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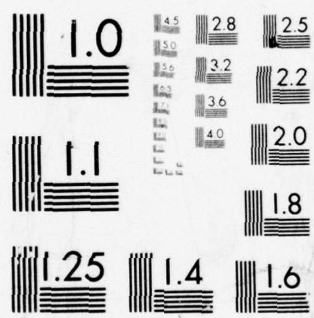

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Bethesda, Md. 20084

MODERATORS' REPORTS FROM THE WORKSHOP ON MECHANICAL TRANSMISSIONS FOR HIGH PERFORMANCE SHIPS

PROPULSION AND AUXILIARY SYSTEMS DEPARTMENT

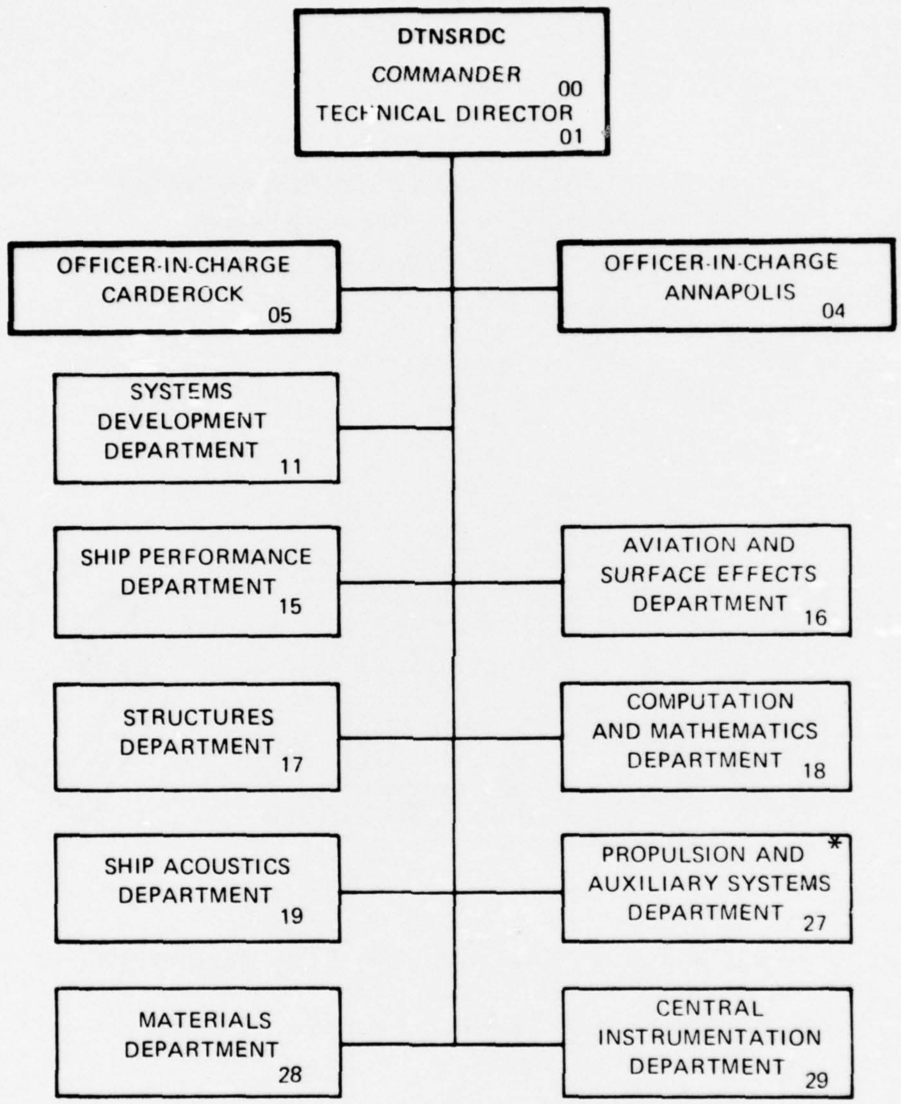
RESEARCH AND DEVELOPMENT REPORT

March 1977

Report PAS-76-47

MODERATORS' REPORTS FROM THE WORKSHOP ON MECHANICAL TRANSMISSIONS FOR HIGH PERFORMANCE SHIPS

MAJOR DTNSRDC ORGANIZATIONAL COMPONENTS



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On September 29, 30 and October 1, 1976 a High Performance Ship Transmission Workshop was conducted at DTNSRDC Annapolis, Md. The Workshop was attended by Defense Department and private sector personnel informed in various aspects of transmission design, development, manufacture and operational problem solving. This report is a summary of information gleaned from this three day effort.

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MODERATORS' REPORTS FROM THE WORKSHOP ON MECHANICAL TRANSMISSIONS FOR HIGH PERFORMANCE SHIPS

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ADMINISTRATIVE INFORMATION

The High-Performance Ship's Power Transmission Workshop was conducted as a part of NAVSEA (033) Task Area SF 43432301 Task 12501 Work Unit 2723-135.

BACKGROUND

The U.S. Navy's interest in high performance craft in recent years has created a requirement for marine transmissions with new constraints imposed. The components of these transmissions must be light weight and, therefore, highly loaded. Additionally, they must operate within a flexible structure and cope with the adverse marine environment. This transmission system is different from former transmissions of aircraft or marine type. Operational problems have resulted, and still further problems are anticipated as we move to higher power and larger craft.

This Mechanical Transmission Workshop, concentrating on applications directed toward high performance ships, was initiated by a letter on 4 August 1976 from the Commander of the David W. Taylor Naval Ship Research and Development Center (Appendix A). The workshop was conducted to bring experts from the government and industry together, to compile necessary information for support of Navy efforts in this area. Respondees to the letter of invitation were then sent further information presenting some background material and the mechanics of the workshop. Both letters are presented in Appendix A.

The decision to conduct a workshop was based on a perceived need as well as the success of recent technology workshops sponsored by the Naval Sea Systems Command. These workshops included a Gas Turbine Workshop held in March of 1974, a Hydromechanics Workshop held in August of 1973 and a Marine Engineering Workshop held in the summer of 1964. These workshops have provided broad-based information for the formulation of research and development programs.

As mentioned in the letter of invitation, the purpose of this Mechanical Transmission Workshop was threefold. A clear definition of the state-of-the-art was required. An evaluation needed to be made of what research and development would be required to solve current problems and meet future needs. Finally, criteria were required to specify and procure future transmissions.

The workshop was divided into three working groups. The first group was an Overview of Mechanical Transmissions. This group looked at future system requirements and the state-of-the-art definition. The group was co-moderated by Mr. Alfred B. Harbage and Mr. Leonard G. Schneider of David Taylor Naval Ship Research and Development Center. An Operational Problems Group was co-moderated by Mr. George Nagorny and Mr. Jules DeBaecke of the Naval Ship Engineering Center in Philadelphia. This group worked towards establishing research and development requirements to help solve future operational problems. Finally, Mr. Edmund J. Gutowski of the Naval Ship Engineering Center, Hyattsville, moderated the Specifications group; their purpose was to establish criteria for specification and procurement of high performance transmission systems.

The notes of moderators are presented as the bulk of the text of this report. The notes are basically as delivered by the moderators rather than attempting to carry a single format throughout the report. This was done to expedite release of this report. No attempt has been made to alter notes taken or to express any final Navy position on what transpired.

A verbatum transcript of the meeting was made. This has been received and is now being reviewed.

In addition to these notes, a breakdown of the attendees and which group they participated in is enclosed as Appendix B.

WORKSHOP

Registration for the Mechanical Transmission Workshop was completed on schedule, by 11:00 a.m. on 29 September. The final agenda (see page 4) was distributed to the participants.

Following the introduction of Capt. R. E. Sugg, Officer-in-Charge DTNSRDC's Annapolis Laboratory, Mr. L. F. Marcous, Head of the Propulsion and Auxiliary Systems Department at DTNSRDC opened the Workshop with welcoming remarks to the participants.

Dr. E. R. Quandt, DTNSRDC, introduced Capt T. L. Albee, Head, Advanced Technology Systems Division of the Research and Technology Directorate, NAVSEA, who presented the keynote address to the Workshop. Capt. Albee spoke on the Navy's interest in high performance craft and the related transmission problems with this type of craft. He pointed out that each craft must have some unique capability to justify its use. Transmission problems result from the constraints of large reduction ratio and confined space requiring highly loaded gears. These gears operate in a flexible structure and the harsh marine environment. Transmissions of 25,000, 50,000 and 100,000 hp were felt to be needed in a reliable configuration so that advanced ships would not be delayed for lack of a propulsion system.

Dr. Quandt then presented the background information and introduction to the workshop. The material presented was a survey of existing advanced ship propulsion systems, similar to that contained in the second letter sent to the participants prior to the conference.

The participants were divided into groups that met for the remainder of the Workshop. The participants of each group are listed in Appendix A. The remainder of this report consists of notes taken in each group. Final conclusions will be made following a review of the transcripts made from tapes that were recorded at each session.

OVERVIEW, ALFRED HARBAGE, MODERATOR

First Working Session

The session began with a discussion on materials. There was broad concurrence in the use of Case Carburizing 9310 and 9310 CEVM as the present state-of-art. Concerning ingot size, the participants concurred that up to 40-inch sizes could be handled using present techniques, quench presses, etc. Above 40 inches, distortion problems arise, i.e. case depth may be ground off. The desired case depth was discussed. For small diametral pitch, i.e. large teeth, a question remains as to just how deep a case is required, and whether it can be achieved. This led into a discussion on spiral bevel gearing. The subject of pitch line velocity was then brought up. Two figures from an AIAA/SNAME paper numbered 76-871 relating pitch, pitch diameter, beam stress, compressive stress, and pitch line velocity at 3,600 rpm was sketched on a display pad to show, figuratively, their interdependence or trade offs. At this point in the discussion, the bevel system of the AGEH was described. This system is a

MECHANICAL TRANSMISSION WORKSHOP

David W. Taylor Naval Ship R&D Center
Annapolis, Md.

AGENDA

Wednesday - 29 September

1030-1100	Registration
1100-1200	Welcome - L. F. Marcous, Head, Propulsion & Auxiliary Systems Department
	Keynote - CAPT T. L. Albee, Head, Advanced Technology Systems Div of the Research & Technology Directorate, NAVSEA
1200-1300	Lunch
1300-1400	Introduction and presentation of background information - Dr. Earl Quandt, DTNSRDC
1400-1630	First Working Session
1630-1700	Moderators' Meeting
1730-1900	Cocktail Party - Hubbard Hall, U. S. Naval Academy

Thursday - 30 September

0830-0930	Presentation to General Meeting - Paul L. Diehl, Diehl & Lundgaard, Inc.
0930-1130	Second Working Session
1130-1230	Lunch
1230-1300	Moderators' Meeting
1300-1630	Third Working Session
1630-1700	Moderators' Meeting

Friday - 1 October

0830-0930	Presentation to General Meeting - Procurement Experiences
0930-1130	Fourth Working Session
1130-1230	Lunch
1230-1300	Moderators' Meeting
1300-1430	General Meeting, Presentation by Moderators
1430-1500	Concluding Remarks

dual mesh system which operates at 1,500 rpm delivering 15,000 hp through the two meshes, i.e. about 7,500 hp/mesh. It was designed for 28,000 hp at 3,000 rpm, i.e. 15,000 hp/mesh. One attendee said that this system was suitable and under control. (This system uses 26-inch bevel gears with a 2 pitch). At this point the discussion shifted to epicyclic gears.

The question was posed as to whether the material problems for the epicyclic gear, required to reduce the speed to 350 rpm for a 50,000-hp reducer, were in hand. The comment was made that more face width is possible in straight gears than in bevel gears. A further comment was made that helical or spur gears are easy to measure and that bevel gears are more difficult to measure until a master exists; also, that above 36 inches in diameter, carburized and ground gears, will present problems.

The cause of failure of the AGEH gears while on test was queried. The response was that the failure was metallurgical. The discussion returned to the figures presented in the AIAA/SNAME paper 76-871 with a description of their value, using as an example a bending stress of 30,000 psi and a pitch line velocity of 30,000 feet/min.

At this point, constraints of an application for a high performance transmission system were presented. This was a hydrofoil having 50,000 total horsepower delivered by two 25,000 hp, 3,000 to 3,600-rpm power turbines, amd a drop to a pod with an inside diameter of 6 feet and a length of 30 feet with a 350-rpm propeller load. It was quickly mentioned here that 7.1 was the limit for planetary reducers with multiple planets and that they were sun gear limited. The discussion returned to bevel gears and there was a consensus within the group that bevel gears were a problem. At this point, some values taken from a letter to Mr. Bob Johnston, Manager (Code 115), Hydrofoil Development Program at DTNSRDC[(1)] were presented. These used limits of 30,000 psi in bending stress, 200,000 psi in contact stress and 25,000 feet/min. pitch line velocity.

