PRELIMINARY DESIGN CONSIDERATIONS OF PRESSURE VESSELS FOR A DEEP DIVING SUBMERSIBLE

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by

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ABSTRACT

Preliminary Design Considerations
of Pressure Vessels for a
Deep Diving Submersible

by

Thomas John Elliott, Jr.

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This report proposes a design methodology for weight limited deep submersibles from a vehicle designer's veiwpoint. Essential vehicle subsystems are considered, emphasizing those subsystems in which little data is available. The data from the subsystem descriptions is used for the establishment of the pressure vessel boundaires. An actual deep submersible design is paralled throughout the presentation of the methodology.

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Title: Visiting Lecturer

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I. Introduction:

During the 1960's, over thirty four submersibles were designed and commissioned, but only a few of those have accomplished their mission objectives. Many of these submersibles are not even operational any longer due to their inability to function as designed or accomplish cost effective tasks.

The author feels that the "All-Purpose Vehicle" philosophy is a major contributing factor to the above problem. The following paper is presented as a possible solution to the "All-Purpose Vechile" problem in that the mission requirements drive the design. In theory this may sound like a statement of the obvious, but in reality this is a major factor in the optimization of the design.

The purpose of this paper is to propose a design methodology, and to present state-of-the-art technology as it pertains to the various vehicle subsystems. An actual deep submergence vehicle, DSV, design will parallel the development of the design methodology. The essential vehicle subsystems are discussed emphasizing those areas which are limited in good design information. The other subsystems are reviewed and a list of references is presented which are felt, by the author, to be some of the best in the field of interest.

The title of the thesis states that the paper will discuss, prelininary design of submersible pressure vessels. At first it may seem logical to start designing the pressure vessels at the beginning of a design since they comprise a significant portion of the vehicles weight, but it will soon become evident that this is not feasible. In reality, the pressure vessel design and material selection are dependent upon the entire vehicle requirements and should come towards the end of the preliminary design stage.

II. Procedure

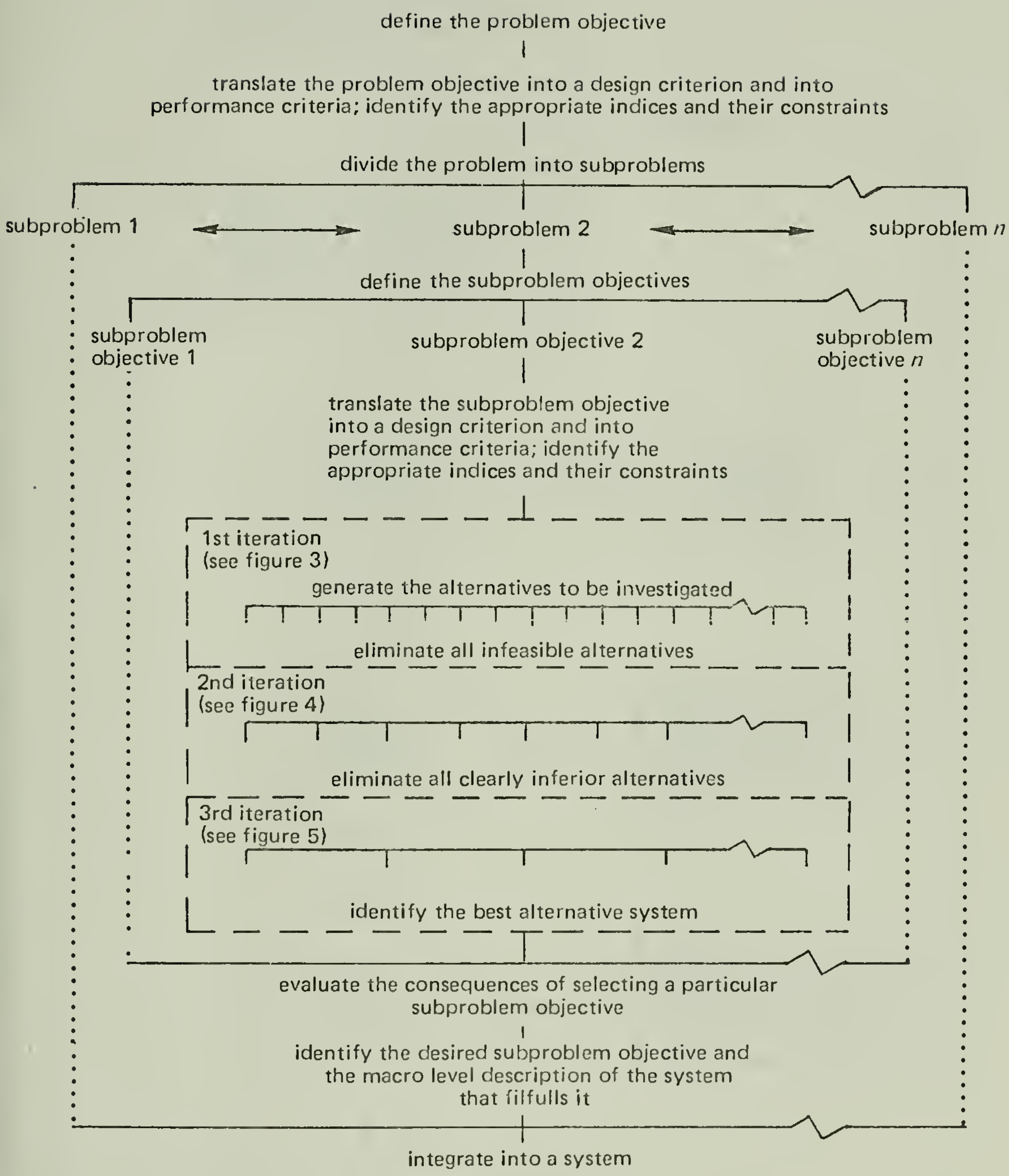
A design of a deep submergence vehicle, DSV, is a complex process involving many parameters. The design methodology proposed by Mandel and Chryssostomidis [1] will be used as the format for the presentation of the DSV design. The methodology is divided into two phases, the exploration phase and the synthesis phase. The exploration phase has a single stage whereas the synthesis phase has three stages, preliminary design, contract design and final design. This thesis will only consider the exploration phase and the first stage in the synthesis phase namely, preliminary design.

The steps involved in the exploration phase are presented in Figure II-1 as a flow diagram. The exploration phase begins with an overall problem objective and ends with a quantitative description of the problem objective and a macrolevel description of the "optimum" system which will satisfy the problem objective. The word "optimum" was placed in quotes since the exploration phase is not a closed form solution, but rather an approximate solution obtained via an iterative process. The optimization is highly dependent upon the number of alternative systems examined and the method used to identify the "best" alternative system. The solution of the "best" alternative system will be accomplished by systematic ,search a direct search method. The iterative process will proceed in a manner similar to the sequence presented in Figures II-2 through II-4.

The preliminary design stage of the synthesis phase begins with the output of the exploration phase, and ends with a micro-level description of the system under investigation. Essentially, the exploration phase identifies the optimum configuration of the optimum system. The steps involved in the preliminary design stage are the same as those in the exploration phase in Figures II-1 through II-4.*

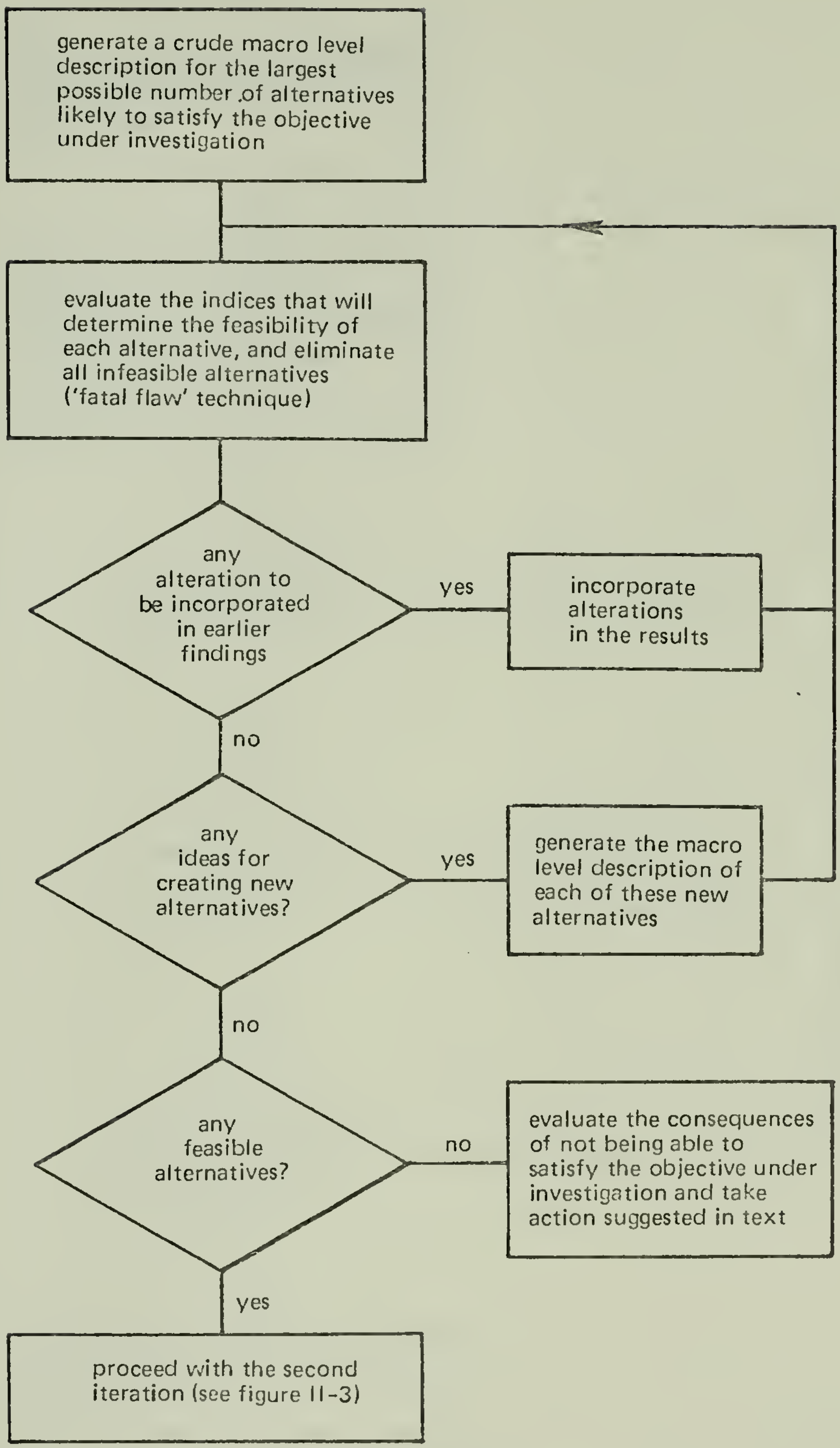
The actual details of the exploration phase and the preliminary design stage will be discussed in the next two sections using an actual DSV design as an example.

*Figures II-1 through II-4 were taken from reference 1.



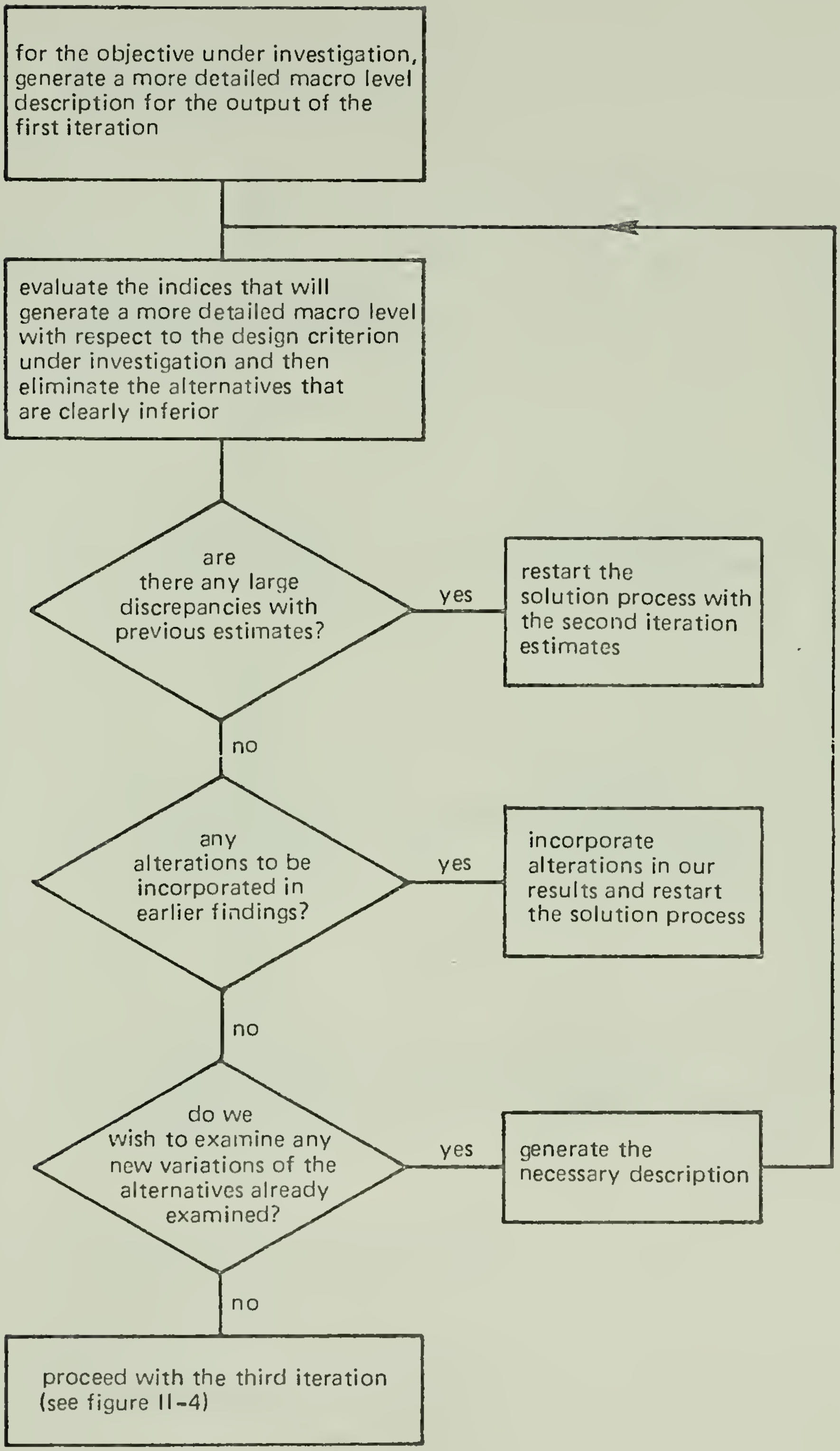
Exploration Phase Flow Diagram

Figure II-1



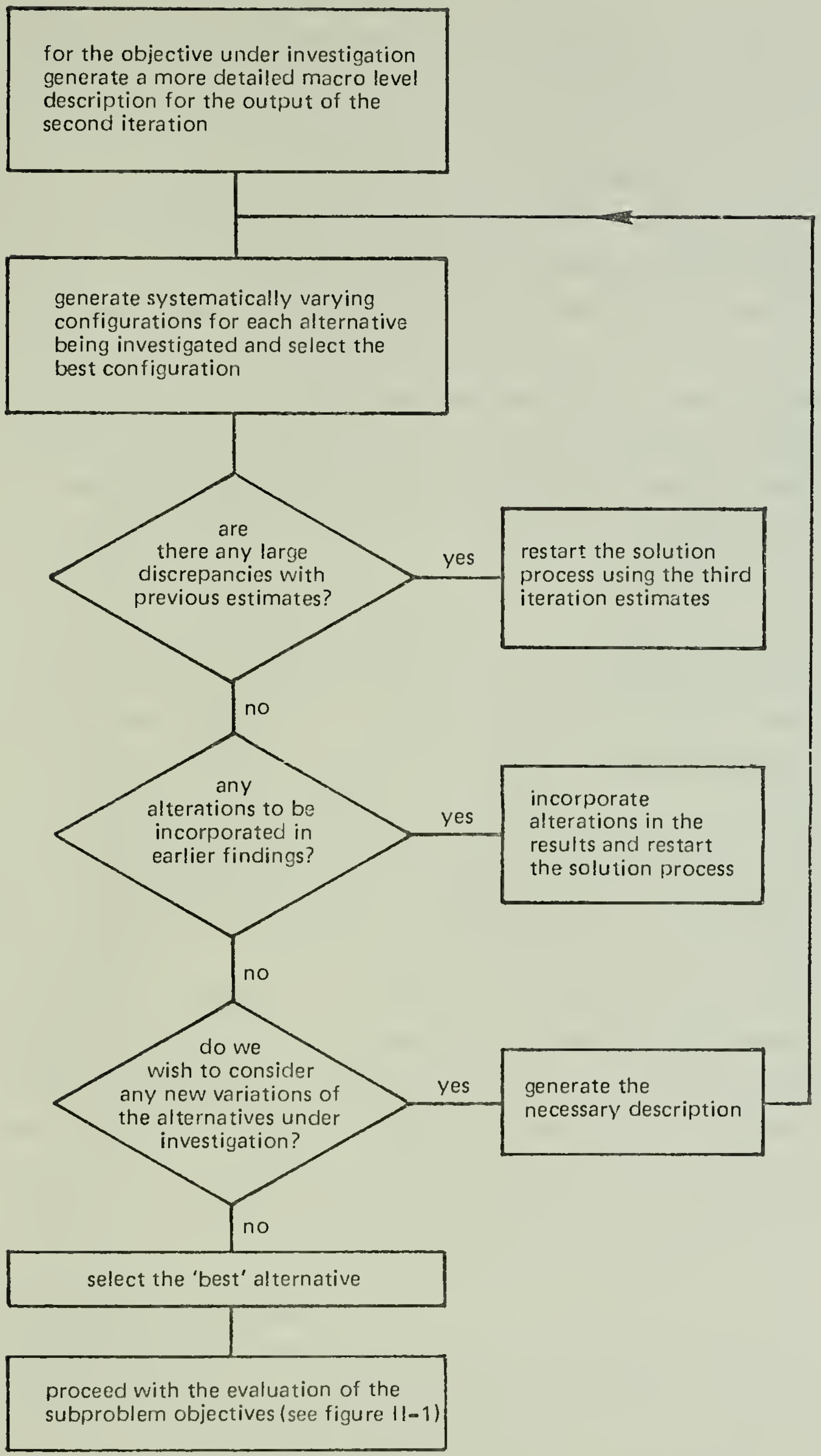
First Iteration

Figure II-2



Second Iteration

Figure II-3



Third Iteration

Figure II-4

III. Exploration Phase

A. Problem and Subproblem Objective

The outline of the exploration phase in Figure II-1, first divides the overall problem into subproblems which are further subdivided until a level is reached where a single person can solve the subproblem with the resources available to him. This subdivision is an approximation, since the subdivision tends to isolate the subproblem. This isolation of the subproblem is reduced by the degree of effective communications among the subproblems. Effective communications is probably the single most important factor in the efficient design of a complex vehicle.

The problem was briefly stated in the Introduction. The problem objective is stated in Figure III-1 along with the subdivision of the problem into subproblems. The example presented in sequence with the design methodology will be a manned untethered submersible. The subproblem objective is also presented in Figure III-1.

The next step in the exploration phase is the translation of the subproblem objective into design criteria and into performance criteria, identifying the various DSV constraints. A design criterion differs from a performance criterion in that the designer attempts to optimize the design criteria whereas the performance criteria is checked to insure that the performance parameter falls within the range of a given DSV constraint. The statement of the design criteria and performance criteria is based upon the initial statements of the subproblem objective. As can be seen in Figure II-1, the examination of the system alternatives provides feedback to the subproblem objective. As the exploration phase progresses the subproblem objective may change due to the change in the state-of-knowledge of the subproblem. In the DSV example, the initial subproblem objective specifies vehicle capabilities of wide area search, work, and inspection and surveillance. The outcome of the DSV exploration phase is a subproblem objective which specifies an inspection and surveillance mission alone.

B. Generation of Alternatives

1. Baseline Design: The following discussion will consist of the generation of a baseline design to establish the feasibility of the DSV subproblem objective stated in Figure III-1 followed by a description of the various systems considered and concluding with a description of the "best" system.

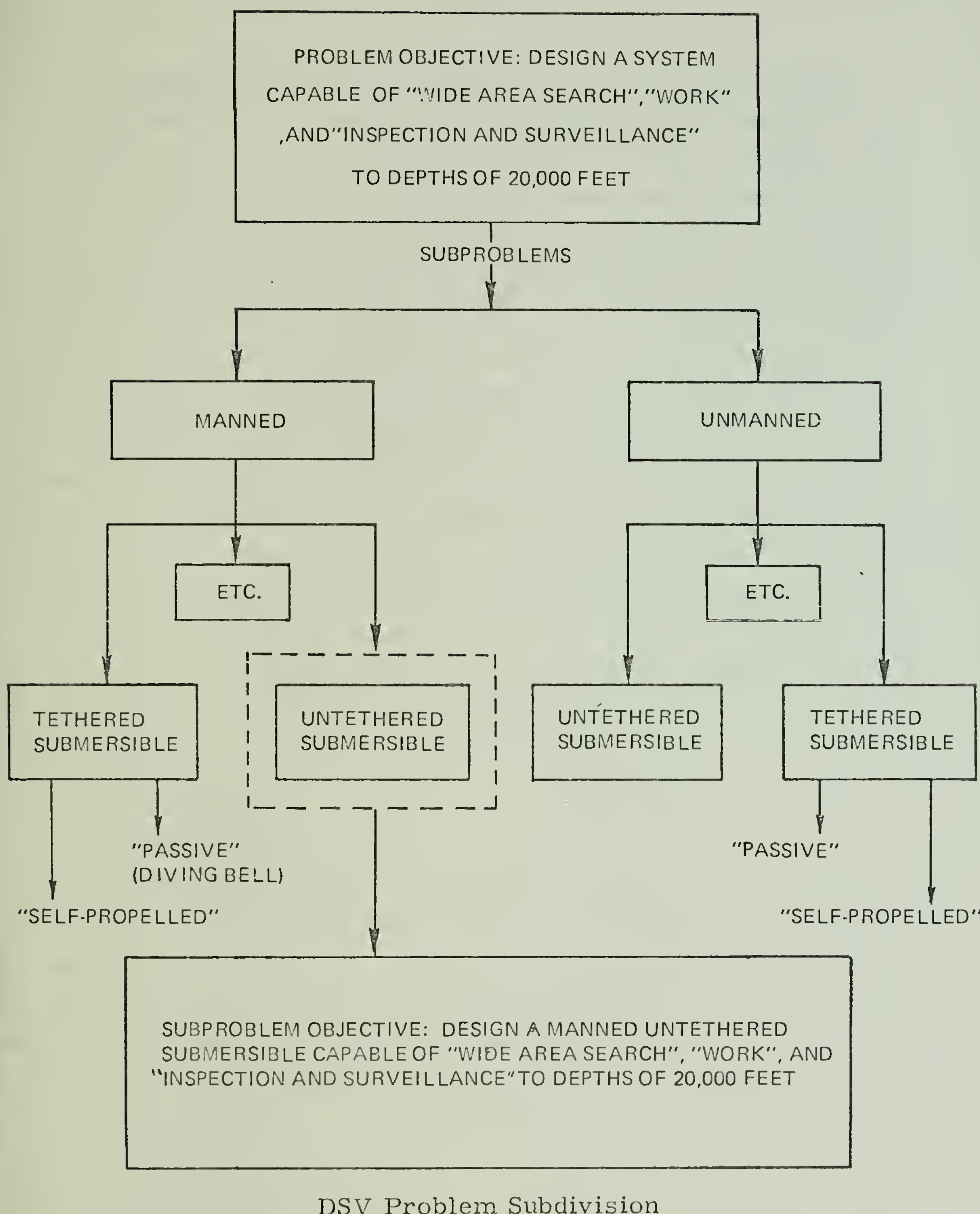


Figure III-1 DSV Problem Subdivision

There can be many inputs into the generation of a baseline design such as "state-of-the-art" constraints, environmental constraints and various constraints imposed by the mission requirements and mission profile.

a. State-of-the-art Constraints: At the outset of a DSV design there will be many "how to" questions. For example, the subproblem objective specified a work capability. What kind of impact does this have on the DSV? A good place to start with is a survey of operational submersible work capabilities. A list of operational submersibles [2] with a brief description of their capabilities is listed in Table III-1.* Table III-1 reflects the hopes and desires of the designers or owners, but does not portray the present utility of that submersible. It has been my experience from interaction with submersible operators, that many of the operational submersibles do not accomplish their problem objectives. Busby and many others have suggested that the problem objectives are too broad[3]. Submersibles like other high technology vehicles have been subject to the "All Purpose Vehicle," concept which usually comes out of the high vehicle cost. In reference 3, Busby talks of this problem and suggests some areas of consideration. Busby examined the operations of four submersibles:

Alvin

Aluminaut

Star III

Deep Star - 4000

Busby conjectures that the lack of agreement between submersible problem objectives and reality lies in the fact that there is no "All-Purpose Submersible," and furthermore, submersibles with multimission capabilities will probably accomplish none of them well. In his review, Busby compares four submersibles in the ten areas listed below:

Viewports
Speed
Atmospheric control
Habitability
Power/Endurance
Payload

*Table III-1 taken from reference 2.

Table III-1

Name of Vehicle	Builder/ Owner/ Operator	Status	Oper Depth (feet)	Propulsion: Speed (knots) Cruise	Propulsion: Speed (knots) Max	Propulsion: Endurance (hours)	Propulsion: No. Units	Propulsion: Energy Source	Dimensions (feet): L_{OA}	Dimensions (feet): Beam	Dimensions (feet): Height	Weighs (long tons): Dry	Weighs (long tons): Disp Sub.	Crew
ALUMINAUT	General Dynamics Corp/Reynolds International, Inc/ Reynolds Submarine Service	Oper 1964	15,000	3	3.8	32 @ 3 kn	Two 5-hp horiz (tail-plane). One 5-hp vert. (amidships).	Four 77-cell, 32,500-amp-hr, silver-zinc (internal)	51	15'5"	15	75	81.1	3 + 3 or 4 obs
ALVIN I	Litton Industries Applied Sciences Div /Office of Naval Research/Woods Hole Oceanographic Institution	Oper 1965	6,000	2	2.8	7.5 @ 1.75 kn	Two 5-hp hydraulic motors powered by pumps driven by 2 oil-encased electric motors	3 external, oil-encased, lead acid pressure comp	22	8	12.5	32,000 lb	13.5	2 + 1 obs
AMERICAN SUBMARINE MODEL 300	American Submarine Co	Oper 1961	300	1	6	3 @ 3 kn	One 3-hp, 12-to 24-v	Four 6-v, lead-acid	13	4.2	4.75	2,200 lb	--	2
AMERICAN SUBMARINE MODEL 600	American Submarine Co	Oper 1963	600	1	6	6 @ 3 kn	One 3.5-hp, oil-immersed	Four 6-v, nickel-cadmium	13	5.5	5.2	3,500 lb	--	2
AQUA-MARINE	Japan	Planning stage. Oper 1969.	2,000	--	--	10	--	--	--	--	--	--	--	--
ARCHIMEDE	French Navy	Oper 1961	36,000	1	2	12 @ 1 kn	One 30-hp, 110-v (horiz). One 5 hp, (vert.). One 6-hp (direction control).	Two 110-v, 860-amp-hr oil-encased, alkaline battery (outside pressure hull)	69	13	26.5	61 (less ballast and hexane)	195.9	3
ASHERAH (Star II Series)	General Dynamics Corp/U. of Penn Museum	Oper 1964	600	1	4	10 @ 1 kn 2 @ 4 kn	2 side-mounted, 2-hp, 24-v	External (24-v, lead-acid)	17	8'7"	7'7"	8,500 lb	9,350 lb	2
ATLANTA I	USSR	--	350	--	--	--	--	--	--	--	--	--	--	--
AUGUSTE PICCARD (PX-8)	Jacques Piccard/ Swiss National Exposition Corp	Oper 1964	2,500 with safety factor of 2	1-2	6	8 @ 6 kn	One 80-hp	Two 2000-amp-hr, 110-v, lead-acid batteries	93.5	19.7	24	167 (metric)	222 (metric)	3 + 40 passengers
AUTEC I	General Dynamics Corp/Woods Hole Oceanographic Institute	Under constr. Compl 1968.	6,500	2	--	8 @ 2 kn	--	Lead-acid battery	26	11	12	22	23.5	3
ARCHER-FISH (AGSS 31)	USN	Oper 1964	400	2	10	--	4 diesels 2 motors 2 propellers	Diesel electric	312	27	--	--	--	67+4
BEAVER MARK IV	North American Aviation, Inc, Ocean Systems Operations	Under constr. Compl 1968.	2,000	--	3	12 cruise	--	55-kw, 120-v, lead-acid battery	25	9.5	9.5	13	32,450 lb	3 + 2 divers
BENTHOS V	Lear Siegler, Inc	Oper 1963	600	1	3	16 @ 1 kn 2 @ 2 kn	Two 1-hp, 24-v	Nickel-cadmium batteries	11.3	6'1"	6'1"	4,200 lb	--	1 + 1 obs
BENTHOS 300	USSR	Oper	984	--	--	360	--	--	--	--	--	--	--	10 crew and obs
CUBMARINE PC3A	Perry Submarine Builders, Inc/ Kentron, Hawaii, Ltd	Oper 1965	300	2	4.25	8 @ 2 kn	One 7-hp	Six 6-v, 21-amp-hr, lead-acid	18 5	3.5	5'9"	4,790 lb	6,580 lb	2
CUBMARINE PC3B	Perry Submarine Builders, Inc/Ocean Systems Inc	Oper 1963	600	2 5	4.5	10 @ 1.5 kn 2 @ 4.3 kn	One 7-hp, 36-v	Lead-acid batteries	22	3.5	6'	5,500 lb	7,750 lb	2
DEEP DIVER (formerly PLC-4)	Perry Submarine Builders, Inc/Ocean Systems Inc	Oper 1967	1,350	2	3	18 @ slow speed	10-hp 240 d-c main motor, 3 hp bow & stern thrusters.	Twenty 12-v, 95-amp-hr, lead acid (main) 10- 12-v, 95-amp-hr, lead-acid, 30-kwhr, d-c (aux).	23	5.5	8.5	8	--	2 + 2 divers
DEEP JEEP	Naval Ordnance Test Station, China Lake, Calif/Scripps Inst of Oceanography	Oper 1964	2,000	2	2	4-6 + 2 kn	Two 0.75 hp, 24-v (variable) oil-immersed	Eight 6,900-whr, lead acid	10	8.5	8	4	--	1 + 1 obs
DEEP QUEST	Lockheed Aircraft Corp/LMSC	Launched 1967. Oper 1968.	8,000	2	4.5	24 @ 2 kn 12 @ 4 kn	2 horiz and 2 vert. 7.5-hp, a-c, main thrusters. Lateral water jets thrusters.	Lead-acid batteries (120-v)	39'10"	19	13.3	50	--	2 + 2 obs
DEEP STAR 2000	Westinghouse Electric Corp	Compl 1968	2,000	--	3	8 @ 2 kn	--	400-amp-hr lead-acid for propulsion. 155-amp-hr for aux	20	7	5.5	6.5	7	2
DEEP STAR 4000	Westinghouse Electric Corp	Oper 1965	4,000	2	3	12 @ 1 kn 6 @ 3 kn	Two 4.5-hp, a-c, free-flooded	3 lead-acid, 400-amp-hr in sixty-two 2-v cells (inverter)	18	11.5	7	17,800 lb	--	2 + 1 obs
DEEP STAR 20000	Westinghouse Electric Corp	Conceptual design. Compl 1969.	20,000	--	--	16-24 hr mission	--	Batteries	28	9	8	20	37	2 + 1 obs
DIVING SAUCER (DENISE)	Cousteau/ Westinghouse Electric Corp	Oper 1959	1,000	1	1	4 @ 1 kn	One 2-hp motor driving centrifugal water-pump, 2 jets provide thrust	Three 104-amp-hr, lead-acid batteries	9.5	9.5	5.5	6,500 lb	--	1 + 1 obs
DOLPHIN AG (SS)	Navy Yard Portsmouth NH/ US Navy/NEL	Under Constr. Compl 1968	Classified	--	--	Classified	1 diesel electric	3 silver-zinc batteries, 2 diesel-electric generators	152	19	20	900	950	22
DOWB	AC Electronics Defense Research Lab, General Motors Corp	Under Constr. Compl 1967.	6 500	2	2.5	26 @ 1 kn	--	Lead-acid batteries (40 kwh)	17	8'8½"	8'2"	15,500 lb	18,830 lb	2
DRV	USN China Lake	Design stage	35,000	--	6	--	--	--	70	--	--	--	--	3
DSRV	LMSC, USN	Under constr. Compl 1968.	3 500	--	5	12 @ 3 kn	One 15-hp main. Four 7½-hp thrusters	Silver-zinc batteries	50	8	9.5	65,000 lb	125,000 lb	3 + 24
DSSV	Undetermined/ USN, LMSC and Westinghouse Electric Corp	Conceptual design. Compl 1972.	20,000	--	5	30	--	Battery, fuel cells or other	50	11	11	70,000 lb	180,000 lb	4
FNRS-3	Piccard	--	--	--	--	--	--	--	--	--	--	--	--	--

Table III-1

Life Supp End. (hours) Norm.	Life Supp End. (hours) Max	Payload (pounds)	Pressure Hull Type	Maneuvering Control	Emergency Features	System Support	Remarks
36	72	6,000	11 forged 40-in. wide cylindrical rings, 2 hemispherical heads of aluminum alloy 7079-T6, 6.5-in. thick. Bolted construction.	Vertical motor, rudder & control planes, variable H_2O trim tanks, puller-type propellers	Fail-safe droppable shot ballast. 4000-lb droppable lead bar. Remote control station forward.	Towable at 5 knots, surface support from 316-ft ex-PCS PRIVATEER. Special designed ship now under development.	Oceanography, mineral & oil survey. 2 mechanical arms under development, capable of taking 4-ft core samples & 1-cu ft grab. 2 trainable TV, extendable boom, lifting ability of 4400 lb. Recently recovered 2100 lb of instruments from 3150-ft depth. Range 60 NMI.
10	24	13,000	7 ft od sphere HY-100 steel 1.33-in. thick	1 trainable stern & 2 rotable reversible propeller, mercury trim system. Variable ballast system.	Emergency sphere release batteries & mechanical arm droppable, emergency expulsion of mercury	Towable at 8 knots, surface support from 96-ft catamaran barge	Range 15-20 NMI. Designed to support multi-purpose scientific program. Mechanical arm, sample trays and jars. Two additional ALVIN-type vehicles in planning stage.
8	8	450	Welded dimetcoted A-36 steel, 0.375-in thick	Stick-type diving planes & rudder, H_2O trim tank	High pressure system for blowing ballast & trim tanks, releasable lead keel	Trailer and air transportable boat hoist, lifting rings	Wrap around plexiglass window. Range 9 NMI.
8	16	750	Welded dimetcoted 0.5-in. thick, A-36 steel	Stick-type diving planes & rudder, mercury trim system, mineral oil trim tanks for vernier weight control	2 high pressure tanks for blowing ballast, releasable lead keel	Trailer and air transportable	Range 18 NMI. Single conning tower with wrap around window, 8-in. port in bottom forward.
--	--	--	--	--	--	--	--
12	32	4,000	Ni-Cr-Mo forged steel sphere, 7.87-in. od, 6.9 id, 5.9-in. thick	2 motors provide vertical movement & direction control. Trailing rope for near-bottom control.	Shot-ballast automatically released on power failure	Towable at 8 knots max by tending vessel MARCEL LE BIHAN	Bathyscape. Has explored Kuriles Trench (31,320 ft), Puerto Rican Trench & Tyrrhenian Sea. Uses hexane (42,000 gal) for buoyancy. Range 12 NMI.
10	24	175	A-212 GR B steel, 0.625-in. thick, 5-ft dia sphere	Variable speed rotable propulsion motors. Water ballast, fixed stern stabilizers.	Emergency release, 360-lb skids	Trailer and air transportable. Ship supported (charter).	6 viewports forward, downward on both sides. Range 10 NMI. Inspection, photographic & scientific work.
--	--	--	--	Wings and stabilizers with rudders	Blow ballast	Towable by tender	Glider vehicle towed behind mother ship. Equipped with 5 lights, still & movie cameras
8	48 (with pass)	20,000	Steel cylinder, capped by steel hemispheres 33-mm thick	Forward & stern planes, aprs luyere, variable H_2O trim system	Shot ballast automatically released on power failure	Towable at 6 knots in good weather	Tourist version used to transport 40 pass. to 900 ft in Lake Geneva during the 1964 Swiss National Expo at Lausanne. Has scientific capability. Range 50 NMI.
--	24	1,200	Sphere with reinforced inserts for penetration. HY-200 steel, 6½-in. od sphere.	--	--	None designated	Oceanographic research. 2 bow manipulators. Range 16 NMI.
--	--	--	--	--	--	--	Oceanographic research & survey. Converted fleet sub to URV. Range 12,000 NMI.
6+	--	2,000	2 steel spheres connected by trunk. HY-100 steel. Fwd sphere 0.481-in. thick, 7-ft o.d. Aft sphere 0.387-in. thick 5.5-ft dia	Mercury trim system	--	Undetermined	Range 25 NMI @ 3 knots, diver lockout. 2 manipulators. Undersea construction projects. Rescue and salvage operations, surveying, mapping and sampling.
8	16	400	5-ft od medium steel sphere, 0.625-in. thick fiberglass fairing.	Reversible propulsion motors trainable 180 deg from horiz. variable H_2O trim system.	Droppable lead ballast. Blow main ballast	Trailer & air transportable	6 viewing ports. Range 4 NMI.
--	--	--	--	--	--	--	Equipped with special device enabling sub to stand on bottom. Has equipment to analyze water. Creates sound, light & electric fields. Endurance 360 hr.
8	20	750	Welded rolled plate. A-285 steel, 0.25-in. thick	Rudder & bow planes, variable H_2O trim system	Droppable 185-lb keel, high pressure air for emergency blowing of main ballast tanks	Trailer & air transportable. Charter ships.	Range 16 NMI. Search, observation, survey & recovery. Propulsion endurance may be increased with optional silver-cadmium or silver-zinc batteries. Hovering motors available. Lifting capacity of 2100 lb.
12	20	750	Welded rolled plate, 0.5-in. thick, A-212 steel	Rudder & bow planes, variable H_2O trim system	Droppable 220-lb keel, high pressure air for emergency blowing of main ballast tanks	Trailer & air transportable. Charter ships.	Lifting capacity 2200 lb. Range 20 NMI. Search, observation, survey & recovery.
8	--	2,000	Two 53-in. cylindrical compartments, 0.50 in. thick (HSS 1-1 type), welded rolled plate.	Rotates on own axis. Hover in any attitude at 2 speeds. Tilt a max of 30 deg. from horiz. along roll & pitch axis in a cross cur. of 2 kn.	Droppable battery compartment	Sea diver	7 view ports fwd compt. 5 view ports diving compt. Lock out, lock in to 1250. Control compt features all around visibility. 112 hull penetrations. Range 9 NMI.
6	48	200	5-ft od HSS(50) sphere	Rotable motors aft	Droppable batteries, air-filled toric emer marker	Ship supported	Range 10 NMI. Neutrally buoyant vehicle. Underwater range tool & ocean research instr. Single viewport & fixed periscope.
24	48	7,000	8 spherical 7-ft dia HSS(200) maraging steel, 0.89-in. thick spheres	Hover in 0.5-knot current	Droppable batteries, lead shot, arm and TV units, mercury trim	Ship supported (TRANSQUEST). Trailer trans.	Operable from submerged submarine. Aluminum (5083) outer hull. Range 48 NMI. Oceanographic research and recovery and man-in-sea support
--	24	1,000	Spun hemispheric heads and rolled cylinder. HY 80 steel, 0.75-in. thick. Sphere dia 5 ft.	--	Jettisonable descent weight, jettisonable emergency battery weight, jettisonable mercury ballast	Undetermined	Manipulator. Range 10 NMI. Scientific instrument carrier.
12	48	350 + crew	6.5-ft od, 1.2-in. thick HY-80 steel sphere	10 small, floodable spheres outside hull. Mercury trim system. Reversible var speed motors. 2 weights: 330 lb for descent, 173 lb for ascent.	Battery (forward) & mercury supply droppable, inflatable conning tower for emer exit on surface	Air transportable, ship supported	Direct underwater observation, bottom coring, water & bottom sampling. Helicoidal over pattern permits 60° ascent & 50° descent. Hydraulic claw & specimen basket. Range 10-16 NMI @ 2 kn.
--	24	1,000	Welded higher strength steel, 1.85-in. thick, 7-ft od sphere	--	--	Undetermined	Transportation scientists & equipment to 20,000 ft. Stops at intermediate depths with station keeping ability. Range 15-20 NMI.
4	24	100	0.75-in. thick medium steel ellipsoidal, 6.5-ft major dia, 4.9-ft minor dia	Rotable jets. Two 55-lb weights for ascent and descent. Fine buoyancy control by internal ballast tank. Mercury trim system	420-lb. releasable emergency weight. Inflatable conning tower for surface exit.	Air transportable. Supported by 136-ft boat-truck	Range 4 NMI. Hydraulically controlled arm. Specimen basket. Excellent maneuverability.
--	--	24,000	HY-80 steel cylinder, hemispheric ends	Water ballast, rudder & diving planes, hovering control	--	None required	This deep diving submarine will be operated in support of naval oceanographic research requirements
--	65	1,000	Spun & machined HY-100 steel, 0.935-in thick, 80.12-in. id sphere	--	--	Research vessel SWAN or charter ship	Range 26 NMI @ 1 knot. Search and recovery. Placement and relocation of small objects
--	--	--	--	--	--	--	Range 48 NMI. Endurance 100 hr.
--	--	24 rescuees	HY-140 steel, 3 intersecting 7½-ft dia spheres	--	--	Air, land, ship, & sub transport. ASR & ASRD.	Range 36 NMI. Primary mission, rescue; secondary mission, sonar research, bottom coring, mapping, etc
--	--	--	Welded sections	--	--	ASR & ASRD	Range 50 NMI. Search, recovery of small objects
--	--	--	6.25-ft id sphere	--	--	--	Bathyscaph

