62-4-5

ELECTRIC BOAT GROTON, CONNECTICUT
A DIVISION OF GENERAL DYNAMICS

DESIGN STUDY REPORT

FIRST INTERIM REPORT ON PROPULSION MACHINERY AND PITCH CHANGING SYSTEMS FOR THE TANDEM PROPELLER

Contract NOnr 3383(00)

by

T. J. Gerken, A. C. McClure, M. Wieser

Research and Development Department

and subcontractors

General Electric Company

Elliott Company

GENERAL DYNAMICS/ELECTRIC BOAT

Groton, Connecticut

Approved: E S Dennison

E. S. Dennison
Chief Development Engineer
Development Engineering Division

U413-62-092

June 30, 1962

ABSTRACT

A study is made of a turboelectric submarine propulsion system incorporating propellers at both ends of the ship. The propellers are driven by free-flooding motors located in the propeller hubs. Propeller blades are arranged to be controllable in pitch individually, and hence provide directed thrust for ship control. Design requirements for machinery and pitch changing systems are developed, and a feasible design is evolved.

TABLE OF CONTENTS

LIST OF FIGURES

		Page
28.	GENERAL ELECTRIC Pitch Changing System Overall Schematic	107
29.	GENERAL ELECTRIC Pitch Changing System Detailed Schematic	108
30.	GENERAL ELECTRIC Assumed Propeller Blade Dimensions	110
31.	GENERAL ELECTRIC Propeller Blade and Actuator, Plan View	114
32.	GENERAL ELECTRIC Propeller Blade and Actuator, Elevation	115
33.	GENERAL ELECTRIC Propeller Blade Actuator, Plan Section	116
34.	GENERAL ELECTRIC Propeller Blade Actuator, Elevation Section	117
35.	GENERAL ELECTRIC Typical Propeller Blade Actuator Motor Torque and Current vs. Speed Curves	120
36.	GENERAL ELECTRIC Pitch Changing System Composite Block Diagram	123
37.	GENERAL ELECTRIC Signal Transfer from Hull to Hub via Ring on Hull and Pickups on Hub	125
38.	GENERAL ELECTRIC Synchronous Signal Generation (Signals change once per revolution of hub)	126
39.	GENERAL ELECTRIC Signal Transfer from Hull to Hub via Single RF Antenna	127
40.	GENERAL ELECTRIC Static Signal Generation (Signals do not change once per revolution of hub)	128
41.	GENERAL ELECTRIC Signal Transfer from Hull to Hub via Multiple RF Antennas	129
42.	GENERAL ELECTRIC Electro-Hydraulic Pitch Changing System	131
43.	GENERAL ELECTRIC Blade Drive Motors	133
44.	GENERAL ELECTRIC Motor-Blade Couplings	134
45.	ELLIOTT Propulsion Motor Cross Section	139
46.	ELLIOTT Turbine-Generator Outline	140
47.	ELLIOTT-ELECTRIC BOAT Hydraulic Pitch Changing System	145
48.	ELLIOTT-ELECTRIC BOAT Signal Generation and Transmission	147
49.	ELLIOTT-ELECTRIC BOAT Typical Hydraulic System Layout in Propeller Hub	148
50.	Tandem Propeller Motor - Space Allocation	167
51.	Propeller Blade Lift, Drag, L/D, and Center of Pressure vs. Angle of Attack	177
52.	Axial and Tangential Forces vs. Angle of Attack	180
53.	Axial and Tangential Forces vs. Pitch Angle	181
54.	Propeller and Motor Torque vs. Speed Curves	184

I

INTRODUCTION

This contract is a part of an investigation being performed by several contractors on a tandem propeller system for submarine propulsion which has been devised by Cdr. F. R. Haselton, USN.

The tandem propeller system consists of two complete motor and variable pitch propeller systems; one near the bow and the other near the stern of the ship, as shown in Figure 1. The motors are mounted outside the pressure hull and are free-flooding. Propeller blades and pitch changing mechanism are arranged to enable the pitch of all blades to be altered uniformly or to superimpose periodically varying pitch on the average pitch. The pitch of the propeller blades can be changed at will while the ship is being operated. By this means, control of propeller thrust in magnitude and direction ahead or astern or at an angle to the ship's centerline is possible. By a suitable coordination of bow and stern propellers the ship can be made to move bodily upward, downward, or sideways as well as to perform the usual maneuvers.

The portion of this investigation covered by this report consists principally of the study of main propulsion machinery and means for controlling propeller blade pitch. The scope of this investigation is evaluation of feasibility, and this was done by preparing preliminary designs and then reviewing the results and also any problems encountered in preparing the designs.

PROPULSION MACHINERY AND PITCH CHANGING SYSTEMS

The main propulsion machinery is similar to that developed under a previous study for the Office of Naval Research, the Novel Electric Propulsion System, Figure 2, and this report draws upon that background. In the Novel Electric Propulsion System study the main motors were

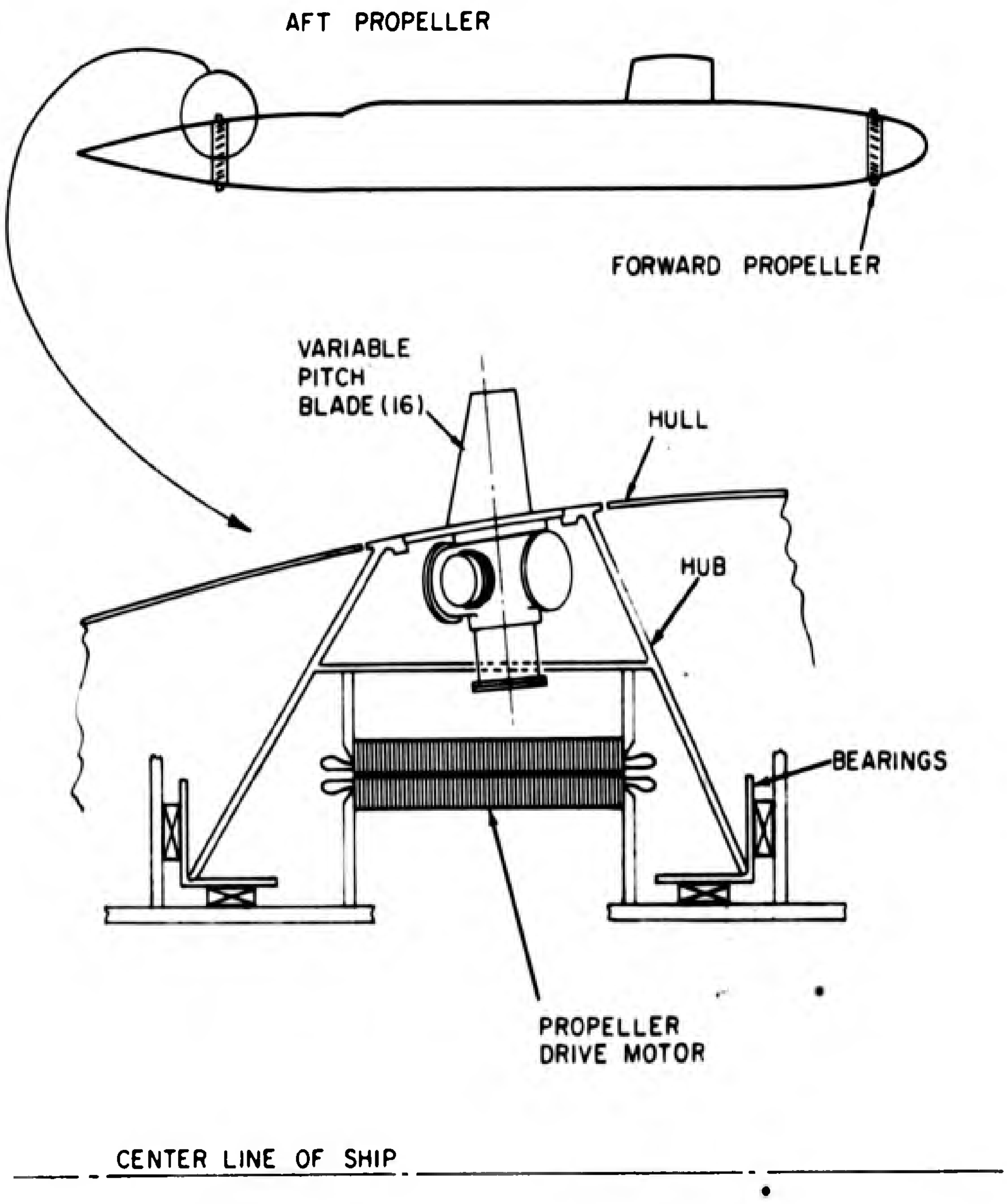


Figure 1. General Propeller Arrangement

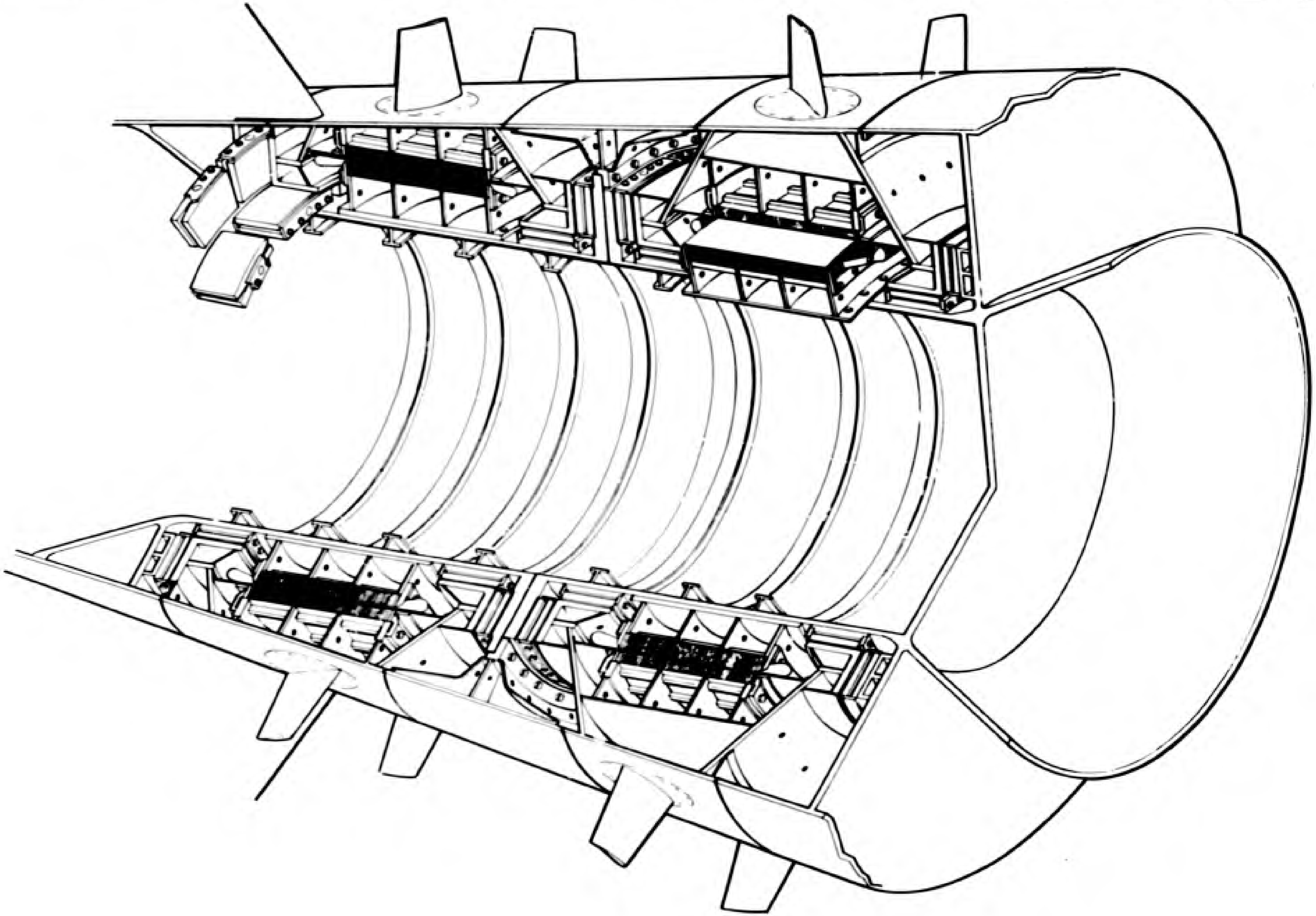


Figure 2. Novel Electric Propulsion System Motors and Propellers

limited to squirrel cage induction type, but in the present study the synchronous motor is also considered. The portion of this study dealing with main propulsion machinery includes:

Study and selection of motor type.

Preliminary design of the motor.

Outline design of propulsion turbine generator sets and associated equipment.

Pitch control is assumed to begin with a set of signals from the ship control system which define the required propeller pitch in terms of collective pitch, superimposed cyclic pitch, and orientation of the cyclic pitch. The pitch control system converts these signals to the form appropriate to the propeller blade actuators and brings each blade to the required pitch. Suitable feedback systems are provided to insure correct pitch and time constants. This study includes:

Analysis of pitch variation schemes to select the most promising method.

Investigation of the parameters of simplicity, reliability, silent operation and response time as they affect each actuating system.

Preparation of diagrams of the systems investigated with an analysis of the advantages and disadvantages of each.

Preparation of a preliminary design incorporating the most promising system selected.

Assisting Electric Boat in developing hardware for the tandem propeller system are two subcontractors: General Electric Company and Elliott Company. General Electric has studied all aspects of the propulsion system: transmitting pitch information into the hub, providing electric power on the hub, pitch control mechanisms, feedback control systems, and propulsion motor and turbine-generator set design. Emphasis in pitch control studies has been on electrically controlled and operated systems, but not to the exclusion of other systems.

The Elliott Company has applied primary effort to main propulsion motor design and means of transferring electric power to the hub. A team from Electric Boat worked with Elliott to develop an electrohydraulic pitch control system compatible with the motor to form a complete system. The Elliott propulsion machinery work is reported to date, but is not complete. The remaining work will be reported later in a supplement to this report.

Electric Boat developed a mechanical pitch control system which is compatible with either the Elliott or General Electric motor.

The efforts of Electric Boat, General Electric and Elliott thus constitute three separate and complete systems. In this way, three different approaches to the pitch changing and propulsion system were developed in sufficient detail to form a firm basis for a decision at the end of the program.

HYDRODYNAMIC STUDY

Before a quantitative study of the propeller pitch changing mechanism could be undertaken, a knowledge of the forces and moments on the blades was required. Accordingly, a study of hydrodynamic loading on the propeller blades during normal operation and maneuvering was conducted. Since hydrodynamic studies are the principal concern of other contractors, our work was limited to that necessary to provide approximate data.

SHIP DESIGN STUDY

A brief ship design study incorporating the propulsion and pitch changing equipment was also undertaken. An existing submarine design formed the basis for this study, and minimum modifications in hull design and arrangement were made to adapt the vessel to tandem propeller propulsion. This design provides some indication of what can be expected, but is a cursory, "first cut" and does not represent an optimized design.

II

CONCLUSIONS

PROPULSION MACHINERY

Turboelectric propulsion machinery with flooded, inside-out, squirrel cage induction motors was determined to be feasible in an earlier study of the Novel Electric Propulsion System. In the present study, in which the General Electric Company again participated by investigating the propulsion machinery, the same conclusion is reached for propulsion machinery with flooded, inside-out, synchronous motors. This conclusion is concurred in by the second machinery subcontractor, the Elliott Company, in a concurrent but independent investigation. In addition, it is concluded that propulsion machinery incorporating synchronous motors has, although not grossly so, a significant advantage over machinery with induction motors. For general orientation, a cutaway view of the General Electric synchronous motor is shown in Figure 3.

Machinery preliminary designs by the two subcontractors are different in many respects, and are presented in Sections VI and VII of this report. No comparative evaluation is made since the Elliott work is not yet complete, but this evaluation will be included with the supplement reporting the remaining Elliott work.

PITCH CHANGING SYSTEMS

With respect to outright feasibility, all three systems studied for positioning the propeller blades are determined to be feasible. However, it is concluded that the electric system is most promising, the mechanical system less promising, and the hydraulic system decidedly unpromising. The investigation is primarily directed toward outboard equipment, for although the inboard equipment is developmental hardware there is no question of feasibility involved. Development of both inboard and outboard equipment is within the present state-of-the-art.

GENERAL

The foregoing conclusions on feasibility apply to obtaining workable propulsion machinery and pitch changing systems. The hydrodynamic performance of the ship, its overall control, and its operational uses are beyond the scope of this report and are the subject of investigation by other contractors.

SHIP DESIGN STUDY

The brief ship design study, consisting of fitting the tandem propeller propulsion system to an existing design submarine with minimum modification, shows significant increases in length, displacement, and cost, and a decrease in speed. This cursory effort, however, does not represent an optimized design nor does it utilize the full potential of the propulsion system.

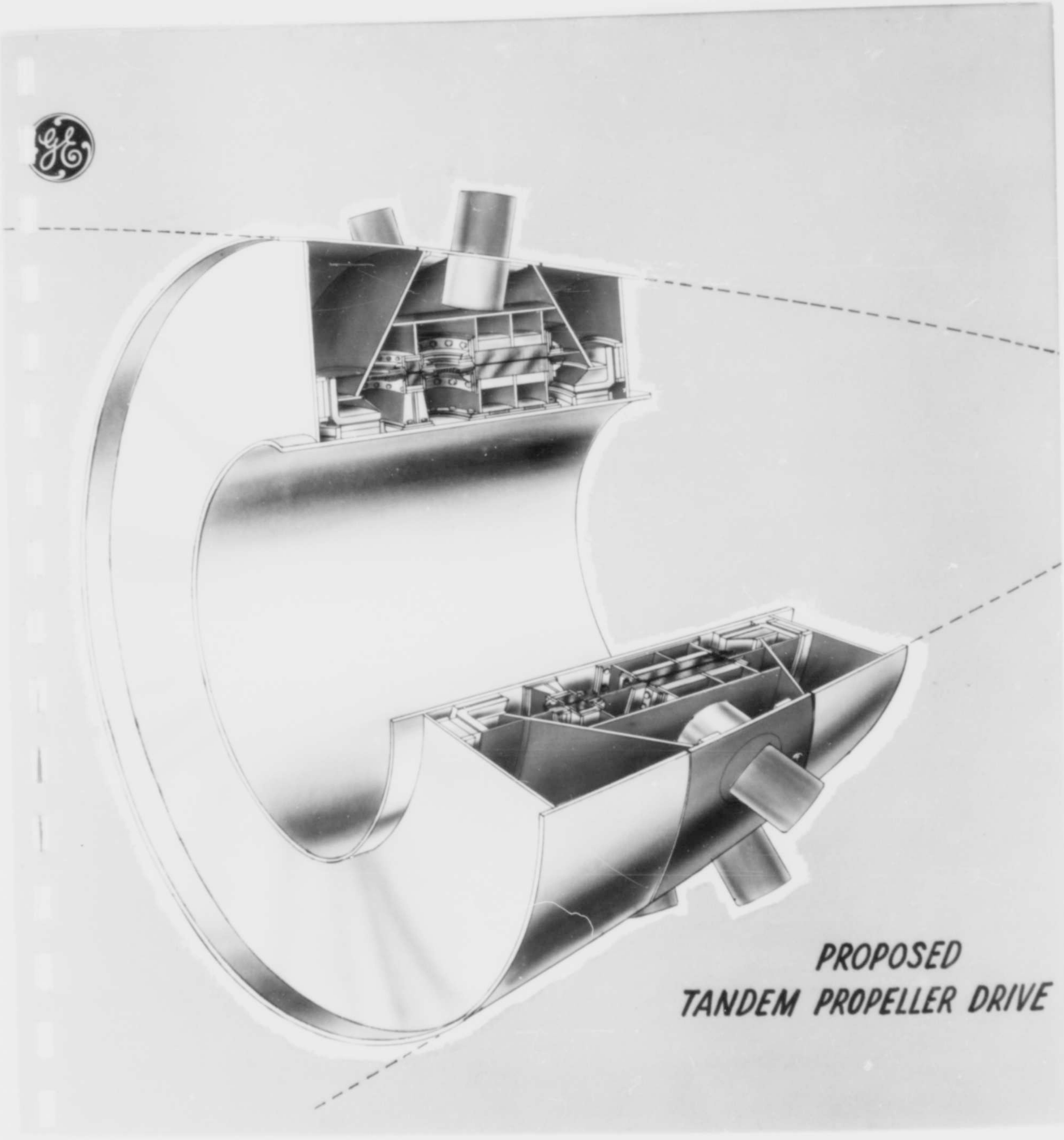


Figure 3. Cutaway View of General Electric Synchronous Propulsion Motor

III

SHIP DESIGN INCORPORATING TANDEM PROPELLERS

In order to obtain some indication of the ramifications of the Tandem Propeller propulsion system installed in a ship, a brief ship design study was undertaken. Starting with a current FBM submarine design, SSB(N) 616, the propulsion system was fitted with minimum modification of the ship. The resulting ship arrangement is shown in Figure 4, and is discussed later in this section.

Since the basic concept of the Tandem Propeller system is to generate thrust in any desired direction and, hence, control the motion of the vessel in all directions, the control surfaces (rudders, stern planes, and sail planes) have been omitted. Similarly, the hovering system for depth control and the gyroscope for roll control have been omitted. It is most probable however that the installation of either automatic control or fixed fins at the stern, vertical and horizontal, will be necessary to maintain directional stability. Fixed fins would be of about the same area as the existing stern appendages.

This study is a "first cut" at ship arrangement and size, sufficient to establish feasibility and penalties and/or gains from a ship design viewpoint, but it is not a completely engineered contract design.

ARRANGEMENT

Referring again to Figure 4, the forward propeller has been designed to fit around the torpedo tube bundle.* In this position, however, the

* Propulsion motors and propellers shown in Figure 4 are of slightly different diameter and length than those discussed elsewhere in this report, in order to better fit the ship.

present sonar suit of the 616 is displaced. The most likely place for the AN/BQR7 conformal array is around the sail, and that for the AN/BQS4 hydrophone array is in front of the present sail with the sail extended and faired around it. No investigation has been made into the practicability of these sonar locations.

The anchor and chain locker have been moved to the stern clear of the after propeller. There should be no problem with this location although an interference might exist with the reinstatement of stern appendages. These could be added in an X-configuration, however.

Since it is desirable that the after propeller be the same diameter as the one forward, and since the rotating element should be supported on pressure equalized structure to avoid distortion, the after body lines required modification to provide the required diameter and reasonably fine run.

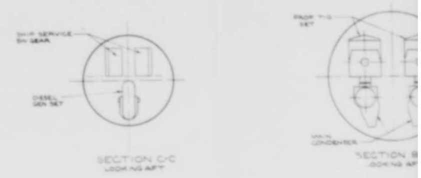
The elimination of the propeller shaft and propeller at the stern permitted the installation of hind-sight sonar in the aftermost position.

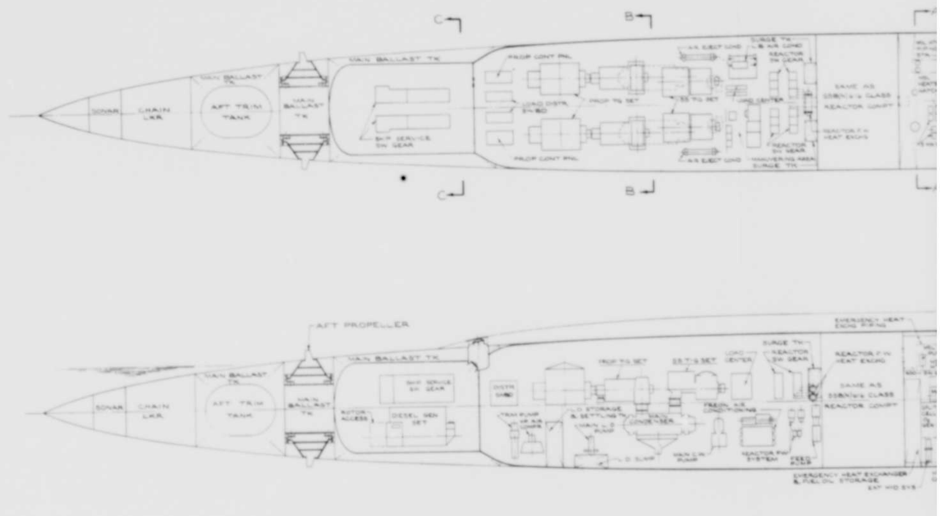
The propulsion TG sets are so large that they will not fit in the existing 616 engine room. Hence, the auxiliary machinery space containing the diesel generator set was relocated aft of the engine room, permitting the engine room to be moved relatively forward to take advantage of the increasing hull diameter. The reactor systems, normally in this auxiliary machinery space, have been relocated to the forward end of the engine room.

The after ballast tanks have been moved to the extended stern from between the reactor compartment and engine room. This arrangement eliminates two pressure hull transitions, an improvement from the fatigue aspect.

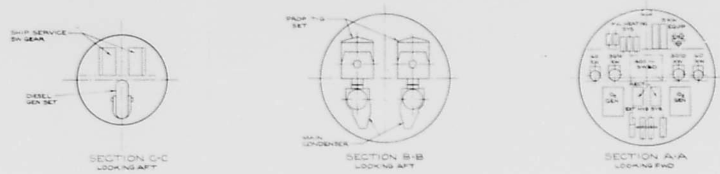
The auxiliary machinery space forward of the reactor compartment has been reduced by the elimination of the gyro stabilizer and depth control tanks.

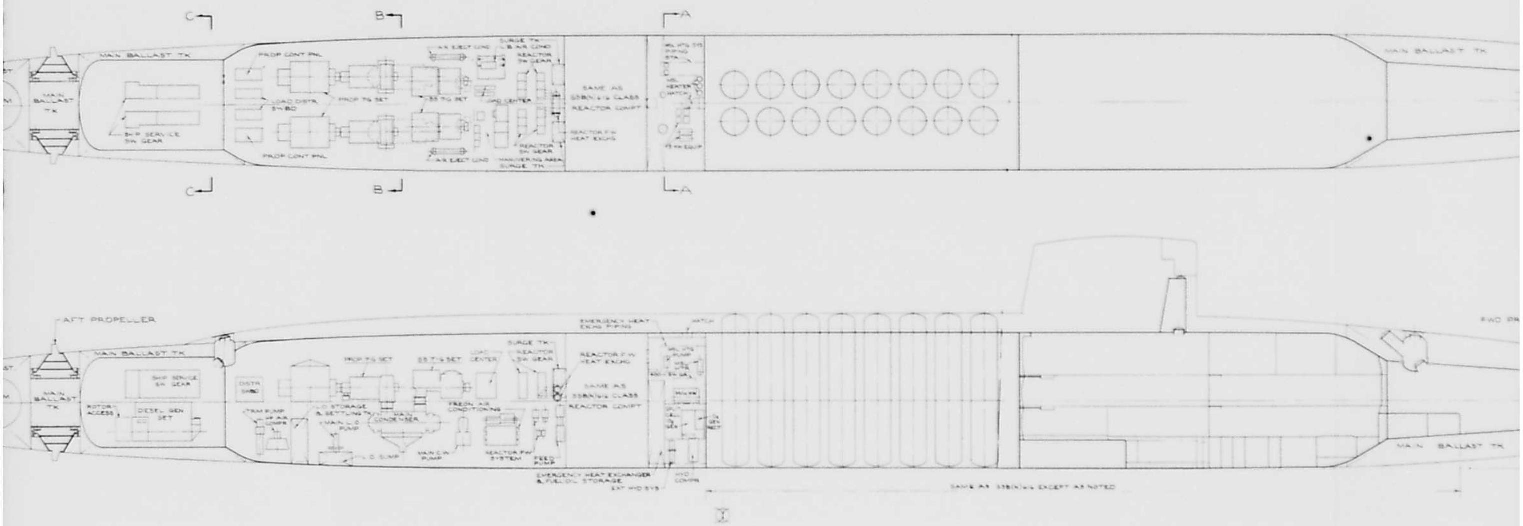
This drawing represents a first attempt to fit the Tandem Propeller propulsion system to SSB(N) 616 with minimum modification. It does not represent a final design. Machinery arrangements demostrate adequate space allocation but are not necessarily final arrangements. Longitudinal balance has not been determined.



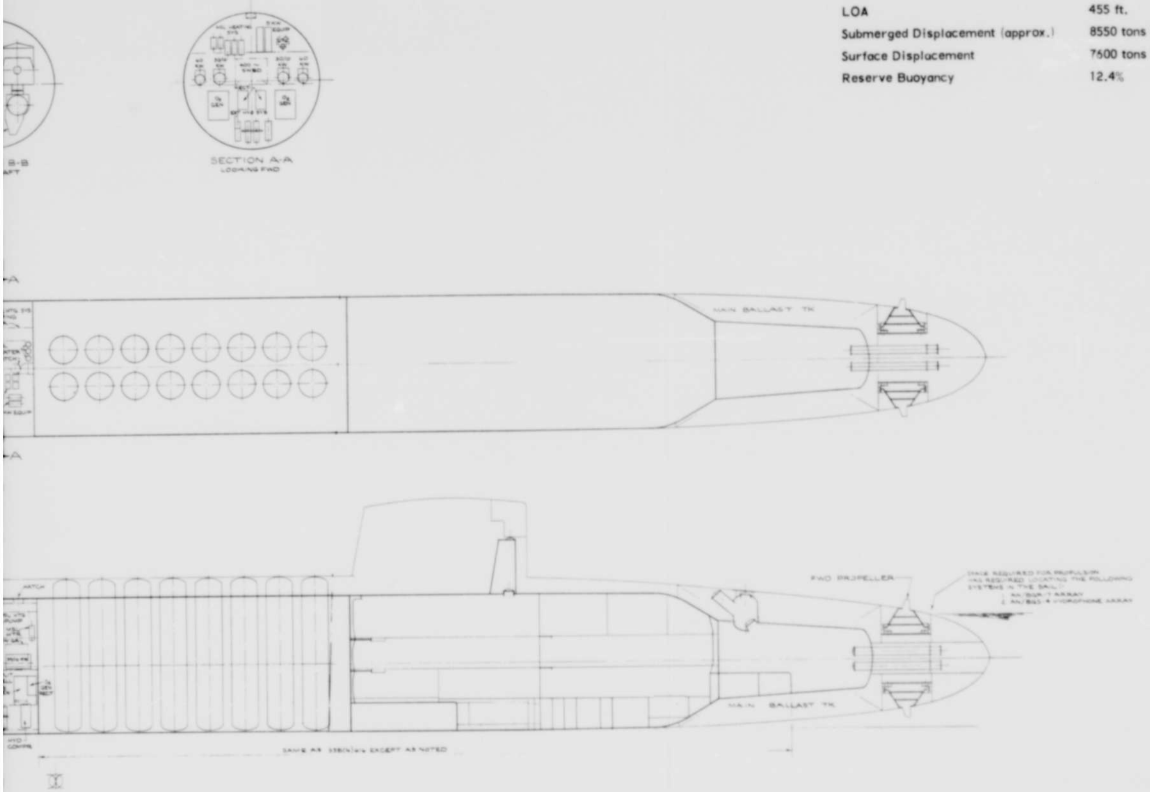


the Tandem Propeller
n modification. It does
ngements demostrate
rily final arrangements.





Figu



LOA	455 ft.
Submerged Displacement (approx.)	8550 tons SW
Surface Displacement	7600 tons SW
Reserve Buoyancy	12.4%

Figure 4. General Arrangement of Ship

The steering and diving instrument panel and control sticks have been replaced by new ship control equipment, in substantially the same space.

COMPARISON

The following is a tabulation of the approximate ship characteristics as compared with the SSB(N) 616:

	Tandem Propeller Ship	SSB(N) 616
Length, OA	455 ft.	425 ft.
Submerged Displacement	8550 tons	8250 tons
Surface Displacement	7600 tons	7320 tons
Reserve Buoyancy	12.4%	12.7%

Without stern appendages and without sail modifications to accommodate displaced sonar equipment, the skin area is about 1400 square feet less than SSB(N) 616 with appendages. This is about 3% less and by itself results in about a $1\frac{1}{2}$% increase in speed, all other things being equal. This condition will be reversed if the stern appendages are reinstated. However, there most likely will be an increase in resistance in either event due to the accelerated flow over most of the body by the forward propeller and over the afterbody surface by the after propeller. The evaluation of this effect can only be accomplished by model tests.

The mismatch between the power rating of the SSB(N) 616 propulsion machinery and the Tandem Propeller machinery studied will naturally affect the speed, but will not be discussed here since classified information is involved. It is evident, however, that if 616 power at the turbine shafts is retained, a speed reduction will accrue, since about 20% of the power is lost in transmission from turbine shaft to propeller, compared to about 3% loss for a geared drive. Improved propeller performance (compared to a screw propeller) may recover a small part, but not all, of this loss.

A rough weight estimate indicates that there will be a net weight increase of about 400 tons. Since this is greater than the increase in displacement, there will probably be a loss of 100 tons of lead ballast or a further increase in volume and displacement. No work has been done to evaluate the longitudinal center of gravity versus center of buoyancy which, in turn, will determine whether the center of lead ballast is reasonable. Further work would, of course, include a detailed weight and moment study to insure that weight and buoyancy are balanced.

It is also emphasized that the arrangement of the machinery components is by no means a final arrangement; its purpose is to establish that the space allocation is sufficient.

In summary, it has been established that the physical installation of the Tandem Propeller system in an FBM submarine is feasible but will result in an increase in length of about 30 feet and an increase in displacement of 300 to 400 tons compared with SSB(N) 616. There will also be a reduction in speed.

IV

MACHINERY SPECIFICATION AND HYDRODYNAMIC STUDY

MACHINERY SPECIFICATION

In order to provide a sound basis for the propulsion machinery and pitch changing system preliminary designs, a brief specification was prepared. This was written in the general form of a hardware purchase specification, describing the functional requirements, performance requirements, environmental conditions, and life. It is reproduced in its entirety in Appendix A. The salient points are:

> The two propellers are each driven by a large diameter inside-out flooded motor, located within the propeller hub. Choice between induction or synchronous type is optional. Full speed is 50 rpm.
>
> Each motor is energized from its own steam turbine generator set. The TG sets are speed governed in accordance with signals from the ship control system. Each turbine is rated approximately 7500 shp.
>
> There are sixteen blades on each propeller, variable in pitch both collectively and cyclicly, in accordance with three signals from the ship control system (collective pitch angle, cyclic pitch angle, and angular orientation of cyclic pitch variation with respect to the hull).

Range of blade position (with respect to plane of propeller disc)	+90° to -50°
Design ahead pitch	+35°
Collective pitch (with respect to plane of propeller disc)	+90° to -30°
Collective pitch accuracy	± 1°
Cyclic pitch (with respect to collective position)	+20° to -20°
Cyclic pitch accuracy	± 2°
Cyclic pitch axis	Infinitely variable
Cyclic pitch axis accuracy	± 10°

Maximum torque on blade during normal operating conditions	± 1500 lb-ft
Maximum torque on blade during abnormal operating conditions	± 7000 lb-ft

Cyclic pitch waveform is not critical

Submergence pressure is 600 psig, but all outboard equipment is to be inherently insensitive to ambient pressure.

Minimum period for extensive maintenance or overhaul is one year.

Since the optimum cyclic pitch waveform was not known, no restriction was placed on cyclic pitch waveform, except of course that it be periodic and that its period correspond to the propeller speed. Since the waveform was indeterminate the accuracy at all points could obviously not be specified. The accuracy was therefore specified only at the maximum and minimum peaks, and it was assumed for this study that whatever the intervening waveform might be it would be adequately reproducible.

With respect to control, a completely flexible approach was taken. Of the four parameters which determine propeller performance:

Collective pitch
Cyclic pitch
Cyclic pitch axis
Propeller speed,

each can be varied continuously and independently of the other three. However, when the hydrodynamic study is completed by others, it may be possible to simplify or restrict the control system. For example, operation with only two discrete collective pitch angles might be satisfactory.

HYDRODYNAMIC STUDY

In order to prepare this specification, it was necessary to obtain a knowledge of the forces and moments on the propeller blades as well as the extent and accuracy of required collective and cyclic pitch changes.

Other contractors were investigating these items in detail, but the results were not to be forthcoming for several months. We therefore performed our own brief study, limited to relatively simple approximations, so as to obtain results early in the program with accuracy sufficient for our purposes. The hydrodynamic study consisted of:

Selection of blade section and plan form, and determination of lift, drag, and center of pressure as a function of angle of attack.

Determination of axial and tangential force on a blade running at a given rpm and ship's speed, but with varying angle of attack.

Determination of maximum possible blade loading and moment.

Summation of forces on the propeller running at a given rpm, mean pitch, and ship's speed, but with a sinusoidally varying pitch increment of several amplitudes.

This study is reported in detail in Appendix B, and the desired results are incorporated in the previously discussed machinery specification.

For this study a position type of system was selected as the most obvious and straightforward approach, but other approaches should be considered by those making the major hydrodynamic study. For example, in a position type of system, the positioning accuracy must be in the neighborhood of $\pm 1°$ because the load assumed by the propeller blade is very sensitive to angle of attack. However, if instead of a position being imposed upon the blades a torque were imposed, the desired loading would be more readily attained because the loading is a much less sensitive function of torque than position. This could lead to simpler hardware and less critical control systems for pitch changing.

Similarly, a sinusoidal cyclic pitch waveform was selected as the most convenient waveform. Here again, other approaches should be considered by those making the major hydrodynamic study, so as to determine the optimum waveform. For example, cyclic pitch might be changed in step fashion, by quadrants. In two opposite quadrants, in which the blades have substantial velocity components in the direction in which transverse force is desired, the blades have off design pitch (one increased pitch, one decreased pitch). In the other two quadrants, in which the blades have relatively small velocity components in the desired transverse direction, the blades have design pitch and maintain maximum longitudinal propulsion efficiency. Another area of possible gain may be an irregular pitch variation to enable the propeller to accommodate any irregular wake present.

V

MECHANICAL PITCH CHANGING SYSTEMS

PITCH CHANGING MECHANISM CLASSIFICATION

It was noted early in the program that the very large number of different pitch control systems which can be conceived are really multifarious combinations of a smaller number of parts. The entire pitch control system can be subdivided into several functional areas. Equipment for each function can at first be considered more or less independently of other parts of the system. For example, the rack and pinion, sector and pinion, or simple crank can be considered to move a blade whether power is supplied by a mechanical link, a hydraulic cylinder, or an electric motor and screw. The functional areas are:

(1) Controller: Interprets ship control system output signals and converts them to required form for blade pitch control.

(2) Information transfer to the rotor: Transfers blade pitch information from the controller to the rotor.

(3) Power transfer to the rotor: Transfers mechanical or electrical power for pitch changing from the hull to the rotor.

(4) Blade actuator: Moves the blades by means of the power provided by (3) and in accordance with the information provided by (2).

(5) Feedback systems: As required for accurate control.

The controller, located inside the ship, is an electric or electromechanical system. A great variety of types of systems is possible, according to the type of information and power transfer systems selected. Two information transfer systems are considered feasible: mechanical, and electrical. (Optical and hydraulic systems are ruled out because of marine growth in the case of the former and the impossibility of maintaining an adequate seal for the latter.) There are several ways of accomplishing the results by each method, however.

Similarly, mechanical or electrical systems can be used for power transfer. Blade actuators can be mechanically, electrically, or hydraulically powered. Twelve combinations can thus be formed, but of these seven are not possible since, for example, hydraulic blade actuators require electrical power transfer. The permissible combinations are:

Information transfer	Power transfer	Blade actuator
Mechanical	Mechanical	Mechanical
Mechanical	Electrical	Electrical
Mechanical	Electrical	Hydraulic
Electrical	Electrical	Electrical
Electrical	Electrical	Hydraulic

Various manifestations of each of these systems are discussed below, with reasons for or against each one. In judging each system some consistent set of standards must apply. Each system is conceived in such form that it can, in all probability, meet the minimum requirements of the specification.* The following qualitative requirements will then provide the standards upon which the systems can be rated relatively:

Reliability
Noise
Compatibility with submarine environment
Simplicity
Flexibility
Accuracy compatible with system requirements
Life
Efficiency
Casualty control
Ease of installation
Maintenance
Size
Weight
Cost

---

* Appendix A

Conceptual design of pitch changing systems was carried out by three separate groups under the direction of Electric Boat: An Electric Boat team emphasizing mechanical systems, a General Electric Company group emphasizing electrical systems, and an Elliot' Company-Electric Boat team emphasizing electrohydraulic systems. It was our intention in assigning team emphasis not to limit any group in their thinking, but only to insure serious consideration of each of the three major types. The results of Electric Boat mechanical studies are discussed in this section. Sections VI and VII deal with the work of the General Electric electrical system group, and the Elliott-Electric Boat electrohydraulic group, respectively.

Primary effort of Electric Boat has been applied to all-mechanical systems, although some consideration has been given to electric and hydraulic systems. A preliminary survey of system components narrowed the possible types of mechanisms down to a manageable number. Preliminary design studies of complete systems were then carried out to consider in more detail such factors as size, weight, potential noise sources, accuracy, and packaging to reduce hydrodynamic losses.

MAIN PROPULSION MACHINERY

A ship set of main propulsion machinery consists of two flooded main propulsion motors, two turbine generator sets, and two propulsion control panels. These panels contain control equipment associated with the main motors and TG sets, but not the pitch changing system. They are normally unmanned stations. During normal operation, the main propulsion machinery operates fully automatically, with its speed responding to signals from the ship control system.

Specific Machinery Designs

The propulsion machinery designs by the General Electric Company and the Elliott Company are discussed extensively in Sections VI and VII of this report, and this discussion will not be repeated here. The pitch changing system drawings in this section are based upon the General Electric propulsion motor; however, the system is also adaptable to the Elliott propulsion motor. Both of these motors are radial air gap machines.*

Axial Air Gap Machine

For use with the mechanical pitch changing system, the axial air gap machine was also considered since it allows radial access from the stationary hull through the motor to the interior of the rotating propeller hub. This, however, turned out not to be a very promising approach.

Because of the very large axial magnetic force between the rotor and stator, this machine must be divided into two half-size machines, mounted back to back so as to balance the axial forces. This requires exceedingly rigid cantilever type of structure, as compared to hoop type of structure in the axial gap machine. Since it consists of two machines instead of one machine, there is some reduction in efficiency, and a first look showed no saving in space. In addition, a formidable manufacturing problem exists, since the distance between slots in the punchings varies continuously with the radial position of the punchings in the stack.

Since there was no outstanding advantage to the axial air gap machine and since satisfactory radial access to the propeller hub was possible with the radial air gap machine, the axial air gap machine was not studied further.

Hull Electrical Penetrations

An item essential to the propulsion machinery but not directly a part of it is the hull penetration for the motor power cabling. This was investigated during the NEPS study, and the results are directly applicable here, although some changes in dimensions are required to

* Throughout this report conventional motor terminology such as "air gap" and "windage loss" is used, although the motors are in fact immersed in water.

suit the different voltage and current ratings. The penetration consists of a metallic web with three large pins glass-sealed to the web to form the basic water barrier. The outboard side of the penetration is suitably potted to exclude sea water from energized parts. A separate penetration and cable is used for each motor circuit, keeping the penetration current rating manageably low and providing system advantages discussed in other sections of this report. A more extensive description of the penetration can be found in the NEPS report.* The penetration is a development item, but the level of confidence in satisfactory development is very high.

RECOMMENDED MECHANICAL PITCH CHANGING SYSTEM

The result of Electric Boat Division's study of mechanical pitch changing systems indicates that the most favorable device is that represented in Figure 5.

This mechanism utilizes a nonrotating wobble plate, positioned by hydraulic cylinders. A pivoted arm associated with each blade terminates in a sliding bearing which bears on and is positioned by the wobble plate. Each lever arm has a sector gear machined on its outer rim transmitting motion to a bevel gear on its respective blade spindle.

This device is extremely simple and has a minimum of parts, bearings, and joints. It is arranged to be installed or removed as a package unit, including blade, gears, lever arm and slider bearing. It can be provided with mechanical back-up and indicator features, and allows minimum hub diameter, although causing a lengthening of the entire motor structure. The bearings and gears are expected to have a life in sea water consistent with the specification.

Gear noise can be reduced by insertion of a strip of hard rubber in the gear face to reduce possible backlash noise. The packaging of this

---

* General Dynamics/Electric Boat Report C-411-62-024, "Feasibility of a Novel Electric Power Propulsion System for a Submarine," Volume I of III, Engineering Study, CONFIDENTIAL.

unit can easily be adapted to enclose the gears and bearings in a pressure compensated oil-filled housing. This will reduce gear noise and wear, and increase reliability.

It is felt that a pitch changing system of this type suitable for this application can be developed.

PRELIMINARY SURVEY

Mechanical System Components

Information Transfer

Three methods of information transfer have been considered:

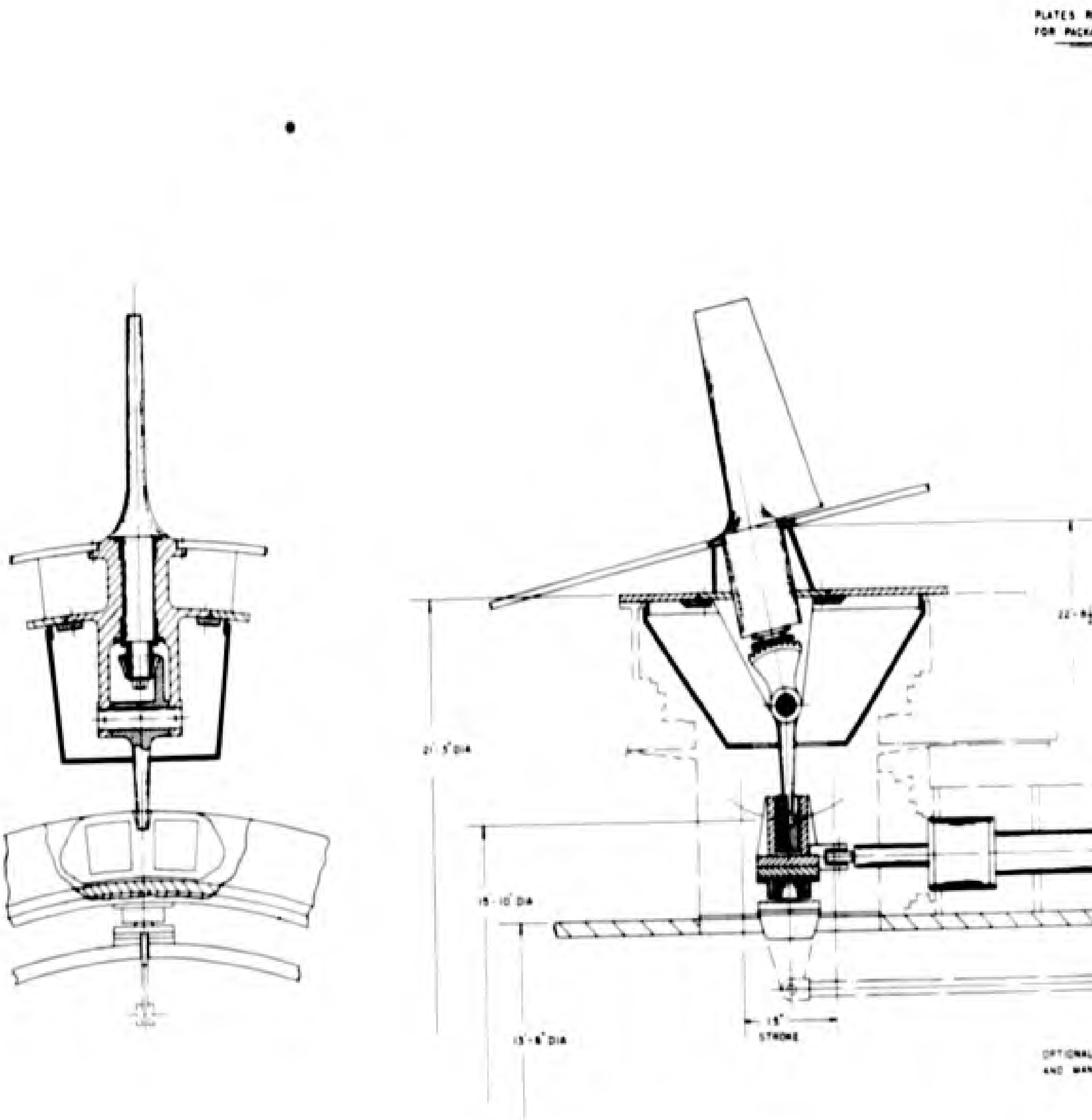
(1) Wobble plate, Figure 6A. This system consists of a plate or ring around the hull near the propeller hub which can be moved forward or aft, and rotated about any transverse axis. Blade pitch is proportional to the position of the plate at the location of the blade. The system is thus a direct mechanical analog of the required pitch.

A "feeler " or push rod provided for each blade moves in or out depending upon wobble plate position as the propeller rotates. In its simplest form the feeler is fitted with a bearing slider which runs on the wobble plate. Since the feeler must transmit load in both directions, a pair of bearings is required.

Either type of bearing shown in Figure 6D can be used. However, since the propeller blade push rods are rotating with the hub, consideration must be given to their hydrodynamic drag. The internal slide type is preferred for this reason.

Layouts have demonstrated that it is possible to provide the required motions of the parts without ambiguity and maintaining all parts fully captive at all times. The equipment involved is simple and rugged, but may lead to large hydrodynamic losses and difficult bearing design. This system appears to be feasible, and to possess to a high degree the qualities desired as far as they can be assessed in a subsystem.

(2) A variation of this (Figure 6B) consists of a wobble plate mounted on the rotor, positioned by slipper bearings which are in turn mounted from the hull. This system has similar characteristics to the nonrotating wobble plate but involves simpler links between the propeller blades and the wobble plate. Hydrodynamic eddy-making may be reduced since the blade actuator links can be contained within the hub and the wobble plate presents a smooth surface to the flow. A disadvantage is that the wobble plate, being mounted on the rotor, can contribute to rotor unbalance if it is not maintained in accurate concentricity.