Diameter (in.)	Velocity (FPM)	RPM	Initial Rating (hp)	Potential Rating (hp)
34	32,044	3600	43,015	50,184
	25,000	2809	33,564	39,158
30	28,274	3600	36,268	42,313
	25,000	3183	32,067	37,412
26	24,504	3600	26,911	31,397

The potential rating was based on 35,000 psi bending stress and 250,000 psi contact stress limits, and, in all cases, used a 1.1 load distribution derating factor. The pitch of the teeth was 1.267 for the 34-inch gears, 1.265 for the 30-inch gears, and 1.423 for the 26-inch gears. All these figures refer to miter ratio and are per/mesh values. A copy of the letter was submitted to the moderator; other ratios were included. A comment was made that "Solar" had in operation smaller, more lightly-loaded gears operating at 37,000 feet/min. and that there may be a load trade-off for speed. The size of other bevel gears mentioned were: Marad 23", Gleason 24", AGEH 26", Canadian 14", HLH ?, and that these have seen very little operational time. Again, it was stated (probably in response to a question) that a

(1) Gleason Works ltr, File 2-4-5 of 4 Feb 1974

33,500 hp/mesh is required for 34-inch gears. Also, (probably in response to a question), the AGEH gears were based on root stress built for and tested at 3,600 rpm.

The comment was made that too much weight has been removed from the rotating parts, and that increasing the weight will improve its reliability. More attention must be given to stronger wheels and cases. It was then stated by one attendee that the state-of-the-art was a bending stress of 30,000 psi, a pitch line velocity of 25,000 feet/min., a compressive stress of 200,000 psi, and scoring index of 30,000, and a development program should include contact pattern at low speed, including stress measurements. Another member, referring to the material in the letter to Bob Johnston, mentioned the potential of 40,000 hp/mesh of the 34-inch bevel gear at 25,000 feet/min. In response to the possible value of scale testing, there was a very firm consensus that for the bevel gear problem there would not be anything gained in an R&D program based on scaling. A comment was made that bearings were failing at 1/10 B10 life. The question as to positioning accuracy of bevel gears was brought up. The response was 5-10 thousandths from one member, and 2 thousandths of an inch from another. One member of the group asked whether a 1.2 pitch gear could be made.

Second Working Session

This session began by directing discussion toward considerations required to permit motion between components or within the system of components (flexibility) or what is the state-of-the-art toward flexibility of hard transmissions.

The discussion began with the comment that the tooth type couplings were designed to take up to 1/2° misalignment in one particular application, and possibly the misalignment is greater than 1/2°. A comment followed that sliding velocities greater than 5 in./sec. will result in problems even for fully submerged fully crowned tooth forms. In answer, it was stated that in the single shaft (referring to a particular Grumman application) hydrofoil there was no problem. Another attendee stated that there seems to be no problem in maintaining alignment. Another comment, directed toward helicopter applications, was made that for long runs where flexibility is required, intermediate bearing, coupling, and bulkhead mounts are utilized, but at the expense of complexity and weight. It was then stated Bendix type couplings were all designed to take up to 1/3° to 1/2° misalignment per disc, but series arrangements increase this and arrangements have been operated at up to 8° misalignment. The comment was made that Bendix couplings were larger, and in a pod where they are located off the pod axis, there might not be enough room for such couplings, as Bendix couplings are simpler but larger. The question was raised, as to flexurable type couplings, whether possibly fatigue was due to water or saltwater environment. The response to this was that little practical experience existed, but that they could be protected by paint and possibly other materials such as a titanium, 6 aluminum, 4 vanadium alloy could be used. More work is needed but good fatigue and corrosion resistance could be obtained. Titanium may be a good compromise having a light weight. It was then brought out that Zurn has a 5° misalignment coupling. It was also brought out that the more misalignment the couplings can handle the easier the overall design is. It was brought out by one member of the group who had experience in the design of the AGEH that the motions in service were not as severe as were calculated. There was consensus that the coupling problem was not nearly as severe as the large bevel gear problem. It was stated that deflections must be determined. In a particular case, it was stated that bearing failure was observed but not failure of the coupling (what this was in reference to was missed by the recorder). The question was asked if there were any torque limiting aspects to couplings. Another attendee stated, no. Another member stated that Bendix couplings had operated to 100,000 hp at 3,600 rpm with no problem. It was stated that the system can be designed to limit the deflection to 1/2°. It was stated that flexurable couplings cannot take much axial motion. It was then stated by an attendee that couplings still are a problem, needing close attention during the design process, and possibly require R&D work.

The discussion was then directed toward casing mounting and structure. It was stated that light weight was not all that important and weight should not be skimped on at the cost of decreased reliability. The comment was made that the subject of casing design needs attention, but finite element analysis is not needed, common sense is the need. The further comment was made that a welded steel casing is better protection inside of gear case and more compatible with bearings and gears. Another attendee responded that large gears require exacting designs, not artistic judgments. Still another attendee expressed the surprises they received on using finite element analysis, and supported its use at times. It was stated that if weight is not important, what flexibility is needed. It was stated by another member that gear boxes have not used sophisticated enough design methods. A member of the group conveyed his disagreement with the statement. It was stated by another member that the methods for sophisticated design techniques exists. It was stated by another that the AGEH boxes were made of welded T-1 plate, that no finite element analysis was used, that stress was not a primary concern but that deflection was, and the boxes were fine. He went on to say, the FHE used 1/8-inch plate and was still OK. Another attendee again advocated the use of finite element design techniques along with old time trial tests. The subject of aluminum versus steel was discussed and it was stated that aluminum has some advantages such as stiffness through thickness increase. A response to this included the statement that attachment problems for bearings and fasteners existed in aluminum casings. There was a consensus that the finite element design technique was an available tool in design. Where and when it should be used was not fully agreed upon.

A discussion of the necessity of weight reduction followed. The comment was made that reduction in shafting weight helps critical speed and further reduces weight of supporting structure, and further mentioned composite shafting in this line. Someone else, at this point, stated that weight was not important and another said that you do not sacrifice reliability for weight. The comment was introduced that more latitude in so far as weight allocation is needed for gears (transmissions) and, also, the allocation of time for problem correction is at times arbitrary. Another member suggested that cost per pound should be used, or a cost per pound of reduction should be used.

The discussion then was directed toward arrangements of gears. Comments suggested that combining could be done with bevel or offset gears. The idea of using vertical drive turbines was brought up and for a 50,000 hp combining power turbine, and further that bevel gears will be required in the "Z" part of the arrangement in any case. It was a near consensus that no major problems exist in combining gears except in using large bevel gears.

Third Working Session

This session opened with the discussion of a "Z" arrangement in which a single input bevel meshes with two bevel gears. The shafts connected to these gears are concentric, rotate in opposite directions, and go down to the pod. There they connect with bevel gears which mesh with a single output bevel. The comment was made that the bearings between shafts would rotate at double speed. It was further commented that strut deflection must be dealt with, creating a difficult coupling problem, and further, that the difficult bearing mounts would make the shafting and struts larger. Someone stated that cumulative gear error causes a cyclic load, and also there would be difficulty in load sharing. Someone else said it was a locked train system and both shafts need similar spring constants. The final words on the subject were that the simplest system should be used. The situation should not be made worse by complicating the problem as such a system would.

The next subject brought up was the possibility of using a DuPont Vespel plastic insert of finite thickness between the teeth of a tooth-type coupling. One attendee stated that lubricant would still be required for the removal of heat. Another said it would be OK for small coupling work, still another said that it would change torsional stiffness.

The question of the use of solid film lubricants for couplings was raised. The response was given that the Hertzian stresses between teeth were too high for solid lubricants.

The discussion was then directed toward the final reduction gear, epicyclic and/or differential types. The comment was made that the biggest problems with planetary gears were the bearings and centrifugal force. There was a consensus at this point that there was no insurmountable problem with the final reducer. It was stated that the Curtiss-Wright gear box 40,000 hp 3,600 rpm to 900 rpm built for the Navy had about 900 psi bearing load on the planet bearings, and that the MARAD planet bearings had about 500 psi. It was suggested that R&D effort should be undertaken in the area of high pressure sleeve-type bearings which are the type used.

The next item discussed was the torsional vibration which may be encountered with propellers. It was stated that a semisubmerged propeller may have torsional variations in excess of 30-40% of full torque, and that other propellers also have considerable torsional variations, although not so large. The comment was made that an attempt had been made on propeller blade tests, but the instrumentation was destroyed at the very outset of the test. Someone else mentioned that it is the last gears that see all of the torque variation, but the bevel gears might also be affected. It was generally agreed that some R&D effort should be directed toward a 50,000-hp vibration reducer. The "Geislinger" coupling was mentioned.

The main thrust bearing was then discussed for a load of the order of 350,000 lb. It was stated that a 40,000-lb thrust bearing of the tapered-roller type ran to 40% overload tested at "Aerojet." There was general agreement that there was not a great deal of trouble in this area. It was commented that rolling contact bearings are available possibly staged, and that you could fall back to hydrodynamic types. There was consensus on the subject of thrust bearings.

In the area of seals and keeping water out, it was stated that dual seals were used on one application supplying oil between the seals, also that O-ring seals, lip seals, and carbon face seals are used. Another comment was to use Waukesha seals, which are OK and very reliable low speed seals, for the propeller shaft, and to separate the seal oil system from the rest of the oil system. It was generally agreed that shaft seals (propeller penetration) were not a major problem. It was suggested that a closed venting system be used and that oil should be cleaned after shutdown. It was also suggested that the oil be heated to remove water. The response to this was that the salt may be retained. It was brought up that the whole lube system was a problem in practice and it was difficult to maintain pressure and temperature to the desired level in an extended, complex system. It was also mentioned that in as large a system as this will be, the system could be broken up into completely separate smaller systems, such as having the pod oil system completely separate and the shaft seal system completely separate. Another member mentioned it might be a good idea to pressurize the entire cavity with nitrogen or dry air. It was further stated that this method is sometimes used in the chemical industry. In response to the suggestion of nitrogen pressurization, it was brought up that oxygen is needed for oxide films to protect sliding or rolling surfaces and for satisfactory life of carbon seals.

A short discussion on polygon couplings followed. The manufacturing capabilities were questioned. Few machines are available and size is limited at present. The significant comment was made that splines were satisfactory when torque only is a problem, however, when a rotating force vector normal to the splined shaft surface is encountered, troubles arise, and the polygon does not change this. Hoop stress in thin shells of polygon couplings was mentioned. It was generally agreed that polygon splines were applicable in some particular applications.

The discussion returned to epicyclic gearing. The comment was made that differential planetaries may be smaller than conventional planetaries. Another comment was that scaling up was not strictly valid, flexibility and heat rejection were given as examples. It was suggested that for non-conventional types, small units should be built to prove design principles and establish design criteria. The comment was made that something can be learned through scaling. The further statement was made that you have a better chance of scaling up parallel shaft gearing. Another member stated that for new types such as some of the so-called bearingless differential systems, 1/5 scale models should be made. This was followed by another's voice of approval. This was followed by the statement that small scale allows shrinking things, which will help provide information. (Breaking something small is better than breaking something big). Another member voiced disagreement with the overall use of scaling. Another comment was made that high speed, high contact stresses, and high load is really what complicates the problem. During this discussion, it became obvious that some members were speaking toward scaling of conventional planetaries, some toward scaling bevel gearing, and others toward new concepts. The discussion was continued by the question put forth by one member, i.e. could the AGEH test facility be resurrected for evaluation of bevel gears? Another stated that to extend the state-of-the-art, use models, but application requires full-scale. Another person stated that two tests are required, critical component tests and proof of principle tests. Someone else stated that upping the size just a little with parallel helical gears was not a problem. This line of discussion was terminated with the criticism made by one member that in the past we scaled upward both technology and size.

The workshop then was directed toward what values in gearing represent the state-of-the-art. The following list was presented by one member:

Bending Stress (unit load/J)	30,000 psi
Scoring index	30,000 psi
Contact Stress	180,000 psi
K factor	500
Pitch Line Velocity	25,000 fpm
Hardness	58 to 62 Rc
L/D including relief space for cutter	1.5 for double Hel
	1.2 for single Hel
Helix angle	25° for double Hel
	10°-15° for single Hel

The comment was made that the AGMA level could be adjusted downward for longer life. AGMA provides 50,000 for bending and 180,000 in contact. Another member stated that "Gleason" can meet the AGMA 13 standard for a gear when shipped; AGMA 12 can be produced consistently (bevel gears). It was generally agreed that K factor was probably a poor criteria in comparison of stress value method, but that if it were not used, a modifier or other application factor would be required.

