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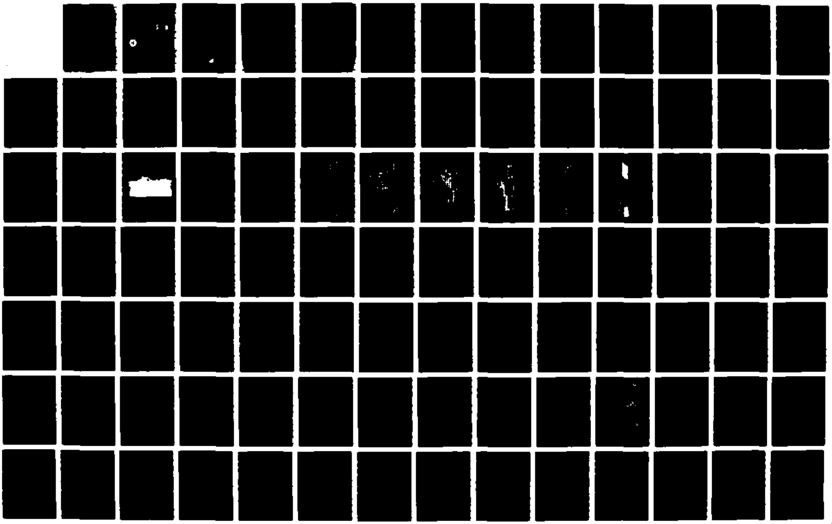

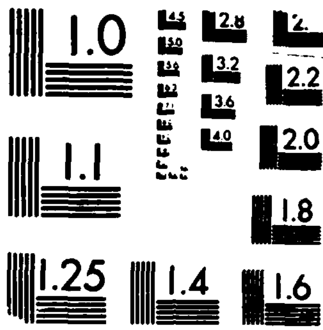

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SURVEY OF TECHNOLOGY WITH POSSIBLE APPLICATIONS TO UNITED STATES COAST GUARD BUOY TENDERS

VOLUME I - TECHNOLOGY ASSESSMENT

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FINAL REPORT
SEPTEMBER 1987

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16. Abstract

This report is divided into three volumes. ~~Volume I~~, "Technology Assessment", contains state-of-the-art summaries and projected trends for major technology areas pertinent to buoy tender design. Volume II, "Literature Abstracts", contains an annotated bibliography of the citations obtained during the technology survey. Volume III, "Technology Characterization", contains a description of the relational model and documentation of the computerized database used for storage and analysis of buoy tender data.

Volumes I, II, and III are contained within separate binders due to size considerations. Detailed abstracts of Volumes II and III may be found within each volume. What follows is the abstract for only Volume I.

In Volume I, "Technology Assessment", narrative summaries of the state-of-the-art in the following areas are presented:

1. foreign aids to navigation vessels;
2. aids to navigation; foreign practices;
3. offshore supply support/work vessels;
4. hull forms for seakeeping;
5. propulsion systems;
6. (6) Buoy ~~6.~~ weight handling systems; and
7. vessel automation, navigation, control and monitoring.

An assessment of candidate technologies within the above areas, most appropriate to new buoy tender designs, is provided.

17. Key Words

buoy tenders	seakeeping
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METRIC CONVERSION FACTORS

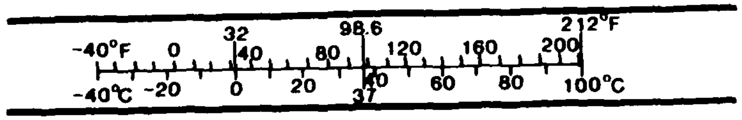
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (WEIGHT)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (EXACT)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures. Price $2.25. SD Catalog No. C13.10.286.

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (WEIGHT)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	0.125	cups	c
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (EXACT)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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1.0 INTRODUCTION

1.1 BACKGROUND

Over forty years have passed since the last WLBs (180 foot buoy tender) were constructed during World War II and sixteen years since the last WLM (157 foot buoy tender) was delivered by the Coast Guard Yard. Because of the advancing age of these units and the length of the acquisition process, even though Service Life Extension Programs (SLEPs) will keep these vessels in service into the 1990s, the Commandant of the Coast Guard has initiated the WLB/WLM Capability Replacement Project within the Office of Acquisition (G-A).

Since WLB/WLMs were constructed, major components of the Short Range Aids to Navigation (SRA) servicing task have changed very little. Only minor servicing details (e.g. battery replacement) have changed and those due to developments such as installation of solar power. While major tasks have changed very little, the technology to accomplish major tasks has changed considerably. This new technology provides alternatives for servicing the SRA system. None have been adopted in the past due to budgetary constraints, manpower limitations, and multi-mission requirements. Replacement of the WLB/WLM fleet should consider these technology alternatives. To accomplish this, a wide range of technologies must be assessed and candidate options that provide cost-effective solutions must be studied closely.

Identifying the need for technology assessment, the Coast Guard's Office of Acquisition assigned task 205.06.4.1, WLB/WLM Capability Replacement Project, to the Office of Engineering and Development. Since 1973 the Research and Development Program has been responsible for keeping the Coast Guard informed of new developments in the marine field. The work was performed by the Ocean Engineering Branch of the Coast Guard Research and Development Center as task 9207.1.1.3.4 of the Marine Vehicles Technology project. The R&D Center is a Headquarters Unit located in Groton, Connecticut.

1.2 OBJECTIVE

The objective of this Technology Survey is to compile, document and review the state of the art in specific areas of marine technology that apply to buoy tending. In addition, projected trends in each area are surveyed. The surveys are not intended to be all-encompassing; rather they are critical in nature; citations were appraised and reviewed to identify conventional and unconventional vessels and subsystems for possible inclusion in the WLB/WLM Capability Replacement Project. Areas of technology identified by G-A as needing assessment were:

Foreign Aids to Navigation Vessels

Aids to Navigation; Foreign Practices

Offshore Supply/Support/Work Vessels

Hull Forms for Seakeeping

Propulsion Systems

Weight Handling Systems

Vessel Automation/Propulsion/Navigation/Control and Monitoring

1.3 SCOPE

The present study stops short of making recommendations on optimal designs for hull configuration and supporting systems for buoy tenders. The purpose of this study is only to identify candidate technologies.

Access to information on foreign tenders and practices was limited, for the most part, to published reports, manufacturers' literature, telephone contacts, and data provided through official U.S. Coast Guard liaison contacts with foreign services.

Although there may be future design changes in aids to navigation which may result in smaller or lighter buoys, the present survey has reviewed only those candidate technologies needed to tend existing buoy designs and mooring hardware.

1.4 APPROACH

To make assessments of the available technology, it was necessary to obtain a working knowledge of the missions required for the

WLB/WLM Capability Replacement, the philosophy for the introduction of technology, and the constraints under which the project is structured. This was accomplished by reviewing the available materials on the WLB/WLM Capability Replacement Project such as the Mission Needs Statement and the Acquisition Paper, and by reading about and observing buoy tender operations. The R&D Center has long been involved in the development of new equipment and techniques in the Aids to Navigation (ATON) field, and ship test and evaluation work. This has given project personnel a broad exposure to the ATON tasks and missions. Project personnel made several trips to observe and study buoy tender operations in the fall and winter of 1986 to renew this experience.

It was clear that the best way to present the information gathered to program managers and other decision makers was in the form of a report. In case the information obtained is needed at a later date in either more detail or different formats, a computerized database was created.

After determining the form of information presentation, the available sources of information were reviewed to find which would be the most beneficial to search. Appendix A has more details about the sources of information used in this study. Promising automated databases, indexes, journals, books, and personal contacts were accessed, often in two steps, first to obtain abstract data, then the most promising references were obtained and reviewed to add to the survey.

1.5 REPORT ORGANIZATION

This report is divided into three volumes. Volume I, "Technology Assessment", contains state-of-the-art summaries and projected trends for major technology areas pertinent to buoy tender design. Current status in each area is presented along with recent and projected changes.

Volume II, "Literature Abstracts", contains an annotated bibliography of the citations obtained during the survey.

Volume III, "Technology Characterization", contains a description of the relational model and documentation of the computerized database used for storage and analysis of buoy tender data.

2.0 FOREIGN AIDS TO NAVIGATION VESSELS

2.1 PURPOSE

The purpose of this chapter is to present a review of foreign aids to navigation vessels, their equipment, and mission requirements.

2.2 BACKGROUND

In order to document the current state-of-the-art in foreign buoy tenders, an extensive literature survey was conducted. Initially, automated searches were conducted through the data bases listed in Appendix A. A very limited amount of information was acquired using this approach, so a manual search was invoked with emphasis on sources such as International Association of Lighthouse Authorities (IALA) Conference Proceedings and Bulletins, and marine trade journals. Information was compiled on approximately 28 different vessels from 11 different countries.

In order to put the international population of buoys and corresponding buoy tender fleet sizes in perspective, Tables 2.1 and 2.2 are presented. These statistics are compiled by IALA in the 1986/3 Bulletin, and represent data from the calendar year 1985. The 15 countries listed all have more than 500 navigational buoys.

TABLE 2.1

IALA BUOY TOTALS
(COUNTRIES WITH GREATER THAN 500 BUOYS)

SOURCE: IALA BULLETIN 1986/3

Country	Lighted Buoys	Unlighted Buoys	Total
Brazil	487	360	847
Canada	2841	9358	12199
China	799	70	869
Denmark	370	1300	1670
England	452	170	622
Finland	283	5620	5903
France	1040	1333	2373
German Dem. Rep.	274	1500	1774
Germany, Fed. Rep. of	672	2242	2914
Indonesia	407	403	810
Japan	1370	84	1454
Netherlands	565	1600	2165
Norway	126	1974	2100
Poland	124	418	542
United States	4219	19606	23825

TABLE 2.2
WORLD-WIDE AIDS TO NAVIGATION SERVICING RESOURCES
(COUNTRIES WITH MORE THAN 500 BUOYS)

Country	Tenders >30 m	Tenders <30 m	Other Vessels	Helicopters
Brazil	6	4	22	--
Canada	30	16	30	27
China	9	50	--	--
Denmark	3	5	--	--
England (Trinity House)	5	2	2	2
Finland	5	1	116	2
France	7	4	48	--
German Dem. Rep.	2	11	--	--
German, Fed. Rep. of	11	16	18	--
Indonesia	23	--	44	--
Japan	5	74	1	--
Netherlands	5	8	6	--
Norway	5	8	6	--
Poland	4	1	9	1
United States	51	30	145	--

It is apparent from Table 2.1 that the U.S. maintains the most extensive system of navigational buoys in the world. Our total number of buoys is nearly twice that of second ranked Canada, and an order of magnitude greater than most other nations. The 4219 lighted buoys are the larger, more important buoys in the total population, which mark offshore locations, channel entrances or turns. The necessity to service these larger buoys which are often placed in more exposed locations is a driving factor in the servicing platform design. Table 2.2 shows the resources used by these same countries in servicing their navigational aids. Categories listed include buoy tenders greater than 30 meters in length, tenders less than 30 meters in length, other vessels, and helicopters.

The category of principal interest in this section is tenders greater than 30 meters in length. The current fleet of U.S. offshore (WLBs) and coastal (WLMs) buoy tenders totals 42. Thirty of these are WLBs, 2 of which are currently in shipyards undergoing extensive renovation, or SLEP (Service Life Extension Program). These vessels were built in 1942-1944. The WLM fleet is comprised of 5 of the 157′ Redwood Class, built during the period 1964-1971; 6 of the 133′ White Sumac Class, built in 1942-1944; and one 175′ Fir Class tender, built in 1939. The balance of the 51 vessels tabulated in 1985 are mostly inland and construction tenders servicing rivers and shallow inland waterways, and vessels since decommissioned. The average age of our WLBs and 133′ WLMs is nearly 44 years. The average age of

157′ WLMs is 20 years. The single remaining 175′ Fir Class vessel is the second oldest vessel currently in Coast Guard service at 48 years. For the most part, buoy tender ages are well in excess of buoy tender replacement cycles of other countries, which is typically 25-30 years.

2.3 DESCRIPTION OF EXISTING TENDERS AND THEIR GEAR

Information on 28 foreign buoy tenders representing 11 different countries was compiled. Open literature references were typically descriptive texts of foreign replacement vessels. Other sources were general arrangement drawings of proposed vessels. In any case, the information available was condensed into a chart form and is presented in Table 2.3. A typical U.S. Coast Guard WLB (class C) is included in this table for reference and comparison. Each buoy tender was assigned a ship number. To facilitate cross-referencing with the data and references provided in Table 2.3, this number follows the ship name in parentheses as it is cited in the text. Outboard profiles of several of the buoy tenders are presented in Figures 2.1-2.13 immediately following Table 2.3.

The vessels surveyed fell into three general categories: traditional design offshore buoy tenders, coastal buoy tenders, and offshore supply vessels (OSV) converted for buoy tending operations.

Buoy Tender (name or class)	Number	Launch Date	LOA (ft)	LBP (ft)	Beam (ft)	Draught (ft)	Displacement (l. tons)	Deadweight (l. tons)	Speed (kts)	Fuel (gal)	Range (nm)	Crew	Buoy Deck	Powerplant	Thrusters	Propellers	Weight Handling Equipment	Other missions and capabilities	Comments	Hull Builder [References]
United States WLB	1	1944	180	170	37	13	1088	411	Max. 13.0 Econ 7.5	28,720	13,500	48	fwd well deck 37' x 48' 50 ton cap.	Diesel electric 2 x Cooper Bessemer GN8 diesels 2x700hp for 1200 shp	Transverse duct bow thruster driven by GM 671, 200 hp	1x5 blade FPP 8' 6" dia. 7' pitch	20 ton derrick crane stepped aft in buoy deck	SAR, ELT, Light icebreaking Lighthouse supply	Description of WLB (class C) for reference and comparison. Variations in these attributes exist in some vessels in the fleet	Duluth Ironworks Duluth, MN
Canada CCGS Jackman (pre-1050 class)	2	1973 purch 7/80	184		45	15-17			Max. 14 Cruise 12	176,000		Max 30	aft deck 100'x36' 520 ton cap.	6560 hp	Bow: 360 hp		Mobile Terrain crane (temp) 12 tons mntd stbd side aft	Icebreaking, towing, logistics and supply, constuction support, SAR	Offshore supply vessel purchased and tested for SAR and navaids service work. Successful testing lead to 1050 class design. Aft deck unworkable in typical offshore conditions	[2.10]
CCGS Samuel Risley (1050 class)	3	1985	227	207	50	17	2982	755	Cruise 12	200,000		31	aft deck 91' x 42' pedestal crane on stbd side	Diesel electric 4 x Wärtsilä diesels 2210 bhp ea., geared in pairs 2x1000kw, 575v shaft generators	Bow: 750 hp jet pump type Stern: 400 hp CPP	2 x 9.8' dia. 4 blade CPP in Kort nozzles	Liebherr 20 ton motion comp crane; 360° swing. Double drum towing and anchor winch	SAR (limited) Firefighting Icebreaking Lighthouse supply	This design resulted from testing of Jackman OSV design with large working deck aft. Vessel has good maneuverability and speed in handling buoys, but low freeboard aft limits operations in even moderate sea states. Can't back in ice as Kort nozzles tend to jam, very square stern	Vito Steel Boat and Barge Construction, Ltd Delta, B.C [2.16, 2.17]
CCGS Earl Grey (1050 class)	4	1986	227	207	50	17	2982	755	Cruise 12	200,000		30	aft deck 91' x 42' pedestal crane on stbd side	Diesel electric 4 x Deutz MWM diesels. 2200hp each	Bow: 750 hp jet pump type Stern: 400 hp CPP	2 x 9.8' dia. 4 blade CPP in Kort nozzles	Liebherr 20 ton motion comp crane; 360° swing. Towing and anchor winch	SAR (limited) Firefighting Icebreaking Lighthouse supply	Same class vessel as Samuel Risley with Deutz diesels. Second and last vessel of this class.	Pictou Industries Ltd Pictou, Nova Scotia [2.17]
1100 class	5	1986-1987	272	246	53	18.9	4736	1384	15.3		6500	51	fwd well deck 57' x 50'	Diesel electric AC/AC system w/ cycloconverter 3 main generators, 1 aux and 1 emergency. 5250kw; 7000shp	Transverse duct bow thruster	2 x FPP	20 ton Speed-crane derrick crane stepped fwd or aft of buoy deck. 15 ton towing winch on aft deck	Icebreaking SAR Helicopter ops	Canada's newest offshore navaids/light icebreaking vessel. Crane stepped aft allows helicopter sling ops from foredeck. Carries a 30' self-propelled barge and 24' workboat	Six vessels built in 4 Canadian shipyards
Denmark Inspections-fartej 80	6	pro-posed 1982	154		33.1	11.8		525	est max 15 est econ 12			14 to 16	fwd well deck 4.9' fbd	Diesel 1250 hp	Transverse duct bow thruster	1	12 ton derrick crane stepped fwd of buoy deck	Icebreaking	Cost approx $50 million (1982)	
Finland Seili	7		140	133	40	Max. 12.5		150		28,000		18	aft deck 50' x 37'	Diesel 1 x Wärtsilä 12V 22B, 2100 bhp @1000 rpm	Bow	1 x CPP Liaaen ACG 75/500	Hydraulic crane rated 6 tons @ 52.5' max reach. Stepped in stbd fwd corner of buoy deck	Icebreaking Towing in heavy ice	Vessel is of stout tug / supply boat configuration with aft working deck	Rauma-Repola [2.11]
Japan Hokuto	8	1979	180		34.8	Light 8.7	800.8	619.2	Max. 14 Econ 13.2	16,300	3460 @ 13 kts	25 max 31	fwd flush deck 6.6' fbd	Diesel 2 x Hanshin 6L 24SH Total 1085 bhp @400 rpm	Bow: electric 220V, 80 kw motor, 1.2 tons thrust	2 x 5.2' diameter 3 blade CPP, aluminum-bronze	15 ton Thompson ship crane stepped aft of buoy deck	Lighthouse support	Vessel has offshore capability. Cost equipped: $4.4 million (1982). Can store up to 3 x 7 ton buoys and 4 x 4 ton sinkers fwd of buoy deck	Sasebo Heavy Industries Co Ltd [2.8]

TABLE 2.3 Foreign Buoy Tender Characteristics

Buoy Tender (name or class)	Number	Launch Date	LOA (ft)	LBP (ft)	Beam (ft)	Draught (ft)	Displacement (l. tons)	Deadweight (l. tons)	Speed (kts)	Fuel (gal)	Range (nm)	Crew	Buoy Deck	Powerplant	Thrusters	Propellers	Weight Handling Equipment	Other missions and capabilities	Comments	Hull Builder (References)
West Germany Gustav Meyer & Otto Treplin	9	1966	160	147	31.2	10.5	650		Max 13.8	9,300		11	fwd well deck	Diesel: electric 2 x Deutz SBA 8M528, 750 bhp each @ 750 rpm, 2 x BBC 428 KVA, 231/400V, 3Φ generators	Bow: transverse duct, powered by mains. Vessel also has an active rudder	1 x 4 blade CPP	DC electric drive jib crane, 11 tons @ 21.3' radius, 6 tons @ 37.7' rad. 66' lift, 100 ft/min at full load; stepped fwd of buoy deck	Resupply capabilities: 34 ton water, 20 ton fuel; firefighting	Designed for river and coastal buoy tending. Engines mounted fwd under buoy deck to reduce noise aft (accomodations, bridge, inspectors cabin). Exhaust stack inside main crane pillar fwd.	[2.4]
Scharhörn	10	orig. 1974 recom 1982	184		47	12.1							large wkng deck aft 3.3 -5.9' freeboard	2 x 1726 bhp	Bow: 430 hp	2 x FPP	12 ton crane with 59' reach motion compen.	Pollution control Oil Skimming Firefighting Icebreaking	Offshore supply vessel converted to pollution control / buoy tending vessel for testing. Overall design rated reliable. Further changes required to improve maneuverability, bridge visibility, and working conditions on aft deck which was very wet.	[2.14]
Mellum	11	comm 8/84	234	209	49.5	17.2			Max 16 Econ 11			14 + 6	large wkng deck aft - approx. 3,440 ft²	Semi-diesel electric 4 x Krupp MAK 8M332; 1656kw each geared in pairs	Bow: 8v 396 TC 53, 700kw @ 1500 rpm driving 1 x 4 blade CPP, 6' diameter	2 x 4 blade CPP, 10.8' diameter in nozzles, 2 rudders	Mobile gantry crane rated 12 tons @ 42.6' reach, 8 tons @ 49' reach; can reach 9.8' overboard	Pollution control Oil Skimming Towing-salvage Lightering Firefighting Icebreaking	OSV type design resulted from tests with Scharhörn. Primarily a pollution control vessel with particular attenetion to hazardous environment safety features. Germans operate 3 such combination vessels	[2.14]
Walter Korte	12	1957	180		29.7	11.5			Max 14		3500	18	fwd well deck 1786 ft²	4 x Maybach 6 cylinder geared dieses, total 1800 blp	Bow: Jastram thruster in transverse duct with 2 counter-rotating props 250 hp DC motor drive; plus 200 hp Pleuver active rudder	1 KaMeWa-Atlas CPP Diameter: 7.7' 4 blades	12 ton jib crane by Kampangel 65 ft/min @ full load motion compen. Sinkers hoisted from bow pulley 60 ton pull by buoyancy lift.		Operation of bow thruster and active rudder allow vessel to siddle without turning. Vessel recently decommissioned.	Jadewerft Wilhelmshaven [2.18]
India M.V. Pradeep	13		161		31	9.2				15,140	2000	30	fwd flush deck	2 x GRW MAN 8V, 490 bhp each @ 1500 rpm	None	2 x 4.6' dia. 4 blade FPP Mang-bronze	5 ton derrick crane stepped aft of buoy deck; 7 ton winch	Logistics and supply, wreck location & marking small towing	Vessel is equipped with 23' landing craft type work boat for construction support and landing supplies ashore. Large buoys are towed, not handled aboard.	[2.15]
Norway 2946	14	1975	137		32.8	12.5		850	Max 13 Econ 11		3000			1 x Winchmann SAX, 1250 blp	Bow, 100 hp	1 x CPP	12 ton (2 x 6 ton) derrick crane	Lighthouse support Icebreaking	Cost est. $3.8 million (1982)	
7488	15	1985	144	124	32.8	12.5														[2.11]
Netherlands Breeveertien (B1)	16		201	180.4	37.1	11.2	1000		14.2	13,250 + 3,150 for lightship	2000 @ 12 kts	16 2 x 11 alternate weekly	fwd well deck, 4.3' fbd	Diesel-electric main electric motor gives 1373 blp	Bow: 420 hp Stern: 420 hp Transverse ducts		Damag Kampnagle electric pivoting crane; 15 tons, 9 ton chain winch	Resupply Mine laying Research	Offshore vessel with special consideration to noise abatement. Can do buoy operations in Beaufort 6, with 2 kt current	[2.2]

TABLE 2.3 (Continued)

Buoy Tender (name or class)	Number	Launch Date	LOA (ft)	LBP (ft)	Beam (ft)	Draught (ft)	Displacement (l. tons)	Deadweight (l. tons)	Speed (kts)	Fuel (gal)	Range (nm)	Crew	Buoy Deck	Powerplant	Thrusters	Propellers	Weight Handling Equipment	Other missions and capabilities	Comments	Hull Builder [References]
Sweden Baltica	17	1981	180	164	39.4	12.1	1238		15	44,150		12 + 12 ex	fwd well deck 50' x 30'	2 x Hedemora V16a/12 rated 1760 bhp each @ 1200 rpm	Bow & Stern: KaMeWa elec., transverse ducts, 300 hp each	1 x KaMeWa-Seffle CPP, 8.2' diameter Ni-Al bronze	Hydrolift electro-hydraulic crane with 12 ton hook @ 30', & 6 ton hooks @ 46' & 59' reach	Construction support Logistics Icebreaking Pollution control Towing	Vessel has pumped heeling-tank system used during icebreaking and heavy lifts. Also has integrated maneuvering joystick control.	A B Asiverken Åmål, Sweden [2.3]
Trinity House (England) THV Mermaid	18	1987	263	246	47.6	13.1	3080	870	12	120,693		24	fwd well deck Approx 40' x 40'	Diesel-electric: 4 x Ruston 8 cyl turbo diesel/alt. 900kw ea; 2 x Brush Electrical Machines dc motors 860kw ea. @ 220rpm	Bow: "U" tube by WhiteGill 925 shp @ 480 rpm; 6.9 tons thrust in any direction (360°)	2 x FPP	20 ton John Hastie Speed-crane (derrick); 15 ton towing winch; 14 ton buoy chain capstans	Towing Helipad	Most recent design for Trinity House. Similar but less extravagant than 'Patricia'. Buoy deck has vertical stowage pockets for high focal plane buoys. Bridge has tiller steering and joystick controls.	Hyundai Heavy Industries Co. Ltd. Ulsan, Korea [2.6, 2.7]
THV Patricia	19	1982	283	253	45.3	14.1		1042	14			21	fwd well deck Approx. 50' x 45'	Diesel-electric: 4 x Ruston 6RKCZ 4000 bhp @ 750 rpm each drive Nebb alt. 750 kw; 2 x BBC dc propulsion motors - 1,500 shp each @ 250rpm direct to shaft	Bow: Ellon Turbomachinery 925 bhp @ 480 rpm gives 7 ton thrust	2 x Stone manganese 4 blade FPP	20 ton John Hastie Speed-crane with 57.4' jib, 82 ft/min @ 12 tons, ±70° arc of operation; plus 2 buoy chain capstans, 1 tow winch	Towing Helipad Replenishment Facility inspect. Royal escort	Vessel is designed to work in Beaufort 5-6 (max waves to 13'). Elegant accomodations for inspectors and royal escort duties. Cost approx £9 million (1982)	Henry Robb Leith, Scotland [2.12]
England Relume	20	Deliv 5/79	249	230	40.1	12.3		773				55	fwd well deck, stowage for 8 buoys	2 x APE-Allen 512-DX turbo diesels 860 bhp each @ 750 rpm derated (for heat) to 815 bhp each	Bow: Brown Boveri reversible electric motor, 216 kw; 3.5 tons thrust	2 x Bamford Ajax 4 blade CPP	20 ton Stot & Pitt electro-hydraulic crane stepped fwd; can rotate 360°	Tends buoys, 1 lighthouse, 1 radio beacon, and 7 Decca land stations	Vessel run by Middle East Navigation Aids Service (MENAS) homeported in Bahrain. All machinery runs on central fresh water cooling system with heat exchanger.	Ailsa Shipbuilding Co. Ltd. Troon, Scotland [2.13]
MV Wilton	21	1983	131	121.4	30.3	6.3			8.5	11,500		7	fwd flush deck 65' x 27'	1 x British Polar F-20, 1200 bhp @ 900 rpm powers 3 hydraulic pumps which drive 3 x Rexroth 300 hp hydraulic motors which drive 3 rotatable Schottel thrusters (1 fwd, 2 aft)	Maneuverability provided by 3 rotatable Schottle drives	3 x FPP on Schottel drives	Hydraulic A-frame rated 30 tons with 11.5' reach plus 22 ton deck winch and 8 ton HIAB 180 hydraulic knuckle crane	Obstruction clearance Minor salvage	Vessel has computer-controlled dynamic positioning / navigation system GEC GEM 80, linked to Motorola MK3 radio position reference system. Manual control is by portable control box or single joystick control allowing motion in any direction. Buoy handling is over the blunt bow. Service areas are rivers, harbors and bays.	Tees & Hartlepool Port Authority Eng. Dept (After George Brown defaulted) [3.1]
Canada Provo Wallis 900 Class (Typical Existing)	22A	1969	189		42	12	1750		Max cruise 12.5	32,170	3300	30		Diesel electric 1565 kw 2100 shp					Typical design for 900 class — basic design for new ships of this class. Considered a small nav-aids tender / ice strengthened	

TABLE 2.3 (Continued)

TABLE 2.3 (continued)

Buoy Tender (name or class)	Number	Launch Date	LOA (ft)	LBP (ft)	Beam (ft)	Draught (ft)	Displacement (l. tons)	Deadweight (l. tons)	Speed (kts)	Fuel (gal)	Range (nm)	Crew	Buoy Deck	Powerplant	Thrusters	Propellers	Weight Handling Equipment	Other missions and capabilities	Comments	Hull Builder [References]
Canada																				
Type 900 (Proposed)	22	Prop 1988-1989	≤164		est 35	≤9.8	345-493		max 12 econ ≥10		1600 @ econ speed	17-23 + 8 extra	fwd well deck min length: ~50' min area: ~1250 ft ^{2}	500 - 1000 kw 670 - 1340 hp	Multi-directional waterjet bow thruster	2 screws and 2 rudders	Pedestal type crane with max 10 ton lift and 2 x 5 ton aux. lifts, stepped aft of buoy deck	Light icebreaking (up to 6") SAR	Estimated annual operational days requirement is 250-270 days. Unreplenished endurance requirement is 20 days. Sea worthiness requirement is for positive operational control in Beaufort wind force 5, sea state 4. CCG plans for 20 year life-span.	
Type 1000 (Typ. Existing) Tupper	23A	1959	204		42	13.9	1402		max cruise 13.5	65,000	5000	38		2162 kw 2900 bhp Diesel-electric				Light icebreaking Resupply	Typical design for 1000 class-basic design for new ships of this class. Considered a medium navaid tender / ice strengthened	[2.5]
1000 Class (Proposed)	23	Prop 1989	196-213		est 42	≤11.8	985-1970		12		6000 @ 12 kts	23-29 + 6 extra	fwd well deck min length: ~56' min area: ~2150 ft ^{2} fbd ≤4.3'	2000 - 3000 kw 2700 - 4000 hp	Multi-directional waterjet bow thruster	2 screws and 2 rudders	Derrick or crane stepped aft of buoy deck 25 ton main lift, 10 ton second lift, 2 x 5 ton aux lifts	Lightstation resupply Light icebreaking (up to 12") Helipad w/ Portable fueling module Firefighting	Estimated annual operational days requirement is 250-270 days. Unreplenished endurance requirement is 30 days. Seaworthiness requirement is for maintaining control in Beaufort wind force 6, sea state 4. CCG plans for 30 year lifespan.	
Denmark																				
Argus	24	1971	224		39	14			15.5		2500	17	fwd well deck 36' x 54'	Diesel - electric B&W / 1 Alpha 12 cyl 3300 hp	Transverse duct bow thruster 5 kw	1	15 ton Halenborn derrick crane stepped fwd of buoy deck	Lightstation resupply Icebreaking	The "Argus" was built primarily to service the Loran stations in Greenland. Danish government is currently evaluating the multi-mission capabilities of its vessels for cost reduction	
Ireland																				
Atlanta	25	1959	232		40	est 12				~55,500				Diesel-electric by English Electric Co. — 1500 hp		2	12 ton Velle Ship-Shape Derrick/Crane			Philip & Son Ltd; Dartmouth, Ireland
Granuaile	26	1970	265		42	est 14			13	85,000		2 x 34 on/off 28 days	fwd well deck	2 x W.H. Allen 8 cyl diesels 2367 kw total 3200 hp	Bow: Stone-Vickers 300 kw, 5.2 tons thrust	2 x Stone KaMeWa CPPs	12 ton Velle Ship-Shape Derrick/Crane stepped fwd of buoy deck		Vessel is similar to Patricia and Mermaid with luxury accomodations for commissioners on lighthouse inspection tours; producing a large vessel with a small working deck. Buoy chain is pulled with a wildcat on a vertical capstan and fed directly into a chain locker below deck.	Ferguson Bros., Port Glasgow Ireland
Netherlands																				
Rotterdam (B3)	27	1986	146	131	34	9.8			11		2000	1 x 9	aft working deck, 59' x 33' carriers 6 x 8800# buoys w/6 x 6600# sinkers		Bow and stern thrusters	2	10.2 ton crane stepped immediately aft of bridge; wave following	SAR Pollution control Sounding	Vessel is designed for offshore and some inland water work. Total of 3 x B3 buoy tenders. Vessel has dynamic positioning system using Hifix 6 or Syledis. Designed to work buoys in wind force 6, sea state 5 conditions	

TABLE 2.3 (Continued)

Buoy Tender (name or class)	Number	Launch Date	LOA (ft)	LBP (ft)	Beam (ft)	Draught (ft)	Displacement (l. tons)	Deadweight (l. tons)	Speed (kts)	Fuel (gal)	Range (nm)	Crew	Buoy Deck	Powerplant	Thrusters	Propellers	Weight Handling Equipment	Other missions and capabilities	Comments	Hull Builder [References]
Norway																				
Skomvaer	28	1985	144	125	33	12.5		457	11.4	55,000	10,500 @ 10.5 kt	2 x 9 on/off 28 days	fwd well deck; area approx 1500 ft^2	Bergen Diesel KRM-6 gives 995 kw @ 750 rpm	Brunvoll hydraulic fixed pitch tunnel thrusters Bow: 150 kw Stern: 75 kw	1 x Volda-Liaaen type CP 62/450 Diam: 6.5' 4 blades	Moelven DK 150 hydraulic crane rated 2 x 6 tons @ 39', 4 tons @ 79'; telescoping	SAR Pollution control Lighthouse resupply	PTO on main engine drives thrusters and main hydraulic system. Aft thruster too small. Ship has Kort rudder nozzle and Becker flap, electronic remote control system for main propulsion, and a joystick main helm control.	
England																				
Vigilant	29	1978	174	157	36	11.5	765	445	12.5	28,500		16	fwd well deck approx 48' x 34'	2 x G.E.C. / Ruston type 6AP 230 x 6 cyl diesels 680 BHP each @ 750 rpm	Transverse bow thruster 250 BHP motor gives 3 tons thrust	2 x FPP	15 ton Speed Crane derrick stepped aft in buoy deck, 110 ton bow roller	First Aid Salvage Diving ops Firefighting Hydrographic surveying		[2.9]

TABLE 2.3 (Continued)

FIGURE 2.1 US Coast Guard Offshore Buoy Tender: 180' WLB (Ship 1)

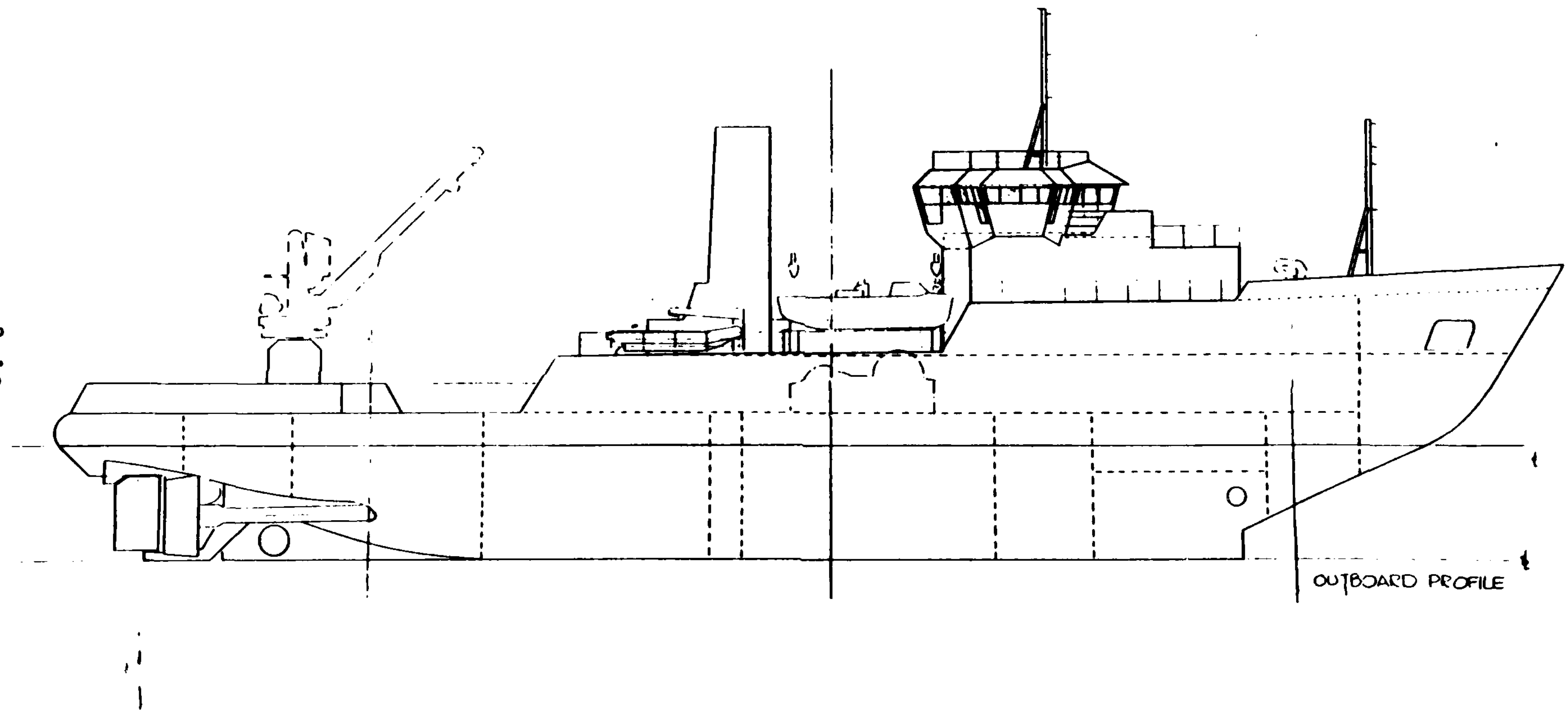

FIGURE 2.2 Canadian Coast Guard Type 1050 Navaid Vessel (Ship 3)

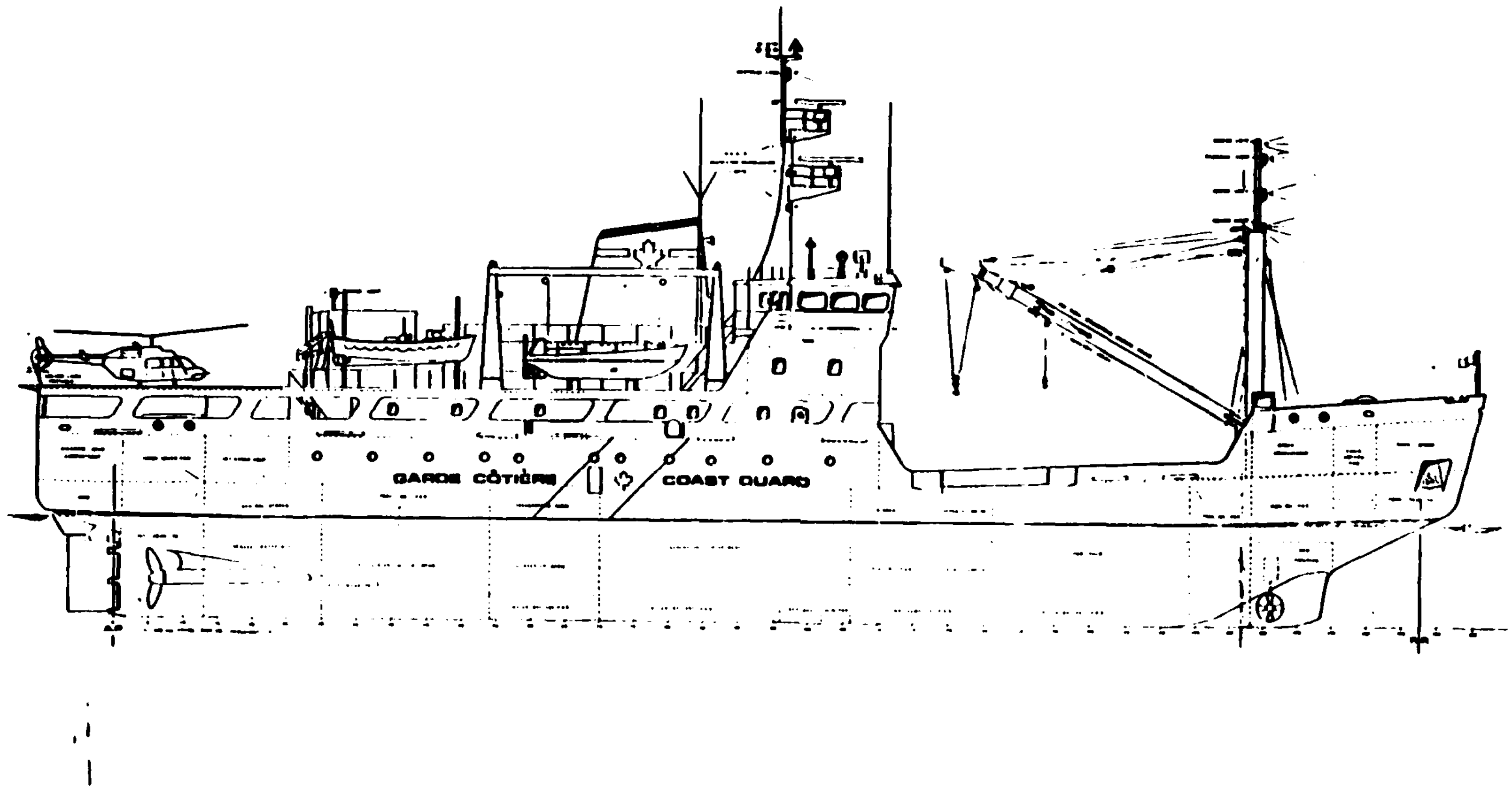

FIGURE 2.3 Canadian Coast Guard Type 1100 Navaids / Light Icebreaking Vessel (Ship 5)

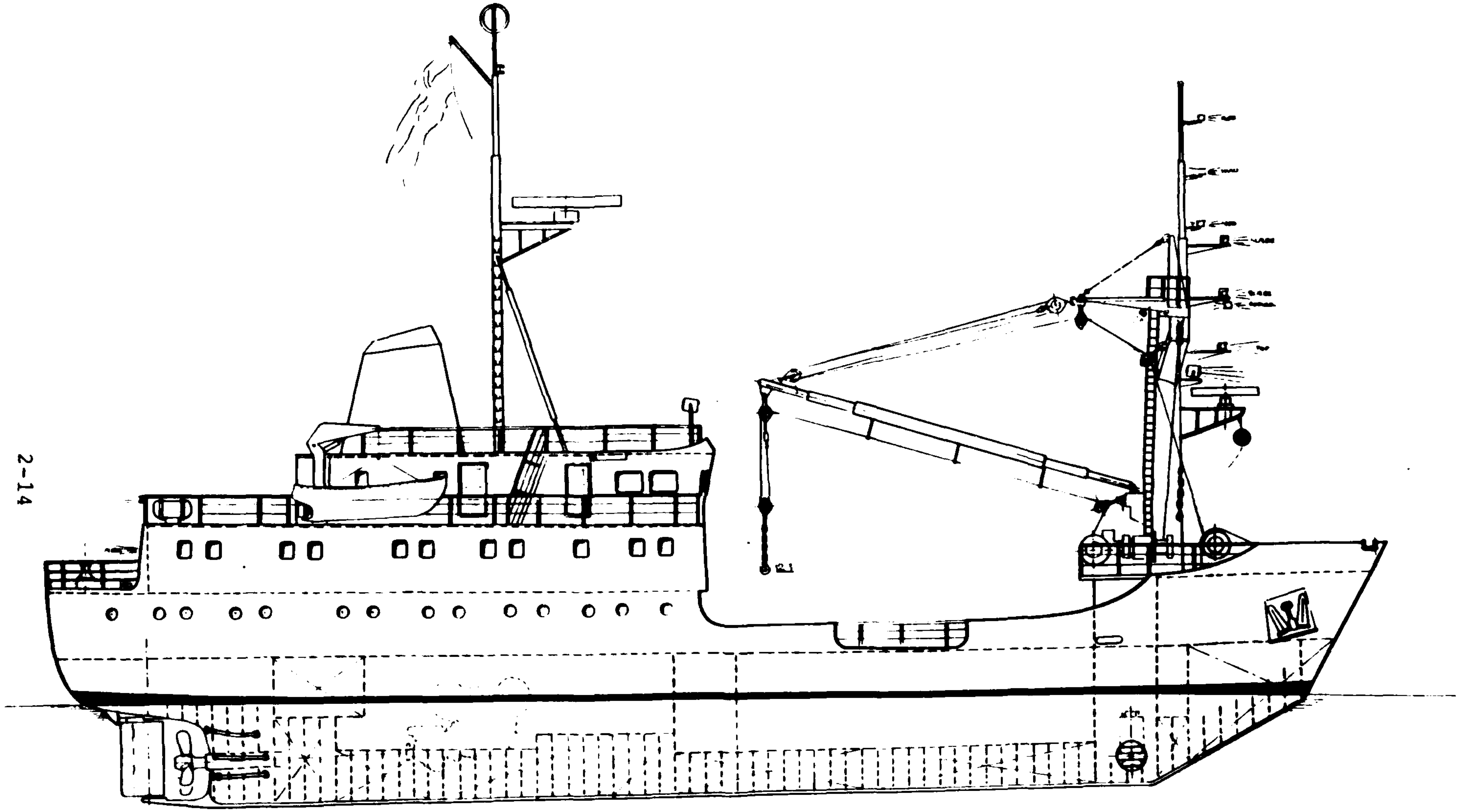

FIGURE 2.4 Danish Proposal 'Inspektiosfartoj 80' (Ship 6)

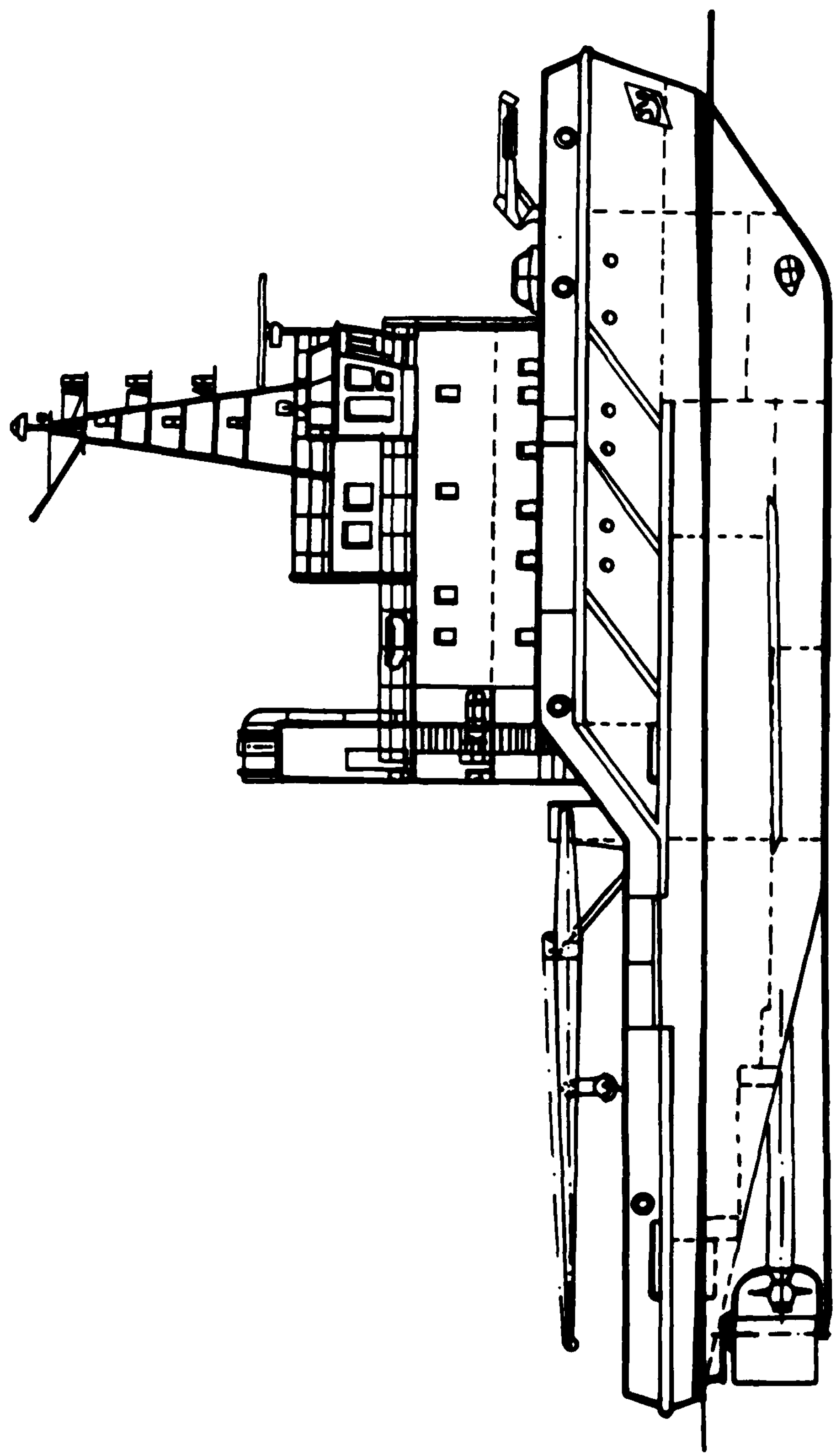

FIGURE 2.5 Finnish Channel Servicing Vessel *Seili* (Ship 7)

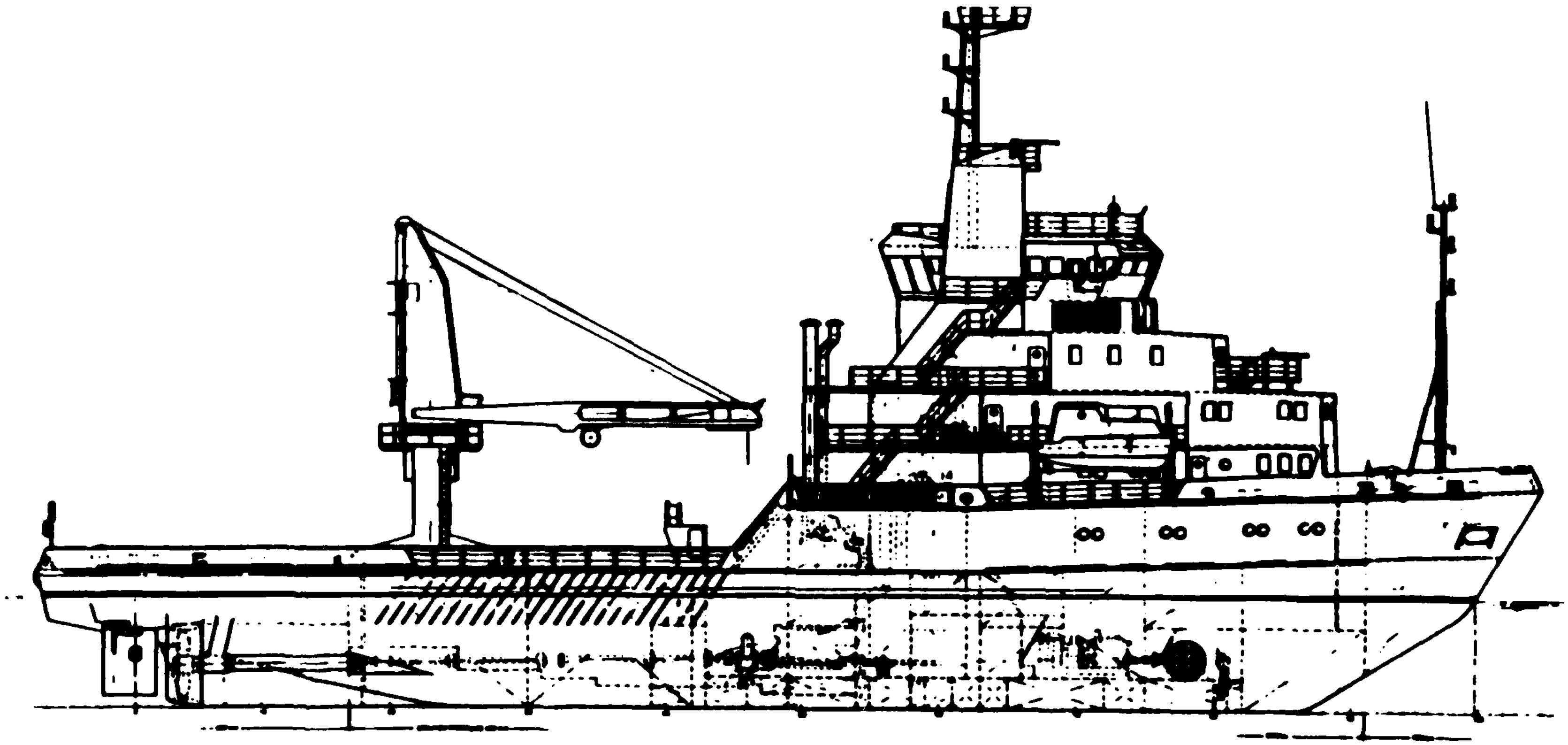

FIGURE 2.6 West German Pollution Control / Buoy Tending Vessel *Mellum* (Ship 11)

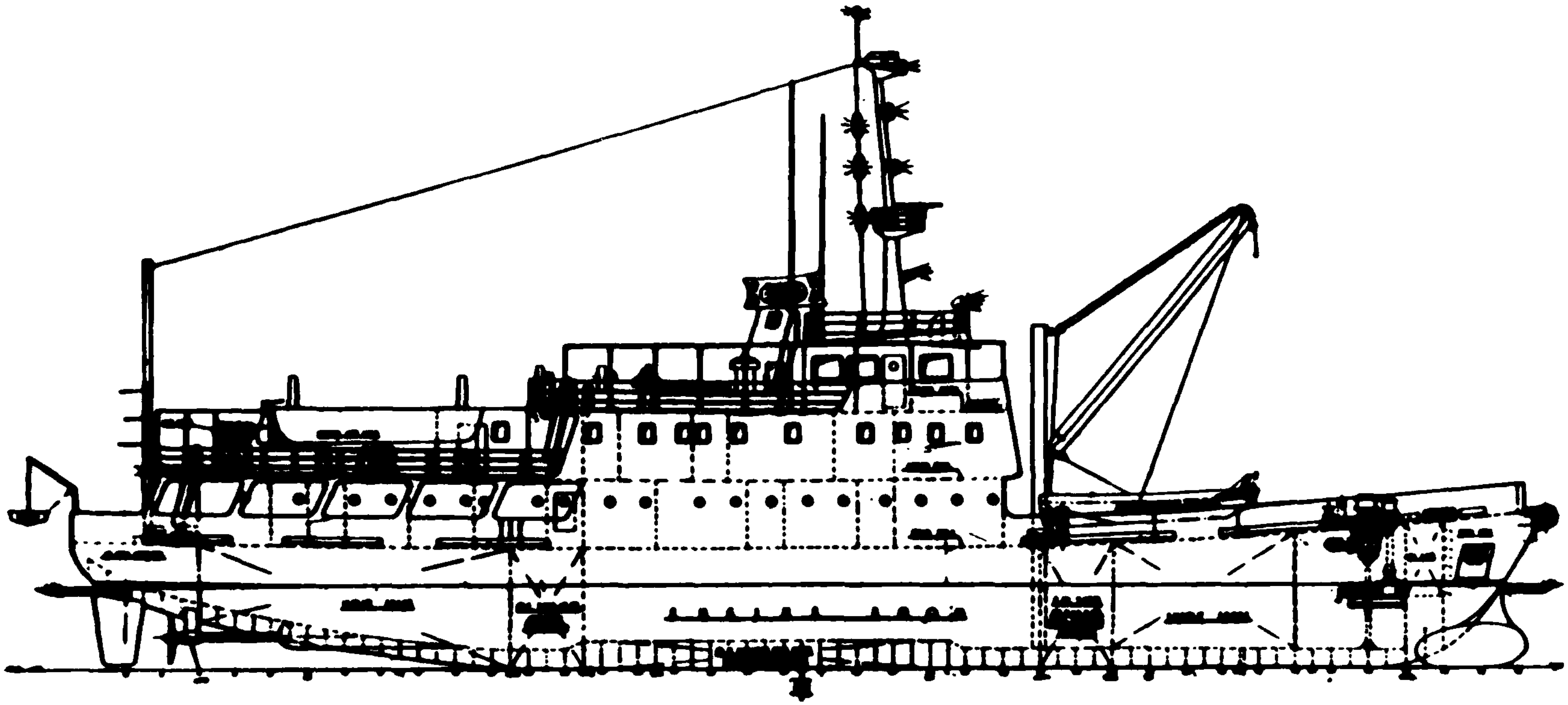

FIGURE 2.7 Indian M. V. *Pradeep* (Ship 13)

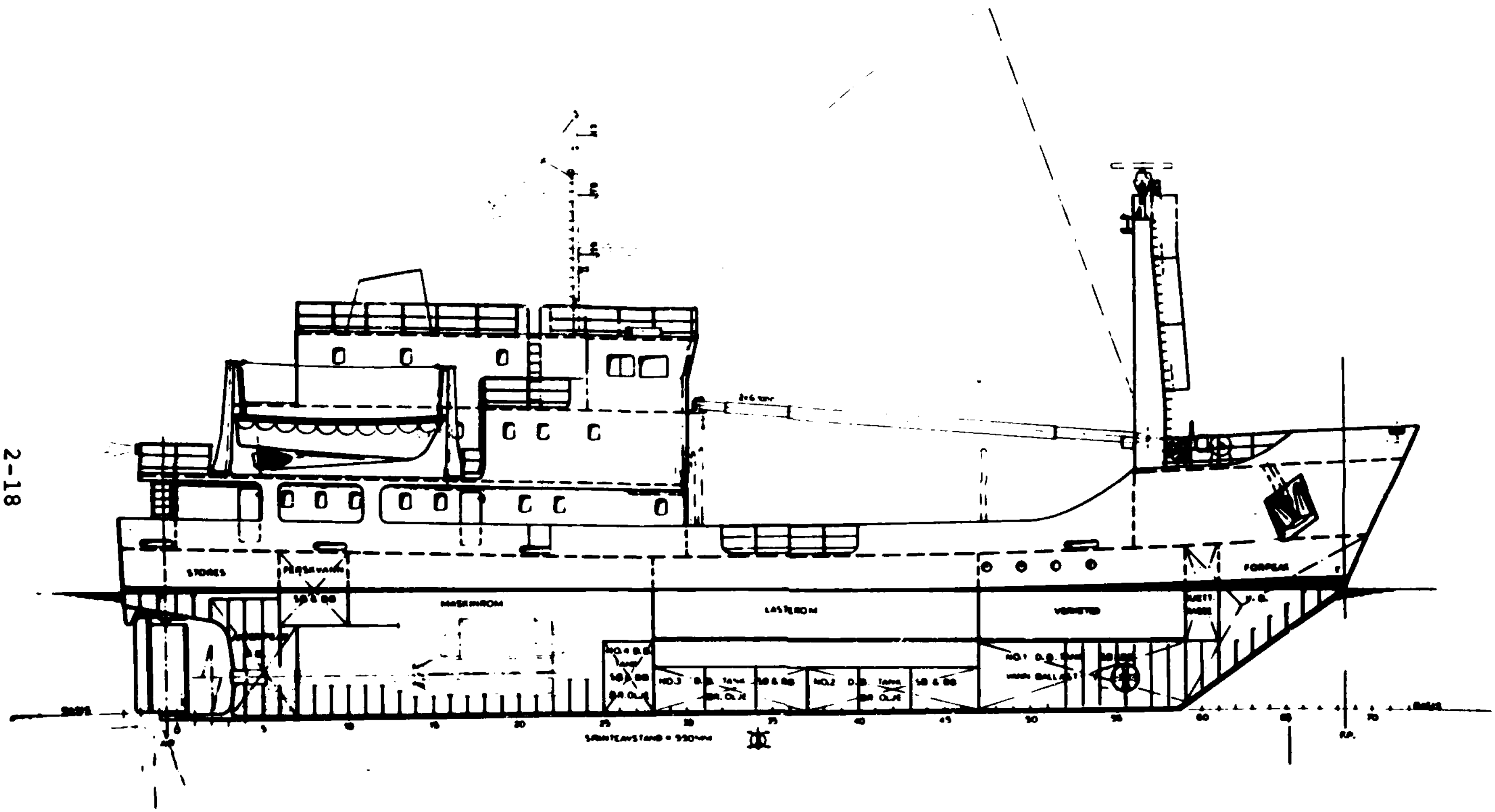

FIGURE 2.8   Norwegian Lighthouse Support / Icebreaker 2946 Class (Ship 14)

