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Selection of a Hydrofoil Transmission and Propeller System for the General Electric LM-2500 Gas Turbine

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SELECTION OF A HYDROFOIL TRANSMISSION AND PROPELLER SYSTEM FOR THE GENERAL ELECTRIC LM-2500 GAS TURBINE

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Abstract

This paper presents the results of propulsion system design studies for large hydrofoils utilizing the General Electric LM-2500 Gas Turbine drivinq mechanical transmissions. The effects of propeller operating characteristic on ship performance and transmission design are discussed. It is shown that at the power and torque level of the LM-2500 engine, the design of the spiral bevel gears and associated bearing is the most critical desiqn problem. The design of epicyclic gears, in the final reduction, are not expected to present any siqnificant problem, other than selection of the most efficient tvoe. A method for rapidly comparing efficiencies of epicyclic gear designs is explained. Several transmission configurations for a large conceptual hydrofoil are compared. Finally, it is shown that a variable pitch propeller can be effectively used not only for performance improvements at off-design conditions, but also in reducing the transmission design torque level.

I Introduction

Considerable effort is being devoted by the Navy to the design and evaluation of future high speed surface vehicles. One concept being seriously considered is a large ocean-going hydrofoil in the 1000-2000 ton displacement category. In addition to the already stringent powering requirements of hydrofoils in qeneral, the power levels required by hydrofoils of this size, present a challenge to the propulsion system designer.

This paper addresses the propeller selection and transmission design options available to the propulsion and power transmission designers. Specifically, the discussion centers on the design of a geared mechanical transmission system powered by the General Electric LM-2500 marine gas turbine engine for a propeller driven hydrofoil.

II. Discussion

Ship Requirements

The paramount criterion in the selection of a propulsion system is overall vehicle performance and efficiency, assuming that reliability and risk can be shown to be acceptable. Unlike conventional displacement ships, hydrofoils have at least three performance requirements that must be satisfied by the propulsion system as shown in Fiqure 1. A minimum thrust margin at maximum engine power is required at takeoff for acceleration and for variations in takeoff drag due to sea state conditions. Overall propulsive efficiency should be maximized and fuel consumption minimized at the foilborne cruise condition for maximum range (or minimum fuel). Finally, the propulsion system must provide the thrust required for maximum or dash speed, at or near maximum engine power and RPM. In actuality, the propulsion system will usually have to be a compromisebetween the three requirements.

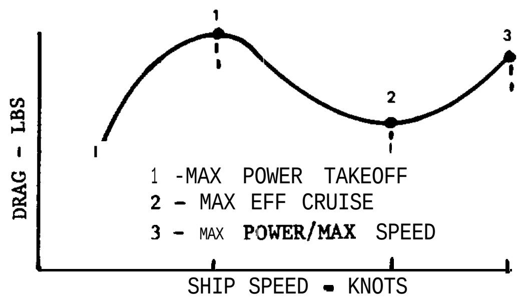


Figure 3. Powering Design Points of Hydrofoils

Transmission Requirements.

Since ship and propeller characteristics were and still are subjects of study, it was decided to proceed with conceptual transmission design studies independently. Therefore two requirements necessarily imposed on the transmission system were:

1) To absorb full power and RPM of one (1) GE LM-2500 Gas Turbine Engine; and,

2) To provide flexibility of design so that any selected propeller could be matched to it.

Maximum power and torque characteristics of the LM-2500 engine are shown in Figure 2. For in-house studies, the transmission design torque was chosen to be 480,000 lb-in, corresponding to 27500 HP at 3600 RPM. During these studies it was found that constant engine speed shafting down to the final propeller gear reduction results in near-maximum transmission1 weight. This fact led to the design concept of using a minimum risk and weight planetary gear box for the final and only gear reduction in the system. Furthermore, this concept provides the flexibility mentioned above, since development of higher risk components can be initiated long before the final propeller and planetary gearbox selection. A schematic of the resultant transmission concept is shown in Figure 3.

