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FOR

SCIENTIFIC AND TECHNICAL INFORMATION

CAMERON STATION, ALEXANDRIA, VIRGINIA

CONCEPT STUDY REPORT

TRACKED AMPHIBIAN PERSONNEL AND CARGO CARRIER

Prepared For

CHIEF, BUREAU OF SHIPS
DEPARTMENT OF THE NAVY
WASHINGTON 25, D.C.

ATTENTION: CODE 529V

Contract: NObs-4466

Senior Engineer

Chief Engineer

Project Engineer

Executive Engineer

2 NOVEMBER 1961

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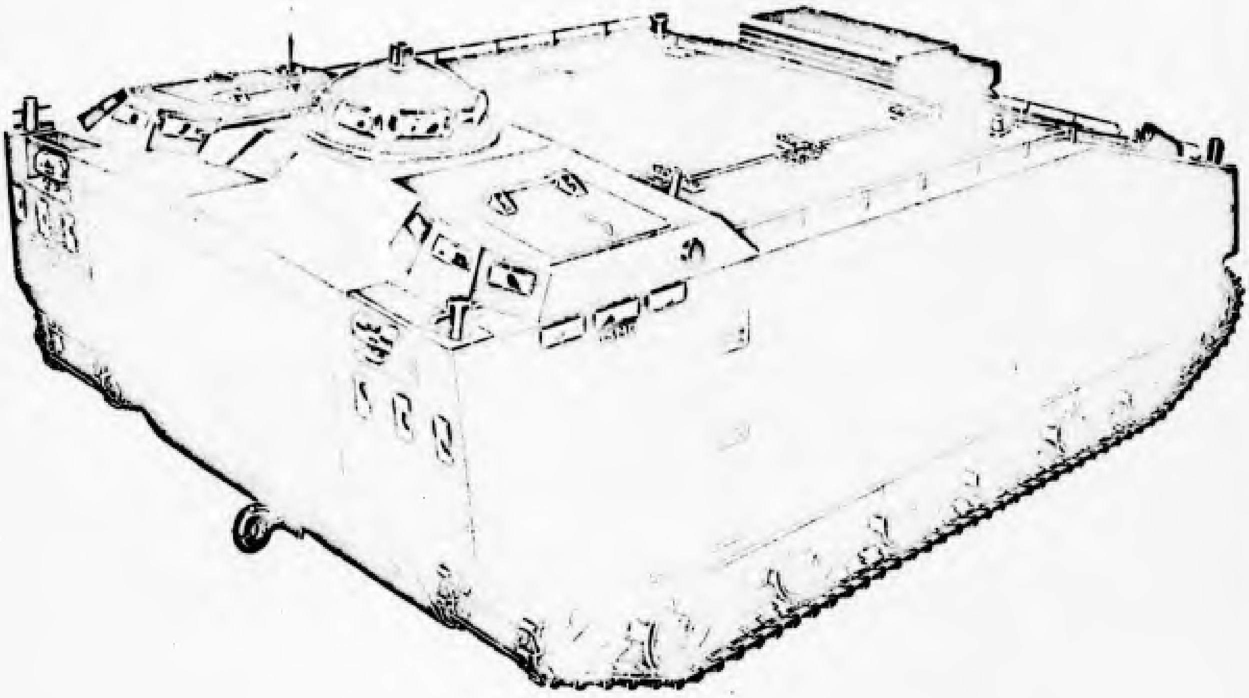

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1. INTRODUCTION

This report has been prepared under the authority of Paragraph 1.g. of the referenced contract, which states that the Contractor shall prepare and submit ten (10) copies of a Concept Study Report describing all of the work performed under the contract.

Under this program, Chrysler Corporation agreed to conduct an intensive design study to be completed within a period of four months. The purpose of the program was to establish the concept design and over-all configuration of a Tracked Amphibian Personnel and Cargo Carrier, designated as the LVTPX-11. The Bureau of Ships stated that the vehicle will be used to transport infantry, infantry weapons and cargo from ship to shore and to inland objectives, and to provide fire support in amphibious operations. The basic vehicle hull, power train and suspension system were required to be adaptable to the following special purpose vehicles:

(1) Command Vehicle
(2) Recovery Vehicle
(3) Amphibian Assault/Anti-Mechanized Weapon Vehicle
(4) Amphibian Field Artillery Weapon Vehicle
(5) Amphibian Light Air Defense Weapon Vehicle
(6) Engineer Mine Clearance Vehicle

The Bureau of Ships specifically required that the Contractor perform the following items of work:

(1) The selection of components.
(2) The preliminary design of new components.
(3) The preparation of preliminary arrangement drawings.
(4) The design of the structure and form of the hull in sufficient detail to support the over-all study.
(5) The preparation of a lines drawing of the hull.
(6) Determine the land and water performance of the vehicle in sufficient detail to show compliance with the stated Development Characteristics, (see Appendix A).

The study program was carried out by the Contractor's Defense Engineering staff. Scale model tow tank testing was performed at the University of Michigan under the direction of Professor R. B. Couch.

2. DESIGN APPROACH

2.1 BACKGROUND

The concept design described in this report has been developed in an entirely practical manner so as to make possible the fabrication of a prototype vehicle within the next 12 to 18 months.

While the requirement to produce a vehicle concept of not more than 35,000 pounds gross weight has been given maximum attention, the mission requirements of the vehicle have been kept in mind, particularly with regard to armor protection. It is our opinion that in order to provide the Bureau of Ships and the Marine Corps a vehicle having an acceptable degree of armor protection, a 40,000 pound configuration might offer a far more useful vehicle in terms of combat effectiveness. Therefore, we have taken a dual approach to the design problem and have studied both the 35,000 and the 40,000 pound vehicle configurations.

It ~~is our purpose, in the body of~~ this report, ~~to~~ outline the design of a vehicle which provides the required payload capacity and which meets the land and water performance specifications in a 35,000 pound package. Having done this, we will then discuss some trade-offs which have occurred to us during the Concept Design Program in Section 8 titled "Design Alternatives".

Chrysler has taken the approach that only those components which are well advanced in their development should be considered for prototype installation. While pursuing this design policy, care has been taken to provide a configuration which could take advantage of newly developed components when they become available.

The vehicle weight limitation of 35,000 pounds has been used as the yardstick by means of which a fairly large number of engineering decisions have been judged. In each case, where a design or performance feature could be optimized if an increase in weight could be accommodated, the requirement for meeting the weight target has been the overriding consideration.

After satisfying the weight requirement, each component and design feature has been selected on the basis of its ability to meet the following general design considerations:

(1) Simplicity
(2) Long trouble-free life
(3) Ease of maintenance
(4) Economy of operation

Chrysler's proposed design stresses simplicity from the standpoint of manufacture, operation, and maintenance.

Power train components, in particular, have been selected so as to have reserve capacity so that the machinery will not be called upon to operate for long periods of their maximum output. High engine loadings have been avoided so that a long power plant life can be assured. A review of current transmissions indicated that two units were of appropriate capacity for this vehicle; one was approved by the manufacturer for vehicles up to but not exceeding 35,000 pounds, the other was approved for vehicles of 40-45,000 pounds. The larger capacity unit is recommended by Chrysler for this vehicle in order to insure a long service life and also to provide some margin for growth. It is improper design practice, in Chrysler's opinion, to install major components at the limit of their capacity and thus invite a major modification program early in the service life of the vehicle.

Ease of maintenance has been stressed with two thoughts in mind; the ease with which components can be serviced, and the total number of points and the frequency with which they must be serviced.

Economy of operation has been provided for primarily from the standpoint of fuel consumption. However, simplicity of design can also contribute to economy of training time and economies in the cost of spare parts supply. Rigid adherence to the weight requirement can contribute to economies of transportation with respect to both space and weight. All of these factors have been borne in mind throughout this concept study phase.

2.2 HULL CONFIGURATION

The size and shape of the hull influence the gross weight of the vehicle, the speed of the vehicle in the water, the cargo-carrying capacity of the vehicle, and the ease with which the vehicle can be manufactured and repaired. Chrysler set about to design a hull of the smallest possible dimensions in order to limit the weight of the hull plating and structure. The width of the vehicle was of particular importance in that, for a given vehicle length, width has a considerable influence on effective horsepower. After making several trial layouts of the hull with a beam of about 9-1/2 feet, it was decided to increase the beam to 10-1/2 feet in order to accommodate the truck, utility 1/4 ton 4 x 4, lightweight, M422. While the development characteristics for the Tracked Amphibian do not require that the M422 be carried as cargo, it was Chrysler's opinion, after discussing the matter with knowledgeable Marine Corps Officers, that it would be a mistake not to provide space for that vehicle. The additional vehicle width has, no doubt, increased the effective horsepower, and in turn decreased water speed. The actual decrement of the water speed is not known as only one width (the larger) was tested in the towing tank program.

2.3 POWER PLANT

The power plant recommended for installation in the prototype Tracked Amphibian was chosen from a number of engines which are now in existence. During the time in which the power plant decision was being formulated, consideration was given to a gas turbine. While a suitable regenerative gas turbine is not yet available, several recent Government sponsored programs have been initiated which may produce an attractive power plant for this vehicle. With this thought in mind, the engine compartment and the air induction system were worked out so as to be compatible with gas turbine requirements. This process of reasoning argued against the employment in the prototype of either a liquid cooled power plant with sea water heat exchanger, or air to sea water cooling system.

It was Chrysler's opinion that an air cooled compression ignition engine and an engine compartment air induction system which would provide air for both combustion and cooling without the use of a keel cooler or heat exchanger would offer maximum flexibility regarding a future gas turbine installation.

This decision carried with it the need for developing an engine compartment air induction system which would admit a large volume of combustion and cooling air and at the same time prohibit the entry of water when the vehicle is immersed in surf. While such an air induction system represents a design problem of some difficulty, the advantages to be gained were sufficiently attractive to justify the effort, and the design was undertaken accordingly.

2.4 TRACK

The matter of track block design was analyzed initially from the standpoint of weight. The track blocks of the LVTP-5, for example, represent 7.6% of that vehicle's gross weight. If the track block which is used on the LVTP-5 were to be used on the LVTPX-11, the track blocks would represent 14.2% of the latter's gross weight (calculated on the basis of 134 track blocks per side for the LVTP-5 and 100 track blocks per side for the LVTPX-11).

Looking at the problem from another point of view, if the total weight which could be budgeted for track blocks were to be computed on the basis of the relative gross vehicle weights of the LVTP-5 and the LVTPX-11, we found that the total weight of the track blocks should not exceed 2,660 pounds on the LVTPX-11. Assuming a total of 200 track blocks per vehicle, the weight per pitch which we could allow was 13.3 pounds. When compared with a weight per pitch of 24.9 and 38 pounds for the LVTP-5 and 6 respectively, the need for a new approach to the design of the track was clearly indicated.

In order to achieve the lighter track block weight, the designer might remove the water propulsion grouser, or make the track narrower than that used on the LVTP-5 and 6, or make the track from a lighter weight material.

If the grouser were to be removed from the track block, some other means would have to be provided for center guiding the track. Assuming that this could be done satisfactorily and at very little expense in terms of weight, some other means of water propulsion would have to be provided. Water jets and screw propeller-kort nozzle combinations suggest themselves, but such a solution would only relocate the weight rather than remove it.

The total projected area of the track in contact with ground has an important effect on vehicle performance in that it determines the ground pressure of the vehicle. Once the length of track in contact with the ground has been selected to satisfy approach and departure angles, the designer must calculate the track width needed to satisfy mobility requirements. In order to negotiate the various terrains described by the Bureau of Ships in the Development Characteristics for this vehicle, and in particular, muskeg, a ground pressure of approximately 5 psi was established as a goal by Chrysler. Thus the possibility of achieving a reduction in track block weight by making the blocks narrow in width was eliminated.

We were left there with the apparent need for making the track block from a light weight material in order to satisfy the requirements of the weight budget. A one piece aluminum forging with hard metal overlays at points of wear was decided upon after detailed discussion with representatives of the aluminum industry.

2.5 ARMOR PROTECTION

Note was taken of the statement in the Development Characteristics that sufficient armor is required to protect the crew and personnel from small arms fire and shell fragments, "Commensurate with maximum vehicle weight specified". Chrysler has interpreted this statement to mean that whatever weight allowance is available after all other components of the vehicle have been provided for shall be invested in thickness of armor. We are not entirely satisfied with the ballistic protection which has resulted from this design approach, and we will return to the subject in a later section of this report.

2.6 SURFABILITY

Particular emphasis has been placed upon those elements of the design which contribute to the negotiation of plunging surf; i.e., a high degree of static stability, a rugged, water tight hull and hatch structure, a power plant which can operate in a completely flooded engine compartment, a configuration which will remain afloat with the engine compartment completely flooded, and a large capacity bilge system.

Chrysler adopted the philosophy that there must be no communication of air or water between the engine and personnel compartments and that the bilge systems for the two compartments must be entirely divorced from one another. It was also our goal to provide a power plant having fan clutches which would automatically disengage should the engine compartment flood. If the engine can thus continue to operate, the bilge system will clear the water from the engine compartment and the driver continue to maneuver the vehicle and make headway through the surf zone.

However, should the power plant fail, we wanted to be certain that the vehicle could survive with a completely flooded engine compartment. Consequently, a water tight bulkhead and two separate bilge systems were decided upon. In addition to the hydraulically driven bilge pumps, each compartment is provided with an electrically powered bilge pump which can be switched on or off independently of one another and operated by battery power. Since flooding of the engine compartment will result in a down by the stern attitude, the electric emergency bilge pumps have their intakes situated in the rear of each compartment.

2.7 TRANSITION FROM LAND TO WATER OPERATION

At the outset of the program it appeared essential to Chrysler that any special "Land" or "Water" control actuation be completely eliminated. We have been particularly careful to insure that there is absolutely no difference in control function whether the vehicle is on land or in the water. The only control actuation which the operator will have to make when going into the water will be the shifting of the transmission into Second Gear to obtain clutch-brake steering and proper track speed for water travel.

2.8 WATER PROPULSION SYSTEM

The statement in the Development Characteristic to the effect that the water speed be the highest "Consistent with the power required for the specified land performance" indicates clearly that the power plant capacity is to be determined by the land speed and grade climbing requirements rather than speed in the water.

The selection of water propulsion machinery has been left entirely up to the designer. One way to approach the propulsion machinery problem is on the basis of weight. If the weight of the water propulsion grouser of the track as it is used on the LVTP-5 were to be calculated for the aluminum track proposed for the LVTPX-11 we would find that 438 pounds has been invested in water propulsion "machinery". The entire weight of the grousers cannot be attributed to water propulsion as the grouser is also the track center guide and the cross beam which ties together the two ends of the track block. So actually, considerably less than the 433 pounds cited has been invested in a water propulsion device. Therefore, for sake of discussion, we will assume that half of that figure, or approximately 220 pounds, is available in the weight budget for water propulsion without incurring a weight penalty.

The most efficient device which could be employed would be a screw propeller and Kort nozzle. We would then have to include a set of reduction gears, a reversing gear, a clutch, a rudder, and a set of operator controls. It seems unlikely that this much mechanism could be obtained in a 220 pound package, and if a retraction mechanism is included to raise the propeller above the ground clearance line, the situation becomes still more unlikely.

A hydro jet is often suggested for amphibious vehicle applications and the use of such a device has been considered by Chrysler for the LVTPX-11. A review of the fluid mechanics of the water jet reveals that the thrust produced is equal to the product of the mass flow through the pump times the change of velocity (vectorially) which occurs between the intake and exit of the jet. Jet thrust can obviously be increased by either of two means; increasing the flow rate of the pump (in gpm), or by increasing the acceleration of the water through the pump.

Before making a decision as to whether to increase the flow rate of the pump (which means using a larger pump) or increasing the fluid acceleration through the pump (which means increasing the speed of the impeller within cavitation limits) we should investigate the factors which determine the propulsive efficiency of the water jet. The propulsive efficiency of any fluid jet is given by the expression:

$$E_p = \frac{2V}{W + V}$$

where: E_p = propulsive Efficiency

V = craft speed of advance in fps

W = jet wake velocity in fps

In order to operate at 100 per cent efficiency, the craft speed and the jet exit velocity would have to be equal. It is apparent, therefore, that a fluid jet is efficient when used in high speed craft. It is also apparent that the jet becomes less and less efficient as the speed of the craft becomes slow (with respect to the jet exit velocity). Numerical examples and actual experience with hydro jets support this conclusion.

Returning to the matter of the jet pump characteristics, we can provide a hydro jet of reasonable propulsive efficiency (15 to 20%) only if we are able to install a pump of very large capacity, with a relatively low exit velocity. For a craft moving in the water at 6 to 7 mph, a very large (and very heavy) pump would be required in order to equal the efficiency of track grouser propulsion.

This rationale, which is supported by experience which Chrysler has had with a U.S. Army (Corps of Engineer) amphibian propelled by hydro jets, led us to the decision to use track grouser water propulsion on the LVTPX-11.

3. DESCRIPTION AND DATA

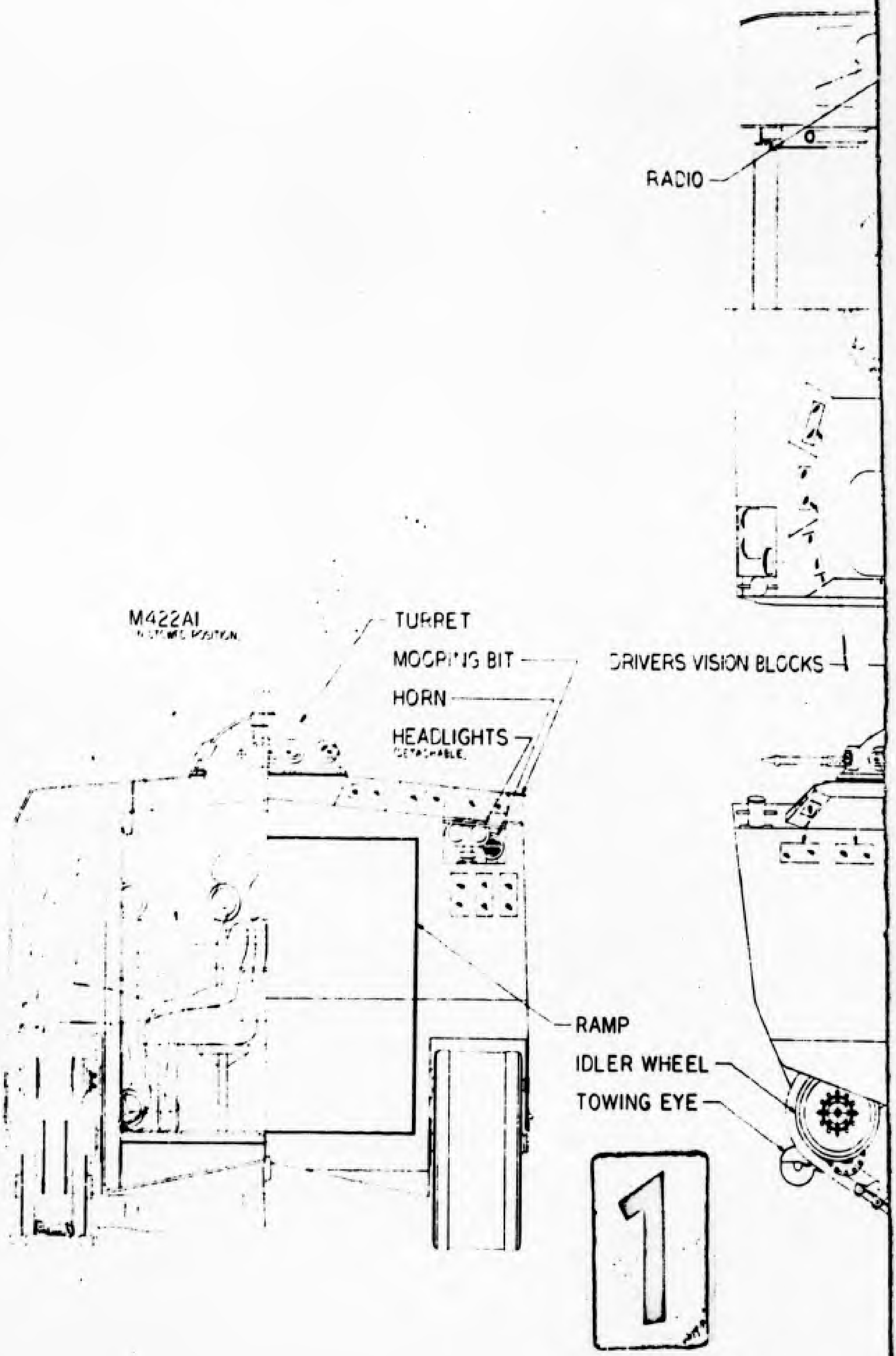
The LVTPX-11 is a full tracklaying amphibious vehicle, designed to land and transport personnel and/or cargo to beachhead or inland objectives and to support logistically troops in a landing operation.

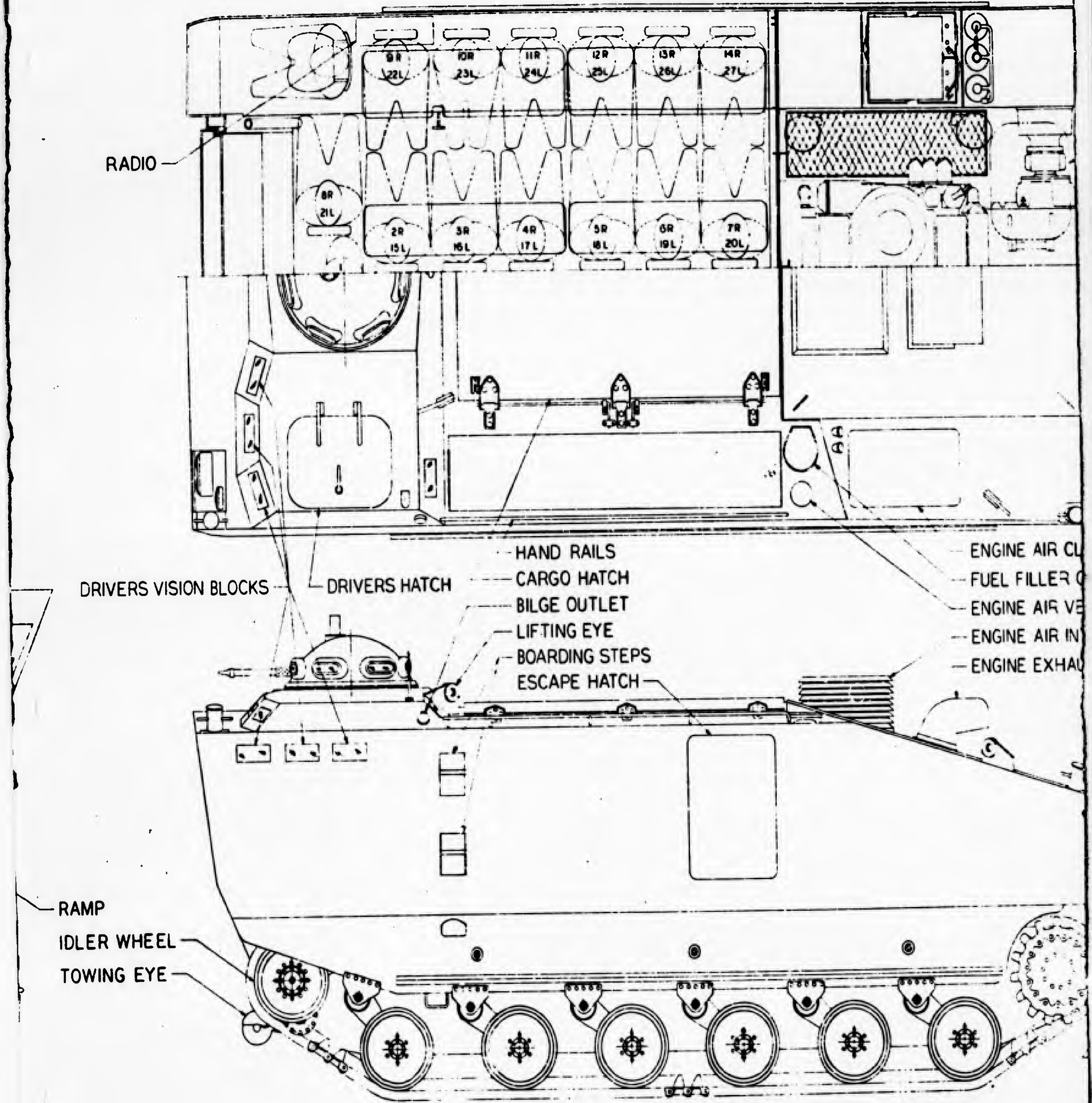
The vehicle is intended to operate ashore 80 per cent of its life and afloat 20 per cent of its life. The design has been optimized for land performance accordingly.