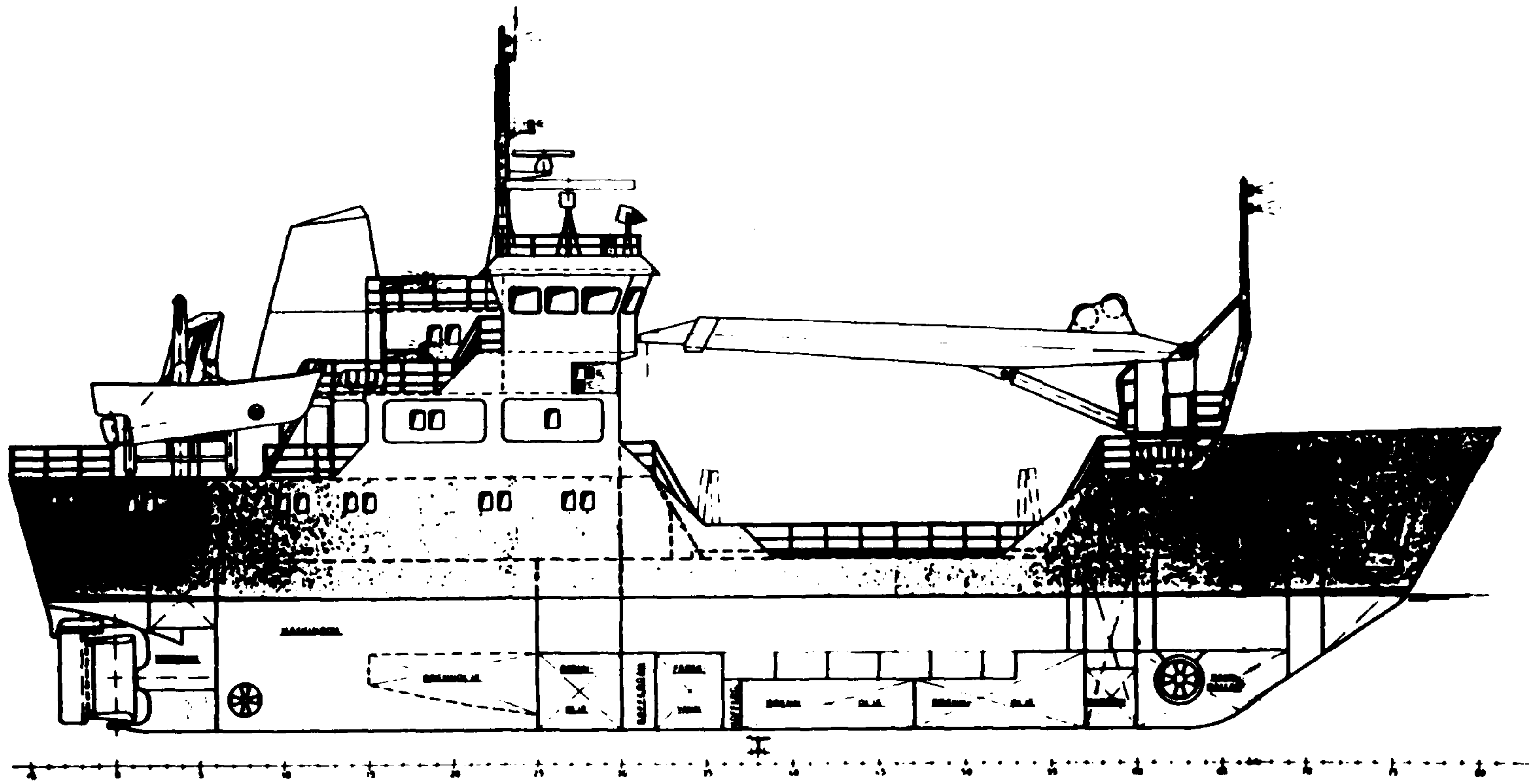

FIGURE 2.9 Norwegian Lighthouse Support / Icebreaker 7488 Class (Ship 15)

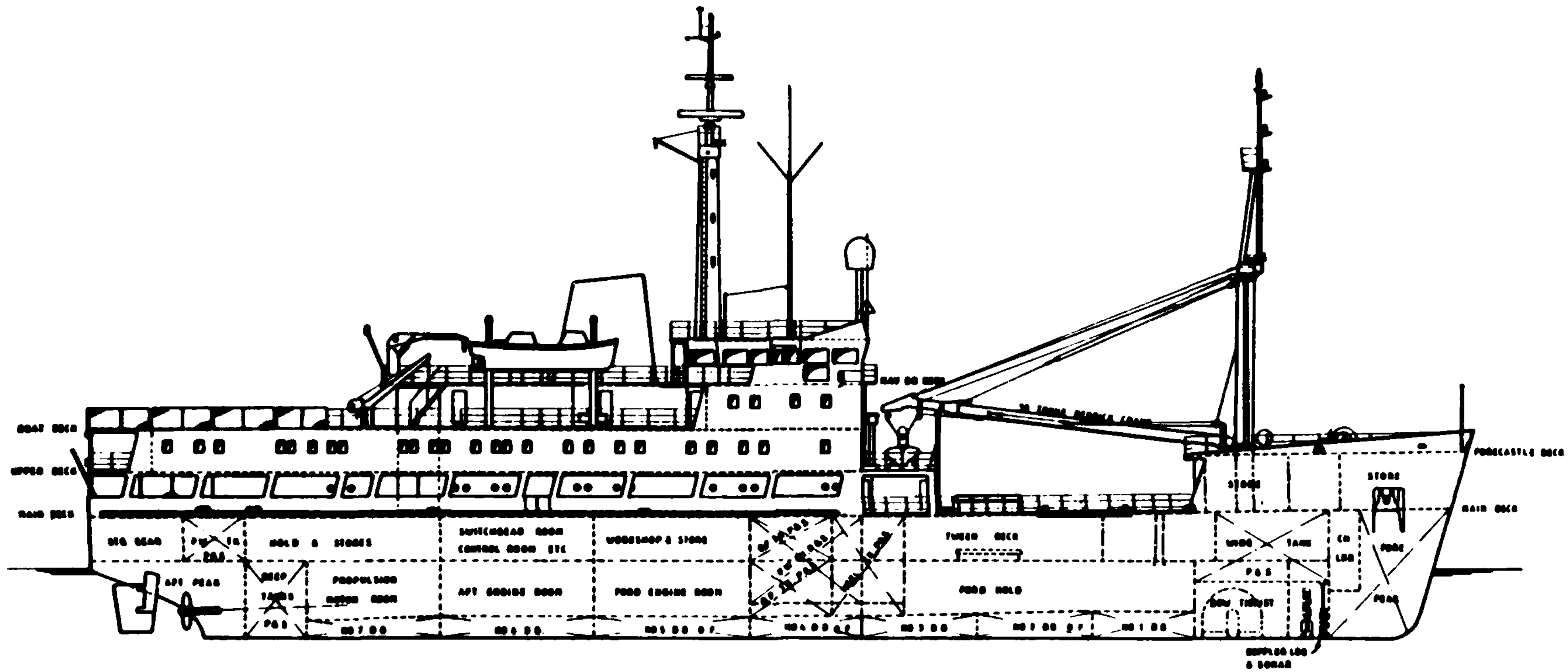

FIGURE 2.10 THV *Mermaid* (Ship 18)

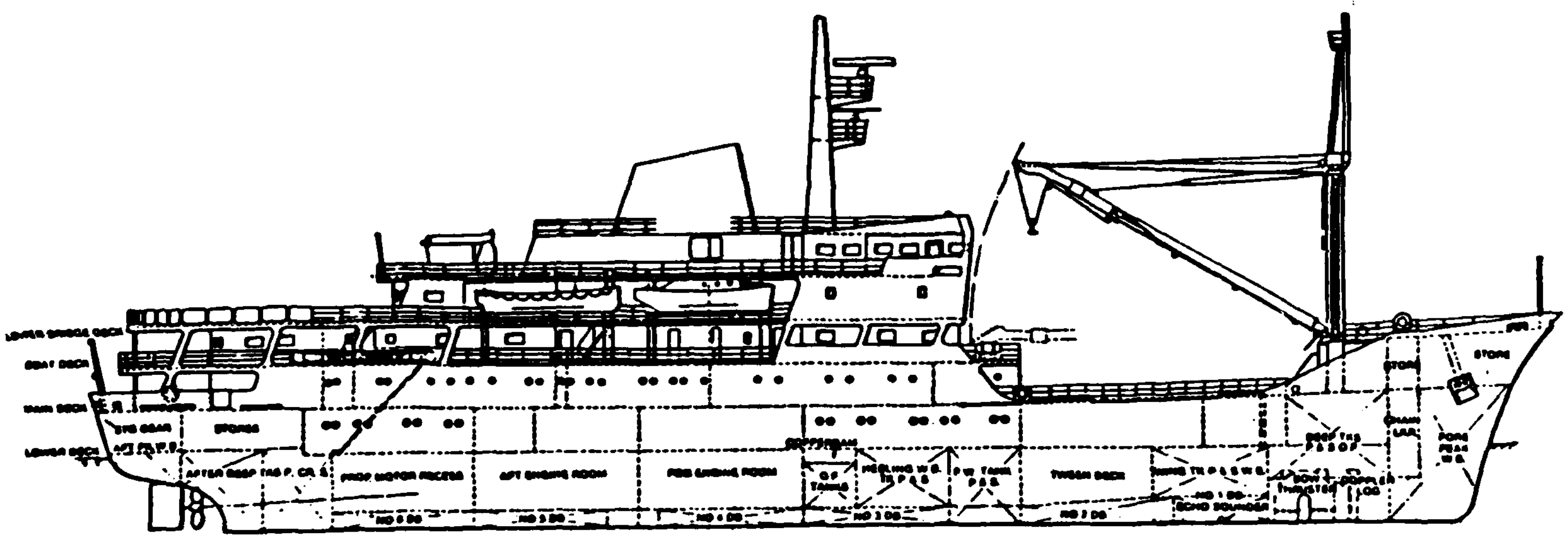

FIGURE 2.11 THV *Patricia* (Ship 19)

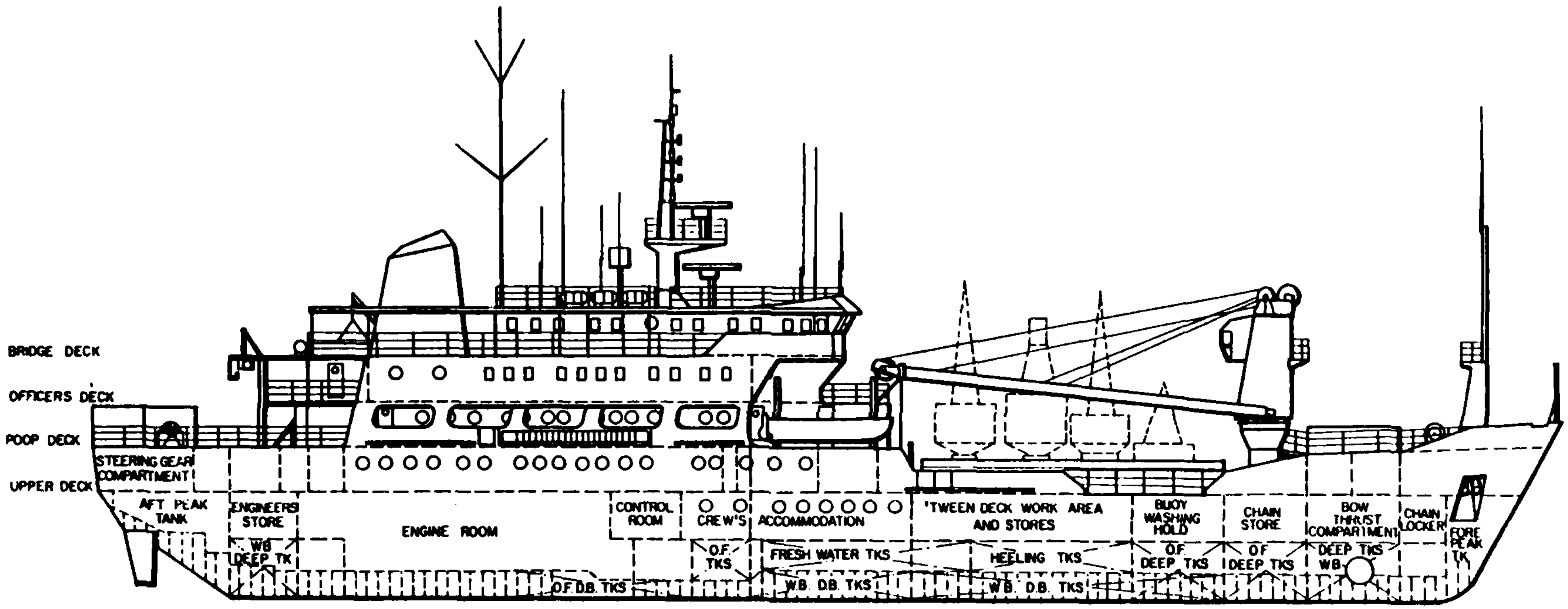

FIGURE 2.12 Middle Eastern Navigation Aids Service Vessel *Relume* (Ship 20)

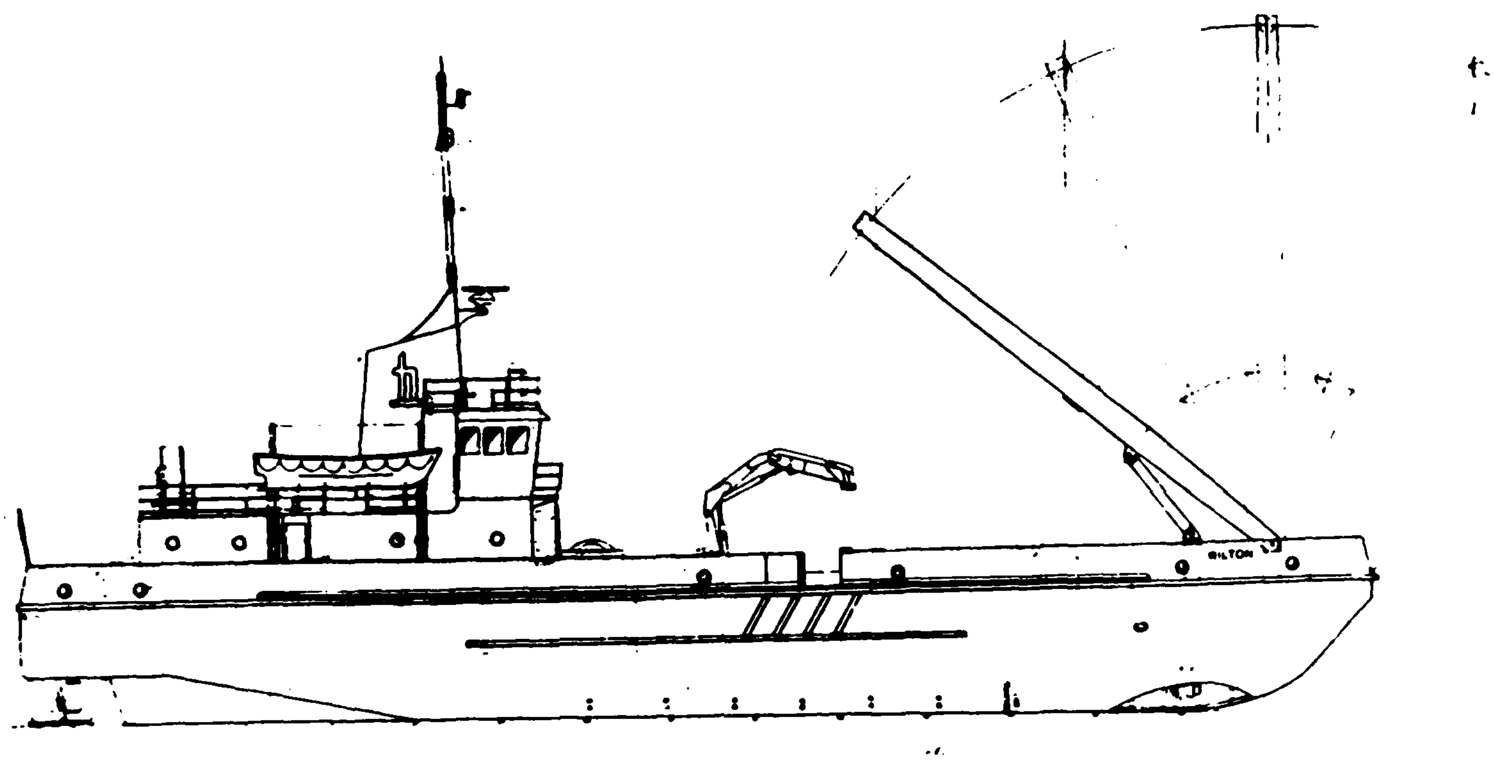

FIGURE 2.13 Tees and Hartlepool Port Authority M. V. *Wilton* (Ship 21)

2.3.1 Offshore Buoy Tenders

Sixty percent (60%) of the vessels fell into the first category of offshore buoy tenders, and are similar in form and function to the current USCG WLBs. The Canadian 1100 class (5), Trinity House vessels, "PATRICIA" (19) and "MERMAID" (18), and Sweden's "BALTICA," (17) are all representative of this type. These vessels range in length from 140' to about 280' with an average length of 200'. Vessel beams range from 30' to just over 50' with a 38' average. The draft of offshore tenders ranges from a maximum of almost 19' to a minimum of 9' with an average of 12.4'. Generally speaking, the larger the principal dimensions of length, beam and draft, the more suitable the vessel may be for transiting and working in offshore conditions, and handling heavy weights over the side. Conversely, the larger vessels must compromise their speed, maneuverability, and economy. (Note: larger vessels may have increased speed with a relatively small increase in power.)

The trend in recently built offshore buoy tenders is towards ships 200 feet and greater in length. This is driven by the desire to maximize operational effectiveness of these vessels. The larger ships offer a more stable platform for a given sea state, and thereby increase the number of days per year on which the buoy tender can operate.

The greater length of foreign buoy tenders also provides a larger capacity for carrying buoys, sinkers and chain and other supplies. This allows the buoy tender to work buoys for a longer period of time without spending time in transit to resupply. As a point of clarification, several buoy tenders in Table 2.3 are longer than might be necessary for the primary buoytending missions. The PATRICIA (19) is also the escort for the Royal yacht and makes annual inspection tours of lighthouses carrying many VIPs for whom generous accommodations are provided. The Canadian Type 1100 (5) buoy tender has an icebreaking function that may also result in the addition of some length and displacement above that required for a single-mission buoy tender.

The Danish buoytender, ARGUS (24), both of the Trinity House buoy tenders (18,19), and the Canadian Type 1100 (5) and 1000 (23) buoy tenders are all equipped for helicopter operations. The ARGUS helicopter capability is intended for servicing lighthouses and Loran stations in Greenland. The Trinity House buoy tenders operate regularly with two helicopters supplying lighthouses. The Canadians routinely use helicopters for servicing lightstations and lighthouses in remote locations inaccessible from shore. The ships have landing light systems and cargo handling systems to move supplies onto the flight deck for the helicopters. The support logistics and helipad both contribute to the increased length of these vessels.

The general configuration of the offshore tenders is with the buoy deck forward and bridge, accommodations and machinery spaces aft. The buoy deck is usually a well deck, aft of a raised forecastle, which gives considerable protection from wind and waves for the working crew. The raised forecastle arrangement also provides convenient storage for servicing equipment and buoy supplies. The flush deck forward arrangement, such as found on Japan's "HOKUTO" (8), is much less common. In this case the area near the bow and forward of the buoy deck is used for buoy storage and deck winch placement. In either case, the buoy deck freeboard is from 4 to 8 feet and the buoys are usually worked over the side of the vessel.

The forward buoy deck configuration is traditionally preferred because the captain can look forward to watch the buoy work, maneuver the ship, and watch other ship traffic in the area. This is particularly important in congested areas and ship channels. It also reduces the possibility of fouling the buoy mooring in rudders and propellers. In addition, buoy work frequently requires the tender to work near shoal water or other navigational hazards. Ship operators usually prefer to approach a buoy marking such a hazard bow first, for ease of maneuverability and further minimizing the risk to propellers and rudders. France, The Netherlands, West Germany, and Canada have tried or are using buoy tenders with the buoy deck aft of the bridge. This configuration has the advantages of providing a larger buoy deck and more protection to the crew. However, this arrangement does require adjustments in operations as conducted by the bridge and buoy deck crews. Buoy tenders with an aft deck are discussed further in Section 2.3.2.

The cranes and derricks on the vessels surveyed are rated between 12 and 20 tons safe working load (SWL). Most of these are stepped forward of the buoy deck. This arrangement precludes the problem of wrapping the boom around the bridge wing (commonly referred to as "right shoulder arms") by allowing a buoy to get too far astern while still attached to the main hook. However, having the derrick or crane stepped aft of the buoy deck (closer to amidships) provides for a more stable lifting hook, since less pitch motion is translated through the mast and boom. In addition the crane mast can be less massive by tying into the existing ship's superstructure for support. Power is either electro-hydraulic or straight electric. The powered derrick or derrick crane configuration is the most prevalent, but the newer, fully pivoting cranes are beginning to be used more frequently in this type of service. Chapter 7 on Buoy Weight Handling Equipment describes the various cranes and derricks on existing tenders in greater detail.

The cargo capacity of some buoy tenders is increased by storing buoys, sinkers and chain in a hold under the buoy deck. The larger numbers of buoys and supplies on board extends the time between port calls for resupply. The MERMAID (18) is an example of a ship with this feature. The three British designed vessels, PATRICIA (19), MERMAID (18), and RELUME (20) have provision for vertical stowage of buoys on the buoy deck. The buoy crane on MERMAID also has a forked head which allows a two-point attachment to the buoy. The buoy can then be lifted vertically with the upper structure passing between the forks until the buoy can be set into the "pocket" on deck. They have found this arrangement improves access to the buoy being serviced and increases the deck stowage capacity.

The main powerplants of offshore buoy tenders are either diesel or diesel-electric. Straight diesel is a more economical installation, but it does not offer the close control, particularly at lower speeds, and flexibility of power distribution found in the diesel-electric power plant. While diesels outnumbered diesel-electrics by about 2 to 1 in the vessels surveyed, more recent designs favor diesel-electric propulsion. Such systems reduce the need for ship's service generators, and may prove more economical in the long run than straight diesel propulsion with 3 or more auxiliary generator sets to run cranes, thrusters, and supply hotel load power.

The number of main engines varied from 1 to 4, with 2 and 4 being preferred. This provides for greater reliability and flexibility of operation. Total horsepower ratings ranged from 980 to 4830. The average is 2175 hp.

A slight preference for twin screw versus single screw propulsion is observed. The single propeller with the shaft bearings and seals all housed within the hull is the least vulnerable to damage, least expensive, and most efficient arrangement hydrodynamically, while twin screws offer greater maneuverability, reliability and (if arranged with counter-rotating shafts and screws) no sideways forces when power is first applied with no headway. Three-blade and four-blade propellers are employed, with near equal distribution. The selection of fixed pitch (FPP) or controllable pitch propellers (CPP) is also equally mixed. The variety of propulsion system installations, shafting and propeller selections produces speeds ranging from 12 to 15 knots in the offshore buoy tenders.

Nearly all of the vessels in this category have bow thrusters for improved maneuverability. These are typically transverse mounted, ducted propellers with horsepower ratings from 200 to 925. They can provide up to 7 tons of thrust. Bow thrusters were installed on our own WLBs during the major renovation. A more recent development is the inclusion of a stern thruster. Sweden's BALTICA (17) and the BREEVEERTIEN (16) from The Netherlands both have stern thrusters having 300 hp and 420 hp, respectively. A unique approach to stern thrust is used by the West Germans, in the form of an active rudder found on the WALTER KORTE (12). The active rudder contains a 200 hp thruster mounted in the normal thrust direction. However, the thrust direction follows the rudder direction and it can rotate $\pm$ 90 degrees. When operated with the bow thruster, it allows the vessel to sidle without rotation.

In order to better portray the range of designs employed in the international buoy tender fleet we can compare the cubic volume versus the installed horsepower. The cubic volume parameter, in this case a simplified calculation, relates to the volume moving through the water and is found by

$$V = \frac{\text{Length x beam x draft}}{100}$$

otherwise expressed as $L^3/100$. These values are plotted against horsepower in Figure 2.14. This plot provides a simple means for observing the range of these two parameters. Generally, the larger the vessel, the greater the $L^3/100$, and the further to right it falls in the plot. The higher the installed horsepower, the further up the plot the vessel falls. Vessels with more efficient or economical installations therefore fall to the lower right, while the less efficient and more powerful designs are up and to the left. The lines plotted are straight line fits to the trends for each of the three types of vessels. The offshore buoy tenders have $L^3/100$ values ranging from 450 to 1800 ft^3 and horsepowers in the 1000 to 5000 hp range. The 180′ WLB (1) is shown to be one of the more conservatively powered and efficient vessels for its size.

The desire to maximize operational effectiveness has led several lighthouse authorities to incorporate automated and work-saving

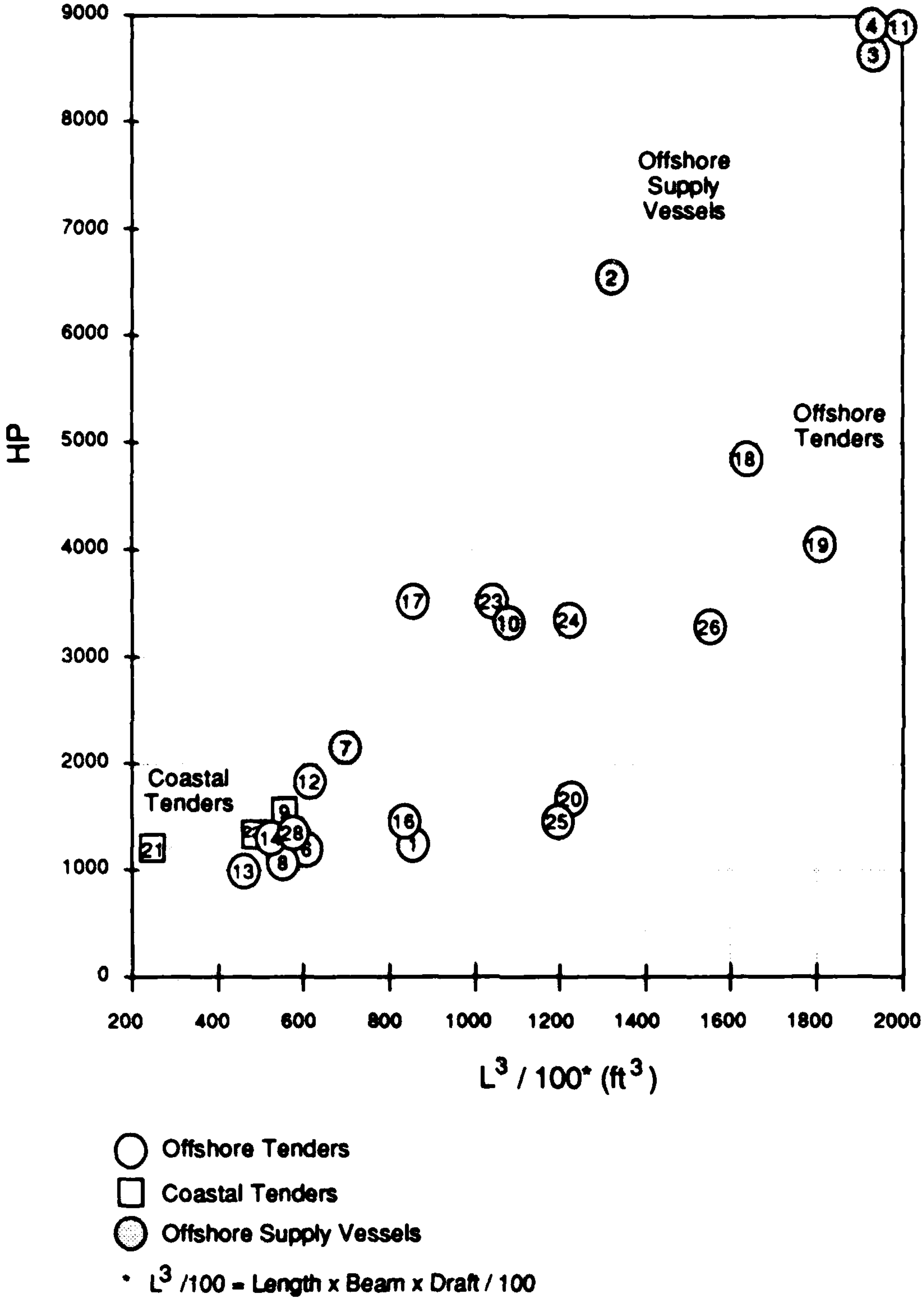

FIGURE 2.14 Buoy Tender Horsepower vs. Volume

systems in the newer buoy tenders. Weight handling systems are a prime example of this. Improved systems for boom rigging, self-contained cranes, motion compensation, cargo hold gantries, multiple whip lines, chain winches and specialized boom heads are some of the innovations recently introduced on foreign tenders to reduce manpower requirements and improve operational capabilities. The Netherlands has installed a dynamic positioning system (DPS) on each of their offshore tenders. The DPS uses one of the navigation systems in the area (Hifix 6, Syledis or Decca) as a reference and controls the thrusters to maintain the ship's position within several yards while the buoy is being serviced. Automated engine rooms with bridge readouts and alarms, remote auxiliary monitoring systems, and integrated steering and control systems are being incorporated into most modern buoy tenders. Taking advantage of these systems allows reductions in crew size and therefore reductions in operating costs as well. Smaller crew sizes are reflected in the data in Table 2.3. While some of these improvements have potential in our own fleet, the current multi-mission tasking of our offshore buoy tenders necessitates a crew larger than that required solely for ATON service work.

Two related factors should be emphasized at this point. First, the U.S. Coast Guard is the only military force to operate and maintain a major aids, i.e., greater than 500 buoys, to navigation system in the world. Second, the Coast Guard is responsible for a multitude of mission areas in addition to its longstanding role in providing maritime aids to navigation.

Due to these factors, plus increasing economic pressures in the past several years to realize "full utilization" of the Coast Guard's limited resources, the U.S. offshore buoy tenders are employed as multi-mission platforms, at times engaging in activities unrelated to aids to navigation. This is in definite contrast with foreign authorities who employ their vessels as single-mission or focused-mission assets. For the most part, the foreign offshore buoy tenders perform floating aids to navigation service work, and any other areas of responsibility are minimal and closely related to the primary mission.

To summarize, the general trend in foreign offshore buoy tenders is toward:

- -Large ships (in excess of 200 feet)
- -Ships with increased deck and cargo space for working and carrying more buoys, sinkers and chain
- -Ships able to operate in higher sea states
- -Ships employing automated and labor-saving devices for reducing crew levels

Foreign offshore buoy tenders are built to maximize buoy working time by minimizing transit time, extending time between resupply, increasing the buoy working weather limit, and reducing operating costs. They are built primarily as buoy tenders, with limited responsibilities in related areas.

2.3.2 Offshore Supply Vessel Conversions

The development and characteristics of offshore supply vessels (OSVs) are described in detail in Chapter 4. This section presents information on OSVs converted to buoy tenders and vessels built from a modified OSV design, employed in navaids servicing. There are a number of reasons why these vessels are attractive alternatives to the traditional buoy tender design. First, they are rugged, seagoing vessels built to carry and handle heavy loads. For a given length of vessel the usual OSV aft working deck is larger than the typical forward buoy deck. The working deck aft and superstructure forward arrangement also provides a better weight distribution for the ship as well as an unobstructed view forward, even when transporting large navigational aids. Recently many OSVs have become available for conversion at relatively low cost due to the slump in the oil market.

This category includes all vessels in Table 2.3 with an aft working deck (ship numbers 2, 3, 4, 7 , 10, 11 and 27). These vessels range in size from 140 feet to 234 feet, with beams from 34 feet to 50 feet and drafts between 10 feet and 17 feet. The installed horsepower ranges from 2100 to 8800 bhp. The power plant is usually a diesel-electric system with two or four main engines driving twin controllable pitch propellers in Kort nozzles. The superstructure causes considerable windage forward.

All vessels surveyed have a bow thruster. Power ranges from 360 to 900 hp for the bow units. About half of the vessels also have a stern thruster for improved maneuverability. A variety of crane configurations have been installed in these vessels. The cranes may be stepped forward of the working deck (nearly amidships), in the center or off to one side. The cranes can usually reach overboard at any point around the working deck and have lifting capacities which range from 6 to 20 tons. Buoys are typically worked over the side of the aft deck.

The JACKMAN (2) was a 184-foot OSV purchased by the Canadian government as a primary SAR resource for the Newfoundland Region. Subsequent testing and evaluation showed the vessel to have good icebreaking capabilities, adaptability to navaids service work and relatively low cost. Design improvements were incorporated into the Canadian Type 1050 and two vessels were built in 1985-86. The vessels (3,4) are somewhat larger than the JACKMAN, with an overall length of 227 feet and are rated as Medium Navaids Tenders/Light Icebreakers. They retain the typical OSV profile with a fully enclosed forecastle and deckhouse forward and a large flat deck stretching from roughly amidships aft. The hull is a single chine form with an icebreaking bow, complete with an ice-knife at the forefoot.

Experience with the Canadian vessels to date has shown them to be adequate buoy tenders for the coastal regions of the Maritimes (Prince Edward Island) and the Central Region (Great Lakes).

However, the low aft buoy deck is extremely wet in any significant seaway. Therefore the Type 1050 is not considered an offshore buoy tender. The Canadians have successfully adapted their operations to the buoy deck aft design. A large hydraulic pedestal type crane weighing 64 tons is located on the starboard side of the buoy deck. Manufactured by Liebherr, the crane has a 20-ton lift capacity, 360° swing and motion compensation. While the crane does provide the required handling capabilities, it is extremely heavy and the large pedestal structure restricts visibility aft from the bridge deck. The Type 1050s are heavily powered, and have very good icebreaking abilities while proceeding ahead, but the square stern and Kort nozzle arrangement make it difficult to operate in the back-and-ram mode necessary for heavy ice. Their high horsepower and deep draft put these vessels in the upper right corner of Figure 2.14.

Overall the Canadians feel the Type 1050 is a good icebreaking vessel which was relatively inexpensive, and has the ability to function as a buoy tender in coastal and Great Lakes areas. However, they continue to rely on the traditional design, that is buoy deck forward of the bridge, for vessels whose primary mission is servicing aids to navigation.

The Germans also bought an OSV, the 184-foot MS OSTERTOR in 1980. Their principal requirement was to develop a vessel for oil pollution control and oil recovery, as required by agreement with other countries bordering the Northern and Baltic Seas. They decided to use the OSV as a test vessel for the layout and design details, before building a new vessel for this purpose. It was necessary to use the vessel for other purposes in order to achieve sufficient benefit from the high cost of the project. After studying the alternatives, it became evident that due to its dimensions, working space, crane, stout construction, and maneuverability, the vessel could operate as a buoy tender and icebreaker. Since the German coastal area has infrequent need for an icebreaker, the emphasis was put on its function as a buoy tender. The oil recovery duties required the spacious aft working deck, and this implied a fundamental change in the handling of buoys. This compromise was accepted, the ship was converted for the new multi-mission duties and recommissioned as the SCHARHORN (10) in 1982.

After successful testing of the SCHARHORN, the Germans initiated construction of a new vessel to carry out the oil recovery, towing and salvage, buoy tending and icebreaking missions. The vessel was equivalent to SCHARHORN, but somewhat longer (234 feet) in order to satisfy the increased requirements for tankage capacity, endurance, speed and towing capability. The new vessel was commissioned as the MELLUM (11) in 1984. A 12-ton mobile gantry crane was fit on the working deck. The engine room was automated, and an integrated joystick control was installed on the bridge in addition to the conventional steering systems. This configuration allows the vessel to operate with a crew of 14. All electrical devices used for oil recovery are explosionproof or intrinsically safe. The Germans now operate three such pollution control/buoy tending vessels and are generally satisfied with the overall performance of the compromise design.

Several countries have experimented with offshore supply vessel conversions and aft deck buoy tenders. This configuration can be made to work for buoy tending. However, in the cases studied, it is apparent that this is a compromise design and the principal motivation for using this configuration is not because it is optimum for buoy tending. The underlying reasons for employing this design in buoy tending include:

- A vessel of this design with another principal mission (such as icebreaking, search and rescue, or pollution control) was available and required fuller utilization to be cost effective.

- A vessel of this design was available at low cost due to the slow-down in the oil market.

The vessels using this configuration are generally employed as coastal tenders due to wetness of the aft deck in a seaway. For the most part, those countries which have used aft buoy deck tenders, still prefer the traditional design, placing the working deck forward of the bridge when the principal requirement is to service floating aids to navigation. Only the Dutch seem to prefer the aft working deck arrangement.

2.3.3 Coastal Buoy Tenders

As indicated earlier in this chapter, coastal buoy tenders similar to the U.S. Coast Guard's WLMs are not frequently employed by foreign lighthouse authorities. This may be due in part to the fact that most countries, particularly in Europe, have significantly smaller buoy tender fleets than the United States. Consequently the tenders they do have must be capable of handling the largest components in their system, in the most exposed locations. The buoy tender designs are driven by the extremes in servicing requirements. Additionally, larger vessels are more affordable, when the overall fleet requirements are small.

Nonetheless the literature survey revealed three vessels in the category of coastal buoy tenders. While The Netherlands type "B3" (27) and the Canadian type 1050 (3,4) function as coastal tenders, they were included in the OSV category due to their working deck aft configuration.

In 1966-68 the West Germans built the GUSTAV MEYER, the OTTO TREPLIN (9) and two more sister ships. Length of these ships is 160 feet, beam is 31 feet, and draft is 10.5 feet. They were built to service the navigational aids in the coastal North Sea and river regions of northwestern Germany. These ships are basically a scaled-down version of the German offshore buoy tenders as a result of their positive experience with this type of design. They are single screw vessels with a controllable pitch propeller, active (powered) rudder and bow thruster. An 11-ton jib crane is stepped forward of the buoy deck. In order to improve the accommodations layout and reduce the machinery noise in the bridge, living spaces and inspector's room, the engine room was placed forward under the buoy deck. The exhaust piping for the main and auxiliary engines as well as the heating boiler, is routed up through the buoy crane pillar post. Two turbo-exhausters were fit and there has been no annoyance due to exhaust fumes, or noise on the buoy deck. A fire monitor was also installed on the crane pillar. This engine room placement has eliminated the possibility of a cargo hold below the buoy deck (not so important on a short-range coastal tender), but has given full satisfaction in accomplishing the noise reduction.

The WILTON (21), a coastal tender owned by the Tees and Hartlepool Port Authority in England, services buoys in rivers, harbors and bays, including one 14-ton fairway buoy. The vessel is 131 feet long with a 30-foot beam and 12-foot draft. It has a large (65 ft x 27 ft) flush buoy deck forward and a hydraulically operated A-frame gantry which has a maximum lift capacity of 30 tons. Buoys are worked over the blunt bow which also has a 125-ton bow roller. This buoy tender has a microprocessor control system which provides automatic maneuvering between buoy stations and dynamic positioning control during buoy work. The GECGEM 80 system is linked to the vessel's Motorola MK3 radio position reference system. The system has a normal 3-axis joystick control and selectable center of rotation - either the bow gantry for buoy servicing, or the vessel midships point for standard maneuvering. The vessel can also be controlled by a single portable control box. Propulsion is provided by three azimuthing thruster units, one at the bow and two at the stern. This arrangement gives exceptional maneuverability, allowing the vessel to move sideways or turn through 90° on short notice. The sophisticated control equipment, together with the relatively simple design of the vessel, allows for satisfactory operation with a crew of 7.

The Canadian Type 900 (22a) is rated as a Small Navaids Tender/Ice-strengthened. The Canadian operating environment tends to be more severe than in the U.S., particularly with respect to the ice season. As a result the buoy tenders are generally larger than required simply for the buoy tending mission. As an example the typical Type 900 in the existing fleet is 189 feet long with a 42-foot beam and 12-foot draft. The tender has a forward well deck, bridge aft configuration. The proposed configuration for the new standard Type 900 (22) is of similar configuration, but about 164 feet long with a draft of less than 10 feet. The forward buoy deck is to have a minimum length of 50 feet, minimum area of approximately 1250 ft^2, and a pedestal type crane rated for 10 tons stepped aft of the buoy deck. The diesel-electric propulsion system will provide between 2700-4000 hp through twin screws. Twin rudders will also be fit. Canadian Coast Guard plans for a 20-year lifetime for the new vessels.

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3.0 AIDS TO NAVIGATION; FOREIGN PRACTICES

3.1 PURPOSE

The purpose of this chapter is to review practices of foreign aids to navigation authorities to determine how offshore and coastal buoy tenders are used within their ATON systems, and how buoy tender design has been driven by present and future operational requirements.

3.2 SCOPE

Buoy tenders are managed with other components of an aids to navigation (ATON) system to provide a service to the mariner. How the buoy tender is to be used within the system drives the design of new tenders. This chapter discusses overall trends in the use of buoy tenders; detailed operational practices or procedures in tending buoys are not discussed. Features and characteristics of foreign buoy tenders are described in Chapter 2 of this report. These features are repeated here only when necessary to support design features resulting from operational requirements of the ATON system.

3.3 BACKGROUND

3.3.1 Data Collection Methods

A literature search was conducted through several information data bases to find articles and reports by lighthouse authorities, offshore operators and others who may have experience in operating buoy tenders or performing similar operations from vessels such as offshore supply vessels. That method of investigation yielded very little information germane to this study.

The publications of the International Association of Lighthouse Authorities (IALA) were searched; IALA Bulletins from 1960-1986 were reviewed as well as the proceedings of the technical conferences, but few articles appeared that discuss the operation of aids to navigation systems with respect to buoy tender design and configuration.

Because of the lack of published information, (both in IALA publications and the open literature) direct contact with IALA members was initiated. The questionnaire in Appendix B was developed and sent to eight countries. Responses were received from Canada, Denmark, England, France, Japan, The Netherlands and Norway. Follow-up visits were arranged to discuss the questionnaires as well as other topics. These countries were selected because: 1) they maintain 70% (excluding the USA) of the buoys on the list of the world's 15 largest ATON systems as shown in Table 2.1, 2) the coastal area of these countries provide conditions that reflect the variety of operational environments encountered by the U.S. Coast Guard's WLB/WLM fleet and, 3) several authorities have recently built new buoy tenders and have analyzed their aids to navigation system with respect to offshore buoy tender requirements. The discussion of foreign practices that follows is largely based on the results of the questionnaire.

Not all IALA countries were contacted and there is undoubtedly other information that could be included in this report. However, the information presented is considered representative and serves to illustrate the major trends.

3.3.2. Foreign Authorities And Aids To Navigation Systems

A few introductory remarks regarding the characteristics and nature of the foreign authorities will provide a background and framework for interpretation of information that follows.

The force mix of most foreign buoy tender fleets does not have the depth of the USCG fleet, thus direct comparisons or projections must be made with care. There are several reasons for this disparity. Most systems have at most several thousand buoys and need only a few tenders. These tenders must be suitable for offshore conditions and handling large buoys. Often the only other ATON craft in the system besides the offshore tenders are small craft that have little or no lifting capability that are used for light maintenance and discrepancies. There are rarely vessels of intermediate capability.

In some countries local harbor or port authorities operate small buoy tenders in their harbors independent of the larger national authority. This means that while there are coastal buoys and buoy tenders, they do not appear as part of the overall ATON resources of the country.

Most foreign ATON authorities are not multi-mission in nature. With some exceptions, they are structured and managed for only one reason, the maintenance of buoys and structures. How they design, staff and operate their buoy tenders is a function solely of ATON requirements. There is very little involvement with search and rescue, pollution control, law enforcement, fisheries patrol or other missions.

3.4 DEPLOYMENT OF OFFSHORE AND COASTAL BUOY TENDERS

Offshore buoy tenders are deployed to maintain large buoys, especially those in offshore areas where work platform stability and heavy lift capability are essential. In ATON systems with several sizes of buoy tender, the maintenance of near-shore buoys is handled by smaller near-shore tenders of less lifting capability, perhaps 60-90 feet in length. Servicing of in-shore aids is handled by service craft up to 50 feet in length that have little or no lifting capability. Some ATON systems have few intermediate capacity buoy tenders and large tenders fulfill all heavy lift requirements whether offshore or in-shore.

To maximize the use of the offshore buoy tenders, these vessels operate with two crews that may rotate on for 14-30 days and work extended work days. The trend of multiple crews on many large buoy tenders is shown in Table 3.1. This is discussed in more detail in the following section. Transit times are reduced by putting into a convenient port near the current work area where the crew and supplies meet the vessel rather than returning to homeport for resupply and crew change. This method of operation is enhanced by the larger size and extended operating capacity of that vessel.

Extensive buoy maintenance is performed in central buoy maintenance facilities located at appropriate locations around the country. This is intended to relieve the tender of this kind of work and keep it underway as much as possible to maximize the use of its unique capacity (i.e., heavy lift capacity in a seaway). Buoy tenders perform light maintenance (e.g., refuel and refresh paint or retroreflective tape) during regularly scheduled visits to the buoys. The Netherlands has recently organized the ATON system around four modern buoy maintenance facilities where buoys are moved from the vessel and efficiently refurbished at the dockside facility. This releases the buoy tender for working buoys.

Most foreign authorities use conventional aids to navigation. Trinity House and the Netherlands use articulated beacons and jetted piles; however, these have not had an effect on the design

TABLE 3.1. Foreign Practices Survey Summary

	Canada	Denmark	England	France	Japan	Netherlands	Norway	U.S.[1]
Requirements for Buoy Tender Ships								
No. of buoys serviced	N/AV[2]	326	194	200	939	200	700	141
Coastline services (miles)	300 - 800	184	370	200	957	200	570	268
Utilization (days/year/ship) ATON	230[3]	300	299	114	192	205	—	110
Ship maint.	N/AV	30	15	N/AV	135	20	14	115
Other missions	N/AV	30	—	N/AV	3	—	—	60
Training	N/AV	—	—	N/AV	35	—	2	20
Crew size	40	17	25	18	24	16	9	48
No. of crews	1[4]	2	2	N/AV	N/AV	2	2	1
Work schedule	8hrs/day	14 days 9 hr/day	14 days 10 hr/day	N/AV	N/AV	14 days 10 hr/day	30 days, 10 hr/day 6 days	(6)
Service Requirement per Buoy								
Buoy serviced (Year)	1	.5	.5 - 1.5	.5 - 2	2	.5	1	1
Mooring inspected (Year)	1	.5	1	1	2	.5	1	2
Relief for major service ashore (Year)	3 - 4	2	4	4	2	2	N/AV	6
Buoy power supply	Battery	Battery	Battery	Acetylene	Propane; going to solar	Battery WATG (5)	Battery	Battery

(1) All US data in this Table refers to WLBs on the East coast
(2) N/AV — Information not available
(3) Utilization with one crew. Going to two crews on same vessels; also icebreaking in winter.
(4) For Canadian buoys not removed every year for ice conditions
(5) WATG — Wave Activated Turbine Generator
(6) The USCG is the only military organization on this Table. Because of the difficulty of comparing civilian and military work schedules, none is reported for USCG

or use of their buoy tenders. Solar powered buoys are being implemented routinely or on trial bases by several authorities. No special handling procedure is required; the two hooks on the boom can be used to lift a solar buoy straight up, thereby reducing the likelihood of damaging the solar panels. The real impact of solar power may be in extending the service cycle for the buoy. Information presented in the following section suggests that many buoys are visited most often to service lighting equipment. Increased efficiency and reliability in lighting equipment may require fewer visits with the buoy tender and thereby reduces total utilization of the buoy tender.

The replenishing of lighthouses is not a predominant function for most offshore buoy tenders. A notable exception is Trinity House which does replenish lighthouses and lightvessels with buoy tenders, often in concert with two helicopters. The buoy tenders have specific design features for landing, refueling and supplying helicopters. Helicopters dedicated to ATON work rotate crews and provide supplies to lighthouses and lightvessels.

Some offshore buoy tenders carry work boats, small craft or inflatables that can be launched from the buoy tender to service several buoys at once if no lift of the buoy is required. In the few situations where the tender is used to replenish lighthouses or structures, launches are available to ferry personnel and supplies ashore.

Horizontal sextant and visual bearings are used as the primary method of positioning buoys in many areas, with electronic systems used secondarily, as available. France, the Netherlands and, to some extent, Denmark, use electronic navigation systems like Hifix 6, Decca and Syledis. Position accuracies of 3-30 meters are given depending on location and time of day. Generally, there are no standards for the position accuracy of the buoys nor are there widespread guidelines for the correction of discrepancies. Since virtually all authorities are civilian operated and are usually required to operate within a union contract, responding to discrepancies on weekends and outside normal operating hours is a contractual matter.

Generally, it is very difficult to discriminate between the offshore and the coastal buoy tender function in foreign fleets because both types of tenders are often performing the same operations and managed as one resource group. The vessel of the next lower capability is far below the offshore/coastal tender group capability; this lack of vessels of intermediate capability leads to the observation earlier in the report that the force mix is thin. It is not usually appropriate, therefore, to draw direct comparisons between U.S. and other ATON systems.

The Netherlands and Canada are notable exceptions to the general observation that most foreign authorities do not have a depth of force mix. The Netherlands has three classes of tenders: B1, B3, B4. The B1 class is the large offshore buoy tender (the

BREEVEERTIEN) at 205 feet in length and 12-1/2 tons lifting capacity. The B3 class (there is no B2) is 144 feet long and can lift 10 tons. The B3 class normally would work nearshore buoys but can, under certain conditions, do everything that the B1 class can do but it carries a smaller number of buoys. The B4 class (99 feet long, 7-1/2 tons lift) is intended for river and harbor use; however, it can work up to 15 miles offshore. This force mix is the result of a comprehensive study of the overall ATON system. The new modern buoy maintenance facilities discussed above are intended to complement this mix of tenders.

The Canadian ATON fleet has depth of force mix also. The coastal buoy tender class is a grouping of tenders with similar characteristics rather than a single design. Requirements documents have been developed for two new classes of buoy tender that will standardize one design. The Type 900 is limited to 150 feet and 10 tons lifting capacity and the new Class 1000 is limited to approximately 190 feet and 25 tons lifting capacity. Both vessels will have the buoy deck forward of the bridge.

3.5 UTILIZATION

The information in Table 3.1 is an indicator of the usage of offshore buoy tenders. It must be understood that even though most of these figures were furnished by the foreign authorities, they may be approximations and there is much latitude in interpretation in some cases due to differences in terminology, language and utilization accounting procedures. Accepting the limitations of the information, it is still useful as a whole as an insight to how the buoy tenders are used. All USCG data refer to the WLB because the overwhelming trend in foreign ATON systems is toward large buoy tenders and it would be *misleading* to include WLM characteristics in a table with much larger buoy tenders. There are few coastal buoy tenders in the foreign fleet as compared with offshore buoy tenders and there is insufficient information to warrant providing a separate table for coastal buoy tenders.

3.5.1 Buoy Tender Area Of Responsibility And Workload

Even though the size and extent of the ATON systems of the foreign authorities interviewed are different, it appears that the offshore buoy tenders have comparable usage across the various systems. The number of buoys worked by a typical buoy tender in each ATON system and the amount of coastline within the buoy tender's responsibility are shown in Table 3.1. With a few exceptions that could not be verified, use of buoy tenders is reasonably consistent. The information shown for the U.S. in Table 3.1 is an average of six WLBs operating on the East Coast of the U.S. This area is representative of the conditions in which most of the foreign authorities operate.

3.5.2 Mission Utilization

The data in Table 3.1 show the approximate number of days per year that offshore buoy tenders of the foreign authorities spend on various missions. Not surprisingly, ATON work is the primary mission of virtually all the authorities except Canada. Icebreaking is the major mission for the largest Canadian buoy tenders during the winter season; most buoys being removed prior to ice season. The Canadian data, then, are a combination of summer ATON work and winter icebreaking.

Ship maintenance time shown in Table 3.1 is often taken in large blocks when the crews are on holiday with the intent of disrupting the ATON schedule as little as possible. Comparison of the WLB utilization with other offshore buoy tenders is complicated by the fact that WLBs are operated by military personnel and all others discussed in this report are operated by civilians whose time is accounted for in different manners.

As discussed earlier in this report, many ships have two crews in an effort to increase the days of utilization and therefore receive maximum benefit from the large capital investment in buoy tenders. The crew number, size and workday appear in Table 3.1.

The data in Table 3.1 suggest that some authorities visit their buoys more often, usually to service lighting equipment (often powered by gas). Improved efficiency of lighting equipment is considered by many authorities to be essential for improved efficiency of the ATON system. Some authorities are moving toward battery power (perhaps with solar power also) in an effort to extend service intervals of lighting equipment and reduce the need to service the buoy with the offshore buoy tender. (Since most authorities prefer to lift the buoy on deck to service the lighting equipment, this also provides an opportunity to inspect the top of the mooring.)

4.0 OFFSHORE SUPPLY/SUPPORT/WORK VESSELS

4.1 PURPOSE

The purpose of this section is to survey the existing fleet of Offshore Support Vessels (OSVs) and their technology with an eye towards adapting the applicable characteristics of the vessels, or perhaps the vessels themselves, to the short range aids to navigation (SRA) task. There are several reasons to focus on the OSV fleet. First, relative to buoy tenders, it is the largest fleet of vessels of a similar size and operating environment. Second, the missions performed by OSVs resemble those of buoy tenders in many ways. The market for offshore services is also highly competitive, which encourages innovation, cost reduction, and increases in productivity from designers, builders, and operators. Finally, other governmental authorities with SRA responsibilities have adapted OSVs to perform buoy tending and other Coast Guard missions.

While the main thrust of this section is to examine OSVs; oceanographic and research vessels, and certain patrol, pollution control and other vessels performing over-the-side operations in the coastal environment were also examined.

4.2 BACKGROUND

The following two sections rely heavily on Reference 4.1, an excellent introduction to field.

4.2.1 What is an OSV?

Public Law 96-378 of October 6, 1980 defines an Offshore Supply Vessel thus:

1. is propelled by machinery other than steam,
2. is not within the description of passenger carrying vessels in Section 1 of the Act of May 10, 1956 (70 Stat. 151),
3. is of more than 15 and less than 500 gross tons and
4. regularly carries goods, supplies, or equipment in support of exploration, exploitation, or production of offshore mineral or energy resources.

The variety of vessels employed in support of the offshore oil industry can be confusing, but the majority of the vessels can be classified into a few major types. The first type is the Inshore Crewboat, a light scantling vessel from 30 to 65 feet in length, generally an aluminum planing hull design. It is generally used to transport men and lightweight supplies, such as spare parts and consumables, to and from offshore rigs within 25 miles of shore.

The next major type is the Offshore Crew/Utility Boat. This class ranges from 60 to in excess of 120 feet and often has a light scantling planing hull. Its size allows it to carry fuel and drill water to the rigs in integral tanks, and a large flush deck aft enables it to carry greater quantities and larger packages of cargo. Both classes of crewboats are subject to competition from helicopters (in spite of the very high cost of helo operations), particularly in rough weather areas, and where distances from shore to platform are great. Neither class has much relevance to WLB/WLM replacement and they will not be discussed further.

The real workhorses of the OSV fleet are the supply vessels. They are sturdy, beamy vessels with heavy scantlings and displacement hulls. They are single deck vessels with the house and superstructure well forward, devoting the aft two-thirds of the deck to cargo space. Cargo handling gear is generally not fitted, as the vessels are unloaded by platform mounted cranes. Integral tanks are provided below decks (if needed, additional portable tanks will be carried on deck) for the carriage of fuel, potable and drill water, dry and wet muds, and special drill fluids like acid or calcium chloride. They are capable of carrying large quantities of drill stem, casing, tubing, and other heavy, bulky items of oilfield equipment.

There are many variations on the basic supply vessel. They are often fitted with towing winches for short rig moves, or with stern rollers and "tugger" winches for handling the anchors of pipe lay barges and semi-submersible rigs, leading to the designation of "Tug/Supply Vessels" and "Anchor Handling/Tug/Supply Vessels". Such vessels have been included in this survey; however, pure tugs have been largely excluded (references to seagoing salvage tugs have been included, but no special effort has been made to look for them). Many specialized vessels have been built on or converted from the basic OSV hull, including Geophysical, Construction and Diving Support, Well Stimulation, and Workover Vessels, and they have been included in the survey.

4.3 CHARACTERISTICS OF EXISTING VESSELS

The two sections which follow rely heavily on Reference 4.1 and Reference 4.2, both of which should be consulted for further information.

Today's supply vessels range in length from less than 160 feet to as great as 240 feet, although the 180 foot boat is more or less standard in the U.S. Beam runs from 30 to 50 feet, and depth from 10 to 16 feet. Power is from twin 1000 to 2500 horsepower diesel engines. Vessels are twin screw for added maneuverability and backup power in the event of engine failure. Two generators are generally installed. Each one is capable of handling the average daily electrical requirement on board. The other provides backup capability, or through a split bus, added power when required. (Non-U.S. vessels tend to have much higher generator capacities, and a third emergency genset, probably due to the greater use of electrical auxiliary equipment, bow thrusters, winches, etc.) The size of the average supply boat has increased over the years as drilling moved into deeper, rougher waters further offshore. Its arrangement continues to be much as described for the earlier vessels, except for the increase in size, and the placement of the pressurized bulk tanks below decks, inset into the fuel and ballast tanks.

Recently built supply boats fall into two basic categories: the straight cargo carrying vessel, and the tug/supply vessel. The latter is of the same basic design as any supply vessel, but with the addition of a large towing winch and stern roller. Sometimes they have additional chain lockers for carrying the anchor chain of the rigs they service. The towing winch is placed just aft of the forecastle on the main deck and is usually independently diesel driven (as are most other auxiliaries on U.S. vessels). These vessels are usually the larger hulls, (180 ft and up) with power ranging from 3000 to 5000 hp. They are used for towing drill rigs of various types on long distance moves and they handle the rig anchors when positioning the rig on location. The smaller vessels are sometimes classed as tug/supply, with winches and stern rollers, but are used primarily for cargo service with some anchor handling duties. Figure 4.1 shows typical straight supply boats.

Data on OSVs were collected as part of this survey, and entered into an automated data base. Selected characteristics on several vessels are presented in Table 4.1. Tables 4.2, 4.3, 4.4 present breakdowns of the fleet by size, horsepower and age.