Table III-1

Name of Vehicle	Builder/ Owner/ Operator	Status	Oper Depth (feet)	Propulsion: Speed (knots) Cruise	Max	Endurance (hours)	No. Units	Energy Source	Dimensions (feet) L_OA	Beam	Height	Weight (long tons) Dry	Disp Sub.	Crew
FNRS-3	Piccard French Navy	Replaced by Archimedes	20,000	--	--	--	Two 1-hp	--	57.5	--	--	--	--	2
GROUPER (AGSS 214)	General Dynamics Corp/USN	Unknown	--	--	--	--	--	Diesel electric	311	15	27	--	1,816	80
GSV-1	Grumman Aircraft Engrg Corp	Conceptual design. Compl 1969.	2,000	--	--	--	--	Lead-acid batteries (> 750 kwh)	61'9"	13	22'10"	173	130	3 + 6
KUROSHIO I	Japan	Oper	660	--	--	--	--	--	36.8	7.5	10.4	11.5	--	4
KUROSHIO II	Hokkaido Univ Japan	Oper 1960	650	--	--	Tethered	One 440-v, a-c motor	Surface power electric through cable	36.7	7.15	10.4	11.5	--	4-6
MORAY (TV-1A)	US Naval Ordnance Test Sta, China Lake, Calif.	Compl 1964. Not in oper now.	3,500	6	15	3.6 @ 6 kn	One 90-hp battery operated torpedo motor, counter-rotating propellers	Bank of 240 silver-zinc secondary cells	33	5.3	5.3	10	16	1 + 1 obs
PAULO I	Marine Technology Inc/Anautics Inc	Oper 1967	600	--	--	10	--	Lead-acid batteries	13.5	4.5	6'4"	5,200 lb	6,025 lb	2
PISCES	International Hydro, Canada	Oper 1966	5,000	1	6	12 @ cruise	--	Lead-acid batteries (66 kwh)	16	11.5	9	6.5	--	2-3
PX-15	Grumman Aircraft Engrg Corp, Cousteau	Under constr. Compl 1967.	2,000 with safety factor of 2	3	5	--	Four 25-hp, variable frequency, a-c motors	2,500-amp-hr @ 300-v capacity, lead-acid	48	13'4", 18'6" with motors	20	--	130	6
SEVER I	USSR	--	Tested 1,500	--	--	2	--	--	--	--	--	--	--	--
SEVER II	USSR	Under constr	6,000	--	5	--	--	--	20	--	--	--	--	2
SEVERYANKA	USSR/All-Union Institute of Marine Fishery and Oceanography (VNRIO)	Oper 1958	550	15	--	16,500 mile range	Diesel electric	Diesel-snorkel, lead-acid batteries	240	22	15	--	1,180	60 6-8 scientific party
S-10	Link	--	432	--	--	49	--	--	8	--	--	--	--	2
SPORTSMAN DRY SUB	American Submarine	--	300	2	6	8	2 hp	--	12	4.2	--	1.1	--	2
STAR I	Electric Boat Co. General Dynamics Corp	Oper 1964. No longer being operated.	200	0.75	1	3 @ 0.75 kn	2-side mounted, 0.25-hp, 18-v	2 external, 18 v, d-c, lead acid	10.1	6	5.8	2,750 lb	2,890 lb	1
STAR II	Electric Boat Co. General Dynamics Corp	Oper 1966	1,200	1	4.5	8 @ 1 kn 2 @ 4 kn	2-side mounted 2-hp One 2-hp in sail.	Battery (108-v)	17'9½"	5'4"	7'7¾"	9,400 lb	10,340 lb	2
STAR III	Electric Boat Co. General Dynamics Corp	Oper 1967	2,000	1	4	12 @ 1 kn 2 @ 4 kn	7.5-hp, single screw, stern drive, 2-hp bow thruster, 2-hp vert. hovering motor	Battery (120-v)	24.5	6	8	8	19,800 lb	2
STAR IV	Electric Boat Co. General Dynamics Corp	Design Stage	2,000	--	--	--	--	60-kwh lead-acid battery	30	7	10	14	--	--
SUBMAR I	Boris K. von Danielowsky, USA	Prelim design	20,000	--	7	--	Two 10-kw electric motors	Fuel cells or nuclear generators	--	--	--	--	--	4
SUBMARAY	Hydrotech Co	Oper	300	--	2.5	16	--	Battery (24 v)	14	--	--	--	3,200 lb	2
SUBMAR-BELL	Divers International	--	900	--	--	--	--	--	--	--	--	--	--	--
SURV	Lintott Engrg, England	1967	1,600	--	2.5	--	--	--	--	--	--	6	--	2
TOP 15	USN/Nortronics	Design stage	3,500	3	5	12+	--	--	44	--	--	--	--	3
TOP 16	USN	Design stage. Compl 1968.	6,000	--	5	--	--	--	44	--	--	--	--	4
TINRO I	USSR	Oper	984	6	8	--	--	--	36	--	--	--	--	5
TINRO II	USSR	Under constr	--	--	--	--	--	--	--	--	--	--	--	--
TRIDENT	USN	Design stage	36,000	--	--	--	--	--	--	--	--	--	--	2
TRIESTE II	USN/COMSUBPAC	Oper 1964	(Terni) 20,000 (Krupp) 36,000	2	--	5 @ 2 kn	Two 10-hp aft. One 2-hp bow thruster.	External 145-kwh lead-acid	67	15	18	50 less ballast and avgas	220	(Terni) 1 + 2 obs (Krupp) 1 + 1 obs
TUNA RESEARCH SUB	General Dynamics Corp/Bureau of Fisheries	Design stage	1,000	3	6	--	--	--	--	--	--	--	--	7
TV-2	USN	Design stage	6,000	--	--	--	--	--	44	--	--	--	--	--
UTILITY SUB	USN/China Lake	Design stage	--	--	--	--	--	--	--	--	--	--	--	2
WATER-COUP	Weber Aircraft, Div of Walter Kidde & Co	Prototype under constr	600	--	--	48	--	Batteries	9	--	--	3,300 lb	--	2
YOMIURI	Mitsubishi/Yomiuri Shimbun Newspaper, Tokyo Japan	Oper 1964	1,000	4	--	6 @ 4 kn	1 electric a-c, 2 diesel	Diesel-electric, auxiliary diesel for battery charging, alternator for a-c power.	48	8.2	9.2	10.5	35	6

Table III-1

Life Supp End (hours) Norm.	Max	Payload (pounds)	Pressure Hull Type	Maneuvering Control	Emergency Features	System Support	Remarks
--	--	--	6.75-ft id sphere	--	--	--	Bathyscaph. Buoyancy by 20,000 gal gasoline.
--	--	--	--	--	--	--	Range 10,000 NMI & 10 kn. Endurance 60 days. Conversion of a GATO class sub, built in 1942 by Electric Boat. First conv June 1958 and designated (AGSS) sub. surveying vessel. Later conv 1964.
--	--	18,000 +	Ring stiffened cylinder (isobaris & hyperbaris chambers), 38-mm thick, 10-ft dia, steel sphere	--	--	Undetermined	Range 75 NMI. Endurance 9 men, 2 weeks. Utility version of PX-15 adapted for commercial use. Diver egress at 1000 ft.
--	--	--	--	--	--	Power sup. from surf. ship	Endurance 24 hr
24	--	--	Medium steel plate	Rudder, water ballast system	Cable can be cut so vehicle can power to surface on batteries	Power supply cable from surface ship	Vehicle tethered to surface by 1500-ft (35-mm dia) cable, 16 viewing ports, 5 exterior lights, Manipulator, bottom sampler, TV, phone to surface
24	--	200	Two 5-ft od, cast aluminum (A-356-T6), bolted spheres (1 for crew, 1 for instr)	Tail-mounted control surfaces, mercury trim system autopilot	Vehicle is positively buoyant, automatic mercury release, automatic buoy marker release	Air transportable, ship supported	Range 21 NMI. Ring stiffened fiberglass hull contains syntactic foam. First vehicle to demonstrate feasibility of positively buoyant URV design. MORAY is "flown" through water. Oceano-acoustic research. No viewports, 2 TV cameras.
--	--	450 normal, 3000 special	Welded cylinder & hemispherical end bells, A-212 steel, 0.75-in. & 0.5-in. thick	--	--	Charter ships	Primarily for inspection of offshore pipelines, rigs & sewer outfalls. Range 30 NMI.
24	72	1,500 + crew	6 welded segments. ALGOMA 44 steel, 0.750-in. thick., 76.5-in dia sphere.	Motor driven ballast for trim control	Droppable weights -- 2200 lb	Barge	Range 24 miles & 1 kn. Endurance 72 hr. 2 manipulators, 2 sonars, 1 clamp arm.
--	42 days	10,000 min	10-ft dia, 1.4-in steel plate with reinforcing rings spaced 27½ in. along longitudinal axis	Speed regulated by changing current frequency	--	Undetermined	Endurance 6 weeks, 6 men, 2 weeks, 15 men. Used initially for a drifting mission in the Gulf Stream. Range 60 miles @ 3 kn. 25 viewing ports, 20 floodlights, TV sets. Liquid oxygen tanks to supply air. CO_2 scrubbers, hull is less compressible than water enabling craft to achieve depth stab. without mech ballasting.
--	--	--	--		--	--	
--	--	--	--	Vertical & horizontal screws	--	--	Endurance 24 hr. Similar to ALVIN. Research submersible reinforced plastic superstructure.
--	--	--	--	Rudder, diving planes & water ballast	--	None	Range 16,500 miles. Converted WHISKEY class submarine for fisheries oceanography. Forward torpedo room converted into scientific lab. 3 observation stations with viewing ports on each side of hull and overhead. TV in bow, exterior illum, extendable bottom sampler through hull
--	--	--	--	--	--	--	Underwater tent
--	--	450 crew + inst	--	--	--	--	Sports activities. Survey work for salvage, etc. Range 10-15 miles.
4	18	200	A-212 Grade B steel, 0.375-in. thick, 4 ft dia sphere	Differential operation of rotatable motors with planes attached, water ballast.	Emergency release of lead ballast	Trailer & air transportable, ship supported	Range 3 NMI. Operated on SEA LAB I. First URV to test fuel cell propulsion.
12	24	310	HY-80 steel, 0.625-in. thick, 5-ft dia sphere	Variable speed rotatable products of motors water ballast fixed stern stabilizers	Emergency release 360 lb skids	Trailer & air transportable, ship supported	6 viewpoints forward, downward on both sides Highly maneuverable. Bow manipulator. Range 8 NMI. Performed cable inspections
12	24	1,050	5.5-ft dia, HY-100, 0.5-in. thick	Hovering & bow thrust motors Servo-powered rudder. Mercury trim system. Variable ballast tanks.	Emergency release of mercury. Mechanical arm with droppable equipment mounting plate	Trailer & air transportable, ship supported	Scientific work, inspecting & photography. Retractable mech arm 270° viewing through 4 ports. High payload-to-displ ratio and manuv. Range 12 NMI.
24	48	--	Three intersecting spheres	--	--	--	Conceptual design
--	--	--	--	Rotate horiz on own axis. Hover in any attitude. Nitrogen & pyrotechnic devices, blown ballast tanks. Vertical adj propellers	--	--	Offshore oil fields, search survey, mining, salvage, ocean engineering. Manipulators operated by fluidic controls.
1.6	--	--	3-ft dia	Rudder & bow planes	--	--	Survey work. Fisheries, cables, sewer outfalls. Range 15 NMI.
--	--	--	--	--	--	--	Crawler type. Self-propelled, double-hulled diving bell.
36	50	--	--	--	--	--	Resembles Deep Jeep. Commercial use in offshore oil field in North Sea
--	--	--	--	--	--	--	Range 50 miles. Planned for submarine rescue work.
--	--	--	--	--	--	--	Range 60 miles. Search & recovery of small objects.
--	--	--	--	--	--	--	Range 40 miles & 3 kn. 20 day capability. Surface range 1200 miles & 8 kn. Communication equip, cameras & TV. Looks like ALVIN in overall configuration but will have cylindrical manned hull.
--	--	--	--	--	--	--	Same purpose as TINRO I but a limited capability. Same equipment as TINRO I. Resembles DEEP STAR 4000. Fiberglass constr manipulator.
--	--	--	--	--	--	--	Planned for bottom work
10	24	(Terni) 20,000	Terni sphere. 6-ft. od, 4-in. thick, Ni-Ce-Mo forged steel, 7.2 ft dia Krupp hull	2 shot ballast silos, 47,000 gal avgas flotation. Drag rope. Gas maneuvering tank amidships.	Snorkel for emergency at surface. Fail-safe ballast release, two 11 6-ton droppable shot silos	Tug or LSD supported	Presently used for test & training. Range 10 NMI. Vehicle presently equipped with Tern sphere. Krupp sphere interchangeable. Mechanical arms, 3 TV camr, High payload cap. TR.&S & III, an evolutionary improvement over I-II, built at Mare Island Naval Shipyard for navies. (1967) Gas. add. 100 tons.
--	--	--	8-ft dia sphere	--	--	--	Range 40 miles & 3 kn. Endurance 1000 hr, 29 & 65-sq-ft labs.
--	--	--	2 glass spheres	--	--	--	Designed for positive buoyanc,
--	--	--	--	--	--	--	Research
--	--	400	Glass fiber hemisphere	Propelled by 2 cycloidal propellers P&S	--	--	--
--	--	--	Higher strength steel spheres, 0.6-in. thick, 6'-8"-ft dia sphere	Manual control of rudder & diving planes H_2O trim, buoyancy tanks	--	Ship supported. Tabuse model No. 25.	Range 24 NMI. Miniature submarine for fisheries oceanography and seafloor investigation. Mechanical arm 4 floodlights. 7 optical glass windows, TV camr net and press. tank for specimen preservation

Buoyancy control

Manueverability and flight control

Operating performance and maintenance

Busby's recommendations are listed below[3]:

*The concept of an all-purpose submersible is a delusion that only serves to frustrate the user and retard development and application of manned submersibles to oceanographic studies and tasks. The prospective builder/owner should first decide what primary mission the vehicle will perform and then design to fill these requirements. This approach will result in a vehicle designed to accommodate pertinent instrumentation and perform the required tasks, rather than the present situation which requires the user to modify his instruments and tasks to accommodate the vehicle.

*Physiological and anatomical comfort of pilot and observers must be a first-order consideration in present and future vehicles to obtain full effectiveness for even the short 8 to 12 hr. submerged duration now available. The novelty of diving is quick to wear off and, unless improvements are made in providing comfortable vehicles, there will be little benefit received in longer duration dives. The greater need for improving comfort is in the viewing arrangement where either a lying or sitting posture is required if the dives are to exceed 2 or 3 hr. duration. In the smaller submersibles,fixed seats should be avoided and replaced by folding stools. Covering the deck of the submersible with padding allows for greater comfort and a wider variety of viewing positions.

*Standardization of hatch dimensions in future vehicles should, where feasible, be required of all submersibles. This would aid the user in designing for internal instruments, and more important, if the hatch can be made compatible with the Deep Submergence Rescue Vehicle's personnel transfer hatch, it would offer some means of rescue. At present, there is no method of rescuing personnel from existing deep submersibles other than by bringing the vehicle to the surface.

*Electronic interference is a present problem which will undoubtedly increase as multiple tasks are pursued on the larger vehicles. Consequently, electrical leads close to or paralleling high-energy sources should be well shielded. An alternate solution is provided in Alvin, where sensitive electronic leads are in hull penetrations physically removed from cables carrying

heavy loads.

*Two forward looking viewports, with the viewing area overlapping, similar to the Deep Star-4000, are needed in order to obtain full viewing effectiveness and the teamwork between the pilot and observer required for a successful dive. A third, smaller port solely for cinephotography is mandatory for successful and simplified photographic documentation. A still camera can be mounted in this port, if motion pictures are not required. It is granted that such cameras can be mounted externally; however, the inability to reload film or change camera settings while the equipment is external to the sphere is a severe handicap. The most preferable viewing capability is that of incorporating complete glass spheres or glass hemiheads, such as is under development in the Navy Undersea Warfare Center's Deepview. This capability, when achieved, will provide a system many times preferable to any arrangement now available or under construction.

*Incorporation of an upward looking viewport into future submersible design would greatly enhance the vehicle's safety while operating in the presence of overhanging cliffs, cables, and when surfacing.

*Displays of vehicle depth/altitude, compass heading, pitch, roll, and other information required by the pilot for navigation should be mounted where he can view them without changing position in the vehicle. Similar mounting of environmental sensor displays should be provided by the observer. In the case of large vehicles with co-pilot and co-observer, these tasks can be performed and the information relayed elsewhere as required by them.

*Automatic on-board recording of all environmental and operational information required for the mission should be incorporated into the instrument design. Electronic design must also allow for the great variations in submersible temperature and humidity present during tropic or subtropic operations. These variations are restricted to a great degree on Alvin and Deep Star by blowing in cool air between dives. Internal temperature may be lowered during shallow, warm water dives by directing blowers or fans to blow against the generally cooler pressure hull.

*Although no ill effects have been noticed, care should be taken in the selection of recording paper. The fumes generated from such recorders may produce a noxious or, in the extreme, toxic atmosphere in the limited confines of the smaller submersibles.

*Variable ballast systems, such as are used on <u>Alvin</u>, offer far greater operational versatility than shot ballast systems, such as <u>Aluminaut</u>'s, and have been more dependable.

*Fixed side thruster propeller such as on <u>Deep Star-4000</u> can be made more useful for maneuvering, if they are modified to rotate 360 deg. as on <u>Alvin</u>.

*The mechanical arms with the dexterity of <u>Aluminaut's</u> would greatly enhance the capability of any submersible and are minimal requirements when working in a fixed position under strong currents. Future design should allow for feedback to the operator which indicates the pressure or torque being applied.

*Sampling baskets should be designed to hold the sample on the surface as well as underwater, as sea or swell can often wash the sample out of its basket when the vehicle has surfaced. The arrangement of the basket on the vehicle should be such that it can be observed directly or through TV.

*A reasonable mounting rack, such as on <u>Star III</u>, offers distinct advantages in the design of instrument mounting brackets and ease of equipment installation. Standardization of these racks would greatly assist users who employ the same equipment on various vehicles.

*Homing in on bottom beacons has been accomplished by many submersibles and can easily become routine. This procedure can be used to advantage in many operational aspects, particularly so in recharging batteries, as proposed by Westinghouse Corporation, while the submersible is submerged. Similar to aircraft in-flight refueling, the submersible can be designed to mate with a tethered battery charger from the surface and recharge batteries, while the submersible crew rests. This system offers the potential of increasing state-of-the-art submersible power endurance, while negating costly development of fuel cells or other exotic power sources.

b. <u>Environmental Constraints</u> are another input to the generation of a baseline design. Reference 4 is an invaluable source of potential hazards which a DSV could encounter. The hazards are divided into two groups:

<u>Man-Made Hazards</u>

1. Cables
2. Wrecks
3. Bottom-mounted hardware/buoyed arrays
4. Surface traffic

5. Sub-surface traffic
6. Explosive ordinances
7. Miscellaneous hazards

<u>Natural Hazards</u>

1. Sea State
2. Currents
3. Bottom sediments
4. Topography
5. Visibility
6. Marine organisms
7. Miscellaneous

The recommendations from reference 4 are listed below:

*Inspection or repair of bottom cable and hardware should be preceeded by a thorough briefing on the description of the particular hardware as well as the method used to plant and subsequently retrieve these items in order to ascertain the presence of lowering and retrieval lines in the area.

*Lateral visibility limits of 30 to 50 feet and a general lack of near-bottom current information encourages an up-current approach to anchored hardware or instrumentation by the submersible in the event of propulsion loss and by virtue of the limited viewing area available.

*Exploration of large-scale wrecks or sunken ships should be undertaken only when required by the mission, and identify of the vessel should be ascertained, when possible, so that the deck plan is known or can be anticipated to the highest degree possible.

*Coordination between Naval surface and subsurface operations should be investigated prior to an operation and maintained throughout the diving program.

*Unless it is their specified task and they are adequately prepared for mission, submersible pilots should, under no condition, voluntarily make contact with or manuever in close proximity to any piece of explosive ordinance detected on or anchored to the ocean floor.

*All submersibles should be equipped with obstacle avoidance equipment capable of detecting and, to some extent, classifying bottomed objects in

the size range of mines, torpedoes and depth charges at a range sufficient to prevent inadvertent contact.

*The location, type and condition of any piece of explosive ordinance observed during the course of submersible operations should be reported to the U. S. Naval Oceanographic Officer or the U. S. Coast Guard.

*Rather than allow the accumulation of potentially dangerous explosive ordinance on the ocean floor to continue, the armed nations of the world should seek practical means whereby any weapon expended at sea which fails to fire on a target, renders itself harmless through self-detonation or through rendering inert its explosive charge.

*Predictions of near-bottom currents in topographically rough areas can only be assumed to represent that spot in which the current meter was planted. Variations of several knots can occur within short time periods and distances.

*Topographic (bathymetric) data obtained from surface platforms should be treated as a general guide to bottom roughness and not taken verbatim as a true index of small scale (tens of feet) relief.

*An agency of the Federal Government should be appointed to provide pre-dive environmental information in the prospective area of interest upon request and to coordinate Naval and civilian activities to preclude operational interference on the part of both parties.

*The same agency should be responsible for accumulating post-dive information from the submersible operators which will be available to future submersible or engineering operations upon request.

c. MR and MP Constraints: Now that a brief review of "state-of-the-art" constraints and environmental constraints have been presented, the various constraints imposed by the mission requirements and mission profile will be discussed.

(1) Mission requirement: The mission requirement is to locate and investigate objects of interest in the deep ocean. These objects of interest could be natural or man-made. The subproblem objective specified "Wide Area Search," "Work," and "inspection and surveillance" capabilities, but what does this mean in terms of a DSV system or subsystem requirements? A sensor trade-off study is presented in Appendix B. Since many DSV wide area search trade-offs have been conducted, a literature review will be provided. Appendix B also con-

tains a section on optical sensor design, since very little has been published on DSV optical sensor design. Appendix C, provides an outline on the present status of deep submergence work capabilities.

(2) Mission profile: The proposed mission profile will begin at the support base for the submersible, proceeding to the on-site mission and back to support base again. The efficiency and economy of a search mission is closely related to the on-site time to at-sea time ratio. Therefore, it is essential to determine the constraints imposed by the various modes of transportation and by the various modes of support which will help to maximize this ratio. The discussion below will propose on-site support craft and modes of submersible transportation in which will help to maximize the on-site time to at-sea time.

(a) Transportation

[1] Mother Sub: Depending on the design of the submersible, the mother sub should be able to transport the submersible at surface speeds of 15 knots. This imposes a wave slap requirement on the submersible superstructure and skin. The wave slap requirement traditionally has been taken between 500 and 1000 psi.

[2] ASR: The ASR can transport the submersible at speeds from 15-18 kts depending upon the sea state. Acceleration loads created while on the deck of the ASR must be determined.

[3] Air Transport: Although air transport is not essential to the mission it would be convenient and help cut down total mission time. The capabilities of a C-5A are listed below:

C-5A Capabilities:

[a] Range-Payload

5500 nautical miles with a 100,000 lb payload

2700 nautical miles with a 200,000 lb payload

[b] Cargo Compartment Profile

(See Figure III-3)

The restrictions imposed on the submersible are basically depth restrictions (i.e. the cradle and submersible must be less than 13.5 feet). Another restriction placed on the submersible is the acceleration loading due to aircraft operations.

[4] Truck: Overland transport can be accomplished by truck. Acceleration loadings and possibly wind loadings may be the only real restriction.

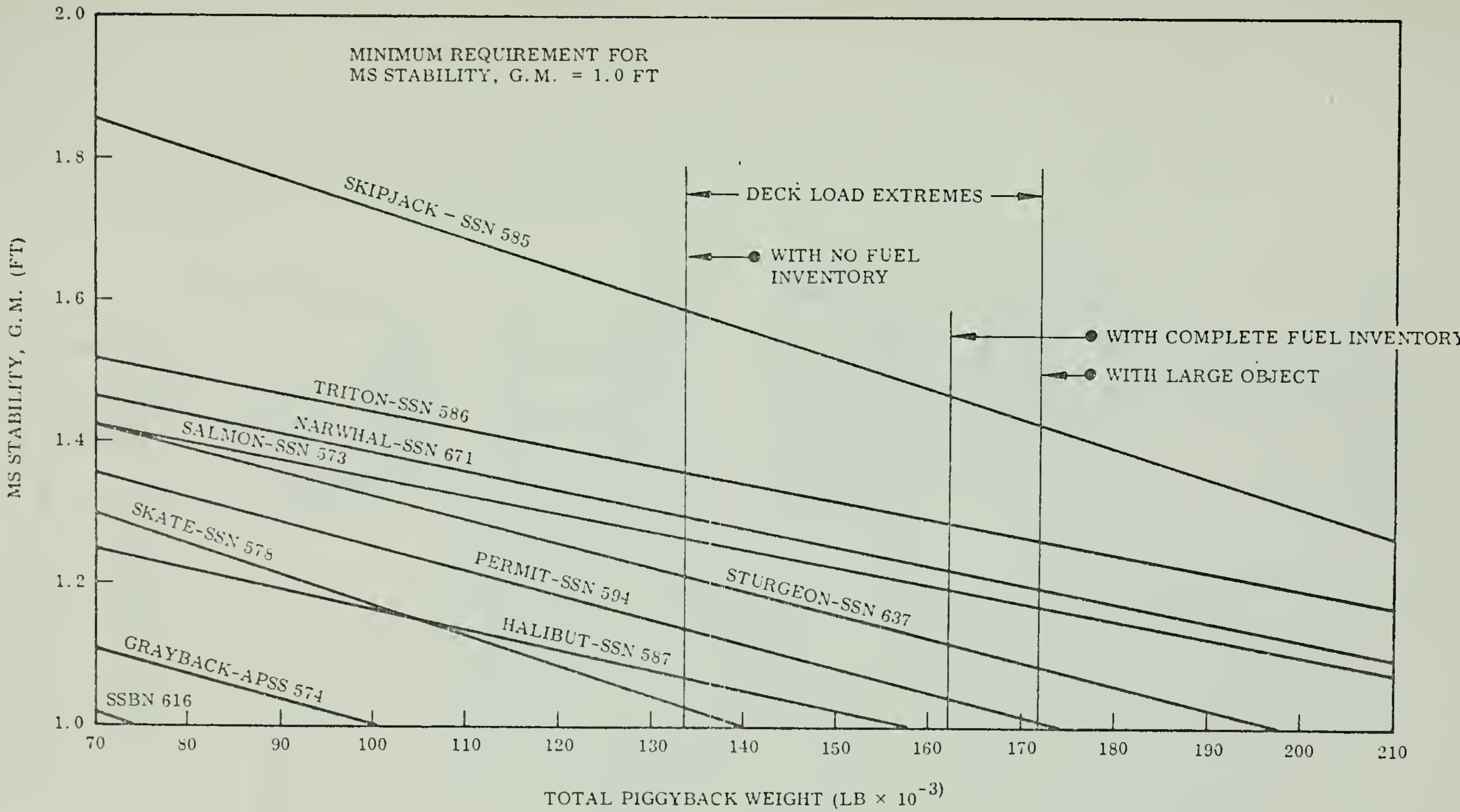
(b) On-Site Support:

[1] Nuclear Sub: An examination of Table III-1 will show that the endurance of deep submersibles ranges from 7 hours to 32 hours and varies with operational speed and mission tasks. In order to optimize the on-site mission to on-site time, it is desirable to have a nuclear submarine acting as mother craft. Busby points out that the greatest natural hazard and hinderance to at sea operations is sea state. There are locations which have acceptable sea states only a small portion of the year [4].

Surfaced and submerged operations from a mother submarine impose limitations on the manned submersible. The surface-stability of the mother submarine/manned submersible system restricts the dry weight of the submersible to about 80,000 lbs. As can be seen in Figure III-2 the limitation does not come from the surfaced stability condition, but from the transient stability condition while surfacing where the submersible weight approaches 160,000 lbs. due to entrained water.

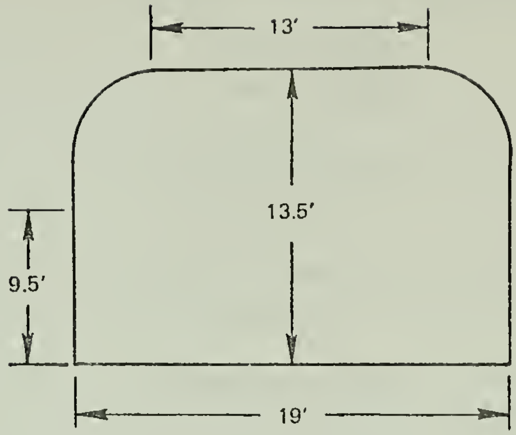
The submerged mating requirement places other restriction on the submersible in terms of sensors, optics, and diving plane configuration.

[2] ASR: Another option for an on-site system is the Navy's ASR. The ASR can operate as a mother craft up to moderate sea states. The ASR capabilities are listed below:



Mother Sub/DSV Surface Stability

Figure III-2



C-5A CARGO COMPARTMENT

Figure III-3

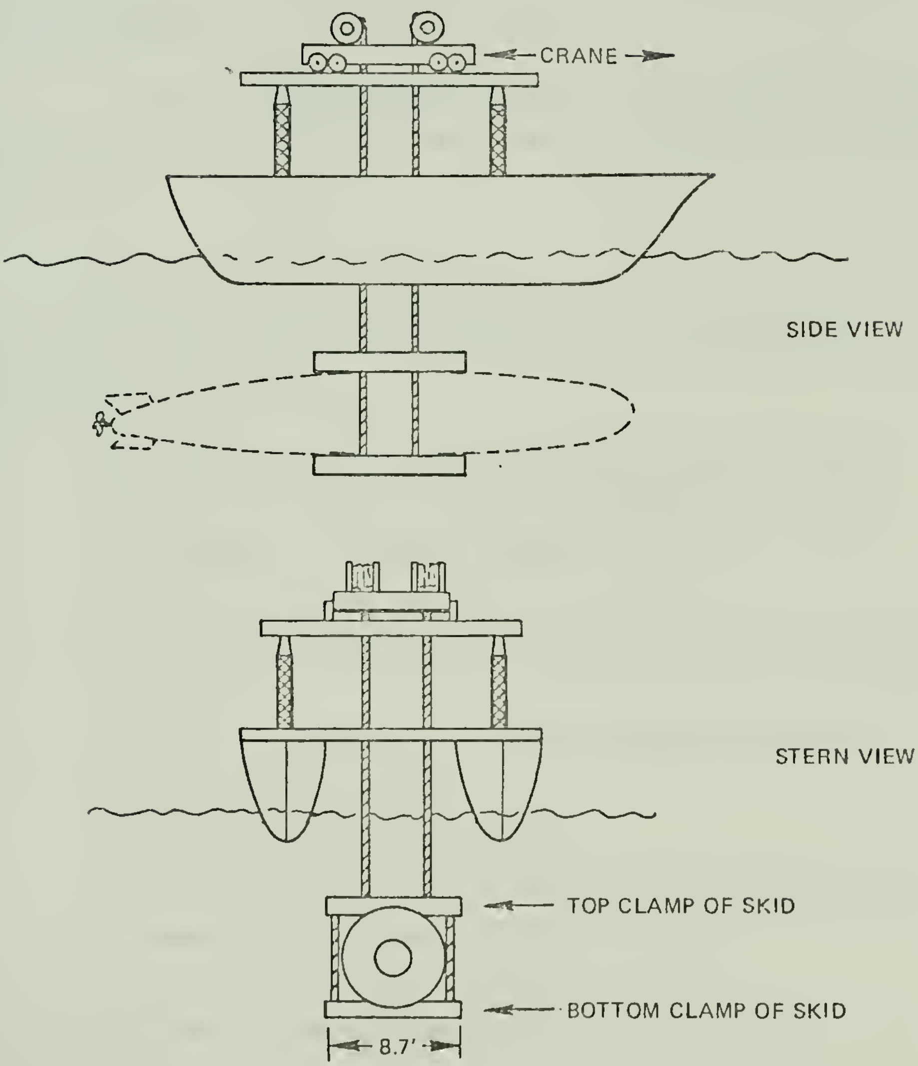


Figure III-4

* Center well opening (clear)

  16 feet wide
  55 feet long

* After cable spread - 9 ft. 8 in. centerline to centerline

* Forward cable spread 8 ft. 9 in. centerline to centerline

* Maximum weight of any vehicle including all water which the ASR can lift is 110, 000 lbs.

If the present ASR configuration is used, the beam of the submersible will be limited to 8.7 ft. due to cable spread and skid arrangement. (See Figure III-4).

[3] Other Support Ships: Other surface support vehicles could be used, but would tend to reduce the mission/on-site time.

So far, the basic mission profile has been established for selected areas. Listed below for sake of continuity is a skeleton mission profile:

* Notification that search mission is required
* Travel to site with appropriate mother craft
* Establish navigation system and grid
* Survey the environment
* Conduct acoustical search and return for evaluation of data
* From acoustical data conduct optical search and target classification
* Analyze optical search data
* Conduct inspection and surveillance missions with possibility of small object recovery
* Conduct work if necessary
* Mother craft/submersible returns to port

d. <u>Summary of Constraints:</u> The above discussions have established some system constraints. These constraints may change as the design progresses, but for now they are as listed below:

Depth:	20,000 ft.
Length Overall:	50 ft.
Maximum Beam:	13 ft.
Air Weight:	80,000 lbs.
Maneuverability:	straight line stability
Sea Transport and Support:	SSN piggyback ASR
Life Support:	Assume 3 men from case history and discussion with operations. 20 hr. mission, plus 20 hr. emergency power and lift support

From the above constraints, the initial design criteria and initial performance criteria were derived, and are listed below:

<u>Design Criteria</u>

*Air Weight

*Maneuverability

*Payload

*Actual search time to on-site time ratio

*Probability of detection

*Good optics

<u>Performance Criteria</u>

* Depth 20,000 ft.

*Overall length up to 50 ft.

*Beam up to 12 ft.

*Transport by surfaced ship

*Air weight less than 30,000 lbs.

e. <u>Design Process:</u> The objective of the baseline design was to establish the feasibility of a vehiclecapable of "wide area search," "work," and "inspection and surveillance." The design process used is outlined in Appendix A. From the design criteria, performance criteria and certification criteria, (See Appendix A), a set of subsystem requirements was developed. For the baseline design , all subsystems except the sensor and work subsystems were selected for minimum in-air weight and held fix throughout the design process. This was done so that an upper limit could be set on "search," "work" and "inspection and surveillance" subsystem requirements. The design process required many iterations resulting in the Basic Vehicle portrayed in TABLE III-2. The outcome of this vehicle trade-off study was that the DSV cannot possess simultaneous capabilities for "wide area search," "work", and "inspection and surveillance." The Basic Vehicle design is described in the following paragraphs.

As can be seen in TABLE III-2, the Basic Vehicle has a bisphere manned pressure envelope with the observation sphere positioned for optimum viewing capability. This arrangement has a number of distinct

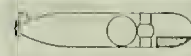
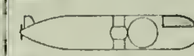
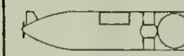
advantages. First, the small observation sphere allows an observer easy access within a small region to many viewing directions, it keeps the viewing function independent of the other crew functions, it provides good visibility for the assigned manipulator work area, and finally it lends itself to optimum hydrodynamic contouring of the submersible.

All of these factors are important for a DSV that is designed for maximum mission time. Crew comfort and efficiency for extended duration missions are highly dependent on crew station layout. Surveillance, inspection and work efficiency again depends upon the observers ability to see what he is doing. Mission duration is directly related to the energy consumption of the vehicle, which is in turn dependent upon the vehicle size and shape. The access sphere group consists of an access sphere, top trunk and bottom trunk and is located aft of the control sphere. The bottom access trunk mates to the mating skirt that will be attached to the mother submarine. A seal will be made when the mother submarine pumps the water from the access sphere and lower trunk. During surface operations, the access sphere group will be blown dry by compressed air from the main ballast tank.

The power plant, the H_2/O_2 fuel cell system, is located in the aft section of the vehicle. Two power modules have been positioned close to the fuel reactant and produce water tanks. The reactant and product water tanks are located so as to minimize any significant shift in the center of gravity as the fuel is converted to water.

Optical and acoustical sensors have been attached to various portions of the submersible. The final choice of the number, type and mounting of these sensors will depend upon the final hydrodynamic envelope.

Table III-2 DSV Alternatives, Modular Concept

	Basic ADDS 20 Vehicle	Basic Vehicle Module Configuration	Fwd Module Veh	Aft Module Veh	Sectionalized - Bottom Module	Sectionalized Top Module	Topside Module Vehicle	
								LMSC Basic DSSV
Size	50' length 10' max dia 8' dia p.s.	same as basic, increased in nose area	50' length 11' max dia 9'6" dia p.s	50' length 11' max dia 9'6" dia p.s.	53' length 10' max dia 8' dia p.s.	53' length 10' max dia 8' dia p s.	present size & arrangement unstable	52' length 11'9" height 9'6" dia p s.
Weight	80,000 lbs	84,000 lbs	110,000 lbs	111,000 lbs	95,000 lbs	95,000 lbs	Heavy	137,000 lbs (based on 30 lb foam)
Direct Viewing	Excellent	Overall reduced good in work area	Satisfactory	Excellent poor in work area	Poor for work area, good for mating	Satisfactory	Excellent, poor in work area	Visibility constraint to viewing beneath vehicle fair
Speed at Constant HP	4.0 knots	3 5 knots	3.9 knots	3.8 knots	4.0 kn[illegible]	4.0 knots	3 8 knots	3.75 knots
Transportation	High speed sub Air	Must remove module for air transport	High speed sub Air	High speed sub Air	High speed sub Air	High speed sub Air	High speed sub Air	Not designed for hi speed sub transport, air transport (C 5A only)
Maneuverability & Stability	Good	Less stable	Good	Less trim control, not as maneuverable	Good	Good	Unstable	Stable, less maneuverable
Mating	Good	More difficult	Excellent	Excellent	Good	Good	Good	Fair
Bottom Sitting	No pods required	Pods required	No pods required	Pods required	No pods required	No pods required	No pods required	No pods required
Work Capabilities	Two manipulators	Wide range depending on module	Wide range, depending on module	Poor for work module, good for lifting	Wide range; Good test vehicle	Excellent test vehicle	Poor	One manipulator
Work Dexterity	Limited	Good below vehicle	Good below vehicle	Poor	Good below vehicle	Good above vehicle	Poor	Limited
Crew Size	3	3	4	4	3	3	4	4
Module Interchangeability	None	Attaches from bottom	Good slide in from front	Good slide in from side	Good slide in from front	Good - slide in from front	Excellent	None
Maintainability	Good	Congested in bow area	Fwd - good hatch - congested aft - good	Fair	Excellent	Excellent	Fair	Was a design consideration
Module Location	None	Good, espec for test bed	Excellent, good for test bed	Excellent for test bed, poor for work module	Excellent	Good; poor for lifting	Poor	None
Lifting Capability	250 lbs	250 lbs	250 lbs	4,000 lbs	4,000 lbs	Has lift capability, but difficult to implement	Very poor	Small object recovery basket
Structures	Minimal	Requires add'l structure to support module	Increased structure fwd not as complicated as nose pod	Increased structuring req'd due to max bending moment at module	Increased structuring fwd req'd	Increased structuring fwd req'd	Poor because of module location	Contains extensive structuring
Cost	Nominal	Slight increase	Slight Increase	Slight Increase	Increase due to sections	Increase due to sections	Increased	Very high

Propulsion is provided by the fixed propeller electric motor drive system shown on the aft section. At this same section are the stern planes which basically control the vehicle's depth. For control during h over, the two horizontal ducted thrusters, one in the forward section, and one in the aft section will generate yaw moments and sway forces, and a vertical thruster located in the mid-section will generate heave forces.

A trim tank is located at each end of the vehicle. Sea water is pumped between these tanks to obtain any desired trim angle up to twenty degrees. The trim system is coupled to the variable ballast system which has a tank located on the port and starboard side near the midship section. During ascent and descent, sea water will be flooded or pumped into these tanks to obtain neutral buoyancy at the operating depths.

The shaded area in the upper mid-body indicates the general location of the buoyancy material. The vehicle weighs 80,000 pounds and is 50 feet long and 10 feet in diameter. The main control sphere has a nominal diameter of eight feet.

The Basic Vehicle was used as a starting point in the design and a comparative evaluation of all other vehicles designed during the feasibility studies. As previously mentioned, this submersible has excellent viewing capability. For work capability it has two manipulators and is considered to have limited work dexterity. For object retrieval it can lift 250 pound objects, but it does not have the capability of accepting internal modules. Due to the arrangement of the major components, maintainability of this submersible is good. The submersible will be capable of a speed of four knots and will have good maneuvering and stability characteristics.

2. DSV Alternatives:

a. Modular Concept: Since a single DSV with complete capabilities was determined unfeasible, a modular design was proposed which would allow the basic vehicle to act as a platform for modular missions. The modules would provide their own power and mission hardware. The power and mission modules could be integrated or separate. Many module configurations were attempted. None of the modular vehicles satisfied all the vehicle design criteria and performance criteria. The best configurations are presented in TABLE III-2. At this stage, the modular concept was rejected. The following paragraphs describe the modular concept trade-offs.

