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AAI REPORT NO. R60011-00013

AUTOMOTIVE TEST RIG FINAL DESIGN REPORT

JANUARY 1986

VOLUME I – DESIGN AND FABRICATION

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prepared for

U.S. NAVY DAVID TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER
CODE 1240, MARINE CORPS PROGRAMS OFFICE
BETHESDA, MARYLAND 20084

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This report documents the design and fabrication of a tracked automotive test rig vehicle developed to investigate the feasibility of advanced automotive components for potential use in future high water speed amphibians. Advanced components included in the design are a hydrostatic drive train for land and water propulsion, a hydropneumatic suspension system, two speed final drives and micro computer control systems.

Volume I includes detailed descriptions of the various test bed major systems and components along with engineering analyses to support their design development. Characteristics of major test bed elements are defined along with overall test bed characteristics. Estimated land and water performance and weight and mass properties are also provided for the test bed

Volume II provides the details of the hydrostatic drive train control system.

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1.0 INTRODUCTION

1.1 Automotive Test Rig Program

1.1.1 Background

Under the provisions of the National Security Act of 1947, the United State Maine Corps is authorized to pursue the development of unique equipment that is peculiar to their primary mission of conducting the amphibious assault. In this regard and with the technical assistance of the United States Navy, the Marine Corps has fielded numerous, successful, assault amphibians in support of their global commitment to the projection and forcible entry of naval forces ashore.

The assault amphibian vehicle (generally known as the Landing Vehicle Tracked or LVT) represents various technological challenges due to the dual requirements for high levels of both waterborne and land performance with ever increasing emphasis placed upon survivability and mobility aspects. In view of these seemingly divergent criteria, it has become a prudent effort on the part of amphibian developers to explore new and emerging technologies. Using exploratory development funds, Navy and Marine Corps technical personnel, with direct access to Government laboratory assets and industry, are able to concentrate their efforts on the development of vehicle components which will eventually enhance amphibian capabilities. This important program is conducted under the auspices of the Marine Corps Surface Mobility Exploratory Development program.

1.1.2 Scope

The Automotive Test Rig (ATR) has been developed to demonstrate the technical feasibility of automotive systems and components which may be employed in the Marine Corps' Advanced Light Armored Amphibious Vehicle System (ALAAVS) Technology Demonstrator. The advanced automotive systems and components of primary significance to the ATR are:

1. A hydrostatic, microcomputer controlled propulsion system;
2. Lightweight, wire-link, band track;
3. Hydropneumatic suspension units which permit full retraction of the track and suspension components during seaborne operations;
4. Two-speed final drives (with integral service brakes) which extend the hydrostatic propulsion system speed and torque characteristics.

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This report documents the design and fabrication of the ATR during the period 25 January 1984 to 19 December 1985. The report is presented in five volumes: Volume I, Design and Fabrication; Volume II, Control System; and Volumes III A,B and C, Engineering Drawings.

1.2 Automotive Test Rig Configuration

External views of the ATR are presented in Figures 1.2-1, 1.2-2, 1.2-3 and 1.2-4. The ATR is 207.25 inches in length (5264mm), 106 inches in overall width (2692mm) and has a top deck height of 78 inches (1981 mm). The 17 inch wide wire-link track provides a 115 inch (2921mm) contact length and the 88.5 inch (2248mm) track centerline distance yields an L/T ratio of 1.3:1. At a test weight of 28,000 pounds (12,700 kg) the ATR develops an average ground pressure of 7.2 psi (49.6 kPa).

The hull is constructed of aluminum alloy 5083. Ballistic welds are not employed. The majority of the interior materials are aluminum alloy 6061. Steels, where used, are stainless or coated alloys and all fasteners incorporate protective coatings.

The interior arrangement of the ATR is presented in Figure 1.2-5. The driver is positioned on the port side of the forward glacis region and the commander is located on the starboard side. With the crew hatches closed the driver and commander are seated in a semi-supine posture. In the open-hatch mode the crewmen are elevated such that their heads are above deck for the purpose of maximum visibility. The seats provide adequate fore and aft adjustment and may be positioned at any point between maximum and minimum vertical travel. The seat geometry is configured to maintain a constant operator hip-to-pedal distance during vertical transitions.

The power plant is positioned in the bow between the driver and commander and is centered on the vehicle's longitudinal axis. The engine (Caterpillar 3208T, 320 brake horsepower) drives the primary hydrostatic propulsion system pumps by means of a directly coupled transfer gearbox. Electrical power is provided by a 28V, 300A Niehoff alternator, which is belt driven from the forward end of the engine, and has been specifically designed for the marine environment. The auxiliary system hydraulic pumps are mounted on the aft drive pads of the primary hydrostatic pumps. The cooling system heat exchangers (engine coolant and hydraulic oil) are located immediately above the engine. The cooling system fans (hydraulically driven mixed flow

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Figure 1.2-1 External View of ATR Port Side

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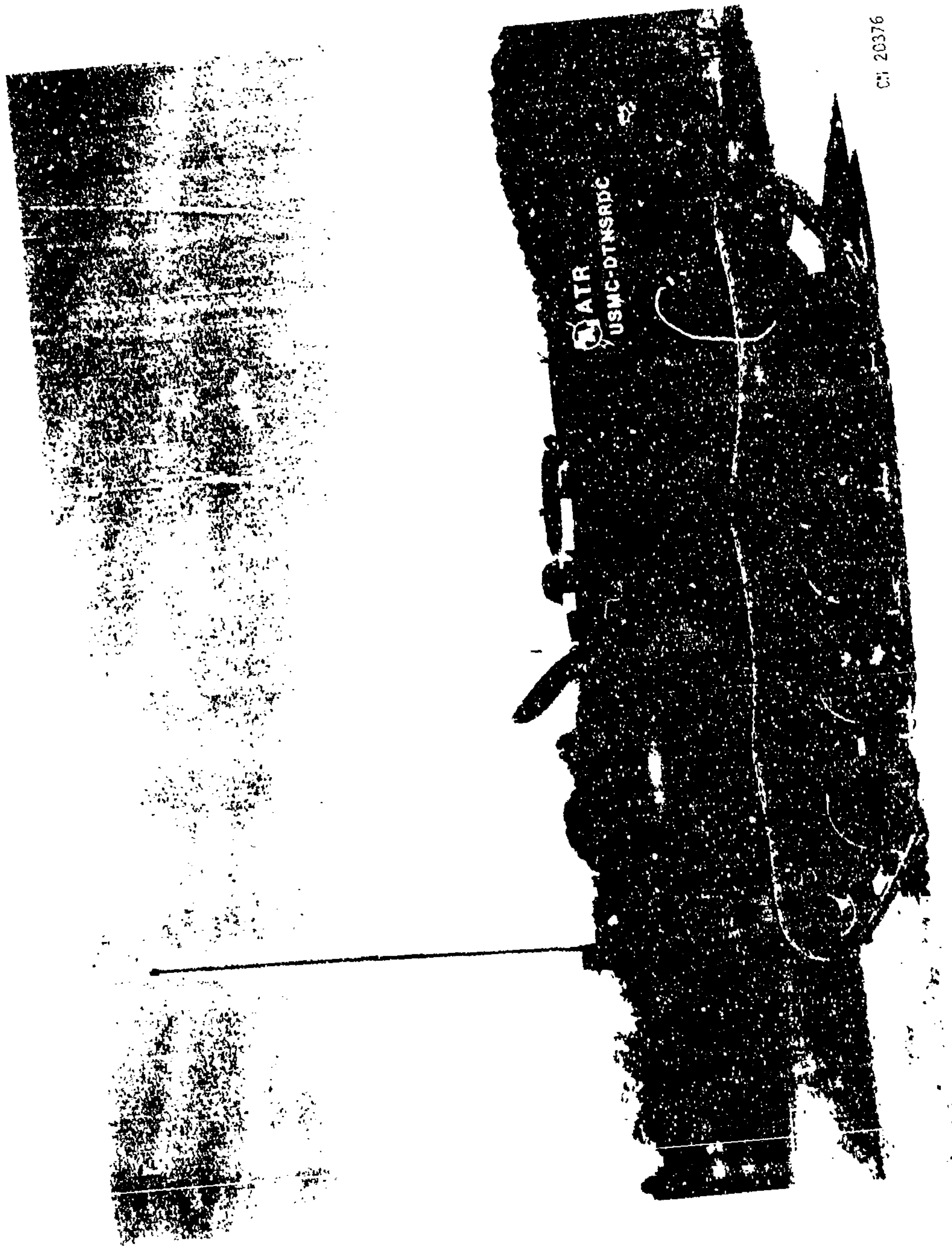

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Figure 1.2-2 External View of ATR Starboard Side

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Figure 1.2-3 External View of ATR Bow

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Figure 1.2-4 External View of ATR Stern

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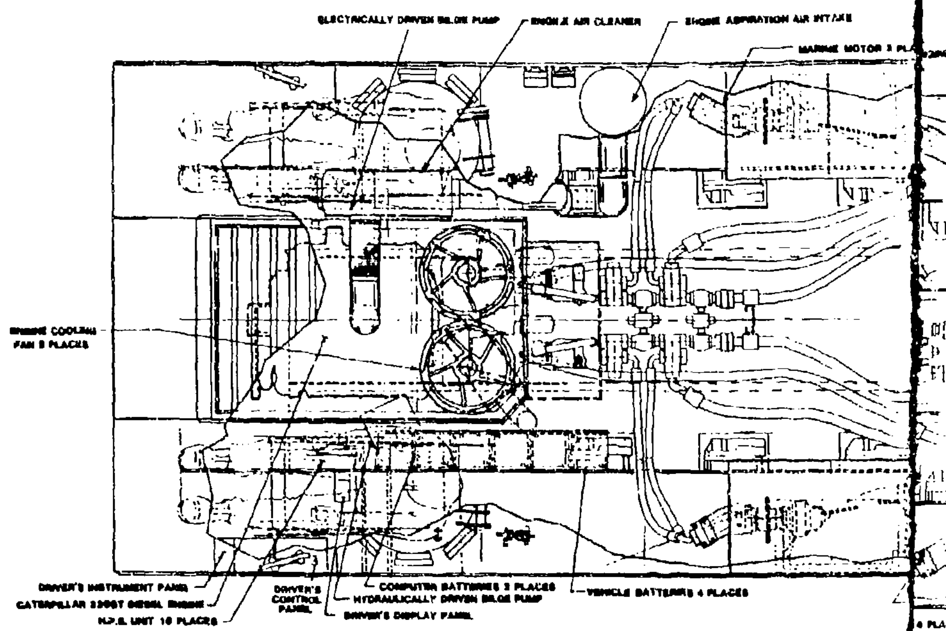

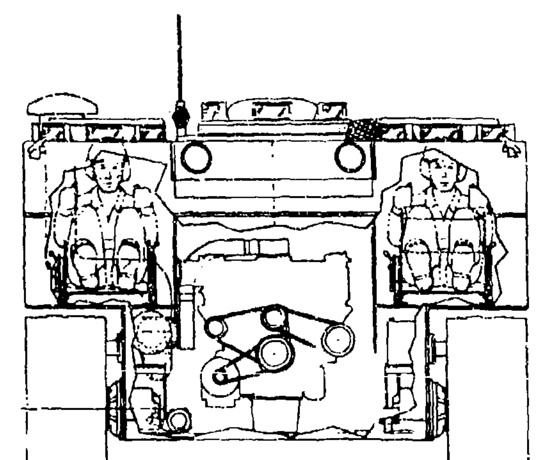

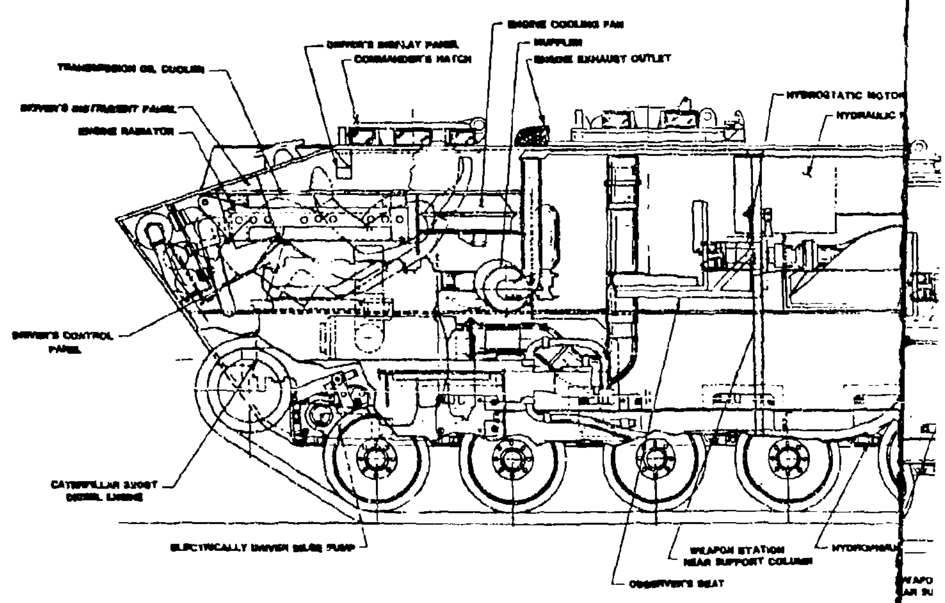

INTERNAL VIEW (STARBOARD SIDE)

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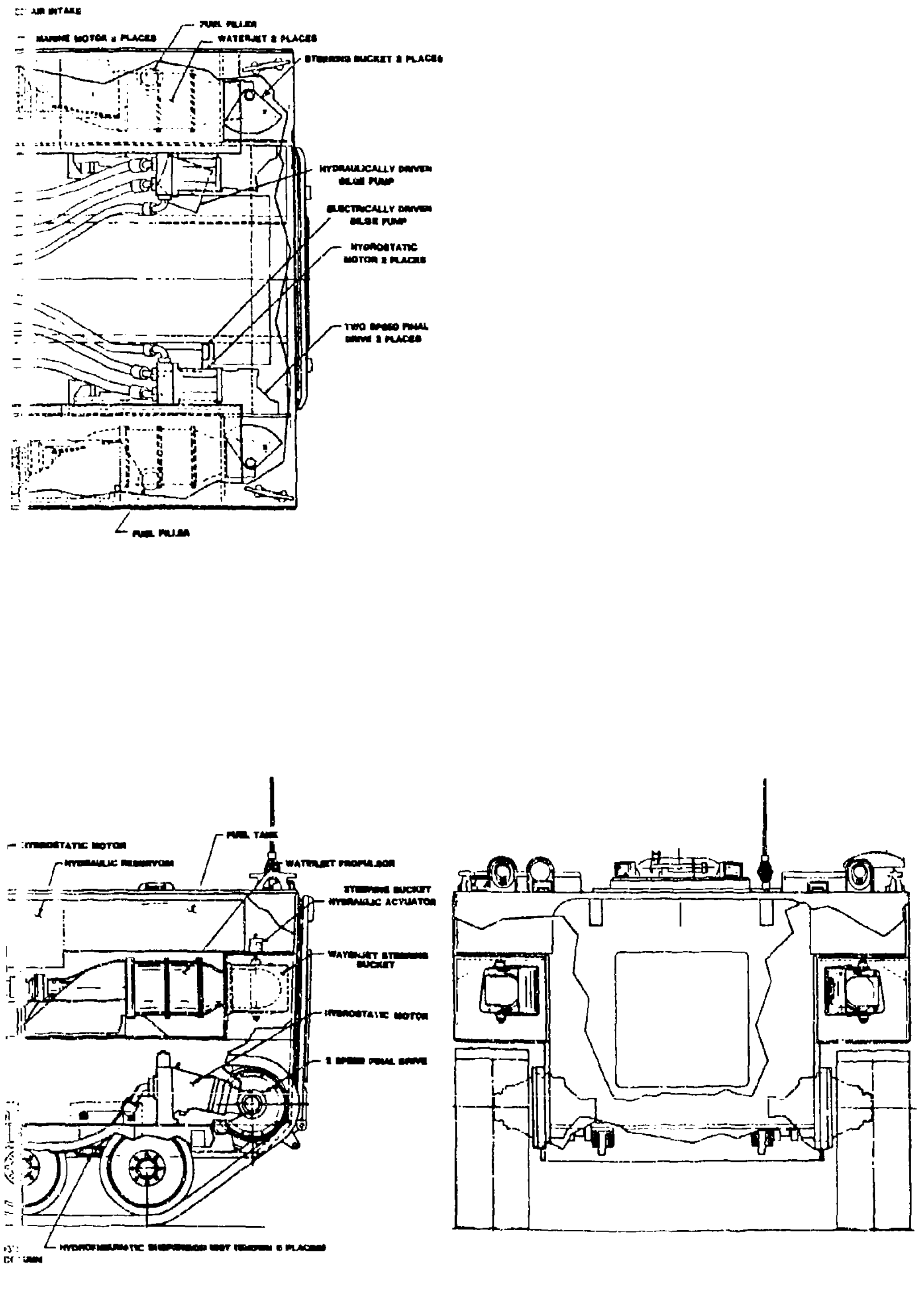

Figure 1.2-5 ATR Internal Arrangement

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units) are horizontally mounted aft of the engine. During landborne operations the airflow path is through the deck mounted, hydraulically actuated louvers, through the heat exchangers, to the fans and out the aft region of the deck louvers. In the seaborne mode the louvers are closed and cooling is provided by a pumped seawater system with a single cooling fan operating at reduced speed and venting through a modified louver to provide engine compartment and troop space ventilation. In both land and sea operating modes, the engine compartment is below atmospheric pressure to preclude the possibility of engine exhaust gases and fuel/lubricant vapors entering the crew and troop spaces. The engine-gearbox-pump assembly is hard mounted in a three-point arrangement with mounts on the port and starboard sides of the gearbox and a mount on the forward end of the engine. The deck region above the engine compartment can be pivoted up and forward for topside access to the power plant (the cooling system heat exchangers are also pivot mounted), and a removable panel on the lower glacis provides access to components mounted on the front of the engine. The engine compartment is totally enclosed by means of an insulated, aluminum, firewall which incorporates engine and hydraulic component access panels. The engine aspiration/filter system is positoned on the starboard side immediately below the commander's position (the aspiration system includes a seawater shut-off mechanism), and the exhaust system is placed in the port side region of the power plant enclosure, aft of the driver's compartment.

An observer's station is positioned aft of the engine compartment. A one-meter diameter opening on the top deck is provided to accommodate the planned addition of a remotely operated, twenty-five millimeter, chain gun weapon station. The present configuration consists of a cupola and seat for the observer, and a top deck reinforcing ring with column supports. support structure and deck reinforcement have been designed for wea. stations on the order of three-thousand pounds (1361 kg).

Two, fifty gallon (189 l) fuel tanks and two, twenty-five gallon (96 l) hydraulic oil reservoirs are positioned in the port and starboard sponsons. The fuel tanks are constructed of stainless steel. The fuel tank vents are external to the vehicle thereby eliminating the accumulation of fuel vapors within the troop space. The hydraulically driven seaborne propulsors (two-stage, twelve-inch (300 mm) diameter waterjets) are located directly

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below the fuel tanks in the port and starboard sponsons. Steering is accomplished by means of hydraulically actuated deflectors which are positioned just aft of the waterjet outlet nozzles. The waterjet inlets are located on the lower surfaces of the sponsons directly above the tracks. The maximum forward water speed of the ATR (with the tracks retracted) is six miles-per-hour (9.6 km/h).

The two-speed final drives with variable displacement hydrostatic motors are positioned on the lower hull side plates, at the stern. The hydraulic pressure and return lines to the final drives and the waterjets are routed below the bilge covers to preclude personnel injury in the event of a primary line rupture. Aluminum shields and fabric protective liners are additionally provided to enclose all hydraulic propulsion system lines above the bilge covers. The maximum forward land speed of the ATR is forty miles-per-hour (64.4 km/h). A hydraulically actuated personnel and equipment ramp is positioned in the stern plate. The ramp incorporates a smaller, manually operated personnel hatch for emergency egress.

The hydropneumatic suspension units are positioned on the lower hull side plates as shown in Figures 1.2-1 and 1.2-2. The suspension geometry provides 16.5 inches (419 mm) of roadwheel jounce (upward travel measured vertically) and 4 inches (102 mm) of rebound (travel below the static position measured vertically). The suspension is provided with a compensating idler mechanism which additionally serves to take up excessive track sag when the suspension units are retracted for seaborne operation.

A major safety feature of the ATR is the provision of an automatic, Halon® fire detection and suppression system. The system is supplied by Santa Barbara Research Inc., and represents the state-of-the-art in detecting and extinguishing explosive fuel and oil fires.

The automotive control system is functionally portrayed in Figure 1.2-6. The microcomputer and controls software developed by Southwest Research Institute form the nucleus of the control system. The microcomputer (SwRI Model SC-1) receives the operator input signals (steering, speed, range selection, etc.) and feedback signals from the propulsion system components (engine, motor and sprocket speeds, pump and motor displacement and pressure, etc.) and, through numerous calculation schemes, schedules the system components for optimum performance levels.

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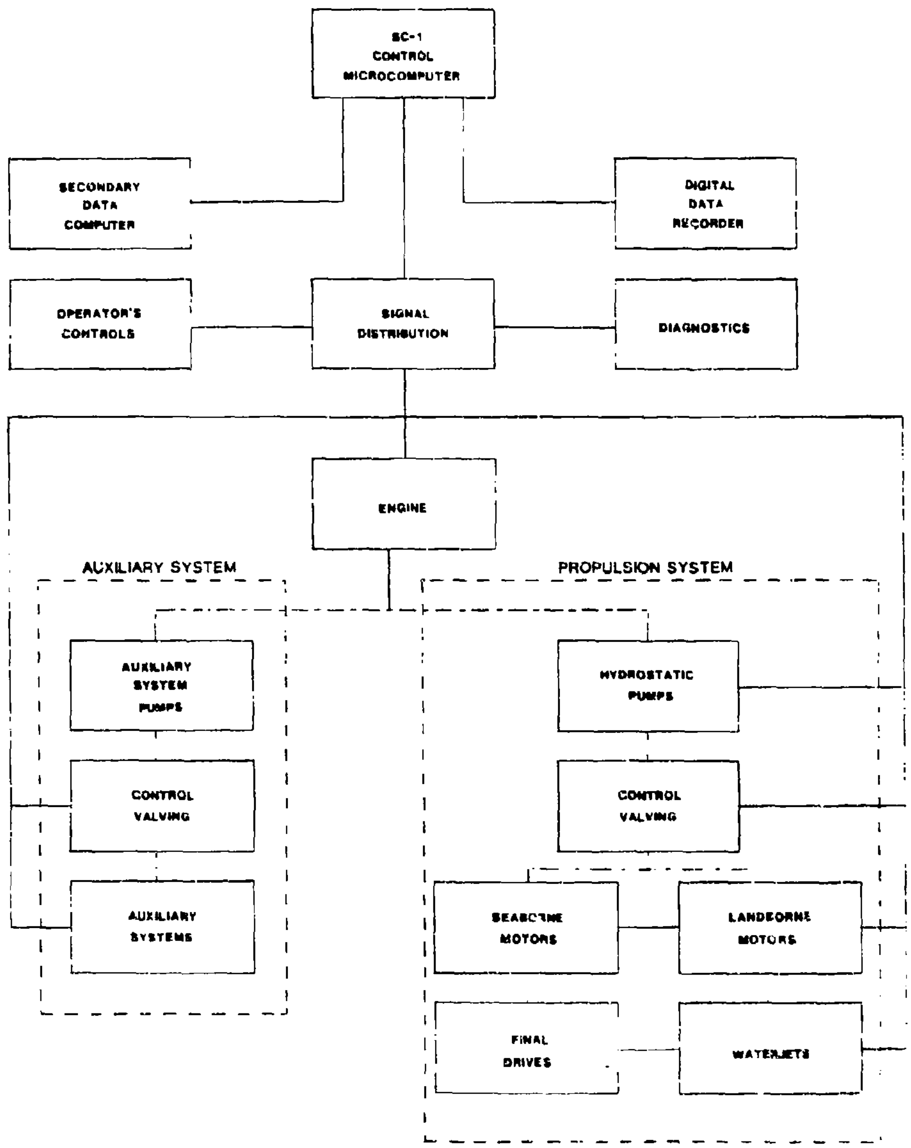

Figure 1.2-6 Automotive Control System

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The computer-based control system also incorporates a digital data recorder for automotive performance analyses, a secondary computer for real time monitoring of selected propulsion system components, an analog backup controller which provides an effective "get home" capability in the event of a primary control system failure and, an electronics diagnostics device.

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2.0 HULL AND FRAME

2.1 General Arrangement

This section contains a general overview of the hull and frame configuration of the ATR and also provides a basic description of some of the features of this structure. A general layout of the hull configuration is presented in Drawing No. 60011-10002, "ATR Overall Dimensions", found in Volume III of this report.

2.1.1 External Structure

An exploded view of the external hull structure major components is presented in Figure 2.1.1-1. All of the components are flat, 5083 aluminum alloy plates. No forming operations are required for the fabrication of these plates.

The majority of the hull structural area is composed of the top plate, the sponson sides, the sponson bottoms, the hull sides, and the bottom plate. Two sloping glacis plates form the ATR nose. Skid plates are located at the forward and rear ends of the hull bottom plate to provide a transition between the horizontal hull bottom and the nearly vertical lower front glacis and hull rear plates. The use of these plates eliminates the formation of sharp corners, which would be vulnerable to damage from obstacles.

The center section of the top plate profile is extended forward, above the slope of the upper front glacis, to accommodate the cooling air inlet/exhaust grille.

The grille area extends rearward along the hull top deck, between the commander's and driver's hatch locations. These hatches are positioned on the outside areas of the top deck, immediately behind the line of intersection between the upper front glacis and hull top plate. The hatches used are standard M113A1 APC driver's hatches. Providing hatches for the ATR was accomplished by removing plate sections from an M113A1. These plate sections included the hatches and vision block mounts. The ATR top plate was cut out to accommodate the hatch area plates.

Immediately aft of the hatches and grille area is a 39.37 inch diameter circular cutout centered laterally on the top plate. In the initial ATR configuration, this area is utilized as an observer's station. A six inch high turret ballast ring is bolted on the upper surface of the top plate, about the opening. An observer's cupola is bolted on top of the ballast ring.

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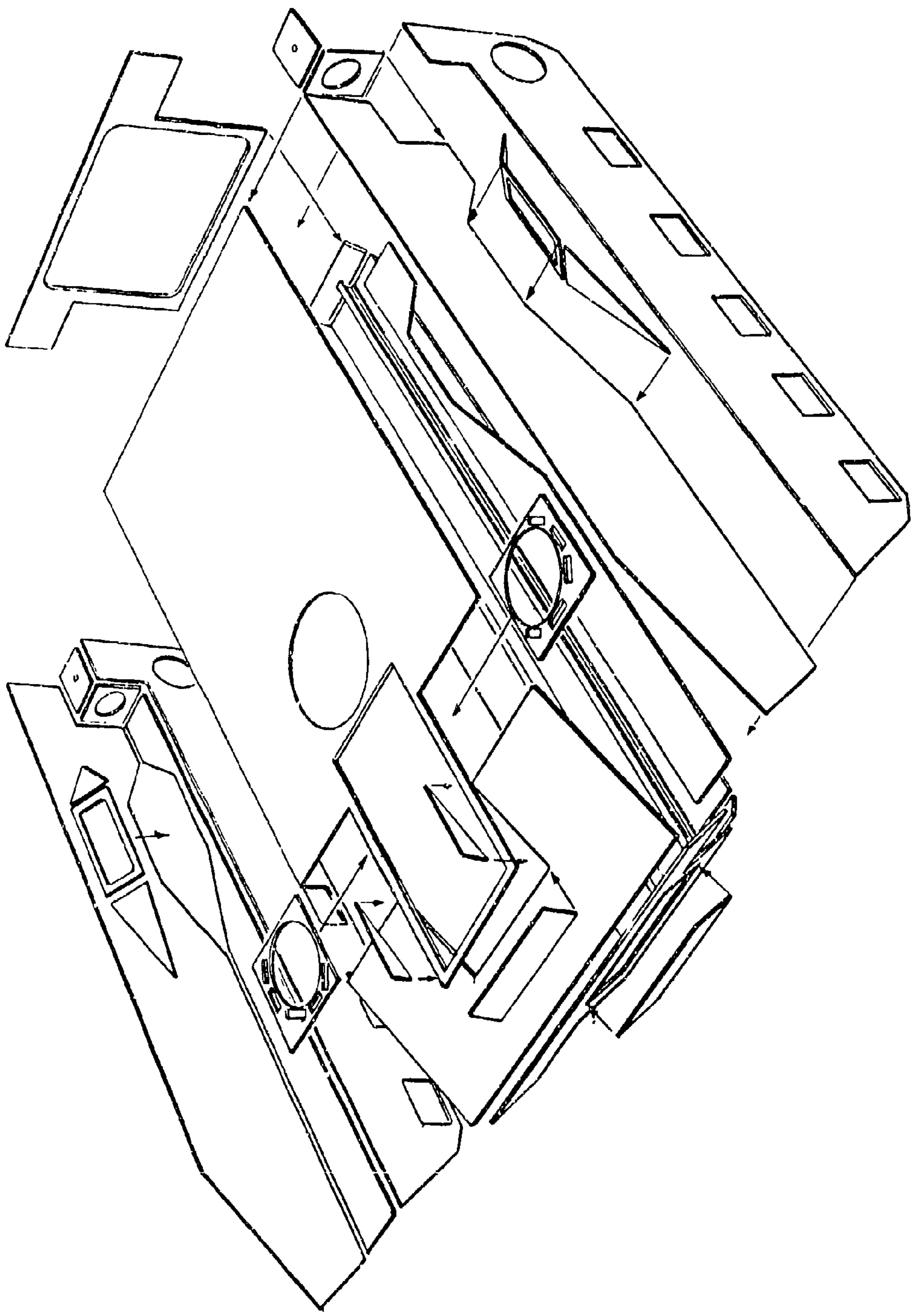

Figure 2.1.1-1 Exploded View of ATR Hull

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The cupola used is the commander's cupola from an M113A1 APC. In the future, the ballast ring and cupola may be removed for the mounting of a weapons station on the ATR.

Features on the ATR sides include the waterjet inlet and outlet port areas. Each of these ports is configured of three small plates that are integral to the hull weldment. The machining of these plates, necessary to accomplish mounting of the waterjets, is performed following welding, as explained in Section 2.3. Guards are bolted to the hull rear about the waterjet outlet ports to prevent damage occurring to the rear of the waterjets.

Also included on the hull sides are several mounting plates whch accommodate the various suspension components mounted exterior to the hull. These plates provide the proper lateral positioning of the suspension components, and also add strength to the most highly stressed areas of the lower hull. Again, these plates are welded in place prior to being final machined.

The hull rear plate is inclined inward from the top at a 2° angle. A large cutout is included in the rear plate. This cutout accommodates a rear ramp and accompanying escape hatch, as discussed in Section 2.6.

2.1.2 Internal Structure

The internal structure of the ATR hull consists primarily of three elements, the weapons station support structure, the firewall, and the floor and accompanying floor structure. Each of these elements is detailed in the following sections. See Drawing No. 60011-40230, "ATR Hull Internal Arrangement" for an overall view of these elements.

2.1.2.1 Weapons Station Support Structure

The weapons station support structure consists of a support ring and three vertical support columns, as illustrated in detail on Drawing No. 60011-40232, "Turret Support Structure". This structure is included on the ATR to provide support of the top deck in the event a weapons station is mounted in the ATR.

The support ring, one inch high by three inches thick in cross section, is welded to the underside of the hull top deck, about the circumference of the weapons station cutout. This ring serves to increase the

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cross-sectional area of the top plate in the immediate region of the weapons station and thus provide support against firing loads and both lateral and longitudinal road shock loads.

The three vertical columns are mounted between the underside of the support ring and the hull floor. They provide resistance to vertical road shock loads, as well as to firing loads imposed by an elevated weapon. The forward two columns are cylindrical, with a two inch outer diameter and a 0.25 inch thick wall. The rear column is a "C" channel. In the initial ATR arrangement, this column is utilized to mount the observer's seat.

2.1.2.2 Firewall

A 0.188 and 0.25 inch thick aluminum firewall encloses the engine compartment. The firewall is divided into two sections. The main firewall section spans longitudinally from the front of the ATR to a point immediately behind the engine block. Vertically, this portion of the firewall extends from the ATR ceiling to the level of the sponson bottoms, on either side of the engine. At the bottom point, the firewall sides extend outward and adjoin the inner edges of the sponson bottoms. The rear part of this firewall section extends downward to the ATR floor. An extension is built rearward from the lower part of the main firewall, to encase hydraulic pumps mounted to the rear of the engine. This extension protrudes beneath the weapons station opening, but is sufficiently low so as not to interfere with the observer's station or with a weapons station, should one be installed.

2.1.2.3 Floor and Floor Structure

The ATR floor is located six inches above the inside surface of the hull bottom plate. It is comprised of three sections of 0.25 inch aluminum diamond tread plate material. Each of the floor sections is lightweight and of sufficiently small size so that it may be readily removed in the event that access must be gained to the bilge area below it.

The floor is supported by the two main channels which run fore and aft over the entire length of the ATR. In addition to supporting the floor plate, these channels provide the base for the engine mounting structure. Only minimal fastening of the floor sections is required, to prevent movement during operation over rough terrain.

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2.2 Hull Weldment

The hull weldment drawing, Drawing No. 60011-40201, is presented in Volume IIIA, page 2.6 of this report. The component part drawings of the weldment are presented on succeeding pages of Volume IIIA. A list of the component parts and their corresponding drawing numbers is provided in Table 2.2-1.

Table 2.2-1 Component Parts of the ATR Hull Weldment and Corresponding Drawing Numbers

Component Part	Drawing Number
Top Plate	60011-40202
Upper Front Glacis	60011-40203
Lower Front Glacis	60011-40204
Front Skid Plate	60011-40205
Hull Bottom	60011-40206
Rear Skid Plate	60011-40207
Rear Plate	60011-40208
Sponson Bottom	60011-40209
Sponson Side	60011-40210
Hull Side	60011-40211
Forward Waterjet Mounting Plate	60011-40212
Upper Waterjet Mounting Plate	60011-40213
Inner Waterjet Mounting Plate	60011-40214
Final Drive Mounting Ring	60011-40215
Hydropneumatic Unit Mounting Plate	60011-40216
Channel Floor/Engine Support	60011-40217
Waterjet Inlet Mounting Plate	60011-40218
Waterjet Inlet Rear Transition Plate	60011-40219
Waterjet Inlet Forward Transition Plate	60011-40220
Radiator Cover Side Plate	60011-40221
Radiator Cover Front Plate	60011-40222
Weapon Station Support Ring	60011-40223
Weapon Station Forward Support Columns	60011-40224
Weapon Station Rear Support Column	60011-40225

The material used for the hull weldment is MIL-A-46027, 5083 aluminum alloy armor plate. The majority of the plates are 0.75 inches thick. Exceptions include the front skid plate, hull bottom, and rear skid plate, which are 1.125 inches thick, and the hull sides, which are 1.25 inches thick. These plates form the lower part of the hull structure and are, therefore, subject to severe shock loads from the suspension system, necessitating their

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greater thickness. Another exception is the sponson bottom plates, which are 0.50 inches thick.

As the ATR does not require ballistic welds, all weld joints are either constructed of fillet welds, groove welds, or a combination of the two. Any machining operations necessary to prepare the component plates for welding are performed during the plate manufacturing process so that no machining is required during the welding procedure. Several of the weldment plates do require such a process, which generally entails cutting an angle on a plate edge so that there is sufficient space for a welding rod when that edge is joined to another plate edge.

Note that some of the larger hull component plates are either longer or wider than the available stock sizes of aluminum plate. These plates are fabricated by joining smaller plates with butt welds, as shown on the drawings of these plates. These butt welds are done per MIL-STD-21A, number 3 type weld. Plates manufactured in this manner are the top plate, lower front glacis, hull bottom, rear plate, sponson bottom, sponson side, and hull side.

All welds are done via a gas metal-arc process, using aluminum alloy 5356 as a filler material. Inspection of the welds is accomplished with a penetrant method as per MIL-I-6866.

Special plates included on the hull weldment are the hydro-pneumatic suspension unit mounting plates, the front idler mounting plates, the track support roller mounting plates, the bump stop mounting plates, and the final drive mounting rings. Each of these parts is welded to the hull side and then final machined, as described in Section 2.3 of this report, to provide a mounting surface for an item that is bolted to the hull. In addition to providing a mounting surface, these plates serve to increase the strength of the hull side in areas of high stress and, with the exception of the bump stop mounting plates, provide the proper positioning of the components mounted to them with respect to the track.

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2.3 Hull Machined

This section contains descriptions of the machining processes that are performed on various areas of the ATR hull, following welding. The waterjet inlet and outlet plates were machined prior to welding. The hull machining drawing, Drawing No. 60011-40200, is presented in Volume IIIA, page 2.1.

2.3.1 Suspension Component Mounting Surfaces

Areas machined for the mounting of suspension components include the final drive mounting rings, the hydropneumatic suspension unit mounting plates, the front idler mounting plates, the track support idler mounting rings, and the bump stop mounting pads. Two individual machining operations are performed on all of these areas. The first of these operations is to face machine the mounting surface so that the component mounted to it is positioned the correct lateral distance from the hull and is parallel to the other suspension components. The second operation is to drill a hole pattern that is identical to the mounting bolt pattern of the component into the mounting surface. Rosan inserts are placed into these holes to provide anchors for the mounting bolts. All machining operations of the individual suspension components are listed below.

2.3.1.1 Final Drive Mounting Ring

1. Face machine mounting surface on a plane that is parallel to a vertical plane through the longitudinal centerline of the hull. This operation is critical, as the face machining of all other suspension component mounting surfaces is based on this surface. The nominal thickness of the ring after machining is 1.75 inches.

2. Machine the 16.0 inch diameter hole to finished size. This hole is also cut through the hull side plates at this time.

3. Drill mounting hole pattern of eighteen holes to accommodate 1/2-20 locking ring type Rosan inserts.

2.3.1.2 Hydropneumatic Suspension Unit Mounting Plates

1. Face machine mounting surfaces (5 per side) parallel to the final drive mounting surface. The nominal thickness of the plates after machining is 0.5 inches.

2. Machine out center opening of plates to finished size. These openings are also cut through the hull side plates at this time.

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3. Drill mounting hole pattern of ten holes to accommodate 5/8 - 18 Rosan slimserts.

2.3.1.3 Idler Mounts

1. Face machine mounting surface parallel to the forward hydropneumatic suspension unit mounting surface.

2. Machine the pilot boss, which provides for accurate alignment of the idler support spindle during mounting, to finished size.

3. Drill mounting hole pattern of eight holes to accommodate 3/4-16 locking ring type Rosan inserts.

2.3.1.4 Track Support Roller Mounting Rings

1. Face machine mounting surfaces (2 per side) parallel to the nearest hydropneumatic suspension unit mounting surface. The nominal thickness of the rings after machining is 0.5 inches.

2. Drill mounting hole pattern of six holes to accommodate 1/2-20 locking ring type Rosan inserts.

2.3.1.5 Bump Stop Mounting Pads

1. Face machine mounting surface parallel to mounting surface of #4 hydropneumatic suspension unit. The nominal thickness of the pads after machining is 0.5 inches.

2. Machine 0.75 inch wide shear key groove into pad.

3. Drill mounting hole pattern of four holes to accommodate 1/2-20 locking ring type Rosan inserts.

2.3.2 Waterjet Mounting Surfaces

Both the inlet and outlet waterjet mounting areas required machining of the plates prior to welding. During welding these plates were held in place by a fixture which duplicated the waterjet mounting hole locations. The machining details of each area area listed below.

2.3.2.1 Waterjet Inlet Area

1. Machine the inlet opening into the waterjet inlet mounting plate.

2. Drill and chamfer a seventeen hole pattern, matching the mounting bolt pattern of the waterjet inlet, about the inlet opening.

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2.3.2.2 Waterjet Outlet Area

1. Machine the 8.5 inch diameter outlet opening into the waterjet outlet mounting transome, using the waterjet centerline as a reference to obtain the outlet openings' true position.

2. Drill and prepare a four hole pattern, matching the mounting bolt pattern of the waterjet rear, about the outlet opening.

2.3.2.3 Waterjet Inlet and Outlet Positioning

A steel fixture which duplicated the waterjet, including inlet and outlet mounting hole locations, was fabricated. The inlet and outlet plates of the hull structure were machined and bolted to the fixture prior to welding these plates into the hull structure. Correct alignment of these plates was achieved by employing this method.

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2.4 Personnel Accommodations

The personnel accommodations include seating arrangements for t.e ATR driver and commander which are similar to the HSTV(L), RDF-LT, and M1 driver's stations. For the observer's station located at the weapon station mount, a seating arrangement similar to the RDF-LT, 76mm turret is employed.

2.4.1 Driver's and Commander's Stations

The driver's and commander's stations are configured to accommodate the 5th and 95th percentile crewman. Each is seated in a semi-supine position as shown in Figure 2.4.1-1. The seat assembly is similar to the RDF-LT driver seat with several improvements:

1. Several of the seat back adjustments, which are not used on the RDF-LT, have been eliminated. This simplifies seat construction and operation, while reducing the number of parts required;

2. The crewman generates the force required to elevate the seat by pushing his legs and feet forward. The RDF-LT requires the crewman to pull up the seat while raising his weight off of the assembly.

The ATR driver's and commander's seats are supported by cam-rollers which ride inside two "C" channels attached to each of the seat side frames. The front channels are horizontal while the rear channels are inclined at a 30° angle from the horizontal. As the seat is pushed back by the crewman it can be raised 12.5 inches from its lowest position. Locking mechanisms located on both sides of the seat frame allow the operator to set the seat at 11 different positions of elevation. The elevation lock release lever is located on the left side of each seat.

The ATR driver's and commander's seats can also be set at 9 equally spaced fore/aft positions over a 8 inch span. The seat side frames are supported by cam-rollers which ride in two "C" channels which are attached to the bottom of the sponson. The fore/aft lock release is located on the right side of each seat.

Both the fore/aft and elevating mechanisms have spring assists to reduce the effort required to raise or move the seat forward.

A pivot is located at the base of the seat back which allows the back to be folded toward the front or back of the vehicle after released. The seat back release mechanism consists of a "U" shaped spring-loaded plunger mounted on each side frame which engages a cam-roller on each side of the seat

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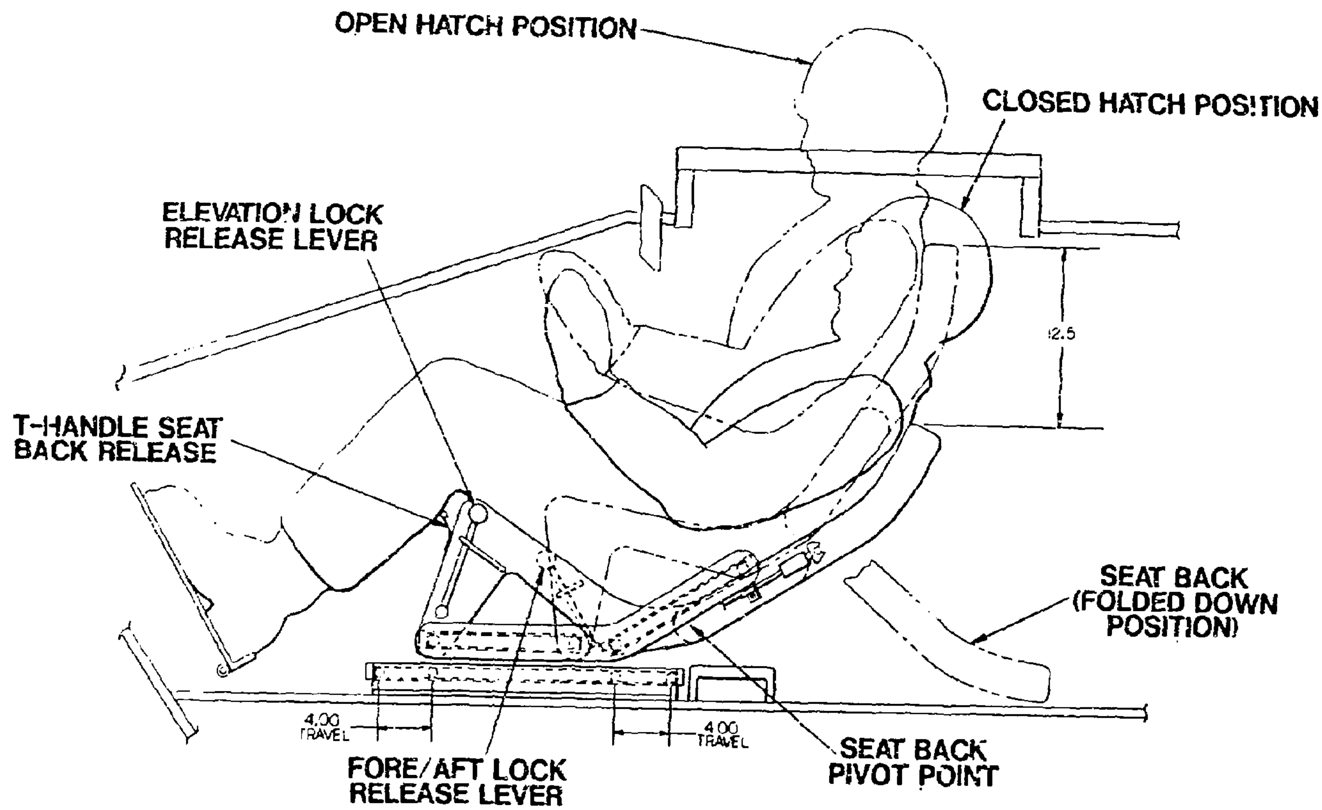

Figure 2.4.1-1 Driver's and Commander's Seat Configuration

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back. A pull handle located at the center front edge of the seat cushion is used to release the plunger assemblies via a cable routed underneath the seat. The seat and seat back cushions will be fabricated from foam rubber pads covered with nylon parachute pack cloth.

2.4.2 Observer's Station

The observer's station is equipped with a seat which is similar to RDF-LT 76mm turret seat. The seat bolts into evenly spaced holes located vertically along the rear support column of the weapon station support structure as shown in Figure 2.4.2-1. It is attached to the column with double cam handle eye bolt and pin which engages a vertical row of evenly spaced holes on the support column. The seat is adjusted in height by removing the double-cam handle eye bolt and spherical washers. The seat is manually lifted or lowered to the desired position. The spherical washers and handle eye bolt are reinstalled. The handle eye bolt is secured by turning until it is snug with the handle pointing to the rear of the vehicle, then folding the handle so that it is parallel to the support column.

The seat cushion and seat back both consist of foam rubber pads, backed by support plates and covered with Naugahyde®. The seat is designed to facilitate its removal from the vehicle. Removal is accomplished by removing the double-cam handle eye bolt and spherical washers, and removing the seat.

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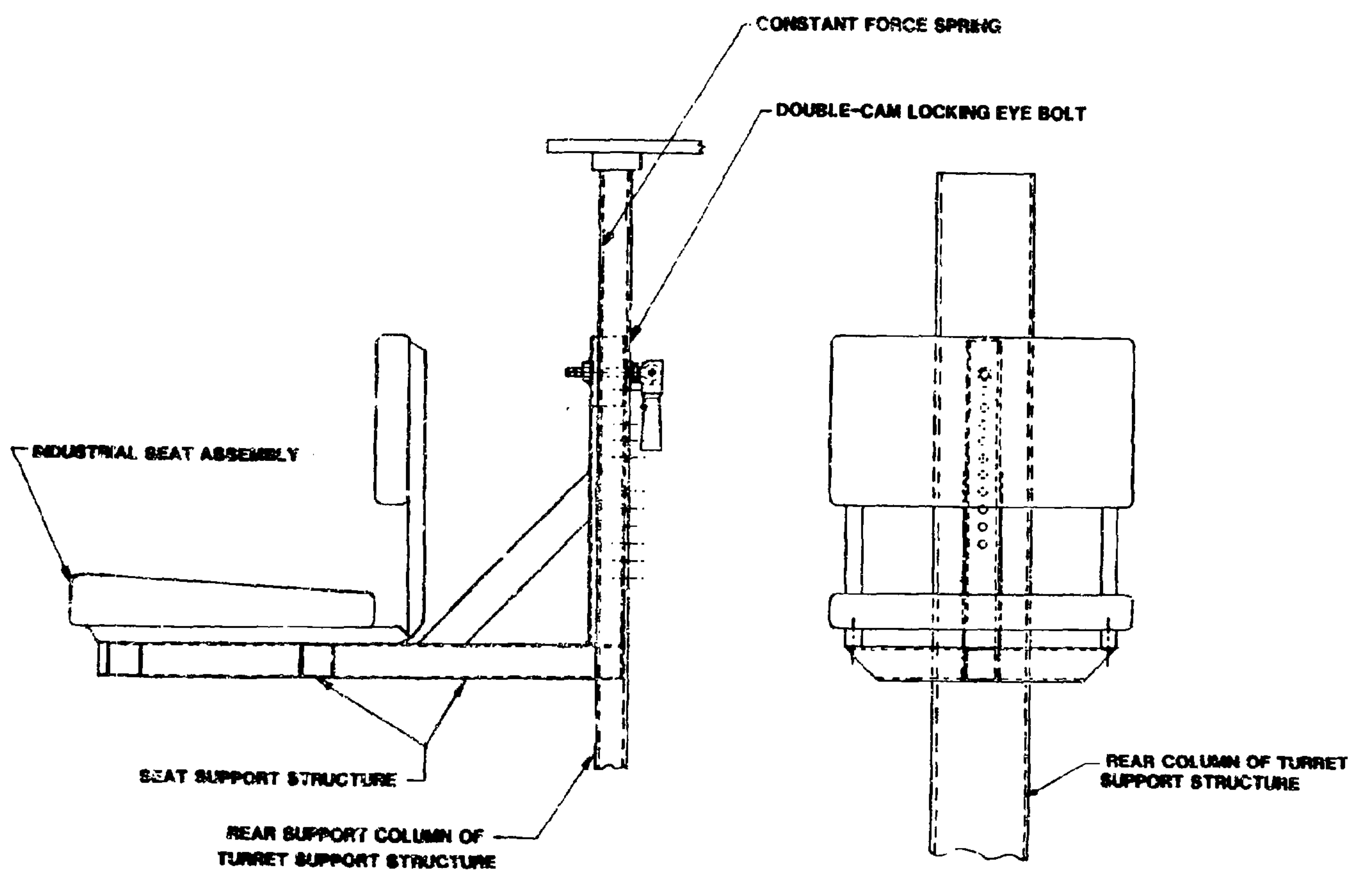

Figure 2.4.2-1 Observer's Seat Configuration

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2.5 Appendages

2.5.1 Towing Lugs

The ATR is equipped front and rear with towing lugs as shown in Figure 1.2-1. The lugs conform to M113 drawing number 8763384. Each lug is bolted to the hull with six 5/8 inch, grade 8 bolts. The center to center spacing is 47 inches at the front and 37 1/4 inches at the rear. This spacing and configuration is compatible with either towing cables or the standard US Army tow bar, medium duty, MS500048. The integrity of the hull structure at the towing lug attachment points has been verified during the final hull stress analysis as discussed in Section 2.8. Reinforcing plates, which are welded into the hull, are employed at each tow lug location.

2.5.2 Lifting Eyes

A combination lifting eye and mooring bitt is attached near each of the corners of the ATR hull top plate. These lifting eye/mooring bitts conform to LVTP7 drawing number 2584217. Each lifting eye is bolted to the hull with four, 3/4 inch, grade 8 bolts. The integrity of the hull structure at the lifting eye attachment points has been verified during the final hull stress analysis as discussed in Section 2.8. Reinforcing plates, which are welded into the hull, are employed at each lifting eye location.

The lifting eyes also permit tiedown of the ATR to the floor (or deck) of the transport medium in such a manner as to prevent shifting or movement in any direction.

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2.6 Hatches and Ramp

This section presents a detailed discussion of the various areas of ingress and egress of the ATR vehicle. These areas include the driver's and commander's hatches, the rear ramp and accompanying escape hatch, and the observer's station.

2.6.1 Driver's and Commander's Hatches

The driver's and commander's hatches of the ATR are adapted driver's hatches from the M113 APC, including the M113 unity vision periscopes. These hatches were chosen over hatch configurations employing "straight through" vision blocks, as these latter configurations require the crewman's eye to be directly in line with the vision block, resulting in the crewmen's head protruding above the top deck of the vehicle. Thus, the crewman would be subject to injury from threats which would be defeated by the hull armor, but would not be defeated by the vision blocks. Examples of such a vision block configuration are shown on Drawing No. 60011-10032, "Hatch Configuration, Vision Block" and Drawing No. 60011-10038, "Flat Clear Armor Vision Block/Hatch Configuration". This consideration, although minor for the initial test bed vehicle, will be of great importance should a combat vehicle based on the ATR design be built. Details of the hatch installation are depicted on Drawing No. 60011-40235, "ATR Hatch Installation".

The hatches are located at the forward edge of the hull top plate, in positions that afford maximum forward and side visibility, but still allow adequate leg room for the crewmen. In conjunction with the adjustable contour seat design being used (see Section 2.4), operation of the vehicle is possible in both open and closed hatch positions. An additional seat position is included to facilitate ingress/egress from the hatch.

2.6.2 Ramp

The rear ramp of the ATR is shown in Drawing No. 60011-40233, "Ramp Installation". The outer dimensions of the ramp are 60.75 inches wide by 52.00 inches high, centered about an opening 54.75 inches wide by 46.00 inches high.

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The ramp is constructed of three layers, each made of aluminum. The outer layer is a .75 inch thick plate, while the inner layer is a 0.25 inch thick diamond tread plate. A layer of 1.50 inch square structural tubing is sandwiched between the inner and outer layers. In addition, two wear pads, a 1.25 inches thick by 6.0 inches wide, are attached to the outside upper corners of the ramp.

Raising and lowering of the ramp is accomplished via a vertically mounted hydraulic rotary cylinder that is attached to the ramp by a roller chain and sprocket system. This system is illustrated in detail on drawing No. 60011-40233, "Ramp Installation". When in the closed position, the ramp is secured by two hydraulically actuated latches. These latches are controlled by a switch on the driver's control panel. A safety switch on the control panel prevents accidental operation of the ramp through the actuator system. The latches are shown on Drawing No. 60011-40233. In addition, limit switch interlocks are utilized to indicate that the ramp is closed and sealed prior to ATR amphibious operation.

A 31.0 inch square emergency escape hatch, with a 26.0 inch square opening, is located near the center of the ramp. Four manually actuated latches are used to secure the emergency hatch. These latches are designed so as to be flush with the tread plate on the inside of the ramp.

2.6.3 Observer's Station

The observer's station of the ATR consists of a commander's cupola of an M113A2 vehicle with vision blocks, which is welded to a mounting ring bolted to the hull top plate. A seat, discussed in Section 2.4, is mounted below the cupola and is adjustable to allow seating during both open and closed hatch observation. The hatch of the cupola provides the crew with an alternate means of ingress/egress, as well as with an alternate escape route.

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2.7 Grille Assemblies

The grille assembly for engine cooling air intake and exhaust is shown in Figure 2.7-1. This assembly is located on the hull top plate between the driver's and the commander's stations and extends forward over the upper front glacis in a section raised to the same level as the hull top plate. The front portion of the grille assembly is used for cooling air intake and the rear portion is used for cooling air exhaust. A louvered closure mechanism which spans the entire area of the grille is provided to seal the assembly during waterborne operations.

The exhaust area is divided into two sections with three louvers covering each section. The port side (driver's side) section covers the primary cooling fan exhaust and the starboard side section covers the secondary cooling fan exhaust. Each of these sections operate independently. The primary fan exhaust louvers open and close with the five air intake louvers and are operated by two hydrualic cylinders and their associated linkages. The secondary fan exhaust louvers are controlled by a separate hydraulic cylinder and linkage. The opening and closing of the grille louvers is controlled by the SC1 computer. The air intake and primary exhaust louvers are open during land mode and closed during transition and sea modes. The secondary exhaust louvers are closed at all times except during land mode when additional cooling is required (coolant temperatures above 200°F or transmission oil temperatures above 140°F). The center louver of the secondary exhaust section has a slotted linkage connected to it and will be raised in transition and sea modes of operation. During these modes of operation the secondary fan is operated in low speed for ventilation of the engine and crew compartments. The ventilation air is exhausted through this louver. A screen assembly is located across the top of the grille to prevent entry of debris or foreign objects.

The grille assembly is mounted at a level of approximately 8 inches below the hull top plate. With the louvers open, weapon depression is not compromised and crew member fields-of-view are not affected. A sloping front plate and two side plates prevent water from entering this recessed area

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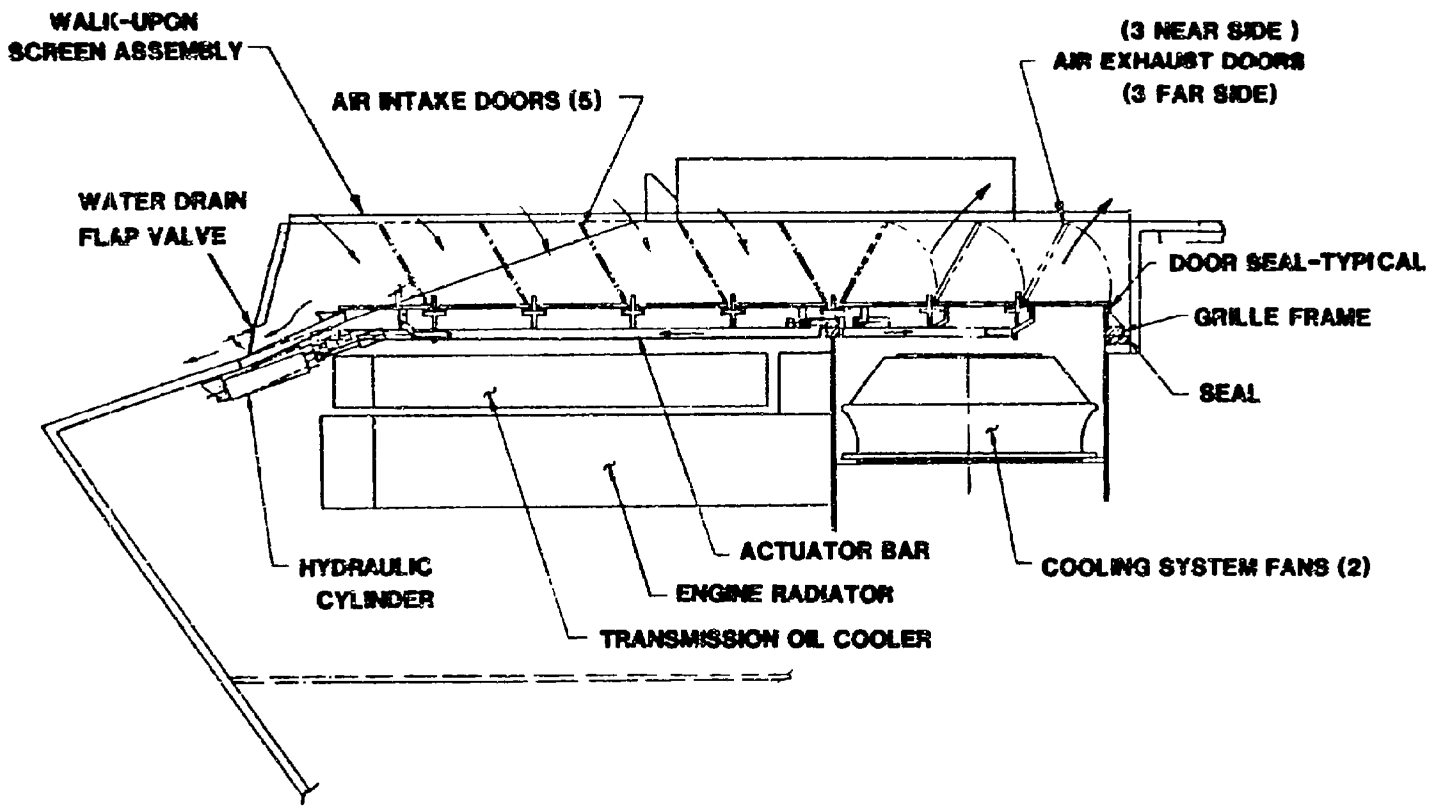

Figure 2.7-1 Grille Assembly With Louver Door Closure Mechanism

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until it exceeds the level of the hull top plate. A one-way flap type valve located along the lower edge of the front plate allows any water trapped in this recessed area to drain forward. This design has a significant advantage, in that, with the louver doors located on top of the grille assembly, any pressure exerted by the presence of water tends to increase the sealing pressure rather than relieve it as is the case of the interior doors used on LVTP7 plenum assemblies.

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2.8 Hull Stress Analysis

This section contains a description of both the analysis methods used and the results obtained from the various stress analyses performed on components of the ATR hull. Many of the analyses presented here were originally discussed in two published reports: "ATR Preliminary Stress Analysis" (R-60011-00003) and "ATR Hull Structural Analysis - Suspension Induced Loads" (R-60011-00005). In addition to discussing the analyses of these earlier reports, this section will present revisions of the original analyses, as well as new analyses related to the ATR hull. Stress analyses relating to the components of the suspension system will be discussed in Section 3.11. Note that the loading conditions examined in the following sections are based on the requirements defined in the report "ATR Design Criteria" (R-60011-00002).

2.8.1 Hull Analysis

This analysis was originally performed in February 1984 to determine the effects of the maximum design road loads on the ATR preliminary hull weldment design. Although the basic hull weldment configuration has been altered somewhat since the analysis was performed, these alterations are not of a magnitude which would substantially effect the results of the analysis. This is especially true at Section C-C (see Section 2.8.1.1.2, below, for an explanation of the cross-sections analyzed), the most critical hull cross-section analyzed, which is essentially unchanged from its initial configuration. A report of this analysis, which was originally presented in R-60011-00003, is given below.

2.8.1.1 Analysis Method

The ATR hull was stress analyzed by the application of various combinations of maximum design roadwheel loads to the hull and the determination of the stresses, at a certain vehicle section, resulting from these applied loads. A drawing showing the location of the applied loads and of the sections at which the stresses were analyzed is presented in Figure 2.8-1.

2.8.1.1.1 Applied Loads

The maximum applied load at each roadwheel station was determined by the method described in Chapter 4 of the Engineering Design Handbook "Automotive Bodies and Hulls" (AMCP No. 706-357, 20 April 1970). The resulting loads were as follows:

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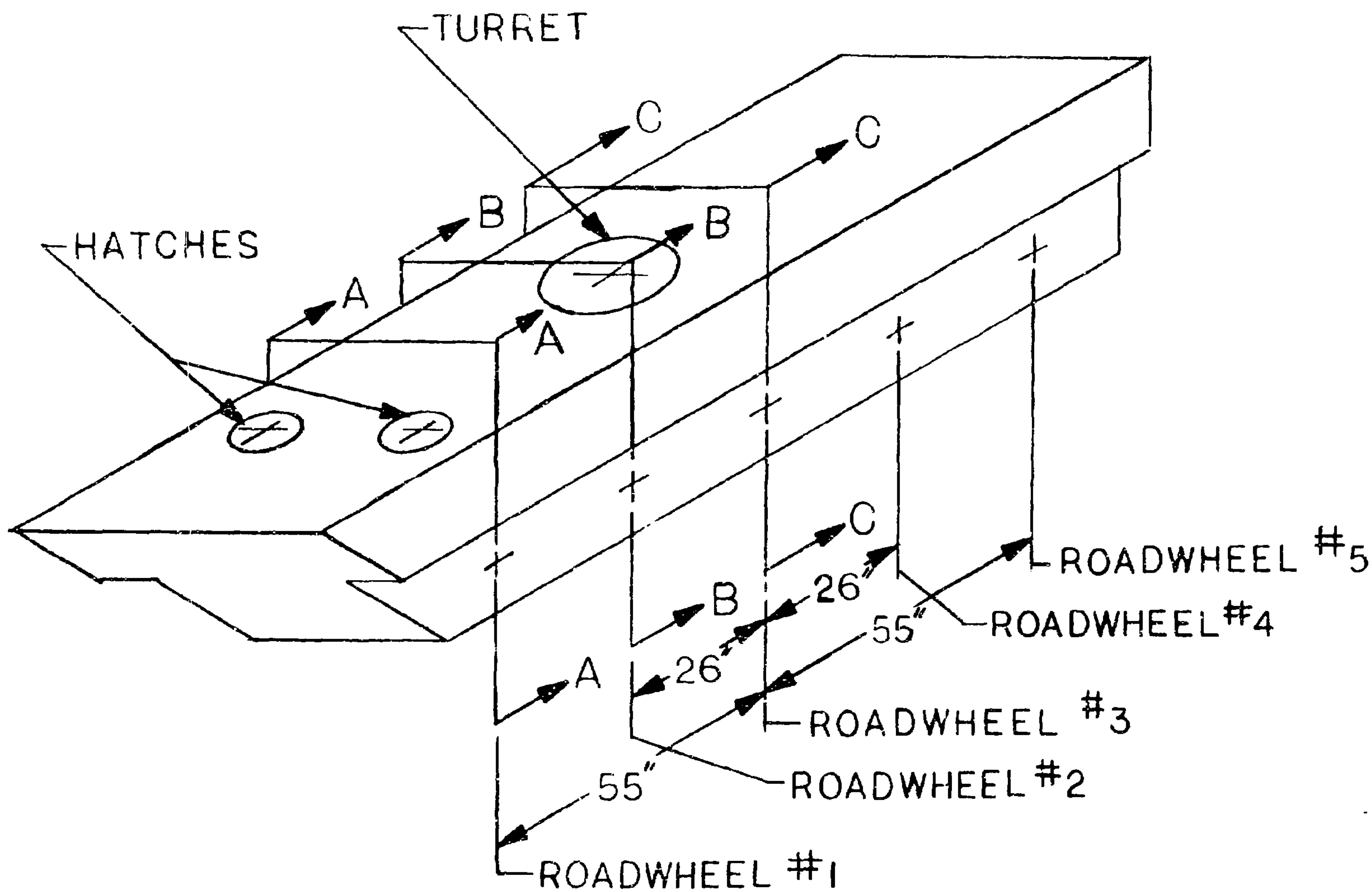

Figure 2.8-1 Locations of Roadwheel Stations and Sections Where Stresses are Determined on ATR Hull

Load at Station #1 = 101,920 lbs.

Load at Station #2 = 50,960 lbs.

Load at Station #3 = 50,960 lbs.

Load at Station #4 = 50,960 lbs.

Load at Station #5 = 50,960 lbs.

2.8.1.1.2 Sections

Cross-sectional views of the areas that were stress analyzed are presented in Figures 2.8-2, 2.8-3, and 2.8-4. Sections A-A, B-B, C-C are located at roadwheel stations #1, #2, and #3, respectively. Each section was treated as a closed, thin-walled, hollow member.

2.8.1.1.3 Load Cases

The load cases analyzed during this preliminary structural analysis are presented in Table 2.8-1.

Table 2.8-1. Preliminary Structural Analysis Load Cases

Load Case	Applied Load	Section Where Stresses Are Determined
1	P_1 to one vehicle side	A-A
2	P_1 and P_2 to one vehicle side	B-B
3	P_1, P_2 and P_3 to one vehicle side	C-C
4	P_1 to both vehicle sides	C-C
5	P_1 and P_2 to both vehicle sides	C-C
6	P_1, P_2, P_3, P_4 and P_5 to one vehicle side	C-C

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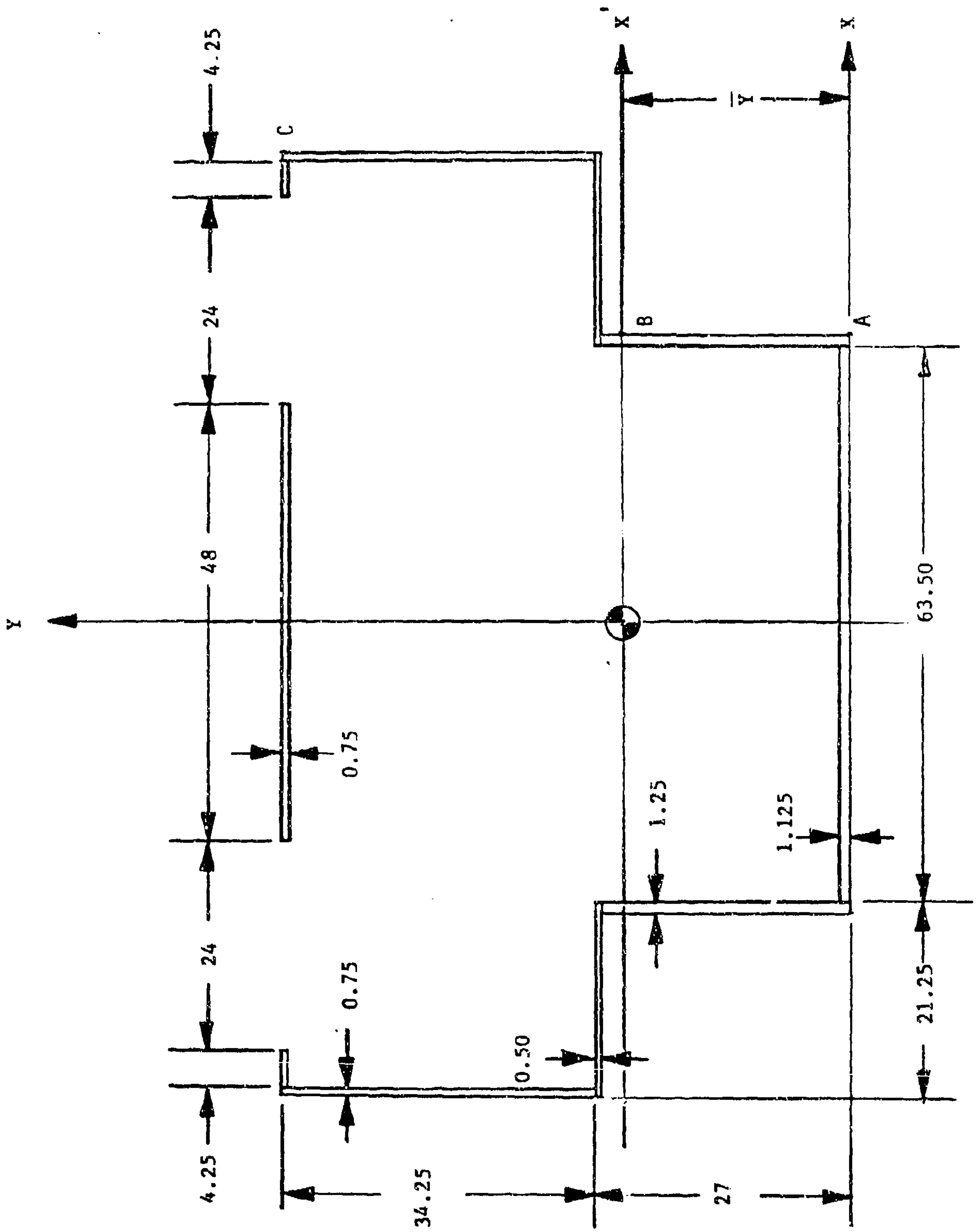

Figure 2.8-2 Section A-A

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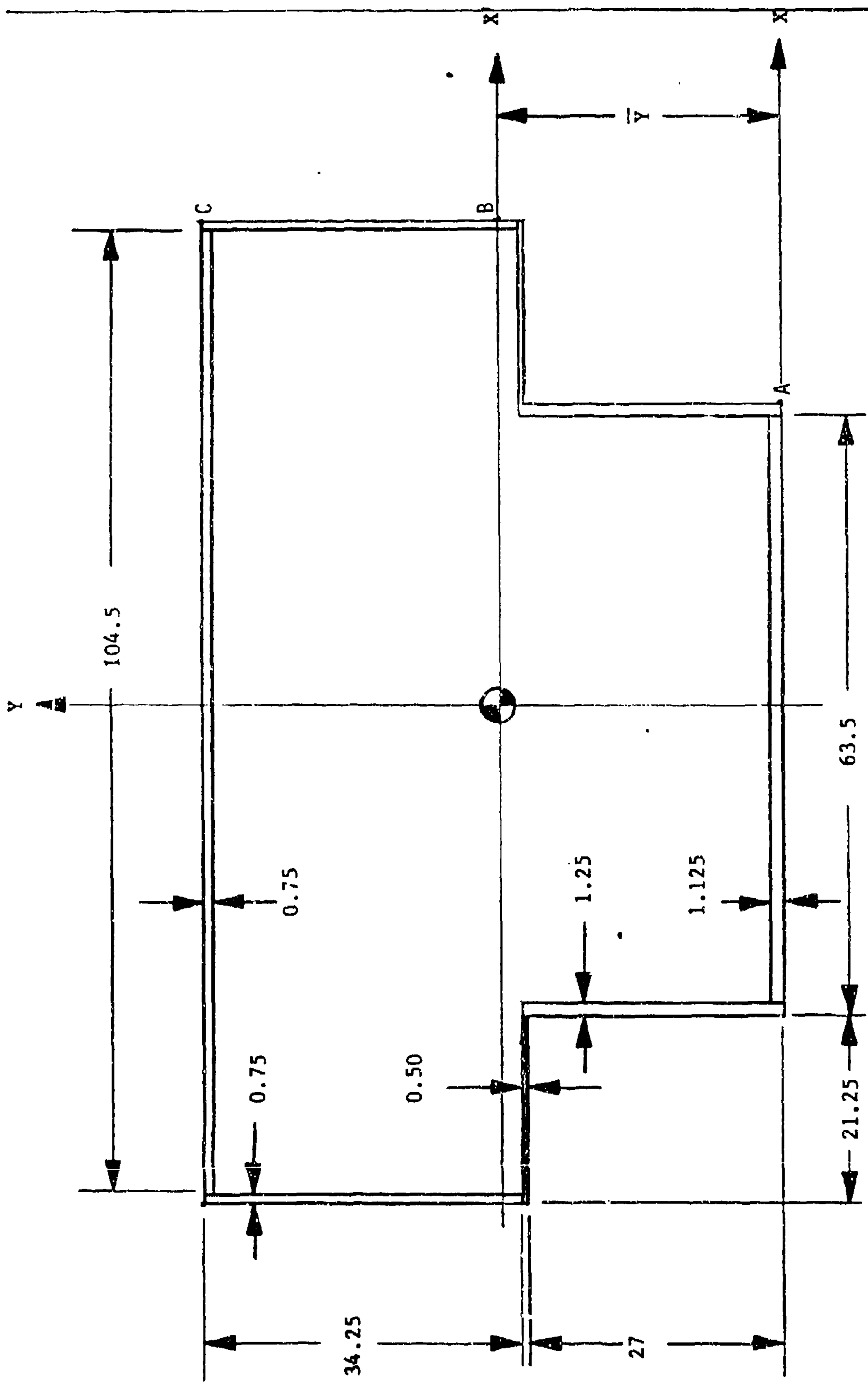

Figure 2.8-3 Section B-B

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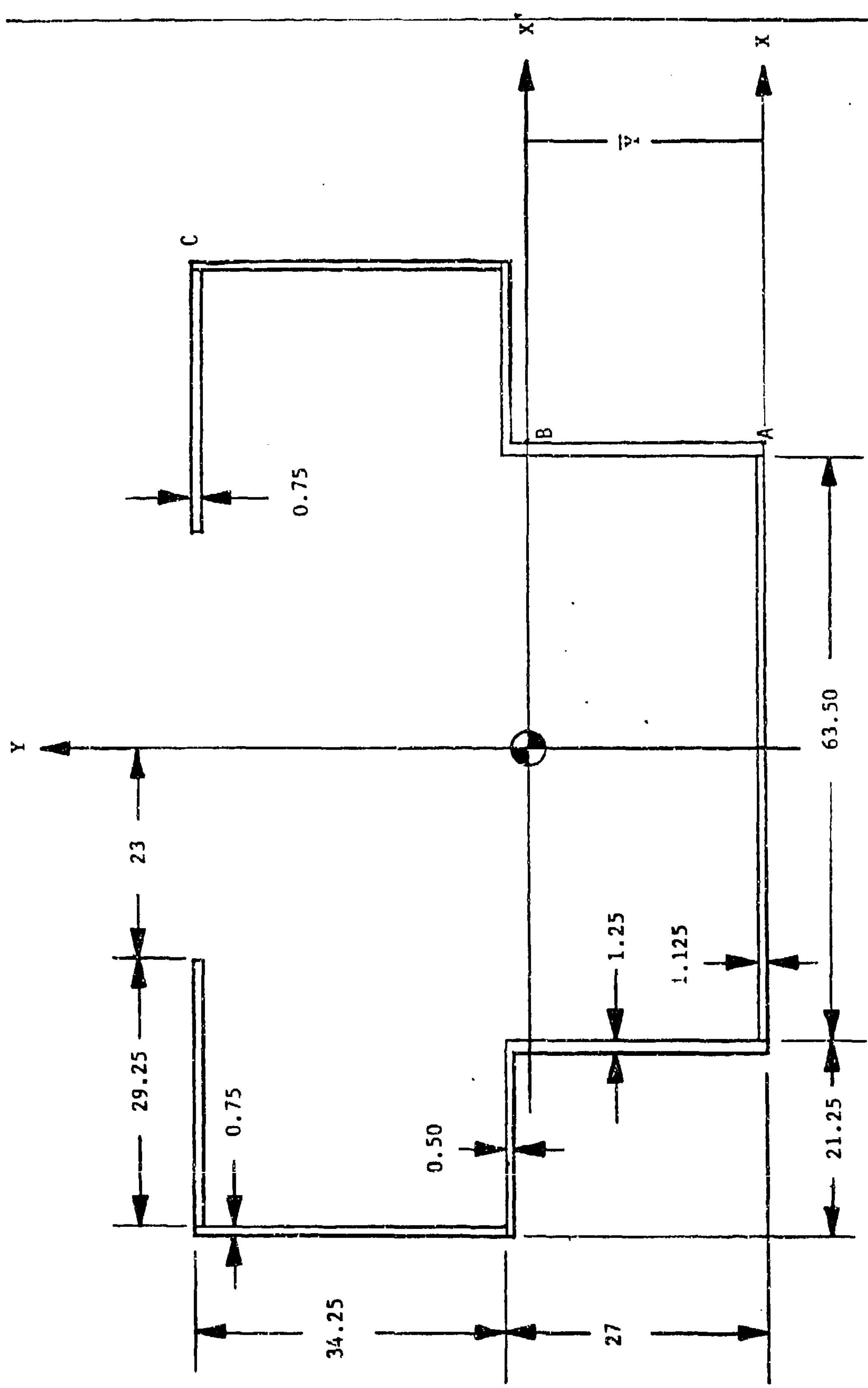

Figure 2.8-4 Section C-C

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In each case the hull was assumed fixed at the section being analyzed. Stresses were determined at Points A, B and C (see Figures 2.8-2, 2.8-3, and 2.8-4) for each load case.

2.8.1.1.4 Formulas Used

Torsional Stress - $f_s = \frac{T}{2 A t}$

where,

T = applied torque

A = area within the perimeter defined by the centerline of the structure

t = thickness of beam being analyzed

Direct Shear Stress - $f_s = \frac{F}{A}$

Where,

F = applied shear load

A = area of cross-section

Bending Stress - $f_b = \frac{Mc}{I_{x'}}$

Where,

M = applied bending moment

c = distance fron neutral axis to point being analyzed

$I_{x'}$ = area moment of inertia about neutral axis

Total Shear Stress - $f_{s\ total} = f_{s\ torsion} + f_{s\ direct}$

Principal Stresses - $f_{s\ max} = \sqrt{\left(\frac{f_b}{2}\right)^2 + f_{s\ total}^2}$

$$f_{n\ max} = \frac{f_b}{2} \pm f_{s\ max}$$

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$$\text{Margin of Safety, Bending - M.S.}_{Bending} = \frac{F_B}{f_{nmax}} - 1$$

$$\text{Margin of Safety, Shear - M.S.}_{Shear} = \frac{F_s}{f_{smax}} - 1$$

where,

F_B = allowable bending stress

F_s = allowable shear stress

2.8.1.1.5 Allowable Stresses

The allowable bending stress used was 40,000 psi, the ultimate strength of welded 5083 Aluminum Alloy. The allowable shear stress used was 20,000 psi, one-half of the allowable bending stress.