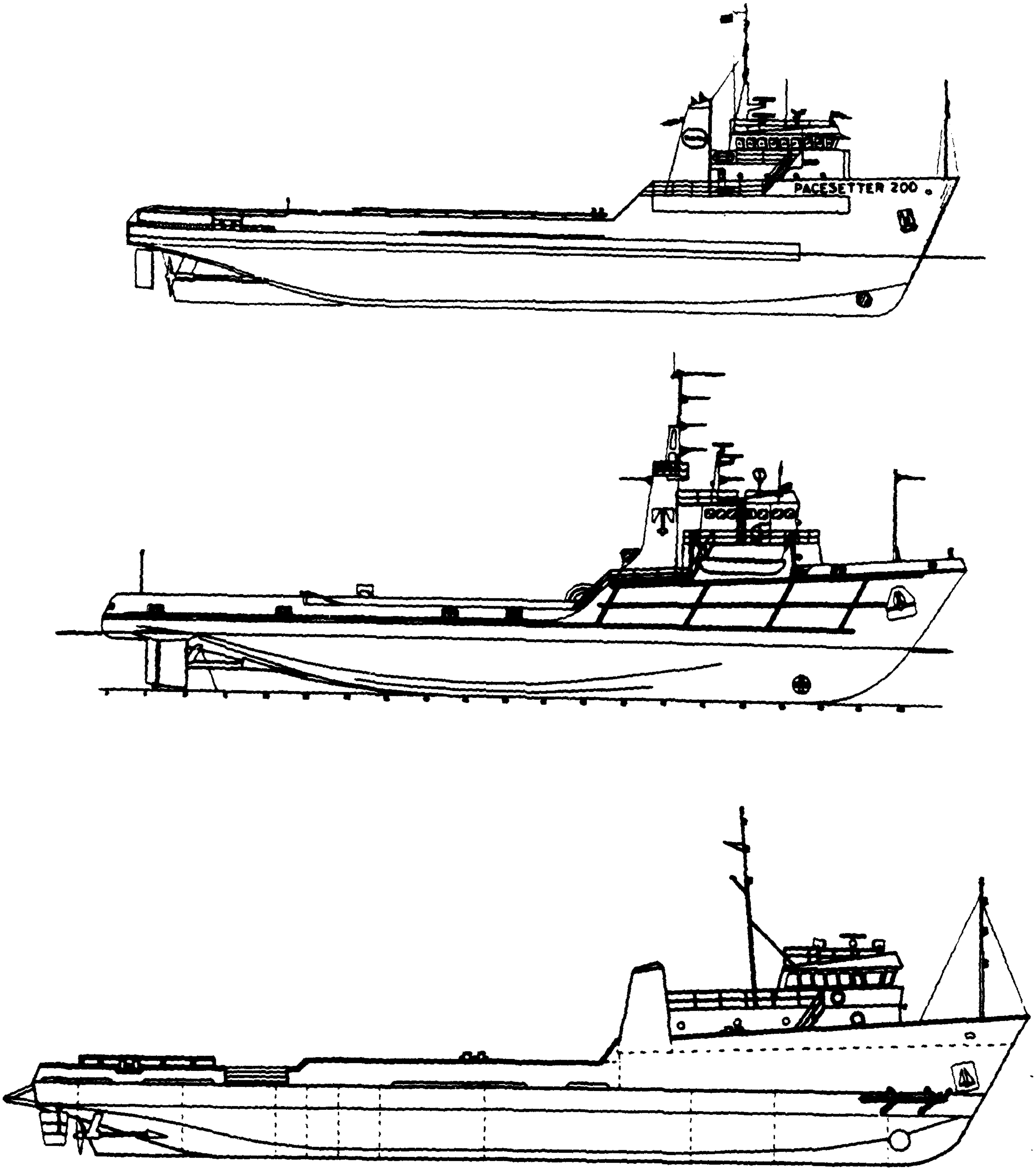

FIGURE 4.1 Typical Offshore Supply Vessels

TABLE 4.1

DESIGN PARAMETER RANGES FOR OSVS

	SUPPLY VESSELS		TUG/SUPPLY VESSELS	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
Brake Horsepower	675	6000	1700	9450
Cubic Number	225	2350	700	2525
Length/Beam Ratio	4.05	5.04	4.30	5.24
Beam/Depth Ratio	2.00	3.35	2.00	3.17
Block Coefficient	0.64	0.70	0.60	0.65
Prismatic Coefficient	0.71	0.78	0.66	0.73
Lightship VCG/Depth	0.76	0.89	0.76	0.91
	(ave 0.82)		(ave 0.83)	

TABLE 4.2

U.S. OSV FLEET BREAKDOWN BY SIZE

TYPE =	Supply	AH/Supply	Tug/Supply	Total
size(ft)	Number of Vessels in 1986 Fleet			
160-169	156	10	4	170
170-179	61	18	11	90
180-189	214	34	99	347
190-199	36	32	111	179
200-209	4	5	39	48
210-219	6	3	22	31
220-229	0	2	12	14
230-239	0	0	1	1

TABLE 4.3

U.S. OSV FLEET BREAKDOWN BY HORSEPOWER

TYPE =	Supply	AH/Supply	Tug/Supply	Total
Horsepower	Number of Vessels in 1986 Fleet			
1000-1499	7	0	0	7
1500-1999	195	1	1	197
2000-2499	197	16	16	229
2500-2999	18	4	1	23
3000-3499	61	20	58	139
3500-3999	15	30	57	102
4000-4499	6	7	28	41
4500-4999	0	7	34	41
5000-5499	0	10	1	11
5500-5999	2	2	46	50
6000-6499	0	1	21	22
6500-6999	0	0	0	0
7000-7499	0	2	16	18
7500-7999	2	3	6	11
>8000	0	1	7	8

TABLE 4.4

U.S. OSV FLEET BREAKDOWN BY AGE

TYPE =	Supply	AH/Supply	Tug/Supply	Total
Year Built	Number of Vessels in 1986 Fleet			
1964	1	0	0	3
65	8	0	0	11
66	15	2	0	2
67	11	1	0	17
68	13	3	0	20
69	19	4	0	25
70	7	10	2	27
71	11	9	4	27
72	20	3	7	32
73	22	3	32	49
74	19	2	40	66
75	15	3	35	56
76	15	1	28	53
77	31	5	16	58
78	40	2	23	74
79	49	8	18	82
80	45	5	10	73
81	56	12	9	93
82	72	18	42	145
83	20	12	32	74
84	10	0	7	22
85	3	1	5	13
86	0	0	4	8

Note: This table includes Geophysical and Miscellaneous vessels over 150 feet in length as well as Supply, Anchor Handling/Supply and Tug/Supply vessels.

Source: Fleet Data Service, "**Offshore Service Vessels, A Guide to the American Fleet**", 1986.

4.4 DESIGN PRACTICES AND TRENDS

The supply vessel is a classic example of a craft designed and built for a specific need and market. Some of the factors which drive OSV design have been noted in the proceeding sections, and others will be discussed in this section.

For example, there are significant differences between U.S. supply vessels and European designs. Table 4.5 lists several foreign and U.S. OSVs' characteristics. U.S. boats are nearly always chine hulls, with developable sections, even on vessels up to 220 ft in length. The conviction that a chined hull is easier and faster (and therefore cheaper) to produce has stood the test of time to the present. Augmented by its motion dampening characteristics, particularly in roll (ref. 4.3), the chined hull retains a secure position with builders for the near future.

Molded forms are frequently used in European construction, and bulbous bows have often been used there as well. The added cost of a bulbous bow in molded construction is relatively less than that for chined construction and can be more readily justified by the lower still water resistance it affords. Overall, the engineering and construction of continental vessels is much more sophisticated and often augmented by advanced equipment (i.e., more frequent use of controllable pitch propellers, active rudders, multiple thrusters, stabilization tanks, icebreaker bows, etc.).

TABLE 4.5

DEPTH, DRAFT AND FREEBOARD VERSUS LENGTH

U.S. AND FOREIGN OSVs

Name	Flag	LOA (ft)	Depth (ft)	Draft (ft)	Freeboard (ft)	Depth/draft
Sentinel	foreign	151.90	16.73	12.14	4.59	1.38
Flinders Tide	f	169.65	17.39	15.26	2.13	1.14
Stirling Imp	f	170.96	15.42	13.45	1.97	1.15
UT 711	f	173.88	22.15	14.76	7.38	1.50
UT 714	f	192.25	22.15	18.70	3.44	1.18
UT 713	f	193.57	21.00	18.04	2.95	1.16
Retiever	f	198.32	22.97	20.51	2.46	1.12
Solvbas	f	201.77	18.04	13.94	4.10	1.29
Shelf Express	f	202.16	19.36	15.09	4.27	1.28
Seaforth Vis.	f	205.18	20.18	16.40	3.77	1.23
Stirling Esk	f	212.92	20.51	17.72	2.79	1.16
UT 734	f	213.25	24.28	16.99	7.28	1.43
UT 704	f	213.25	22.64	18.86	3.77	1.20
Maersk Rover	f	219.81	24.61	21.16	3.44	1.16
Seaforth Monarch	f	220.47	23.29	19.98	3.31	1.17
Seafoth Emperor	f	220.47	23.29	19.36	3.28	1.17
Salvageman	f	223.91	22.31	19.68	2.62	1.13
UT 706	f	224.73	23.95	16.40	7.55	1.46
Livita	f	226.38	23.79	16.40	7.38	1.45
Wimpey Seahorse	f	227.36	23.69	20.67	3.02	1.15
UT 712	f	247.70	22.64	18.37	4.27	1.23
Ikaluk	f	258.69	31.82	24.61	7.22	1.29
UT 705	f	264.99	23.29	14.11	9.19	1.65
Explorer MSV	f	270.00	25.00	22.00	3.00	1.14
Robert LeMeur	f	270.67	24.61	18.60	6.00	1.32
Kalvik	f	288.64	32.80	26.24	6.56	1.25
Canmar Kigoriak	f	298.75	32.81	27.99	4.82	1.17
Osam Eagle	US	110.00	10.50	8.00	2.50	1.31
Midnight Alaskan	US	125.00	11.50	9.50	2.00	1.21
Point Au Fer	US	130.50	12.00	10.00	2.00	1.20
Boat 'A'	US	165.00	12.50	10.20	2.30	1.23
Bishop Rock	US	166.00	13.00	11.00	2.00	1.18
Boat 'B'	US	176.00	13.00	11.00	2.00	1.18
Safaniya Five	US	180.00	14.00	11.00	3.00	1.27
HOS Bold Venture	US	180.00	14.00	12.00	2.00	1.17
Marsea 1-6	US	180.00	14.00	12.00	2.00	1.17
Marsea 11-25	US	180.00	14.00	12.00	2.00	1.17
PBR/330	US	180.00	13.50	12.16	1.34	1.11
Marsea 7-10	US	184.67	14.00	12.00	2.00	1.17
Hawke Seal	US	185.00	14.00	12.00	2.00	1.17
State Spirit	US	192.00	14.00	11.00	3.00	1.27
K Marine No. 1	US	192.00	14.00	11.67	2.33	1.20
State Power	US	192.00	14.00	12.00	2.00	1.17

Why the difference when the missions are identical? The first factor is the environment. The North Sea, the locale for nearly all European offshore development, is significantly rougher than the Gulf of Mexico. Operators in this area have always favored a stout hull, and until recently, it was felt that molded hull forms were superior to chine forms in rough seas.

The second factor is customary practice. The European countries bordering the North Sea have large fleets of coastal freighters and fishing boats, therefore their shipyards and marine suppliers are highly developed. The advantages of Controllable Pitch Propellers (CPP) for fishing boats have long been recognized in Europe, and there are many competing manufacturers of such gear. The molded hulls, CPPs, etc., on European supply vessels reflect common practice for any type of small ship in that area, where the chine hulls and fixed-pitch propellers of Gulf Coast OSVs reflect fishing boat and tug practice for that area.

The final, and perhaps most influential factor are the business arrangements under which the vessels are employed. When the offshore oil business first started in the U.S., there were few people in the industry with experience in traditional maritime operations. Contracts between the oil companies and their contractors bore more resemblance to service contracts already common in the shore-based oil industry than to the "Demise Bareboat Charters" and "Time Charters" of the steamship industry. The periods were much shorter, for instance, and the contracts were much more easily revoked than was customary in traditional maritime practice. Often contracts were based on "day rates", and if the boat was not working for any reason, the owner received no compensation. Also, the charterer (the oil company), provided all fuel, which effectively meant that there was no incentive for the owner to conserve it. Overall, this meant that to maximize the owners return on investment, capital costs (i.e., cost of construction and finance) needed to be cut to the bone, reliability and low maintenance were essential and everything else was superfluous. Boats were built and managed in the Gulf of Mexico using a short term, bottom line perspective.

In Europe, a more traditional maritime approach was followed. Many supply boats were run by existing shipping lines, and they tended to favor doing business the way they always had, building boats with a 20-25 year life in mind, introducing somewhat more sophisticated systems to increase efficiency, and holding out for longer term contracts which allowed them to recover their higher capital costs. With this background in mind, it is easier to reason out some of the differences between U.S. and European OSV practices.

So it is clear that the owner's need to make a profit drives the design. What are the factors that charterers look for which increase the market value of the owner's assets?

1. The total amount of supplies it can carry under the main deck and the capacity and versatility of each type of cargo, i.e.:
   A. Bulk Mud/Cement
   B. Drill Water
   C. Potable Water
   D. Fuel Oil
   E. Calcium Chloride/Bromide
   F. Liquid Mud
   G. Combustibles and Hazardous Cargoes
   H. Rig Chain (Tug/Supply Vessels only)
2. The maximum amount and types of deck cargoes it can carry and the available clear deck space to accomplish this.
3. Maximum cargo it can carry at load water line, in combination of items 1. and 2. above (total deadweight).
4. Cruising speed and range
5. Total Number of personnel in addition to the crew the vessel can carry and accommodate.
6. Other factors which are not related to payload, but do affect the versatility and marketability of the vessel:
   A. Maneuverability (CPP, CRP, Thrusters)
   B. Shallow Draft Capability (*Where required*)
   C. Sea Keeping Characteristics
   D. Ice Classification (where required)
   E. Fire Fighting Capabilities
   F. Auxiliary Equipment (deck cranes, moonpools, tugger winches, deep water mooring equipment)

Supply vessel operations bear close analogy to liner shipping, where speed and turnaround times are important factors. The supply vessel hull should be aimed at carrying the maximum cargo with the use of minimum horsepower for its speed. Free running speed/length ratio is an important consideration. In the past, the majority of hulls have been overpowered for the speeds attained. This has been an indication of the owner's concern to maximize cargo capacity, which has resulted in hulls rather too full and too short for anticipated speeds. It has been suggested (ref. 4.4) that a reasonable compromise between speed and cargo capacity would be achieved by employing a prismatic coefficient of 0.65.

Most supply vessels constructed in this country have open fixed pitch propellers. However, Controllable Pitch Propellers do have clear advantages, since they allow the main propulsion engines to operate in their most favorable loading condition both when free-running and when towing at high bollard pulls and low speeds. This increases fuel efficiency and reduces wear and tear on the engines. They were tried by several American operators, particularly on Anchor Handling or Tug/Supply Vessels, but the added maintenance and capital costs, and the decreased reliability of the mechanically complex CPPs rendered them non-cost effective for U.S. owners. (Some of the maintenance and reliability problems may have been due to lack of familiarity among U.S. industry personnel with the systems.)

Tug/Supply Vessel design has to be done with consideration for the functions both of tugs and of supply vessels. This means that increases in maneuverability and bollard pull are sought over straight Supply Vessel designs while retaining as much carrying capacity as possible. Thus bow and stern thrusters are used to approach the maneuverability of pure Tugs, and Kort nozzles, which augment bollard pull, are frequently used in U.S. construction and are nearly universal in Europe. The advantages of CPPs for vessels of this nature which require both high bollard pulls and high cruising speeds are realized by the Europeans in their wide use of CPPs with Kort nozzles. Alternative propulsion solutions to the conflicting requirements of Tug/Supply Vessels include multiple engine Father/Son configurations, Diesel Electric propulsion and 2-speed reduction gears.

Table 4.1 shows the ranges for some of the design parameters of straight Supply Vessels and Tug/Supply Vessels, and is adapted from Reference 4.2.

4.5 ADAPTATION TO BUOY TENDING

The adaptation of OSVs to SRA duties has been examined in detail by many authorities with responsibilities in this area, and the Germans and Canadians have converted existing hulls or built new vessels based on OSVs as discussed in the chapter on foreign tenders. The success, or lack of it, of these vessels has depended in part upon the design characteristics of OSVs, and this area bears further examination.

It seems clear than an OSV can be converted into a vessel which can service navigational aids. The existing foreign conversions attest to this. Additionally, the work of Bowling, et al. (ref. 4.5) has shown that a typical 180 ft. supply vessel has the necessary volume and displacement to accomodate conversion to the SRA servicing mission. What then are the drawbacks of doing so, given the ready availability and low cost of OSV hulls in the current depressed market?

The biggest problem the Canadians have encountered with their conversions and OSV-like newbuilds has been the working condition on the aft deck. Due to the forward pilothouse of the designs, they tend to station-keep with their sterns into the wind and seas, which affords little shelter to the crew attempting to work the aid. Stern slamming is a real possibility, and the low aft freeboard characteristic of OSVs leads to a lot of water on the deck.

Can these faults be alleviated, and if so, at a cost that does not obviate the advantages of conversion? There are several approaches. One is to install a Dynamic Positioning System of sufficient power to force the vessel to ride head to wind and waves. With the protection of the forecastle in front of them, and the bow-on attitude reducing motions compared to stern-to, the working conditions may be better than for a conventional arrangement, although the power required to hold the vessel in position would increase substantially. Another possible approach would be to reduce forward windage and/or increase it aft so that the vessel would lie head to wind/waves. Finally, the freeboard aft could be increased or some sort of shelter deck could be fitted to protect the working deck.

Another problem facing potential conversions is the high installed power of many existing OSVs. Hull forms optimized for carrying capacity rather than propulsive efficiency and requirements for high bollard pull for rig moves lead to typical OSV installations of two to three times the horsepower that would be found in a similar size buoy tender. Even operating at reduced power (bad for diesel life and time between overhauls), a OSV conversion will have poor fuel economy compared to a purpose-built tender.

One troublesome problem facing potential conversions is the lack of two-compartment subdivision, which is generally a requirement of any Coast Guard or Naval vessel. Bowling, et al. (ref. 4.5) found this to be the only requirement a conversion could not meet without radical reconstruction. In some cases, due to their large engine rooms, OSVs cannot even meet an any-one-compartment flooding criteria (although they must be able to withstand a given area and depth of side shell penetration). It should be pointed out that Bowling, et al., dismissed the lack of ice breaking capability as a deficiency that would not prevent the converted design from fulfilling its primary mission. However, ice breaking ability is a mission requirement for WLBs/WLMs which must service buoys in the ice.

It seems that an "austere" conversion of an OSV (adding berthing, shops and buoy handling gear, without major structural or mechanical modifications) will not be able to be considered as a fully capable Offshore Buoy Tender. Further modifications to existing hulls to make them fully capable will be extensive and therefore costly, reducing the benefits of such a conversion. The strengths and weaknesses of the converted vessels would need to be carefully considered. Perhaps a force mix of conversion for milder weather areas and purpose built tenders for more exposed areas would be advantageous.

But the benefits of the OSV type of vessel could also be realized in new construction as well. Low aft freeboard became a characteristic of OSVs because it increased stability for a given deckload, or allowed a larger deckload for a given beam and displacement, while it minimized the gross tonnage of the vessel, and therefore had an economic advantage in a highly competitive market. For a vessel less driven by considerations of deck load and tonnage, the freeboard could be increased without hampering the good qualities of the OSV (as it often is for North Sea OSVs). Similarly, installed power need not be as high as previous U.S. practice has seen it, for without the context of the contractural arrangements with the oil companies, it no longer makes sense (again, current practice is towards reduced horsepower).

By taking the best, most innovative and applicable features from the latest U.S. and foreign OSVs, such as the STIRLING IMP, and the ACADIAN MARINER, the low cost, good seakeeping and ruggedness of the typical OSV could be applied to the SRA mission in the most satisfactory manner, although conversions of existing OSV may prove unsuitable for the mission requirements of a WLB/WLM.

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5.0 HULL FORMS FOR SEAKEEPING

5.1 PURPOSE

In this chapter, the influence of several candidate hull forms upon seakeeping will be surveyed. The importance of hull form selection to the effectiveness of naval units and merchant shipping has been widely recognized, particularly in the last 20 years. This has expanded the interest and the research into this complex field. Thus it has been necessary to greatly restrict the scope of this survey to only those articles and developments which the project personnel felt to have direct applicability to Offshore Buoy Tenders. In some cases, references oriented towards Naval Combatants (a significant portion of the work in seakeeping) have been included, but only if the method, theory, or program presented is applicable outside the range of long, slender, high speed frigates, cruisers and destroyer hull shapes.

5.2 SEAKEEPING PREDICTION, EVALUATION AND OPTIMIZATION

In this report, Prediction means the ability to predict the behavior of a vessel in a seaway. Evaluation refers to the measurement of the performance of the ship while conducting given mission(s), and Optimization is the process of maximizing the performance of the marine vehicle, in this case trading off seakeeping with other performance factors.

5.2.1 Prediction

The seakeeping of ships has been a matter of concern to designers, builders and operators since the time of Archimedes, but it wasn't until the 19th century that any real progress was made towards understanding the motions of ships in waves. The development of steam ships heightened interest in seakeeping, since without the damping effect of sails, they tended to roll heavily. William Froude published two landmark papers in 1861 and 1862 which set forth the mechanics of the rolling of ships in waves. Froude continued his work through the 1870s until his death, and like so much of his other work, further studies were carried out by his son, R.E. Froude. A. Kriloff began attacking the problems of pitching, and the equations of motion developed by Froude and Kriloff are very nearly the same as those used today (at least to the first order).

It is beyond the scope of this survey to develop even the rudiments of ship motions theory, but it may help to think of a ship in a seaway as a damped, forced, harmonic oscillator. The motions of this oscillator (ship) depend on the magnitude and frequency of the forcing function (waves) and the damping of the oscillator. If the forcing function frequency is near the system's natural frequency, resonance can develop, resulting [illegible] very large responses. Damping of the oscillator (ship [illegible] major role in determining the ship's motions, particular[illegible]

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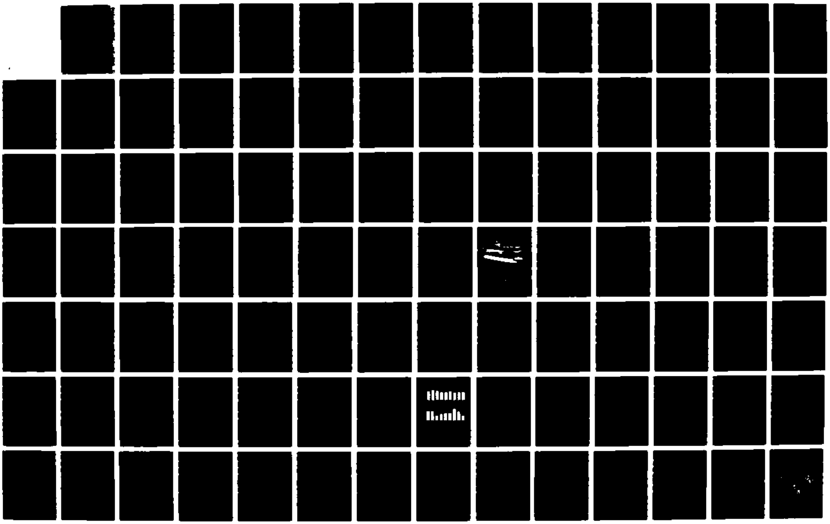
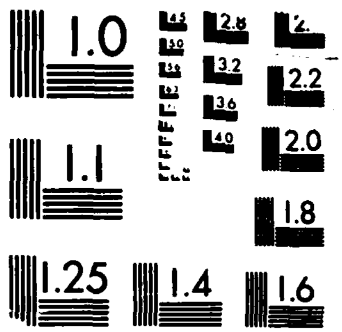




MICROCOPY RESOLUTION TEST CHART
REAU STANDARDS-1963-A

resonance, and a minor role in determining the ship's natural frequency.

For motions in regular, sinusoidal waves, the Froude-Kriloff approach, given the proper terms for the virtual mass and damping of the system, work well for predicting linear (i.e., small amplitude) motions.

A primary problem with the early work in seakeeping prediction was the the inability to realistically describe the sea environment. For a number of years after the publication of Stokes' work on gravity waves, the theoretically unsound, but mathematically convenient trochoidal waveform was used. The fundamental problem of the random nature of real sea states vice the regular waves naval architects were using in their theories and predictions remained. This lead to the classic remark, "It is hoped that the author will continue his efforts to coordinate what seas look like to seafarers with what naval architects imagine them to be", (ref. 5.1).

In 1913 R. Froude pointed out that an irregular wave form could be developed by the superposition of regular waves with varying frequency, amplitude and phase. As in so many other areas, this early insight was not fully appreciated until electronic computers made possible the calculations required. An example of the superposition of four waves and the resulting spectrum is shown in Figure 5.1.

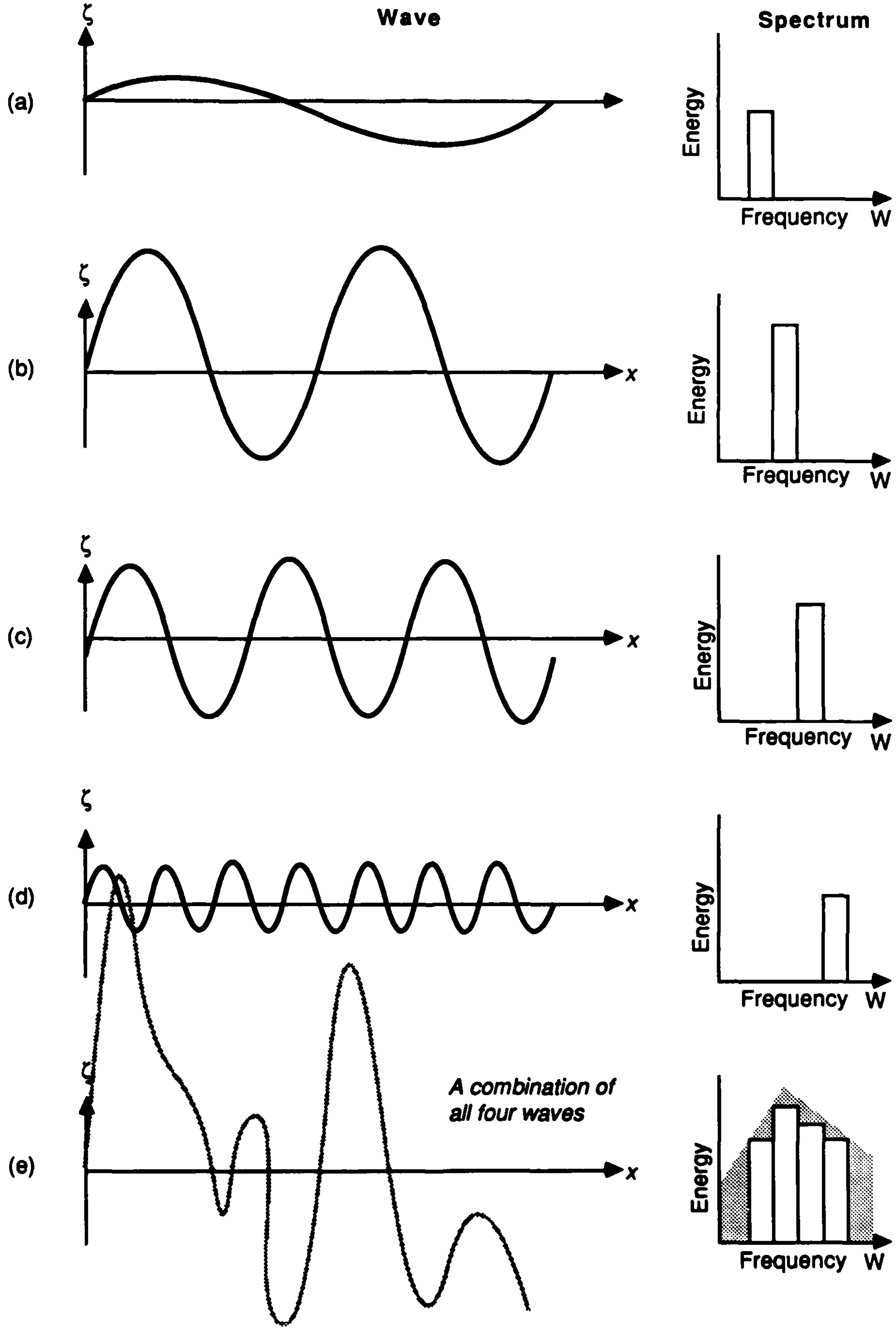


FIGURE 5.1 Illustration of Superposition Principle

The modern era in seakeeping prediction starts with two papers presented in the SNAME Transactions in the mid-1950s. The first was by St-Denis and Pierson. It used techniques for the analysis of random signals developed in the communications industry to analyze irregular seaways, and develop their spectra. Transfer function techniques could then be used to predict the motions of a vessel in that seaway, as illustrated in Figure 5.2.

The second paper by Korvin-Kroukovsky developed the needed transfer functions for pitch and heave by cutting the vessel into transverse sections or strips (giving rise to the name of this method, Strip Theory), calculating the added mass, damping, restoring and forcing properties at each strip, and then integrating over the length of the vessel for the rigid body motions. This theoretical base was expanded by others over the years to cover roll, surge, sway and yaw, as well as accounting for the effects of forward speed, the diffraction of the incident wave due to the ship hull, appendages, etc.

The model testing of Dalziell and Gerritsma (and many others), along with full scale trials, again by Gerritsma, provided correlation between calculated results and the real world, and provided increased confidence in the strip theory procedure.

Many refinements of the basic approach have been proposed, discussed, programmed and tested over the years. Some have advanced the state of the art, others have been discarded, and

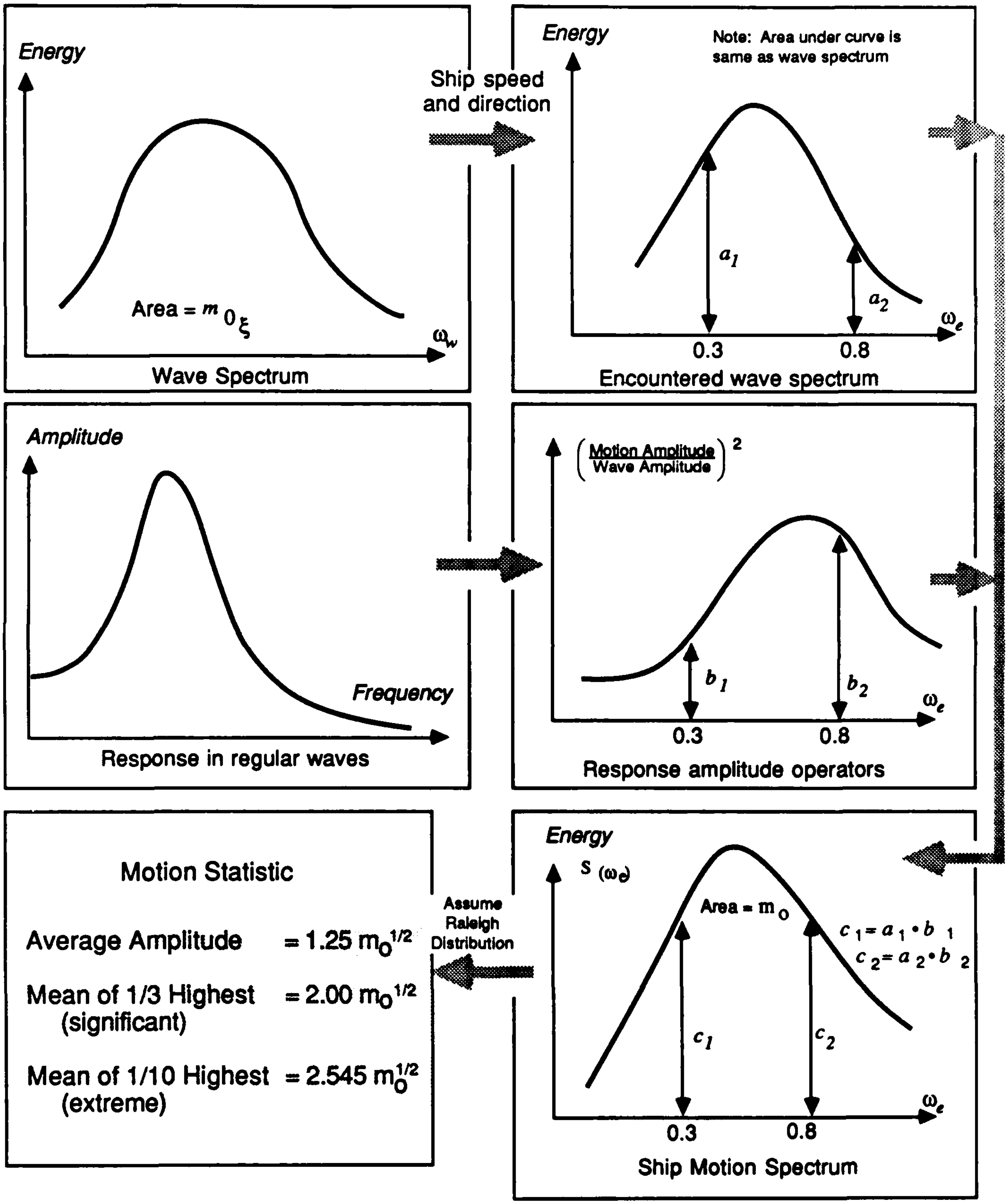


FIGURE 5.2 Prediction of Ship Motion in an Irregular Seaway

still others provided alternate methods to the same end. From the initial capability for calculating vertical plane motions in head seas at zero speed for fine-lined ships, the theory was developed to account for forward speed, shapes other than the simple "Lewis Forms", lateral motions, oblique seas, second order effects, relative motions, etc.

Odabasi and Hearn (ref. 5.2) and Hearn and Donati (ref. 5.3) have compared several different two-dimensional (strip theory) and three-dimensional (source distribution or finite element) methods of seakeeping prediction and found that: 1) there is little difference which 2-D method is chosen so long as the mathematical instabilities of the certain programs are avoided; 2) the 3-D method consumes far more labor and computer time; 3) the 3-D results differ significantly in phase from the 2-D results, while the magnitudes of the responses are comparable. The conclusion is that 3-D methods are needed only for geometries and circumstances which cannot be handled using 2-D approximations.

Other reviews (ref. 5.4) have shown that while the state-of-the-art prediction of vertical plane (pitch, heave, hull bending) responses is very good, the results for lateral plane motions are not as good. Recent developments in the field have concentrated on this area, and have improved prediction significantly (ref. 5.5, 5.6, 5.7). Other areas of research include improvements in the predictions of relative motions (relative to the free surface, between multiple objects) and many efforts to streamline the process of predicting seakeeping performance.

5.2.2 Evaluation

It took an enormous amount of development to be able to predict a ship's response to a seaway, but this is not the bottom line. Once it is possible to predict ship motions, and it is recognized that improvements in seakeeping performance are desired, there must be a way of choosing between alternatives in providing that upgrade. That is the purpose of seakeeping evaluation.

We need far more than just the transfer function between the waves and the ship's response (although this is a vital part of the analysis) for this evaluation. We also need to know:

- What missions and operations the ship needs to perform.
- The limit states for those missions.
- The environment in which they are performed, including the frequency of occurrence of severe wind, wave and other factors.
- The effects of directionality, what heading the ship takes to the prevailing weather.

The amount of information needed at the outset and the amount generated during the analysis depends largely on the scope of the desired evaluation. The closer to an assessment of the whole ship system over its lifetime the analysis gets, the more information is needed.

Saunders (ref. 5.8) notes that in 1960 there were no acceptable quantitative requirements for ship seakeeping performance. He did note that such requirements could be developed, and that they would in general fall into three areas relating

-"First, safety of the ship and crew. This means that the ship must remain afloat and topside up, within safe limits, and that the well-being and lives of the personnel must be preserved. Under these conditions there shall be no major structural failure and no major slam damage. The watertight integrity and stability of the hull shall not be threatened by the water taken aboard.

-Second, performance of its mission in the specified limiting sea condition, not necessarily the same as for the first requirement. This requirement involves good behavior and sea-kindliness under the weather conditions laid down.

-Third, maintenance of schedule, in the limiting sea and condition specified for the second item, or in some other specified condition."

Probability was only mentioned in a general way in Saunders' discussion of seakeeping requirements, and no account is taken of the likelihood of actually encountering the design sea condition.

The development of evaluation techniques was accelerated by the realization in the early 1970s that our naval combatant ships could not keep up with those of our adversaries or our allies in moderate to heavy weather conditions. Out of this realization came the first halting efforts at true evaluation (ref. 5.9). The earliest efforts involved comparing the actual freeboard of combatants with an empirically developed minimum freeboard. This was taken as a relative measure of deck wetness. A refinement was to assign a figure of merit to each vessel based on its percentage of the empirical minimum freeboard. This technique did not directly address the effect of ship motions on the required freeboard, although it was a start. Kehoe's paper also reports on an effort using analytical predictions of ship motions and a slamming criteria to develop limiting speeds in various sea states. These efforts represent the first, limited developments in a method that would be refined by a number of researchers over the next fifteen years to develop sensitive, consistent evaluation tools.

The programs of Comstock and Covich (ref. 5.10) appear to be the first comprehensive application of this approach. They do build on earlier work by Ochi and others on prediction of slamming occurrences, and they in turn depend on efforts such as that of Hogben and Lumb (ref. 5.11) to characterize the statistics of ocean waves. Further work in this area by Olson (ref. 5.12), various researchers in Italy (ref. 5.13, 5.14, 5.15) and a number of efforts by workers at David Taylor Naval Ship

Research and Development Center (DTNSRDC) have developed tools which are useful at a surprisingly early design stage to evaluate the seakeeping performance of prospective designs, of both conventional and advanced hull types. Figure 5.3 illustrates the steps in this procedure.

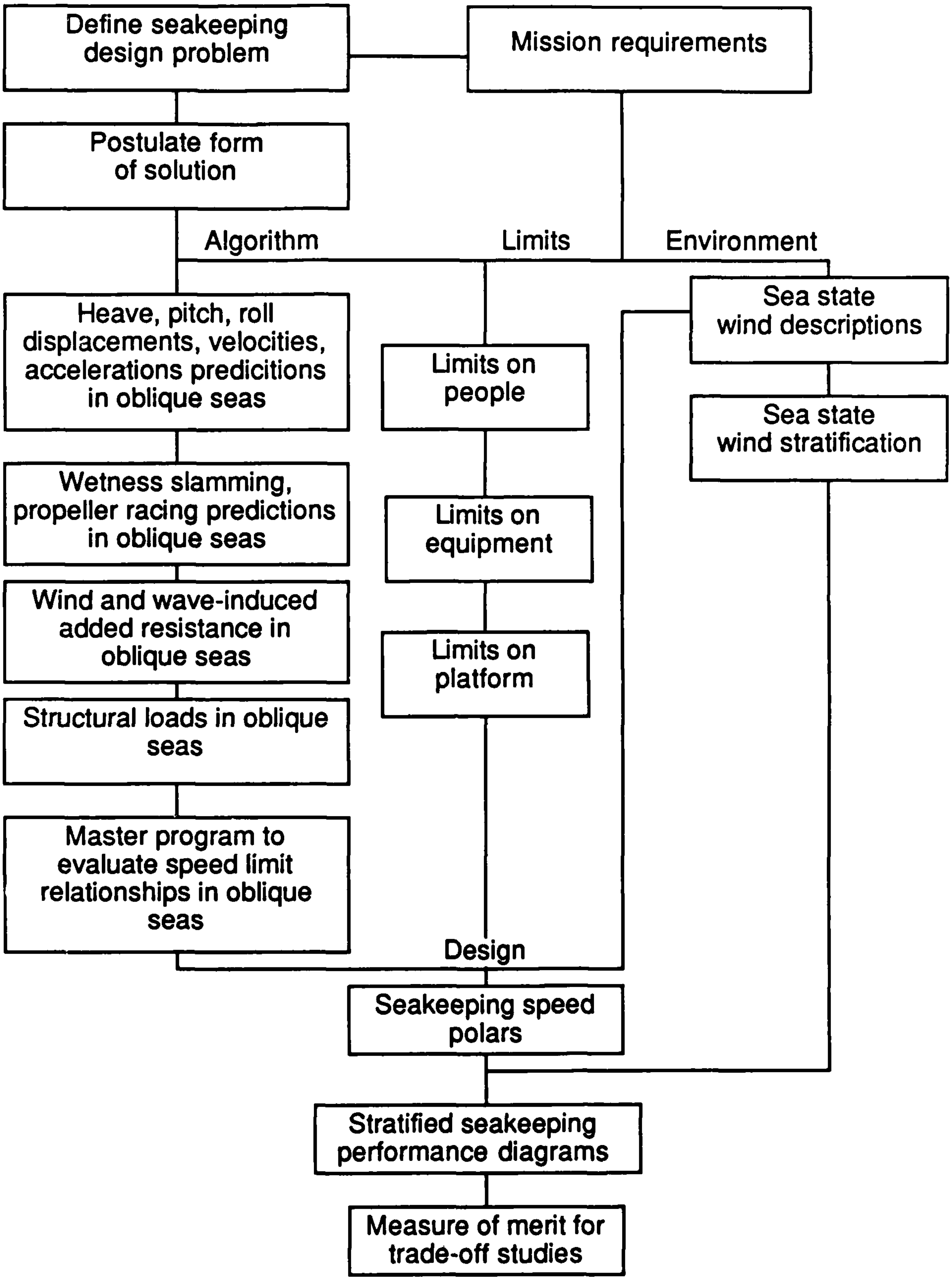
While there are a number of differences in the methodologies employed by the various researchers in the field, they all strive for a numerical index of relative performance. (called a "Box Score", "Figure of Merit", "Rank Factor", "Performance Index", etc). Most missions (searching, transiting) are speed dependent, and indices for these missions essentially reduce to:

$$\frac{\text{Average Sea Speed}}{\text{Calm Water Speed}}$$

But there are certain missions which are not speed dependent, or which are carried out at zero speed. The general form of indices for these missions (actually, the general form for all seakeeping indices, including speed as a measure of effectiveness) is:

$$\frac{\text{Mission Effectiveness in Rough Seas}}{\text{Mission Effectiveness in Calm Seas}}$$

The current state-of-the-art as practiced by the Navy can be seen in articles such as Kennell, et al. (ref. 5.16) and McCreight and Stahl (ref. 5.17). The Navy's Spectral Ocean Wave Model (SOWM) is used to obtain detailed information on wave heights and periods such as Table 5.1, and a generalized performance index is



Source: Comstock & Eevich (1975)

FIGURE 5.3 Flow Chart for Seakeeping Evaluation in Design

TABLE 5.1
ANNUAL SEA STATE OCCURRENCES IN THE OPEN OCEAN, NORTHERN HEMISPHERE

Sea State Number	Significant Wave Height (m)		Modal Wave Period (Sec)		Percentage Probability of Sea State
	Range	Mean	Range	Most Probable	
0-1	0 - 0.1	0.05	-	-	0
2	0.1 - 0.5	0.3	3 - 15	7	5.7
3	0.5 - 1.25	0.88	5 - 15.5	8	19.7
4	1.25 - 2.5	1.88	6 - 16	9	28.3
5	2.5 - 4	3.25	7 - 16.5	10	19.5
6	4 - 6	5	9 - 17	12	17.5
7	6 - 9	7.5	10 - 18	14	7.6
8	9 - 14	11.5	13 - 19	17	1.7
>8	>14	>14	18 - 24	20	0.1

employed which can utilize multiple limiting criteria depending on the missions of the vessel being evaluated. In some cases, two evaluations are developed at each grid point of the SOWM, and contours of constant operability are drawn for geographical areas. Such an approach is illustrated in Figure 5.4.

Another approach has been to relate the rather abstract performance indices to something operators and program managers understand - cash. Gatzoulis and Keane (ref. 5.18) computed the cost effectiveness of a frigate in helicopter operations with and without fin stabilizers, and Brown (ref. 5.19) assigns a return on investment to seakeeping improvement. Both of these studies come to the conclusion that investing in seakeeping provides a return that any banker or industrialist would applaud.

As Lewis (ref. 5.4) notes in his semi-annual review of seakeeping technology, the development of the limiting criteria is the area of seakeeping evaluation that is the least well-developed. We really know very little about how ship motions degrade the performance of ship systems, particularly its most complex system, its crew. For operations such as buoy tending, it is doubtful that measurements of the limiting systems criteria have ever been attempted, although information developed from studying helicopter operations may be of some use.

Another needed advance in the methodology of seakeeping evaluation is the development of techniques that allow for

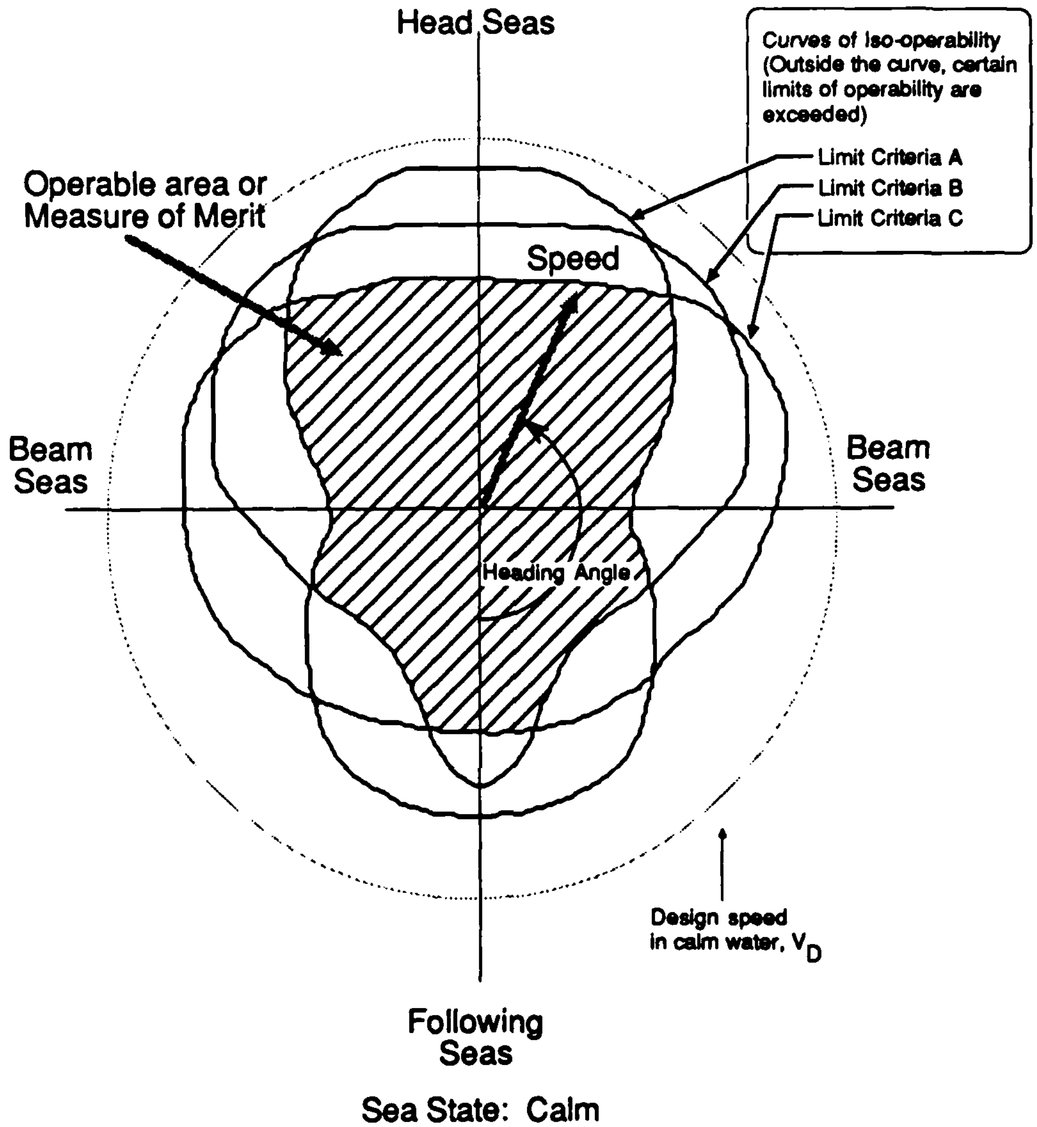


FIGURE 5.4 Seakeeping Speed Polar Program

gradual degradation in effectiveness. Current methods generally assume a ship or a sub-system is either 100% effective or it is inoperable. Obviously, most real situations are not so clear cut, and the ability to model a more graceful decline in effectiveness would be an improvement.

The methods of seakeeping evaluation may yet be only a crude analog of the real operations of ships and their systems, and the percentages of operability they generate may not be congruent with experience, but it must be remembered that the true worth of the evaluation process is not its ability to mimic nature, but its utility in making choices between alternatives.

5.2.3 Optimization

Evaluation techniques allow the designer to quantify the performance of a vessel; they do not help him design the best vessel. Optimization seeks to find the best alternative given a set of requirements and a set of constraints. Optimization involves performing evaluations on a wide range of alternatives and making choices based on those evaluations. Thus it can be seen that seakeeping optimization is a direct descendant of seakeeping evaluation.

Two primary methods have evolved for performing optimization studies. The first depends on the evaluation of regression analyses developed from a database of existing vessels and is typified by the Bales Seakeeping Rank Factor, developed at DTNSRDC by the late Nathan Bales, and his methods of developing freeboard requirements (ref. 5.20, 5.21). Criteria similar to that used by Bales are also described in reference 5.21a. The second approach involves a direct search of a range of alternative ships, evaluating each to find the best combination of performance within the constraints of the problem. This is the approach used by Walden and Grundman (ref. 5.22), and is illustrated in Figure 5.5.

Each approach has its merits. The Bales approach is much simpler to evaluate, once the initial analysis of the database has been performed. The regression equations are differentiated to find minima or maxima which define the optimum value. Any alternative can easily be ranked by evaluating the equations using its design parameters. However, this method depends on a substantial bank of information on a reasonably large number of existing vessels, and has little applicability outside the range of parameters enclosed by the vessel database. It is doubtful that enough data on buoy tender type vessels can be accumulated to match the database used by Bales and others to develop the Rank Factor for Frigates and Destroyers.

The approach of Walden and Grundman (ref. 5.22 - in their original work, they also worked on an extension of Bales) also depends on a database of seakeeping performance on a variety of vessels. The data are either obtained directly by calculation as the analysis proceeds or are obtained from previously computed

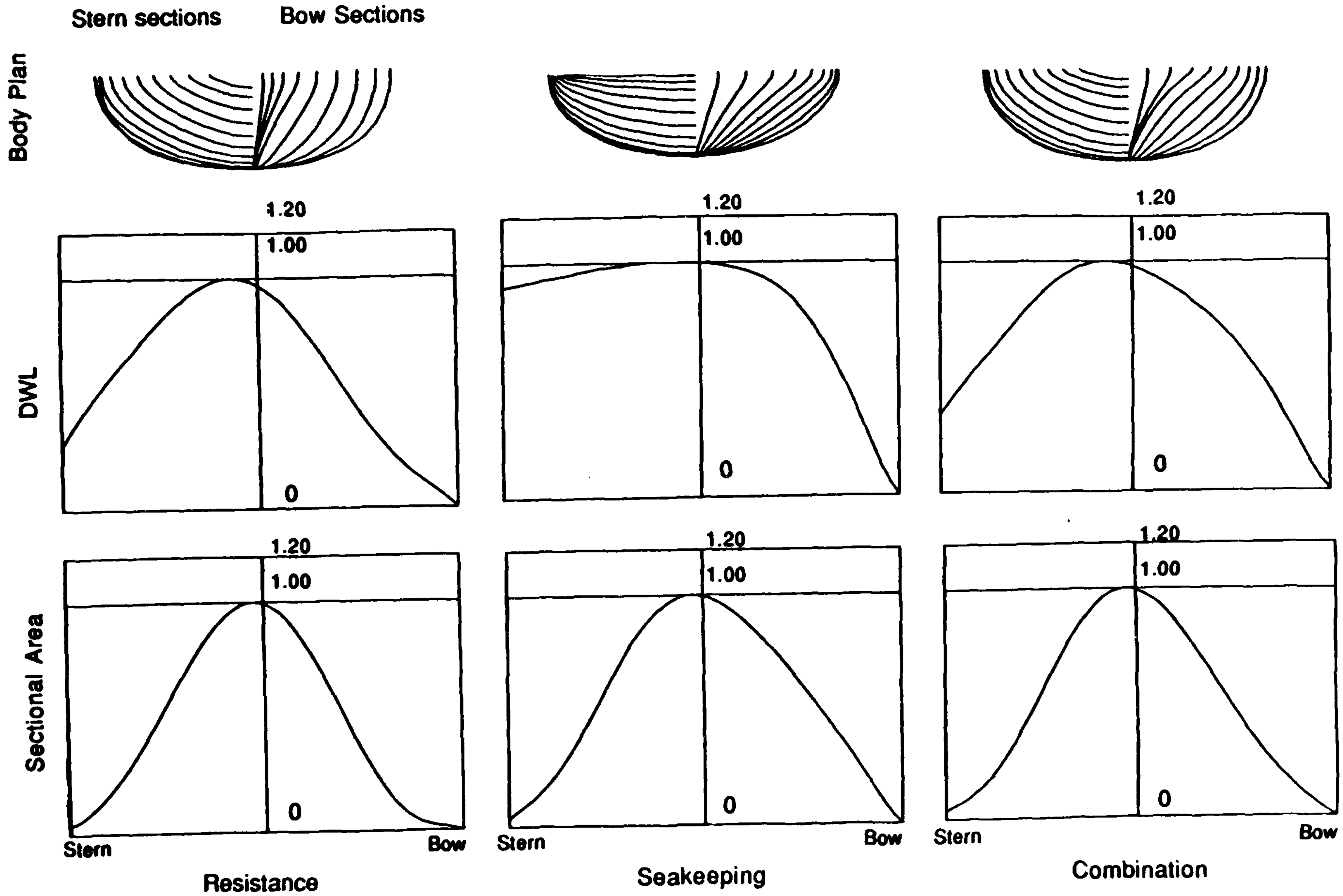


FIGURE 5.5 Optimized Hull Forms (from Walden & Grundman, 1985)

results of a seakeeping series. If the results are to be calculated, the usual procedure is to synthesize a possible ship, generate its lines, then perform a seakeeping prediction and evaluation on those synthesized parameters. This operation is computationally intensive, particularly since this synthesis, prediction, and evaluation procedure needs to be performed at every step in the optimization search. If the seakeeping performance is pre-computed in series results (analogous to the Taylor Standard Series and Series 60 results used for resistance prediction) the computational effort is much reduced, but unfortunately only a very few seakeeping series exist (ref. 5.21, 5.23, 5.24).

As a result of the effort and expense required to perform optimization studies, they are not generally used as part of the ship design process. They are used to identify trends in classes of vessels so that the alternatives chosen during specific design studies are more nearly optimum than they would be without the insight the optimization studies provide.

5.3 MOTION STABILIZATION

Motion Stabilization in this context is taken to refer to any system which seeks to reduce the wave induced rigid body motions of a vessel. Thus neither improvements in hull form alone (which will be discussed in a later section) nor systems which stabilize other mission equipment (just guns for instance) are to be considered in this section, although both can greatly improve the overall performance of the ship system.

For conventional ships, motion stabilization most often deals with roll, for several reasons. Most ships are naturally lightly damped in roll, and roll motions are larger than other motions. The ship's overall performance in seas is often limited by its roll, without ever reaching a corresponding limit of pitch. In head sea operations, heave is generally the limiting motion. For conventional hulls, the buoyancy forces which generate heave and pitch are also so large that a stabilizing system capable of generating offsetting forces and moments would also be very large, heavy and power consuming. Thus, while they have been tried, anti-heave and anti-pitch systems for conventional hulls hold very little promise.

However, anti-yaw systems have been developed for some conventional hulls. An autopilot is a yaw stabilizer, and the coupling between yaw and roll often needs to be addressed, and in fact is taken advantage of in the design of Rudder-Roll Stabilizers. Small Waterplane Area Twin Hull (SWATH) ships, due to their decreased waterplane area, have much lower wave excitation forces under most circumstances than conventional hulls, and stabilization in roll, pitch, and heave is possible and routine. Pitch stabilization is generally required at high speeds for stability. Ride control systems are required for fully-submerged hydrofoils and desired for surface piercing hydrofoils. Surface Effect Ships (SES) and air cushion vehicles have high frequency (2 Hz) vertical acceleration motions which contribute to human fatigue. Air cushion ride control systems have been effective in reducing those accelerations in up to five foot seas.

One of the first attempts at ship stabilization using free surface tanks was made on the HMS INFLEXIBLE in 1880. It was known that ships with low transverse metacentric height had longer, more comfortable roll periods; the free surface tanks reduced the metacentric height. It is not clear whether the designers of the time also realized that in certain cases the motion of the water in such a tank was out of phase with the ship motion, producing a correcting moment acting in opposition to the roll of the ship. After their initial popularity, the lack of understanding of their design (including sensitivity to tank ullage and the possibility of introducing instability), their noise, and their weight and space requirements caused them to lose favor.

Much the same fate befell the other main type of stabilizing tank, the U-Tube or Frahm Tank. Frahm presented his paper on "Results of Trials of the Anti-Rolling Tanks at Sea" to the Institution of Naval Architects in 1911, but again, poor design and the lack of appreciation of the tuning needed for successful operation slowed the pace of installation down to a crawl by the 1930s (the decline in the shipping industry during this time period also retarded this and many other maritime innovations).

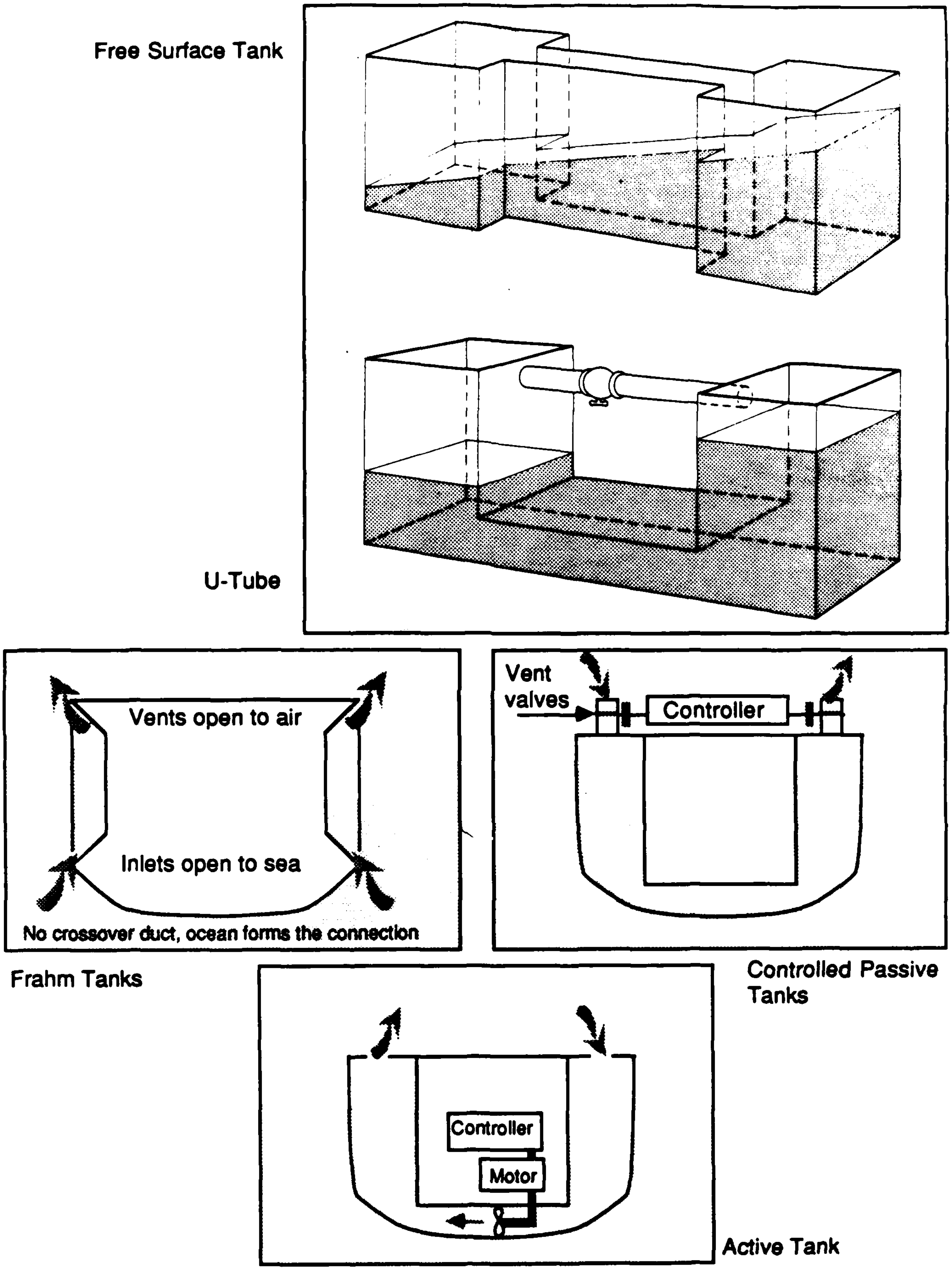


FIGURE 5.6 Schematic of Roll Stabilization Systems

Figure 5.6 illustrates several types of roll stabilization systems involving tanks.