The configuration of the basic personnel and cargo-carrying vehicle is depicted on the Figures and Drawings which appear on the following pages.

(1) Vehicle General Arrangement Drawing
(2) Vehicle Characteristic Sheet
(3) 3/4 Starboard Stern View
(4) 3/4 Starboard Bow View

Illustrations and a brief technical description of each of the six special vehicle adaptations may be found in Section 9 of this report.

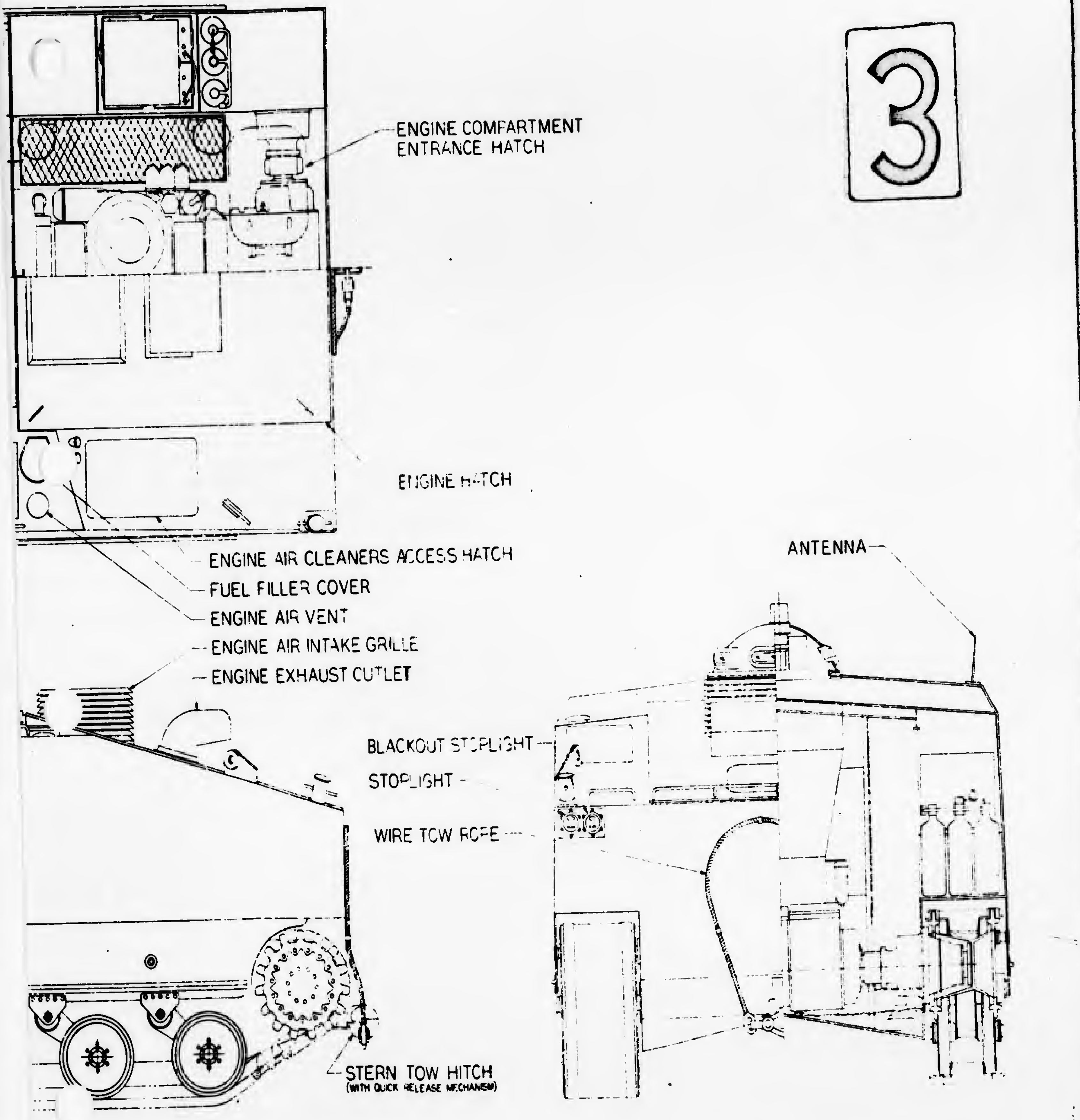

RADIO
M422A1
TURRET
MOORING BIT
HORN
HEADLIGHTS
DRIVERS VISION BLOCKS
RAMP
IDLER WHEEL
TOWING EYE
1



RADIO
9R 22L
10R 23L
11R 24L
12R 25L
13R 26L
14R 27L
8R 21L
2R 15L
3R 16L
4R 17L
5R 18L
6R 19L
7R 20L
HAND RAILS
CARGO HATCH
BILGE OUTLET
LIFTING EYE
BOARDING STEPS
ESCAPE HATCH
DRIVERS VISION BLOCKS
DRIVERS HATCH
ENGINE AIR CL
FUEL FILLER
ENGINE AIR VE
ENGINE AIR IN
ENGINE EXHAU
RAMP
IDLER WHEEL
TOWING EYE

2

3

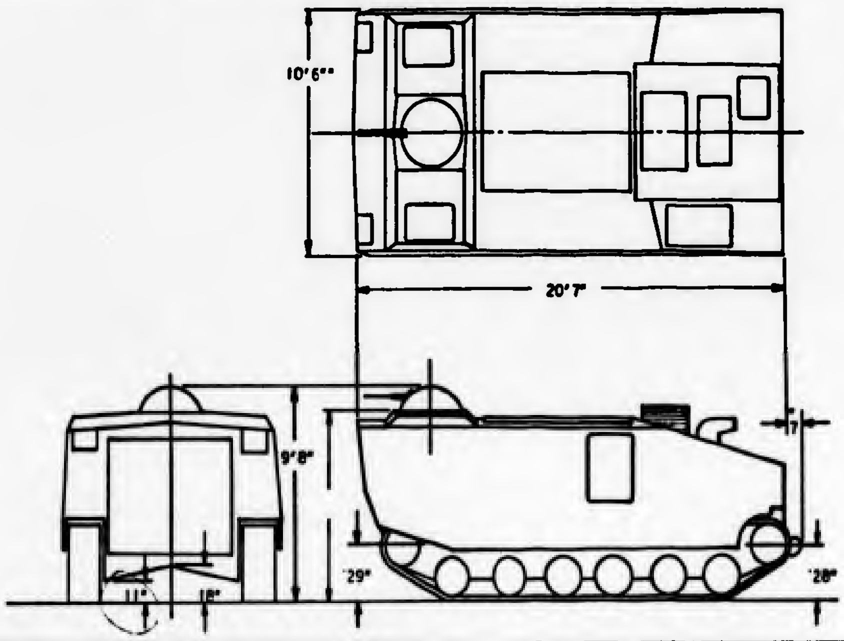


VEHICLE GENERAL
ARRANGEMENT TD 106521

CHARACTERISTICS SHEET **BUREAU OF SHIPS** **CODE 518**

VEHICLE LVTPX-11

TYPE LANDING VEHICLE, TRACKED; PERSONNEL, MODEL XII (LVTPXII) (TRACKED AMPHIBIAN PERSONNEL AND CARGO CARRIER)



21.167 x 10.5 x 9.66 =
2145 x 2000 / 35000 = 172

1. GENERAL

Weight (Combat loaded) 35000 lbs. No. Crew (Including Gunners) 2
Weight (Less crew, stowage and fuel) 23000 lbs. Max. Cargo Cap. 10000 lbs.
C. of G. 38.88 inches above ground, 137.39 inches from 0
Unit Ground Pressure 5.4 psi at 0 inches penetration and 35000 lbs. weight
TRACKS CONSIDERED SOLID BAND.

2. DIMENSIONS

Length Overall 21 ft. 2 inches Width Overall 10 ft. 6 inches
Height Overall 9 ft. 8 inches Cubic Volume 2089 cubic feet
Tread Width (Center to Center) 100 in. Tread Length 12 ft. 10 inches
Angle of Approach 31 Angle of Departure 30°
Ground Clearance C_L 18 inches Draft, Bow 42 3/4 inches. Stern 46 1/4 inches
SIDE 11 INCHES ABOVE BASE LINE

3. PERFORMANCE

Gross HP to Weight Ratio 24.3 HP/Ton Net HP to Weight Ratio 20.5 HP/Ton
Max. Drawbar Pull 28000 lbs. Cruising Range, Land 16.7 Water 11.7 HRS.
Max. Land Speed, Forward 40 mph. Reverse 8.4 mph.
Max. Water Speed, Forward 6.5 mph. Reverse 2.2 mph.
Max. Grade 70+ % Max. Side Slope 60 %
Max. Trench 8 feet 0 inches Max. Wall 3 feet 0 inches
Turning Diameter, Land 22 feet Turning Diameter, Water -- feet

4. ENGINE

Make CONTINENTAL Model AVDS-750-1 Type 90° UPRIGHT VEE
No. of Cylinders 8 Bore 4.875 inches. Stroke 5.000 inches. Speed 2800 rpm
Total Displacement 746 cubic inches. Comp. Ratio 17.5:1 No. of Engines 1
Fuel MULT. FUEL Rating - Fuel Capacity 190 US Gallons
Max. Gross HP 425 at 2800 rpm. Max. Net HP 366 at 2800 rpm
Max. Net Torque 870 ft. lbs. at 2000 rpm. Cooling System AIR
Oil Capacity 9 gals. - quarts. Carburetor - NONE
Weight, Dry 2503 lbs. Wet 2573 lbs. Wet Weight/Net BHP 7 lbs./bhp.

5. POWER TRAIN

Transmission ALLISON XTG411-7
Transfer Case NONE
Differential INTEGRAL WITH TRANSMISSION
Torque Converter ALLISON 2.8:1 STALL RATIO
Other Units
Overall Ratio (Engine to Sprocket) 21.05:1
Steering Control HYDRAULIC Brakes MECHANICAL
Oil Cooling System OIL-AIR HEAT EXCHANGER Transmission Oil Capacity 46 quarts

6. FINAL REDUCTION

Type CONCENTRIC PLANETARY Ratio 4.5:1
Sprocket Pitch Diameter 23.85 inches. No. of Teeth 15

7. RUNNING GEAR

Type of Suspension System RUBBER TORSION
No. of Bogies 6 per side. Size 18 INCH DIA.
No. of Return Idlers 3 per side. Size 9 INCH DIA.
Type of Track PERMANENTLY LUBRICATED-FLEX. HINGE PIN Mfg.
Pitch 5 inches. Pin Diameter .875 inches. Width 20.75 inches
Weight per pitch 10.3 lbs.; per foot 24.7 lbs.; per vehicle 2054 lbs.
Grouser Type CENTER BUCKET Materials FORGED ALUMINUM 2014T6

8. FIRE EXTINGUISHERS

Fixed 3 Location STBD SIDE - ENGINE COMPARTMENT Weight 10 lbs. EA
Portable 1 Location CREW CHIEF'S STATION Weight 5 lbs.

9. ELECTRICAL

Nominal Voltage 24 D. C. Amperage 100
Generator, Main 100 AMP ALTERNATOR
Generator, Auxiliary NONE
Batteries, Make MS35000-3 No. 4 No. of Plates 23 100 Amp. hrs.
Magnetos -- Distributor -- Coil --
Spark Plugs -- Ignition Harness --
Starting Motor DELCO - REMY

10. COMMUNICATIONS

Radio Set AN/PRC-47 Location CREW CHIEF'S STATION
Radio Set - Location -
Intercomm. Set AN/VIA-1 No. of Outlets 3
Tank-Infantry Telephone AT VEHICLE STERN

11. ARMAMENT

Primary (7.62MM MG (M73) in CUPOLA Mount. Recoil Mech. --
Traverse, Right 360° Left 360° Rate -- Power MANUAL
Elevation 60° Depression 15° Stabilizer System
Gun Control MANUAL Mounting Ring Diameter 29 3/4 inches
Secondary --
Other --

12. AMMUNITION

Type 7.62MM Rounds 1000 Weight 59 lbs.
Type - Rounds - Weight - lbs.
Type - Rounds - Weight - lbs.

13. ARMOR

Permanent Hull INTEGRAL ALUMINUM ARMOR: RAMP, OUTER 1/2", INNER 3/4", SIDES 5/8" TOP 3/8", BOTTOM 1/2", STERN 3/8"
Pin-on Hull NONE
Turret 3/8 IN. STEEL

14. OPTICAL DEVICES

Driver's Vision -BLOCK, DIRECT VISION Quantity 6
Gunner's Vision -CUPOLA, VISION BLOCKS Quantity 5
Other Vision SIDE AND REAR VISION Quantity 10
Telescope NONE
Periscope M28C
Sights DRIVER'S LINE SIGHT QUANTITY 1
INCLINOMETER QUANTITY 2

15. CARGO COMPARTMENT

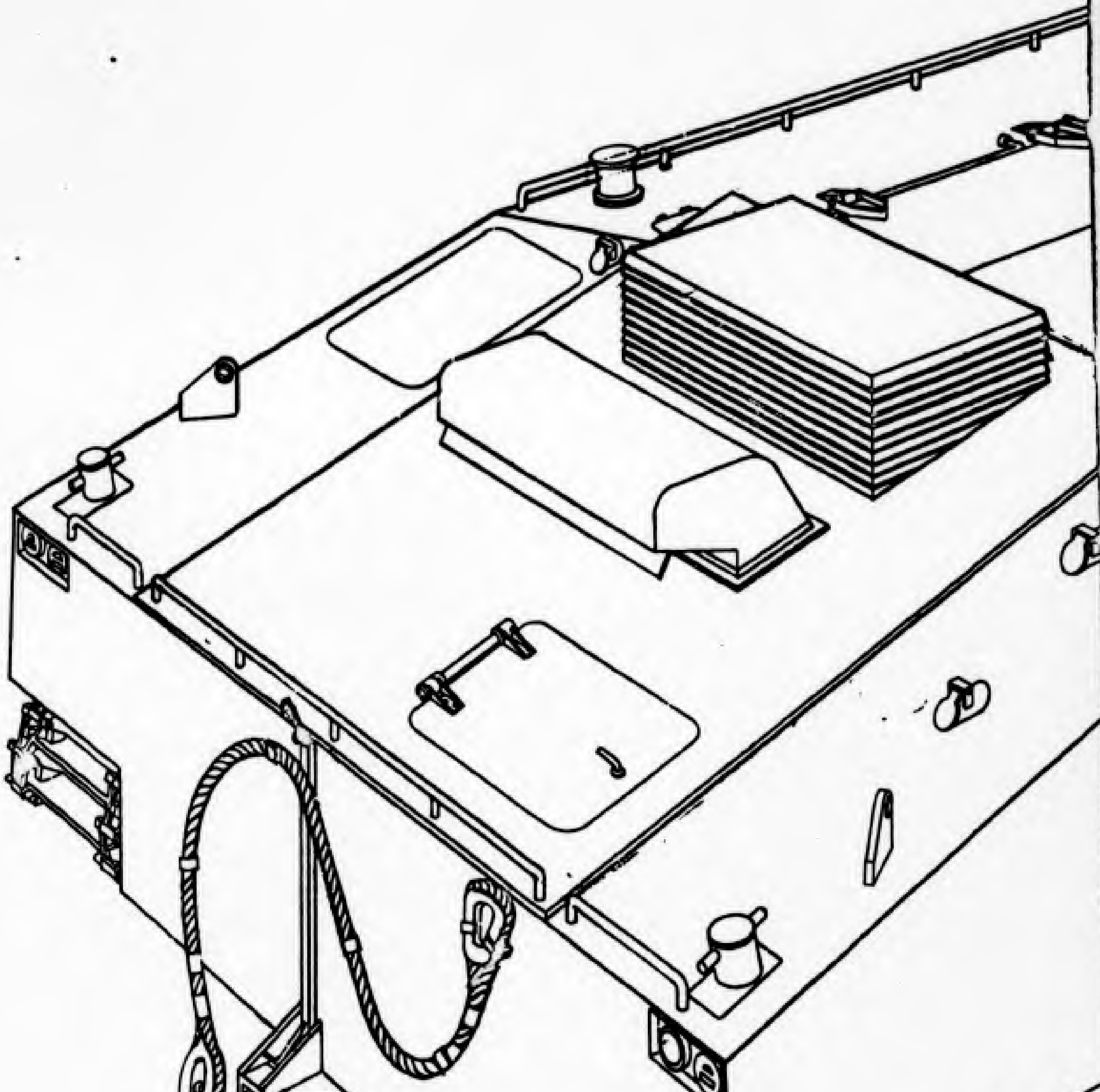
Length 11 feet 8 inches Width 5 feet 8 inches
Height 5 feet 6 inches Area 64 square feet
Volume 352 cubic feet Ramp Location BOW
Capacity, Equipment 27 Personnel 27 COMBAT EQUIP. MARINES
Cover Opening, Length 7 feet 0 inches. Width 5 feet 0 inches
Height to covers, Max. feet inches. Min. feet inches

16. OTHER

Contractors CHRYSLER DEFENSE ENGINEERING, DETROIT, MICH.
Contracts NOBS. 4466
Publications FINAL REPORT
Date Model Built NONE Date First Production Vehicle
Date Prototype NONE Date Last Production Vehicle
No. Built 0 as of 11-2-61

17. STATUS DESIGN STUDY Date 1 NOV. 1968

Figure 3-1 Vehicle Characteristics Sheet



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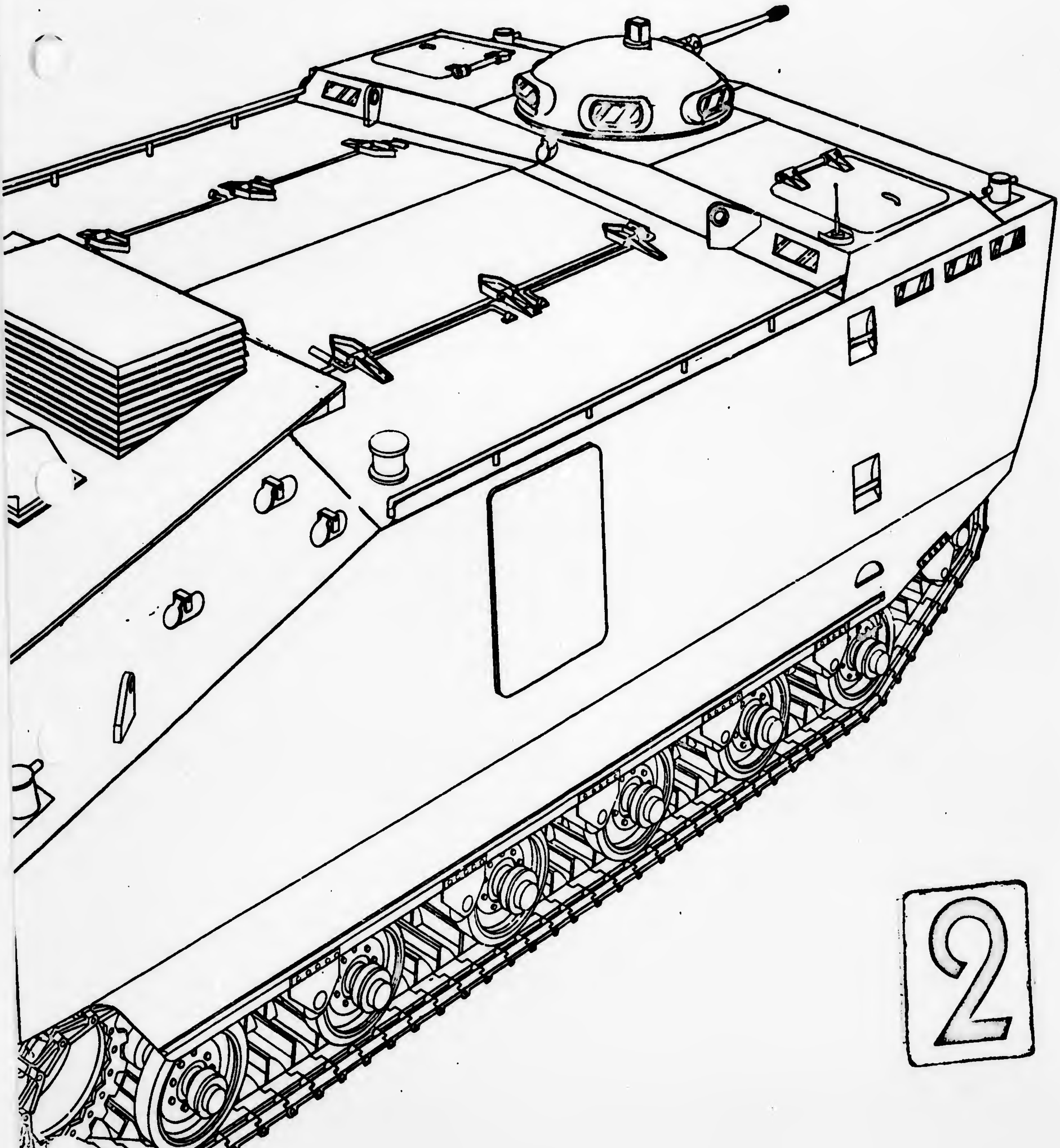


3/4 STARBOARD
STERN VIEW

TD 106522

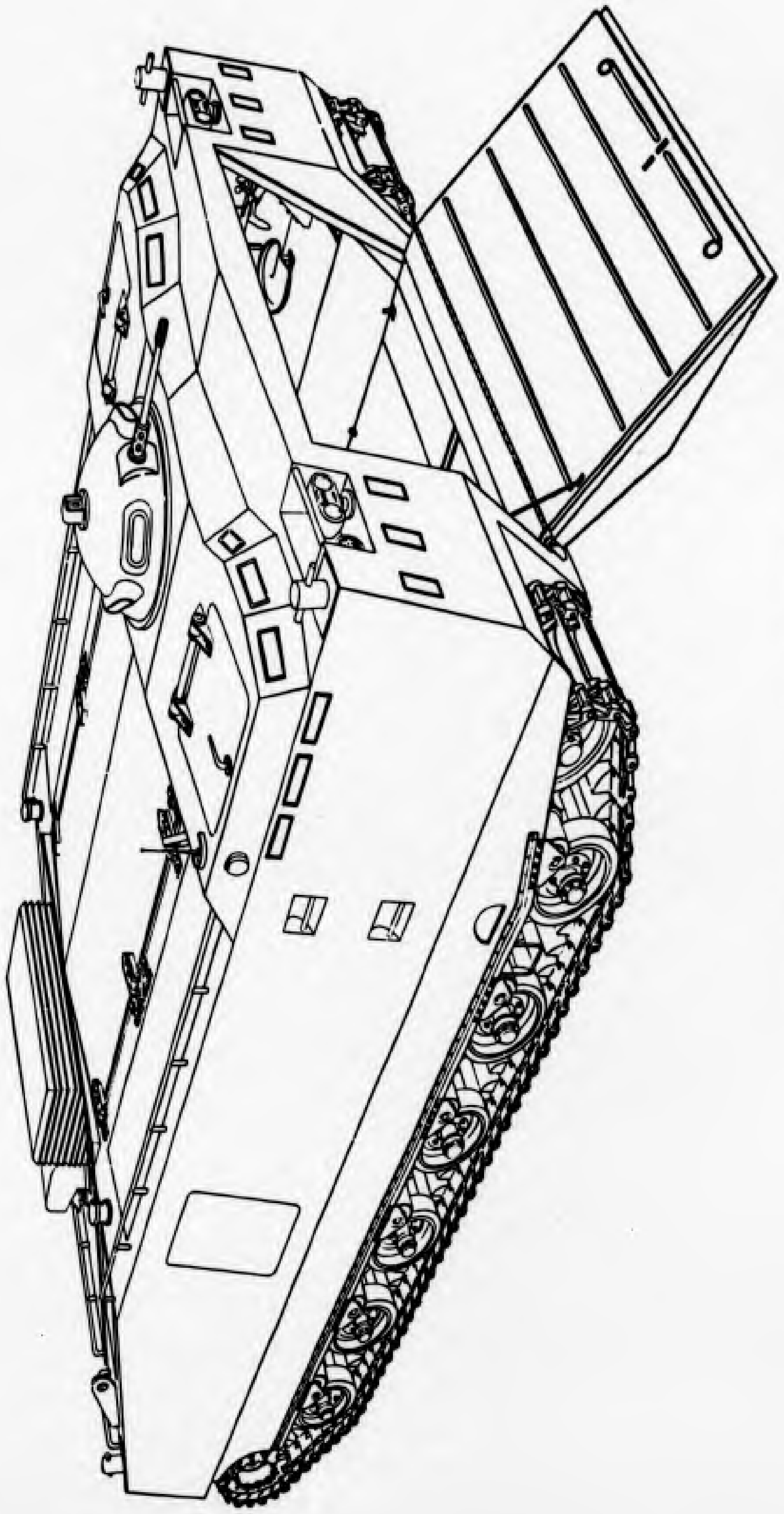


Figure 3-2   3/4 Starboard Bow View

4. DESCRIPTION OF MAJOR COMPONENTS

4.1 ENGINE AND RELATED SYSTEMS

4.1.1 STUDY OF ENGINE REQUIREMENTS

The purpose of this study is to present the basis on which the Power Train System, with related components and accessories, was selected to meet the requirements of the LVTPX-11 vehicle specifications.