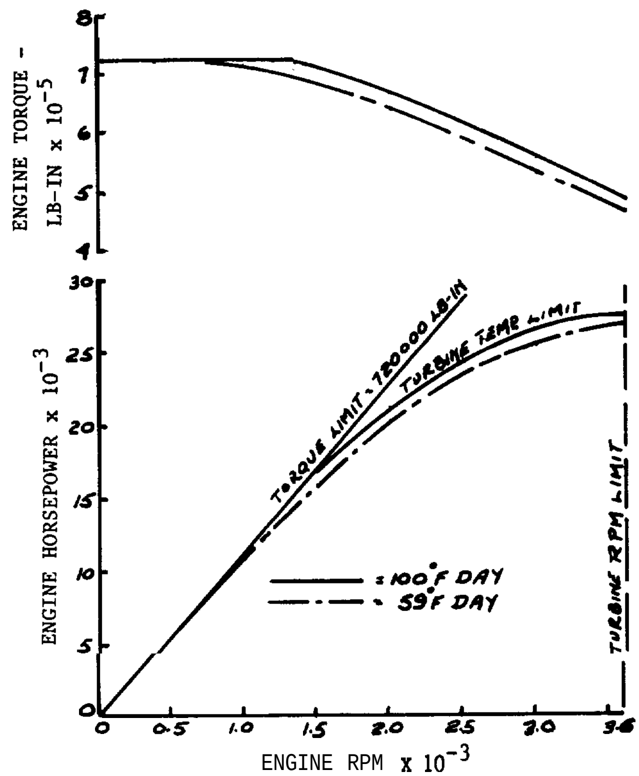


Figure 2. General Electric LM2500 Engine Power and Torque Characteristics

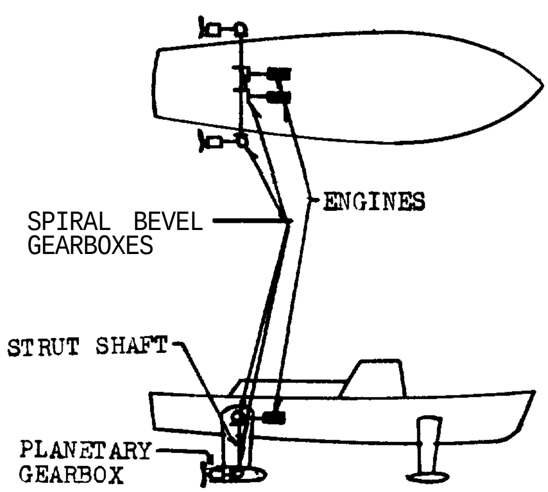


Figure 3. Conceptual Transmission System

Transmission Design

The transmission component designs presented below are the results of over two years of design effort on highpower geared mechanical transmission systems for hydrofoils.

Single Mesh Bevel Gear Box System

Figure 4 shows a simple straightforward system utilizing single mesh spiral bevel gears. Engine power is directed aft via shafting to the bevel gearboxes which in turn redirect power athwartship to shoulder bevel boxes at the tops of the struts. From these points power is directed down through the struts to bevel gearboxes in the pods. The pod bevel gearboxes direct power aft to the propeller shafts via reduction planetary gearboxes.