Active fin stabilizers were also undergoing development. First proposed by Thorneycroft and patented in 1889, installations were made on several passenger ships in the 1920s. In an interesting coincidence, Motora of Japan independently developed similar equipment in the 20s and early 30s, apparently with no knowledge of the British work. Massive gyroscopes were also tried as roll stabilizers, but their size, weight and the fearsome amount of potentially destructive energy they stored made them unattractive. Improvements in fins slowly won adherents, first in passenger liners and ferries, and very slowly, with the world's naval powers, starting with Great Britain. Gyroscopic stabilizers were used in early ballistic missile submarines.

In the U.S., Minorsky had done pioneering work in Control Theory in the 1930s in conjunction with an effort to develop actively controlled stabilizing tanks, but sea trials of the system were aborted at the start of World War II. Some of the equipment was later used in a disastrously mismanaged post-war trial which hindered tank stabilization in particular, and ship stabilization in general for another 10 years in the U.S. By 1956, when the U.S. Navy made its first experimental fin stabilizer installation, such equipment had become standard equipment on British and Russian combatants.

The following sections discuss the specifics of each stabilization system and their recent development.

5.3.1 Fins

An actively controlled fin stabilization system consists of a sensor, a processor, a power source and the control surfaces themselves. The sensor picks up the ship's motion and provides data to the processor, which generates a control signal (usually proportional to roll rate) to the actuators (generally hydraulic) which drive the control surfaces. The ship reacts according to its particular dynamics, and the resulting motions are picked up by the sensor to close the loop.

There are several variations of fin stabilizers. Fins can either be retractable or fixed in place. They can be high or low aspect ratio. They can be flapped or simple, semi-balanced surfaces. Two examples of active fin systems are shown in Figure 5.7.

Since fins depend on the forward motion of the ship to generate lift, they are more effective at high speeds, and tend to be fitted on ships with high sustained speeds like naval combatants, passenger vessels, and a few cargo liners and container ships. As the speed of the vessel increases, the fin angles must be limited to avoid cavitation, which can cause severe vibration and noise, presenting a detection problem to naval vessels. The U.S. Navy's more recent installations have a low noise mode which

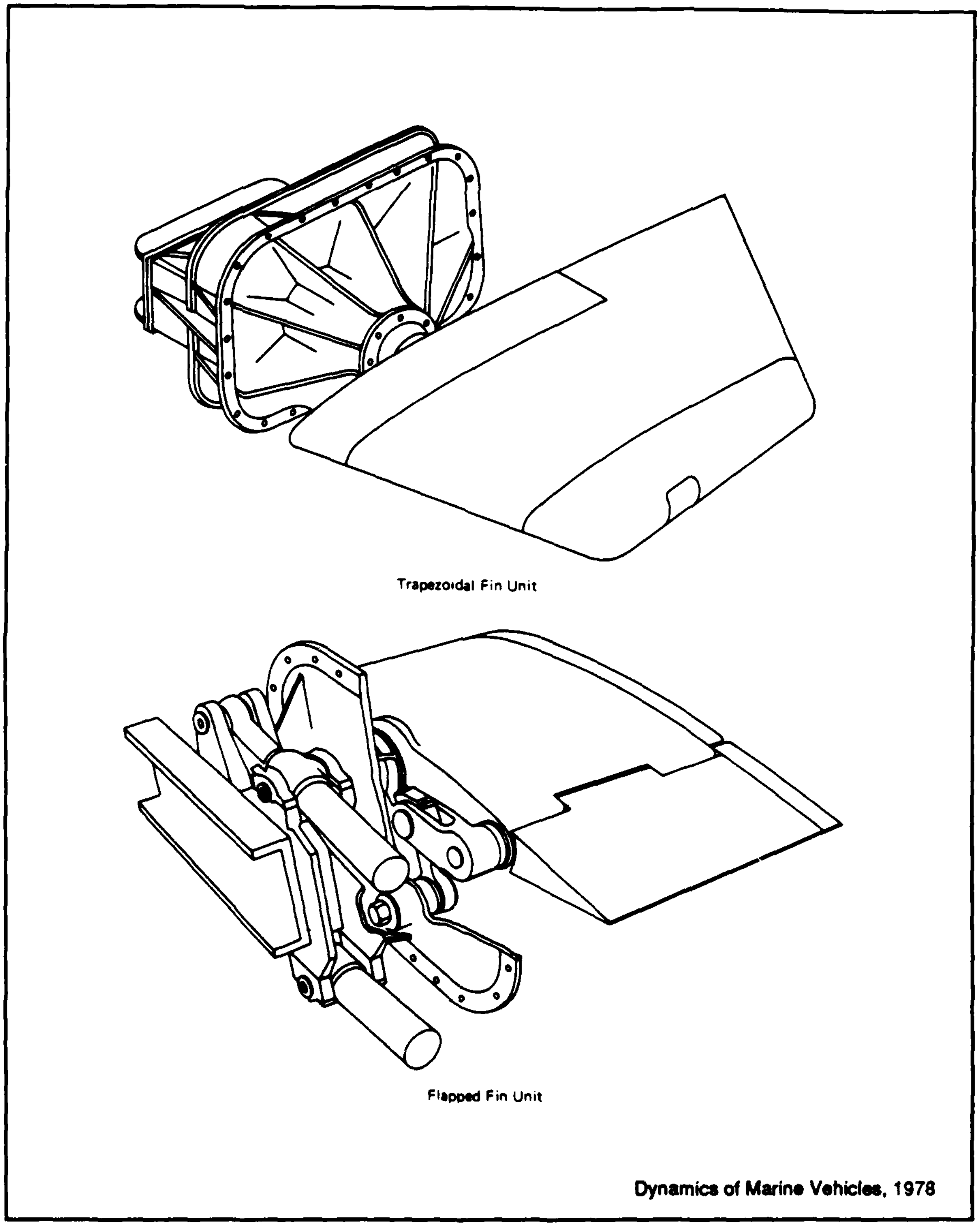


FIGURE 5.7 Typical Active Fin Stabilizers

further limits fin cavitation-induced noise. Limiting fin angles also limits effectiveness. Saturation is reached when the system cannot supply a sufficiently large moment to counteract the roll of the ship. Additionally, as the roll of the ship increases, the fins approach the surface of the water, causing them to ventilate and broach, further reducing effectiveness.

The key recent developments in Fin Stabilization have been in analytical prediction of their performance for design purposes. Conolly (ref. 5.25) developed the first rational method of specifying the size and number of roll stabilizer fins necessary to achieve a given level of performance. His approach (based on a simple linear equation in one degree of motion and lift curve slopes determined by cavitation model testing) has been shown to over-estimate the performance of the fins.

Lloyd (ref. 5.26) presents the results of an extensive analytical and experimental effort to determine the the source and magnitude of the losses of fin effectiveness. He shows that the losses can be attributed to three distinct sources: interaction effects between stabilizer fins and bilge keels, losses due to the fins being immersed in the ship's viscous boundary layer, and the effects of coupling between roll, sway and yaw motions. The procedure he develops to account for these losses and the recommendations made form the basis for stabilizer design in both the U.K. and U.S. Navies to this date. The recommendations are:

- Multiple fin configurations and arrangements with bilge keels aft of the fins should be avoided. (Unfortunately, just such an arrangement was fitted to the *U.S. Coast Guard's 270'* Medium Endurance Cutter (WMEC), designed prior to the dissemination of this information).

- There is no clear optimum position for the fins, unless low frequency performance is a dominant consideration, in which case the fins should be as far forward and as nearly horizontal as possible.

- High aspect ratio fins, apart from their inherent efficiency in producing lift, are less subject to the losses due to interference and boundary layers.

Using the predictive method of Lloyd, later extended in Cox and Lloyd (ref. 5.27) and Bales, et al. (ref. 5.28), fin stabilization effects were added to the seakeeping evaluation methods previously discussed. The abstract section has several references which employ the complete method to size fins for a particular effectiveness in a given operation and area.

Product improvements have also been made in the fin systems. Reliability and maintainability was a real problem for the U.S. Navy's early fin systems, leading to a detailed requirement for reliability and an extensive shore-based testing effort prior to the fitting of fins to the FFG-7 class of frigates. Nelson (ref. 5.29) and Donahue, et al. (ref. 5.30) document this very successful program. The advent of microprocessors has made the control processors simpler, easier to design, test and fix, and has allowed an increase in system capabilities through more complex control laws and extensive self-test and trouble-shooting functions. Inertial sensors are also starting to replace gyros in gathering motion information for use by the controller.

It is to be noted that fins are ineffective when the vessel is stopped, as when servicing buoys. Their function on a buoy tender is limited to stabilizing the vessel during transit.

5.3.2 Tanks

As shown in Figure 5.6, there is a wide variety of possible arrangements for roll stabilization tanks and many of them have been tried over the years and have found proponents. The main distinction is between the free-surface and the U-tube type (although ref. 5.31, shows they can be treated by the same equations of motions, the free surface tank being a special case where the inter-connecting branch has zero length). There are some further variations which apply almost exclusively to U-tube tanks; they may be passive, controlled or active. Passive implies that no external force or power acts on the water in the tank. Controlled tanks use valves to regulate either the transfer of fluid (usually sea water, but occasionally potable water, fuel or cargo oil) from one side of the vessel to the other or the exchange of air above the water columns from side to side or to the atmosphere. This slows down the transfer of water and extends the applicability of U-tube tanks to lower frequencies. Active tanks impart energy to the system using pumps or blowers to move or retard the water in the system.

Free surface tanks are the more common type since they are effective over a wider range of frequencies (although both types work best near their resonant frequency), and they can be tuned by varying the depth of water in the tank. On the other hand, U-tube tanks have less adverse effect on the ship's stability, and if they are active, can offset static heel due to asymmetric loading or crane lifts, or perform a GM test by inclining the vessel with a known moment.

Design methods for tanks have progressed greatly, but design is still a rather knotty problem, and unless the expense is absolutely critical, bench testing of a good-sized model of the tank system seems to be a very good way to establish the proper period and optimum damping for the planned configuration. Zdybek (ref. 5.31), Field (ref. 5.32), Miller, et al. (ref. 5.33), Barr, et al. (ref. 5.34), Lewison (ref. 5.35), Cox and Lloyd (ref. 5.27) and McCallum (ref. 5.36) give the designer valuable guidance on the design of roll stabilizer tanks. In general, tanks should be as wide and deep as possible, located as high in the ship as practicable. Damping (for free surface tanks) is best achieved by making the tank C-shaped, or like a dog bone, rather than by placing restrictions in the cross over.

Some recent developments include new variations of active, controlled U-tube tanks, promoted in Halden (ref. 5.37) as particularly suitable for Ro-Ro vessels. Another variation is what might be called an active, uncontrolled system, where fans pressurize or depressurize open-bottom tanks to change their natural period, but no attempt at control is made, see Ocean Industry (ref. 5.38) or Marine Engineer's Review (ref. 5.39). There are many patents in this area that protect various details of different designers' systems, and it can be a chore to avoid infringing them.

5.3.3 Other Systems

Gyroscopes were mentioned as one system tried and discarded for roll stabilization; moving solid weights on a pendulum is another system that was examined (by the Coast Guard as it turns out, on a 95 ft WPB) and found deficient compared to other methods. Since a weight of about about .5% of the vessel's displacement is needed for effective stabilizatio., on any vessel larger than a small boat (where such a tuned stabilizer is relatively ineffective), the problems of mounting, controlling and dampening such a huge mass of lead become prohibitive. There are at least three other methods of reducing roll in effective use today. The first is the oldest, simplest and cheapest form of roll reduction: bilge keels. They work by increasing the dampening, primarily the viscous dampening, of the ship and thus have the most effect near resonance. Unfortunately, the overall effectiveness of bilge keels is low, and degrades with speed in some cases. Bilge keels are vulnerable to damage due to slamming, over-the-side operations and ice, and may cause substantial resistance penalties if poorly located or damaged. Guidelines for choosing bilge keel size are contained in Cox and Lloyd (ref. 5.27). Though they do have some drawbacks, they are comparatively very cheap, and naval architectural practice is to fit them unless there is a reason not to do so.

The next form of stabilization is another low-tech approach, paravanes. These are weighted, horizontally winged bodies towed from long poles on either side of the vessel. Their effectiveness, due to the long moment arm through which the paravane forces act, can be quite impressive, and the system is cheap to install and maintain, even as a retrofit. Developed initially by troll fisherman in the Pacific Northwest, they were first adapted for power yacht use, then for small oceanographic vessels. Two papers, Koelbel, et al. (ref. 5.40) and Fuller, et al. (ref. 5.41) codified the procedure of designing the systems and represent the only technical references on the subject. Paravanes have not yet been used for roll stabilization on large vessels, although their use on the 157 ft WLMs has been investigated by the Coast Guard (written discussion of Koelbel, 1979).

The other alternative system, Rudder Roll Stabilization (RRS), is one of the most technically elegant methods for reducing roll. It has long been recognized that rudder action could induce roll in a ship. In the early 1970s Taggart noted a particularly bad case where a poorly tuned autopilot induced synchronous rolling in a fast cargo liner. This seems to have set off a flurry of activity to use the same phenomena, with the proper control, to reduce roll. In practice, a control signal proportional to the roll rate is added to the steering signal supplied by a conventional autopilot. Since the natural roll and yaw periods are well separated for conventional ships, this relatively high frequency signal does not degrade course keeping. Depending on the rudder rate available, it does reduce roll significantly without adding any additional appendages or heavy equipment to the ship, and with no loss of useful volume. Several of the Coast Guard's High Endurance Cutters (WHEC) have been fitted with such systems to improve effectiveness in helicopter operations. Baitis, et al. (ref. 5.42) describes this first operational use of RRS. The increased rudder activity has not caused undo wear on steering system components in service. Significant reductions in roll with minimal impact on ship design, can be achieved if the ship is designed with enlarged, faster acting rudders from the outset.

5.3.4 Selection and Design

If the roll response of the ship hull as designed is unsatisfactory and the decision is made to fit a stabilization system, then two tasks must be undertaken: the selection of the appropriate system and the sizing and design of the selected system to achieve the desired level of performance.

Each stabilization system has its costs, in dollars, in added weight and lost volume, in resistance, complexity, and impact on ship's operation. As noted in previous sections, however, lost effectiveness of the ship also has its costs, which run high. The choice of which system (including no system) is like any other design decision facing the naval architect, requiring careful analysis of the missions of the ship, and the strengths and limitations of the competing alternatives. There is no one system best for all ships in all missions at all times.

Fin stabilizers promise the greatest reduction in roll motions while underway; since they do not depend on the roll motion of the vessel itself to generate a restoring moment, they are theoretically capable of completely extinguishing roll. Their effectiveness is also relatively independent of frequency, which is less true of active tanks and RRS, the other systems theoretically capable of complete suppression of motions. Active tanks, because they must move very large masses of water rather quickly, require large amounts of power as they move away from their resonant range. RRS can run into stability problems (that is to say stability of control, not transverse stability) and adverse yaw/roll coupling at low frequencies of encounter, such as in following seas. On the other hand, fin systems (and RRS) are ineffective at low speeds (at what speed fins are ineffective is a point of much contention, but below six knots is probably a good first estimate), acting only as rather tiny bilge keels, while the tank systems and paravanes retain much or all of their effectiveness down to zero speed. On a relative basis, at the ship's design speed, a ranking of the systems for effectiveness at reducing roll would be fins, RRS, active tanks, paravanes, passive tanks, bilge keels. At speeds below design speeds , down to zero speed, fins and RRS would drop out completely, while the rest of the systems would roughly retain their place, except for some loss of paravane effectiveness.

The "costs" of the various systems are examined in Table 5.2. The high price paid for the effectiveness of fins can be seen, but the cost in terms of weight and space for tanks may be just as high, if the ship must be 80 tons larger to fully accommodate the tanks. RRS looks very good in these comparisons, as long as the vessel's speed is high enough to make use of the system. No data on bilge keels or paravanes comparable to that presented in Table 5.2 were available, but some preliminary studies (discussion of Koelbel, 1979) indicate that paravanes are about 20% less expensive than bilge keels for an initial installation of the same or better effectiveness, i.e., a 30% reduction of roll. Bilge keels in turn are substantially cheaper than fins or tanks. Maintenance on these two systems will probably be greater than for tanks, due to the cost of drydocking to repair bilge

TABLE 5.2
ROLL STABILIZATION SYSTEM ECONOMICS

	PASSIVE TANK	ACTIVE FINS	RUDDER ROLL STABILIZATION STD RATE	RUDDER ROLL STABILIZATION HI RATE
ADDED WEIGHT	80 tons S.W. (0 if fuel used) 10 tons structure	20 tons	<1 ton	est 5 tons
LOST VOLUME	6000 ft^3	1000 ft^3	negligible	<500 ft^3
ACQUISITION COSTS	$60K (design)	$400K	$20K	$65K
INSTALLATION COSTS	$50K	$110K	$10K	$20K
ANNUAL MAINTENANCE	Negligible	$15-20K	<$5K	<$10K

Notes:

- Assumes 4000 ton Destroyer/Frigate
- Production systems assumed, systems integrated at early stages of design.
- Data on fins and tanks from McCallum (1976). Fin to RRS comparison from Baitis, et al. (1983). Costs converted to December 1986 estimates using method of Appendix C. Weights for RRS are author's own estimates.

keels and wear on various elements of the paravane system. With bad luck, maintenance to bilge keels due to accidental damage might exceed that for fins.

Various systems also entail a resistance penalty in terms of percentage of total resistance, ranging from essentially zero for tanks to about 1% for fixed fins, 1.5% (minimum) for properly located bilge keels, 3% for retractable fins (this is for the case where the fins are extended, the pockets into which they retract being rather disruptive), perhaps 5% for paravanes, and as much as 10-15% for poorly aligned or damaged bilge keels. This needs to be contrasted with the ship added resistance due to roll, which is about 0.5-0.9% per degree of roll.

What is not shown on the chart is the impact on ship operations. For a buoy tender, any system which adds underwater appendages is probably out of the question for use while working aids, although it could be tolerated during transit. This eliminates fixed fins as a potential candidate, and makes paravanes less attractive. Bilge keels would also be a dubious choice for this application due to the possibilities for damage inherent in a buoy tender's mission. Rudder Roll Stabilization remains a cost-effective candidate.

Of the currently available systems, only passive tanks are known to have been fitted to buoy tenders, namely several classes of Canadian Coast Guard vessels. This is not surprising, given the low speed effectiveness of tanks, their lack of underwater appendages, and the relatively roomy and burdensome hulls of buoy tenders, which can easily accommodate the space and weight of tank systems. Most of these installations have been the patented "Flume" free surface tanks, but there have been U-tube type installations as well.

Once a system has been chosen, it needs to be designed to achieve the required level of performance. This entails sizing the components of the system and tuning the system to achieve the best results. It is vital, particularly for active systems, to examine the dynamics of the whole installation. Increasing the sensitivity of the sensors does no good, for example, if the fin actuators are so sloppy they induce phase lags and inaccuracy into the system. Careful analytical study, breadboarding, model testing and shore-based testing of the complete system are some of the tools which should be used to avoid poor performance of the installed system.

Most design tools for stabilizer systems start with one-degree-of-freedom models of the roll of the ship (and if need be, the motion of the fluid in the system). Damping and restoring forces are assumed linear in the simpler models, which is a fairly inadequate assumption in general, but for small angles (which is what the system is supposed to maintain) and in the absence of anything else (even linear damping coefficients backed up by test results on real ships are hard to find), it provides useful results. Non-linear equations are available and the Navy's state-of-the-art procedures for fins and bilge keels use such a model (ref. 5.27, 5.28). Only single degree-of-freedom models are necessary for fins and bilge keels, since studies have shown little influence from yaw and sway coupling. Yaw, and particularly sway, do affect tank stabilizers and RRS. The tools used to design them usually involve the coupled motions in two or three degrees-of-freedom (ref. 5.27, 5.36, 5.43).

5.4 RECENT ADVANCES AND TRENDS IN SEAKEEPING HULL FORMS APPLICABLE TO BUOY TENDING

With the so-called "Maritime Strategy" of the NATO navies increasing the emphasis on operations in the high latitudes of the North Atlantic (in order to deny Soviet submarines their bastions), research into seakeeping continues at a fairly rapid pace. Most of the improvements in seakeeping are evolutionary rather than revolutionary; none of the advances seen in the last 20 years can compare with the stupendous changes in electronics for instance. The most important advance has been in attitudes, with the realization that seakeeping is a vital part of ship design and operation.

The work of Kehoe and others mentioned in previous sections has helped to accelerate this process, leading to the development of the design tools already discussed. These are the most important developments in seakeeping, for almost every "new" idea in ship design has been tried at least once. The tools developed in the last 10-20 years make it possible to test these concepts and evaluate them; given the conservatism inherent in the marine world, few concepts ever reach the water without substantial evidence that they will work.

Another important advance has been in the increase in seakeeping information available to guide ship operators. Comstock, et al. (ref. 5.44) discuss some of the guidance available to the Master through Optimal, Tactical Operations, Heavy Weather, and Survival Ship Routing. These methods use different criteria, but they all seek to avoid areas were ship performance or safety will be degraded. Bales, et al. (ref. 5.45) discusses the development of a catalog of operator guidance to avoid heavy weather damage, essentially establishing an "operating envelope" for a given class of vessels. There have also been several efforts to develop real-time monitors which measure ship's motion, acceleration, stress and the encountered sea state, and provide warnings when the operating limits of the vessel are approached. All of these efforts depend on the prediction and evaluation methodologies noted in the previous sections.

While the numerous advanced concepts for improved seakeeping brought forward in recent years all have their advocates, it is well to remember that a conventional displacement monohull is very cost-effective for missions which do not require increased speed or significant improvements in seakeeping. Advances in monohull roll stability performance made in recent years make an even better case for using a conventional hull unless there are overriding reasons for not doing so.

The primary reasons for choosing an alternative to a displacement monohull is high speed, sustained speed in the seaway and improved seakeeping. Above a certain point, increased speed in a displacement monohull becomes very expensive. Hull forms such as planing craft, SES and hydrofoils become attractive. The SWATH provides sustained moderate speed in the seaway. However, these vehicles are all extremely weight sensitive, and expensive. It has been shown that an enlarged monohull designed to the same rigid standards required of these Advanced Marine Vehicles will give them a run for their money (ref. 5.46). Thus, the need for speed and seakeeping must be strong to favor advanced hull forms.

Sustained speed in the seaway and seakeeping ability, especially while tending buoys, do apply to SRA missions in rough water offshore scenarios. Variations of speed and seakeeping capabilities will be investigated through the use of a vessel ATON simulation model under development at the USCG R&D Center.

The Coast Guard's research and development community has been studying the entire spectrum of marine vehicle concepts for a number of years, and only three hull forms clearly show promise for the SRA mission: displacement monohulls, displacement multihulls and SWATHs. High speed capabilities (speed/length ratios greater than 1.5), around which most other concepts center, are irrelevant to buoy tenders. Recent advances in these hullforms are discussed in the next three sections.

5.4.1 Displacement Monohulls

The ideal seakeeping monohull is a bigger monohull. A longer, higher displacement hull will have better seakeeping performance, and will result in smaller motions and accelerations which allow cost savings in terms of crew fatigue and ability to carry out missions in higher sea states. The additional procurement expense of larger vessels should be considered in a cost trade off study which compares the above factors. For many vessels, especially naval combatants, the hull structure is a very small part in the overall ship cost. Thus, a small increase in steel cost incurred by a larger monohull construction may be very cost effective in terms of improved seakeeping performance (ref. 5.46, 5.47). In the following paragraphs, the effect upon monohull performance of changes in ship proportions and design are examined.

Looking at vertical plane responses first, the evidence on the influence of ship proportions on seakeeping is somewhat mixed. Aside from increasing length and displacement, the early studies seem to indicate increasing the length/beam, length/draft, beam/draft and longitudinal separation of the Center of Buoyancy (LCB) and Center of Flotation (LCF) improved the motion properties of ships (ref. 5.48, 5.49, 5.50). A study on full merchant ship forms (ref. 5.23) supported V-shaped forebody sections and high block coefficients, while concluding that longitudinal radius of gyration had little effect. Recent research has focused on naval combatant hulls. While affirming the importance of V-shapes forward, length to beam ratio and length to draft ratio, it has shown a marked preference for low prismatic coefficients. This also means low block coefficients, as the two are related by the midship section coefficent (ref. 5.51, 5.21). Further, reducing longitudinal gyradius has been shown to markedly reduce the pitch, heave and added resistance of a typical frigate hull (ref. 5.52, 5.53).

Some of these differences can be explained by development in method. The later research used evaluation methods to examine the overall effectiveness of the ship, including criteria for deck wetness and slamming along with simple motions. This pointed out that the previous valued separation of LCB and LCF, while reducing motions, greatly increased deck wetness. Some other differences may be attributable to the different ranges of the parameters examined, i.e., the merchant vessels of the early work differ greatly in form from frigates. The key variables identified in the later work were not examined previously.

It appears that further research, particularly on hull forms other than high speed naval combatants is needed. The following factors are generally accepted as leading to improved seakeeping performance in displacement monohulls:

- Increased
  - Length
  - Displacement
  - Speed
  - Waterplane Coefficient
  - Freeboard
- Decreased
  - Draft to Length Ratio
  - Vertical Prismatic Coefficient
  - Radius of Gyration

Turning to roll motions, the greatest improvement in monohull seakeeping has come from the widening acceptance of the benefits of stabilization, primarily active fin stabilization. Since studies (ref. 5.6) have shown stabilization to be by far the most influential factor in improved roll motions, this is an important development. A moderate GM, about 8% of the beam, provides adequate roll performance for a monohull ship. Lower values strongly degrade performance; higher values offer little improvement. When roll period is also considered, high GMs may prove detrimental. The influence of other factors such as displacement, and gyradius is nil to slight. A high waterplane coefficient and a low block coefficient give the best results (these factors also improve vertical plane motions).

Effort has also been directed to gain the benefits of improved seakeeping performance without paying too high a cost in still water resistance. The optimization efforts of Walden and Grundman (ref. 5.22) have been one approach. They used a "cost function" for optimization that includes resistance. Another approach is used by Lin, et al. (ref. 5.51). Their efforts involved taking a hull optimized for seakeeping performance and attempting through detail design to improve its resistance characteristics without sacrificing too much of its seakindliness. Both of these approaches, the first based on general proportions, the second on hull details, have shown good results.

5.4.2 Displacement Multihulls

Catamarans, the most common form of multihulls, have been the focus of increased interest in recent years, but largely as high speed craft. Ferries built by International Catamarans of Australia, represent one example of this resurgence of interest. The interest in catamarans is primarily where speed/length ratios (speed in knots divided by the square root of the waterline length) from about 1 to as high 3.2 are required. In this range a high Length to Beam catamaran has lower resistance than other competing hull forms. Low resistance combined with the simplicity of construction (compared to vessels like Hydrofoils and SES), makes a catamaran an excellent platform for carrying fairly light loads at moderately high speeds.

As noted previously, high speed is not as important as seakeeping ability in the SRA mission. Nevertheless, there are reasons why a slow-speed catamaran might make a good buoy tender. The hull form offers very good deck space and hull volume for a given length and displacement, and a high transverse stability. This means space on deck for buoys, space below for shops, and low heel angles when lifting. Unfortunately, catamarans are highly weight sensitive, both for best resistance and to ensure adequate cross structure clearance. They are always more expensive than monohulls for the same displacement due to the high cross structure loads, and the larger amount of skin surface area needed to enclose the same volume of ship. All in all, a catamaran hull form should not be chosen unless the ship's mission requires it.

Since there is not a large base of catamarans in service in missions related to buoy tending from which trends could be discerned, the experience of specific vessels needs to be closely examined. The most instructive examples are the USNS HAYES, AGOR-16 and its near-sister ships, USS PIGEON and ORTOLAN. Designed in the mid-1960s and commissioned in the early 1970s, they suffered from severe cross-deck slamming and a sickening corkscrew motion in which the vessel rolled and pitched simultaneously in a resonant manner. Significant design advances have been made since then and there are many catamarans serving as ferry vessels in the marine industry.

Part of the problem in the 60s was the lack of a technology base. "Despite the lack of prototype experience and the relatively large size and cost of these vessels, little effort has been devoted to their development, particularly to their behavior in a seaway" (ref. 5.54). Model tests were run only on the larger Auxiliary Submarine Rescue (ASR) hull and then only to determine cross structure loads at a far lighter displacement, and hence larger cross structure clearance than the final design of the ships. The narrow hulls had little pitch damping and the natural periods of pitch and roll were rather close together, and were easily excited by the prevailing conditions in the North Atlantic. Effective operating time on station during the first winter was less than 50%.

A substantial effort was undertaken to find a remedy for the ship's severe seakeeping problem. Full scale and model tests, along with analytical studies of changes in parameters and details were run, culminating in the fitting of a hydrofoil between the hulls of all three vessels, and a modification of the cross structure on the ASRs to increase under-deck clearance. These modifications greatly improved the seakeeping of the vessels so that they became acceptable platforms for their open-ocean missions. The HAYES was removed from service in 1983 due to its high cost of operation (it is nearly twice the displacement of the next largest oceanographic vessels), but is currently being converted to an Acoustic Research ship. The ASRs are still in active service.

The use of a hydrofoil with the fine, high length to beam ratio catamaran hulls has proven to be a very favorable development. The foil increased pitch damping considerably, and increased the pitch natural period about nine percent. A somewhat unanticipated result was the influence on roll. Roll damping was increased, lengthening the roll period by fifteen percent. The increase in the separation of the natural periods, along with their increase somewhat out of the range of commonly encountered wave periods, greatly improved the vessel's motions.

The analytical studies revealed for the first time some of the effects of various proportions on the loads and motions of catamarans. Variations in displacement, length, length to beam, and beam to draft ratios, and in hull separation were examined with and without hydrofoils. It was seen that changes in hull proportions had little effect on cross structure loads, while reductions and increases in hull separation from the design condition increased the loads. The hydrofoil decreased loads, both through reduced motions and by providing additional structural support. Beam to draft and cross structure clearance strongly influenced slamming incidence and pressure. Cross structure slamming was seen to almost always limit operations before other motions.

Catamarans with hydrofoils need to be modified to avoid broaching the foils, such as using deep and narrow hulls to increase foil immersion and separating the hulls as far as possible. The greater the span of the foil, the greater the damping, and the less the chance of foil emergence. Due to the roll damping of the foil, such vessels can tolerate increases in hull separation and still keep the roll and pitch natural periods sufficiently spread. A catamaran without a foil must reduce hull separation for this reason.

The somewhat unfortunate experience of the HAYES greatly accelerated the development of analytical tools for the design of catamarans and lead to a greater understanding of the strengths and weaknesses of the type. Using the resources available today, catamaran designers can now be confident of achieving acceptable results.

The Royal Australian Navy's new catamaran minehunters represent the latest development in slow-speed catamarans, as well as being a totally new concept in mine-countermeasures vessels. The modular concept in weapons systems adopted by the Australians put a premium on deck space. A catamaran configuration maximizes deck space for a given displacement. Low displacement generally means lower cost and reduced pressure signature. (Important when dealing with certain types of mines). This design further exploits the high deck area and metacenter of the catamaran configuration by placing the main propulsion engines in the deckhouse. Power is supplied to propulsion and steering units through a hydraulic system which greatly reduces the vessel's underwater acoustic signal. The hulls are constructed using sandwich skin glass-reinforced-plastic in the manner of the Swedish monohull minehunters. The hull form somewhat resembles the HAYES, but also includes an anti-pitch fin. Just entering trials, if they are successful they will represent a very innovative solution to the mine warfare needs being faced by many nations. Monohulls recently designed for the same mission are at least twice the displacement of the Australian design.

5.4.3 Small Waterplane Area Twin Hull (SWATH)

The term SWATH refers to a family of vessels in which most of the volume of the buoyant hull has been submerged below the water's surface, leaving only a few relatively thin struts piercing the air/water interface. This action reduces rather substantially the exciting force that wave action is able to induce on the forced, damped, harmonic oscillator the ship at sea represents. Two typical SWATH designs are shown in Figure 5.8.

Keeping this simple concept in mind is often difficult when considering the SWATH ship, since it is perhaps the most widely discussed and little used naval development since the early days of the development of the submarine. Partly this stems from the innate conservatism the sea imparts to all mariners, partly from the long-held supremacy of speed as a maritime virtue, and the attendant lack of understanding of the value of seakeeping in ship effectiveness. Many of the developments in seakeeping evaluation (ref. 5.12) were driven by the need to quantify the seakeeping improvements the SWATH concept promised.

Although the ideal of separating the buoyant and the working sections of a vessel is simple, the reality may not be, for this separation creates problems of stability, arrangements, powering, etc., to be solved, often in a manner alien to normal displacement ship practice. The benefits of SWATHs can also be exploited in ships with more or fewer hulls, and combined with conventional displacement hulls, or with dynamic lift devices such as hydrofoils and air cushions to form various hybrid concepts. The reader is referred to the Advanced Marine Vehicle chapter in Bhattacharya (ref. 5.49) for a brief review of the various hybrid concepts, which will not be further discussed due to their lack of applicability to the SRA mission.

There are a number of characteristics which separate SWATH ships from ordinary displacement vessels, both advantages and limitations, and an understanding of these characteristics is vital to understanding the SWATH concept and how best to utilize it.

The first and most important characteristic is the outstanding seakeeping performance and sustained speed in the seaway of SWATH vessels. This is the main reason for their existence as a viable alternative to conventional vessels. Comparing vessels of the same displacement, SWATH ships display as little as 10% of the motions of conventional hulls. Stating the comparison a little

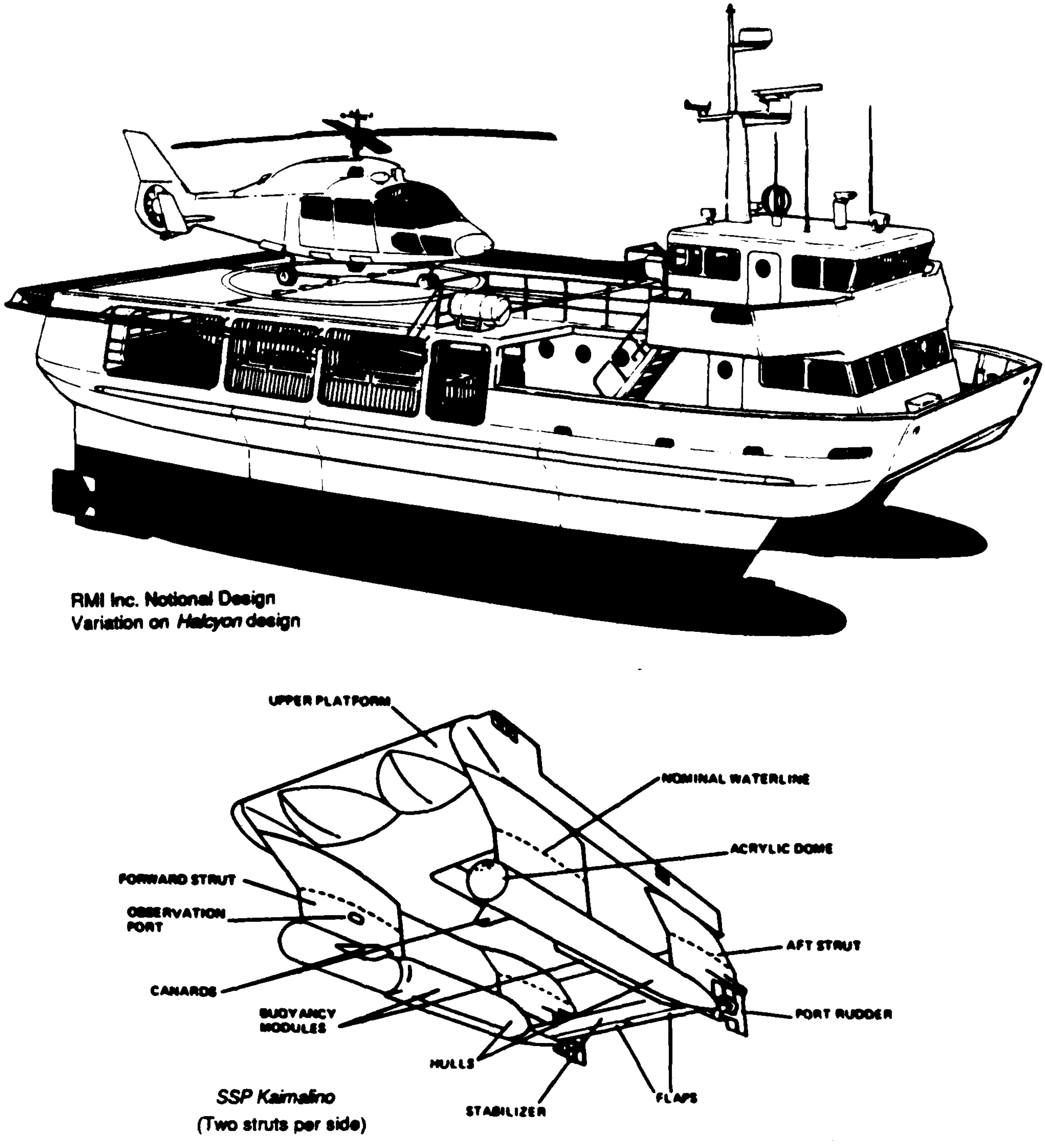


FIGURE 5.8 The SWATH Configurations

differently, a SWATH vessel will have motions comparable to a vessel perhaps four times its length and ten times its displacement. This advantage holds for high speeds as well as when dead in the water, and nearly regardless of heading, although following seas can present problems for SWATHs.

There is a cost for this increase in performance, from several factors. Pound-for-pound, SWATH ships tend to be more expensive than conventional ships, due to the increased complexity of their structure and engineering plant, and the loads on the cross structure and struts imposed by their arrangement. The displacement of a SWATH to carry the same payload as a monohull will be higher due this added structure, as well as the extra surface area needed to enclose a given volume with the SWATH platform, which increases outfit as well as structural weight fractions. These differentials have been shown (ref. 5.47) to decrease with increasing SWATH size, and numerous studies (refs. 5.55, 5.56, 5.57, 5.58) have shown that a small SWATH in rough seas can have the mission effectiveness of a much larger and more expensive conventional platform, so that on an equal-effectiveness basis, a SWATH is comparatively cheap, if it can be smaller than the conventional hull it replaces.

Operators of SWATH vessels must pay much closer attention to the loading of their vessels, since the low waterplane area which defines them as SWATH also leads to low values of Tons-Per-Inch-Immersion, Moment-to-Trim and Moment-to-Heel. With careful loading, and possibly counter-ballasting, this effect can be minimized. Once again, large SWATHs have less of a problem due to the the lower ratio of the weight of individual loads they carry to their overall displacement (a helicopter weighs the same on a small SWATH as a large one, but is a much larger portion of the overall ship displacement). Due to the relatively finite size limit of ocean waves, a large SWATH can have a proportionally larger waterplane area than a small SWATH and still retain excellent seakeeping (ref. 5.58).

SWATH resistance for equivalent displacement monohulls tends to be higher than for displacement monohulls. The underwater surface area of a SWATH hull is higher, leading to greater frictional resistance, and the wave-making drag is far from absent. As with all displacement vessels (although perhaps a bit more sensitive than most), high speed (above 25 knots) in a SWATH is very costly, reducing range and payload, forcing even more structural complexity, and/or exotic materials, hull distortions, or various expensive combinations of all of the above.

SWATH high speed turning diameters of two ship lengths are readily achieved, and are comparable to five ship lengths for conventional ship performance. Zig-zag tests show SWATH ships have half the overshoot of conventional ships. At slow speeds, the widely separated propulsion units aid maneuverability. Lack of ship motions and reduced drift due to wind and wave action make station keeping performance far higher than for comparable monohulls (ref. 5.84).

A proportion of unusable volume is a problem for the SWATH designer. Some spaces, such as the struts, are of such a size and shape to make them impractical for any use, and even though areas such as the underwater hulls may be used for tankage, there tends to be more enclosed volume available than there are profitable ways to use it. This also leads to greater hull weight for a given payload as noted for structural reasons. Also, high volume and surface area and numerous flat surfaces lead to high radar, visual and infrared signatures for SWATH vessels.

Axi-symmetric bodies translating through a fluid tend to generate a de-stabilizing moment that increases with speed. This was first treated analytically by the airship designer Munk, and is sometimes referred to as the "Munk Moment". Torpedoes and submarines also are subject to these de-stabilizing effects, as well as other contributing phenomena due to the effect of the free surface when running shallow. SWATHs too face these problems, and this makes stabilizing fins aft virtually mandatory for successful SWATH operations at reasonable speeds (although shaping of the struts and auxiliary hulls just above the water's surface have been tried as alternatives). Underwater appendages are always somewhat vulnerable, and add to the drag of the vessel. Most SWATH designers have decided that as long as fins are necessary, they may as well be active (see the sections on motion stabilization for general discussion of the techniques) in order to further decrease ship motions, improve maneuverability, and make minor trim corrections. However, fixed fins provide the necessary pitch stability and damping to improve motions.

Many tools for the design and analysis of SWATH vessels have been developed in recent years, largely by the U.S. Navy, but important contributions to the state-of-art have been made by the British, Japanese and Canadians.

Resistance of SWATH vessels is still primarily predicted by the method of Chapman which dates back to 1972 (described in ref. 5.59). The SWATHGEN program (ref. 5.60) and others developed at DTNSRDC have enabled the evaluation and optimization of hull forms with contoured and cambered hulls and struts to be easily made. Seakeeping performance can be evaluated using the programs developed by Lee (ref. 5.61) and also Nethercote (refs. 5.62, 5.63), Hosoda (ref. 5.64), Chryssostomidis (ref. 5.65) and McGregor (ref. 5.66). The U.S. Navy's Ship Structural Design Synthesis program has been used to examine variations in SWATH structural parameters (ref. 5.67), but perhaps the most important development in SWATH design tools has been the integration of the various elements of design calculation into synthesis models such as ASSET/SWATH (ref. 5.68). Starting with values of basic parameters such as speed and required payload, they are capable of producing a range of feasible SWATH designs. This enables the ship designer to make rapid studies of parametric variation to find the most suitable vessel for a given set of requirements. Since ASSET/SWATH was developed originally for combatant type vessels with specialized weight groups, it may need modification for use on buoy tender designs.

The results of use of these tools, backed up by extensive model and full scale testing, has been a change in the parameters of the "typical" SWATH (which is still largely a creature of paper studies). An increase in GM_L in recent designs has improved seakeeping in following seas substantially, with only mild degradation in performance at other headings. The depth of submergence of the underwater hulls has been reduced versus earlier designs, due to experimental evidence that propeller emergence is not the problem it was thought to be, the realization that increasing hull submergence did not significantly decrease wavemaking unless taken to radical extremes, and the improvements in wavemaking resistance of shallowly-submerged hulls possible with highly contoured hulls. This is good news for buoy tenders and other vessels with draft limitations. The prismatic coefficient, once held rather high for resistance reasons, has been lowered due also to hull contouring.

These changes in gross parameters have in some cases been driven by changes in the detail design of SWATHs; there are many other detail changes which have not had such a great effect on the overall parameters, but which nonetheless have done much to make

SWATH a viable concept. Contoured hulls and struts have already been mentioned. Present practice is to use a large midship bulge to reduce the low and moderate speed wavemaking hump present in purely prismatic SWATH lower hulls, while fore and aft bulges attempt to minimize the high speed penalty the midship bulge exacts. The result is a hull form with superior resistance characteristics over a wide range of speed/length ratios. These complex shapes would be replaced by simpler rolled cylindrical and conical sections on actual ships using a flat baseline for ease of drydocking (i.e., they would not be axi-symmetrical). The advantages of contoured hulls, long shown to be theoretically superior (ref. 5.60 is the latest reference), has recently been borne out by model testing (ref. 5.69).

Another change in lower hull design has been to use elliptical or oval sections for the lower hulls. This reduces the draft for a given displacement, while slightly increasing surface area and thus drag, but the primary benefit is the increase in damping it provides. This decreases ship motions at most operational wave periods, while increasing motions at long wavelengths, allowing the SWATH to wave-follow in longer, larger waves, thus increasing effective cross-structure height and reducing cross-structure slamming. Another way of achieving an increase in damping is to use canted struts (ref. 5.70). Canted struts also reduce the dynamic bending moment, and may be cambered to correct bow down trim and high speed immersion experienced by most SWATHs.

Rudders have been a problem for SWATH designers. For best effectiveness, most existing SWATHs have placed them aft of the propeller in its slipstream. This increases the length of the vessel, and the large rudder forces necessitate a large and heavy structure. This aft protrusion usually pierces the water's surface, but has little buoyancy, leading to a offset between the LCB and LCF, causing trim problems and possibly degrading seakeeping. Attempts at using rudders at the forward edge of the struts and other arrangements have lead to unsatisfactory performance. Canting the aft stabilizer fins and properly configuring an automatic control system to use these angle stabilizers and the forward canards, causes equal or greater turning ability than achieved by conventional rudders (ref. 5.70). An early manned model of the TRISEC used combined rudder/stabilizers. This elimination of the rudders means two fewer underwater appendages, and allows for more freedom in hull and strut design, as well as improving propeller efficiency and reducing vibration.

Perhaps the most important recent development has been the building of SWATH ships around the world, and the accumulation of operating experience. In the United States, three small SWATHs have been constructed by interests in the San Diego area. The first, the SUAVE LINO was conceived as a sport fishing vessel with improved seakeeping abilities. This 65 ft, 53 tonne, aluminum hull vessel was successfully tested by the Navy (ref. 5.71) and was chartered by the U.S. Army Corps of Engineers for use as a survey vessel. Ocean Systems Research has recently completed a somewhat larger vessel, the CHUBASCO, which is unique in that the main engines are contained in the lower hulls.

RMI Inc. in National City, California undertook the construction of the demonstrator craft HALCYON, launching it in March of 1985. This vessel is also aluminum, and is roughly the same size as the SUAVE LINO. HALCYON is notable for its careful structural design to reduce fatigue, and the sophistication of its systems, including counter-rotating variable pitch propellers (CRP) propulsion and micro-processor controlled motion stabilization. Extensive trials of the HALCYON were conducted by the U.S. Coast Guard, DTNSRDC and NAVSEA (ref. 5.72). RMI is now in receivership and the vessel has been purchased by Ocean Systems Research.

The KAIMALINO, the oldest in the fleet, has been the beneficiary of continued development over the years (ref. 5.73). Its automatic motion control system has undergone several revisions to increase its effectiveness, even at low speeds, and to aid in maneuvering by controlling heel during high speed turns. Its propulsion systems has been upgraded by redesigning some deficiencies in its chain drives, and by providing a hydraulic low-speed auxiliary propulsion system driven by its generators. The addition of strap-on steel and foam buoyancy modules (installed in the water using SCUBA divers) enabled an increase in displacement from 190 to 220 tonnes without reducing cross structure clearance.

At least one SWATH fishing vessel has been constructed in the U.S., the F/V CHARWIN constructed by St. Augustine Trawlers in Florida (ref. 5.74). It was used primarily as a scallop dredger, which seems an unlikely mission for a weight sensitive vessel. Even though its deck load of scallops often immersed its aft deck when returning from the grounds, its owners felt it had significant advantages over conventional hulls, particularly when setting and retrieving the unwieldy dredges in rough waters off the Atlantic Coast of Florida near the edge of the Gulf Stream.

The newest U.S. SWATH is the T-AGOS-19. This ship is designed to steam slowly in the open ocean, towing a linear array of hydrophones to detect submarines. This primarily slow speed design utilizes canted stabilizer fins for steering and oval hull sections. It displaces about 1700 tonnes on a length of 175 ft, with a 60 ft beam and 18 ft draft. Presently under construction by McDermott Marine in Texas, it is due to be launched within the year.

The latest Japanese SWATH launchings have included the largest SWATH ship built to date and some of the smallest; the Support Vessel KAIYO, and a series of SWATH recreational powerboats. The KAIYO is extensively outfitted to support underwater operations, with a saturation diving system, Dynamic Positioning System as well as 4-point mooring equipment, several cranes and gantries, etc. It has diesel-electric propulsion for quieter running, and displaces about 3000 tonnes on its 60 meter length, 28 meter beam and 6.3 meter draft. The 15 meter pleasure boat being marketed by Mitsui is a futuristically styled, highly powered recreational craft designed to compete in the top end of the yachting market, and follows a series of smaller prototypes.

Sea trials have been conducted on all of the world's existing SWATH vessels (11 vessels larger than 10 meters as of August 1987), and there is a large body of available information resulting from these trials. The KAIMALINO has been the most extensively tested, with Fein (refs. 5.75, 5.76, 5.77, 5.78) Kallio (ref. 5.79), Woo (ref. 5.80), and Stenson (ref. 5.81) being the essential references. Two series of comparative trials where run by the U.S. Coast Guard in cooperation with the U.S. Navy comparing the performance of the KAIMALINO to other conventional vessels. The first set of trials compared the seakeeping and the physiological response of the crews of the KAIMALINO and 2 Coast Guard Cutters, a 95 ft Patrol boat, and a 378 ft High Endurance cutter. The KAIMALINO far outperformed the patrol boat, and was superior on some headings for some motions to the much larger High Endurance Cutter. Woolaver (ref. 5.55) and Wiker (ref. 5.82) document these tests and the methodology used to gauge physiological response (fatigue, motion sickness, etc.).

The second set of trials compared the buoy tending performance of the KAIMALINO with that of a Coast Guard 180 foot buoy tender (Coe, ref. 5.83) and Strickland (ref. 5.84) documents this series of tests. Both ships simultaneously worked a second-class can buoy (2CR). The 180 foot buoy tender MALLOW was scheduled to work a larger 8-26LR buoy, but did not. Two quotations from Strickland (ref. 5.84) are worthy of note. First,

"After completion of the buoy-tending exercises, discussions were held with the participating officers and crew covering their experiences. Two notes were taken in this regard. The crew of the MALLOW was noticeably tired at the end of the day. The SSP buoy deck-crew leader said his men were not tired when the trials ended."

"When it became evident that it would not be possible to work the larger 8-26LR buoy, the Commanding Officer of the MALLOW was asked to comment on the differences he would have expected to see relative to working the smaller buoy. His response was that he would not have worked the larger buoy at any direction other than head seas because ship motions were excessive and the safety hazard too great."

Second,

"This study has shown that the SWATH concept offers the possibility of improved buoy-tending productivity while providing a safe environment in which to conduct these hazardous operations. The high freeboard and deeper draft of the SWATH concept must be taken into proper consideration when evaluating its suitability as a buoy tender."

Test results for the SUAVE LINO can be found in Jones (ref. 5.71), and for the HALCYON in Coe (ref. 5.72). The Japanese have not been as forthcoming in publishing their test results, but some information can be found in their rather general articles such as Mabuchi (ref. 5.85).

It is clear that SWATH vessels work, and hold real promise for missions where seakeeping has a high value. Limited tests have shown that they can perform buoy tending tasks on light weight aids to navigation as well as, or better than, conventional hulls. In recognition of this fact, the Naval Ocean Systems Center has published a report in which possible buoy tender SWATH designs are proposed (ref. 5.86).

SWATH buoy tending feasibility studies are currently being done at the David Taylor Naval Ship Research and Development Center (DTNSRDC). These studies indicate that draft requirements rule out consideration of a SWATH for the coastal, WLM, buoy tending missions (ref. 5.87). Feasible WLM SWATHs would have too great a draft. Although the ability to break ice is a requirement for the seagoing, WLB, buoy tending mission, DTNSRDC decided to look at a non-ice breaking SWATH WLB first, and then study the feasibility of building an ice breaking SWATH WLB. There is good reason to believe that a non-ice breaking SWATH WLB would be a candidate for the seagoing buoy tending mission, if ice breaking were not essential. No conclusions have been drawn as to the feasibility of building an ice breaking SWATH WLB at this time. The studies at DTNSRDC have yet to address this case.

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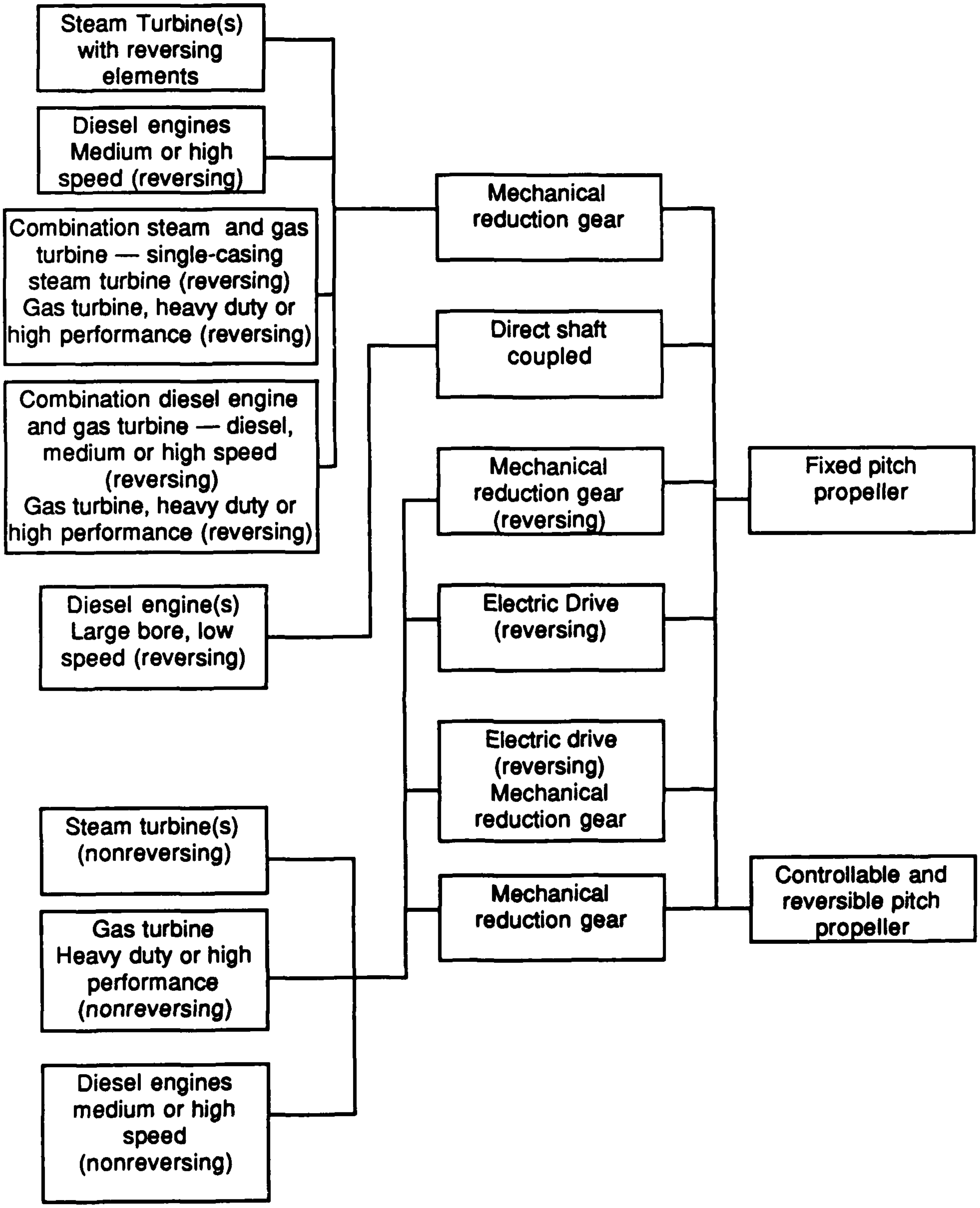
6.0 PROPULSION SYSTEMS

The basic operating requirement of the propulsion system is to provide the thrust necessary to propel the vessel at the required sustained speed. This should be achieved as economically and reliably as possible while providing suitable maneuvering capabilities. Typically, a propulsion system consists of prime movers, a transmission system, shafting, and propulsor. A variety of prime movers, transmissions, and propulsors are designed to satisfy specific performance requirements such as size, weight, maintenance and fuel consumption. The alternatives for each of these will be described in further detail in this chapter and are summarized in Figure 6.1

6.1 PRIME MOVERS

The purpose of the prime mover is to convert fuel into usable mechanical energy, usually in the form of a rotating shaft. The majority of prime movers have a rated shaft horsepower (SHP) which is the power available at a determined speed. The most common prime movers at present are diesel engines, gas turbine engines, and steam turbines. Descriptions and analysis of current systems, discussion of their advantages and disadvantages, and future trends will be provided for these prime movers. Other exotic power plants which utilize nuclear and fuel cell technology will be discussed briefly.



Marine Engineering SNAME, 1971

FIGURE 6.1 Alternatives in the Selection of a Main Propulsion Plant

6.1.1 Diesel Engines

The diesel engine is utilized on all types of marine vehicles both in the merchant marine and in the navies of the world. It is by far the most commonly used marine propulsion system in modern vessels. As stated in Reference 6.1, the high percentage of diesel engines is due mainly to their low initial and operating costs, reliability, and high adaptability to ship operation.

Diesel engines are referred to as being high, medium, or low speed and usually categorized as follows:

Engine Speed Classifications	Piston Speed (ft/min)	Shaft Speed (rev/min)
Low Speed	1000-1500	100-514
Medium Speed	1200-1800	700-1200
High speed	1800-3000	1800-4000

The arrangement of the cylinders and pistons also differs to achieve certain performance characteristics. In-line, V-block, and vertically opposed are presently available, and operate on either the two or four-cycle principle.

A sample of the state-of-the-art in diesel engines is listed in Table 6.1. The engines were selected from a variety of

TABLE 6.1

DIESEL ENGINE DATA

Manufacturer	Model	Cost ($K)	Continuous Power (SHP)	Speed (rev/min)	Cycle (2/4)	Weight (tons)	Fuel Consumption (lb/SHP-hr)
Ruston	6AP230	172	1,030	720	4	10.1	0.345
GM	8-645E6	285	1,050	900	2	10.2	0.392
Caterpillar	3512-V12	109	1,060	1200	4	7.2	0.312
Krupp MaK	6M 282	160	1,060	750	4	8.3	0.348
MAN B&W	6L23/30	269	1,086	825	4	13.6	0.307
GMT	B230.4	185	1,140	1200	4	5.7	0.323
Wartsila	6R22	200	1,260	900	4	9.7	0.312
Cummins	KTA 50-M	122	1,250	1,800	4	6.0	0.351
Bergen Diesel	KRM-6	325	1,500	750	4	14.0	0.311

manufacturers, each having an output of approximately 1000 shaft horsepower (SHP). This particular output was chosen based on the historic power requirements of Coast Guard buoy tender engines. The main reason for obtaining information on several engines of the same rating is to allow for comparisons of various aspects of engine design.