Basic Vehicle Modular Configuration: Six of the many module-vehicle designs studies are illustrated in TABLE III-2. In TABLE III-2, the first module-vehicle studies, used the basic boat design and modified the lower bow section to accept a module. This module contains two grappers, port and starboard, and a manipulator between them. Located to the rear of this module is an indexing tool storage bin containing interchangeable tools. Because of its location below the observation sphere, good viewing for work functions is provided. Adding the module to the basic boat does change a number of its characteristics. The added protrusion increases the vehicle drag and decreases overall stability. The boat weight increases by about two tons and the external structures must be increased in the module attachment area. Mating with this vehicle will be difficult. The module section of the bow must be made with skids capable of bottom sitting. For air transportation the module will have to be removed.

Forward Module Vehicle: In TABLE III-2 a Forward Module Vehicle is also shown. The module subsystem was designed as an integral part of the vehicle with the module section conforming to the hydrodynamic shape of the vehicle. With the module located in the bow, the observation sphere has been eliminated. The pressure hull is now a single nine and one-half foot diameter sphere.

The Forward Module Vehicle provides the best modular configuration capabilities. The bow location offers an unobstructed interface to the forward, side and down looking directions. Small object recovery devices, bottom samplers, and manipulators can also be considered for this module.

Direct forward viewing is not possible, but good viewing under the module can be achieved. Mechanisms that will be lowered from the module should be designed to be located in the aft portion of the module for favorable viewing. For mating a backward looking viewport is provided. The large control sphere provides ample equipment and four crew members.

The Forward Module Vehicle has good hydrodynamic and mating characteristics because of the clean lines of the outer hull. However, the weight has increased by about 15 tons. This increase is primarily due to the overall dimensional growth of the boat to accommodate the large sphere, the increase in buoyancy material required, the weight of the large pressure sphere and the increased exostructure. The hull has a maximum diameter of eleven feet, an increase of one foot over the Basic Vehicle.

Missing in this design is the specific functions of the modules which can either effect the overall design of the boat or

cause mismatches between the boat characteristics and the module characteristics. For example, a boat that has excellent hovering and bottom characteristics for a work module may have poor characteristics for a high resolution search type of sonar module. Also missing from the description of modules is their degree of autonomy. The amount and type of power required from the vehicle by each module can effect the vehicle/module interface and the overall mission duration. Also, controls and displays inside the pressure hull will interface differently for each type of module; a universal interface is not practical. Clever schemes for equipment layouts, penetrators and cabling will be required to accomodate different modules, but always at the expense of complexity and weight, the penalties for interchangeability.

Aft Module Vehicle: These comments on modular design impact on overall vehicle design also apply to the Aft Module Vehicle design which is shown in TABLE III-2. For this design the module area has been located in the lower midbody of the boat. This location is excellent if the boat will be used to lift objects, having a designed lift capability of two tons. Viewing of the module area is rather poor because the rear looking viewport is approximately ten feet from the module. To compensate for this, a pan and tilt mounted TV camera is located just forward of the module.

Since the pressure sphere is located in the bow of the vehicle, excellent forward viewing can be achieved. The forward location of the pressure sphere also changes some of the vehicle characteristics. The blunt nose causes the vehicle to be slower and less

maneuverable. The trim tanks must be relocated further aft than in the previous vehicles, resulting in reduced trim capability and increased power consumption.

Maintainability of this vehicle is complicated due to the concentration of equipments in two areas, namely aft of the pressure sphere and of the module.

Sectionalized Vehicle: In an effort to increase maintainability of the vehicle subsystems, "sectionalized" hull designs have been investigated, in which the various sections may be separated to allow access to equipments contained in that area. TABLE III-2 shows the design for a sectionalized vehicle with the module located in the bottom bow section. This vehicle has been provided with an eight foot diameter sphere which is positioned aft of the access trunk. With the viewport directed towards the access truck area, excellent visual mating capability is provided. Viewing under the module area, however, is rather poor because of the ten foot separation.

This vehicle has a 4,000 pound lift capability and has the same module location advantages previously discussed for the Forward Module Vehicle. Vehicle weight sensitivity to sphere size is quite evident when we compare the weight of this boat, 95,000 pounds, to the weight of the Forward Module Vehicle, 110,000 pounds. Although there is a penalty in structures weight for the sectionalized vehicle, the sphere size is the predominant factor in the weight difference in these vehicles which are similar in other respects. The other difference of course is that maintainability of this vehicle is better because of the sectioning.

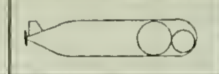
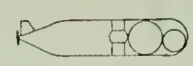
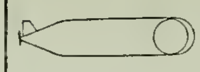
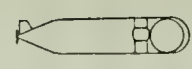
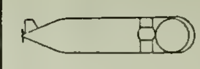
Top Module Vehicle: An attempt to design a vehicle with the module located in the upper midbody is shown in TABLE III-2. Disregarding the fact that the vehicle is unstable, the following things were found. The boat became exceedingly heavy, and the length increased by at least four feet. In order to compensate for the high center of gravity, the other components were located as low as possible, however, this failed to work, and further design on this configuration was not pursued.

b. Single Mission Concept: The major lesson of this design study of module-compatible vehicles was that allowances made in weight, volume, and prime real estate locations for ill defined future module capability: (1) detracted greatly from the basic vehicle capabilities; (2) increased size, weight and cost; and (3) made the design marginal if not unsuitable for submarine transport. Again, the vehicle characteristics were heavily influenced by the mission requirements. As requirements grew, the vehicle grew; and such characteristics as speed, maneuverability, operator direct viewing, and transportability were degraded.

It was therefore decided to further investigate the relationship between mission requirements and vehicle characteristics to indicate what was paid in size, weight, and cost for each anticipated mission requirement. The single mission vehicles considered will be described in the following paragraphs and is summarized in TABLE III-3.

Mini-Sub: The driving motivation for this design was the generation of the requirement for the smallest vehicle that will have mating capability, and will provide a two man crew with a

Table III-3 DSV Alternatives, Single Mission Concept

	SEARCH SURVEILLANCE & INSPECTION (MATING)	SEARCH SURVEILLANCE & INSPECTION (NON MATING)	SEARCH		REVISED BASELINE (8I–SPHERE)	BIG SPHERE	10 KNOT BOAT	WORK BOAT
								
SIZE	35' - 0" Length 9' 4" Dia 6' - Sphere	26' 6" Length 6' - 0" Dia 2 - 4' Pressure Sphere	35' - 0" Length 8' - 0" Dia 4½ - 6" Pressure Sphere		50' - 0" Length 11' - 0" Dia 6 & 8 FT SPHERES	50' - 0" Length 11' - 0" Dia. Mono Sphere 9' - 10"	50 12' 6" Mono 10'	50 12' 6" Mono 10'
WEIGHT	56,895 lbs	27,777 lbs	54,628 lbs		88,348 lbs	89,775 lbs	165,000	165,000
DIRECT VIEWING	Yes	Yes	Yes		Yes - For 2 People	Yes - For 2 People	Yes 2	Yes 2
SPEED AT CONSTANT HP	6½ knots @ 12 HP	4 knots @ 1 HP	8 knots @ 12 HP		3½ knots @ 12 HP	3½ knots @ 12 HP	10	3
TRANSPORTATION	Very Good Air - Land - Water	Excellent Air - Land - Water	Very Good Air - Land - Water		Good Air - Land - Sub.	Good Air - Land - Sea	Fair Yes, Land, Water	Fair Yes, Land, Water
MANEUVERABILITY & STABILITY	Very Good 2 Hor Thrusters	Less Than Optimum 2 Hor Thrusters	Good 2 Hor Thrusters		Fair 2 Hor Thrusters 1 Vert Thruster	Fair 2 Hor Thrusters 1 Vert Thruster	Very Good	Fair
MATING	Yes	SURFACE SHIP ONLY	SURFACE SHIP ONLY		Yes	Yes	Yes (No Surface)	Yes (No Surface)
BOTTOM SITTING	No	No	No		No	No	No	Yes
WORK CAPABILITIES	Camera - TV SLS - Winch	Camera - TV	SLS & Camera TV		Camera - TV SLS 2 Arms Hos	Camera - TV SLS 1 Arm HOS Aft Pan	Camera TV SLS HOS /'It	Super
WORK DEXTERITY	Poor	None	None		Very Good	Fair	NA	Excellant
CREW SIZE	2	2	3		4	4	4	4
MODULE INTERCHANGE-ABILITY	None	None	None		None Provided	Difficult	NA	Excellant
MAINTAINABILITY	Poor As Conceived	Poor	Very Poor As Conceived		Fair	Very Good	Good	Good
MODULE LOCATION	None	None	None		None	Front	NA	Forward
LIFTING CAPABILITY	500 lbs	None	None		1,000 lbs	1,000 lbs or More	1,000 lbs	
STRUCTURES	Good	Good	Good		Good	Good	Good	Good
COST	Nominal	Slight Reduction	Nominal		Slight Increase	Increased	Large Increase	Very High

highly maneuverable vehicle which can rapidly bring them in direct visual contact with the ocean bottom.

This design is not shown. The vehicle is nine feet four inches in diameter, 35 feet long and weighs slightly less than 57,000 pounds.

The six foot diameter pressure hull provides space for a two man crew, and is equipped with three viewports.

The hydrodynamic envelope is configured for low drag. The vehicle can achieve a velocity one and a half times that of the Basic Vehicle.

Although it is not equipped with a manipulator, it does have a winch system, and can lift a 500 pound object.

The price of achieving the small overall size is realized in the small size pressure hull, and in the high packing density of external equipment. These two factors will complicate the detailed designing of this type of vehicle. The maintainability will also be effected.

Non-Mating Inspection & Surveillance Vehicle: Here again, the major design objective was to configure a small vehicle. The function of the vehicle is to provide good visual capability from a small pressure capsule capable of carrying a two man crew. To reduce size, the mating requirement was eliminated, and the pressure hull was sized to a minimum value. Two, four foot diameter spheres were chosen for this hull.

Since the vehicle is designed principally for inspection, all work and sonar search functions were eliminated. Two large viewports

are positioned on the forward pressure hull located in the bow of the vehicle. This will give an observer good forward, downward and upward coverage. Photographic cameras and lights are also integrally mounted in the bow section of the boat.

TABLE III-3 shows the design of this vehicle. Its dimensions are six foot in diameter, twenty eight and a half feet long, and the weight is of the order of fourteen tons.

This vehicle has the same high packing density problems as the Mini-Sub. The four foot bi-sphere space will require further verification as to human factors and equipment mounting adequacy.

A major drawback of this vehicle is that it is designed for surface support, and is therefore, weather dependent, a problem not present with the vehicles that have mating subsystems. A further problem caused by the elimination of surface access trunk is the fact that the access hatch can only be used when the vehicle is removed from the water.

Before the reader gets the impression than an unfeasible design is being pursued, it should be pointed out that the overall intent of these studies was to show the impact of different functional requirements on vehicle design. What has to be weighed, is the fact that if certain features can be sacrificed, a vehicle with direct viewing can be made small and lighter than the Basic Vehicle. The weight savings in this case is almost a factor of three. This adequately points out the sensitivity of a vehicle to functional requirements.

High Speed Search Vehicle: This vehicle shown in TABLE III-3 was designed for high speed, 8 knots sonar search missions. Achieving search area coverage comparable to the coverage of the slower 4 knot boats, and using the same type of power plant, the high speed vehicle drag must be about a quarter of the drag of the slower vehicles. This resulted in a vehicle thirty-five feet long with an eight foot diameter.

For the three man crew and search mission equipments required, a bi-sphere hull was chosen. The bi-sphere (4.5 ft. ID/6.0 ft. ID) lends itself to a narrow hull diameter and is more weight efficient for equipment arrangement and viewport locations than a single sphere.

Because of the dimensional constraints on this vehicle, the mating capability was eliminated. The resulting vehicle has a length and a weight comparable to the Mini Sub, has good high speed maneuvering and control characteristics for search, but must be supported by a surface ship.

Bi-Sphere Work Vehicle: The design criteria for this vehicle was to provide a vehicle with good work capability. This was accomplished by modifying the Basic Vehicle configuration as shown in TABLE III-3. Two retractable manipulators have been located on the lower bow section. The bi-sphere (6.0 ft. ID/8.0 ft. ID) has the observation sphere positioned so that the sphere center is below the center line of the vehicle. This results in excellent viewing of the manipulator work area. The observation sphere contains two sets of viewports which permits simultaneous viewing for the observation sphere's two man crew. This is an important capability during manipulator work periods. During this period, not only two

manipulators but also the boats position must be controlled. Having two people side by side who can look at the same work area simultaneously and who are devoted to manipulator and vehicle control will enhance the overall work capability. An additional two men can also be accommodated in the main sphere.

Review of this boat shows that the outside diameter has increased to eleven feet and the weight has gone up by over four tons. Because of the blunt bow, the speed has gone down by about one-half a knot, and also the control and maneuverability has been reduced.

Big Sphere Vehicle: The design objective here was to generate a vehicle that performs the same type of functions as the Basic Vehicle, but utilizes a single sphere as opposed to the bi-sphere pressure hull. The arrangement for this vehicle is shown in TABLE III-3. The hull has a nine-foot, ten-inch outside diameter and is positioned in the forward section of the bow. Two large viewports with overlapping fields are located to give simultaneous viewing. Because of the reduction of space between the outer hull and pressure hull, a retractable manipulator must be employed. This close proximity to the windows is a definite advantage for work functions. The details of the retraction and storage of the arm, however, have not been worked out, and a high degree of complication is anticipated to accomplish this.

To accommodate the large sphere, the vehicle weight has increased by almost five tons and the vehicle diameter has increased to eleven feet. The speed has been reduced, and the maneuverability and

stability of this vehicle are not as good as they are for the Basic Vehicle which will result in degradations for sonar search and photographic mapping functions.

Ten Knot Boat: For modern steerable beam side scan sonar systems a platform speed of the order of ten knots is required to realize the rapid area coverage of these systems. The Ten Knot Boat was configured in order to determine the sonars impact on the vehicle. The arrangements for this vehicle are shown in TABLE III-3. The main features of this design are an outer hull configured for low drag, a pressure hull increased in size to ten feet, a propulsion system power increased to 150 horsepower, and a full cell system fuel tanks increased for 2000 kilowatt hour capacity. The high power and energy are required to sustain the high velocity for the duration of the sonar search missions.

The major impact of these items on the vehicle are an outer hull that is twelve and one-half feet in diameter, larger than all the previous designs, and a weight that is more than double the Basic Vehicle.

A mating trunk is part of this design, but it is doubtful that the mating mode of operation is feasible. This results from the instability of the mother sub while on the surface with the vehicle mounted piggyback.

Building the propulsion motor to achieve the required horsepower presents its own unique problems.

Finally, the logistics of providing the amount of fuel required for this boat, especially from a mother sub appears prohibitive.

Work Boat: The final boat studied is the work boat shown in TABLE III-3. This boat was configured primarily for the work function. It has five large viewports on the forward section of the ten-foot diameter pressure sphere. Mounted on the bow is a two-arm, rotatable manipulator system. This arrangement of windows and arms allows for work functions to be performed in a large region around the bow of the boat.

To meet peak power demands for the many work functions anticipated, four fuel cells are used for the prime power. The fuel required for this boat has been increased to meet the energy demand of this boat.

The propulsion system for this boat is the same as on the Basic Vehicle. Speed and a high degree of maneuverability are not required for this boat. It has a lift capability of four thousand pounds.

The weight, size, mating, and logistics problems discussed for the ten knot boat are again present in this design. A vehicle designed to perform significant work functions will not be configured as a submersible which can operate from a support mother submarine.

Summary: The exploration phase began with a subproblem objective and is ended with the mating inspection and surveillance vehicle as the "best" system. The design progressed from a feasible baseline design to various configurations attempting to find a configuration which would have the greatest mission capability. In retrospect, Buzby's proposition that "All-Purpose" submersibles cannot accomplish multipurpose missions was found to be correct. The subproblem objective is now, "design a manned untethered submersible capable of inspection and surveillance." This subproblem objective will now act as input to the synthesis phase particularly the preliminary design stage.

IV. Preliminary Design Stage:

A. Introduction: The preliminary design stage begins with the output of the exploration phase. The basic purpose of the exploration phas was to establish the feasibility of the problem objective and to measure the impact of the problem objective on the vehicle. In the previous section, a weight limited design was assumed. The mission impact studies basically assumed a minimum weight criteria for all subsystems. The mission subsystems were varied in order to establish ranges of variation on the mission variables. The result of the exploration phase was the limitation of the subproblem objective to inspection and surveillance and a baseline DSV design.

The objective of the preliminary design stage is to provide a micro level description of the optimum DSV configuration. The design methodlogy for the preliminar design stage is similar in format to the exploration phase, but differs in the generation of alternatives and the subsystems varied. The exploration phase examines the system alternatives whereas the preliminary design stage examines configuration alternatives. In the exploration phase, the mission subsystems were varied but in the preliminary design stage optimum mission subsystems will be held fixed. In the DSV under consideration, the inspection and surveillance subsystem was optimized and fixed for each iteration of the configuration trade-offs. If a requirement or constraint was violated, adjustments were made to eliminate the violation with minimum change to the mission subsystem. The non-mission subsystems were optimized within their individual subsystems and combined into an optimum DSV using the systematic search method.

B. Trade-offs: The DSV under examination is a weight limited design. The approximate weight distribution for various subsystems are listed below:

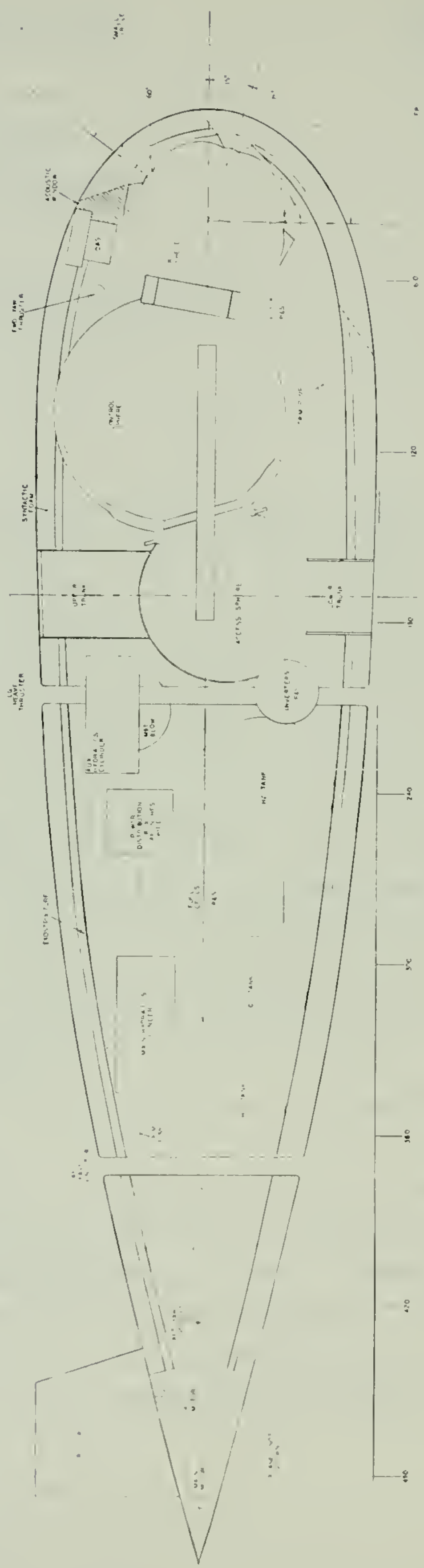


Figure IV-1 DSV Inboard Profile

Weight Groups	$\frac{\text{Group weight}}{\text{Vehicle weight}}$
Manned pressure envelope	.28
Ballast subsystem	.05
Framing/drop weights and weight margin	.08
Propulsion	.12
Control and work subsystems	.05
Electrical distribution	.08
Sensors	.01
Foam	.33

The two largest weight groups are the foam and the pressure envelopes which form between 60%-70% of the entire vehicle weight. The natural inclination is to work on these two weight groups first because of their influence, but it will soon be discovered that these two weight groups can best be optimized at the end of the design. The pressure vessel configuration is basically specified by the subsystem requirements outlined in Appendix F. The material selection process can be narrowed down to a few choices even if the designer has a limited background in materials. For a weight limited design, a lower limit can be set on the allowable stress for a given material by calculating the minimum allowable stress which coincides with the maximum vehicle weight. An upper limit can be established by determining the minimum required notch toughness. The following discussion will use a DSV as an example of the above methodology.

Figure IV-1 is an inboard profile of the final DSV design.

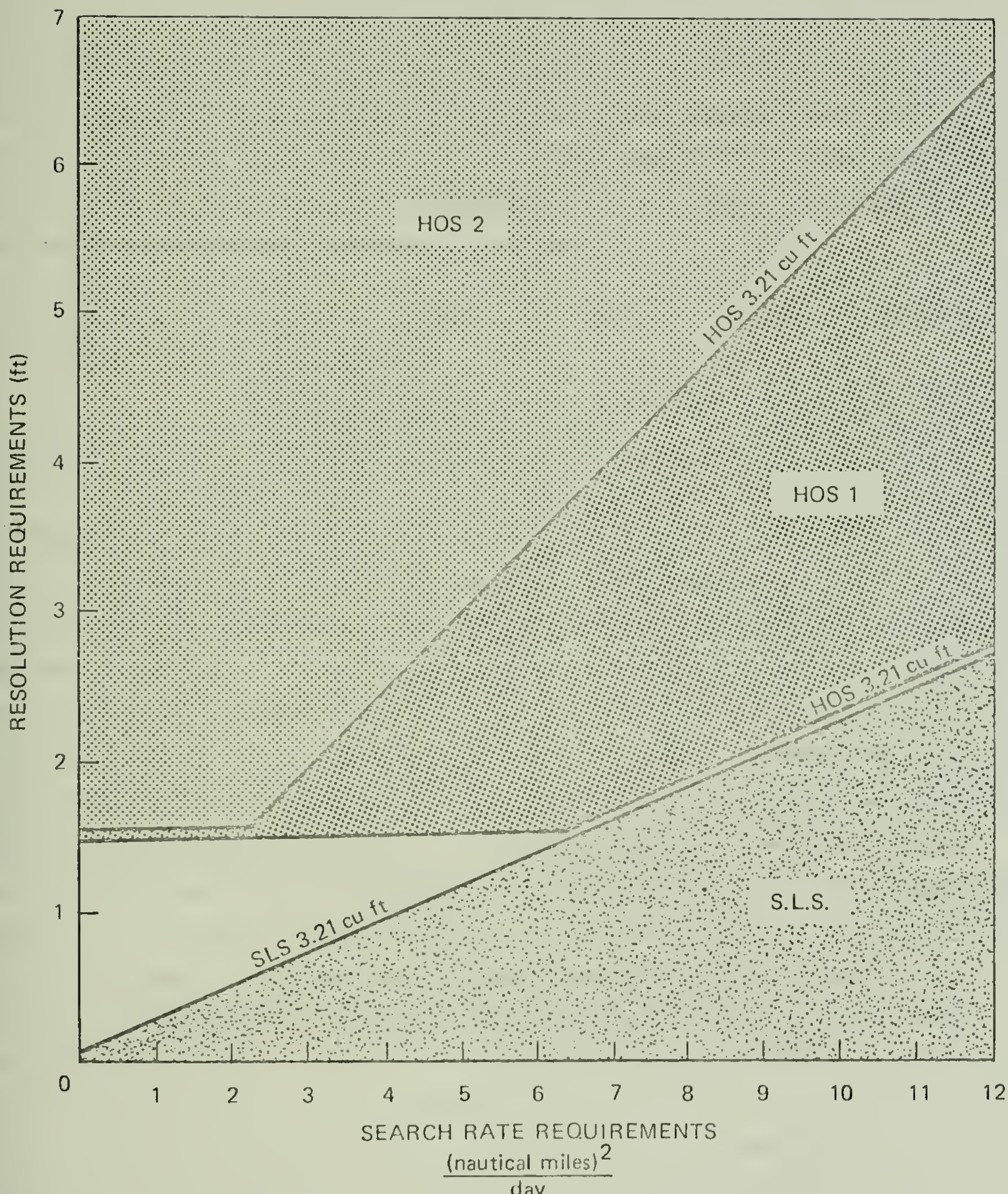
1. Sensors: Appendix B discusses the trade-offs for the "search" mission and "inspection and surveillance" mission.

a. Search Mission: Sonar, magnetic Anomoly and optical sensors were considered for the search mission. From the various trade-offs in Appendix B, it was decided that the "best" type sensors for the search mission were either forward or side looking sonars, SLS.

The selection of either type of sonar depends upon the resolution and search rate requirements. Figure IV-2 represents the basic mission requirements tradc-off. Other factors which may be considered are displays, search patterns, type of resolution elements (mechanically or electrically scanned), range resolution, filter requirements and reliability. For the stated DSV requirements, the SLS was chosen because of its resolution capability and size effectiveness.

The various navigation systems in Appendix B were considered, but a dead reckoning system was chosen because of its ability to receive and process various inputs, outputing the (X, Y) position with respect to an established grid. The dead reckoning system utilizes a doppler sonar and a directional gyro. The doppler sonar with four transducers can compute the body axis velocities, (u, v, w). The directional gyro senses heading and the local vertical from which roll, pitch and true heading are calculated. This vehicle oricnted data is placed into a processor which outputs the (X, Y) position of the vehicle. This (X, Y) position is relative to the grid system chosen. To date, the pilot can input a grid or select a grid from a beacon network. The DSV transmits and the beacons respond, fixing the DSV position relative to the beacons. The mother craft fixes the beacon positions relative to the earth.

b. Inspection and Surveillance Mission: Appendix B divided an inspection and surveillance mission into two basic missions, mapping and close-up photography. Conventional imaging techniques were selected for both missions. In order to conscrve power, strobe lights were used for mapping along with a mapping camera located inside the observation sphere (See Figure IV-1). The mapping camera automatically takes pictures to insure 60% overlap along the mapping path. The camera switching system senses the vehicle attitude and activates the camera switching system when the 60% overlap criteria is satisfied. The normal flying altitude for deep ocean water is about 35 feet. For close-up photography continuous and strobe lamps are both uscd. The majority of the close-up viewing will be through veiwports. Image intensifiers, telescopes and cameras can be used at the inner facc of the vciwports to enhance viewing or photography. The observation sphere has fivc viewports. The four downward looking veiwports are for photography and



Sonar Search Trade-Off

Figure IV-2

mapping whereas the upward looking veiwport is primarily for collision avoidancc during ascent.

2. Work: Appendix C summarizes the trade-offs for a vehicle work system. Small object recovery, large object recovery and work were considered in the trade-offs. It is strongly recommended that the DSV work system be limited to one manipulator capable of lifting objects up to 250 lbs. These objects must have attachment points and be frec. If any task above simple objcct recovery is required, then the whole work system increases radically.

3. Control: Appendix D reviews the possible trade-offs for static and dynamic stability.

a. Static Stability requires that the submcrged DSV be neutrally buoyant and longitudionally balanced at all depths to 20,000 feet. The static stability trade-offs were reviewed in Appendix D, resulting in the ballast system pictured in Figure IV-1. The overall vehicle trim is compensated for by trim tanks forward and aft. The tank size was basically determined by thc vehicle's compressibility and ascent/descent conditions. At the bcginning of thc prcliminary design stage, it was attempted to match the bulk modulus of water and the submersible, but the syntactic foam bulk modulus was too high. Lower bulk modulus 34 pound syntactic foam is not yet available. As a result, the vehicle will require about 650 pounds of additional ballast to obtain ncutral buoyancy and shift weight internal to thc vehicle to maintain longitudinal balance. The ascent-descent ballast conditions require a large trajectory velocity and slope to keep ascent and descent times to a minimum. A computer program was fabricated which input thc basic vehicle bulk modulus and projected mission profile and output required pumping times the ballast conditions. Two problems arosc from this simulation. The first problem involves the large encrgy and power consumption requirements of the trim pumps. This was solved by a mission scenario trade-off which resulted in long pumping periods but tolerable peak power requirements. The second problem resulted from the limited available space forward and the required trim tank displacement. If a toroidal tank were

feasible, it could be placed around the bi-sphere reinforcement ring. At present, it is felt that from a fabricability standpoint the spherical tanks are the most practical, but also the most difficult to locate without violating the hydrodynamic envelope. The final trade-off resulted in two spherical trim tanks forward as shown in Figure IV-1.

b. Dynamic Stability: The dynamic stability of the DSV in Figure IV-1 is excellent. The vehicle's bare hull is dynamically stable without the planes. The free propeller/planes/ thruster system was chosen from a maneuverability standpoint (see Appendix D). The propeller diameter WAS limited to 5 feet and the plane span to 3.5 feet due to the requirement that nothing project beyond the maximum vehicle beam for mating. The propeller rotates at about 100 rpm and is driven by a 7 HP hydraulic motor. The DSV has four ducted thrusters, two vertical and two hoirzontal. The two horizontal thrusters were placed as far forward and aft as possible to reduce the power required for a given moment. It was not feasible to locate a vertical ducted thruster forward so a large thruster was placed as close to the overall vehicle center of pressure as possible so that a heave displacement could be initiated without any pitch. The aft thruster was placed as far aft as possible and is used to produce vehicle pitch.

4. Power: The power system trade-off are considered in Appendix E and the results are presented below:

a. Energy Source: The energy source was specified from two vehicle characteristics, peak power and vehicle endurance. Table IV-1 lists peak power demands of each equipment and its power consumption at each stage of the mission. This type of peak load and endurance can best be assommodated by a fuel cell. The final fuel cell system consisted of two Pratt and Whitney H_2-0_2 fuel cells (20KW each) and enough fuel for a 500 KW-HR mission. The hydrogen and oxygen can be stored as a solid, liquid or gas. The gas storage was chosen due to the relative simplicity of the storage and the submerged refueling capability. A storage pressure of 7500 psi was chosen for the hydrogen tank to reduce the tank size and 4500 psi for the oxygen tank to satisfy the dual stress criteria mentioned in Appendix G.

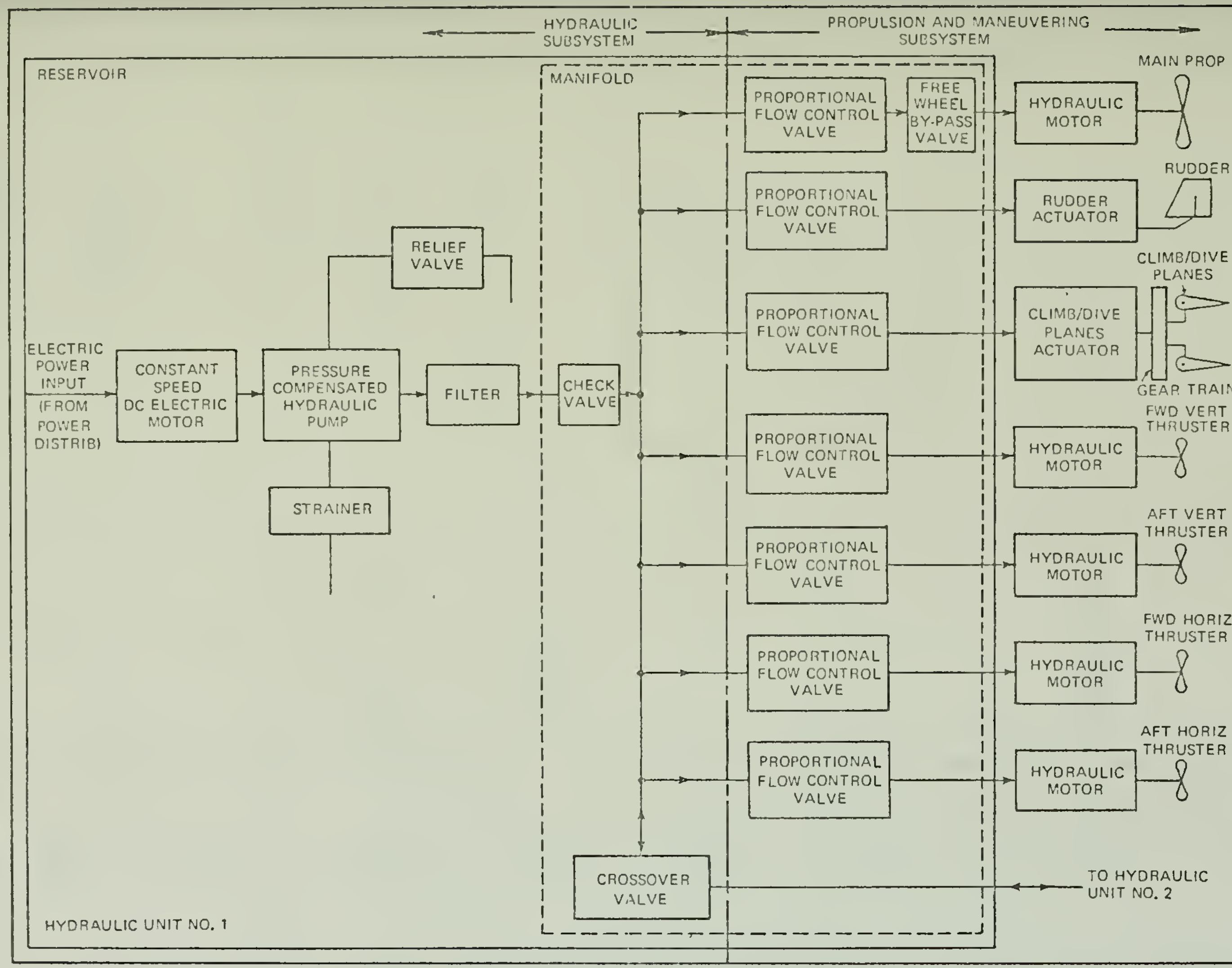
b. Hydraulic vs. Electric trade-offs were considered in Appendix E. A hydraulic/electric system was chosen for the DSV, because of the savings in component size for components located in space critical areas. The weight savings due to components size reduction was lost when the increase in power requirements were considered. The hydraulic system consists of two hydraulic units, the main hydraulic unit and the auxiliary hydraulic unit displayed in Figure IV-1. Either unit can power the entire vehicle, but during normal operations the main hydraulic unit powers the systems in figure IV-3. The auxiliary hydraulic unit operates the manipulator, pan and tilt, valves, and other retractable mechansims.

5. Configuration trade-offs were conducted in Appendix F and the structural analysis of these configurations was considered in Appendix G. Figure IV-4 summarizes the results of the configuration trade-offs. In the past, many configuration trade-off studies have used internal volume as a criterion wich will represent equivalent internal utility. An attempt was made to correlate various configurations with a utility function, but this was deemed as inefficient with regards to time. Therefore, the required pressure envelope equipments were arranged for the configurations in Figure IV-4 to determine the required internal dimensions. The computer program mentioned in Appendix G was used to optimize the weight to displacement ratio for the governing failure criteria. The final configuration selection was made primarily from the pressure envelopes in-air weight, veiwport utility and the pressure envelope effect on the hydrodynamic envelope. The resulting configuration was a bi-sphere, with a 78 inch inside diameter control sphere and a 56 inch inside diameter observation sphere.

6. Material Selection trade-offs have been considered in Appendix H. Steel, titanium, aluminum, glass and GRP were considered and their effect on the DSV weight was analyzed. Lower bound stress levels were established and are listed below:

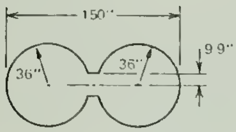
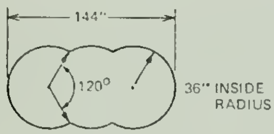
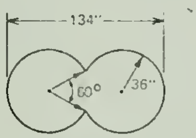
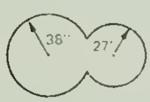
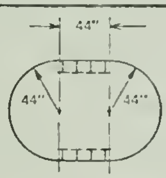
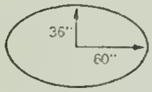
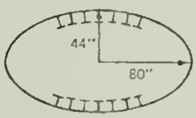
Table IV-1 Power Profile

EQUIPMENT	Peak Power Bus. No. 1 kW	Peak Power Bus. No. 2 kW	Actual Power kW	Predive Check 0.25h kWh	Demate 0.25h kWh	Descent 0.5h kWh	Bottom Trans. 0.5h kWh	Sonic Search 13h kWh	Ascent 2h kWh	Rendez 1h kWh	Mate/Retrieval 0.5h kWh	TOTAL kWh
DOPPLER		0.200	0.200	0.050	0.100	0.400	0.100	2.600	0.400	0.200	0.100	3.950
ATTITUDE REF		0.135	0.135	0.034	0.068	0.270	0.068	1.755	0.270	0.135	0.068	2.668
PRESSURE GAGE		0.025	0.025	0.006	0.013	0.050	0.013	0.324	0.050	0.025	0.013	0.494
PROCESSOR & DISPLAY		0.080	0.080	0.020	0.040	0.160	0.040	1.040	0.160	0.080	0.040	1.580
STATE DISPLAY		0.070	0.070	0.019	0.035	0.140	0.035	0.910	0.140	0.070	0.035	1.384
TV MONITOR		0.060	0.030	0.008	0.015	0.060	0.015	0.390	0.060	0.030	0.015	0.593
ALARM SYSTEM		0.030	0.030	0.008	0.015	0.060	0.015	0.390	0.060	0.030	0.015	0.593
SHIP CONTROL		0.020	0.020	0.005	0.010	0.040	0.010	0.360	0.040	0.020	0.010	0.495
C&D POWER SUPPLY		0.100	0.100	0.025	0.050	0.200	0.050	1.300	0.200	0.100	0.050	1.975
PLOTTER		0.085	0.085	0.021	0.041	0.170	0.041	1.105	0.170	0.085	0.041	1.674
FUEL CELL CONTROL		0.050	0.010	0.003	0.005	0.020	0.005	0.130	0.020	0.010	0.005	0.198
UNDERWATER TELEPHONE		0.495	0.150	0.037	0.075	0.300	0.075	1.950	0.300	0.150	0.075	2.962
UHF			0.005	0.001								
INTERCOM		0.010	0.010	0.002	0.005	0.020	0.005	0.130	0.020	0.010	0.005	0.197
SLS & RECORDER		0.445	0.445	0.011				5.785				5.796
HOS		0.196	0.196	0.049	0.098	0.392	0.098	2.548	0.392	0.196	0.098	3.871
TV CAMERAS		0.150	0.150	0.026	0.075	0.060	0.015	0.390	0.060	0.150	0.075	0.851
MAPPING CAMERA			0.025									
LIFE SUPPORT		0.035	0.035	0.009	0.018	0.070	0.018	0.455	0.070	0.035	0.018	0.693
INTERNAL LIGHTING		0.150	0.150	0.037	0.075	0.300	0.037	1.950	0.075	0.150	0.075	2.699
AUXILIARY HYDRAULICS (incl. var. Ballast)		16.500	16.500		0.250	33.000	3.000	15.000	33.000	7.500	0.250	92.000
PAN/TILT UNIT	0.400		0.080	0.020	0.040					0.200	0.200	0.460
PAN UNIT	0.050		0.010	0.003							0.005	0.008
STROBES	9.000		3.400									
QUARTZ IODIDE	2.800		1.870		1.400		1.400	14.151	1.600	0.935	0.935	19.161
METAL VAPOR	3.000		2.880			2.880	1.440	5.800		1.440	1.440	13.000
PROPULSION HYDRAULICS	16.500		5.000		2.500	4.500	1.125	55.500	4.500	2.500	5.000	75.625
MANIPULATOR HYDRAULICS	7.500		7.500									
EQUIPMENT COOLING	0.500		0.500	0.125	0.250	1.000	0.250	6.500	1.000	0.500	0.250	9.875
POWER LOSSES		0.200	0.200	0.050	0.100	0.400	0.100	2.600	0.400	0.200	0.100	3.950
TOTAL	32.200	19.036		5.569	5.278	44.492	7.955	123.063	42.987	14.751	8.918	248.013



Hydraulic System

Figure IV-3

CONFIGURATION	W/D	THICKNESS	WEIGHT
 SINGLE SPHERE	0.88	2.19"	23,648.
 CONNECTED SPHERE	0.93	1.00" CYLINDER 1.42" SHELL	16,123.
 NESTED SPHERE	0.93	1.42"	11,344.
 EQUAL BI-SPHERE	0.93	1.42"	12,902.
 UNEQUAL BI-SPHERE	0.96	1.58" LEFT 1.14" RIGHT	12,751.
 CYLINDER HEMI ENDS	0.93	t_{ENDS} =1.74" $t_{CYLINDER}$ =2.48	22,697.
 PROLATE SPHEROID	0.88	2.07"	11,472.
 STIFFENED PROLATE SPHEROID	1.05	2.36"	26,886.

NOTE: 1. ALL DIMENSIONS ARE INSIDE DIMENSIONS.
2. WEIGHT OF REINFORCEMENTS ARE INCLUDED.

Figure IV-4 HY 180 Steel Configuration Trade-Off

Steel	155 ksi
Tit anium	95 ksi
Aluminium	65 ksi

From this study, aluminium was rejected since the maximum availabe stress for aluminium is 60 ksi. Glass and GRP were not bounded by the vehicle weight.

In the establishment of the upper bound stress level, it was determined that present glass and GRP do not possess sufficient notch toughness. Notch toughness for steel and titanium are listed in Appendix H.

The final trade-off was reduced to steel (155 ksi-200 ksi) and titanium (95 ksi-115 ksi). Hy 180 steel was chosen as the best material from an in-air wieght, stress corrosion cracking, hydrogen embrittlement, material property degradation with thickness, and commercial availability viewpoint.

The vehicle will use Hy180 for all pressure vessels except the fuel cell hydrogen tank. From the references in Appendix H, it was determined that Hy100 should be used due to hydrogen embrittlement.

C. Vehicle Configuration: The final vehicle configuration requires many iterations and arrangements of the component locations and interactions. The trade-offs presented to date do not complete the preliminary design but they do provide the designer with enough data to utilize the skills of experts in each field.

V. Conclusion:

Effective communication is probably the single most important factor in the proposed design methodology. With it the design can proceed efficiently towards an optimum. Without it, the design rapidly approaches a summation of optimum subsystems which may or may not be an optimum design.