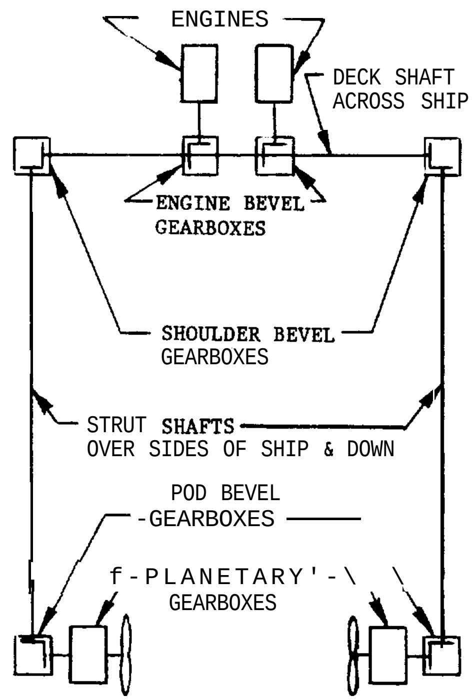


Figure 4. Single Mesh Bevel Gear System

Although the system is straightforward, the magnitude of the loads and shaft speed imposed on the spiral bevel gears by the power level transmitted, namely 27500 HP @ 3600 RPM per gear mesh, places this system well "outside the state-of-the-art". The few tests that were performed with bevel gears approaching this size, load and speed ended in swift disaster and were never pursued further.

Selection of bearings to support these gears presents an equally challenging problem. Tapered roller bearings appear to be the best choice but the larger of the bearings needed for this design are outside present experience, and would necessitate development testing. Qualified experts in tapered roller bearing design nevertheless feel that there would be a high probability of success.

The bearing design used in most gear driven hydrofoils to date has been of the cylindrical roller, ball thrust design. This design is impractical for the single mesh gear box because of the limitation of the ball thrust bearing. Hydrodynamic bearings have also been proposed for this application. The requirement to accurately position spiral bevel gears for good tooth contact under all load conditions, however, is incompatible with the design of hydrodynamic bearings which need adequate clearances.

Dual Mesh Bevel Gearbox System

Spiral bevel gear design can be brought "within the state-of-the-art" by utilizing dual mesh gears as in the transmission systems aboard the Denison, FHE and AGEH hydrofoils as shown in Figure 5.

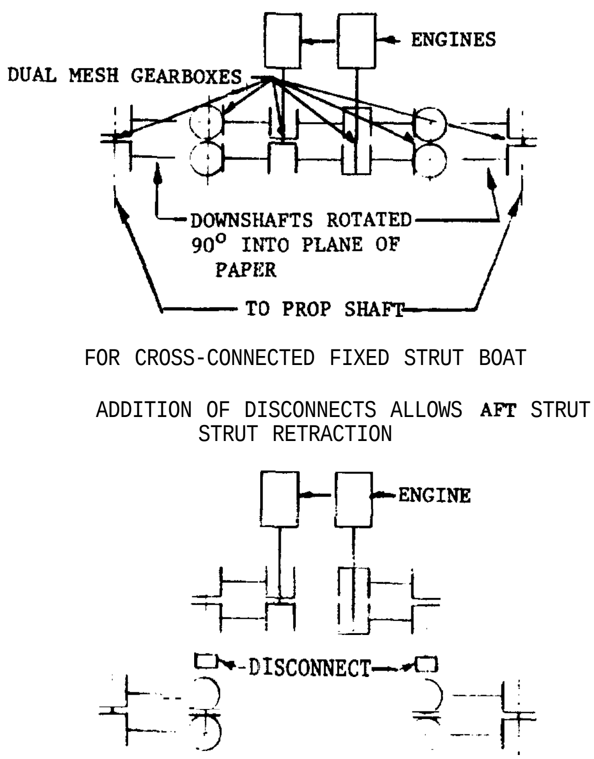


Figure 5. Dual Mesh Bevel Gear System - Adaptation of Denison. AGEH and FHE Design Type

Figures 6 and 7 are presented for comparison of single and dual mesh gear design parameters. These charts are on identical scales, but the range of values for single mesh are significantly higher than those for dual mesh design. Both figures represent transmissions with the same total torque capability, but the design torque of each part of the dual mesh system is based on a 52.5/47.5% torque distribution between the two paths.

