overload becoming too great.

Dynamic Tests

Dynamic water-transfer tests were made during Drydock Test 1 in September 1949 with the impeller blades uncut and were repeated to some extent during Drydock Test 2 in February 1950 with the blades cut. The purpose of the dynamic tests was to determine the pumping characteristics of each of the two tank systems with respect to the frequency of reversals of the impeller pitch. The smoothness of operation, amplitudes of water transfer, phase lag of the pitch of the impeller blades, and the phase lag of the height of water with respect to the initial signal were determined.

During the transfer tests each impeller was rotated continuously, and the pitch of the impeller was changed sinusoidally. The pitch mechanism including its amplidyne drive was actuated by a low-frequency calibrator through voltage reducers and the mixer unit. The sinusoidal signal with 400-cps carrier was generated by a synchro rotated by the low-frequency calibrator which was capable of continuous adjustment in a frequency range of 1/18 to 10 cps, or a period range of 18 to 1/10 seconds. Various pitch amplitudes were obtained by reduction of the synchro signal through a combination of variac voltage divider and potentiometer before supplying the signal to the mixer unit. The mixer unit was used to demodulate the 400-cps carrier and to match the signal to the input of the amplidyne system. The modulator unit was used as a feedback circuit in that a signal proportional to the water level in one tank of a system was added to the mixer signal before it was fed to the amplidyne 5CT input circuit. This signal was always a small proportion of the sinusoidal signal and was used in the proper sense to keep the water oscillating about the midheight of the tank. Without the modulating signal, the oscillating water would gradually creep to the upper or lower portions of the tank.

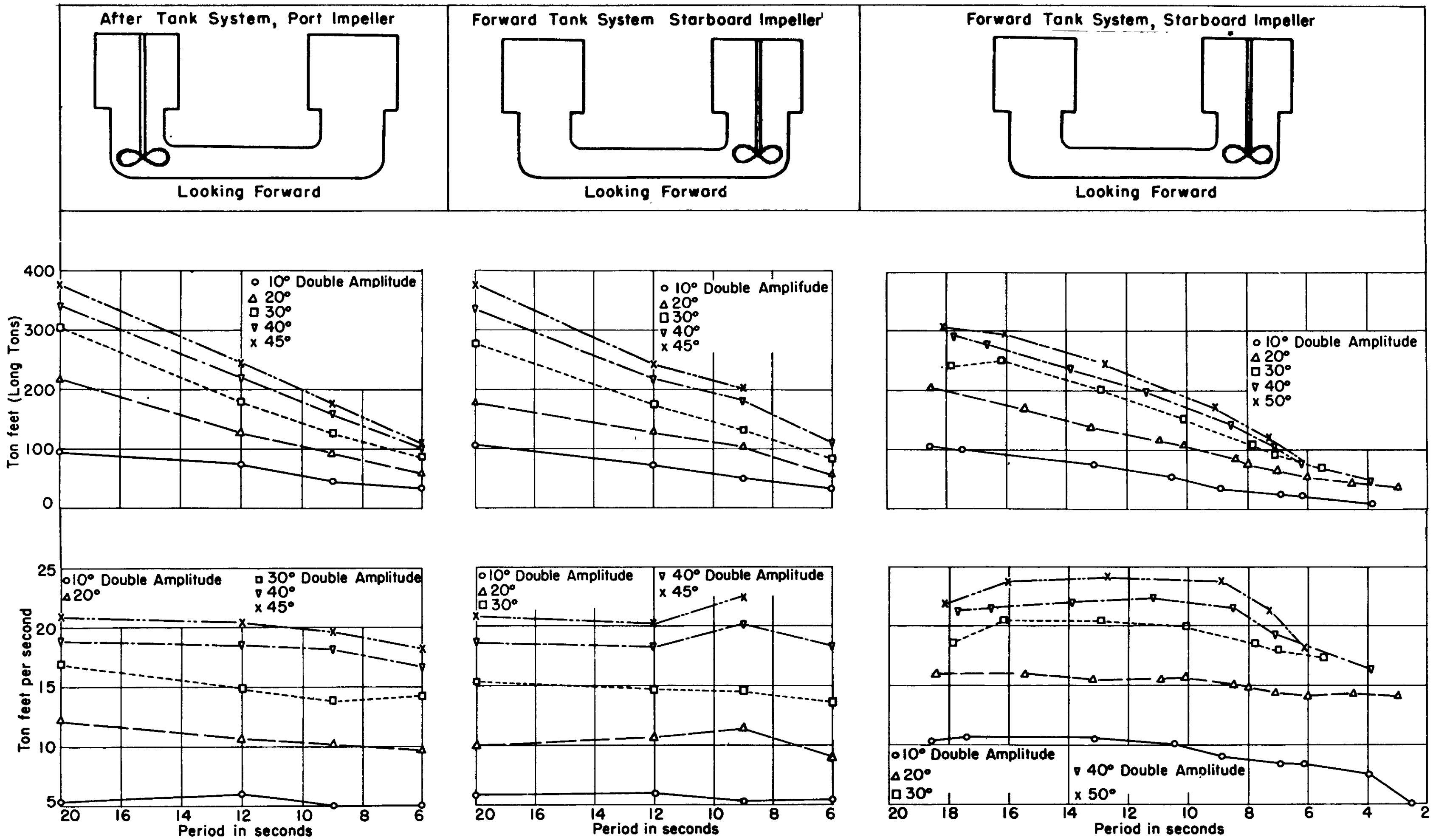
Data were obtained at each of the pitch double-amplitudes of 10°, 20°, 30°, 40°, and 45° during Drydock Test 1, September 1949, at periods of 18, 12, 9, and 6 seconds. The forward system with starboard impeller and the aft system with the port impeller were calibrated separately. During Drydock Test 2, February 1950, the forward system with starboard impeller was calibrated with reasonable thoroughness at about eight periods for 10°, 20°, 30°, 40°, and 50° pitch double amplitudes. This calibration covered a wide range of periods from about 18 seconds to about 3 seconds. Then a limited number of runs at two pitch-amplitude settings were made with the forward and aft systems operating together. Readings were taken of maximum and minimum levels of water in the tank (using the window opposite the one where the impeller was installed), pitch angles were read from the pitch indicator, and the period of rotation of the calibrator was timed. These were taken as a check during the run and as a cross-check against the results obtained from the oscillograms.

Recorded on the oscillograms were the calibrator output, the mixer output, blade pitch, water level, and a timing trace. Each condition was set, and readings and records were taken after the water level settled down to a regular oscillation between two fixed points.

The results of these drydock tests are plotted in Figure 48. Here the effective moment of the water transfer in ton-feet, the moment-per-second in ton feet per second, the moment-per-degree-pitch in ton-feet per degree, and the moment-per-degree-pitch-per-second are plotted for different periods of input signal. The moment in ton-feet is in units of long tons. The results for the after system with the port impeller and for the forward system with the starboard impeller for dynamic Drydock Test 1 and for the forward system for dynamic Drydock Test 2 are plotted side by side in Figure 48 for ease of comparison.

The moments in ton-feet due to the oscillation of water are plotted against period for a number of increments of pitch amplitude, see Figure 48a. The moments are obtained by the difference of maximum and minimum water level multiplied by the moment arm of the water in the tanks, giving a change in moment for one cycle of impeller-pitch oscillation. The periods of sinusoidal oscillation of the actuating signal generated by the calibrator were set successively at 18, 12, 9, and 6 seconds for Drydock Test 1. Since on the calm-water tests and early sea tests it was discovered that the period of roll was in the range of 7 to 8 seconds rather than the expected 9 to 10 seconds, on the dynamic Drydock Test 2 runs were made at periods as low as 3 seconds and at more frequent intervals. For each run the pitch double-amplitude was usually set to within 1° or 2° with somewhat greater error at the lower pitch amplitudes and especially at 45° double-amplitude because of the touchiness of the controls in the dynamic drydock test set-up. Also, the amplitude was not always equal on both sides of zero; for example, the 30° setting at one period may mean a pitch of +15° and -13°—or a double amplitude of 28°—and at another period +14° and -16°—or a double amplitude of 30°. Nevertheless a moment-period curve was plotted for each nominal pitch setting since it showed the trend for approximately equal increments of pitch.

During the drydock tests, too, the voltage did not always remain constant. Yard power had to be used, and considerable variation occurred especially at the end of labor shifts. The voltage output of the motor-generator which furnished the d-c power to the impeller motors varied as much as from 108 to 118 v when set at 114 v. This voltage was checked and adjusted from time to time, but it was not found practicable to adjust continuously the output of the motor-generator. A change of a few volts made some difference


After Tank System, Port Impeller
Looking Forward
Forward Tank System Starboard Impeller
Looking Forward
Forward Tank System, Starboard Impeller
Looking Forward
Ton feet (Long Tons)
400
300
200
100
0
o 10° Double Amplitude
Δ 20°
□ 30°
▿ 40°
x 45°
o 10° Double Amplifude
Δ 20°
□ 30°
▿ 40°
x 45°
o 10° Double Amplitude
Δ 20°
□ 30°
▿ 40°
x 50°
Ton feet per second
25
20
15
10
5
o 10° Double Amplitude
Δ 20°
□ 30° Double Amplitude
▿ 40°
x 45°
o 10° Double Amplitude
Δ 20°
□ 30°
▿ 40° Double Amplitude
x 45°
o 10° Double Amplitude
Δ 20°
□ 30°
▿ 40° Double Amplitude
x 50°
20 16 14 12 10 8 6
Period in seconds
20 16 14 12 10 8 6
Period in seconds
20 18 16 14 12 10 8 6 4 2
Period in seconds

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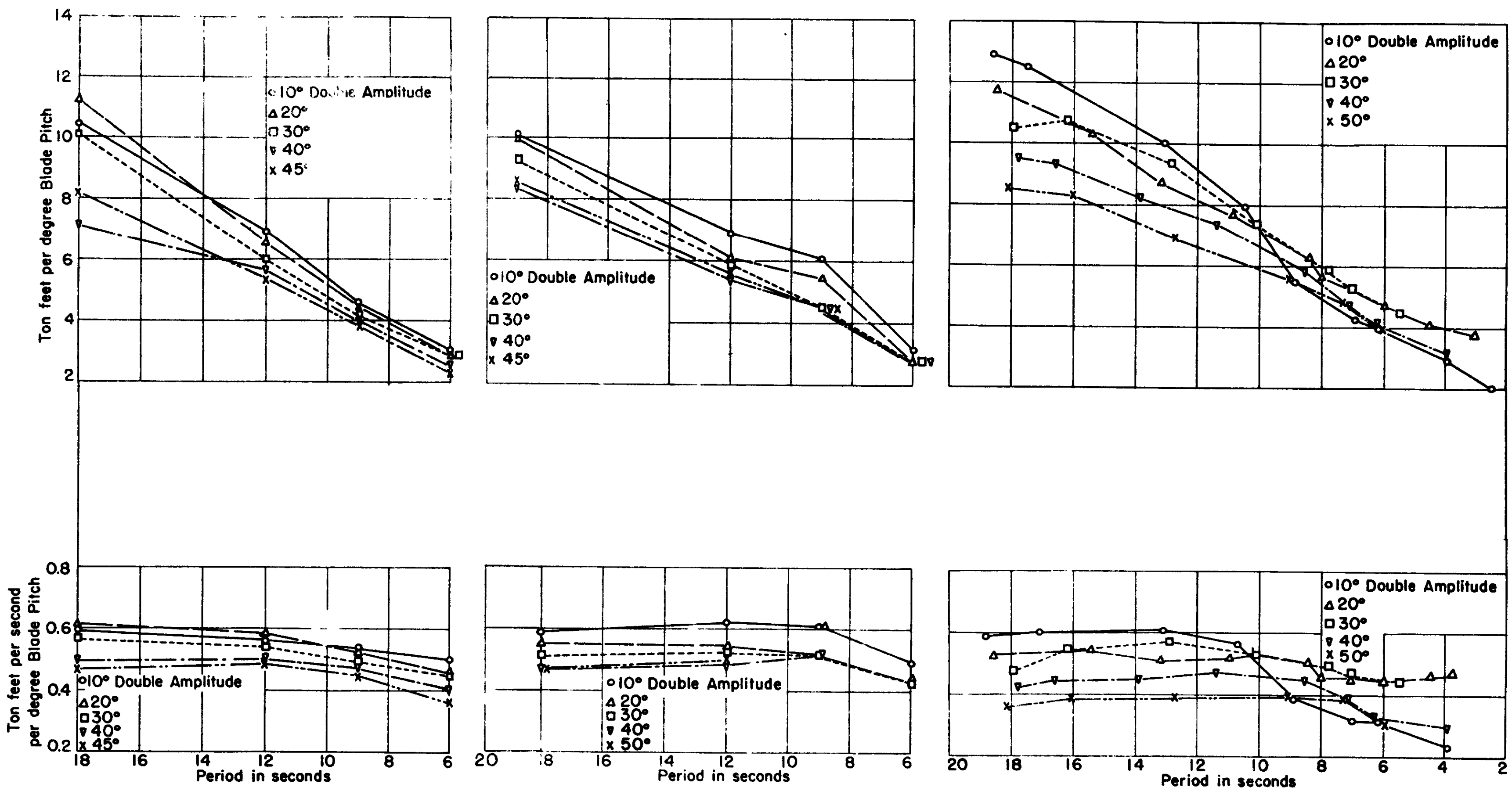

Figure 48 - Performance of the Forward and Aft Tank Systems as Determined from Dynamic Drydock Tests 1 and 2

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in the speed of the impeller so that the comparison of moments of water transfer was not on an absolute basis. As a consequence of this voltage fluctuation, there were temporary variations in the calibrator speed. This was indicated by the different periods of successive cycles occasionally obtained on the oscillograms. The voltage change did not immediately affect the impeller speed and the calibrator speed because of inertia of the flywheel of the impeller motor and because of slack in the V-belts and inertia of the pulleys of the calibrator. These lags were not the same and were greater for the impeller. This type of effect may account for the moment being somewhat out of line on the test with the starboard impeller at the 9-second runs during Drydock Test 1; Figure 48a.

Another factor affecting the smoothness of the moment curve was the fact that the maximum and minimum water heights were not exactly the same for successive cycles but varied about mean maximum and minimum levels. The amount of modulator-feedback signal introduced was adjusted for each run; the

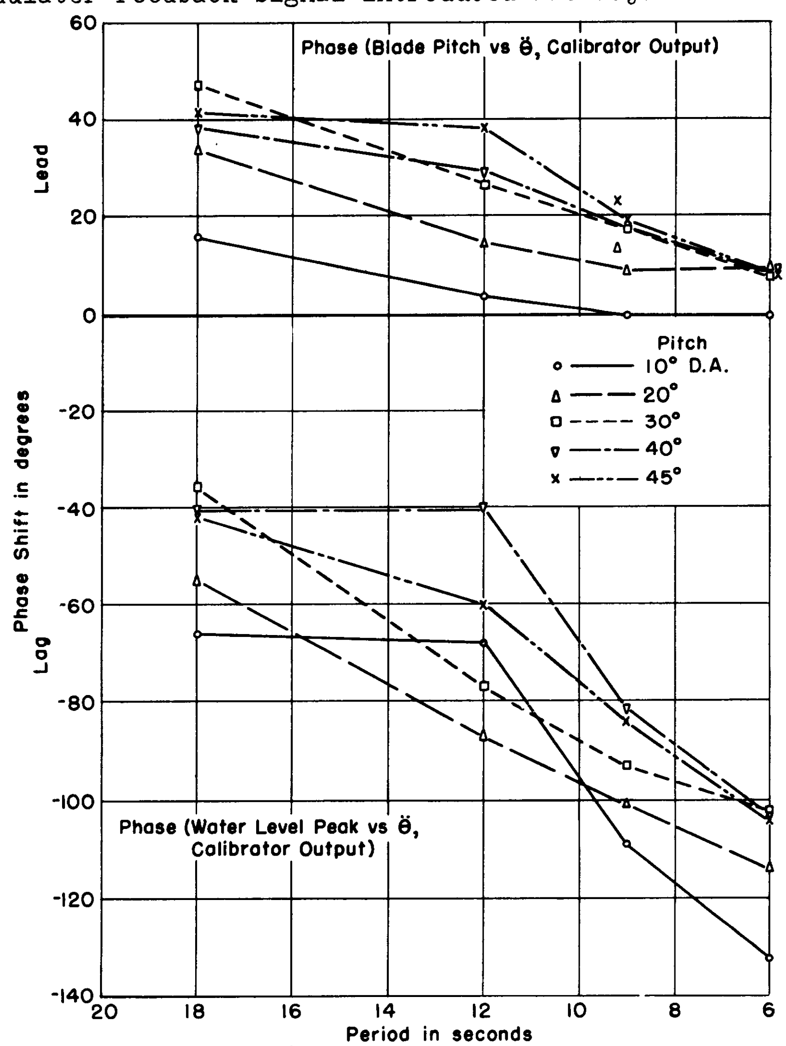

Figure 49a - Port Impeller, Aft System, Drydock Test 1

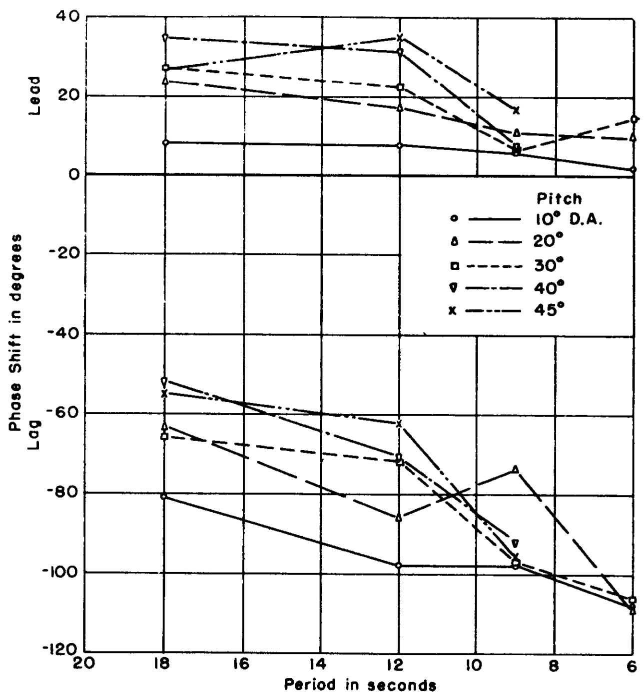

Figure 49b - Starboard Impeller, Forward System, Drydock Test 1

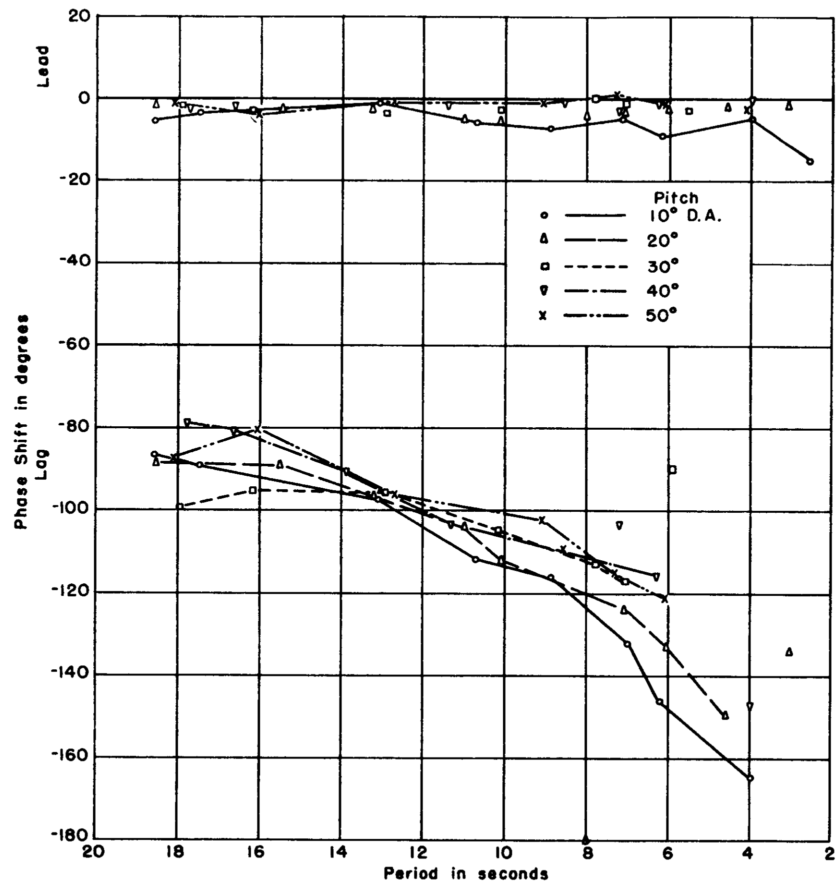

Figure 49c - Starboard Impeller, Forward System, Drydock Test 2

Figure 49 - Phase Relations between Water Level, Pitch, and Calibrating Signal for dynamic Drydock Tests 1 and 2

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minimum introduced was just sufficient to keep the water oscillating between mean values without drift. The effect of the small modulator signal was therefore somewhat different each time. Since the modulator feedback signal was proportional to and in phase with water level, it was out of phase with the mixer output—upon which it was superimposed by about 90°—and tended to distort the sinusoidal signal of the mixer. On Drydock Test 2 somewhat less of the modulator signal was needed since the amplidyne system was balanced better in that equal peak power was needed for each qaarter cycle of impeller blade position. This was accomplished by improved balancing of the amplidyne electronic circuits.