21'-5" DIA
15'-10" DIA
13'-8" DIA
15"
STROKE

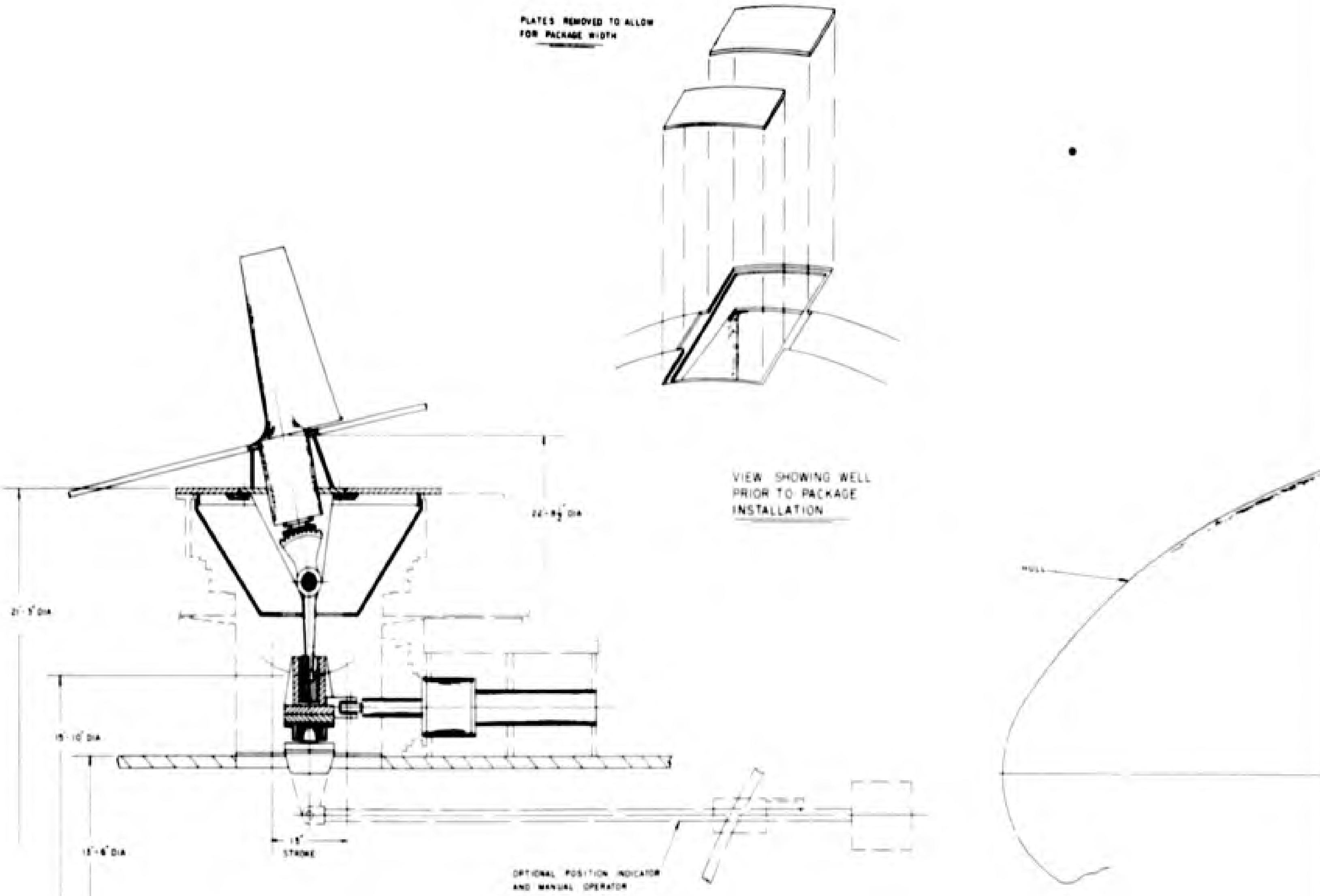


Figure 5. Stationar

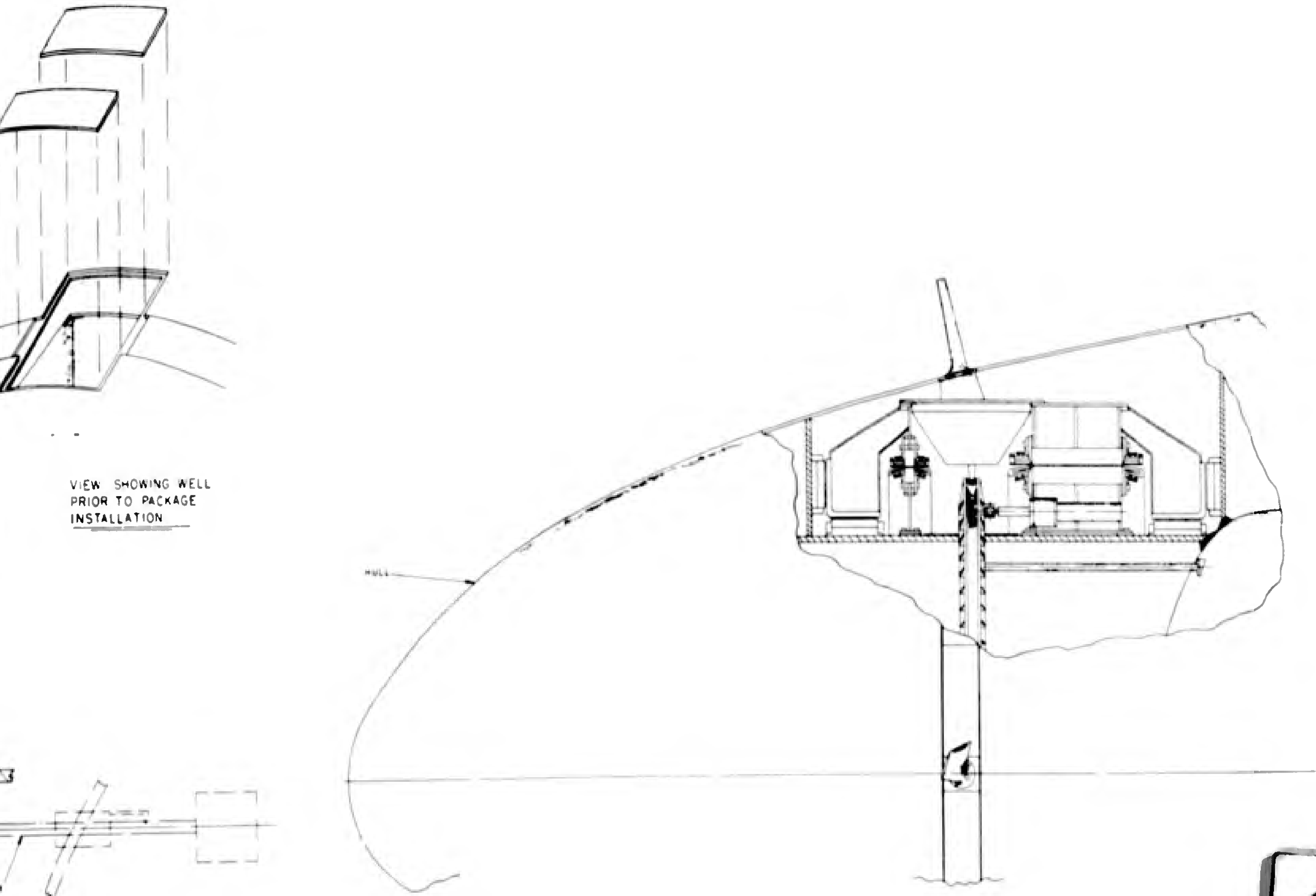


Figure 5. Stationary Wobble Plate, Sector and Bevel Gear Drive

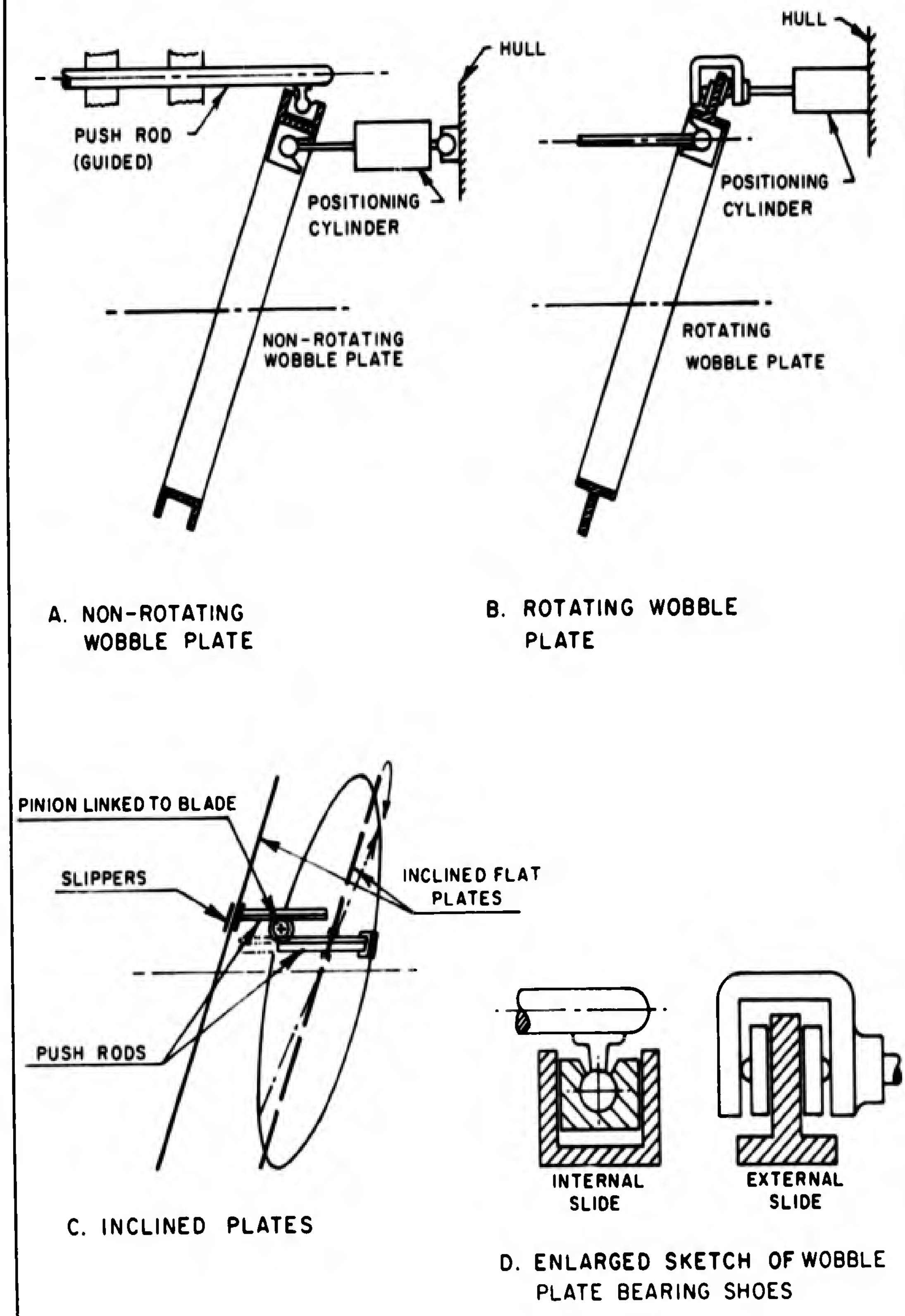


Figure 6. Mechanical Information and Power Transfer System

(3) A third mechanical system (Figure 6C) consists of a pair of plates or rings inclined a fixed amount and separately rotatable. The distance between the two plates can be varied by rotating one with respect to the other, while the location of the maximum (or minimum) distance can be varied by rotating both together. A follower for each blade can be devised which moves the blade according to the separation of the plates. This system is much more complicated than the first two, and cannot operate as rapidly. Since no advantages accrue to this system, it was dropped from further consideration.

Power Transfer

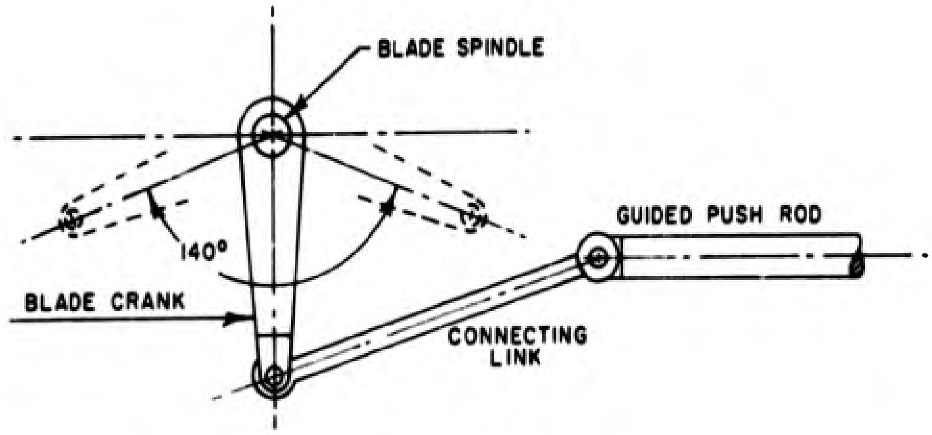
In the case of the wobble plate systems, the power transfer is accomplished by the same mechanism that provides information transfer. Direct linkage from the wobble plate to each blade is provided. With a non-rotating wobble plate, each blade link has a pair of slipper bearings running on the plate. In the case of a rotating wobble plate, the blade links can be attached by means of a ball joint. The wobble plate positioners are then provided with slipper bearings. Either system will fulfill the requirements.

Blade Actuator

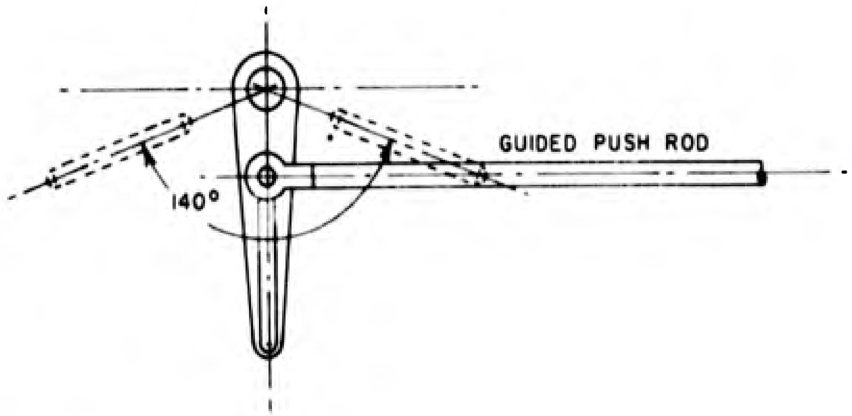
With an analog of blade pitch available in the form of longitudinal motion of a link, it is a simple matter to devise means for rotating the blades. Initially the specified minimum travel was 140°. Later it became evident that a greater range of travel, perhaps 160°, would be desirable to permit use of the propeller blades for steering when the rotor is not turning. Possible mechanisms involve gears, racks, or cranks.

(1) Simple crank, Figure 7A. The design pitch of the propeller is at approximately the center of the total range of pitch. For this reason the region of maximum accuracy, where the crank is at right angles to the push rod, fortuitously occurs near the design pitch. The mechanical advantage and hence, also the positioning accuracy, would be about a third as great at the extremes of pitch which is limited to about 140°.

The travel of the push rods must be relatively long with the simple crank. Allowing a minimum moment arm of 4 inches, the travel to accommodate 140° would be 22 inches. The fact that



A. SIMPLE CRANK, SHOWN WITH GUIDED PUSH ROD FOR NON-ROTATING WOBBLE PLATE. WITH ROTATING WOBBLE PLATE, CONNECTING LINK WOULD BE ATTACHED DIRECTLY THROUGH BALL JOINTS.



B. CRANK AND SLIDE

Figure 7. Blade Actuators - Sheet 1

the pitch change is not a linear function of travel complicates the pitch control signal generator.

(2) Crank and slide, Figure 7B. If instead of joining the connecting link to the blade crank with a pin, a slider is provided working in a groove in the crank, the accuracy is reversed. Here the least accuracy occurs at the mid-position where mechanical advantage is least. Push rod travel is as great as or greater than in the case of the simple crank. The relationship between pitch and push rod travel is also nonlinear, but follows a simpler pattern.

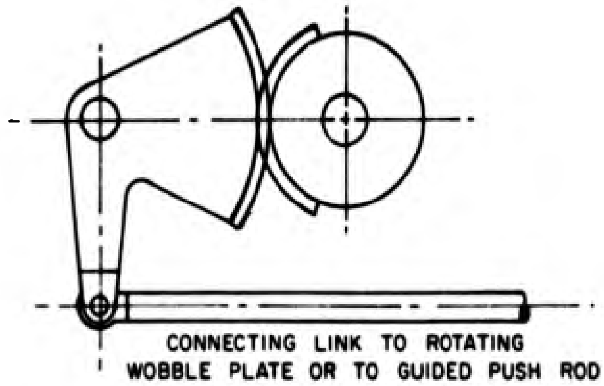
(3) Bell crank with sector and pinion, Figure 8C. By suitable selection of the gear ratio and crank length, push rod travel can be freely chosen. The mechanical advantage is very nearly uniform throughout the range of travel. Systems of this type can be devised in which no sliding members are required. It is expected that pin joints or ball and socket joints would be less susceptible to looseness and rattling than sliding members.

(4) Rack and pinion, Figure 8D. This mechanism has the simplest kinematics of all those described. It generates angular changes proportional to push rod movement, and has constant mechanical advantage. Push rod travel can be selected freely. Used in conjunction with a nonrotating wobble plate, the rack can be directly attached to the push rod. With a rotating wobble plate a link is required between the guided rack and the wobble plate. Tooth loading on the pinion is slightly lower meshing with a rack than with a gear sector. This may be important in these sea water lubricated gears.

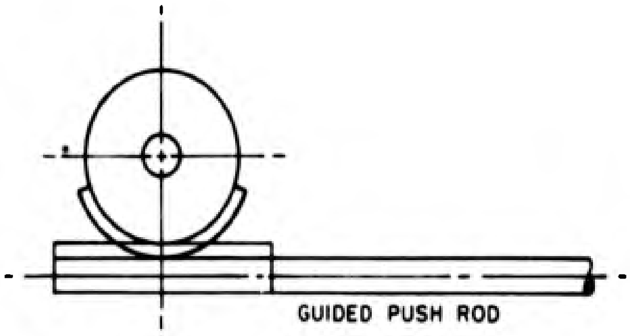
Wobble Plate Positioning System

Total force on the wobble plate may reach 50,000 lbs. during normal pitch changes, and may be even higher during transients. Speed of motion of the wobble plate is in the order of 1/10 foot per second. Required positioning accuracy is in the order of 1/10 inch. The demands of this system, relatively large forces, low speeds, and high accuracy strongly indicate a hydraulic piston positioning system, although a rack and pinion or lead screw could be used. Also, hydraulic power for applications of this sort is standard submarine practice.

Occasional transients may impose loads 4 or 5 times the normal loading on the wobble plate. A system which is not reversible, such as a lead screw, must be built strongly enough to withstand these occasional forces. The rack and pinion system requires a very large speed



C. BELL CRANK WITH SECTOR AND PINION



D. RACK AND PINION

Figure 8. Blade Actuators - Sheet 2

reduction which would probably have to be a worm and gear, also a nonreversing device. A reversible system "rolls with the punch," and can be designed for lower peak loads.

The feature of reversing, or overhauling, is important for another reason too. The minimum number of positioners to determine the position of the wobble plate is three. Any additional positioners are redundant. It appears necessary, however, to use four or six so as to distribute the loads on the wobble plate and reduce its deflection. The load-sharing tendency of a hydraulic system is advantageous in the event of minor discrepancies in control signals, and prevents serious overloading as, for example, in the event of malfunction in control of one positioner.

Hydraulic cylinders or rams have given satisfactory service in submarines both inside and outside the hull. Either location can be used for this application. The type of wobble plate, rotating or nonrotating, may influence the choice of interior or exterior mounting of the cylinders, however. In the case of the nonrotating wobble plate the positioner shafts may be attached thru ball joints. A small degree of translation of the shafts must be provided for as the plate is tilted. This can be accomplished by attaching exterior positioning cylinders to the hull through ball joints, allowing the cylinder to rotate slightly to accommodate the extraneous movements, or a crosshead can be used to connect a rigidly-mounted cylinder to a connecting rod. Alternatively, the positioning cylinders may be attached to the wobble plate mounting fixtures and not to the wobble plate itself. These mounting fixtures may have simple translational motion, permitting cylinders to be rigidly mounted. The rotating wobble plate is positioned by bearing shoes mounted on a guided shaft. The translational motion of the point of contact on the wobble plate is accommodated by sliding along the bearing shoes. The cylinder can therefore be rigidly mounted inside or outside the hull without need to resort to a crosshead.

Wobble Plate Mounting

The wobble plate, whether rotating or not, requires three degrees of freedom relative to its supporting structure: Translation along the ship's centerline, and rotation about the transverse axes. It must be prevented from rotating about the ship's centerline or moving off the centerline.

A simple support system which fulfills the requirements consists of a spherical member which can slide back and forth, and upon which the wobble plate rotates (merely a large ball joint), Figure 9A. The spherical surface can be splined to prevent rotation about the longitudinal axis. This type of support is not practicable, however, in view of the large diameter of the hull on which the propeller and motor are mounted. The spherical surface is of too small an arc length to provide against jamming, and the clearance required in a sliding bearing some 15 feet in diameter will render the location of the plate uncertain.

A satisfactory support system, Figure 9B is provided by a number (at least 3) of radial pins or sliders, each arranged to slide radially in guides attached to the plate. A link with ball joint at the slider end and a hinge at the other end connects the sliders to the hull. The hinge is oriented tangentially to the hull, so that the wobble plate is restrained from moving tangentially. It can be seen that a series of such supports prevent the undesired rotation or translations. Tilting of the plate to produce cyclic pitch requires a small degree of flexibility in the mounting which can be provided by stiff rubber if built-in tolerances do not allow sufficient motion.

Since the angularity of the wobble plate is very small, not over 2° at extreme cyclic pitch, elastomer units shown in Figure 9C can be used in conjunction with sliding guides in place of the pivoted members described above.

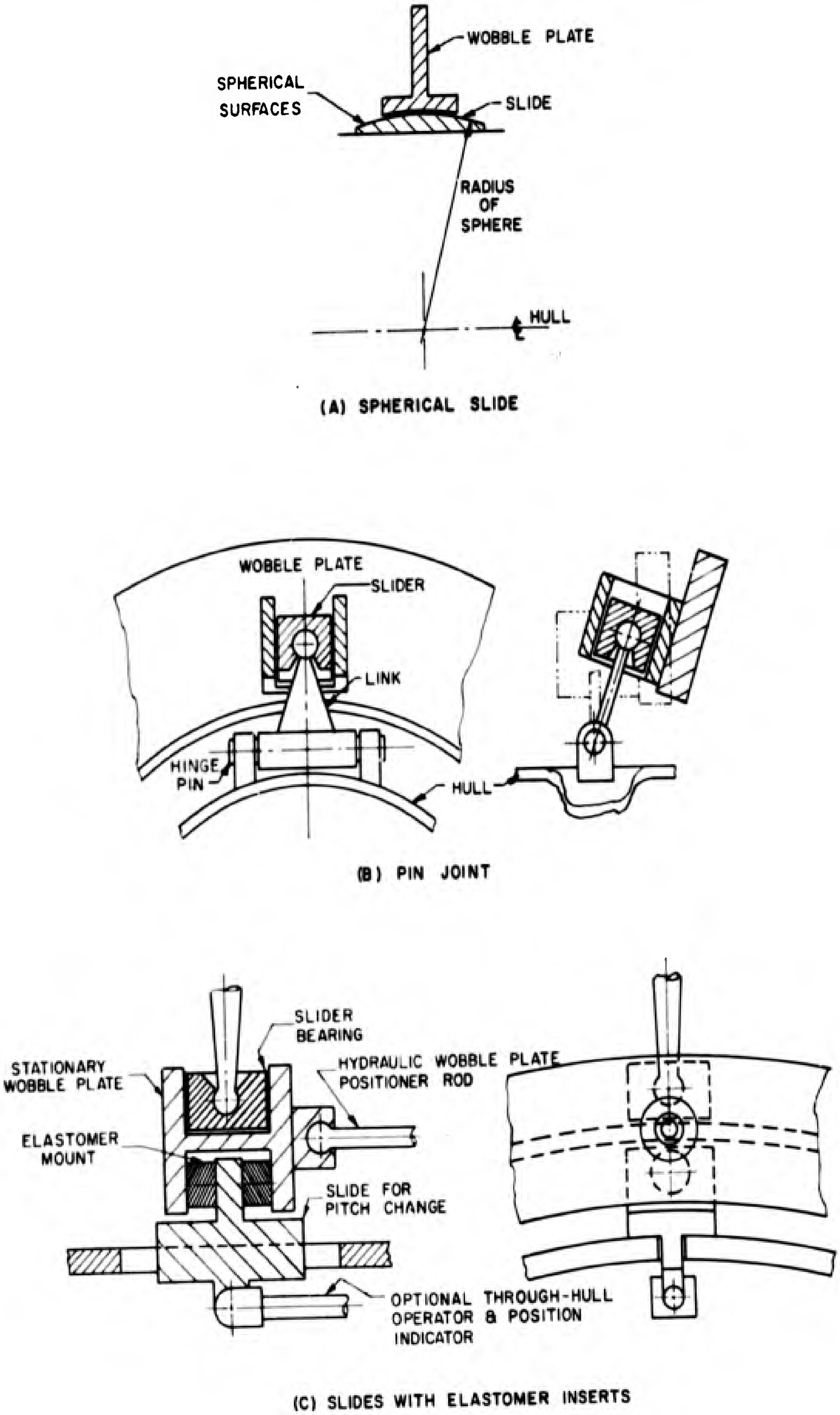


Figure 9. Wobble Plate Mounting

Propeller Blade Mounting

Consideration of blade mounting led to the conclusion that the blades should be arranged with their axes perpendicular to the hull surface at the point of mounting. This results in a minimum of blade clearance or hub unfairness upon blade rotation as shown in Figure 10. An ideal mounting utilizes the outer surface of the rotor formed as a spherical arc generated on the same diameter as the hull diameter. The lower edge of the blade also has this arc, resulting in constant clearance of the blade from the hub throughout blade rotation. This follows practice of variable pitch hydraulic turbines and marine propellers.

Complete Systems

The separate mechanical parts combine into two potentially useable systems:

Nonrotating Wobble Plate System

A wobble plate supported on the hull is moved forward or aft or tilted by means of hydraulic cylinders. Slipper bearings running between two bearing surfaces transmit wobble plate position to the blades. For the latter purpose, a crank attached to the blade spindle connected through a ball joint to the slipper provides the simplest actuator. If space or loading precludes the use of a crank, as determined by more detailed design study, a rack and pinion or sector and pinion may be substituted.

Rotating Wobble Plate System

A wobble plate mounted on the rotor is positioned through slipper bearings by hydraulic cylinders which are mounted on the hull. Blade actuator rods are attached to the wobble plate through ball joints and similarly to a rack which drives the blade pinions.

The most critical parts of both systems are push-rod bearing shoes, rack and pinion gear teeth, and blade spindle bearings. Bearings can readily be tested to establish their performance, hence this is mainly

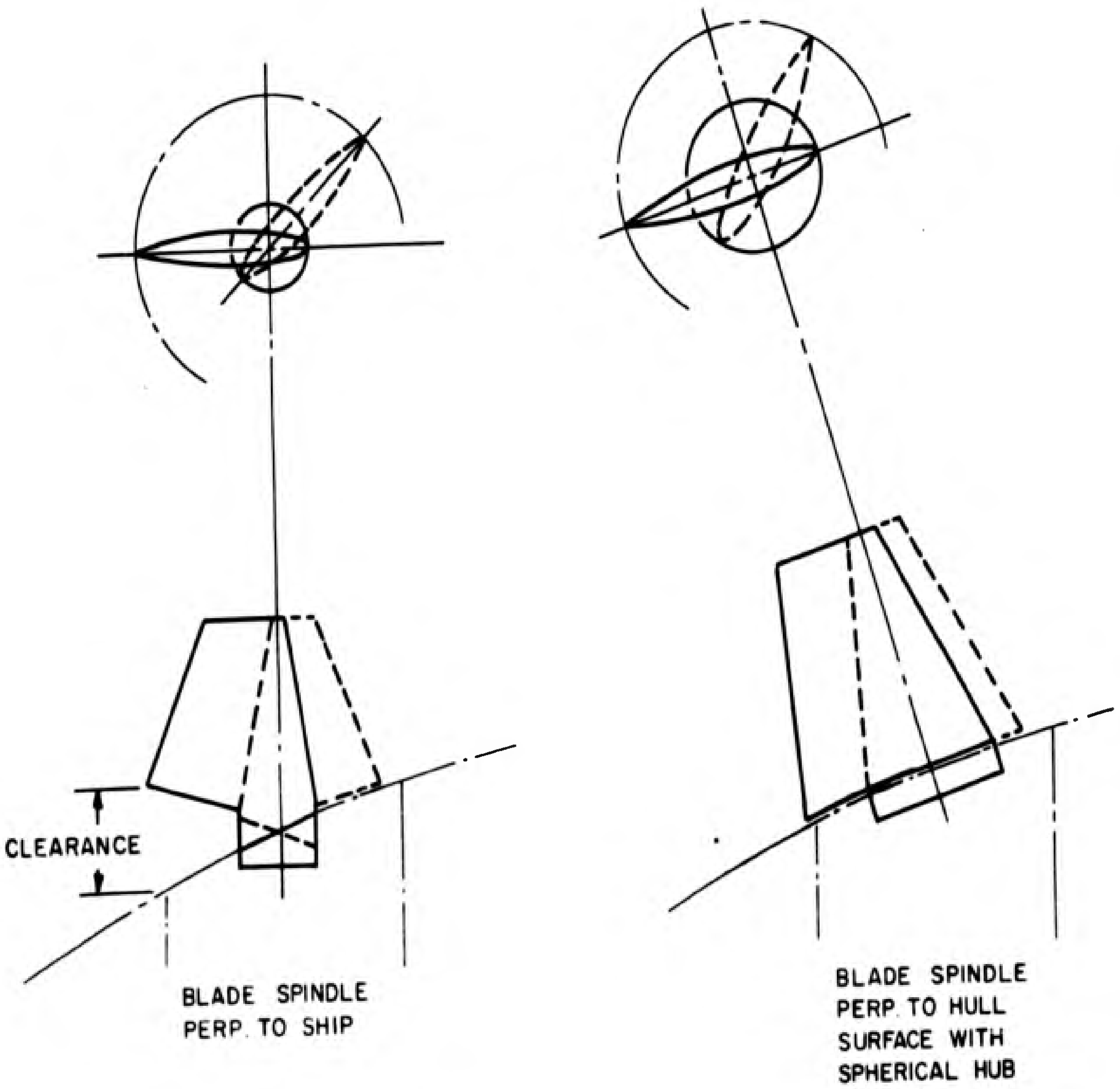


Figure 10. Propeller Blade Mounting

a question of development and laboratory verification. Gears exposed to sea water have been used in submarines with excellent results, but the conditions do not match those encountered here in speed or magnitude of load. Laboratory tests are needed to establish confidence in this area. The wobble plate positioning cylinders are similar in size and duty to rudder and diving plane actuators. Smaller cylinders outside the hull are used for mast hoists. Extension of this experience to large cylinders outside the hull requires careful engineering but does not raise doubt as to feasibility. Further design study may indicate, however, that the cylinders should be placed inside the hull. This would add complication, but would render the system more nearly similar to present practice.

In addition to the motor as a source of vibration and noise, the propeller actuating mechanism itself is a possible noise source. Rattling is a potential offender in pin joints, ball joints and gear meshes. Bearings may be noisy during transients in which loading is suddenly increased. Close clearances and preloading are expected to minimize these problems. Consideration of noise during preliminary design studies consisted mainly of effort to reduce the number of parts and the number of joints.

When mechanical systems are compared with other types of systems it is well to consider flexibility or adaptability to development changes. The kinematics of the mechanical systems are for all practical purposes identical and fixed. The blade pitch variation is approximately sinusoidal, and this cannot be changed in either system. Furthermore, the ratio of pitch change to plate movement is constant and cannot be changed without extensive modifications of the system. The external equipment is thus seen to be very inflexible. The only parts of the system which are amenable to modifications are the hydraulic system which operates the positioning cylinders and the control system which interprets the ship control signals in terms of hydraulic system requirements.

The reason for building flexibility into a system is to enable adjustments to be made during service such as changes in rate of movement, time lags, damping, or possibly more drastic modifications such as change in the order of the servo system, or type of feedback system. Except that the pattern of blade motion is confined to a sinusoid, the mechanical pitch control systems provide the required flexibility in the control and hydraulic systems.

Preliminary studies have served to emphasize the importance of reducing hydrodynamic drag of the rotating equipment. In the Novel Electric Propulsion Study this consideration alone dictated a reduction in propeller speed to 50 rpm. This factor was considered only qualitatively in this preliminary study, but is dealt with quantitatively in the section on Electrical Main Propulsion Machinery.

Accuracy of blade position will be affected by deflections of the members, and clearance in all of the joints and bearings in the system, including the main motor thrust bearing, as well as the hydraulic positioning system. In this respect the several mechanical systems are about equivalent, and are probably slightly less accurate than the electrical or electric-hydraulic systems. Further design development is needed to establish quantitative values of pitch error, but it is estimated that it will be possible to approach the $\pm 1°$ required by the specification.

In the foregoing is the reasoning which has led to the selection of systems for further study, together with some of the pitfalls anticipated. It was not desirable at that time to restrict the possibilities to one system, since certain factors such as hydrodynamic drag can have overriding significance and must be known with some certainty before any design can be accepted. For this reason, a series of design studies was undertaken, enabling more accurate engineering estimates of losses, pitch accuracy, noise, etc. to be made. These are included in a subsequent part of this section, Development and Conceptual Design of Mechanical System.

Mechanical System Model

In order to facilitate study and demonstration of the mechanical pitch control system, a working model about 12 inches in diameter was built. Transparent plastic is used for most of the structure to facilitate viewing of the mechanism during operation, Figure 11.

The demonstration model uses the principle of the rotating wobble ring. The hydraulic positioning cyclinders are simulated by a manually positioned plate thereby setting the position and tilt of the wobble ring. The four blades are operated by racks attached to the wobble ring, moving gears fixed to the blade spindles.

Rotation of the hub is provided by a 1/10 hp gear motor turning the propeller at 30 rpm.

Mechanical-Electric and Mechanical-Hydraulic Systems

Mixed systems can be devised in which power is transmitted to the hub electrically while blade pitch information is transmitted through a mechanical system. A wobble plate system can directly position a hydraulic servo valve, or a linear electrical transducer for each blade. These systems were not given primary emphasis since the elements being developed could be incorporated later into a combined system, and also the principal drawbacks of both type systems are retained: Relatively poor accuracy in mechanical positioning, potentially high water drag, and complexity of electrical or electrohydraulic equipment in the hub.

Electrical Systems

Figure 12 shows a system incorporating electric information transfer, electric power transfer, and electric pitch changing devices.

Both information and power are transferred by a set of four rotating transformers. There are sixteen rotor segments in each transformer, one for each blade. Four stator segments per transformer are shown for simplicity, yielding position control by quadrants, but any reasonable number can be used. Each group of four axially adjacent transformer stationary segments is excited with four synchronous signals, three of which are variable in magnitude.

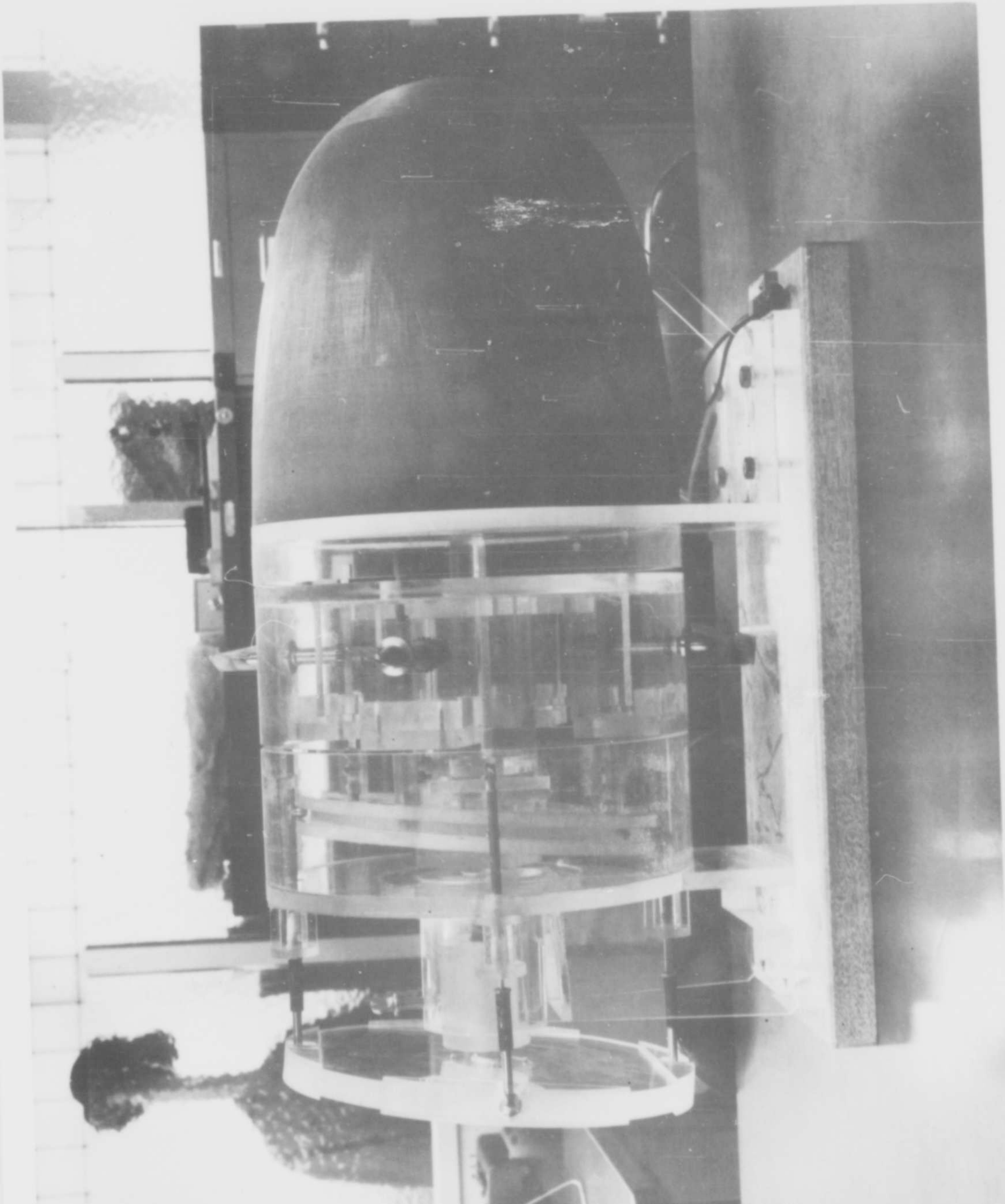


Figure 11. Mechanical Pitch Changing System Model

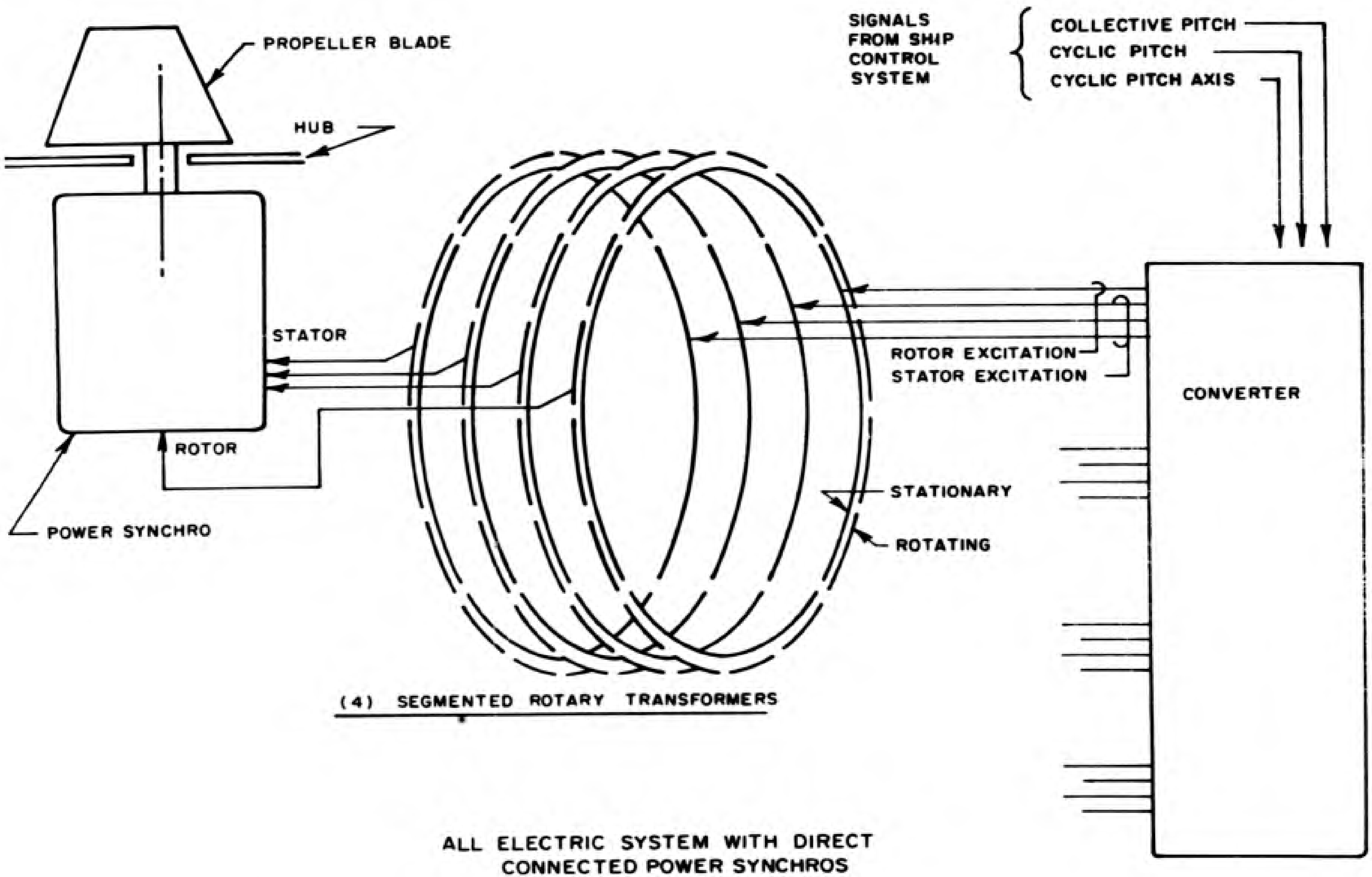


Figure 12. All-Electric System with Direct Connected Power Synchros

Each group of four axially adjacent transformer rotating segments excite the rotor and stator of a power synchro directly connected to one propeller blade. The three variable voltages establish the angular orientation of the magnetic field in the synchro. The single constant voltage energizes the rotor, which aligns itself with the stator field. Since the synchro operates over a range of only 140°, slip rings for the rotor are not required.

This is a simple system, without closed loop control, and with all equipment exposed to sea water being of substantial construction. A good degree of isolation between pitch changing devices is attained, so that failure of one device will not readily affect others, and upon electrical failure the affected blade is free to align itself with the stream and thus minimize interference. The system additionally lends itself to modular construction.

The synchros are large and heavy, but not impossibly so. However, to develop torque the stator mmf must be displaced from the rotor mmf, and thus the required positioning accuracy cannot be attained for a reasonably sized synchro. If it should be ultimately found that a force, rather than position, type of blade control would be satisfactory this scheme (or one using torque motors in a similar manner) would merit review.

Figure 13 shows another all-electric system. In this system the electric power and information are transferred to the rotor separately. Power is transferred by a simple rotary transformer, while information is transferred by another segmented rotary transformer, again yielding stepwise pitch control.

Each blade is positioned by a separate electric motor, driving through a pair of eddy current couplings. The motor runs continuously, and the couplings are geared to drive the blade in opposite directions. The entire package of motor, couplings and gearing is immersed in oil, encouraging modular type construction. A closed loop control system in the hub energizes the eddy current couplings as appropriate to position the blade.

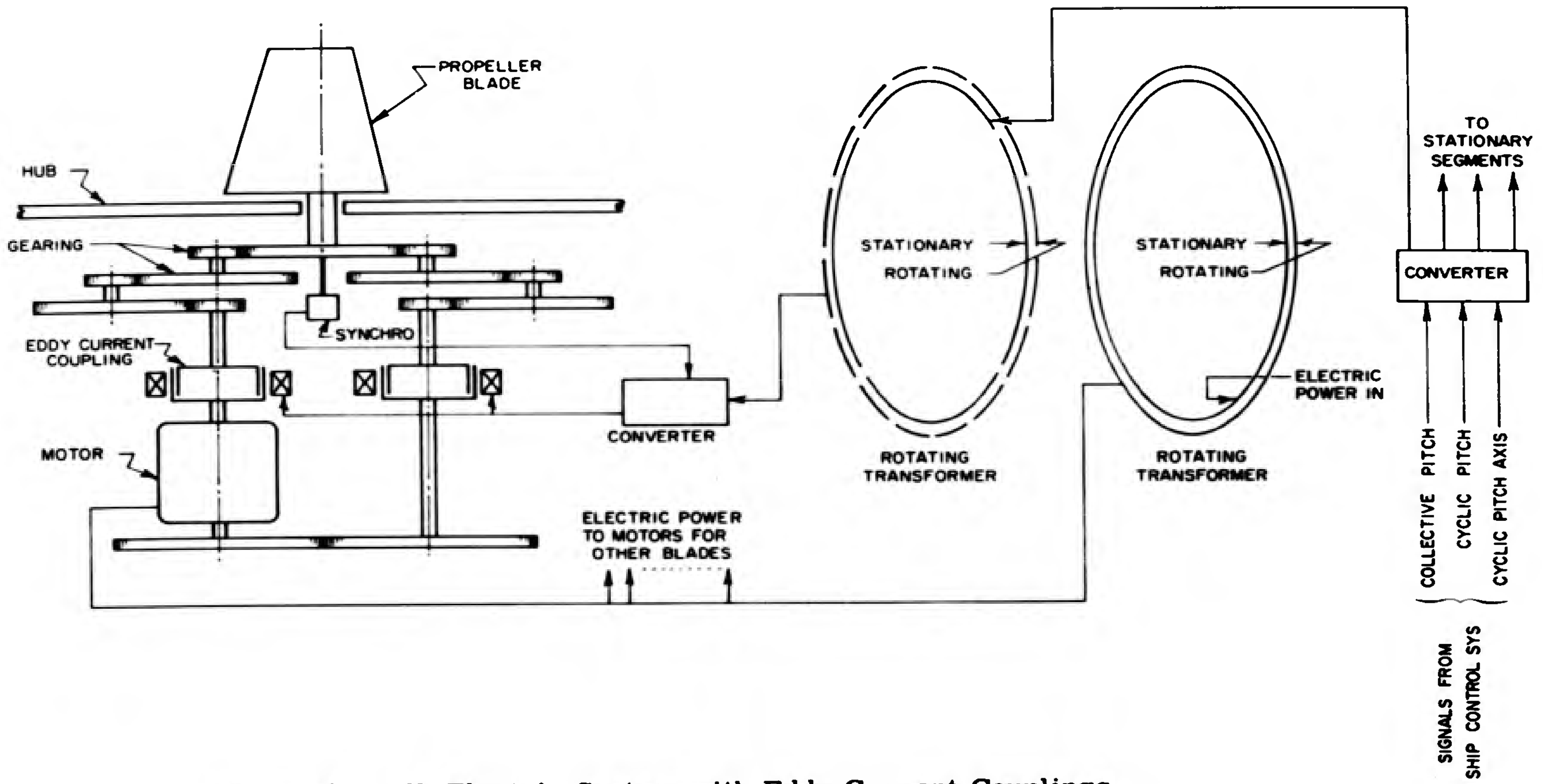


Figure 13. All-Electric System with Eddy Current Couplings

This system, like the previous one, provides a good degree of isolation between positioners for each blade, and allows the blade to align with the flow upon electrical failure. It requires the motors to run continuously, which is undesirable from a noise viewpoint and also results in continuous loss from viscous drag in the couplings. In addition, electronic control equipment must be located in the propeller hub. It is nevertheless not without merit, but was not pursued because better schemes were devised.

Figure 14 shows a third all-electric system. Electric power is transferred to the rotor by a single rotary transformer. Information is also transferred by another single rotary transformer.

The cyclic pitch variation is generated by a crank which rotates synchronously with the propeller hub. The magnitude is controlled by the length of the crank and the orientation is controlled by the phase angle between the crank and proepller hub positions. The collective pitch is controlled by the length of the link between the crank and blade spindle.

The system also involves continuously rotating motors in the hub, and is of such mechanical and electrical complexity that it was discarded.

The principal useful feature to come from these three all-electric systems is the application of segmented transformers for information transfer. In return for accepting stepwise cyclic pitch control, the difficult and complex task of continuously determining the instantaneous hub position and the corresponding position for each propeller blade is eliminated. The blade pitch angle is established directly by the location of the blade with respect to the hull, and is independent of time and of speed of hub rotation.

Electrical schemes are further discussed in Section VI of this report, which covers the General Electric Company work.

DEVELOPMENT AND CONCEPTUAL DESIGN OF MECHANICAL SYSTEM

Utilizing the basic considerations described in the Preliminary Survey

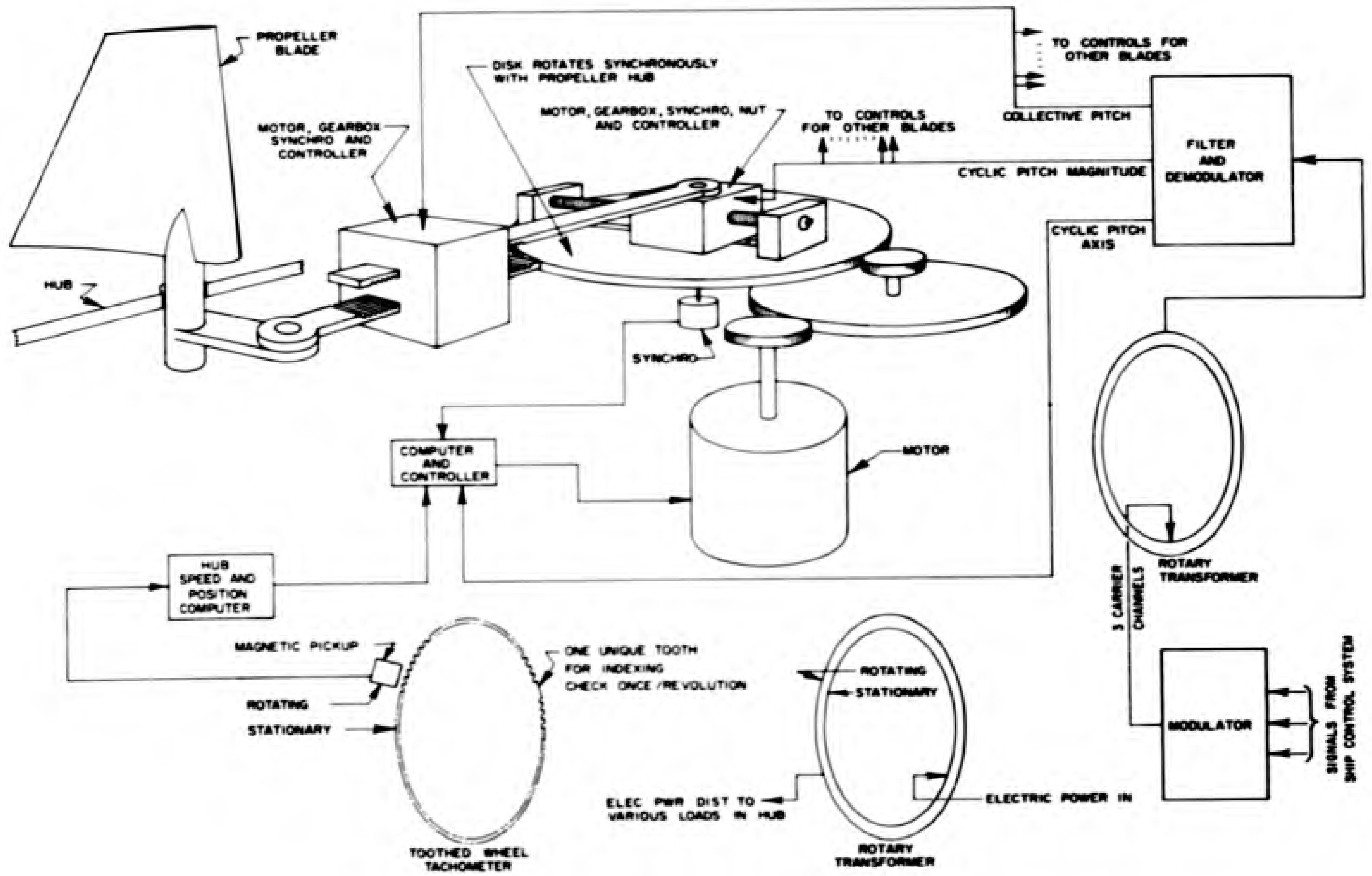


Figure 14. All-Electric System with Variable Length Crank

part of this section, preliminary design layouts were made to obtain engineering data as to feasibility, cost, fluid drag, and to realistically define the size, shape and other characteristics of a mechanical pitch changing system as it would be designed and installed on a submarine. As expected, each layout suggested certain favorable features and certain undesirable features of the particular concept under study. In turn, each study suggested a revision, so that finally five schemes were laid out and analyzed.

