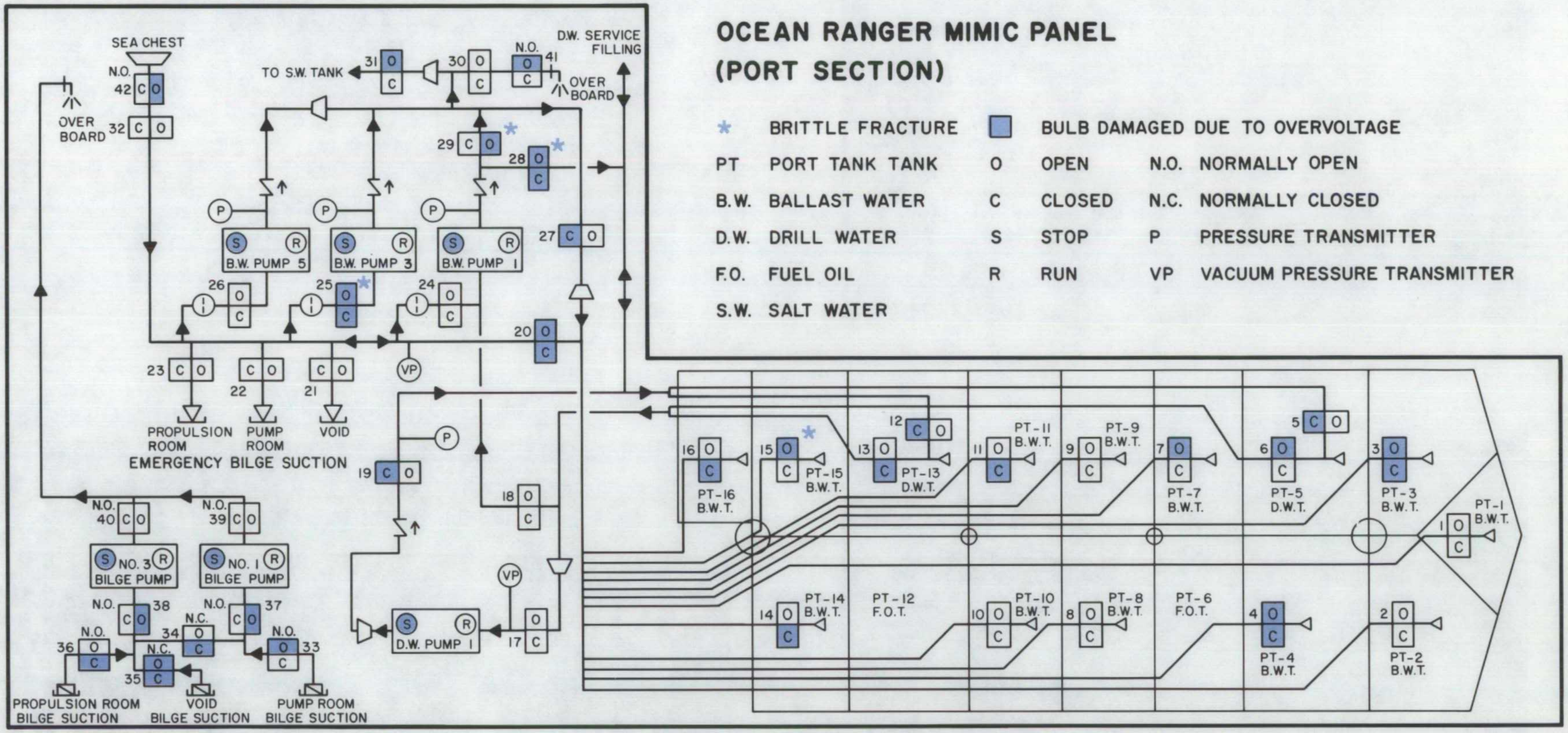


FIGURE 1 Port Mimic Panel.

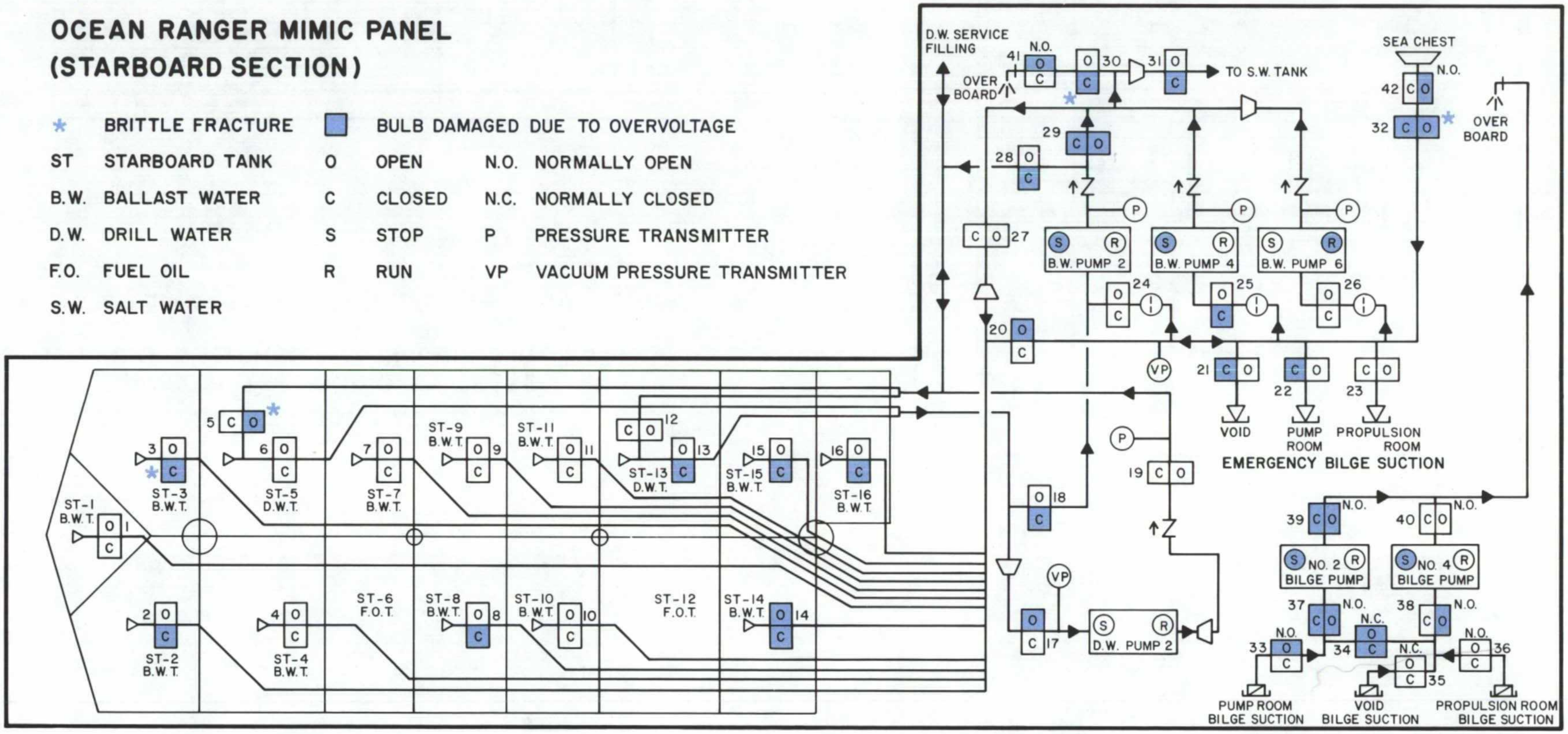


FIGURE 2 Starboard Mimic Panel.

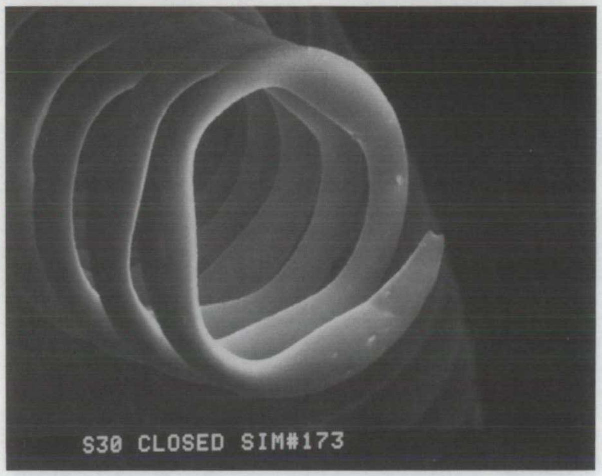


PHOTO 9 Filament from simulation panel flooded with water.

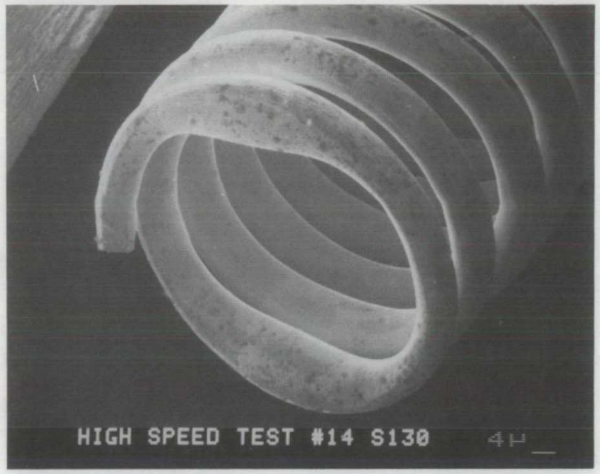


PHOTO 10 Filament experiencing 70 volts AC in the laboratory.

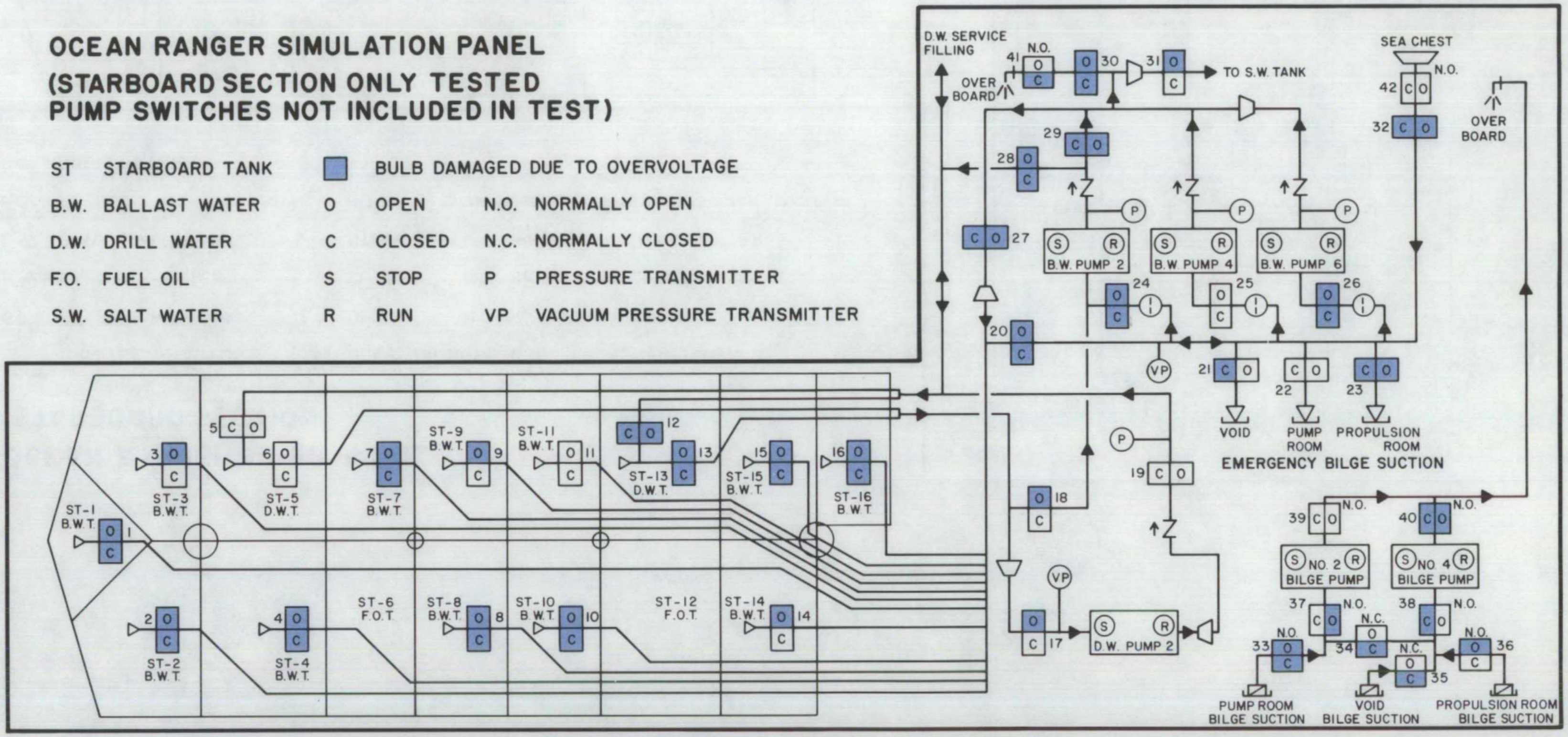


FIGURE 3 Starboard Mimic Panel (Simulator).

REPORT "F"
ENGINEERING REPORT EP 333/83
BALLAST CONTROL PANEL TESTS
8 September 1983

INTRODUCTION

1.1 The Royal Commission investigating the *Ocean Ranger* Marine Disaster requested the Aviation Safety Engineering (ASE) Facility of the Aviation Safety Bureau, Transport Canada to assist in the investigation by conducting certain tests and analyses on a ballast control (mimic) panel and the micro command switches used in these panels to determine the effects of salt-water flowing over the panel and entering the switches.

TEST EQUIPMENT

2.1 A control panel "Simulator" was constructed as depicted in Photo 1. It consisted of the two starboard mimic panels recovered from the rig mounted at a slant of 12° on 18 cm height boxes, see Photos 2 and 3, and a "Display and Monitoring Panel" mounted vertically on a box 80 x 30 x 50 cm in height, Photo 4. The three boxes were bolted together to form the test panel as shown in Photo 1.

2.2 New micro command switches and indicator lights, identical to those from the rig, were obtained and installed in the tank and pump room mimic panels as shown in Photos 2 and 3. The six pump switch pairs were different from those on the rig in that their buttons did not contain lights. The monitoring and display panel, Photo 4, consisted of the following:

1) a mimic panel of 32 green and 32 red lights, see Photo 5, which simulated the actual positions of the butterfly valves; 6 green and 6 red larger lights which simulated actual pump run or stop condition and ten toggle switches simulating the actual manual valve positions;
2) 115 volts main voltage and current meter, see Photo 6;
3) 24 volts voltage and current meters which monitor the lights on the mimic panel;
4) main switch and fuses;
5) relay fuses;
6) switch panel power supply fuses;
7) light test switch simulating the lamp test relay in the original panel.

The two stainless steel mimic panels were directly connected to ground for safety.

2.3 The wiring of the switches and indicator lights on the mimic panel was identical to that on the panels of the rig. However the components that were controlled via the panel, such as control valves, butterfly valve and pumps, were replaced by relays, lights and switches in the display panel. The solenoid control valves were replaced by relays, and the limit switches on the butterfly valve pistons were replaced by these relay contacts and lights showing their open and closed position. The manual valves were simulated by toggle switches and pumps by red and green indicator lights. The basic wiring diagram is shown in Figure 1.

2.4 The ballast control panel was designed to operate as follows:

1) when connected to an 115 volt AC supply and switches on, all 32 red lights on the switches and the 32 red butterfly valve lights on the display panel were lit. The manual valve indication lights 33 to 42 on the panel were lit red or green as a function of the toggle switches' position on the display panel;
2) the pump lights were also lit as a function of the pump switch positions on the panel. The meters indicated the main voltage and current and the voltage and current of the mimic panel lamp circuit.

2.5 When the light test switch is thrown, all non-lit lights light up at half power. This test serves as a check as to whether any light has burned out.

2.6 When a micro command switch green button is depressed the adjacent red light will go out and subsequently the green light will go on. The same thing will happen to the corresponding light on the display panel. On the rig, where a large butterfly valve had to open completely, there is a period of about 30 seconds duration when both red and green lights are off, which is an indication that the butterfly valve is "in transit".

2.7 The command switches and relays on the rig were wired in such a manner that a short circuit in the green switch could only open the control valve while a short circuit in the red switch could *not* close the valve. The test control panel was wired in a similar manner except that the corresponding green light on the display panel would light.

2.8 The switch indication lights' circuit was monitored by a voltage meter, an ampere meter and two ten ampere fuses. Since the test panel simulated only the starboard side, the fuses were half the value of the 20 ampere fuses used on the *Ocean Ranger*.

TESTING

3.1 The intent of the test was to apply a quantity of sea water over the ballast control mimic panel and to observe the effects. The tests were performed at the Engineering Facilities of Memorial University, St. John's, Newfoundland.

3.2 Appropriate scaffolding and a 50 gallon capacity trough was constructed to douse the panel with sea water in a manner similar to how it was believed to have occurred on the night of the capsizing. Photo 7 shows the test set-up just before the test.

3.3 Fifty gallons of sea water were poured on the panel over a three second period while all valve and switch lights were red. The observed effects were immediate. The 24 volt mimic panel light circuit fuse blew. On checking the blown fuse it was discovered that it was a five ampere fuse instead of the intended ten ampere fuse. The fuse was replaced with a ten ampere fuse which also blew after a few minutes. The fuse blew as a result of salt-water entering most of the switch lamp housings shorting out the lights and causing an increased load on the circuit.

3.4 Within minutes ten valve lights on the display panel turned to green, indicating that water had entered the microswitch and shorted it out. Most of these ten green lights did not go out during the one hour test period. The ten green lights corresponded to switch numbers: 1, 10, 18, 19, 21, 23, 25, 26, 28, and 30.

3.5 During the one hour test period that the power was left on the panel, most of the lights (typically one or two randomly distributed at a time) would flicker and light up very brightly momentarily and then die out. Also sparking was heard continuously, but also randomly distributed over the panel, as was the observation of smoke coming from the switches. At one point, one switch housing even caught fire. After about an hour it was decided to cut the power, since the damage observed was much more extensive than that observed on the *Ocean Ranger* ballast control panels.

TEST RESULTS

4.1 After the power was cut, some switches were removed from the panel and it was noted that the burning damage was similar to, but more extensive than, the damage observed on switch P-19 of the *Ocean Ranger* panel.

4.2 Analysis of the switches determined that shorts created by sea water between the 115 volt circuit and ground (via the leaf spring and the panel) caused more sparking, which provided sufficient heat energy to burn and melt the plastic housing of the switch.

4.3 The 115 volt circuit also leaked in a similar manner into the 24 volt circuit, causing the same burning damage on the manual valve indicator light housing. The damage to nearly all switches and indicators was severe, as is evident from Photo 8.

4.4 All 84 light bulbs were removed from the switches and indicator lights and microscopically examined for broken filaments. Ten bulbs were found to be relatively undamaged and twelve were too badly damaged by heat for proper examination of the filament, while 62 bulbs were found to have broken filaments with "hot" fractures indicating failure due to over-voltage. The analysis of the *Ocean Ranger* test light bulbs are covered in the Light Bulb Analysis Report "E", (EP 332/83).

DISCUSSION

5.1 The extensive burning damage to the panel switches made it clear that the test scenario sequence contained a basic difference from the actual events in the ballast control room prior to the capsizing of the rig, although shorting of switches and failure of light bulbs did occur as expected. The differences were considered to be due to one or more of the following:

a) the quantity of water used in the test did not compare closely to that flooding the mimic panel in the actual drill rig;
b) whereas in the test, power was left on the panel for a period of one hour, the drill rig crew may have cut power to the mimic panel shortly after the initial water flood;
c) the grounding of the test panel, and the AC polarity used, may not have been identical to that in the drill rig.

5.2 With respect to point (a) of Paragraph 5.1, the quantity of water used in the test was decided on the basis that a wave large enough to burst the porthole glass must have driven large quantities of water in the control room, of which a substantial part must have flooded the panel. The light bulb failures on the actual control panels were relatively evenly distributed, as is evident from report "E", (EP 332/83). This is evidence that water covered all areas of the panel, even around obstructions such as the upper part of the console, indicating that substantial quantities of water must have flowed over the panel.

5.3 With respect to point (b) of Paragraph 5.1, from communications it was known that the crew made a mopping-up effort after the bursting of the porthole. Circuit breaker NFB1, located behind the left-hand door in the upper part of the console, may have been pulled during the clean-up. A report that all systems were functioning normally again was put out around 22:00 hrs.

5.4 With respect to point (c) of Paragraph 5.1, when the test panel was constructed it was assumed that the stainless steel mimic panels should be grounded. This grounding provided the electrical pass for the sea water shorted sparking, which caused all the damage not generally observed on the *Ocean Ranger* panel switches (except for Switch P-19). Lack of grounding of the test panel would most likely have prevented this damage. It should be noted that no reference to grounding was found in any of the electrical schematics. It was also considered possible that all switches were wired to the neutral line of the 115 volt AC supply on the panels. If this was the case, then "shorting" to ground of the neutral line on the *Ocean Ranger* panel would not have created a potential and therefore no sparking damage would have occurred. The hot line of the 115 volt supply would then pass through the relay coil before being connected to the switch terminal. Shorting of this line to ground could have possibly energized the relay in an irregular manner, causing red lights to "flicker". In this configuration a short between the 24 volt circuit and the 115 volt hot line which passed through the relay coil first could still cause over-voltage in the light bulbs and burn them out.

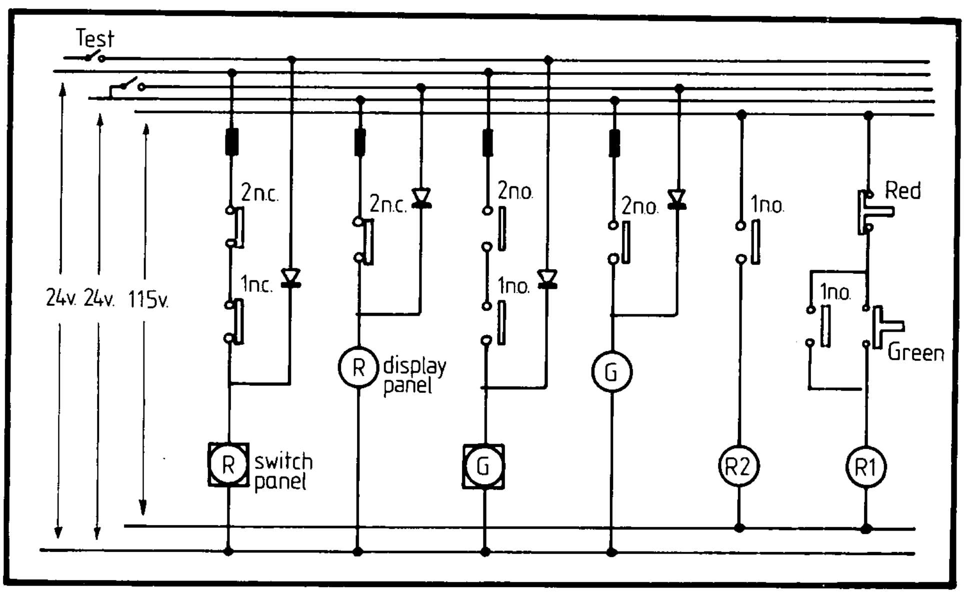


FIGURE 1 Basic Test Panel Schematic

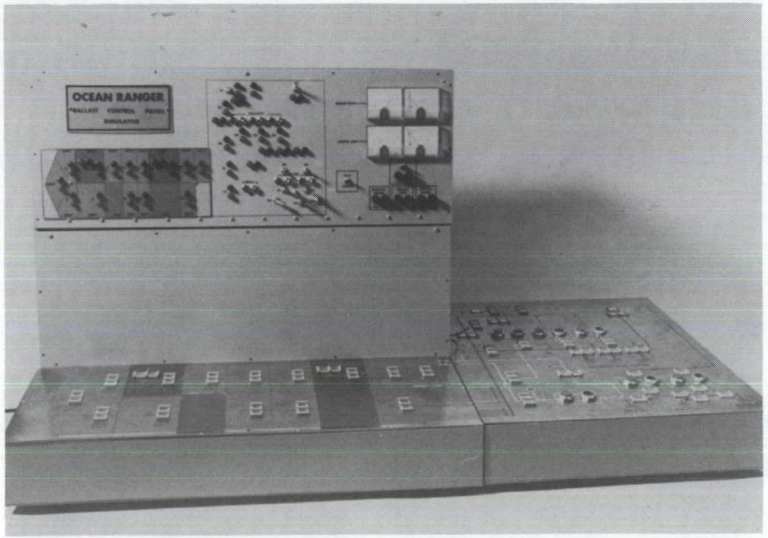


PHOTO 1 Complete Test Panel

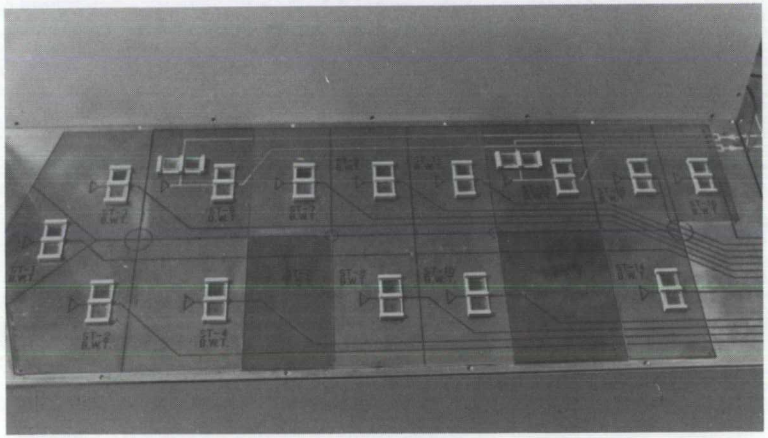


PHOTO 2 Starboard Tank Valve Switch Mimic Panel

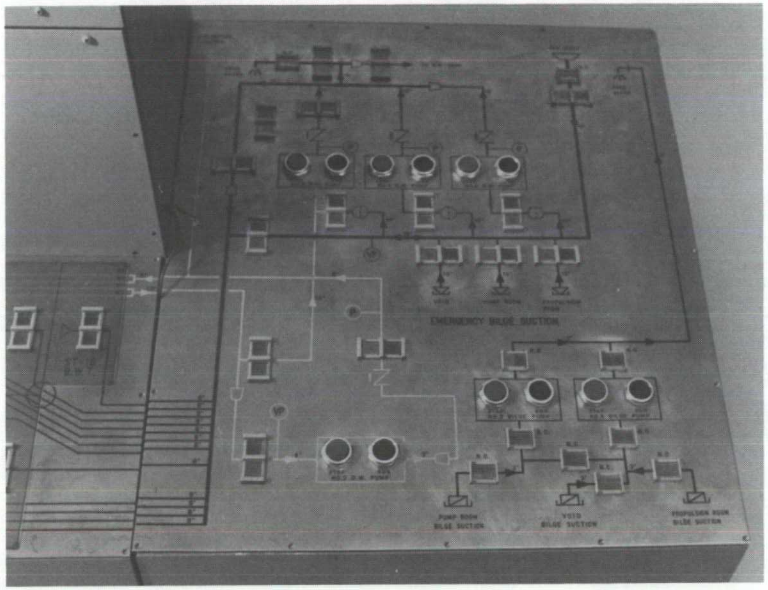


PHOTO 3 Starboard Pump Room Valves Switch Mimic Panel

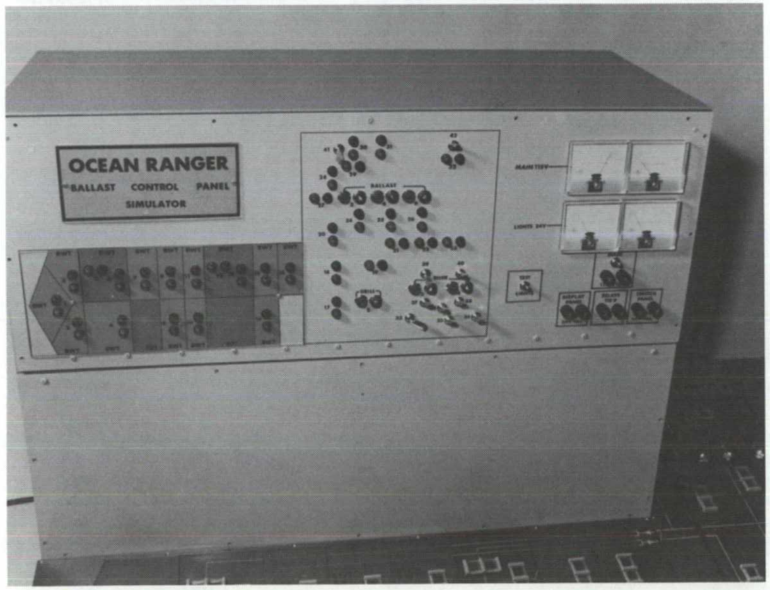


PHOTO 4 Display Panel

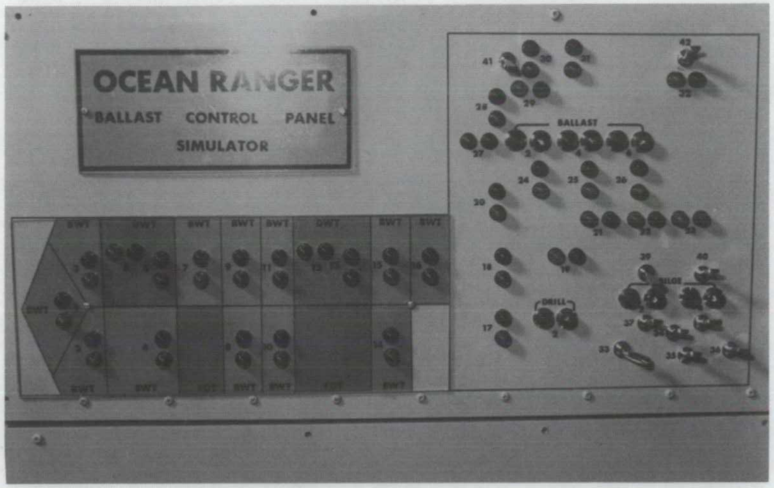


PHOTO 5 Display Mimic Panel

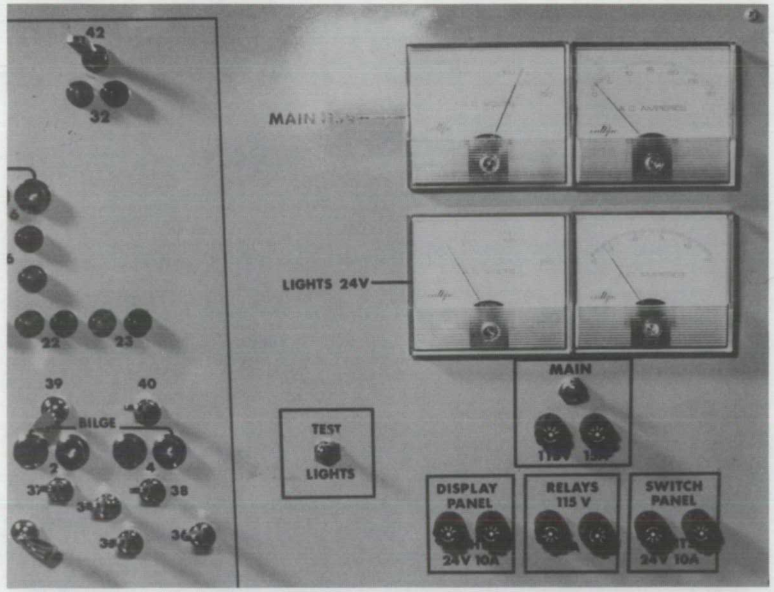


PHOTO 6 Meter and Fuse Panel

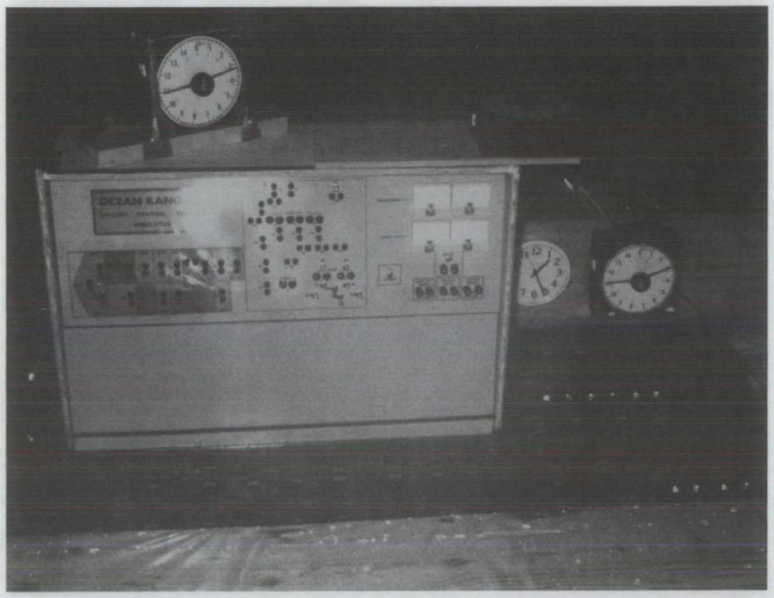


PHOTO 7 Panel set-up just prior to test.

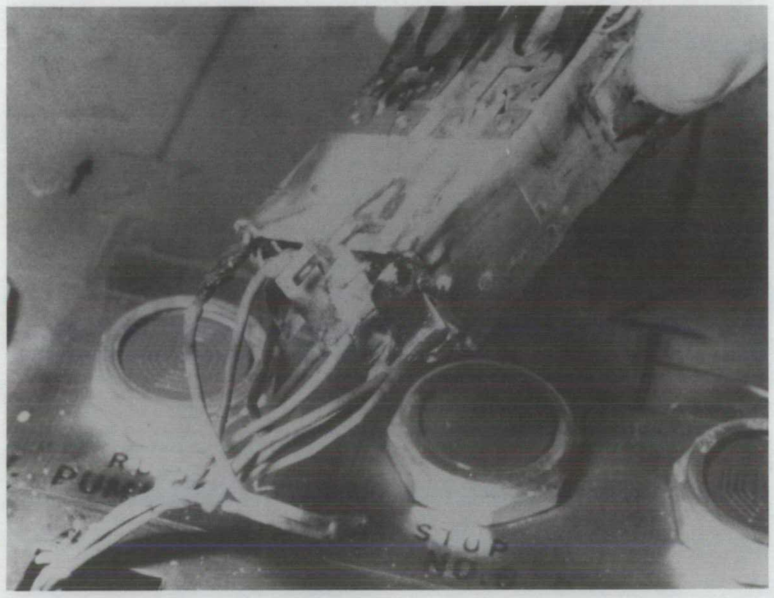


PHOTO 8 Micro command switch showing arcing damage after test.

REPORT "G"
ENGINEERING REPORT 195/82
BALLAST CONTROL ELECTRICAL SYSTEM
AND OVERALL ANALYSIS
8 September 1983

INTRODUCTION

1.1 The Royal Commission investigating the *Ocean Ranger* Marine Disaster requested the Aviation Safety Engineering (ASE) Facility, of the Aviation Safety Bureau, Transport Canada to assist in the investigation by conducting certain tests and analyses on the ballast control room portholes, the ballast control (mimic) switch panels and the ballast control solenoid valves.

1.2 Three portholes, four switch panels and six valve banks were forwarded to ASE with the following list of requests:

1) prior to analysis and testing, photograph and identify all portholes, control valves, switches and indicator lights;
2) determine the mode of failure of two portholes with broken glass;
3) pressure test the undamaged porthole to determine wave force required to fail the glass;
4) examine all control valves for evidence of possible manual operation;
5) determine significance of presence or absence of the rubber plugs on the solenoid valves housing;
6) determine the valve positions of the 18 valves found with manual actuator rods inserted;
7) examine all switches for evidence of burning or arcing to terminals and contacts;
8) examine all indicator lights for evidence of burning, arcing and light bulb failure;
9) determine mode of all light bulb failures;
10) analyse the switches and indicator lights in terms of their susceptibility to salt-water damage;
11) determine the effects of salt-water flow over the control panel to the ballast control system through testing on a reconstructed ballast control panel;
12) analyse the ballast control electrical system in terms of safety, reliability and susceptibility to salt-water damage and electrical failure.

1.3 The Royal Commission Counsel provided ASE with the following information related to the accident:

1) on 14 February at approximately 19:30 hours a porthole in the ballast control room was reportedly smashed by a wave and quantities of water entered the control room;
2) the crew reported that the influx of sea water had affected the ballast control panels and that a cleaning operation was in progress;
3) around 22:00 hours it was reported that all systems were functioning normally again and that mopping-up was completed;
4) on 15 February at around 01:00 hours a severe and uncontrollable forward list was reported, together with a Distress call.

1.4 After receipt of the ballast control room components and the Royal Commission's requests, ASE divided the necessary work to be performed into the following separate projects:

A – Porthole Analysis
B – Porthole Glass Testing
C – Ballast Control Valve Analysis
D – Ballast Control Panel Switch Analysis
E – Light Bulb Analysis
F – Ballast Control Panel Test
G – Covering Report with Ballast Control Electrical System and Overall Analysis

Each of the projects is covered in a separate report which are assembled in this covering report "G" and will be referenced by their assigned designator letter.

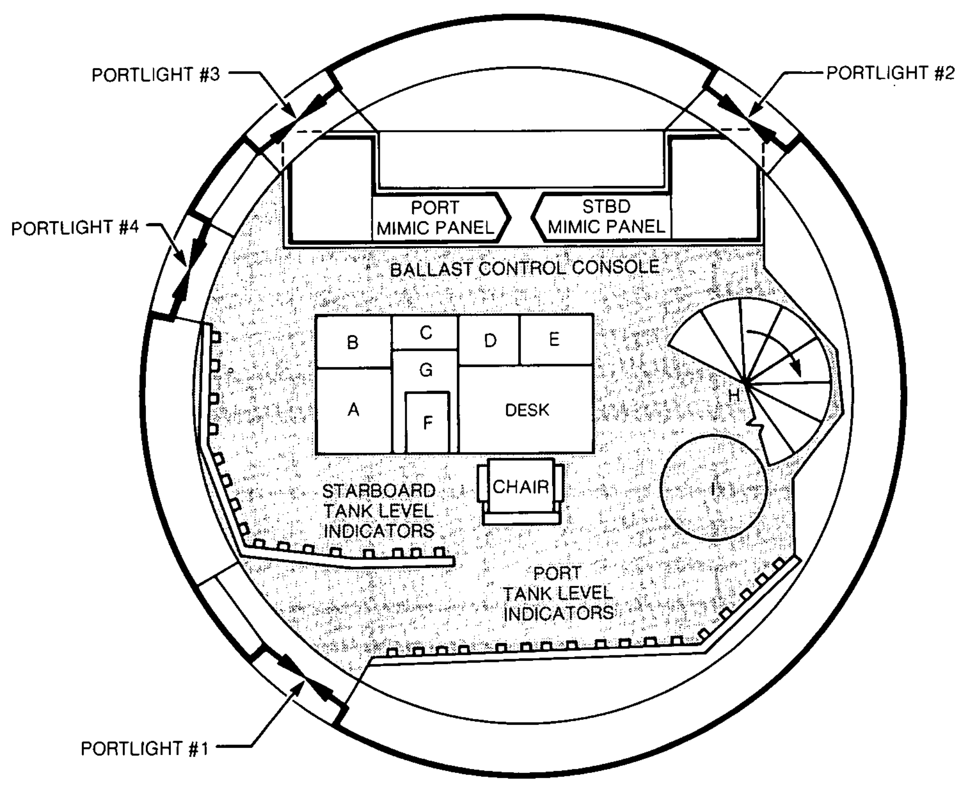


FIGURE 1 Ballast Control Room

A Hydrophone Control Unit
B Hydrophone Electronics Panel
C CO_2 Actuating Cabinet
D Sliding Door Control
E Smoke Detection Cabinet
F Teleprinter
G Display Terminal
H Spiral Staircase
I Watertight Manhole

SYSTEM DESCRIPTION

2.1 The ballast control room was located in the after centre starboard column SC3 about 33 meters above the keel baseline. The room was circular, approximately 5 meters in diameter and could be entered only from above, through a spiral staircase from the upper control room. The room had four portholes located as described in Figure 1 and ASE Report "A", (EP 266/82). The portholes were about 50 cm in diameter and could not be opened, each having a deadlight hinged from the top. The furnishings and equipment layout is also shown in Figure 1.

2.2 The ballast control console was about 3.5 meters wide, 1 meter deep and mostly counter high except for the relay cabinet and meter panel.

2.3 The ballast control console consisted of:

1) service tank level alarm panel;
2) port and starboard watertight alarm panels;
3) port and starboard meter and gauge panels;
4) port and starboard relay and terminal racks;
5) port and starboard control switch mimic panels;
6) port and starboard control valve banks.

Only the control switch mimic panels and the six control valve banks were recovered from the wreckage.

2.4 The ballast control system functioned roughly as follows: in each pontoon, the twelve ballast tanks are connected with butterfly valves to the ballast water manifold which is located in the pump room aft in the pontoon, where routing valves, pumps and piping are located to accommodate the ballast and level requirements. Two drill water tanks can also become part of the system in an emergency and are controlled by a four valve manifold. The "overboard" and "sea chest" each have a manually operated valve for emergencies and these are normally open.

2.5 Thirty-two tanks and routing valves for each pontoon are operated by one-way pneumatic pistons with spring return, and are controlled by the solenoid operated control valves located in the base of the ballast control console. Air supply lines approximately one centimeter in diameter run the air from the control valves to the butterfly valve pistons. The air supply was typically 90 psi, and removal of the air supply would cause all thirty-two pneumatically operated valves to close.

