3KSES
PRODUCIBILITY IMPROVEMENT STUDY
Propulsion and Lift Systems
Summary Report

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1 / INTRODUCTION AND SUMMARY

This summary report contains the documentation of work performed under Tasks C, E and I of the 3000-Ton Surface Effect Ship Producibility Improvement Plan and Statement of Work (Reference 1). The principal objective of these tasks is to identify propulsion and lift system concepts that will impart to the improved producibility SES, maximized performance and economy of operation consistent with minimized risk.

The work accomplished in this effort supports the ship design concept development reported in the Ship Characteristics and Performance Report (Reference 2). Two basic propulsion/lift systems are identified for ships in the nominal displacement range of 1000 to 1800 LT established for the study.

In order to minimize the producibility risk in the main machinery area, propulsion gas turbine engines are selected that are of current U.S. manufacture and suitable for marine service. The propulsion gas turbines selected together with both the present and projected power ratings are shown in Table 1-1.

The main machinery selections for the two basic propulsion/lift system configurations are summarized below.

Table 1-1. Propulsion Gas Turbines

ENGINE	SHAFT HORSEPOWER*		STATUS
	MIP	MCP	
DDA 570 KF (Allison)	6524	5823	In Production
DDA 571 (Allison)	7160	6500	Projected
LM2500 (G.E.)	27,000	22,500	In Production
LM2500 (G.E.)	30,000	27,000	Projected Uprating

*Quoted powers are for the following conditions: -

- $80^{\circ}F$ ambient temperature
- 4 in. H_2O and 6 in. H_2O inlet and exhaust losses

	Configuration I	Configuration II
Propulsion Engines	DDA 570KF (4)*	GE LM2500 (2)
Lift Engines	SACM 195V12RVR (2) SACM 175V8RVR (2)	SACM 195V20RVR (2) SACM 195V12RVR (2)
Reduction Gear	Cinti CODOG (2)	Cinti CODOG (2)
Lift Fans (Aerophysics Co.)	RD135-.65-.13-70 DWDI (6)	RD154-.65-1.3-70 DWDI (6)

*Numbers in parenthesis are number of units per ship.

Propellers are not listed in the above summary of the propulsion/lift configurations. Each propeller is a custom design specifically prepared for the ship/powerplant combination on the basis of required thrust and ship speed. All the propeller preliminary designs are outlined in Appendix A. These include both ventilated (partially submerged, supercavitating) and fully submerged (conventional, high speed) types.

The machinery selected is consistent with the features of both logistics vessels and the Medium Displacement Combatant (MDC) design presented in PMS-304 Report, Reference 3. These features include high on-cushion speeds for sprint and rapid transit, and suitable off-cushion speed for patrol and economical transit at fleet speeds. Both gas turbine and diesel prime movers are provided. For high-speed on-cushion operation the gas turbines provide the necessary power with good economy, in a high power to weight installation. For cruise off-cushion, the diesel provides the required power with excellent economy. The diesel engine is also used for lift system power when operating on-cushion.

Light weight power transmissions are selected to transmit either the gas turbine power or diesel power to the propeller (CODOG drive). Clutches are used to isolate the prime movers.

Propulsors evaluated in this study included partially submerged supercavitating propellers, generally referred to as ventilated propellers, and submerged (high speed) propellers. Ventilated propellers were baselined for the 1500 LT ship.

Waterjet pump type propulsors were eliminated early in the study for reasons of poor economy and high installation weight. In today's fuel conscious environment the lower propulsive efficiency of the waterjet is considered inappropriate for the improved-producibility SES.

The lift fans selected are Aerophysics rotating diffuser fans which show good performance, and insensitivity to wave induced cushion pressure fluctuations.

Configuration I utilizes two Allison 570KF gas turbine engines per sidehull, originally in an over/under arrangement. This arrangement was dictated by the hull lines in use at the onset of the program, which would not permit a more desirable side-by-side arrangement. It was recognized that access for maintainability was hampered by the over/under engine arrangement. As the program progressed model tests led to a modified sidehull configuration with greater beam-wise space in the engine area. An updated Configuration I arrangement was subsequently prepared in which the engines are arranged side-by-side in the new sidehull cross-section.

In the area of lift fan arrangements the early design approach taken was to minimize the number of lift fans and lift engines, using only the diesel engines in the CODOG propulsion plant for lift power. This arrangement used long ducts to route air flow to the bow seal. There were valid reasons for doing this, as discussed in Appendix C of this report. Tow tank model tests conducted during the program, however, mandated a change in the lift system design approach. The test results disclosed required lift airflow in the order of 70 percent greater than had been indicated in the Navy MDC Report (Reference 3) which was accepted as the baseline document at the outset of the study. As the design effort converged on the 1500 long ton ship, the airflow and cushion pressure dictated a level of lift horsepower for that ship beyond that available with the two diesel engines originally conceived for Configuration II. Aft-mounted diesel engines developing sufficient horsepower to meet the new requirement would have been prohibitively large for the installation. The solution was to retain the selected aft diesel engines to drive the cushion and stern-seal fans while adding two smaller diesel engines up forward, each driving a fan for bow seal inflation. This approach brought the required total lift horsepower on board, retained sufficient horsepower in the aft location for hullborne propulsion via CODOG, and provides redundancy for bow seal inflation while eliminating the long trunks.

Three appendices are included in this summary report:

- Appendix A - Propulsor Studies: Documents work performed to date to select the optimum propeller system for the various sizes of MDC ship. Technical descriptions are given. Sizing and performance calculations are presented. A propeller performance computer program and results is described. Drive shafting design parameters are outlined. Technical risk, reliability, maintainability, and producibility are discussed.
- Appendix B - Propulsion Machinery Configuration: Addresses arrangements for the ship main propulsion and lift machinery. Technical descriptions are given. Layouts are presented. Reduction gearing and CODOG arrangements are examined in detail. Cost data and recommended sources for all machinery including lift are furnished. Technical risk, reliability, maintainability, and producibility are discussed.
- Appendix C - Lift Machinery Configuration: Presents machinery selections, arrangements, and performance for the various ships

in the study. Technical descriptions are given. Sizing and performance calculations are presented. Lift fan drawings are provided. Technical risk, reliability, maintainability, and producibility are discussed.

2 / TECHNICAL SCOPE

2.1 PROPULSOR STUDIES - TASK C - SUBTASK 1

The studies, presented in Appendix A, show the suitability and applicability of the selected propulsors for the MDC and its variants, with the objective of maximized efficiency and economy. The task includes development of concepts, layout drawings, performance calculations and system definition. Candidate systems include controllable pitch propellers and waterjet pumps. 3KSES developed support systems and components, standards and design tools are used for the selected designs where applicable.

2.2 PROPULSION MACHINERY CONFIGURATIONS - TASK C - SUBTASK 2

The studies, presented in Appendix B, show the suitability and applicability of the selected prime movers and gearing systems for the MDC and its variants with the objective of maximized efficiency and economy. The task includes development of concepts, layout drawings, performance calculations and system definition. Candidate propulsion systems include a four DDA 570K gas turbine plus two SACM 195 V12 RVR diesel CODOG combination and a two LM2500 gas turbine plus two SACM 195V20 RVR diesel CODOG combination. 3KSES developed support systems and components, standards and design tools are used for the selected designs where applicable.

2.3 LIFT MACHINERY CONFIGURATION - TASK E - SUBTASK L

The task comprises identification of the fan characteristics, concept formulation, layout drawings, performance calculations and system definition. Candidate fans include three single or double inlet lift fans per sidehull driven through a gearbox from one or more diesel engine(s). 3KSES developed support systems, and components, together with 3KSES standards and design tools, are used where applicable.

2.4 MAIN MACHINERY ARRANGEMENT UPDATE - TASK I AND L

To solidify and develop the designs performed under the above task further studies were conducted to develop main machinery arrangements to be compatible with the new sidewall geometry. Emphasis was placed on simplicity, reliability, accessibility and maintainability. Layout drawings, performance calculations and system definition studies result in forward and side diesel driven fan systems and separation of the gas turbines and diesels.

Preliminary design level propulsion and machinery arrangement drawing and a corresponding three digit level weight estimate comprise the Task L work.

3 / CONCLUSIONS AND RECOMMENDATIONS

The following conclusions and recommendations are made.

1. The CODOG system for propulsion power is a viable concept and matches the broad requirements of the SES studies.

2. Controllable pitch propellers are the most suitable propulsors for the improved-producibility SES.

3. Submerged high speed propellers show good efficiency at speeds up to about 50 knots. The potential for cavitation damage at the higher speeds and the generally more complex transmission systems add elements of risk for these propellers.

4. Ventilated propellers are the preferred propulsors for the improved-producibility SES. They exhibit good efficiency at on-cushion speeds and satisfactory efficiency for hullborne cruise. Model test data over the complete range of advance coefficient and immersions are required to predict performance with high confidence.

5. Methods of obtaining increased low ship speed (off-cushion) propeller immersion for the ventilated propeller have been identified qualitatively.

6. The transmissions identified are state-of-the-art, conservatively rated, compact, relatively light weight and high reliability units.

7. The four DDA 570 gas turbines used in Configuration I have been rearranged in horizontal pairs to improve accessibility.
8. The four DDA 570 gas turbines are adequate for an SES in the order of 1000 LT using ventilated propellers.
9. The two LM2500 gas turbines provide adequate power for ships in the displacement range of 1200 LT to 1800 LT.
10. The use of aft and forward lift fans is preferred from the standpoint of reliability.

4 / REFERENCES

1. 3000-Ton Surface Effect Ship Producibility Improvement Plan, Appendix A - Statement of Work for 3000-Ton Surface Effect Ship Producibility Improvement, 28 May 1981.

2. 3KSES Producibility Study, Ship Characteristics and Performance Report (Preliminary), 26 June 1981.

3. Medium Displacement Combatant Surface Effect Ship (MDC) Technical Report, Surface Effect Ship Acquisition Project - Naval Sea Systems Command, April 1981.

APPENDIX A

PROPULSOR STUDIES

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1 / TECHNICAL DESCRIPTION

1.1 GENERAL

The selection of a propulsor for the modified 3KSES is governed by the primary requirements of performance and economy at both low and high speeds. For the modified 3KSES, the top speed requirement is in the range of 40-60 knots. Additionally, it is anticipated that the majority of operating time of the ship will be at fleet speeds in the range of 15-20 knots. The need for good performance at both high and low speeds indicates that a propeller propulsion system will show distinct advantages of economy over a waterjet propulsor system. Furthermore, there are advantages to be realized by the judicious choice of propeller type.

The two propeller types investigated are the supercavitating propeller and the subcavitating propeller. The supercavitating propeller has two subdivisions: surface piercing (partially submerged) and fully submerged. Partially submerged supercavitating propellers have the propeller blade back surfaces ventilated to atmosphere and for the remainder of this appendix will be referred to as ventilated propellers.

Figure 1.1-1 shows the approximate installed efficiency relationships between various propulsors as a function of craft speed. Figure 1.1-1 is taken from Reference 1. The efficiency curve for conventional and high performance subcavitating submerged propellers was extended into the higher speed range according to data given in Reference 4. Each

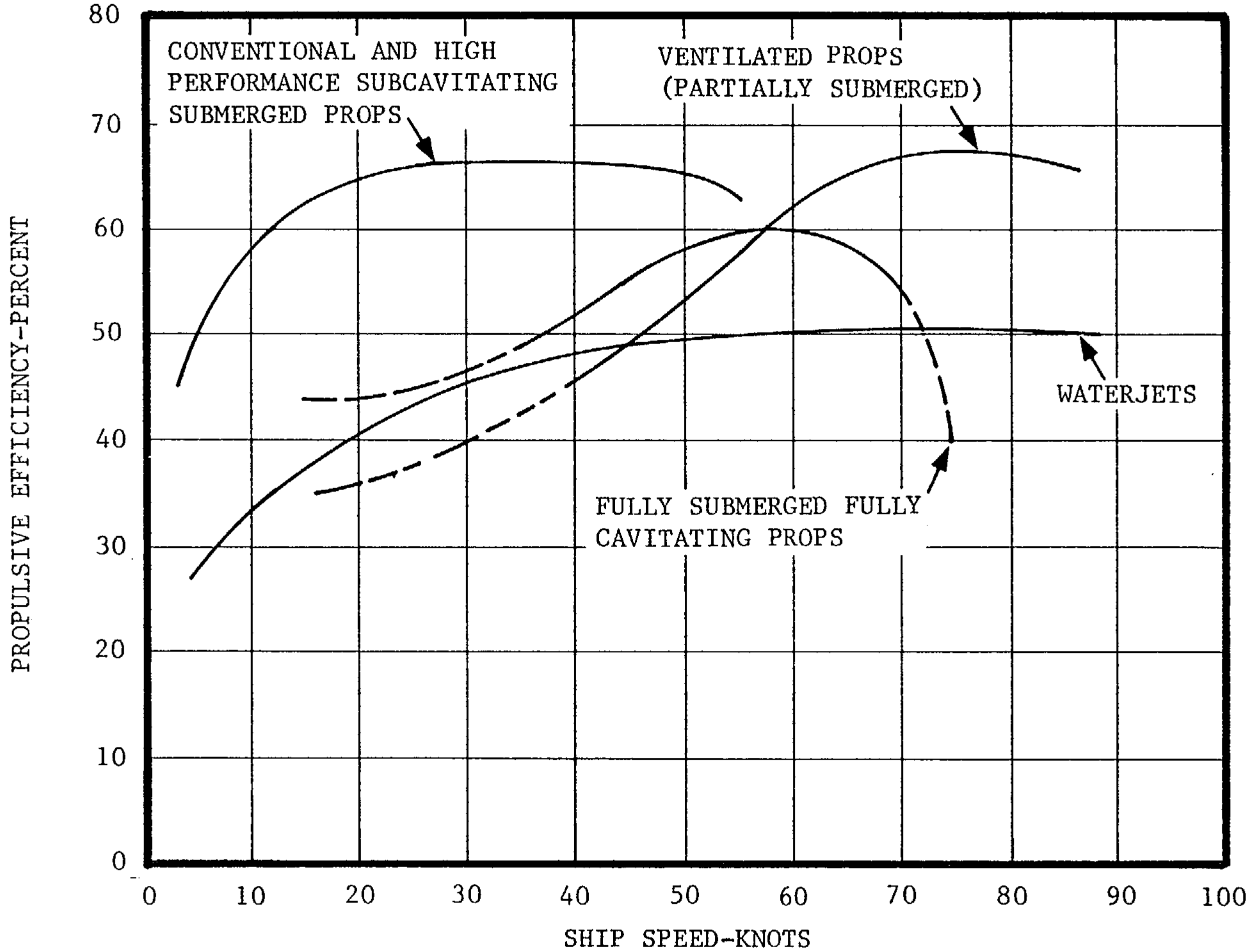


Figure 1.1-1. Approximate Maximum Installed Efficiency Envelopes for Propellers and Waterjets(Current Design)

curve represents a composite of data and should only be used for comparative purposes. As can be seen, conventional high performance submerged propellers are more efficient that other propulsors up to a speed of 55 knots. This type propeller will be referred to as a submerged propeller for the remainder of this appendix. Waterjet propulsors show, in general, the lowest overall efficiency but have the advantages of simplicity, and in some cases, availability, over propellers which are usually custom sized and designed for the application. Fully submerged, fully cavitating propellers were not evaluated because of generally lower efficiency.

The requirement for maximized efficiency over the complete speed range and particularly at the off-cushion speeds, combined with the need for backing thrust, led to selection of controllable pitch (CP) for evaluation ob both the ventilated propeller and the submerged propeller. The controllable pitch propeller is a fully developed system used almost universally by ship builders to maximize overall ship propulsive efficiency.

The thrust bearing and drive shaft are sized and located as dictated by the propeller type. Since the ventilated propeller is to be mounted close to the sidehull transom, and as it is known that these propellers produce large side and vertical loads, a thrust bearing/radial bearing unit for the propeller is mounted as close to the propeller plane as possible. This arrangement absorbs the propeller thrust and transmits it into the structure without loading the drive shaft in compression, and will thereby avoid dynamic problems.

The thrust bearing uses rolling element bearings for compactness and rigidity. The drive shaft will be hollow to reduce weight and to provide a conduit for the pitch change oil tubes.

1.2 VENTILATED PROPELLER

The ventilated propellers studied are controllable pitch partially submerged types. Performance data for the propellers is given in Section 2.2.

The propeller blades are made from forgings machined all over to achieve accurate profiles and sections. Candidate blade materials are stainless steel, nickel based superalloy, titanium, and nickel-aluminum bronze.

Each blade is replaceable externally via a bolted joint with no other disturbance of the propeller or pitch change mechanism required. The propeller blades can be changed while the ship is afloat. The blade section is a wedge with an annex section as shown in Figure 1.2-1. This blade shape and section is similar to that used successfully on the SES-100B.

The propeller derived from this study is 4-bladed with a diameter of 12.49 feet and a hub to tip ratio of .40. Preliminary input from Tacoma Boat/Escher Wyss, who supported the study with technical and fiscal data, is that a 5-bladed propeller with a diameter of 12.5 feet and a hub to tip ratio of .336 would be required. This is a good correlation for a preliminary estimate. The hub is an approximately cylindrical body made from corrosion resistant material which contains the pitch change mechanism. Attached to the after-end of the hub is a fairing. The propeller is mounted behind the sidehull such that 50 percent of its disc area is masked by the sidehull transom. From the propeller, the drive shaft is installed at an inclined angle up and forward 8 degrees. Immediately adjacent to the propeller hub is the thrust bearing module.

For the arrangement drawings of the propeller and propulsion machinery, see Appendix B, Sections 1.2 and 1.3.

1.3 SUBMERGED PROPELLER

The submerged propellers studied are conventional propellers with modified sections after Newton and Rader to allow operation in the cavitation regime at high ship speeds. Performance of the propellers is discussed in Section 2.3.

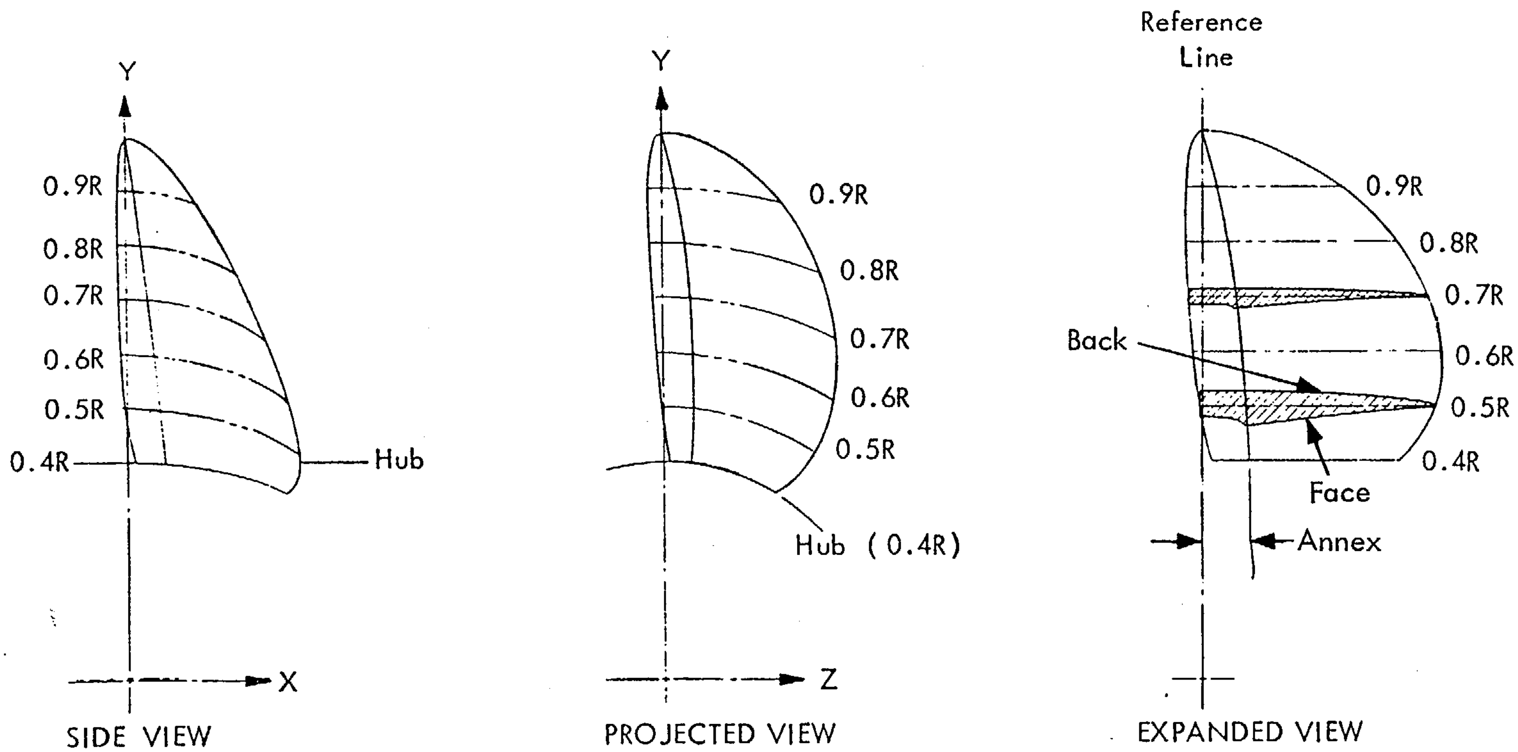


Figure 1.2-1. Ventilated Propeller Shapes and Sections

The propeller blades are made from forgings machined all over to achieve accurate profiles and sections. Candidate blade materials are stainless steel, nickel based superalloy, titanium, and nickel-aluminum bronze.

Each blade is replaceable externally via a bolted joint with no other disturbance of the propeller or pitch change mechanism required. The propeller blades can be changed while the ship is afloat. The blade section is a double circular arc with modified leading and trailing edges of the shape developed by Newton and Rader, Reference 4, for high speed vessels. A sketch of the section is shown in Figure 1.3-1.

The propeller derived from this study is 3-bladed with a hub to tip ratio of .300. The hub is an approximately cylindrical body made from corrosion resistant material which contains the pitch change mechanism. Attached to the after-end of the hub is a fairing.

The propeller is mounted beneath the sidehull such that the clearance between the blade tip and sidehull is 3 feet. Ahead of the propeller a streamlined propeller bossing is provided to contain the stern bearing for the propeller shafting. The shaft inclines upward 8 degrees. It should be noted that the hull lines in this area require further development and will be modified if a submerged propeller is utilized.

Ahead of the stern bearing, where the shaft penetrates into the sidehull is a combined thrust and radial bearing. A coupling connects the shaft to the reduction gearbox.

1.4 PITCH CHANGE MECHANISM AND CONTROLS

Many modern commercial and naval ships use controllable pitch propellers to achieve operating economy and to suit the special requirements of the particular ship. Pitch change mechanisms and their controls have been fully developed as a consequence.

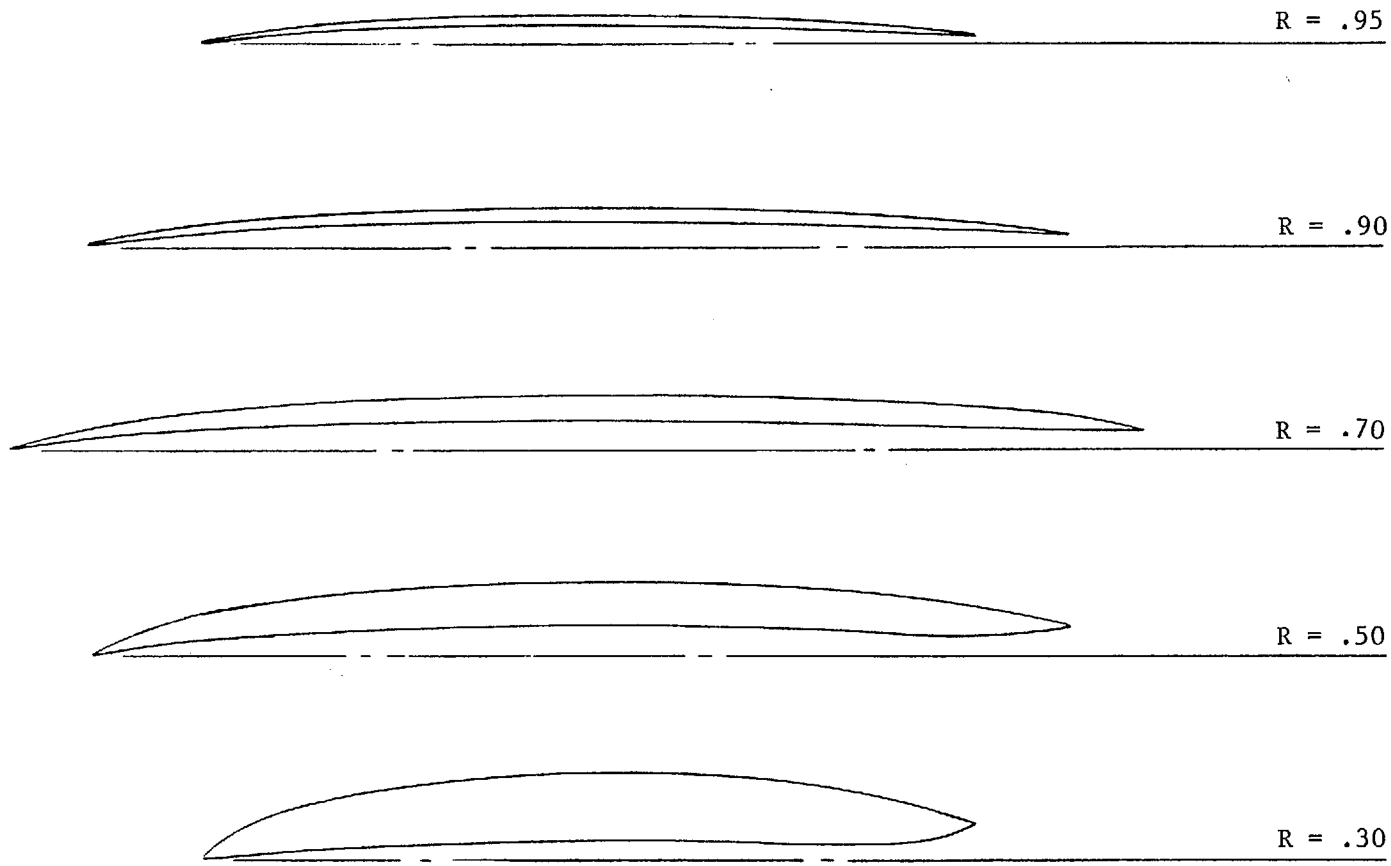


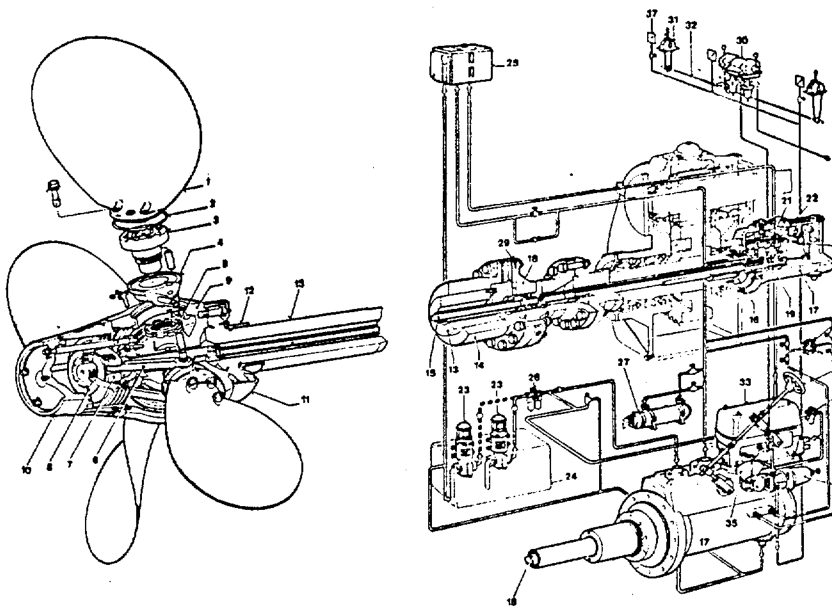
Figure 1.3-1. Blade Sections of High Speed Submerged Propeller

The Tacoma Boat/Escher Wyss type of pitch change mechanism and controls is fully described in Reference 2. The mechanism, as shown in Figure 1.4-1, is briefly described as follows.

Each propeller blade is bolted to a trunnion which carries a crank arm. Each crank arm is linked to a servomotor piston. Hydraulic oil is fed to the piston via a double oil tube from an oil distribution unit mounted within the hull either on the end of the propeller shafting or around the propeller shaft. The double oil tubes are nested inside the propeller shaft. A control valve and feedback system is integral to the oil distribution unit. In response to a pitch change signal, the control valve directs pressurized oil to the servomotor piston. As the piston moves, and causes the propeller blades to rotate on their trunnions, a feedback signal is transmitted back mechanically to the control valve (via the double oil tube) so that when the input signal and propeller blade pitch agree, the control valve spool is centered and no further blade movement occurs. Pressurized oil is supplied from a tank having main and standby electric pumps. The tank is connected to a head oil tank so that when the system is unpressurized there is sufficient head in the propeller pitch change mechanism and hub to prevent ingress of the surrounding water. This mechanism plus the command transmitter, pitch indicator, and normal hydraulic system components comprise a reliable and simple pitch change mechanism and controls. Provisions are made to allow emergency local control in the event of hydraulic failure.

1.5 THRUST BEARING

For the ventilated propeller installation, the sidehull construction in the area of the stern and transom permits placement of the thrust bearing assembly entirely within the sidehull and much closer to the propeller hub than in conventional displacement craft. The proximity



I. Propeller

1 Propeller blade
2 Blade seal
3 Double supported blade trunnion
4 Adjusting crank
5 Trunnion nut
6 Link
7 Cross head with double supported adjusting rod
8 Servomotor piston
9 Propeller hub
10 Servomotor cylinder

II. Propeller Shaft

11 Protecting hood for propeller shaft flange
12 Bushing of sterntube seal
13 Propeller shaft
14 Coupling flange
15 Double oil tube

III. Oil Distribution Unit

16 Oil distribution shaft
17 Oil distribution box housing
18 Intermediate shaft
19 Control housing
20 Feedback system
21 Control valve
22 Pilot valve

IV. Hydraulic Control System

23 Main and standby control oil pump with electric motor
24 Suction tank (yard's supply)
25 Head oil tank
26 Oil filter
27 Oil cooler
28 Hand pump for mechanical locking device
29 Mechanical locking device

V. Remote Control and Pitch Indication

30 Command transmitter
31 Slave unit
32 Mechanical connection (yard's supply)
33 Pitch setter
34 Handwheel for local emergency control
35 Mechanical pitch indication
36 Actual pitch transmitter
37 Pitch indicator

Figure 1.4-1. Typical Pitch Change Mechanism and Controls of the stern bearing to the thrust bearing permits mounting both bearings in a common housing, with both bearings being lubricated and cooled by circulating oil. This assembly is called the Thrust bearing Module. See Figure 1.5-1.

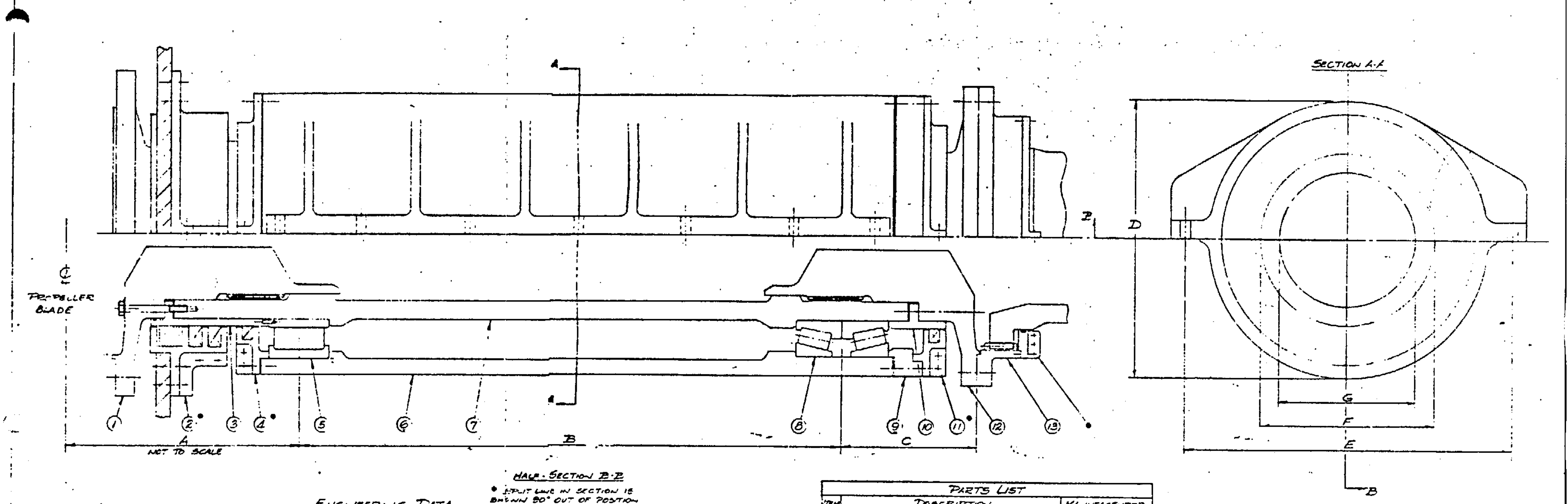
Loads are generated by the ventilated propeller in the lateral and vertical directions, as well as in the axial direction. In addition, there is a bending moment genrated by the eccentric thrust of the partially immersed propeller, and the weight load of the propeller itself.

By designing the stern bearing and the thrust bearing into a common housing, the two bearings are made to react the lateral loads, the thrust loads, and the bending moments generated by the propeller, while minimizing the risks due to wear and misalignment normally associated with water-lubricated stern bearings.

The design speed of the propellers makes possible the use of a high capacity, tapered-roller thrust bearing to absorb both the forward and reverse thrusts of the propeller, and a high-capacity cylindrical roller bearing to react the loads and moments due to the combined effects of propeller weight, vertical thrust, lateral thrust, and offset axial thrust.

The radial bearing is located aft, closest to the propeller, to minimize the propeller shaft bonding moment. The thrust bearing is located at the forward end of the housing to react the axial thrust of the propeller and the lower radial loads that occur at this location in the drive line.

The propeller hub is flange-bolted directly to the aft flange of the thrust bearing module. The aft half of the gear coupling that connects the propeller to the gearbox is flange-bolted to the forward end of the bearing module.



ENGINEERING DATA

DIMENSIONS · INCHES		
	CONFIGURATION 1	CONFIGURATION 2
A	42	48
B	60	72
C	15	17
D	31	40
E	36	46
F	18	23
G	15	15

CYLINDRICAL ROLLER BEARINGS ⑤		
PARAMETER	CONFIGURATION 1	CONFIGURATION 2
RADIAL LOAD · LBS	134800	178000
SPEED RPM	322	255
B-10 LIFE · HOURS	23000	28500
SIZE	480RU80	230RJU758
MANUFACTURER	TORRINGTON	TORRINGTON

TAPERED ROLLER BEARINGS ⑧		
PARAMETER	CONFIGURATION 1	CONFIGURATION 2
RADIAL LOAD · LBS	46600	48000
THRUST LOAD · LBS	61000	86000
SPEED RPM	322	255
B10 LIFE · HOURS	17700	18800
SIZE	188TS655	230TS755
MANUFACTURER	TORRINGTON	TORRINGTON

PARTS LIST		
ITEM	DESCRIPTION	MANUFACTURER
1	SHAFT FLANGE, PROPELLER HUB	
2	STERN SEAL ASSEMBLY, SPLIT	TYTON
3	SHAFT SLEEVE, SEALS, AFT	
4	OIL SEAL, AFT, SPLIT	TYTON
5	AFT BEARING, CYLINDRICAL ROLLER	TORRINGTON
6	HOUSING, BEARING MODULE	
7	SHAFT, BEARING MODULE	
8	FORWARD BEARING ASSEMBLY, THRUST	TORRINGTON
9	LOADING RING, THRUST BEARINGS	
10	SHAFT SLEEVE, SEAL, FORWARD	
11	OIL SEAL, FORWARD, STL	TYTON
12	SHAFT FLANGE, FORWARD	
13	GEAR COUPLING, AFT HALF	[illegible]

CONFIGURATION 1 (SHOWN): 2 · 690 KF GAS TURBINES, 1 · SACM 195 V12 DIESEL

CONFIGURATION 2: 1 · LM2500 GAS TURBINE, 1 · SACM 195V20 DIESEL

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THRUST BEARING MODULE - PROPELLER DRIVE

SIZE D | CODE IDENT NO 55917 | DRAWING NO PIA3-2A4001

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SHEET 1 OF

Appendix A. Figure 1.5-1. Thrust Bearing Module

The thrust bearing module is foot-mounted in the horizontal plane that contains the shaft centerline, to minimize the loading on the hold-down bolts from the axial thrust of the propeller. Since both bearings that support the propeller are oil lubricated, rolling contact assemblies, progressive wear and resulting misalignment that occur in water-lubricated stern bearings are eliminated. This feature is of particular importance when considering the complex loading into the bearings and hull structure that occur with ventilated propellers, and the compounded adverse effects that wear in the stern bearing would produce.

The module incorporates one cylindrical roller bearing that accommodates only radial loads, and a pair of tapered roller bearings in a 'TDI' mounting, described in Reference 5, to accommodate axial thrust in either direction while providing for slight angular misalignments resulting from shaft deflections. All bearings are lubricated by circulating oil that is filtered and cooled by the system that serves the propeller gearbox. Lube oil to the bearings is supplied by a number of jets evenly spaced around one side of each bearing annulus. The oil to the thrust bearings is fed by separate jets on each side of the assembly, from where it flows through the bearing to the space between the outer races, then to the groove and a gravity drain to a sump; the oil is pumped from this sump to the sump in the gearbox.

Horizontally split oil seals are installed at each end of the module. All parts in each seal assembly (see References 7 and 8) may be inspected and replaced without disturbing the bearings or removing the module from its mounting.

Bearings are secured and clamped by hardened steel rings that are shimmed to the proper axial fitup at assembly.