In the exploration phase, the multimission concept was shown to be detrimental to the successful completion of the DSV mission. As a result, the inspection and surveillance mission was selected and used as the revised mission requirement for the preliminary design stage. The output of the preliminary design stage iterations was a bi-sphere manned pressure envelope made of Hy 180 steel and a set of feasible subsystem alternatives. The detailed design of each subsystem should be done by experts in that field, but the synthesis of the vehicle subsystems should be a cooperative effort of all subsystem heads.

VI. References

1. "A Design Methodology for Ships and Other Complex Systems," Phil. Trans. of Royal Society, A.273, 85-98 (1972), Mandel and Chryssostomidis, 1972 also in Technology of Ship Design Symposium, Naval Ship Engineering Center, Feb. 1972.

2. D'Arcangelo, A.M., "Ship Design and Construction," SNAME, 74 Trinity Place, New York, N.Y., 1969.

3. Busby, R. F., "Design and Operational Performance of Manned Submersibles," ASME Winter Annual Meeting and Energy Systems Exposition New York, N.Y., Dec. 1-5, 1968.

4. Busby, R. F., Hunt, L. M. and Rainnie, W. O., "Hazards of the Deep" Parts I, II, III, Ocean Industry Vol. 3, No. 7, July 1968, Vol. 3, No. 8, Aug. 1968, Vol 3, No. 9, Sept. 1968.

Appendix A
Design Process

Establishment of a pressure boundary in a manned submersible is the driving point in most deep submersible designs. Shallow submersibles tend to place most of the equipment within the pressure boundary, but as depth increases the pressure boundary design becomes increasingly more influencial due to materials, fabrication and structural design. In deep submergence, there has been a trend to maximize the amount of machinery and equipment, that is free flooded, pressure compensated, or contained in a separate pressure vessel outside the manned pressure boundary. The sole exception to this trend in deep submergence is the Aluminaut.

Figure A-1, graphically describes the design process which will be used during the exploration phase. A presentation of the design process will be concurrent with the DSV design.

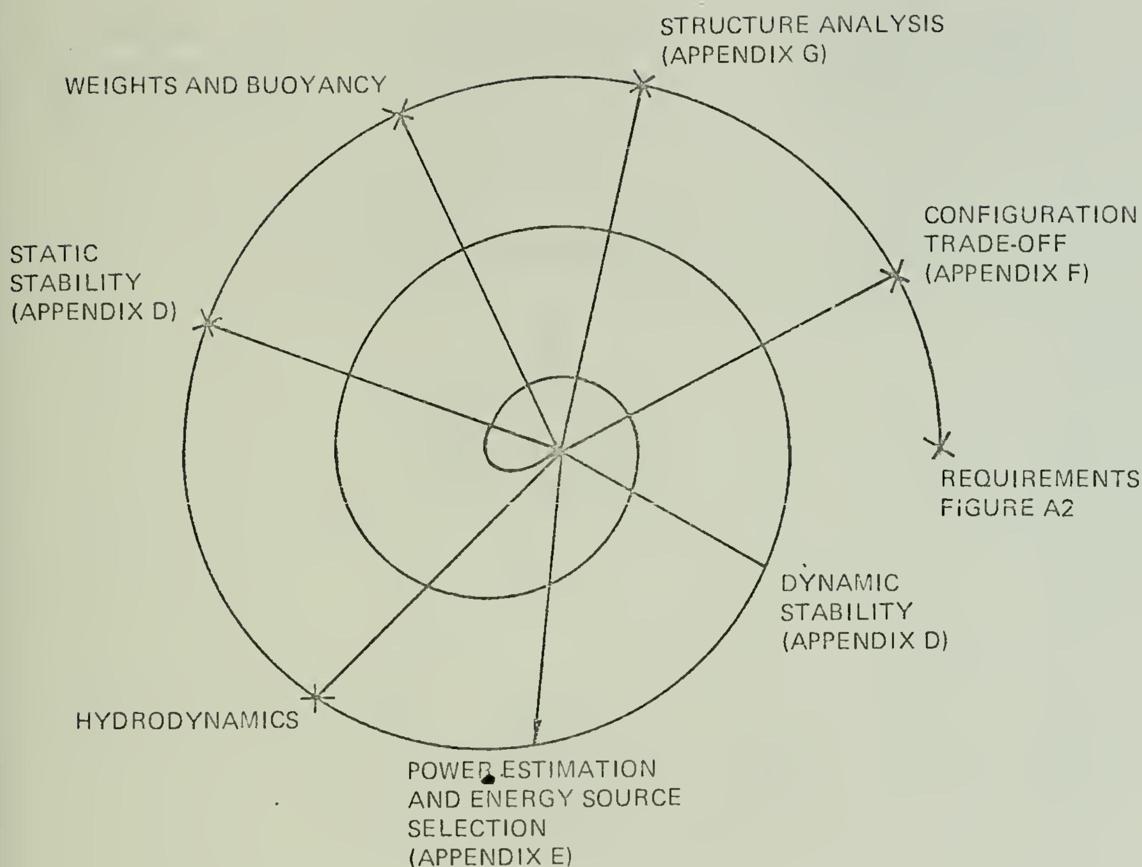
From the outset of a DSV design, it is important to consider the impact of certification on the entire design. As seen in Figure A-2, the MR and MP are inputs to the selection of certification criteria which is, in turn, input to the system and subsystem specifications. In this design methodology, the system and subsystem requirements will be established from our design criteria, performance criteria, and certification criteria.

There are two fundamental certification guidelines for manned submersible design. The ABS Certification Manual,[1] and the Navy's Deep Submergence System Certification Manual[2]. The ruling certification manual for the example DSV is reference 2. Reference 3, provides a check list for certification subsystem requirements and is recommended as a starting point for a submersible design.

The basic subsystems common to most deep submersibles are listed below:

Power subsystem
Ballast subsystem
Sensor subsystem
Control subsystem
Work subsystem
Environmental subsystem

Associated with each of these subsystems will be subsystem requirements which satisfy our subproblem objective. Some of these subsystems and



DESIGN SPIRAL

Figure A-1

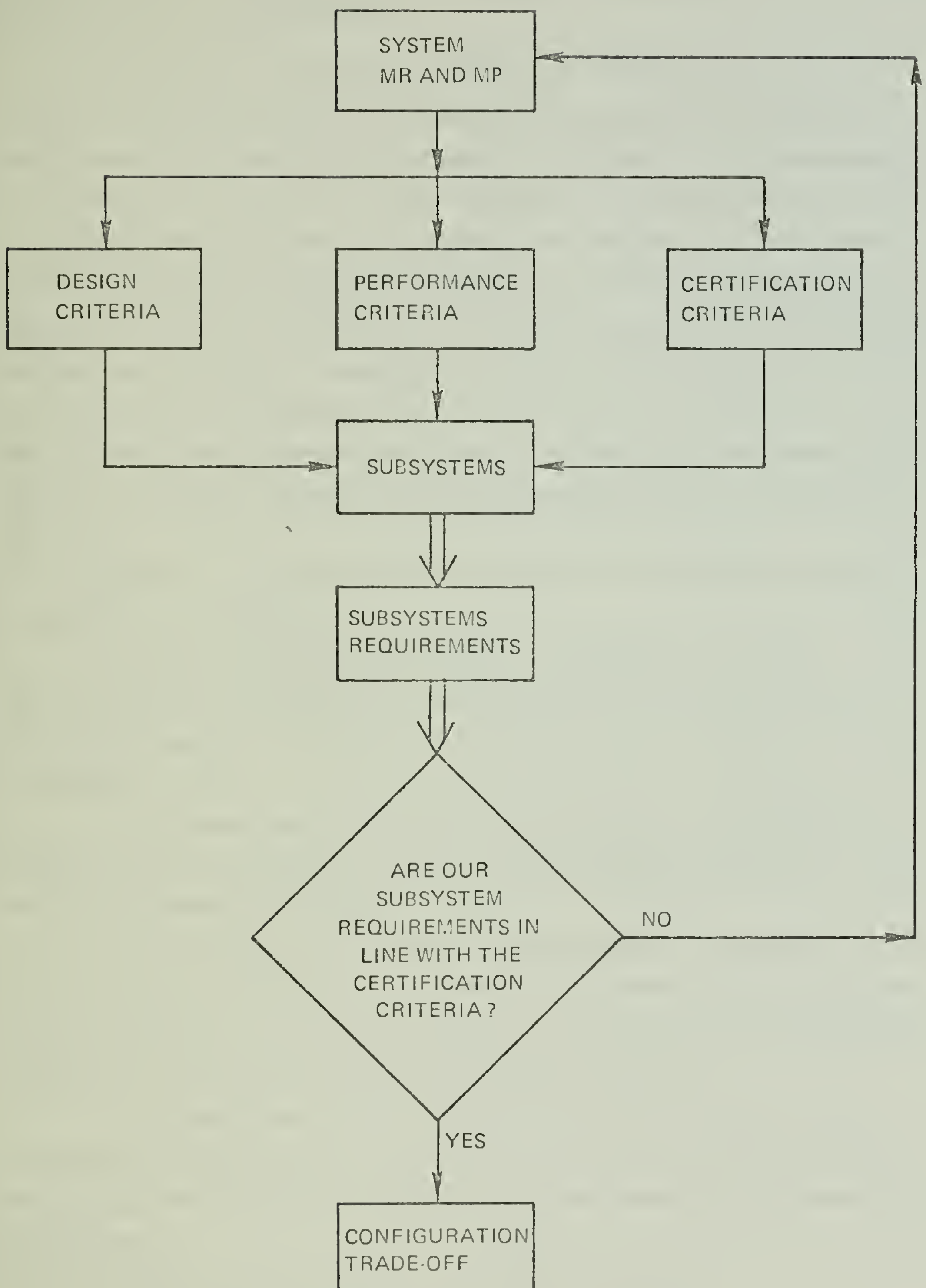


Figure A-2 Subsystem Requirements

subsystem requirements will be specified by the certification criteria[2].

Listed below are subsystems and subsystem requirements which fall within the scope of certification:

> The subsystem certification scope of a DSV is a list of those subsystems required to insure and preserve the safety and well-being of its operators, divers, or occupants. It encompasses "life critical elements" of all subsystems, components and portions of the DSV including normal operating and maintenance procedures which are needed to insure the continuous physical well-being of the operators, divers, or occupants. It also encompasses those emergency systems and procedures required to return them safely from any depth, down to the maximum operating depth, back to the surface, or to a submerged base under abnormal conditions following any non-catastrophic accident or casualty which precludes continued normal operation of the DSV.
>
> Components, subsystems, and portions of the DSV that require certification include:
>
> a. All components, subsystems and portions of the DSV which, through malfunction or failure, could prevent the return of the DSV operators, divers, or occupants to the surface or to a submerged base.
>
> b. All components, subsystems and portions of the DSV required to keep operators, divers, and/or passengers safely on the surface following any ascent.
>
> c. All components, subsystems and portions of the DSV provided to rescue personnel from the DSV and return them to the surface, support ship, a submerged base; or, in the case of hyperbaric chambers, to ambient conditions outside the chamber.
>
> d. All subsystems and components including temporary test equipments affecting trim and stability conditions, both surfaced and submerged, which could prevent the safe recovery of personnel from a DSS.
>
> e. Normal and emergency operating procedures.
>
> f. Maintenance procedures.
>
> Examples of areas of a DSV involved in system certification are

listed below:

It is recognized that individual DSV designs will vary to the extent that no single list can adequately define the subsystem certification scope for all cases. The following list of areas which could require certification is given for purposes of illustration and should not be considered all inclusive or universally applicable.

a. The pressure hull, pressure vessels, hard structure and appurtenances.

b. The ballast systems which can be used for maintaining adequate freeboard when operating a submersible capsule or habitat on the surface or that can be used for emergency surfacing.

c. Jettisoning and emergency blow systems which can be used to return the DSV to the surface in the event of an emergency.

d. Normal and emergency life support subsystems which provide an acceptable atmosphere to the DSV personnel.

e. Non-compensated equipment, subject to pressure, which may implode or explode.

f. Release devices for external appendages.

g. Fire fighting devices or systems.

h. Communication subsystems that enable personnel utilizing the DSV to communicate with support personnel.

i. Monitoring detecting devices which will be depended upon to assure that the DSV does not exceed specified limits.

j. Obstacle avoidance subsystems, such as active sonars, fathometers, passive sonars, TV viewing systems, optical viewing devices and periscopes.

k. In the case of a submersible, the propulsion subsystem may be included when the submersible operates under or near overhangs, cliffs, canyons, etc.

l. Accessibility to vital equipment which actuates recovery systems or is involved in life support systems. These should include systems which may be required for recovery of personnel from the DSV following a casualty.

m. Flotation or buoyancy systems whose failure or inadequacy could prevent the return of the DSV personnel to the surface.

n. Electric power systems which include internal and external

electrical protective devices whose failure could result in malfunction of a critical component or system.

o. Written operating and maintenance procedures including pre-and post-dive procedures for the particular DSV.

p. Support ship handling system components such as cranes, brakes, and cables when the DSV is handled with personnel aboard.

q. Components, systems, and portions of the DSV that protect the DSV personnel directly or indirectly against the effects of accidents and hazards.

Certification and non-certification subsystem requirements are listed below according to the respective subsystem.

A. Power System

1. Propulsion for speeds compatable with mission (reference is made here to forward or asternpropulsion) including:

(a) energy source

(b) power transmission

(c) propulsion

2. Power sources for

(a) work subsystems

(b) control subsystems

(c) ballast subsystems

(d) sensor subsystems

(e) atmosphere subsystems

B. Ballast System

1. Surfacing Capability and safety while on surface

2. Neutral Buoyancy Control

(a) Payload

(b) Compressibility effects

(c) Various loadings Conditions

(d) Change in water density

3. Trim Control

(a) Surfaced

(b) Submerged

4. Emergency Ballast Control

C. Sensor System

1. Navigation "See Busby's paper on Navigation"
   (a) Position
   (b) Course
   (c) Speed
   (d) Time
   (e) Depth
   (f) Altitude

2. Communications with Mother Craft
   (a) Surfaced
   (b) Submerged

3. Visibility
   (a) Optical mapping or close-up
      1. Photography
      2. Video systems
      3. Direct viewing
      4. Mating
   (b) Acoustical
      1. Search
      2. Mating
      3. Collision avoidance

D. Control System

1. Surfaced
   (a) obstacle avoidance
   (b) maneuverability commensurate with surface operations.

2. Submerged
   (a) Low speed (define) Yaw Heave
      Pitch Sway
      Surge

      For
      1. Mating
      2. Close-up photography
      3. Collision avoidance
      4. Hovering

(b) Dynamic Stability (above low speed)

1. Straight line stability

2. Maneuverability

3. Obstacle avoidance

E. <u>Work System</u>

1. Salvage

(a) Small object recovery 250 lbs

(b) Large object recovery 250 lbs

2. Mechanical Work

(a) Must define

F. <u>Environmental System</u>

1. Life Support System

(a) Oxygen supply

(b) Carbon dioxide removal

(c) Emergency breathing

(d) Contamination control

(e) Temperature and Humidity Control

(f) Instrumentation

(g) Food and water supply

(h) Waste Management

2. Fire extinguishing System

(a) Must be compatible with human life

References

1. "Guide for the Classification of Manned Submersibles," American Bureau of Shipping, New York, N.Y., 1968.

2. "Material Certification and Criteria Manual for Manned Non-Combatant Submersibles," Naval Ship Systems Command, Navships 0900-028-2010, Sept. 1968.

3. "Pre-Survey Outline Booklet for Manned Non-Combatant Submersibles," Naval Ship Systems Command, Navships 0900-028-2020, Sept. 1968.

Appendix B

Sensors

1. Introduction: This section will briefly discuss the "search" and the "inspection and surveillance" missions. Within the past ten years, the "search" mission has received a great deal of attention, due to incidents like the Thresher, Scorpion, Alvin and Palomares. As a result, much information has been published which can greatly reduce the work load of the designer. The "search" mission discussion will briefly define the "search" problem and list some helpful references. The "inspection and surveillance" mission has received little attention and therefore technical information and design trade-offs are difficult to obtain. The "inspection and surveillance" discussion will basically propose a sensor subsystem design methodology and list references which will help to provide technical information and design trade-offs.

2. Search Mission: "Search", may be defined as the act of locating an object "on the ocean floor." "On the ocean floor," was placed in quotes since the precise description of the environment and object is essential to the efficient design of a DSV with a search mission. The ability of a DSV to perform a search mission can be measured by the vehicle's probability of detection for a specified mission. A problem with the specification of the probability of detection is that it does not imply a mission duration. This problem was prominent in the Palomares search[1] and was circumvented by the specification of a search effectiveness probability, SEP. The following statement is from reference 1:

> "In conducting a large-scale operation, it is important to have a measure of effectiveness which will allow past search efforts to be analyzed and evaluated, and which will provide a

sound basis for determining future allocation of effort. The measure of effectiveness employed during the search phase of Salvops Med was search effectiveness probability, defined as the probability that if a target were in a specified area, then it would have been detected and identified with a specified amount of search effort. The SEP provided means for organizing and placing into perspective the search and identification of data relating to all aspects of the operation."

he degree of correlation between the SEP and probability of detection is ighly dependent upon the accuracy of the navigation subsystem.

The following references are provided since it is felt that their coverge of the design trade-offs is sufficient for the mission specification.

"Instrumentation for a Deep Submergence Search Vehicle," MIT Instrumentaion Laboratory Report E - 2446, Sept. 1969.

"This report investigates the instrumentation required for a DSV capable of search missions of the ocean bottom at 20,000 foot depth for objects ranging in size from a basketball to a nuclear submarine. A general conceptual framework for submersible design is given emphasizing a functional breakdown of equipment and software. Functional requirements are derived from the stated search mission objectives for Search Control, Navigation, and Perception Functions. Hardware alternatives for search sensors, processors, and displays are delineated. Digital processing needs with a computational profile are given. A tentative list of instrumentation equipment is proposed. The last three chapters are devoted to the pressure sphere internal arrangement, power profile, power management and thermal control."

*"DSSV Final Report, "Lockheed Missile and Space Company Report LMSC-T-14-68-/ Part VI., Section 10.

"The purpose of this effort was to conduct an investigation of the parameters of the search missions which could ultimately impact the design of the vehicle. Various search sensors and search methods were considered. The search missions were divided into small-object, large-object and average-target search."

*Myers, J. J., Holm, C. H. and McAllister, R. F., "Handbook of Ocean and Underwater Engineering." McGraw-Hill, 1969.

"Pages 3-40 through 3-57 provide a good description of magnetic anomaly principles and search methods."

*Craven, J. P., "The Design of Deep Submersibles," SNAME Paper No. 9, Diamond Jubilee International Meeting, New York, N.Y. June 18-21, 1968.

"This reference was included since it lists the DSRV search and navigation subsystem components and their basic functions. A review of the sensor suits on board operational submersibles may be helpful from a feasibility standpoint."

The following references are provided since it is felt that their scope is sufficient for a navigation subsystem specification.

*Busby, R. F., "Ocean Surveying from Manned Submersibles," Marine Technology Society Journal, Vol. 3, No. 1 Jan. 1969.

"This reference is an outstanding presentation of the feasible navigation systems which could be employed by manned submersibles engaged in ocean bottom surveying."

*McCloskey, L. M., "Integrated Navigation System Design for Deep Submersible Vehicles," MIT Instrumentation Laboratory Report R-594, Oct. 1967.

"This reference describes the design of a navigation subsystem which makes optimum use of all sensor information and the recent history of vehicle motion to provide the best indication of vehicle position and velocity. The mathematical theory underlying the development of the navigation subsystem is discussed. The computations and equations necessary to implement the system are discussed and summarized for easy reference."

*Lowenstein, C. D., "Position Determination Near the Sea Floor," Marine Physical Laboratory of the Scripps Institute of Oceanography Report MPL-U-22/67, University of California, San Diego.

The search mission significantly impacts the DSV design and should receive careful consideration. The search sensors and navigation system should be tailored so that one fulfills the needs of the other. It is recommended that at the outset of a DSV design study, a careful analysis be made of the system objectives to establish the feasibility of the search mission.

3. Inspection and Surveillance Mission:

a. Introduction: From the search mission trade-offs, it was decided that optical sensors would not be used for a search mission, but would instead be employed as a high resolution, high information recording system. In this context, two basic optical missions were proposed. The first optical mission would be to map an area already located. The vehicle would return to the mother craft, process the mapping information, and then specify future mission profiles. The second optical mission would be close-up photography.

(1) The mapping mission requires a complete coverage of a specified area. In order to optimize on-station time, it is assumed that the object has already been localized to a small area--perhaps several acres.

Once detecting the object, the vehicle would proceed to optically map the specified area. A map intended for photo interpretation utilization should have the following information.

Optical requirements for a good mosaic are listed below:

a. Submersible requirements:

(1) Straight line dynamic stability

(2) Maneuverability at low speeds

(3) Position known to plus or minus five feet

(4) Altitude known to be within six inches

(5) Reliable collision avoidance sonar

(6) Maximum rc¹l and pitch while taking pictures, 25°. (This comes out of the photo-interpreter's ability to rectify a photo.)

b. Camera requirements:

(1) The following information should be recorded on the film:

(a) Real time

(b) Altitude

(c) Depth

fixes vehicle's position in the in the Z-direction and helps the photo-interpreter to determine bottom slope

(d) True Course

(e) Speed. over bottom

helps to fix vehicle location relative to navigation system

(f) Vehicle attitude of inertial aim (local vertical)

(g) Vehicle position relative to navigation system

(h) Optical alpha reading

(2) Need automatic focus to give

40% overlap on sides

60% overlap Fore and Aft

(3) For optimum stereo pictures, the following ratio should be

satisfied:

$$\frac{b}{h} = .6$$

b = camera base-line separation
h = camera altitude

(This comes from aerial photography requirements.)

(4) Automatic strobe control, possibly run off an alpha meter.

(5) Stereo: a fixed camera base-line is highly desirable if possible.

The mission requirements and mission profile can be translated into vehicle constraints, once a particular photographic technique is chosen for mapping.

(2) The close-up mission basically involves recording objects of interest, but a problem arises in recording what the pilot or observer sees, and ensuring adequate image information and object coverage. In the past, close-up missions have basically been failures, not from the standpoint of picture quality as much as coverage. A pilot or observer will see an interesting object and photograph it with ten times the required coverage, at the expense of other more important objects of interest. This over-emphasis on coverage has stemmed from inexperience on the pilot's part, but more from the pilot's uncertainly that the object was actually recorded.

b . Optical Techniques: Within the last decade much work has been done to extend conventional imaging techniques.[2] The basic extended range techniques are listed below:

(1) Range-gating

(2) Polarization discrimination

(3) Volume scanning

Each of these extended range techniques were examined in detail and it was decided that with the present state-of-the-art, these techniques

would not afford the DSV adequate fulfillment of the optical mission requirements. It was also felt that much work could be done with the conventional imaging techniques without expense to the mission requirements.

c. Trade-offs

(1) Location: Both the mapping and close-up mission place certain requirements on the optical system. It is essential to determine early in the preliminary design and possibly in the exploration phase the location of the optical sensors. Listed below are some arguments for locating the optical sensor inside or outside the manned pressure envelope.

(a) Sensors Inside Manned Pressure Envelope:

Advantages

[1] Provides flexibility in camera operation, like camera setting. Various cameras can also be employed.

[2] Easy reloading and possibility of on-site camera repair if film advance mechanism jams. Can also change film for varying water conditions. Inside loading also allows mission repeatability from a mother sub.

[3] Allows the pilot or observer to record what he observes.

[4] Provides a small in-air weight savings, assuming that the same number of viewports are employed.

Disadvantages

[1] Possible image degradation due to viewport deformation.

[2] Cameras and viewports take up inside volume, and may make pilot viewing difficult.

(b) Sensors Outside Manned Pressure Envelope:

Advantages

[1] Camera location becomes less of a problem. Can also use one camera on a pan and tilt to cover an entire hemisphere.

[2] Vehicle certification may be made easier if exterior location reduces the number of viewports.

[3] Concentric dome windows on an exterior camera are thinner than a large viewport. This not only helps reduce optical transmission losses, but also reduces the thick lens problem. The thick lens problem basically states that thick windows remap object space. The remapped object plane not only appears closer but undergoes severe distortion off the central axis. See Figures 8-9.

Disadvantages

[1] A slight weight penalty is sensed from the camera pressure envelopes and the increase in hull penetrations.

[2] Making repeated missions from a submerged mother craft is extremely difficult.

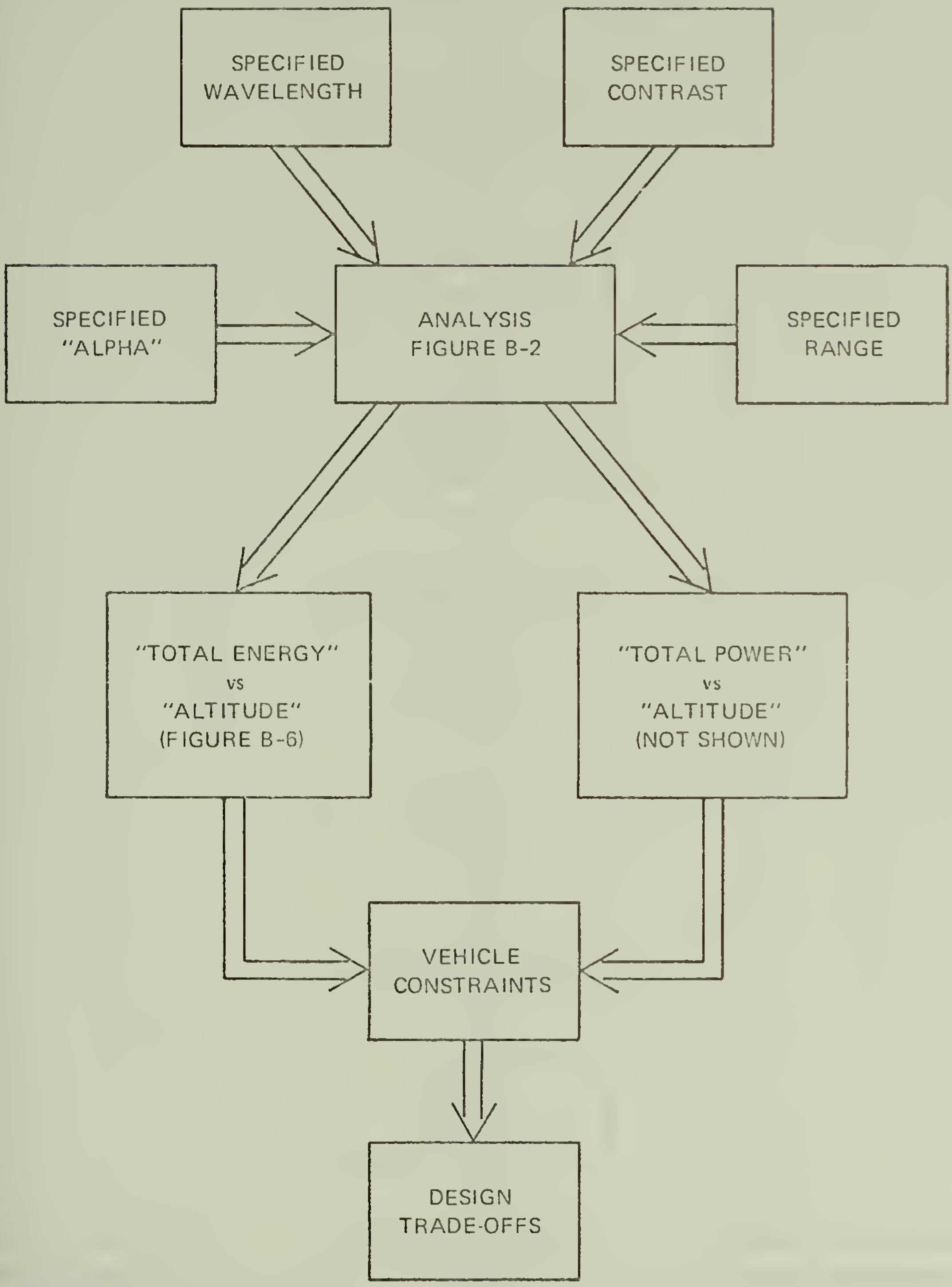
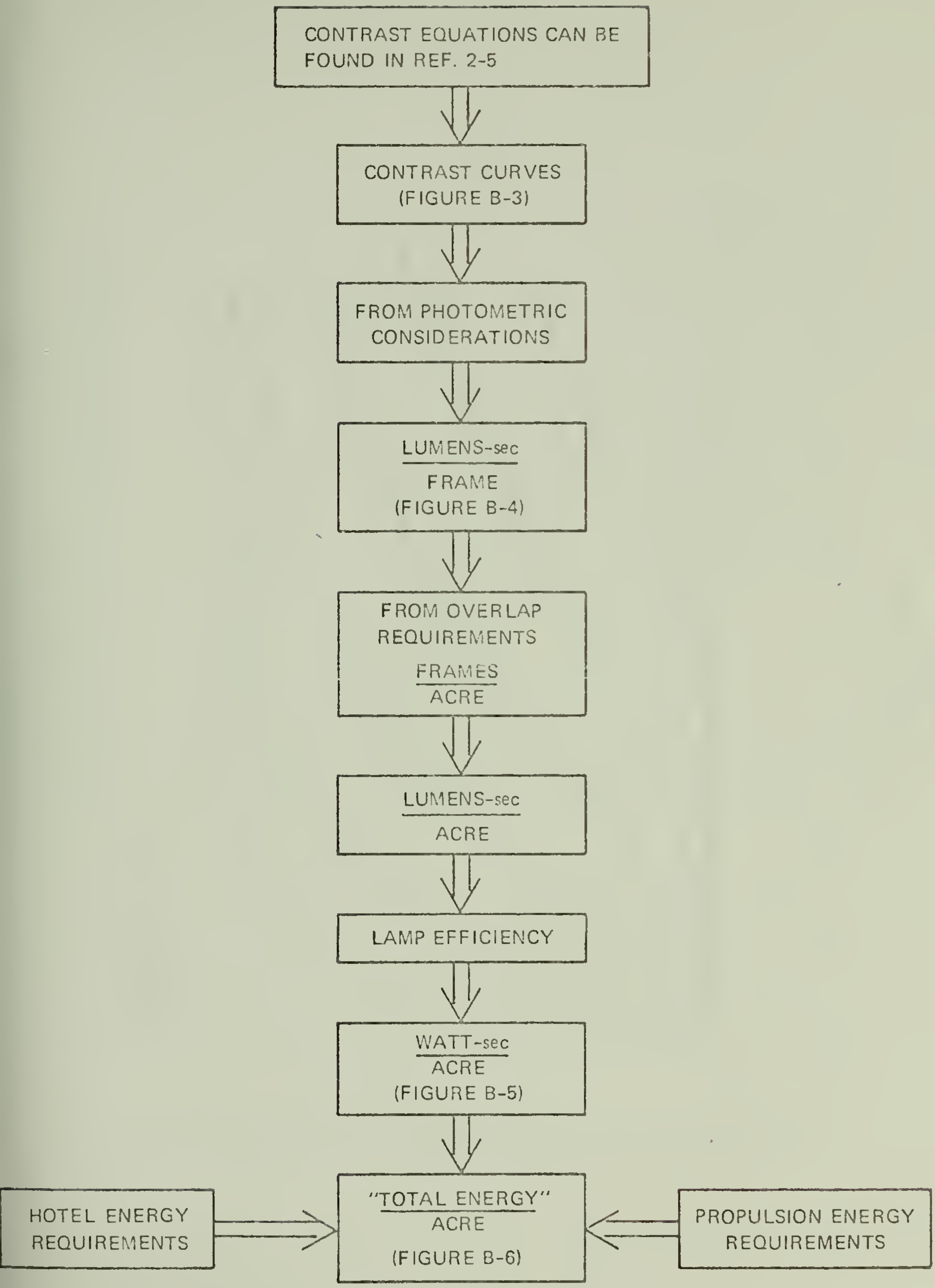
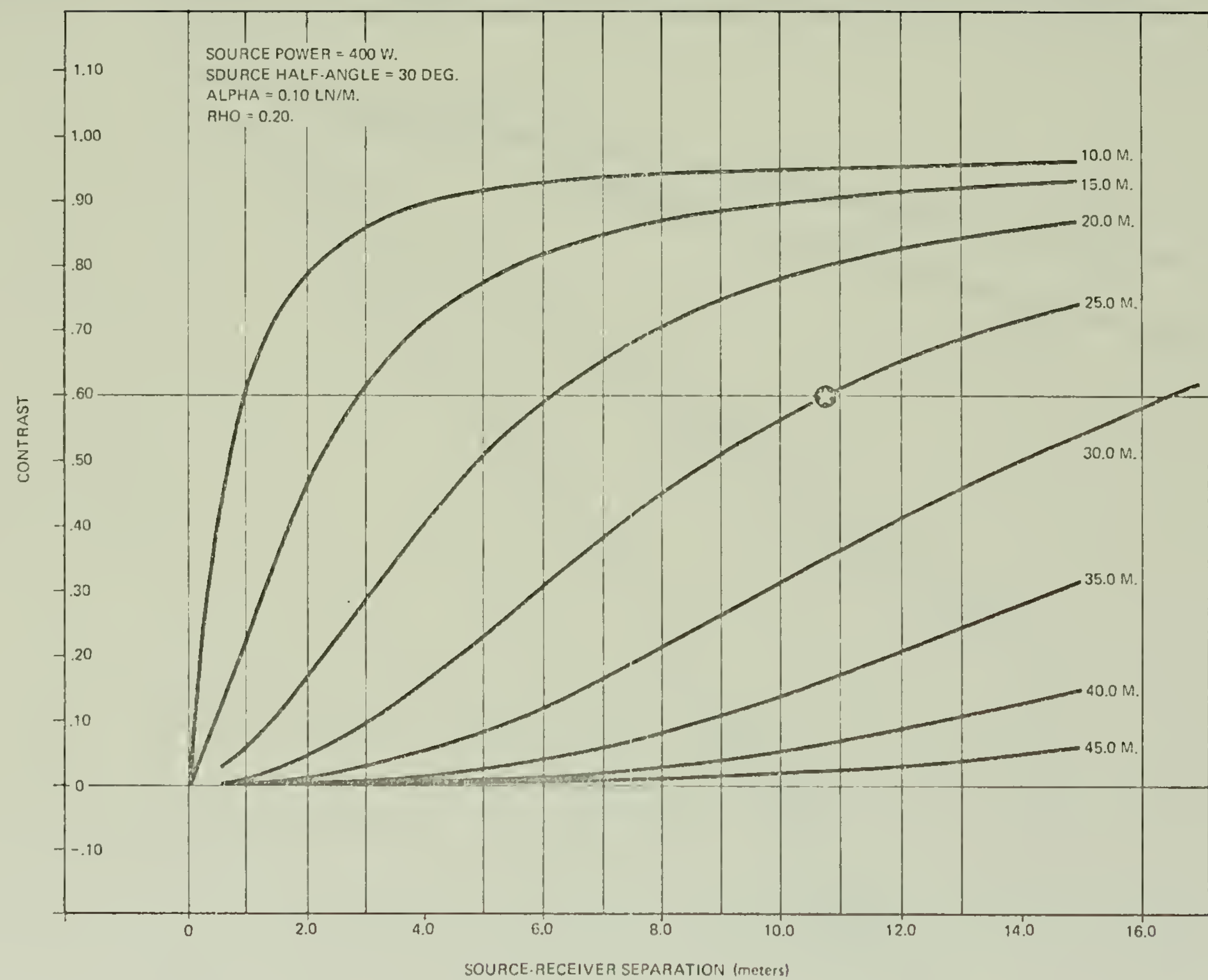


Figure B-1 "All Purpose System" Design Sequence



DESIGN PROCEDURE

Figure B-2



Contrast Curves

Figure B-3

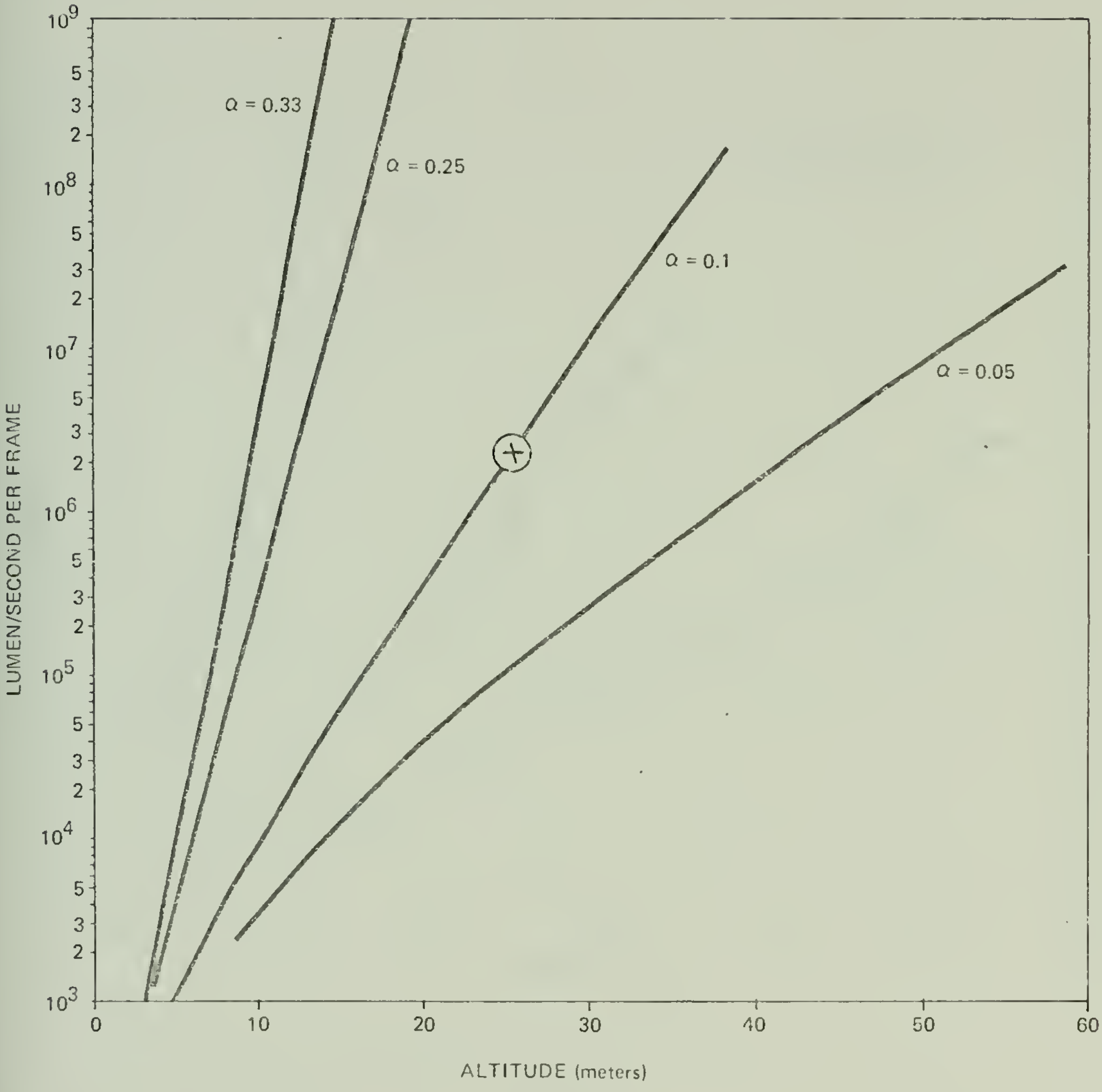
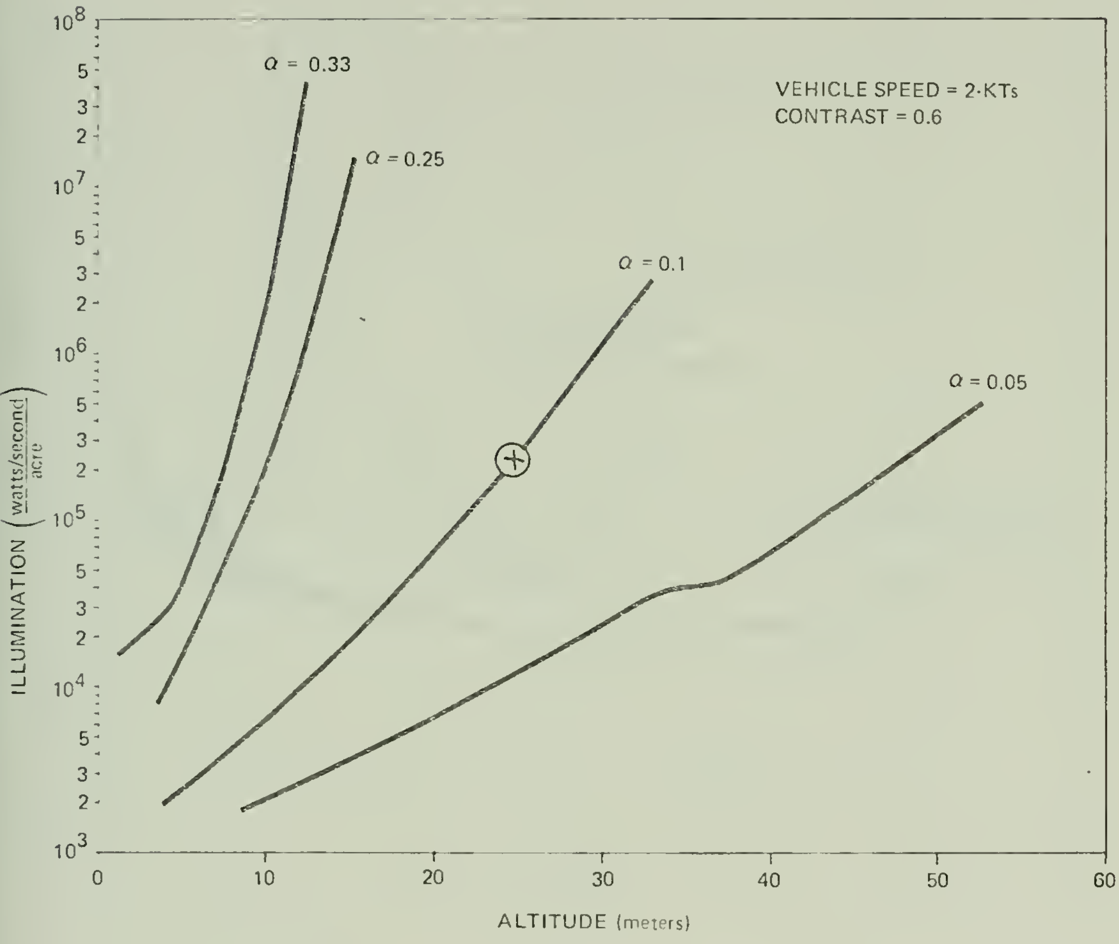
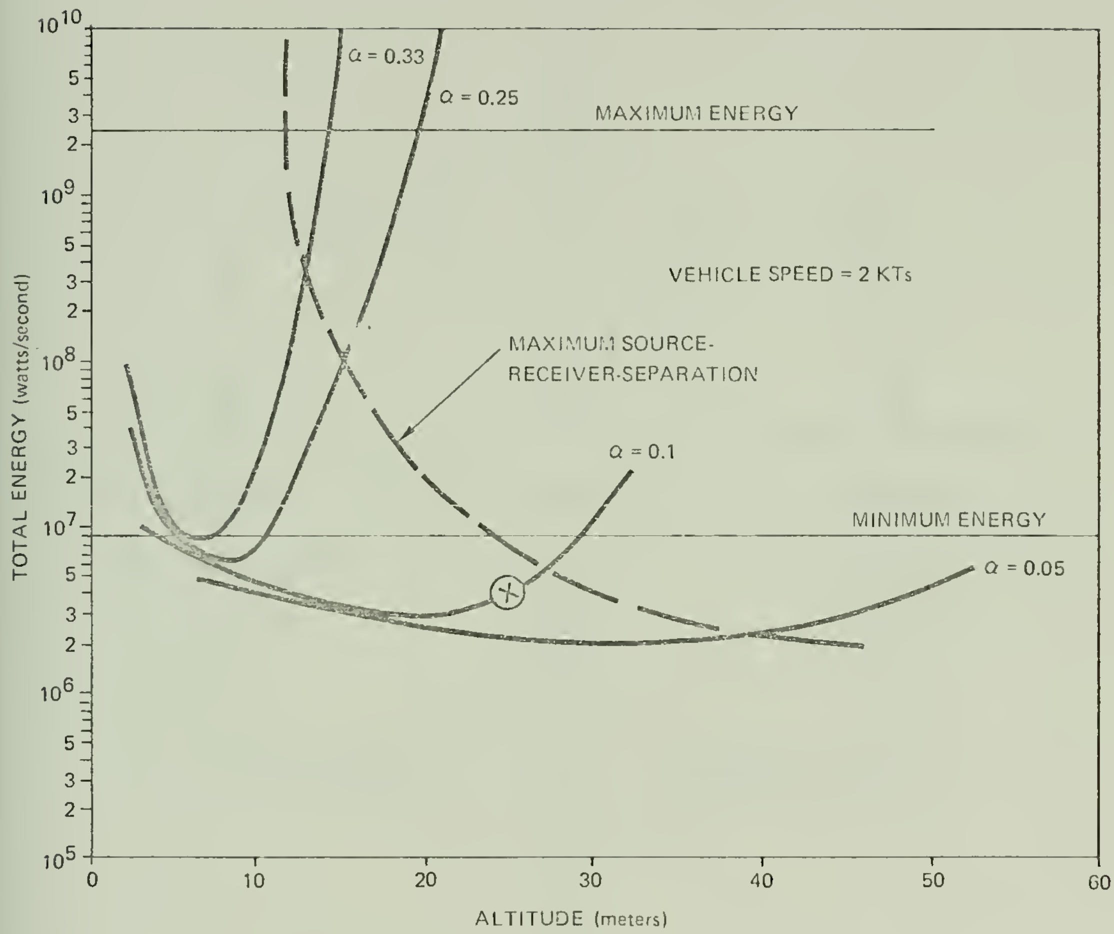