Rotating Wobble Plate, Rack and Pinion Drive, Figure 15 (page 55)

This system consists of a rotating wobble plate mounted to the rotor by a series of eight trunnions, which allow axial movement. Tilt of the wobble plate is provided by differential movement of four stationary hydraulic cylinders, which also cause the axial movement when positioned collectively. Blade movement is provided by transmission of wobble plate motion through individual connecting rods to racks and pinions, driving the blade spindles. Pin and ball joints are provided where necessary.

The drawing shows the pitch changing system built into the General Electric synchronous motor. This motor outline was used for convenience in the five studies discussed in this section, but the Elliott motor is also suitable for this application.

On closer study of the mechanics of this system, it is seen that allowance for vertical movement of the various support points of the wobble plate as it is tilted is not provided. This movement is proportional to the cosine of the angle of the wobble plate with respect to the ship's centerline. At the maximum angular tilt of about 2°, required to obtain ± 20° cyclic pitch variation, this vertical movement is only about 0.020". It is felt this is within machining or manufacturing tolerances for a structure of this size. However, rubber bushings could be provided in the trunnion axles to relieve this potential binding condition. After the degree of wobble plate tilt was recognized, the need for the large trunnion type mounting was felt to be unnecessary. The design would be greatly simplified by the use of sliding guides with elastomer connections to the wobble plate to permit the required motions.

A further unsatisfactory feature of this design is the large slider bearing required to take the forces of four blades. This results in poor bearing efficiency and resultant high bearing drag losses. The high bearing speed caused by location near the outer hull diameter also contributes to this factor. Rotor fairing is complicated and unbalance is possible.

The rotating wobble plate has an interesting feature not possessed by nonrotating wobble plate systems in that it can be assembled as a unit along with the main motor rotor, thereby allowing machining and line-up to be accomplished on the shop floor.

However, the direction of the present design study has tended toward the stationary wobble plate. There are several reasons for this course:

(1) Using the stationary ring, it is possible to simply mount a minimum diameter wobble plate on the sturdy surface of the inner hull. This results in slipper bearing speed consistent with main motor bearing speed.

(2) It allows the positioning hydraulic cylinder to be arranged through the motor stator structure, reducing overall volume requirements.

(3) Since the stationary wobble plate requires a slipper bearing for each blade, it first appears to be at a disadvantage as compared to the rotating wobble plate which requires a fewer number, although larger bearings at each positioning cylinder. However the larger bearings are taking the load of four blades at a single point, which requires an arrangement of a number of bearing pads in order to maintain desired design bearing loadings.

<u>Stationary Wobble Plate, Rack and Pinion Drive</u>, Figure 16 (page 57)

This system consists of a hull mounted wobble plate, moving axially over guides which also prevent it from rotating. Movement is effected by hull-mounted hydraulic cylinders as described previously. Wobble plate tilting is allowed by clearance in the hull guides.

Blade motion is transmitted through individual slider bearings connected through a rack and pinion to the blade spindle. This system, while fairly simple, suffers from high bearings speeds, bulkiness, and high appendage drag due to the sixteen large protruding racks. An effort was made to reduce the diameter by building the mechanism into the motor rotor structure. This alleviated conditions somewhat; however, the system is considered to be unfavorable.

Stationary Wobble Plate, Lever Arm Drive; Figure 17 (page 59)

This system incorporated several features to eliminate items shown to be undesirable in the prior studies. It was decided to mount the wobble plate on a bearing surface on the minimum hull diameter, in order to reduce slider bearing speeds to that of the main motor bearings. This places the bearing question in the same perspective for the mechanical pitch changer as for the main motor.

The pitch changer bearings differ from the motor bearings in two significant respects, however. In order to provide accurate pitch control the bearings on both sides of each slider have a minimum of clearance. Sand or grit trapped in these bearings would cause severe abrasions. The main motor bearings will have clearance appropriate to the diameter of about 13 feet, allowing room for solid particles to roll through.

Another major simplification integrated in this concept is the use of slides with rubber mounts to support the wobble plate, instead of the previous trunnion system. These mounts will allow the angular tilt of the wobble plate necessary for $\pm$ 20° blade cyclic pitch change and also serve as sound and shock mounts.

Blade movement is provided by a crank fixed to the blade spindle having a suitable slider bearing following the wobble plate. The lever arm drive for the blade limits the degree of rotation to less than 140° with poor efficiency at the extreme positions and is not capable of providing the large range of blade movement required for steering and diving use of the blades of a stopped rotor. However, blade accuracy in the normal operating zone is greater than that obtained by a rack and pinion drive. This type of actuator is not adaptable to placing the blade axis perpendicular to the hull surface.

Stationary Wobble Plate, Rack and Pinion Drive, Figure 18 (page 61)

This scheme is essentially an extension of the previous one, using a rack and pinion drive to allow sufficient blade movement (0 - 160°) for steering and diving use of blades on a stopped rotor. Unfortunately the use of the rack driven by a rod attached perpendicularly may if not carefully proportioned, be subject to jamming or binding. This is dependent on the ratio of the length of the vertical arm to the bearing span. The positioning hydraulic cylinder is mounted in the motor stator structure, and the pitch changing system is inserted between the motor and the transformer to reduce volume requirements. The pitch changer is also designed into a package unit for each blade for ease of removal and installation. To remove a unit suitable plates and bolts are removed and the entire package, blade mechanisms, arm, and bearing can be lifted out as a unit.

In this study a mechanical link is shown as an option which is attached to the wobble plate and goes through a gland into the pressure hull. Alternatively this arm could be the primary means of positioning the wobble plate, or it could be included as a secondary means of positioning in case of failure of the outboard system. This closely follows existing submarine practice in providing backup features on ship control systems. It also gives a direct mechanical blade positioning indicator.

Stationary Wobble Plate, Sector and Bevel Gear Drive, Figure 5 (page 27)

This scheme represents Electric Boat Division's recommendation as the most favorable mechanical pitch changing device evolved in the current study period. It eliminates the propensity toward binding of the rack as in Figure 18 and allows the use of simple lever operated sector and bevel gear for blade rotation from 0 - 160°. This range could be extended by extending the sector. It is extremely simple and has a minimum of parts, bearings, and joints. This unit is more fully described on page 25.

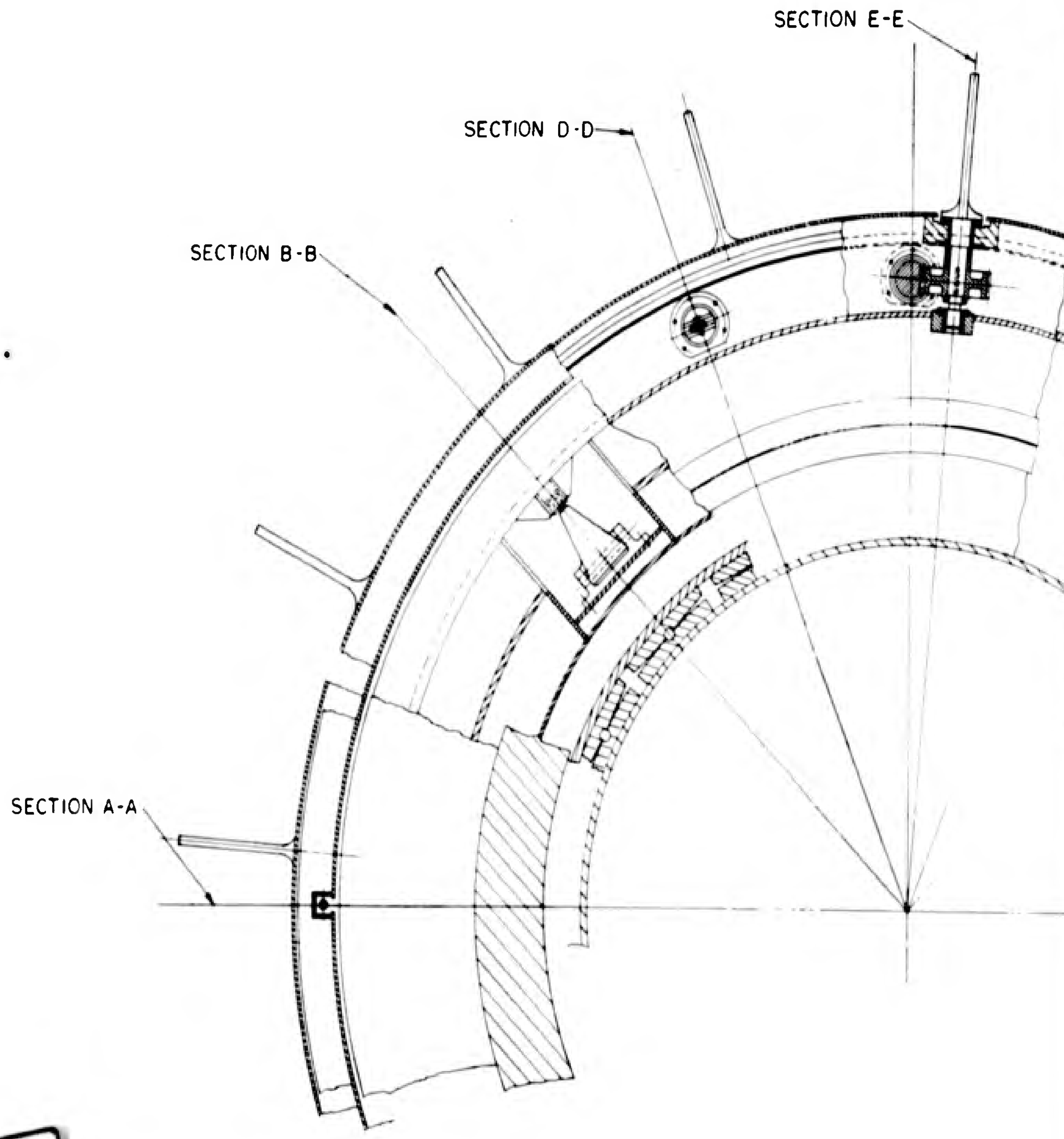
In order to compare the various mechanical concepts, various estimates are tabulated as shown in Table I. The weight of the five mechanical concepts vary between 30,000 and 40,000 lbs. The average weight of rotating parts in the four stationary wobble plate concepts is about 16,000 lbs. Figure 15, using a rotating wobble plate has 26,000 lbs. of rotating weight.

The enclosed volume of the five mechanical pitch changing schemes averages 1900 ft^3. A preliminary cost estimate for the selected pitch changing mechanism resulted in a figure of about $200,000 per propeller for outboard equipment.

The other factors, such as number of bearings, limits of movement, bearing velocity, and friction drag are tabulated in Table I.

Table I - Comparison of Mechanical Systems

	Blade Angular Movement	Number of Slider Bearings	Bearing Speed fps	Pitch Changer Bearing Drag hp	Appendage Drag hp	Surface Drag hp	Motor Bearing Drag hp	Total Drag hp
Figure 15 Rotating Wobble Plate, Rack and Pinion Drive	0-140°	4	57.5	244	142	710	200	1296
Figure 16 Stationary Wobble Plate, Rack and Pinion Drive	0-140°	16	59.0	219	594	600	200	1613
Figure 17 Stationary Wobble Plate, Lever Arm Drive	0-140°	16	41.8	137	15	600	200	952
Figure 18 Stationary Wobble Plate, Rack and Pinion Drive	0-160°	16	41.4	137	77	650	200	1064
Figure 5 Stationary Wobble Plate, Sector and Bevel Gear Drive	0-160°	16	41.4	137	77	660	200	1074



1

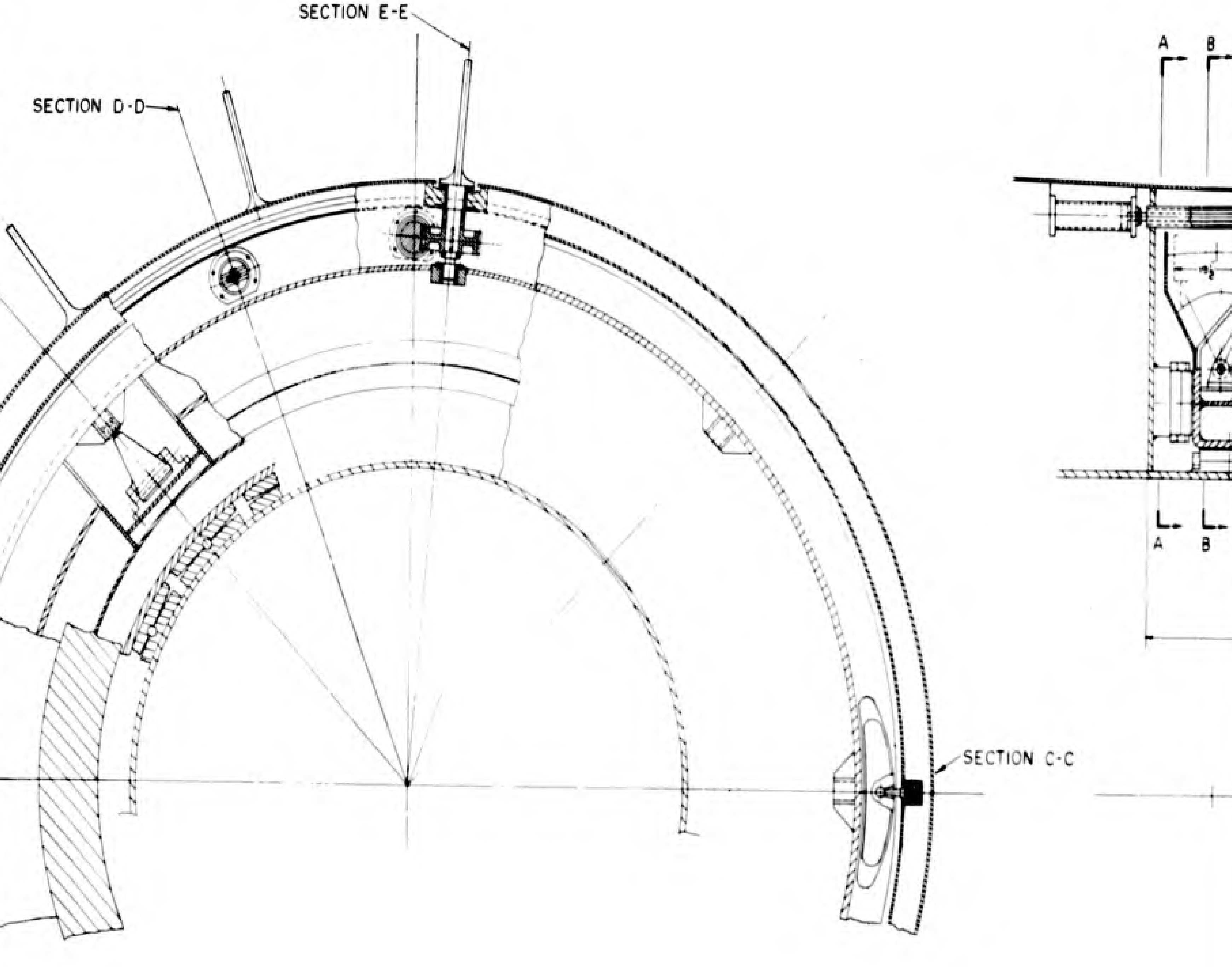


Figure 15.

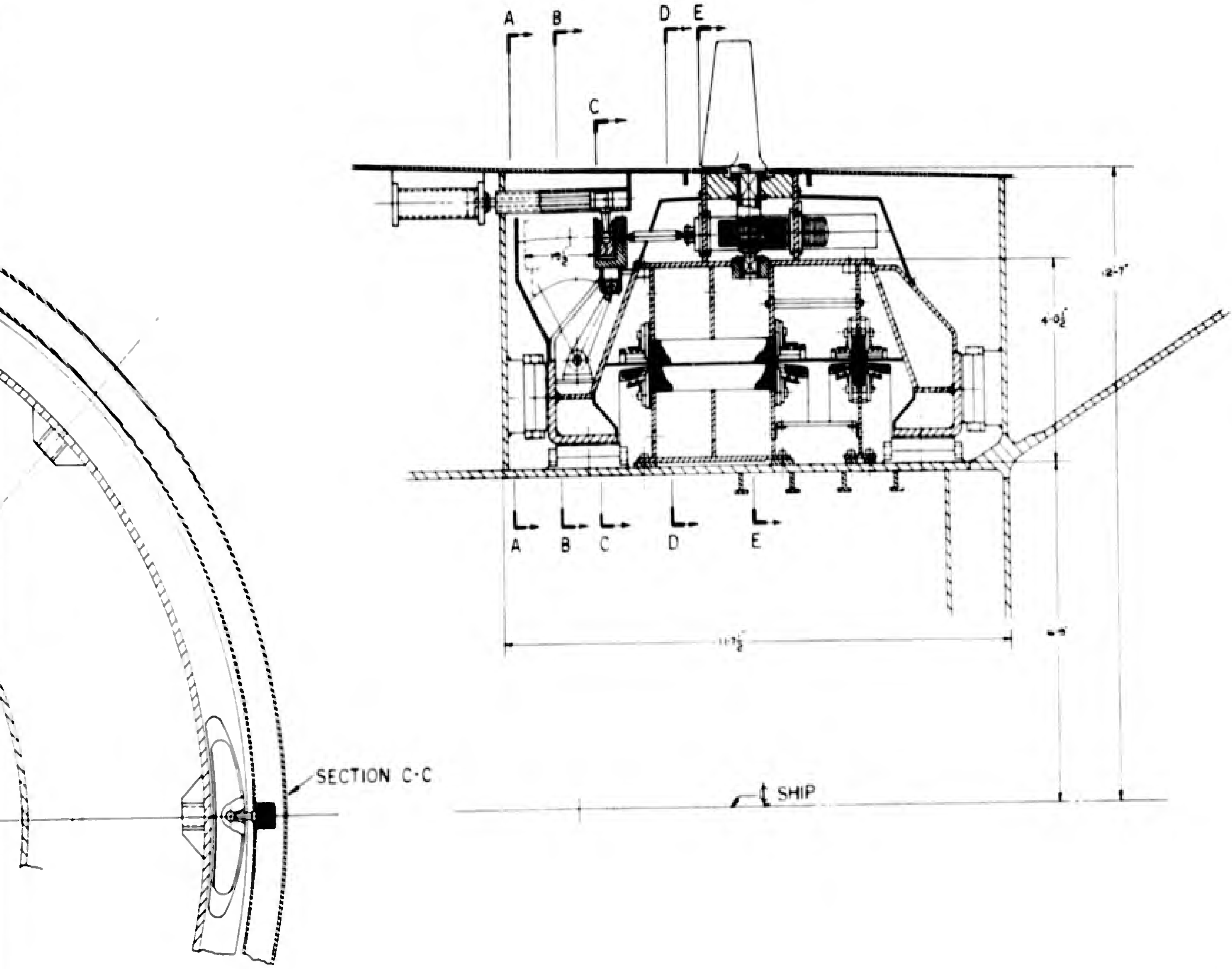
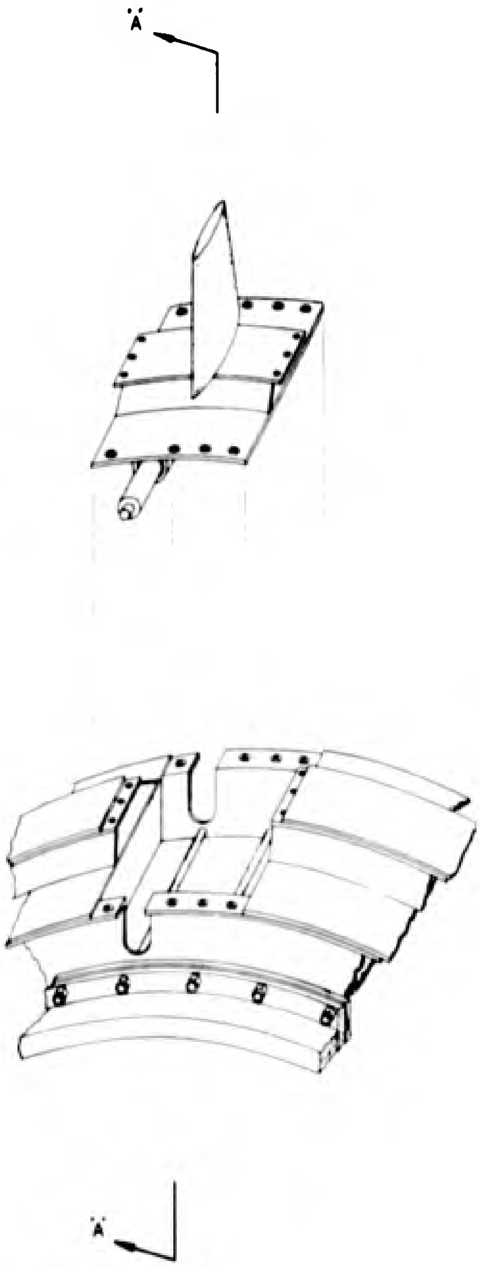
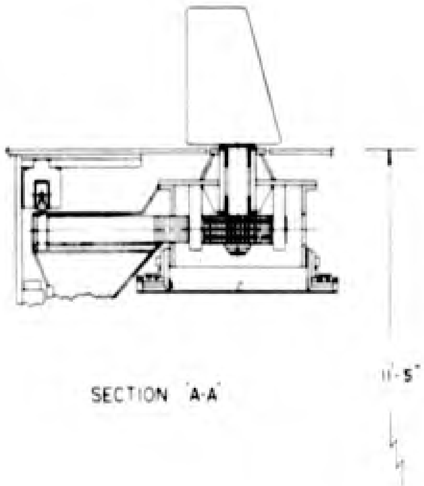
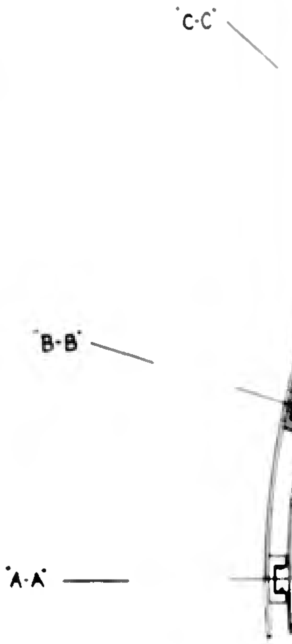


Figure 15. Rotating Wobble Plate, Rack and Pinion Drive

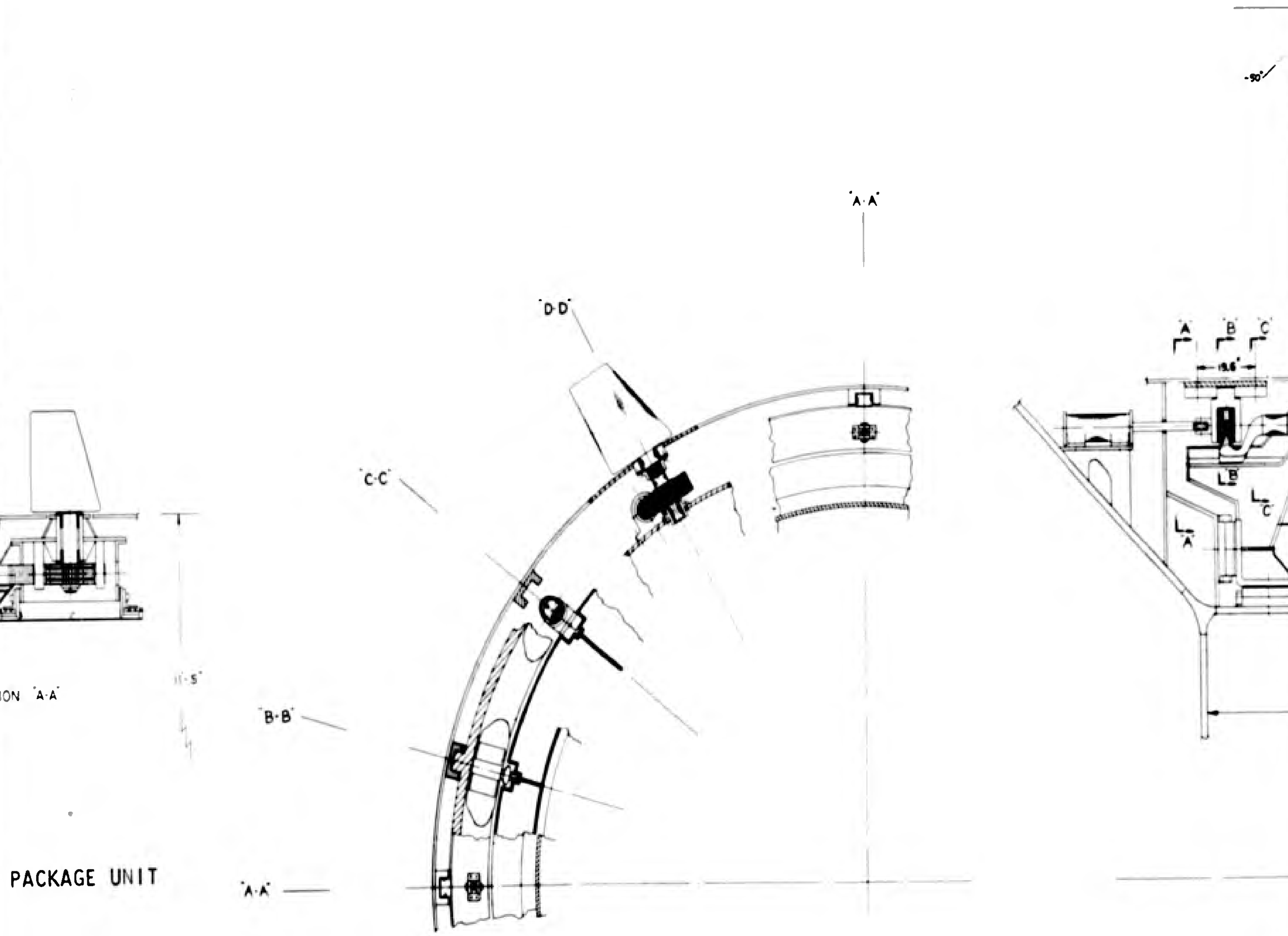






ALTERNATIVE PACKAGE UNIT

1



2

Figure 16. Stationary Wobbl

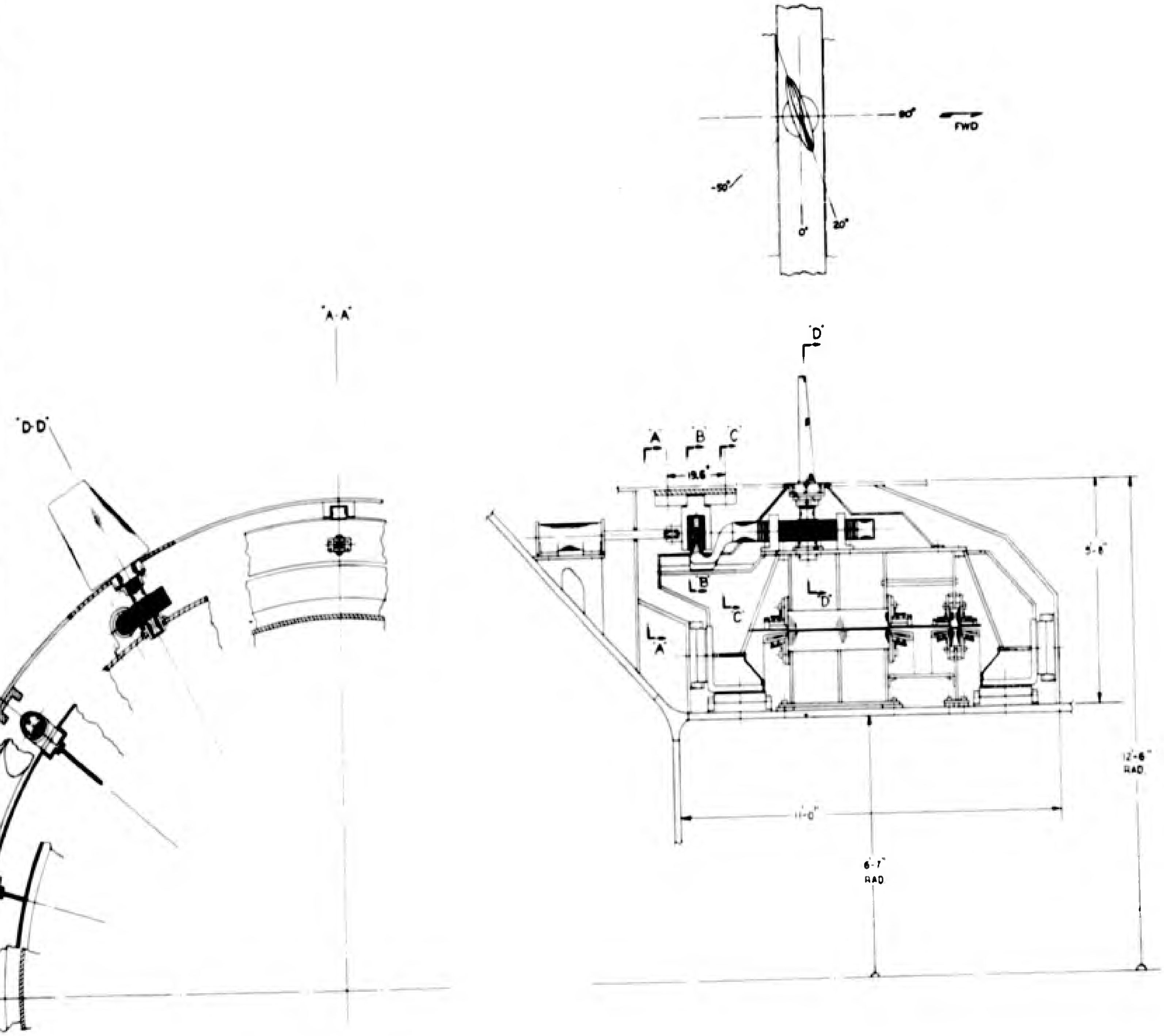
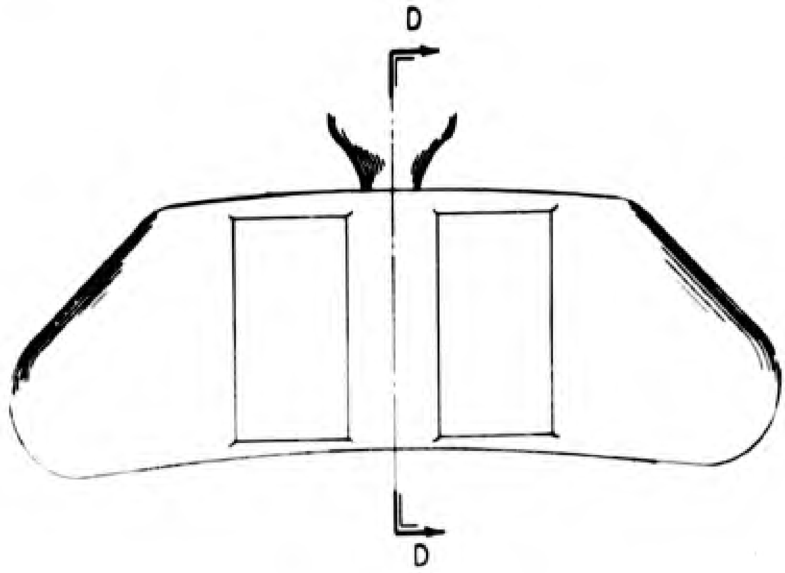
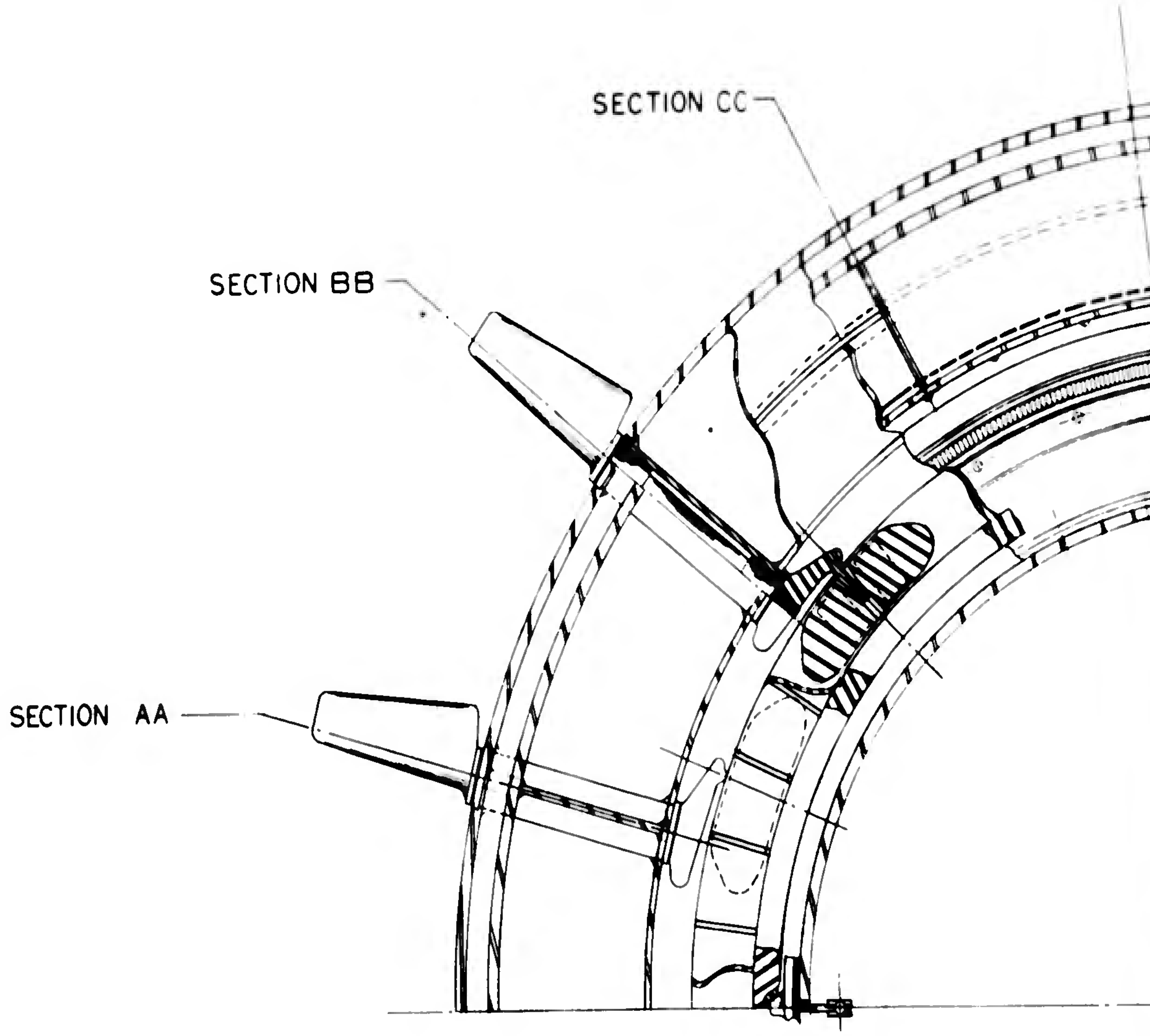


Figure 16. Stationary Wobble Plate, Rack and Pinion Drive

1





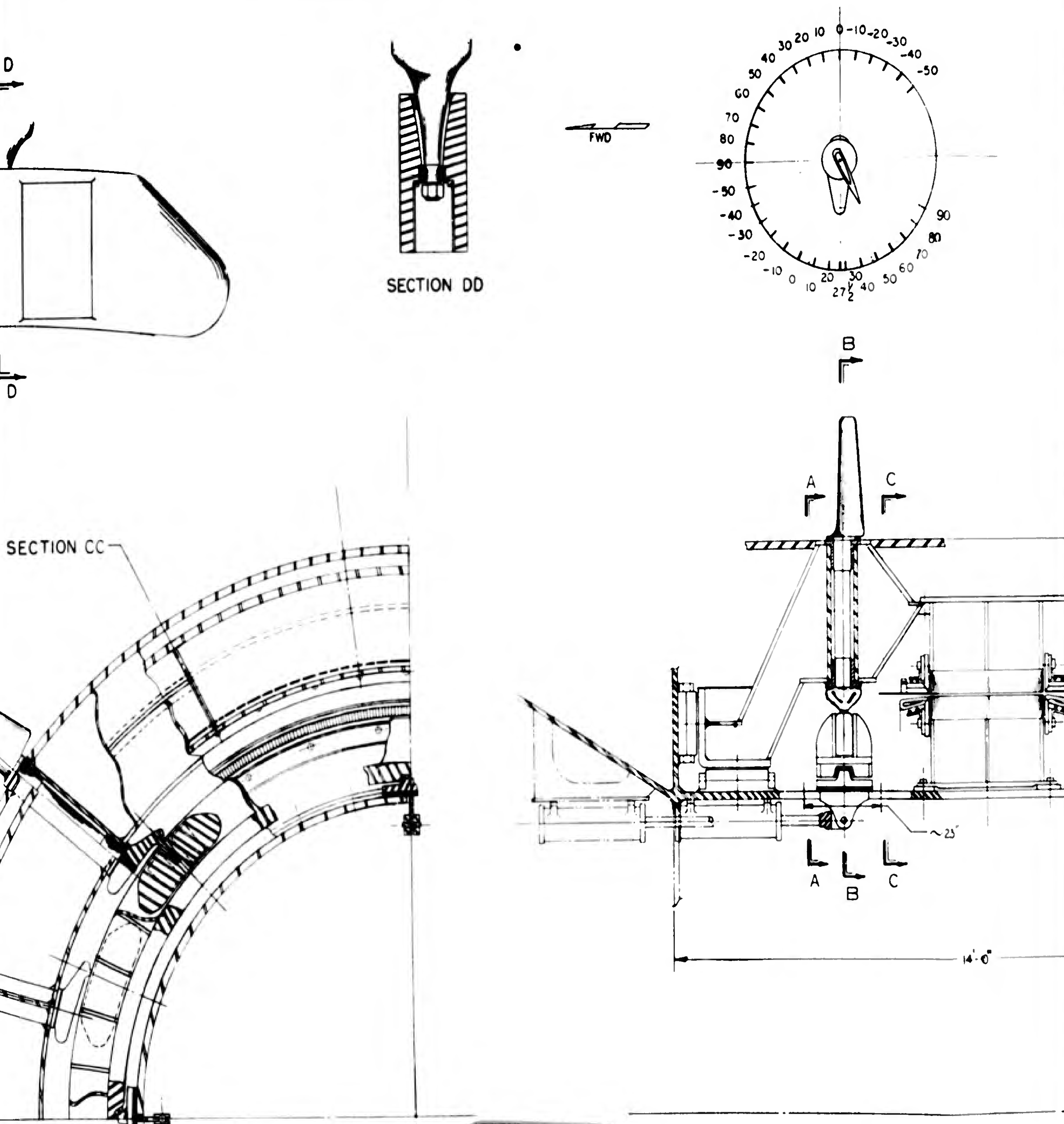


Figure 17. Stationary W

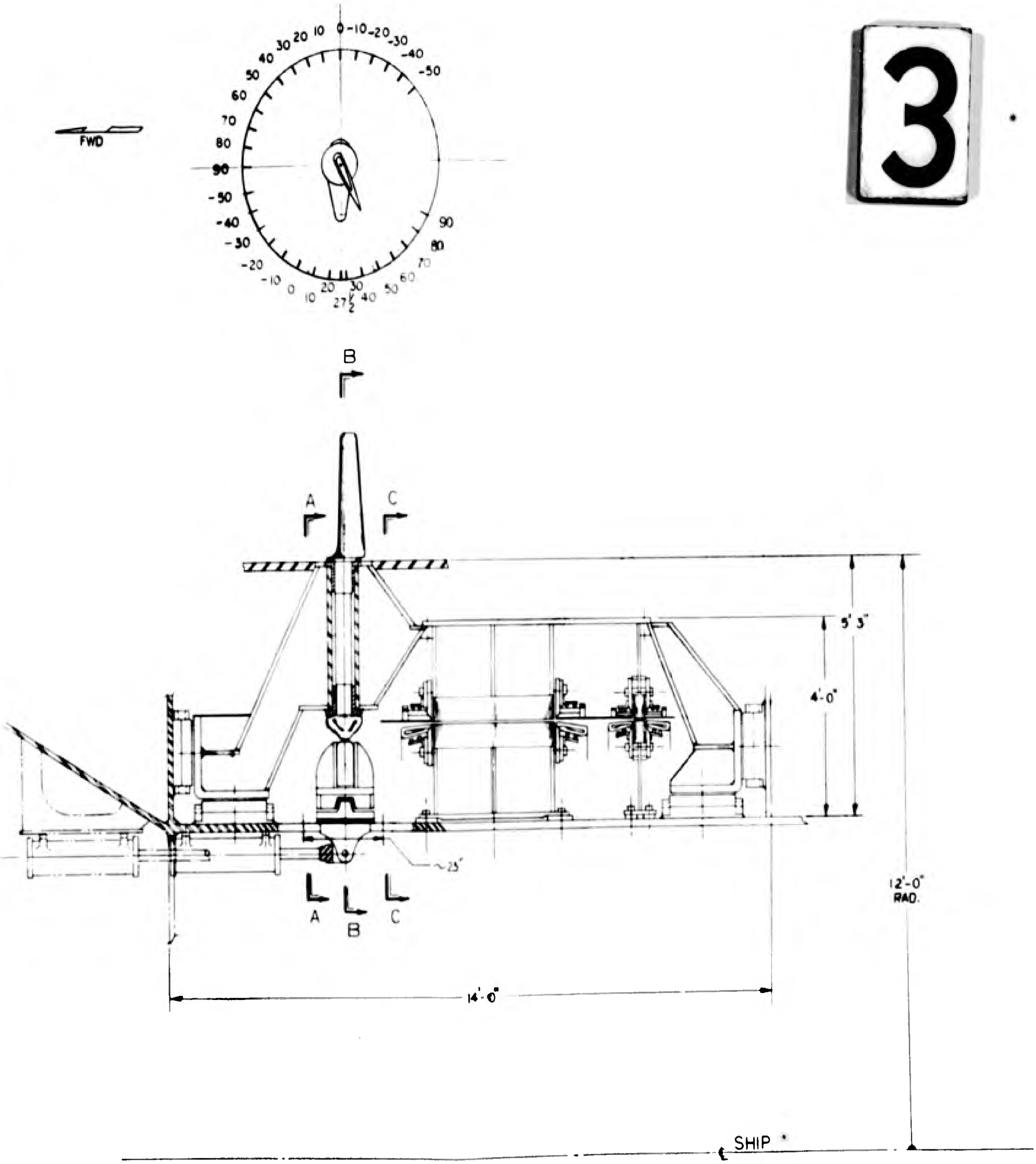
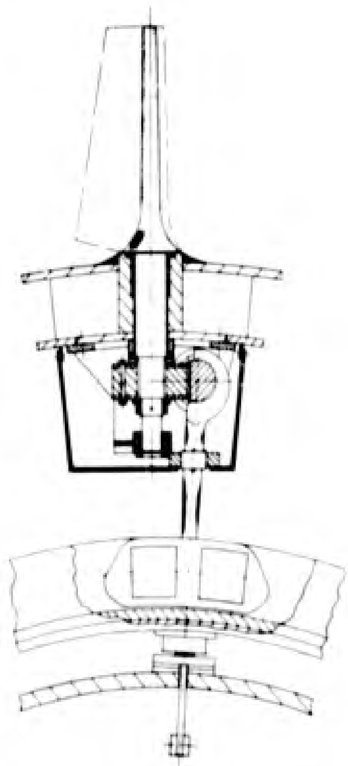
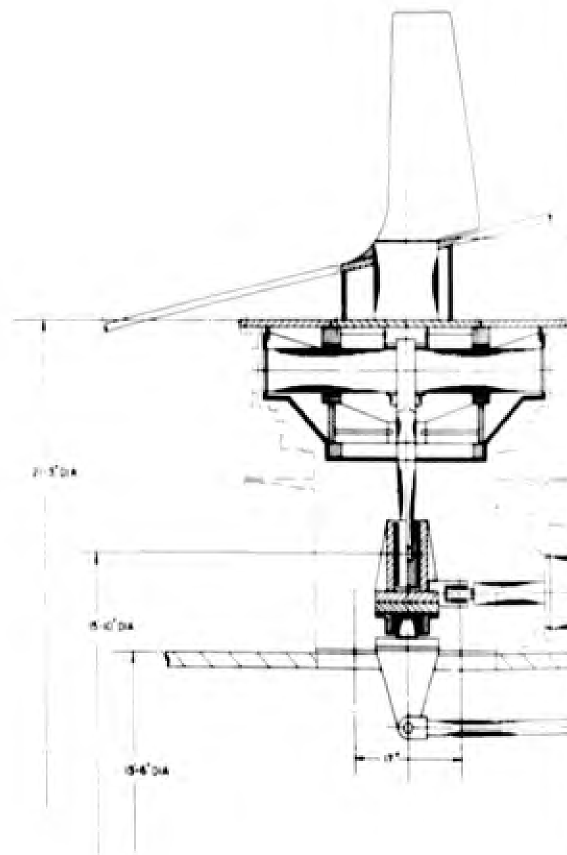
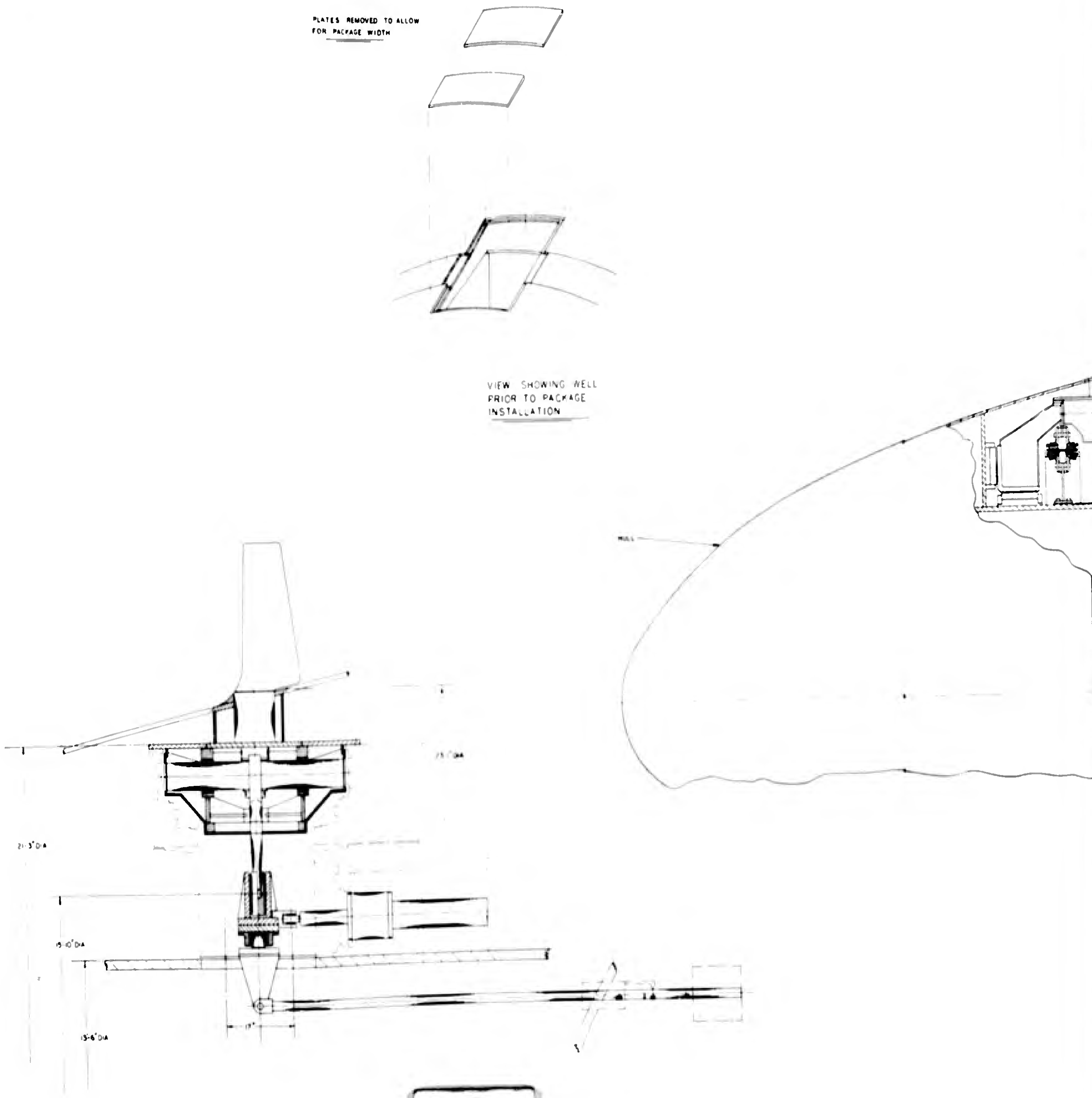


Figure 17. Stationary Wobble Plate, Lever Arm Drive





1



2

Figure 18. Stationary

3

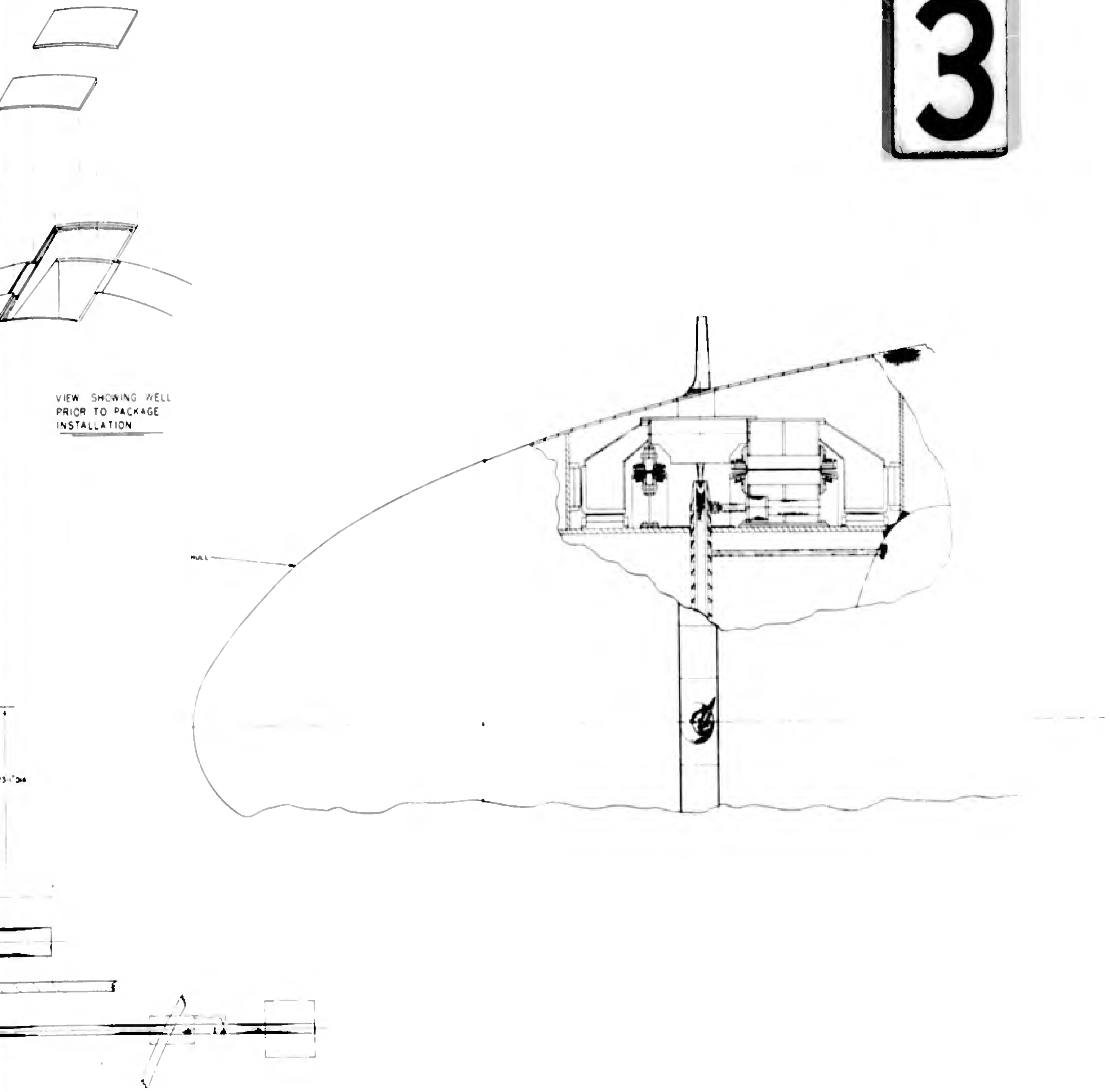


Figure 18. Stationary Wobble Plate, Rack and Pinion Drive

VI

ELECTRICAL PITCH CHANGING SYSTEM

During 1961, the General Electric Company participated, under subcontract to Electric Boat, in the feasibility study of a Novel Electric Propulsion System, termed NEPS. This study involved the use of two wrap-around inside-out induction motors, operating totally immersed in sea water and driving sets of fixed pitch contra-rotating propeller blades. The feasibility of such a system was adequately demonstrated by the NEPS study.