4.1.1.1 VEHICLE LAND PERFORMANCE ANALYSIS

The specifications state that land performance should be the guiding factor for determining engine power requirement. To determine land performance, it was first necessary to establish vehicle rolling resistance. Considerable background information was derived from the "Research, Investigation and Experimentation in the Field of Amphibian Vehicles" report. This information is tabulated as follows:

Vehicle	GVW-Lbs	Surface & Lb/Ton Resistance Factor	
LVT-1	17,300	Dirt Surface:	276 @ 3 mph 340 @ 11.5 mph
		Concrete Surface:	196 @ 3 mph 232 @ 12.5 mph
LVT-3	18,000	Hard Surface:	158 @ 11.75 mph
LVT-4 (With steel dry pin track)	15,700	Hard Surface:	192 @ 15.75 mph 172 @ 16 mph
LVT-4 (With rubber-bushed Borg-Warner track)		Hard Surface:	128 @ 12 mph
M-3 Light Tank	25,225	Hard Surface:	94 @ 12 mph
M-5 " "	30,800	Hard Surface:	90 @ 16 mph
M-24 " "	37,750	Hard Surface:	80 @ 34 mph

Other approximate vehicle rolling resistances were determined from maximum speed data quoted in this same reference report, all on hard-surfaced roads.

Vehicle	Weight	Lb/Ton Factor	Type of Track
LVTPX-3	62,980	187.0 @ 30 mph	Steel Pin
LVT-A	75,000	142.5 @ 33.4 mph	" "
LVTP-5	81,780	145.0 @ 30 mph	Type VIII
LVTH-6	86,600	135.0 @ 30 mph	" "
M-59	41,800	135.0 @ 32 mph	Rubber Bushed
M-113	22,900	141.0 @ 40 mph	" "

Other vehicle rolling resistance factors were obtained from test data on medium tanks to complete the survey. These factors are for rubber bushed track only.

Vehicle	Weight	Lb/Ton Factor
M-48	89,000	90 @ 5 mph - 108 @ 25 mph
M-48A2	96,000	90 @ 5 mph - 110 @ 25 mph
M-60	103,000	92 @ 5 mph to 109 @ 25 mph (Extrapolated to 40 mph = 120 lb/ton)

At this point, reference should be made to Section 4.4 on suspension and track selection. This section states that Type VIII track was selected, and this corresponds to that used on the LVTP-5. Taking into consideration such differences between the LVTP-5 and LVTPX-11 as shock absorbers, suspension, track length, number of road wheels and air resistance, a figure of 143 lbs/ton rolling resistance at 40 miles per hour was derived. This point was plotted on Figure 4-1 and a curve drawn shows the slope of which parallels that obtained from various actual test reports of other vehicles.

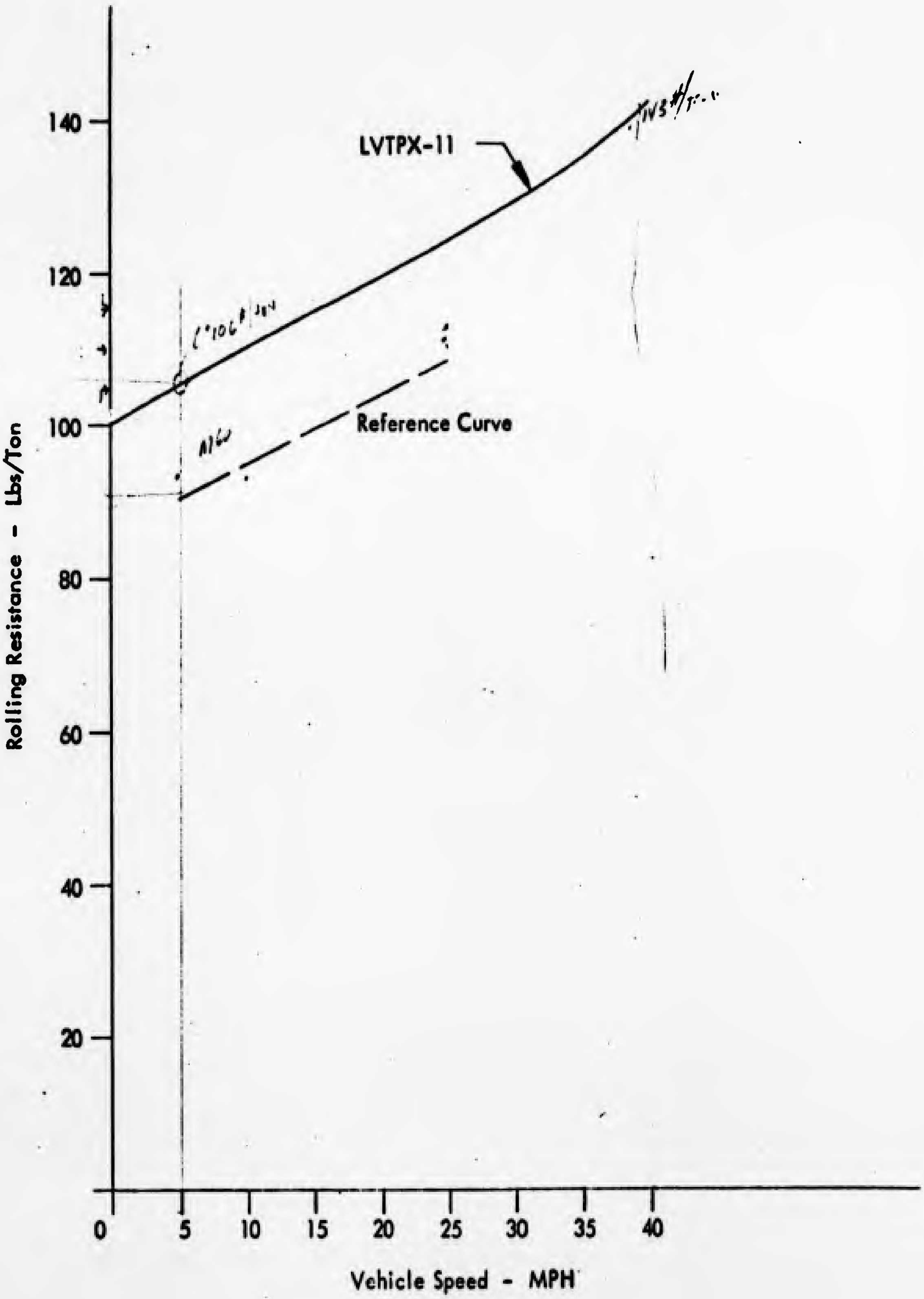


Figure 4-1 Rolling Resistance Curve

Using this data, rolling resistance horsepower values were calculated. It should be noted that air resistance was included in these calculations because all the resistance values were obtained from towing tests of complete vehicles. The results are tabulated in Table 4-1.

Table 4-1

Rolling Resistance Horsepower

Speed MPH	R.R. Lb/Ton	R.R. Lbs.	R.R. HP
1.0	101.5	1780	4.74
1.5	102.0	1786	7.14
2.0	102.5	1794	9.57
2.5	103.0	1804	12.05
5.0	105.5	1850	24.65
10.0	110.0	1925	51.4
15.0	115.0	2014	80.5
20.0	120.0	2100	112.0
25.0	125.0	2190	146.0
30.0	130.0	2280	182.5
35.0	135.5	2370	221.0
40.0	143.0	2500	266.5

A determination of grade resistance horsepower was then made out and is tabulated in Table 4-2.

The results of these calculations show that 266.0 hp are required at the sprocket to meet the 40 mph vehicle speed requirement. Starting with this figure and allowing for 90% final drive efficiency, 82.5% transmission efficiency and the necessary accessory losses, a 350 bhp engine is required. In addition to this, power for engine and transmission cooling would have to be provided.

Table 4-2

Grade Horsepower

Speed MPH	Per Cent Grade 5	10	15	20	25	30	35	40	45	50	55	60	65	70
1.0	4.66	9.3	14.0	18.3	22.6	26.8	30.8	34.6	38.2	41.7	45.0	48.0	51.8	53.5
1.5	7.0	13.9	21.0	27.4	33.9	40.2	46.2	52.0	57.4	62.6	67.5	72.0	76.2	80.2
2.0	9.3	18.6	28.0	36.6	45.2	53.6	61.7	69.2	76.5	83.4	90.0	96.0	101.5	107.0
2.5	11.7	23.3	35.0	45.8	56.5	67.0	77.1	85.6	95.6	108.5	112.5	120.0	127.0	134.0
5.0	23.3	46.6	70.0	91.5	113.0	134.0	154.2	173.2	191.5	208.5	225.0	240.0	254.0	267.5
10.0	46.6	93.0	140.0	183.0	226.0	268.0	308.0	346.0	382.0	417.0	450.0	480.0	518.0	535.0
15.0	70.0	139.0	210.0	274.0	339.0	402.0	462.0	520.0	574.0	626.0	675.0			
20.0	93.0	186.6	280.0	366.0	452.0	536.0	617.0	692.0						
25.0	117.0	233.0	350.0	458.0	565.5	670.0								
30.0	140.0	279.0	420.0	548.0	678.0									
35.0	163.5	325.0	490.0	640.0										
40.0	187.0	372.0	560.0											

$$GHP = \frac{\text{Veh. Wt(Lbs)} \times \text{MPH} \times \text{Sine of Grade Angle} \times 88 \text{ ft/min/mph}}{33{,}000 \text{ ft-lb/min/hp}}$$

4.1.1.2 ENGINE SELECTION

The fuel specifications given allowed consideration of three basic types of engines; namely, spark ignition, compression ignition, and gas turbine. To begin with, a survey was made of all applicable engines of these three types.

(1) Spark Ignition. No high speed, lightweight gasoline, spark ignition engines are commercially available in the 350-400 bhp range required. Obsolescent Ordnance designs are available, but should not be considered at the beginning of a new development program. A list of presently available commercial gasoline engines which might be utilized by "twinning" is shown in Table 4-3.

(2) Compression Ignition. The review of compression ignition engines falls into two categories. Those of the 300-400 hp range and those in the 150-200 hp range which would be applicable to "twinning". These are shown in Table 4-4.

(3) Gas Turbine. Gas turbine engines which would have possible application to this project were selected from the 300-400 hp range. These are shown in Table 4-5.

A comparison of the relative advantages of utilizing twin engines with individual transmissions connected to a controlled differential for steering purposes, against the use of a single integrated engine-transmission power package, was then made. This study showed that a twin engine package could result in a lighter weight power package. However, it is more complex, requiring more installation lines, controls, wires and supports, and resulted in undesirable space arrangement and vehicle stability problems when the troop compartment seating arrangement was finalized. Also, air cooled engines were not available. Based on the above findings, the single engine-transmission power package was selected.

Table 4-3

Commercial Spark Ignition Engines

Make & Model No.	HP of Bare Engine	Torque (BE)	WT	Cyl.	Dimension W H L
Chrysler					
A710B	215 @ 4000	332 @ 2800	688	8	27 x 33 x 34
Chevrolet					
Workmaster	230 @ 4400	335 @ 2800	802	8	30 x 34 x 42
Special	185 @ 4000	315 @ 2200	810	8	30 x 37 x 42
Continental					
R6602	232 @ 2800	482 @ 1200	1525	6	
R6513	192 @ 2800	420 @ 1200	1525	6	
Ford					
EDL-2V	206 @ 3400	274 @ 2400	963	8	35 x 45 x 36
EDL-4V	226 @ 3600	343 @ 2300	963	8	35 x 45 x 36
GMC					
401	210 @ 3400	377 @ 1400		6	30 x 40 x 40
International					
UV401	206 @ 3600	340 @ 1800	1310	8	37 x 41 x 41
461	226 @ 3600	378 @ 1900	1320	8	37 x 41 x 41
Reo					
OV207	207 @ 3400	354 @ 2400	1211	8	28 x 37 x 40
OV235	235 @ 3400	412 @ 2400	1211	6	26 x 34 x 46
Roline					
TH570	238 @ 2600	508 @ 1800	1250	8	31 x 42 x 45
H884	330 @ 3000	780 @ 1600	2000	V8	33 x 47 x 51
Waukesha					
145G2	250 @ 2400	630 @ 1100	1810	6	25 x 46 x 57

Table 4-4

Compression Ignition (Diesel) Engines

Make & Model No.	HP of Bare Engine			Torque (BE)			WT	Cyl.	Dimension W H L
Allis-Chalmers									
11000	210	@	2200	540	@	1800	1980	6	30 x 47 x 51
21000	350	@	2100	930	@	1500	3040	6	38 x 49.5 x 57
Chrysler LVMS-377	230	@	3400	420	@	2200	900	V8	32 x 31 x 39
Caterpillar									
D333T	205	@	2200	555	@	1600	2140	6	32 x 39 x 58
D333 (Future)	320	@	2400	770	@	1600	1675	6	32 x 38 x 58
LDS750	425	@	2400	1130	@	2000	1945	5	36 x 43 x 46
Continental									
SD6802	225	@	2200	620	@	1300	2150	6	
VD6803	200	@	2800	460	@	1700	1640	8	34 x 47 x 44
Continental Military									
AVDS750	425	@	2800	870	@	2000	2552	V8	38 x 32 x 41
Cummins									
NT380	380	@	2300	955	@	1600	2750	6	40 x 45 x 58
VT8-430	430	@	2500	1005	@	1750	3020	V8	41 x 46 x 37
P & H									
487H-18AT	220	@	1800	630	@	1600	1480	4	40 x 49 x 46
687H-18AT	320	@	1800	980	@	1600	1894	6	31 x 34 x 60
GM									
6V53	195	@	3800	423	@	1500	1540	6	36 x 39 x 35
8V71	336	@	2300	805	@	1200	2505	V8	
8V71T	390	@	2300	840	@	1800	2500	V8	41 x 40 x 41
Hercules									
D426T	180	@	2600	406	@	1900	1450	6	28 x 28 x 45
DV8662	200	@	2600	400	@	1700	1450	V8	35 x 24 x 52
International									
VDT817	385	@	2100	1090	@	1300	3500	6	40 x 57 x 61
Lycoming									
AVMT-625	413	@	2800	850	@	2000	1100	V8	35 x 29 x 49
Waukesha									
148DKB	200	@	2100	584	@	1000	2320	6	29 x 47 x 56
148DKBS	280	@	2100	706	@	1800	2420	6	29 x 55 x 56

Table 4-5

Gas Turbine Engines

Make & Model No.	HP of Bare Engine	Regen.	Free Power Turbine	Wt.	W.H.L. Dimension	Type
Allison						
GMT305	225 @ 3400	Yes	Yes	760	31x28x40	Automotive
Boeing						
502-10-W	325 @ 2900	No	Yes	325	24x24x42	Industrial
520	400 @ 6200	No	Yes	325	25x26x58	"
Chrysler	293 @ 3600	Yes	Yes	859	30x32x41.5	Automotive
Continental						
T72-T-2	450 @ 5900	No	Yes	210	21x21x42.5	Aircraft
Garrett (Air Research)						
DVP331	330 @ 3000	No	Yes	950	30x32x57	Automotive
Pratt & Whitney						
PT6	466 @ 2200	No	Yes	250	19x19x61.5	Aircraft
Solar						
T350	350 @ 6000	No	Yes	195		Industrial

4.1.1.3 ENGINE TYPE SELECTION

After listing applicable engines of the three types, namely spark ignition, combustion ignition, and gas turbine, the final selection was made. Following is a relative characteristic chart with an explanation of each characteristic considered.

Relative Characteristic Rating

	Performance	Weight	Vehicle Adaptation	Multifuel Capabilities	Use in Other Heavy Duty Vehicles	CO Safety	Fire Safety	Fuel Consumption	Acceleration Responses	Engine Braking	Cost	Ease of Maintenance
Spark Ignition	2	2	2	3	2	3	3	2	1	1	1	2
Compression Ignition	1	3	1	1	1	1	1	1	1	1	2	2
Gas Turbine	3	1	3	1	3	1	1	3	3	3	3	1

Key: 1 - Most Desirable
2 - Intermediate
3 - Least Desirable

Definitions of engine rating characteristics:

(1) Performance -- A combination of engine capabilities to meet all specified vehicle performance requirements.

(2) Weight -- To allow for increased ballistics, maximum stability, and vehicle balance.

(3) Vehicle Adaptation -- Engine most adaptable to vehicle (size) configuration and related components.

(4) Multifuel Capabilities -- Engine capable of using varying fuels from Diesel No. 2 to gasoline.

(5) Use in Other Heavy Duty Vehicles -- Engine most used in heavy duty vehicles, both military and commercial.

(6) CO Safety -- Engine exhaust containing lease amount of CO, regardless of fuel burned.

(7) Fire Safety -- Based on capability of engine to burn fuels of low volatility.

(8) Fuel Consumption -- Lowest fuel consumption at both full and part load.

(9) Acceleration Response -- Engine capability for instant acceleration response.

(10) Engine Braking -- Engine capability for providing vehicle braking.

(11) Cost -- Lowest cost per engine, both for development and in production.

(12) Ease of Maintenance -- Least requirement for maintenance and most easily performed maintenance.

The preceding tabulation of characteristics shows that the compression ignition engine is the best selection for the LVTPX-11 vehicle. It is principally superior to the spark ignition engine, as noted in the chart, by its multifuel capability, greater fuel economy, and CO and fire safety. The very low idle fuel consumption of the compression ignition engine permits its use during standby operation, and removes the need for an auxiliary engine.

The gas turbine engine has several outstanding characteristics, but its development has not yet reached the point where it can be recommended for the initial design of the LVTPX-11. At this time, the compression ignition engine is superior to the turbine in cost, fuel economy (particularly at light loads and idle), engine braking and acceleration response, due to the fact that a regenerative turbine of suitable power does not exist. The results of such current test and development programs as the Jered-GE turbine installation in a modified LVTP-5 and the Army-Navy 600 hp turbine developments by Ford and Solar should provide answers regarding the suitability of the turbine for tracked vehicles, and perhaps provide the engine as well. As was indicated earlier in this report, ample consideration was given in the basic design to provide for a future turbine installation.

4.1.1.4 FINAL ENGINE SELECTION

Preliminary review and discussion with U.S. Army Ordnance personnel and engine manufacturers narrowed down the field of C.I. engines to three which required further study at this time. The characteristics of the Lycoming engine listed in the table is also very attractive for this application, but it is a future development item not yet under test. When the test program is carried out and the engine proven, then it should be considered as a possible replacement for the engine selected below. The three present engine choices are:

(1) General Motors (Detroit Diesel) 8V71T
(2) Continental Motors AVDS 750-1
(3) Caterpillar LDS 750-1

Both the GM 8V71T and Caterpillar LDS750-1 are liquid-cooled, while the Continental engine is air-cooled. Again, a relative characteristic rating chart is provided with an explanation given for each characteristic listed.

Relative Characteristic Rating

	Weight	Size	Vehicle Adaptation	Performance	Cost
GM 8V71T	2	3	2	2	1
Continental AVDS 750-1	1	1	1	1	2
Caterpillar LDS 750-1	1	2	2	2	3

$5000 today

Key: 1 - Most Desirable
2 - Intermediate
3 - Least Desirable

Definition of engine rating characteristics:

(1) Weight -- Total weight of engine, including necessary cooling equipment.

(2) Size -- Based on total cubage of engine, including cooling equipment.

(3) Vehicle Adaptation -- Engine rating, based on the proposed engine compartment configuration and the possible future incorporation of a gas turbine.

(4) Performance -- Engines rated as most capable of meeting specified land and water performance requirements.

(5) Cost -- Based on preliminary information from manufacturer as to prototype vehicle installation only.

To complete the above study, it was necessary to review the pros and cons of air-cooled versus water-cooled engines, as follows:

Air-Cooled	Water-Cooled
(1) Lower weight -- including cooling equipment.	(1) Higher weight -- including cooling equipment.
(2) Coolers integral with engine.	(2) Coolers not made integral with engine.
(3) Larger basic engine size, but includes coolers.	(3) Smaller basic engine size, but does not include coolers.
(4) Smaller cooling air flow required (13,000 cfm).	(4) Larger cooling air flow required (21,000 cfm).
(5) Smaller air inlet and exhaust grill area.	(5) Larger air inlet and exhaust grill area.
(6) Cooling air flow must be routed to engine.	(6) Cooling air flow pattern can be varied.
(7) Intricate air inlet and exhaust grill automatic mechanisms for water operation.	(7) Must provide water to water radiators, pump system and mechanically-actuated dampers for water operation.

All the above characteristics were very carefully considered as to their effect on over-all vehicle performance. From this, the Continental AVDS 750-1 air-cooled engine was selected for the LVTPX-11.

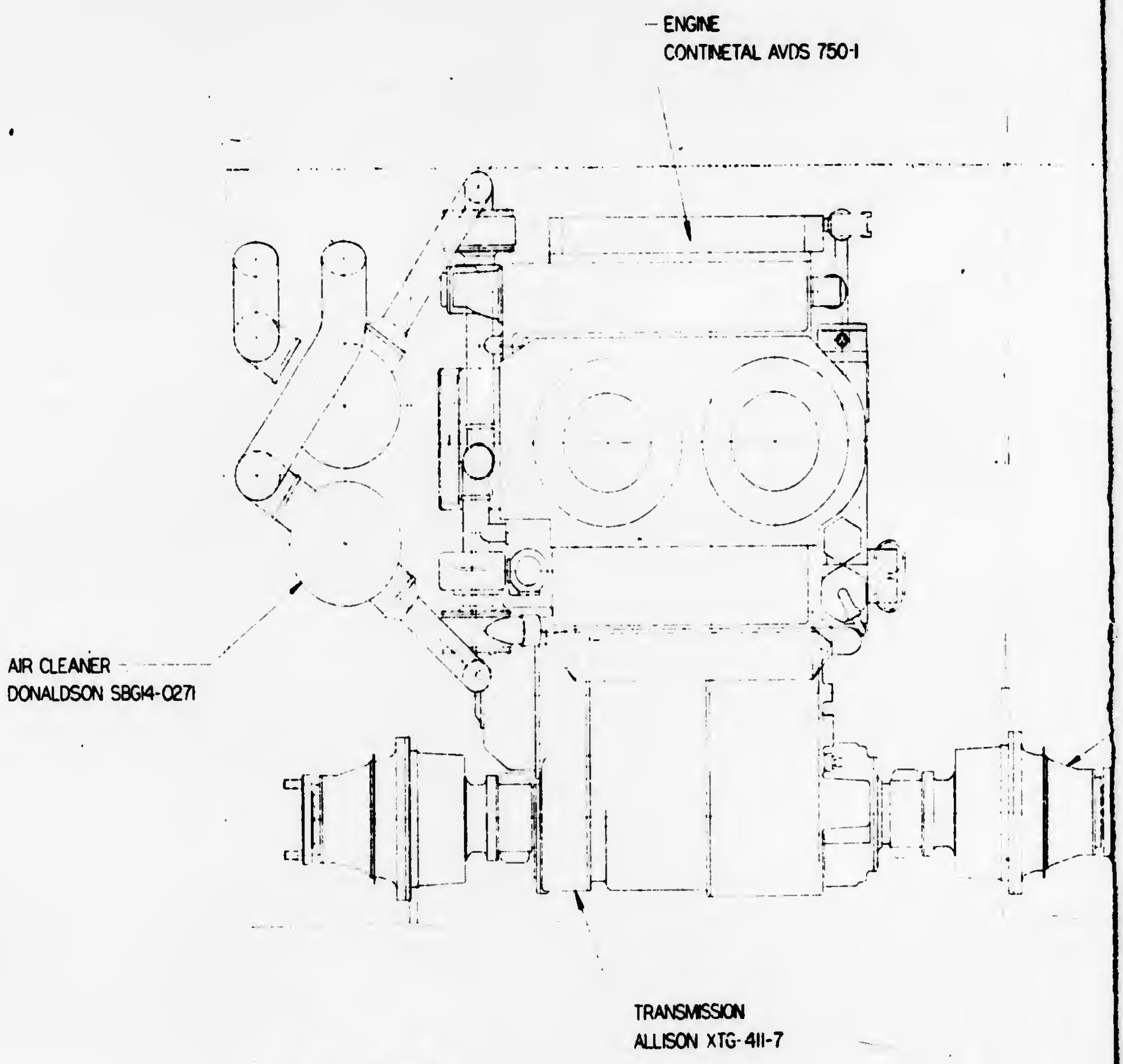
4.1.1.5 ENGINE DESCRIPTION

The Continental AVDS 750-1 engine is a compression ignition, eight cylinder, 90 degrees upright Vee-type engine, with a bore of 4.875 inches and stroke of 5.0 inches to provide a total displacement of 746 cubic inches. It has a 17 to 1 compression ratio, is air-cooled, turbo-supercharged and after-cooled, and rated at 425 gross brake horsepower at 2800 rpm. The minimum brake specific fuel consumption is 0.41 lbs/bhp-hr. It has a wet weight of 2573 pounds, which includes 9 gallons of oil and both engine and transmission oil coolers.

