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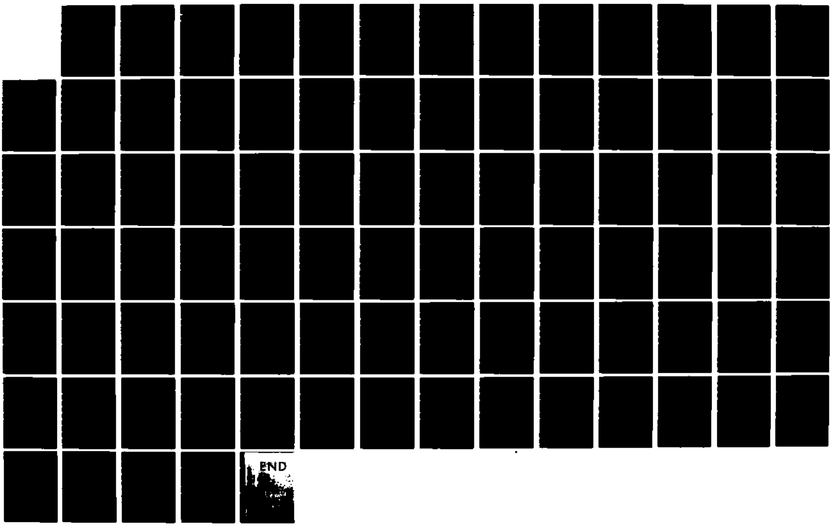

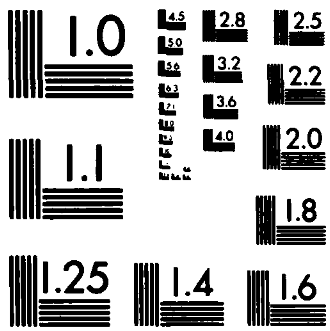
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REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER	2. GOVT ACCESSION NO.
3. RECIPIENT'S CATALOG NUMBER	
4. TITLE (and Subtitle) AN ALTERNATIVE PROPULISION PLANT FOR NAVAL AUXILIARY SHIPS	5. TYPE OF REPORT & PERIOD COVERED THESIS
	6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) STONE, EDWARD LAWRENCE	8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS MASSACHUSETTS INSTITUTE OF TECHNOLOGY CAMBRIDGE, MA.	10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS CODE 031 NAVAL POSTGRADUATE SCHOOL MONTEREY, CA. 93940	12. REPORT DATE JUNE 1980
	13. NUMBER OF PAGES 84
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)	15. SECURITY CLASS. (of this report) UNCLASS
	15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE; DISTRIBUTION UNLIMITED

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

ALTERNATIVE PROPULSION
AUXILIARY SHIP PROPULSION

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

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ABSTRACT

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The results of this analysis show that it is feasible to convert the AO-177 class fleet oiler to a diesel powered vessel provided that the conversion is performed prior to the end of the ship's third year of life. Two candidate diesel plants were studied with the most economical plant being selected for use in the conversion.

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AN ALTERNATIVE PROPULSION PLANT FOR NAVAL AUXILIARY SHIPS

by

Edward Lawrence Stone

B.S., Mechanical Engineering
Purdue University (1973)

SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS OF THE
DEGREES OF

MASTER OF SCIENCE IN
NAVAL ARCHITECTURE AND MARINE ENGINEERING

and

OCEAN ENGINEER

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

June 1980

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May 9, 1980

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AN ALTERNATIVE PROPULSION PLANT FOR NAVAL AUXILIARY SHIPS

by

EDWARD LAWRENCE STONE

Submitted to the Department of Ocean Engineering on May 9, 1980, in partial fulfillment of the requirements for the Degrees of Master of Science in Naval Architecture and Marine Engineering and Ocean Engineer.

ABSTRACT

Due to the rising fuel cost, marine engineers and ship operators alike are searching for more fuel efficient propulsion plants to power ocean-going vessels. This thesis describes an alternative propulsion plant for the U.S. Navy's newest fleet oiler, the AO-177, in order to solve the problem of rising fuel costs. An analysis of converting the present geared steam turbine propulsion plant is performed, and the economic benefits of such a conversion are examined.

The results of this analysis show that it is feasible to convert the AO-177 class fleet oiler to a diesel powered vessel provided that the conversion is performed prior to the end of the ship's third year of life. Two candidate diesel plants were studied with the most economical plant being selected for use in the conversion.

Thesis Supervisor: Dr. A. Douglas Carmichael

Title: Professor of Power Engineering

ACKNOWLEDGEMENTS

The author wishes to acknowledge the contributions made to the content and scope of this thesis by Professor A. D. Carmichael. A special thanks goes to Ms. Gwen Johnson for her efforts in editing and typing the final copy.

To my wife and children goes the greatest measure of thanks for the love and understanding they have provided during these past three years.

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NOMENCLATURE

BHP	brake horsepower
CRP	controllable reversible pitch
EAR	expanded area ratio
EHP	effective horsepower
J	advance ratio
K_t	thrust coefficient
P/D	propeller pitch to diameter ratio
PC	propulsive coefficient
q_t	dynamic pressure
R	resistance
RPM	revolutions per minute
SFC	specific fuel consumption
SHP	shaft horsepower
T	thrust
t	thrust deduction factor
V	ship's speed
V_a	speed of advance
w	wake fraction
η_H	hull efficiency
η_O	open water propeller efficiency
η_R	relative rotative efficiency
ρ	dens'ty of salt water

NOMENCLATURE (Continued)

σ	local cavitation number
x	model correlation factor

CHAPTER I

INTRODUCTION

1.1 Background

The oil crisis of 1973 and the subsequent dramatic increase in world oil prices has driven marine engineers to search for more fuel efficient propulsion systems to power ocean-going vessels. In the United States, this has meant a growing interest in the use of diesel engine propulsion plants. There are increasing numbers of commercial ship owners selecting diesel engine propulsion plants to power their vessels. In some cases, the ship owners are converting their less efficient steam powered vessels to diesel propulsion.

Diesel propulsion has found wide acceptance, for many years, with foreign ship owners for use in most, if not all of their ocean-going fleets and as a result, they have developed a good reputation. The diesel engines used for high horsepower marine applications fall into two general classes: the medium speed engine with horsepower in the 5,000 to 12,000 BHP range operating at 400 to 500 RPM; and the low speed engine with up to 4,000 BHP per cylinder operating at 110 to 130 RPM. These engines have demonstrated excellent fuel economy and good reliability. The diesel engine provides the additional advantage that it lends itself readily to automated operation, thus allowing a reduction in crew size.

In light of the world oil situation and the necessity to conserve fuel, the U.S. Navy is facing the same problems as the commercial ship owners. In the past the principle main propulsion unit for combatant and non-combatant vessels in the U.S. Navy has been the geared steam turbine. It is a highly reliable, proven source of power whose selection was more than justified at the time and given the circumstances.

For many years the U.S. Navy has rejected the diesel engine for use as a main propulsion unit because of its high self-generated noise levels, poor slow speed operation, and high specific machinery weight. In addition there were no major U.S. diesel engine manufacturers capable of producing a reliable engine that could provide the horsepower required in U.S. naval ships. Thus, naval application of the diesel engine was limited to small craft, patrol boats, and a few auxiliary ships.

In light of the recent developments and improvements made in modern medium speed engines, it would appear that these engines could find wide application aboard U.S. naval ships. Obviously, not every ship in the Navy's fleet would be a candidate for diesel propulsion; but for ships in which economy of operation overrides operational considerations (such as noise reduction and machinery specific weight) diesel engines would present a very attractive alternative. Specifically, ships that provide logistic support to the combat units of the fleet would appear to be prime candidates for this type of propulsion plant.

Logistic support ships such as oilers, stores ships, and ammunition ships are required to transit long distances, at their most economical speed, to rendevous with the combat units. Upon completion of the replenishment of the combat units, they must return to their supply bases to repeat the cycle once again. In order to do this economically, the ships should be relatively inexpensive to build, operate, and maintain.

It is the intent of this thesis to investigate the feasibility of converting the steam propulsion plant of the U.S. Navy's newest fleet oiler to a medium speed diesel engine propulsion plant in order to improve its overall cost of operation and maintenance. The following factors will be considered in this study:

... compatibility of the propulsion plant with the entire ship system;

... technical risk;

... annual operating costs;

... acquisition costs; and

... manning.

The engines selected for consideration in this study will be such that there is no degradation of the original performance requirements of the existing ship. In fact it is anticipated that significant improvements will be realized in ship performance by installing a diesel main propulsion plant.

This study will begin by establishing what the propulsion requirements for the AO-177 are and how they are currently met. Several candidate medium speed engine arrangements will then be selected to meet the AO-177's propulsion requirements. The candidate machinery plants will then be compared with each other and the steam plant in terms of the following factors:

... acquisition costs;

... annual fuel costs;

... manning costs; and

... maintenance costs;

with the best plant being selected to power the AO-177.

1.2 AO-177 Description

The AO-177 is the U.S. Navy's new fleet oiler design. The mission of the AO-177 is to transport and deliver petroleum products to the operating forces of the U.S. Navy at sea. The principle characteristics of the ship are as follows:

Length overall	590 ft
Length between perpendiculars	550 ft
Beam	88 ft
Draft	33.5 ft
Full load displacement	27,000 tons
Cargo capacity	120,000 tons
Clean ballast capacity	8,000 tons
Type of propulsion	geared steam turbine

The ship was designed with two principle directives in mind:

1. Design to cost.
2. Design for reduced manning.

These two directives meant that all design elements had to be carefully examined from a cost effective point of view and the ship had to be designed with a high degree of machinery automation and centralized control.

The propulsion plant of the AO-177 is presently a 600 psi/ 850°F geared steam turbine capable of developing 24,000 shaft horsepower at 100 RPM for full power operation. The single cross compounded steam turbine drives a 21 foot fixed pitch propeller through a double reduction, double helical articulated reduction gear.

The boilers are top fired, natural circulation, watertube boilers fitted with automatic combustion controls which allow for unattended fireroom operation.

The ship's electrical plant consists of three steam driven 450 volt, 3 phase, 60 Hz generators each rated at 2,500 kw, and one emergency diesel generator rated at 750 kw. Two ship's service generators will be capable of providing the maximum electrical load, which will occur during underway replenishment.

The auxiliary machinery plant consists of two 12,000 gallons per day distilling plants, two one ton refrigeration plants, and two 75 ton air conditioning plants.

The machinery plant is based on a two compartment standard because of damage control requirements. An enclosed operating station is provided for centralized control of all machinery during operation. A watertight door in the engine room/fireroom bulkhead provides access to the firing aisle in the fireroom from the engine room.

The endurance range has been established at 6,000 nautical miles at 20 knots. At the design endurance speed of 20 knots, the main engine will develop 80% of the installed horsepower. There were no maximum speed requirements established for this ship.

CHAPTER II

DIESEL ENGINE SELECTION

2.1 Introduction

The selection of a particular diesel engine for use in the AO-177 was based on the following factors:

... horsepower developed;

... overall engine length; and

... manufacturer.

Diesel engines are manufactured for discrete horsepowers; therefore, the engines selected must closely match the required installed horsepower in order that the penalty for purchasing more or less horsepower than required is reduced or eliminated.

The current machinery space lengths will dictate which engines are selected as the engines must fit into the space provided on the ship without affecting U.S. Navy damage control and damage stability requirements. In addition, engine sizes will have a direct impact upon machinery arrangements.

Due to the political climate and government regulations concerning the purchase of machinery for government-owned vessels, only United States manufacturers of medium speed diesel engines will be considered.

Therefore, the engines to be considered are the Enterprise engines manufactured by Delaval Turbine, Inc., and the Pielstick engines manufactured by Colt Industries under license. Both these engines have seen application in ocean-going vessels and have proven their reliability in at-sea operation both in U.S. and foreign ships.

Figure 2.1 shows the basic options considered for use in the AO-177.

The electric drive option was rejected in the very early stages of this study as it was discovered that an electric motor capable of developing 24,000 SHP was not available. Such a main propulsion motor could be manufactured on special order, but its size and weight would be so large as to make it very unattractive for this application. Therefore, a geared driven propulsion system was selected.

For the gear driven plant, two basic options were selected - one with a shaft driven ship's service generator and one without. Table 2.1 shows the candidate engines selected for consideration for these two options. A controllable reversible pitch propeller was selected for the reasons outlined in Chapter III entitled "Propeller Selection".