Fourth Working Session

This working session returned to a discussion of the reduction gear box because some members felt that it was not given enough consideration. A member of the group started by saying that 1,500 hours had been accumulated on each of two "Curtiss-Wright" gear boxes tested. These boxes were 40,000-hp planetaries with a 4:1 ratio and 900-rpm output. The planets were carburized and ground, the sun was carburized and ground, and the ring shaved and nitrided. There had been some planet bearing problems initially but were resolved early. The "Curtiss-Wright MARAD," a 4:57 ratio 40,000 hp planetary, with 105-rpm output, has accumulated 400 hours.

The comment was made by someone else that in these boxes, the babbitt is located on the inner surface of the gears and might present a problem in fatigue because it sees a once per revolution load. He further stated that the British have used babbitted pins with good success for years and had the technique for babbitting the pins in hand.

The subject of the "Curtiss-Wright free planet differential" gear arrangement was discussed and ratios around 8:1 and 20:1 had been tested. This type of gear was originally designed for helicopters. It was stated by someone else that flexible gears require a higher degree of accuracy and that manufacturers were getting away from the use of rolling contact bearings in planetaries.

The discussion was then directed to monitoring in service. A member stated that vibration monitoring on the SES 100A was provided. Web failure in the drop box was found just prior to complete failure through this system. Lots of harmonics indicate impending failure of the above web and the spline fretting at the bevel gear attachment. Foreign-object damage is not predicted on this system. It was also stated that high frequency, structure-borne noise is not that good an indicator. Someone added that one can tell a lot from vibration, but that a computerized system providing a trace is better, especially if a base line comparison is available. The question of the value of chip detectors was raised. There was consensus that they were very valuable. It was stated also that acoustic vibration might be valuable. A comment was made that microprocessors were getting less expensive and could find a place in monitoring system.

The remainder of this session was directed toward presentation of a priority list for R&D programs which would increase general confidence in producing a reliable transmission system for a 50,000-hp/shaft high-performance ship such as a hydrofoil from a low to a moderate to a high degree of confidence. These were listed on a chart (see figure 1) which was presented at the general meeting.

Very high priority (an absolute must) should be toward the full-size evaluation of 26- to 34-inch bevel gears, incorporating protection devices using state-of-the-art technology. Strain gaging would be required, employing a four square set-up with each corner a single mesh arrangement, each corner having varied geometry such as pitch, diameter, spiral angle, lubricant, material, etc.

Second on the list was bearing R&D which anticipates high load in bevel gear applications for rolling contact bearings, life requirements for highly loaded high speed bearings, race retention, and thrust absorbing bearings. Also included in this area is sleeve-type bearing work associated with the high loads incorporated in planet bearings. Corrosion of bearings would also be considered under this heading.

Third on the list for R&D was the known operational problems of general corrosion and fretting. Development of coatings such as black oxide, system watertight integrity, lubricant maintenance, system isolation, press fits, and geometry in shafts under races are to be considered. Overlap exists between the second and third priority items.

The fourth item deals with couplings, covering compatibility of diaphragm types in a marine environment, increasing the sliding capability of 5 inches/second in tooth-types, and investigation of alternate types.

The fifth item deals with development of application standards through full-scale experimental evaluation of such devices as clutches, brakes, over runners, disconnects, etc.

The sixth item deals with the R&D of vibration, temperature, chip detection, etc. devices and incorporation of these into an on-line monitoring system.

Within the seventh item, it is intended that for new epicyclic arrangements, an analysis of the design should be conducted in depth and stop just short of the small parts-detailing stage prior to building of hardware.

The eighth item considers the R&D work for propeller variable torque reduction.

The final item is minimal in that at the very least, a list should be compiled of available lubricants, their characteristics, and the requirements of the lubricant.

1. BEVEL GEARS
   26-34" Miter Ratio
   Experimental Investigation with State-of-the-Art Technology
   Application of Protection Devices

2. BEARINGS
   APPLICATION TO
   a. Bevel Gears
   b. Reduction (Epicyclic)
   c. Thrust Absorption
      Capacity – Life
      Corrosion
      Race Retention

3. CORROSION & FRETTING
   Coatings
   System Integrity
   Lubricant Maintenance
   System Isolation

4. COUPLINGS
   Marine Environment
      On Diaphragm
      Couplings
   Sliding Velocity of Tooth Couplings
   Alternate Configurations

5. CLUTCHES, BRAKES, OVERRUNNERS, DISCONNECT
   Experimental Evaluation of Full Scale Devices
   Development of Application Standards

6. APPLICATION OF PROTECTION DEVICES
   Vibration Monitoring
   Temperature Monitoring
   Micro Processors

7. EPICYCLIC GEARS
   Design Studies of Full-Size Arrangement stopping just short of detail design

8. TORSIONAL VIBRATION REDUCER
   FOR SEMISUBMERSIBLE
      PROPELLER & OTHER LOADS

9. LUBRICANTS
   Load and Temperature Characteristics
      Improve Washability without Loss of Washing Characteristics.
      Characterize Requirement. (What is needed?)

Figure 1. R&D Program Priority List Presented to the General Meeting, 1 Oct 1976

OVERVIEW, LEONARD SCHNEIDER, MODERATOR

Charts 1 and 3 (see figures 2 and 4) were developed from the discussions which are summarized in the sections headed Wednesday Session and Thursday Session. In the Friday session, the group developed Chart 2 (figure 3), R&D for Benefits from Lightening. There are no notes on this session, as Chart 2 is the result of the discussion and represents a consensus of the group.

Wednesday Session

Gearing. Present aircraft gear steel 9310 (AMS 6265) is applicable to Navy needs for high-performance gears up to 50,000 hp, with Vasco steels applicable to the next step where higher temperature operation could be required.

A problem lies in translating aircraft practice to large marine gearing where R&D is needed in processing and quenching to control heat treatment (case depth and distortion control). (The 36-in. diameter is OK, anything over that presents a control problem.)

One source reported successfully producing large turbine gears (40- to 60-in. diameter) from 4145 induction hardened under water (NATCO process) to 52-56 Rc case with good control of case depth and distortion control. This case hardness does not, however, reach value reached in 9310 carburizing steel used for high output aircraft gearing.

Cleanliness in processing steel is extremely important on performance. All inspection methods to insure cleanliness should be employed.

Use of corrosion resistant gear materials for marine environment would be an advantage but there is no knowledge of carburizing materials which would be effective. NASA reported a BG42 bearing steel (corrosion-resistant) which can be through-hardened and has good fatigue resistance. The use of newer corrosion-resistant bearing steels for gears is a possibility and needs to be pursued. With present high output gearing, scoring, not breakage or pitting, is the main problem. Bearings are often the limiting factor rather than gear materials.

The designer needs to know good values for material properties at long life (at high number of stress cycles) because high speeds representative of gas turbines mean very large number of stress cycles at relatively low hours of operation; also, properties under marine environment. The problem is usually more in good design (light-weight requirement drives one into a design problem) than in materials themselves, i.e., maximized design efficiency based on adequately cataloged material is the goal.

Couplings. – For slower speed shafting, dental couplings with hardened teeth can do job; limit misalignment to 5 in./sec. or less. Thomas disc coupling is very successful; it is reliable, requires no lubrication, has no wearing parts, and the shaft is held on center so it can run at high speed (properly balanced); it is limited to 2° misalignment with 3/4° good working value per disc pack. This type has also been used for high torque (4×10^6 lb. in. at several hundred rpm). Solid diaphragm couplings (Bendix type) are used by one manufacturer on all high speed applications. R&D is needed in notch sensitivity of disc materials (titanium). The diaphragm coupling will have a weight advantage over the dental. Coupling R&D must consider the effect of thermal growth of the machine set-up on the diaphragm, heat generation from flexing, and the effect on any isolation washers or mounts.

Thursday Session

Needed R&D
Couplings & Shafting. –

Shafting. – Fiber (graphite) reinforced drive shafts for aircraft operate over longer spans; stiffness to weight ratio is good. Reduced support bearing loads and weight as well as reduced number and size (weight) support struts are required. Costs are high in the aircraft R&D area, however, there is possibly good applicability to Navy since the shaft problem is to get steel with high yield strength and adequate stress corrosion capability.

Couplings. – The lower modulus of non-metallics (fiber-epoxy composites) are being considered because they have good fatigue strength and are corrosion free. In aircraft this is in the R&D area, but the technology is advancing. There is a possible future applicability to Navy.

Gear Arrangements. – Bevel gears represent smallest, lightest, most efficient right angle transmission. AGEH represents the highest capability now available, 20-25K hp per mesh. Presently the maximum grinding capacity is 36-in. diameter. Gleason is reported to be working on the development of a 40-in. diameter spiral bevel gear generating machine.

Gleason program can optimize tooth design, but there is a need to optimize design of webs and rim by finite element analysis. Furthermore, there is a need to do the same to identify case (mounting) deflections. The producing of bevel gears is now an art rather than a science, requiring successive regrinding and tooth contact pattern checks to get desired result.

Improved metrology capability is needed for spiral bevels (similar to what can now be done with involute spur and helical gears) to know what you have and assure the desired result.

New tooth forms for angle drives: Twin Disc has the manufacturing capability for "conical-helical-involute" V-drive gears (20-25° included angle) which gives involute form with all the advantages this offers for checking on standard machines. It cannot be used for large angles (e.g. 90°), but is of possible Navy use. Oerlikon system of gearing is worthy of consideration for 90° drive (Boeing Vertol).

Straight Thru Reductions: Multiple path drive train (of which epicyclic is an example) is the way to go for high hp reductions. 50,000 hp is within present state-of-the-art, but must be done with good design and care in all aspects. Flexibility in design is needed to allow load sharing and free planet gears (latter type of transmission in development by

Curtiss-Wright) are needed for high-capacity high-speed craft gear boxes. In pursuit of light weight, one company has gone to a titanium carrier.

Reversing capability in planetary needs R&D.

Lowenthal, NASA, stated some work on torque splitting drives, which also supply some redundancy in the drive system, has been done. It is worth looking into for new arrangement ideas.

Epicyclic ring gears. – No trouble getting up to 36-in. diameter internally ground; but above 36-in. there is a shortage of machine capability.

Vibration: Work is needed to study the effect of increasing machining accuracy on improvement in dynamic load capability.

CHART 1

R&D: Arrangements — 50Khp

Bevel Gears (≈90°) — 25Khp/mesh & up

- Tooth forms to accommodate displacement
- Improved metrology
- Design to maintain gear alignment in structure (gear box)
- Better definition of dynamic tooth stresses
- Vibration characteristics of bevel gears
- Definition of service factors for 50Khp bevels

Helical Gears (single)

(Design programs and measurement capability exist)

Double Helical & Herringbone

- Define dynamic effects with floating elements

General (applicable to all arrangements)

- Improved review of bearing application design
- Improved bearing rolling element and cage design
- Analysis for resonant frequency and provision for damping
- Analysis for web and rim stresses

Figure 2. Research and Development: Arrangements.

CHART 2

R&D for Benefits from Lightening	50Khp
(includes development of analytical methods in approach)	25Khp/mesh & up

1. Fiber (graphite) reinforced shafting (for operation over longer spans)
2. Non-metallic (fiber-epoxy composites) for couplings
3. Steels with high yield strength and improved stress corrosion properties for shafting
4. Use of titanium (define limitations for design configurations; methods of combating limitations)
5. Improved design criteria for shafting from fatigue considerations (refer to ABS Regs. for info)
6. Improved review of bearing application design (also listed under R&D Arrangements – General)
7. Improved corrosion protection for materials applicable to weight saving designs (including fretting)
8. Lubrication concepts for lightweight designs
9. Establish noise and vibration criteria (allowable levels) for light weight machinery concepts
10. Establish system integration design criteria for light weight craft machinery systems

Figure 3. Research and Development: Benefits from Lightening.

CHART 3

Desired Requirements, General Materials

50Khp
25Khp/mesh & up

Fatigue (bending) strength $\geqslant$ 9310 (AMS 6265)*

Surface (pitting) capacity $\geqslant$ 9310 (AMS 6265)

Stability (distortion in processing) $\geqslant$ Nitriding Stls.

Corrosion resistance $\geqslant$ Ti

Notch sensitivity resistance $\geqslant$ 9310 (AMS 6265)

Hot hardness = VASCO, CBS, M50, SUPERNITRALLOY

Cost (incl. fabrication) $\leqslant$ 8620

Availability in super quality $\geqslant$ M50

Weldability (to dissimilar metals) $\geqslant$ 9310 (AMS 6265)

*Most important (biggest payoff in attainment)

Figure 4. Desired Requirements of General Materials.

OPERATIONAL PROBLEMS, GEORGE NAGORNY, MODERATOR

Recommendations

The following list represents the recommendations of the working group:

Recommendation 1 – Conduct a workshop specifically on computer-aided design techniques. The purpose of the workshop would be to identify the state-of-the-art in computer-aided design of transmissions and to relate experience. Workshop to be made up of several meetings of a small group. Examples of necessary work:

- Fits required to eliminate fretting
- Bevel gear rim/support design
- Effect of changes in tooth contact on load distribution

Recommendation 2 – Develop a means to communicate such information as experience (problems encountered and solutions) to those concerned. (This could be done in the manner of NASA Monographs, for instance).