On Drydock Test 1, because of the potentiometer method of obtaining records of water level used at that time and because of possible error in measuring amplitudes of waver level from the small oscillogram trace, it was believed that the use of the visual data would give more representative curves for Drydock Test 1. Therefore the moment curves and the curves derived from them were plotted for this test from computations based on visual data. There was another advantage in this—the average value of water level for a particular run was more easily obtained by observation than by analysis of oscillograms. However, for Drydock Test 2 the curves were plotted from data obtained by the analysis of oscillograms since the test party at this time was too small to obtain complete visual data. For these reasons broken-line curves have been drawn in Figure 48 to indicate trends. It is believed that the particular points on the curves are accurate to about ±10 percent.

Now that the basis upon which the dynamic water-transfer data were obtained has been stated and the known and possible errors in the data accounted for, or at least explained away, we are now ready to show the characteristics of the water-transfer process as indicated in the curves of Figure 48. On Drydock Test 1 the moment decreased quite linearly with period for both port and starboard impellers, Figure 48a. In the same figure, on Drydock Test 2 after the impeller blades were cut, the moment curves were similar except that below 6 seconds the curves started to level out, At 18 seconds, for large pitch amplitudes, the moments were a little less than for Test 1. At 12 seconds the moments were about the same except for the 50° pitch amplitude where the moment was less; at 6 seconds the moments were again less, for all pitch amplitudes.

In order to plot all the runs for different amplitudes and periods on the same basis, values for derived curves were computed. In Figure 48b the moments per second were plotted against period. For Drydock Test 1 the moment per second was about the same, decreasing slightly with decreasing period; for Drydock Test 2 the moment per second was about the same for period between 17 and 9 seconds decreasing above 17 seconds and from 9 to 4 seconds. On all the curves the moment per second increased at a decreasing rate with increased pitch amplitude. In Figure 48c the moment-per-degree-pitch was plotted against period and the values decreased linearly with a decrease of period. The curves in this figure for different pitch amplitudes indicate that at any period the rate-per-degree-pitch at which water was pumped decreased as the pitch amplitude was increased. Finally, in Figure 48d the moment-per-degree-pitch-per-second was plotted against period. Here we have a performance factor, independent of the pitch amplitude and of period, by which the performance at different pitch amplitudes for various periods could be compared. For the port impeller, Drydock Test 1, there was a small decrease in the factor from 18 to 9 seconds and then a greater decrease to 6 seconds. For the starboard impeller on the same test there was a slight increase in the factor from 18 to 9 seconds and then a rapid decrease. On both impellers for a particular period the factor decreased with increased pitch amplitude. For the starboard impeller, Drydock Test 2, the factor was about the same between 18 and 9 seconds and then a rapid decrease occurred from 9 to 4 seconds at some pitch amplitudes. Again for a particular period the factor decreased with increased pitch amplitude except for the 10° amplitude, where below 11 seconds the factor became less than for the large pitch amplitudes.

No mention has been made as yet of the phase relationships between the calibrator signal and the pitch of the impeller blades and the water level in the tank system; these are shown in Figure 49. The values have been determined from oscillograms, and the possibility of error (especially with the water-level trace and its potentiometer signal device on Drydock Test 1) have been discussed previously. The phase of the mixer output with respect to that of the calibrator has not been plotted because essentially it was about the same but had slightly greater phase lead than the phase of the impeller pitch with respect to this calibrator signal.

On Drydock Test 1, before the impeller blades were cut, both pitch phase and water-level phase changed more with period for the port impeller (Figure 49a) than for the starboard impeller (Figure 49b). For the port impeller at 18 seconds the phase lag of the water level to the calibrator signal was about 50°, at 6 seconds about 110°. The phase lead of pitch was about 35° at 18 seconds and 10° at 6 seconds. This gives a lag of water level with respect to pitch of 85° at 18 seconds and 120° at 6 seconds. For the starboard impeller the phase lag of water level was 65° at 18 seconds and 110° at 6 seconds, and the phase lead of pitch was about 25° at 18 seconds and 10° at 6 seconds. This gives a lag of water level to pitch of 90° at 18 seconds and 120° at 6 seconds. Therefore the lag of water level to pitch of the port and

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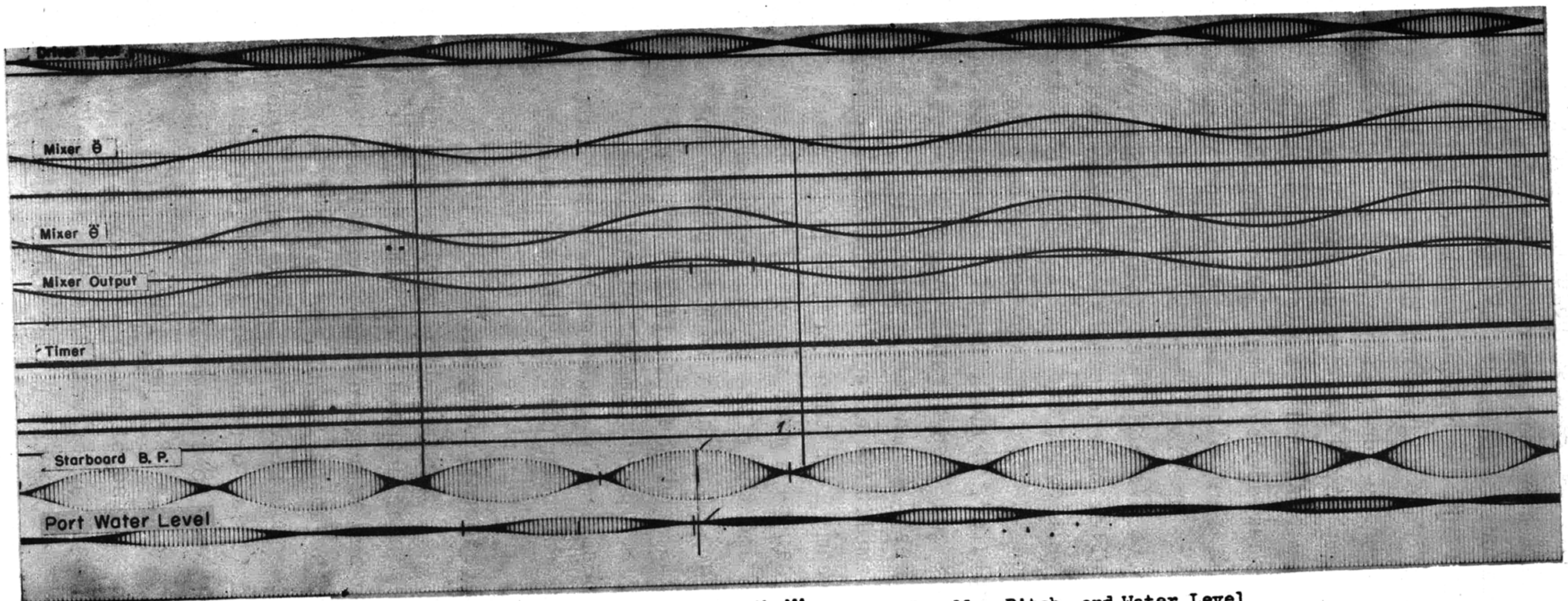

Figure 50a - Control System - Input, $\dot{\theta}$, $\ddot{\theta}$, Output, Impeller Pitch, and Water Level

(Consolidated 5209)

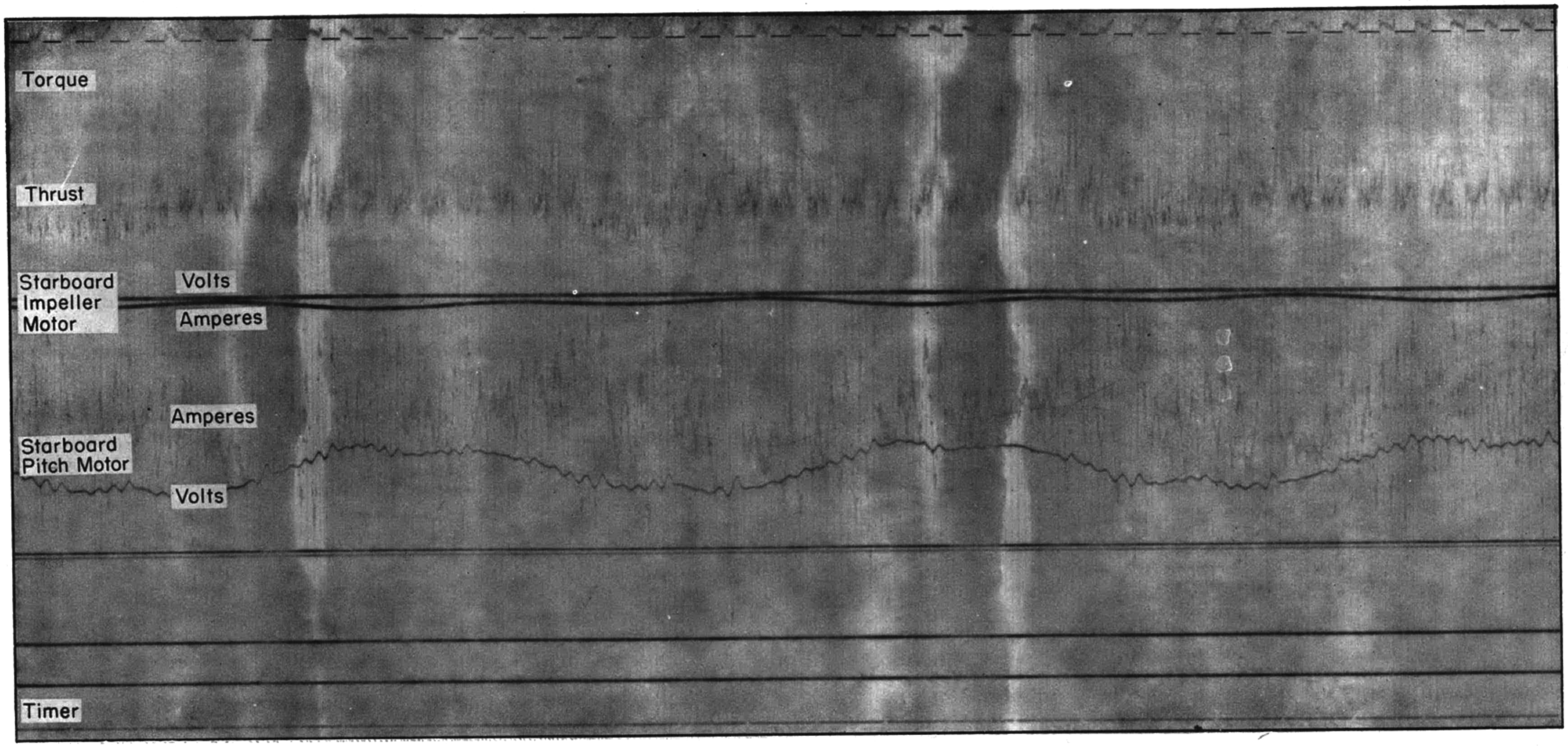

Figure 50b - Volts and Amperes of Pitch and Impeller Motor and Torque and Thrust of Impeller Shaft (Hathaway 176)

Figure 50 - Typical Oscillograms Obtained during Drydock Test 2 for 30° Double Amplitude Pitch Setting and 10-Second Period

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starboard impellers is about the same. On Drydock Test 2 after the impeller blades were cut, the pitch phase with respect to the calibrator signal remained practically constant but the water-level phase changed considerably with period especially at 6 seconds and below. For the starboard impeller (Figure 49c) at 18 seconds the phase lag of the water level was about 90° and at 6 seconds about 130°. The phase lag of pitch was about 2° at 18 seconds and about 4° at 6 seconds. This gives a lag of water level to pitch of about 88° at 18 seconds and 127° at 6 seconds. This lag is somewhat greater at 6 seconds than for the port and starboard impellers in Drydock Test 1. This change in phase shift with period is not as bad as it looks since a ship tends to roll at periods that fall within a narrow band, and the optimum phase lead can be obtained by adjustment of the amount of $\ddot{\theta}$ introduced into the signal at the mixer. The stabilizer effectiveness would be greater if the phase plotted against period were flatter within the range of rolling periods of a ship, since the average setting would be closer for over-all performance.

Typical oscillograms showing the performance of the control system and showing power and strain are reproduced in Figure 50 for Drydock Test 2. These are believed representative since not only was the control system in its best working order but the amplidyne system was better balanced, thus reducing the power consumed. The mechanical filter was not in the control circuit for these drydock tests. Depending upon the pitch variation and the period of forced oscillation for the various drydock runs, each impeller motor consumed from about 6 to over 30 peak horsepower. The higher power consumptions were for the longer periods and the larger pitch variations. The average power for each cycle of oscillation was on the order of one-third of the peak power. For the pitch motor the power consumption increased with pitch and with a decrease in period. The peak powers within a cycle varied from 2 to 45 hp, the thermal relays having been reset to permit greater than 40 hp swing. The average power was considerably less, one-fifth or more of peak power for the most part. Records of torque and thrust were obtained in the starboard impeller shaft during the drydock tests. Both reached values as high as 12,000 psi; however, the number of peak values this great was fewer for thrust than for torque. The average values for thrust too were less than half those for torque.

CALM-WATER TESTS

During the calm-water tests, records of self-rolling and self-quenching of the ship were obtained in addition to oscillograms of the performance of the control and tank system. However, the first task during these calm-water tests was to be sure the wiring hook-ups and switching arrangements for the accelerometers and for the low-frequency calibrator were such that the control signal was in the proper sense so that the ship could be rolled by either of these and, after the maximum angle of rolling was obtained, it could be stabilized by means of the accelerometers. A number of difficulties were revealed by this series of tests in calm water and had to be solved, such as insufficient isolation of the accelerometers from vibration of their supports (necessitating a filter), errors in wiring and non-isolation of the two parts of the driver unit, and the necessity for readjusting the potentiometers in the amplidyne circuits to prevent hunting.

After these deficiencies were corrected, the results during the calm-water tests were obtained under two principal conditions, first with the impeller blades uncut and second after the blades had been reduced in area. On each run the ship was rolled, using the sinusoidal calibrator signal for actuation. Its period was adjusted until the maximum angle of roll was obtained. When the maximum angle of roll was obtained, the actuating signal was shut off and the ship was allowed to roll freely. The impeller blades returned to zero pitch after the signal was shut off. In this manner the oscillating motion of the water in the tanks was effectively stopped, and in the test after the blades had been cut the oscillatory motion was quickly reduced. The period of free roll was about the same as the period of forced roll since the period of forced roll was adjusted until the maximum angle of roll was obtained.

With the uncut blades a maximum roll of 5 1/2° single amplitude was obtained with both impellers operating and a roll of 3 1/4° single amplitude with one impeller operating at a period of 7.8 seconds. After the impeller blades were reduced in area, a maximum roll of 4 3/4° single amplitude with both impellers operating was obtained at a period of 7.5 seconds. Tests were made rolling the ship with one system operating and with the impeller of the other system fixed, first at 30° and then at 0° pitch. No difference could be determined in amplitude or period. This indicated that no benefit was derived from the free oscillation of water in the latter system. This was not surprising since the natural period of oscillation of water in the tanks was about 12 seconds and damping was large. This natural period of oscillation was obtained by imposing an air pressure on the water surface in one tank of a system (creating a difference in head of about 1/2 foot), releasing the pressure suddenly, and recording the oscillation of the water level on a time base.

Another test procedure during the calm-water tests, principally for checking operation of the controls and the behavior of the tank systems, was to initiate a small angle of rolling by means of the calibrator signal, build up the roll to maximum by means of the accelerometer signal, and then reverse the accelerometer switch to quench the roll. This was done with and without the mechanical filter in the control circuit. Within the limitations of switching arrangements from rolling to stabilizing it was possible a good number of times to obtain dead beat stabilization in less than one-half cycle of roll. Now there was assurance that the control and pumping systems were operating properly and that all extraneous signals were minimized.

The results of the calm-water tests indicated, as did the drydock water-transfer tests, that the transfer of water was insufficient for full stabilization of the PEREGRINE. A number of factors were involved. A full head of water transfer could not be obtained at the periods of roll of 7 to 8 seconds since at these periods not only was there insufficient time for full transfer but the pumping efficiency was reduced. This factor was counteracted in part by the larger restoring moments possible per foot of transfer and was augmented by the fact that in conjunction with the increased metacentric height the exciting moment for a given sea condition was also increased due to the submersion of the sponsons. As indicated by the air test the oscillation of water was considerably damped, indicating much friction in the crossover ducts and elbows principally because of their length. Also, no advantage was taken of the natural period of oscillation of the water. The power consumption of the pitch and impeller motors was of the same order of magnitude as on the drydock tests except that the values were those of the upper end of the range since large impeller pitches and periods of about 7 seconds were the conditions of operation to obtain maximum roll.