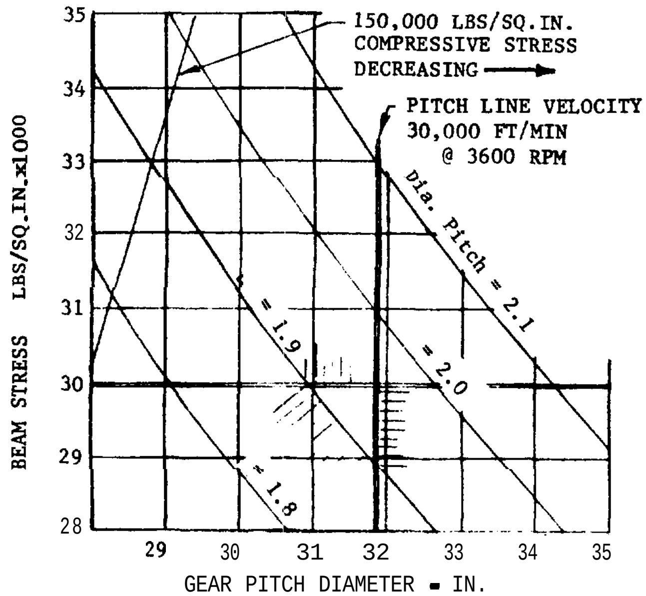


Figure 6. Single Mesh Spiral Bevel Gear Design Parameters - Torque = 481,440 LB-IN

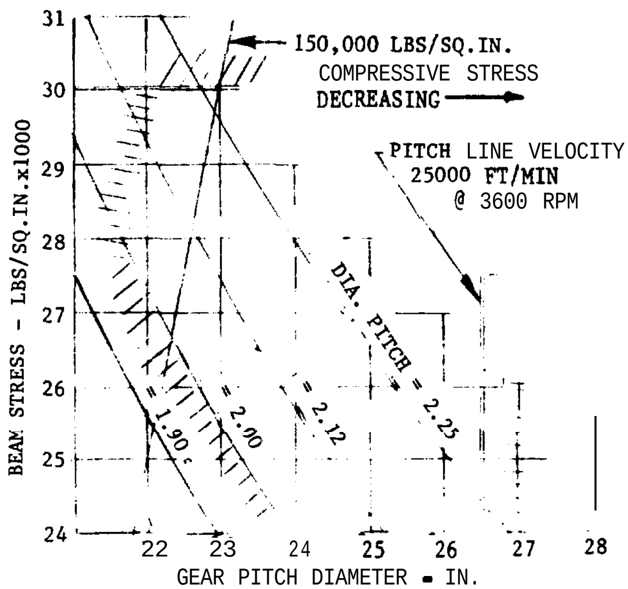


Figure 7. Dual Mesh Spiral Bevel Gear Design Parameters - Torque = 252,525 LB-IN

The shading on both figures encloses areas of possible interest, and the applicable range of design parameters, if these gear systems are to be built. To make comparisons between the figures easier, lines at 150,000 psi compressive stress and lines of 25,000 and 30,000 fpm pitch line velocities for dual and single mesh gears respectively, have been entered. Without implying any state-of-the-art limits on the various design parameters, it can, nevertheless, be seen by anyone experienced in hydrofoil transmission systems, that the Single mesh design needs a large amount of developmental work, whereas the dual mesh system is much closer to present capability.

The required bearings for the dual mesh gearboxes have a much higher confidence level than those for the single mesh design because of their lower loading and diameter.

Operational Requirements

It has been established in the above design discussions, that the component development of the dual mesh transmission is closer to the present "state-of-the-art" and should result in a lower risk system. Additional ship operational requirements, however, will influence the design of the dual mesh system affecting its complexity and reliability. Two such requirements are for retraction and cross-connection of engines. Figures 8 through 11 show some additional dual mesh transmission configurations designed for both strut retraction and engine cross-connection. These dual mesh systems are almost as unattractive in their complexity as the single mesh system is in the level of its bevel gear and bearing design parameters. The configuration shown in Figure 11 is considered the most attractive dual mesh system, and detailed hardware design layouts of the engine gearbox with two helical splitter gears, and pod and shoulder bevel gear boxes have been completed.