One of the most important factors in diesel engine selection is cost. Cost includes initial cost, recurring costs and contingency costs. The initial or installed cost is dependent on factors such as material and labor cost, the similarity of a plant with those previously produced, and manufacturer's existing work backlog. As a result, prices are subject to fluctuations which depend on the current status of the industry. The initial price per rated shaft horsepower is shown in Figure 6.2 for each selected engine. Generally, the initial price includes vital auxiliaries, control features, and delivery. From the research and engine data, the following trends with regard to initial price per horsepower output have been established:

- Decreases with increased engine speed
- Lower for 4-cycle engines of similar design and output than for 2-cycle engines, due to smaller engine component sizes

As mentioned in the previous paragraph, total life cycle engine costs also include recurring and contingency costs. Recurring

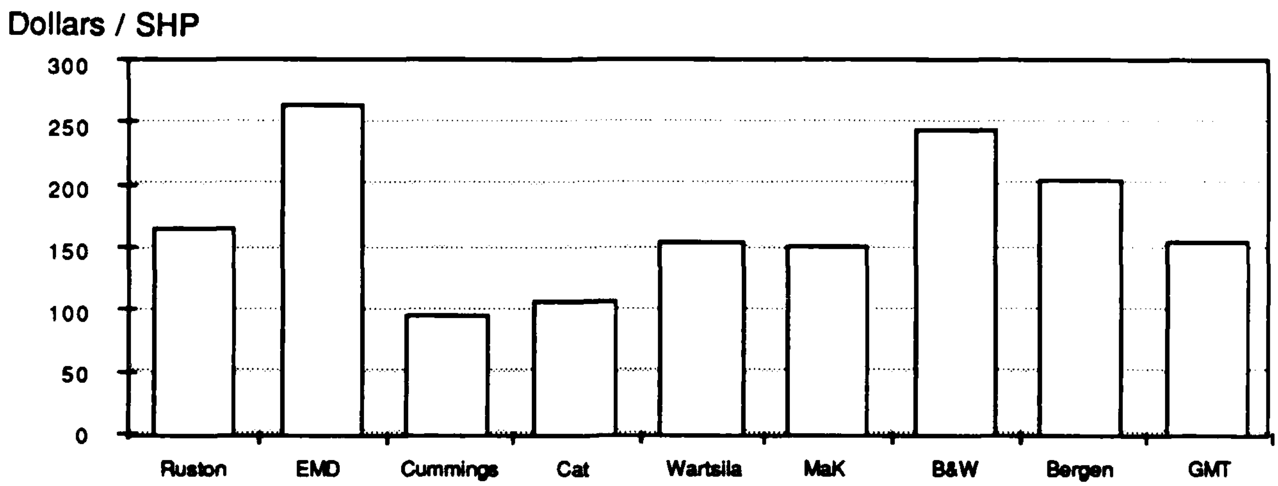


FIGURE 6.2 Diesel Engines — Initial Cost per Shaft Horsepower

costs include expenses for fuel oil, lube oil and routine maintenance. Contingency costs are dependent on engine reliability. Although actual dollar amounts from these costs are difficult to obtain, information such as specific fuel consumption and time between engine overhauls may be used to compare their relative operating costs. Specific fuel consumption for each of the selected engines is shown in Figure 6.3. The following fuel consumption trends have been identified:

- Increases with increased engine speed
- Decreases with larger stroke/bore ratio
- Decreases with larger bore and fewer cylinders

Maintenance costs show the following trends:

- Increase with increased engine speed
- Higher for 2-cycle engines than for 4-cycle engines due to the higher loads on the engine components.

Figure 6.4 shows a chart of diesel engine weight per shaft horsepower output, referred to as specific weight, for each of the selected engines. Specific weight trends are as follows:

- Decreases most for increased engine speed
- Lower for 2-cycle engines of comparable size and design than 4-cycle engines.

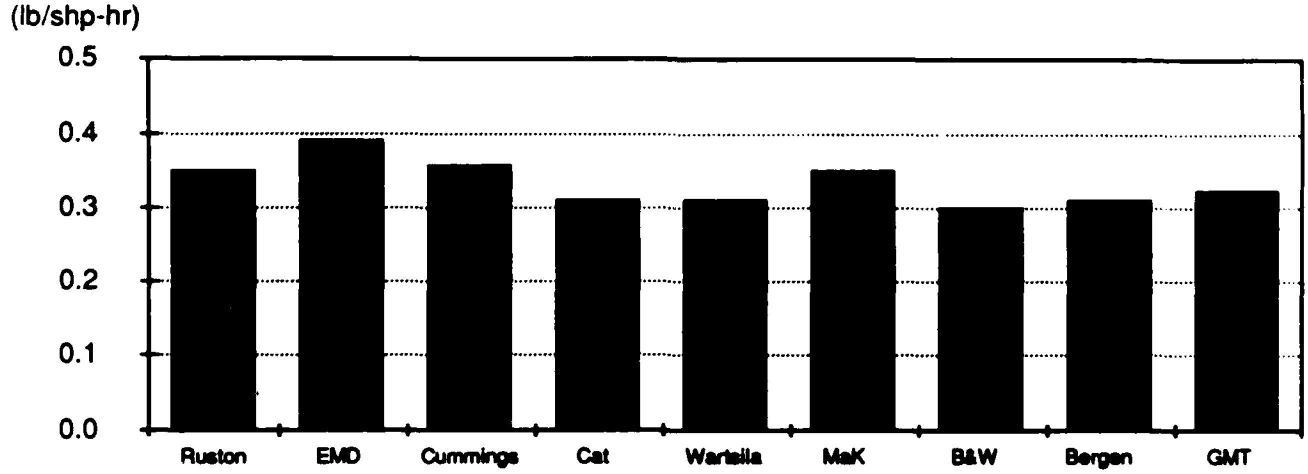


FIGURE 6.3 Diesel Engines — Specific Fuel Consumption

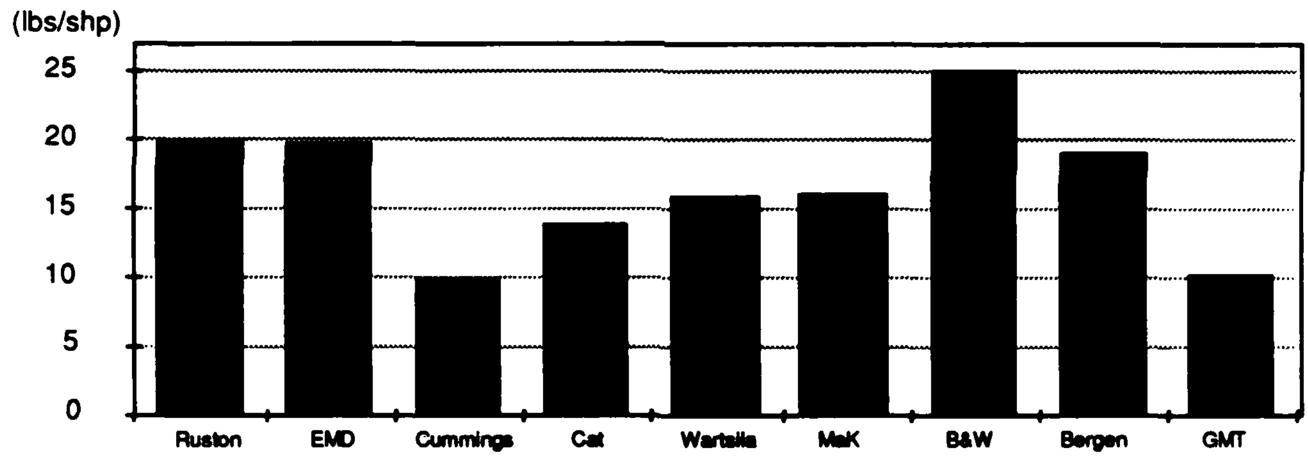


FIGURE 6.4 Diesel Engines — Specific Weight

In general, high-speed diesel engines are more practical for situations where vessel speed is important and therefore engine specific weight is critical. Slow-speed diesel engines are larger than high-speed engines although this difference diminishes somewhat for 2-cycle engines compared to 4-cycle engines. Although initially less expensive, the high-speed engine has higher fuel, operating and maintenance costs than slow-speed engines.

Improvements in diesel engine technology are being pursued by manufacturers in the areas of engine efficiency, waste heat recovery, use of lower grade fuels, performance monitoring, exhaust gas emissions, and noise control. These topics are discussed in several journal articles which are summarized below.

In modern diesel machinery, higher engine efficiencies have been obtained primarily by increasing cylinder temperatures and working pressures, reducing exhaust gas temperatures, constant pressure turbo-charging and variable injection timing. Further increases in engine efficiency through raising of the maximum cylinder pressure must be such that the pressures do not lead to undesirable deformations, stresses or wear rates. Progress in this area has been achieved through the use of alloys and heat resistant coatings for cylinder linings and other engine components. A general discussion of improvements in piston performance of recent General Electric marine diesels is given in Reference 6.2. A detailed analysis of testing on materials being performed by Mitsui Engineering and Shipbuilding Company is given in Reference 6.3. Reductions in fuel consumption are also possible through the utilization of waste heat energy. About thirty percent of the available energy from the fuel is lost to exhaust energy. As described in Reference 6.1 and Reference 6.4, the waste heat recovered at present is used primarily for heating of bunker fuel and the generation of electricity. A comparison of a turbo generator with other electricity generation alternatives is shown on page 70 of Reference 6.4. A likely future use is augmentation of the propulsion power. This is discussed further in Section 6.1.4. Development of more effective heat recovery systems will continue, primarily satisfying ship electrical power requirements; however, cost effectiveness is the largest obstacle particularly when maintenance cost and availability are brought into the analysis. Ironically, another setback to waste-heat recovery is the improvement in engine efficiency. Less fuel is required for propulsion, thus less available waste heat energy.

The most successful use of exhaust gas energy has been through the use of turbochargers. Turbochargers increase the mean effective pressure and therefore the outputs of the engine. Recent improvements in turbocharger design, as reported in References 6.5, 6.6 and 6.7, have resulted in efficiencies as high as 72.5 percent. The improvement of 4.5 percent over previous models provided a reduction in fuel consumption of 3 g/bhp-h. A significant development is the use of compound turbochargers which may offer an additional 4 percent in fuel savings. A comparison of fuel consumption between compound turbocharged systems and conventional engines is given on Page 42 of Reference 6.8. The continued growth of compound turbo systems is expected.

Continued use of lower grade fuels is anticipated in future engines. Fuel viscosities of up to 700 cSt at 50 deg. C are permitted for use in both 2 and 4-stroke engines. The unit of kinematic viscosity is the stoke ($1 \times 10^{-4} M^2/S^2$), but for convenience, the centistoke, cSt (1/100 stoke), is widely used. Fuel viscosities are commonly measured at 50 degrees Celsius. Changes in the combustion characteristics that might be expected with future low-grade fuels are discussed in Reference 6.9. If prices or availability of conventional diesel fuel deteriorates, the use of alternative fuels may have to be considered. References 6.1 and 6.10 discuss some of the possibilities including the use of crushed coal combined with heavy oil along with some of the performance aspects which must be considered when using alternative fuels.

Improvement in engine efficiency through the use of performance monitoring, especially in the part-load condition, is expected. References 6.1 and 6.4 point out some of the advantages that may be achieved through the use of appropriately designed engine components, selection of suitable sensors, and proper software utilization. Engine condition monitoring may also be useful in planning a cost efficient maintenance schedule, reducing the number of personnel needed to man machinery spaces, extending engine life, and reducing the possibility of engine failure.

The disadvantages of diesel engines, such as exhaust gas emission and noise, will be reduced in future designs. Products of incomplete combustion that cause concern are the nitrogen oxides, carbon monoxide, and unburned hydrocarbons. The worst of these, nitrogen oxides, can be reduced presently by injecting a fuel/water emulsion into the combustion chamber. A future possibility as reported in U.S. News and World Report, February 16, 1987, page 72, is an inexpensive chemical elimination process; however, this has only been demonstrated in the laboratory at present. Increasing attention is also being paid to noise levels. The reduction of engine noise is expected in the future through the use of exotic materials and more efficient silencers as described in Reference 6.1. This is important in the WLB/WLMs performance of defense missions.

6.1.2 Gas Turbines

The basic gas turbine operates on the Brayton thermodynamic cycle. The simple Brayton cycle consists of the following elements:

- Compression of air
- Heating of air under a constant elevated pressure in a combustion chamber.

- Expansion through a turbine

The power produced by the turbine is greater than the power required by the compressor. The excess power is available at the drive shaft for propulsion.

At moderate turbine inlet pressures and temperatures and with the component efficiencies attainable when gas turbines were first developed, the simple cycle gas turbine operating with atmospheric air and burning light distillate fuel was limited in output and specific fuel consumption. However, during the subsequ^nt stages of progressive development, the cycle efficiency has been greatly improved by the following changes:

- Higher compressor pressure ratios
- Higher turbine inlet temperatures which were permitted by metallurgical and cooling developments.
- Improved compressor and turbine stage efficiencies
- Increased compressor pressure loading per stage
- Improved combustion efficiency
- The introduction of intercooling in the process of compression.
- The introduction of regeneration (recovery of waste heat from the turbine exhaust and subsequent addition to the compressor discharge air flow before it enters the combustion chamber).
- The introduction of reheating (a second combustion chamber between the compressor turbine and power turbine).
- Further waste heat recovery

The simple cycle and improved efficiency cycle arrangements are shown schematically in Figure 6.5.

Several selected gas turbines which are suitable for marine propulsion are listed on Table 6.2. All the selected gas turbines have power outputs in the vicinity of 1000 HP in order to compare turbine characteristics with diesel engines. The information on the gas turbines featured in this section was obtained from their respective manufacturers. It is used to show general trends of gas turbines.

The initial cost of selected gas turbines per horsepower output is shown in Figure 6.6. These values are higher than diesel engines since a higher degree of workmanship and materials are used on the construction of these engines.

Specific weight (lbs/SHP) of the selected gas turbines are shown in Figure 6.7. Gas turbines have the ability to provide a relatively high output for a given engine weight.

Shown in Figure 6.8 are the specific fuel consumption ratings of the gas turbines. Gas turbine engines generally require the use of light distillate fuel such as JP-4 or JP-5. Fuel consumption can be greatly improved through the use of regenerative or intercooling processes.

In summary, the main advantages of gas turbines for marine appli-

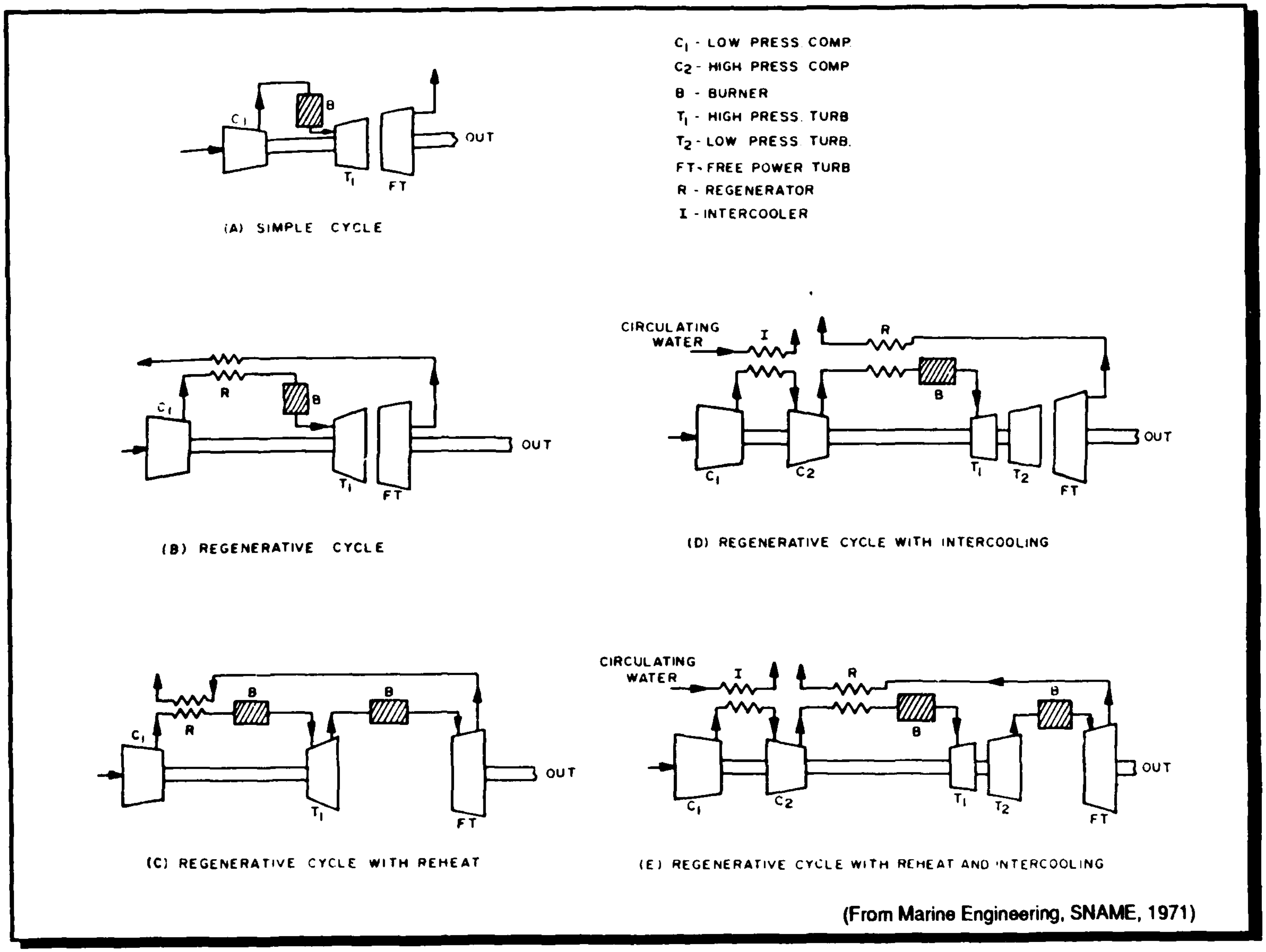


FIGURE 6.5 Brayton Cycle Arrangements

TABLE 6.2

GAS TURBINE ENGINES

Manufacturer	Model	Cost ($K)	Continuous Power (SHP)	Speed (rev/min)	Weight (lbs)	Fuel Consumption lb/Hp-hr
Allison (GM)	250 KD	143	550	6,016	240	0.612
Avco Lycoming	TF15	375	1,500	3,000	2,500	0.859
Garrett	IM 831-800	-	1,000	1,500	1,750	0.624

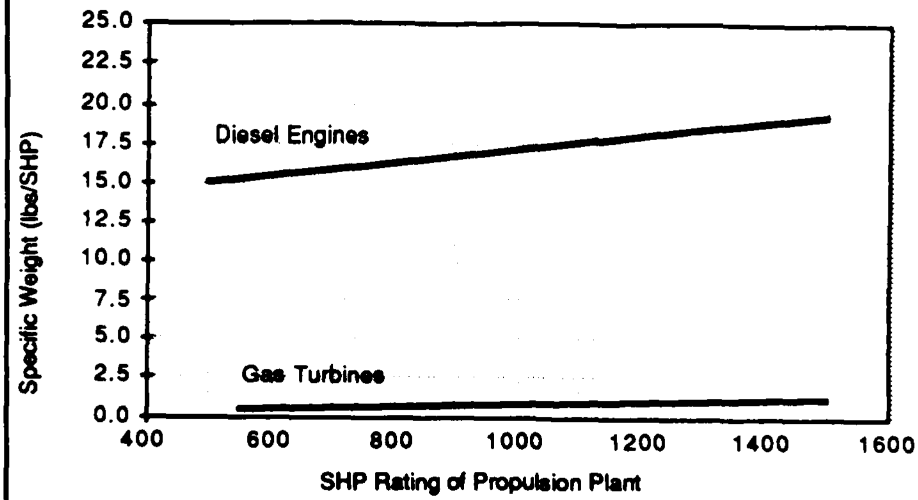


FIGURE 6.6 Initial Cost vs. Shaft Horsepower

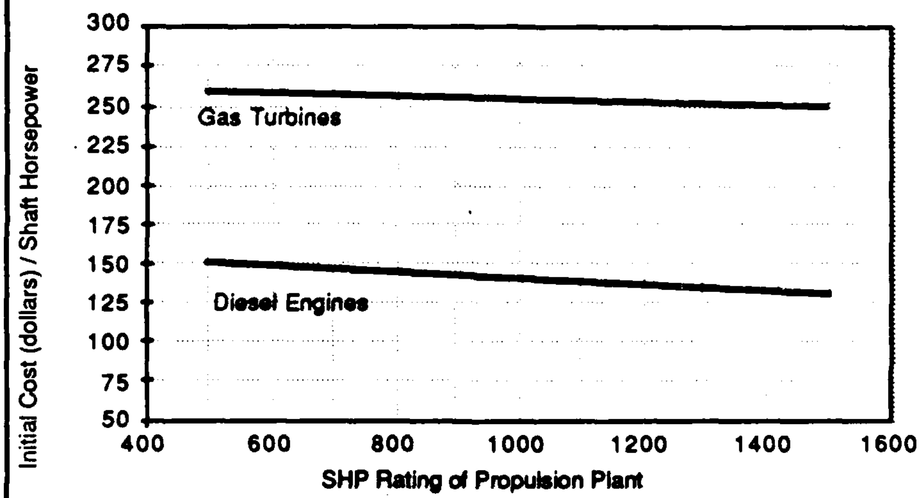


FIGURE 6.7 Specific Weight vs. Shaft Horsepower

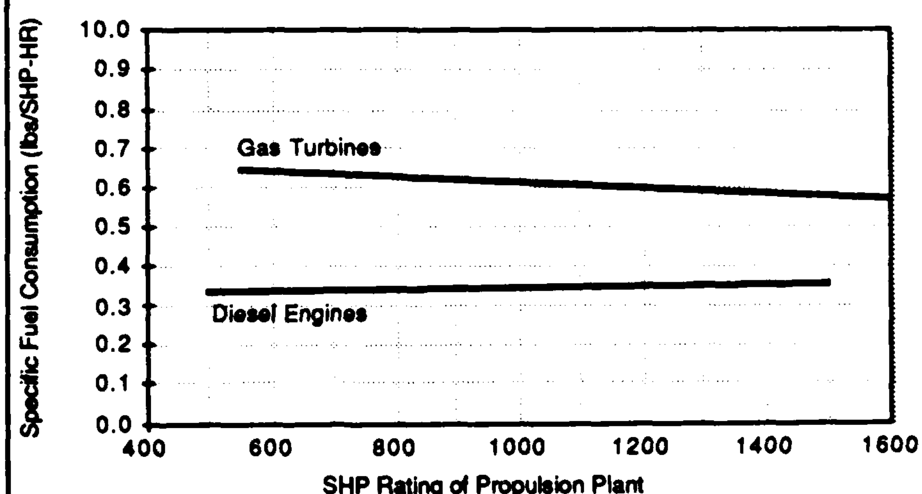


FIGURE 6.8 Specific Fuel Consumption vs. Shaft Horsepower

cations are its simplicity and light weight. As an internal-combustion engine, it is a self-contained power plant in one package with a minimum number of large supporting auxiliaries. It has the ability to start and go on line very quickly. Having no large masses that require slow heating, the time required to reach full speed and accept the load is limited almost entirely by the rate at which energy can be supplied to accelerate the rotating components to speed. Additional advantages are its low personnel requirement and ready adaptability to automation. Its major disadvantages are its high initial maintenance costs and light distillate fuel requirement.

Improvements in fuel efficiency will be the major trend in the gas turbine industry. Leading the way will be the utilization of intercooled regenerative cycles. Descriptions and analyses of intercooled regenerative turbines that the Navy is considering are given in References 6.11 and 6.12. The Navy is also conducting an R&D program on ship propulsion dynamics and control systems for gas turbines. At the present this program is used for new ship design, propulsion plant R&D and for fleet hardware analysis and improvement. Details of this program are contained in Reference 6.13.

6.1.3 Steam Turbines

The basic steam propulsion plant consists of main boilers, steam turbines, a condensate system, a feedwater system and numerous auxiliary components necessary for the plant to function. As opposed to diesel engines and gas turbines which utilize existing designs, steam plants are designed for specific ship needs and requirements. This makes it difficult to do analyses of existing systems; therefore, only trends will be discussed.

The principal trend in steam technology has been the improvement of steam cycle efficiency through the increase in temperatures and pressures, introducing reheat, and reducing boiling exhaust gas temperature and excess air level. This has resulted in higher power installations for a given space or a reduction in the size and weight necessary for a given propulsion requirement. Obstacles which prevent further improvements are cost and materials. Boiler materials must withstand both high and low temperature corrosion and have the necessary strength at high steam pressures and temperatures. A thorough analysis of these fuel-saving features is contained in Reference 6.4.

The main advantages of steam turbines have been their reliability and low maintenance cost. Future developments will focus on increasing cycle efficiency with operating reliability remaining comparable to the present standard. Reference 6.14 discusses future developments in steam technology. The author concludes that the future of marine steam propulsion lies in the utilization of inexpensive fuel which diesel engines cannot use.

Buoy tenders need to be maneuverable vessels. One of the major disadvantages of steam plants is their lack of controllability. Without a technological breakthrough in steam plant design, installation of steam plants on buoy tenders would be impractical. Reference 6.15 contains a comparison between the operating life cycle costs of an efficient coal fired steam plant and a slow speed diesel concluded that the coal fired steam plant may be a viable future propulsion alternative if diesel fuel prices rise or supplies become scarce.

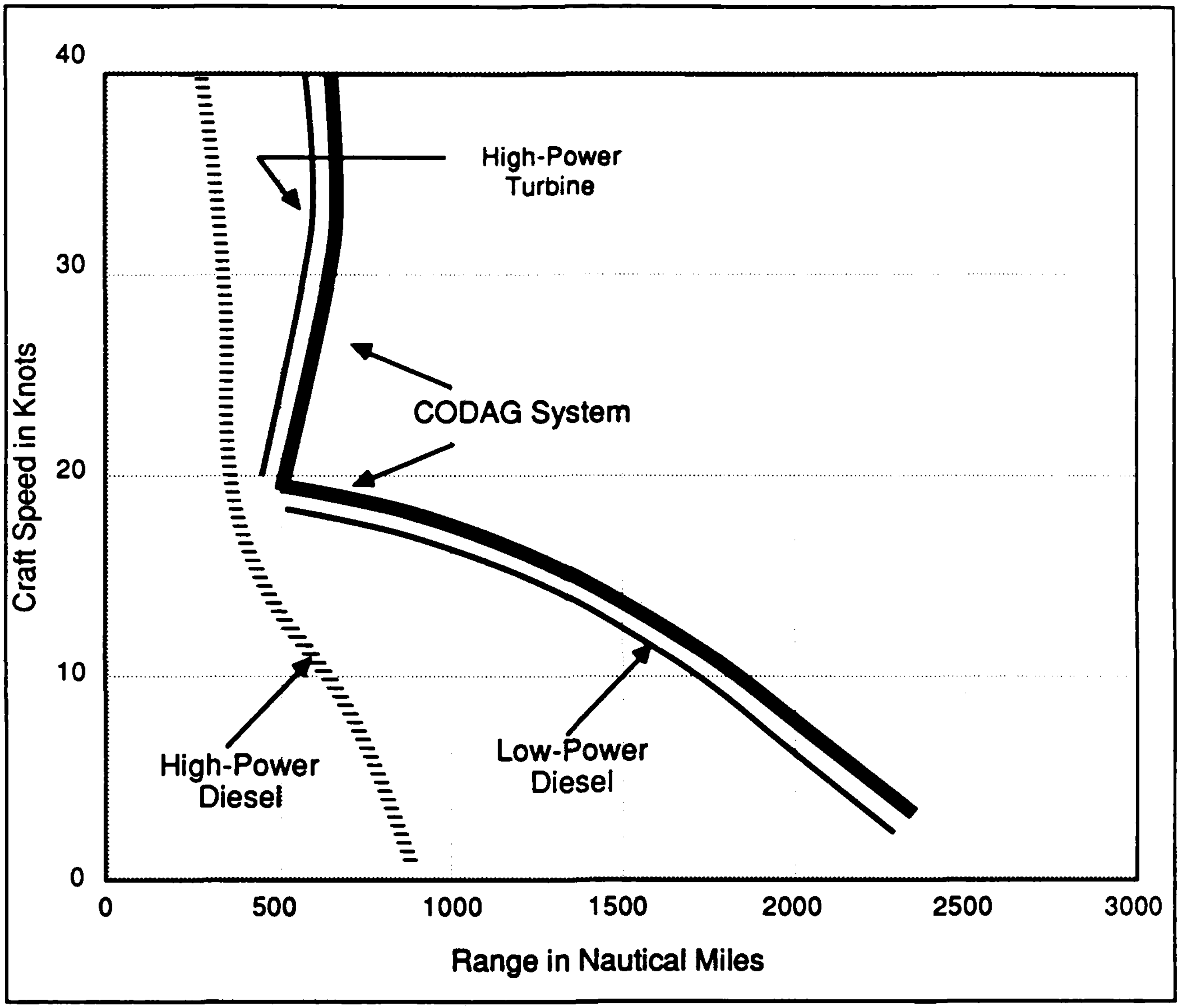
6.1.4 Combined Systems

In some shipboard applications, diesel engines, gas turbines, and steam turbines can be employed effectively in various combinations. The prime movers may be combined mechanically, thermodynamically, or both.

The gas turbine is a very flexible power plant and consequently figures in most possible combinations which include combined diesel and gas turbine plants (CODAG); combined steam and gas turbine (COSAG or COGAS); and combined gas turbines (COGAG). In these arrangements, gas turbines and other engines or gas turbines of two different sizes or types are combined in one plant to give optimum performance over a very wide range of power and space requirements, Figure 6.9. In addition, combinations of diesel or gas turbine engines (CODOG), or gas turbines (COGOG), where one engine is a diesel or a small gas turbine for use at low or cruising powers, and the other is a large gas turbine which operates alone at high powers, are also possibilities.

The combination of diesel and gas turbines, the most popular choice of naval vessels, has also been employed on Coast Guard cutters (378′ Hamilton Class). Danish Navy experience with a CODOG propulsion plant is given in Reference 6.16. Analysis of diesel and gas turbine combinations for use on U.S. Navy ships is provided in Reference 6.17. This study concluded that the benefits of the CODAG system exceeded any other for their propulsion requirements. The advantage of an "and" fit over an "or" is that both the cruising and boost engines can be used for the highest speed possible.

As previously mentioned, engine combinations may be achieved both mechanically and thermodynamically. Two possible future applications of this concept are detailed in References 6.18 and 6.19. The first paper presents the combination of a diesel engine and gas turbine. The gas turbine is driven by the exhaust gases from the diesel engine. Both are mechanically connected by a gear box. The advantages of this system would be a reduction in specific weight and in fuel consumption. Reference 6.19 discusses combinations of diesel engines and gas turbines with a steam cycle. Exhaust gasses from the combustion engines are used to produce steam which drives a steam turbine. Both the steam turbine and diesel or gas turbine are mechanically connected. The Navy is currently conducting full-scale tests and evaluation of this system, called RACER (Rankine Cycle Energy Recovery).



(From *Modern Ship Design*, T. Gilmer, 1972)

FIGURE 6.9 Engine Comparison — Range vs. Speed

6.1.5 Nuclear Energy

The most significant characteristic of nuclear power for marine propulsion is the compact nature of the energy source. The power produced by fission of one gram of Uranium is equivalent to about 900 tons of fuel oil. Nuclear power permits the utilization of very large power plants on board ships without the necessity for very large bunker storage or frequent refueling.

Economic studies indicate that the cost penalties associated with nuclear power are sufficiently high that further innovations will be required before nuclear power for ship propulsion will be able to economically compete with fossil-fueled power systems. Only when the advantages of high power and endurance override purely economical considerations is this power source on attractive option.

6.1.6 Fuel Cell Technology

Fuel cells, which directly convert the chemical energy of a fuel and oxidant into electrical energy, have been under development for about two decades. Marine applications for their use is discussed in Reference 6.20. Their advantages over existing systems could include higher efficiencies, lower cost, reduced emissions, low noise, and fewer maintenance problems. Despite these potential benefits, commercial marine applications are not in the near future.

6.1.7 Selection of a Prime Mover

Prior to the selection of the prime mover, the required plant output must be established. In addition, the selection criteria and their relative importance should also be decided. Typically, the selection criteria of importance in the early stages of the design spiral are:

- Initial and life-cycle costs
- Specific weight
- Specific fuel consumption

Other selection criteria normally considered are:

- Reliability
- Maintenance and Repair Requirements
- Maximum-to-continuous power ratio
- Fuel Requirements (including fuel treatment)
- Space requirements
- Interrelations with auxiliaries

The three top selection criteria will be discussed in more detail below as they relate to various types of prime movers.

The initial cost of the prime movers, discussed in Section 6.1.1, are shown as a function of shaft horsepower output in Figure 6.6. Although initial costs are not usually the highest costs of prime movers over their life cycle, they are considered to be of great importance. Operating costs which include fuel, lube oil and maintenance costs contribute to the majority of the life cycle cost. One of the factors in maintenance cost is the time between overhauls. Shown in Figure 10-5 of Reference 6.21 are the maintenance and lifetime characteristics of various engines.

The importance of the weight of a main propulsion plant varies depending upon the application. A parameter that is used to define its importance is the speed-displacement ratio. As this increases, so does the importance of the specific weight characteristics of the propulsion machinery. Thus, high performance vessels attach a higher importance to specific weight than do ordinary displacement vessels. Shown in Figure 6.7 are the specific weights for propulsion systems considered in this study. The general trend is for propulsion systems weights to continue declining.

Fuel costs make up a substantial portion of the propulsion system life cycle cost. Specific engine fuel cost may vary depending on availability and fuel price. The specific fuel consumption of the power plants is compared in Figure 6.8. The general trend is for an increase in efficiencies for power plants, therefore reducing fuel consumption. The types of fuel needed to run each particular plant must also be considered. In the case of gas turbines the cost of the fuel may be as much as 30% higher than that used by diesel or steam plants.

6.2 TRANSMISSION SYSTEMS

The majority of propulsion systems, with the exception of slow-speed diesels, require the use of a transmission system. The purposes of the transmission system are to reduce the engine shaft speed to a range that can be more efficiently used by the propeller and in some circumstances to provide a means for reversing the direction of propeller shaft rotation. Transmission systems widely used at present include reduction gears, electric drive, and semi-electric drive.

6.2.1 Reduction Gears

The development of propulsion gearing has been characterized by continuous improvements in reliability and service life. These improvements can be attributed to refinements in materials, manufacturing techniques, and equipment. Shown in Figure 6.10 are the typical gear arrangements available today. Reduction gears (a) and (b) are more common for diesel engine transmission. Turbine propulsion systems normally use the arrangement shown in (d). The other gears are utilized for more specialized requirements. Many other reduction gear arrangements are possible and have been used. Specialized gears are more common with combined propulsion systems.

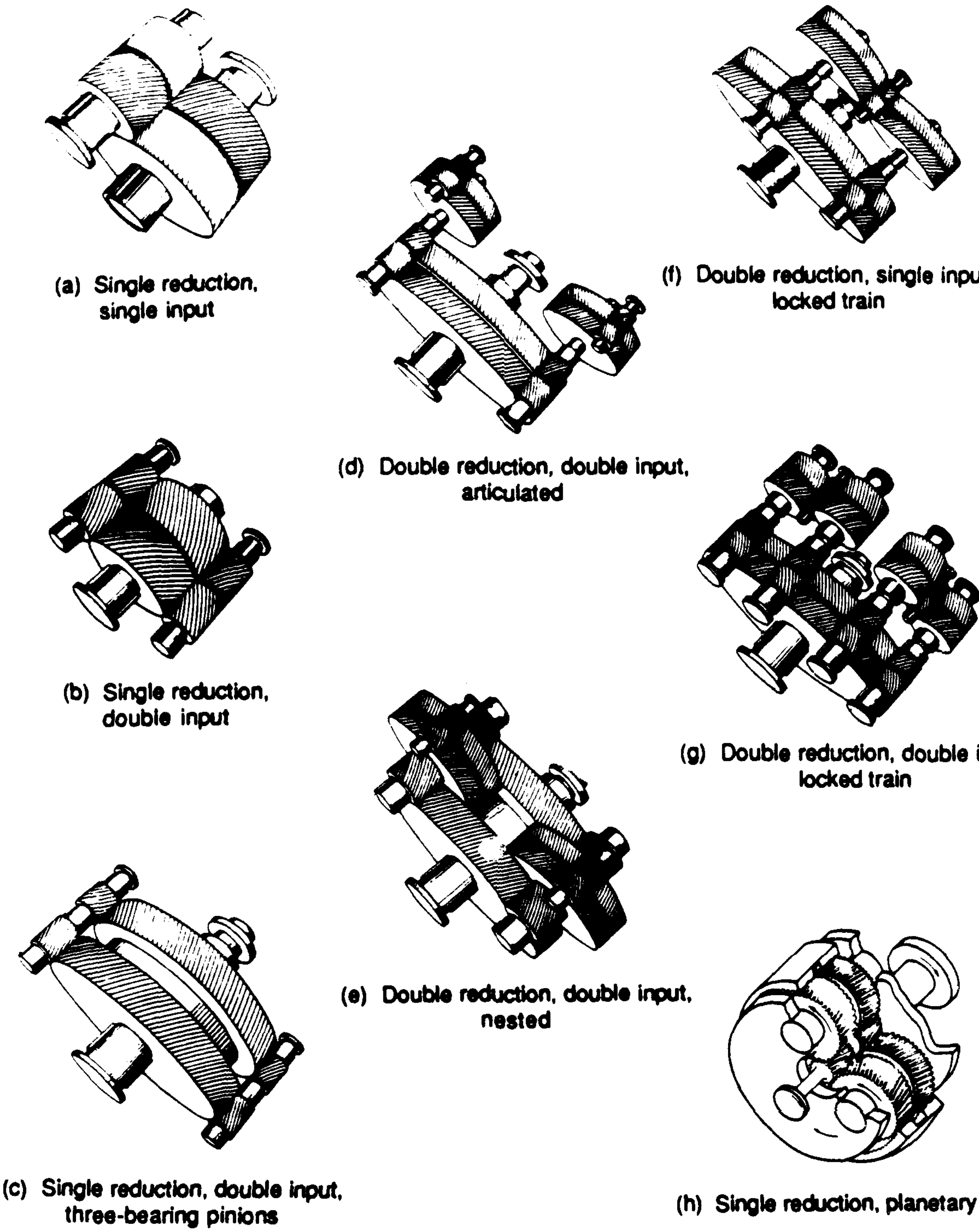
For selection of single-input/single-output gear box the following parameters are needed:

1. Engine power (BHP)
2. Engine speed (rev/min)
3. Propeller speed (rev/min)
4. Classification: American Bureau of Shipping (ABS), Lloyds Registry Service (LRS), etc.
5. Ice Class: 1, 2 or 3
6. Configuration: Vertically, horizontally, or diagonally offset or co-axial.

Manufacturers provide selection tables into which these parameters can be entered. Selection is on the basis of the power/speed ratio corrected by safety factors depending on the duty and classification, and the speed reduction required.

Twin-input single-output gears use the same parameters for selection, but it is necessary only to use the power/speed ratio of one engine if both have equal power. In addition, the center distance of the engine crankshafts must be given to allow sufficient maintenance working space between engines. The selection is made in a similar manner for the nearest standard gear with the requisite center distance.

In all configurations it is possible to provide an additional pinion and shaft to drive electric generators, pumps, etc. Power



(a) Single reduction, single input

(b) Single reduction, double input

(c) Single reduction, double input, three-bearing pinions

(d) Double reduction, double input, articulated

(e) Double reduction, double input, nested

(f) Double reduction, single input, locked train

(g) Double reduction, double input, locked train

(h) Single reduction, planetary

(From *Marine Engineering*, SNAME, 1971)

FIGURE 6.10 Gear Arrangements

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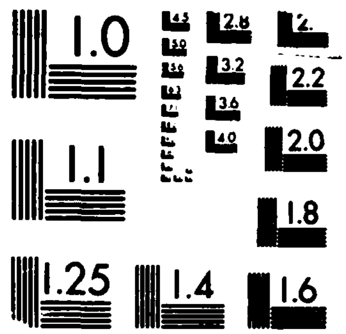


MICROCOPY RESOLUTION TEST CHART

NATIONAL BUREAU OF STANDARDS-1963-A

take-off may be primary, turning with the engine, or secondary, turning only when the propeller is turning.

Advancements in marine gearing technology have occurred in two areas, surface-hardening and clutches. The use of surface hardened gears instead of through-hardened gears can result in reductions of 50% in gear size, 30% in weight, and 35% in cost. Hydraulic clutches are now replacing pneumatic clutches due to their higher efficiency and reliability. Slip clutches are now available which allow shaft speeds lower than engine idling speeds, providing a greater degree of maneuverability. Further information on modern gearing concerning trends in materials, gear construction and configuration can be obtained from Jackson, Reference 6.22. Transmission weight reduction is further discussed in Reference 6.23.

6.2.2 Electric Drive

Electric drive propulsion systems have typically been selected when their advantages such as ease of control, flexibility of machinery arrangement, and "soft" coupling between propulsors and prime movers outweigh its higher initial cost, weight and space requirements, and lower transmission efficiency.

Initially, electric systems were entirely direct current (DC) systems. Later, as required propulsion powers exceeded the capabilities of the conventional DC machines, alternating current

(AC) machines were developed. An important difference was that the motor/generator speed ratio with a DC system was continuously variable while, with an AC system, it was fixed by the pole-ratio and synchronous operation. Therefore, controllable pitch propellers were required for AC systems, whereas DC systems could use fixed-pitch propellers. Recently, changes in technology have eliminated this restriction.

A large number of work boats today use SCR (Silicon Controlled Rectifier) electric propulsion systems. Power is supplied through AC generators driven by diesel engines, operating at their rated continuous speed, and delivered to a main switchboard. The AC current is used directly to power auxiliaries and rectified to DC current for use by the propulsion motors. The advantage of these systems is that each component is used at its most efficient range. Other features of the SCR system are presented in References 6.24 and 6.25. Descriptions of vessels outfitted with SCR propulsion are contained in References 6.26, 6.27 and 6.28.

A number of technological developments have made variable speed, total AC drives possible in sizes exceeding DC drives. These developments include solid-state adjustable frequency AC power supplies with power capabilities in the tens of megawatts, the widespread use of direct water cooling to improve the power density of motors and generators, and the introduction of microcomputer-based controls for large electric drive systems.

Further information can be found in References 6.29 and 6.30. A modern Finnish icebreaker utilizing this technology is described in Reference 6.31.

A revolution in the future of electric drive systems most likely will result from the development of superconductive machinery. Superconductors are materials which, when cooled below a critical temperature, exhibit zero electrical resistance and thus are able to sustain very high current densities. The material must also be able to tolerate a high magnetic field. The superconductor presently adopted for motors is niobium-titanium alloy. The superconducting DC motor is significantly lighter than conventional motors and offers high efficiency and performance characteristics particularly suited to high efficiency propellers, with no practical power limit. At the present, the main drawbacks are economics and the requirement for a helium refrigeration system. Superconducting machinery information and analysis is presented in References 6.32 through 6.36. Installation of a 400 HP superconducting electric drive system in a Navy test craft, described in Reference 6.37, proved it to be successful. A recent discovery of new materials for superconductors was reported in U.S. News and World Report, March 2, 1987. Made up of lanthanum, barium, copper and oxygen, these superconductors allow the use of liquid nitrogen which is much less expensive, less volatile and more plentiful than helium.

In addition, it is very likely that new materials will be discovered which act like superconductors at room temperature. Once this is achieved, the performance advantages of these new materials will outweigh their economic costs.

6.2.3 Semi-Electric Drive

A recent trend in work boat propulsion technology has been the introduction of Semi-Diesel-Electric Drive. A common installation is shown in Figure 6.11. Any combination of the four diesel engines can be used to drive either or both of the shafts. The advantages of this drive system are its flexibility, redundancy and economy. The main penalty is its higher initial cost. An analysis of semi-diesel-electric drives is contained in Reference 6.38.6.3.

PROPULSOR SYSTEMS

The purpose of the propulsor is to transfer the rotational power of the main engines into thrust horsepower through the fluid medium. Propellers generate their propulsive force from the lift and drag of their blades acting on the water. In addition to the usual screw propeller there are many other options available today which may offer improved performance or efficiency for particular requirements. Unconventional propeller arrangements include highly skewed, tandem, contrarotating and overlapping/interlocking twin screws. Water-jet, vertical axis,

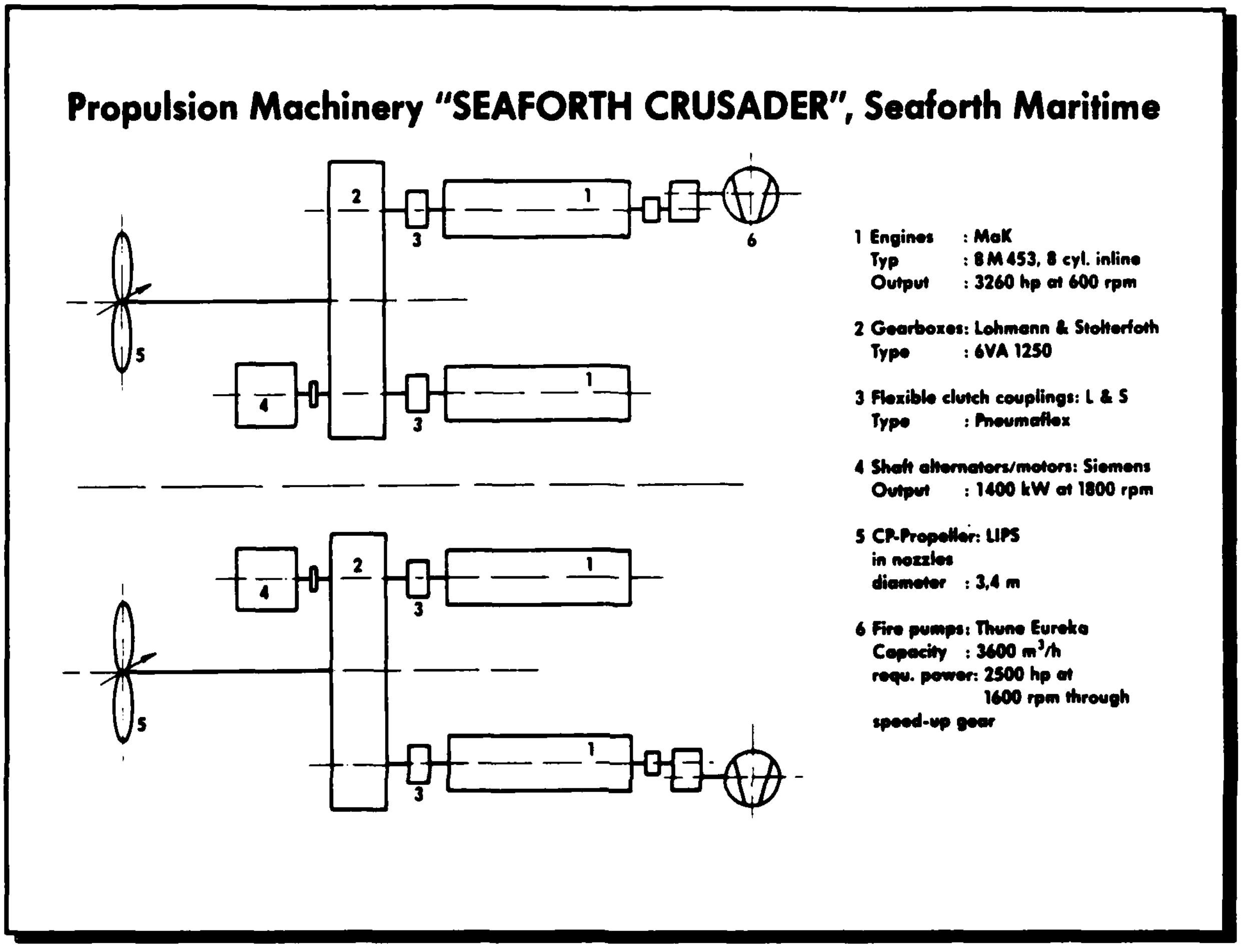


FIGURE 6.11 Semi-Diesel-Electric Drive

and nozzle arrangements offer further choices. A section of this report will also cover present thruster technology and its application. Selection of the propulsor will be discussed in the last section.

6.3.1 Screw Propellers

The most commonly used propulsor of vessels operating at speeds below 35 knots is the screw propeller. General terms and definitions, as well as propulsive theories of screw propellers, can be found in References 6.21, and 6.39 through 6.41. Screw propellers can be classified as fixed-pitch, variable-pitch and controllable-pitch. These propeller types will be discussed in this section along with the factors which affect their efficiency. Further trends and developments of screw propellers will be presented at the end of the section.

Conventional fixed-pitch propellers are the most commonly selected propellers due to their efficiency, cost and simplicity advantages over the other types. They are usually made from a single casting, although some are constructed with the blades cast separately from the hub and connected with bolts. The advantages of the separately cast component propeller are that damaged blades can more easily be replaced and that small adjustments in pitch can be made by turning the blades on the hub. Their disadvantages as compared with the solid propeller are higher first cost, greater weight and somewhat smaller effectiveness because of a larger hub.

Variable-pitch propellers change in pitch from the root to the tip. These are most commonly used on single screw ships where water approaches each section of the propeller at a different velocity due to wake effects. To better match this, the pitch at each radius is varied. It is usually lower at the inner root sections and higher toward the tip. Some propellers have increasing pitch from the root to the 0.5 R (50% radius point on the blade) and are constant from the 0.6 R to the tip. The root pitch may be 15%-20% lower than the 0.6R pitch. Variable-pitch may also be used to delay the onset or reduce the severity of tip-vortex cavitation.

There are different kinds of controllable pitch propellers on the market today. The majority are fully controllable, which means the blades can be adjusted to any position and can go from ahead to astern operation without reversing shaft rotation, thereby eliminating the need for a reversing gear. Others are partially controllable and may have predetermined settings of blade position. An example of this is the two-pitch propeller, which has one low-pitch setting for trawling and one high-pitch setting for free running. The two-pitch propeller still requires a reverse gear for astern operation.

The main advantage of controllable-pitch propellers are that they allow the engines to operate at their rated speeds which are most efficient. The amount of thrust delivered by the propeller can be varied by controlling the pitch angle of the blades. Other advantages, described in References 6.42 and 6.43, include easier reversing capabilities, smoother transition of power, and handier operation with auxiliaries. The disadvantages are the higher initial and maintenance costs. The approximate relative fixed price of fixed-blade propellers and controllable-pitch propellers are shown in Figure 6.8.7 of Reference 6.41.

Screw propellers have a maximum propeller efficiency of about 70%. The 30% losses can be split into the following three parts:

- -- Approximately 10% is due to loss in momentum.
- -- Approximately 10% loss due to friction.
- -- Approximately 10% loss due to the rotation in the propeller race.

The individual losses can be decreased in different ways, but then these "solutions" always cause one of the other losses, perhaps both of them, to increase or new losses to be added, reducing the gain. Typically, propellers operate at less than 70% efficiency. Maximizing fuel economy is dependent on approaching this level. Factors which affect efficiencies are detailed in Reference 6.44 and are summarized below:

-The propeller diameter should be as large as possible, with corresponding low revolutions; however, this may be limited by clearance and engine rpm considerations.

-The number of blades, in general, should be as low as practical for acceptable vibration levels. In some cases, however, particularly if the diameter is restricted, a high number of blades may be preferable.

-The blade arc should be as low as possible to suit the main operating condition at sea. Accepting higher cavitation risk at other less frequently employed conditions might be considered in the interest of efficiency.

-The propeller boss or hub diameter should be as small as possible.

-Clearances around the propeller should be as large as possible.

A major future trend in propellers will be an increase in their strength-to-weight ratios. The possible approaches for achieving this can be classified into one or more of the following categories:

-New or modified fabrication techniques such as the design of hollow blades and hubs.

-Modification of blade design to better utilize the strength properties of the material, such as the use of high strength alloys which allow for thinner blade sections.

-Use of lighter weight material of equal strength such as fiber-reinforced plastics (FRP).

These developments are discussed further in Reference 6.45.

A recent development in screw propellers is the highly skewed propeller. A propeller is termed skewed when its outline is asymmetrical with respect to a straight radial reference line in the plane of the propeller. Skew is usually introduced by successively displacing the blade sections away from the direction of rotation. Highly skewed blades were developed to handle very high horsepower input to propellers of limited size. The highly skewed design is able to utilize the power without cavitation and extreme vibration characteristic of conventional propellers under similar loading conditions. Additional information is available in Reference 6.46.

Many other methods for improving propeller efficiency, and therefore decreasing vessel operating costs, have been proposed. The use of pre-swirl vanes and reaction fans have been investigated for Coast Guard 41 foot utility boats in References 6.47 and 6.48. Proposed in Reference 6.49 are the use of tunnel wedges. The tunnel wedges accelerate the flow of water into the propeller causing an increase in pressure behind the propeller. Described in Reference 6.50 is the Additional Thrusting (A.T.) Fin developed by IHI of Japan. The A.T. fin, which was fitted on a 238,400 DWT oil tanker, converts the rotational flow energy behind a propeller into propulsive work providing a fuel savings of 4 to 5%. Reference 6.51 reports a High Efficiency Flow Adapted (HEFA) propeller being developed in Spain. It is expected that the propeller, which has end plates at the blade tips, will deliver a fuel savings of around 18% over the whole of a ship's power range. The blade tip plates are arranged to be tangential to the flow through the propeller disc, reducing drag on the top plates and providing for the theoretical load distribution of the propeller to be obtained.

An entire transmission system located outside the ship's hull is proposed in Reference 6.52. The hydraulic transmission contains an axial flow pump and turbine and uses sea water as the transmission fluid. The propulsive thrust is divided between the propeller, which is driven by the turbine, and fluid discharged from the turbine.

6.3.2 Multiple Screw Arrangements

When large propulsive power is required relative to the size of the ship in which it is installed, a twin or multiple screw arrangement may be considered. The typical twin screw arrangement involves the use of two transversely separated shafts which generally rotate outward when viewed from astern. A significant advantage of a twin screw vessel is its improved maneuvering characteristics. Twin screws also provides redundancy which may be a desired characteristic. These advantages are countered by the additional initial and life-cycle costs. Described in this section also are less typical arrangements of multiple screws.

Overlapping and interlocking propellers are twin-screw installations arranged to utilize the energy available in the wake behind the hull. The propellers are either arranged in the same longitudinal position, but with the blades in phase by means of an interlocking gear box to avoid interference problems, or arranged in different longitudinal positions such that the blades overlap. The transverse separation of the shafts is far less than in the typical twin screw arrangement. In addition to the wake gain, the advantages of lower revolutions and load sharing are also applicable.

Shown in Figure 4 of Reference 6.53, it is estimated that overlapping propellers could give a power savings of 16% with the optimum speed being 90 rev/min. Modifications to the hull afterbody are necessary as is the case with all multiple screw arrangements.

Interlocking propellers are discussed further in Section 6.8.2 of Reference 6.54. An efficiency improvement of 10% over a single screw arrangement on a large bulk carrier is reported.

Two or more propellers arranged on the same shaft are used to divide the increased loading factor when the diameter of a propeller is restricted. Propellers turning in the same direction are termed tandem, and in opposite directions, contrarotating.

In tandem, the rotational energy on the race from the forward propeller is augmented by the after one. From References 6.54 through 6.56, the following conclusions concerning tandem propellers can be drawn: a slight increase in propulsive efficiency of up to 4% over conventional propellers can be obtained; the pitch of the after blades should be higher than the forward ones; for equal diameters, tandems optimize at about 5/6 the RPM of conventionals; the forward propeller should develop slightly more than half the thrust; axial spacing should be small; and phasing should be such as to minimize interference-induced velocities.

Contra-rotating propellers work on coaxial, contrary-turning shafts so that the after propeller may regain the rotational energy from the forward one. The after propeller is of smaller diameter to fit the contracting race and has a pitch designed for proper power absorption. The advantages of these propellers are increased propulsive efficiency, improved vibration characteristics, and higher blade frequency. Disadvantages are the complicated gearing, coaxial shafting, and sealing problems. These advantages may be an important consideration on life cycle costs. Studies on contrarotating propellers are described in References 6.57, 6.58 and 6.59. A comparison of propulsive efficiencies with twin, single, tandem, and contrarotating propellers are given in Table 44 of Reference 6.38.

The vane wheel is a freely rotating device installed on the propeller shaft behind the propeller to provide additional thrust with no increase in power. The inner portion of the wheel (which is larger in overall diameter than its companion propeller) functions as a turbine, recovering energy from the otherwise wasted propeller slipstream to generate the extra thrust using a propeller element at the tip of each blade of the wheel. In Reference 6.60, model tests on conventional, contrarotating and vane wheel propellers were performed to compare their efficiencies. An improvement of over 9% over a conventional propeller was obtained. Installations of the Grim Vane Wheel on a 75,000 DWT cargo vessel showed a 10% improvement in efficiency. It was determined that the fuel savings would pay for the addition of the vane wheel on 300 operating days. Additional studies on the Grim-Wheel are provided in References 6.61 and 6.62. Recently, Grim-Wheels were installed on the twin screw Queen Elizabeth 2. As reported in Reference 6.63, problems have occurred during full-speed trials. Five of the seven vanes of the Grim-Wheel broke off. An investigation is now underway to find the cause of the failure.

6.3.3 Ducted Propellers

With conventional propellers, high thrust loadings yield low propulsive efficiencies. A method to change the thrust loading is to construct a duct or shroud around the propeller. There are two types of arrangements which fall into this category; water jets and nozzles.

There are several variants of nozzles (Reference 6.491), the most common being the Kort nozzle. The Kort nozzle arrangement consists of a propeller located in a nozzle of relatively short length (the length/diameter ratio of the nozzle is in the range of 0.5 to 0.8). Their principal advantage is found in tugs, where the towing force or bollard pull for a particular shaft horsepower may be increased by as much as 40% as compared with that given by an open propeller. At low towing speeds, a considerable advantage is still found. At high speeds and/or low speeds, the drag of the nozzle results in a loss of speed and efficiency. In ships other than tugs, the advantage can be extended to higher speeds by using thinner nozzles, with some loss of thrust at low speeds. Nozzle profiles need to be carefully designed, as discussed in Reference 6.64, in order to avoid adverse pressure distributions, particularly pressure or suction peaks at the nose (entry) on either the inner or outer surface. Pressure peaks may lead to cavitation.

The Tip Vortex Free Propeller (TVF) has been used on several cargo carriers and reported in References 6.65 and 6.66 to provide substantial fuel savings. By adding closing plates to the ends of the propeller blades, the tips can be heavily loaded without the flow jumping from the front to the back of the blade, thereby avoiding the formation of tip vortices. Unlike the HEFA propeller (see Section 6.3.2.5), the TVF propeller is installed inside a duct. In addition to the economic advantages provided by this arrangement, service reports suggested other bonuses from the propeller. Rudder response, and therefore coursekeeping, was improved after installation of the TVF propeller. Vibration and noise were also reduced.

A novel type of ducted propeller is presented in Reference 6.67. This arrangement, called the Mitsui Integrated Duct Propeller (MIDP), differs from conventional ducts in a number of ways: it is mounted entirely forward of the propeller and overlaps the stern hull, intrinsically it has no risk of cavitation erosion on the duct so no countermeasures are necessary, and its retrofitting allows utilization of the existing propeller. A fuel savings of 5% was achieved on a 250,000 DWT tanker fitted with the device.

A highly skewed propeller in a nozzle was selected for the Swedish hydrographic survey vessel "JOHAN NORDENANKAR." The highly skewed propeller was chosen to reduce the potential for vibration problems. The ship is described in Reference 6.68.

In the water or pump jet arrangement the propeller is placed in a long duct with guide vanes either forward, aft, or at both positions relative to the propeller. The motive force is produced by drawing in water at one end and discharging it at the other at a higher velocity. A rotatable deflector is placed at the end of the duct which allows thrust to be directed in any direction providing good maneuverability. The water jet is often considered where propeller noise is important. Due to the resistance of the duct and guide vanes, the overall efficiency of the system is strongly dependent on the particular arrangement. In general, water jets become more competitive with other propulsors as ship speed increases. A disadvantage of the system is the loss of internal volume due to ducting. A concise and thorough report on water jets is given as Reference 6.69. An analysis of a water jet propulsor performed by the David Taylor Naval Ship R&D Center is given in Reference 6.70.

6.3.4 Vertical Axis Propellers

The vertical axis propeller has four or more blades of airfoil section connected perpendicularly to a disk with its axis of rotation approximately vertical. The disk is geared to the propeller drive shaft and, as it rotates, the blades, by means of cam action, adjust to provide thrust in one direction. The position of the cam with respect to the disk can be varied so that reverse or side thrust may be produced. This system has maneuverability characteristics superior to all other propulsors; however, its efficiency is lower and its expense higher than the screw propeller.

There are two types of vertical axis propellers dominating the market. Kirsten-Boeing has blades interlocked by gears such that the blades make a half revolution about their axis for each revolution of the disk. The orbit will be a cycloid. The second type, Voith-Schneider, have blades which complete a full revolution for each revolution of the disk. Their orbit is an epicycloid.

Vertical axis propeller systems are used mainly on tugs, oceanographic research vessels, short haul passenger ferries, double-ended car ferries, floating cranes, and recently the large Italian dynamically positioned drill ship "SAIPEM DUE." Descriptions of their installations are given in References 6.71 through 6.76.