SEA TESTS

A number of trips were made at sea with the impeller blades uncut, and one trip was made with the impeller-blade area reduced. Rolling and pitching records and motion pictures of the ship were obtained with and without the stabilization equipment in operation, so that the effectiveness of the system could be evaluated. Records were also obtained to determine the behavior of the control and pumping systems and the power consumed. In no case was over 50 percent stabilization obtained and in many instances less than 20 percent was obtained.

The first consideration is whether the installation of the stabilization equipment affected the behavior of the ship in a seaway or affected the handling of the ship in general. The change in metacentric height, as revealed in the inclining experiment, was expected to result in shorter periods of roll, but the effect of the sponsons on the handling of the ship was uncertain. Handling in a seaway proved, however, to be about the same as before the installation of the stabilization equipment. The ship seemed to steer as easily, the rudder sensitivity and the turning circle were about the same. The sponsons increased the metacentric height; however the diagonal bottoms of the sponsons were almost parallel to the sea upon contact when the ship rolled. Therefore the time of motion at the ends of the rolls were shortened, so that almost immediately the ship rolled back because of the slap of the bottom of the sponson on the water. The helmsman noticed only to a slight extent the slap of the waves against the sponsons. Relations between rolling, pitching, and yawing were definite. Generally, when the ship had large pitching amplitudes the rolling amplitudes were small, and vice versa. Also the amplitudes increased and decreased cyclically, so that as the pitch increased, roll decreased and as pitch decreased, roll increased. The greatest yawing occurred when there was the greatest rolling, especially with a quartering sea. On the trips to sea the yawing was worst with a quartering sea and a 10° maximum single amplitude of rolling. Under these conditions the helmsman had great difficulty trying to keep the ship on course; the single amplitudes of yawing were 3° to 5° with the helmsman compensating. Without compensation these single amplitudes may have been greater than 10°.

The greatest pitching occurred with the ship heading directly into the waves. The tank and motor-room structure made the ship somewhat more susceptible to heel owing to wind. The speed was reduced by the sponsons approximately 2 knots at 15-knot shaft speed and 1/2 to 1 knot at 5-knot shaft speed. The wave lengths during the sea tests were of the order of 150 to 200 ft, usually about 150 ft. The full extent of bilge keels was retained on the ship after the installation of the stabilization equipment except that the sponsons were located immediately above them on both port and starboard sides, probably making these portions of the bilge keels relatively ineffective. It was noticed on several occasions in regions of squalls that rolling of a few degrees was rapid; in less than 15 minutes it had built up to 25° to 30° single amplitude, and then diminished as rapidly.

At first the PEREGRINE operated within the capes at Norfolk, Virginia, with short trips outside the capes. During this period the operating difficulties with the stabilizing equipment were overcome. Finally three longer trips were made, at least several hundred miles eastward and southward. These three trips (November 16 to 19, November 28 to December 2, and March 3 to 7) are designated Sea Tests 1, 2, and 3. Sea Test 1 was essentially a shakedown cruise, and results are presented only for Sea Test 2—before the impeller blades were cut—and for Sea Test 3 after the blades were cut and other modifications made.

Before the over-all results of the effectiveness of stabilization are presented, it is well to present not only a sample record of the rolling of the unstabilized ship in a seaway but also an analysis of the rolling periods of the ship under various sea conditions. A sample record of rolling and pitching in a seaway is shown in Figure 51.

It is noted that the amplitude of roll builds up from less than a degree to the maximum and then decays. The same is true for pitching except that the maximum pitch amplitude occurs when rolling is minimum, and the minimum pitch amplitude occurs when rolling is maximum. At various aspects of sea to ship and at various ship speeds the rolling periods were timed. These are presented, in the form of incidence diagrams together with pertinent sea information, in Figure 52. In these diagrams there is a more definite peaking of periods of roll with quartering seas than with bow or astern seas. With quartering seas almost astern and with increases in ship speed, the dominant period of rolling increased. This variation was used to advantage in lengthening the period of roll for Sea Tests 2 and 3 so that the effect of stabilization could be measured. On Sea Test 1 the period of roll was close to 7 seconds so that stabilization effects were small because the time was too short for much water to be pumped back and forth, particularly with the water-transfer rate decreasing sharply in the vicinity of 7 seconds.

The stabilization achieved at various intervals on Sea Tests 2 and 3 is compared with that for unstabilized intervals on a time base in Figure 53. Typical roll and pitch records showing stabilized and unstabilized intervals for Sea Tests 2 and 3 are reproduced in Figure 54. The percent stabilization obtained in these various intervals under different sea conditions is tabulated together with the durations, the total angle of roll, the average angle of roll, and the average period for each run in Table 10. Also listed are the conditions of stabilization, the stabilization units in operation, the phase advance in the control system, and the peak amplitudes of roll for each run.

Each day the procedure in obtaining data was varied somewhat in order to be able to evaluate the variabilities encountered in obtaining data in such an erratic medium as a seaway and with a vessel whose amplitudes and periods of rolling were continually varying. First, from the roll and pitch records obtained with the ship's motion recorder, the angles of roll were obtained by measuring the swings from peak to peak. Thus for the run the total angle of swing was obtained together with the number of swings and time of the run. From this the average period of stabilization was computed. There were as many as five successive runs all under the same conditions with the stabilization system either on or off. The purpose here was to see the variation with time that might have been hidden in averages taken over longer periods of time. The results of these individual runs with the same conditions were summarized in Table 10 for the over-all period. The general

(Text continued on page 130.)

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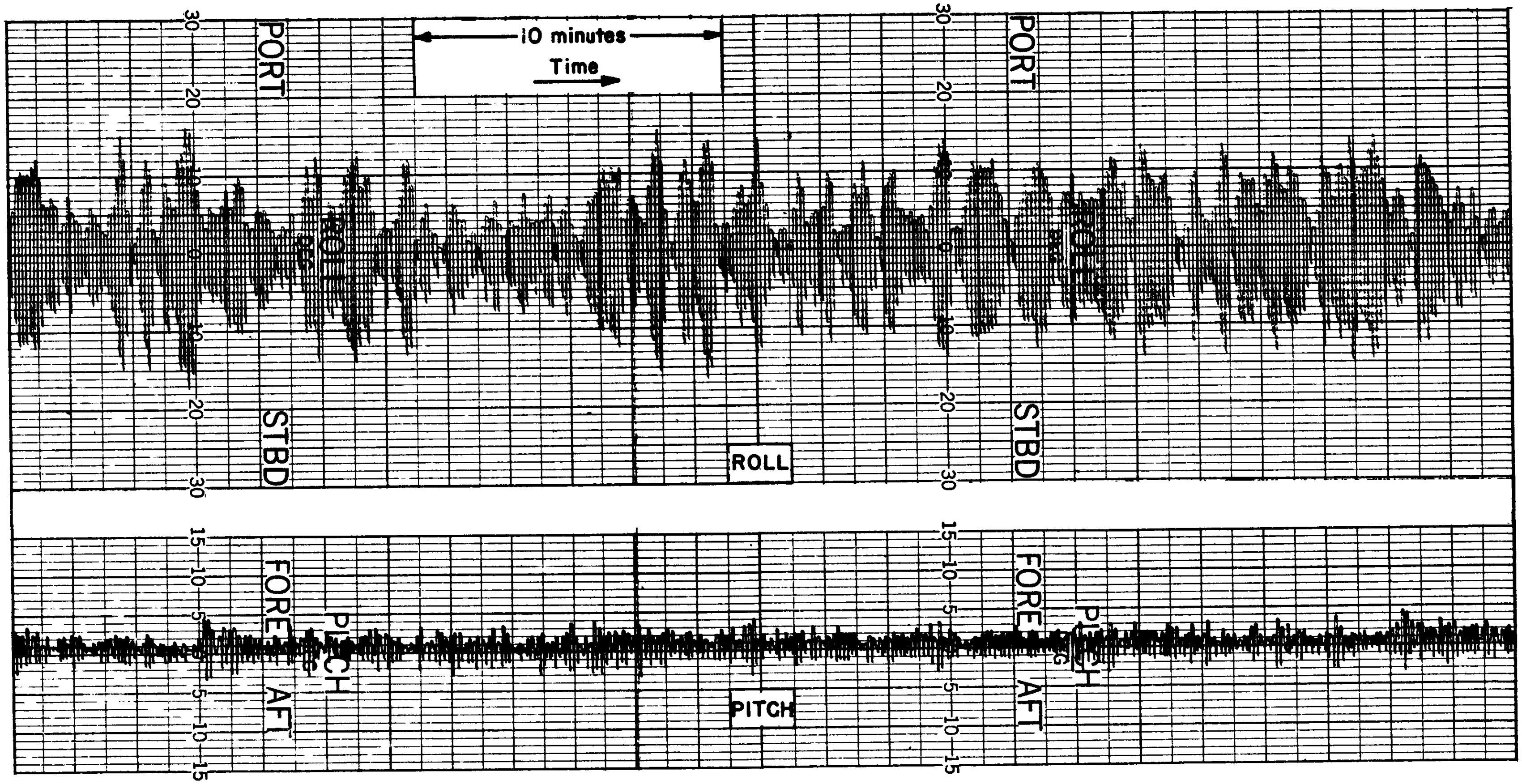


Figure 51 - Record of Roll and Pitch of the Unstabilized USS PEREGRINE (E-AM373) in a Seaway

TABLE 10

Average Angles and Periods of Roll and Percent Stabilization for Various Runs during Sea Tests 2 and 3

Run	Time: Nominal	Time: Seconds	Stabilizing Conditions: Impeller Units	Stabilizing Conditions: Control Phase Setting $\theta + \theta$ degrees	Total Angle of Roll degrees	Cycles	Average Period of Roll seconds	Average Amplitude of Roll °SA	Stabilization: Percent	Stabilization: Relative to Run:	Maximum Roll ° single amplitude	Difference Between Stabilized and Unstabilized Runs °SA	Ship Speed knots	Course	Wind T	Wind Velocity knots	Sea T	Sea Amt.	Swell T	Swell Amt.
Sea Test 2						30 November 1949					Impeller Blades Uncut									
D_0	10^{30}-10^{32}	216	S	40	216.5	30	7.2	1.81	36 7	F	4 1/2	1.05	6.5	120	315	8	293	2	281	1
D	10^{32}-10^{40}	559	S	40	610.8	71	7.88	2.15	24.8	F	5 1/2	0.71								
D_0+D		775			827.3	101	7.67	2.04	30.8	$F+F_1$	5 1/2	0.91								
E	10^{40}-10^{47}	554	S	50	480.9	73	7.45	1.64	42.6 44.4	F $F+F_1$	5	1.22 1.31	"	"	"	"	"	"	"	"
F	10^{47}-10^{57}	567	Off		868.6	76	7.45	2.86			8		"	"	"	"	"	"	"	"
F_1	10^{57}-11^{08}	607	Off		992.8	81.5	7.45	3.05			8									
$F+F_1$		1174			1861.4	157.5	7.47	2.95			8									
F_2	11^{08}-11^{12}	302	S	50	300.3	38.5	7.87	1.95	36.0	F_1	4 1/2	1.10	"	"	"	"	"	"	"	"
G_1	11^{14}-11^{22}	487	S	50	460.9	65.5	7.43	1.76	42.3 29.6	F_1 G_3	5 1/2	1.29 0.74								
G_2	11^{23}-11^{31}	465	S	50	551.9	64	7.26	2.15	14.0	G_3	6	0.35								
$F_2+G_1+G_2$		1254			1313.1	168.5	7.48	1.96	33.6 5.0	$F+F_1$ $G_3+G_4+G_5$	6	0.99 0.23								
G_3	11^{31}-11^{39}	461	Off		611.5	61	7.57	2.50	21.6	G_3	6 1/2	0.54	"	"	"	"	"	"	"	"
G_4	11^{39}-11^{50}	716	Off		725.5	92	7.79	1.97			5 1/2									
G_5	11^{50}-12^{03}	794	Off		887.4	100.5	7.90	2.21			6									
$G_3+G_4+G_5$		1971			2224.4	253.5	7.78	2.19			6 1/2									
H_1	12^{18}-12^{27}	482	Off		539.9	63	7.64	2.14			6 1/2		"	"	"	6	"	"	"	"
H_2	12^{27}-12^{32}	447	Off		467.4	58	7.72	2.01			5									
H_1+H_2		929			1007.3	121	7.67	2.08			6 1/2									
I	12^{33}-12^{44}	562	S	50	475.3	82.5	6.81	1.44	30.7 30.4	H_1+H_2 J	4	0.64 0.22	"	"	"	"	"	"	"	"
J	12^{45}-12^{50}	262	Off		293.5	35.5	7.38	2.07			4 1/2		"	"	"	"	"	"	"	"
K	12^{50}-12^{5}	370	S	0	320.9	51.5	7.19	1.56	24.6	J	3 1/2	0.51	"	"	"	"	"	"	"	"
L	12^{56}-1^{00}	572	S	40	467.7	78.5	7.28	1.48	28.5	J	3 1/2	0.59	"	"	"	"	"	"	"	"
L_1	1^{06}-1^{10}	312	S	40	273.1	43.5	7.18	1.57	24.2	J	3	0.56	6.8			4				
L_2	1^{17}-1^{24}	586	S	40	558.7	78.5	7.47	1.78	14.0	J	4	0.29								
$L+L_1+L_2$		1470			1299.5	200.5	7.43	1.62	21.7	J	4	0.45								
L_3	1^{42}-1^{52}	519	S	40	329.1	74.5	6.97	1.10	46.9	J	2 1/2	0.97	"	"	"	"	"	"	"	"
M	1^{54}-1^{58}	386	S	40	302.6	53.5	7.22	1.42	31.4	J	3 1/2	0.65								
L_3+M		905			631.7	128.0	7.07	1.23	40.6	J	3 1/2	0.84								
$L+L_1+L_2$ $+L_3+M$		2375			1931.2	328.5	7 23	1.47	29.0 24.6	J $O+O_1+O_2+O_3$	4	0.60 0.48								
N	2^{01}-2^{09}	490	Off*		583.8	78	6.28	1.87			5		6.9	120	270	4	236	2	225	2
N_1	2^{09}-2^{15}	310	Off*		318.5	41.5	7.47	1.92			4 1/2									
$N+N_1$		800			902.3	119.5	6.69	1.89			5									
O	2^{15}-2^{22}	492	Off		422.3	63.5	7.76	1.66			4		"	"	"	"	"	"	"	"
O_1	2^{22}-2^{30}	506	Off		461.1	64.	7.91	1.80			5									
O_2	2^{30}-2^{41}	542	Off		608.1	72	7.53	2.11			6									
O_3	2^{41}-2^{50}	624	Off		715.8	83	7.52	2.16			5 1/2									
$O+O_1+O_2+O_3$		2164			2207.3	292.5	7.41	1.95			6									
P	2^{50}-2^{54}	208	Off		263.5	26.5	7.85	2.49			5		"	"	"	"	"	"	"	"
P_1	2^{54}-2^{57}	216	S	40	253.1	31	6.97	2.04	18.1	P	5	0.45	"	"	"	"	"	"	"	"
Q	$2^{57\frac{1}{2}}$-3^{05}	458	S	40	348.8	65	7.06	1.48			3 1/2		6 9	120	320	5	315	2	315	"
Q_1	3^{05}-3^{16}	636	S	40	511.8	83.5	7.63	1.53			3 1/2									
$Q+Q_1$		1094			896.6	148.5	7.38	1.51	39.4 72.6	P $O+O_1+O_2+O_3$	3 1/2	0.98 0.44								
Q_2	3^{16}-3^{26}	552	Off		632.7	73	7.56	2.17	32.5	Q_2+Q_3	5 1/2	0.73	"	"	"	"	"	"	"	"
Q_3	3^{26}-3^{35}	617	Off		747.4	81	7.62	2.31			5									
Q_2+Q_3		1169			1380.1	154.7	7.59	2.24			5 1/2									
Sea Test 2						1 December 1949							**Impeller Blades Uncut**							
A_{00}	12^{00}-12^{12}	807	Off		1457.9	103	7.83	3.54			10 1/2		9.0	170	340	18	315	3	315	1
A_0	12^{12}-12^{19}	461	Off		905.8	59	7.81	3.84			8 1/2									
A_1	12^{19}-12^{28}	569	Off		1098.2	77	7.40	3.57			10									
$A_{00}+A_0+A_1$		1837			3461.9	239	7.69	3.62			10 1/2									
A_2	12^{36}-12^{44}	576	P & S	40	827.4	73.5	7.84	2.81			8		"	"	"	"	"	"	"	"
A_{2-1}	12^{44}-12^{55}	629	P & S	40	875.9	86.5	7.27	2.53			7									
A_{2-2}	12^{55}-1^{05}	610	P & S	40	772.8	82	7.44	2.36			7		9.0		350	18	326	3	326	3
A_3	1^{05}-1^{13}	456	P & S	40	614.7	60	7.60	2.56			6 1/2									
A_2+A_{2-1} $+A_{2-2}+A_3$		2271			3090.8	302.0	7.52	2.56	29.3 19.5	$A_{00}+A_0+A_1$ A_4 - A_8	8	1.06 0.62								
A_4	1^{14}-1^{24}	567	Off		755.5	65.5	8.67	2.88			7 1/2		"	"	"	"	"	"	"	"
A_5	1^{24}-1^{36}	675	Off		884.5	84.5	7.99	2.62			8 1/2									
A_6	1^{36}-1^{43}	485	Off		803.8	58	8.36	3.46			7 1/2									
A_7	1^{43}-1^{55}	756	Off		1258.4	89	8.49	4.45			10									
A_8	1^{55}-2^{24}	976	Off		1610.3	120.5	8.10	3.34			9							4		3
A_4-A_8		3459			5312.5	417.5	8.28	3.18			10									
B_1	2^{24}-2^{35}	569	S	50	881.9	72	7.90	3.06	3.8 12.3	A_4 - A_8 B_2 - C_1	8 1/2	0.12 0.43	"	"	"	"	"	"	"	"
B_2	2^{35}-2^{43}	569	Off		1026.9	71	8.01	3.62			10		"	"	"	"	"	"	"	"
B_3	2^{43}-2^{55}	648	Off		1161.9	84	7.71	3.46			8									
B_4	2^{55}-3^{05}	576	Off		1048.6	79.5	7.25	3.30			9 1/2							3		
C_1	3^{05}-3^{14}	569	Off		1116.9	77	7.39	3.63			9									
B_2-C_1		2352			4354.3	311.5	7.56	3.49			10									
C_2	3^{14}-3^{23}	572	S	50	898.7	79	7.24	2.84			7 1/2		"	"	"	"	"	"	"	"
C_3	3^{23}-3^{30}	358	S	50	551.3	46	7.83	3.00			7									
C_2+C_3		930			1450.0	125	7.44	2.90	16.9 8.8	B_2 - C_1 C_4	7 1/2	0.59 0.28								
C_4	3^{32}-3^{53}	1200	Off		1909.0	150	8.00	3.18			9		8 9	"	"	"	"	"	"	2

*Unstabilized except for small float level signal

TABLE 10 (continued)