2.2 Candidate Diesel Plants

The arrangement selected without the shaft generator consists of two Enterprise RV-20-4 engines driving a single reduction gear and developing 24,008 shaft horsepower. Ship's service electrical power is provided by three Fairbanks Morse 12 cylinder 38 D 8 - 1/8 series diesel driven 450 volt, 3 phase, 60 Hz generators rated at 2,500 kw. The auxiliary plant will be identical to that presently installed in the AO-177

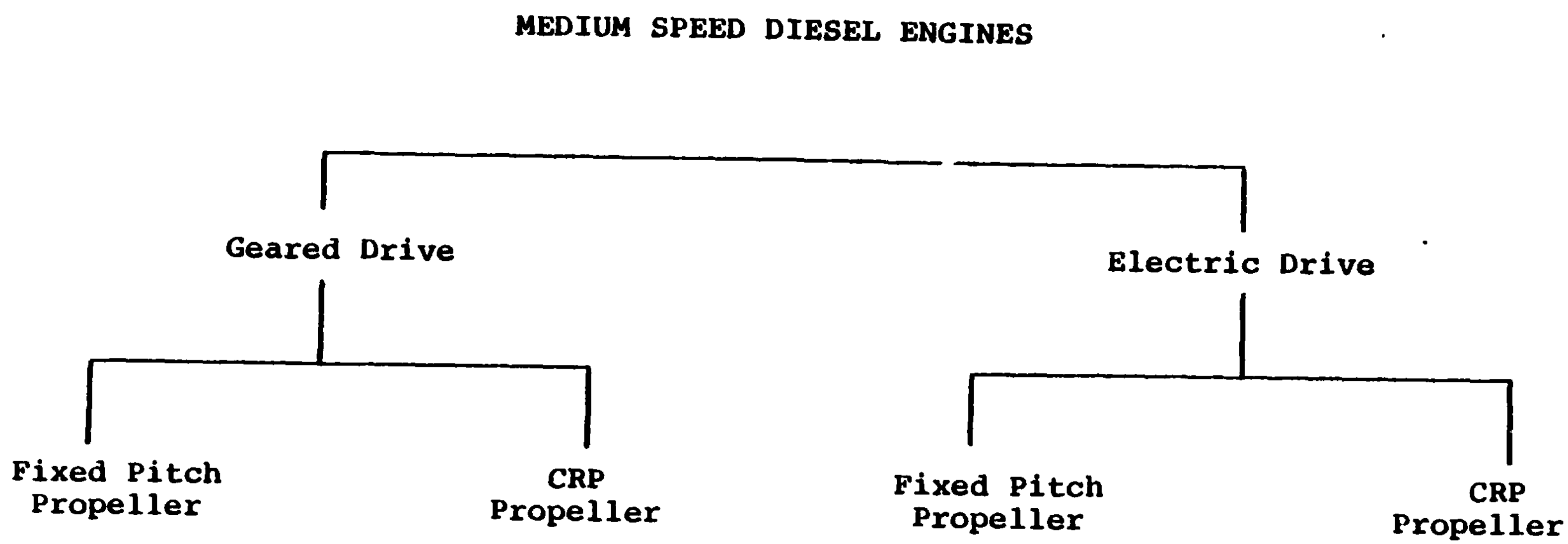

MEDIUM SPEED DIESEL ENGINES
Geared Drive
Electric Drive
Fixed Pitch Propeller
CRP Propeller
Fixed Pitch Propeller
CRP Propeller

FIGURE 2.1

ENGINE SELECTION

	With Shaft Generator	Without Shaft Generator
	Required SHP = 27,353 Assumed Gear Eff = 0.985 Installed BHP = 27,770	Required SHP = 24,000 Assumed Gear Eff = 0.985 Installed BHP = 24,365
Colt-Pillstick Engines	3-16 PC 2.5v engines Tot. BHP = 28,080 @ 502 RPM L=30.5' H=11.5' W=11.5' Wt = 71.38 tons/engine SFC = 0.371 lb/bhp-hr L.O Consumption = 5,000 hp-hr/gal	3-14 PC 2.5 v engines Tot. BHP = 24,570 @ 502 RPM L=28.0' H=11.5' W=11.5' Wt = 64.77 tons/engine SFC = 0.371 lb/bhp-hr L.O Consumption = 5,000 hp-hr/gal
Enterprise Engines	3-RV-16-4 engines Tot. BHP = 29,253 @ 450 RPM L=27.0' H=13.6' W=11.1' Wt = 104.9 tons/engine SFC = 0.370 lb/bhp-hr L.O Consumption = 1 gal/15,000 hp-hr	2-RV-20-4 engines Tot. BHP = 24,374 @ 450 RPM L=30.1' H=13.6' W=11.1' Wt = 126.9 tons/engine SFC = 0.370 lb/bhp-hr L.O Consumption = 1 gal/15,000 hp-hr

TABLE 2.1

oiler with the addition of two waste heat recovery boilers fitted on the main engine exhaust and one auxiliary, separately fired boiler to provide the ship's steam requirements.

The arrangement selected for use with a shaft driven generator consists of three Colt-Pielstick 16 PC 2.5v engines driving a single reduction gear and developing 27,659 shaft horsepower. Ship's service electrical power is provided by one shaft driven 450 volt, 3 phase, 60 Hz generator rated at 2,500 kw and two Fairbanks-Morse 12 cylinder 38 D8-1/8 series diesel driven 450 volt, 3 phase, 60 Hz generators rated at 2,500 kw. The same auxiliary plant would be utilized as previously described.

These two particular arrangements were selected based on the criteria outlined in section 2.1. They both provided an excellent match in horsepower, and the engine lengths are compatible with the machinery space provided in the existing hull. The machinery arrangements for each configuration are shown in Figures 2.2 through 2.9.

It should be noted at this point that in order to arrange the machinery for the shaft driven generator configuration, the watertight bulkhead at frame 48 had to be moved aft to frame 40. In addition the centerline fuel oil storage tank 4-70-0 was reduced to one half of its original size and the watertight bulkhead at frame 69 was extended up to the first platform deck. This necessitated the enlargement of the fuel