Will this cause a response problem

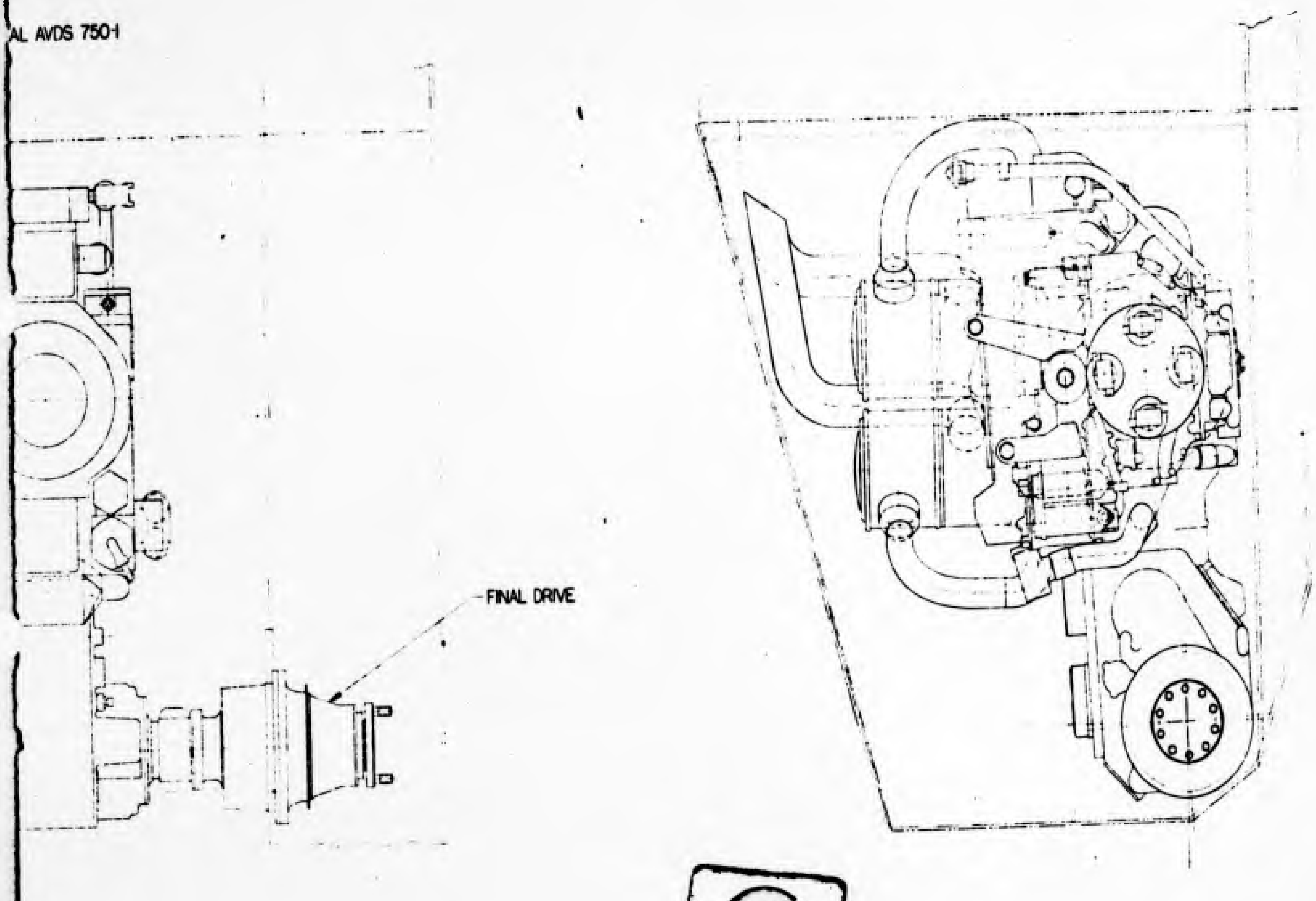
The fuel injection unit and associated drive system is mounted in the engine Vee. Each cylinder bank is equipped with an exhaust-driven turbocharger. The engine is provided with a wet sump, pressure lubrication system.

The engine is designed for military vehicles where compactness, air cooling, low fuel consumption and low weight are of primary importance and, therefore, is ideally suited to the LVTPX-11 vehicle. (Refer to TD-106523).



ENGINE
CONTINETAL AVDS 750-1
AIR CLEANER
DONALDSON SBG14-0271
TRANSMISSION
ALLISON XTG-411-7

1



411-7

2

GENERAL MACHINERY
ARRANGEMENT TD106523

4.1.2 ASPIRATION AND EXHAUST SYSTEM

The engine compartment air induction system for the LVTPX II provides for both combustion air and cooling air. A cross sectional view of the system is shown on TD-106524 (following Page 4-18). The entire system, including both intake and exhaust shrouding, is an integral part of the engine compartment cover.

Two separate air intake points are provided; one for engine and transmission cooling air. It should be noted that the system does not employ any water to air or water to water heat exchangers or heel coolers.

The combustion air intake is an open vent which has been sized to provide the air flow required for engine combustion. The intake pipe is concentric with a bilge pipe so that sea water which enters will be directed to the bilge pump. Incoming air must make a 180 degree turn in its flow path which will effectively purge incoming water from the air flow. The engine air cleaner inlets are located as close as possible to the combustion air intake vent, and both units are positioned in the forward area of the engine compartment so as to be above the water line even if the engine compartment becomes completely flooded (refer to TD-106535 following Page 4-64). Since the engine is capable of operating even though completely submerged in water, it will be possible to operate the LVTPX II with a flooded engine compartment.

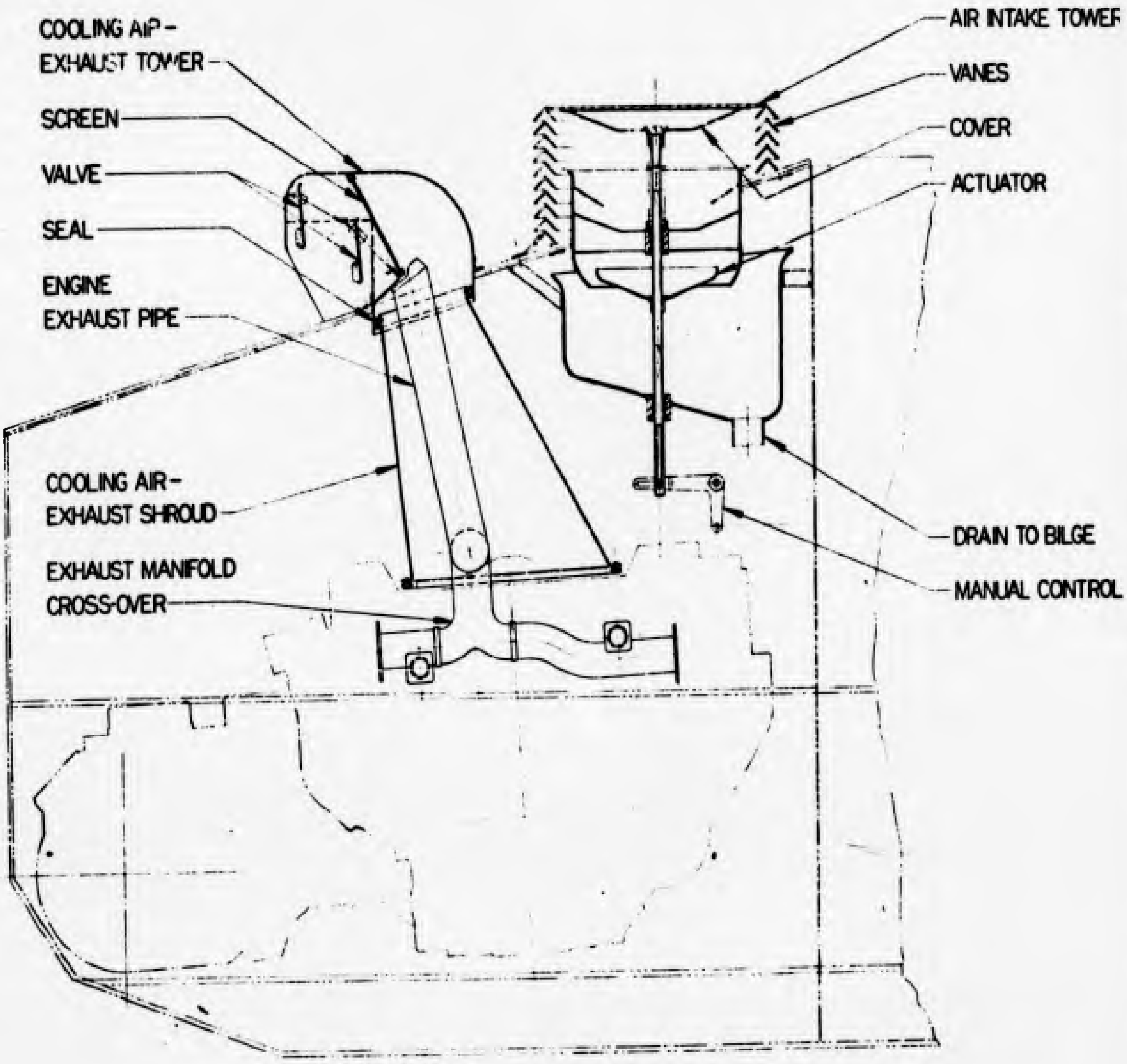
The main air intake tower on the engine compartment cover has an effective area of 4 square feet to accommodate an air flow rate of 13000 CFM to provide for engine and transmission cooling. External to the engine compartment hatch are a set of vanes which will prevent the entry of casual water. Inside the tower and vane structure is a cover which is spring loaded so as to keep the cover in the open position. When the vehicle becomes immersed in surf, water will enter the system through the vanes and will be directed by a collector ring into an actuator which is mounted on the cover support shafts. The weight and dynamic pressure of the incoming water impinging on and filling the actuator will overcome the spring force and close the cover. The actuator is provided with weep holes so that the water it collects will drain to bilge. However, should the vehicle remain immersed after the actuator has drained, the static pressure of the water on the top of the cover will seal the air intake cover until the vehicle re-surfaces.

The actuator and cover shaft are provided with a linkage which will enable the operator to override the spring and keep the cover in either the closed position, which might be desirable if the vehicle is to be parked for a long period in rain or snow, or in the open position should the spring or linear ball bushings on the actuator shaft fail at a critical time. The incoming air must change the direction of its flow path 180 degrees, which will purge entrained water from the air flow.

To provide clean air to the engine two Donaldson SBG 14-0271 dry type air cleaners are provided. One cleaner per engine bank will provide air to the turbosupercharger. This heavy duty cleaner has a reusable paper filter element along with a centrifugal cleaning stage to provide clean engine combustion air under all vehicle operating conditions.

The exhaust system terminates in an exterior tower, directly to the rear of the main air intake tower. Its outlet faces rearward to prevent recirculation. Engine exhaust is piped from each manifold into a tee and on through a single pipe to the tower. The exhaust pipe connects to an extension which has a spring-loaded, normally closed cover.

Engine and transmission cooling air, which is brought into the engine compartment through the intake tower, is circulated past the oil coolers mounted on the engine, by two integrally-mounted engine fans. The fans then force the air through the shrouded area between the engine and top deck into the exhaust tower. The tower incorporates a screen and two normally closed damper valves. The valves are forced open by the blast of the engine fans. They will be closed by wave action, or when the vehicle is submerged. The exhaust area through the damper valves is 2.5 sq. ft.



ENGINE AIR
INDUCTION SYSTEM TD 106524

4.1.3 FUEL SYSTEM

Incorporation of a compression ignition engine with a relatively low specific fuel consumption, as noted under engine selection, results in a weight savings of not only fuel quantity but also in the fuel container. That is, the fuel container is smaller, taking up less space and weight. Calculations performed to determine the amount of fuel necessary to meet the LVTPX-11 water and land range specifications are presented in Section 6.1.1. From these calculations it is seen that 180 gallons of usable fuel are to be provided in the fuel tank. Actual fuel tank capacity was increased to approximately 190 gallons to allow for unusable fuel, fuel pumps and gauges.

Location for fuel storage was arrived at after a thorough study of vehicle stability, vehicle component weight distribution and troop and engine compartment space requirements. This resulted in a design using two fuel tanks, one on either side of the fore and aft centerline of the vehicle under the floor of the troop compartment. Integral fuel tanks were considered to utilize the hull structure; however, this presented a critical field maintenance problem in case of fuel tank puncture. Based on ease of maintenance, simplicity, weight, and maximum utilization of available space, rubber fuel cells were selected for the LVTPX-11 vehicle. The fuel tanks would be designed to fit the hull structure with allowances for routing of necessary filler, vent and return lines.

The fuel system consists of four major sections. Fuel filler and venting components; fuel tank, its integral components and the fuel supply line between the fuel tanks and engine; the on-engine components and fuel return line; and the primer pump system. (See Drawing No. TD 106525 following Page 4-22).

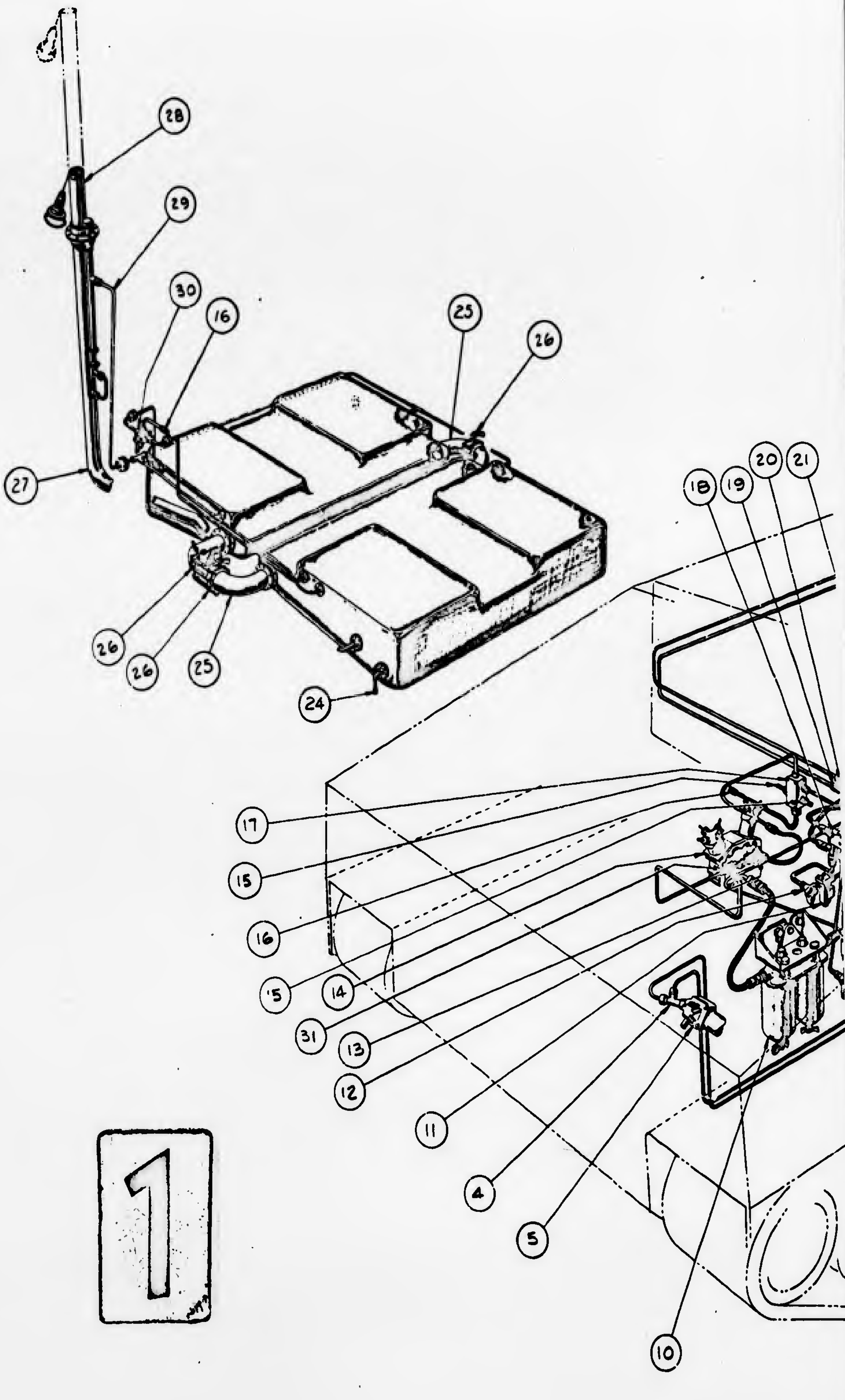
The fuel filler tube is comprised of the stationary main tube running from the top deck to the fuel tank and an extension tube with cap which telescopes inside the main filler neck. This extension tube is utilized during sea refueling operations. The filler neck would be sized to provide a minimum fuel tank filling rate of 50 gallons per minute. From experience it has been found that a properly sized and arranged venting system is the determining factor in meeting this filling rate. The filler neck extending from the top deck to a cross over at the center of the vehicle floor would be straight where possible and gradual bends would be provided as necessary to keep fuel flow smooth. The rear cross over incorporates two 2-inch butterfly valves and the front cross over has one 2-inch butterfly valve for fuel tank isolation. That is, if one tank is punctured that tank can be isolated from the other which can still be refueled and utilized. Breather tubes are connected to both ends of both tanks to prevent air pockets during filling and slope operation. The vent tubes are connected together and terminate at the top end of the main filler neck.

The fuel tanks will be made of rubber meeting all construction and test requirements of Specification MIL-T-6396. They have been designed so that either side is interchangeable with the other. A single electric fuel pump is inserted in each tank, so positioned as to positively supply fuel during all the specified slope operations and to prevent vapor lock when using highly volatile fuels. A fuel gage sending unit of the float type will be inserted in each tank near the center. A fuel tank drain tube with valve will be at the rear of each tank. It will be possible to drain the fuel tanks to the bilge area or to the outside through removable hull drain covers. Fuel from the two electric pumps is routed to a double check valve which allows fuel flow from either or both fuel tanks into a single line. The single line from the check valve terminates in a quick disconnect fitting which allows for easy fuel line connection or disconnect from the engine for power package installation and removal.

From the quick disconnect fitting fuel flows through a flexible hose to the "on engine components." All on engine components are a part of the engine as received from the manufacturer, however, they are described here to complete the fuel system. Fuel first flows to the primary filter which has a cleanable type filter element. From here the fuel flows to a combination pressure-check valve. This check valve is incorporated as part of the primer pump system and its purpose is described later. Fuel is then routed to an engine driven fuel pump which increases the pressure to that necessary for proper metering in the injector pump. The fuel then travels to two replaceable secondary filter elements, then through the main fuel solenoid valve prior to entering the injector pump.

Since excess fuel is supplied to the injector pump, injectors and manifold heaters, a return fuel line system is required. A check valve is incorporated in the fuel return line from the injector pump to prevent fuel re-entering the injector pump after the fuel supply solenoid has been shut off. The fuel return lines are then connected together at a manifold and one single line returns to the fuel tank. A quick disconnect is also placed in this line for ease of engine removal. A selector valve is provided near the fuel tank to route the return fuel to either or both tanks. This again is to allow isolation of one tank during emergencies.

A primer pump which also serves as a purge pump is located at the left of the driver's position. The primer pump is used to provide fuel to the manifold heaters for cold weather starting. A swtich in the handle opens the solenoid valves in the manifold heater fuel supply and return lines to allow fuel to flow to the heaters during pumping. The primer pump can also be utilized to purge the fuel lines of air. This is accomplished by not opening the manifold heaters solenoid valves but pumping until pressure builds up to approximately 90 psi, to open the pressure valve in the combination pressure check valve. Fuel then flows through all the engine supply lines and components and on through the fuel return lines to the fuel tanks.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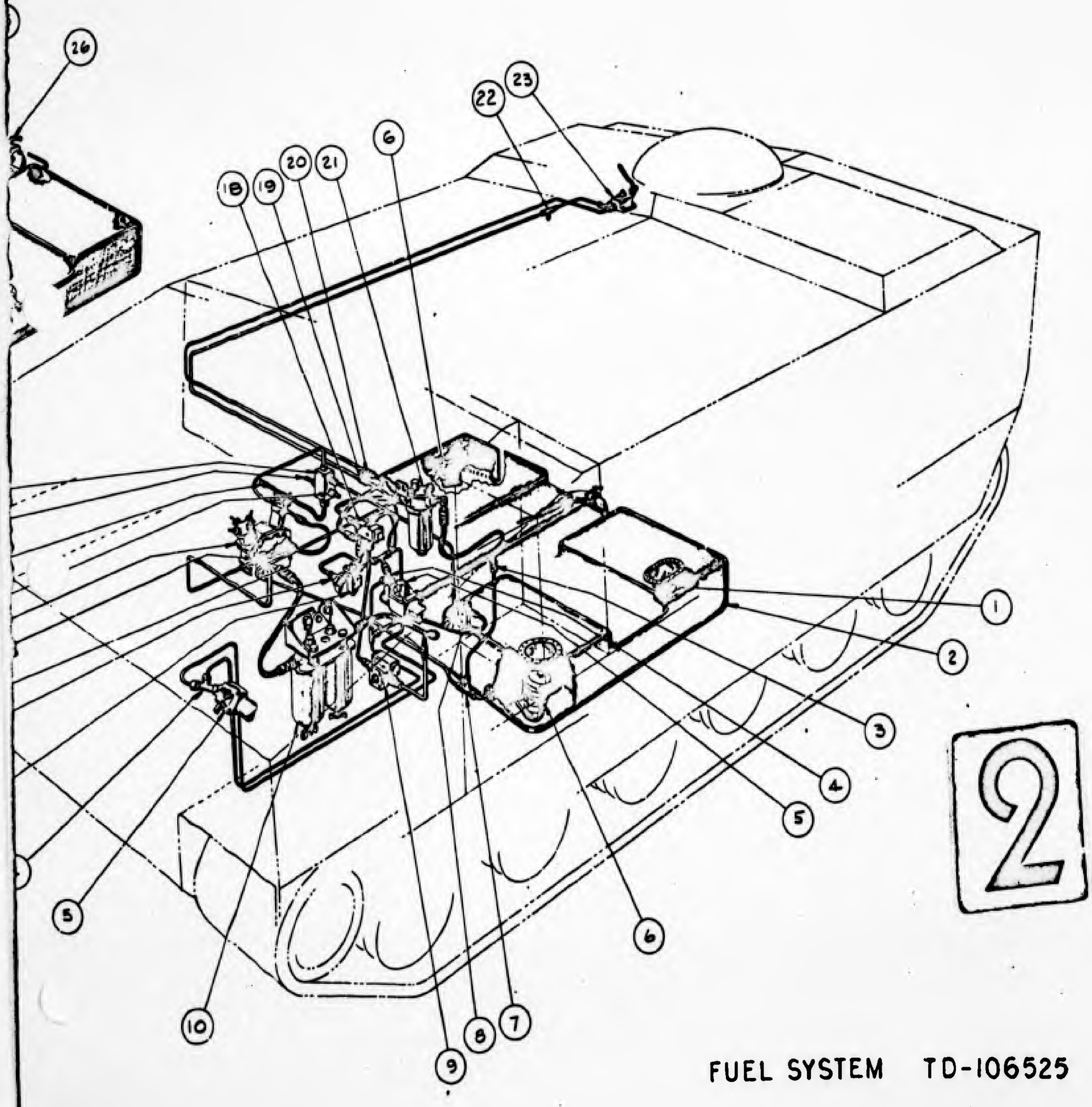


FUEL SYSTEM TD-106525

4.2 TRANSMISSION AND FINAL DRIVE

4.2.1 TRANSMISSION

The number of automatic transmissions suitable for military tracked vehicles, incorporating steering, braking and gearing, all in one single unit, is quite limited. Of this limited number, only the Allison cross-drive type is available from current production at a low unit cost. The cross-drive transmission makes possible automotive-type steering and braking control instead of steer and brake levers as required with a separate transmission and controlled differential combination. Several Allison transmission models were reviewed, including the XTG-250 and XTG-411. The XTG-250 transmission has pivot steer and is lighter in weight than the XTG-411 series transmission; however, it is applicable to vehicles of approximately 30,000 pounds gross weight and is rated for 250 net input horsepower.

how low? before you recommend XTG-250

Keeping in mind the relatively high speed of the LVTPX-11, it appears risky to recommend the XTG-250 for this 35,000 lb. vehicle. The XTG-411 is now in production for Ordnance self-propelled artillery in the 40,000 lb. weight class, insuring adequate capacity for the present application.

cost 5000 vs 12000
weight 1150 vs 2320* ← estimate dry
* has no final drive

The Allison XTG-411-7 transmission was selected for the proposed LVTPX-11 Vehicle. The dash 7 is the model of the XTG-411 series which mates with the Continental AVDS 750-1 engine. This transmission includes a hydraulic torque converter with lockup clutch in combination with planetary gearing giving four forward and two reverse speed ranges. It also incorporates clutch brake steering in first, second and reverse "1" ranges. Geared steer is provided in the remaining ranges. The maximum torque converter multiplication is 2.1:1. The dry weight of this transmission is 2320 lbs.