It was natural, then, for the General Electric Company to participate in the extension of this work, incorporating the Tandem Propeller concept. Many of the conclusions of the NEPS study could be incorporated in the present study without modification, and the background work was invaluable where extension and modification were required. Many of the same personnel contributed to both studies, representing the following General Electric departments:

Large Motor and Generator Department
Medium Steam Turbine, Generator, and Gear Department
General Engineering Laboratory
Materials and Processes Laboratory

In addition, a number of other departments were consulted in areas of specific technology or product experience.

The initial phase of the Tandem Propeller System study involved a brainstorming effort to investigate various ways and means of controlling the pitch of the individual propeller blades. This resulted in an informal interim report in December, 1961, outlining many ways in which such control could be achieved. On the basis of this report, in agreement with Electric Boat, it was decided that General Electric would concentrate further effort on the development of an all-electrical system of propeller drive and blade control.

The result is the system described in the following parts of this section. In all respects, from the standpoint of the propeller drive and blade control portions of the overall design, the conclusion reached is that the system is feasible. Additional work is required to optimize the design and to establish manufacturing processes; however, no major road-blocks are anticipated.

All reported data represent calculated values and not guaranteed values.

MAIN PROPULSION MACHINERY

The NEPS study was used as a foundation from which to begin the investigation of the Tandem Propeller Propulsion System. This earlier system was based on the use of an induction motor, because of the special requirement of operating in sea water and the anticipated problem of transferring power to the rotor if a synchronous machine were used. However, study indicated that there would be advantage both in weight and space saving if a synchronous machine could be used, and electric power on the rotor was required in any event for some of the pitch changing schemes. The feasibility of such a machine has been established by the previous study and the continuation of this work under the present contract.

Description of System

The system now considered, and illustrated in simplified form in Figure 19, utilizes 50 rpm synchronous motors, and maneuvering is primarily accomplished by control of the propeller blades. This method of control eliminates non-synchronous operation of the propulsion motors (except, of course, initial starting), and results in faster maneuvering since propulsion power is not lost in rapidly accelerating or decelerating the high inertia of the turbine generator sets and wrap-around propulsion motors.

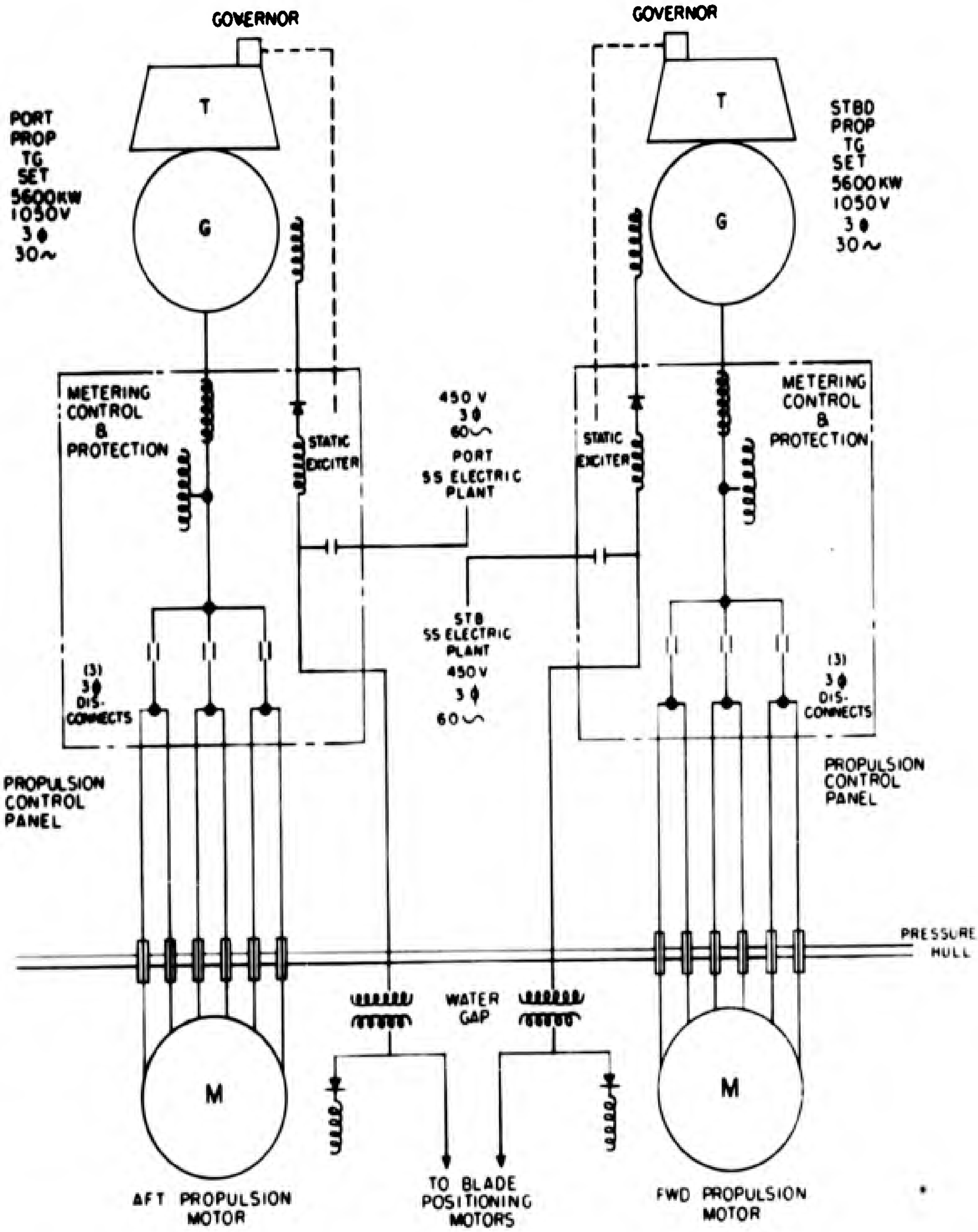


Figure 19. GENERAL ELECTRIC Electric Propulsion System One-line Diagram

With the exception of the special propulsion motors and blade control system, the propulsion plant is a conventional AC synchronous propulsion system. The AC propulsion power is obtained from steam turbine generator sets rated 5600 kw, 1800 rpm, 1050 volts, 1.0 power factor. The speed of the turbine generator sets and synchronous motors can be varied from 20% to 100% (6 to 30 cycles) by controlling the speed of the turbines through the speed governors. This speed control is a function of the ship control system and is coordinated with the blade control. Generators and motors always rotate in one direction, and do not reverse.

The propulsion motors are started and synchronized at 33% of rated frequency (10 cycles). During starting, the blade position and motor and generator excitation and speed are controlled locally. Once the motors have been started and synchronized, control is transferred to the ship control system, and from this point on all maneuvering is accomplished by a combination of blade control and motor speed.

Power for exciting the fields of the synchronous motors and generators is obtained from the 450 volt ship's service system.

System Performance

The system is fully automatic and requires no adjustments once the propulsion motors are started and synchronized. From this point on the motor speeds and blade positions are determined by the ship control system. Electric power can be supplied or absorbed by the turbine generator sets as conditions dictate. Under certain conditions, the turbines can produce torque in excess of the synchronous motor torque. This could cause the motor to pull out of step. However, although a maneuvering transient analysis was not made, it is not expected that these conditions can arise in normal operation. If they do arise, this problem can be solved easily by the addition of an automatic steam control on the turbines to limit the turbine torque to the maximum pull out torque of the motor and generator. The system should respond faster and permit more rapid maneuvers than a conventional propulsion system, since energy is not lost in rapidly overcoming the inertia of the motors and the turbine generator sets.

Description of Major Components

The propulsion machinery is sized using assumed steam conditions and flow.

Turbine Generator Sets

These machines are of conventional design, and no development is necessary. Additional work is required to assure a minimum size and low noise system. However, this work is of a standard nature, and no detailed effort has been extended in this direction under the present study.

Each machine is rated:

Steam pressure	285 psig (625 psig no load)
Steam temperature	417°F (493°F no load)
Steam flow	90,000 lb/hr.
Condenser pressure	7 in Hg
Water rate	12.2 lbs/turbine shp hr. maximum
Speed	1800 rpm
Voltage	1050 volts
Frequency	30 cps
Phases	3
Power	5600 kw
Power factor	1.0
Voltamperes	5600 kva
Efficiency	98.0%
Insulation class	B
Enclosure	Total, with air to water HX
Length	375 inches
Weight	178,500 lbs.

An outline of one of the turbine generator sets appears in Figure 20.

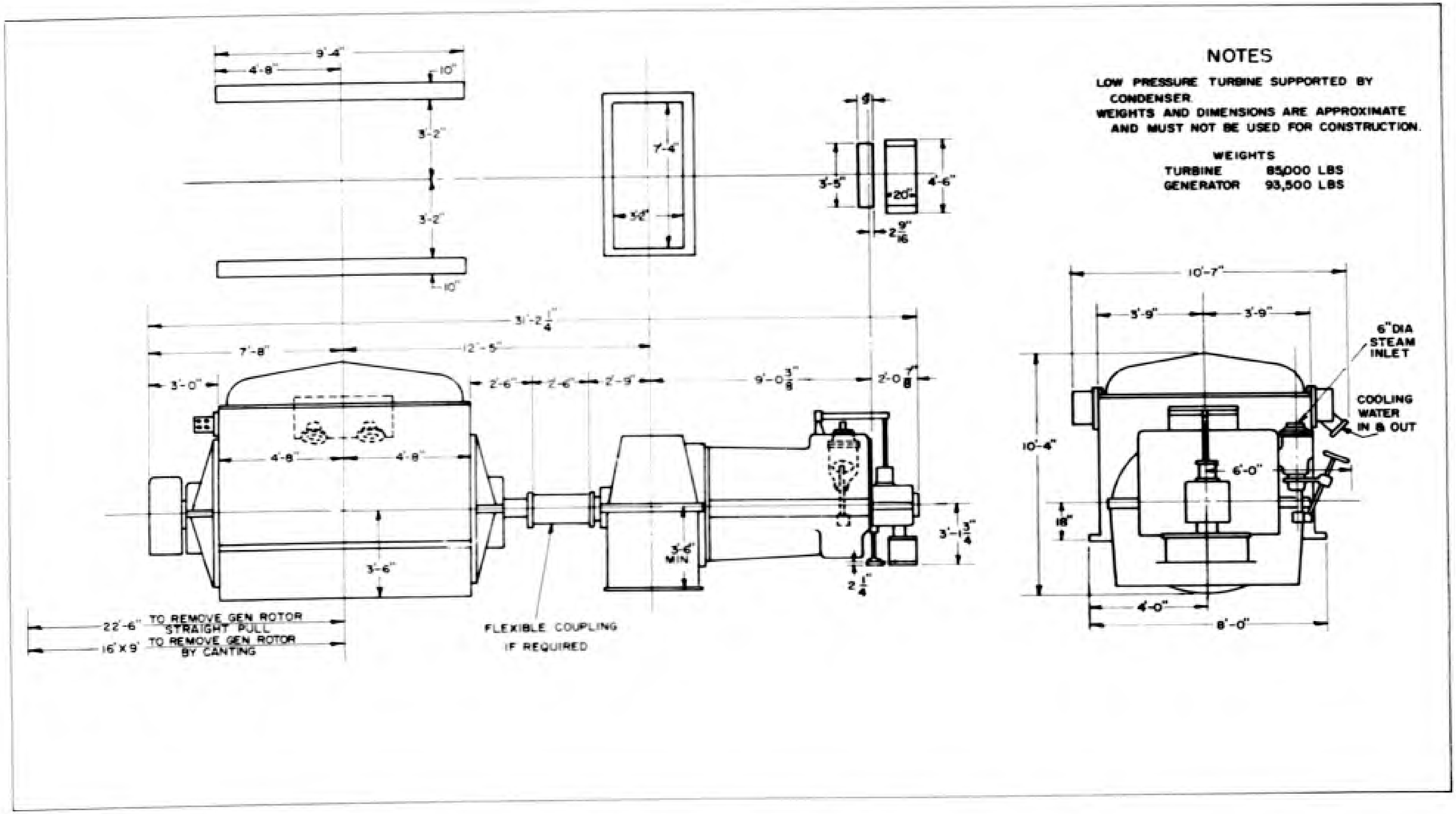


Figure 20. GENERAL ELECTRIC Turbine-Generator Outline

Propulsion Control Panels

Two propulsion control panels are required (one for each propeller) to house the components necessary for the starting, synchronizing, and protection of the propulsion motor and generator. The excitation supplied to the generators is automatically controlled such that the voltage is maintained proportional to speed. The propulsion motor is permanently connected to the generator through three sets of disconnects. These manual disconnects are used only during maintenance and in the event of a motor casualty requiring that a portion of the motor armature winding be disconnected.

Each propulsion control panel is approximately 78 inches wide, 60 inches high, 40 inches deep, and weighs approximately 3500 lbs.

The propulsion bus voltage is regulated by a static exciter with power supplied from the ship's 450 volt system. This static exciter maintains the proper bus voltage over the complete speed range and assures that maximum pull out torque is always available. The static exciter is a combination of transformers and silicon controlled rectifiers. These static components minimize noise and maximize reliability. The exciter for each generator is rated 25 kw continuously, with a ceiling output of 175 kw. This ceiling output is required only during starting of the propulsion motor and is not required during normal operation of the propulsion plant.

Fault protection, consisting of ground and phase unbalance detection, is provided by relaying which operates to remove generator and motor field excitation. Since the propulsion system is normally ungrounded, emergency operation is permissible with one ground on the system.

For maintenance and casualty purposes, disconnects for the motor are provided. These disconnects can be changed only with the system deenergized. The six stator circuits for each motor are paralleled in the propulsion control panel, thereby permitting continued operation at reduced power with a portion of the propulsion motor disconnected.

The propulsion control panels contain local metering for maintenance and log keeping but are not manned stations. A local-remote transfer is provided to permit starting of the propulsion motor under local control and transfer of control of the blades and motor speed once the motor is started and synchronized. All of the equipment required is standard hardware, and no major development is required.

Propulsion Motors

Figure 21 (also Figure 3, page 9) shows a cross-sectional view of one of the propulsion motors, together with its associated rotating transformer. Contrary to usual practice, the inner member of the motor is the stationary part, and the outer member rotates. The motor is basically a smooth rotor (as opposed to a salient-pole) synchronous machine, with a polyphase winding on the stator and a direct current winding on the rotor. It is of the wet winding type, with windings and iron suitably insulated and coated but otherwise exposed to the sea.

Either an induction motor or a synchronous motor could be selected for this application. An induction motor was used for the NEPS study, which preceded this one. This type of motor has the advantage of simplicity of construction, particularly on the rotor. The rotor winding is a simple squirrel cage type, and does not require insulation. In addition, an induction motor does not require transfer of electrical power to the rotating member, except through the normal medium of the air gap flux.

In the Tandem Propeller design, in order to cyclicly vary the pitch of the propeller blades, some form of power on the rotor is required. Conceptually, this may be mechanical, hydraulic, or electrical. Of these three, the simplest and most flexible mechanism for power transfer to the rotor is electromagnetic in nature, which results in electrical power availability on the rotating structure. This immediately suggests the possibility of a synchronous motor, rather than an induction motor. A synchronous motor has a number of distinct advantages:

Improved power factor. The large air gap required for mechanical reasons dictates that an induction motor will have a relatively

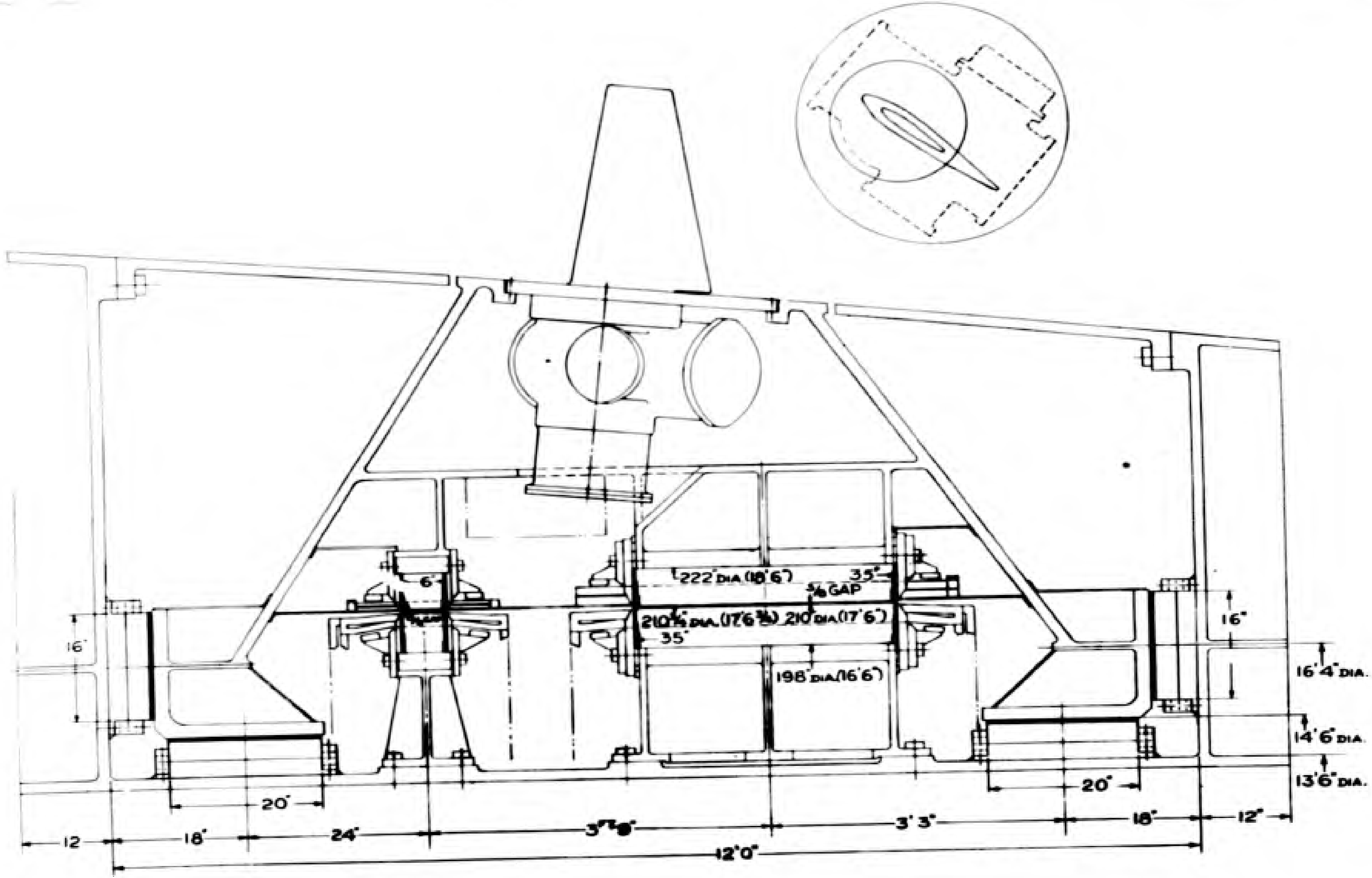


Figure 21. GENERAL ELECTRIC Propulsion Motor and Rotating Transformer Cross Section

poor power factor. In the NEPS design, for example, the power factor under rated conditions of operation is 56%. The synchronous motor for the present design will operate at unity power factor.

Higher efficiency. Some increase in efficiency is obtained because of the lower stator current required. This is partially offset by an increase in copper loss in the rotor winding.

Lower current. The substantially reduced current leads to smaller cabling and smaller hull penetrations.

Higher pull-out torque. The pull-out torque for the synchronous motor is 228%, compared with 160% maximum torque for the NEPS induction motor.

Smaller size. At approximately the same diameter, the stacked length of the NEPS induction motor is 55 inches, compared to 35 inches for the synchronous motor.

Smaller turbine generator. In comparison with the NEPS turbine generator, that required for the Tandem Propeller design is 33% lighter in weight and approximately two feet smaller in each major dimension. This amounts to a saving of over 90,000 lbs/ship.

Higher starting torque. Because the machine remains locked in step as frequency is decreased, it is capable of higher stability in the very low frequency region.

Low speed machines are normally built with salient poles. This results in an improved space factor, since more mmf per pole can be produced in a given volume using concentrated coil design than can be provided by a distributed field winding. However, such construction does not seem to be practical for a machine of this type for the environmental conditions anticipated. Insulation and sealing of the field winding would be much more difficult with salient pole construction, and the windage loss would tend to be much higher. For these reasons, the propulsion motors are designed as round rotor synchronous machines.

The motor is located between the hull and the propellers, and the rotor structure directly supports and drives the propellers, together with the mechanisms for controlling the pitch of each blade. A water-lubricated journal and thrust bearing at each end of the motor provides complete constraint of the rotating parts. The motor in turn is foundationed on a part of the hull which is both internally and externally free-flooding, and therefore, unaffected dimensionally by submergence pressure.

The design of both the motor and the rotating transformer is basically insensitive to submergence pressure, and while a 600 psig design pressure (about 1300 feet) has been used for this study, there is good potential for operation at much greater submergence pressures.

Each motor is rated:

Synchronous speed	50 rpm
Voltage	1050 volts
Frequency	30 cps
Phases	3
Power factor	1.0
Line current	3080 amperes
Voltamperes	5600 kva
Power input to stator[1]	5600 kw
Power at air gap	7030 hp
Bearing loss[2]	200 hp
Windage loss[3]	390 hp
Mechanical power to transformer	150 hp
Power at propeller hub	6290 hp
Field current	680 amperes
Field power	200 kw
Pull out torque	228%
Locked rotor torque	127%
Locked rotor current	388%
Air gap diameter	210 in.
Air gap	0.375 in.

Notes:

1 Does not include input to transformer stator

2 Includes loss for entire rotating assembly

3 Includes loss for entire rotating assembly except surface of propeller hub fair with hull

Torque and current during non-synchronous operation are shown as a function of speed in Figure 22.

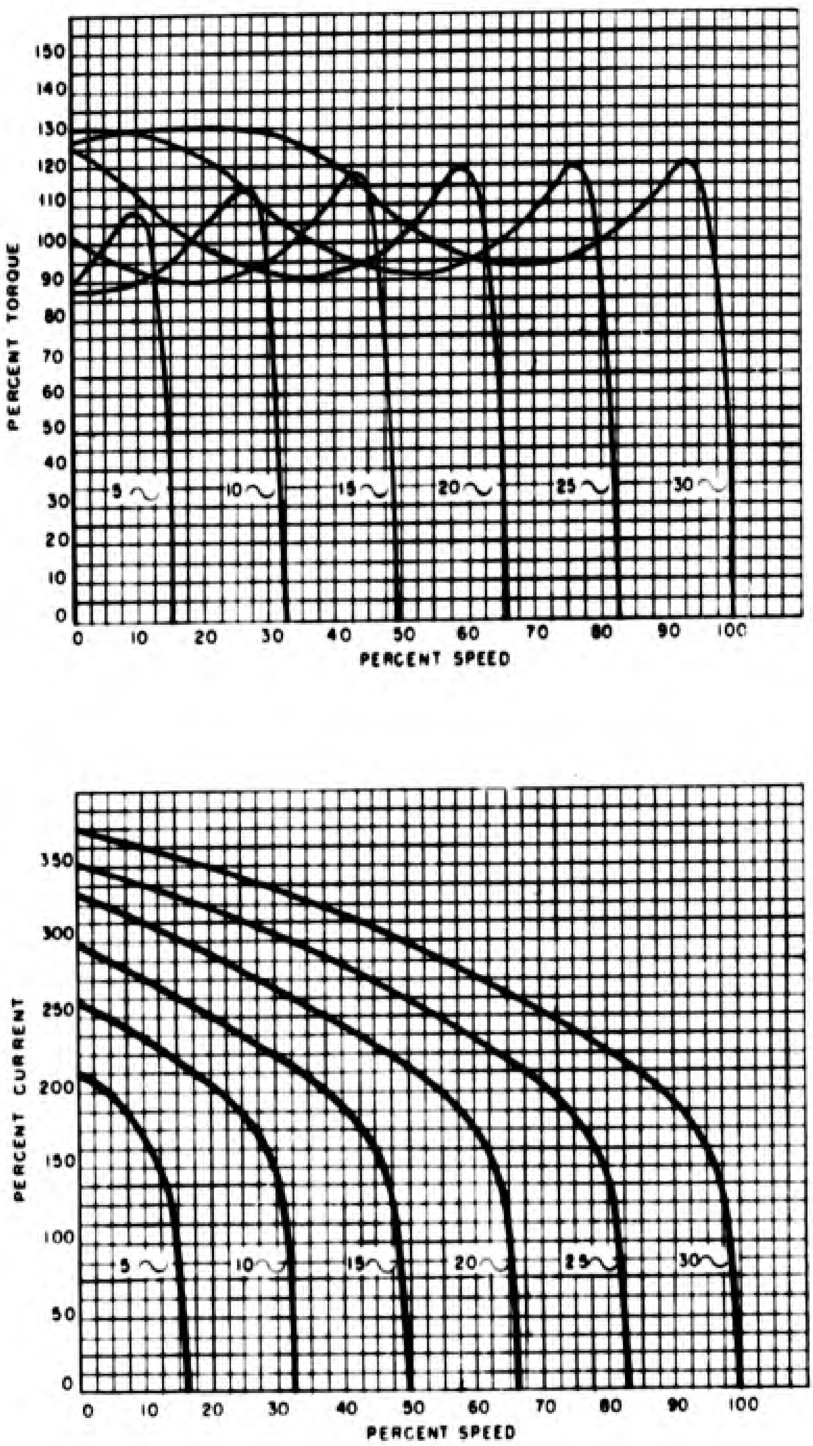


Figure 22. GENERAL ELECTRIC Propulsion Motor Torque vs. Speed and Current vs. Speed Curves

Calculation of efficiency is complicated by the multitude of ways in which losses can be charged between motor proper, transformer, propeller hub, and pitch changing system, and also the interchange of power between the propulsion motor and transformer. (At 50 rpm, two-thirds of the transformer output results from transformer action and one-third from mechanical shaft power.)

There is sufficient information in the foregoing table and in the table of data for the rotary transformer on page 99 to calculate efficiencies on almost any basis. To bracket the range, however, a gross full load electrical efficiency for the motor proper, consisting of

$$\frac{\text{gross air gap mechanical power}}{\text{stator electric power input + rotor electric power input}},$$

is 91%. A gross overall efficiency, consisting of

$$\frac{\text{net mechanical power to propeller (not including blade actuators)}}{\text{motor stator electric input + transformer stator electric input}},$$

is 80%. Other efficiencies will fall between these two values.

Dry weights in air are:

Stator of synchronous motor	60,300 lbs
Stator of rotating transformer	18,300 lbs.
Bearings	32,600 lbs.
Total stationary portion	111,200 lbs.
Rotor of synchronous motor	44,100 lbs.
Rotor of rotating transformer	11,700 lbs.
Rotating structural elements, including pitch-changing mechanisms and propeller blades	168,200 lbs.
Total rotating portion	224,000 lbs.

The weights of the submerged parts are reduced about 15% due to buoyancy. Approximately 73,000 lbs. of water are enclosed by the rotor when flooded.

Electrical Design - The fundamental electrical design of the motors is similar to that of a conventional round rotor synchronous motor. In normal practice round rotor constructions is limited to two and four pole high speed synchronous machines. The basic theory, however, is general and well advanced, and may be applied equally well to a low speed machine such as that used in the Tandem Propeller design. The inside-out configuration is unusual, but is otherwise of no electrical significance. The flux density in the iron is maintained at a level considerably below that used in industrial machines, thereby directly reducing noise generation and indirectly (by increasing size) improving heat dissipation. The heat transfer characteristics of the machine are excellent, because of the high heat transfer coefficients from the active materials to the sea water. This permits maintenance of the low maximum temperature required by the wet type of insulation without serious penalty in terms of larger machine size.

The stator is constructed in three segments to facilitate handling and assembly. Such construction is not unusual (for example, in large hydraulic turbine-driven generators), but these stator sections are unusual in that they are never interconnected either magnetically or electrically. The design of the stator winding is such that local flux distorition at the three joints is negligible, and since there are 72 poles, what flux distortion exists is of minor consequence. In addition, each segment is wound with two separate circuits, giving a total of six circuits per stator. Each circuit is energized separately, so that if any circuit fails it can be isolated to permit continued service of the motor. In this event, the diametrically opposite circuit must also be isolated in order to maintain balance in the magnetic forces.

The rotor winding is a straightforward direct current distributed field winding, with all active coils connected in series. Certain selected rotor coils are inactive (that is, they do not carry the DC field current). These are short-circuited. This helps to provide a more satisfactory distribution of flux in the machine during synchronous operation, and the short circuited turns provide a quadrature axis amortisseur which improves the starting performance considerably.

The rotor is constructed in two segments to facilitate assembly. The windings on the two halves of the rotor must be electrically connected after assembly, to provide a single DC current path through the field winding.

For minimum generator size and weight, the operating frequency should be as high as possible. On the other hand, the higher the operating frequency, the larger the number of poles required in the motor winding for a given rotational speed. The choice of frequency then represents a compromise which results in minimum weight and size of the combination. In this preliminary design, it appears that about 72 poles is the maximum number that can be wound in a machine of this size without seriously affecting the winding space factor. This results in a frequency at full speed of 30 cps. Additional design refinement may well indicate that the number of poles can be increased somewhat, with a corresponding increase in frequency and a more nearly optimum overall system design. It should be noted, however, that the low operating frequency is of some acoustic advantage.

Winding and Iron Protection - In building a motor suitable for total immersion in sea water it is essential that the magnetic members (stator and rotor laminations) be isolated from sea water, in order to prevent corrosion. It is equally essential to isolate the copper windings from sea water. The system used to isolate these active materials must be capable of substaining high cyclic pressure variations, must not corrode itself, and must be resistant to abrasion and attachment of marine growth. Two methods of isolating the windings and iron from the sea are recognized; enclosure of the active rotor and stator materials in metal envelopes, and coating of the windings and iron with non-metallic material. These are conveniently referred to as canned and wet designs, respectively.

Pump motors for use in atomic reactor loops have been built for many years and are designed for total immersion in primary coolant water. These have usually been built with very thin Inconel cans on the rotor and stator acting as barriers between the active materials and the water. Such motors are usually high speed, with small diameter and large length to diameter ratio. Even with the very thin cans used, the eddy current losses in the stator can are appreciable. Reference to the literature on can losses of machines of this type indicates that, for a machine the size of the Tandem Propeller propulsion motor, the eddy current loss in a can of practical dimensions becomes excessive. The resulting reduction in efficiency is sufficient to render a canned motor impractical for a machine this size.

On this basis the wet design approach was adopted as the most practical. In such a design the coating of the stator and rotor laminations and the insulation of the windings may be considered to be separate problems which can be treated independently. In an earlier investigation,* no magnetic materials were found which were adequately corrosion resistant. A literature search was conducted in connection with the Tandem Propeller project to be sure that this conclusion was still sound. No additional information was uncovered which would indicate that practical magnetic materials were available which were sufficiently corrosion resistant. It is, therefore, essential that the individual iron laminations be adequately coated to provide the necessary isolation. To accomplish this, the laminations are coated with a bondable Epoxy resin. After assembly into a core, this Epoxy is cured to bond the laminations together. One coat of an Epoxy primer and several layers of Epoxy anti-corrosion coating are then applied to the assembled core and cured. A number of Epoxy resins are satisfactory for this application. A preparation with the trade name "Copon" has been used successfully in similar applications.

This coating technique is applied to the rotor and stator structures of the main propulsion motor before installation of the windings. The

---

* Corrosion Investigation Report for Submergence of Induction Motors in Sea Water, Contract NObs 74446, General Electric Company.

same technique applies to the rotating transformer. The number of layers of Epoxy applied are selected to provide adequate sealing under the environmental conditions anticipated, without seriously impeding the flow of heat from the active materials to the water.

The application of insulation to windings which will operate in direct contact with sea water has been under active study in the General Electric Company since at least 1954. During this time considerable experience has been gained in the evaluation and application of materials for this type of service. This insulation material must have low moisture permeability, good electrical properties, good abrasion resistance, mechanical strength and, equally important, the material must be readily applied to a formed coil and processed to produce a reliably sealed insulation wall in the completely wound machine.

A polyethylene based insulation system was selected, but throughout the program consideration was continually given to alternate materials and systems. Among those considered were polyvinylchloride, silicone rubber, epoxies, polychlorotrifluoroethylene and polychlorotetrafluoroethylene. While the polychlorotrifluoroethylene and polychlorotetrafluoroethylene have lower moisture permeation rates than polyethylene, they were not adopted because of processing considerations. The polyvinylchloride presented problems in sealing, and the moisture permeability of silicone rubber was too great. The epoxy based insulations on the other hand were too rigid to withstand the flexing required in winding.

The insulation system selected has evolved from the experience gained in the development, evaluation, manufacture, and service operation (up to 4 years) of systems designed for this type of service and is considered to represent the best overall balance of electrical, mechanical and processability characteristics. It is similar to the insulation systems that have been used successfully in small rudder motors immersed in sea water and in wet motors designed for boiler feed pump applications. The insulation system consists of:

Strand insulation - Heavy Formex enamel, covered by cross-linked polyethylene, over-served with polyester filaments.

Coil insulation - multiple layers of cross-linked polyethylene, armored with a layer of woven polyester fabric filled with cross-linked polyethylene.

Series connections and leads - a layer of polyester film, over-taped with multiple layers of cross-linked polyethylene, armored with woven polyester fabric filled with cross-linked polyethylene.

This structure cures to a sealed system which is essentially impervious to sea water. Although long term data are lacking, this system appears to be satisfactory for operation at greatly increased submergence depths. 450 volt coils, insulated with the coil insulation system described above, passed a routine quality control test requirement of greater than 10,000 megohms between conductor and water after 48 hours of immersion in tap water at 3000 psi and 160°F. Other limited laboratory tests performed on this type of insulation system indicate a leveling off of decreasing insulation resistance after approximately 10 hours when submerged in tap water at 4000 psi and 100°C, and approximately 100 hours after submersion in salt water at 1800 psi and 100°C.

The coils are placed in the stator slots in the conventional manner, without additional slot insulation. Wedges of epoxy resin are cast in place so as not to damage the iron protective coating. Any voids in the slots between the coils and iron are free flooding.

The temperature rating of this insulation system is relatively low. At steady state full load the maximum stator copper temperature does not exceed 80°C, and the same design limitation applies to the rotor, as well as both windings of the rotating transformer.

The insulation system described above is most suitable for low voltages, when the environment is taken into account. Essentially all experience with windings of this type in water is at 500 volts or below. On the other hand, for this size machine the propulsion generator is most easily designed and constructed for higher voltages, of the order of

5000 volts or above. The resulting operating voltage, 1050 volts, represents a compromise and is about the minimum practical voltage for the generator and is also sufficiently low for the motor insulation system.

Both the insulation and the protective coating for the iron show little attraction for marine growth. The motors are advantageously located in that they are always completely submerged, and thus any marine growth which might appear would not harden. Marine growth will not appear while the motors are operating, and if it does become troublesome during prolonged periods of idleness, the propeller hubs furnish a substantially closed volume of water subject to convenient poisoning (a procedure used even in open water by some merchant ships).

It is conceivable that the water around the top portion of the motors might freeze during idle periods in cold weather. While this will not necessarily cause any damage if the motors are not started while still frozen, it is a situation best avoided. Freezing can be prevented by using the motor windings as heaters.

Stray Electromagnetic Fields - Since the propulsion motors are intimately attached to the hull, they are in a position particularly conducive to production of stray electric and magnetic fields. For completeness, the following discussion of this topic is substantially reproduced here from the NEPS report, except that numbers have been changed slightly as required.

In the vicinity of the supports for the rotor stacking the leakage flux through the back of the laminations and out through the structural members and propeller hub is conservatively calculated as 0.026 milligauss at the hub surface. The corresponding electric field is 0.038 microvolts/meter, and both decay as $e^{-0.324x}$, where x is in inches. Since this decay is equivalent to a reduction in magnitude by a factor of one million for approximately every three feet, no difficulty would be expected either with respect to effects on equipment on the same ship or detection of the ship by the enemy.

This, however, may be an unduly optimistic approach. The hull of a more conventional ship, for example, should not exhibit any external fields, but it nevertheless does. Therefore it is qualitatively assumed that the propulsion motors make some additional contribution to the already existing electric and magnetic fields around the hull, and from this assumption several qualitative observations can be made:

> Well balanced and shielded sonar transducers will not be affected by electrical noise pickup. The current use of electrostrictive instead of magnetostrictive transducers is helpful in this respect. As is presently required, careful attention to shielding of sonar cables will be necessary.
>
> Some increase in interference with VLF communications receiving is possible.
>
> The motor frequencies are above the frequency range of present airborne magnetic anomaly detectors. If the detectors were made sensitive to higher frequencies detection might be enhanced, but conversely the deep submergence for which these propulsion motors are particularly adapted serves to effectively increase the distance between ship and detector and to impose considerable attenuation of the higher frequency field.
>
> The motor frequencies are above the frequency range of present mines. If the mines were made sensitive to higher frequencies detection might be enhanced, and in this case the distance could not be increased by deeper submergence because these devices are used in harbors and shallow water.
>
> Electromagnetic detection by ship-borne detectors is of no consequence since acoustic detection is possible at much greater ranges.

The operational significance of these observations is beyond the scope of this report.

Noise - The generation and radiation of audible noise is one of the most critical problems in the design of submarine propulsion equipment. While considerable progress has been made in designing extremely low noise machinery of conventional construction and predicting noise levels to be expected of such machines, the unusual size and mechanical construction of the Tandem Propeller motors renders extrapolation of such data of uncertain reliability. While it is difficult to predict with accuracy the nature and level of the output from the propulsion motors, certain steps can be made during the design stages to minimize the noise production and radiation.

The stator is wound with an integral number of slots per pole. In addition a winding pitch was chosen which will minimize the fifth and seventh mmf harmonics. Both of these contribute to minimization of magnetic noise production by smoothing out the air gap flux wave. In selecting the number of stator slots per pole, the maximum possible number was used. Because of the lower stator current required, a larger number of slots per pole is possible in the Tandem Propeller design than in the case of the NEPS design. This should result in a lower level of noise output.

In general, the larger the air gap, the lower the level of slot frequency noise developed. In this respect, the use of the synchronous motor with its larger permissible air gap is highly advantageous relative to the induction motor. In addition, some slip noise will be produced in the NEPS induction motor as a result of the slight distortion of flux in the vicinity of the splits in the core. This effect will be entirely absent in the synchronous machine used in the TAPS design.

The average flux density in the motor air gap is approximately 32 KL/sq. in. This is conservatively low, and corresponds favorably with values used in present-day emergency propulsion motors, which are subject to stringent noise requirements.

Methods presently exist for calculating, with reasonable accuracy, the magnitude and distribution of the driving forces due to air gap flux in synchronous machines. If the machine is submerged in sea water, the gap flux and the resulting driving forces should be essentially unchanged. The forced deflection of the structural parts, when acted upon by these forces, should also be essentially unchanged.

Measurement and/or analysis of underwater sound generated by large submerged low speed propulsion motors is apparently non-existent. The sound levels of objects having similar shape and size have been measured and analyzed, but in general the vibration modes and number of stationary nodes are of low order. Extrapolation could lead to gross error for the case involving a high number of rotating nodes.

A brief "first cut" acoustic study of the NEPS propulsion system is included in the NEPS feasibility study report,* much of which also applies to the Tandem Propeller system. A comprehensive analysis, supported by test data, is essential before a complete evaluation of the Tandem Propeller system as a noise source can be made.

Structural Design - The general structural design of the propulsion motors can be seen in Figure 21 (page 71).

Geometry and Assembly - It is essential for successful and quiet operation that the geometry of the motors be accurately controlled and preserved. In order to insure good rigidity, circularity, concentricity, and generally sound construction, the following overall manufacturing and assembly methods are used:

> The sections of hull to which the motors are attached are shipped to the motor manufacturer, with all major internal structure and welding completed so as to avoid subsequent distortion due to welding. The outer surfaces are machined suitably for mounting the stators and bearing pads. The hull sections are then returned to the shipyard and welded to the remainder of the hull. These welds are made sufficiently far from the machined surfaces to avoid distortion.
>
> The three stator segments for each motor are bolted together to form a complete circle, and are then machined as a unit. The inside diameter is sized to fit directly on the previously machined hull outer surface without any shimming. The segments are then unbolted for shipping.
>
> The propeller hubs are shipped to the motor manufacturer, or are alternately fabricated directly by him. The motor rotors and bearing runners are welded in, and the two segments for each motor are bolted together to form a complete circle and are then machined as a unit. The segments are then unbolted for shipping.

* General Dynamics/Electric Boat Report C-411-62-024, "Feasibility of a Novel Electric Power Propulsion System for a Submarine," Volume I of III, Engineering Study, CONFIDENTIAL.

At the shipyard, the stator segments are bolted directly to the hull and to each other, and the bearing pads are installed. The rotor segments are assembled around the stators and bolted together.

It is intended that no shimming be required in assembly. As a backup means, however, some shimming can be done by adjusting the thickness of each bearing pad.

The heaviest single piece, 112,000 lbs., is a 180° segment of the rotor-hub assembly. If necessary, this can be reduced about 20% by removing the propeller blades and pitch-changing "packages."

Interchangeability and Repair - The two motor stators are identical, and each stator in turn consists of three identical 120° segments. The motor rotors and bearing runners are also identical, and each in turn consists of two identical 180° segments; however, the propeller hubs to which they are welded may be different due to the shape of the hull. Depending on the requirements of the ship design, it may be possible to design the rotor segments to be identical for the forward and aft motors. This is true in the case of the ship design in this report.

While this minimizes the number of drawings required and somewhat simplifies manufacturing, it does not follow that the segments are individually interchangeable. The stator and rotor-hub assembly are finish machined assembled as complete circles, and must be re-assembled identically on the boat. While perhaps not impossible, interchangeability of individual segments is not practical due to their large size and weight and the small dimensional tolerances involved. Thus if replacement were required, a complete stator or a complete rotor-hub assembly would be replaced. (Useable parts can, however, be returned for machining with new parts for subsequent use as a set.)

The need for replacement is remote, however. The iron protective coating and the winding insulation can be field repaired, permitting replacement of coils while in drydock. For a more extensive undertaking, such as rewinding the entire stator or rotor, it would probably be desirable to return the stator or rotor-hub assembly to the factory.

Distortion - A thorough analysis was made of the stresses and deflections in the motor structure caused by centrifugal loading, gravity, magnetic pull, thrust and turning moments. The rotor and stator structures are very stiff, and all stresses and deflections under normal operating conditions are well within reasonable limits.

The most serious stress and deflection conditions exist under shock loading. A shock analysis of the rotor-hub assembly was made, using the method of Belsheim and O'Hara, assuming step-function velocity inputs of 8 ft/sec. transversely and 4 ft/sec. longitudinally. A simple three-mass equivalent system was used in the analysis. Calculated loadings on the order of 85 g's were obtained on some individual parts of the rotor, and the resultant loading on the journal bearings was 36 g's. Stresses on the order of, but no exceeding, the yield strength were obtained on some parts, which indicates the adequacy of the rotor structure.

Average bearing pressures under shock loading are of the order of 1250 psi and are well within the compressive strength of the bearing material.

A shock analysis of the stator was made using a loading of 70 g's transversely and 35 g's longitudinally. Again, stresses, in the structure were within the yield strength of the material.

Windage - It can be seen in Figure 21 (page 71) that the entire outer surface of the rotor-hub assembly is smooth, with the exception of a few small flooding holes. Where the structural members are not already smooth, they are plated over. This serves to minimize the windage loss, which nevertheless is still about 390 hp for each rotor-hub assembly. This figure includes loss over the entire surface with the exception of the surface of the propeller hub fair with the hull.

In this design, and in the NEPS study which preceded it, the windage loss has a pronounced effect on the determination of the optimum rotational speed. In the NEPS case, 50 rpm was found to be essentially the optimum speed. Since the Tandem Propeller construction is dimensionally similar, 50 rpm was also selected for the present design. It is not necessary for the feasibility study to carefully optimize the speed, and this was not done, but 50 rpm appears to be substantially the optimum speed. The optimum speed of course depends upon the criteria selected, but if the speed is lowered, at best a few hundred horsepower/propeller can be gained, and this comes at the cost of a progressively more severe weight and space penalty. If the speed is raised, weight and space will be reduced, but this comes at the cost of a progressively more severe reduction in propulsion power. It is possible, however, that a modest increase in speed might be advantageous on some overall basis.

Cooling - Cooling water for the motors enters through the clearance behind the aft edge of the propeller hub, passes through the rotating transformer and motor in series, and leaves through the clearance ahead of the forward edge of the propeller hub. In passing through the machine, the water travels between the bearing pads and then both through the air gaps and (through holes in the structure) past the backs of the stators of the rotating transformer and motor. Some water will also circulate through the rotor-hub assembly via flooding holes, which will help to cool the rotor. Pressure head for circulating the water is derived from the differential pressure across the propellers. There is ample pressure head available to provide the approximately 150 gpm flow required.

Access and Foundationing - It is evident that the extra plating on the rotor to minimize windage loss additionally serves to make access to the motors even more difficult than it would otherwise be. Access for inspection was recognized as a significant topic, but was not pursued since it was considered to be resolvable in a detailed design and would not affect feasibility. Similarly, the hull section forming the foundation for the motor was not investigated in detail, since, while not necessarily a simple task, a foundation can certainly be designed when the need arises.

Bearing Design - Motor bearings constitute one of the critical feasibility areas in the Tandem Propeller concept. In connection with the NEPS study mentioned earlier, an intensive investigation of the bearing problem was undertaken by General Electric, Electric Boat, and a private consultant to Electric Boat, Mr. Stanley Abramovitz. At the conclusion of the earlier study, all those involved concurred in the general conclusion that there is a high probability that a satisfactory bearing system can be found. The general arrangement, materials, and operating characteristics and requirements of the Tandem Propeller bearings are quite similar to those of the NEPS bearings, and the same conclusions hold.

It was further concluded in the NEPS study that the bearings should consist of separate pads arranged to tilt so as to promote film formation. Some fraction of the load would be carried on boundary lubricated surfaces, with the remaining amount, probably smaller, carried on hydrodynamic film. A relatively simple bearing system was devised which was believed to have a reasonable chance of success. Variations of this basic design were also studied which would enhance the fluid film support mechanism at the expense of some extra complication in the bearing design.

Design Background - Sea water lubrication has been used successfully for a long time in ships' stern tube and strut bearings. Generally, these journal bearings consist of a large number of long, narrow staves separated from each other by axial grooves for water circulation and to flush out dirt and wear debris. Because of the low viscosity of water and the long, narrow stave design, stern tube and strut bearings do not operate on a full hydrodynamic film. Practice has been to allow low bearing loads generally in the range of 20 to 50 psi, based on projected area. Within this load range the calculated film thicknesses are about 0.0002" to 0.0004". Local asperities, sand particles and wear debris are larger than this, so that at least a fraction of the load is always carried by direct contact between wetted surfaces; i. e., under boundary lubrication. Because of this, a small amount of wear occurs continuously in operation. Material selection is critical, the materials used being those which have a low wear rate, no pickup of wear debris by the mating surface, and no extension of local wear or dirt score marks. Laminated phenolics, lignum vitae and Buna N rubber have all been successfully used as stern tube and strut bearing materials. The mating surface is generally a monel metal or bronze sleeve.