2.8.1.2 Sample Calculations

The example calculations presented here are for Case 3, Points B and C. These calculations will include examples of all stress calculations used. The calculations of the location of the neutral x axis and of the area moment of inertia are not presented as they are standard calculations and require large amounts of space to perform. The results of these calculations are as follows:

$\bar{Y}$ = distance from hull bottom to neutral axis = 25.509"

I_x' = area inertia about neutral axis = 138,602.42 in^4,

2.8.1.2.1 Stresses at Point B, Load Case No. 3

Point B occurs on the neutral axis and, therefore, encounters both torsional and direct shear stress, but no bending stress.

$$\text{Torsional Shear Stress - } f_s = \frac{T}{2(A)t}$$

$$T = (P_1 + P_2 + P_3)\ (43.375")$$

$$= (101{,}920 \text{ lb} + 50{,}960 \text{ lb} + 50{,}960 \text{ lb})\ (43.375")$$

$$= 8.84156 \times 10^6 \text{ in lbs}$$

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$$(A) = 31.125'' (105.25'') + 26.6875'' (64.75'')$$

$$= 5319.672\ in^2$$

$$t = 1.25$$

$$f_s = \frac{8.84156 \times 10^6\ in\ lbs}{2(5319.672\ in^2)x(1.25\ in)} = 664.82\ psi$$

Direct Shear Stress - $f_s = F/A$

$$F = P_1 + P_2 + P_3$$

$$= 101{,}920\ lbs + 50{,}960\ lbs + 50{,}960\ lbs$$

$$= 203{,}840\ lbs$$

$A = 255.440\ in^2$ (Determined with $\bar{Y}$ and I_x')

$$f_s = \frac{203{,}840\ lbs}{255.440\ in^2} = 798.00\ psi$$

Total Shear Stress - $f_{s\ Total} + f_{s\ Torsion} + f_{s\ Direct}$

$$fs_{Total} = 664.82\ psi + 798.00\ psi = 1{,}462.82\ psi$$

Since there is no bending, $f_b = 0$ and $f_{s\ max} = f_{s\ Total}$

$$M.S._{Shear} = \frac{20{,}000\ psi}{1{,}462.82\ psi} - 1 = 12.67$$

2.8.1.2.2 Stresses at Point C, Load Case No. 3

Point C occurs at the maximum distance from the neutral axis. At this point, the direct shear stress is 0.

Torsional Stress - Same calculation as 2.8.1.2.1, except

$$t = 0.75\ in$$

$$f_s = 1108.03\ psi$$

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Bending Stress $f_B = \frac{Mc}{I_{x'}}$

$M = P_1\ (55") + P_2\ (26")$

$= 101{,}920\ \text{lbs}\ (55") + 50{,}960\ \text{lbs}\ (26")$

$= 6.93056 \times 10^6\ \text{in lbs}$

$C = 36.24\ \text{in}$

$I = 138{,}602.42\ \text{in}^4$

$$f_B = \frac{6.93056 \times 10^6\ \text{in lbs}\ (36.24\ \text{in})}{138{,}602.42\ \text{in}^4} = 1{,}812.16\ \text{psi}$$

Principal Stresses - $f_{smax} = \sqrt{\left(\frac{f_b}{2}\right)^2 + f_s^{\,2}}$

$$f_{smax} = \sqrt{\left(\frac{1{,}812.16\ \text{psi}}{2}\right)^2 + (1{,}108.03\ \text{psi})^2} = 1{,}431.33\ \text{psi}$$

$$f_{nmax} = \frac{f_b}{2} \pm f_{smax}$$

$$= \frac{1{,}812.16\ \text{psi}}{2} \pm 1{,}431.33\ \text{psi}$$

$$= 2{,}337.41\ \text{psi},\ -525.25\ \text{psi}$$

Margins of Safety

$$M.S._{Bending} = \frac{40{,}000\ \text{psi}}{2{,}337.41\ \text{psi}} - 1 = 16.11$$

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$$M.S._{Shear} = \frac{20{,}000 \text{ psi}}{1{,}431.33 \text{ psi}} - 1 = 12.97$$

2.8.1.3 Results

The results for each load case described in Section 2.8.1.1.3 are detailed in Table 2.8-2, below.

Table 2.8-2. Results of Hull Structural Analysis

Load Case	Point	$M.S._{Bending}$	$M.S._{Shear}$
1	A	-	53.15
	B	-	26.26
	C	-	35.10
2	A	45.63	31.90
	B	-	13.72
	C	34.03	21.90
3	A	23.79	19.49
	B	-	12.67
	C	16.11	12.97
4	A	18.39	-
	B	-	24.06
	C	12.65	-
5	A	14.68	-
	B	-	15.71
	C	10.04	-
6	A	27.66	16.58
	B	-	8.11
	C	18.34	10.75

In none of the cases checked does the structure approach failure.

2.8.1.4 Revised Analysis of Sections A-A and B-B

The analysis detailed in the above sections assumed that the top plate of the hull was a solid member except for the hatch and turret areas. This is not true in that the engine grille, located in the forward area of the top plate, provides no structural support. This fact effects the analysis done for cases #1 and #2. These case analyses were redone, using revised cross-sectional configurations. The sections were again treated as closed,

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thin-walled, hollow members. The support of the firewall, which was increased in thickness to add structural support to the top hull plate between the date of the initial hull analysis and the date of this analysis, was neglected. The results of this analysis are presented in Table 2.8-3.

Table 2.8-3. Revised Results of Cases No. 1 and No. 2 of Hull Structural Analysis

Load Case	Point	$MS_{Bending}$	MS_{Shear}
1	A	-	50.36
	B	-	23.78
	C	-	33.24
2	A	42.22	29.91
	B	-	16.94
	C	29.10	19.98

Again the structure does not approach failure for either case. Note that the M.S. in shear actually increases for Point B of Load Case #2, as compared to the initial analysis. This is due to the fact that Point B, which is located on the netural axis, is shifted from a beam 0.75 in. thick to one that is 1.25 in. thick due to a lowering of the neutral axis. This lowering is the result of the removal of area from the uppper part of structure.

2.8.2 Analysis of Top Plate

hould a weapons station be mounted to the ATR, the firing loads and road shock loads would be transmitted into the top plate of the hull. This analysis was performed to determine if the hull top plate is capable of supporting these loads. It was assumed that the weapons station would be a remotely con ⁻olled, unmanned turret, mounting a 25mm cannon.

When this analysis was initially performed, it was assumed that the top plate was supported by welds on each side and had no additional reinforcing structure. In this configuration, the plate failed when subjected to the design loads. The addition of a weapon station support ring, a lateral support beam immediately aft of the weapon station, and a structural firewall forward of the weapons station allowed the structure to survive the design loads.

Since the time of this analysis, which was reported in R-60011-00003, both the weapons station support structure and the design loads have

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changed. The new structure is illustrated on Drawing #60011-10042, "Turret Support Structure". The structure now consists of a turret support ring and three vertical support columns. Note that the firewall is no longer considered a part of the structure. A new analysis has been performed to assure the ability of this structure to withstand the design loads. This analysis is presented below.

2.8.2.1 Applied Loads

The following loads are applied to the top plate:

1. A 21,000 lb. firing load, applied at any angle between -8° and +60° with the horizontal (includes safety factors).

2. An 18 g vertical road shock load, applied to the turret weight at the turret c.g.

3. A 9 g longitudinal road shock load, applied to the turret weight at the turret c.g.

4. A 7.5 g lateral road shock load, applied to the turret weight at the turret c.g.

The loads are applied independently. The design turret weight is estimated as 3000 lbs, which includes the weight of the armament and ammunition, as well as the turret structure.

2.8.2.2 Analysis Method

Two analysis methods were utilized in examining the top plate. The primary method used was a NASTRAN finite element computer analysis, utilized to determine the stresses and deflections resulting in the top plate and turret support structure due to the application of the various loads. The second analysis method utilized standard buckling formulas to assure that none of the support columns would buckle under the maximum vertical loads imposed on them.

2.8.2.3 Results

Several load cases were examined with the NASTRAN stress analysis method. It was found that the top plate and support structure did not fail in any of the cases examined.

The most critical load case examined proved to be the application of the 18 g vertical road load. The results of the analysis of this case are presented in Figures 2.8-5, 2.8-6, and 2.8-7. Figure 2.8-5 presents the model of the top plate, along with the grid point identification numbers of

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Maximum Deflection = 0.690 inches at Point 112

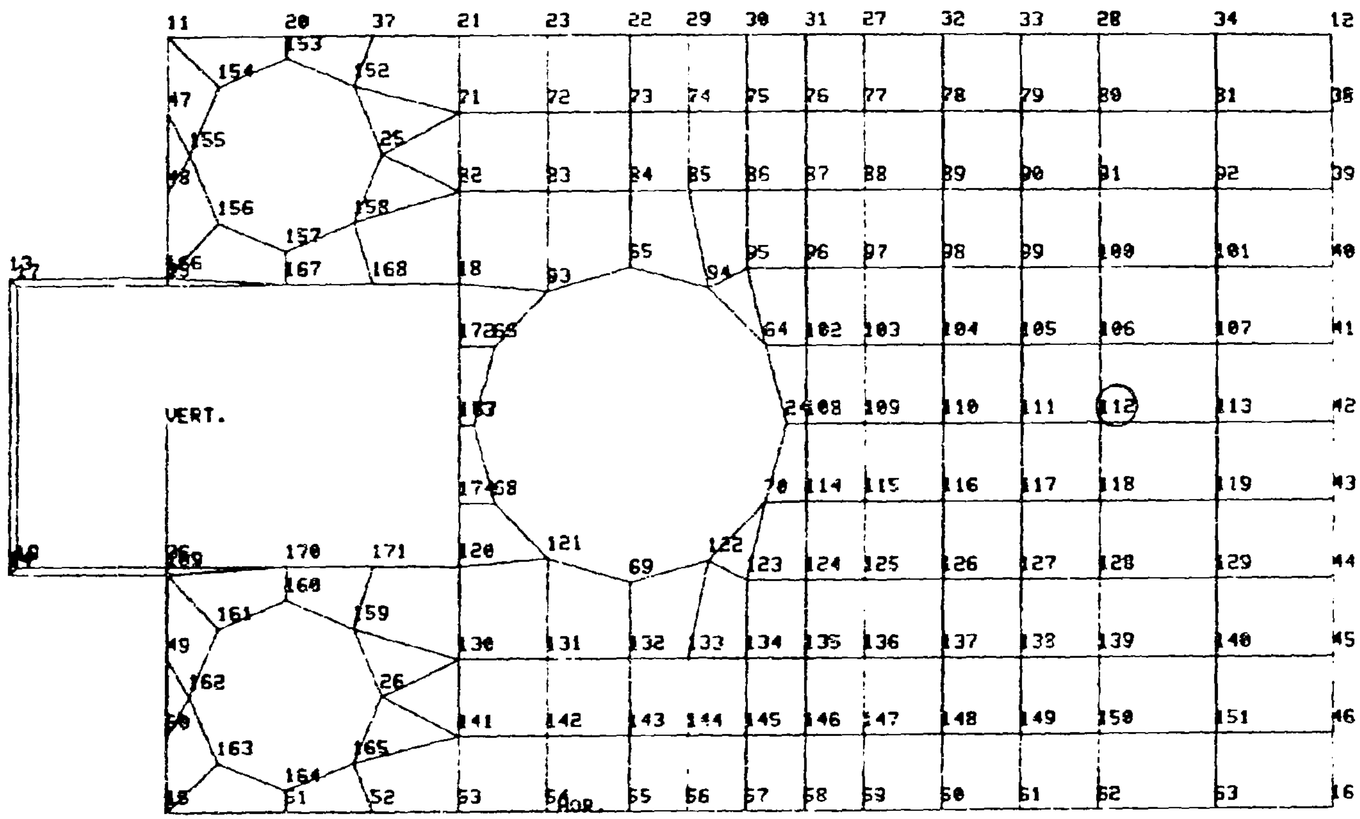

Figure 2.8-5 Top Plate Model and Maximum Deflection

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Maximum Stress = 6800 psi in Element 164

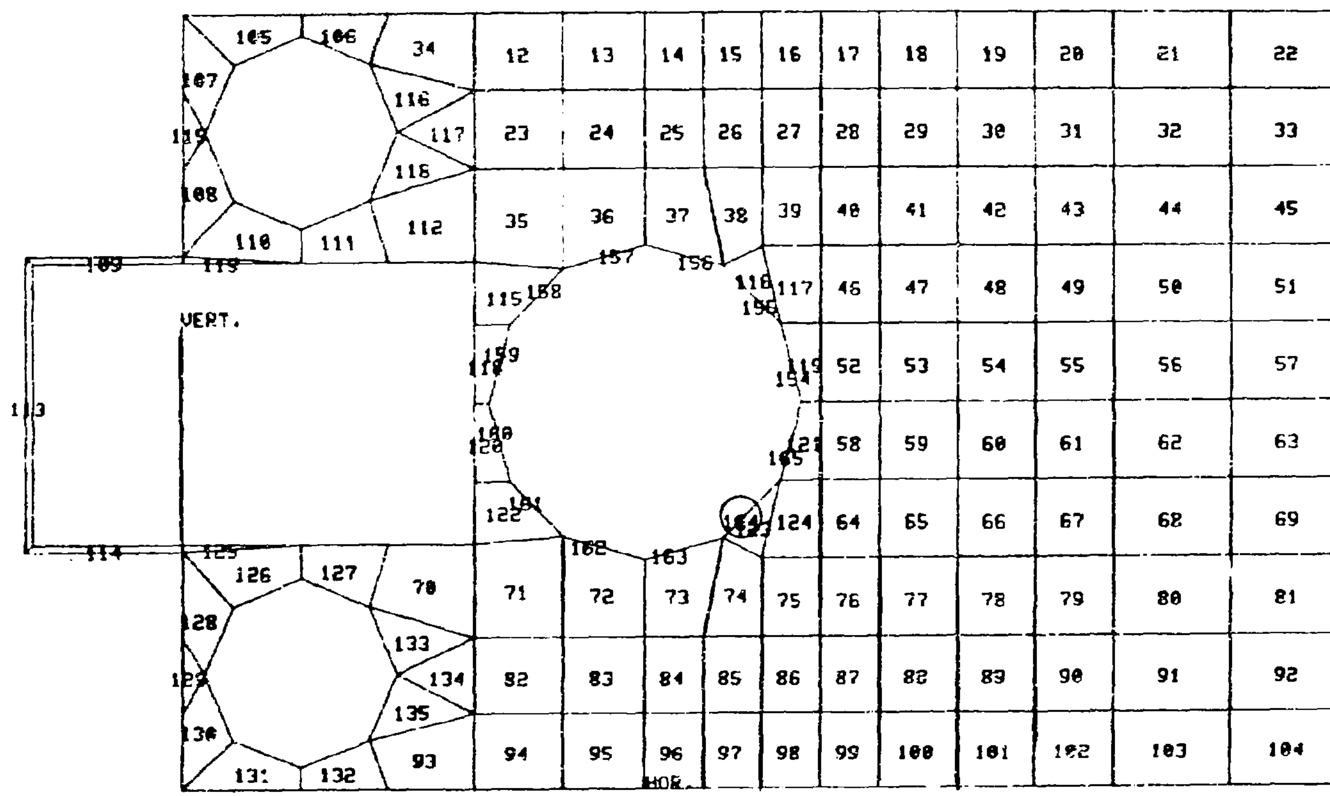

Figure 2.8-6 Top Plate Model and Maximum Stress

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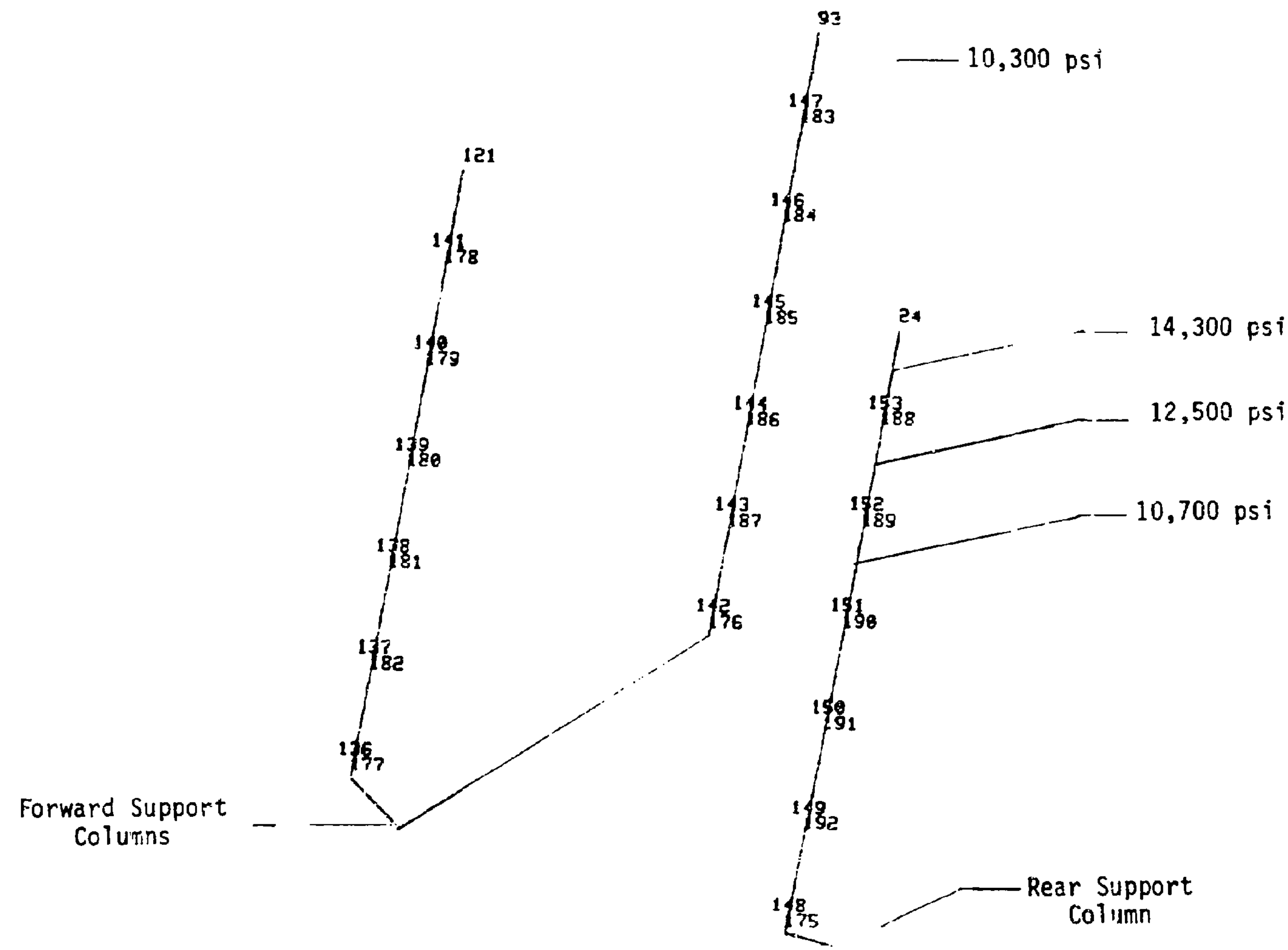

Figure 2.8-7 Turret Support Columns and High Stress Areas

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the model, and indicates the point where the maximum plate deflection of 0.690 inches occurs. Figure 2.8-6 presents the model of the top plate, along with its element identification numbers, and indicates the element in which the maximum stress of 6800 psi occurs. Figure 2.8-7 presents the models of the support columns and indicates areas of high stress. Note that at no point does the stress resulting in the structure approach the ultimate strength of 5083 aluminum (40,000 psi).

Studying the allowable critical loads before buckling for the vertical support columns showed that the original forward columns considered were inadequate to support the loads imposed on them. A proper size was determined for these columns and the new column design was incorporated into the structure. The new column design has a 2.0 inch outer diameter and a 0.25 inch thick wall. The original rear column design was adequate to support the load applied to it.

The forward columns were initially examined in two ways. The first way treated the columns as fixed at one end and pinned at the other, while the second way treated the columns as fixed at both ends. The actual behavior of the columns is somewhere between these ideals. The stricter criteria, one end fixed and the other pinned, was used to choose the needed column size and also to examine the rear column.

2.8.2.4 Reanalysis of the Top Plate

The NASTRAN analysis of the top plate and weapons station support structure was conducted again following the design of the intake/exhaust grille. This reanalysis was required since the final grille design created changes in the hull top plate and the relocation of the weapons station towards the rear end of the ATR. The applied loads and analysis method utilized were identical to those described in the preceeding sections.

2.8.2.5 Results of the Reanalysis

Upon examination of the revised top plate and weapon station support structure, it was determined that the forward turret support columns were not sufficient to support some of the applied loads. To correct this deficiency, columns having a 3.0 inch outer diameter and a 0.375 inch wall thickness were used to replace the previous forward support columns design. Following this design change, all of the structural components analyzed were found to be of sufficient strength to support each of the design loads imposed

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on them. The maximum stresses incurred in the individual structural members are summarized in Table 2.8-4.

The maximum deflection of the structure was 0.411 inches. This deflection occurs in the top plate, midway between the rear of the weapons station and the rear of the top plate. The deflection results from application of the 18g vertical shock load.

2.8.3 Floor Analysis

This analysis was performed to determine the load carrying characteristics of the ATR floor. The analysis was performed on the floor plates as they were initially configured within the hull. This configuration consisted of three plates. Two of these plates spanned the length of the vehicle and extended laterally from the main floor/engine support channels to the vehicle sides. The third plate covered the center area, between the two channels and spanned from the firewall to the vehicle rear. The plates were supported at each edge. It was found that these plates would withstand loads well above the design floor load of 100 lbs/ft^2 and would have minimal deflection at a load of 100 lbs/ft^2.

The current floor plate configuration consists of nine plates. Each plate is supported on all sides, as shown on Drawing # 60011-40230. None of the plates approaches the overall size or the span of the plates of the original design. It is unnecessary, therefore, to revise the floor analysis, as the new floor configuration is obviously superior to the initial design in supporting loads. The original analysis is presented below.

2.8.3.1 Floor Description

The floor plate of the hull is a 0.25 in. thick aluminum plate, supported on four sides and by two lateral beams running fore to aft over the entire vehicle length. These beams will also be used to support the engine.

2.8.3.2 Analysis Method

The lateral beams divide the floor into three sections of rectangular plates, supported on each side. This analysis consisted of determining the maximum allowable load for an outboard section and for the center section of the floor, the deflection resulting from the maximum allowable load, and the deflection resulting from a load of 100 lbs/ft^2.

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Table 2.8-4 Resulting Maximum Stresses For Reanalysis of Hull Top Plate and Weapons Station Support Structure

Component	Load Condition	Stress (psi)	Margin Of Safety
Top Plate	0^0 Lateral Firing Load	16,800	0.96
Turret Support Ring	0^0 Lateral Firing Load	24,500	0.35
Upper Front Glacis	18g Vertical Shock Load	2,300	13.3
Forward Turret Support Column	18g Vertical Shock Load	17,800	0.12
Rear Turret Support Column	18g Vertical Shock Load	18,900	0.06
Engine/Floor Support Channel	18g Vertical Shock Load	1,100	29.0

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2.8.3.3 Formulas Used

$$\text{Allowable Load} - q = \frac{f_b \, t^2}{B \, b^2}$$

$$\text{Deflection} - y = \frac{A \, q \, b^4}{E \, t^3}$$

Where,

f_b = bending stress
t = plate thickness
b = length of short side of plate
E = Young's Modulus
A,B = constants

These formulas were taken from:

Roark, Raymond J. and Young, Warren C., Formulas for Stress and Strain, (New York: McGraw-Hill Book Company), 1975, p. 386.

The allowable bending stress was taken as 33,000 psi, the yield stress of a thin, 5083 aluminum plate.

2.8.3.4 Results

The results of this analysis are presented in Table 2.8-5.

Table 2.8-5. Results of Floor Analysis

Floor Area	Allowable Load (psf)	Deflection at Allowable Load (in.)	Deflection at Load of 100 psf (in.)
Outboard	1222.6	0.81	0.07
Center	1549.4	0.64	0.04

The floor will easily support the load imposed on it, which is not expected to exceed 100 psf.

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2.8.4 ATR Nose

This analysis was performed to insure that the welded plates configuring the nose of the ATR would not fail under the load encountered when the ATR overrides a three inch diameter tree. The force generated in accomplishing this is estimated as 500 lbs. The initial hull configuration easily withstood this load.

Since this analysis was performed, the hull configuration has been changed by the addition of an upper front glacis plate which slopes down from the ATR top plate until it intersects the lower front glacis plate. This addition lowers the initial contact point between the ATR and tree, but does not alter the fact that the contact point is along the welded joint of two hull plates. In the initial analysis, as discussed in R-60011-00003, this weld was the most critical area examined, but still maintained a margin of safety of 6.0 under the 500 lb. load. As the weld joining the upper and lower glacis plates in the present configuration is thicker than that examined in the original analysis, it may be safely assumed that the current nose structure is more than adequate to withstand the overriding of a three inch diameter tree.

2.8.5 Shock Factor Diagram

In conjunction with the vehicle stress analyses, a shock factor diagram was derived to determine the design g load applied to vehicle mounted items. The method used for drawing this diagram is presented in the design manual "Automotive Bodies and Hulls," Chapter 4, page 4-14 (AMCP No. 706-357, 20 April 1970). This diagram, which is used in designing the mounting structure for all vehicle mounted components, is presented in Figure 2.8-8.

2.8.6 ATR Hull Structural Analysis - Suspension Induced Loads

Finite element analyses have been performed to determine the effects of suspension induced loads. These analyses were originally discussed in R-60011-00005.

2.8.6.1 Loading Conditions

Three loading conditions have been evaluated in the analysis:

Load Case A:

From, "ATR Design Criteria", R-60011-00002 a road load applied at the centerline of the track of 102,000 pounds. The load will be reacted by the road wheel impacting the sponson bottom together with the hydropneumatic suspension unit resistance (4 g's avg.).

[illegible]

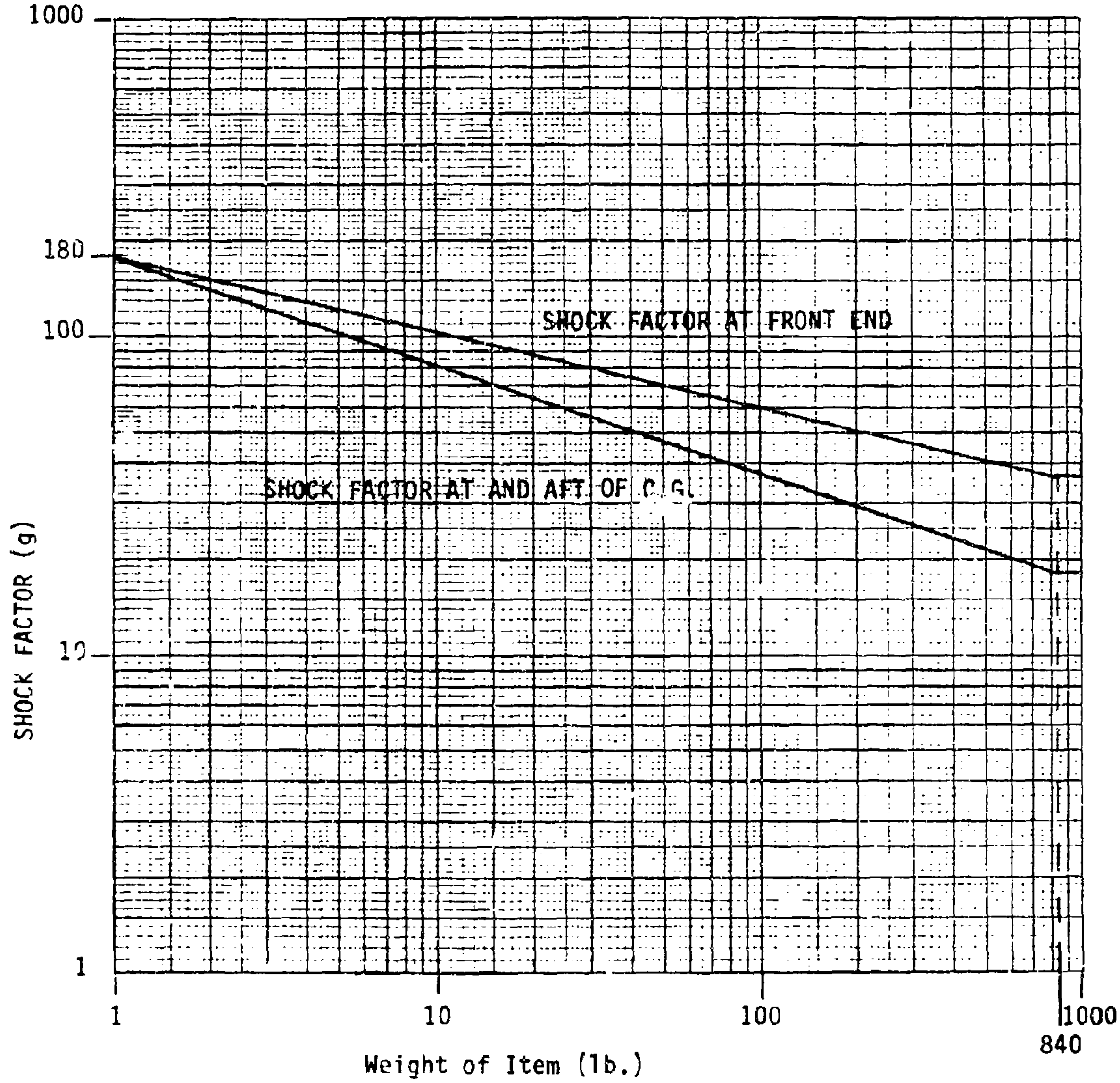

Figure 2.8-8 Road Shock Design Factors for Hull, Turret and Vehicle Mounted Components

Load Case B:

From, "ATR Design Criteria", a road load applied laterally against the bottom edge of the road wheel. The load will be reacted into the hull side in the area of the bolted connection to the hydropneumatic suspension unit.

Load Case C:

A road load applied at the centerline of the track equivalent to 4 g's. The load will be reacted into the hull side in the area of the bolted connection to the hydropneumatic suspension unit.

2.8.6.2 Assumptions

Several assumptions have been made in order to simplify the structural model. These are as follows:

1. A vertical cross section of the vehicle was analyzed which represents one wheel station.

2. Symmetry was assumed. Therefore, one half of the cross section was analyzed.

3. The hydropneumatic suspension unit mounting flange was assumed to be rigid.

4. The road arm and wheel assembly were assumed to be rigid.

5. The top deck was assumed to be a flat plate without cutouts and stiffeners.

6. All elements used are flat plates.

7. The hull side plate and sponson side plate edges are restrained in the vertical direction.

8. The sponson bottom plate edges are restrained in the lateral direction.

2.8.6.3 Model Description

The finite element model used in this analysis is depicted in Figure 2.8-9. This model is one half of a 40 inch long cross section of the ATR hull. The length of the section was selected such that the suspension unit mounting location and the wheel bump stop could be effectively modeled.

Figures 2.8-10 through 2.8-12 show the magnitude and location of the loads applied to the model for load cases A, B, and C.

Figure 2.8-13 shows the boundary constraints which were imposed on the model.

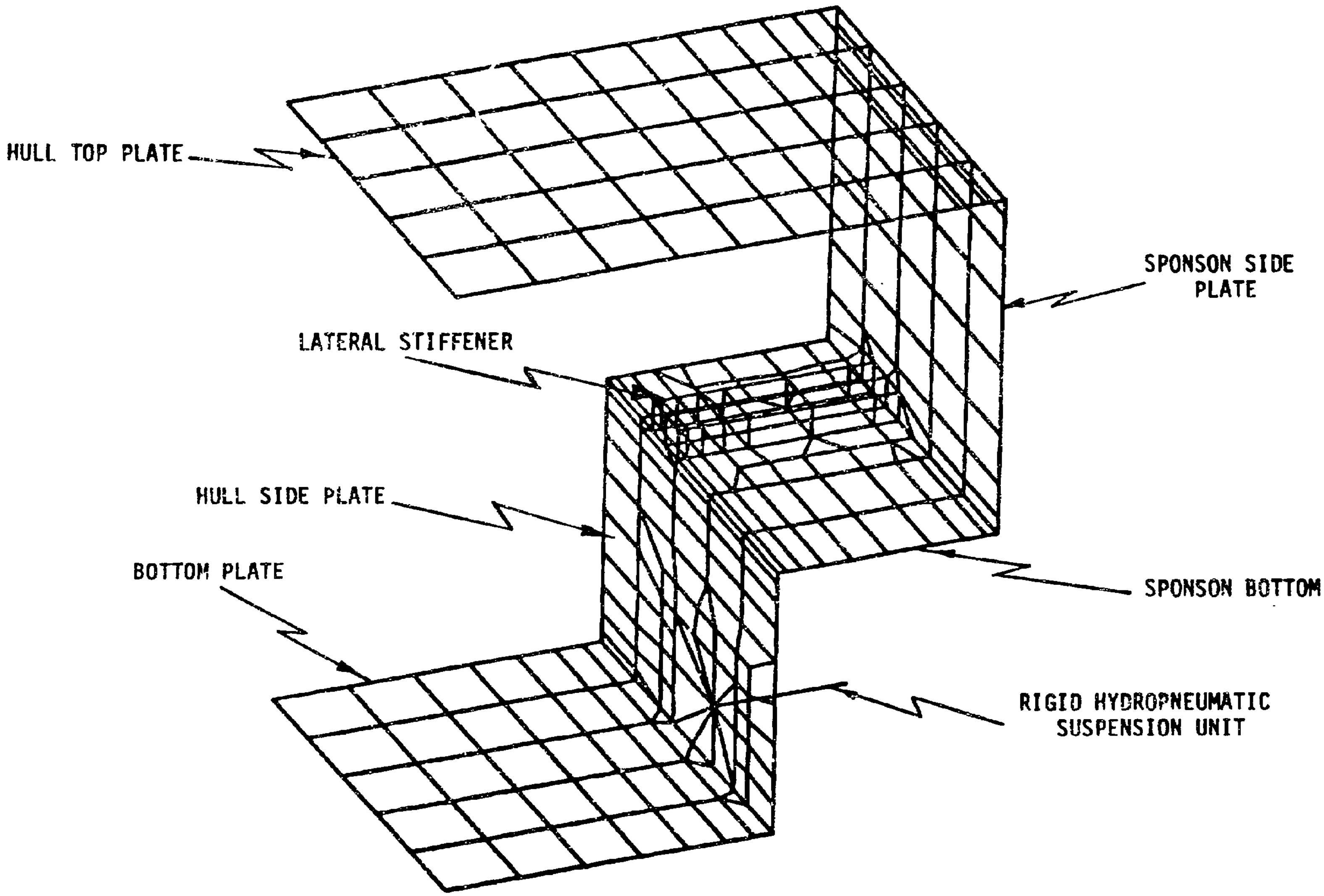

Figure 2.8-9 Finite Element Model

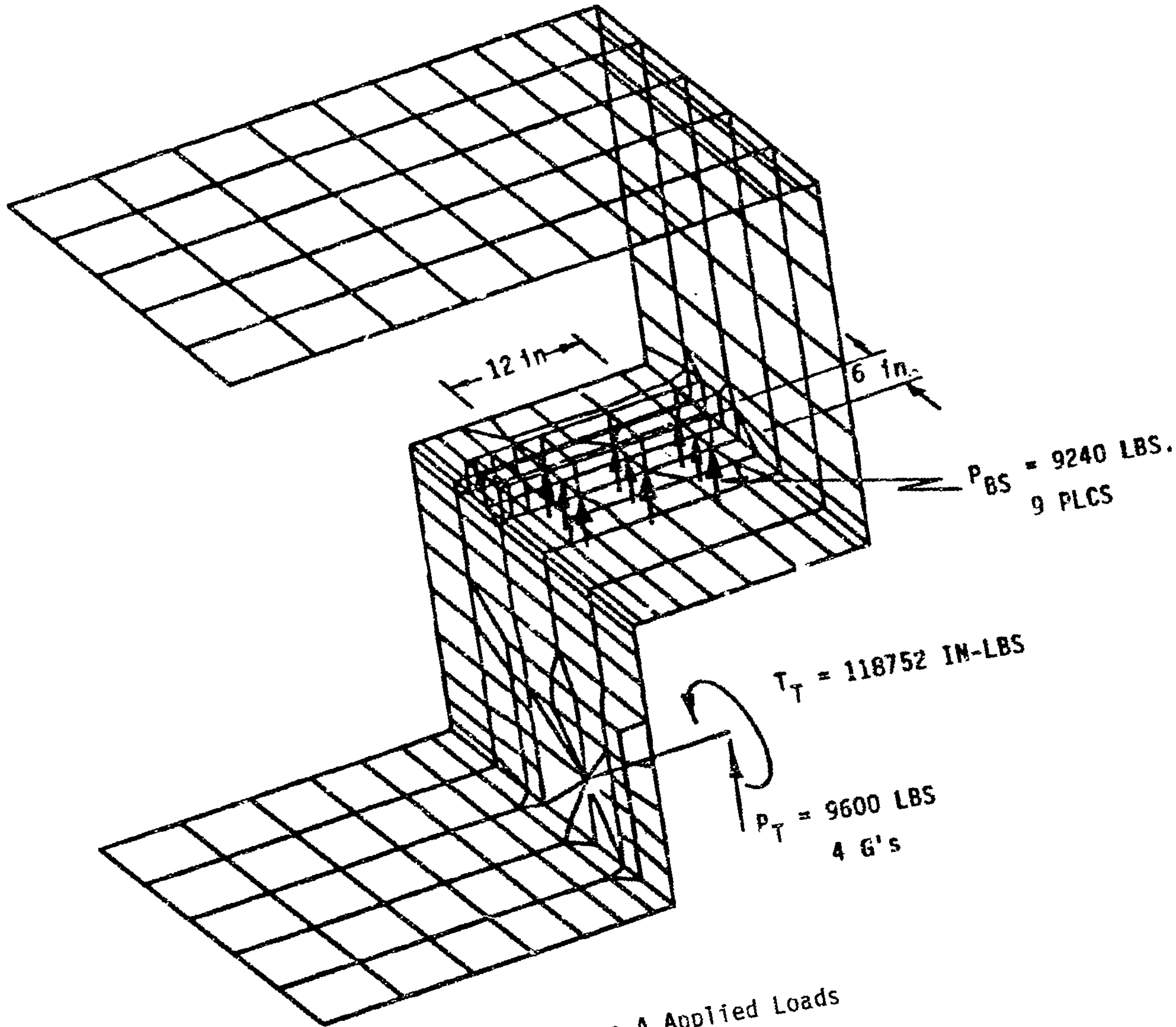

Figure 2.8-10 Case A Applied Loads

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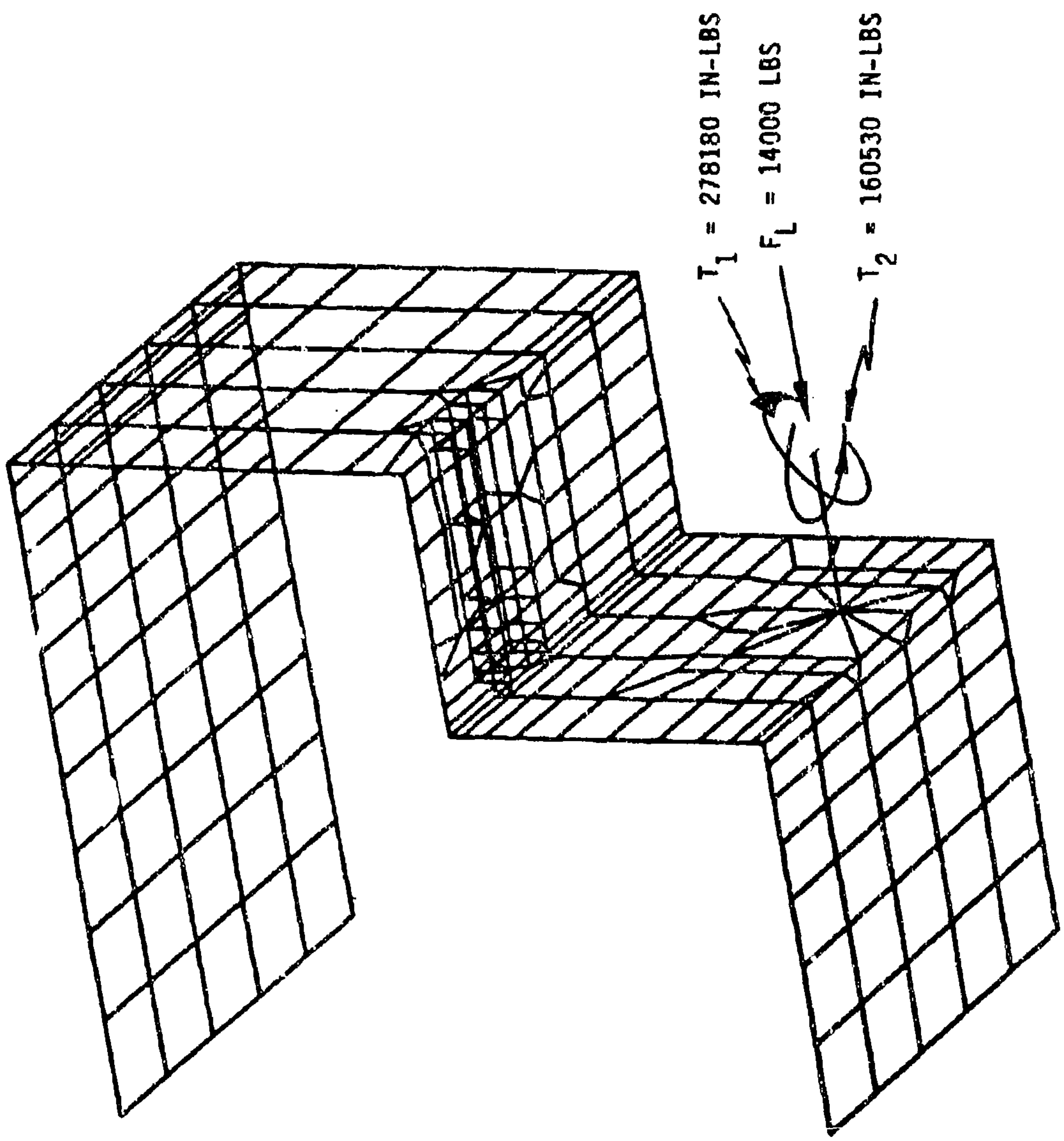

Figure 2.8-11 Case B Applied Loads

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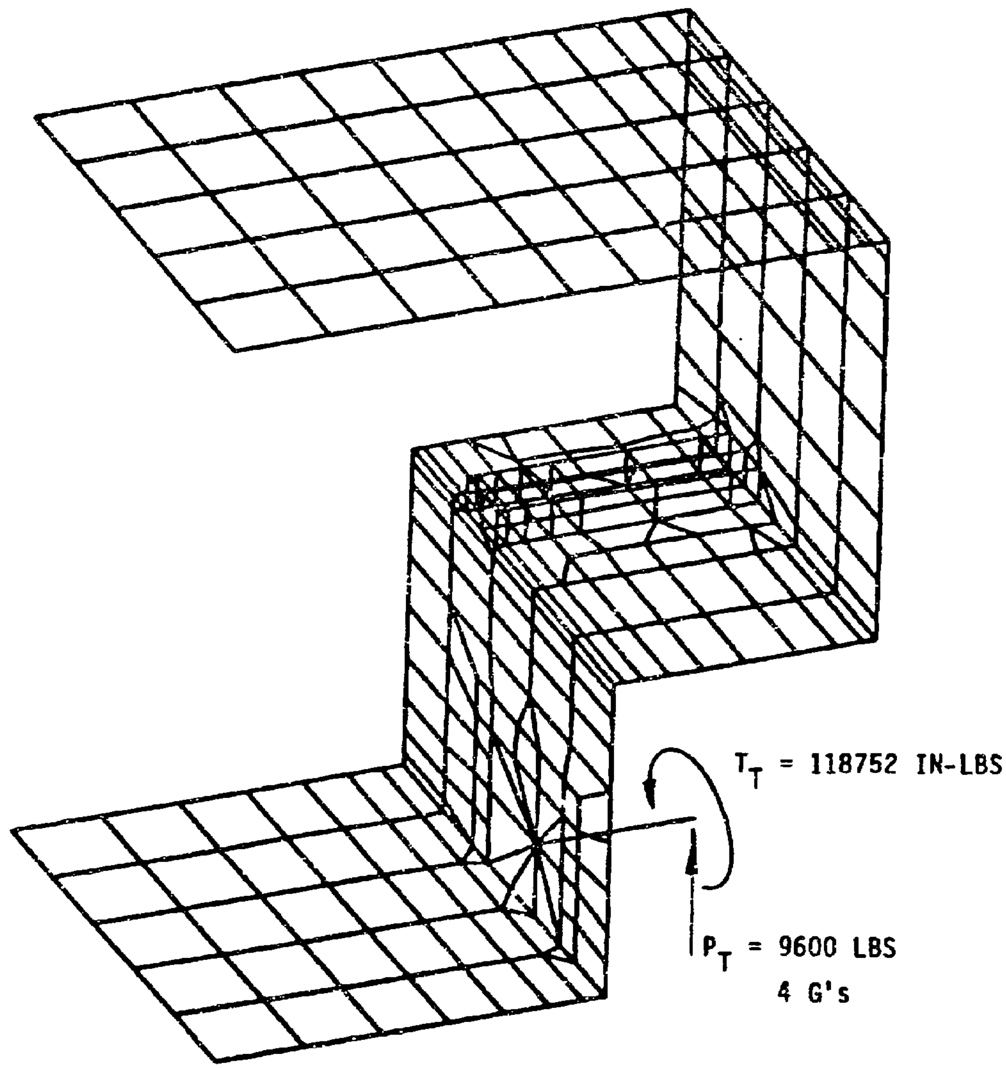

Figure 2.8-12 Case C Applied Loads

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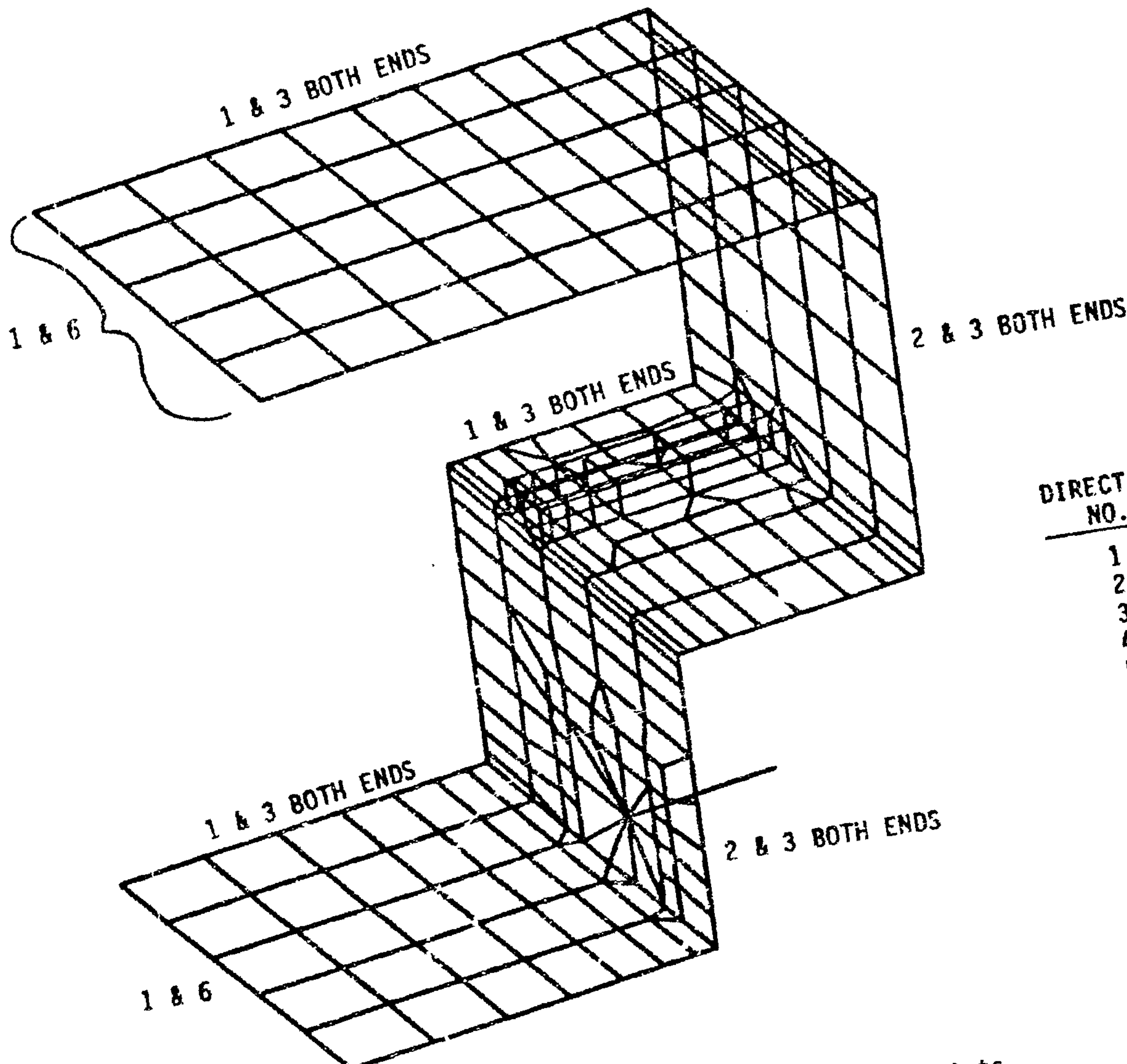

DIRECTION NO.	DIRECTION
1	X
2	Y
3	Z
4	X ROT
5	Y ROT
6	Z ROT

Figure 2.8-13 Boundary Constraints

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All hull members are modeled using flat plate elements and all joints are assumed to be full penetration or equivalent welds.

2.8.6.4 Material Properties

Material: 5083 aluminum alloy armor per AAI Engineering Report No. R-60011-00002, ATR Design Criteria with as welded properties: F_{tu} = 40,000 psi, F_{ty} = 33,000 psi.

2.8.6.5 Results

Prior to this analysis the lateral stiffener, which is shown in Figure 2.8-9, was not in place. Without this stiffener the sponson bottom plate experienced 170 ksi stress due to load case A. Due to these stresses a stiffener was sized, as detailed in R-60011-00005.

The maximum stress in each plate and the stiffener are provided in Tables 2.8-6, 2.8-7 and 2.8-8 for each load case.

Table 2.8-6 Stresses Due to Load Case A

Plate	Principle Stress (psi)	Margin of Safety (yield)	Margin of Safety (ultimate)
Hull Top	734	44	53
Sponson Side	11720	1.8	2.4
Sponson Bot.	30530	0.1	0.3
Hull Side	15100	1.2	1.6
Bot. Plate	2060	15	18.4
Stiffener	23710	0.4	0.7

Table 2.8-7 Stresses Due to Load Case B

Plate	Principle Stress (psi)	Margin of Safety (yield)	Margin of Safety (ultimate)
Hull Top	92	358	434
Sponson Side	786	41	50
Sponson Bot.	14700	1.2	1.7
Hull Side	24100	0.4	0.7
Bot. Plate	5650	4.8	6.1
Stiffener	8200	3.0	3.9

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Table 2.8-8 Stresses Due to Load Case C

Plate	Principle Stress (psi)	Margin of Safety (yield)	Margin of Safety (ultimate)
Hull Top	534	60.8	74
Sponson Side	504	64.5	78
Sponson Bot.	9470	2.5	3.2
Hull Side	15600	1.1	1.6
Bot. Plate	2400	12.8	15.7
Stiffener	5310	5.2	6.5

The maximum deflections due to each load case are provided in Table 2.8-9.

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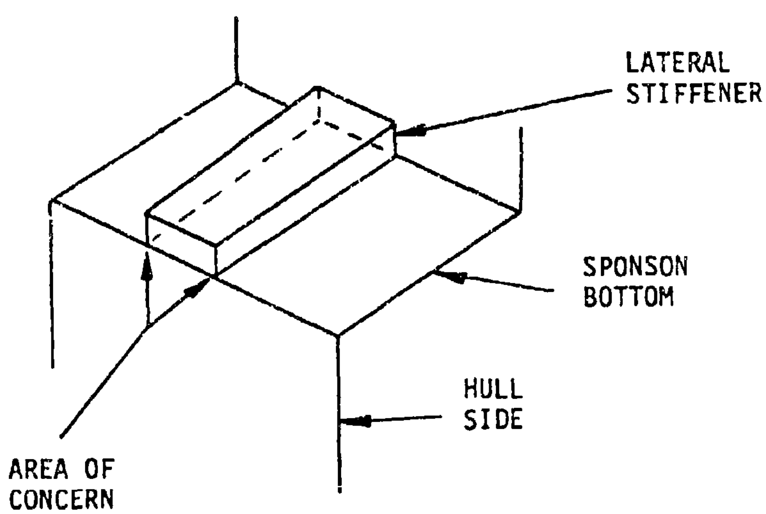
Table 2.8-9 Maximum Deflections

Load Case	Direction	Magnitude (in)	Plate
A	X		
A	Y	.12	Sponson Bot.
A	Z		
B	X	.12	Hull Side
B	Y	.15	Hull Bot.
B	Z		
C	X		
C	Y	.09	Rigid Element
C	Z		

2.8.6.6 Weld Analysis

Examine the stresses in the weld areas between the sponson bottom and the hull side plate under load case A. Highly localized stresses are expected at the end of the lateral stiffener.



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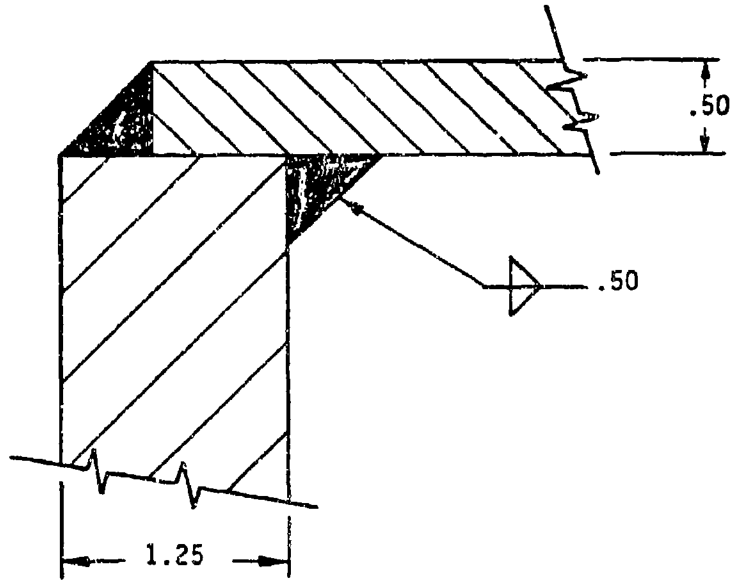
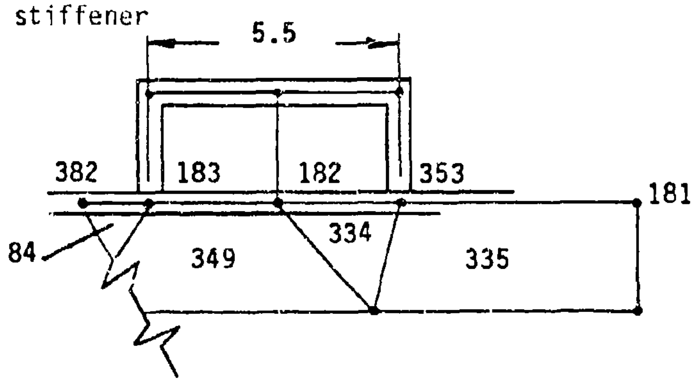
Cross Section of hull side to sponson bottom interface

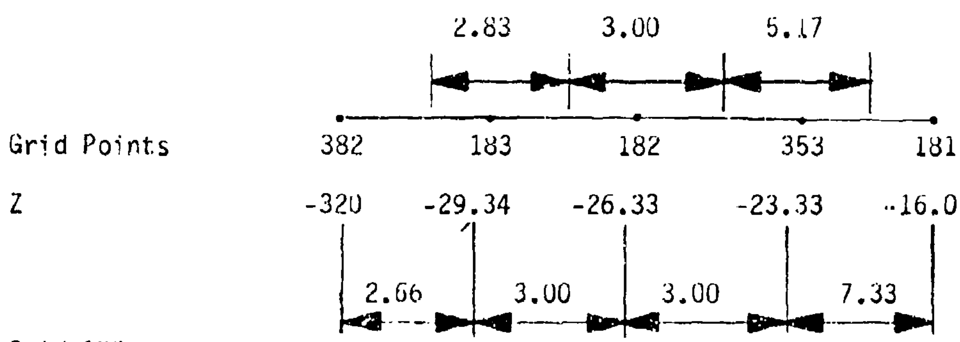


End view of lateral stiffener



Determine the forces and moments applied at grid points 183, 182, and 353. This is done by using the grid point force balance and adding the forces and moments taken by elements 84, 349, 334 and 335. The grid point identifiction numbers and element identification numbers are those used in the NASTRAN stress analysis, discussed above.

Grid	EID	F_x (lbs)	F_y (lbs)	F_z (lbs)	M_{x-x} (in-lbs)	M_{y-y} (in-lbs)	M_{z-z} (in-lbs)
183	84	-450	-690	-220	0.3	640	-170
	349	-3700	-13000	-820	11	-313	-8050
	Total	-4150	-13690	-1040	11.3	327	-8220
182	349	3000	-3000	-1200	-8.8	470	-6300
	334	-81	-27	290	0	-500	-986
	Total	2919	-3027	-910	-8.8	-30	-7286
353	334	-1050	-2040	-1500	0	704	-2980
	335	-3300	-13150	-3850	0	-1063	-8840
	Total	-4350	15190	-5350	0	-359	-11820



Grid 183

$F_x = -4150$

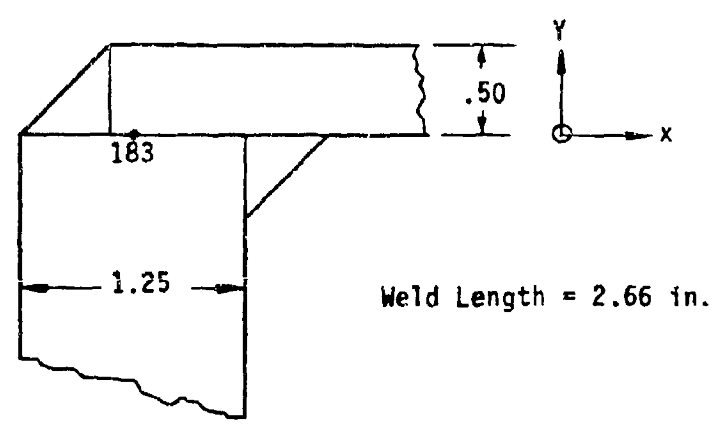
$F_y = -13690$

$F_z = -1040$

$M_x = 0$

$M_y = 0$

$M_z = -8220$



Due to F_x

$$f_s = \frac{F_x}{1.414\ (.50)\ (2.66)} = 2207 \text{ psi}$$

Due to F_y

$$f_s = \frac{F_y}{1.414\ (.50)\ (2.66)} = 7280 \text{ psi}$$

Due to F_z

$$f_s = \frac{F_z}{1.414\ (.50)(2.66)} = 553 \text{ psi}$$

Due to M_{z-z}

$$Iu = \frac{bd^2}{2} \qquad b = 2.66 \text{ in} \qquad d = .75 \text{ in}$$

$$= \frac{2.66\ (.75)^2}{2}$$

$$= 0.748 \text{ in}^3$$

$$I = 0.707\ hIu$$

$$= 0.707\ (.5)\ (.748)$$

$$= .264 \text{ in}^4$$

$$f_b = \frac{Mc}{I} = \frac{8220 \times .375}{.264} = 11676 \text{ psi}$$

$$f = \frac{fb}{2} + \sqrt{\left(\frac{fb}{2}\right)^2 + (\ \ fs)^2}$$

$$= \frac{11676}{2} + \sqrt{\left(\frac{11676}{2}\right)^2 + (2207 + 7280 + 553)^2}$$

$$= 17452 \text{ psi}$$

$$M.S. = \frac{33000}{17452} - 1 = 0.89$$

Grid 182

F_x = 2919 lbs M_{x-x} = -8.8 in-lbs

F_y = -3027 lbs M_{y-y} = -30 in-lbs

F_z = -910 lbs M_{z-z} = -7286 in-lbs

$$l_w = 3.00$$

Grid 182 is longer than grid 183 and loads are less

Grid 353

F_x = -4350 lbs M_{x-x} = 0

F_y = 15190 lbs $M_{y-y} \approx 0$

F_z = -5350 lbs M_{z-z} = -11820 in-lbs

Due to F_x

$$f_s = \frac{-4350}{1.414\ (.50)(5.17)} = 1190 \text{ psi}$$

Due to F_y

$$f_s = \frac{15190}{1.414\ (.50)\ (5.17)} = 4156 \text{ psi}$$

Due to F_z

$$f_s = \frac{5350}{1.414\ (.5)\ (5.17)} = 1475 \text{ psi}$$

Due to M_{z-z}

$$Iu = \frac{5.17\ (.75)^2}{2} = 1454 \text{ psi}$$

$$I = 0.707\ (.5)\ (2.454) = .514$$

$$f_b = \frac{Mc}{I} = \frac{11820 \times .375}{.514} = 8623 \text{ psi}$$

Combining stresses

Combining stresses

$$f = \frac{f_b}{2} + \sqrt{\left(\frac{f_b}{2}\right)^2 + (\ f_s)^2}$$

$$= \frac{8623}{2} + \sqrt{\left(\frac{8623}{2}\right)^2 + (1190 + 4156 + 1475)^2}$$

$$= 12380 \text{ psi}$$

$$M.S. = \frac{33000}{12380} - 1$$

$$= 1.67$$

2.8.6.7 Recommendations

1. Use lateral stiffener on top of sponson bottom plate at wheel station #1.

  x 2.5" x .75" TK Channel Section

2. Use lateral stiffener on top of sponson bottom plate at wheel stations #2, 3, 4 & 5.

9" x 1.25" TK Doubler Plate

3. The local stresses in the welds attaching the sponson bottom to the hull side have been analyzed to determine the local weld stress levels. The analysis shows an adequate margin of safety. A plate connecting the end of the lateral stiffener to the hull side is optional.

4. No additional structure to support the HPS unit mounting flange is required. The results of load case B show a margin of safety of 0.4. Therefore, a reduction in hull side thickness is not recommended.

2.8.7 Analysis of Ramp Hinges

This analysis was performed to examine the strength of the hinge pin supporting the rear ramp and also of the welds ing the lower portion of the hinge to the rear skid plate of the hull. This mounting structure is illustrated in Drawing No. 60011-40233, "Ramp Installation".

2.8.7.1 Loading Conditions

The load placed on the hinges was the weight of the rear ramp (481 lbs.) multiplied by the vertical g load factor (22), which was determined from the shock factors diagram presented in Section 2.8.5 of this report. The resultant load is 10,582 lbs., uniformly distributed laterally across the ramp.

2.8.7.2 Analysis Method

2.8.7.2.1 Resultant Reaction Loads

The problem was treated as if the ramp were a beam, simply supported in three places. This configuration is statically indeterminant. To solve the problem, the force method was utilized. This method involves the following steps:

1. Remove one of the redundant reaction equations.
2. Determine the resulting deflection at the point of removal.
3. Replace the reaction force and remove the distributed load.
4. Determine the new resulting deflection at the original point of load removal.
5. Equate the two deflections found and solve for the reaction force. This is possible as the total deflection at the support point equals 0.
6. Solve statically for the remaining reaction forces.

2.8.7.2.2 Stress Analysis

Stress analyses of the hinge pin and of the welds joining the hinge to the rear skid plate were performed with standard shear stress, normal stress, and bending stress equations. In both cases, the load applied was the maximum r tion load determined by the method described in Section 2.8.7.2.1, above.

The hinge pin is in double shear. It was analyzed by determining the shear stress produced in it and the margin of safety for this load.

Stresses in the welds in the plane of the rear skid plate were found from the normal and shear force components and bending moment produced in this plane by the application of the downward vertical load at the pin to hinge contact point. The principal stresses produced in the welds were then determined, as were the resulting margins of safety for these stresses.

2.8.7.3 Results

2.8.7.3.1 Resultant Reaction Loads

Analysis of the rear ramp hinge reaction forces produced the following results:

Load on center hinge = 6613.75 lbs.

Load on end hinges = 1984.12 lbs.

The center hinge load was used for the stress analysis.

2.8.7.3.2 Stress Analysis

The stress analysis showed that the hinge pin and hinge welds easily survived the applied load. Table 2.8-10 presents the resulting margins of safety.

Table 2.8-10. Results of Ramp Hinge Stress Analysis

Area Analyzed	$M.S._{Normal}$	$M.S._{Shear}$
Hinge pin	13.55	12.34
Welds joining hinge to rear skid plate	--	8.08

2.8.8 Analysis of Lifting Eye Mooring Bitts

Four lifting eye/mooring bitts are mounted to the top surface of the ATR. This analysis was performed to determine if the mounting bolts and the surrounding hull structure to which the eyes are bolted are structurally adequate to carry the loads incurred during transport and hoisting of the ATR.

2.8.8.1 Analysis Method

The analysis was performed in accordance with MIL-STD-209E. Standard shear, normal, and bending stress equations were utilized. For those load cases where a normal force component was applied to a flat hull plate, the formula used to approximate the resulting stress was:

$$f_b = \frac{BP}{T^2}$$

where, f_b = the resulting bending stress

B = a constant whose value depends on the dimensions of the area to which the load is applied and to the overall dimensions of the stressed plate

P = the applied load

T = the thickness of the stressed plate

This formula is actually intended for the determination of the stresses resulting in a large rectangular plate upon application of a uniform load to a small central area of the plate. For the configurations analyzed, the formula provided a reasonable approximation of the magnitude of the resulting stress.

2.8.8.1.1 Applied Loads

Four load conditions were applied to both a forward and a rear eye, as detailed below:

I Lifting Load

A. 2.3 times the portion of the vehicle weight supported by the eye, applied in the direction of a pick-up point above the c.g. of the vehicle. A detailed explanation of the determination of the pick-up point is provided in MIL-STD-209E. The vehicle weight was assumed to be 28,417 pounds, as per the report R-60011-00004D. For the tie down loads, it was assumed that one fourth of the total weight was applied at each eye. For the lifting load, the actual portion of the total weight supported by each eye was determined.

II Tie Down Loads

A. 4.0 times the portion of the vehicle weight supported by the eye, applied longitudinally.

B. 2.0 times the portion of the vehicle weight supported by the eye, applied vertically.

C. 1.5 times the portion of the vehicle weight supported by the eye, applied laterally.

2.8.8.1.2 Design Criteria

The design criteria used, for each component analyzed, was that the ultimate allowable stress of a component be greater than or equal to 1.5 times the working stress applied to that component.

2.8.8.1.3 Areas Analyzed

As stated previously, both the securing bolts and the weldment areas to which the eyes are bolted were analyzed. For bending stressed induced in a hull plate by forces applied perpendicular to that plate, an effective area over which the stress could act was assumed. In cases where a hull plate failed under one of the design loads, the necessary plate thickness for support of the load was determined.

2.8.8.2 Results

The results of this analysis are presented in Table 2.8-11. In all cases, the securing bolts are of adequate strength to support the applied loads. The hull weldment plates, however, failed under application of several of the loads, indicating that an increase in plate thickness is required. This was provided by the addition of reinforcement plates, internal to the hull, in the areas of the eyes. The necessary reinforcement plate thicknesses are 0.28 inches for the rear lugs and 0.24 inches for the forward lugs, as indicated in Table 2.8-11.

Table 2.8-11 Results of Lifting Eye/Mooring Bitt Analysis

Loading Condition	Rear Lugs					Forward Lugs				
	Bolts		Hull Top Plate			Bolts		Upper Front Glacis		
	Normal Stress F.S.	Shear Stress F.S.	Normal Stress F.S.	Needed Thickness For F.S.=1.5	Doubler Plate Thickness	Normal Stress F.S.	Shear Stress F.S.	Normal Stress F.S.	Needed Thickness For F.S.=1.5	Doubler Plate Thickness
Longitudinal Tie Down	-	3.94	10.39	-	-	7.02	4.09	1.24	0.84"	0.09"
Vertical Tie Down	15.75	-	Fails	0.97"	0.22"	15.10	13.86	Fails	0.96"	0.21"
Lateral Tie Down	-	10.50	13.59	-	-	-	10.50	13.46	-	-
Lifting Load	9.61	6.95	Fails	1.03"	0.28"	8.33	5.46	Fails	0.99"	0.24"
Required Reinforcement Plate Thickness For Support Of All Loads				0.28"					0.24"	

2.8.9 Analysis of Towing Lugs

Two towing lugs are located at each end of the ATR in order to provide attachment points for recovery equipment, should the ATR require towing. The forward of towing lugs are bolted to the lower part of the lower front glacis and the rear lugs are bolted ot the rear skid plate. The purpose of this analysis was to determine if the bolts anchoring the tow lugs and the hull plates to which the lugs are bolted will withstand the loads induced during towing of the ATR.

2.8.9.1 Analysis Method

The method of analysis used in examining the towing lugs was similar to that used in examining the lifting eye mooring bits, as discussed in Section 2.8.8. Standard shear, normal and bending stress equations were used. Also used was the equation for approximating the bending stress induced in a flat plate by a concentrated normal load, as presented in Section 2.8.8.1.

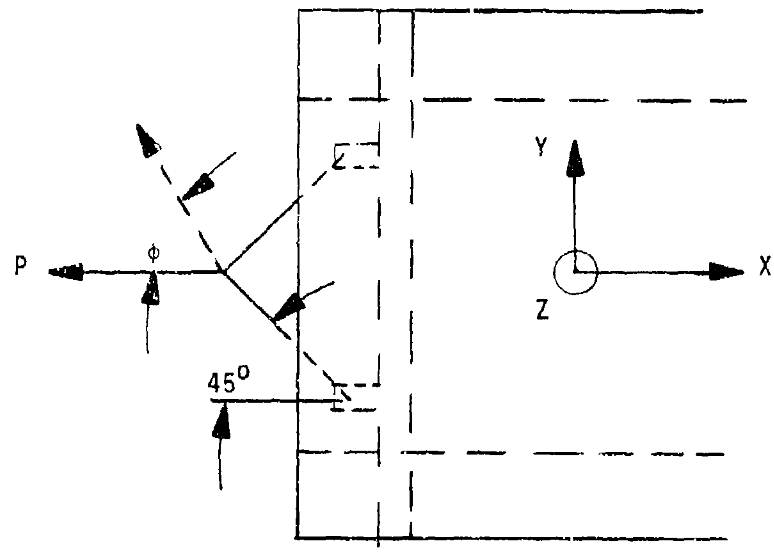
2.8.9.1.1 Applied Loads

The design load for towing lugs, as stated in the "ATR Design Criteria", R-60011-00002, is "a static equivalent ultimate load, equal to the gross vehicle weight..., applied at any position within a 120° cone with the horizontal axis, times a safety factor of five." Since two lugs are normally employed during towing operations, the load was proportioned equally between them. Figure 2.8-14 depicts the assumed load application geometry used for this analysis.

The following four load cases were examined:

1) P applied along X axis, $\theta = 0°$, $\emptyset = 0°$
2) P applied in X-Z plane, $\theta = 60°$, $\emptyset = 0°$
3) P applied in X-Z plane, $\theta = -60°$, $\emptyset = 0°$
4) P applied in X-Y plane, $\theta = 0°$, $\emptyset = 60°$

It was assumed that the vehicle weight, P, equaled 28,417 pounds, as per the report R-60011-00004D. Rather than applying a load equal to five times P to the lugs, the actual value of P was applied and the resulting stresses were determined. Then, the factors of safety for the stressed components, based on the ultimate strength of the component being analyzed, were determined, with the desired result being a safety factor in excess of 5.0.



$\theta_{max} = 60^{o}$

$\phi_{max} = 60^{o}$

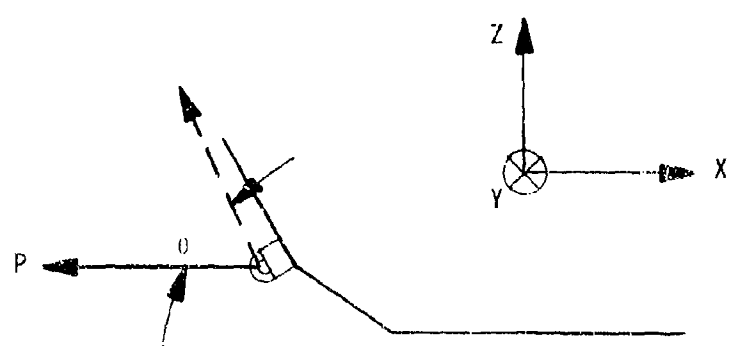


Figure 2.8-14. Load Application for Analysis Of Towing Lugs

2.8.9.1.2 Areas Analyzed

As with the lifting eye mooring bitt, both the anchoring bolts and the hull plates to which the lugs are bolted were analyzed. An effective area of stress was assumed for bending stresses caused by forces acting perpendicular to the hull plates. If a weldment plate was found to fail under a given load, the plate thickness was increased until structural integrity was achieved.

2.8.9.2 Results

The results of this analysis are presented in Table 2.8-12. In three cases, the anchoring bolts did not have the desired safety factor of 5.0. The minimum value was 4.64, which was deemed to be sufficiently close to 5.0 to be acceptable.

The hull plates, however, were not sufficient to support the applied loads in most of the cases examined. In addition, it became evident early in the analysis that very thick doubler plates would be required to allow the hull plates to withstand the applied loads and maintain a safety factor of 5.0. This fact, coupled with past experience in mounting towing lugs, indicate that the criteria of a safety factor of 5.0 is unduely conservative. As a result, the plate thickness required to maintain a safety factor of 2.0 while supporting the design loads was also determined.

It was found that a 0.79 inch thick reinforcement plate would be required behind the forward lugs in order to attain a safety factor of 2.0, while the rear lugs do not require any reinforcement. To achieve a safety factor of 5.0, reinforcement plate thicknesses of 1.35 inches and 0.83 inches would be required behind the forward and rear lugs, respectively.

Table 2.8-12 Results of Tow Lug Analysis

Loading Condition	Forward Lugs							Rear Lugs						
	Bolts		Lower Front Glacis					Bolts		Rear Skid Plate				
	Normal Stress F.S.	Shear Stress F.S.	Normal Stress F.S.	Needed Thickness For F.S.=2	Doubler Plate Thickness	Needed Thickness For F.S.=5	Doubler Plate Thickness	Normal Stress F.S.	Shear Stress F.S.	Normal Stress F.S.	Needed Thickness For F.S.=2	Doubler Plate Thickness	Needed Thickness For F.S.=5	Doubler Plate Thickness
1	9.93	6.63	1.49	0.88"	0.13"	1.46"	0.71"	12.67	4.98	2.64	-	-	1.61"	0.49"
2	11.14	8.13	7.46	-	-	-	-	10.59	5.82	5.76	-	-	-	-
3	9.95	6.89	Fails	1.10"	0.35"	1.82"	1.07"	9.99	7.13	1.97	-	-	1.86"	0.74"
4	7.27	4.85	Fails	1.54"	0.79"	2.10"	1.35"	7.30	4.64	1.94	1.15"	0.03"	1.95	0.83"
Required Doubler Plate Thickness For Support Of All Loads			M.S. = 2 : 0.79" M.S. = 5 : 1.35"							M.S. = 2 : 0.03" M.S. = 5 : 0.83"				

2.8.10 Analysis of Vehicle Rear

This analysis was performed as a result of the large cutout made in the hull rear plate to accommodate the rear ramp of the ATR. The rear area of the hull is subjected to severe suspension system and final drive loads. It was thought that the large area removed for the ramp might render the hull rear structurally unstable upon application of these loads.

2.8.10.1 Applied Loads

The design loads were taken from the "ATR Design Criteria" Report (R-60011-00002). These loading conditions are described below:

1) A 71,000 pound radial load applied at the centerline of the sprocket hub of the final drive. For this analysis, this load was applied in the longitudinal direction, toward the vehicle rear.
2) An 8,400 pound radial load applied at the centerline of the sprocket hub of the final drive and a 4,200 pound load applied laterlly at the O.D. of the sprocket tooth, in either direction. Again, the radial load was directed longitudinally rearward. Individual load cases were performed with the 4,200 pound load oriented in each lateral direction and applied at the bottom of the final drive sprocket ring.

2.8.10.2 Design Criteria

The maximum stresses resulting from the application of the design loads were compared to the yield strength of the hull plate in which they occurred in order to determine a margin of safety for the plate. Since the applied loads included a safety factor, any resulting positive margin of safety was deemed acceptable.

2.8.10.3 Analysis Method

This analysis was performed using an NASTRAN finite element computer stress analysis. It was assumed that the rear ramp provided no structural support to the hull rear plate. Figure 2.8-15 presents the finite element model utilized in the analysis. The model encompasses the entire rear portion of the ATR hull structure, forward to road wheel station #5.

2.8.10.4 Results

Review of the computer analysis results indicates that application of the 71,000 pound radial load resulted in the highest levels of stress.

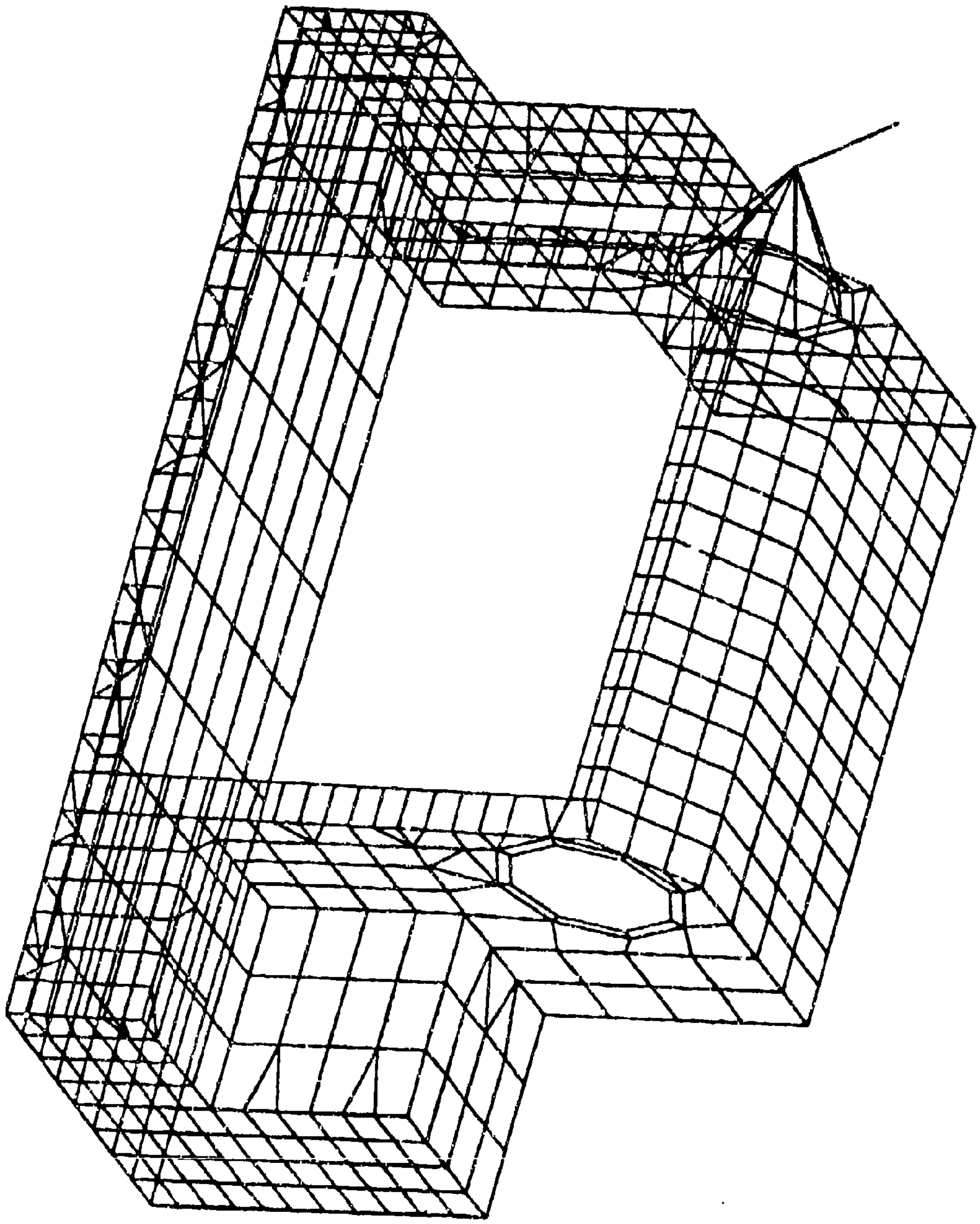


Figure 2.8-15. Finite Element Model of Vehicle Rear

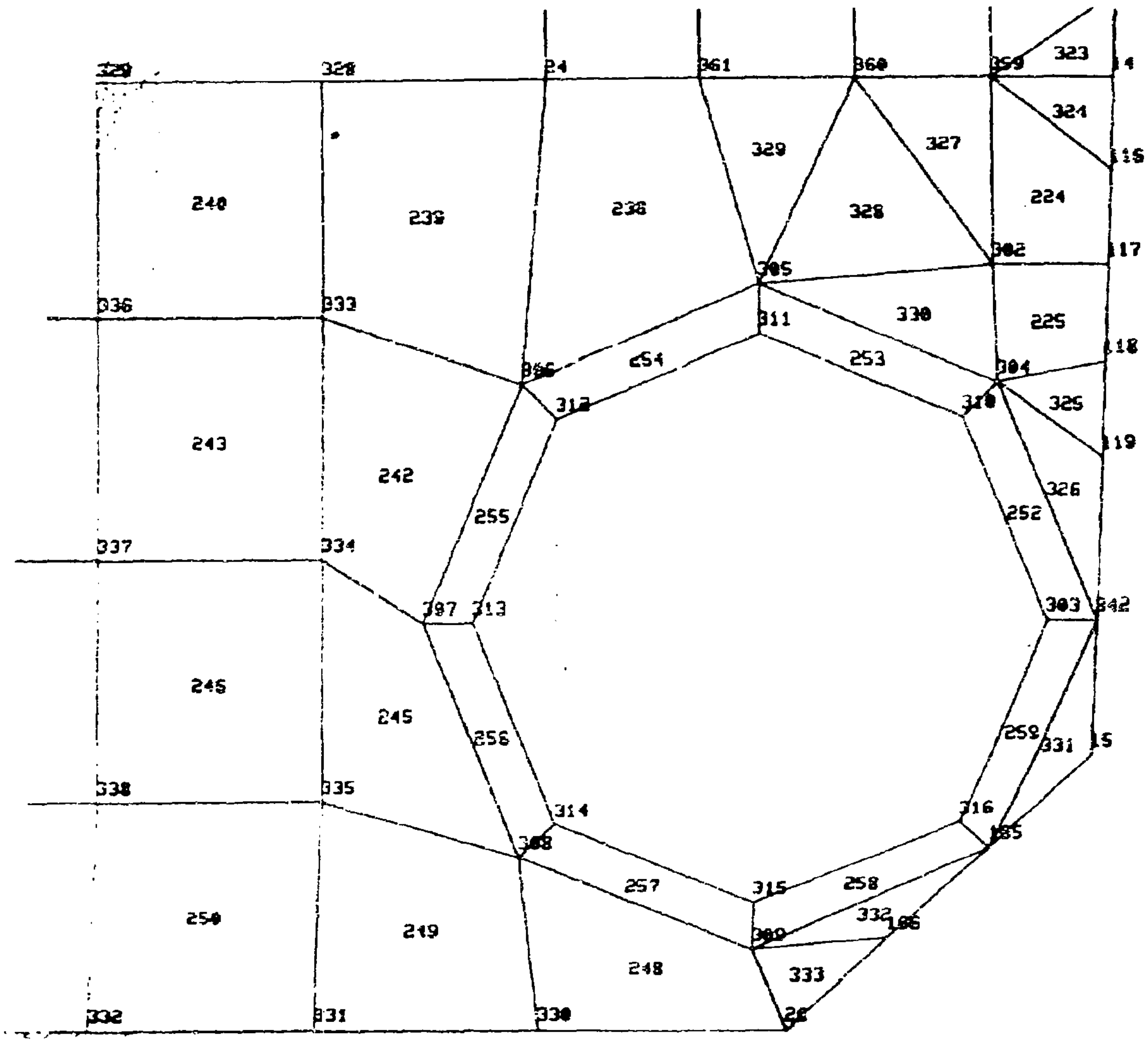
Table 2.8-13 provides a list of the hull weldment plates affected by this load, the stress generated in the plates by application of the load, and the margins of safety obtained. The margins of safety are based on the yield strength of the component, following welding.