6.3.5 Thrusters

The thruster is a device with the purpose of assisting vessels in maneuvering and docking as well as for dynamic positioning of special purpose vessels. Therefore, they must be optimized for low-speed operation. There are many types of thrusters which use various methods of moving water but all have some common elements and problems. Among these, cavitation is one of the limiting factors. The two categories in which thrusters can be classified are lateral and azimuthing.

The design and selection of thrusters can be as complex as that of the main propulsor. Typically, they are located as far forward or aft as possible to maximize the turning moment. Determination of required thrust may be done analytically, where great accuracy of positioning is essential, or empirically, where information is obtained from past experience. If multiple thrusters are used, spacing between them must be adequate to reduce interactions. Additional information on thruster design and selection is contained in References 6.76 through 6.79. Additional factors which must be considered when ice operations are anticipated are discussed in Reference 6.80.

The simplest form of lateral thruster is the tunnel thruster containing either a fixed or controllable-pitch propeller. Factors which influence selection of a propeller and prime mover are given in References 6.77 and 6.78. As the demand for larger and more efficient thrusters has increased, so have the variety of available options. Reference 6.81 describes other lateral thrusters such as the Schottel anti-suction tunnel (AST), the Orenstein & Koppel K-bow jet, the Alsthom-Allantique Y-thruster, and the Omnithruster T-thruster. Although these new thrusters claim to possess advantages over the ordinary tunnel thruster, they are seldom used due to their complexity and higher cost. Ulstein Propell A/S has recently developed a tunnel thruster designed to reduce noise and vibration. This is achieved through the insulation and suspension of the tunnel and use of a forward skew-bladed propeller as reported in Reference 6.82.

Azimuthing thrusters are advantageous in that they can direct thrust over 360 degrees. Since they are normally fitted on towing vessels, most are fitted with nozzles. The thrusters may be deck or hull-mounted and retractable or non-retractable. The various types and manufacturers of azimuthing thrusters are presented in References 6.83 through 6.87. Retractable units have the advantage of allowing the vessel to operate in shallow water. They are most suitable for vessels with dynamic positioning. Examples and descriptions of vessels operating with azimuthing thrusters are given in References 6.88 through 6.93, and 6.65.

6.3.6 Selection of the Propulsors

Once the ship speed requirements and resistance have been tentatively established, it is necessary to select the type of propulsor. For particular vessel requirements, there are certain propulsors which operate more efficiently. In Figure 6.12, taken from Reference 6.41 (page 218), the optimum open-water efficiencies of some propulsor types are expressed in terms of the Taylor power coefficient, b_q, which is defined as:

$$b_q = NP^{0.5}/V_q^{2.5}$$

where

N = propeller rpm

P = power, hp

V_q = speed of advance, knots

The selection of the propulsor may not be a simple process, particularly in marginal cases, because in order to establish the type of propulsor it may be necessary to at least tacitly select the type of main propulsion machinery. Similarly, the selection of the number of propellers may be a multifaceted problem. Table 6.3 from Reference 6.94 presents a comparison of the features of available propulsors.

A trade-off study must be made between the propeller rpm which is required from a maximum propulsive efficiency viewpoint and propeller constraints imposed by the prime mover/transmission size, weight and cost considerations. The propeller rpm which is necessary to achieve a maximum propulsive efficiency is frequently considerably lower than that which is feasible from the viewpoint of the prime mover/transmission (due to the greater torque and therefore, machinery size associated with lower propeller speeds). Furthermore, attainment of the maximum propulsive efficiency does not necessarily constitute the most cost effective system. Propeller characteristics are in general such that the propeller can be designed to operate at an rpm somewhat greater than that corresponding to the maximum efficiency without incurring a serious efficiency penalty. While no significant penalty in efficiency is incurred with propeller rpms slightly greater than that for peak efficiency, significant savings on the first costs, size and weight of the prime mover/transmission can be realized due to the lower torque rating (with the power remaining the same). The most cost-effective

Symbols:

T - thrust of propeller
C_T - thrust coefficient
σ - cavitation number
V - speed of ship
N - propeller rpm

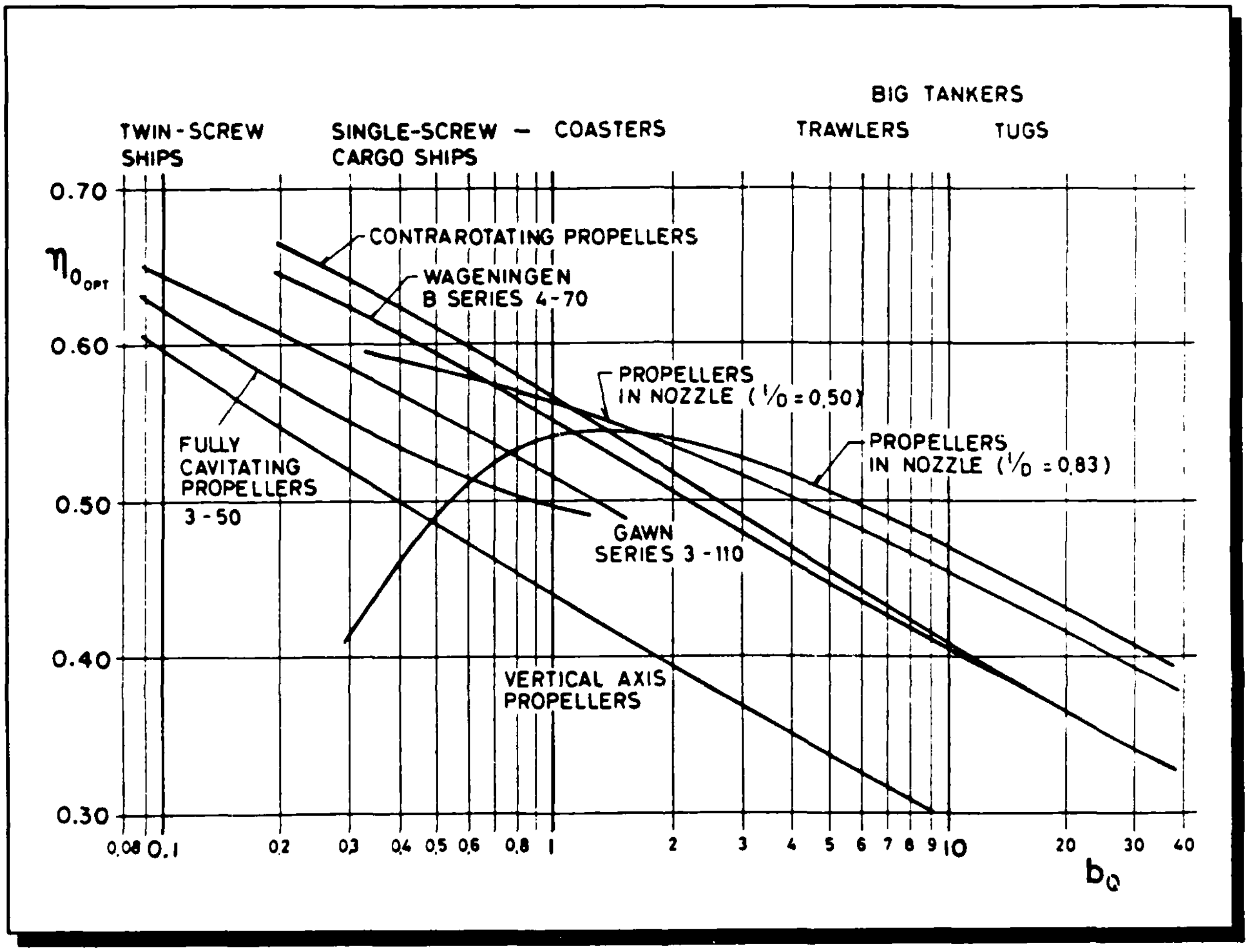
Symbols

t - thrust deduction fraction
η_T - total propeller efficiency
η_H - hull efficiency
η_D - propulsive efficiency

TABLE 6.3
COMPARATIVE FEATURES OF UNCONVENTIONAL MARINE PROPULSORS

TYPE	OPEN				DUCTED		VERTICAL AXIS
	Controllable Pitch	Fully Cavitating	Contra-Rotating	Tandem	Nozzle	Pumpjet	
Characteristic Features	Pitch Variable in Action	Complete Cavity over Blade Back	Co-Axial Contre-Turning Shafts	Two screws on single Shaft	Accelerating Duct	Diffusing Duct	Blade Pitch Variation With Rotation
Principal Purpose	High Efficiency at Varying Loading	Good Performance at High Speeds	Reduced Screw Loading. Regain Rotational Energy	Reduced Loading on each screw	Increase η By part T on duct, so Reducing T on Screw	Reduce C_T Raise Cavitation Inception Speed	Varying Direction of Thrust
Relative to Equivalent Normal Propeller η_T	Very slight reduction	Almost equal at very low σ and high V	Up at low C_T, Down at high C_T	Slight reduction	Gain, increasing with C_T		As much as 30% reduction
η_H	Unchanged	Perhaps better ($t \longrightarrow o$)	Down at low C_T, up at high C_T	Slight increase	Reduced		
η_D	Almost unchanged	Can be better at low σ, high V.	Up at most C_T, best at low C_T ($\sim$10%)	Little Change	Increase less than in η_T, best at high C_T ($\sim$10%)	Probably reduced	Much reduced
Advantages	Better matching of engine & hull	High T at high V & N Less blade erosion	Torque balance. Diameter reduction.	Higher thrust & power on a single shaft	Screw size & weight Inflow more uniform	Noise reduction	Steering & stopping at constant N & at low V
Disadvantages	Higher capital costs Complexity? Reliability?	Strength difficulties Off-design performance	Much higher cost Mechanical complexity	Higher cost	Duct weight Increased cost. Tip clearance	Duct weight and drag	Low efficiency Higher cost
Likely Applications	Ships with varying operating conditions (Tugs, trawlers ferries)	High speed craft Ventilated sections for lower speed ships?	Torpedoes High-Speed cargo liners?		Tugs Trawlers Large tankers	Naval Vessels	Tugs, ferries vessels for crowded waters

From: "Prospects for Unconventional Marine Propulsion Devices", A. Silverleaf, Paper presented at the Seventh Symposium on Naval Hydrodynamics, Rome, August 1968.



(From *Resistance and Propulsion of Ships*, Wiley & Sons, 1983)

FIGURE 6.12 Comparison of Optimum Efficiency Values for Different Types of Propulsors propeller rpm is selected by conducting a trade-off study which balances the propulsive efficiency against the size, weight and cost of the prime mover/transmission.

6.4 PROPULSION SYSTEMS SUMMARY

The purpose of this chapter was to document the present options available for propulsion systems. Trends and possible future developments were also presented to identify technological changes which occur in certain areas. The most dramatic example of this being the rapid progress in the development of superconductors which can operate at room temperature (see Section 6.2.2). Comparisons were made between systems in order to highlight their major advantages and disadvantages.

The design of a propulsion system, like many other general design projects, consists of combining a number of units and elements into a functioning system which yields a desired performance. This entails selecting components, adjusting each to the constraints imposed by all others, and arranging them so as to achieve the required system performance, a satisfactory configuration, and an acceptable life cycle cost.

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7.0 BUOY WEIGHT HANDLING EQUIPMENT

7.1 SCOPE

In this section weight handling equipment used for the launch and retrieval of buoys is discussed. Equipment requirements including lifting capacity, outreach, and speeds as well as operational considerations such as manpower, maintenance, and reliability are presented. Deck arrangements for buoy handling, maintenance, and storage are reviewed.

7.2 PRESENT BUOY TENDER DESIGNS

Traditionally, U.S. buoy tenders of the WLM and WLB classes are designed with a forward buoy deck arrangement. A typical arrangement is shown in Figure 7.1. Although buoys may be handled either on the port or starboard sides through cutouts in the gunwale (buoy ports), they are more commonly handled on the port side. Both WLM and WLB class tenders utilize swinging derricks (defined in a following section) operated either from consoles on the port and starboard bridge wings, or from a boom shack above and behind the base of the boom. The derrick's boom is generally stepped aft on the buoy deck and faces forward. Present storage capacity on deck for the larger WLB (180') tender is 50 tons. Maximum lifting capacity of the WLB class is approximately 20 tons (ref. 7.1). A survey of weight handling systems on both U.S. and foreign buoy tenders is presented in Table 7.1. Included in this table are deck arrangements and

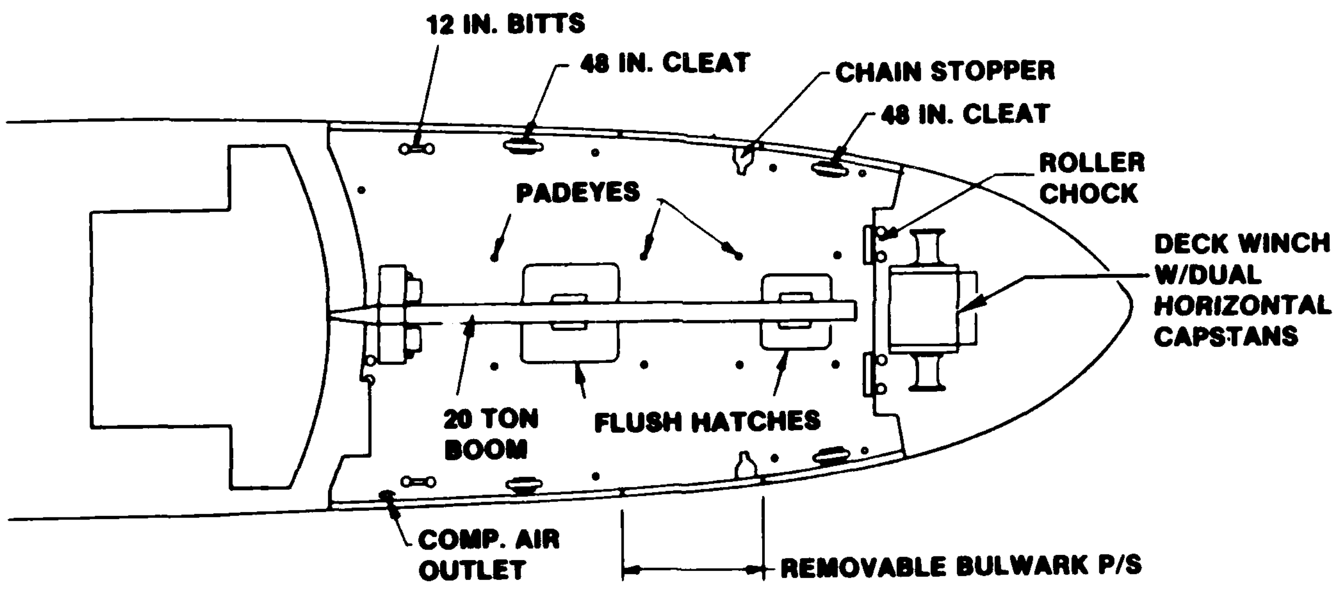

FIGURE 7.1 Typical U.S. Buoy Tender Deck Layout

TABLE 7.1

BUOY HANDLING SYSTEMS OF EXISTING TENDERS

Country	Vessel Name	Buoy Lifting System
UK	STENA SEAWELL	The STENA SEAWELL has the work deck aft of the bridge house with the turn cranes mounted in its center. Twin 65 metric ton Maritime Hydraulics heavy lift cranes with computer-controlled heave-compensation system are fitted. Lifting operations can continue in 2 m wave heights (sea state 5). Accelerometers at the job tips are linked to the control cab computer which also notes inputs from the ship's dynamic positioning system, drive control station and sea condition measurements. (Reference from Marine Engineers Log, 1986)
UK	PATRICIA	Single 20-ton Speedcrane manufactured by John Hastie on forward buoy deck. Two lifting hooks are provided on wires connected to a topping drum. Each wire is capable of lifting a 15-ton load under dynamic conditions when working in a seaway. The derrick has a lifting speed of at least 25 meters/min at full load and 50 meters/min with a light hook load. The Speedcrane derrick has a maximum radius of operation of about 17 m and can work around an angle of 140 degrees, 70 degrees each side of centerline. It is designed to operate with the vessel heeled to 5 degrees and rolling in a seaway to a maximum angle of 10 degrees either side of the vertical, with an eccentric load of 10 degrees on the head of the job in any direction. Switches, housed in the trunnion, limit excessive slewing and topping, and two control positions, having watertight consoles, are located at the forward end of the working deck together with a wandering lead facility for use on the after end.

TABLE 7.1 (cont'd)

Country	Vessel Name	Buoy Lifting System
UK	PATRICIA (cont'd)	Two deck capstans, supplied by Norwinch, are located on the main deck; port and starboard of the main hatch, and two more are situated in the hold. The upper capstans are used for mooring and steadying buoys during lifting operations. These capstans are each capable of pull of 5 tons at 20 m/min.
		The Speedcrane derrick is served by three electric winches from Clarke Chapman. These control topping, slewing and lifting operations. Two of these winches are located on the main deck sufficiently clear of all other equipment and acetylene stowage, while the topping winch is positioned on the after end of the forecastle deck. Control is from one of two consoles on the forecastle deck or from a wandering lead. Reference: Motor Ship (1982).
UK	WILTON	A-frame gantry crane manufactured by Welin Davit & Eng. hydraul., 30 t S.W.L. at 3 m outreach. 2 deck winches by Welin Davit & Eng. 1 buoy winch, 22 ton by Vickers. 8 t S.W.L. HIAB 180 hydraulic deck crane. The hydraulically operated A-frame can handle 14-ton buoys. Buoy deck storage is limited to 2 such buoys. Vessel is also equipped with a 22+ deck winch which can be rigged to lift 100 tons over a bow roller. Buoy deck area is forward of bridge. Reference (Ship and Boat International, 1983).

TABLE 7.1 (cont'd)

Country	Vessel Name	Buoy Lifting System
Japan	MYOJO	Thomson derrick with two cargo hooks. Capacity of 15 tons. Four winches in winch room consist of: 2 hoisting winches: 4 tons at 22 m/min; A.C. 18 KVA 1 topping winch: 6.5 tons at 7.2 m/min; A.C. 10 KVA 1 slewing winch: 4 tons/2 tons at 6/12 m/min; A.C. 5.5 KVA. Reference: Buoy Tender MYOJO (1968).
Finland	SEILI	One hydraulic crane of 6 tons S.W.L. at maximum reach of 16 m. Buoy deck is aft of bridge. Reference: Rauma-Repola Shipbuilding Division, undated.
U.K.	THV MERMAID	Forward buoy deck with 20-ton S.W.L. Speedcrane by John Hastie. 14-ton buoy chain capstans by Clarke Chapman.
Menas	RELUME (U.K. Design)	Forward buoy deck with vertical buoy storage. 20-ton Stott and Pitt electro-hydraulic crane mounted on forecastle and capable of 360 degree rotation. Heaviest buoy weight is 16 tons. Reference: The Motor Ship. 1979.
Netherlands	BREEVEERTIEN	Electric revolving crane by Demag Kampnagel with wave-following tensioning device. Forward buoy deck capable of storing 6X10 m3 buoys and chain. Crane is capable of 15 tons at 9 meters. Able to handle buoys in SS6. Reference: IALA Bulletin 1976/x.

TABLE 7.1 (cont'd)

Country	Vessel Name	Buoy Lifting System
Sweden	BALTICA	Hydrolift electro-hydraulic crane with rigid luffing jib. Maximum lifting capacity is 12 tons at a 9 m outreach. Crane may be controlled from heated cab or electric remote control. Full load may be handled at up to 10 degrees list. Crane is forward on buoy deck. Buoy deck is forward of bridge. Tidan electro-hydraulic buoy winch of 7-ton capacity is also included. Reference: IALA Bulletin 1984/1.

arrangements and winches based upon available data.

7.3 CRITERIA FOR SELECTION OF LIFTING EQUIPMENT

Detailed criteria for lifting equipment must be developed from an examination of the present lifting requirements taking into consideration projected changes to the Short Range Aids to Navigation (SRA) system. These criteria will normally be selected in the course of the new buoy tender design. Actual numerical values of loads, speeds, dimensions, etc. will be arrived at only after a compromise between these criteria and cost has been made.

The scope of this report has been limited to a survey of existing handling equipment attributes where such data are readily available from manufacturers. Because of the custom nature of buoy lifting system design, many manufacturers were reluctant to provide detailed specifications. As a result, what follows is a descriptive list of design criteria:

- Safe Working Load (SWL). This is the load, usually in short tons, which the lifting (or handling) gear can accommodate with an adequate margin of safety. The range of SWL is normally from 0 to 20 tons, although this value may be required to be 35 tons to fulfill a multi-mission requirement.

-Maximum outreach from the side of the vessel. This value depends upon the boom length and angle, vessel beam, and vessel roll angle, among other factors. Normally a minimum outreach of 10 to 25 feet is required.

-Buoy lifting speed. High buoy lifting speeds are necessary to work buoys in rough water. The buoy must be lifted rapidly from the water to prevent shock loads from occurring while the buoy is partially supported by its buoyancy and the hoisting wire. Typical lifting speeds are from 50-150 ft./min.

-Additional criteria for the handling system relate to the vessel motions in a seaway. One method for the specification of lifting gear is to require that it be capable of performing a lift of a given load during a specified sea state. Currently, WLB class buoy tenders are capable of lifting 20 tons SWL. As sea state increases, this figure will decrease depending upon the judgment of the tender's commanding officer.

-Power requirements for the equipment need be compatible with ship's power.

-Coverage. The lifting gear must be capable of reaching all areas of the working deck.

-Vertical lift. The handling system must be able to lift the tallest buoy high enough to set on the working deck.

-Ability to lift chain. It is desirable to be able to lift as much chain as possible in one hoist of the boom (or crane). In this case a telescoping crane might be an advantage.

7.4 CLASSES OF WEIGHT HANDLING EQUIPMENT

Classes of weight handling equipment include derricks, cranes, gantry cranes, and A-frames.

7.4.1 Derricks

Derricks comprise a class of weight handling equipment which is characterized by a rigid boom of fixed length attached to the ship structure by wire rope in one or more locations. A derrick is a non self-contained system in that all drive winches, controls, and guys are external to the derrick boom. Derricks are classified as either fixed or swinging.

Fixed derricks are the type of lifting gear found almost exclusively on older break bulk cargo ships. Commonly, two booms work in tandem, one being positioned over the cargo hold and the other over the deck. The booms are positioned by several lines attached at various points to the ships structure. The exact nomenclature and positioning of these lines varies with the specific installation. Once positioned, these booms are fixed in place for the remainder of the cargo operation. The cargo is lifted jointly by two lines, one from each boom. This system is effective in reducing swinging of the load (pendulation) due to its multiple point lifting, but it cannot precisely spot cargo. Furthermore, the possibility exists that excessive loads may inadvertently be applied to one of the booms when lifting a single load with both booms. An example of a fixed derrick is shown in Figure 7.2.

As described by Beattie, et al. (ref. 7.2), swinging derricks are the primary lifting gear found on the WLM/WLB classes of buoy tenders. As with the fixed derrick, the swinging derrick makes use of a boom of fixed length which is positioned by several lines attached at various points to the ship's structure. Again, as with the fixed derrick, the location of attachment and nomenclature of these lines varies with the particular type of swinging derrick. Unlike the fixed derricks, the swinging derrick uses only one boom which is continuously repositioned throughout the lifting operation. Generally, movement of the boom in the horizontal and vertical planes is independent of each other. There will usually exist a single topping lift for lifting and lowering the boom, while side to side motion is controlled by vangs. Lifting of loads may be from either a main hook or a whip attached near the boom head.

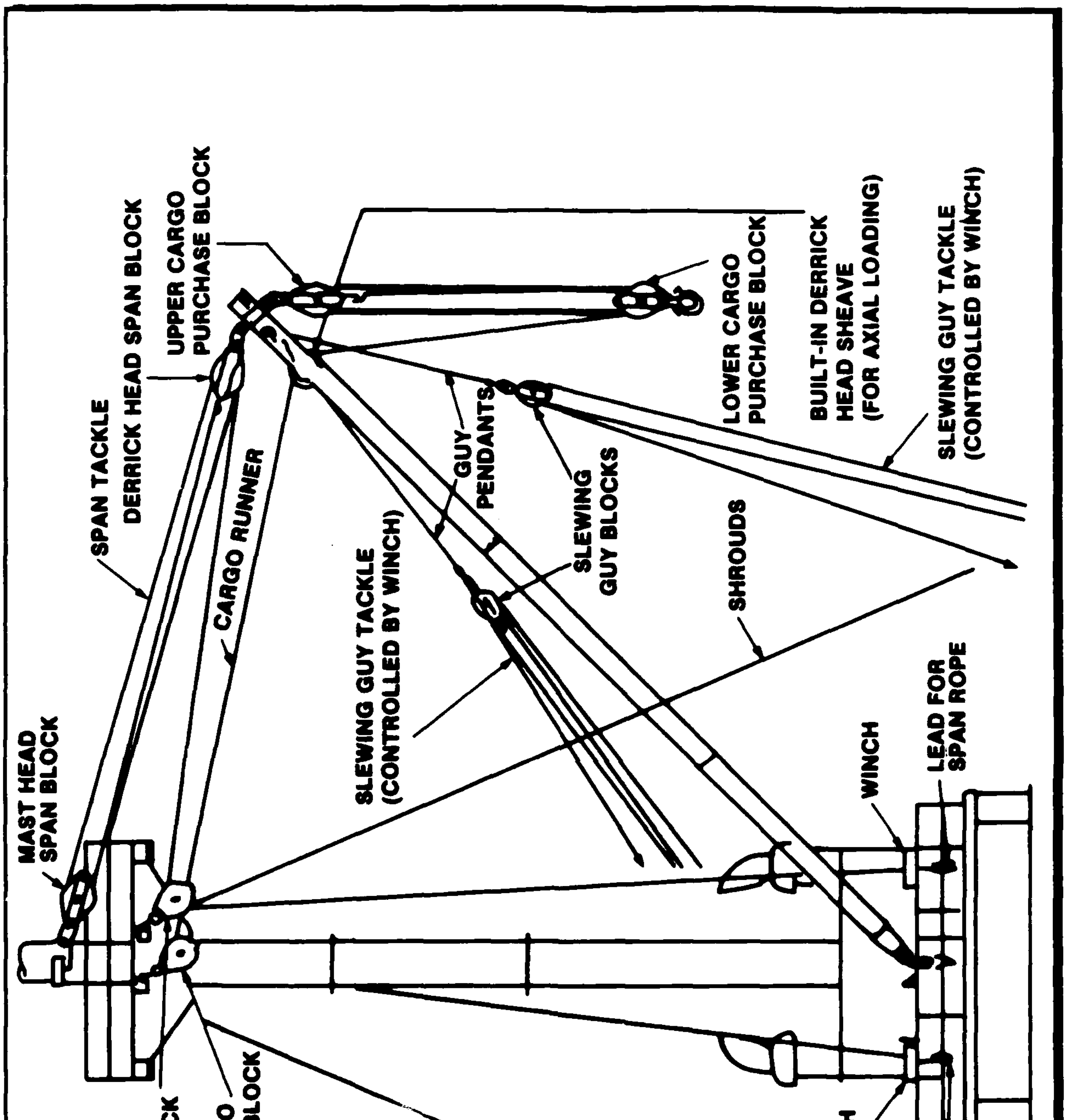

FIGURE 7.2 Heavy Lift Derrick (Beattie, et al., 1979)

A type of swinging derrick which is used by several countries is the Speedcrane Derrick manufactured by John Hastie, Ltd. of Glasgow, Scotland and shown in Figure 7.3. Limited information is available on the Speedcrane design. However, one author of this report (K. Bitting) was able to examine a Speedcrane in a recent visit to the United Kingdom. The Speedcrane has recently been installed on the Trinity House replacement buoy tender Patricia (ref. 7.3) and Canadian Coast Guard Type 1100 (272') Major Navaids Tender/Light Icebreakers. In the Speedcrane design, light loads are lifted by a single wire at maximum winch speeds. Heavier loads can then be lifted using a multi-purchase system. The switch from light to heavy loads may be accomplished in fewer than four minutes without re-rigging the hoisting ropes as discussed in Reference 7.2. The Speedcrane system generally has one cargo winch, one topping winch, and one slewing winch. The vertical and horizontal motions of the boom are integrated into a single joystick control system. Each Speedcrane design is a custom installation on the buoy tender. In the Trinity House design, the boom is fitted with a forked head to accommodate high-focal plane buoys used by the United Kingdom.

Several advantages of derricks and swinging derricks have been listed in Reference 7.3 and these include:

- The construction of the derrick boom is simple and therefore inexpensive.

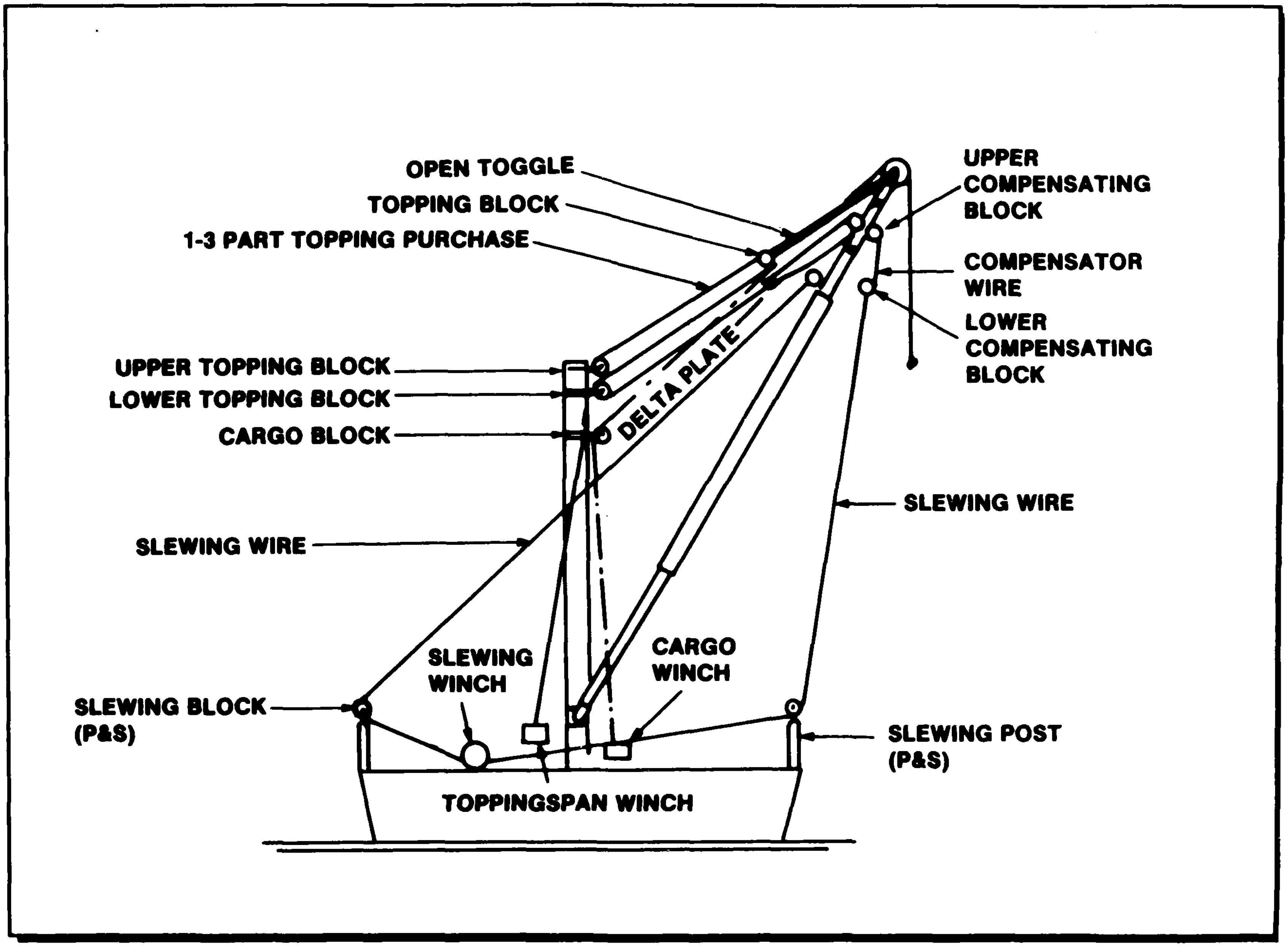

FIGURE 7.3 Speedcrane Derrick (Beattie, et al., 1979)

-Because of its simplicity, maintenance of the derrick requires little specialized training.

-Access for maintenance is good because of the exposed nature of its components.

-The winches used for hoisting, slewing, topping, etc. may be selected from several manufacturers, and there is flexibility in their powering, i.e., hydraulic or electric.

-There is a flexibility in the placement of winches on (or below) working deck.

Disadvantages of swinging derricks as identified in an article in Motor Ship (ref.7.4) include:

-High manpower requirement for operation.

-Limited operating arc, usually of 140 degrees.

-Lack of boom dynamic stability when working off the centerline of the ship or when fully luffed (raised).

-Standing rigging or guys may be fouled by the load.

-Luffing, topping, and slewing operations are generally slow because of their individual control. (The Speedcrane swinging derrick has supposedly overcome this disadvantage through an integrated control system). Unfortunately, the integration of all controls results in added complexity to the system and may result in difficulty of maintenance.

7.4.2 Cranes

Deck cranes represent the most recent method for moving cargo on ship. This, however, does not mean that deck cranes are replacing derricks, only that the crane technology is more recent. The biggest difference between cranes and derricks is that cranes are self-contained. They have no external guys or control lines which must be attached to the structure of the ship. This feature allows for 360 degree rotation and an increase in the deck area that a crane can reach. Cranes can be subdivided into fixed boom, telescoping boom, and articulated (knuckle) boom cranes as shown in Figures 7.4, 7.5, 7.6. Each has certain advantages and limitations in terms of performance and complexity. In general, the limitations of cranes include:

-Access for maintenance is difficult due to the self-contained nature of cranes.

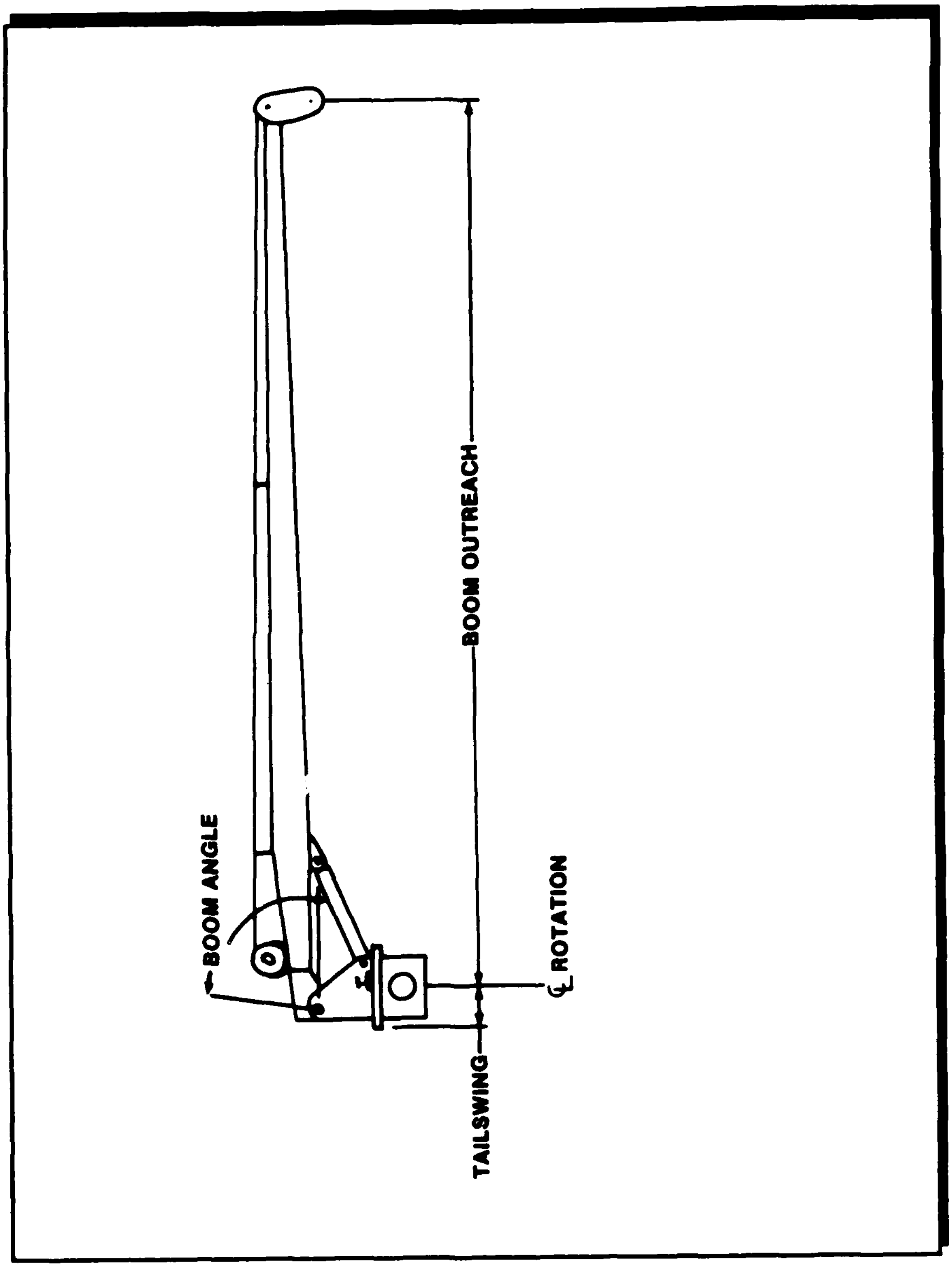

FIGURE 7.4 Fixed Jib Crane (Appleton Marine)

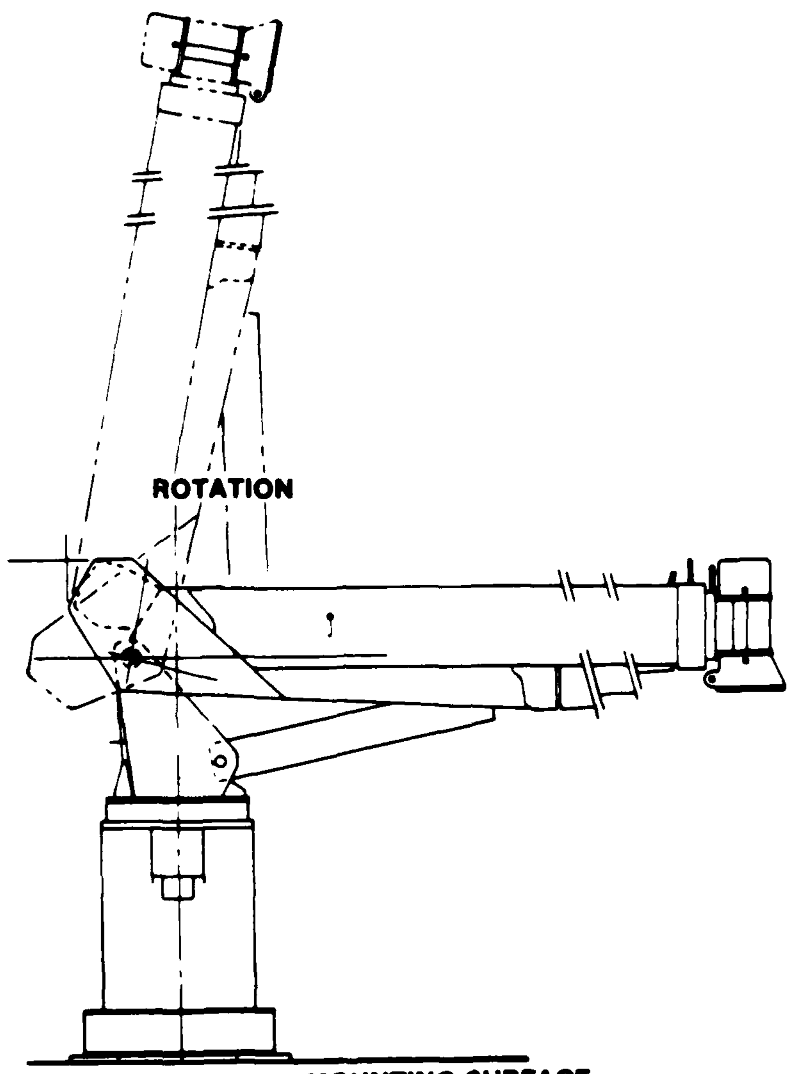

FIGURE 7.5 Telescoping Crane (National)

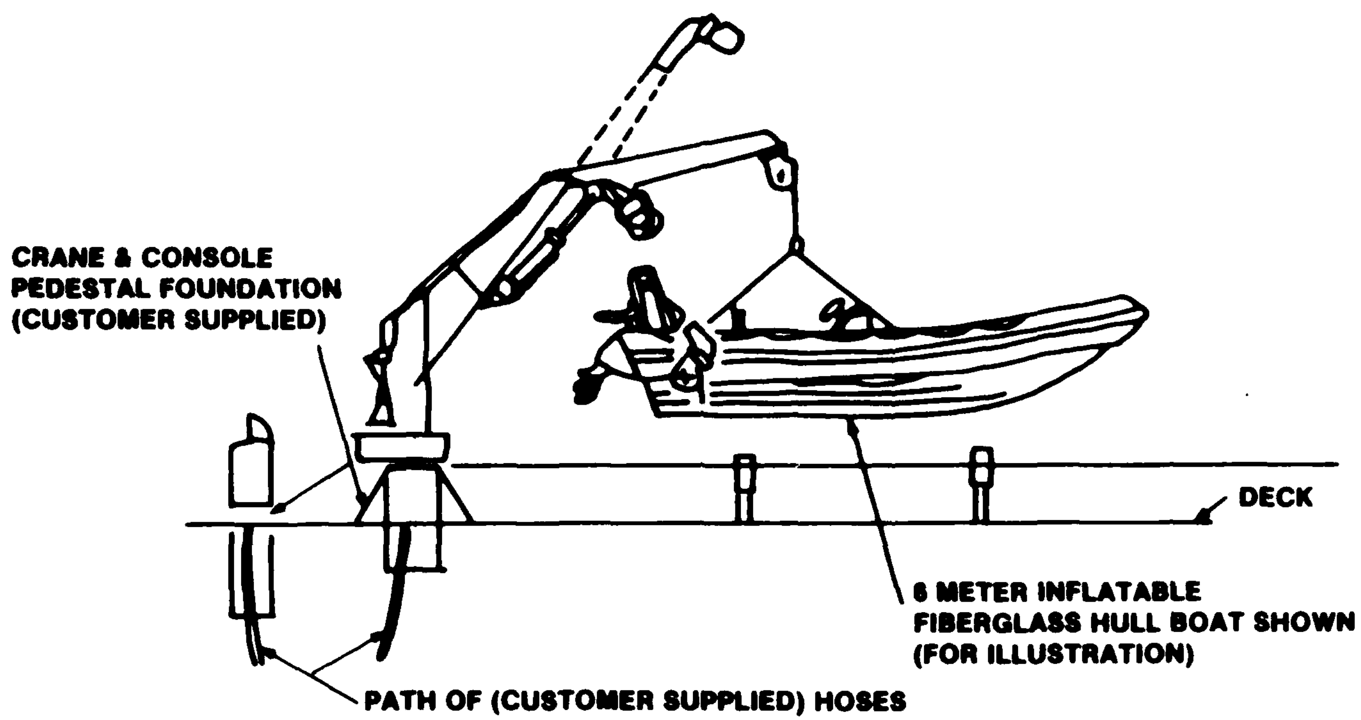

FIGURE 7.6 Knuckle Crane (Allied Marine Crane)

-The boom is more massive than on derricks due to the fact that it must now take all the loading. The added weight of the crane boom makes the crane system heavier than a derrick system capable of lifting the same SWL.

-A higher level of expertise for maintenance is required since most cranes utilize hydraulic control systems.

-Higher cost than conventional derricks.

7.4.3 Gantry Cranes

Gantry cranes consist of a lifting device suspended from an overhead track or framework which travels on rails. This arrangement, while not common on ships is found in limited numbers on container ships. The major advantage of this type of crane is the elimination of the boom and all its associated guys. The advantages of gantry cranes include precise linear control, high load capacity, and high throughput. Gantry cranes are often used in situations requiring repetitive handling and high tons/hour throughput. Figure 7.7 shows a typical gantry crane installation. Because lifting buoys is not a high throughput operation and is not repetitive, gantry cranes were not investigated in this survey.

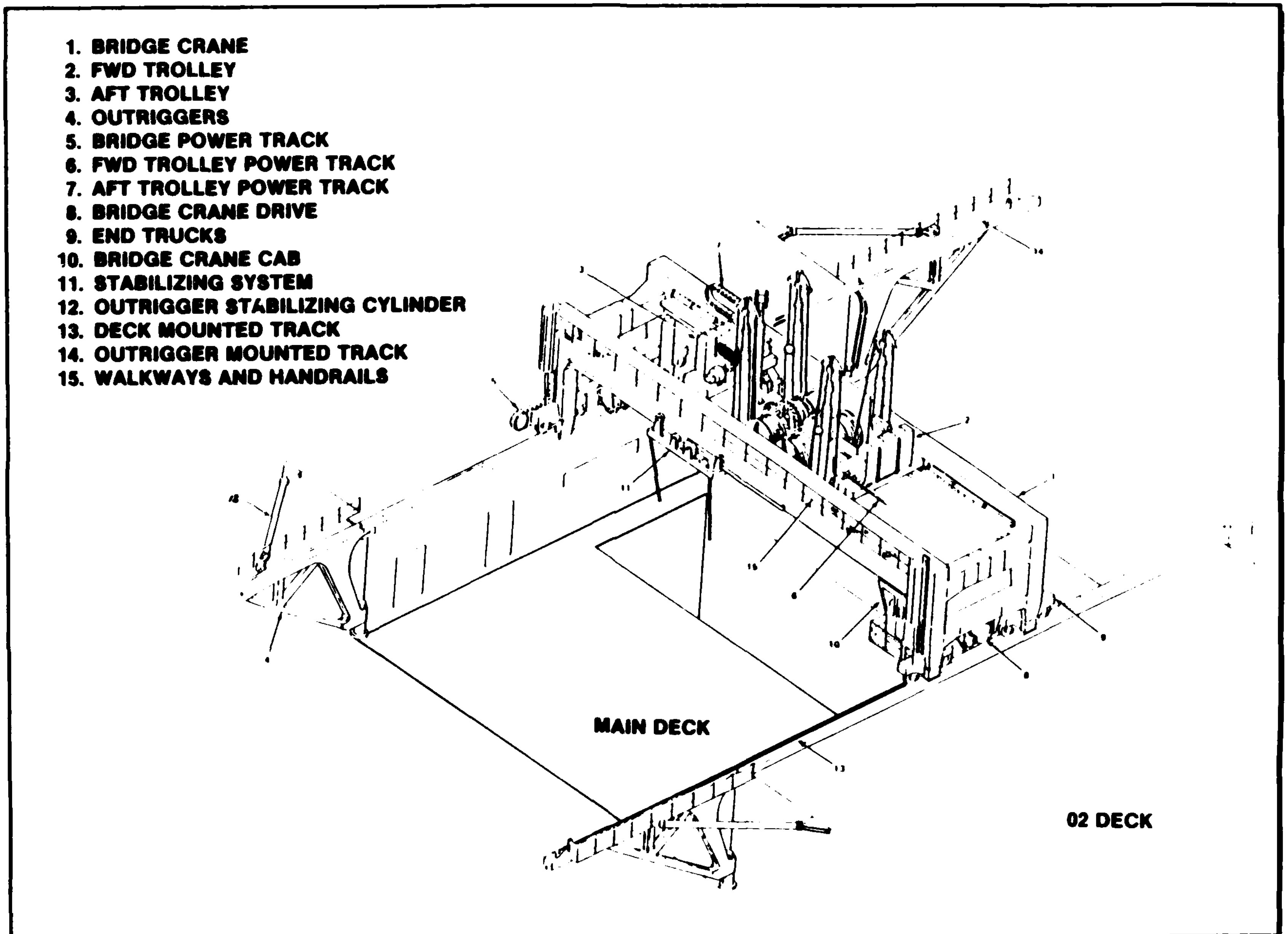

FIGURE 7.7 Typical Gantry Crane Installation

7.4.4 A-Frames

For applications in which very heavy equipment such as a submersible or other heavy equipment must be deployed and towed astern, an A-Frame shaped lifting arrangement is often used. The actual shape may often be rectangular, as well as an "A" shape. Such an arrangement was selected for the U.S. Navy's Motion Compensating Deck Handling System (MCDHS) used to deploy the Remote Unmanned Work Station (RUWS). Development of the MCDHS was accomplished at the Naval Oceans Center by Techwest Enterprises, Ltd. of Vancouver, B. C. (ref. 7.5).

This system, shown in Figure 7.8, was designed to maintain the heavy RUWS at a specified depth in spite of surface excitation of the towing cable. Another A-Frame system has been described by Daidola and Griffin (ref. 7.6) for the proposed design of an oceanographic research vessel shown in Figure 7.9. The major advantage of A-Frames is their ability to lift very heavy objects over the stern. Their major disadvantage is their inability to spot objects precisely on the working deck. Although few buoy tenders utilize A-Frames, this system might be capable of lifting buoys for tender designs which utilize an aft working deck.

7.5 DRIVES

Several different drive systems are used to power weight handling machinery. The speed control, speed vs. load characteristics,

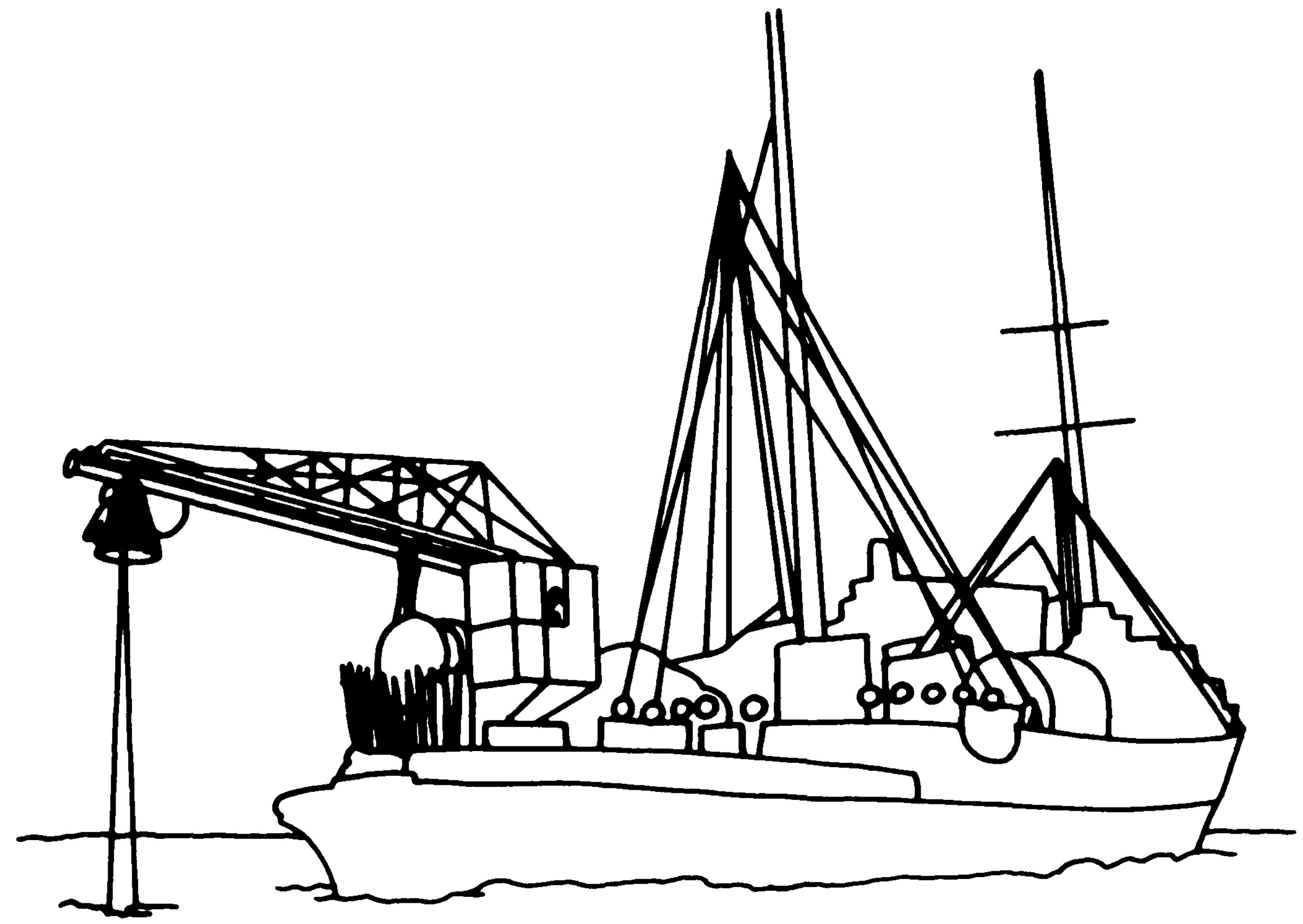

FIGURE 7.8 Motion Compensating Deck Handling System
(Techwest, Ltd.)

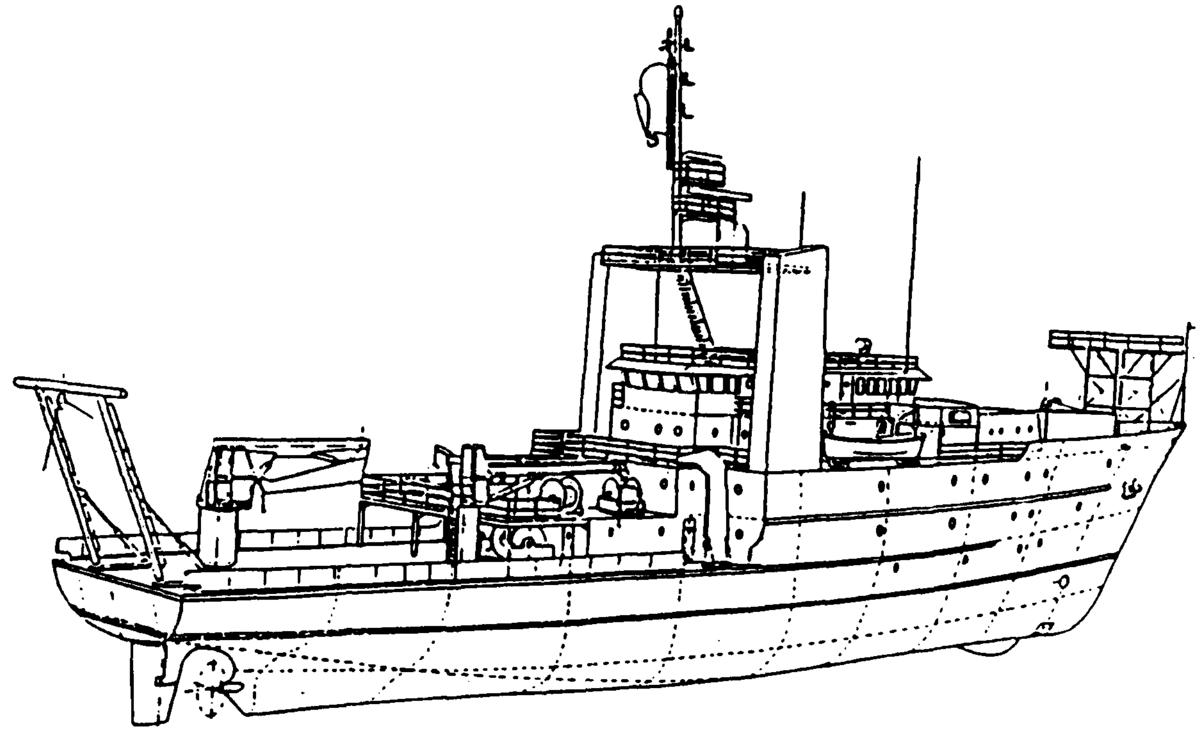

FIGURE 7.9 Oceanographic R/V Design with A-Frame
(M. Rosenblatt & Son, Inc.)

and precision of control of each alternative are unique. For some applications this may dictate one system over another. However, cost, maintenance, and the personal prejudices of the designer seem to be of equal importance.

In general the choice is between either electric or hydraulic drives. Electric drives are either of the DC or AC type. Hydraulic drives are either of the high pressure or low pressure type.

7.5.1 <u>DC Electric Drives</u>

DC electric drive provides very precise, infinitely variable speed control through simple and reliable devices. Some control schemes allow for variable speeds, however, the minimum speed is significantly greater than zero. DC power on ships has traditionally been generated by motor-generator sets. In recent years, however, DC power has been provided through the use of solid-state inverters and associated controls. More maintenance can be expected with DC motors than their AC counterparts.

7.5.2 <u>AC Electric Drives</u>

AC electric drives exist in two forms, discrete stepped speed drives which are known as squirrel cage motors, and continuously variable speed drives which are known as wound round motors. The stepped speed version is mechanically controlled and subject to a very high transient startup current. This is not a problem provided that such current is anticipated and provided for in the design. The continuously variable speed version is electronically controlled. It provides very precise control with high reliability and low maintenance.

7.5.3 Hydraulic Drives

Most crane manufacturers favor hydraulic drives. More than three-quarters of all cranes being built are hydraulically driven (ref. 7.7). The reasons for this preference include compactness, smoothness of operation, durability and possibly ease of maintenance. The claim for ease of maintenance however is disputed by several manufacturers (of electric drives) who point out the need for cleanliness of the hydraulic oil which is also prone to leakage.

All hydraulic drives consist of a hydraulic pump which is itself driven by an electric motor. The most common installation consists of an AC induction motor running at constant speed, with the hydraulic motor either controlled by a piston pump or a fixed delivery pump. A controlled amount of hydraulic oil is allowed to pass to the hydraulic motor with the excess diverted to the hydraulic oil reservoir (ref. 7.8).

Hydraulic drives may be divided between high and low pressure. High pressure drives allow higher hoisting speeds with smaller lines, but are also prone to more leakage and problems associated with contamination of the oil. Low pressure hydraulic drives require high oil flows, and consequently heavier lines, resulting in higher installation costs.

7.6 APPLICATION OF WEIGHT HANDLING EQUIPMENT TO BUOY TENDING

7.6.1 Fixed Derricks

This type of derrick does not appear to have any application to buoy tending operations. While it would offer the advantage of reduced pendulation, the lack of precise spotting, involved setup procedures, and operator skill requirements make this type of lifting gear unfeasible.

7.6.2 Swinging Derricks

Until recently this was the only equipment used for buoy tending. This class has, however, taken on a great many variations with the arrangement of vangs, slewing wires, topping lifts, trunnion gears, etc. Additionally, these derricks have been mounted on well deck vessels both facing forward and aft. The major advantage to this method of weight handling appears to be simplicity and ruggedness, low topside weight (improved stability), ease of serviceability and low skill level required for maintenance. Some significant limitations of this technology include:

- -Susceptibility of boom to instability or damage from non-vertical loads, e.g. buoy tending aft of the normal working area resulting in an overturning moment on boom. Also instabilty at high topping angles.

- -Limited slewing angle due to the attached guys.

- -High operator skill level to coordinate separate topping, slewing, and lifting functions.

- -Exposure of operating machinery to weather.

Generally, existing swinging derricks have not been designed to incorporate any motion compensation, although there is no reason why they could not be equipped with constant tension winches to reduce shock loading.

7.6.3 Cranes

While there are very real reasons to choose between a fixed, telescoping, or articulated boom, the characteristics are so dependent on the manufacturer and control system that no attempt to compare them will be made here. Cranes are a fairly recent entrant into the field of buoy tending. Their major assets which relate to the buoy tending operations include:

- -Totally self-contained mechanisms. While this minimizes corrosion of internal parts, there is a trade-off in that cranes become harder to service.

- -Most cranes have very sophisticated control systems which often can include some form of motion compensation.

- -Ability to rotate 360 degrees. This feature is most advantageous when working from an open stern deck.

- -Cranes can usually reach a greater height above deck which is very useful for pulling chain onto deck.

- -Operation is easier and faster due to the integrated control systems.