INBOARD PROFILE - TWIN ENGINE PLANT

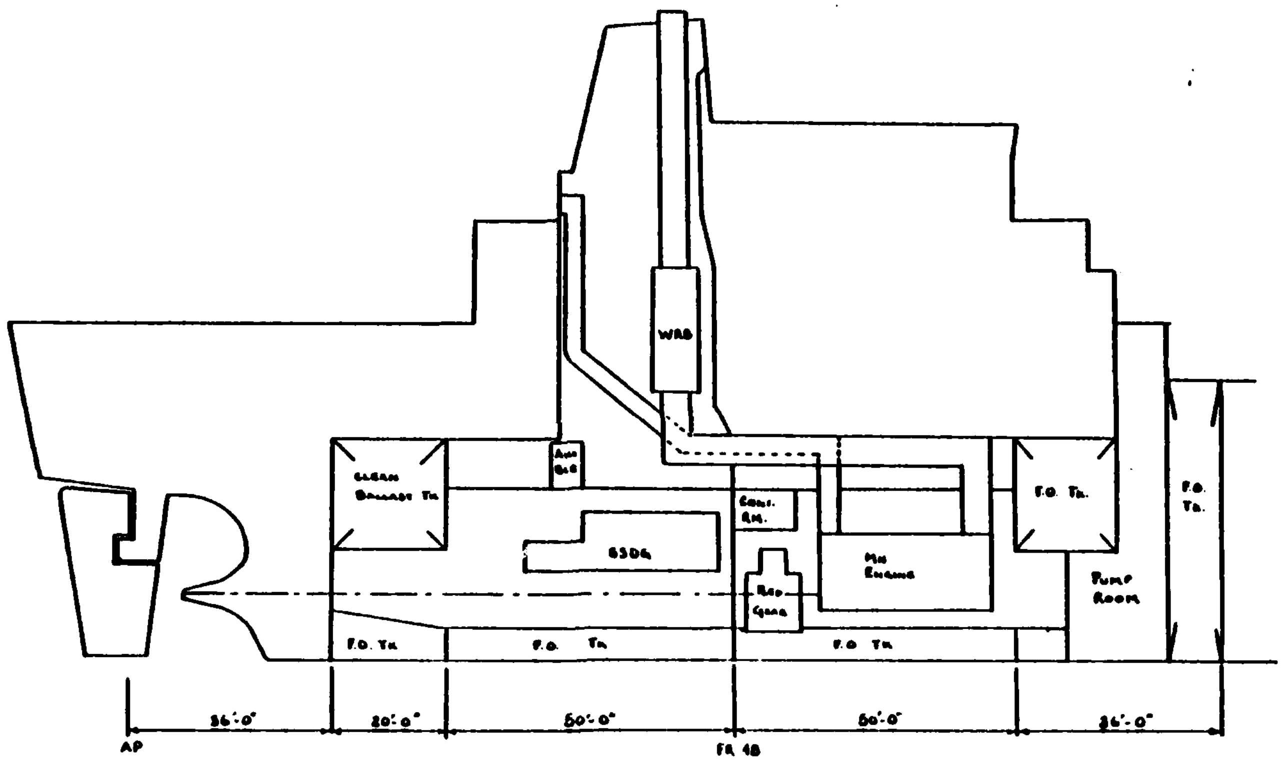


FIGURE 2.2

SECOND PLATFORM - TWIN ENGINE PLANT

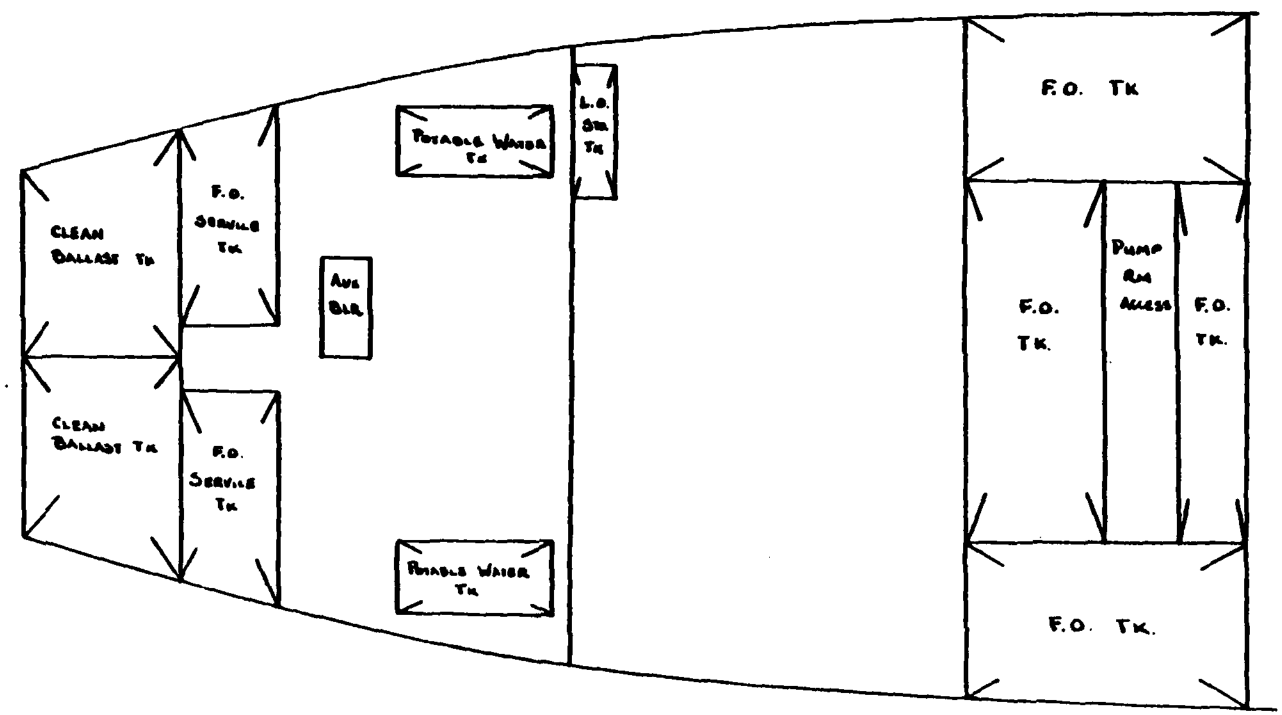


FIGURE 2.3

THIRD PLATFORM - TWIN ENGINE PLANT

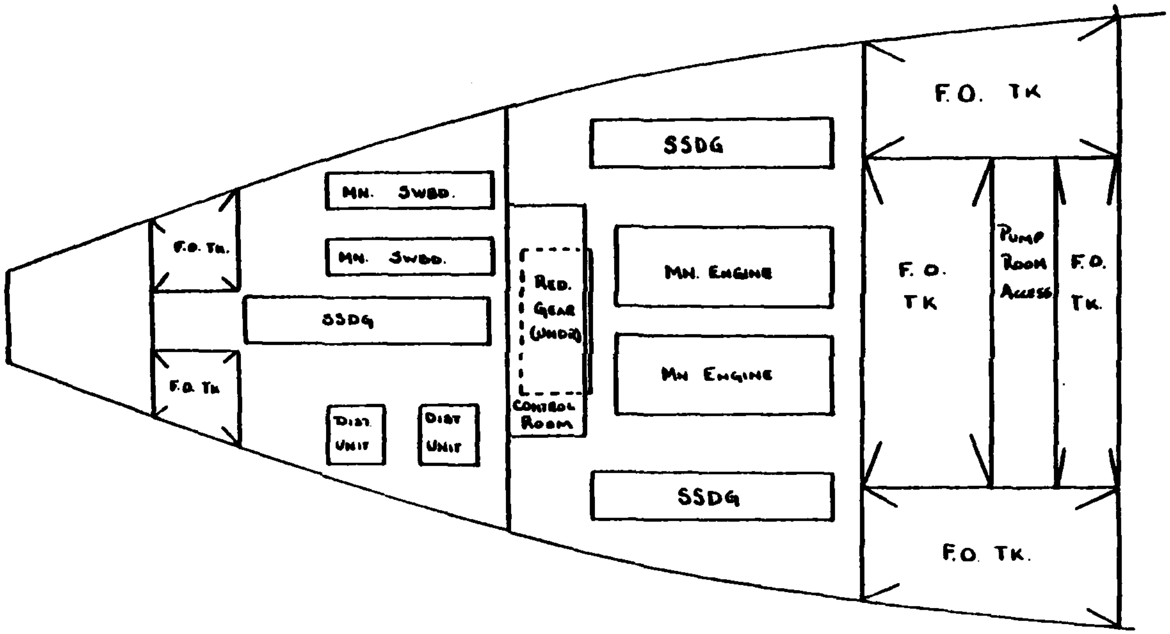


FIGURE 2.4

HOLD PLANT - TWIN ENGINE PLANT

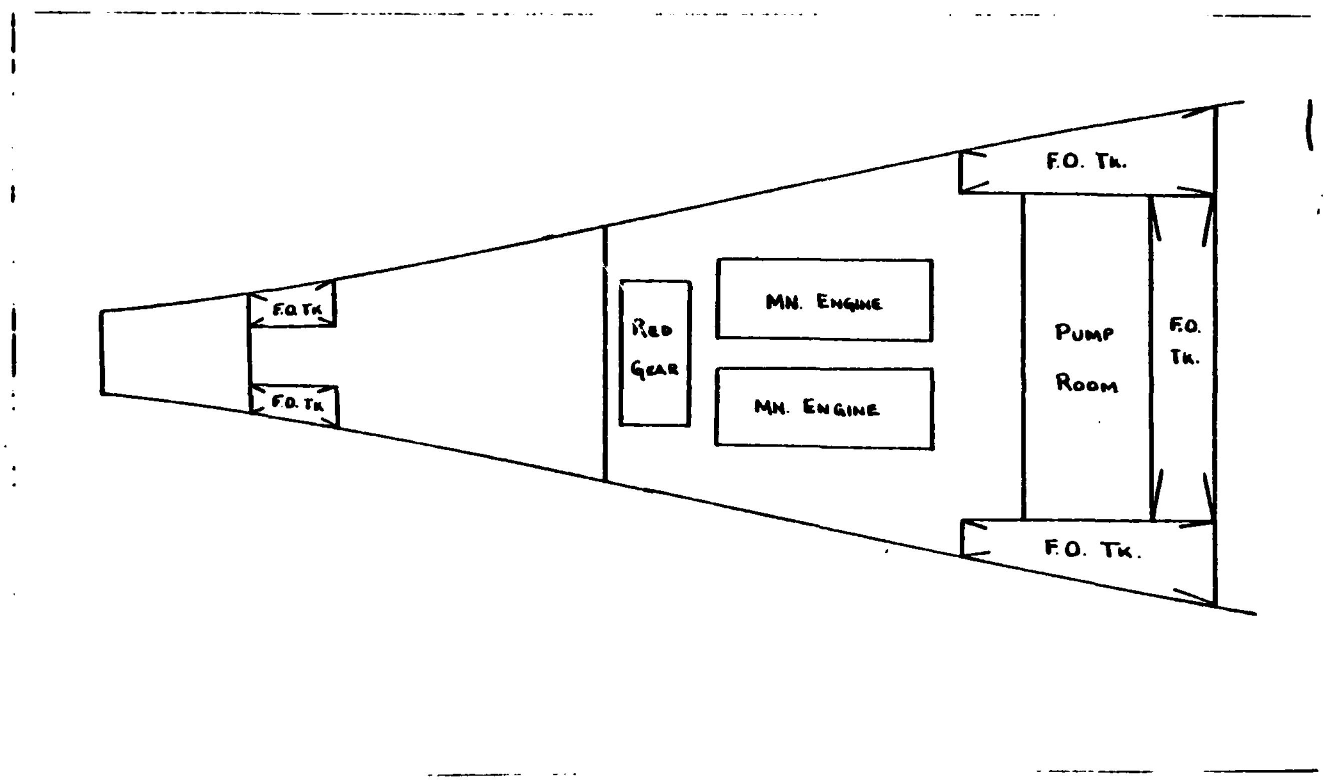


FIGURE 2.5

INBOARD PROFILE - TRIPLE ENGINE PLANT

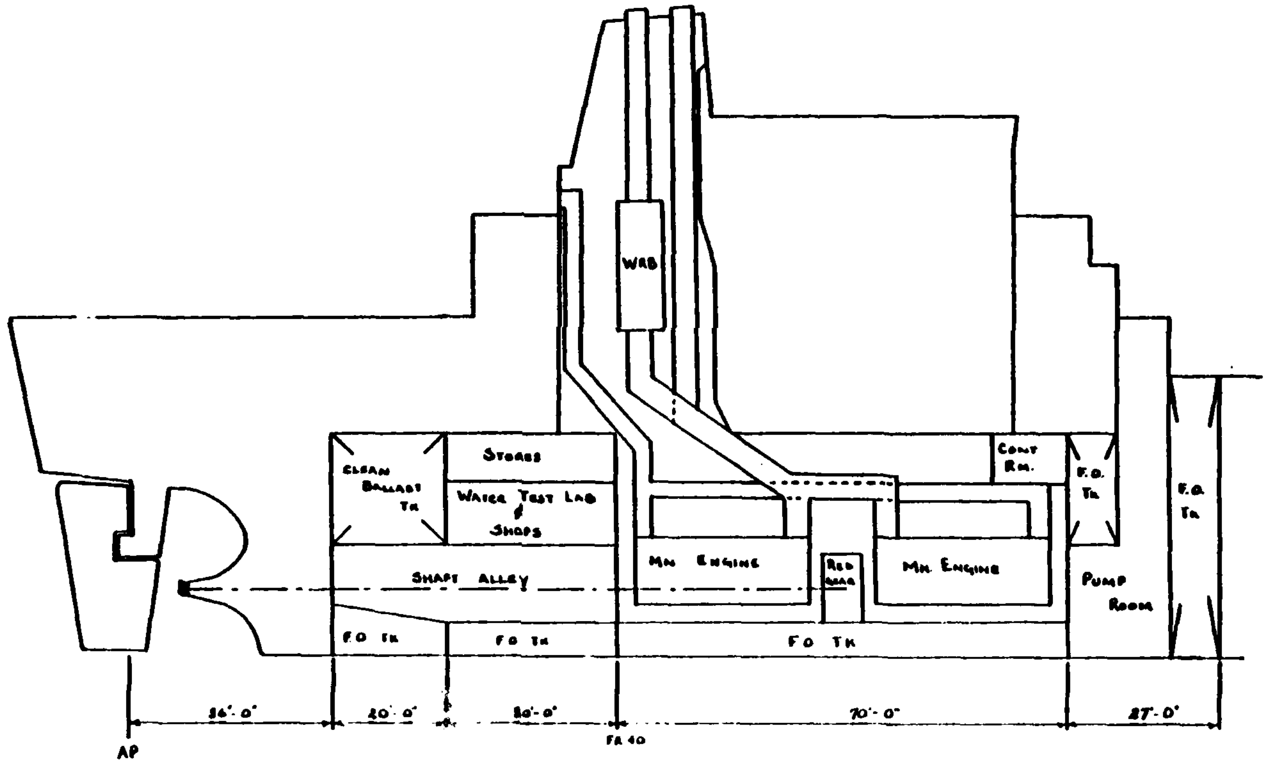


FIGURE 2.6

SECOND PLATFORM - TRIPLE ENGINE PLANT

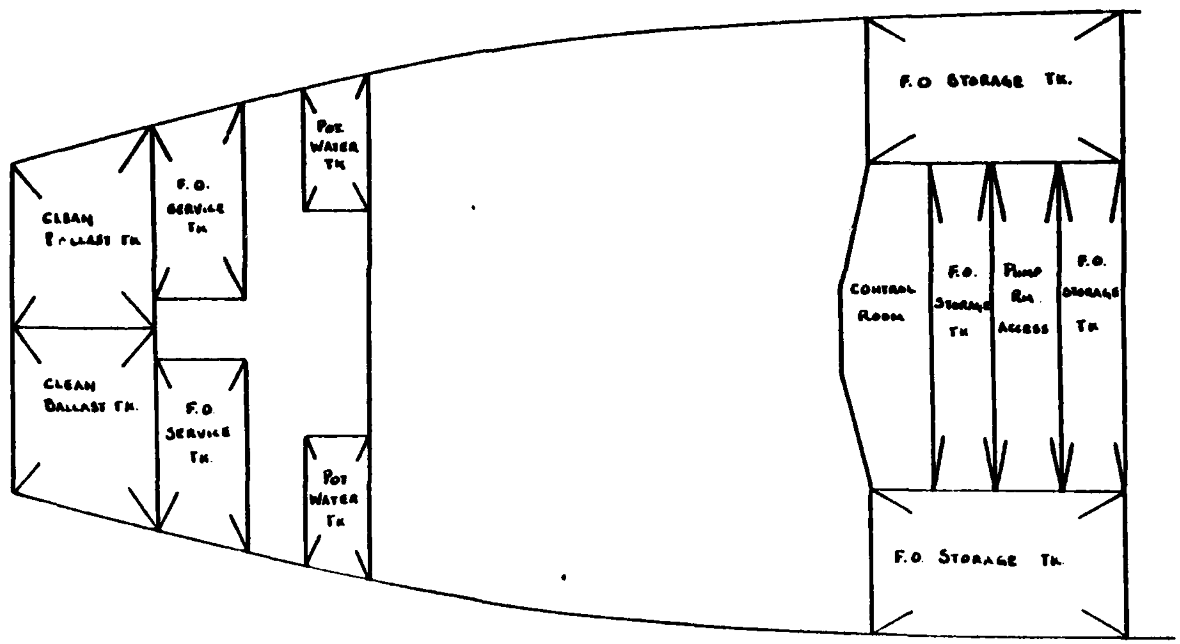


FIGURE 2.7

THIRD PLATFORM - TRIPLE ENGINE PLANT

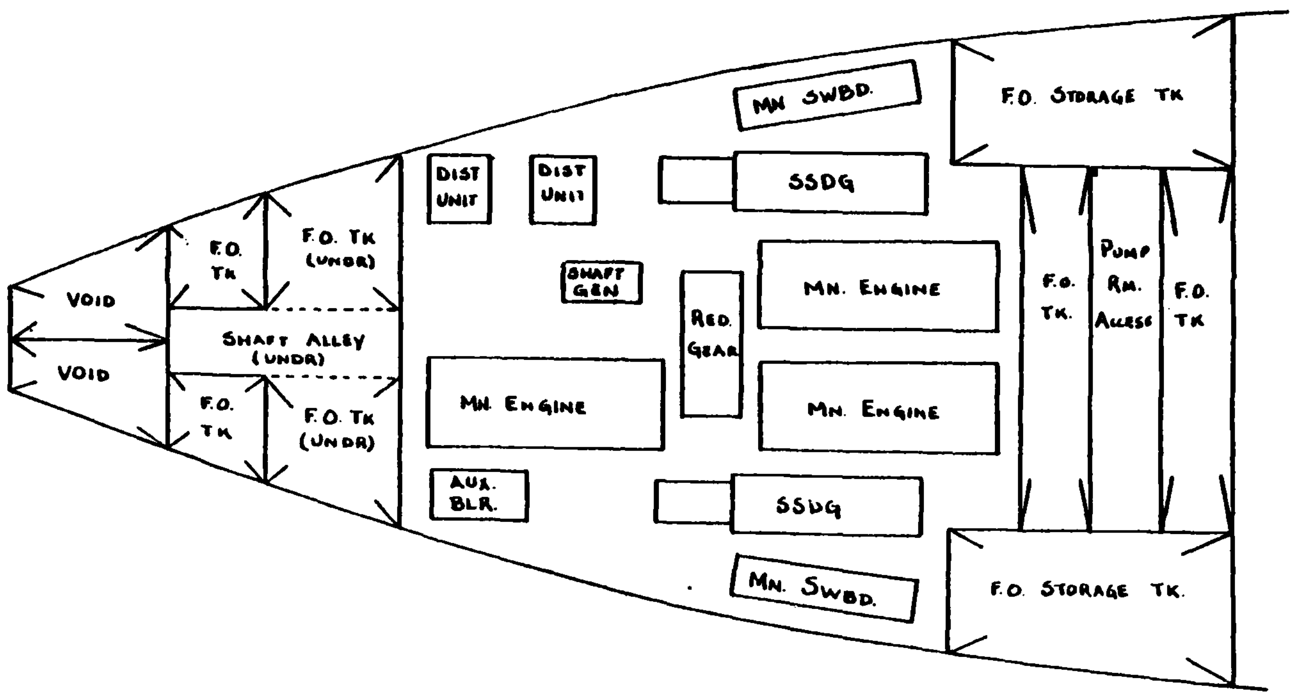


FIGURE 2.8

HOLD PLAN - TRIPLE ENGINE PLANT

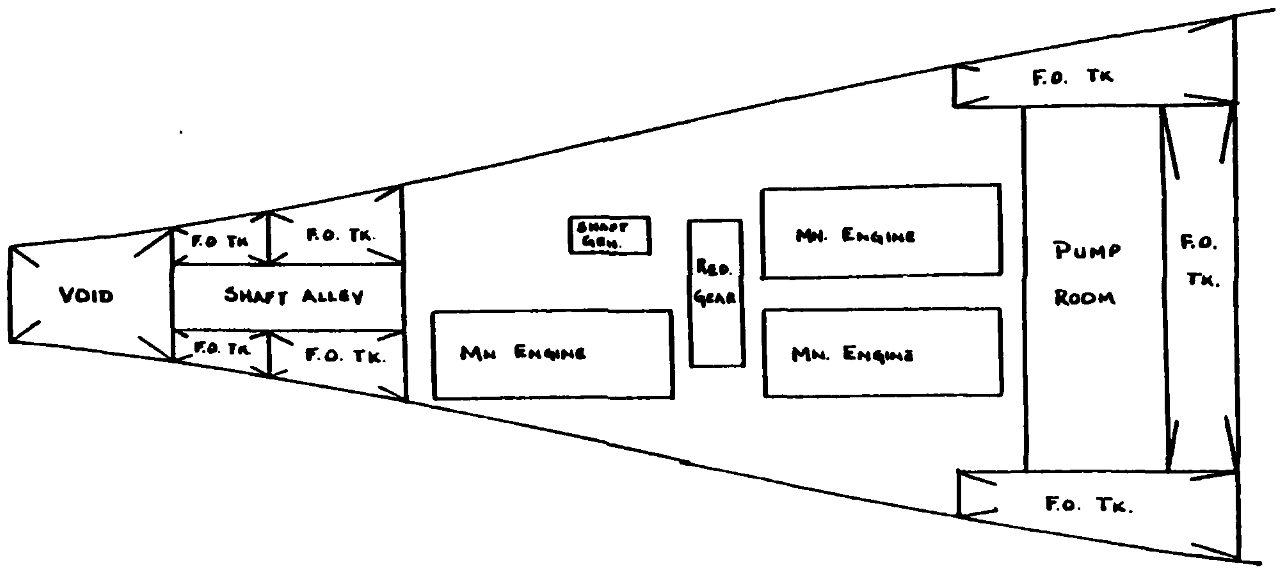


FIGURE 2.9

oil tanks 5-33-1 and 5-33-2 on the third platform and relocating the orientation of the potable water tanks. These changes allowed the installation of the Pielstick engines without impacting the fuel load or potable water capacity of the ship.