Water lubricated bearings of the type discussed above have been manufactured in sizes up to about 40" diameter (the largest sizes have been used in rolling mills, where water lubrication is also commonly used). The Tandem Propeller system, however, requires journal bearings about 174" diameter and thrust bearings about 196" mean diameter, which is considerably larger than any previously used. The large size of the bearings is important in that manufacturing, assembly and alignment tolerances, out of roundness, and structural deflections are many times larger than any fluid film thicknesses that can be generated in the bearings. Misalignments and structural deflections cause high local loads, which are relieved in operation by corresponding local wear. Wear rate is critical and must be kept low, since it affects the motor gap symmetry.

Requirements - Bearings for the Tandem Propeller system must fulfill the requirements of the general specifications for submarine machinery as well as particular requirements for this application. They should be satisfactory for the life of the submarine with a reasonable amount of maintenance. Replacement of bearing pads should not be more frequent than at yearly intervals.

Since the entire motor is outside of the hull of the vessel, no lubricant can be used other than sea water or dry lubricants.

In order to maintain the necessary tolerance in the air gap, the rotor must be held central with respect to the stator within 0.0375 inches. The fore and aft positioning of the rotor is much less critical, and no initial requirements are imposed.

The two motors are essentially identical, and therefore have identical bearing loading. The bearing loads consist of rotor-hub weight and propeller blade forces plus gyroscopic moments developed when the vessel is turned.

Rotor weight: 224,000 lbs.

Thrust: 80,000 lbs.

Transverse force due to cyclic pitch variation: 40,000 lbs.

Moment about transverse axis due to cyclic pitch variation: 750,000 lb-ft.

Gyroscopic moment due to rotation of the ship about a transverse axis: 854,000 lb-ft.

Calculation of the gyroscopic moment assumes a turning rate of 0.17 radians per second, with the rotor turning at 50 rpm. The direction of the gyroscopic moment is, of course, at right angles to the direction of rotation of the vessel and can be about any transverse axis.

Basic Bearing Design - The approach used in the present feasibility study was to determine, through calculations and review of sea water lubrication experience, whether journal and thrust bearings for the Tandem Propeller system could be designed which tolerate misalignment, structural deflections and sand passage without failure and with low wear rate, so that long term operation can be achieved with minimum maintenance. The other criteria which had to be satisfied were quiet operation and simplicity of manufacture and assembly.

During the course of the NEPS study, a number of different arrangements of thrust and journal bearings was investigated. Each of these was ultimately discarded in favor of an arrangement consisting of journal bearings close to the ends of the rotor, and thrust surfaces just outside of the journal bearings. The reasons for the selection of this configuration were:

Simplicity of manufacture, assembly and maintenance.

Compactness and structural rigidity.

Low reactions and balanced loading both under linear ship motion and during turns.

Minimum allowable bearing diameter, within the restrictions imposed by the ship design.

On the basis of the NEPS study, a similar bearing design was selected for the Tandem Propeller. This is illustrated in cross-section on the motor layout drawing, Figure 21 (page 71). Additional details of construction are presented in Figures 23 and 24.

Bearing Materials - The materials most commonly used in water-lubricated bearings are phenolic based plastics, lignum vitae and rubber. A great deal of experience has also been obtained with graphite bearings, but only in relatively small sizes. The first three are considered to be the most suitable materials which are known to have the proper characteristics for application to large water-lubricated bearings. Of these, graphite-filled phenolic is considered to be the most satisfactory for the following reasons:

It can be manufactured in the preferred pad type configuration.

Considerable experience has been gained with it with sea water lubrication in stern tube and other applications.

The modulus of elasticity of laminated phenolics is in the range of 600,000 to 1,000,000 psi, so that deflections under load are extremely small and do not affect motor gap symmetry.

Compressive strength of laminated phenolics is high (30,000 to 40,000 psi) and resistance to shock is excellent (for example, tank turret ball bearings employ plastic balls because of their ability to withstand the high firing shocks).

Lignum vitae possesses some of these advantages, but it cannot readily be manufactured in the large pad configuration required. Rubber must be rejected because of its low modulus of elasticity, which would lead to excessive deflections under load (on the order of 0.060").

The phenolic material will be bonded to bronze backing plates to provide the necessary rigidity. Present bonding techniques will permit adequate bond shear strength, but if need be, additional strength can be obtained by means of dovetails or grooves. Monel metal has been selected for the mating surfaces due to the extensive successful experience with this metal.

Design and Performance Details - Each journal and thrust bearing consists of a number of roughly square pads with graphite-filled phenolic

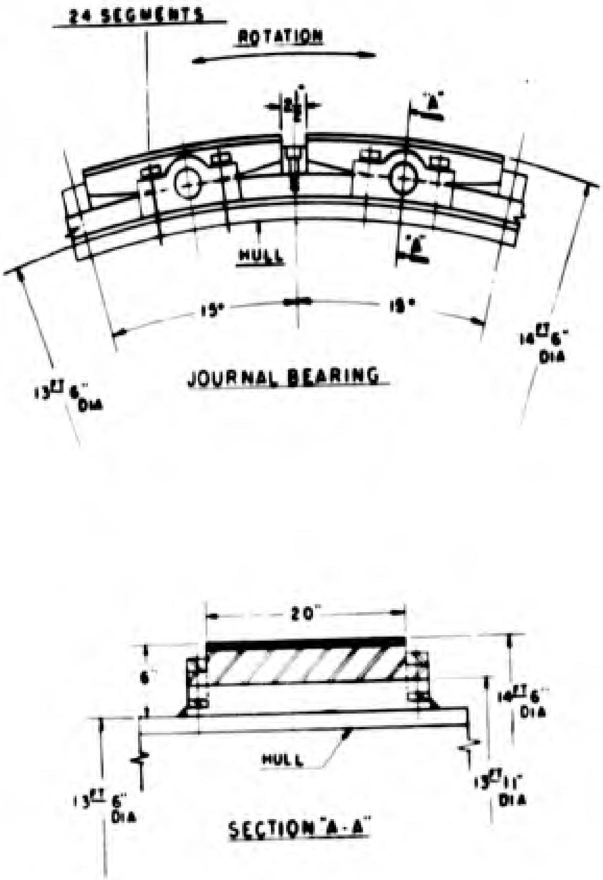


Figure 23. GENERAL ELECTRIC Propulsion Motor Journal Bearing

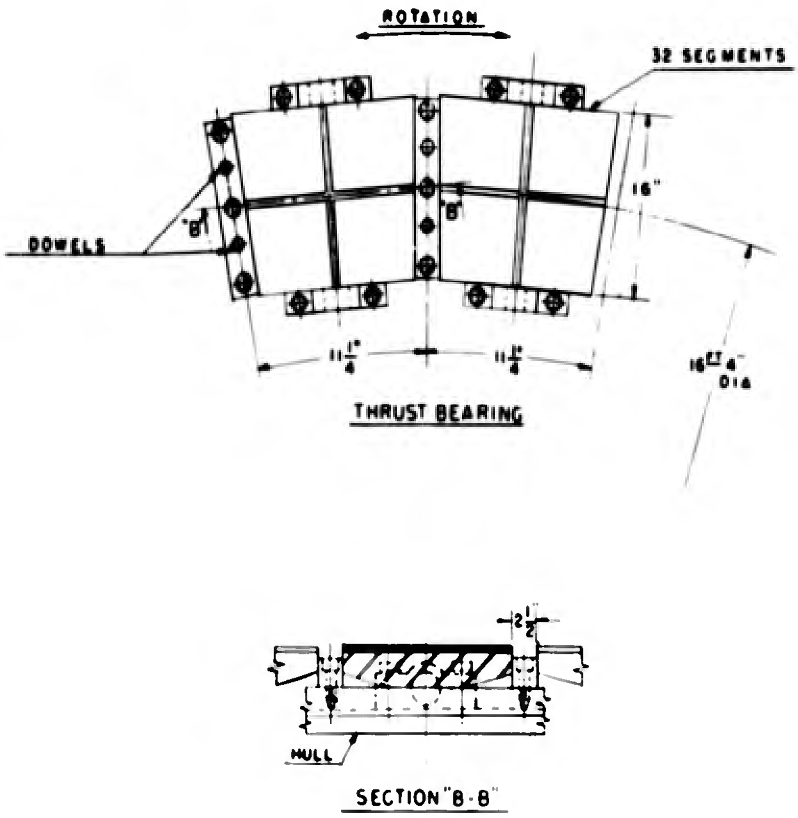


Figure 24. GENERAL ELECTRIC Propulsion Motor Thrust Bearing

facing running on a monel surface. The pads are arranged to roll slightly on a large radius cylindrical surface in order to promote proper alignment and to assist in starting after prolonged shutdown. The following data pertain to the bearings for one motor and rotating transformer assembly:

Journal bearings:

Bearing diameter	14 ft. 6 in.
Number of pads	24
Size of pads	20 in. x 20-1/4 in.
Maximum bearing loading	
Linear motion	34 psi
Maximum ship turn	72 psi
Power loss at 50 rpm	125 hp
Film thickness (approximate)	0.001 in.
Break-away torque	50%

Thrust bearings:

Bearing mean diameter	16 ft. 4 in.
Number of pads	32
Size of pads	16 in. x 16-3/4 in.
Maximum bearing loading	
Linear motion	10 psi
Maximum ship turn	60 psi
Power loss at 50 rpm	75 hp

It is clear that the irregularities in machining such large bearings as these and deflections of the structure will greatly exceed the expected film thickness. It is not considered possible, therefore, to design for full hydrodynamic lubrication. As in the case of ship's stern tube bearings, some continuous wear is expected. The present bearings are designed such that the amount of wear will not require replacement of the bearing pads more frequently than about once a year. Stern tube bearings normally wear in rapidly when new and at a lesser rate thereafter. Over a 24 month period, they may average

0.002" per month. In consideration of the allowable maximum motor gap asymmetry of 0.0375", it appears that replacement of worn pads may be required at intervals of about 18 months. However, in several respects, the present bearings are more sophisticated and should operate with less wear than stern tube bearings. The pivoted journal bearing pads tend to align themselves favorably for film formation during steady operation. When starting, the slight tilting action permitted by the pivot assists in breaking the pads away from the bearing surface, permitting water to enter the gap. In this way, operation without water film is practically eliminated. Starting torque and wear during startup are thus sharply reduced.

The extent to which sand and silt contribute to wear in stern tube bearings is not known. However, an improvement in bearing life would certainly accrue if entry of foreign matter could be prevented. In the Tandem Propeller system design, the bearings are located adjacent to the inner hull, with water for lubrication and cooling supplied by flow through the gap between the rotating portion of the hull (propeller hub) and the outer hull proper. At least the larger particles of dirt can be expected to be expelled by the combination of the axial flow of the slip stream past the gap in the hull and centrifugal force imparted to the particles by the rotating propeller hub. With these points in mind, somewhat better performance can be expected for the Tandem Propeller System bearings than for usual stern tube bearings.

Maximum breakaway torque is required when the motor is started after a prolonged shutdown. Assuming a coefficient of friction of 0.3, which should be quite conservative, this maximum breakaway torque is approximately 50% of rated full-load torque. The locked rotor torque of 90% at 17% frequency and voltage provides adequate margin for starting.

It should be noted that this dry starting friction coefficient of 0.3 applies only during start up after a prolonged shutdown. Routine starts after a short shutdown encounter much lower breakaway torque. In addition, the friction coefficient drops rapidly as the rotor turns over, reaching the running value of about 0.007 within the first one or two revolutions.

Breakaway torque may be further reduced in a motor of this design by means of a relatively simple expedient, at the expense of a slight complication in the switchgear. At starting, one of the three stator sections (two of the parallel circuits of the stator winding) at the top of the machine is left temporarily unexcited. This has the effect of producing an unbalanced magnetic pull tending to lift the rotor vertically upward. The magnetic force available is of the order of magnitude of the weight of the rotor. The remaining stator sections which are excited produce about two-thirds of the nominal motor starting torque, which is more than enough to break the rotor away and start to bring it up to speed, after which the final section is connected to the line.

Vibration, Shock and Noise - Due to the importance of shock resistance and quietness, careful attention was paid to vibration of the rotor and excitation by the bearings.

A shock analysis of the rotor was made, using the method of Belsheim and O'Hara, assuming step-function velocity inputs of 8 ft. per second transversely and 4 ft. per second longitudinally. A relatively simple three-mass system was used in the analysis. Calculated loading on the journal bearings was 36 g's, resulting in average bearing pressures of about 1250 psi. This loading causes no distress to the plastic bearing surfaces or their supporting members, and the brief duration of loading prevents any adverse wear effects.

Experience has shown that sea water lubricated rubber or plastic stern tube bearings sometimes produce a high pitched noise under bearing load, at creep speeds up to about 10 to 15 rpm. This phenomenon is not yet fully explained, and conflicting opinions exist to explain its occurrence in some instances and absence in others. It is generally accepted, however, that the noise occurs only at very low speeds and that its principle causes are stick-slip friction at low speeds that induce torsional vibrations in the bearing shell or in the shaft. Here, the size of the Tandem Propeller System is advantageous, because the surface speeds are high, even at creep rotation, thus inhibiting stick-slip, and because the extremely high torsional stiffness of the system should prevent torsional vibrations. For these reasons, it is felt that creep speed noise, akin to that experienced in stern tube bearings, is unlikely to develop in the Tandem Propeller System. In the present design, it is estimated that above approximately 3 rpm film formation will be substantially as complete as at full speed. Squealing is, therefore, not expected to be a problem above this speed. Furthermore, operation of the propulsion motors at such low speeds is rare and of short duration. Bearings are not an important factor in noise at higher speeds.

Installation and Maintenance - A previous part of this section has discussed the assembly of the motor in the submarine by building the rotor in halves that may be placed around the hull with the parts fastened together in a relatively straightforward manner. Reference to the drawings of the bearings, Figures 23 and 24 (pages 92 and 93), shows that the bearing pads are maintained captive by keeper pieces. The journal bearing may, therefore, be installed on the hull structure before the rotor is brought into place. With the rotor in place, the journal bearings may not be removed. Access to the bearings and to check the motor air gap may be provided by removing a portion of the plating on the rotating member. To replace the journal bearings, the rotor must be removed from the ship. The thrust bearings may be slipped into place after the rotor is installed and, of course, they can be inspected or replaced without removal of the rotor. To provide this feature for the journal bearings, a great deal more space would be required, as well as a much more complicated structure. Emphasis is placed on the development of journal bearings with satisfactory operating life to avoid the penalties attendant in a design for easy removal.

Design Variations - It is not known exactly what characteristics will result from service, especially in the presence of sand and silt. For example, the design described above provides a cylindrical support surface for the bearing pads which will promote, to some degree, formation of a hydrodynamic film. The regime of operation of this bearing is a matter of conjecture. It is entirely possible that greatly improved wear rates would accrue if a more sophisticated support arrangement were used. This may take form of a spherical support of small radius rather than a large cylindrical surface. Loading on the various pads may require adjustment by means of jacking screws or shims, although the concept proposed is that initial wear compensate for manufacturing inaccuracies. Deflections occurring in service may not satisfactorily be accounted for by wear, and it may be necessary to provide an equalization system for each of several groups of pads in both journal and thrust bearings.

In the initial bearing design, the pivot has been located at the centerline of each pad. Since each motor will turn in only one direction, improved bearing performance will undoubtedly result if the pivot is more nearly optimally located with respect to the center of pressure. In addition, better wear characteristics will result if the pads are arranged to tilt about an axis normal to the direction of relative motion.

In addition to considering different types of arrangements and support, experimentation with various pad geometries, such as flat and crowned shoes and with rounded or sharp leading and trailing edges is desirable. Crowning of the pad surfaces is beneficial to film formation. The surface is also kept wetted to a greater extent than with flat shoes, and tilting is assisted during startup which permits all of the surface to be wetted.

Sand particles in the water flowing into the bearings may be a very severe problem. There is some experimental evidence that at least the larger particles (greater than the leading edge film thickness) are excluded by giving each pad a sharp leading edge. The effect of this sharp corner on film forming tendency is not known with certainty, but it is not expected to adversely affect crowned pads.

A test program to study these questions and to assess the merits of suggested variations to the basic design could be carried out successfully on a somewhat reduced scale. Full size test rigs would be necessary for complete quantitative answers to all of the possible questions, but it is believed that tests at about one-half of full scale would provide sufficient data to lend complete confidence to the design. Tests would initially establish the performance of the basic design with respect to wear rates, sensitivity to machining tolerances or structural deflections, and susceptibility to chatter or squealing. Testing would then proceed to refine this design by addition of the various features described above as they are indicated. A test would also determine the extent of film formation in order to establish the extent of possible gain from modifications intended to improve film formation.

POWER TRANSFER TO THE ROTOR

Power on the rotor for field excitation of the propulsion motor and for supplying the pitch changing mechanisms is supplied by a machine similar to a wound rotor induction motor. The stator of this machine is energized by a three-phase, 60 cycle power supply in the ship's hull. Exciting currents establish a rotating magnetic field in the gap between rotor and stator, generating a voltage in the secondary or rotor windings. At standstill, power is transferred by transformer action alone. When the propulsion motor is rotating, the rotor power is a result of both transformer and generator action. The field rotates in the direction opposite to that of the motor.

Machine Rating

This machine, termed a rotating transformer in this report, has an output on the rotor rated:

Speed	0 - 50 rpm
Voltage	129 - 193 volts
Frequency	60 - 90 cycles
Voltamperes	267 - 400 kva
Power	190 - 300 kw
Current	1200 amperes

Phases	3
Poles	72

Stator Excitation

The stator of the rotating transformer requires three-phase, 60 cycle excitation, approximately 800 kva. There is some variation of the input active and reactive voltamperes with rotor power requirements, but because the rotating transformer needs 600 kva to establish its gap flux, the input kva is relatively constant and characterized by low power factor, on the order of 0.3 to 0.5. The active input power is:

Speed	0 - 50 rpm
Electric power	270 - 290 kw
Mechanical power	0 - 112 kw
Total power input	270 - 402 kw

In these figures a constant 100 kw load for the pitch changing system has been assumed for convenience, although the load at reduced speed is actually lower.

Rotating transformer excitation is supplied from the ship's service generator with no adjustment of the transformer rotor voltage. With constant exciting voltage the rotor voltage varies directly from 129 volts AC at zero shaft speed to 193 volts at 50 rpm. The corresponding rectified DC voltage applied to the propulsion motor varies from 173 to 260 volts with increasing speed. This automatically holds approximately unity power factor on the propulsion motor. The AC output is essentially constant volts per cycle, which is ideal for input to the blade actuating motors.

Alternately the rotating transformer may be supplied from a separate generator inside the ship's hull, permitting independent adjustment of transformer voltage by generator field control. This, however, does not presently appear to be necessary.

Motor Field Excitation

Propulsion motor field excitation is rectified rotating transformer power. The circuit arrangement is shown in Figure 25. Transformer rotor windings connect to a three-phase full wave bridge represented by diodes D_1 - D_6. The bridge output supplies the propulsion motor field.

Each rectifier represented as a diode in Figure 25 actually consists of an assembly of elements as shown in Figure 26. There are four paralleled silicon rectifier cells, each having a series fuse and resistor. The resistor is a short strip of stainless steel connector, with a full load voltage drop of about one volt to insure equal current division among the parallel paths. In the event of a diode random failure a fuse serves to isolate the defective diode, permitting uninterrupted operation with the three remaining diodes having adequate capacity to carry full load field current.

Inverse Field Current Protection

The propulsion motor field generates reverse field current whenever the motor is not in synchronism with its generator. This occurs at start and may occur during unusual running conditions. For example, if the generator voltage falls and reappears due to opening and reclosing the generator field, the two machines would very likely lose synchronism. Under conditions of reverse field current, a field voltage protective circuit senses rising field voltage and connects a resistor across the field. This circuit is a simple arrangement of silicon controlled rectifiers and other static devices. When synchronism is restored the field resistor is open circuited to avoid loss of field power. If a return path for reverse field current is not provided, the field voltage will rise to a value likely to destroy the diodes in the rectifier bridge during asynchronous operation.

Rotor mounted equipment designed on these principles has been thoroughly tested by the General Electric Company and is now successfully operating on G-E industrial motors as large as 2500 horsepower.

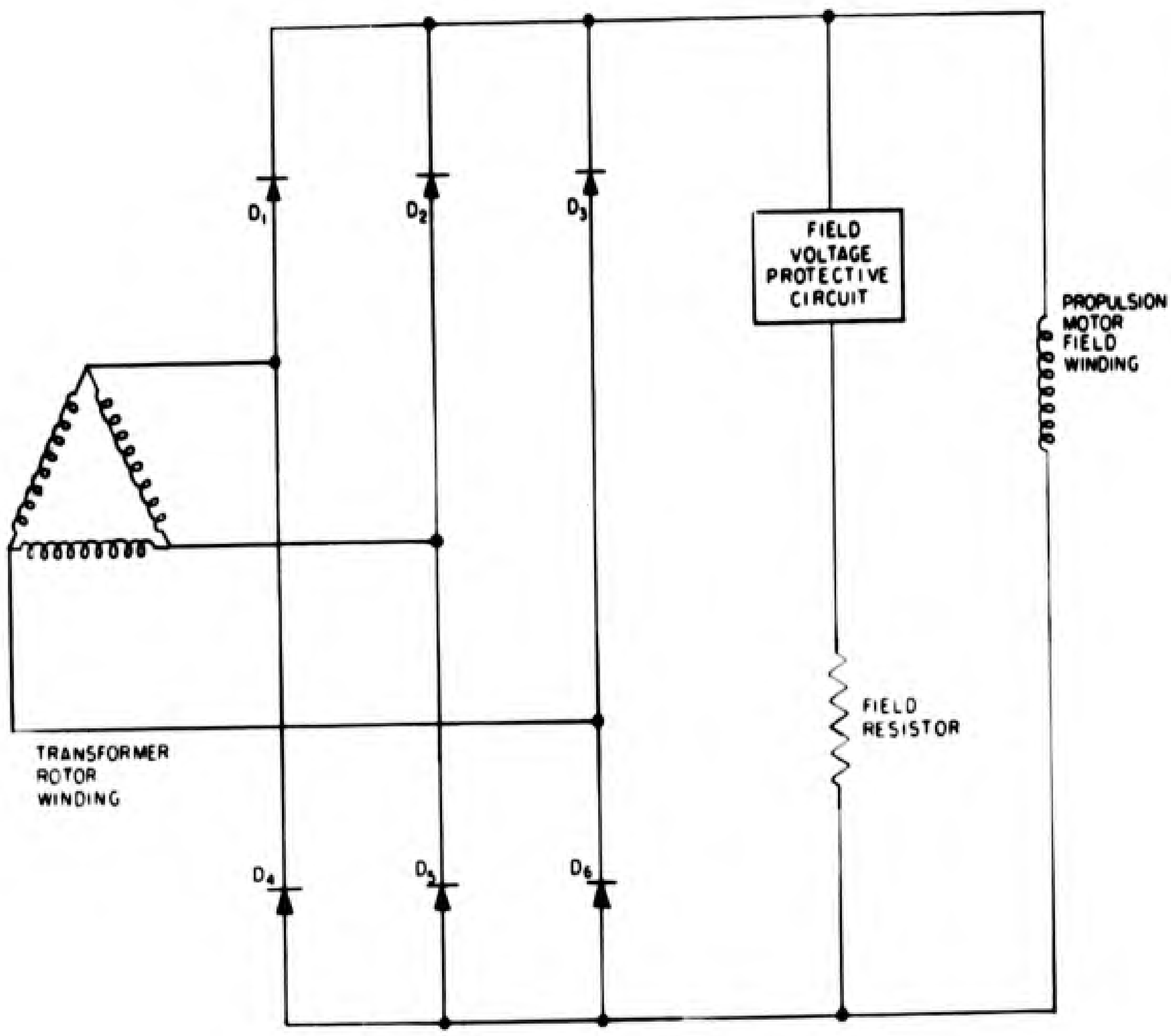


Figure 25. GENERAL ELECTRIC Synchronous Motor Excitation Circuit

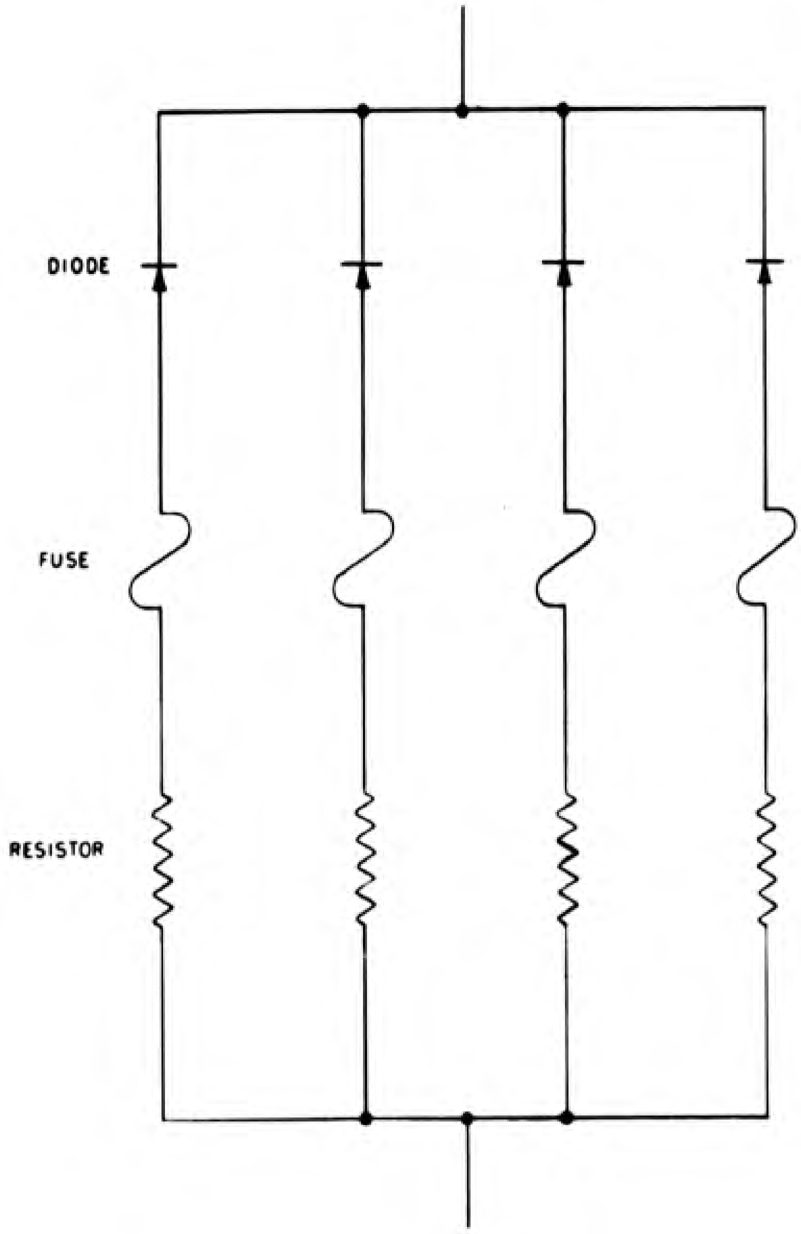


Figure 26. GENERAL ELECTRIC Diode Connection Details for Synchronous Motor Excitation Circuit

Rectifier Assembly

The silicon rectifiers, GE's high current, hermetically sealed, hard solder cells, are very efficient rectifying units. With proper application, known failure mechanisms are avoided, and the silicon rectifiers can be expected to operate for many years without a single cell failure.

The entire rectifier assembly, mounted in a metal enclosure, is potted in cured epoxy. An accessible outside surface of the enclosure has a test terminal board. By removing a watertight cover over the board the condition of rectifier cells, fuses, and resistors is readily determined by electrical tests at scheduled maintenance periods.

Magnetic Cores and Windings

The rotor core is split into two segments and the stator core in three to facilitate assembly on the ship. Cores are protected with several layers of cured epoxy, identical to the treatment given the propulsion motors. The insulation system is also identical to that described for the propulsion motors.

Alternate Transformer Configuration

A transformer with cross section as shown in Figure 27 was also studied for this application. The three phases of both the stator and rotor are wound around the large teeth. This machine requires considerably more stacking length and increased weight to obtain the low reactance of the wound rotor induction machine arrangement with approximately the same exciting current. It poses the problem of winding, insulating, and installing the large coils, but once installed they are almost completely enclosed by the punchings. This machine is a true transformer in the sense that its output voltage is independent of rotor speed. It is feasible to build a machine of this design, but the wound rotor induction machine seems more attractive, particularly because of its electrical characteristics.

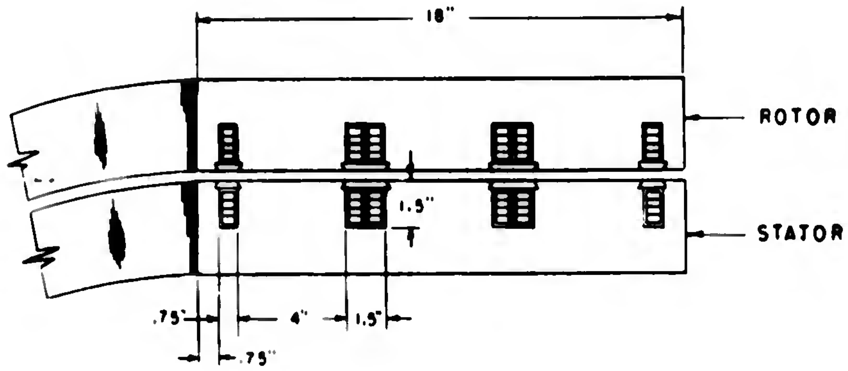


Figure 27. GENERAL ELECTRIC Alternate Rotating Transformer Configuration

INFORMATION TRANSFER TO THE ROTOR

The initial conceptual design study, presented in summary form in a later part of this section, resulted in a large number of possible systems for transfer to the rotor of the information required for blade control. From this list, a single scheme was selected for further, more detailed investigation. The selection of this particular system was based on the best combination of simplicity, reliability, inherent accuracy, availability of technology, and flexibility.

Figure 28 shows in schematic form the pitch changing system, based on a squirrel cage induction motor actuator for each blade. The motor voltage is varied using silicon controlled rectifiers, control signals are transferred to the propeller hub through transformer-like units, and electric power is transferred to the hub by a rotating transformer. The schematic in Figure 29 is a more detailed version of the system in Figure 28. The salient features of the system are:

All controls use static elements.

The blade drive package is oil filled and pressure biased 1/2 psi above ambient pressure. However, there is nothing in the system that would prohibit temporary operation in sea water if the control components were potted against exposure.

The control signals are frequency modulated in proportion to the pitch angle they are to represent. The basic frequencies are in the audio frequency range. Frequency modulation provides a system with excellent inherent accuracy, since frequency is unaffected in the transfer to the hub.

The desired blade position signals are commutated magnetically to the hub, and it is possible to operate with non-sinusoidal pitch programs. The smoothness of such a program is determined by the number of transmitter elements on the hull, as well as by the inertia of the blade motor and reduction gear.

Abbreviated Specifications

The following criteria were used in the preliminary design of the blade control system and the individual pitch-changing devices:

Minimum maintenance interval	One year
Hub rotational speed	50 rpm maximum

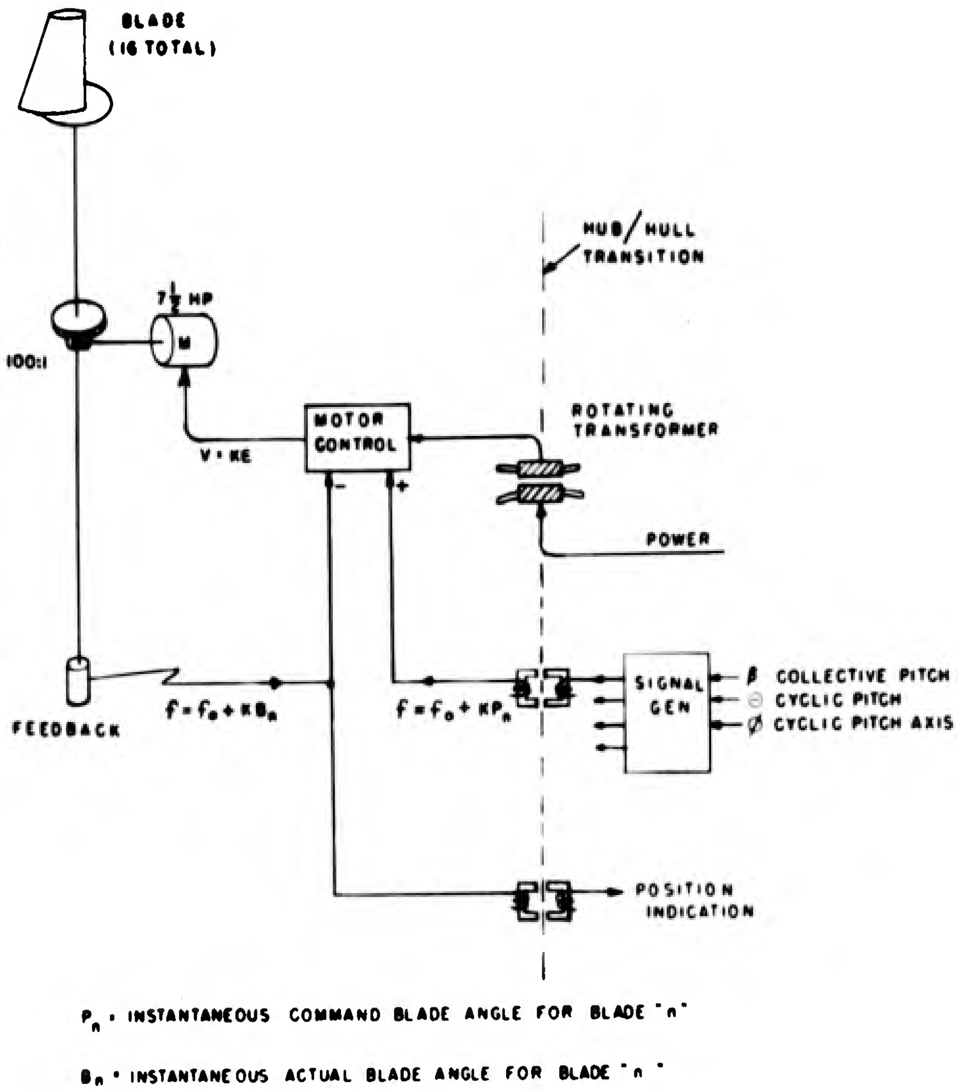


Figure 28. GENERAL ELECTRIC Pitch Changing System Overall Schematic

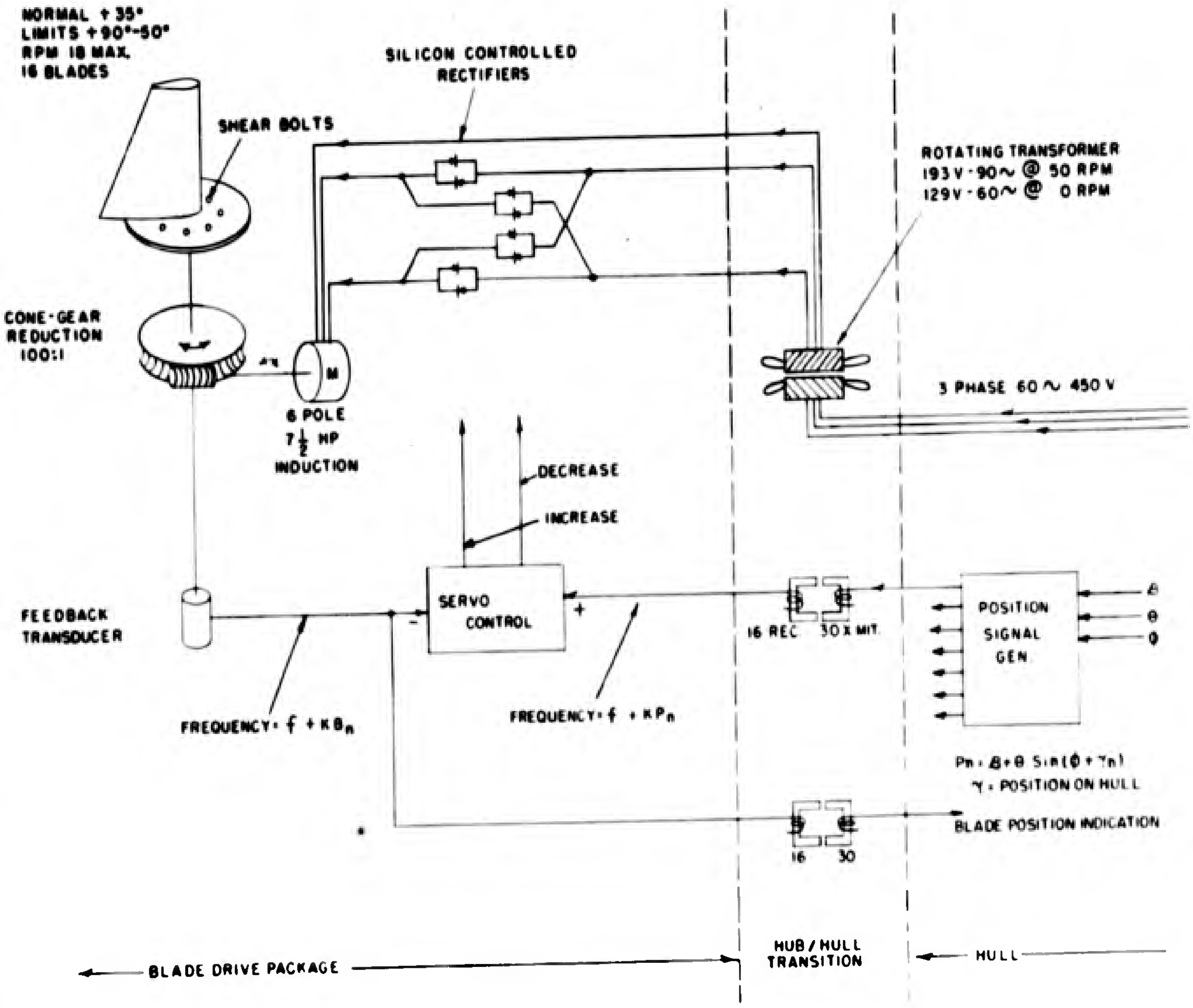


Figure 29. GENERAL ELECTRIC Pitch Changing System Detailed Schematic

Number of propeller blades	16 per hub
Blade control range	+90° to -50°
Design ahead pitch	+35°
Collective pitch accuracy	± 1°
Maximum cyclic pitch	± 20°
Cyclic pitch accuracy	± 2°
Cyclic pitch phase accuracy	± 10°
Maximum torque on blade	± 7000 lb-ft
Maximum normal torque during pitch change	± 1500 lb-ft

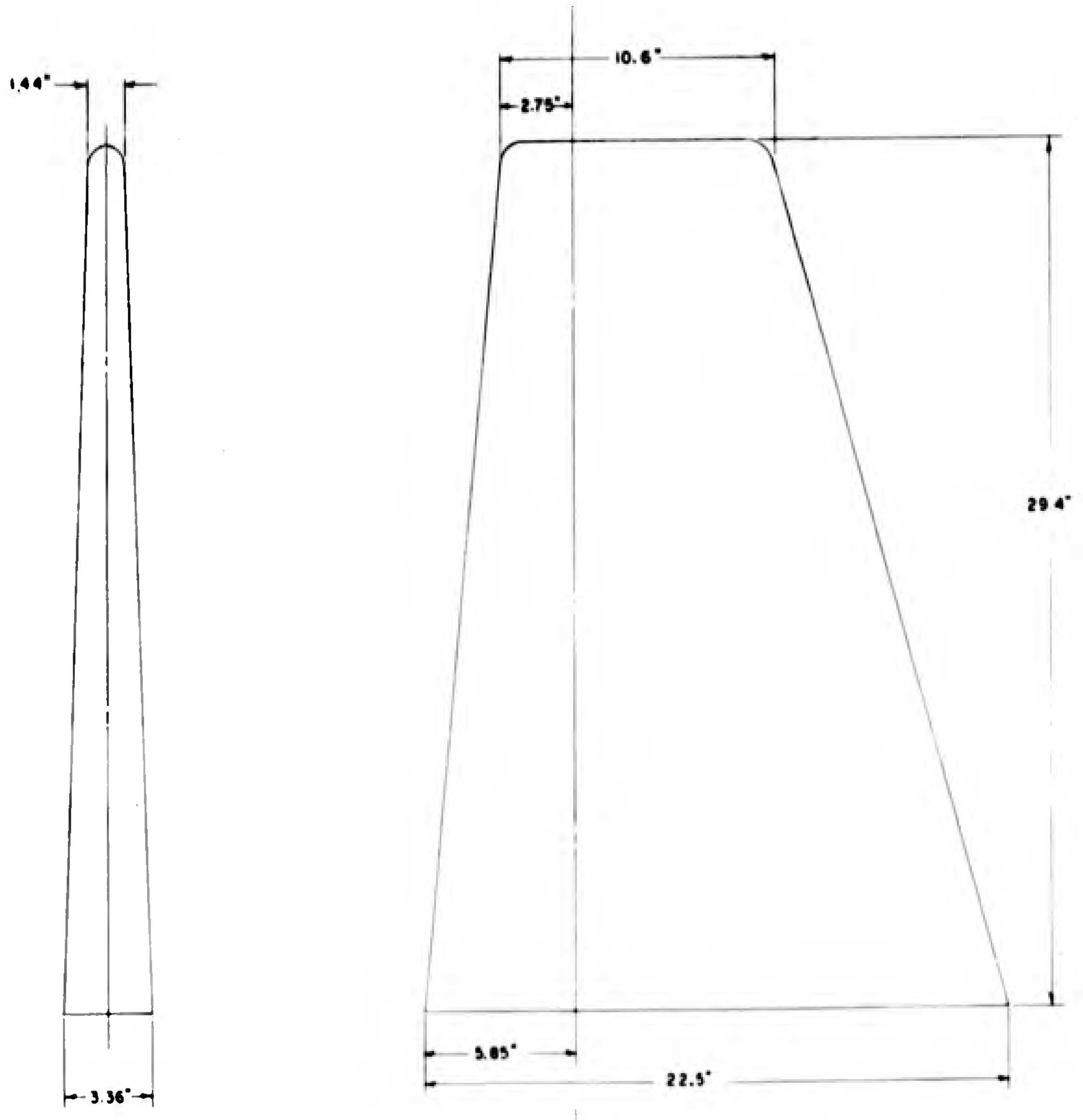
The system is to be inherently unaffected by pressure changes. Blade dimensions assumed are shown in Figure 30.

Control Circuitry

All of the control signals are frequency modulated with the base frequency in the audio frequency range. Signal transmission from the stationary hull to the hub is accomplished by transmitters and receivers which together form the equivalent of a primary and a secondary of a transformer with an air gap in the magnetic circuit. The frequency of the secondary signal is not affected by the factors which affect the voltage of the secondary signal.

Inside the hull, the control signals from the ship control system are converted into frequency modulated signals to feed the stationary magnetic transmitters. These transmitters are located around the hull, in sea water, adjacent to the propeller hub. The number of these transmitters is determined by the degree of refinement desired in the blade pitch program. Thirty transmitters should be adequate.

The receivers (one for each blade) pass by the transmitters and receive the position signal appropriate for the blade at that phase position around the hull. This signal is then the input signal to a servo loop consisting of the motor control, motor, reduction gear, and feedback transducer.



ZONE	BASE THICKNESS
0	0
1	2.500
2	3.250
3	3.375
4	3.250
5	3.000
6	2.625
7	2.125
8	1.500
9	0.875
10	0

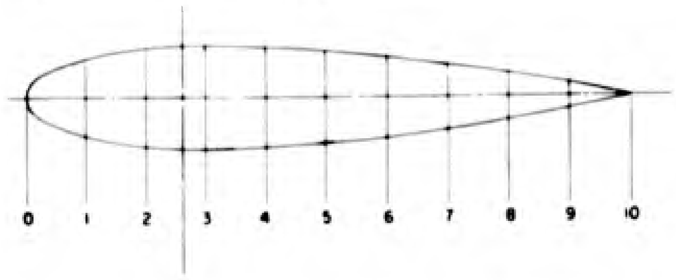


Figure 30. GENERAL ELECTRIC Assumed Propeller Blade Dimensions

The blade must be controllable with the propeller hub stationary. This means that the transmitters must span the arc over which they control so that the hub can never stop with the receivers out of correspondence with the transmitters. Normal cruise operation of the submarine does not require more power than the capacity of either the forward or aft propeller, so that if ship control is adequate the other propeller may be kept stationary, with the blades feathered for minimum drag and flow noise.

The diagrams have shown separate units to feed back position information into the hull, in order to keep the signals indicating completion of action independent of those initiating the action. There are, however, other approaches which might be considered when more detailed hardware investigations are made. The control signal transmitters and receivers can be used simultaneously for control and indication, with the signals far enough apart in frequency to be separated by filtering. Another alternative is to add supplementary windings on the propulsion motor or on the rotating transformer, and use these windings to convey control and/or indication signals in the same way as the transmitters and receivers outlined above. While this would simplify the physical construction of the information transfer apparatus, it would considerably complicate the main propulsion machinery.

Design and Selection of Components

Technology already exists for the design and selection of most of the elements of the blade control system. It was felt, therefore, that detailed design of the information transmission circuitry would add little or nothing to the determination of feasibility of the overall system, although such detailed design would be essential as part of a more complete investigation.

An analogue computer study of the complete control system would be invaluable in selecting and sizing components, investigating system response, and establishing conformance with specifications. This study would have the added advantage of demonstrating the flexibility of the electrical control system. In addition, a breadboard model of the system, including a full-power blade drive package and transmission of control information through sea water, would help to demonstrate the characteristics and feasibility of the system and would furnish a check on the adequacy of the various components selected.

A detailed design was not made of the information transmitters and receivers, since no major problems are foreseen in establishing an adequate design. The elements would be essentially segmental E-frame transformers, wound and potted to provide adequate corrosion protection, and mounted circumferentially around the hull. The problems involved in sealing these elements, and in providing satisfactory hull penetrations for the lead wires, are much simpler than those associated with the main propulsion motors, and no difficulties are anticipated.

BLADE ACTUATING DEVICE

Considerable attention was devoted, during the latter part of the study, to a more detailed design of the blade actuating packages. This was deemed to be the most critical area in the blade control system, particularly in view of the environmental requirements.

The electric motor pitch changing system lends itself to the construction of a packaged unit for each propeller blade. This provides an exceptional advantage from an operations standpoint, since these packages may be individually removed without drydocking for such reasons as inspection, replacement, or preventive maintenance. With appropriate spares, such operations would minimize out-of-service time on the submarine. Reliability is also enhanced by the multiple package approach, since the loss of one unit does not render the propeller inoperative. The shear bolts holding the propeller blade to the drive package prevent any damage to the package resulting from the blade striking an object. The blades are also replaceable, possibly even at sea.

Layout of the Actuator Package

Details of the actuator package are shown on Figures 31, 32, 33 and

34. The general envelope of the package is such as to house the motor and the worm reduction. To this have been added two cylindrical volumes to house control components (shown along the vertical centerline of Figure 33). These openings are also necessary to provide access to the worm reduction during original alignment of the two gears.

Operating the actuator oil filled is not considered a problem, since the system is very similar to commercial submersible pump motors built by the General Electric Company. In fact, the duty on the actuator at 18 rpm maximum is less severe than the 3600 rpm, 10 year, continuous duty for the submersible pump motor.

Two rotating seal units are used. The outer unit keeps contamination out of the seal area, and the inner unit actually provides the sealing. It is felt that an elastomeric chevron type of seal would be adequate. An interesting inovation would be the use of an inert ceramic material for the separable sealing element, provided that shock would do it no damage.

The "oil" in the drive package is actually polyalkylene glycol, which is miscible with sea water and would thus not disclose the position of the ship if the drive package were ruptured by combat damage.

Sleeve bearings have been used on the motor-worm shaft, since Navy experience has been that they are inherently more quiet on faster shafts than anti-friction bearings.

Pressure equalization is provided by the piston-springs-Bellofram assembly shown at the bottom of Figure 34. This is best located at the innermost radius when the drive package is mounted in the propeller hub. At 50 rpm there is a nine "g" field produced, and the mass of the piston will then add to the pressure bias inside the drive package.