Low stall gives better eff

Only one modification must be made to adapt the transmission for an amphibious vehicle. The transmission normally has a second gear ratio of 4.0 to 1 (in relation to 4th gear), and geared steer in this ratio. This would provide a maximum track speed of only 10 mph, and a steering system with inadequate control in the water. By compounding the intermediate range of the main transmission with the low range geared steer planetary, a ratio of 2.95 to 1 and 13 mph track speed is achieved, together with the clutch-brake steering desired. This can be accomplished merely by a change in the valve body, and Allison has concurred with the feasibility of this change.

At the present time, extensive development work is being conducted on hydrostatic transmissions for all types of vehicles. Sunstrand Aviation Division of Sunstrand Corporation has a transmission presently under development for use in an Engineer Corps dual purpose crawler tractor, and they made a proposal to Chrysler for a transmission for the LVTPX II.

The hydrostatic transmission is a system which functions as the transmission, steer mechanism, and service brakes for the vehicle. These functions are accomplished with positive displacement, high pressure, piston-type hydraulic units. The advantages offered include infinitely variable speeds and torques to each output of the transmission, which is the key to obtaining improved mobility. Optimum torque is always available for acceleration and deceleration. Steering can be accurately controlled by modulating the output torques so that all turns from an infinite radius to pivot turns are possible.

Another possible advantage of the hydrostatic transmission compared to the torque convertor-mechanical type is in fuel programming. The hydrostatic transmission is capable of permitting the engine to run at its most efficient speed for a given vehicle load and speed.

Full throttle efficiency curves of the XTG-411-7 transmission and proposed Sunstrand hydrostatic transmission have been plotted on Figure 4.2 to show relative efficiencies versus vehicle speed. The hydrostatic transmission must be able to compete with the cross-drive transmission in overall efficiency, weight, space, cost and availability. As noted above, a hydrostatic transmission qualifies for the first three; however, the designer must also consider cost and availability.

Because this type of transmission does show promise for the future, its characteristics were considered and provision has been made in the design of the vehicle to incorporate it should a developed hydrostatic transmission become available. No actual pumps and motors of the size and type proposed have yet been tested, so that it is premature to recommend the incorporation of a hydrostatic transmission into a vehicle at this time.

Electric drive has also been discussed frequently in recent years. Several months ago, Chrysler Defense Engineering acted as vehicle consultant to two other firms which submitted proposals to Army Ordnance for electric drive systems for a 45 ton tracked military vehicle. The information obtained at that time showed the electric drive to have lower efficiency and to be heavier than the torque convertor-mechanical type. It is our opinion that electric drive should not be considered for the LVTPX II.

4.2.2 FINAL DRIVE

Much work has been done in the design and application of final drives to combat vehicles. The engine and transmission installation for the LVTPX II vehicle allows the use of a simple concentric planetary type final drive. For a maximum vehicle speed of 40 mph, the final drive ratio and sprocket diameter had to be selected concurrently to produce the

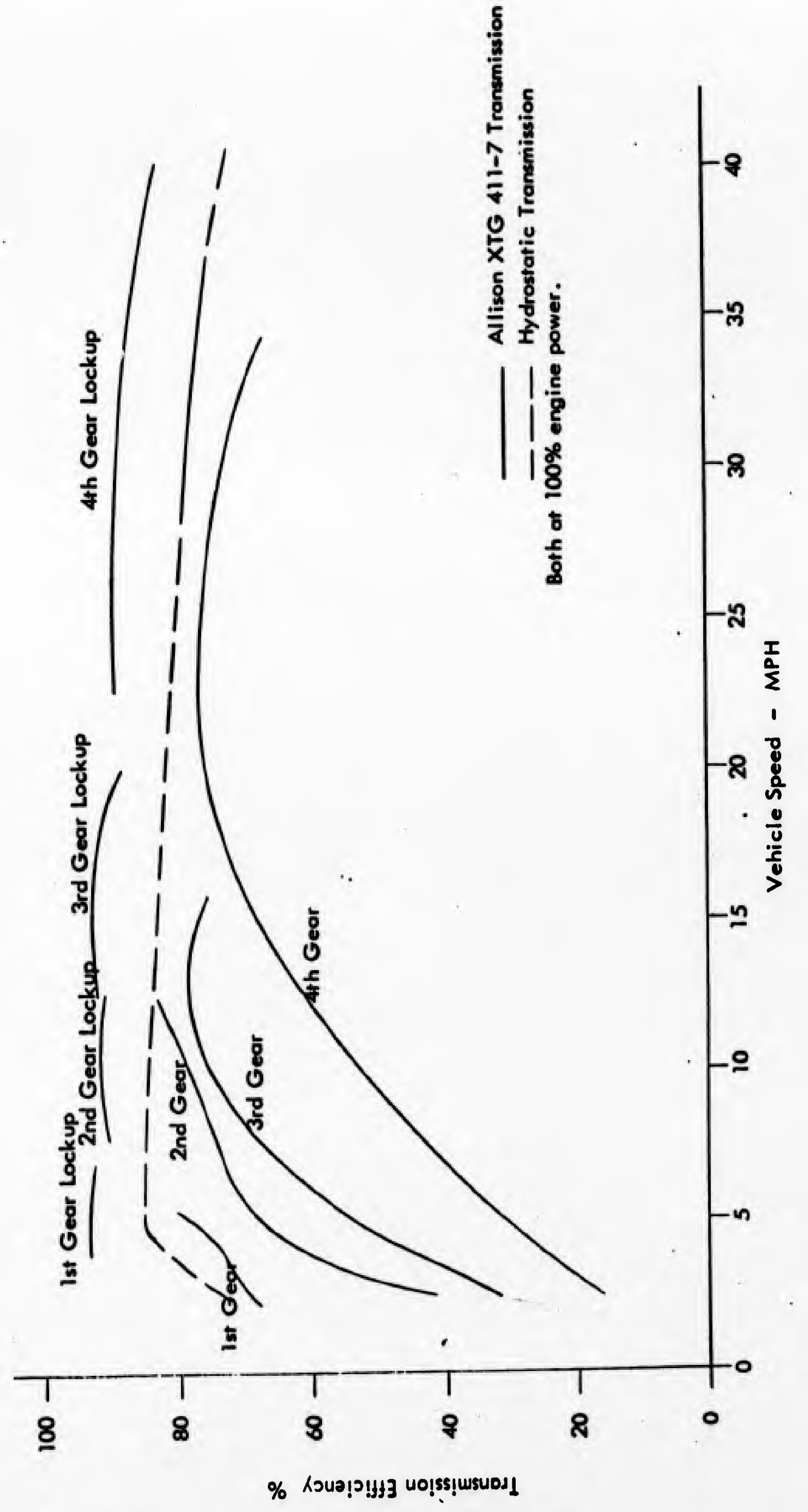


Figure 4-2 Transmission Efficiencies vs Vehicle Speed

optimum size and performance characteristics. After several trial calculations, the final combination of sprocket diameter and final drive ratio was determined. These are presented below:

Maximum vehicle speed of 40 mph. Sprocket pitch diameter of 23.85 inches with 15 teeth.

$$\frac{23.85 \text{ inches}}{12 \text{ inches/ft.}} \times 3.1416 = 6.25 \text{ ft/rev.}$$

$$\frac{88 \text{ ft/min.}}{6.25 \text{ ft/rev.}} = 14.1 \text{ rpm @ 1 mph}$$

at 40 mph sprocket rpm = 14.1 x 40 = 564.0

Utilizing performance curves of Continental AVDS-750-1 engine and Allison XTG-411-7 transmission, it can be seen that a transmission output speed of 2530 rpm is obtained at 2800 rpm engine speed.

$$\frac{2530 \text{ rpm}}{564 \text{ rpm}} = 4.49{:}1 \text{ Reduction ratio of final drive.}$$

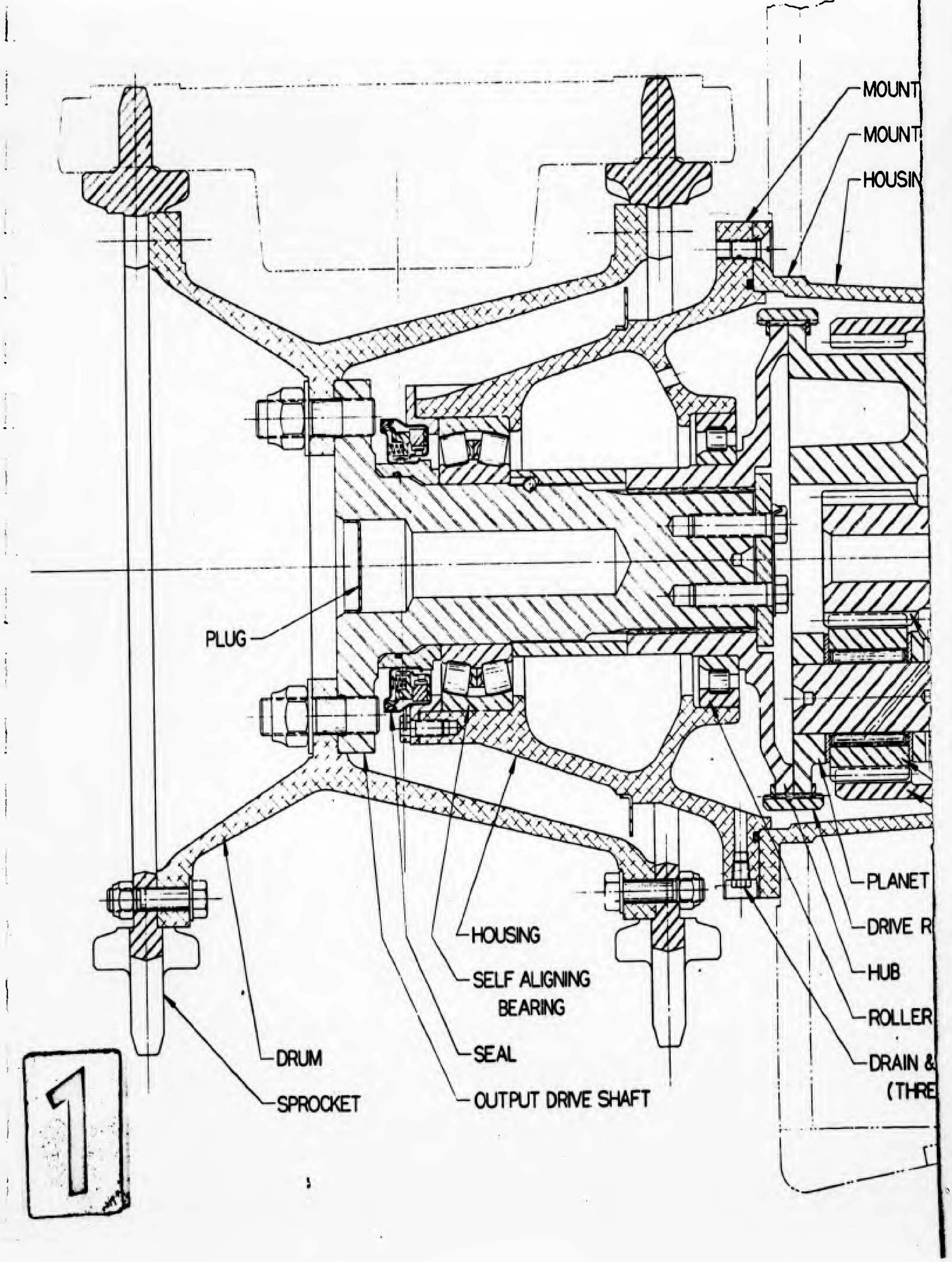
The current production Ordnance final drives reviewed were unable to transmit the necessary power and could not provide the required reduction ratio. One final drive, Ordnance No. 8351824, which best fits our installation was used as a design and guide in the preparation of the preliminary design of the final drive which Chrysler would propose to build for the LVTPX II. THIS MEANS NEW DESIGN FINAL DRIVE

The referenced Ordnance final drive has a reduction reversing planetary set for front sprocket drive vehicles. The sun pinion teeth are part of the drive shaft which is splined to the transmission output. The planet pinions are held stationary and power is taken off the ring gear which is driven in the opposite direction at a reduction of 5.35:1.

A 4.5:1 reduction ratio, non-reversing, was accomplished by driving the sun gear as before but holding the ring gear stationary and taking the drive power off the planet carrier. Standard gear design criteria was incorporated in determining gear teeth number and diameters to provide the proper ratio.

		No. Teeth	Diameter
N_s	Sun Gear	20	3.33"
N_l	Ring Gear	70	11.67"
N_p	Planet Gear	25	4.17"

With the new gear sizes and teeth number the planet set was designed as shown in Drawing No. TD-106526. Since the Ordnance version final drive is utilized with the same power range transmission as selected for the LVTPX II, shaft sizing and bearing loads were checked but need not be revised. It is proposed, however, that an aluminum housing be used to reduce the overall weight.



MOUNT
MOUNT
HOUSIN
PLUG
PLANET
DRIVE R
HOUSING
HUB
SELF ALIGNING
BEARING
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OUTPUT DRIVE SHAFT

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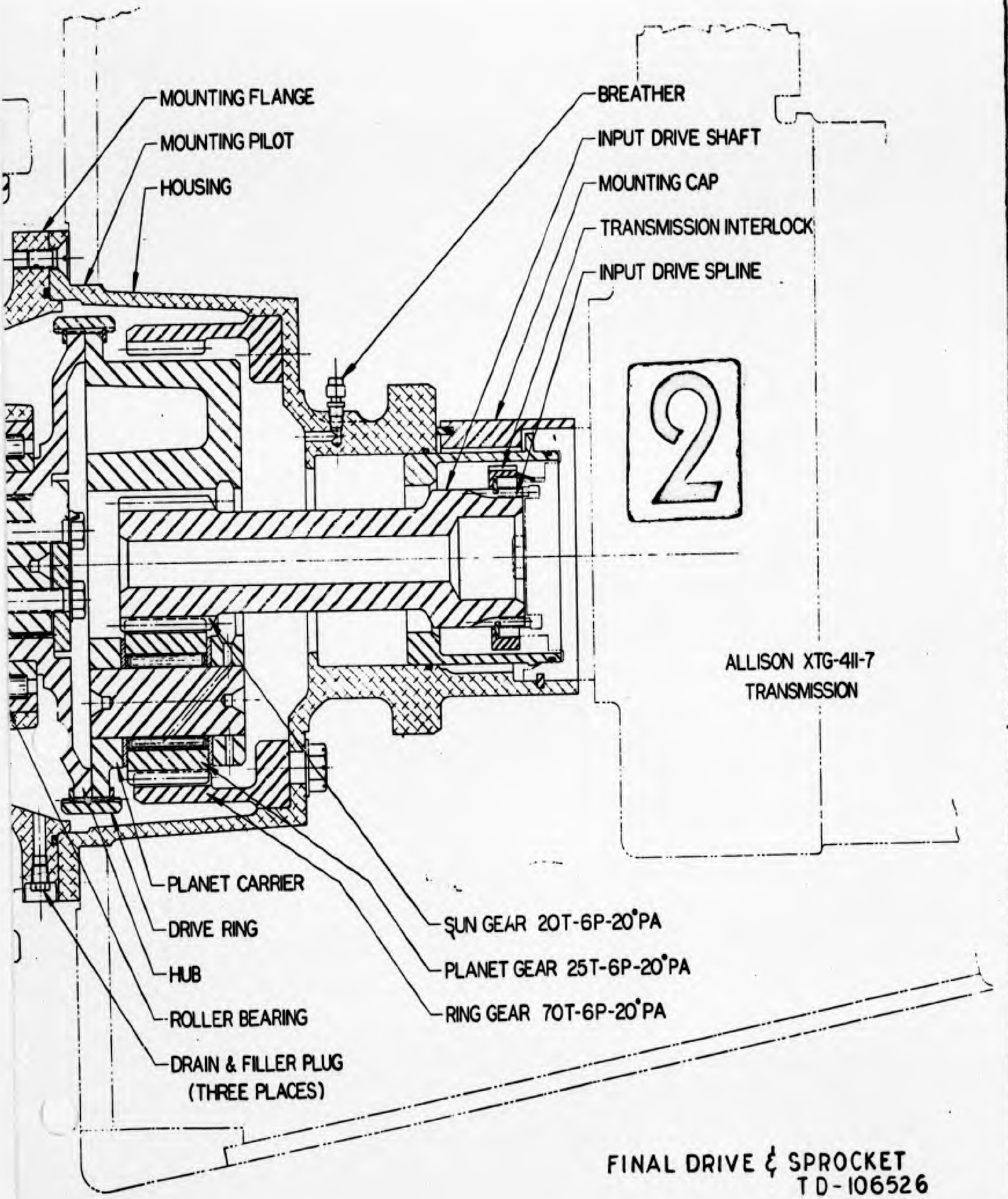


FINAL DRIVE & SPROCKET
TD-106526

4.3 POWER PACKAGE

4.3.1 REMOVAL AND INSTALLATION

The power package is removed and installed as an assembly made up of the engine, the cross drive transmission, and the engine accessories. The engine and transmission with final drives is supported at four points by two trunnion mounts, one under each final drive, and two engine mounts. The trunnion mounts are saddle type with bolt down caps. The engine mounts will be the shoe type, which are automatically engaged during the power package installation.

The engine compartment access cover is one piece and is bolted down at twelve points. The engine air intake tower and internal ventilation shrouding, as well as the exhaust tower and pipes, are an integral part of this access cover. The cover is removed by a lifting sling attached to the lifting eyes.

All connections to the power package have quick-disconnect type fittings, this includes fuel and fire extinguisher lines, electrical harnesses, air cleaner hoses, and exhaust tubes. The mechanical control rods are readily accessible and easily detached at the control arms and bell cranks.

4.3.2 LUBRICATION AND SERVICING

Chrysler's general design philosophy for this vehicle placed emphasis on ease of maintenance. One approach has been, first, to reduce the number of lubrication and service points to a minimum, and second, to make the lubrication and service points readily accessible. The selection of proven components also insures a reduction in servicing time required for components.

The engine compartment has been designed to position all service and lubrication points in one area. This area is located near the engine compartment access door. Upon entering the engine compartment through this door, the batteries, hydraulic reservoir, fire extinguisher, and electrical control box are located on the left for servicing. The engine accessories, engine and transmission dip sticks, and oil fill service points are located on the right. The dip stick and oil fill tubes have been clustered to allow these items to be checked and serviced from one location. The control rod adjustments may also be checked at the right rear. The hydraulic motors, alternator, fuel and oil filters are located on the accessory end of the engine.

The engine and transmission drains are connected to flexible hoses which are long enough to insert into the hull drain plug holes, to facilitate drainage.

4.4 TRACKS AND SUSPENSION

4.4.1 DESCRIPTION

The suspension system, (see TD 106527 following Page 4-33), is of the "torsilastic" spring-trailing arm type, with six dual bogie wheels per side. The drive sprocket is in the rear and the torsion bar spring-loaded compensating idler with a spring bump stop is in the front. Shock absorbers are provided for the front and rear suspension arms. There are three dual track support rollers on each side. The track is of the open-center type, similar to the Type VIII (as described in the Report, "Research, Investigation and Experimentation in the Field of Amphibian Vehicles"), but with one-piece forged aluminum track blocks. It is clevis-hinged, with center water grousers and guides. It has the rubber-in-torsion type of seals and Super Oilite "16" sintered metal bushings.

The suspension arrangement design has been based on the following considerations:

(1) A target ground pressure of 5 psi, for the full ground contact area of the track of a vehicle having 35,000 pound gross weight.

(2) A track width of 20-3/4 inches was selected, the same as the Type VIII track, because, together with the 5 psi ground pressure, the required length of track in contact with the ground (169 in.) was a reasonable length compared with the 21 ft. 6 in. over-all vehicle length. With a spacing of 151.5 inches between the forward and aft torsilastic suspension assembly mounting points and a ground penetration of 2-1/2 inches, the ground pressure requirement was satisfied.

(3) A 20-inch diameter road wheel, (see TD 106528 following Page 4-37), was selected, based on a study of similar vehicles which have been proven in field service.

(4) A 16-inch long trailing arm was selected. It is the shortest arm possible which still provides a minimum 2-inch clearance between the road wheels (in full jounce) and the torsilastic spring housings.

(5) Six pairs of road wheels per side were selected because six was the maximum number which allow a proper clearance between each road wheel and the next torsilastic spring housing when the wheels are in full jounce.

(6) A compensating idler wheel, (see TD 106529 following Page 4-42), 20 inches in diameter, was selected and positioned 18 inches forward of the front torsilastic spring mounting point and 30.4 inches above the ground for the static unloaded vehicle configuration. This is the minimum distance forward which permits no part of the vehicle (with the exception of the pintle hook) to extend forward and below the angle of approach of the track. The height was selected to permit the climbing of a 36-inch wall as determined by studying the capabilities of previous vehicles.

(7) The torsilastic springs, (see TD 106528 following Page 4-37), all but the forward one, mounted on the vehicle at a height such that the trailing arms make an angle of 47 degrees with the vertical for the static unloaded (25,000 pounds net weight) condition. In the static loaded (35,000 pounds gross weight) condition, the ground clearance to the hull at the vehicle centerline is 18 inches. This large angle to the vertical was selected in order to prevent the possibility of reversing the torsilastic springs when the vehicle travels in reverse over obstacles. The forward spring is mounted in somewhat higher position, with the trailing arm at 30-1/2 degrees to the vertical for the following reasons:

(a) With the torsilastic spring mounted in a higher position, the forward road wheels can travel upward farther before the tops of the water grousers on the track interfere with the forward torsilastic spring housing.

(b) The resultant force on the front road wheels, due to the track tension, has a shorter effective moment arm about the torsilastic spring, leaving more reserve spring torque to resist ground loads transmitted through the track. These ground loads have a smaller effective moment arm about the spring and, therefore, for a given reserve torque, a larger resistance to the ground loads results.