In order to access the effect of these structural changes on the damage stability of the ship, these bulkhead locations were plotted on the floodable length curve. As can be seen in Figure 2.10 with the relocation of the watertight bulkhead at frame 48 to frame 40, the ship is still a 3 compartment ship that can withstand a length of damage equal to 82.5 ft (0.15L).

Thus the candidate engines selected for installation in the AO-177 meet the selection criteria of section 2.1 and are compatible with the existing hull. The following sections will describe in more detail each of the proposed propulsion plants.

2.3 Endurance Fuel Calculation

With the candidate propulsion plants selected, the next step in the analysis is to determine the required endurance fuel load. The endurance fuel calculation was performed for each of the candidate propulsion plants based on an endurance range of 6,000 nautical miles at an endurance speed of 20 knots. This calculation is shown in Table 2.2 and is self-explanatory.

FLOODABLE LENGTH CURVE

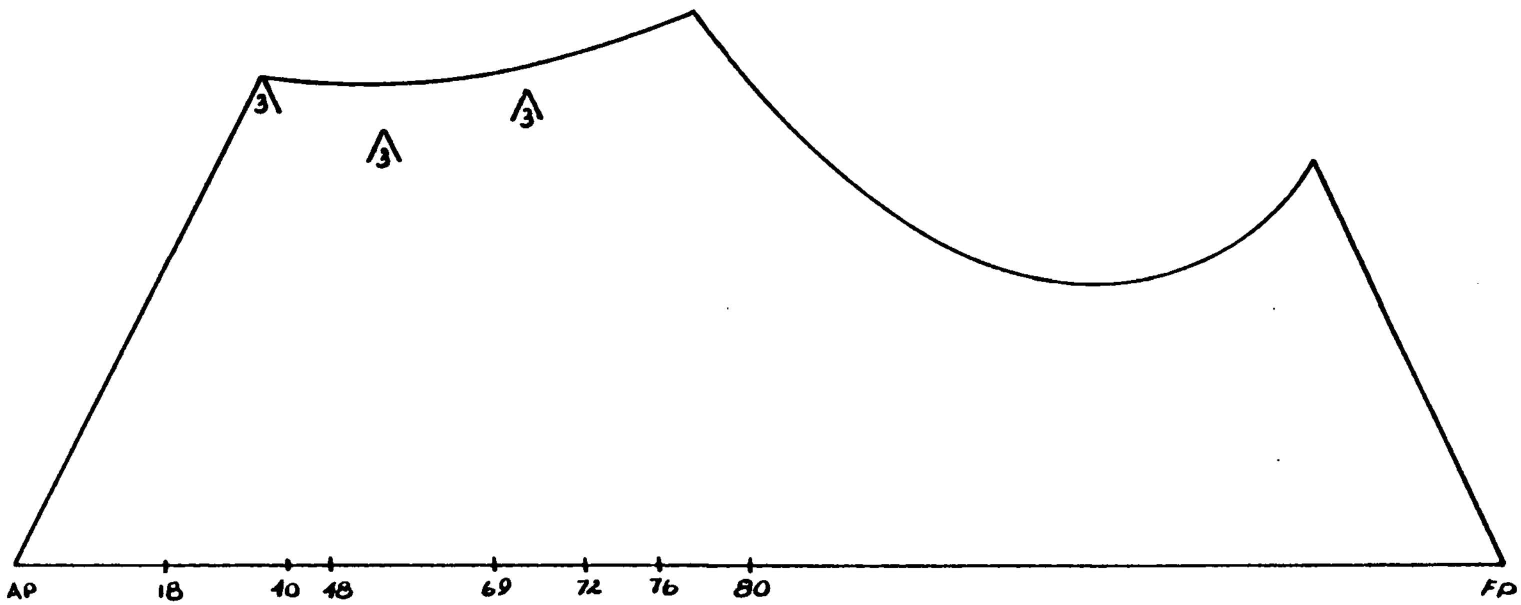


FIGURE 2.10

ENDURANCE FUEL CALCULATION

		RV-20-4	16PC 2.5v
(1)	endurance required, miles	6,000	6,000
(2)	endurance speed, knots	20	20
(3)	full load displacement, tons	26,271	26,271
(4)	rated full power, BHP	24,374	28,080
(5)	design endurance power @ (2) & (3), BHP	17,315	19,420
(6)	avg. endurance power, BHP (5) x 1.10	19,046.5	21,362
(7)	ratio, avg. end. BHP/ rated FP BHP (6)/(4)	0.781	0.760
(8)	cruising electrical load, kw	1,570	1,570
(9)	calc. propulsion fuel rate @ (6), lbs/BHP-hr	0.371	0.371
(10)	calc. propulsion fuel consumption, lbs/hr (9) x (6)	7,066.25	7,925.3
(11)	calc. aux. gen. fuel rate @ (8) lbs/kw-hr	0.509	--
(12)	calc. aux. gen. fuel consumption, lbs/hr (11) x (8)	799.13	--
(13)	tot. calc. all purpose fuel rate lbs/hr (10) + (12)	7,865.38	7,925.3
(14)	calc. all purpose fuel rate lbs/BHP-hr (13)/(6)	0.413	0.371
(15)	fuel correction factor based on (7)	1.02	1.02
(16)	specified fuel rate, lbs/BHP-hr (15) x (14)	0.421	0.378
(17)	avg. end. fuel rate lbs/BHP-hr (16) x 1.03	0.433	0.389
(18)	endurance fuel (burnable), tons (1)x(6)x(17)/(2)x2,240	1,104.5	1,115.1
(19)	tail pipe allowance	0.95	0.95
(20)	endurance fuel load, tons (18)/(19)	1,162.6	1,173.8

Allowing a 15% margin for tank size:

	RV-20-4	16PC 2.5v
tankage capacity required =	1,137 tons	1,350 tons

TABLE 2.2

This calculation resulted in a specific fuel consumption of 0.413 lbs/BHP-hr and an endurance fuel capacity of 1,337 tons for the Enterprise engines and a specific fuel consumption of 0.371 lbs/BHP-hr and an endurance fuel load of 1,350 tons for the Colt Pielstick engines. The current installed fuel capacity of the AO-177 is 1,940 tons.

Thus there are two alternatives since the required endurance fuel load is less than the present fuel capacity. One could reduce the fuel capacity of the ship and maintain the current endurance range of 6,000 nautical miles or one can maintain the fuel capacity and increase the endurance range.

The latter alternative, that of increasing the ship's endurance range, is more desirable because the increased range is obtained for no additional cost. In addition no major structural changes are required to remove existing fuel tanks or convert them into spaces for other uses. Finally, the impact on the total ship is reduced by keeping alterations to the existing ship to a minimum.

The endurance range obtainable by using medium speed diesel engines is calculated in the following manner using the figures obtained in Table 2.2:

for the Enterprise engines:

$$\text{Range} = \frac{(1940)(20)(0.95)(2,240)}{(0.433)(19,046.5)} = 10,011 \text{ NM}$$

for the Colt-Pielstick engines:

$$\text{Range} = \frac{(1940)(20)(0.95)(2,240)}{(0.389)(21,362)} = 9,936 \text{ NM}$$

It can be seen from the preceeding calculations that the endurance range of the AO-177 can be extended by some 4,000 nautical miles over the existing range of 6,000 nautical miles.

This increased endurance range provides improved flexibility in the operation of the ship without increasing the basic costs of the ship.

2.4 Engine Operating Profile

A plant configuration and engine load analysis was performed based upon the horsepower and electrical load requirements for cruise and underway replenishment operations. The speeds selected for these operations were those speeds at which the U.S. Navy most commonly conducts underway replenishment operations; 12 and 15 knots and finally a 20 knot cruise condition was considered.

As can be seen in Tables 2.3 and 2.4 on the following pages the twin engine plant is capable of performing underway replenishment operations (at 12 and 15 knots) on one engine. For the 20 knot cruise condition two engines are required to be operated. A minimum of one ship's service generator is required for all ship's operations with a second generator required for underway replenishment operations.

TRIPLE ENGINE MN. PROPULSION PLANT

	12 Knot Cruise	12 Knot Unrep	15 Knot Cruise	15 Knot Unrep	20 Knot Cruise
Propulsion (kw)	3,190	3,190	5,720	5,720	14,201
Elect Load (kw)	1,570	3,900	1,570	3,900	1,570
Total	4,760	7,090	7,290	9,620	15,771

Unit	Description	12 Knot Cruise	12 Knot Unrep	15 Knot Cruise	15 Knot Unrep	20 Knot Cruise
1	MN. Eng. (6,978 kw)	R	R	R	R	R
1	M.E. Gen (2,500 kw)	R	R	R	R	R
	Load on #1 ME	68%	74%	54%	57%	75%
2	MN. Eng (6,978 kw)	S	S	R	R	R
	Load on #2 ME	0	0	54%	57%	75%
3	MN. Eng (6,978 kw)	D	D	S	S	R
	Load on #3 ME	0	0	0	0	75%
1	SSDG (2,500 kw)	S	R	S	R	S
2	SSDG (2,500 kw)	D	S	D	S	D

R = Running
S = Standing
D = Down, available for repairs/maintenance

TABLE 2.3

TWIN DIESEL MN. PROPULSION PLANT

	12 Knot Cruise	12 Knot Unrep	15 Knot Cruise	15 Knot Unrep	20 Knot Cruise
Propulsion (kw)	3,790	3,190	5,720	5,720	14,201
Elect Load (kw)	1,570	3,900	1,570	3,900	1,570
Total	4,760	7,090	7,290	9,620	15,771

Unit	Description					
1	MN. Eng. (9,159 kw)	R	R	R	R	R
	Load on #1 M.E.	35%	35%	62%	62%	77%
2	MN. Eng. (9,159 kw)	S	S	S	S	R
	Load on #2 M.E.	0	0	0	0	77%
1	SSDG (2,500 kw)	R	R	R	R	R
2	SSDG (2,500 kw)	S	R	S	R	S
3	SSDG (2,500 kw)	D	S	D	S	D

R = Running
S = Standing
D = Down, available for repair

TABLE 2.4

The triple engine plant requires only one engine to be run for 12 knot underway replenishment operations and cruise; two engines are required for a 15 knot underway replenishment and cruise operation, and all three engines are required for the 20 knot cruise condition. The only time the ship's service generator is required is during underway replenishment operations. During all other times, the shaft driven generator has sufficient capacity to provide the electrical load requirements.

Thus it can be seen that both these plants provide excellent flexibility and ample time, at sea, for preventive maintenance and repairs to be performed. This is an important feature as these ships will be required to spend extended periods of time at sea which will require that most preventive maintenance be completed while the ship is underway.

2.5 Steam Generating Equipment

The steam generating equipment for each alternative propulsion plant consists of two waste heat recovery steam generators combined with an oil fired auxiliary boiler arranged as shown in Figure 2.11. It is normal design practice to place two waste heat steam generators on a ship and then size each for approximately two-thirds of the normal heating load. The normal heating load for the AO-177 is 7,500 lb/hr of steam.