Table 2.8-13 Results of Analysis of Hull Rear Subjected to 71,000 Pound Radial Load

Component Description	Stress Type	Stress (Ksi)	Margin of Safety
Rear Plate	Bending	11.61	1.84
Rear Plate	Shear	6.67	1.97
Bottom Plate	Bending	18.91	0.75
Rear Skid Plate	Bending	11.04	1.99
Sponson Side Plate	Bending	1.12	28
Hull Side Plate	Bending	24.94	0.32
Final Drive Mounting Ring	Bending	32.39	0.02
Waterjet Outlet Mounting Plate	Bending	9.05	2.65

Note that the final drive mounting ring maintains a margin of safety of only 0.02 in bending. This indicates that, while the ring itself is adequately strong to withstand the applied load, the weld joining the ring to the hull side must be of equal strength. The weld joining these components consists of two bevel welds and a fillet weld, as shown on Drawing #60011-40201. This weld configuration provides the strength required.

The maximum deflections occurring in the hull structure, as a result of the 71,000 pound radial load, also occur in the region of the final drive mounting ring. Figure 2.8-16 illustrates the computer finite element model of the hull area including the final drive mounting ring. The points of highest deflection are also indicated. It should be noted that the highest deflection is only 0.154 inches, which is well within an acceptable level.



Point	Deflection
306	0.132"
306	0.150"
312	0.139"
313	0.154"
314	0.114"
334	0.140"
337	0.105"

Figure 2.8-16. Maximum Deflections Resulting From Application of 71,000 Pound Load

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2.8.11 Analysis of Commander's/Driver's Seats

The commander's and driver's seats are located in the forward sponson areas. These seats are of a new design, which include several channels and rollers to facilitate seat adjustment. With the exception of the Grade 8 steel bolts, used in the seat assemblies, the entire seat frame is constructed of aluminum. A stress analysis was conducted for several areas of the seat, in order to determine their adequacy during road shock loads and the loads imposed during seat adjustment.

2.8.11.1 Analysis Method

2.8.11.1.1 Areas Analyzed

Ten individual areas of the seat frame were examined, as described in Table 2.8-14. Figure 2.8-17 identifies the location of the areas analyzed on the seat frame. The identifying numbers in the figure, correspond to those of the numbers listed in the table.

Table 2.8-14 Stress Analyzed Areas of Commander's/Driver's Seats

Identifying Number	Type of Stress Analyzed	Load Applied	Description of Area
1	Shear	Vertical Road Shock	Bolts joining lower roller channel to lower frame
2	Tension	Vertical Road Shock	Bolts anchoring seat to sponson bottom
3	Bending	Vertical Road Shock	Sponson bottom
4	Bending	Lateral Road Shock	Lower frame
5	Shear	Vertical Road Shock	Bolts joining upper roller channel to central frame
6	Shear	Vertical Road Shock	Weld joining the central frame side to the central frame bottom
7	Bending	Longitudinal Adjustment	Upper frame
8	Shear and Bending	Vertical Road Shock	Roller Axles
9	Shear and Bending	Longitudinal Adjustment	Teeth on position locking plates
10	Shear	Assumed Max. Shear Load	Upper seat locking pin

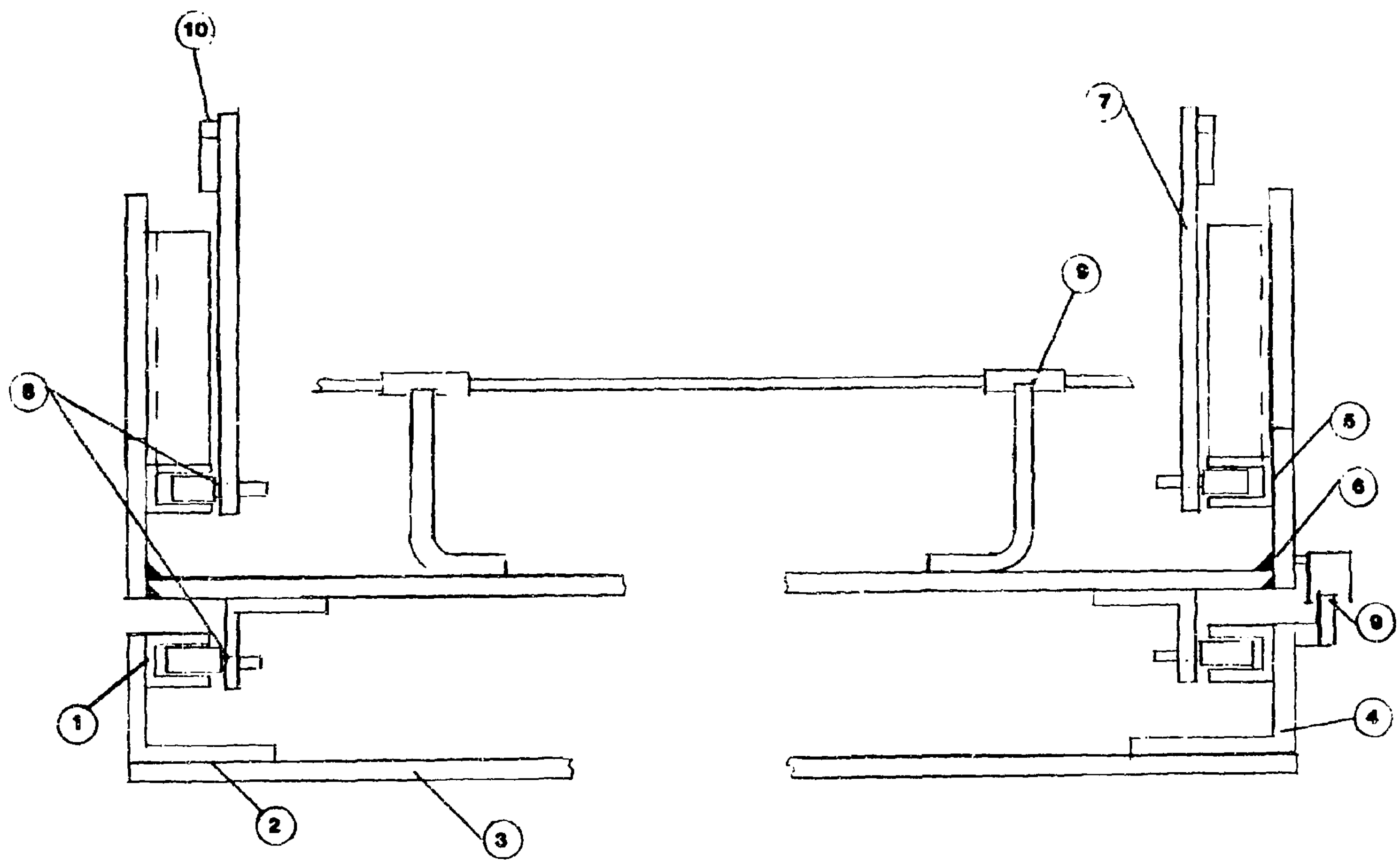

Figure 2.8-17. Stress Analyzed Areas Of Commander's/Driver's Seats

2.8.11.1.2 Applied Loads

The following four design loads were utilized in the analyses of the areas previously discussed:

1) Vertical Road Shock Load = weight of man, gear, and seat x g load
= 237 lbs x 5 = 1,185 lbs
2) Lateral Road Shock Load = weight of man, gear, and seat x g load
= 237 lbs x 5 = 1,185 lbs
3) Longitudinal Seat Adjustment Load = 250 lbs
4) Maximum Shear Load on Upper Seat Lock Pin = 250 lbs (assumed)

2.8.11.2 Results

Table 2.8-15 details the resulting margins of safety for each analyzed. The identifying numbers correspond to those in Table 2.8-14 and Figure 2.8-17. All margins of safety were determined based upon the yield strength of the material of the component being analyzed. An exception to this was area 8. In area 8, the ultimate strength of the roller axle shaft was used as the yield strength. Since the yield strength could not be determined from the data supplied by the manufacturer of the part.

Table 2.8-15. Results of Analysis of Commander's/Driver's Seats

Identifying Number	Margin of Safety
1	3.36
2	7.50
3	8.76
4	2.68
5	0.74
6	9.15
7	5.21
8	0.81 (Bending) 3.01 (Shear)
9	6.73 (Bending) 5.35 (Shear)
10	9.88

It should be noted that this type of roller is commonly used to support loads of magnitudes equal to those applied in this analysis.

All areas analyzed were found to be structurally adequate. The most critical area analyzed is area 5. A high shear stress level occurs in the bolts connecting the upper roller channel to the seat side frame upon application of a vertical road shock load. The load, in this case, is transmitted to the upper roller channel via a single roller, which may be located at any position along the channel, including a position forward of the first mounting bolt. In this forward position, the majority of the load is transmitted to the first bolt. The analysis was conducted assuming the entire shear load was applied to one bolt. Even under this severe load the bolt has a 0.74 margin of safety. This indicates that the five bolts that are used to attach the channel are more than adequate.

2.8.12 Analysis of Engine Mounting Structure

This analysis was performed to determine if the engine mounting structure which supports the ATR engine, transfer gear box, and hydrostatic and auxiliary pumps is sufficiently strong. Engine support is provided by three hard mounts, one located at the forward end of the engine and the other two at the rear of the engine where the transfer gear box is attached.

2.8.12.1 Analysis Method and Applied Loads

A hand stress analysis of the engine mounting structure was performed. The load applied at each mount was equal to the portion of the engine weight supported by that mount, multiplied by a g factor of 4.5. This load was applied independently in the vertical, lateral, and longitudinal directions and the resulting stresses were determined in suspect areas for each of the loading conditions.

2.8.12.2 Results

The results of the analysis indicate that the engine mounting structure is sufficiently strong to support each of the design loads. In no case examined was the stress produced in a component of the structure near the yield strength of that component.

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3.0 SUSPENSION

3.1 General Arrangement

The overall configuration and arrangement of the Automotive Test Rig (ATR) suspension system is shown in Figure 3.1-1. The suspension system consists of five dual roadwheel stations per side with springing and damping provided at each station by a Bird-Johnson Co. rotary hydropneumatic suspension unit. The AAI Corporation lightweight wire link track is employed along with two double support roller assemblies and a rear drive sprocket. A front compensating idler assembly is provided for track take-up during roadwheel travel. The compensating link connects the roadarm of number one roadwheel station and the front idler mount arm. Contained within the compensting link is a screw jack mechanism for adjusting track tension. The compensating link is equipped with a fusible link, shear pin, to prevent overloading of the number one station hydropneumatic unit output shaft and the track during compensation.

The front idler mount is a modified M113A2 type with a 17 1/4 inch diameter idler wheel. Roadwheel mounting spindles and hub assemblies are M113A2 type. The roadarms are 14 1/2 inches in length and have a 5.19 inch offset. They are fabricated of 4140 alloy steel martempered to obtain a tensile strength between 130 KSI and 150 KSI. Roadarm attachment to the hydropneumatic suspension unit output shaft is accomplished by a spline. The hydropneumatic suspension units are mounted to the ATR hull structure using bolts that engage structural inserts in the hull mounting flange. Sealing of the mounting surface is accomplished by a gasket installed between the hydropneumatic suspension unit mounting flange and the hull mounting flange.

Hydraulic supply and return lines to each hydropneumatic suspension unit are contained within the ATR hull.

As shown in Figure 3.1-1 the ATR suspension system is capable of being retracted such that the bottom surface of the track is approximately flush with the ATR hull bottom. Full retraction is accomplished for water operations to reduce vehicle water drag, and requires the front idler wheel to move forward approximately 5 inches. Idler forward movement is accomplished by movement of the compensating link due to rotation of the number one hydropneumatic suspension unit's output shaft.

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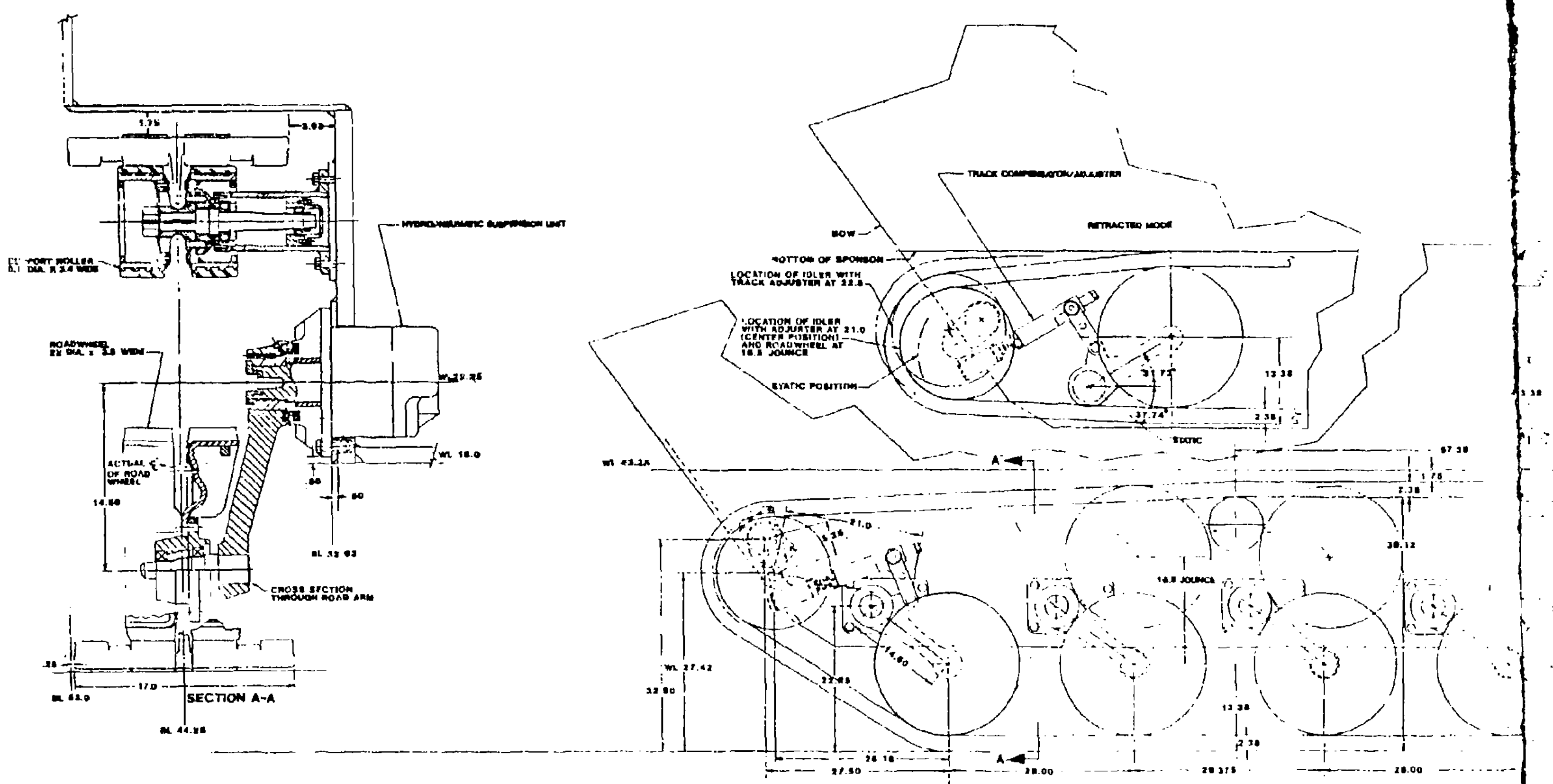

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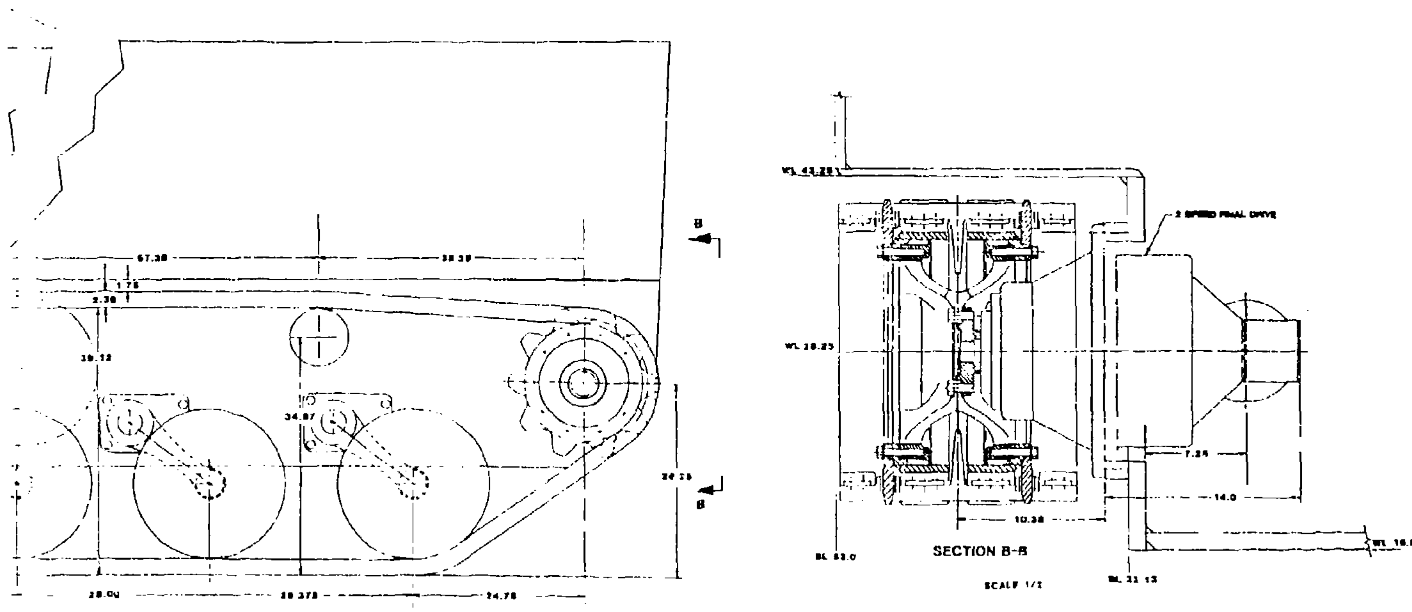

Figure 3.1-1 Automotive Test Rig
Suspension System

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Dual roadwheels of 22 inch diameter and 3 1/2 inch nominal widtn are employed at each roadwheel station. Each roadwheel is filled, within the disc and flange elements, with a foam material to provide buoyancy and reduce water drag.

Each roadwheel station provides for a jounce travel of 16.5 inches and a rebound travel of 4 inches. Jounce travel for stations #1, #2, #3, and #5 is limited by roadwheel/track contact with the hull sponson bottom plate. Station #4 is provided with a hull mounted bump stop that engages the roadwheel spindle extension at full jounce.

Retraction and extension of the ATR suspension system is accomplished by the operation of a switch located at the driver's station. Hydraulic fluid is supplied to the hydropneumatic suspension units from the ATR's auxiliary hydraulic system. Hydraulic fluid is supplied to each HPS unit by operation of a three position electrically controlled solenoid valve.

The time estimated for full retraction or extension of the ATR suspension system is 30 seconds. The total weight of the ATR suspension system, Group No. 3 of the ATR detailed weight breakdown provided in Appendix A of this report, is 6447.2 pounds. This is 22.7 percent of the ATR gross weight.

Engineering drawings of the various components of the ATR suspension system are provided in Volume III of this report.

Table 3.1-1 provides a listing of the M113A1/A2 suspension components and the Government furnished components used in the ATR suspension system.

3.2 Suspension/Mobility Analysis

3.2.1 Suspension Analysis

Based on discussions held with DTNSRDC personnel at the Automotive Test Rig Program kickoff meeting, held on 25 January 1984, it was agreed that suspension station configuration planned for the ATR in the proposal phase would be reconfigured to provide a ATR ground clearance of 16 inches. As shown in Figure 3.2.1-1 the originally proposed configuration used M113A1 type roadarms to reduce ATR development costs and provided a ATR ground clearance of 15 inches. With the hydropneumatic suspension unit rotational

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Table 3.1-1 Existing Vehicle Components Used In Automotive Test Rig Suspension System

COMPONENT NAME	PART NUMBER	QUANTITY	CURRENT APPLICATION	SOURCE
Roadwheel Assys.				
Spindle	10866123	8	M113A2 APC	AAI
Inner Bearing	712893	10		
Outer Bearing	712532	10		
Washer, Key	8756256	10		
Nut, Slotted	10866118	10		
Gasket	12253161	10		
Hub Cap	12253130	10		
Washer	10910174-33	40		
Hub Housing	12253131	10		
Bolt, Ribbed, Shoulder	8756378	80		
Seal	12253286	10		
Seal	11669372	10	M113A2 APC	AAI
Hydropneumatic Suspension Units				
HPS Unit	RD-HPS-101	5(LH)/5(RH)	None	DTNSRDC (GFE ITEM)
Roadarm/HPS Unit Shaft Cap	RD-HPS-157	10		
HPS Cap Seal, Outer	Parker 2-152	10		
HPS Cap Seal, Inner	Parker 2-131	10		
HPS Unit Cap Bolt, 7/8 - 9 Unc x 2" lg Cap Screw	Unbrako, Socket HD (Grade 8)	10	None	DTNSRDC (GFE ITEM)
Road Arm Assys.				
Shaft Seal	C10-80-6277	10	LVTPX12/LVTP7	DTNSRDC (GFE ITEM)

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Table 3.1-1 Existing Vehicle Components Used In Automotive Test Rig Suspension System (cont.)

COMPONENT NAME	PART NUMBER	QUANTITY	CURRENT APPLICATION	SOURCE
<u>Sprocket Assys.</u>				
Sprocket Wheel	55769-40018	4	M113 Wire Link Track Test Bed	DTNSRDC (GFE ITEM)
Sprocket Tire, Out.	55769-40015	4		
Sprocket Tire, Inn.	55769-40017	4		
Sprocket Carrier	10942567	2	M113 Wire Link Track Test Bed	DTNSRDC (GFE ITEM)
<u>Track Assy.</u>				
Track Pitch	55769-40024	138 Pitches		
<u>Front Idler Assy.</u>				
Inboard Idler	11669373	2	M113A2	DTNSRDC (GFE ITEM)
Outboard Idler	10907799	2		DTNSRDC (GFE ITEM)
Spindle Idler Support	11669356	2		AAI
Packing, Preformed	11669364	2		
Spindle, Idler	11669367	2		
Bearing Sleeve, Inner	11669365-2	2		
Bearing Sleeve, Outer	11669365-1	2		
Washer	11669363	2		
Cover	11669357	2		
Retaining Ring	11669368	2		
Bearing, Inner	712705	2		
Washer	10910174-33	8		
Hub Cap	12253130	2		
Gasket	12253161	2		
Bearing, Outer	11669370	2	M113A2	AAI

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Table 3.1-1 Existing Vehicle Components Used In Automotive Test Rig Suspension System (cont.)

COMPONENT NAME	PART NUMBER	QUANTITY	CURRENT APPLICATION	SOURCE
Front Idler Assy. (cont.)				
Hub Body	11669361	2	M113A2	AAI
Bolt, Ribbed, Shoulder	8756377	16		
Seal	11669372	2		
Nut, Slotted	10866118	2		
Washer, Key	8756256	2		
Washer	8763458	2	M113A2	AAI

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ϕ = 29.25°
Θ = 38.36°
α = 32.39°
ψ = 28.36°

TOTAL HYDRO UNIT ROTATION =100° (REBOUND TO JOUNCE)

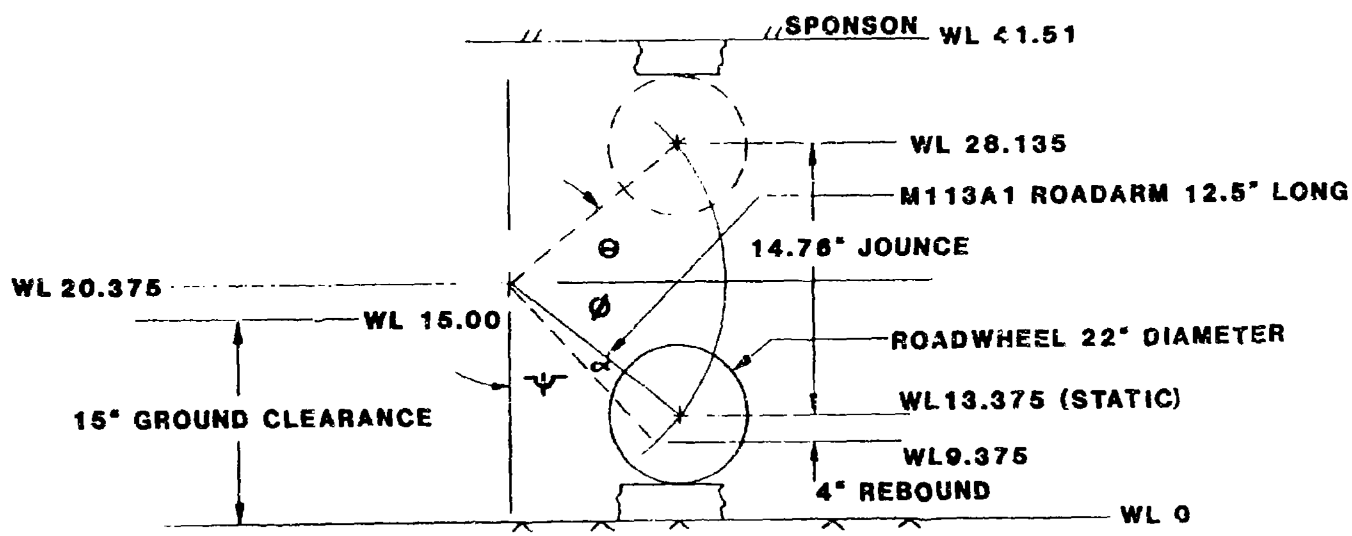

Figure 3.2.1-1 Original ATR Suspension Station Configuration

limit of a total of 100 degrees this configuration provided a total jounce travel of 14.76 inches. With this jounce travel full suspension retraction would result in a 15.00-14.76 = .24 inch track protrusion below the hull bottom, neglecting track sag and buoyancy effects in the water.

It was also agreed at the kick-off meeting that it would be desirable to move the suspension center of force more forward of the sprung mass center of gravity to provide better suspension performance in the vehicle pitch mode.

Detail analysis and investigations resulted in the final ATR suspension station configuration shown in Figure 3.2.1-2. This configuration provides for a 16-inch ATR ground clearance and requires the development of a new roadarm design. As shown, the configuration provides a total jounce travel (full retraction) of 16.5 inches. This jounce travel was selected to hopefully overcome the track sag effect and thus provide at full retraction in the water mode a track bottom surface that was flush with the ATR hull bottom. The roadarm length selected, 14.5 inches, requires a total rotational travel, full rebound to full jounce, of the hydropneumatic suspension unit of 94.35 degrees.

Design investigations conducted to evaluate the feasibility of shifting the suspension system forward, for better pitch motion characteristics, resulted in the revised roadwheel/idler/sprocket spacings shown in Table 3.2.1-1. The final configuration reflects the space required for packaging of the front idler compensating linkage and positioning of the support roller assemblies for clearance at full roadwheel jounce.

For the original ATR configuration, the sprung mass center of gravity location was estimated to be 53 inches aft of the first roadwheel. Thus, with the original ATR suspension configuration and equal spring rates at all stations, the suspension center of force is 55-53 = 2 inches aft of the sprung mass center of gravity. For the final ATR configuration the sprung weight characteristics are as follows.

ATR Sprung Weight = GVW - [1/2 roadarm weights + roadwheel weights + roadwheel hub weights + track segment on ground weight]

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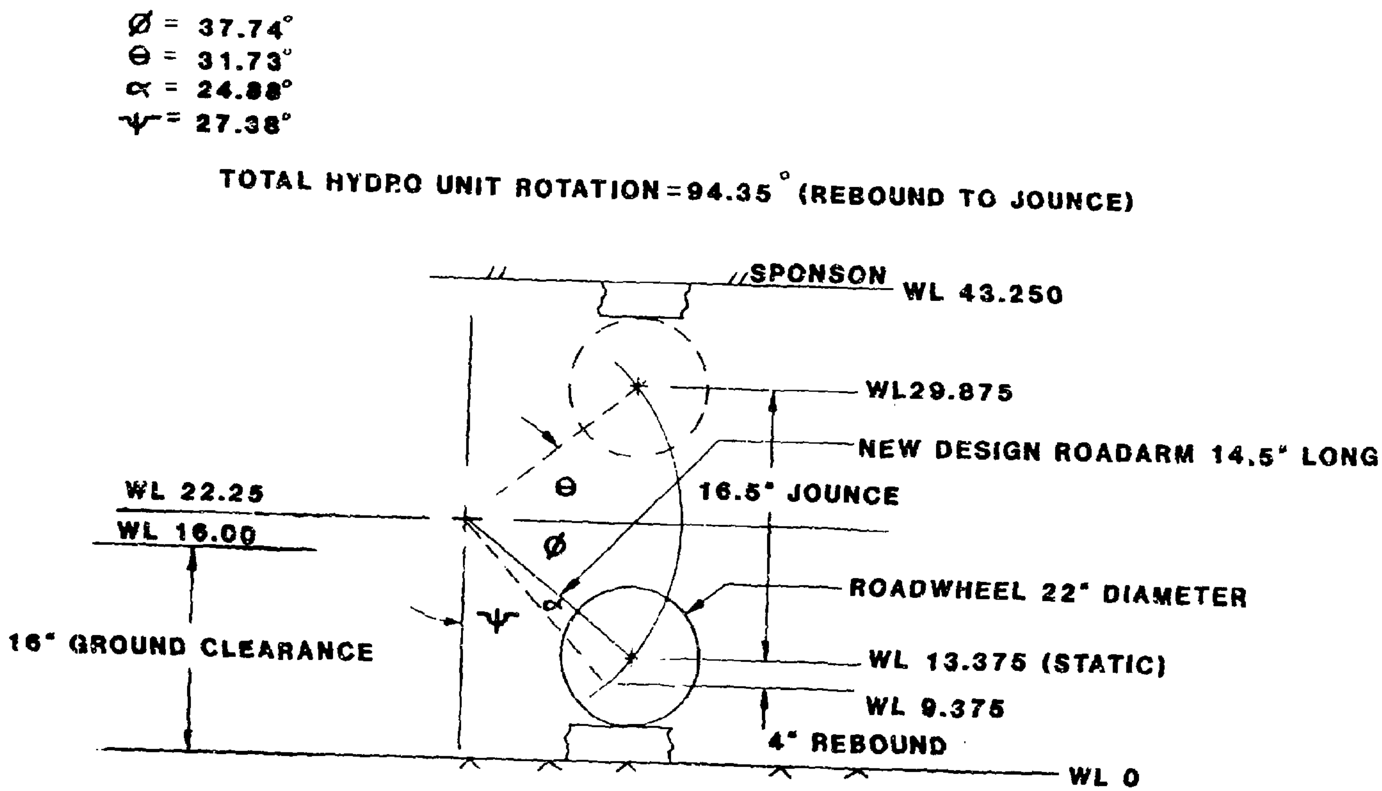

Figure 3.2.1-2 Final ATR Suspension Station Configuration

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Table 3.2.1-1 ATR Suspension System Spacings

SPACING (Centerline to Centerline)	ORIGINAL CONFIGURATION	FINAL CONFIGURATION
Front Idler to Roadwheel #1	26.00 inches	26.16 inches
Roadwheel #1 to Roadwheel #2	29.00	28.00
Roadwheel #2 to Roadwheel #3	26.00	29.38
Roadwheel #3 to Roadwheel #4	29.00	28.00
Roadwheel #4 to Roadwheel #5	26.00	29.38
Roadwheel #5 to Sprocket	24.75	24.75
Track Ground Contact Length (Roadwheel #1 to Roadwheel #5)	110.0	114.75

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ATR sprung Weight = 28417 -[1/2 x 427.5 + 876.9 + 216.3 + 627.3]
= 28417 - 1934 = 26483 lbs

ATR Sprung Mass Center of Gravity Longitudinal Location

$$\bar{x}_{sw} = \frac{(28417 \times 167.9) - (1934 \times 167.09)}{26483} - 85.82$$

$$= 167.96 - 85.82 = 82.14 \text{ inches forward of sprocket}$$

centerline or

$$\bar{x}_{sw} = (165.66 - 82.14) - 26.16 = 57.36 \text{ inches aft of}$$

roadwheel #1.

For the final ATR suspension system arrangement shown in Figure 3.2.1-3 and assuming equal spring rates at each roadwheel station the resulting suspension center of force location is as follows.

$$K_1 = K_2 = K_3 = K_4 = K_5$$

Summation of moments about roadwheel #1.

$$\bar{x}_{sf} = \frac{2[28K + 57.4K + 85.4K + 114.8K]}{2(K + K + K + K + K)}$$

$$= \frac{285.6}{5} = 57.12 \text{ inches aft of roadwheel \#1}$$

Therefore the resultant force location of the ATR suspension system is 57.36 - 57.12 = .24 inches forward of the ATR sprung weight center of gravity.

Providing greater spring rates at roadwheel stations #1 and #2 results in the following resultant force location relative to the ATR sprung weight center of gravity.

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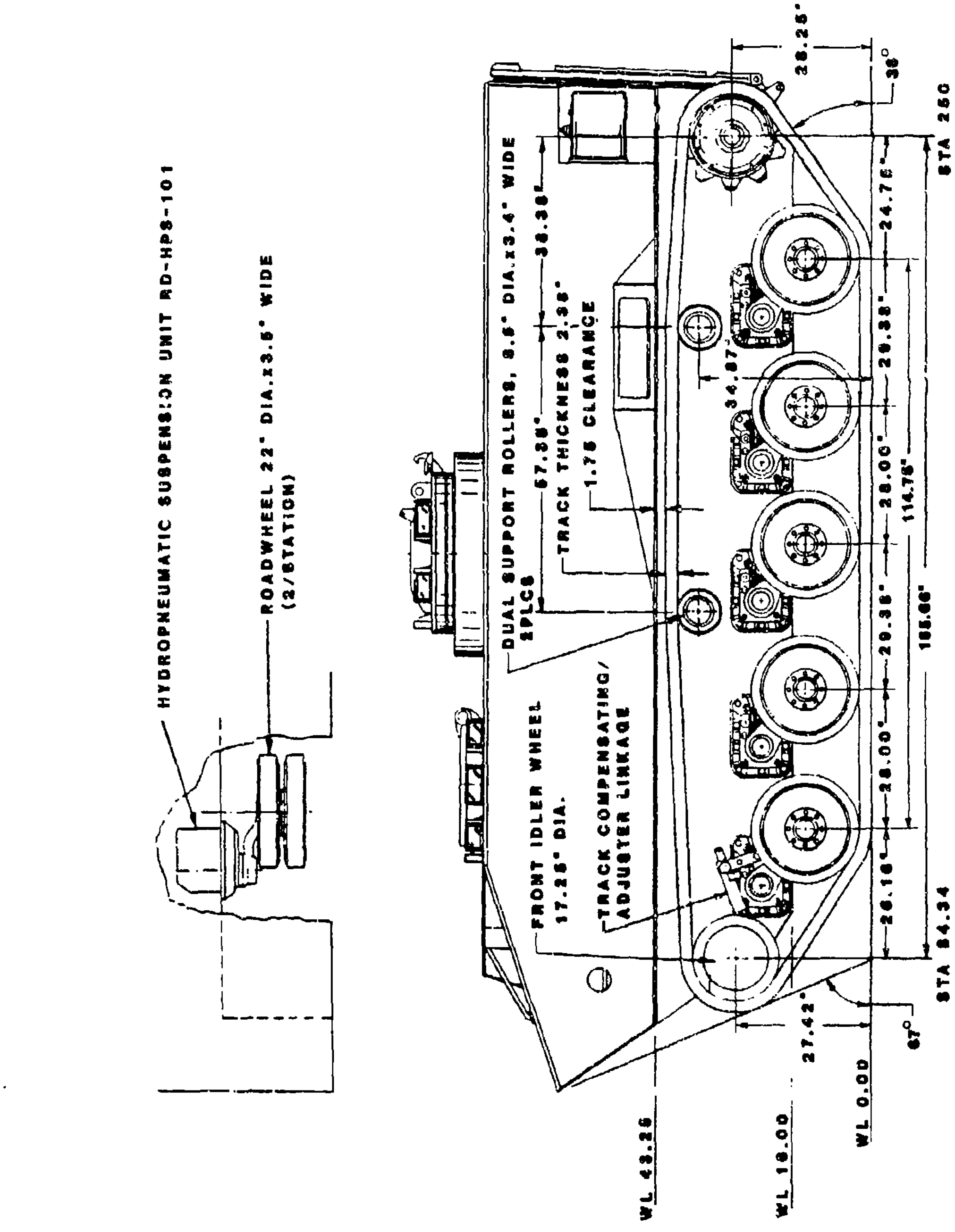

Figure 3.2.1-3 Final ATR Suspension System Arrangement

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Case I $K_1 = K_2 = 1.1K$ (10% greater on stations 1 and 2)

$K_3 = K_4 = K_5 = K$

$$\bar{x}_{sf} = \frac{K(30.8 + 57.4 + 85.4 + 114.8)}{K(1.1 + 1.1 + 1.0 + 1.0 + 1.0)}$$

$$= \frac{30.8 + 57.4 + 85.4 + 114.8}{5.2}$$

$$= \frac{288.4}{5.2}$$ = 55.46 inches aft of roadwheel #1 or 57.36 - 55.46 = 1.9 inches forward of sprung weight center of gravity.

Case II

$K_1 = K_2 = 1.2K$

$K_3 = K_4 = K_5 = K$

$$\bar{x}_{sf} = \frac{K(33.6 + 57.4 + 85.4 + 114.8)}{K(1.2 + 1.2 + 1.0 + 1.0 + 1.0)}$$

$$= \frac{291.2}{5.4}$$ = 53.9 inches aft of roadwheel #1 or 57.36 - 53.9 = 3.46 inches forward of the sprung weight center of gravity.

Therefore, to provide an ATR suspension system with a resultant force location several inches forward of the sprung weight center of gravity location the first and second hydropneumatic suspension units must be adjusted to provide between 10 and 20 percent greater spring rates than the remaining units.

Bounce motion of the ATR can be predicted on a simplified basis, neglecting damping and assuming linear spring rates for the hydropneumatic suspension units, using the following equation.

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$$f_b = \frac{1}{2} \sqrt{\frac{K_T}{\text{sprung mass}}}, \text{ CPS}$$

where:

f_b = bounce natural frequency, cycles per second

K_T = suspension spring rate, lbs/inch

= $2[K_1 + K_2 + K_3 + K_4 + K_5]$ for five suspension stations per side

= 10K for equal spring rates at all stations

$$\text{Sprung Mass} = \frac{28417 - 1934}{386}$$

$$= \frac{26483}{386}$$

$$= 68.61 \text{ lb-sec}^2\text{/inch}$$

Therefore for equal spring rates f_b equals the following:

$$f_b = \frac{1}{2\pi} \sqrt{\frac{10K}{68.61}}, \text{ cps}$$

$$f_b = .0608 \sqrt{K}, \text{ cps}$$

Solving for various spring rates yields the results shown in Table 3.2.1-2. Jounce capacity is based on an average station static load of 28417/10 = 2842 lbs and 16. inches of jounce travel.

$$\text{Jounce Capacity} = \frac{K \times 16.5}{2842}, \text{ g's}$$

From a tracked vehicle suspension performance standpoint a bounce frequency in the range of 1.0 to 1.5 cps and a jounce capacity between 2.5 and 3.5 g's is

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Table 3.2.1-2 Predicted ATR Bounce Natural Frequencies with Equal Spring Rates

STATION SPRING RATE, lbs/inch	f_b, CPS	JOUNCE CAPACITY, G's
300	1.053	1.74
350	1.137	2.03
400	1.216	2.32
450	1.290	2.61
500	1.360	2.90
550	1.426	3.19
600	1.489	3.48
650	1.550	3.77
700	1.609	4.06
750	1.665	4.35
800	1.720	4.64
850	1.773	4.93
900	1.824	5.23

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generally desirable. Considering now higher (+20%) spring rates on stations #1 and #2 results in the following bounce frequency equation.

$$K_T = 2[1.2K + 1.2K + K + K + K]$$

$$= 10.8K$$

$$f_b = \frac{1}{2\pi}\sqrt{\frac{10.8K}{68.61}}\text{, cps}$$

$$= .0631\sqrt{K}\text{ , cps}$$

Solving for various spring rates yields the bounce natural frequency values shown in Table 3.2.1-3.

The Bird-Johnson Company hydropneumatic suspension units provide nonlinear spring rates. Based on spring rate data available on 10 February 1984 for possible LVTPX12 application, the following approximate linear spring rates were estimated.

Basis: LVTPX12 GVW = 43,000 lbs
Roadarm length = 13 inches
Travel: jounce = 12 inches
rebound = 4 inches

8958 lbs
+ 4" (jounce)
Static position (2471 lbs) (2842 lbs AV. ATR static load)
-4" (Rebound)
460 lbs

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Table 3.2.1-3 Predicted ATR Bounce Natural Frequencies with Stiffer Spring Rates on Stations #1 & #2

STATION #1 & #2 SPRING RATES, LBS/INCH	STATIONS #3, #4 & #5 SPRING RATES, LBS/INCH	f_b, CPS	JOUNCE CAPACITY, STATIONS #1 & #2, G's
360	300	1.093	2.09
420	350	1.181	2.44
480	400	1.262	2.78
540	450	1.339	3.13
600	500	1.411	3.48
660	550	1.480	3.83
720	600	1.546	4.18
780	650	1.609	4.52
840	700	1.669	4.87
900	750	1.728	5.22
960	800	1.785	5.57
1020	850	1.840	5.92
1080	900	1.893	6.28

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Approximate spring rate over 8 inches of travel

$$K_{rebound} = \frac{2471 - 460}{4} = 503 \text{ lbs/inch}$$

$$K_{jounce} = \frac{8958 - 2471}{4} = 1622 \text{ lbs/inch}$$

Average K value over 8 inches of travel = 1063 lbs/inch

If no adjustments of the original hydropneumatic suspension unit operating pressures were made, the following approximate bounce frequency is estimated for the ATR with equal linear spring rates at all stations.

$$f_b = \frac{1}{2\pi}\sqrt{\frac{10 \times 1063}{68.61}}$$

$$= 1.98 \text{ cps}$$

$$\text{Jounce Capacity} = \frac{16.5 \times 1063}{2842}$$

$$= 6.17 \text{ g's}$$

Based on these approximate results, the original adjustments of the hydropneumatic suspension units provided spring rates that were too high for the ATR application.

Pitch motion of the ATR can be approximated, again assuming linear spring rates, by the following equation:

$$f_p = \frac{1}{2\pi}\sqrt{\frac{\sum K_i \quad L_L^2}{I_{\text{pitch of sprung mass}}}}, \quad \text{CPS}$$

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where:

K_L = spring rate of each station, i = 1 to 5

L_i = longitudinal distance of station relative to sprung weight center of gravity, inches

I_{pitch} of sprung mass = mass moment of inertia of ATR sprung weight, lb-sec^2-inch, about its center gravity.

For the ATR configuration suspension unit distances are as follows: (see Figure 3.2.1-3)

L_1 = 57.36 inches (fwd)

L_2 = 57.36 - 28.00 = 29.36 inches (fwd)

L_3 = 57.36 -[28.00 + 29.38] = .20 inches (aft)

L_4 = 57.36 -[28.00 + 29.38 + 28.00] = 28.02 inches (aft)

L_5 = 114.75 - 57.36 = 57.39 inches (aft)

I_{pitch} of Sprung mass = 202,166 lb-sec^2-inch.

Solving yields the following equation for the ATR pitch natural frequency with equal spring rates at all stations.

$$f_p = \frac{1}{2\pi}\sqrt{\frac{2K[3290 + 862 + .4 + 785 + 3294]}{202{,}166}}$$

$$= \frac{1}{2\pi}\sqrt{\frac{16462K}{202{,}166}}$$

$$= .0454\sqrt{K}\quad\text{, cps}$$

Solving for various spring rates yields the pitch natural frequency values shown in Table 3.2.1-4.

Considering again higher spring rates on stations #1 and #2 (+20%) results in the following pitch frequency equation.

$$f_p = \frac{1}{2\pi}\sqrt{\frac{2K[1.2 \times 3290 + 1.2 \times 862 + .4 + 785 + 3294]}{202\ 166}}$$

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Table 3.2.1-4 Predicted ATR Pitch Natural Frequencies with Equal Spring Rates

SPRING RATE, LBS/INCH	f_p, CPS
300	.786
350	.849
400	.908
450	.963
500	1.015
550	1.065
600	1.112
650	1.157
700	1.201
750	1.243
800	1.284
850	1.324
900	1.362

$$= \frac{1}{2\pi} \sqrt{\frac{18122K}{202\ 166}}$$

$$= .0477 \sqrt{K} \quad , \quad \text{cps}$$

Resulting pitch frequencies are shown in Table 3.2.1-5.

For the linear spring rate estimated for the Bird-Johnson Co. hydropneumatic suspension units, the following approximate pitch frequency is predicted for the ATR with equal spring rates at all stations.

$$f_p = \frac{1}{2\pi} \sqrt{\frac{16462\ X1063}{202,166}}$$

$$= 1.48 \text{ cps}$$

For the ATR configured with higher spring rates on suspension stations #1 and #2 to achieve a bounce frequency between 1.0 and 1.5 cps the spring rates shown in Table 3.2.1-6 are required. These spring rates result in the pitch frequencies and suspension jounce capacities also shown in Table 3.2.1-6. The comparison data indicates that as originally configured, for the LVTPX12 installation, the units will provide undesirable suspension characteristics for ATR application. Therefore, the hydropneumatic system pressures were adjusted to suit the ATR.

3.2.2 Mobility Analysis

As shown in Figure 3.2.1-3 the ATR suspension arrangement provides a track ground contact length of 114.75 inches. With the 17 inch width wire link track, the following nominal ground pressures result.

$$\text{Ground Pressure} = \frac{\text{GVW}}{2 \text{ X TWX TGCL}}$$

$$= \frac{28417}{2 \times 17 \times 114.75}$$

Table 3.2.1-5 Predicted ATR Pitch Natural Frequencies with Stiffer Spring Rates on Stations #1 & #2

STATIONS #1 & #2 SPRING RATES, LBS/INCH	STATIONS #3, #4 & #5 SPRING RATES, LBS/INCH	f_p, CPS
360	300	.826
420	350	.892
480	400	.954
540	450	1.012
600	500	1.067
660	550	1.119
720	600	1.168
780	650	1.216
840	700	1.262
900	750	1.306
960	800	1.349
1020	850	1.391
1080	900	1.431

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Table 3.2.1-6 ATR Suspension Characteristics

BOUNCE FREQUENCY, CPS	SPRING RATE, LBS/INCH[Δ]	JOUNCE CAPACITY, G's (16.5" Travel)	PITCH FREQUENCY, CPS
1.0	251	1.46	.756
1.1	304	1.76	.832
1.2	362	2.10	.908
1.3	425	2.47	.983
1.4	493	2.86	1.059
1.5	565	3.28	1.134
Comparison LVTPX12 Configured HPS Units			
2.06	1063	6.17	1.555

Δ - Stations #1 & #2 with 20% higher spring rates.

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= 7.28 psi. with troop payload

= 7.65 psi. with troop payload and 5% weight growth

The resulting fine grain soil strength in terms of cone index, VCI_1, required for one straight line pass is the following:

$$\text{Contact Pressure Factor} = \frac{28417}{2 \times 17 \times 114.75} = 7.28$$

Weight Factor = 1.0

$$\text{Track Factor} = \frac{17}{100} = .17$$

Grouser Factor = 1.0

$$\text{Bogie Factor} = \frac{28417/10}{10 \times 17 \times 5.81} = 2.88$$

$$\text{Clearance Factor} = \frac{16}{10} = 1.6$$

Engine Factor = 1.0

Transmission Factor = 1.0

$$\text{Mobility Index (MI)} = \left[\frac{7.28 \times 1.0}{.17 \times 1.0} + 2.88 - 1.6\right] \times 1.0 \times 1.0$$

$$= 44.10$$

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$$VCI_1 = 7.0 + .20\ MI - \frac{39.2}{MI + 5.6}$$

$$= 7.0 + 8.82 - .79$$

$$= 15$$

The ATR hull and suspension geometry provides the following L/T ratio.

$$\frac{L}{T} = \frac{114.75}{88.50} = 1.30$$

The hull and suspension arrangement provides the following approach and departure angles neglecting local protrusions.

Angle of Approach = 67°
Angle of Departure = 36°

With the front idler height of 27.42 inches, see Figure 3.2.1-3, and its approach angle of 67° degrees the estimated forward vertical wall capability of the ATR is 27 inches. Rear direction vertical wall climbing is estimated to be approximately 18 inches.

For bridging the following ATR capacity is estimated.

Ditch Length = 4/9 [165.66 + .7 (11 + 11)]

= 81 inches

Table 3.2.2-1 summarizes the estimated ATR mobility characteristics and provides comparison data for the LVTP7A1 and M113A1 vehicles.

The longitudinal slope stability of the ATR is demonstrated by the following analysis. As shown in Figure 3.2.2-1 for tipping on a 60% slope the moments about point A, the centerline of the rear roadwheel are the following:

Moments about point A^{+} = (43.89 x 14536) - (57.35 x 24358) + RF_y x d = 0

Since RF_y = 24358 lbs solving for d yields the following result:

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Table 3.2.2-1 ATR Mobility Characteristics

CHARACTERISTIC	ATR	LVTP7A1	M113A1
Ground Pressure, PSI.	7.28	7.72	7.80
L/T Ratio	1.30	1.45	1.23
Ground Clearance, inches	16.0	16.0	16.0
GHP/Ton	22.5	15.2	17.5
Angle of Approach, degrees	67	40	70
Angle of Departure, degrees	36	54	40
Bridging Capability, inches	81	96	66
Vertical Wall Capability, inches			
Forward	27	36 (hull contact allowed)	24
Rearward	18	-	17.0
Vehicle Cone Index, Fine Grain	15.0	16.0	17.0
Soils - one pass			
Roadwheel Travel, inches			
Jounce	16.5	10.0	7.5
Rebound	4.0	4.0	3.5

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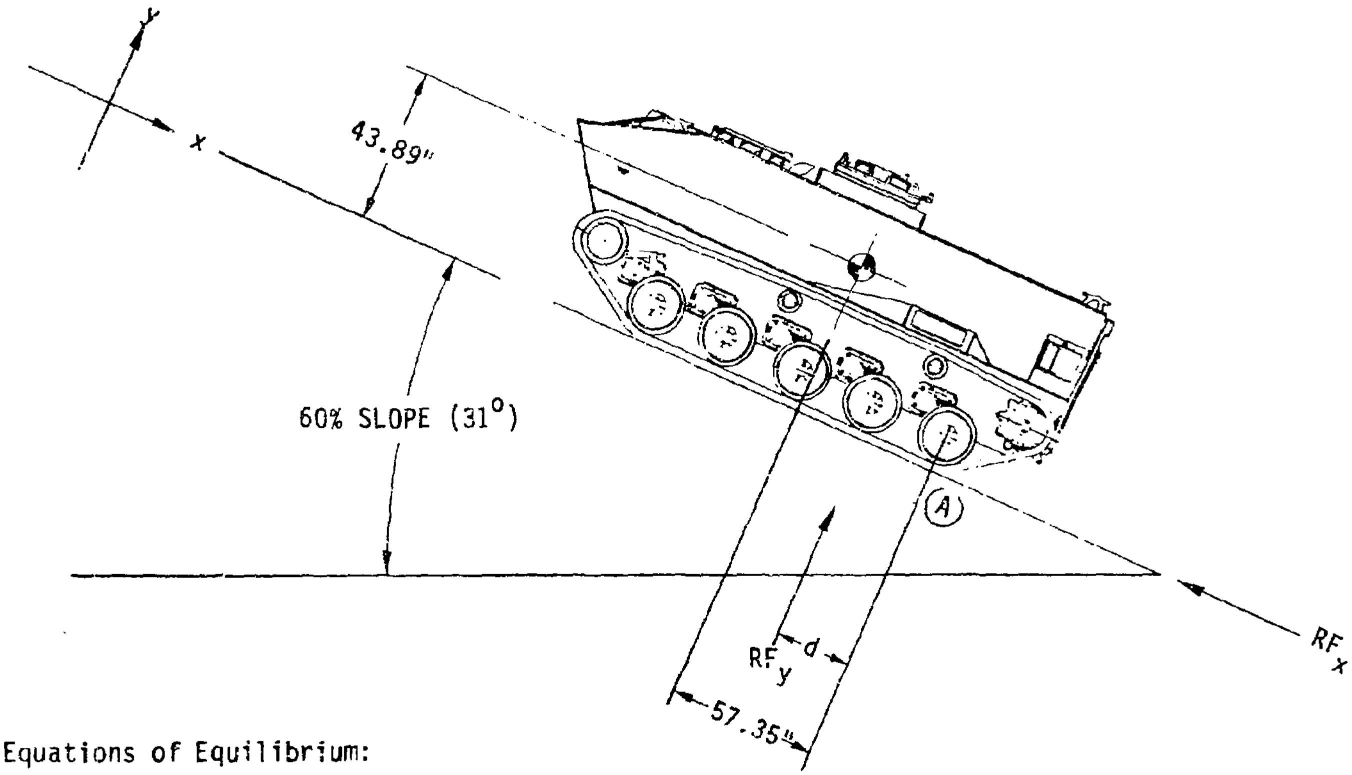

Equations of Equilibrium:

$$\Sigma F_y = RF_y - \cos 31^\circ \times GVW = 0$$

$$\Sigma F_x = RF_x - \sin 31^\circ \times GVW = 0$$

For GVW = 28417 lbs

$RF_y = \cos 31^\circ \times 28417 = 24358$ lbs

$RF_x = \sin 31^\circ \times 28417 = 14636$ lbs

Figure 3.2.2-1 ATR Longitudinal Slope Stability

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$$d = \frac{1396931 - 642374}{24358}$$

$$d = \frac{754557}{24358} = 30.98 \text{ inches}$$

Since this distance is within the track ground contact length the ATR will not tip over.

For sliding a rubber track pad sliding coefficient of friction, $\mu_{sliding} = .75$, is assumed. For sliding down a slope, the RF_x that can be developed at the track/ground interface equals

RFy x .75 = 24,358 x .75 = 18269 lbs

For no sliding on a 60 percent slope

RFx developed $\geq$ RFx required = sin 31° x 28417

Since 18,269 lbs is greater than 14,636 lbs, the ATR will not slide down a 60 percent slope.

The margin of safety for sliding is:

$$MS = \frac{18,269}{14,636} - 1 = +.25 \text{ (60\% slope)}$$

For stability on a 40 percent (21.8°) side slope the following anlaysis demonstrates the ATR stability. Solving the equilibrium equations shown in Figure 3.2.2-2 yields the following results:

$$R_{LH(SW)} + R_{RH(SW)} = \cos 21.8° \times 26,483 \text{ or}$$

$$R_{LH(SW)} = 24,589 - R_{RH(SW)} \quad . . . . \quad (2A)$$

$$8.5(R_{LH(SW)} + 967) + 97 (R_{RH(SW)} + 967) =$$

$$-(43.89 \times \sin 21.8° \times 26,483) + (52.75 \times \cos 21.8° \times 26,483)$$

$$8.5 R_{LH(SW)} + 8220 + 97R_{RH(SW)} + 93799 =$$

$$-431655 + 1297075$$

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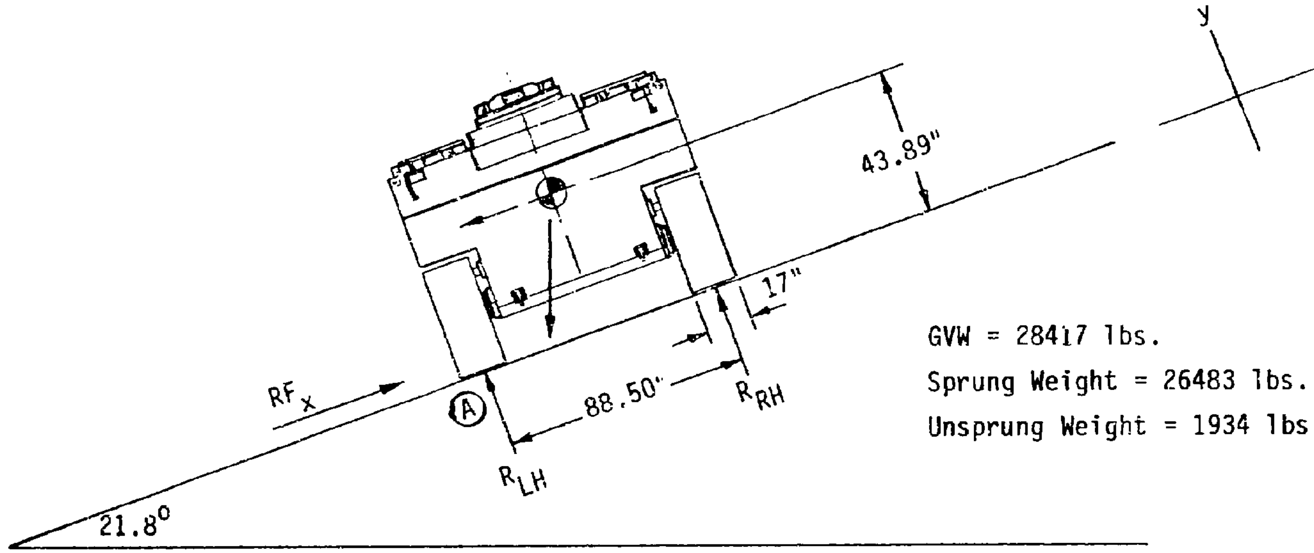

Equations of Equilibrium:

(1) ΣF_x: $RF_x = \mu\text{sliding}\ (R_{LH} + R_{RH}) = \text{Sin}\ 21.8^{o} \times 28417$ (For no sliding)

(2) $\Sigma F_y = 0 = R_{LH(SW)} + R_{RH(SW)} - \text{Cos}\ 21.8^{o} \times \text{Sprung Weight (SW)}$

(3) $\Sigma M_{A} = 0 = -(R_{LH} \times 8.5) - (43.89 \times \text{Sin}\ 21.8^{o} \times \text{Sprung Weight}) +$
$(52.75 \times \text{Cos}\ 21.8^{o} \times \text{Sprung Weight}) - (R_{RH} \times 97.0)$

Where $R_{LH} = R_{LH(SW)} + R_{RH(UNSW)}$ $\qquad R_{LH(UNSW)} = 967$ lbs

$R_{RH} = R_{RH(SW)} + R_{RH(UNSW)}$ $\qquad R_{RH(UNSW)} = 967$ lbs

Figure 3.2.2-2 ATR Lateral Slope Stability

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$$8.5R_{LH(SW)} + 97\ R_{RH(SW)} = 763401 \quad . . . . \quad (3A)$$

Substituting equation 2A into 3A yields

$$8.5[24589 - R_{RH(SW)}] + 97\ R_{RH(SW)} = 763,401$$

$$88.5\ R_{RH(SW)} = 763,401 - 209,007$$

$$R_{RH(SW)} = 6,264 \text{ lbs}$$

Therefore

$$R_{LH(SW)} = 24,589 - 6264 = 18,325 \text{ lbs}$$

The suspension deflection resulting from these sprung weight distributions can be calculated on a simplifed linear basis as follows:

$$\Delta F_{LH(SW)} = \frac{26483}{2} - 24589 = +11348 \text{ lbs}$$

$$\Delta F_{RH(SW)} = \frac{26483}{2} - 6264 = -6978 \text{ lbs}$$

Assuming a suspension vertical spring rate of 5 x 550 lbs/inch = 2750 lbs/inch per side yields the following deflections.

$$L_{LH} = \frac{11,348}{2750} = 4.13 \text{ inches}$$

$$L_{RH} = \frac{6978}{2750} = 2.54 \text{ inches}$$

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This results in an ATR roll angle of

$$\text{ATR Roll Angle} = \tan^{-1} \frac{4.13 \quad + 2.54}{88.50}$$

$$= 4.31°$$

This small roll angle will increase the effective side slope angle to 26.1 degrees. This increased slope will not significantly alter the ATR load distribution. Since $R_{RH(SW)}$ remains a positive quantity, a positive moment exists to prevent the ATR from tipping over on the 40 percent side slope.

For sliding down the slope, the force RFx that can be developed equals the following:

$$RFx \text{ (developed)} = \mu_{\text{sliding}} [R_{LH} + R_{RH}]$$

$$= .75 [18325 + 6264 + 1934]$$

$$= 19892 \text{ lbs}$$

For no sliding R_{Fx} (developed) $\geq$ RFx (required)

RFx (required) = sin 21.8° x 28417

= 10,553 lbs

Since 19,892 lbs is greater than 10,553 lbs, the ATR will not slide down the 40 percent side slope.

$$MS = \frac{19892}{10553} - 1 = +.88 \text{ (40\% slope)}$$

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3.3 Hydropneumatic Suspension Units

The ATR employs hydropneumatic suspension units rather than the more commonly used torsion bars. Hydropneumatic suspension units offer the performance benefits of nonlinear spring rates and variable damping over their wheel travel range. They also provide the capability for vehicle height reduction or, in the case of the ATR, retraction of the suspension system for reduced drag during amphibious operation. The HPS units have been specified as Government Furnished Equipment (GFE) and are depicted in Figure 3.3-1. These units are currently being developed for the Marine Corps Programs Office at DTNSRDC by the Bird Johnson Company of Walepole, Massachusetts. Bird Johnson has designed, built, and tested a similar type of suspension for the US Army M9 armored tractor.

The Bird-Johnson hydropneumatic units have several unique features. Each hydropneumatic unit contains a double vane rotary actuator. A splined shaft is connected to the rotating member to which a roadarm can be attached. Oil links the rotary actuator to a gas filled piston type accumulator with several internal passage ways in the unit body. The passages incorporate valving which controls dampening, leakage compensation, unit configuration (port or starboard rotation), and overpressure relief. All valve subassemblies can be replaced without major disassembly. The entire assembly is bolted together to form a rugged and compact unit which can be bolted directly to the hull side plate.

The suspension unit has the following specifications:

Actuator Displacement	23.3 cu. in./rad
Total Maximum Angular Travel	105 degrees
Operating Pressure	6500 psi max.
Accumulator Volume	46.8 cu in.
Unit Weight w/o Roadarm (Wet)	144 pounds

In order to properly position the roadarm an open spline space has been provided on the output shaft of the HPS units. The angular travel limitations of the missing spline space as the output shaft is rotated is shown in Figure 3.3-2.

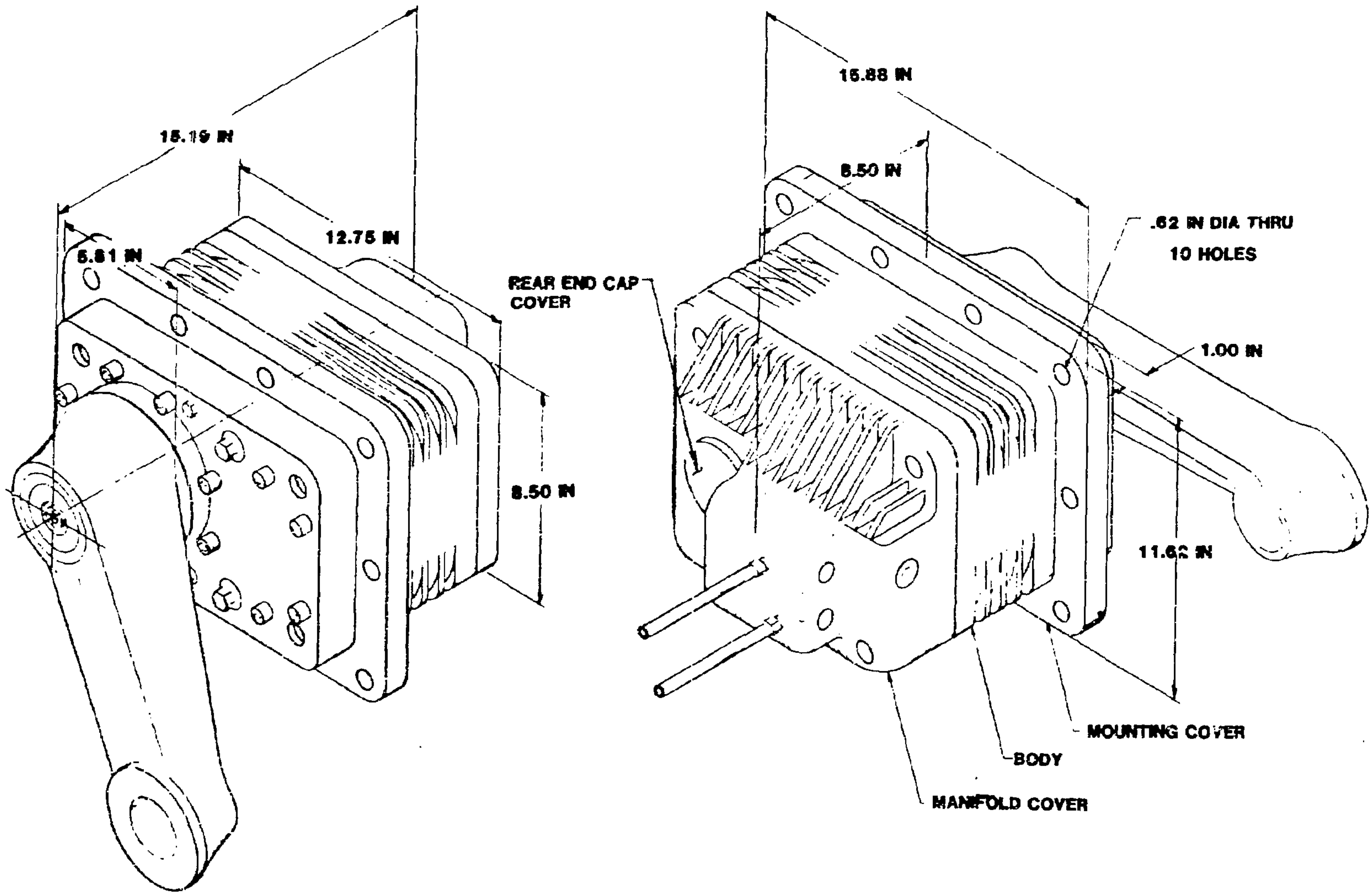

Figure 3.3-1 Bird-Johnson Hydropneumatic Unit

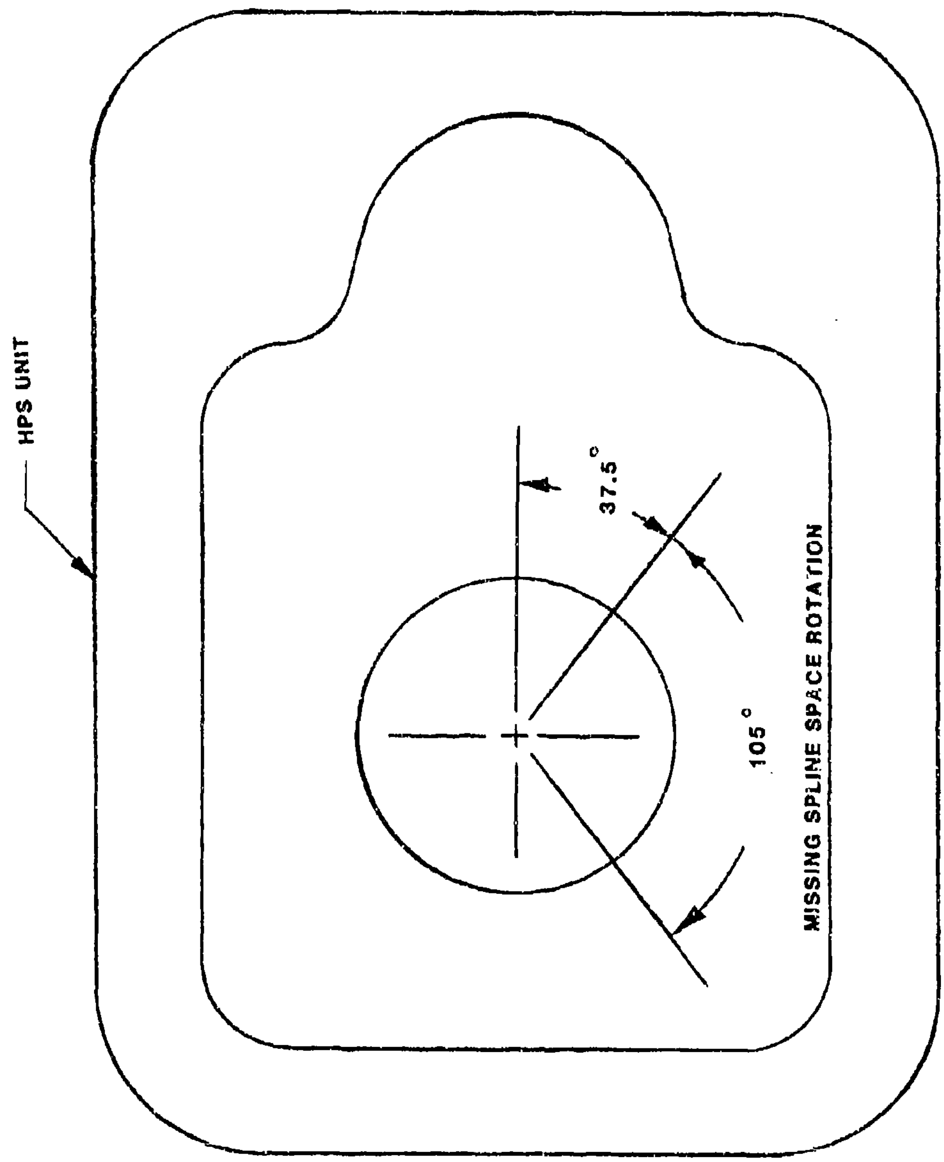

Figure 3.3-2 Rotation of Missing Spline Space on HPS Unit Output Shaft

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The HPS units are capable of being mounted on either side of the ATR. However, several modifications are required. The primary modification is to switch two pair of configuration cartridge valves. Other modifications include switching the inlet and outlet piping as well as providing the proper roadarm with the correct missing space to align with the missing spline. The hull machining and bolt pattern for mounting the HPS units is shown in Figure 3.3-3. For sealing purposes a 1/32" thick vellomoid gasket is required between the mounting flange and the hull.

3.4 Roadarms

The initially proposed ATR suspension system provided only 15 inches of ground clearance as opposed to the requested 16 inches. This was done in order to utilize the existing 12.5 inch long M113A1 roadarms with only minor modifications required to mount the arms to the HPS units. The reduced ground clearance was necessary due to the short roadarm combined with the 105 degree rotational travel limitation of the HPS units. The proposed suspension wheel station roadarm geometry is shown in Figure 3.2.1-1.

At the kick-off meeting held at AAI on 25 January 1984, the Marine Corps Programs Office requested that the 16 inch ground clearance be retained. To comply with this request the M113A1 roadarm was again evaluated for possible use on the ATR. It was determined that a roadarm length of 12.5 inches will be inadequate and a longer arm would be required. Further investigations were conducted to determine if another existing roadarm such as the 13 inch LVTP7 arm could be used. The evaluation indicated that a new roadarm design is required because of length and offset requirements.

AAI has designed a 14.5 inch roadarm for the ATR. The resulting suspension geometry is shown in Figure 3.2.1-2.

The roadarm design is shown in Figure 3.4-1. The 14.5 inch roadarm has an offset of 5.19 inches. An M113A1 spindle is installed in the lower end to permit an M113A2 roadwheel hub assembly to be mounted. On the upper end of the roadarm is a splined hole to accept the HPS output shaft. The location of a missing spline for alignment with the output shaft missing tooth is shown in Figure 3.4-1.

The ATR roadarm characteristics are compared to the M113A1 and the LVTP7 roadarms in the following table:

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Table 3.4-1 Roadarm Characteristics

Characteristics	ATR	M113	LVTP7
Length, inches	14.5	12.5	13
Offset, inches	5.19	4.25	5.81
Weight	42.0	24.0	30.0
Material	4140 Steel, Rc = 28-32	4140 Steel Rc = 32-37	4140 Steel Rc = 30-32
Method of Fabrication	Machined	Forged	Forged

It can be seen that the weight of the ATR roadarm is quite high by comparison. The roadarm weights could be reduced by providing a contoured shape such as the forged arms. This was not done in order to avoid high fabrication costs.

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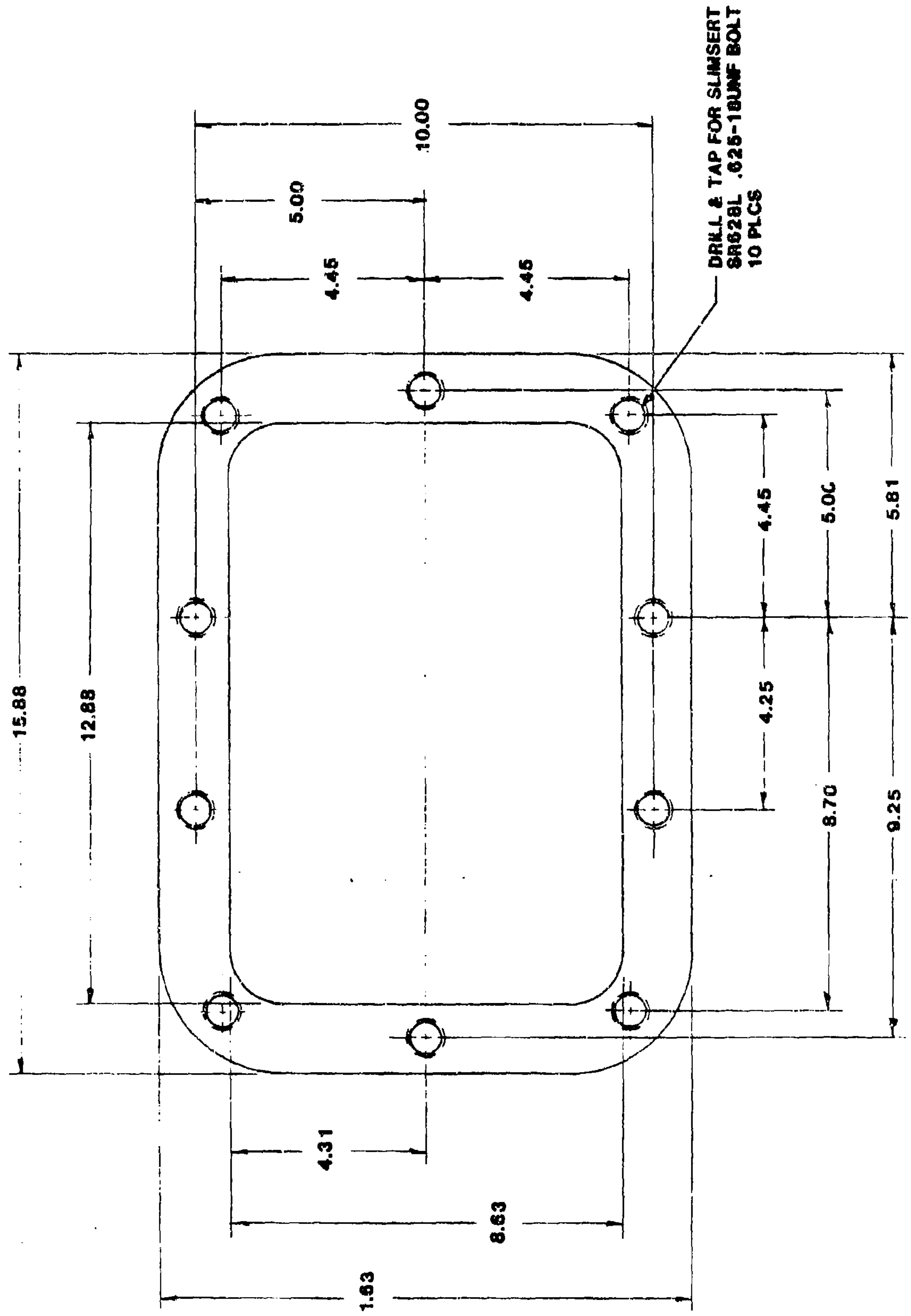

Figure 3.3-3 HPU Unit Required Hull Interface Dimensions

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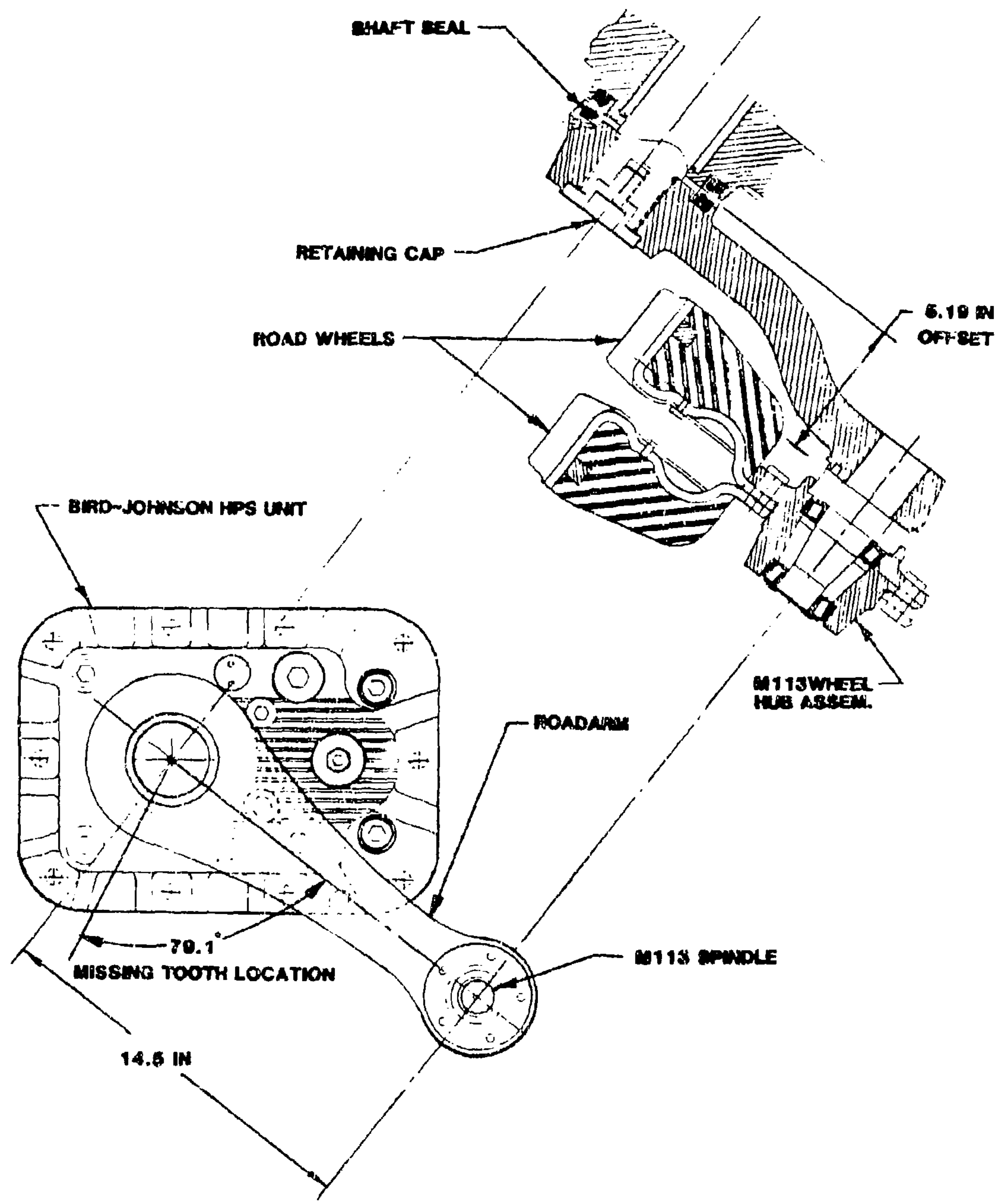

Figure 3.4-1 ATR Roadarm Installation

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3.5 Roadwheels

The roadwheel design developed for the Automotive Test Rig centered on maximum use of existing roadwheel designs so tooling and fabrication costs would be acceptable. The roadwheel design selected for the ATR was developed jointly by AAI and the Motor Wheel Corporation of Lansing, Michigan. The design employs the existing M113A2 roadwheel mounting surface so that the M113A2 roadwheel spindle and hub assemblies can be used.