The bearing housing is made from a steel weldment that is stress relieved prior to final machining. The housing is bolted to the ship's structure at two flanges that extend radially outward from the housing, with the bolting surface lying in the horizontal plane that contains the shaft centerline. This construction minimizes eccentric loading of the module housing as well as of the ship's structure under the action of the propeller thrust.

All exposed parts of the bearing module are treated and coated to resist corrosion.

1.6 DRIVE SHAFTS AND COUPLINGS

In order to reduce weight, forged tubular drive shafts are used to transmit torque from the gearboxes to the propeller. Shaft weights are reduced to about 50 percent of the weight of a solid shaft operating at the same design stress. In the case of the propeller shafting, the central hole is used to route the hydraulic oil feed tubes for the controllable pitch propeller from the control unit, located on the forward side of the gearbox, to the propeller hub outboard of the thrust bearing module.

All shafts are machined from through-hardened alloy steel, and are designed for combined stresses in bending and torsion that do not exceed 10,000 psi at the maximum load conditions. Exposed sections of shafting are coated to resist corrosion.

Torque through the thrust bearing module is transmitted by means of flanges splined, piloted, and bolted to the shaft ends. All splines are registered by means of a long pilot on each side of the spline. In the thrust bearing module the replaceable seal sleeves are clamped by the bolting interface for the spline to minimize the number of loose pieces; the seal sleeves serve also as the clamping rings for the inner races of the radial and thrust bearings.

Between the thrust bearing module and the gearbox, the propeller torque is transmitted by gear-type flexible couplings of a standard design. The gear couplings are grease-lubricated and incorporate horizontally split grease seals for simplified installation and access. The use of couplings of special design has been avoided, relying instead on couplings of proven design with the reliability required for these components.

1.7 WATERJET PERFORMANCE

Conceptual waterjet installations were reviewed to provide comparative inputs to the propulsor selection process. Figure 1.1-1 shows that, in general, waterjet propulsors have a lower efficiency than propellers. This reflects directly on range and speed performance. The conclusion is confirmed by comparison of the ALRC PHM waterjet pumps and ventilated propellers shown in Table 1.7-1.

For low speeds, i.e., about 18 knots, the thrust/efficiency values for waterjet propulsion were calculated using two PJ-24 pumps powered by SACM195V20RVR diesel engines. This is in accordance with current practice for waterjet propelled ships whereby a separate cruise or low speed propulsion mode is furnished by a separate system. Examples of these cruise systems include the PHM hydrofoil, the "American Enterprise" crewboat and "HMS Speedy", a hydrofoil patrol ship very similar to the "Jetfoil" hydrofoil ferry. For the MDC the cruise waterjet diesel would drive the lift fans in the cushion-borne mode when propulsion is by the main LM2500 engines.

The values of thrust given in Table 1.7-1 show that for the 1500-ton MDC, two waterjets would give approximately 5 knots less off-cushion top speed when compared to two propellers and approximately 10 knots less on-cushion top speed. These differences do not take into account the weight differences of the two installations which are judged to be in favor of the propeller ship propulsion machinery largely because of the entrained water in the inlet ducting and waterjet pump. This weight difference would affect, primarily, range since less fuel could be carried.

In view of these results, waterjets were eliminated early in the study phase.

Table 1.7-1. Comparison of Waterjets and Propellers

Ship Speed, Knots		Engine Power, SHP (3)	Two Waterjets		Two Propellers	
			Thrust Lb	Efficiency	Thrust Lb	Efficiency
(1)	15	3700	46,000	.29	72,134	.46
(1)	18	3700	42,000	.31	72,127	.56
(1)	22	3700	38,000	.35	67,159	.63
(2)	30	22,500	153,776	.33	207,749	.44
(2)	40	22,500	141,920	.41	206,791	.58
(2)	50	22,500	130,847	.48	189,788	.67

(1) Off-Cushion

(2) On-Cushion

(3) Rated engine power - 2 engines per ship

2 / PERFORMANCE AND SIZING

2.1 ESTIMATING PROPELLER SIZE AND PERFORMANCE

The procedure to be shown derives and applies the K_T/J^2 method of propeller selection to calculate the diameter needed to satisfy a thrust requirement for a given speed. In a similar way, the K_Q/J^3 method is derived and applied to predict maximum thrust obtainable for a chosen propeller absorbing maximum continuous power. Equations needed for these methods, as well as computational steps, are as follows:

Equations for thrust, power delivered to the propeller (propulsion power), and propeller efficiency are:

$$\text{thrust,}\quad T = \rho n^2 d^4 K_T, \tag{1}$$

$$\text{propulsion power,}\quad \eta_G x SHP = 2\pi\rho n^3 d^5 K_Q/550, \quad \text{and} \tag{2}$$

$$\text{propulsive efficiency,}\quad \eta = \frac{TxV/550}{\eta_G x SHP}, \tag{3}$$

Using Equations (1), (2) and (3), η becomes

$$\eta = \frac{J}{2\pi} \cdot \frac{K_T}{K_Q} \qquad \text{where } J = \frac{V}{nd} \tag{4}$$

K_T/J^2 and K_Q/J^3 follows by rearranging Equation (1) to obtain:

$$K_T/J^2 = \frac{T}{\rho (Vd)^2} \tag{5}$$

and

$$K_Q/J^3 = \frac{550\ \eta_G\ SHP}{2\pi\rho (V^3 d^2)} \tag{6}$$

Definitions are:

K_T	Thrust coefficient, $T/\rho n^2 d^4$
K_Q	Torque coefficient, propeller torque$/\rho n^2 d^5$
Torque	η_G x 550 x SHP/($2\pi n$) ft-lb.
J	Advance ratio, V/nd
V	Ship speed, ft/sec
n	Propeller rotational speed, rev/sec
d	Propeller diameter, ft
SHP	Engine output power
η_G	Transmission gear efficiency (0.97)
η	Propeller efficiency
ρ	Density of water, slugs/ft^3

The propeller diameter is selected using the following steps:

1. Select or predict thrust required at maximum design speed.
2. Select several propeller diameters and for each, calculate K_T/J^2 using Equation 5.
3. Each K_T/J^2 determines the operating advance ratio, J, from the propeller data of K_T vs J.

4. Calculate propeller RPM using the definition of J.

5. For the ventilated propeller (50 percent submergence) use the design pitch curve (Reference 3, page 5-36), restricted to a maximum J of about 1.2, to avoid entering the J region where full ventilation may not occur. Calculations are made for several J values to find an acceptable combination of diameter, efficiency and RPM.

6. For the Newton-Rader fully submerged propellers, select propeller data at the cavitation index for the design speeds of Step 1. Trial calculations are made for different propeller pitch/diameter ratios to find an acceptable combination of diameter, efficiency, and RPM.

7. Calculate the low speed performance of the selected propeller, using the propeller data at lower pitch angles.

8. Modify the selected propeller design to improve low speed performance, if necessary. This is an iterative procedure between the low and maximum speed requirements.

9. Observe engine torque limits with selection of a commensurate propeller RPM.

Thrust versus velocity with maximum continuous power is calculated using the following steps.

10. Calculate K_Q/J^3 versus ship speed from Equation 6. K_Q is not available from the ventilated propeller data in Reference 1, but can be obtained from Equation (4). The equation equivalent to Equation 6 is:

$$\frac{K_T}{\eta J^2} = \frac{550\ \eta_G\ SHP}{\rho (V^3 d^2)} \tag{7}$$

11. Each K_Q/J^2 or $K_T/\eta J^2$ determines the operating advance ratio, J, from propeller data at a given pitch setting.

12. Select pitch setting for which the propeller performance is acceptable.

13. Observe engine torque limits with selection of a commensurate propeller RPM.

All computations assume a zero wake fraction, a zero thrust deduction, and unity for the relative rotational efficiency. These aspects are believed to exert a secondary effect on results and are to be taken into account for a final design.

For the conventional propeller data, Tables 2.3-1 and 2.3-2, a 6 percent appendage drag is assumed at all speeds. An examination of results with double this value (12 percent) shows the same propeller efficiency but with a slightly increased propeller diameter, slightly decreased RPM and an additional 6 percent power required.

2.2 VENTILATED PROPELLER PERFORMANCE

Ventilated propellers were sized using data given in Reference 3 which gives designs with the following characteristics:

Number of Blades	4
Expanded Area Ratio	0.6
Hub/Tip Diameter Ratio	0.4
Maximum Stress	18000 psi (fatigue limit)

Tables 2.2-1 and 2.2-2 show the propeller size and performance estimated for the ship concepts addressed in the study. Performance data for high speed (on-cushion) is shown in Table 2.2-1; corresponding data for off-cushion cruise is shown in Table 2.2-2. Table 2.2-3 shows the variation of propeller thrust with ship speed at MCP engine powers.

Comparison between Tables 2.2-1 and 2.2-2 show that while there is good efficiency for the propeller on-cushion (higher speeds) the efficiency at low speed is poor under hullborne speed conditions.

2.3 SUBMERGED PROPELLER PERFORMANCE

Submerged propellers are selected using cavitation water tunnel data given in Reference 4. This data contains the effect of blade area ratio, pitch angle and cavitation index of the water velocity approaching the propeller. An advantage of the submerged propeller is good performance both at speeds of at least 50 knots and at low speeds. Characteristics of the chosen design are:

Number of Blades	3
Blade Area Ratio	0.48
Hub/Tip Diameter Ratio	0.3
Thrust/Disc Area	Less than 1000 psf

Table 2.2-1. Ventilated Propeller Characteristics, On-Cushion

SHIP WEIGHT, LT	SPEED KT (1)	REQ. SHIP THRUST LB (2)	ONE PROPELLER (EACH SIDEHULL)				
			POWER AVAILABLE SHP (3)	REQ. POWER SHP	DIA FT	RPM	EFFICIENCY
940	40	108,000	11,297	10,205	11.60	322	.65
1060	40	122,000	12,610 (4)	11,528	12.34	303	.65
1200	50	171,000	21,825	19,306	12.49	350	.68
1380	50	190,000	26,190 (4)	21,451	13.16	332	.68
1800	40	172,000	21,825	16,252	14.64	255	.65

(1) Design speed - Sea State 3
(2) Equal to ship drag at design speed - Sea State 3
(3) Gas Turbine MCP X .97 (Transmission Efficiency)
(4) Uprated Gas Turbines

Table 2.2-2. Ventilated Propeller Characteristics, Off-Cushion

SHIP WEIGHT, LT	SPEED KT (1)	REQ. SHIP THRUST LB (2)	ONE PROPELLER (EACH SIDEHULL)				
			POWER AVAIL-ABLE SHP (3)	REQ. POWER SHP	DIA FT	RPM	EFFICIENCY
940	17	42,000	2,134	2,134	11.60	242	.47
1060	15.5	42,500	2,134	2,134	12.33	225	.48
1200	19	78,000	4,365	4,365	12.49	275	.48
1380	17	72,200	4,365	4,365	13.16	254	.44
1800	17.5	77,500	4,365	4,365	14.64	210	.48

(1) Maximum Speed - Sea State 3
(2) Equal to ship drag at speed quoted - Sea State 3
(3) Diesel MCP X .97 (Transmission Efficiency)

Table 2.2-3. Ventilated Propeller Thrust vs Speed at Maximum Cruise Power, On Cushion

SHIP WEIGHT, LT	PROPELLER DIAMETER, FEET	SHIP SPEED, KNOTS		
		30	40	50
		THRUST, LBS(1)		
940	11.60	132,408	119,558	91,221
1060	12.33	147,797	133,454	101,824
1200	12.49	198,996	202,550	190,468
1380	13.13	237,380	234,532	221,057
1800	14.64	236,854	222,094	194,761

(1) Thrust at MCP powers shown in Table 2.2-1, Column 4

Tables 2.3-1 and 2.3-2 show propeller size and performance estimated for the MDC and its variants. Performance data for high speed (on-cushion) are shown in Table 2.3-1; corresponding data for off-cushion cruise is shown in Table 2.3-2. Table 2.3-3 shows the variation of propeller thrust with ship speed at MCP engine powers.

Comparison between Tables 2.3-2 and 2.2-2 show that the submerged propeller maintains high efficiency at off-cushion operating speeds.

2.4 THRUST BEARING SIZING

The thrust bearing module shown in Figure 1.5-1 (Drawing PIA 3-244001) is sized to support both the radial and the thrust loads developed by the ventilated propellers designed for Configurations I and II, when the respective turbine power plants are operating at maximum continuous power. Configuration I is the twin DDA570 gas turbine propulsion plant; Configuration II is the LM2500 gas turbine propulsion plant. These are described in Appendix B.

The span between the radial bearing and the thrust bearing for each configuration was determined such that the resulting combined radial and thrust load into the thrust bearing was within the capacity of a commercially available tapered roller thrust bearing (Reference 5) and that the B10 life would be in excess of 15000 hours. The thrust bearing configuration is designed to absorb maximum continuous propeller thrust in either direction. This configuration requires a pair of bearings, thereby distributing the radial load between the two bearings, and results in the smallest bearing size for the combined load.

The thrust bearing is located forward of the radial bearing to minimize the radial loads into the thrust bearing. This reduces the combined load into the thrust bearing and the thrust bearing size. Calculations show that the thrust bearing is the more critically loaded of the two bearings; therefore the thrust bearing size will determine the dimensions

Table 2.3-1. Submerged Propeller Characteristics, On-Cushion

SHIP WEIGHT, LT	SPEED KT (1)	REQ. SHIP THRUST LB (2)	ONE PROPELLER (EACH SIDEHULL)					
			POWER AVAIL- ABLE SHP (3)	REQ. POWER SHP	DIA FT	RPM	EFFICIENCY
940	40	117,021	11,297	9,758	9.64	248	.737
1060	40	132,979	12,610 (5)	11,029	10.54	224	.741
1200	50	170,213	21,825	18,060 (4)	11.61	249	.724
1380	50	200,000	26,190 (5)	21,700	11.61	269	.708
1800	40	186,170	21,825	15,868 (4)	11.61	214	.721

(1) Minimum top speed - Sea State 3
(2) Equal to 1.06 X ship drag at design speed - Sea State 3
(3) Gas Turbine MCP X .97 (Transmission Efficiency)
(4) Significant overload capability
(5) Uprated Gas Turbines.

Table 2.3-2. Submerged Propeller Characteristics, Off-Cushion

SHIP WEIGHT, LT	SPEED KT (1)	REQ. SHIP THRUST LB (2)	ONE PROPELLER (EACH SIDEHULL)				
			POWER AVAIL-ABLE SHP (3)	REQ. POWER SHP	DIA FT	RPM	EFFICIENCY
940	18	53,191	2,134	1,956	9.64	180	.752
1060	17.5	63,830	2,134	2,134	10.54	165	.752
1200	18	74,468	4,365	2,713	11.61	147	.759
1380	18	79,787	4,365	2,934	11.61	150	.752
1800	18	90,426	4,365	3,398	11.61	155	.736

(1) Design speed - Sea State 3
(2) Equal to 1.06 X ship drag at design speed - Sea State 3
(3) Diesel MCP X .97 (Transmission Efficiency)

Table 2.3-3. Submerged Propeller Thrust vs Speed at Maximum Cruise Power, On Cushion

SHIP WEIGHT, LT	PROPELLER DIAMETER, FEET	SHIP SPEED, KNOTS		
		30	40	50
		THRUST, LBS(1)		
940	9.64	142,929	131,240	102,744
1060	10.54	164,279	149,229	114,313
1200	11.61	242,223	218,076	195,856
1380	11.61	268,303	239,906	222,781
1800	11.61	218,022	237,390	195,856

(1) Thrust at MCP powers shown in Table 2.3-1, Column 4.

of both the shaft and the housing. In order to minimize the size and weight of the shafting and housing it is essential that the radial loads into the thrust bearing be reduced to the maximum extent practicable.

The length and proportions of the bearing module and its mounting flanges are adequate to distribute the radial and thrust loads of the propeller into the ship's structure.

The shaft size is determined from the thrust bearing bore and is considerably larger than the diameter of a solid shaft required to transmit the design torque at a 6000 psi torsional shear stress. The larger diameter allows a hollow shaft to be used. For a 6000 psi torsional shear stress the weight of the hollow shaft for each configuration is about half the weight of a solid shaft, when the outside diameter of the hollow shaft is based on the thrust bearing size. In order to utilize a shaft of constant section over the length of the drive line the bore size of the cylindrical roller bearing is the same as the bore size of the thrust bearing. The resulting cylindrical roller bearing has excess capacity, for the radial loads imposed. This results in a greater B10 life for this bearing than for the tapered roller bearing.

Since the tapered roller thrust bearing and the cylindrical roller radial bearing selected for each configuration have comparable bore sizes and outer diameters, the bearing housing configurations are cylindrical and free of large section changes, reducing any tendencies for thermal distortion during load transients or under a wide range of steady state loads. All bearings in each module have line-contact rolling elements.

The bearing loads, speeds, sizes and B10 lives are given in Figure 1.5-1.

2.5 DRIVE SHAFTING SIZING

The size of the drive shafting in each configuration was determined by the bore size of the thrust bearing required to react the propeller thrust load. An additional factor was the need for a hollow shaft in which to install the piping and control transfer mechanism for the controllable pitch propeller.

The weight of the drive shafting was determined by the outside diameter and the design stress level. The shafting for these applications is designed primarily to transmit the torque from the gearbox to the thrust bearing module, with a very minimum of bending. The radial loads and bending moments due to the ventilated propeller are reacted by the bearings in the thrust bearing module (thrust block) at the transom. The shaft bore size was determined on the basis of the allowable torsional shear stress, which was set at a maximum value of 6000 psi at maximum engine torque.

The shaft sizes for both propulsion plant configurations, as determined from these considerations, are tabulated in Figure 1.5-1.

2.6 SHAFT COUPLINGS

Because of the low propeller speeds in both configurations, the large propeller shaft torques are beyond the range of most commercially available flexible couplings. Flexible couplings are considered necessary to accommodate misalignments, thermal differential expansion between the steel machinery and the aluminum sidehulls, and distortions in the sidehulls due to conditions that include off-cushion and on-cushion operation under wide ranges of loading in various sea states. A preliminary survey of available couplings indicated that the propeller torques for both propulsion plant configurations exceed the capacities of commercially available disc and diaphragm couplings. Available grease-lubricated gear couplings manufactured by Zurn (Reference 6) were found to have torque capacities adequate not only for the present torque loads, but for torque loads well in excess of present requirements. In the torque ranges in which disc couplings are available, the weights of the disc and gear couplings are comparable. Therefore, it is expected that no weight penalties will develop when gear couplings are applied to the subject drive lines. From the layouts made in the course of developing the concepts for the thrust bearing modules, the proportions of the gear couplings are consistent with those of the shafting and the thrust bearing modules.

3 / TECHNICAL RISK

Of the propulsors evaluated, the submerged (conventional) propeller for high speed craft, is considered to be the lowest risk. Many have been, and are currently being used, for high speed naval craft such as the PC and PCG. Vendors are therefore experienced in the manufacture of these propellers and much is known about propeller performance. The ventilated propeller does not enjoy the benefit of previous service experience except comparatively briefly in the SES-110B testcraft. It follows that manufacturing experience for the ventilated propeller is lacking. Performance data for ventilated propellers needs expansion into the areas of off-design, low pitch ratio, side forces and backing. Complete experimental/model data will reduce this risk to near that of the submerged propeller.

Pitch change mechanisms and controls have minimal technical and producibility risk when associated with submerged propellers. These systems are now employed extensively on commercial and naval ships. The same mechanisms and controls associated with ventilated propellers have some risk because of the unknown effects of the larger lateral forces and cyclic loading inherent to ventilated propellers. Thrust bearings, shafting, and couplings are rated similarly to the pitch change mechanisms for the same reasons.

4 / RELIABILITY

This reliability discussion is confined to the propeller and associated equipment. The reliability of other propulsion machinery elements is addressed in Appendix B. The Tacoma Boat/Escher Wyss propeller design includes the pitch controls, instruments, sensors, the hydraulic system which contains pumps, hydraulic controls, oil coolers, filters, valves and fittings, the propeller and the pitch change mechanism.

Since the ship thrust and the ship maneuvering capability are dependent on this system, high reliability of these parts are mandatory. The Tacoma Boat/Escher Wyss propeller and associated equipment, has proven capabilities on previous designs. The layout drawings of the various configurations are presented in Appendix B, Section 1.5. The highest probability of failure occurs in the elements of the hydraulic system. For this reasons redundancy in pump and controls is provided. Additional redundancy is provided in the manual control. In the event of a complete breakdown of the hydraulic system, the propeller can be mechanically locked in an ahead position by means of the linkage between the propeller blades and the double oil tube.

5 / MAINTAINABILITY

This discussion of maintainability is confined to the propeller and associated equipment. Maintainability of other propulsion machinery is addressed in Appendix B.

The Escher Wyss propeller is a highly maintainable design. The mechanical pitch change mechanisms in the propeller hub is designed for the life of the ship and requires no maintenance. The hydraulic activation system which requires some maintenance is installed in the shafting within the ship. Accessibility is provided for all scheduled and unscheduled maintenance required in the mechanism itself. Access from the engine room around the equipment is satisfactory for the various installations.

Escher Wyss has estimated a 6.9 hour MTTR for the hydraulics and controls. Ready access for these systems on all the machinery arrangements presented assure that the baseline MTTR will hold at 6.9 hours.

All similar parts, including repair parts, are interchangeable without additional machining or selective assembling. Daily maintenance is minimal involving oil level and oil filter checks.

6 / PRODUCIBILITY

Propeller size and materials for fully submerged propellers do not pose producibility problems since much larger conventional propellers in similar materials have been manufactured and are in service both in commercial and naval ships. For the ventilated propeller, there may be a need for manufacturing development since the blades will be made of harder and tougher alloys to resist the generally higher steady state and alternating stress levels used for ventilated propellers. It is felt, however, that overall, the task of producing either a conventional submerged propeller or a ventilated propeller is about the same providing some manufacturing development is performed on the ventilated propeller.

The other aspects of the propeller, the CP mechanism and controls, present no producibility problems, being state-of-the-art and familiar to vendors.

The thrust bearing module and shafting are custom designed. They present no producibility problems. The bearings and couplings are catalogue items and seals are similar to items in service.

3. Ship speeds for which output is reuired.

4. Propeller immersion.

5. Propeller diameters for parametric studies.

6. Propeller charts for a range of cavitation indicies (fully submerged propeller) -- the program interpolates to obtain a propeller chart for each ship speed (cavitation index).

Table 7.1-1A, B and 7.1-2A, B are sample printouts of the propeller program for the ventilated propeller for the 1500 LT ship at high and low speeds showing thrust versus velocity for a fixed power input. The propeller diameter is 12.5 feet. Engine SHP is 22,500/propeller (25 to 50 knots) and 3700 SHP (15 to 22 knots).

Table 7.1-1A shows the maximum possible thrust for two 12.5 foot diameter propellers (1500 LT ship) with two x 22,500 engine SHP. Such a maximum requires both blade angle and propeller rpm change with ship speed. The rpm change versus ship speed, converted to engine rpm, is more than the allowable change for the engine. Consequently, Table 7.1-1B shows thrust attainable for a constant 358 rpm versus ship speed; this rpm is near the value for 50 knots in Table 7.1-1A. SHP in Table 7.1-1B is the same as used in Table 7.1-1A. Thrusts in Table 7.1-1B are somewhat less than values in Table 7.1-1A. Table 7.1-2A shows the maximum possible thrust for two 12.5 foot diameter propellers (1500 LT ship) with two engines of 3700 SHP. The rpm change is more than that allowable for operation of the engine. Consequently, Table 7.1-2B shows thrust attainable for a constant 187 rpm versus ship speed; this rpm is near the value for 18 knots in Table 7.1-2A. Again, SHP is the same and thrust is somewhat lower than maximum obtainable. Tables 7.1-1 and 7.1-4 contain 12 columns; namely:

7 / PROPELLER STUDIES

7.1 PROPELLER SIZING AND PERFORMANCE COMPUTER PROGRAM

Propeller selection for the 1500 ton ship is obtained using an RMI computer program. Pertinent aspects of this program, analysis and propeller data follow. The program is applicable to all ships and propellers studied.

7.1.1 TYPE OF PROGRAM CALCULATIONS, INPUT AND OUTPUT DATA

A. Program Calculates (for each input SHP and propeller diameter):

1. Maximum possible thrust for each speed (low and high speeds).
2. Corresponding values of thrust, rpm, pitch angle, J, K_T, K_Q and N (defined in Section 2.1).
3. Highest thrust obtainable versus velocity for a selected blade diameter and rpm (constant) and corresponding pitch angles.*

B. Input Data:

1. Propeller chart values of J, K_T, K_Q.
2. Engine horsepower, SHP and transmission efficiency.

---

* The blade diameter is selected by considering (1) available space for installing the propeller, (2) a good low speed efficiency to obtain an acceptable range, (3) a good high speed efficiency to obtain the required maximum speed and (4) an acceptable propeller rpm.

Table 7.1-1A and B. Ventilated Propeller On-Cushion Performance

VENTILATED PROPELLER PROGRAM

HIGH SPEED DIAMETER=12.5 22500 HP

ENGINE HORSEPOWER = 22500.0
SHIP WEIGHT = 1500.0 LT
PROP DIAMETER = 12.500 FT
NUMBER OF PROPELLERS = 2

	VELOCITY KNOTS	THRUST LBS	T/A	DESIGN-X	RPM	ADV RATIO J	THRUST COEFF KT	TORQUE COEFF KQ	PROP EFFIC ETA	KQ/J3	KT/J2	PROP TORQUE FT-LB
A	25.0	194603.6	792.9	-10.000	496.77	0.408	0.0292	0.0055	0.3408	0.0818	0.1757	237879.70
	30.0	207749.5	846.4	-10.000	463.32	0.525	0.0358	0.0068	0.4385	0.0473	0.1303	255058.68
	35.0	211779.8	862.9	-7.005	410.23	0.691	0.0467	0.0098	0.5205	0.0298	0.0978	288062.95
	40.0	206800.3	842.6	-5.220	392.24	0.826	0.0498	0.0113	0.5844	0.0200	0.0730	301278.16
	41.1	205069.3	835.5	-5.076	392.76	0.848	0.0492	0.0112	0.5960	0.0184	0.0685	300876.93
	50.0	** OUTSIDE AVAILABLE PROPELLER DATA **										
	55.0	** OUTSIDE AVAILABLE PROPELLER DATA **										
	60.0	** OUTSIDE AVAILABLE PROPELLER DATA **										

	VELOCITY KNOTS	THRUST LBS	T/A	DESIGN-X	RPM	ADV RATIO J	THRUST COEFF KT	TORQUE COEFF KQ	PROP EFFIC ETA	KQ/J3	KT/J2	PROP TORQUE FT-LB
B	25.0	109861.2	447.6	0.000	358.00	0.473	0.0318	0.0086	0.2788	0.0474	0.0992	330090.98
	30.0	191832.9	781.6	-0.383	358.00	0.679	0.0554	0.0148	0.4058	0.0472	0.1203	330090.98
	35.0	207604.1	845.9	-2.560	358.00	0.792	0.0600	0.0148	0.5121	0.0298	0.0956	330090.98
	40.0	204859.6	834.7	-2.554	358.00	0.905	0.0592	0.0148	0.5805	0.0199	0.0723	330090.98
	49.1	210771.9	856.8	0.000	358.00	1.150	0.0609	0.0164	0.6219	0.0156	0.0587	330090.98
	50.8	189627.0	772.6	0.000	358.00	1.150	0.0548	0.0148	0.6673	0.0102	0.0428	330090.98
	55.0	** OUTSIDE AVAILABLE PROPELLER DATA **										
	60.0	** OUTSIDE AVAILABLE PROPELLER DATA **										

Table 7.1-2A and B. Ventilated Propeller Off-Cushion Performance

VENTILATED PROPELLER PROGRAM

LOW SPEED INPUT DIAMETER=12.5 MULTIPLIER=1.5 3700 HP

ENGINE HORSEPOWER = 3700.0
SHIP WEIGHT = 1500.0 LT
PROP DIAMETER = 12.500 FT
NUMBER OF PROPELLERS = 2

A

VELOCITY KNOTS	THRUST LBS	T/A	DESIGN-X	RPM	ADV RATIO J	THRUST COEFF KT	TORQUE COEFF KQ	PROP EFFIC ETA	KQ/J3	KT/J2	PROP TORQUE FT-LB
15.0	72134.2	293.9	-10.000	219.84	0.553	0.0553	0.0105	0.4621	0.0622	0.1809	88394.82
16.0	72454.5	295.2	-10.000	219.62	0.590	0.0556	0.0105	0.4934	0.0513	0.1597	88484.34
17.0	72721.0	296.3	-6.758	194.68	0.707	0.0713	0.0152	0.5282	0.0428	0.1424	99817.95
18.0	72169.5	294.0	-6.100	191.36	0.762	0.0731	0.0160	0.5563	0.0360	0.1259	101551.14
19.0	71208.4	290.1	-5.227	187.48	0.821	0.0751	0.0169	0.5810	0.0306	0.1114	103654.78
19.7	70461.3	287.1	-5.076	187.98	0.848	0.0739	0.0168	0.5960	0.0276	0.1028	103378.18
21.0	** OUTSIDE AVAILABLE PROPELLER DATA **										
22.0	** OUTSIDE AVAILABLE PROPELLER DATA **										

B

VELOCITY KNOTS	THRUST LBS	T/A	DESIGN-X	RPM	ADV RATIO J	THRUST COEFF KT	TORQUE COEFF KQ	PROP EFFIC ETA	KQ/J3	KT/J2	PROP TORQUE FT-LB
15.0	70968.7	289.2	-4.863	187.00	0.650	0.0752	0.0171	0.4548	0.0622	0.1780	103918.84
16.0	72381.5	294.9	-5.406	187.00	0.693	0.0767	0.0171	0.4941	0.0513	0.1596	103918.84
17.0	72653.3	296.0	-5.553	187.00	0.736	0.0770	0.0171	0.5276	0.0428	0.1419	103918.84
18.0	72171.4	294.1	-5.422	187.00	0.780	0.0764	0.0171	0.5561	0.0360	0.1257	103918.84
19.0	71222.4	290.2	-5.154	187.00	0.823	0.0754	0.0171	0.5810	0.0306	0.1113	103918.84
20.0	91470.1	372.7	0.000	187.00	1.150	0.0969	0.0261	0.5892	0.0402	0.1291	103918.84
23.3	88098.7	358.9	0.000	187.00	1.150	0.0933	0.0252	0.6108	0.0334	0.1127	103918.84
23.6	84653.7	344.9	0.000	187.00	1.150	0.0897	0.0242	0.6313	0.0279	0.0987	103918.84

Column 1 Ship velocity, knots

Column 2 Maximum obtainable ship thrust for each velocity, pounds

Column 3 Thrust/wetted disc area -- a propeller loading parameter, pounds per square foot

Column 4 Blade pitch angle in the form, design angle minus X^o, where X is the output value, degrees

Column 5 Propeller rpm

Column 6 Propeller advance ratio, J (= V/nd)

Column 7 Thrust coefficient, K_T

Column 8 Torque coefficient K_Q

Column 9 Propeller efficiency, η

Column 10 K_Q/J^3

Column 11 K_T/J^2

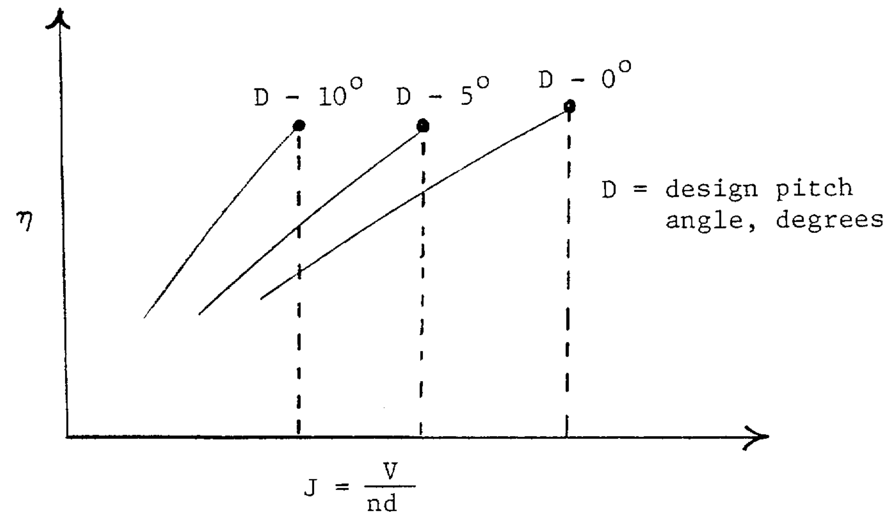
Column 12 Torque on propeller shaft, ft lb

See Section 2.1 for considerations related to the propeller parameters in columns 7 to 11.

7.1.2 ANALYSIS -- The computer program procedure uses methods similar to those described in Section 2.1. Steps 1-7 pertain to maximum possible thrust versus ship speed for a given diameter and SHP; rpm is allowed to vary. Steps 8-15 are similar except rpm is held fixed with ship speed.

1. Calculate propeller diameter that will produce the maximum possible thrust for a designated ship speed, the corresponding blade pitch angle and rpm. Use this diameter at all ship speeds to obtain thrust versus velocity with constant SHP. RPM will varry with ship speed. Steps 2 to 6 describe this procedure.

2. Blade diameter d is calculated using the J value (and designated ship speed) where the efficiency is maximum for each pitch angle, as illustrated below for the ventilated propeller.



3. At each J value read corresponding K_T, K_Q and η. Calculate K_Q/J^3 for each J value read.

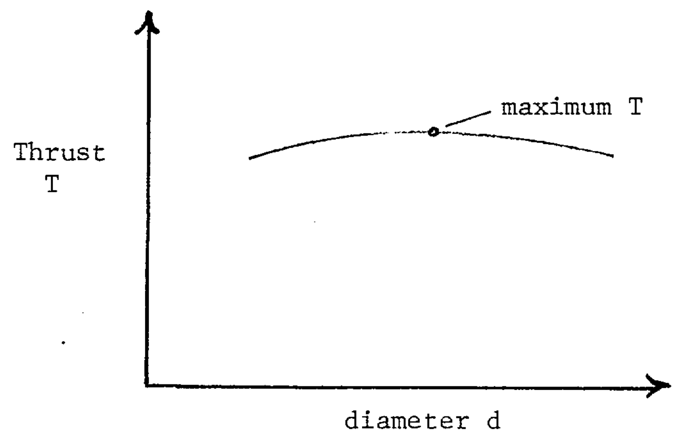
4. Calculate blade diameters d from

$$d = \sqrt{\frac{550\ \eta G\ SHP}{2\pi\rho V^3 \quad K_Q/J^4}}$$

This formula is derived from Equation 6 of Section 2.1.

5. Calculate propeller revolutions/second from J = V/nd for each diameter. Calculate corresponding thrust T versus d from Equation 1 of Section 2.1.

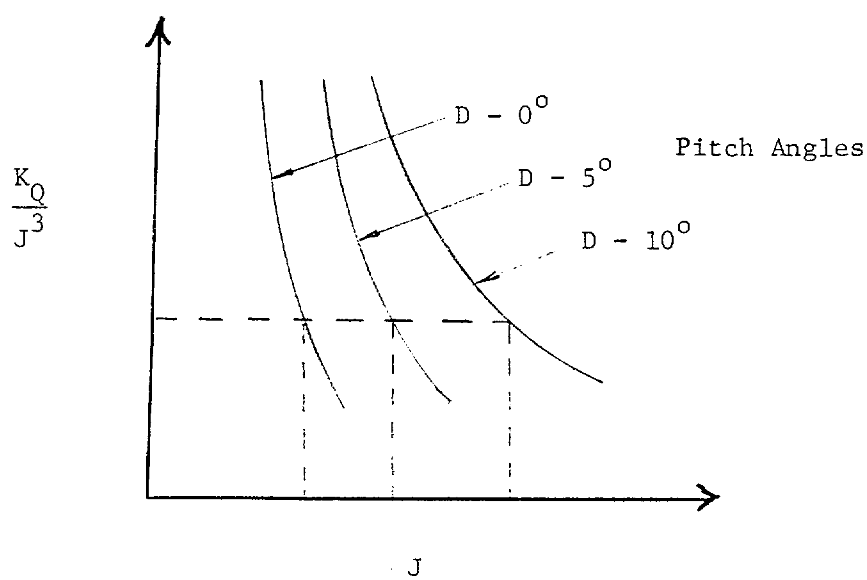
6. Find diameter where thrust is maximum as illustrated.



Find corresponding J, K_T, K_Q, η and pitch angle from propeller data.

7. Select a design rpm to be held constant with ship speed and calculate maximum possible thrust with the selected rpm and corresponding blade pitch angle needed at each ship speed.

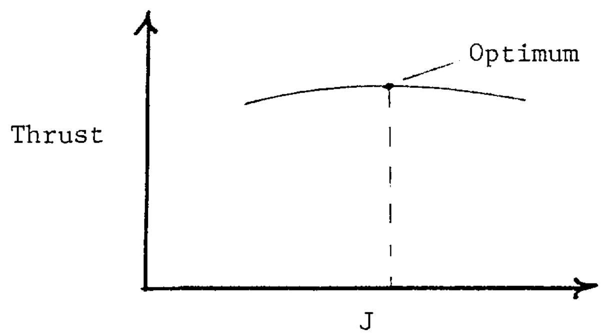
8. Steps 1 and 7 require the calculation of K_Q/J^3 versus ship speed according to Equation 6 of Section 2.1. Find the advance ratio J from K_Q/J^3 to enter in input propeller data shown in the following sketch for the ventilated propeller.



9. For each calculated K_Q/J^3 of step 8 read a J value at each pitch. Enter the propeller data at each J and read a thrust coefficient K_T.

10. Calculate thrust from Equation 1 for each K_T value.

11. Plot thrust versus J and find J for optimum thrust as illustrated



12. Find corresponding pitch angle from data in step 9; also find corresponding K_T, K_Q and η from propeller charts.

To calculate maximum thrust versus ship speed for a fixed SHP and rpm:

13. Calculate advance ratio J at each ship speed for a fixed SHP and rpm using Equation 4, Section 2.1.

14. Calculate K_Q/J^3 versus ship speed as noted in step 8.

15. Repeat steps 9 to 12.

7.1.3. PROPELLER DATA -- Table 8.1-3 lists the ventilated propeller data, for 50% diameter submergence, as used in the computer program. This data is taken from Reference 3 and shows K_T, K_Q and η versus J for blade angles D, D-5° and D-10°. No extrapolation of the data is allowed in the computer program.