2.6 All butterfly valves, and all the manually operated valves, have limit switches at the extremes of their stroke which control the indicator lights on the ballast control mimic panels, ie: "Green" means fully open, "Red" means fully closed and no light means "valve in transit". The control valves consist of a "one-way" shuttle valve which on electrical activation of the solenoid allows high pressure air from a compressor into the air supply line for the individual butterfly valve, activating its piston. De-activation of the solenoid allows the air in the supply line and piston to exhaust into the control valve exhaust manifold.

2.7 The solenoid in turn is controlled, through a relay, by the micro command switch pair in the ballast control mimic panel. These are momentary microswitches controlling the self-holding relay. The push button micro command switches on the mimic panel are directly controlled by the ballast control operator, who would select the appropriate valve and pump configuration, in response to the requirements from the drill crew.

ANALYSIS

3.1 The random distribution of light bulb failures and non-failures over the whole mimic panel suggested that water covered the whole panel at the time of the porthole failure.

3.2 For a light bulb to blow due to overvoltage the following conditions were required:

1) the 115 volt supply must have leaked into the 24 volt lamp circuit (the damage to P-19 "green" attests to that having occurred, as also does the physical evidence of the 68 blown light bulbs); and,
2) the bulb would have to be set in the "ON" condition by the limit switch or the light test switch; and,
3) the bulb contacts could not be shorted out by sea water.

Only switches that are shorted out locally by arcing can possibly burnout on their own light bulbs when not in the "ON" condition. This condition only occurred at P-19 "green", which light was not damaged.

3.3 For a light bulb *not* to have blown, the following conditions were required:

1) the bulb would have to be set in the "OFF" condition by the limit switch; and/or
2) the bulb contacts would have to have shorted out by sea water; and/or,
3) no 115 volt leakage into the 24 volt circuit had occurred (the 68 burned out bulbs attest against that having occurred).

3.4 Considering the distribution of bulb failures, the above conditions suggest that the panel had been operated while the sea water was affecting the panel. From the above conditions, it was also clear that evaporation or the draining or cleaning away of the water in the lamp socket could cause its light to burnout.

3.5 Removal of electrical power from the mimic panel could only be effected by pulling circuit breaker NFB1. This would have left only the pump switches functional although disabling the pump switch indicator lights. Removal of the electrical power would cause all open pneumatically operated valves to close. The only reasonable way to operate these valves after power is removed is with the brass actuator rods as described in Report "C".

3.6 It should be noted that the only valves that showed evidence of having been operated by the brass rods were tank valves, as is known from Report "C". Therefore if electrical power was removed from the panel, the manual operation of the tank valves alone appear to be totally non-effectual because other valves should have also been open to effect and control flow.

3.7 Portholes or side scuttles have mostly been designed and standardized with ships in mind and not drilling rigs. In general a ship is rarely anchored stiffly against the waves as is the typical condition for a drilling rig. Therefore, it is considered reasonable to expect that a porthole in a rig would get a much more severe pounding than in a ship. In view of its application it should not have been considered unlikely that this porthole would have failed.

3.8 Damage to the ballast control electrical system removes from the operator his only source of information in relation to the stability control of the rig. There was no back up system in the ballast control room, other than the manual (brass rod) operation of the solenoid control valves. However it is considered that the inconspicuous identification and the awkward-to-reach location of the valves would make manual operation in an emergency very difficult, if not impossible.

3.9 It is considered that most of the ballast control electrical equipment appeared to be an undesirable complexity in the system. The whole pneumatic system could have been easily and conveniently controlled by hand operated spigot type valves located directly in the mimic panels, eliminating 64 relays, 128 switches and 64 solenoid valves. Two or more separate "red" and "green" indicator light mimic panels directly wired to the limit switches on the valve pistons would then provide status information in various locations. The valve panel should also have been duplicated on the bridge for emergencies.

CONCLUSIONS

Answers to the requests of the Commission as listed in 1.2 are as follows:

4.1 All photographs requested will be submitted to the Commission together with this report.

4.2 A full analysis of the two broken portholes is contained in Report "A" attached.

4.3 The results of the porthole glass pressure tests are contained in Report "B" attached.

4.4 Evidence of manual operation of the solenoid control valves is contained in Report "C" attached.

4.5 An analysis of the rubber plugs in the solenoid valve housing is contained in Report "C" attached.

4.6 The valve positions of the valves with manual actuator rods inserted are listed in Report "C" attached.

4.7 An analysis of all switches is contained in Report "D" attached.

4.8 An analysis of all indicator lights is contained in Report "D" attached.

4.9 An analysis of all light bulbs is contained in Report "E" attached.

4.10 An analysis of the switches and indicators' susceptibility to salt-water damage is contained in Report "F" and Report "G" attached.

4.11 The results of the control panel salt-water tests are contained in Report "F" attached.

4.12 The analysis of the ballast control electrical system is covered in Report "G" attached.

REPORT "H"
ENGINEERING REPORT EP 72/84
PUMP SWITCH FAILURE DEMONSTRATION
01 March 1984

INTRODUCTION

1.1 The Royal Commission investigating the *Ocean Ranger* Marine Disaster requested the Aviation Safety Engineering (ASE) Facility, of the Aviation Safety Bureau, Transport Canada to assist in the investigation by conducting certain tests and analyses on the ballast control room portholes, the ballast control (mimic) switch panels and the ballast control solenoid valves.

1.2 The results of these tests and analysis were submitted to the Commission in September 1983 in ASE Reports "A" through "G". However, in order to clarify the proposed scenario that sea water entered the pump switches from below the ballast control panel, after the porthole failure, it was requested that ASE carry out supplementary testing on new pump switches to demonstrate that water can indeed run along the bottom of the control panel, enter the pump switches and eventually cause 115 volts AC to leak into the 18 volt AC switch light bulb circuit and burnout the light bulb filament by an over-voltage as proposed in paragraph 3.2 of ASE Report "E", (EP 332/83).

TESTING AND ANALYSIS

2.1 For recording convenience, the pump switch light failure tests were split into two separate tests:

1) a demonstration that water can flow along the bottom of a slightly inclined (12 degrees) horizontal panel;
2) that sea water, once entered into the switch near its light bulb terminals, can in fact cause the light bulb to burn out.

2.2 The pump switches tested were identical to the switches recovered from the *Ocean Ranger*. They were manufactured by Tokyo Electric Co. under part number 4031E-11R for the red "Stop" switch and 1031E-11G for the green "Run" switch.

2.3 The "Stop" switch is a self-latching push button switch which changes state every time it is pushed. The "Run" switch, Photo 1, is a momentary push button switch and activates a self-holding relay to keep the associated pump running. Both switches have a 115 volt to 18 volt transformer built in, to power the light bulb located in the clear plastic green or red button. Photo 2 shows a disassembled "Run" switch with its major parts identified.

2.4 Photo 3 shows the run switch split at the transformer / push button interface. It shows the bottom of the light socket at "A", the secondary transformer winding and terminals at "B", the primary winding and terminals at "C" and the external 115 volt transformer connection terminals at "D".

2.5 The push button part of the switch is constructed and sealed in such a way that water flowing on top of the panel cannot enter the switch body. However if water could reach the switch body from underneath the panel then it can readily flow into the switch near the light bulb connection terminals as indicated in Photo 1.

2.6 The transformer terminals of the primary windings "C" and the light bulb socket "A" are in very close proximity, as is evident from Photo 3. This close proximity would facilitate the leakage of the "primary" 115 volts into the "secondary" light bulb contacts, if sea water were to enter the transformer area within the switch body.

2.7 To demonstrate that water can flow under a horizontal or near horizontal panel a small test fixture was constructed, as depicted in Photo 4. It consisted of a 10 x 24 inch sheet metal panel supported by a simple plywood frame to accommodate a 12 degree slope. A narrow slot about 0.050 inches wide and 6 inches long was cut in the sheet about 4 inches above the pump switch position. A video camera was placed so as to show the bottom of the plate and that portion of the switch that is mounted below the plate.

2.8 The short videotape accompanying this report clearly shows how the sea water, which was poured on top of the panel above the slot, flows through the slot and adheres to the bottom of the panel while running down the incline, and then flows down the side of the switch and into the opening near the light terminals.

2.9 The slot in the test panel was considered to be an acceptable simulation of the small gaps between the rectangular valve switches and the ballast control panel mounting holes that were noted to exist in the ballast control panel assembly.

2.10 For the second test, which demonstrated that the ingestion of sea water can cause an over-voltage in the light bulb causing it to burn out, the switch light bulb terminals were directly wired with 115 volts AC into the transformer primary. Some sea water was injected into the two cavities near the light bulb terminals, Photo 1, while being videotaped. After a few minutes the light changed brightness, flickered, went out and back on again, then flashed very brightly, indicating an over-voltage and failure by burnout. The accompanying videotape has this event recorded for demonstration.

2.11 It was noted that new bulbs did not burn out under the above described test conditions. However the primary transformer windings burned out every time the 115 volt supply was left on for more than fifteen minutes after the application of sea water. The voltage "surges" observed in the "secondary" light bulb circuit were only up to approximately 25 volts and were of relatively short duration; therefore, only light bulb filaments weakened by age were susceptible to burnout.

2.12 It must be noted that the increase in incandescence of a light bulb filament, when it burns out, is not necessarily the result of an increase in the supply voltage but rather can be due to a release of inductive energy as a result of the filament fracture.

2.13 Of the 24 pump switches recovered with the *Ocean Ranger* ballast control panels, only one (the port side #1 bilge pump stop switch) transformer had a burned out primary winding. All other pump switch transformers were electrically undamaged. This would indicate that either not much water entered the switches or, more likely, that the power was removed from the pump circuits not long after water damage was noted. Twelve of the 24 lights were found burned out in the *Ocean Ranger* pump switches (11 red and 1 green). Only lights that were on at the time that water entered the switch bodies could possibly have burned out.

2.14 During the tests it was also noted that considerable amounts of condensation formed in the push buttons, due to the heat of the light bulb. The videotape shows evidence of this type of condensation. This may explain why 14 of the 24 red and green buttons were missing when the control panels were recovered. The crew could well have removed the buttons in their clean up efforts, after noticing erratic behaviour of the lights and/or condensation in the buttons.

CONCLUSIONS

3.1 It has been demonstrated that sea water could easily have (and most likely did) run along the underside of the ballast control panel and entered the pump switch bodies near the light terminals.

3.2 It has also been demonstrated that the pump switch indicator lights can burn out after the entry of sea water, providing the light bulb filament was substantially damaged by age.

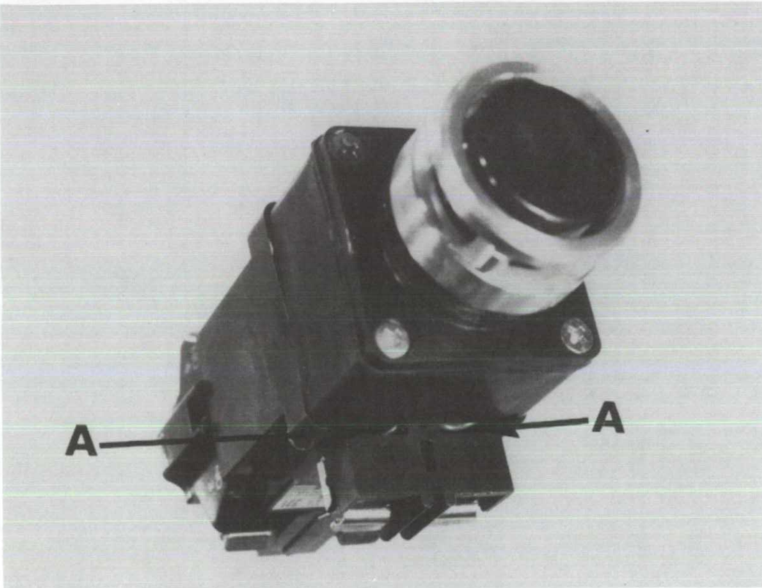


PHOTO 1 Pump switch *green* or *run*, showing the cavity on the side of the body where water can enter at *A*.

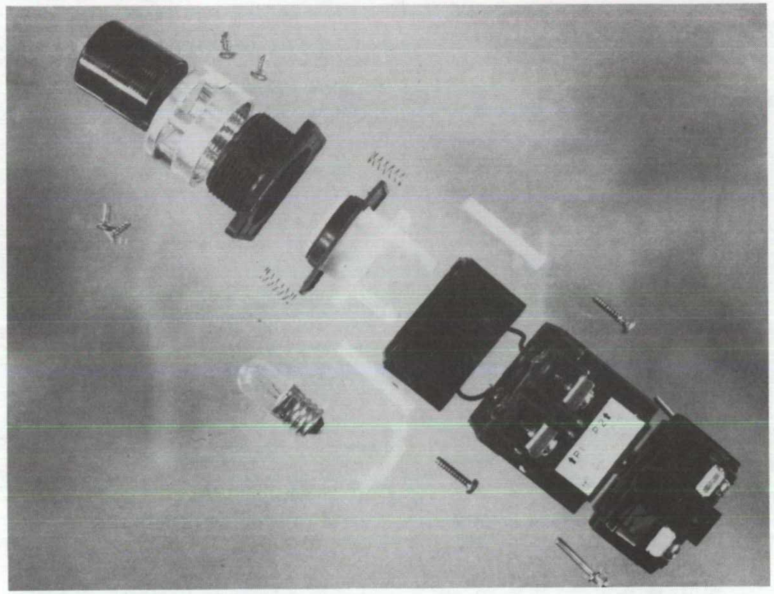


PHOTO 2 The same pump switch disassembled.

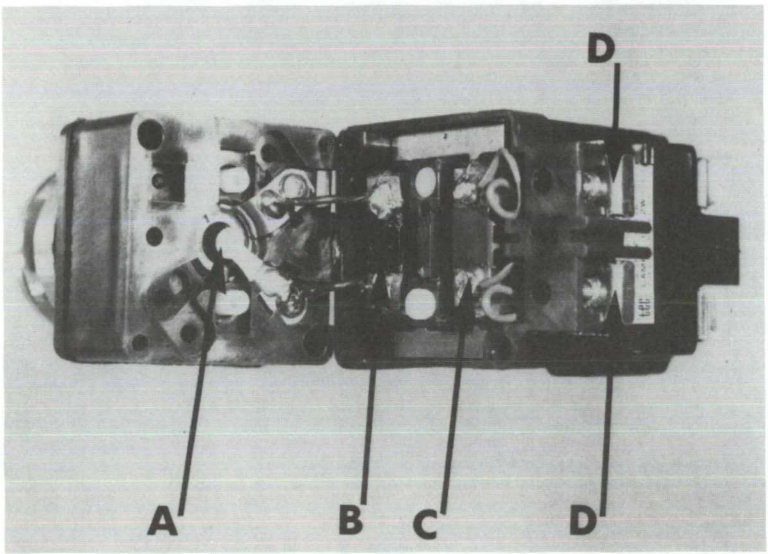


PHOTO 3 The same switch split at the transformer and button light interface. *A* is the bottom of the light bulb socket, *B* secondary winding, *C* is the primary winding and *D* is the external transformer connections.

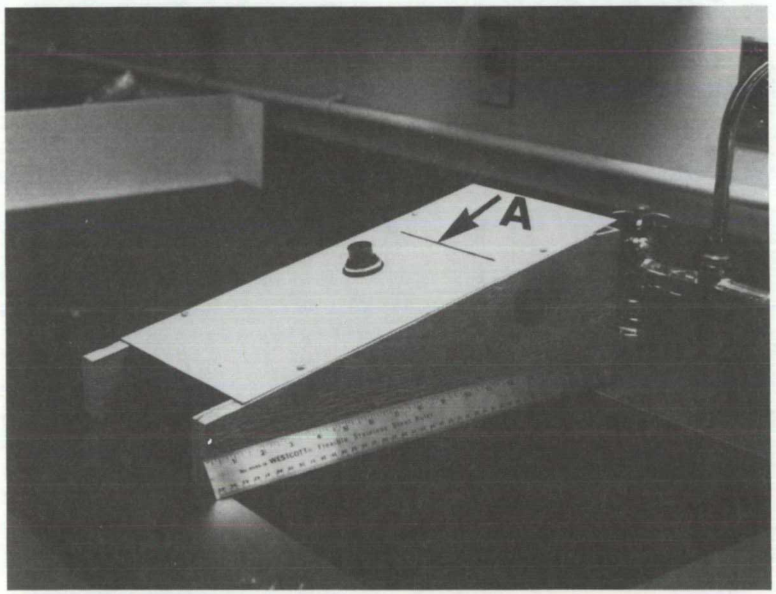


PHOTO 4 The test fixture for water ingestion test, showing the slot at A.

REPORT "I"
ENGINEERING REPORT EP 73/84
MICROSWITCH FAILURE ANALYSIS
01 March, 1984

INTRODUCTION

1.1 The Royal Commission investigating the *Ocean Ranger* Marine Disaster requested the Aviation Safety Engineering (ASE) Facility, of the Aviation Safety Bureau, Transport Canada to assist in the investigation by conducting certain tests and analyses on the ballast control room portholes, the ballast control (mimic) switch panels and the Ballast Control solenoid valves.

1.2 These tests and analyses were completed and the results were submitted to the Royal Commission in September 1983 in ASE Reports "A" through "G". ASE's Report "F", (EP 333/83) titled "Ballast Control Panel Tests", covered testing of a reconstructed ballast control panel for its susceptibility to electrical damage and failure, when doused with sea water. During this test ten control valve "open" (green) switches failed, as is evident from Photo 1 and paragraph 3.4 in ASE Report "F".

1.3 The green light display on the special monitoring panel shown in Photo 1 was the basic evidence that these switches had failed at the time of the test. The manner of failure of these 10 switches, as postulated in paragraph 3.4 of ASE Report "F", was a logical deduction from indirect evidence. At the time of this test it was decided not to open the microswitches to look for sea water, because of the risk of losing the evidence (i.e., the sea water) in the process of opening the switches.

1.4 The valve control switches were removed from the test panel, identified and stored in open plastic bags, in a low humidity environment, for a period of three months, to allow any sea water to evaporate and leave identifiable salt deposits within the microswitch, if sea water had indeed entered the switches.

1.5 For positive and direct proof that sea water can enter the valve control microswitches it was requested by the Commission Counsel that ASE open a selection of microswitches from the test panel described in ASE Report "F", after an appropriate drying period, and determine if there indeed was any evidence to show sea water had entered the microswitches.

EXAMINATION AND ANALYSIS

2.1 The microswitches from the ten valve control "open" (green) push button switches, referenced in paragraph 3.4 of ASE Report "F", were removed and one side of each was gently abraded away to reveal its mechanism, as is shown in Photo 2. A detailed microscopic examination showed small white specks in varying numbers in all of the ten switches. Photo 3 shows typical white specks in one of these switches.

2.2 Scanning electron microscopic analysis revealed the white specks to be largely of a spikey crystalline shape as shown in Photos 4 and 5. Energy dispersive X-ray spectrometric analysis revealed that these specks were various crystalline compounds of sodium and/or chloride, as described in more detail in the ASE Report by the Physical Analysis Specialist, attached.

2.3 The most likely place for sea water to have entered the microswitch was around the red activation button, as shown in Photo 2. However, one of the switches showed that sea water had entered through a space in between the two halves of the switch housing which was apparently not properly sealed with cement, as is evident in Photo 6.

CONCLUSION

3.1 It was determined through energy dispersive X-ray spectrometric analysis that sea water had entered the valve control "open" (green) switches indicated in paragraph 3.4 of ASE Report "F".

Department of Transport
AVIATION SAFETY
ENGINEERING LABORATORY
Internal Request for
Technical Analysis

REQUIREMENTS

Please identify the white deposits found on some microswitch components.

4 February 1984 M. Vermij

FINDINGS

Scanning Electron Microscope (SEM) examination of the "white" deposits on three locations of the microswitch, including two of the gold plated bus bars and one silver contact surface, indicates a wide variation in precipitate morphology – as per attached photomicrographs, see Photos 7 and 8.

Energy dispersive X-ray analysis confirms the presence of a corresponding number of different chemical species, although common to all analyses are high concentrations of sodium, chlorine and other typical constituents of sea water.

Attached spectra #1-5 refer as follows:

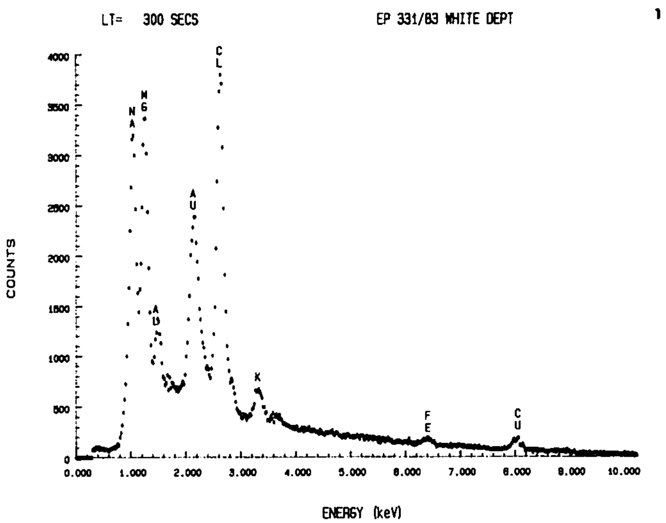
#1, 2 and 3 – random spectral analysis of general background deposits with no well defined crystallographic habits.

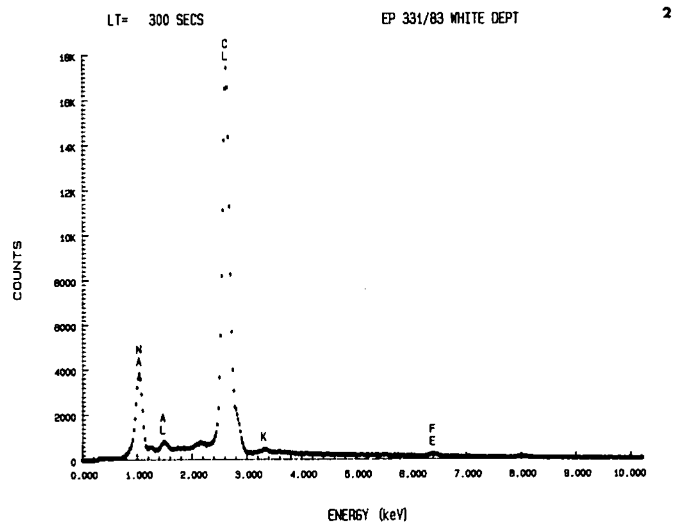
#4 – spectral analysis of typical spikey clusters. Shows sodium salt with minor trace elements suggesting these growths may be sodium compounds with low atomic radicals such as the carbonate, nitrate or oxide which are not detectable by the non-dispersive technique.

#5 – spectral analysis of well developed crystalline phase shown by the longer arrows on Photo 7 and identified as silver chloride.

It can be concluded that all compounds present were derived from reaction with salt-water.

8 February 1984 K.M. Pickwick


LT= 300 SECS
EP 331/83 WHITE DEPT
1
COUNTS
ENERGY (keV)
NA
MG
AL
AU
CL
K
FE
CU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EP 331/83 WHITE DEPT
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COUNTS
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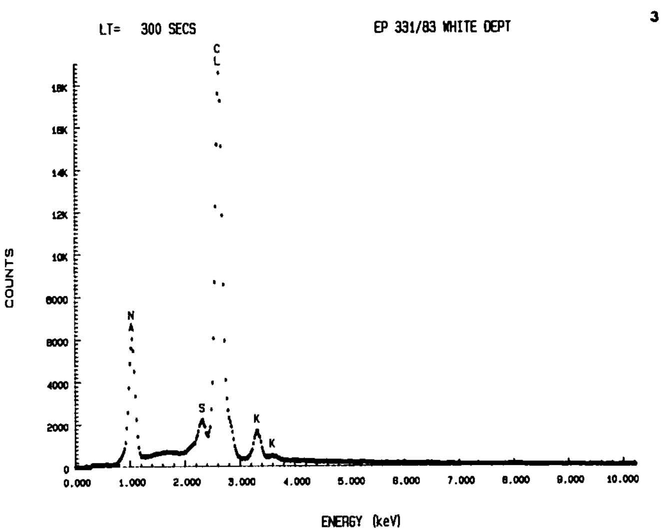
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EP 331/83 WHITE DEPT
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COUNTS
ENERGY (keV)
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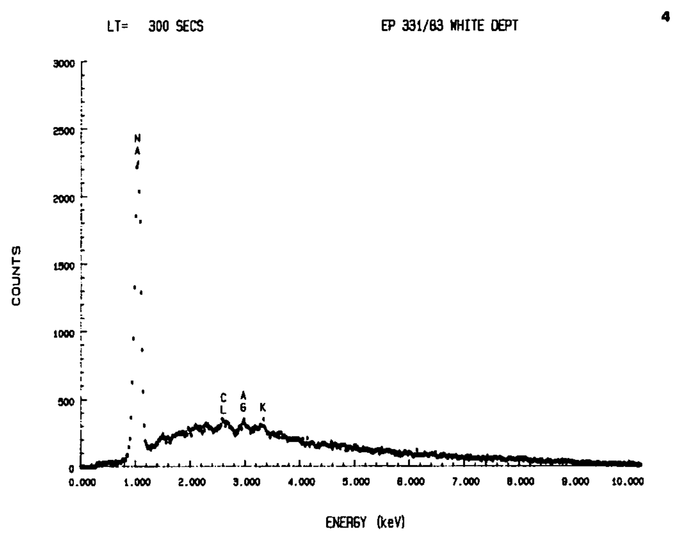
K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EP 331/83 WHITE DEPT
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COUNTS
ENERGY (keV)
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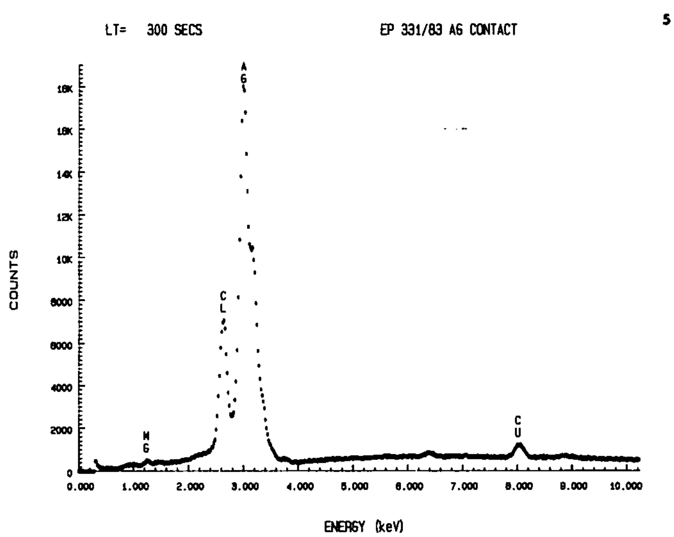
CL
AG
K


LT= 300 SECS
EP 331/83 AG CONTACT
5
COUNTS
ENERGY (keV)
MG
CL
AG
CU

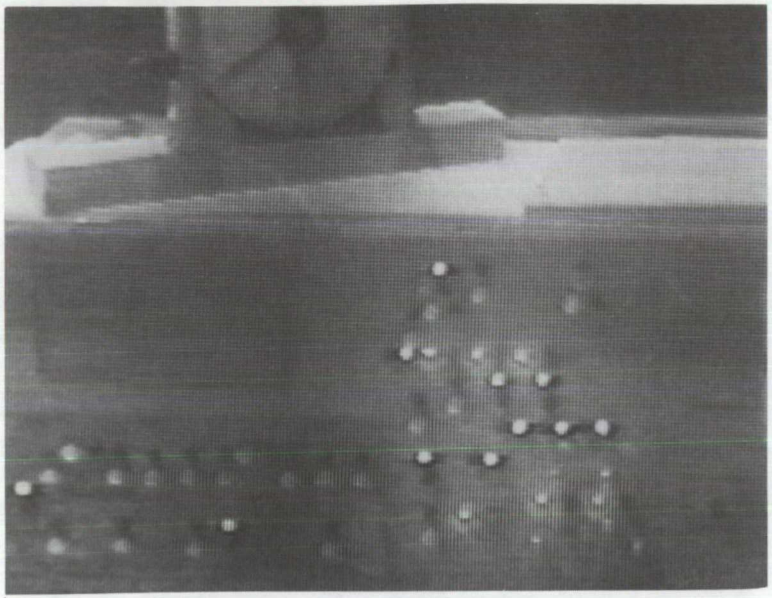


PHOTO 1 The test panel at the time that the monitor panel showed the 10 green lights, indicating switch failures.

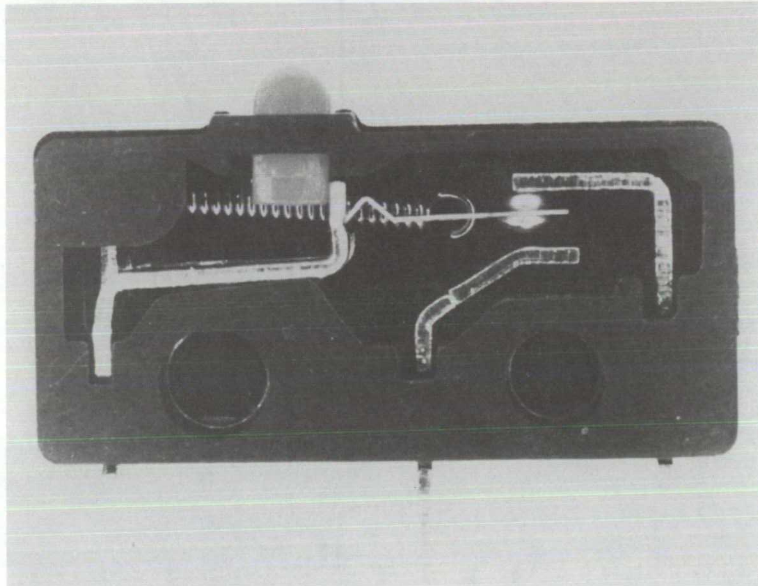


PHOTO 2 A typical microswitch with the side of its housing removed, showing its mechanism.

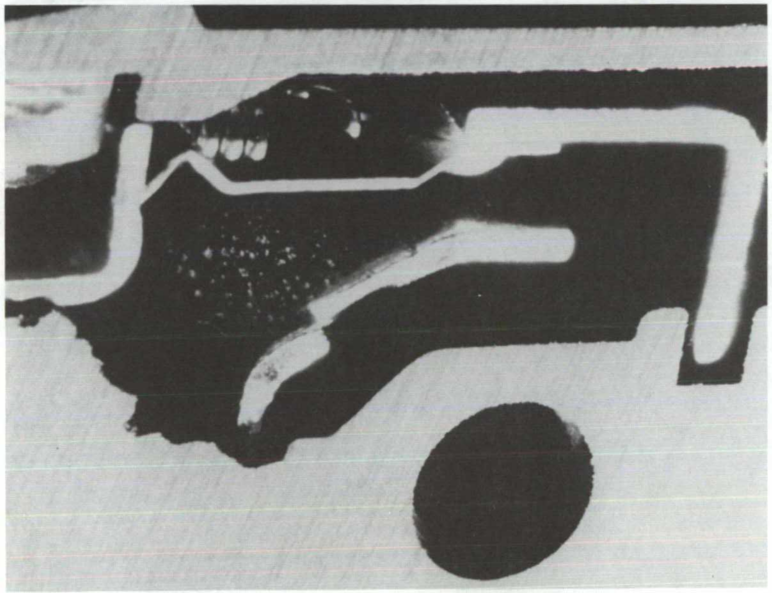


PHOTO 3 Detail of the microswitch mechanism showing the white specks deposited on the gold plated contact bar.

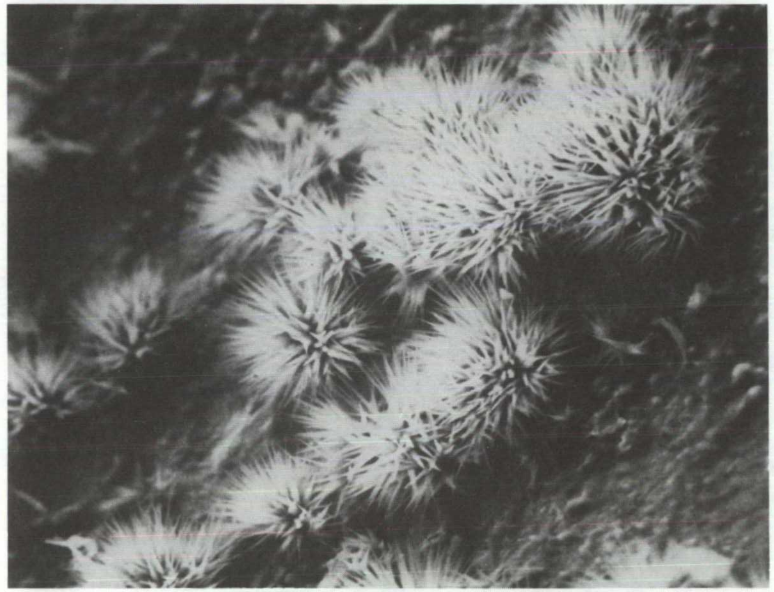


PHOTO 4 Micrograph of the white specks found to be sodium compound crystal clusters.

PHOTO 5 Micrograph of a sodium compound crystal cluster surrounded by small sodium chloride crystals.

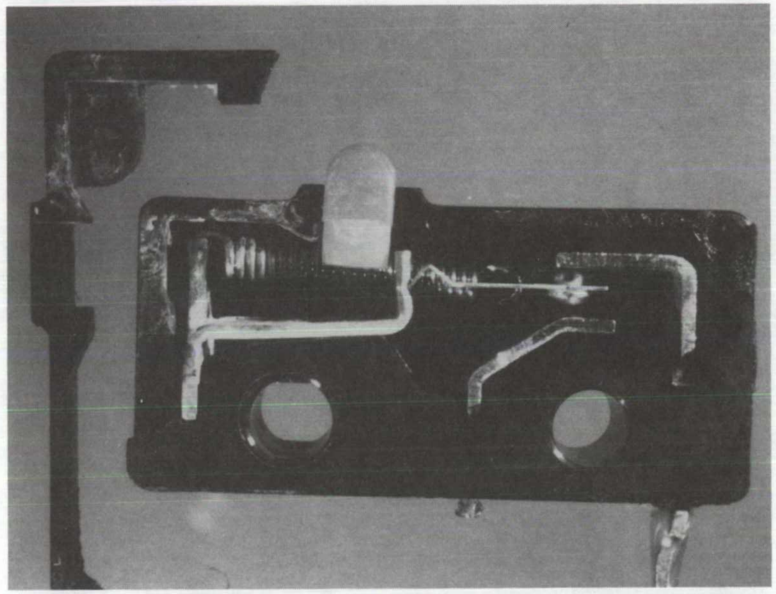


PHOTO 6 Microswitch showing evidence of sea water having entered through a separation in the casing halves.

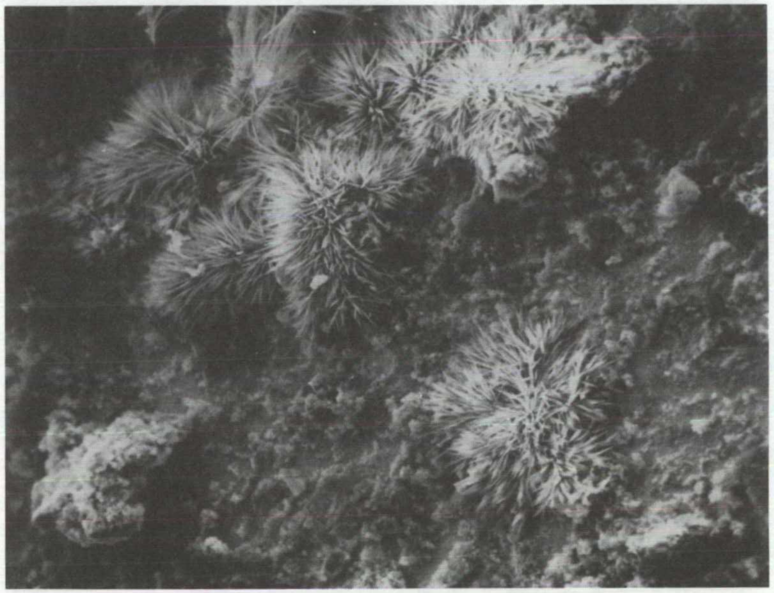


PHOTO 7 Micrograph showing the various crystalline deposits on a switch component.

PHOTO 8 Micrograph details of some typical spikey crystalline deposits.

Item F-4

Analysis and Calculations of the *Ocean Ranger* Ballast Pumping System Capability

SUMMARY

It is very likely that the *Ocean Ranger* experienced ballast control problems on the evening of 14th February 1982 which contributed to the total loss of the unit. The following analysis considers some characteristics of the ballast system and its ability to rectify a forward trim.

Since the pump rooms were situated at the after ends of the pontoons, the suction head required when the vessel is trimming forward was increased to such an extent that at 12°, no pump could lift more than about 35% of the number 1 tanks, and no more than about 10% of tanks 2 and 3. At this angle of trim, this is the capacity at which the static suction head is equal to the vapour pressure of the water being pumped. Figure 13 plots the angle of forward trim against the capacity of tanks 1, 2 and 3 showing the point at which the ballast system becomes inoperable.

The constant-speed pump motors cause cavitation at the impellers when pumping with 1 pump from tanks 2 or 3 separately for all angles of forward trim and at zero trim even when the tanks are nearly full. When drawing from tanks 2 and 3 together using 1 pump, cavitation occurs at all forward trims in excess of about 5° when the tanks are full. At zero trim cavitation will occur at all tank levels below approx. 10 ft. or about 38% of their total capacity. It follows that the pumping of the number 1 tanks will also cause cavitation at all levels below approximately 26.5 ft. or about 65% of their total capacity. The exact relationship between cavitation and pumping rate is not known, though in general the pumping rate will decrease in proportion to the degree of cavitation. Cavitation will increase with increasing suction head levels.

In Figure 11 it can be seen that pumping from tanks 2 and 3 on a one-pump/one-tank basis, the allowable flow rate for no cavitation calculated for zero trim ranges from approximately U.S. 1630 GPM when the tank is empty to approximately U.S. 2750 GPM when the tank is full. Analysing the actual design flow rate, i.e., that rate at which the ballast pumps would operate if there was no constraint due to cavitation, it is calculated that the comparable flow rate would range from approximately U.S. 2325 GPM with the tank empty, to U.S. 2590 GPM with the tank full (see Figure 8).

It is normal practice to design a pumping system for zero impeller cavitation, since operating at or beyond the cavitation point produces noise, vibration and rapid erosion of the impeller and the surrounding metal surfaces. In this respect the ballast system on the *Ocean Ranger* was not designed in accordance with good practice, since the pumping capacity was too high to avoid cavitation when pumping from the forward tanks. The reason was, in part, the long run of relatively small-bore piping from the tanks to the pump rooms. This pipe was near the minimum recommended diameter for the design flow rate of the pumps, and caused a considerable dynamic loss in the suction head.

When operating at the design flow rate of a ballast pump (U.S. 2000 GPM), the total suction head in the suction system is calculated to rise to nearly 27 ft. when pumping empty from tanks 2 and 3 on a single tank basis. However, due to the constant speed characteristic of the pumps, the flow rate would be 2300-2600 GPM and the head loss around 39 ft. Pumps such as those fitted in the *Ocean Ranger* would normally be expected to operate on a total maximum suction head (static + dynamic) of no more than about 20 ft. Consequently, the design rate would not be achieved. The total maximum suction head, pumping from two tanks, would be around 24 ft. at achievable flow rates of approximately 2400-2700 GPM.

Part of the problem in pumping from the forward tanks was due to the location of the pump rooms at the after end of each pontoon. When the rig is trimmed, the static suction head is directly related to the horizontal distance between pump and tank bellmouth. Placing the pump room amidships would reduce this suction head considerably. On the *Ocean Ranger* there would be some difficulty in obtaining access to a midships pump room, but this could be achieved from one of the central columns via a watertight tunnel or passage inside the pontoon tanks. A pump room located at both ends of each pontoon would give a positive static head to at least one pump under any condition of trim. Obviously, this would increase the complexity of pipework and control systems, and the operator could well argue that the unit was not designed to operate in a trimmed condition.

Nevertheless, the *Ocean Ranger* has been shown to be easily trimmed through relatively small transfers of ballast. Once in a considerably trimmed-forward condition it is very difficult, with the configuration as built, to see how the pumping system could rectify the problem unless positive air pressure was applied to the tanks. Furthermore, the pump-room-aft configuration placed the centre of gravity of the tank block well forward of the centre of buoyancy. Thus, for normal operating drafts, the after ballast tanks had to be substantially full at all times which limited the trimming aft capability of the rig.

The ballast system of *Ocean Ranger* was arranged so that all pipes from the ballast tanks led aft to a common manifold in the pump room. This manifold was in turn accessed by the pumps and could be used either to fill ballast tanks from the sea or discharge the contents of the tanks overboard. What was not possible, however, was to pump the contents of one ballast tank to another. Instead, it was necessary to pump the first tank overboard and then fill the corresponding or balancing tank with the appropriate amount of water. This arrangement is considered to be unnecessarily limiting and potentially dangerous in the event of malfunction of any valves.

If the forward tanks had been connected by one manifold and the after tanks by another, it would have been possible to pump ballast between forward and after tanks. Figure 14 shows the arrangement as fitted and Figure 15 a suggested alternative, perhaps not optimum, which would be more flexible and permit transfer rather than discharge and replacement. This lack of internal transfer capability must be considered to be a defect in the ballast piping system.

It is reiterated that, due to cavitation, the capacity of the ballast pumps on the *Ocean Ranger* was too large to empty effectively the forward tanks under conditions of forward trim. Cavitation has been shown to occur even under conditions of zero trim.