Run	Time: Nominal	Time: Seconds	Stabilizing Conditions: Impeller Units	Stabilizing Conditions: Control Phase Setting $\dot{\theta}+\theta$ degrees	Total Angle of Roll degrees	Cycles	Average Period of Roll seconds	Average Amplitude of Roll °SA	Stabilization: Percent	Stabilization: Relative to Run:	Maximum Roll ° single amplitude	Difference Between Stabilized and Unstabilized Runs °SA	Ship Speed knots	Course	Wind T	Wind Velocity knots	Sea T	Sea Amt.	Swell T	Swell Amt.
Sea Test 3										5 March 1950			**Impeller Blades Cut**							
1	$1^{58}-2^{01}$	290	P & S	50	326.2	37	7.84	2.20	39.6	2	6 1/2	1.24	7.5	315	115	7	90	1	68	1
2	$2^{01}-2^{06}$	192	Off		364.4	25	7.68	3.64			8 1/2									
3	$2^{06}-2^{11}$	360	P & S	50	590.1	46	7.83	3.24	11.0 28.5	2 4	7 1/2	0.40 1.29								
4	$2^{11}-2^{16}$	204	Off		444.3	24.5	8.32	4.53			8 1/2									
5	$2^{16}-2^{24}$	526	P & S	60	789.4	67	7.85	2.95	34.9 26.3	4 6	8	1.58 1.05								
6	$2^{24}-2^{29}$	302	Off		640.1	40	7.55	4.00			8 1/2									
7	$2^{41}-2^{53}$	710	P & S	60	771.5	86.5	8.21	2.23	44.3 17.8	6 8	5	1.77 0.58	8.2	325	"	9	236	2	225	2
8	$2^{53}-3^{09}$	986	Off		1411.1	125.5	7.86	2.81			8 1/2									
9	$3^{09}-3^{23}$	840	Off*		1525.9	106.5	7.89	3.58			8 1/2									
10	$3^{26}-3^{49}$	980	P & S	60	1067.3	118.5	8.27	2.25	37.2 32.6	9 11	5 1/2	1.33 1.09								
11-1			Off		1448.6	105		3.45					7.5	"	"	"	23	"	236	"
11-2			Off		1527.9	117.5		3.25												
11	$3^{49}-4^{21}$	1815			2976.5	222.5	8.16	3.34			10 1/2									
12	$4^{21}-4^{26}$	348	P & S	60	672.7	48	7.25	3.70	-9.7 24.8	11 14	10 1/2	-0.36 1.22	"	"	"	"	"	"	"	"
13-1			P & S	60	1504.4	105		3.58					"	"	"	"	"	"	"	"
13-2			P & S	60	1182.6	97		3.05												
13	$4^{30}-4^{55}$	1573			2687.0	202	7.79	3.33	0 32.4	11 14	9	0 01 1.59								
14	$4^{55}-5^{01}$	537	Off		1347.7	68.5	7.84	4.92			10 1/2		"	"	"	"	"	"	"	"
Sea Test 3										6 March 1950			**Impeller Blades Cut**							
1-1			Off		1124.4	75		3.75					5.5	145	320	15	90	1	68	1
1-2			Off		854.8	60		3.56												
1-3			Off		1016.3	70.5		3.61												
1	$10^{15}-10^{41}$	1487	Off		2995.5	205.5	7.24	3.69			9									
2-1			P & S	50	869.4	75		2.90		1			"	"	"	"	"	"	"	"
2-2			P & S	50	771.8	75		2.57												
2-3			P & S	50	920.1	65		3.54												
2	$10^{41}-11^{07}$	1483	P & S	50	2561.3	215	6.90	2.98	19.2	1	8	0.71								
3-1			Off		1015.3	75		3.38					7.8	"	"	"	"	"	225	"
3-2			Off		930.3	60		3.88												
3-3			Off		1123.5	60		4.68												
3-4			Off		1402.4	75		4.67												
3	$11^{07}-11^{41}$	2045	Off		4472.5	270	7.59	4.14			10 1/2									
4-1			P	50	1306.6	75		4.36					"	"	"	"	"	"	"	"
4-2			P	50	1090.5	60.5		4.51												
4	$11^{41}-11^{57}$	982	P	50	2397.1	135.5	7.24	4.42	5.4 13.2	3-4 5-1	9 1/2	0.25 0.67								
5-1			Off		1525.7	75		5.09					4.5	"	"	"	236	"	236	"
5-2			Off		1136.3	77.5		3.67												
5	$11^{57}-12^{15}$	1090	Off		2662.0	152.5	7.15	4.36			10 1/2									
6	$12^{15}-12^{26}$	569	S	50	1660.1	82	6.94	5.06	-27.5 -7.9	5-2 7-1	11 1/2	-1.39 -0.40	"	"	"	"	"	"	"	"
7-1			Off		1677.1	90		4.66					"	"	"	"	"	"	"	"
7-2			Off		2102.0	88		5.97												
7	$12^{26}-12^{49}$	1341	Off		3779.1	177	7.58	5.34			13 1/2									
8-1			P & S	50	920.6	45		5.11					"	"	"	"	"	"	"	"
8-2			P & S	50	1524.3	59		6.46												
8	$12^{49}-1^{01}$	769	P & S**	50	2444.9	104	7.40	5.88	-9.2 -14.6	7 9+10	14 1/2	-0.54 -0.86								
9	$1^{01}-1^{07}$	341	Off		796.5	43.5	7.83	4.58			8 1/2		"	"	"	"	"	"	"	"
10	$1^{07}-1^{18}$	776	Off***		2106.1	94	8.26	5.60			13		6.5	"	"	"	"	"	"	"
11-1			P & S	50	1457.6	90		4.05					"	"	"	"	"	"	"	"
11-2			P & S	50	1921.7	105		4.58												
11	$1^{18}-1^{42}$	1484	P & S	50	3379.3	195	7.62	4.33	22.7	10	12 1/2	1.27								
12	$1^{42}-1^{52}$	582	P & S	40	1587.4	71	8.20	5.59	1.0	10	12	0.01	"	"	"	"	"	"	"	"
13	$1^{52}-2^{02}$	536	P & S	30	1368.3	67.5	7.95	5.07	9.5	10	12	0.53								
14	$2^{02}-2^{13}$	715	P & S	20	2082.9	91	7.86	5.72	-2.1 1.9	10 15	16	-0.12 0.11								
15-1			Off		1641.4	75		5.47					"	"	"	"	"	"	"	"
15-2			Off		1531.3	66.5		5.59												
15	$2^{13}-2^{34}$	1159	Off		3172.7	141.5	8.19	5.61			12									
16	$2^{34}-2^{41}$	463	Off†		1027.6	55.5	8.36	4.63			13		"	"	"	"	"	"	"	"
17-1			Off		1150.1	90		3.19					10.0	165	"	"	"	"	"	"
17-2			Off		1130.9	93		3.04												
17	$2^{41}-3^{07}$	1560	Off††		2281.0	183	8.53	3.10			10 1/2									
18-1			P & S	50	823.3	75		2.74					"	"	"	"	"	"	"	"
18-2			P & S	50	933.9	81		2.88												
18	$3^{11}-3^{34}$	1340	P & S	50	1757.2	156	8.59	2.82	9.0	17	8	0.28								
19	$3^{34}-3^{43}$	516	P & S†††	50	738.3	62	8.33	2.98	3.9 19.9	17 20	8	0.12 0.74	"	"	"	"	"	"	"	"
20	$3^{43}-3^{55}$	771	Off		1374.7	92.5	8.33	3.72			9 1/2									
21	$4^{03}-4^{09}$	434	P & S	30	674.9	51	8.51	3.31	11.0 -4.8	20 22	9	0.41 -0.16	9.8	"	"	"	"	"	"	"
22	$4^{09}-4^{20}$	718	Off		1166.0	92.5	7.78	3.15			8 1/2									
23	$4^{20}-4^{25}$	161	P & S	60	296.0	20.5	7.86	3.61	-13.7 0	22 26	9	-0.46 0								
24	$4^{25}-4^{29}$	247	P & S	40	475.1	32	7.72	3.71	-15.1 - 3.7	22 26	8	-0.56 -0.10								
25	$4^{29}-4^{44}$	977	P & S	50	1417.4	121.5	8.04	2.91	7.6 19.3	22 26	7 1/2	0.24 0.70								
26	$4^{44}-4^{59}$	873	Off		1631.3	113	7.72	3.61			10		9.9	"	"	"	"	"	"	"

*Impellers off, changed course and reduced speed

**Decreased damping of mechanical filter

***Changed speed

†Changed speed during the run (changing engines)

††Changed course and speed

†††Increased amplitude setting of control system

RESTRICTED

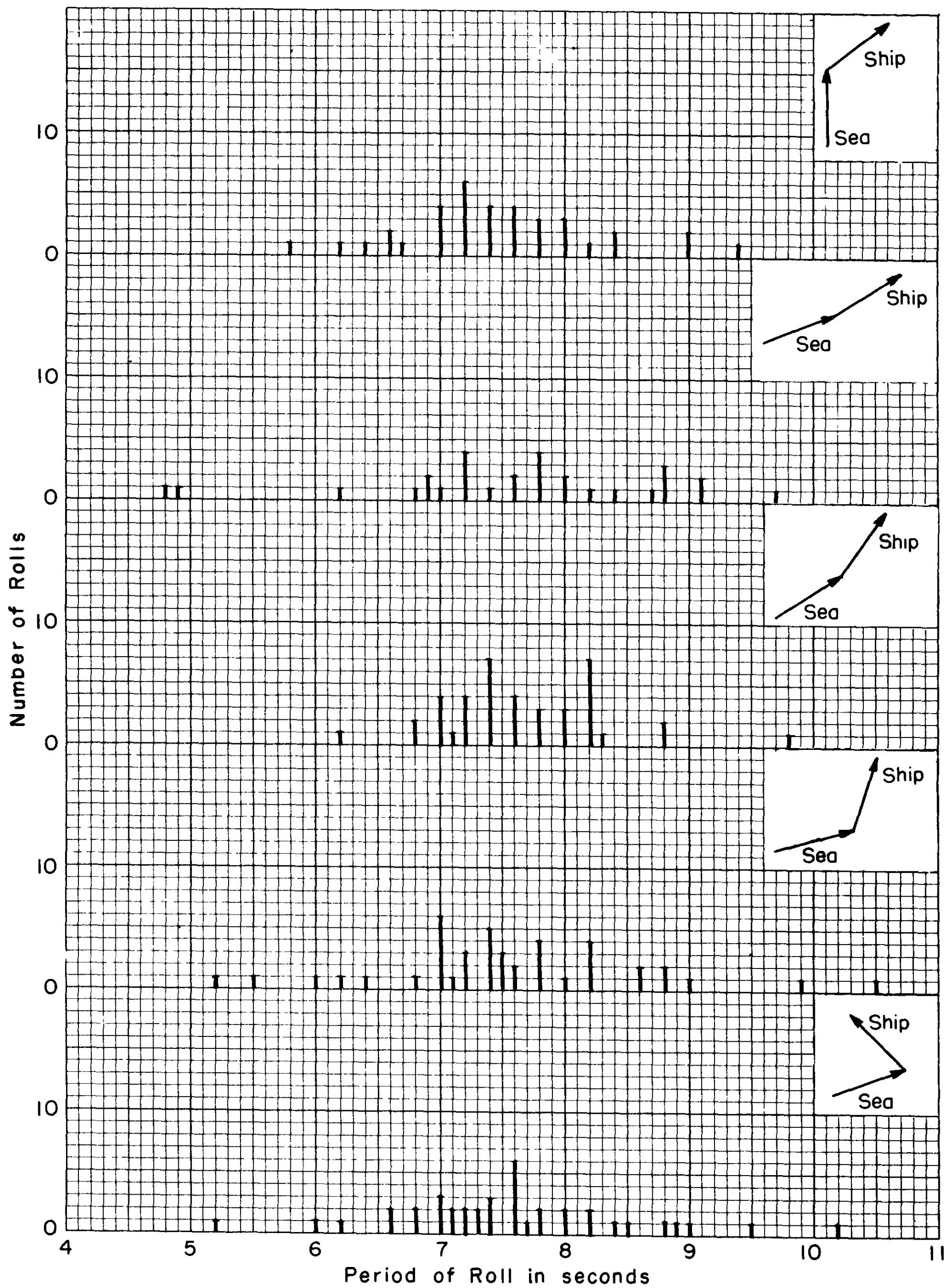
Time 8:55 - 9:00
Course 065° T
Speed 5 knots
Wind 180° T
Sea 180° T
Swells 1-2
Sea 2
36 cycles

Time 9:31 - 9:55
Course 045° T
Speed 5 knots
Wind 280° T
Sea 265 T
Swells 1
Sea 2
40 cycles

Time 9:50 - 9:55
Course 025° T
Speed 5 knots
Wind 270° T (20 kt)
Sea 240 T
Swells 1-2
Sea 3
40 cycles

Time 10:21 - 10:27
Course 005° T
Speed 5 knots
Wind 260° T (15 kt)
Sea 260 T
Swells 1-2
Sea 2
41 cycles

Time 10:49 - 10:55
Course 352° T
Speed 5 knots
Wind 260° T
Sea 260 T (15 kt)
Swells 1-2
Sea 2
40 cycles



Time 11:10 - 11:15
Course 300° T
Speed 5 knots
Wind 240° T (20 kt)
Sea 260 T
Swells 1-2
Sea 2
40 cycles

Time 12:16 - 12:21
Course 045° T
Speed 5 knots
Wind 230° T (13 kt)
Sea 230 T
Swells 2
Sea 2
41 cycles

Time 12:34 - 12:39
Course 045° T
Speed 7 knots
Wind 240°T (13 kt)
Sea 230 T
Swells 2
Sea 2
41 cycles

Time 13:05 - 13:10
Course 045° T
Speed 9 knots
Wind 245° T (13 kt)
Sea 245 T
Swells 3 (235 T)
Sea 2
41 cycles

Time 13:29 - 13:36
Course 045° T
Speed 11 k ots
Wind 250° T (16 kt)
Sea 235 T
Swells 3
Sea 3
56 cycles

Time 14:03 - 14:10
Course 045° T
Speed 3 knots
Wind 245° T (18 kt)
Sea 235 T
Swells 3
Sea 3
52 cycles

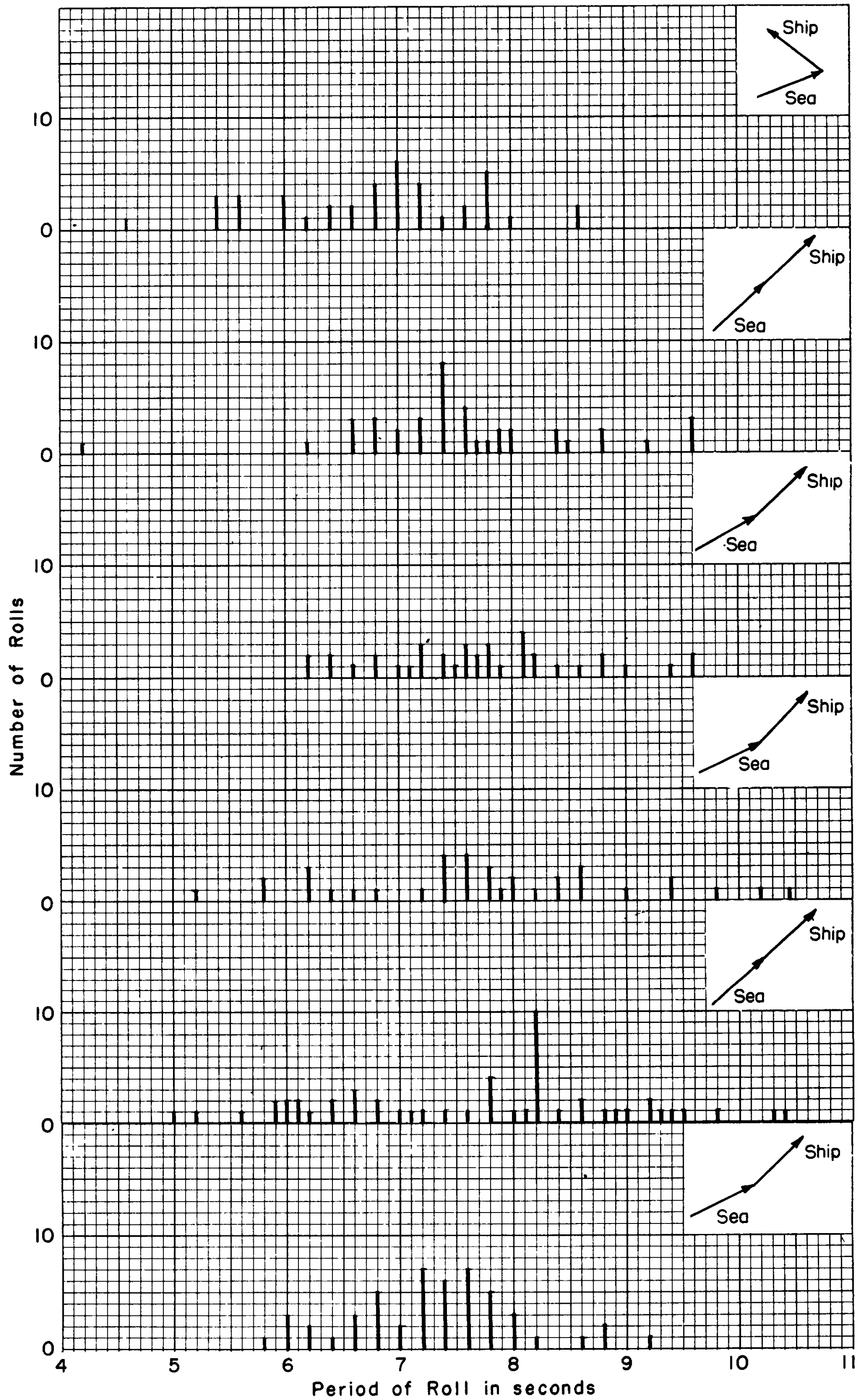


Figure 52 - Incidence of Rolling Periods for Various Aspects of Sea to Ship and for Various Ship Speeds