For 100% blade diameter submergence, Table 7.1-3 data is factored as follows:

a. K_T and K_Q values at each J for 50% submergence are multiplied by 1.5.

Table 7.1-3. Propeller Input Data

DESIGN - 0. (degrees)

FREESTREAM ADVANCE RATIO	NET THRUST COEFF	TORQUE COEFF	PROP EFF
0.050	0.0010	0.0003	0.0295
0.100	0.0020	0.0005	0.0590
0.150	0.0035	0.0009	0.0885
0.200	0.0040	0.0011	0.1180
0.250	0.0090	0.0024	0.1475
0.300	0.0110	0.0030	0.1770
0.350	0.0175	0.0047	0.2065
0.400	0.0238	0.0064	0.2360
0.450	0.0295	0.0080	0.2655
0.500	0.0345	0.0093	0.2950
0.550	0.0410	0.0111	0.3245
0.600	0.0471	0.0127	0.3540
0.650	0.0530	0.0143	0.3835
0.700	0.0575	0.0155	0.4130
0.750	0.0620	0.0167	0.4425
0.800	0.0650	0.0175	0.4720
0.850	0.0669	0.0180	0.5015
0.900	0.0670	0.0181	0.5310
0.950	0.0675	0.0182	0.5605
1.000	0.0645	0.0174	0.5900
1.050	0.0612	0.0165	0.6195
1.100	0.0575	0.0155	0.6490
1.150	0.0530	0.0143	0.6785

DESIGN - 5. (degrees)

FREESTREAM ADVANCE RATIO	NET THRUST COEFF	TORQUE COEFF	PROP EFF
0.050	0.0015	0.0005	0.0352
0.100	0.0035	0.0008	0.0703
0.150	0.0055	0.0012	0.1055
0.200	0.0085	0.0019	0.1406
0.250	0.0115	0.0026	0.1758
0.300	0.0150	0.0034	0.2109
0.350	0.0210	0.0048	0.2461
0.400	0.0265	0.0060	0.2812
0.450	0.0322	0.0073	0.3164
0.500	0.0370	0.0084	0.3515
0.550	0.0420	0.0095	0.3867
0.600	0.0460	0.0104	0.4218
0.650	0.0500	0.0113	0.4570
0.700	0.0522	0.0118	0.4921
0.750	0.0530	0.0120	0.5273
0.800	0.0520	0.0118	0.5624
0.850	0.0495	0.0113	0.5960

DESIGN - 10.

FREESTREAM ADVANCE RATIO	NET THRUST COEFF	TORQUE COEFF	PROP EFF
0.050	0.0030	0.0006	0.0418
0.100	0.0050	0.0010	0.0836
0.150	0.0073	0.0014	0.1254
0.200	0.0103	0.0020	0.1672
0.250	0.0140	0.0027	0.2090
0.300	0.0185	0.0035	0.2508
0.350	0.0233	0.0044	0.2926
0.400	0.0285	0.0054	0.3344
0.450	0.0325	0.0062	0.3762
0.500	0.0348	0.0066	0.4180
0.550	0.0368	0.0070	0.4598
0.600	0.0370	0.0070	0.5016
0.650	0.0356	0.0068	0.5434
0.700	0.0310	0.0059	0.5852

b. Efficiency η is kept the same as for 50% submergence.

c. The 1.5 factor is based on test data in Reference 10.

Table 7.1-6 lists the 100% submergence data. This data is used to predict thrusts and efficiency for low speed (15 - 22 knot) 3700 SHP, 12.5 diameter propeller.

7.1.4 PROPELLER SUBMERGENCE MULTIPLIER -- The relationship between the propeller submergence multiplier and the calculated propeller efficiency (from the computer program), is shown in Figure 7.1-1. Correspondence between the submergence of the propeller and the propeller submergence multiplier is shown in the inset to Figure 7.1-1.

Use of these two curves enables the designer to determine the low ship speed propeller efficiency based on the measured propeller submergence area ratio, propeller area immersed divided by total propeller area. Both of these areas exclude the area of the hub. This data is not used for higher ship speeds (on-cushion).

7.2 PROPELLER INSTALLATION

Machinery requirements for the ventilated propeller are found to be simpler mainly in respect to the reduction gearbox size and complexity. Since the propellers and engines are set at an angle of 8 degrees to the ship baseline for both submerged and ventilated propellers, the ventilated propeller installation always requires a reduced offset distance between the centerline of the engines and the propeller shaft, using parallel gears. Concepts using a bevel gear, crossed axis helical gears or universal joints were examined (see Appendix B, Section 8.5 for a discussion of these). These could allow installation of the propeller at a steeper angle to the ship baseline while keeping the engines at 8 degrees or less. However, the angles drive and gearing would require much development testing to reduce risk in the manufacturing and life areas. This risk is felt to be incompatible with the improved producibility of the MDC and therefore parallel gearing only is used.

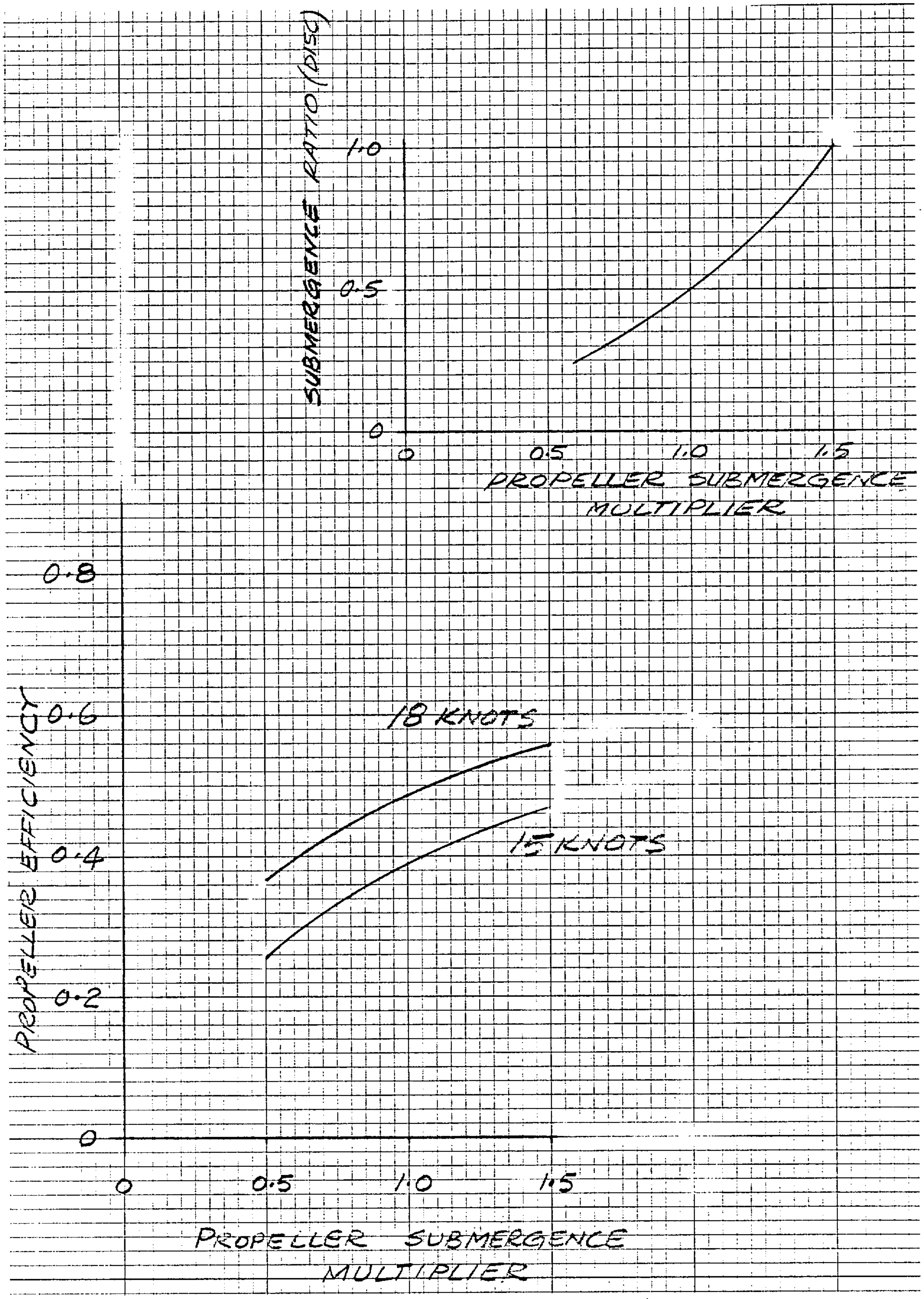


Figure 7.1-1. Submergence and Efficiency

Comparison between the ventilated propeller and submerged propeller concept installations, shown in Appendix B, Figures 1.3-1 and 1.3-3 respectively, illustrate the differences in machinery between these propeller types. Clearly, within the confines of the space given to the machinery spaces, the ventilated propeller machinery is simpler and thus inherently more reliable, more maintainable and less costly.

7.2.1 PROPELLER PERFORMANCE -- The performance of submerged and ventilated propellers and other types of propulsors is shown in Figure 7.2.1. This figure is similar to Figure 1.1-1 but is modified by the addition of the two curves for the 12.5 feet diameter propellers selected for the 1500-ton MDC. These curves, obtained by use of the RMI propeller sizing and performance computer program described in Section 7.1, are representative of the ventilated propeller envisaged for the MDC. The propeller characteristic data for use in the computer program are taken rom Reference 3, the Navy MDC report. Use of this data enables prediction of ventilated propeller performance which is better at lower ship speeds than was originally concluded from Figure 1.1-1, which is a composite of data for general cases of ventilated propellers.

Comparison of the submerged and ventilated propeller efficiencies shows the submerged propeller to be superior by approximately 5 percent at a ship speed of 18 knots and at 50 knots to be the same. Below 18 knots (off-cushion) and below 50 knots (on-cushion) the submerged propeller is superior. However, the efficiency losses associated with each propeller type installation and the uncertainties associated with each installation differ. For the ventilated propeller there is little drag associated with its installation. A small fairing before the propeller loss and a rudder forward of the propeller are used on each sidehull (see Appendix B, Figure 1.3-1); for the submerged propeller there is judged to be more drag associated with its installation. A summary of rudder and appendage drag aspects follow.

*Curve A - 12.5 feet diameter ventilated propeller, 50 percent submerged.
*Curve B - 12.5 feet diameter ventilated propeller, maximum submergence.

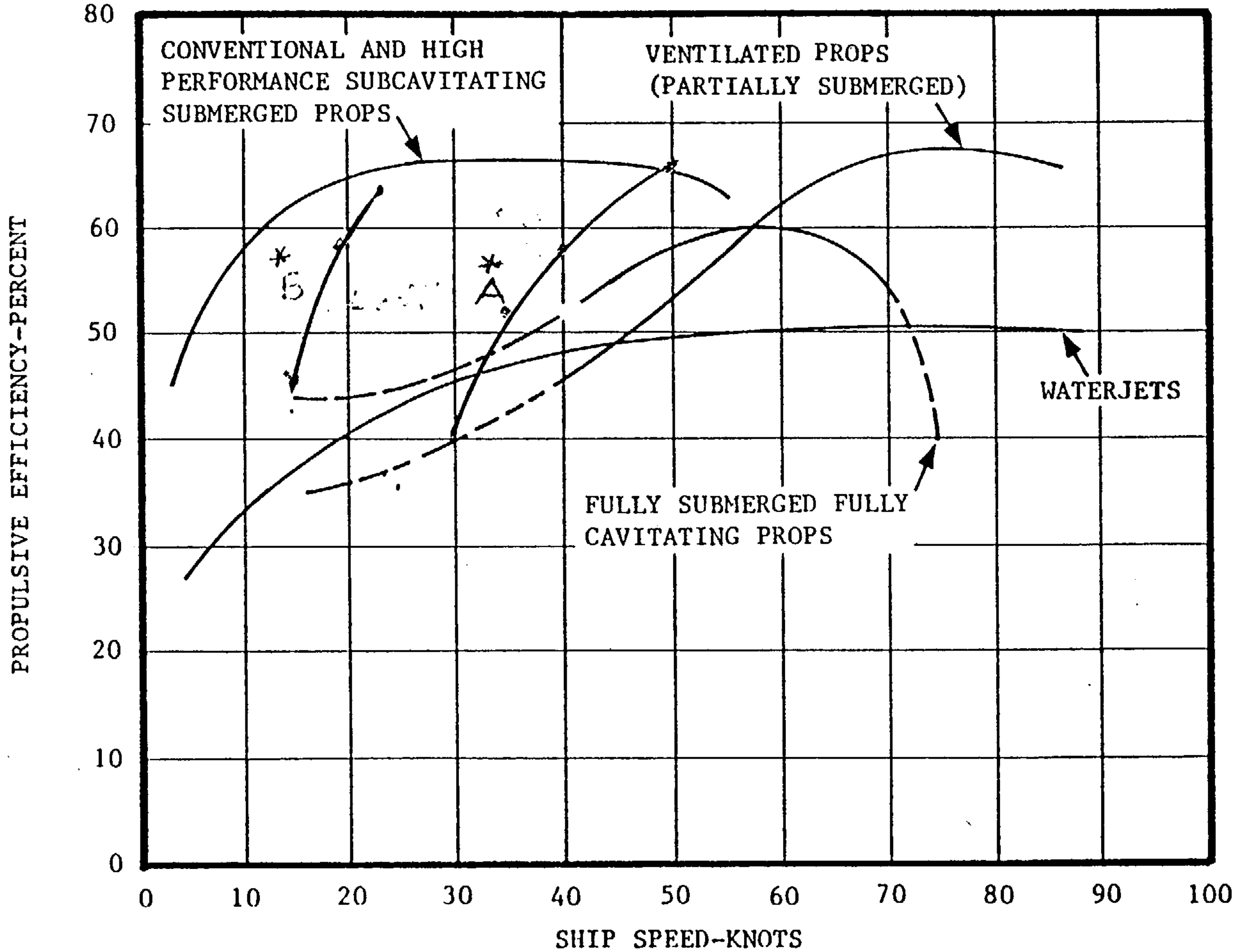


Figure 7.2-1. Approximate Maximum Installed Efficiency Envelopes for Propellers and Waterjets (Current Design) with Efficiency of 12.5 Feet Diameter Propeller

RUDDER DRAG

Ventilated Propeller	Submerged Propeller
• Larger area rudder not in propeller jet.	• Smaller rudder in propeller jet
• Propeller jet swirl not reduced by rudder ahead of propeller disc.	• Propeller jet swirl reduced by rudder; can increase propeller efficiency.
• No propulsion benefit from rudder. Effect of rudder flow into propeller requires evaluation but estimated small.	• Possible efficiency benefit requires evaluation but estimated small

OTHER APPENDAGE DRAG

• Propeller hub drag is assumed included in Navy performance data.	• Propeller hub drag included in Newton and Radir data.
• No struts, bossings, shaft fairings.	• Drag associated with struts, bossings and shaft fairing.
• Drag of hub/thrust bearing module fairing ahead of the propeller not included in performance.	• Appendage drag of 9 percent included in submerged propeller performance predictions based on planing craft data. Requires further evaluation for SES craft.
• No appendage drag used in ventilated propeller performance. Requires evaluation.	

Prediction of added drags are possible by analytical methods but testing is needed. Such work is not included in the scope of this effort documented herein.

Both propeller types have been successfully demonstrated at high ship speeds. The ventilated propeller is fitted to the SES-100B. The submerged propeller type has been fitted to 50 knot craft and has performed well. At this time adequate performance for the MDC is indicated using either propeller type based on available data. However, any propeller and later the propeller installation selected for the MDC will require orderly testing to determine performance, cavitation, installation flow interferences, vibration and engine matching characteristics.

7.3 METHODS OF INCREASING VENTILATED PROPELLER IMMERSION

The following describes modified sidehull shapes ahead of the ventilated propeller studied for the purpose of improving low ship speed propeller performanced by increasing propeller disc area immersion. The work is performed in response to Task L-3 of the SOW for the extension/update period. Increasing the immersion is discussed in terms of both mechanically and hydrodynamically controlled sidehull shaping. Four modifications to the sidehull are described.

7.3.1 BACKGROUND -- Increasing the propeller immersion is desirable to improve ventilated propeller efficiency during off-cushion operation, i.e., ship speeds up to about 22 knots. Adequate ventilated propeller performance is predicted for higher ship speeds on cushion using Reference 3 data and the RMI propeller performance program described in Section 7.1. Reference 3 data is based on 50 percent disc immersion. At low ship speeds performance is judged inadequate using the 50 percent immersion data yielding low propeller efficiencies. Section 7.1.4 describes a method to estimate increased submergence/immersion from the 50 percent submergence data, as well as performance sensitivity to different percentages of propeller immersion. To improve propeller efficiency, a reduction of propeller disc loading (thrust/immersed or submerged area) is needed. Usually, with a given power input and ship speed and for a given propeller diameter, the disc loading is reduced by increasing the propeller submergence. Currently, without sidehull shaping, 50 percent of the disc area of the propeller is seen to be immersed, using the hull lines as tested and shown in companion reports. Reference 11, Section 3.6, provides some insight into submergence control for a tandem propeller installation. A discussion follows of some stern sidehull shaping modifications with the potential to give an increase in propeller immersion at ship speeds up to 22 knots.

7.3.2 MODIFICATION 1 - SINGLE RAMP WITH FLAP -- This, sketched in Figure 7.3-1, has a moveable longitudinally arranged low aspect ratio flap attached to the sidehull deadrise area. Aft of the flap attach

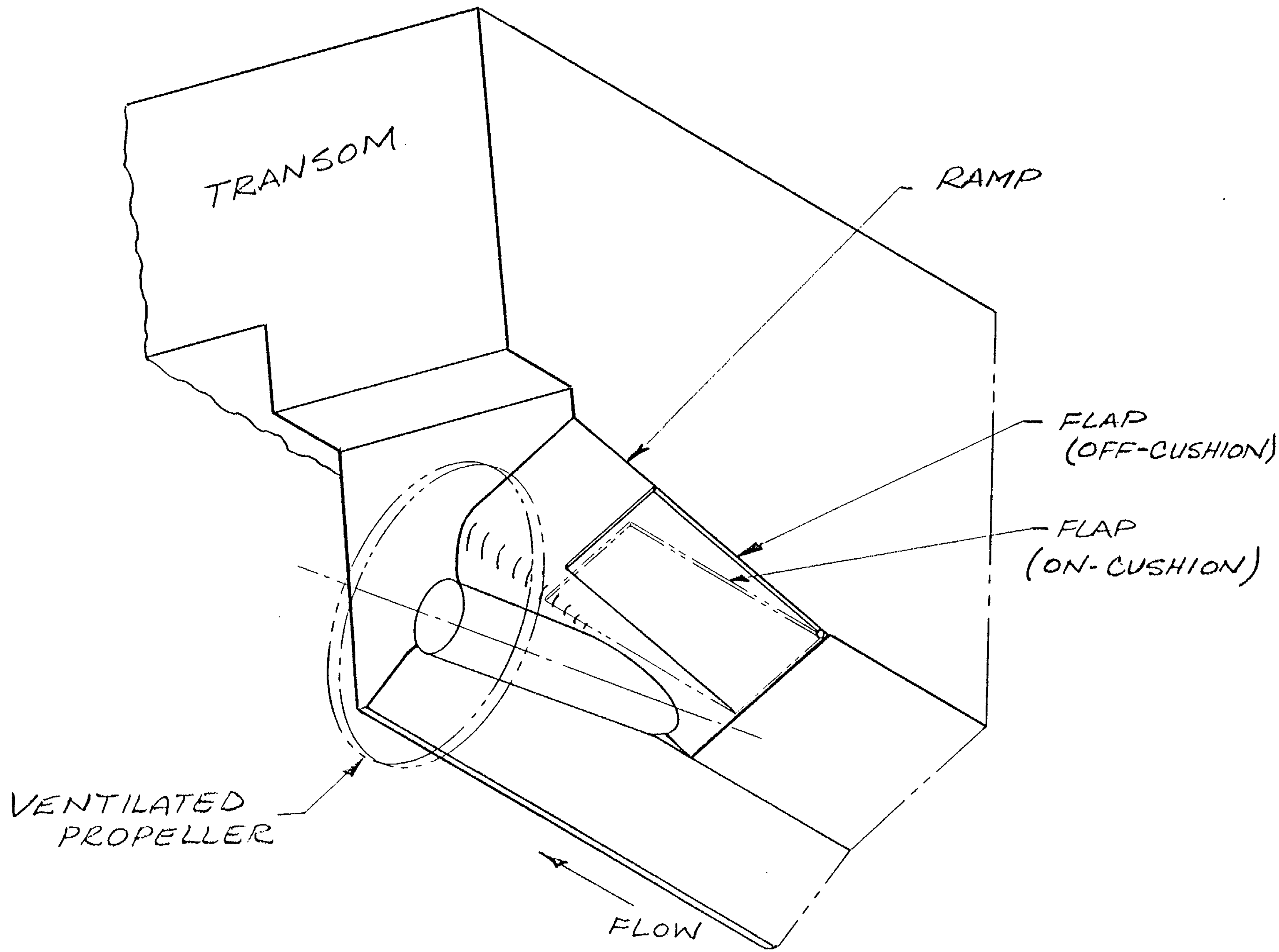


Figure 7.3-1. Modification 1 - Single Ram with Fence

points, the hull is swept upward in a ram shape. At high speed the flap is positioned horizontally, parallel to the keel, and the propeller approach flow is thereby prevented from moving into the swept upward hull region; thus half the disc area receives flow.

At low speeds, the flap is positioned up against the swept up hull structure allowing the approach flow to enter the entire propeller disc. A maximum of 75 percent of the disc is wetted. Controllable positioning of the flap will give proportionate decreases in disc immersion. This method is used on the SES-100B and appears to work satisfactorily. Particular details of this installation are not at hand and are required for a more detailed analysis.

Tests are needed to determine both the thrust deduction, t, which may result from lowered pressure on the upswept surface, and wake fraction, w. Power to propel the ship is proportional to the ratio $(1 - t)/(1 - w)$.

7.3.3 MODIFICATION 2 - FLOW CHANNELS -- As illustrated by Figure 7.3-2, flow channels in the deadrise and sidehull outer wall are provided to allow flow access to the propeller and thereby increase the immersion. The resulting side strut provides structural support to the propeller thrust bearing/transom area. About 80% propeller disc immersion at low speeds is estimated using this method. The flow channel method is similar to that described in Reference 11, Section 3.6. The effectiveness of this modification is difficult to judge and depends like Modification 1 on the obtainable effective immersion and the ratio $(1 - t)/(1 - w)$ noted in 7.3-2.

In operation at high speeds, experience indicates the flow will shear past the cut-outs to give the 50 percent immersion as needed. At low ship speeds the flow will tend to enter the cut-out regions.

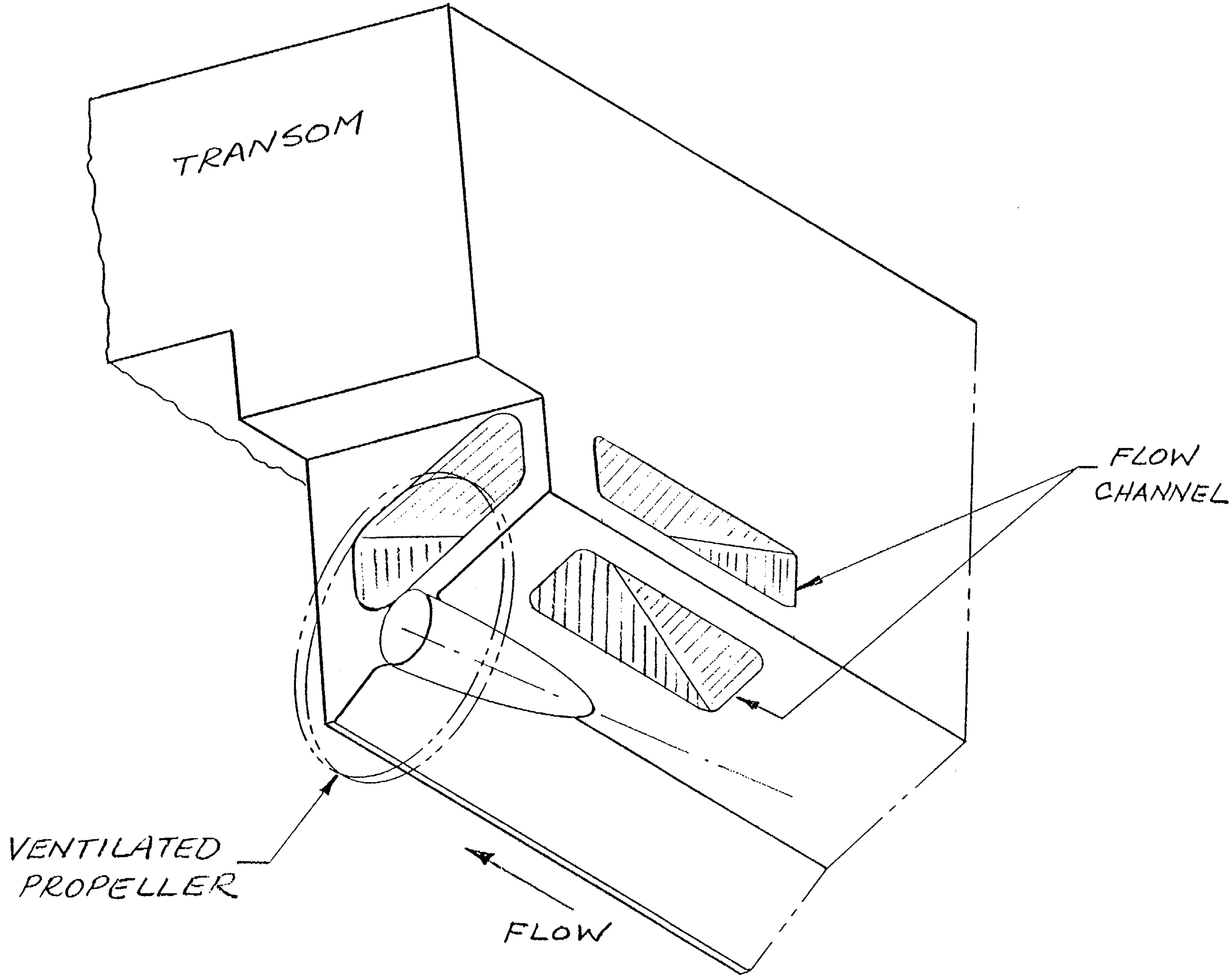


Figure 7.3-2. Modification 2 - Flow Channels

Gradual shaping of the passage walls is needed to avoid flow separation and losses. Propeller suction may permit wall angles larger than the usual practice of $\sim$ 18 degrees; however, the decelerative aspect of cavitating propellers needs to be considered.

No estimates are made at which ship speed the transition from the increased to the nominal 50 percent submergence takes place and the manner of transition. Model testing is necessary to provide this data.

7.3.4 MODIFICATION 3 - DOUBLE RAMS WITH FENCE -- A sketch of Modification 3 appears in Figure 7.3-3. The keel is upswept (cut away), the inner sidewall of the sidehull in way of the aft seal is also in the cut away region. A fence is provided at the inner sidewall to permit seal action and prevent cushion air leakage. At low speed, the flow follows the upswept region and can provide 100 percent propeller immersion. There will be appendage drag on the thrust bearing fairing and a thrust deduction due to suction on the upswept hull caused by propeller suction. These require model test evaluation. The net effect of (1 - t)/1 - w) will again (like Mod. 1 and 2) be an important judgement factor for evaluating the net performance of this concept.

At higher speeds it is anticipated the flow will tend to separate at the start of the ramps. Such separation can be enhanced by a sharp edge or tripping plate or by injection of compressed air. Once separated initially ambient air ventilation is needed to sustain separation and the required 50 percent immersion. With ventilation, the ramp pressures are ambient and (1 - t)/(1 - w) will probably be unity.

The means to initiate and maintain separation needs further study; possibilities include a trip plate or air injection as previously noted, or a thin jet of water. Hydrodynamic tests are required with the scaled cavitation number being very important. Full size tests may be more appropriate and cost effective.

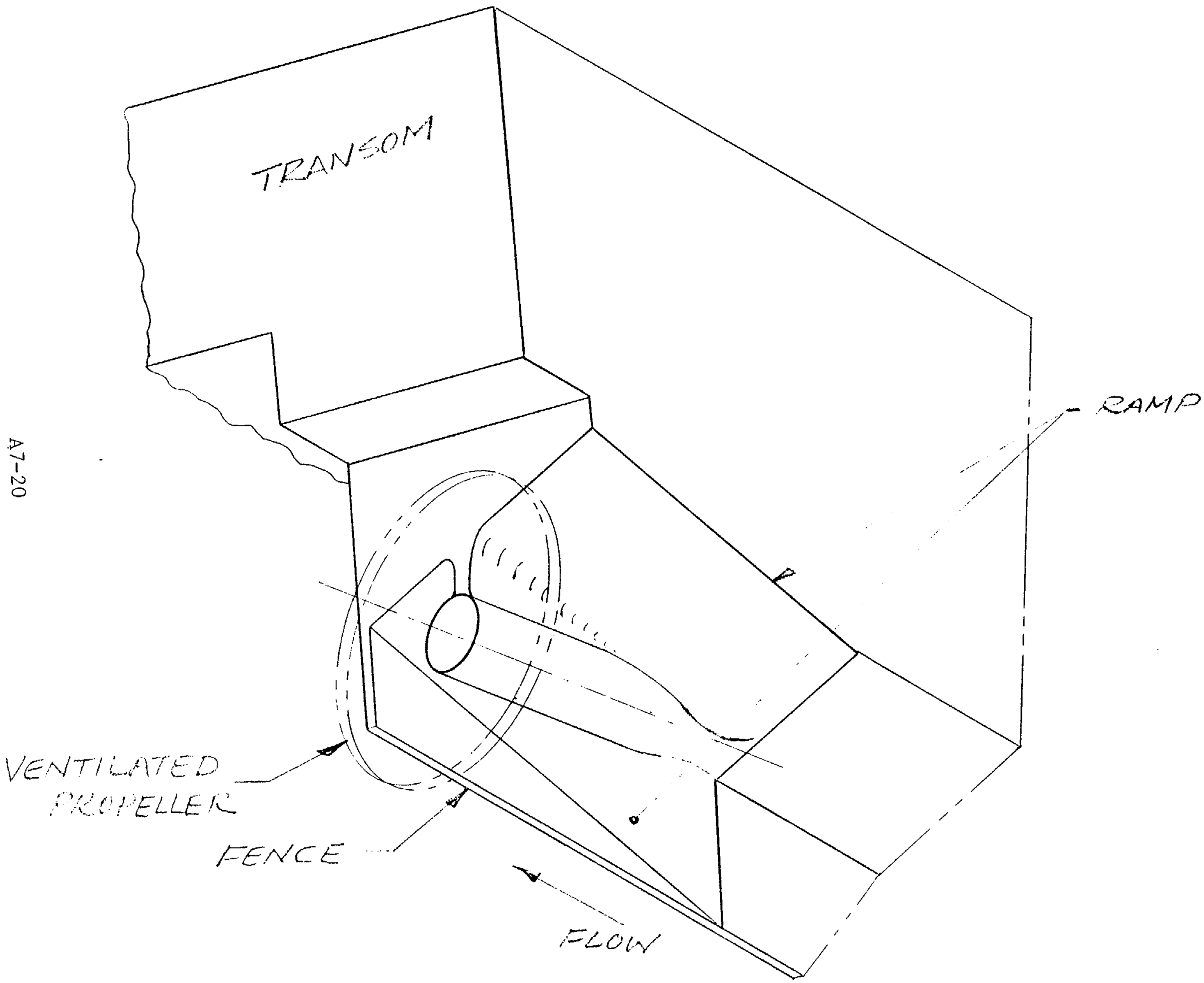


Figure 7.3-3. Modification 3 - Double Ram with Fence

7.3.5 MODIFICATION 4 - SINGLE RAMP -- As shown by Figure 7.3-4, Modification 4 is similar to Modification 1 except the flap is omitted. For high speed the flow is separated as described for Modification 3 and a 50% submergence is anticipated as noted for Modification 1. At low speeds, 75 percent submergence is anticipated.

7.3.6 RESULTS -- The results of these investigations show various methods of maximising propeller immersion at low speed and of maintaining 50 percent immersion at high speeds. No one method is favored although the need to provide this capability with no moving parts, such as ramps (SES 100B) or flow trippers is advocated. Air or water injection to complete flow separation so that ventilation can occur is attractive from the performance aspect but requires additional systems and complication. Separation which relies purely on the hydrodynamic phenomena is favored but data on the subject will need to be determined from model test.

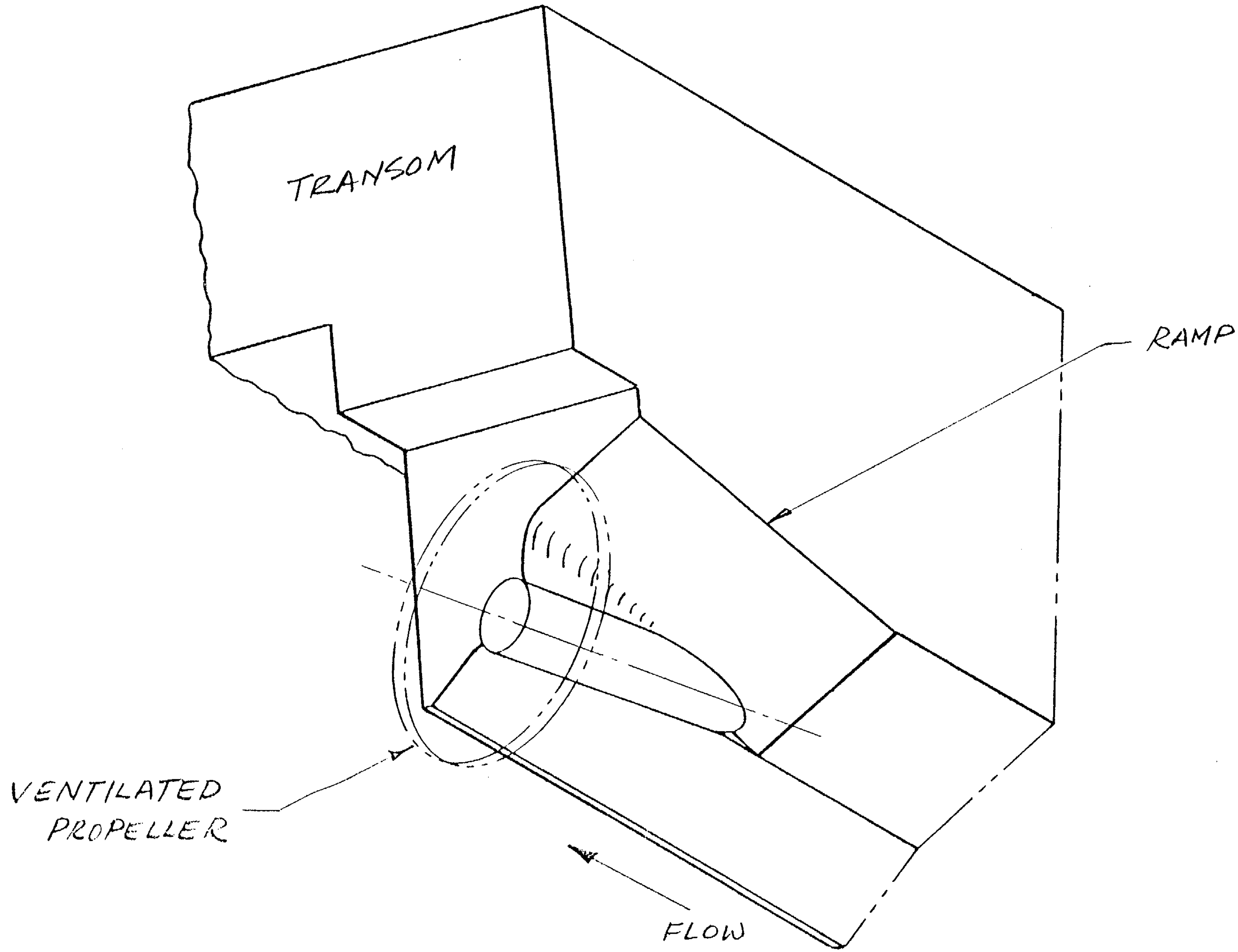


Figure 7.3-4. Modification 4 - Single Ramp

7.4. SELECTED PROPELLER FOR THE 1500-TON SHIP

A 12.5 feet diameter ventilated propeller was selected for the 1500-Ton MDC. This selection was based on installation consideration, propeller rpm, available thrust and efficiency. See Appendix B, Figure 8.3-1 for the arrangement drawing of the 1500-Ton MDC machinery.

From consideration of the sidehull sections both at the transom and at the machinery rooms and machinery sizes, the centerline of the propeller shafting was determined. The distance from this centerline to the inside of the inner sidehull interface with the seal gives the approximate radius of the rpopeller blade tip since much overlap of the blade into the seal area was not desired.

Within the constraints of the sidehull geometry various diameter propellers were investigated to arrive at an optimum size. The rpms of the propeller were to be about 350 at 22,500 SHP and 187 at 3700 SHP to suit a transmission previously sized and described in Appendix B.

The 12.5 feet diameter propeller selected as the result of this approach rotates at 358 rpm at 22,500 SHP and 187 rpm at 3700 SHP. It has four blades. The performance of this propeller is shown in Tables 8.1-1 and 8.1-2 and is adequate for the ship. A drawing of the propeller and machinery arrangement in the ship is shown in Appendix B, Figure 8.3-1.

8 / REFERENCES

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APPENDIX B

PROPULSION MACHINERY CONFIGURATION

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1 / TECHNICAL DESCRIPTION

1.1 GENERAL

Propulsion Machinery Configurations selected for the Medium Displacement Combatant and its variants are Combined Diesel or Gas Turbine driven, CODOG, configurations employing diesel drive for low speed, off-cushion, operation and gas turbine drive for high speed, on-cushion operation. The diesels have double usage. When the ship is off-cushion, they are used for propulsion and when it is on-cushion, they drive the lift-fans. In this way, the most economical off-cushion operating mode is achieved giving a large cruising range to the MDC at a top speed of about 18 knots. For sprint and high speed transit, the gas turbines are used for propulsion and the diesels drive the lift fans to give top speeds in the range of 40-60 knots.