CONCLUSIONS

1. The ballast system of *Ocean Ranger* was not totally satisfactory in a number of respects. The main deficiencies are considered to be as follows:

i. It was not possible to pump ballast from a forward tank to an after tank or vice versa. A simple modification to the manifold would have made this possible.

ii. The piping from the forward tanks to the after pump rooms was too small in diameter, in relation to the length of run and the pumping capacity. This resulted in cavitation even at zero trim.

iii. The characteristics of the pumping system and location of the pump room limited the capability of the pumps to deballast the forward tanks with the vessel trimmed by the bow. Indeed, with the crucial tanks 1, 2 and 3 in each pontoon, total suction loss would occur at a trim of 6° with the tanks nearly empty and at a trim of 12° with the tanks virtually full. With a trim in the region of 8-10° the pumps were not able to draw from numbers 1, 2 or 3 tanks if they were less than about 45% full.

2. The design of the pumping system did not follow good practice. The total dynamic suction head loss exceeds acceptable limits when pumping individually at design capacity from tank 2 or 3. It was marginally acceptable when pumping from two forward tanks simultaneously, using a single pump.

3. The location of the pump rooms aft placed the centre of gravity of the tank block well forward of the centre of buoyancy. Consequently, at normal operating drafts, the aft ballast tanks had to be substantially full at all times, limiting the trimming aft capability of the rig.

PERFORMANCE CALCULATION

This section describes the method used to determine the ability of the ballast system on the *Ocean Ranger* to deballast the vessel under conditions of both level trim and at varying degrees of trim by the bow.

Since the ballast pumps were situated in the pump rooms at the aft end of the vessel, it follows that the most arduous pumping conditions were imposed by suction from the forward pontoon tanks, i.e., tanks 1, 2, 3 and 4.

Pumping conditions for tanks 2 and 3 are investigated, including some calculations to extend the application of the method to tanks 1 and 4.

PUMP CHARACTERISTICS

Pump specification: Layne & Bowler 'Veriline' close-coupled axial flow centrifugal pump, driven by U.S. 125 HP three-phase electric motor (constant speed 1770 rpm). Design capacity U.S. 2000 GPM at 170 ft system head.

The output of a ballast pump is dependent upon two factors:

a) Total system head: The head-capacity curve is reproduced in Figure 1. As the head increases so the flow decreases and efficiency drops from the design point.

b) Incidence of cavitation: Any axial flow pump, which is impeller driven, will cavitate at some point, dependent upon the rate of flow and the suction lift required. The positive pressure of water around the impeller, for conditions of no cavitation, and expressed as a head of water, is the Net Positive Suction Head (NPSH). This is defined as the difference between the total suction head (including the dynamic head in the suction line) and the head corresponding to the vapour pressure of the liquid pumped. The NPSH curve for the impeller fitted to the *Ocean Ranger* ballast pumps is reproduced in Figure 2.

The effect of pump impeller cavitation on pumping capability is examined numerically in a later section.

BALLAST SYSTEM

Any pumping system is made up of three components: suction line, pump and discharge line. For each pontoon the *Ocean Ranger* could use either one, two or three pumps in parallel, drawing water from one or more tanks and discharging through a common main.

The function of a pump is basically to lift liquid from one level to a higher one. The difference in these two levels is defined as the static head. Losses in pressure due to friction of piping, valves, etc., is defined as the dynamic head. The sum of these two components is the total system head. Since the characteristics of a pump are different for the suction and discharge, the total system head must be divided into total suction head and total discharge head in order to examine its capabilities.

The total system head will vary on the *Ocean Ranger* according to tank level, which affects the static head. The number of tanks pumped simultaneously, and the corresponding flow rate through the suction lines will vary the total friction loss which affects the dynamic head.

This report details two hypothetical pumping conditions:

a) 1 pump acting on 1 tank (1P/1T)

b) 1 pump acting on 2 tanks simultaneously (1P/2T)

It determines the cavitation point of the pump impeller under varying levels of water in tanks 2 and 3; also flow rates from these tanks under conditions of zero cavitation for the two pumping conditions above.

DETERMINATION OF STATIC HEAD

Under any trim condition, the total static head is measured from the level of water in the tank to the point of overboard discharge.

Overboard discharge level
= 32.00m above baseline in the upright condition.

Height of tank bellmouth above baseline
= 0.09m.

Height of pump suction
= 0.915m. This height is taken as the position of the priming propeller, since the priming propeller must develop enough head to reach the second stage suction in order for the pump to function.

Longitudinal position of Nos. 2 and 3 tank bellmouths
= 800mm forward of frame 7.

Longitudinal position of discharge riser
= 3.548m forward of frame 53.

Distance from tank bellmouth to frame 53
= 69.684m.

Distance from bellmouth to discharge riser
= 69.684 − 3.548 = 66.14m.

If angle of trim by the bow = α

Then the vertical distance between the tank bellmouth and the discharge point (maximum static head), is calculated as:

$h_1 = 66.14 \sin\alpha + (32.00 - 0.09) \cos\alpha$ (metres)

Horizontal position of pump centreline
= 5.60 m. aft of frame 53.

Horizontal distance of Nos. 2 and 3 tank bellmouths to pump centreline
= 5.60 + 69.684 = 75.28m

Then the vertical distance between the tank bellmouth and the pump suction (maximum static suction head), is:

$h_2 = 75.28 \sin\alpha + (0.915 - 0.09) \cos\alpha$ (metres)

Then for tanks 2 and 3:

α Degrees	h_1	h_2	h_3 (metres)
0	31.91	0.83	31.08
2	34.20	3.45	30.75
4	36.45	6.07	30.38
6	38.65	8.69	29.96
8	40.80	11.29	29.51
10	42.91	13.88	29.03
12	44.96	16.46	28.50

The values of h_1 and h_2 represent the static heads in the system when the level of water in the tank is level with the bottom of the bellmouth, i.e., the point at which all suction will cease. At intermediate tank levels the head of water in the tank can be deducted from the maximum static head to give the actual static and static suction head. Curves comparing the head of water in the tank against tank percentage capacity for varying trim angles have been prepared for tanks 1, 2, 3 and 4 (see Figures 3-5). For these graphs the head of water has been calculated with the waterplane in the tank trimmed relative to the baseline for all corresponding angles of trim. The static discharge head h_3 will remain constant at selected angle of trim for all tank levels.

DETERMINATION OF DYNAMIC HEAD

Each of the ballast tanks leads to the forward pump room bulkhead by a single ballast line of 200mm diameter standard grade steel pipe. Aft of the pump room bulkhead the ballast lines are fed through a manifold into a common main which varies in diameter from 250mm to 450mm. The discharge main is generally 400mm diameter, connected to the ballast pump with a short length of 250mm diameter pipe.

Length of suction to pump room bulkhead
= 78.25m. (averaged for tanks 2 and 3).

Constrictions to flow:
1 x Bellmouth
3 x 90° bends (assumed R/r = 6)
14 x 45° bends (" " " ")

Length of nominal 200mm suction in pump room
= 1.8m each for tanks 2 and 3.

Constrictions to flow:
1 x Butterfly valve
1 x Expansion (200mm – 450mm)

Length of nominal 450mm suction in pump room

a) Manifold
= 2.60m (averaged for tanks 2 and 3)

Constrictions to flow:
1 x 90° branch (1P/1T)
2 x 90° branch (1P/2T)

b) Branch to strainer
= 2.40m

Constrictions to flow:
2 x 90° branch
1 x Butterfly valve

Length of nominal 250mm suction in pump room (assumed for pump with shortest route)
= 0.95m

Constrictions to flow:
1 x Contraction (450mm – 250mm)
1 x Strainer
1 x Butterfly valve

Length of nominal 250mm discharge
= 3.9m

Constrictions to flow:
1 x Non-return valve
1 x 90° bend
1 x Expansion (250mm – 400mm)

Length of nominal 400mm discharge
= 49.0m

Constrictions to flow:
1 x 90° branch
9 x 90° bends (assumed R/r = 2)
2 x 45° bends (" " " ")
1 x non-return valve
1 x discharge to atmosphere

FRICTION LOSSES

The pressure loss in a piping system due to pipe friction is generally expressed in the form:

$$\text{Pressure loss} = f.\frac{L}{d}.p.\frac{V^2}{2}\ \text{KN/m}^2$$

where:
f = friction coefficient (dimensionless)
L = length of pipe (m)
d = diameter of pipe (mm)
V = Velocity of water through pipe (m/s)
P = density of liquid (kg/m³)

The friction coefficient is dependent upon Reynold's Number, and for this report has been obtained from the *British Standard; Marine Series Specification for Salt-Water Piping in Ships*, which has also been used (unless stated otherwise) for all the foregoing calculations of head loss due to pipe friction and constrictions to flow.

In the foregoing calculations of dynamic friction losses the internal diameters of all piping have been taken from *Ocean Ranger Ballast System Analysis* by Ralph W. Loomis, an ODECO engineer.

Friction factors used throughout apply to new steel pipes. The friction factor corresponding to the design flow of 2000 gpm has been used for all flow rates under investigation.

The pressure loss due to friction can be equated to a head loss by P=pgh.

$$\text{Head loss } h = f.\frac{L}{d}.\frac{V^2}{2g}$$

1 GPM = 0.0631 litres/sec.

$$\text{and by } V = \frac{Q}{A}$$

$$V = \frac{0.0631 \times Q}{1000 \times \frac{kd^2}{4}} = 8.034 \times 10^{-5} \times \frac{Q}{d^2}\text{m/s}$$

where
Q = flow rate (GPM)
d = internal pipe diameter (m)
h = $3.290 \times 10^{-10} \times \frac{L}{d^5} \times Q^2 \times f$
L = pipe length (m)

All elements of friction loss were calculated on a microcomputer using the above formulation. Input for each element was length, diameter and friction factor. Results are shown in Tables 1-14 for both 1P/1T and 1P/2T combinations.

CONSTRICTIONS TO FLOW

The pressure loss in a piping system due to constrictions is expressed in the form:

$$P = K \times \frac{V^2}{2g} \times \frac{p}{1000}\ \text{KN/m}^2$$

where
K = dimensionless coefficient
V = fluid velocity (m/s)
p = density of fluid (kg/m³)

This can be equated to a head loss, where

$$\text{Head loss } h = K.\frac{V^2}{2g}$$

and since, as shown before,

$$V = 8.034 \times 10^{-5} \times \frac{Q}{d^2}$$

$$\text{then } h = K \times 3.290 \times 10^{-10} \times \frac{Q^2}{d^4}$$

Again each element of constriction loss was calculated by a microcomputer. Input for each element was the constriction coefficient K as set out below, and results are shown in Tables 1-14.

Loss in suction line forward of pump room:

1 x Bellmouth	K = 0.10*
3 x 90° bends	K = 3 x 0.12
14 x 45° bends	K = 14 x 0.07
TOTAL	K = 1.44

*Source: Kempes Engineers Year Book 1977.

Loss in nominal 200mm suction in pump room:

1 x Butterfly valve	K = 0.42
1 expansion 200-450mm	K = 0.65
TOTAL	K = 1.07

Loss in nominal 450mm suction:

a) Manifold 1 x 90° branch	K = 0.90
TOTAL	K = 0.90
b) Branch to strainer	
2 x 90° branches	K = 2 x 0.90
1 x Butterfly valve	K = 0.42
TOTAL	K = 2.22

Loss in nominal 250mm suction:

1 Contraction 400-250mm	K = 0.38*
1 x Strainer	K = 1.31
1 x Butterfly valve	K = 0.42
TOTAL	K = 2.11

*Assumed 1 contraction + 1 expansion + 50% for strainer basket, etc.

Loss in nominal 250mm discharge:

1 x Non-return valve	K = 1.60
1 x 90° bend	K = 0.22
1 expansion 250-450mm	K = 0.38
TOTAL	K = 2.20

Loss in nominal 400mm discharge:

1 x 90° branch	K = 0.90
9 x 90° bends	K = 9 x 0.30
2 x 45° bends	K = 2 x 0.16
1 x Non-return valve	K = 1.60
Discharge to atmosphere	K = 1.00
TOTAL	K = 6.52

TOTAL LOSSES

The total head loss in the system is the summation of friction and constriction loss and is termed the dynamic head of the system. To allow for computation of 1P/2T losses, both the length of pipe and constriction coefficients were doubled in those parts of the system not common to both 1P/1T and 1P/2T combinations. Similarly, in these elements the velocity was halved relative to the flow rate through the pump and common suction/discharge.

The total dynamic head of the system is added to the total static head to give the total system head. Similarly, the dynamic and static heads on the suction side are summed to give the total suction head. The total system head and the total suction head are the two factors affecting the performance of the ballast pump.

The system head is maximum when the water in the tank is at the bellmouth level. From this maximum head may be deducted the levels of liquid in the tanks (see Figures 3-5) at varying capacities and trims, and the resultant net system head compared to the head-capacity curve of the ballast pump to give the design flow rates at various tank levels. Similarly the suction head can be analysed for varying tank levels to determine the cavitation limits on pumping capability.

PUMP CAVITATION

Ultimately, the suction is limited to the point at which the pressure drop is such as to cause the water to vaporize. At a water temperature of 5°C this vapor pressure is approximately 0.9 KN/m². Under stable conditions the pressure at the water surface is equal to atmospheric pressure. At the *Ocean Ranger* site on the night of 14th/15th February 1982 the atmospheric pressure was at its lowest point approximately 975 mb., which is equal to 97.5 KN/m² (1000 mb = 100 KN/m²)

Thus, the maximum pressure loss in suction is equal to 97.5 – 0.9 = 96.6 KN/m²

$$\text{and by the form } h = \frac{P}{\rho g}$$

the equivalent suction head

$$\text{for S.W.} = \frac{96.6 \times 1000}{1025 \times 9.81}$$

$$= 9.61\text{m } (31.53 \text{ ft})$$

As mentioned earlier, any axial flow impeller-driven pump is limited by cavitation to a suction head less than the static vapor pressure of the liquid being pumped. This reduction in head is termed the NPSH (net positive suction head) and bears a relationship to "flow times speed squared" in the form:

$$\frac{\text{litres/sec} \times (\text{rev/min})^2}{\text{NPSH}^{1.5}} = \text{constant}$$

This can also be expressed as:

$$\frac{\text{GPM X rpm}^2}{\text{NPSH}^{1.5}} = \text{constant}$$

Reference to the Layne and Bowler Propeller 10P NPSH curve (reproduced in Figure 2) shows that at 4000 GPM NPSH = approximately 28 ft and at 3000 GPM NPSH = approximately 19 ft.

Using 3000 GPM as the starting point

$$\frac{3000 \times 1770^2}{19^{1.5}} = 1.135 \times 10^8$$

On this basis:

$$\frac{4000 \times 1770^2}{\text{NPSH}^{1.5}} = 1.135 \times 10^8$$

NPSH AT 4000 GPM = 23 ft

This discrepancy in the calculated and actual NPSH figures is thought to be due to the discharge conditions influencing cavitation at high flow rates and the consequent low system heads.

Since further information on NPSH is not available, it is proposed to use NPSH = 19 ft at 3000 GPM as a basis for computing NPSH at lower flow rates using the above relationship and hence computing available suction lift.

GPM	NPSH (m)	Available Suction Lift (m)
1000	2.78	6.83
1250	3.23	6.38
1500	3.65	5.96
1750	4.04	5.57
2000	4.42	5.19
2250	4.78	4.83
2500	5.13	4.48
2750	5.46	4.15
3000	5.79	3.82

Since the flow rate is proportional to the total system head, the flow from the tanks will conform to that relationship up to the point where the required suction lift becomes greater than the available suction lift, i.e., until cavitation occurs.

ANALYSIS OF PUMP CAPABILITY

Tables 1-14 set out the total system and total suction heads for flow rates from 0-3000 GPM at varying bow trims of 0°-12° and at varying tank levels. Tables 1-7 are for 1P/1T pumping and Tables 8-14 for 1P/2T.

The total system head when pumping tanks 2 and 3 was compared graphically with the Layne and Bowler Head Capacity Curve for 1P/1T and 1P/2T combinations at varying tank levels and at forward trim angles. Figures 6 and 7 show these curves for zero trim; similar curves have been prepared for 2° to 12° bow trim. Thus the intersection points give curves of design flow rate vs percentage of total tank capacity for 1P/1T and 1P/2T combinations at all forward trims, reproduced in Figures 8 and 9. As previously stated, these curves are valid only at zero cavitation, i.e., so long as the required suction lift is less than the available suction lift at the point of cavitation of the pump impeller.

In order to determine the cavitation point of the pump impeller, it is necessary to plot the curve of available suction lift for zero cavitation against the curves of required suction lift at varying tank levels. For tanks 2 and 3 at 100% capacity this plot is shown in Figure 10 for 1P/1T and 1P/2T combinations at zero trim. This graph shows that for the 1P/1T combination cavitation will occur at any pump flow rate in excess of 2750 GPM, while the 1P/2T combination will produce pump cavitation under these conditions only at a pump flow rate in excess of the range considered.

Referring back to Figure 8, it can be seen that at zero trim, the 'no cavitation' flow rates through the pump on a 1P/1T combination range through 2325 GPM with the tank empty to 2590 GPM with the tank full. Thus, since this design flow rate is less at full capacity than the flow rate which will cause cavitation, it can be concluded that the pump will not cavitate under these pumping conditions when the tank is full and the rig at zero trim. This also applies to a 1P/2T combination at zero trim, since design flow rates range through 2430 GPM with the tank empty to 2700 GPM with the tank full. In order to determine at which combination of tank capacity and forward trim cavitation is likely to occur, it is necessary to repeat the exercise illustrated in Figure 10 for other tank levels and trims. These calculations are reproduced graphically for all trims in Figures 11 and 12 for both 1P/1T and 1P/2T combinations.

Figures 11a and 12a cross plot the design flow rate curve against the allowable flow curve for tanks 2 and 3 using 1P/1T and 1P/2T combinations at zero trim, zero cavitation. Similar cross plots were prepared for other bow trim angles. Provided that the design flow rate does not exceed the allowable flow rate, cavitation will not occur until the level of water in the tank drops to the intersection of the allowable and design curves. Percentage of tank capacity can be equated to tank level by reference to Figure 4.

Cross plotting in the manner outlined above between Figures 8 and 11, and between Figures 9 and 12, then yields the following results:

A) 1 PUMP/1 TANK

At 0° trim, cavitation will commence when tanks 2 and 3 are at approximately 93% capacity, corresponding to a tank level of approximately 8.1 metres (26.6 ft.).

At and in excess of 2° bow trim cavitation will be present at all tank levels.

B) 1 PUMP/1 TANK

At 0° trim, cavitation will commence when tanks 2 and 3 are at approximately 38% capacity, corresponding to a tank level of approximately 3.1 metres (10.2 ft.)

At 2° bow trim, cavitation commences at approximately 77% capacity, corresponding to a tank level of approximately 5.8 metres (19.0 ft.).

At 4° bow trim, cavitation commences at approx. 94% capacity, corresponding to a tank level of approximately 8.5 metres (27.9 ft.).

At and in excess of 6° bow trim, cavitation will be present at all tank levels.

As can be seen, in order to avoid pump cavitation at all tank levels and zero forward trim, the pump should operate at no more than approximately 1630 GPM with the tank empty on a 1P/1T configuration, and 2030 GPM with the tank empty on a 1P/2T configuration. Owing to the constant-speed nature of the ballast pump motor it is not possible for the pump to achieve these lower pumping rates. The only way of achieving lower pumping rates is to throttle the discharge, producing an artificially large discharge head while leaving the suction head unaltered. Since the pumping rate is proportional to the total system head, this would create the desired effect on pumping rates. No information was available at the time of writing to suggest that any means of throttling the discharge was fitted to the *Ocean Ranger*. On this basis this report concludes that the *Ocean Ranger*'s ballast pumps were of too great a capacity given their location and the piping characteristics. This is true when pumping one tank with one pump, or even two tanks simultaneously. It is normal practice for a ballast pump to be sized according to the required suction lift while retaining some margin above the NPSH for the pump, since operating at or beyond the cavitation point produces noise, vibration and rapid erosion of the surrounding metal surfaces. The net operating suction lift is usually designed to be in the region of 5m to 7m with a safety margin on NPSH of 2m and upwards. Any suction pump will be ultimately limited to the point at which the static (zero flow) suction head is equal to the vapor pressure of the liquid being pumped. For water this has been shown to be a suction head of 9.61m. Figure 13 shows a plot of angle of forward trim against percentage capacity of nos. 1, 2 and 3 tanks beyond which no suction is possible. It should be borne in mind that these curves show ultimate loss of suction at zero flow, and no inference should be drawn that the curves are in any way representative of the actual capability of the *Ocean Ranger* Ballast System, which, even new, would be less than the theoretical figures. The theoretical figures would be even less representative of the system after 6 years in service.

[Editors note: Editorial changes have been made to this report, with the author's approval, to assist in publication.]

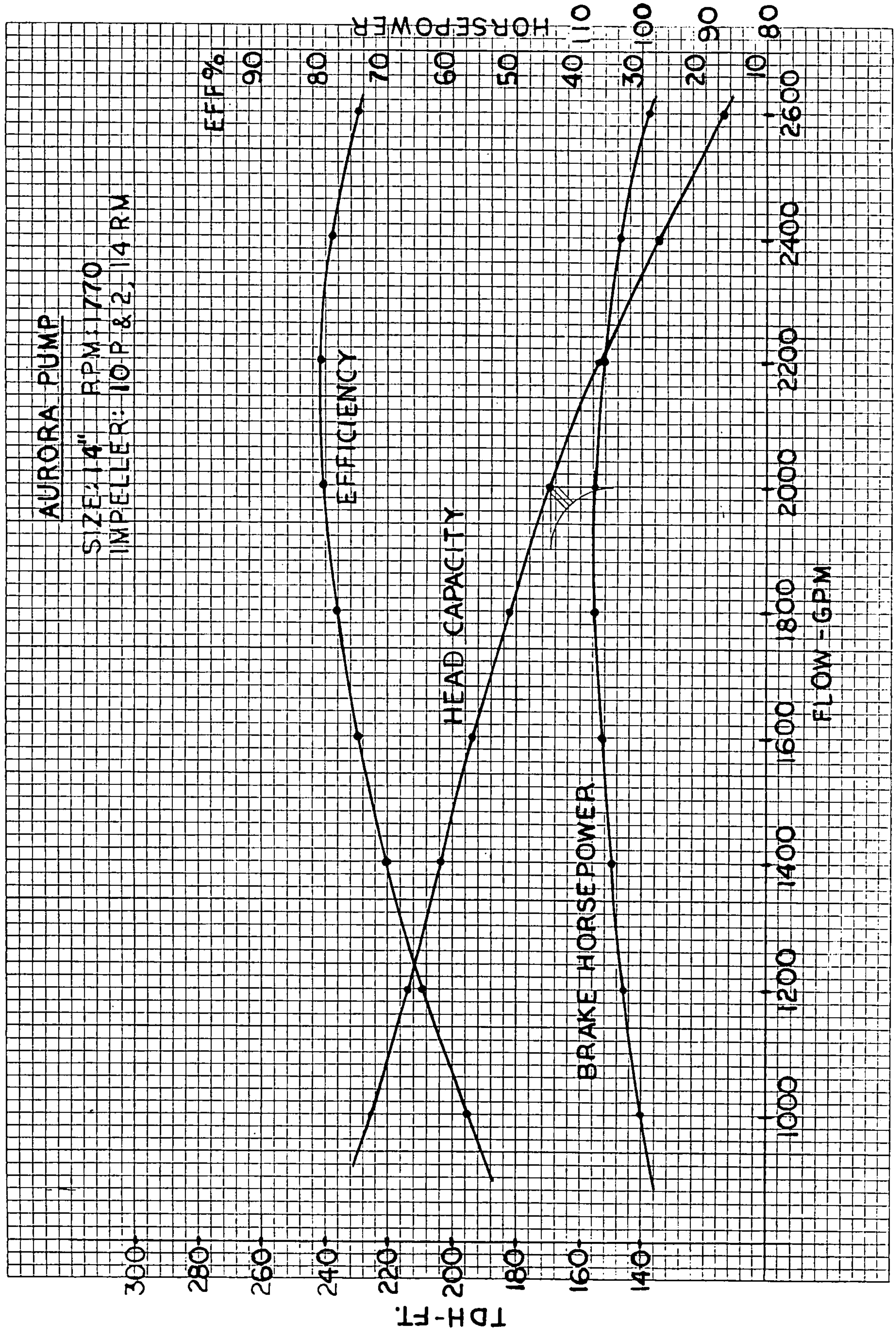

AURORA PUMP
SIZE: 14" RPM: 1770
IMPELLER: 10P & 2, 14RM
EFFICIENCY
HEAD CAPACITY
BRAKE HORSEPOWER
FLOW - GPM
TDH-FT.
HORSEPOWER
EFF %

FIGURE 1

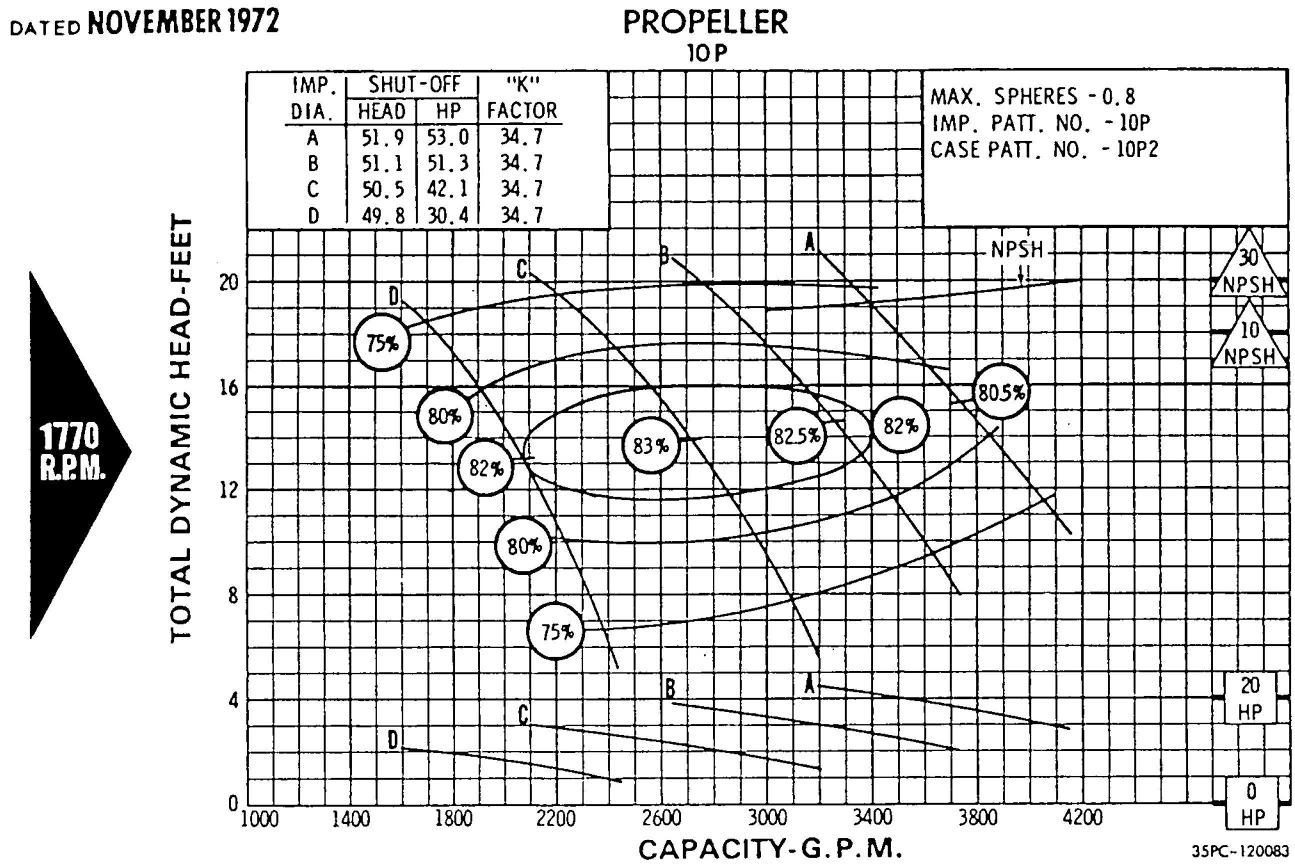


FIGURE 2

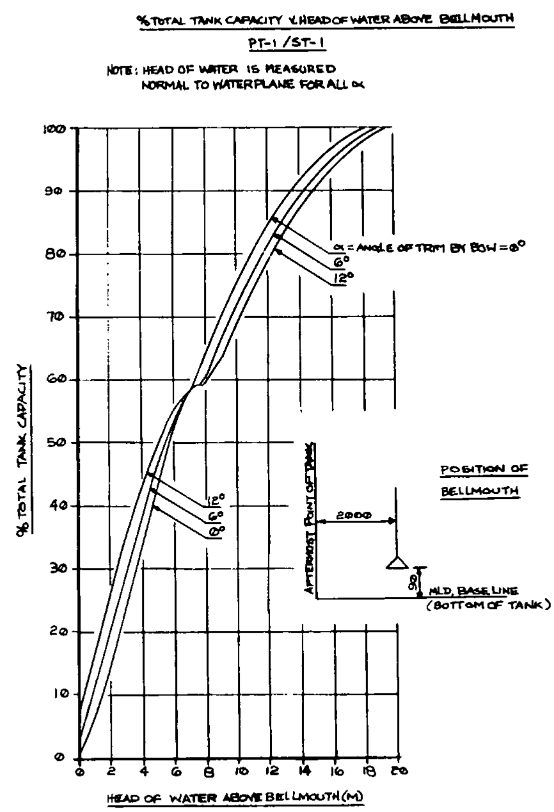


FIG.3

% TOTAL TANK CAP. V. HEAD OF WATER ABOVE BELLMOUTH

PT-2, ST-2/PT-3, ST-3

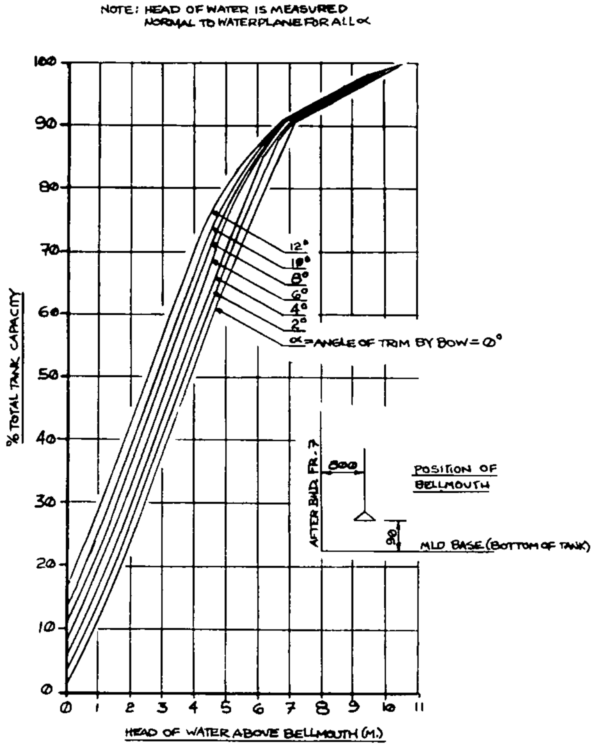


FIG.4

% TOTAL TANK CAPACITY V. HEAD OF WATER ABOVE BELLMOUTH

PT-4, ST-4

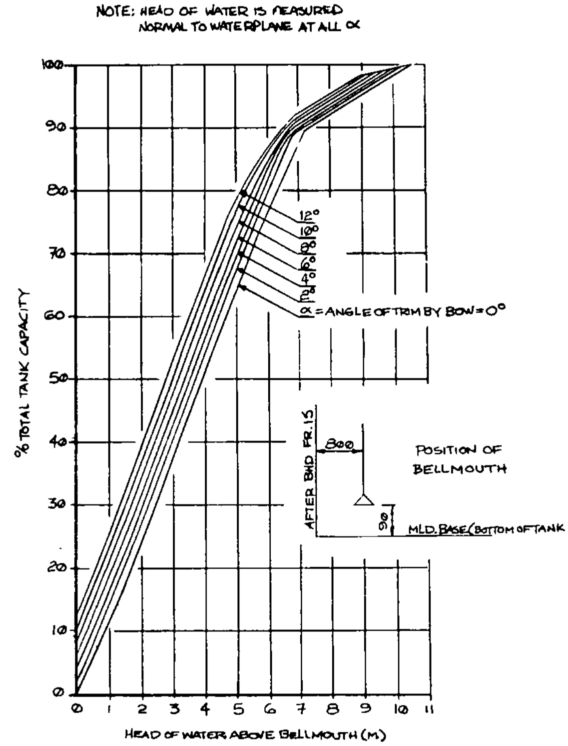


FIG.5

SYSTEM HEAD V. FLOW RATE

NOS. 2 & 3 TANKS

IP/IT

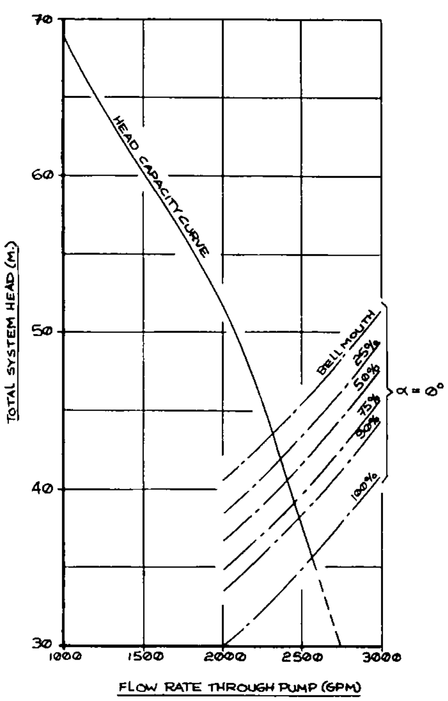


FIG.6

SYSTEM HEAD V. FLOW RATE

NOS. 2 & 3 TANKS

IP/2T

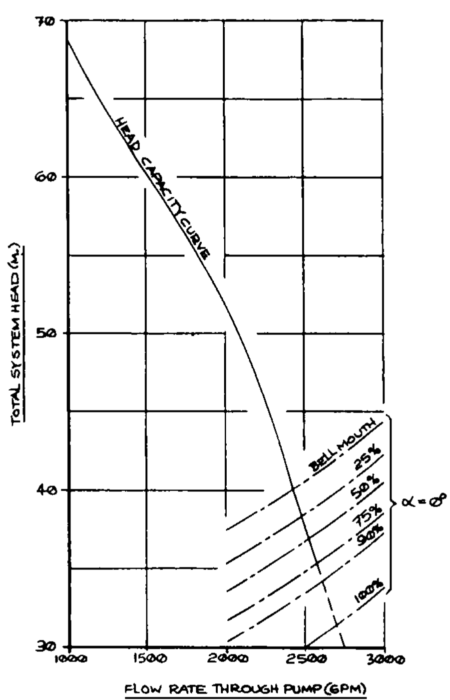


FIG.7

DESIGN FLOW RATES
Nos. 2 & 3 TANKS
IP/IT

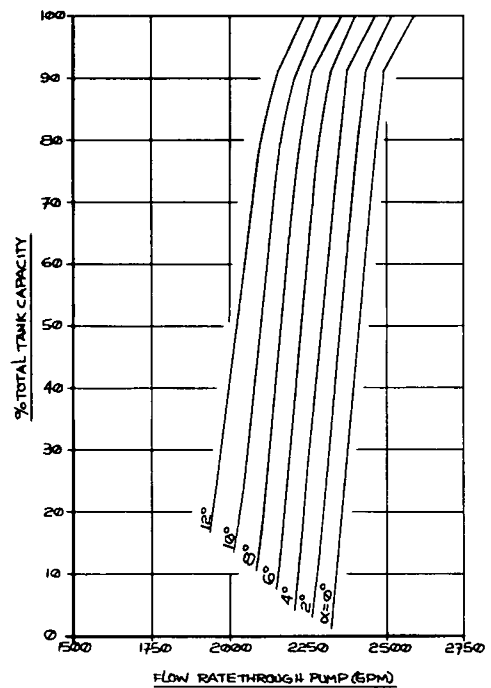


FIG. 8

DESIGN FLOW RATES
Nos. 2 & 3 TANKS
IP/2T

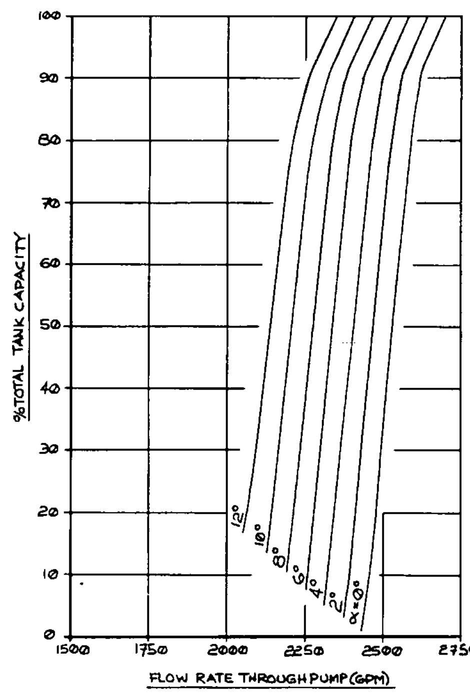


FIG. 9

DETERMINATION OF CAVITATION POINT
PUMPING FROM NOS 2 & 3 TANKS

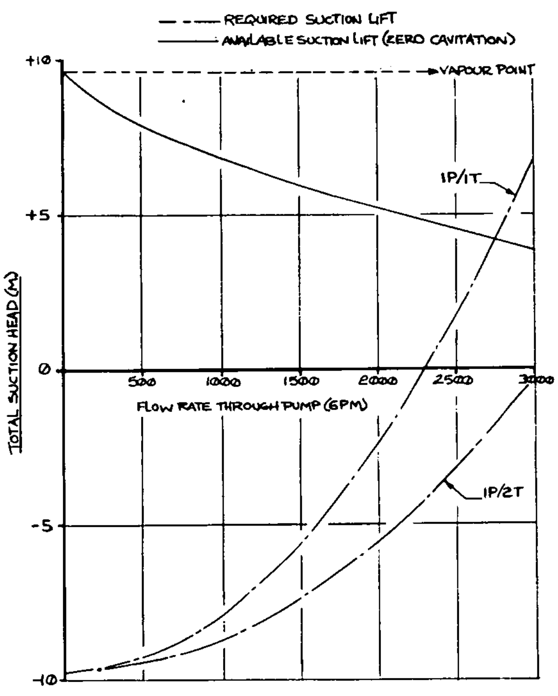


FIG. 10

ALLOWABLE FLOW RATES FOR ZERO CAVITATION
PUMPING FROM Nos. 2 & 3 TANKS
IP/IT

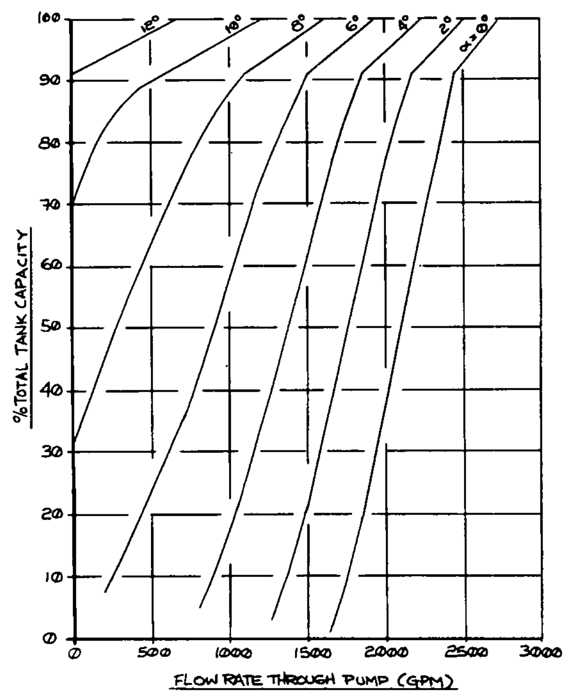


FIG. 11

COMPARISON OF DESIGN AND CAVITATION-FREE FLOW RATES

IP/1T α=0°

PUMPING FROM NOS. 2 & 3 TANKS

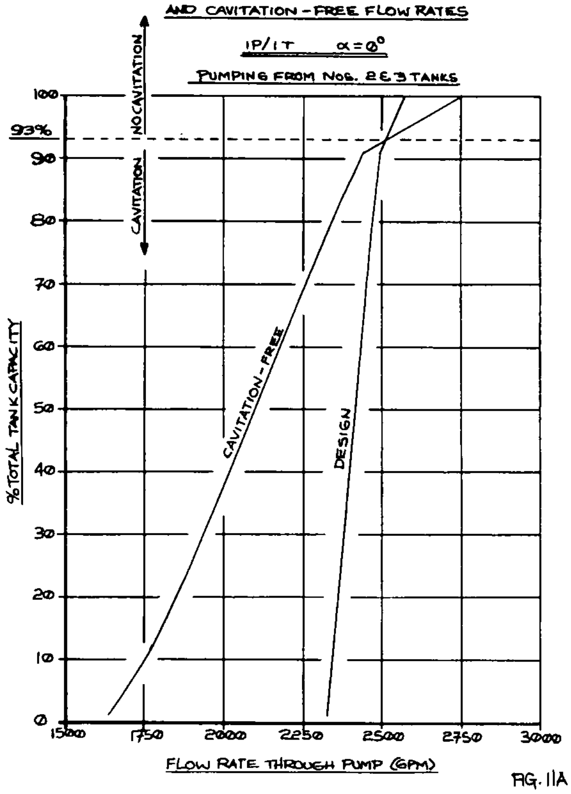


FIG. 11A

ALLOWABLE FLOW RATES FOR ZERO CAVITATION

PUMPING FROM NOS. 2 & 3 TANKS

IP/2T

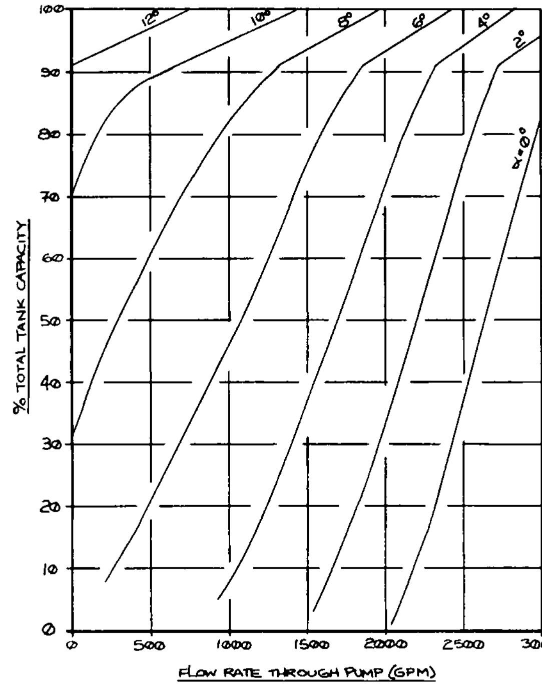


FIG. 12

COMPARISON OF DESIGN AND CAVITATION-FREE FLOW RATES

IP/2T α=0°

PUMPING FROM NOS 2 & 3 TANKS

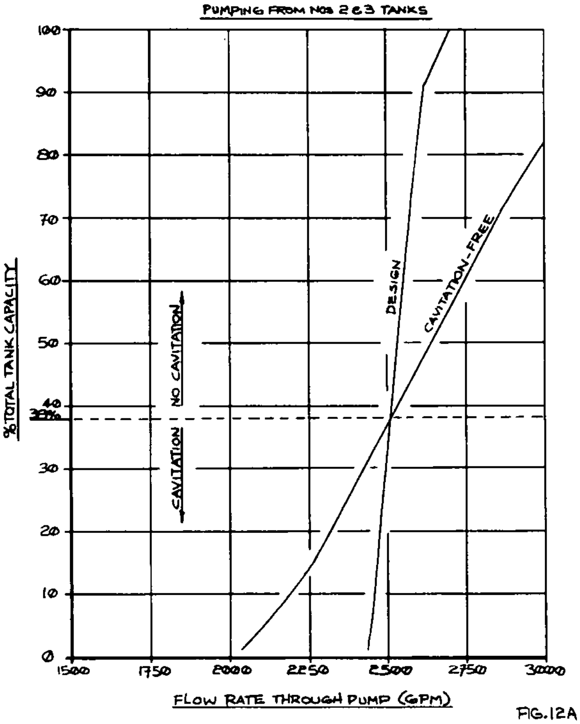


FIG. 12A

CAPACITY OF TANKS BELOW WHICH BALLAST SUCTION IS INOPERABLE

(BASED ON MAX. SUCTION HEAD = 9.61 M.)