As shown in Table 3.5-1 roadwheel designs currently employed on tracked vehicles have a wide range of configurational loading characteristics. Initially the M113A2 roadwheel, 24" diameter x 2 1/8" wide, was considered for the ATR. ATR roadwheel loads are expected to be in the following range.

$$\text{Average load} = \frac{28417}{2x5x2} = 1421 \text{ lbs}$$

$$= \frac{29838}{2x5x2} = 1492 \text{ lbs (5\% weight growth)}$$

The maximum load is expected to be on the order of 25 percent greater than average loads. Selection of this increase is based on maximum loads demonstrated by the M113A1 and LVTPX12 vehicles, see Table 3.5-2, which have weight distributions similar to that of the ATR. Therefore, the estimated ATR maximum roadwheel loadings are as follows:

1421 x 1.25 = 1776 lbs

1492 x 1.25 = 1865 lbs (5% weight growth)

With these estimated ATR roadwheel loads, the M113A1 roadwheel would have the loading characteristics shown in Table 3.5-3. As shown, these loading characteristics are significantly greater than the similar characteristics associated with the current M113 vehicle family applications of this roadwheel. In addition, the use of this 24 inch diameter roadwheel would require an ATR hull sponson bottom height of the following.

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Table 3.5-1 Current Roadwheel Loading Characteristics

VEHICLE	ROADWHEEL SIZE, ODXWIDTH	ROADWHEEL LOADS		LOAD/INCH OF WIDTH		Z FACTOR = LBS/INCH/DIAMETER	
		AVERAGE	MAXIMUM$^{\Delta}$	AVERAGE	MAXIMUM	AVERAGE	MAXIMUM
XM723 (M2)	24" x 4"	1875 lbs.	2155 lbs.	469 lbs./in.	539 lbs./in.	19.5	22.4
M113A1	24" x 2-1/8"	1171	1390	551	654	23.0	27.3
LVTPX12	26" x 4"	2056	2550	514	638	19.8	24.5
M551	28" x 2-3/4"	1675	2425	609	882	21.8	31.5
HSTV(L)	22" x 3"	1783	2430	594	810	27.0	36.8
M60A2	26" x 6"	4696	5625	783	938	30.1	36.1
M109A2	24" x 4"	1818	3050	455	763	19.0	31.8
M741	24" x 2-1/8"	1198	1745	564	821	23.5	34.2
M110	32" x 4"	2925	3650	731	913	22.8	28.5

Δ - Maximum values based on measured station loads.

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Table 3.5-2 Existing Tracked Vehicle Maximum Roadwheel Loads

VEHICLE	ROADWHEEL LOADS[Δ] AVERAGE	ROADWHEEL LOADS[Δ] MAXIMUM	RATIO OF MAX. LOAD TO AV. LOAD
M113A1	1390 lbs	1171 lbs	1.19
LVTPX12	2056	2550	1.24
XM723(M2)	1875	2155	1.15
M551	1675	2425	1.45
HSTV(L)	1783	2430	1.36
M741	1198	1745	1.46
M60A2	4696	5625	1.20
M109A2 SPH	1818	3050	1.68
M110 SPH	2925	3650	1.25

Δ - USATECOM Test Data.

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Table 3.5-3 M113A1 Roadwheel Loading Characteristics in ATR Application

LOADING CONDITION	LOADING/INCH OF WIDTH	Z FACTOR = LBS/INCH/DIAMETER
Average Load	669 lbs/inch	27.9
Average Load (5% Weight Growth)	702	29.3
Maximum Load	836	34.8
Maximum Load (5% Weight Growth)	878	36.6
Comparison M113A1 Loadings		
Average	551 to 564$^{\Delta}$	23.0 to 23.5$^{\Delta}$
Maximum	654 to 821	27.3 to 34.2

Δ - Data show reflects M113A1 & M741 vehicles of M113 family of vehicles.

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For 12 inches of jounce travel:

Sponson Height = 2(2.375) + 12 + 24 = 40.75 inches

For 14 inches of jounce travel:

Sponson height = 2(2.375) + 14 to 24 = 42.75 inches

For 16 inches of jounce travel:

Sponson height = 2(2.375) + 16 + 24 = 44.75 inches

Based on these loading characteristics and the suspension packaging penalties associated with use of a 24 inch diameter roadwheel AAI rejected the use of the M113A1 roadwheel.

To comply with the desired ATR sponson height of 44 inches, as shown in the DTNSRDC ATR concept, a roadwheel diameter of 22 inches was selected. This results in a ATR hull sponson height of 43.25 inches with jounce travel of 16.5 inches.

To provide realistic roadwheel loading characteristics and reduce roadwheel rubber tire loadings a roadwheel nominal width of 3.5 inches was selected.

ATR expected roadwheel loading characteristics are shown in Table 3.5-4.

The ATR roadwheel employs 2024 Aluminum Alloy and the mounting arrangement used on the current M113A1 roadwheel.

Compared to the current minimum loading characteristics of the M113A1 roadwheel the ATR roadwheel exhibits the following margins of safety.

$$MS = \frac{551}{406} - 1 = +.36 \text{ (average load)}$$

$$MS = \frac{551}{426} - 1 = +.29 \text{ (average load +5\% weight growth)}$$

$$MS = \frac{654}{507} - 1 = +.29 \text{ (maximum load)}$$

$$MS = \frac{654}{533} - 1 = +.23 \text{ (maximum load +5\% weight growth)}$$

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Table 3.5-4 ATR Roadwheel Loading Characteristics

LOADING CONDITION	LOADING/INCH OF WIDTH	Z FACTOR = LBS/INCH/DIAMETER
Average Load	406 lbs/inch	18.5
Average Load (5% Weight Growth)	426	19.4
Maximum Load	507	23.0
Maximum Load (5% Weight Growth)	533	24.2

Roadwheel Size: 22 inch diameter x 3.5 inch nominal width.

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In addition to reducing the roadwheel rubber tire loading per inch of width, the 3.5 inch width provides better load distribution into the track roadwheel pad area and track center misguiding tendencies will be reduced.

From a structural loading standpoint, the diameter of the ATR roadwheel is 2 inches smaller than the M113A1 roadwheel, thus the moments imparted due to lateral loads are approximately 8 percent smaller. Tire width is 1.6 times that of the M113A1 roadwheel.

The ATR roadwheel configuration is shown in Figure 3.5-1. It consists of a formed 3/8 inch thick aluminum disc with a welded reinforcing ring. A steel wear ring is attached with rivets to the inside surface of the wheel assembly. The interior of the wheel assembly is filled with polyurethane foam, density 10 to 15 pounds/ft^3, except in the area required for mounting on the roadwheel hub assembly. The foam is covered with a skin .060 to .090 inch thick that resists fracturing when a 150 pound load is applied by a 1 inch diameter spherical plunger. The foam installed in the ATR roadwheel provides buoyancy and reduces the water drag of the wheels in the amphibious operating mode. A detailed drawing of the ATR roadwheel assembly is provided in Volume III of this report.

The ATR roadwheel is fabricated of 2024-0 Temper ALCLAD Aluminum Alloy. After forming and welding it was heat treated to Temper T42. Although the welding of 2024 Aluminum Alloy is not generally recommended, the Motor Wheel Corporation has successfully fabricated roadwheels by this process for both the HSTV(L) and the DTNSRDC wire link track test bed. HSTV(L) roadwheels operated for over 3000 miles during testing at Aberdeen Proving Ground and Fort Knox without weldment problems. To insure the quality of the welding associated with the ATR roadwheels the reinforcing ring welds on all wheels were inspected using dye penetrant per MIL-I-6866. A welding sample was also be provided to verify the weld quality and penetration.

Alternate aluminum alloys for the ATR roadwheels, with better welding characteristics, were investigated. The alloys identified had strength properties that would require increased material thickness and significantly increase the roadwheel weight. The only alternate approach to the fabrication of the ATR roadwheels was a design with a formed lip, to replace the welded reinforcing ring. This would require the development of

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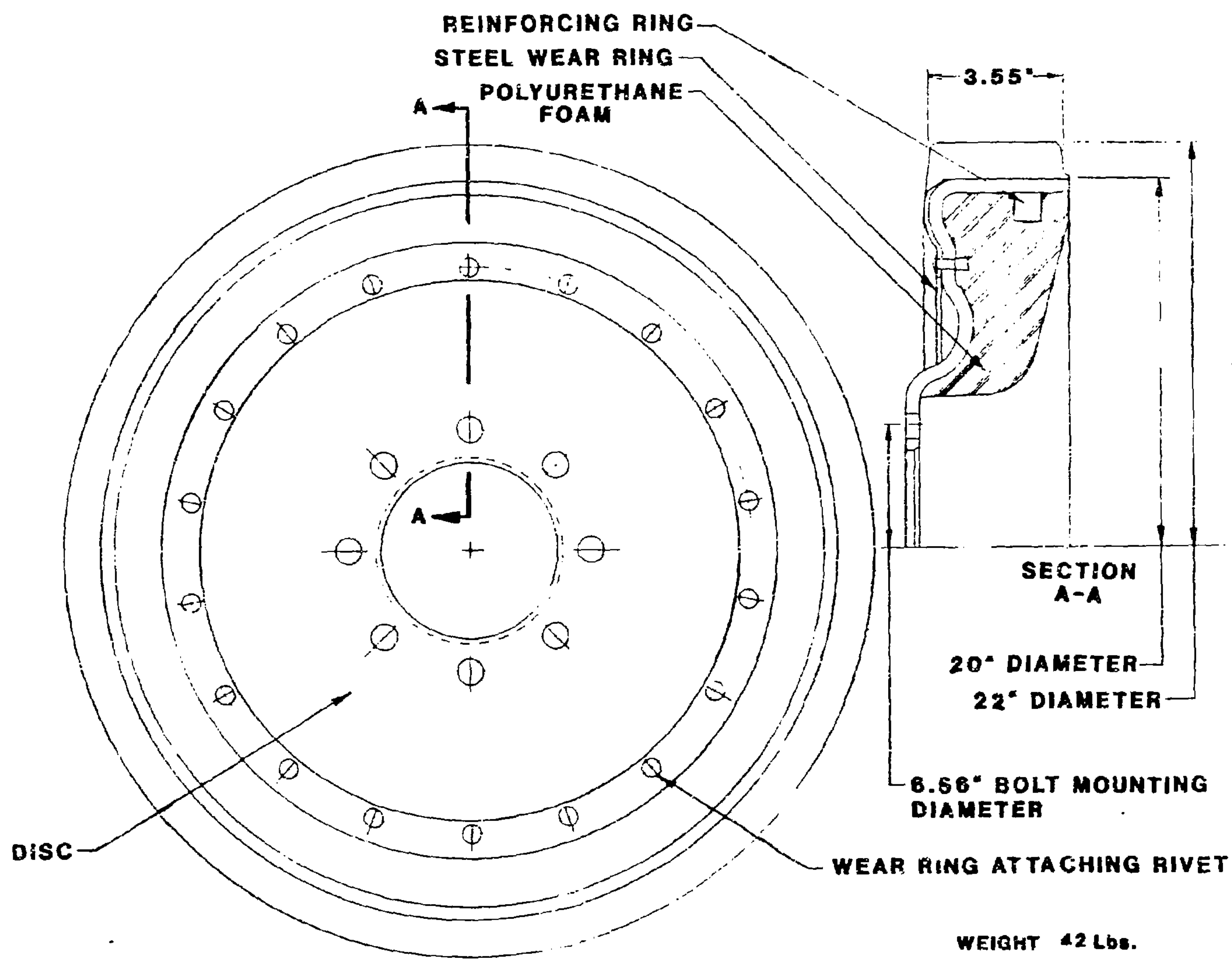

Figure 3.5-1 ATR Roadwheel Configuration

additional tooling with its associated costs and schedule impacts. The rubber tire vulcanized to the metal elements of the ATR roadwheel are per the requirements of MIL-W-3100.

The weight of the ATR roadwheel is 42 pounds. Its applied polyurethane foam will displace 25 pounds of seawater.

3.6 Support Rollers

As shown in Figure 3.2.1-3 the ATR suspension system employs two dual support roller assemblies per side. Two support roller assemblies were selected to provide good alignment of the track return segment relative to the ATR hull side plates, the front idler wheels and the rear drive sprocket assembly. In the vertical direction they provide track support to minimize track sag and constraint the track return segment from impacting the roadwheels during jounce travel. Their vertical positioning also provides 1 3/4 inches of clearance between the hull sponson bottom plate for passage of debris. The longitudinal spacings selected for the support roller assemblies were based on providing the clearances required for full track retraction, 16.5 inches of jounce, in the amphibious mode of operation.

Figure 3.6-1 shows the overall configuration and internal details of the support roller assembly. The support roller assembly is mounted to the ATR hull side plate by bolts. Support of the rollers is provided by a welded housing that contains bearings supporting a spindle. The roller assembly is attached to the spindle. Two rollers 8.5 inches in diameter by 3.4 inches in width are provided. Their mounting provides a slot for constraint of the center guide of the wire link track. Sides of the slot are provided with a replaceable steel wear ring.

The rollers are provided with urethane tires that are molded to the roller wheels. Lubrication of the roller assembly spindle shaft bearings is accomplished by SAE 30 weight oil that is contained within the housing assembly. A sight gage is provided for the checking of the oil level. Oil lubrication was selected for the design based on the rotational speeds of the roller assembly.

The weight of the support roller assembly with mount is 41.5 pounds (dry).

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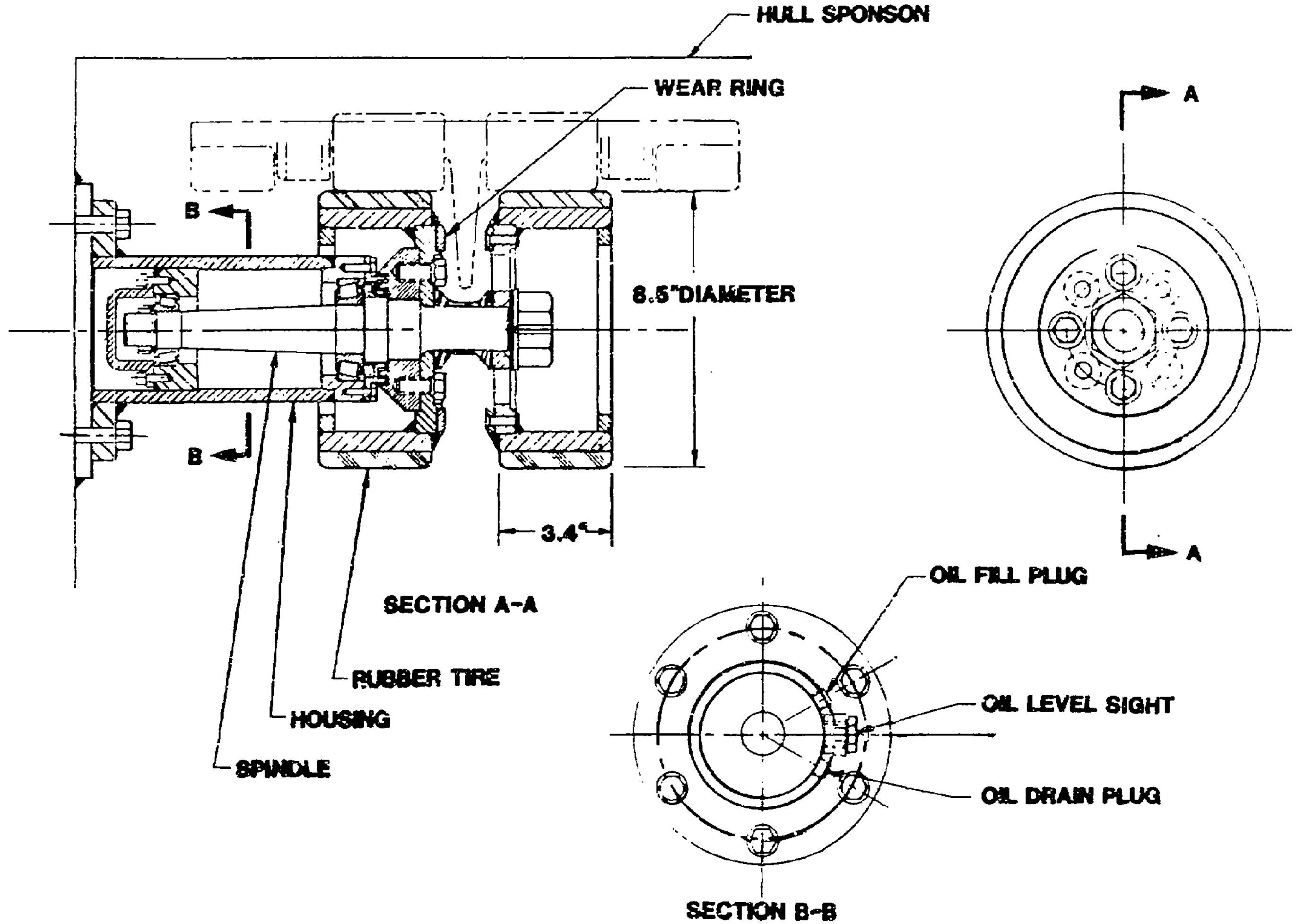

Figure 3.6-1 ATR Support Roller Assembly

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3.7 Compensating Idler

The ATR suspension system utilizes a front compensating idler arrangement as shown in Figure 3.1-1. This arrangement provides the following functions:

1. Manual adjustment of track tension.
2. Reduction of track slack due to roadwheel travel in landborne operations.
3. Retraction of the track to a nearly flush position with the hull bottom during waterborne operations, to reduce vehicle water drag.

As shown in Figure 3.1-1 the compensating idler assembly consists of an adjustable link that connects the number one roadarm to the front idler arm. Movement of the link due to roadarm jounce rotation causes the idler arm to rotate and move the idler wheel forward.

Initially, AAI proposed to use a standard M113A1 idler mount and track adjuster to configure the compensating idler arrangement. However, during the ATR detailed design effort, two undesirble characteristics associated with this configuration were disclosed. The first was that an interference was found to occur between the track adjuster and the front idler mounting hub. This interference occurs when the track adjuster is in the fully extended position and movement of the compensating idler is required to provide for track retraction in the waterborne mode.

The second undesirable characteristic involved the relationship between roadwheel vertical movement and idler wheel movement. As shown in Figure 3.7-1 for the initial compensating idler design, the idler wheel movement is the greatest around the roadwheel static position and its rate of increase declines as roadwheel travel increases. This is prohibitive since it would result in over-tensioning of the track. The desirable characteristic is a very small idler wheel travel for small roadwheel movements and large idler travel for large roadwheel movement.

Based on these design deficiencies, a new compensating link arrangement, shown in Figure 3.7-2, was developed. The link is configured to employ a turnbuckle type track adjuster rather than the previously considered M113A1 grease type adjuster. The use of the grease filled adjuster was eliminated because the offset mounting lugs required for attachment to the roadarm and idler arm would cause eccentric loadings. These loadings would result in leakage of the grease filled adjuster.

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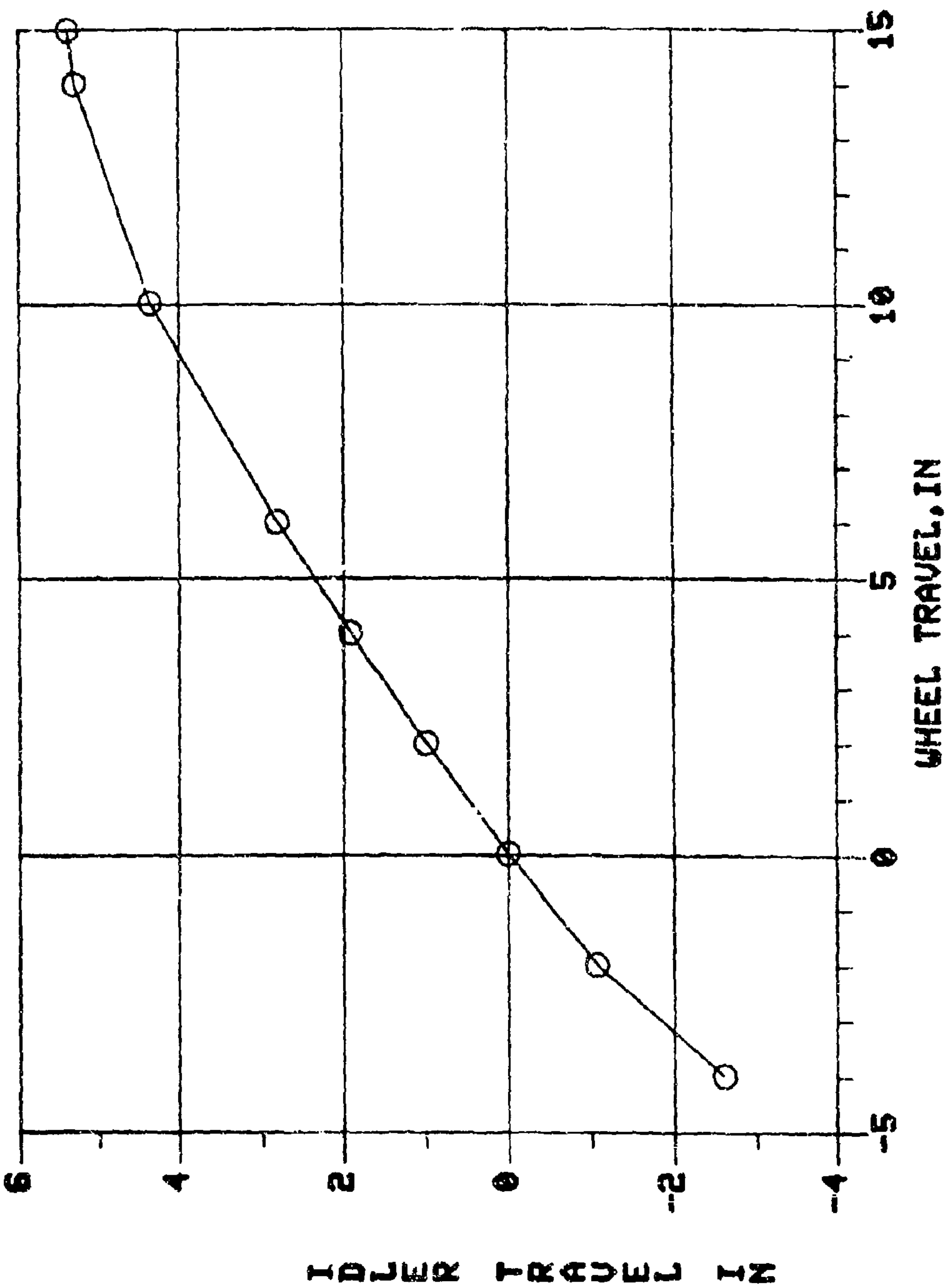

Figure 3.7-1 Proposed Compensating Idler Movement

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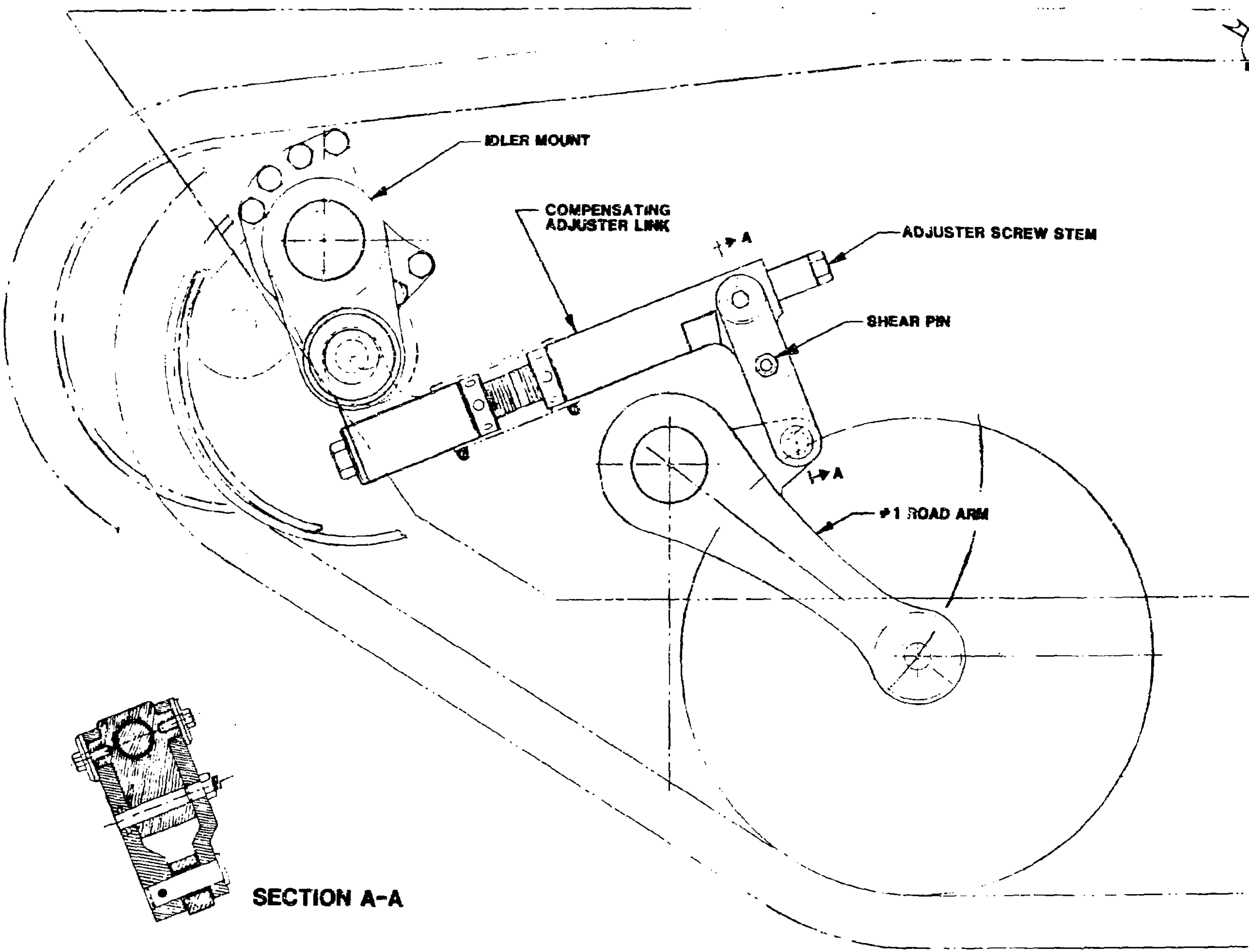

Figure 3.7-2 Compensating/Adjuster Linkage

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To perform adjustment of track tension, the turnbuckle screw is rotated from the end to obtain the desired track tension. Once adjusted, the two locking nuts are seated with a specially designed spanner wrench. The turnbuckle design provides 5.79 inches of adjustment along the pitch line of the track. If the track adjuster is fully extended and does not provide the proper track tension, the adjuster must be fully retracted to permit the removal of one track pitch.

The idler mount employed in the idler arrangement is an M113A2 type with the idler arm length increased from 4.35 to 5.35 inches. When the number one roadwheel is moved vertically the idler wheel is forced to rotate forward about the idler arm spindle. This provides track slack compensation during landborne operations. Figure 3.7-3 shows the horizontal idler movement as a function of roadwheel travel. As shown, idler movement is initially less amplified than that resulting from the initial proposed design. This design is less likely to overload the track, due to over tensioning, than the originally proposed design.

During the track retraction mode the idler wheel movement has been designed to pull the track nearly flush with the ATR hull bottom. Analyses were performed to examine the idler movement and the resulting change in overall track length. This was done to determine the resulting track tension during waterborne operations and to estimate the amount of track sag that is expected below the hull bottom. A summary of this analysis is presented below. The details of this analysis are presented in AAI Corporation Engineering Report No. R60011-00009.

The pitch line track length was determined with the suspension in the static and the retracted modes and the track adjuster in three positions, 1) retracted, 2) centered, and 3) extended. Table 3.7-1 shows the results of this length analysis.

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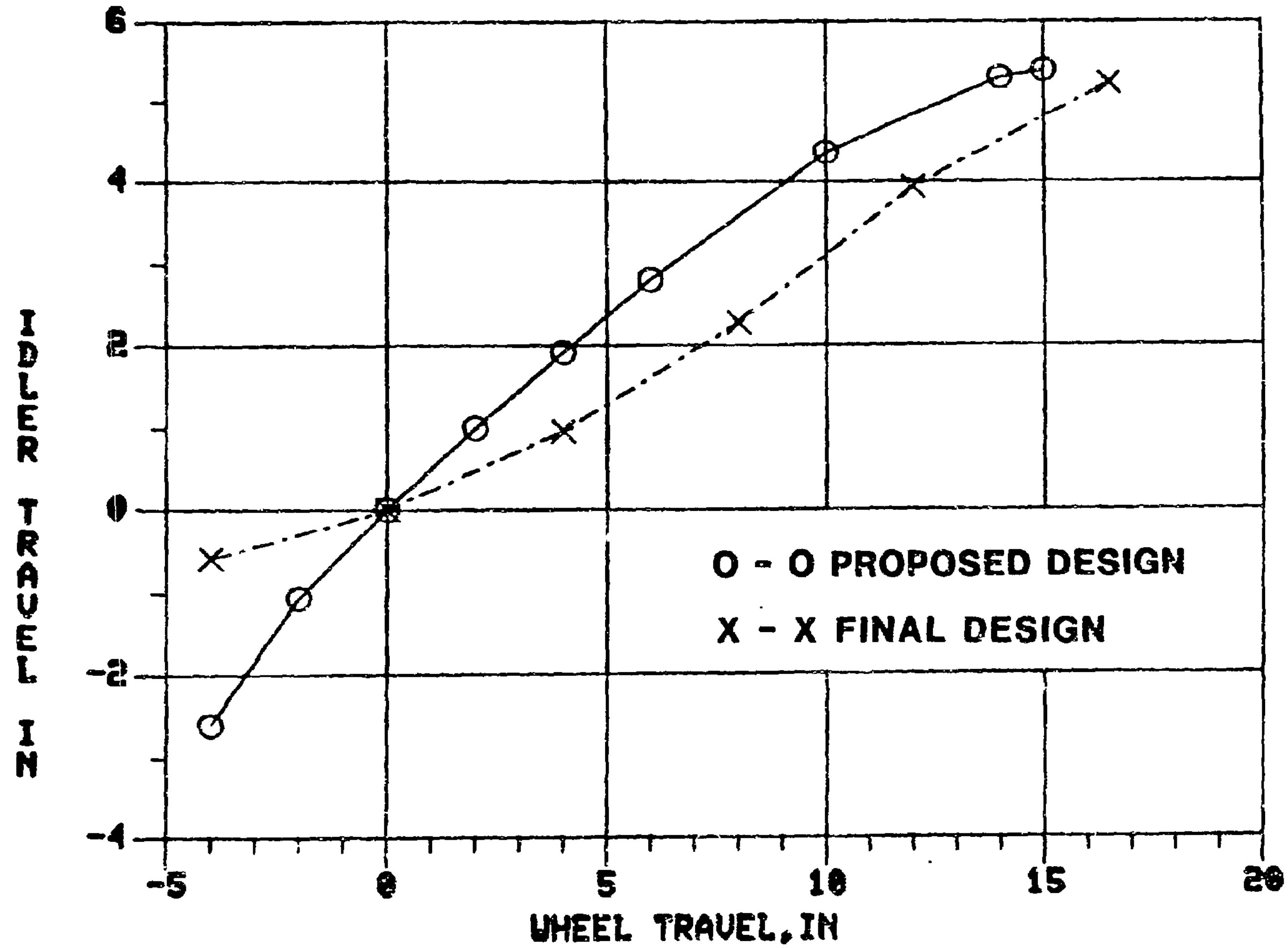

Figure 3.7-3 Compensating Idler Movement

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Table 3.7-1. Track Length

Track Adjuster Position	Adjuster Length, inches	Track Length, inches		Track Stretch, inches
		Static Position	Retracted Position	
Retracted	19.38	397.47	397.86	.39
Centered	21.0	400.42	400.95	.53
Extended	22.5	403.21	403.58	.37

The amount of track stretch shown will increase the tension in the track according to the spring rate of the track. The spring rate of the wire link track, which is installed on the ATR, has been determined by using laboratory pull test data which is presented in AAI Corporation Engineering Report No. ER-13302. The track spring rate (K) determined from this data is: K = 313840 lbs/in/5.81 inch section.

The increase in track tension can be determined from the amount of stretch using the following formula:

$$T_I = K \times \frac{LP}{L_{TS}} (L_S)$$

where:
T_I = tension increase, lbs
K = track spring rate per section, lbs/in/section
L_P = length per pitch, inches
L_{TS} = length track in static position, inches
L_S = length of stretch, inches

The sag at the track can be determined by assuming that the track assumes a parabolic shape with the weight uniformly distributed along the horizontal. This is a valid assumption for this case since the track is expected to be quite taught. The formula to determine the sag of the track is:

$$SAG = \frac{w \quad x^2}{2 (T_s + T_I)}$$

where:
w = unit weight 2.75 lbs/inch
x = half the span 83 inches
T_s = track static tension, 2000 lbs
T_I = tension increase, lbs

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Applying the formulas results in the predicted track sag shown in in Table 3.7-2.

Table 3.7-2. Predicted Track Sag

Track Adjuster Position	T_I, LBS	Sag, inches
Retracted	1789	2.56
Centered	2400	2.20
Extended	1673	2.64

Further analysis of the retracted suspension arrangement was required to determine the capability of the Bird-Johnson hydropneumatic suspension (HPS) unit to supply the required torque for retraction. The details of this analysis are published in AAI Engineering Report No. R60011-00001. The results of this analysis are shown in Table 3.7-3. The maximum roadarm trunnion reaction is 54477 in-lbs and occurs with the track adjuster in the center position. The HPS must be capable of delivering this torque in order to fully retract the track. A supply system pressure of 2342 psi will be required to generate this torque. This is well within the capabilities of the HPS unit and the ATR 3000 psi auxiliary hydraulic system.

Table 3.7-3. Track Adjuster Load Summary

Adjuster Position	Suspension Mode	Adjuster Link Force, lbs	Idler Arm Radial Force, lbs	Road Arm Trunnion Reaction in-lbs
Collapsed	Static	3760	108	
Collapsed	Retracted	7644	148	45057
Center	Static	3671	371	
Center	Retracted	9292	631	54477
Extended	Static	3782	404	
Extended	Retracted	617	6742	3698

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Since the idler movement is designed to provide full track retraction for the waterborne mode, the amount of track compensation during land operation will be excessive. When encountering a large obstacle at high speed, the roadwheel is stroked upward as the wheel rolls over it. The linkage connecting the roadarm to the idler arm will force the idler wheel forward. Because the linkage is designed to provide full retraction the track tension will increase when large obstacles are being crossed.

An analysis was conducted to examine the magnitude of the track tension increase as the ATR passes over two obstacle sizes (16 and 12 inches in height) with two suspension configurations. The configurations considered are shown in Figure 3.7-4. The first configuration is with the first roadwheel on the obstacle with the second roadwheel remaining on the ground. The other configuration is with the first roadwheel on the obstacle causing the second roadwheel to raise and the three trailing roadwheels remaining on the ground. This analysis assumes the vehicle remains level during the obstacle negotiation. For each of the configurations the pitch line track length was calculated with the track adjuster setting in the centered position. Table 3.7-4 presents the required amount of track stretch together with the resulting track tension.

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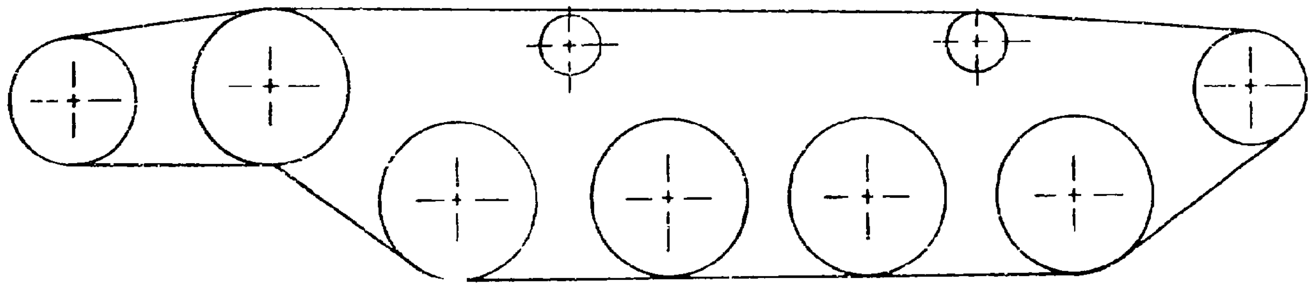
CONFIGURATION 1 - RW #1 ON OBSTACLE, ALL OTHERS ON GROUND



CONFIGURATION 2 - RW #1 ON OBSTACLE, RW #2 RAISED, ALL OTHERS ON GROUND

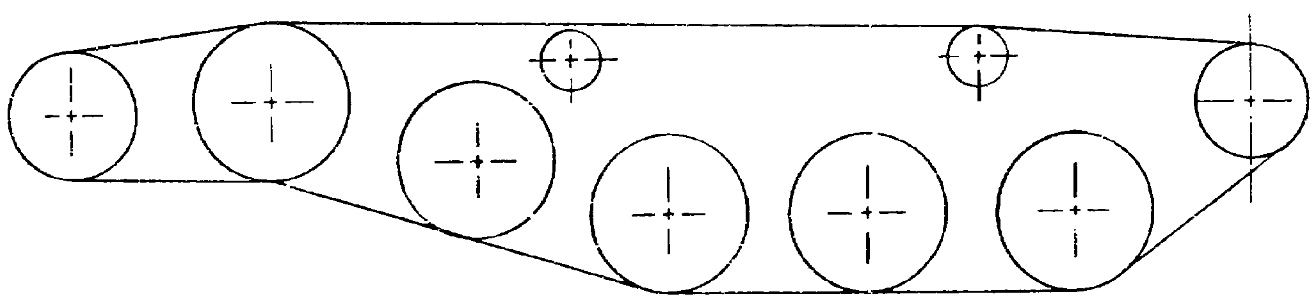


Figure 3.7-4 Loading Analysis Suspension Configurations

Table 3.7-4. Track Tension Under Severe Conditions

Configuration	Static Track Length Inches	Obstacle Crossing Track Length Inches	Track Stretch Inches	Tension Pounds
#1 RW @ 16 in., #2 Static	400.42	411.83	11.41	107300
#1 RW @ 16 in., #3 Static	400.42	409.28	8.86	83900
#1 RW @ 12 in., #2 Static	400.42	405.17	4.75	43898
#1 RW @ 12 in., #3 Static	400.42	403.54	3.12	30834

The load developed due to overcompensation can potentially be quite high. The operational scenario which could cause such overcompensation is considered to be quite severe. The frequency of occurrence will be rare even during cross-country mobility testing. However, to safeguard against track overload, a shear pin has been incorporated into the compensating linkage. The shear pin will be a fusible link to protect the track and all other effected suspension components from excessive loading.

To determine the maximum allowable shear pin loadings that are required to protect the suspension system components, a loading analysis was conducted. This analysis, presented in detail in AAI Corporation Engineering Report No. R60011-00009, considers the effects of track tension on the major components of the Wire Link Band Track, the Two Speed Final Drive the M113A2 Idler Mount, and the Bird-Johnson Hydropneumatic Suspension Unit. The

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allowable shear pin loadings determined by each component is given in Table 3.7-5. The loads presented carry a 1.2 safety factor on yield or a 1.5 safety factor on ultimate capacity. The table shows that the hydropneumatic suspension unit is the weakest of the suspension components considered and will establish the shear pin setting at 33,300 pounds.

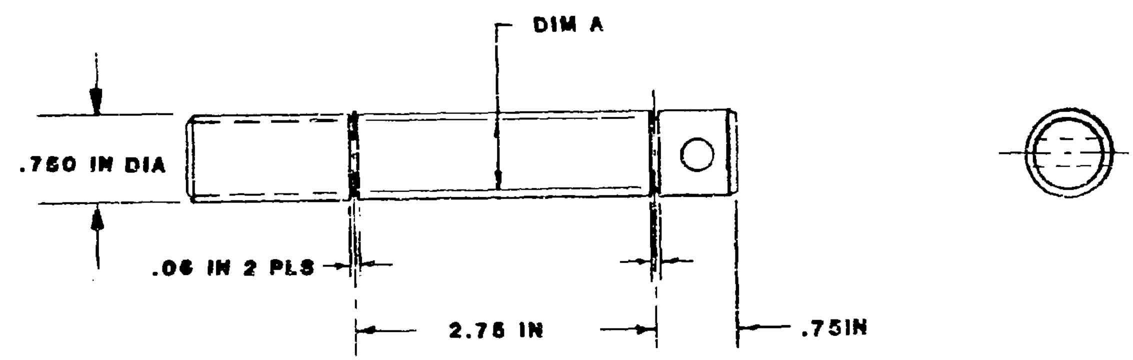
The shear pin, shown in Figure 3.7-5 is fabricated from 4140 steel heat treated to a Brinel Hardness of 495. As shown in Figure 3.7-5 two grooves are cut in the pin to facilitate "clean" double shear failure. Several groove depths are designed and sample destructive testing is planned for validation prior to installation on the ATR.

Table 3.7-.5 Allowable Shear Pin Loadings

Component	Track Tension Limit, lbs	Shear Pin Loading Limit, lbs
Wire Link Band Track	30,000	60,000
Two Speed Final Drive	23,100	46,200
Idler Mount	26,500	53,000
Hydropneumatic Suspension Unit	16,650	33,300

The revised compensating idler linkage design was presented to the Marine Corps Programs Office personnel at a preliminary design review at AAI on 1 May 1984. As a result of the meeting, the Marine Corps representatives requested that alternate idler mount designs be considered. The primary purpose for this investigation was to prevent track tension overload while maintaining all the capabilities of the existing design. In response to this request, AAI conducted a trade-off study to identify potential alternate design concepts. The trade-off study results and recommendations were presented to the Marine Corps Programs Office personnel on 22 May 1984, and are repeated in the following paragraphs.

To begin the trade-off study, required and desired idler mount capabilities were determined. Table 3.7-6 shows the two required and six desired capabilities that were identified. The capabilities of track tension



LINK CAPACITY (LBS)	DIMENSION A
30000	.462
45000	.566
60000	.598
75000	.750

Figure 3.7-5 Shear Pin

Table 3.7-6 ATR Idler Mount Capabilities

REQUIRED:

- o Track Tension Adjustment
- o Track Retraction in Waterborne Mode

DESIRED:

- o Landborne Track Slack Compensation
- o Energy Absorption Capability
- o Simplicity
- o Low Cost
- o Light Weight
- o Low Risk

adjustment and track retraction in the waterborne mode were determined to be requirements of an alternate design.

As part of this trade-off study a modification to the existing design was generated. The modification, which is shown in Figure 3.7-6, incorporates an "outrigger" bearing support to the existing design. The purpose of this support is to protect the Bird-Johnson HPS unit output shaft. This would permit the allowable shear pin setting to be increaed to 46,200 pounds from 33,300 pounds.

Four alternate idler concepts were evaluated relative to the existing compensating/adjuster linkage design. The alternate designs that were considered were:

1. Compensating/adjuster link w/energy absorption
2. Hull mounted hydraulic cylinder (inactive)
3. Hull mounted hydraulic cylinder (active)
4. Bird-Johnson Co. HPS unit

The first alternate concept, a compensating/adjuster link with energy absorption, would be configured much the same as the existing link with the addition of an elastic energy absorbing feature on the link body. The mounting points of this device would be identical to the existing design and would therefore have the same motion characteristics during normal operation. The advantage of this concept is that the energy absorbing device would be self contained. The disadvantages of the concept are: added weight, saltwater sensitivity, size increase and possible heat build-up problems.

The second alternate concept, a hull mounted hydraulic cylinder (inactive), is depicted in Figures 3.7-7 and 3.7-8. The cylinders are internally mounted in the vehicle hull to protect the cylinder rod from saltwater and to reduce the risk of cylinder rod bending. As can be seen from the figures the cylinder could be capable of actuating a sector gear or a lever. To provide water to land transition, a means to sense the position of the cylinder is required. The cylinder rod will remain inactive (or stationary) and will not provide track compensation during landborne operations. The advantage of this concept is the use of simple hydraulics. The disadvantages are increased weight, no compensation during landborne operations, new mounting required with through spindle, and an increase in system complexity.

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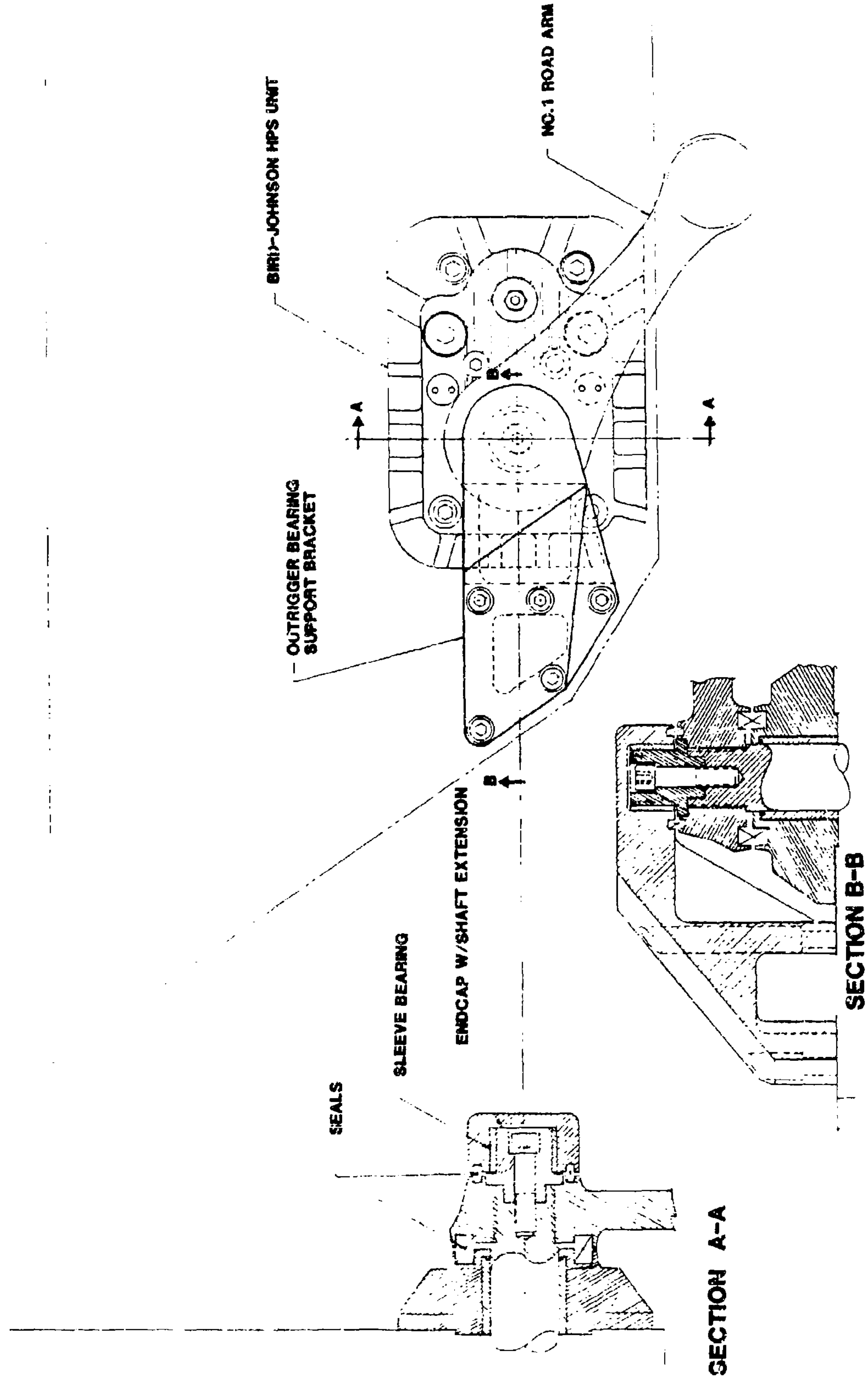

Figure 3.7-6 Outrigger Bearing Support for HPS Unit

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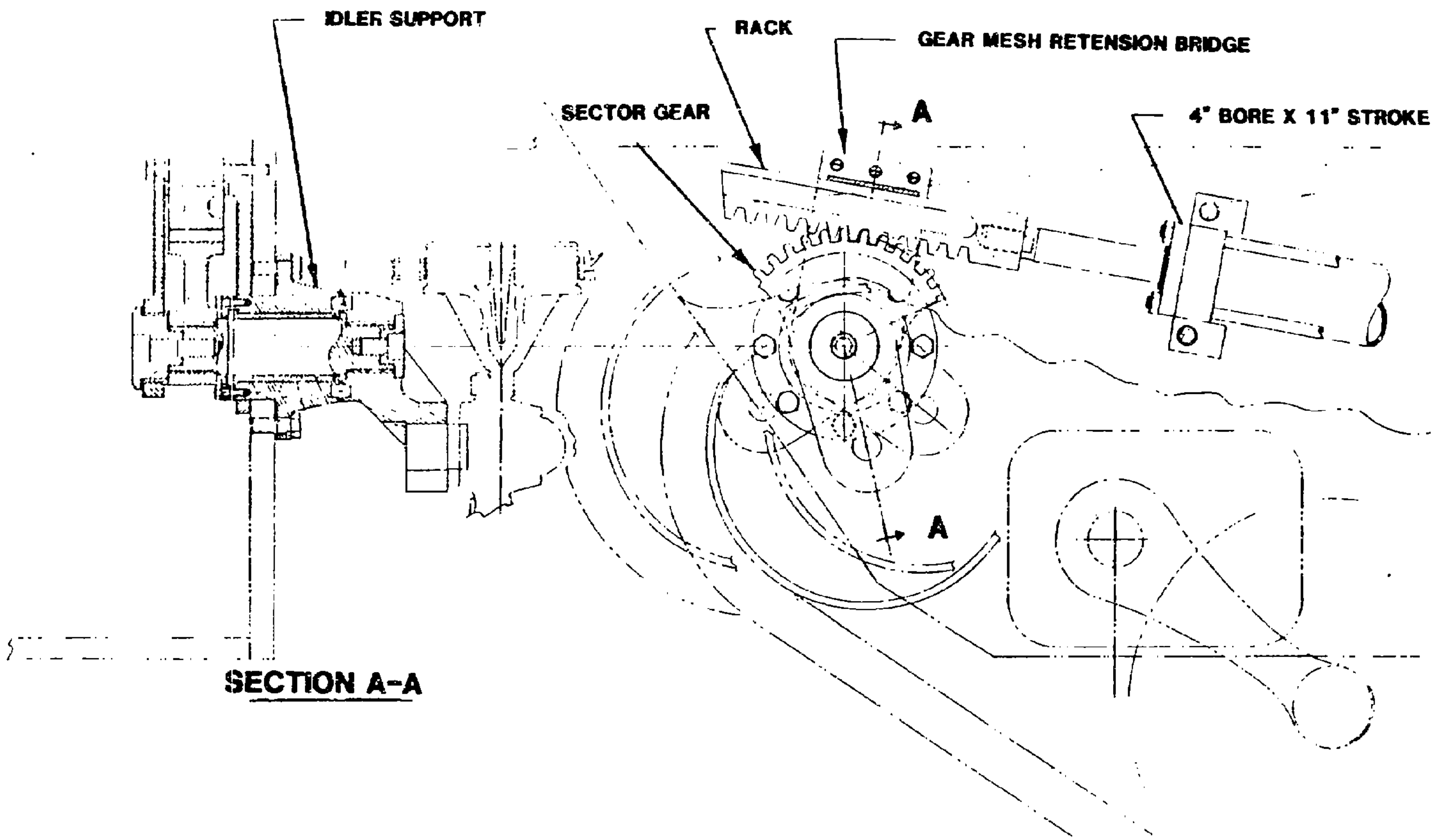

Figure 3.7-7 Hull Mounted Hydraulic Cylinder: Sector Gear

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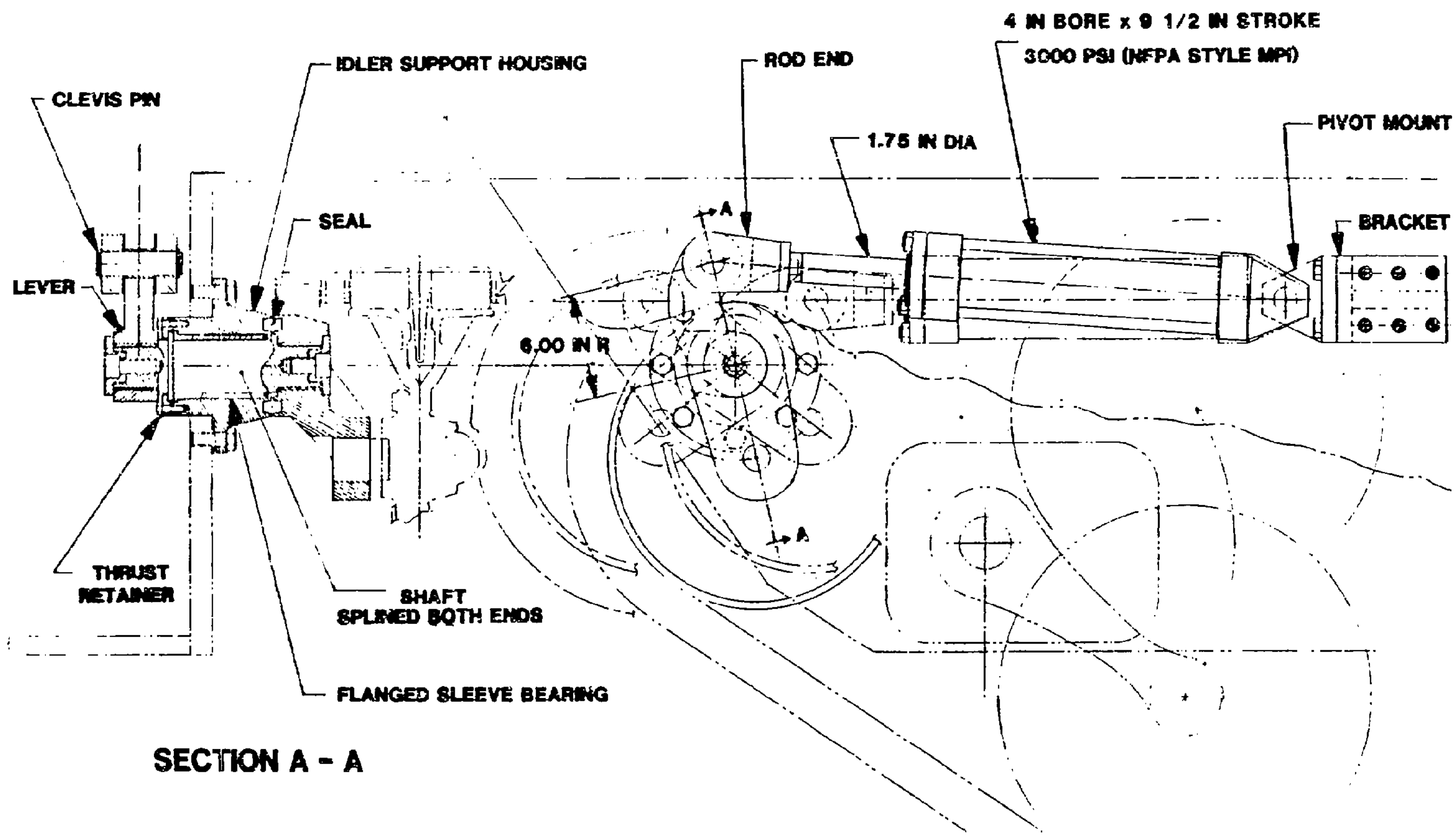

Figure 3.7-8 Hull Mounted Hydraulic Cylinder: LEVER

The third alternate concept, hull mounted hydraulic cylinder (active) could use the same arrangement as that of the inactive cylinder as shown in Figures 3.7-7 and 3.7-8. The active cylinder would be used to provide track compensation during landborne operations. To provide an active capability position and load sensing hydraulic valving would be required. The advantages of this concept are full control of idler position and energy absorption capability. The disadvantages are high development risk, high cost, a large weight increase and a complex hydraulic system.

In the fourth alternate concept, the HPS unit is employed as an idler mount as shown in Figure 3.7-9. This concept utilizes a GFE Bird-Johnson HPS unit, mounted to an idler arm. To provide the full range of required capabilities the HPS units must be modified to permit track tension adjustment. The advantages of this concept are: low cost, light weight, and self contained damping and valving. The disadvantages are: high risk (developmental item), unknown heat build-up characteristics, and limited track compensation during landborne operations.

For evaluation purposes, each of the concepts were compared by using a rank ordering technique for each of the selected evaluation criteria. No weighting factors were applied to any of the evaluation criteria. Rank varied from 1 to 5 which corresponded from poor to good performance. The numerical rankings assigned to each concept appears in Table 3.7-7. All concepts received essentially the same ranking for the two required capabilities. When comparing the ranking total, the original compensating/adjuster link received the highest total rank. The primary reasons the original design is the highest ranked are is its advantages in the areas of complexity, cost, weight and risk.

As a result of the idler trade-off study the following recommendations were made:

1. Use the compen ting/adjuster linkage with a fusible shear pin.
2. If required, a compensating/adjuster linkage with energy absorbing capabilities can be retrofitted.

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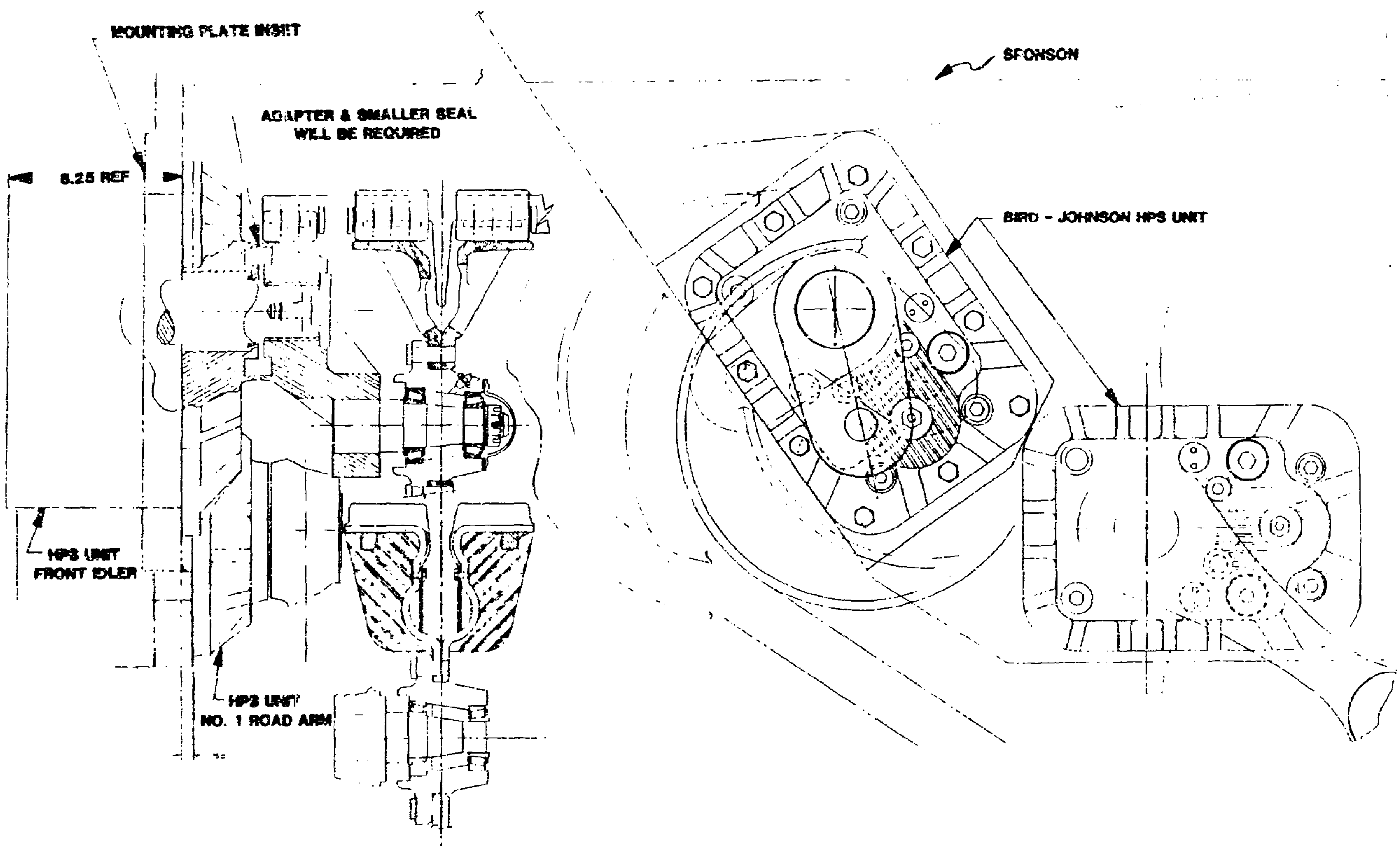

Figure 3.7-9 Bird-Johnson HPS Unit Idler Mount

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Table 3.7-7 Ranking of ATR Idler Mount Alternatives

Concept	Evaluation Criteria, Rank 1-5 (Poor-Good)								
	Track Tension	Track Retrac.	Track Compen.	Energy Absorb.	Complexity	Cost	Weight	Risk	TOTAL
Compensating/ Adjuster Link	4	4	3	1	5	5	4	4	30
Comp./Adj. w/ Absorbtion	4	4	3	4	3	3	3	3	27
Hull MTD Hyd. Cylinder, (in-active	4	4	1	2	3	3	2	3	22
Hull MTD hyd. Cylinder, (active)	4	4	4	4	1	1	1	2	21
Bird-Johnson HPS Idler Mt.	3	4	2	3	2	4	3	2	23

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3.8 Track

The ATR employs a lightweight band track rather than a conventional block type. The track was specified as Government Furnished Equipment (GFE) and is depicted in Figure 3.8-1. This track is being developed for the Marine Corps Programs Office by the AAI Corporation.

Prior to installation on the ATR, the track was subjected to acceptance testing on an M113A2 armored personnel carrier. The vehicle was ballasted to 14-tons for the testing which was conducted in October 1985.

The characteristics of the track are as follows:

Tread:	Rubber
Roadwheel path:	Rubber
Weight:	15.9 lbs/pitch, 32.84 lbs/feet
Pitch:	5.81 in
Width:	17.0 in
Grouser height:	.81 in
Guide:	Integral with crossmember
Pitches required for ATR:	138

The lightweight track is basically a band type track which uses wire mesh rather than steel cables. The track is constructed of two basic members: a molded link and a steel crossmember.

Within each molded link is an assembly of wire mesh, connecting pins and spacers linked to end connectors on either end. All metal components are stainless steel with the exception of the spacers within the wire mesh. The end connectors are joined end to end to form a continuous rubber band composed of individual rubber covered links.

Four bands, two deep treads and two shallow treads, are used to make a complete track pitch. The steel crossmembers bolt to the bands and provide sprocket-engaging surfaces as well as a centerguide and grouser.

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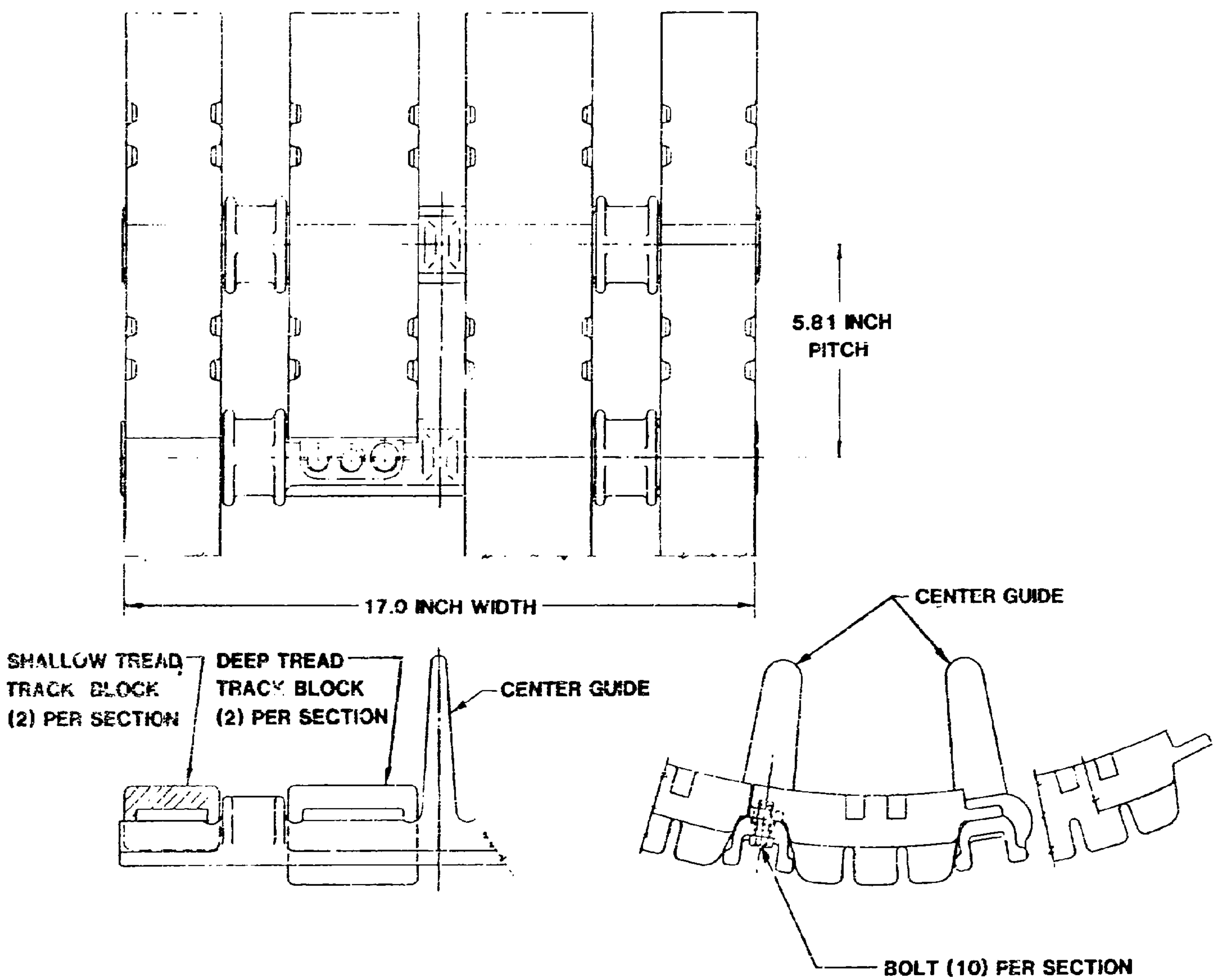

Figure 3.8-1 AAI Wire Link Track

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3.9 Bump Stops

ATR suspension travel in jounce is limited by roadwheel/track contact with the ATR hull sponson bottom plate at suspension stations #1, #2, #3 and #5 as shown in Figure 3.1-1. The sponson bottom plates are locally reinforced with channel sections in the impact area of each roadwheel.

For suspension station #4, jounce travel is limited by a hull mounted bump stop. A separate bump stop is required at this station because the hull sponson area is removed to enhance water flow to the sponson mounted waterjet. At full jounce the hull mounted bump stop assembly impacts with the rear extension of the number four roadwheel spindle.

The bump stop, shown in Figure 3.9-1, is a steel weldment which is bolted to the ATR hull side plate. The stop is designed to contact a special roadwheel spindle which is equipped with a 2 inch inboard extension. This design minimizes the load transmitted to the roadarm and HPS unit by absorbing the roadwheel impact load as close to the roadwheel spindle as possible.

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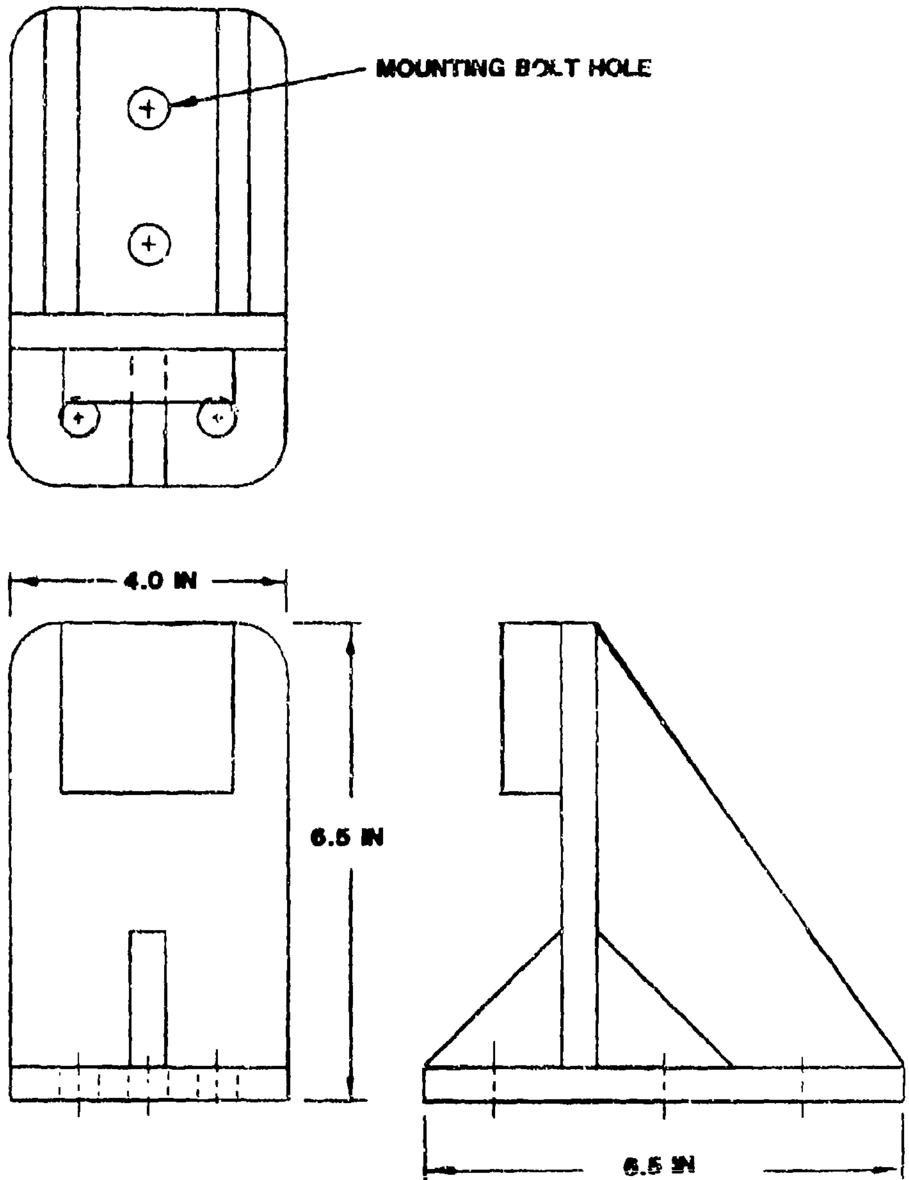

Figure 3.9-1 Bump Stop Weldment

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3.10 Suspension Controls

The ATR suspension system is capable of maintaining two operating positions; extend and retract. The extend position is used for land and transition operations. The retracted position is used for sea operations. For the control of the suspension system, the ATR is equipped with the hydraulic control and distribution system shown schematically in Figure 3.10-1. The system is fully integrated into the ATR auxiliary hydraulic system.

The control and distribution system consists of ten three position directional control valves (DV14-DV23), one pressure regulator (PR1), one pressure switch (PS11) and supply return lines. In the de-energized position, the directional control valves block the supply pressure and open the ports on the HPS units to the reservoir. In this position suspension actuation is not provided. In the solenoid A position the suspension system is extended. In the solenoid B position the suspension system is retracted. The regulator is used to limit the supply pressure to 2,000 psi to minimize HPS unit internal leakage. The electrical pressure switch is used to indicate a low suspension supply pressure. The switch signal is entered into the SC-1 computer causing a master warning fault.

During normal vehicle operation the SC-1 computer controls the suspension system position according to the drive mode selected (i.e., land, sea, or transition) by the operator. For additional control the operator is provided with a separate suspension position switch. This switch is only active during the land mode of operation. In the land mode, the operator can select suspension extend, suspension off, or suspension retract. Should the operator select the suspension off position from the suspension extend position the vehicle will begin to settle. Complete settling occurs in 2 to 4 minutes. In the sea and transition modes of vehicle operation the computer is in complete control of the suspension regardless of the suspension position selected by the operator.

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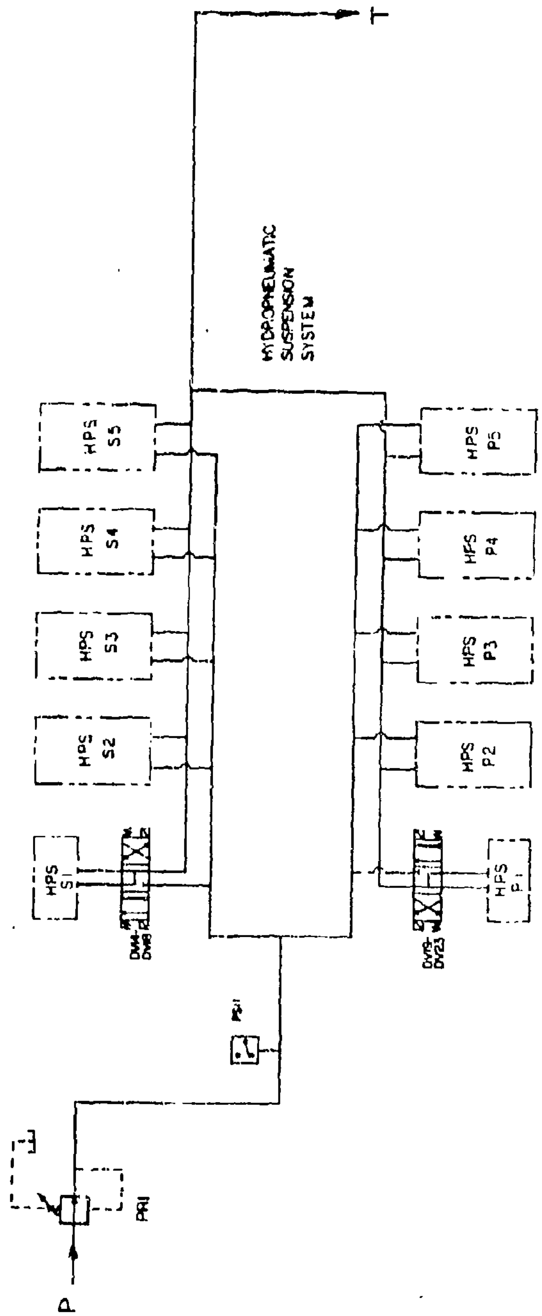

Figure 3.10-1 Suspension System Hydraulic Control Schematic

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3.11 Suspension Component Stress Analyses

Stress analyses has been performed on the roadarm assembly, the compensating adjuster linkage, and the hull mounted bump stop. The detailed stress analyses are contained in AAI Report No. R60011-00008 and a summary is presented below.

Roadarm Assembly

The roadarm and spindle have been evaluated for two loading conditions:

1. Vertical load applied perpendicular to arm;
2. Lateral load applied at wheel rim.

The magnitude of the vertical load for roadarm assemblies at stations 1, 2, 3 and 5 are limited by the motion resistance, damping and spring, capabilities of the Bird-Johnson HPS unit. Higher wheel impact loads will be transmitted through the roadwheel to the sponson bump stop. For the roadarm at station 4 the inboard end of the spindle will impact the hull mounted bump stop which allows the maximum roadwheel impact force to be transmitted through the spindle. The magnitude of the radial load is then:

Road Wheel Station	Vertical Load, Lbs
1,2,3,5	13600
4	51000

The maximum lateral load is the same for all roadarm stations The magnitude of this load is specified in AAI Corporation Report No. R600011-00002 "ATR Design Criteria", to be 14000 lbs.

Figure 3.11-1 shows a loading diagram of the roadarm assembly under a vertical loading condition. The worst case is considered to be with the applied force acting perpendicular to the roadarm. The maximum stresses have been determined to occur at Section A-A of the figure for all wheel stations. The maximum stresses have been determined to occur at Section B-B for the roadarms at Station 1,2,3 & 5 and at section C-C for roadarms at Station 4. The magnitudes of these stresses are presented in Table 3.11-1. As can be seen, low stresses are experienced at all roadarm assemblies except those at station 4 due to the bump stop. The negative margin of safety

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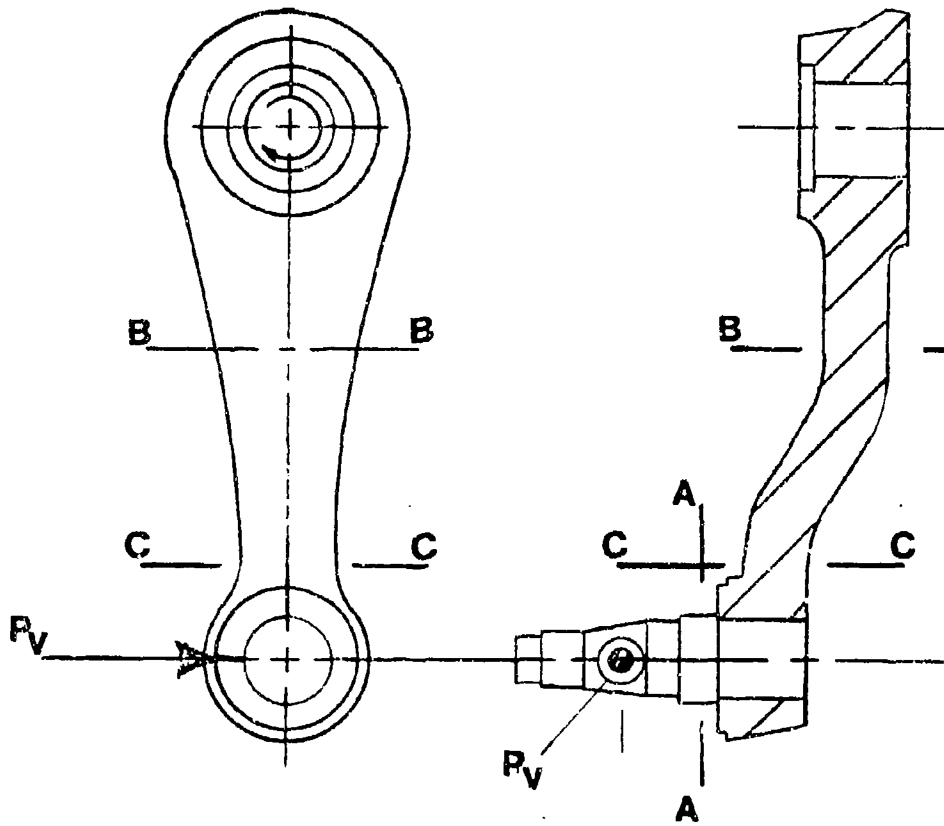

Figure 3.11-1 Vertical Loading Of Roadarm Assembly

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Table 3 11-1 Roadarm Assembly Stress Due To Vertical Loading

Roadwheel Station	Spindle			Roadarm		
	Section	Max Stress (Ksi)	Margin* of Safety	Section	Max Stress (Ksi)	Margin of Safety
1, 2, 3 & 5	A-A	33.8	3.23	B-B	53.7	+ 1.67
4	A-A	125.7	.13	C-C	174.4	-.18

$* \text{ M.S.} = \frac{F_{tu}}{f} - 1$, F_{tu} (Spindle) = 143 Ksi, F_{tu} (Roadarm) = 143 Ksi

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experienced by the roadarm at roadwheel station No. 4 at section C-C is considered permissible. It is permissible because the loading condition considered is not likely to be experienced by this roadwheel station.