Each configuration is varied by the selection of propeller; the ventilated propeller (semi-submerged, supercavitating) or the submerged propeller (high speed conventional). Table 1.1-1 lists the arrangements investigated and descriptions and performance of the propellers, thrust bearing and shafting are contained in Appendix A.

1.2 MACHINERY CONFIGURATION I

Configuration I employs four gas turbines and two diesels arranged so that two gas turbines and one diesel are in each sidehull driving one propeller through a reduction gearbox. This configuration is applicable to the USCG-940 LT ship. The gas turbines and diesel selected for this study are the Detroit Diesel Allison DDA 570 gas turbine and the Société Alsacienne de Constructions Mécaniques, SACM, 195V12 CZSHR high speed diesel. The diesel is used to provide lift system power for the on-cushion condition as well as hullborne propulsive power. This machinery configuration has two arrangements, one for the ventilated propeller and one for the submerged propeller. The axes of rotation of all the machinery is 8 degrees to the baseline, down aft.

1.2.1 CONFIGURATION I WITH VENTILATED PROPELLER -- The arrangement is shown in Figure 1.2-1. All machinery is located in the sidehulls and is similar port and starboard. The two DDA 570 gas turbines are situated at Frame 251 in an over/under arrangement. The over/under arrangement of the gas turbine engines is necessary to fit the engines in the space available. Aft of Frame 251 the sidehull narrows abruptly in way of the stern seal to provide a vertical inner sidewall for the seal action.

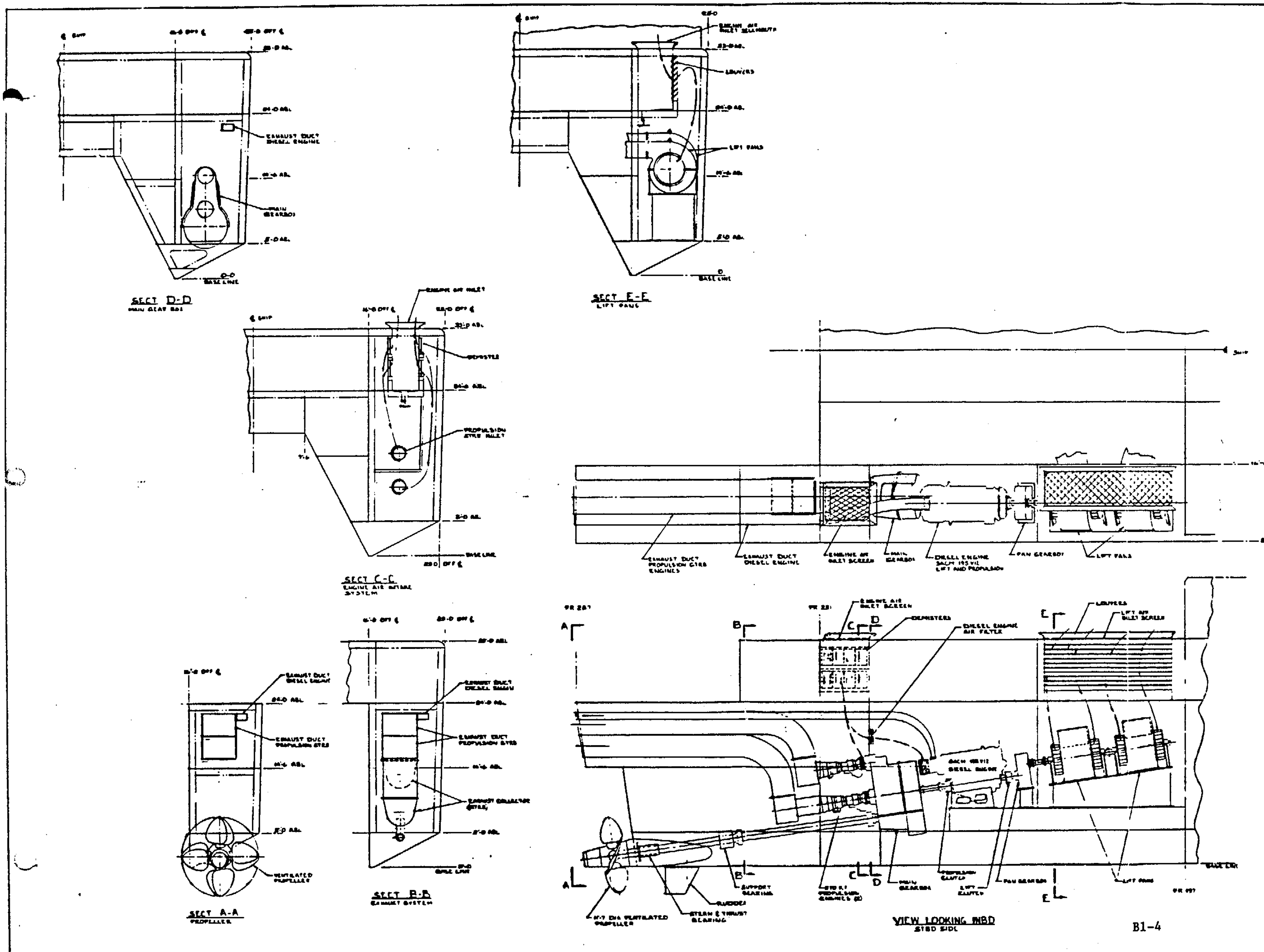
Access is maximized by spacing the engines far apart vertically to make best use of the space but commensurate with provision of space for the gas turbine exhuasts. The gas turbines drive through the inlet ends of the engines and are connected to the reduction gearbox with disc type flexible couplings. Each gas turbine is face mounted off its compressor from the reduction gearbox and supported at its turbine section.

Table 1.1-1. Propulsion Machinery Arrangements

WEIGHT, LT	MACHINERY CONFIG.	GAS TURBINE ENGINE (1)	DIESEL ENGINE	PROPELLER TYPE DIA FT
940	I	2 X DDA 570	1 X 195V12 CZSHR	VENTILATED 11.60
940	I	2 X DDA 570	1 X 195V12 CZSHR	SUBMERGED 9.64
1200	II	1 X LM2500	1 X 195V20 RVR	VENTILATED 12.49
1200	II	1 X LM2500	1 X 195V20 RVR	SUBMERGED 11.61
1800	II	1 X LM2500	1 X 195V20 RVR	VENTILATED 14.64
1800	II (2)	1 X LM2500	1 X 195V20 RVR	SUBMERGED 11.61

(1) UPRATED ENGINES WERE CONSIDERED BUT ARE NOT SHOWN

(2) WITH BEACHING SKEG SHOWN



Appendix B

Figure 1.2-1.

Configuration I

RMI, Inc.
MACHINERY ARRGT - USCG / 340 LT VENTILATED PROPELLER
A3-245001

The air inlet for the gas turbine is radial. Air is drawn from the deck opening through a screen and demisters located vertically above the engines. Inlet air heating for anti-icing is incorporated in the air inlet screens and uses spray bars to distribute a portion of the engine exhaust gas into the inlet air stream. The demister banks have a by-pass door feature for emergency use to allow the engine to aspirate air directly if the demisters are blocked for any reason. The air inlet also serves as the removal route for the gas turbine engines.

The exhaust gas from the turbines are routed up and aft from the exhaust collectors mounted on the engine. Square section exhaust gas uptakes are shown. These are welded/mechanically fastened structures using double wall construction of heat resistant Inconel 625 alloy. The inner wall of the exhaust duct is perforated to attenuate turbine noise below the required level.

The reduction gear, described in detail in Section 2.1, is mounted to the ship structure on a rail mount. Input flanges are provided for the two gas turbines on the aft face, and on the forward face for the diesel. The reduction gearbox is the main mount of the gas turbines. The output flance for the propeller shaft is located below the lower DDA 570 gas turbine. It connects to the propeller shafting through a Zurn gear type coupling. Technical descriptions and performance for the ventilated propeller, thrust bearing and shafting are given in Appendix A.

The diesel engine is located forward of the reduction gearbox and is connected to the reduction gear through an air clutch and a flexible coupling. The diesel draws its combustion air from the diesel compartment; the air enters the compartment through a filter located on the wall of main air inlet plenum. The diesel is mounted on its own foundation aligned with the lift fan gearbox (for details of the lift machinery see Appendix C). The diesel exhaust is routed up and aft to the transom by an insulated duct.

1.2.2 CONFIGURATION I WITH SUBMERGED PROPELLER -- This arrangement is similar to that described for the ventilated propeller except for the position of the diesel and the reduction gearbox. It is shown in Figure 1.2-2. The submerged propeller is located beneath the sidehull to give adequate submergence for the propeller tip. To minimize the depth of the propeller installation, the aft portion of the hull is raised gradually to the transom. A propeller bossing is provided forward of the propeller to house the shaft and bearings. The shaft passes up through the hull into a reduction gearbox similar to that used for the ventilated propeller. To allow a shaft angle of 8 degrees, the reduction gears were moved forward and the lift machinery slightly rearranged. It is felt that with further design study,rearrangement of the two gas turbines in the horizontal plane is possible.

1.3 MACHINERY CONFIGURATION II

Configuration II employs two gas turbines and two diesels arranged one in each sidehull, driving one propeller through a reduction gearbox. This configuration is applicable to the MDC-1200 LT ship and the Logistics 1800 LT ship. The gas turbine is a General Electric LM2500 and the diesel is an SACM 195V20 RVR high speed diesel. The diesel is used to provide lift system power for the on-cushion condition and propulsive power in the hullborne mode. This machinery configuration is used in four arrangements, a ventilated propeller and a submerged propeller each for the MDC/1200 LT ship and the Logistics/1800 LT ship. The machinery arrangements for the submerged propellers differ in some details from the machinery arrangement for the ventilated propeller mainly in the area of the reduction gearbox and shafting.

B1-7

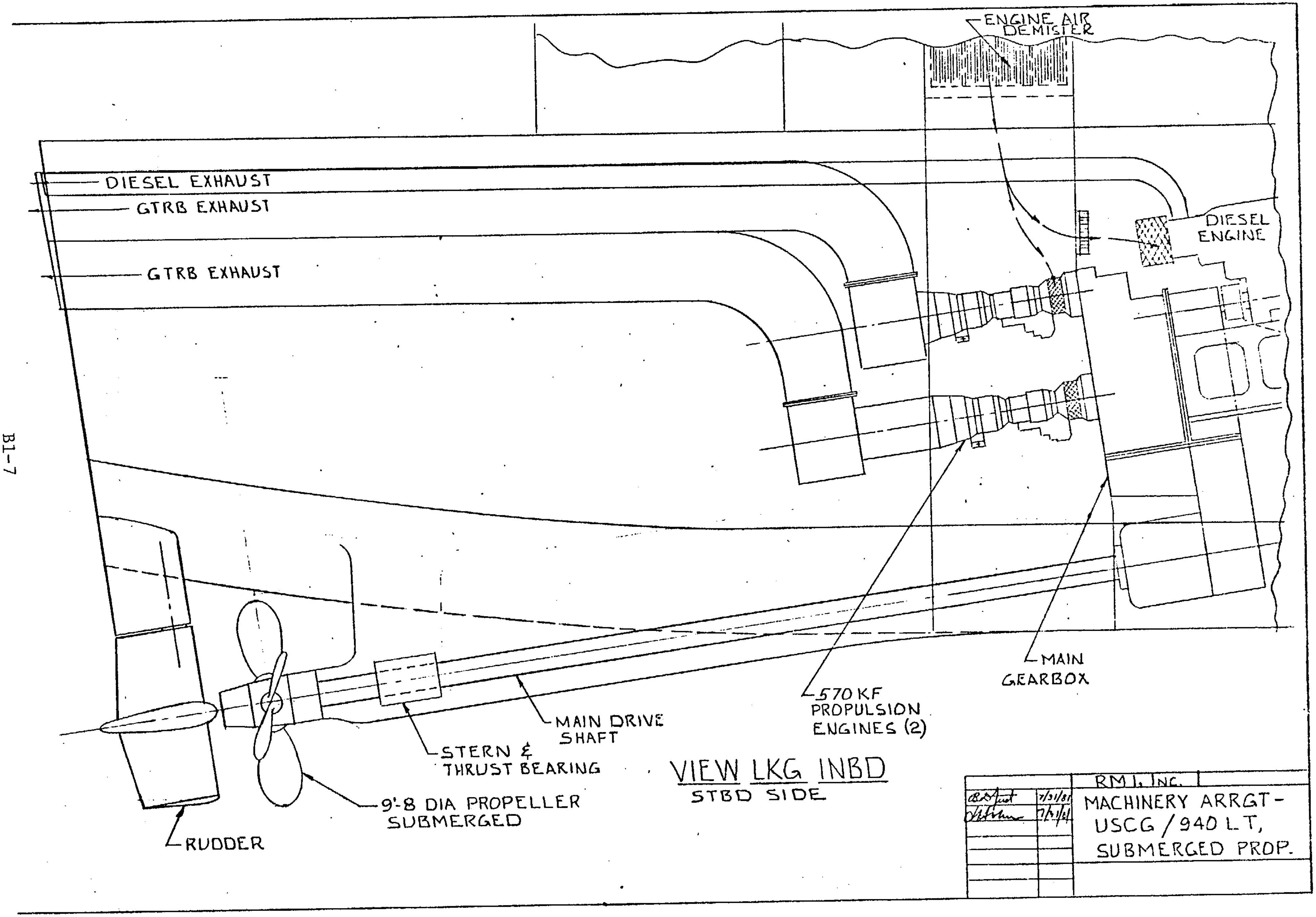


Figure 1.2-2

All of the above arrangements have the machinery at an angle of 8° to the baseline, down aft.

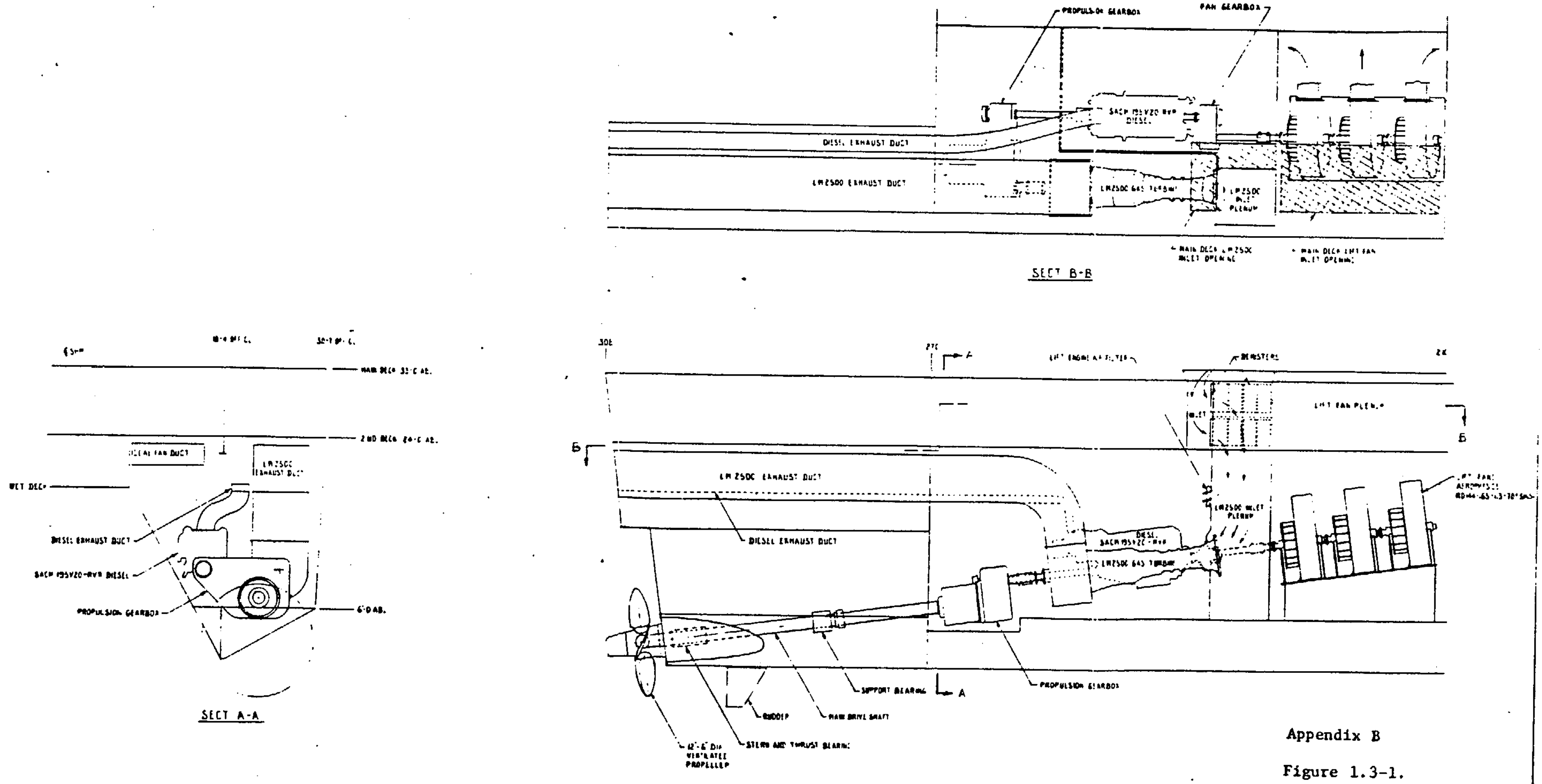
1.3.1 CONFIGURATION II WITH VENTILATED PROPELLER -- As with Configuration I all machinery is located in the sidehulls and is similar port and starboard. The LM2500 gas turbine is mounted outboard of the diesel, forward of the reduction gearbox and is supported on its own mount system. The gas turbine exhaust is taken up and aft in the MDC/1200 LT arrangement and vertically upwards in the Logistics/1800 LT arrangement. Construction of the uptake is the double walled sound attenuating sheet metal. Air inlets for both arrangements are similar. Air enters at the upper deck opening and is aspirated via demisters from an inlet air plenum. The inlet has anti-ice capability, using reingested gas turbine exhaust gases, and the demisters have a blow-in door feature to allow emergency running of the gas turbine if the demister is blocked for any reason.

Power from the LM2500 gas turbine is taken into the reduction gearbox by a Synchro Self Shifting (SSS) clutch. Support for the gas turbine output shaft is provided by a bearing module located between the engine and the SSS clutch. The reduction gearbox, described in detail in Section 2.1, is mounted to the ship structure on a rail mount. It has input flanges for the gas turbine and the diesel on its forward side. On the aft lower side, is the output flange which connects to the propeller shafting by a Zurn gear coupling. Technical description and performance for the ventilated propeller, thrust bearing and shafting are given in Appendix A.

The diesel engine is located forward of the reduction gearbox, inboard of and parallel to the gas turbine. It is connected to the reduction gear through an air clutch and a flexible coupling. The diesel draws its inlet air from the diesel compartment; the air enters the compartment through a filter located on the wall of the main air inlet plenum. The diesel is mounted on its own foundation aligned with the lift fan gearbox (for details of the lift machinery see Appendix C). The diesel exhaust is routed alongside the gas turbines by an insulated duct. The Configuration II machinery arrangement with ventilated propeller for the MDC and logistics ship applications is shown in Figures 1.3-1 and 1.3-2 respectively.

1.3.2 CONFIGURATION II WITH SUBMERGED PROPELLER -- These arrangements are similar to those described for the ventilated propellers except that for the power transmission provisions. In these cases (for the 1200 LT and 1800 LT ships), the submerged propeller is located beneath the side hull to give adequate submergence for the propeller tip. To minimize the depth of the propeller installation the aft portion of the hull is raised gradually to the transom. A skeg structure is provided to support and house the propeller shaft and bearings. In the case of the Logistics/1800 LT ship, the skeg structure is extended to below the propeller tip to provide protection to the propeller in the case of beaching, for unloading and loading purposes. Without a beaching requirement, the skeg can be reduced to a fairing around the propeller shaft and bearings. This machinery installation for the 1200 ton MDC is shown in Figure 1.3-3. The installation for the 1800 ton Intra-Theater Logistics Ship concept is shown in Figure 1.3-4.

The arrangement drawings show more complex transmission systems for the submerged propeller in order to size the gearing within the constraints of the sidehull and yet not put the reduction gearing too far forward beneath the lift fans. In the arrangements an intermediate shaft connects a drop box to the reduction gear. It is felt that with further design study, this additional drop box and shaft could be eliminated.



Appendix B

Figure 1.3-1.

Configuration II

RMI, Inc.

MACHINERY ARRGT -
MDC/1200 LT.
VENTILATED PROPELLER

Appendix B

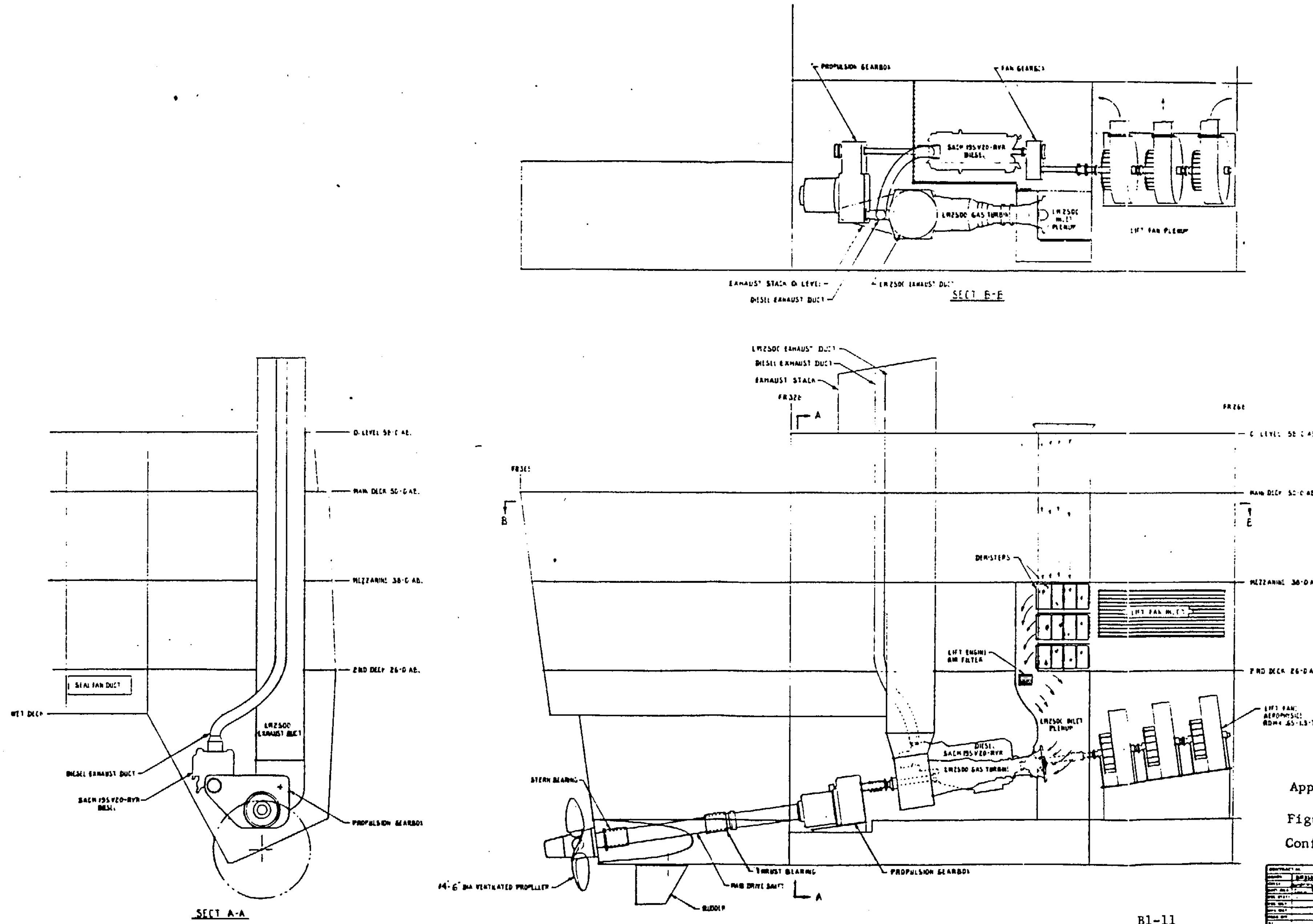
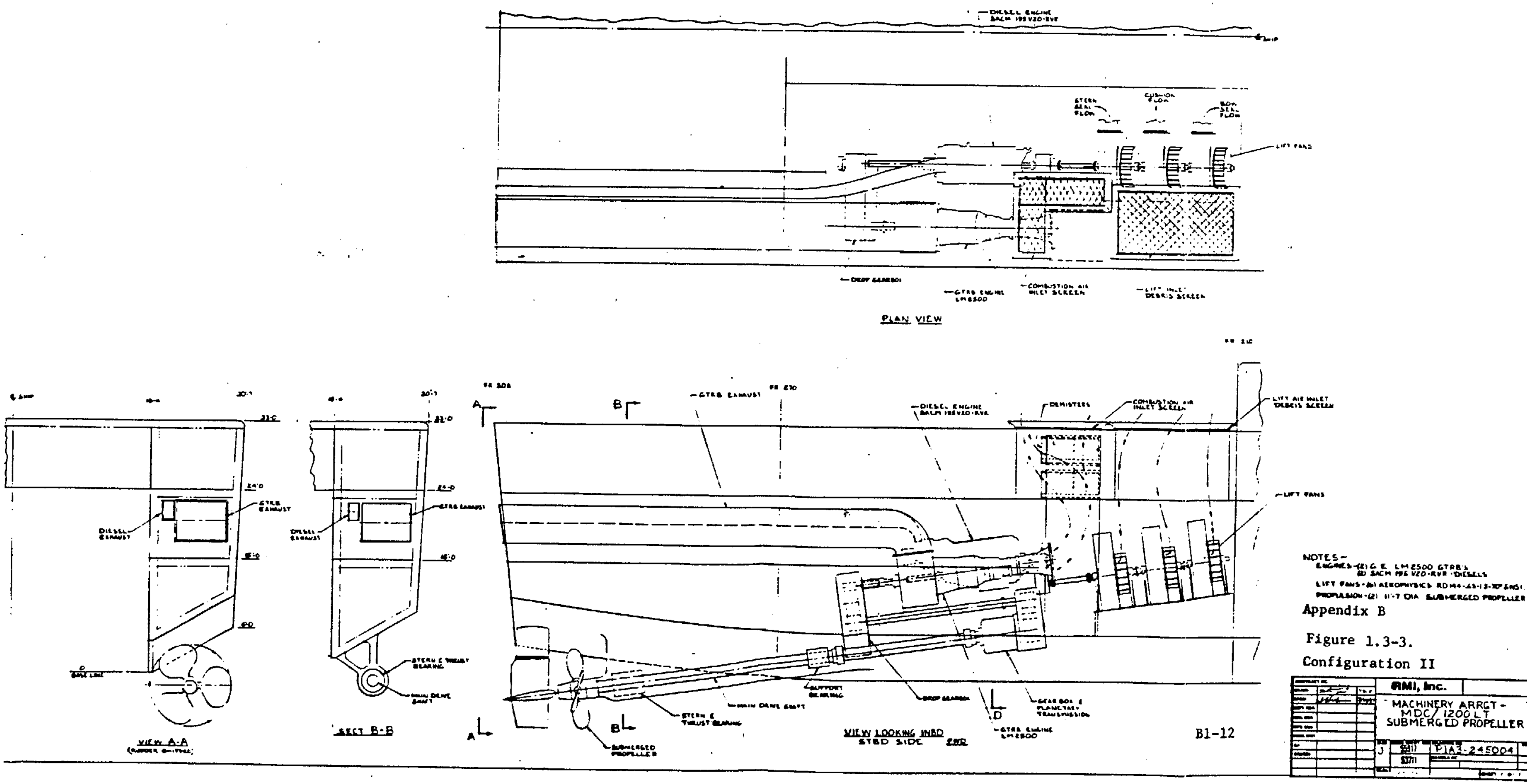


Figure 1.3-2.
Configuration II

RMI, Inc.
MACHINERY ARRGT - LOGISTICS/1800 LT, VENTILATED PROPELLER

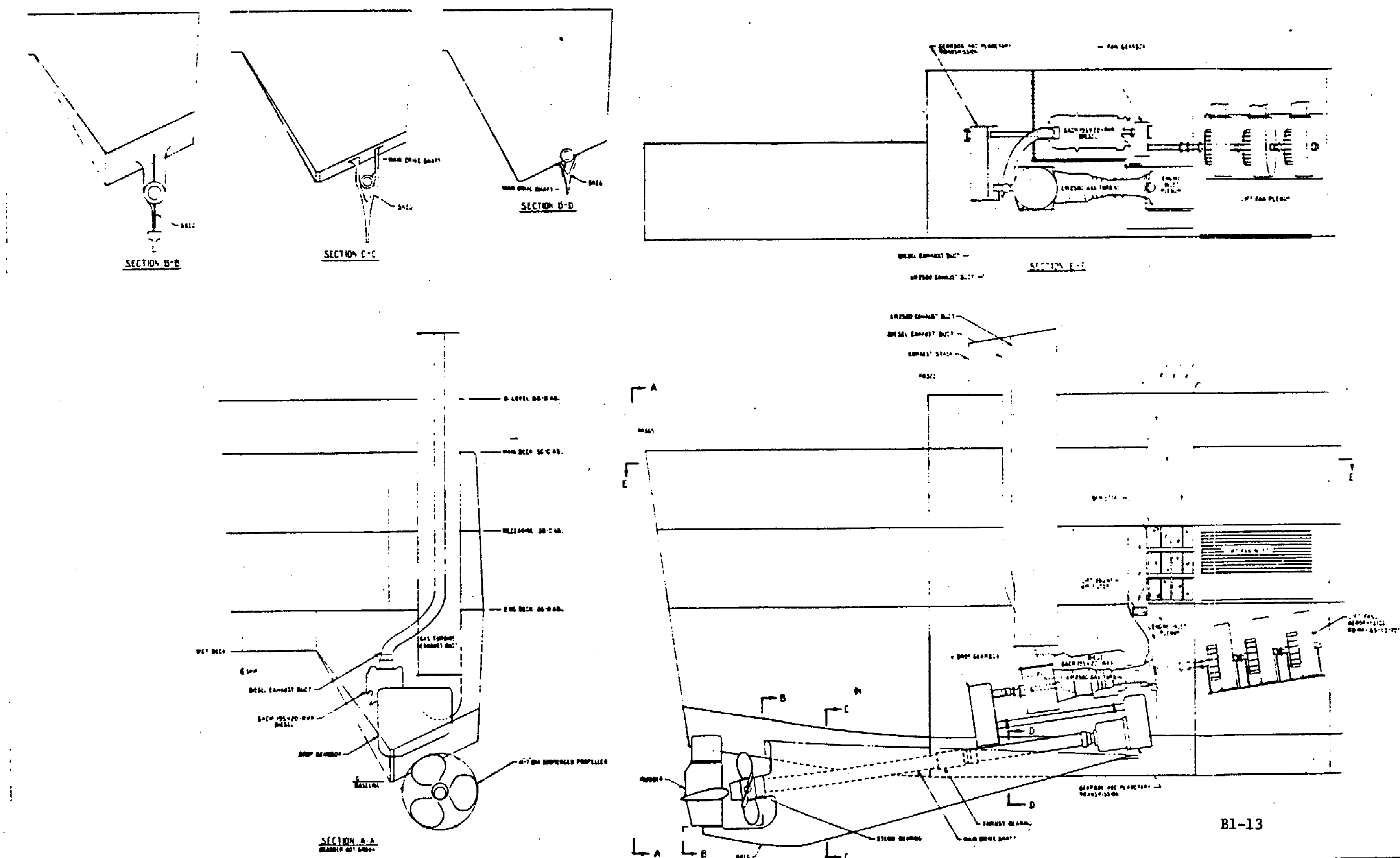


Appendix B

Figure 1.3-3.

Configuration II

B1-12



Appendix B

Figure 1.3-4.

Configuration II

RMI, Inc.

MACHINERY ARRGT - LOGISTICS/1800 LT SUBMERGED PROPELLER

2 / REDUCTION GEARING

2.1 MACHINERY CONFIGURATION I

The power system installed in each sidehull of Configuration I comprises two DDA 570 KF gas turbines that are geared to the propeller shaft for on-cushion operation and a diesel engine (SACM 195V12CZSHR) that drives, through suitable gearing, either the lift fans for on-cushion operation or the propeller for off-cushion operation. When the diesel drives the propeller the gas turbines and their gear train are decoupled from the propeller drive.

The gas turbines, the diesel engine and the reduction gearing to the propeller are installed in a narrow sidehull that precludes side-by-side placement of the turbines and the diesel. The drive shafts for the propeller and the lift fan lie approximately midway between the walls of the sidehull to obtain the proper location for the propeller aft of the gearbox and for the large plenums for the lift fans forward of the diesel engine. Both of these requirements are met by placing one gas turbine directly above the other on the aft side of the gearbox and the diesel engine on the forward side of the gearbox; the centerlines for all three engines lie approximately on the vertical center of the sidehull. The centerline of the diesel is offset from the centerline of the compartment by the distance required to accommodate a parallel-shaft speed increaser that is to be installed between the diesel engine and the lift fans situated forward of this engine.

2.1.1 GEAR DESIGN -- The propeller reduction gearing is divided into two sections comprising a combining box for the two gas turbine engines and a drop box to the propeller. The diesel engine shaft is input to the drop box through an air clutch that is external to the gearbox. Parallel shaft gearing is used since there are adequate center distances between the two gas turbines and in the drop box to the propeller to develop the required speed reductions. Parallel-shaft gearing makes optimum use of the gearing required to connect the gas turbine shafts to a common output shaft while at the same time reducing speed. This shaft is then coupled through the drop box to the propeller shaft obtaining the final stage of speed reduction for the propeller. The diesel engine input to the propeller is through an idler in the drop box. An additional idler is installed in one of the two drop boxes in the ship to obtain opposite rotations for the propellers in the two sidehulls to accommodate one direction rotation for both the gas turbine and diesel engines.

In the off-cushion mode, when the diesel engine drives the propeller, the gas turbine and the gearing in the combining section of the gearbox is decoupled from the propeller drop box by a SSS clutch that is mounted on the diesel side of the drop box housing. In the on-cushion mode, when the gas turbines drive the propeller, the diesel engine is decoupled from the entire propeller reduction gear by an air clutch external to the gearbox; in this mode either or both gas turbines can drive the propeller. When one gas turbine is shut down, an overrunning clutch on the pinion shaft of the idle engine prevents the engine that is running from windmilling (back driving) the power turbine section of the idle engine.

The transmission driveline is designed such that all propeller loads will be accommodated by bearings external to the gearbox. Adequate access and space are provided on the forward side of the bull gear shaft for installation and inspection of the mechanism and controls required for the variable pitch propeller. The gearbox is foot mounted in one plane and incorporates an oil sump of adequate capacity that is integral with the gearbox housing. The gas turbines are flange-mounted to the gear housing using split, piloted torque spools for ready access to the flexible couplings between the engine and gear input shafts.

All of the above requirements were submitted to the Cincinnati Gear Company with the request that they integrate these requirements into a feasible design. In response to this request, which was formalized in a Statement of Work under Subcontract 3K02276, the Cincinnati Gear Company prepared the preliminary design shown in schematic form in Figure 2.1-1. The outline drawing for this design is shown in Figure 2.1-2. A summary of data relating to gear design dimensions and stresses is given in Table 2.1-1. AGMA standards and derating factors were used in sizing all gearing. Gear efficiencies and lubrication requirements are summarized in Table 2.1-2.

All gearing is single helical and is surface hardened and ground except for the bull gear on the propeller shaft which is through-hardened and machined to final dimensions. Each high-speed stage utilizes thrust collars that bear on the gear rims, to cancel out the axial thrust in the helical mesh. The selection of thrust collars is aided by use of medium mineral oil for lubricant and the use of journal bearings. These collars also eliminate highly loaded thrust bearings, reduce the number of thrust bearings required, and permit the use of higher helix angles in single helical gears than are feasible with conventional thrust bearings.

A comprehensive list of the features designed into the propeller reduction gearing for Configuration I by Cincinnati Gear is presented in Section 2.3.