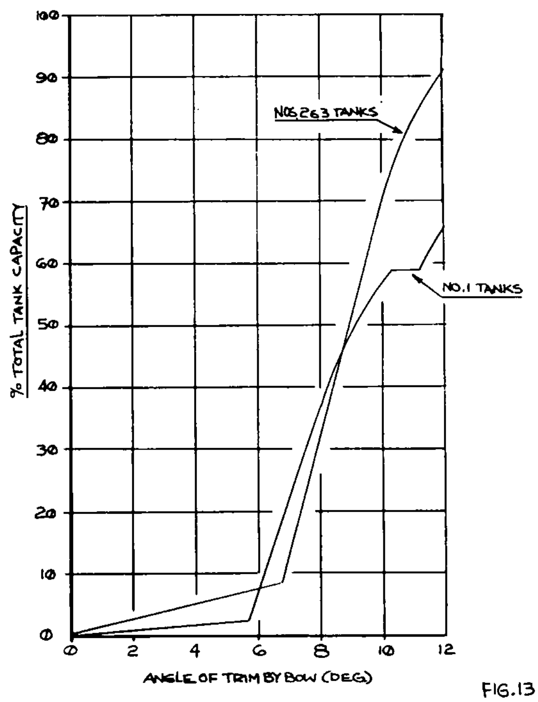


FIG. 13

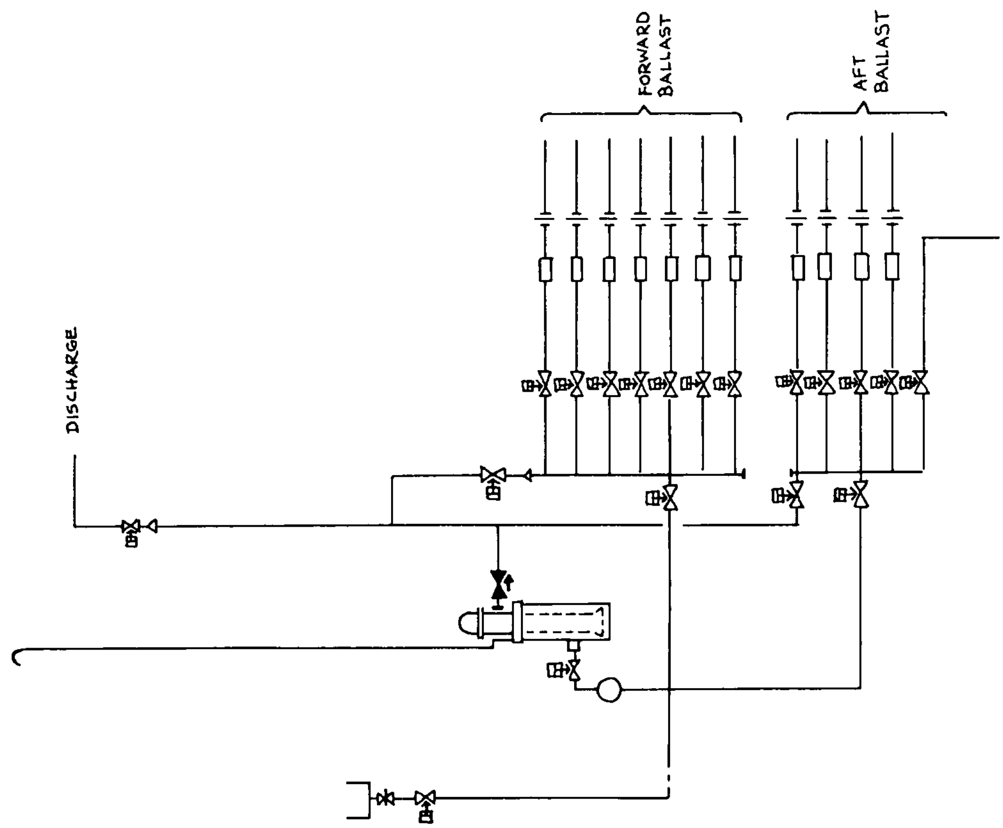


FIGURE 15

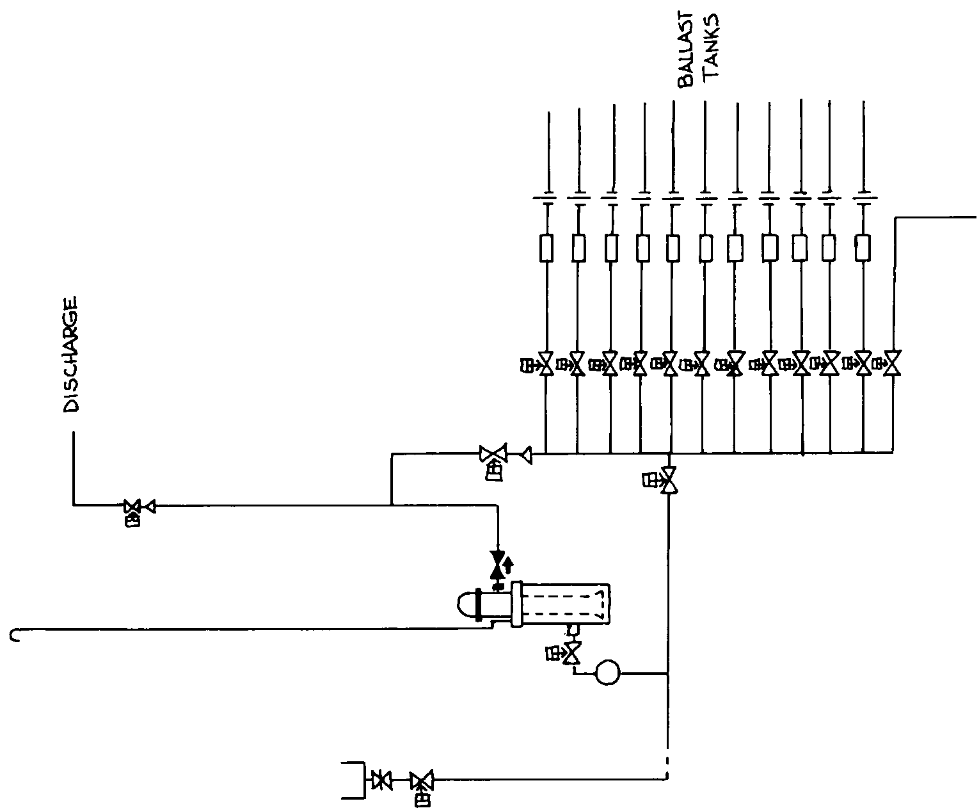


FIGURE 14

TABLE 1

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length (metres)	78.25	1.80	2.60	2.40	0.95	3.90	49.00
diameter (m)	.2047	.2047	.4318	.4318	.254	.254	.381
velocity (m/s)	3.83	3.83	0.86	0.86	2.49	2.49	1.11
reynolds *10^-6	0.50	0.50	0.24	0.24	0.40	0.40	0.27
friction factor	.016	.016	.016	.016	.016	.016	.016
k(f) friction	6.12	0.14	0.10	0.09	0.06	0.25	2.06
k(c)constriction	1.44	1.07	0.90	2.22	2.11	2.20	6.52
k(t) total	7.56	1.21	1.00	2.31	2.17	2.45	8.58
dynamic head (metres)	5.65	0.91	0.04	0.09	0.68	0.77	0.53

static total head = 31.91   static suction head = .825

pumping tank 2 or 3

1 Pump / 1 Tank

Trim in degs. = 0

flow rate q (gpm)	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	31.91	31.91	0.00	0.83	0.83
250	0.14	31.91	32.05	0.12	0.83	0.94
500	0.54	31.91	32.45	0.46	0.83	1.29
750	1.22	31.91	33.13	1.04	0.83	1.86
1000	2.17	31.91	34.08	1.84	0.83	2.67
1250	3.39	31.91	35.30	2.88	0.83	3.70
1500	4.88	31.91	36.79	4.14	0.83	4.97
1750	6.64	31.91	38.55	5.64	0.83	6.46
2000	8.67	31.91	40.58	7.36	0.83	8.19
2250	10.97	31.91	42.88	9.32	0.83	10.14
2500	13.54	31.91	45.45	11.50	0.83	12.33
2750	16.39	31.91	48.30	13.92	0.83	14.75
3000	19.50	31.91	51.41	16.57	0.83	17.39

flow rate q (gpm)	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	31.91	29.81	28.06	26.13	24.79	21.33	0.83	-1.28	-3.03	-4.96	-6.30	-9.76
250	32.05	29.95	28.20	26.27	24.93	21.47	0.94	-1.16	-2.91	-4.84	-6.13	-9.64
500	32.45	30.35	28.60	26.67	25.33	21.87	1.29	-0.81	-2.56	-4.49	-5.83	-9.29
750	33.13	31.03	29.28	27.35	26.01	22.55	1.86	-0.24	-1.99	-3.92	-5.26	-8.72
1000	34.08	31.98	30.23	28.30	26.96	23.50	2.67	0.57	-1.18	-3.11	-4.45	-7.91
1250	35.30	33.20	31.45	29.52	28.18	24.72	3.70	1.60	-0.15	-2.08	-3.42	-6.88
1500	36.79	34.69	32.94	31.01	29.67	26.21	4.97	2.87	1.12	-0.61	-2.15	-5.61
1750	38.55	36.45	34.70	32.77	31.43	27.97	6.46	4.36	2.61	0.68	-0.66	-4.12
2000	40.58	38.48	36.73	34.80	33.46	30.00	8.19	6.09	4.34	2.41	1.07	-2.39
2250	42.88	40.78	39.03	37.10	35.76	32.30	10.14	8.04	6.29	4.36	3.02	-0.44
2500	45.45	43.35	41.60	39.67	38.33	34.87	12.33	10.23	8.48	6.55	5.21	1.75
2750	48.30	46.20	44.45	42.52	41.18	37.72	14.75	12.65	10.90	8.97	7.63	4.17
3000	51.41	49.31	47.56	45.63	44.29	40.63	17.39	15.29	13.54	11.61	10.27	6.81

TABLE 2

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length (metres)	78.25	1.80	2.60	2.40	0.95	3.90	49.00
diameter (m)	.2047	.2047	.4318	.4318	.254	.254	.381
velocity (m/s)	3.83	3.83	0.86	0.86	2.49	2.49	1.11
reynolds *10^-6	0.50	0.50	0.24	0.24	0.40	0.40	0.27
friction factor	.016	.016	.016	.016	.016	.016	.016
k(f) friction	6.12	0.14	0.10	0.09	0.06	0.25	2.06
k(c)constriction	1.44	1.07	0.90	2.22	2.11	2.20	6.52
k(t) total	7.56	1.21	1.00	2.31	2.17	2.45	8.58
dynamic head (metres)	5.65	0.91	0.04	0.09	0.68	0.77	0.53

static total head = 34.1986   static suction head = 3.45154

pumping tank 2 or 3

1 Pump / 1 Tank

Trim in degs. = 2

flow rate q (gpm)	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	34.20	34.20	0.00	3.45	3.45
250	0.14	34.20	34.33	0.12	3.45	3.57
500	0.54	34.20	34.74	0.46	3.45	3.91
750	1.22	34.20	35.42	1.04	3.45	4.49
1000	2.17	34.20	36.37	1.84	3.45	5.29
1250	3.39	34.20	37.58	2.88	3.45	6.33
1500	4.88	34.20	39.07	4.14	3.45	7.59
1750	6.64	34.20	40.84	5.64	3.45	9.09
2000	8.67	34.20	42.87	7.36	3.45	10.81
2250	10.97	34.20	45.17	9.32	3.45	12.77
2500	13.54	34.20	47.74	11.50	3.45	14.96
2750	16.39	34.20	50.59	13.92	3.45	17.37
3000	19.50	34.20	53.70	16.57	3.45	20.02

flow rate q (gpm)	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	34.20	32.31	30.53	28.67	27.20	23.82	3.45	1.56	-0.22	-2.08	-3.55	-6.93
250	34.33	32.44	30.66	28.80	27.33	23.95	3.57	1.68	-0.10	-1.96	-3.43	-6.81
500	34.74	32.85	31.07	29.21	27.74	24.36	3.91	2.02	0.24	-1.62	-3.09	-6.47
750	35.42	33.53	31.75	29.89	28.42	25.04	4.49	2.60	0.82	-1.04	-2.51	-5.89
1000	36.37	34.48	32.70	30.84	29.37	25.99	5.29	3.40	1.62	-0.24	-1.71	-5.09
1250	37.58	35.69	33.91	32.05	30.58	27.20	6.33	4.44	2.66	0.80	-0.67	-4.05
1500	39.07	37.18	35.40	33.54	32.07	28.69	7.59	5.70	3.92	2.06	0.59	-2.79
1750	40.84	38.95	37.17	35.31	33.84	30.46	9.09	7.20	5.42	3.56	2.09	-1.29
2000	42.87	40.98	39.20	37.34	35.87	32.49	10.81	8.92	7.14	5.28	3.81	0.43
2250	45.17	43.28	41.50	39.64	38.17	34.79	12.77	10.88	9.10	7.24	5.77	2.39
2500	47.74	45.85	44.07	42.21	40.74	37.36	14.96	13.07	11.29	9.43	7.96	4.58
2750	50.59	48.70	46.92	45.06	43.59	40.21	17.37	15.48	13.70	11.84	10.37	6.99
3000	53.70	51.81	50.03	48.17	46.70	43.32	20.02	18.13	16.35	14.49	13.02	9.64

TABLE 3

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length (metres)	78.25	1.80	2.60	2.40	0.95	3.90	49.00
diameter (m)	.2047	.2047	.4318	.4318	.254	.254	.381
velocity (m/s)	3.83	3.83	0.86	0.86	2.49	2.49	1.11
reynolds *10^-6	0.50	0.50	0.24	0.24	0.40	0.40	0.27
friction factor	.016	.016	.016	.016	.016	.016	.016
k(f) friction	6.12	0.14	0.10	0.09	0.06	0.25	2.06
k(c)constriction	1.44	1.07	0.90	2.22	2.11	2.20	6.52
k(t) total	7.56	1.21	1.00	2.31	2.17	2.45	8.58
dynamic head (metres)	5.65	0.91	0.04	0.09	0.68	0.77	0.53

flow rate q (gpm)	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	36.45	36.45	0.00	6.07	6.07
250	0.14	36.45	36.58	0.12	6.07	6.19
500	0.54	36.45	36.99	0.46	6.07	6.53
750	1.22	36.45	37.66	1.04	6.07	7.11
1000	2.17	36.45	38.61	1.84	6.07	7.91
1250	3.39	36.45	39.83	2.88	6.07	8.95
1500	4.88	36.45	41.32	4.14	6.07	10.22
1750	6.64	36.45	43.08	5.64	6.07	11.71
2000	8.67	36.45	45.11	7.36	6.07	13.44
2250	10.97	36.45	47.42	9.32	6.07	15.39
2500	13.54	36.45	49.99	11.50	6.07	17.58
2750	16.39	36.45	52.83	13.92	6.07	19.99
3000	19.50	36.45	55.95	16.57	6.07	22.64

static total head = 36.4456   static suction head = 6.07387

pumping tank 2 or 3

1 Pump / 1 Tank

Trim in degs. = 4

flow rate q (gpm)	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	36.45	34.82	32.99	31.13	29.57	26.28	6.07	4.44	2.61	0.75	-0.81	-4.10
250	36.58	34.95	33.12	31.26	29.70	26.41	6.19	4.56	2.73	0.87	-0.69	-3.98
500	36.99	35.36	33.53	31.67	30.11	26.82	6.53	4.90	3.07	1.21	-0.35	-3.64
750	37.66	36.03	34.20	32.34	30.78	27.49	7.11	5.48	3.65	1.79	0.23	-3.06
1000	38.61	36.98	35.15	33.29	31.73	28.44	7.91	6.28	4.45	2.59	1.03	-2.26
1250	39.83	38.20	36.37	34.51	32.95	29.66	8.95	7.32	5.49	3.63	2.07	-1.22
1500	41.32	39.69	37.86	36.00	34.44	31.15	10.22	8.59	6.76	4.90	3.34	0.05
1750	43.08	41.45	39.62	37.76	36.20	32.91	11.71	10.08	8.25	6.39	4.83	1.54
2000	45.11	43.48	41.65	39.79	38.23	34.94	13.44	11.81	9.98	8.12	6.56	3.27
2250	47.42	45.79	43.96	42.10	40.54	37.25	15.39	13.76	11.93	10.07	8.51	5.22
2500	49.99	48.36	46.53	44.67	43.11	39.82	17.58	15.95	14.12	12.26	10.70	7.41
2750	52.83	51.20	49.37	47.51	45.95	42.66	19.99	18.36	16.53	14.67	13.11	9.82
3000	55.95	54.32	52.49	50.63	49.07	45.78	22.64	21.01	19.18	17.32	15.76	12.47

TABLE 4

analysis of ocean ranger ballast system
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calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length (metres)	78.25	1.80	2.60	2.40	0.95	3.90	49.00
diameter (m)	.2047	.2047	.4318	.4318	.254	.254	.381
velocity (m/s)	3.83	3.83	0.86	0.86	2.49	2.49	1.11
reynolds *10^-6	0.50	0.50	0.24	0.24	0.40	0.40	0.27
friction factor	.016	.016	.016	.016	.016	.016	.016
k(f) friction	6.12	0.14	0.10	0.09	0.06	0.25	2.06
k(c)constriction	1.44	1.07	0.90	2.22	2.11	2.20	6.52
k(t) total	7.56	1.21	1.00	2.31	2.17	2.45	8.58
dynamic head (metres)	5.65	0.91	0.04	0.09	0.68	0.77	0.53

flow rate q (gpm)	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	38.65	38.65	0.00	8.69	8.69
250	0.14	38.65	38.78	0.12	8.69	8.80
500	0.54	38.65	39.19	0.46	8.69	9.15
750	1.22	38.65	39.87	1.04	8.69	9.72
1000	2.17	38.65	40.82	1.84	8.69	10.53
1250	3.39	38.65	42.03	2.88	8.69	11.56
1500	4.88	38.65	43.52	4.14	8.69	12.83
1750	6.64	38.65	45.29	5.64	8.69	14.33
2000	8.67	38.65	47.32	7.36	8.69	16.05
2250	10.97	38.65	49.62	9.32	8.69	18.01
2500	13.54	38.65	52.19	11.50	8.69	20.19
2750	16.39	38.65	55.04	13.92	8.69	22.61
3000	19.50	38.65	58.15	16.57	8.69	25.26

static total head = 38.6482   static suction head = 8.68881

pumping tank 2 or 3

1 Pump / 1 Tank

Trim in degs. = 6

flow rate q (gpm)	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	38.65	37.23	35.45	33.62	31.87	28.68	8.69	7.27	5.49	3.66	1.91	-1.28
250	38.78	37.36	35.58	33.75	32.00	28.81	8.80	7.38	5.60	3.77	2.02	-1.17
500	39.19	37.77	35.99	34.16	32.41	29.22	9.15	7.73	5.95	4.12	2.37	-0.82
750	39.87	38.45	36.67	34.84	33.09	29.90	9.72	8.30	6.52	4.69	2.94	-0.25
1000	40.82	39.40	37.62	35.79	34.04	30.85	10.53	9.11	7.33	5.50	3.75	0.56
1250	42.03	40.61	38.83	37.00	35.25	32.06	11.56	10.14	8.36	6.53	4.78	1.59
1500	43.52	42.10	40.32	38.49	36.74	33.55	12.83	11.41	9.63	7.80	6.05	2.86
1750	45.29	43.87	42.09	40.26	38.51	35.32	14.33	12.91	11.13	9.30	7.55	4.36
2000	47.32	45.90	44.12	42.29	40.54	37.35	16.05	14.63	12.85	11.02	9.27	6.05
2250	49.62	48.20	46.42	44.59	42.84	39.65	18.01	16.59	14.81	12.98	11.23	8.04
2500	52.19	50.77	48.99	47.16	45.41	42.22	20.19	18.77	16.99	15.16	13.41	10.22
2750	55.04	53.62	51.84	50.01	48.26	45.07	22.61	21.19	19.41	17.58	15.83	12.64
3000	58.15	56.73	54.95	53.12	51.37	48.18	25.26	23.84	22.06	20.23	16.48	15.29

TABLE 5

analysis of ocean ranger ballast system
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calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length [metres]	78.25	1.80	2.60	2.40	0.95	3.90	49.00
diameter [m]	.2047	.2047	.4318	.4318	.254	.254	.361
velocity [m/s]	3.83	3.83	0.86	0.86	2.49	2.49	1.11
reynolds *10^-6	0.50	0.50	0.24	0.24	0.40	0.40	0.27
friction factor	.016	.016	.016	.016	.016	.016	.016
K[f] friction	6.12	0.14	0.10	0.09	0.06	0.25	2.06
K[c]constriction	1.44	1.07	0.90	2.22	2.11	2.20	6.52
K[t] total	7.56	1.21	1.00	2.31	2.17	2.45	8.58
dynamic head [metres]	5.65	0.91	0.04	0.09	0.68	0.77	0.53

static total head = 40.8037   static suction head = 11.2932

pumping tank 2 or 3

1 Pump / 1 Tank

Trim in degs. = 8

flow rate q [gpm]	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	40.80	40.80	0.00	11.29	11.29
250	0.14	40.80	40.94	0.12	11.29	11.41
500	0.54	40.80	41.35	0.46	11.29	11.75
750	1.22	40.80	42.02	1.04	11.29	12.33
1000	2.17	40.80	42.97	1.84	11.29	13.13
1250	3.39	40.80	44.19	2.88	11.29	14.17
1500	4.88	40.80	45.68	4.14	11.29	15.43
1750	6.64	40.80	47.44	5.64	11.29	16.93
2000	8.67	40.80	49.47	7.36	11.29	18.66
2250	10.97	40.80	51.77	9.32	11.29	20.61
2500	13.54	40.80	54.35	11.50	11.29	22.80
2750	16.39	40.80	57.19	13.92	11.29	25.21
3000	19.50	40.80	60.31	16.57	11.29	27.86

flow rate q [gpm]	total system head bellmouth	25%	50%	75%	90%	100%	suction system head bellmouth	25%	50%	75%	90%	100%
0	40.80	39.61	37.83	35.98	34.12	31.04	11.29	10.10	8.32	6.47	4.61	1.53
250	40.94	39.75	37.97	36.12	34.26	31.18	11.41	10.22	8.44	6.59	4.73	1.65
500	41.35	40.16	38.38	36.53	34.67	31.59	11.75	10.56	8.78	6.93	5.07	1.99
750	42.02	40.83	39.05	37.20	35.34	32.26	12.33	11.14	9.36	7.51	5.65	2.57
1000	42.97	41.78	40.00	38.15	36.29	33.21	13.13	11.94	10.16	8.31	6.45	3.37
1250	44.19	43.00	41.22	39.37	37.51	34.43	14.17	12.98	11.20	9.35	7.49	4.41
1500	45.68	44.49	42.71	40.86	39.00	35.92	15.43	14.24	12.46	10.61	8.75	5.67
1750	47.44	46.25	44.47	42.62	40.76	37.68	16.93	15.74	13.96	12.11	10.25	7.17
2000	49.47	48.28	46.50	44.65	42.79	39.71	18.66	17.47	15.69	13.84	11.98	8.90
2250	51.77	50.58	48.80	46.95	45.09	42.01	20.61	19.42	17.64	15.79	13.93	10.85
2500	54.35	53.16	51.38	49.53	47.67	44.59	22.80	21.61	19.83	17.98	16.12	13.04
2750	57.19	56.00	54.22	52.37	50.51	47.43	25.21	24.02	22.24	20.39	18.53	15.45
3000	60.31	59.12	57.34	55.49	53.63	50.55	27.86	26.67	24.89	23.04	21.18	18.10

TABLE 6

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length [metres]	78.25	1.80	2.60	2.40	0.95	3.90	49.00
diameter [m]	.2047	.2047	.4318	.4318	.254	.254	.381
velocity [m/s]	3.83	3.83	0.86	0.86	2.49	2.49	1.11
reynolds *10^-6	0.50	0.50	0.24	0.24	0.40	0.40	0.27
friction factor	.016	.016	.016	.016	.016	.016	.016
K[f] friction	6.12	0.14	0.10	0.09	0.06	0.25	2.06
K[c]constriction	1.44	1.07	0.90	2.22	2.11	2.20	6.52
K[t] total	7.56	1.21	1.00	2.31	2.17	2.45	8.58
dynamic head [metres]	5.65	0.91	0.04	0.09	0.68	0.77	0.53

static total head = 42.9095   static suction head = 13.6833

pumping tank 2 or 3

1 Pump / 1 Tank

Trim in degs. = 10

flow rate q [gpm]	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	42.91	42.91	0.00	13.88	13.88
250	0.14	42.91	43.04	0.12	13.88	14.00
500	0.54	42.91	43.45	0.46	13.88	14.34
750	1.22	42.91	44.13	1.04	13.88	14.92
1000	2.17	42.91	45.08	1.84	13.88	15.72
1250	3.39	42.91	46.30	2.88	13.88	16.76
1500	4.88	42.91	47.79	4.14	13.88	18.03
1750	6.64	42.91	49.55	5.64	13.88	19.52
2000	8.67	42.91	51.58	7.36	13.88	21.25
2250	10.97	42.91	53.88	9.32	13.88	23.20
2500	13.54	42.91	56.45	11.50	13.88	25.39
2750	16.39	42.91	59.30	13.92	13.88	27.80
3000	19.50	42.91	62.41	16.57	13.88	30.45

flow rate q [gpm]	total system head bellmouth	25%	50%	75%	90%	100%	suction system head bellmouth	25%	50%	75%	90%	100%
0	42.91	41.94	40.14	38.29	36.29	33.35	13.88	12.91	11.11	9.26	7.26	4.32
250	43.04	42.07	40.27	38.42	36.42	33.48	14.00	13.03	11.23	9.38	7.38	4.44
500	43.45	42.48	40.68	38.83	36.83	33.89	14.34	13.37	11.57	9.72	7.72	4.78
750	44.13	43.16	41.36	39.51	37.51	34.57	14.92	13.95	12.15	10.30	8.30	5.36
1000	45.08	44.11	42.31	40.46	38.46	35.52	15.72	14.75	12.95	11.10	9.10	6.16
1250	46.30	45.33	43.53	41.68	39.68	36.74	16.76	15.79	13.99	12.14	10.14	7.20
1500	47.79	46.82	45.02	43.17	41.17	38.23	18.03	17.06	15.26	13.41	11.41	8.47
1750	49.55	48.58	46.78	44.93	42.93	39.99	19.52	18.55	16.75	14.90	12.90	9.96
2000	51.58	50.61	48.81	46.96	44.96	42.02	21.25	20.28	18.48	16.63	14.63	11.69
2250	53.88	52.91	51.11	49.26	47.26	44.32	23.20	22.23	20.43	18.58	16.58	13.64
2500	56.45	55.48	53.68	51.83	49.83	46.89	25.39	24.42	22.62	20.77	18.77	15.83
2750	59.30	58.33	56.53	54.68	52.68	49.74	27.80	26.83	25.03	23.18	21.18	18.24
3000	62.41	61.44	59.64	57.79	55.79	52.85	30.45	29.48	27.68	25.83	23.83	20.89

TABLE 7

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length [metres]	78.25	1.80	2.60	2.40	0.95	3.90	49.00
diameter [m]	.2047	.2047	.4318	.4318	.254	.254	.381
velocity [m/s]	3.83	3.83	0.86	0.86	2.49	2.49	1.11
reynolds *10^-6	0.50	0.50	0.24	0.24	0.40	0.40	0.27
friction factor	.016	.016	.016	.016	.016	.016	.016
K[f] friction	6.12	0.14	0.10	0.09	0.06	0.25	2.06
K[c]constriction	1.44	1.07	0.90	2.22	2.11	2.20	6.52
K[t] total	7.56	1.21	1.00	2.31	2.17	2.45	8.58
dynamic head [metres]	5.65	0.91	0.04	0.09	0.68	0.77	0.53

flow rate q [gpm]	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	44.96	44.96	0.00	16.46	16.46
250	0.14	44.96	45.10	0.12	16.46	16.57
500	0.54	44.96	45.50	0.46	16.46	16.92
750	1.22	44.96	46.18	1.04	16.46	17.49
1000	2.17	44.96	47.13	1.84	16.46	18.30
1250	3.39	44.96	48.35	2.88	16.46	19.33
1500	4.88	44.96	49.84	4.14	16.46	20.60
1750	6.64	44.96	51.60	5.64	16.46	22.09
2000	8.67	44.96	53.63	7.36	16.46	23.82
2250	10.97	44.96	55.93	9.32	16.46	25.78
2500	13.54	44.96	58.51	11.50	16.46	27.96
2750	16.39	44.96	61.35	13.92	16.46	30.38
3000	19.50	44.96	64.47	16.57	16.46	33.02

static total head = 44.9631   static suction head = 16.4574

pumping tank 2 or 3

1 Pump / 1 Tank

Trim in degs. = 12

flow rate q [gpm]	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	44.96	44.26	42.44	40.56	38.41	35.61	16.46	15.76	13.94	12.06	9.91	7.11
250	45.10	44.40	42.58	40.70	38.55	35.75	16.57	15.87	14.05	12.17	10.02	7.22
500	45.50	44.80	42.98	41.10	38.95	36.15	16.92	16.22	14.40	12.52	10.37	7.57
750	46.18	45.48	43.66	41.78	39.63	36.83	17.49	16.79	14.97	13.09	10.94	8.14
1000	47.13	46.43	44.61	42.73	40.58	37.78	18.30	17.60	15.78	13.90	11.75	8.95
1250	48.35	47.65	45.83	43.95	41.80	39.00	19.33	18.63	16.81	14.93	12.78	9.98
1500	49.84	49.14	47.32	45.44	43.29	40.49	20.60	19.90	18.08	16.20	14.05	11.25
1750	51.60	50.90	49.08	47.20	45.05	42.25	22.09	21.39	19.57	17.69	15.54	12.74
2000	53.63	52.93	51.11	49.23	47.08	44.28	23.82	23.12	21.30	19.42	17.27	14.47
2250	55.93	55.23	53.41	51.53	49.38	46.58	25.78	25.08	23.26	21.38	19.23	16.43
2500	58.51	57.81	55.99	54.11	51.96	49.16	27.96	27.26	25.44	23.56	21.41	19.61
2750	61.35	60.65	58.83	56.95	54.80	52.00	30.38	29.68	27.86	25.98	23.83	21.03
3000	64.47	63.77	61.95	60.07	57.92	55.12	33.02	32.32	30.50	28.62	26.47	23.67

TABLE 8

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length [metres]	156.50	3.60	5.20	2.40	0.95	3.90	49.00
diameter [m]	.2047	.2047	.4318	.4318	.254	.254	.381
velocity [m/s]	1.91	1.91	0.43	0.86	2.49	2.49	1.11
reynolds *10^-6	0.25	0.25	0.12	0.24	0.40	0.40	0.27
friction factor	.017	.017	.018	.016	.016	.016	.016
K[f] friction	13.00	0.30	0.22	0.09	0.06	0.25	2.06
K[c]constriction	2.88	2.14	1.80	2.22	2.11	2.20	6.52
K[t] total	15.88	2.44	2.02	2.31	2.17	2.45	8.58
dynamic head [metres]	2.97	0.46	0.02	0.09	0.68	0.77	0.53

flow rate q [gpm]	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	31.91	31.91	0.00	0.83	0.83
250	0.09	31.91	32.00	0.07	0.83	0.89
500	0.34	31.91	32.25	0.26	0.83	1.09
750	0.78	31.91	32.69	0.59	0.83	1.42
1000	1.38	31.91	33.29	1.05	0.83	1.88
1250	2.16	31.91	34.07	1.65	0.83	2.47
1500	3.10	31.91	35.01	2.37	0.83	3.20
1750	4.23	31.91	36.14	3.23	0.83	4.05
2000	5.52	31.91	37.43	4.21	0.83	5.04
2250	6.99	31.91	38.90	5.33	0.83	6.16
2500	8.62	31.91	40.53	6.58	0.83	7.41
2750	10.43	31.91	42.34	7.97	0.83	8.79
3000	12.42	31.91	44.33	9.48	0.83	10.31

static total head = 31.91   static suction head = .825

pumping tanks 2 + 3

1 Pump / 2 Tanks

Trim in degs. = 0

flow rate q [gpm]	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	31.91	29.81	28.06	26.13	24.79	21.33	0.83	-1.28	-3.03	-4.96	-6.30	-9.76
250	32.00	29.90	28.15	26.22	24.88	21.42	0.89	-1.21	-2.96	-4.89	-6.23	-9.69
500	32.25	30.15	28.40	26.47	25.13	21.67	1.09	-1.01	-2.76	-4.69	-6.03	-9.49
750	32.69	30.59	28.84	26.91	25.57	22.11	1.42	-0.68	-2.43	-4.36	-5.70	-9.16
1000	33.29	31.19	29.44	27.51	26.17	22.71	1.88	-0.22	-1.97	-3.90	-5.24	-8.70
1250	34.07	31.97	30.22	28.29	26.95	23.49	2.47	0.37	-1.38	-3.31	-4.65	-8.11
1500	35.01	32.91	31.16	29.23	27.89	24.43	3.20	1.10	-0.65	-2.58	-3.92	-7.38
1750	36.14	34.04	32.29	30.36	29.02	25.56	4.05	1.95	0.20	-1.73	-3.07	-6.53
2000	37.43	35.33	33.58	31.65	30.31	26.85	5.04	2.94	1.19	-0.74	-2.08	-5.54
2250	38.90	36.80	35.05	33.12	31.78	28.32	6.16	4.06	2.31	0.38	-0.96	-4.42
2500	40.53	38.43	36.68	34.75	33.41	29.95	7.41	5.31	3.56	1.63	0.29	-3.17
2750	42.34	40.24	38.49	36.56	35.22	31.76	8.79	6.69	4.94	3.01	1.67	-1.79
3000	44.33	42.23	40.48	38.55	37.21	33.75	10.31	8.21	6.46	4.53	3.19	-0.27

TABLE 9

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length [metres]	156.50	3.60	5.20	2.40	0.95	3.90	49.00
diameter [m]	.2047	.2047	.4318	.4318	.254	.254	.381
velocity [m/s]	1.91	1.91	0.43	0.86	2.49	2.49	1.11
reynolds *10^-6	0.25	0.25	0.12	0.24	0.40	0.40	0.27
friction factor	.017	.017	.018	.016	.016	.016	.016
K[f] friction	13.00	0.30	0.22	0.09	0.06	0.25	2.06
K[c]constriction	2.88	2.14	1.80	2.22	2.11	2.20	6.52
K[t] total	15.88	2.44	2.02	2.31	2.17	2.45	8.58
dynamic head [metres]	2.97	0.46	0.02	0.09	0.68	0.77	0.53

flow rate q [gpm]	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	34.20	34.20	0.00	3.45	3.45
250	0.09	34.20	34.28	0.07	3.45	3.52
500	0.34	34.20	34.54	0.26	3.45	3.71
750	0.78	34.20	34.97	0.59	3.45	4.04
1000	1.38	34.20	35.58	1.05	3.45	4.50
1250	2.16	34.20	36.35	1.65	3.45	5.10
1500	3.10	34.20	37.30	2.37	3.45	5.82
1750	4.23	34.20	38.42	3.23	3.45	6.68
2000	5.52	34.20	39.72	4.21	3.45	7.67
2250	6.99	34.20	41.18	5.33	3.45	8.78
2500	8.62	34.20	42.82	6.58	3.45	10.04
2750	10.43	34.20	44.63	7.97	3.45	11.42
3000	12.42	34.20	46.62	9.48	3.45	12.93

static total head = 34.1986   static suction head = 3.45154

pumping tanks 2 + 3

1 Pump / 2 Tanks

Trim in degs. = 2

flow rate q [gpm]	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	34.20	32.31	30.53	28.67	27.20	23.82	3.45	1.56	-0.22	-2.08	-3.55	-6.93
250	34.28	32.39	30.61	28.75	27.28	23.90	3.52	1.63	-0.15	-2.01	-3.48	-6.86
500	34.54	32.65	30.87	29.01	27.54	24.16	3.71	1.82	0.04	-1.82	-3.29	-6.67
750	34.97	33.08	31.30	29.44	27.97	24.59	4.04	2.15	0.37	-1.49	-2.96	-6.34
1000	35.58	33.69	31.91	30.05	28.58	25.20	4.50	2.61	0.83	-1.03	-2.50	-5.88
1250	36.35	34.46	32.68	30.82	29.35	25.97	5.10	3.21	1.43	-0.43	-1.90	-5.28
1500	37.30	35.41	33.63	31.77	30.30	26.92	5.82	3.93	2.15	0.29	-1.18	-4.56
1750	38.42	36.53	34.75	32.89	31.42	28.04	6.68	4.79	3.01	1.15	-0.32	-3.70
2000	39.72	37.83	36.05	34.19	32.72	29.34	7.67	5.78	4.00	2.14	0.67	-2.71
2250	41.18	39.29	37.51	35.65	34.18	30.80	8.78	6.89	5.11	3.25	1.78	-1.60
2500	42.82	40.93	39.15	37.29	35.82	32.44	10.04	8.15	6.37	4.51	3.04	-0.34
2750	44.63	42.74	40.96	39.10	37.63	34.25	11.42	9.53	7.75	5.89	4.42	1.04
3000	46.62	44.73	42.95	41.09	39.62	36.24	12.93	11.04	9.26	7.40	5.93	2.55