Recommendation 3 – Develop computer analysis techniques to establish optimal trade-off between bevel gear case deflections and tooth profile modifications.

Recommendation 4 – The computer techniques, mentioned above, should be specified in future contracts.

Recommendation 5 – Contracts should include design reviews to be conducted by qualified Navy engineers.

Recommendation 6 – Land-based testing should be conducted prior to shipboard installation that includes, first, the component only and then the component in the system. These tests should be performed with extensive instrumentation and should include environmental simulation. The environmental simulation should include torsional excitation, oil contamination, deflections, etc.

Recommendation 7 – Available techniques for performance monitoring should be documented. New devices should be developed as required.

Recommendation 8 – Available techniques for vibration monitoring should be documented. Develop recommended equipment for on-line monitoring and trend monitoring.

Recommendation 9 – Investigate alternative bevel gear geometries that are available and recommend development if indicated by studies.

Recommendation 10 – Pursue the use of sleeve bearings for spiral bevel gears.

Recommendation 11 – Document lube oil research within industry and government.

Recommendation 12 – Study coatings for corrosion protection.

Recommendation 13 – Document case-depth requirements of hardening for spiral-bevel gear applications.

Recommendation 14 – Study coatings to eliminate fretting.

I. Gear Tooth Problems

A. Item/Area – Reduction and combining gearing/separate helices

1. Cause/failure/problem – loosening and fretting
2. Solution/recommendations/comments – integral design shall be used unless analytical techniques show that the use of separate helices will not be a problem.

B. Item/Area – Right angle gearing/carburized teeth

1. Cause/failure/problem – tip cracking
2. Solution/recommendation/comments – define and document end chamfering to z-dimensional case. Develop for 3-dimensional case and document. This may be done with a document similar to NASA monograph.

C. Item/Area – Nitrided and carburized

1. Cause/failure/problem – case crushing
2. Solution/recommendation/comments – document desired case depth relative to tooth design in order to eliminate overlap in hardening.

D. Item/Area – High vibration/noise

1. Solution/recommendation/comments – consider use of "octoid" profile which is very sensitive to deflections. Other available and conceived profiles should be studied and the best selected for development. Example: constant relative radius at curvature profile, used in some commercial applications. It has less change in load distribution with variations in profile, lower sliding velocities and, in tests to date, substantially less noise. Can be made with minor changes to current Gleason machines and may be used immediately for parallel axis applications. *Noise studies* – NASA Langely – helicopter noise Hughes Helicopter, LOH Model Army Interim Report (Math Model to Reduce Noise).

E. Item/Area – Material

1. Cause/failure/problem – higher strength
2. Solution/recommendation/comments – VASCO tool steel is one of the best (must be metalurgically clean since inclusions at the surface can result in non-uniform carburization).

F. Item/Area – Bearings

1. Cause/failure/problem – use of sleeve-type
2. Solution/recommendation/comments – Would provide more damping and also give a stiffer support than rolling contact. If a less sensitive tooth profile is used likelihood of success with sleeve bearings is higher. Use of deaerator can reduce sump tank capacity to 1/3 of that required with natural deaeration. Designs are needed that can go to 1000 to 2000 psi. Pad type could minimize clearance/movements significantly over plain sleeve bearings.

G. Item/Area – Rolling element bearings

1. Cause/failure/problem – larger sizes
2. Solution/recommendation/comments – At larger diameters and high speeds internal loads due to centrifugal force become limiting. Tapered roller bearings can handle large loads, but lubrication must be handled by drastically different means (such as holes in inner race). Advantages of tapered rollers include lower power loss than ball bearings and the fact that two tapered roller bearings can be used instead of three ball bearings.

H. Item/Area – Design methods
   1. Cause/failure/problem – hacking/inadequate
   2. Solution/recommendation/comments – require analytical techniques for proper proportioning of rim cross-sections. Currently thickness equals three times tooth depth, this gives too heavy a gear for weight sensitive craft.

I. Item/Area – Tooth contact problem
   1. Cause/failure/problem – how to obtain better contact
   2. Solution/recommendation/comments – require analytical technique to permit good understanding of total deflection (gear case and tooth deflection) and their relationship to contact patterns. Present "development" method is too costly.

J. Item/Area – Size
   1. Cause/failure/problem – limitation
   2. Solution/recommendation/comments – presently Gleason can grind 34-inch diameter. Larger than 36-inch diameter would require four to five million-dollar investment.

K. Item/Area – Epicyclics
   1. Cause/failure/problem – high centrifugal loads on planet bearings/planet bearing deflections
   2. Solution/recommendation/comments – with proper analysis this should not be a problem. Must be carefully analyzed in light weight designs.

II. Lower Gear Case Corrosion

A. Item/Area – Environmental control
   1. Cause/failure/problem – salt water entry/fresh water condensation
   2. Solution/recommendation/comments – dry nitrogen with a sealed system. Use of deaerator or accumulator or both.

B. Item/Area – Seals
   1. Cause/failure/problem – salt water entry
   2. Solution/recommendation/comments – carbon face type used successfully on SES-100B and PCH-1 (two with 5 p.s.i. oil pressure between; monitoring of pressure drop and lube oil in accumulation indicates leakage). Seals should be located in area of low hydrodynamic pressure, where possible, so oil goes out rather than water in. Static seals must be used on lower housing joints. NSRDC seal handbook available.

C. Item/Area – Lube oil
   1. Cause/failure/problem – salt water entry/fresh water condensation
   2. Solution/recommendation/comments – better lube oil. MIL-L-24467.900772 2190 TEP with vapor phase inhibitor. NAPTC XAS 2354 has high corrosion resistance.

D. Item/Area – Monitoring techniques
   1. Cause/failure/problem – salt water entry/fresh water condensation
   2. Solution/recommendation/comments – monitor percentage water and replace oil if water exceeds 0.2%. If chloride content exceeds 12-15 ppm, look for salt water entry.

E. Item/Area – Materials

1. Cause/failure/problem – salt water entry/fresh water condensation
2. Solution/recommendation/comments – investigate use of black oxide coatings (can't be used on M-50). Black oxide not compatible with some lube oil additives. Phosphate coating and rust ban paint (applied within 24 hours, over sand-blast finish).

III. Fretting Corrosion

A. Item/Area – Fretting corrosion

1. Cause/failure/problem – high vibration/improper design
2. Solutions/recommendations/comments – need analytical method to assure optimum fit. One recommended approach for bearings is to analyze interference between inner bearing race and O.D. of shaft. Shaft thickness must be increased if hoop stresses exceed 20,000 psi. Use of coatings for marine use is considered to be only a temporary measure to inhibit start of fretting. Splines should be used with a double pilot. Analytical work is required to define use of splines when rotating force vector (as in case of spiral bevel or single helical) is present.

IV. Problem Detection Devices

A. Item/Area – Vibration monitoring

1. Cause/failure/problem – optimal types should be selected
2. Solution/recommendation/comments –
   a. many times elaborate vibration instrumentation of great cost, complexity has not provided necessary warning to prevent failure. Often airborne noise changes are the signal to impending failure. Simple, rugged system would be preferred.
   b. several monitoring levels are required
      (1) on line monitoring – simple, rugged, and effective
      (2) trend analysis – long range maintenance indications should be collected

B. Item/Area – Performance monitoring

1. Solution/recommendation/comments – lube oil pressure, bearing temperature, temperature of discharge oil, chip detectors with in-line screen, suspended particle detection (for particles smaller than in-line screen can detect), oil conductivity change, percentage of H_2O and Cl in lube oil, spectrographic oil analysis. Input torque monitoring was suggested as possibility, it had been found to change as a bearing fails.

SPECIFICATIONS & PROCUREMENTS, EDMUND J. GUTOWSKI, MODERATOR

I. Approach: Since it was impossible to cover Mil-G-17859, "Military Specification Gear Assembly, Propulsion (Naval Shipboard Use)" in every detail, major areas were selected for review by the Specifications Group. The following major areas were selected:

   a. arrangements
   b. gearing details
   c. tooth design details
   d. component fits and measurements
   e. lubrication
   f. gear components such as:
      1. clutches
      2. brakes
      3. couplings
      4. shafting
      5. bearings
      6. housing/casing
      7. seals
   g. test

   The procedure was to discuss the requirements of existing specifications (Mil-G-17859C) in the above mentioned areas; then describe the requirements in the proposed specification (Mil-G-17859D) and then to elicit comments as to the application of these requirements to high performance craft.

II. A summary of the recommendations for each area covered is as follows:

   a. arrangements – exclude specific arrangement requirements; instead, provide functional and environmental requirements such as weight, size, reliability, as per horsepower, etc.
   b. gearing details:
      1. materials – add aircraft material specifications, specify vacuum remelt steels especially in highly stressed areas
      2. life – 10^{10} cycles may be harsh requirement, rather than stress cycles, indicate allowable stress limits, provide operational spectrum
      3. forgings – no change recommended
      4. welding – no change recommended
      5. fasteners – no change recommended
      6. noise and vibration – deferred
      7. shock – deferred until hull resistance is determined
      8. piping – no change recommended
      9. balance – no change recommended
   c. tooth design details
      1. accuracies
         – K factor – eliminate and apply contact stress within the allowable limits of the materials
         – helix angle – do not specify
         – contact pattern – no change recommended
         – binding stress – material stress limit should control

– surface finish – 20 rms or better
– scoring – flash temperature and elasto-hydrodynamic film should be included in the calculations
– surface durability – no change recommended

general comments – accuracies as required by revised AGMA-12 should be applied. Stress limits should be derated by safety factors.

d. component fits and measurements – Do not specify fits of components, instead provide adequate interface requirements. Fits and tolerances of internal parts of a transmission should be left to the manufacturer.

e. lubrication – Recommendations were deferred until attendees have an opportunity to review the details of the proposed specification. One general recommendation was to add Mil-L-23699 gas turbine oil as an acceptable transmission oil. It was further stated that as a result of the higher operating temperatures, greater consideration should be addressed to incompatibility of the oil and materials.

f. gear components

1. clutches – Mil-C-18087 dated 1955 must be revised. If new types of oil are used, one must consider the effects on friction clutches.
2. brakes – The selection of the brake and the construction details should be left to the manufacturer
3. couplings – Mil-C-23233 would be acceptable with necessary exceptions for specific applications
4. shafting - no change recommended, however, stress limits should be reviewed
5. bearings – no change recommended; however, temperature sensing for rolling contact bearings should be added
6. casing/housing – no change recommended
7. seals – no change recommended

g. testing: The only apparent concern to the group was in regard to the spin test. It was recommended that a minimum load be applied to provide operational stability and the extent of the applied load would be determined by the manufacturer.

III. Conclusions: It was quite obvious to all participants that the present Mil-G-17859C is inadequate for high performance ship transmissions and the proposed Mil-G-17859 is too severe and restrictive. More specific conclusions are as follows:

a. Measurements should be specified as well as measurement techniques and fits should be avoided

b. Tooth accuracies and stress requirements should be provided, but adjustments should be made for specific applications. Tooth geometries should not be specified.

c. Specific arrangements should not be required, but environmental and functional parameters should be provided.

d. Navy design reviews should be initiated within two weeks of contract execution. The first review is to assure a thorough understanding of the requirements and subsequent reviews depending on the critical milestones established.

IV. Recommendations:

a. Develop new specifications for High Performance Ships (defining high performance)

b. Establish a specification committee for the purpose of developing a new specification. The committee to consist of Navy and Industry (both aircraft and marine)

c. That present procurement policy (shipbuilder as prime contractor) be continued, but with greatly increased Navy involvement in the preparation of specifications and periodic manufacturing reviews.

APPENDIX A

LETTERS TO PARTICIPANTS

DEPARTMENT OF THE NAVY
NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER
HEADQUARTERS
BETHESDA, MARYLAND 20084

ANNAPOLIS LABORATORY
ANNAPOLIS, MD. 21402
CARDEROCK LABORATORY
BETHESDA, MD. 20084

IN REPLY REFER TO:

2721:KTP
5050
4 Aug 1976

From: Commander, David W. Taylor Naval Ship R&D Center
To:

Subj: Mechanical Transmission Workshop

Encl: (1) Workshop Goals
(2) Working Group Subjects
(3) Tentative Agenda

1. The U. S. Navy's interest is moving increasingly toward high performance craft, as reflected by our current involvement with the surface effect ship, assault landing craft and hydrofoils. The propulsion systems of these craft require complex transmissions of high power density. In order to assure the availability of such transmissions for future craft, the U. S. Naval Sea Systems Command is sponsoring a three-day workshop on the subject of high performance mechanical transmissions. This workshop will be conducted at the David W. Taylor Naval Ship Research and Development Center, Annapolis, Maryland, from 29 September to 1 October 1976. The purpose of the workshop is to provide: (1) a clear definition of the state of the art of transmission systems, gears, and related transmission components, (2) an evaluation of what research and development is required to meet the projected needs of future ships and to solve current operational problems, and (3) setting of criteria to be used in specifications and procurement procedures for these transmissions.