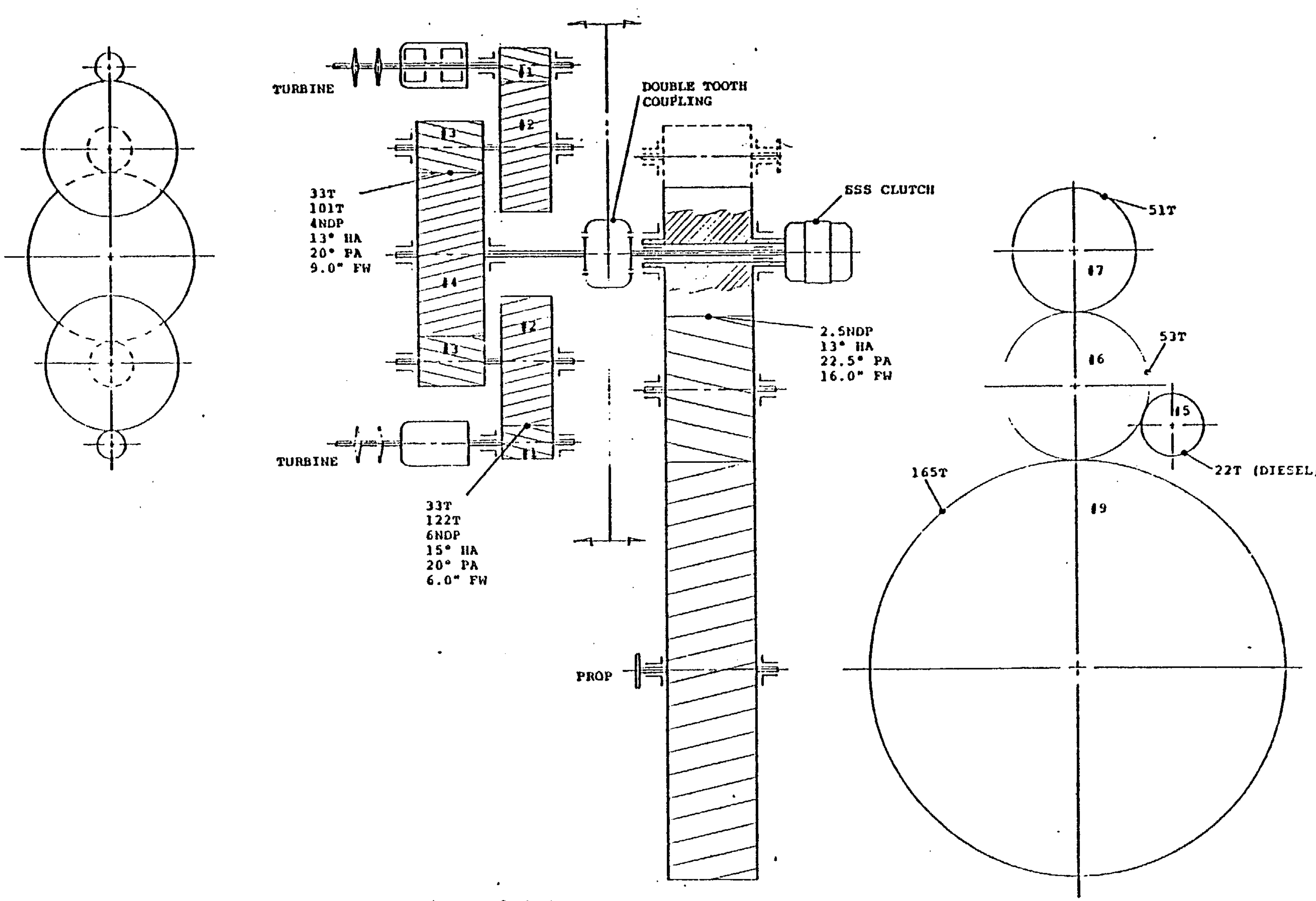


Figure 2.1-1. Gearbox Schematics Configuration 1

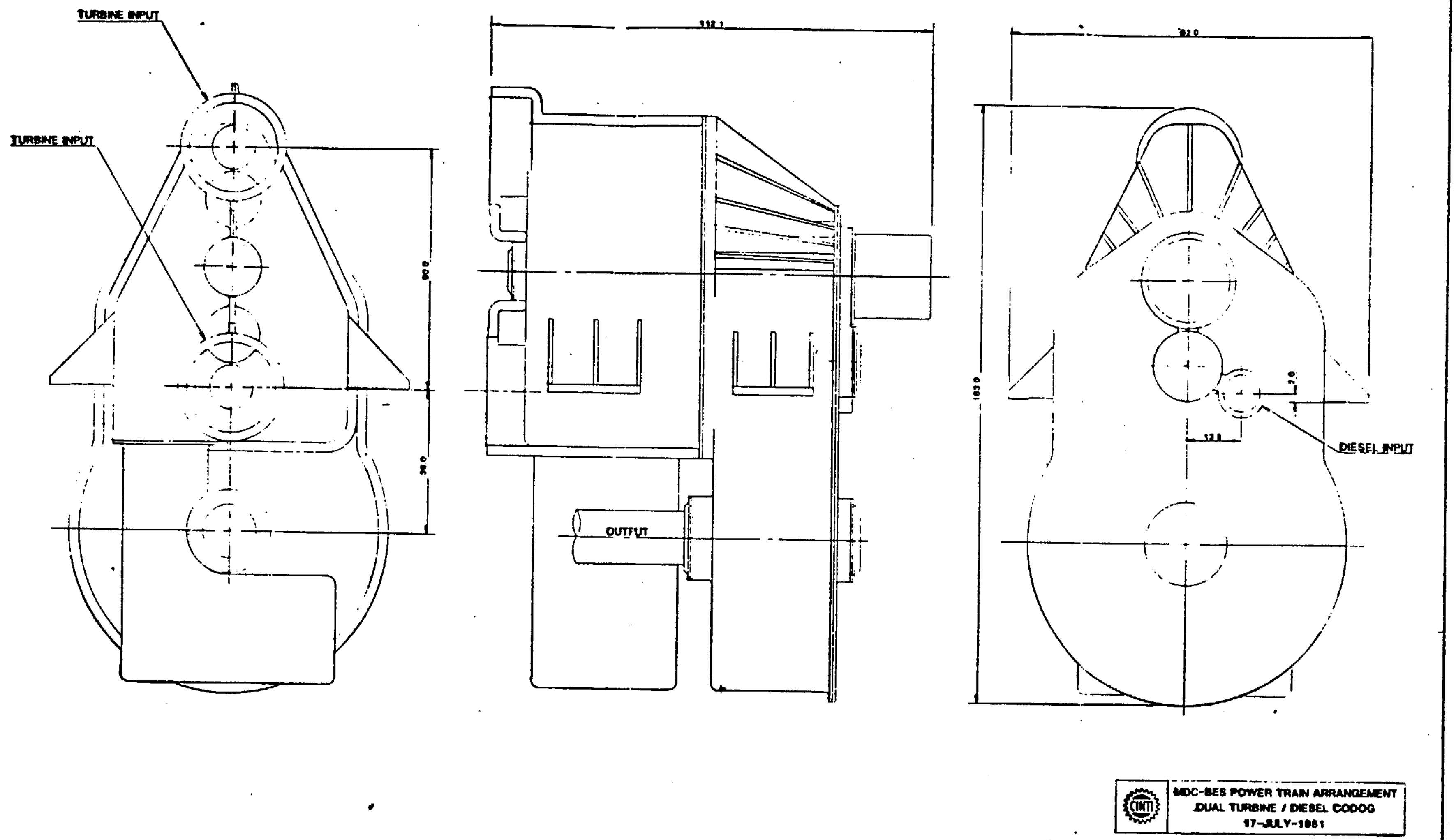


Figure 2.1-2. Configuration I

Table 2.1-1. Gear Design Data - Configuration I - Ref. Figure 2.1-1.

MESH	1-2		3-4		6-7		6-9	
UNITS (1)	PINION	GEAR	PINION	GEAR	PINION	GEAR	PINION	GEAR
Horsepower	6300		6300		12600		12600	
RPM	11500	3110	3110	1050	1050	1010	1010	324
Pitch Dia.	5.694	21.051	8.467	25.914	20.937	21.758	21.758	67.736
Face Width	6.000		9.000		16.000		16.000	
Normal DP	6		4		2.5		2.5	
No. Teeth	33	122	33	101	51	53	53	165
Pressure Angle	20.00		20.00		22.50		22.50	
Helix Angle	20.64		13.00		13.00		13.00	
Contact Ratio	1.68		1.70		1.58		1.63	
Pitch Line Vel	17142		6894		5316		5320	
K Factor	450		525		458		297	
Allow. Hertz Stress	151000		151000		173000		135000	
Hertz Stress, Max.	81276	115469	88972	124123	131864	131864	105276	105276
Durability HP	9342		7791		21687		20719	
Allow. Bend.Str.	50000		50000		55000		38500	47500
Bending Stress	37979	34315	43457	39470	38203	38071	37454	34876
Bending HP	8294	9179	7248	7981	18139	12741	12952	17161
AGMA Quality No.	11		11		11		11	

(1) • All dimensions are in inches • Stresses are in PSI

• Pitch-line vel. is in ft/min • Angles are in degrees

Table 2.1-2. Gearing Efficiencies, Configuration I

PERFORMANCE SUMMARY

(DUAL TURBINE/DIESEL CODOG)

OPERATIONAL MODE	INPUT POWER (HP)	POWER LOSS (HP)	EFFICIENCY	OIL REQUIREMENT (GPM)
TWIN TURBINE	12600	421	.9666	202
ENGINE OUT	9000	366	.9593	183
DIESEL	2200	77	.9650	54

1) DTE MEDIUM OIL

2) 120° F. OIL INLET TEMPERATURE

3) DESIGN W/O EXTRA IDLERS FOR ROTATION CHANGE

2.2 MACHINERY CONFIGURATION II

The power system installed in each sidehull of Configuration II comprises one General Electric LM2500 gas turbine engine that is geared to the propeller shaft for on-cushion operation and a diesel engine (SACM 195V20 RVR) that drives either the lift fans for on-cushion operation or the propeller for off-cushion operation; both through reduction gearing. When the diesel drives the propeller the gas turbine output shaft is decoupled from the propeller drive.

The design of the sidehull for Configuration II permits side-by-side placement of the gas turbine and the diesel engine, and results in a compact installation of the two engines and the propeller reduction gearing, with ample space between and around both engines for installation, inspection, and maintenance activities.
The positions of the centerlines of the propeller shaft and of the two engines are resolved within the gearbox by means of parallel-shaft gearsets that serve as vertical, lateral and inclined drop boxes while reducing the engine speeds to those required by the output planetary gearing coupled to the propeller shaft. Since both the power and the speed of the two engines are quite different the engines require separate reduction gearsets. In addition there is the requirement that the port and starboard propellers rotate in opposite direction. This requirement calls for additional idler gears in one of the reduction gearboxes, since the gas turbine is available with only one rotation and it is desirable from a logistics standpoint that the port and starboard diesels have the same rotation. All of the above considerations were implemented in the gearbox design developed as a result of this study.

2.2.1 GEAR DESIGN -- The propeller reduction gearing for Configuration II is divided into three sections that are integrated into a single gearbox assembly. There are two sets of parallel-shaft gears, one for each engine, that drive common output gears on the same shaft as the planetary gearset. Each set of parallel-shaft gears serves as the drop-box gearing from its driving engine to the propeller shaft centerline while reducing engine speed to that required at the planetary input sun gear.

In order to locate the propeller shaft as low as possible in the ship's sidehull it is essential that the final stage of propeller gearing, particularly the bull gear or its equivalent, be of a relatively small diameter. For the particular power and propeller speed involved in Configuration II this objective is achieved by utilizing a planetary gearset as the final stage of speed reduction from engine to propeller speed, for both the diesel and the gas turbine. By selecting a gear ratio of the order of 4:1, it is possible to install five planet gears in a planet gear carrier and distribute the large torque load among five gears of relatively small diameter and modest face width. This results in a housing diameter for the planetary gearset that is considerably smaller than that of a bull gear in which all of the torque load would have to be transmitted through a single mesh. In this arrangement the first stage of speed reduction, in which the torques are low and the speeds are high, is taken in single mesh, parallel-shaft gears, while the final stage of speed reduction, involving high torques and low speeds, is taken in a multiple-path planetary gearset of relatively small diameter and low weight. This results in lower weight for the entire gearbox installation as well as in satisfying the primary objective of propeller-shaft location.

With the diesel engine driving the propeller, the gas turbine shaft is decoupled from the propeller reduction gearbox by a SSS overrunning clutch mounted external to the gearbox just aft of the outboard bearing module that supports the gas turbine engine output shaft. When the gas turbine engine is driving the propeller, the diesel engine is decoupled from the reduction gear by a dry-plate, air-actuated clutch mounted outside the gearbox housing on extensions of the diesel pinion and shaft.

The transmission shafting is designed such that all propeller loads will be accommodated by bearings external to the gearbox, thereby avoiding any tendency for the propeller loads to distort the gear housing or the shafts and compromise gear tooth action. Adequate access and space are provided on the forward side of the drop box, opposite the planetary, for installation and inspection of the mechanisms and controls required for the variable pitch propeller. The gearbox is foot-mounted in one plane and incorporates an oil sump of adequate capacity that is integral with the gearbox housing. The engine shafts of the diesel and the gas turbines are flexibly coupled to the gearbox shafts.

All of the above requirements were submitted to the Cincinnati Gear Company with the request that they integrate these requirements into a feasible design. In response to this request, which was formalized in a Statement of Work under Subcontract 3K02276 the Cincinnati Gear Company prepared the schematic design shown in schematic form in Figure 2.2-1. The outline drawing for this design is shown in Figure 2.2-2. A summary of data relating to the gear dimensions and stresses for the parallel-shaft, drop box gearing as given in Table 2.2-1; technical data for the planetary gearset is given in Table 2.2-2. AGMA standards and derating factors were used in sizing all gearing. Gear efficiencies and lube requirements are summarized in Table 2.2-3.

All parallel-shaft gearing is single-helical, and is surface hardened and ground. Each high speed stage utilizes thrust collars that bear on the gear rims, to cancel out the axial thrust in the helical mesh; these collars eliminate highly loaded thrust bearings, reduce the number of thrust bearings required, and permit the use of higher helix angle in single helical gears than are feasible with conventional thrust bearings.

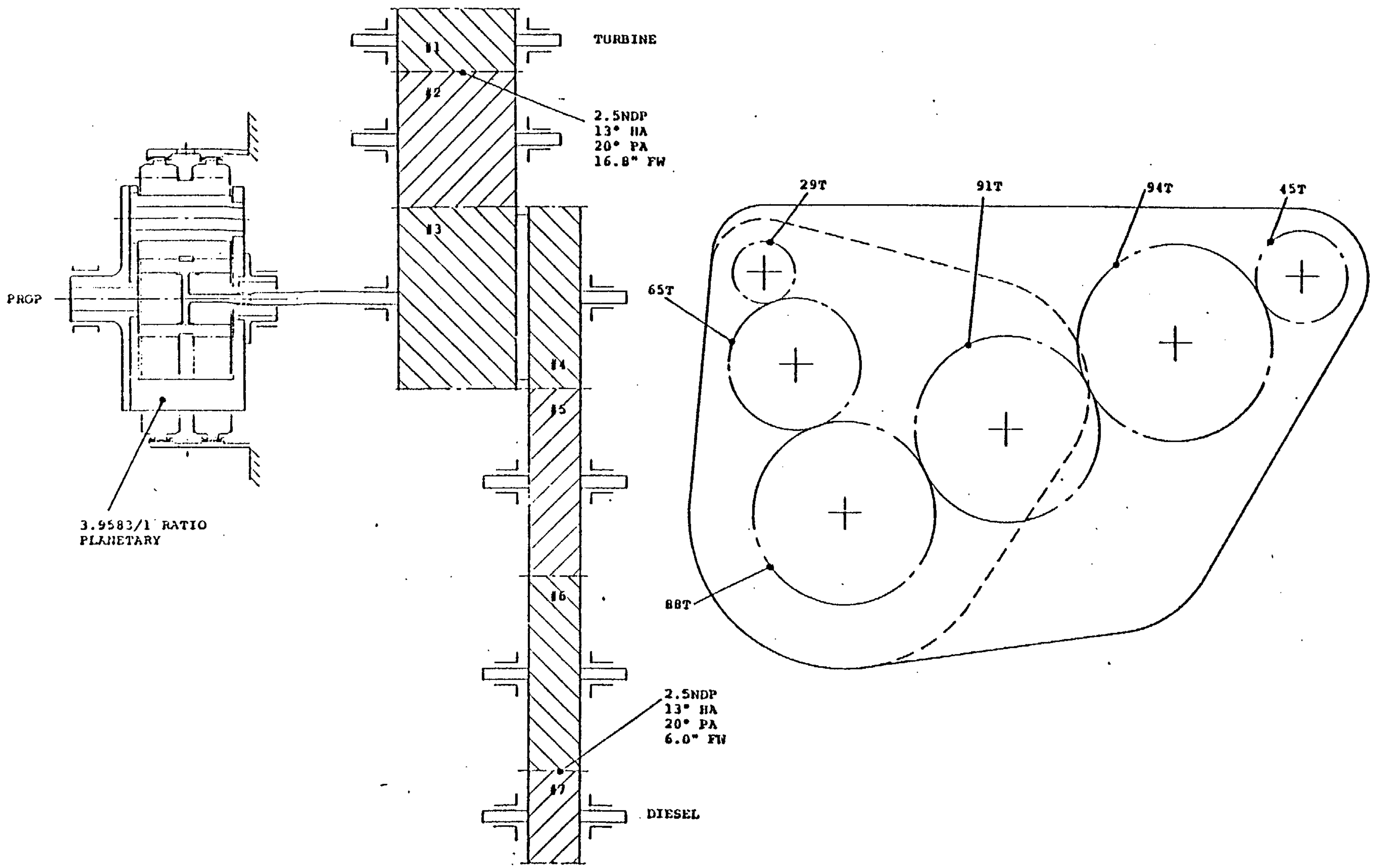
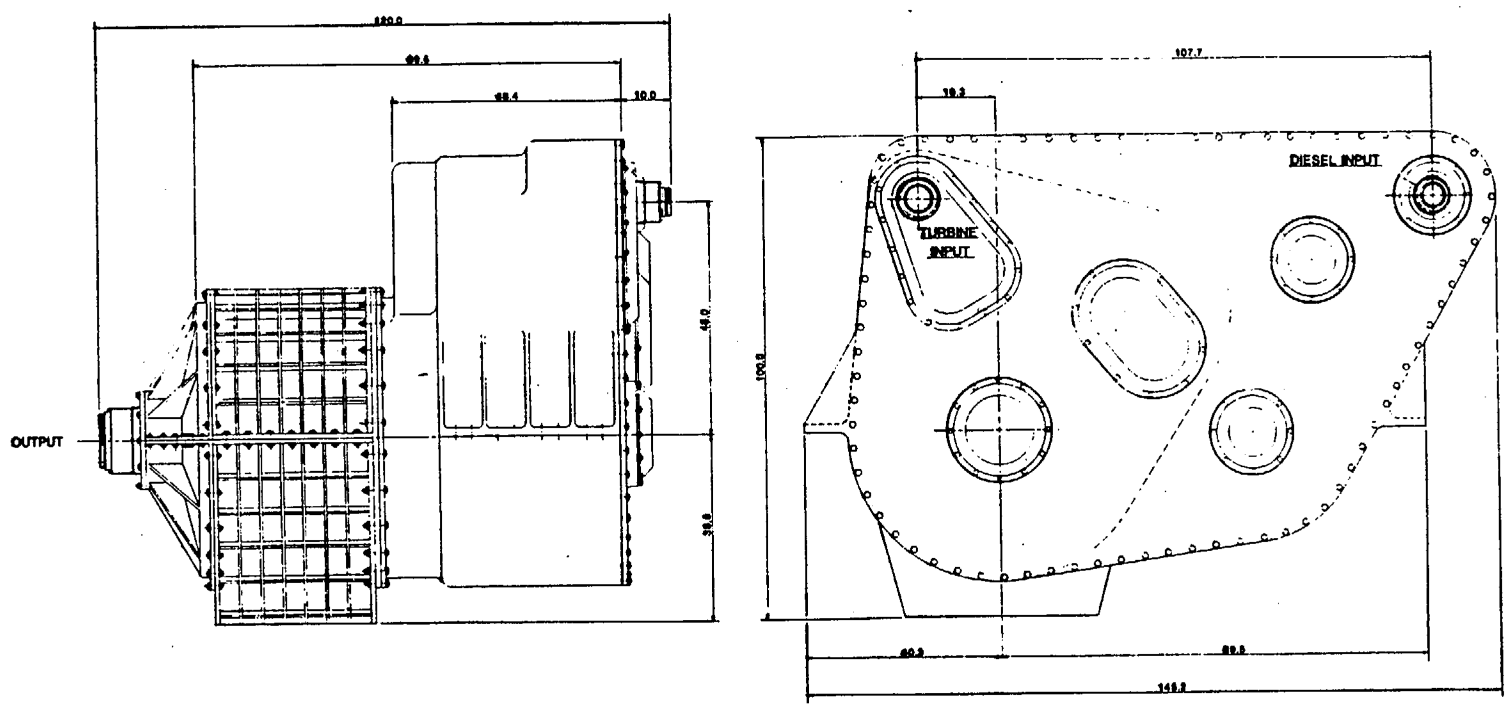


Figure 2.2-1. Gearbox Schematic, Configuration 2.



MDC-SES POWER TRAIN ARRANGEMENT
SINGLE TURBINE / DIESEL CODOG
17-JULY-1981

Appendix B

Figure 2.2-2. Configuration II

Table 2.2-1. Gear Design Data - Configuration II. Reference Figure 2.2-1

MESH / UNITS (1)	1-2		2-3		4-5		5-6		6-7	
	PINION	GEAR	PINION	GEAR	PINION	GEAR	PINION	GEAR	PINION	GEAR
Horsepower	27000		27000		4500		4500		4500	
RPM	3600	1605	1605	1187	798	771	771	746	1560	746
Pitch Dia.	11.905	26.684	26.684	36.126	36.126	37.358	37.358	38.589	18.474	38.589
Face Width	16.800		16.800		6.000		6.000		6.000	
Normal DP	2.5		2.5		2.5		2.5		2.5	
No. Teeth	29	65	65	88	88	91	91	94	45	94
Press. Angle	20.00		20.00		20.00		20.00		20.00	
Helix Angle	13.00		13.00		13.00		13.00		13.00	
Contact Ratio	1.66		1.75		1.77		1.77		1.72	
Pitch Line Velocity	11220		11219		7547		7550		7545	
K Factor	574		308		179		173		262	
Allow. Hertz Str.	173000		131000		131000		131000		131000	
Hertz Str. Max.	141144	141144	102305	102305	85017	85017	83735	83735	101313	101313
Durability Hp	40563		44270		10684		11013		7523	
Allow Bend Stress	55000	38500	38500	42500	42500	29750	29750	29750	55000	29750
Bending Str.	38464	35168	34446	33798	27930	27873	27946	27905	28382	26886
Bending HP	38607	29559	30178	33952	6847	4803	4741	4798	8720	4979
AGMA Quality No.	11		11		11		11		11	

(1) o All dimensions are in inches o Stresses are in PSI
o Pitch-line velocity is in ft/min o Angles are in degrees

Table 2.2-2. Planetary Gear Design Data - Configuration II

Type	Single Stage Planetary, Double-Helical, Stoeckicht System		
Horsepower		27000	
Input Speed - RPM		1187	
Gear Ratio		3.958	
Output Speed - RPM		299	
	SUN	PLANET	ANNULUS
Pitch Dia.-In	14.547	14.244	43.036
Number of Teeth	48	47	142
Number of Planets		5	
Normal Diametral Pitch		3.81	
Pressure Angle		22.5°	
Helix Angle		30.0°	
1/2 Effective Face Width-In.		5.906	
Face Width of Sun Pinion-In.		14.173	
K Factor, Sun/Planet		464	
K Factor, Planet/Annulus		157	
Unit Pressure, Planet Brg. PSI		610	
Planet Bearing Diameter-In.		9.25	

Table 2.2-3. Gearing Efficiencies, Configuration II

OPERATIONAL MODE	INPUT POWER (HP)	POWER LOSS (HP)	EFFICIENCY	OIL REQUIREMENT (GPM)
TURBINE	27000	752	.9721	360
DIESEL	4500	288	.9360	220

1) DTE MEDIUM OIL

2) 120° F. OIL INLET TEMPERATURE

3) DESIGN W/O EXTRA IDLERS FOR ROTATION CHANGE

The planetary gearset utilizes double-helical gearing; all external gears are surface hardened and ground. The annulus gears are through hardened and machined to final dimensions. This gearset is designed in accordance with the Stoeckicht principle, with floating sun and annulus gears to achieve efficient load sharing among the five planets utilized in Configuration II.

A comprehensive list of the features designed into the propeller reduction gearing for configuration II by Cincinnati Gear is presented in Section 2.3.

2.3 DESIGN FEATURES AND REQUIREMENTS

The following is a summary of the salient features and design requirements prepared by the Cincinnati Gear Company for application to the propulsion reduction gears for Configuration I and II.

2.3.1 MATERIALS -- The use of materials of high cost, low availability, or high strategic index shall be kept to a minimum. All materials shall be corrosion resistant in a marine atmosphere or shall be suitably protected against corrosion, by lubricant, plating or other coating.

2.3.2 INTERCHANGEABILITY -- Assemblies and details with the same part number shall be completely interchangeable. Gears shall not be matched to gear cases.

2.3.3 IDENTIFICATION AND MARKING -- All parts, where practical, shall be permanently and legibly marked for identification. An identification plate shall be affixed to the primary gear case having the following information:

a. Manufacturer's name and code identification number

b. Manufacturer's part number

c. Manufacturer's serial number

d. Date of manufacture

e. Unit weight (dry) in pounds

2.3.4 MAINTAINABILITY -- The transmission shall be designed for maintenance with modest skill levels and standard tools. Removal and replacement of major components shall be simplified as far as practical.

2.3.5 EXTERIOR DIMENSIONS -- The exterior dimensions of the transmission, the direction of rotation, the clutch locations, the interface mounting and the shaft coupling requirements shall be in accordance with the RMI configuration control drawings.

2.3.6 DIRECTION OF ROTATION

Turbine	*Output Shaft	Configuration
DDA 570	CCW	I
GE LM2500	CCW	II
SACM 195V12CZSHR	CW	I
SACM 195V20RVR	CW	II
Propeller Shaft	**	

*View direction looking toward engine from end of engine output shaft.

**Port and starboard propellers are to rotate in opposite directions for both turbine and diesel operation.

2.3.7 MODES OF OPERATION -- The modes of operation are as follows:

Configuration I

Mode of Operation	Prime Mover
1 - Cruise	Diesel
2 - Dash	Twin Turbine
3 - Engine Out	Single Turbine

Configuration II

Mode of Operation	Prime Mover
1 - Cruise	Diesel
2 - Dash	Turbine

Overrunning and decoupling clutches shall be provided as applicable to allow CODOG or engine out operation.

2.3.8 DESIGN PERFORMANCE REQUIREMENTS -- The design load conditions and life requirements shall be in accordance with Table 2.3-1.

2.3.9 LIFE -- The transmission is to have a design life of 20 years.

Main power gear to be designed for hours given in Table 2.3-1 at rated power and speed. Journal bearings are used to support main power gears. Anti-friction bearings (if employed) supporting overrunning clutches or accessory gears shall have a calculated B-10 life of 10,000 hours.

Anti-friction bearings, seals, O-rings and gaskets are to be replaced at overhaul intervals of 2000 hours initially with a maturity goal of 5000 hours.

2.3.10 MATERIALS -- The main transmission casing is to be A356-T71 cast aluminum. All castings are to be impregnated per MIL-STD-276 using impregnants per MIL-I-6869.

All aluminum components are to be hard anodized per MIL-A-8625, Type III.

All aluminum parts requiring threads shall incorporate "Keensert" brand steel inserts, 300 Series Passivated CRES.

Table 2.3-1. Design Load Conditions and Life Requirements

CONFIGURATION	MODE OF OPERATION	PRIME MOVER	RATED INPUT POWER AND SPEED (HP @ RPM)	GEAR DESIGN LIFE (HRS.)	PROPELLER SPEED (RPM)
I	1 - CRUISE	DIESEL	2200 @ 1560	100,000	200
	2 - DASH	TWIN TURBINES	6300 @ 11,500	20,000	300
	3 - ENGINE OUT	SINGLE TURBINES	9000 @ 11,500	1,000	300
II	1 - CRUISE	DIESEL	4500 @ 1560	100,000	200
	2 - DASH	TURBINE	27,000 @ 3600	20,000	300

2.3.11 ENVIRONMENT -- The transmission shall meet operational requirements of this specification under the following:

a. Ambient air temperature 0-140°F.

b. Relative humidity to 100% including conditions where moisture condenses on the exterior surfaces of unit.

c. Exposure of fungi and bacterial growth as encountered in tropical regions.

d. Exposure to sea or fresh water.

e. Accelerations as follows: TBD

2.3.12 WEIGHT -- The transmission including input and output couplings shall not exceed TBD lbs. (dry).

2.3.13 LUBRICATION -- The transmission lubrication system shall be as follows:

a. Gearbox Driven Lubrication Pump - The transmission shall have provisions for driving a positive displacement lubrication pump which is to provide lubricant to transmission bearings, seals, and gears.

b. Filtration - A duplex filter is required as part of the ship system - 10 micron nominal (25 micron absolute).

c. Sump - The transmission case shall provide a minimum capacity of 1.5 times normal flow rate and a settling and collection point.

d. Auxiliary Electric Driven Pump - The ship system will provide lubricant for pre-post, and supplemental requirements of the transmission.

e. Heat Exchanger - The ship system will incorporate a heat exchanger for removal of heat from the lubricant.

f. Deaerator - The ship system will incorporate a deaerator to insure entrapped air in lubricant does not exceed 10%.

g. Lubricant Type - The transmission shall be designed to operate on mineral oil for maximum corrosion protection.

2.3.14 INSTRUMENTATION -- The transmission shall contain provision for monitoring:

a. Inlet Oil Temperature

b. Outlet Oil Temperature

c. Inlet Oil Pressure

d. Metallic Debris in Gearbox Sump

e. Overall Vibration Level

2.3.15 GEAR INSPECTION -- Provisions shall be made for inspecting all gear meshes without transmission disassembly by means of readily removable inspection port covers.

2.3.16 COMPONENT RETENTION -- All rotating components shall be secured with positive locking devices. Externally mounted fasteners are to be lock-wired.

2.3.17 GEAR DESIGN -- Life calculations are to be per applicable AGMA Standards and derating factors.

3 / TECHNICAL RISK

3.1 SCOPE OF DISCUSSION

The propulsion system is comprised of gas turbine engines, diesel engines, and their ancilliaries, the air inlets, exhaust and systems, reduction gearing, shafting, thrust bearing and propeller. A discussion of risk for the shafting thrust bearing and propeller is included in Appendix A.

The following three subsections are a discussion of the technical risk for the prime movers and the reduction gearing.

3.2 PRIME MOVERS

The ancilliary systems for the prime movers are all within the state-of-the-art. The air inlet system, a combination of ducting, demisters with by-pass provisions, anti-ice systems and noise suppression measures have all been applied to modern ships in commercial and Navy service with excellent results. The inlet system for the 3KSES is a completed design which was fully model tested for performance. All gas turbine and diesel powered craft and ships have similar air inlet systems. Similarly, exhaust/uptake systems, usually comprised of gas turbine and diesel connections, ducting, sound suppression material/items and weather closures, are well known, within the state-of-the-art and show good service records. Systems for supply and management of fuel, lube oil, starting, etc., are low-risk due to the many examples in service.

3.2.1 DIESEL ENGINES -- The SACM 175V8RVR, 195V12 CZSHR, 195V12RVR, and 195V20 RVR are representative of modern, reliable, high specific power, high speed diesels being used in increasing numbers for all applications world-wide. These engines are fully developed rugged units with superior economy at part and full load and are considered low risk.

3.2.2 GAS TURBINES -- The GE LM2500 gas turbine is a fully developed free turbine unit currently in Navy inventory. Service experience includes the DD 963, the FFG 7, the PHM and ships of foreign Navies. There is low risk associated with its selection.

The DDA 570 gas turbine is marine (salt) qualified with a rapidly increasing industrial base with current orders for 95 with 34 delivered. The risk associated with selection of this engine is minimal and is expected to be low risk at the time of MDC build.

3.2.3 REDUCTION GEAR -- The reduction gear design is a low risk conservative, state-of-the-art design furnished by the Cincinnati Gear Company. The design has not been compromised by small size, weight or low cost approaches which would reflect on reliability. Premium grade steels and conservative stress levels combined with a rigid casing, generous bearing areas, high capacity lube supply and fully factored loadings give confidence in the design. The reduction gear designs which are proposed do not require developmental activity or unusual manufacturing techniques.

Comparable in-service reduction gears designed and manufactured by the Cincinnati Gear Company include the parallel offset reduction gearbox for the Boeing Jetfoil and the combined drop/epicyclic gearbox for the American Enterprise craft.

4 / COMPONENT AND SYSTEM RELIABILITY

The selection process for the equipments for the propulsion machinery configurations used reliability as one of the driving parameters. The high reliability of the chosen equipments with the diesel engines used for lift power and for off-cushion operations assures a very reliable ship. Redundancy for off-cushion is provided with the gas turbines, however, they will be operating inefficiently on the low end of their power curve. This case then, would be for emergency operation only.

The SACM diesels, the DD570 gas turbine and the LM2500 gas turbine have been proven reliable. The LM2500 is currently in Navy inventory. The diesels and the DD570 are in commercial/industrial use. The combination of engines chosen for the arrangements offers an efficient reliable match.

The propulsion transmission trains feature Cincinnati Gear Company gearbox designs. These designs utilize conservatively loaded gears and journal bearings which result in gearboxes that are durable and reliable.

A typical combat mission profile was used as a basis for reliability evaluation (Reference: Medium Displacement Combatant Surface effect Ship Technical Report, April 1981). This profile does not have any operational conditions that would affect the inherently high reliability of the proposed gearboxes, which feature conservatively designed helical gears mounted in journal bearings.

The SSS overrunning clutches and the air clutches that are used to engage/disengage the diesels are mounted on the outside of the gearbox. These clutches are reliable and commonly used in marine and industrial applications. The demisters and exhaust system in all configurations are of straightforward design and are not considered to be reliability-critical. These power trains will be very reliable in any of the configurations presented.

5 / MAINTAINABILITY ASSESSMENT

The maintenance philosophy for the SES is minimum preventive maintenance while operational. Corrective maintenance onboard ship will be limited to removal and replacement of mission-essential equipment. This philosophy dictates a total maintenance concept that demands high reliability parts with most maintenance performed while the ship is in port.

Removal and replacement of major items is discussed in Section 1. The following maintainability criteria were applied in developing the concepts described in this report.

a. No secondary equipment removals required for access.

b. Adequate access doors for all equipment with sliding or hinged covers.

c. No structural cutouts for equipment access.

d. Adequate access around all equipments for inspections, checks, adjustments and corrective in-place maintenance.

e. Cathodic protection for dissimilar metal joints.

f. Standardization of fasteners/parts/materials.

g. Quick disconnect latches, cables, lines, etc.

h. Minimizing special tools and test equipment.

The machinery arrangement in the 1800 LT Logistic ship is the most critical with respect to removal and replacement of major equipment. The propulsion equipment is in the sidehull 30 feet below the main deck. The access and removal paths will provide further study and the best methods for removal determined before configurations are firmed up.

All the heavy components such as gear boxes, engines, etc. will have lifting lugs. Maintenance rails are provided on special removal paths through combustion air inlet or the fan air inlet for major equipment.

6 / PRODUCIBILITY

All components and assemblies for the Propulsion System Configuration have a solid manufacturing base relative to state-of-the-art practices, tooling and machines.

The gas turbine and diesel engines are in series production.

The reduction gear designs as selected by RMI and Cincinnati Gear do not present producibility problems beyond learning curve development during manufacturing and test of the first reduction gear set.

The producibility of the propeller, bearings and drive shafting is discussed in Appendix A.

7 / COST AND WEIGHT ESTIMATES

Budgetary estimates of costs for the major elements of the MDC machinery systems, including the lift system, are shown in Table 7-1 for Configuration I and in Table 7-2 for Configuration II. It should be noted that budgetary prices are not commitments on the part of the suppliers but are useful for comparative purposes.

Included in the table are corresponding weight estimates and the recommended sources. Weights are estimated on the basis of vendor information, scaled values from the 3KSES, and calculations using available drawings. The budgetary costs and recommended sources are from selected data obtained during the course of the design study. Items denoted RMI as the recommended source are those which involve make-to-print items or are fluid systems.

Table 7-1. Weights, Budgetary Costs and Suggested Sources for Configuration I Machinery

ITEM		Required per Ship	Weight LBS	Cost Dollars	Suggested Sources
1	Gas Turbine Engine, DDA 570KF	4	9580	2682000	Allison
2	Diesel Engine, 195V12 CZSHR	2	32120	620000	SACM
3	Propeller Gearbox	2	45779	1550000	Cincinnati Gear
4	Lift Fan Gearbox	2	7600	100000	Philadelphia Gear
5	Propeller, Shaft & Controls	2	62200	1900000	Tacoma Boat
6	Fan Assembly	4	14120	420000	Aerophysics
7	Demister	4	8000	100000	Peerless
8	Exhaust Duct, Turbine	4	18656	93280	RMI
9	Exhaust Duct, Diesel	2	1600	8000	RMI
10	Lube System, Prop. Gearbox	2	12858	59592	RMI
11	Lube System, Fan Gearbox	2	2900	15776	RMI
12	Thrust Block, Prop. Shaft	2	14272	126718	RMI
13	Stern Seal, Prop. Shaft	2	1680	54880	Tyton
14	Propeller Shaft	2	(27200)	*	See Item 5
15	Flex. Coupling, Prop. Shaft	2	4630	34980	Zurn
16	Overrunning Clutch, Turbine	-	Included in Item 3		--

( ) Weight included in Item 5
* Cost included in Item 5

Table 7-2. Weights, Budgetary Costs and Suggested Sources for Configuration II Machinery

ITEM		Required per Ship	Weight LBS	Cost Dollars	Suggested Sources
1	Gas Turbine Engine, LM2500	2	22410	4510000	General Electric
2	Diesel Engine, 195V20 RVR	2	57860	1500000	SACM
3	Propeller Gearbox	2	54555	1790000	Cincinnati Gear
4	Lift Fan Gearbox	2	13000	144000	Philadelphia Gear
5	Propeller, Shaft & Controls	2	99920	1900000	Tacoma Boat
6	Fan Assembly	6	18540	416000	Aerophysics
7	Demister	2	10000	80000	Peerless
8	Exhaust Duct, Turbine	2	25800	129000	RMI
9	Exhaust Duct, Diesel	2	3248	16240	RMI
10	Lube System, Prop. Gearbox	2	19800	88084	RMI
11	Lube System, Fan Gearbox	2	5558	28008	RMI
12	Thrust Block, Prop. Shaft	2	24078	190000	RMI
13	Stern Seal, Prop. Shaft	2	2020	56516	Tyton
14	Propeller Shaft	2	(29120)	*	See Item 5
15	Flex. Coupling, Prop. Shaft	2	6240	47080	Zurn
16	Shaft Brg. Module, Turbine	2	1234	49400	RMI
17	Overrunning Clutch, Turbine	2	TBD	TBD	SSS

( ) Weight included in Item 5

* Cost included in Item 5

8 / PROPULSION MACHINERY CONFIGURATION UPDATE

8.1 GENERAL

The update of the 3KSES Producibility Improvement Study included changes to the machinery arrangements in response to Navy inputs, Reference 3, and Task L statement of work. These changes revised the propulsion machinery arrangements to:

a. Suit the revised MDC sidehull lines

b. Improve accessibility and maintainability

c. Reflect lift machinery changes in the sidehull as the result of the adoption of forward lift machinery and the elimination of long lift air ducting.

Described are two updated MDC propulsion machinery arrangements, one for the 940-Ton MDC which was modified to a 1000-Ton MDC and one for the 1500-Ton MDC. Both propulsion systems feature Combined Diesel Or Gas Turbine, CODOG, propulsion machinery with the capability of diesel drive to the aft lift fans.