TABLE 10

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length [metres]	156.50	3.60	5.20	2.40	0.95	3.90	49.00
diameter [m]	.2047	.2047	.4318	.4318	.254	.254	.381
velocity [m/s]	1.91	1.91	0.43	0.86	2.49	2.49	1.11
reynolds *10^-6	0.25	0.25	0.12	0.24	0.40	0.40	0.27
friction factor	.017	.017	.018	.016	.016	.016	.016
K[f] friction	13.00	0.30	0.22	0.09	0.06	0.25	2.06
K[c]constriction	2.88	2.14	1.80	2.22	2.11	2.20	6.52
K[t] total	15.88	2.44	2.02	2.31	2.17	2.45	8.58
dynamic head [metres]	2.97	0.46	0.02	0.09	0.68	0.77	0.53

flow rate q [gpm]	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	36.45	36.45	0.00	6.07	6.07
250	0.09	36.45	36.53	0.07	6.07	6.14
500	0.34	36.45	36.79	0.26	6.07	6.34
750	0.78	36.45	37.22	0.59	6.07	6.67
1000	1.38	36.45	37.83	1.05	6.07	7.13
1250	2.16	36.45	38.60	1.65	6.07	7.72
1500	3.10	36.45	39.55	2.37	6.07	8.44
1750	4.23	36.45	40.67	3.23	6.07	9.30
2000	5.52	36.45	41.96	4.21	6.07	10.29
2250	6.99	36.45	43.43	5.33	6.07	11.41
2500	8.62	36.45	45.07	6.58	6.07	12.66
2750	10.43	36.45	46.88	7.97	6.07	14.04
3000	12.42	36.45	48.86	9.48	6.07	15.55

static total head = 36.4456   static suction head = 6.07387

pumping tanks 2 + 3

1 Pump / 2 Tanks

Trim in degs. = 4

flow rate q [gpm]	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	36.45	34.82	32.99	31.13	29.57	26.28	6.07	4.44	2.61	0.75	-0.81	-4.10
250	36.53	34.90	33.07	31.21	29.65	26.36	6.14	4.51	2.68	0.82	-0.74	-4.03
500	36.79	35.16	33.33	31.47	29.91	26.62	6.34	4.71	2.88	1.02	-0.54	-3.83
750	37.22	35.59	33.76	31.90	30.34	27.05	6.67	5.04	3.21	1.35	-0.21	-3.50
1000	37.83	36.20	34.37	32.51	30.95	27.66	7.13	5.50	3.67	1.81	0.25	-3.04
1250	38.60	36.97	35.14	33.28	31.72	28.43	7.72	6.09	4.26	2.40	0.84	-2.45
1500	39.55	37.92	36.09	34.23	32.67	29.38	8.44	6.81	4.98	3.12	1.56	-1.73
1750	40.67	39.04	37.21	35.35	33.79	30.50	9.30	7.67	5.84	3.98	2.42	-0.87
2000	41.96	40.33	38.50	36.64	35.08	31.79	10.29	8.66	6.83	4.97	3.41	0.12
2250	43.43	41.80	39.97	38.11	36.55	33.26	11.41	9.78	7.95	6.09	4.53	1.24
2500	45.07	43.44	41.61	39.75	38.19	34.90	12.66	11.03	9.20	7.34	5.78	2.49
2750	46.88	45.25	43.42	41.56	40.00	36.71	14.04	12.41	10.58	8.72	7.16	3.37
3000	48.86	47.23	45.40	43.54	41.98	38.69	15.55	13.92	12.09	10.23	8.67	5.38

TABLE 11

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length (metres)	156.50	3.60	5.20	2.40	0.95	3.90	49.00
diameter (m)	.2047	.2047	.4318	.4318	.254	.254	.381
velocity (m/s)	1.91	1.91	0.43	0.86	2.49	2.49	1.11
reynolds *10^-6	0.25	0.25	0.12	0.24	0.40	0.40	0.27
friction factor	.017	.017	.018	.016	.016	.016	.016
K(f) friction	13.00	0.30	0.22	0.09	0.06	0.25	2.06
K(c)constriction	2.88	2.14	1.80	2.22	2.11	2.20	6.52
K(t) total	15.88	2.44	2.02	2.31	2.17	2.45	8.58
dynamic head (metres)	2.97	0.46	0.02	0.09	0.68	0.77	0.53

flow rate q (gpm)	:	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	:	0.00	38.65	38.65	0.00	8.69	8.69
250	:	0.09	38.65	38.73	0.07	8.69	8.75
500	:	0.34	38.65	38.99	0.26	8.69	8.95
750	:	0.78	38.65	39.42	0.59	8.69	9.28
1000	:	1.38	38.65	40.03	1.05	8.69	9.74
1250	:	2.16	38.65	40.80	1.65	8.69	10.33
1500	:	3.10	38.65	41.75	2.37	8.69	11.06
1750	:	4.23	38.65	42.87	3.23	8.69	11.91
2000	:	5.52	38.65	44.17	4.21	8.69	12.90
2250	:	6.99	38.65	45.63	5.33	8.69	14.02
2500	:	8.62	38.65	47.27	6.58	8.69	15.27
2750	:	10.43	38.65	49.08	7.97	8.69	16.66
3000	:	12.42	38.65	51.07	9.48	8.69	18.17

static total head = 38.6482   static suction head = 8.68881

pumping tanks 2 + 3

1 Pump / 2 Tanks

Trim in degs. = 6

flow rate q (gpm)	:	total system head: bellmouth	25%	50%	75%	90%	100%	:	suction system head: bellmouth	25%	50%	75%	90%	100%
0	:	38.65	37.23	35.45	33.62	31.87	28.68	:	8.69	7.27	5.49	3.66	1.91	-1.28
250	:	38.73	37.31	35.53	33.70	31.95	28.76	:	8.75	7.33	5.55	3.72	1.97	-1.22
500	:	38.99	37.57	35.79	33.96	32.21	29.02	:	8.95	7.53	5.75	3.92	2.17	-1.02
750	:	39.42	38.00	36.22	34.39	32.64	29.45	:	9.28	7.86	6.08	4.25	2.50	-0.69
1000	:	40.03	38.61	36.83	35.00	33.25	30.06	:	9.74	8.32	6.54	4.71	2.96	-0.23
1250	:	40.80	39.38	37.60	35.77	34.02	30.83	:	10.33	8.91	7.13	5.30	3.55	0.36
1500	:	41.75	40.33	38.55	36.72	34.97	31.78	:	11.06	9.64	7.86	6.03	4.28	1.97
1750	:	42.87	41.45	39.67	37.84	36.09	32.90	:	11.91	10.49	8.71	6.88	5.13	1.94
2000	:	44.17	42.75	40.97	39.14	37.39	34.20	:	12.90	11.48	9.70	7.87	6.12	2.93
2250	:	45.63	44.21	42.43	40.60	38.85	35.66	:	14.02	12.60	10.82	8.99	7.24	4.05
2500	:	47.27	45.85	44.07	42.24	40.49	37.30	:	15.27	13.85	12.07	10.24	8.49	5.30
2750	:	49.08	47.66	45.88	44.05	42.30	39.11	:	16.66	15.24	13.46	11.63	9.88	6.69
3000	:	51.07	49.65	47.87	46.04	44.29	41.10	:	18.17	16.75	14.97	13.14	11.39	8.20

TABLE 12

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads   flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length (metres)	156.50	3.60	5.20	2.40	0.95	3.90	49.00
diameter (m)	.2047	.2047	.4318	.4318	.254	.254	.381
velocity (m/s)	1.91	1.91	0.43	0.86	2.49	2.49	1.11
reynolds *10^-6	0.25	0.25	0.12	0.24	0.40	0.40	0.27
friction factor	.017	.017	.018	.016	.016	.016	.016
K(f) friction	13.00	0.30	0.22	0.09	0.06	0.25	2.06
K(c)constriction	2.88	2.14	1.80	2.22	2.11	2.20	6.52
K(t) total	15.88	2.44	2.02	2.31	2.17	2.45	8.58
dynamic head (metres)	2.97	0.46	0.02	0.09	0.68	0.77	0.53

flow rate q (gpm)	:	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	:	0.00	40.80	40.80	0.00	11.29	11.29
250	:	0.09	40.80	40.89	0.07	11.29	11.36
500	:	0.34	40.80	41.15	0.26	11.29	11.56
750	:	0.78	40.80	41.58	0.59	11.29	11.89
1000	:	1.38	40.80	42.18	1.05	11.29	12.35
1250	:	2.16	40.80	42.96	1.65	11.29	12.94
1500	:	3.10	40.80	43.91	2.37	11.29	13.66
1750	:	4.23	40.80	45.03	3.23	11.29	14.52
2000	:	5.52	40.80	46.32	4.21	11.29	15.51
2250	:	6.99	40.80	47.79	5.33	11.29	16.63
2500	:	8.62	40.80	49.43	6.58	11.29	17.88
2750	:	10.43	40.80	51.24	7.97	11.29	19.26
3000	:	12.42	40.80	53.22	9.48	11.29	20.77

static total head = 40.8037   static suction head = 11.2932

pumping tanks 2 + 3

1 Pump / 2 Tanks

Trim in degs. = 8

flow rate q (gpm)	:	total system head: bellmouth	25%	50%	75%	90%	100%	:	suction system head: bellmouth	25%	50%	75%	90%	100%
0	:	40.80	39.61	37.83	35.98	34.12	31.04	:	11.29	10.10	8.32	6.47	4.61	1.53
250	:	40.89	39.70	37.92	36.07	34.21	31.13	:	11.36	10.17	8.39	6.54	4.69	1.60
500	:	41.15	39.96	38.18	36.33	34.47	31.39	:	11.56	10.37	8.59	6.74	4.89	1.80
750	:	41.58	40.39	38.61	36.76	34.90	31.82	:	11.89	10.70	8.92	7.07	5.21	2.13
1000	:	42.18	40.99	39.21	37.36	35.50	32.42	:	12.35	11.16	9.38	7.53	5.67	2.59
1250	:	42.96	41.77	39.99	38.14	36.28	33.20	:	12.94	11.75	9.97	8.12	6.26	3.18
1500	:	43.91	42.72	40.94	39.09	37.23	34.15	:	13.66	12.47	10.69	8.84	6.98	3.90
1750	:	45.03	43.84	42.06	40.21	38.35	35.27	:	14.52	13.33	11.55	9.70	7.84	4.76
2000	:	46.32	45.13	43.35	41.50	39.64	36.56	:	15.51	14.32	12.54	10.69	8.83	5.75
2250	:	47.79	46.60	44.82	42.97	41.11	38.03	:	16.63	15.44	13.66	11.81	9.95	6.87
2500	:	49.43	48.24	46.46	44.61	42.75	39.67	:	17.88	16.69	14.91	13.06	11.20	8.12
2750	:	51.24	50.05	48.27	46.42	44.56	41.48	:	19.26	18.07	16.29	14.44	12.58	9.50
3000	:	53.22	52.03	50.25	48.40	46.54	43.46	:	20.77	19.58	17.80	15.95	14.09	11.01

TABLE 13

analysis of ocean ranger ballast system
****

calculation of total system and suction system heads flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length (metres)	156.50	3.60	5.20	2.40	0.95	3.90	49.00
diameter (m)	.2047	.2047	.4318	.4318	.254	.254	.381
velocity (m/s)	1.91	1.91	0.43	0.86	2.49	2.49	1.11
reynolds *10^-6	0.25	0.25	0.12	0.24	0.40	0.40	0.27
friction factor	.017	.017	.018	.016	.016	.016	.016
K(f) friction	13.00	0.30	0.22	0.09	0.06	0.25	2.06
K(c)constriction	2.88	2.14	1.80	2.22	2.11	2.20	6.52
K(t) total	15.88	2.44	2.02	2.31	2.17	2.45	8.58
dynamic head (metres)	2.97	0.46	0.02	0.09	0.68	0.77	0.53

flow rate q (gpm)	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	42.91	42.91	0.00	13.88	13.88
250	0.09	42.91	43.00	0.07	13.88	13.95
500	0.34	42.91	43.25	0.26	13.88	14.15
750	0.78	42.91	43.69	0.59	13.88	14.48
1000	1.38	42.91	44.29	1.05	13.88	14.94
1250	2.16	42.91	45.07	1.65	13.88	15.53
1500	3.10	42.91	46.01	2.37	13.88	16.25
1750	4.23	42.91	47.14	3.23	13.88	17.11
2000	5.52	42.91	48.43	4.21	13.88	18.10
2250	6.99	42.91	49.89	5.33	13.88	19.22
2500	8.62	42.91	51.53	6.58	13.88	20.47
2750	10.43	42.91	53.34	7.97	13.88	21.85
3000	12.42	42.91	55.33	9.48	13.88	23.36

static total head = 42.9095

static suction head = 13.8838

pumping tanks 2 + 3

1 Pump / 2 Tanks

Trim in degs. = 10

flow rate q (gpm)	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	42.91	41.94	40.14	38.29	36.29	33.35	13.88	12.91	11.11	9.26	7.26	4.32
250	43.00	42.03	40.23	38.38	36.38	33.44	13.95	12.98	11.18	9.33	7.33	4.39
500	43.25	42.28	40.48	38.63	36.63	33.69	14.15	13.18	11.38	9.53	7.53	4.59
750	43.69	42.72	40.92	39.07	37.07	34.13	14.48	13.51	11.71	9.86	7.86	4.92
1000	44.29	43.32	41.52	39.67	37.67	34.73	14.94	13.97	12.17	10.32	8.32	5.38
1250	45.07	44.10	42.30	40.45	38.45	35.51	15.53	14.56	12.76	10.91	8.91	5.97
1500	46.01	45.04	43.24	41.39	39.39	36.45	16.25	15.28	13.48	11.63	9.63	6.69
1750	47.14	46.17	44.37	42.52	40.52	37.58	17.11	16.14	14.34	12.49	10.49	7.55
2000	48.43	47.46	45.66	43.81	41.81	38.87	18.10	17.13	15.33	13.48	11.48	8.54
2250	49.89	48.92	47.12	45.27	43.27	40.33	19.22	18.25	16.45	14.60	12.60	9.66
2500	51.53	50.56	48.76	46.91	44.91	41.97	20.47	19.50	17.70	15.85	13.85	10.91
2750	53.34	52.37	50.57	48.72	46.72	43.78	21.85	20.88	19.08	17.23	15.23	12.29
3000	55.33	54.36	52.56	50.71	48.71	45.77	23.36	22.39	20.59	18.74	16.74	13.80

TABLE 14

analysis of ocean ranger ballast system
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calculation of total system and suction system heads flow rate = 2000 gpm

description of pipe system	suction in pontoon	suction in pumproom	suction in manifold	suction in pumproom	suction in pumproom	discharge	discharge
length (metres)	156.50	3.60	5.20	2.40	0.95	3.90	49.00
diameter (m)	.2047	.2047	.4318	.4318	.254	.254	.381
velocity (m/s)	1.91	1.91	0.43	0.86	2.49	2.49	1.11
reynolds *10^-6	0.25	0.25	0.12	0.24	0.40	0.40	0.27
friction factor	.017	.017	.018	.016	.016	.016	.016
K(f) friction	13.00	0.30	0.22	0.09	0.06	0.25	2.06
K(c)constriction	2.88	2.14	1.80	2.22	2.11	2.20	6.52
K(t) total	15.88	2.44	2.02	2.31	2.17	2.45	8.56
dynamic head (metres)	2.97	0.46	0.02	0.09	0.68	0.77	0.53

flow rate q (gpm)	total dynamic head m	total static head m	total system head m	suction dynamic head m	suction static head m	suction system head m
0	0.00	44.96	44.96	0.00	16.46	16.46
250	0.09	44.96	45.05	0.07	16.46	16.52
500	0.34	44.96	45.31	0.26	16.46	16.72
750	0.78	44.96	45.74	0.59	16.46	17.05
1000	1.38	44.96	46.34	1.05	16.46	17.51
1250	2.16	44.96	47.12	1.65	16.46	18.10
1500	3.10	44.96	48.07	2.37	16.46	18.83
1750	4.23	44.96	49.19	3.23	16.46	19.68
2000	5.52	44.96	50.48	4.21	16.46	20.67
2250	6.99	44.96	51.95	5.33	16.46	21.79
2500	8.62	44.96	53.59	6.58	16.46	23.04
2750	10.43	44.96	55.40	7.97	16.46	24.42
3000	12.42	44.96	57.38	9.48	16.46	25.94

static total head = 44.9631

static suction head = 16.4574

pumping tanks 2 + 3

1 Pump / 2 Tanks

Trim in degs. = 12

flow rate q (gpm)	total system head: bellmouth	25%	50%	75%	90%	100%	suction system head: bellmouth	25%	50%	75%	90%	100%
0	44.96	44.26	42.44	40.56	38.41	35.61	16.46	15.76	13.94	12.06	9.91	7.11
250	45.05	44.35	42.53	40.65	38.50	35.70	16.52	15.82	14.00	12.12	9.97	7.17
500	45.31	44.61	42.79	40.91	38.76	35.96	16.72	16.02	14.20	12.32	10.17	7.37
750	45.74	45.04	43.22	41.34	39.19	36.39	17.05	16.35	14.53	12.65	10.50	7.70
1000	46.34	45.64	43.82	41.94	39.79	36.99	17.51	16.81	14.99	13.11	10.96	8.16
1250	47.12	46.42	44.60	42.72	40.57	37.77	18.10	17.40	15.58	13.70	11.55	8.75
1500	48.07	47.37	45.55	43.67	41.52	38.72	18.83	18.13	16.31	14.43	12.28	9.48
1750	49.19	48.49	46.67	44.79	42.64	39.84	19.68	18.98	17.16	15.28	13.13	10.33
2000	50.48	49.78	47.96	46.08	43.93	41.13	20.67	19.97	18.15	16.27	14.12	11.32
2250	51.95	51.25	49.43	47.55	45.40	42.60	21.79	21.09	19.27	17.39	15.24	12.44
2500	53.59	52.89	51.07	49.19	47.04	44.24	23.04	22.34	20.52	18.64	16.47	13.67
2750	55.40	54.70	52.88	51.00	48.85	46.05	24.42	23.72	21.90	20.02	17.87	15.07
3000	57.38	56.68	54.86	52.98	50.63	48.03	25.94	25.24	23.42	21.54	19.39	16.59

Item F-5
A Review of the *Ocean Ranger* Hydrodynamic Model Testing

It was recommended that model studies of the *Ocean Ranger* be undertaken to assist in examining the possible causes of the disaster. This suggestion was weighed against time domain computer simulations and was judged to have certain distinct advantages. These advantages included a more accurate simulation of the mooring system, breaking waves, chain locker flooding, ballast valve runaway, behaviour at large trim angles, flooding of the deckhouse and dragging of the anchors.

The choice of scale for the test was considered very carefully and was established at 1:40 in order to minimize scale effects. It was also concluded that such a large model would facilitate internal water ballast transfer, modelling of chain locker flooding and simulation of the mooring system, including breaking and dragging. The disadvantage of such a scale was that the mooring pattern became immense, even if not taken right out to the anchors but only well past the touch down points.

Technical and commercial proposals were sought from a number of basins in Canada, the United States, Britain, The Netherlands, Denmark, Sweden and Norway. The two basins that best fulfilled the technical requirements of the Commission were the Norwegian Hydrodynamics Laboratory (NHL) at Trondheim and the Hydraulics Laboratory of the National Research Council of Canada (NRC) at Ottawa. The basins were able to work with the model scale of 1:40 and were able to provide teams of highly qualified and experienced technical personnel. The Norwegian laboratory had an extensive track record of contract semisubmersible model testing experience and the Canadian laboratory was internationally known for its work on realistic simulation of wave conditions.

The decision to employ two wave basins, provided the advantage of cross-checks on a number of parallel results. NHL was able to offer the possibility of tests being run at 400 ft. depth as well as two-directional waves. These two features were not available at NRC. Certain other differences existed between the two laboratories. The first major difference was that NRC felt that it was necessary to reproduce the Smoothed Instantaneous Wave Energy History (SIWEH) distribution as well as the wave spectra from the time series obtained from the *Zapata Ugland* wave rider buoy. NHL on the other hand were of the opinion that wave grouping was a function of the peakedness of the spectrum and hence by producing the correct spectra they would automatically reproduce the groupiness.

The other major difference in approach between the two laboratories was in the use of results from the wind tunnel testing carried out by the National Aeronautical Establishment (NAE). NRC used loading filaments attached to the model, the forces in which were computer controlled to model the force and moment spectra. NHL used computer programmed speed controlled fans to apply the force, and hence moments, to the model directly as wind loads.

A number of investigations were required to precede the hydrodynamic model tests. One of these studies, conducted at NRC, was the hydraulic modelling of chain locker flooding. It was realized from the beginning that downflooding due to forward chain locker flooding would play a major role in the behaviour of the rig at large angles of trim, and probably in the subsequent capsize. The flooding of the chain lockers by waves breaking onto the deck and water flooding down the navel pipes is a complex phenomena governed by Froude, Reynolds and Weber number scaling laws. To reduce scale effects a 1:15 model of the forward column SC1, the deck facilities of which could easily be converted to simulate the mirror image arrangement of column PC1, was built. The deck items and all six chain locker openings on the column-deck and the associated navel pipes were modelled in detail.

Wave and rig motions were simulated by repeatedly immersing the model in still water, for various depths and periods of submergence, by means of a hydraulic actuator (Fig. 1). After measuring flooding rates using the 1:15 model, a 1:40 scale model was tested in the same manner to determine the single navel pipe size required to model the flooding rates correctly. A set of different initial pitch angles and depths of water over the deck for several time cycles was investigated for both models. An orifice was found that gave similar flooding rates through the single navel pipe in the 1:40 model to that of the six navel pipes in the 1:15 model. Few immersions were required at large trim angles to fill the chain lockers and navel pipes. When full, almost 1,200 tons (1,219 tonnes) of water are contained in either SC1 or PC1 chain lockers and navel pipes.

Another important study, conducted prior to the hydrodynamic test program, was the aerodynamic model test series, (Fig. 2). The National Aeronautical Establishment (NAE) of NRC used a 1:100 scale model to determine the mean and fluctuating components of the wind loads. Force and moment data were measured for various combinations of vessel attitude, draft and wind directions. A complete dynamic analysis of the model was carried out, using the structural analysis computer program SAP4, to ensure that the lowest natural frequency of the model was sufficiently high. The fabrication of the major structural components of the model from thin wall aluminum and deck fittings from wood and styrofoam kept the mass of the model, including the mounting beams, to 33 lb. (15 kg). The existing balance of the 30 ft. x 30 ft. (9 m x 9 m) wind tunnel of NAE at Uplands was judged to be unsatisfactory for tests of fluctuating loads on the extremely light model. A new dynamic six degree of freedom balance was designed, built, calibrated and used to measure all force and moment data.

Values of wind speed and direction for the storm were provided in the synoptic weather reports from the *Zapata Ugland*, the *SEDCO 706* and the *Ocean Ranger* (last report of the latter at 2330 NST). Values of the parameters were also available from the hindcast study by V.J. Cardone of Oceanweather Inc. made available by Mobil Oil (Canada) Ltd.

Estimates of the effects of waves on the wind loads were measured using stationary, rigid waves mounted on the floor of the wind tunnel. These simulated waves had prototype wave heights of 59 ft. (19 m) and wave lengths of 1312 ft. (400 m). Forces and moments for the model were measured in a number of positions relative to wave crest. The profile of the main wind speed, the turbulence intensity, turbulence scale and frequency content of the turbulence were modelled to simulate the conditions preceding and at the time of the capsize. Most of the wind tunnel data were collected for wind directions between 220° and 310°, for drafts between 32.8 ft. (10 m) and 131.2 ft. (40 m), and for pitch and roll angles between -20° to +20°. The spectra of drag forces, lift forces and overturning moments for several tests were compared. From the analysis, average spectra for the drag force, the lift force and overturning moment were obtained (the other degrees of freedom contained only small fluctuating components). These average spectra were used as models for the spectra of the wind loads to be applied.

Based on the test data, numerical models for predicting coefficients for drag, drag standard deviation, lift, lift standard deviation, overturning moment and overturning moment standard deviation were developed using multiple linear regression techniques. These results were given to the Hydraulics Laboratory (NRC), but were obtained too late for the initial use of NHL. Analytical gust spectra, due to Van Karman, were initially supplied to NHL.

A third study preceding the hydrodynamic model was also conducted at NRC. This study was undertaken to investigate the possibility of realistic simulation of the sea state at the time of the accident. The only wave data available from this time came from the *Zapata Ugland* site, 20 nautical miles away from the *Ocean Ranger*. In order to see if this data could be used with reasonable confidence as being representative for the *Ocean Ranger* site at the time of the disaster, data from the three rig sites at an earlier date were analyzed.

Records from the site of the *SEDCO 706*, the *Zapata Ugland* and the *Ocean Ranger* for the period between 16 to 20 January 1982 and 1 to 2 February 1982 were subjected to spectral and zero crossing analysis. The objective of this analysis was to establish that while all three stations were recording more or less concurrently, the average statistical descriptions of the sea state were sufficiently similar. If this could be shown, then it could be reasonably assumed that the sea state analysis from the *Zapata Ugland* at the time of the disaster would be descriptive for the sea state which prevailed at the *Ocean Ranger* site during the same time period.

The results of the analysis indicated that all wave parameters (without mean values removed) showed extremely high correlation. The day to day variations in sea state parameters about the mean, showed high cross-correlations for wind direction, wind velocity, wave power and characteristic wave height, and moderately high correlations for maximum wave height, peak period, peakedness factor and average steepness. However, the variational cross-correlations for the maximum to significant wave height ratio, the groupiness factor and the average horizontal wave asymmetry are almost zero.

Having established that the *Zapata Ugland* wave recording could be trusted to supply fairly reliable parameter descriptions of the sea state for westerly winds and significant wave heights greater than 5 m, at the *Ocean Ranger* site, NRC undertook a wave synthesis. The simulation was undertaken for the storm of the night of 14 February 1982 from 1830 NST to 15 February at 0430 NST. NRC re-analyzed the data and provided ten variance spectral densities for the ten consecutive hours. These ten spectra were then used by both NHL and by NRC to form ten separate one hour full-scale records, equivalent to ten separate 9 minute 29 second records in model scale. NRC used a synthesis procedure that they had previously developed in-house that exercises greater control over such wave parameters as groupiness factors and maximum wave height. A typical result of this procedure is shown in Fig. 3.

During the time of the hydraulic modelling of chainlocker flooding, the aerodynamic tests and the wave climate analysis and synthesis, work on the 1:40 scale hydrodynamic models proceeded. The design and construction of the model at NHL was undertaken by the staff of the Ship and Ocean Laboratory. The NHL model had pontoons fabricated from watertight plywood 16 layers thick, covered on both sides with a layer of epoxy and fiberglass matt. The construction of the pontoons is shown in Fig. 4. All vertical columns were constructed of aluminum sheet welded into tubes (Fig. 5). Tubes of rolled aluminum and reinforced plastic were used for the horizontal braces and the vertical trusses. The decks were made of watertight plywood and except for the centre area the entire deck section was watertight, but could be opened for water flooding.

The anchor bolsters were made of brass tubes and the stability cones were made of brass plates. The same material was used for the derrick, which was mounted with magnets. The three cranes were made of aluminum, the propeller ducts were made of PVC plastic and the boat bumpers were made of dyvinicell plastic. The helicopter deck, with the accommodation quarters, the winch control houses, the drill floor and pilot house were all modelled according to the general arrangement. The two forward columns had three winches each that were modelled.

The pontoons were subdivided into tanks according to the tank capacity plan of the rig and were filled with a salt-water solution to compensate for the slighter smaller tank volumes of the model than the prototype. A vented ballast system was installed, consisting of a longitudinal brass pipe at the center of each pontoon with cross-connections to 10 tanks on the starboard side and to 5 tanks on the port side. During the dive survey, manual control rods had been found to have been inserted in solenoid valves corresponding to these tanks.[1] The cross-connections to the tanks were opened or closed from the underside of the pontoons by a diver. This procedure was followed in a number of subsequent tests to simulate a specific ballast water transfer.

The chain lockers were modelled on the basis of the previous NRC chain locker flooding tests. All pipes going down to the chain lockers from the upper level were modelled as one tank at the centre of each corner column with a dyvinicell lining of varying diameter. The chain lockers were divided in three by wash bulkheads and were vented to the upper deck.

The deck was constructed of watertight plywood and was subdivided according to the principal accommodation arrangements. The watertight deck-volume was constructed to be flooded through small openings in all internal bulkheads and from the outside through openings in the accommodation quarter down through a stairwell opening in the upper deck.

The vertical centre of gravity was adjusted by rearranging lead weights in the four centre columns, (see Fig. 6). A pendulum method was used to determine the radii of gyration of the lightship and to adjust it to the specified values by moving onboard weights. All five loading conditions were achieved by filling the tanks in the pontoons.

Onboard instrumentation included light emitting diodes mounted on the ends of two booms attached to the derrick and at the

[1]Of the 18 manual control rods recovered, 3 were inserted in solenoid valves associated with the drill water system and, consequently, did not affect the ballast system.

foot of the derrick. These diodes were tracked by two horizontal and one vertical onshore cameras. The optical positioning system (OPTOPOS) based on these components provided motion measurement for six degrees of freedom. Three linear accelerometers were mounted inside the machine house to measure surge, sway and heave motion. Twelve force transducers were mounted at the fairleaders to measure the vertical and horizontal components of mooring line tensions. Two twin wire resistance wave probes were attached to the bow of the model to measure freeboard, and flooding of the forward chainlockers was measured by pressure cells mounted on the bottom of each chainlocker.

The NRC 1:40 scale hydrodynamic model was designed by the staff of the Arctic Vessel and Marine Research Institute (AVMRI) who also supervised its construction. The model was constructed completely in aluminum by the Manufacturing Technology Centre. The chainlockers in the four corner columns were included in the hydrodynamic model and the navel pipes were constructed such that the flow rate of water through them into the chainlockers was modelled correctly. Each pontoon was subdivided into sixteen tanks and piping was installed between ballast tanks (5 on the port and 10 on the starboard side) with remotely controlled pneumatic valves to initiate subsequent ballast transfer tests.

The vertical centre of gravity as well as the longitudinal and transverse radii of gyration were determined using standard inclining and swinging tests respectively, on a specially designed frame (Fig. 7).

The free floating longitudinal metacentric height (GM_L) and transverse metacentric height (GM_T) was checked by inclining the model in the water. Metacentric heights were also measured with all the mooring lines pretensioned to prototype values 235,000 lb.(1,045 kN). These larger values of GM are not to be confused with the smaller values of GM with "mooring pull-down" as used by ODECO. Natural periods of oscillation of the model were also measured.

A general layout of the hydrodynamic model showing the locations of onboard instrumentation is shown in Fig. 8. Each of the twelve mooring lines passed over a fairleader pulley, A1 to A12, which incorporated a load cell to measure the angles of the mooring lines to the model. Load cells, F1 to F12, measured the tension in each mooring line. Four capacitance wire wave probes were installed in perforated tubes in the chainlockers to measure volumes of water in the chainlockers and navel pipes. A reference accelerometer was mounted inside the drill house to measure heave acceleration. Eight light emitting diodes were mounted on a frame on the deck of the model. These diodes were monitored by two cameras that formed part of the Selspot system (similar to the NHL OPTOPOS System) used to measure six degrees of freedom motion response of the model.

The tests of the NHL model were carried out in the Ocean Basin shown in Fig. 9. The dimensions of the basin were 263 ft. x 164 ft. (80 m x 50 m) with a maximum depth of 33 ft. (10 m). Long crested waves are generated by a hydraulically operated, double flap type wave maker along the 164 ft. side. A second system of wave makers, along the 263 ft. side, consists of 144 individually-controlled elements. Each element is an electro-mechanically driven single flap unit. This system of wave generators has been designed primarily for generating short-crested waves.

The general test set-up is shown in Fig. 10. Wind was generated by four fans located as shown in the figure. In general, waves were generated by the wave maker (BM 2), but the multiflap machine (BM 3) was used for three tests. A photograph of the basin and model is shown in Fig. 11.

The test basin at NRC had dimensions of 164 ft. by 98 ft. by approximately 10 ft. (50 m x 30 m x 3 m). A water depth of 6.4 ft. (1.95 m) was used to simulate the water depth at the Ocean Ranger site of 256 ft. (78 m) at a scale of 1:40. Figures 12 and 13 show the general arrangement of basin and model. Three computer controlled high speed DC servo motors mounted on the wall of the basin were connected to the model by braided nylon filaments. These motors acted to produce resultant fluctuating forces and moments on the model, equivalent to the required wind loading.

For both wave directions (240° and 280°) used in the program, it was necessary to truncate ten of the twelve mooring lines at the walls of the test basin. At each truncation point, the mooring line passed over a fixed pulley, vertically to a spring which simulated the elasticity of the lines and then to a reel to permit the adjustment of pretension in the line. A static analysis of both the full-scale and the model mooring systems was carried out at NHL, using the ANKAN program, for the mooring systems at both the NHL and NRC basins. The physical tests were conducted by measuring the forces required to move the moored model in the surge and sway directions.

The overall test objectives for both basins included various phases either known or potentially applicable to the situation of the *Ocean Ranger* during the period from approximately 2200Z (1830 NDT) on the 14 February 1982 to 0800Z (0430 NDT) on the morning of February 15th., (171 tests in all).

The first objective was to investigate the behaviour of the unit, hung-off, from 2200Z. The envelope of tested parameter variance, for 81 tests, was: Draft of 79 ft.; free floating GM_L of 8.86 ft. to -0.54 ft.; initial trim from 0° (level) to +8° (by bow); fully pretensioned moorings or ones with leeward lines slackened; wind direcion of 280° True; time period from 2200Z – 0918Z (tests of varying duration of 1 to 10 hours). In some tests a transient wave(s) of approximately 90 ft. height was run.

During the July 1982 dive survey soundings of No. 10 tanks indicated the possibility of a deballasting operation. This second objective used the following envelope of tested parameters for 13 tests: Draft of 72 ft.; free floating GM_L of 5.82 ft. or 2.54 ft.; initial trim of 0° (level) to +4° (by bow); fully pretensioned moorings or ones with leeward lines slackened; wave direction 240° True; wind direction 280° True; time period 2300Z to 2400Z or 0500Z to 0600Z.

The third objective was to simulate a hypothetical inadvertent transfer of ballast that would have led to the as sounded lower hull tank contents. The envelope of test parameter variance for 30 tests was: Draft of 93 ft.; free floating GM_L of 8.11 ft.; initial trim of -4° (by stern) to +12° (by bow); fully tensioned moorings or ones with leeward lines slackened; wind direction of 280° True; wave direction of 240° True or 280° True; time period from 2300Z to 2400Z or 0500Z to 0600Z with transient waves injected into some tests.

The fourth objective was to simulate a hypothetical inadvertent transfer of ballast leading to "minimum contents" of lower hull tanks based on possible errors in the dive survey tank soundings. The parameter variance in the sequence of 5 tests was: Draft of 86 ft.; free floating GM_L of 6.51 ft.; wind direction of 280° True and wave direction of 240° True.

The fifth objective examined a free transfer of water ballast in lower hull tanks whose solenoid valves were found with manual control rods inserted. A total of 15 tests were conducted with drafts of either 72 ft., 79 ft., 86 ft., or 93 ft.; free floating GM_L of 4.64 ft. to 8.11 ft.; initial trims of -4° (by stern) to 0° (level); fully tensioned moorings; wind direction of 280° True; wave direction of 240° True; time period from 0500Z to 0700Z (in one hour tests).

A single test of simulated impact of the rig's pontoons with the seabed was carried out. The starting conditions for the test (deckhouse and chain lockers flooded) corresponded to the final conditions of the previous 93 ft. draft tests in Objective 5. The tests specified a free floating GM_L of 8.11 ft., ballast valves open; wind and waves from 280° True and additional transient waves.

A final set of model tests were performed within the framework of conditions thought to have had some possible bearing on the loss. The first six tests were carried out a draft of 72 ft.; with a free floating GM_L of 5.82 ft.; with moorings allowed to drag; with ballast pipes and deckhouse open; with no wind; with waves from 280° True; with test times from 0500Z to 0700Z (in one hour intervals). This series of tests included two important tests in which the entire lower deck was allowed to flood as were the two forward columns. It was these tests that produced the only capsizes of the model.

Three tests were conducted at a draft of 79 ft. and a free floating GM_L of 0.96 ft. with an initial trim of 0°, with ballast valves and deckhouse closed and with no wind. These tests used a JONSWAP Spectrum, one with short crested waves, one with long crested waves and one with a cross sea of regular long crested waves. Four tests were also carried out at this draft with a reference of GM_L of 1.36 ft. to 4.64 ft. with no wind and with waves from 240° True from 2300Z to 2400Z or from 0600Z to 0700Z. Two tests at a free floating GM_L of 4.64 ft. and a draft of 79 ft. were conducted. The moorings were allowed to drag, there was no wind, and the ballast pipes, deckhouse, lower deck and forward columns were open. These tests were analogous to the two tests conducted at a draft of 72 ft. but were both carried out with a test time of 0500Z to 0700Z.

A series of four tests were also conducted at 79ft. draft with a reference GM_L of 2.86 ft. or 4.64 ft. with wind only (from 280° True) from a test time of 2200Z to 2300Z.

A group of seven tests were carried out to investigate the phenomenon of wave induced tilt. A draft of 79 ft. was used, for three tests, with a free floating GM_L of 0.30 ft., waves from 240° and no wind. Regular waves or a JONSWAP spectrum were utilized. The remaining four tests were carried out at a draft of 72 ft. with a free floating GM_L of 1.0 or 1.76 ft. at similar test conditions.

Part 2 objectives of the model test program included a series of eighteen tests that investigated the behaviour of the rig at conditions other than those that were thought to be applicable to the loss.

A series of four tests were conducted with a draft of 58 ft., free floating GM_L of 1.87 ft., fully tensioned or slack mooring, wind and waves from 280° True with a test time of 2300Z to 2400Z. A single test was conducted with a 64 ft. draft, a free floating GM_L of 4.92 ft. and fully tensioned moorings, wind from 280° True, waves from 240° True and a test time of 0500Z to 0600Z.

Five tests employed all-chain moorings at a draft of 80.8 ft. or 65.3 ft. with a free floating GM_L of 4.92 ft. to -0.54 ft., wind from 280° True and waves from 240° True with a test time of 0500Z to 0600Z.

The final eight tests were the only ones carried out at a water depth of other than 255 ft., namely 400ft. The draft was 79 ft. with a free floating GM_L of 4.64 ft. or 1.36 ft., trim was from either 0° (level), +4° (by bow) or -4° (by stern). The waves were from 240° True and wind was from 280° True, test times were 2200 to 0342Z or 03V37Z to 0918Z or 0500Z to 0600Z. Moorings of wire/chain were pretensioned or slack (for one test).

These latter sets of tests were carried out in collaboration with the Mobile Platform Stability (MOPS) research project funded by the Norwegian Maritime Directorate (NMD), the Norwegian Offshore Association (NOF) and the U.K. Department of Energy (DOE).

Under the normal conditions that applied to the *Ocean Ranger* on February 14, 1982, that is, corresponding to a moored draft of 79 ft. and a free floating GM_L of 4.64 ft., the response or motions of the hydrodynamic model did not indicate any stability problems. However, in such storm conditions it would have been advisable for the rig to deballast to a survival draft, in order to avoid damage to the deck structures, or to prevent flooding of the chain lockers by waves of extreme height.

The rig did not appear susceptible to downflooding at either 79 ft. or 86 ft. draft unless it was subject to large transient waves or had an initial trim of more than 10° by the bow. This last condition was only true with slack moorings and with the lowest metacentric height of the test series.

At a draft of 93 ft. the incidence of chain locker flooding increased markedly. The tests indicated that there was some water on the deck at a bow trim of 4°. However, significant flooding occurred at approximately 8° of bow trim with slack mooring and at approximately 12° of bow trim with fully tensioned moorings. Once waves began to spill on the deck of the hydrodynamic model, the chain lockers filled with water extremely rapidly because of the large navel pipe openings and also because the rate of flooding increased with increasing forward trim. Once the chain lockers began to flood, it was unavoidable that they would eventually flood completely.

The tests conducted at drafts 72 ft., 79 ft and 93 ft. allowing the free flow of ballast water between tanks, found to have solenoid valves containing manual control rods, resulted in dangerous trims by the bow. Bow trims exceeded 15° and resulted in progressive and critical flooding of the bow chain lockers.

The hydrodynamic model capsized by the bow during a test for which the moored draft was 72 ft. and the free floating GM_L was 5.82 ft. However, it was necessary for the ballast valves to be open and the chain lockers, deck spaces and forward columns to be flooded. Similar tests from a draft of 79 ft. and with free floating GM_L of 4.64 ft. resulted in the model sinking to a final position with the pontoons resting on the bottom at an angle of approximately 56°.

A bottom impact test was conducted to determine if the hydrodynamic model would capsize from an initial moored draft of 93 ft. and a free floating GM_L of 8.11 ft. The test simulated the flow of ballast in the pontoons and the flooding of the bow chain lockers and deck spaces as well as the port trim tank (PT-I). Transient waves with heights of up to 90 ft. were simulated to see if the model would capsize. During the test, the model rested heavily on the bottom at a pitch angle of approximately 52°, and it was apparent that it would have been unlikely for the *Ocean Ranger* to have capsized from this draft.

The wave induced tilt test showed that the hydrodynamic model at a moored draft of 72 ft. and a very low metacentric height acquired steady angles of tilt even in irregular waves.

Tests at the survival draft of 58 ft. with a free floating GM_L of 1.87 ft. showed a reduced stability, due to the location of the stability cones, for bow trim angles greater than approximately 1°. The motions of the hydrodynamic model during the irregular wave test were erratic, and steady trim angles by the bow up to a maximum of 4° were measured.

It was concluded by the Royal Commission's Chief Technical Advisor that capsize of the rig was possible at all drafts up to and including 82-83 ft., but not possible at a draft of 93 ft. The analysis referred to a time dependency whereby, in the 79 ft. test, the model continued to flood in the 'tween deck before a large wave created capsize conditions. Alternatively, once the rig attains a possible capsize condition, a suitable large wave permitting capsize must occur before additional flooding of the 'tween deck increases the displacement such that capsize is prevented.

The model test program provided insight into the general motion response of the rig at a number of drafts and clearly demonstrated the importance of the mooring system in shallow water. Significant response was observed at rig-natural frequency in addition to that at wave frequency for a number of different values of GM. The downflooding angles were investigated at a number of drafts and were found to be less than what was predicted by the USCG study. In general, the model tests clearly established that at level trim the *Ocean Ranger* had quite favourable motion response. However, prior to this set of tests the behaviour of semisubmersible rigs at various trims or in a damaged condition has not received exhaustive attention. The data obtained in the present series of tests should give further understanding in this important area.

Technical advances in model testing techniques included model construction, simulation of realistic wave and wind fields, modelling of ballast transfer and free surface effects, modelling of chain locker flooding and modelling of mooring systems and of second order wave effects. The test basins at NHL and NRC successfully completed a very sophisticated and exhaustive test program whose data should provide a basis for subsequent further analyses. Such analyses should complement existing understanding of semisubmersible behaviour.