2. Because of your background and expertise in the area of mechanical transmission systems we wish to invite you to participate in this workshop.

3. Enclosures (1), (2) and (3) indicate the goals of the workshop, a tentative agenda, and the basic areas of discussion. No security clearance will be required for the workshop, however, participants must be U.S. Citizens. The preparation expected of participants is a review of the enclosures herein from the standpoint of their technology, so as to be ready to contribute to the technical areas to be covered. The results of the workshop will be documented and distributed to the participants. It is expected that this workshop will aid in the solution of design, development and procurement problems of power trains for advanced ships with single shaft horsepowers up to 50,000.

4. Your acceptance to participate in the workshop must be received prior to 3 September 1976 for planning purposes.

2721:KTP
5050

Acceptance for and questions about the workshop should be directed to Dr. Earl R. Quandt, Head, Power Systems Division, Code 272, David W. Taylor Naval Ship Research and Development Center, Annapolis, Maryland 21402, telephone (301)267-2564.

Sincerely yours,

J. R. WALKER, CAPTAIN, USN
COMMANDER (ACTING)

Encl:
(1) Workshop Goals
(2) Working Group Subjects
(3) Tentative Agenda

HIGH PERFORMANCE MECHANICAL TRANSMISSION

WORKSHOP GOALS

It is the intent of the workshop to achieve five specific goals. These are as follows:

1. To define the state of the art of mechanical transmissions and determine what it will do for future Navy craft.

2. To determine what development is suggested by the needs of current and future Navy craft.

3. To determine what supporting research is required to aid this development.

4. To establish what development is required to solve current problems with gearing related transmission components, and transmission systems.

5. To provide guidelines for Navy procurements, specification and development of mechanical transmission systems.

Enclosure (1)
DTNSRDC ltr 2721:KTP 5050

HIGH PERFORMANCE MECHANICAL TRANSMISSION WORKSHOP

WORKING GROUP SUBJECTS

The workshop will be organized into three different working groups, each of which will address one of the three topics below. The subjects under each major heading will be covered during four working group sessions.

I. OVERVIEW OF MECHANICAL TRANSMISSIONS

1. System arrangement concepts and gearing types (parallel shaft, epicyclic, angle)
2. Component weight and size reduction
3. Flexibility of systems
4. Manufacturing quality
5. Materials

II. SOLUTIONS TO OPERATIONAL PROBLEMS

1. Gear tooth scoring
2. Corrosion
3. Fretting at interfaces
4. Bearing performance
5. Vibration
6. Problem detection
7. Noise

III. SPECIFICATION AND PROCUREMENT

1. Tooth form and gear geometry
2. Gear train and other mechanical design
3. Fits and measurements
4. Lubrication
5. Seals
6. Acceptance testing criteria
7. Design review questions to be satisfied

Enclosure (2)
DTNSRDC ltr 2721:KTP 5050

HIGH PERFORMANCE MECHANICAL TRANSMISSION WORKSHOP

DAVID W. TAYLOR NAVAL SHIP R&D CENTER

Annapolis, Maryland

TENTATIVE AGENDA

Wednesday - 29 September

1030-1100	Registration
1100-1130	Keynote and Welcome
1130-1230	Lunch
1230-1400	Introduction and presentation of background information
1400-1630	First Working Session
1630-1700	Moderator's Meeting

Thursday - 30 September

0830-0930	Presentation to general meeting
0930-1130	Second Working Session
1130-1230	Lunch
1230-1300	Moderator's Meeting
1300-1630	Third Working Session
1630-1700	Moderator's Meeting

Friday - 1 October

0830-0930	Presentation to general meeting
0930-1130	Fourth Working Session
1130-1230	Lunch
1230-1300	Moderator's Meeting
1300-1430	General meeting, presentation by moderators
1430-1500	Concluding remarks

Enclosure (3)
DTNSRDC ltr 2721:KTP 5050

DEPARTMENT OF THE NAVY
NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER
HEADQUARTERS
BETHESDA, MARYLAND 20084

ANNAPOLIS LABORATORY
ANNAPOLIS, MD. 21402

CARDEROCK LABORATORY
BETHESDA, MD. 20084

IN REPLY REFER TO:

2721:KTP
5050

From: Commander, David W. Taylor Ship R&D Center
To: Distribution List

Subj: Mechanical Transmission Workshop

Encl: (1) Group of Sketches showing various high performance craft and their transmission systems
(2) Sketches and data on hypothetical reference craft
(3) Topic Questions
(4) List of Groups (Deleted for this report)

1. Thank you for your response to our invitation to participate in the High Performance Mechanical Transmission Workshop to be held at this Center from 29 September through 1 October 1976. Enclosed is a package of information that will give a better understanding of the intent of the workshop.

2. Enclosure (1) is a group of sketches showing various high performance craft and their transmission systems. The PCH-1, AG(EH)-1, PG(H)-1, PG(H)-2, SES 100-A, SES 100-B and CPIC shown are all operational craft. The hydrofoil small waterplane area ship (HYSWAS) is merely one example of many possible future configurations. The power levels of the operational craft are lower than those future craft of interest in this workshop. However, their configurations are similar and give an indication of the type of transmission required.

3. Enclosure (2) gives sketches and data on hypothetical reference craft of future interest to the Navy. The craft sizes and power levels are greater than for the existing craft. The three craft shown are an SES (Surface Effects Ship), a SWATH (Small Waterplane Area Twin Hull) and a hydrofoil. The hydrofoil poses the most difficult transmission design due to size constraints in the foil pod. These are not intended to represent proposed craft but merely to define example systems that may require 50,000 h.p. high performance transmissions for water propeller drive.

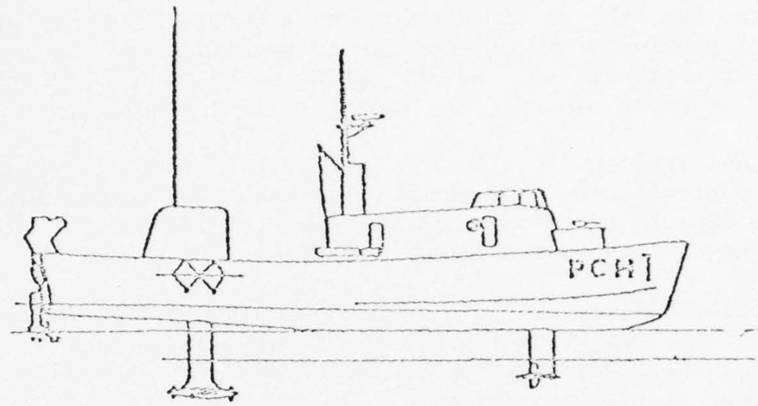
4. The approach then to meeting the workshop goals will be for the moderators of the various groups to use topic questions for stimulation of discussion within the group. The questions are not expected to be all inclusive, but should bring up relevant discussion in the necessary problem areas. From this discussion, data may be filled into a matrix of transmission components and the workshop goals. Enclosure (3) indicates the topic questions to be used, and a matrix. On the matrix, the topic questions

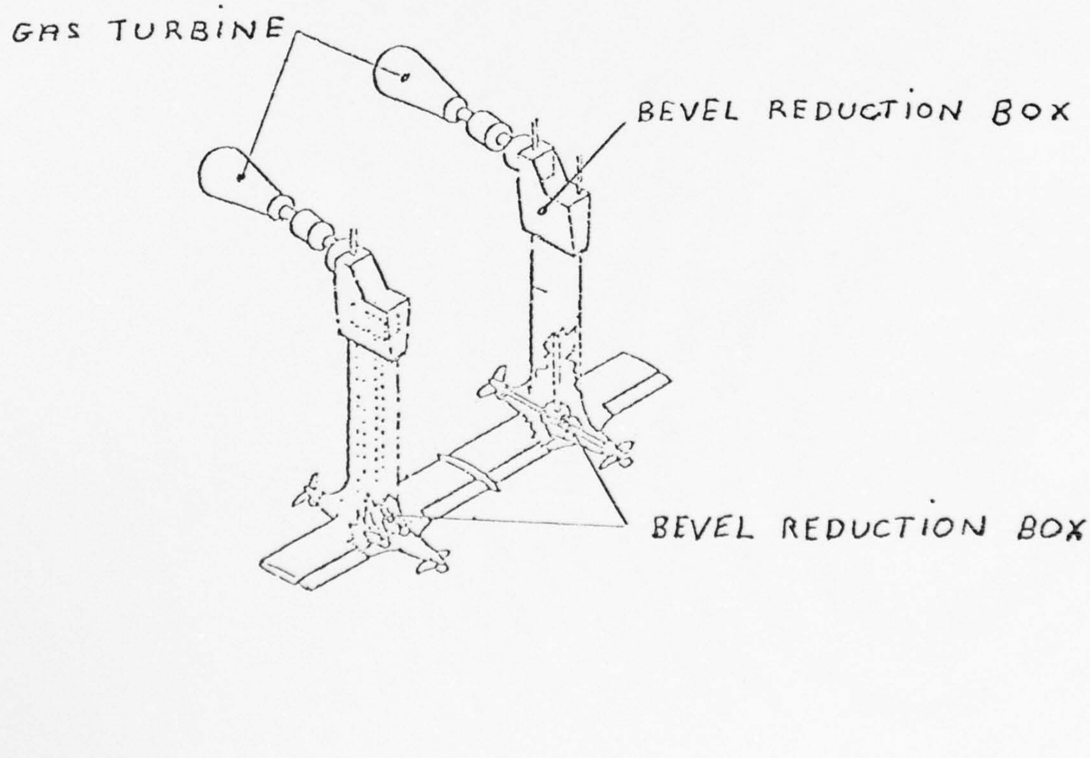
(to the left) are input. The components are listed across the top. The goals are indicated at the right. The final matrix should indicate the state of the art, the required R&D to meet future needs, an approach to operational problems, and an approach to specification and procurement for the components listed. By attempting to solve the problems in this manner, it is hoped that all the topics will be adequately discussed.

5. The final enclosure, (4), is a list of the groups in which people will be asked to participate. An attempt has been made to provide knowledge in all the aspects of transmission systems to each group. The list is based on those who have responded to date.

6. We are looking forward to a rewarding workshop. Plans are to prepare and distribute a report of the results to all participants. If there is any question please contact our Mr. A. B. Neild at (301)267-2263 or Mr. A. B. Harbage at (301)267-2845.