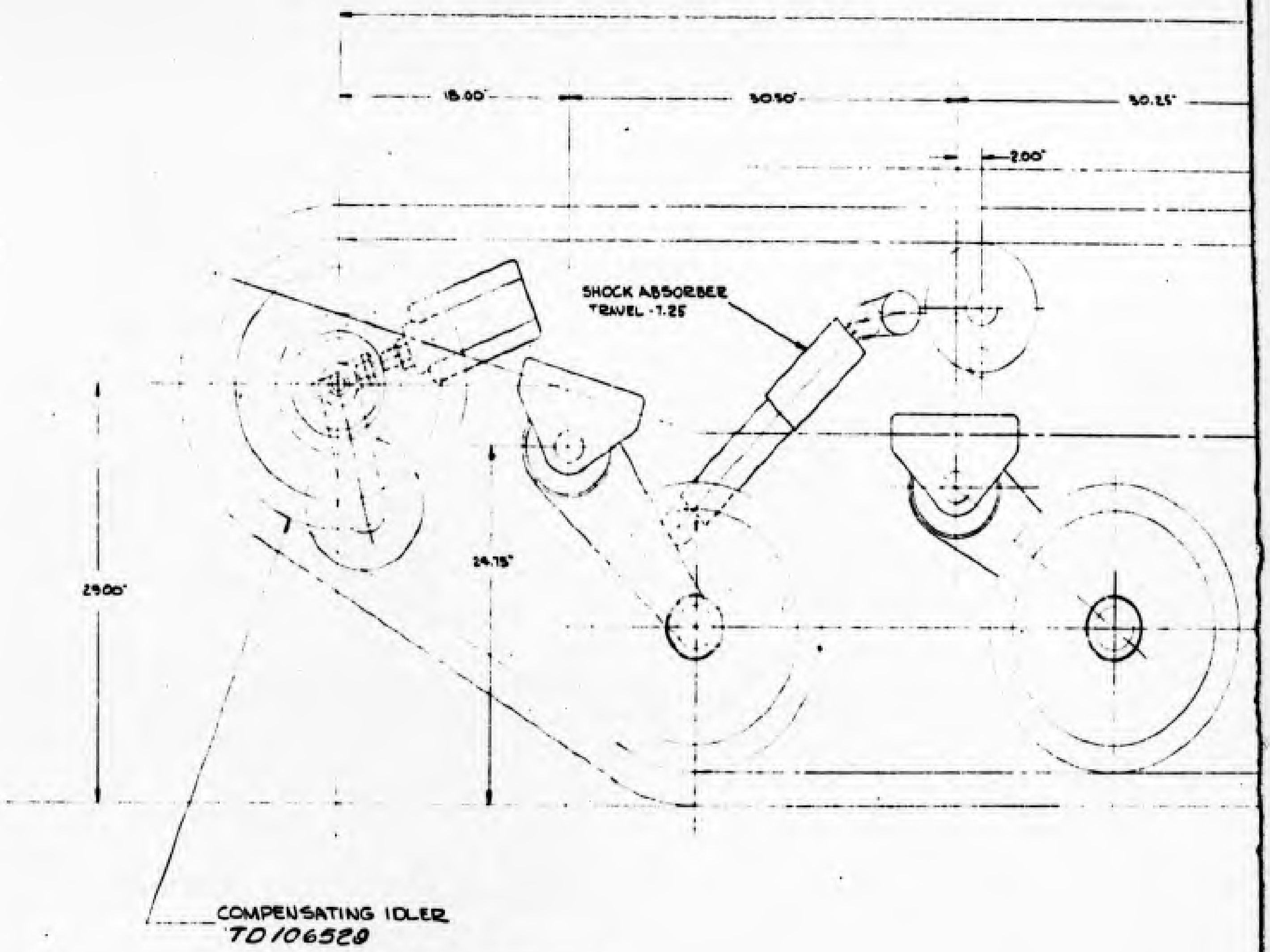
(c) There is less chance of the forward arms being reversed for a given angle to the vertical because of the moment about the torsilastic spring of the resultant load from the track tension.

(8) A drive sprocket, (see TD 106526), with 15 teeth was selected in order to provide a minimum of clearance between the sprocket and the final drive. The longitudinal position, 38.75 inches aft of the aft torsilastic spring mounting point, was, of course, determined by the position of the final drive. The vertical location was determined such that the top of the track is parallel to the ground and the bottom of the transmission did not interfere with the bottom of the hull for this final drive height.

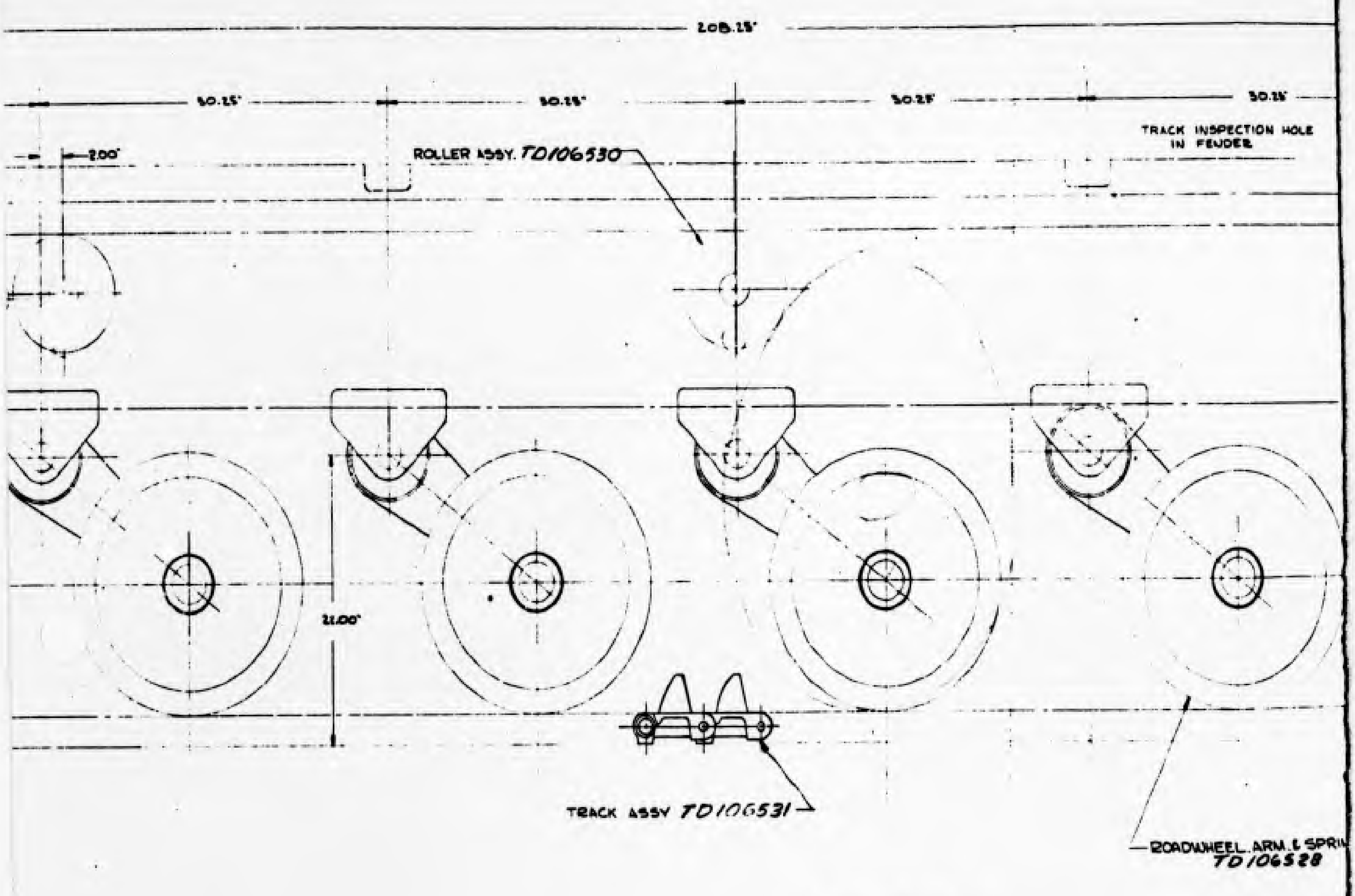
(9) Three pairs of track support rollers per side, (see TD 106530 following Page 4-43), were selected each having a diameter of 9 inches, which will permit clearance between the track water grousers and the track support roller axles. The positioning of these rollers was determined by the areas through which the road wheels travel.

(10) Shock absorbers were provided for the forward and aft trailing arms to prevent excessive pitching of the vehicle. They were positioned so that they are most effective in the normal static deflection range of the suspension arms, but for the larger vertical travels they rotate to a position where they become ineffective, thereby reducing the travel requirements of the shocks, compared to the large travel of the road wheels, and limiting the retracted length to a reasonable value. The locations of the shock absorber mounting beams were restricted by several factors, including areas swept through by the road wheels, the location of the track support rollers, the shock absorbers' potential interference with the road wheel axle, the track water grousers, and the sprocket.

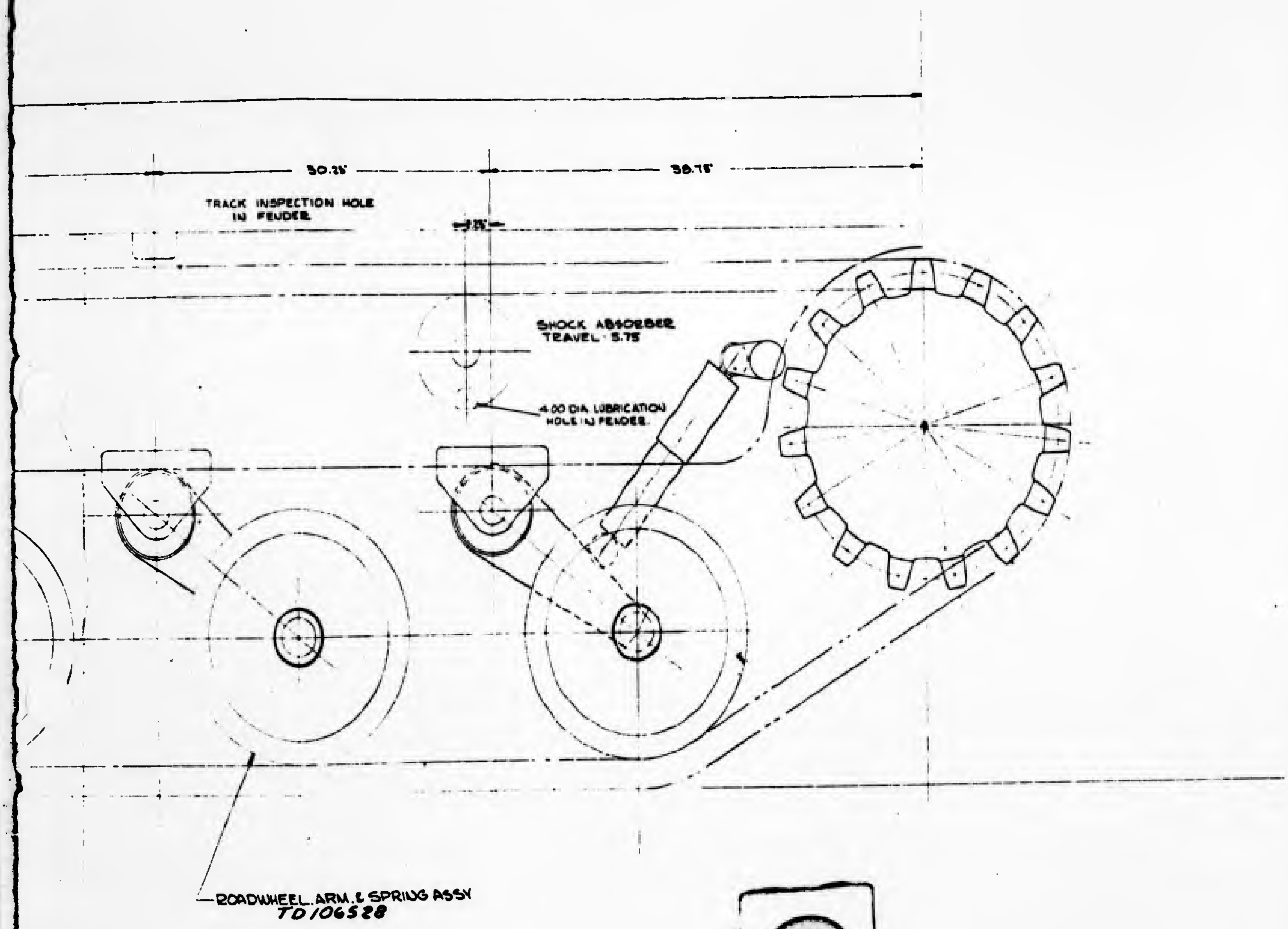
(11) The clearance between the track and the surrounding structure and the track skirt was kept to a minimum to give an optimum water propulsion efficiency.


18.00"
30.50"
30.25"
2.00"
SHOCK ABSORBER
TRAVEL - 7.25
29.00"
24.75"
COMPENSATING IDLER
TD106520

1


208.25"
30.25"
30.25"
30.25"
30.25"
2.00"
ROLLER ASSY. TD106530
TRACK INSPECTION HOLE IN FENDER
21.00"
TRACK ASSY TD106531
ROADWHEEL, ARM & SPRI
TD106528

2



3

SUSPENSION SYSTEM TD 106527

4.4.2 TORSILASTIC SUSPENSION ASSEMBLIES, (See TD-106528 following Page 4-37)

The choice of the torsilastic suspension in preference to a torsion bar spring type is based on several factors:

(1) It is located outside the hull, which has permitted placing the fuel cells in the area where the torsion bars would have been. This has conserved the area above the sponsons for personnel or cargo.

(2) Sealing of the hull is much simpler, requiring only sealing around the attaching bolts through the hull wall.

(3) No moving parts are used, eliminating wearing surfaces and the need for lubrication.

(4) Not sensitive to damage due to rough handling.

(5) Light weight.

(6) Simple construction involving no splines or bearings, make it less costly.

(7) A change in the spring rate requires only the substitution of rubber of a different durometer.

(8) Inasmuch as a track skirt is required to increase the propulsive efficiency of the track in the water, this provides an outboard support for the straddle mounting.

(9) The wheels are cushioned not only for vertical bumps, but also in sustaining impacts in any other direction, as in skidding sideways into a ditch.

The trailing arm assemblies are 16 inches long and are steel weldments. Into a tubular sleeve at the upper end, the torsilastic spring is pressed and keyed to the arm. The steel axle shaft is pressed into a tubular sleeve at the other end and is prevented from rotation by a dowel pin. Tapered roller bearings are used in the hubs which are secured to either end of the common axle with hardened steel tongue washers, castellated nuts and cotter pins. Knock out slots are provided in the hub to ease removal of bearing cups. A triple rubber seal with integral metal excluder ring seals the inner end of the hub, while a cast aluminum cap and gasket seals the outer end. The forged aluminum road wheels are 20 inches in diameter and are bolted on to the hub. Their inner diameter pilots onto a machined surface on the hubs to resist the radial loadings. Double clevis-type brackets are welded to the forward and aft arms for mounting the shock abosrbers.

The entire assemblies are attached to the vehicle as follows: On the inboard end, a steel plate which is welded to the inner tube of the torsilastic spring is bolted to the outside of the hull. The outer end of this tube has a steel plug welded to it which pilots into a forged aluminum bracket which is bolted to the bottom of the track skirt. A bolt, with a washer, in a blind tapped hole in this plug, prevents outboard deflection of the skirt.

The torsilastic spring consists of a cylinder ot 43 durometer rubber 19 inches long, with an inner diameter of 3 5/8 inches and an outer diameter of 7 1/16 inches. Figure 4-3 shows a curve of spring wind-up versus static torque for this spring. The rubber is bonded to an inner steel tube and to an outer split aluminum sleeve. The spring design is based on a stress of 75 psi on the inner diameter of the rubber for the static unloaded configuration of the vehicle (25,000 pounds).

The stress on the inner diameter of the rubber will be 300 psi for the front road wheel when it is deflected 13 1/2 inches up from the static unloaded position and for the other road wheels when they are deflected 15.2 inches up. However, since no jounce stops are provided, all of the road wheels can travel upwards at least 16 1/2 inches without mechanical interference, or more than 4 inches above the bottom of the hull. Since rubber can take on occasional loading well above 300 psi without deleterious effects, a usable wheel travel from static unloaded to full jounce of 16 1/2 inches is provided.

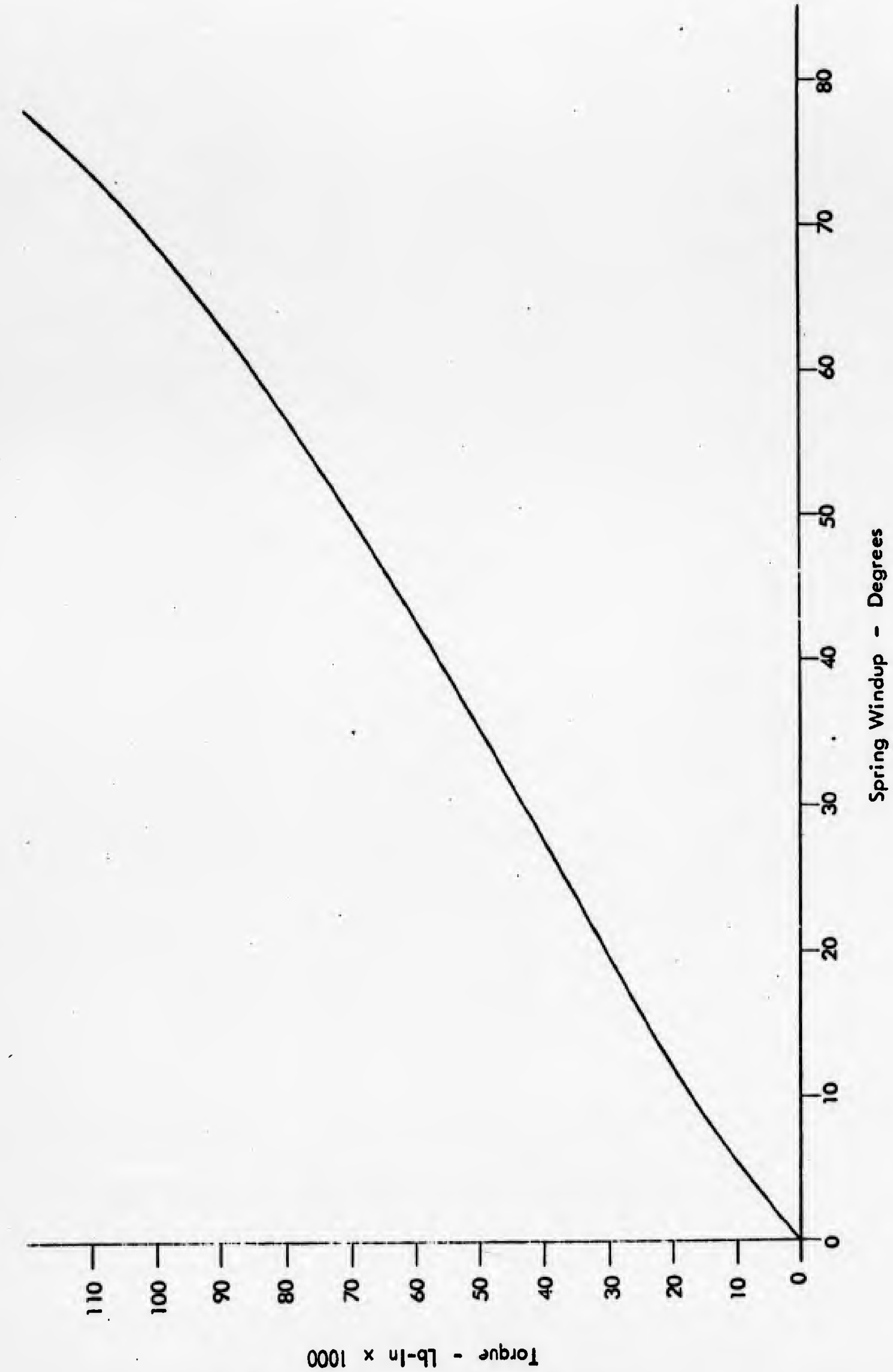


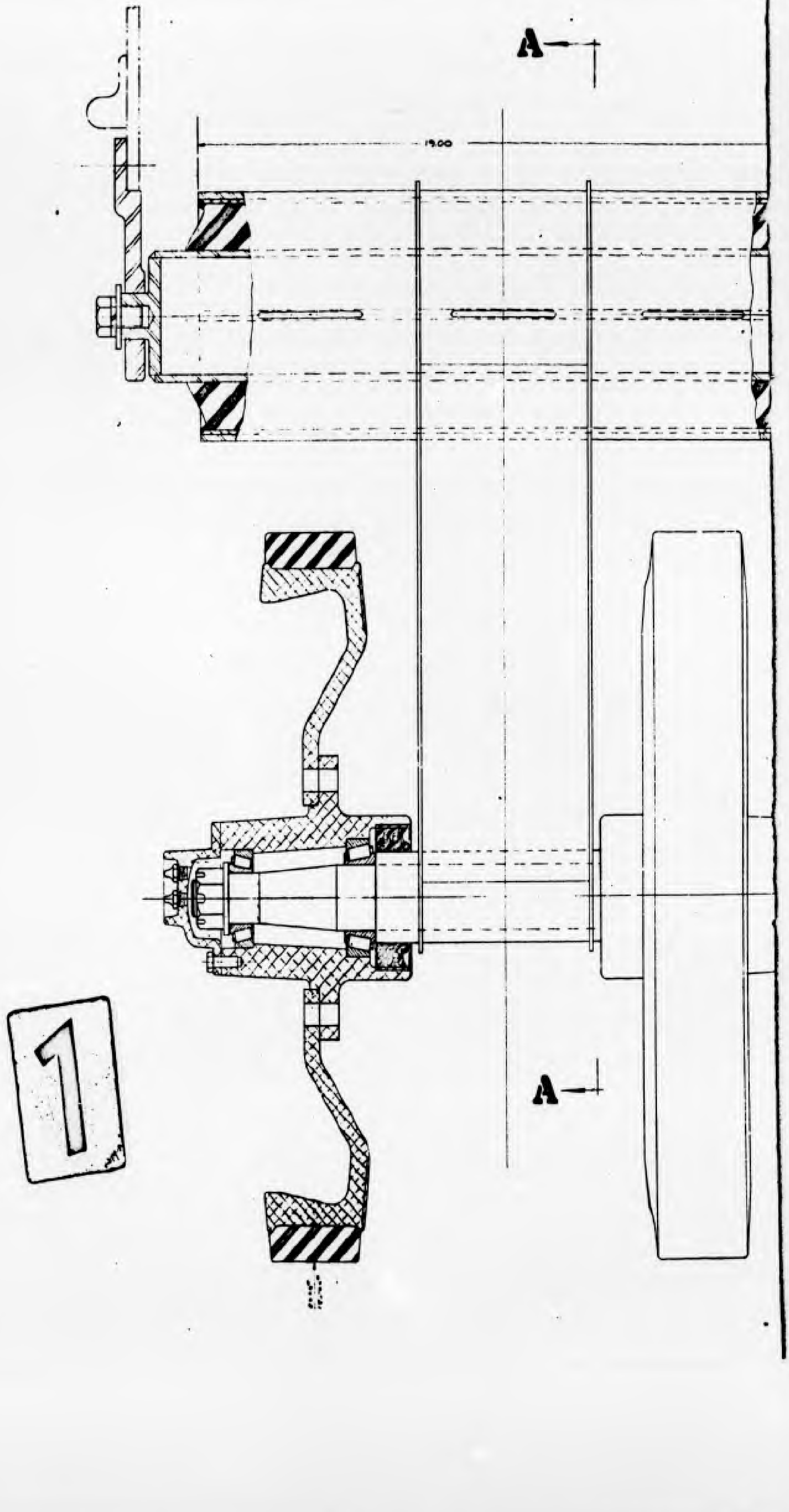
Figure 4-3 Spring Wind-up vs Static Torque

The unloaded positions of the arm assemblies in which they are initially installed are: 11 degrees aft of vertical for the front arm and 27 degrees aft of vertical for the others.

The road wheels are of forged 2014T6 aluminum, with 62 durometer natural rubber tires and a sprayed stainless steel wear surface where the wheel contacts the guides on the track. Small flanges are provided on the rim to mechanically retain the tire in addition to the bond between the rubber and the aluminum.

Before spraying with stainless steel, the surface is grooved and sandblasted to permit a mechanical interlocking between the sprayed metal and the aluminum.