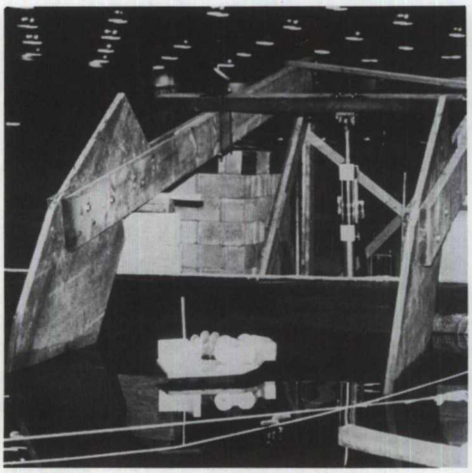

FIGURE 1 Hydraulic actuator – controlled frame for immersing 1:15 and 1:40 model decks – NRC

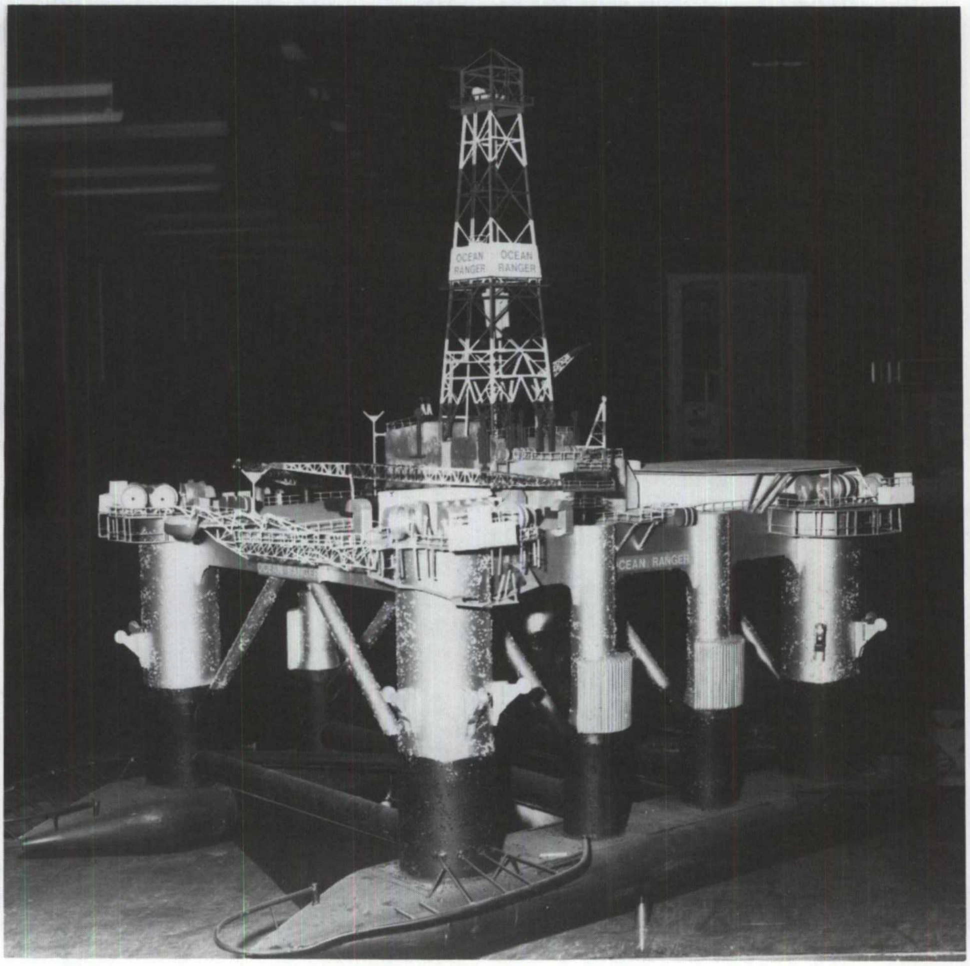

FIGURE 2 View of model showing the deck and column detail – NAE

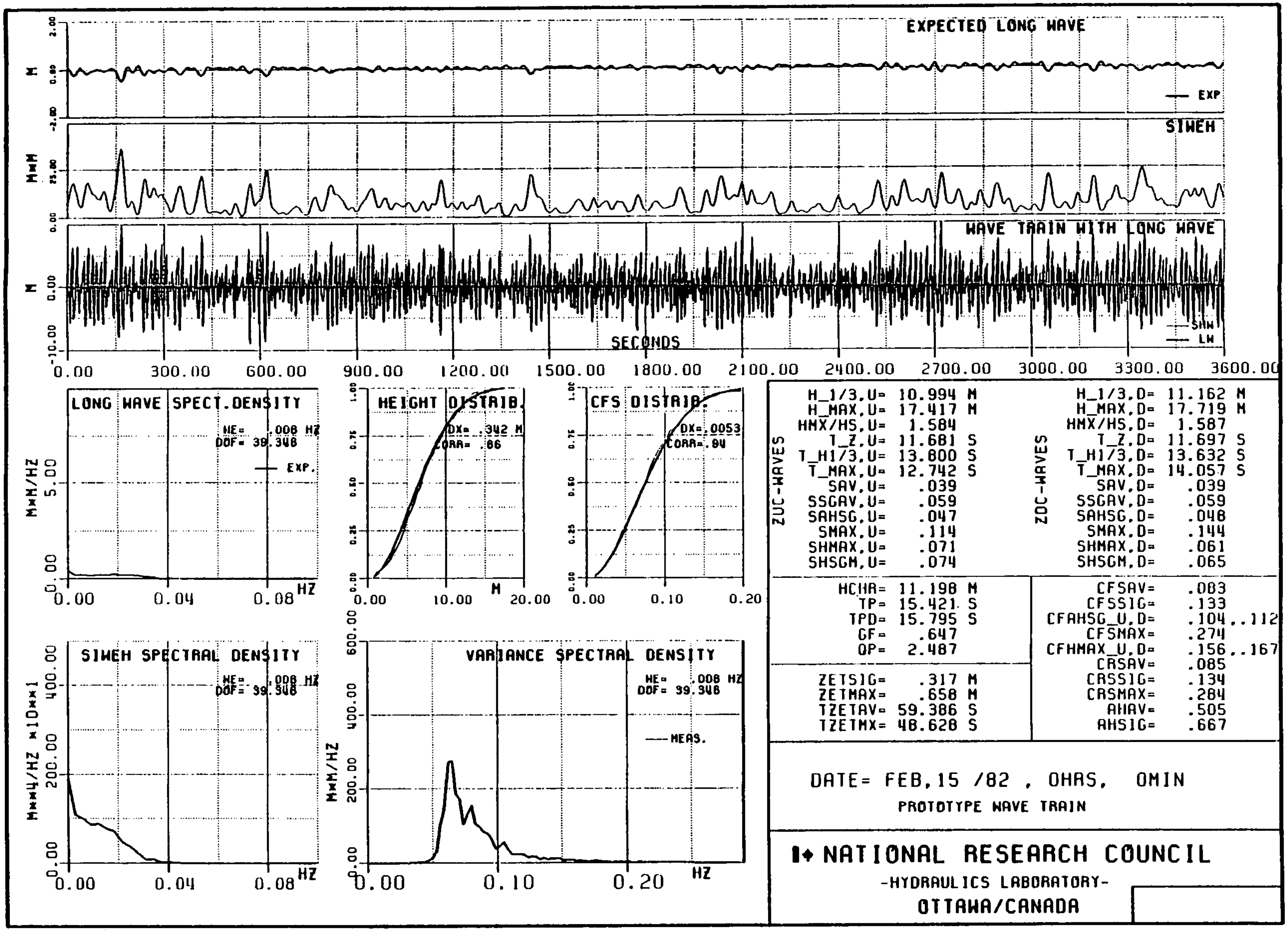

FIGURE 3 Prototype wave train – NRC

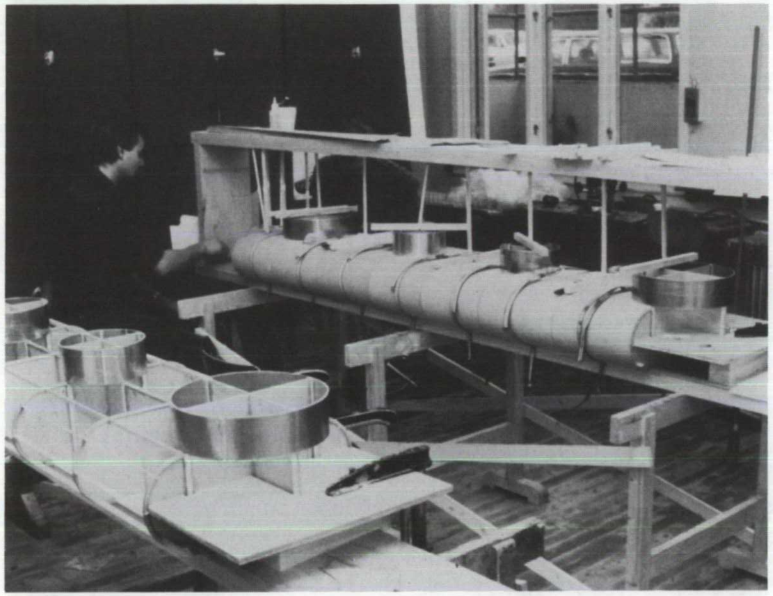

FIGURE 4 Construction of the pontoons – NHL

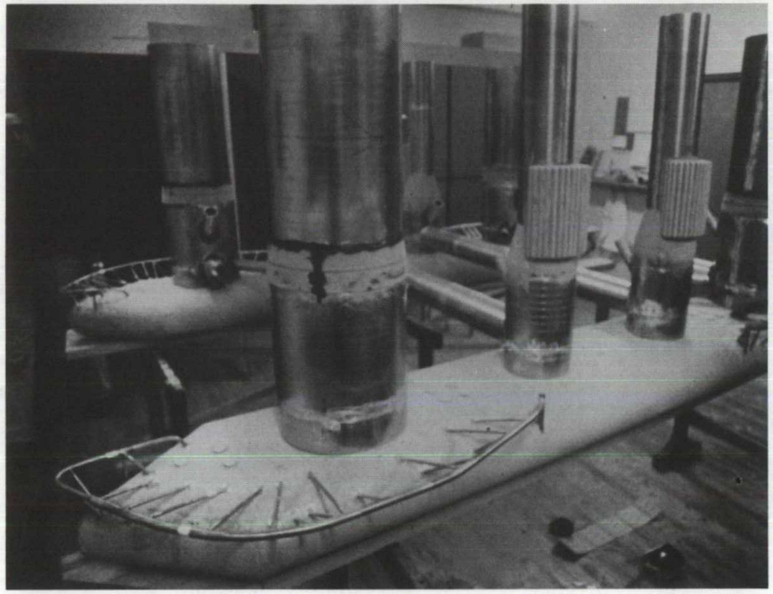

FIGURE 5 Model assembly – NHL

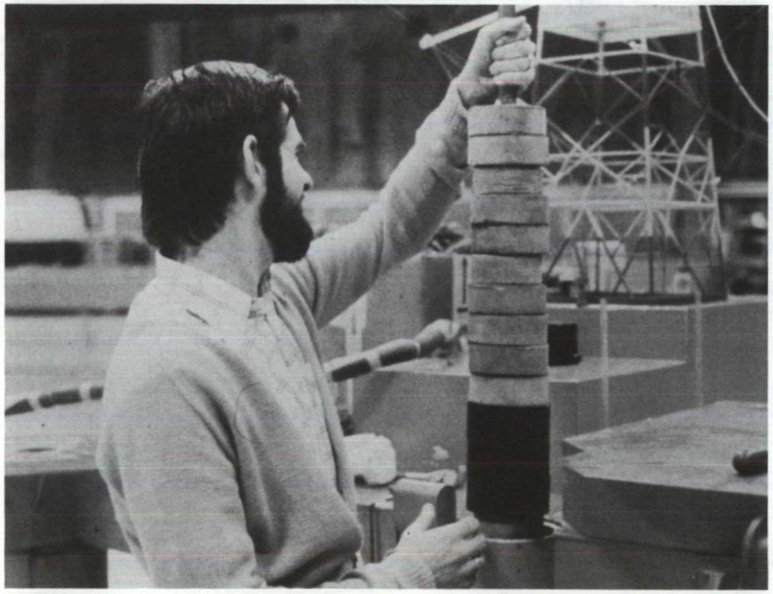

FIGURE 6 Adjusting arrangement for center of gravity – NHL

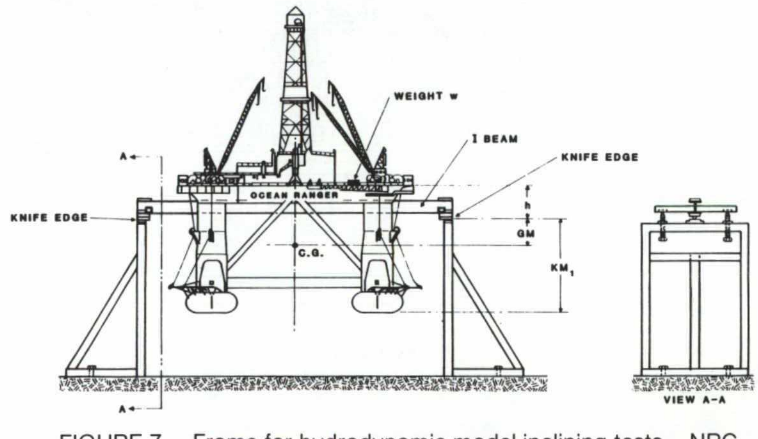

FIGURE 7 Frame for hydrodynamic model inclining tests – NRC

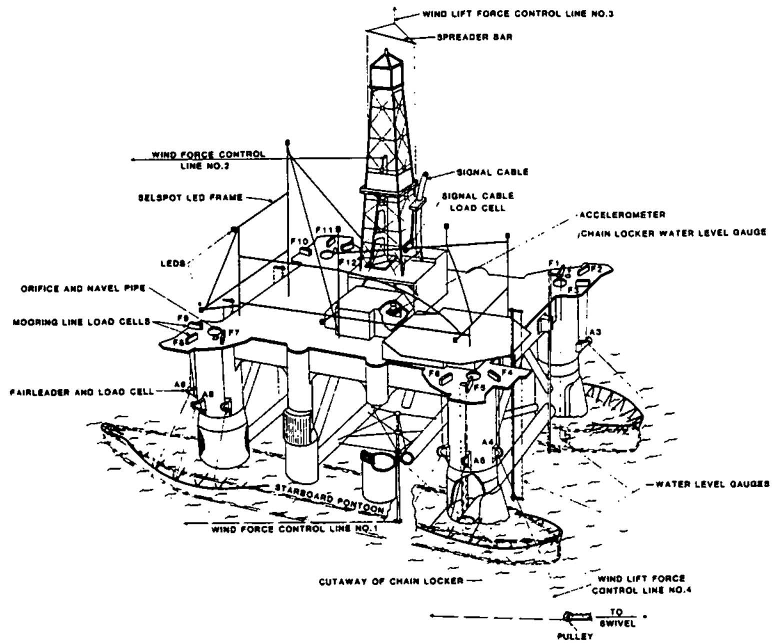

FIGURE 8 Instrumentation on-board the hydrodynamic model – NRC

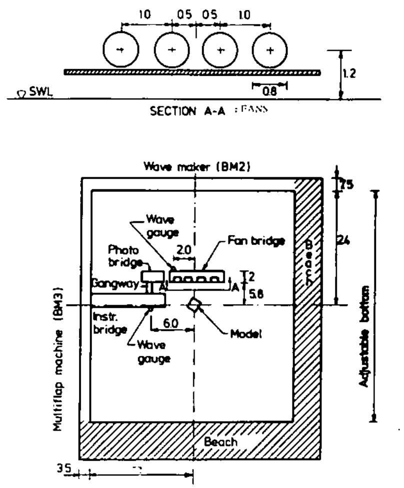

FIGURE 10 General test set-up – NHL

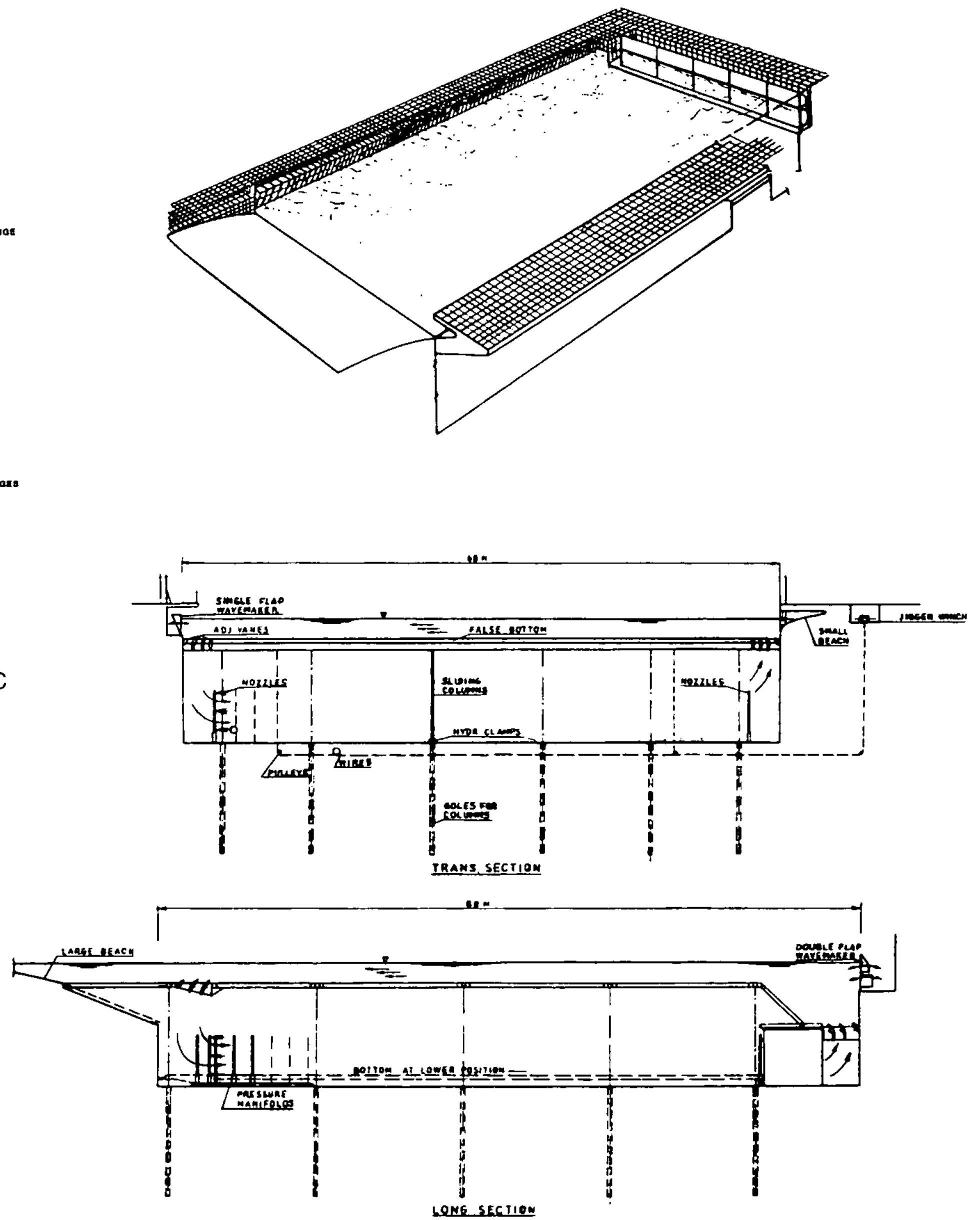

FIGURE 9 Design of the ocean basin – NHL

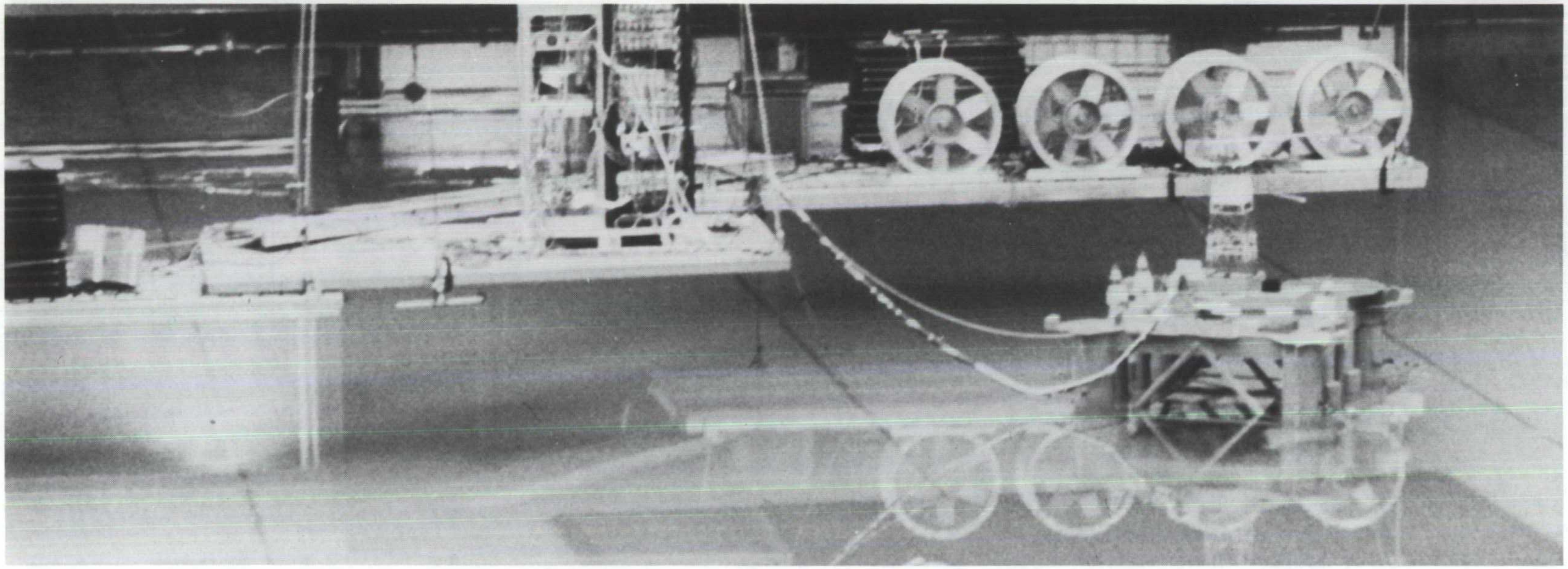

FIGURE 11 General test set-up – NHL

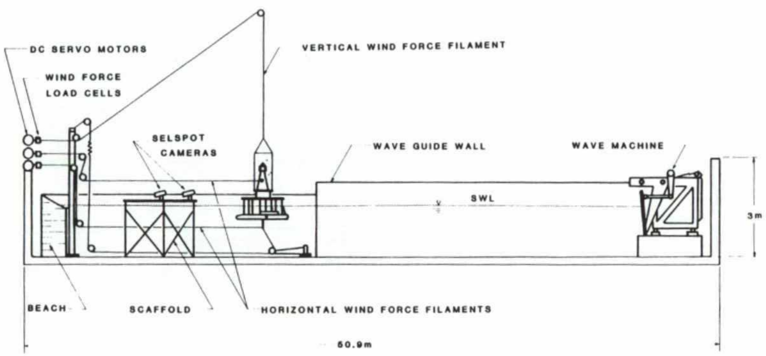

FIGURE 12 Schematic elevation view of the test basin – NRC

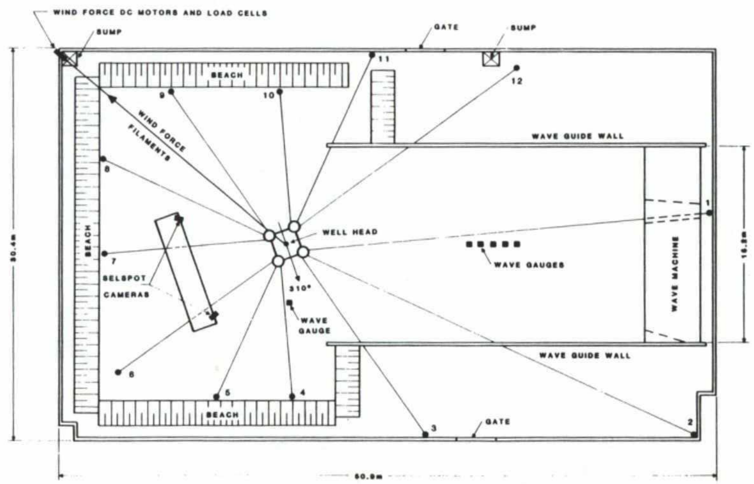

FIGURE 13 Mooring arrangement in the test basin for waves from 240° and wind from 280° – NRC

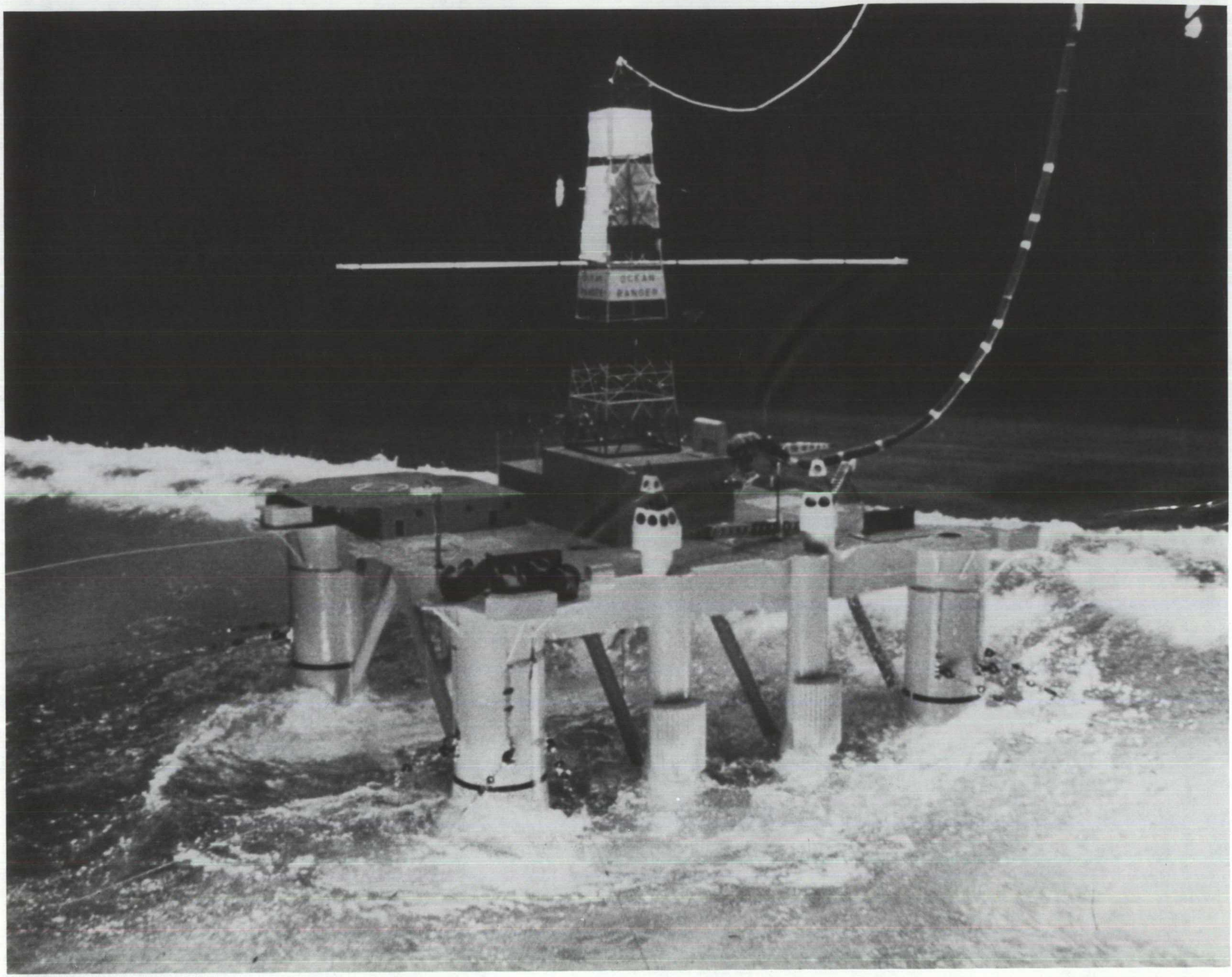

FIGURE 14 The *Ocean Ranger* hydrodynamic model in the NHL test basin

Item F-6
Analysis of Lifesaving Equipment Performance

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INTRODUCTION

The Mobile Offshore Drilling Unit *Ocean Ranger* sank in the early morning hours of 15 February 1982 in the Atlantic Ocean about 175 nautical miles east of St. John's, Newfoundland. All 84 persons aboard are presumed to have died as a result of the casualty; 22 bodies were recovered. The major contributing cause of death for all 22 was identified as hypothermia (loss of body heat, in this case due to immersion in cold water). The prevailing water temperature at the time of the casualty was approximately 31°F (-0.7°C). As a result of this casualty, both the U.S. Coast Guard Marine Board of Investigation and the National Transportation Safety Board have recommended that exposure suits be provided for all persons on board such units that operate in waters where hypothermia is a severe hazard.

The *Ocean Ranger* was built in Japan, initially for Panamanian registry. As such, the lifesaving equipment on board did not necessarily comply with U.S. Coast Guard requirements. In 1979, it was registered as a U.S. vessel, and at that time it would have been required to comply with U.S. Coast Guard requirements for lifesaving equipment (46 CFR 108.501 – 108.527, and Navigation and Vessel Inspection Circular (NVC) 3-78). For the *Ocean Ranger*, these regulations require totally enclosed lifeboats for 100% of the persons on board (100 persons), davit launched liferafts for 100% of the persons on board (or additional totally enclosed lifeboats for 100% of the persons on board), and life preservers for 125% of the persons on board. (A number of other items which were not factors in the survival aspects of the casualty are also required.) The Coast Guard Marine Inspection Office in Providence, RI issued a letter dated 18 December 1979 after the initial inspection for certification that required the *Ocean Ranger* to be equipped with the required U.S. Coast Guard approved totally enclosed lifeboats and davit launched liferafts prior to the next inspection for certification due December 1981 (reference 15 – references listed on page 363). At the time of the casualty, the lifesaving equipment included:

2 unapproved totally enclosed lifeboats installed in davits and operational – total capacity 100 persons;

1 U.S. Coast Guard approved totally enclosed lifeboat installed in davits and operational (this installation had not been inspected or accepted by the Coast Guard at the time of the casualty) – total capacity 58 persons;

1 U.S. Coast Guard approved totally enclosed lifeboat stowed on deck, not operational – total capacity 58 persons;

10 U.S. Coast Guard approved inflatable liferafts (not davit launched) – total capacity 200 persons;

127 life preservers labeled as U.S. Coast Guard approved (see section on LIFE PRESERVERS), equipped with lights and retroreflective material

– U.S. Coast Guard approved work vests (quantity unknown).

In light of the failure of this equipment to save anyone on board the *Ocean Ranger*, the Marine Board of Investigation requested that this analysis of the performance of the equipment be prepared. This analysis was made through examination of exhibits and records of the Coast Guard Marine Board of Investigation, and through inspection and testing of the lifesaving equipment recovered from the *Ocean Ranger*.

LIFEBOATS

At the time the *Ocean Ranger* was constructed, it was equipped with two Harding totally enclosed lifeboats built by Bjõrke Båtbyggeri (now Harding AS) of Rosendal, Norway. These boats were identical, 26 ft. long and had a rated capacity of 50 persons. This lifeboat design has a fibrous glass reinforced plastic (FRP) hull and cover made using methods and materials that are typical for this type of construction. Power is provided by a Sabb diesel engine capable of propelling the boat at a speed of approximately 6 knots. The boat is nominally self-righting, in that if capsized it returns to an upright position, provided that all persons inside are secured to their seats with the seat belts and that there is no significant accumulation of water inside the boat.

The release gear on the Harding boats was of the Mills type, allowing the boat to be disengaged only when the weight of the boat is not supported on the falls (off-load release). The purpose of this arrangement is to prevent the boat from being released before it is waterborne. A single handle located near the release gear support bar inside the boat at the aft end controls this release gear. Cables are attached to this handle which are connected to both the fore and aft release hooks. When the load of the boat is off of the hooks, pulling on the handle overcomes the force of the hook counterweights and opens the hooks simultaneously. When the load of the boat is on the release gear, the force required to open the hooks exceeds that which can be applied manually, so the release does not work in the on-load mode.

One of these boats (#1) was installed on the forward end of the *Ocean Ranger*, just to the port side of center. The other boat (#2) was installed on the aft end, also on the port side of center. In order to comply with the regulations requiring 200% capacity in a combination of lifeboats and davit launched liferafts, the owners of the *Ocean Ranger* contracted with Watercraft America to provide Coast Guard approved boats (reference 6). At the time of the casualty, one of these boats (#4) had been installed on the aft end of the unit, just to the starboard side of the centerline. The other boat (#3) was to have been installed on the forward end just to the starboard side of the centerline, but this installation had not been completed and this boat was stowed on the deck of the *Ocean Ranger* at the time of the casualty (reference 11c).

The Watercraft America lifeboats were built by Watercraft America, Inc. of Edgewater, Florida. These boats were identical, 28 ft. long and had a rated capacity of 58 persons. This lifeboat design is similar to the Harding in that FRP is used in construction of the hull and cover. Power is provided by a Westerbeke (marinized Perkins) diesel engine capable of propelling the boat at a speed of approximately 6 knots. The boat is nominally self-righting to the same degree as the Harding boat. The release gear in this boat is a Rottmer Gear which is an on-load release. On-load release gear allows the boat to be disengaged from the falls at any time, even with the weight of the boat on the falls.

In October 1981, the U.S. Coast Guard published NVC 10-81 on certification and inspection of certain categories of existing vessels, including foreign flag vessels brought under U.S. flag. This NVC contains a section on acceptance of existing lifeboats which were not built under Coast Guard approval and inspection. It lists the features which are regarded as critical to satisfactory lifeboat performance. If the lifeboats on an existing vessel comply with all of these critical requirements, the lifeboats can be used on the vessel as long as they remain in good and serviceable condition. Had this NVC existed at the time the *Ocean Ranger* was brought under U.S. registration and the lifeboats reviewed under its provisions, the following deficiencies would have been noted:

a. The release gear is of the Mills type (see preceding discussion). NVC 10-81 requires that the release gear be controlled from a single point, providing simultaneous release of the hooks while supporting the full weight of the boat (on-load release). The most common release gear of this type is the Rottmer mechanical disengaging apparatus, but recently other types of release gear have been approved that perform the same function. This type of release gear has been required on U.S. Coast Guard approved lifeboats for ocean-going vessels since the 1940s because it allows the boat to be released if the vessel is underway or stationary in a current, and it also allows a carefully timed release for rising and falling water in heavy seas. Retrofit of an on-load release for the Harding boats would have been a major modification.

b. Compared with similar Coast Guard approved boats, the rated capacity of the Harding boat appears to be slightly high at 50 persons. Application of NVC 3-79 (referenced in NVC 10-81) could possibly have resulted in a reduction in capacity of 1 to 3 persons.

c. Under NVC 10-81, the engine is required to start by hand or by a hand-energized system at 20°F without starting aids. Alternatively, engine starting depending on cold starting aids is permitted if the aids are of the permanently installed type and if starting can be accomplished at 5°F with aids and 40°F without aids. The Sabb engine is equipped with a hand crank starting system, but it is not known if it would function at 20°F without aids. If aids were necessary, the type provided on the engine would not be acceptable as a permanently installed type because two screw-in plugs on the side of the engine block must first be removed with a wrench, followed by injection of oil into the holes or insertion of a "cigarette" into the hole, and then replacement of the plugs. Testimony before the Marine Board indicated that on the *Ocean Ranger*, heat lamps were kept in the lifeboat engine boxes to facilitate cold starting, and that a can of ether was also kept available (reference 11g).

LIFEBOAT #1

When lifeboat #1 was first sighted and recovered the day after the casualty, it was flooded, right side up, and down by the stern. There was a large hole in the bow where the forward release gear support cut through the hull and was torn out, and there was a hole in the cover in the area where the rear hatch and helmsman's tower should be. No one was inside the boat when it was recovered and there were no signs of bodies or lifejackets in the vicinity (references 11d, 11h). Only 8 of the required 12 hand flares were found in this boat, but testimony indicates that the flares sighted by the standby boats were probably from boat #2 (references 11g, 12).

In the process of recovering the boat with cables, the boat suffered additional damage. This is apparently when the cover was crushed and the hull damaged in a number of places (reference 11d). In addition to the damage caused by the release gear, there were two other areas of damage that apparently did not occur during recovery. These are two "L" shaped inward fractures on either side of the hull several feet aft of the bow. These fractures match the position of the davit chocks on the launching platform and indicate that the launching sequence for this boat may not have begun, or had just begun when it was separated from the launching platform. The boat and its release gear arrangement are shown in Figure 1.

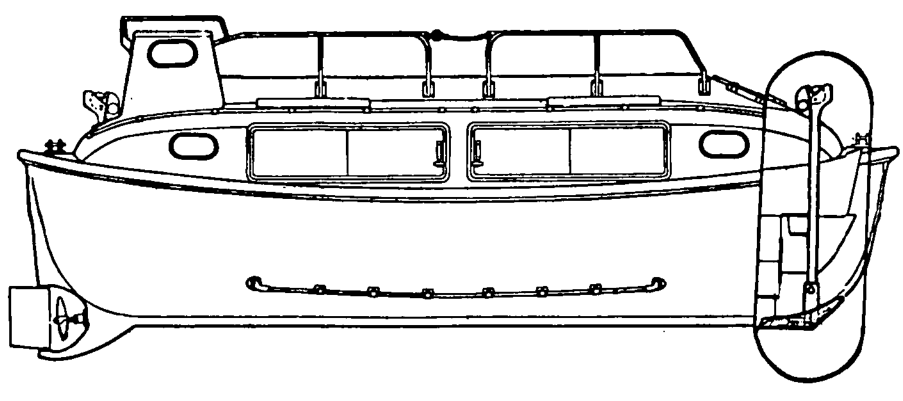


FIGURE 1 Harding 26 ft. totally enclosed lifeboat. Internal view at forward end shows release gear arrangement. The hook is attached to a support bar which is in turn attached to the keel shoe by a pin joint. The keel shoe is "glassed in" at the keel and is the means of transferring the load of the boat to the release gear. The support bar is held vertically by a flange bolted to the fiberglass at the point where the support bar penetrates the cover.

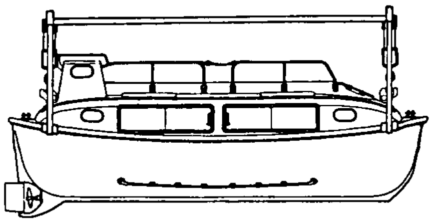


FIGURE 2A Harding lifeboat #1 shown in normal stowage position in davit.

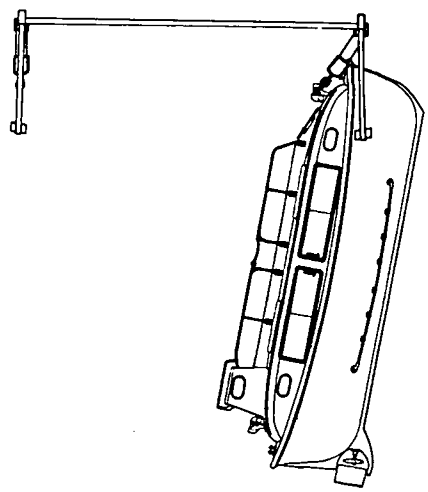


FIGURE 2B Aft release hook has been opened, allowing aft end of boat to fall. Davit chocks at forward end normally in contact with gunwale dig into hull, leaving "L" shaped inward fractures on both sides of the hull.

Figures 2a through 2d depict a series of events which could account for the damage sustained by this boat. Note that there were no surviving witnesses to the release of this boat or any of the other boats, and consequently no testimony to support this scenario. It is deduced from the damage found during the post-casualty inspection of the boat, and in the opinion of the author represents the most probable series of events. The following is a description of the events depicted in Figures 2a through 2d:

a. Boat #1 was at the port bow, the area of the *Ocean Ranger* which is believed to have been the first area of the main deck to enter the water. Seas were heavy at the time, so as the launching platform with the boat approached the water, it would have been struck by a series of waves. The waves were such that the boat would have been subject to severe forces as is evident by the distortion and damage in the aft release hook supporting structure and surrounding FRP laminate. The waves would have lifted and dropped the boat repeatedly, and when the boat was supported by a wave the load would be off the release hook and it could be easily moved to the open position by overcoming the force of the counterweight on the hook. This apparently happened to the aft hook while the boat was being battered by the waves resulting in release of the aft hook. The damage to the rear helmsman's tower and hatch could have occurred at this point since the aft release gear is adjacent to this area.

b. Supported only by the forward hook, the davit chocks on the launching platform lost contact with the gunwale and dug into the hull below and behind their normal position, as the boat was wedged between the chocks on either side. This caused the "L" shaped fractures discussed above. Had the launching sequence been started, the davit chocks would not have contacted the hull in this manner.

c. Hanging vertically from the forward hook, and possibly aided by leverage on the hull by the davit chocks as well as continued battering by the waves, the forward release gear structure began to slice through the bow.

d. Finally, the support shoe was torn out of its keel connection. This allowed the boat to separate completely from the unit and float away. Damage to the helmsman's tower could also have occurred at this point since the boat dropped stern first.

In this damaged condition, the boat would have been open to the sea and flooded, and would have been stable floating either right side up or capsized due to the arrangement of the foam filled flotation compartments along either side of the hull. Because of the immediate flooding of the boat as soon as it fell from the launching platform and entered the water, it would have been very difficult for anyone inside to start the engine or keep the engine running and get underway.