Figure B-4



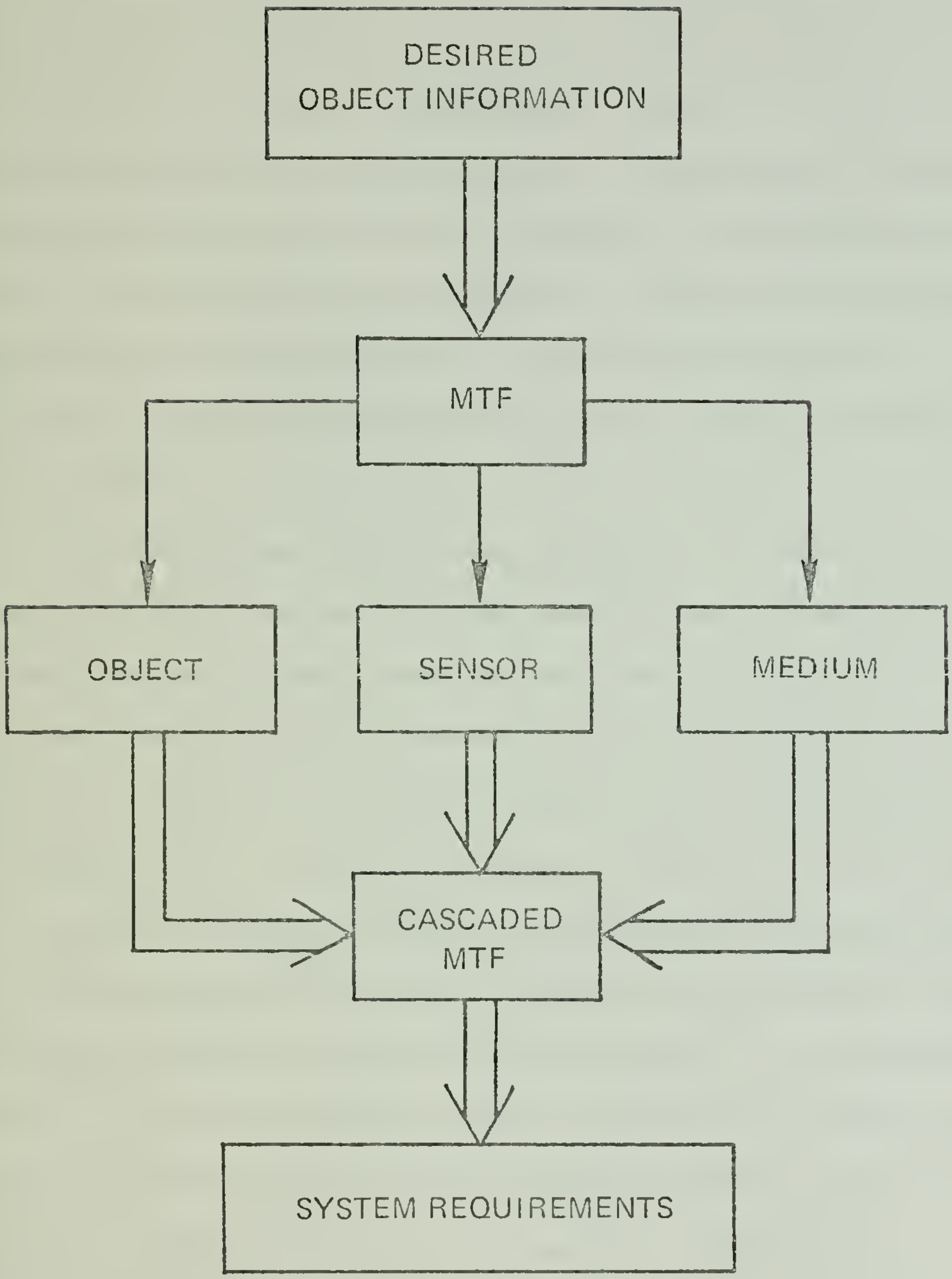
ILLUMINATION (watts/second / acre)
ALTITUDE (meters)
VEHICLE SPEED = 2-KTs
CONTRAST = 0.6
α = 0.33
α = 0.25
α = 0.1
α = 0.05

Figure B-5



Optical System Trade-Off

Figure B-6



Specified Mission, Optical Design
Figure B-7

(2) Optical System's Effect on Vehicle: A primary problem in determining the capability of the optical sensor is identification of specific mission requirements. For example, specifying that the optical suit will only be looking at golf balls helps the designer to tailor the design instead of having to provide an "all-purpose" system. Two analyses are presented below because of the difficulty in locating usable "design techniques" for underwater optical sensor systems. The methods presented in FIGURES B-1 and B-7 can be used for either a mapping or close-up photography mission. FIGURE B-1 presents a design methodology for an all-purpose optical system, whereas FIGURE B-7 presents a design methodology for an optical system requiring specific object information.

(a) All-Purpose Mission Optical Design: In this design methodology, the designer assumes an optimum contrast and a threshold contrast determined by his environment and mission. This process will probably be iterative in nature unless specific contrast data is provided or available. FIGURE B-2 outlines a general method for development of the vehicle power curves. The contrast curves in FIGURE B-3 can be developed from references 2 through 5. FIGURE B-4 was plotted from general photometric considerations and was plotted mainly for interpolation. FIGURE B-5 is useful because it shows two discontinuities. The discontinuity on the α = .05 curve corresponds to the single picture coverage of an area at a given altitude at which the 60% overlap increment becomes so small that the power usage to cover one acre increases significantly.

Finally, FIGURE B-6 represents the compilation of the last three figures. The dotted circular arc represents the limit on source receiver separation imposed due to the finite submersible length. The lower horizontal line represents the minimum energy required to fulfill a given

alpha mission. The upper horizontal line represents the maximum vehicle energy available. There are other constraints which can be imposed such as the maximum number of strobes, but these will not be considered here, since it is felt that this is adequate for this stage of the design. Any futher work should be done by optical designers.

In FIGURE B-1, the design process branches and develops energy and power curves. The power curve was not discussed but should be considered if a large number of strobes are anticipated, since the power output of the vehicle may limit the recycle time of the strobe capacitors.

(b) Specific Mission Optical Design: FIGURE B-7 describes the method employed by optical systems with a specific task. Accurate object descriptions are essential input to this methodology. Not much work has been done in the area of water or viewport Modulation Transfer Functions (MTF). References 3 through 8 provide helpful information on MTF's. The output of the methodology is a system tailored to the mission requirements, which is in a sense an optimization in the optical sensor system.

(3) Viewports:

(a) Overlapping Viewing: The need for overlapping viewing has been expressed by many pilots and observers. Busby[9] in his review of four submersibles points out that overlapping fields of view are essential for vehicles conducting any type of work or analysis. There are a couple of variables in this trade-off which are discussed below.

[1] For a given viewport separation, the curvature of the hull basically determines the extent of the blind spot between the two viewpoints.

[2] The single most influential factor in determining overlapping fields of view is the effective cone angle of the

viewport.

The first trade-off will not be discussed, as it is basically one of geometry and mission requirements.

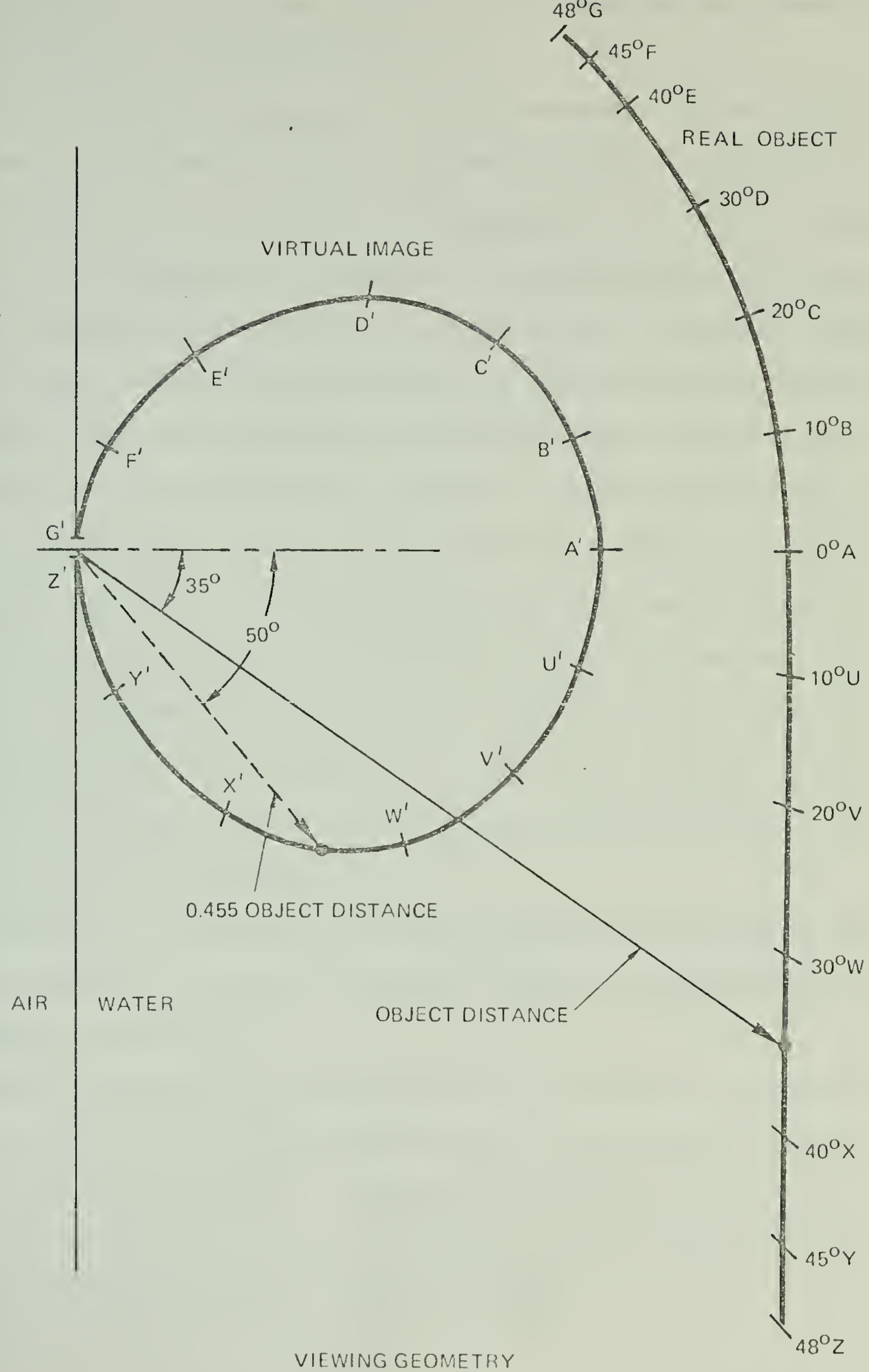
The second trade-off will be approached by trying to establish the minimum cone angle required for overlapping fields of view for the entire forward half of the manned pressure envelope. This analysis was carried out using a basic computer polar projection plot which examined the degree of intersection for various cone angles and ranges. The outcome of this trade-off was that a viewport must have a minimum cone angle of 110 . A word must be said about the use of this result. It is based upon a given spherical hull radius of 24 inches where the spacing was determined by the following criteria:

(b) <u>Viewport Spacing Criteria</u>:

(1) Viewports cause stress concentrations in the hull envelope, which must be reduced to membrane stress levels. The decay length of the stress field is dependent upon the geometry of the hull and can limit the spacing of the viewports.

(2) From a fabrication standpoint, a symmetrical distribution of viewports about the nadir of the forward hemisphere greatly reduces the fabrication residual stresses in the shell. The DDS viewports were located about about an axis 18 depressed from the horizontal to provide good forward viewing for the pilot. The four remaining viewports were placed on a small circle whose radius was determined by the mapping which looks straight down. See FIGURE IV-1 for a picture of the viewing sphere.

In order to fulfill the 110 cone angle requirement, spherical viewports must be used. A spherical viewport can not only provide cone angles



VIEWING GEOMETRY

Figure B-8

up to 180°, but also eliminates the off-axis distortion experienced by a flat viewport as illustrated in FIGURE B-9.

(c) Viewport Aperture: As in all trade-offs, the mission requirements are essential input. In the DSV design, the viewports will be used for piloting, observation, and photography, each possibly requiring different inside apertures. Experiments have shown that a one-to-one ratio, inside diameter to thickness, should be used in order to provide proper impact and strength requirements for a plane-truncated conical viewport.[10] Clearly, the larger the inside aperture, the thicker the viewport. As viewports increase in thickness, image space tends to move towards the observer, causing severe degradation in depth perception.[11] This means that for a pilot or observer to perceive depth of field, the viewport must accomodate both of the observers eyes and provide a correction system (caused by the thick lens problem) to restore realistic depth perception. On the other hand, if all that is required is a window for mapping with a fixed internal camera aperture, helping to alleviate the thick lens problem.

Reference 12 provides statistical information on interocular separation. This can be helpful in sizing the viewport to accommodate a certain percentage of observers.

The following acrylic specification are provided for convenience:

Properties vary according to the type of optical acrylic employed. Specifications below are for a frequently-used standard type. Special acrylics are available for high or low transmittance of selected wavelengths, including ultr-violet, and/or infrared.

*Optical:

<u>Thickness vs. Transmittance</u>
(400 to 1100 millimicrons):

Thickness	% Transmittance
0.25	92%
0.50	88%
1.00	86%
1.50	85%
2.00	84%
3.00	83%
4.00	82%

<u>Refractive Index</u>

1.489 at 589 millimicrons (Sodium, D)

1.496 at 486 millimicrons (Blue, F)

1.487 at 656 millimicrons (Red, C)

<u>Minimum Focal Length</u>: $F = \frac{c^2}{4t} + t$

F = focal length

c = diagonal of lens in inches

t = thickness = 7.5 inches, maximum

<u>Optical Quality</u>: Equal or superior to Grade B optical glass, which exhibits only light and scattered striae when viewed with a striaescope.

<u>Subsurface Inclusions</u>: Only one with an average diameter of less than 0.0008 inches permitted per 20cc of material. Otherwise, lens or optical component is required to be completely free of bubbles, dust, and other foreign matter larger than 0.002 inches maximum.

Surface Finish: Only one imperfection between 0.004 and 0.010 inches is permitted per square foot; and twenty between 0.0002 and 0.0001 inches per square foot.

Surface Haze: When viewed with the unaided eye in reflected light, polished surfaces must be completely free of "orange peel" or "sleak". Surface haze, or that percentage of transmitted light which deviates from the incident beam by scattering, is measured with a hazemeter or recording spectrophotometer as described in ASTM methods D1003-59T and is held to 3% or less.

Impact Strength: 6 to 17 times greater than glass.

Specific Gravity: 1.19

Tensile Strength: 10,500 psi

Compressive Strength: 18,000 psi

Modulus of Elasticity: 450,000 psi

Rockwell Hardness: M93

Maximum Optical Sizes Available: Sixty inches in longest dimension and 8 inches in thickness are standard.

Guaranteed Minimum Tolerances: Focal length of lenses shall be within $\pm$ 2% of customers' specification; when a 3 ball spherometer ($2\frac{1}{2}$ - $9\frac{1}{2}$ diameter) is moved over the spherical surface of a convex (or concave) lens, the maximum and minimum reading shall be within $\pm$ 0.5% of the average reading; plano side of lenses shall not vary more than $\pm$ 0 - 6' from parallel to a plane that is tangent to the spherical surface at the thickest (convex) or thinnest (concave) portion of the lens; all linear dimensions shall be held to $\pm$ .015 inches.

Flammability: Non-Flammable

*Thermal:

Continuous Surface Temperature: $180^{\circ} - 200^{\circ}$F.

Coefficient of Thermal Expansion at 70°F.: .000042 inches/in./ F.

Coefficient of Thermal Conductivity: 1.3 BTU/hr./sq. ft./ $^{\circ}$F/in.

Specific Heat at 70°F: 0.35 BTU/lb./$^{\circ}$F.

*Chemical:

Chemical Resistance: Oxidizing and reducing agents do not affect acrylics; however, immersion in acetone, alcohols, lacquer thinners, and carbon tetrachloride must be avoided as these agents cause optical distortion of the surface. Water gain after seven days immersion at 77°F. is negligible (1% or less).

d. Mapping: As mentioned in the requirements for a good stereo mosaic, the following two criteria should be satisfied if quality photo interpretation is expected.

The first criteria states:

$\frac{b}{h} = .6$ where b = camera baseline separation
h = flying height above target

This relationship can be achieved two ways. The first method is to have two cameras fixed to the submersible which takes pictures simultaneously. Another method uses one camera and achieves the baseline separation by measuring the distance required to fulfill the above criteria.

A second criteria requires a 60% picture overlap along the search path.

If these two criteria are equated and solved for the full cone angle of the optical sensor, then both criteria can be satisfied with one picture if the optical sensor full cone angle is between 75° and 90°.

A practical problem evolves out of this, in that wide angle lenses (i. e., greater than 60°) allow severe light losses at the outer portions of the lens. If a 60° lens is used, this means that to achieve a 60% overlap, the b/h drops. As mentioned above, the b/h criterion is a "nice-to-have" criterion, whereas the 60% overlap is essential. The 60° lens will require more pictures than a larger cone angle lens, but will save on power consumption.

4. Conclusion: There are many problems associated with optical sensors, but it is felt that if the methodology presented herein is followed, many of these problems can be better defined and therefore designed for at early stages in the design process. It is recommended that future work be concentrated in the following areas:

a. Much work needs to be done in reflector design. Present reflectors produce hot spots on the optical sensor, thus destroying information. Reflectors are generally very inefficient.

b. MTF's need to be developed for various water conditions and various viewport configurations. A deep ocean MTF meter incorporated into the vehicle would be invaluable to the extension of conventional imaging techniques.

c. A method for recording environmental information on the film would greatly reduce the work-load of the photo interpreter.

d. Better acrylics for viewports, need to be developed.

The information provided in this discussion should be analyzed from the beginning each time a new design is initiated, since technology is advancing at such a rapid pace. Today, it is felt that optimization of conventional viewing methods will produce the "best" system, but great strides are being made with extended range techniques.

Resolution-limited systems were not discussed in this section since it was felt that references 3,4, and 7 cover the conventional techniques sufficiently.

It is recommended that more time be spent in perfecting the conventional imaging techniques, before rushing on to extended range imaging techniques.[13]

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2. "Handbook of Underwater Imaging Systems Design," NUC TP 303. "An outstanding design manual for estimating photographic power requirements."
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   "A good discussion of MTF, but the results are questionable."
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11. McNeil, G. T., "Optical Fundamentals of Underwater Photography," Edwards Brothers Inc., Ann Arbor, Michigan, 48104, 1968.

12. Morgan, C., Cook, J., Chapanis, A., and Lund, M., "Human Engineering Guide to Equipment Design," McGraw-Hill Book Co., New York, 1963.

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    2. "Discusses various photographic techniques and underwater viewing systems."

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1. McNeil G., "Optical Fundamentals of Under Water Photography," Edwards Brothers Inc., Ann Arbor, Michigan, 48104, 1968.

2. Mertens, L., "In-Water Photography, Theory and Practice," Wiley Series on Photographic Science, 1970.

3. Jensen, N., "Optical and Photographic Reconnaissance Systems," Wiley Series on Photographic Science, 1968.

4. "Handbook of Underwater Imaging Systems Design," NUC TP303.

The first three references are helpful in setting a basic ground work in vocabulary and underwater optical problems. Reference four is helpful in determining the energy and power requirements for a given mission. Reference four also has outstanding bibliographies on the following topics:

1. Optical Properties of sea water.
2. Conventional underwater television systems.
3. Extended range underwater imaging systems.
4. Introduction to underwater imaging system design and work sheets.

5. Duntley, S. Q., "Light in the Sea," Journal of the Optical Society of America, Vol., 53, pp. 214-233, Feb. 1963.

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   "An experiment-oriented paper, evaluating various parameters in spherical acrylic plastic windows."

2. Snoey, M. R., and Stachiw, J. D., "Windows and Transparent Hulls for Man in Hydrospace," in a Critical Look at Marine Technology, Transactions of the 4th Annual Marine Technology Society Conference, Wash., D.C., July 1968, pp. 419-463.

   Purpose of paper:

   1. A historical review of viewports and properties of viewports materials (outstanding).
   2. NCEL's work in hydrospace window design.
   3. A summation of viewports in actual use (outstanding).
   4. A brief literature review of window research being done by other organizations.

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Appendix C

Work

1. Object Analysis:

An object analysis has been done by Lockheed and is presented below as an aid to future object recovery analysis. An extensive overview of possible recovery objects is presented on TABLE C-I.* An object definition summary was prepared from this table. The known inputs that must be given consideration in this summary are as follows:

* Density of salt water at 20,000 ft. is 65.9 lb/cu ft, or 0.038 lb/cu in.

* Weight of aluminum is 169 lb/cu ft or 0.100 lb/cu in.

a. Small Object Definition: A basic requirement is that the small object have a water weight of 250 lbs. at 20,000 feet.

The basic assumptions are listed below:

* The sample object is composed of aluminum skin and internal structure with enclosed instruments or payload. The displacement is about 50 percent of total volume. Fifty percent of the total volume will be void and floodable.

* Other external materials and attachment methods were considered for stainless steel, beryllium, and other materials.

* The sample object will have a smooth outer surface.

* The outer skin thickness will range from 0.090 to 0.120-in.

* The object has either ruptured or an open end to permit flooding.

In establishing weight and volume limits for the sample object, it was decided that the resultant weight per cubic inch at 20,000-ft depth would equal the material weight of the object per cubic inch in air minus 66 lb/cu ft. or 0.038 cu in., the weight of water displaced at the above depth.

Aluminum was selected for the study since the diameter of a spherical steel shape based on the same object assumptions, would be only 7 in.

*Table C-1 taken from reference 1

Table C-1
SUMMARY OF POTENTIAL OBJECTS FOR RECOVERY

Object or Task	Recovery Motivation	Shape and Dimensions	Limiting Weight (LW), Limiting Dimensions (LD)
Airframes	Economic, Intangible, Security	Generally tubular or cellular	Wing Density – based on wt, 15% of total P-3 density – 7 lb/ft^{-2} limiting area 36 ft^2 C-141 density – 6 lb/ft^{-2} limiting area 42 ft^2
Bombs, Rockets, Projectiles, and Torpedos	Security	Spherical to cylindrical $1 < D < 3$ ft $1 < L < 10$ ft	Solid – may have lifting or towing eye LW: 5 in. proj. ~ 30 lb; torpedo, neutrally buoyant; bombs and rockets: 100 lb LD: 3 ft up to 12 ft for torpedos
Hull Debris	Economic, Security	Linear to tubular	Plates and Shapes – LD: 1 in. steel plate, 2.0 × 20 ft LD: 6 in. × 2.3 in. I-beam 16 ft length
(Radioactive) Disposal Canisters	Intangible	Cylindrical barrels Dia. ~ 1 ft, length 2 ft	Solid, must not be punctured, LD: barrel 13 ft^3, 2 × 4 ft
Space Hardware	Intangible	~ Equidimensional, cones, cylinders Dia. = 1 to 15 ft	Weight: 50 to 20,000 lb. Dim: Various from small (~5 ft) cylinders or spheres to large (~15 ft) cones.
Electronic Devices	Security, Intangible	Tetragonal W×L×T: (2×2×6 ft) max.	Relay racks, etc., open construction. Weights: 5 to 250 lb.
Oceanographic Instrument Arrays	Intangible	Cylindrical R = 1 to 3 ft L = 1 to 10 ft	Frameworks, Sealed cylinders ~ 100 lb.
Grabs and Corers	Intangible, Economic	Cubic or cylindrical 1×1×1 ft or 1×1×10 ft	Lifting shackles attached; embedded; LW: 100 lb w/o sediment weight.
Geologic Specimens	Intangible	Tubular or equidimensional 1×1×1 ft	High density, embedded. LW: ~ 100 lb/ft^3. LD: cube 1.3 ft
Baggage and Crates (loose books)	Economic, Security	Tubular or equidimensional. Dia. < 5 ft	Weighted sacks, LD: 2 ft^3 with 50 lb wt (weighted bag). Paper: Neutrally buoyant.
Wall Head Devices	Intangible, Economic	Linear to equipment. Dia. < 3 ft	LW: 6 in. HP gate valve: 650 lb LD: 2×2×3 ft 6 in. HP flanged spool: 200 lb LD: 1×1×1 ft
Cables and Piping	Economic	Linear to cylindrical	Pipe { 8 in. dia. double extra strong, 63 lb/ft LD: 4 ft 3 in. dia. double extra strong, 16 lb/ft LD: 15 ft Cables: ≤ 15 lb/ft
Engines	Economic, Security	Equidimensional or cylindrical 2×3×3 ft or 2×10 ft and up	Aux. gas turb: 2×2×2 ft; 200 lb and up Prime movers: 500 to 20,000 lb
Artifacts	Economic, Intangible	Generally, elongate Dia. ~ 2 ft, Length ~ 6 ft	High or low density, extremely fragile. LD: Marble 1.5 ft^3
Module Removal, (as Electronic Components from Aircraft)	Security	Tetragonal to equidimensional 2×2 10 ft	Involves cutting hole in bulkhead to remove module. LW: < 5 lb to 250 lb.

Since the requirements state that the object will be larger than a 9-in. diameter shape, steel obviously had to be eliminated. Of interest is the fact, that 4,032.2 cu. in. of aluminum in any shape is required to make an object of 250-lb. weight at 20,000 ft. (The aluminum sphere shape studied has a 25-in. diameter.)

The study objects selected were limited to the diameter range of 9.0 in. to 25.0 in. Dimensions for the three chosen shapes are as follows:

* Sphere	25-in. diameter
* Cylinder	9-in. diameter, 126-in. length
* Cone	24-in. diameter base, 53 in. height

Each of these objects has a total volume of 8,064 cu. in., a solid displacement of 4,032 cu. in., and an aluminum structure density of 0.100 lb/cu. in.

b. Large Object Definition:

*For non covert operations, it was decided to adopt the following upper limits for the recovery object:

maximum envelope - 20 ft. long cylinder with a 5-ft. diameter; maximum water weight of 5,000 lb. at 20,000 ft.

*For covert operations, the following limits were set:

maximum envelope - 15 ft. long cylinder with a 5-ft. diameter; maximum water weight of 5,000 lb. at 20,000 ft.

The following assumptions were made for the objects to be used in both operations:

* Thickness of outer skin: 0.090 to 0.120 in.

* Materials are: steel, aluminum, magnesium, titanium, beryllium, or a ceramic or fiber glass

* Texture of outer surface: assumed to be smooth

* Condition of objects: ruptured or with holes or an open end to permit flooding

* Burial depth prior to recovery: 40 in.

2. Environmental Analysis

A deep ocean environmental analysis was conducted by Lockheed. TABLE C-2 represents the average bottom characteristics. For a more detailed coverage see Reference 1. References 3-26 are just a few of the references available, but it is felt that these references at least will provide a good starting point for most environmental studies.

3. Small Object Recovery

a. Small Object Recovery Devices: The small object recovery devices listed below were considered to be the most feasible alternatives of those considered.

(1) Suction cup

(2) Stud gun

(3) Bonding

(4) Claw

(5) Clam shell

(6) Snare

(7) Power driven net and winch system

(8) Magnetic lift

The "Oceanographic Instrumentation" section of Reference 2, discusses small object recovery devices and provides a brief description of each device. The number of recovery devices is limited only by the designers imagination and the various configurations of objects to be recovered. It is recommended that the small object recovery devices be designed for a specific task, although multiple task devices should be considered in the trade-off.

TABLE C-2

DSSV ENVIRONMENTAL PROBABILITY BASED ON ANALYSIS OF BOTTOM SURFACE[(a)]

Condition	Probability (%)	Description
I	60	• Relatively flat or gentle slopes, primarily clay sea floor (ocean basins 20,000 ft or less) object partially buried, current less than 1 knot. * • Relatively flat or gentle slopes, primarily mud sea floor (globigerina ooze), object partially buried, current less than 1 knot. • Gentle slopes, boulders and cobbles, adjacent to steep slopes, object may be between boulders and rocks, current less than 1 knot.
II	30	• Steep slopes to 45 deg; primarily mud (globigerina ooze) sea floor, object partially buried, current less than 1 knot. *
III	10	• Submarine canyons (vee-shaped valley) rough walls of bedrock, canyon floor varies from boulders, rocky gorges, smooth sand and thick deposits of surf grass and kelp. Object may be partially buried in kelp or between rocks, current less than 0.5 knot. *

NOTE: The asterisk-marked items are the three most probable environments selected for recovery analysis and approximate percentage of probability. These three environments were selected because they represented the following conditions:

1. Most representative of existing environmental conditions
2. Offerred opportunity to study retrieval on a slope
3. Offered worst retrieval conditions

Note: Taken from reference 1

b. Small Object Storage: The design of small object storage systems is primarily controlled by the following:

(1) The configuration of the objects to be recovered.

(2) The quality of direct or aided viewing.

The closure system for the storage system.

(a) Deep Star - 4000 - retractable sample bag

(b) Aluim-non rigid sample bag

(3) The ability of the storage system to retain an object while the DSV is surfaced.

A brief description of small object recovery devices is covered in Reference 27.

c. One Versus Two Manipulators: This trade off will confront most designers engaged in submersible design for work missions. Reference 27 reviews the manipulator systems on board four operational submersibles and recommends that two manipulators be provided if the work task requires anything more than picking up a free object on the bottom.

4. Large Object Recovery:

a. Large Object Recovery Devices: The design of large object recovery devices is governed by the designers imagination and the particular mission requirements. Lockheed[1] has investigated a few alternatives which are listed below:

(1) Suction cup

(2) Clam shells

(3) Nets

(4) Explosive studs, anchors or harpoons

(5) Hay rake or tine system

Lockheed conducted trade offs on the above devices and decided that the hay rake or tine system would be best since it provided positive attachment and water jet capability to reduce the breakout force.

b. Object Transport: Transporting the recovered object from the bottom to the mother craft is not a simple problem. The transport can be made via:

(1) A surface lift system, but this has problems in rough seas and object stability during ascent.

(2) A submersible direct lift, but this can be dangerous for for the submersible and also increases the size of the DSV significantly.

(3) A work module lift system which could be tailored for each mission.

The submersible direct lift is not recommended because of its effects on the DSV characteristics.

c. Object Handling by Support Craft: Another restriction on the large object recovery system is the mother craft lift system interface. Object handling by the ASR of SSN should be examined for each of the transport methods mentioned above.

d. Operational Problems:

(1) Breakout Forces: The force required to extract an object embedded in the ocean bottom is dependent upon the type of soil, the duration of the applied force and the object shape. When the object is released by the bottom it generally "pops" out. This could cause some vehicle stability problems. The breakout problem has been investigated and References 1 and 22-26 can provide the designer with sufficient information for a preliminary investigation of the problem. The output of the breakout force trade off should be input into the control subsystem. These references also suggest methods of reducing the breakout forces, such as

water jetting.

(2) Sediment Distrubance: As can be seen in TABLE C-2, the major part of the ocean bottom is mud. The local currents vary but in general they are small in magnitude causing a long particulate clearing time. It is imperative that sediment disturbance be kept to a minimum. This will require a wake analysis for low altitude mapping missions and vehicle control simulation for work mission.

There are many operational problems but these are felt to be the two most pressing to date. This probably reflects the state of the art in large object recovery. (i.e. still on the bottom!!)

5. Work:

Work is defined in this as any task other than recovery. Listed below are probably work tasks and various tools which could complete the tasks.

1. Attachment:
   a. Welding
   b. Drill/tap
   c. Stud gun

2. Joining:
   a. Bolting
   b. Adhesive
   c. Welding

3. Cutting:
   a. Mechanical such as a saw or drill
   b. Gas/electric gas systems are not feasible for deep ocean work

4. Drilling:
   a. drill; for metal or bottom rock
   b. water jet for mud drilling

5. Cable Cutting:
   a. Explosive cutting tool
   b. Mechanical cutting tool

6. Assembly or Disassembly:
   a. Impact wrench
   b. Screw driver
   c. Impact hammer
   d. Plus all tasks mentioned above

6. Tools:

There is little information in open literature on underwater tools. This is probably a result of the problems encountered with manipulator task completion times and the poor results encountered with manipulators onboard operational submersibles.[27]

A comprehensive search was made of the tool industry to determine the status of deep submergence tools. An explosive cable cutter made by Atlas Acrospace (Valley Forge, Pa.; sea data sheet 808, Sept. 1972) was the only tool commercially available and it would require extensive adaptation for use with a manipulator. Basically, if work is to be accomplished the designer must adapt present in aim tools for underwater use. A few submersible manufacturers have fabricated tools for their own use. Some of these companies are listed below.

a. Vickers Oceanics Ltd.
   P. O. Box 8
   Barrow-in-Furness
   Lancashire LA 14 1AD
   ENGLAND

b. Perry Submarine Builders
   Rivera Beach
   Florida 33404

c. Kawasali Heavy Industries Ltd.
   24-1 Hamamatsu-Cho
   Minato-Ku
   Tokyo, JAPAN

There are many more submersible owners and builders who use or have built manipulator tools. See TABLE III-1 under "Remarks" for those submersibles with manipulators. To date, hydraulic adapted tools have proved to be capable of significant underwater work.

7. Manipulators:

There are three major groups of manipulators, hand powered master-slave, powered master-slave and powered rate controlled. The first part

of the name designates the power mode whereas the second part designates the method of control. Hand powered master-slave manipulators are not used on board submersibles due to the problems with hull penetrating seals and sufficient internal volume for full operator arm swing. Powered master-slave (i.e. servo) manipulators are not used on board submersibles due to the problems involved with pressure compensation of the servo mechanism and the requirement for full arm swing of the operation. There are two basic types of powered rate-controlled manipulators, rectilinear and spot mounted. Spot mounted powered rate-controlled manipulators are used on board operational submersibles for very simple tasks due to the high task completion time. The task completion times are on the order of one hundred times greater than tasks done by hand. The high task completion times are due to the uncoupled nature of each degree of freedom. References 39 discusses the theory behind decoupling the manipulation motion for use in automated assembly. This has possible application for deep submergence work. (See note under Reference 39).

The selection of hydraulic or electric power for the manipulator and tools should be part of the design trade off.

The manipulator power level will be dependent upon the type of work, the rate of work and the required degrees of freedom.

Listed below are the names of companies which have made underwater manipulators.

a. Remotion Company
330-E South Kellogg Avenue
Goleta, CA 93017
(805) 964-5418

Remotion Company is now developing a Servoarm Force Reflecting Manipulator. The arm is operated by water hydraulics. Some

of the manipulator characteristics are listed below:

Size - approximately human arm

Capacity - 20 pounds outstretched

Operating Media - fresh water this model, seawater available.

Operating depth - pressure compensated - no real limit.

Force reflection - 1:1 or ratios on all motions

Tests have shown that a force reflecting servo manipulator works about twenty times faster than a rate controlled manipulator (switch box controlled). In addition, good visibility is not as necessary because some operations can be done by feel.

b. Programmed and Remote Systems Corporation

899 West Highway 96

St. Paul, Minnesota 55112

(612) 484-7261

PAR makes two manipulators, PAR 3000 and PAR 3500. The PAR 300 is hydraulically operated and has the following specifications: Two PAR 3000 have been used on board Deep Quest since 1968. These manipulators were not outfitted with tools, but did have explosive cable cutters.

Motion	Travel	Speed	Capacity
Shoulder Rotate	370 degrees	1 RPM	7000 lb. in.
Shoulder Pivot	210 degrees	1 RPM	*
Elbow Pivot	270 degrees	1 RPM	*
Wrist Pivot	270 degrees	1 RPM	**
Wrist Side Pivot	180 degrees	3 RPM	*
Wrist Rotate	Continuous both directions	3 RPM	900 lb. in.
Grip	0 to 8 in.	1 in./sec.	500 lbs.

These motions are rated for a 100 pound load in the hand with the manipulator elements in any position.

The load capacities given are based on a hydraulic supply pressure of 1750 psi. All motions are protected from overloads by either slip clutches or overload valves. The manipulator will operate at hydraulic pressures up to 3000 psi.

Air Weight (dry) - 425 lbs.

Air Weight (oil filled) - 490 lbs.

Submerged Weight (oil filled) - 295 lbs.

Controller Weight - 10 lbs.

The PAR 3500 is electrically operated and has the following specifications. The DOWB used two PAR 3500 for two years until they were accidently dropped during an ascent. These manipulators were not outfitted with tools.

Motion	Travel	Speed	Capacity
Shoulder Rotate	370 degrees	2.5 RPM	*50 lbs. Manipulator
Elbow Pivot	320 degrees	2.1 RPM	*
Wrist Pivot	328 degrees	2.2 RPM	*
Wrist Rotate	Continuous both directions	5.0 RPM	80 16/inches
Grip	0-4"	35 in/min	0-100 lbs.

*. These motions are rated for a 50 pound load in the hand with the manipulator elements in any position.

The load capacities given are based on a constant supply voltage of 120 volts D. C. All motions are protected from overloads by mechanical slip clutches and individual electrical circuit breakers for each manipulator motion.

Unit Weight

In Air (dry weight) - 153 lbs.

In Air (fluid filled) - 185 lbs.

In Water (fluid filled) - 105 lbs.

Supply voltage - 120 volts D.C.

c. Westinghouse Electric Corporation

Box 1488

Annapolis, Md. 21404

Westinghouse has built manipulators for the following submersibles:

Vehicle	Number of Manipulators
Trieste II	I
N R-I	I
Deep Star 2000	I
Deep Star 4000	I
Deep Star 20,000	I
DSRV (bought but not used)	2

Tools were made for the manipulators but were not used.

d. North American Rockwell has outfitted at least the DSRV and the Beaver MK IV. The DSRV is presently using North Americans manipulators.

e. General Milk constructed Alvin and made manipulator arms in 1964. Alvin is equiped with water samplers, coring and other scientific tools. Alvin is located in Woods Hole, Mass.