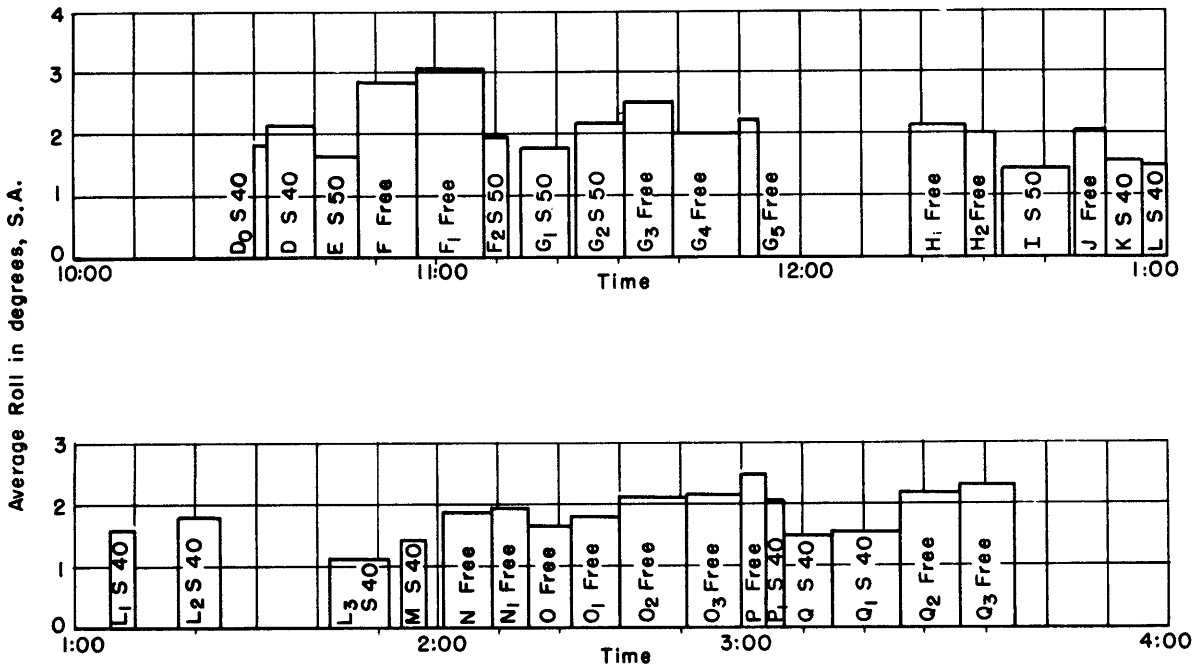


Figure 53a - Sea Test 2, 30 November 1949, Before Cutting of Impeller Blades

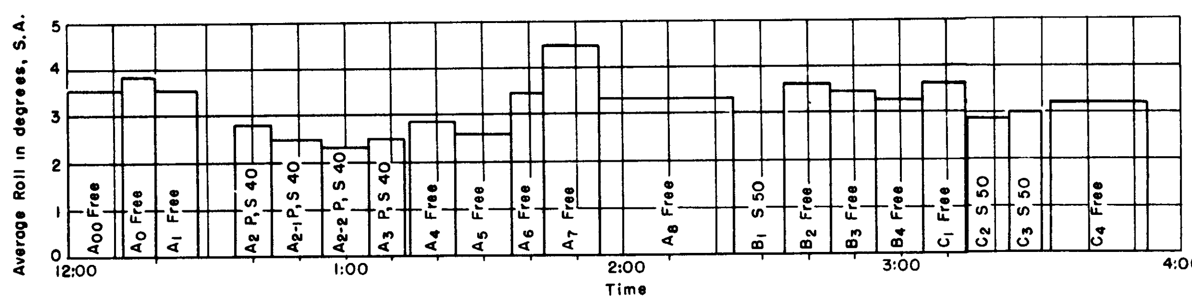


Figure 53b - Sea Test 2, 1 December 1949, Before Cutting of Impeller Blades

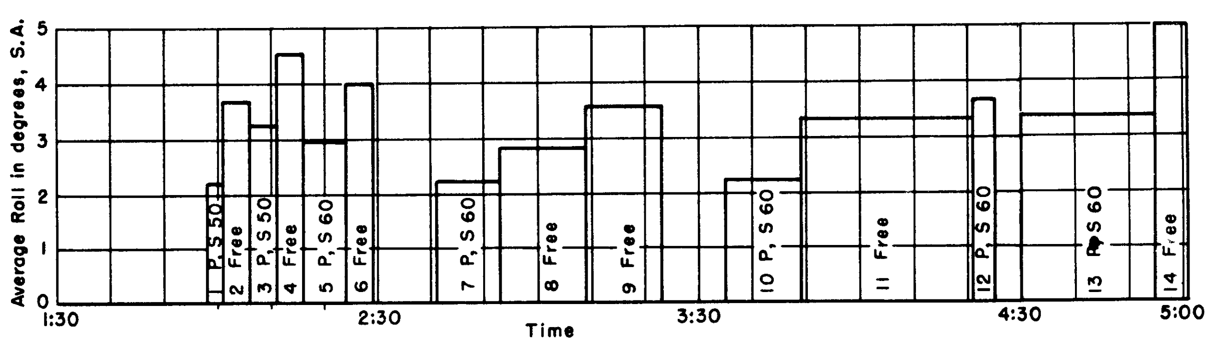


Figure 53c - Sea Test 3, 5 March 1950, After Cutting of Impeller Blades

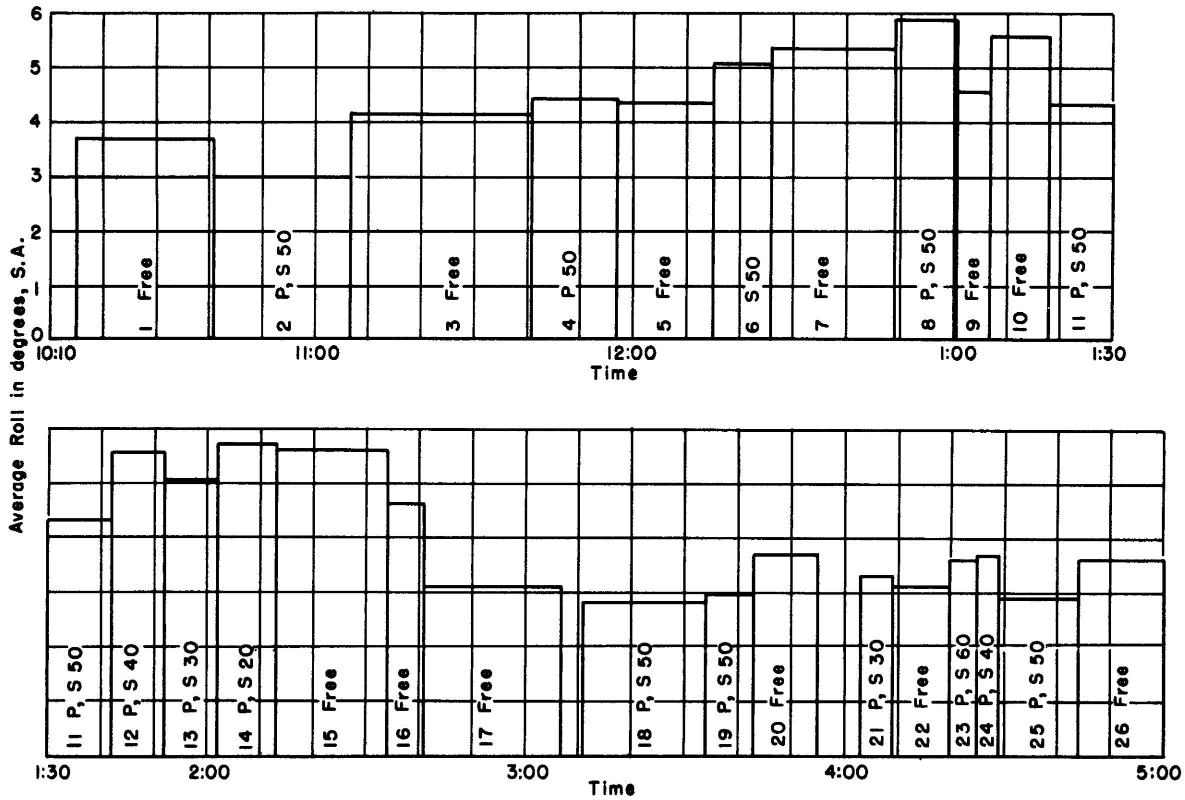


Figure 53d - Sea Test 3, 6 March 1950, After Cutting of Impeller Blades

Figure 53 - Graphical Summary of Effect of Stabilization during Sea Tests 1 and 2

The average roll in single amplitude for each run and averages for similar contiguous runs are plotted as step graphs on a nominal time base. The runs and the conditions of stabilization or of free roll are noted on each graph. Separate graphs are plotted for each day that data were analyzed.

procedure was to run alternately with stabilizing equipment operating and with it off. On Sea Test 2 there was considerable difficulty in trying to keep the pitch mechanism of the port impeller operating. The breaker for this unit kept kicking out, and when it did run it did not operate as smoothly as the other unit. The difficulty was caused by the shifting of bolts and consequent misalignment of the mechanism; this was corrected for Sea Test 3. This malfunction caused erratic time lags during a cycle of rolling. On Sea Test 2, 1 December 1949, it was possible to run both impellers for a short time, but the percent stabilization was less than for the starboard impeller alone. On this day, too, the equipment was run for longer times without any change in operating condition. The transfer of water was in no case greater than 2 1/2 ft double amplitude. On Sea Test 3, both units were operated together, but the blades had been cut, resulting in less effective stabilization per unit. All this time the phase advance was set at optimum setting, which was usually a nominal 40° or 50°. On 6 March records were obtained with variation of this advance greater and less than the optimum setting.

The final evaluation of the ship-stabilizing system is its behavior in a seaway. It is difficult to assess the performance of this system because of the variation of the many parameters involved. Even the counting of cycles on the roll and pitch records, in order to determine the average period of roll for an interval, is uncertain since the questions arise: "What are cycles and what are erratic variations within a cycle?" It was assumed that any variation greater than 1/2° single amplitude was a cycle unless it was superimposed on cycles of larger amplitudes or occurred in intervals of time short compared with those of the larger-amplitude cycles.

The range of average periods encountered for different intervals on the two sea tests was from 8.6 to 6.3 seconds. It must be borne in mind that the ship's course and speed were altered to make this period as long as possible to get greater stabilization. This was necessary since in the vicinity of a period of 7 seconds the pumping effectiveness of the impellers dropped rapidly. The rolling periods for the various intervals give a clue to the uniformity of the sea on the various days of the sea tests. 1 December 1949 of Sea Test 2 and 5 March 1950 of Sea Test 3 were days when the sea conditions and the response of the PEREGRINE to them did not change much throughout the period of recording. 6 March 1950 of Sea Test 3 was especially bad in that, superposed over a sea of more or less regularity, were the effects of squalls of different intensities that appeared from time to time. Another measure of the uniformity of sea conditions is the maximum angle of roll for the various intervals. The squally condition may well account for the so-called negative stabilization obtained by comparing certain average amplitudes of rolling in