Motor to Blade Gear Reduction

It is desirable that a reversible reduction be used between the blade motor and the blade. Consequently, with the propeller blade pivoted ahead of the center of pressure, the blade is permitted to align itself

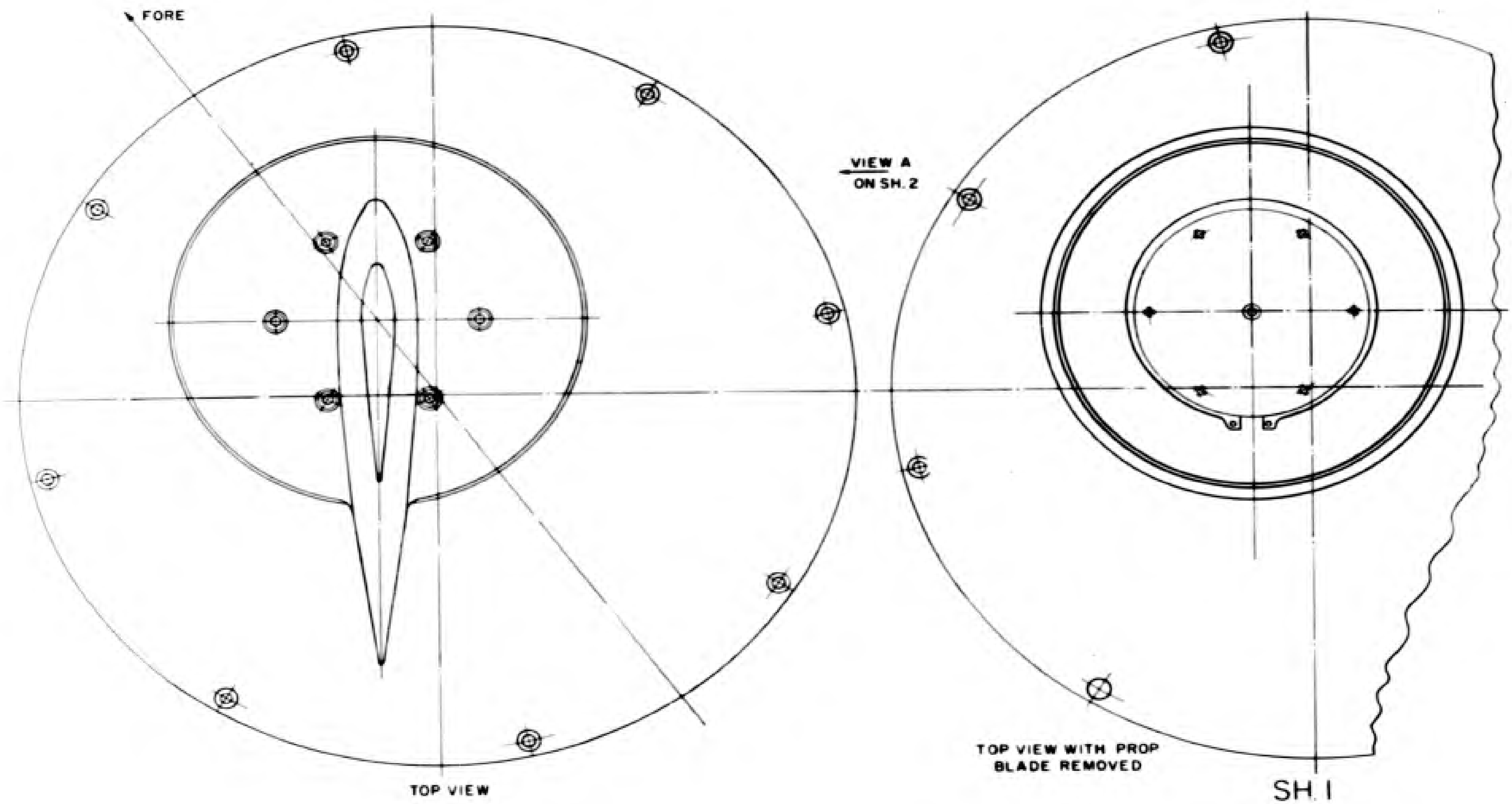


Figure 31. GENERAL ELECTRIC Propeller Blade and Actuator, Plan View

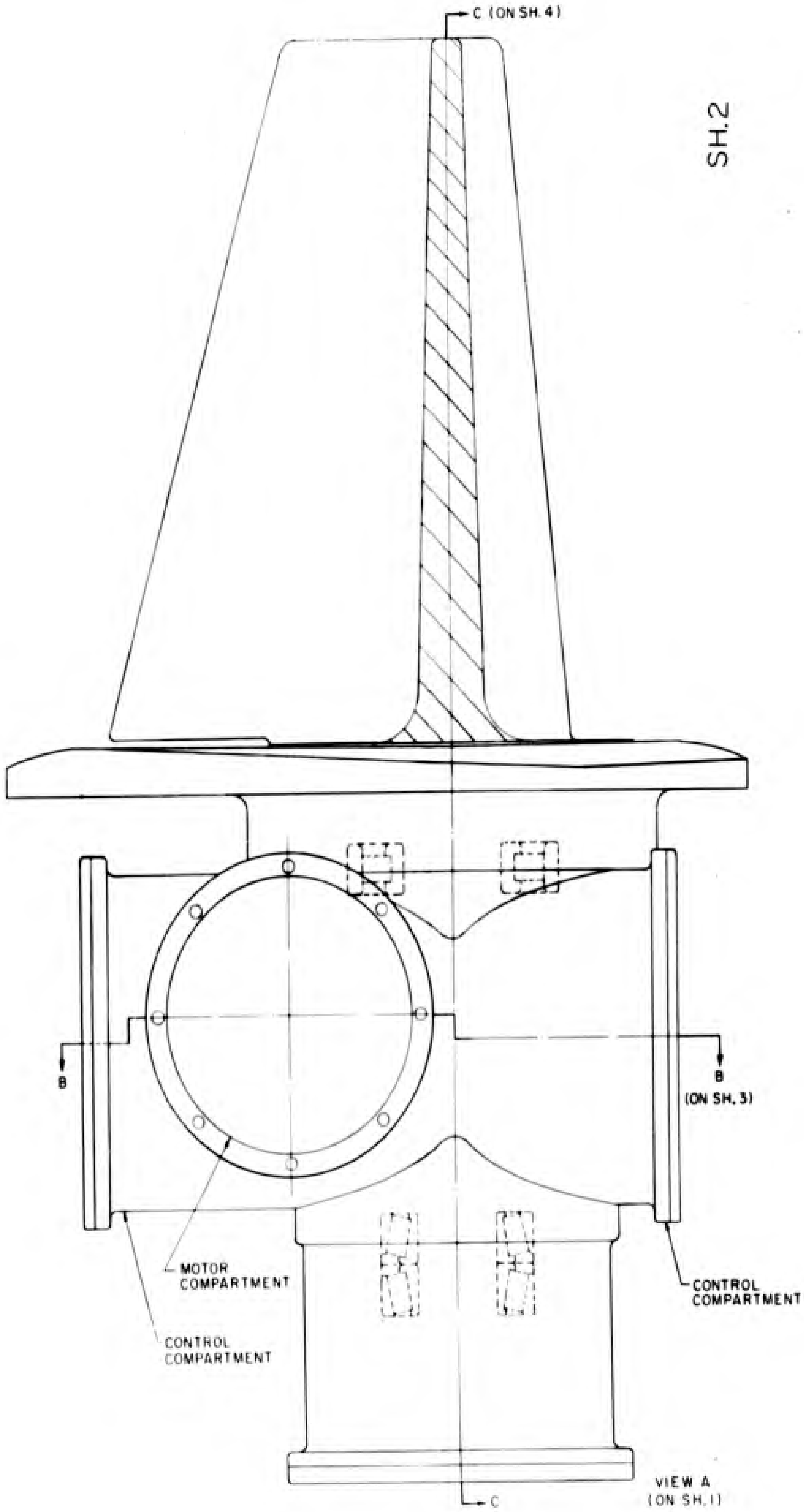
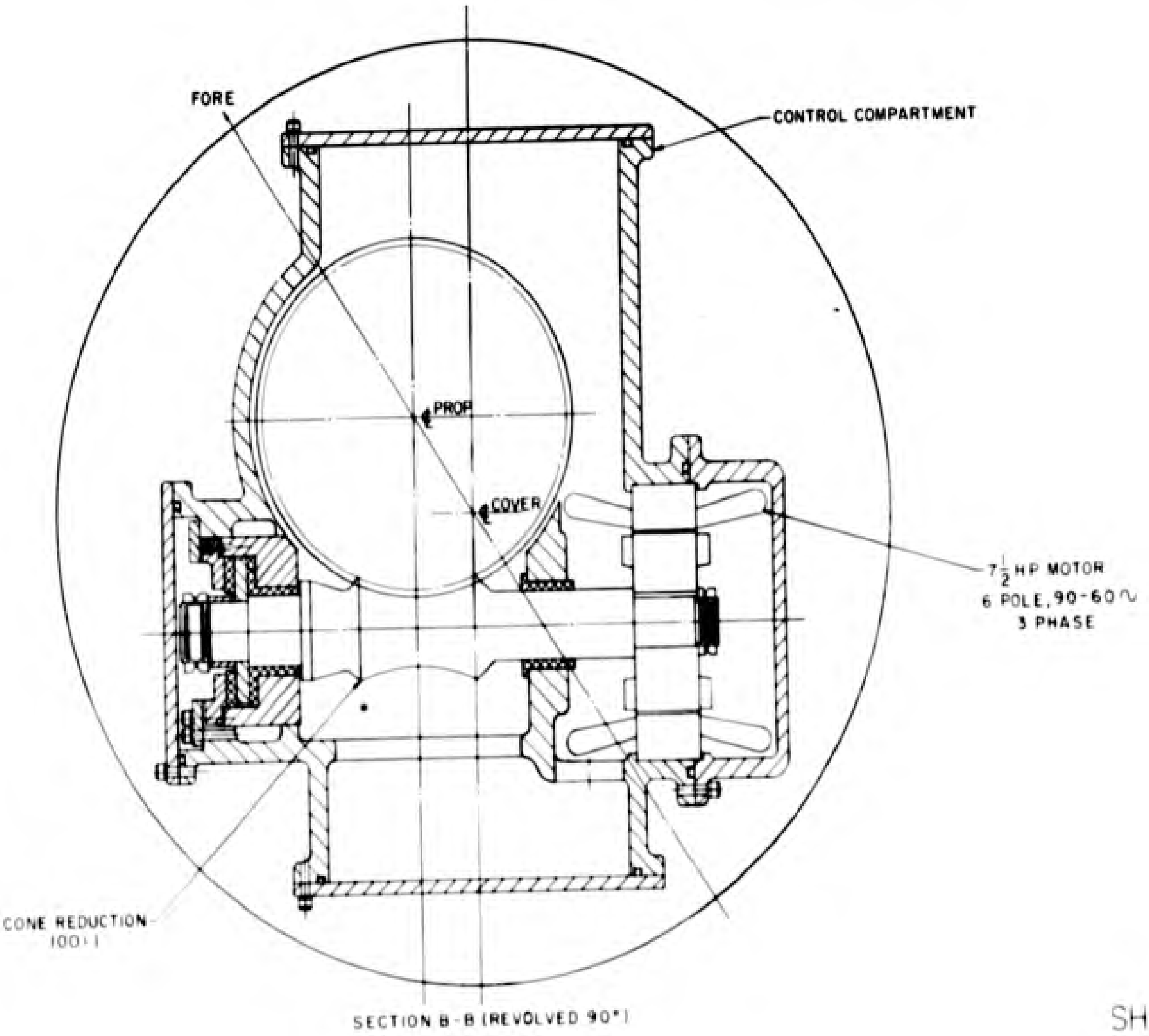


Figure 32. GENERAL ELECTRIC Propeller Blade and Actuator, Elevation



SH 3

Figure 33. GENERAL ELECTRIC Propeller Blade Actuator, Plan Section

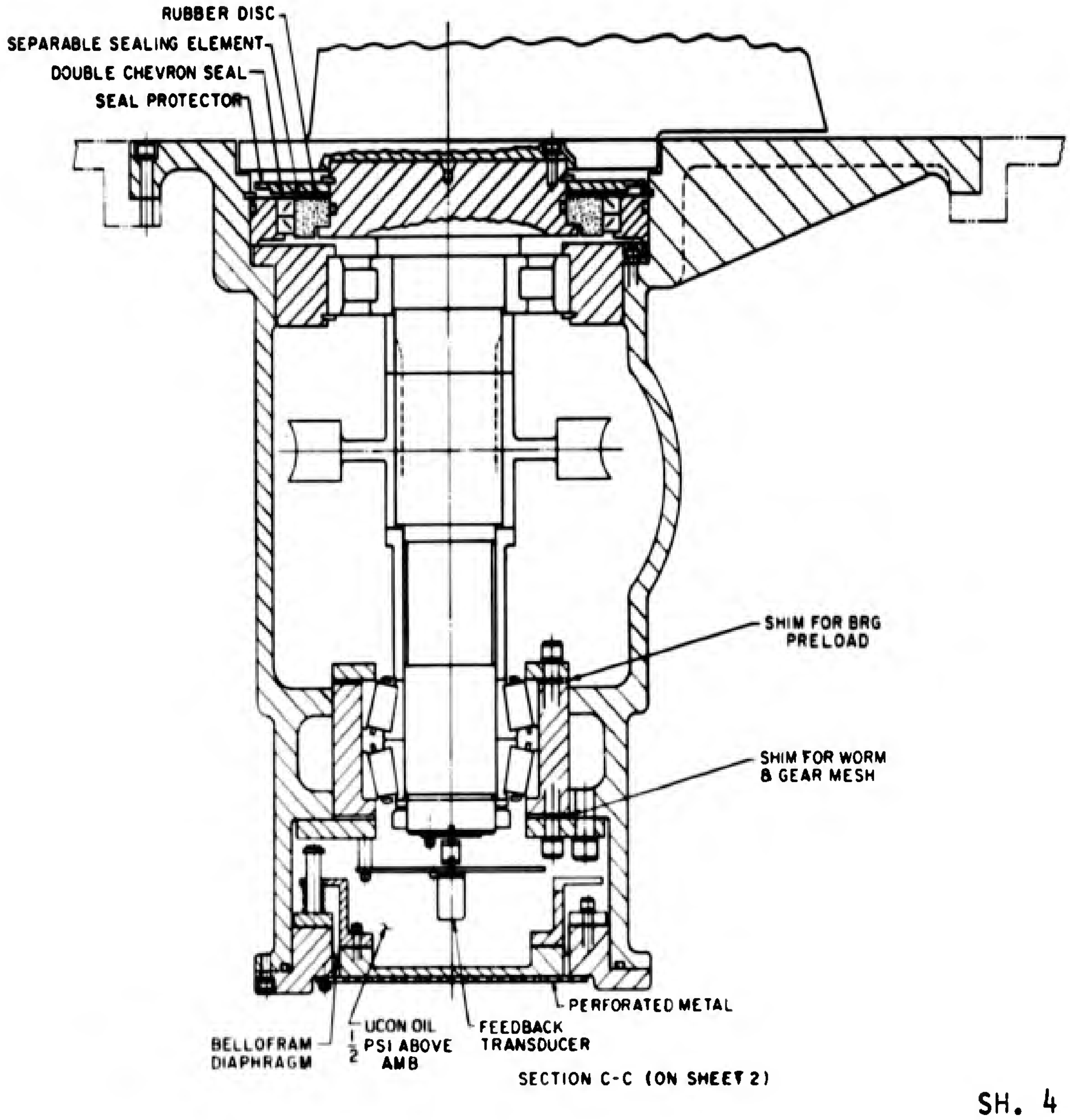


Figure 34. GENERAL ELECTRIC Propeller Blade Actuator, Elevation Section

with the flow upon loss of power to the blade motor. In addition, the reversible reduction and motor peak torque capability combine to provide an effective torque limit.

Theoretically, the dividing line between reversible reductions and non-reversible reductions is at 50% efficiency. Above 50% efficiency a reduction is reversible. For this reason, a double-enveloping worm reduction was chosen as having the highest efficiency of the worm reduction types. This reduction bears the trade name "Cone-Drive" and is manufactured by the Cone-Drive Gears Division of Michigan Tool Company, Detroit, Michigan.

To keep the fluid friction losses down, an 1800 rpm motor was selected in preference to a 3600 rpm motor. This then required a 100:1 gear reduction. To handle the specified maximum torque without damage, a reduction having a 10" center distance was selected (3.75" pitch diameter worm and 16.28" pitch diameter gear). The expected efficiency of this reduction running in antifriction bearings is 70 to 75% at 1800 rpm on the worm, and 55 to 60% at 10 rpm on the worm. The efficiency of the reduction could be increased by resorting to a 1200 rpm motor.

Blade Drive Motor

A standard three-phase squirrel cage motor was selected for the blade drive motor because of its simplicity and reversibility. Based on preliminary data on torque requirements, the rating chosen was 7-1/2 horsepower. This will be subject to change, of course, as hydrodynamic data are refined.

The motor is actually used in a very normal servo motor mode. However, when viewed from an industrial motor designer's standpoint, it is subject to the rather heavy duty of two starts a second and no significant time at full speed. The problem is one of heat dissipation, and for this reason the stator is mounted directly in a part of the case that is exposed on the opposite side to sea water. In the final design, a thorough heat transfer analysis will be essential.

The original calculations, on which the motor selection was based, were made assuming a 4 pole 60 cycle motor. Later, as the design of the rotating transformer progressed, it became apparent that operation at 90 cycles at full propeller hub speed would be required. The motor was then changed to a 6 pole machine which would have comparable performance at 90 cycles. This change has not been reflected in the blade actuator layouts, which still include the 4 pole motors. However, the dimensional changes are small and do not affect conclusions regarding feasibility.

A high starting torque is advantageous in a motor with a duty cycle such as this. A 5 to 8% slip motor (Type KR) has an excellent speed-torque curve for this purpose, as illustrated in Figure 35.

DEVELOPMENT AND CONCEPTUAL DESIGN OF ELECTRICAL SYSTEM

The basic concepts involved in the design of a propulsion motor for operation totally immersed in sea water were investigated in detail in connection with the NEPS study. It seemed appropriate, therefore, to concentrate on the conceptual design of the pitch changing mechanism during the early stages of the present study.

As a first step, a brainstorming effort was undertaken, with no restrictions imposed on methods or types of hardware. The approach taken was to determine the various component functions required by a blade positioning system, and then to explore the methods and techniques available to produce these functions.

The systems studied involved mechanical, hydraulic, and electrical components, in various combinations. These were reviewed in the course of several General Electric-Electric Boat meetings, and it was decided that General Electric would devote particular attention to electrical schemes. This part of this section summarizes the work performed during the initial conceptual design phase of the study.

The most stringent requirement of the blade control system is its compatibility with sea water under considerable pressure. In this respect, the systems shown have the conventional problems associated with operating

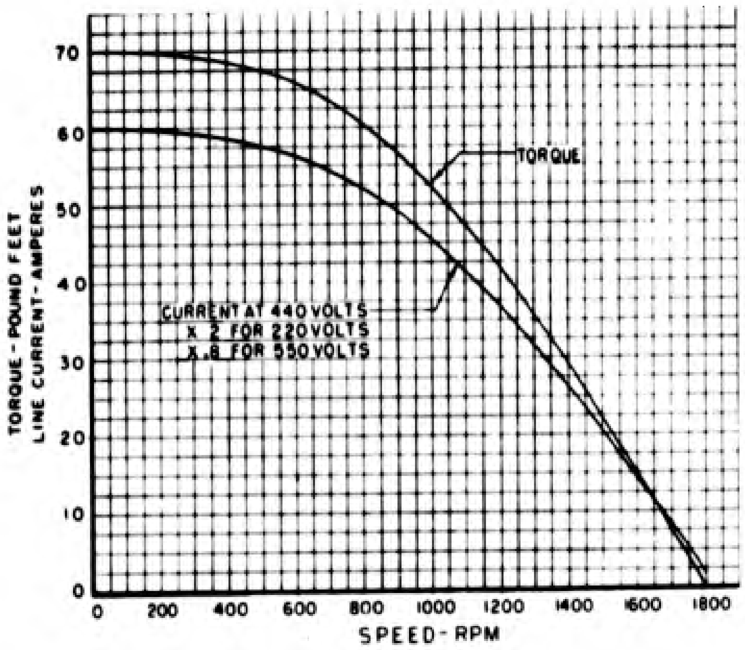


Figure 35. GENERAL ELECTRIC Typical Propeller Blade Actuator Motor Torque and Current vs. Speed Curves

flooded, operating in oil with pressure equalization and shaft seals, and encapsulation. Freedom from detectable noise generation is the second most stringent requirement. Along these lines, an attempt has been made to present components that have the greatest potential for being quiet.

Besides ability to operate quietly in the sea water environment, performing according to the demands in accuracy and time, the system must have an exceptionally high degree of reliability over periods of at least a year between overhauls.

Conclusions

The following general conclusions were reached, and these formed the basis for the preliminary design work which followed:

> Signal generation will be accomplished within the hull to the maximum practical extent, and static control hardware will be used.
>
> Information will be transferred to the rotor by frequency modulated radio frequency propagation, using multiple antennas distributed peripherally. (This was later changed to frequency modulated audio frequency magnetic coupling.)
>
> Power will be transferred to the rotor by a transformer with stationary primary and rotating secondary windings.
>
> Each blade will be driven by a separate electric motor located in the propeller hub.

Within each of these areas there are several alternate approaches, which are discussed subsequently in this section. Further resolution took place in the course of the preliminary design.

Pitch Changing Systems

The general configuration of the propulsion system relative to the ship, and the more detailed requirements for the pitch changing equipment are discussed in a previous section of this report. However, a few aspects to be considered in developing and evaluating a system are listed below.

> Compatibility with sea water under pressure.

Minimum radiation to the sea providing means of detection or tracking of the submarine.

Maximum overall reliability.

Minimum complexity.

Minimum size and weight.

Good system flexibility commensurate with a first-of-its-kind system.

Torque limiting to eliminate overload damage to the drive.

Enhanced reliability by keeping the maximum amount of the system inboard.

Accuracy at least as high as called for in the specification.

Greater accuracy in collective pitch than cyclic pitch.

Maintenance free operation for one year.

Figure 36 is a composite pitch changing system block diagram for general orientation in the discussion which follows. Numbers adjacent to the various blocks correspond to figure numbers in this part of the report which elaborate on the contents of the respective blocks. These figures all indicate the use of individual control on the sixteen blades, but it may ultimately be practical to couple two adjacent blades to have the same motion and thereby decrease the number of components.

Signal Generation and Transmission

Individual blade positions are composed of two components: Collective pitch β, and cyclic pitch Θ at a phase relationship ϕ to the hull. Assuming for illustration purposes a sinusoidal pitch waveform, the position of the Nth blade of a 16 blade propeller has the form:

$$P_N = \beta + \Theta \sin \left(\gamma - \phi + N\frac{2\pi}{16} \right),$$

where γ is the instantaneous hub position with respect to the hull.

Note first that it is very difficult electrically to generate the phase differences ($N\, 2\pi/16$) between the individual blade control signals. This is so because it is a position difference and electronic equipment can only generate time phase displacements. If the propellers were operated at constant speed, the position differences could be directly derived

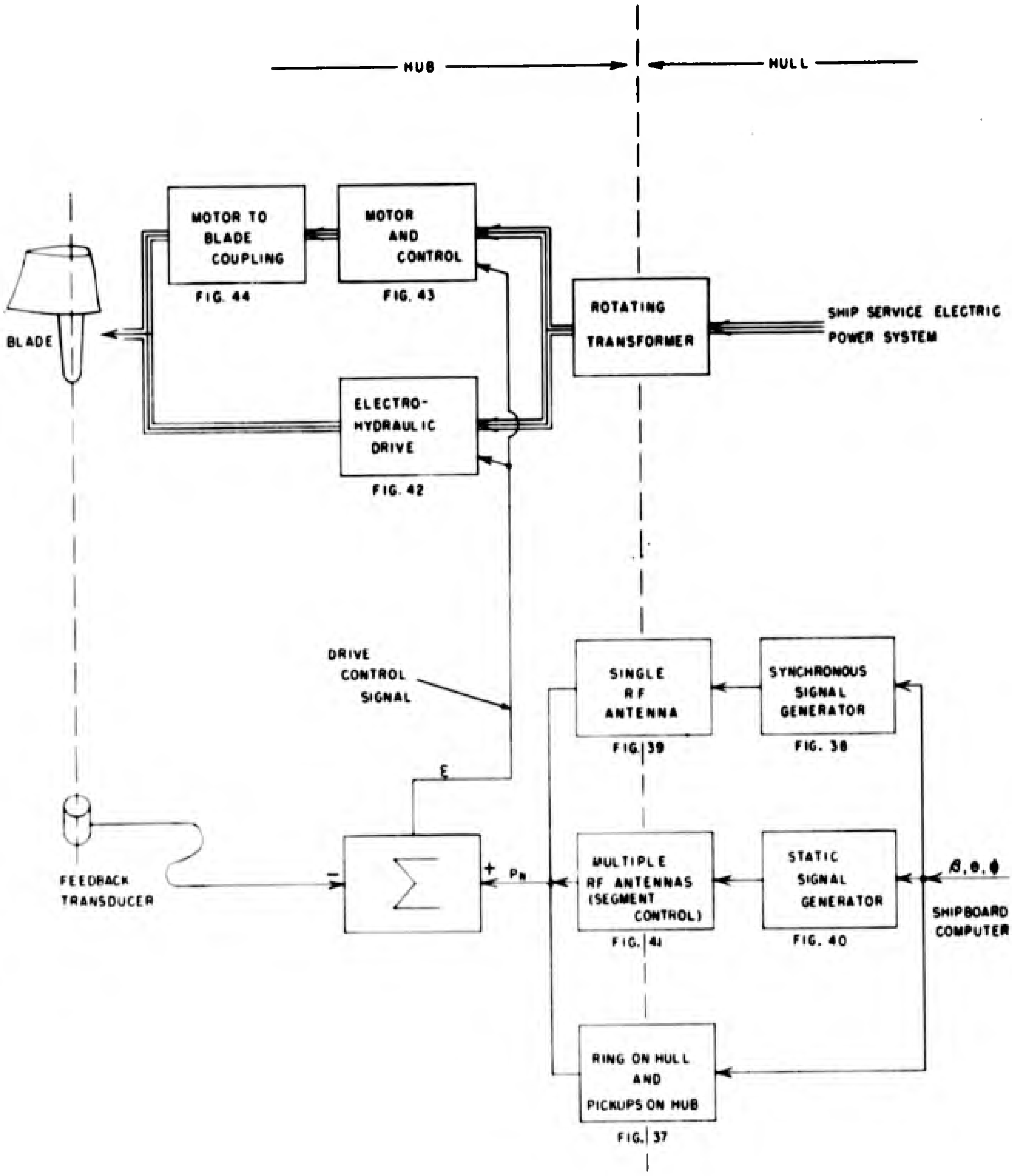


Figure 36. GENERAL ELECTRIC Pitch Changing System Composite Block Diagram

electronically from time differences, but for varying speeds this becomes practically impossible. The basic difference in the three methods shown for transferring blade position signals to the hub is in the means of obtaining this phase difference ($N\ 2\pi/16$).

> Figure 37 shows the use of a ring similar to a wobble plate mounted on the hull with non-contacting pickups mounted on the hub. This scheme has the disadvantage of placing more of the system in sea water, with consequent inaccessibility and the possibility of contamination. However, if the problems associated with the sea water are solved for the blade drive devices the same solutions would apply to the pickups and the ring positioning equipment.
>
> Figure 38 shows the signals generated in a similar way as Figure 37, except that it is done with a model hub inside the hull. This requires the model to run synchronously with the hub, which is difficult unless the hub drive motor is synchronous, and in either event adds complexity and continuously moving parts. The generated signals are transmitted to the hub on a radio frequency carrier, as shown in Figure 39.
>
> Figures 40 and 41 show an approach having both the advantage of generating the signals inside the hull as in Figure 38, and the advantage of being self synchronizing as in Figure 37. The compromise here is that the blade pitch would not be uniformly changing but would approximate a series of step functions. This could possibly be an asset if it should be desirable to alter the blade pitch program, as in Figure 40-A, to be a two position control. Again, the signals are transmitted to the hub on an RF carrier.
>
> A modification of this would be a combination of single and multiple RF antennas wherein collective pitch is transmitted to the hub for all blades on a single antenna as in Figure 39, with the cyclic pitch information transmitted on multiple antennas as in Figure 41. It is quite possible this could provide the system with the best inherent accuracy since the cyclic pitch system would not affect the collective pitch accuracy. However, it is anticipated that even for steady sailing, minor, but almost continuous, directional control will be required, and thus the cyclic pitch signal will be non-zero for most of the time.

Signal transmission to the hub at radio frequency has the following advantageous features: It is noiseless, is easily shielded, requires only the simplest of antennas, and requires no critical mechanical alignment; a frequency modulated signal has great inherent accuracy and can withstand considerable abuse without loss or change of information.

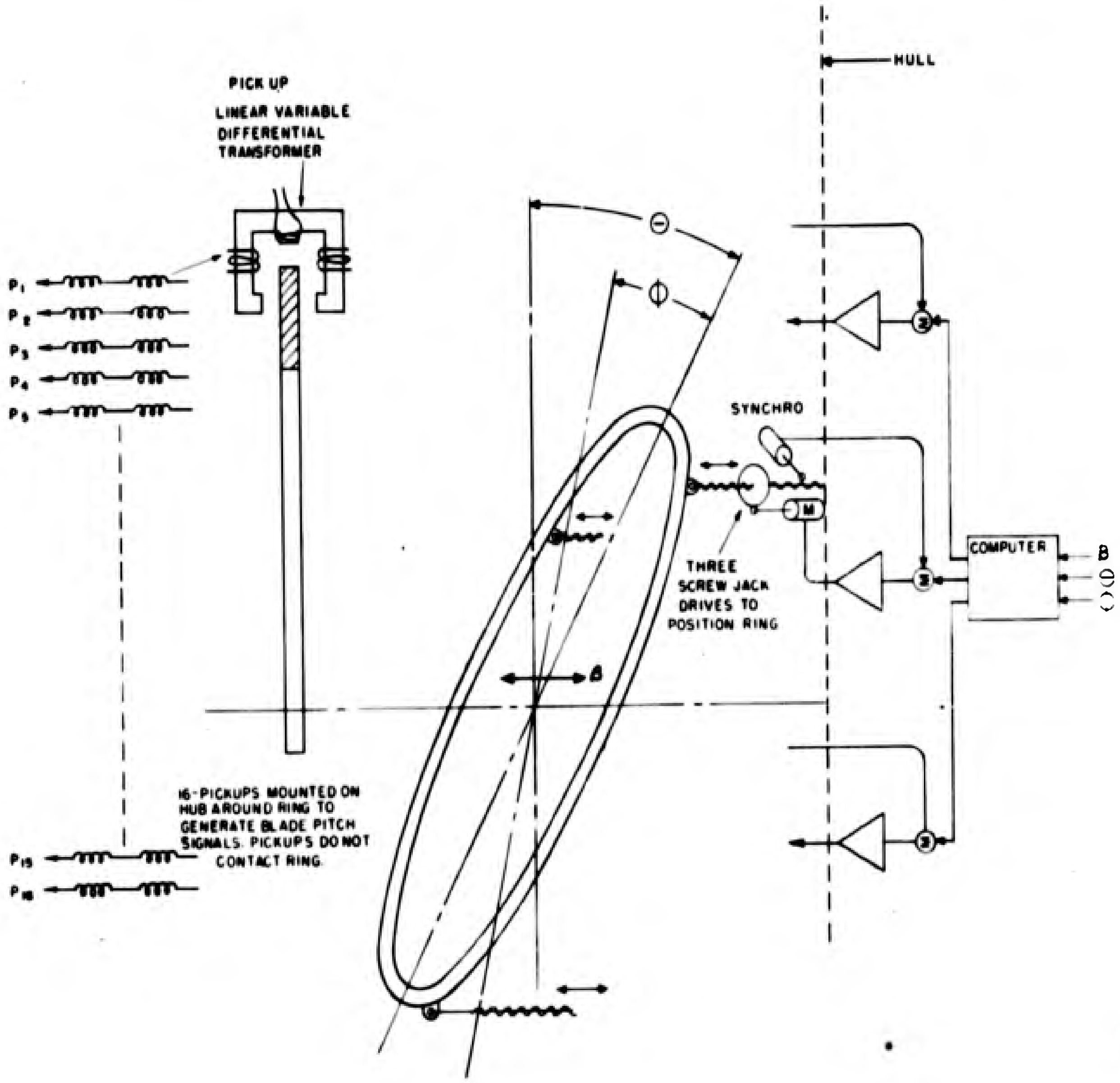
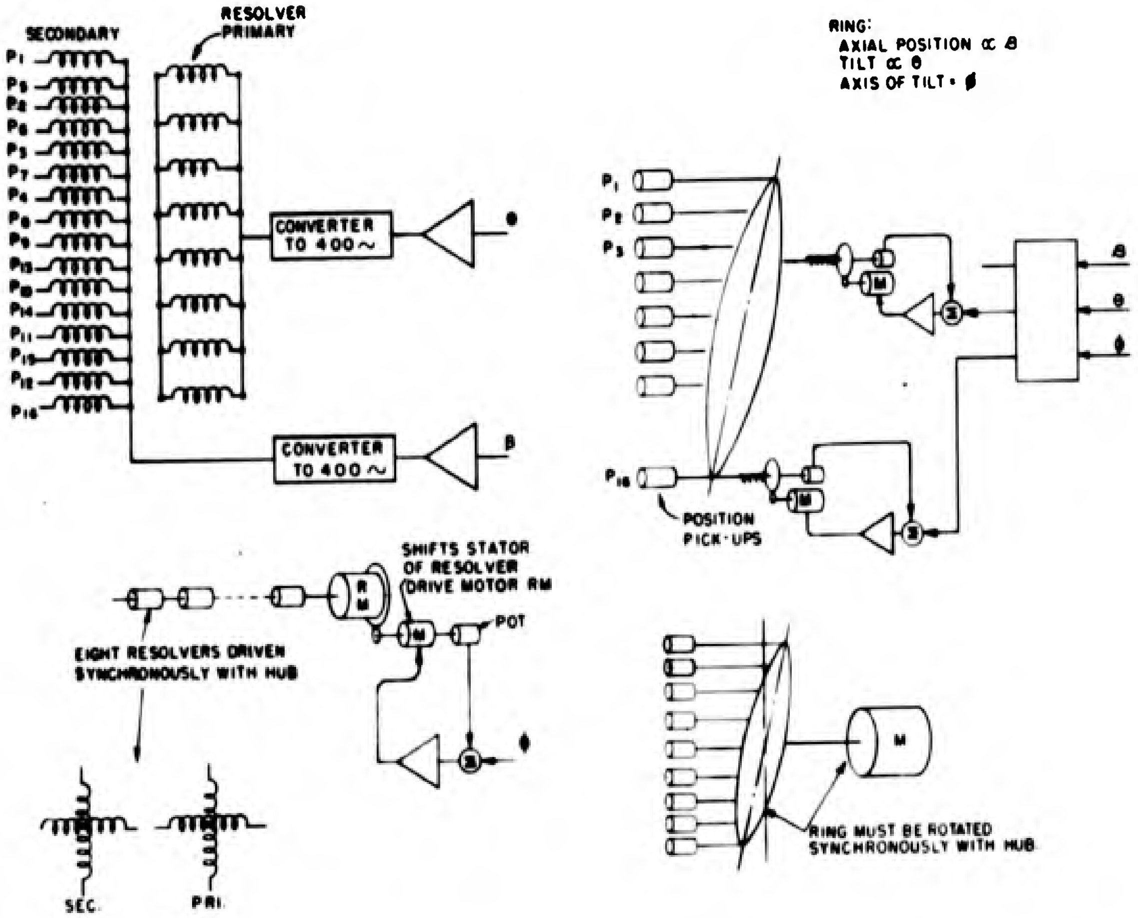


Figure 37. GENERAL ELECTRIC Signal Transfer from Hull to Hub via Ring on Hull and Pickups on Hub



A. CYCLIC PITCH GENERATED SEPARATELY.

B. CYCLIC AND COLLECTIVE PITCH GENERATED COMBINED.

Figure 38. GENERAL ELECTRIC Synchronous Signal Generation (Signals change once per revolution of hub)

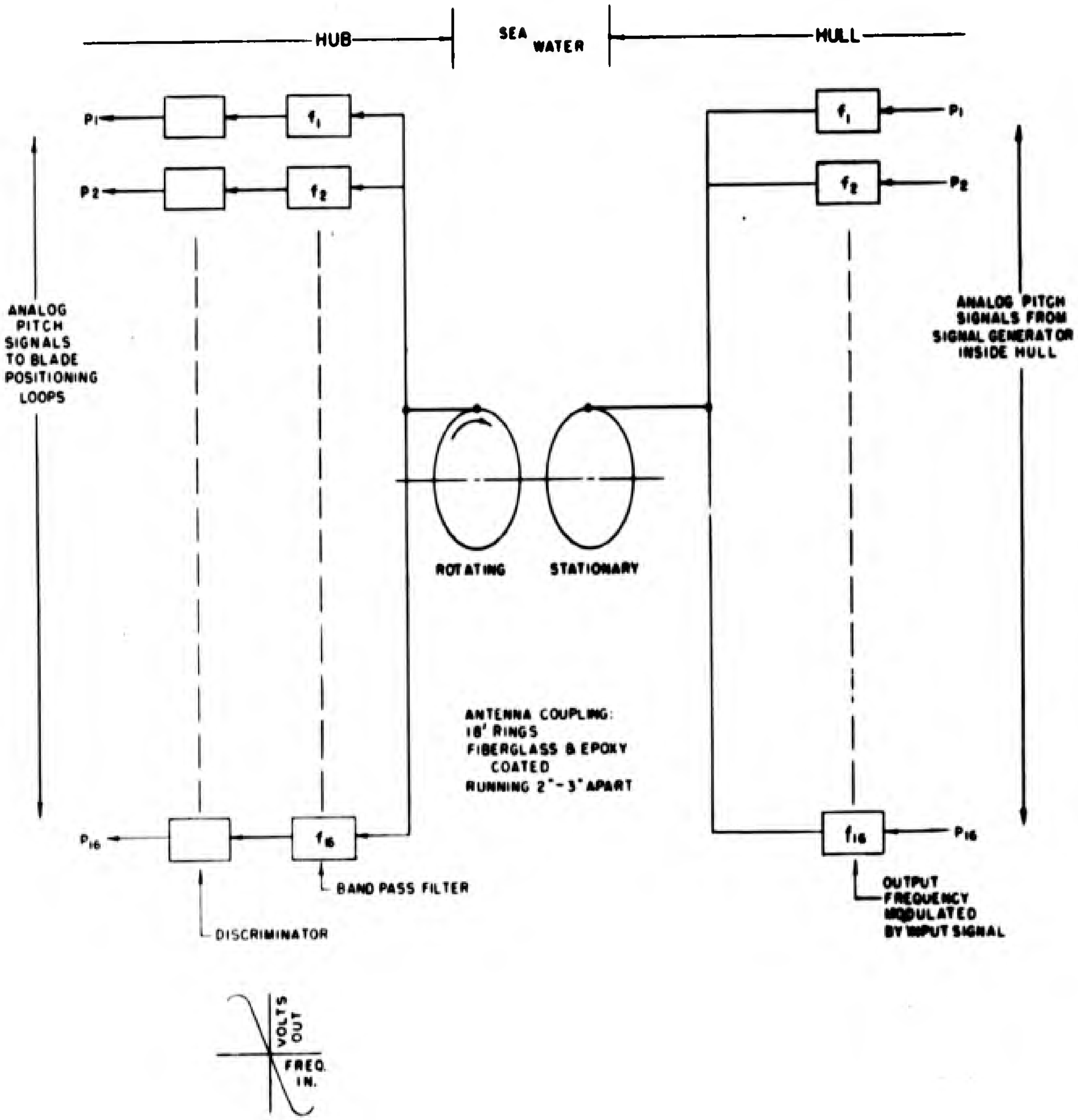


Figure 39. GENERAL ELECTRIC Signal Transfer from Hull to Hub via Single RF Antenna

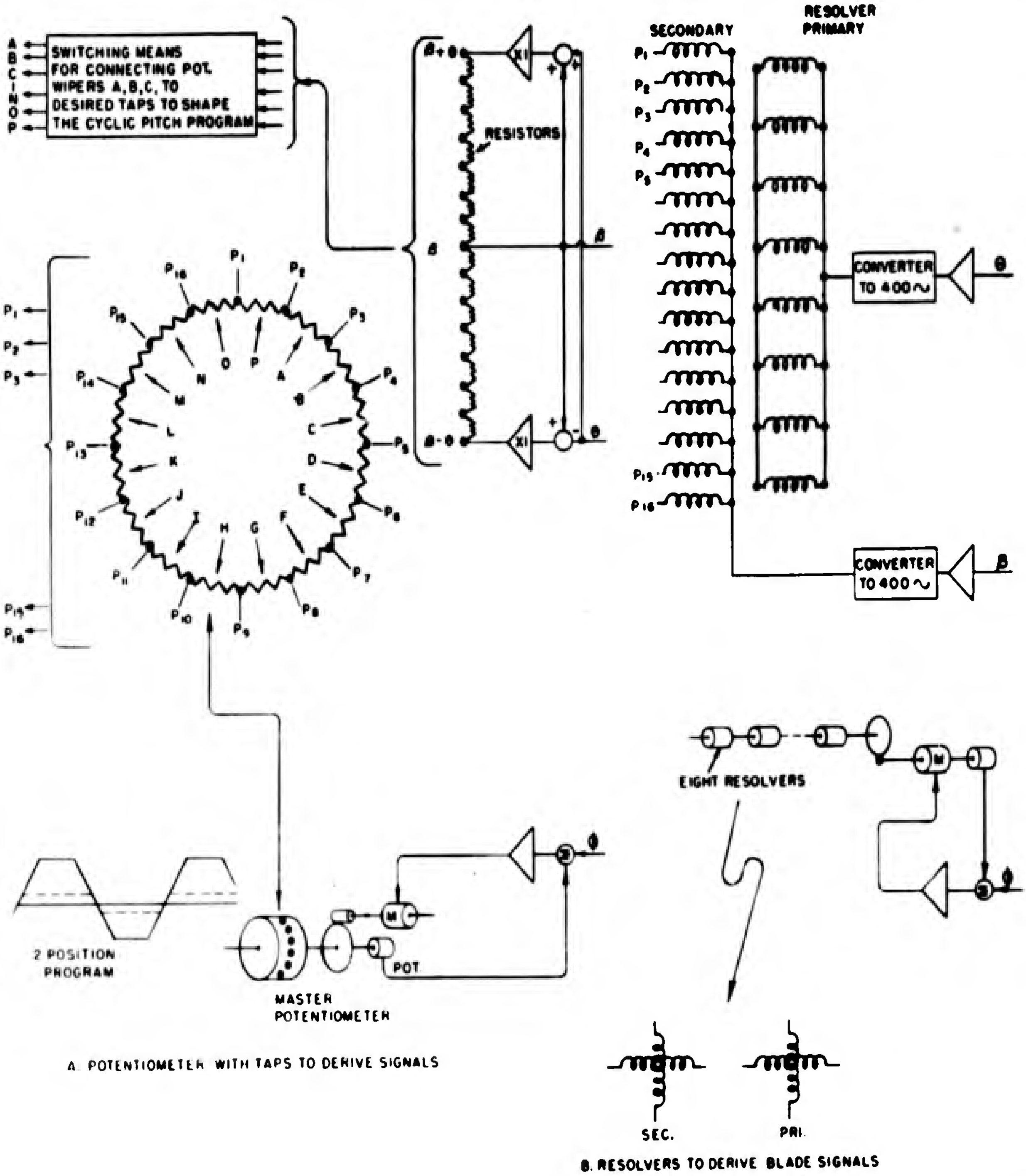


Figure 40. GENERAL ELECTRIC Static Signal Generation (Signals do not change once per revolution of hub)

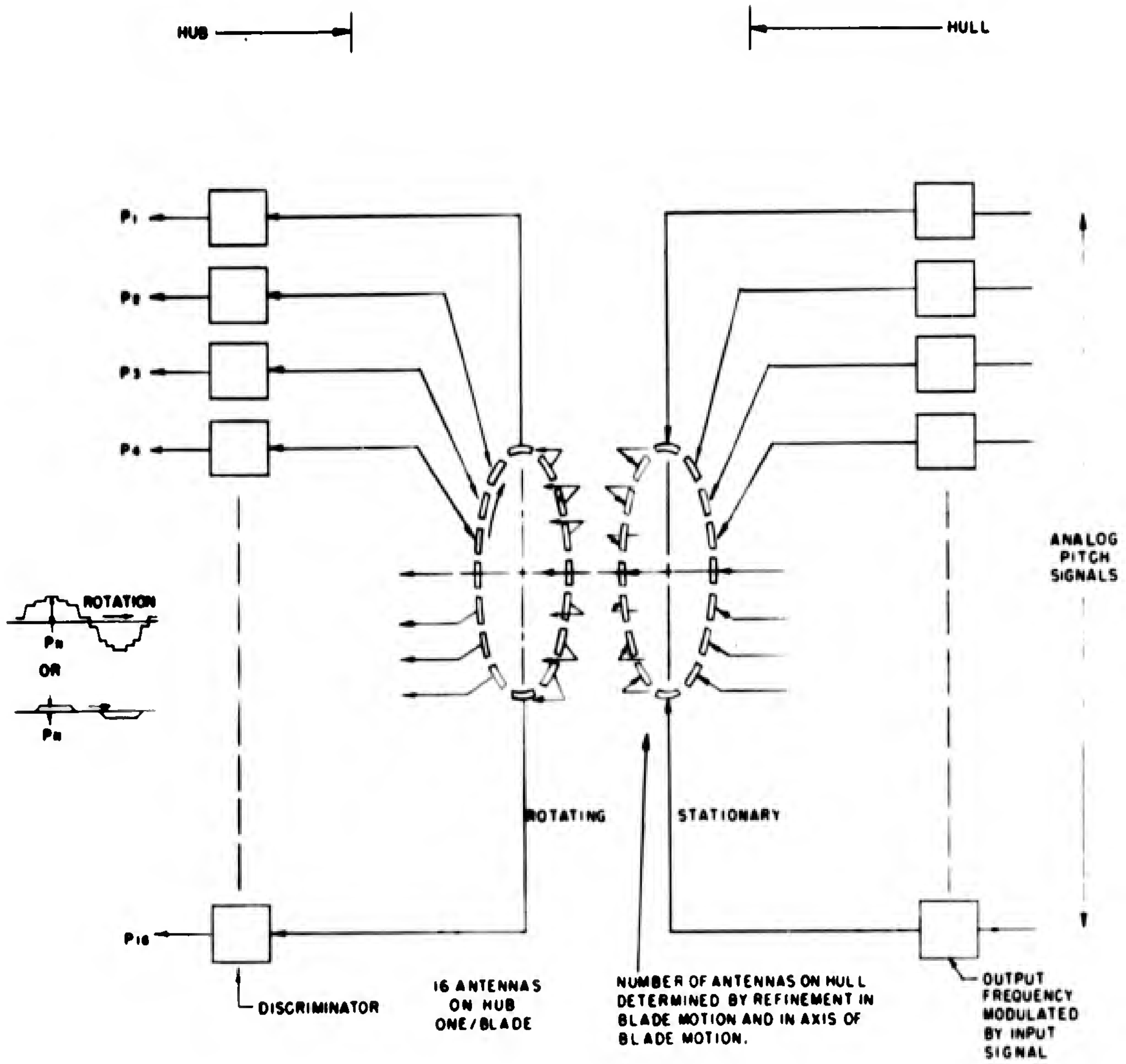


Figure 41. GENERAL ELECTRIC Signal Transfer from Hull to Hub via Multiple RF Antennas

In the various figures, potentiometers, synchros, and resolvers can be used interchangeably, but in practice potentiometers would be limited to the static systems of Figure 40.

This first exploratory phase of the study has been conducted on a basis of sinusoidal cyclic pitch changing. If the rate of motion is controlled to an optimum speed, it may well be desirable to have a three position program wherein only a 90 degree sector of the blades will have an increase in pitch and the opposite 90 degree sector would have a similar decrease in pitch. Figure 40-A shows this possibility using multiple wipers in a potentiometer. Another scheme not shown as a figure would be the use of a mechanical cam to generate a program, and the cam could be replaceable to change the program. Another aspect not evaluated is the use of punched cards to describe the program as the switching means of Figure 40-A. However, using the multiple antenna transmission scheme allows for this later possibility since only signal generation equipment within the hull need be changed, and this can be done even after the ship is built.

Pitch Changing Devices

Two general approaches are considered: hydraulic and electric. The hydraulic pitch changing device is shown in Figure 42. A central hydraulic system supplies fluid to all 16 devices from an electric motor driven pump, and thereafter each blade has its own operating cylinder and control valve. While the figure shows the control valves operated by electric motors, the control valve can also be operated by mechanically following a ring such as that in Figure 37. Actual rubbing on the ring can be avoided by a bleed flow of water or a suitable feedback system.

The electric pitch changing device is shown in Figure 43. Except for the common electric power supply, each blade is actuated and controlled completely independently. Several types of motor are shown, but all of the motor controls use silicon controlled rectifiers. This element alone is the breakthrough that makes possible a static control that is smaller

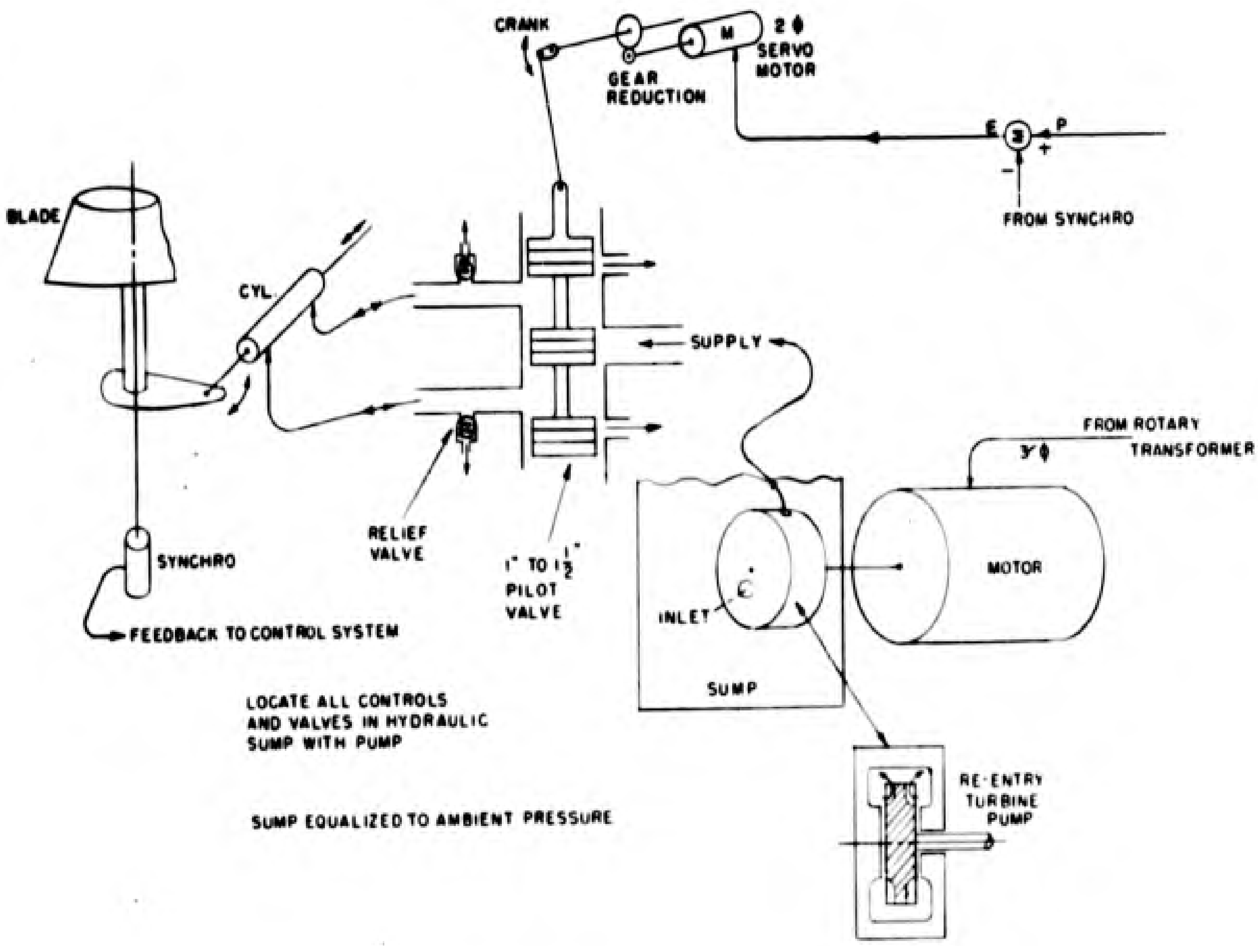


Figure 42. GENERAL ELECTRIC Electro-Hydraulic Pitch Changing System

than the motor itself. Within the General Electric Company, a 10 hp unit is in operation in the General Engineering Laboratory and 40 hp units are being developed by the Medium A. C. Motor Department.

The power input to the motor control is DC obtained from a common rectifier supplying all motor controls.

The advantage of a wound rotor induction motor is in an application as a power synchro. Coupled with a one-speed control synchro, the system can be very accurate. The slip rings might need to be replaced by a rotary transformer, however, making it much less desirable. The wound rotor induction motor is larger than the squirrel cage motor.

The synchronous motor has the same position accuracy as the power synchro, except at low speed. The rotor is probably permanent magnet excited.

The squirrel cage induction motor offers the utmost in simplicity, and if the ultimate in position accuracy is found to be mandatory, a two-speed feedback system would suffice.

Several types of coupling between the motor and blade are shown in Figure 44:

> Scheme A may be inherently quieter than Scheme B; however, the torque-limiting clutch is easier to realize in Scheme B.
>
> Scheme C provides a very natural torque limiting means.
>
> Scheme C is less efficient than Schemes A and B due to pump losses, so the motor size is larger than in Schemes A and B.