Cranes also possess some significant disadvantages when compared to derricks:

- -Significantly greater topside weight and height for the same SWL. This additional weight drives up the size of the buoy tender hull needed to support the crane.

-Cranes occupy otherwise usable deck space especially on well deck vessels.

-Maintenance is complicated by close quarters.

-Crane systems are more complex and may not be able to be serviced by ship's personnel without special training.

-Crane booms are as susceptible to damage from lateral loads as their derrick counterparts.

7.6.4 A-Frames

These cranes are in limited service for buoy tending operations. This concept of buoy handling is intimately related to working the buoy over the stern of the vessel. While it is not inconceivable that an A-frame could be used to work buoys over the side, it is doubtful that the arrangement could compete with either derricks of cranes. The advantage of using the A-frame over the stern is that extremely good control of the buoy can be maintained with minimal effects from vessel roll. This combination should enable buoy operations in higher sea states than are now possible. This potential has not been realized because conventional approaches to the buoy require the ship with an A-frame to back upwind toward the buoy, making the stern of the vessel a very hazardous place tc be in even in moderate seas.

It may be possible to eliminate this problem in a properly configured vessel (propulsion and control), by backing down on the buoy from upwind, keeping the bow into the seas. A-frames may also be effective on non-conventional hull forms such as catamarans SWATHs, or surface effects ships.

7.6.5 Survey of Weight Lifting Equipment and Manufacturers

Table 7.2 lists major manufacturers of lifting gear along with brief descriptions of their products. Appendix D contains a more detailed list of specifications from major manufacturers. This information has been obtained from trade journals and manufacturers' responses to inquiries. Because of the highly custom nature of lifting gear installations, very limited data on specifications and cost were forthcoming from manufacturers.

7.7 ADAPTATION OF MOTION COMPENSATION TO BUOY TENDING

The primary impetus for the incorporation of motion compensated weight handling systems on buoy tenders is to extend the operational "window". On present USCG tenders the decision on whether or not work buoys is based upon the judgement of the tender's CO who must consider the influences of wave height, wave direction, wind speed, wind direction, vessel response to these environmental factors, and type of buoy to be worked, its location, presence of ice, nature of the service (e.g. setting or

TABLE 7.2

SURVEY OF MANUFACTURERS OF WEIGHT HANDLING GEAR

Company Name	Location	Description of Equipment
Wes Tech Gear Corp.	Lynwood, CA	Custom manufacturers of heavy lifting gear. Developed lifting systems for Navy's DSRV, Glomar Explorer, ASR. Much experience with motion compensation.
TechWest	Vancouver, B.C.	Custom manufacturer of heavy lifting gear. Developed Navy's MCDHS system. Designs motion-compensated systems. Developed Navy's FACS crane.
Asea Hogglunds	New Jersey U.S. location	Manufactures fixed boom deck cranes of capacity up to 40 tons. Hydraulically driven. Uses low-speed, high torque hydraulic motors for both luffing and hoisting drivers. Has introduced motion compensation systems "Steadyline" and "Swing Defeater" for cargo-aligning and anti-swing. Cranes are designed to work to -40 degrees C. Also manufactures hydraulic winches.
Liebherr Werk	Nenzing, Austria	Manufactures cranes. Line of slim-line cranes with capacities of up to 80 tons. Has high-pivoted jib and drivers cabin below jib. Electro-hydraulically operated, computer controlled. Load capacities of 5-35 tons and radius of up to 72 feet.
Take Shore	Iron Mountain, Michigan	Turn-boom pedestal cranes. Each crane capable of lifting 33 short tons with effective outreach of 108 feet from side of ship. Can operate cranes with maximum list of 5 degrees and under environmental conditions of winds up to 30 knots and temperatures from 0 degrees F to 120 degrees F. Hoisting speeds of up to 120 ft/min, luffing minimum to maximum in 60 secs and slewing at .7 rpm. Contains pendulation control device in form of rider block tagline system. Also includes a load orientation control device that is a powered rotator. Reference: ME Log, February 1987.

TABLE 7.2 (cont'd)

SURVEY OF MANUFACTURERS OF WEIGHT HANDLING GEAR

Company Name	Location	Description of Equipment
John Hastie	UK	Manufactures Speedcrane Derrick. SWL = 20 ton, Outreach = 3 m over ships side, Jib length = 17.5 m, Lifting speed = 25 m/min, Able to operate with + 10 deg. heel.
NEI Clarke Chapman	UK	Challenger deck crane, electrically driven, motion compensated.
Hydrolift		SWL=6 tons at 18 m outreach, 12 tons at 9 m outreach.
Denag Kampnagel		SWL = 12 tons at 22 m outreach Wave-following device.
Pitt		Electro-hydraulic crane SWL = 20 tons.
Welin Davit & Eng.	UK	SWL = 30 tons at 3 m outreach.
Maritime Hydraulics		Heavy lift crane, 130-ton capacity, Hook stabilized to + 20 cm in 10 secs.
Appleton Marine	Appleton, WI	Manufactures series of fixed boom, telescoping, and knuckle boom cranes to 10 tons SWL.
Allied Marine Crane	Sherwood, Oregon	Manufacturers of marine cranes.
National Marine Cranes	Waverly, Nebraska	Manufacturers of marine cranes fixed boom and telescoping boom.

relieving, recharge, mooring check, and/or replacement), and of course, crew experience and condition. If incorporation of motion compensation is to be cost effective, there must be a clearly defined benefit in terms of more "go" decisions by the CO. In addition, there may also be non-economic considerations such as enhanced safety of the deck crew. Because such a detailed cost/benefit analysis is beyond the scope of this survey (and also detailed quotations were difficult to obtain from manufacturers due to the highly customized nature of most installations) this survey only discusses the instances where motion compensation would benefit the buoy handling operations.

In its most sophisticated form, through the use of accelerometers and other sensors, motion compensation can enable a crane to transfer a heavy object such as a tank from one moving vessel to another vessel which may be moving in a different manner. The relative velocity between the transferred object and the receiving vessel is essentially zero both vertically and horizontally. At the other end of sophistication, motion compensation can be nothing more than a constant tension device which will serve to reduce the shock loading on the derrick or crane. In general, the promise of motion compensating devices is operations in higher seastates; the price is increased cost and complexity (maintenance).

An examination of the operations conducted in the process of lifting a buoy must be made to see just where motion compensation can be useful. In the following discussion the assumption is made that the present system of aids-to-navigation remains unchanged. Thus the size, weight, and designs will not drastically change. The following is based upon the authors collective observations of buoy tending operations and discussions with foreign (especially Canadian) buoy tending personnel.

7.7.1 Hooking the Buoy

Throughout the world current technology dictates that this operation must be done manually. Perhaps in the future there might be some laser guided tag line which can be fired at the buoy and will eliminate the need for human intervention, but, for the present, the only way to hook the buoy is for the vessel to maneuver close by the buoy and a line to be passed through the cage of the buoy to steady the buoy while a tag line to the main lifting hook is then passed through one of the buoy bails. This operation requires that the tag be passed manually usually with the aid of a device called the "Happy Hooker". The tag line is then manually pulled until the lifting hook is properly positioned in the buoy bail. While the lifting hook is being positioned, its wire is slack. This phase of the operation may be weather limiting since passing of the tag line can be quite difficult. Motion compensation of the lifting device would be of no value in this phase of the operation since the lifting wire is slack. Motion compensation could only be valuable in this phase if the crane and sensors were sufficiently fast and accurate enough to track the buoy precisely and the buoy hooking device was redesigned to resemble those used for in-flight refueling of aircraft.

7.7.2 Hoisting the Buoy Aboard

This phase of the operation is characterized by high dynamic loads on tne lifting gear. As the lift begins, the buoy is alternately floating and suspended by the lifting cable as each wave passes. Also, until the buoy is totally suspended and the lifting device brought inboard to pin the buoy against the side of the ship, the buoy will swing to and from the side of the ship, creating additional dynamic loads on all parts of the system. The buoy, with the aid of one or more lateral lines, is then half dragged, half lifted to the location at which it will be "gripped down". Motion compensation in the form of a constant tension hoisting winch could be helpful in this state by reducing the high shock loading on the lifting gear. Thus the lifting gear could be reduced in size or its SWL increased. More sophisticated motion compensation devices might be useful in this stage by allowing the buoy to be hooked away from the side of the ship. The buoy could then be lifted from the water clear of the side of the ship thus reducing the incidence of damage to the buoy from crashing into the side of the vessel. Such motion compensation devices would need to reduce both the dynamic shock loads and the pendulation tendency of the buoy. At present there exists at least one crane manufacturer with both such devices (Haaglund's "Steady Line" and "Swing Defeater").

7.7.3 Buoy on Deck

One of the most critical periods in buoy operations occurs when the buoy is on deck but not gripped down. This condition occurs both when the buoy is being brought aboard and when it is being readied to set. During either of these periods the buoy is in part resting on the deck and being held by the lifting wire and at least one "crossdeck" line. The anchor chain acts to some extent as a counter force to the crossdeck line. The objective at these times is to minimize the athwartship sliding of the buoy due to rolling of the ship while personnel are securing it, and to some extent releasing it. It is doubtful that any sort of motion compensating device on the lifting gear would be helpful during this phase of the operation because there is little relative motion between the boom and the deck. However, constant tension devices in the crossdeck winches may reduce the shock loads in crossdeck lines at the risk of allowing some buoy movement on deck. The most effective motion compensation system at this phase would be anti-roll devices on the vessel hull itself.

7.7.4 Setting the Buoy

Depending upon the amount of chain on deck, this operation involves picking the buoy up partially or completely clear of the deck. Using the lifting wire and the crossdeck line, the buoy is positioned over, but against, the side of the ship. At this time the buoy is partially afloat. When the final position is reached the sinker is set, the buoy is fully floated, and the lines tripped. Motion compensation would be of some value here since, as the buoy clears the deck, pendulation can become a problem even with the crossdeck line attached. More importantly, constant line tension would significantly reduce the dynamic loads experienced by the weight lifting gear while the buoy is partially floating.

7.7.5 Summary of Motion Compensation Applications

Limited motion compensation in the form of a constant tensioning device appears to be a viable method for reducing dynamic shock loads on weight handling devices. It is suitable for both cranes and winches. The primary advantage of motion compensation would be to increase the SWL for a given lifting systems. No longer would massive structures be required to resist peak dynamic loads. The result would be a decrease in overall crane (or derrick) weight.

Chapter 7 References

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8.0 VESSEL AUTOMATION, CONTROL AND MONITORING

8.1 DYNAMIC POSITIONING

8.1.1 Background

Dynamic positioning (DP) can be described as the technique of maintaining the position and heading of a floating vessel by means of an active thruster controlled by a computer. The use of DP for vessels is not new. The first dynamic positioning systems appeared in the early 1960s on ships designed for coring, cable laying or surface support of underwater work. The success of these initial DP vessels has encouraged expansion of DP applications into areas such as offshore loading, dredging, precision dumping, pipe laying and floating hotels.

In many DP applications, anchor deployment had been considered the norm. So why use DP over anchoring? DP offers several advantages (refs. 8.1, 8.2 and 8.3) such as:

- -Effectiveness in all water depths
- -Ability to begin operations immediately upon arrival
- -No need for auxiliary vessels
- -Mobility of position and heading
- -Controlled traverse where needed
- -Ability to work close to platforms or other anchored objects

-Ability to work in adverse weather

However, DP is not without its disadvantages, which include:

-High capital cost

-High Fuel cost

-High maintenance cost

-High manpower for operation and maintenance

-Reliance on an active rather than passive positioning system

DP is an effective means of stationkeeping when:

-the water depth is too great to anchor

-only short times will be spent on station (as in buoy tending)

-all weather stationkeeping is required

-or some factor other than cost is most important

For a ship to be dynamically positioned it must employ the following special equipment; thrusters, sensors, computer, and a control and display system. The thrusters must be able to provide forward-aft, athwartships and rotational thrust. The sensors will provide physical information including position, heading, and wind-wave conditions. The computer uses the sensor information and calculates the required thruster commands that will position the vessel at the desired location and heading. The controls and displays are required to allow an operator to monitor and control the DP system. In addition, an interface system converts sensor output to computer input, and output of the computer to control signals for the thrusters, main propulsion, and rudder control.

8.1.2 Sensors

There are several sensors that are employed in a DP control system. For example, there can be a wind sensor, gyro or flux gate compass, vertical and horizontal position sensors. Both the wind sensor and compass systems are fairly standard items and need no further discussion. There are three types of vertical reference sensors. The first uses two single-axis inclinometers at right angles to each other, aligned with the vessel's roll and pitch angles. The second is a vertical gyro. The last type contains a gravity stabilized, gimballed sensor that acts as a perpendicular with a natural period of 40 seconds. This sensor is not sensitive to lateral forces. It is most desirable since it can be used to minimize wave-induced thruster activity.

Position sensors are the most important and diversified of the required sensors. They include sensors that determine position, velocity, or acceleration in some common reference system. A combination of position sensors is usually needed by the control system. The basic position sensor types are taut wire, optical, acoustic beacon, and radio systems.

Before discussing each sensor type, it is important to understand the role they play in the DP control system. The maneuvers of a vessel can be broken into three phases. The duration of each maneuver phase is determined by the vessel mission and local conditions.

The first maneuver phase is general navigation of the vessel in the classic sense. The accuracy, speed and safety needed for this phase is dependent on vessel mission objectives and local conditions. The frame of reference used can be nonlinear, like the hyperbolic Loran-C grid which can contain irregularities between the predicted and actual position. It can be unrelated to actual surface position at all, such as searching for a wreck using the fathometer. This maneuver phase has little relationship to DP. How the vessel gets somewhere or why it is there is of no concern to the DP control system. During this phase, the DP control system would be in standby. The requirement for frequent position updates with a high degree of accuracy tends to be relaxed relative to the DP requirements. What sensor is best for vessel navigation and what sensor is best for the DP control system should not be assumed to be the same. This simple realization is often overlooked when DP sensors are being selected.

The second phase is the close-in maneuvering of the vessel to the location that the DP control system will have to hold. This maneuvering phase is characterized by the loss of steerage and complete maneuver dependence on the propulsion system. Again the final vessel position may or may not require the use of the precise positioning sensor(s) depending on the vessel mission objectives. If the desired vessel position and orientation are known upon entering this maneuver phase, the DP control system should be designed to assume the job of maneuvering during this phase. If position and orientation must be adjusted by the vessel operator, the DP control system should be designed to assist during this phase until the vessel has been properly positioned.

The third phase is holding a stationary position. This function should be done completely by the DP control system. Any operator intervention should be viewed as a temporary change back to the second phase of maneuvering.

Distinguishing among the three phases of maneuvering makes it easier to understand why certain sensors are needed and how they should be used during execution of a vessel mission. It also makes it easier to understand why a sensor may be perfectly adequate for one phase of maneuvering and unacceptable for another. No single sensor possesses all the data needed for all three maneuver phases and this is why the DP control system design is complex. Its design must be approached at a systems level with a clear definition of the vessels mission objectives.

All the sensor types listed above can be used in the third maneuver (DP) phase. The optical and radio position systems can be used to assist in all three maneuver phases, but the importance of their different characteristics changes in the three phases. The acoustic beacon technology can also be used to a limited degree in the second maneuver phase.

A key to minimizing plant fatigue and fuel consumption caused by DP control system operation is noise free rate of change information. After all, if there were no forces on the vessel, the DP control system would not have to expend any energy to hold a position. Other than the present position and heading of the vessel, the velocity and the rate of heading change are the most important information for the control system. This is why direct doppler measurements (radio or acoustic) provide the best overall short term velocity information to the DP control system. As an alternative, velocity change information can be obtained from integrating ship acceleration with accelerometers or an inertial platform. Velocity can also be obtained by timing position changes measured by the positioning system. However, the short term noise in the position measurements can be very high causing excessive plant adjustments. Heavy filtering of position information can smooth plant operation and reduce fuel consumption but the ability to hold a specific position is also hampered. The optimal mix of sensors and filtering is the key to DP control system design and must be taken into account early in the design of the overall DP vessel.

8.1.3 Position Reference Systems

There are four types of position systems presently available for dynamic positioning; taut wire, optical, acoustic beacon, and radio. The use of positioning systems introduces a source of noise into the DP control system that can be as significant as the actual vessel motion. For this reason, the position reference system must be accurate and have good short term stability. In general, accuracy determines the mission efficiency of the vessel (accuracy requirement of mission) while short term position stability determines the power efficiency (power consumed by thruster action). Incorporation of an inertial reference system within the control system shows promise in improving the power efficiency of the DP control but can be very expensive. This makes the choice of an appropriate positioning sensor a tradeoff between absolute accuracy and short term (low noise) stability.

The taut wire system works by lowering a weight to the sea floor by wire rope. The wire is held in constant tension, either by a constant tension winch (preferred) or by a counterweight running in vertical guides through a form of gimbal-mounted jaws that decreases the angle in any direction from the vertical. Taut wires are usually accurate to within 2-3% of water depth and the repeatability is better than 2%. Taut wire systems are very simple, free from erratic interruptions and very robust. However, taut wire sensors have some limitations. Their horizontal range is about ± 25% of water depth and their maximum water depth for operation is limited to about 300 meters. Also the system, like other mechanical devices, is subject to wear and requires preventive maintenance. In particular, the rope which is heavily stressed and continually used, requires strict attention.

Optical positioning systems can be of passive or active design. Active optical systems make use of laser technology. The most common positioning method aboard USCG buoy tenders involves the use of sextants. Horizontal angle sightings (resection) between objects at known shore locations are used to position many of the buoy anchors placed by the USCG each year. A limited number of electronic sextants that can transmit angular measurements directly to a computerized positioning system such as AAPS (see 8.2.1 Radionavigation) have been developed. Active laser systems can provide range, azimuth, and elevation to the position of a known target. In order to derive good position information, several sextants or laser systems must be in simultaneous operation. The minimum is two. Each system in operation must be manned. Fatigue of the operator is a significant limitation for applying this technology to DP vessels.

Acoustic positioning systems are the most commonly used in the offshore industry. There are several types, but most have a processor/display unit and hull-mounted (or "stalk" mounted) hydrophones to detect acoustic signals from transponders mounted on the sea floor. These systems may have a short baseline (multiple hydrophones), ultra short baseline (single hydrophone) or long baseline (single hydrophone, multiple transponders). Acoustic systems use either pulse time-of-arrival or phase shift comparison techniques for determining vessel position. Some systems have the capability of depth and tilt measurements.

Acoustic systems are generally accurate to within 2.5% of water depth and are repeatable within 2%. Their operating range varies significantly with the manufacturer, but can be as great as 500% of the water depth. The advantages of acoustic systems are that there is no actual physical link to sea bed and the transponders can be left in place to locate a returning vessel.

The major disadvantages of acoustic positioning systems are erratic signal transmission and limited range. This is most often due to underwater ship noise and propeller operation. This is a significant disadvantage aboard a DP vessel when the thruster system is creating a significant area of water turbulence. Thruster propellers encased in tunnels or nozzles seem to reduce the noise problem. There is some evidence that fixed-pitch propellers running at reduced speed cause less noise than the controllable pitch propellers at high speed and reduced pitch (ref. 8.4). Nevertheless, it must be up to the user to realize that acoustic positioning systems are sensitive to the environment and operation of the propulsion system.

Radio positioning is best done using special purpose electronic ranging systems operating on frequencies between the VHF and radar bands. There are also short range systems that measure the range and bearing of a fixed radio beacon (ref. 8.5). These frequencies limit the separation between the vessel and known reference positions to about 20 miles. A sampling of coverage and accuracy for some popular systems are listed in Table 8.1. In a typical application the control unit and position computer are aboard the vessel. Ranges to remote units located at known sites are measured electronically every second. The measured ranges are used to calculate the position of the shipboard antenna by triangulation.

Generally, microwave ranging systems are the most commonly used with DP because of their high accuracy and good all-weather capabilities. The concept of having this type of system mounted on a floating structure that is being serviced, so that a constant relative position with respect to the structure is kept, has been tested (ref. 8.6).

It is generally felt that radio navigation signals (Loran-C, OMEGA, and Transit) do not give enough accuracy or short term signal stability for DP applications. However, the NAVSTAR GPS scheduled for full 2-D marine operation by 1990 may have sufficient accuracy and stability to support DP (see Section 8.2 Navigation Aids).

TABLE 8.1
SHORT RANGE RADIO NAVIGATION SYSTEMS

System	General Range (km)	DP Range (km)	Accuracy (m)	Measurement Method
ARTEMIS Standard	30	5-10	2-5	Distance/ bearing 1 cm
ARTEMIS Short Range	5	2	0.5-1	Distance/ bearing 1 cm
AUTOTAPE	50		0.5-1	Distance triangulations/ phase measurement
TELLUROMETER	50	20-30	0.5-1	Distance triangulations/ phase measurement
TRISPONDER	50	20-30	5-10	Distance triangulations/ 1 pulse measurement
MINIRANGER	50	20-30	5-10	Distance triangulations/ 1 pulse measurement
TRIDENT III	100	30-50	5-10	Distance triangulations/ 1 pulse measurement
SYLEDIS	100	50	5-10	Distance triangulations/ 1 pulse measurement
RADACTOR		2-3	2-3	Distance/bearing/ pulsed
DIFFERENTIAL GPS/SPS	*	*	2-8	Distance triangulation/ communications

* Subject of present research program at USCG R&D Center, presently thought to be greater than 400 km.

8.1.4 Thrusters and Main Propulsion Units

The significant difference between conventional and DP duty for thrusters is the number of thrust changes required. A DP system will inevitably require a continuously changing value from each thruster. The magnitude and frequency of the changes can be reduced by careful design, but the thrusters and their control systems must be built to withstand upwards of 750 thrust changes per hour, often 24 hours a day (ref. 8.4).

Thrusters for DP take many forms. Fore-and-aft thrusters are usually provided by the vessel's main propulsion, which can be thrusters rather than screws. The screw type is often of controllable pitch. Athwartships and rotational thrust is provided by groups of tunnel thrusters, water jet nozzles, or azimuth thrusters; the latter are sometimes retractable for maintenance or docking. Thrusters are operated by diesel engines through a gear box, or more likely electrically driven. Some thrusters are even hydraulically driven, such as the DP thrusters in the Buoy Tender Wilton, (ref. 8.7).

An often overlooked design aspect of DP thrusters is that they should be able to provide thrust repeatably at a level of about 2% of the maximum rated thrust. The lack of repeatability is in most cases due to the controls, such as hydraulic, containing dead bands or lost motion. If the DP ship cannot obtain the required thrust setting, the whole ship might oscillate about the required position while the thrusters are moved alternately above and below the required level. Other potential areas for poor design are the servo systems for azimuth thrusters and engine speed control. These are examples of the need to carefully inspect the overall design details.

8.1.5 Computer and Control Systems

The computer is essentially the heart of the DP system. Here all the sensor data merges and the current position and heading are established. These are compared with the desired position and heading. The necessary differences between these positions initiate corrective thruster movements taking into account the magnitude of deviation from desired location and effect of wind on the vessel.

In the case of multiple sensors, there are two distinct methods in which a computer can use sensor data. The first is a queue method, in which each type of sensor system is allocated a priority. In this method, if the first priority sensor system fails, then the second priority sensor is used. This means many sensors are ignored a majority of the time. The second method is a pooling method where every available sensor system contributes input to a pool from which a "best estimate" of position is determined. The pooling method is generally preferred since it provides a more consistent position (ref. 8.1).

Some computers carry out "reality checks" on incoming sensor data. If the data is out of the realm of the ship's behavior, the data is automatically discredited.

It has been found that a single computer is sufficient to provide control (ref. 8.8). However, some users, where the penalties of losing position are great, have employed a second computer in case of failure.

DP systems almost always have three operating modes, automatic, semiautomatic and manual. Automatic operation lets the computer hold position and heading automatically. Semiautomatic lets the computer automatically maintain the heading and the position is set manually. Manual operation is done by an operator using a joystick with the computer doing the thruster allocation. Some DP systems, such as Kongsbergs 503 DP system, also have automatic tracking which ensures that the ship will automatically follow a predetermined irregular course at a preselected speed.

8.1.6 Field Experience/Miscellaneous

It has been found that the major source of failure in DP systems is the position measurement equipment (ref. 8.4). Thus relying on only one measurement system is not recommended.

DP thrusters are used more frequently than thrusters in "conventional" service, and thus require more maintenance. It is suggested that ease of maintenance should be a factor in selecting a thruster system. A good example of this is the Buoy Tender Wilton's easily removed azimuth thrusters (ref. 8.7).

A simulation analysis can save many trial and error type selections for thruster size selection and vessel response. The Glomar Challenger is one vessel for which a simulation study was used to evaluate its DP abilities (ref. 8.8).

Additional operational hints are as simple as air conditioning the computer and waterproofing cabling and junction boxes of the DP equipment. A good maintenance schedule is strongly recommended for successful DP operations.

8.1.7 Conclusions/Summary

DP is no longer a new technology. It is fully established and recognized as a working tool. Modern DP systems are increasingly reliable and effective, yet improvements still continue to be made. One of the areas where there is vast room for improvement is in position reference sensors. The question about DP systems is no longer "will it work?", but "is it needed or cost effective in this situation?".

8.2 NAVIGATION AIDS

Typical on-board navigation aids in current use are radio navigation equipment and collision avoidance radars. These aids have been in use over the last couple of decades but are always undergoing changes and improvements. The coverage and accuracy of radio navigation varies with each system. Collision avoidance radars, also known as Automatic Radar Plotting Aids (ARPAs) or Collision Avoidance Systems (CAS), with recent improvements have been shown to increase ship operating safety.

8.2.1 Radionavigation

There are presently many radionavigation and radio positioning systems available with different uses, capabilities, (ref. 8.9) and operating costs. The federal radionavigation systems are described in the "Federal Radionavigation Plan" (FRP, ref. 8.25). The distinction between radionavigation and radio positioning has nothing to do with the specific accuracy of a system. It is governed by the operator of the system and the frequency spectrum utilized. International agreements have set aside and protected specific radio frequencies for radionavigation signals. In the United States all radionavigation signals are operated by the federal government. All other radio "positioning" systems not contained in the FRP are considered radio positioning systems.

The major radionavigation systems are Loran-C, OMEGA, NAVSTAR GPS, TRANSIT, Radiobeacons, VOR, VOR/DME, VORTAC, TACAN, ILS, and MLS. Of these, the wide area continuous coverage marine systems are Loran-C, OMEGA, and NAVSTAR GPS. The U.S. Navy's TRANSIT Satnav, which uses six satellites, is available world-wide and can give accurate positioning within 25 meters. Although a highly reliable all-weather service, there are long gaps between reliable fixes. Most TRANSIT navigation is based on dead reckoning sensors. The radiobeacon system provides continuous position information, but is used primarily as a "homing" system. The VOR, VOR/DME, VORTAC, TACAN, ILS, and MLS are designed for aircraft use.

Loran-C was originally developed by the U.S. Department of Defense during World War II. Loran-C stations are land based and have a useful range of about 1000 miles. Loran-C is not implemented for worldwide coverage, but does cover most of the U. S. coastline. Loran-C gives predictable accuracy within 460 meters, with a 95% probability. Construction of new Loran-C stations in the U. S. is underway. The purpose is to provide complete coverage of the U. S. for aviation. Research has shown that the local area (30 mile radius of monitor) accuracy of Loran-C can be improved to about 20 meters by applying local "corrections" to the measurements. This technique is called Differential Loran-C. Implementation of the technique requires a stationary monitor station that determines the local correction, a communications link from the monitor to the vessel, and a

Loran-C receiver on the vessel that has been modified to accept the local corrections. Differential corrections do not reduce the short term measurement noise aboard the vessel but do remove large bias errors.

The land-based Omega system provides world-wide positioning, but its accuracy is limited to 4 nautical miles with a 95% probability. Some experimentation to improve local area accuracy by using monitor stations and a correction method similar to that described for Loran-C above has been done. The accuracy can be improved to about 500 meters using this technique.

The NAVSTAR GPS system is presently available using prototype satellites. The system is being built by the DOD and is scheduled to be "operational" for the marine user by 1990. This system is the most complex radio position system ever built and can and will be used in a variety of ways. In addition to a variety of position accuracies that will be discussed below, this system allows vessel velocity to be measured directly using the Doppler principle to accuracies of about 0.1 meter per second. This capability could be significant when using GPS as a velocity sensor in a DP control system.

Two signal formats are broadcast by each GPS satellite. The precise signal format is called the Precise Precision Service (PPS). The predictable horizontal positional accuracy of this signal will be about 18 meters. The velocity accuracy will be

0.1 meters per second. The satellite signals are continuously available making the "fix" rate simply a function of receiver processing speed. Most receiver designs will provide a "fix" every second.

The other less precise signal from the satellite is called the Standard Precision Service (SPS). The predictable horizontal positional accuracy of this service will be about 100 meters. The velocity accuracy is expected to be very poor. An objective of the SPS is to provide some positional accuracy while providing as little velocity information as possible. It would not be wise to use SPS velocity information as input to any control system until DOD changes this policy. The USCG will have military receivers making both formats available for operations. Outside DOD and the USCG access to the precise signal format will be very difficult to obtain.

Experimentation to improve the position accuracy of the SPS is being done at the USCG R&D Center. The technique being explored is similar to the Differential Loran-C approach. A demonstration system has been constructed and used to test the concept. Both static and dynamic vessel tests show that the SPS position accuracy can be improved to better than 8 meters in the area of the monitor equipment (10 km). Future tests to explore the service range of a single monitor are scheduled. Initial findings suggest that the range will be on the order of hundreds of kilometers. Note that the position accuracy of the experimental differential SPS is better than PPS, but the velocity information from PPS is better than SPS or differential SPS.

The appropriate radionavigation or radio positioning system for a vessel is mission dependent. In general radionavigation systems are cheaper to use and maintain, but the accuracy is not as good as radio positioning systems. This distinction may change when GPS is operational.

An example of a radionavigation system which combines a variety of sensors and is designed to assist buoy tender operations is the Automated ATON Positioning System (AAPS). This system was developed in an effort to automate some of the tasks performed aboard a tender while positioning buoys. The computer based system presents sensor data (Loran-C, sextants, etc.) and maneuvering information in the form of a visual positioning grid. The conning officer can us this grid to assist in maneuvering to the position for buoy set. AAPS has been installed aboard five buoy tenders and has demonstrated a great potential for supporting buoy tender mission objectives. The USCG R&D Center is currently expanding the sensor input and capabilities of the AAPS.

8.2.2 Collision Avoidance Radars

Presently, most collision avoidance assessments are made by standard radar contacts. The assessment is made by interpreting the relative motion of the radar contacts on the radar display. Grease pencil reflection plotting marks are used to estimate closest point of approach. A skilled deck officer can competently plot and maintain 6 to 10 contacts using this method. It can be generally stated that reflection plotting is a time-consuming and tedious task.

CASs or ARPAs are automated equipment that can perform these tedious monitoring and control tasks more accurately and expeditiously. In high density traffic and limited visibility, CASs or ARPAs ability to free the deck officer of these tasks has obvious merits.

CASs or ARPAs have two basic parts (ref. 8.10). The first part extracts data from the radar video and other signals presented to it and uses a correlation and smoothing (tracking) process to estimate the position and velocity of specific echoes. The second part displays the information that the first part, the extractor tracker, has produced and provides controls for the operator. The extractor/tracker part of CAS is the most important, since it is the source of information for the deck officer making possible collision assessments.

There are two classes of extractor/tracker systems currently in use (ref. 8.10). The first system type employs "Global" target extraction that monitors the entire regional situation. This procedure which is essential for automatic target acquisition is often employed by many manual target acquisition systems. The second system employs the limited area extraction technique. This is a detailed analysis of a small area of interest in order to accurately determine the position of a target.

The majority of global systems are manually directed, due to the two distinct disadvantages of fully automated systems. The first disadvantage is the high threshold required during operation in order to compensate for variations in overall sensitivity and to avoid acquiring unwanted echoes. This high operating threshold reduces the number of weak but visible targets acquired. A second problem, related to the first, is that marginal targets near the threshold are not always continuously extractable. These compromises generally make a fully automatic system difficult, if not impossible, to use.

Limited area extraction/tracking systems, unlike global extractors, are software controlled. They can therefore be designed for a specific radar type, incorporating logarithmic video with a wide dynamic range, to optimize the extracting and tracking criteria for each target. This increases the system's sensitivity to changing target echo and movements, which results in less noise and clutter.

The coordinate systems in which the CASs or ARPAs track data is sorted fall into two categories. The data may be stored in relative motion (contact's motion relative to own ship's course and speed) or true motion (contact's motion with respect to earth coordinates) terms. Manufacturers of both types of systems claim improved accuracy. The relative motion method is claimed to ensure maximum accuracy in closest point of approach calculations by eliminating the need to convert from true motion to relative motion prior to calculation. The relative motion method of tracking loses credibility during ship maneuvers, since its velocity data must be reestablished after the ship has steadied from each maneuver. Depending on the sensitivity, a relative motion system may need to reestablish velocity data even after a large yaw. The true motion method doesn't have this requirement and is therefore a superior tracking method during ship maneuvering.

CASs or ARPAs have two general display formats which use contact motion vectors to indicate heading, speed and location. The vectors are displayed in either relative or true motion format. The true motion format is the most accepted and considered easiest to use. Some CAS systems even display a "predicted area of danger" enclosed in an ellipse. This ellipse is to indicate a course which should not be taken to avoid collision.

When CASs or ARPAs first appeared in the mid-1970s, there was widespread, and probably well-founded, skepticism regarding their reliability and accuracy. A field study performed by Shell International Marine concluded that CASs or ARPAs had difficulty tracking targets under conditions of clutter. Today's systems have significantly improved on their predecessors.

A recent field study comparing deck officer performance with and without automated information displays (ref. 8.11) found several positive conclusions on use of CASs or ARPAs. These are:

1. The range at which approaching ships were detected as threats was doubled.
2. The distance between ships at the closest point of approach resulting from evasive maneuvering was doubled.
3. The deck officer's workload was reduced by factors of 2 to 4 depending upon the situation.
4. Saturation workloads for officers in critical maneuvering situations were eliminated.

8.3 AUTO PILOTS

The typical autopilot of today has not changed much since the 1950s. Autopilots are "proportional, integral and derivative" (PID) controllers which implement solid-state electronics. There are several adjustable control functions, such as a "rate multiplier" and a proportional gain that can be varied according to load, weather, direction of waves and speed of the ship (ref. 8.12).

The optimum manual setting of PID control functions varies under different ship conditions. Determination of the optimum setting involves a degree of uncertainty in a case-by-case basis despite the experience of the operator. Operators tend to set the control functions to reduce heading error. It has been found that this practice is not necessarily the most economically efficient.

It is not surprising that autopilots which require no manual tuning of the control functions have recently been developed. These autopilots are termed "adaptive autopilots" since they are continuously tuning or "adapting" the control functions to changing environments and operating conditions.

Adaptive auto plots were introduced primarily for fuel economy rather than improved controllability. The priority of the adaptive autopilot is a function of what controller optimization technique was selected or programmed into the hardware (ref. 8.13). Optimization techniques vary from controller to controller with no one accepted standard as of yet. Several techniques have been reviewed (ref. 8.14). However, few manufacturers state what techniques are used in their adaptive autopilot.

Most adaptive autopilots have three modes of operation: open sea course keeping, confined sea course keeping and course changing. The only difference between the two seakeeping modes is that the confined seakeeping mode minimizes cross-track error for increased safety. The course changing mode will direct the ship to the new course specified at a turning rate previously set.

The actual monetary benefit of adaptive autopilots is fuel economy, claimed to be between .5-1.5%. The fuel economy is dependent upon controller type and sea states. Other, non-monetary, benefits are improved safety and seakeeping.

A recent simulation study (ref. 8.15) showed fuel savings of Adaptive Autopilots can be in excess of .5% over a typical well-tuned PID Autopilot. The simulation was conducted at a ship's design speed with random heading in varying sea states.

Mitsubishi published a report (ref. 8.16) on their Tonas-Pilot-I adaptive autopilot which gave a fuel saving rate of .5% in limited actual test conditions. Sperry has claimed (ref. 8.17) fuel savings of 1.5% in sea states 4-5 with their Adaptive Autopilot, the ASM.

It has been shown that at least one existing conventional PID controller has manual control settings that, chosen unwittingly, can cause directional instability in an otherwise stable ship (ref. 8.13). There is at present no proof that adaptive autopilots are foolproof to instability.

Adaptive Autopilots, though relatively new, will be on the increase for reasons of fuel saving, vessel stability and increased vessel safety.

8.4 MACHINERY MONITORING AND CONTROL

8.4.1 General

Engine rooms were one of the first areas of vessels to be automated. Originally this meant remote control of the propulsion machinery. However, automation in this area presently encompasses much more.

Generally the benefits derived from the application of automation are reduced personnel, increased safety of ship and machinery, increased efficiency, and less down time. Machinery Monitoring and Control usually addresses six critical areas which concern h overall safety of the ship. These are (ref. 8.17):

- -Alarm and safety systems for ship's systems
- -Remote indicating of alarms on the bridge and in engineer's accommodations
- -Remote control of main propulsion machinery from bridge
- -Fire detection and prevention in machinery space
- -Bilge flooding detection
- -Supply of electric power

A typical automation system which aims at some unmanned periods in the engine room is shown in Figure 8.1.

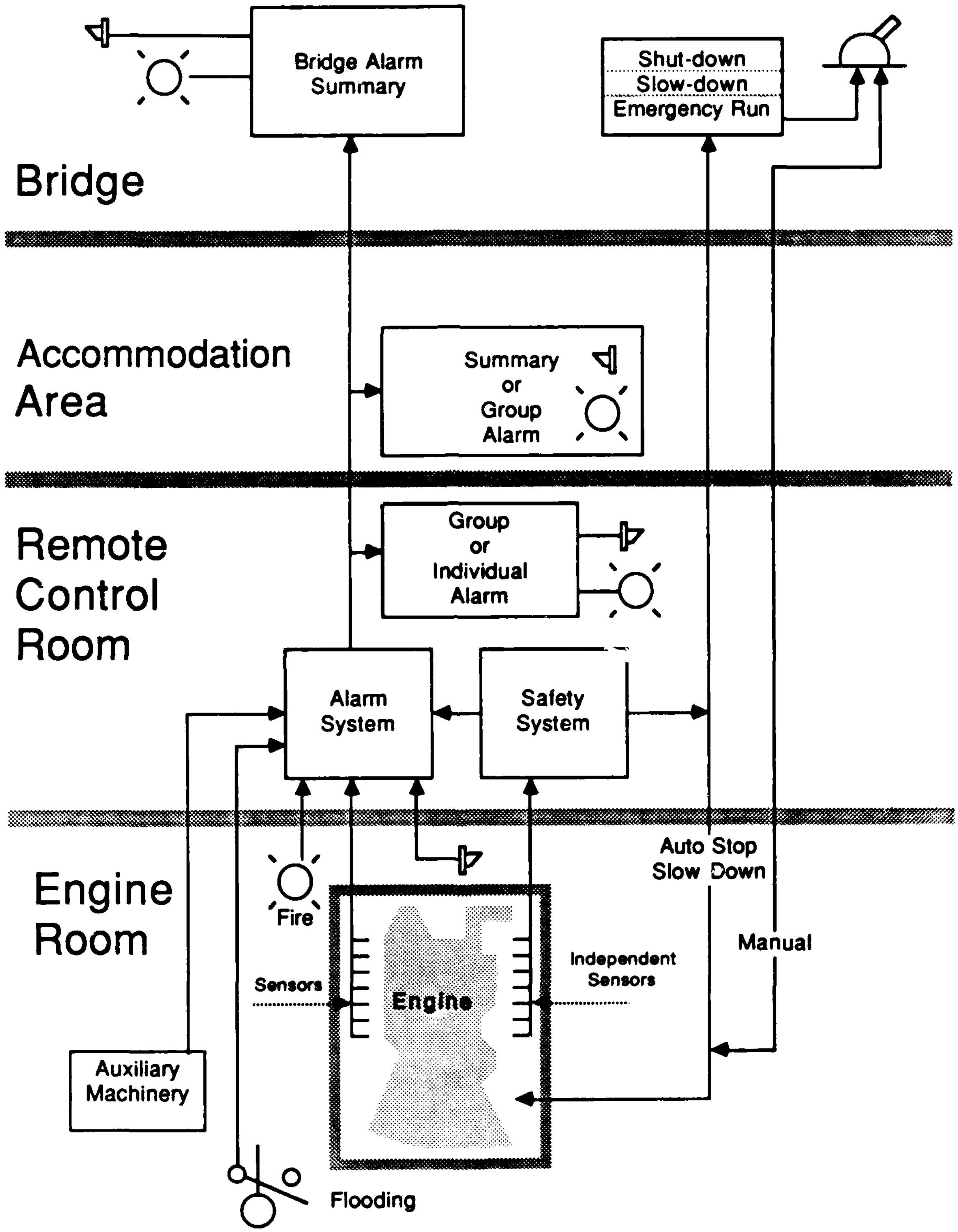

FIGURE 8.1 Unattended Engine Room Operation

AD-A193 918 SURVEY OF TECHNOLOGY WITH POSSIBLE APPLICATIONS TO UNITED STATES COAST GU..(U) COAST GUARD RESEARCH AND DEVELOPMENT CENTER GROTON CT S ALLEN ET AL. SEP 87 4/4

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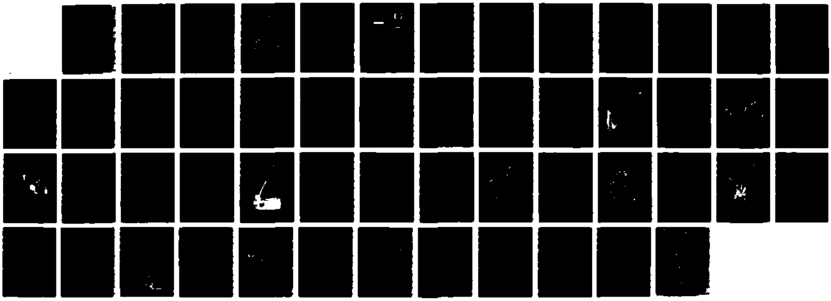

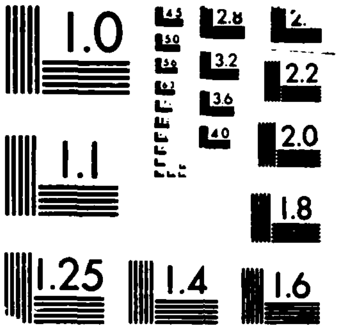
1.0
2.8
2.2
3.2
3.6
4.0
2.0
1.1
1.8
1.25
1.4
1.6

8.4.2 Components

Today the choice of automation systems is not so much whether to automate or not, but how much automation and what type or level complexity of automation (ref. 8.19).

In the past all remote controls were of the pneumatic type. Recently a mix is being used in control and automation systems, pneumatics for remote control of the engine and an electrical/electronic mix for alarm and monitoring functions. Most Automation and Control system manufacturers use pneumatics because of their reliability, insensitivity to heat and vibration and because the engine manufacturers who offer engine control packages have based their on-engine systems on pneumatics.

However, not all vessel operators belong to that school of thought, in particular those with Naval applications operating gas turbines. The British HM ships chose electronic controls on the grounds that they were cheaper, easier to maintain and more flexible than pneumatics (ref. 8.20).

The alarm and monitoring functions are almost exclusively electrical/electronic. Simple binary sensor systems are usually electrical fail-safe-open type of circuitry. Pressure and temperature points (such as oil temperature and pressure) which are monitored for out-of-limits functions are usually analog devices with self-checking circuitry or sampling interfaces with a computer-based system. Some systems, such as Racal-Decca's ISIS 250 and Tano's T-MAC have a modular design with local scanning units which can operate autonomously in case of system failure.

Microprocessors are being applied to specific subsystem control and monitoring. Prior to microprocessors, control equipment was selected from a wide range of very different devices including relay equipment, static logic equipment, programmable logic controllers, and analogue controllers in various forms. The introduction of the microprocessor has made control and monitoring systems more flexible and given the ability to integrate them with other systems. Typical microprocessor control loops are shown in Figure 8.2. In many alarm, monitoring and control systems, it is often desirable to have a mix of digital and analog inputs. This allows digital signals to be readily introduced into analog systems and vice versa. A microprocessor such as Tano's T-MAC will provide central engine room monitoring, alarm and control, throttle control, boiler combustion, feed water control and burner management (on steamships), and automatic bell loggers that record conditions and changes by time.

Microcomputers with multiplexed transmissions are in use on some of the newer, more advanced commercial ships. The use of multiplexing through a single cable becomes particularly attractive when the costs of routing a cable for every point monitored are considered.

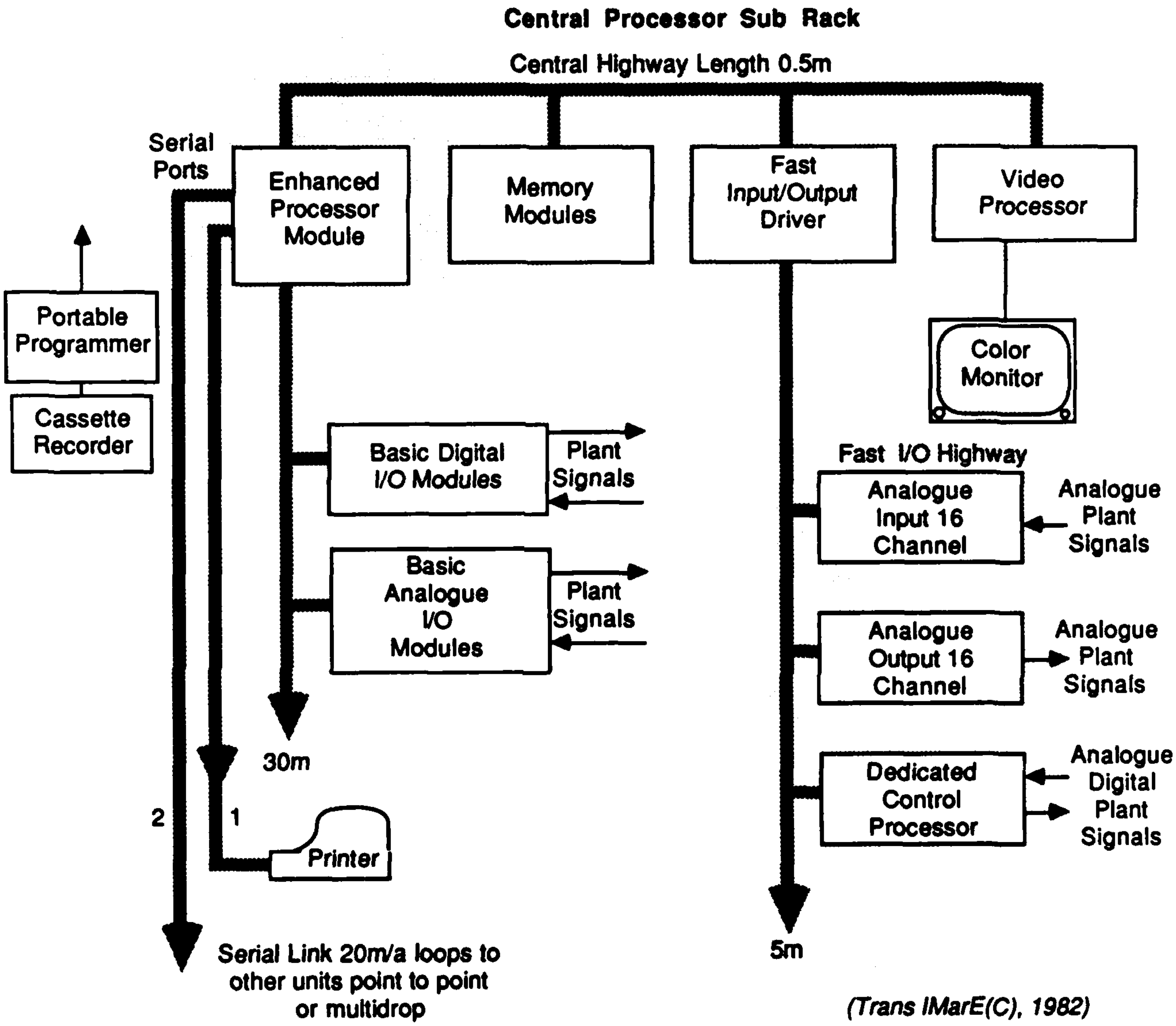

FIGURE 8.2 Typical microprocessor system hardware configuration

Video display units (VDU) for alarm and monitoring are becoming more commonplace. VDUs are more compact than light boards and are used by more aggressive commercial ship owners to maximize the amount of system information at minimum cost per point monitored. A typical distributed processing configuration for an automated monitoring system is shown in Figure 8.3.

Data loggers have also been introduced with microprocessors. Data loggers are used in an attempt to establish trends in wear and tear and set maintenance standards.

8.4.3 Future Trends

In addition to just monitoring engine characteristics, the monitored signals will be used in optimizing engine efficiency and life. Areas of interest have been engine rpm and propeller blade angle for optimal load conditions, engine injection timing for optimum operating economy, and electronic governors for limiting mechanical wear.

As ship automation increases, so does the need for skilled people to run the automated equipment (ref. 8.21). While overall personnel requirements may drop, the level of skill required of the personnel will rise. It is important to design the equipment such that it is usable, and the operator is trained for its use (ref. 8.22). If these factors are ignored it would result in disaster. Much more thought, planning, analysis and simulation should be going into automation.

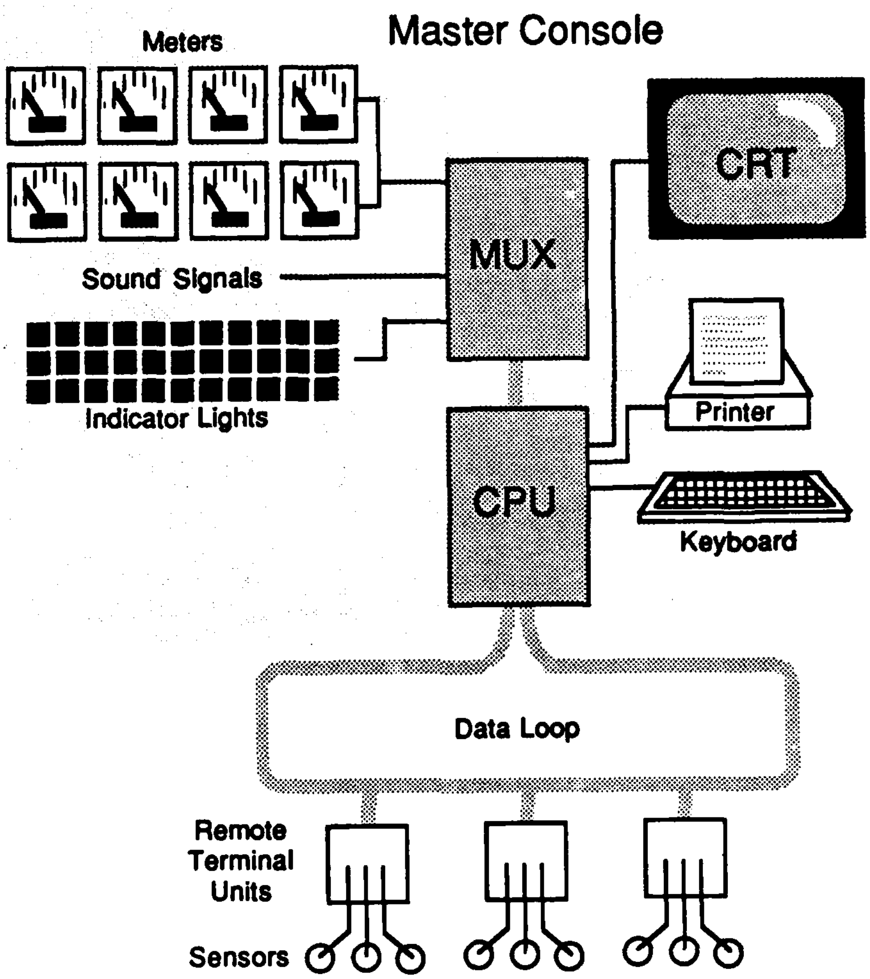

FIGURE 8.3 Typical Distributed Processing Configuration

8.4.4 Experiences

A study of the Automation of British Naval Ships reached several important conclusions (ref. 8.20):

- The initial acquisition cost of the controls for full automatic option was some 300% of the costs for the non-automated option.

- Automation at the lowest levels gives rise to the biggest savings.

- Generally surveillance equipment costs are greater than control equipment costs.

A reliability analysis of large commercial vessel engine room automation systems (ref. 8.23) revealed that sensors caused most of the problems, and that:

- Improving the control system quality by using military grade parts would decrease the basic failure rates by 53% (predicted).

- Increasing operating temperature from 35 to 50 deg. C would increase basic failure rates by 22% (predicted).

- Of three ships, the most complicated automation system has the highest predicted basic failure rate.

There is a group of common themes in most papers when discussing the failure of automation systems, or the reason for their lack of acceptance (ref. 8.24). These reasons were:

- -Systems were unnecessarily sophisticated and complicated
- -Systems required higher skill levels than available
- -Manual backup capability inadequate
- -Manpower reduction not as great as predicted
- -Insufficient shore support

However, where these pitfalls were avoided, the success was resounding.

8.4.5 Conclusions

The trend is towards the highly automated engine room, with control and monitoring functions. The main benefits from this type of automation are reduced personnel requirements and increased ship efficiency. However, this trend must be tempered by keeping these systems simple to operate and maintain.

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APPENDIX A

NOTES ON SOURCES

In gathering materials for this report, a combination of automated searching using computerized databases and traditional library searches was used. Computerized searches are very thorough and efficient, but exclude many specialized journals and books. Obtaining this material plus manufacturers' advertisements and literature made less sophisticated, but no less thorough, library work necessary.

The technology survey was initiated by contacting NERAC to do a search of the databases available to them on-line in all eight technology areas. The search included appropriate key words suggested by R&D Center personnel and inferred by the information specialists at NERAC from information provided about each subject area. The databases searched by NERAC for this study were (not all were searched for each topic):

NTIS/DTIC Government Technical Information

Engineering Index

ISMEC (Mechanical Engineering)

Oceanic Abstracts

EIM (Engineering Meetings)

NASA

INSPEC

DOE

Upon receipt of the lists of citations in each area, the researcher for that area reviewed the abstracts to see which ones were worthy of inclusion in the study, and obtained a copy of the article or book, from NTIS (for government publications), the U.S. Coast Guard Academy Library, British Ship Research Abstracts (for foreign periodicals) or through Interlibrary Loan.

The second line of inquiry centered on published indexes specifically marine oriented. These sources were examined to find articles of interest:

Society of Naval Architects and Marine Engineers
Publications Index

American Society of Naval Engineers Index

British Ship Research Abstracts

Marine Research Information Service Abstracts

Naval Abstracts

Finally, the following periodicals which were felt to be especially valuable to the survey (based on the researcher's previous familiarity with them), but which were not well represented in the previously searched abstracts and indexes, were searched issue-by-issue.

International Shipbuilding Progress

Marine Engineer's Review

Marine Technology Society Journal

Maritime Reporter

The Motorship

Ocean Engineering

Ocean Industry

Small Ships

Workboat

Copies of all citations mentioned in this report are on file at the R&D Center.

APPENDIX B

AIDS TO NAVIGATION; FOREIGN PRACTICES

FOREIGN PRACTICES QUESTIONNAIRE

U.S. COAST GUARD

1.0 Aids to Navigation Overview

The intent of this se:tion is to characterize your aids to navigation system in size and scope of responsibility. It will allow us to understand your system and to interpret the information in the following sections.

1. What is the jurisdiction of your authority?
2. How many buoys, lighthouses and structures are there?
3. Is the system operated by civilians, military or contractors?
4. What other missions are performed by buoy tenders (e.g., search and rescue, towing, fisheries patrol, environmental monitoring/clean-up)?

2.0 Aids to Navigation System Description

The intent of this section is to describe the buoys, lighthouses and structures that your authority maintains as well as the vessels and other craft used to maintain them.

Offshore buoy tenders:

1. How many offshore buoy tenders are operating?
2. Are the buoys worked fore or aft of the bridge?
3. What is the area of the buoy deck?
4. How many and what size buoys and sinkers can be carried on the buoy tender?
5. What is the size of the buoy tender crew?
   Is the crew that operates the ship the same crew that works buoys?

Buoys:

6. How many buoys are:

   a. Lighted and unlighted?
   b. In exposed waters and sheltered waters?

7. What is the power source for lighted buoys?
8. How many aids are solar, wind or wave powered?
9. What are the general capabilities of the buoy handling equipment ?

Lighthouses and Structures:

10. How many lighthouses and structures are serviced by offshore buoy tenders?

11. Are other vessels or craft used to assist buoy tenders in servicing the lighthouses and structures (e.g., helicopters, work boats, etc.).

12. Are there any unconventional structures or buoys serviced by buoy tenders?

13. Do these impose different servicing requirements on the buoy tenders?

3.0 Aids to Navigation System Operation and Practices

This section describes the policies and practices that your authority has developed to control how the individual components of the aids to navigation system act together to provide a complete system for the mariner. The emphasis is on how offshore buoy tenders are used within the system.

1. How often is a buoy serviced by an offshore buoy tender?

2. Is most of the buoy maintenance performed on scene or is the buoy taken to a shore facility for most maintenance functions?

3. If applicable, what kind of maintenance is performed by the tender crew while on scene?

4. Are the buoys lifted on board on each visit?

5. How often are the moorings lifted and inspected?

6. Do solar/wind/wave-powered buoys require special servicing practices?

7. Do solar/wind/wave-powered buoys have extended or reduced servicing intervals as compared with conventionally powered buoys?

8. How many buoys are serviced by a typical offshore buoy tender?

9. What is the operating radius of the typical offshore buoy tender?

10. For an offshore buoy tender, how many days per year are planned for:

a. Aids to navigation operations?
b. Other missions (e.g., search and rescue, pollution control).
c. Ship maintenance?
d. Training?

11. What system is used to position the buoy (e.g., sextant, electronic)?

12. What is the accuracy of the positioning system?

13. Does the authority have a buoy position reliability standard that is advertised to the mariner?

14. Does the mariner have the ability to recover damages from the authority if loss or damages are attributed to negligence on the part of the authority?

15. How many lighthouses are serviced by a typical offshore buoy tender?

16. How frequently are lighthouses replenished by a buoy tender?

17. What is the role of smaller service craft?

a. Assist larger buoy tenders?
b. Service smaller buoys not served by the buoy tenders?
c. Service discrepancies?

18. Are there any plans to modify the aids to navigation system, either the equipment or the operation, to optimize the system or to incorporate new technology?

19. Has your authority recently performed any studies to optimize the use of your present buoy tenders?

20. What are the primary factors that resulted in the size, configuration and use of your current buoy tenders?

APPENDIX C

NOTES ON PRICE COMPARISON METHODOLOGY

Since the information gathered for this report spans roughly the last five years in the literature, with sources both in the U.S. and several foreign countries, some method was needed to compare cost information on ships and their equipment which took care of inflation, fluctuations in foreign exchange values, and the market factors which influence ship production.

First it should be noted that cost information is scarce in the shipbuilding industry. Perhaps in no other industry is there so much secrecy concerning contract prices. In other aspects of shipping, such as charter fixtures and freight rates, it is routine to quote current market prices, but in shipbuilding it is the exception rather than the rule.

The second difficulty is the numerous market distortions present in the industry. It is hard to establish a "Free Market Price" when governments erratically establish and abolish subsidy programs for their shipbuilders and their shipping companies. Due to the intense competition in the industry, in order to preserve jobs and huge capital investments, subsidized prices must be met or bettered by other yards to keep their order books full. Thus social and political considerations tend to influence what would otherwise be straightforward economic calculations (admittedly this happens in many industries, but it is especially acute in shipbuilding).