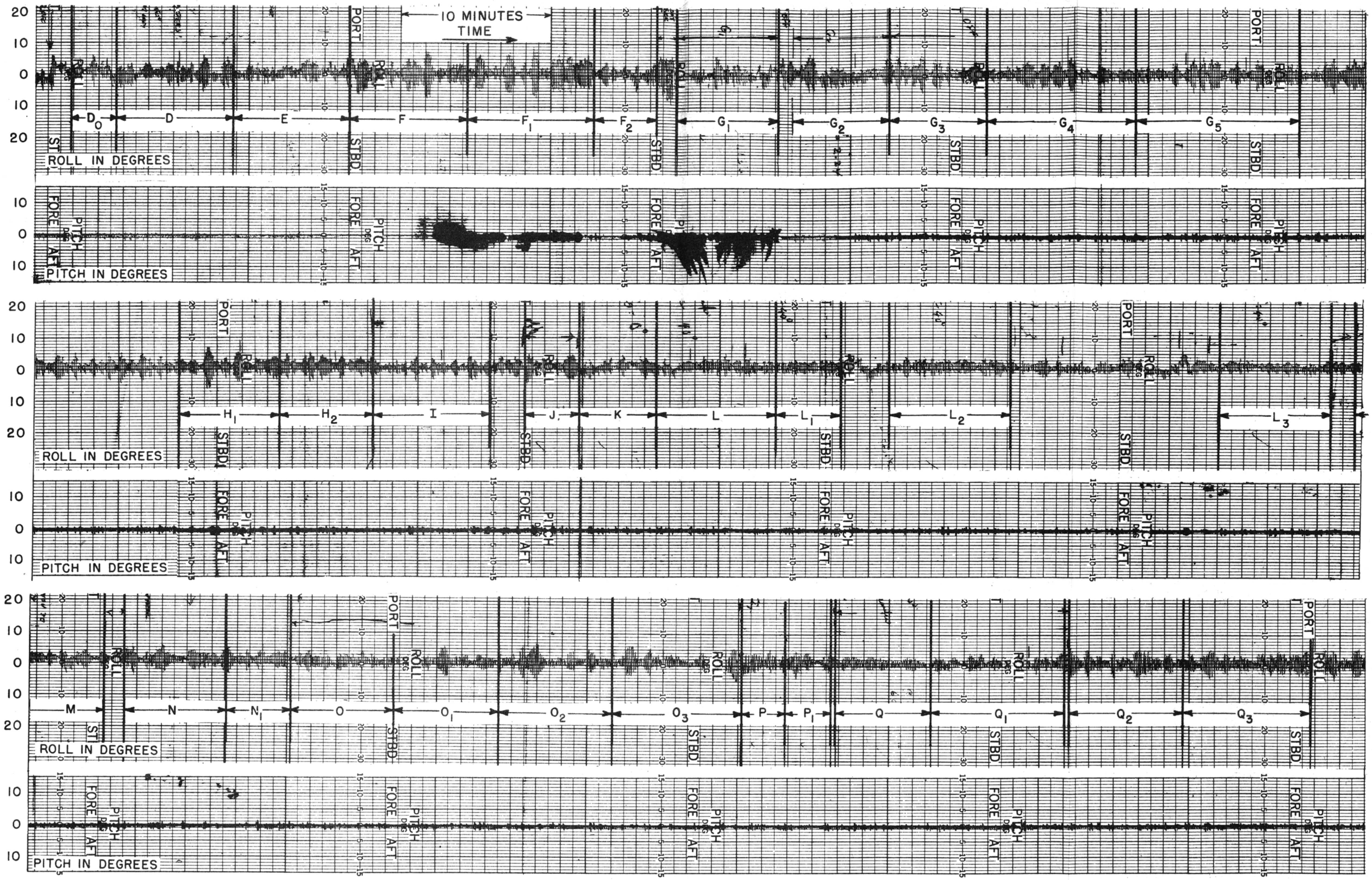


Figure 54a - Sea Test 2, 30 November 1949

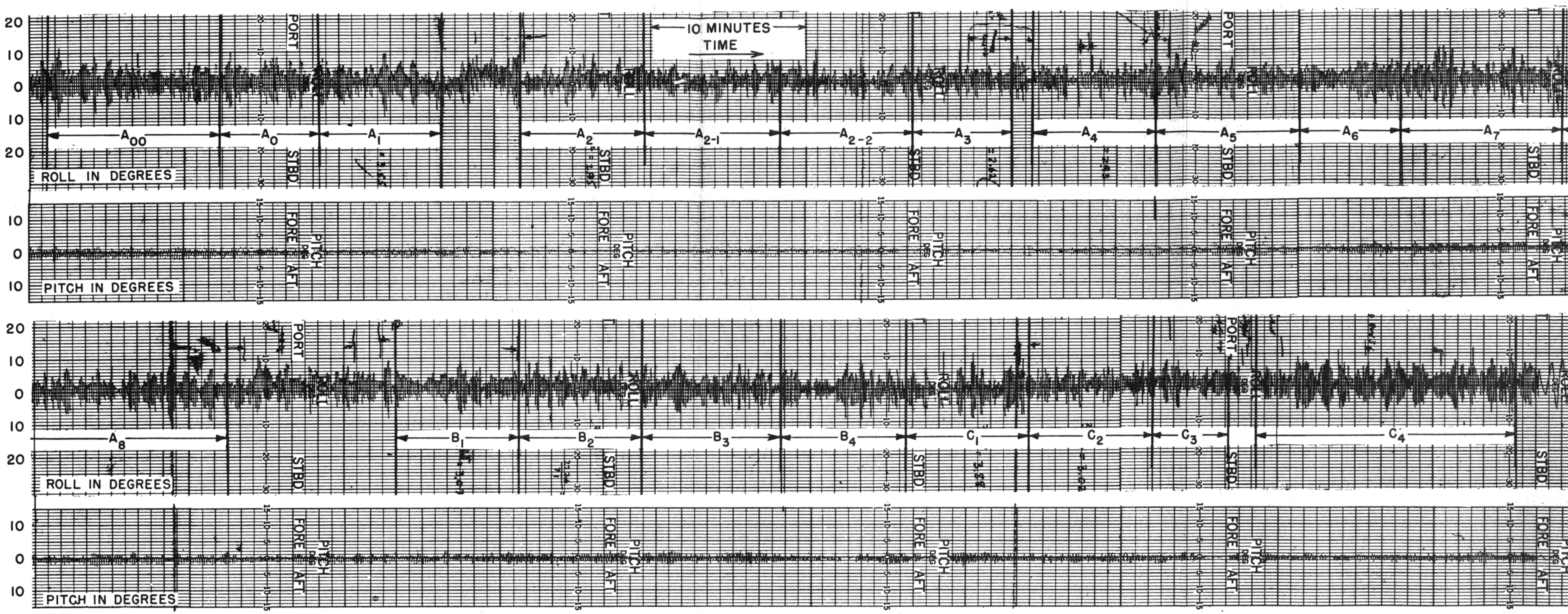


Figure 54b - Sea Test 2, 1 December 1949

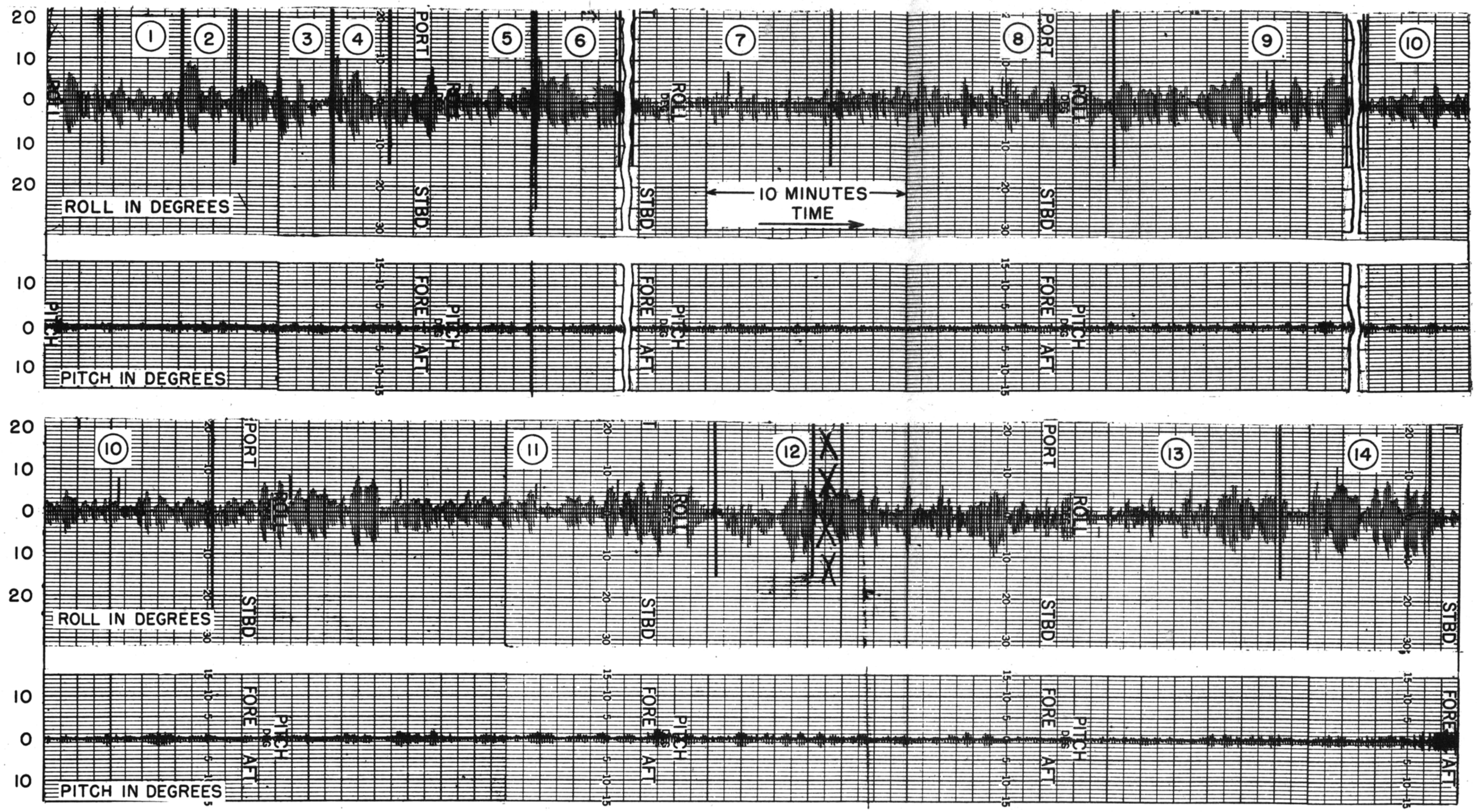


Figure 54c - Sea Test 3, 5 March 1950

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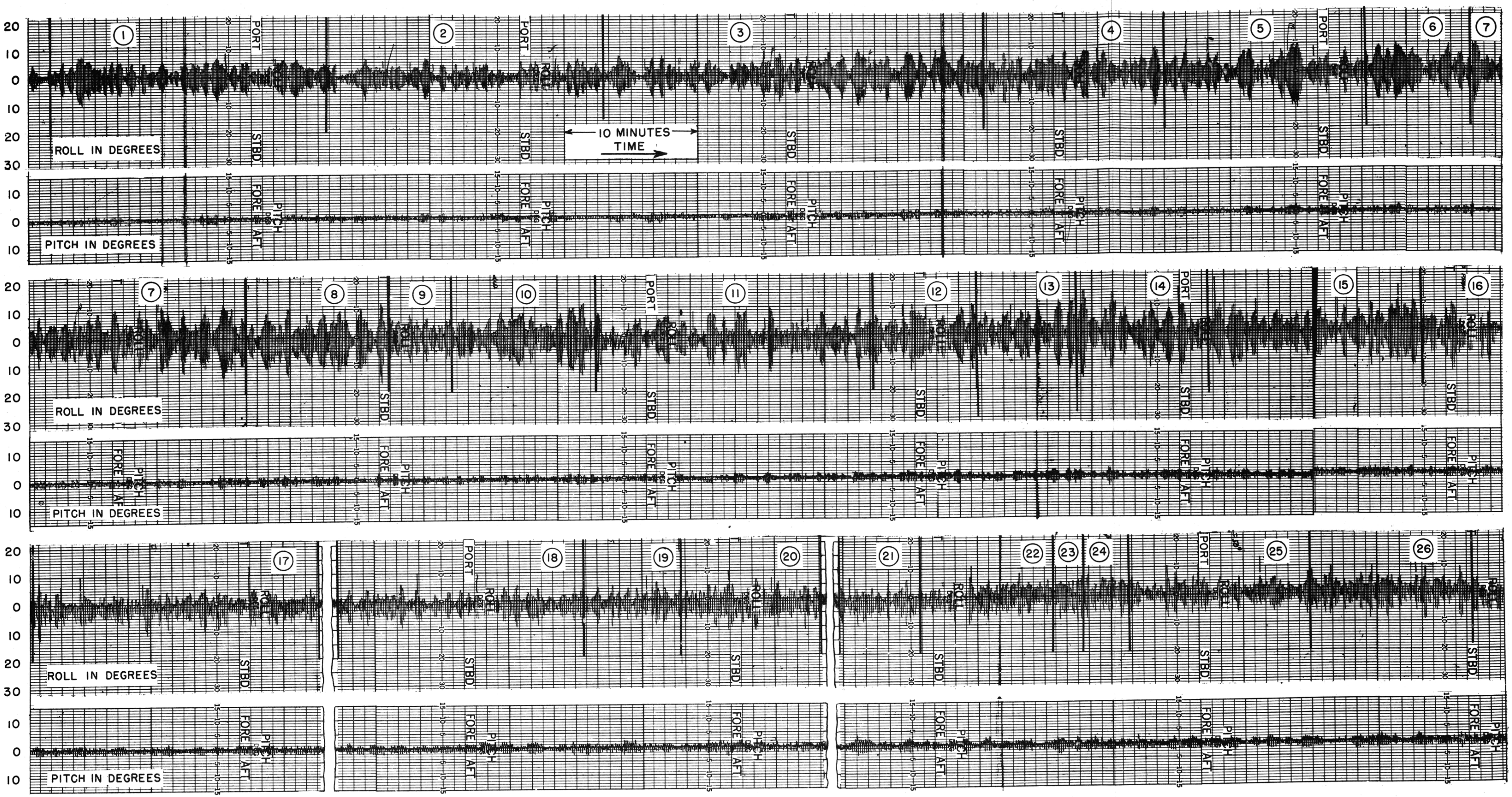

Figure 54d - Sea Test 3, 6 March 1950

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stabilized intervals with those of unstabilized intervals. Also, some of these negative intervals were too short and were at other than optimum phase advance. The average periods and amplitudes are listed in Table 10. Gradual changes in the sea conditions can be seen by examining the graphical summary of the effect of stabilization, Figure 53.

The effectiveness of stabilization was computed in two ways. In both, a comparison was made of the average roll during a stabilized interval with the average rolls of unstabilized intervals immediately before or after the stabilized interval. In the one method, the average difference was obtained. In the other, the average difference was divided by the larger of the two and percentage stabilization was computed. Both of these are listed in Table 10. The days when the sea conditions were not changing too rapidly are believed to have resulted in data that are better criteria for evaluating the activated tank system. These were 30 November 1949 and 5 March 1950. On the first day, 30 November, only one impeller was operated most of the time. On the percent basis, there was an improvement varying from 14 to 47 percent, mostly about 30 percent. On the difference basis, there was a reduction in rolling of about 0.7° single amplitude with a number of values clustered at 0.6° and 1.1°. On the second day, 5 March, both impellers were running, but the blades had been cut to shift the center of pressure. On the percent basis, there was an improvement varying from 10 to 45 percent, mostly about 30 percent. On the difference basis, the rolling was reduced about 1.3°, and the values were clustered about this figure. These difference values show that almost twice the stabilization was obtained with both impellers operating as with one. The percent basis does not show this because the average roll was about twice as great on 5 March as on 30 November.

These values for the sea tests indicate that stabilization was effective, but not nearly as much as it could have been because of the high metacentric height and the low rolling period of the ship, causing the impellers to work outside their range of maximum efficiency. The transfer of a maximum of 2 1/2 ft double amplitude of water in the tanks shows the effects. Distribution diagrams, Figure 55, show the effectiveness better perhaps than Table 10. In Figure 55 "percent stabilization" and "differences in rolling amplitude," taken from Table 10, are plotted for each day. Omitted are the negative percentages of stabilization which are believed to be meaningless and to be caused by fluctuations of the sea, such as occurred, particularly, on 6 March. Also omitted are certain values on 6 March when the phase advance was varied at too frequent intervals to give valid comparison data. Duplicates on the basis of several adjoining free intervals and the runs not at optimum phase advance were also omitted.

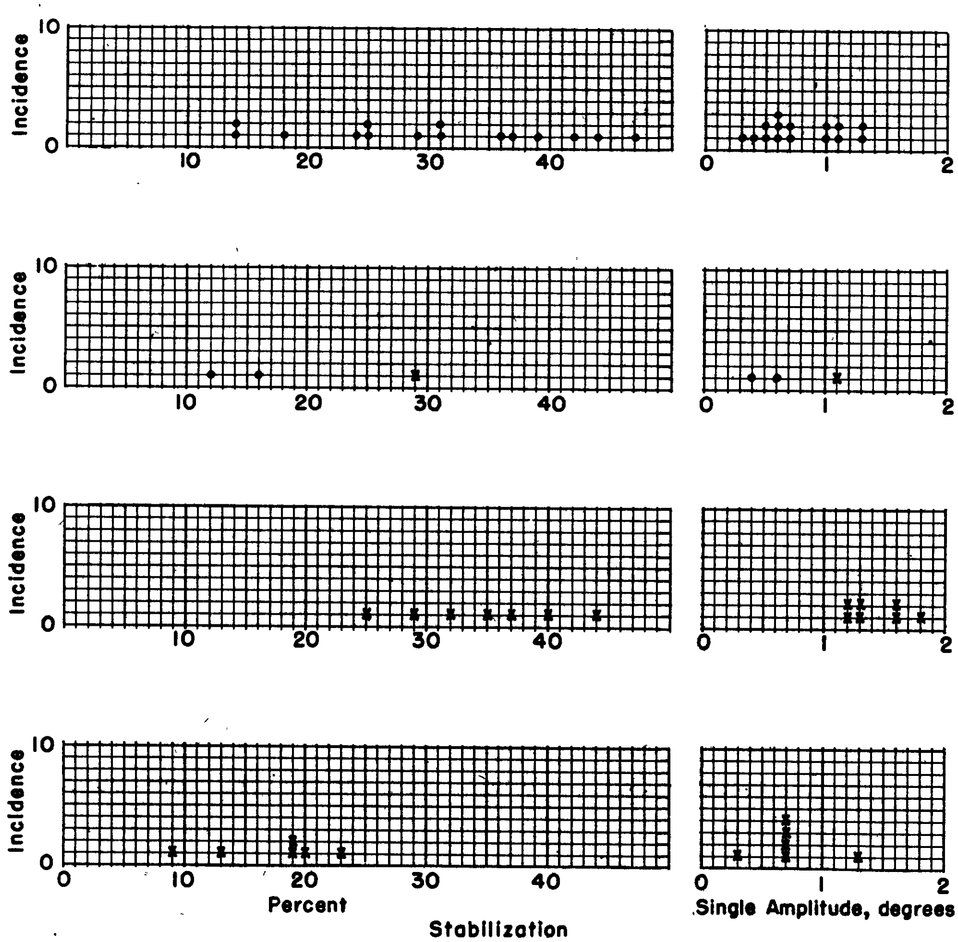

Figure 55 - Percent Stabilization and Reduction of the Average Angle of Roll on Sea Tests 2 and 3

One system operating •; Two systems operating x

To complete the presentation of representative data obtained during this project, an oscillogram, Figure 56, is reproduced showing the performance of the control system, pitch, and water level for Sea Test 3; see also Figure 54, Record 11 for the stabilized interval 1:18 to 1:42 P.M. and Table 10. Oscillograms of power and strain were obtained during the sea tests. However, the resulting data are similar to that obtained during the drydock tests, Figure 50b, except that the values were more erratic. The range of values was about the same, since the maximum pitch permitted during the sea tests was never greater than that during the drydock tests. Actually, it was kept somewhat lower to prevent the pitch motor from kicking out whenever an occasional large signal occurred.

Now that the characteristics of the sea, the principles of rolling and stabilization, the activated-tank installation on the PEREGRINE, and the performance of the installation with the ship in drydock, in calm water, and in rough water have been described, it is possible to discuss the advantages

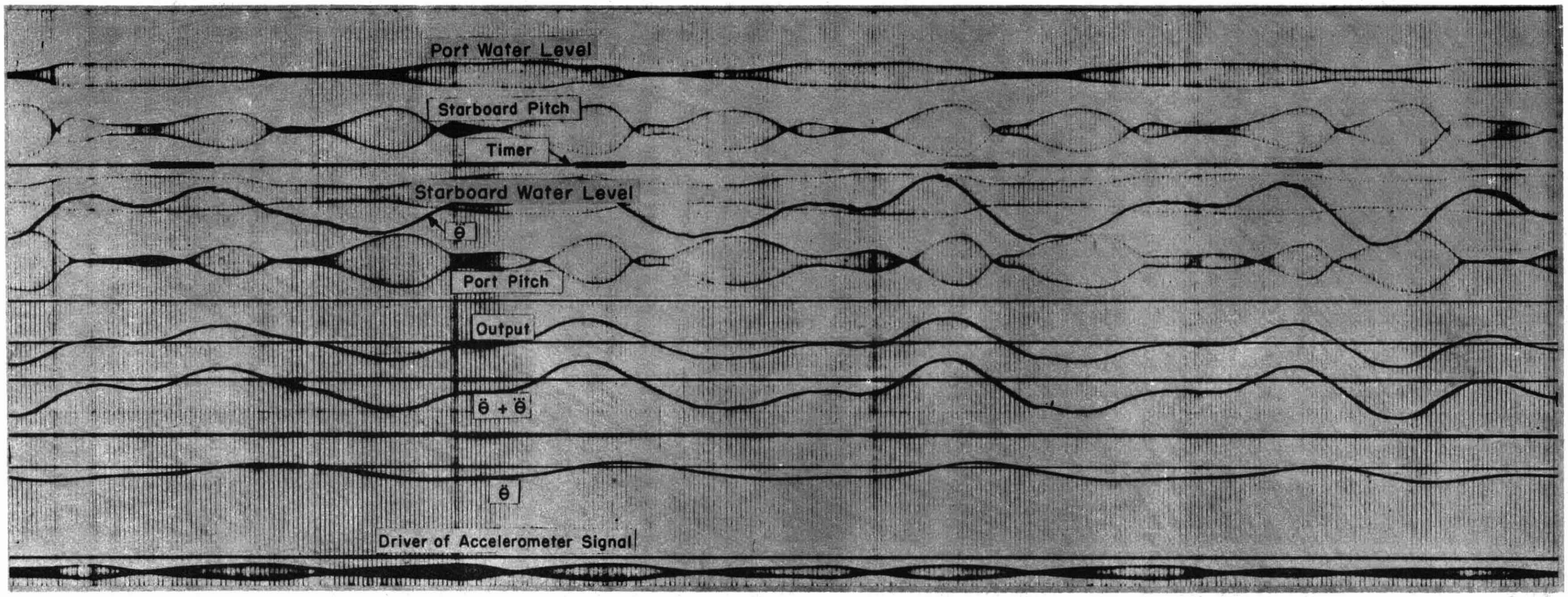

Figure 56 - Typical Oscillogram with Both Impellers Operating, Recorded on Sea Test 3, 6 March 1950, in the Interval 1:18 to 1:42 PM

$\ddot{\theta}$ (input), $\dddot{\theta}$, $\ddot{\theta} + \dddot{\theta}$, output, and port and starboard pitch, and water level were recorded.

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and shortcomings of this particular installation, to discuss its possibilities, and to suggest requirements for a permanent installation.

DISCUSSION

The calibration of the equipment and the results of the drydock, calm-water, and rough-water tests have indicated undesirable as well as desirable characteristics. The tests on the PEREGRINE have indicated the practicability of the activated-tank system of stabilization but have also indicated many changes and modifications—none of a fundamental nature—that should be made prior to future installations. The faults inherent in the PEREGRINE installation are all believed to have been discovered, and solutions are offered for them. It is therefore believed that enough is known now to be able to proceed to a permanent installation. Over-all features of the control system and the pumping system are discussed; then specific features are discussed. This is followed by a discussion of the various tests of the entire installation, drydock, calm-water, and rough-water tests. Other possible systems that appear practicable, such as the activated fins, are discussed and compared with the activated-tank system.

The control and pumping systems had been designed utilizing available components such as synchros, two-phase low-inertia motors, gun-mount train amplidyne and gearing systems, and the impellers and motors that had been used on the HAMILTON test. The amplidyne amplifier, motor, and generator system with its own feedback circuits were rather complicated. In addition there were feedback circuits in the mechanical filter and driver system as well as a feedback water-level signal to the output of the mixer unit. It was difficult to adjust the amplidyne system to avoid hunting and nonlinear response because the equipment was being used differently than in the application for which it was designed.

Minorsky provided a foundation for servomechanism theory and there has been great advance in recent years (References 44 through 48) so that the theory has become a powerful tool in the design of servomechanisms. However, a coordinated complete solution of the stabilization installation, including the controls, is needed to make the design of the system completely rational and to be able to design components for optimum operation of the over-all system. A particular servo system can then be analyzed to determine the simplest mechanism to obtain the control desired. The analysis can reveal nonlinearities, phase change with frequency, and possible hunting or parasitic oscillations in the system. This is predicated on there being adequate knowledge of the characteristics of the components proposed to be introduced in the system. In the case of stabilization this will involve knowledge of power requirements and frequency and amplitude response for both transient and steady-state conditions. It is, therefore, strongly urged that such analysis be made in the preliminary-design stage of future installations.

CONTROL SYSTEM

In addition to the applications of servomechanism theory to the over-all design of the activated tank system, there are certain considerations that apply specifically to the control system. First, all electronic equipment needs to be designed so that it is stable. There is no objection for most applications if as long as an hour is needed for the equipment to reach a steady temperature. However, when the equipment does reach this temperature it should be insensitive to surrounding temperature changes. If that is not possible, then the equipment should be protected by proper housing or even by temperature-control equipment. It may be that a combination is necessary—designing toward temperature insensitivity and temperature protection and control. The individual components such as tubes, resistors, and condensers should be of rugged types and they should be used considerably below their power ratings so as to give the equipment long, trouble-free service with the minimum of replacements. The tubes should operate within their linear ranges and should be selected for good operating characteristics.

The equipment, too, should be engineered as is much of the radio and radar equipment now used on shipboard, with adequate shock and vibration protection not only of the chassis but also of those sub-assemblies that need special protection. The wiring should be arranged in such fashion, aided by the use of sub-assemblies, that troubles can be quickly determined and parts readily replaced. It cannot be overemphasized that only the best components should be used and that workmanship, particularly of connections, should be of the best and must be sturdy (this does not imply that the equipment need be heavy).

The Schaevitz accelerometers, used as the transducers actuating the control system, were satisfactory. They were eminently satisfactory in maintaining constant characteristics and gave hundreds of hours of trouble-free service. However, a considerable amount of adjusting and auxiliary-circuit changing had to be done before the accelerometers could give satisfactory service at sea. First, the noise level had to be reduced to a minimum. A balancing circuit of resistance, inductance, and capacitance was used for this at each accelerometer. Since the accelerometers were connected push-pull, considerable juggling between the balance boxes was necessary to obtain minimum noise level on both accelerometers. A judicious use of connecting cable between the control room and the accelerometers could have reduced the amount of balancing considerably. Wires in cables usually are insulated with black or white rubber. The black rubber has carbon mixed with it which not only affects the noise level but the effect is not uniform through the length of a wire. If wire insulated with white rubber only is used, the noise level could be reduced to close to zero rather than to 2 to 5 millivolts. The accelerometers had a nominal natural frequency of 50 cps. At sea, with some of the ship's auxiliary equipment running, it was discovered that the felt mounts of the accelerometers did not isolate them adequately from vibration. Since the vibration signal was of much higher frequency than the roll of the ship, the acceleration signal caused by vibration, which was small, often was much greater than the acceleration signal caused by rolling. Since, of course, signal caused by rolling only was wanted, the vibration signal had to be eliminated. Some of it was eliminated by inserting sponge rubber in the felt-lined mounts, and the remainder by converting a strain indicator to an R-C push-pull filter and demodulating the accelerometer signal prior to the filter and remodulating to 400 cps after the filter. It would have been better to have accelerometers with natural frequency of 1 to 5 cps, which would have had much more sensitivity.