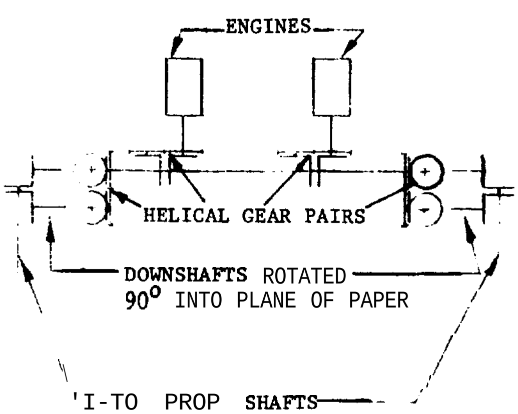


Figure 8. Dual Mesh Gear System - Power Split by Helical Gear Pairs

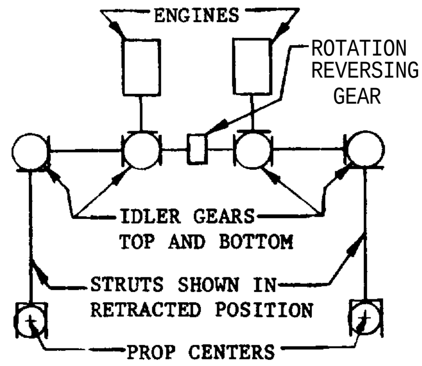


Figure 9. Dual Mesh Gear System - One Half Load per Tooth Mesh

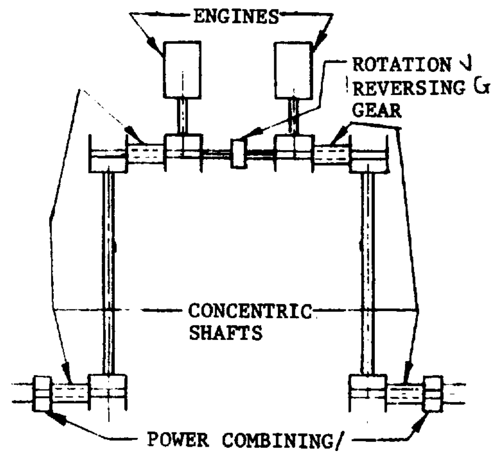


Figure 10. Dual Mesh Gear System - Concentric Counterotating Shafts

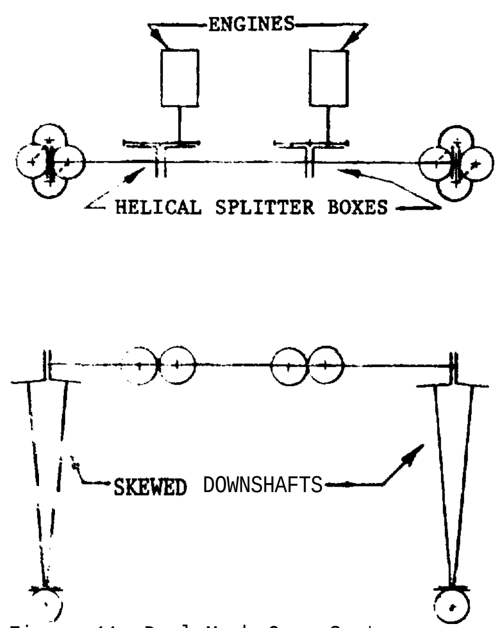


Figure 11. Dual Mesh Gear System - Helical Splitter/Skewed Shafts

Planetary Gearbox Design

Planetary gear design is considered to be within the present "state-of-the-art" for this application. The type of epicyclic gear - simple planetary, star, compound star, solar or free planet - best suited for a hydrofoil transmission is, however, subject to debate. In house studies show that the multiple stage simple planetary type is the most attractive because of its small diameter and light weight.