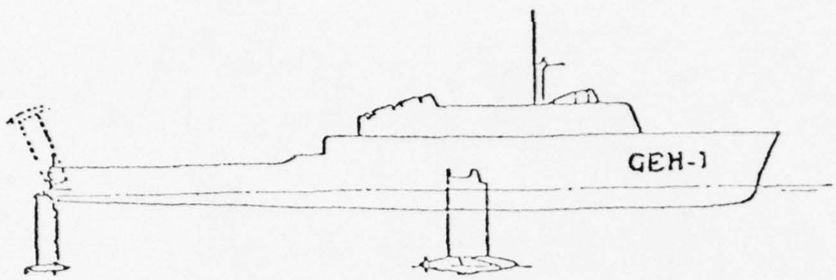
PCH-1 PROPULSION SYSTEM ARRANGEMENT

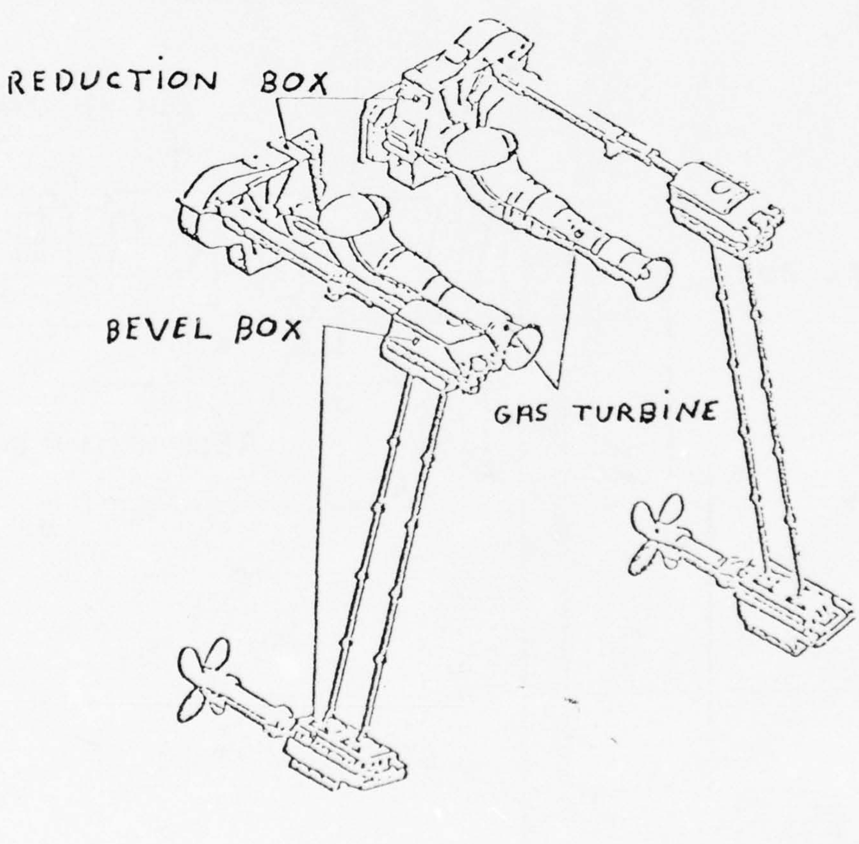




Enclosure (1), Page 1

AG(EH)-1 PROPULSION SYSTEM ARRANGEMENT





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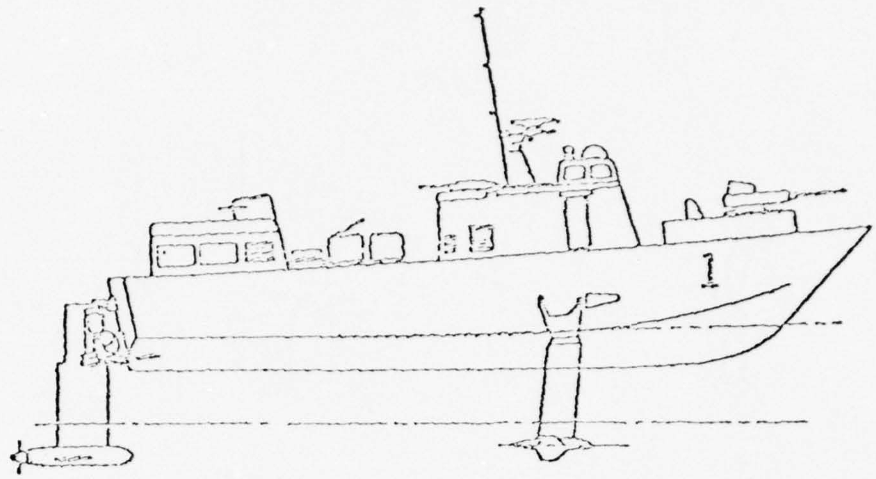

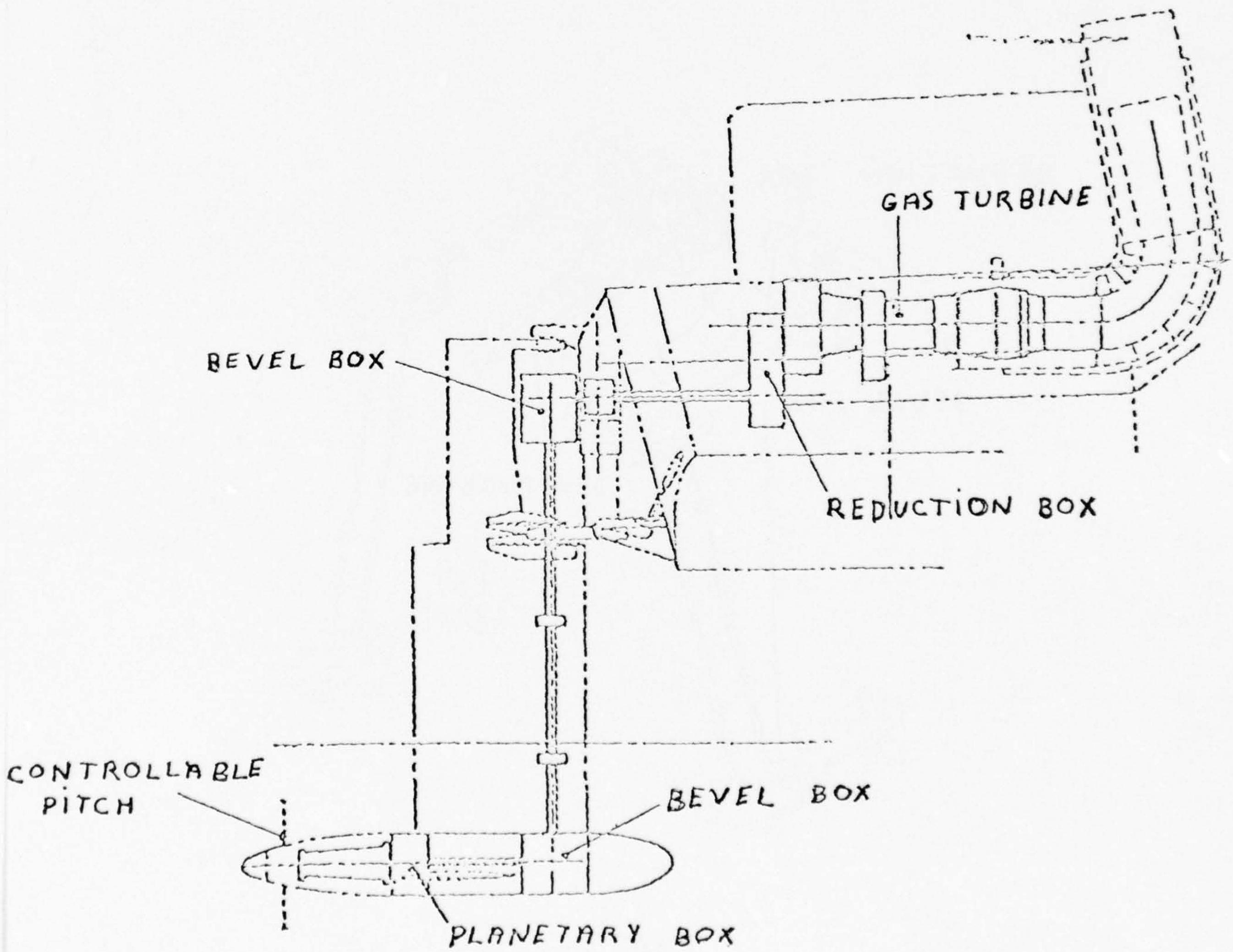

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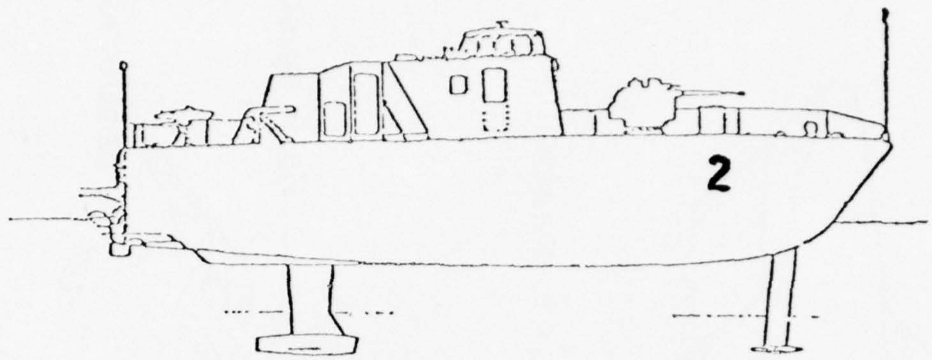

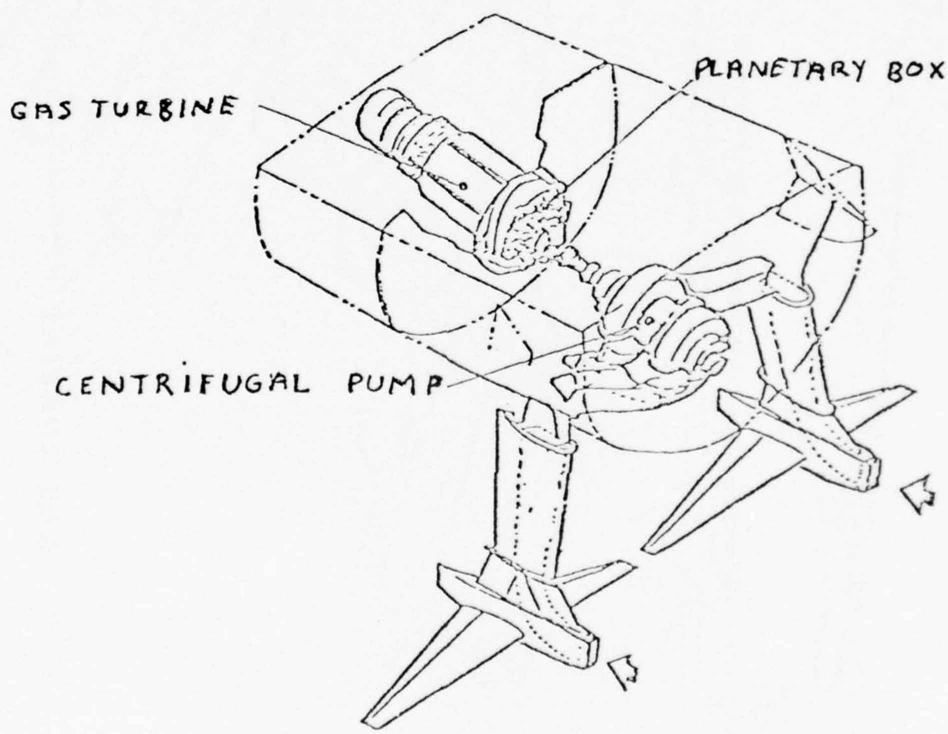

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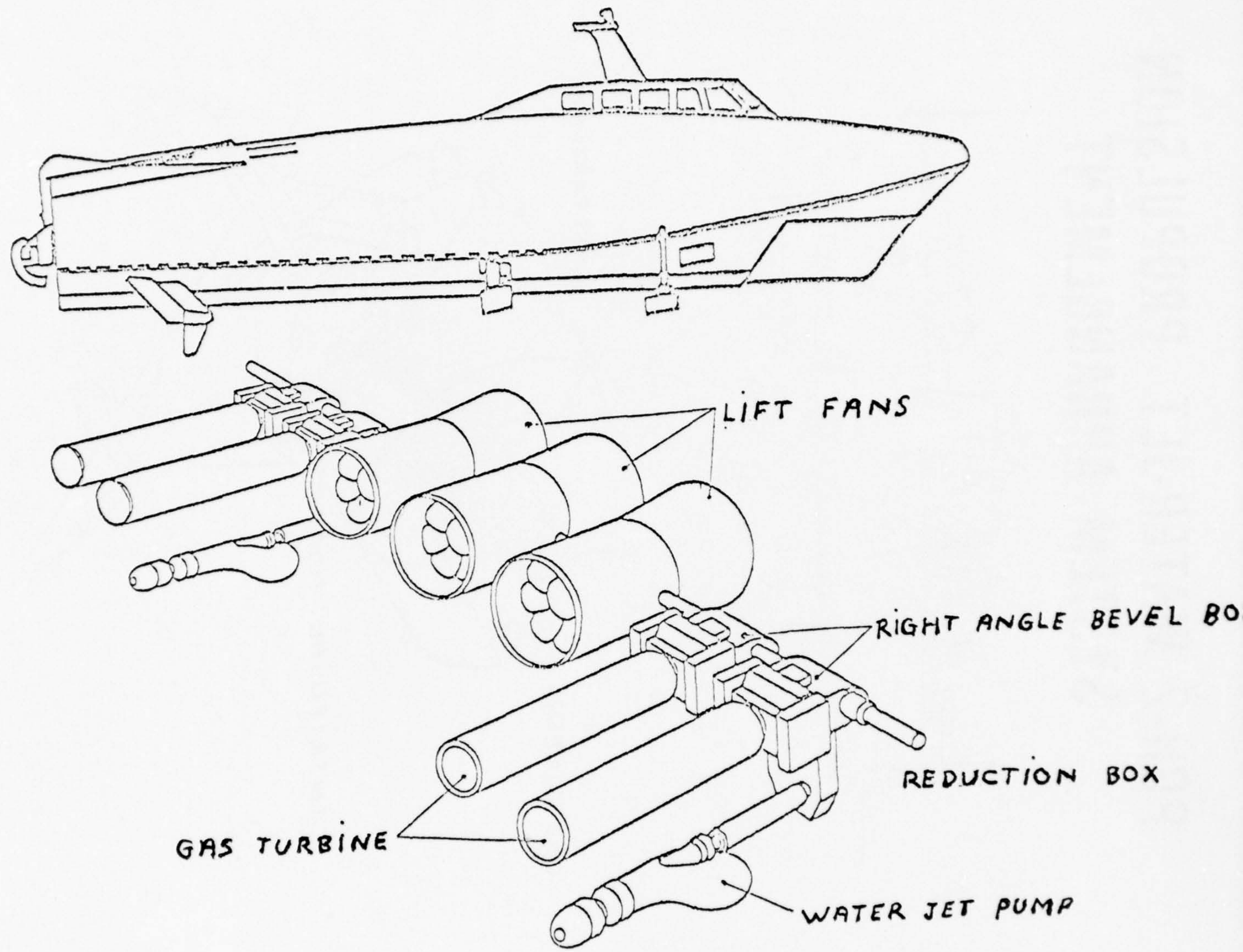