Lubrication of both inboard and outboard wheel bearings is accomplished by means of two grease fittings located on the outboard cast aluminum hub cover. A hole through the center of the axle is the passageway for the lubricant from one of the grease fittings to the inboard hub.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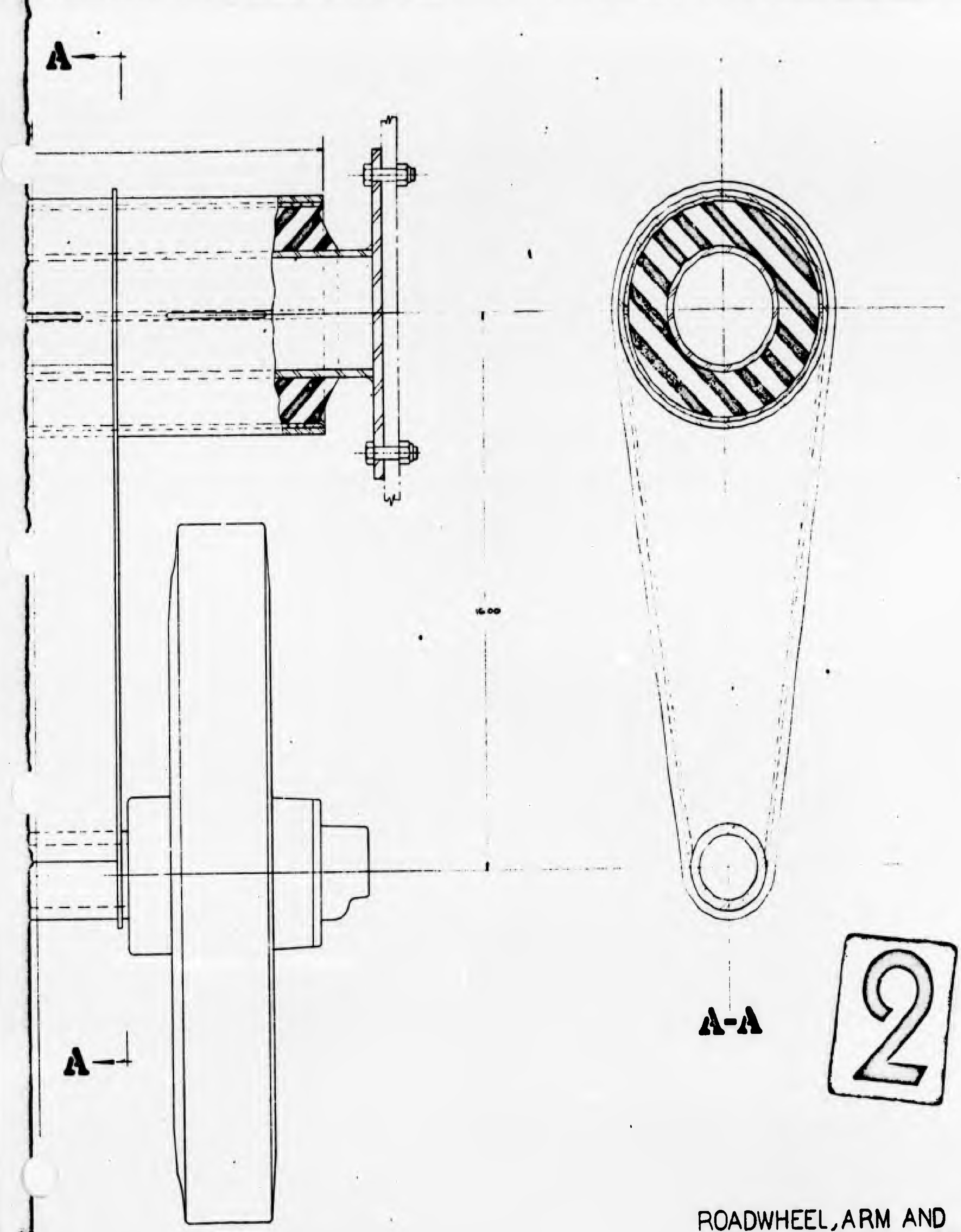


ROADWHEEL, ARM AND SPRING ASSEMBLY TD 10652

4.4.3 COMPENSATING IDLER (See TD-106529 following Page 4-42)

A spring loaded compensating idler was selected because of the success of this type on the LVTP-5 vehicles which have a similar suspension system.

The idler wheels are mounted on a common hub. A pair of tapered roller bearings together with a castellated nut, hardened tongue washer, and cotter pin retain the hub and wheel assembly on a spindle. Knockout slots are provided in the hub to ease removal of the cups. The spindle which is part of a forged steel cantilevered arm pivots on an 8 1/2 inch radius around the centerline of the torsion bar. This arm rides in sintered metal bushings inside the torsion bar anchor tube assembly and side mounting ring. The arm, torsion bar anchor tube assembly and side mounting ring are held together with a hardened washer, nut and lock nut on the end of the arm. A flanged, bolted joint in the torsion bar anchor tube gives access to these nuts. This assembly is supported at its inboard end near the vehicle center line in a sintered metal bearing pressed into an aluminum forging which also supports the front towing hitch. The outboard end, that is, the flange of the side mounting ring, pilots into a hole and is bolted to the hull. The torsion bar anchor tube is free to rotate with respect to both the arm and the side anchor ring. An anchor arm bolted to a flange on the outboard end of the torsion bar anchor tube, together with an eye bolt, lock nut and spherical seat washer provide a means of torquing up the torsion bar and adjusting the torque when necessary. Removing this anchor arm permits the removal of the entire compensating idler assembly by removing the bolts which attach the side mounting ring to the hull.

The torsion bar is inserted into the outboard end of this assembly through a tapped hole in the arm. A threaded plug is inserted with a spanner wrench to retain the torsion bar in position and together with a nylon insert prevents the entry of water. The torsion bar can be removed through the same hole.

The torsion bar is of alloy steel, shot peened and pre-set to give a beneficial residual stress pattern which permits the super-position of higher working stresses. The splines at both ends are the same, their pitch diameter being 1 1/2 times the bar diameter with a gradual change of diameter in the transitional area to reduce the stress concentration. Pre-setting the bars produces lefts and rights and care must be taken to see that the bars are installed on the proper side of the vehicle. Either end of the bars can, however, be installed first into the anchor tube. Adequate travel of the anchor arm is provided: to twist the torsion bar to its maximum torque; to allow a rotation of the anchor equal to the angular spacing between the spline teeth, due to the fact that the splines on both ends of the torsion bars have no specified angular orientation with respect to each other; and to allow for enough wear in the track pin joints up to the point when an entire 5 inch track block section would be replaced.

Lubrication of the tapered roller bearings in the hub is accomplished through a grease fitting on the cast aluminum hub cover. A triple rubber seal with integral metal excluder ring seals the inboard end of the hub.

The idler wheels are of similar construction to the road wheels. They are of forged 2014T6 aluminum, with 62 durometer rubber tires and a sprayed stainless steel wear surface where they contact the guides on the track. Small flanges are provided on the rim to mechanically retain the tire in addition to the bond between the rubber and the aluminum.

The torsion bar is designed to provide a maximum tension in the track of 4000 pounds. The tangential travel of the spindle from the unloaded position of the torsion bar to this maximum torque position for the bar is 4-1/2 inches.

A track tension of 3000–4000 pounds is adequate whenever the vehicle is being driven forward. However, when it is being backed up a slope or when it is being steered, much higher tensions occur in the top of one or both tracks. For these conditions a bump stop has been provided to prevent the torsion bars from being overstressed. This stop consists of a stack of steel Belleville spring washers, encased in a cast housing, with a piston type plunger, to transmit the load from the arm to the springs. The housing is filled with oil through a tapped hole in one end. As the plunger compresses the springs, the oil moves from one side of the piston to the other through an orifice. At a load of 24,000 pounds, the springs are compressed flat, giving a total travel of about 5/8ths of an inch.

An adjustable bump stop bolt with lock nut is screwed into a tapped hole in the arm. This bolt has a convex spherical head which contacts the concave spherical end on the plunger. These spherical surfaces provide a good load transfer surface throughout the full adjustment of the bolt. The bump stop bolt adjustment is provided to allow for wear of the track pin joints up to the point when an entire 5 inch track block would be replaced, and to permit some additional slack in the track when it is being installed. Shims are provided in the stock of Belleville spring washers to compensate for stock and manufacturing tolerances.

Track tension for a new unworn track is initially set when the vehicle is in the unloaded configuration on level ground by following these steps:

(1) Before attaching hub and wheel assembly onto spindle, install torsion bar into the torsion bar anchor tube just short of engaging the splines. A special tool which screws into the tapped hole in the end of the bar facilitates this operation.

(2) Screw bump stop bolt lock nut all the way up to the bolt head and position bolt in arm so that 1/2 inch of bolt is visible between arm and nut.

(3) Position the lock nut and its spherical seat washer on the torsion bar anchor eye-bolt with only 2/3 of the threads engaged. This eye-bolt protrudes through the inside floor of the vehicle next to the inside wall of the passenger compartment. If the nut and washer are not in contact with the top of the bracket which is at floor level force the eye-bolt down until this is the case.

(4) Engage the torsion bar splines at both ends of the bar if possible. If not, rotate lock nut on torsion bar anchor eye-bolt in the direction of engaging more threads until the torsion bar can be pushed into position with the splines engaged at both ends. Unscrew torsion bar installation tool and screw torsion bar retainer plug into position with a spanner wrench.

(5) Screw bump stop bolt all the way into arm.

(6) Install hub and wheel assembly on spindle.

(7) Install track.

(8) Screw bump stop bolt out 1/2 inch and lock into position with lock nut. The bump stop bolt head is, of course, in contact with the plunger because in this position, before the torsion bar is torqued up, the bump stop is providing the full track tension.

(9) Turn lock nut on torsion bar anchor eye-bolt until an additional 4 1/4 inches of threaded bolt protrudes above the floor.

(10) Check whether head of bump stop bolt is still in contact with bump stop plunger. If not, adjust bump stop bolt and lock nut so they are in contact.

After following this procedure, the track is properly tensioned. The sag in the track between track support rollers will be approximately 1/4 inch. This sag can be checked by stretching a chord between return rollers and holding it tangent to the top of the track at the rollers.

The amount of sag in the track can then be observed through the inspection hole in the track skirt halfway between rollers.

To check track tension after some wear has taken place, check amount of track sag between track support rollers with the vehicle on solid level terrain in the static unloaded condition. Also check clearance between bump stop bolt and bump stop plunger. If the track sag exceeds 1/2 inch follow procedure below. If it does not exceed 1/2 inch adjust bump stop bolt and lock nut so that bolt head contacts plunger.

(1) Turn lock nut on torsion bar anchor eye-bolt until sag of track between track support rollers is 1/4 inch.

(2) Adjust bump stop bolt and lock nut so that bolt head is in contact with bump stop plunger.

If, after adjusting track tension, about 3 inches of bump stop nut thread is exposed between the bolt head and its lock nut, a full track block must be removed from the track as follows:

(1) Turn lock nut on torsion bar anchor eye-bolt until lock nut is only engaged for 2/3 of its threads.

(2) Screw bump stop bolt all the way into the arm.

(3) Remove one track block from track.

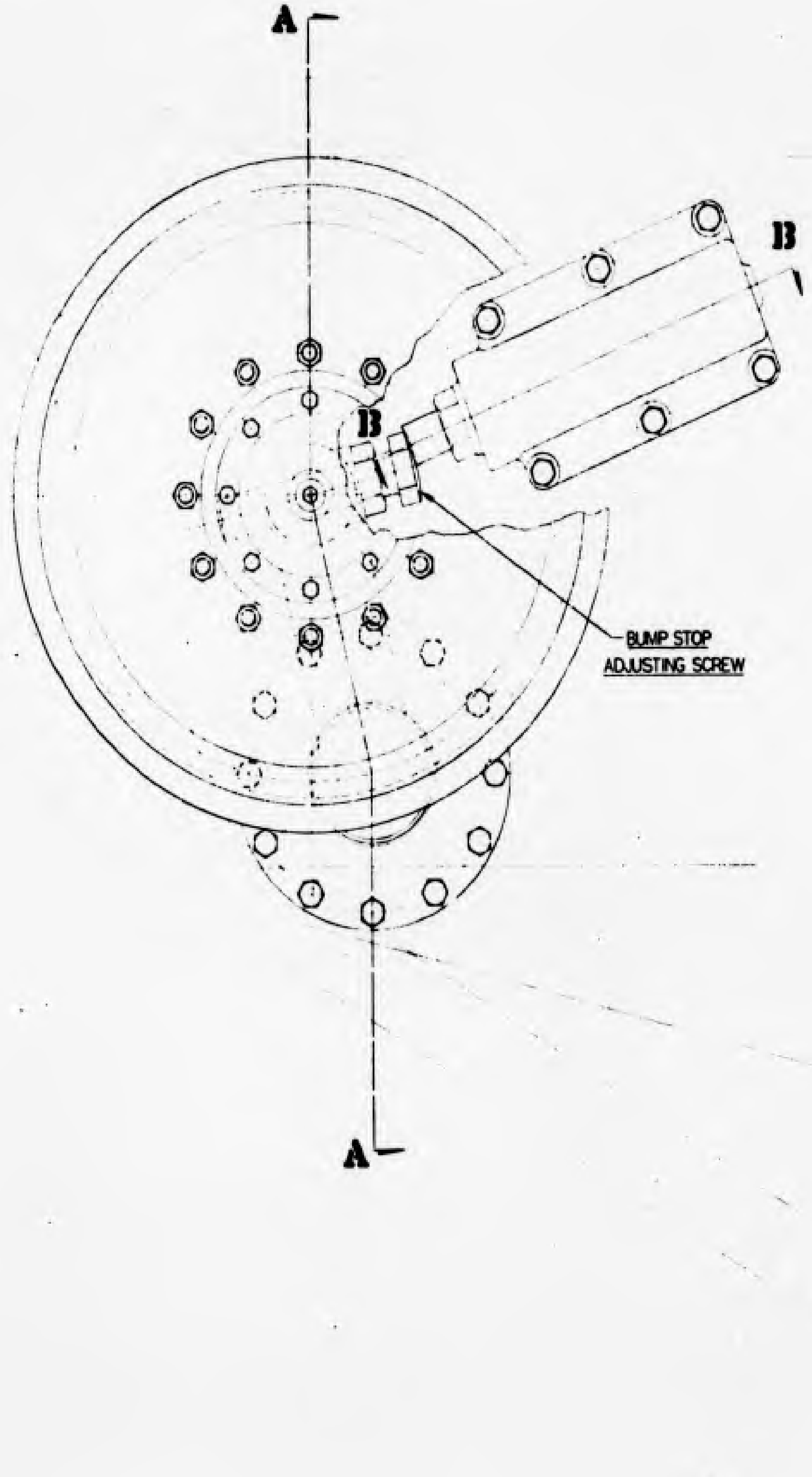
(4) Rotate idler wheels as far aft as possible.

(5) Install track.

(6) Screw bump stop bolt out 1/2 inch and lock into position with lock nut.

(7) Turn lock nut on torsion bar anchor eye-bolt until it just touches the top of its support bracket. Then turn until an additional 4 1/4 inches of threaded bolt protrudes above the floor.

(8) Check for proper track sag and whether head of bump stop bolt is in contact with bump stop plunger. Correct if necessary.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BUMP STOP
ADJUSTING SCREW
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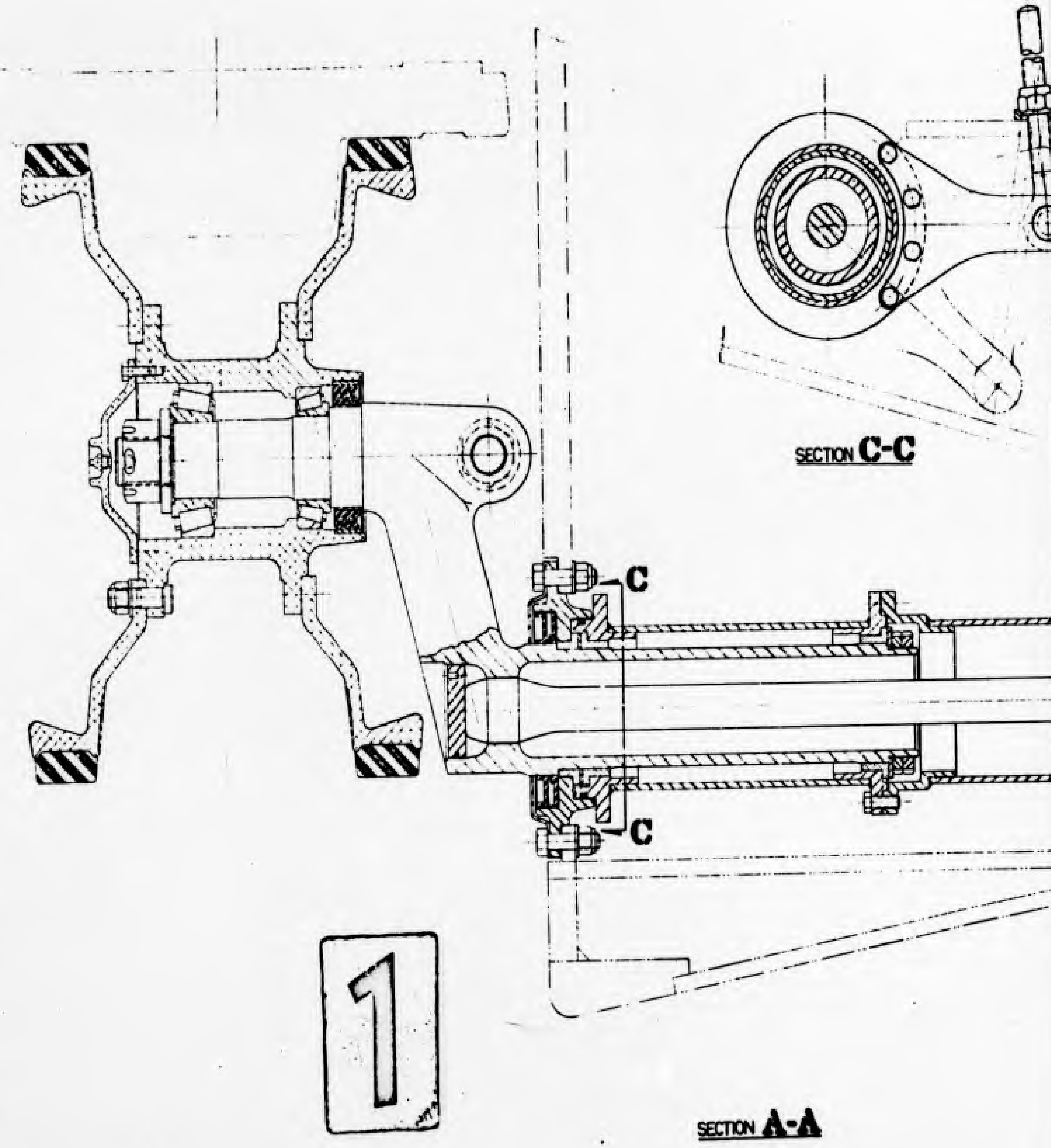
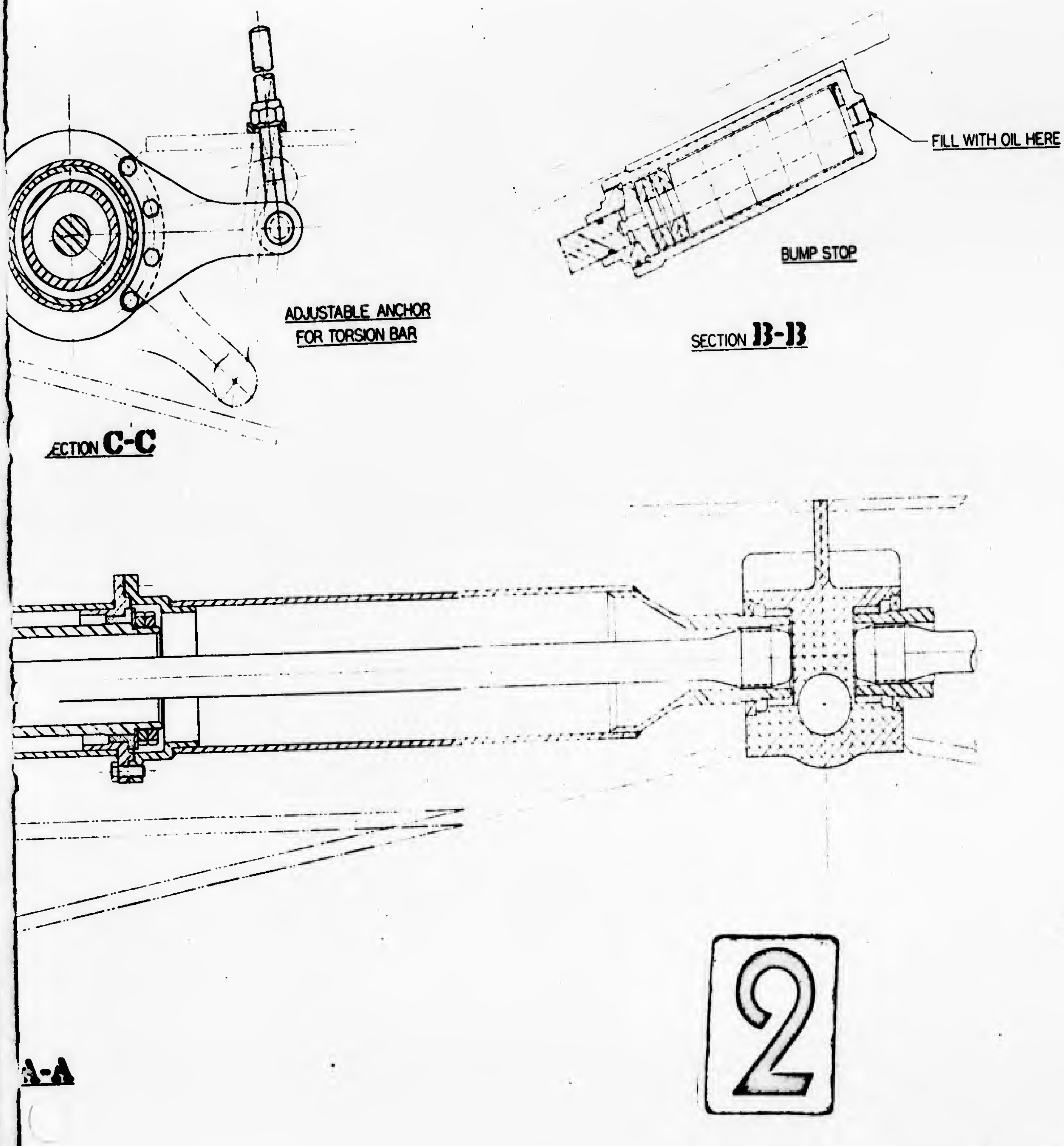


SECTION C-C

SECTION A-A



2

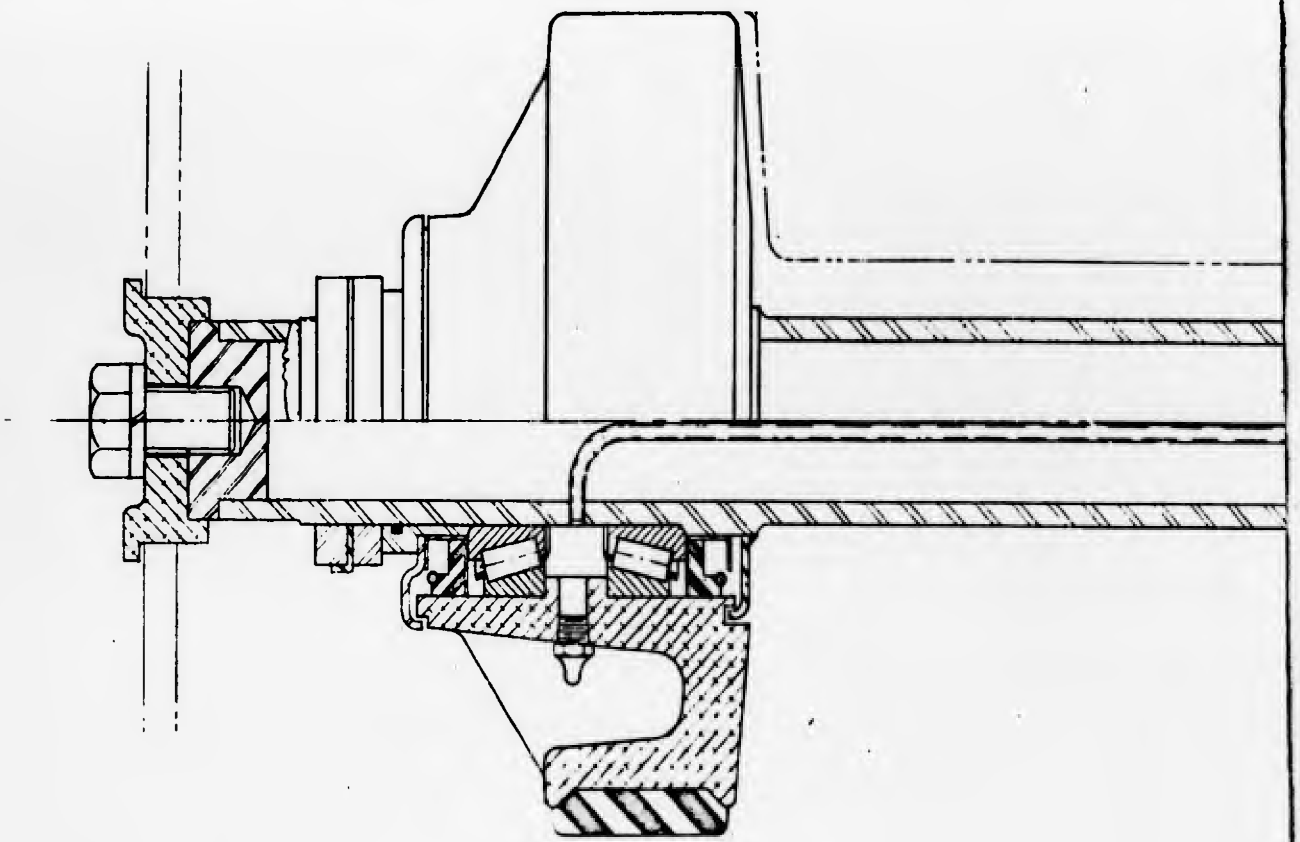
COMPENSATING IDLER TD 106529

4.4.4 TRACK SUPPORT ROLLERS (See TD-106530)

The track support roller wheels and hubs are one-piece aluminum castings with 62 durometer rubber tires. The wheels are mounted on their axles with a pair of tapered roller bearings which are retained by a spacer and a double nut and tang washer combination. Single rubber seals and metal excluder flanges are used on both sides of the wheels. A rubber "O" ring provides a seal between the axle and the spacer.