But as pointed out in section 2.4, there are several

STEAM GENERATION SYSTEM SCHEMATIC

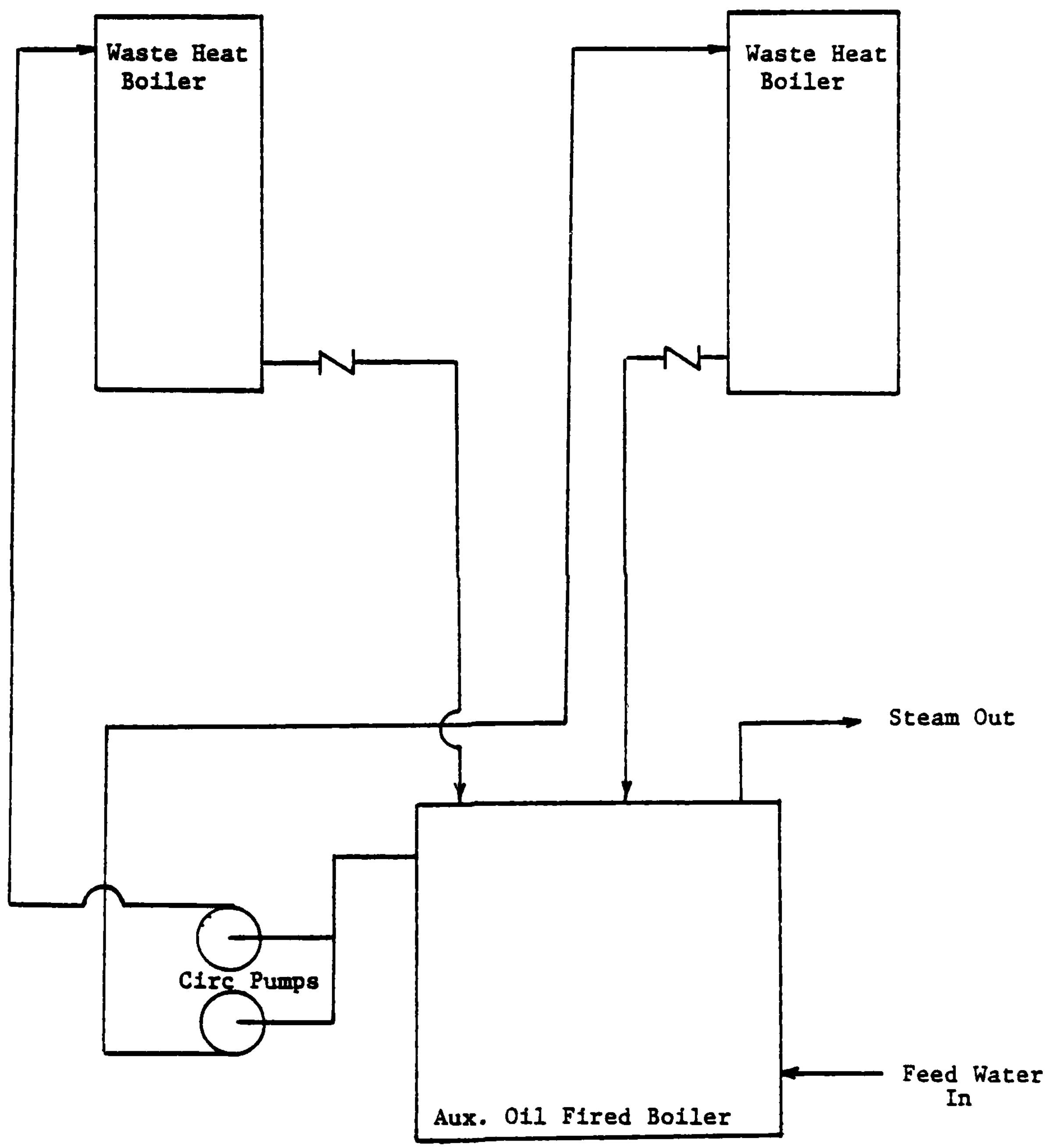


FIGURE 2.11

operating conditions which require only one engine, and it is anticipated that the ship will be operated at these conditions fully 50% of its operating time. In addition there is a requirement that the ship be capable of performing hot tank cleaning operations. Therefore, the capacity of each waste heat steam generator sized to produce 10,000 lb/hr at 60 psig and the auxiliary oil fired boiler was sized to produce 18,500 lb/hr at 60 psig in light of the preceeding requirements. The ship's tank cleaning heater can be arranged as a steam dump condenser if necessary in order to handle excess steam produced by the waste heat recovery units during periods of high power and low steam demand. The waste heat units will also act as an exhaust silencer for the engines. Note that in the three engine arrangement, where there are only two waste heat recovery units installed the third engine must be equipped with an exhaust silencer, but its exhaust may be fed into either of the waste heat units if desired during periods of operation when less than three engines are required.

The following calculations show that with the engines operating at their maximum continuous rating, there is ample heat in the exhaust gases to generate the required steam flow. These calculations are based on the following assumptions:

... exhuast gas temperature at boiler outlet = 300°F;

... boiler efficiency = 72%;

... outlet steam conditions are saturated;

... feed water to boiler is a saturated liquid at a temperature of 200°F and a pressure of 80 psig; and

... C_p = 0.275 BTU/lb-°F for exhaust gas from engines.

for the Enterprise engines:

$$\rho_{air} = 0.0288 \text{ lb/ft}^3 \text{ at } 885°F$$

$$\mathring{m}_{air} = 65{,}000 \text{ ft}^3\text{/min} = 112{,}320 \text{ lb/hr}$$

$$\mathring{Q}_{exh.gas} = mC_p\Delta T = (112{,}320)(0.275)(885-300)$$

$$= 1.806 \times 10^6 \text{ BTU/hr}$$

$$\mathring{Q}_{exh.gas} = \mathring{Q}_{BLR}\eta_{BLR} = (\Delta h)_{STM}\mathring{m}_{STM}$$

$$1.806 \times 10^7(0.72) = (1{,}177.6-168.09)\mathring{m}_{STM}$$

$$\mathring{m}_{STM} = 12{,}880.7 \text{ lb/hr}$$

for the Pielstick engines:

$$\rho_{air} = 0.0280 \text{ lb/ft}^3 \text{ at } 820°F$$

$$\mathring{m}_{air} = 60{,}639 \text{ ft}^3\text{/min} = 104{,}784 \text{ lb/hr}$$

$$\mathring{Q}_{exh.gas} = mC_p\Delta T = (104{,}784)(0.275)(820-300)$$

$$= 1.498 \times 10^7 \text{ BTU/hr}$$

$$\dot{Q}_{exh.gas} = \dot{Q}_{BLR}\eta_{BLR} = (\Delta h_{STM})\ \dot{m}_{STM}$$

$$(1.498x10^7)(0.72) = (1,177.6-168.09)\dot{m}_{STM}$$

$$\dot{m}_{STM} = 10,686.9\ lb/hr$$

The steam generation system will be fitted with appropriate alarms and controls to allow all functions to be controlled and monitored from the enclosed operation station in the engine room. In addition controls will also be provided for the auxiliary oil fired boiler to allow it to start automatically and supply the necessary steam to meet requirements should the waste heat boiler pressure drop below 45 psig.

2.6 Auxiliary Plant

As stated in section 2.2, the installation of diesel engines as main propulsion units will have little or no impact on the remaining auxiliary plant. The two 12,000 gallons per day distillation plants will be retained. It is recognized that the diesel plant does not have the make-up feed requirements that a steam plant does, but since the tankage for this water was not removed, it could be used for transfer to ships alongside during underway replenishment operations. Smaller combatant ships have limited fresh water producing abilities, but high consumption rates, so the ability to deliver fresh water by underway replenishment ships is a very desirable feature.

The air conditioning and refrigeration plants would remain as presently installed as would the electric drive cargo pumping system. The emergency diesel driven generator rated at 750 kw will also be retained.

2.7 Final Machinery Weights

In order to determine the machinery weights for each of the candidate propulsion plants, the first step was to obtain the individual machinery weights for the diesel plants. These weights, presented in Table 2.5, were obtained from manufacturers' data.

The next step was to identify that steam machinery to be removed and determine its weight. This was done using the final weight report for the AO-177. Table 2.6 identifies the machinery to be removed and its weight.

The resulting machinery weight for the twin engine plant was 917.3 tons and the triple engine plant was 802.2 tons. This proved to be a significant savings in weight over the steam plant which weighed some 958.5 tons.

DIESEL MACHINERY WEIGHTS

Machinery	Weight in Tons
Three 16 PC 2.5v Engines	214.14
Reduction Gear	55.80
Two Ship's Service Generators	86.00
One Shaft Driven Generator	4.00
	359.22
Two RV-20-4 Engines	253.8
Reduction Gear	91.5
Three Ship's Service Generators	129.0
	474.3

TABLE 2.5

STEAM PLANT MACHINERY REMOVED

Propulsion		Electrical	
Item	Weight (Tons)	Item	Weight (Tons)
Boilers	177.31	SSTG	47.10
Turbine	44.31	SSTG L.O. SYS	2.68
Reduction Gear	100.43		
Combustion System	14.94	Total	49.8
MN Steam Piping	20.58		
MN Condenser	45.17		
Fd. & Cond. System	38.09		
Boiler Uptakes	25.04		
Total	465.7		

Total steam machinery wt (GP2+GP3) = 958.5 tons
steam machinery removed = 515.5 tons
443.0 tons

Diesel machinery added:

twin engine arrangement = 474.3 tons
triple engine arrangement = 359.2 tons

Total medium speed diesel machinery weights:

twin engine arrangement = 917.3 tons
triple engine arrangement = 802.2 tons

TABLE 2.6

CHAPTER III

PROPELLER SELECTION

3.1 Introduction

When medium speed diesel engines are used for main propulsion, there are two widely accepted methods used to obtain reverse rotation of the propeller shaft. One method is to use a direct reversing engine and the other is to employ a controllable reversible pitch (CRP) propeller. Each method has its own very distinct advantages and disadvantages and the decision of which method to adopt must be made carefully.

The direct reversing engine connected to a fixed pitch propeller through a suitable reduction gear provides low initial costs, improved propeller reliability and less maintenance. The vessel's speed, in this case, is controlled by varying the speed of the main engine. Therefore, the main engines do not operate at their most efficient speed under varying load conditions. In addition, the slow speed performance of diesel engines is very poor and on the low end of the speed range, accurate speed control is difficult.

The CRP propeller on the other hand allows the main engines to operate at their most economical speed under a wider variation of loads. The CRP propeller provides ease of maneuvering in that all speed changes can be accomplished with one lever; ahead/astern operation can be accomplished without stopping the engines and dead slow operation of the ship can be achieved with an excellent degree of control. Use of a CRP propeller allows the main engines to operate at a constant speed over a wide speed range, thus allowing the use of a shaft driven generator to provide ship's service power. The principle disadvantages of the CRP propeller are its high initial cost, lower reliability and increased maintenance requirements.

In order to select the method to provide line shaft reversal for the AO-177 using medium speed diesel engines as a main propulsion unit, the operational requirements of the ship must be carefully considered. The AO-177 will be required to transit out from rear area support depots, at its most economical speed, and rendevous with the fleet to provide underway replenishment services. During underway replenishment (UNREP) operations, the ship will be required to maintain a near constant course and speed. Therefore, for these reasons, the CRP propeller was selected because of the inherent excellent speed control that can be obtained. In addition, the use of the CRP propeller allows the main engine to be operated at its most efficient speed under a wide variety of loading conditions, thus providing for excellent fuel economy. The excellent degree of speed control during slow speed operations is necessary in port where maneuverability is of importance. The high initial cost of the CRP is more than compensated for in terms of life cycle costs for the ship.

3.2 Controllable Reversible Pitch Propeller Design

In the design and selection of a particular propeller for any ship, the most dominant factor in the determination of propulsive coefficient is propeller efficiency. The following axioms of propeller selection were used to select a propeller for the AO-177:

1. Maximum efficiency at endurance speeds.
2. Minimum cavitation at maximum sustained speeds.

Since the AO-177 is an existing ship, data is available on speed and power requirements. In addition, because of the existing hull, there is a constraint on the maximum propeller diameter. A full power speed was not specified for the AO-177. An endurance speed of 20 knots was specified and this speed is to be attained using 80% of the installed horsepower. The installed horsepower requirement was set at 24,000 shaft horsepower. A margin of 6% was imposed on shaft horsepower to attain endurance speed to ensure that endurance conditions were met. This means that the ship must be capable of attaining 20 knots with 6% less than the required endurance power; or 19,200-0.06(19,200) = 18,048 SHP. Using these requirements and data obtained from model tests, the propeller selection analysis was performed using standard Troost Curves.

The first step in this calculation is to determine the ratio K_t/J^2 for the endurance speed of 20 knots and a maximum propeller diameter of 21 feet.

PROPELLER CALCULATION INPUTS

- resistance (R) = 191,688 lbs @ 20.0 knots
- model correlation factor (x) = 0.0005
- propeller diameter (D) = 21.0 ft
- V_{end} = 20.0 knots (full power speed)
- number of shafts (N) = 1
- 1-W = 0.78
- 1-t = 0.815
- head of water at prop CL = 21.5 ft

$$\frac{K_t}{J^2} = \frac{R}{[ND^2 V_{FP}^2 \; \rho (1-W)^2 (1-t)]}$$

$$= \frac{191{,}688}{(1)(21)^2[(20.0)(1.69)]^2(1.99)(0.78)^2(0.815)}$$

$$= 0.385$$

TABLE OF CONSTANT K_t/J^2 VALUES AT 20 KNOTS

J	K_t
0.1	0.00385
0.2	0.0154
0.3	0.03465
0.4	0.0616
0.5	0.0962
0.6	0.1396
0.7	0.1886
0.8	0.2464
0.9	0.311
1.0	0.385
1.1	0.4658
1.2	0.554

The next step in the propeller analysis consists of plotting the K_t/J^2 relationship on various propeller curves and constructing Table 3.1. A typical Troost Curve with the K_t/J^2 relationship plotted on it is presented in Figure 3.1. The values for P/D, J, K_t and η_o were taken from the plots of K_t/J^2 on the propeller curves. The values for n, A_p, T, q_t, P_o-P_v and % cavitation came from the following formulae and the cavitation diagram shown in Figure 3.2.