The Equivalent Mesh Method was used to rapidly compare the efficiencies of the several types of planetary gears. In this method the total power transmitted through all the gear teeth in the gearbox is calculated for 100% gear efficiency and divided by the input horsepower. In a parallel shaft spur gear mesh the engagement velocity and tooth load and, therefore input or output power for each gear are identical. Hence this gear system represents the simplest case for the method.

This single mesh unit is used as the basis for comparison of planetary gears in which the calculated power in all the tooth meshes is always greater than a single mesh unit. A step by step procedure outlining the use of the equivalent mesh method follows:

1) The power being generated at every tooth mesh in the gearbox is calculated by determining the tooth load and engagement velocity at each mesh. The tooth engagement velocities for complicated epicyclic systems are best determined by applying a single fictitious RPM to the entire assembly so-that, theoretically, the planet gear centers are stationary. In this manner the actual engagement velocities are clearly evident. Multiple stages are handled separately. Tooth loads are determined by treating the gears as levers.

2) The product of tooth load and engagement velocity is the power in ft.-lbs/min. at each mesh and the sum of all meshes is the total power transmitted by the gears in the gearbox. Buckingham calls this the "potential power" of the gearbox. Other terms such as "locked in power" and and "induced power" refer to the same'value.

3) The total horsepower generated within the gearbox divided by the input horsepower yields the equivalent number of gear meshes in that particular gear box.

4) The equivalent number of gear meshes multiplied by an appropriate loss factor (.5% to .75% for high quality gears) yields the probable power losses in the gear box.

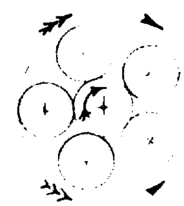
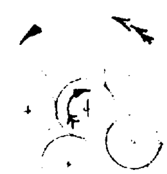
In the following example a 4.0:1 ratio simple planetary is compared with a star gear reduction of identical size, by means of the equivalent mesh method.

COMMON CRITERIA

Sun drives in each
Input is 15,232 horsepower @ 4,000 R.P.M.
Input torque = 240,000 Lb. In.
Dia. Sun = 12 In.
Dia. Planets = 12 In.
Dia. Ring = 35 In.

Total tooth load between sun and 5 planets = $\frac{240,000}{6}$ = 40,000 lbs.

Total tooth load between ring and 5 planets = 40,000 lbs.

Simple Planetary	Star
Carrier Output	Ring Output
	
Fictitious RPM = 1,000 on whole assembly	
4.0:1 RATIO	3.0:1
1,000 RPM OUTPUT SPEED	1,333 RPM
Fictitious sun RPM = 4,000 - 1,000 = 3,000	
Fictitious ring RPM = 0 - 1,000 = -1,000	
Fictitious carrier RPM = 1,000 - 1,000 = 0	
Engagement velocity = $\frac{12}{12}$ X X3000=9425 ft/min.	Engagement velocity = $\frac{12}{12}$X X4000=12566 ft/min.
Total power at sun and ring mesh = 2(40000X9425)= $7.54X10^8$ ft. lbs. =22,848 HP	Total power at sun and ring mesh = 2(40000X12566)= $1.00528X10^9$ ft. lbs. = 30,463 HP
Equivalent meshes = $\frac{22848}{15232}$ = 1.5	Equivalent meshes = $\frac{30463}{15232}$ = 2

The total expected loss for these two gear systems with a .75% loss per mesh is 1.125% for the simple planetary and 1.5% for the star. The results show that the simple planetary is more efficient, provides more ratio for a given size and the planet speeds about their own Centers are lower. If the above comparison were to be made on the basis of equal ratio, that is 4:1 for the star, with the same input conditions, the star ring and planets would be larger and with space for only four planets. This would necessitate an increase in the face width of the star if stresses were to be kept the same as the planetary. On this basis the weight and size of the star gearbox would be greater than that of the simple planetary.