The CODOG arrangement gives off-cushion cruise up to 20 knots (diesel power) with maximized range for convoy, cruise and fleet operations. The gas turbines give speeds up to 50 knots for high speed transit with the propulsion diesels driving the aft lift system. Separate lift diesels drive the forward fans. The propulsion machinery arrangement described gives good accessibility and maintainability. The diesel and gas turbine are spaced apart in separate compartments to conform to more usual practice.

8.2 1000 LT CONFIGURATION

The 940 LT configuration, as described in Reference 2, was modified to a 1000 LT configuration as the result of RMI review and with the Navy letter (Reference 3) regarding CDRL No. E06C, Ship Characteristics and Performance Report, dated 26 June 1981. The results of this work, which included other changes, is documented as follows.

The gearbox design change for the over and under installation of the 570 KF gas turbine engines to the side-by-side installation consists of rotating the lower engine input gear 90 degrees and modifying the gearbox casing. Section 2.3.1 has a description of the gear design produced for the original engine configuration. In all other respects, it is applicable for the current configuration.

The side-by-side configuration utilizes the available space to achieve a practical layout for both operation and maintenance. Attachment of the engine to the gearbox, using the gearbox casing as the engine front mount, minimizes alignment procedures for fast installation and removal of the gas turbine engines. The configuration also minimizes structureborne noise and vibration.

The 8 degree slope for the ventilated propeller installation is maintained throughout the machinery train to utilize conventional gearbox design, shafting, and flexible couplings. Figure 8.2-1 shows the arrangement of the aft machinery for propulsion and lift. The diesel engine used to provide lift system power for the on-cushion condition as well as hullborne propulsive power is the SACM 195V12RVR. It is mounted forward of the reduction gearbox and with the DDA-570 gas turbine engines mounted aft conforms to current practice.

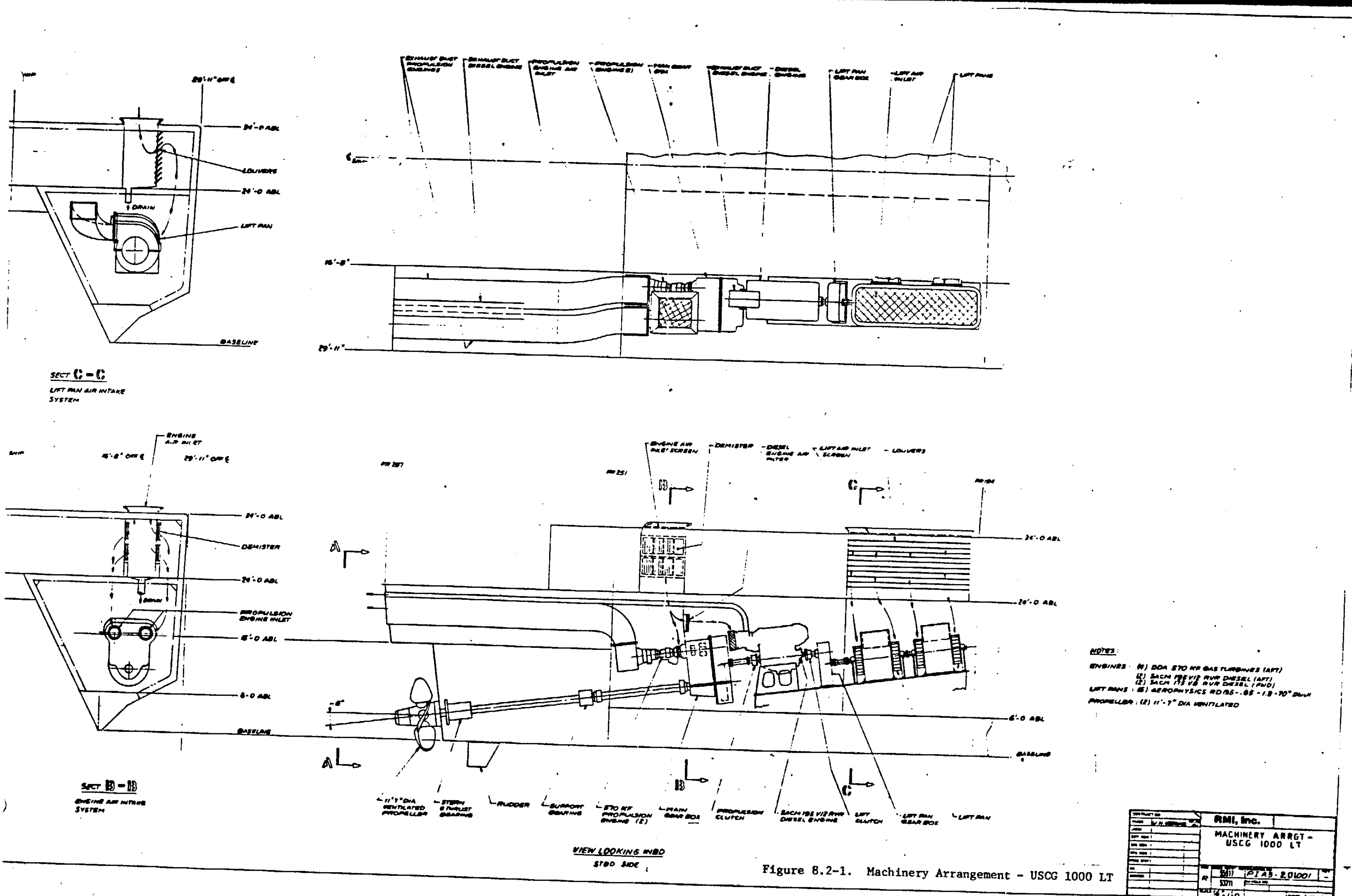


Figure 8.2-1. Machinery Arrangement - USCG 1000 LT

8.3 1500 LT CONFIGURATION

For the 1500 LT configuration the propulsion machinery is arranged port and starboard in the sidehulls well aft between the transom, frame 308, and bulkhead, frame 210. Except for the engine combustion air intake systems, all propulsion equipment is located below the second deck. Figure 8.3-1 shows the machinery arrangement.

Propulsion power is provided by two LM 2500 gas turbines and two SACM 195 V20 RVR diesel engine installed one in each sidehull. The LM 2500 is located approximately mid-way between Frames 270 and 231. It is mounted at 8 degrees to the decks, compressor end up. Its combustion air inlet is located forward and above the engine inlet bellmouth. Air enters the combustion intake from a bellmouth at the main deck level and is aspirated through demister banks located in the air inlet plenum. The inlet has anti-ice provisions which used reingested gas turbine exhaust cases for surface heating and mixing with the ambient air. As further protection against combustion inlet icing or blockage, blow-in doors are provided integral with the demister banks. Noise attenuating panels and surface treatment are used in the inlet plenum and trunk to limit compressor noise on the main deck.

The LM 2500 gas turbine exhaust trunk, a rectangular structure of double walled Inconel, routes up and aft beneath the second deck to the transom. An exhaust closure door is provided to prevent spray, moisture and salty air from depositing in the exhaust uptake and engine. The double wall structure of the exhaust is used for exhaust duct wall cooling air flow. The inner duct wall is treated with noise attenuating treatment to reduce community and ship noise levels.

The LM 2500 gas turbine engine is supported on its own mounts from the ship structure foundations. Adjustment is provided for alignment purposes. Removal of the engine is via the combustion air intake using rails to translate the engine gas generator and turbine separately, forward and upward through the previously cleared inlet plenum. This is similar to the method used on the DD 963 and similar to that designed

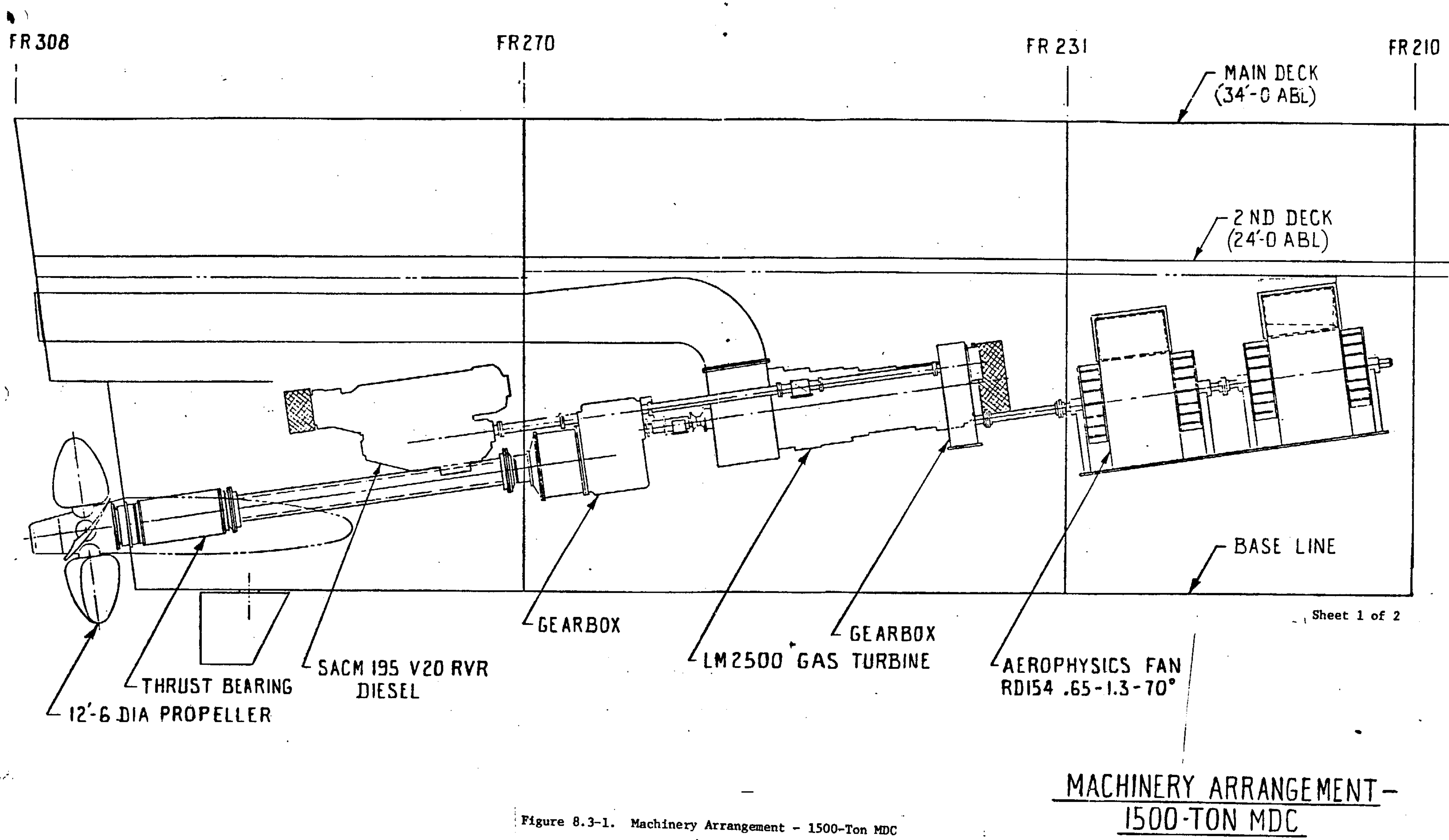
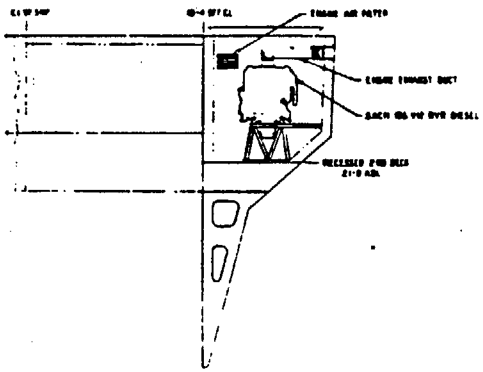
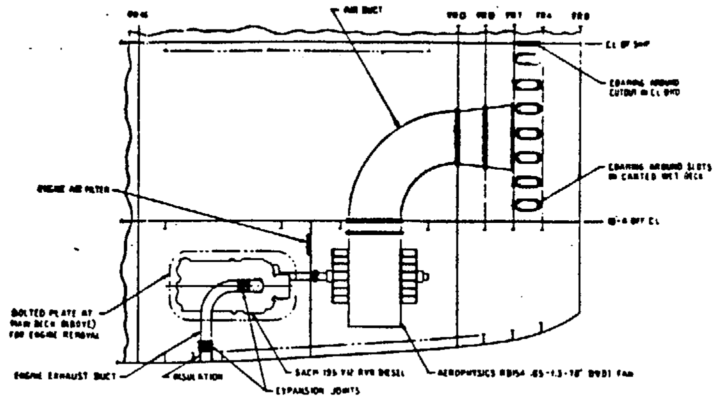


Figure 8.3-1. Machinery Arrangement - 1500-Ton MDC



SECTION C-C



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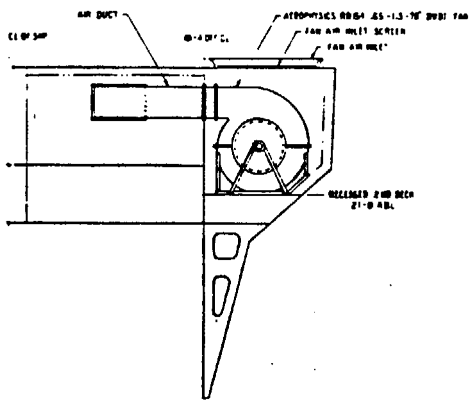

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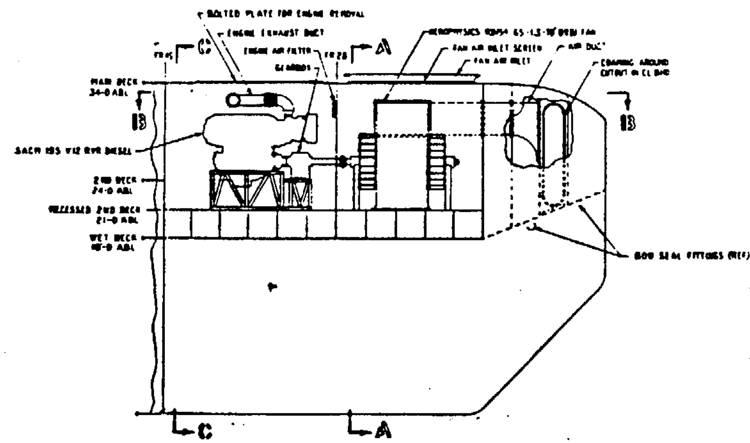

BOW SEAL MCHRY ARRGT

SHELL PLATING REMOVED FOR CLARITY. STBD SHOWN PORT OPPOSITE EXCEPT FOR ENGINE SHAFT ROTATION.

for the 3KSES. The LM 2500 turbine shaft drives into the reduction gearbox through a shaft support bearing mounted externally to the gearbox and an SSS clutch. The reduction gearbox has an overall reduction ratio of 10.06. The first reduction stage is by helical gears and the final stage is with an epicyclic reduction gearset. The reduction gearbox is described more fully in Section 8.6.* From the epicyclic gearbox the drive is transmitted by the propeller drive shaft to the thrust bearing module and the propeller. Zurn flexible couplings on the propeller drive shaft allow the shaft to absorb misalignments and deflections. Technical descriptions and performance for the propeller are given in Appendix A, Section 8. Technical descriptions of the propeller, thrust bearing and propeller shafting are given in Appendix A, Sections 1 and 2.

Positioned aft of Frame 270 is the SACM 195 V20 RVR Diesel engine. It also drives into the reduction gearbox where it is clutched into its drive pinion by a friction clutch actuated by air pressure. Combustion air for the diesel is taken from the compartment through a filter. The diesel exhaust is taken up and aft out of the transom. The diesel engine is supported on elastomeric mounts to minimize ship vibration and noise. As stated previously, the diesel drives either the propeller or the lift fans. Drive for the lift fans is taken from the reduction gear and into the fan drive shaft alongside of the LM 2500 engine. The drive shaft connects to the lift fan speed increaser gearbox which is mounted adjacent to the engine inlet.

* Although much of this section is common to Section 2.2, sufficient difference exists to justify a separate section.

8.4 DIESEL ENGINES

For the purposes of this study SACM diesel are selected to provide low speed propulsion power and lift power. Catalog data gathered to aid selection of the exact diesel engine indicated that the SACM line of diesel engines is appropriate for the MDC designs.

During the extension period, a meeting was held with SACM representatives at RMI. The SACM representatives had been reviewing the MDC application and recommended the reduced volumetric ratio, RVR, diesel line as being the most appropriate. Further discussion of the MDC application resulted in SACM recommending ratings published by them as the most suitable for tugboats and supply vessels. This rating is based on the expected long duration power requirements of the propulsion and lift system as opposed to the ratings previously selected by RMI which were, in essence, high power/short duration ratings - fast patrol boat usage.

The ratings shown in Table 8.4-1 are those recommended and adopted for use in the MDC ships

Table 8.4.1 SACM Diesel Engine Ratings, SHP/RPM

SACM ENGINE DESIGNATION	NUMBER OF CYLINDERS/HP/RPM				
	6 (In Line)	8 (Vee or In Line)	12 (Vee)	16 (Vee)	20 (Vee)
175 RVR	700/1500	925/1500	1400/1500	1700/1400	
195 RVR			2220/1450	2960/1450	3700/1450

Fuel consumption 0.36 pounds/SHP

The RVR engines rated per Table 8.4-1 are expected to give 24,000 hours between major overhauls. If required, and at the expense of the engine overhaul period, the rated powers can be increased to maximum intermittent power which is up to 46 percent more (for 1/2 hour in 6 hours) than the ratings shown in Table 8.4-1.

For the 1500 LT MDC, the diesel engines selected are the SACM 195V20RVR for the propulsion and lift and the 195V1ZRVR for the lift engines forward. For the 1000 LT MDC, the diesel engines selected are the SACM 195V1ZRVR for propulsion and lift and the 175V8RVR for the lift engines forward. Fuel consumptions at MCP are .36 pounds/SHP/hr. This selection gives diesel engines of adequate power and life with relatively low weight compared to the slower speed diesels more commonly used for the Navy.

8.5 ALTERNATE LIFT MACHINERY ARRANGEMENTS

The sidehull configuration as recently revised is sufficiently narrow as to require in-line placement of the propulsion and lift machinery. Space restrictions in the vicinity of the keel require that the propeller gearbox be elevated above the keel to obtain working clearances for the gearbox installation. Adequate clearances for the gearbox can be most readily obtained by inclining the propeller shaft upward, proceeding forward from the propeller into the sidehull; the propeller shaft angle is limited by the maximum angle of inclination at which the gas turbine will operate, since the gas turbine shaft is parallel to the propeller shaft. The propeller shaft angle determined from these considerations is 8 degrees.

The in-line placement of the lift and propulsion machinery at an angle of 8 degrees results in this equipment extending into the upper deck spaces, which could be allocated to other functions. In addition, the center of gravity of the machinery is raised considerably above where it would be if some or all of the machinery were parallel to the keel.

By introducing non-parallel shafting between the lift and propulsion sections, the lift machinery could be installed parallel to the keel, avoiding the intrusion of this equipment into the upper deck spaces while reducing the center of gravity of the machinery installation. The propeller shaft, the gearbox, the gas turbine and the diesel engine used for cruise power would be installed with shaft angles of 8 degrees, and could be accommodated within the space allocated for the propulsion equipment.

To obtain an arrangement that uses non-parallel shafts for the propulsion and lift machinery sections requires the use of either bevel gears, crossed-axis helical gears or universal joints.

The design of a bevel gearset sized for the lift fan drive and for an angle change of only 8 degrees requires that the cone distance for the gearset be appreciably longer than that for which gear generating equipment is available within the United States. Overseas sources for suitable equipment to generate these gears are being investigated.

In the case of a crossed-axis helical gearset, there is the problem of developing the data and procedures to design a gearset for an 8 degree shaft angle. Design information on these gears is sketchy and incomplete. Gearsets to date have been designed for shaft angles of 90° which is far beyond the MDC requirements and involves data and ratings which are not directly applicable to small shaft angles, i.e., 8 degrees. A leading manufacturer of helical gears has been contacted on this problem and is "guardedly optimistic" regarding a feasible design. Once designed the gearset could be manufactured with the same equipment used for helical gears.

Data on commercially available universal joints indicate that existing units can accommodate the torques, speeds and shaft angularities required for a horizontal installation of the lift fan machinery. For the 1500-Ton MDC lift system requirements a 20,000 hour B_{10} life for the joint bearings is expected based on manufacturers' data (Zurn-Voith).

The advantages to be gained in improved machinery installations justify continued efforts to provide drives that use bevel gears, crossed-axis helical gears or universal joints to make viable candidates for use in the lift and propulsion transmissions of surface effect ships.

8.6 UPDATED MACHINERY CONFIGURATION II (1500-TON)

The power system installed in each sidehull of this configuration comprises one General Electric LM2500 gas turbine engine that is geared to the propeller shaft for on-cushion operation and a diesel engine (SACM 195V20 RVR) that drives either the lift fans for on-cushion operation or the pro-propeller for off-cushion operation; both engines through reduction drive gearing. When the diesel drives the propeller the gas turbine output shaft is decoupled from the propeller drive.

The design of the sidehull for updated Configuration II requires in-line placement of the gas turbine and the diesel engine, and results in a compact configuration for the propeller reduction gearing, with ample space around the engines and gearboxes for installation, inspection, and maintenance activities. The positions of the centerlines of the propeller shaft and of the two engines are resolved within the gearbox by means of parallel-shaft gearsets that serve as vertical, and lateral drop boxes while reducing the engine speeds to those required at the input of the planetary gearing coupled to the propeller shaft. Since both the power and the speed of the two engines are quite different, the engines require separate reduction gearsets. In addition there is the requirement that the port and starboard propellers rotate in opposite directions regardless of whether the diesel or the gas turbine is driving; this requirement calls for additional idler gears in one of the reduction gearboxes, since the gas turbine is available with only one rotation and it is desirable for logistics reasons that the port and starboard diesels have the same rotation. All of the above considerations were implemented in the gearbox design developed for the propeller drive as a result of this study.

8.6.1 GEARBOX DESIGN -- The propeller reduction gearing is divided into three sections that are integrated into a single gearbox assembly. There are two sets of parallel-shaft gears, one for each engine, that drive a common gear on the input shaft of the planetary gearset. Each set of parallel-shaft gears serves as the drop box gearing from its driving engine to the propeller shaft centerline while reducing engine speed to that required at the planetary input sun gear.

In order to locate the propeller shaft as low as possible in the ship's sidehull, it is judged essential that the final stage of propeller gearing, particularly the bull gear or its equivalent, be of a relatively small diameter. For the particular power and propeller speed involved, this objective is achieved by utilizing a planetary gearset as the final stage of speed reduction from engine to propeller speed, for both the diesel and the gas turbine. By selecting a gear ratio of the order of 4:1, it is possible to install five planet gears in a planet gear carrier and distribute the large torque load among five gears of relatively small diameter and modest face width. This results in a housing diameter for the planetary gearset that is considerably smaller than that of a bull gear in which all of the torque load would have to be transmitted through a single mesh. In this arrangement, the first stage of speed reduction, in which the torques are low and the speeds are high, is taken in single mesh, parallel-shaft gears, while the final stage of speed reduction, involving high torques and low speeds, is taken in a multiple-path planetary gearset of relatively small diameter and low weight. This results in lower weight for the entire gearbox installation as well as in satisfying the primary objective of propeller-shaft location.

With the diesel engine driving the propeller, the gas turbine shaft is decoupled from the propeller reduction gearbox by a SSS overunning clutch mounted external to the gearbox just aft of the outboard bearing module that supports the gas turbine engine output shaft. When the gas turbine engine is driving the propeller, the diesel engine is decoupled from the reduction gear by a dry-plate, air-actuated clutch mounted outside the gearbox housing on extensions of the diesel idler gear and shaft.

The transmission shafting is designed such that all propeller loads will be accommodated by bearings external to the gearbox, thereby avoiding any tendency for the propeller loads to distort the gear housing or the shafts and compromise gear tooth action. Adequate access and space are provided on the forward side of the drop box, opposite the planetary for installation and inspection of the mechanisms and controls required for the variable pitch propeller. The gearbox is foot-mounted in one plane and incorporates an oil sump of adequate capacity that is integral with the gearbox housing. The engine shafts of the diesel and the gas turbines are flexibly coupled to the gearbox shafts.

A summary of data relating to the gear dimensions and stresses for the parallel-shaft, drop box gearing, as given in Table 8.6-1; technical data for the planetary gearset is given in Table 8.6-2. AGMA standards and derating factors were used in sizing all gearing. Gear efficiencies and lube requirements are summarized in Table 8.6-3. The schematic design and outline drawing are similar to those shown in Section 2, Figures 2.4-1 and 2.4-2.

All parallel-shaft gearing is single-helical, and is surface hardened and ground. Each high speed stage utilizes thrust collars that bear on the gear rims, to cancel out the axial thrust in the helical mesh; these collars eliminate highly loaded thrust bearings, reduce the number of thrust bearings required, and permit the use of higher helix angles in single helical gears than are feasible with conventional thrust bearings.

The planetary gearset utilizes double-helical gearing. All gears are surface hardened and ground, except the annulus gears are through hardened and machined to final dimensions. This gearset is designed in accordance with the Stoekicht principle, with floating sun and annulus gears to achieve efficient load sharing among the five planets utilized.

Table 8.6-1. Gear Design Data - Updated Configuration II

MESH UNITS (1)	1-2		3-4		5-6		6-7		7-8	
	PINION	GEAR	PINION	GEAR	PINION	GEAR	PINION	GEAR	PINION	GEAR
Horsepower	27,000		3,700		3,700		3,700		3,700	
RPM	3,600	1,388	1,450	1,450	1,450	1,517	1,517	1,763	1,763	786
Pitch Diameter	13.136	34.075	18.473	18.473	18.473	17.653	17.653	15.190	15.190	34.075
Face Width	16.800		6.000		6.000		6.000		6.000	
Normal DP	2.5		2.5		2.5		2.5		2.5	
No. Teeth	32	83	45	45	45	43	43	37	37	83
Pressure Angle	20.000		20.00		20.00		20.00		20.00	
Helix Angle	13.00		13.00		13.00		13.00		13.00	
Contact Ratio	1.71		1.64		1.65		1.68		1.78	
Pitch-Line Velocity	12,412		7,030		7,030		7,030		7,030	
K Factor	452		315		323		356		277	
Allow. Hertz Stress	17,300		131,000		131,000		131,000		131,000	
Hertz Stress	TBD		TBD		TBD		TBD		TBD	
Durability HP	TBD		TBD		TBD		TBD		TBD	
Allow. Bend Stress	55,000	55,000	55,000	29,750	55,000	29,750	29,750	29,750	29,750	42,500
Bending Stress	TBD		TBD		TBD		TBD		TBD	
Bending HP	TBD		TBD		TBD		TBD		TBD	
AGMA Quality Number	11		11		11		11		11	

- All dimensions are in inches.
- Pitch-line velocity is in feet/minutes.
- Stresses are in PSI.
- Angles are in degrees.

Table 8.6-2. Planetary Gear Design Data - Updated Configuration II

Type	Single Stage Planetary, Double-Helical, Stoeckicht System		
Horsepower		27000	
Input Speed - RPM		1388	
Gear Ratio		3.958	
Output Speed - RPM		351	
	SUN	PLANET	ANNULUS
Pitch Dia.-In	14.547	14.244	43.036
Number of Teeth	48	47	142
Number of Planets		5	
Normal Diametral Pitch		3.81	
Pressure Angle		22.5°	
Helix Angle		30.0°	
1/2 Effective Face Width-In.		5.906	
Face Width of Sun Pinion-In.		14.173	
K Factor, Sun/Planet		397	
K Factor, Planet/Annulus		157	
Unit Pressure, Planet Brg. PSI		610	
Planet Bearing Diameter-In.		9.25	

Table 8.6-3. Gearing Efficiencies, Updated Configuration II

OPERATIONAL MODE	INPUT POWER (HP)	POWER LOSS (HP)	EFFICIENCY	OIL REQUIREMENT (GPM)
TURBINE	27000	752	.9721	360
DIESEL	3700	236	.9360	182

1) DTE MEDIUM OIL

2) 120° F. OIL INLET TEMPERATURE

3) DESIGN W/O EXTRA IDLERS FOR ROTATION CHANGE

A list of the features designed into the propeller reduction gearing by the Cincinnati Gear Company is presented below:

a. Materials -- The use of materials of high cost, low availability, or high strategic index shall be kept to a minimum. All materials shall be corrosion resistant in a marine atmosphere or shall be suitably protected against corrosion, by lubricant, plating, or other coating.

b. Interchangeability -- Assemblies and details with the same part number shall be completely interchangeable. Gears shall not be matched to gear cases.

c. Identification and Marking -- All parts, where practical, shall be permanently and legibly marked for identification. An identification plate shall be affixed to the primary gear case having the following information:

    1. Manufacturer's name and code identification number.

    2. Manufacturer's part number.

    3. Manufacturer's serial number.

    4. Date of manufacture.

    5. Unit weight (dry), in pounds.

d. Maintainability -- The transmission shall be designed for maintenance with modest skill levels and standard tools. Removal and replacement of major components shall be simplified as far as practical.

e. Exterior Dimensions -- The exterior dimensions of the transmission, the directions of rotation, the clutch locations, the interface mounting, and the shaft coupling requirements, shall be in accordance with the RMI configuration control drawings.

f. Direction of Rotation

Turbine	*Output Shaft	Configuration
GE LM2500	CCW	II
SACM 195V20RVR	CW	II
Propeller Shaft	**	

* View direction looking toward engine from end of engine output shaft.

** Port and starboard propellers are to rotate in opposite directions for both turbine and diesel operation.

g. Modes of Operation -- The modes of operation for updated Configuration II are as follows:

Mode of Operation	Prime Mover
1 - Cruise	Diesel
2 - Dash	Turbine

Overrunning and decoupling clutches shall be provided as applicable to allow CODOG operation.

8.6.2 DESIGN PERFORMANCE REQUIREMENTS -- The design load conditions and design life of the gearbox shall be in accordance with Table 8.6-4. The design performance requirements are presented below:

a. Life -- The transmission shall have a design life of 20 years. Main power gear to be designed for hours given in Table 8.5-4 at rated power and speed. Journal bearings are used to support main power gears. Anti-frction bearings (if employed) supporting overrunning clutches or accessory gears shall have a calculated B-10 life of 10,000 hours.

Anti-friction bearings, seals, O-rings and gaskets are to be replaced at overhaul intervals of 2000 hours initially with a maturity goal of 5000 hours.

Table 8.6-4. Design Load Conditions and Life Requirements of Updated Configuration II

OPERATION	PRIME MOVER	RATED INPUT POWER AND SPEED (HP AT RPM)	GEAR DESIGN LIFE (HRS.)	PROPELLER SPEED (RPM)
1 - Cruise	Diesel	3,700 at 1,450	100,000	200
2 - Dash	Turbine	27,000 at 3,600	20,000	358

b. Materials -- The main transmission casing is to be A356-T71 cas aluminum. All castings are to be impregnated per MIL-STD-276 using impregnants per MIL-I-6869.

All aluminum components are to be hard anodized per MIL-A-8625, Type III.

All aluminum parts requiring threads shall incorporate "Keensert" brand steel inserts, 300 Series Passivated CRES.

c. Environment -- The transmission shall meet operational requirements of this specification under the following:

1. Ambient air temperature 0-140 degrees F.
2. Relative humidity to 100 percent including conditions where moisture condenses on the exterior surfaces of unit.
3. Exposure of fungi and bacterial growth as encountered in tropical regions.
4. Exposure to sea or fresh water.
5. Accelerations as follows: TBD

d. Weight -- The transmission including input and output couplings shall not exceed TBD pounds (dry).

e. Lubrication -- The transmission lubrication system shall be as follows:

1. Gearbox Driven Lubrication Pump -- The transmission shall have provisions for driving a positive displacement lubrication pump which is to provide lubricant to the transmission bearings, seals, and gears.
2. Filtration -- A duplex filter is required as part of the ship system - 10 micron nominal (25 micron absolute).
3. Sump -- The transmission case shall provide a minimum capacity of 1.5 times normal flow rate and a settling and collection point.

4. Auxiliary Electric Driven Pump -- The ship system will provide lubricant for pre-lube, post-lube, and supplemental requirements of the transmission.

5. Heat Exchanger -- The ship system will incorporate a heat exchanger for removal of heat from the lubricant.

6. Deaerator -- The ship system will incorporate a deaerator to ensure entrapped air in lubricant does not exceed 10 percent.

7. Lubricant Type -- The transmission shall be designed to operate on mineral oil for maximum corrosion protection.

f. Instrumentation -- The transmission shall contain provision for monitoring:

   1. Inlet oil temperature.

   2. Outlet oil temperature.

   3. Inlet oil pressure.

   4. Metallic debris in gearbox sump.

   5. Overall vibration level.

g. Gear Inspection -- Provisions shall be made for inspection all gear meshes without transmission disassembly by means of readily removable inspection port covers.

h. Component Retention -- All rotating components shall be secured with positive locking devices. Externally mounted fasteners are to be lock-wired.

i. Gear Design -- Life calculations are to be per applicable AGMA standards and derating factors.

8.7 WEIGHTS

Weight estimates for the lift and propulsion equipment have been made based on manufacturers' data for standard off-the-shelf machinery, responses to requests for quotation where special equipment was involved- and on designs and calculations based on the installation layouts for the MDC program and on related 3KSES activities. All components and sub-systems have been sized in accordance with the performance requirements for the 1500 ton ship for the MDC program. Equipment weights have been grouped under appropriate SWBS numbers and are listed in Table 8.7-1. The total weight of the equipment for the lift and propulsion systems is 177.0 long tons. Descriptions of the items in each SWBS 3-digit group follow.

Diesel Engines (SWBS 233)

Four engines in two sizes are used for the propulsion and lift fan drives; these are the SACM 195V12RVR's and two are SACM 195V20RVR's. The engine weights, including their respective standard accessories packages are from catalog information.

Turbines (SWBS 234)

An LM2500 gas turbine engine in production by the General Electric Company is installed in each sidehull. The exhaust collectors are included as part of the engine weights. The weights of the lube oil systems external to the engine are included in SWBS item number 262.

Gearing (SWBS 241)

There are six gearboxes in the lift and propulsion system; four are modified versions of standard parallel-shaft increase and are used in the lift system drives; the other two are special combination designs of parallel-shaft and planetary gearing and are used in the propulsion system. Weight estimates for the lift fan gearboxes are based on catalog data; the weights of the gearboxes for the propeller driven are based on preliminary design layouts developed by the Cincinnati Gear Company for the MDC program. The lube system weights for all gearing are included in SWBS item number 262.

Table 8.7-1. Propulsion and Lift Estimated Weight Breakdown - Configuration II

SWBS NO.	ITEM	WT. PER SHIP (L.T.)
233	Diesels	41.70
234	Turbines	9.30
241	Gearing	29.60
242	Clutches & Couplings	3.55
243	Shafting & Seals	13.90
244	Bearings	11.30
245	Propellers & Controls	19.65
248	Lift Fans and Ducts	17.70
251	Combustion Air System	4.46
252	Propulsion Control	0.50
259	Uptakes	12.00
262	Lube Oil System	6.34
298	Operating Fluids	7.00
200	Propulsion and Lift	177.00

Clutches and Couplings (SWBS 242)

The weight of equipment in this category comprises those components installed or mounted remotely from the engines and gearboxes. Included in this group are the flexible couplings between the propeller gearboxes and the propeller assemblies, and the overrunning clutches installed between the gas turbines and the propeller gearboxes.

Shafting and Seals (SWBS 243)

The weights in this category include only the shafting and seals that are separate from equipment such as the engines, gearboxes, fans, etc., and involve primarily the lift fan and propeller shafting and the stern seals.

Bearings (SWBS 244)

Included are the special bearing assemblies designed to support the propellers and the aft ends of the turbine engine shafts. The weights of all other bearings are included in the weights of the equipment in which they are installed.

Propellers and Controls (SWBS 245)

The weights tabulated comprise those of the controllable pitch propellers and the associated hydraulic actuators and controls, including piping and valves. The weights of the propeller shafting and thrust bearings are listed under SWBS 243 and 244, respectively.

Lift Fans (SWBS 248)

Six lift fans are installed, two forward and two in each sidehull. All fans are the same size and capacity. The fan weights include inter-connecting shafting, couplings, and the ductwork required for installation.

Combustion Air System (SWBS 251)

This equipment comprises two sets of demisters and filters, one set per sidehull, to condition the combustion air for the gas turbines. Included are the weights of the acoustic treatment and the anti-ice equipment for the duct walls.

Propulsion Control (SWBS 252)

The propulsion control equipment comprises the controls and monitoring equipment for the gas turbines and the diesel engines, but does not include the controls for the controllable pitch propellers, which are included in SWBS item number 245.

Uptakes (Swbs 259)

The weights of the uptakes include the weights of the exhaust ducting for the gas turbines and for the diesel engines.

Lube Oil Systems (SWBS 262)

The lube oil systems comprise the main pumps, auxiliary pumps, heat exchangers, filters, valves, and piping for the gas turbines and the lift fan and propulsion reduction gears. The lube systems for the diesel engines are included as part of the engine accessories packages in SWBS item number 233.

Operating Fluids (SWBS 298)

The weights of the operating fluids include the lube oil and cooling water for the turbines, diesel engines, and the lift fan and propulsion gearboxes.

9 / REFERENCES

1. 3000-Ton Surface Effect Ship Producibility Improvement Plan, Appendix A - Statement of Work of 3000-Ton Surface Effect Ship Producibility Improvement, 28 May 1981.

2. Propulsion and Lift Systems Summary Report (Preliminary), CDRL No. E06C, RMI, Inc., 1 September 1981.

3. Navy Letter Reference PMS304-20:RRB:SVS, Ser. 3046, 29 July 1981.

APPENDIX C

LIFT MACHINERY CONFIGURATIONS

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1 / TECHNICAL DESCRIPTION

1.1 GENERAL DESCRIPTION

Two basic lift system configurations were developed for the series of ship concepts covered by this study, described in the Introduction to this report. These are designated simply as Configuration I and Configuration II, where Configuration I is applicable to the smaller Cutter-type ships, and Configuration II applies to the larger Combatant and Logistic type ships. The lift system configurations are described in detail in the subsections which follow.