Figure 3.11-2 shows the loading diagram of the roadarm assembly under lateral loading conditions. Two lateral cases are considered:

1. Lateral force exerted at roadwheel rim along the radial axis of the roadarm;
2. Lateral force exerted at road wheel rim 90 degrees off the radial axis of the roadarm.

The maximum stresses have been determined to occur at section A-A in the spindle. In the roadarm the maximum stress will occur at section B-B for the on axis loading case and at section C-C for the off axis loading case. The magnitude of these stresses are presented in Table 3.11-2. As can be seen the lateral loading case greatly stresses the assembly. The negative margin of safety is small in all cases and is acceptable considering the severity of the loading condition.

Compensating Linkage

The compensating linkage has been sized to accept a shear pin with a 50,000 pound capacity. Due to the limitations of the HPS unit, the shear pin will be limited to a capacity of 33,300 pounds. The stress exerted on each of the compensating linkage components is given in Table 3.11-3 with a shear pin capacity of 33,300 pounds.

Hull Mounted Bump Stop

The hull mounted bump stop is designed to accept a load from the roadarm spindle of 39800 pounds. This is the maximum vertical wheel loading for roadwheel number four of 51000 pounds minus the HPS unit resistance of [4 (2800) = 11200 pounds.] Figure 3.11-2 shows the bump stop loading diagram. The bump stop is fabricated of T-1 structural steel which has a weld strength of 100 ksi. The stress levels experienced by the bump stop are shown in Table 3.11-4.

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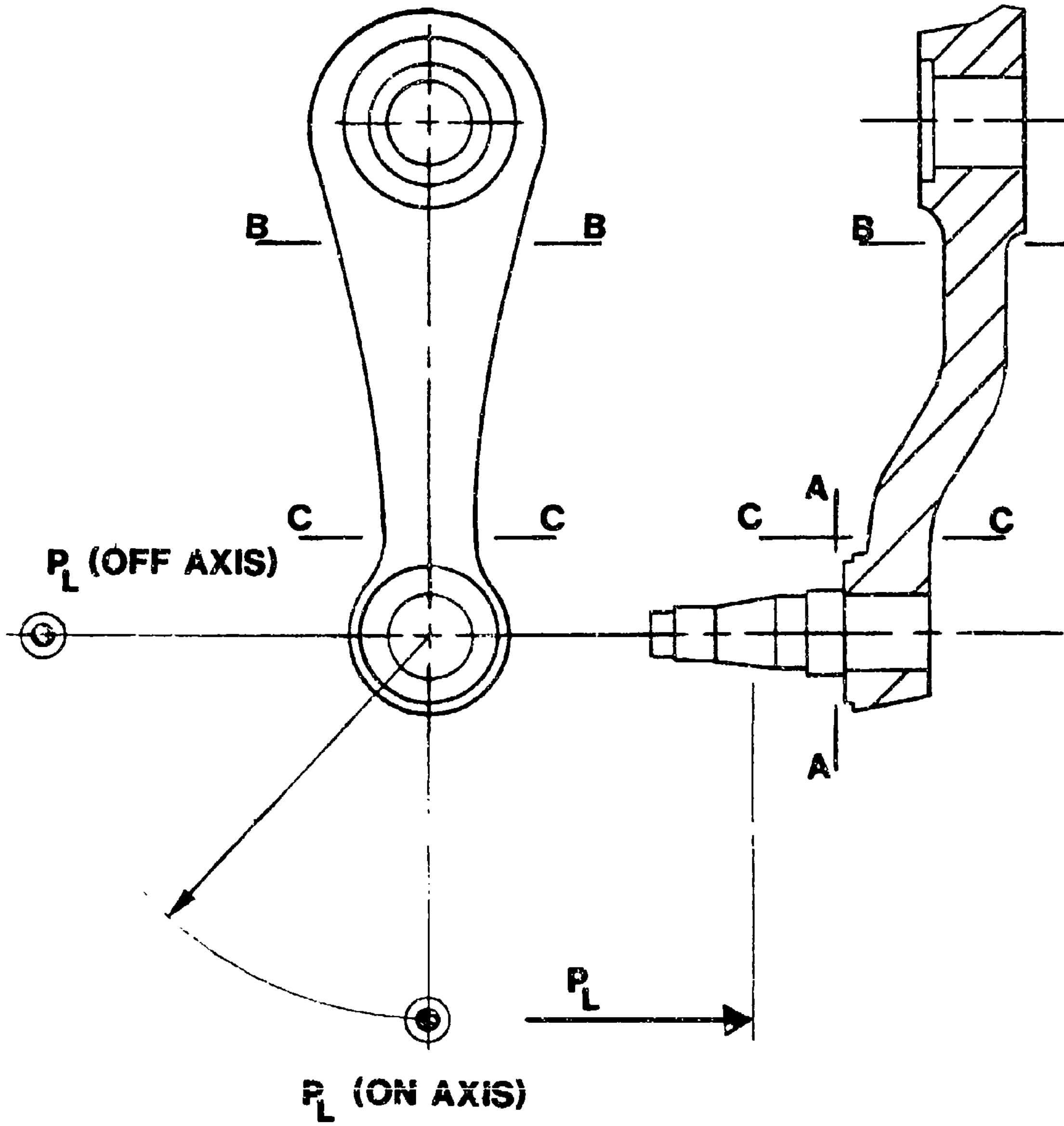

Figure 3.11-2 Lateral Loading of Roadarm Assembly

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Table 3.11-2 Roadarm Assembly Stress Due To Lateral Loading

Loading Case	Spindle			Roadarm		
	Section	Max Stress (Ksi)	Margin* of Safety	Section	Max Stress (Ksi)	Margin of Safety
Along Roadarm Axis	A-A	149.5	-.043	B-B	152	-.059
90° Off Roadarm Axis	A-A	149.5	-.043	C-C	149.5	-.043

* $M.S. = \frac{F_{tu}}{f} - 1$, F_{tu} (Spindle) = 143 Ksi, F_{tu} (Roadarm) = 143 Ksi

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Table 3.11-3 Compensating Linkage Stress Levels

Component	Material	Max Stress Ksi	Margin of* Safety
Adjusting Screw	4340	83.9	+0.5
Idler End Brg.	AMPCO 18	24.8	+2.5
Roadarm End Brg.	AMPCO 18	37.6	+1.3
Clevis Pin, Roadarm	4340	18.0	+5.9
Side Bar	4340	97.2	+0.3
Link, Roadarm End	4340	85.9	+0.5
Link, Idler End	4340	37.0	+2.4
Shear Pin	4140	149,100	0.0

$$* \text{ M.S.} = \frac{F_{tu}}{f \times 1.2} - 1$$

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Table 3.11-4 Hull-Mounted Bump Stop Stress Levels

Component	Max Stress Ksi	M.S.*
Weld	35.5	+1.8
Bolts	101.5	+.47
Hull	25.6	+.45

$$*M.S. = \frac{F_{tu}}{f} - 1$$

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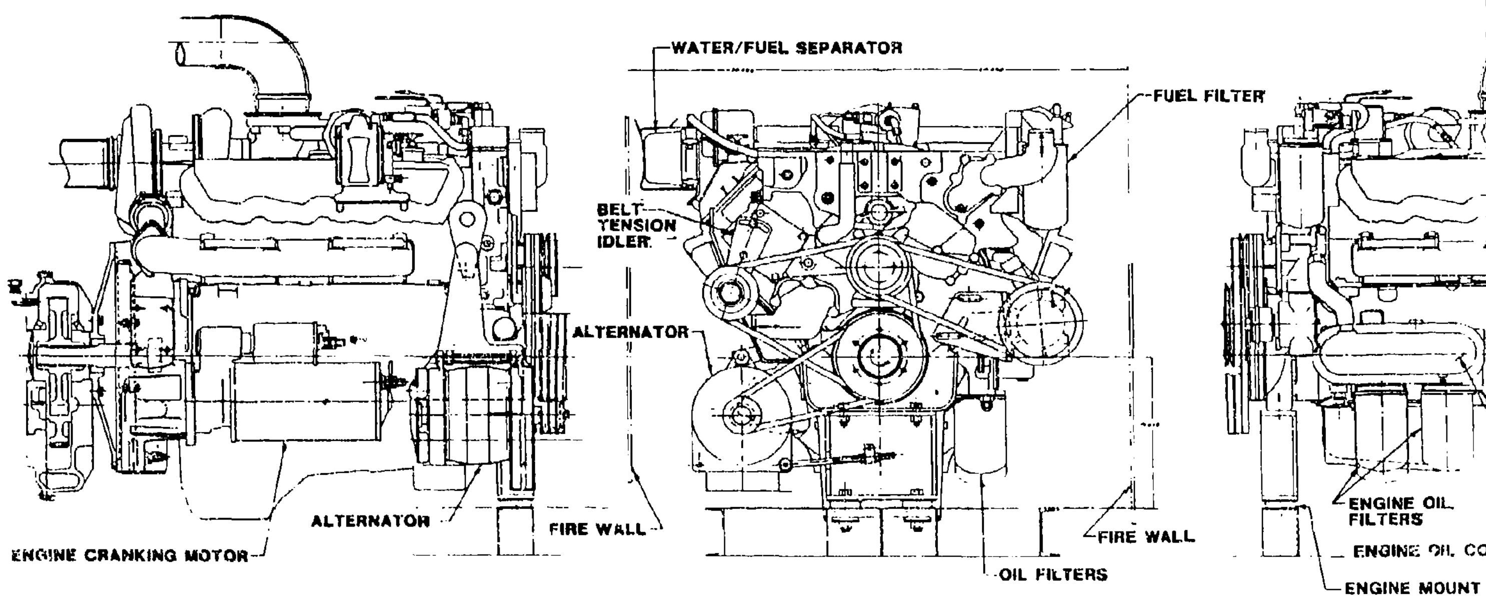
4.0 PROPULSION SYSTEM

4.1 General Arrangement

The propulsion system is positioned in the ATR bow and centered on the vehicle's longitudinal axis as shown in Figures 1.2-2 and 4.1-1. The engine is installed with the flywheel aft. The primary hydraulic pumps (hydrostatic drivetrain) and auxiliary systems hydraulic pumps are driven by a gearbox assembly which is directly attached to the engine flywheel housing. Service access to the front of the engine and the cooling system is available through a panel in the lower front glacis plate. The landborne cooling system heat exchangers are positioned horizontally above the engine. The heat exchangers can be rotated, as an assembly, to a near vertical position for service access to the top of the engine. The cooling air inlet/exhaust grille assembly and the cooling system heat exchanger assembly can be removed from the ATR to provide a large opening through which the engine and pump drive gearbox can be removed, if required. The cooling system fans are located immediately below the air exhaust grille and serve to ventilate the engine compartment and the troop compartment in addition to their primary role of drawing cooling air through the heat exchangers.

The marine cooling system heat exchangers are located in the ATR bow. Seawater is pumped through these heat exchangers and discharged overboard. The engine combustion air cleaner is located in the starbord side of the engine compartment and the engine combustion exhaust system is contained within the engine compartment near the cooling fans. Engine controls and instruments are located in the driver's station, and the vehicle battery box is located below the driver's station. The engine cranking motor and vehicle alternator are located on the lower right hand side of the engine. Two, fifty-gallon capacity fuel tanks are located above the waterjet units in the aft sponsons and the ATR fuel system is equipped with a water separator in addition to, and separate from the primary fuel filter. A mechanical fuel shut-off valve, intended for emergency engine shut-down, is controlled manually from the driver's station. Normal engine shut-down is accomplished via an electric fuel shut-off solenoid valve incorporated in the engine fuel governor/pump assembly. The engine, the combustion gas exhaust system, the vehicle cooling system, and the hydraulic pumps and drive gearbox are totally separated from the troop compartment by the insulated firewall. Removable panels provide service access to the engine compartment components.



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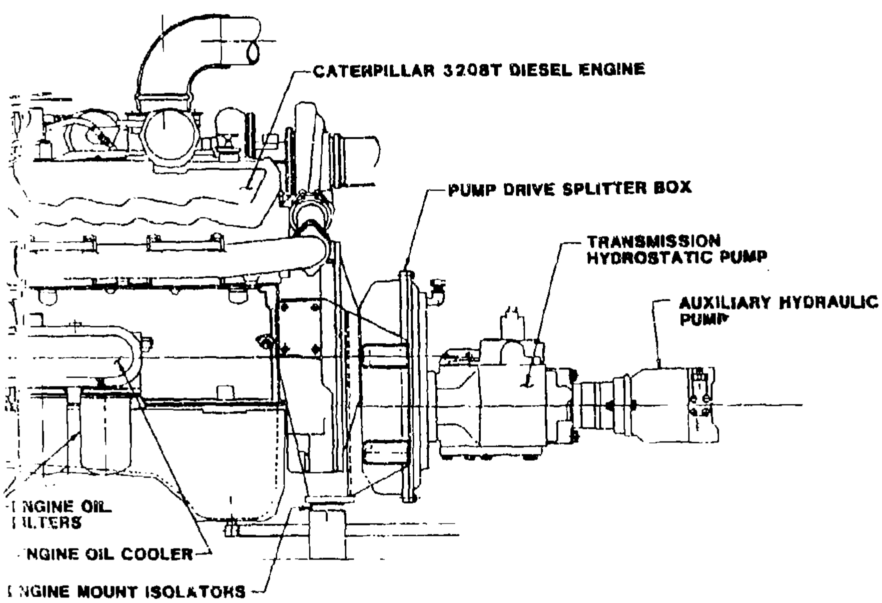

Figure 4.1-1 Propulsion System Arrangement

4.2 Engine

The ATR is powered by a Caterpillar 3208T engine as shown in Figure 1.2-2 and Figure 4.1-1. This four stroke cycle, compression ignition, turbocharged engine has a power rating of 320 BHP. The engine specifications are shown in Table 4.2-1, the engine power and torque curves are shown in Figure 4.2-1.

4.3 Induction/Exhaust System

The Caterpillar 3208T engine requires up to 735 CFM of air flow for combustion. Air enters the vehicle through a seawater shut-off valve. The air cleaner assembly is located in the engine compartment below the commander's station. Easy access to the air cleaner is provided from the rear via the troop compartment to facilitate filter element inspection and replacement. The air cleaner prevents dirt and foreign debris from entering the turbocharger and engine combustion chambers and is, therefore, of vital importance to engine life and performance. The ATR is equipped with a Donaldson Konepac™ air cleaner, Model EBA09-2001 weighing 18.5 pounds. This is a dry type air cleaner system that provides high filtration efficiency and minimum intake restriction. Air enters the end of the air cleaner housing and contaminents are removed by the filter element. The filter element is constructed of chemically stabilized filter media that is pleated and embossed for maximum surface area. A perforated metal shell protects the filter media inside and out, and gives the element strength and rigidity. The air cleaner outlet duct is fitted with a port for measuring engine inlet air restriction. A visual indication of excessive restriction is provided.

The engine combustion exhaust gases flow through a muffler provided by Donaldson. The exhaust gases then exit the vehicle through an exhaust pipe located behind the driver's hatch.

The exhaust system components are constructed of corrosion resistant steel and the entire interior exhaust system is wrapped with insulation to minimize radiant heating within the vehicle and to provide for safety protection. The exhaust system is enclosed in the engine compartment, and the engine compartment is maintained under negative air pressure by the cooling fans to prevent accumulation of dangerous fumes.

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Table 4.2-1 Caterpillar 3208T Engine Specification

GENERAL ENGINE DATA

Rated BHP (29.38 in Hg at 85°F)	320
Full Load rpm	2800
Bore and Stroke - inches	4.5 x 5.0
Displacement - cubic inches	636
Combustion system	Direct Injection
Aspiration type	Turbocharged
Compression ratio	16.5
Number of cylinders	8
Cylinder arrangement	90° "V"
Direction of rotation as viewed from flywheel end	Counterclockwise

GENERAL DIMENSION DRAWING 5N8329

ATTACHMENT INSTALLATION DRAWING . . . 9N1929

MOUNTING

Maximum allowable static bending moment at rear face of flywheel housing - pound inches	6700
Dry weight - Engine - pounds	1525
Power density - pounds/BHP	4.8
Length - inches	38
Width - inches	36
Height - inches	36

PERFORMANCE DATA

Low idle rpm	650
High idle rpm	3015
Altitude capability - feet	1000
Peak torque - pound foot	767
Peak torque rpm	1400
Torque rise - percent	28

AIR INTAKE SYSTEM

Combustion air flow - CFM	735
Maximum allowable system restriction:	
Clean dry element - inches water	15
Dirty element	25
Air inlet adapter diameter - inches	4

EXHAUST SYSTEM

Exhaust stack air flow - 2040 CFM @ 1045°F	
Maximum allowable back pressure - inches water	40
Exhaust pipe diameter - inches	4

FUEL AND GOVERNOR SYSTEM

Type	Sleeve
Maximum fuel flow to transfer pump - gph at rated conditions	25.2
Maximum fuel consumption - gph at rated condition	16.7
Fuel return line flow - gph	8.5
Maximum allowable restriction to pump - inches Hg	8.0
Maximum allowable return line restriction - inches Hg	8.0
Shutoff:	
Energized to run	Yes
Energized to shutoff	No

ELECTRICAL SYSTEM

Starting torque - lb-ft		537
Minimum recommended battery capacity:		
30°F and above 24V starter -	CCA @ 0°F	450
0°F to 30°F 24V starter -	CCA @ 0°F	730
Below 0°F 24V starter -	CCA @ 0°F	800

COOLING SYSTEM

Specific heat rejection - BTU/BHP minute:	
Full load	28
Peak torque	27
Top tank temperature:	
Maximum allowable - °F	210
Cooling capability for:	
Maximum ambient temperature - °F	110
Heat rejection - BTU/minute:	
Rated	8400
Peak torque plus 100 rpm	5535
Coolant flow-gpm at 5 foot water head	
Rated rpm	90
Coolant capacity - quarts	25
Minimum recommended pressure cap-psi	7
Coolant regulator:	
Start to open - °F	180
Fully open - °F	197
Top tank temperature:	
Minimum recommended - °F	160
Fill rate-gpm-average capable of interrupted fill	5
Pump cavitation temperature °F - minimum	199
Coolant low level sensitivity:	
Minimum percent of total system	9
Maximum percent of pump pressure rise loss	10
Air venting capability:	
Pints/minute	1.1
At pump pressure rise loss-percent	35

LUBE OIL SYSTEM

Capacity-refill-filter lines-quarts	20
Sump capacity-low-quarts	12
Sump capacity-high-quarts	16
Type oil - API class	SE/CC, CC, CD
Oil pressure-psi-SAE	10W30 @ 210°F
Normal	65
Low idle-minimum	14
Oil Temperature - Maximum	225°F

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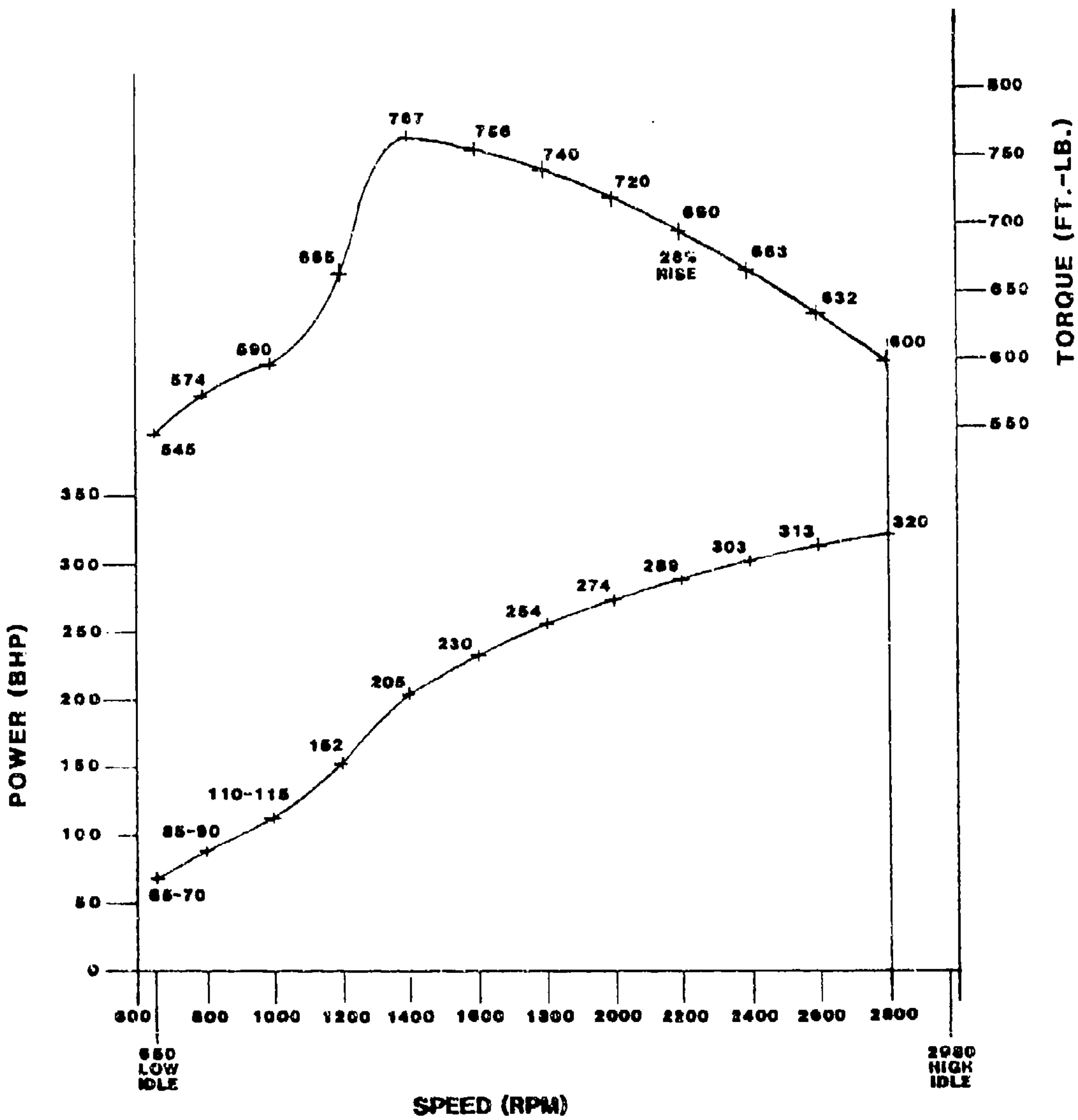

Figure 4.2-1 Caterpillar 3208T Engine Performance

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4.4 Automotive Cooling System

The ATR automotive cooling system must be capable of dissipating heat heat generated by the liquid cooled diesel engine and the oil cooled hydraulic system during both landborne and amphibious operations.

Conventional hot liquid to forced air heat exchangers (radiators) are incorporated into the ATR for landborne operation. Forced air cooling is not viable for amphibious operation, therefore, hot liquid to sea water heat exchangers must also be provided in the ATR.

Two approaches were considered for transferring heat between the hot fluids and the sea water. The first method employs a pump to circulate fresh sea water through shell and tube bundle heat exchangers, and then discharges the heated sea water overboard. The second method, a keel contact cooler, circulates the hot fluid along the inside surface of the hull bottom plate transferring the heat to the surrounding sea water.

Both marine cooling system approaches were evaluated for application in the ATR. The keel contact cooling system was rejected because of the excessive hull area required to provide sufficient heat rejection. Although not incorporated in the ATR vehicle, the keel contact cooling system design analysis is summarized in Section 4.5. The shell and tube bundle analysis is included in the overall vehicle cooling system analysis.

4.4.1 General Arrangement

The automotive cooling system is designed to maintain engine, hydrostatic transmission, final drive and auxiliary hydraulic equipment operating within the component manufacturer's specified temperature limits in an environment of 125°F ambient air and 70°F sea water temperatures.

The ATR includes two interrelated cooling systems, one for marine operations and the other for landborne operations. Engine coolant and hydraulic oil flow through both the landborne cooling system and the marine cooling system, in series, whenever the engine is running. There are no shut-off or diversion valves to be operated when transitioning to/from marine and landborne operations.

During amphibious operation, an electric pump circulates cooling sea water through shell and tube bundle heat exchangers located in the ATR bow. The entire sea water flow path is double enclosed to protect against uncontrolled sea water entry in the event of a component leak.

During landborne operation, two hydraulically driven mixed flow fans draw cooling air through the inlet grille, the oil cooler, and the radiator located above the engine. The air flow continues through the engine compartment, the fans, and exits through the exhaust grilles. The operation of the grilles and fans is controlled by the microprocessor. Negative air pressure is maintained in the engine compartment to prevent the accumulation of dangerous fumes. Additionally, the fans will exhaust ventilation air from the troop compartment. During amphibious operation, the starboard fan continues to operate at a reduced speed to maintain engine compartment ventilation. This ventilation air exits through a free-floating louver in the starboard exhaust grille.

To provide accessibility to the engine compartment and cooling system, the inlet and exhaust grill assembly can be rotated up and forward to an over-vertical position above the upper front glacis. Additionally, the radiator/oil cooler assembly can be rotated in a similar manner to permit engine accessibility. During this procedure, it will not be necessary to disconnect any cooling system hoses or to disturb the cooling fans. The engine will remain operational with the radiator/oil cooler in the raised service position. However, since the radiator is not in a forced air path, engine operation with the radiator open will be temperature limited. The cooling fans are not driven when the grille assembly is open.

The engine lubricating oil heat exchanger, an integral part of the Caterpillar engine assembly, is located on the port side of the engine above the oil filters. The final drive heat exchanger is located on the starboard side of the engine compartment above the number one suspension unit. Engine coolant is pumped through these heat exchangers to provide rapid oil warm-up and to maintain oil temperatures close to that of the engine coolant temperature. Table 4.4.1-1 presents a summary of the ATR cooling system performance.

4.4.2 Hydraulic System Cooling

The ATR hydrostatic transmission system predicted heat rejection is based upon several analyses. The first of which is the "Overall Transmission Efficiency Analysis Calculations" presented as Appendix "A" in Southwest Research Institute's report entitled, "Design and Integration of a Hydrostatic Transmission in a 300 HP Marine Corps Amphibic Vehicle". Data

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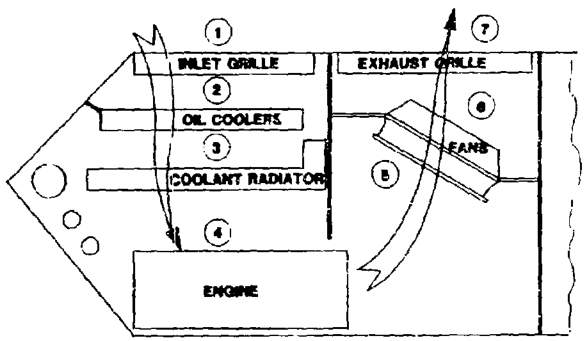
Table 4.4.1-1 ATR Cooling System Summary



				1	2	3	4	5	6	7
				TO INLET GRILLE	TO OIL COOLERS AND SHELL/TUBES	TO RADIATOR AND SHELL/TUBES	TO ENGINE COMP'T.	TO FANS	TO EXHAUST GRILLE	TO AMBIENT
AIR:	Temperature	^oF	Ta	125	125	135	165	180	180	180
	Density	lbm/ft^3	ρa	0.068	0.068	0.0668	0.0636	0.062	0.062	0.062
	Specific Heat	BTU/lbm^oF	Cpa	0.24	0.24	0.24	0.24	0.24	0.24	0.24
	Volume Flow Rate	CFM	CFM	18292	18292	18593	19528	20000	20000	20000
	Mass Flow Rate	lbm/min	Ma	1242	1242	1242	1242	1242	1242	1242
OIL:	Temperature	^oF	To		190	162				
	Density	lbm/gal	ρo		6.93					
	Specific Heat	BTU/lbm^oF	Cpo		0.56					
	Volume Flow Rate	GPM	GPMo		30.0					
	Mass Flow Rate	lbm/min	Mo		170					
SEAWATER:	Temperature	^oF	Tsw		70	78.1	102.3			
	Density	lbm/ft^3	ρsw		64	64				
	Specific Heat	BTU/lbm^oF	Cpsw		0.94	0.94				
	Volume Flow Rate	GPM	GPMsw		46	46				
	Mass Flow Rate	lbm/min	Msw		394	394				
COOLANT:	Temperature	^oF	Tc			210	195.5			
	Density	lbm/ft^3	ρc			63.28				
	Specific Heat	BTU/lbm^oF	Cpc			0.862				
	Volume Flow Rate	GPM	GPMc			85				
	Mass Flow Rate	lbm/min	Mc			719				

			INLET GRILLE	OIL COOLERS & SHELL/TUBES	RADIATOR & SHELL/TUBES	ENG. COMP'T.	FANS	EXHAUST GRILLE
Heat Rejection	BTU/min	Q	-	3000	8960	4470	-	-
Face Area	ft^2	A	7.3	7.3	6.25	-	-	8.0
ΔAir Pressure	in H_2O	ΔPa	-1.8	-5.5	-4.5	TBD	+5.4	-1.6
ΔOil Pressure	PSI	ΔPo	-	-15.6	-	-	-	-
ΔSea Water Pressure	ft H_2O	ΔPsw	-	-1.6	TBD	-	-	-
ΔCoolant Pressure	PSI	ΔPc	-	-	-4.7	-	-	-

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from the five highest engine output power conditions in the analysis was reviewed to determine the highest hydrostatic transmission heat rejection rate. This extracted data and the conversion to "lost power" is shown in Table 4.4.2-1. Condition "E" yields the highest "lost power" summation at 68.9 HP. Although the hydrostatic transmission oil cooler will not receive all of this "lost power" as heat to be rejected, it is, however, a close approximation. The second analysis was provided by Linde Hydraulics. This analysis assumes that the hydrostatic transmission system operates at 80% efficiency. Thus, if the net transmission input power is 295 HP, the heat rejection will equal 59 HP.

The third ATR hydrostatic transmission heat rejection analysis was conducted by the Marine Corps Program Office at DTNSRDC. This analysis was based upon the performance of the M113 Hydrostatic Vehicle, and was adjusted for final drive efficiency-differences between the two vehicles. It was recommended that the ATR hydrostatic transmission cooling system be based upon 69 HP of heat rejection. Considering these analyses for transmission heat rejection and the fact that the ATR auxiliary hydraulic systems will also add some heat to the transmission system oil, it was concluded that the ATR transmission oil cooling system must be capable of rejecting 70.7 HP (3000 BTU/MIN) of heat.

From the oil cooling viewpoint, the ATR hydrostatic transmission and auxiliary hydraulic system on the ATR is divided into two basic subsystems; a port side subsystem, and a starboard side subsystem. Each subsystem consists of a hydrostatic transmission pump with its respective hydrostatic motors, an auxiliary hydraulic pump with its respective hydraulic motors and actuators, a hydraulic oil reservoir and an oil cooler. There is no mixing of oil between these two subsystems. Since each of the subsystems is essentially equal in heat load, each oil cooler must be capable of rejecting 50% of the previously discussed heat load value.

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Table 4.4.2-1 Hydrostatic Transmission Lost Power Analysis

System No.	Vehicle Speed (MPH)	Engine Output Power (HP)	Splitter Box		Boost Pump Power (HP)	Hydrostatic Pumps		Hydraulic Line Lost Power (HP)	Hydrostatic Motors		Summation of Lost Power (HP)
			Lost Power (HP)	Output Power (HP)		Lost Power (HP)	Output Power (HP)		Lost Power (HP)	Output Power (HP)	
B	2.86	240.20	4.80	235.40	7.0	27.06	201.34	0.24	17.76	183.34	52.60
C	5.71	249.12	4.98	244.14	7.0	23.04	214.10	0.80	22.82	190.48	53.66
D	11.42	274.78	5.50	269.28	7.0	25.66	236.62	1.10	25.24	210.28	59.0
E	16.64	256.14	5.12	251.02	7.0	24.06	221.96	1.32	36.52	184.12	68.9
H	25.70	263.38	5.26	258.12	7.0	22.22	228.90	1.38	22.16	205.36	52.76
M	40.0	211.38	4.22	207.16	7.0	33.02	167.14	0.68	27.58	138.88	68.28
*	*	*		*	*		*	*		*	

*Denotes data extracted from Southwest Research Institute's report entitled "Design and Integration of a Hydrostatic Transmission in a 300 HP Marine Corps Amphibious Vehicle", Appendix "A".

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ATR Vehicle Hydrostatic Transmission Oil
Cooler Single Unit Requirements

1. Heat Load: 1500 BTU/MIN

2. Oil: Quintolubric 822-220
   Viscosity 13.5 c St @ 190°F
   70 SuS @ 190°F
   Pressure 280 PSI regulated (including cold start)
   Temperature @ cooler in 190°F maximum
   @ cooler out 170°F maximum
   Flow rate 12.3 GPM

3. Air: Ambient air flows through oil coolers first
   Ambient air temperature 125°F
   Air volume flow rate 18292 CFM maximum available

4. Sea water: Temperature 70°F
   Flow rate 46 GPM at 6 foot head

The port side and starboard side landborne l coolers are combined into one assembly, however, they each maintain their separate oil flow circuits. The custom made tube and fin oil cooler assembly is bolted to the engine radiator structure. Dunham Bush was selected to manufacture the oil cooler assembly because their Kool-Mor® Thin-Fin® heat exchangers feature a unique internal fin turbulator. The turbulator, located within the circular tubes, forces the fluid over and through aluminum fingers to resist the formation of a boundary layer. This provides an increased rate of heat rejection and resulted in a smaller and lighter oil cooler package. The entire two unit oil cooler assembly has a dry weight of 100 pounds and holds about 3 gallons of oil, therefore the wet weight is about 120 pounds.

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Single Side Unit Oil Cooler Performance

Air Flow Rate (CFM)	9146	6860	4573	2500
Air Pressure Drop (in. H_2O)	5.5	3.5	1.8	0.68
Air In Temperature (°F)	125	125	125	125
Air Out Temperature (°F)	135	137	141	139
Oil Flow Rate (GPM)	15	15	15	4.12
Oil Pressure Drop (PSI)	14.8	14.8	14.8	4.6
Oil In Temperature (°F)	190	190	190	190
Oil Out Temperature (°F)	162	164	167	149
Heat Rejection (BTU/MIN)	1527	1408	1232	597

The marine oil cooling system includes two shell and tube bundle heat exchangers manufactured by the Young Radiator Company as part number HF-302-EY-IP-CNTB. These single pass units are installed into the ATR with the sea water flowing through the 90-10 copper nickel tubes and the oil flowing through the shell around the tubes. The single side unit heat exchanger performance is as follows:

Sea Water Flow Rate (GPM)	46
Sea Water Pressure Drop (PSI)	1.6
Sea Water In Temperature (°F)	70
Sea Water Out Temperature (°F)	74
Oil Flow Rate (GPM)	15
Oil Pressure Drop (PSI)	0.8
Oil In Temperature (°F)	190
Oil Out Temperature (°F)	162
Heat Rejection (BTU/MIN)	1500

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4.4.3 Final Drive Oil Cooling

The ATR is equipped with a shell and tube bundle oil to engine coolant final drive heat exchanger. A Young Radiator Model Number F-302-HY-2 pass unit was selected.

Engine Coolant Flow Rate (GPM)	10
Engine Coolant Inlet Temp. (°F)	185
Engine Coolant Outlet Temp. (°F)	191
Oil Flow Rate (GPM)	11
Oil Inlet Temp. (°F)	211
Oil Outlet Temp. (°F)	200
Heat Rejection (BTU/MIN)	424

4.4.4 Engine Cooling

Requirements

1. Heat Load: (320 HP) x (28 BTU/HP-MIN) = 8960 BTU/MIN
2. Coolant: Aqueous solution of ethylene glycol (50/50% by volume)
   Coolant volume in block - 6.5 gal
   Top tank temperature - 210°F maximum
   Coolant jacket flow rate - 85 GPM at 2800 RPM at 9.8 ft H_2O head
3. Air: Ambient air flows through oil coolers first, therefore, the radiator in air temperature is 134.4°F
   Air volume flow rate 18292 CFM maximum available
4. Sea Water: Temperature 78.1°F
   Flow rate 46 GPM at 6 FT head for oil coolers and radiator systems combined

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The engine coolant radiator selected for use in the ATR is manufactured by the Young Radiator Company in Racine, Wisconsin. This radiator includes the coolant deaeration system in the top tank. The 6 tube row and 10 fin per inch core section is 30 inches by 35.5 inches by 5.3 inches deep.

Heat rejection	8960 BTU/MIN
Coolant in temperature	210°F
Air in temperature	135°F
Air out temperature	165°F
ΔP coolant	4.7 PSI
ΔP air	4.5 in H_2O
Dry weight	260 lb
Coolant volume	10 Gal.
Wet weight	330 lb.

The marine engine cooling system includes one shell and tube bundle heat exchanger manufactured by the Young Radiator Company as Part Number F-602-ER-1P-CNTB. This single pass unit is installed with the sea water flowing through the 90-10 copper nickel tubes and the engine coolant flowing through the shell around the tubes. The heat exchange performance is as follows:

Sea Water Flow Rate (GPM)	46
Sea Water In Temp (°F)	78
Sea Water Out Temp (°F)	101
Coolant Flow Rate (GPM)	85
Coolant In Temp (°F)	210
Coolant Out Temp (°F)	195
Heat Rejection (BTU/MIN)	8960

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4.4.5 Cooling Fans

The ATR is equipped with two NOA Airscrew Howden Part Number 41200-1 mixed flow fan units as shown in Figure 1.2-2 and Figure 4.4.5-1. This design has non-stall, non-overloading characteristics. The backward inclined blades remain clean in dusty conditions, thus maintaining efficiency. The fans are driven by fixed displacement hydraulic motors combined with variable delivery pumps for efficient control by thermostatic devices. The rated performance of a single fan unit is shown in Figure 4.4.5-2. The following cooling fan calculations combine the rated performance of two fan units and adjust this performance for corrected air density based upon anticipated fan inlet air temperature.

Air Temperature at Fan Inlet, 180°F = Ta_5

Air volume flow rate through 2 fans,

$2(10000) \quad = 20000 \text{ CFM} = CFM_{5-6}$

Air density at fan inlet, temperature corrected,

$$0.075\left(\frac{460+70}{460+180}\right) = 0.062 \text{ lbm/ft}^3 = \rho a_5$$

Air mass flow rate through 2 fans

$20000(0.062) \quad = 1242 \text{ lbm/min} = Ma$

Air pressure rise, density corrected,

$$6.5\left(\frac{0.062}{0.075}\right) = 5.4 \text{ INH}_2\text{O} = \Delta Pa_{5-6}$$

Fan power, $2(17) \quad = 34 \text{ HP} = HP_F$

The operation of the grilles and fans is controlled automatically by the microprocessor. After the engine is started, the inlet and port side exhaust grilles open and the port fan will operate at high speed. Operation of the starboard side fan is based upon engine coolant or hydraulic oil temperature per the following schedule:

Hydraulic oil Temperature	or Engine coolant Temperature	Starboard Fan Speed	Fan Drive Motor Oil Flow Rate
140°F	200°F	Low	4 GPM
165°F	205°F	Medium	8 GPM
180°F	210°F	High	12 GPM

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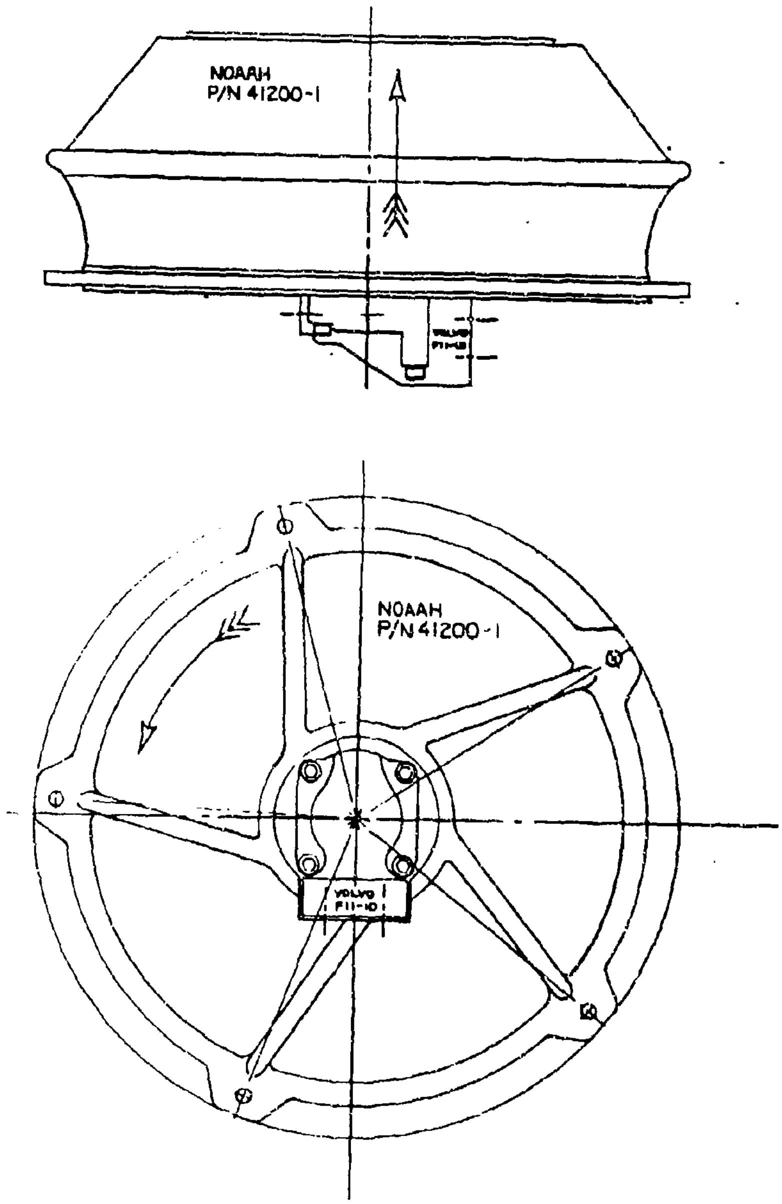

Figure 4.4.5-1 Mixed Flow Cooling Fan

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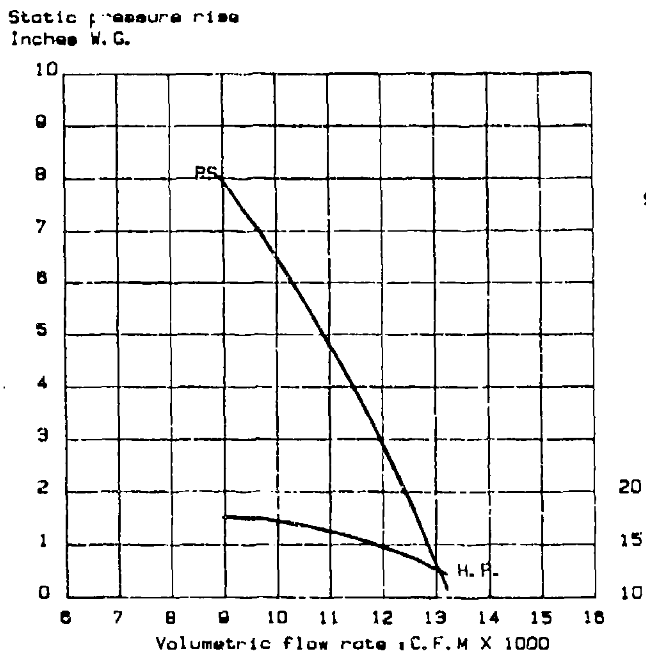

Customer : AAI
Inquiry # : C1140
Date : 05/14/84

Specified data -
Fan speed : 4150 R.P.M.
Standard data -
Fan type : MIXED FLOW
Part # : 41200-1
Nominal supply : HYDRAULIC DRIVE
Impeller dia. : 15 Inches
Nominal weight : 53 lbs

Figure 4.4.5-2. Mixed Flow Cooling Fan Performance

4.4.6 Fan Drive Motors

The fixed displacement hydraulic motors driving the cooling fans are manufactured by Volvo. The motor shaft is designed to fully support the rotating fan impeller. The 0.60 cubic inch per revolution displacement motor will operate at 2000 psi and 4150 rpm when full fan air volume flow is desired. The calculated torque required is 187 in-lbs. The volumetric efficiency of the motor is about 98%, therefore, a hydraulic flow rate of about 11 GPM is required.

4.4.7 Inlet and Exhaust Grilles

The inlet and exhaust grilles must each allow the passage of 1242 lbm/min of cooling air with a minimum of air flow restriction. The available combined inlet and exhaust grille dimensions are 58 inches long and 38 inches wide, thus providing 15.3 square feet of face area. This face area must be allocated between the inlet and exhaust sections based upon the air volume flow rate through each section.

Air mass flow rate through inlet grille, = 1242 lbm/min = Ma

Air temperature at inlet grille = ambient, = 125°F = Ta_1

Air density at inlet grille,

$$0.075 \left(\frac{460 + 70}{460+125}\right) = \qquad = 0.068 \text{ lbm/ft}^3 = \rho a_1$$

Air volume flow rate through inlet grille,

$$\frac{1242}{0.068} \qquad = 18292 \text{ CFM} = \text{CFM}_1$$

The air volume flow rate through the exhaust grille is about equal to the air volume flow rate through the fans, 20000 CFM. Therefore, the total inlet and exhaust air volume flow rate through the combined grille is equal to 38292 CFM.

Inlet grille allocated area

$$15.3 \left(\frac{18292}{382921}\right) \qquad = 7.3 \text{ FT}^2 = A_I$$

Exhaust grille allocated area

$$15.3 \left(\frac{20000}{38292}\right) \qquad = 8.0 \text{ FT}^2 = A_E$$

The standard air pressure drop through the inlet grille and the exhaust grille is predicted to be 2.0 inches of water for each section. Adjusting this value for air temperature and corrected air density yields the following:

Air pressure drop through inlet grille,

$$2.0\left(\frac{0.068}{0.075}\right) = 1.8 \text{ in } H_2O \qquad = \Delta Pa_{1-2}$$

Air pressure drop through exhaust grille,

$$2.0\left(\frac{0.062}{0.075}\right) = 1.6 \text{ in } H_2O \qquad = \Delta Pa_{6-7}$$

4.4.8 Engine Compartment Cooling

The engine compartment, being in the cooling air flow path, is ventilated by the fans. Negative air pressure is maintained in the engine compartment to prevent the accumulation of dangerous fumes.

The estimated heat radiated to engine compartment from the engine, the hydraulic pumps and the splitter box is 4470 BTU/MIN (105 HP). This increases the cooling air temperature by 15°F, and results in a cooling fan inlet air temperature of 180°F.

4.4.9 Sea Water Circulating System

During the transition from landborne to amphibious operation, an electric sea water pump is turned on by the microprocessor. The cooling sea water flow path is as follows:

1. Inlet/Strainer - a pattern of 18 slots (1/4 inch x 1 inch) passing through the hull bottom plate near the port side number one roadwheel station.
2. Wet Box - a double walled enclosure about 6 inches long, 5 inches wide and 5 inches high. This box is used as a transition between the turbulent inlet/strainer and the hose leading to the pump.
3. Pump Inlet Hose - enclosed in a water tight compartment

4. Sea Water Pump - an electric bilge pump with dry run capability and a 46 GPM capacity against a 6 foot head. Enclosed in a water tight compartment.
5. Pump Outlet Hose - enclosed in water tight compartment
6. Sea Chest - a water tight enclosure, in the ATR bow, containing the three shell and tube bundle marine heat exchangers. Two access panels are provided for inspection of the sea water circulating hoses.
7. Discharge Hose - enclosed in water tight compartment
8. Discharge Port - located above the starboard side number one roadwheel station

4.5 Keel Contact Cooler Evaluation

Although a keel contact cooler is not employed on the ATR, this section provides a summary of the analyses performed to predict keel contact cooler performance and to determine the optimum design applicable to the ATR.

Known or Assumed Data:

- Heat transfer and fluid flow properties of engine coolant, hydraulic oil, sea water, and aluminum
- Heat transfer fouling factors
- Engine coolant and hydraulic oil flow rate and maximum allowable operating temperature
- Engine coolant and hydraulic oil heat load

Design variables:

- Fluid tube shape, size, and wall thickness
- Fluid tube length, and return bend diameter
- Fluid tube layout pattern, number of tubes per pass, and number of return bends
- method of integrating fluid tubes into hull

Resulting Factors to be Optimized

- Hull surface area required
- Engine coolant and hydraulic oil flow resistance
- Keel cooling system wet weight
- Keel cooling system integration and durability

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Fundamental heat transfer and fluid flow considerations must be balanced with the difficulty of integrating the keel contact cooling system into the ATR hull, and with the durability that the cooling system must provide.

- Higher hot fluid flow velocity increases turbulence and the rate of heat transfer, however, it also increases fluid flow resistance (or pressure drop) which, in turn, requires additional fluid pumps.

- Longer fluid tubes, with fewer flow reversals, reduces pressure drop, however, integration of the tubes into the hull becomes more difficult.

- Hot fluid tubes or passages with large width to height ratios increase the heat transfer rate, however, durability and integration considerations become limiting factors.

The optimum ATR keel contact cooling system configuration is shown in Figure 4.5.1. The predicted performance for this system is listed in Table 4.5.1.

Based upon the keel contact cooler predicted performance evaluation, it appears this cooling system could provide adequate heat rejection during amphibious operations. However, developmental tests would be necessary to guide the detail design and to reduce risk. Additionally, this keel contact cooling system, requiring considerable hull bottom area and additional pumps, must be compared to the shell and tube bundle cooling system that can be packaged in the ATR bow area and requires only one pump.

4.6 Engine Controls/Instruments

Of prime importance when designing a vehicular control panel is the basic premise that all controls requiring operation while the vehicle is in motion be located so that the driver can manipulate them with one hand, thus permitting the other hand to remain on the steering control. It was determined that the ATR steering control be of a "joystick" design rather than of a full or abbreviated "wheel' design because of space limitations. Furthermore, it was determined that the joystick steering unit could be best controlled by the right hand of most drivers. This, therefore, places the driver's control and instrumentation panels to the left of the driver.

The driver's instrument panel, shown in Figure 4.6-1, includes automotive analog type gauges to indicate engine oil pressure, engine coolant

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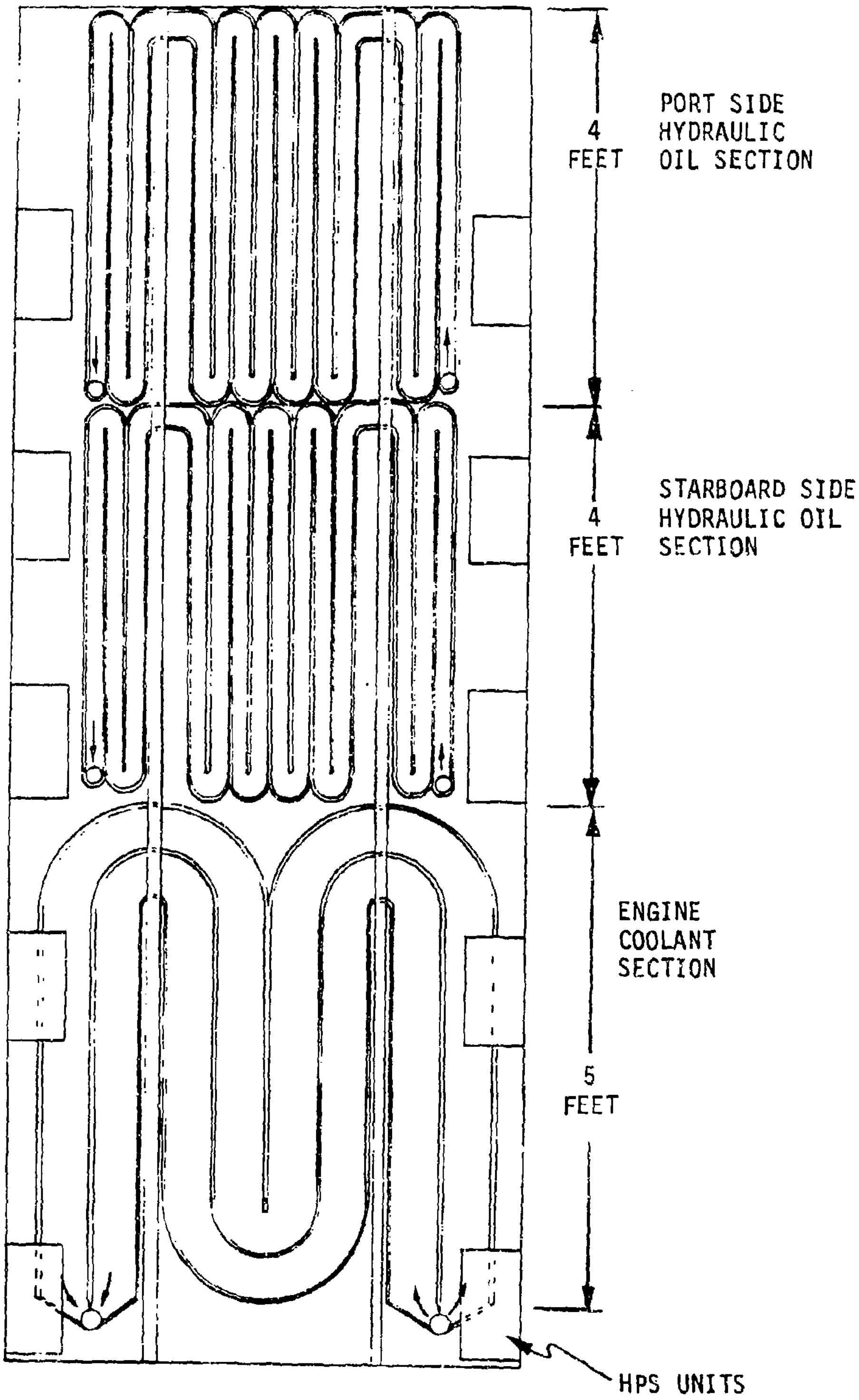

Figure 4.5-1 Keel Contact Cooler

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Table 4.5-1 Keel Contact Cooling Predicted Performance

		Engine Coolant System	Hydraulic Oil "One Side Only" System
Tube Width	Inches	5.5	2.0
Tube Height	Inches	0.37	0.25
Tube Length	Inches	60	48
Tubes Per Pass		2	1
Hot Fluid Flow Rate	GPM	85	12.2
Hot Fluid Reynolds No.		47147	3479
Hot Fluid Convection	BTU/HR ^{o}F FT^{2}	912.5	113.9
Sea Water Velocity	MPH	6	6
Sea Water Convection	BTU/HR ^{o}F FT^{2}	827.4	843.7
Heat Rejection Per Tube	BTU/MIN	1052	103
Quantity of Tubes Required		8	14
Quantity of Passes Required		4	14
Hot Fluid Pressure Drop	PSI	8	45
Cooler Wet Weight	Pounds	121	62
Cooler Surface Area	Sq. Feet	21	11
Hull Bottom Area Required for Integration	Sq. Feet	23.5	15
Available Heat Rejection	BTU/MIN	8416	1442
Required Heat Rejection	BTU/MIN	8960	1500

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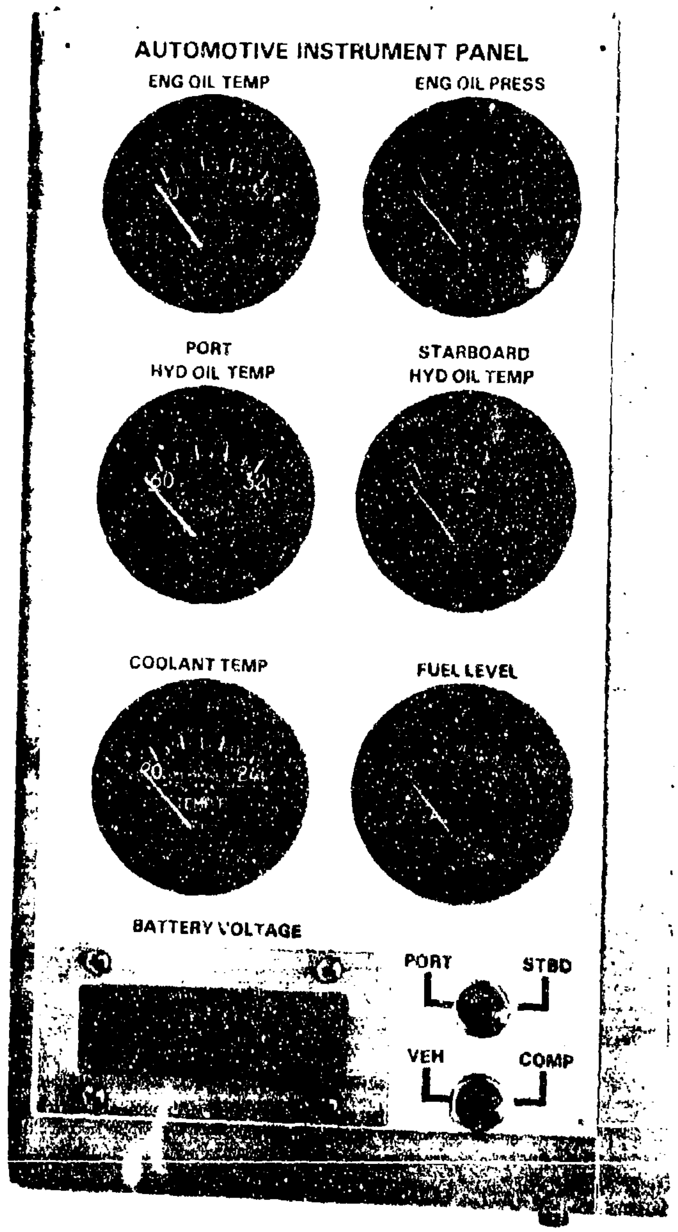

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Figure 4.6-1. Driver's Instrument Panel

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temperature and hydraulic oil temperature in both reservoirs. Via a switch, the vehicle battery or computer battery voltage levels can be monitored with a digital display. The fuel level in one tank is indicated via an analog gauge with a switch to select the tank to be monitored.

The driver's control panel, shown in Figure 4.6-2, includes vehicle lights, bilge pumps, radio and ramp operational control switches. This panel also includes the pushbutton transmission and final drive range selection switch labeled "gear select". The master power and automotive power on/off switches, the land/sea mode selection, and the suspension extent/retract switches are located close to the gear select control. The engine starter (cranking) control is provided as a toggle type switch, and the normal engine stop switch, controlling the engine fuel shut-off solenoid, is a spring loaded pushbutton.

The emergency engine stop control is located below the driver's right leg. This toggle valve will shut off fuel supply to the engine, independently of the engine's fuel shut-off solenoid.

Directly in front of the driver is a display panel, shown in Figure 4.6-3. This panel provides digital display of vehicle ground speed. The horn button, the direction signal switch and indicator lamp, the ramp open, grille malfunction, and master warning lamps are also included in this panel. Additionally, this driver's display panel includes the coolant loss lamp to indicate that the engine radiator coolant volume has decreased to a level below which effective engine cooling can be sustained.

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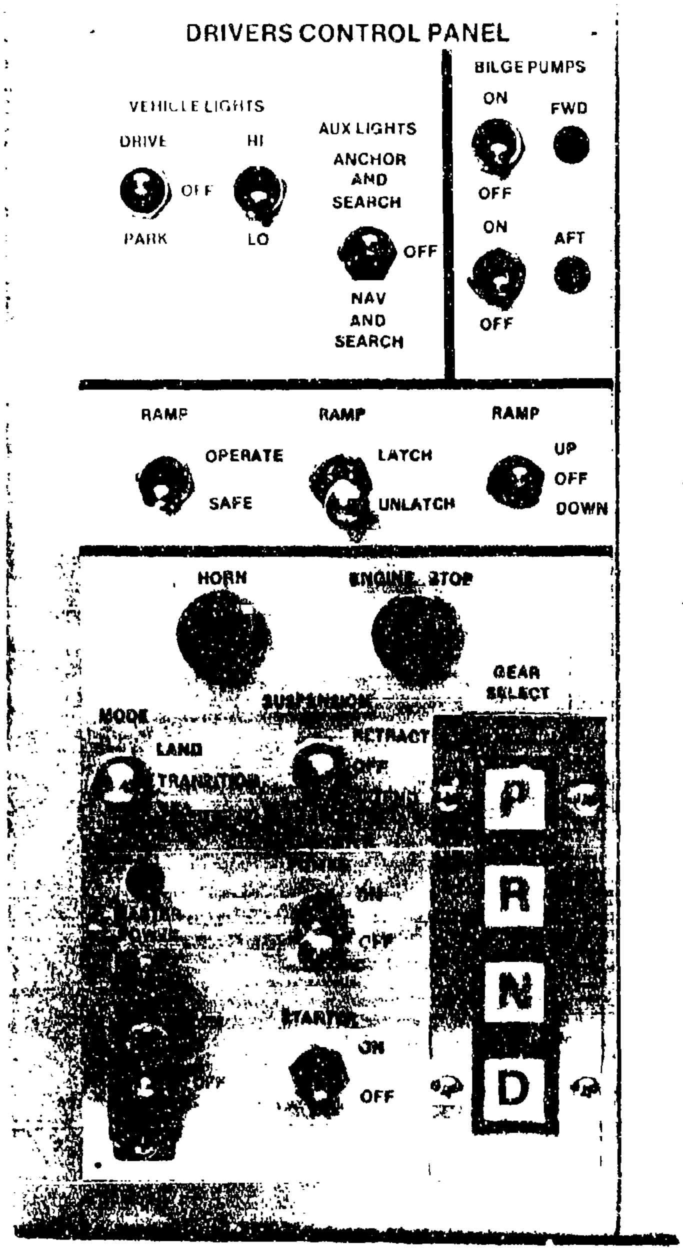

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Figure 4.6-2. Driver's Control Panel

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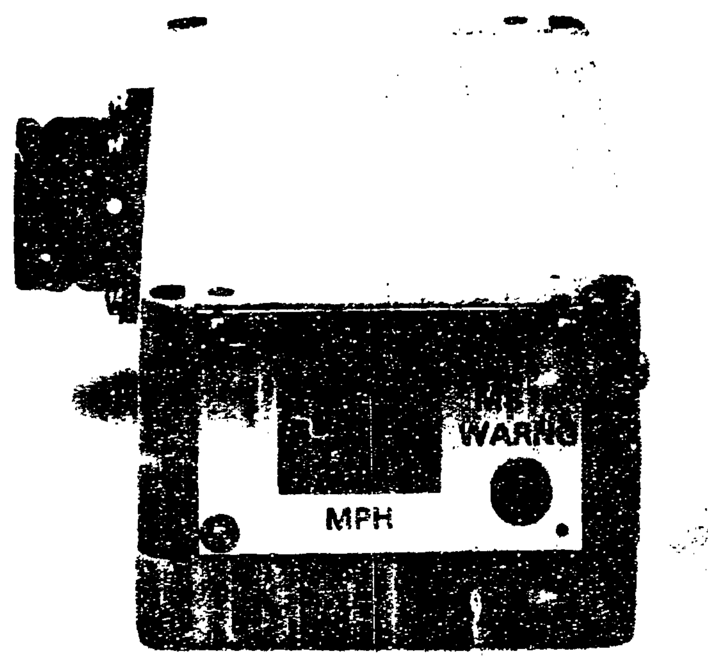

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Figure 4.6-3 Driver's Display Panel

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4.7 Fuel System

The ATR includes two, fifty-gallon capacity fuel tanks positioned above the waterjets in the aft sponsons. The fuel tank fill ports are fitted with locking protective covers, non-vented fuel caps and fuel strainers. Along side each protective cover is a small fuel tank pressure relief check valve vent that opens at 1.5 $\pm$0.5 PSI. Each tank is fitted with a fuel level transmitter, an inspection access panel and a vacuum relief check valve vent that opens at 1.0 +0.1-0.2 PSI of vacuum. This arrangement precludes the venting of fuel vapors to the interior of the vehicle and ensures that fuel spillage during refueling does not enter the troop space.

The tanks are constructed of 16 gauge stainless steel (selected for its corrosion resistance and durability) and incorporate internal baffles to minimize fuel sloshing. Below each fuel tank are supply and return line shut-off valves and a fuel tank drain hose. Fuel can be supplied from either of the single tanks or from both tanks by opening the appropriate supply and return valves. This arrangement permits the isolation of a contaminated tank until cleanout can be effected.

Additional fuel supply and return line shut-off valves are located in the engine compartment. The supply shut-off valve is operated from the driver's station. The fuel system water separator is located in the engine compartment above the port side number one suspension unit. The engine fuel filter and fuel priming pump is located above the front of the engine on the port side. Since the fuel tanks are well above the level of the engine fuel pump, auxiliary fuel boost pumps are not required. All fuel lines are routed behind covers to provide maximum protection from inadvertent damage.

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4.8 Engine Electrical System/Alternator

4.8.1 Engine Cranking Motor

The Caterpillar 3208T engine is equipped with one 24 volt cranking motor manufactured by the Delco-Remy Divison of General Motors Corp. This motor is identified as a series 40 MT Type 400 Model 1114845. This unit is sealed from the flywheel housing. The cranking motor is located on the right hand side of the engine.

4.8.2 Engine Fuel Shut-Off Solenoid

The Caterpillar 3208T engine is equipped with a fuel shut-off solenoid, mounted on and operating inside of the fuel system governor assembly. For the ATR, an "energize to run" solenoid was selected instead of an "energize to shut-down" solenoid. De-energizing of this solenoid causes the solenoid plunger to depress a lever in the fuel pump housing, which rotates the injector sleeve levers and terminates fuel delivery to the combustion chambers.

Additionally, the ATR is not equipped with an electric fuel transfer pump. The camshaft driven mechanical fuel transfer pump will draw the fuel from the fuel tanks, through the mechanical fuel shut-off valve and through the water separator and fuel filter into the governor/transfer fuel pump assembly whereupon fuel, under presure, fills the housing to fuel the injection pumps.

4.8.3 Vehicle Alternator

The vehicle electrical system alternator is powered by the engine and is triple belt driven at 2.46 times the engine speed. The alternator is mounted on the right hand side of the engine, forward of the engine cranking motor. TRW-Niehoff Division supplied this 24 volt/300 ampere model N1326-1 alternator.

4.8.4 Engine Speed Sensor

The engine magnetic speed sensor is located in the flywheel housing at the 7 o'clock position as viewed from the flywheel end.

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4.9 Propulsion Plant Enclosure

The ATR is equipped with an acoustically and thermally lined enclosure for the engine/splitter box/pump area. This enclosure is lined with Soundfoam® acoustic material having a Tedlar® film covering to protect the foam from moisture and oil. At 1000 Hz, one-inch thick foam with facing has a sound absorption coefficient in excess of 90 percent. Removable panels are provided in the enclosure to facilitate component inspection and servicing.

4.10 Propulsion Plant Mounts

The pump drive gear box is attached to the engine with the circular bolt pattern on the end of the flywheel housing. Additionally, the gear box is tied to the engine with the left and right rear engine support members. These rear supports and the front engine support are attached rigidly to the ATR hull.

4.11 Batteries and Mounts

AAI originally planned to use four (4), 6TN military batteries. However, commercially available "maintenance free" batteries have been selected during the detailed design phase. Below, in Table 4.11-1, the 6TN battery is compared to the alternate battery.

Table 4.11-1. Battery Comparison

	6TN (military type)	Commercial (GR 27)
Cold cranking amps	400	650
Length, in	11.25	12
Width, in	10.5	6.88
Height, in	8.04	9.0
Volume, cu in	946.2	742.5
Weight, lbs	71	36.5
Cost, $	100	60

The compactness and lower weight of the commercial batteries are the primary reasons for selecting them.

The batteries are located under the floor behind the driver. The batteries are contained within a battery box for short circuit protection. The battery hold downs are firmly fastened to the battery support frame in the bottom of the box. The batteries are supported on a resilient pad to reduce the chance of damage due to road shock.

To allow for routine maintenance, the floor is equipped with an access panel with quick release fasteners. Easy access is provided with adequate clearance for battery removal.

Ventilation holes are provided to eliminate the chance of moisture and hydrogen gas build-up. All holes in the battery box for power cables are fitted with grommets to prevent chaffing which may cause a short.

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5.0 HYDROSTATIC DRIVETRAIN

5.1 General Arrangement

The ATR hydrostatic drivetrain consists of a pump drive gearbox, variable displacement hydrostatic transmission pumps and motors, two-speed final drive units, a hydraulic system and the powertrain microcomputer control unit. The general arrangement of these components in the ATR are shown in Figure 5.1-1. Based on the ATR driver controlled inputs (from the steering control unit, the accelerator and brake pedals, the vehicle operational mode selector, and the transmission/final drive range selector), the microprocessor continually determines and controls; the engine speed (for most efficient operation), the hydrostatic pump and motor displacements (for desired vehicle performance), and the clutch apply and brake release pressures (when and as required).

The ATR is equipped with a Twin-Disc pump drive gearbox attached directly to the engine flywheel housing and driven via a flex-plate package without disconnect clutch. The two pad Twin-Disc unit, Model F11-PMD2, was selected for its desirable characteristics of which power capacity, pump mounting pad spread, and weight were of primary importance. The Linde BPV 100 hydrostatic pumps, with through shaft drive and mounting provisions for auxiliary hydraulic pumps, are used in the ATR. These variable displacement, axial piston pumps will hydrostatically drive the Linde BMV 186 variable displacement bent axis piston motors for landborne operation, and the Linde BMF 105 fixed displacement bent axis piston motors for seaborne operation. The BMV 186 motors are attached to the right angle drive input section of the GFE two-speed epicyclic final drive units. The right angle drive input section is rotated 14° above the horizontal to permit installation of the two aft bilge pumps. The Linde BMV 186 motors are provided, for the ATR, with end and side ports, to permit nigh pressure line installation. The two-speed epicyclic final drive units are installed on the ATR hull from the outside. The final drive units are interchangable side-for-side as are the hydr transmission pumps and motors.

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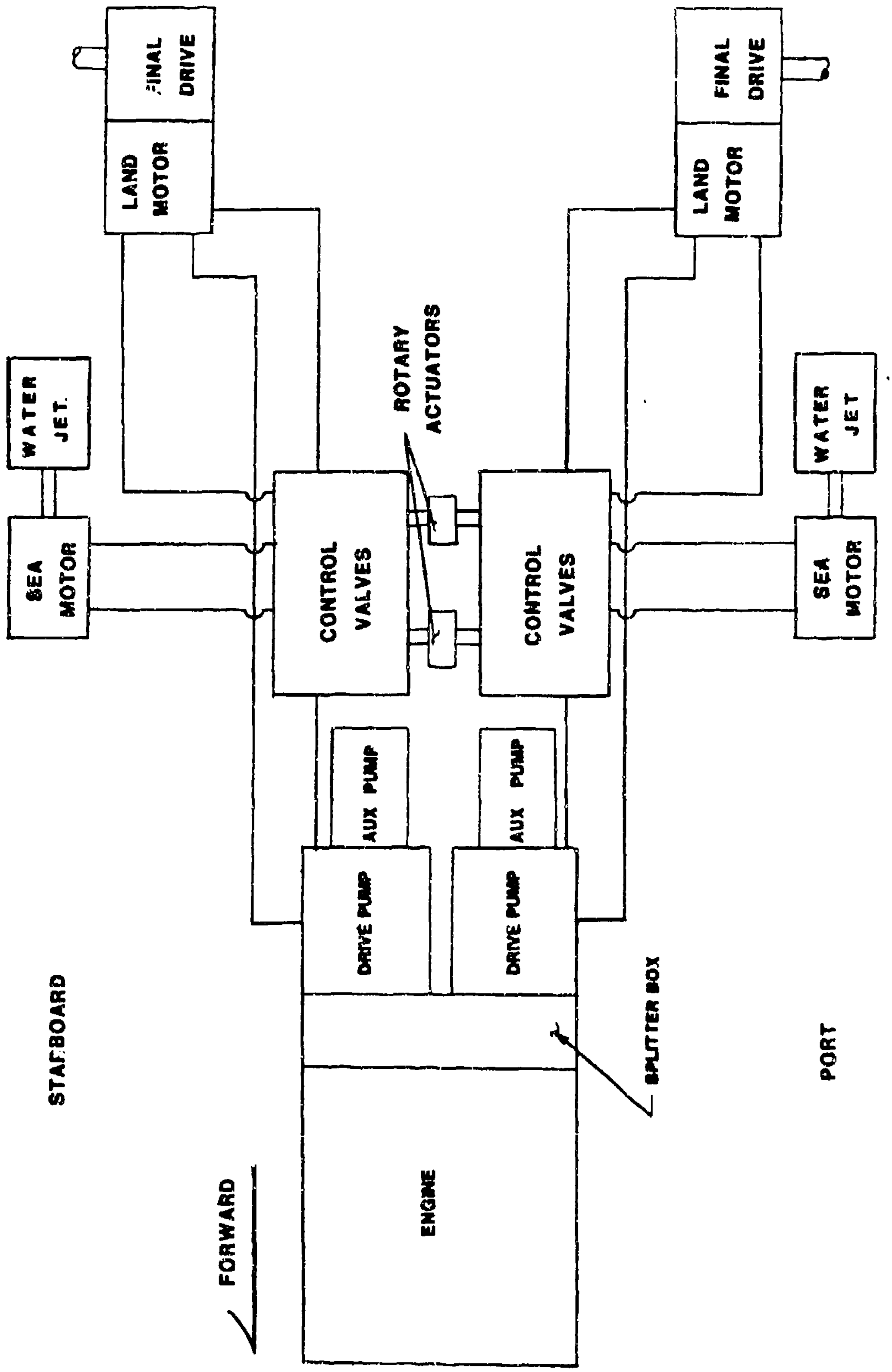

Figure 5.1-1 Hydrostatic Drivetrain General Arrangement

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5.2 Transfer Gearbox

A gearbox, attached to the engine flywheel housing and supporting the hydrostatic transmission pumps, is employed to transfer or split the engine output power between the port and starboard hydrostatic transmission systems.

It is desirable that the hydrostatic transmission pumps be installed in a symmetrical manner so that the control valves are positioned on top of both pumps, with all solenoids horizontal. To accomplish this, the distance between the two pump mounting pad centers must be almost 14 inches allowing for service of the solenoids, without pump removal, since the solenoids are the widest parts of the present pump/control valve assembly.

Several pump drive gearboxes are available with a 14 inch pump mounting pad spread. However, in an effort to reduce the gearbox weight, smaller units were reviewed for possible consideration. The smaller pump drive gearboxes typically have a 12 inch pump pad spread, and the gearbox in the M113 Hydrostatic Test Vehicle, has an 11.1 inch spread. The Linde Hydraulics BPV 100 hydrostatic transmission pumps cannot be mounted in a symmetrical manner on an 11.1 inch spread because the integral oil filter arrangement requires a pump pad spread of 12 inches, and remotely mounted oil filters are prohibited by Linde. Linde, however, did agree to design a new control valve with solenoids reconfigured to permit pump installation on a 12-inch mounting pad spread. Redesigning the control valve would considerably increase the control valve cost and would increase the risk associated with transmission performance control, but it could permit the use of a smaller pump drive gearboxe. If developmental efforts toward the design of a new control valve were to be considered, the design should address improvements of transmission control functions well beyond the basic capabilities of the present or reconfigured control valve. It was, therefore, determined that the pump mounting pad spread of about 14 inches would remain a requirement to permit installation and service of the present control valve design and to allow for the installation and service of a significantly improved control valve, should such a development effort be pursued.

In addition to the two Linde BPV 100 hydrostatic transmission pumps, the transfer grearbox must drive the ATR auxiliary hydraulic pumps. Initially considered were gearboxes with three pads in a triangulated

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configuration, with the transmission pumps on the lower two pads and the auxiliary pumps tandemly mounted on the upper pad. However, to allow for additional weapon station space claim, the two auxiliary hydraulic pumps are now removed from the third mounting pad and are installed with one auxiliary pump tandemly mounted to each transmission pump. This permits the pump drive gearbox to be of a two pad design. Furthermore, a pump drive gearbox with two output pads offset below the input shaft position (engine crankshaft centerline level) will provide greater weapon station volume and is therefore desirable.

The following transfer gearbox general physical and functional requirements were determined with the assistance of Linde Hydraulics, Southwest Research Institute and Caterpillar Tractor Co.:

1. Adaptable to the Caterpillar 3208T engine: via direct mount to SAE No. 2 flywheel housing; flex-plate drive without clutch.
2. Sufficient horsepower capability: 320 bhp input; 280 hp output per pad.
3. Minimum weight and size.
4. Sufficient spread between the two pads to mount both Linde BPV 100 pumps (with control valves on top and horizontal preferred): 13.5 inch minimum.
5. Output mounting pads offset below the input shaft centerline (preferred).
6. Output shaft spline and mounting pad compatibility with the Linde BPV 100 pumps: 15T 16/32 DP involute spline; SAE"B" mounting pad.
7. Sufficient self-lubrication capacity (externally supplied pressure lubrication not required).
8. Sufficient self-cooling capacity via radiation (external oil cooling not required).

After comparing the physical and functional characteristics of fourteen pump drive gearboxes from five manufacturers, as shown in Table 5.2-1, it was determined that the AL will incorporate the Twin-Disc Model FD11-PMD2 unit with custom output splines, providing 14.25 inches of spread between the pump mounting pads and 4.16 inches of vertical offset between the input shaft and the output pads.

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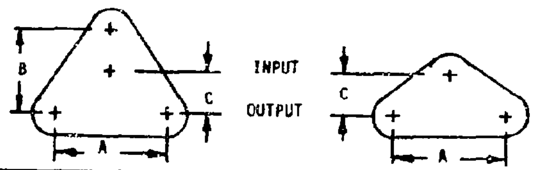
Table 5.2-1 Transfer Gearbox Candidates



MANUFACTURER	MODEL	NO. OF PADS	INPUT POWER HP	WEIGHT, DRY LBS	SPREAD "A" INCHES	"B" INCHES	OFFSET "C" INCHES	COMMENTS
FUNK	28211XA	3	475	230	11.1	8.3	2.3	THIS MODEL USED ON M113 HYDROSTATIC VEHICLE
FUNK	593XXP	3	475	310	12.0	13.3	5.3	
FUNK	592XXP	2	475	210	12.0	--	0.0	REQUIRES TANDEM PUMP SUPPORT
TWIN DISC	F11PMT2	3	500	417	14.25	12.41	4.16	1.106:1 REDUCTION OR 0.904:1 STEP UP RATIO
TWIN DISC	F11PMD2	2	650	300	14.25	--	4.16	1.106:1 REDUCTION OR 0.904:1 STEP UP RATIO
COTTA	PD220	3/2	600	285	16.0	0.0	0.0	3 PADS IN LINE
COTTA	PD230	3	600	325	13.8	12.0	5.3	
MARCO	DG26	2	250	350	12.0	--	0.0	
MARCO	DG37	3	400	400	12.0	11.2	4.0	
MARCO	DG38	3	448	515	13.0	13.2	5.0	
MARCO	DG39	3	448	320	12.2	6.0	6.0	NOT IN PRODUCTION, INPUT IN LINE WITH TOP PAD
FEDERAL	762F	2	425	185	12.0	--	0.0	REQUIRES TANDEM PUMP SUPPORT
FEDERAL	763F	3	600	240	10.8	8.5	2.5	
FEDERAL	803F	3	800	480	13 0	11.25	3.75	

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5.3 Hydrostatic Pumps

The ATR is equipped with two Linde BPV100 axial piston, closed loop, variable displacement pumps which have the following characteristics:

Displacement, cu. in./rev:	6.12
Maximum speed, rpm:	2800
Maximum pressure - intermittent, psi:	6000
- continuous, psi:	3600
Weight, dry, lbs	165.5

For the ATR, the pumps are equipped with through shaft PTO provisions for mounting and driving auxiliary hydraulic pumps. The auxiliary pump mounting flange conforms to SAE "B" dimensions and the drive shaft spline conforms to SAE "B" (13 teeth) dimensions.

The BPV100 incorporates: 1) an integral gear pump for make-up flow and servo control flow, 2) a built-in 10 micron filter, 3) high pressure relief valves (6000 psi), 4) make-up flow check valves, and 5) a by-pass valve for cold start. An electro-hydraulic servo valve mounted directly to the top of each pump is used to control the displacements of the pump and motors.