Accelerometers of the same transformer type with lower natural frequency and much more sensitivity are now available. On larger ships it may be possible to find vibration-free locations where accelerometers could be installed a considerable distance apart, but on smaller ships such as the PEREGRINE or on destroyers such a condition may be impossible to find. A device such as a light long bar with vibration mounts could be installed transversely in the ship at a position fore and aft that has the least amount of hull vibration. The bar then would impart the same motion and phase to both accelerometers and the vibration effect would cancel out; pitching and heaving signals cancelled in the way the accelerometers were wired.

For the driver unit it would have been more convenient to have control steps of equal increments, and more of them, rather than steps of increasing increments on the voltage amplifier. For power amplification on the same unit rather than the continuous control provided, all that is needed is an on-off switch. The power supply for the driver unit—which is also the power supply for the damping unit—needs redesigning to reduce the overloads, particularly on the tubes, some of which had to be replaced a number of times.

The mixer unit has been modified at the output end to tie in the water-level feedback signal externally instead of internally, therefore it should be rebuilt. The design, particularly of this unit and its power supply, should be re-examined with respect to overloads and thermal sensitivity.

Another change that would help in checking the balances of the mixer unit is the use of larger meters on the cathode-follower circuits.

The mechanical filter proved to be a difficult problem in design, construction, and maintenance. A number of cut-and-try alterations were necessary to obtain the proper natural frequency of that portion of the filter at the output end of the torsional springs. There was also difficulty in determining where on the mechanical filter the velocity damping and derivative damping should act. Finally, results were obtained about as desired with rapid attenuation at about 5- to 3-second periods and negligible phase shift until this point was reached. Certain difficulties were also encountered in keeping the shafting and the gearing properly aligned to minimize friction. The pinion gears probably had too few teeth to perform with the minimum of friction and without backlash. At all the support points of the shafting, unsealed bearings were used to keep the friction down. However these unsealed bearings picked up enough dirt to increase the friction to the point where the entire mechanism had to be cleaned, even though it was kept covered when it was not being calibrated. Rusting was another problem. The springs broke once, and their connections loosened a number of times.

These difficulties necessitated much attention and adjustment during the calibration of the control system and during the drydock and calm-water tests. Fortunately the mechanical filter was completely trouble-free during the sea tests. Upon the replacement of the filter springs, the new springs had a slightly lower natural frequency than the previous ones. This, together with the PEREGRINE's rolling frequency proving to be higher than anticipated, no doubt resulted in the water transfer being out of phase with the velocity of roll when those rolls occurred that were of highest frequency in a series of rolls. This reduced the over-all effectiveness of the stabilization.

The importance of having a mechanical filter that is always in prime mechanical condition indicates that a new filter should be designed and built in which the friction can be kept to a minimum with little or no maintenance. This can be done only by constructing a highly precise mechanism. Consideration should be given too to the complete redesign of the spring sub-assembly—providing for easy adjustment so that the natural frequency of the output end of the filter can be changed a reasonable amount. Torsional springs with linear characteristics may prove more convenient than the double, preloaded, helical springs that have been used. The use of nonlinear springs may even have an advantage in causing the output end of the filter to respond with less overshoot and therefore less correction by damping if the output gears reach their peak positions in each direction more gently.

Another factor influencing friction in the filter is the stiffness and alignment of the six shafts. There is considerable angular acceleration as the signal changes sign, and the pinion gears particularly make many turns in a short time. The two pinion gears are fastened to the shaft ends of the inertia two-phase motors, one at the input end driving the system and the other at the output portion used for damping. If the shafts are too limber, friction will vary depending on load. But there are restrictions that necessitate keeping their size and therefore their mass as small as possible; this applies to the gears too. The input power to drive the system is limited. This is true to a lesser extent concerning the power for damping. In addition the period of the output portion of the filter beyond the spring must be kept to relatively low values, on the order of 1/3 second.

None of these difficulties is unsurmountable, as was proved by the operation of the mechanical filter in a fairly satisfactory manner during the sea tests. However, from the viewpoint of maintenance-free operation and ease of adjustment another approach to the design of a filter may give better results. Apparently standard electrical-filter design cannot be utilized in the frequency range needed for ship stabilization without the filter becoming enormous in size, weight, and power requirements. It may be that by inventive ingenuity an all-electronic filter design may be evolved that will eliminate the disadvantages of the mechanical portion of the present filter.

The stabilizing system was tried with the mechanical filter bypassed in the early stages of the sea tests. The mechanical filter was found essential to the operation of the system, however, because as the frequency of roll increased, the signal for the same amplitude of roll increased. The felt-and-sponge-rubber combination at the accelerometer supports together with the R-C filter reduced the effect of the vibration frequencies to negligible proportions. But there still remained the acceleration signal immediately beyond the range of useful frequencies which had to be eliminated by the mechanical filter.

The stabilizing system was next tried without the water-control signal. Without this signal the water piled up in one tank or the other after starting, even in a relatively short time, so this signal proved to be essential for the successful continuous operation of the stabilizer. This feedback signal operated by tank floats comes from each of the two tank systems to each of the two modulator units which receive equal signals from the mixer unit. The water-level signal should have fine control for each tank system so that the minimum amount necessary to keep the water oscillating about the mid-height of the tank can be used, for the water-level signal reduces the effectiveness of stabilization.

PUMPING SYSTEM

The amplidyne units for changing pitch and the impeller and its drive, were used elsewhere and were adapted for stabilization on the PEREGRINE. The impellers, their drive motors, and connecting gearing were those used over a decade ago on the HAMILTON tests, while the old hydraulic-pitch mechanism was replaced by an adaptation of the Five-Inch Train Power Drive Mark 14 Mod 0. This expediency necessitated the acceptance of characteristics inherent in these devices that were not necessarily the best for ship stabilization.

The impeller unit, including gearing and motor, was originally of necessity a compromise design. The blades at zero pitch almost completely filled the cross section and were pivoted and symmetrical about their centers. Before the end of the tests the blades were cut at the leading edge to move the center of pressure on the blades closer to the pivot axis, which reduced the amount of power needed to change pitch. It was hoped, too, that the incomplete closure at zero pitch would enable better utilization of the natural period of oscillation in each tank system. The natural frequency of the water was so low and the damping was so great that it is doubtful whether the reduction in blade area helped at all in this respect. Actually, the drydock water-transfer tests showed a reduction in the amount of water transfer after the blades were cut. The blades for a new installation should therefore almost completely fill the cross section at zero pitch, unless other aspects of pump and control design preclude this, but their pivot axes should pass through an average center of pressure of the blades.

The drive motor and the gearing were a compromise design. A d-c motor turning at 1900 rpm was used; the herringbone reduction gearing reduced this to 140 rpm at the impeller. The faster the motor the lighter it would be for the same power output; soon, however, a "break point" is reached where the reduction gearing and its housing increase in weight faster than the motor weight decreases. Increased gear reduction necessitates, too, a stiffer housing and supports as well as better fitted gears so that rapid deterioration does not occur. Another ratio which should be chosen at the most advantageous value is that between the diameter and the rpm of the impeller blades. There is a limiting factor here: The amount of water pumped per unit blade area and per rpm. It may be that blades of smaller diameter could turn faster and be as efficient up to a certain point. This would permit a somewhat lighter power drive. Here again, however, the impeller weight per unit output would increase, and this increase would have to be compared with the decrease of the weight of the drive.

For a new installation the hydrodynamic characteristics of the impellers should be carefully reevaluated. In the present installation a heavy flywheel was fastened to the motor shaft to store energy for the peak loads. Alternating-current motors would be more desirable in the future to save weight and to simplify the power requirements for the installation, since d-c power in the amount needed is generally not easily available on shipboard. The new insulations permit use of higher currents—therefore higher ratings for the same motor frame. This advantage of weight reduction is lost in the use of a flywheel, but the space requirement would be reduced. In any case, for military reasons, the flywheel should be eliminated since there are the same objections to it as to the gyro stabilizer. In all cases a ship stabilizer will be working when the ship will be operating at cruising speed or less, since foul weather precludes a ship traveling at full speed because of the great strain on the hull. The power increase would not be great in abandoning the flywheel since there is a reduction in ship resistance when a ship's roll is decreased.

The location of the impeller in the tank-and-duct arrangement is dictated by the available space in a ship and by the need for keeping the water friction to a minimum. To be certain there is always adequate head of water, the impeller blades should be as low in the system as possible and the drive also should be kept low—eliminating long shafts and the need for weather protection. Two possible arrangements are shown in Figure 57. Two

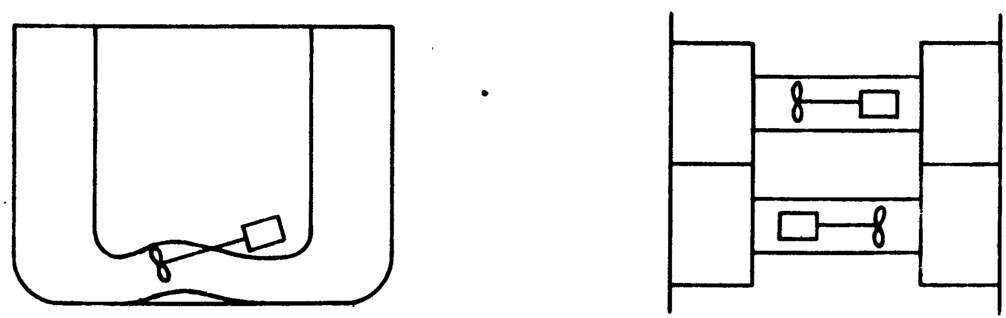

Figure 57a - Vertical Bend

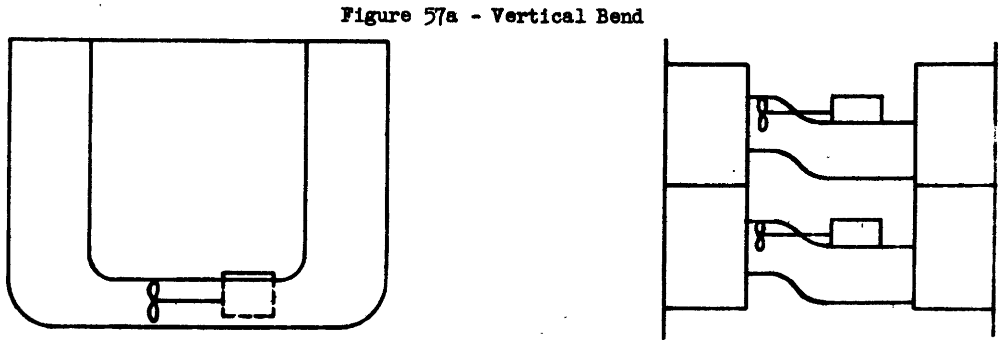

Figure 57b - Horizontal Bend

Figure 57 - Two Possible Arrangements of Impellers in the Crossover Ducts

tank systems are shown in each case, in Figure 57a with a vertical bend in the crossover duct and in Figure 57b with a horizontal bend in the crossover duct. The latter has the advantage of having one bend while the former has two. This results in less water friction and also permits more convenient arrangement of the drive, utilizing what otherwise would be dead space.

The mechanism for changing pitch, including the train gearing, the motor drive and its reduction gearing, the amplidyne motor generator, the amplifier and its power supply, and the magnetic regulator, took up an inconvenient amount of space, was complicated to operate, and had to be nursed along.

The train gearing was retained on the PEREGRINE installation to permit zeroing the impeller blades and returning them to zero pitch by means of the manual control when, during adjustment, the amplidyne system kicked out with the impeller blades at large pitch angles. It permitted, too, preliminary operation by means of hand control before the control system signal was injected to the pumping system. The manual feature can be incorporated in a simpler way if the amplidyne system were designed specifically for the stabilizer: The manual control could be tied in to the reduction gearing by means of a clutch and might be simply a pair or two [illegible] gears. If a shut-down switching circuit were installed to reduce the input signal to zero, then the manual gearing could be very simple since the manual control would not be frequently used and therefore effortless operation need not be a major consideration. Another zeroing circuit could be incorporated in the thermal relay cut-out so that the blades would return to zero before the amplidyne system kicks out. This would reduce the use of the manual control to a minimum. Included in the train gearing were elements for correcting for parallax and other effects ininvolved in gun firing. The only element needed on redesign is the control transformer which transmits an error signal to the amplidyne amplifier, a necessary and basic feedback circuit in the amplidyne system.

With the manual control worked into the reduction gearing which connects the d-c pitch motor to the pitch shaft, space and weight are greatly reduced. The magnetic controller can be reduced to less than half its size, since only the train portion without some of the gun corrections is needed. In this unit, the rating of the thermal relay can be increased safely as was proved on the PEREGRINE, where this helped tremendously in reducing the kicking out of the system on the occasional rapid changes of acceleration.

The amplidyne amplifier too can be reduced by more than half since only the train portion was used on the PEREGRINE: therefore the elevation amplifier and that portion of the power supply needed for it can be eliminated, and the remainder can be simplified since most of the corrective feedback controls needed in gun mounts are unnecessary here. The proper balancing of the amplidyne potentiometers completely eliminated kick-outs. Not only was there no hunting, but for a sinusoidal signal the power drawn by the d-c motor was equal for swings in both directions. This eliminated the overload peaks of power that had been drawn previous to this adjustment. On the PEREGRINE double electrical stops were installed to prevent the pitch mechanism from jamming. A stop at each end of travel was installed to kick out the amplidyne system, but the point of kicking out depended upon the speed of approach to these stops; there was variable overshoot. In addition, ahead of these, positive micro-switch kick-out stops were installed on the pitch shaft. In redesign, a circuit replacing the first electrical stop should be built-in, limiting the current so that at no time does the mechanical system go beyond predetermined points. Adjustment should be provided for this stop. Also, into the thermal and other necessary overload relay circuits should be inserted a circuit that would return the pitch to zero just prior to kicking out the amplidyne circuit. The micro-switch safety kick-out should be retained as the final safety device. All these devices as well as timing and balancing controls in the amplifier should be built sc that adjustment is fine and sensitive, preferably with scales to be able to compare results of changes. In the design of an amplidyne system for stabilization, provision should be made for the starting of the amplidyne motor-generator without overloading the ship's power facilities, particularly when obtaining power for these sets from relatively small ship service units.

After the difficulties with the amplidyne pitch-control system on the PEREGRINE were corrected, operation was good. It is believed that a redesign using standard components such as the amplidyne d-c motors and motor-generators, and with redesign and reconstruction of the magnetic controller and the amplifier and power supply units, excellent control will be achieved that, once adjusted, will be trouble-free for long periods of time. It is believed that a minimum of maintenance will be necessary and would involve primarily following a standard schedule of lubrication and the cleaning of electrical contacts.

With a complicated electro-mechanical array,such as this stabilizing system is, the questions of simplification always arise: Why is a pitch mechanism necessary? Can it be eliminated? It can, by driving a fixed-blade impeller directly with the amplidyne system and with speeds proportional and reversals in accordance with the actuating accelerometer signal.

An alternative installation is shown in Figure 58. A straight radial-bladed impeller is inserted in a section of the crossover duct with the impeller shaft horizontal. Using the amplidyne system to pump directly eliminates the manual control, the impeller motor and its flywheel and reduction gear, and all of the protective stops and zeroing circuits. The gear reduction for the drive motor will not be as great and may even be omitted. The design of the crossover ducts will be simpler. The control transformer feedback circuit would still be needed. A clipping-protective overload circuit would also be required to prevent the load from exceeding a set amount yet which will still permit the impeller to operate momentarily at a level less than proportional to the input signal. The advantage of the flywheel will be lost, but this is more apparent than real since two motors, impeller and pitch, are replaced by one and the peak power is such that no strain will be placed on ship power, while the average power consumption is considerably less. Care will be necessary in order to design the impeller with minimum inertia so that the power needed to stop and reverse it can be reduced. A certain amount of flexibility will be needed in the connection between the motor drive and the impeller to ease off the starting and stopping of the impeller. Then finally, and perhaps most important, the sacrifice in pumping efficiency with this type of impeller should not be great enough to reduce the final over-all efficiency.

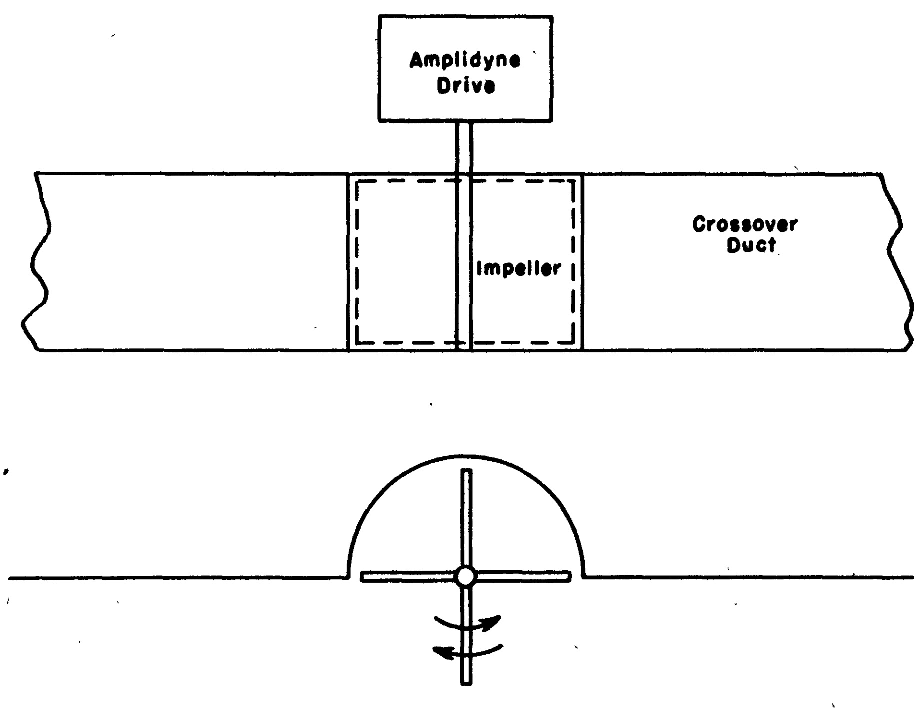

Figure 58 - Alternate Method of Pumping Water for Ship Stabilization with the Amplidyne System

Some reduction in the over-all efficiency can be tolerated because of the greatly simplified mechanism. Perhaps helical blades may improve the pumping characteristics of this type of impellers. A fixed-pitch two-directional propeller pump, with its axis essentially parallel to the duct, also merits investigation.

The crossover ducts installed on the PEREGRINE were of the greatest possible cross-sectional area permissible in the space available. This was 10.9 sq ft for the most part—almost square—with a minimum of 8.9 sq ft at the elbows. The ducts were lined with steel plate to reduce friction. In the elbows at both ends of the ducts were splitters to smooth out the flow. With all these precautions the natural frequency of oscillation of the water in the tank system was about 12 seconds. So, for the short rolling periods on the PEREGRINE, no help was gained from the free oscillation of the water. On a permanent installation the tanks would be inboard just within the hull, which shortens the lengths of the ducts considerably. Even this would not be enough in the case of the PEREGRINE. It would be necessary to increase the depth of the crossover duct from 2 3/4 to about 4 ft, giving a cross-sectional area of about 16 instead of 11 sq ft, in order to reduce the period of oscillation to about 7 seconds.