<u>FORMULAS USED IN PROPELLER ANALYSIS</u>

$$V_a = V(1-W) \qquad EHP = \frac{RV}{326}$$

$$n = (V_a/JD)^{60} \qquad \sigma_{0.7R} = \frac{P_o-P_v}{q_t}$$

$$T = K_t \cdot \rho \cdot n^2 \cdot D^4$$

$$A_D = (EAR)(\pi D^2/4)$$

$$q_t = (V_a/7.12)^2 + (nD/329)^2$$

$$P_o-P_v = 14.45 + 0.45h$$

$$A_p = A_D[1.067-0.229(P/D)]$$

$$T/A_p = \frac{2.26(EHP)(1+x)}{(1-t)VA_p}$$

Based on the selection criteria discussed previously in this section, Table 3.1 suggests that a B-5-75 propeller is the best choice for a CRP propeller for this application. This is also in keeping with the fact that the expanded area ratio

Propeller	P/d	J	n	q_t	A_p	T/A_p	T/A_{pqt}	$\sigma_{0.7R}$	%Cav	η_o
B-4-40	0.9	0.65	115.8	59.43	119.27	12.64	0.212	0.406	9	0.66
B-4-55	0.9	0.65	115.8	59.43	164.00	9.19	0.154	0.406	5	0.66
B-4-70	0.9	0.65	115.8	59.43	208.72	7.22	0.121	0.406	4	0.65
B-5-45	1.0	0.75	100.4	45.86	130.61	11.54	0.251	0.526	17	0.65
B-5-60	1.1	0.76	99.0	44.73	169.39	8.90	0.199	0.539	5	0.65
B-5-75	1.0	0.70	107.6	51.97	217.68	6.92	0.133	0.464	1	0.65

TABLE 3.1

PROPELLER CURVE B5-75; 5 BLADES EAR = 0.75

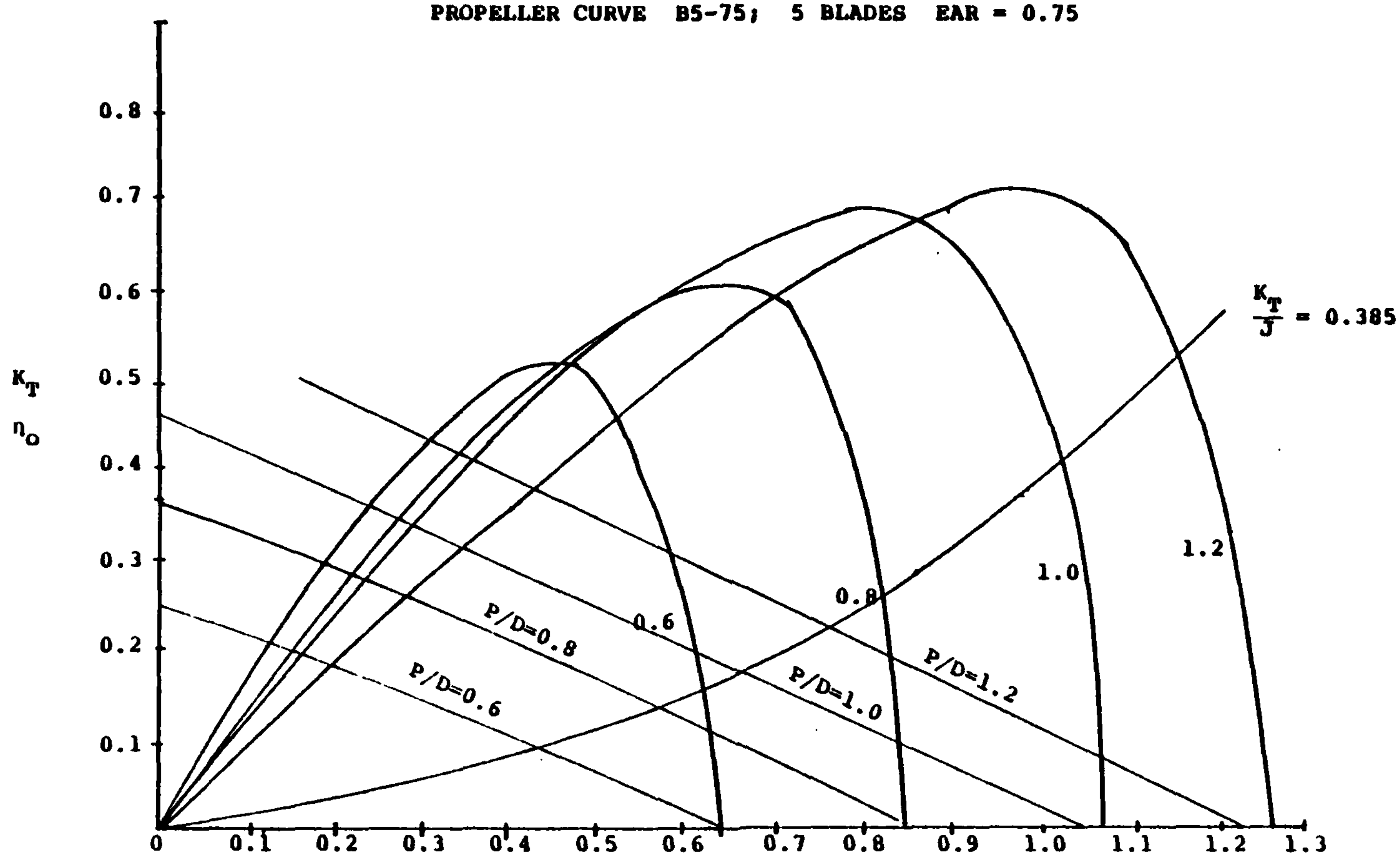


FIGURE 3.1

CAVITATION DIAGRAM

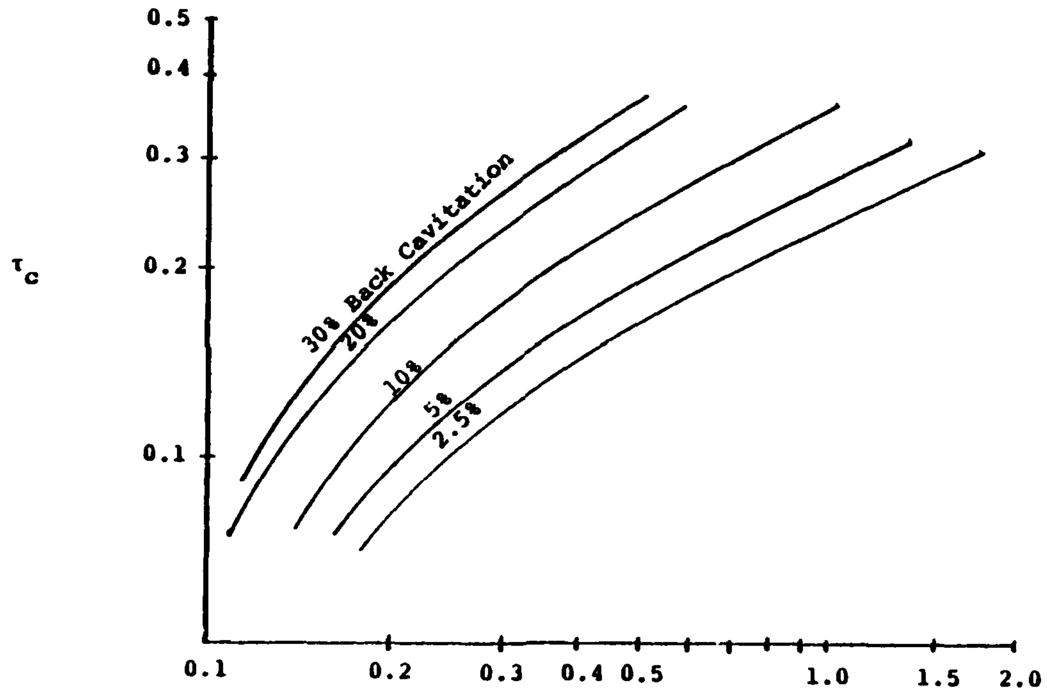


FIGURE 3.2

for CRP propellers must be less than 0.78 in order to allow sufficient blade clearance for blade reversal during astern operations.

Once the propeller is selected and its efficiency is known the propulsion coefficient (PC) and shaft horsepower required can be calculated as follows:

$$PC = \eta_O \cdot \eta_R \cdot \eta_H$$

$$\eta_R = 1.0,\ \eta_O = 0.65,\ \eta_H = \frac{1-t}{1-W} = \frac{0.815}{0.78} = 1.04$$

$$\therefore PC = (0.65)(1.0)(1.04) = 0.679$$

$$\text{and} \quad SHP = \frac{EHP}{PC} = \frac{RV}{326(PC)}$$

$$SHP = \frac{(191,688)(20)}{(326)(0.679)}$$

$$SHP = 17,320$$

Therefore, in selecting a B-5-75 CRP propeller, the required shaft horsepower to make a speed of 20 knots is 17,320. This results in a 9.8% margin under the required shaft horsepower of 19,200. This more than insures that endurance conditions will be met.

CHAPTER IV

RELIABILITY

4.1 Introduction

The reliability of a particular propulsion plant is an important factor in the performance of trade-off studies. In order to discuss reliability in any detail, the following definitions are presented to eliminate any confusion that might arise.

RELIABILITY, OPERATIONAL: Operational reliability is the reliability demonstrated by an equipment under actual field use. It is the probability that a system will give a specified performance for a given period of time, when used in the manner and for the purpose intended.

MAINTAINABILITY: A characteristic of design and installation which is expressed as the probability that an item will be retained in, or restored to, a specific condition within a given period of time, when the maintenance is performed in accordance with prescribed procedures and resources.

AVAILABILITY, INHERENT: The probability that a system or equipment, when used under the stated conditions without consideration for any scheduled or preventative maintenance in an ideal support environment, will operate satisfactorily at any given time. It excludes ready time, preventative maintenance downtime, supply downtime, and waiting or administrative downtime.

AVAILABILITY, OPERATIONAL: The probability that a system or equipment, when used under stated conditions and in an actual supply environment, will operate satisfactorily at any given time.

An analysis of reliability, maintainability, and availability (RMA) requires the calculation of the following factors based on statistical data provided for a particular price of equipment:

MTBF - mean time between failure

MTTR - mean time to repair

MTBM - mean time between maintenance

MDT - mean downtime

RMA requirements for the AO-177 were established during the initial design phase of the ship. It is the intent of this thesis not to degrade any of the original design parameters and, therefore, the requirement of having the reliability of 0.904 and an availability of 0.990 for a 30 day mission will be retained as requirements for the diesel engine plant also.

The reliabilities and availabilities presented in Table 4.1 were calculated based on a 30 day (720 hours) mission duration using the following equations:

. Reliability (repairable)

$$R = \frac{\mu+\lambda e^{-(\lambda+\mu)t}}{\mu \ \lambda}$$

RELIABILITIES & AVAILABILITIES OF VARIOUS
PROPULSION PLANT COMPONENTS

COMPONENT	MTBF(HRS)	MTTR(HRS)	R	A
Diesel Engine	8,000	8	0.999	0.999
Clutch	50,000	NR	0.986	1.0
Reduction Gear	200,000	NR	0.996	1.0
Shaft and Bearings	200,000	NR	0.996	1.0
CRP Propellers	25,000	15	0.999	0.999
Fuel Oil Motor	7,500	18	0.998	0.997
Fuel Oil Pump	5,500	4.5	0.999	0.999
Fuel Oil Purifier	10,000	4	1.0	0.999
Lube Oil Motor	7,500	7.8	0.999	0.998
Lube Oil Pump	4,000	5	0.999	0.998
Jacket Water Pump	27,000	7.6	0.999	0.999
Fresh Water Pump	12,500	12	0.999	0.999
F.O. Booster Pump	5,500	4.5	0.999	0.999

NOTES: Above reliabilities are based on 30 days operational time (720 hours).

TABLE 4.1

. Reliability (non-repairable)

$$R = 1-\lambda t$$

. Availability

$$A = \frac{MTBF}{MTBF+MTTR}$$

where

$\lambda = 1/MTBF$, $\mu = 1/MTTR$, and

t = mission duration time in hours

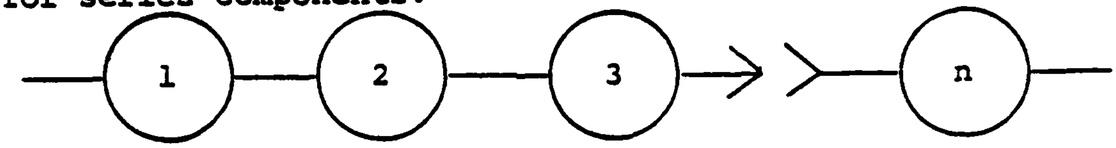
4.2 Twin Engine Reliability Calculations

The first step in the performance of a reliability/availability calculation is to develop a system and subsystem model. These models consist of functional schematics for the system under study.

Figure 4.1 shows the functional schematics for the twin engine arrangement. For the model to be operational, a complete "operational path" must exist between points 1 and 2. In order to determine the overall system reliability/availability the individual component/subsystem reliabilities must first be calculated and those resulting reliabilities/availabilities combined into the entire system to obtain the overall propulsion system reliability/availability.

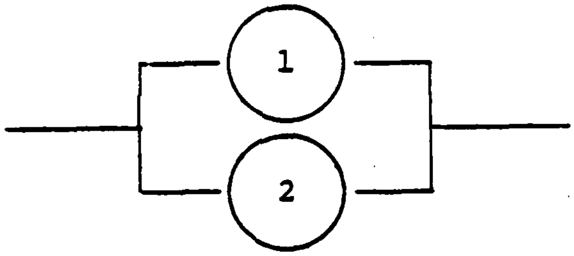
The method used to combine the reliabilities of individual components that make up a particular system/subsystem is as follows:

for series components:



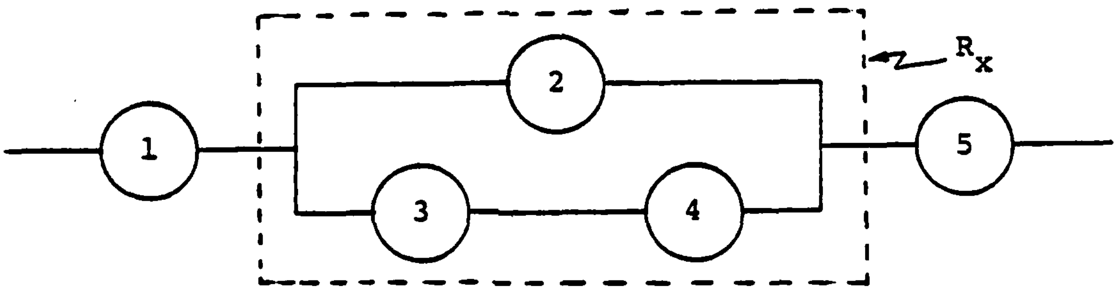
$$R_{SYS} = R_1 \cdot R_2 \cdot R_3 \cdot \ldots \cdot R_n$$

for parallel components:



$$R_{SYS} = R_1 + R_2 - R_1 \cdot R_2$$

for series-parallel components



$$R_{SYS} = R_1 \cdot R_x \cdot R_5$$

$$R_x = R_2 + (R_3 \cdot R_4) - R_2 \cdot (R_3 \cdot R_4)$$

$$R_{SYS} = R_1 \cdot [R_2 + (R_3 \cdot R_4) - R_2 \cdot (R_3 \cdot R_4)] \cdot R_5$$

The same rules apply in the calculation of system and subsystem availabilities for series, parallel, and series-parallel components. The difference lies in the definition of availability.

$$A(\text{inherent}) = \frac{MTBF}{MTBF+MTTR}$$

The following calculations, using the preceeding rules applied to the functional schematic diagrams of Figure 4.1, determine the reliability and availability of the twin engine arrangement. The reliability and availability figures used were obtained from Table 4.1. These figures were obtained from statistical data available from reference (1).