Reference 37, provides an excellent review of operational submersible manipulators. For more detailed information the vehicle operators or manipulator builder should be contacted.

Reference 28 and 32 - 39 were provided as background information on

underwater manipulators.

8. Conclusion:

It is felt that unless manipulators of the type mentioned in reference 39 are developed, work missions will be long and frustrating. It is recommended that, "work" be kept to small object recovery for the present. If a work vehicle is desired, it should be designed for a specific task.

Future work efforts should be centered on developing force reflecting decoupled manipulators which can accommodate tools.

REFERENCES

1. Lockheed Missiles and Space Company, "DSSV Final Report," LMSC-T-14-68-1, Part VI, System Studies June 1968.

2. Myers, J. J., Holm, C. H. and McAllister, R. F., "Handbook of Ocean and Underwater Engineering." McGraw-Hill, 1969.

A good summary of Basic Oceanography with good bibliography.

3. Major, J. W. and Peyser, R. E., "WHOI Ocean Engineering Course 13.995, Lecture Notes." 1973.

The section titled "Environmental Considerations for Ocean Engineering" is an excellent start and sufficient for preliminary design of a DSV. References 4-23 were taken from Major's work.

4. "Guide For the Classification of Manned Submersibles," American Bureau of Shipping, New York 1968.

5. Williams, A. J. III, "Progress Report on Sea Grant GH104 for Oceanographic Engineering Academic Development," Woods Hole Oceanographic Institution Ref. 71-4 Dec. 1971.

6. Busby, R. F., L. M. Hunt and W. O. Rainnie, Jr. "Hazards of the Deep" Parts I, II, III. Ocean Industry, July 1968, Vol. 3, No. 7., August 1968, Vol. 3, No. 8. Sept. 1968, Vol. 3, No. 9.

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10. Neu, H. J. A., "Extreme Wave Height Distribution Along the Canadian Atlantic Coast." Ocean Industry July 1972. "Wind Drift Currents," Geo-Marine Technology, Vol 2, No. 8, Sept. 1966.

11. "Material Performance and the Deep Sea," ASTMSTP 445, ASTM Feb. 1969.

12. Monney, N. T. "The Engineering Properties of Marine Sediments" MTS Journal, Vo. 5, No. 2, March-April 1971.

13. Kuenene, P H., "Marine Geology" Wiley, N.Y. 1950

14. Rinehart, F. M. "Corrosion of Materials in Hydrospace", Parts I, II, III, IV, 1967 Naval Civil Engineering Laboratory, Port Hueneme, Ca.

15. Wiids, K. and A. Assur, "The Mechanical Properties of Sea Ice," U. S. Army Cold Regions Research and Engineering Laboratory, Report IIC3, Sept. 1967.

16. Peyton, H. R., "Sea Ice Strength," University of Alaska, Arctic Environmental Engineering Laboratory, Geophysical Institute Report UAG R-182, Dec. 1966.

17. "Serial Atlas of the Marine Environment," American Geophysical Society.

18. "Project Seabed," Prepared by the Oceanography Sub-panel of the Undersea Technology Panel. W. H. O. I. Ref. 64-4 Sept. 1964.

19. Wiegel, R. L. Oceanographical Engineering, Chapter 9. Prentice-Hall. 1964.

20. Bradner, H. and J. D. Isaccs. "Final Technical Report of Overpressure Due to Earthquake Project, "Scripps Institute of Oceanography Ref. 72-18, 1969.

21. "Clinatological and Oceanographic Atlas for Mariners," U. S. Navy Oceanographic Office.

22, Vesic, A. S. "Breakout Resistance of Objects Embedded in Ocean Bottom," U. S. N. C. E. L. Report No. CR, 69.031 May 1969.

23. Inderbitzen, A. L., F. Simpson, and G. Goss "A Comparison of In Situ and Laboratory Vane Shear Measurements," MTS Journal Vol. 5, No. 4 July-August 1971.

24. Lee, H.J., "Unaided Breakout of Partially Embedded Objects from Cohesive Seafloor Soils." U, S. N. C. E. L. TR R-755, Feb., 1972.

25. DeHart, R. C. and Ursell, C. R., "Force Required to Extract Objects from Deep Ocean Bottom." Prepared for the Structure Mechanics Branch Office of Naval Reasearch by Southwest Research Institute, San Antonio, Texas.

26. Taylor, R. J. and Lee, H. J., "Direct Enbedment Anchor Holding Capacity," U. S. N. C. E. L. TN N-1245, Dec. 1972.

27. Busby, R. F., "Design and Operations Performance of Manned Submersibles," ASME, 68 WA/UMT-11, Jun. 1968.

28. Myers, J. J. Holm, C. H. and McAllister, R. F., "Handbook of Ocean and Underwater Engineering," McGraw-Hill, 1969.

It defines various work functions and suggests possible in air tools which could be adapted for underwater use. It provides a good overview of manipulators and a design methodology for work adaptation. The bibliography at the end of each chapter is helpful.

29. Liffick, B. L. and Guthrie, G. S., "Evaluation fo Three Diver-Installed Fastener Systems," U. S. N. C. E. L. TN N1262; Apr. 1973.

Compares three fastening systems capable of 20,000 lbs. pullout force:

1. Drilling and tapping
2. Power Velocity Studs
3. Combination Drill/Taps

30. Black, S. A. and Barrett, F. B., "Technical Evaluation of Diver-Held Power Tools," U. S. N. C. E. L. TR R729, June 71.

Report summarizes tradeoff studies for impact wrenches and evaluates drilling and tapping work times. The report can be used for load estimation.

31. Vagi, J. J., Mishlen, H. W. and Randal, M. D., "Underwater Cutting and Welding State-of-the-Art," Battelle Memorial Institute Columbus Laboratories, Columbus Laboratories, Columbus, Ohio.

Discusses various welding and cutting techniques and gives enough technical data for a load estimation. A condensed version of the report can be found in Masubuchi, K. "Materials for Ocean Engineering," M. I. T. Press.

32. D'Arcangelo, A. M., "Ship Design and Construction," S. N. A. M. E., New York, 1969.

Lists operational submersibles with manipulators (pp. 562-565) Also see Table III-1 of this thesis.

33. Buzby, R. F., "Design and Operational Performance of Manned Submersibles," A. S. M. E. 68WA/UNT-11, June 1968.

Compares four submersible manipulations and work systems, Alvin, Aluminaut, Deep Star 4000 and Star III.

34. Karimen, R. S., "Land-Based Remote Handling Background of Underwater Handling Equipment," A. S. M. E. 65-UMT-7, March 1965.

35. Froehlich, H. E., "Underseas Manipulators," Programmed and Remote Systems Co., St. Paul, Minnesota., Apr. 1969.

A brief note on manipulations, but excellent on Mr. Froehlich's background in manipulators.

36. Froehlich, H. E. "Underseas Manipulators Technical Report, " Remote Programmed Systems Corp., St. Paul, Minnesota, Apr. 1969.

A hardware description of rate-controlled manipulators. Enough specifications are presented for preliminary load estimation.

37. Wischoefer, W. and Jones, R., "Submersible Manipulator Developments," Undersea Technology, March 1968.

A brief review of manipulators on present submersibles and a discussion of their capabilities and limitations.

38. Rockwell, P. K., "Underwater Manipulative Construction System," U. S. N. C. E. L. TN N-115B, May 1971.

39. Nevins, J. L., Whitney, D. E. and Simunouic, S. M., "Report on Advanced Automation System Architecture for Assembly Machines," Charles Stark Draper Laboratories, Inc., Cambridge, Mass. 02139, Report R-764, Nov., 1973.

This work was basically concerned with designing a system which could perform complicated work functions. The functions could be programmed for on-the-spot work. For example, if a spot mounted powered rate controlled manipulator was given a job of reaching for an object; this system could decouple the input motion and provide signals for required manipulator motion so that a 1:1 or selected motion ratio could be achieved. This work was not done for underwater work, but could have possible applications.

Appendix D

Control

This appendix was written to aid the designer in the trade-offs necessary for the development of a control system for a DSV. The discussion has been divided into two main sections, static stability and dynamic stability Each section will present background information, references and a set of trade-offs.

1. Static Stability

The static stability of submersibles can be divided into three general areas, namely surfaced, during submerging and emerging, and submerged. During the surfaced condition, all stability principles in reference 1 apply. The transient condition of submerging and surfacing is also covered in reference 1. The fully submerged condition basically requires two conditions for static stability, neutral buoyancy and longitudinal balance. The achievement of these two conditions in a DSV is highly dependent upon the designers ability to predict various loading conditions. A graphical design tool which can help the designer size the submerged trim system is called the equilibrium polygon. This is also presented in reference 1.

In DSV design, the achievement of neutral buoyancy and longitudinal balance has produced many different trim systems in operational submersibles [2]. Buzby has analyzed the trim systems of four submersibles and recommends that dropable weights not be used except as emergency ballast [3], and that a pumping system be utilized for a reliable trim system. A trim system like that on ALVIN was recommended. Before discussing pumping systems, it is necessary to examine the conditions which will determine the trim system requirements. An examination of the DSV mission profile might produce the following requirements on the trim system.

a. Compensate for buoyancy changes of the DSV as it descends into water of increasing density. See reference 4, for a standard listing of

water density to 36,000 feet. Buzby, in reference 3 points out that "hot spots" or a sudden decrease in buoyancy must be compensated for either with the trim system or the thrusters. The buoyancy changes experienced with a DSV are not only overall trim changes, but also fore and aft trim changes.

b. Aid in control, particularly in creating bow a heavy condition while diving and a bow light condition for ascent.

c. Compensate for vehicle load changes caused by additional equipment, object recovery and loss of equipment or buoyancy material.

In considering the first condition, the designer must look at the compressibility of the sea water and the submersible. It is highly desirable to match the compressibility of the sea water and submersible, but with present buoyancy materials this does not seem feasible. The bulk modulus of water can be found in reference 5. Reference 6, discusses buoyancy materials which may apply to deep ocean uses. The discussion only covers low density liquids and low density solids, the most significant of which is syntactic foam. Reference 2 also discusses gaseous systems. Syntactic foam characteristics can be obtained from literature[6-9], but if large quantities of foam are being used in the DSV design it is recommended that commercial foam manufacturers be contacted for the specific foam characteristics.

In looking at the compressibility of the entire submersible, it can be seen that the majority of the compressibility changes occur with the pressure vessels. The compressibility of the pressure vessels can be computed from the radial deflection of the pressure vessel[10]. The loss of buoyancy due to the compression of the pressure vessel, and the change in the water density combine to give a change in buoyancy. If there is unsymmetrical pressure vessel distribution fore and aft, then there will also be a relative change in the trim conditions fore and aft.

In considering the second and third conditions, close analysis of the mission profile is essential. Most DSV's today become light-over-all since

the effective bulk modulus of the submersible is greater than the bulk modulus of water. This requires that the submersible take on water as it descends keeping neutral buoyancy or to start heavy on the surface in order to accelerate its descent rate. There are many combinations of this trade-off, all effecting the vehicle in a different manner.

In order to more intelligently approach the trade-offs, the following discussion on trim systems is presented. Many trim systems have been proposed but to date the most reliable and flexible systems is the sea water pump and hard tank systems which will be discussed below. Two ever present problems are a selection of a pump and the solubility of gasses into solution. The trade-offs involved here require specific data, so the following references are provided.

DSV Sea Water Pumps

(1) Robbins, R. W., Schneider, W. E., and Sasse, J. A., "Design and Development of a 9000 psi Sea water pump," NSRDC Annapolis Report 7-614, May 1971.

(2) Robbins R. W., Schneider, W. E. and Sasse, J. A., "Variable Ballast System Development for Deep-Diving Submersibles." NSRDC Report 3555, Nov. 1971.

(3) Lebowitz, R. K., Schneider, W. E., Sasse, J. A., and Mc Pherson, S. E., Evaluation of DSV Alvin 12,000 Foot Sea-Water Variable - Ballast System." NSRDC Annapolis Report 27-302, Nov. 1972.

Solubility of Gasses in Sea-Water

(1) K. Frolich, E. J. Tauch, J. J. Hogan and A. A. Peer, "Solubilities of Gasses in Liquids at High Pressure," Industrial and Engineering Chemistry Vol. 23, No. 5, May 1931.

(2) W. C. Eichelberger, Solubility of Air Brine at High Pressures," Industrial and Engineering Chemistry, Vol. 47, No. 11, October, 1955.

(3) I. R. Kirchevsky and J. S. Kasarnovsky, "Thermodynamical

Calculations of Solubilities of Nitrogen and Hydrogen in Water at High Pressures."

(4) The National Bureau of Standards at Boulder Dam Colorado has computerized thermodynamic packages for various gasses. These packages include everything needed for the ballast subsystem trade-offs.

Due to the problems encountered with gas solubilities, brine foaming and freezing, the recommended trim system should use an Alvin type sea water pump with a hard tank. Pre-changing the hard tanks has been suggested,which would greatly reduce the pressure head seen by the pump. This is another area which should undergo a trade-off analysis. The input should be trim system requirements and the output a trim system configuration with an operational profile.

The characteristics of the Alvin trim system are presented in Table D-1 and 20,000 foot pumps in Table D-2.

Table D-1

ALVIN VARIABLE BALLAST SYSTEM

ITEM	WEIGHT (lb)	VOL ($in.^3/ft^3$)
PUMP	86	400/.23
PUMP MOTOR (OIL FILLED)	150	2400/1.39
SHUT-OFF VALVE	11.7	59/.03
FLOAT SWITCH	18	52/.03
TANK RELIEF VALVE	1.5	NEG.
VALVE FRAME & PIPING	20	NEG.
4 WAY VALVE	5	NEG.
BY-PASS VALVE	11.7	59/0.3
P. B. VALVE	5.4	NEG.
FLOWMETER	21	80/.05
PUMP RELIEF VALVE	1.5	NEG.
CHECK VALVE	1.9	NEG.
HYD. CONT BOX (OIL FILLED)	195	3840/2.22
MOTOR-PUMP FRAME	15	NEG.
TOTAL	523	7000/4

Table D-2

20,000 FOOT SEA WATER VARIABLE BALLAST SYSTEM PROJECTIONS

ASSUME: PUMP EFFICIENCY 70%, D. C. MOTOR EFFICIENCY 68%, OVERALL SYSTEM EFFICIENCY 47%
(TAKEN AT 9,000 psi, MOTOR SPEED ABOUT 1800 rpm)

gpm.	PUMP hp. (THEO.)	MOTOR hp. (OUT)	KW (TO MOTOR)	No. PISTONS	BORE (in.)	STROKE (in.)	WEIGHT (LB.)
* 1.1	5.8	7.8	-----	5	.336	.486	90
2.0	10.5	15.0	16.7	7	.360	.540	100
2.5	13.1	18.8	20.8	7	.403	.540	110
2.5	13.1	18.8	20.8	9	.324	.684	130
3.0	15.8	22.5	25.0	9	.355	.684	130
4.0	21.0	30.0	33.3	9	.395	.700	160
5.0	26.3	37.5	41.7	11	.384	.756	200

* ACTUAL DATA, A. C. MOTOR—1735 rpm

IF FURTHER TESTING SHOWS THAT THE PUMP CAN OPERATE AT 3,000 rpm, THEN WITH 5 PISTONS A .5 in. STROKE AND .375 PISTON DIA. AND ASSUMING 70% VOLUMETRIC EFF. THE FLOW AT 9,000 psi. SHOULD BE ABOUT 2.5 gpm. (21.5 LB/min.). PUMP WEIGHT WOULD BE ABOUT 100 LB. AND POWER TO THE D. C. MOTOR SHOULD BE ABOUT 22 KW.

2. Dynamic Stability

a. Dynamic Stability: In the proposed design methodology, it is suggested that the design spiral be utilized. The final stage on each loop of the design spiral is the dynamic stability analysis. In early stages of the exploration phase, it may be necessary to assume that a certain control mechanism and degree of dynamic stability exist until a vehicle configuration can be drawn and examined. Table III-1 may be useful in selecting a feasible configuration. On the next design loop, the vehicle should be checked for stability in the horizontal and vertical plane. The equations of motion for the DSV can be derived from references 1 or 12. To examine the motion stability of the DSV, some infinitesimal disturbance from the equilibrium condition of stright ahead motion is applied to see if the DSV returns to the original equilibrium condition. Once we make this assumption we can linearize the equations of motion.

For the motion stability analysis in the horizontal plane, the yaw, sway and roll linearized equations of motion are used. The analysis basically involves the solution of three linear differential equations with constant coefficients (constant with respect to time). The solution is characterized by three normal modes which are listed below:

*Roll convergence: a single degree of freedom motion about the roll axis (x axis).

*Spiral Mode: Primarily a planar yaw-sway motion.

*Dutch Roll: Basically a roll-sway oscillation.

The roll convergence mode is typically a first order root of the fourth order stability determinant. Submersibles tend to be stable in this mode due to the roll damping of the fins, appendages and the action of the center of gravity. The spiral mode is typically a first order root. If the first order root is negative then the submersible has stable motion stability or has straight line stability. The Dutch roll mode is typically a second order root and is generally stable for submersibles. An explanation of these modes with examples can be found in reference 13 and Appendix

B of reference 12.

For the motion stability analysis in the vertical plane, the surge, heave and pitch linearized equations of motion are used. The analysis is similar to that for the horizontal plane. The form is about the same except that the BG may be the dominating effect. An aside note, the performance of the vehicle in the vertical plane during ascent and descent will be largely governed by the compressibility effects.

b. Maneuverability: An analysis of the maneuverability of a DSV can be accomplished with linearized theory for moderate maneuvers, for more realistic maneuvers where large angles of attack or toe-in angles would occur it is recommended that higher order non-linear equations be utilized in order to provide meaningful results. The maneuverability of a DSV is critical during mating, collision avoidance, close-up photography and backing out of caves or overhangs which the vehicle has accidently entered[3]. In some of these examples, the vehicle speed is so small that a different approach should be taken, like backing down or use of thrusters.

The stability and maneuverability analyses of present submersibles have tended to follow the analysis outlined above and when higher order non-linear quantities of motion were needed simulation was utilized. A word of caution, many of the analyses published are loaded with errors. This is not meant to be a criticism, but a word of warning as to the complexity of the problem.

The trade-off involved is basically one of determining the degree of stability and maneuverability desired. In DSV design, the bare hull tends to be inherently unstable, but becomes progressively more stable as the stern planes are increased in size. Bow planes tend to have a destabilizing effect. Another trade-off with the stern planes involves the relationship between the surface area the the aspect ratio. The span of

the stern planes is generally limited by operational constraints, such as potential stern plane collision with the mother submarine during mating which limits the stern plane's span to the maximum radius of the boat. This implies that the stern plane chord must be increased to increase the surface area. For some DSV's, this presents the vehicle with a low aspect ratio stern plane. At this stage the reduction in aspect ratio caused by an increase in fin chord may decrease the lift curve slope of the fin enough to make an increase in surface area useless.

Another trade-off involves the selection of control surfaces. The basic choices are a shroud, stabilizer-flap assembly (like on aircraft) and a full plane. The trade-off should look at operational constraints, dynamic stability and maneuverability implications.

c. Propulsion and Maneuvering Devices: The selection of propulsion and maneuvering devices for a submersible is largely dependent upon the mission requirements and the associated speed regime. For example, the requirement for good maneuverability from 0-4 knots will require different devices at different speeds. The forces and moments generated by the planes varies as the square of the velocity. At speeds below a knot, the effectiveness of the planes on vehicle control drops significantly, and devices such as thruster are usually employed.

There are many propulsion and maneuvering devices available to the designer. Some of the more feasible devices are listed below:

(1) Free propellers

(2) Nozzled propellers

(3) Contra-rotating propellers

(4) Cyclic propellers

(a) Haselton

(b) Voith - Schneider

(5) Ducted propellers

(a) Axial flow impeller

(b) English inlet

(6) Hydrojets (radial flow impeller)

(7) Plane deflected hydrojets (Coanada effect)

(8) Free planes

(9) Augmented planes (jet flaps)

(10) Shrouds

(11) Variable Trim System (Discussed under static stability)

Many of the above devices have been used on operational submersibles. As mentioned before, Table III-1 summarizes maneuvering and control systems used on submersibles. A few are mentioned below:[14]

*Star 1 and Star II use two ducted propellers capable of rotation through 360 degrees in the vertical plane to provide force and moment in four degrees of freedom and a rudder and bow thruster to provide yaw and sway control.

*Aluminant uses conventional propeller, rudder and planes for longitudinal thrust and control and a vertical thruster for control of heave.

* Soucoupe uses multiple rotable jets of water for propulsion and control.

* Deep Quest and DSRV use a conventional propeller with a movable shroud for cruise control and ducted jets for hovering control.

The selection of one of these systems depends largely on configuration convenience and the required hydrodynamic efficiency or more specifically the trade-offs investigated should be:

* Power requirements: This should include peak load requirements established from the mission profile and total power required for a given mission endurance.

* Maximum thrust and moment capability: These are determined by device location, efficiency, vehicle constraints, and mission constraints. For example, maximum speed over the bottom for an optical mapping mission is limited to two knots because of the strobe recycle

time, therefore an efficient two knot cruise capability is desirable. Another example, if planes are selected as control devices, then the type of plane allowed will be highly dependent upon the designers ability to select an adequate shaft and shaft location for the plane. (ie. adequate from a strength analysis).

*Weight and size of the hardware: The actual weight and size of the propulsion has not been a problem as much asweight and size of the power train. The power train consists of gear reduction boxes, motors and converters or inverters. In general, hydraulic systems are better than electric systems in this trade-off.

*Total weight of system for a given mission profile: In this trade-off nothing can be said about the selection of hydraulic vs. electric, since they come out about equal overall. The hydraulic systems have lighted components, but are less efficient requiring more power.

*Mud stirring characteristics: This is an area essential to the successful completion of some missions. A photographic mission is terminated if the bottom is stirred up. If there are bottom currents, the problem is reduced, but it is felt (see section on environmental analysis Appendix E) that in general, the deep ocean bottom currents are small, requiring long particulate settlement times.

*Vulnerability: This is highly dependent upon the mission requirements and profile. Restricting requirements are bottom sitting, submerged mating and collision avoidance.

*Dynamic Stability: As discussed previously each of the above devices effects the dynamic stability of the vehicle. As the design progresses it is imperative that some sort of vehicle simulation be undertaken.

Many trade-off studies have been done in this area, but references 15 and 16 are two of the complete trade-offs done for vehicles requiring high degrees of manuevera bility and control. Reference 15, is Lockheed's DSRV control trade-off study and compares the shrouded propeller-ducted thruster to the Haselton tandem propeller configuration. The tandem propeller configuration consisted of two rotating rings of variable pitch

propellers mounted fore and aft. The shrouded propeller-ducted thruster configuration consisted of a shrouded propeller aft with 2 horizontal and 2 vertical thrusters. Reference 16 is a summary of the Lockheed DSSV control trade-off. It examined four basic configurations:

* Open propeller with ducted thrusters and planes
* Knot nozzles located amidships
* Haselton propellers; two equal units sized to maneuvering requirements.
* Haselton propellers, two units, forward unit sized to maneuvering requirements, aft unit sized for 5.0 kt fwd speed.

From the trade-offs mentioned above the free propeller with planes and ducted thrusters was best overall. This will be the system used for the DSV design.

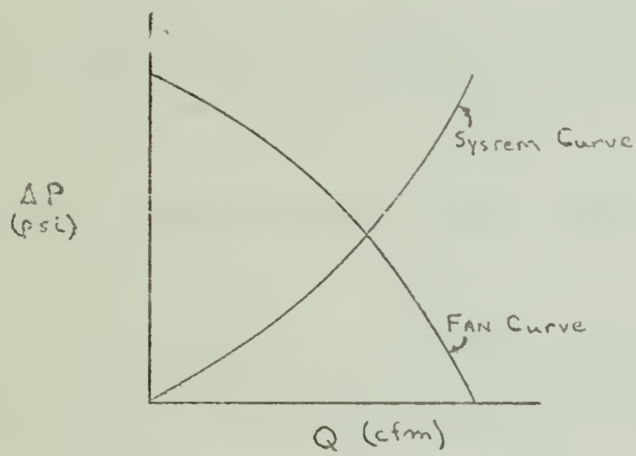
d. Thruster Design - The effect of thrusters on the dynamic stability of a DSV can best be analyzed in the simulation phase of the stability analysis and will not be discussed. The discussion below breifly outlines how thrusters can be sized for a given mission profile using cross flow drag theory. References17-19, describes the theory necessary for cross flow drag analysis. Once the cross flow drag coefficient is calculated, the analysis can proceed as does any drag calculation. Due to the nature of calculation the area term will be projected area instead of wetted surface area.

If continuous speed control is not provided, the thrusters can be sized so that no moment is produced about the vehicle center of pressure for a required sway or heave velocity. This can be very helpful to the pilot since it helps to reduce some of his control problems at low speeds.

The trade-offs involved in thruster design are basically those of power transmission, location and duct size vs. motor size. The thruster motors can be either hydraulic or electric. These trade-offs were discussed in Appendix B. The thrusters can be located internal to the hull in ducts

or external to the hull. Alvin and Aluminant use external thrusters whereas the DSRV uses internal ducted thrusters. The thruster location can be effected by the vehicle drag, stability and maneuverability considerations. A 360 degree gimbeled thruster like that on Alvin is extremely useful for maneuvering in tight spots. The duct size vs. motor size trade off is the same one that is involved in fan design where the motor characteristics are matched to the duct characteristics to provide an optimum design. Overlapping plots of the fan characteristics and system characteristics provide the deisgner with a graphical trade-off tool.

The figure below is an example of this trade-off.



It is felt that there is enough open literature and data available for optimum propeller and plane design, therefore no discussion will be presented in this paper.

References

1. "Principles of Naval Architecture," Edited by J. P. Comstock, SNAME, 1967.

2. Nobaica, E. C., "Buoyancy Systems for Deep Submergence Structures," Naval Engineers Journal, Oct., 1964.

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10. Roark, R. J., "Formulas for Stress and Strain," McGraw-Hill Book Company, 1965.

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16. Zieden, A., "Alternate Propulsion Systems, Lockheed Missile and Space Company "DSSV Final Report LMSC-T-14-68-1 Appendix A, Vol II, Part 3, Section 11, Study 1 April 1968.

References (Cond't)

17. Hoerner, Ing S. F., "Fluid-Dynamic Drag," Published by Author, Dr. Ing S. F. Hoerner, 148 Busteed Drive, Midland Park, New Jersey, 07432.

18. Thwaites, Bryan, "Incompressible Aerodynamics," Oxford University Press, 1960.

19. Flax, A. H. and Lawrence H. R., "The Aerodynamics of Low Aspect Ratio Wings and Wing Body Combinations," Third Anglo-American Conference, 1961.

Appendix E

Power

The purpose of this appendix is to provide the designer with a review of the conceptual trade-offs and literature which can aid in the optimization of the over-all system. The discussion will be divided into two sections:

1. Energy Sources:
   a. Storage
   b. Conversion
2. Power transmission

Specific data will not be provided since the presentation of this data would seem redundant with the references listed.

1. Energy Sources

In keeping with the proposed design methodology it is essential that feasible energy sources be developed from a set of alternatives. The generation of the energy source alternatives can be achieved by dividing the problem into two parts, energy storage and energy conversion.

a. Energy Storage: Energy may be stored in many forms such as mechanical, thermal, electrical, fuel/oxidant and nuclear. Possible trade-offs will be listed by the energy storage forms above. This list is not all inclusive; but appears to be the state-of-the-art to date.

(1) Mechanical Storage Devices

(a) Spring motors

(b) Hydraulic accumulators

(c) Flywheels

(d) Compressed gas

At present the super flywheel has the highest energy storage capacity. The references listed below may be helpful in this trade-off.

*. Dugger, G. L. et al, "Flywheel and Flywheel/Heat Engine Hybrid Propulsion Systems for Low-Emission Vehicles, "Paper 71949, Proceeding of 1971 Intersociety Energy Conversion Engineering Conference, Boston, Mass., 3-5 Aug. 1971

*. Lawson, L. J. and Hellman, K. H., "Design and Testing of High Energy Density Flywheels for Application to Flywheel/Heat Engine Hybrid Vehicle Drives," Paper 719150, Proceeding of 1971 Intersociety Energy Conversion Engineering Conference, Boston, Mass. 3-5 Aug. 1971

*. Rabenhorst, D. W., "Potential Applications for the Superflywheel," Paper 719148, Proceedings of 1971 Intersociety Energy Conversion Engineering Conference, Boston, Mass., 3-5 Aug. 1971.

(2) Thermal Storage Devices: A suitable heat storage material should have a high heat capacity, low vapor pressure, high density, be chemically stable and compatible with containment and heat-transfer materials. Possible alternatives for thermal storage services can be found in the following references:

*. "Proceedings of AIAA Second Propulsion Joint Specialist Conference (U)," Technical Report TRD-TR-66-1, Volume V, Sep. 1966.

*. Morrison, T. D., et al, "Thermal Energy Storage Systems for Porpulsion of Small Submarines (U)," MEL R&D Rept. 167/65, Aug. 1965. (Note: MEL R & D is now NSRDC Annapolis, Md.)

*. Lynch J. F., et al, "Engineering Properties of Selected Ceramic Materials," Battle Memorial Institute study published by the American Ceramic Society, Inc. 1966.

*. "Handbook of Thermophysical Properties of Solid Materials," Goldsmith, A., et al, Revised Edition Volume III: Ceramics, The Macmillan Company, New York (1961).

*. "High Temperature Ceramic Heat Exchangers, A Summary of FluiDyne Capability for Development of Thermal Energy Storage Systems "FluiDyne Engineering Corp., Minneapolis, Minn. (1971)

(3)Electrical Storage Devices: There are two relatively long term storage devices, capacitors and electrochemical batteries. The storage capacity of the electro chemical batteries is an order of magnitude greater than capacitors, and therefore the capacitors will not be considered. The possible batteries under consideration are listed below with capacities.

Type	Capacity $\left(\frac{\text{Watt-hr}}{\text{lb}}\right)$
Zine Silver oxide	30-80
Silver-Cadmium	15-40
Nickel-Iron	10-18
Lead-Acid	10-15

The references below contain specific data on these systems.

*. "Sea space Power Systems, " Data catalog, Exide Power Systems Division, E. S. B. Incorporated, Philadelphia, Penn.

*. Myers, J. J., Holm, C. H. and McAllister, Rr. R., "Handbook of Ocean and Underwater Engineering," McGraw-Hill, 1969.

*. Friedland; N. "Propulsion of Deep Submergence Vehicles," Society of Automotive Engineers, Cleveland Ohio Meeting, Oct. 21, 1965. (Gives discharge times of batteries, outstanding)

* Lead Acid
* Silver Zinc
* Nickel Cadmium
* Silver Cadmium

*. "DSSV Phase 1 Final Report," Westinghouse Underseas Division, Annapolis, Md., Vol. 5, Part 5, June 1968.

"Trade-offs considered the following systems for a 20,000 ft. search vehicle."

H_2/O_2 Fuel cell

Hydrazine/H_2O_2 Fuel cell

Secondary batteries

Closed Cylce Internal Combustion Engines

(4) Fuel/Oxidant Energy Storage: The reactions of fuel with oxidants and their suitability for use in a submersibles has been studied extensively. In the trade-off studyies, comparisons should be made with respect to cost, storage stability, supply, distribution, energy density and effects on the DSV operating characteristics.

The references listed below present the specific trade-offs listed above.

*. Morrison, T. D., et al, "Power Plants for Deep Ocean Vehicles," Paper No. 5, Society of Naval Architects and Marine Engineers, Philadelphia, Penn. May 1966.

*. "Proceedings of the Conference on Energy Sources of Extended Endurance in the 1-100 KW Range for Naval Applications (U)," Technical Report TRD-TR-1, Volume V, Sept. 1966.

*. Moore, R. W., "Submarine Propulsion System Study (U)," MEL-sponsored Rept. 14/65 by Westinghouse Electric Corporation, East Pittsburgh, Penn. under contract N600 (61533) 61394, 5 Jan 1965.

(5) Nuclear Sources: Nuclear sources can be divided into two basic systems, fission reactors and radioisotope systems. There are a large number of possible alternatives which will not be listed, but can be

*. Jaffer, H., Buck, K., McCluer, H., Montgomery H., and Finkle, E., "The Application of Nuclear Reactors and Radioisosopes to Under-water - Vehicle Propulsion," ASME Paper No. 65 UNT-6, May 1965. "Good review of potential application for submersible propulsion systems."

*. Myers, J. J., Holm, C. H. and McAllister, R. F., "Handbook of Ocean and Underwater Engineering" McGraw-Hill, 1969. "Has a small bibliography which may be helpful."

*. Carmichael, A. D., "Ocean Engineering Power Systems," MIT Sea Grant Program, Cambridge, Mass. Report No. MITSG74-15, Index No. 74-115 No. "Describes various types of nuclear energy sources."

b. Energy Conversion: Energy Converters may be classified into two basic categories, Q-engines and E-engines.

(1) A Q-engine is a cyclic device which requires that the internal energy of the fuel be converted to heat and then transferred to a working fluid in the engine. Steam engines, thermionic and thermoelectric generators, not in engines such as the Stirling engine and closed-cycle gas turbines are examples of Q-engines.

*. Balukjian, H., "A Closed Brayton Cycle Power Plant for Under-water Applications and Comparison with a Fuel Cell," Seventh Annual Technical Symposium, Association of Senior Engineers, Naval Ship Systems Command (1970).

*. Mattavi, J. N., et al, "The Stirling Engine for Under-water Vehicle Applications," SAE Paper 690731, National Power plant meeting, Cleveland, Ohio (27-29 Oct. 1969).

*. "The Stirling Engine for Underwater Applications," brochure K. B. United Stirling (Sweden) AB & CO, Fack, S-20110 Malmo 1, Sweden.

*. Milligan, H. H., Brandes, P. J., "The Development of the Closed-Brayton Cycle Power Conversion System," AIAA Paper 66-889 2 Dec. 1966.

*. Mock, E. A., Balukjian, H., "Undersea Applications of a Closed Brayton Cycle Powerplant," SAE Paper 710828, St. Louis, Mo. (26-29 Oct. 1971.)

*. Klann, J. L., and Wintucky, J. W., "Status of 2-15 KW Brayton Power System and Potential Gains from Component Improvements," Paper 719027, Intersociety Energy Conversion Engineering Conference, Boston, Mass. (3 - 5 Aug. 1971).

*. Secunde, R. R., et al "Experimental Evaluation of the Electric Subsystem of the 2-15 KW Brayton Power Conversion System," Paper 719031, Intersociety Energy Conversion Engineering Conference, Boston, Mass. (3-5 Aug. 1971).

(2) An E-engine utilizes the internal energy of the fuel directly in the E-engines. The second reference under fuel/oxidant energy storage has a summary of the information contained above.

* "Fuel Cell Power Systems for DSSV," Pratt and Whitney Aircraft Report, Contract N00024-69-C-0242, 12 Aug. 1969.

*. Warszawski, B., and Dumas, Jl, "Alsthom Fuel Cells for Marine and Submarine Applications," Marine Technology Society Journal, Jan.-Feb. 1971.

*. "Final Technical Report for Advanced Submarine Concept Study (U)," by Westinghouse Ocean Research and Engineering Center for Office of Naval Research under Contract No. N00014-71-C-0085

(dated 16 June 1971), Secret.

*. Sanderson, R. A., et al, "Fuel Cell Powerplant for Deep-submergence Cehicles," SAE Paper 710826, St. Louis, Mo. (26-29 Oct. 1971).

*. Gormley, D. R. and J. H. Harrison, "A liquid Reactant Fuel Cell Power System Design for Underwater Applieation," Paper 719073, Intersociety Energy Conversion Engineering Conference, Boston, Mass. (3-5 Aug. 1971).

*. Buechler, L. W., and Klots, C. E., "Fuel Cells and their Application to Undersea Power," SNAME, Marine Power Plants, where are they Healed 1966.

So far a brief sketch of the possible alternatives has been presented. In each new vehicle design, the specific vehicle power requirements must be fed into this aspect of the design process. Feasible alternatives can be generated and a final system or systems chosen for more detailed trade-off studies.

Within the past ten years, many trade-off studies have been conducted. References 1 through 4 listed below may be helpful to the designer in their graphical presentation of the energy source trade-offs. Figures E-1 and E-2 are representative of the graphical trade-offs. A rough rule of thumb which summarizes the energy sources trade-off for minimum weight design is as follows:

1. Up to one hour missions   zinc-silver oxide battery
2. 1-15 hour missions   thermal power plants
3. 15-140 hour missions   fuel cell
4. Above 140 hr. missions   radioactive isotopes.

*. Huschildt, H. "Project Seabed Panel III, Sub panel 5, Propulsion/

Machinery Report," DTMB-C-1931, May 1965. "Comprehensive Review of Promising Alternatives for Power Systems."

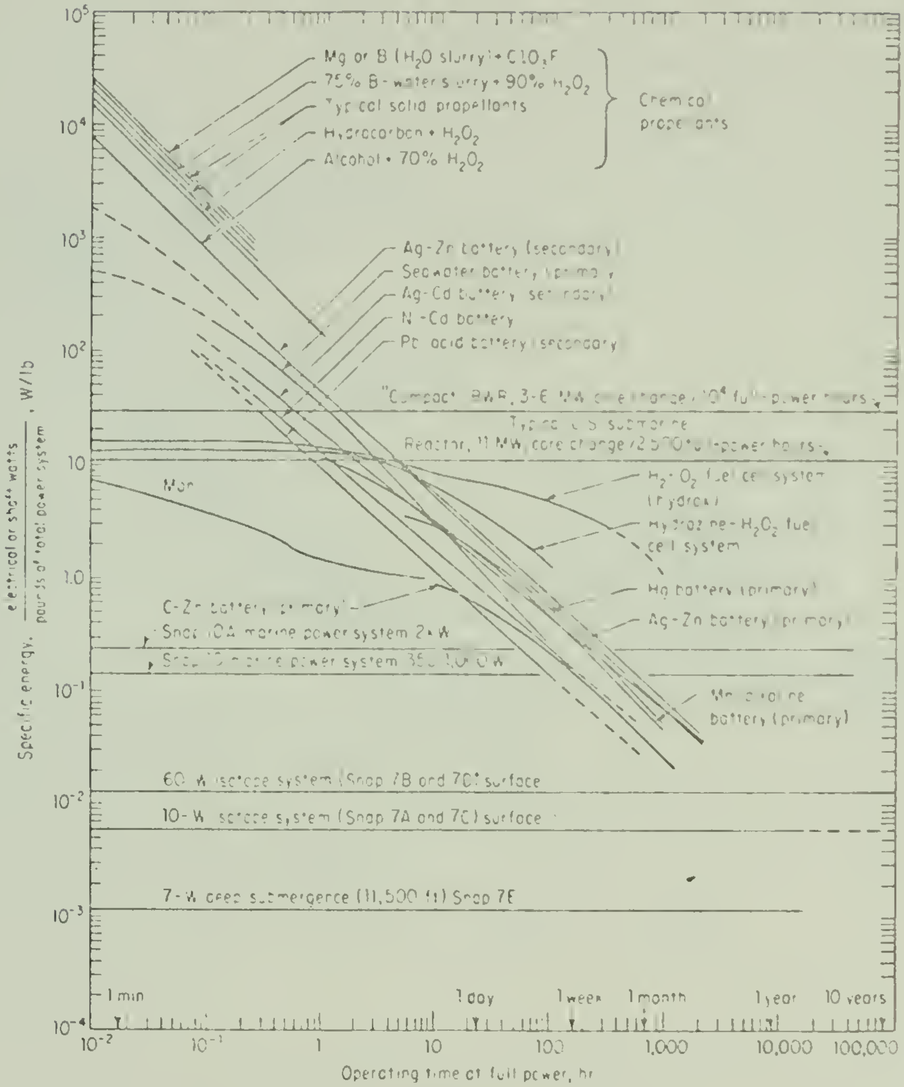
*. Spadone D. M., "Power for Deep Submergence Vehicles," Astor-nautics and Aero Nautics, July 1967. An overview of promising power systems summarizing the potential of these systems specific weight as a function of mission time.

*. Kinsimger, Walter W., "Propulsion of Deep Diving Submersibles," Naval Engineers Journal Aug. 1965. A good review of present sub-mersible power sources and a review of power sources and their trade-offs.

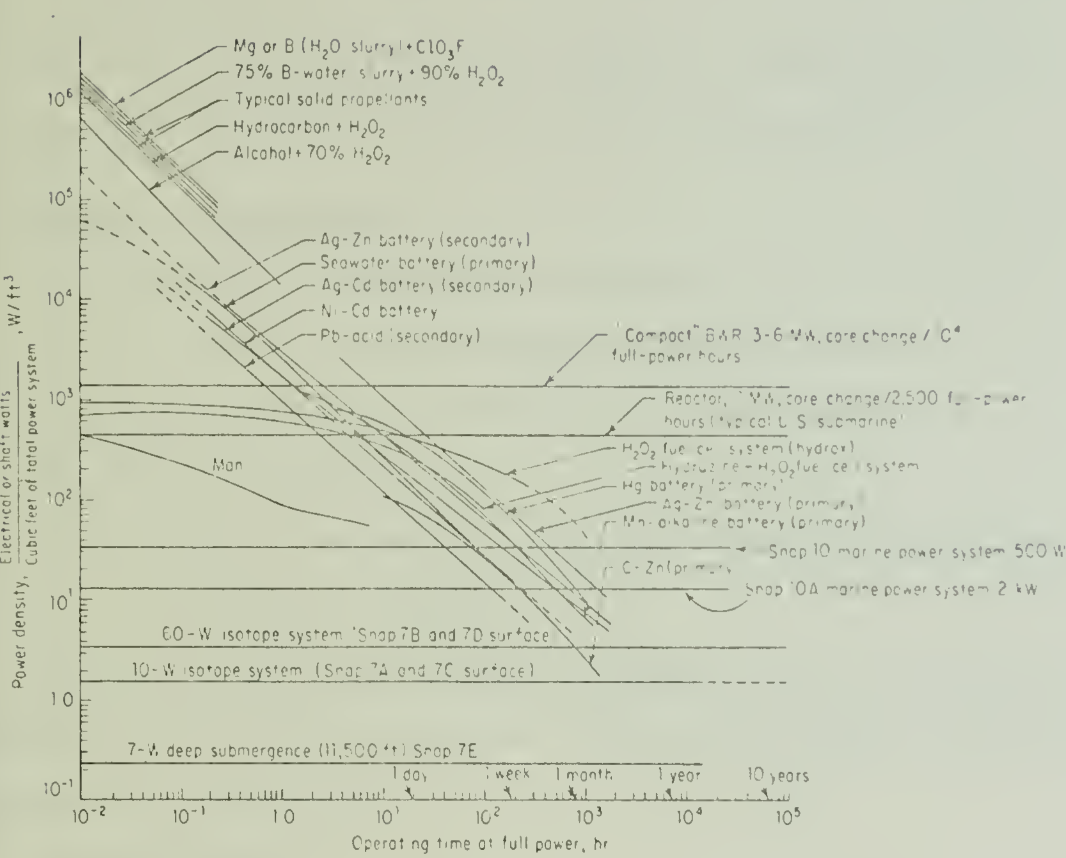
*. Myers, J. J., Holm, C. H. and McAllister, R. F., "Handbook of Ocean And Underwater Engineering." McGraw-Hill, 1969.