In addition to the damage, another item that suggests that launching preparations had not been completed is the battery charger. This was connected to power aboard the rig by a conventional extension cord. The cord was apparently led out through one of the hatches and the hatch closed over the cord. The charger was found in the boat still plugged into the extension cord, and the extension cord was severed at approximately the place where it would have been led through the closed hatch. Apparently the closed hatch severed the cord as the boat separated from the launching platform. There was no trace of a heat lamp in the engine box or its electrical supply, however.

A telex from the *Ocean Ranger* to ODECO on 11 January 1982 indicated that there was a problem with the lowering control wire on boat #1 chafing on an obstruction. This is the wire that leads inside the boat which must be pulled and held to cause the boat to lower. The telex stated that a modification to rectify the problem could be carried out aboard, but there was no subsequent verification that this modification was completed, and there was no discussion about how or if this interfered with the lowering of the boat (reference 5). There was no discussion found in testimony as to whether or not this was a problem.

The seat belts in the boat would have been useless in their primary role as part of the re-righting system since the boat was flooded, however, the seat belts could have lessened injury during the time the boat was separating from the launching platform. One seat belt mounting plate in this boat has been bent inward, and the FRP structure that secures the stud for the mounting shows evidence of distress from this inward pull. This seat is near the engine box and the boat operator's position where one of the first few persons aboard the boat might sit. There is, however, no way to determine if this damage to the seat belt mounting

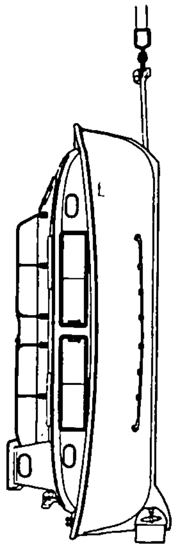


FIGURE 2C (Davit omitted for clarity.) Release support bar connection to cover is intended to stabilize the support bar in the vertical position in normal circumstances. It is unable to support the boat hanging from one end, so it pivots on the pin connecting it to the keel shoe, ripping out the stem area of the hull as it goes.

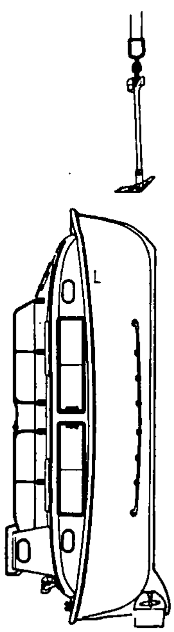


FIGURE 2D The glassed-in keel shoe is unable to support the boat in this position and is torn out, allowing the release gear to separate from the boat which enters the water stern first.

occurred during the abandonment of the *Ocean Ranger*.

The seat belt and mounting designs appear to have shortcomings. The buckles are of a conventional aircraft design with a lift latch buckle that appears to operate easily. This attaches to the other belt-half that includes a sliding adjuster. This adjuster belt does not have a tab at the end, and the adjuster can easily be slipped off the end of the belt by holding the belt and shaking it. It was also noted that it is easy to replace the adjuster mechanism on the belt incorrectly, and if this is done, the adjuster will slide off the belt easily as well. Many of the belt adjusters were found in the boat separated from the belts. Other than simply falling off the belts, another possible explanation for the separation of so many belt adjusters could be that the adjusters were not adequate for holding the passengers in place. There are no known standards that apply to lifeboat seat belts, but there are standards that apply to automotive seat belt assembly strength. In order to determine the suitability of the adjuster mechanism, three belt sets were removed from the boat and sent to United States Testing Laboratory to be subjected to the belt assembly test from Federal Motor Vehicle Safety Standard (FMVSS) 209 of the National Highway Traffic Safety Administration (NHTSA). This involves application of a 5000 lb. load to a loop formed by the belt. One of the seat belt sets passed the test, and the other two failed in the stitching, but not in the adjuster mechanism (reference 17). Since there was no evidence of stitching failure in any of the belts that were examined in the boat, it is probable that the belt adjusters did not fail under load.

The seat belt mounting arrangements on the thwarts appear to be inadequate. These are simply studs threaded into a blind hole in the FRP thwart structure and a backing plate which appears to be about 1/8 in. to 3/16 in. thick, so that only two or three stud threads would be engaged in the backing plate. The FRP would have little value in holding the stud threads. The studs had been torn out of a number of these holes and the threads were stripped. One thwart recovered from boat #2 showed similar damage to these mountings, and one of the stripped holes had been drilled all the way through to the inside of locker underneath the thwart and a bolt used to replace the stud. This indicates that these mountings were a problem before the casualty, and that it can not be concluded that all of these mountings failed in the course of the casualty.

In summary, there is no physical evidence sufficient to draw a conclusion as to whether or not boat #1 was ever occupied.

LIFEBOAT #2

Boat #2 was first sighted underway. It came alongside the *Seaforth Highlander* and capsized slowly as four to five men scrambled out of the boat. Between four and nine men were seen shortly after clinging to the overturned boat. None of these persons were able to be recovered because of the heavy seas and their inability to assist in their own rescue (references 11b, 12). In a later recovery attempt, seven bodies floated out through the hole in the bow and approximately 20 more bodies were seen through an open hatch still belted to their seats. It is known that this was the same boat because the *Seaforth Highlander* ring buoy that had been secured to the boat just before it capsized was still attached (reference 11e). This boat was therefore launched with approximately 31 people or more aboard.

The slow capsizing suggests that the boat was partially swamped as does the testimony indicating that the boat was being bailed as it approached (references 11b, 12). The shift of the weight of the persons leaving the boat on one side was apparently enough to capsize the boat which had diminished stability due to the water inside. If dry inside, a boat like this would not be expected to capsize due to the weight of extra persons on one side. Partial flooding is also suggested by the damage to the bow area that was noted. Witnesses aboard the *Seaforth Highlander* recalled the damage being on the waterline one each side of the bow, "smashed inward", but the top deck appeared okay. None of the witnesses before the Marine Board stated whether or not the release hook was present in the bow (references 11b, 12). After the boat capsized, a crack was noted in the hull running fore and aft, parallel to the keel with water passing through (reference 11e). The cause of the damage to boat #2 cannot be determined from the information available for this analysis, but the damage was probably not as extensive as that to boat #1 since #2 was observed to be underway and "riding high" (references 11b, 12). Bailing a boat as extensively damaged as boat #1 would also have been a futile effort since the bow was open from gunwale to keel. Boat #1 had assumed a position in the water that would have swamped its engine.

During the *Nordertor's* attempt to recover boat #2, a rope was passed around the prop shaft resulting in the shearing of the pin that held the shaft to the engine coupling, allowing the shaft to pull out of the boat. The boat was not recovered (reference 11e). Later, two pieces of flotation foam and a thwart with its attached locker were recovered. These items were definitely identified as coming from a Harding boat since they were identical to similar components in boat #1. Boat #1 was also found not to be missing any of these components. In addition, a checklist was found in the thwart locker that contained identification of boat #2. The only way that the thwart and locker and the flotation foam could have been separated from the boat is if the boat hull had been broken apart. Since it was intact when the attempt was made by the *Nordertor* to recover the boat, it must be concluded that some time after the recovery attempt, this boat suffered extensive damage. During the two days following the attempt to recover boat #2, several sightings of half of a lifeboat were reported (references 7). This wreckage may have been part of boat #2.

LIFEBOAT #3

This is the Watercraft boat that was stowed on deck. This boat was discovered with hull intact and capsized. The cover of this boat was almost totally torn away. Recovery was accomplished by cables wrapped around the boat, and during various moves, one cable eventually cut through the hull and severed it about 1/3 length aft of the bow. This boat contained no fuel, provisions or other equipment. Many of the seat belts were still rolled-up and secured by rubber bands. The boat shows no evidence of having been occupied. It appears likely that it slid or rolled off the deck as the *Ocean Ranger* pitched forward, and that the cover was destroyed in the process. This is an opinion based on the examination of the boat and the knowledge that the boat was stowed on deck, not in its launching platform. None of the witnesses giving testimony to the Marine Board of Investigation saw this boat enter the water. Once in the water, the boat would have behaved essentially as an open lifeboat, flooding in the heavy seas and eventually capsizing. Like the Harding boats, this boat would be relatively stable in the capsized position.

LIFEBOAT #4

No trace has been found of boat #4. It could possibly still be secured to its launching platform, although one witness reported seeing no lifeboats on the stern of the *Ocean Ranger* (reference 11d). The testimony of the alternate Master of the *Ocean Ranger* stated that as of three weeks before the casualty, boat #4 had not been included in the muster list (reference 11c). If the boat had been released, or if it had broken free of its launching platform, the boat or large portions of the boat would have floated to the surface due to its inherent buoyancy. The only sightings of a lifeboat that could be connected with boat #4 were the half lifeboat sightings, although the circumstances suggest that this wreckage was in fact part of boat #2.

LIFEBOAT DESIGN AND PERFORMANCE

The primary purpose of an off-load release gear such as the Mills Gear on the Harding boats, is to allow the boat to be released when the weight of the boat is off the falls. One characteristic of the Mills Gear design is that when the weight of the boat is taken off a hook, the hook can be easily moved to the open position (even independently of the other hook) by overcoming the force of the hook counterweight. In the case of a Rottmer gear and other on-load releases approved by the U.S. Coast Guard, the hook is locked in the closed position until the operator throws the release handle. Additionally, no manufacturer of U.S. Coast Guard approved lifeboats uses a "glassed-in" connection for the keel shoe as in the Harding boat. All keel shoes are connected to the keel by through-hull bolts. The Mills type release gear operating characteristic and method of construction may have therefore led to the premature release of the aft hook of boat #1 with subsequent separation of the forward release mechanism, along with the severe damage it caused to the bow. It can not be definitely concluded that a Rottmer gear would not have failed under the same circumstances, but it would not have failed in the same way. There have been reports of lifeboats on U.S. vessels being swept away by boarding seas, so failure of a Rottmer gear under similar circumstances can not be ruled out. Even if boat #4 which is equipped with Rottmer gear is found still on the *Ocean Ranger*, it must be noted that this boat was on the aft end of the unit, and would not have been subject to the same kinds of forces experienced by boat #1.

The lifeboat installation drawings for the *Ocean Ranger* show that the boats would clear the transverse tube connecting port and starboard columns up to an adverse trim of 12°. Since the *Ocean Ranger* is believed to have gone down by the bow, boat #2 on the stern would have had to be launched against the adverse trim. If the trim exceeded 12°, or if the boat was swinging as it approached the transverse tube, some impact damage might have occurred and might account for the damage noted to boat #2. The length of the falls at the level of the transverse tube would have been approximately 100 ft. which in combination with the heavy seas would have made some swinging a realistic possibility.

In March, 1980, the Norwegian semisubmersible *Alexander L. Kielland* suffered a broken column, heeled to 30° – 35°, continued to heel until 20 minutes later when it capsized. This unit had seven totally enclosed 50 person lifeboats on board which are believed to have been essentially identical to boats #1 and #2 on the *Ocean Ranger*. The following is extracted from a summary of the report prepared by the Norwegian Government Commission investigating the casualty:

> Four of the boats were lowered without problems. However, there were problems with the release of the lifeboat hooks. The hooks, equipped with simultaneous release mechanisms, could not be disengaged under load, a circumstance difficult to avoid because of the rough seas on the day of the accident. For this reason three of the boats were blown against the platform and damaged. On the fourth boat, the after part of the wheelhouse was crushed. Through an opening caused by the impact, a man managed to release the aft hook by hand. Before that, someone had somehow succeeded in releasing the forward hook. A fifth boat fell into the water bottom-up when the platform capsized. In some unknown way, the hooks had been released. People in the boat and people outside it, managed by common effort to right it (reference 4).

The type of problems experienced with the off-load release gear and the subsequent damage to the boats in the *Alexander L. Kielland* case may be relevant in explaining the damage to *Ocean Ranger* boats #1 and #2.

Some concern was expressed in testimony that the FRP structure of the lifeboats was inadequate due to the extent of damage that was incurred (reference 11d). There is no reason to conclude this when all the damage is analyzed. The damage to the FRP in the bow of boat #1, the damage around the rear release hook, the "L" shaped fractures on either side of the bow, and possibly the damage to the helmsman's tower and hatch were apparently directly and indirectly the result of the premature release of the aft release hook. The crushing of the cover occurred when the boat was retrieved by cables. Other damage to the hull also appeared to be cable damage, some of which could have been caused by the lashing cables on the launching platform.

Boat #2 had some damage to the bow of the boat, but the reason for this can not be conclusively determined. It may have been associated with the characteristics of the release gear, impact on the transverse tube on launching, or some other unknown reason. The reason for the apparent subsequent destruction of the hull has not been determined.

The cover of boat #3 was completely torn away, but since this boat was not in a launching platform, this damage probably occurred as the boat slid or rolled off the deck. The hull was subsequently cut in two by a cable used in recovery. The hull is significantly damaged in only one other place, which was a fracture that did not penetrate the buoyancy foam and inner hull. No loss of integrity would have resulted from such damage. This damage may also have occurred when the boat came off the *Ocean Ranger*, or upon recovery.

SELF-RIGHTING OF FLOODED LIFEBOATS

After the loss of the *Ocean Express* in 1976, the U.S. Coast Guard approached the Lifesaving Appliances Subcommittee of IMCO (Inter-Governmental Maritime Consultative Organization, now International Maritime Organization – IMO) and lifeboat builders with a proposal that would require totally enclosed lifeboats to provide an above-water escape in the event of a capsizing in the flooded condition. In most cases, this would be accomplished by the addition of flotation foam to the inside of the cover, so that it would not remain underwater in the event of a capsize. This would raise the hatches on one side out of the water, and in some cases might result in re-righting of the boat. This would prevent persons inside the boat from being trapped underneath with no way out. This approach seems to be accepted by the boat builders and will probably be part of the requirements of a revised lifesaving chapter of the International Convention for the Safety of Life at Sea (SOLAS). This feature might have allowed more of the people inside the lifeboat that capsized alongside the *Seaforth Highlander* to get out of the boat, or it might have caused the flooded boat to re-right itself.

ALTERNATE LAUNCHING METHODS

The damage to lifeboat #2 may have been caused by contact with some part of the rig structure during the launching sequence. This possiblity seems even more likely when the events during the abandonment of the *Alexander L. Kielland* are considered. The type of release gear used on boats #1 and #2 is not Coast Guard approved because it will not release the boat when there is a load on the falls. Nevertheless, Coast Guard approved systems still depend on lowering by wire which can result in the lowering of the boat onto some part of the lower structure of the rig, or swinging into some part of the structure. At the present time, alternatives to lowering by wire are limited.

One new system developed in Norway allows a specially designed lifeboat to slide down a short ramp and free fall into the water. The shape of the boat, its angle of entry into the water, and the motion imparted by the ramp all work to cause the boat to move away from the casualty, even if the engine is not operating. Persons in the boat are secured in specially designed, energy absorbing, aft-facing seating. A number of these systems have been installed on Norwegian ships. The current state of the art limits this system to a launching height of approximately 20m (66 ft). Another version of the system is being developed for use on rigs. This system may be able to be used at heights of up to 30m (99 ft.). Unlike the shipboard system, no ramp would be used and the boat would drop vertically. The shape of the boat and its angle of attack would still result in movement away from the rig. The vertical drop would eliminate the swinging problem of wire systems, but it could still allow the boat to be dropped onto some part of the structure especially in the case of a boat on the high side of a listing rig. Also, if the launch is on the weather side, the boat can be driven into or under the rig as in wire launch systems.

Another system that has been considered would involve the use of some type of boom or slide that allow the survival craft to be launched well away from the structure of the rig. Such a system was proposed in the mid 1970s by the Red Adair Co., and a similar system has been recently proposed by Conoco. Such systems would seem to offer a significant improvement in the ability to launch survival craft from rigs under adverse conditions, however, neither of these systems is beyond the conceptual stage. Development of the Adair system stopped when it became evident that there would be significant structural problems. Inflatable slides have been used to launch inflatable liferafts, however, tests and observations of these systems made it evident they were not suitable for use in heavy winds and seas. At the present time, there are no known raft slide installations on any U.S. registered vessels. Nevertheless, slide or boom launch systems may offer a good launching alternative if the present problems can be overcome.

Another type of release system has been developed by the Whittaker Corp. for their survival capsules launched on single fall systems. This type of release can best be described as semi-automatic. Like the Mills gear, it uses a counterweighted hook that is designed to open when there is no load on the hook, but it is set during lowering by pulling a handle which is connected to a pin that holds the hook in place. When the boat enters the water, the load is momentarily off the hook, and it releases at that instant. If the hook is not set, and the boat becomes waterborne, or if the operator intentionally wants to release the boat before it reaches the water, a lever is provided that can be used to release the boat under load. This design is intended to combine the best features of off-load and on-load release gears. Model tests in a wave tank have shown this system to reliably provide automatic release of the boat. It is of course still a wire launch system, and therefore subject to the same limitations as other systems of that type.

LIFERAFTS

Soon after the casualty, four inflatable liferafts were recovered. One raft was complete with some damage to its canopy and damage to one of the inflation tubes which occurred during recovery. Another raft was complete, but the upper and lower tube had separated from each other over about 75% of the circumference of the raft and some damage to the canopy. The third raft was complete with its floor separated about 80% of its circumference. The floor became completely separated in the process of moving and inspecting the raft. The fourth raft consisted only of an upper buoyancy tube and canopy support, and a floor which was completely separated from the tube except for the inflation hose connection. This raft's canopy and lower buoyancy tube are missing. One of the witnesses reported seeing one partially inflated raft and two fully inflated rafts, one of which was blowing over and over. It is not known if any of these rafts were recovered. One raft was observed to sink the day after the casualty, and another five days after (references 7, 8). A sunken raft was recovered in June 1982 about 60 miles from the scene of the casualty at a location different from the sites where the other two rafts were seen sinking. The five recovered rafts and the two sunken rafts not recovered account for seven of the ten rafts aboard the *Ocean Ranger*, although there is a chance that one of the rafts sighted but not recovered was one lost from the *SEDCO 706* several hours before the sinking of the *Ocean Ranger* (reference 11a).

Three of the rafts and the separated floors had separated at the joints that hold the floor to the buoyancy tubes and that hold the buoyancy tubes to each other. Only one of the painter lines was complete from the raft to the point of the weak link. The other painters were severed at a point short of the weak link. Some damage to the rafts was incurred on recovery. Testimony from persons on-scene indicates that some rafts were properly inflated and others were damaged before they were picked up. One was described as being a few bubbles of jumbled liferaft material with ropes wrapped around it (references 11d, 11e).

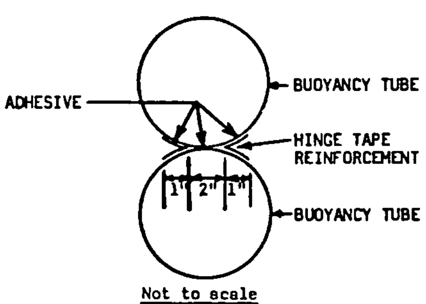


FIGURE 3 Normal construction of liferaft.

There was no evidence that suggests that the rafts were ever occupied. Some equipment bags were open, but since they were made to be readily opened, this is not significant. There was no evidence of the use of flares. None of the liferaft relief valves had plugs screwed into them. While this would not necessarily be done by survivors, any plug found in a relief valve would suggest that the raft had been occupied since the rafts are packed with the plugs out of the valves. All doors were tied in the open position the way they should be when packed.

LIFERAFT DESIGN AND PERFORMANCE

Nine of the ten rafts involved were built in 1974 for C.J. Hendry Co., of San Francisco, California. The tenth raft was a B.F. Goodrich raft which was one of the rafts not recovered. Inflatable liferafts are typically considered to last roughly 10 years, so these rafts may have been nearing the end of their useful lives. Because of the extent of joint separation, attention was focused on the performance of the joints and adhesive. Raft seams are required to have a strength greater than that of the base fabric, however, these requirements are intended primarily for the seams in the buoyancy tubes, rather than the joints that assemble tubes, floor, and canopy into a complete raft. The joints between upper and lower buoyancy tubes and between buoyancy tube and floor were the primary problem areas. Joint samples have been cut from the recovered rafts and tests are to be performed on them by Technitrol Canada, Dorval, Quebec. At this writing, those tests have not been completed. *[Editors Note: The Technitrol Canada Ltd. report, Exhibit # 224, revealed that none of the samples taken from the Ocean Ranger Liferafts met the British Department of Trade specifications for joint strength. No American specification existed for this type of test.]*

Examination of the areas of the raft that had joint separation showed in most cases, adhesive adhered to one side of the joint, but not the other. Failure appeared to be in the peel mode, but it could not be conclusively determined by examination where the peeling began or why. Glued joints are generally weakest in the peel mode. Figure 3 illustrates the normal method of joining upper and lower tubes. On raft 715, the central area shown as 2 in. wide in the figure was actually much narrower and did not have any evidence of adhesive joining upper and lower tubes directly. Adhesive was evident only on the reinforcement. If the tubes are directly joined, forces tending to separate upper and lower tubes would be resisted by a tensile load on the adhesive joining the tubes. As built, the forces pulling the tubes apart are resisted by the reinforcing tape in the peel mode. If the tubes had been joined in the central area, the resulting structure may have been more resistant to separation.

In its examination of the rafts, Technitrol Canada repaired some of the ripped tubes and attempted to inflate the rafts. Several rafts showed blistering where inner and outer coating had separated from the base fabric. Some of these blisters exhibited pinhole leaks. It has not been determined how or when these blisters occurred, or if they contributed to deflation of some of the rafts soon after the casualty.

In order that inflatable liferafts function properly when needed, they are required to be serviced annually by an approved service station. According to the records, the rafts on the *Ocean Ranger* were serviced between 20 April 1981 and 31 July 1981 by an organization in St. John's, Newfoundland (reference 3). This organization was not an approved servicing facility for either C.J. Hendry or B.F. Goodrich rafts and as such would probably not have had the necessary repair parts, manuals, servicing bulletins and packing instructions. A raft which is improperly serviced may not inflate or deploy properly, leading to rafts which can not be used. There were and are no approved servicing facilities in St. John's for U.S. Coast Guard approved rafts. The closest facility was in the Boston, Massachusetts area.

One of the problems with inflatable liferafts that has been recognized for some time is their tendency to be carried away from the scene of an accident before survivors can reach them, and to capsize in high winds and heavy seas. In recent years, a new type of "heavily ballasted" liferaft has been developed and promoted primarily for its resistance to capsizing in heavy seas. In an Advance Notice of Proposed Rulemaking dated 29 June 1981, the U.S. Coast Guard announced that it was considering amendment of the approval requirements for inflatable liferafts to include requirements for such ballast systems. Capsizing of liferafts has been recognized as a problem, but if no one can reach the raft in the first place it is only an academic interest. Perhaps a more important characteristic of such rafts is their tendency to drift with the current rather than being carried away at high speed by wind and waves. Survivors in the water will also drift with the current, so the probability that survivors could reach the rafts is increased.

Even if all of the rafts had floated free, inflated, and had been in the vicinity of persons in the water, it is doubtful that many persons would have been able to reach and board them, although those wearing helicopter-type immersion suits would have had a better chance (see following discussion of exposure protection). The paralyzing effect of the cold water would have made it difficult for anyone in the water without exposure protection to pull themselves aboard a raft. This was illustrated by the inability of any of the persons that entered the water alongside the *Seaforth Highlander* to board the liferaft deployed by that vessel or to assist themselves in any way (references 11b, 12). Some type of effective personal hypothermia protection would have to be provided in order for these persons to help themselves to the extent necessary to board a liferaft.

The fact at least three rafts sank should not be taken as conclusive evidence that they were severely damaged. These rafts are equipped with relief valves to prevent the raft from exploding due to a pressure build-up from excess inflation gas. Once inflated and boarded, occupants should plug the relief valves to prevent loss of gas as the raft flexes in the waves. Unoccupied rafts may eventually deflate even if undamaged. It is not possible to conclusively determine what happened to the liferafts. In the opinion of the author, the available evidence suggests one or a combination of the following may explain why some rafts were damaged before they were recovered:

a. The liferafts may have floated free of their stowed positions as the *Ocean Ranger* sank. A few became entrapped in the rigging and appendages of the unit and never got to the surface. Others did inflate and rise to the surface, but some were damaged as they came in contact with various parts of the structure. This would account for damage to the raft joints and severed painter lines.

b. The liferafts may have floated free of their stowed positions, inflated, and risen to the surface. Some of the rafts had aged sufficiently to cause deterioration in the glued joints. These rafts then suffered damage in the heavy seas.

c. The liferafts may have floated free of their stowed positions, inflated, and risen to the surface. The joints had not significantly deteriorated, but the joint design was not adequate for the stresses encountered. These rafts then suffered damage in the heavy seas.

d. The rafts may not have been properly serviced and repacked, leading to noninflation in some cases, and damage upon inflation in other cases.

e. Rafts damaged as described above would have been readily swamped. When swamped, these rafts would have behaved in a manner similar to heavily ballasted liferafts, drifting with the current and staying near the site of the casualty. Undamaged rafts would have been quickly carried away from the scene by the wind and waves, so that they were difficult to locate by the time daylight arrived.

DAVIT LAUNCHED LIFERAFTS

Under Title 46 of the Code of Federal Regulations, SS108.506 and NVC 3-78, sec. 3.d.(8), the *Ocean Ranger* was required to have a combination of lifeboats and davit launched inflatable liferafts sufficient to accommodate 200% of the persons on board. The owner intended to comply with this requirement by the addition of the Watercraft lifeboats, which in combination with the Harding lifeboats would bring total lifeboat capacity to 200% (references 11b, 6). This solution did not address the fact that the Harding lifeboats were not acceptable under Coast Guard regulations or under NVC 10-81.

In order to fully comply with the Coast Guard requirements, the owner would have had to replace or upgrade the Harding lifeboats, or else remove them and replace the liferafts with davit launched liferaft installations. Had davit launched liferafts been on board, these could have been boarded and launched from the deck in a manner similar to the lifeboats. The approved release hook system automatically releases the raft when the hook is set during lowering and the raft becomes waterborne. Operation of the hook is similar to the system described for the Whittaker survival capsules in a preceding section, except that it may not be possible to release the raft when the hook is loaded. The davit launching system would have made the liferafts more readily available for use since the conventional liferafts could not be boarded until they were waterborne and inflated. On a rig like the *Ocean Ranger* or any vessel with a high freeboard, this is a very difficult operation, made more difficult by the weather, sea state, and sea temperature. On the other hand, the davit launched liferafts are subject to the same launching problems on MODUs as the lifeboats are. The air gap under the rig results in full exposure to wind and sea regardless of where located, and there is the risk that the raft will be driven into some part of the structure during or after launching. Nevertheless, since davit launched liferafts would have been more likely to have been boarded than the conventional rafts, it follows that they could possibly have saved some lives.

LIFE PRESERVERS

Of the bodies recovered after the casualty, 21 were wearing Billy Pugh Model 200 life preservers and one was wearing a Billy Pugh Model WV0-100 work vest. All but two of the life preservers were equipped with ACR model L8-2 water-activated personal flotation device lights. The lights apparently worked well and were useful for locating persons in the water. Many of the bodies (actual number unknown) were found facedown and some were underwater, hanging by the body strap underneath the floating life preserver (references 11b, 11d). Under the latter circumstances, the life preserver apparently came off over the head of the wearer who did not put it back on, indicating that when the life preserver came off, the wearer was already dead or was unable to help himself due to the effects of hypothermia.

The Billy Pugh Model 200 life preservers that were recovered were examined and were found to fall into two distinctly separate groups. One group of devices that came from lot 1A were noticeably heavier than the other devices and were of a different design. The other group was comprised of devices from various lots produced later than lot 1A. The initial certificate of approval for the Model 200 was issued 17 February 1977, however, the lot 1A devices were inspected and passed by a Coast Guard inspector from the Corpus Christi, TX, Marine Safety Office on 15 July 1976. These devices had the Coast Guard approval number on them because the manufacturer had been told what the approval number would be. This is frequently done in advance of actual approval so that the manufacturer can plan equipment markings and promotional material. The fact that they were inspected and passed by a Coast Guard inspector would indicate that they were found to have the proper buoyancy and to conform with the manufacturer's plans and specifications, although this inspection marking is usually not applied until a device is actually approved. Nevertheless, the lot 1A devices were a pre-approval design of 98 units and would not normally have been sold or used as Coast Guard approved devices. It is not known how these devices came to be released.

One pre-approval Model 200 was tested by Coast Guard Headquarters personnel in May 1976. At that time, a tendency for the device to come off over the wearer's head when jumping into the water was noted, but the turning moment (the force that turns the wearer from a face-down to a face-up position) appeared to be acceptable (reference 13). In August, 1976, the company was informed that the device fell short of life preserver performance requirements in that it had a lack of turning moment and that it did not keep the wearer's head far enough out of the water (reference 14). The differences in the designs tested at these two times and their exact relationship with the lot 1A design are not known, however, sketches enclosed with the August 1976 letter show a design similar to the lot 1A design. The design finally approved in February 1977 resolved these problems sufficiently to allow its approval (reference 9). The Model 200 devices from the *Ocean Ranger* that were from lots other than 1A appear to conform with the approved design. No correlation between bodies found face-down and those wearing lot 1A devices can be made from the information available for this analysis.

Rough water performance of life preservers has recently become a matter of concern to the Coast Guard. The person in the water will not rise as fast as the water on the face of a wave and therefore may be submerged momentarily. Depending upon the combination of person, life preserver and sea state, this may develop into a plunging action. One witness reported the heads of the persons in the water constantly washing underwater (reference 12). On yoke-type life preservers like the Billy Pugh devices, this action may result in the life preserver being pulled off over the head if the device is not secure under the chin or around the body. One of the tests that has been used to determine the acceptability of life preservers is a jump test from a height of 3 m into a pool. Although this is intended as a test of the performance of the life preserver when the wearer is jumping into the water, it may also prove to be useful in evaluating the tendency of the device to come off in rough water. During the approval testing of the Model 200 (approved version), 26 persons performed the jump test in the device. It came off over the heads of three of the test subjects and tended to ride up on a fourth. These subjects jumped a second time wrapping their arms around the device (a procedure generally recommended for jumping into the water in any life preserver), and in each case it stayed on. The test report does not record the way in which the body strap was adjusted (reference 9). Recently, as part of the *Ocean Ranger* lifesaving analysis, a Model 200 (approved design) was subjected to the jump test on five different test subjects. With the body strap secured tightly, the device tended to rise to the subject's eye or ear level, but did not come off. With the body strap adjusted to a "comfortable" position as judged by the subject, the device came off over the heads of four out of the five subjects. The same test was performed with a yoke-type life preserver of "standard" design which was found to stay on the same subjects with the body strap in the tight and also in the comfortable positions.

Samples of the Model 200 life preservers from the *Ocean Ranger* were obtained and subjected to further examination and a buoyancy test (reference 10). Examination of the devices and Coast Guard files indicates that the lot 1A devices are made of polyvinyl chloride (PVC) flotation foam rather than polyethylene (PE) foam as prescribed for the approved design. The PVC has a higher density which accounts for the apparent weight difference in the two groups of devices. The neck opening in the lot 1A devices is of a different design and slightly larger than the approved design. PVC foam is also more flexible than PE foam, and the flotation pads on the lot 1A devices are thinner than on the approved devices. All of these factors would contribute to the tendency to allow the wearer's head to slip out of the lot 1A devices. The buoyancy test showed that the lot 1A devices had a buoyancy loss of about 6-1/2% as compared to their original buoyancy. One of the three lot 1A devices tested was 1 oz. under the 22 lb. minimum buoyancy required for new devices. The other two were 6 oz. under the minimum. Some degradation of life preserver buoyancy is expected with age, and the losses on these devices would not be considered critical. The other three devices of the approved design were all above the 22 lb. minimum by 1 oz., 27 oz., and 28 oz.

As a result of these findings, the manufacturer of the life preservers was advised that the unapproved devices had been discovered to be in use and should be recalled or destroyed. The manufacturer's approval of the device was suspended pending improvement in its performance in the jump test (reference 16). The manufacturer did institute a voluntary recall of devices from lots 1 and 1A, comprising 172 unapproved devices (reference 1). The design of the approved device was also altered so that it performs properly in the jump test. The approval certificate was subsequently reinstated.

EXPOSURE PROTECTION

At least two of the bodies recovered were wearing some type of exposure protection garment. In photographs, these appeared to be uninsulated immersion suits of the type sometimes used on offshore helicopters. A quantity of these suits issued by the helicopter operator were normally kept on board the *Ocean Ranger*. These devices were apparently returned as personal effects and were not available for examination. It was reported that at least one person in one of these suits sank when he came out of his life preserver (reference 12). Unlike the U.S. Coast Guard approved exposure suits, these devices do not have the buoyancy and insulation provided by flotation foam. They are waterproof garments that must be used in conjunction with a life preserver. The purpose of these garments is to keep the wearer dry, so that loss of body heat through direct contact with the water is prevented. To protect from conductive heat loss through the suit, as much clothing as possible should be worn underneath the suit.

One recent study compared heat loss rates of different types of exposure protection in calm 11.8°C (54°F) water. All of the test subjects wore the same type of clothing – underwear, long sleeve shirt, denim trousers, socks and sneakers. The average cooling rate for the subjects wearing only a life preserver in addition to the basic clothing ensemble was 2.30°C/hr. Subjects wearing uninsulated immersion suits averaged 1.07°C/hr. loss rate (2.15 time "better" than the subjects with only a life preserver). Those wearing insulated exposure suits averaged a loss rate of 0.31°C/hr. (7.35 times "better" than the subjects with only a life preserver). This study also estimated the time to "incipient death" with different types of exposure protection in the 11.8°C water. For those in life preservers, this time was 3.4 hr. For those in uninsulated immersion suits, it was 7.0 hr. For insulated exposure suits, it was 23.1 hr. (reference 2).

From this data, it can be seen that those persons wearing the immersion suits should have been able to survive perhaps twice as long as those with life preservers alone. These suits obviously did not provide the margin of exposure protection needed in the conditions that existed following the abandonment of the *Ocean Ranger*. Insulated exposure suits of the type that are U.S. Coast Guard approved might have extended survival time six or seven times that of persons wearing life preservers alone.

EMERGENCY RADIO COMMUNICATION EQUIPMENT

An ACR RLB-14 Emergency Position Indicating Radio Beacon (EPIRB) was on board the *Ocean Ranger*. It was recovered after the casualty indicating that it had floated free. The signal from the EPIRB was received by rescue aircraft flying to the site of the casualty, however, since the standby boats had already been alerted to the problems being experienced by the *Ocean Ranger* and since its position was known, the EPIRB did not appear to be a factor in this casualty.

A JVC portable lifeboat radio (Japanese – not FCC approved) was found in boat #1. There was no evidence that indicates any attempt was made to use this radio.

A VHF-FM two-way radio was also found inside boat #1. There were no radio transmissions during the casualty identified as having come from this radio.

REFERENCES

1. Billy Pugh Co., letter, "Recall Model #200 Life Preservers Lot 1 and Lot 1A," 12 October 1982.

2. J.S. Hayward, et al., "Survival Suits for Accidental Immersion in Cold Water: Design-Concepts and their Thermal Protection Performance", University of Victoria, Victoria, B.C., Canada, January 1978.

3. IMP Group Ltd., Liferaft Certificates of Inspection, *Ocean Ranger* Marine Board of Investigation file 4.57A.

4. Torgeir Moan, "The *Alexander L. Kielland* Accident," proceedings from *The First Robert Bruce Wallace Lecture*, Massachusetts Institute of Technology, June 1981. p. 12.

5. *Ocean Ranger* telex to ODECO, St. John's and New Orleans offices, dated 11 January 1982, U.S. Coast Guard Marine Board of Investigation Exhibit 47.

6. Ocean Drilling and Exploration Co., letter to Commander (mmt), Eighth Coast Guard District, dated 14 January 1980, U.S. Coast Guard Marine Board of Investigation Exhibit 12p.

7. Rescue Co-ordination Center Halifax, Nova Scotia, "Search and Rescue Special Report, SAR *Ocean Ranger*," p. 9/2, 20/6, undated.

8. *SEDCO 706* Radio Log, U.S. Coast Guard Marine Board of Investigation Exhibit 11.

9. Underwriters Laboratories, letter, "Performance Testing of Billy Pugh Adult Life Jackets," 12 November 1976.

10. Underwriters Laboratories, letter, "Test Results: Weight Determination and Buoyancy Tests on Six Billy Pugh PFD's," 19 August 1982.

11. U.S. Coast Guard, Marine Board of Investigation. *Certified Transcript of Proceedings in the Matter of Investigation of the Sinking of the Mobile Offshore Drilling Unit Ocean Ranger in the Atlantic Ocean on 15 February 1982.*

a. Volume III, p. 15, testimony of Donald King, 20 April 1982.

b. Volume IV, pp. 15, 17, 18-20, 40, 41, 45, 74, 78, 156, testimony of Rolf W. Jorgensen, 21 April 1982.

c. Volume V, pp. 24-25, 36, testimony of Geoffrey Dilks, 22 April 1982.

d. Volume VIII, pp. 27, 33, 37, 47, 84-85, testimony of James Davison, 27 April 1982.

e. Volume VIII, pp. 97-100, 101, 106-107, testimony of Baxter Allingham, 27 April 1982.

f. Volume XI, testimony of Kelvin Gernandt, 7 June 1982.

g. Volume XII, pp. 31-34, 38, testimony of Ronald Green, 8 June 1982.

h. Volume XVI, p. 34, deposition of Thomas Kane, 21 July 1982.

12. U.S. Coast Guard, Marine Board of Investigation. *Investigation of the Sinking of the Mobile Offshore Drilling Unit Ocean Ranger in the Atlantic Ocean on 15 February 1982* transcript, Exhibit 53A, pp. 10, 11-12, 25-26, 29, 31, 33, deposition of Ronald Duncan, 21 May 1982.

13. U.S. Coast Guard (G-MMT-3), memorandum, "Swim Test Results," file 5946/160.055/113, 6 May 1976.

14. U.S. Coast Guard (G-MMT-3), letter to Billy Pugh Co., file 16714/160.053/GENERAL, 2 August 1976.

15. U.S. Coast Guard Marine Inspection Office, Providence, RI, letter to Ocean Drilling and Exploration Co., dated 18 December 1979, U.S. Coast Guard Marine Board of Investigation Exhibit 12i.

16. U.S. Coast Guard (G-MVI-3), letter to Billy Pugh Co., file 16714/160.055/113, 22 September 1982.

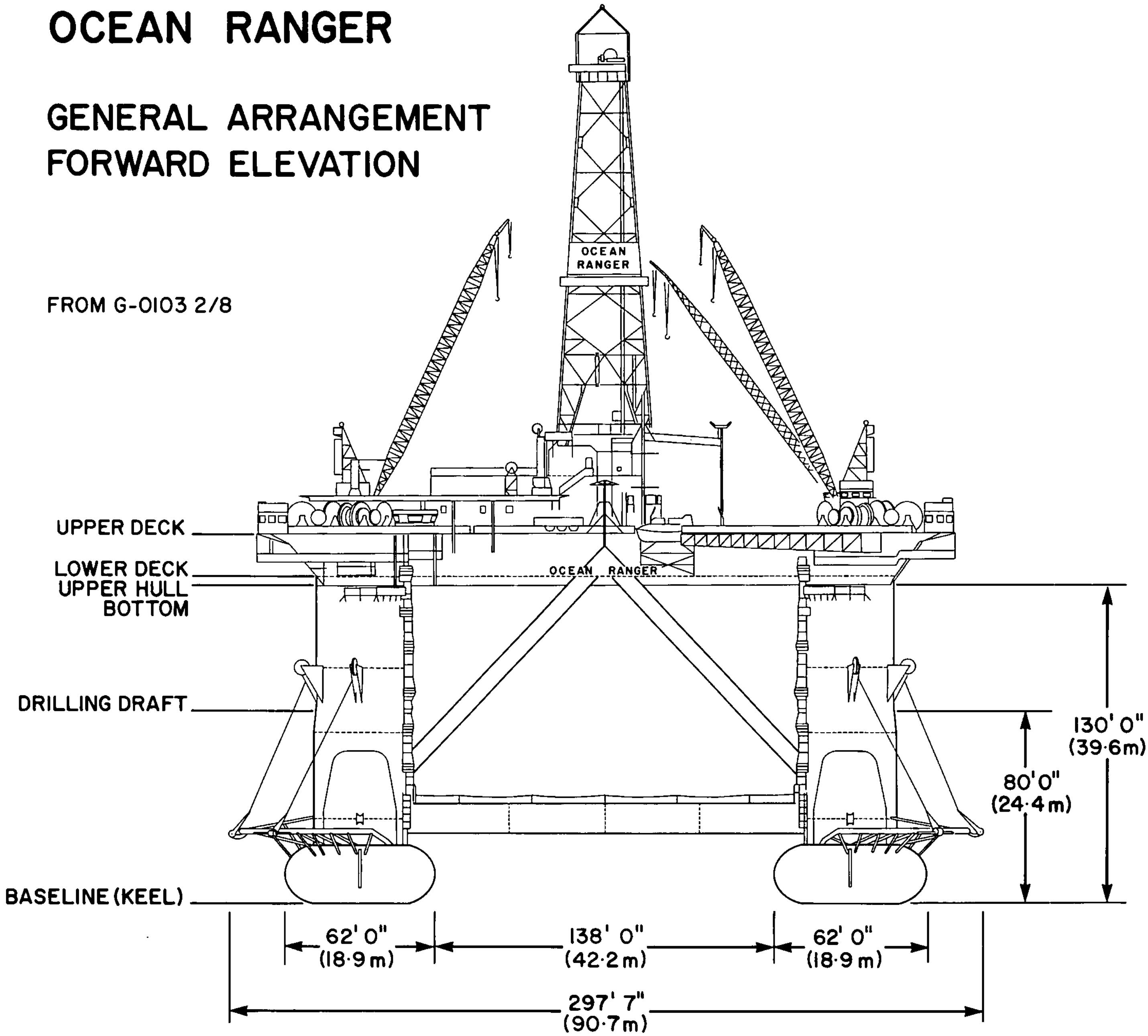
17. United States Testing Co., Inc., "Report of Test; Seat Belt Assemblies," Test Number 83428-82, 19 August 1982.