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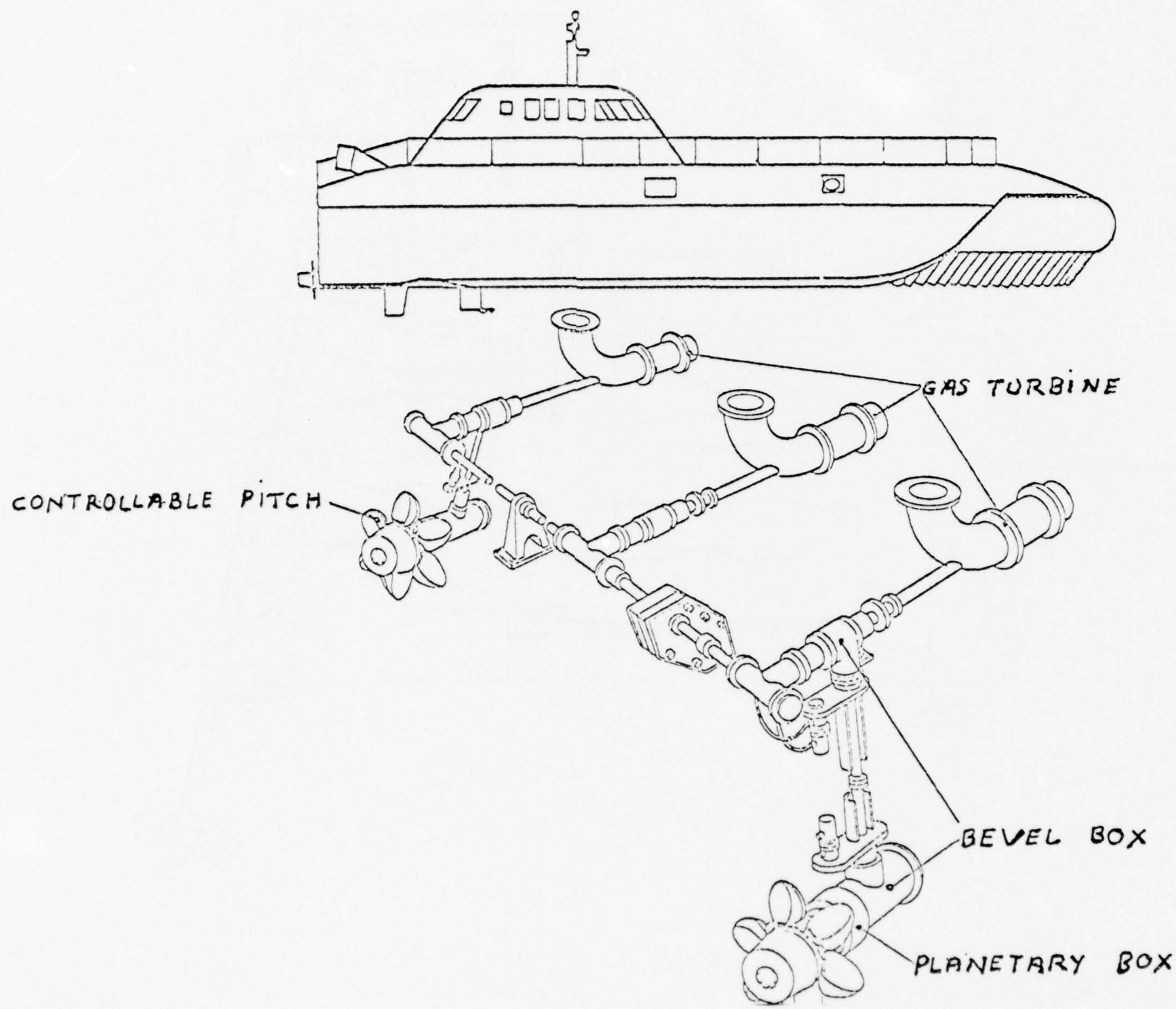

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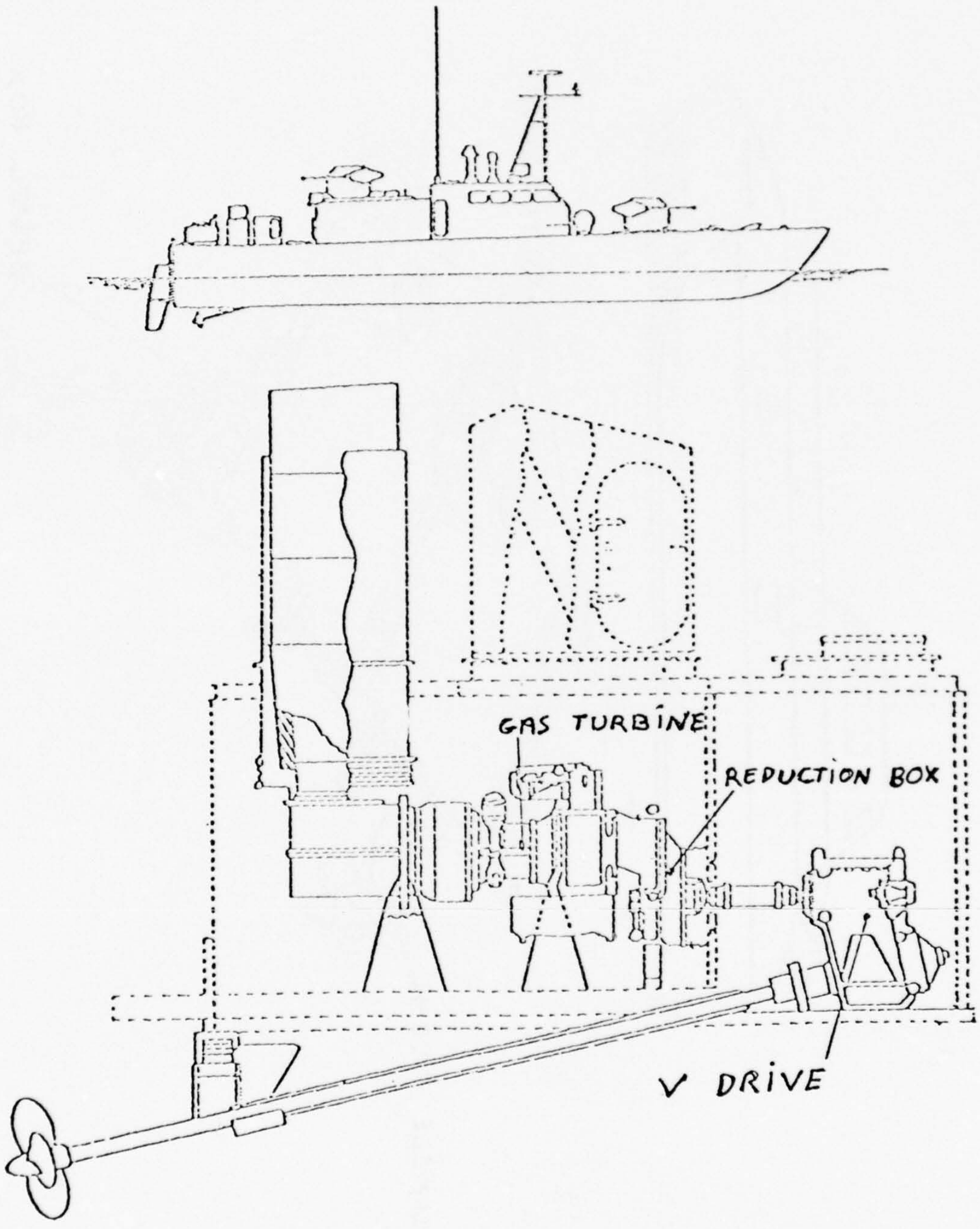

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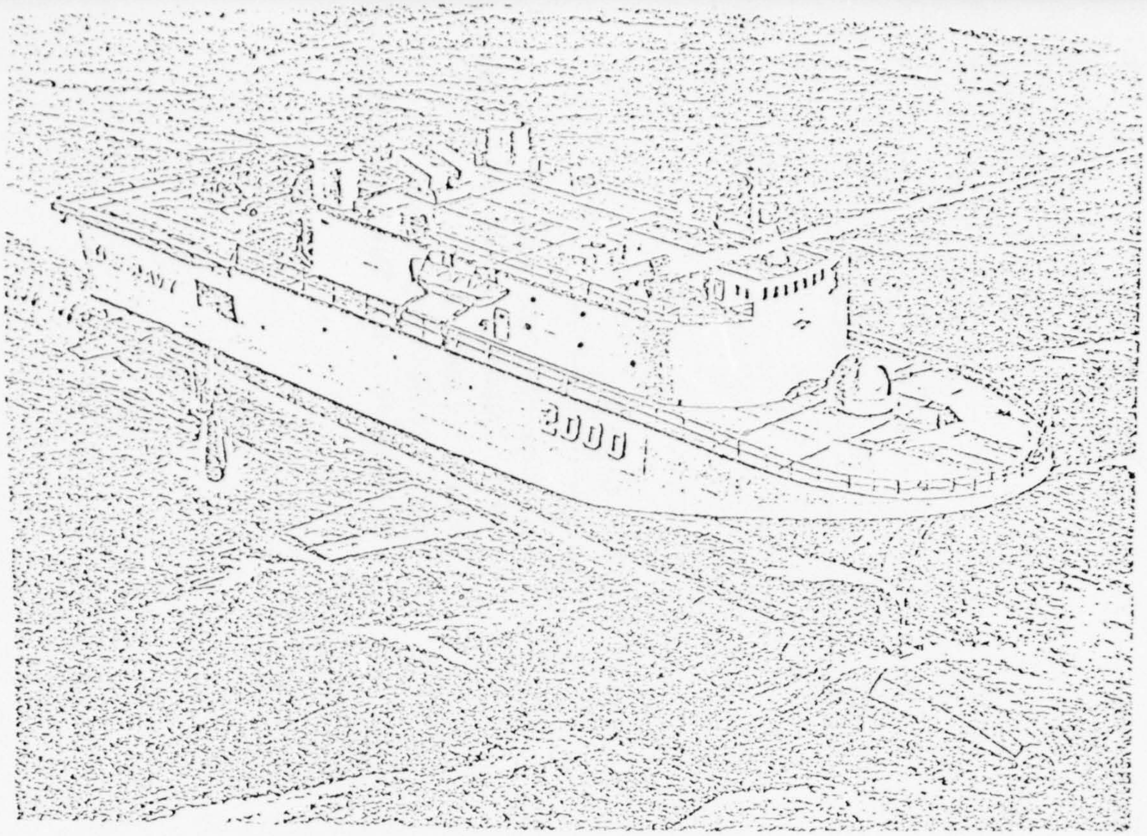
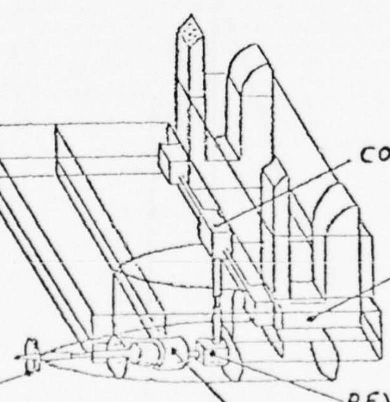
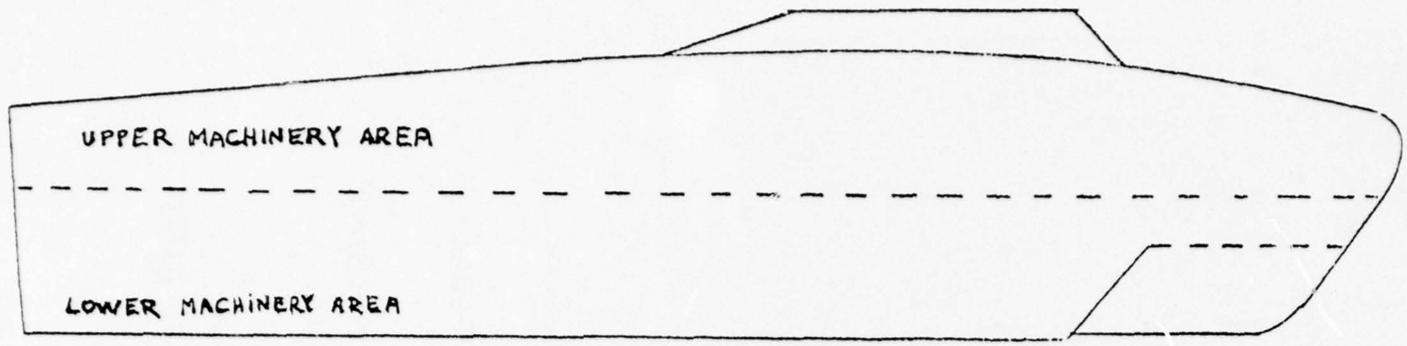