5.4 Hydrostatic Motors

The ATR is equipped with two Linde BMV 186 variable displacement, bent axis piston motors to drive the vehicle during landborne operations. Motor characteristics are as follows:

Displacement - maximum 28°, cu. in/rev:	11.35
- minimum 8°, cu. in/rev:	3.36
Maximum pressure - intermittent, psi:	6000
- continuous, psi:	3600
Maximum output torque - intermittent, lb-ft:	812
- continuous, lb-ft:	489
Maximum output speed - 28°, rpm:	2800
- 8°, rpm:	3500
Flow at maximum speed and displacement, GPM:	133
Weight, dry, lbs:	188.0

For cooling the closed loop hydrostatic drive oil the motor incorporates a "hot oil" shuttle valve. This valve permits a 1.5 GPM cooling oil flow to enter the motor case from the low pressure side of the motor working ports.

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For the ATR, application, the motors are provided with end ports and side ports to permit high pressure hydraulic line installation.

5.5 Final Drives

The ATR is equipped with government furnished two-speed, epicyclic final drives. The two-speed epicyclic final drive units meet the following requirements:

Gearbox Input Ratings

- Maximum input speed, rpm:	3500
- Maximum intermittent input power, hp:	450
- Maximum continuous input power, hp:	125
- High gear ratio, reduction:	4.480:1
- Low gear ratio, reduction:	10.4765:1
- Maximum input torque, lb-ft:	966

- Brake input torque, as required to stop the 14-ton vehicle from 5 mph on a 60% downgrade. The brake is linked to the output shaft under all operating scenarios and will not become disengaged during shifting.

Gearbox Output Ratings

- Maximum output speed, rpm:	788
- Maximum intermittent output torque, lb-ft:	10000
- Maximum side load capacity, lbs:	30000
- Minimum efficiency-low gear, percent:	94
-high gear, percent:	96

Weight

Overall wet weight per unit, lbs:	305.0

The two-speed final drives require a hydraulic system to supply lubrication, brake release pressure and clutch control pressure. Four supply and three drain ports are provided on each drive unit. The supply ports supply internal cavities with: (1) brake lubrication, (2) brake release pressure, (3) main lubrication and high clutch application pressure, and (4) low clutch application pressure. The drain ports are provided for oil scavenging to maintain the desired case fluid level. The specific hydraulic flow and pressure requirements to operate the final drive are shown in Table 5.5-1.

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Table 5.5-1 Final Drive Unit Hydraulic Requirements

	Low Range		Neutral		High Range		Emerg. Stop	
	PSI	GPM	PSI	GPM	PSI	GPM	PSI	GPM
Brake Lube	10	.7-1.0	10	.7-1.0	10	.7-1.0	10	.7-1.0
Brake Release	275 ± 25	0	275 ± 25	0	275 ± 25	0	0	
Main Lube	10	1.3-1.5	10	1.3-1.5	225 ± 25	18*	10	1.3-1.5
High Clutch					225 ± 25	3.3-3.8		
Low Clutch	225 ± 25	4.0* 0	10	0	10	0	10	0

* For first 200 milliseconds

- Input Oil Requirements: MIL-L-2104 30 weight only, filtered to 10 micron and less than 180^{0}F
- Output oil from brake and clutch compartments must be scavenged

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The schematic for the Control and Lubrication System is shown in Figure 5.5-1. The scavenge system schematic is shown in Figure 5.5-2. The pumps P1 and P2 are combined in one tandem pump which is hydraulically driven from the auxiliary hydraulic system. The pump drive system description is presented in Section 7.2.

A summary of the hydraulic valve function and operation for the Control and Lubrication and Scavenge System is as follows:

1. Accumulator (A1) - This component provides the flow required to quickly engage the high and low range clutches. During clutch engagement this accumulator supplements the flow from the supply pump. For this function the one quart bladder type accumulator is precharged to 100 psi.

2. Check Valve (CV1) - Permits 10 psi pressure to be applied to the low range clutch port when drive is in the high range. It also prevents the high pressure, required for low clutch engagement, from entering the brake lubrication ports. The cracking pressure of this valve is 3 psi.

3. Check Valve (CV3) - The check valve has a cracking pressure of 15 psi. When clutches are released the compartments will be allowed to depressurize quickly thus pemitting rapid declutching.

4. Directional Valve (DV1) - Permits fluid to enter and exit the brake release compartments. The deenergized position of the valve permits the brakes to be applied. The energized position allows control pressure to enter the brake release compartments thus holding the spring applied brakes from engaging.

5. Directional Valve (DV2) - Directs clutch apply flow to the high and low range clutch compartments.

6. Filter (F2) - Filters supply oil to 10 microns absolute. An electric differential pressure switch is used to signal that the element should be replaced. A light bar on the engineer's panel accompanied by a master warning fault will occur.

7. Filter (F3) - Filters the scavenge system oil to 25 microns absolute. This provides coarse filtration to protect the scavenge pump (P2).

8. Flow Control (FC1) - Controls the emergency brake apply time. This is a non-pressure compensated flow control.

9. Flow Control (FC2) - This is a pressure compensated flow control. This valve regulates the flow to the brake lubrication ports.

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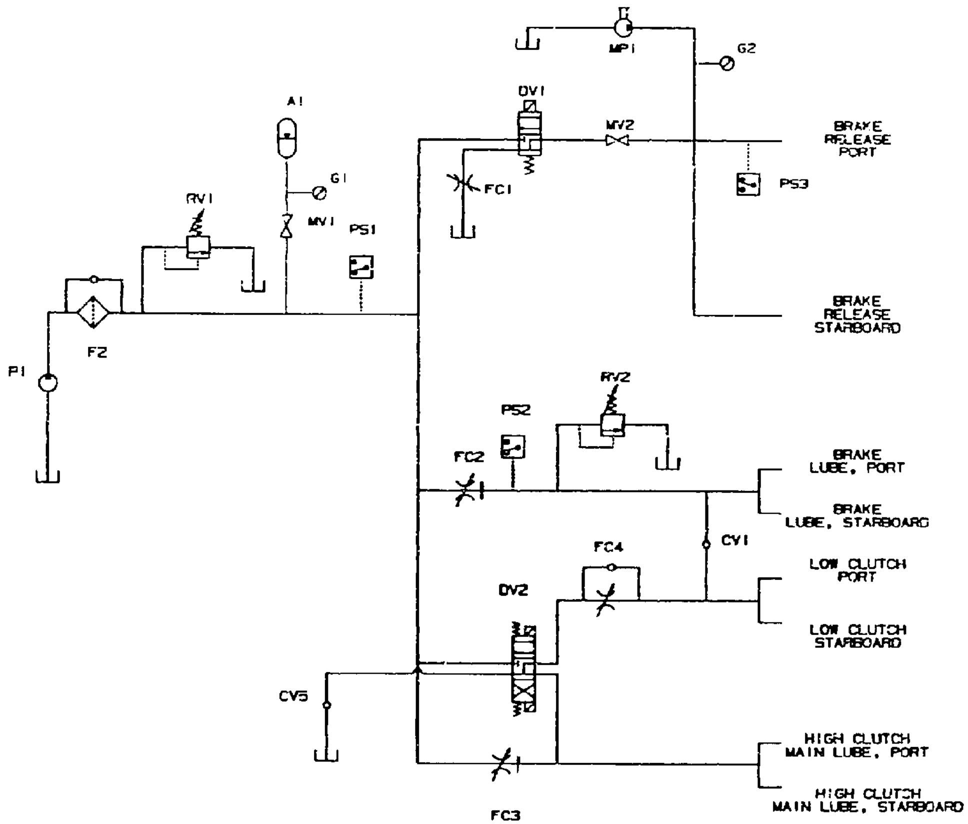

Figure 5.5-1 Final Drive Control and Lubrication System Schematic

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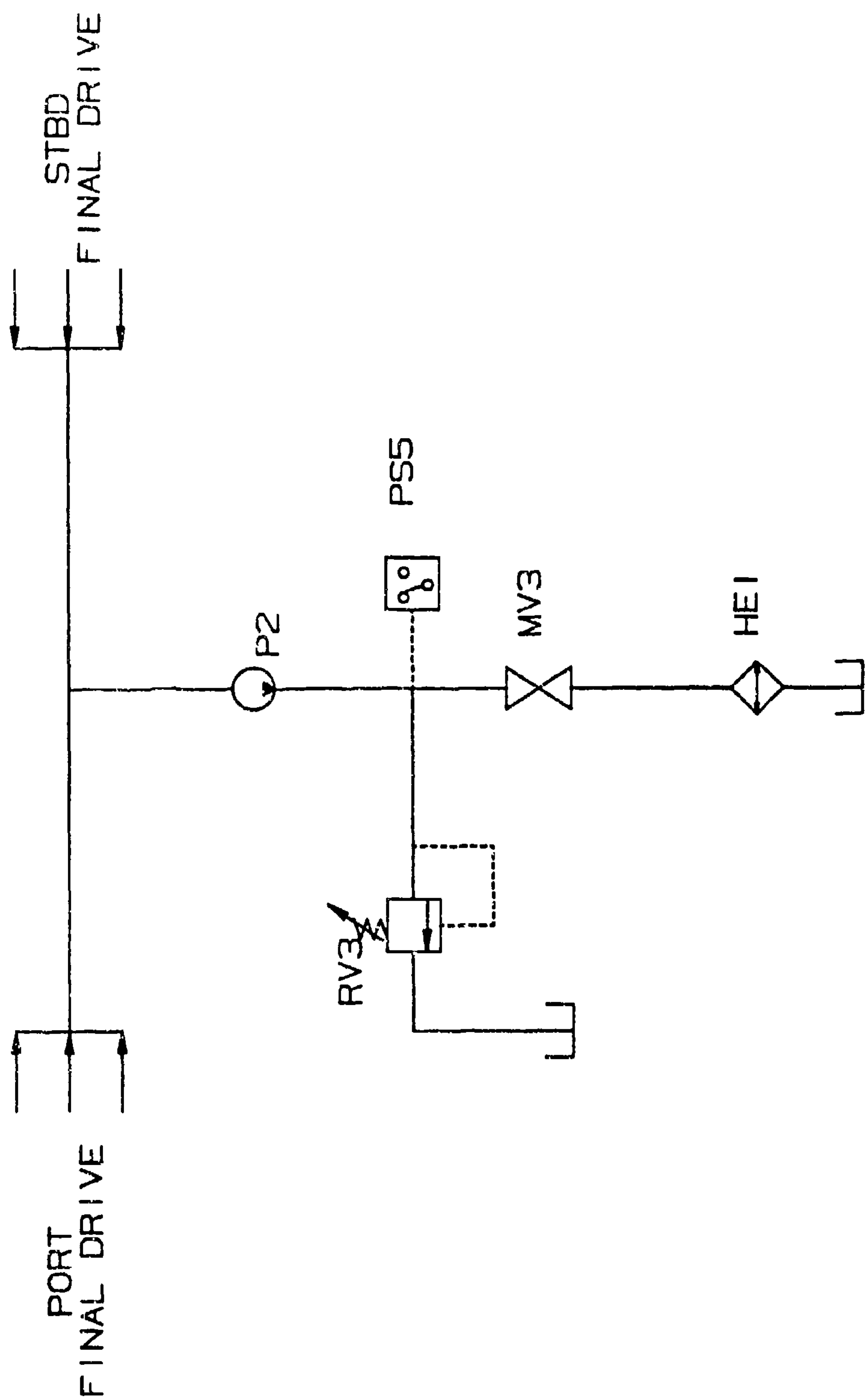

Figure 5.5-2 Final Drive Scavenge System Schematic

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10. Flow Control (FC3) - This is a pressure compensated flow control. This valve regulates the flow to the main clutch lubrication flow.

11. Flow Control (FC4) - This valve is used to regulate the low clutch engagement times.

12. Manual Pump (MP1) - Used to manually release the brakes for towing purposes.

13. Manual Valve (MV1) - Blocks accumulator flow during service of pressure lines. The normal position is open. The valve should be closed prior to final drive system maintenance.

14. Manual Valve (MV2) - Required for manual brake release. The normal position is open. For manual brake release the valve should be closed prior to pumping.

15. Manual Valve (MV3) - Required for servicing of scavenge circuit. Valve position is normally open. The valve should be closed to prevent excessive fluid loss.

16. Pump (P1) - This is the final drive control and lubrication supply pump. It supplies flow and pressure for lubrication, shift control, and brake release.

17. Pump (P2) - This is the final drive scavenge pump. This draws the fluid from the final drives thus maintaining the desired sump level.

18. Pressure Switch (PS1) - Low supply system pressure switch. Detects pressure below 200 psi. The switch will signal a master warning fault.

19. Pressure Switch (PS3) low brake release pressure. Detects pressures below 200 psi. The switch is used to detect an inadvertent brake application. The hydrostatic drive i capable of overcoming the brakes in low gear. This switch connects to the master warning system.

20. Pressure Switch (PS5) - Low scavenge system pressure. Detects the absence of pressure in the scavenge return to tank line. Pressure below 10 psi will signal the master warning system.

21. Relief Valve (RV1) - Prevents supply system pressure from exceeding 250 psi.

22. Relief Valve (RV2) - Prevents brake lube pressure from exceeding 10 psi.

23. Relief Valve (RV3) - Prevents cooler pressure from exceeding 100 psi.

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5.6 Hydrostatic Drivetrain Hydraulics

The hydrostatic drivetrain of the ATR consists of two closed loop hydraulic circuits. One system operates the port track and water drives and the other operates the starboard drives. The circuits are segregated to prevent total system contamination. Each system contains a high and a low pressure network of hoses and valves. The high pressure network is used to channel the power, hydraulic pressure and flow, from the pumps to the motors. The low pressure circuit is used for makeup oil, for the closed loop drive system, and for servo control of the pump and and motor displacements. The port and starboard drive systems are serviced by separate hydraulic reservoirs. For cooling purposes a cooler circulation system has been used which draws hot oil from the case drains and the reservoirs. The overall schematic for each of the hydrostatic drive systems is shown in Figure 5.6-1.

5.6.1 Hydraulic Fluid

The hydraulic fluid used in the ATR hydrostatic drive system and the auxiliary hydraulic systems is Quaker Chemical Corporation Quintolubric 822-220. The characteristics of this fluid are given in Table 5.6.1-1. This fluid was selected because of its high degree of fire resistance and lubricity. This fluid can be handled without operator discomfort, and if burned, generates non-toxic, non-noxious smoke.

Table 5.6.1-1 Typical Properties of Quintolubric 822-220

Appearance		Light amber, clear fluid
Specific Gravity	77°F	0.9088
	122°F	0.8925
Moisture, wt % max		0.1
Pour Point (ASTM D97)		-15°F
Viscosity	0°F	5500 SUS
	32	1500
	77	380
	100	220
	122	140
	167	77
	210	55

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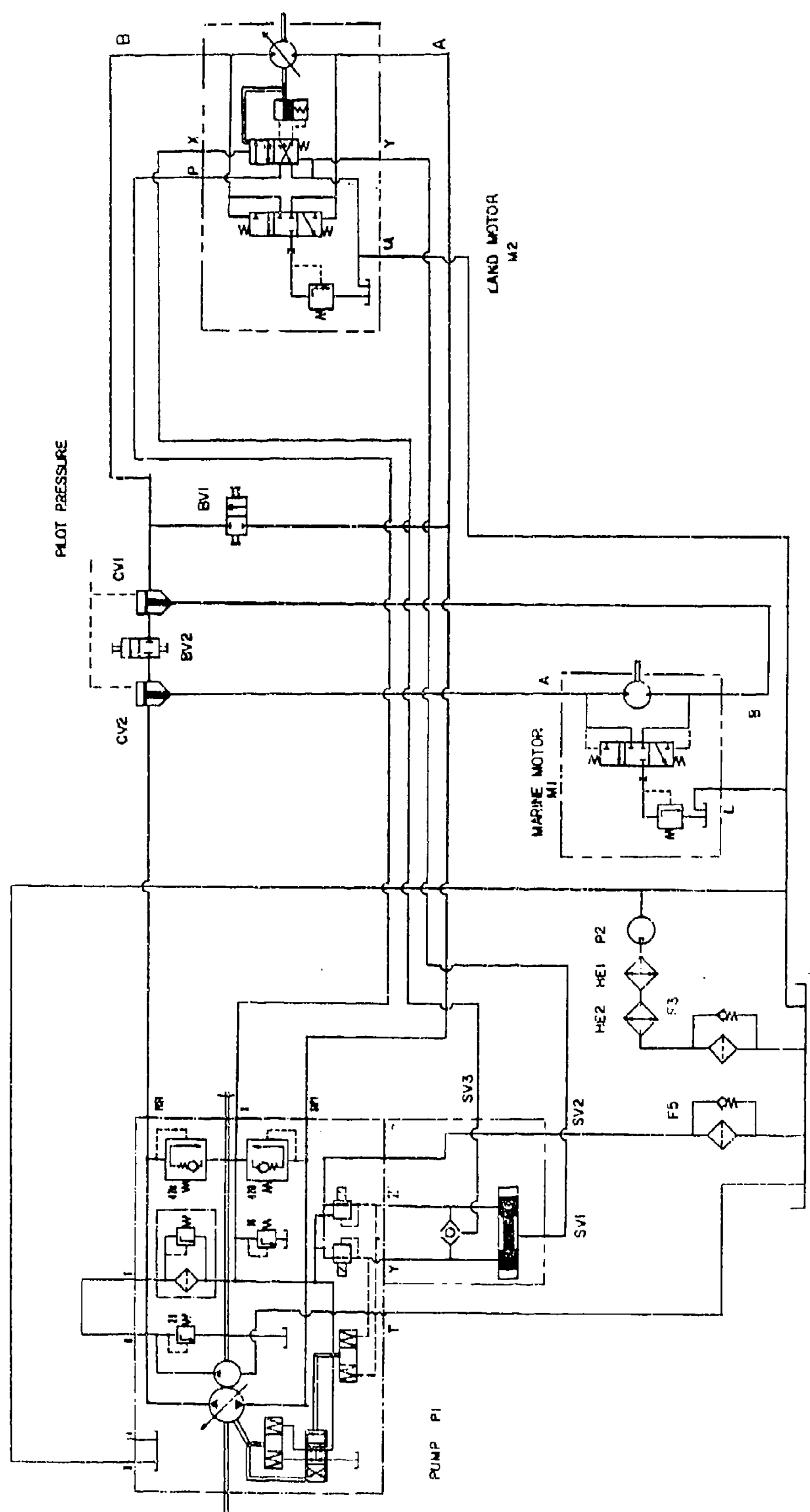

Figure 5.6-1 Hydrostatic Drive System Schematic

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Viscosity Index .185

ISO Grade 46

Specific Heat @ 68°F .49 BTU/LB -°F

Flash Point 500°F

Fire Point 550°F

Auto ignition 900°F

5.6.2 High Pressure Hydraulics

The closed loop hydrostatic drive system on the ATR operates at pressures of up to 6000 psi depending on the drive load. Through the simple hydraulic circuity, the operator has the capability of selecting various drive modes. With this hydrostatic drive system there are three modes of operation:

1. land
2. sea
3. land and sea

Figure 5.6.2-1 shows the high pressure hydraulic circuitry used in the port and starboard systems. The drive motors are in a series type arrangement thereby eliminating the need for flow divider valves. To facilitate forward propulsion the fluid flows first through the land motor and then to the sea motor. For motor selection, large high pressure ball valves are used as motor bypass valves. These valves, when in the open condition, provide an alternate path for fluid flow with little or no pressure drop. Since the motor experiences no pressure drop across the working ports, rotation does not occur. With the ball valve closed the alternate flow path is eliminated and motor rotation occurs.

The auxiliary hydraulic system is used to control the position of the ball valves. The port and starboard sea motor ball valves are activated by a single rotary actuator. This insures symmetrical mode change operation. The same arrangement is provided for the land motor ball valves.

The cartridge valves shown in Figure 5.6.2-1 are capable of blocking flow and pressure to the sea motor working ports when pilot pressure is applied from the auxiliary hydraulic system. During the land mode of operation the cartridge valves are energized. This prevents the high return line pressures, which are experienced during dynamic braking and steering, from entering both working ports of the sea motor. Without the cartridge

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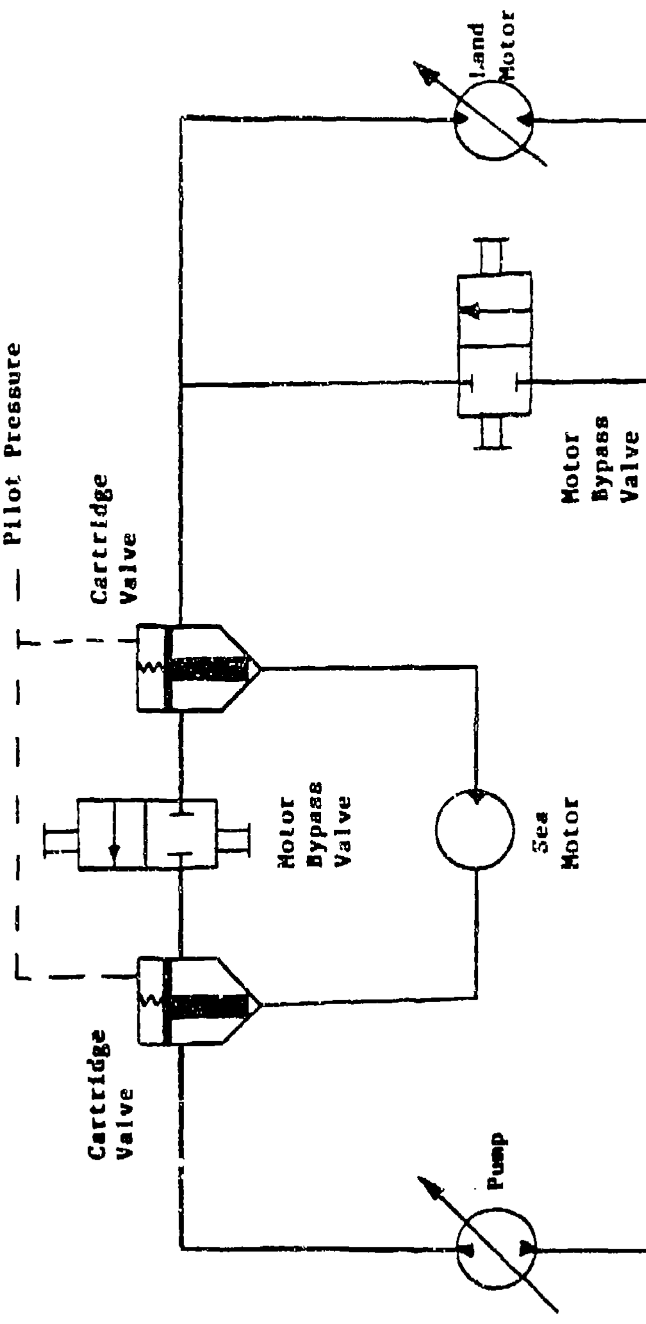

Figure 5.6.2-1 High Pressure Hydrostatic Drive System Circuitry

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valves a summated pressure of 12000 psi would be applied to the motor which exceeds the structural limitation of the housing. During the sea and land/sea modes the cartridge valves are vented to the tank thus permitting sea motor flow.

The ball valves and cartridge valves for port and starboard transmissions are mounted as an assembly shown in Figure 5.6.2-2. This assembly consists of 4 ball valves, two rotary actuators, 4 manifolds and 4 cartridge valves all mounted to a single plate. This assembly is installed in the ATR below the floor plates under the weapon station.

The primary hose routing and the overall installation of the hydrostatic components is depicted in Figure 5.6.2-3. With the exception of the mode control valve assembly, all connections between motors and pumps are made with high pressure hose. The hose selected is Caterpillar XT-6 (6000 psi continuous rated) and XT-5 (5000 psi continuous rated). The XT-6 hose used has 1 1/4 inch inside diameter and is used for land drive connections with the exception of one 1-inch diameter XT-5 hose used on the return line to the pumps. One inch diameter XT-5 hose is also used to connect the control valve assembly to the sea motors.

5.6.3 Low Pressure Hydraulics

For control of the pump and motor displacements an electro-hydraulic control valve is used. The servo valve receives a variable voltage from the microcomputer. Using the charge pump as a controlled pressure source of 400 psi, the servo valve converts the voltages into control pressure. This control pressure can vary the pump displacement from -6.12 cu. in/rev to plus 6.12 cu. in/rev and the land motor displacement from 3.36 cu. in/rev to 28 cu. in/rev.

For control of the pump displacement, all fluid connections are made internal to the pump case. For control of the land motor displacement, three low pressure hoses connecting the pump to the motor are required.

5.6.4 Oil Cooler Circulation System

The oil cooler circulation system pumps fluid from auxiliary hydraulic and and hydrostatic drive case drain lines through the sea and land heat exchangers as shown in Figure 5.6.4-1. The tandem cooler circulation pumps are driven by a hydraulic motor on the primary fan auxiliary hydraulic system. This system is capable of maintaining constant cooler flow of 12 to 15 GPM to both the port and starboard hydraulic reservoirs.

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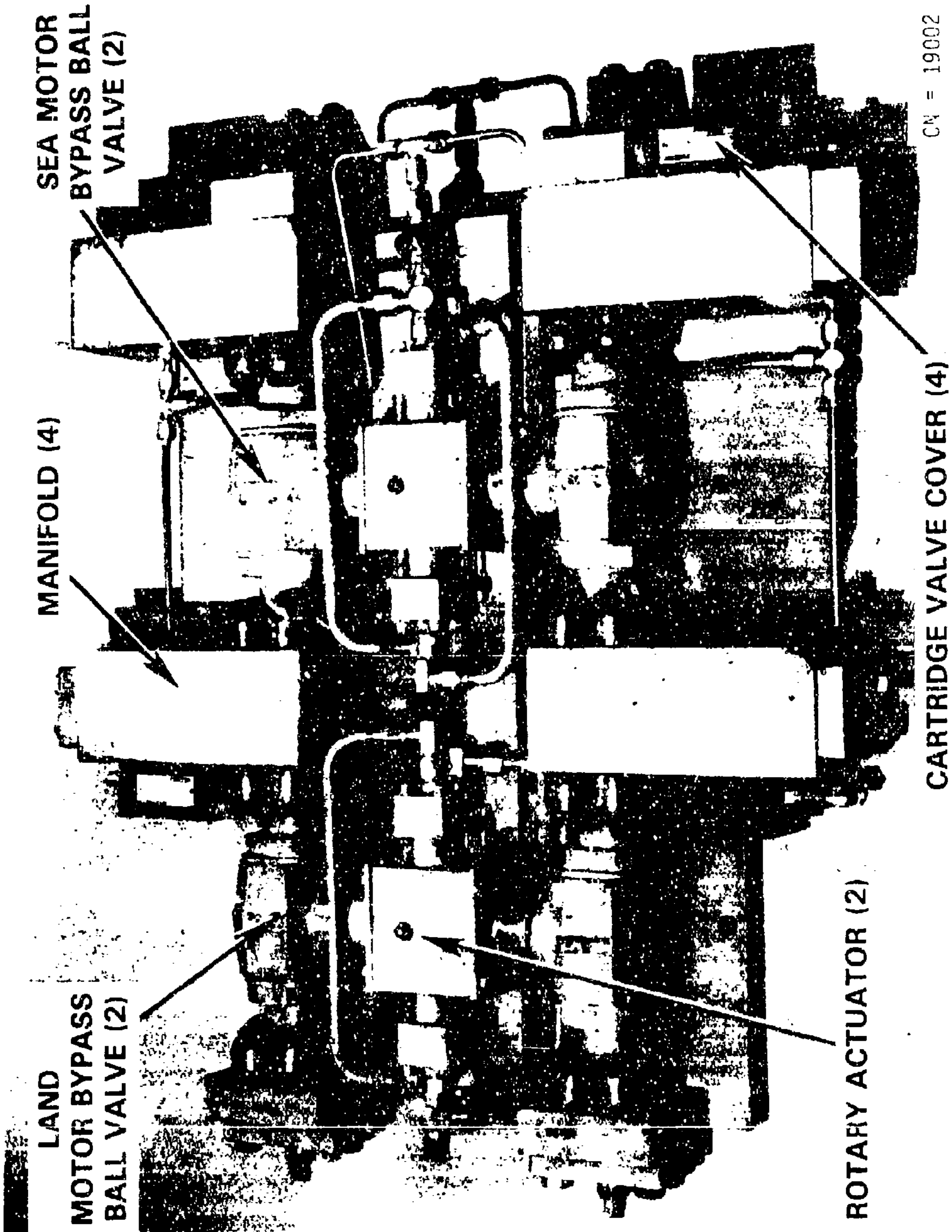

Figure 5.6.2-2 Hydrostatic Drive Mode Control Valve Assembly

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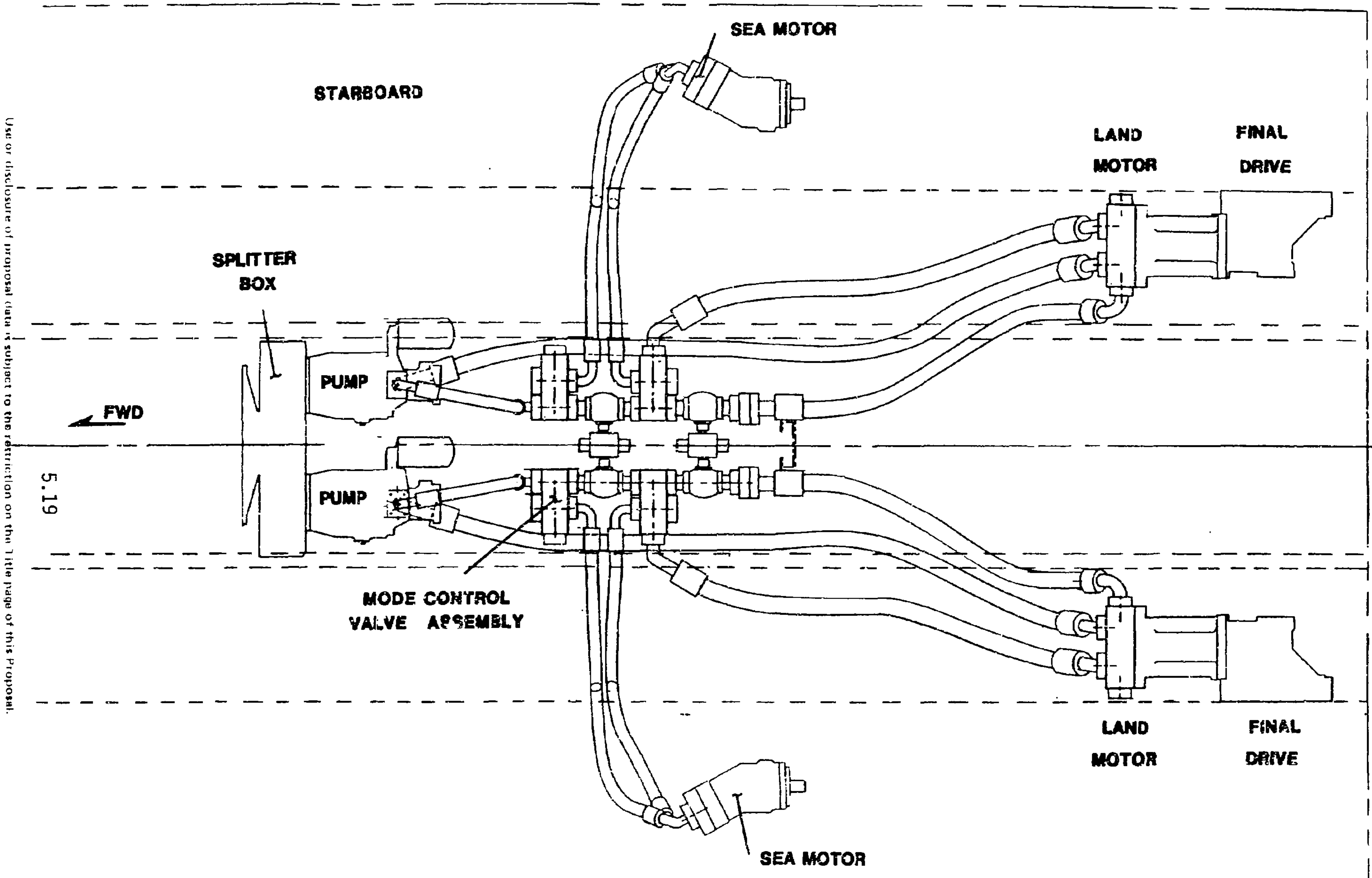

Figure 5.6.2-3 Hydrostatic Drive Hose Routing

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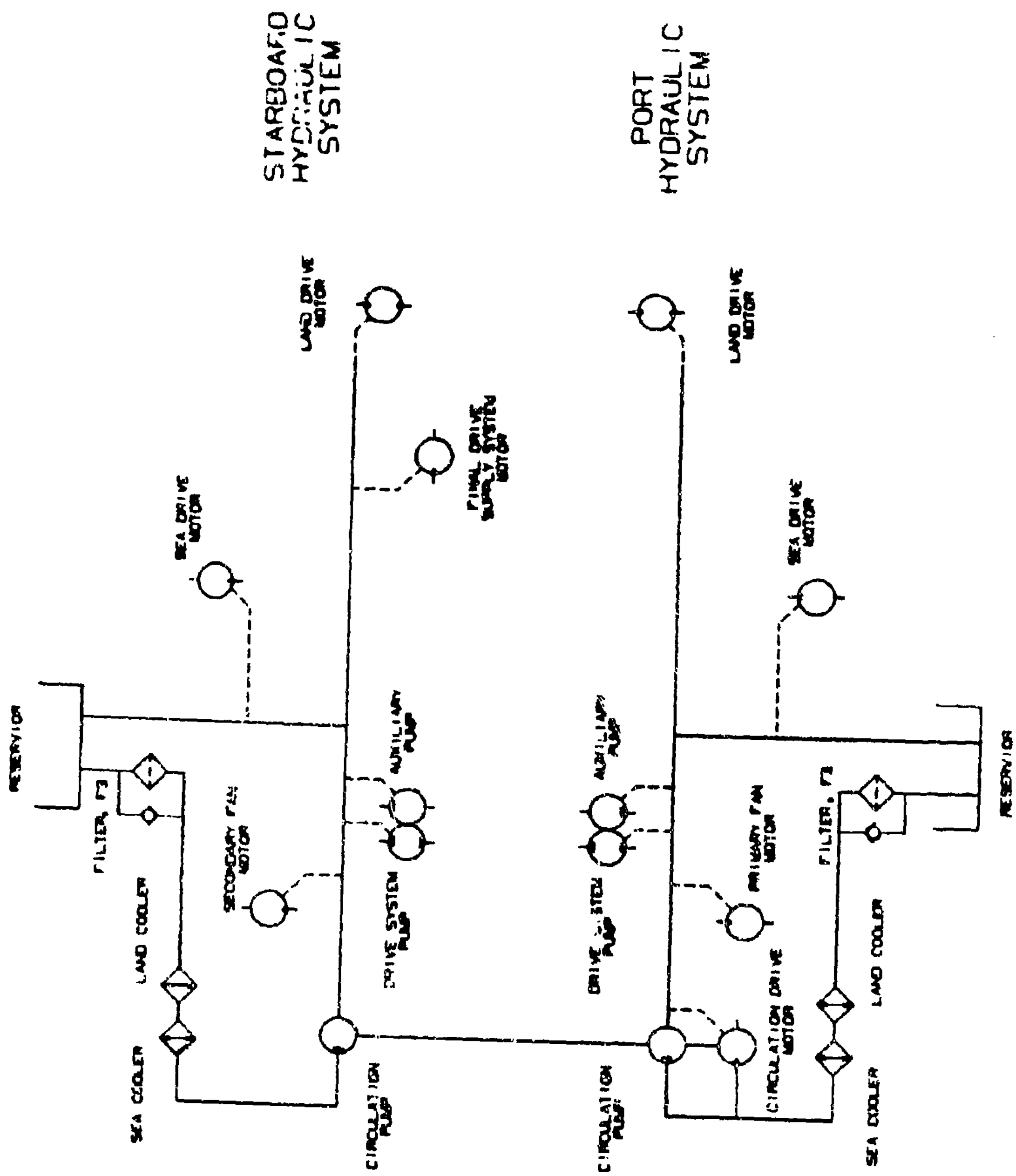

Figure 5.6.4-1 Oil Cooler Circulation System Schematic

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6.0 MARINE DRIVETRAIN

6.1 General Arrangement

The ATR marine drivetrain consists of waterjet propulsion units powered by fixed displacement hydrostatic motors. The waterjets and motors are connected with flexible element couplings. The Linde BMF 105 bent axis piston motors are driven hydrostatically by the same Linde BPV 100 pumps that drive the land transmission system. Through a series of control valves, pump flow is directed to either the land motors or the sea motors, depending upon the vehicle propulsion mode selected by the driver. Additionally, a third mode of operation, transition, permits simultaneous operation of both land and sea motors. The marine propulsors selected for use on the ATR are Dowty in-line two-stage axial flow waterjets. These units are located in the aft sponson areas and discharge through the ATR's stern. Steering of the ATR, in marine operations, is accomplished by varying the position of the waterjet discharge deflecting buckets.

6.2 Marine Propulsors

Waterborne propulsion is provided by two, aft sponson mounted, hydraulically powered waterjets. The units are manufactured by Dowty Hydraulics of England, and are presently employed in such military applications as the FV180 Combat Engineering Tractor (U.K.), the AMX10 (Fr.) and the U.S. Army Combat Support Boat.

The waterjets are two-stage, axial impeller devices. The impellers, pump reaction cases and main shafts are constructed of stainless steel. The forward and aft ends of the mainshaft (with impellers) are supported by needle bearings. The forward shaft end is additionally provided with a pair of thrust bearings to react the high forward and lower reverse thrust loads.

Details of the waterjet performance are provided in Section 8.2 of this volume. The overall waterjet envelope is presented in Figure 6.2-1. General characteristics data are as follows:

Type:	Two-stage Axial Impeller
Impeller Diameter:	12 inches (300 mm)
Steering:	Integral Scoop (bucket)
Weight:	205 pounds (93 kg)

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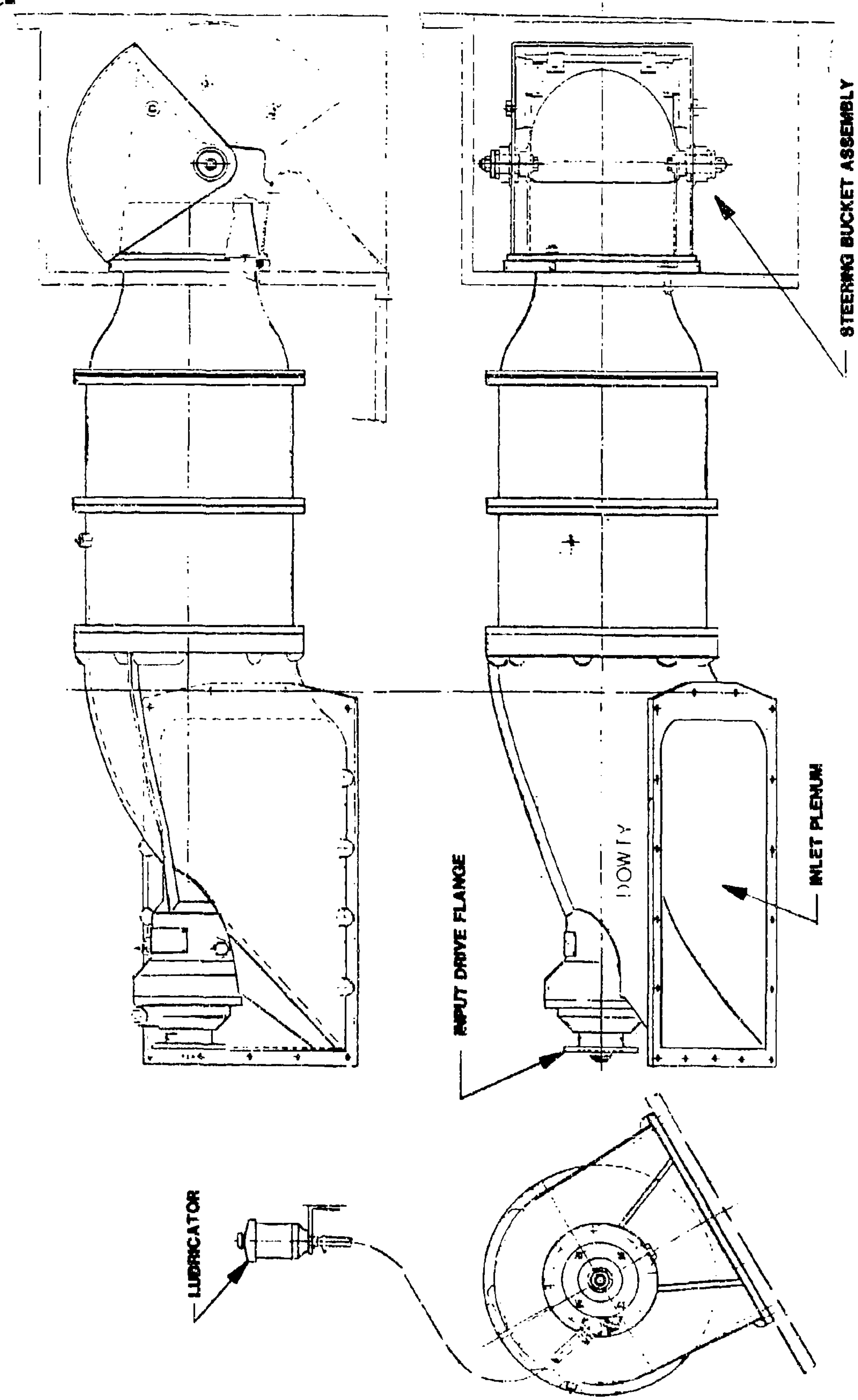

Figure 6.2-1 Dowty Waterjet

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Dowty Part Number	
Port:	8377-000-0L9
Starboard:	8377-000-0RU
Propulsive Efficiency:	21% @ 160 hp (8 mph)

6.3 Propulsor Drives

The waterjets are powered by axial piston, bent axis, fixed displacement, hydraulic motors (Linde Hydraulics Model BMF 105). Waterjet/motor coupling is achieved through Lovejoy Centaflex Couplings Model 1 Size 30. These couplings include rubber flexible elements and hub spline as follows: 23 tooth, 16/32 pitch, involute spline (fillet root side fit), 30 degree pressure angle, and a major diameter of 1.500 inch. Characteristics data regarding the drive motors are presented below.

Displacement:	6.40 in^3/rev
Working pressure range:	0-6000 psi
Max. output torque:	461 lb-ft
Max. output speed, continuous	3,000 rpm
-intermittent	3,400 rpm
Max. Flow:	94 gpm
Weight:	83.5 lbs.

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6.4 Steering Mechanism

Waterborne steering is accomplished by means of the cast aluminum rotating outlets (steering buckets or scoops) fastened to the aft portions of the waterjets at the outlet nozzles. The rotating outlets also support the reverse scoops which can be controlled to assume an infinite range of positions from forward, through neutral, to reverse thrust.

Control of the bucket position is provided by a hydraulic rotary actuator (Flow Tork Model 900) affixed to the stainless steel bucket pivot shaft. The rotary actuator additionally provides a mounting location for a rotary potentiometer which generates the feedback position signal for the microprocessor control. A block diagram of the steering control system is presented in Figure 6.4-1.

Actuator torque applied to the steering bucket is relatively low due to the balanced design of the bucket. Dowty has calculated that a maximum value of 20 lb-ft is required to achieve the full reverse bucket position. The rotary actuators are capable of providing 75 lb-ft of applied torque at a system pressure of 3000 psi. Details of the Flow Tork actuators are as follows:

Length, in:	8
Width, in:	3
Height, in:	3
Weight, lbs:	9
Displacement cubic inch:	.57
Travel, degrees:	90
Max. Pressure Rating, psi:	3000

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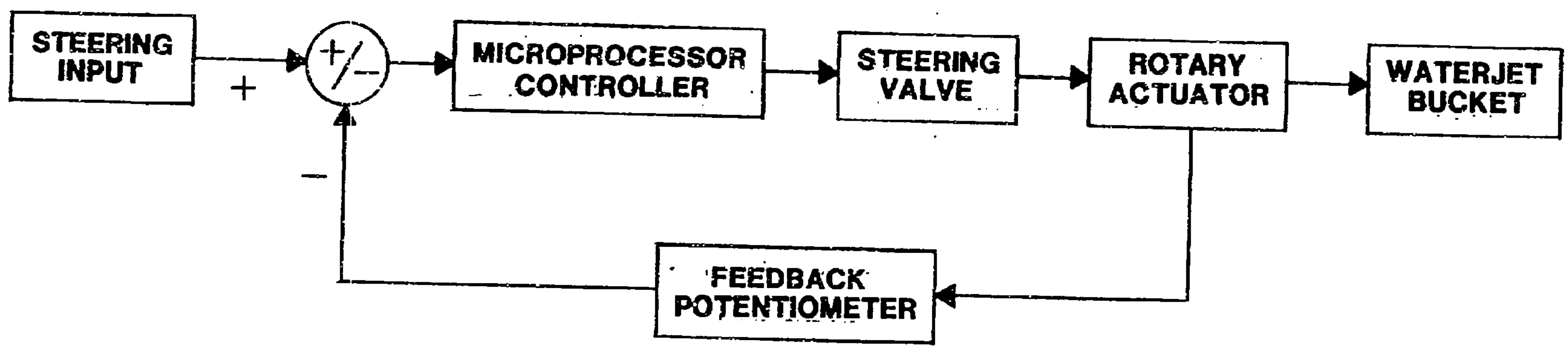

Figure 6.4-1 Waterjet Steering Control

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7.0 AUXILIARY SYSTEMS

7.1 Vehicle Electrical System

The electrical system for the ATR has been designed with respect to subsystem requirements as well as the overall operational requirements of the vehicle. The supply and distribution of electrical power has been considered for each mode of vehicle operation and over the range of specified environmental conditions. The wiring system is a two wire system, as opposed to the vehicle chassis being used as a ground return wire. The two wire system reduces the effects of ground loops and electromagnetic interference. Reliability and maintainability have been given a high degree of consideration. A diagram depicting the electrical elements of the ATR is shown in Figure 7.1-1.

7.1.1 Power Requirements

A study has been conducted to determine the power requirements under each mode of vehicle operation. The power requirements for land operation, transition (land/sea), and sea modes are depicted in Table 7.1.1-1.

7.1.2 Electrical System Components

Each of the system components are discussed in detail in the following subsections.

7.1.2.1 Automotive Batteries

The ATR is equipped with four, maintenance-free, commercial automotive batteries. Each of these batteries produce over 600 amps of engine cranking current at 0 degrees F.

The batteries are connected in series parallel to provide 24 VDC at an approximate rating of 300 ampere-hours.

These batteries have a greater capacity and are smaller and lighter than the 6TN military battery.

7.1.2.2 Alternator

The ATR is equipped with a Niehoff Co., Division of TRW, alternator with an output capacity of 300 amps at 24 VDC. Voltage regulation is provided by a modular unit mounted to the alternator frame.

7.1.2.3 Master Power Switch

The master power switch is located on the drivers control panel and provides a means of disconnecting the batteries from the vehicle ground.

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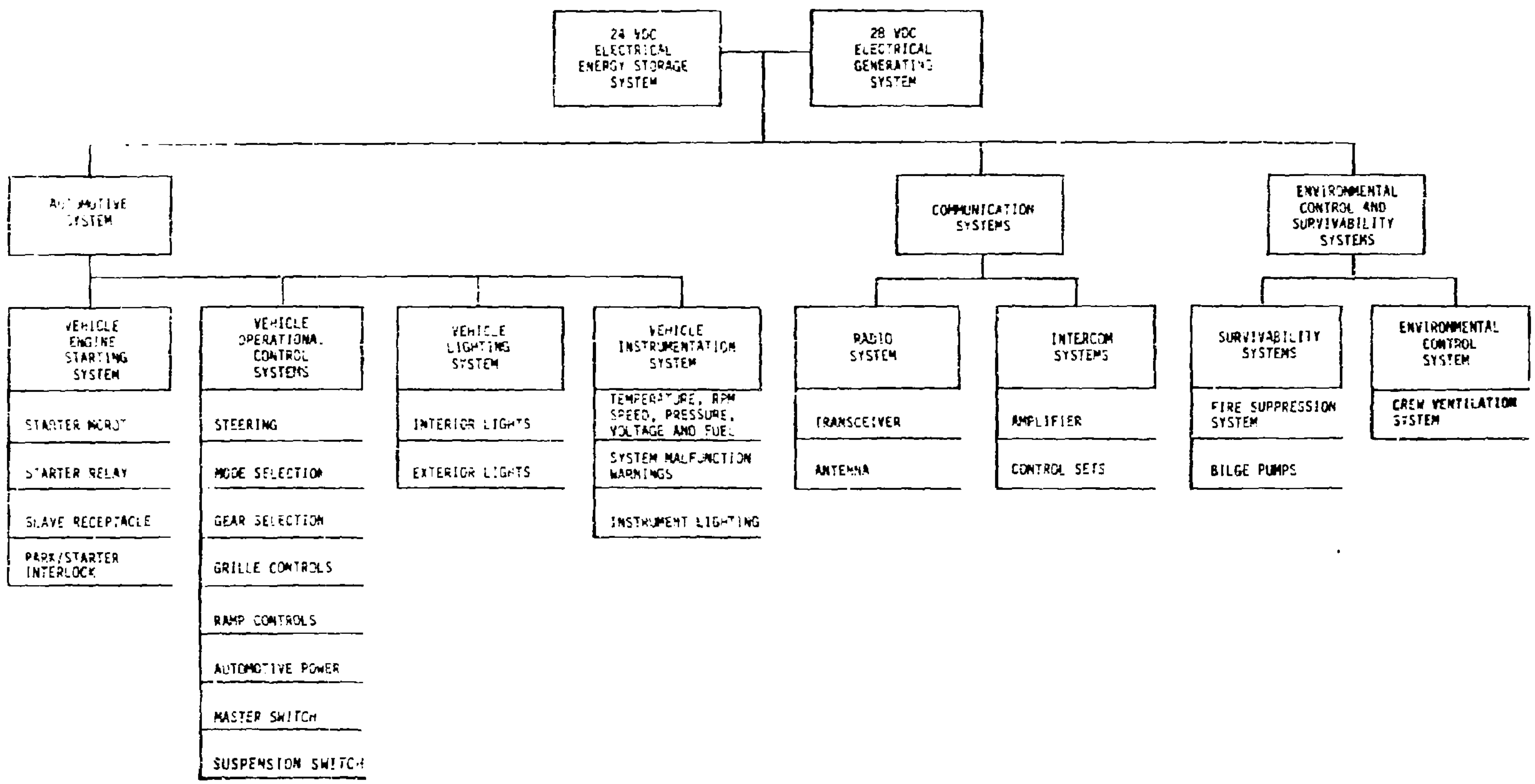

Figure 7.1-1 Element Diagram of ATR Vehicle

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Table 7.1.1-1. ATR Electrical Systems Requirements

APPLICATION	LAND	LAND/SEA	SEA	INTERMITTENT
	AMPERES AT 28 VDC			
Automotive Systems				
Starter Motor	--	--	--	500.0
Eng. Stop Solenoid	--	--	--	1.0
Master Relay	1.0	1.0	1.0	--
Alternator	5.0	5.0	5.0	--
Horn	--	--	--	4.0
Starter Solenoid	--	--	--	60.0
Eng. Instruments	1.5	1.5	1.5	--
Driver's Display Panel	1.5	1.5	1.5	--
Panel Lights	3.0	3.0	3.0	--
Exterior Lights	10.0	14.0	14.0	--
Interior Lights	3.0	3.0	3.0	--
Master Fault Panel	1.0	1.0	1.0	5.0
Driver's Control Panel	1.5	1.5	1.5	--
Auxiliary Systems				
Bilge Pumps Electric	--	110.0	110.0	240.0
Bilge Pumps Hydraulic	--	2.0	2.0	--
Fire Suppression	1.0	1.0	1.0	10.0
Ramp Control	--	--	--	1.5
Grill Control	2.0	2.0	2.0	--
Vent Control	2.0	2.0	2.0	--
Seawater Pump	--	19.0	19.0	63.0
Communications				
Radio (PRC-68)	1.0	1.0	1.0	11.0
Intercom (VIC/1)	2.0	2.0	2.0	--
Suspension System				
Suspension Controls	12.8	12.8	12.8	--
Hydrostatic Drive				
Computer	14.0	14.0	14.0	--
Control Valves	15.0	15.0	15.0	--
Control Relays	8.0	8.0	8.0	
Charging System				
Batteries (4)	40.0	40.0	40.0	100

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Table 7.1.1-1. ATR Electrical Systems Requirements (continued)

APPLICATION	LAND	LAND/SEA	SEA	INTERMITTENT
		AMPERES AT 28 VDC		
Turret Systems				
Elevation Drive	10.0	10.0	10.0	50.0
Azimuth Drive	10.0	10.0	10.0	50.0
Sights/FCS	5.0	5.0	5.0	--
TOTAL	240.3	303.3	304.3	1095.5

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A semiconductor diode is connected in series with the coil of the master power relay to provide reverse polarity protection. This protects the vehicle and equipment from inadvertent reverse connection of the batteries or the slave receptacle.

7.1.2.4 Slave Receptacle

The ATR is equipped with a military NATO type receptacle. This receptacle is coaxial in construction and eliminate the possibility of reverse polarity when the mating plug is inserted.

7.1.2.5 Engine Electrical

The engine is wired with several sending units such as, coolant temperature, oil pressure, and oil temperature. There is also an electrical solenoid for shutting off the engine fuel. All engine wiring, such as sending units, starter motor, and alternator are routed through an engine quick disconnect bracket. This greatly reduces the time required to remove and install the engine. In addition to the aforementioned items, several temperature and pressure switches are incorporated and connected to the master warning system.

7.1.2.6 Power Distribution

The main vehicle wires from the batteries and alternator are connected in a Power Distribution Panel. The Power Distribution Panel contains the master power relays that connect and disconnect the battery systems from the vehicle chassis. Also, the panel contains the main power busses and circuit breakers that distribute power throughout the system. This panel also supplies electrical power directly to the vehicle control system. This system reduces the time required to remove or install cables, when troubleshooting or performing maintenance.

7.1.2.7 Master Warning System

A master warning light is located on the Driver's Display Panel and on the Engineer's Panel. When a fault occurs, these lights will flash and a audio tone will be heard in the CVC helmets. The test engineer in the vehicle will observe the fault message on the Terra computer display to determine its source.

7.1.2.8 Digital Speedometer

The driver's instrument panel contains a digital speedometer which is readable in full sunlight. The two digit readout of speed in miles

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per hour which is displayed to the driver and is updated approximately every three seconds.

7.1.2.9 Tachometer

The engine speed is displayed to the driver by an electronic analog meter. The sending unit for this meter is located on the front of the engine.

7.1.2.10 Odometer

An odometer records the total distance traveled by the vehicle. The electrical signals for determining the distance traveled are generated by the speedometer sending unit and processed in the SwRI computer.

7.1.2.11 Elapsed Time Meter

An elapsed time meter is connected to the main power bus of the vehicle. This meter records the total time that electrical power is applied to the vehicle through the master power relay.

7.1.2.12 Battery Voltage-High/Low Detector

A battery low and high voltage detector is provided and will activate the master warning system in the event of a failure.

7.1.2.13 Circuit Breakers

The ATR is equipped with thermal resetting circuit breakers for protection of the vehicle electrical systems. If a circuit is shorted or heavily overloaded the circuit breakers interrupt the circuit for a short period of time and then automatically reset themselves. If the overload is temporary the circuit will be restored to normal operation. If the circuit remains shorted or overloaded the circuit breaker will continue to interrupt the circuit.

7.1.2.14 Transmission Gear Selector Switch

The transmission gear selector switch contains a series of mechanically interlocked, illuminated pushbuttons. When any of these buttons are depressed a corresponding signal is routed to the SwRI computer. In addition, tne "park" pushbutton is electrically interlocked with the starter switch to preclude starting the vehicle in any "gear" position other than park. The park position ensures that the service brakes are applied when the vehicle is started.

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7.1.2.15 Mode Selector Switch

The mode selector switch is a three position locking toggle switch. This switch selects the Land, Land/Sea (Transition), and Sea modes of operation.

7.1.2.16 Rear Ramp

Raising and lowering of the rear ramp is controlled by a toggle switch that operates momentarily in the up and down directions with a spring return to the "off" position. This switch is located on the driver's control panel. A safety limit switch is incorporated on the ramp latch mechnism to indicate when the ramp is closed and locked. Operating in conjunction with the ramp switch is a momentary toggle switch that must be placed and maintained in the operate position while the ramp is being raised or lowered.

7.1.2.17 Auxiliary Lights

Although auxiliary lights (navigation, search and anchor) are not provided on the ATR, a switch has been provided to facilitate the addition of auxiliary lights if they are required in the future.

7.1.2.18 Crew Lights

Dome and map lights are provided for the crew to operate the vehicle during periods of reduced visibility.

7.1.2.19 Horn

A horn is provided on the vehicle for warning surrounding personnel of vehicle maneuvering or the raising or lowering of the ramp.

The horn button is located on the Drivers Display Panel.

7.1.2.20 Automotiv Lights

Night driving is aided by the use of two waterproof Halogen headlights mounted on the upper glacis plate above the water line.

7.1.2.21 Intercom System

Communication between crew members is maintained through a VIC/1 intercommunication system. This system consists of a 1780 audio amplifier, headsets and control units. The system is integrated with the transceiver and the master warning system.

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7.1.2.22 Radio System

The vehicle electrical system is designed to accommodate a radio system such as an AN/VRC-12 which consists of receiver-transmitter RT-246/VRC, an antenna, 1729/VRC, and control units C-2298/VRC. An alternate system would consist of a PRC/68 and vehicle applique.

7.1.2.23 Fire Suppression System

The wiring harnesses used for interconnecting the fire suppression system are shielded. This shielding protects the system from false triggers caused by electromagnetic and radio frequency interference.

7.1.2.24 Fuel Level Monitors

The fuel level is monitored by float-type sensors mounted in each tank which detect the fuel level and display the level detected on a gauge on the drivers control panel.

7.2 Vehicle Hydraulic System

This section addresses the auxiliary hydraulic system. The hydrostatic drivetrain hydraulic system is presented in Sections 5.5 and 5.6.

The auxiliary hydraulic system provides the fluid power required for:

(1) the power plant cooling system fan drives
(2) cooling system grille actuators
(3) seaborne steering system
(4) bilge pumps
(5) the hydropneumatic suspension system
(6) rear ramp actuator
(7) rear ramp latch actuator
(8) hydrostatic drive mode control
(9) final drive supply system drive
(10) final drive oil cooler circulation system drive

The auxiliary systems are powered by two, Rexroth A10V25, variable displacement, axial piston pumps. The pumps are tandemly mounted on the aft drive pads of the primary hydrostatic drive pumps. The auxiliary system is divided by function as shown in Figure 7.2-1. The starboard side pump only powers the primary cooling system drive which, in effect, establishes a closed system as there is no exchange volume. The port side pump services the remaining subsystems (suspension, ramp, grilles, secondary cooling system fan drive, etc.).

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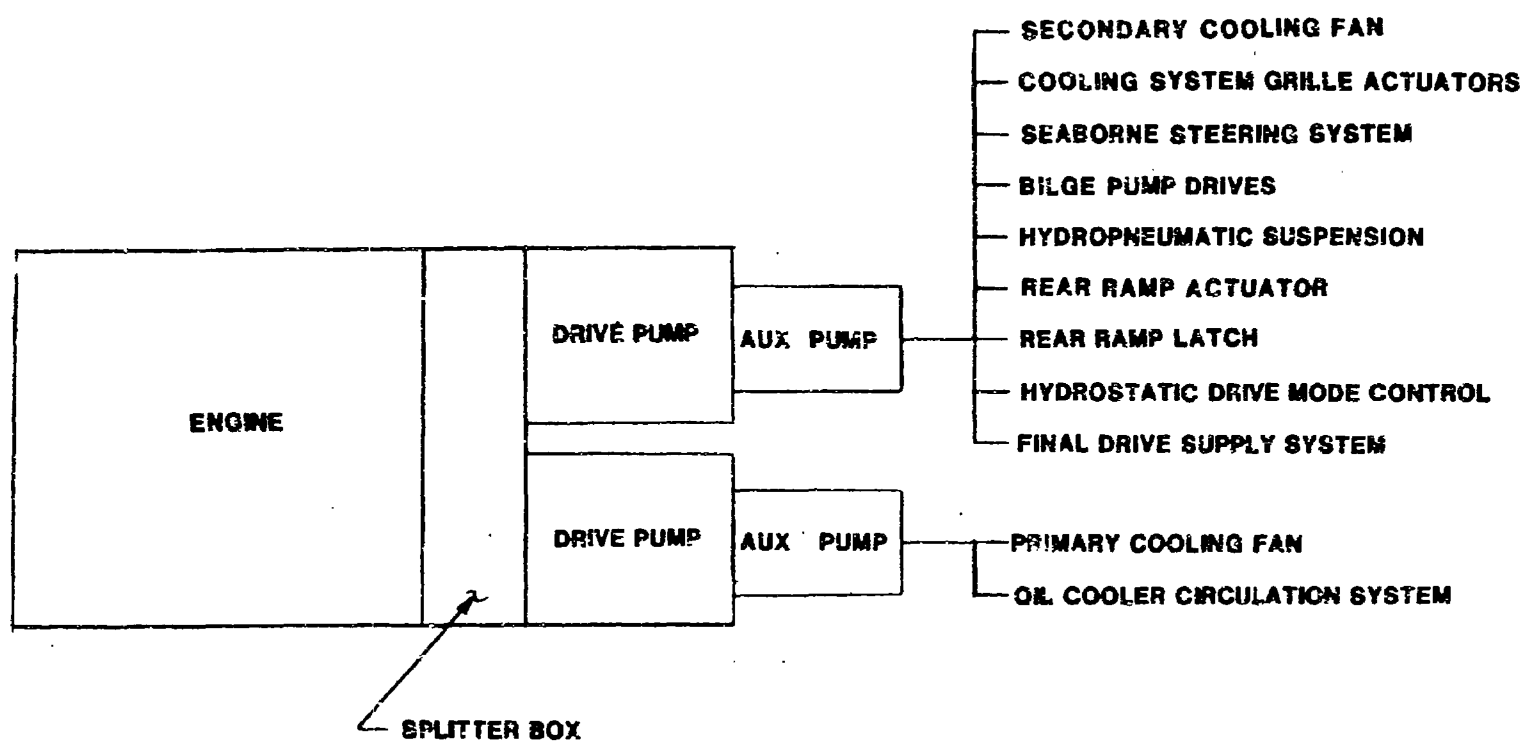

Figure 7.2-1 Auxiliary Hydraulic System Functions

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The hydraulic requirements of the subsystems are shown in Table 7.2-1. Characteristic pump data are presented in Table 7.2-2.

Table 7.2-1 ATR Auxiliary Hydraulic System Requirements

Application	Continuous Flow Rate	Continuous Pressure	Intermittent Flow Rate	Intermittent Pressure
Primary Cooling Fan	11.0	2000		
Secondary Cooling Fan	11.0	3000		
Grille Actuators			1.0	3000
Waterjet Steer			.3	3000
Bilge Pump Drives	6.0*	1400		
Hydropneumatic Suspension	1.0-2.0	2000	4.5	2000
Ramp Actuator			1.0	3000
Ramp Latch			1.2	3000
Drive Mode Control			1.0	3000
Final Drive Supply System	1.5	3000		
Oil Cooler Circ. System	1.9	2000		

* Sea and Transition Mode Only

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Table 7.2-2 Rexroth A10V25 Pump Specifications

Displacement, cu.in./rev.	1.52
Flow at 1000 rpm, gpm	6.60
Maximum Drive Speed, rpm	3250
Rated Pressure (continuous), psi	3000
Peak Pressure (Intermittent), psi	3625
Minimum Inlet Pressure, psi	-3.0
Maximum Inlet Pressure, psi	215
Weight (dry), lbs.	24.2
Filling Capacity of Case, cu.in.	43
Overall Length, in.	7.68
Overall Diameter, in.	6.30
Mounting	SAE B PAD
	SAE B SPLINE

Two, twenty-five gallon capacity, aluminum reservoirs are positioned in the port and starboard sponsons directly above the waterjet drive motors. The port side reservoir supplies hydraulic fluid to the port side primary hydrostatic drivetrain pump, and its tandemly mounted auxiliary pump. The starboard side reservoir, in a like manner, supplies only the starboard side pumps. The split-system arrangement reduces the possibility of total system fluid contamination in the event of a major hydraulic component failure.

The reservoirs are provided with sight gauges for visual inspection of fluid levels and also incorporate solid-state level sensors to provide low fluid level indications to the master warning system. The fill ports are located such that hydraulic fluid cannot be directly poured into the reservoirs, any required oil must be pumped in. This arrangement reduces the possibility of accidental contaimination or use of improper fluids.

The primary cooling fan system schematic is shown in Figure 7.2-2. This system drives only the primary cooling fan and the final drive

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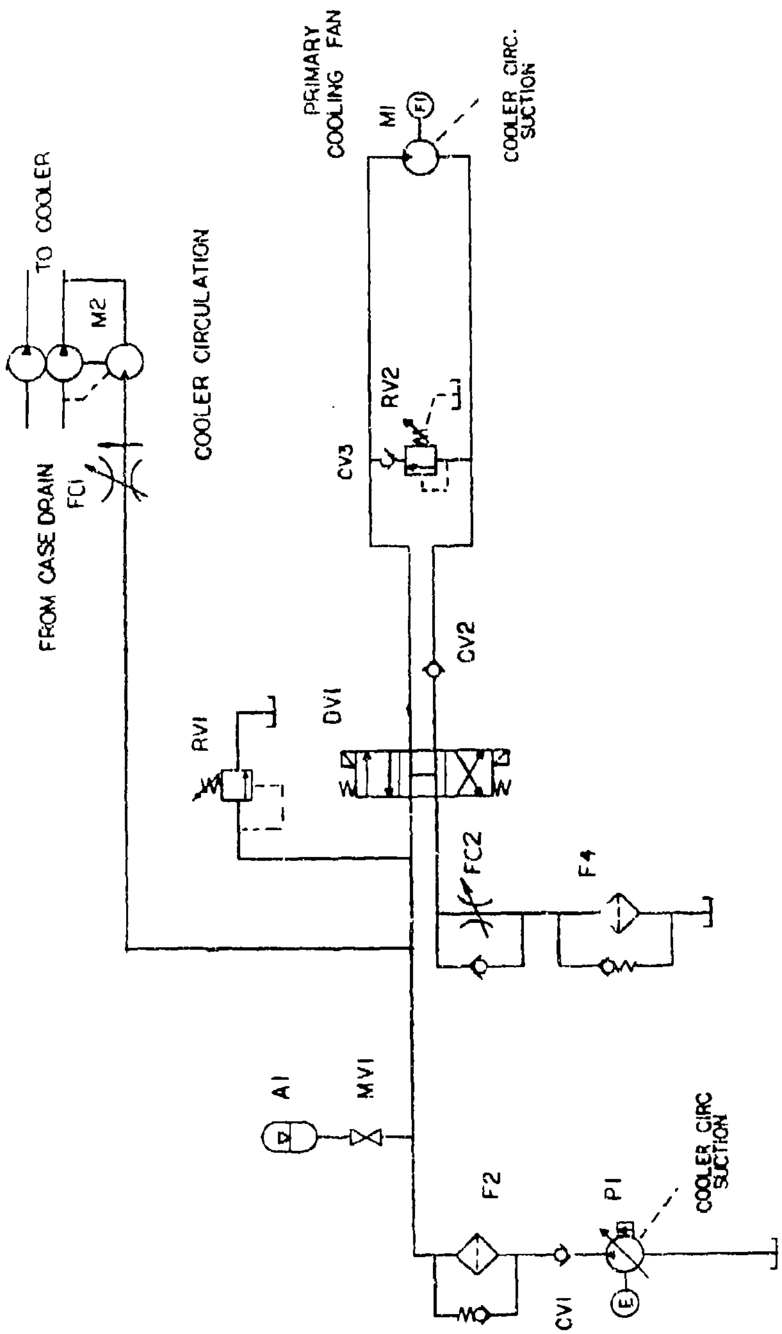

Figure 7.2-2 Primary Cooling Fan Hydraulic System Schematic

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oil cooler circulation system. For description purposes the circuit will be divided into three subsystems: 1) supply and return, 2) fan drive, and 3) cooler circulation drive.

The components included in the supply and return group are: one accumulator (A1), one check valve (CV1), two filters (F2, F4), one flow control (FC2), one manual valve (MV1), one pump (P1) and one relief valve (RV1). The component operation is as follows:

1) Accumulator (A1) contains a 1000 psi precharge. It acts to dampen shocks due to opening and closing control valves and to attenuate pressure pulses generated by the piston pump.
2) Check valve (CV1) is included to prevent system drainage should pump discharge hose require removal for maintenance.
3) Filter (F2) cleans the pump discharge oil to 10 microns absolute level. The filter housing incorporates a 50 psi bypass valve and an electrical pressure differential indicator. This indicator signals the Engineer's Panel and the master warning system of impending filter bypass. This notifies the operator when it is time for element replacement.
4) Filter (F4) cleans the system return line flow to the reservoir. This filter is identical to F2 and is also reservoir mounted.
5) Flow Control (FC2) is used as a manual shut off valve for the system return line. During normal vehicle operation this valve must remain fully opened which permits fluid to enter and exit the reservoir freely. Exit flow is required during the fan braking mode. During system maintenance such as F2 element replacement FC2 should be closed to prevent reservoir drainage.
6) Manual valve (MV1) is used as a shut-off valve to the accumulator. During system servicing the valve should be closed.
7) Pump (P1) is a Rexroth A10V25 which was previously described in this section. The pressure setting of this pump is 2000 psi.

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8) Relief valve (RV1) provides system overpressure protection. The valve has been set to open at 3200 psi.

The components dedicated to the fan drive group of this system are: two check valves (CV2, CV3), one directional control valve (DV1), one motor (M1) and one relief valve (RV2). During land operation this fan operates continuously. The component operation is as follows:

1) Check valve (CV2) prevents reverse flow to the fan when the directional valve is shifted to the fan stop position.
2) Check valve (CV3) prevents supply pressure directed to fan motor from leaking through relief valve (RV2).
3) Directional control valve (DV1) is a three position four-way valve. The three positions correspond to: Solenoid B for fan on, system bypass, and Solenoid A for fan stop.
4) Motor (M1) is the fan drive motor which is an integral part of the cooling fan. The fan speed is regulated by the pump pressure compensator. The desired maximum fan RPM is 4150 RPM.
5) Relief valve (RV2) determines the decleration rate of the cooling fan when the fan is stopped. This relief valve has been set for 500 psi.

The final drive oil cooler circulation system drive group, of this system, are: one flow control (FC1) one motor (M2). A description of the operation of these components is as follows:

1) Flow control (FC1) is a pressure compensated flow control. This valve has been set to deliver a flow of 1.9 GPM.
2) Motor (M2) drives a tandem gear pump used for the oil cooler circulation. To generate the proper flow in the cooler the motor speed must be approximately 2800 RPM.

The auxiliary hydraulic system schematic is shown in Figure 7.2-3. This system drives all auxiliary hydraulic functions with the

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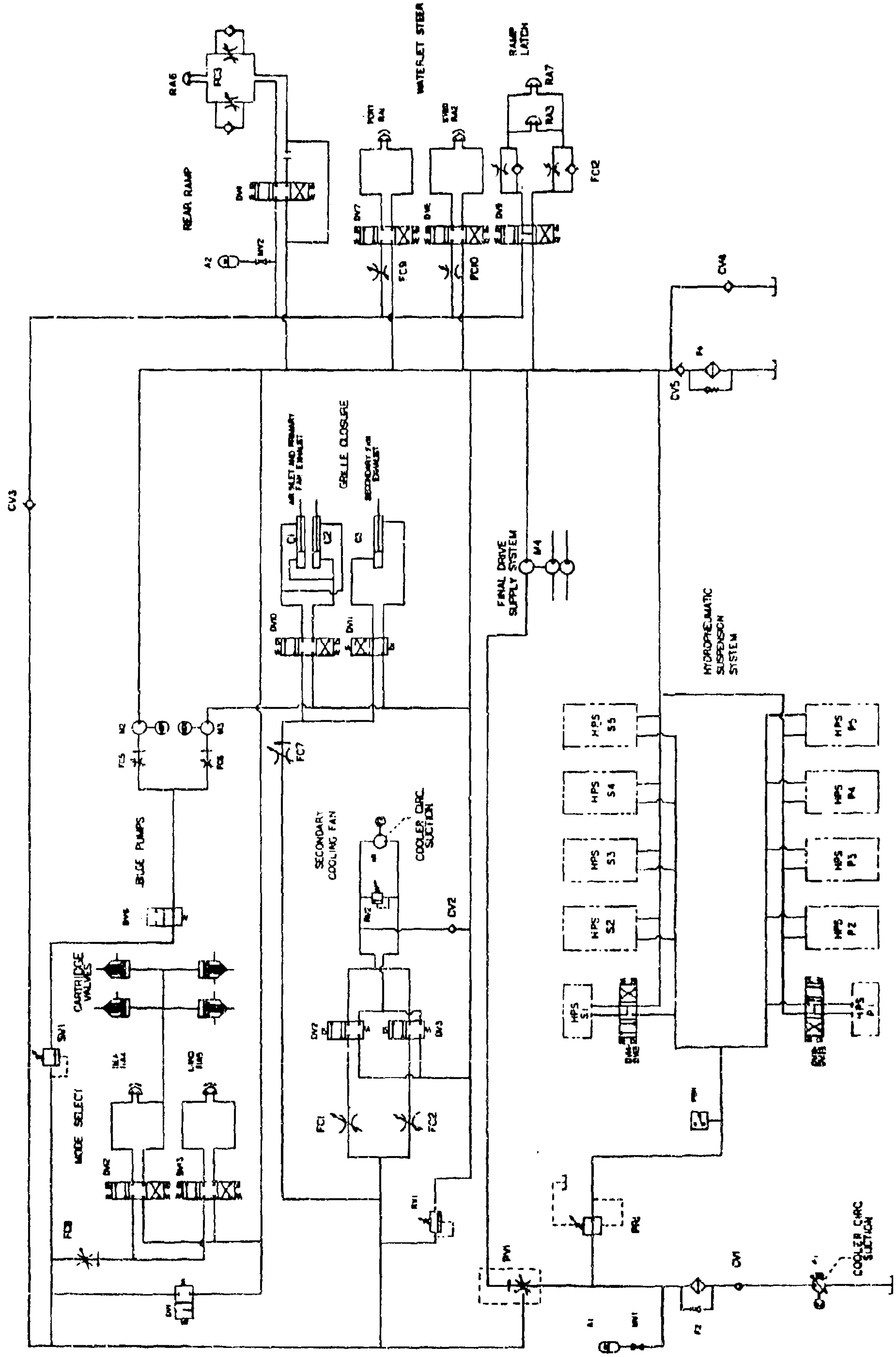

Figure 7.2-3 Auxiliary Hydraulic System Schematic

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exception of the primary fan and the final drive cooler circulation drive. For description purposes, the system will be divided into subsystems by function: 1) supply and return, 2) secondary cooling fan, 3) grille actuators, 4) seaborne steering, 5) bilge pump driver, 6) hydropneumatic suspension, 7) rear ramp actuator, 8) rear ramp latch, 9) hydrostatic drive mode control, and 10) final drive supply system drive.

The components of the supply and return subsystem of the auxiliary hydraulic system are similar to that of the previously described primary cooling fan system. The components included in the supply and return subsystem are: one accumulator (A1), three check valves (CV1, CV4, CV5), two filters (F2, F4), one directional control valve (DV1), one manual valve (MV1), one pump (P1), and one relief valve (RV1). The components that differ from the primary cooling fan system are discussed below:

1) Check valve (CV4) has a cracking pressure of 3 psi. This valve permits backflow from the reservoir to the return line. Backflow capability is needed to accommodate the suspension system during rebound.
2) Check valve (CV5) has a cracking pressure of 15 psi. The valve protects the filter (F4) from reverse flow surges.
3) Directional valve (DVI) is a system bypass valve. In the normal position the valve is closed and all systems are functional. With DVI energized the flow downstream of the priority valve (PV1) is relieved to tank.

The secondary cooling fan circuit is composed of one check valve (CV2), two directional control valves (DV2, DV3) two flow controls (FC1, FC2), one motor (M1) and one relief valve (RV2). This system provides the secondary cooling fan with the capability of operating at four speeds: off, low, medium, and high. In the land mode, the secondary fan is used to supplement the primary cooling fan. This supplement only occurs when the microcomputer identifies a high temperature condition of the engine coolant or hydraulic reservoirs. A second function of the auxiliary cooling fan is to ventilate the engine compartment and vehicle during the sea and transition modes of operation. In these modes the low speed control valve is activated. The function of the components in secondary cooling fan circuit are as follows:

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1) Check valve (CV1) is supplied to prevent cavitation of the fan motor during the fan braking mode.

2) Directional control valve (DV2) controls the 4 GPM flow to the secondary cooling fan. This valve is activated for low and high speed fan operation.

3) Directional control valve (DV3) controls the 8 GPM flow to the secondary cooling fan. This valve is activated for medium and high speed fan operation.

4) Flow control (FC1) is a non-pressure compensated flow control. This valve has been set for a 4 GPM flow with a 2000 psi pressure drop through the valve.

5) Flow control (FC2) is a non-pressure compensated flow control. This valve has been set for a 8 GPM flow with 1000 psi pressure drop through the valve.

6) Motor (M1) is an integral part of the cooling fan. This motor is a bent axis piston type with a .60 cu. in./rev. displacement. The motor is manufactured by Sundstrand and has the following characteristics:

Model	F11-10
Max. operating pressure	6000 psi
Max. continuous speed	6800 RPM
Weight	17 LBS

7) Relief valve (RV2) is used to set the decel(ration rate of the secondary fan motor.

The ATR is equipped with louver type cooling fan inlet and exhaust grilles. The grille closure system is divided into two controlled sub-systems: air inlet and primary fan exhaust; and secondary fan exhaust. Actuation of the grille closure requires: three hydraulic cylinders (C1, C2, C3), two directional control valves (DV10, DV11) and one flow control (FC7). The function of these components is as follows:

1) Hydraulic cylinders (C1, C2) actuate the air inlet and primary fan exhaust grille closure. These are double acting cylinders.

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2) Hydraulic cylinder (C3) actuate the secondary fan exhaust grille closure. This is a double acting cylinder.
3) Directional valve (DV10) directs the flow to air inlet and primary fan exhaust grille actuator. This valve is a four-way, 3-position solenoid valve. In the deactivated position, the cylinders are locked in place. With solenoid A energized, the grille opens. With solenoid B energized the grille closes.
4) Directional valve (DV11) directs the flow to the secondary fan exhaust actuator. This valve is a four-way, two-position solenoid valve. In the deactivated position, the grille is in the closed position. In the solenoid B activated position, the grille is opened.
5) Flow control (FC3) controls the speed that the grilles open and close. The secondary cooling fan exhaust grille opens and closes half as fast as the air inlet and primary fan exhaust grille.

The waterborne steering system controls each of the waterjet outlet diverter scoops called the steering buckets. Rotary actuators are mounted inside the hull and are connected to the buckets using a watertight linkage connection. To control the buckets, the following hydraulic components required are: two directional control valves (DV7, DV8), two flow controls (FC9, FC10) and two rotary actuators (RA1, RA2). The components operate as follows:

1) Directional control valve (DV7) controls the flow to the port waterjet steering bucket actuator. This valve is a four-way, three position solenoid control valve. The deactivated position locks the buckets in place. The solenoid A position rotates the buckets in the counter-clockwise or forward direction. The solenoid B position rotates the buckets in the clockwise or reverse direction.
2) Direction control valve (DV8) controls the flow to the starboard waterjet steering bucket actuator. This valve is identical to DV7. However, the solenoid A position rotates the buckets in the clockwise or forward direction and the

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solenoid B position rotates the bucket in the counterclockwise or reverse direction.

3) Flow control (FC9) is a non-pressure compensated flow control. This valve controls the flow rate to the port rotary actuator. The speed of actuator rotation is 45 deg/sec.
4) Flow control (FC10) is identical to (FC9).
5) Rotary actuator (RA1) rotates the port waterjet steering bucket. This is a rack and pinion type unit capable of 900 in-lbs of torque output.
6) Rotary actuator (RA2) rotates the starboard waterjet steering bucket.

The hydraulic bile pump drive subsystem is composed of one directional valve (DV6), two flow controls (FC5, FC6), and two motors (M2, M3). The hydraulic bilge pumps are required to operate continuously in the sea and transition modes of vehicle operation. The operation of the hydraulic components is as follows:

1) Directional valve (DV6) is a two-way, two-position solenoid control valve. The de-energized position corresponds to hydraulic bilge pumps on. The solenoid B position shuts the bilge pumps off.
2) Flow control (FC5) is pressure compensated. This valve controls the flow rate to the aft hydraulic bilge pump.
3) Flow control (FC6) is identical to (FC5). This valve controls the flow rate to the forward hydraulic bilge pump.
4) Motor (M2) is an integral part of the aft hydraulic bilge pump
5) Motor (M3) is an integral part of the forward hydraulic bilge pump.