It can be seen readily from the calculations that the twin engine arrangement exceeds the reliability and availability requirements for this ship.

RELIABILITY CALCULATIONS FOR TWIN ENGINE ARRANGEMENT

Fuel Oil System Reliability

$$R_{MTR} = 0.998 \qquad R_{PUMP} = 0.999 \qquad R_{PURIF} = 1.0$$

$$R_{PUMP+MTR} = (R_{MTR} \cdot R_{PUMP}) + (R_{MTR} \cdot R_{PUMP}) - (R_{MTR})^2 (R_{PUMP})^2$$

$$R_{PUMP+MTR} = 1.0$$

$$R_{PURIF} = (R_{PURIF} + R_{PURIF}) - R^2_{PURIF} = 1.0$$

$$\therefore R_{F.O.SYS} + R_{PUMP+MTR} \cdot R_{PURIF} = 1.0$$

Engine Reliability

$$R_{\substack{LO \\ MTR}} = 0.999, \quad R_{\substack{LO \\ PMP}} = 0.999, \quad R_{\substack{JW \\ PMP}} = 0.999, \quad R_{\substack{FW \\ PMP}} = 0.999$$

FUNCTIONAL SCHEMATIC FOR TWIN ENGINE ARRANGEMENT

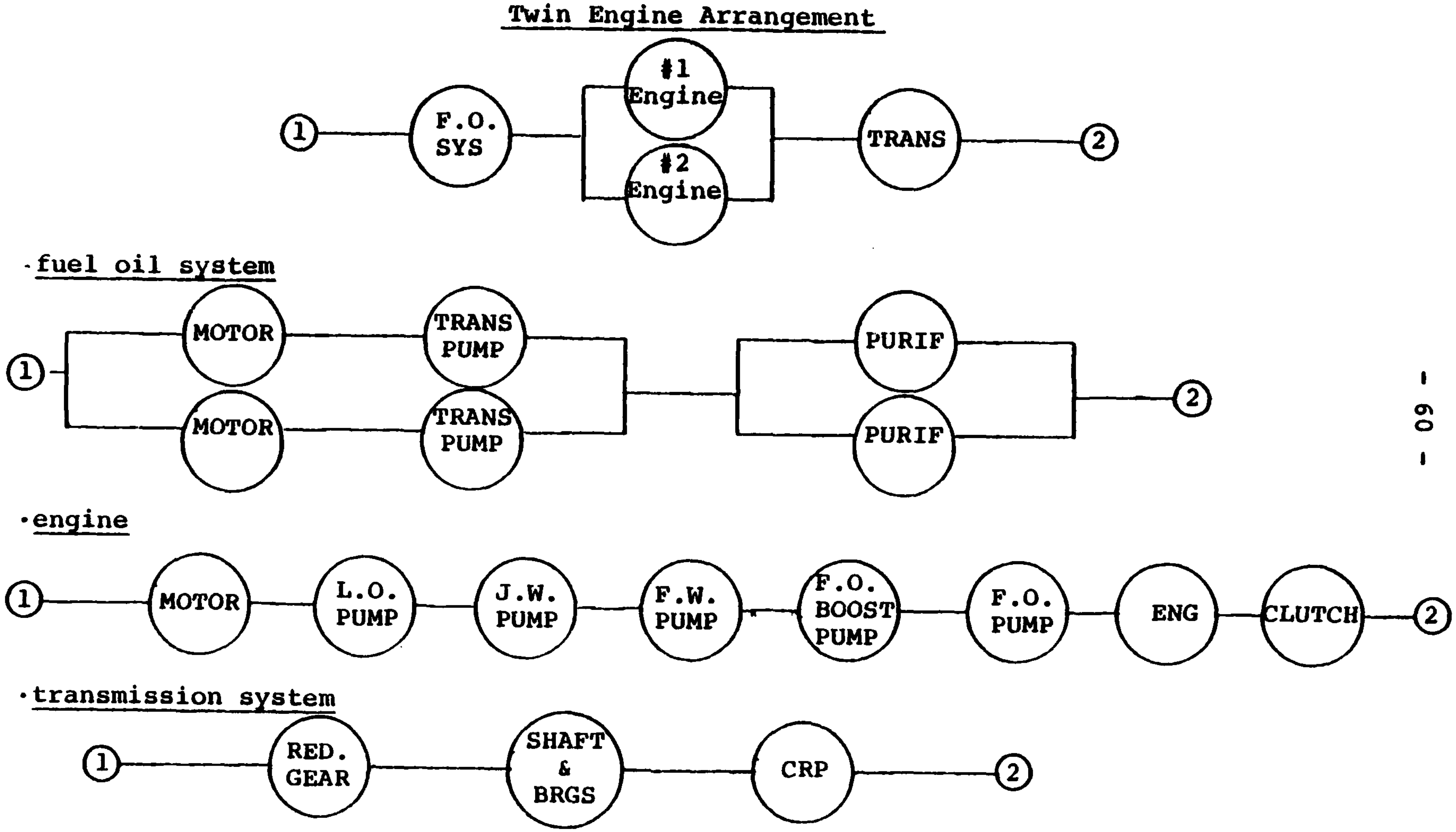


FIGURE 4.1

$R_{\substack{F.O.BOOST\\PMP}} = 0.999,\qquad R_{\substack{F.O.\\PMP}} = 0.999,$

$R_{ENG} = 0.999,\qquad R_{CL} = 0.986$

$$R_{ENGINE} = R_{\substack{LO\\MTR}} \cdot R_{\substack{LO\\PMP}} \cdot R_{\substack{JW\\PUMP}} \cdot R_{\substack{FW\\PUMP}} \cdot R_{\substack{F.O.BOOST\\PUMP}} \cdot R_{\substack{F.O.\\PUMP}} \cdot R_{ENG} \cdot R_{CL}$$

$$R_{ENGINE} = (0.999)(0.999)(0.999)(0.999)(0.999)(0.999)(0.999)(0.986)$$

$$R_{ENGINE} = 0.979$$

Transmission Reliability

$R_{RED.GEAR} = 0.996,\qquad R_{S\&B} = 0.996,\qquad R_{CRP} = 0.999$

$$R_{TRANS} = R_{RED.GEAR} \cdot R_{S\&B} \cdot R_{CRP}$$

$$R_{TRANS} = (0.996)(0.996)(0.999) = 0.991$$

Twin Engine Plant Reliability

$$R_{TWIN\ ENG} = R_{ENG} + R_{ENG} - R^2_{ENG}$$

$$R_{TWIN\ ENG} = (0.979)+(0.979)-(0.979)^2 = 0.999$$

$$R_{\substack{TWIN\ ENG\\PLANT}} = R_{\substack{F.O.\\SYS}} \cdot R_{TWIN\ ENG} \cdot R_{TRANS}$$

$$R_{\substack{TWIN\ ENG\\PLANT}} = (1.0)(0.999)(0.991) = 0.990$$

AVAILABILITY CALCULATIONS FOR TWIN ENGINE ARRANGEMENT

Fuel Oil System Availability

$$A_{MTR} = 0.997, \qquad A_{PUMP} = 0.999, \qquad A_{PURIF} = 0.999$$

$$A_{PUMP\&MTR} = (A_{MTR} \cdot A_{PUMP}) + (A_{MTR} \cdot A_{PUMP}) - (A_{MTR})^2 \cdot (A_{PUMP})^2$$

$$A_{PUMP\&MTR} = 0.999$$

$$A_{PURIF} = A_{PURIF} + A_{PURIF} - A^2_{PURIF} = 0.999$$

$$\therefore A_{F.O.\ SYS} = A_{PUMP\&MTR} \cdot A_{PURIF} = 0.998$$

Engine Availability

$$A_{\substack{LO\\MTR}} = 0.998, \quad A_{\substack{LO\\PMP}} = 0.998, \quad A_{\substack{JW\\PUMP}} = 0.999,$$

$$A_{\substack{FW\\PMP}} = 0.999$$

$$A_{\substack{F.O.BOOST\\PUMP}} = 0.999, \quad A_{\substack{F.O.\\PMP}} = 0.999, \quad A_{ENG} = 0.999,$$

$$A_{CL} = 1.0$$

$$A_{ENG} = A_{\substack{LO\\MTR}} \cdot A_{\substack{LO\\PMP}} \cdot A_{\substack{JW\\PMP}} \cdot A_{\substack{FW\\PMP}} \cdot A_{\substack{F.O.BOOST\\PMP}} \cdot A_{\substack{F.O.\\PMP}} \cdot A_{ENG} \cdot A_{CL}$$

$$A_{ENG} = (0.998)(0.998)(0.999)(0.999)(0.999)(0.999)(0.000)(1.0)$$

$$A_{ENG} = 0.991$$

Transmission Availability

$$A_{RED.GEAR} = 1.0. \quad A_{S\&B} = 1.0, \quad A_{CRP} = 0.999$$

$$A_{TRANS} = A_{RED.GEAR} \cdot A_{S\&B} \cdot A_{CRP}$$

$$A_{TRANS} = (1.0)(1.0)(0.999)$$

$$A_{TRANS} = 0.999$$

Plant Availability

$$A_{TWIN\ ENG} = A_{ENG} + A_{ENG} - A^2_{ENG}$$

$$A_{TWIN\ ENG} = (0.991)+(0.991)-(0.991)^2$$

$$A_{TWIN\ ENG} = 0.999$$

$$A_{PLANT} = A_{\substack{F.O.\\ SYS}} \cdot A_{TWIN\ ENG} \cdot A_{TRANS}$$

$$A_{PLANT} = (0.998)(0.999)(0.999)$$

$$A_{PLANT} = 0.996$$

4.3 Triple Engine Reliability

The functional schematic diagram for the triple engine arrangement is presented in Figure 4.2.

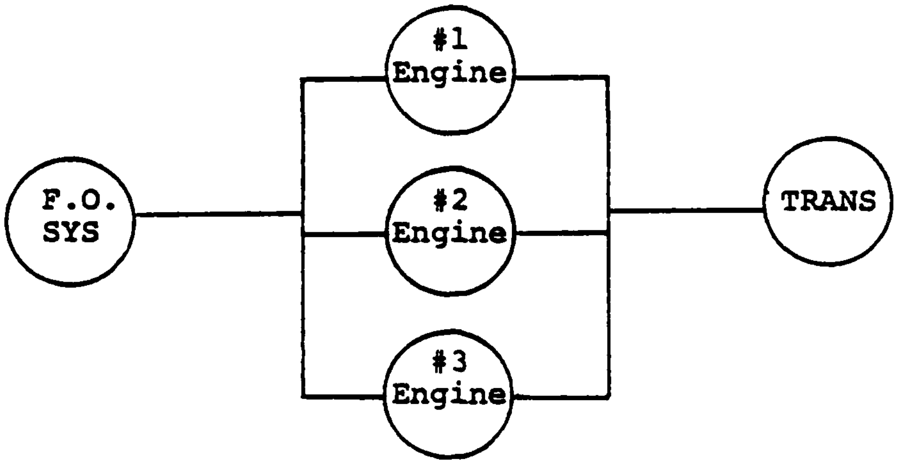


FIGURE 4.2

The subsystem components are identical to those used in the twin engine reliability and availability analysis; thus, the subsystem reliabilities/availabilities will be identical to those previously calculated in section 4.3. The values are as follows:

$R_{F.O.SYS} = 1.0$ $\qquad A_{F.O.SYS} = 0.998$

$R_{ENG} = 0.979$ $\qquad A_{ENG} = 0.991$

$R_{TWIN\ ENG} = 0.990$ $\qquad A_{TWIN\ ENG} = 0.999$

$R_{TRANS} = 0.991$ $\qquad A_{TRANS} = 0.996$

Using the rules outlined in section 4.2 for the combination of series, parallel, and series-parallel components of a functional schematic diagram, the reliability and availability of the triple engine arrangement is calculated in the following manner.

Reliability

$$R_{TRIPLE\ ENG} = (R_{TWIN\ ENG} + R_{ENG}) - R_{TWIN\ ENG} \cdot R_{ENG}$$

$$R_{TRIPLE\ ENG} = (0.990+0.979)-(0.990)(0.979)$$

$$R_{TRIPLE\ ENG} = 0.999$$

$$R_{PLANT} = R_{\substack{F.O.\\SYS}} \cdot R_{TRIPLE\ ENG} \cdot R_{TRANS}$$

$$R_{PLANT} = (1.0)(0.999)(0.991)$$

$$R_{PLANT} = 0.990$$

Availability

$$A_{TRIPLE\ ENG} = (A_{TWIN\ ENG} + A_{ENG}) - A_{TWIN\ ENG} \cdot A_{ENG}$$

$$A_{TRIPLE\ ENG} = (0.999+0.991)-(0.999)(0.991)$$

$$A_{TRIPLE\ ENG} = 0.999$$

$$A_{PLANT} = A_{\substack{F.O.\\SYS}} \cdot A_{TRIPLE\ ENG} \cdot A_{TRANS}$$

$$A_{PLANT} = (0.998)(0.999)(0.996)$$

$$A_{PLANT} = 0.993$$

Again, the availability and reliability of this arrangement exceeds the original design requirements for this ship.

4.4 Summary

In summary, both medium speed diesel engine machinery arrangements exceed the design requirements in terms of reliability and availability. Table 4.2 presented below shows the calculated values and the required values for reliability and availability. It should be noted that all calculations were based on a 30 day mission as called for in the original design requirements.

	REQUIRED	TWIN ENGINE	TRIPLE ENGINE
Reliability	0.904	0.990	0.990
Availability	0.990	0.996	0.993

TABLE 4.2

CHAPTER V

COST ANALYSIS

5.1 Introduction

In the design or conversion of any naval ship, the one question that is continuously on the lips of the customer and designer alike is, "How much will it cost?". In light of the rapidly fluctuating world economic situation of today, this question becomes, on one hand, more and more important; and on the other hand, more difficult to answer. Therefore, it is nearly impossible to make any general statements about the cost of any type of propulsion plant available which will be true in all cases.