Item F-7
Technical Drawings

OCEAN RANGER

GENERAL ARRANGEMENT FORWARD ELEVATION

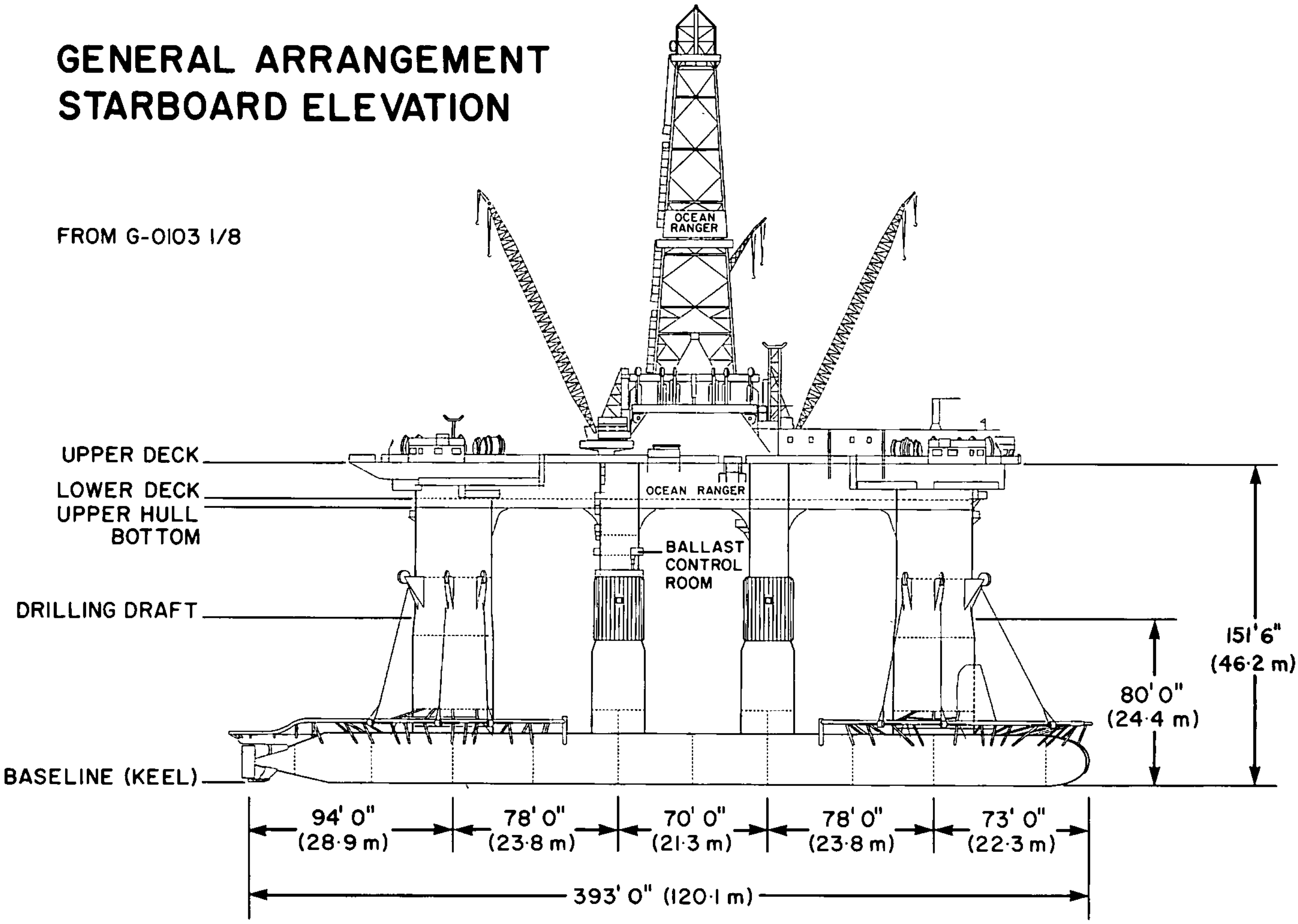
FROM G-0103 2/8



OCEAN RANGER

GENERAL ARRANGEMENT STARBOARD ELEVATION

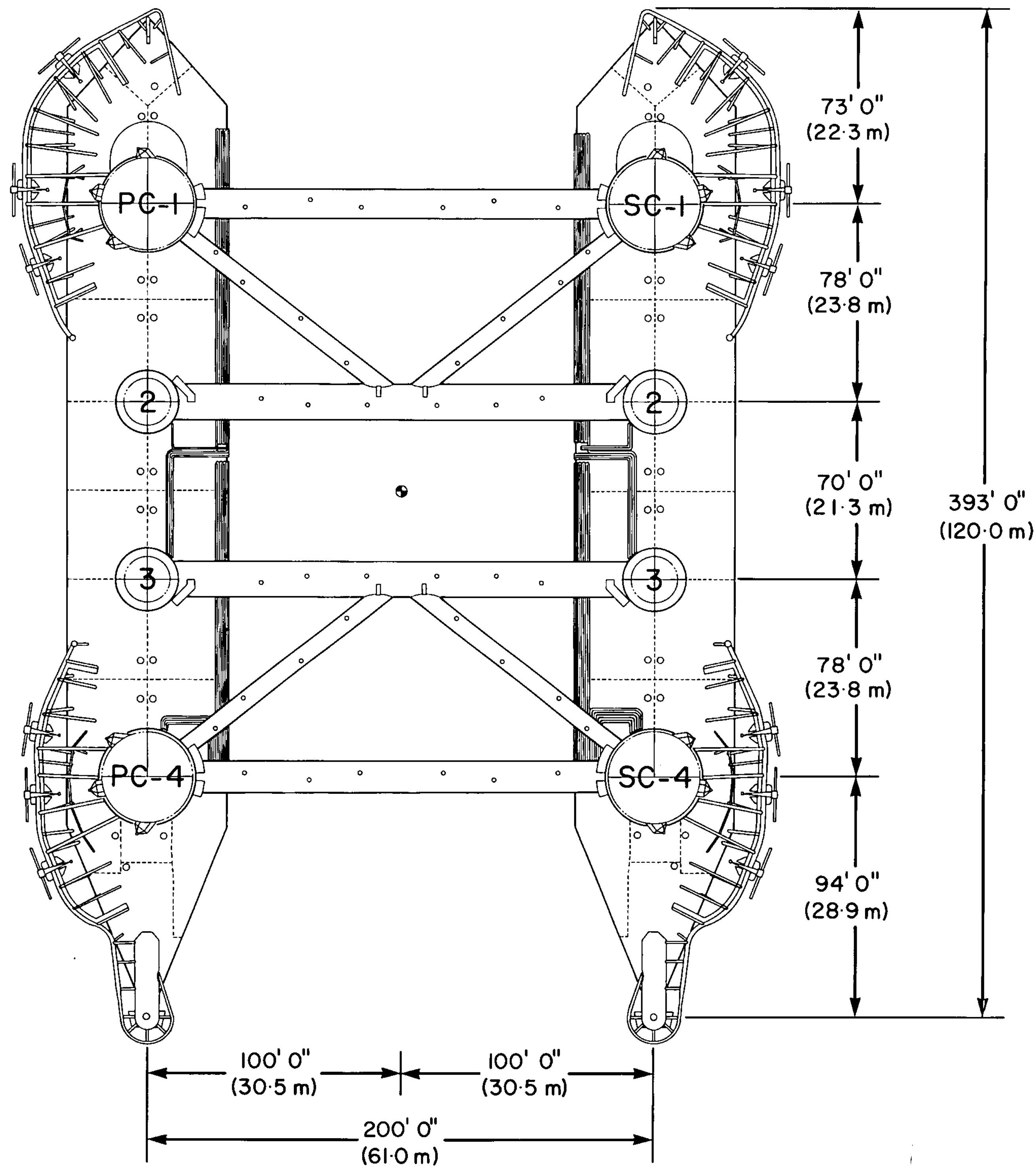
FROM G-0103 1/8



OCEAN RANGER

GENERAL ARRANGEMENT
LOWER HULL EXTERIOR FITTING

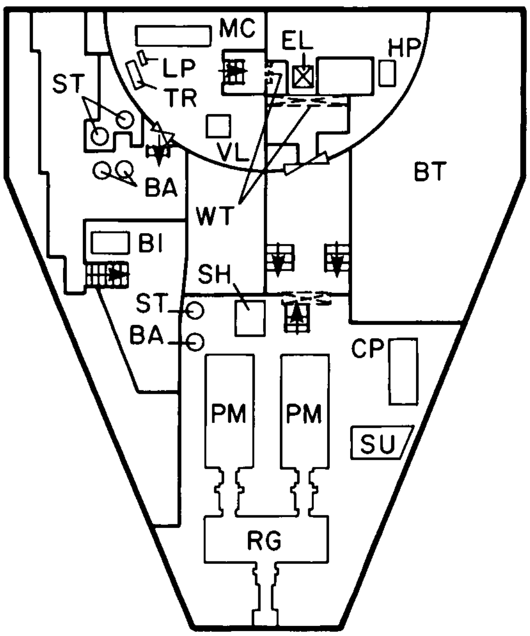
FROM G-0103 3/8



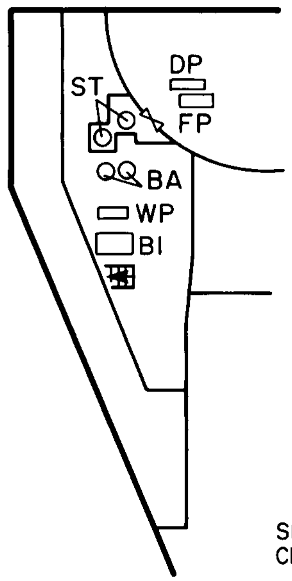
OCEAN RANGER

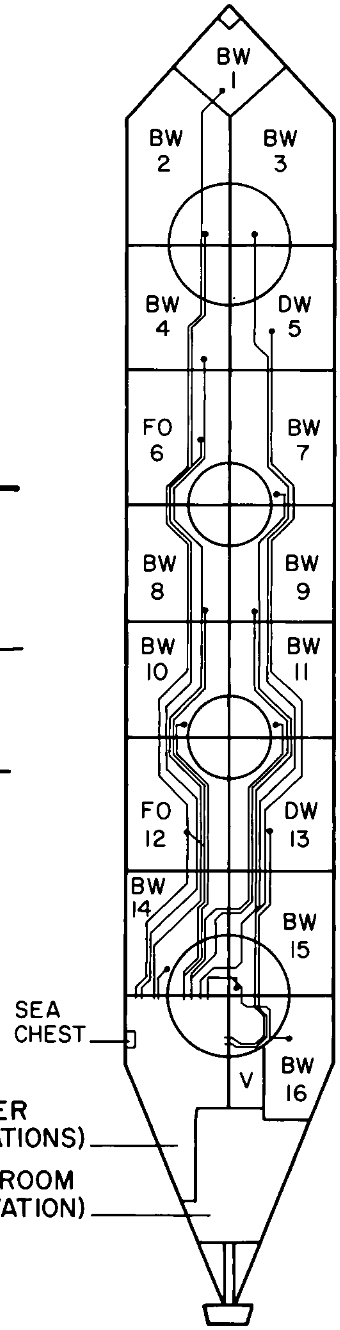
PUMP AND PROPULSION ROOMS (STARBOARD PONTOON)

UPPER ELEVATION



LOWER ELEVATION

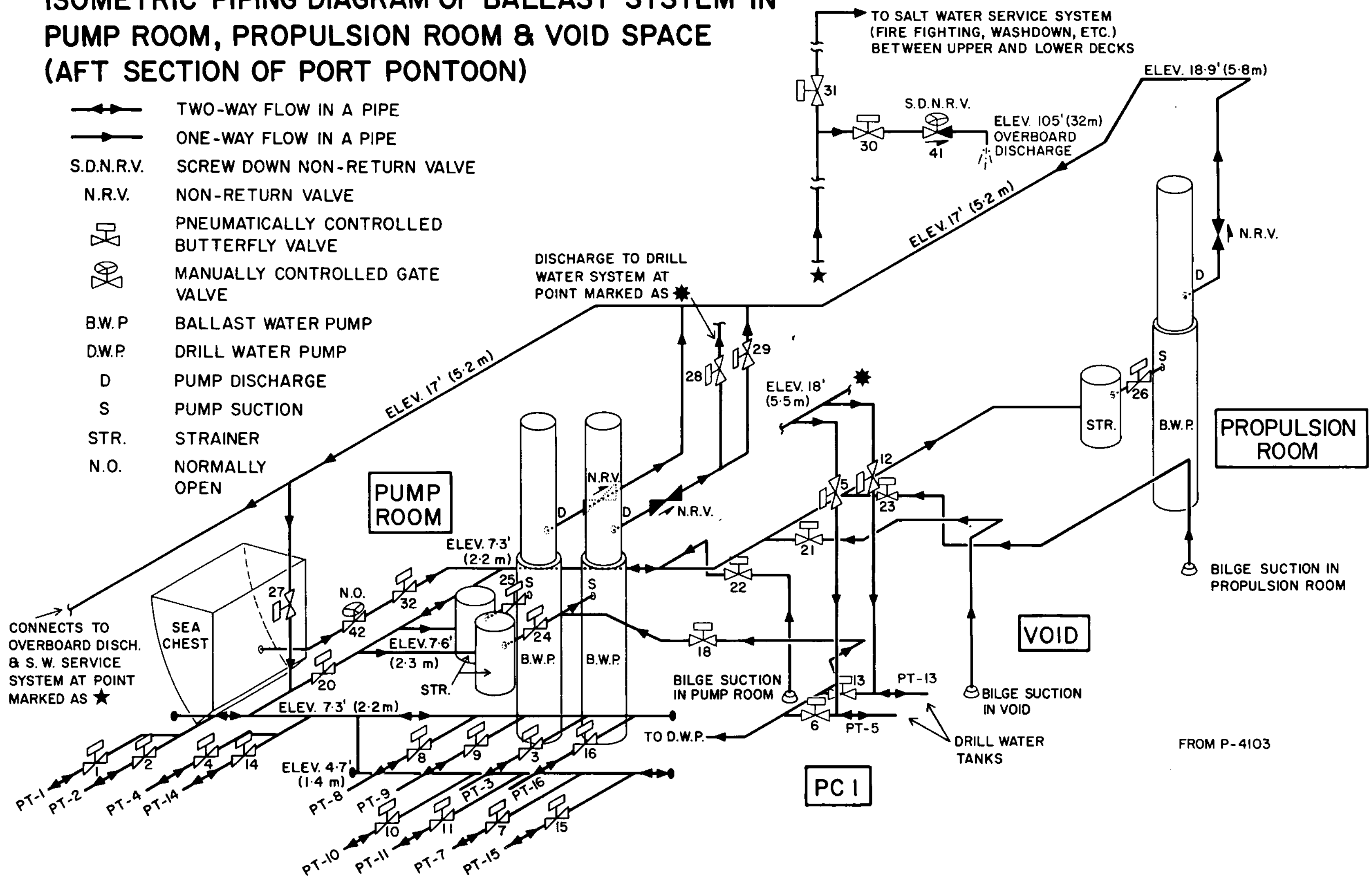


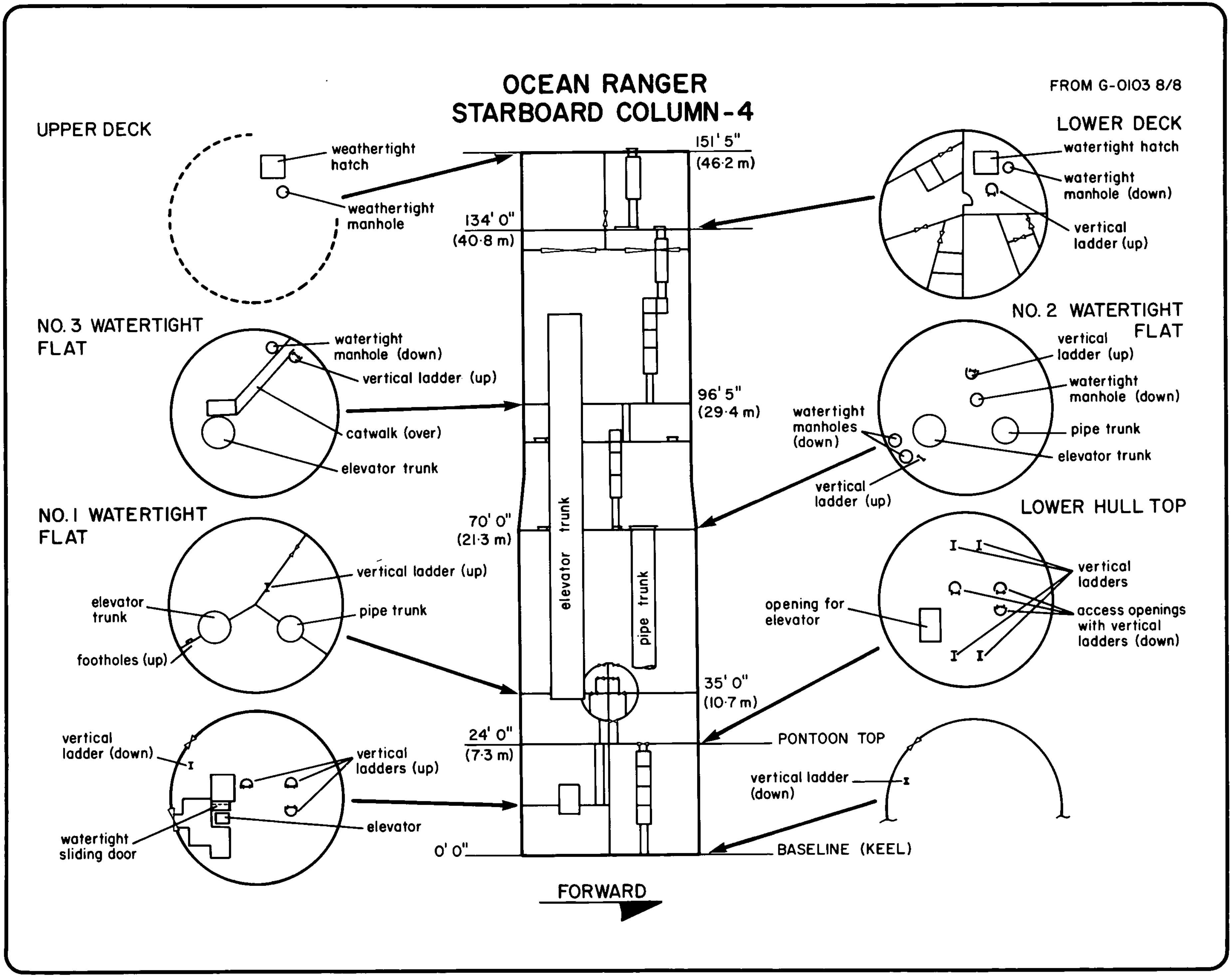


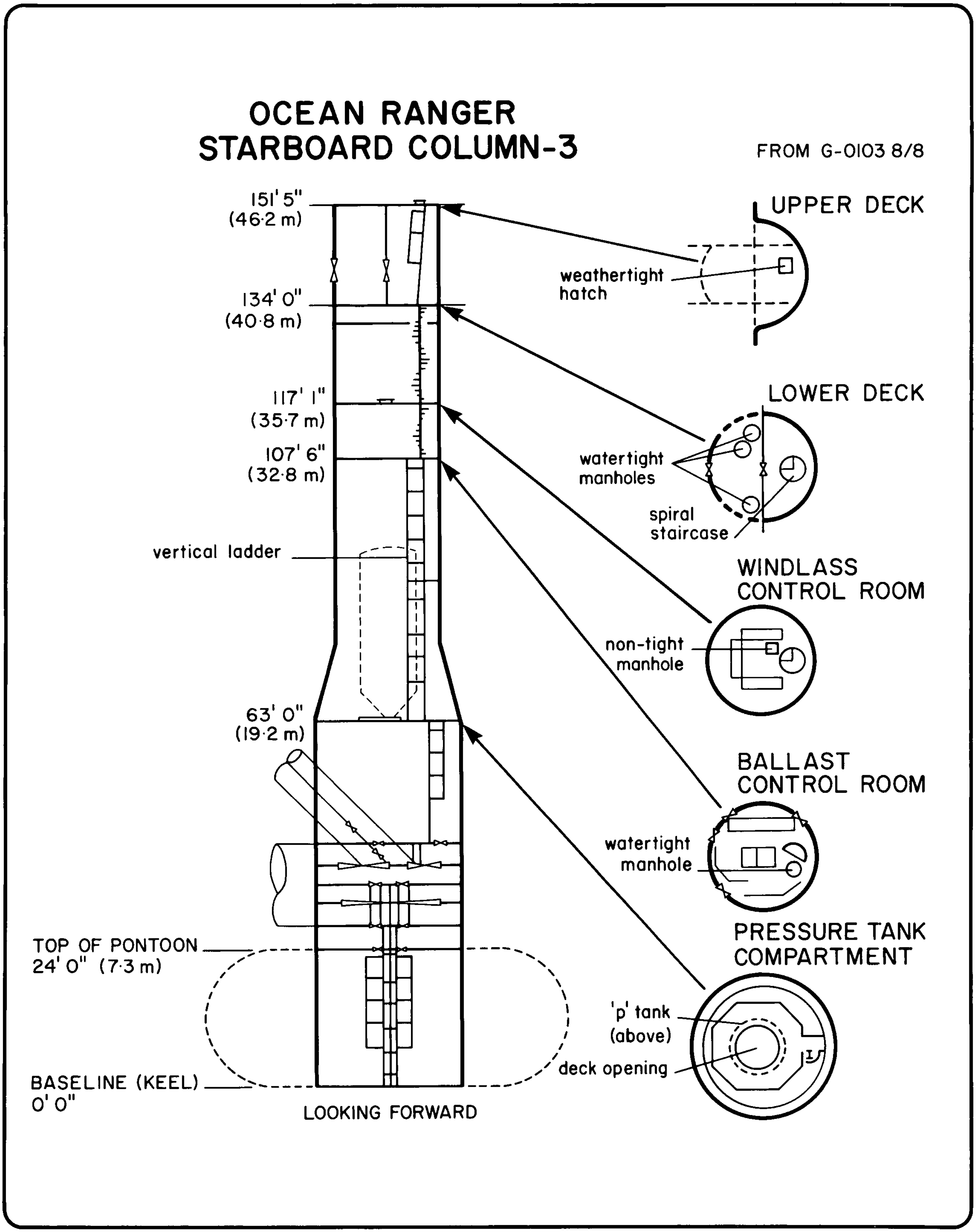
BA	BALLAST WATER PUMP
BI	BILGE PUMP
BT	BALLAST TANK
CP	CONTROL PANEL
DP	DRILL WATER PUMP
EL	ELEVATOR
FP	FUEL PUMP
HP	HYDRAULIC PUMP UNIT
LP	LIGHTING PANEL
MC	MOTOR CONTROL CENTRE
PM	PROPULSION MOTOR
RG	REDUCTION GEAR
SH	STEERING HYDRAULIC UNIT
ST	STRAINER
SU	SUMP TANK
TR	TRANSFORMER
VL	VERTICAL LADDER
WP	PROPULSION MOTOR COOLING WATER PUMP
WT	WATERTIGHT SLIDING DOOR
•	SUCTION BELL MOUTH
BW	BALLAST WATER
DW	DRILL WATER
FO	FUEL OIL
V	VOID

OCEAN RANGER

ISOMETRIC PIPING DIAGRAM OF BALLAST SYSTEM IN PUMP ROOM, PROPULSION ROOM & VOID SPACE (AFT SECTION OF PORT PONTOON)



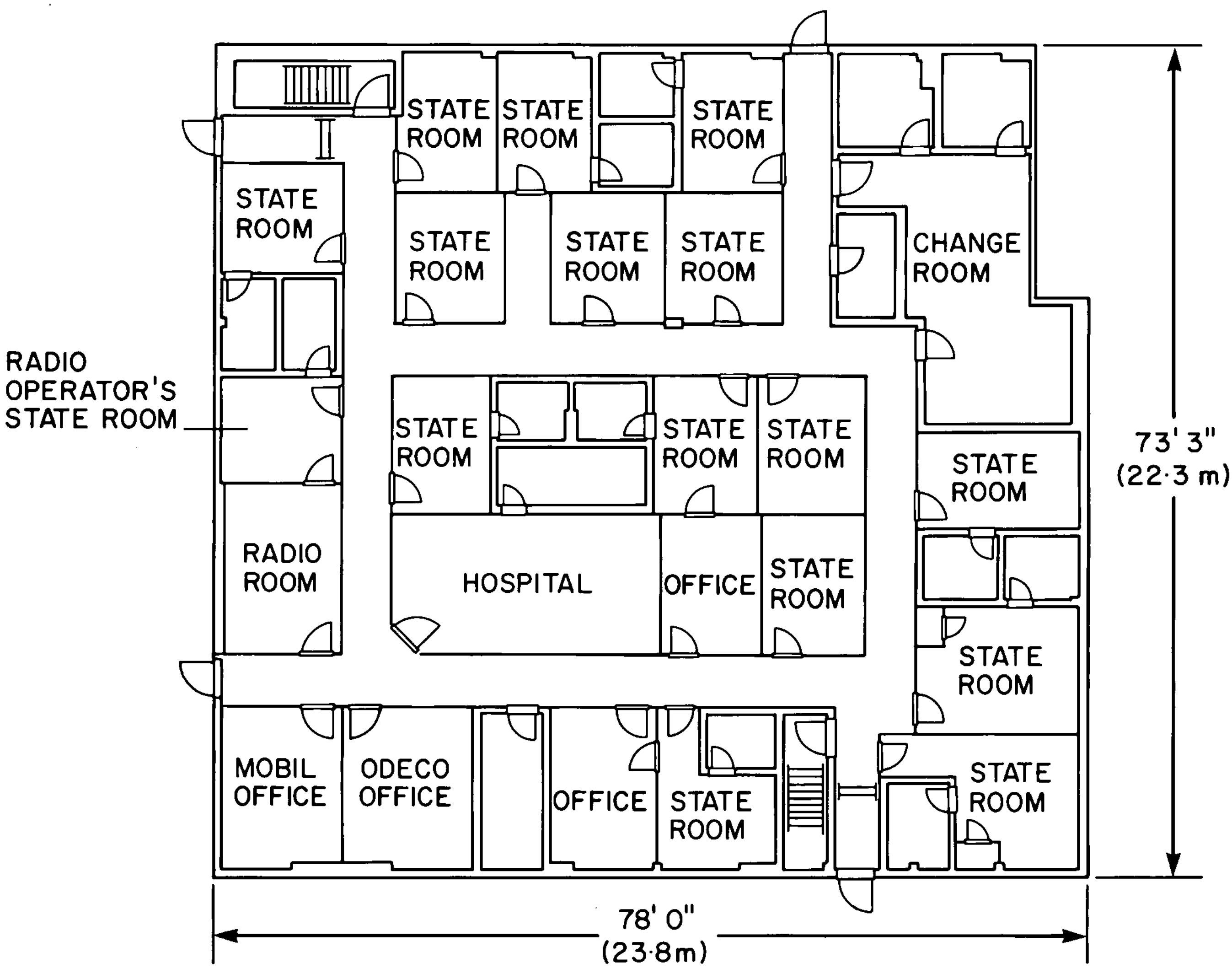

OCEAN RANGER
STARBOARD COLUMN-4
FROM G-0103 8/8
UPPER DECK
weathertight hatch
weathertight manhole
NO. 3 WATERTIGHT FLAT
watertight manhole (down)
vertical ladder (up)
catwalk (over)
elevator trunk
NO. 1 WATERTIGHT FLAT
vertical ladder (up)
elevator trunk
pipe trunk
footholes (up)
vertical ladder (down)
vertical ladders (up)
elevator
watertight sliding door
151' 5"
(46·2 m)
134' 0"
(40·8 m)
96' 5"
(29·4 m)
70' 0"
(21·3 m)
35' 0"
(10·7 m)
24' 0"
(7·3 m)
0' 0"
elevator trunk
pipe trunk
PONTOON TOP
BASELINE (KEEL)
FORWARD
LOWER DECK
watertight hatch
watertight manhole (down)
vertical ladder (up)
NO. 2 WATERTIGHT FLAT
vertical ladder (up)
watertight manhole (down)
pipe trunk
elevator trunk
watertight manholes (down)
vertical ladder (up)
LOWER HULL TOP
vertical ladders
access openings with vertical ladders (down)
opening for elevator
vertical ladder (down)

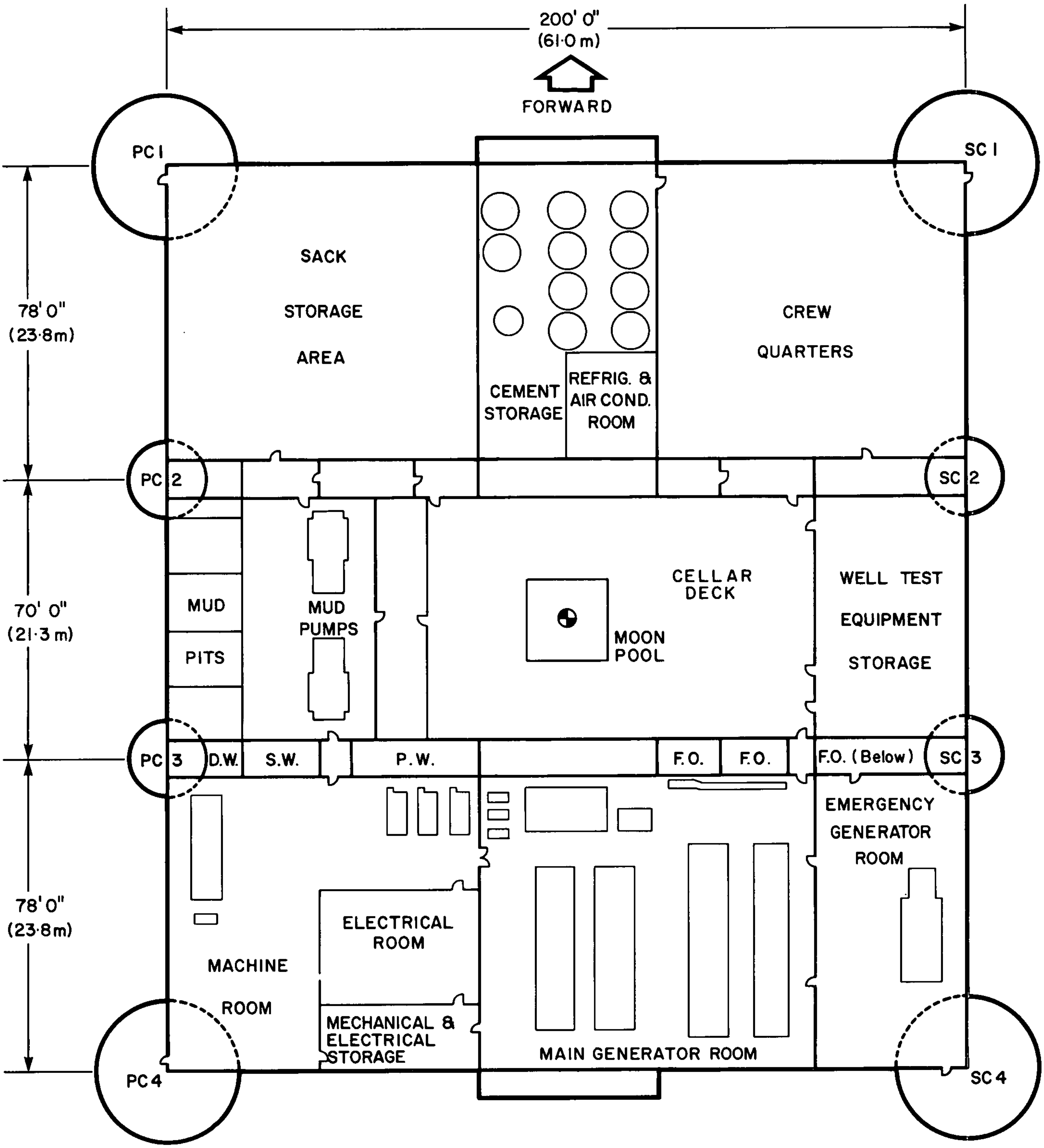


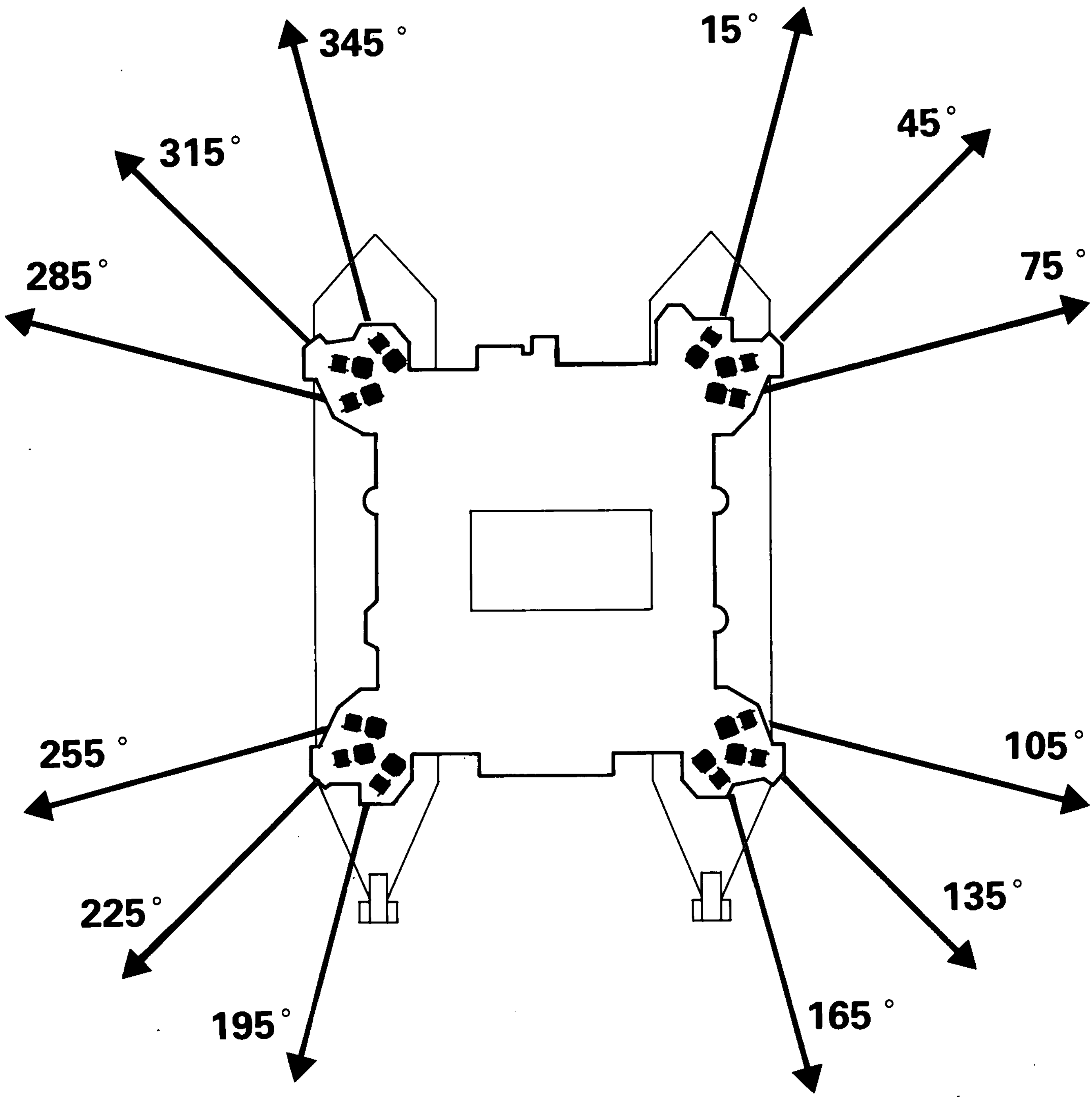
OCEAN RANGER
STARBOARD COLUMN-3
FROM G-0103 8/8
151' 5"
(46·2 m)
134' 0"
(40·8 m)
117' 1"
(35·7 m)
107' 6"
(32·8 m)
vertical ladder
63' 0"
(19·2 m)
TOP OF PONTOON
24' 0" (7·3 m)
BASELINE (KEEL)
0' 0"
LOOKING FORWARD
UPPER DECK
weathertight
hatch
LOWER DECK
watertight
manholes
spiral
staircase
WINDLASS
CONTROL ROOM
non-tight
manhole
BALLAST
CONTROL ROOM
watertight
manhole
PRESSURE TANK
COMPARTMENT
'p' tank
(above)
deck opening

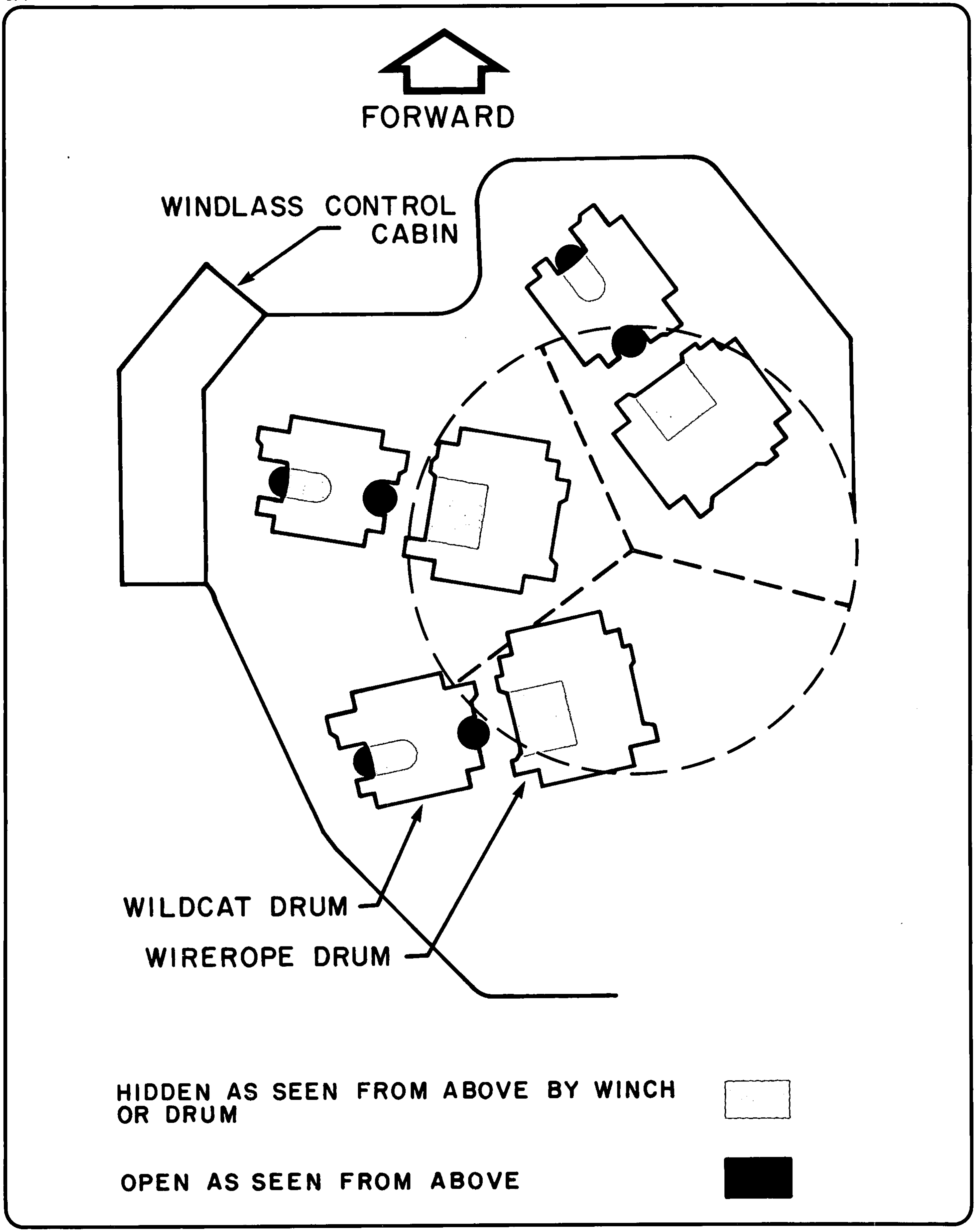
OCEAN RANGER

GENERAL ARRANGEMENT OF QUARTERS
THIRD FLOOR ELEV. 155' 0" (47·2 m)




200' 0"
(61·0 m)
FORWARD
PC 1
SC 1
SACK
STORAGE
AREA
CEMENT
STORAGE
REFRIG. &
AIR COND.
ROOM
CREW
QUARTERS
78' 0"
(23·8m)
PC 2
SC 2
70' 0"
(21·3 m)
MUD
PITS
MUD
PUMPS
CELLAR
DECK
MOON
POOL
WELL TEST
EQUIPMENT
STORAGE
PC 3
D.W.
S.W.
P.W.
F.O.
F.O.
F.O. (Below)
SC 3
78' 0"
(23·8m)
EMERGENCY
GENERATOR
ROOM
ELECTRICAL
ROOM
MACHINE
ROOM
MECHANICAL &
ELECTRICAL
STORAGE
MAIN GENERATOR ROOM
PC 4
SC 4


345°
15°
315°
45°
285°
75°
255°
105°
225°
135°
195°
165°


FORWARD
WINDLASS CONTROL CABIN
WILDCAT DRUM
WIREROPE DRUM
HIDDEN AS SEEN FROM ABOVE BY WINCH OR DRUM
OPEN AS SEEN FROM ABOVE