Blade position detection for use in a closed loop positioning system can be accomplished with synchros or resolvers.

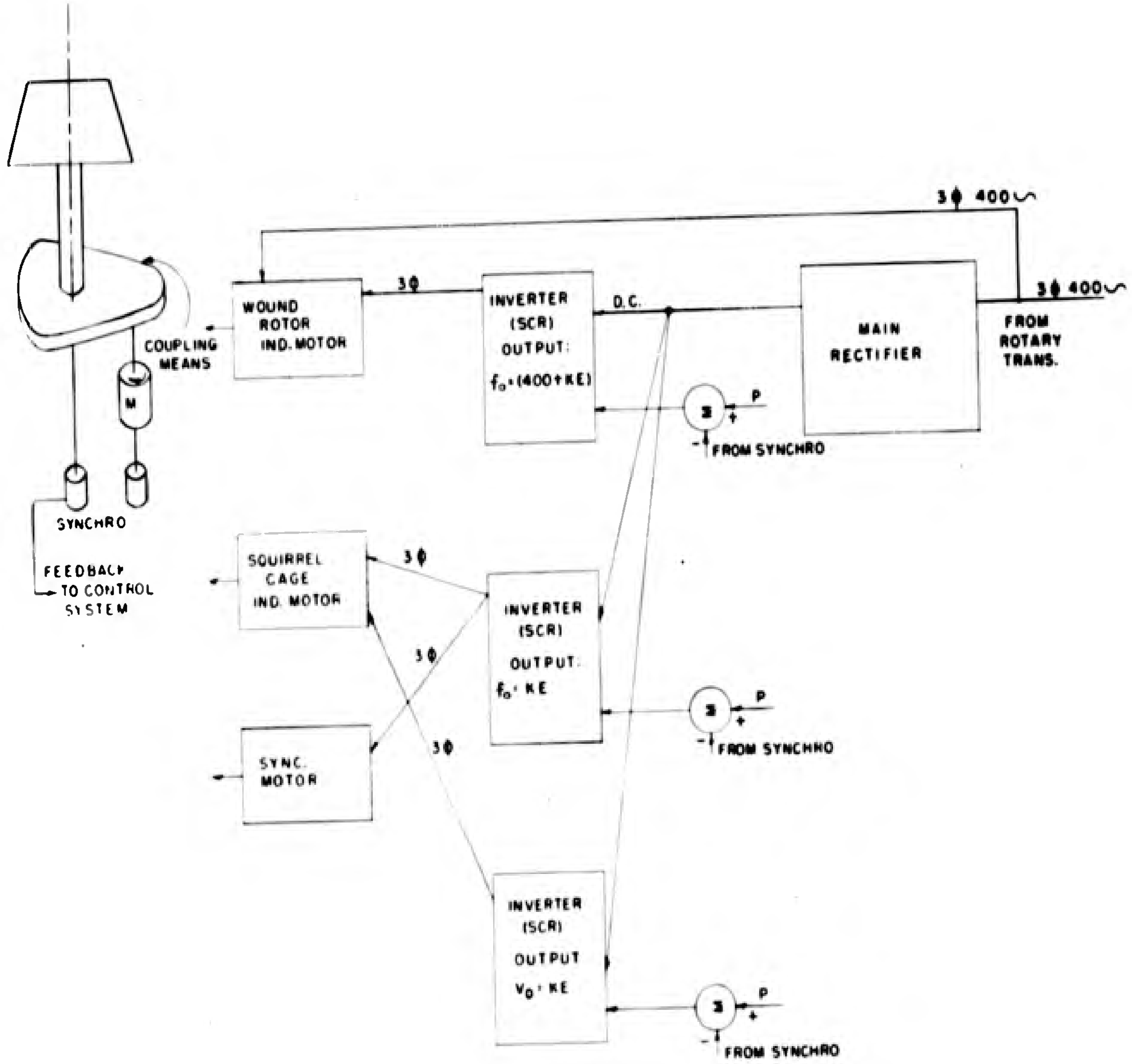


Figure 43. GENERAL ELECTRIC Blade Drive Motors

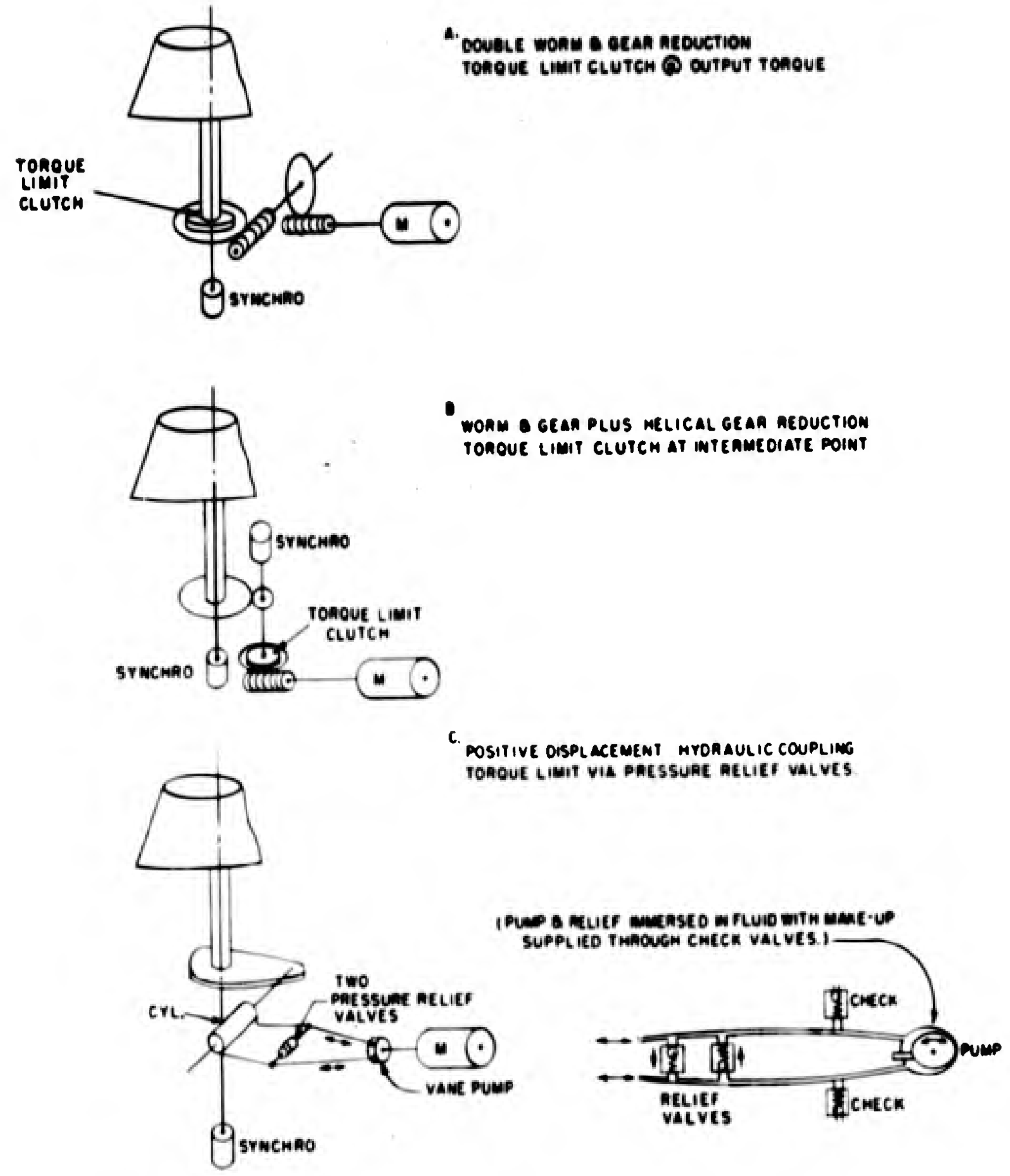


Figure 44. GENERAL ELECTRIC Motor-Blade Couplings

VII

HYDRAULIC PITCH CHANGING SYSTEM

Although it was originally anticipated that the Elliott Company would study both the propulsion machinery and pitch changing systems, it soon developed that better use of available talent would accrue if Elliott concentrated on the propulsion machinery. The pitch changing system study was therefore undertaken by a separate team at Electric Boat, to work with Elliott in producing a complete propulsion package. Since the original Electric Boat and General Electric work had already commenced with mechanical and electrical approaches, respectively, this third effort was directed specifically toward hydraulic systems. This section accordingly covers Elliott studies of propulsion machinery and Electric Boat studies of pitch changing systems. The Elliott work is reported to date but is not yet complete. The remaining work will be reported in a supplement.

MAIN PROPULSION MACHINERY

Because of the greater size and weight of the turbine generator and motor combination, as well as the need for larger interconnecting cables and hull penetrations when induction motors are employed as compared to synchronous motors, most of the effort has been concentrated on the synchronous propulsion drive. Studies conducted to date indicate no basis exists for denying the feasibility of the tandem propeller concept insofar as problems associated with the propulsion motor are concerned.

Due to the simplicity of the normal squirrel cage induction motor, employing only one insulated winding, it seemed desirable to investigate the possibility of using this type of propulsion drive. It has the distinct advantage over the normal synchronous motor that no electric power need be transmitted across the air gap to the rotor, nor is an insulated rotor winding required.

The normal synchronous motor, on the other hand, is usually somewhat smaller in physical size than an induction motor since the induction motor must furnish its own magnetizing current. As an alternate to the normal synchronous motor employing an energized field winding, the possibility of using a permanent magnet field was also considered.

Squirrel Cage Induction Motor

As a starting point a design was first made for a 50 rpm induction motor developing 6550 hp at 60 cycles (144 poles), 3 phase, 2300 volts. The results are summarized in the first column of Table II. Because of the very poor power factor of the 60 cycle machine, an alternate design was made using 25 cycles (60 poles), 3 phase, 2300 volts, and these results are summarized in the second column of Table II. Although the power factor is improved by a factor of two, the motor weight is also increased by about the same factor. Attention was then directed toward the synchronous machine.

Synchronous Motor

For comparison, a design was made for a 50 rpm synchronous motor of the same horsepower at 60 cycles, 3 phase, 2300 volts. The results are summarized in the third column of Table II. From inspection of the table, which also shows the size and weight of the associated turbine generator sets, it is evident that the synchronous machine offers (with some increase in complexity):

A modest weight saving of about 20 tons/ship, plus additional saving in weight of the motor structure and TG set foundationing.

A saving of about 5 feet in turbine generator length, which directly affects the ship length.

A somewhat larger air gap, which eases the bearing alignment requirements.

TABLE II - Characteristics of 6550 hp, 50 rpm, 2300 volt, 3 phase Propulsion Motors and Associated Turbine Generator Sets (Data shown is for one motor and one TG set)

Motor Type	Induction	Induction	Synchronous (4)
Frequency, cps	60	25	66.7
Power factor	0.328	0.677	1.0
Efficiency (1)	0.895	0.939	0.937
Core length, in.	63	63	60
Air gap, in.	3/16	1/4	1/4
Stator ID (2), in.	190	180	182
Stator OD (2), in.	200	199.5	197
Rotor OD (2), in.	210	220	218
Stator weight (2), lbs.	55,000	106,000	65,000 (5)
Rotor weight (2)	56,000	117,000	95,000 (5)
TG set speed, rpm	3,600	1,500	3,600
TG set length, ft-in.	30-7	31-5	25-5
TG set width, ft-in. / TG set height, ft-in.	Sized for two to fit 31 ft. diameter circle		
TG set weight (3), lbs.	218,000	279,000	147,000
Motor weight (2), lbs.	111,000	223,000	160,000
Total machinery weight	329,000	502,000	307,000

Notes:

1 Not considering friction and windage

2 Active material only

3 Includes sub-base and condensers

4 Includes integral provision for power transfer to the rotor

5 Including encapsulation and ordinary frame structures the stator is about 90,000 lbs. and the rotor about 125,000 lbs.

The synchronous motor was therefore used in all further machinery design work.

The synchronous propulsion motor is shown in cross section in Figure 45. The hydraulic pump also shown is a part of the pitch changing system, described later. The associated turbine generator set is shown in outline in Figure 46. As a consequence of incorporating the means for transferring power to the rotor in the main core (which is also discussed later), the frequency was raised from 60.0 to 66.7 cycles, and the resulting 4000 rpm turbine generator sets are therefore slightly smaller than the 3600 rpm sets shown in Figure 46.

In the interest of simplicity, an effort was made to use a permanent magnet type rotor. It was discontinued when it was found that the flux density which could be developed by these magnets was only about 50% of what could be developed using the more conventional electrically energized field member. To have employed permanent magnets would have increased the size of the motor about 300%. Due consideration was also given to the possibility of accidentally demagnetizing the magnets through faulty operation, and the possibility of breaking the magnets (which are quite brittle) in service due to shock.

Environmental Protection

Protection of the electrical windings and cores of the motor from sea water is necessary, and is the principal item of development in the electrical design of this machine. Several different materials and arrangements for protection of the windings and core are being studied, and it is expected that the final result will be a composite. One arrangement is the enclosure of the active parts in metal cans which are hermetically sealed and filled with either a suitable liquid or a solid, and arranged to operate successfully in the range of temperature and pressure which will be encountered in service. In the case of a liquid-filled system, the liquid used must be of such a nature that leakage could not be detected on the surface of the water. The second principal arrangement is one using non-metallic materials throughout

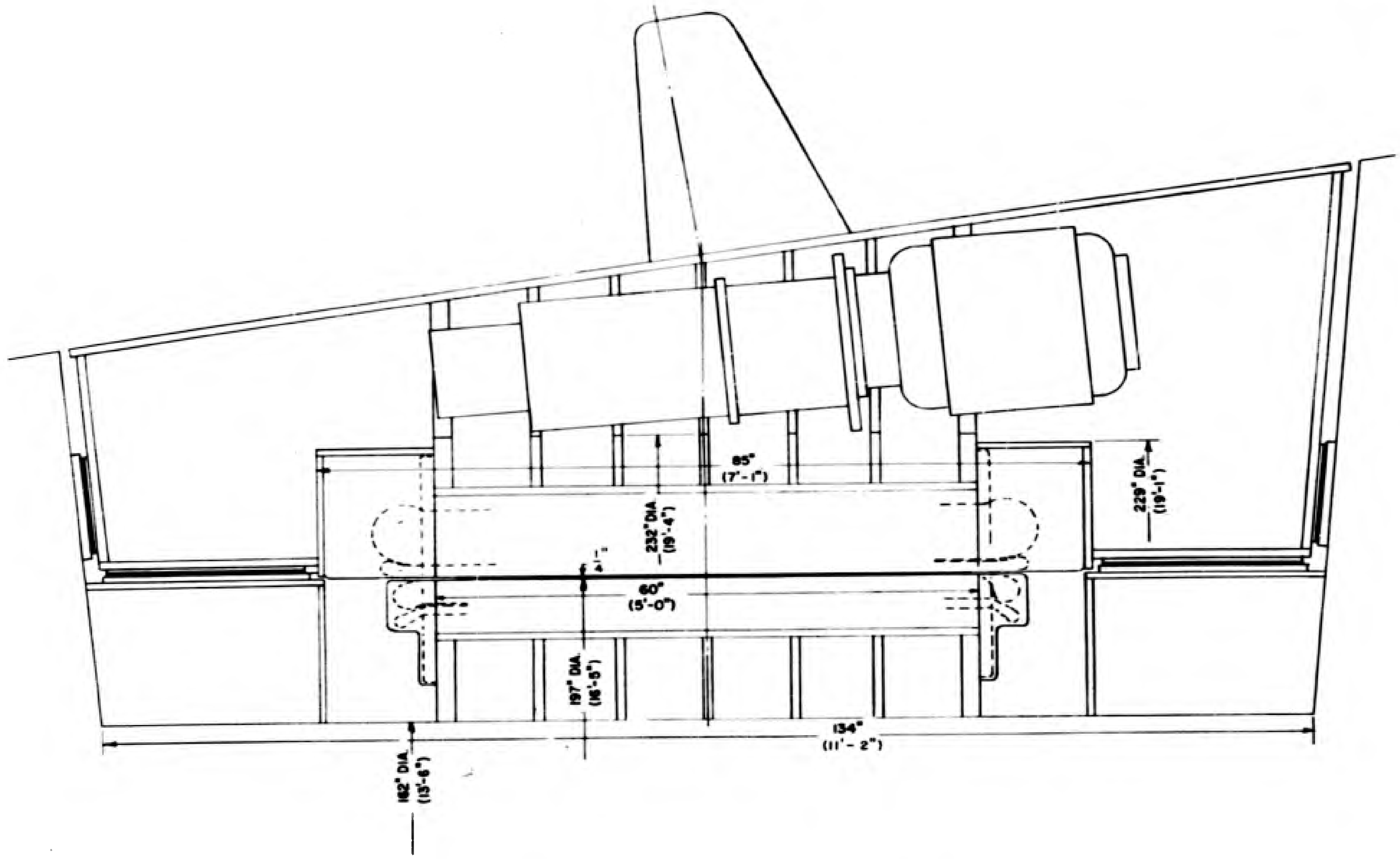


Figure 45. ELLIOTT Propulsion Motor Cross Section

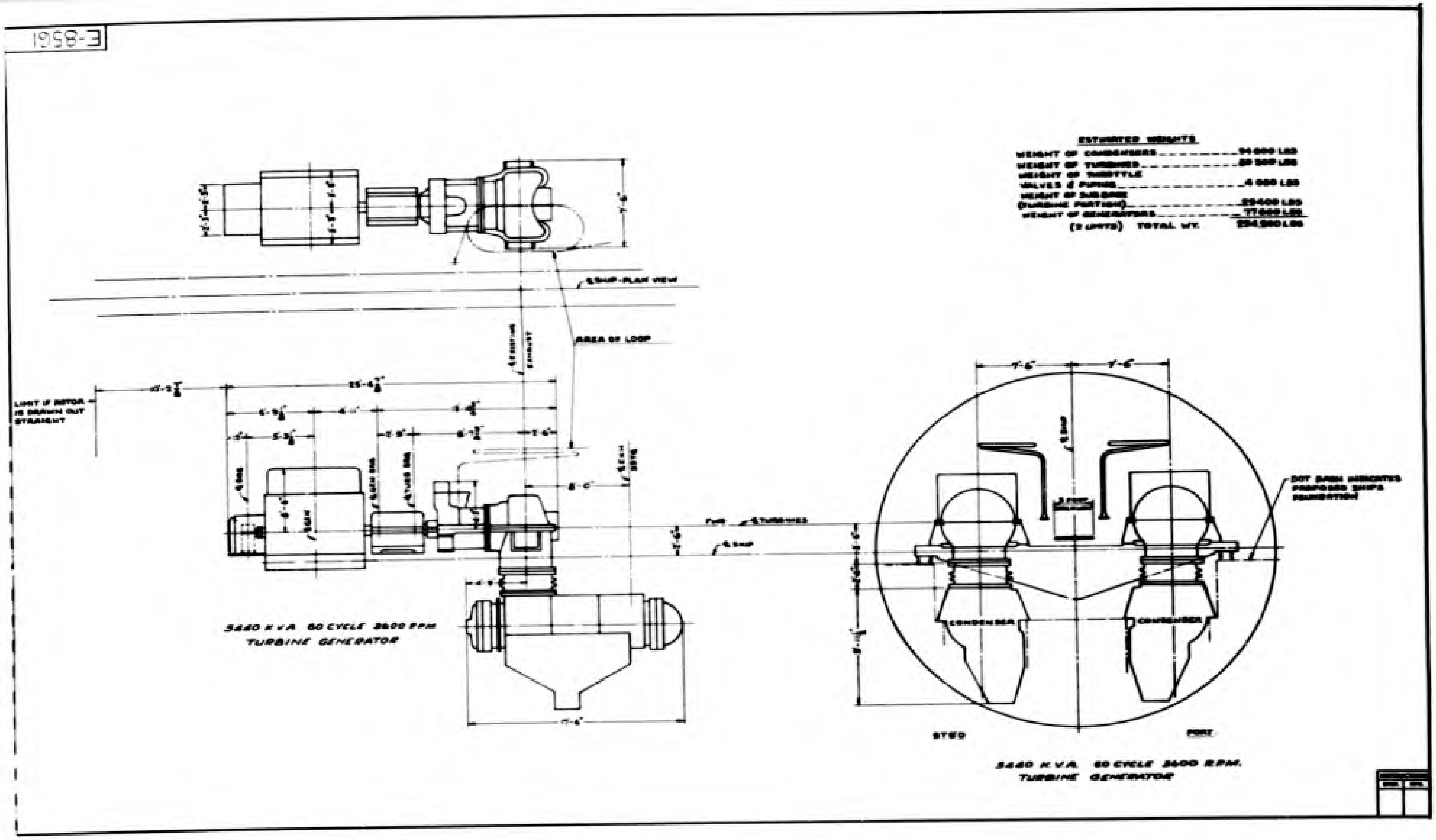


Figure 46. ELLIOTT Turbine-Generator Outline

with a single material for the entire encapsulation, if such a material can be found. More likely, a combination of materials will be required to provide the necessary properties such as flexibility, toughness, resistance to abrasion, and thermal endurance, giving due consideration to both the low and high temperature limits which will be encountered. Practical considerations of application of the materials in the construction of the machine and the possibility of repairs of minor damage due to handling or other accidental rupture of the enclosure are among the requirements.

In connection with the encapsulation, under serious consideration is the possibility of including practical means for leak detection inside the enclosure to give suitable warning before the leak can cause failure of a main winding. Heat transfer in an encapsulated machine is a matter of prime consideration in the design and will be given additional attention as the study proceeds.

In the construction of this motor both the rotor and the stator are split into two halves to permit assembly on the hull of the vessel. Each half of each of the two motor elements is completely encapsulated and sealed. Each part is provided with the necessary sealed leads and with the necessary mechanical handling devices and shipping and installation cradles.

Excitation power for the synchronous motor field winding is obtained from auxiliary power transfer windings in the main rotor and stator cores, and is rectified by silicon rectifier units mounted in the rotor structure. The rectifier system and the rectifier units are being studied to determine the types most suitable for the conditions; i. e., encapsulation and operation at the ambient pressures that will be experienced, and to determine the degree of margin and protection that will be necessary.

Of utmost importance in this entire study is the reliability and life of the components in every detail of the entire design. It is recognized that no one component which will cause the machine to become inoperative can be permitted to fail under any anticipated condition of service, including combat. This means that in the case of units such as the rectifiers, protection in depth and the use of seasoned rectifier elements will be employed rather than the use of a fused circuit. Insulation levels and the endurance of the encapsulation system are considered to be of the utmost importance.

The current program includes making and arranging for calculations and tests to determine electrical losses and to determine the feasibility of specific encapsulation materials and methods. Component testing is underway in the current program, and systems testing would be undertaken in a subsequent program.

Motor Bearings

The radial bearing length is a conservative 22" for each bearing. Rubber has been tentatively selected as the most suitable material for the radial bearings, and is used in the form of 3" wide staves. However, an entirely new plastic material is being studied, with possibly more desirable overall properties than rubber. Phenolic plastic has been tentatively selected as the most suitable material for the thrust bearings, and is used in the form of 6" square pads.

Tentative bearing loss figures are 175 hp for each radial bearing and 50 hp for one active and one idle thrust bearing, resulting in a total bearing loss of approximately 400 hp.

POWER TRANSFER TO THE ROTOR

A design of synchronous motor employing a round rotor type of field member (instead of the more conventional salient pole construction) permits independent transfer of electric power to the rotor through separate additional windings in the main stationary and rotating cores. Thus, by a relatively small increase in size and weight of the propulsion motor over a conventional synchronous motor, a means is readily available for transfer to the propeller hub of AC power, part of which is rectified and employed to excite the field winding, and part of which is used without rectification by the pitch changing system. Consideration of winding combinations, speed, and power generation leads to a main motor winding having 160 poles and a frequency of 66-2/3 cycles for the maximum speed of 50 rpm. The power transfer winding is energized from the 450 volt 60 cycle ships service system. It has 80 poles and a phase rotation opposite to that of the main winding. The voltage induced in the secondary (or rotor winding) is at 60 cycles per second at standstill and will vary linearly to 93.3 cycles at full speed. This variation in frequency with speed is actually desirable in this application, since it is advantageous that the hydraulic pump speed increase with increasing propeller speed. About 300 kw is required for the pitch changing system and 170 kw for motor excitation.

The use of the core of the motor for both main propulsion power and transfer of auxiliary power results in a more compact and lighter weight unit than would be the case if the functions of main propulsion power and control power transfer were handled by separate units. Although the application is new, there are no new principles involved in the design and application of multiple winding single core machines. For example, many multi-speed induction motors have two or more separate armature windings having different numbers of poles which do not interact with each other electrically. An additional advantage of the single core construction is a reduction in the amount of environmental protection for windings and cores compared to that which would be required with more than one set of cores and windings.

It is also possible to transfer amplitude modulated information to the hub in a similar manner, with still more windings, giving the advantage of a high signal power level and a reliability commensurate with that of the main motor. However, while this approach is amenable to rigorous machine design and analysis, it does not appear that the required signal transmission accuracy of about ± 1/4% could be accomplished with confidence because it would be significantly affected by such subtleties as material non-uniformities, dimensional variations, and flux leakage.

INFORMATION TRANSFER TO THE ROTOR

A variety of means of information transfer is available. The one selected has the advantage that the bulk of the equipment is available on the industrial market and provides no need for development of new components except to make them suitable for Navy service. The only exception to this is the coupling device that bridges the gap between the hull and propeller hub.

The scheme is shown in Figures 47 and 48. Inductive coupling is used between a large loop on the hull and a pick-off coil on the propeller hub. Both elements are encapsulated in a plastic and are held in place by non-magnetic supports. The vehicle for transmission is a block of telemetering channels ranging in frequency from 5.4 to 70 kilocycles per second. These were selected to be high enough to be free of interference from the power machinery. Eight channels are used to carry the cyclic pitch information and one channel is used to carry the collective pitch information. Each channel is frequency modulated to prevent erroneous information from being generated by geometric irregularities or dynamic perturbations of hub or hull.

Hub position is determined by an array of stationary magnetic pickups located peripherally around the hull, and actuated by a permanent magnet so located on the hub that it passes close to each of the pickups. A voltage divider network is tapped at each of the pickups, and this provides a signal that indicates the position of the hub at any time. This signal is fed to a servo motor in the hull which duplicates the rotational motion of the hub both in velocity and phase. This motor drives a bank of four resolvers phased in $2\pi/16$ steps. These resolvers then supply the proper phase relation for half of the propeller blade cyclic pitch signals. The other half are supplied by phase inversion later. The amplitude of the cyclic pitch is controlled by the excitation voltage of the resolvers. The pitch axis angle is adjusted by varying the relationship between the motor and resolvers by a differential or equivalent. The eight cyclic pitch signals frequency modulate their eight respective telemetering carriers. The collective pitch signal modulates the ninth channel.

The signals are then combined and amplified to power the antenna loop. They are received and reconstituted in the hub, after which the cyclic

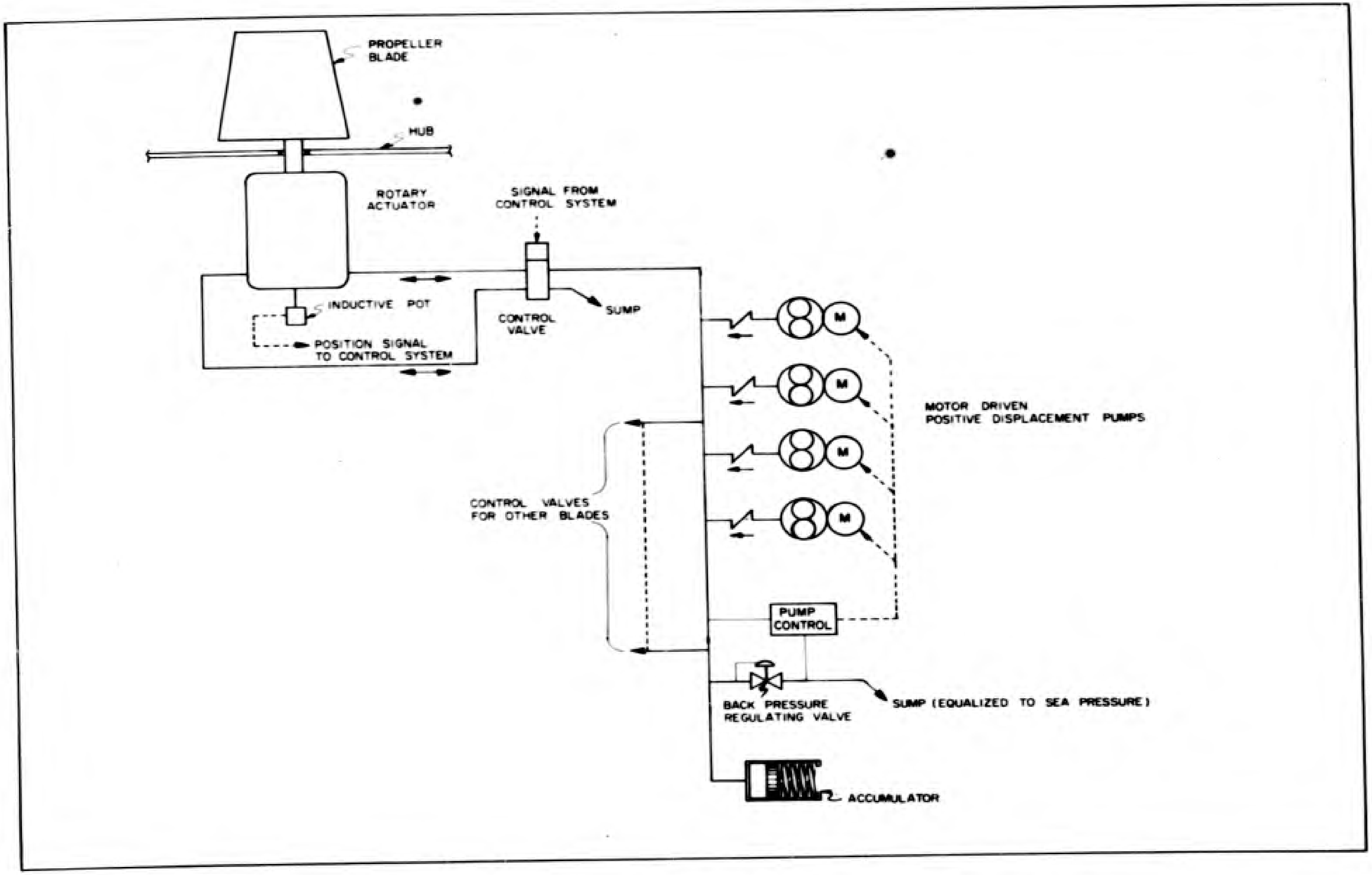


Figure 47. ELLIOTT-ELECTRIC BOAT Hydraulic Pitch Changing System

pitch signals are inverted to provide a total of 16 signals from the 8 supplied. In 16 separate summing amplifiers they are combined with collective pitch and the feedback signals from the blades. The amplifier outputs actuate servo valves to keep each blade following its proper signal as originally generated in the appropriate resolver on the inboard side of the system.

BLADE ACTUATING DEVICE

The blade actuating device is a hydraulic rotary actuator of the type produced by Flo-Tork Corporation. This consists of a pinion on a shaft which is driven by a pair of counter-moving racks which in turn derive their motion from hydraulic cylinders. Referring again to Figures 47 and 48, the flow of fluid is controlled by servo vlaves and the power is derived from positive displacement electrically driven pumps. Mounted on the same shaft as the propeller blade is an inductive potentiometer to provide feedback for the position control system.

The hydraulic power plant consists of a battery of four hydraulic pumps of varying sizes. The pumps are the IMO type, manufactured by DeLaval, and are driven by submersible electric motors. A switching network based on current average demand selects the pumps which are to be energized at any particular time. This feature, coupled with the frequency vs. rpm relationship of the electric power supply, aids in minimizing noise generation by the pumps. The reservoirs contain provisions for equalizing the base pressure of the system to the ambient water pressure. Accumulators are available to provide for surges of power when needed, such as for collective pitch changes, and to provide a buffer between the pumps and valves. A typical arrangement of the various parts within the propeller hub is shown in Figure 49.

DEVELOPMENT AND CONCEPTUAL DESIGN OF HYDRAULIC SYSTEM

In order to control the ship using variable pitch propellers in tandem it is necessary to provide the propeller hub with adequate power and control signals to vary the pitch of the propeller blades, both collectively and in a cyclic pattern properly oriented to the vertical

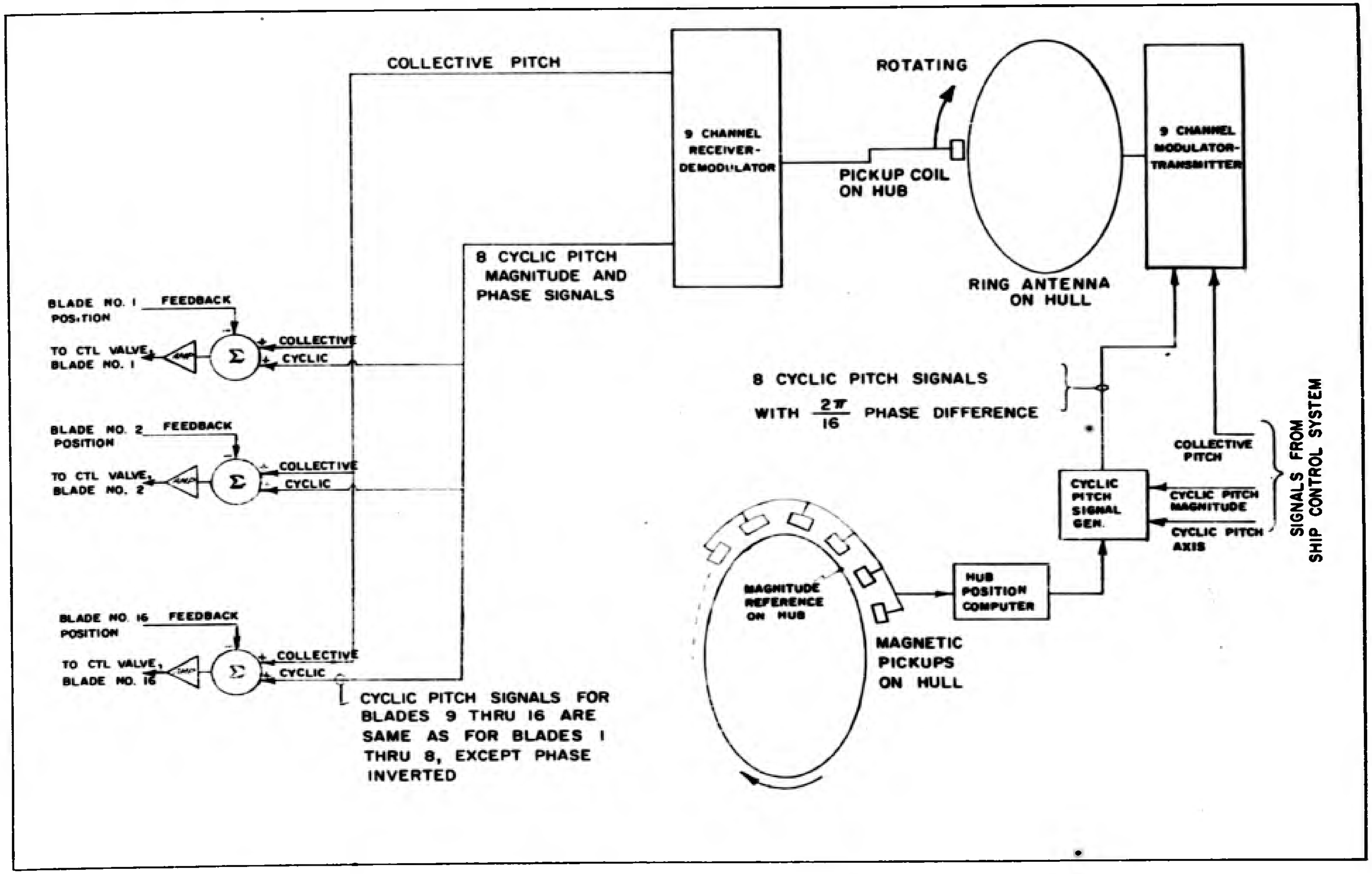


Figure 48. ELLIOTT-ELECTRIC BOAT Signal Generation and Transmission

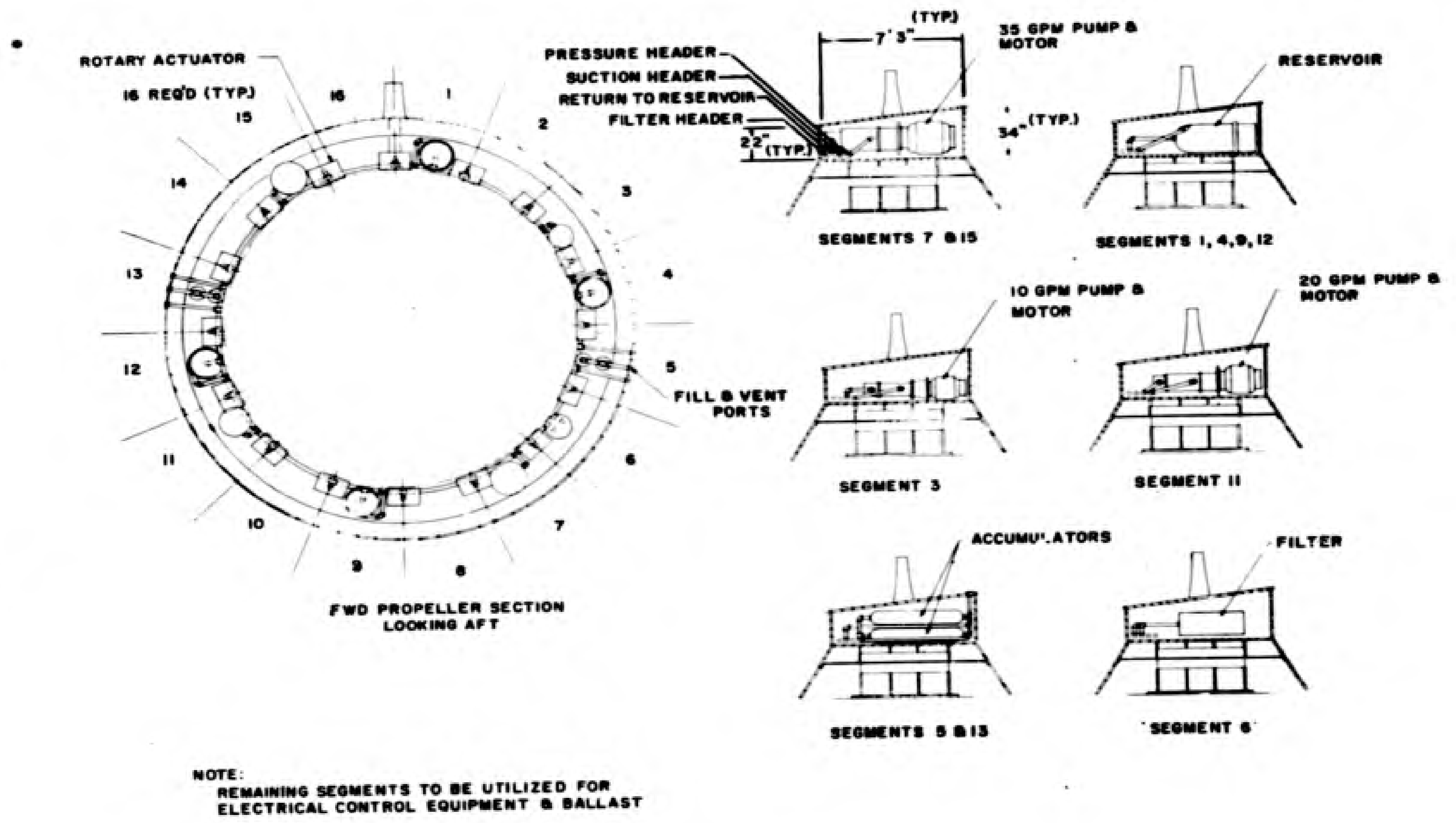


Figure 49. ELLIOTT-ELECTRIC BOAT Typical Hydraulic System Layout in Propeller Hub

centerline of the ship. Because of the basic hydraulic approach, there can be no direct link between the hull of the ship and the propeller hubs. Thus it is necessary that the hydraulic plant be entirely self contained within the hub, and that the control system be partially contained within the hub. In addition, the equipment in the hub must be capable of operating when exposed to chemical and abrasive influences of natural sea water.

The problem can be resolved into the following items:

Bridging the gap between hull and hub with control signals and power.

Providing an accurate control system capable of translating commands into corresponding blade motions.

Making the system compatible with its environment and isolated condition.

During the course of this study many possible alternatives were investigated. For various reasons, most were disqualified; one was selected as most promising and another was considered as a possible concept to be considered in the event that the first choice must be abandoned.

The following general conclusions were reached, and these formed the basis for the preliminary design work which followed:

Information will be transferred to the hub at radio frequency (VLF), using a ring shaped antenna and a pickup coil, with conventional telemetering techniques.

Power will be transferred to the hub by additional windings in the main propulsion motor stackings.

Each blade will be driven by a hydraulic actuator, supplied with fluid from a central hydraulic plant through an electrically controlled valve.

Signal Generation and Transmission

The variety of methods of transferring information from the hull to the rotating hub is almost limitless, but it can be rapidly narrowed down to a few likely candidates for active consideration. Electrical contact is obviously out of the question. Mechanical systems similar to those described in a preceding section could be used. However, since the information gained in the mechanical studies could be applied directly to a hydraulic system (for example, to position a servo valve), and since wider knowledge is gained by taking a different approach, effort was directed toward non-mechanical signal transmission schemes.

In the absence of conducted information transfer one must rely on a propogated medium. These can be divided into acoustic and electromagnetic. The acoustic is eliminated because no transducer is immediately evident which could provide a continuous link of communication for all propeller positions, and because of the disrupting effects of extraneous noise. This leaves electromagnetic transmission as the remaining candidate. But electromagnetic energy can take many forms depending on its frequency. One can consider optical, radio, and power frequencies, and the varying forms in which each can be modulated.

Optical frequencies can first be eliminated, not because of any inability to perform the task, but because marine growth would soon provide a filter with more than enough attenuation to suppress the signal. A cleaning device was considered as a solution to this problem, but any cleaning device would in all probability pick up enough abrasive particles to ruin the transducers in a short period of time.

Radio frequency has its problems, but they seem to be the most amenable to solution and the variety of modulation schemes makes this part of the spectrum most attractive. There is the question of generation and propogation of unwanted energy that can be used for homing purposes by enemy installations, but only enough energy is transmitted to be picked up by a receiver less than an inch away, and the antennae are well shielded by the hull and propeller hub. A scheme using a radio frequency telemetering link is described more fully below.

Power frequencies can also be used with a different modulating system. However, this requires more elaborate hardware and is put in a lower suitability category for this reason. In the event that the radio frequency link proves to be unfeasible for any reason a description of this system is given as an alternate.

Radio Frequency Communication Link

This system uses a hoop-shaped antenna and a pickup coil to transmit information from the hull to the hub. Nine signals are transmitted as frequency modulated channels on VLF carriers.

The collective pitch signal is handled in a straightforward manner. The cyclic pitch signal is conveniently assumed to be sinusoidal (since no specific waveform is presently required), and the magnitude of this signal is readily handled. The phase of this signal, however, must be controlled with respect to the propeller hub angular position, in order to establish the desired cyclic pitch axis. This is done by continuously measuring the hub speed and position and feeding this information to the cyclic pitch signal generator, which then appropriately synchronizes the cyclic pitch signal. The cyclic pitch signal is generated by a bank of four resolvers and transmitted to the hub via eight VLF channels. Once the properly phased cyclic pitch signals are on the rotor, they must be divided from 8 into 16 separate signals, each phased differently to account for its corresponding blade's different location around the hub. This is done by reversing the phase of the first 8 signals to supply the remaining 8 required.

Power Frequency Communication Link

As an alternative to the above system, this system uses a transformer coupling between the stator and the rotor with a series of primaries spaced around the circumference at equal intervals. To achieve the required accuracies approximately twenty primary segments are required. There is one secondary or pick-off for each blade of the propeller. The signal is in the form of a square wave with a basic repetition rate of approximately 100 cycles per second. The information is contained in the pulse length of the square wave. All computation is done inside the pressure hull and appropriate signals are fed to each of the primary coils. For collective pitch only, all primaries have the same pulse length. For cyclic pitch, the primary in the position corresponding to maximum forward pitch has the longest pulse length signal;

the one opposite, the shortest; the two in quadrature, the average value; and the intervening ones the proper value to approximate a sinusoidal progression.

As each blade passes a primary coil position its secondary picks up the appropriate signal, which is processed into an amplitude proportional to the pulse length and then fed to the blade positioning servo. The main drawbacks of this system are its more elaborate hardware requirements and its reliance on a precision power supply for its accuracy. Both of these may well provide more difficulty than the system using radio frequency to transmit the signal.

Hydraulic Fluid

The hydraulic fluid selected for this system must meet the following requirements:

It must be compatible with the materials used and it must have suitable viscosity and lubricity to permit proper functioning of the system.

It must be such that it will not leave traces to reveal the location of the boat in the event of a leak or failure.

Of the several fluids investigated, only three, in addition to water, merited further consideration:

Cellulube, manufactured by Celanese Corporation

Silicon fluid by Dow Corning

UCON fluid by Union Carbide Corporation

A brief description follows:

Water - Water was investigated because it offered the possibility of a limitless supply, but it lacks both the lubricity and viscosity to be suitable at the power levels required.

Cellulube - This is a phosphate ester fire resistant fluid previously used aboard submarines but since replaced by petroleum base fluids. The chances of making a previously rejected fluid meet customer acceptance cancelled this candidate.

Silicone Fluid - There are two types that were considered. The chlorinated silicone fluid could possibly rise to the surface and betray the position of the boat. The fluorinated fluid seemed to meet all requirements but one. It is prohibitively expensive (about $80,000 per system filling).

UCON or Polyalkylene Glycol - These fluids were suggested by the manufacturer and appear to satisfy our requirements. They are good lubricants, are miscible with water without leaving any tell-tale clues, and can be purchased over a range of viscosities to suit specific needs. The fluid tentatively selected is 50-HB-280-X. It has a viscosity of 280 SSU at 100°F and a density of 1.031 at 60°F.

Accuracy and Speed of Response

Accuracy in the collective pitch mode requires that each blade be positioned to within plus or minus one degree of the commanded position. After a very preliminary analysis of the system, it appers that an overall accuracy of one degree can be achieved for the collective pitch.

Since the accuracy specified for the collective pitch mode can be attained, and since substantially the same components are used when operating in the cyclic mode where the accuracy requirements are less stringent, the required cyclic performance can also be attained under static conditions. This being the case, it then remains to provide sufficient dynamic response to meet the cyclic pitch accuracy specification.

The command signal is sinusoidal and has a maximum frequency of fifty cycles per minute, or less than one cycle per second. To follow this signal closely the most elementary of servomechanisms would suffice. The disturbances of the blade have the same peak frequency, but because of the presence of harmonics they require a better response than a pure sine wave does. Increasing the response of the servo to five cycles per second imposes no severe design problems and this response rate should be ample to meet the cyclic pitch accuracy requirements.

VIII

COMPARATIVE EVALUATION OF PITCH CHANGING SYSTEMS

This section evaluates the three pitch changing systems, and does so principally relative to each other rather than on an absolute basis. This section does not evaluate the propulsion machinery, since the Elliott work is not yet complete. The machinery evaluation will, however, be included in the supplement reporting the remaining Elliott work.

The three pitch changing systems compared are described in detail in the preceding sections. Briefly, the mechanical system considered here consists of a large wobble plate, with the propeller blades rotated through a bevel gear by a crank which follows the wobble plate. The hydraulic system consists of a complete hydraulic power and positioning system located within the propeller hub. The electric system consists of an individual motor and worm gear for each blade, also located within the propeller hub.

Each system also has associated electrical control equipment located within the pressure hull, and while this is different for each system, taken as a whole there is no gross difference which would seriously affect the choice of pitch changing system. Thus in this comparison principal attention is focused on outboard equipment.

In addition, the main propulsion motor is similar in all cases, and it therefore is not a variable. The synchronous type machine was first considered because two of the three pitch changing systems required electric power in the propeller hub. Although the mechanical pitch changing system does not require electric power in the hub, and can therefore use the simpler squirrel cage induction motor, it now appears that the synchronous motor has sufficient advantage in itself to also be used with the mechanical system. The mechanical system can be built into either the General Electric or the Elliott motor. Also, no inherent design features of the motors preclude the use of the Elliott motor with an electrical pitch changing system similar to that developed by General Electric, or the use of the General Electric motor with a hydraulic pitch changing system as developed by Electric Boat and Elliott.

Listed below are a variety of topics significant in the comparison of the systems, followed by brief comments on each system with respect to each particular topic. Following the list is a general appraisal, leading to the conclusion that the electric system is most favorable, the mechanical system next most favorable, and the hydraulic system least favorable.

Compatibility With Environment

Mechanical - Structural parts are painted. Gears, rams, bearings, and other moving parts are made of corrosion resisting materials. All bearings and gears are sea water lubricated. The rams have slow speed high pressure linear seals between hydraulic fluid and sea water.

Hydraulic - All stationary parts are painted or made of corrosion resisting materials. The pump motors operate flooded with sea water. The pump and blade actuating shafts penetrate the hydraulic system and are made of corrosion resisting materials. All other moving parts are within the hydraulic fluid filled system. The pump shafts have high speed high pressure rotating seals between hydraulic fluid and sea water. The blade actuator shafts have low speed low pressure rotating seals between hydraulic fluid and sea water. Information transfer devices are unstressed and encapsulated. Outboard control equipment is in sealed enclosures.

Electrical - Stationary parts are painted or made of corrosion resisting materials. The blade actuating shafts penetrate the fluid filled enclosure and are made of corrosion resisting materials. All other moving parts are within the fluid filled enclosure. The blade actuator shafts have low speed low pressure rotating seals between fluid and sea water. The information transfer devices are unstressed and encapsulated. Outboard control equipment is within the fluid filled blade actuator enclosure.

Equipment Inboard and Outboard

M All outboard equipment is of substantial structural type construction. All control equipment except the hydraulic rams and servo valves is inboard.

H Information transfer devices, signal processing equipment, closed loop electrohydraulic positioning equipment, and hydraulic plant are outboard. Remaining control equipment is inboard.

E Information transfer devices and closed loop electric positioning equipment are outboard. Remaining control equipment is inboard.

Torque Limit

M Relief valves on rams provide torque limit.

H Relief valves on blade actuators provide torque limit.

E Available motor torque provides torque limit.

Simplicity

M Outboard equipment is basically very simple. Inboard equipment for determining required wobble plate ram positions and for closed loop electrohydraulic ram positioning is roughly equal in complexity to that of other systems.

H Outboard equipment is most complex of the three systems. Inboard equipment is roughly equal in complexity to that of other systems.

E Outboard equipment is mechanically very simple, but also includes closed loop electric position control equipment. Inboard equipment is roughly equal in complexity to that of other systems.