The purpose of this chapter is not to provide absolute costs for the propulsion plants considered, but rather to provide some preliminary comparative cost estimations in order to allow a more intelligent decision to be made in the evaluation of each propulsion plant considered. In this case, the cost of the AO-177's steam propulsion plant is calculated and then compared with the costs calculated for each of the candidate medium speed diesel propulsion plants.

The method used to obtain the costs for each propulsion plant is that outlined by Femenia in reference (6). This basic method was adapted for use with the computer as outlined in reference (7) to provide flexibility and ease of calculation.

5.2 Description of Approach and Basic Inputs

The life cycle cost approach was used in evaluation of each of the propulsion plants. In this approach, the annual operating costs are expressed in terms of net present worth and added to the initial acquisition cost. This approach was selected as it represents current U.S. Navy practice for the determination of costs for new ships. In addition, annual costs for fuel, lubricating oil, and maintenance and repair costs were calculated. By summing these individual costs, a good comparison of the short term operating costs can be made. All cost figures are for 1979 dollars.

Table 5.1 shows the inputs to the computer program which are common regardless of plant type. The operating days per year were set at 245 because this represents 68% of one year which is a typical operational time for a U.S. Naval ship. The discount rate of 13% was selected because, at the time of writing, this was the annual inflation rate. Fuel cost per barrel was chosen to be $28 as this represents a good average cost for oil, considering the price fluctuations in the world oil market. The cost of lubricating oil was based on current commercial prices at the time of writing. The cost per man per year was established from current U.S. Navy data.

5.3 Manning

An input to the cost analysis is personnel costs. A cost per man per year was established but now a preliminary manning

INPUTS TO COMPUTER COST ANALYSIS

Year Dollars	1979
Operating Days/Year	245
Discount Rate	13%
Fraction at Maximum Speed	0.05
Fraction at Cruise Speed	0.95
Number of Shafts	1
Fuel Cost in $/bbl	$ 28
Cruise Speed (knots)	20
Endurance Range (NM)	6,000
Cost of Lube Oil, $/gal	$ 5
Cost/Man/Year	$ 26,000
EHP	11,760
Propulsion Coefficient	0.679

TABLE 5.1

schedule for each propulsion plant must be established.

In the case of the original steam propulsion plant, the manning table presented in Table 5.2 was developed from preliminary manning studies performed in the cause of the original AO-177 design and operational experience with single screw, two boiler steam plants. The manning table for the diesel plants was developed from manning documents from diesel propelled ships currently in service in the U.S. Navy and from operational experience.

It was assumed that the twin engine and triple engine diesel plants will require the same number of watchstanders and maintenance personnel, as there are approximately the same number of cylinders in each plant. The number of cylinders in a diesel engine are a good indication of the amount of maintenance required for a particular installation which can be directly related to the number of men required to man a particular ship.

5.4 Determination of Salvage Value

In order to determine the salvage value of the steam plant at the time of conversion, the following method was developed. It was assumed that if a shipowner were to purchase a used steam propulsion plant for installation in a ship he would operate for a period of one year, this would represent the best price that could be obtained for this machinery. In addition, it was also assumed that the lowest price obtainable for a used

MANNING TABLE

Personnel	Steam	Diesel
Machinist mates (MM)	24	0
Boiler technicians (BT)	23	0
Enginemen (EN)	5	26
Electrician mates (EM)	8	8
Interior communications Electrician (IC)	5	5
Hull technicians (HT)	9	9
Machinery repairmen (MR)	2	2
Officers	4	4
Totals	80	54

TABLE 5.2

steam plant would be 17% of the acquisition cost as stated by Femenia in reference (6).

Therefore, the salvage value of the steam plant is calculated using the following equation:

$$A_{STM}-(A_{STM}-A_{DIESEL})-N(AC_{STM}-AC_{DIESEL}) = SC_{STM} \quad (5.1)$$

where

A_{STM} = acquisition cost of steam plant

A_{DIESEL} = acquisition cost of diesel plant

AC_{STM} = annual cost of steam plant

AC_{DIESEL} = annual cost of diesel plant

SC_{STM} = salvage value of steam plant

N = number of years plant to be operated

This equation would be applied until the salvage value figure fell below 17% of the original acquisition cost for the steam plant; at this point, it was assumed the salvage value was constant.

Table 5.3 shows the cost figures for each of the propulsion plants under consideration in this study developed using the method described in reference (7). Table 5.4 shows the salvage value of the steam plant as calculated using equation 5.1. Note that by the fourth year, the salvage value has fallen to 17% of the original acquisition cost.

	COSTS IN 1979, MILLIONS OF DOLLARS		
	TWIN DIESEL	TRIPLE DIESEL	STEAM
Fuel	3.710	3.692	6.590
Lubricating Oil	0.116	0.339	0.006
Repair/Maintenance	0.119	0.121	0.162
Personnel	1.404	1.404	2.080
Annual	5.350	5.556	8.839
Annual Cost Over Life	39.214	40.728	64.787
Acquisition	14.751	14.993	16.072
Life Cycle Cost	53.895	55.650	80.783

TABLE 5.3

SUMMARY OF COST FIGURES

SALVAGE VALUE OF STEAM PLANT

Year	Steam Salvage Value in Million Dollars
1	11.26
2	8.24
3	4.28
4	2.73

TABLE 5.4

5.5 Comparison of Costs

The cost comparison was made by comparing the life cycle of the ship, over a 25 year life, operated as a steam ship to the cost of operating the ship for a given number of years as a steam ship and then converting it to a diesel ship. Again the total life of the ship was assumed to be 25 years. Thus, the expression used to determine the life cycle for the converted ship is:

$$LCC_{CONV} = (LCC_{STM} \text{ for N yrs}) + (LCC_{DIESEL} \text{ for 25-N yrs})$$

where

LCC_{CONV} = life cycle cost of converted ship

LCC_{STM} = life cycle cost of steam ship

LCC_{DIESEL} = life cycle cost of diesel ship

N = number of years to conversion

Tables 5.5 and 5.6 summarize the costs developed and the resulting savings using the above expression.

5.6 Cost Comparison Results

The most obvious conclusion that can be drawn from the results presented in Tables 5.5 and 5.6, regardless of which diesel plant is selected, is that the conversion must be performed prior to the end of the third year of life of the ship. Conversion beyond this point in time results in

TWIN DIESEL CONVERSION SAVINGS

Convert at End of Year	STM Life Cycle Cost	Twin Diesel Life Cycle Costs	Total Life Cycle Cost of Conversion	Savings
1	13.65	53.71	67.36	13.42
2	22.57	53.42	75.99	4.78
3	32.66	53.10	85.76	(4.97)

TABLE 5.5

TRIPLE DIESEL CONVERSION SAVINGS

Convert at End of Year	STM Life Cycle Cost	Triple Diesel Life Cycle Costs	Total Life Cycle Cost of Conversion	Savings
1	13.65	55.45	69.10	11.68
2	22.57	55.16	77.73	3.05
3	32.66	54.25	86.91	(6.127)

TABLE 5.6

increased life cycle costs over that of operating the ship as a steam ship for twenty-five years. The comparison also indicates that if the conversion is performed prior to the end of the ship's third year of service that the twin engine diesel plant will provide the greater cost savings in terms of life cycle costs. This can be directly attributed to the twin engine plant's slightly lower operating and acquisition costs.

In order to check the accuracy of the cost estimations developed in this chapter, other methods of cost estimation were examined. The Military Sealift Command of the U.S. Navy provided a figure for the cost per kilowatt used by their design division to obtain preliminary acquisition cost estimations. Their estimated cost per kilowatt of installed power (main propulsion and electrical generating capacity) is \$500/kw for diesel engine propulsion plants.

Using this figure, the twin engine diesel plant with a rated total power of 25,673.25 kw will have an acquisition cost of \$12.83 million and the triple engine diesel plant, rated at 25,936.44 kw will have an acquisition cost of \$12.96 million. These estimates compare favorably with the acquisition costs predicted by the computer analysis in that the difference in these costs are of the same order of magnitude as those predicted by the computer analysis. In addition, the order of magnitude of the absolute costs in each case are comparable considering the fact that these costs represent estimates only.

CHAPTER VI

RESULTS AND CONCLUSIONS

6.1 Candidate Plant Comparison

Table 6.1 shows the principle characteristics of the two candidate medium speed diesel engine plants considered in this study. In order to determine which of these two configurations to select for use in the AO-177 the following conclusions were drawn from the detailed analysis of each plant.

TWIN ENGINE PLANT:

(1) good engine loading under all conditions

(2) good plant flexibility

(3) ship can perform 15 KT UNREP on one main engine

(4) little or no structural changes required in ship

(5) a 4% savings in weight realized over steam plant

(6) improved savings if installed prior to the end of the third year of ship's life

TRIPLE ENGINE DIESEL:

(1) fair engine loading under all conditions

(2) excellent plant flexibility

(3) major structural changes are required

(4) ample maintenance opportunities available during underway periods

(5) a 16% savings in weight realized over steam plant

DIESEL PLANT COMPARISON

No. Engines/Rating	2/12,187 BHP	3/9,360 BHP
Installed BHP	24,374	28,080
Back-up Engine Available @ 15KT UNREP	Yes	Yes
Weight (Tons)	917.3	802.2
SFC (lb/BHP-hr)	0.433	0.389
Endurance (NM)	10,011	9,936
ACQ Cost ($)	14.751×10^6	14.993×10^6
Life Cycle Cost ($)	53.895×10^6	55.650×10^6
Maintenance Cost ($)	1.19×10^5	1.21×10^5
Fuel Cost ($)	3.710×10^6	3.692×10^6
L.O. Cost ($)	1.16×10^5	3.39×10^5
Reliability	0.979	0.988

TABLE 6.1

As can be seen from the above conclusions, both candidate medium speed diesel engine plants have very distinct advantages over the steam propulsion plant in the areas of machinery weight, life cycle costs, and fuel costs. In addition to these savings, the endurance range of the ship can be extended by approximately 4,000 nautical miles as demonstrated in Chapter II. The question now arises, "Which diesel plant configuration should be selected?" In order to answer this question, a more detailed examination of the two proposed plants is required.

6.2 Candidate Plant Selection

In the determination of which candidate plant to select, three advantages that the twin engine candidate has over the triple engine candidate are:

(1) no major structural changes are required;

(2) engine loading characteristics superior; and

(3) greater savings realized if converted.

The triple engine plant showed the following advantages over the twin engine plant:

(1) greater weight savings;

(2) more compact machinery arrangements; and

(3) superior plant flexibility and maintenance opportunities.

In light of the above advantages, the twin engine plant appears to be the more attractive option for use in the AO-177.

The lack of structural changes makes the twin engine plant more attractive not only from a cost basis, but it also will have a much lower impact on the overall ship in terms of strength, damage stability, and arrangements.

The improved engine loading achieved with the twin engine plant not only improves fuel economy but has an impact on engine maintenance problems and costs. With higher engine loadings the cylinder liners and pistons are subjected to fewer maintenance problems associated with the accumulation of carbon deposits which develop when these engines are operated at low speeds over extended periods of time. At higher engine loading conditions (70%-80% of rated horsepower) a slight improvement in overall fuel consumption rates will also be noticed.

6.3 Conclusions

The results of this study show that it is not only feasible to convert the geared steam propulsion plant currently installed in the AO-177 to a twin engine medium speed diesel propulsion plant, but the economic savings in terms of life cycle costs are significant. The savings are reflected in all of the factors that go to make up the life cycle cost of a ship. The most striking savings are made in terms of fuel costs. The medium speed diesel engine has a much better specific fuel consumption rate than that obtained in the steam plant. This has the additional advantage of allowing an extension of the ship's endurance range providing greater operational flexibility.

Another factor which greatly influences life cycle costs is personnel costs. This factor is important today and can be expected to grow in importance in the future as far as the U.S. Navy is concerned. The reduced manning of the diesel driven ship represents a significant savings over the steam plant and in addition, two skill areas (the machinist mate and boiler technician) are eliminated. This not only reduces direct ship related costs, but training and recruitment costs will also be significantly reduced.

In the determination of the actual horsepower required to drive the AO-177 at a sustained sea speed of 20 knots, it was discovered that considerably less than the installed 24,000 SHP was required. The margins imposed on the original design require that the sustained speed be made using 74% of the installed horsepower. The actual margin obtained by the use of diesel propulsion with a CRP propeller was 70.2%. Thus, the AO-177, as presently configured, has installed nearly 1/3 more horsepower than actually required.

While margin policy is not a subject addressed in this thesis, there would be a savings, in terms of acquisition costs, if a less conservative margin policy were used. A smaller main propulsion engine would reduce this cost. This could prove to be an area for further investigation; the trade-off between acquisition costs and margins applied to naval auxiliary ships.

The obvious question that now arises is that since it is feasible and economical to convert the AO-177's steam propulsion plant to a diesel propulsion plant prior to the end of the third year of the ship's service life, what are the savings obtained in building the ship from the start as a diesel ship? Table 5.3 of Chapter V presents a summary of the cost figures obtained for all three propulsion plants. From this data it can readily be seen that the twin engine diesel plant provides a 33.2% savings in life cycle costs over the steam plant and the triple engine diesel plant provides a 31% savings in life cycle costs over the steam plant. These savings can be directly attributed to the savings in the cost of fuel for the diesel plants. These savings also confirm the choice of the twin engine plant as the most economical one to select for conversion.

It is felt that this study shows that the U.S. Navy must take a long hard look at diesel propulsion for use in its auxiliary ships in light of the savings that can be realized across the board. The rising costs of fuel is the driving factor, and the improved specific fuel consumption obtainable with diesel engines make them appear more and more attractive as the price of fuel rises.

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