Both lift system configurations employ a set of lift fans in each sidehull, each set driven by a single diesel engine. The diesel engines are used for propulsion in a CODOG arrangement in the hullborne mode of operation, where lift power is not required, using appropriate gearing and clutches. It will be noted that the arrangements of both Configuration I and Configuration II differ from the Navy MDC concept in that no forward fan set is provided and airflow is routed to the bow seal by means of ducts from the fans mounted in the sidehulls. This approach was explored in that it was felt appropriate to offer an alternative with potential for reducing cost, weight, and complexity.

A lift system arrangement without the third fan set offers the following features, albeit at the expense of some reliability.

- Two fan sets already provide airflow which exceeds nominal ship requirements by approximately 80 percent.
- Deleting the third fan set results in a weight saving of approximately 12 LT per ship.
- Deleting the third fan set results in a cost saving of approximately $1 million per ship.
- The diesel engine and rotating diffuser fans are mature equipment (Reference 1, Paragraph 5.4.3.3) not requiring the added redundancy of a third set for satisfactory reliability.
- The need for long runs of services, or for additional equipment for fuel, starting air, ventilation, etc., are eliminated.
- The diesel engine of the bow fan set would not contribute to CODOG propulsion.

In Configuration I, each fan set contains two fans (a total of four on the ship). Since airflow is distributed to three compartments (bow seal, stern seal, and cushion), this arrangement requires the airflow from each fan to be split. Pressure control is accomplished by throttling the leg supplying the cushion so that the fan feels equal pressure in each leg. Configuration II employs a total of six fans per ship. In summary, advantages of the two-fan set are lower cost of fans and less machinery to align while the advantages of the three-fan set are lower risk of dynamic instability and elimination of the trim valves necessary for throttling the cushion leg.

Lift fans selected for the improved-producibility SES were the rotating diffuser type. A substantial data base for SES application was generated under Navy test programs with these fans. High efficiencies were obtained (approximately 0.85 peak). A good tolerance for downstream pressure fluctuations (as in an SES cushion) was observed, with minimal hysteresis effect on the pressure-flow characteristic. A capability for effecting ride control by using dynamically activated inlet guide vanes to attenuate downstream pressure disturbances was also demonstrated. Given the background of this data base and the short schedule for this study, it was decided to use the rotating diffuser lift fans rather than undertake a broad study of the various fan types - centrifugal, mixed-flow, and axial. The rotating diffuser fan is essentially a centrifugal type with the rotor shroud and back-plate extended radially beyond the blade tips, forming a diffusing passage which is integral with the rotating assembly. This "rotating diffuser" is responsible for some degree of efficiency gain over conventional centrifugal fans since it permits recovery of velocity head beyond the rotor blade tips prior to encountering the losses associated with discharge into the stationary housing. The rotating diffuser may also act as a "buffer" against downstream pressure fluctuations, possibly contributing to more orderly behavior of flow in the rotor blade passages under these conditions. Aerophysics Company of Washington, D.C., provided technical support in the area of rotating diffuser fans during this study in response to the Statement of Work under subcontract 3K02273.

Diesel engines were selected as prime movers for the lift fans due to their mature status as marine powerplants and their inherent fuel consumption advantage over gas turbines where space, weight and power requirements are compatible with their use. The lift diesels were sized for their alternate role as propulsion powerplants in the hullborne mode of operation. This function is accomplished through the CODOG transmission, as described in Appendix B of this report. When advanced diesel and gas turbine engines are both operated at their rated power levels, the diesel specific fuel consumption is in the order of 30 percent lower. An example is the DDA 570KF gas turbine, sfc approximately 0.49, versus the SACM 195V12, sfc about 0.35. The advantage of the diesel for hullborne propulsion is even more dramatic. For 18 knots hullborne cruise, the power requirement is 2200 horsepower. For the same two engines at this power level, the diesel again operates at an sfc of 0.35 (being near its rated power) but the gas turbine (operating at a low part power setting) exhibits an sfc of about 0.6. The diesel sfc is therefore more than 40 percent lower for this condition. In view of the advantages described, diesel engines were the choice for lift system prime movers.

1.2 CONFIGURATION I DESCRIPTION

Configuration I is integral with the machinery arrangement which incorporates 570KF gas turbine propulsion engines. The entire machinery set is shown in Figure 1.2-1 of Appendix B. The lift system portion consists essentially of the lift fans, the lift engines, the lift system gearboxes, and the lift air distribution system. It can be seen that these components are arranged in two identical sets, one in each sidehull. Two Aerophysics lift fans, in-line on a common shaft centerline, are driven by a single SACM diesel engine in each set. Air is discharged into the cushion and the bow and stern seals through a duct network which splits the flow from each fan into two legs. The cushion leg is throttled by a trim valve which balances pressures in the system. Components of the lift system are described in detail in the following sections.

1.2.1 LIFT FANS -- The lift fans are Aerophysics RD112-.65-1.3-70° DWDI units. The designation defines the major characteristics of the fan. "RD112" indicates that the fan is a rotating diffuser type (the concept described in Section 1.1) with a rotor blade tip diameter of 112 centimeters. The ".65" value is the ratio of the rotor blade tip diameter in the inducer portion to the blade discharge diameter. The next value, "1.3", defines the extent of the rotating diffuser passage, 1.3 times the rotor blade discharge diameter in this case. The "70°" figure defines the angle that the rotor blade camber line at the blade discharge makes with the tangential. The complement of this angle, in this case 20°, is the amount of "backsweep" of the blade from radial, a parameter effecting the head produced and the nature of the pressure-flow characteristics. Finally, "DWDI" signifies double-width, double-inlet, meaning the rotating assembly is two mirror-image rotors assembled back-to-back with air entering both inlet portions at opposite ends. In sizing the fans, all the parameters described must be evaluated in light of service requirements. Aerophysics Company worked closely with the Navy in the MDC design, resulting in the fan described here. As described previously, it was selected early-on for the ships in this study in the same weight range as the Navy MDC. Design speed for the RD112-.65-1.3-70°-DWDI fan is 2474 RPM, giving a blade tip speed of 476 fps. Performance will be discussed in Section 2.1 of this appendix.

The fans are equipped with variable inlet guide vanes (IGV's) which are capable of dynamically varying the fan airflow for ride control. The ride control requirements are not defined at this time, but the capability will be available when needed. The IGV's also provide a means of adjusting the fan steady state operating point. While the required cushion flow has been determined analytically, the fans are sized for much higher flows, recognizing the uncertainties in the predictions. When the ship is operated there will no doubt be a need to operate at different steady state flows and pressures. This will be accomplished by combinations of fan speed and IGV setting, as appropriate.

1.2.2 LIFT ENGINES -- The prime movers selected for Configuration I are 195V12 CZSHR marine diesel engines manufactured by SACM of Mulhouse, France. The lift system employs one of these engines per fan set or two for the ship. The engine is a turbocharged V-12 with a continuous power rating of 2400 HP at 1560 RPM, with a two-hour and peak power ratings of 2640 HP at 1610 RPM and 2880 HP at 1660 RPM, respectively. The engine was selected on the basis of its alternate role as a propulsion engine where it will be used at 2200 HP for hullborne cruising. Specific fuel consumption at this condition is 0.35 pounds/HP/hour. Operating as a lift prime mover at nominal cushion airflow and pressure, the engine will be called upon for approximately 1300 HP at 1350 RPM where the sfc is again very close to 0.35 pounds/HP/hour. The engine, therefore, offers excellent fuel consumption in both the propulsion and lift modes. Since it is working at moderate powers, long life is also expected. More than 500 of the 195 series engines are in service at this time. An overall performance plot for the SACM 195V12 CZSHR diesel engine is given in Figure 1.2-1. Note that the lines of constant specific fuel consumption are in units of grams/HP/hour.

The SACM engine is suitable for this application but competitive engines are available. An example is the 12V956TB91 diesel manufactured by MTU of Friedrichshafen, Germany, rated 2500 HP at 1455 RPM. Budgetary price was $480,000 for this engine versus $310,000 for the SACM. Weight for the MTU is 18,820 pounds compared to 13,980 pounds for the SACM. This comparison favored the SACM but a more rigorous competition should be undertaken when the ship requirements become firm. Other manufacturers, such as SEMT Pielstick of Saint Davis, France, and MAN of Nurenberg, Germany, would be candidates in such a competition.

The major drawback of the diesel engine is its high weight. For example, the weight-to-power ratio for the DDA 570 gas turbine engine is approximately 0.34 lbs/HP while the equivalent value for the SACM 12 cylinder diesel is 5.8 lbs/HP. Using Configuration I, where a 2400 HP engine is required, as an example this translates to a weight increase of

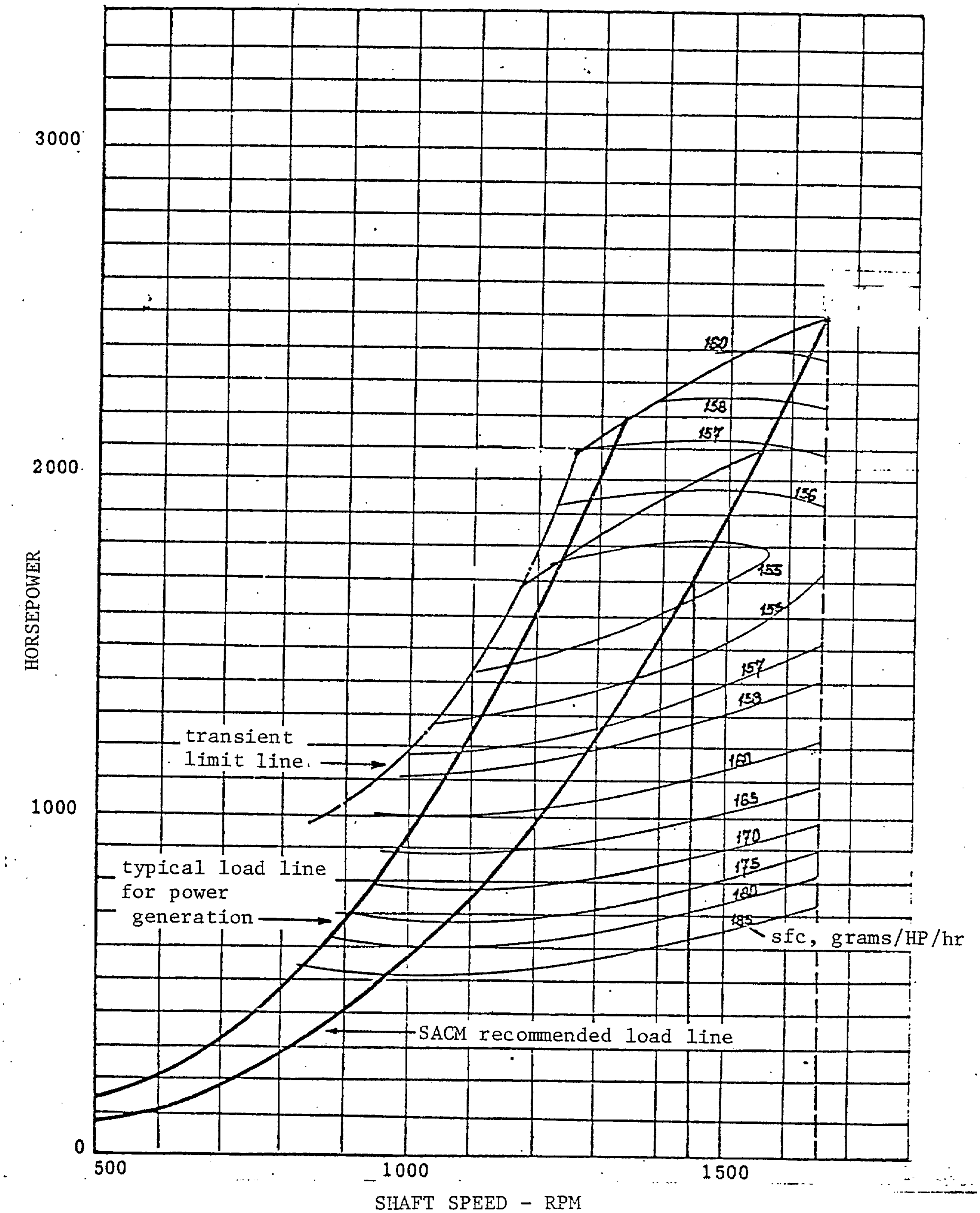


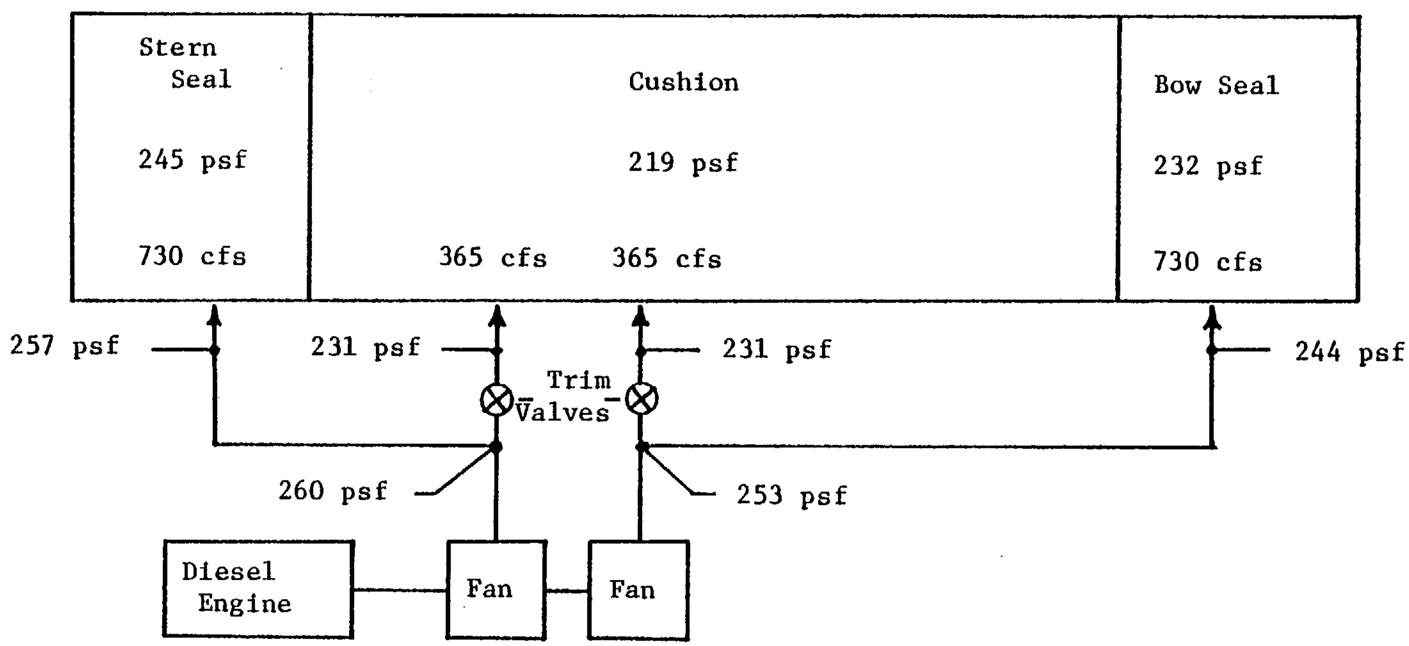
Figure 1.2-1. S.A.C.M. 195V12CZSHR Characteristic Curves

13,200 lbs. using the diesel instead of a gas turbine of the same horsepower. For the present application such a weight increase can be justified on the basis that it is offset by the lower weight of fuel which is needed for a given mission. Using the case given in the previous paragraph for operation as a propulsion engine at 2200 horsepower, the fuel consumption of the diesel is 550 lbs/hour lower than the gas turbine. The diesel would therefore make up the 13,200 lbs weight difference, in fuel saved, in 24 hours of cruising. For any mission requiring 24 or more hours of such cruising the diesel engine provides a net weight saving for the ship, in addition to continuing dollar savings for the lower fuel usage.

1.2.3 LIFT GEARBOXES -- One gearbox is employed in each fan set, or two per ship. Its function is to increase the diesel engine shaft speed of 1560 RPM to the required fan shaft speed of 2474 RPM, a step-up ratio of 1.586. This is a very simple gearbox, along the lines of commercial speed increasers, and minimum effort was devoted to it during this study. It is conceived as an offset, parallel-shaft unit probably utilizing single helical gears with an idler set to control center distance and minimize gear size. It is probable that an off-the-shelf unit would fulfill the requirements for the lift gearbox.

1.2.4 LIFT AIR DISTRIBUTION SYSTEM -- The manner in which lift air is routed to the seals and cushion is shown schematically in Figure 1.2-2. Pressure levels and airflows are given at key points in the system. Airflow from each fan is split between the cushion and one of the seals. The air which inflates the seals passes to the cushion so that all fan airflow eventually reaches the cushion. Ducts for the long runs to the bow and stern seals are 3 feet diameter which results in a flow velocity of just over 100 fps. Pressures are slightly higher in the legs which supply the seals, due to losses in the long ducts and the need to operate the seals at prescribed pressure ratios. A means of balancing the flows and pressures in the two legs is therefore provided.

Values Shown are for Ship "A"
See Table 2.2-1



Machinery and Ducting Typical
– Port and Starboard

Figure 1.2-2. Schematic Diagram for Lift Air Distribution System, Configuration I

This is achieved by a trim valve in each of the ducts supplying the cushion. This valve is envisioned as a "venetian blind" or butterfly type, and would accomplish the small degree of throttling required to balance the system.

1.3 CONFIGURATION II DESCRIPTION

Configuration II incorporates the machinery arrangement utilizing LM2500 gas turbine propulsion engines, as shown in Figure 1.3-1 of Appendix B. Aerophysics lift fans are again used but in this arrangement, there are three per fan set. A larger SACM diesel engine is the prime mover for the fans, which it drives through the lift gearbox. Each of the fans supplies a seal or the cushion separately without splitting of the air-flow. Components of this configuration are described in the following sections.

1.3.1 LIFT FANS -- The lift fans are Aerophysics RD144-.65-1.3-70°-SWSI. Differences from the fans of Configuration I are the larger diameter rotor, 144 centimeters versus 112 centimeters, and the Single-Width Single Inlet (SWSI) arrangement instead of double inlets. The fan matches the flow requirements for all the larger ships on the basis of six fans per ship, with the same substantial margin. Design speed is 1846 RPM (rotor tip speed 457 fps) and variable IGV's are again incorporated for ride control and tuning of steady state operating conditions.

1.3.2 LIFT ENGINES -- Configuration II utilizes SACM 195V20RVR diesel engines for lift power. This is a turbocharged V-20 cylinder unit with a continuous power rating of 4500 HP at 1560 RPM. Two-hour and peak power ratings are 4950 HP at 1610 RPM and 5400 HP at 1660 RPM. Specific fuel consumption is 0.34 to 0.36 pounds/HP/hour. The engine was sized for propulsion where the continuous rating is applied for hullborne cruise.

1.3.3 LIFT GEARBOXES -- The same comments as made for the Configuration I lift gearbox are applicable here but the gear ratio is slightly different. For Configuration II, the lift gearbox increases speed from 1560 RPM to 1846 RPM, a step-up ratio of 1.183.

1.3.4 LIFT AIR DISTRIBUTION SYSTEM -- Lift air distribution for Configuration II is shown schematically in Figure 1.3-1. Each fan supplies a single compartment with no splitting of flows. The forward fan of each set supplies the bow seal, center fan supplies the cushion, and rear fan supplies the stern seal. Ducts to the bow and stern seals are 4 feet in diameter, again giving a flow velocity of approximately 100 fps. Each fan has a different pressure requirement due to differences in losses and seal pressures. At this time it is anticipated that these relatively small differences (see schematic) can be accommodated through selective setting of the fan IGV's.

Values Shown are for Ship "C"

See Table 2.2-1.

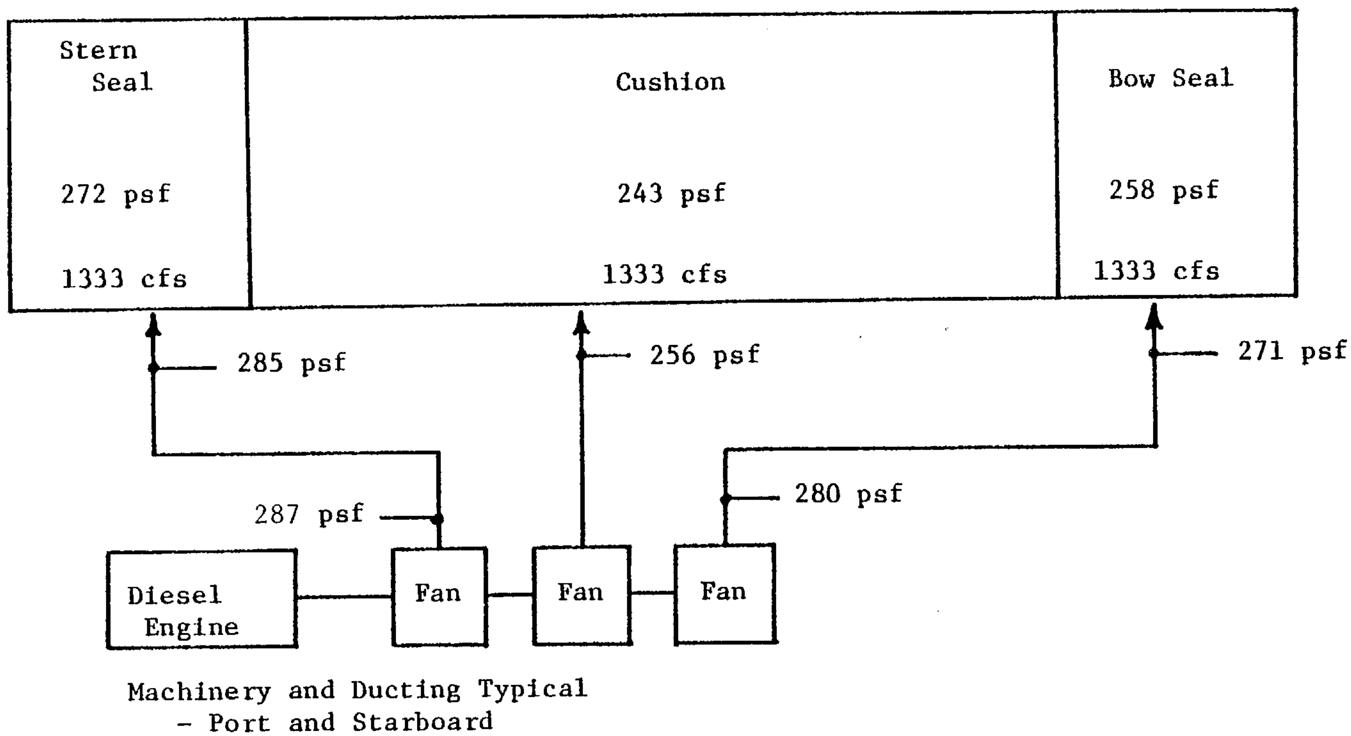


Figure 1.3-1. Schematic Diagram for Lift Air Distribution System, Configuration II

2 / PERFORMANCE AND SIZING

2.1 CONFIGURATION I

As previously described this configuration utilizes two fans per side-hull, with the airflow from each divided between the cushion and one of the seals. The general procedure for sizing the lift fans is to take the given values for airflow and pressure in the seals and cushion, and work upstream through the loss system of the air distribution network to determine required fan output. Configuration I is applicable to the 940 LT vessel (Ship A) and the 1060 LT vessel (Ship A'). The calculation procedure for Ship A will be shown here, and is representative of the calculations for Ship A' which operates at slightly different airflow and cushion pressure. For Ship A the total lift airflow is 4380 cfs and the cushion pressure is 219 psf above atmospheric. The seals are inflated by air at higher than cushion pressure, which subsequently passes to the cushion through orifices in the seals. Pressure ratios $\left(P_{seal}/P_{cushion}\right)$ in use at this writing are 1.06 for the bow seal (TSM type) and 1.12 for the stern seal. Air is routed from the fans to the seals by means of 3 foot diameter ducts. Friction losses are experienced in the ducts and sudden-expansion or "dump" losses are encountered where the airflow enters the seals. As stated, part of the airflow from each fan passes directly to the cushion. The fraction is 1/3 to the cushion and 2/3 to one of the seals. This proportioning results in equal flows to each of the seals and the cushion, 1/3 of total flow to each. The flow passing directly to the cushion does not experience the losses of the seal flow path and thus requires less pressure. A trim valve is situated in the short duct supplying the cushion which essentially throttles the flow in this leg to equalize pressures at the bifurcation where the fan flow splits. Pressures and flows throughout the system are shown in the lift air distribution schematic, Figure 1.2-2. Calculations for this sytem are as follows:

Definition of terms:

Q	=	Airflow, cfs
P_t	=	Total pressure, psf
4f	=	Friction factor
L	=	Length, ft
d	=	Diameter, ft
ρ	=	Weight density, lb/ft^3
V	=	Velocity, fps
A	=	Duct flow area, ft^2
μ	=	Air viscosity, lb/ft-sec
Re	=	Reynolds Number
q	=	Velocity head, psf
η	=	Efficiency
HP	=	Horsepower

$$Q = 4380 \text{ cfs}, \; P_{cushion} = 219 \text{ psf}$$

$$Q \text{ per fan} = \frac{4380}{4} = 1095 \text{ cfs}$$

Ship flow split is given as 1/3 bow seal, 1/3 cushion, 1/3 stern seal
Flow split for each fan is 1/3 cushion, 2/3 to one of seals

$$1/3 \times 1095 = 365 \text{ cfs (cushion)}$$

$$2/3 \times 1095 = 730 \text{ cfs (seal)}$$

Bow Seal Pressure Ratio = 1.06, giving

$$P_t \text{ bow seal} = P_t \text{ cushion} \times 1.06$$

$$= 219 \text{ psf} \times 1.06 = 232.1 \text{ psf}$$

Stern Seal Pressure Ratio = 1.12 giving

$$P_t \text{ stern seal} = P_t \text{ cushion} \times 1.12$$

$$= 219 \text{ psf} \times 1.12 = 245.3 \text{ psf}$$

Seal Duct Friction Pressure Losses

$$\Delta P \text{ bow seal duct} = 4f \frac{L}{d} \frac{\rho V^2}{2g}$$

$$\text{Air velocity, } V = \frac{Q}{A} = \frac{730}{.7854(3)^2} = 103.3 \text{ fps}$$

$$\text{Reynolds No., Re} = \frac{\rho V d}{\mu} = \frac{.0735 \times 103.3 \times 3.0}{125 \times 10^{-7}}$$

$$= 1.82 \times 10^6$$

Friction Factor, $4f = f\,(R_e) = .011$ (Ref. 2, Figure 2.2)

$$\text{Bow Duct } \Delta P = 4f \frac{L}{d} \frac{\rho V^2}{2g}$$

$$= .011 \times \frac{200}{3} \times \frac{.0735 \times (103.3)^2}{2 \times 32.17} = 8.94 \text{ psf}$$

Similarly for the stern duct

$V = 103.3$ ft/sec

$R_e = 1.82 \times 10^6$

$4f = 0.11$

$$\Delta P = .011 \times \frac{55}{3} \times \frac{.0735 \times (103.3)^2}{2 \times 32.17} = 2.46 \text{ psf}$$

Sudden Expansion Losses for Flow Entering Seals and Cushion

assume 1 velocity head (q) loss

$$q = \frac{1}{2} \frac{\rho}{g} V^2$$

$$= \frac{1}{2} \times \frac{.0735}{32.17} \times (103.3)^2 = 12.2 \text{ psf}$$

These calculations define the nominal or typical operating point for the lift fans. In the case of the forward fan the required drive horsepower is given by:

$$HP = \frac{Q \quad P_t}{550\ \eta} \quad \text{where } \eta \text{ is assumed as } 0.8$$

$$= \frac{1095 \times 260}{550 \times .8} = 629.6$$

Power required by the aft fan, calculated in the same way, is 647.0 horsepower and therefore the power requirement for the set is approximately 1277 horsepower. It will be recalled that the lift prime mover is a diesel engine developing 2400 horsepower continuous rating, sized for its alternate role of hullborne propulsion. This permits selection of fans with substantially higher capacity to cover the contingencies of ride control (if vent valves are eventually employed) or non-typical cushion operating conditions. The fan selected is the Aerophysics RD112-.65-1.3-70° DWDI, described in some detail in Section 1.2.1. The estimated performance of this fan is shown in Figure 2.1-1. The figure is a typical fan map which plots total pressure rise against airflow for a shaft speed of 2474 rpm on an 80°F day. The pressure rise characteristics are given for various inlet guide vane closure angles, up to 70° closure, as well as the wide open condition. Horsepower and efficiency curves are also included on the map for the wide open and 20° closed IGV settings. Attention is called to the operating point of 260 psf pressure rise and 1095 cfs airflow which is spotted on the map (the aft fan operating point is used as typical since the two fans operate at nearly the same point). It can be seen that the operating point is well below the capacity of this fan. In practice the fan could be slowed down to operate at the point shown. The characteristic for wide open IGV's would pass through this point at approximately 86% of rated shaft speed. Various higher speeds combined with closure of the IGV's may also be used. The performance map, then, indicates that the selected fan will meet requirements with substantial margin and good efficiency.

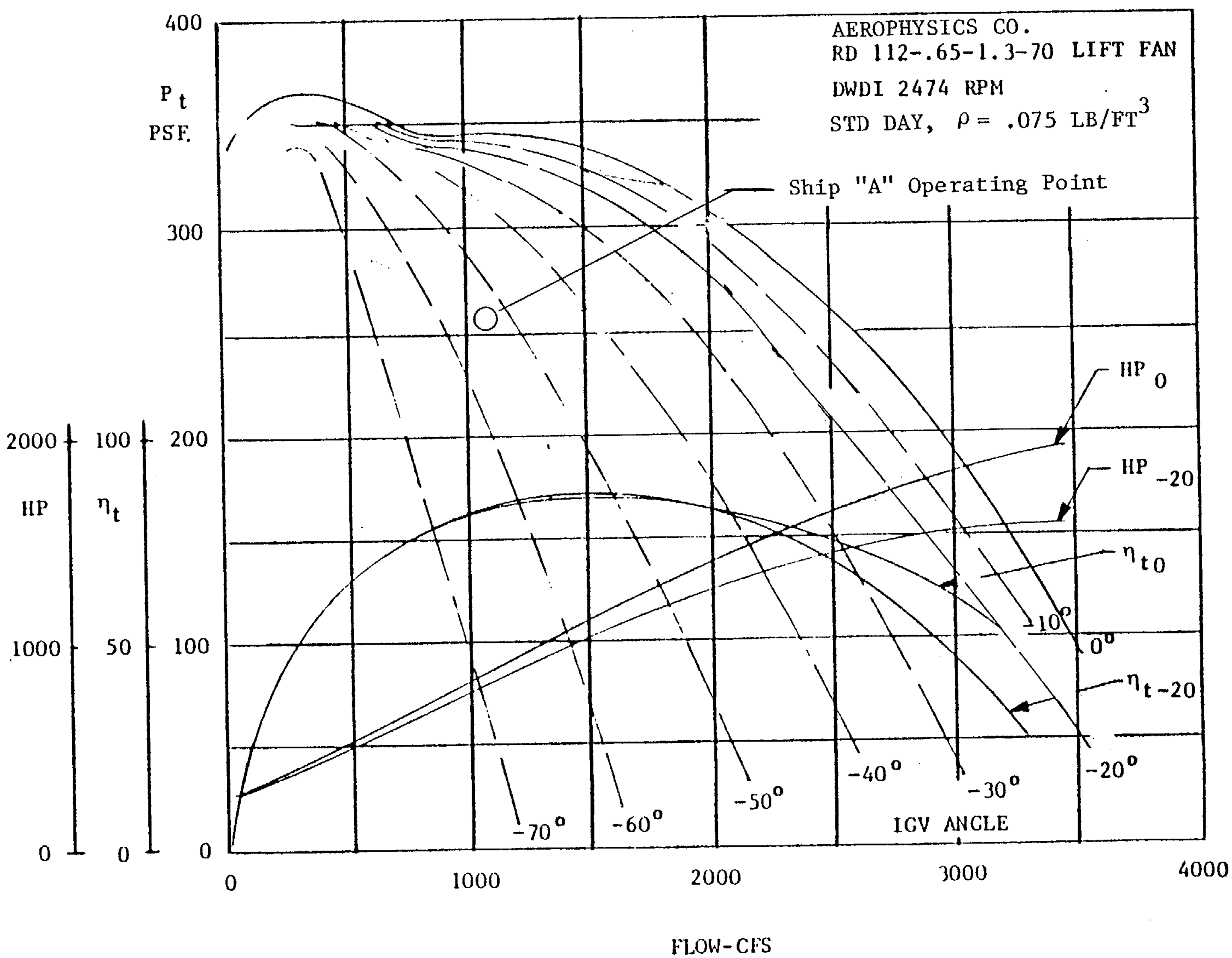


Figure 2.1-1. Configuration I Lift Fan Performance

2.2 CONFIGURATION II

This arrangement employs three fans per sidehull, with the airflow from each dedicated to the bow seal, the stern seal and the cushion. It is shown schematically in Figure 1.3-1. The fans are in-line on a common shaft centerline. The forward fan inflates the bow seal, the center fan supplies the cushion, and the aft fan inflates the stern seal. Fan Configuration II is used on Ships B, B', C, and C'. The calculation procedure for ships C and C' (the ships are identical except for cushion-borne propulsive power) will be shown in this section. The procedure is the same for the B and B' ships with different airflows and cushion pressures. Air is carried to the seals by 4 foot diameter ducts in Configuration II to accommodate the higher flows. Since fan flows do not split in this arrangement, there are no bifurcations in the discharge ducts and no trim valves are anticipated at this time. Calculations for Configuration II, as applied to Ship C, are as follows. The same definitions of terms as used for Configuration I apply.

$$Q = 8000 \text{ cfs}, \; P_{cushion} = 243 \text{ psf}$$

$$Q = \text{per fan} = \frac{8000}{6} = 1333 \text{ cfs}$$

Bow Seal Pressure Ratio = 1.06, giving

$$P_{t \text{ bow seal}} = 1.06 \times P_{t \text{ cushion}}$$

$$= 1.06 \times 243 = 257.6 \text{ psf}$$

Stern Seal Pressure Ratio = 1.12, giving

$$P_{t \text{ stern seal}} = 1.12 \times 243 = 272.2 \text{ psf}$$

Seal Duct Friction Pressure Losses

Bow Seal:

$$V = \frac{Q}{A} = \frac{1333}{.7854\ d^2} = 106.1 \text{ fps}$$

$$R_e = \frac{\rho V d}{\mu} = \frac{.0735 \times 106.1 \times 4}{125 \times 10^{-7}} = 2.49 \times 10^6$$

$4f = f\ (R_e) = .01$ - (Reference 2, Figure 2.2)

$$\Delta P = 4f\ \frac{L}{d}\ \frac{\rho V^2}{2g}$$

$$= .01 \times \frac{270}{4}\ \frac{.0735 \times (106.1)^2}{2 \times 32.17} = 8.68 \text{ psf}$$

Stern Seal:

Values of V, R_e, and 4f same as for bow seal

$$\Delta P = .01 \times \frac{50}{4} \times \frac{.0735 \times (106.1)^2}{2 \times 32.17} = 1.61 \text{ psf}$$

Sudden Expansion Losses for Flow Entering Seals and Cushion

assume 1 velocity head (q) loss

$$q = \frac{1}{2}\ \frac{\rho}{g}\ V^2$$

$$= \frac{1}{2} \times \frac{.0735}{32.17} \times (106.1)^2 = 12.9 \text{ psf}$$

Drive horsepower for the front fan is given by:

$$HP = \frac{Q\ \Delta P_t}{550 \times \eta} = \frac{1333 \times 280}{550 \times .8} = 848.3$$

Horsepower values for the center and aft fans, calculated in the same manner, are 775.6 and 869.5. Drive power required for the fan set is therefore just under 2500 horsepower. The lift prime mover for Configuration II develops 4500 horsepower continuous, again facilitating selection of fans with substantial margin. The fan selected for this application is the Aerophysics RD144-.65-1.3-70° SWSI. The estimated performance map is given in Figure 2.2-1. The operating points calculated for the three fans are shown on the map. The points again fall inside the fan wide open IGV capability. The three fans will achieve their different operating points using a reduced speed and appropriate IGV settings. At 96% of rated speed, IGV settings of approximately 5° closed, 20° closed, and 0° on the forward, center, and aft fans respectively, should produce the proper fan delivery conditions.

A tabulation has been prepared summarizing the calculations (shown here for ships A and C as examples) for all six ships in the study. Table 2.2-1 gives the lift system flows, pressures, and velocities pertinent to sizing and performance of the lift fans for the six ships.