Propeller Requirements

A summary of propeller parametric studies for large hydrofoils is presented in Figures 12 and 13. In Figure 12 it can be seen that propeller RPM's and consequently overall transmission gear ratio vary significantly with propeller selection and ship design speed. Weight variation due to gear ratio selection is minimal in the proposed transmission concept, since the only component weight change is in the lightweight planetary gearbox. Furthermore, any weight increase is negligible when compared to the improvements in propeller efficiencies at low propeller RPM's and the resultant increases in ship size and range, as shown in Figure 13. Although propeller selection is not based solely on efficiency, the transmission concert can accomodate the full range of expected propeller speeds, from subcavitating to supercavitating operation.

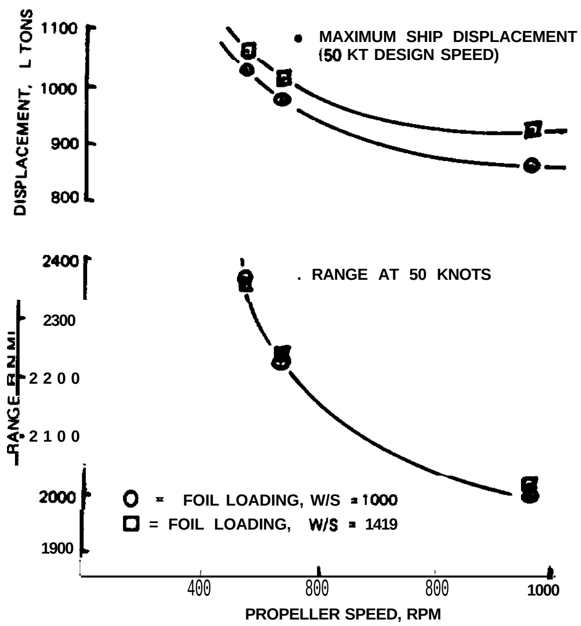


Figure 13. Hydrofoil Size and Range Summary

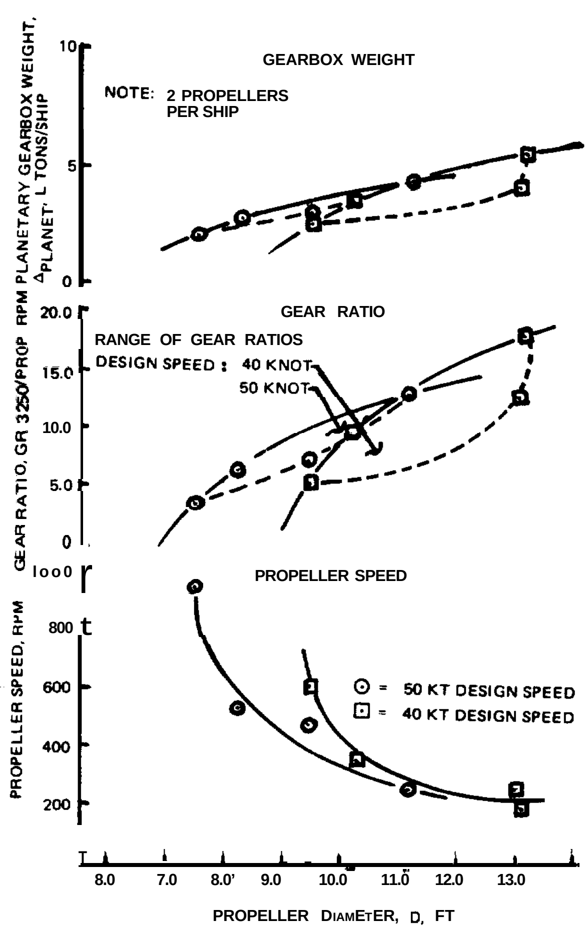


Figure 12. Propeller/Gear Ratio Summary and Planetary Gearbox Weights

Propeller/Engine Match

Since the design torque of the transmission is considerably below the maximum available from the engine, selection of the propeller and gear ratio must ensure the avoidance of overtorque conditions during normal ship operations at or near maximum engine power. To illustrate this point, a propeller and gear ratio are selected for a 1000 ton hydrofoil with drag characteristics as shown in Figure 14. An 8.14 foot diameter propeller based on the KaMeWa 398-B series was selected as being the optimum size for the maximum range cruise condition. The engine match at the three design condtions with this propeller are presented in Figure 15. It can be seen that a fixed pitch propeller of this design cannot