(HYSWAS)

HYDROFOIL SMALL WATERPLANE AREA SHIP



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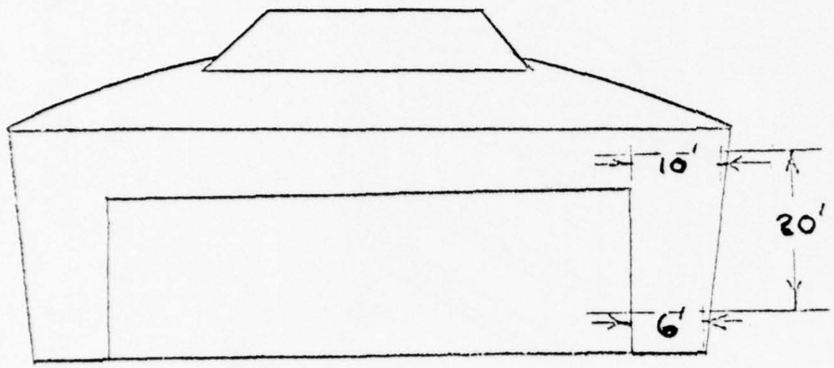
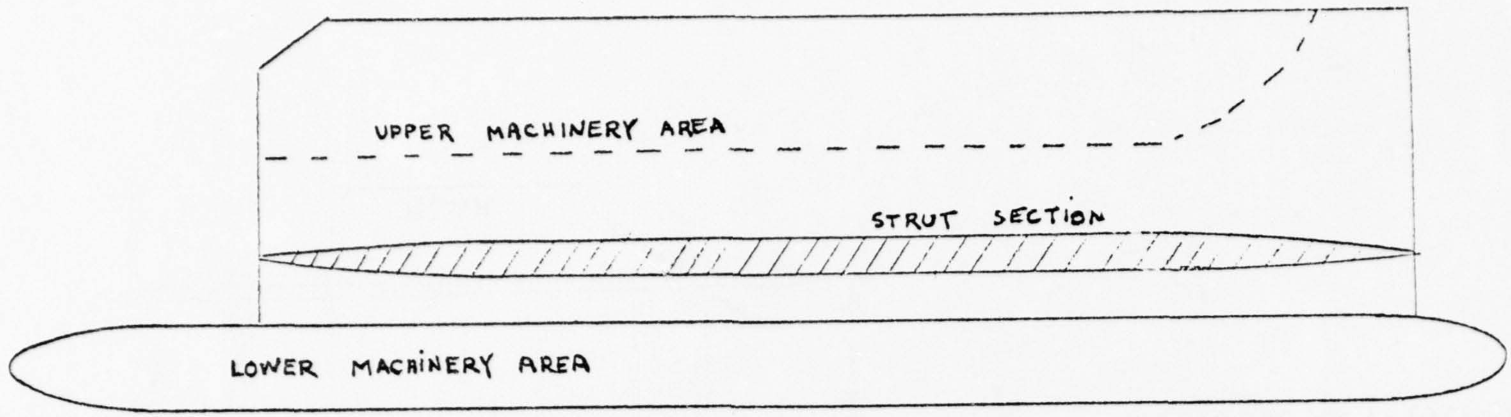


REAR

SES

- 50,000 HP PER SHAFT
- TWO ENGINE INPUT PER SHAFT
- 3600 ENGINE RPM
- 350 PROPELLER RPM
- ~ 350,000 LB THRUST BEARING



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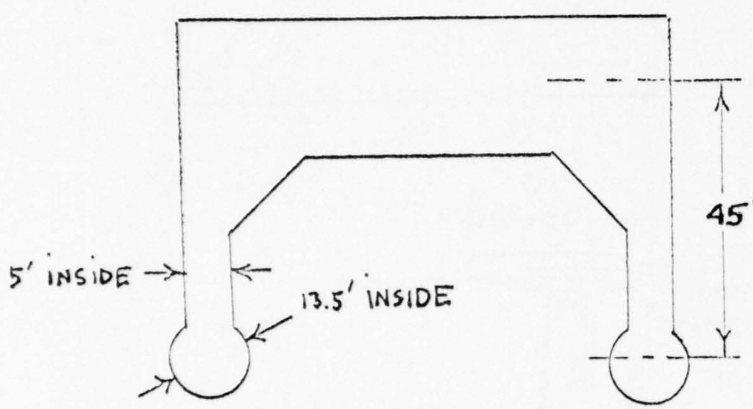
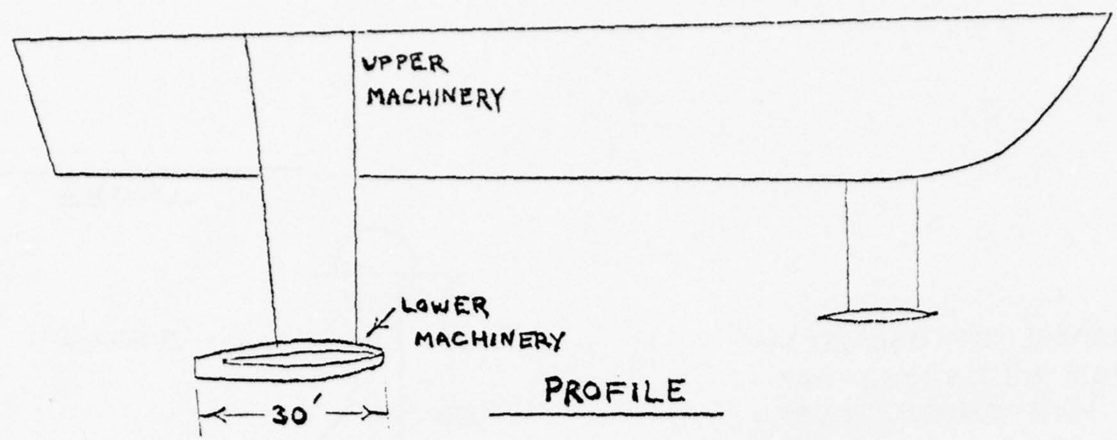


FRONT

SWATH

- 50,000 HP PER SHAFT
- TWO ENGINE INPUT PER SHAFT
- 3600 ENGINE RPM
- 200 PROPELLER RPM
- ~350,000 LB THRUST BEARING



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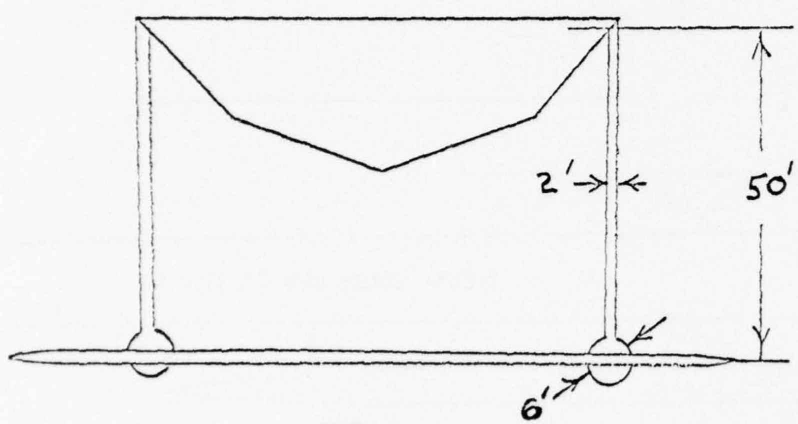

REAR

HYDROFOIL

- 50,000 HP PER SHAFT
- TWO ENGINE INPUTS PER SHAFT
- 3600 ENGINE RPM
- 300 PROPELLER RPM
- ~350,000 LB THRUST BEARING

MECHANICAL TRANSMISSION WORKSHOP
TOPIC QUESTIONS

I. Overview of Solid Transmissions

1. What arrangement concepts and specific gearing types should be pursued for naval applications?

The reference ships described define the general type of transmission required, lightweight and compact, consequently highly loaded. There are many questions as to approach to these transmissions. These include such things as defining the level to which lightweight reduction gearing (light combining gears with epicyclic reduction) should be used. How far should the effort be carried in the lightening of components?

What is required to provide the necessary right angle gearing? Should development be done on contrarotating epicyclic arrangements? What additional approaches to arrangement would show promise?

2. What benefits could result from lightening of components?

Discussion of various components is intended. Should lightening of shafts through use of carbon/epoxy composites be developed? Should new coupling, clutch, torsional vibration reducer, etc., designs be sought (from a lower weight standpoint)? Should lightening of cases and mounting structure be attempted? What can be accomplished in the area of lighter weight bearing designs?

Are lighter weight gear materials a practical approach? If so, is material development suggested?

3. What is the state of the art for system flexibility for hard transmissions?

What areas of coupling and soft mounting need to be pursued in order to achieve troublefree gearing in structurally flexible craft? Are new component designs required or will judicious design and arrangement be adequate?

Are the limits on misalignments and deflections in couplings representative of what can be achieved or only what has been required to this point?

4. What is the state of the art in manufacturing quality and gear size?

Can the necessary size gears be cut, ground and heat treated such that the necessary accuracy can be maintained? Does this require special techniques such as soft grinding and then heat treating or is production within state of the art?

Is the same true for bevel gears, how different are limits?

Are manufacturing tolerances adequate to produce high performance gears, how important are the tolerances to quality gears?

5. <u>Define the state of the art of gear materials.</u>

What are the weaknesses and strengths of the various materials currently used?

What properties are desired of gear materials that can't be provided with available materials?

What is likelihood of solving gear problems with improved material properties, i.e., how much material development effort is warranted?

II. Solutions to Operational Problems

1. <u>What can be done to eliminate failure of highly loaded gear teeth?</u>

What are the causes of gear tooth distress? Are better lubrication, new materials and/or new manufacturing techniques required to overcome failure of highly loaded gear teeth? Can design load limits be kept to a level that will eliminate failure and still permit gears small enough to meet light weight demands of high performance craft?

2. <u>What techniques can be used to eliminate corrosion problems within gear cases?</u>

What are the causes of this type of corrosion (purely water?) and what surfaces corrode?

What solutions are offered through the use of different materials, different seal designs, different lubricants and environmental control within areas containing gears and bearings.

3. <u>What can be done about fretting corrosion at bearing/shaft interfaces?</u>

Can different materials be used that are more resistant to fretting?

Can specification of fits be set up that would eliminate causes?

Can design of systems utilizing integral gears or combinations of integral and shaft fitted gears be provided that will eliminate fretting?

4. <u>What can be done to improve bearing life?</u>

What techniques can be employed for cleaning and cooling bearing areas?

What can be done with hydrostatic and hydrodynamic rather than currently used antifriction bearings?

What can be done with new antifriction bearing designs (such as hollow ended compliant rollers)?

Is development warranted in the area of thrust bearings?

5. What can be achieved with problem detection in gearboxes and what techniques can be used?

Use of proximity probes for deflections?

The use of chip detection devices?

Temperature monitoring?

Vibration monitoring?

In all cases, what can be concluded from data available and what are the best methods to obtain data accurately?

6. What improvements in gear noise are of sufficient magnitude to warrant the required development?

Reduction in tooth forces by torsional vibration absorber.

Better case damping through use of coatings or noise attentuating material.

Reduction of shaft deflections at bearings by controlling dynamic response through stiffness and mass and inertia distributions.

III. Specification and Procurement

1. What should be specified in the area of tooth form and gear geometry?

How closely should this be controlled?

Are standards within the industry such that no controls on tooth form are necessary?

2. What should be specified as far as gear train and associated mechanical design is concerned?

Should the type of gearbox (epicyclic for instance) be specified or merely requirements that a unit must meet? Should design aspects (such as integral gears) be specified?

3. What should be specified in area of component fits and component measurements?

Should specific component fits be called out or should problems be avoided through some sort of design evaluation after manufacturer has a design?

How should components be measured to assure proper profiles of teeth, clearances, fits, etc.

4. How should lubrication of high performance transmission be specified?

Do you specify a lubricant and a type of system, a required set of lubrication properties under given loading conditions (such as film thickness), or do you specify only the required performance of the transmission by whatever method necessary to provide that performance?

5. How should transmission seals be specified?

Should seal specifications be made to type or design or merely sealing requirements?

6. How should components such as clutches, brakes, couplings, shafting, bearings, housing structures be specified?

Should specific design data be called out or should performance requirements alone be used?

7. What form should Navy design review take?

Should frequent design reviews be conducted throughout the design and building process in order to assure meeting of requirements or should specification be depended on to assure this with lesser degree on review?

8. How should component and system acceptance testing be performed?

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APPENDIX B

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