2. Power Transmission

In the field of power transmission from the energy source to the propulsor, there are only a few feasible trade-offs available to the designer. The system should be compact, light in weight, have a high efficiency at its principal operating speed and loads, generate as little noise as possible and be capable of providing mechanical, hydraulic and electric power to energize the various vehicle systems mentioned in Appendix A. The propulsors under the power system, the sensor system, the control system and the environmental systems will utilize AC electrical power, but the propulsors and controls can use hydraulic, electric (AC-DC) or hydraulic/electric power. The energy sources mentioned previously output either AC or DC electric energy. The trade-offs involve locating the power system (ie. wet or dry) and selection of power conversion mode. The dry systems have problems with shaft seals, but the wet systems generally have lower efficiencies.



Specific Energy Trade-Off
Figure E-1



Power Density Trade-Off

Figure E-2

In the selection of the motor type, the trade-off is basically electric versus hydraulic. The various advantages are listed below:

Hydraulic:

1. Simpler overall system
2. Small motor at propulsor location where space is usually at a premium.
3. No electric controller is required to regulate speed.
4. Requires no gear reduction box for main propulsion.

Electric (AC-DC Motors)

1. Generally a higher overall efficiency.
2. Direct drive, no intermediate conversion system.
3. Cables are easier to arrange then hydraulic lines.
4. Easier to maintain.
5. Custom made components, are easier to purchase commerically than hydraulic components.

When selecting either system, the degree or type of speed control desired will be the deciding factor. The various problems and trade-offs involved are listed below.

Electric Motors

1. Direct use of energy source power eliminates inverter losses.
2. Easy speed control if only a few speeds are required. The controls basically change the voltage supplied to the motor through a simple switching system.
3. DC motors have had commutation problems along with contamination of the compensating fluid in wet systems.

AC-Motors

1. If the energy source produces DC power, it must be converted to AC along with a loss in efficiency and the generation of a potential problem area. In the past, inverters have had a history of problems. In order to avoid future problems with inverters, sufficient containment space should be allowed for future growth in the vehicles electrical loads. To date free flooding AC-DC inverters have not become feasible, producing a containment weight penalty.

2. Squirrel cage AC induction motors are desirable since they eliminate the commutation problem associated with DC motors.

3. Precise speed control is a problem.

Hydraulic Motors

1. Precise speed control is available with a hydraulic motor.
2. DC signals can be used to control the hydraulic valves eliminating a sophisticated electronic control system.
3. A hydraulic motor system still requires an electric pump to provide the hydraulic power. The pump is usually powered by an AC or DC motor.

A good reference on practical experience with electrical drive systems is listed below.

*. Bloomquist, D. L., "Experience with Electric Drive Systems for Deep Submergence," Paper 719077, Proceedings ff1971 Intersociety Energy Conversion Engineering Conference, Boston, Mass. (3-5 Aug. 1971).

Appendix F
Configuration

As covered in Appendix A, Design Process, trade-offs must be carried out on each system in order to allocate equipments to fulfill the various system requirements. If each subsystem was independent of all others, then optimization would not be a problem, but the subsystems are dependent upon one another. One example, is the interrelationship between the pressure envelopes and the overall vehicle weight. At the beginning of any submersible design, the pressure envelope configuration must be established.

It is proposed that a realistic configuration trade-off cannot be achieved by comparing equal internal volumes. To a large extent, packing factors of various geometry do not allow an equivalent internal volume study. The ba 'c strategy involved in this methodology is to establish a baseline of equipments which satisfy the subsystem requirements. Within each subsystem the various subsystems requirements should be ranked so that the design criteria are optimized. As the subsystems are combined, a continuous evaluation should be made as to the compatibility of all subsystems. Once compatibility has been established pressure envelopes should be generated which can accommodate equipments and personnel.

The proposed design criteria and performance criteria should be used as a guide for determining if equipment should be located inside or outside the pressure envelope.

<u>Design Criteria</u>

*Mission safety
*Mission reliability
*Maintainability
*Vehicle weight
*Habitability

Performance Criteria

*Repeatability of mission

*Three man crew

*Vehicle weight less than 80,000 lbs.

The following list is given as an outline for the deisgner to use in determining if an equipment system is placed inside or outside the manned pressure envelope.

Equipment Inside If:

*Mission success depends upon specific components.

*On board maintenance is possible and feasible.

*Reliability is low or marginal.

*The mission requires it; such as submerged mission repeatability which requires.

(1) Inside photography film/camera

(2) Waste disposal

(3) Life support replenishment

(4) Power supply replenishment

*Sub-systems require

(1) Mannual attendance

(2) Manual control

Equipment Outside If:

*Weight or volume advantage is achieved

*Free flooding, pressure compensated or independent containment is a weight savings and therefore desirable.

Using these guidelines and criteria, a listing of inside equipments and displays can be produced. At this stage, man must be placed into the design spiral, figure A-1, in order to achieve the optimum man-machine interface. The process is iterative due to the interdependence of the systems. Man is placed into the design spiral in the form of space requirements for various aspects of the mission such as mapping, close-up photography and mating.

In order to optimize the design criteria mockups should be made, and criteria such as habitability and maintainability should be simulated Figures F-1 and F-2 have been provided to aid the designer in the drawing and mock-up stage. The 95th percentile represents the limits on the operators head and leg room whereas the 5th percentile represents the maximum reach attainable and therefore the position of the upper control panels. The operators position whether standing, seated, bending or lying down should be determined by the mission requirements and endurance.

The mission requirements for men should be determined from the following.

System requirements imply equipments and displays. From this the number of men should be determined. It is recommended that actual operators be used to determine if the configuration is feasible.

Human Engineering

The number of men required and mission profile determines the rest space and workspace which should be provided. Work includes, viewing, piloting, navigation, system monitoring, and possibility of task rotation.

The configuration should be capable of handling any specified percentile. It is felt that a certain percentile should not be specified as in the space program, unless the deep submergence program developes a large training program for selected individuals.

Structural Limitations

By now the designer should have sufficient information to determining the required internal volume. The pressure vessel configuration should be driven by a number of criteria:

*W/ Dratio and in-air weight or effective wieght.

*Number of viewports, viewport location, and minimum viewport spacing

*Penetrators

*Flooring

*Access hatch accessability into control space

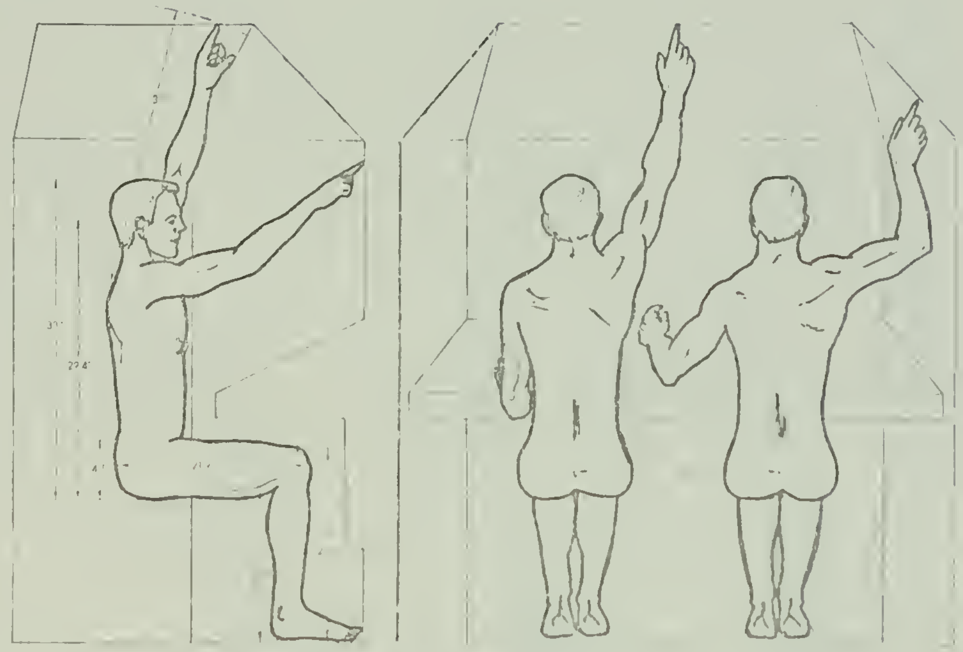


Figure F-1
5th Percentile

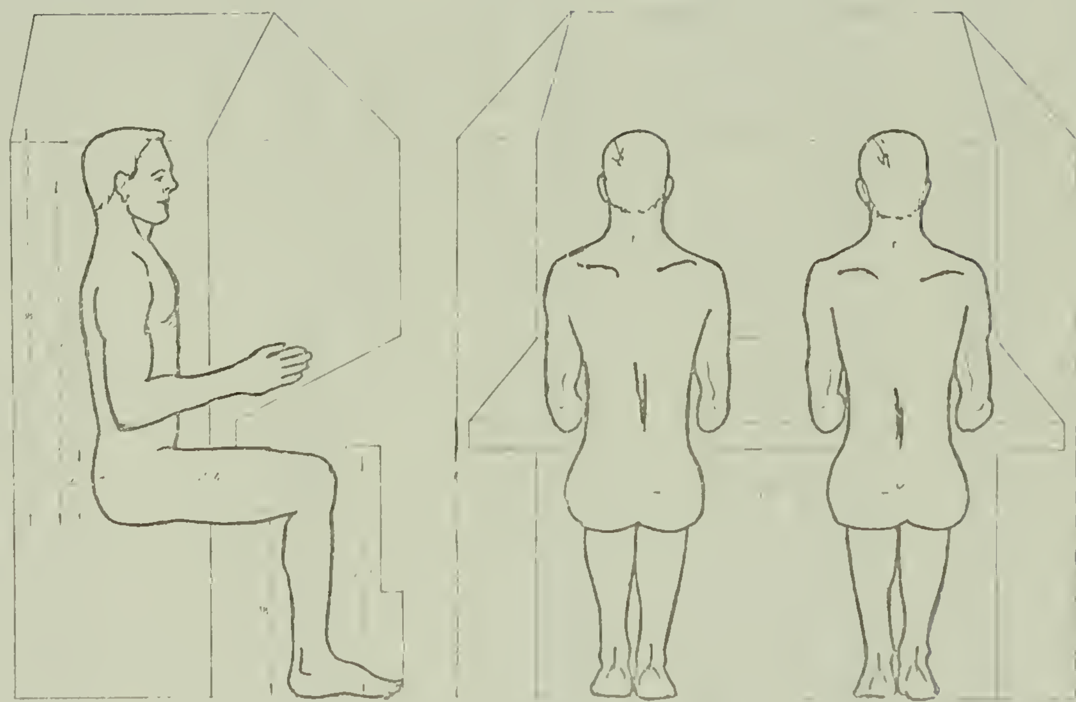


Figure F-2. 95th Percentile

*Insulation

*Heat exchanger space requiring inside surface of pressure envelope

*Equipment supports

*Framing and web-flange of stiffened shell.

The following paragraph will discuss the above items in more detail.

1. The requirement for overlapping viewing for pilot and observer is limited by:

a. The minimum viewport spacing which is determined by an acceptable viewing overlap and the stress concentration decay length.

b. Accessibility to viewports for the duration of a mission without excessive degradation of the pilot or observer effectiveness.

c. The type of mission

(1) mapping requires down looking viewports

(2) surfacing or ascent requires an upward looking viewport.

d. The type of viewports and viewing aids

(1) Flat window

(2) Spherical windows

e. Sphere size and its effect on curvature, and minimum viewport spacing.

f. Fabrication limitations.

2. <u>Penetrators</u>

A build-up of material around the penetrators is required to cope with the stress concentrations. The space loss due to the feedthrough dimensions can be significatn. Penetrator location should be near the internal electrical distribution boxes to aid in the reduction of cabling. Build-up around hatches or viewports should be considered as a possible sight for penetrators.

3. Flooring is not essential, but can be helpful for storage of certain types of equipment. It should be remembered that unless the thermal control system is powerful, the bilge area will collect condensate. Electronics equipment should not be stored under the flooring.

4. Access space should be compatable with the hatch dimension so that equipment can be maintained. Anything more than the compatability should be carefully evaluated since entering or exiting the vehicle only involves a small part of the mission.

5. Insulation is essential for a deep submersible. The temperature below 1000 meters is a constant 32° F whereas temperatures on the surface can get up to 110° F. Therefore, insulation is essential for control of the inside environment. To date the insulation thickness averages about 1 inch. This adds 2 inches to the inside diameter of the pressure envelope.

6. Heat Exchanger Spacing - Most operational submersibles use a hull heat exchangers. This type of cooling system requires a certain amount of inside hull area. This hull area is a function of the:

*Thermal conductability of hull
*Efficiency of heat exchanger
*Heating load
*Circulation around outside of pressure envelope.

This necessity for direct hull contact places a restriction on possible vehicle configuration especially systems with internal stiffeners.

7. Equipment supports add additional weight to the hull structure. These weights are generally not very large.

8. The internal framing can either help or hinder the internal arrangement. Preliminary estimates should be made to observe the effects of web-flange-assembly and its utility.

In the design of the DSV, and inboard profile should be made on each possible pressure vessel configuration. The configuration trade-off has been simplified by the fabrication of a computer program. The input to the program is material properties and pressure vessel dimension and outputs optimum W/D, shell thickness, framesize and frame spacing. The configurations considered are listed below

*Sphere
*Unstiffened Spheroid
*Stiffened Spheroid
*Cylinder (spherical segment ends)
*Stiffness cylinder (prolate spheroidal ends)
*Nested sphere
*Connected sphere

Appendix G considers the possible configuration analysis and failure mode. In making the above comparison it must be insured that each configuration is being compared by the governing failure mode. This failure mode can be determined by the criteria in Appendix G.

Configuration trade-offs should be calculated as soon as possible in the exploration phase as they will help to bracket the material and configuration which can be considered. As discussed in Appendix H, the vehicle in-air weight is the limiting factor on the configuration of a weight limited design which will be used to establish a lower bound on the various materials stress ranges. Since the design process is iterative in nature the configuration should be checked for optimization at each loop. For example, a single sphere may be chosen as the optimum configuration in the exploration phase but as the design progresses it is found that more internal volume is required or a more efficient geometry on the present volume is required. At this point, a new configuration may be chosen subject to the afore mentioned criteria.

Appendix G
Structural Analysis

1. Introduction

In Appendix F, the pressure vessel envelope was established for various configurations. In analyzing the envelope, various failure modes and loading conditions are considered to determine to governing loading condition for a given failure mode. The general modes of failure are instability, yielding, and fracture. The possible loading conditions experienced by the pressure envelope are listed below.

a. Static pressure loading extremes:

(1) Envelopes subjected to external static loading

(2) Envelopes subjected to internal static loading.

b. Cyclic loading only becomes a problem if the pressure vessel at some time experiences tensile stresses. A vessel with internal pressure or with residual stresses in the shell would experience cyclic loading.

c. Dynamic loading:

(1) In transportation

(a) Vibrations

(b) G-loading (acceleration loads)

(c) Bending loads

(2) Impact

(a) Mechanical impact or collision could occur where the structure or shell itself sustains an impact load. This type of loading is a localized elastic or plastic deformation.

(b) Internal or external superstructure shock or explosive loading. This type of loading could occur from tanks exploding, or other reasions listed in Busby's paper on the hazards of the deep. (See reference 4 of main paper.)

In this design methodology, it is proposed that for a DSV, the governing loading condition in the exploration phase and the initial iterations of the preliminary design stage is the hydrostatic loading condition.

In the sequence of design, it is natural to consider hydrostatic loading first, not only because of themagnitude of the load, but also because the other two loading conditions require a vehicle configuration with some type of mission profile to model the possible loading conditions.

This appendix will evaluate the hydrostatic and static loading conditions seeking to establish a performance criterion for the pressure envelope. The structural analysis will examine failure by instability and yielding. Two criteria will be established, one for external pressure and one for internal pressure.

Appendix H will examine failure by fracture as an input into the material selection process. The cyclic and dynamic loading conditions will also be examined in Appendix H.

Various external loadings: like hydrodynamic loads, wave-bending, vibration, and g-loadir for transportation, will not be considered in this paper, but should be checked as the design progresses in the preliminary design stage.

2. Safety Analysis:

At the outset of any analysis, assumptions are made in order to execute the analysis. These assumptions represent possible uncertainties in the analysis which in the past have been grouped together and labelled: "factor of safety." The factor of safety has incorporated the following uncertainties.

* Inaccuracy of strength analysis tools
* Material inhomogenity
* Inaccuracies in fabircation and out-of-roundness
* Exceeding operation depth:
    (i)instrumentation
    (ii)operator error
* Fatigue
* Corrosion
* Dynamic loading

In the past, the factor of safety has varied from 3.0 for TRIESTE to 1.4 for rockets. Most shallow submersibles, less than 5000 feet, have applied a 1.5 factor of safety. The question is, do the above partial factors remain constant with depth ?

A qualitative analysis of each partial factor should be conducted at the outset of the structural analysis. A proposed analysis would assign a proportional overpressure factor if the factor is a function of depth and a fixed overpressure factor if the factor isnot a function of depth.

The inaccuracies associated with the strength analysis tools and depth measurement should be assigned a fixed percentage. The overshoot problem caused by operation error should be analyzed based on a casualty recovery analysis, which is in turn based on vehicle dynamics and hydrostatic recovery.

Fatigue is more cyclic-dependent than pressure-dependent. The analysis in Appendix H will not consider fatigue as a governing partial factor in the factor of safety analysis, but will evaluate it as something which can be designed against later in the design.

Corrosion is more time-dependent than pressure-dependent, but can cause a degradation in the vehicles factor of safety in time. A fixed overpressure could be assigned, if general corrosion is a problem. Since corrosion is material dependent, the corrosion analysis will be examined separately, later in the design.

Dynamic loading is not a function of depth. For a DSV, the dynamic loads on the pressure envelope are probably decreased because of the compressive nature of the shell loads. At slow speeds it may be difficult to initiate an impact of sufficient magnitude to relieve the compression. As mentioned previously, Appendix H will discuss this in more detail.

The uncertainties involved in material inhomogenity, inaccuracies in fabrication and out-of-roundness can be accounted for by proper quality control.

This qualitative analysis is not proposed as the best method, but as a possible method.

a. <u>Factor of Safety for a DSV Pressure Envelope Subject to External Pressure:</u>

In the United States of America, it has been traditional to apply a constant factor of safety on the maximum design operating depth. Prior to World War II, the factor of safety for submarines was greater than 1.5 while the ships were new. After several years of service and some corrosion of the hull structure, the factor of safety was reduced. Many ships required structural repairs in order to restore an adequate factor of safety. During World War II, the maximum design operating depth was increased to 400 feet with a collapse depth of 600 feet. The factor of safety was then 1.5 at the yield stress of high tensile steel. Since World War II many combatant and non-combatant submarines have been built. The factor of safety for most of these vehicles has been 1.5 of the yield stress of the hull material. Recently, there has been a tendency, particularly when designing for very deep diving submarines to discount all variables involved in the factor of safety consideration except the overshoot problem. It has been assumed that the margin provided for by the recovery of an accidental overshoot from design to collapse depth would be adequate to cover the other uncertainties.[1]

In the preliminary design of the Navy's Deep Submergence Rescue Vehicle, the certification procedures outlined in reference four were used. This certification procedure utilized two safety criteria to ensure a minimum factor of safety of 1.5. The first criteria required that, if the shell was buckling limited, it would buckle at 1.5 times the vehicle's maximum operating depth.[2] The second criteria required that the total stresses in the shell (i.e., membrane plus bending) would not exceed 75% of the compressive yield stress at maximum operating depth.[3]

In approaching the certification procedure for the DSS, the Navy Deep Submergence Certification Manual[4] shows itself to be insensitive to depth in regards to the factor of safety of 1.5 with 75% total compressive yield stress at maximum operation depth of the vehicle. The uncertainties regarding strength decrease for increasing collapse depth, because the magnitude of a depth excursion beyond operating depth is not likely to increase with increasing hull depth capability. A certain

measure of safety is assumed by the practical limitation that only 2% of the ocean bottom lies at depth greater than 20,000 ft. These two factors actually give the ADDS-20 a greater margin of safety than shallower diving submersibles, resulting in an inefficient certification criteria.

NSRDC has proposed a certification procedure for the factor of safety problem, covering the depth overshoot problem and the reduction in the uncertainties in predicting collapse depth.

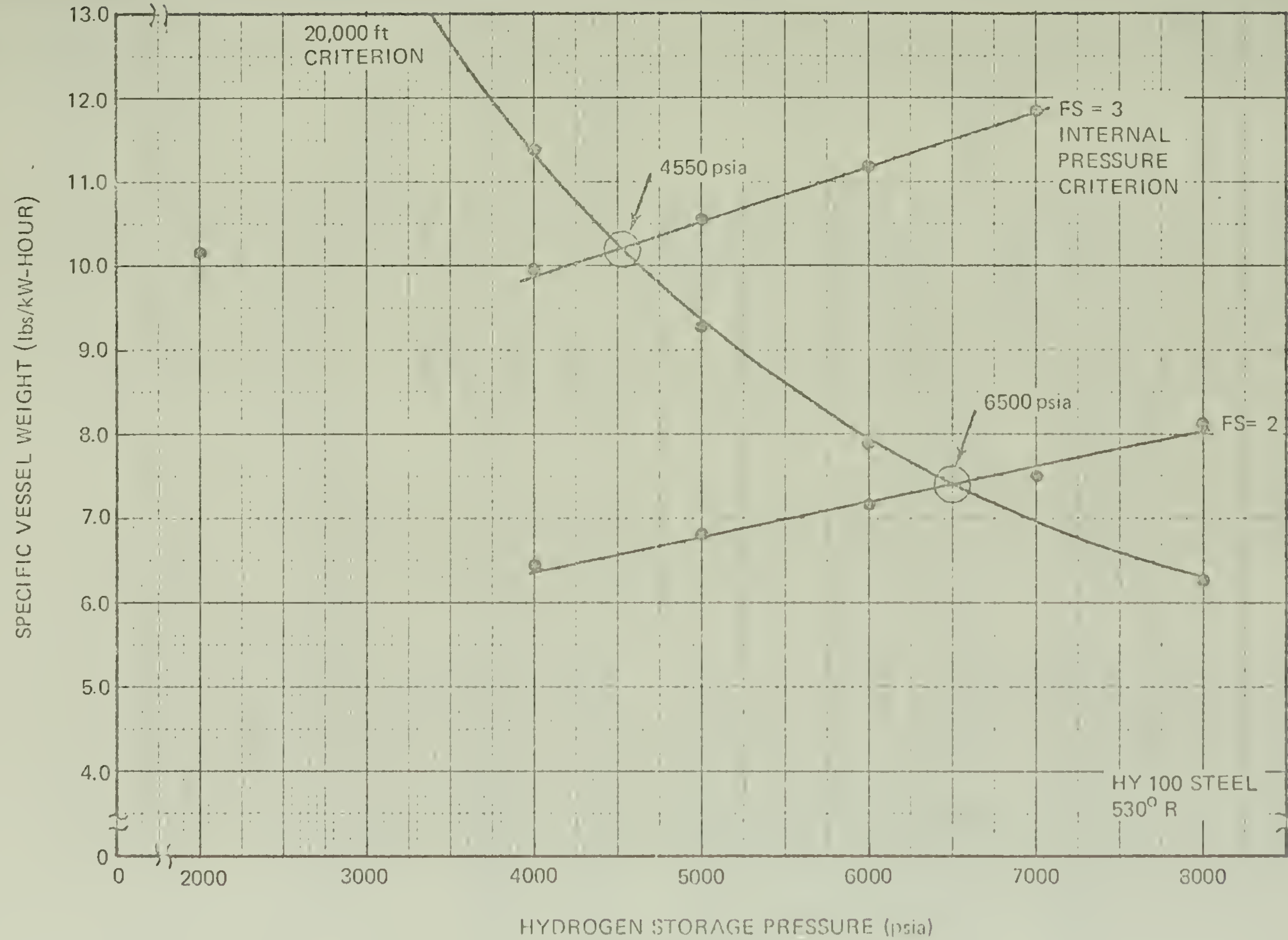
In regards to the overshoot problem, NSRDC proposed a depth overshoot of 2,000 feet, which is greater than many shallow-diving submersibles. Therefore, the factor of safety from an overshoot consideration has been reduced to 1.1 on the maximum operating depth of 20,000 feet.

When considering the uncertainties in predicting collaspe depth, it is essential to provide a wide margin of safety. NSRDC recommended that, the lower bound estimate be set at a depth of 25,000 feet, which is a factor of safety of 1.25. This lower bound was arrived at after considering the dispersion in yield strength, creep, geometrical tolerances, and scatter in supporting structural collapse data.

In order to prevent permanent deformation during the acceptance tests, a recommended maximum stress, bending plus membrane, be held to 75% of compressive yield stress at the maximum operating depth of 20,000 feet.[3]

b. Factor of Safety for a DSV Pressure Envelope Subject to Internal Pressure:

A factor of safety of 3.0 is required[4] for a DSV pressure vessel of subjected to internal pressure. This criterion applies while the vehicle is on the surface, since this imposes the largest internal oriented stresses. This pressure vessel must also satisfy the safety criterion in the previous paragraph. In the DSV considered, the design is weight limited. Figure G-1 shows the optimum point from a weight standpoint. Figure G-1 represents a trade-off for the hydrogen fuel cell tank. It was drawn in order to establish the specific vessel weight and the optimum storage pressure for a minimum weight design.



13.0
12.0
11.0
10.0
9.0
8.0
7.0
6.0
5.0
4.0
0
SPECIFIC VESSEL WEIGHT (lbs/kW-HOUR)
0
2000
3000
4000
5000
6000
7000
8000
HYDROGEN STORAGE PRESSURE (psia)
20,000 ft
CRITERION
4550 psia
6500 psia
FS = 3
INTERNAL
PRESSURE
CRITERION
FS= 2
HY 100 STEEL
530° R

Figure G-1

3. Structural Analysis:

The feasibility of various deep submergence pressure envelopes has been investigated by several authors. The following list is provided in order that future deep submergence design literature searches be kept to a minimum.

* Wenk, E., Jr., "Feasibility Studies of Pressure Hulls for Deeply Submerged Submarines," National Academy of Science - National Research Council, AD No. 305685-L, August 1958.

* Wenk, E., Jr., "Feasibility of Pressure Hulls for Ultra-deep Running Submarines," Paper No. 61-WA-187, winter annual meeting of ASME, Underwater Technology Professional Group, New York, Nov. 26, 1961.

* Krenzke, M., Hom, K., and Profitt, J., "Potential Hull Structures for Rescue and Search Vehicles of the Deep Submergence Systems Project," David Taylor Model Basin Report 1985 (Mar. 1965).

* Krenzke, M., "Structural Aspects of Hydrospace Vehicles, "Naval Engineers Journal, Aug. 1965.

* Trimble, L., Nickell, E., "Unique Design Considerations for the Underwater Environment," ASME paper 64-MD-39, March 1, 1965.

* Rockwell, R. D., "A Computer Program for Conducting Trade-off Studies for Deep Submergence Vehicles of Various Materials and Shapes (u)," Naval Ship Research and Development Center Report C-3026(July 1965).

The Structural Analysis of each configuration will not be presented, but rather the most complete literature on each configuration. The literature listing will be followed by a presentation of the latest spherical shell analysis, since the DSV example uses a spherical configuration.

a. Monocoque Sphere

*Timoshenko, S., "Theory of Elastic Stability," McGraw-Hill Book Co., Inc., New York (1936).

*Krenzke, M. and Kiernan, T., "The Effect of Initial Imperfections on the Collapse Strength of Deep Spherical Shells," David Taylor Model Basin Report, 1757 (Feb. 1965).

*Kiernan, T., "Predictions of the Collapse Strength of Three HY100 Steel Spherical Hulls Fabricated for the Oceanographic Research Vehicle ALVIN," David Taylor Model Basin Report 1792(March 1964).

*Krenzke, M. and Kiernan, T., "Tests of Stiffened and Unstiffened Machined Spherical Shells under External Hydrostatic Pressure," David Taylor Model Basin Report 1741 (Aug. 1963)

*Dadley, A. "Tests of Machined High Strength Steel Spherical Shells Subjected to External Hydrostatic Pressure." David Taylor Model Basin Report 1854 (Aug. 1964).

*Krenzke, M., Hom, K., and Profitt, J., "Potential Hull Structures for Rescue and Search Vehicles of the Deep Submergence Systems Project," David Taylor Model Basin Report 1985 (March 1965).

*Krenzke, M. and Schwartz, F., "Recommended Approach to the Structural Certification of the Proposed Trieste Replacement Vehicle," NSRDC LTR Ser 72-172-176, (Oct. 1972).

*Schwartz, F., "Preliminary Design and Cost Estimates of Pressure Hulls for the Proposed Trieste Replacement Vehicle," NSRDC LTR. Ser 73-172-11, (Feb. 1973).

b. Stiffened Spheres

*Krenzke, M. and Kiernan, T., "Tests of Stiffened and Unstiffened Machined Spherical Shells under External Hydrostatic Pressure," David Taylor Model Basin Report 1741, (Aug. 1963).

*K. von Klöppel and O. Jungbluth, "Beitroy zum Durchschlagproblem dunnevandiger kugelschalem," Der Stahlbau 22 Jahrgang, Berlin, June 1953, Heft 6.

Kloppel and Junglbuth have shown that stiffened spherical caps required a $^t/R$ about 3/4 that of the classical monocoque solution. Their investigation was not done for minimum weight structures and improvement might be possible by using more efficient stiffened patterns.

*Krenzke, M., "Structural Aspects of Hydrosphere Vehicles," Naval Engineers Journal, (Aug. 1965).

Briefly summarizes the most likely configuration for deep submergence pressure hulls. The stiffened spherical shell offers some strength advantages over monocoque spheres in the unstable region, but is not considered because of unreliable design criteria.

c. Cylindrical Shells

*Krenzke, M., Hom, K., and Profitt, J., "Potential Hull Structures for Rescue and Search Vehicles of the Deep Submergence Systems Project," David Taylor Model Basin Report, 1985 (March 1965).

"Appendix A of this report contains a bibligoraphic summary of deep submergence cylindrical hull design up through 1965."

*Krenzke, M., "Structural Aspects of Hydrosphere Vehicles." Naval Engineers Journal, (Aug. 1965).

"A brief summary of NSRDC Report 1985."

*Trimble, L and Nickell, E., "Unique Design Considerations for the Underwater Environment," ASME paper 64-MD-39, March 1, 1965.

"Mr. Trimble has arranged much published information into a very usable format for configuration trade-offs. In the process of setting up his trade-offs, he has presented a literature review of various stiffened analyses. References

The efficiency factors and associated references are listed below:

Type of Stiffening	"ϵ" Efficiency Factor	Reference
Angle.	0.250	7
Waffle	0.300	4
Truss-cone sandwich . . .	0.423	5
Rectangular	0.479	4
Vee torrugation	0.555	5
Channel or zee	0.590	7
Tee.	0.750	7
"┌┐".	0.707	10
Trapezoidal corrugation .	0.860	7

*Nickell, E. and Crawford, R., "Structural Shell Optimization Studies, Final Report," vol. 2, "Optimization of Cylindrical Shells Subjected to Uniform External Hydrostatic Pressure," LMSC Report 3-42-61-2, June 30, 1961 (AD 267624).

*Nickell, E., and Crawford, R., "Optimum Ring-Stiffened Cylinders Subjected to Hydrostatic Pressure, "LMSC 6-90-62-57, July 1962, (Presented to SAE Material Aerospace Engineers and Manufacturing Meeting, Los Angeles Calif., Oct. 8-12, 1962).

*Stuhlman, C., "Optimization Analyses for Long Cylindrical Shells Stiffened with Rectangular Rings and Subjected to an Uniform External Pressure," unpublished Lockheed Report.

*Nickell, E. and Burns, A., "Optimization of the Polaris Interstage," LMSC Report N. 8-40-63-10, June 28, 1963.

*Crawford, R. and Stuhlman, C., "Minimum Weight Analysis for Truss-Cone Sandwich Cylindrical Shells under Axial Compression, Torsion, and Radial Pressure," LMSC Report 2-47-61-2, April 1961.

*Nickell, E., "Design Procedure for Deep Launch Capsules," LMSC Report 6-62-63-2, December 31, 1963.

*Gerard, G., "Minimum Weight Design of Ring-Stiffened Cylinders Under External Pressure," Journal of Ship Research, vol. 5, No. 3, Sept. 1961, pp. 44-49.

*Pappas, M. and Allentuch, A., "Automated Optimal Design of Frame Reinforced, Submersible, Circular, Cylindrical Shells," Journal of Ship Research, vol. 17, No. 4, Dec. 1973, pp. 208-216.

*Comments on "Automated Optimal Design of Frame-Reinforced, Submersible, Circular, Cylindrical Shells," by M. Pappas and A. Allen Tuch: H. Berka, Vol. 18, No. 2, June 1974, p. 139.

The last two papers were not in Trimble's paper, but are useful for automated optimal cylindrical design. Reference 12 provides a helpful analysis which can be done with a slide rule in a few minutes.

d. Monocoque Prolate Spheroidal Shells

*Krenzke, M., Hom, Kl, and Profitt, J., "Potential Hull Structures for Rescue and Search Vehicles of the Deep Submergence Systems Project, "David Taylor Model Basin Report 1985 (Mar. 1965).

In Appendix A of the above paper Krenzke reviews the state of the art in prolate spheroidal design.

*Krenzke, M., "Structural Aspects of Hydrospace Vehicles," Naval Engineers Journal, (Aug. 1965).

Repeats information in less detail, but overs the same ground.

The references mentioned in the papers above are listed below for reference:

* Flugge, W., "Stresses in Shells, "Springer-Verlay OHG, Berlin/Gottengen/Heidelberg (1960).

* Mushtari, Kh. and Galinov, "Non-Linear Theory of Thin Elastic Shells," Talknigoigoat Kuzare (1957)

or

Mushtari, Kh. and Galinov, "Non-Linear Theory of Thin Elastic Shells," published for the N. S. F. and NASA by the Isreael Program for Scientific Translations, NASA-TT-F62, 1962.

* Hyman, B. I., "Elastic Instability of Prolate Spheroidal Shells Under Uniform External Pressure," DTMB Report 2105.

*Healey, J. J., "Parametric Study of Unstiffened and Stiffened Prolate Spheroidal Shells under External Hydrostatic Pressure," David Taylor Model Basin Report 2018 (Aug. 1965).

e. Stiffened Prolate Spheroidal Shells:

*Krenzke, M., Hom, K. and Profitt, J., "Potential Hull Structures for Rescue and Search Vehicles of the Deep Submergence Systems Project," David Taylor Model Basin Report, 1985 (March 1965).

Report suggests that the elastic stresses in a ring stiffened prolate spheroidal shell may be approximated by the Salerno and Pulos Analysis below.

*Salerno, V. L, and Pulos, J. G., "Stress Distribution in a Circular Cylindrical Shell under Hydrostatic Pressure Supported by Equally Spaced Circular Ring Frames," Polytechnic Institute of Brooklyn Aeronautical Laboratory Report 171-A (June 1951).

*Healey, J. J., "Parametric Study of Unstiffened and Stiffened Prolate Spheroidal Shells under External Hydrostatic Pressure," David Taylor Model Basin Report 2018 (Aug. 1965).

f. Monocoque Oblate Spheroidal and Torodial Shells

* Nickell, E., Structural Shell Optimization Studies-Final Report, vol. 4, "Experimental Buckling Tests of Magnesium Monocoque Ellispoidal Shells Subjected to External Hydrostatic Pressure," LMSC Report 3-42-61-2, June 30, 1961.

* Lovell-Jameson, "Thrust Vector Control from Tanks Tested Under External Pressure," LMSC Reports 37639, 37640, 37641, June 1963.

. * Maehnig, "Über Stabilitats Probleme von Torusfermigen Schalen," Z. Wissda Hachsch f. Verkehrsw, Dresden 4, 1956, H213.

4. Governing Criteria Analysis:

On e output of the structural analysis is the criterion or criteria which govern one pressure envelope for a given material. This is an essential input to the configuration trade-off, since the comparison of various configurations is dependent upon the governing criterion. Trimbell has composed a table which is very useful in the early stages of the structural design because it presents a means of estimating the intersection point of the instability and stress criteria for a given material and configuration. Trimble's results are displayed in G-2. The optimum design point, or the point of simultaneous fulfillment of both criteria can be determined by equating the stability and strength terms and solving for the desired parameter. For example, the strength term for the monocouque sphere is the hoop-stress or membrane stress analysis where the stability term is an approximation of the buckling data. To find the strength required for a steel sphere, to sustain 75% compressive yield and 1.25 operating depth for buckling the following is presented:

Monocoque Sphere

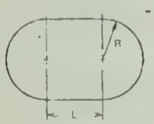
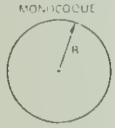
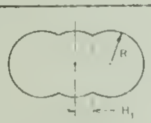
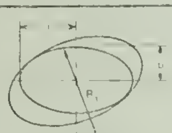
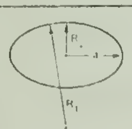
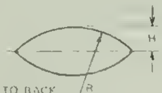
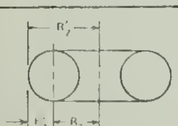
1. Stability term

$$\frac{t}{R} = 1.217\left(\frac{d^1}{E}\right)^{\frac{1}{2}}$$

Elastic buckling does not include out-of-roundness

Table G-1

Hull Weight/Displacement Estimation

CONFIGURATION	$\frac{\Delta R}{V}$ TERM	$\frac{t}{R}$ STABILITY TERM	$\frac{t}{R}$ STRENGTH TERM
MONOCOQUE 	$2\left[\frac{2+\frac{L}{R}}{\frac{4}{3}+\frac{L}{R}}\right]$	$\left[\frac{0.747\left(\frac{L}{R}\right)^{\frac{7}{15}}\cdot 2.434\,\frac{d}{t}^{\frac{1}{100}}}{\frac{L}{R}+2}\right]\left(\frac{d}{E}\right)^{\frac{2}{5}}$	$0.445\left(\frac{\frac{L}{R}+1}{\frac{L}{R}+2}\right)\frac{d}{\sigma_{STR}}$
UNSTIFFENED CYLINDER WITH HEMI ENDS	$2\left[\frac{2+\frac{L}{R}}{\frac{4}{3}+\frac{L}{R}}\right]$	$\left[\frac{\frac{0.698\left(\frac{L}{R}\right)^{\frac{17}{13}}}{t\ 13}+0.608\left(\frac{E}{d}\right)^{\frac{1}{26}}}{\frac{L}{R}+2}\right]\left(\frac{d}{E}\right)^{\frac{7}{13}}$	$0.445\left[\frac{\frac{L}{R}+1+4.5\left(\frac{\sigma_{STR}}{d}\right)\frac{A_S}{d_S\,R}}{\frac{L}{R}\ 2}\right]\frac{d}{\sigma_{STR}}$
MONOCOQUE  SPHERE	3	$1.217\left(\frac{d}{E}\right)^{\frac{1}{2}}$	$0.2225\left(\frac{d}{\sigma_{STR}}\right)\quad \sigma = \frac{PR}{2t}$
 NESTED SPHERE	$2\left[\frac{1-\left(\frac{m-1}{m}\right)\frac{H_1}{R}}{\frac{2}{3}-\left(\frac{m-1}{m}\right)\left(\frac{H_1}{R}\right)^2\left(1-\frac{H_1}{3R}\right)}\right]$	SAME AS SPHERE	SAME AS SPHERE
 OBLATE SPHEROID	$2\left[\frac{\left(\frac{a}{b}\right)^2+\frac{\ln\left(\frac{1+e}{1-e}\right)}{2a}}{\frac{4}{3}}\right]$	SAME AS SPHERE	SAME AS SPHERE
 PROLATE SPHEROID	$2\left[\frac{\frac{R_2}{a}+\frac{1}{e\sin e}}{\frac{4}{3}}\right]$	$0.667\left(\frac{2-\frac{R_2}{R_1}}{\frac{R_2}{R_1}}\right)^{\frac{1}{2}}\left(\frac{d}{E}\right)^{\frac{1}{2}}$	SAME AS SPHERE
 (BACK TO BACK SPHERICAL CAPS) LENTOID	$2\left[\frac{1}{\frac{H}{R}\left(1-\frac{H}{3R}\right)}\right]$	SAME AS SPHERE	SAME AS SPHERE
 TOROID	2	$1.505\left(\frac{R_1}{R_2}\right)^{\frac{5}{8}}\left(\frac{d}{E}\right)^{\frac{1}{2}}$	$0.223\,\frac{\left(2\frac{R_T}{R_2}-1\right)}{\left(\frac{R_T}{R_2}-1\right)}\,\frac{d}{\sigma_{STR}}$

$$\frac{W}{D} = \left(\frac{\Delta R}{V}\right)\left(\frac{t}{R}\right)\left(\frac{W_P}{W_S}+1\right)\left(\frac{\rho_h}{\rho_w}\right)$$

Nomenclature for Table G-1

A = hull surface area

A_s = stiffener area

E = Young's modulus of elasticity

H = lentoid half height

H_1 = intersecting spherical segment height

L = cylinder length

R = cylinder and sphere radius

R_1 = oblate spheroid maximum radius

R'_1 = toroid maximum radius

R_2 = toroid shell radius

R'_2 = toroid shell radius

R_3 = prolate spheroid maximum radius

R_T = toroid mean radius

V = hull volume

W_p = structural weight penalty

W_s = shell weight including stability stiffeners

W_r = ring weight

W_{ir} = intersecting ring weight for intersecting spheres

W_{jr} = joining ring weight for lentoidal hull

a = ellipse major half axis length

b = ellipse minor half axis length

d = collapse depth

d_s = stiffener spacing

m = number of intersecting spheres

p = hydrostatic pressure

t = monocoque hull thickness

$\overline{t}$ = weight equivalent hull thickness

σ_{str} = material compressive strength

$(\ )_{opt}$ = optimum values

P_h = hull material density

P_w = sea-water density

ϵ = efficiency factor
see page G-10

$e = \sqrt{1 - \left(\frac{R_2}{a}\right)^2}$ for prolate spheroid

$e = \sqrt{1 - \left(\frac{b}{a}\right)^2}$ for oblate spheroid

e = eccentricity

ρ_h = material density

ρ_w = water density

d^1 = buckling depth in feet

2. Strength term

$$\frac{t}{R} = .223 \left(\frac{d}{\sigma STR} \right)$$

d = maximum operating depth in feet

3. Equating terms

$$.223 \left(\frac{d}{\sigma STR} \right) = 1.217 \left(\frac{d^1}{E} \right)^{\frac{1}{2}}$$

$$\sigma STR = \frac{.223}{1.217} \frac{d}{\sqrt{d^1}} \sqrt{E}$$

$$= .18$$

σ STR = 123,736 psi for the DSV 20,000 ft. operating depth and 25,000 ft. buckling depth.