satisfy the takeoff thrust requirement at when operating at the transmission torque limit. A variable pitch version of this propeller, however, meets or exceeds all three requirements, when used with a 6.7:1 planetary gear reduction. This example emphasizes the fact that,in addtion to the performance gains resulting from increased propeller efficiency at extreme off-design conditions such as takeoff, a variable pitch propeller can be used to match torque limited systems, without sacrificing cruise efficiency.

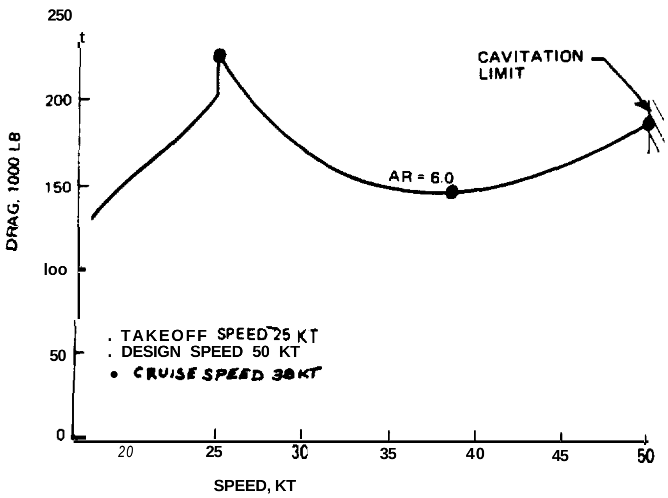


Figure 14. Drag Characteristics for 1000 Ton Hydrofoil

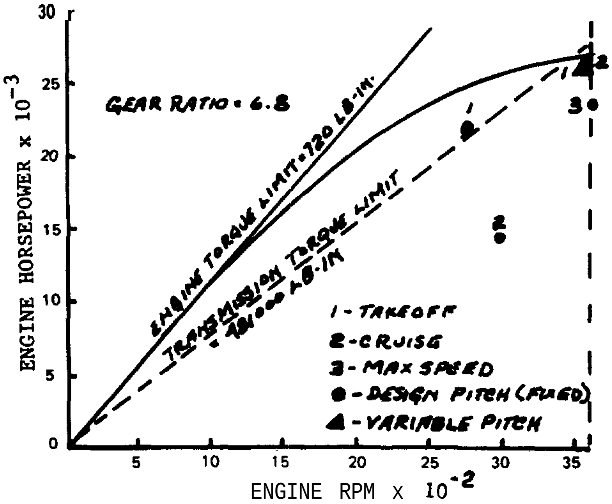


Figure 15. Engine/Propeller Match

III. Conclusions

Although the effort needed to develop a 25,000 horsepower mechanical transmission system for large hydrofoil application cannot be minimized, it has been shown that definite minimum risk paths are available for the design. The concept of "tuning" the transmission to the propeller requirement with-a planetary gearbox, allows the remainder of the system to be "universal" (for this particular power level). Furthermore, the proposed concept allows initiation of design and development of the high risk components of the system independently of the ship and propeller requirements. It has also been shown that the dual mesh design has lower component development risk but greater complexity. Although the single mesh bevel gear operating at 25,000 horsepower is beyond present experience, there is merit in developing this technology since new higher power (50,000 HP) engines are presently being developed. Finally, it has been shown that variable pitch propellers can be used to match torque limited systems without the performance penalties usually associated with fixed pitch propellers.