Flexibility

M Cyclic pitch program is inherently restricted to a sinusoidal waveform. Pitch angle range can be increased if required.

H Cyclic pitch program is limited to a sinusoidal waveform by the information transfer scheme (which could be replaced by a more flexible one). Pitch angle range can be increased if required.

E Cyclic pitch program is completely flexible within the context of a multiple step type of waveform. Pitch angle range is unlimited.

Accuracy

M Required pitch angle accuracy is marginally attainable, and will deteriorate with bearing wear. Required pitch axis accuracy is attainable.

H Required pitch angle accuracy is attainable. Required pitch axis accuracy is also attainable.

E Required pitch angle accuracy is attainable, and there is potential for further improvement if required. Required pitch axis accuracy is also attainable.

Response Time

M Overall system time from maximum ahead collective pitch to maximum astern collective pitch is about 5 seconds. More normal changes in pitch angle and/or axis require less time. Shorter time is practical if required.

H Overall system time from maximum ahead collective pitch to maximum astern collective pitch is about 2 seconds. More normal changes in pitch angle and/or axis require less time.

E Overall system time from maximum ahead collective pitch to maximum astern collective pitch is about 2 seconds. More normal changes in pitch angle and/or pitch axis require less time.

Reliability

M Reliability of inboard control equipment is roughly equal to that of the other systems. The sea water environment will degrade performance of the exterior system with time, but the outboard parts will only stop functioning altogether upon jamming or structural failure. Severe local wear can make the system inoperable because of the resulting noise, even though it is still physically capable of operation.

H Reliability of inboard control equipment is roughly equal to that of the other systems. Outboard electrical control equipment is all solid state. Except for the pump motors, which operate in sea water, most parts operate in a favorable environment of hydraulic fluid. The blade actuator shaft seal presents an elementary sealing problem and should be troublefree. The pump shaft seals will be more susceptible to failure. Within the capacity of the reservoir, outward leakage is not harmful. Inward leakage is harmful but not catastrophic.

E Reliability of inboard control equipment is roughly equal to that of the other systems. Outboard electrical control equipment is all solid state. Most of the parts operate in a favorable environment of lubricating fluid. The blade shaft seal presents an elementary sealing problem and should be troublefree. Within the capacity of the reservoir, outward leakage is not harmful. Inward leakage is harmful but not catastrophic.

Casualty Control

M The wobble plate position can be manually controlled by manually valving hydraulic fluid to the rams. Direct connected mechanical position indication for the wobble plate is available inboard. Electrical control equipment is of modular design to facilitate rapid restoration to service.

H There is little that can be done at sea on outboard equipment. If the casualty affects only one actuator, the blade can be unbolted so as not to interfere with the others. It is possible to control the ship with only one propeller when it is underway. Inboard electrical control equipment is of modular design to facilitate rapid restoration to service.

E There is little that can be done at sea on outboard equipment, but each actuator is independent of the others and a casualty will therefore affect only one actuator. The actuator can be electrically disconnected and the blade allowed to feather, or its blade can be removed altogether. Inboard electrical control equipment is of modular design to facilitate rapid restoration of service.

Installation

M Blade actuators are fitted to the propeller hub in the shop. Wobble plate requires assembly and critical alignment on propulsion motor foundation after installation on ship.

H Blade actuators, hydraulic system, and half of information transfer device are fitted to the propeller hub in the shop. Stationary half of information transfer device is assembled around propulsion motor, but alignment is not critical.

E Blade actuators and half of information transfer device are fitted to the propeller hub in the shop. Stationary half of information transfer device is assembled around propulsion motor, but alignment is not critical.

Maintenance

M Blade actuators are of modular construction. Wobble plate and positioning cylinders are accessible only when the entire rotating assembly is removed.

H Blade actuators are of modular construction. Filling and venting are accomplished from outside the propeller hub. Other hydraulic system parts are accessible through openings in the propeller hub. Information transfer devices are accessible with the rotating assembly in place.

E Blade actuators are of modular construction and include all outboard parts except the information transfer devices which are accessible with the rotating assembly in place.

Life

M Wobble plate follower bearings are running and loaded continuously, even when there is no cyclic pitch variation. Load is varying for cyclic pitch change and may even reverse for large changes. Actuator bearings only move as required. Sea water environment is unfavorable. Bearing and gear replacement is required annually. System imposes decreased load on propulsion motor thrust bearing.

H One or more hydraulic pumps run continuously. Actuator and control valves only run as required. Environment for most parts is favorable. Motor bearing, pump seal, and servo valve replacement is required annually.

E Actuators only run as required. Environment for most parts is favorable.

Size

M -

H Largest

E Smallest

Weight

M Heaviest

H -

E Lightest

Cost

M Roughly comparable to electric system

H Highest

E Roughly comparable to mechanical system

Efficiency

M Comparable to electric system

H Least efficient

E Comparable to mechanical system

Noise

M This system is more prone to large backlash than the others. Bevel gears are a relatively noisy type of gear. Flow noise from wobble plate positioning hydraulic servo valves can be expected.

H Some or all pumps and motors run continuously. Considerable flow noise can be expected from the hydraulic servo valves, and pumps will also contribute significant noise.

E The worm gear is a relatively quiet type of gear. Actuator moves only as required. Some motor noise can be expected. Flow noise from standard ship control surface positioning hydraulic servo valves is eliminated.

Development Required

M All development is within the state-of-the-art.

H All development is within the state-of-the-art.

E All development is within the state-of-the-art.

Perhaps the most immediately evident features of this list are that the various topics are not of equal importance and that there is not always a clear 1-2-3 ranking of the systems with respect to any one topic.

In exploring this list the most obvious implication is a negative one--that the hydraulic system can be dispatched as the least favorable of the systems. This type of actuation was initially thought to be inherently suitable due to the blade positioning requirements for small displacements, large forces, and high accuracy. However, further development of the entire pitch changing system yielded a system which is second or third best in most of the topics listed and is not outstandingly good in any. Since this system has hydraulic pumps and servo valves--noisy components even inboard--in intimate contact with the sea, it can almost be discarded for purely acoustic reasons. It is conceivable that in some areas extensive hardware development might improve the situation, but comparable effort would be even more fruitful if applied to hardware for one of the other two systems. The hydraulic system is clearly the least desirable of the three systems; the choice between the two remaining systems, however, is not as obvious.

One of the major differences between the mechanical and electrical systems, and a factor which pervades the comments on a number of topics in the list above, is environment of the blade actuators. The mechanical system operates entirely in sea water (although the gears and blade spindle could be enclosed in a fluid filled chamber, leaving only the wobble plate bearings, like the motor bearings,* exposed to the sea).

---

* However, these bearings are not directly comparable with the motor bearings, since in order to avoid backlash they must run with substantially zero clearance, and are therefore particularly susceptible to jamming.

The electrical system operates entirely in a lubricating fluid filled enclosure, except of course for the connection to the propeller blade; the information transfer devices are encapsulated and have no moving parts. The controlled environment of the electrical system provides a basic advantage with respect to accuracy, reliability, maintenance, life, and noise generation. Even if a failure should cause dilution of this favorable environment, performance will deteriorate gradually rather than abruptly.

The difference between the two systems with respect to simplicity is not as great as might be expected; in fact, the electrical system is simpler mechanically than the mechanical system. However, considering also the control equipment included in the electrical system, the mechanical system is overall somewhat simpler.

Flexibility is often the antithesis of simplicity, and such is the case here also, although not to a large extent. The simple mechanical system is inherently limited to a sinusoidal cyclic pitch distribution. The somewhat more complicated electrical system is inherently unlimited in cyclic pitch waveform. The value of this flexibility depends upon studies beyond the scope of this report, and while the sinusoid is a convenient waveform, it is not known to be optimum. Section IV of this report notes, for example, that different waveforms may possibly improve propulsion efficiency or adapt the propeller to accommodate wake irregularities. In general, however, flexibility is a particularly desirable feature for a first-of-a-kind ship since it allows obtaining more varied experience at full scale and immediately profiting from this experience.

Reliability is a particularly difficult topic to resolve, since the systems are not much alike. The mechanical system will stop functioning altogether only upon jamming or structural failure. This, however, should not be tacitly assumed to be impossible. Furthermore, any severe local wear can require stopping the propeller for acoustic reasons even though it is still physically capable of operation. The electrical system is also subject to mechanical failure, and electrical failure in addition. This is mitigated by the favorable environment for the mechanical parts and the use of inherently long lived solid state electrical control equipment. Thus while not completely clear, it is probably correct to say that the electrical system will have more frequent failures than the mechanical system. What is clear however, are the consequences of a failure once it occurs. The mechanical system, consisting of one large assembly, is susceptible to a failure disabling the entire mechanism either physically or acoustically. The electrical system, consisting of sixteen independent blade actuators, effectively confines a failure to one actuator and blade. Depending upon the nature of the failure, it can be ignored, a diver at shallow submergence can electrically isolate the actuator, or a diver can unbolt the propeller blade. The remaining fifteen blades are unaffected.

With respect to noise, the electrical system has significant advantages. The mechanical system is more prone to develop large backlash and incorporates relatively noisy bevel gears, while the electrical system has closely fitted parts in a lubricating fluid and incorporates relatively quiet worm gears. The more viscous lubricating fluid will also be of some small aid in reducing noise generation from any clearance which may exist. Some motor noise can be expected, which is completely absent in the mechanical system. Although the tandem propeller concept eliminates hydraulic controls for the ship control surfaces, with the mechanical system there is roughly no net improvement because the wobble plates are controlled hydraulically, and the servo valves with their attendant noise are retained. With the electrical system the servo valve noise is completely absent, and in addition one main hydraulic pump and its attendant noise is eliminated.

Required pitch angle accuracy is marginally attainable with the mechanical system, and will be degraded by wear. The electrical system meets and has the potential for even exceeding the required accuracy.

Range of pitch angle can be increased in either system, but only at the expense of accuracy in the mechanical system.

The mechanical system parts for each blade are of modular design, but the wobble plate is large, heavy, accessible only with the rotating assembly removed, and requires critical alignment on the ship. The electrical system parts for each blade are of modular design, except for the information transfer devices which are small, light, accessible with the rotating assembly in place, and do not require critical alignment.

Size, weight, cost, and efficiency are not sufficiently accurately determined to permit a quantitative comparison, but are qualitatively ranked in the previous listing.

It is thus evident that while the difference between mechanical and electrical systems is not as great as that between these two and the hydraulic system, the electrical system does possess significant advantage over the mechanical system, and it is concluded that the electrical system is therefore the most promising.

APPENDIX A

ELECTRICAL MACHINERY SPECIFICATION
FOR
FEASIBILITY STUDY,
TANDEM PROPELLER SYSTEMS

From Specification U413-61-192, Revision C
December 19, 1961
Electric Boat Division, General Dynamics Corporation

I. SCOPE

This specification covers a complete package of AC electric propulsion machinery, including propeller pitch changing devices, two propulsion motors, two steam turbine generator sets, and necessary excitation, control, and protective equipment. Figure 50, attached, shows the motor and propeller installation in the submarine. The inner part of each motor is stationary, supported by the submarine hull. The outer portion of each motor rotates, and to it is attached a hull-sized propeller hub with blades extending radially outward around the circumference of the hub. Propeller pitch can be changed both collectively and cyclically. The propulsion motors and pitch changing devices are submerged in sea water; remaining equipment is located within the pressure hull.

II. GENERAL REQUIREMENTS

1. A multitude of individual specifications will not be called out. However, the equipment shall generally meet naval specifications applicable to nuclear powered submarine propulsion equipment.

2. Size and weight shall be minimized except where otherwise specified.

3. Where susceptible to control, life of equipment shall be at least 40,000 hours at full power. In addition, sea water immersion life shall be at least 10 years. Overhaul or extensive maintenance shall not be required at intervals of less than one year.

4. Noise shall be minimized.

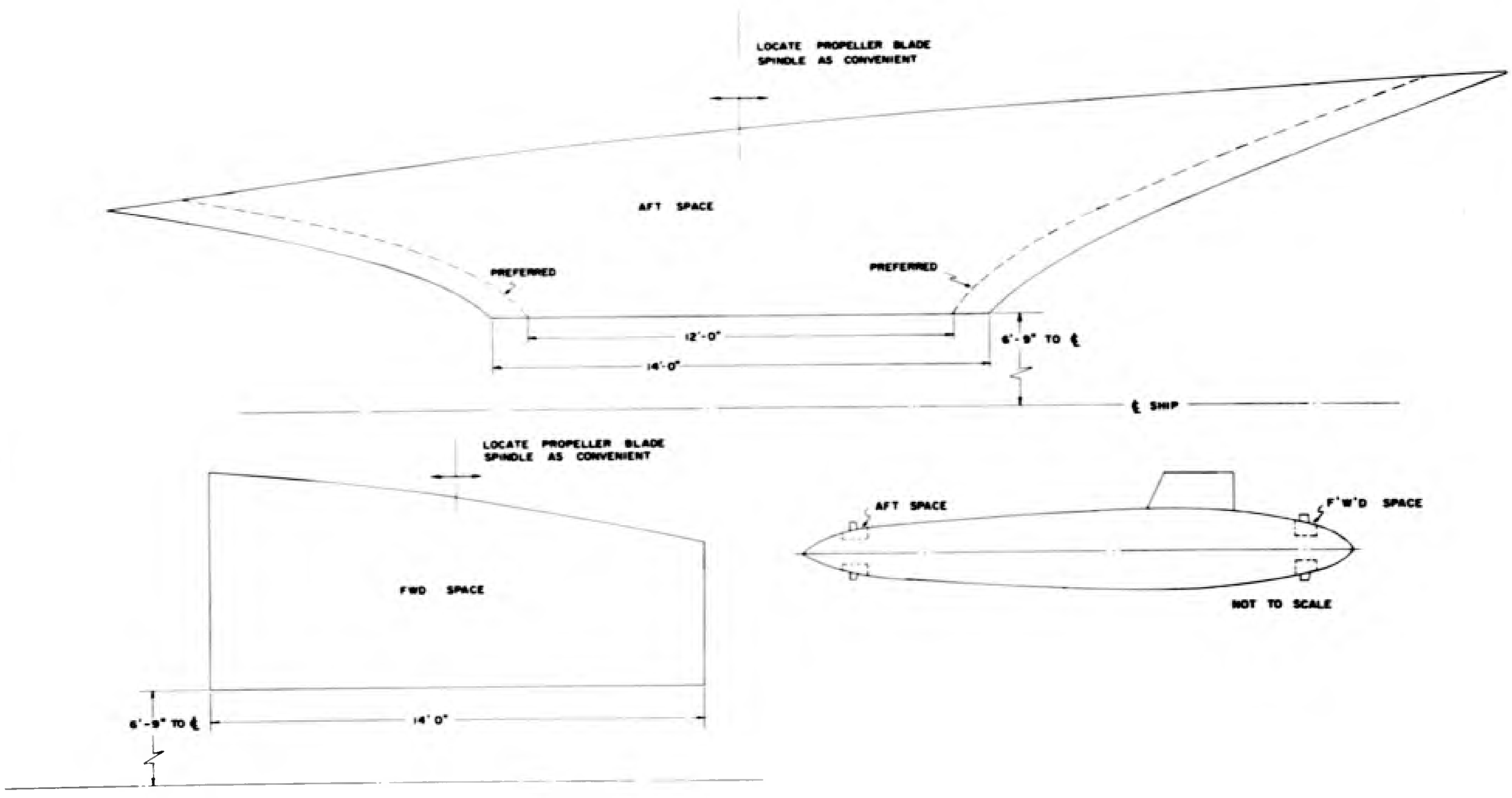


Figure 50. Tandem Propeller Motor - Space Allocation

III. DESCRIPTION OF SYSTEM OPERATION

1. Each TG set is connected directly to its respective motor. No cross connection or switching is provided. Voltage regulators provide a constant voltage/frequency ratio.

2. Motor speed is controlled by varying frequency, which in turn is controlled by adjustable speed turbine governors.

3. The forward motor shall rotate clockwise (viewed from astern) and the aft motor shall rotate counterclockwise. Reversal of direction of rotation is not required.

4. The propeller pitch can be changed both collectively and cyclically.

5. Longitudinal thrust is controlled during transients by varying the pitch collectively, and backing is accomplished with negative pitch. Thrust is controlled during steady operation by adjusting the propeller speed and maintaining design pitch, thereby maximizing efficiency and minimizing noise.

6. Transverse forces are controlled during maneuvering by varying the pitch cyclically about the average collective pitch. This results in unbalanced tangential forces with a net transverse component. For example, consider the forward, clockwise rotating (viewed from astern) propeller in a diving maneuver. The pitch is increased as the blades move upward on the port side of the boat and is decreased as the blades move downward on the starboard side of the boat. The net transverse force is downward. A couple about a vertical axis, tending to steer the boat to starboard is also incidentally produced, due to unbalanced thrust on the two sides of the boat, but it is very much smaller in

magnitude and of minor importance. It can be counteracted by proper articulation of the controls.

7. Transverse forces are controlled during hovering in a similar manner except that the collective pitch of one propeller is reversed so that for the pair of propellers no net longitudinal thrust is produced.

IV. COMPONENTS

A. Propulsion Motors

1. Motors shall be of the synchronous or squirrel cage induction type. Pullout or breakdown torque shall be at least 150% at rated voltage and frequency.

2. Motor full load speed shall be 50 rpm. Motor power shall be determined by the vendor, based upon the steam supply described in Section IV.C.

3. Voltage, frequency, and number of phases are to be selected by the vendor.

4. Motors shall operate submerged in sea water. The submergence pressure for the purposes of this specification will range from near 0 psig to 600 psig. The motors shall, however, be of a design which is inherently unaffected by pressure and therefore could be made for operation at a substantially greater pressure such as 2200 psig. Motors shall also operate satisfactorily at a test pressure 50% greater than design pressure.

5. Means of protecting the windings and iron from the effects of sea water immersion are at the vendor's option.

6. Motors shall not be damaged by occasional operation in water containing sand or ice.

7. Operation shall not be impaired by marine growth, and such growth shall be discouraged where practical.

8. The use of pumps for furnishing sea water to the motor for cooling or bearing operation shall be avoided.

9. Motor bearings shall be sea water lubricated. In addition to loads required to support the rotor and propeller, the bearings of each motor shall withstand the following loads:

   Thrust of 80,000 lbs. forward and 80,000 lbs. aft.

   Forces of 40,000 lbs. in a transverse direction due to cyclic pitch variation and the resulting unbalanced tangential forces.

   Moments of 750,000 lb-ft. about a transverse axis due to cyclic pitch variation.

   Gyroscopic moments due to rotation of the ship at 0.17 radians/second about a transverse axis.

10. Space available for the motors and propellers is shown in Figure 50; however, this represents only an initial estimate and can be changed if the need arises.

11. The motor shall be separable radially into two or more pieces, to facilitate assembly on the ship.

12. No limits need be met, but the order of magnitude of electric and magnetic fields outside the propeller hub arising from the motor shall be stated.

B. Pitch Changing Devices

1. Pitch changing devices shall position approximately 16 blades on each propeller.

2. The overall range of blade movement shall be from +90° to -50° with respect to the tangential direction. Positive sign refers to pitch for ahead propulsion. Design ahead pitch is approximately 35° from the tangential position.

3. The range of collective blade position shall be from +90° to -30° with respect to the tangential direction. Blades shall each be positioned to within ± 1° of the collective pitch ordered by the control system described in Section IV.D.

4. The range of cyclic blade position shall be from +20° to -20° with respect to the collective position, but not to exceed the limits specified in paragraph IV.B.2. Blades shall each be positioned to within ± 2° of the peak cyclic pitch ordered by the control system described in Section IV.D. When a combination of collective and cyclic pitch results in the blades reaching the overall limits of movement in section 2 above, no damage shall occur.

5. The range of cyclic pitch axis position shall be $\pm\infty°$ with respect to the reference axis. The cyclic pitch axis shall be positioned to within ± 10° of the angle ordered by the control system described in Section IV.D. The cyclic pitch axis is a radial line through the location of maximum ahead pitch. The reference axis is a vertical radial line through the top of the hull. Positive angles are measured counter-clockwise (viewed from astern) from the reference axis.

6. The cyclic pitch variation must have a period coinciding with the time for one revolution of the propeller, but the waveform is not critical.

7. The hydrodynamic forces and torques on the individual blades are a complex function of blade position, propeller speed, and boat speed. The following are the maximum anticipated values, and shall be used for design purposes:

   Force of ± 20,000 lbs., perpendicular to face of propeller blade and applied at center of blade span at pivot axis.

   Force of ± 3500 lbs. parallel to face of propeller blade and perpendicular to pivot axis, applied at the center of span.

   Torque of ± 7000 lb-ft. on blade about pivot axis.

   Normal torque during pitch change is ± 1500 ft-lbs.

8. Propeller blade weight is 250 lbs. Propeller blade moment of inertia about the pivot axis is 100 lb-ft^2. (These figures are for the blade only and do not include the spindle or other supporting parts.)

9. Preferred failure aspect for a casualty involving a complete set of blades is design ahead pitch, and the next preferred failure aspect is +90° (feathered at zero propeller speed). Preferred failure aspect for a casualty involving an individual blade is freedom to turn and align with the stream. These are to be regarded as targets, but it is recognized that in order to obtain a feasible pitch changing device it may be necessary to accept some other failure aspect.

10. Requirements of Sections IV.A.4 through 12 pertaining to environment for the propulsion motors apply also to the pitch changing devices.

11. Note that at 50 rpm, equipment at, for example, a 10 foot radius is subjected to a radial acceleration of about 9 g.

C. Steam Turbine Generator Sets

1. Each steam turbine generator set shall be rated in accordance with the inlet steam conditions below. (This corresponds to about 7500 hp at the turbine shaft.)

	No Load	Full Load
Temperature	493 F	417 F
Pressure	625 psig	285 psig
Flow		90,000 lb/hr.
Condenser Pressure		7 in Hg

At full power the water rate shall not exceed 12.2 lbs/turbine shp hr.

2. Reduction gears between the turbine and generator shall not be used.

3. Each turbine shall include a speed governor with remotely adjustable reference speed.

4. Voltage, frequency, number of phases, and power factor are to be selected by the vendor.

D. Control, Excitation, and Protective Equipment

1. Control, excitation, and protective equipment is to be determined by the vendor to operate the propulsion system as described in Section III.

2. Such equipment associated with the TG sets and propulsion motors shall be complete. Such equipment associated with the pitch changing devices shall be suitable for positioning the blades in accordance with input signals of the order of 100 watts from a ship control system not covered by this specification. These input signals will order the three basic parameters: collective pitch, cyclic pitch peak magnitude, and cyclic pitch position with respect to the hull.

APPENDIX B

HYDRODYNAMIC STUDY

Propeller performance estimates were needed at the outset of the propulsion system design study. Knowledge of the forces and moments on the blades was needed in order to design the mechanism and assess its performance. The accuracy required of blade positioning and the extent of change of pitch--collectively and cyclically--were also of vital interest. Since other contractors were investigating these questions in detail, our studies were limited to relatively simple approximations. The results were thus available early in the program with accuracy sufficient for our purposes.

The hydrodynamic studies consisted of:

Selection of blade section and plan form, and determination of lift, drag, and center of pressure as a function of angle of attack.

Determination of axial and tangential force on a blade running at a given rpm and ship's speed, but with varying angle of attack.

Determination of maximum possible blade loading and moment.

Summation of forces on the propeller running at a given rpm, mean pitch, and ship's speed, but with a sinusoidally varying pitch increment of several amplitudes.

BLADE SECTION AND PLAN FORM

The blade selected for study was taken from the Novel Electric Propulsion System study with slight modifications in order to conform with data given in DTMB Report 933 and its extrapolations. The plan form was changed from a taper ratio of .503 to .45, and the rake was changed to correspond to foils for which data was given. Since the blades will work both ahead and astern and at varying angle of attack, a symmetrical section, NACA 0015, was chosen. Table 1 gives pertinent data on the blade. Lift, drag, and center of pressure data are given in Figure 51.

TABLE 1 - Propeller Blade Data

Blade shape				Trapezoid
Blade section				NACA 0015
Taper ratio				.45
Root chord				1.877 ft.
Mean chord				1.36 ft.
Tip chord				.844 ft.
Span				2.45 ft.
Blade area				3.33 ft^2
Effective aspect ratio				3.6
Rake at quarter chord				0
Thickness ratio				.15
Offsets	x/c	y/c	x/c	y/c
	.025	.03267	.50	.06615
	.05	.04442	.60	.05703
	.10	.05852	.70	.04579
	.15	.06680	.80	.03278
	.20	.07170	.90	.01809
	.30	.07500	1.00	.00158
	.40	.07252		

FORCES ON A BLADE

The thrust and tangential forces on a single blade were calculated for a range of angle of attack from near zero to the vicinity of stall.

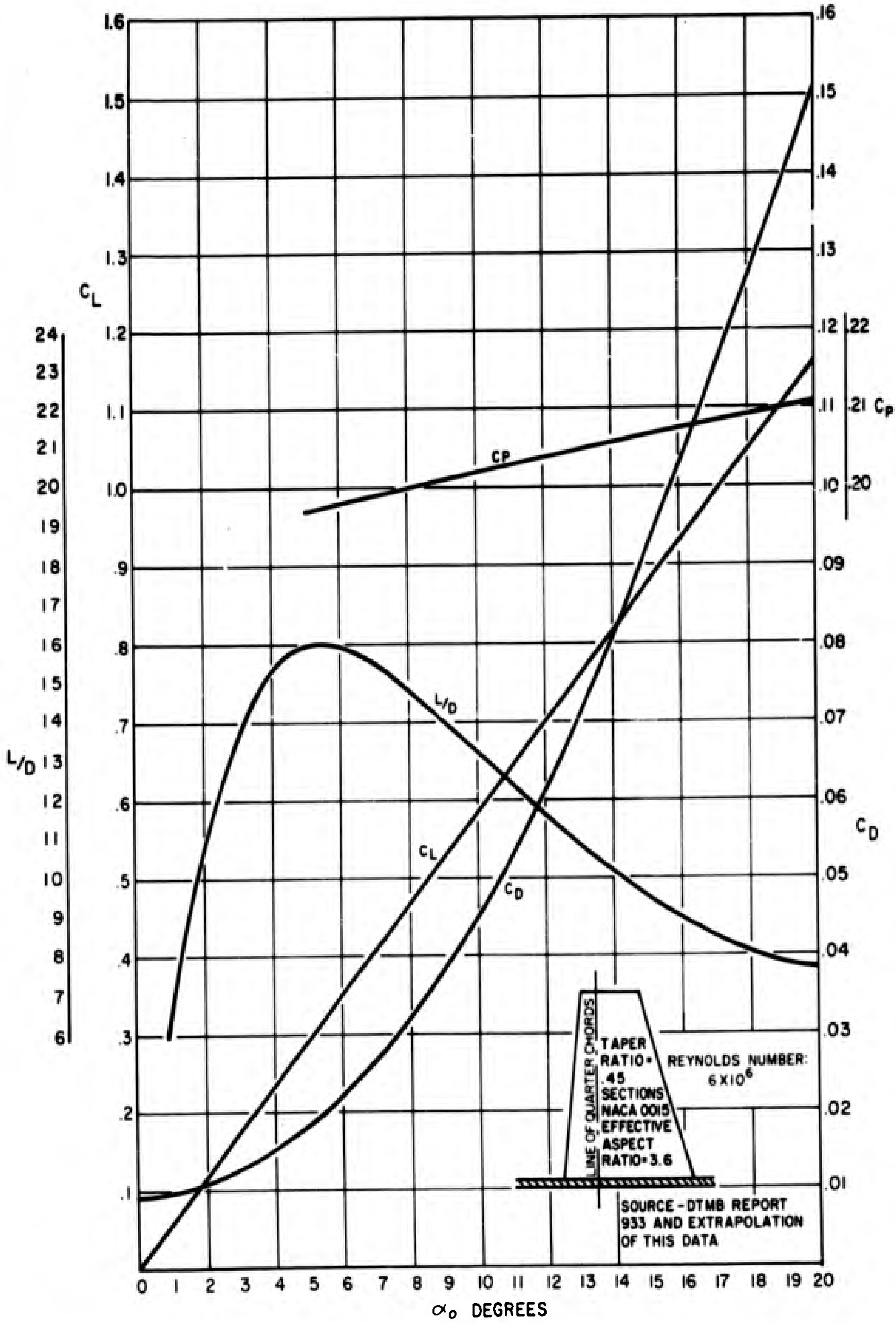
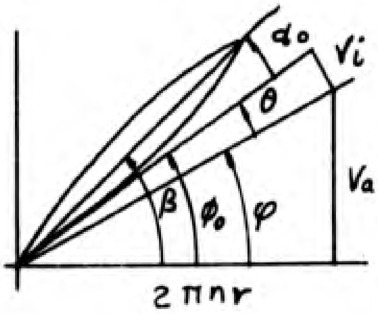


Figure 51. Propeller Blade Lift, Drag, L/D, and Center of Pressure vs. Angle of Attack

Shaft speed and ship speed were held constant at 50 rpm and 19.7 knots, respectively. Wake fraction was assumed to be .15 resulting in a speed of advance of 28.3 ft/sec. It was assumed for purposes of calculation, that flow in the vicinity of the foil was fully developed and that the induced velocity at a foil was the same as if all the blades of the propeller were at that same pitch.

Induced velocities were calculated by means of the equation:*

$$\tan \theta = \frac{B\, b\, C_L}{8 \pi\, r \sin \phi_o}$$



where θ and ϕ_o are defined in the diagram

B	=	number of blades = 16
b	=	mean chord = 1.36 ft.
C_L	=	lift coefficient for angle of attack α_o
r	=	radius to mean chord = 11.92 ft
v_i	=	induced flow vector, assumed at right angles to v_{ro}
n	=	rpm = 50
v_a	=	speed of advance = 28.3 fps

The lift coefficient can be approximated by a straight line function over the range of interest:

$$C_L = a \alpha_o = a(\beta - \phi - \theta)$$

where a is the proportionality constant.

Assuming θ is very small with respect to ϕ, we can take as a first approximation, ϕ equal to ϕ_o, then:

$$\theta = \frac{B\, b\, a\,(\beta - \phi - \theta)}{8 \pi\, r \sin \phi}$$

---

* "Airplane Aerodynamics," Second Edition, by Dommasch, Sherby, and Connolly, Pitman Publishing Corporation.

Then solving for θ:

$$\theta = \frac{\beta - \phi}{\frac{8\pi r \sin\phi}{B b a} + 1}$$

And as a second approximation, θ as calculated above can be added to ϕ to obtain ϕ_o which can be inserted in the correct equation:

$$\theta = \frac{\beta - \phi}{\frac{8\pi r \sin\phi_o}{B b a} + 1}$$

Using the induced velocity angle θ calculated as above, and the lift and drag data from Figure 51, the force on a single blade as a function of pitch angle was determined. The axial and tangential forces are shown as a function of angle of attack in Figure 52, and as a function of blade pitch angle in Figure 53. These forces have been resolved into the directions of the coordinate planes of the ship, and are not to be confused with lift and drag forces.

MAXIMUM BLADE LOADING AND MOMENT

The propeller blade, spindle, bearings, and positioning mechanism must be designed to withstand the highest loads that can be developed by the blade. The highest blade loading occurs just before the stall angle is reached. For the section chosen and the effective aspect ratio of 3.6 the stall angle is about 20°. From Figure 52 we have axial and tangential forces of 12,600 and 12,000 lbs., respectively, at 20° angle of attack. The total resultant force is the vector sum of these two, or 17,400 lbs. The blade pitch angle, β , corresponding to these conditions is 53°. The components parallel and perpendicular to the section chord line are thus:

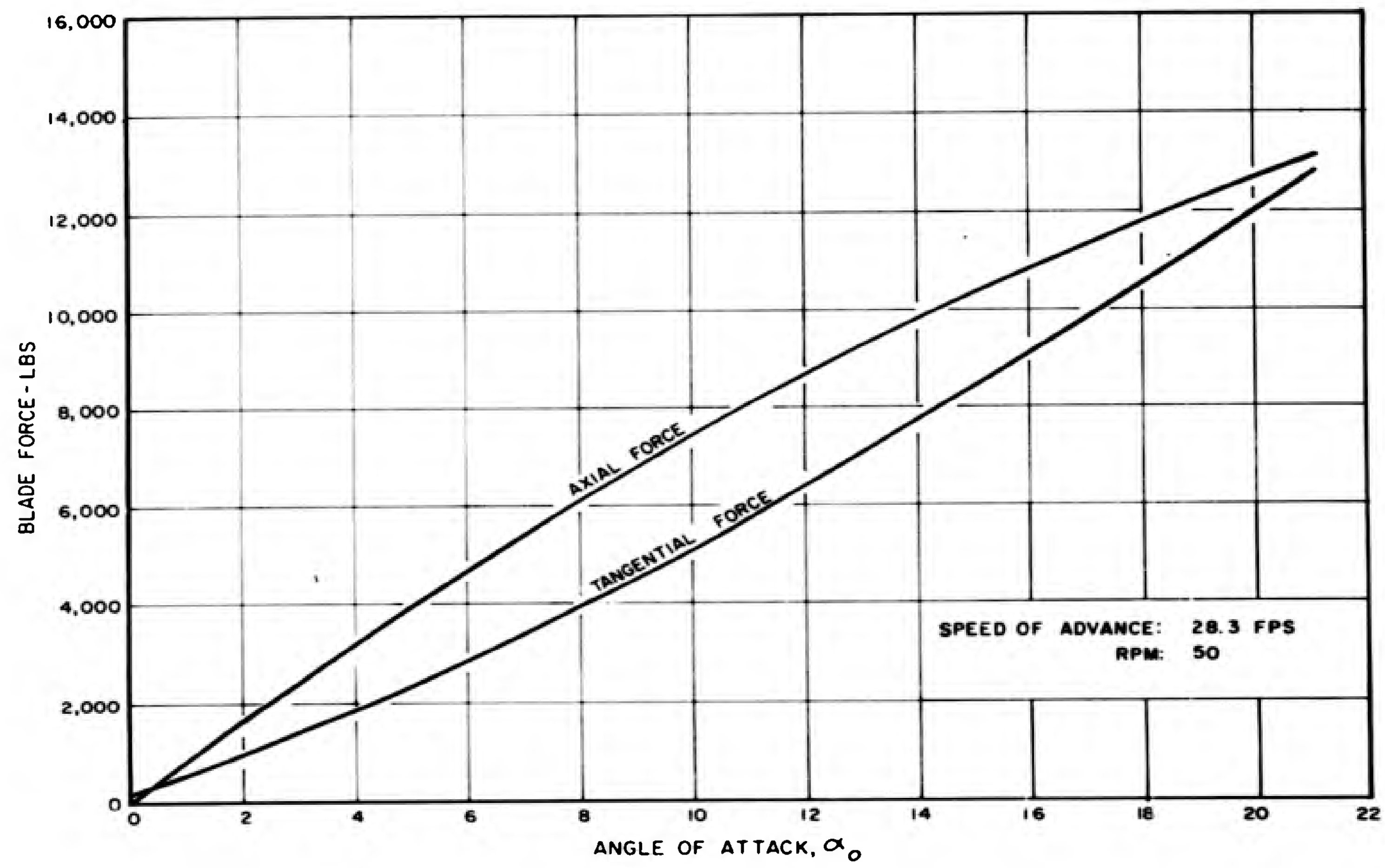


Figure 52. Axial and Tangential Forces vs. Angle of Attack

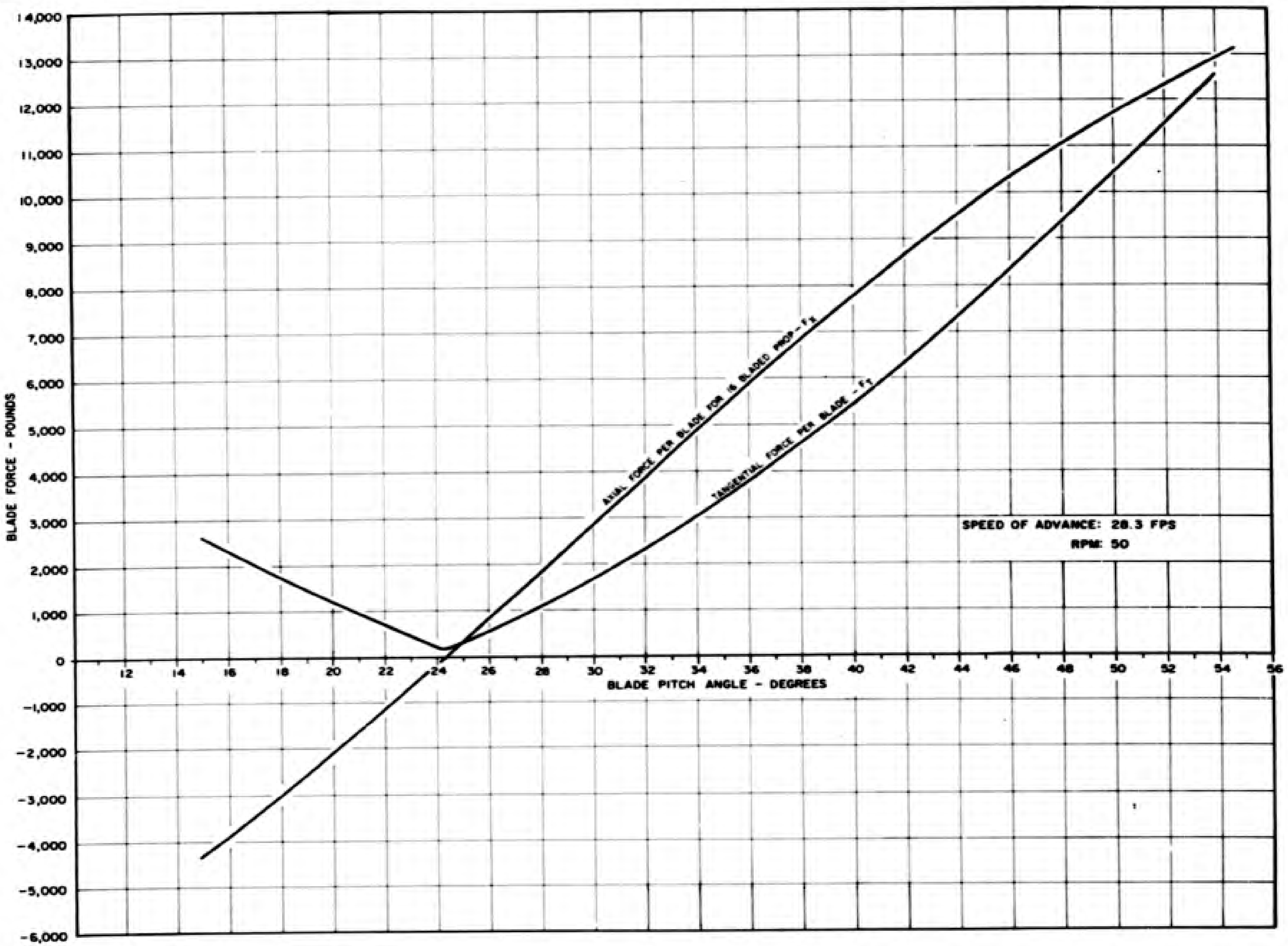


Figure 53. Axial and Tangential Forces vs. Pitch Angle

$$\text{Parallel to chord:} \quad F_{11} = F_x \sin \beta - F_y \cos \beta$$

$$= 8600 - 7220 = 1380 \text{ lbs.}$$

$$\text{Normal to chord:} \quad F_n = F_x \cos \beta + F_y \sin \beta$$

$$= 7580 + 8180 = 15{,}760 \text{ lbs.}$$

In order to minimize torque required for blade control, the spindle axis should be placed near the normal center of pressure. In event of a failure of the mechanism, however, the blade should be arranged so that it will feather with the stream. The axis should therefore be slightly ahead of the most forward position of the center of pressure. For this section this is at about 18% of the chord from the leading edge. The center of pressure departs very little from this position at normal angles of attack. Thus, at 20° angle of attack, the center of pressure is at 23% of the chord from the leading edge. The moment about the axis is then:

$$M_{20°} = 1070 \text{ ft-lbs.}$$

Since the torque is very sensitive to the position of the axis, a conservative figure of 1500 ft-lb. was inserted in the specification.

The maximum moment will occur when the blade is at right angles to the flow. For the same conditions of ship speed and rpm, the relative velocity is 68.6 fps. The normal force coefficient is about 1.2 for an uncambered plate with aspect ratio 3.6. Normal force and moment are then:

$$F_n = \frac{\rho}{2} S v^2 C_n = 18{,}600 \text{ lbs.}$$

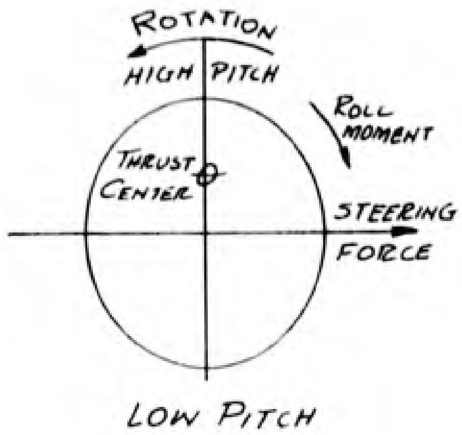
$$M_{90°} = 8100 \text{ ft-lbs.}$$

SUMMATION OF FORCES ON ALL PROPELLER BLADES

As stated above, the hydrodynamic studies were greatly simplified by assuming that each blade performed just as if it were operating in steady state in a propeller all of whose blades were at the same pitch. The summation of forces on the entire propeller was thus a matter of adding the force vectors for all the blades. Sinusoidal pitch variation was assumed, with the maximum cyclic pitch varying from 0 to 18°. At the larger figure, the angle of attack is approximately at the stall point.

A check on the propeller performance reveals that torque increases with an increase in cyclic pitch. The power plant cannot develop sufficient power to maintain design rpm with higher blade pitch. To determine the equilibrium conditions, the motor and propeller characteristics are plotted on torque-speed coordinates, Figure 54. The motor curve is characteristic of an induction motor. (Subsequent selection of the synchronous motor does not alter the curve materially since the main propulsion turbine would exhibit a decrease in rpm with overload.) Propeller torque is assumed to vary as the speed squared. Intersections of the motor and propeller curves indicate maximum operating speed for each value of cyclic pitch amplitude. The right-hand columns of Table 2 give the forces and moments corresponding to these intersections.

The centroid of the axial forces occurs off the center of the propeller for any cyclic pitch amplitude greater than zero. For example, consider the sketch to represent the after propeller, looking forward. The ship is making a turn to port, hence the blades at the top have increased pitch; those at the bottom decreased



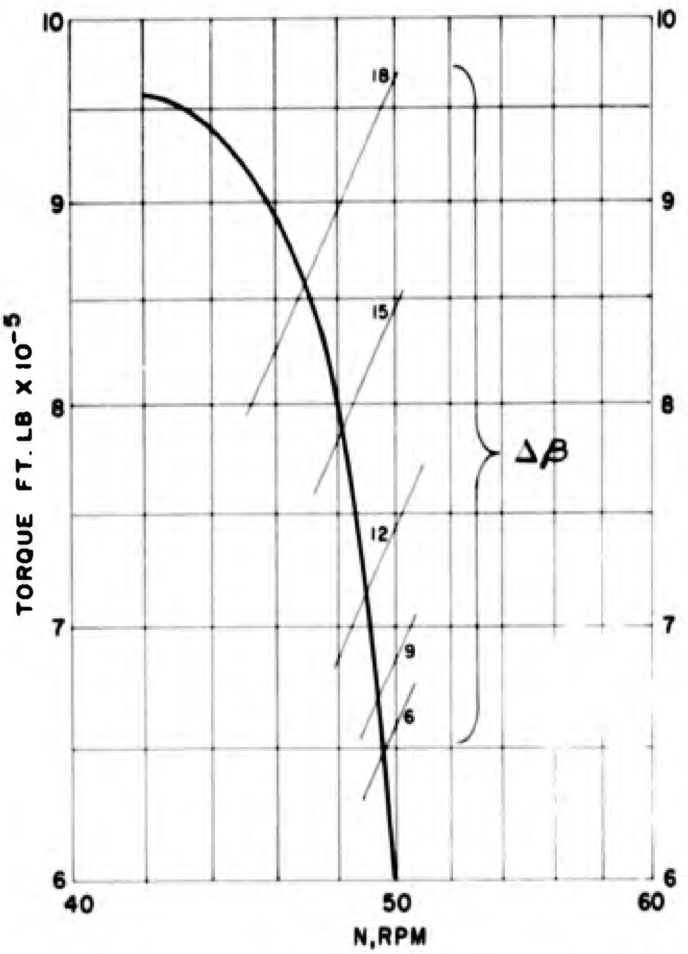
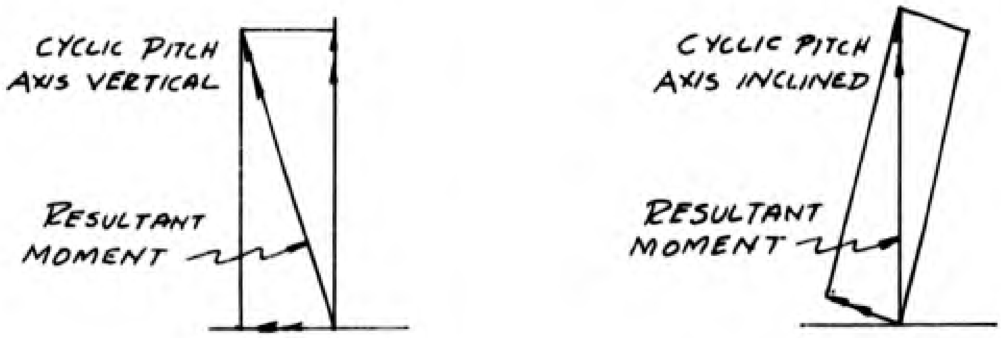


Figure 54. Propeller and Motor Torque vs. Speed Curves

pitch. The transverse component pushes the stern to starboard. The vertical components are cancelled, but the longitudinal force components are larger at the top of the propeller than at the bottom. The center of thrust is therefore above the ship's centerline, producing a pitching moment. Similar consideration of the forward propeller will demonstrate that the pitching moments act in the same direction. The data given in Table 2 refer to one propeller; the total pitching moment on the ship is therefore about two times these figures. The total pitching moment is only about 1/8 of the turning moment, however. (In making a dive or in rising the extraneous moment would, of course, be a turning moment.) In making a maneuver, the cyclic pitch axis can be rotated so that the vector sum of the moments acts about the desired axis, as shown in the diagram below. The ship control computer should be designed to incorporate this correction.



The assumption stated at the beginning of this section to the effect that each blade operated as in steady state must be examined. In the actual case of operation with some cyclic pitch superimposed on the mean pitch, the induced velocity would be somewhere between that for the mean and the instantaneous pitch. The effect of this simplification is that at maximum pitch, the hydrodynamic angle of attack is underestimated, forces would be slightly higher for a given angle than those calculated, but stall would occur at a smaller angle than predicted. At the minimum pitch position, angle of attack would be overestimated.

Thus the performance calculated for ± 10° pitch variation, for example, would actually be obtained with about ± 9° cyclic pitch amplitude.

Table 2 - Forces and Moments Produced by One Propeller

	50 rpm			Motor Torque Speed Curve			
$\Delta\beta$	Thrust lbs.	Steering Force lbs.	Thrust Moment ft-lbs.	Rpm	Thrust lbs.	Steering Force lbs.	Thrust Moment ft-lbs.
0	84,000	0	0	50	84,000	0	0
3	83,420	8,600	135,450	50	83,420	8,600	135,450
6	83,030	17,970	286,260	49.5	81,000	17,530	279,500
9	82,200	26,820	418,590	49.3	79,500	26,000	405,000
12	81,490	34,570	550,210	48.9	77,850	33,000	525,000
15	79,890	39,860	673,100	48.1	73,900	36,850	622,000
18°	77,970	43,380	788,720	47.1	69,000	38,350	698,000

OTHER CONSIDERATIONS IN BLADE POSITIONING

There are a total of four parameters which determine propeller performance:

Collective pitch
Cyclic pitch magnitude
Cyclic pitch axis
Propeller speed

In the absence of a thorough hydrodynamic study when this investigation was started, a completely flexible approach was taken; each of these four parameters can be varied continuously and independently of the other three. However, when the hydrodynamic study is completed, it may be possible to simplify or restrict the control system. For example, operation with only two discrete collective pitch angles might be satisfactory.

The brief hydrodynamic study just described is not strictly within the scope of our work, but was necessary in the absence of other information (which is being obtained in a concurrent study) to provide a sound basis for pitch changing mechanism design. A position type of system was selected as the most obvious and straightforward approach, but other approaches should be considered by those making the major hydrodynamic study. For example, in a position type of system, the positioning accuracy must be in the neighborhood of $\pm 1°$ because the load assumed by the propeller blade is very sensitive to angle of attack. However, if instead of a position being imposed upon the blades a torque were imposed, the desired loading would be more readily attained because the loading is a much less sensitive function of torque than position. This could lead to simpler hardware and less critical control systems for pitch changing.

U413-62-092
General Dynamics/Electric Boat
Groton, Connecticut
First Interim Report On Propulsion
Machinery and Pitch Changing Systems
for the Tandem Propeller, June 30, 1961,
180 p. including figures and tables.

A study of a turboelectric submarine propulsion system incorporating propellers at both ends of the ship. Propellers are driven by free-flooding motors located in the propeller hubs. Propeller blades are arranged to be controllable in pitch individually, and hence provide directed thrust for ship control. Design requirements for machinery and pitch changing systems are developed, and a feasible design is evolved.

U413-62-092
General Dynamics/Electric Boat
Groton, Connecticut
First Interim Report On Propulsion
Machinery and Pitch Changing Systems
for the Tandem Propeller, June 30, 1961,
180 p. including figures and tables.

A study of a turboelectric submarine propulsion system incorporating propellers at both ends of the ship. Propellers are driven by free-flooding motors located in the propeller hubs. Propeller blades are arranged to be controllable in pitch individually, and hence provide directed thrust for ship control. Design requirements for machinery and pitch changing systems are developed, and a feasible design is evolved.

U413-62-092
General Dynamics/Electric Boat
Groton, Connecticut
First Interim Report On Propulsion
Machinery and Pitch Changing Systems
for the Tandem Propeller, June 30, 1961,
180 p. including figures and tables.

A study of a turboelectric submarine propulsion system incorporating propellers at both ends of the ship. Propellers are driven by free-flooding motors located in the propeller hubs. Propeller blades are arranged to be controllable in pitch individually, and hence provide directed thrust for ship control. Design requirements for machinery and pitch changing systems are developed, and a feasible design is evolved.

U413-62-092
General Dynamics/Electric Boat
Groton, Connecticut
First Interim Report On Propulsion
Machinery and Pitch Changing Systems
for the Tandem Propeller, June 30, 1961,
180 p. including figures and tables.

A study of a turboelectric submarine propulsion system incorporating propellers at both ends of the ship. Propellers are driven by free-flooding motors located in the propeller hubs. Propeller blades are arranged to be controllable in pitch individually, and hence provide directed thrust for ship control. Design requirements for machinery and pitch changing systems are developed, and a feasible design is evolved.

UNCLASSIFIED

UNCLASSIFIED