The hydropneumatic suspension system requires a hydrualic supply system for actuation (extension, retraction) and for leakage makeup oil. Supply pressure is limited to 2000 psi. The hydraulic components required for

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this system are: ten directional control valves (DV14-DV23), one pressure regulator (PR1), and one pressure switch (PS11). The description and function of these components is as follows:

1) Directional control valves (DV14-DV23) control the extension and retraction of each of the ten suspension units. The valves are four-way, three-position solenoid control valves. In the de-energized, position supply system pressure is blocked from the suspension unit. This position permits the suspension to collapse due to internal leakages within the suspension unit. The solenid A position extends the suspension and the solenoid B position causes it to retract.
2) Pressure switch (PS11) monitors the suspension system supply pressure. This switch closes at pressures below 2000 psi signaling the master warning system. A detected fault would indicate a supply system problem, a broken line, or a suspension unit failure causing major leakages.
3) Pressure reducer (PR1) regulates the suspension supply system pressure to 2000 psi. This valve was required to minimize the HPSU internal leakage.

The rear ramp actuator circuit is composed of: one accumulator (A2), one check valve (CV3), one directional control valve (DV4), one flow control (FC3), one manual valve (MV2), and one rotary actuator (RA6). The function and operation of the car ramp actuator hydraulic component is as follows:

1) Accumulator (A2) stores hydraulic energy for ramp closure without engine power. This accumulator has a precharge of 500 psi and is capable of raising the ramp one time.
2) Check valve (CV3) blocks back flow from the accumulator to the forward hydraulic subsystems. This prevents leakage from occurring through the forward valves which would reduce the ramp raising capabilities of the accumulator.
3) Directional control valve (DV4) controls the flow to the ramp actuator. This valve is a four-way, three-position

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solenoid valve. The de-energize position holds the actuator position. This gives the operator the capability of temporarily holding the ramp in mid-position. However, it is not intended that ramp actuator hold the ramp in mid-position during loading of cargo. This may cause failure of the actuator or lifing chain. The solenoid A position causes the ramp to raise, and the solenoid B position will lower the ramp.

4) Flow control (FC3) is a sandwich type non-pressure compensated flow control valve. The valve meters the flow from the actuator ports (meter out A&B). The valve has separate adjustments for ramp up and ramp down flow settings.
5) Manual valve (MV2) is a shut-off valve for (A2). This valve should be opened for charging of the accumulator from the auxiliary hydraulic system. Once charged the valve should then be closed in order to hold the charge until needed.
6) Rotary actuator (RA6) is a double rack and piston actuator. It is capable of 7500 in-lbs of torque. A chain sprocket has been directly mounted to the output shaft of the actuator.

The rear ramp latching circuit is composed of: one directional control valve (DV9), one flow control (FC12), and two rotary actuators (RA3, RA7). The operation and function of the ramp latching hydraulic components is as follows:

1) Directional control valve (DV9) controls the flow to the ramp latch actuators. It is a three-position, four-way solenoid valve. the de-energized position blocks system pressure and permits both actuator ports to vent the tank. In this position, a manual override on the latching linkage can be used for latching or unlatching. The solenoid A position unlatches the ramp while the solenoid B position latches it.

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2) Flow control (FC12) is sandwich-type, non-pressure compensated flow control valve. The valve meters the flow to the acturator ports (meter in A&B). The valve has separately adjustable latching and unlatching settings.
3) Rotary actuator (RA3, RA7), operates the port and starboard rear ramp latching linkages. Due to the high latching forces generated by the overcenter mechanism an additional shaft bearing has been installed on the output shaft.

The hydrostatic drive mode control circuit consists of four cartridge valves, two directional control valves (DV12, DV13), one flow control (FC8), and two rotary actuators (RA4, RA5). This system is used for the operation of the hydrostatic drive bypass valves. It also is used for the isolation of the sea motors from the drive system pressure during land operations. The operation and function of the mode control components is as follows:

1) Directional control valve (DV12) is used to control the sea motor bypass valve actuator. In the de-energized position the rotary actuator is locked in the position it was last in. During vehicle operation this position is not normally used. The only time these valves operate in the center position is during an electrical failure to the valves. Also, pilot pressure is applied to the cartridge valves causing the working ports to the sea motor to be blocked from system pressure. In the solenoid B position, the bypass valve is closed. Also, the cartridge valve pilot pressure chamber is vented to tank.
2) Directional control valve (DV13) is used to control the land motor bypass valve actuator. As with DV12 the de-energized position locks the actuator in the position it was last in. The solenoid B position shifts the bypass valve to the closed position. The solenoid A position opens the bypass valve.

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3) Flow control (FC8) is a pressure compensated valve. It controls the flow rate to both rotary actuators.
4) Rotary actuators (RA4, RA5) operate the bypass valve for the sea and land hydrostatic motors.

The final drive supply system drive consists of: one motor (M4) and one priority valve (PV1). The operation and function of these components is as follows:

1) Motor (M4) is a fixed displacement bent axis piston motor with a 0.3 cu. in/rev. displacement. This motor is capable of developing a theoretical torque of approximately 12.0 FT-LBS with a 3000 psi supply.
2) Priority valve (PV1) controls the flow rate delivered to the motor (M4). It is set for 1.5 GPM which develops as speed of approximately 1000 RPM in the motor. All system flow, with the exception of the suspension supply goes through this valve. The priority valve establishes a priority flow to the final drive supply system drive motor prior to all other system requirements. During start up, when the final drive supply system oil is cold, the available torque at the motor (M4) is less than required for pumping cold oil. Therefore, in its attempt to establish the priority flow, the priority valve blocks all flow to the remainder of the system. After a brief warmup period the final drive oil is warmed by the engine and the required torque decreases. This permits the priority flow to be established and the remainder of the system receives oil flow.

For quick reference purposes the electrical connections to the auxiliary hydraulic system and the valve function are summarized in Table 7.2-3.

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Table 7.2-3 Auxiliary Hydraulic System Directional Control Valve Electrical Connections

Application	Function Solenoid Designation A	B
Primary Cooling Fan	Off	On
Secondary Cooling Fan, 4 GPM	On	-
Secondary Cooling Fan, 8 GPM	On	-
Primary Grille Actuator	Open	Close
Secondary Grille Actuator	Open	-
Port Waterjet Steer	CCW	CW
Starboard Waterjet Steer	CW	CCW
Hydropneumatic Suspension	Extend	Retract
Ramp Actuator	Up	Down
Ramp Latch	Unlatch	Latch
Drive Mode Control, Sea	Motor On	Bypass
Drive Mode Control, Land	Bypass	Motor On

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7.3 Fire Suppression System

The ATR is equipped with a Fire Suppression System (FSS) to protect against loss of vehicle and personnel from fires. The system, which is provided by Santa Barbara Research Center, is based upon the reliable FSS currently in production for the Bradley Fighting Vehicles. The system provides fire sensing, KE-HEAT round discrimination and automatic BITE capability in both the engine and crew compartments. A second shot capability is also provided for the engine compartment.

7.3.1 System Hardware

The FSSS consists of PM-34 Fire Sensors, a Control Electronics Amplifier (CEA), a control panel, solenoid activated valve and bottle assemblies filled with Halon 1301, and an electrical harness assembly. The physical characteristics of these components are presented in Table 7.3-1. Descriptions of these components are provided in the following sections.

7.3.1.1 PM-34 Sensors. Four PM-34 CB sensors are used for crew compartment protection and three PM-34 CBEH sensors are used for engine compartment protection.

These sensors are similar to the PM-34C which is currently in use on the Bradley M2/M3 Fighting Vehicle System (FVS), which is completely qualified and field tested. The sensors selected for the ATR also have an internal optical BITE. Sensor operation is not simply monitored for electrical continuity or shorts, but internal infrared emitters are built into the sensor to insure proper operation of detectors and electronic circuitry in the sensor and the CEA.

The PM-34 CBEH has the same capabilities as the PM-34CB, but also has an early warning heat response feature. This feature provides an early warning response to a factory adjusted temperature differential between the sensor and the area being monitored, without triggering the suppression systems to warn the ATR driver of an engine/engine compartment overheat condition.

7.3.1.2 Control Electronics Amplifier. The Control Electronics Amplifier is the standard amplifier now in use on the Bradley M2/M3 FVS vehicles. This CEA, which is completely qualified and field tested, is designed to be compatible with all combat vehicles presently in service. The CEA can drive either solenoid or squib activated valves.

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Table 7.3-1 FSS Component Physical Characteristics

Component	Weight, lbs.	Size, Inches
Bottle and Valve Assembly, 5#	20.0	5.42 dia. x 16.47H
Sensor, PM34	>0.6	3.50L x 2.65W + 1.70H
Control Electronics Amplifier	>5.0	9.38L x 4.50W x 3.50H
Control Panel	5.0 EST	6.50L x 3.75W x 2.63H
Wiring Harness	6.0 EST	N.A.

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7.3.1.3 FSS Control Panel. The FSS control panel is located at the driver's station and provides for complete monitoring and control of both the engine and crew compartments. Automatic BITE testing of both compartments is also provided. The control panel has the same size housing as the LAV FSS control panel, with the panel layout designed specifically for the ATR FSS control functions.

7.3.1.4 Extinguisher Valve and Bottle Assembly. Very low leakage, fast acting, solenoid activated valves assembled with standard receiver bottles of 144 cubic inches capacity, filled with 5 pounds of Halon 1301, and supercharged at room temperature to 750 psig with nitrogen gas, are used for both crew and engine compartment fire suppression. The solenoid valve selected for the extinguisher bottle is Marotta Scientific Controls, Inc., Model MV121KJ-1. Marotta is presently supplying MV121KJ series valves for both the M1 and M2/M3 vehicles.

7.3.2 System Configuration

The crew compartment suppression system is a single shot system utilizing either one or two valve and bottle assemblies. The engine compartment system is a two shot system with one bottle/valve activation per shot. The second shot can be actuated by the vehicle driver from a toggle switch on the FSS control panel. The FSS also provides an output signal from the control panel which automatically shuts down the engine cooling fans anytime a fire is sensed in either the crew or engine compartments.

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7.4 Bilge Pumps

The ATR bilge pump system is configured similar to the system used on the LVTP7 and has the same total system operating capacity of 480 gallons per minute (GPM). The general arrangement of the bilge pump installation is shown in Figure 1.2-2. Two electrically driven, 125 gpm, pumps are located at the right front and left rear corners of the hull. Two hydraulically driven, 115 gpm, pumps are located at the left front and right rear corners. Characteristics of the electrically and hydraulically driven bilge pumps are listed in Tables 7.4-1 and 7.4-2, respectively. Drawings of these pumps are contained in Volume IIIC of this report, reference vendor drawing numbers 2600523 for the hydraulic pump and 012-05002 for the electric pump.

The pump discharge tubes are fabricated from 1.25 inch diameter aluminum tube and flexible hose. The discharge tubes are connected to the pump and to the pump outlet fittings located on the top deck of the vehicle using reinforced rubber hose and hose clamps.

All bilge pumps will automatically operate any time the vehicle is operated in either the transition or the sea modes of operation. The fore and aft electrically driven pumps are independently controlled by two toggle switches located on the drivers control panel as shown in Section 7.6. These switches provide for manual operation of the electrically driven bilge pumps, during land mode, to permit clearing the bilge of water.

The ATR is equipped with four drain plugs located near each corner of the hull bottom plate. The drain plugs conform to MS49005-16 and have a 1-1/2 - 11-1/2 inch pipe thread with a recessed square head drive. The drain plugs are removable and replaceable from the exterior of the vehicle.

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Table 7.4-1 Electrically Driven Bilge Pump Characteristics

Manufacturer:	Enpo Pump Co.
Model:	5-111M
Present Application:	LVTPT7
Rating:	125 GPM against 6 ft. head
Input Requirement:	55 amps max. at 27.5 VDC against 6 ft. head
Overall Dimensions:	15.62"L x 5.62" Dia. - Approx.
Weight:	25 pounds
Operating Capability:	Submerged and dry run
Mounting Configuration:	Horizontal

Table 7.4-2 Hydraulically Driven Bilge Pump Characteristics

Manufacturer:	MP Pumps, Div. of Tecumseh
Model:	26474 (Dept. of Navy #80064-2600523)
Present Application:	LVTP7A1
Rating:	115 GPM against 15 ft. head
Input Requirement:	3 GPM max. at 1400 psi under rated condition
Overall Dimensions:	8.3"W x 8.0"L x 5.4"H
Weight:	10 Pounds

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7.5 Ballast

To reach the GVW of 30285 pounds the ATR must be fitted with 6538 pounds of ballast. This ballast simulates the items which would be required on a combat ready amphibian, but which are not installed on the ATR. Table 7.5-1 lists the ballast items and corresponding weights which were specified in the statement of work together with the updated ballast requirement.

Table 7.5-1 Ballast Items

Item	Weight, lbs. RFP Prediction	Weight, lbs. Current Estimate (6/11/85)	
Turret Assembly	678	2121.4	
Armament	258	335.4	MCWS
Ammunition	350	543.2	
Troops (10)	2250	2250	
Navigation & Communications	330	330	
OEM	730	730	
NBC Equipment	200	200	
Passive Night Viewer	28	28	
Total	4824	6538	

For automotive testing, the ATR has been equipped with 3000 pounds of ballast to represent the weapon station only. The ballast supplied is composed of externally mounted steel plates and the observer's station cupola, cupola mounting ring, and observer's accommodations.

There are four externally mounted steel plates. The characteristics of these are:

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Qty	Thickness x height x length (inches)	Wt (each (LBS)
2	1.5 x 24 x 72	733
2	1.5 x 24 x 60	600

Each plate is designed to mount on the hull side using eight 3/8 inch diameter mounting bolts.

The observer's station is mounted over the weapon station hull opening. These components provide an additional seating position in the absence of the weapon station. The total weight of the observer's station components is 372 pounds, which is removed when the MCWS is installed.

7.6 Operator's Controls and Instrumentation

The operator of the ATR vehicle is provided with the following controls and instrumentation:

a. Driver's Control Panel
b. Driver's Instrument Panel
c. Driver's Display Panel
d. Steering Control Grip
e. Brake Pedal Assembly
f. Accelerator Pedal Assembly

Descriptions of each of these assemblies are provided in the following subsections.

7.6.1 Driver's Control Panel

The Driver's Control Panel, shown in Figure 4.6-2, provides the following controls:

1. Service Lights Switch - Used to illuminate the vehicle headlights for night driving.
2. Hi Lo Switch - Switches headlights to high or low beam. Not functional at present time.
3. Master Power Switch - When placed in the ON position, energize the master power relays that provide electrical power to the vehicle and control system.

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4. Bilge Pump Switches - Operates the FORE and AFT electric bilge pumps when the Mode Switch is in the LAND mode. Two LED lamps are provided to indicate bilge pumps operation.
5. Automotive Power Switch - When placed in the ON position, provides electrical power to the vehicle engine instruments and starting circuits.
6. Navigation Lights - Not presently functional on the vehicle.
7. Starter Switch - When placed in the ON position and held, the engine will crank. Releasing the switch will stop starter motor cranking.
8. Engine Stop Switch - When pressed and held de-energizes the fuel shut off valve located on the engine. The switch must be held on for several seconds to insure a complete engine stop.
9. Horn Switch - The vehicle horn will sound when this switch is pressed.
10. Ramp Latch/Unlatch Switch - This switch energizes the hydraulic valve that latches or unlatches the vehicle ramp. The RAMP SAFETY switch must be held in the OPERATE position for this switch to be functional.
11. Ramp Safety Switch - The Ramp Safety Switch is used to enable the use of the Ramp Latch/Unlatch and Ramp Up/Down switches.
12. Ramp Up/Down Switch - With the ramp unlatched, the Gear Select Switch in PARK, and the ramp Safety switch held in the SAFE position, this switch will operate the ramp up or down.
13. Mode Switch - This switch allows the driver to select the vehicle operating mode as follows:

    Land Mode-allows operation of the suspension switch, provides hydraulic power to the land motors, and opens the louvered cooling system grilles.

    Transition Mode - prepares the vehicle for entering or leaving the water. The louvered grilles are closed,

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hydraulic power is provided to both the land motors and the waterjet motors, and electric power is provided to the marine cooling system circulation pump.

Sea Mode - retracts the suspension, provides hydraulic power to the waterjet motors, and maintains electric power to the marine cooling system circulation pump.

14. Suspension Switch - This switch is used to extend or retract the vehicle suspension for maintenance purposes. The switch is normally kept in the extend position.
15. Gear Select Switch - PARK position is used for starting the vehicle, and enabling the ramp switches. The final drive brake is applied automatically. DRIVE position allows the vehicle to be moved in a forward direction in either low or high transmission ratio. High and low ratios are selected by a slide switch located on the steering grip. Neutral position - vehicle will not move. Low gear ratio and hydrostatic braking are automatically applied. Reverse position - allows vehicle to be operated in a reverse direction in low gear ratio.

7.6.2 Driver's Instrument Panel

The Driver's Instrument Panel, shown in Figure 4.6-1, contains the following gauges and instruments:

1. Engine Oil Temperature Gauge
   Indicates the engine oil temperature
2. Engine Oil Pressure Gauge
   Indicates the engine oil pressure.
3. Port/Starboard Hydraulic Oil Temperature Gauges
   Indicate the temperature of the hydraulic oil in the port and starboard hydraulic reservoirs.
4. Coolant Temperature Gauge
   Indicates the operating temperature of the engine coolant.
5. Fuel Level Meter Gauge
   Indicates the level of fuel in the selected fuel tank.

6. Port/Starboard Fuel Tank Selector Switch
   Selects the fuel tank to be monitored for fuel level.
7. Battery Voltage Switch
   Selects Vehicle or Computer Control system voltage for display on the battery voltmeter.
8. Battery Voltmeter
   Used to monitor the Vehicle and Computer Control System voltages.
9. Built-In Test Switches
   These switches, when pressed, generate high and low, vehicle and computer, voltage faults for testing the master warning system.

7.6.3 Driver's Display Panel

The Driver's Display Panel, shown in Figure 4.6-3, contains the following displays:

1. Speedometer
   Indicates vehicle speed, in miles per hour.
2. Master Warning Lamp
   This lamp flashes whenever a vehicle fault is detected.

7.6.4 Steering Control Grip

The steering control grip is operated by the driver's right hand. The grip moves left or right 28 degrees and is self-centering. As the grip is positioned, a proportional electrical signal is transmitted to the computer and the vehicle steers to the left or right. In the sea mode, vehicle steering is accomplished by the position of steering buckets located at the outlet nozzles of the waterjet propulsers. A switch located on top of the grip, selects the transmission High or Low final drive unit ratio.

7.5.5 Brake Pedal Assembly

This assembly provides three functions:

1. Provides dynamic braking when pedal pressure is applied. As the pedal is pressed, a proportional electrical signal is transmitted to the computer and hydraulic braking is applied to the final drive motors.

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2. A five percent deflection switch is provided to turn on the vehicle brake lights.
3. A ninety percent deflection switch is provided to apply the mechanical brakes in the final drive units.

7.6.6 Accelerator Pedal Assembly

As the driver presses on the pedal, a proportional electrical signal is transmitted to the computer. The computer calculates the required engine speed from the desired vehicle speed, and other inputs, and transmits this signal to the throttle controller and to the hydrostatic drive system control valves.

7.7 Engineer's Panel

The Engineer's Panel, shown in Figure 7.7-1, contains instruments to monitor a number of vehicle parameters and also provides one control function:

1. Master Warning Light

   This lamp flashes, as well as the one on the Driver's Display Panel, whenever a vehicle fault is detected.
2. Elapse Time Meter

   Indicates the total number of engine operating hours. An oil pressure switch located on the engine supplies the signal to operate the meter.
3. Accumulated Miles Indicator

   Indicates the total number of miles accumulated by the ATR. The signal that operates this meter is derived from the speedometer circuit.
4. PORT/STBD Filter EXPIRATION Indicators

   The Port/Stbd. filter bypass indicator consists of a pair of light bars with each individual lamp numbered. These indicate when a particular filter has expired. The master warning system will be activated whenever a filter expires. The test engineer will identify the expired filter(s) by referring to the filter location chart located near the Engineer's Panel. A copy of this chart is shown in Figure 7.7-2.

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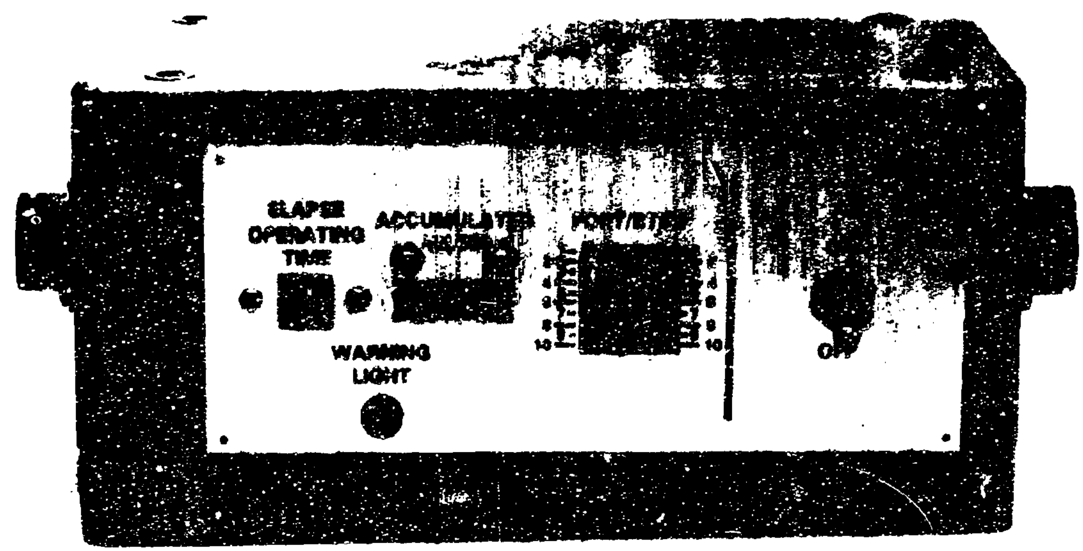

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Figure 7.7-1 Engineer's Panel

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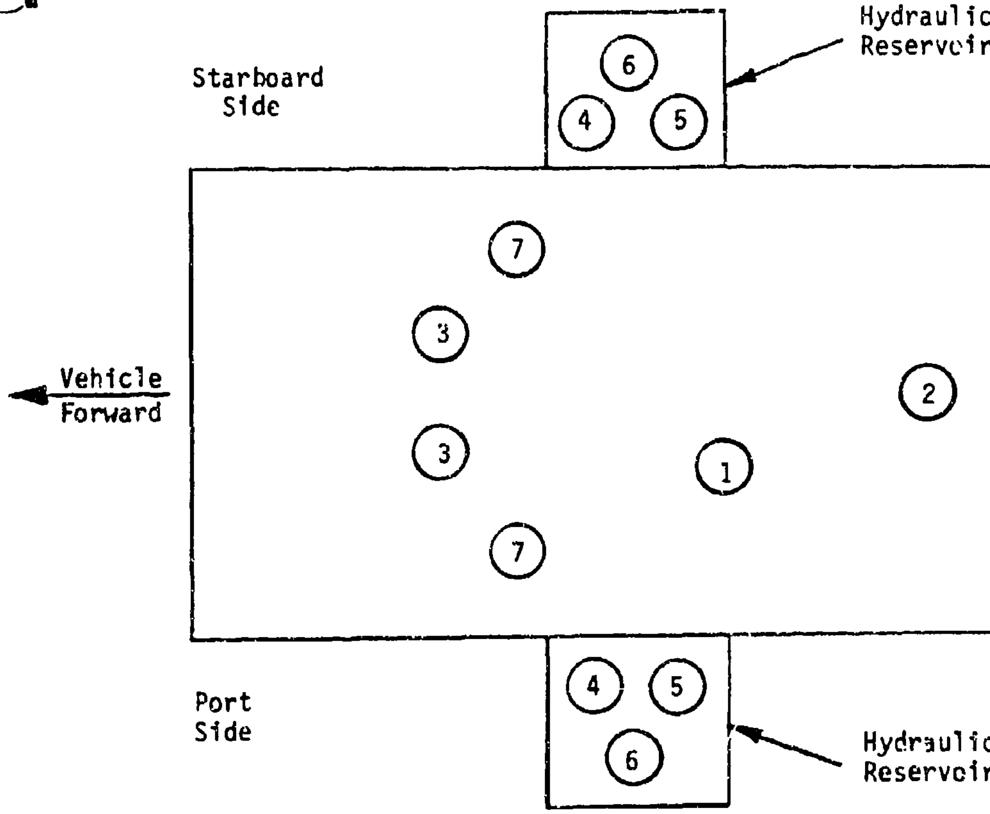

Lamp Number	Filter Location	
	Port Side	Starboard Side
1	Final Drive Supply Discharge	- Unused -
2	Final Drive Scavenge Suction	- Unused -
3	Hydrostatic Drive Charge Pump	Hydrostatic Drive Charge Pump
4	Primary Fan & Hydrostatic Drive Case Drain	Auxiliary & Hydrostatic Drive Case Drain
5	Primary Fan Return	Auxiliary Return
6	Hydrostatic Drive Servo Vent	Hydrostatic Drive Servo Vent
7	Primary Fan Discharge	Auxiliary Discharge
8	- Unused -	- Unused -
9	- Unused -	- Unused -
10	- Unused -	- Unused -

Figure 7.7-2 Filter Location Chart

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5. H.P.S. Auxiliary Power Switch

The H.P.S. Auxiliary Power Switch is used to control the H.P.S. Auxiliary Power Unit, when employed.

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8.0 VEHICLE DATA

8.1 Weight Properties

During the ATR Program, the weight of the ATR has been continuously monitored and updated as additional design features and components have been defined. Actual measurements have been substituted for estimates during the fabrication phase. Composite ATR weight predictions have been provided on a regular basis in program monthly progress reports. Table 8.1-1 shows the progression of ATR weight predictions for each major subsystem and the total ATR. The December 1985 values given in the Table are the "as built" weight figures. The "as built" ATR weight (not including crew and ballast) was verified during acceptance testing on 6 November 1985. The total GVW is 30285 lbs.

The ATR GVW prediction shown in Table 8.1-1 includes ballast for the MCWS weapons station currently under development by DTNSRDC. The substitution of the actual MCWS weapons station for the statement of work specified ballast weight resulted in a 1743 pound weight increase. Without considering the weapon station weight increase, the ATR GVW is 28,542 which is well within the statement of work requirement.

Table 8.1-2 shows the percentage of total ATR weight each of the subsystems contributes, based on the as built weight values.

Table 8.1-2 ATR Weight Breakdown Percentage

Subsystem	Percent GVW %
Hull and Frame	25.7
Suspension System	21.0
Propulsion Plant	11.2
Hydrostatic Drive System	10.0
Marine Drive System	2.1
Auxiliary Systems	4.7
Loaded Items	3.6
Ballast	21.7

A breakdown of the weight of each element of each major subsystem is provided in Table 8.1-3. A breakdown to the component level is presented in Appendix A of this report.

The actual location of the center of gravity has been determined for the ATR and is shown in Table 8.1-4. The center of gravity was determined without ballast and crew aboard.

Table 8.1-1 ATR Weight Progression

SUBSYSTEM	WEIGHT, LBS. PROPOSED	FEB. (84)	MARCH (84)	APRIL (84)	JUNE (84)	*DEC (85)	WEIGHT CHANGE PROPOSAL-DEC (85) LBS.
Hull & Frame	9409	7632	7343	7416	7475	7770	-1639
Suspension	5947	6062	6292	6347	6447	6373	426
Propulsion Plant	2914	3014	3176	3232	3314	3402	488
Hydrostatic Drive	2122	2633	2693	2948	2960	3049	927
Marine Drive	320	576	576	576	606	640	320
Auxiliary System	590	539	538	555	935	1415	825
Loaded Items	1170	1170	1160	1170	1170	1098	- 72
Ballast	4795	5509	5509	5509	5509	6538	1743
TOTAL	27267	27136	27298	27754	28417	30285	3018
Change in Total from Previous Prediction		-131	162	456	660	1868	

*December 1985 values represent the as-built weight.

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Table 8.1-2 ATR As-Built Weight

Item	Weight (Pounds)
Hull and Frame	7770
Hull Welded & Machined	6022
Hull Bolted & Misc.	431
Bulkhead & Internal Covers	304
Accommodations	191
Appendages	343
Ramp	404
Integration & Assembly	75
Bow Flap	0
Suspension	6372
Suspension Devices	1537
Road Wheel Assy	1010
Drive Sprocket Assy	433
Idler Assy	454
Track	2194
Integration, Assy & Road Arms	612
Hydropneumatic Hydraulics	133
Propulsion Plant	3402
Engine	1613
Induction/Exhaust	168
Auto Cooling	690
Marine Cooling	254
Engine Control	25
Fuel System	316
Engine Electrical	102
Batteries and Mount	236

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Table 8.1-3 ATR As-Built Weight (continued)

Item	Weight (Pounds)
Hydrostatic Drive	3049
Pumps (2)	341
Motors (2)	376
Transfer Case	313
Controller	128
Hydraulic Lines	517
Final Drive	859
Marine Drive Train	640
Marine Propulsor	410
Marine Motors (2)	230
Auxiliary System	1416
Vehicle Electric Sys.	124
Vehicle Hydraulic Sys.	716
Fire Detect. & Suppress.	118
Passive Night Sight	0
Bilge Pumps	73
NBC Environment	0
OEM	14
Control & Equipment	372
Turret Assembly	0
Navigation & Comm	0
Loaded Items	1098
Driver	225
Commander	225
Fuel	648
Ballast	6538
Total Vehicle	30285

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Table 8.1-4 ATR Center of Gravity Location

	Location, Inches		
	Longitudinal (FWD of Sprocket)	Vertical (Above Ground)	Lateral (From C.L.)
Combat Loaded (w/o crew or ballast)	85	43	.2 (Port)

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8.2 Hydrodynamic Characteristics

8.2.1 Hydrostatics

The hydrostatic stability of the Automotive Test Rig (ATR) was evaluated for four cases of vehicle configuration: with and without the troop payload, with the suspension system in the landborne mode (tracks extended), and with the suspension system in the waterborne mode (tracks retracted). The roll stability curves for these cases are shown in Figure 8.2.1-1. As shown in all cases, the ATR is predicted to be very stable in roll, with a positive restoring moment up to 120 degrees of roll. In the inverted position, 180 degrees of roll, the ATR is unstable so that it will tend to right itself. With the troop payload it is practically a self-righting vehicle.

ATR pitch stability curves are shown in Figure 8.2.1-2 for the same four cases. As expected, the position of the track has a very minor effect. The ATR configuration used for computer prediction of its hydrostatic stability is shown in Figure 8.2.1-3. The resulting static waterlines for the ATR both with and without troops are shown in Figure 8.2.1-4. As shown in both cases the waterjet inlet area is well submerged in the static condition.

Predicted values of ATR static draft and trim are shown in Table 8.2.1-1. The reserve buoyancy of the ATR is predicted to be 46.2 percent without troops and 34.6 percent with troops aboard.

8.2.2 Water Performance

The water performance of the ATR was predicted based on the results of a series of 1/16th scale model tests conducted by Davidson Laboratory of Stevens Institute of Technology, Hoboken, New Jersey for DTNSRDC. The model configuration used in these tests and depicted in Figure 8.2.2-1 represented the FLASH amphibian. As shown in Table 8.2.2-1 the principal dimensions of the FLASH amphibian and the ATR are very similar and the longitudinal center of gravity locations for both are approximately the same percentage of each vehicle's overall length.

The close similarity between the ATR and the FLASH model allows the FLASH model test results to be used as an indicator of ATR performance with very little modification. The only major difference between the FLASH model and the ATR is that the FLASH was equipped, in all tests but one, with one of several bow flap configurations.

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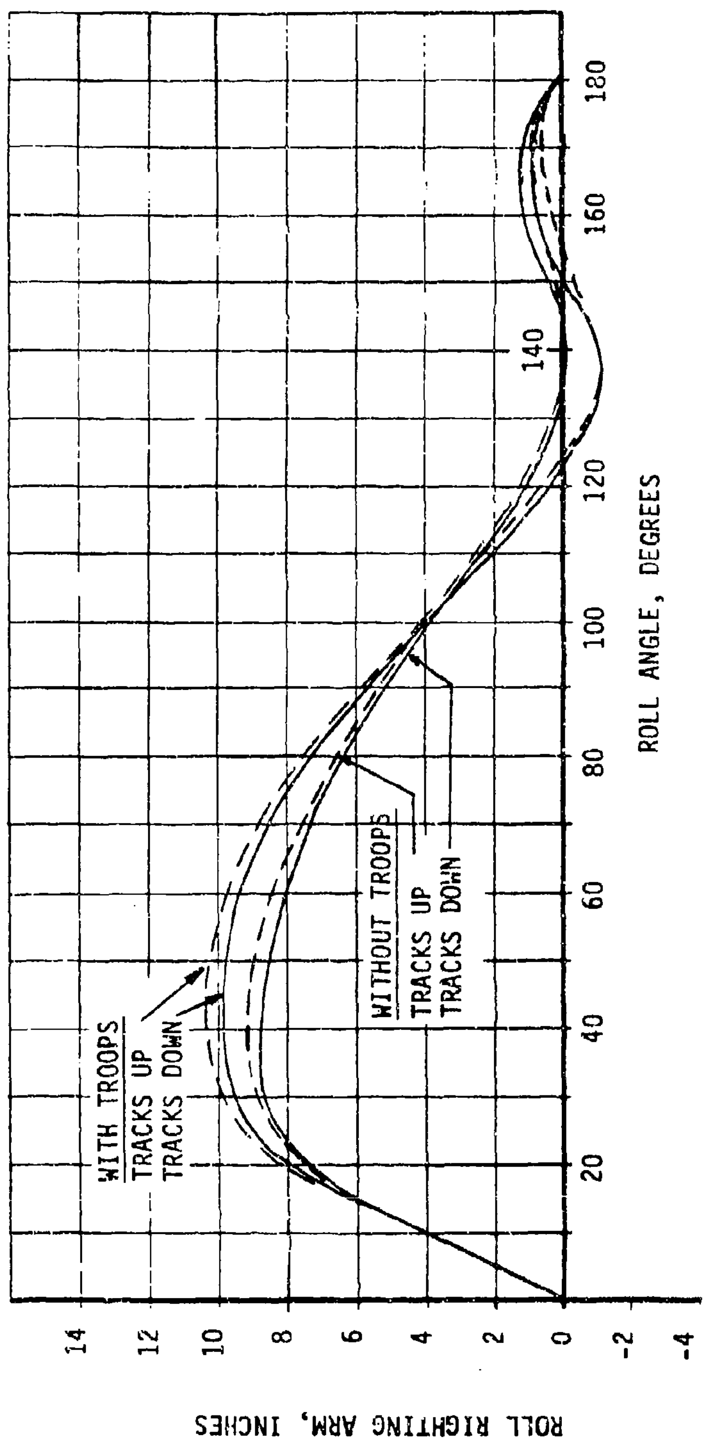

Figure 8.2.1-1 Automotive Test Rig Roll Stability

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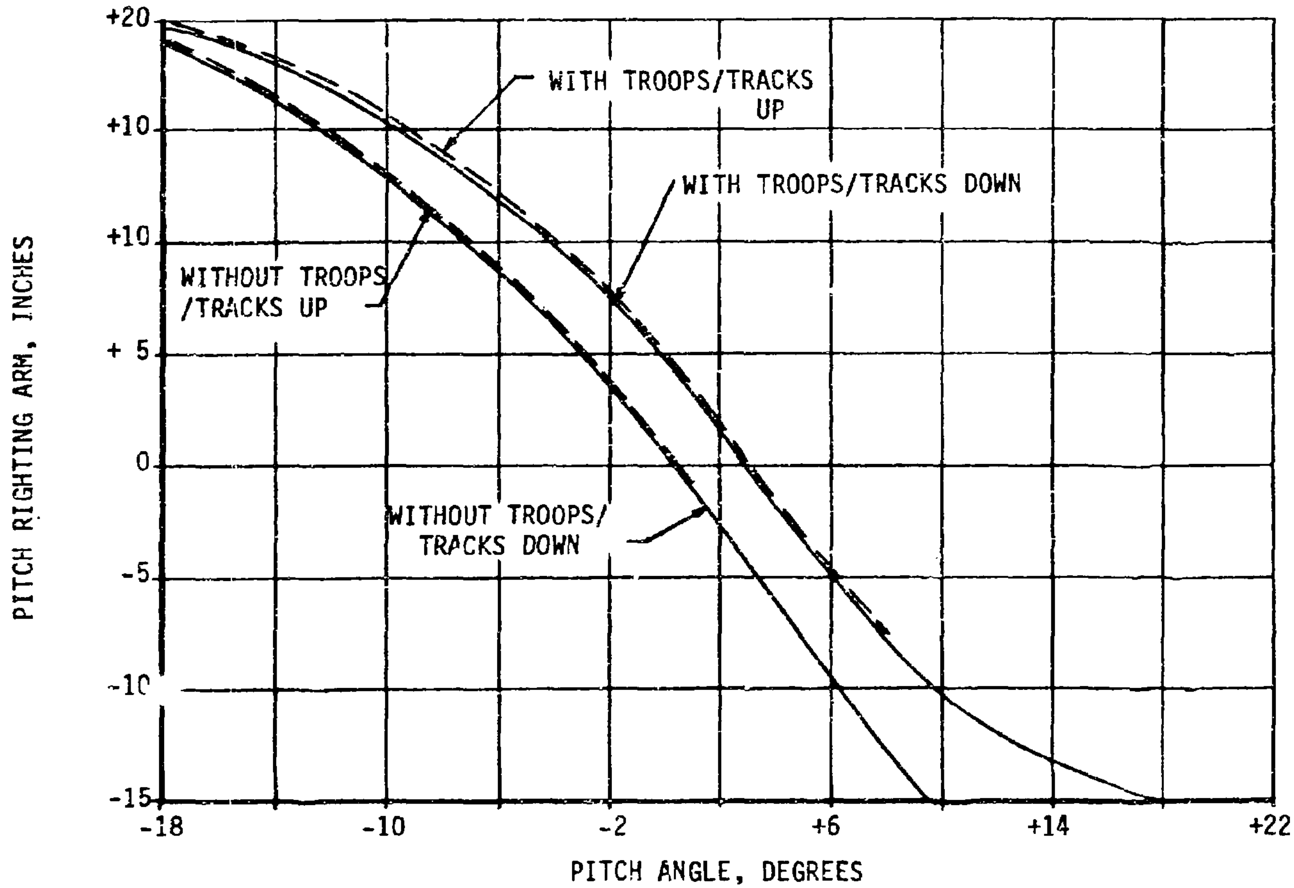

Figure 8.2.1-2 Automotive Test Rig Pitch Stability

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(ALL DIMENSIONS IN INCHES)

XL	= 204.00	Y1	= 32.125	WJH	= 63.00
BEAM	= 106.00	Y2	= 31.625	WJL	= 53.62
HT	= 78.00	BCHIN	= 42.5	WJL2	= 16.753
XBOX	= 204.00	Z0	= 16.00	WETJET	= 0.72
HBOX	= 80.00	Z1	= 20.00	WETTRK	= 0.16 TRACK DOWN
BBOX	= 106.00	Z2	= 23.49	WETTRK	= 0.23 TRACK UP
XB	= 44.113	Z3	= 43.25		
XBU	= 45.75	Z6	= 63.00		
XBL	= 62.12	ZBL	= 38.82		
XS	= 2.72	ZS2	= 25.30		
XS2	= 26.57	ZCHIN	= 69.95		
XCHINL	= 21.2				
XCHINU	= 24.0				

CONFIGURATION:		GVW	XG	YG	ZG
	WITH TROOPS	28,417 lb	110.78	0	43.89
	WITHOUT TROOPS	26,167 lb	106.78	0	43.5

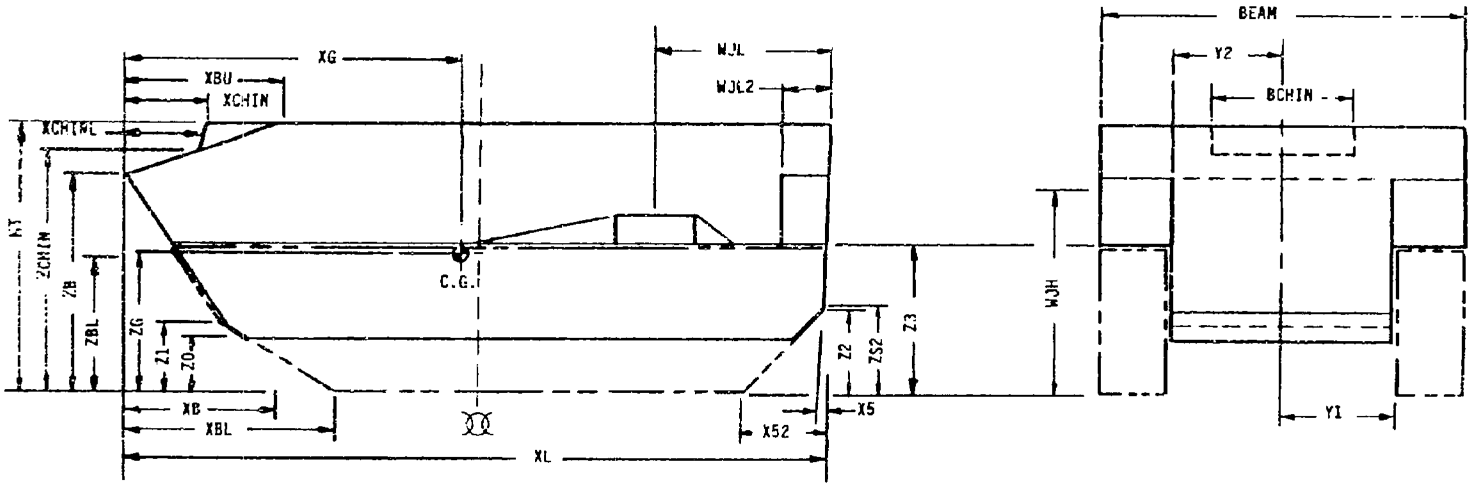

Figure 8.2.1-3 ATR Configuration Used For Prediction of Hydrostatic Stability

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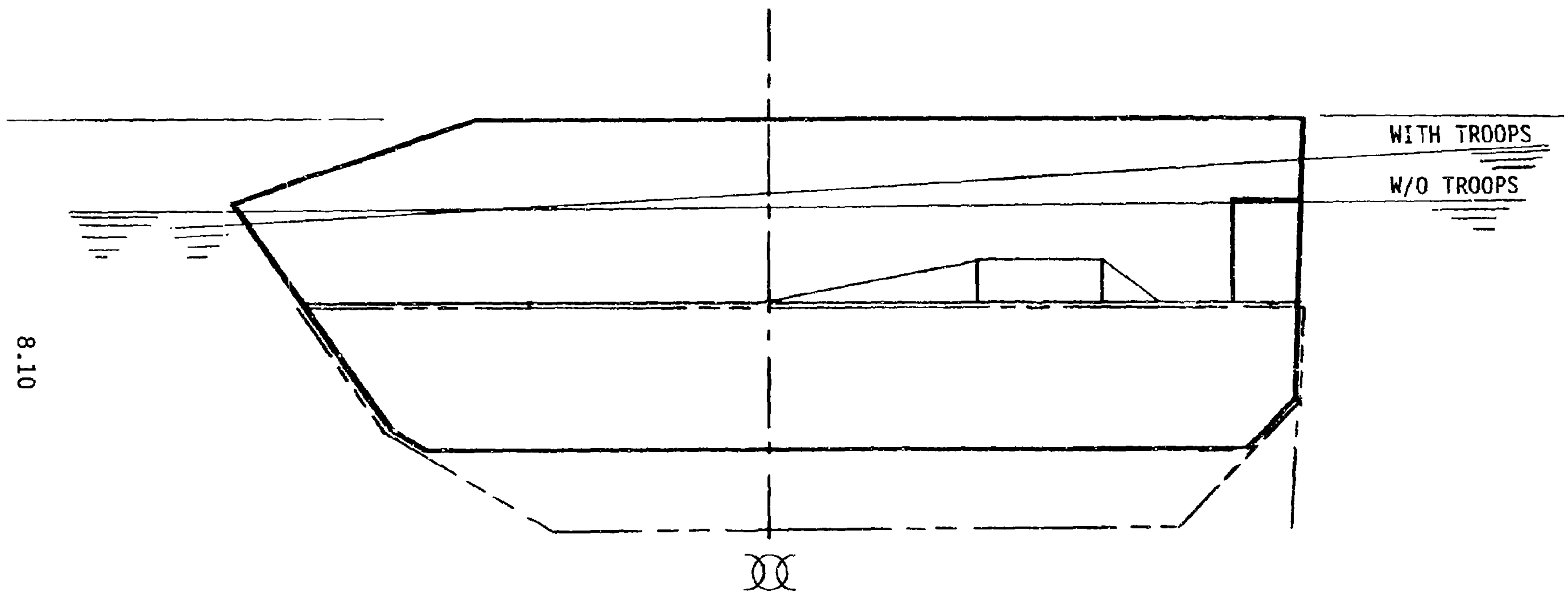

Figure 8.2.1-4 ATR Static Waterlines

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Table 8.2.1-1 Automotive Test Rig Static Trim Data

CHARACTERISTIC	WITH TROOPS GVW = 28,417 lbs.	WITHOUT TROOPS GVW = 26,167 lbs
Center of Gravity Location$^{\Delta}$		
Station, inches	167.90	163.90
Waterline, inches	43.89	43.50
Static Draft		
Forward, inches	58.90	60.80
Midships, inches	64.20	61.30
Aft, inches	69.50	61.80
Trim Angle, degrees	+3.0	+.3

Δ - Note: Sprocket centerline is Station 250.00, vehicle bow is Station 57.12. Waterline 0 is ground with track extended.

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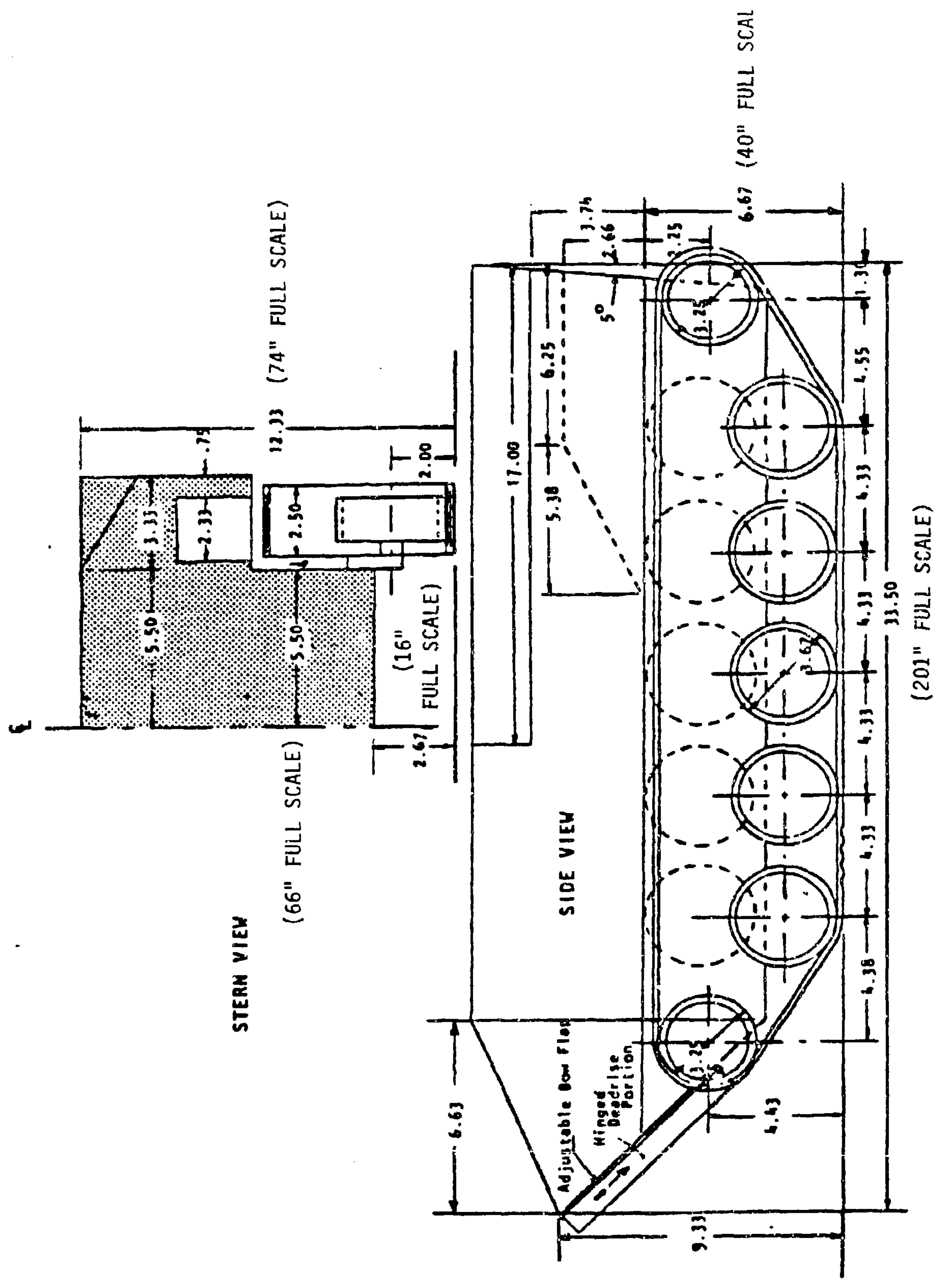

Figure 8.2.2-1 FLASH Model Configuration

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Table 8.2.2-1 Comparison of FLASH and ATR Configurations

CHARACTERISTIC	FLASH		ATR	
DATA SOURCE	DAVIDSON LAB. RPT. SIT-DL-82-9-2250, JANUARY 1932		AAI DRAWINGS NO. 60011-40201, -10002 JULY 9, 1984	
Overall Length, inches	201		204	
Width, inches	106		106	
Height, (ground to top deck), inches	74		78	
Ground Clearance, inches	16		16	
Hull Width Inside Tracks, inches	66		64.25	
Height to Bottom of Sponson from Ground, inches	40		43.25	
Height of Forward Point of Bow, inches	56		63	
	WITHOUT TROOPS	WITH TROOPS	WITHOUT TROOPS	WITH TROOPS
Gross Weight, lbs	25,000	28,000	26,167	28,417
Weight per inch of Overall Length, lbs/inch	127.4	139.3	128.3	139.3
Vertical CG Location, above ground, inches	42.4	42.4	43.5	43.89
Longitudinal CG Location, aft of bow, inches	104.3	107.9	106.78	110.78
Longitudinal CG Percentage of Overall Length,%	51.9	53.7	52.3	54.3
Static Trim Angle, degrees	-1.5	+1.4	+.3	+3.0
Draft @ Midships, inches	-	-	61.3	64.20

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A second source of information used to develop ATR performance predictions, was a video tape of the FLASH tank-test runs which was made available to AAI by DTNSRDC. The video tapes allowed visual judgments to be made as to the acceptability of each condition tested from the point of view of deck wetness and general vehicle behavior. In some of the runs so much water came over the deck that the vehicle's driver would have been unable to see; in others the vehicle went entirely underwater. Both of these conditions were considered to be totally unacceptable.

The FLASH model was tested at two weights (representing the full-scale vehicle with and without troops), with and without wheel covers (circular plates covering the outsides of roadwheels), with the tracks extended (as for land operation) and retracted level with the hull bottom, and with several bow flap configurations. Tests were run at a range of speeds up to 12 mph in calm water and in head seas representing Sea State 2 (significant wave height 2.2 ft). The combinations tested are shown in Table 8.2.2-2 and the bow flap configurations tested are shown in Figure 8.2.2-2. In only one run was the FLASH model operated with its bow flap in the fully retracted position shown in Figure 8.2.2-1. It was run at 6 mph at an equivalent gross weight of 28,000 lb and immediately trimmed down and submerged. The ATR has a slightly higher static trim than the FLASH model and the forward point of its bow is seven inches higher but, based only on this very limited experiment, it must be tentatively assumed that the ATR will not be able to proceed satisfactorily at water speeds higher than 6 mph without a bow flap.

The predicted calm-water drag characteristics of the fully loaded ATR are shown in Figure 8.2.2-3. The three solid lines are derived directly from the FLASH model tests and correspond to the model with tracks down (wheel covers off) and with tracks retracted (wheel covers off and wheel covers on). The original estimate for the ATR resistance which was derived from a formulation developed to determine LVT(X) drag, is shown as the dashed line in Figure 8.2.2-3. The ATR resistance is assumed to be slightly higher than the FLASH due to the effect of the latter's bow flap, so the dashed line is assumed to be a fair representation of the calm-water resistance of the ATR with tracks extended. It differs only slightly from the measured FLASH drag at speeds above 6 mph which will, in any case, probably not be feasible without installing a bow flap on the ATR.

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Table 8.2.2-2 FLASH Model Test Configurations

WATER CONDITION	VEHICLE GROSS WEIGHT, LBS.	TRACK POSITION	ROADWHEEL COVERS	BOW FLAP CONFIGURATION
Calm	25,600	Extended	Off	As Noted
Calm	25,600	Retracted	Off/On	
Calm	28,000	Extended	Off	
Calm	28,000	Retracted	Off/On	
Sea State 2	25,600	Extended	On	With deadrise and gap
Sea State 2	25,600	Retracted	On	With deadrise and gap, maximum up.
Sea State 2	25,600	Retracted	On	With deadrise and gap
Sea State 2	28,000	Extended	On	With deadrise and gap
Sea State 2	28,000	Retracted	On	With deadrise and gap

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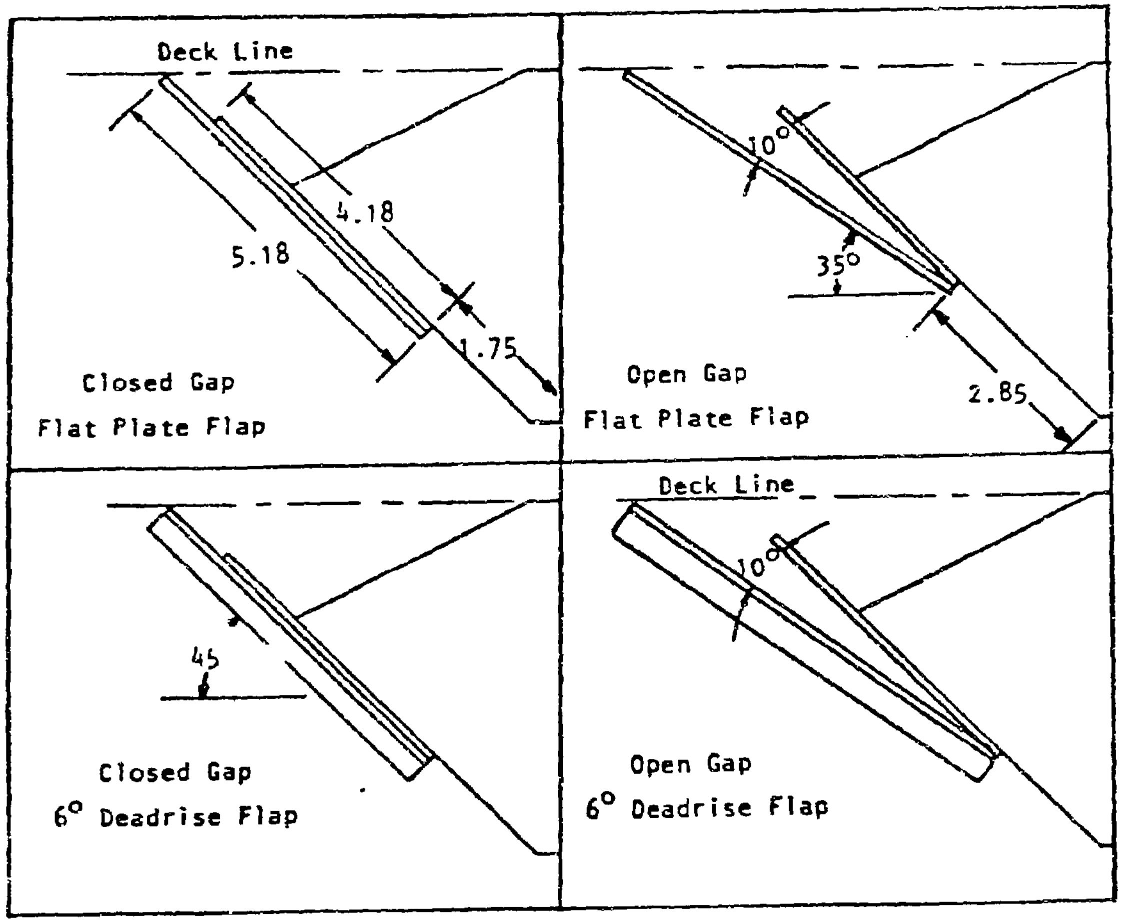

Figure 8.2.2-2 FLASH Bow FLAP Configurations (Dimensions in Full Scale Feet) Open Gap is with FLAP 35° Relative to Baseline

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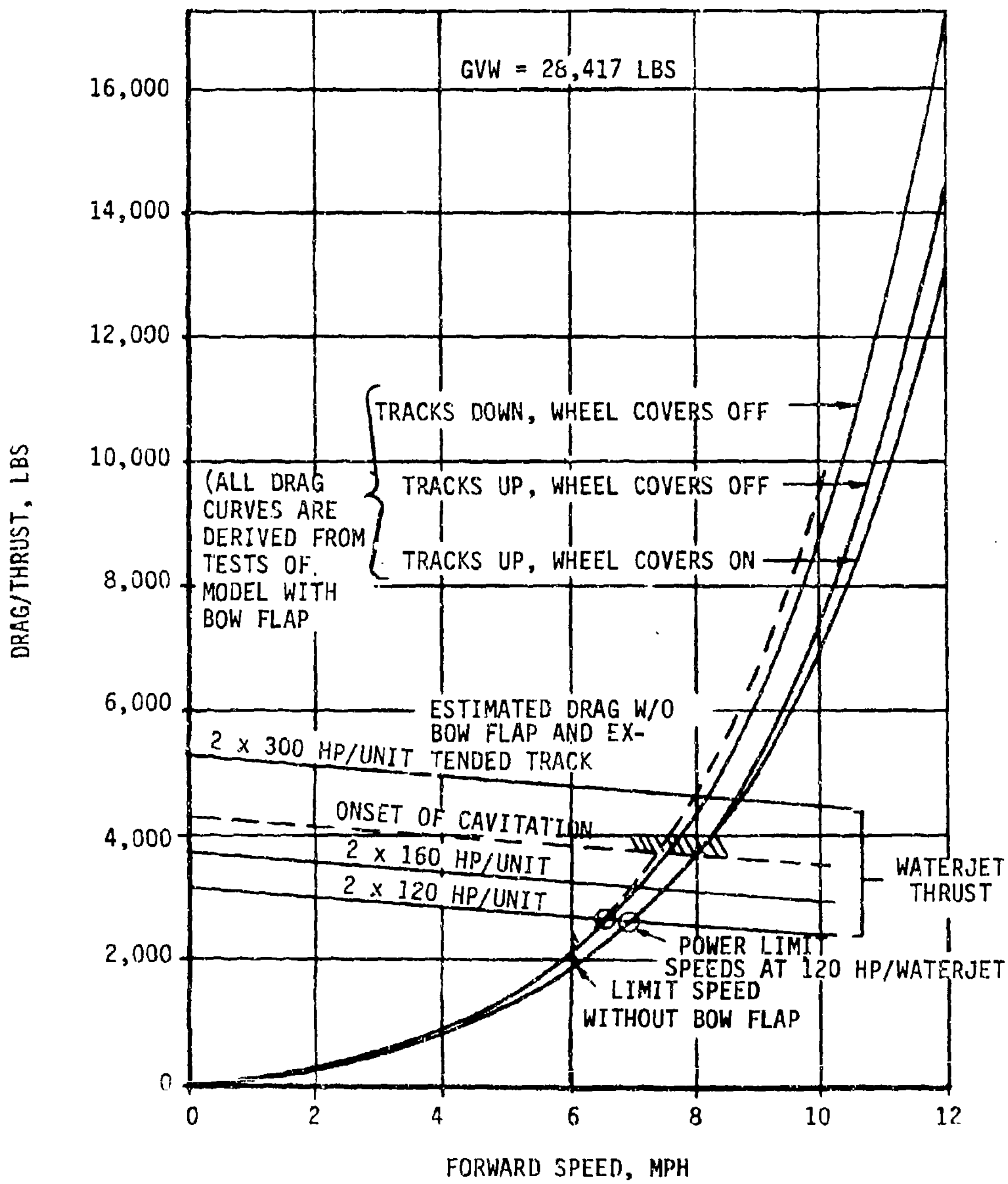

Figure 8.2.2-3 ATR Water Performance

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The effects of track position and ATR gross weight are shown in Figure 8.2.2-4. These relationships are derived based on the FLASH model test results. The variation of ATR trim angle with gross weight and forward speed is shown in Figure 8.2.2-5. The addition of troops, trims the vehicle about 3 degrees up by the bow. The ATR estimated drag in Sea State 2 (significant wave height 2.2 feet) is also shown in Figure 8.2.2-4. These rough water drag curves have also been derived based on the FLASH model test results. Based on these model test results, there is no appreciable difference in rough water drag between the vehicle with its tracks extended or retracted for speeds up to 8 mph.

As discussed in Section 6.2 of this report, the ATR is equipped with two 2-stage axial flow Dowty waterjets with a 300mm (11.8 inch) diameter impeller. The manufacturer's performance characteristics for these waterjets are shown in Figures 8.2.2-6 and 8.2.2-7. The waterjets in the ATR have been installed with angled inlet to allow the incoming water clear access to the pump element of the waterjet with the tracks retracted. Hopefully, this arrangement will allow the waterjets to approach developing the thrust performance shown in Figure 8.2.2-6.

The ATR powertrain is expected to be capable of delivering 120 hp to each waterjet. Curves of the thrust produced by the two waterjets at this power level are superimposed on the drag curves shown in Figure 8.2.2-3. Curves representing 160 hp per unit are also shown as well as the "cavitation limit" from Figure 8.2.2-6. These waterjet units are incapable of absorbing more than about 220 hp each in this application without experiencing cavitation.

A curve of total power, two waterjet units, against forward speed in calm water is shown in Figure 8.2.2-8. The corresponding waterjet impeller speed is also shown. The cavitation limit and the ATR maximum continuous power limit, 240 hp, are also shown.

The estimated ATR maximum forward water speed, based on input power of 120 horsepower to each waterjet, is shown in Table 8.2.2-3 for various sea states and ATR configuration. It should be noted that it may be impossible to attain ATR water speeds above 6 mph if no bow flap is used. This speed limitation may be even more severe for the ATR light load case, without troops, due to the lower static trim angle in this case.

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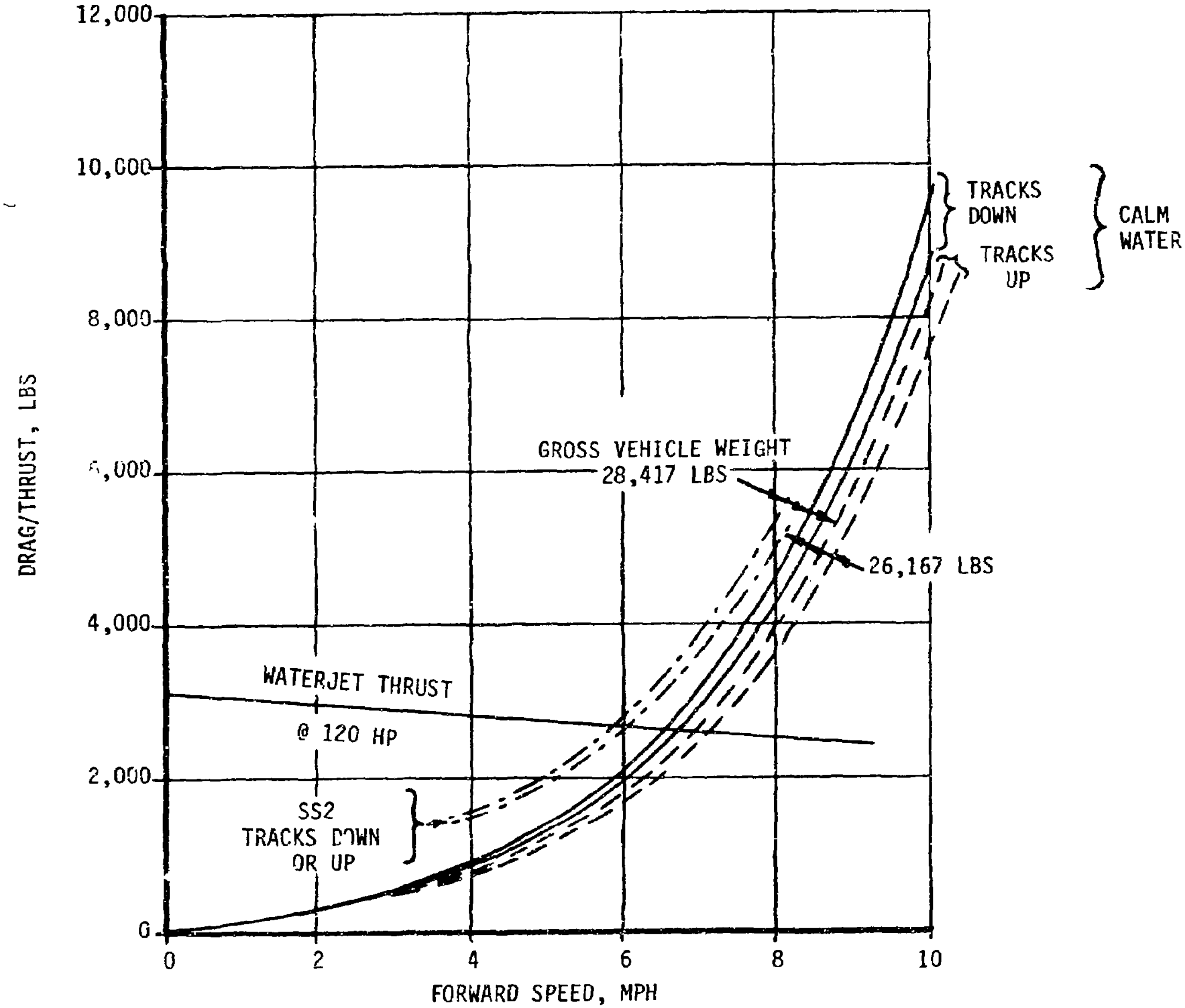

Figure 8.2.2-4 ATR Water Performance in Sea States 0 and 2

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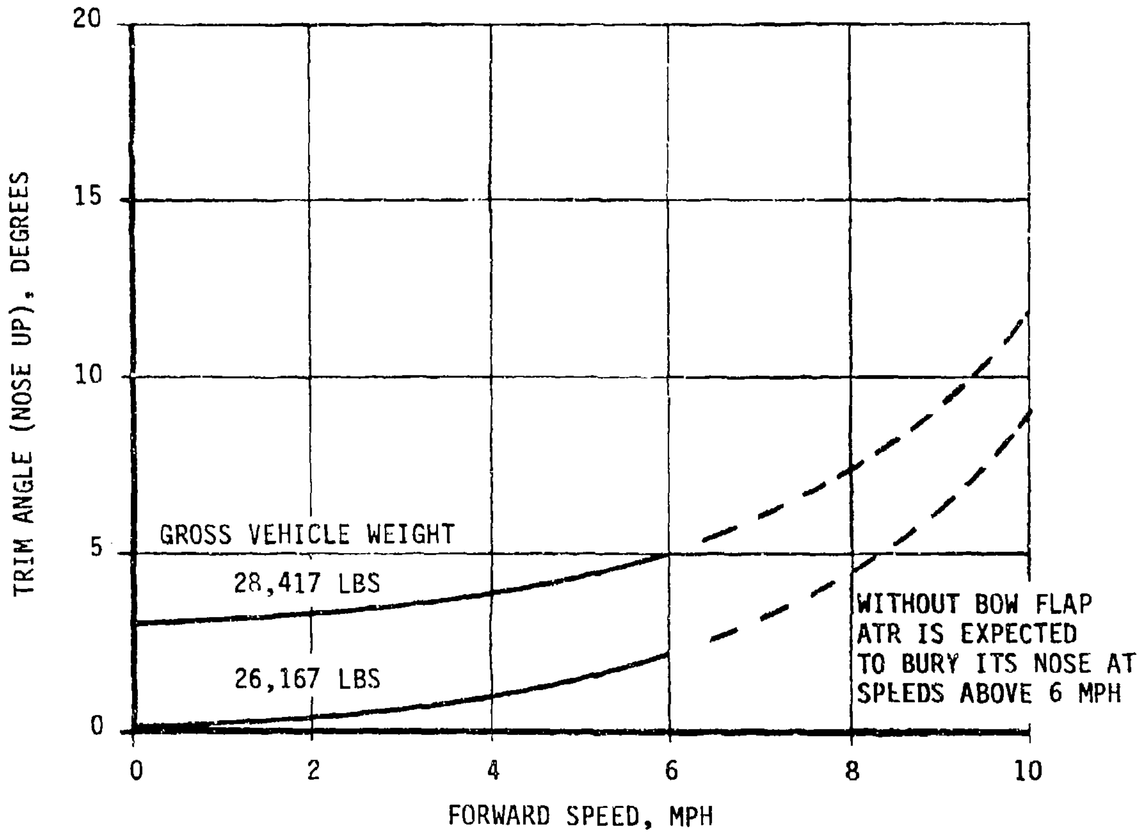

Figure 8.2.2-5 ATR Trim Angle Variation

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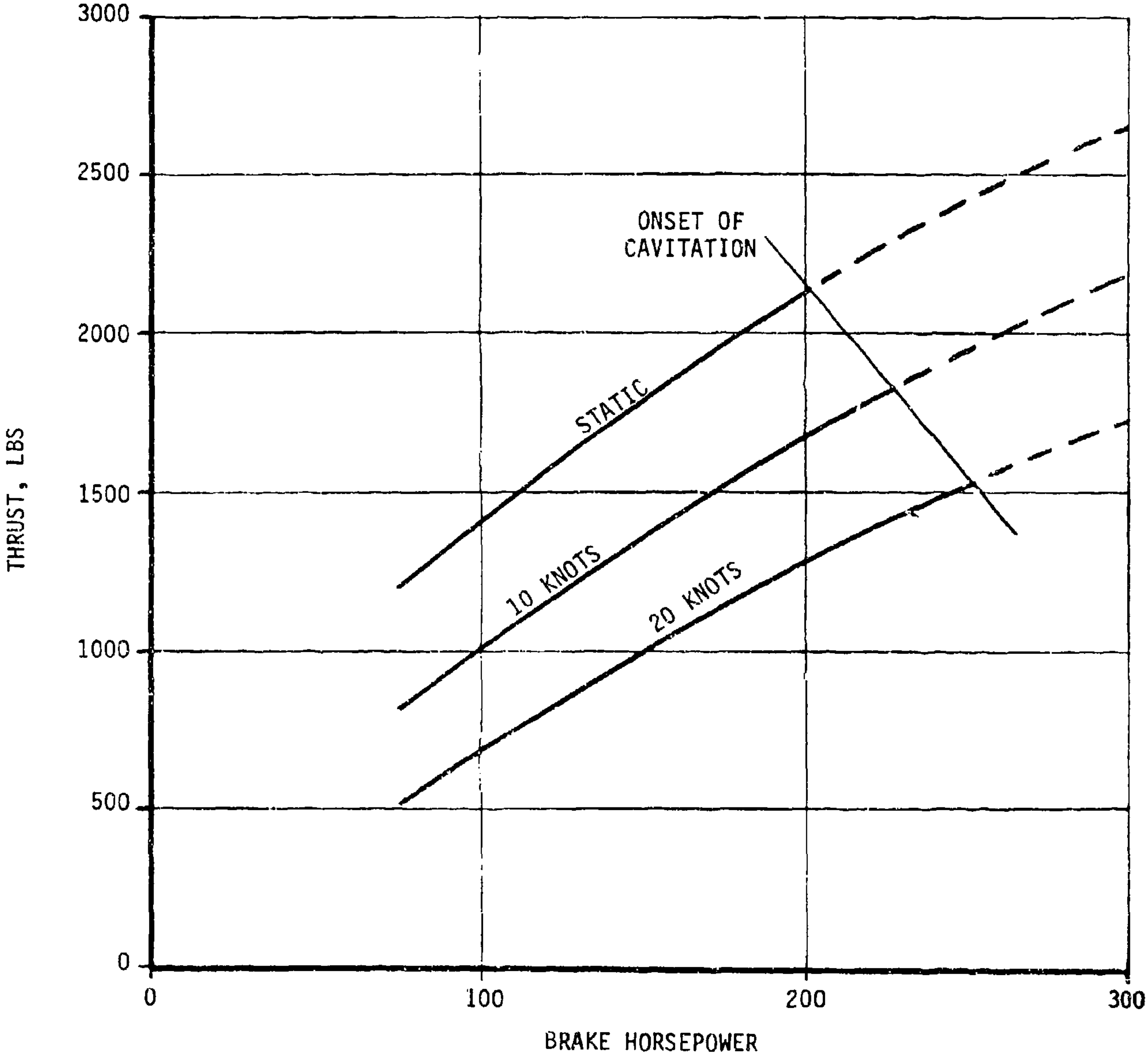

Figure 8.2.2-6 ATR Waterjet Thrust Performance

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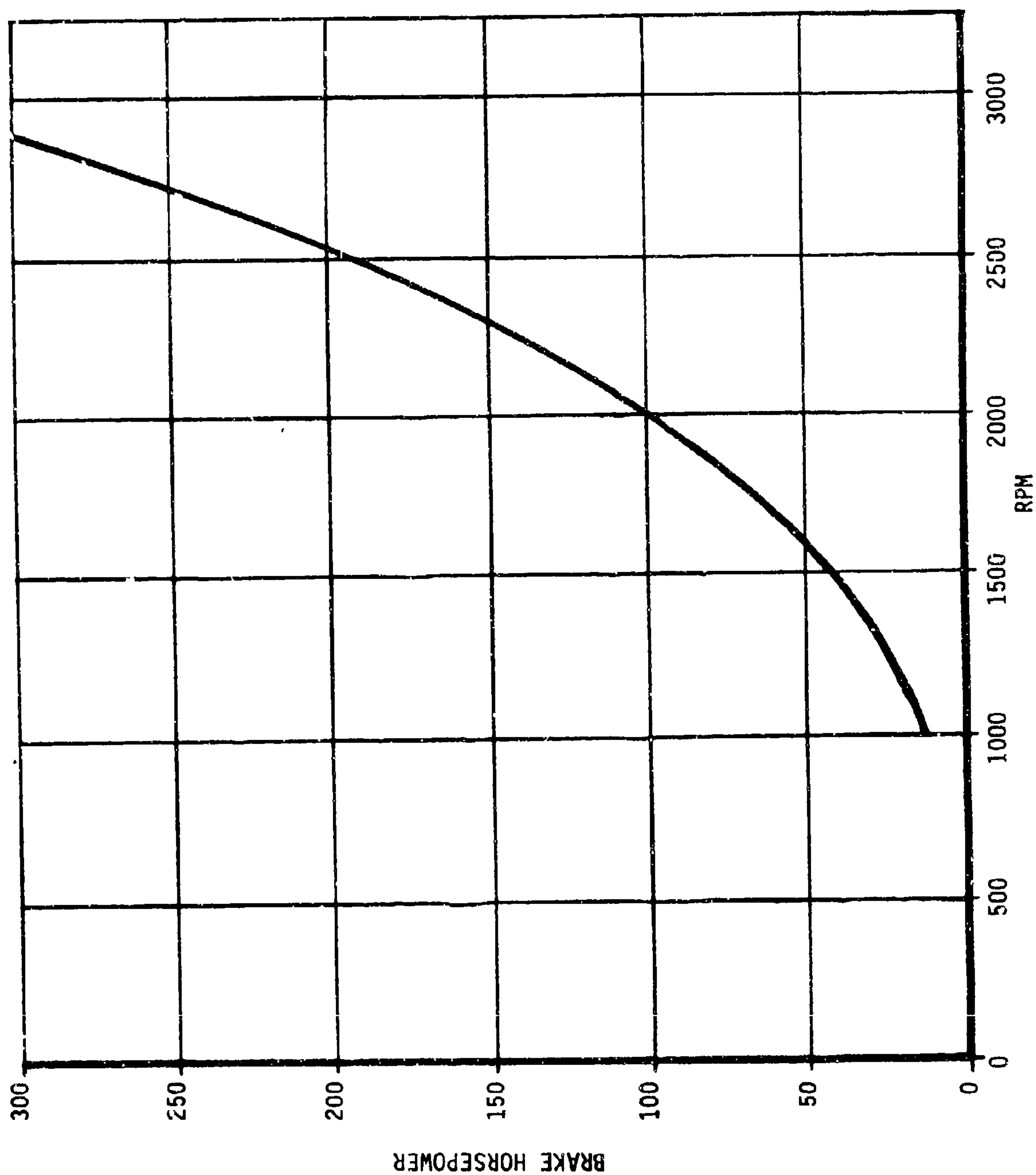

Figure 8.2.2-7 ATR Waterjet Power Absorption

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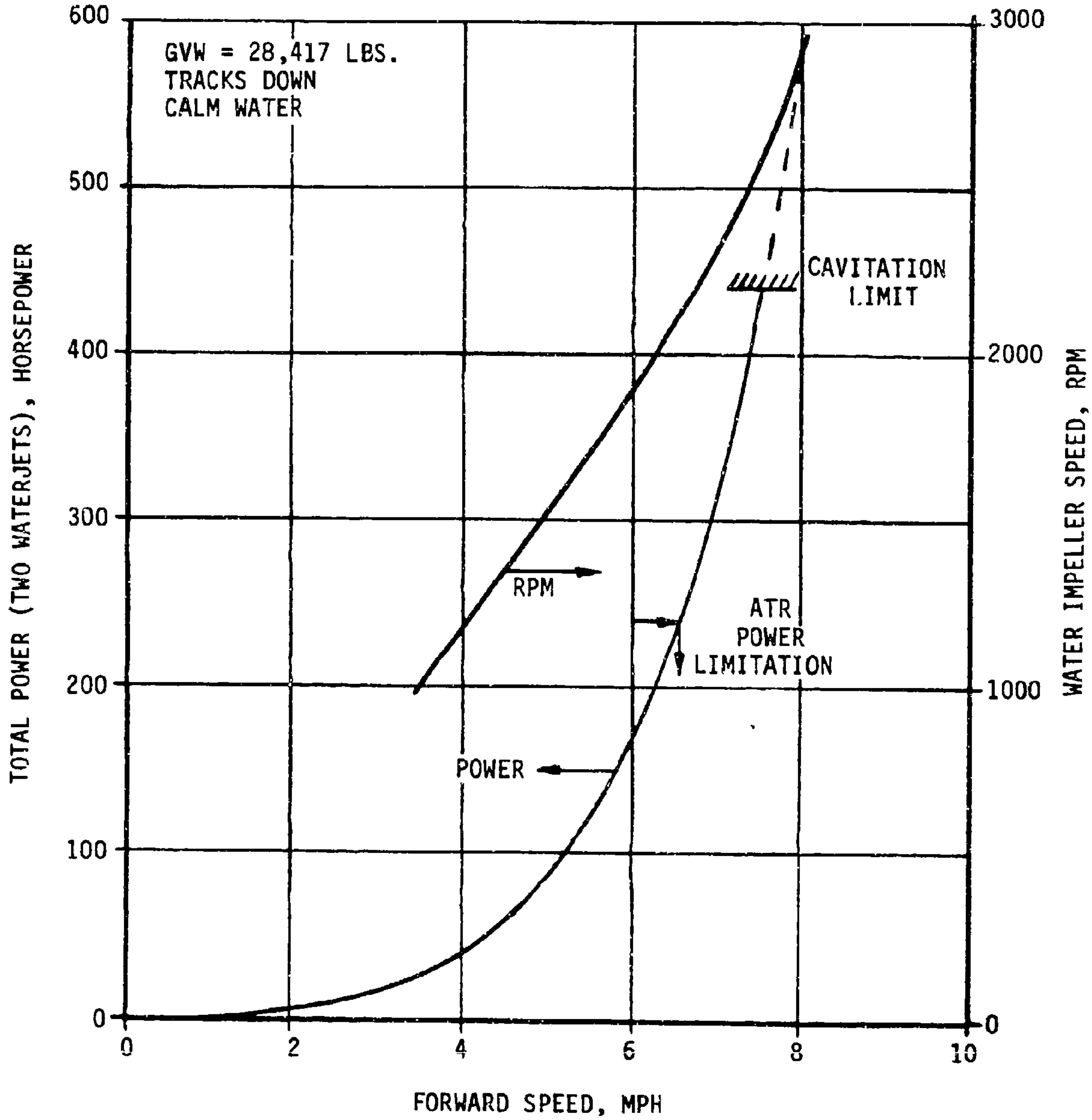

Figure 8.2.2-8 ATR Waterjet Performance Variation

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Table 8.2.2-3 Estimated Automotive Test Rig Maximum Forward Water Speeds

	SEA STATE 0				SEA STATE 2	
PAYLOAD CONFIGURATION	WITHOUT TROOPS		WITH TROOPS		WITHOUT TROOPS	WITH TROOPS
Gross Weight, lbs.	26,167		28,417		26,167	28,417
Track Position	DOWN	UP	DOWN	UP	DOWN UP	DOWN UP
Maximum Forward Speed, mph	6.7	7.1	6.55	6.9	6.05	5.85
Reference Figure	8.2.2-4	8.2.2-4	8.2.2-3	8.2.2-3	8.2.2-4	8.2.2-4

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In conclusion the principal limitation on the Automotive Test Rig's waterborne mobility is its inability to travel at forward speeds higher than 6 miles per hour because of the marked tendency for its bow to trim down and submerge. This tendency was demonstrated in the FLASH model tests. Most of the effort associated with these tests was devoted to developing a bow flap arrangement that would trim the bow up and thus avoid the submergence problem. The bow of the ATR is slightly higher than that of the FLASH model and its longitudinal center of gravity location is slightly further aft. Both of these factors will reduce the tendency of the ATR bow to bury but the amount of this improvement will only be known by conducting model or full scale tests.