Finally, how foreign quotes are converted to U.S. dollars is important. Shipbuilding prices for export in general do not precisely follow the exchange rate fluctuations between the currencies of the contracting nations. The first reason for this is that the contract for export is generally in dollars in the first place, even, for example, on a contract between an Indian shipping line and a Korean builder. Secondly, the supply of materials and machinery for shipbuilding is highly international, and most ships, particularly those built in the third world, have a high percentage of foreign-origin materials and components, bought at the lowest world price, using dollars. As one country's currency goes down, its exports get cheaper, and it conceivably sells more machinery to a country building a ship for export to a third country whose currency is rising, and so on. Thirdly, both shipbuilders and the buyers of ships use a variety of means of minimizing their exposure to currency fluctuations, such as "Currency Cocktails" or "Baskets" (accounts with various proportions of several currencies to average out fluctuations), and Currency Futures. Finally, some of the key shipbuilding countries, notably low cost producers like Korea, Yugoslavia, Taiwan and Hong Kong fix their currencies to the dollar, and since other shipbuilders must compete against them, this also tends to suppress fluctuations in world shipbuilding prices due to currency valuations. None of these mechanisms works perfectly, and there have been times when ship buyers have gotten very badly hurt by currency fluctuations, but in general, such fluctuations do not have a very strong correlation with world shipbuilding prices.

In particular, it was thought desirable to de-emphasize the rapid drop of the dollar with respect to the Yen during 1986.

The method adopted for foreign ships is to convert the price as quoted to US dollars using the exchange rate prevailing at the time of delivery, then using an index based on world shipbuilding costs compiled by "Fairplay Shipping Weekly" the costs are brought up to equivalent costs for January 1987. The costs in "Fairplay" up until January 1983 were estimated by considering the steelwork, main and auxiliary machinery, labor and overhead costs, with a 5% margin for profit on several hypothetical ships typical of types commonly used in world trade. From January 1983 on, the specifications of ships used were changed to reflect the changes in the shipping trade, and more importantly, instead of computing hypothetical ship costs, an average was made using actual ship contract data, which truly reflects ship prices, worldwide, with the effects of subsidies, etc, included. The actual index used in this study attempts at making a transition between these two methods. It uses primarily the container ship (since the high labor and machinery content of container ships resembles offshore work vessels) and the post 1983 5000 dwt general cargo vessel data (although still too large, the closest in size to the vessels under consideration), and uses 1976 as 100.

There are several problems with this method. The vessels considered by "Fairplay" are nowhere near representative of buoy tenders or the offshore work vessels considered in this study, in particular, the labor and machinery content of the "Fairplay" dry cargo, bulker and container ships are low. Also, the "Fairplay" costs prior to 1983 purposely do not include data from the United States, the COMECON countries or South Korea, since the prices quoted in those areas were felt to be either too high or far too low due to circumstances that have nothing to due with actual productivity levels. However, given the scope of this study, it is felt that these and other problems with the method are not crippling, and it allows comparisons to be made on a reasonable basis relatively easily.

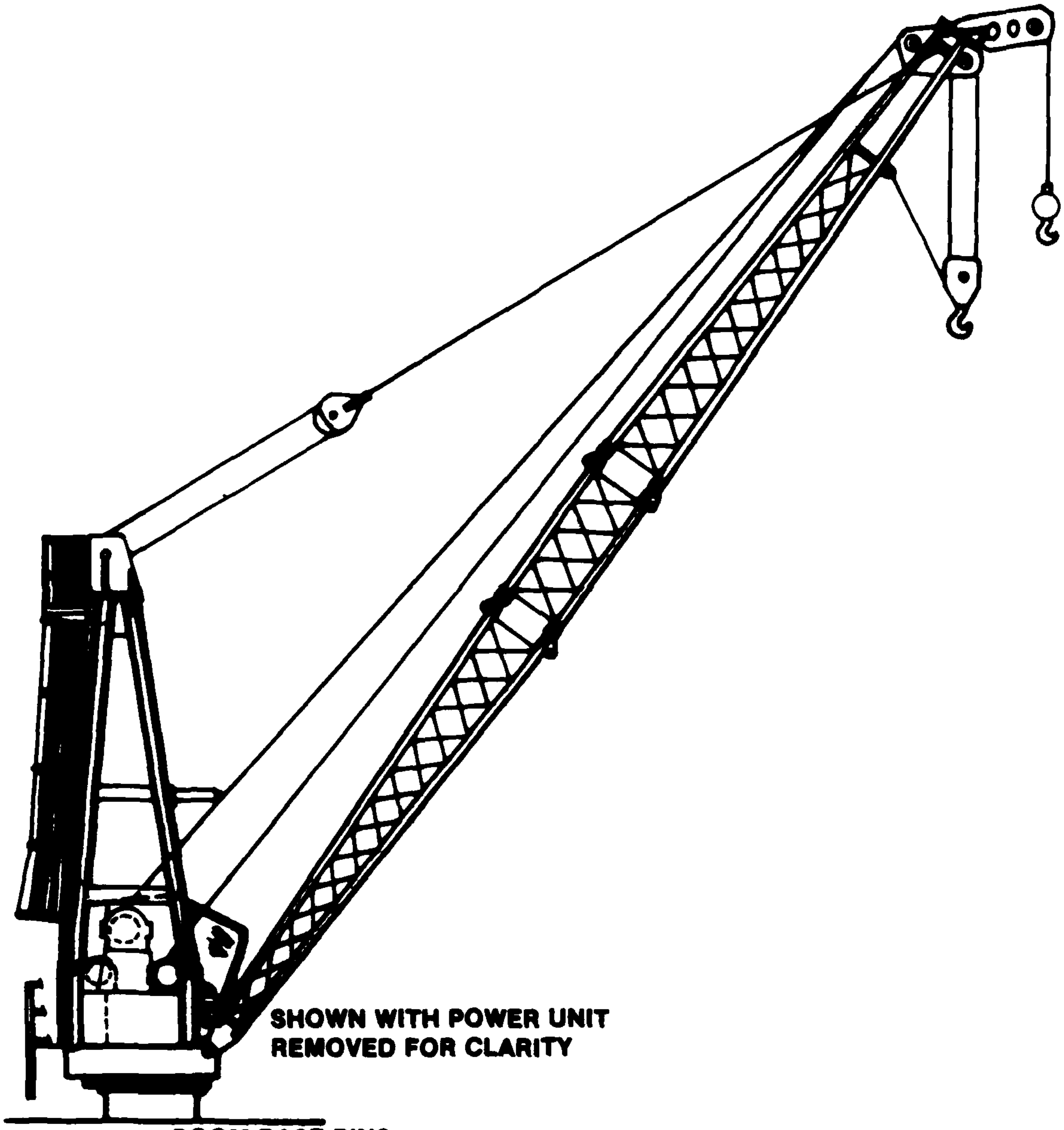
For U.S. built ships, the problem is a little easier since foreign currency exchange is avoided. Also, a different index is used to relate costs from year to year. The Shipbuilding Cost Index prepared by the Maritime Administration's Shipbuilding Cost and Production Office was used, modified so that 1976 is 100 to allow direct comparison with the "Fairplay" world market data. This index has its own limitations, in that it does not include profit or overhead, thus it may not reflect the true condition of the market when gains in productivity are being made, for example. Also, this index is more relevant to large ships than to small, and to large shipyards rather than small boatbuilding companies, a shortcoming it shares with the "Fairplay" data.

Market distortions are less of a problem for machinery costs, and there is less of a need to establish a "world price", since US manufacturers can often supply the needed equipment at competitive prices. Therefore, the foreign cost data was converted to U.S. dollars at the exchange rate prevailing at the time of the quote, then inflated using the Producer's Price Index for General Purpose Machinery, as compiled by the U.S. Bureau of Labor Statistics, once again re-valued to use 1976 as 100. Domestic costs were similarly treated without the foreign exchange calculation.

APPENDIX D

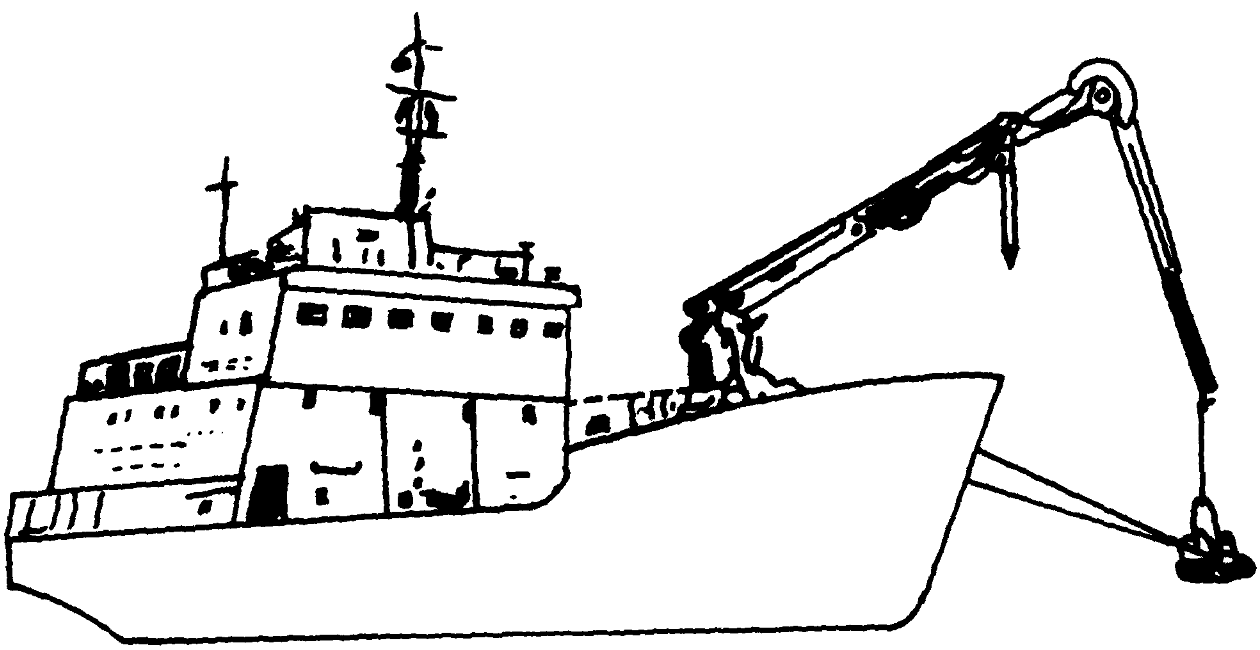
SUMMARY OF WEIGHT HANDLING EQUIPMENT AND MANUFACTURERS

MODEL	ML-2400 SEACRANE
TYPE	LATTICE BOOM CRANE
MANUFACTURER	MANITEX, INC.,
MANUFACTURER ADDRESS	4300 ACAPULCO AVENUE,
	MCALLEN, TEXAS 78503
	(512) 630-2690
LIFTING CAPACITY	5 TONS @ 120', 13 TONS @ 60',
	42 TONS @ 25'
OUTREACH	WORKING RADIUS: 120'
SPEED	
DIMENSIONS	MAX BOOM LENGTH: 116'5"
WEIGHT	28 TONS (APPROX)
DRIVE	HYDRAULIC (DIESEL OR ELECTRIC DRIVEN)
MOTION COMPENSATION	NONE
COST	BY QUOTATION
COMMENTS	



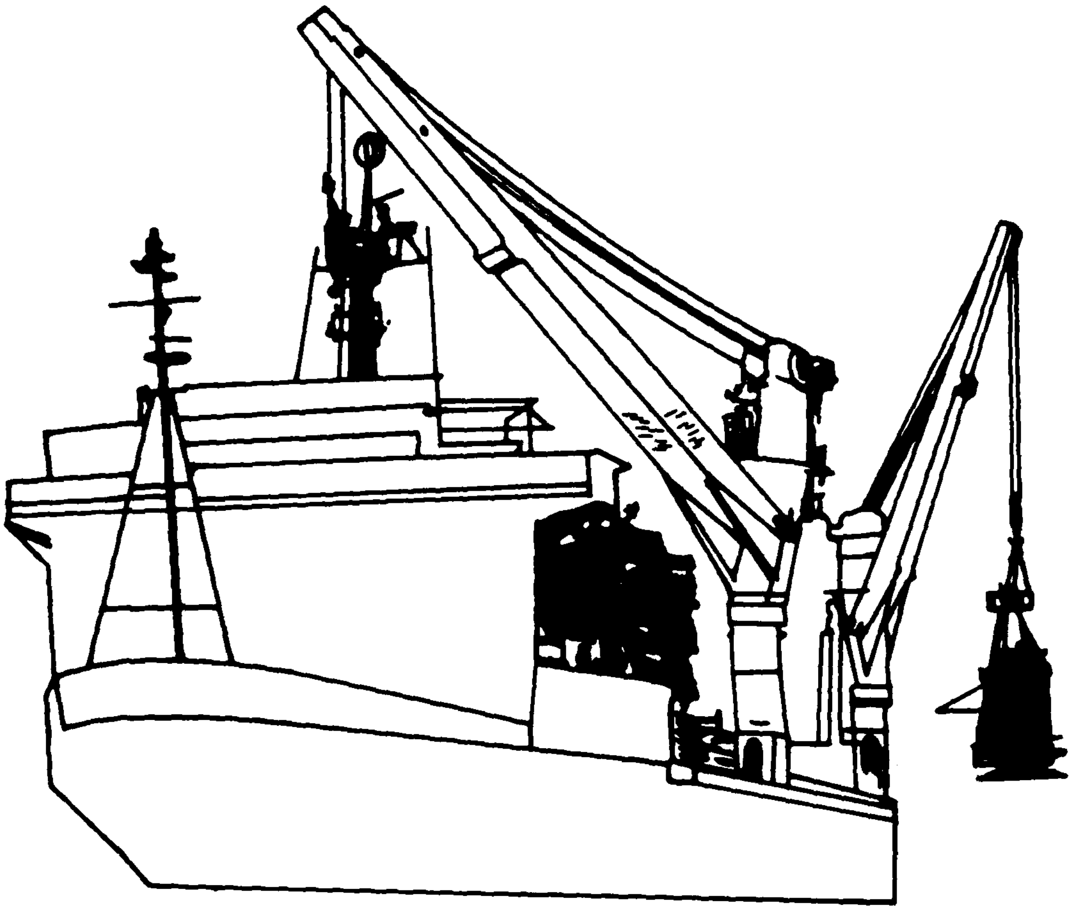
ML-2400
SEACRANE
MANITEX, INC.

MODEL	1102B
TYPE	KNUCKLE BOOM CRANE
MANUFACTURER	EFFER
MANUFACTURER ADDRESS	NORTH AMERICAN DISTRIBUTER
	SMITH BERGER MARINE, INC.
	SEATTLE, WA
LIFTING CAPACITY	35 TONS @ 11'6" OUTREACH
OUTREACH	8 TONS @ 45'11" OUTREACH
SPEED	
DIMENSIONS	BOOM LENGTH: 45'11"
WEIGHT	12 TONS W/O WINCH & POWER PACKAGE
DRIVE	HYDRAULIC
MOTION COMPENSATION	NONE
COST	BY QUOTATION
COMMENTS	



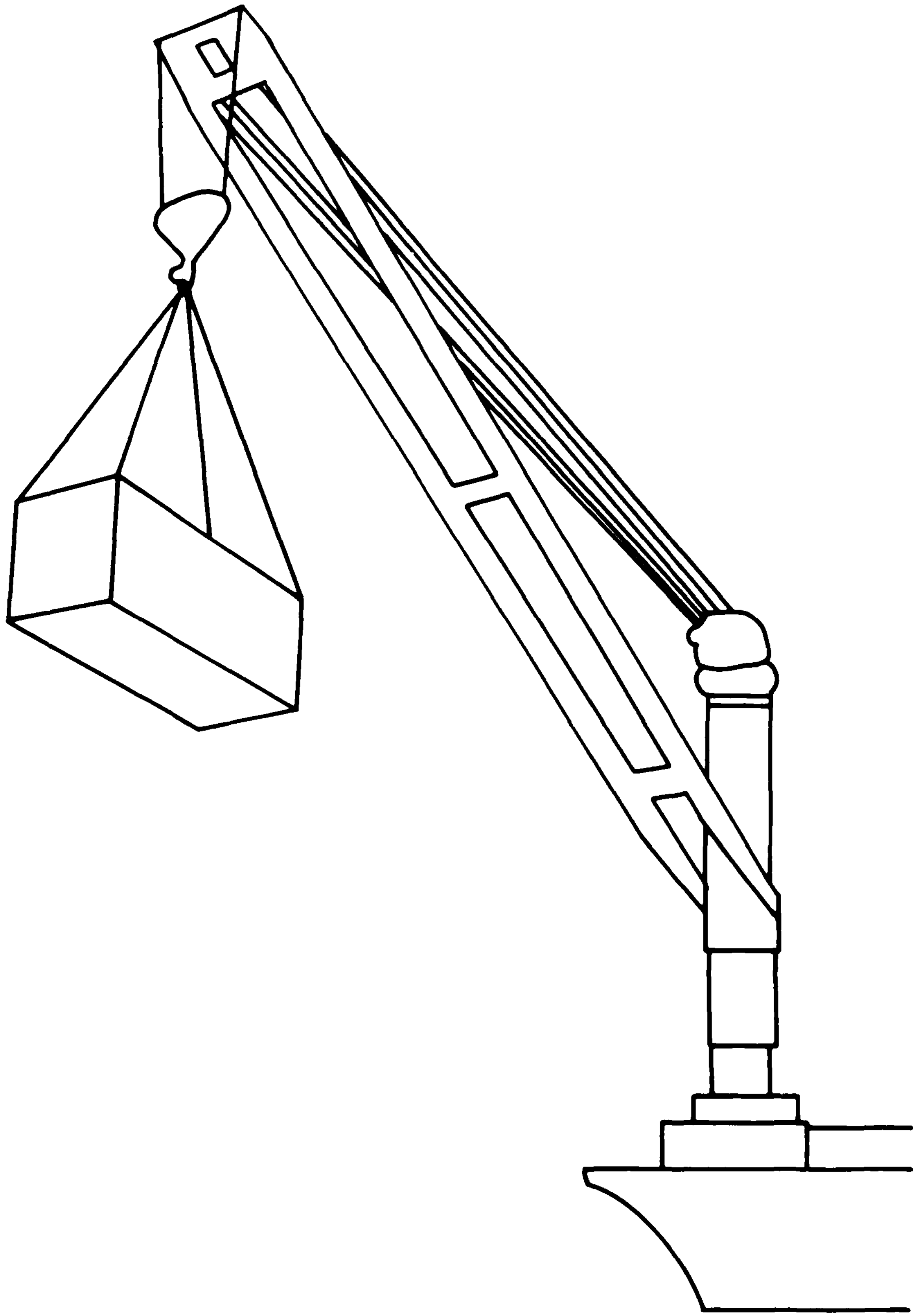
1102B
KNUCKLE BOOM CRANE
EFFER

MODEL	SLIM LINE
TYPE	CRANE
MANUFACTURER	LIEBHERR-WERK NENZING GES.MBH
MANUFACTURER ADDRESS	P.O. BOX 10, A-6710, NENZING
	AUSTRIA (05525) 2480-0
LIFTING CAPACITY	5-35 M.TONS
OUTREACH	MAX RADIUS: 72 FT
SPEED	
DIMENSIONS	MAX WIDTH: 8 FT
WEIGHT	
DRIVE	
MOTION COMPENSATION	NONE
COST	
COMMENTS	DOUBLE-ACTING LUFFING CYLINDERS



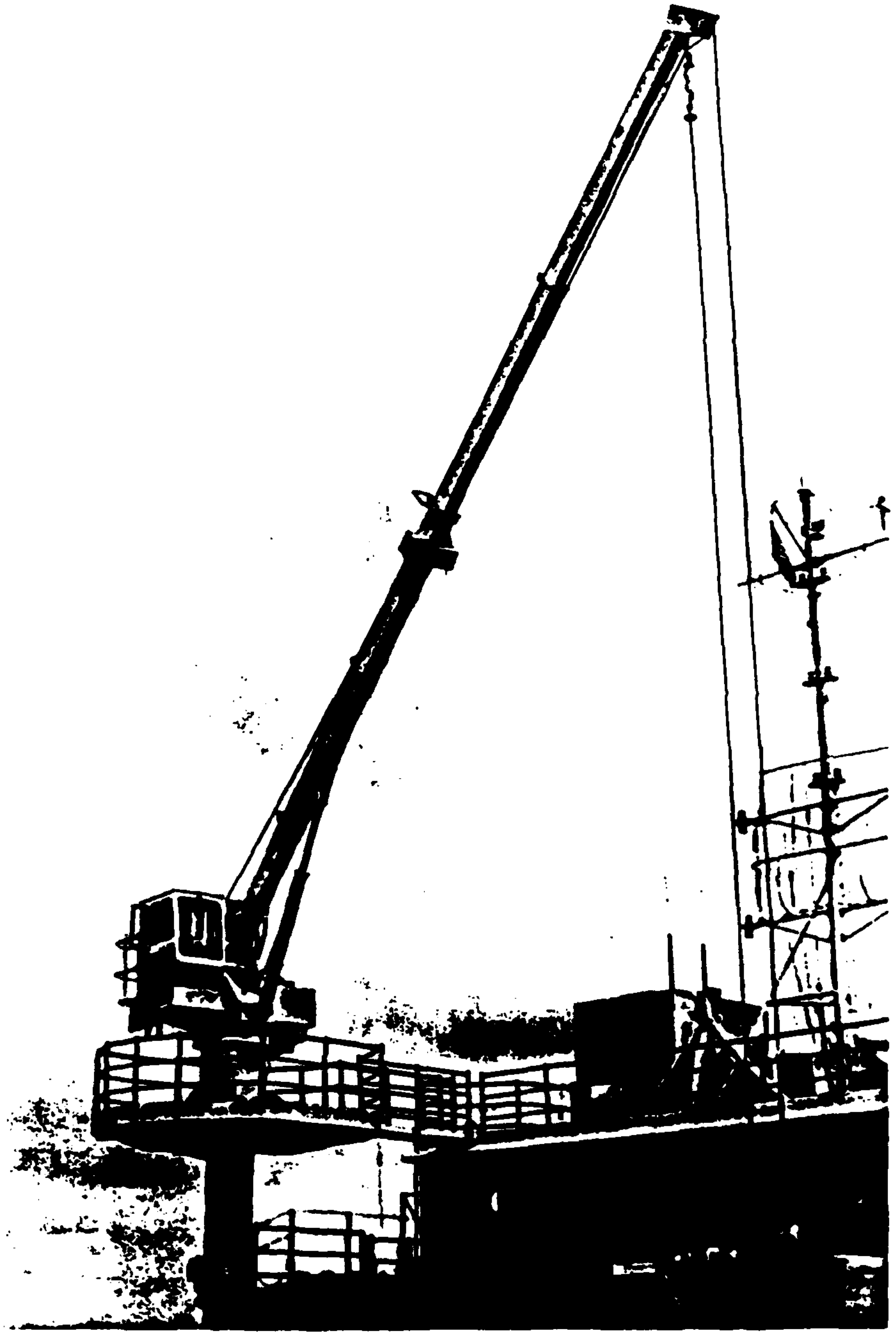
SLIM LINE CRANE
LIEBHERR-WERK NENZING
GES. MBH

MODEL	SERIES E2H
TYPE	CRANE
MANUFACTURER	ORENSTEIN & KOPPEL
MANUFACTURER ADDRESS	LUBECK, GERMANY
LIFTING CAPACITY	36 M.T.
OUTREACH	23.5 M RADIUS MAX; MIN 2.4 M
SPEED	
DIMENSIONS	WIDTH: 2.4 M
WEIGHT	
DRIVE	HYDRAULIC (SELF-CONTAINED)/OPTIONAL
	ELECTRIC
MOTION COMPENSATION	NONE
COST	
COMMENTS	SMALL WIDTH



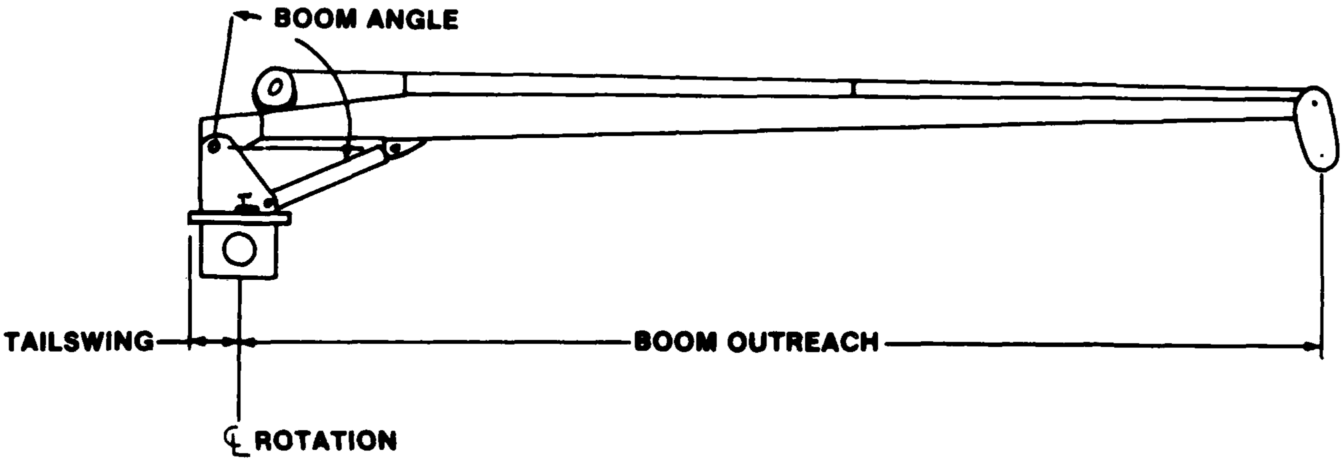
SERIES E2H CRANE
ORENSTEIN & KOPPEL

MODEL	TB SERIES
TYPE	TELESCOPING BOOM CRANE
MANUFACTURER	ALLIED MARINE CRANE
MANUFACTURER ADDRESS	13985 SW TOALATIN-SHERWOOD ROAD
	SHERWOOD, OREGON 97140
	(503) 625-2560
LIFTING CAPACITY	2-75 TONS @ 10 FOOT RADIUS
OUTREACH	20-100 FEET
SPEED	
DIMENSIONS	
WEIGHT	
DRIVE	HYDRAULIC
MOTION COMPENSATION	NONE
COST	BY QUOTATION
COMMENTS	



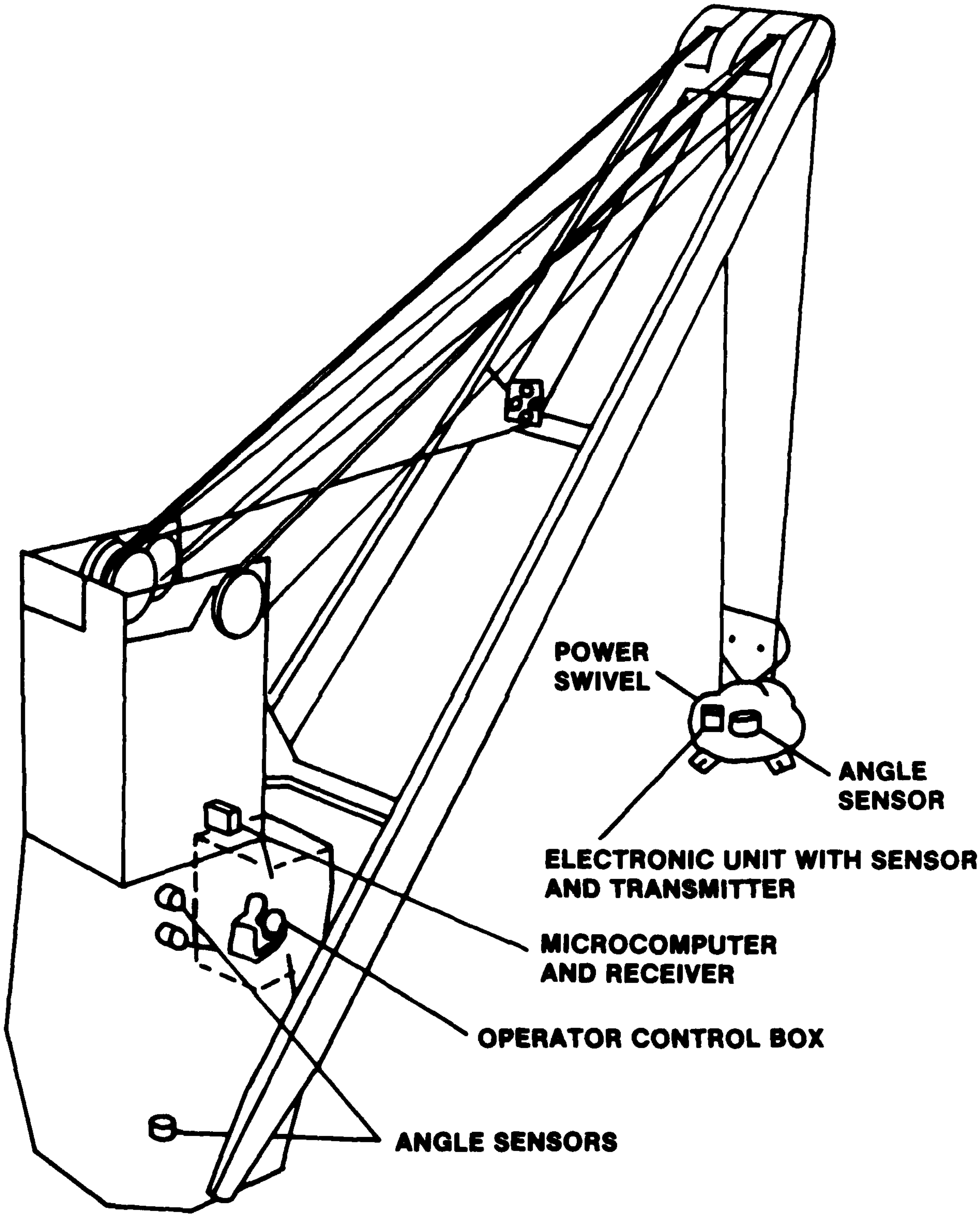
TB SERIES
TELESCOPING BOOM CRANE
ALLIED MARINE CRANE

MODEL	SB 50
TYPE	CRANE
MANUFACTURER	APPLETON MARINE
MANUFACTURER ADDRESS	P.O. BOX 2339
	APPLETON, WI 54913
	(414) 733-7361
LIFTING CAPACITY	36 TONS (SHORT) @ 5 FT OUTREACH;
	16 TONS (SHORT) @ 20 FT OUTREACH
OUTREACH	
SPEED	
DIMENSIONS	11'7" HIGH BASE X 4'2" DIA
WEIGHT	
DRIVE	HYDRAULIC OR ELECTRO-HYDRAULIC
MOTION COMPENSATION	NONE
COST	BY QUOTATION
COMMENTS	CONTROL MAY BE PLATFORM OR REMOTE



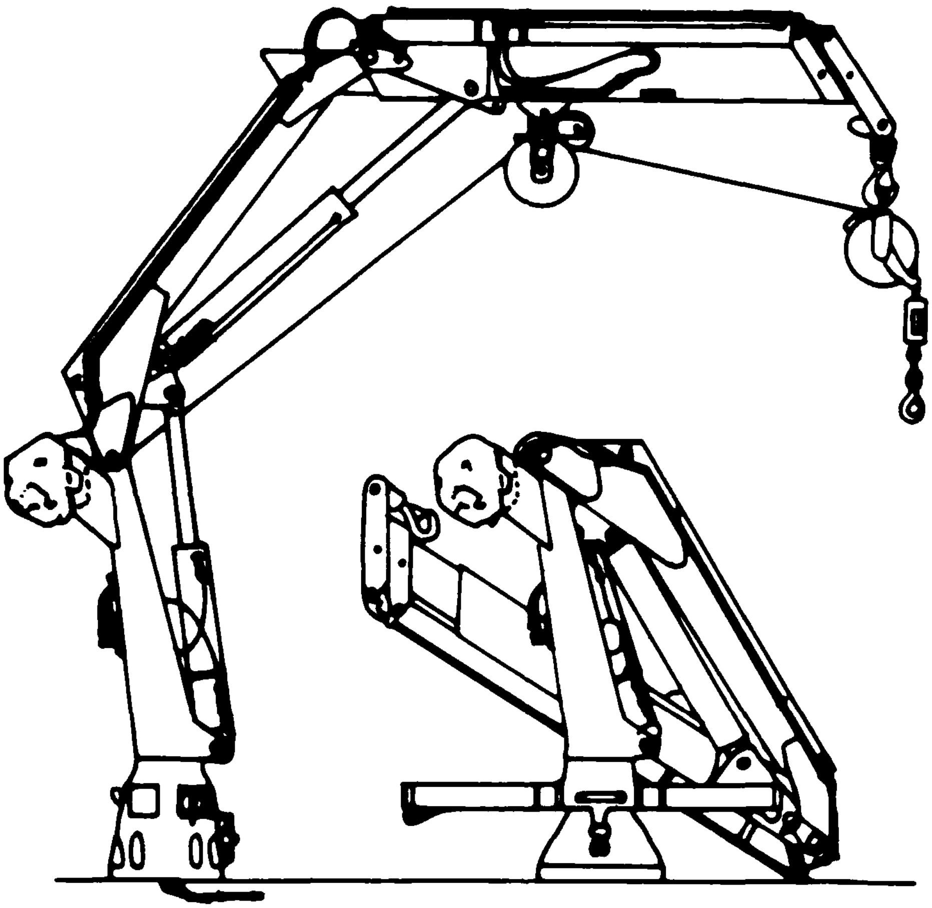
SB 50
FIXED JIB CRANE
APPLETON MARINE

MODEL	G-2
TYPE	CRANE
MANUFACTURER	AB HAGGLUNDS & SONER
MANUFACTURER ADDRESS	MARINE DIVISION BOX 600
	S-89101 ORNSKOLDSVIK
	SWEEDEN (46 660 800 00)
LIFTING CAPACITY	25-50 M.T.
OUTREACH	MIN RADIUS: 20 M; MAX RADIUS: 32 M
SPEED	LOW: 17-25 M/MIN; HIGH: 34-50 M/MIN
DIMENSIONS	
WEIGHT	
DRIVE	LOW SPEED, HIGH TORQUE HYDRAULIC
MOTION COMPENSATION	*STEADY LINE = (CARGO ALIGNING)*
	SWING DEFEATER = ANTI-SWING
COST	
COMMENTS	LOW TEMP (-58°F) OPERATION



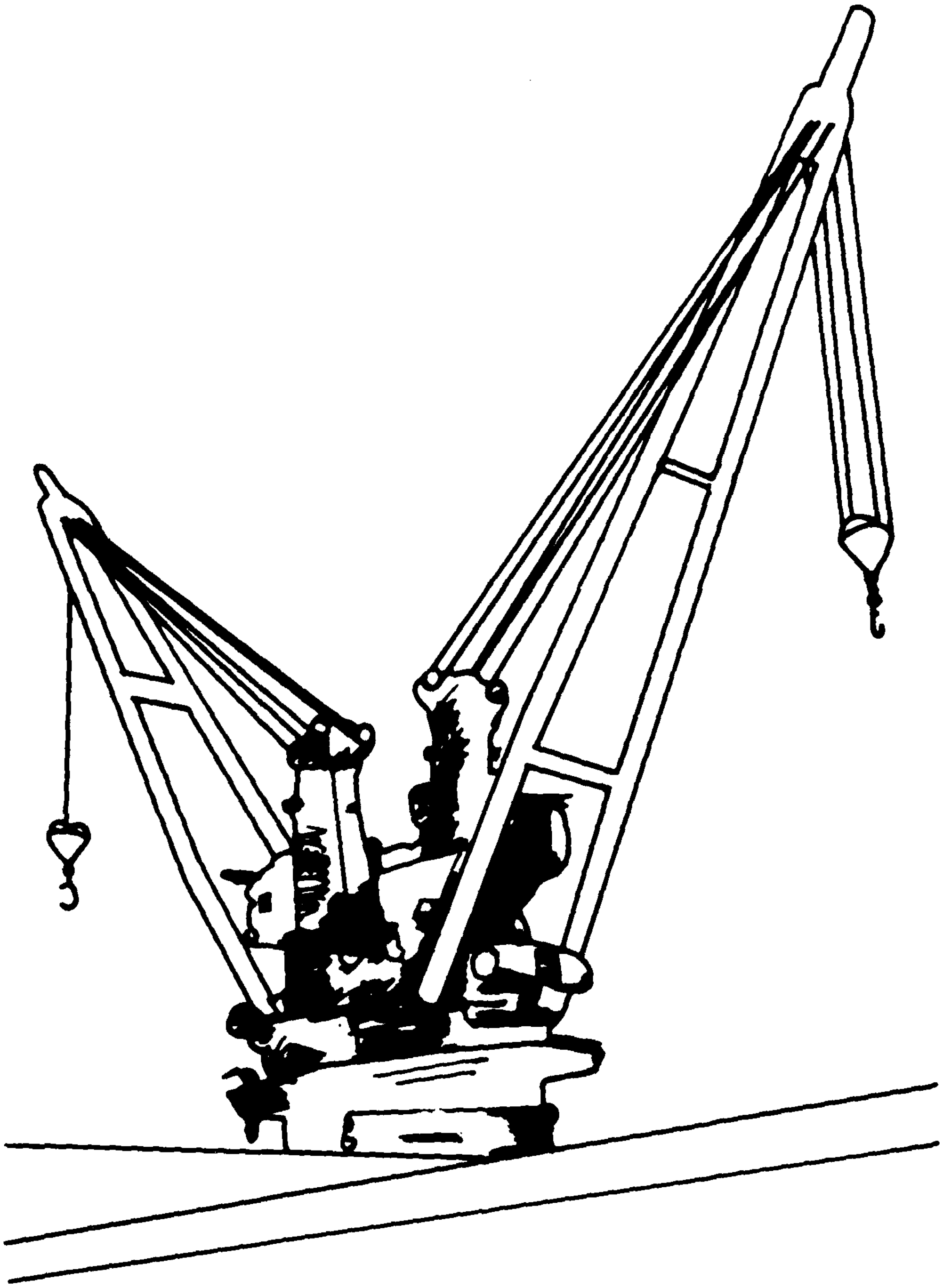
G-2 CRANE
AB HÄGGLUNDS & SÖNER

MODEL	110
TYPE	KNUCKLE BOOM CRANE
MANUFACTURER	HIAB SEACRANE
MANUFACTURER ADDRESS	HIAB CRANES & LOADERS INC.
	AIRPORT IND. PARK, 258 QUIGLEY AVE.
	NEW CASTLE, DE 19720 (302) 328-5100
LIFTING CAPACITY	
OUTREACH	
SPEED	
DIMENSIONS	
WEIGHT	
DRIVE	HYDRAULIC
MOTION COMPENSATION	NONE
COST	
COMMENTS	



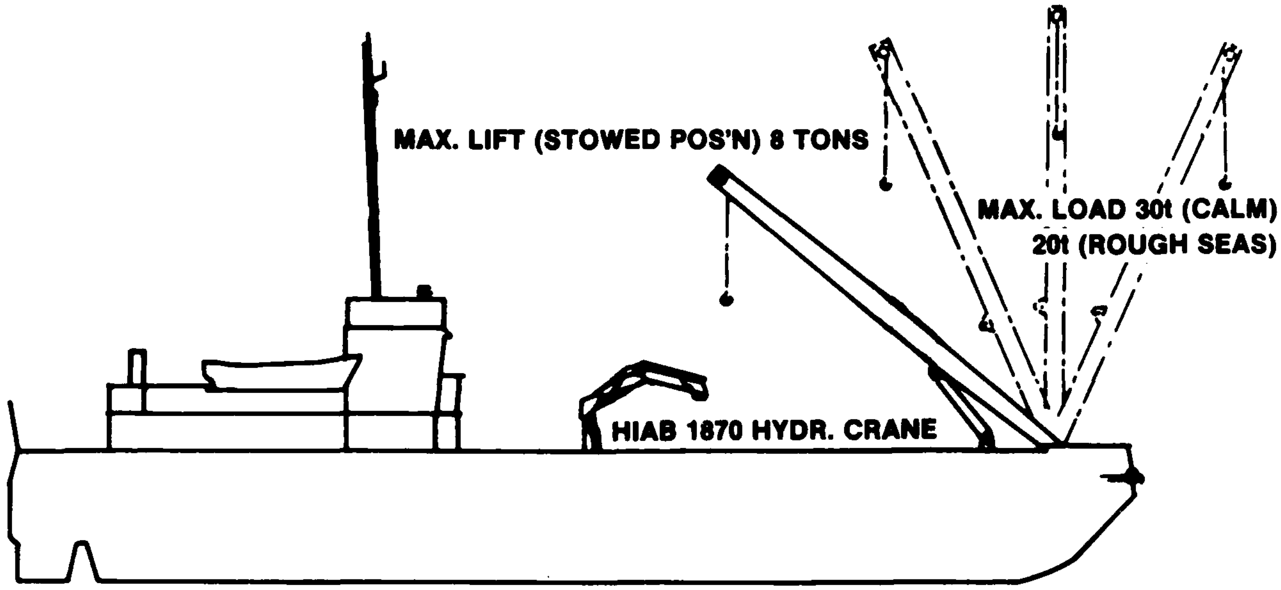
MODEL 110
KNUCKLE BOOM CRANE

MODEL	CHALLENGER
TYPE	TWIN CRANE
MANUFACTURER	NEI CLARKE CHAPMAN
MANUFACTURER ADDRESS	UNITED KINGDOM
LIFTING CAPACITY	40 M.T.
OUTREACH	
SPEED	
DIMENSIONS	
WEIGHT	
DRIVE	ELECTRIC W/ELECTRONIC CONTROL SYSTEM
MOTION COMPENSATION	NONE
COST	
COMMENTS	AUTO-SPOTTING OPTION



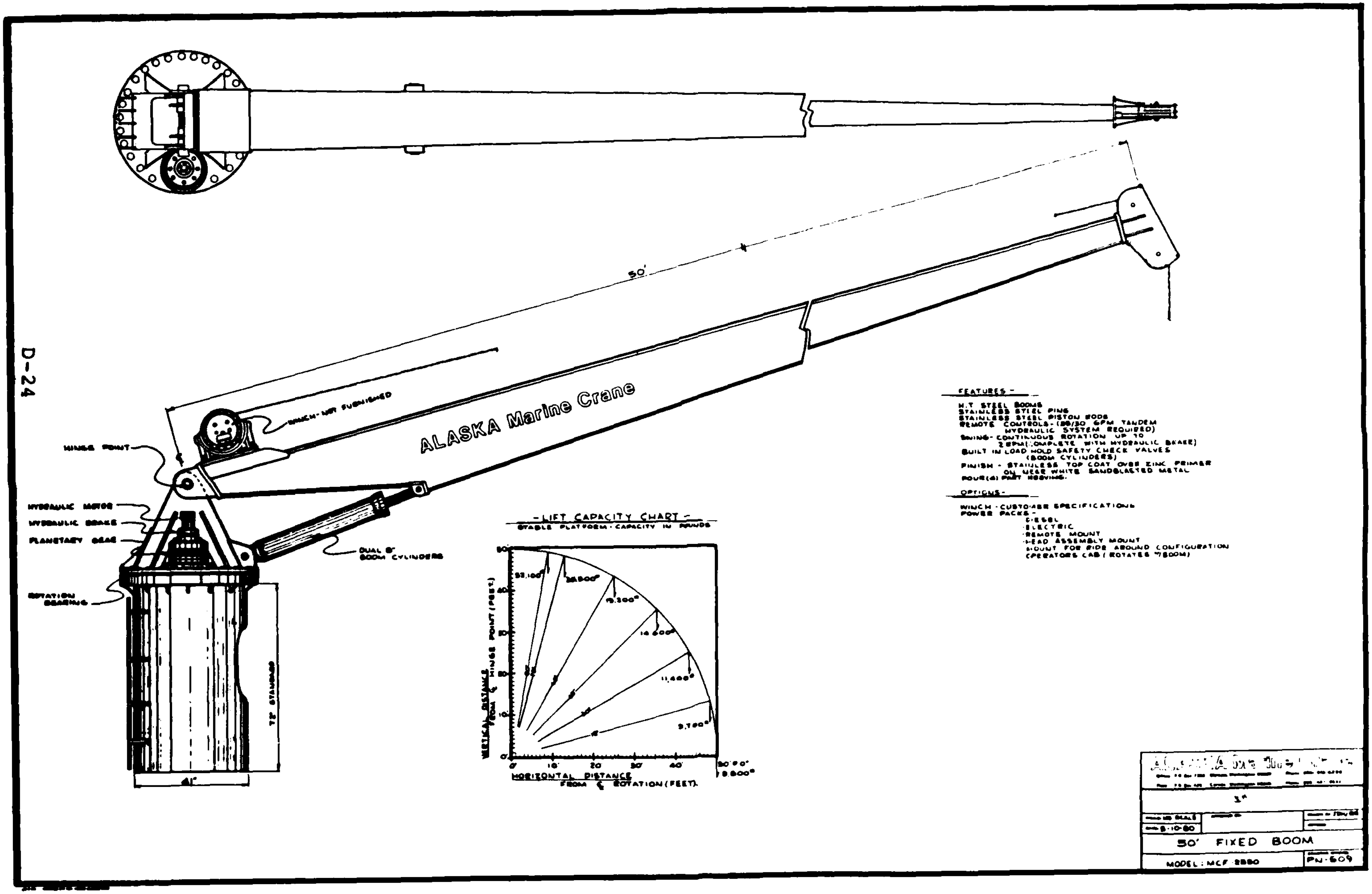
CHALLENGER SERIES
TWIN CRANE
NEI CLARK CHAPMAN

MODEL	
TYPE	A-FRAME GANTRY
MANUFACTURER	WELIN DAVIT & ENG.
MANUFACTURER ADDRESS	UNITED KINGDOM
LIFTING CAPACITY	30 M.T. SWL (CALM) 20 M.T. (ROUGH)
OUTREACH	
SPEED	
DIMENSIONS	
WEIGHT	
DRIVE	HYDRAULIC
MOTION COMPENSATION	
COST	
COMMENTS	ABLE TO LIFT 9M HIGH BUOYS

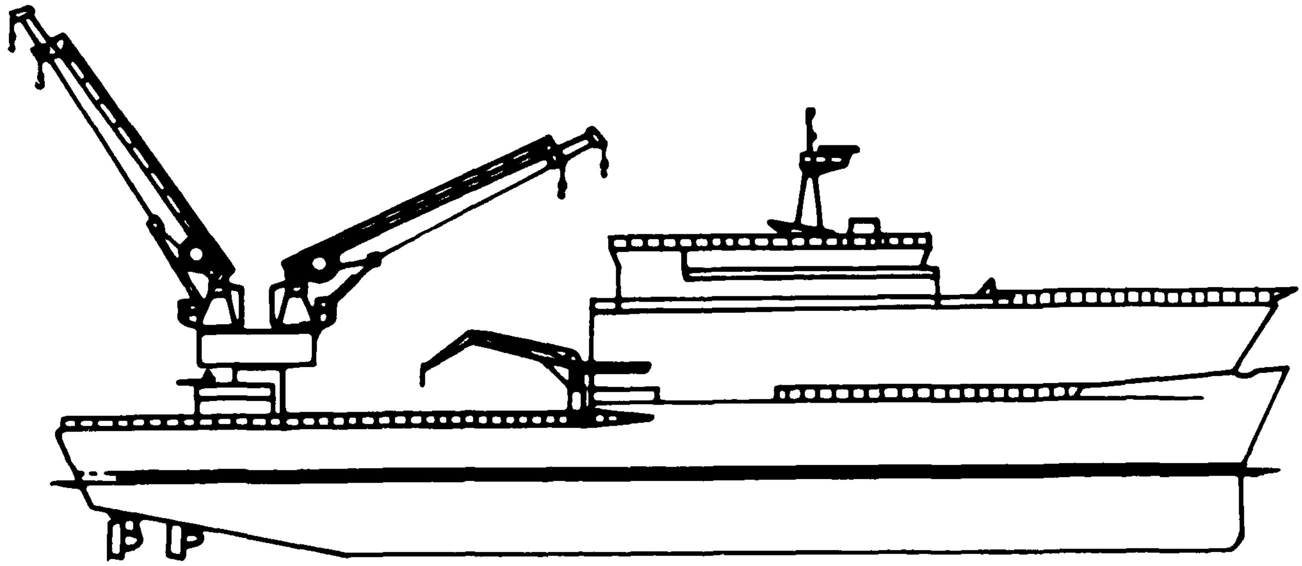


A-FRAME GANTRY
WELIN DAVIT & ENG.

MODEL	MCF 2550
TYPE	BOOM CRANE
MANUFACTURER	ALASKA MARINE CRANE
MANUFACTURER ADDRESS	ALASKA-TICO MARINE CRANE
	4403 20TH STREET EAST
	FIFE, WA 98424 / (206) 922-2272
LIFTING CAPACITY	25 TON @ 10' RADIUS
OUTREACH	
SPEED	ROTATION @ 2 RPM
DIMENSIONS	BOOM LENGTH 60' OR 75'
WEIGHT	
DRIVE	HYDRAULIC (DIESEL OR ELECTRIC POWER)
MOTION COMPENSATION	NONE
COST	BY QUOTATION
COMMENTS	

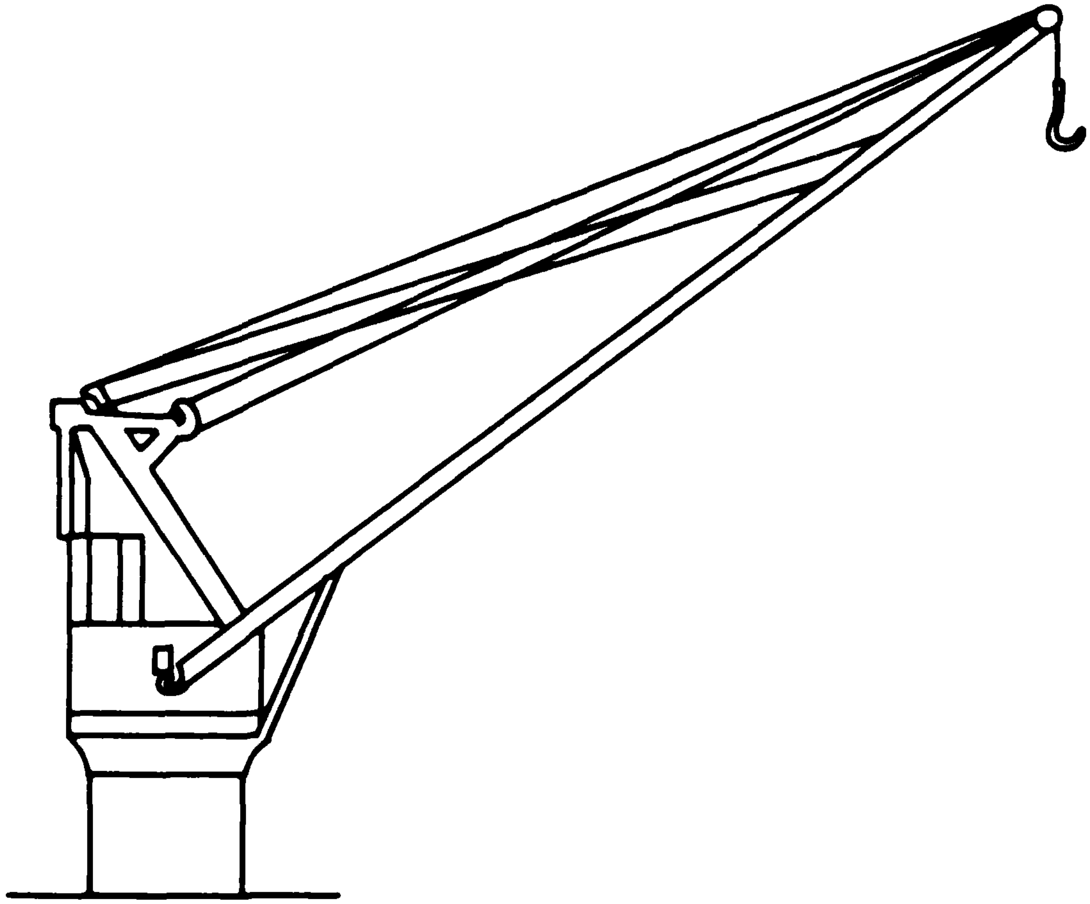

ALASKA Marine Crane
50'
WINCH-NOT FURNISHED
HINGE POINT
HYDRAULIC MOTOR
HYDRAULIC BRAKE
PLANETARY GEAR
ROTATION BEARING
DUAL 8" BOOM CYLINDERS
72" STANDARD
41"
-LIFT CAPACITY CHART-
STABLE PLATFORM - CAPACITY IN POUNDS
VERTICAL DISTANCE FROM ℄ HINGE POINT (FEET)
HORIZONTAL DISTANCE FROM ℄ ROTATION (FEET)
53,100#
28,500#
19,300#
14,600#
11,400#
9,700#
8,800#
FEATURES -
H.T. STEEL BOOMS
STAINLESS STEEL PINS
STAINLESS STEEL PISTON RODS
REMOTE CONTROLS - (28/30 GPM TANDEM HYDRAULIC SYSTEM REQUIRED)
SWING - CONTINUOUS ROTATION UP TO 2 RPM (COMPLETE WITH HYDRAULIC BRAKE)
BUILT IN LOAD HOLD SAFETY CHECK VALVES (BOOM CYLINDERS)
FINISH - STAINLESS TOP COAT OVER ZINC PRIMER ON NEAR WHITE SANDBLASTED METAL
FOUR(4) PART REEVING.
OPTIONS -
WINCH - CUSTOMER SPECIFICATIONS
POWER PACKS -
·DIESEL
·ELECTRIC
·REMOTE MOUNT
·HEAD ASSEMBLY MOUNT
·MOUNT FOR RIDE AROUND CONFIGURATION
OPERATORS CAB (ROTATES W/BOOM)
8-10-80
50' FIXED BOOM
MODEL: MCF 2880
PN-609

MODEL	
TYPE	TWIN CRANE
MANUFACTURER	MARITIME HYDRAULICS
MANUFACTURER ADDRESS	
LIFTING CAPACITY	130 M.TON
OUTREACH	
SPEED	
DIMENSIONS	
WEIGHT	
DRIVE	
MOTION COMPENSATION	COMPUTER CONTROLLED
	HEAVE-COMPENSATED ALLOW LIFTING IN
	SEA STATE 5
COST	
COMMENTS	INSTALLED ON STENA SEAWELL



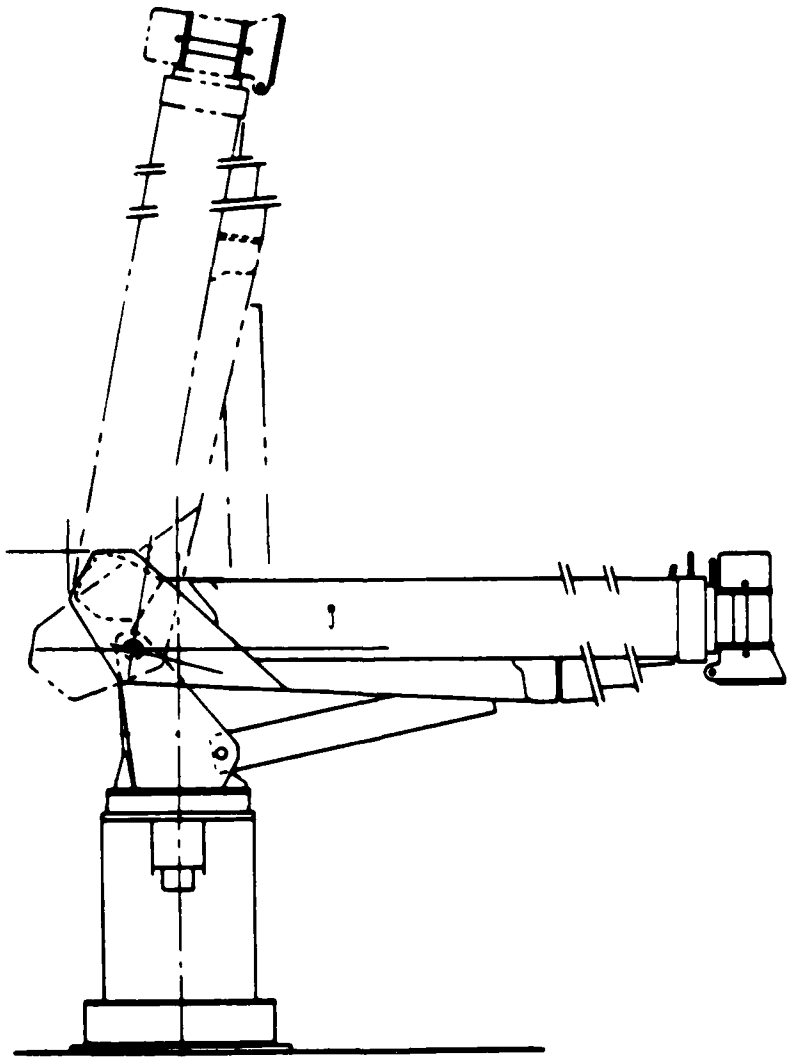
TWIN CRANE
MARITIME HYDRAULICS

MODEL	KE39-2 SUDOSTROYENIYE
TYPE	CRANE
MANUFACTURER	SUDOSTROYENIYE
MANUFACTURER ADDRESS	USSR
LIFTING CAPACITY	16 M.T.
OUTREACH	20 M
SPEED	CARO HOIST: 12 M/MIN; BOOM SWING: 10.5 M/MIN; TURNING: 0.5 RPM
DIMENSIONS	
WEIGHT	50 T
DRIVE	ELECTRIC
MOTION COMPENSATION	NONE
COST	
COMMENTS	ROTATION ANGLE: 335°



KE39-2
SUDOSTROYENIYE

MODEL	MARINE 800 SERIES
TYPE	TELESCOPING CRANE
MANUFACTURER	NATIONAL CRANE
MANUFACTURER ADDRESS	11200 NORTH 148TH STREET
	WAVERLY, NE 68462
	(402) 786-2240
LIFTING CAPACITY	10 TONS @ 10 FT OUTREACH
	17 TONS @ 22 FT OUTREACH
OUTREACH	
SPEED	
DIMENSIONS	BOOM LENGTH = 75 FT (MAX)
WEIGHT	
DRIVE	
MOTION COMPENSATION	NONE
COST	QUOTATION
COMMENTS	



MARINE 800 SERIES
TELESCOPING CRANE
NATIONAL CRANE

MODEL	SPEEDCRANE
TYPE	SWINGING DERRICK
MANUFACTURER	JOHN HASTIE (NOW BROWN BROS., LTD)
MANUFACTURER ADDRESS	BROUGHTON ROAD EDINBOROUGH, SCOTLAND EH7 4LS TEL: (011) 031 557 2008
LIFTING CAPACITY	RANGE 20-35 TONS
OUTREACH	17 M MAX RADIUS 140° SLEWING ANGLE
SPEED	LIFTING: 25 M/MIN @ FULL LOAD; 50 M/MIN @ LIGHT HOOK
DIMENSIONS	JIB LENGTH: 17.5 M
WEIGHT	
DRIVE	ELECTRIC WINCHES (MAY BE WARD LEONARD TYPE)
MOTION COMPENSATION	NONE
COST	
COMMENTS	DESIGNED TO OPERATE WITH VESSEL HEELED TO 5° AND ROLLING IN A SEAWAY TO A MAXIMUM ANGLE OF 10° EITHER SIDE OF THE VERTICAL, WITH AN ECCENTRIC LOAD OF 10° ON JIB HEAD IN ANY DIRECTION. OPTIONAL FORKED JIB HEAD FOR HANDLING HIGH FOCAL PLANE BUOYS.

MODEL	
TYPE	CRANE
MANUFACTURER	HYDROLIFT
MANUFACTURER ADDRESS	
LIFTING CAPACITY	6 M.TON @ 18 M OUTREACH
	14 M.TON @ 9 M OUTREACH
OUTREACH	MAX RADIUS: 9 M FOR 12 T S.W.L.
SPEED	
DIMENSIONS	
WEIGHT	
DRIVE	ELECTRO-HYDRAULIC
CONTROL	HEATED CAB / REMOTE DECK
MOTION COMPENSATION	NONE
COST	
COMMENTS	
OPERATION LIMITS	FULL LOAD UP TO 10° LIST

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