Since in a permanent ship installation the tanks would be of the same general configuration as the normal side tanks on ships, it is necessary to have the impeller somewhere along the crossover ducts. This arrangement reduces the length of the duct and consequently the period of oscillation. A butterfly valve should be installed to serve two purposes—provide adjustment for the period of free oscillation and provide a closure when it is desired to stop the stabilizer and reduce the free surface effect. Insofar as the air crossover duct is concerned, it should be eliminated and risers with valves should be installed over each tank, the risers providing height if the water head on occasion becomes higher than the tank top. The valves provide control of the free period of oscillation. It is important that possible hydrodynamic losses be kept to a minimum for the most effective utilization of the free period of oscillation.

OVER-ALL PERFORMANCE

The advantages, disadvantages, and suggested modifications of the components of the control and pumping equipment have been discussed. Some comments should be made, however, on the over-all performance. In any practical, permanent shipboard installation two or more tank systems will be necessary in order to fit the stabilizer into a ship without major alterations. Adjustment and operation of these multiple-tank systems would be simpler if there were a common amplidyne-amplifier output for all impellers. Then only adjustment at one point would be needed for all amplidyne systems after an initial adjustment between amplidynes is made. The possible merit of a reversible pump without feathering the blades has been discussed previously. The control equipment can be reduced to the size of one bathtub rack and should be placed in a position where it could be operated easily under the cognizance of the navigation department. Aside from starting and stopping and the adjustments necessary upon installation or overhaul, the operating adjustments can be reduced to amplitude, phase, and water-level signal and probably can be done with three knobs on the control equipment and one knob for water level for each tank system. This indicates that in a multi-tank system the float-level signal will have to be added electrically to the input-control signal of each amplidyne motor-generator set.

The drydock tests, which were performed twice (before and after reducing the impeller blade area), consisted of two parts, static and dynamic. In the static test, in addition to the adjustment of the installation and the calibration of the signal control devices, the effectiveness of the impellers at different constant pitches was measured. The cutting of the blades to shift the center of pressure on them toward the blade shaft reduced, for both impellers, the maximum head of water pumped by about 20 percent with an increase in the pitch angle at which the maximum head occurred from about 5° to 15°. The area of the blades was reduced 1/6 so the loss in head was somewhat more than proportional to the reduction in area. In a similar type of impeller design the blade-pitch shaft can be located along the line of center of pressures without reducing the area of the blades. For control purposes, however, some reduction may be necessary to reduce the steepness of the pitch-flow characteristics.

On the calm-water tests confirmation was obtained that the rolling period of the ship was about 7 seconds and that the metacentric height was over 4 ft. The ship could be rolled only about 11° double amplitude maximum at this period. With reduced pumping capacity at 7 seconds the full height of the tank could not be utilized; therefore, it was expected that only incomplete stabilization could be attained with the installation on the PEREGRINE. One question arises at this point. How much of the cross section in the impeller section should be taken up by the impeller blades at zero pitch if the natural period of oscillation of the water in the system is close to the roll period of the ship? Complete closure may not permit full utilization of the natural oscillation of the water. What the effect of the proper natural frequency of oscillation of the water in the tanks would be on the rolling amplitude of the ship is hard to say except that, if the two frequencies are close, greater stabilization will occur. The answer depends upon the amount of damping in the duct system. On the PEREGRINE installation the water in the tanks was critically damped and had a natural period of oscillation of 12 seconds. Doubtless, even if the ducts were large enough and short enough to result in a period of 7 seconds, space requirements would still make the damping fairly high. Relocation of the pump may reduce the damping.

The rough-water tests indicated that for one impeller operating before the blades were cut that there was a reduction in average roll of 0.7° single amplitude, and for both impellers operating after the blades were cut there was a reduction in average roll of 1.3° single amplitude; see Table 10. Now the maximum amount of water transfer obtained during the sea tests was about 2 1/2 ft double amplitude, whereas there was capacity for 8 ft double amplitude. If the rolling period were 9 seconds instead of 7 seconds and if two impellers were running, the upper curves of Figure 50 show that there would have been an increase of about 45 percent in the stabilizing moment, because of greater pumping capacity at this period, and the metacentric height would have been about 3 ft, resulting in the need for less stabilizing moment. Therefore, a reduction in average roll of 1.3° x 1.45 x 4.6/3, or 2.9°, rather than 1.3° single amplitude would have been effected with the same equipment. If the rolling period were 12 seconds and the metacentric height 2 ft, a reduction of average roll of 6° single amplitude would have been obtained. This value, 6°, indicates that if the rolling characteristics had been as originally planned, practically every interval listed in Table 10, including those with maximum angles of roll of 14° single amplitude, would have been stabilized.

Any beneficial effects that might have resulted from passive stabilization with the water oscillating at the right frequency have been neglected in these calculations. Also neglected has been the fact that during the operation of the stabilization system the amplitudes of signal to the amplidyne had to be set for the largest rolling accelerations, otherwise the pitch motor would kick out because of excessive pitch angle. The records of roll show that the magnitude of average roll was on the order of less than half of the peak roll. If clipping circuits had been included, the effectiveness of stabilization would have been increased on the order of 100 percent for this reason alone. For occasional waves the slope angle of the ship would have been greater than the slope limit of the stabilizing system so that on occasion a small amount of roll would have appeared. The other limitation on perfect stabilization would have been the sensitivity of the control and pumping systems and nonlinearities in the righting moments and in control elements.

OTHER SYSTEMS

Advantages and disadvantages of other systems of stabilization have been discussed. The stabilizing fin is the only other system on which serious thought is being given at present, despite the disadvantage that the amount of stabilization depends on ship speed. Nonretracting stabilizing fins are attractive in that they have a very low weight ratio, and the protruding fins do not now seem such a handicap to easy moving or docking.

The control system for the activated tanks can be very simply adapted to the stabilizing fins since the fins are actuated in proportion to velocity of rolling. Considerably less phase advance is needed with the fins than with the tanks since the fins have effective moment as soon as they are positioned. In the activated-tank system after the impeller pitch is changed a certain amount of time is needed to pump water to the proper side. The amplidyne drive can be used to position the fins in a mechanically simple arrangement. A consideration in the application of the fins is their location fore and aft on shipboard since there are positions which can aggravate and positions which can alleviate the yawing tendency of a ship.

CONCLUSIONS

1. The activated-tank system of stabilizing has the advantage over the fin system that stabilization can be obtained for ships while at anchor as well as under way.

2. The results obtained with the USS PEREGRINE (E-AM373) indicate that the activated-tank system is capable of effective stabilization within the limits of a particular system design, provided an adequate control system is devised.

3. With an installation designed for a ship with metacentric height of 2 to 2 1/2 ft and a period of roll from 9 to 12 seconds and with the ship actually having a metacentric height of 4.6 ft and a period of roll of 7 seconds, the PEREGRINE was stabilized an average of 1.3° single amplitude. By extrapolation, for a 12 second-period, 2-ft metacentric height ship, this was calculated to be equivalent to 6° average single amplitude which is greater than the average roll for the intervals measured during the sea tests. Any beneficial effects that might have resulted from passive stabilization with the water oscillating at the right frequency have been neglected in these calculations. Also neglected has been the fact that during operation of the stabilization system the signal amplitude to the amplidyne had to be set for the largest rolling accelerations, otherwise the pitch motor would kick out because of

excessive pitch angle. If clipping circuits had been used on the PEREGRINE, the effectiveness of stabilization could have been increased on the order of 100 percent for this reason alone.

4. An improved form of the present control system with a suitably reduced phase advance, and a modified form of the amplidyne drive can presumably be used with the fin-type stabilizer.

5. Experience with the PEREGRINE installation indicates that the stabilizing system could be simply controlled by three or four knobs after it has been actuated. Only one of these knobs will need slight adjustment from time to time. The control system could be installed in one bathtub rack.

6. Consideration should be given to a fixed-pitch two-way amplidyne-drive impeller to eliminate the impeller motor. From considerations of hydraulic efficiencies a propeller pump may be most appropriate.

7. Hydraulic considerations of friction, free period of oscillation, splitters, ducts, elbows, and pumps should be carefully examined for a new installation.

8. The amplidyne system should be designed for the specific application of changing impeller pitch or running a variable-speed pump. This would simplify the installation both mechanically and electrically.

9. Maximum effective wave slopes greater than the limit of 5° assumed for ship-stabilization design in the past have been encountered. This fact should be considered in future designs of stabilization systems.

RECOMMENDATIONS

1. In view of the expenditure of effort and money by the Navy and in view of the encouraging results obtained toward the solution of the problems encountered on the PEREGRINE, it is recommended that the development be pursued to a successful conclusion by making an installation on a moderate-size naval vessel. The following procedure is recommended:

> A. Redesign the control system with emphasis on simplicity, underloaded components, and sturdiness. Replace accelerometers with those, now available, which have lower natural frequency and increased sensitivity. Improve the mechanical filter, or, better still, replace it with an equivalent electrical filter if possible. Improve the floats.

B. Examine the possibility of driving an impeller at variable speed with an amplidyne drive. Determine the most efficient type of pump for this service. Investigate the frictional losses in the tank-duct pump system, making scale models if necessary.

C. Design an amplidyne system with simplified amplifier and mechanism and the minimum number of feedbacks for this service.

D. Rewrite the dynamical analysis rigorously and state the results in such a way as to be more easily used by those concerned with its application to ship stabilization.

E. A continuing project group for coordination should be set up from the very beginning—including a project head, an electronics expert, and a hydraulics expert. The key men should make themselves thoroughly familiar with the principles and all aspects of an installation. Then a concerted effort can be made toward an optimum, integrated design. A stabilization installation is as good as its weakest link in reliability and performance.

2. If it is desired to install an activated-fin stabilizer on a naval vessel, then the same considerations as in 1 A, C, and E hold. A simplified version of the present control system and an amplidyne or hydraulic drive would operate the activated fins satisfactorily.

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The Bureau of Ships predicted the period and metacentric height, advised on the stability of the ship, and determined the shape of the sponsons. The Norfolk Naval Shipyard was responsible for the design, construction, and installation of the bulk of the stabilization equipment in accordance with specifications, was responsible for the inclining test, and helped the Taylor Model Basin whenever necessary. The PEREGRINE and her crew were completely available for installation and test of the stabilizer. The Taylor Model Basin had over-all cognizance of the project, including design and construction of the control equipment, and responsibility for specifications, testing, and reports.

It is only possible to mention some who participated in the project. From Code 440, BuShips, Lt. Comdr. Henderson, USN, and later Comdr. D.K. Ela, USN, were concerned. In the Design Department of the Norfolk Naval Shipyard may be mentioned Messrs. Sackakeney, Baker, Hardy, Resolute, Need, and their associates and supervisors, and Capt. H.J. Hiemenz, USN, Design Superintendent, and his successor, Comdr. J.C. Dyson, USN. Others at Norfolk Naval Shipyard were Mr. Fentriss, Planning, Chief Progressman Rock, Progressman Lyon, Master Mechanic Warren, Shop Supervisors Reid, MacDaniel, Davidson and their men, Lt. Comdr. C.N. Payne, USN, Docking Officer, Lt. Comdr H.C. Field, USN, Production. Lt. F.L. French, Jr., USN, Commanding Officer of the PEREGRINE, and his officers and men—particularly M.F. Bartscheck, Jr., ETM 2/c, and R. Nolan, Phot. M 3/c, who did much calibrating, computing, and photography for long hours day and night—cooperated wonderfully to expedite this project. From the Taylor Model Basin, Mr. W.S. Campbell and his associates of the Electronics Engineering Branch undertook the design and construction of the control equipment; Mr. W.P. Kiley of the same branch corrected many deficiencies in the shipboard installation of the control system; Mr. R.G. Tuckerman, also of the same branch, participated in the installation and calibration of the control equipment; Messrs. Q.R. Robinson and J.T. Birmingham of the Vibration Division aided in calibration, testing, and analysis, while Mr. R.T. McGoldrick, supervisor of this division, made many valuable observations. Dr. E.H. Kennard, Chief Scientist of the Structural Mechanics Laboratory, made many constructive suggestions on the manuscript of this report.

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Selected working drawings are listed here for convenience:

Drawing Number		Title	Date
BuShips Plan Number E-AM 373-52904-			
SK 628940	Alt. 0	Anti-Rolling Device - Preliminary Arrangement	17 Dec 1948
961899	Alt. 0	General Arrangement of Anti-Rolling Tanks and Equipment	28 Jan 1949
962852	Alt. 0	Anti-Rolling Equipment - Arrangement of Impeller Machinery with Train Amplidyne Pitch Control	7 Jul 1949
962861	Alt. 0	Anti-Rolling Equipment - Impeller Pitch Control - Adapter Assembly	19 Jul 1949
962902	Alt. 1	Anti-Rolling Equipment - Impeller Pitch Control - Clutch Shifting Mechanism - Arrangement and Details	15 Jul 1949
961668	Alt. 2	Anti-Rolling Device - Activated Tanks	8 Feb 1949
961667	Alt. 0	Anti-Rolling Device - Sponson	13 Jan 1949
961646	Alt. 0	Anti-Rolling Device - Anti-Rolling Tank Details	30 Dec 1948
961491	Alt. 1	Anti-Rolling Tank Gear - Electric Power - Schematic Wiring	21 Jun 1949
961727	Alt. 4	Anti-Rolling Tank Gear Cable Layout for Power, Lighting and Instrumentation	29 Jun 1949
BuOrd Drawing Number 480363 G.E. W-9076736		Elementary Wiring Diagram - 5" Train and Elevation Power Drives for 5" Train Power Drive Mk 14 Mod 0 5" Elevation Power Drive Mk 14 Mod 0	16 Apr 1947

Baldwin Southwark Corp., I.P. Morris Division, Drawing 34602	Sectional Elevation of Deck Assembly and Servo-Motor for 55" Dia. Oscillator Units	1 Dec 1939
34501	Arrangement of Hub Mechanism and Lower Bearing	9 Oct 1939
34456	Details of Blades for Impeller	12 Sep 1939
34458	Details of Lower Diffuser Section	14 Sep 1939
34457	Details of Upper Diffuser Section	12 Sep 1939
Taylor Model Basin Electronics Dwg. No. 118	Driver Unit for Mechanical Filter, Type 44-A	17 May 1949
119	Damping Unit, Type 44-A for Mechanical Filter	17 May 1949
120	Mixer Unit - Type 44-A for Amplidyne Control Signals	17 May 1949

APPENDIX 1

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APPENDIX 2

SELECTED LIST OF STABILIZATION PATENTS

This appendix contains a selected list of patents issued by the United States on stabilizing devices. Many German patents exist but quite a few are duplicated by American patents; all other foreign patents have been omitted. Aircraft-control devices and applications to ship and aircraft instruments such as artificial horizons and compasses have been omitted even though their principles may be applicable to ship stabilization.

Sperry, E.A., "Ship's Gyroscope," 1,150,311, August 17, 1915.

Sperry, E.A., "Ship Stabilizing and Rolling Apparatus," 1,232,619, July 10, 1917.

Frahm, H.G.C., "Marine Vessel," 1,427,526, August 29, 1922.

Frahm, H.G.C., "Marine Vessel," 1,480,934, January 15, 1924.

Motora, S., "Steadying Device of the Rolling of Ships," 1,533,328, April 14, 1925.

Lucke, H.F.W., "Device for Reducing the Pitching of Ships," 1,642,163, September 13, 1927.

Hammond, J.H., Jr., "Ship Stabilizer," 1,700,406, January 29, 1929.

Norden, C.A., "Ship Stabilizer," 1,708,679, April 9, 1929.

Dow, H.H., "Stabilizing Device and Method," 1,731,276, October 15, 1929.

Kéféli, M., "Stabilizing Device for Ships," 1,751,278, March 18, 1930.

Hammond, J.H., Jr, "Lateral Stabilization for Torpedoes," 1,772,348, August 5, 1930.

Dow, H.H., "Method of Stabilizing Ships," 1,774,825, September 2, 1930.

Sperry, E.A., "Means for Preventing Pitching of Ships," 1,800,365, April 14, 1931.

Schein, A.E., "Roll and Pitch Reducing Device for Ships," 1,800,408, April 14, 1931.

Sample, P.B., "Method and Apparatus for Controlling Mechanical Oscillations," 1,809,288, June 9, 1931.

Minorsky, N., "Stabilizing Apparatus," 1,853,069, April 12, 1932.

Chessin, A., "Gyrostabilizer," 1,885,414, November 1, 1932.

Hort, H., "Stabilizing Device for Ships," 1,888,667, November 22, 1932.

Hort, H., "Ship Stabilizer," 1,894,272, January 17, 1933.

Chater, W., "Stabilizing Device," 1,899,079, February 28, 1933.

Boykow, J.M., "Stabilizing Apparatus," 1,964,869, July 3, 1934.

Minorsky, N., "Electrical Controlling System," 1,988,458, January 22, 1935.

Fieux, J., "Stabilizing Device," 1,999,897, April 30, 1935.

Minorsky, N., "Stabilizing Apparatus," 2,017,072, October 15, 1935.

Hort, H., "Stabilization of Ships by Means of Liquid Filled Tanks," 2,024,822, December 17, 1935.

Broulhiet, G., "Gyrostat," 2,025,640, December 24, 1935.

Frisch, E., Schaelchlin, W., and Ashbaugh, J.H., "Ship Stabilizing System," 2,046,735, July 7, 1936.

Seligmann, J., "Gyroscope-Controlled Apparatus," 2,047,922, July 14, 1936.

Hort, H., "Ship Stabilizer," 2,066,150, December 29, 1936.

Throndsen, O.A., "Stabilizing Means for Ships," 2,075,594, March 30, 1937.

Carroll, E.R., "Means for Stabilizing Ships with Fuel Oil," 2,077,143, April 13, 1937.

Bazé, W.L., "Stabilization of Ships," 2,098,531, November 9, 1937.

Wallace, W., "Antirolling Apparatus for Ships," 2,099,380, November 16, 1937.

Gonzales, M., "Gyro Stabilizer," 2,104,226, January 2, 1938.

Rocard, Y.A., "Stabilizing Equipment for Vehicles, Particularly Vessels," 2,130,929, September 20, 1938.

von den Steinen, C., "Stabilizing Device for Ships," 2,155,456, April 25, 1939.

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Rosingh, W.H.C.E., "Antirolling Device for Ships," 2,198,295, April 23, 1940.

Manteuffel, G.Z., "Stabilizing Device," 2,200,196, May 7, 1940.

Minorsky, N., "Antirolling Stabilization of Ships," 2,202,162, May 28, 1940.

Giliberty, J.V., "Ship Stabilizer," 2,210,935, August 13, 1940.

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Moody, L F., "Hydraulic Oscillator," 2,258,464, October 7, 1941.

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Stub, J., "Ship Stabilizing and Controlling Mechanism," 2,346,337, April 11, 1944.

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Metcalf, A.G.B., "Recording and Controlling Vehicle Motion," 2,368,673, February 6, 1945.

Rosenstengel, B.L., "Apparatus for Producing Power," 2,385,943, October 2, 1945.

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