Note: σ STR is just membrane stress. Therefore, to backfit to the yield stress of the material.

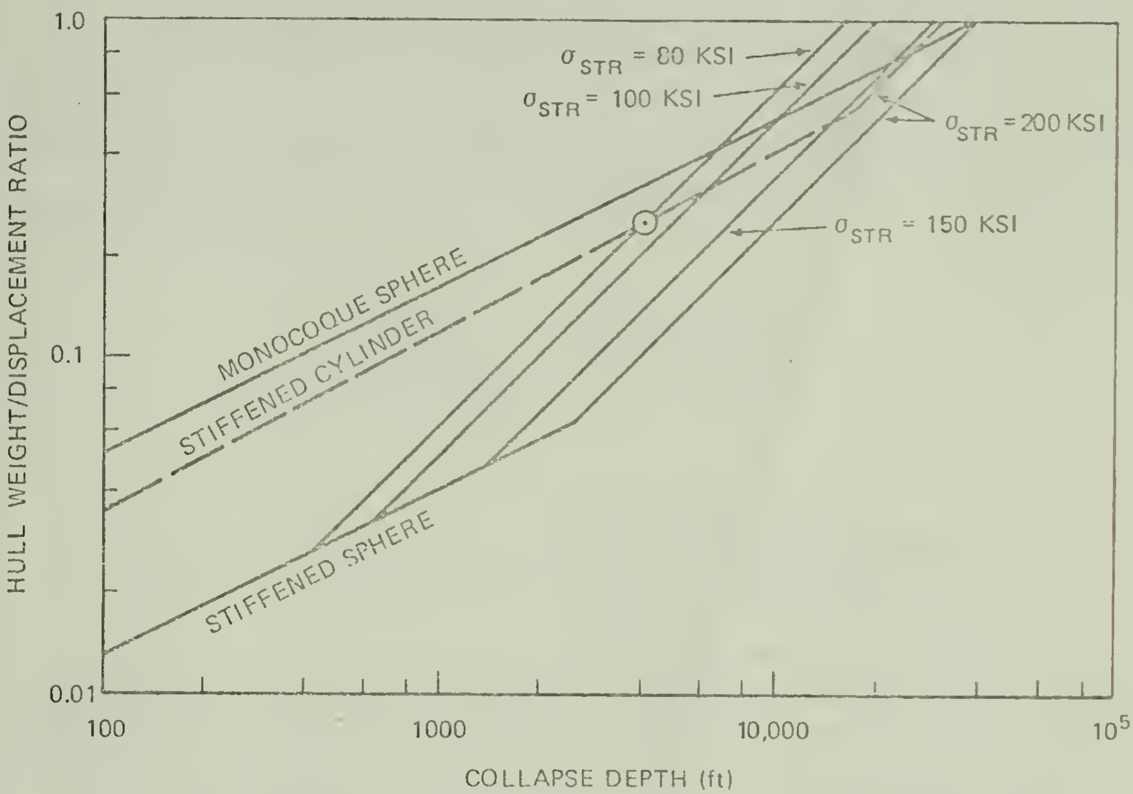
σ STR = .65 σy (from reference 4)

$\therefore$ σy = 190 Ksi

Monocoque spheres made of materials with yield strengths greater than 190 Ksi would be buckling-limited whereas those less than 190 Ksi would be stress limited.

Trimble's table is very helpful to the designer in getting a feel for the failure modes for various materials and configurations. These results should be checked throughout the design process as the governing criteria will change with various criteria changes.

Figure G-2 shows how Trimble's analysis can be helpful in selecting a configuration.



Comparison of Steel Spherical and Stiffened Cylindrical Hulls
Figure G-2

5. Actual DSV Analysis of Spherical Configurations:

From the structural analysis of the DSV, it has been determined that a sphere system (single sphere or bi-sphere) would be used. For this reason, the following section on 1974 state-of-the-art preliminary sphere design is presented.

a. Monocoque Sphere Stability Analysis: The classical small-deflection theory for the elastic buckling of a complete sphere was first developed by Zoelly in 1915[5]. Zoelly's analysis assumes that buckling will occur at the pressure which permits an equilibrium shape minutely removed from a perfect shape. The expression for the classical buckling pressure P_1 may be given by:

$$P_1 = \frac{2E\left(\frac{h}{R}\right)^2}{\sqrt{3(1-\nu^2)}} = 1.210\, E\left(\frac{h}{R}\right)^2$$

E = Young's modulus

h = average shell thickness

R = radius to mid surface of shell

ν = poisson ratio = .3 for steel

Early tests did not support Zoelly's expression, but as of 1965 NSRDC has shown that the early test discrepencies were due to the inattention to initial imperfections in the shells. From the 1965 experiments, an empirical equation for near-perfect spheres was suggested which predicts collapse depth at 70% the classical collapse pressure. The empirical equation for the elastic buckling pressure P_3 near-perfect spherical shells may be expressed as:

$$P_3 = \frac{1.4E\left(\frac{h}{R_o}\right)^2}{\sqrt{3(1-\nu^2)}} = .84E\left(\frac{h}{R_o}\right)^2$$

R_0 = average outer radius

ν = .3 for steel

The inelastic buckling strength of near-perfect spheres was first investigated by Bijlaard[6] and may be expressed as:

$$P_B = \frac{2\sqrt{E_s E_T}}{\sqrt{3(1-\nu_p^2)}}\left(\frac{h}{R_o}\right)^2 = .84\sqrt{E_s E_T}\left(\frac{h}{R_o}\right)^2$$

E_s = secant modulus

E_t = Tangent modulus

ν_p = Poisson's ratio in the plastic range = .5 for steel

ν = .3 for steel in the elastic range

Again NSRDC conducted tests for inelastic buckling, resulting in the following empirical formula

$$P_E = 1.4\sqrt{\frac{E_s E_T}{3(1-\nu^2)}}\left(\frac{h}{R_o}\right)^2 = .84\sqrt{E_s E_T}\left(\frac{h}{R_o}\right)^2$$

ν = .3 for steel

The membrane stress in the elastic buckling case can be found using small deflection thin-shell theory where $\sigma = \frac{PR}{2h}$. In the inelastic case the average stress which satisfies equilibrium conditions for all thicknesses can be found by:

$$\sigma_{Avg} = \frac{P R_o^2}{2Rh}$$

p = hydrostatic pressure

As a result of the 1965 experiments, it was noticed that localized imperfections played a large role, leading to a set of equations which account for these localized imperfections in terms of localized geometry.[5]

These expressions are as follows:[7]

$$P_3'' = .84\,K\,E\left(\frac{h_a}{R_{10}}\right)^2 \quad \text{for } \nu = .3$$

$$P_E'' = .84\,K\sqrt{E_s E_T}\left(\frac{h_a}{R_{10}}\right)^2 \quad \text{for } \nu = .3$$

$$\sigma'_{AVG} = \frac{P(R_{10})^2}{2\,h_a R_1}$$

The primes indicate local geometrical parameters.

where E = Young's Modulus

h_a = Average thickness over a critical arc length

K = empirical coefficient based on test results

R_{10} = local radius of the outside surface of the shell over a critical arc length

P = hydrostatic pressure

R_1 = Local radius to mid-surface of shell over a critical arc length

ν = Poisson's ratio

The use and derivation of the buckling equation is presented in references 3 and 6. The two most promising materials for deep submergence pressure envelopes are HY 180 (10 Ni - 8 Co - 2 Cr -1 Mo) and Ti 110 (6 Al - 2Cb - 1Ta- 0.8Mo).

The information below is presented to extend the information listed in reference 1 and reference 3.

The Material Properties:

Hy-180 (10 Ni - 8Co - 2Cr- 1 Mo)

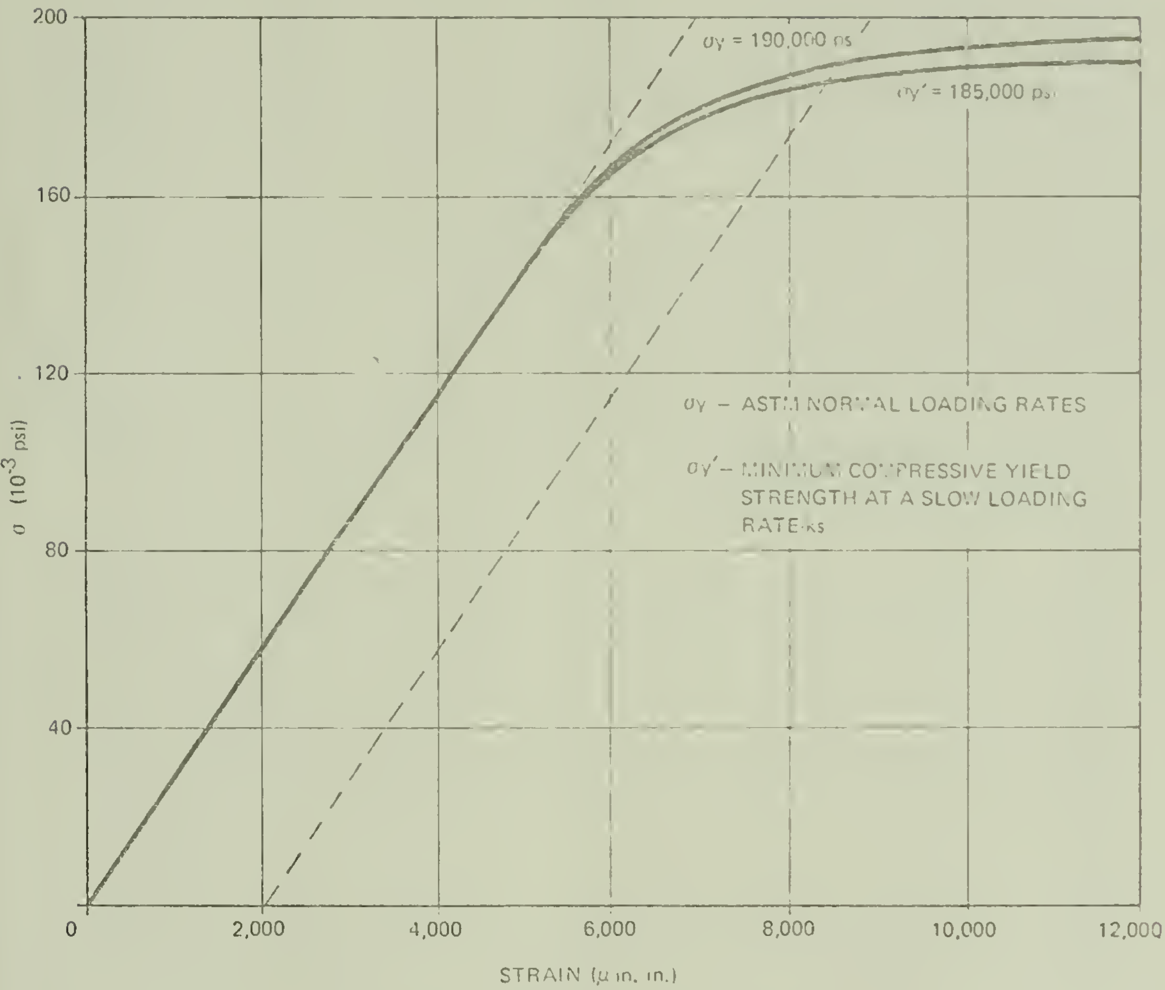
σ_y = 185,000 psi (For slow loading rate 250 ft./min.)

σ_{Pl} = 143,500 psi = proportional limit

ν = .3

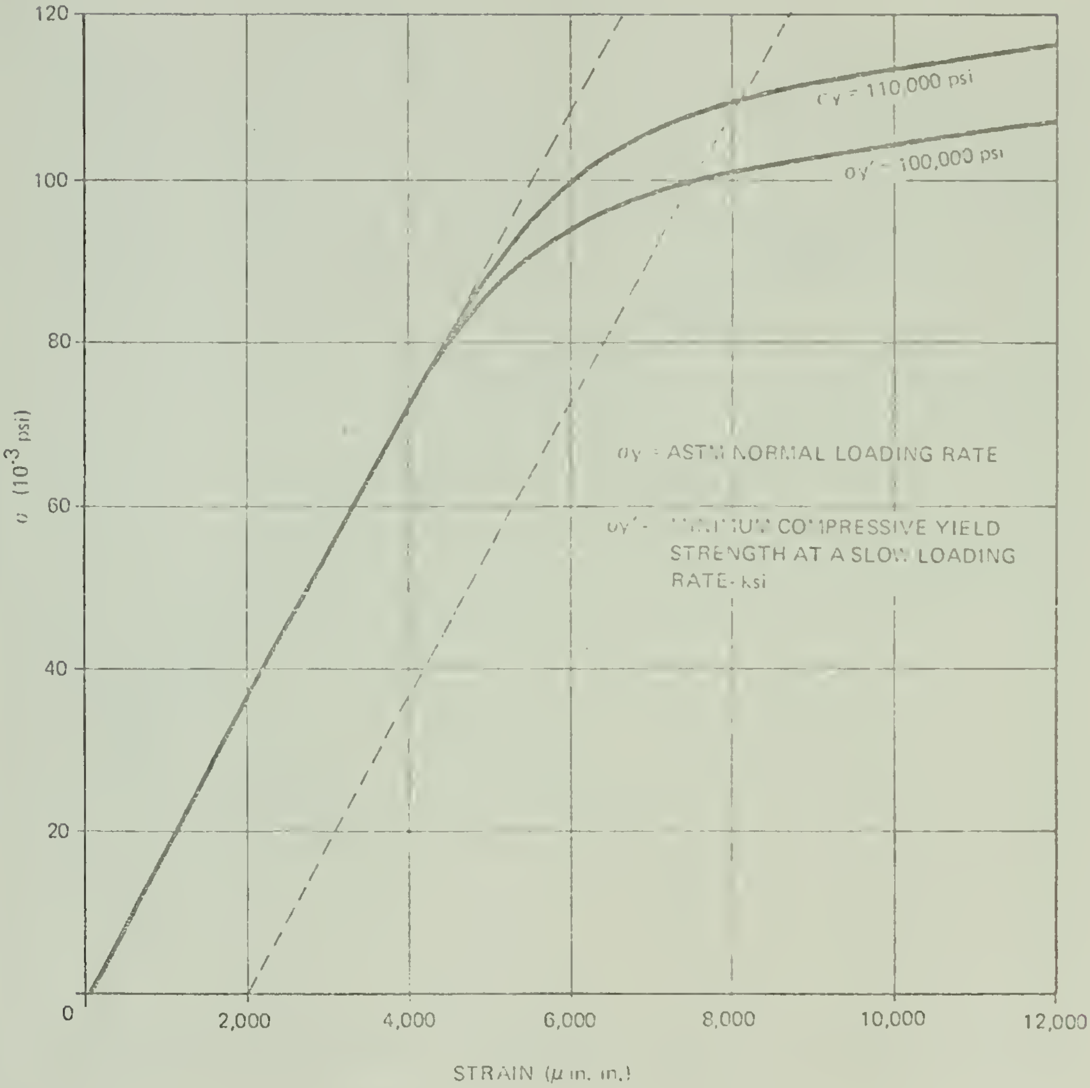
E = 28,500,000 psi

ρ_s = 489 lbf/ft^3



COMPRESSIVE STRESS-STRAIN CURVES
FOR HY180 (10Ni-8Co-2Cr-1Mo)

Figure G-3



COMPRESSIVE STRESS-STRAIN CURVES
FOR Ti 110 (6Al-2Cb-1Ta-.8Mo)

Figure G-4

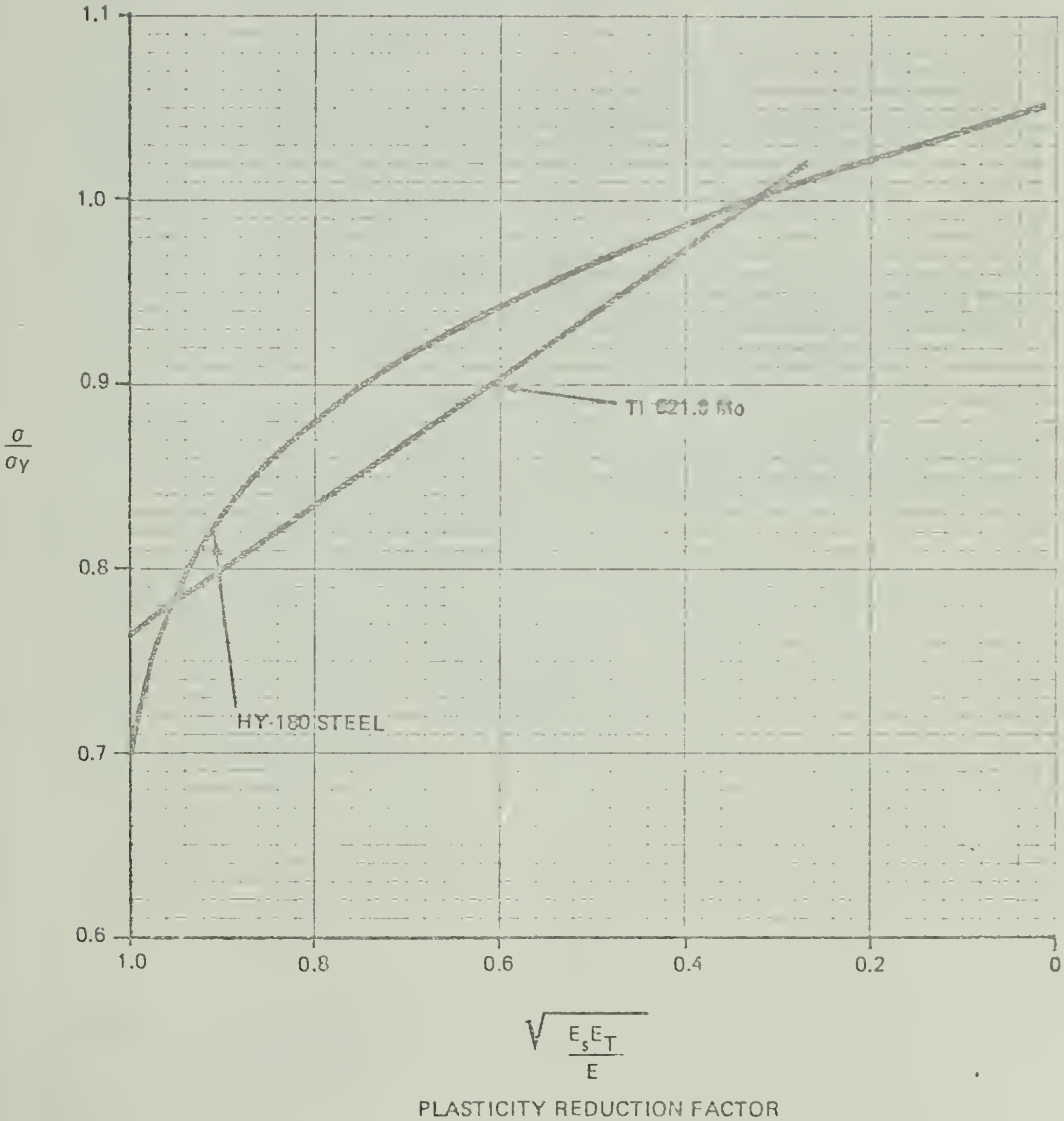
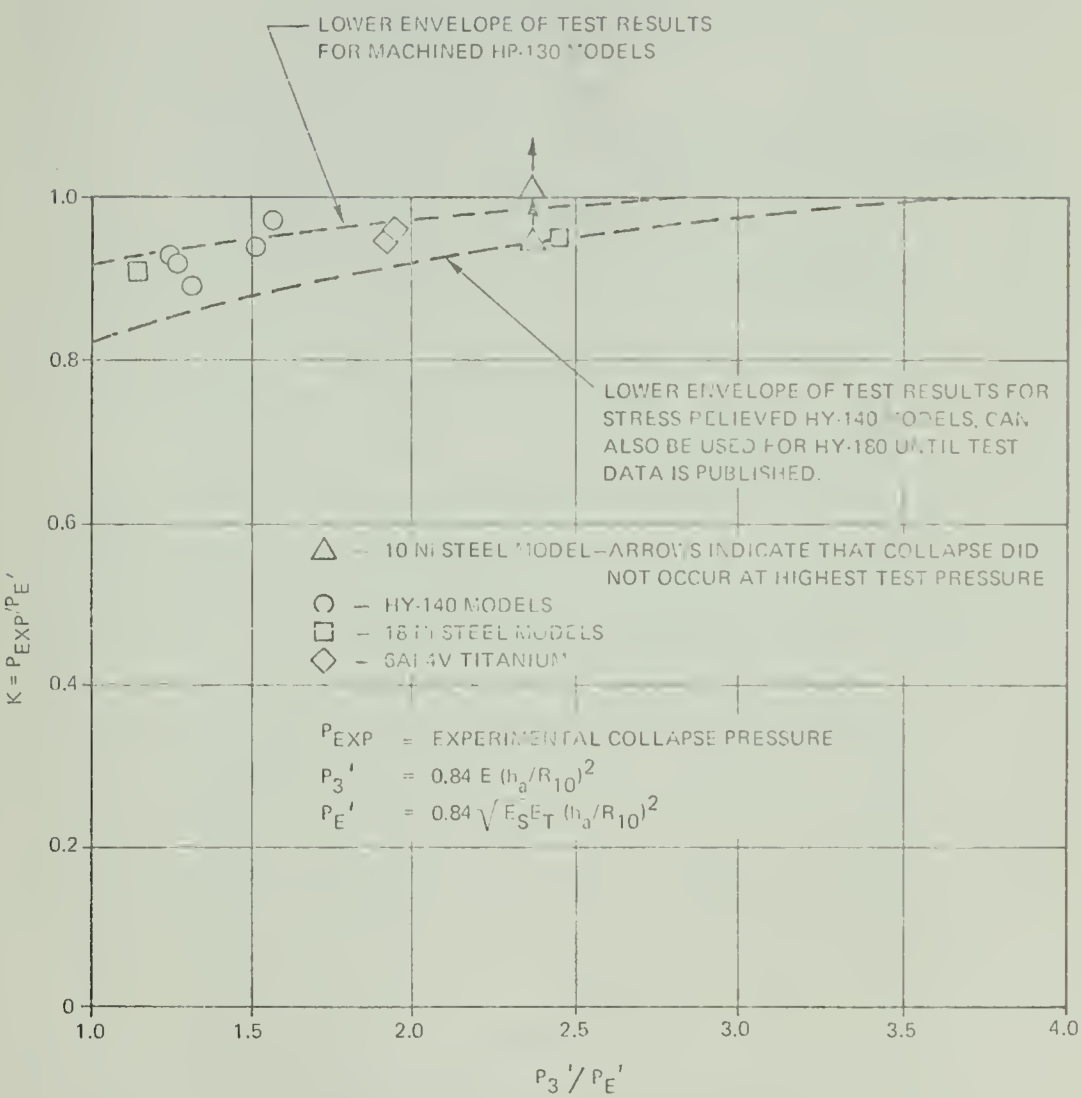
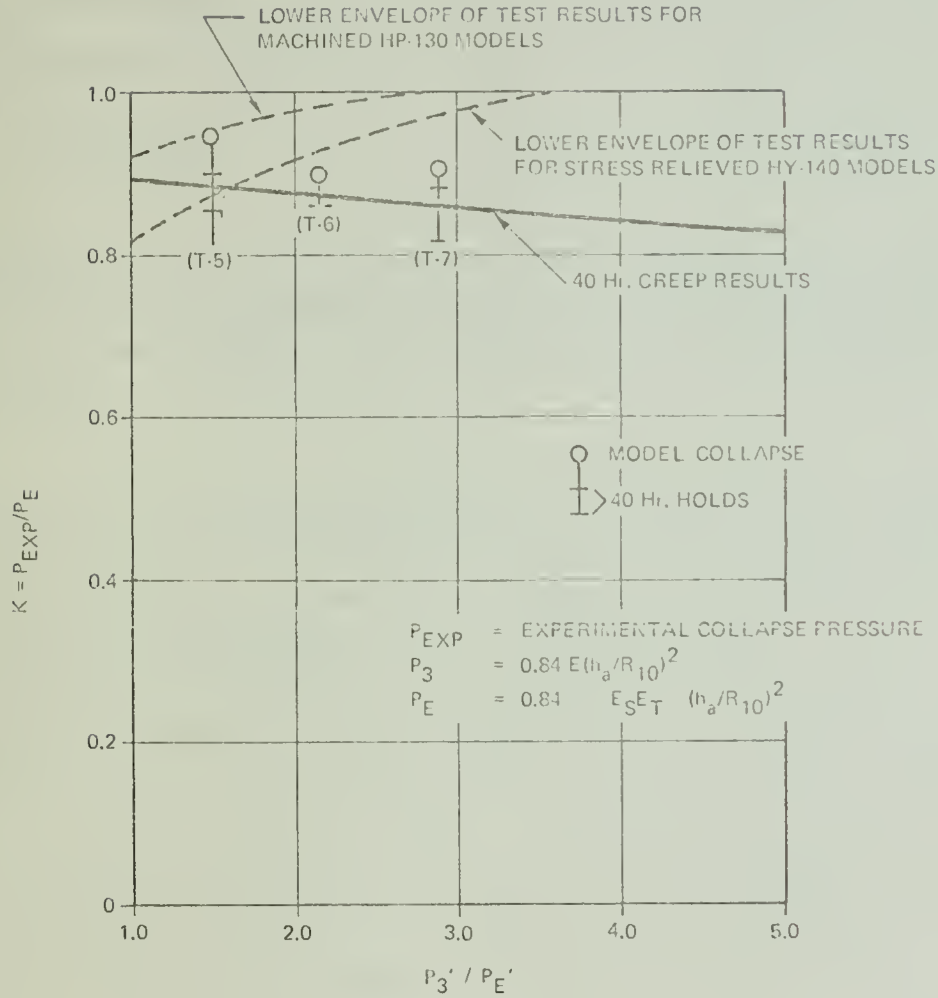

1.1
1.0
0.9
0.8
0.7
0.6
$\frac{\sigma}{\sigma_Y}$
TI 621.8 Mo
HY-180 STEEL
1.0
0.8
0.6
0.4
0.2
0
$\sqrt{\frac{E_s E_T}{E}}$
PLASTICITY REDUCTION FACTOR

Figure G-5



NOTE: REDUCE K VALUE BY 5% TO ACCOUNT
FOR CREEP IN STEEL

Figure G-6



TITANIUM CREEP

NOTES:

1. COLLAPSE PRESSURES
   T-5–COLLAPSED AFTER 1 hr. @ MAX. PRESSURE
   T-6–COLLAPSED AFTER 8 hr. @ MAX. PRESSURE
   T-7–COLLAPSED AFTER 1 hr. @ MAX. PRESSURE
2. FORTY HOUR CREEP CURVE CONSTRUCTED TAKING INTO ACCOUNT THE DATA OF NOTE 1.

Figure G-7

Ti 110 (6Al- 2Cb - 1Ta - 0.8 Mo)

σ_y = 100,000 psi (slow loading rate 250 ft/min for thicknesses $<$ 2 inches)

σ_y = 95,000 psi (slow loading rate 250 ft/min for $2 < h_a < 4$ inches)

σ_{Pl} = 78,000 psi

E = 18,000,000 psi

ν = .3

ρ_T = 278 lbf/ft^3

Figures G-3 and G-4 display the stress-strain curves for these two materials. The governing curve for practical diving pruposes will be the, slow loading rate, which is equivalent to a descent rate of 250 ft/min.

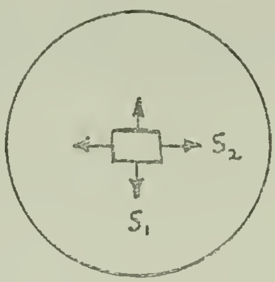
In order to facilitate parametric studies of the inelastic buckling equation, a plasticity reduction factor was introduced. The PRF's for HY 180 and Ti 110 are presented in figure G-5. These were calculated from the respective stress-strain curves.

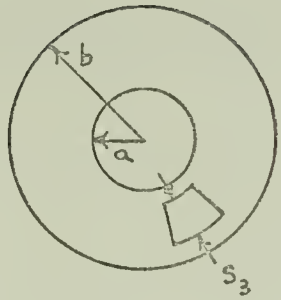
As mentioned previously, NSRDC introduced a "K factor," which correlates the theoretical formulations with the experimental data. Reference 3 contains K factors for STS and HY 80, Hy 140, 10 Ni, 18N; 6Al - 4 Va and 62 -0.8 Mo models. As can be seen in Figure G-b, very little experimental work has been done on Hy 180. NSRDC is at present conducting some model tests on Hy 180, but the information is not yet available. Unitl the information is available, the Hy 140 stress-relieved data can be used for Hy 180 design. A five precent decrease in the "K factor" should be allowed for creep effects in Hy 180 steel.

Figure G-7 presents the titanium creep results and the 40 hr creep line should be used for titanium design.

b . Monococque Sphere Stress Analysis: From the test results mentioned above, NSRDC has recommended, that Lame's thick shell equations be used in stress estimation for a stress-limited shell[3]. Listed below are Lame's thick shell equations for internal and external pressures. A complete derivation of these formulas can be found in reference 8.

Lame Thick Shell formulae:





Internal Pressure:

$$S_1 = S_2 = p\,\frac{b^3(a^3 + 2r^3)}{2r^3(b^3 - a^3)}$$

$$\text{Max } S_1 = \text{Max } S_2 = -p\,\frac{3\,b^3}{2(b^3 - a^3)}$$ at inner surface

$$S_3 = p\,\frac{a^3(b^3 - r^3)}{r^3(b^3 - a^3)}$$

Max S_3 = p at inner surface

External Pressure:

$$S_1 = S_2 = -p\,\frac{b^3(a^3 + 2r^3)}{2r^3(b^3 - a^3)}$$

$$\text{Max } S_1 = \text{Max } S_2 = -p\,\frac{3\,b^3}{2(b^3 - a^3)}$$ at inner surface

$$S_3 = p\,\frac{b^3(r^3 - a^3)}{r^3(b^3 - a^3)}$$

Max S_3 = P at outer surface

The Lame thick shell formulation is a closed form solution if either the inside or outside radius is specified for a given depth and material. Since the main interest is inside volume, the trade-offs will step inside radius and evaluate effects on weight and displacement.

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Appendix H
Material Selection

The purpose of this appendix is to propose a methodology for the material selection of a wieght limited DSV. Many deep submergence material feasibility studies have been done. References 1 through 4 represent only a few of these studies.

The Navy classifies materials according to background and experience. Category one materials include alloys such as Hy80 and Hy100 for which there is an abundance of technical data and operational experience. Category two materials include alloys such as Hy130, Maraging (190) steel, HP 9-4-25 and Annealed Ti 6Al-4Va, for which there is an abundance of technical data but actual operational experience is limited. Category three materials include alloys such as Hy180 and Ti 621.8 Mo for which there is little technical data and experience.

The main problem facing the designer is that most of the materials suitable for a light weight DSV design are category three materials, material selection and certification of a category three material is a formidable task, even for a materials expert, but there is much the DSV designer can do even if he has a limited background in materials. The outline in Table H-1 is proposed as an outline of the overall material selection and certification procedure.

Each of the items in Table H-1 are equally important, but some are considered at different times.

Design Considerations:

Yield strength
Ultimate strength
Young's modulus
Density
Poisson's ratio
Brittle fracture resistance
Fatigue resistance

<u>Fabrication Considerations:</u> (See Table H-1)

Forming techniques

Joining

Total technical confidence factors

<u>Service Considerations:</u> (See Table H-1)

Environmental factors

Creep and stress rupture resistance

Repair

Maintenance

This outline is not meant to imply that the fabrication and service factors are not considered in the design, but rather that the crucial control of these factors occurs during the indicated times. For our pruposes, a methodology will now be presented which allows the designer to make an important input into the material selection.

The five materials under consideration for deep ocean use are listed below and represent the results of the afore mentioned material trade-off studies.

Material	Available Strength ksi	Density lbs/in^3	$\frac{\text{Strength}}{\text{Density}}$ (compared to steel)
Steel	250	.28	1.0
Aluminium	60	.099	.68
Titanium	130	.17	.86
GRP	150	.075	2.24
Cast Glass	300	.090	3.73

The above data represents the state-of-the-art for each of these materials. For a weight limited design it is imperative to use a strong

1. Availability:

   Base material

   Desired shapes and sizes of components

   Weld wier of consistent quality

2. Conventional mechanical properties:

   Elastic limit (compression and tension)

   Ductility (elongation and reduction in area)

   Bauschinger effect

   Poisson's ratio

   Impact energy absorption

   Property variations with thickness

   Temperature dependence

   Drop weight tear test

   Explosion bulge and crack starter

   Density

   Yield strength/weight-ratio

   Elastic, tangent, and secant moduli

3. Fatigue:

   Low cycle, high stress (notched and unnotched)

   Seawater corrosion (notched and unnotched)

   High cycle, low stress (notched and unnotched)

4. Toughness:

   Brittle fracture resistance

   Tearing resistance

   Flaw tolerance

   Resistance to crack initiation and propagation

   Thickness dependence

5. Fracture mechanics parameters:

   Critical fracture toughness, K_{1c}

   Effects of environment, K_{1SCC}

   Crack growth rates as function of K_1, including environmental effects

   Critical comtinations of stress and defect size

6. Environmental factors:

   General Corrosion

   Galvanic corrosion

   Stress corrosion and stress-corrosion cracking

   Contact corrosion

7. Creep and stress rupture resistance: (service temperatures)

8. Metallurgical effects:

   Chemistry control (plate, wire, weld metal)

   Initial heat treatment

   Sensitivity to thermal variations

   Section size effects on attainable properties

   Fabrication effects

   Aging

   Stress relief treatments

   Post-welding heat treatments

Table H-1

9. Forming techniques:

   Rolling, forging, pressing spinning

   Casting

   Formability - fabrication of desired shapes

   Machinability

   Dimensional stability (machining and welding)

   Attainment of critical dimensions within desired tolerance

10. Welding:

   Interrelation to design

   Base metal sensitivity to cracking

   Stress rupture at welding temperatures

   Heat-affected-zone characteristics

   Weld metal grain size and orientation

   Weld metal soundness, porosity and cracking sensitivity

   Applicable processes

   Shrinkage and distortion

   Effects of fabrication

   Effects of stress relief

   Inspectability

   Quality control aspects

11. Costs:

   Mill products - plates and forgings

   Weld wire

   Forming - hemispheres, cylinders, forged rings, etc.

   Fixturing

   Welding

   Machining

   Inspection

   Quality Control

12. Total technical confidence factors:

   Quality control

   Experience (producers and fabricators)

   Inspection capabilities and techniques

   Consistency of Weldability

   Weldment performance

Table H-1 (Cond't)

light material. The strength to density ratio is a good indicator of this property. The lower limit on the allowable stress is determined by the pressure envelopes effect on the vehicle in-air weight. An effective measure of materials effect on the vehicle is the effective weight, which is the weight of the structure plus or minus the weight of any required flotation material. This gives an effective measure if the weight-to-displacement ratio is greater than one, but if the weight-to-displacement ratio is less than one the pressure envelope is actually acting as buoyancy material. Therefore, the effective weight of the pressure envelope should reflect the gain in envelope weight for $W/D > 1.0$ and reductions in envelope weight for $W/D < 1.0$. An upper limit can be set on the allowable stress for a given material, due to the fact that a materials ability to resist crack initiation and propagation decreases with increasing strength[2]. The baseplate material and welded joints must possess sufficient toughness to provide the desired level of brittle fracture resistance, tearing resistance and tolerance for flaws. The high-toughness requirements imposed on materials for combatant submarines are not essential for most DSV designs. However, an acceptable toughness capability should still permit between two and five percent general plastic deformation in the region of a three inch long flaw without excessive tearing or catastrophic brittle fracture. The toughness level should be adequate to allow for both mechanical and shock wave type dynamic loadings. Mechanical loadings would include impact due to an accidental dropping, bumping of docks, tenders, mother sub, or bottom terrain. Shock wave loadings could be from exploding or imploding submersible pressure vessels while at maximum operating depth or explosive loadings from without. Busby's papers mentioned in the epxloration phase on "Hazards of the Deep" discusses other possible loading conditions. A proper impact loading analysis should establish a realistic vehicle toughness criterion. The criterion should be based on the overall structural repsonse to realistic loads rather than on the materials ability to absorb or take permanent deformation in the presence of flaws. The above discussion basically applies to brittle materials. If a material is ductile, the mechanical impact analysis should establish a realistic bucking criterion. A DSV with a ductile

pressure envelope suffering a collision at maximum operation depth may collapse due to the increase in the out-of-roundness of the pressure envelope. The in-service temperature and environment determine whether or not a material will act ductile or brittle[2].

Shock wave dynamic loading should also be investigated for its possible effects on toughness requirements. To date very little has been done in either type of dynamic loading. As a result, we are back to estimating the lowest required toughness from past experience. Both Masubuchi and Westinghouse[5] recommend the following toughness requirements:

Steel:

In steels of the 130,000 to 200,000 psi tensile yield strength class the required toughness (2-5% plastic deformation in the presence of a 2-in. long defect as measured in an explosive tear test) corresponds to a drop-weight tear energy (1-in. thickness) of 1250-1500 ft-lb at 30 F, and to a standard Charpy V-notch impact energy (C_vE) of 30-40 ft-lb at +30 F. The preferred levels are 1500 ft-lb by drop-weight tear and 40 ft-lb Charpy V-notch.

Titanium:

For an equivalent performance in the explosion tear test energy is 2000-2500 ft-lb. An absolute minimum would be 1500 ft-lb which corresponds to 1-2% plastic strain. Therefore, for purposes of subsequent discussion, a minimum of 1500 ft-lb with a preferred level of 2000 ft-lb (measured in weak direction of a 1-in. thick drop-weight tear specimen) will be employed. Since a good correlation does not exist between Charpy V-notch energy and the drop-weight and explosion -tear-tests for the titanium alloys , no minimum C_vE can be stipulated. While final acceptability must be ascertained with a drop-weight-tear test, C_vE in the range of 15-40 ft-lb can be used as a guide. This C_vE range corresponds to approximately 2000 ft-lb in the drop-weight -tear test[5].

Now that criteria have been proposed for bracketing the allowable stress for a given material, actual materials should be chosen for the allowable stress region.

Fatigue

The various parameters associated with fatigue performance are important. Fatigue can be considered from two viewpoints: (1) the invitation of a crack, and (2) the propagation of a crack. In addition, there are two types of cyclic loading to be considered; low-cycle, high stress and high-cycle, low stress. The most significant of these for the DSV system is the low-cycle, high stress fatigue. Conservatively, one should assume that undetected, non-leaking cracks or crack-like defects could exist at the beginning of the life of the vehicle. In this case, attention can be focused on the crack growth rate characteristics of the materials in a sea-water environment under the envisioned service spectrum of low-cycle, high stress loading. While various tests can be employed to rate the relative behavior of materials under these conditions, it is difficult to determine and stipulate a realistic minimum level of fatigue behavior which would be acceptable for service. Unfortunately, the fatigue consideration are extremely complex, and involve design, construction practice, and operation conditions, as well as the initial choice of a material. In the final coordination of design and material selection, these considerations will have to receive careful study and the conclusions reached may require verification by fatigue model tests.

So far only a few of the considerations mentioned at the beginning of this appendix have been discussed since it is felt that this is about the extent to which the designer needs to involve himself in the material selection process.

The output of this methodology should be a material, detailed mission profiles and vehicle life service requirements such as total number of dives to a given depth or a required vehicle life in years.

Summary:

The only feasible materials to date are high strength steels and titanium which are Category Three materials. References 6-24 are provided in order to consolidate some of the more pertinent available data. This reference list should help the designer initiate detailed material investigations. References 17 and 20 are helpful in the material selection of the hydrogen fuel cell tanks.

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