The ability of the ATR to retract its tracks does not have much effect on the water speeds it can achieve. This capability, however, will be important if it is later desired to modify the ATR for higher water speeds. In order to achieve markedly higher (planing) speeds it will be necessary to make the following changes:

- o Increase engine power by a factor of about five.
- o Install larger waterjets lower in the vehicle, perhaps in a transom flap.
- o Install cover plates to close in the underside of the retracted tracks.
- o Install a large bow flap, perhaps backed by an inflatable bow.

8.3 Physical Characteristics

The physical characteristics and major components and systems of the Automotive Test Rig (ATR) are shown in Table 8.3-1.

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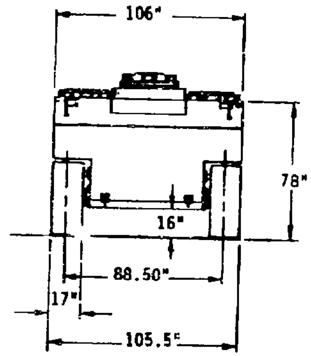
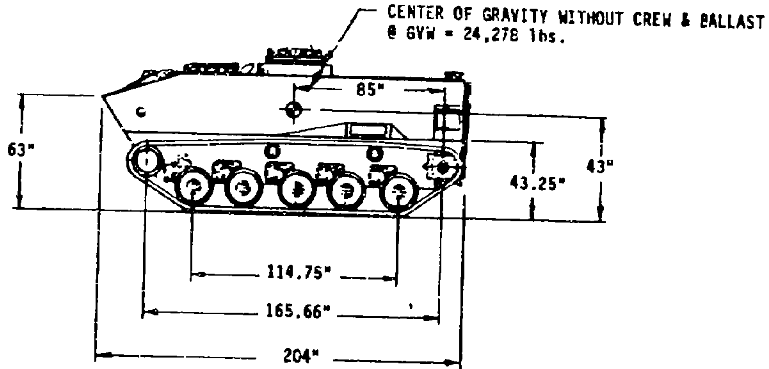
Table 8.3-1 Automotive Test Rig Physical Characteristics





WEIGHT:

With Troops: 30,285 lbs.
Without Troops: 28,035 lbs.

GROUND PRESSURE: 7.76 psi (with troops)

L/T RATIO: 1.30

GHP/TON: 21.1

ANGLE OF APPROACH: 67°

ANGLE OF DEPARTURE: 36°

ENGINE: Caterpillar Tractor Co. 3208T
320 GHP @ 2800 rpm, Four Stroke
Diesel Cycle, 8 Cylinders

TRANSMISSION: Hydrostatic Drive with Microcomputer Control

FINAL DRIVE: Two Speeds, High Range 4.480:1, Low Range 10.4765:1, Brakes

SUSPENSION:

TYPE - Rotary type Hydropneumatic with Front Idler and Rear Sprocket Drive

NO. OF STATIONS - 5/Side, Dual 22"

diameter x 3.5" Wide Rubber Tired Roadwheels

SUPPORT ROLLERS - 2/Side, 8.5" Diameter x 3.4" Wide

SPROCKET - 10 Teeth, 18.65" Pitch Diameter

TRACK - Wire Link Type, 5.81" Pitch, 32.8 lbs/foot, 69 Sections/Side

FRONT IDLER - Compensating with Track Tension Adjuster, 17.25" Diameter Idler Wheel

WATER PROPULSORS:

Two Stage Axial Flow Waterjets with Steering Buckets, 12" Diameter, 1/Side

BILGE PUMPS:

Two electrically driven and two hydraulically driven, 480 GPM total capacity

ELECTRICAL SYSTEM:

24 VDC, (4) Automotive Batteries Connected in Series Parallel

ALTERNATOR - 300 AMP, 24 VDC

FIRE PROTECTION SYSTEM:

TYPE - Automatic Crew & Engine Compartment Protection

SENSORS - 3 Channel, Discriminatory Type w/BITE

BOTTLES - Halon 1301 w/Solenoid Valves

PERSONNEL ACCOMODATIONS:

Driver and Commander Seats, Observer's Position

HYDRAULIC SYSTEM:

Two 1.52 cu. in. variable axial piston 3000 psi pressure compensated pumps to operate; cooling fans, suspension, rear ramp, engine grilles, waterjet steer, bilge pumps

FUEL CAPACITY: 100 Gallons, (1) 50 Gallon Tank each side

CREW ACCESS PROVISIONS:

Driver's & Commander's Hatches, Rear ramp with escape hatch

RECOVERY PROVISIONS: Lifting and towing lugs, front and rear

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Appendix A

Weight Properties Data

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INTRODUCTION

AAI uses its combat vehicle computer weight program to manage, control and monitor weight and its distribution during the ATR design process. The program allows computation of the center of gravity and moments of inertia of a body composed of groups, subgroups and components. Each group is a combination of subgroups and each subgroup is a combination of components. In the case of a combat vehicle, the groups, subgroups and components represent specific levels of the work breakdown structure (WBS). The group names correspond to the third level, and the subgroup names correspond to the fourth level of the WBS. The component name is input to the weight program along with its weight, the coordinates of its center of gravity, and its moment of inertia about its center of gravity. This information is combined and a center of gravity location and inertial properties are calculated for each group. Results for each group are combined and a center of gravity location and inertia properties are calculated for the complete vehicle.

The computer printouts of this analysis appear in the following pages and are organized in accordance with the ATR WBS. For each group, the subgroups with its corresponding components are listed. This is followed by a subgroup summary and is repeated for each group. Following the last subgroup summary is a group summary. A discussion of the weights analysis appears in Section 8.1.

In the ATR, the reference coordinate system used to locate the components was chosen such that the x-direction defines the station locations using the sprocket centerline as Station 250 with the positive direction rearward. The y-direction defines the butt line locations as positive to the right side of the vehicle centerline. The z-direction defines the waterline locations as positive from the ground level upward.

The inertial properties appearing in the computer printouts do not consider the individual inertia of each component about its own axis. For this reason, the inertia of the ATR is actually higher than predicted by the program by an estimated ten percent.

The "*" symbol following an item description indicates that the weight shown is the actual measured weight of that specified component.

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TITLE ATR WEIGHT ANALYSIS

WEIGHT AND C[illegible] R[illegible]

CONTRACT NO.: CW 4598

DATE: 12/20/85

SUMMARY OF OVERALL ASSEMBLY

GROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
INTEGRATION AND ASSEMBLY	0.000	0.000	0.000	0.000
HULL AND FRAME	7769.989	153.426	0.026	46.856
SUSPENSION	6372.301	163.957	0.000	22.234
PROPULSION PLANT	3402.000	114.822	-2.639	46.997
HYDROSTATIC DRIVE	3049.100	138.245	0.710	26.444
MARINE DRIVE TRAIN	640.000	217.986	0.000	50.676
AUXILIARY SYSTEM	1415.900	92.506	-0.246	25.007
TURRET AND ASSEMBLY	0.000	0.000	0.000	0.000
NAVIGATION AND COMMUNICATION	0.000	0.000	0.000	0.000
LOADED ITEMS	1098.000	0.000	0.000	0.000
BALLAST	6538.000	0.000	0.000	0.000

TOTAL WEIGHT (LB) 30285.291

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	109.609	-0.230	26.881

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WEIGHT AND C. R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS   CONTRACT NO.: CW 4598   DATE: 12/20/85

SUMMARY OF GROUP NO. 11 - BALLAST

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN)		
		$\bar{X}$	$\bar{Y}$	$\bar{Z}$
BALLAST	6538.000	0.000	0.000	0.000

TOTAL WEIGHT OF GROUP (LB)   6538.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND C[illegible]R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: BALLAST
GROUP NUMBER: 11

SUBGROUP NAME: BALLAST
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
INSTALLED ITEMS	0.000	0.000	0.000	0.000
PROTECTIVE COVERS	0.000	0.000	0.000	0.000
FINAL DRIVE COVERS	0.000	0.000	0.000	0.000
STBD *	28.000	0.000	0.000	0.000
PORT *	35.000	0.000	0.000	0.000
FORWARD COVERS *	0.000	0.000	0.000	0.000
STBD *	20.000	0.000	0.000	0.000
PORT *	22.000	0.000	0.000	0.000
WATERJET DRIVE HOSE COVERS	0.000	0.000	0.000	0.000
STBD *	5.000	0.000	0.000	0.000
PORT *	5.000	0.000	0.000	0.000
PROTECTIVE LINERS	10.000	0.000	0.000	0.000
OBSERVERS STATION	0.000	0.000	0.000	0.000
HATCH *	66.500	0.000	0.000	0.000
ADAPTER RING	75.000	0.000	0.000	0.000
MOUNTING BOLTS	5.000	0.000	0.000	0.000
COUPLA MOUNT	150.000	0.000	0.000	0.000
VISION BLOCKS (5 @ 7.0 LBS) *	35.000	0.000	0.000	0.000
TORSION BAR AND HARDWARE	5.000	0.000	0.000	0.000
SIMULATOR BOX	12.000	0.000	0.000	0.000
	0.000	0.000	0.000	0.000
PASSIVE NAVIGATION (GOVERNMENT SPECIFIED)	28.000	0.000	0.000	0.000
NBC EQUIPMENT (GOVERNMENT SPECIFIED)	200.000	0.000	0.000	0.000
OEM (GOVERNMENT SPECIFIED) (LESS PRO. COVERS, OBSERVERS STA., AND SIMULATOR BOX)	256.500	0.000	0.000	0.000
NAVIGATION AND COMMUNICATION (GOVERNMENT SPECIFIED)	330.000	0.000	0.000	0.000
TURRET (MCWS PREDICTED WEIGHT)	3000.000	0.000	0.000	0.000
TROOPS (10) (GOVERNMENT SPECIFIED)	2250.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 6538.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND C R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS CONTRACT NO.: CW 4598 DATE: 12/20/85

SUMMARY OF GROUP NO. 10 - LOADED ITEMS

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
FUEL	648.000	0.000	0.000	0.000
CREW	450.000	0.000	0.000	0.000

TOTAL WEIGHT OF GROUP (LB) 1098.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND C. .R CF GRAVITY

TITLE: ATR WEIGHT ANALYSIS   CONTRACT NO.: CW 4598   DATE: 12/20/85

GROUP NAME: LOADED ITEMS
GROUP NUMBER: 10

SUBGROUP NAME: CREW
SUBGROUP NUMBER: 2

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
COMMANDER	225.000	0.000	0.000	0.000
DRIVER	225.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 450.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)

$\bar{X}$	$\bar{Y}$	$\bar{Z}$
0.000	0.000	0.000

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WEIGHT AND C ER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: LOADED ITEMS
GROUP NUMBER: 10

SUBGROUP NAME: FUEL
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
FUEL (90 GAL. @ 7.2 LBS. / GAL.)	648.000	0.000	0.000	0.00

TOTAL WEIGHT OF SUBGROUP (LB) 648.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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TITLE: ATR WEIGHT ANALYSIS

WEIGHT AND [illegible]R OF GRAVITY

CONTRACT NO.: CW 4598

DATE: 12/20/85

SUMMARY OF GROUP NO. 9 – NAVIGATION AND COMMUNICATION

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
NAVIGATION AND COMMUNICATION	0.000	0.000	0.000	0.00

TOTAL WEIGHT OF GROUP (LB) 0.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND C. .R C.AVI.

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: NAVIGATION AND COMMUNICATION
GROUP NUMBER: 9

SUBGROUP NAME: NAVIGATION AND COMMUNICATION
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
NAVIGATION AND COMMUNICATION EQUIPMENT (SEE BALLAST GROUP 12, SUBGROUP 1)	0.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB): 0.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND C[illegible]R OF GRAVITY

TITLE ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

SUMMARY OF GROUP NO. 8 - TURRET AND ASSEMBLY

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
TURRET AND ASSEMBLY	0.000	0.000	0.000	0.000

TOTAL WEIGHT OF GROUP (LB) 0.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND C ER OF GRAVITY

TITLE. ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME. TURRET AND ASSEMBLY
GROUP NUMBER: 8

SUBGROUP NAME: TURRET AND ASSEMBLY
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
MCWS TURRET (SEE BALLAST GROUP 12, SUBGROUP 1)	0.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 0.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND C[illegible]R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO : CW 4598

DATE: 12/20/85

SUMMARY OF GROUP NO 7 - AUXILIARY SYSTEM

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
VEHICLE ELECTRIC SYSTEM	123.500	106.607	1.231	49.692
VEHICLE HYDRAULIC SYSTEM	715.500	122.517	0.000	29.913
FIRE DETECTION AND SUPPRESSION	118.100	147.942	-4.234	49.035
PASSIVE NIGHT SIGHT DEVICE	0.000	0.000	0.000	0.000
BILGE PUMPS	72.500	154.241	0.000	17.628
NBC EQUIPMENT	0.000	0.000	0.000	0.000
OEM	14.000	107.000	0.000	57.000
CONTROL AND EQUIPMENT	372.300	0.000	0.000	0.000

TOTAL WEIGHT OF GROUP (LB) 1415.900

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	92.506	-0.246	25.007

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WEIGHT AND CE R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4596

DATE: 12/20/85

GROUP NAME: AUXILIARY SYSTEM
GROUP NUMBER: 7

SUBGROUP NAME: CONTROL AND EQUIPMENT
SUBGROUP NUMBER: 8

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
ACCELERATOR CONTROL PEDAL	8.000	0.000	0.000	0.0
BRAKE ASSEMBLY *	15.500	0.000	0.000	0.0
JUNCTION BOX #1 *	21.000	0.000	0.000	0.0
JUNCTION BOX #2 *	10.000	0.000	0.000	0.0
JUNCTION BOX #3 *	17.200	0.000	0.000	0.0
MONITOR BOX *	9.600	0.000	0.000	0.0
JUNCTION BOX #4 *	10.500	0.000	0.000	0.0
CABLES *	91.000	0.000	0.000	0.0
AAI FABRICATED CABLE *	17.000	0.000	0.000	0.0
BATTERIES (2 @ 36.5 LBS) *	73.000	0.000	0.000	0.0
DRIVERS CONTROL PANEL *	21.000	0.000	0.000	0.0
DRIVERS INSTRUMENT PANEL *	15.000	0.000	0.000	0.0
DRIVERS DISPLAY PANEL *	2.500	0.000	0.000	0.0
ENGINEERS PANEL *	7.000	0.000	0.000	0.00
POWER DISTRIBUTION PANEL *	40.000	0.000	0.000	0.00
THROTTLE CONTROLLER *	14.000	0.000	0.000	0.00

TOTAL WEIGHT OF SUBGROUP (LB) 372.300

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND C. R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS   CONTRACT NO : CW 4598   DATE: 12/20/85

GROUP NAME: AUXILIARY SYSTEM   SUBGROUP NAME: OEM
GROUP NUMBER: 7   SUBGROUP NUMBER: 7

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
PORTABLE EXTINGUISHER	8.000	107.000	0.000	57.00
GRILL COVER	4.000	107.000	0.000	57.00
PADLOCK (2)	2.000	107.000	0.000	57.00

TOTAL WEIGHT OF SUBGROUP (LB)   14.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	107.000	0.000	57.000

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WEIGH AND C R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO. CW 4598

DATE: 12/20/85

GROUP NAME: AUXILIARY SYSTEM
GROUP NUMBER: 7

SUBGROUP NAME: NBC EQUIPMENT
SUBGROUP NUMBER: 6

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$

NO ENTRIES FOR THIS SUBGROUP

TOTAL WEIGHT OF SUBGROUP (LB) 0.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND CE .R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO : CW 4598

DATE: 12/20/85

GROUP NAME AUXILIARY SYSTEM
GROUP NUMBER: 7

SUBGROUP NAME: BILGE PUMPS
SUBGROUP NUMBER: 5

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN)		
		$\bar{X}$	$\bar{Y}$	$\bar{Z}$
PUMP FORWARD, ENPRO, ELECTRICAL *	25.500	65.000	0.000	18.00
PUMP AFT, ENPRO, ELECTRICAL *	25.500	250.000	0.000	18.00
PUMP, FORWARD, HYDRAULIC *	10.000	65.000	0.000	18.00
PUMP, AFT, HYDRAULIC *	10.000	250.000	0.000	18.00
SCREEN, HYDRAULIC BILGE PUMPS *	1.500	0.000	0.000	0.00

TOTAL WEIGHT OF SUBGROUP (LB): 72.500

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	154.241	0.000	17.628

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WEIGHT AND C.   R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO. CW 4598

DATE: 12/20/85

GROUP NAME: AUXILIARY SYSTEM
GROUP NUMBER: 7

SUBGROUP NAME: PASSIVE NIGHT SIGHT DEVICE
SUBGROUP NUMBER: 4

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$

NO ENTRIES FOR THIS SUBGROUP

TOTAL WEIGHT OF SUBGROUP (LB): 0.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)

$\bar{X}$	$\bar{Y}$	$\bar{Z}$
0.000	0.000	0.000

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WEIGHT AND C. .R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO. CW 4598

DATE: 12/20/85

GROUP NAME: AUXILIARY SYSTEM
GROUP NUMBER: 7

SUBGROUP NAME: FIRE DETECTION AND SUPPRESSION
SUBGROUP NUMBER: 3

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
SENSORS, PM-34 (4 @ .6 LBS.); CREW	2.400	205.000	0.000	70.0
SENSORS, PM-34 (2 @ .6 LBS.); ENGINE	1.200	100.000	0.000	70.0
BOTTLE W/SOL. VAL, 5 LB; CREW *	19.000	205.000	0.000	60.0
BOTTLES W/SOL. VAL, 5 LB; ENG (2 @ 19 LBS) *	38.000	150.000	0.000	60.0
CONTROL ELECTRONICS ASSY	5.000	165.000	-50.000	47.0
CONTROL PANEL	5.000	100.000	-50.000	60.0
WIRING HARNESS *	36.000	165.000	0.000	44.0
BOTTLE MOUNTING BRACKETS (3 @ 1.5 LBS) *	4.509	0.000	0.000	0.0
ANTI-RECOIL LUGS (3 @ 1.0 LBS)	3.000	0.000	0.000	0.0
ANTI-RECOIL PLUG HOLDER	1.000	0.000	0.000	0.0
DISPERSION NOZZLE	1.000	0.000	0.000	0.0
VALVE OUTLET ADAPTER (2 @ 1.0 LBS)	2.000	0.000	0.000	0.0

TOTAL WEIGHT OF SUBGROUP (LB) 118.100

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	147.942	-4.234	49.035

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F2 *	6.500	150.500	0.	34
F3 *	10.000	150.500	0.	34
F4 *	10.000	150.500	0.000	34
FC1	2.600	150.500	0.000	34.
FC2	2.600	150.500	0.000	34.
FC3	3.000	150.500	0.000	34.
FC5	2.500	150.500	0.000	34.
FC6	2.500	0.000	0.000	0.
FC7	2.500	0.000	0.000	0
FC8	2.500	0.000	0.000	0.
FC9	2.500	0.000	0.000	0.
FC10	2.500	0.000	0.000	0.
FC12	2.500	0.000	0.000	0.
M1. INCLUDED IN FAN WEIGHT	0.000	0.000	0.000	0.
M2. INCLUDED IN BILGE PUMP WEIGHT	0.000	0.000	0.000	0.
M3. INCLUDED IN BILGE PUMP WEIGHT	0.000	0.000	0.000	0.
M4. INCLUDED IN PUMP WEIGHT	0.000	0.000	0.000	0.
MV1, PARKER MVB20SV2	2.500	150.500	0.000	34.
MV2, PARKER MVB20SV2	2.500	150.500	0.000	34.
P1, REXROTH A10V23	35.000	150.500	0.000	34.
PR1	3.700	150.500	0.000	34.
RA1, FLO-TORK, 900, 020-0661X180	10.000	0.000	0.000	0.
RA2, FLO-TORK, 900, 020-0661X180	10.000	0.000	0.000	0.
RA3, FLO-TORK, 900, 020-0661X180	10.000	0.000	0.000	0.
RA4, FLO-TORK, 1800, 020-0661X90 (INCLUDED IN MANIFOLD PLATE WEIGHT)	0.000	0.000	0.000	0.
RA5, FLO-TORK, 1800, 020-0661X90 (INCLUDED IN MANIFOLD PLATE WEIGHT)	0.000	0.000	0.000	0.
RA6, SEE RAMP (GROUP 2, SUBGROUP 6)	0.000	0.000	0.000	0.
RA7, FLO-TORK, 900, 020-0661X180	10.000	0.000	0.000	0.
RV1, PARKER RPB00SHV	3.700	150.500	0.000	34.
RV2, PARKER RPB00SMV	3.700	150.500	0.000	34.
OIL COOLER CIRCULATION SYSTEM PUMP *	14.000	0.000	0.000	0.
FITTINGS	75.000	150.500	0.000	34.
HOSES	75.000	150.500	0.000	34.
TUBING	75.000	150.500	0.000	34.
ENTRAINED FLUID	65.000	120.000	0.000	52.
DISTRIBUTION MANIFOLDS	30.000	150.500	0.000	34.
SUBPLATES AND BOLT KITS	15.000	150.500	0.000	34.0
TIEDOWNS	15.000	150.500	0.000	34.0

TOTAL WEIGHT OF SUBGROUP (LB)	715.500		
CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$ 122.517	$\bar{Y}$ 0.000	$\bar{Z}$ 29.913

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WEIGHT AND C   R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS   CONTRACT NO.: CW 4598   DATE: 12/20/85

GROUP NAME: AUXILIARY SYSTEM
GROUP NUMBER: 7

SUBGROUP NAME VEHICLE HYDRAULIC SYSTEM
SUBGROUP NUMBER: 2

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
PRIMARY COOLING FAN CIRCUIT	0.000	0.000	0.000	0.C
A1, PARKER AB01B3TIA1	10.000	150.500	0.000	34.0
CV1, PARKER C1020	2.000	150.500	0.000	34.0
DV1	15.000	150.500	0.000	34.0
F2 *	6.500	150.500	0.000	34.0
F3 *	10.000	150.500	0.000	34.0
F4 *	10.000	150.500	0.000	34.0
M1, INCLUDED IN FAN WEIGHT	0.000	0.000	0.000	0.0
M2	3.000	0.000	0.000	0.0
MV1, PARKER MV820	2.500	150.500	0.000	34.0
P1, REXROTH A10V25	35.000	150.500	0.000	34.0
RV1	3.700	150.500	0.000	34.0
RV2	3.700	150.500	0.000	34.0
FC1	1.500	0.000	0.000	0.0
AUXILIARY HYDRAULIC SYSTEM	0.000	0.000	0.000	0.0
A1, PARKER AB01B3TIA1	10.000	150.500	0.000	34.0
A2 *	6.000	150.500	0.000	34.0
C1	5.000	260.000	0.000	55.0
C2	5.000	150.500	0.000	34.00
C3	5.000	150.500	0.000	34.00
CV1, PARKER, C1220	2.000	150.500	0.000	34.00
CV2	1.300	150.500	0.000	34.00
CV4	2.000	0.000	0.000	0.00
CV5	2.000	0.000	0.000	0.00
DV1 *	5.000	150.500	0.000	34.00
DV2 *	5.000	150.500	0.000	34.00
DV3 *	5.000	0.000	0.000	0.00
DV4 *	6.000	150.500	0.000	34.00
DV6 *	5.000	150.500	0.000	34.00
DV7 *	6.000	0.000	0.000	0.00
DV8 *	6.000	0.000	0.000	0.00
DV9 *	6.000	0.000	0.000	0.00
DV10 *	6.000	0.000	0.000	0.00
DV11 *	5.000	0.000	0.000	0.00
DV12 *	6.000	0.000	0.000	0.00
DV13 *	6.000	0.000	0.000	0.00
DV14, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.00
DV15, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.000
DV16, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.000
DV17, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.000
DV18, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.000
DV19, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.000
DV20, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.000
DV21, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.000
DV22, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.000
DV23, SEE SUSPENSION (GROUP 3, SUBGROUP 7)	0.000	0.000	0.000	0.000

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WEIGHT AND CE R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: AUXILIARY SYSTEM
GROUP NUMBER: 7

SUBGROUP NAME: VEHICLE ELECTRIC SYSTEM
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
WIRING (EST)	45.000	100.000	0.000	40.000
REGULATOR	3.000	36.000	36.000	30.000
L-HEAD LAMP 11589477-1	5.000	21.000	43.200	69.000
R-HEAD LAMP 11589477-1	5.000	21.000	-43.200	69.000
L-TAIL LAMP MS 52125-2 *	2.000	260.000	47.000	70.000
R-TAIL LAMP MS 52125-2 *	2.000	260.000	-47.000	70.000
L-PARK/TURN LAMP MS 52126-2	4.000	21.000	43.200	69.000
R-PARK/TURN LAMP MS 52126-2	4.000	21.000	-43.200	69.000
VEHICLE LIGHT SWITCH MS52128	1.000	45.000	44.000	70.000
DRIVERS DOME LIGHT	0.500	45.000	-44.000	70.000
COMMANDERS DOME LIGHT	0.500	45.000	44.000	70.000
TROOP COMPARTMENT DOME LIGHTS (3)	2.000	0.000	0.000	0.000
CABLES AND HARNESSES *	47.000	150.000	0.000	55.000
HORN *	2.500	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB): 123.500

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	106.607	1.231	49.692

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WEIGHT AND C[illegible]R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/95

SUMMARY OF GROUP NO. 6 - MARINE DRIVE TRAIN

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
MARINE PROPULSOR	410.000	227.390	0.000	51.[illegible]
MARINE MOTORS (2)	230.000	201.222	0.000	48.[illegible]

TOTAL WEIGHT OF GROUP (LB) 640.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	217.986	0.000	50.676

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WEIGHT AND C R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO. CW 4598

DATE: 12/20/85

GROUP NAME: MARINE DRIVE TRAIN
GROUP NUMBER: 6

SUBGROUP NAME MARINE MOTORS (2)
SUBGROUP NUMBER: 2

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
MARINE MOTOR, LINDE BMF100; STBD *	83.500	213.000	0.000	51.800
MARINE MOTOR, LINDE BMF100; PORT *	83.500	213.000	0.000	51.800
SCE (GROUP 7, SUBGROUP 2) FOR HYDRAULICS	0.000	0.000	0.000	0.000
CONTROL LINKAGE	5.000	250.000	0.000	60.000
FLEX COUPLINGS (2) *	43.000	220.000	0.000	51.800
VALENCE ASSEMBLY (2) *	15.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 230.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	201.222	0.000	48.600

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WEIGHT AND C R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: MARINE DRIVE TRAIN
GROUP NUMBER: 6

SUBGROUP NAME: MARINE PROPULSOR
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
HYDROJET ASSEMBLY, DOWTY 300; STBD *	172.500	223.130	0.000	51.840
HYDROJET ASSEMBLY DOWTY 300; PORT *	172.500	223.130	0.000	51.840
HYDROJET STEER; STBD *	32.500	250.000	0.000	51.840
HYDROJET STEER; PORT *	32.500	250.000	0.000	51.840

TOTAL WEIGHT OF SUBGROUP (LB): 410.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	227.390	0.000	51.840

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WEIGHT AND C   R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

SUMMARY OF GROUP NO. 5 - HYDROSTATIC DRIVE

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
PUMPS (2)	341.000	133.953	6.620	28.635
MOTORS (2)	376.000	234.000	0.000	28.250
TRANSFER CASE	312.600	126.766	-0.298	30.371
CONTROLLER	128.000	47.611	0.000	12.852
HYDRAULIC LINES	517.000	0.000	0.000	0.000
HYDRAULIC FLUID	516.000	173.721	0.000	61.767
FINAL DRIVE	858.500	177.635	0.000	20.073

TOTAL WEIGHT OF GROUP (LB) 3049.100

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	138.245	0.710	26.444

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WEIGHT AND C R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO. . CW 4598

DATE: 12/20/85

GROUP NAME: HYDROSTATIC DRIVE
GROUP NUMBER: 5

SUBGROUP NAME. FINAL DRIVE
SUBGROUP NUMBER: 7

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
FINAL DRIVE; STBD *	305.000	250.000	0.000	28.250
FINAL DRIVE; PORT *	305.000	250.000	0.000	28.250
HYDRAULIC SYSTEM	0.000	0.000	0.000	0.000
ACCUMULATOR, A1 (WITH MOUNT) *	8.000	0.000	0.000	0.000
CHECK VALVE, CV1	1.000	0.000	0.000	0.000
CV5	1.000	0.000	0.000	0.000
FILTER, F2	10.000	0.000	0.000	0.000
FLOW CONTROL, FC1	1.000	0.000	0.000	0.000
FC2	1.000	0.000	0.000	0.000
FC3	1.000	0.000	0.000	0.000
FC4	1.000	0.000	0.000	0.000
GAGE, G1	0.500	0.000	0.000	0.000
G2	0.500	0.000	0.000	0.000
HEAT EXCHANGER, HE1	25.000	0.000	0.000	0.000
MANVAL PUMP, MP1	2.500	0.000	0.000	0.000
MANVAL VALVE, MV1	0.500	0.000	0.000	0.000
MV2	0.500	0.000	0.000	0.000
MV3	0.500	0.000	0.000	0.000
PUMPS, P1 AND P2 (WITH MOTOR AND BEARINGS) *	37.000	0.000	0.000	0.000
RELIEF VALVE, RV1	1.000	0.000	0.000	0.000
RV2	1.000	0.000	0.000	0.000
RV3	1.000	0.000	0.000	0.000
DIRECTIONAL VALVES, DV1 *	4.500	0.000	0.000	0.000
DV2 *	15.000	0.000	0.000	0.000
RESERVOIR	25.000	0.000	0.000	0.000
HOSES, FITTINGS, AND TUBES	50.000	0.000	0.000	0.000
FLUID	60.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 858.500

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	177.635	0.000	20.073

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WEIGHT AND C... ER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.. CW 4598

DATE: 12/20/85

GROUP NAME: HYDROSTATIC DRIVE
GROUP NUMBER: 5

SUBGROUP NAME: HYDRAULIC FLUID
SUBGROUP NUMB_R: 6

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN)		
		$\bar{X}$	$\bar{Y}$	$\bar{Z}$
HYDRAULIC FLUID	400.000	180.000	0.000	64.00
RESERVIOR; PORT *	49.000	180.000	0.000	64.00
RESERVIOR; STBD *	49.000	180.000	0.000	64.00
SERVO VENT FILTER, F5 (2)	18.000	0.000	0.000	0.00

TOTAL WEIGHT OF SUBGROUP (LB): 516.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	173.721	0.000	61.767

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WEIGHT AND C R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: HYDROSTATIC DRIVE
GROUP NUMBER: 5

SUBGROUP NAME: HYDRAULIC LINES
SUBGROUP NUMBER: 5

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
MODE CONTROL ASSEMBLY (MTG. PLATE, VALVES, BLOCK, TUBING, ACTUATORS)	255.000	0.000	0.000	0.000
HOSES, FITTINGS AND HOSE	0.000	0.000	0.000	0.000
PORT: PUMP TO LAND MOTOR, XT-6 *	28.000	0.000	0.000	0.000
MOTOR TO CONTROL ASSEMBLY, XT-6 *	20.000	0.000	0.000	0.000
MOTOR BYPASS, XT-6 *	19.000	0.000	0.000	0.000
MARINE MOTOR SUPPLY, XT-5 *	12.000	0.000	0.000	0.000
MARINE MOTOR RETURN, XT-5 *	12.000	0.000	0.000	0.000
PUMP RETURN, XT-5 *	8.000	0.000	0.000	0.000
STBD: PUMP TO LAND MOTOR, XT-6 *	28.000	0.000	0.000	0.000
MOTOR TO CONTROL ASSEMBLY, XT-6 *	20.000	0.000	0.000	0.000
MOTOR BYPASS, XT-6 *	19.000	0.000	0.000	0.000
MARINE MOTOR SSUPPLY, XT-5 *	12.000	0.000	0.000	0.000
MARINE MOTOR RETURN, XT-5 *	12.000	0.000	0.000	0.000
PUMP RETURN, XT-5 *	8.000	0.000	0.000	0.000
SPLIT FLANGES (24 PAIR) *	24.000	0.000	0.000	0.000
MOUNTING BOLTS	25.000	0.000	0.000	0.000
TIEDOWNS	15.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 517.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO : CW 4598

DATE: 12/20/85

GROUP NAME: HYDROSTATIC DRIVE
GROUP NUMBER: 5

SUBGROUP NAME: CONTROLLER
SUBGROUP NUMBER: 4

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN)		
		$\bar{X}$	$\bar{Y}$	$\bar{Z}$
SC-1 COMPUTER *	35.000	174.120	0.000	47.00
SC-1 COMPUTER ENCLOSURE *	61.000	0.000	0.000	0.00
TERRA COMPUTER *	12.000	0.000	0.000	0.00
TERRA COMPUTER MOUNTING BRACKET *	3.000	0.000	0.000	0.00
MILTOPE RECORDER *	17.000	0.000	0.000	0.00

TOTAL WEIGHT OF SUBGROUP (LB) 128.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	47.611	0.000	12.952

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WEIGHT AND C R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO. CW 4598

DATE: 12/20/85

GROUP NAME: HYDROSTATIC DRIVE
GROUP NUMBER: 5

SUBGROUP NAME: TRANSFER CASE
SUBGROUP NUMBER: 3

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
SPLITTER BOX ASSEMBLY, TWIN DISC F11PMD2 *	304.000	127.500	-0.300	30.700
LUBRI..NT, 1 GALLON	6.800	127.500	-0.300	23.700
INSTALLATION HARDWARE *	1.800	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 312.600

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	126.766	-0.298	30.271

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WEIGHT AND C... ..R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: HYDROSTATIC DRIVE
GROUP NUMBER: 5

SUBGROUP NAME: MOTORS (2)
SUBGROUP NUMBER: 2

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
MOTOR, LINDE BMV 186, STBD *	188.000	234.000	0.000	28.250
MOTOR, LINDE BMV 186, PORT *	188.000	234.000	0.000	28.250

TOTAL WEIGHT OF SUBGROUP (LB) 376.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	234.000	0.000	28.250

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WEIGHT AND C. R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO : CW 4598

DATE: 12/20/85

GROUP NAME: HYDROSTATIC DRIVE
G OUP NUMBER: 5

SUBGROUP NAME: PUMPS (2)
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
PUMP, LINDE BPV 100 (DRY) WITH MANN FILTER; STBD *	165.500	138.000	6.820	29.500
PUMP, LINDE BPV 100 (DRY) WITH MANN FILTER; PORT *	165.500	138.000	6.820	29.500
SERVO VALVES (2)	10.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 341.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	133.953	6.620	28.635

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WEIGHT AND CE..ER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

SUMMARY OF GROUP NO 4 - PROPULSION PLANT

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
ENGINE	1612.600	107.822	-0.300	39.317
INDUCTION/EXHAUST	167.500	74.117	15.962	27.681
AUTO COOLING	690.000	103.274	0.000	63.855
MARINE COOLING	253.800	52.227	-0.619	40.581
ENGINE CONTROL	24.600	95.770	-44.683	63.301
FUEL SYSTEM	315.600	211.591	0.000	64.298
ENGINE ELECTRAL	101.900	96.563	0.000	30.682
BATTERIES AND MOUNT	236.000	173.084	-42.000	53.000

TOTAL WEIGHT OF GROUP (LB) 3402.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	114.822	-2.639	46.997

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WEIGHT AND C. .R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS | CONTRACT NO : CW 4598 | DATE: 12/20/85

GROUP NAME: PROPULSION PLANT
GROUP NUMBER: 4

SUBGROUP NAME: BATTERIES AND MOUNT
SUBGROUP NUMBER: 8

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN)		
		$\bar{X}$	$\bar{Y}$	$\bar{Z}$
GROUP 27F BATTERIES (4 @ 36.5 LBS) *	146.000	173.120	-42.000	53.000
COMPARTMENT	20.000	173.120	-42.000	53.000
POWER JUNCTION BOX	20.000	173.000	-42.000	53.000
MISCELLANEOUS POWER WIRING	50.000	173.000	-42.000	53.000

TOTAL WEIGHT OF SUBGROUP (LB) 236.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	173 084	-42.000	53.000

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WEIGHT AND C R OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO : CW 4598

DATE: 12/20/85

GROUP NAME: PROPULSION PLANT
GROUP NUMBER: 4

SUBGROUP NAME: ENGINE ELECTRAL
SUBGROUP NUMBER: 7

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
STARTER MOTOR CABLE	5.000	140.000	0.000	37.000
MISCELLANEOUS INSTRUMENT LEADS	2.000	102.000	0.000	37.000
ALTERNATOR *	77.500	115.300	0.000	37.000
ALTERNATOR BRACKET, ADJUSTING BOLT, AND HARDWARE *	16.500	0.000	0.000	0.000
BELTS (3) (GATES #9430) *	0.900	0.000	0.000	0.000

TOTAL WEIGHT OF S.. GROUP (LB) 101.900

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	96.563	0.000	30.682

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TITLE: ATR WEIGHT ANALYSIS | CONTRACT NO.: CW 459B | DATE: 12/20/85

GROUP NAME: PROPULSION PLANT
GROUP NUMBER: 4

SUBGROUP NAME: FUEL SYSTEM
SUBGROUP NUMBER: 6

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
FUEL TANK, STAINLESS; STBD *	94.100	220.500	0.000	70.000
FUEL TANK, STAINLESS; PORT *	94.100	[illegible] 500	0.000	70.000
LINES, FITTINGS, AND HARDWARE	13.800	[illegible].000	0.000	0.000
MANUAL SHUT OFF VALVES (6) *	3.800	50.000	0.000	50.000
FILLER CAPS, NECKS, STRAINER, AND BALLISTIC COVERS *	29.800	250.000	0.000	58.000
TANK SUPPORT	80.000	220.500	0.000	65.000

TOTAL WEIGHT OF SUBGROUP (LB) 315.600

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	211.591	0.000	64.298

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS   CONTRACT NO.: CW 4598   DATE: 12/20/85

GROUP NAME: PROPULSION PLANT   SUBGROUP NAME: ENGINE CONTROL
GROUP NUMBER: 4   SUBGROUP NUMBER: 5

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
ENGINE TACHOMETER AND SENDING UNIT *	1.400	102.880	-48.000	68.000
OIL TEMPERATURE GAGE	0.250	102.880	-48.000	68.000
COOLANT TEMPERATURE GAGE	0.250	102.880	-48.000	68.000
VOLT/AMMETER	0.250	102.880	-48.000	68.000
SPEEDOMETER	0.500	102.880	-48.000	68.000
FUEL GAGE	0.250	102.880	-48.000	68.000
INSTRUMENT PANEL	5.000	102.880	-48.000	68.000
INSTRUMENT WIRING	15.000	102.880	-48.000	68.000
ENGINE OIL PRESSURE TRANSMITTER, SWITCHES, AND FITTINGS *	1.100	0.000	0.000	0.000
COOLANT TEMPERATURE TRANSMITTER (OMEGA) *	0.600	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB)   24.600

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN) $\bar{X}$ 95.770 $\bar{Y}$ -44.683 $\bar{Z}$ 63.301

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4398

DATE: 12/20/85

GROUP NAME: PROPULSION PLANT
GROUP NUMBER: 4

SUBGROUP NAME: MARINE COOLING
SUBGROUP NUMBER: 4

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
ENGINE HEAT EXCHANGER *	55.000	78.900	0.000	63.000
HYDROSTATIC HEAT EXCHANGER (2 @ 23.9 LBS) *	47.800	78.900	0.000	65.000
HOSES, FITTINGS, AND HARDWARE *	52.100	78.900	0.000	64.000
SEA WATER PUMP *	13.100	78.900	-12.000	30.000
ENCLOSURE ASSEMBLY *	85.800	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB): 253.800

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	52.227	-0.619	40.581

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: PROPULSION PLANT
GROUP NUMBER: 4

SUBGROUP NAME: AUTO COOLING
SUBGROUP NUMBER: 3

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
NOAAH 41200-1 MIX FAN; STBD *	53.000	132.620	0.000	70.000
NOAAH 41200-1 MIX FAN; PORT *	53.000	132.620	0.000	70.000
ENGINE COOLANT RADIATOR TO AIR *	260.000	102.890	0.000	62.000
SWIVEL JOINTS (2) *	28.000	0.000	0.000	0.000
COOLANT MIX (20.75 GAL @ 8.5 LBS/GAL) *	176.400	102.890	0.000	70.000
HOSES, CLAMPS, AND HARDWARE *	19.600	102.880	0.000	70.000
HYDROSTATIC OIL RADIATOR TO AIR *	100.000	102.890	0.000	68.000

TOTAL WEIGHT OF SUBGROUP (LB): 690.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	103.274	0.000	63.85

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO : CW 4598

DATE: 12/20/85

GROUP NAME: PROPULSION PLANT
GROUP NUMBER: 4

SUBGROUP NAME: INDUCTION/EXHAUST
SUBGROUP NUMBER: 2

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
DONALDSON AIR CLEANER EBA09-2001 *	18.500	172.000	43.000	65.750
AIR INLET DUCTING, CLAMPS, AND HARDWARE *	46.300	172.000	43.000	65.750
DONALDSON MUFFLER MUM09-0071	9.400	135.000	-12.000	40.000
INTERNAL EXHAUST PIPES AND HARDWARE *	28.200	0.000	0.000	0.000
INSULATION *	50.800	0.000	0.000	0.000
EXTERNAL EXHAUST PIPES AND HARDWARE *	14.300	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 167.500

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	74.117	15.962	27.681

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: PROPULSION PLANT
GROUP NUMBER: 4

SUBGROUP NAME: ENGINE
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
ENGINE				
CATERPILLAR 3208T (DRY) *	0.000	0.000	0.000	0.000
OIL (5 GAL @ 6.8 LB/GAL)	1540.000	107.600	-0.300	39.700
ENGINE MOUNTING	34.000	107.600	-0.300	39.700
FRONT SUPPORT ASSEMBLY *	0.000	0.000	0.000	0.000
REAR SUPPORT ASSEMBLY (2) *	5.900	90.800	-0.300	23.700
HARDWARE, FRONT MOUNT (LOT) *	24.000	123.500	-0.300	23.700
HARDWARE, REAR MOUNT (LOT) *	1.900	90.800	-0.300	23.700
	6.800	123.500	-0.300	23.700

TOTAL WEIGHT OF SUBGROUP (LB) 1612.600

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	107.822	-0.300	39 317

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

SUMMARY OF GROUP NO 3 - SUSPENSION

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
SUSPENSION DEVICES	1536.301	157.287	0.000	22.250
ROAD WHEEL ASSEMBLY	1010.000	170.182	0.000	16.997
DRIVE SPROCKET ASSEMBLY	432.800	250.000	0.000	28.250
IDLER ASSEMBLY	454.000	87.797	0.000	28.486
TRACK	2194.200	166.500	0.000	23.000
INTEGRATION & ASSEMBLY & ROAD ARMS	612.000	159.395	0.000	19.290
HYD. PNEUMATIC SUSPENSION RESERVOIR	132.500	152.732	0.000	21.826

TOTAL WEIGHT OF GROUP (LB) 6372.301

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	163.957	0.000	22.234

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: SUSPENSION
GROUP NUMBER: 3

SUBGROUP NAME: HYD. PNEUMATIC SUSPENSION RESERVOIR
SUBGROUP NUMBER: 7

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
HYDRAULICS SYSTEM	0.000	0.000	0.000	0.000
ISOLATION VALVES	0.000	0.000	0.000	0.000
DV14, DOUBLE A *	6.000	170.000	0.000	24.000
DV15, DOUBLE A *	6.000	170.000	0.000	24.000
DV16, DOUBLE A *	6.000	170.000	0.000	24.000
DV17, DOUBLE A *	6.000	170.000	0.000	24.000
DV18, DOUBLE A *	6.000	170.000	0.000	24.000
DV19, DOUBLE A *	6.000	170.000	0.000	24.000
DV20, DOUBLE A *	6.000	170.000	0.000	24.000
DV21, DOUBLE A *	6.000	170.000	0.000	24.000
DV22, DOUBLE A *	6.000	170.000	0.000	24.000
DV23, DOUBLE A *	6.000	170.000	0.000	24.000
REGULATOR, 2000 PSI	0.000	0.000	0.000	0.000
FRI	4.000	150.500	0.000	30.000
MANIFOLD BLOCKS (2) *	13.000	0.000	0.000	0.000
BOLT KITS	0.500	170.000	0.000	24.000
TUBING	10.000	170.000	0.000	24.000
FITTINGS	15.000	170.000	0.000	24.000
FLEX HOSES	15.000	170.000	0.000	24.000
ENTRAINED FLUID	15.000	170.000	0.000	24.000

TOTAL WEIGHT OF SUBGROUP (LB) 132 500

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)

$\bar{X}$	$\bar{Y}$	$\bar{Z}$
152 732	0.000	21.826

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: SUSPENSION
GROUP NUMBER: 3

SUBGROUP NAME: INTEGRATION & ASSEMBLY & ROAD ARMS
SUBGROUP NUMBER: 6

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
#1 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); STBD *	64.000	102.870	38.000	19.290
#1 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); PORT *	64.000	102.870	-38.000	19.290
#2 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); STBD *	60.000	130.850	38.000	19.290
#2 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); PORT *	60.000	130.850	-38.000	19.290
#3 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); STBD *	60.000	160.230	38.000	19.290
#3 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); PORT *	60.000	160.230	-38.000	19.290
#4 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); STBD *	62.000	188.230	38.000	19.290
#4 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); PORT *	62.000	188.230	-38.000	19.290
#5 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); STBD *	60.000	217.600	38.000	19.290
#5 ROAD ARM, SPINDLE, HUB, BRGS, NUTS, BOLTS, AND LUB.); PORT *	60.000	217.600	-38.000	19.290

TOTAL WEIGHT OF SUBGROUP (LB) 612.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	159.395	0.000	19.290

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WEIGHT AND CENTER OF GRAVITY

TITLE: ?TR WEIGHT ANALYSIS | CONTRACT NO.: CW 4598 | DATE. 12/20/85

GROUP NAME: SUSPENSION
GROUP NUMBER: 3

SUBGROUP NAME: TRACK
SUBGROUP NUMBER: 5

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
WIRE LINK TRACK (69P @ 15.9 LBS/P);STBD *	1097.100	166.500	43.500	23.000
WIRE LINK TRACK (69P @ 15.9 LBS/P),PORT *	1097.100	166.500	-43.500	23.000

TOTAL WEIGHT OF SUBGROUP (LB) 2194.200

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{Y}$	$\bar{Y}$	$\bar{Z}$
	166.500	0.000	23.000

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: SUSPENSION
GROUP NUMBER: 3

SUBGROUP NAME: IDLER ASSEMBLY
SUBGROUP NUMBER: 4

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
IDLER WHEELS (2), GFE;STBD *	89.000	84.340	0.000	27.420
IDLER WHEELS (2), GFE;PORT *	89.000	84.340	0.000	27.420
IDLER ARM ASSEMBLY (HUB, SPINDLE, BRGS, NUTS, BOLTS, LUB., AND MTG.),STBD *	78.000	84.340	0.000	32.000
IDLER ARM ASSEMBLY (HUB, SPINDLE, BRGS, NUTS, BOLTS, LUB., AND MTG.);PORT *	78.000	84.340	0.000	32.000
TRACK ADJUSTER/COMPENSATOR;STBD *	60.000	97.420	0.000	25.500
TRACK ADJUSTER/COMPENSATOR;PORT *	60.000	97.420	0.000	25.500

TOTAL WEIGHT OF SUBGROUP (LB): 454.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	87.797	0.000	28.486

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS | CONTRACT NO.: CH 4398 | DATE: 12/20/85

GROUP NAME: SUSPENSION
GROUP NUMBER: 3

SUBGROUP NAME: DRIVE SPROCKET ASSEMBLY
SUBGROUP NUMBER: 3

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
SPROCKET ASSEMBLY (WHEELS, CARRIER, AND TIRES); STBD *	215.000	250.000	43.000	28.250
SPROCKET ASSEMBLY (WHEELS, CARRIER, AND TIRES); PORT *	215.000	250.000	-43.000	28.250
CARRIER MOUNTING BOLTS; STBD *	1.400	250.000	43.000	28.250
CARRIER MOUNTING BOLTS; PORT *	1.400	250.000	-43.000	28.250

TOTAL WEIGHT OF SUBGROUP (LB): 432.800

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	250.000	0.000	28.250

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS | CONTRACT NO : CH 4598 | DATE: 12/20/85

GROUP NAME: SUSPENSION
GROUP NUMBER: 3

SUBGROUP NAME: ROAD WHEEL ASSEMBLY
SUBGROUP NUMBER: 2

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
#1 ROAD WHEELS (2); STBD *	84.000	110.500	42.500	13.380
#1 ROAD WHEELS (2); PORT *	84.000	110.500	-42.500	13.380
#2 ROAD WHEELS (2); STBD *	84.000	138.500	42.500	13.380
#2 ROAD WHEELS (2); PORT *	84.000	138.500	-42.500	13.380
#3 ROAD WHEELS (2); STBD *	84.000	167.880	42.500	13.380
#3 ROAD WHEELS (2); PORT *	84.000	167.880	-42.500	13.380
#4 ROAD WHEELS (2); STBD *	84.000	195.880	42.500	13.380
#4 ROAD WHEELS (2); PORT *	84.000	195.880	-42.500	13.380
#5 ROAD WHEELS (2); STBD *	84.000	225.250	42.500	13.380
#5 ROAD WHEELS (2); PORT *	84.000	225.250	-42.500	13.380
FORWARD SUPPORT ROLLER AND MNT ; STBD *	42.500	134.240	39.000	34.870
FORWARD SUPPORT ROLLER AND MNT ; PORT *	42.500	134.240	-39.000	34.870
AFT SUPPORT ROLLER AND MNT. ; STBD *	42.500	211.620	39.000	34.870
AFT SUPPORT ROLLER AND MNT. ; PORT *	42.500	211.620	-39.000	34.870

TOTAL WEIGHT OF SUBGROUP (LB) 1010.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	170.182	0.000	16.997

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: SUSPENSION
GROUP NUMBER: 3

SUBGROUP NAME: SUSPENSION DEVICES
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
#1 HYDRO UNIT; STBD *	150.000	101.030	30.000	22.250
#1 HYDRO UNIT; PORT *	150.000	101.030	-30.000	22.250
#2 HYDRO UNIT; STBD *	150.000	126.030	30.000	22.250
#2 HYDRO UNIT, PORT *	150.000	128.030	-30.000	22.250
#3 HYDRO UNIT; STBD *	150.000	157.410	30.000	22.250
#3 HYDRO UNIT; PORT *	150.000	157.410	-30.000	22.250
#4 HYDRO UNIT, STBD *	150.000	185.410	30.000	22.250
#4 HYDRO UNIT; PORT *	150.000	185.410	-30.000	22.250
#5 HYDRO UNIT; STBD *	150.000	214.700	30.000	22.250
#5 HYDRO UNIT; PORT *	150.000	214.700	-30.000	22.250
#1 HYDRO SHAFT SEAL; STBD, EST	2.000	99.030	0.000	22.250
#1 HYDRO SHAFT SEAL; PORT, EST	2.000	99.030	0.000	22.250
#2 HYDRO SHAFT SEAL; STBD, EST	2.000	127.030	0.000	22.250
#2 HYDRO SHAFT SEAL; PORT, EST	2.000	127.030	0.000	22.250
#3 HYDRO SHAFT SEAL, STBD, EST	2.000	156.410	0.000	22.250
#3 HYDRO SHAFT SEAL; PORT, EST	2.000	156.410	0.000	22.250
#4 HYDRO SHAFT SEAL; STBD, EST	2.000	184.410	0.000	22.250
#4 HYDRO SHAFT SEAL; PORT, EST	2.000	184.410	0.000	22.250
#5 HYDRO SHAFT SEAL; STBD, EST	2.000	213.700	0.000	22.250
#5 HYDRO SHAFT SEAL; PORT, EST	2.000	213.700	0.000	22.250
#1 SHAFT END CAP AND BOLT; STBD	1.680	99.030	0.000	22.250
#1 SHAFT END CAP AND BOLT; PORT	1.680	99.030	0.000	22.250
#2 SHAFT END CAP AND BOLT; STBD	1.680	127.030	0.000	22.250
#2 SHAFT END CAP AND BOLT; PORT	1.680	127.030	0.000	22.250
#3 SHAFT END CAP AND BOLT; STBD	1.680	156.410	0.000	22.250
#3 SHAFT END CAP AND BOLT; PORT	1.680	156.410	0.000	22.250
#4 SHAFT END CAP AND BOLT; STBD	1.680	184.410	0.000	22.250
#4 SHAFT END CAP AND BOLT, PORT	1.680	184.410	0.000	22.250
#5 SHAFT END CAP AND BOLT; STBD	1.680	213.700	0.000	22.250
#5 SHAFT END CAP AND BOLT; PORT	1.680	213.700	0.000	22.250

TOTAL WEIGHT OF SUBGROUP (LB) 1536.801

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN) $\bar{X}$ 157.287 $\bar{Y}$ 0.000 $\bar{Z}$ 22.250

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS   CONTRACT NO.: CW 4598   DATE: 12/20/65

SUMMARY OF GROUP NO 2 - HULL AND FRAME

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
HULL WELDED AND MACHINED	6022.189	162.771	0.034	47.063
HULL BOLTED AND MISCELLANEOUS	431.000	118.566	0.000	66.879
BULKHEAD AND INTERNAL COVERS	304.300	43.822	0.000	5.144
ACCOMODATIONS	191.000	128.288	0.000	58.325
APPENDAGES	343.000	118.788	0.000	63.627
RAMP	403.500	174.501	0.000	34.125
CARGO HATCH	0.000	0.000	0.000	0.000
INTEGRATION AND ASSEMBLY	75.000	157.120	0.000	47.000
BOW FLAP	0.000	0.000	0.000	0.000

TOTAL WEIGHT OF GROUP (LB)   7769.989

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	153.426	0.026	46.856

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: HULL AND FRAME
GROUP NUMBER: 2

SUBGROUP NAME: BOW FLAP
SUBGROUP NUMBER: 9

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$

NO ENTRIES FOR THIS SUBGROUP

TOTAL WEIGHT OF SUBGROUP (LB): 0.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)

$\bar{X}$	$\bar{Y}$	$\bar{Z}$
0.000	0.000	0.000

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS　　CONTRACT NO.: CW 4598　　DATE: 12/20/85

GROUP NAME: HULL AND FRAME　　SUBGROUP NAME: INTEGRATION AND ASSEMBLY
GROUP NUMBER: 2　　SUBGROUP NUMBER: 8

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
PAINT	75.000	157.120	0.000	47.000

TOTAL WEIGHT OF SUBGROUP (LB)　　75.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	157.120	0.000	47.000

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS — CONTRACT NO.: CW 4598 — DATE: 12/20/85

GROUP NAME: HULL AND FRAME
GROUP NUMBER: 2

SUBGROUP NAME: CARGO HATCH
SUBGROUP NUMBER: 7

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN)		
		$\bar{X}$	$\bar{Y}$	$\bar{Z}$

NO ENTRIES FOR THIS SUBGROUP

TOTAL WEIGHT OF SUBGROUP (LB) 0.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	0.000	0.000	0.000

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: .2/20/85

GROUP NAME: HULL AND FRAME
GROUP NUMBER. 2

SUBGROUP NAME: RAMP
SUBGROUP NUMBER: 6

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN)		
		$\bar{X}$	$\bar{Y}$	$\bar{Z}$
REAR RAMP *	271.000	259.820	0.000	50.810
ESCAPE HATCH *	87.500	0.000	0.000	0.000
RAMP ACTUATOR MOUNTING BRACKET *	5.500	0.000	0.000	0.000
RAMP ACTUATOR COVER *	1.000	0.000	0.000	0.000
RAMP LATCH AND LINK PLATE *	2.000	0.000	0.000	0.000
ROTARY ACTUATOR AND SPROCKET *	36.500	0.000	0.000	0.000
SEE (GROUP7, SUBGROUP2) FOR HYDRAULICS	0.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 403.500

	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	174.501	0.000	34.125

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS | CONTRACT NO.: CW 4598 | DATE: 12/20/85

GROUP NAME: HULL AND FRAME
GROUP NUMBER: 2

SUBGROUP NAME: APPENDAGES
SUBGROUP NUMBER: 5

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
LIFTING EYE/MOORING BIT; AFT; STBD	8.000	260.000	22.000	80.000
LIFTING EYE/MOORING BIT; AFT; PORT	8.000	260.000	-22.000	80.000
LIFTING EYE/MOORING BIT; FORWARD; STBD	8.000	59.000	32.000	55.000
LIFTING EYE/MOORING BIT; FORWARD; PORT	8.000	59.000	-32.000	55.000
TOWING LUG; FORWARD; STBD *	10.500	83.300	32.000	22.000
TOWING LUG; FORWARD; PORT *	10.500	83.300	-32.000	22.000
TOWING LUG; AFT; STBD *	10.500	255.000	32.000	22.000
TOWING LUG; AFT; PORT *	10.500	255.000	-32.000	22.000
DRIVERS STATION	0.000	0.000	0.000	0.000
HATCH (M113 DRIVER) *	65.000	118.900	-36.500	79.000
PERISCOPES M17 (5 @ 7.0 LBS) *	35.000	118.900	-36.500	77.000
HATCH MECH AND SEAL	20.000	118.900	-36.500	77.000
COMMANDERS STATION	0.000	0.000	0.000	0.000
HATCH (M113 DRIVER) *	65.000	118.900	36.500	79.000
PERISCOPES M17 (5 @ 7.0 LBS) *	35.000	118.900	36.500	77.000
HATCH MECH AND SEAL	20.000	118.900	36.500	77.000
OBSERVERS STATION (SEE BALLAST GROUP 12, SUBGROUP 1)	0.000	0.000	0.000	0.000
BUMP STOP, RW#4; STBD *	14.500	0.000	0.000	0.000
BUMP STOP, RW#4; PORT *	14.500	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB): 343.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	118.788	0.000	43.627

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WEIGHT AND CENTER OF GRAVITY

TITLE ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: HULL AND FRAME
GROUP NUMBER: 2

SUBGROUP NAME: ACCOMODATIONS
SUBGROUP NUMBER: 4

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN)		
		$\bar{X}$	$\bar{Y}$	$\bar{Z}$
DRIVERS SEAT *	79.500	117.880	-36.500	60.000
COMMANDERS SEAT *	79.500	117.880	36.500	40.000
OBSERVERS SEAT STRUCTURE *	17.000	180.000	0.000	50.000
OBSERVERS SEAT	15.000	180.000	0.000	50.000

TOTAL WEIGHT OF SUBGROUP (LB) 191.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	122.288	0.000	58.325

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS | CONTRACT NO.: CW 4598 | DATE: 12/20/85

GROUP NAME: HULL AND FRAME
GROUP NUMBER. 2

SUBGROUP NAME: BULKHEAD AND INTERNAL COVERS
SUBGROUP NUMBER: 3

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$
ENGINE PUMP COVER *	32.000	0.000	0.000	0.000
FIREWALL	0.000	0.000	0.000	0.000
SIDE; STBD	30.000	0.000	0.000	0.000
SIDE; PORT	30.000	0.000	0.000	0.000
REAR	25.000	0.000	0.000	0.000
FLOOR PLATES	0.000	0.000	0.000	0.000
AFT FLOOR PLATE *	21.000	0.000	0.000	0.000
CENTER FLOOR PLATE *	22.000	0.000	0.000	0.000
FORWARD FLOOR PLATE *	29.000	0.000	0.000	0.000
FLOOR SUPPORTS	0.000	0.000	0.000	0.000
CHANNEL, ENGINE SUPPORT; STBD	38.900	171.400	13.500	20.120
CHANNEL, ENGINE SUPPORT; PORT	38.900	171.400	-13.500	20.120
MARINE COOLING SYSTEM COVERS	0.000	0.000	0.000	0.000
SEA CHEST COVER PLATE *	29.000	0.000	0.000	0.000
SMALL COVER, SEA CHEST (2) *	3.000	0.000	0.000	0.000
PUMP BOX COVER *	3.000	0.000	0.000	0.000
HOSE COVERS (2) *	1.000	0.000	0.000	0.000
INLET BOX COVER *	1.500	0.000	0.000	0.000
PROTECTIVE COVERS (SEE BALLAST GROUP 12, SUBGROUP 1)	0.000	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB) 304.300

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	43.822	0.000	5.144

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WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: HULL AND FRAME
GROUP NUMBER: 2

SUBGROUP NAME: HULL BOLTED AND MISCELLANEOUS
SUBGROUP NUMBER: 2

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN)		
		$\bar{X}$	$\bar{Y}$	$\bar{Z}$
MISCELLANEOUS HARDWARE	75.000	159.120	0.000	47.000
MISCELLANEOUS BOLTS	50.000	159.120	0.000	47.000
INTAKE/EXHAUST GRILLE STRUCTURE *	272.000	102.000	0.000	75.000
GRILLE COVER *	34.000	102.000	0.000	75.000

TOTAL WEIGHT OF SUBGROUP (LB): 431.000

CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
	118.566	0.000	66.879

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MOUNTI... PLATE, RW#4 BUMP STOP; PORT	1.600	195.030	-32.380	32.670
BUMP STOP, RW#5; STBD	17.000	226.730	38.870	44.380
BUMP STOP, RW#5; PORT	17.000	226.730	-38.870	44.380
BUMP STOP, RW#3; STBD	18.100	168.600	38.990	44.380
BUMP STOP, RW#3; PORT	18.100	168.600	-38.990	44.380
BUMP STOP, RW#2; STBD	23.800	139.870	41.560	44.380
BUMP STOP, RW#2; PORT	23.800	139.870	-41.560	44.380
BUMP STOP, RW#1; STBD	15.200	111.370	40.780	45.330
BUMP STOP, RW#1; PORT	15.200	111.370	-40.780	45.330
GRILLE SIDE PLATE; STBD	50.400	109.610	21.000	72.960
GRILLE SIDE PLATE; PORT	46.100	106.760	-21.000	72.960
GRILLE REAR PLATE	27.200	142.820	2.470	73.000
GRILLE REAR CORNER PLATE, PORT	5.200	140.500	-18.690	73.000
GRILLE REAR SEAL PLATE	4.800	141.440	1.770	66.380
GRILLE REAR CORNER SEAL; PORT	0.900	139.190	-17.500	66.380
GRILLE SIDE SEAL PLATE; PORT	6.900	105.780	-19.820	68.380
GRILLE SIDE SEAL PLATE; STBD	7.500	108.550	19.820	68.380
MOORING BIT STIFFENER; STBD, AFT	3.000	254.870	48.750	77.000
MOORING BIT STIFFENER; STBD; FORWARD	3.000	92.250	48.750	73.370
MOORING BIT STIFFENER; PORT; AFT	3.000	254.870	-48.750	77.000
MOORING BIT STIFFENER; PORT; FORWARD	3.000	92.250	-48.750	73.370
SPACER - DRIVERS HATCH	1.500	117.880	-21.810	77.630
SPACER - COMMANDERS HATCH	1.790	117.880	51.750	77.630
ENGINE ACCESS COVER *	47.500	76.540	0.000	36.000
ENGINE ACCESS COVER FRAME	18.800	77.200	0.000	36.380
SUPPORT, ENGINE FRONT *	6.400	0.000	0.000	0.000
SUPPORT, ENGINE REAR *	11.800	0.000	0.000	0.000

TOTAL WEIGHT OF SUBGROUP (LB)	6022.189		
CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	$\bar{X}$ 162.771	$\bar{Y}$ 0.034	$\bar{Z}$ 47.063

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS

CONTRACT NO.: CW 4598

DATE: 12/20/85

GROUP NAME: HULL AND FRAME
GROUP NUMBER: 2

SUBGROUP NAME: HULL WELDED AND MACHINED
SUBGROUP NUMBER: 1

COMPONENT	WEIGHT (LB)	CENTER OF GRAVITY (IN) X̄	Ȳ	Z̄
LOWER FRONT GLACIS PLATE, .75TK	256.200	69.440	0.000	45.690
UPPER FRONT GLACIS PLATE, .75TK	273.900	77.110	0.000	69.700
TOP PLATE, .75TK	890.700	201.600	0.070	77.630
SPONSON SIDE PLATE; STBD, .75TK	420.800	158.310	52.630	61.390
SPONSON SIDE PLATE; PORT, .75TK	420.800	158.310	-52.630	61.390
DRIVERS HATCH COVER MTG RING W/PLT	78.900	118.900	37.250	79.500
COMMANDERS HATCH COVER MTG RING W/PLT	78.900	118.900	-37.250	79.500
SPONSON BOTTOM PLATE; STBD, .50TK	146.900	147.290	43.070	43.500
SPONSON BOTTOM PLATE; PORT, .50TK	146.900	147.290	-43.070	43.500
HULL BOTTOM PLATE, 1.13TK	1086.800	172.490	0.000	16.540
REAR PLATE, .75TK	109.200	259.780	0.000	58.280
HULL SIDE PLATE; STBD, 1.25TK	513.500	167.000	31.500	31.050
HULL SIDE PLATE; PORT, 1.25TK	513.500	167.000	-31.500	31.050
MISCELLANEOUS MOUNTING BOSSES	25.000	162.900	0.000	47.320
FRONT SKID PLATE, 1.13TK	45.500	90.430	0.350	18.000
REAR SKID PLATE, 1.13TK	70.300	255.220	0.000	20.210
FINAL DRIVE MOUNTING RING; STBD	11.800	250.000	33.000	28.250
FINAL DRIVE MOUNTING RING; PORT	11.800	250.000	-33.000	28.250
HYDROPNEUMATIC MTG PLATES; STBD (5)	24.800	158.190	32.500	22.250
HYDROPNEUMATIC MTG PLATES; PORT (5)	24.800	158.190	-32.500	22.250
IDLER MOUNTING PLATE; STBD	15.500	83.370	33.120	33.120
IDLER MOUNTING PLATE; PORT	15.500	83.370	-33.120	33.120
WATERJET MTG PL. FORWARD; STBD, .75TK	26.100	243.990	41.940	53.560
WATERJET MTG PL. FORWARD; PORT, .75TK	26.100	243.990	-41.940	53.960
WATERJET MTG PL. UPPER; STBD, .75TK	23.900	252.480	41.940	63.370
WATERJET MTG PL. UPPER; PORT, .75TK	23.900	252.480	-41.940	63.370
WATERJET MTG PL. INNER; STBD, .75TK	24.500	252.150	31.250	53.380
WATERJET MTG PL. INNER; PORT, .75TK	24.500	252.150	-31.250	53.380
WELD	105.000	159.120	0.000	47.000
WATERJET INLET MTG PL; STBD, .75TK	10.700	207.580	45.410	47.700
WATERJET INLET MTG PL; PORT, .75TK	10.700	207.580	-45.410	47.700
WATERJET INLET REAR TRANSITION PL; STBD, .75TK	7.000	223.550	47.660	46.070
WATERJET INLET REAR TRANSITION PL; PORT, .75TK	7.000	223.550	-47.660	46.070
WATERJET INLET FORWARD TRANSITION PL; STBD, .75TK	23.400	180.750	46.920	46.800
WATERJET INLET FORWARD TRANSITION PL; PORT, .75TK	23.400	180.750	-46.920	46.800
TURRET SUPPORT RING	37.600	167.000	0.000	76.750
FORWARD TURRET SUPPORT COLUMN; STBD	18.300	154.500	19.400	46.690
FORWARD TURRET SUPPORT COLUMN; PORT	18.300	154.500	-19.400	46.690
REAR TURRET SUPPORT COLUMN	13.400	190.040	0.000	46.690
TOW LUG STIFFENER; STBD; AFT	0.800	256.500	22.000	21.000
TOW LUG STIFFENER; STBD; FORWARD	1.100	85.140	22.000	23.000
TOW LUG STIFFENER; PORT; AFT	0.800	256.500	-22.000	21.000
TOW LUG STIFFENER; PORT; FORWARD	1.100	85.140	-22.000	23.000
RETURN IDLER MT.; STBD (2)	6.200	182.930	32.500	34.880
RETURN IDLER MT.; PORT (2)	6.200	182.930	-32.500	34.880
MOUNTING PLATE, RW#4 BUMP STOP; STBD	1.600	195.030	32.380	32.670

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

WEIGHT AND CENTER OF GRAVITY

TITLE: ATR WEIGHT ANALYSIS | CONTRACT NO.: CW 4598 | DATE: 12/20/85

SUMMARY OF GROUP NO 1 - INTEGRATION AND ASSEMBLY

SUBGROUP	WEIGHT (LB)	CENTER OF GRAVITY (IN) $\bar{X}$	$\bar{Y}$	$\bar{Z}$

NO ENTRIES FOR THIS GROUP

TOTAL WEIGHT OF GROUP (LB) 0.000

	$\bar{X}$	$\bar{Y}$	$\bar{Z}$
CENTER OF GRAVITY WITH RESPECT TO REFERENCE AXIS (IN)	0.000	0.000	0.000

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Appendix B

Major Components Listing

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130301

Item Nomenclature: Hydropneumatic Suspension Unit (10)

Dimensions: 15.88 inches long x 11.62 inches high x 15.19 inches deep

Weight: 133.5 pounds without oil

Performance Characteristics: Provides variable spring and damping forces for a roadwheel travel of 16.5 inches in jounce and 4 inches in rebound with a total rotational travel of 94.35 degrees.

Manufacturer: Bird-Johnson Company, Walpole, Mass.

Model Number: RD-HPS-101 (Modified)

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130302

Item Nomenclature: Roadwheel, Rubber Tired (20)

Dimensions: 22 inch diameter x 3.55 inches wide

Weight: 43.81 pounds

Performance Characteristics: Provides support of ATR. Two roadwheels per suspension station. Contains foam to reduce water drag and provide buoyancy.

Manufacturer: Motor Wheel Corporation, Lansing, Mich.

Model Number: X40909

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130303

Item Nomenclature: Drive Sprocket Wheel (4)

Dimensions: 22.2 inch diameter x .75 inch thick

Weight: 28 pounds

Performance Characteristics: Provides power transmission to tracks. Pitch diameter = 19.36 inches, Number of teeth = 10.

Manufacturer: AAI Corporation, Baltimore, Md.

Model Number: 55763-40018

Use or disclosure of proposal data is subject to the restriction on the title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130304

Item Nomenclature: Compensating Front Idler Assembly Mount (2)

Dimensions: 17.25 inch diameter idler wheels

Weight: 216.5 pounds

Performance Characteristics: Provides mounting of idler wheels, track compensation during roadwheel movement and adjustment of track tension.

Manufacturer: AAI Corporation, Baltimore, Md.

Model Number: 60011-40308

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130305

Item Nomenclature: Wire Link Track (138 Pitches)

Dimensions: 17 inches wide x 2-3/8 inches thick x 5.81 inch pitch

Weight: 15.9 lbs/pitch, 32.84 lbs/foot

Performance Characteristics: Provides vehicle floatation and traction in land mode.

Manufacturer: AAI Corporation, Baltimore, Md.

Model Number: 55769-40024

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130306

Item Nomenclature: Support Roller Assembly (4)

Dimensions: 8.5 inch diameter x 3.4 inch wide dual rollers

Weight: 40 pounds

Performance Characteristics: Provides support and alignment of track return segment. Two assemblies per side.

Manufacturer: AAI Corporation, Baltimore, Md.

Model Number: 60011-40306

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130308

Item Nomenclature: Road Arm (10)

Dimensions: 14.5 inches long x 5.19 inches offset

Weight: 46.75 and 41.75 pounds

Performance Characteristics: Connects output shaft of hydropneumatic suspension unit to roadwheels to provide roadwheel travel.

Manufacturer: AAI Corporation, Baltimore, Md.

Model Number: 60011-40309

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130401

Item Nomenclature: Engine

Dimensions: 38"L x 36"W x 36"H

Weight: 1419 pounds

Performance Characteristics: 320 bhp @ 2800 rpm with 28% torque rise

Manufacturer: Caterpillar Tractor Company

Model Number: 3208T

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130401

Item Nomenclature: Cranking Motor

Dimensions:

Weight: 65 pounds

Performance Characteristics: 24 volt, clockwise

Manufacturer: Delco Remy

Model Number: 1114845

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130402

Item Nomenclature: Air Cleaner Assembly

Dimensions: 30.25"L x 9.0" dia.

Weight: 18.5 pounds

Performance Characteristics: Filters high air flow volume with minimum restriction

Manufacturer: Donaldson Company

Model Number: EBA09-2001

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130402

Item Nomenclature: Muffler

Dimensions: 18.1"L x 8.0" dia.

Weight: 9.4 pounds

Performance Characteristics: Silences engine exhaust noise

Manufacturer: Donaldson Co.

Model Number: MUM09-0071

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130403

Item Nomenclature: Radiator, Engine

Dimensions: 60"L x 38"W x 7" THK

Weight: 260

Performance Characteristics: Cools engine coolant via heat transfer to air

Manufacturer: Young Manufacturing Co.

Model Number: E306659

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130403

Item Nomenclature: Oil Cooler

Dimensions: 30"L x 38"W x 1.5" THK

Weight: 100 pounds

Performance Characteristics: Cools transmission oil via heat exchange to air

Manufacturer: Dunham Bush

Model Number: DB-T-32333-2C-2242

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130403

Item Nomenclature: Fan/Motor Assy

Dimensions: 10.82"L x 19.685" dia.

Weight: 53 pounds

Performance Characteristics: High volume and pressure rise mixed discharge air flow.

Manufacturer: NOA Airscrew Howden, Ltd.

Model Number: 41200-1

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130403

Item Nomenclature: Motor, Hydraulic Fan

Dimensions:

Weight:

Performance Characteristics:

Manufacturer: Volvo

Model Number: F11-10-MX-XE-X-104

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130403

Item Nomenclature: Heat Exchanger, Final Drive

Dimensions:

Weight: 21.45 pounds

Performance Characteristics:

Manufacturer: Young Radiator

Model Number: F-302-HY-2P

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130404

Item Nomenclature: Heat Exchanger, Hydraulic Oil Marine

Dimensions:

Weight: 24 pounds

Performance Characteristics:

Manufacturer: Young Radiator Co.

Model Number: HF-302-EY-1P-CNTB

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130404

Item Nomenclature: Heat Exchanger, Engine Marine

Dimensions:

Weight: 55 pounds

Performance Characteristics:

Manufacturer: Young Radiator Co.

Model Number: F602-ER-1P-CNTB

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130404

Item Nomenclature: Pump, Sea Water

Dimensions:

Weight: 13.1 pounds

Performance Characteristics: 46 GPM

Manufacturer: ENPO Pump Co.

Model Number: 3-107 Modified

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130406

Item Nomenclature: Cap, Fuel Tank

Dimensions:

Weight: 0.92 pounds

Performance Characteristics: Non-vented MS53075-1

Manufacturer:

Model Number: FSN 2910-00-753-9118

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130407

Item Nomenclature: Alternator

Dimensions:

Weight: 74.6 pounds

Performance Characteristics: 300 amp 24 volt

Manufacturer: TRW-Niehoff

Model Number: N1326-1

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B S. Number: 130410

Item Nomenclature: Battery, Vehicle (4), Computer (2)

Dimensions: 12.0"L x 6.9"W x 9.0"H

Weight: 36.5 pounds

Performance Characteristics: 150 amp-hrs. rating each battery, 300 amp-hrs. for set of 4 @ 24 VDC

Manufacturer: Delco

Model Number: Group 27F

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130501

Item Nomenclature: Hydrostatic Pump (2)

Dimensions: 18.6"L x 12.1"W x 11.8"H

Weight: 150 pounds

Performance Characteristics: 6.12 CIPR, 74 GPM, 6000 PSI

Manufacturer: Linde Hydraulics

Model Number: BPV-100S

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130502

Item Nomenclature: Hydrostatic Motor (2)

Dimensions: 20.3"L x 8.3"W x 17.3"H

Weight: 165 pounds

Performance Characteristics: 11.35 CIPR to 3.36 CIPR, 6000 PSI, 812 LB-FT

Manufacturer: Linde Hydraulics

Model Number: BMV-186

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130503

Item Nomenclature: Pump Drive Gearbox

Dimensions: 13.04"L x 26.74"W x 22.135H

Weight: 300 pounds

Performance Characteristics: 650 HP, 2 pad, 14.25" between pads

Manufacturer: Twin Disc

Model Number: F11PMD2

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130507

Item Nomenclature: Final Drive (2)

Dimensions: 25"L x 17" dia

Weight: 260 pounds

Performance Characteristics: 2 speed epicyclic gear arrangement

Supplier: G.F.E.

Model Number: Custom

Use or disclosure of proposal data is subject to the restriction on the title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130601

Item Nomenclature: Waterjet (2)

Dimensions: 62.8 inches long x 14.8 inches high x 14 inches width

Weight: 198 pounds (dry)

Performance Characteristics: Two stage axial flow with 300mm (12") diameter impeller and steering bucket.

Manufacturer: Dowty Hydraulics Units Ltd., Cheltenham, England

Model Number: 8377-000-OL9 (PORT), 8377-000-ORU (STBD)

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130602

Item Nomenclature: Propulsor Motor (2)

Dimensions: 14.8"L x 7.2"W x 9.3"H

Weight: 71 pounds

Performance Characteristics: 6.4 CIPR 6000 PSI 510 FT-LB

Manufacturer: Linde Hydraulics

Model Number: BMF-105 TFC

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130602

Item Nomenclature: Flexible Coupling

Dimensions:

Weight: 10 pounds

Performance Characteristics:

Manufacturer: Lovejoy

Model Number: Model 1 Size 30 Custom Spline

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130702

Item Nomenclature: Auxiliary Hydraulic Pump (2)

Dimensions: 7.68"L x 6.3" dia.

Weight: 24.2 lbs.

Performance Characteristics: Variable displacement pump, pressure compensated to 3000 psi max., 1.52 cubic inch/revolution max. displacement.

Manufacturer: Rexroth

Model Number: A10V25

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 13703

Item Nomenclature: FSSS Control Panel

Dimensions: 6.5"L x 3.8"W x 2.6"H

Weight: 5 pounds approx.

Performance Characteristics: Provide complete monitoring and control of crew and engine compartment FSSS.

Manufacturer: Santa Barbara Research Center

Model Number: To be assigned

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 13703

Item Nomenclature: Control Electronics Amplifier

Dimensions: 9.4"L x 4.5"W x 3.5"H

Weight: Less than 5.0 lbs.

Performance Characteristics: Provides logic and control functions for fire sensing and suppression system

Manufacturer: Santa Barbara Research Center

Model Number: 52575 (or similar)

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 13703

Item Nomenclature: Fire Sensor (7)

Dimensions: 3.5"L x 2.7"W x 1.7"H

Weight: Less than 0.6 pounds

Performance Characteristics: Discriminating optical sensor with internal BITE.

Manufacturer: Santa Barbara Research Center

Model Number: PM34-CB (Crew Compartment)

PM34-CBEH (Engine Compartment)

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130703

Item Nomenclature: Bottle and Valve Assembly (4)

Dimensions: 5.42" Dia. x 16.5"H

Weight: 20.0 pounds (charged)

Performance Characteristics: Charged with 5 pounds Halon 1301 fire suppressant, Marrotta Solenoid Valve Model No. MV121KJ-1

Manufacturer: Marrotta Scientific Controls, Inc.

Model Number: 283263-0005

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.

Automotive Test Rig

Major Components Listing

W.B.S. Number: 130705

Item Nomenclature: Bilge Pump (2)

Dimensions: 8.3"W x 8.0"L x 5.4"H

Weight: 10 pounds

Performance Characteristics: Hydraulically driven submerged and dry run capability 115 GPM against 15 ft. head

Manufacturer: MP Pumps, Div. of Tecumseh

Model Number: 26474

Use or disclosure of proposal data is subject to the restriction on the Title page of this Proposal.