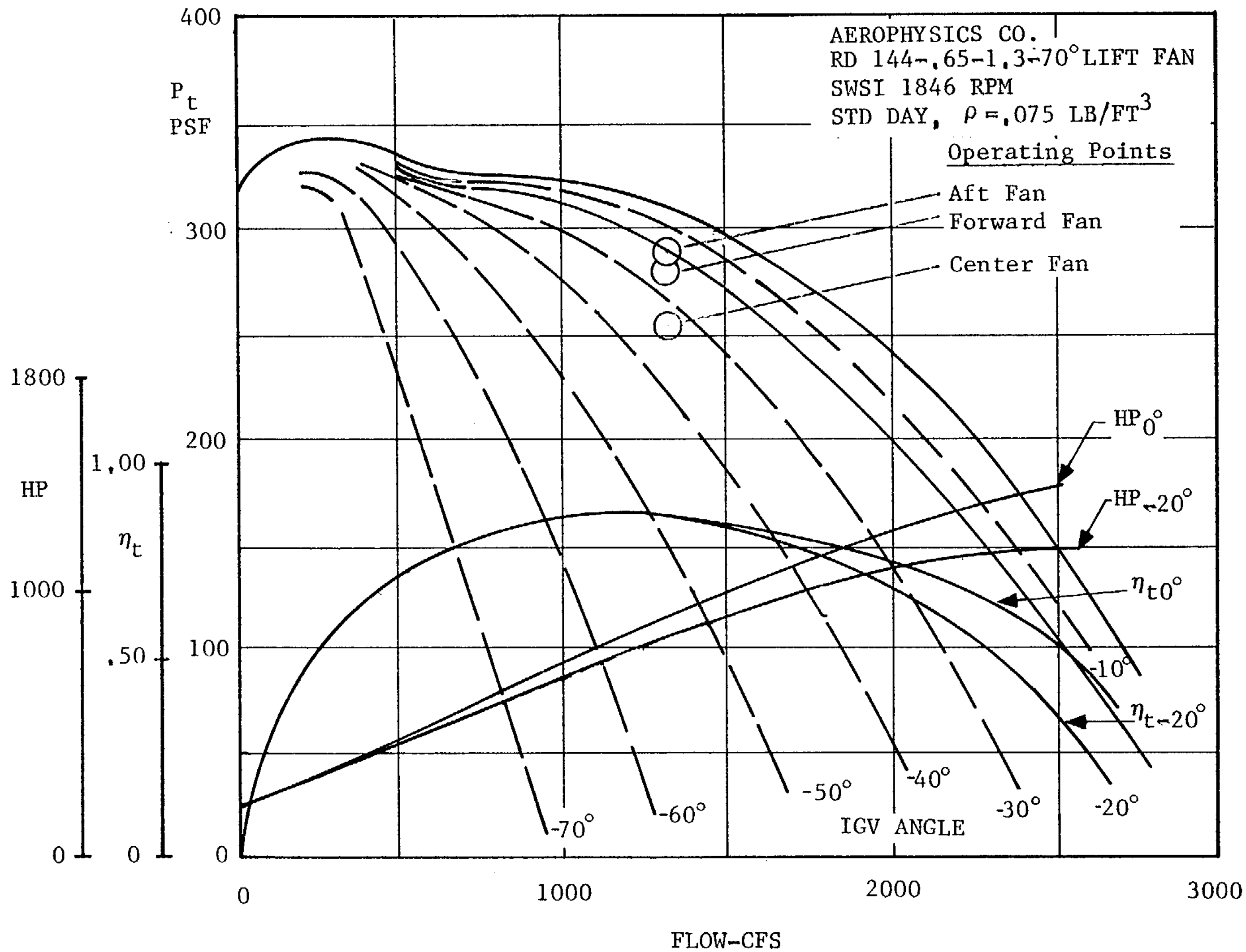


Figure 2.2-1. Configuration II Lift Fan Performance

Table 2.2-1. Summary of Pertinent Lift Air Distribution Parameters

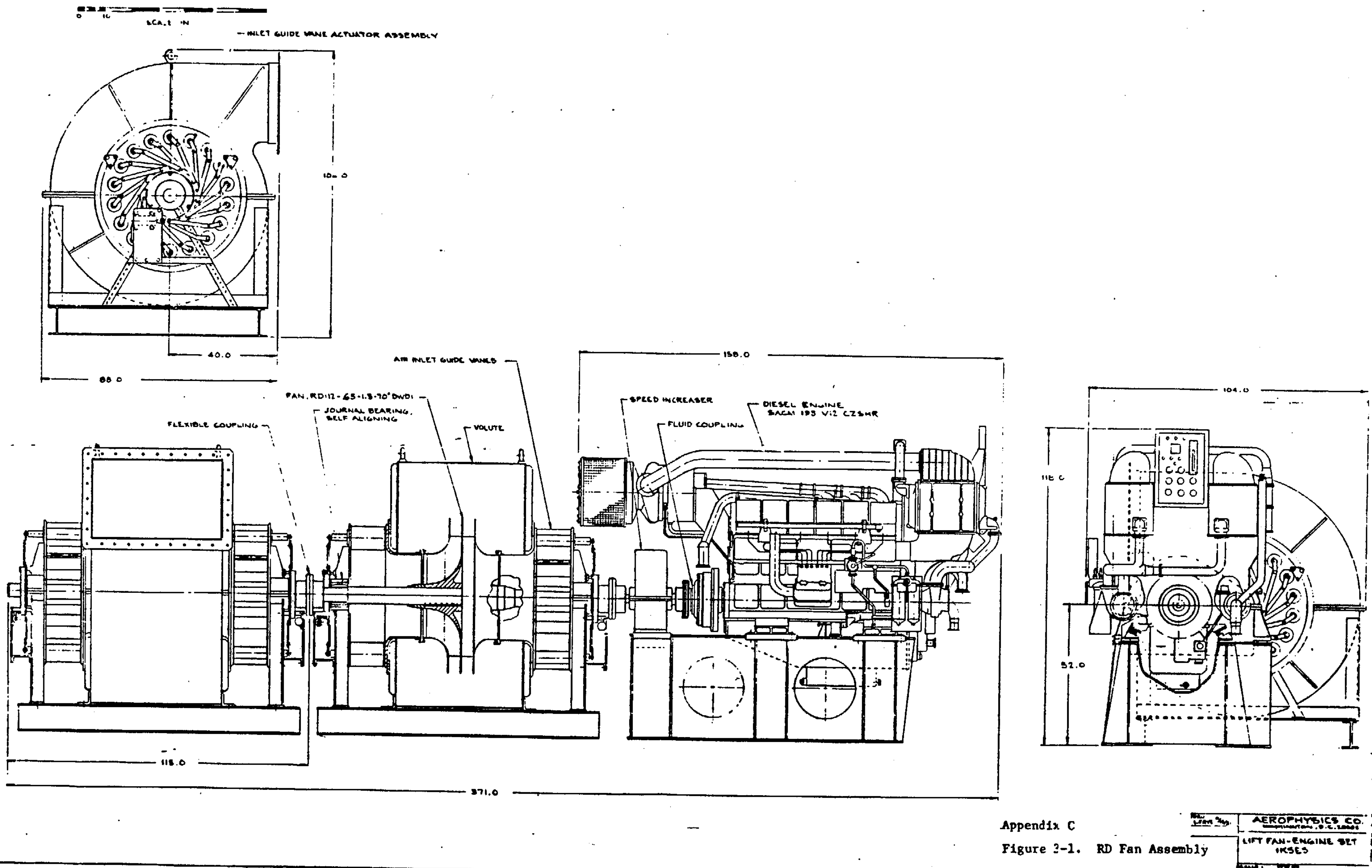
Parameter	Ship A	Ship A'	Ship B	Ship B'	Ship C, C'
Total Airflow (cfs)	4380	4650	7040	7550	8000
Total Press., Cushion (psf)	219	247	235	271	243
Airflow per Fan (cfs)	1095	1163	1173	1258	1333
Airflow, Bow Seal Duct (cfs)	730	775	1173	1258	1333
Airflow, Stern Seal Duct (cfs)	730	775	1173	1258	1333
Total Pressure, Bow Seal (psf)	232	262	249	287	258
Total Pressure, Stern Seal (psf)	245	277	263	303	272
Velocity in Seal Ducts (fps)	103	110	93	100	106
Sudden Expansion P, Seals & Cushion (psf)	12	14	10	11	13
Length, Bow Seal Duct (ft)	200	200	205	205	270
ΔP Bow Seal Duct (psf)	9	9	5	6	9
Length, Stern Seal Duct (ft)	55	55	70	70	50
ΔP Stern Seal Duct (psf)	3	3	2	2	2
Total Pressure, Forward Fan Disch. (psf)	253	285	264	304	280
Total Pressure, Center Fan Disch. (psf)	--	--	245	282	256
Total Pressure, Aft Fan Disch. (psf)	260	294	275	316	287

3 / FAN DRAWINGS

Aerophysics Company furnished several drawings of the rotating diffuser fan configuration. Figure 3-1 shows two RD112-.651.3-70° DWDI fans as a set with the SACM 195V12CASHR diesel engine prime mover. This is not the installation shown on the machinery arrangement drawing discussed in Appendix B, but the engine and fans are the same as in Configuration I used on Ships A and A'. This drawing serves to show the overall configuration of the double inlet rotating diffuser fan. Figures 3-2 and 3-3 are Aerophysics drawings of the RD112-.65-1.3-70° DWDI fan wheel assembly. These show the essential features such as the passages beyond the blade tips which comprise the rotating diffuser, the basic centrifugal blade profile, the rotor blade inlets on opposite ends of the shaft (double inlet) and the back swept rotor blade tips.

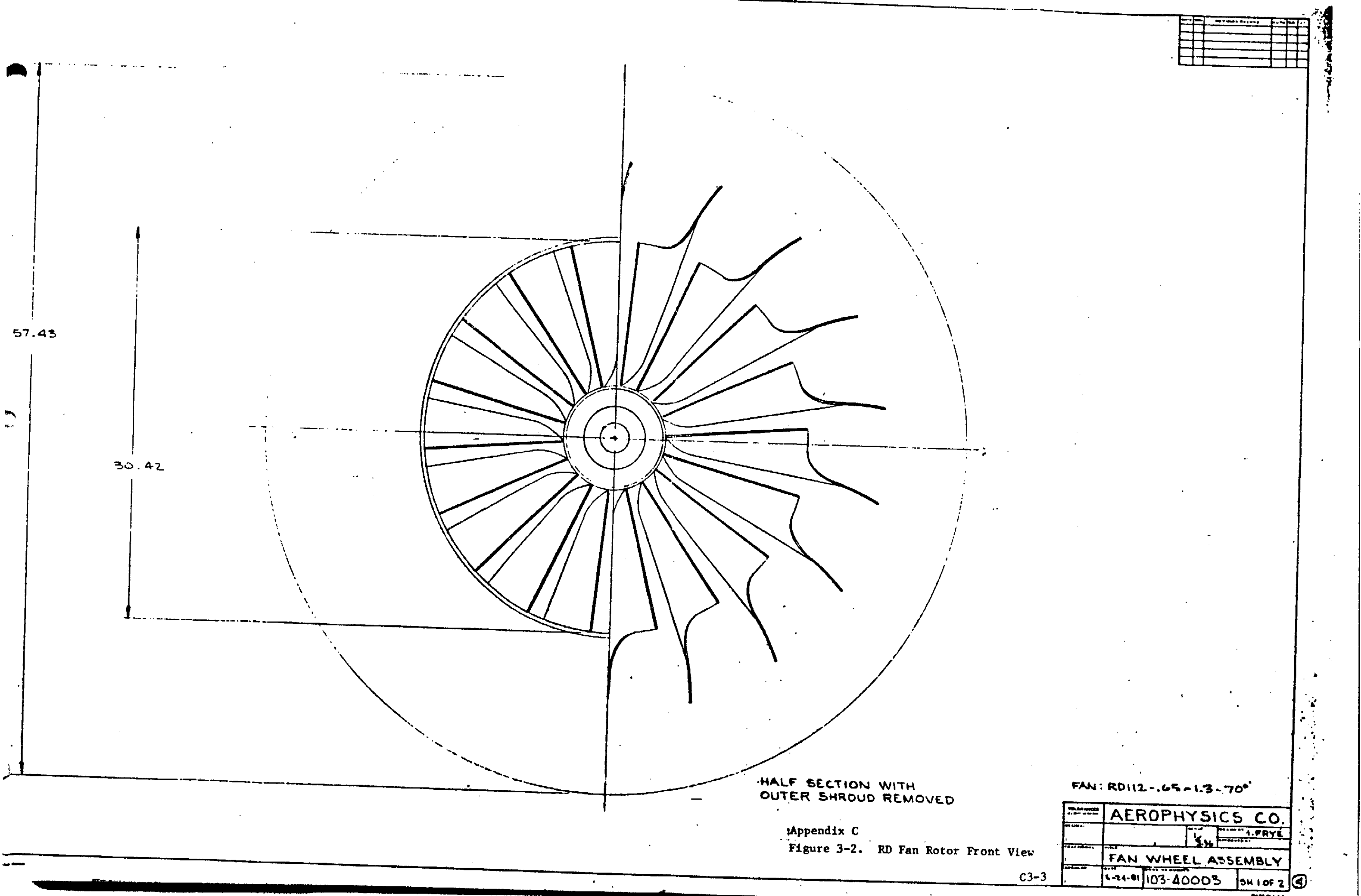
At this writing drawings of the Configuration II fans (Aerophysics RD144-.65-1.3-70° SWSI) were not available.

The drawings which show the lift fans installed in the SES ships under study are provided in Appendix B of this summary report, Figures 1.2-1 and 1.3-1.



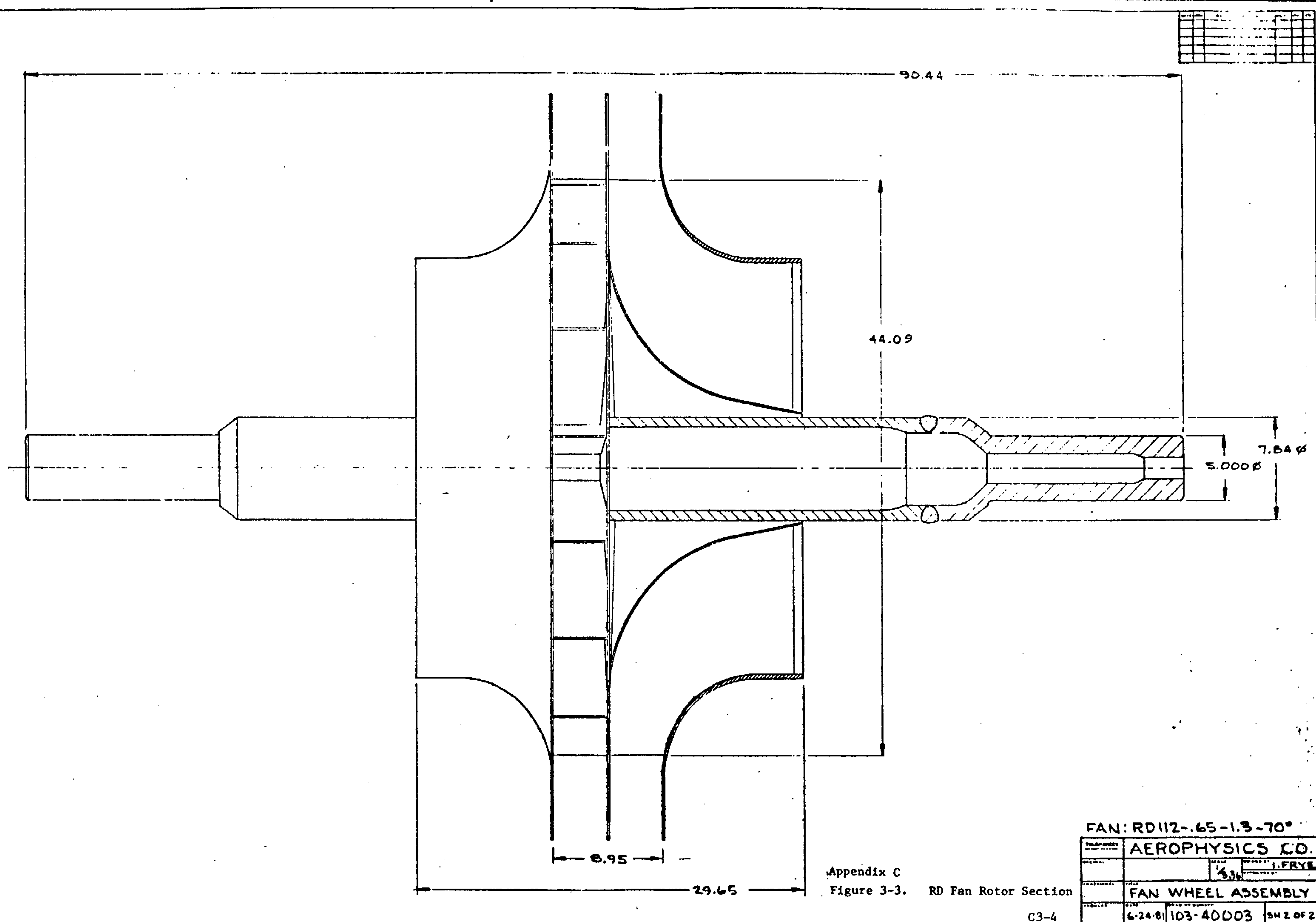
Appendix C

Figure 3-1. RD Fan Assembly



Appendix C

Figure 3-2. RD Fan Rotor Front View



Appendix C

Figure 3-3. RD Fan Rotor Section

4 / TECHNICAL RISK

4.1 SCOPE OF DISCUSSION

The lift system is comprised basically of air intakes, fans, prime movers, gearboxes, and distribution ducts. Of these, only the fans merit a discussion of technical risk. As discussed in Appendix B, the prime movers are established marine diesel engines proven in numerous applications. The gearboxes are simple offset speed increasers transmitting power levels (2400 HP and 4500 HP for the two configurations) well within the state-of-the-art. The air intakes and ducts present no technical problems (other than to require attention in the structural design where they penetrate bulkheads).

4.2 LIFT FAN TECHNICAL RISK

The rotating diffuser lift fans selected for both Configuration I and Configuration II are considered to be low technical risk due to the existence of a good data base and operation at relatively low tip speed.

A rotating diffuser fan model test conducted by the Navy at NSRDC (Reference 3) verified the steady state performance of this type of fan and established its good response to dynamic fluctuations in discharge pressure simulating wave pumping in the SES cushion. There is also a substantial background for the rotating diffuser concept in Europe where Etablissments NEU of Lille, France has been delivering fans of this type for industrial applications for many years.

Conventional centrifugal fan rotor blade profiles are utilized. Flow is axial through the inducer portion and discharges 90° to the shaft centerline. Viewed in a plane normal to the shaft centerline the blades are swept back at the rotor discharge, 20° from radial. All these features represent conventional fan design practice, well within the state-of-the-art.

The comparatively low tip speed of 476 fps results in low velocities throughout the machine. Inlet relative Mach number at the inducer tip is only 0.35 while an absolute Mach number of about 0.4 is experienced at the rotor discharge. The flow is well behaved at these low speeds and no aerodynamic surprises are expected.

The use of inlet guide vanes to adjust the operating point of turbomachines is another well established technique. It is frequently used for flow control in industrial fans and blowers. Inlet guide vanes are also used for stall alleviation in multistage axial flow compressors of gas turbine engines. The use of inlet guide vanes is thus state-of-the-art, but their application for attenuating downstream pressure disturbances by dynamically varying the airflow (as in ride control) could be considered an element of technical risk. The NSRDC test of Reference 3 provides a measure of confidence in this area. In that test program the RD fan model was discharged into a large plenum in which pressure was caused to fluctuate by oscillating the exhaust valve. Reference pressure fluctuations were measured with the IGV's fixed. Then a feedback controller was switched on and the IGV position was changed in response to command signals from the controller, which attempted to maintain a desired pressure in the plenum. The plenum pressure fluctuations were reduced approximately 50 percent in amplitude. The test demonstrated the feasibility of the approach and thus mitigates concerns about risk in this aspect of the design.

In light of the preceding discussion the rotating diffuser lift fans are considered to be a low technical risk, along with the remainder of the lift system.

5 / RELIABILITY

In both Configuration I and Configuration II there is full lift system redundancy with identical systems on port and starboard side. As described in Section 2 of this appendix, the lift engines are sized for their alternate role as off-cushion propulsion engines which provides a large surplus (in the order of 80 percent) over the nominal power required for the lift function. The lift fans are sized to take advantage of this extra power, having a much larger airflow capability than ordinarily required. In the event of a disabling failure of either the port or the starboard side lift system the remaining system can sustain the ship near its rated performance by advancing power level to that system.

In Configuration I the lift engine in each sidehull drives two lift fans on one shaft, while three lift fans are driven on one shaft in Configuration II. If one assumes that a fan failure causes the lift system in the sidehull in which it is situated to be shut down, a 50 percent higher system failure rate can be expected in Configuration II by virtue of the additional fan. This consideration must be balanced against such advantages of a three fan system as separation of flows, without splitting and throttling. Highly reliable fans would minimize the reliability disadvantage of the three-fan system.

The major lift system components are inherently reliable. The diesel engines are proven marine power plants. Lift gearboxes will be a simple straightforward design, conservatively rated. It is possible that a commercial speed increaser, with proven capability, will do the job precluding the need for a new design. The lift fans are a preliminary design at this time. The design is a conservative one, as described in Section 4, and good reliability is anticipated.

6 / MAINTAINABILITY

The key points emphasized in this conceptual phase of the design were the capability for removal and replacement of major equipment and accessibility of equipment for in-place preventive and corrective maintenance.

All major equipment will be removable vertically through the lift fan air inlets in the main deck, normally covered by a safety grating. The lift fans are situated directly below these inlets and can be hoisted directly through them. The diesel engine and lift gearbox must translate forward, to the air inlet location before being hoisted. Appropriate rails and fixtures will be provided for this maneuver to assure minimum maintenance man-hours.

The lift system is situated in a portion of the sidehull which leaves ample room around equipment. This assures access for in-place maintenance tasks. Critical area access for major machinery is generally verified by mockups as the design progresses to a more detailed phase.

7 / PRODUCIBILITY

7.1 SCOPE OF DISCUSSION

This discussion of lift system producibility is limited to the lift fans. The diesel and gas turbine engines are currently in production. The gearboxes are envisioned as straight-forward, readily producible designs, with off-the-shelf procurement a distinct possibility. The fabrication of various ductwork required for the system is also viewed as routine manufacture. The lift fans represent machinery not in current production, for which a preliminary assessment of producibility is appropriate in this report.

7.2 LIFT FAN PRODUCIBILITY

While the Aerophysics rotating diffuser fans selected for the lift system are not in production at this time, they lend themselves to very conventional fabrication methods by virtue of simple construction and shaping. The design point tip speed of 476 fps results in low stresses throughout the rotating structure. This can be seen in Figure 7-1 which presents typical stress distributions in the rotor of a rotating diffuser fan. The example is for a larger fan operating at a somewhat higher tip speed of about 527 fps. Maximum stress is approximately 15,000 psi and occurs in the backplate. Stresses in the blades and shroud both fall within a maximum of 10,000 psi. These values will be lower at the 476 fps tip speed of the selected fans. The low stress levels in the rotor permit

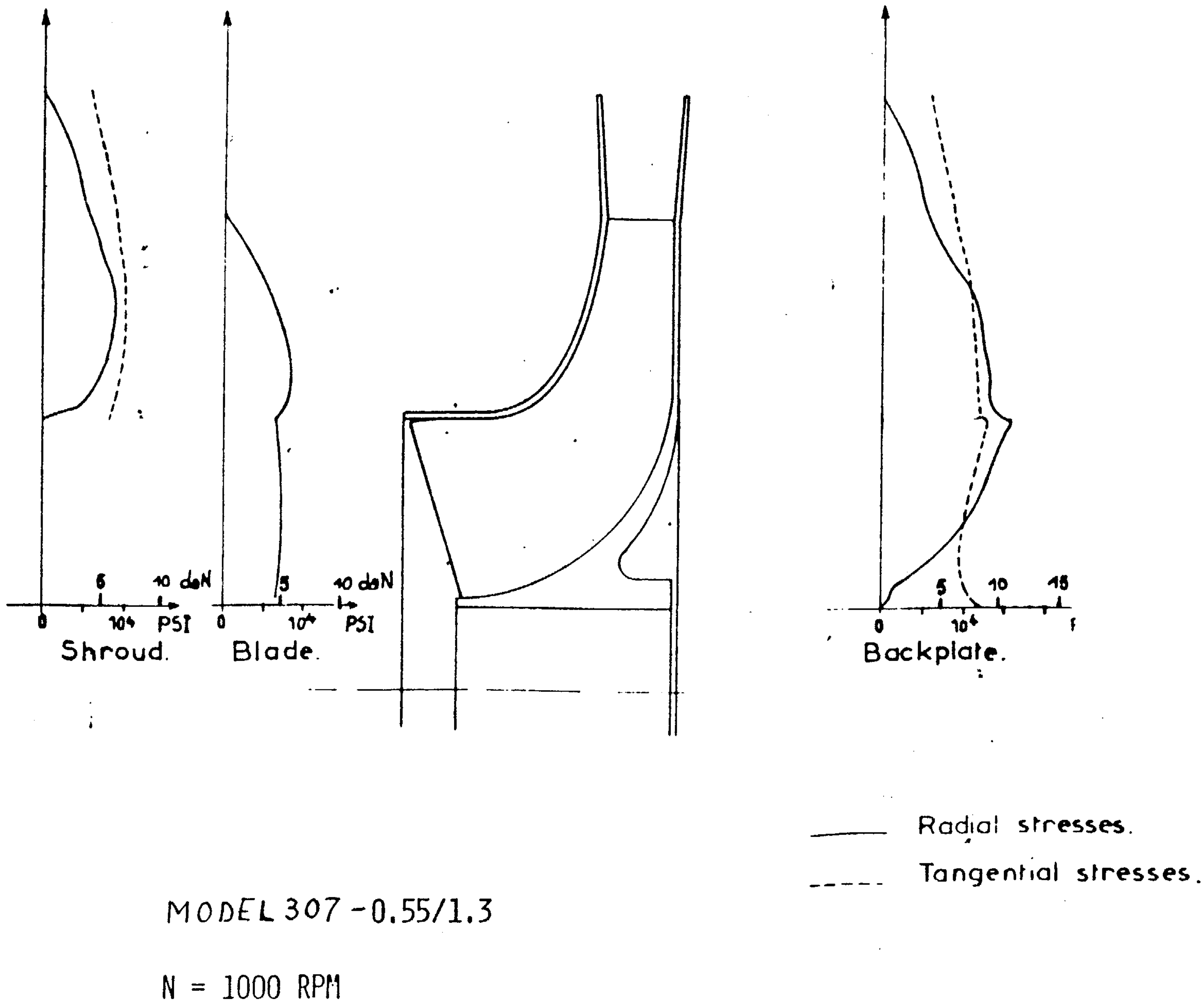


Figure 7-1. Rotating Diffuser Fan Rotor Typical Stress Distribution

the use of low cost materials with low fabrication costs. Aerophysics has selected T-1 steel for the rotating structure of the fans.

USS T-1 steel is a heat treated constructional alloy steel. It is readily available in a large variety of product forms with a yield strength of 100,000 psi and an ultimate strength of 130,000 psi. In addition to high strength and low cost other desirable properties for use in the fan wheel assembly are: weldable without the need of a post heat operation; excellent notch toughness at low temperatures; excellent fatigue life; and a resistance to atmospheric corrosion four times that of carbon steel. It is considered a low risk selection because of the wealth of test data and service experience with the alloy.

Low relative velocities in the fan rotor preclude the need for airfoil-section blades. The expensive fabrication techniques and inspection procedures associated with airfoil sections are thus avoided. The blades, instead, can be fabricated from flatplate stock and formed to the proper inlet and exit blade angles and camber, with appropriately radiused leading and trailing edges.

The fan utilizes a conventional rectangular section volute, fabricated from rolled and stiffened aluminum sheet and plate. No problems are anticipated with the volute.

Aerophysics specifies journal bearings for the lift fans at this time. While these might be preferable for certain applications, RMI feels that rolling element bearings are more appropriate for the SES lift fans. These would be grease packed and sealed, with provisions for repacking at maintenance intervals. Their use would eliminate the need for a complex lubrication/scavenge system. The lower heat rejection of the rolling element bearings precludes the need for special provisions to dissipate heat. A B_{10} life of at least 10,000 hours could be anticipated for properly sized rolling element bearings, which should be more than adequate for the present application. RMI has made this recommendation to Aerophysics.

In summary, while the selected Aerophysics fans are a preliminary design at this time, there is sufficient information available on the design to classify the lift fans as a low producibility risk.

8 / LIFT SYSTEM UPDATE

8.1 GENERAL DESCRIPTION

Engineering efforts, which initially covered a broad range of ship sizes, converged on a 1500 long ton ship with a revised set of sidehull lines.

The lift system consists essentially of the lift fans, lift prime movers, lift system geargoxes, and lift air distribution system. The arrangement is shown in Appendix B, Figures 8.3-1 and 8.3-2, drawings of the aft propulsion/lift machinery and the forward fan installation, respectively. There are three lift fans per sidehull (six per ship). Two of these are situated in the aft part of the ship, approximately at Frames 212 and 220, arranged in-line and driven by a single SACM diesel engine. Another lift fan is located near the bow, at about Frame 20, driven by a single, smaller, SACM diesel engine. All six lift fans on the ship are identical Aerophysics rotating-diffuser double-entry units. Each fan is dedicated to either the cushion or one of the seals, with no splitting of flows. The fan at Frame 20 supplies the bow seal, the fan at Frame 212 supplies the cushion, and the fan at Frame 220 supplies the stern seal. Components of the lift system are described in detail in the following paragraphs.

8.1.1 LIFT FANS -- The lift fans are Aerophysics RD154-.65-1.3-70° DWDI. These are larger units than originally selected due to increased airflow requirements resulting. Rotor diameter at the blade tips is 60.6 inches and the configuration is double-width double-inlet. All six fans on the ship are identical including the sense of rotation. The aft fan shafts are inclined 8° to the horizontal, paralleling the propulsion shafting, while the forward fans are installed with shafts horizontal. The double inlet configuration was chosen to minimize the fan height dimension which proved critical to the installation. Variable inlet guide vanes (IGV's) are incorporated in the fans. These contribute to ride control in the ship by dynamically varying airflow to the cushion to attenuate ship motions. The IGV's also provide a means of adjusting the fan steady state operating point.

8.1.2 LIFT ENGINES -- Two different size prime movers are used to power lift fans, and there are four of them per ship in contrast with the arrangement showing two per ship in the preliminary issue of this report. The reason is that independently powered fans are now provided for bow seal inflaction instead of routing airflow from the aft fan set through long trunks. SACM diesel engines have again been selected for the application. The forward fan (one per sidehull) is powered by a SACM Model 195V12RVR turbocharged 12 cylinder engine. The aft fans (two per sidehull) are powered by a SACM Model 195V20RVR turbocharged 20 cylinder diese lengine. Continuous power ratings for these engines are 2220 HP at 1450 RPM for the 195V12RVR and 3700 HP at 1450 RPM for the 195V20RVR. Both engines deliver a specific fuel consumption of 0.36 pounds/HP/hour at rated power. The 195V20RVR engines double as propulsion engines when the ship is cruising off-cushion, driving the propellers through CODOG gearboxes.

8.1.3 LIFT GEARBOXES -- Two gearboxes in each sidehull increase shaft speed from 1450 RPM at the diesel engine to the required fan speed of 1730 RPM. One such gearbox is required for the aft fan set, transmitting 3700 horsepower to the two fans installed in-line at that location.

This gearbox also provides the offset (approximately 48 inches), allowing lower fans and facilitating installation under the second deck at 24 feet ABL. The gearbox for the bow fan requires the same step-up speed ratio but is required to transmit only 2220 horsepower. It may prove appropriate for logistic reasons but at some weight penalty to make these two gearboxes identical and interchangeable in spite of the different power requirements.

8.1.4 LIFT AIR DISTRIBUTION SYSTEM -- The lift air distribution system for the 1500 long ton ship is shown schematically in Figure 8.1-1. Each lift fan supplies either the cushion or one of the seals. There is no splitting of flows nor are there long trunks for supplying the bow seal. A separate fan (in each sidehull) with its own diesel engine drive, feeds the bow seal. Redundancy is provided in the bow seal fans to preclude loss of seal inflation in the event a forward fan set malfunctions. In the aft fan set the more forward of the two fans feeds the cushion directly. Airflow from the aft fan is routed to the stern seal, which it supplies. This flow passes through the stern seal, exiting to the cushion through a short duct equipped with a valve to establish the proper stern seal pressure ratio.

8.2 PERFORMANCE AND SIZING

Sizing of the lift fans is based on the input from the ongoing tow basin model test on ship airflow requirements and the growth in displacement of the 1200 long ton platform to 1500 long tons. Fan calculations impacted by these changes are given.

Operation of the ship model in the tow basin indicated a required airflow of 13 cfs. The weight of the model is 745 pounds. Applying appropriate scaling laws to scale the model data to a 1500 long ton ship gives the following airflow for the full size ship.

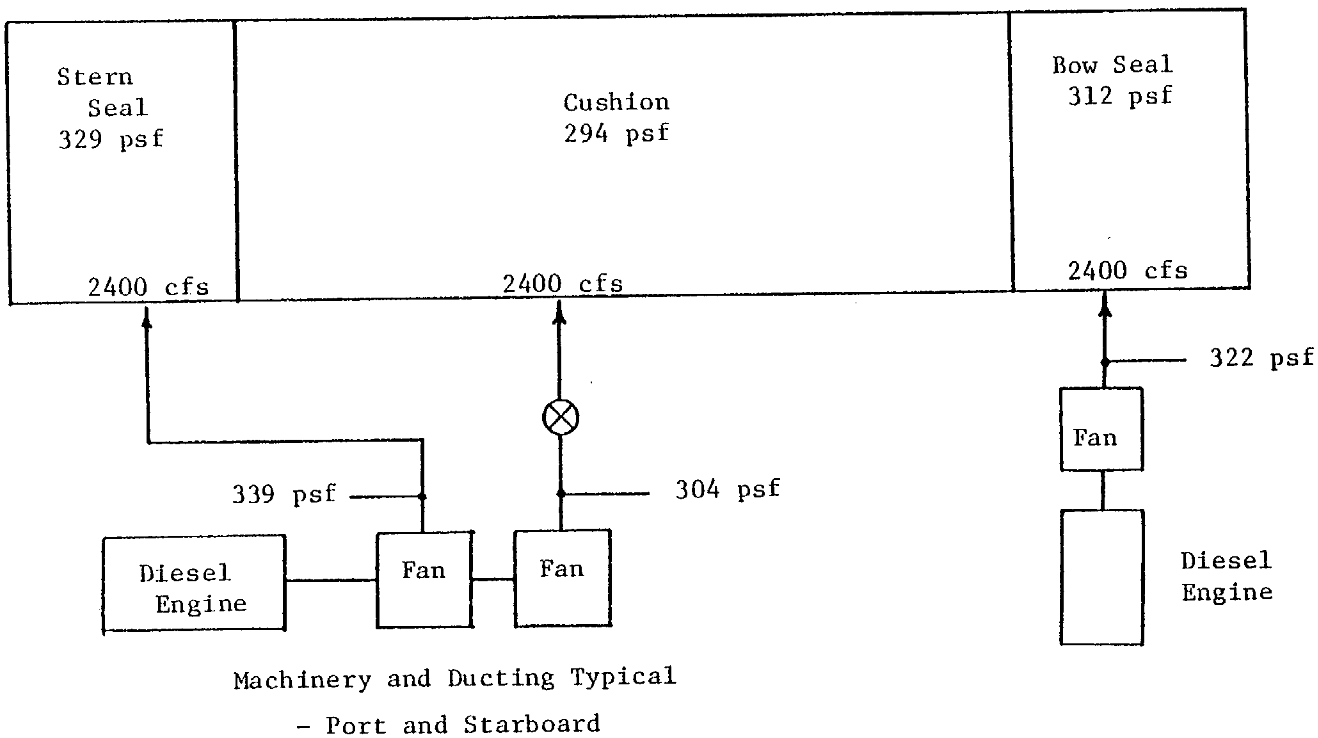


Figure 8.1-1. Schematic Diagram, 1500 L.T. Ship Lift Air Distribution System

$$Q_{ship} = Q_{model} \times \left(\frac{\text{ship weight}}{\text{model weight}}\right)^{5/6}$$

$$= 13.0 \times \left(\frac{1500 \times 2240}{745}\right)^{5/6}$$

$$= 14424.3 \text{ cfs} \quad \text{say} \quad 14400 \text{ cfs}$$

$$Q_{fan} = \frac{Q_{ship}}{6 \text{ fans}} = \frac{14400}{6} = 2400 \text{ cfs}$$

Since the cushion dimensions remain unchanged from the earlier 1200 long ton ship, the cushion pressure for the 1500 long ton version becomes:

$$P_c = P_c, 1200 \times \frac{1500}{1200}$$

$$= 235 \times \frac{1500}{1200} = 293.8 \text{ psf}$$

Given the revised values of airflow and cushion pressure, it is possible to establish a scale factor K_s for sizing a new fan of exact geometric similarity to a fan for which the predicted performance has been established. Scaling up to a geometrically similar fan will preserve the efficiency and general performance characteristics of the basic fan.

Aerophysics Company advised that their approach to determining the appropriate scale factor is embodied in the following expression:

$$K_s = \sqrt{\frac{Q_2}{Q_1}} \times \left(\frac{H_1}{H_2}\right)^{-.25}$$

where: Q_1 = Airflow of original fan

Q_2 = Airflow of scaled fan

H_1 = Pressure rise of original fan

H_2 = Pressure rise of scaled fan

For this computation an RD 125 - .65 - 1.3 - 70 DWDI fan, sized for the 1200 long ton version of the SES by Aerophysics Company, is scaled to the new parameters as follows:

Q_1 = 1760 cfs

Q_2 = 2400 cfs

H_1 = 235 psf

H_2 - 294 psf

$$K_s = \sqrt{\frac{2400}{1760}} \times \left(\frac{235}{294}\right)^{-.25} = 1.235$$

This multiplier is applied to all fan dimensions and the rotor diameter of 125 centimeters therefore scales to:

D_2 = 125 x 1.235 = 154.3 say 154 cm

The scaled fan designation is therefore

RD 154 - .65 -1.3 - 70 DWDI.

As a result of the changes the power requirement for driving the fans increases and is now input with two engines per sidehull, one forward and one aft. Power required is determined as follows:

Pressure rise for the three fans computed by the method of section 2.1 are:

Bow Seal Fan	ΔP	=	322 psf
Cushion Fan	ΔP	=	304 psf
Stern Seal Fan	ΔP	=	339 psf

Horsepower for each fan is calculated using the pressure rise given above, an airflow of 2400 cfs, and an efficiency of 0.78 in the following expression:

$$HP \doteq \frac{\Delta P\ Q}{550\eta}$$

Results are as follows:

Bow Seal Fan Power	=	1801 horsepower
Cushion Fan Power	=	1701 horsepower
Stern Seal Fan Power	=	1897 horsepower

The forward diesel engine in each sidehull drives only the bow-seal fan thus requires 1801 horsepower. For this application, a SACM Model 195 V12RVR diesel engine developing 2220 continuous horsepower is selected. The aft diesel engine drives the cushion and aft seal fans which require a total of 3598 horsepower. For this application, a SACM Model 195V20RVR diesel engine developing 3700 continuous horsepower is selected.

9 / REFERENCES

1. "Medium Displacement Combatant Surface Effect Ship (MDC)" Technical Report, Surface Effect Ship Acquisition Project - Naval Sea Systems Command, April 1981.

2. "Principles of Turbomachinery", D.G. Shepherd, Macmillan Company, New York, 1956.

3. "Experimental Evaluation of a Rotating Diffuser Fan Model", John M. Durkin, Navy Report Number DTNSRDC/ASED - 80/17, August 1980.