Lubrication of both pairs of wheel bearings is accomplished through the grease fittings on the outboard hub. An access hole is provided in the track skirt for this purpose. A tube inside the axle carries lubricant from the outboard to the inboard wheel bearings.

The inboard end of the steel axle is prevented from rotating and restrained from vertical or longitudinal movement by a steel casting which is bolted to the hull. The axle is retained laterally with a bolt through the hull wall into a blind tapped hole in its end. The outboard end of the axle pilots into an aluminum adapter which in turn pilots into a hole in the track skirt. This adapter is bolted to the axle and prevents the track skirt from moving outboard beyond the clearance provided between the outer flange on the adapter and the skirt.



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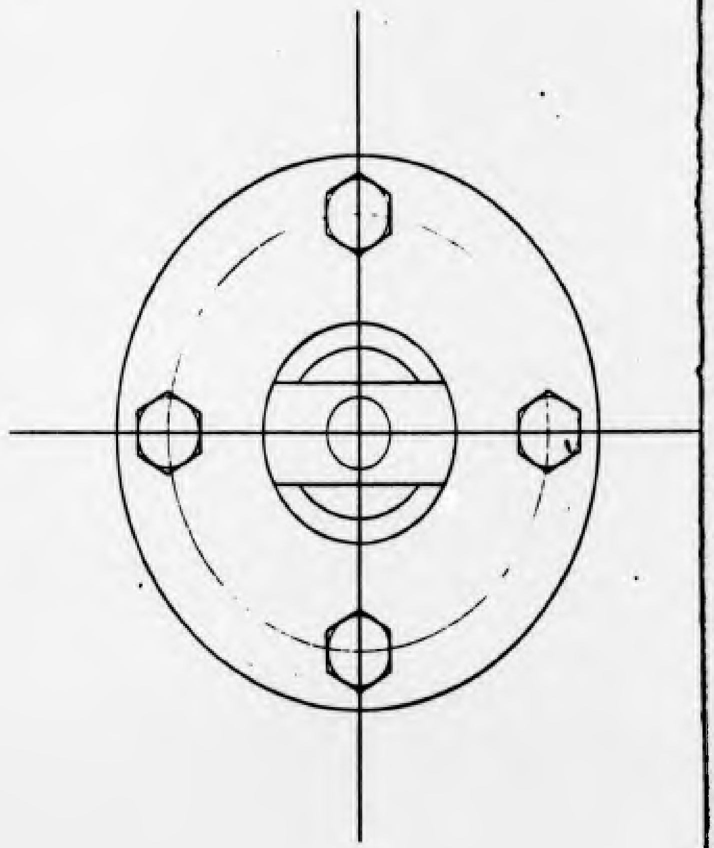

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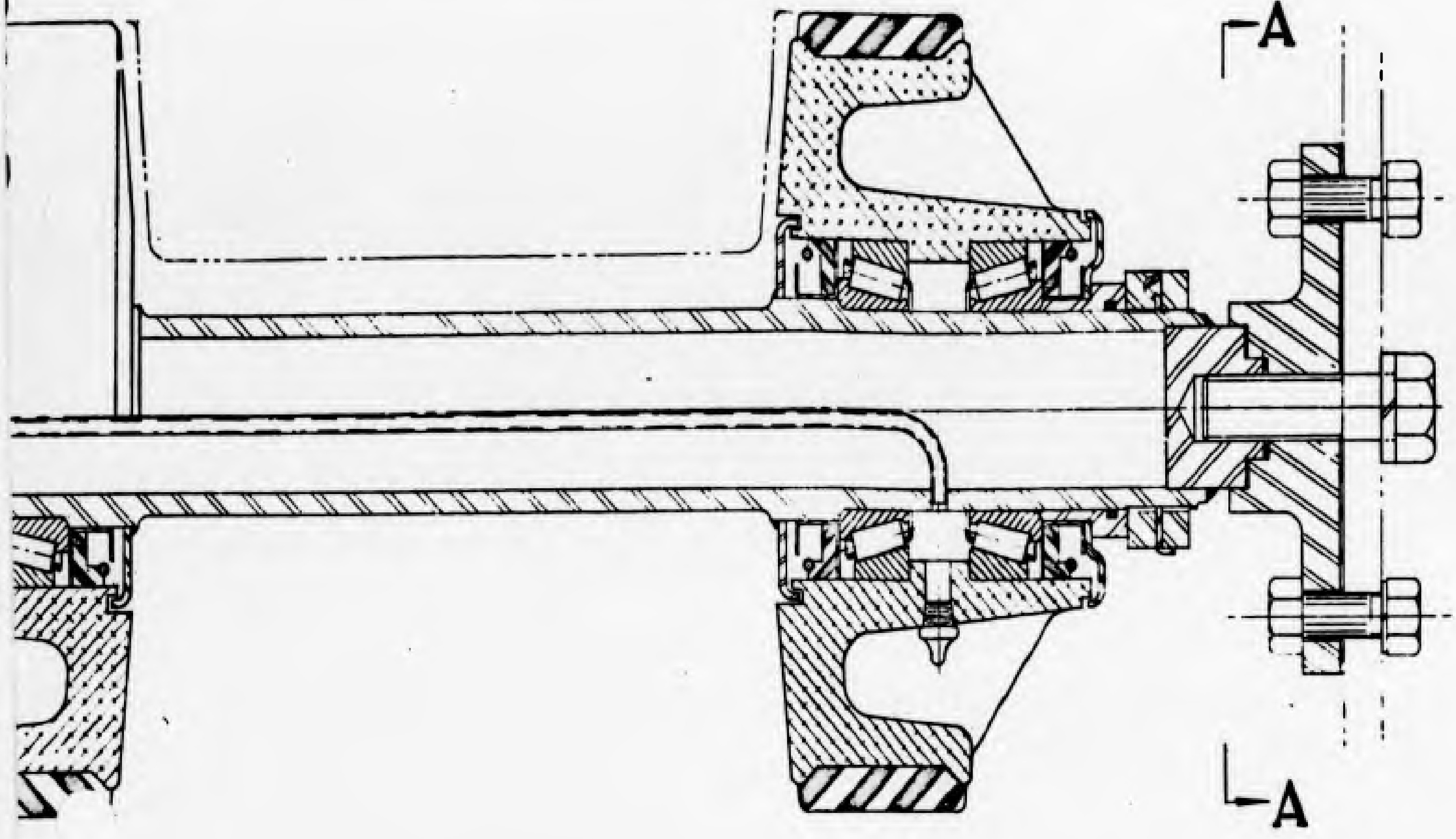
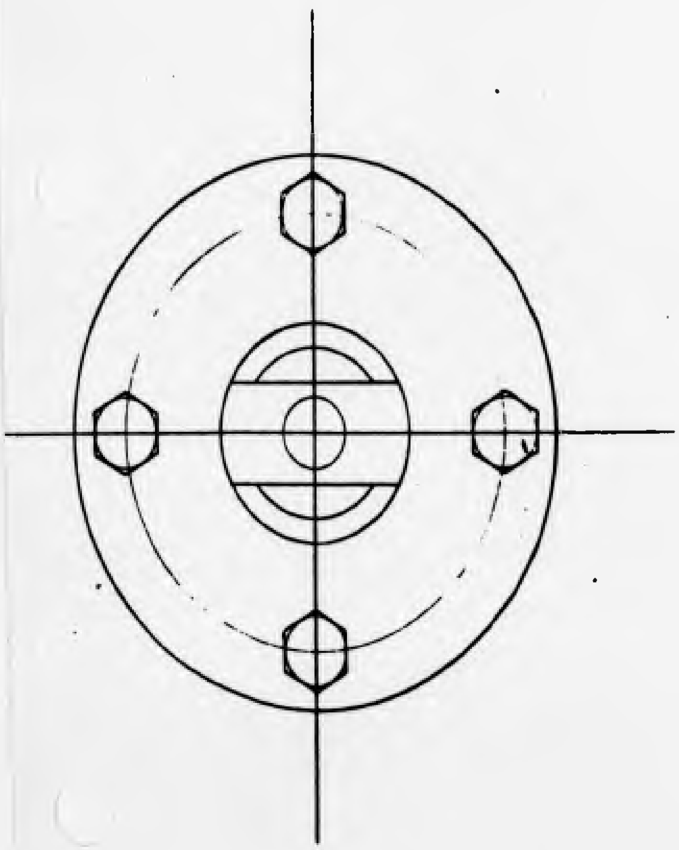




A-A

2

TRACK SUPPORT ROLLERS
TD-106530

4.4.5 TRACK (See TD 106531 following Page 4-47)

A review was made of all of the track blocks currently in service on military tracked vehicles in order to select the one best suited to the LVTPX-11.

Consider first the standard United States Army rubber-bushed type. This track will stand considerable abuse, especially on beach landings where sharp stones and volcanic ash are encountered. Its life and general performance are satisfactory, and rubber on the ground side, either in the form of a pad or molded into the shoe itself, gives a certain amount of protection for road operation. On the other hand, the weight characteristics, plus the poor performance of this type of track in water, hardly make it the desired type for this vehicle.

Next, consider the band type. From a standpoint of weight, this type of track enjoys a high place on the desirability list. This is also true regarding its quietness of operation, freedom from friction and power losses. However, this type of track also has some undesirable features, such as its inability to take abuse, its instability on side-slope operation, and its lack of known characteristics for water operation. Probably the principal shortcoming is its lateral instability. Band tracks, in general, have been used only on lightweight vehicles in the range of 10,000-20,000 pounds.

Another possibility would be the link-type or band-link, currently in the development stage. Insufficient information is available about the performance characteristics of this type to recommend it at this time.

Consider now the sintered metal bearing, sealed type VIII track presently used on the LVTP-5 vehicles. This type of track was first used on the prototype in 1950, and the articulated joint consisted of a hardened pin in a hardened hole with a lip-type seal. The seal proved to be ineffective, and the life of the track was only 40 to 50 hours. As improvements were made in the powdered metal bearing and the torsion seal was developed, the life increased to 250 to 300 hours. These figures are based on tests conducted at Camp Pendleton, California, where the average speed on test vehicles is 14 miles per hour. Using these figures, the average life of these tracks is in the range of 4,000 miles.

In comparison to other types of tracks, this type is relatively light, rugged, and has considerable lateral stability. Due to its lubricated joint, it has low rolling resistance, and hence the power losses are low. Because the pitch of the track is constant throughout its life, the wear on the sprocket is reduced and is almost double that of the rubber-bushed type. From a standpoint of track water propulsion, it is far better than any other type.

This type of track has at present a weak point, and this is the clevis. As the design of the articulated joint incorporates no resistance to transverse movement, there is a tendency to develop excessive wear in the clevis, which ultimately causes failure of the seal and in turn the bearing.

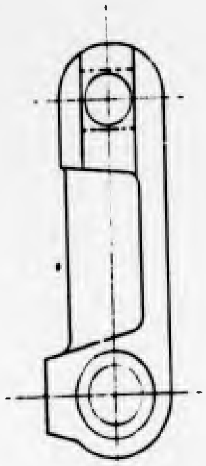
After reviewing the favorable and unfavorable characteristics of the various tracks, the type VIII Track was selected as being the most satisfactory for this vehicle.

Since the LVTPX-11 is a relatively light vehicle, the tracks will be subjected to much smaller loadings than they are on the LVTP-5. Therefore, a much lighter track design should be possible than the present 60 pounds per foot of the Type VIII Track. However, since the width will not be reduced because of ground pressure requirements, it would be impossible to reduce the weight of the present forged steel design without making the sections too thick for practical fabrication and ruggedness. For these reasons and because of the very restrictive weight requirements of this vehicle, a single-piece forged aluminum track block is recommended. Some of the sections have been made thicker, in particular the water grouser blade, which is 1/8 inch thicker on all surfaces. The areas of unusual wear, that is, the track guide and the external surface of the center clevis where the sprocket teeth engage the track blocks, must be overlaid with a hardened wear surface. The preferred method, as shown in TD 106531, is to spray these areas with metal, the track guide with stainless steel and the sprocket wear surface with molybdenum. In spraying the molybdenum, the track block would be rotated 360°, giving a hardened wear surface to the bottom of the cleat which comes in contact with the ground, as well as to the sprocket wear surface. An optional method is to rivet hardened steel overlays in both positions. The one to be used at the sprocket wear surface would be shaped like a horseshoe in cross-section, 1/8 inch thick, and would be sprung into position and then held there with a 1/4 inch double countersunk rivet.

A smaller hardened steel pin (7/8 inch diameter) is used because of the smaller loadings. The pin has a square head which fits into a milled groove in the outer clevis, preventing rotation of the pin with respect to the clevis. The pin is retained in position by a rubber-filled roll pin through the inner clevis which also prevents rotation of the pin with respect to this clevis.

The Super Oilite "16" sintered metal bushings and the rubber-in-torsion type seals have been used because of their success to date. However, a hardened steel sleeve with a flange is pressed into the center clevis with the seal in it. The flange protects the center clevis from wear due to transverse loadings and abrasive materials which work their way into this area. The flange extends down almost to the pin to act as an excluder and provide protection for the seal from abrasive materials.

The track weight has been reduced to 24.7 pounds per foot with this new aluminum design, which is less than one-half of the weight of the present Type VIII Track.



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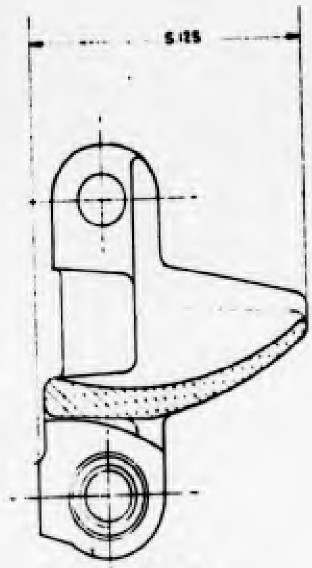

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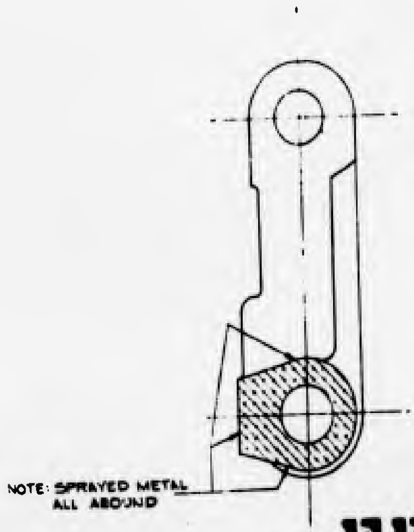

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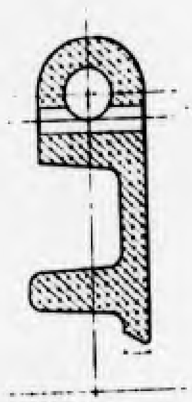

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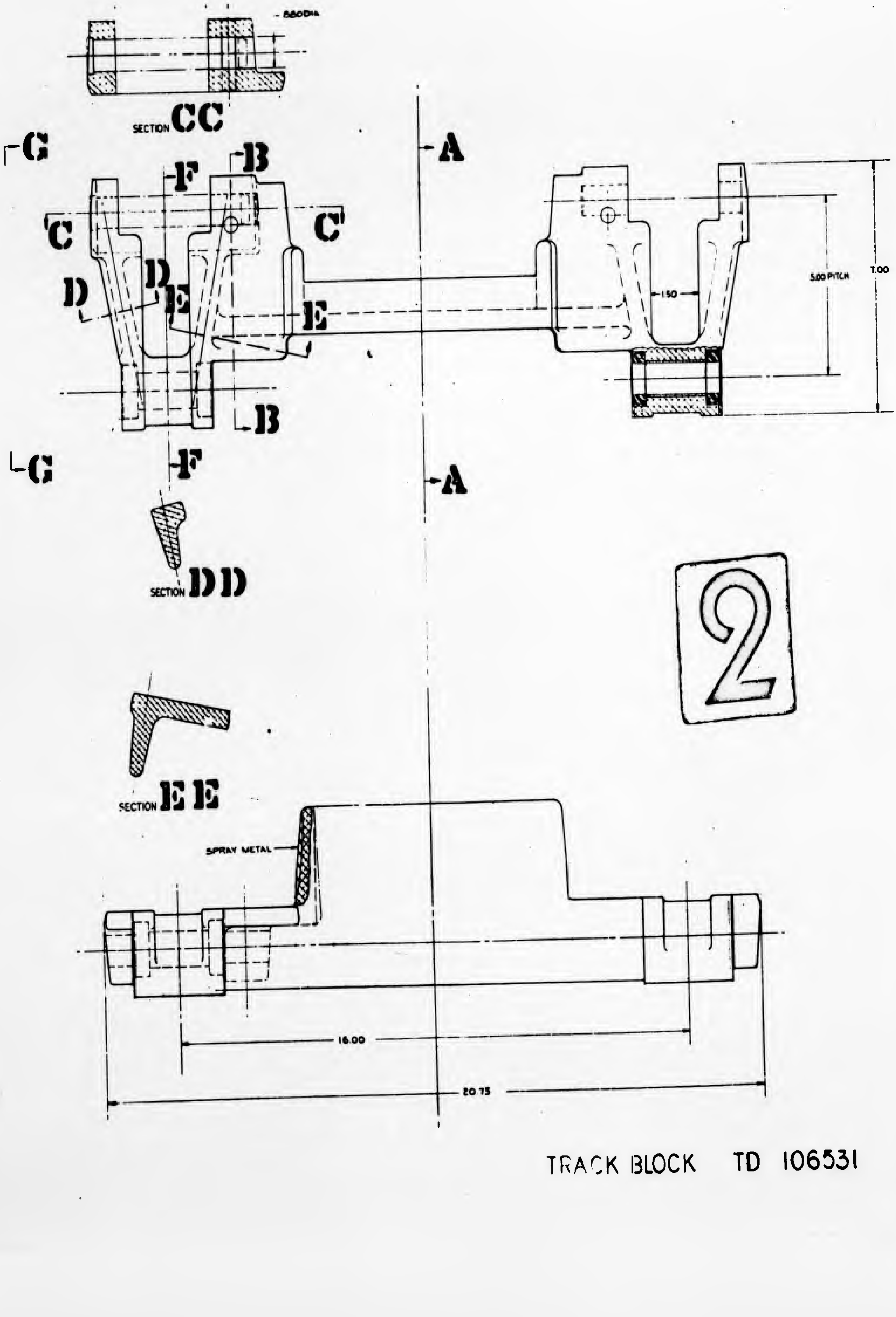


TRACK BLOCK TD 106531

4.4.6 SPROCKET (See TD-106526 following Page 4-28)

The sprocket selected for the LVTPX 11 has 15 teeth, with a pitch diameter of 23.85 inches. The sprocket diameter was reduced to lower the height of the return track, yet be compatible with a final drive design based upon the required torque and gear reduction. The tooth profile was developed by the track pin as it leaves the sprocket with a wiping motion on the tooth proper. The design is an overpitched arrangement that has the last tooth driving with a progressive clearance with the maximum clearance being at the point where the track enters the sprocket. This arrangement allows a gradual pickup of the track load by the sprocket teeth since the clearance diminishes as the sprocket rotates until contact between the track and sprocket is made close to their disengagement.

The sprocket is a forged steel ring which bolts to a forged aluminum drum. The wear surfaces of the teeth and ring forging are case hardened. This drum, in turn, bolts to the final drive shaft.

4.5 HULL

4.5.1 DESCRIPTION (TD 106532 following Page 4-55)

The hull is of all-welded aluminum construction. The external shell as well as all other sheet and plate are of 5456-H343 aluminum alloy. Where the thickness is 3/4 inch or more, it conforms to MIL-A-21170. All extrusions are 5456-H311 aluminum alloy conforming to MIL-A-21170. The welding is of the inert gas shielded metal arc type using a 5356 weld filler alloy.

The hull is composed of two major compartments separated by a watertight bulkhead, the personnel-cargo compartment forward and the engine compartment aft. A ramp at the bow provides access on land to the cargo compartment through an opening 68-1/4 inches wide and 58-3/4 inches high. Actually, however, somewhat more head room is provided because the top of the opening extends forward over the ramp. In the water, access to the cargo compartment is through a large hatch on the top which is 84 inches long and 56 inches wide and through the driver's and assistant driver's hatches. An escape hatch is located on each side of the vehicle at the rear end of thepersonnel-cargo compartment. Access to the engine compartment is through a hatch in the top of the vehicle above the starboard final drive or through a hatch in the bulkhead between the cargo and engine compartments. An access hatch is located above the air cleaners for servicing them. Another hatch is provided in the watertight bulkhead for access to stowage cabinet above the sponson on the starboard side of the vehicle. A removable panel permits removal of the engine and transmission assembly as a unit through an unrestricted opening 68 inches wide and 82 inches long. Six circular vision ports are located in the main bulkhead with fixed panes of safety glass to permit observation of the engine compartment from the crew area.

Sealing around holes cut into the hull to accommodate removable bolts, adapters, fittings, etc., is accomplished by rubber gaskets (where applicable) or a sealing compound equivalent to sealer "EC-801" as manufactured by the Minnesota Mining and Manufacturing Company which has been successfully applied by Chrysler to other amphibious vehicles which we have built for the United States Army.

An elevated section of the top of the vehicle which extends from one side to the other provides a pedestal for mounting the turret as well as headroom for the driver and crew chief who are located above the sponson. On the front and back surfaces of this structure are mounted sight blocks to provide fore and aft visibility.

The floor of the personnel-cargo compartment consists of four removable integrally stiffened aluminum panels. The rear two provide access to the fuel cells while the front two provide access to the bilge area. The floor panels are held down on their outboard edges by angle clips welded to the hull sides and on their inboard edges by dog-type latches which engage the flange of the longitudinal tee section stringer on the vehicle centerline. These latches are operated with a screwdriver.

Holes are provided in the floor panels for the attaching of cargo hold-down devices.

Towing eyes are located at the bow and stern and four hoisting eyes are provided on the top of the vehicle. A towing cable attached to the towing hitch is stowed outside on the vehicle's stern. Mooring bitts are located on the hull top in the four corners.

A track skirt on the outboard side of the sponson extends downwards 16-1/2 inches on the outside of the track.

Rubber head protection pads are provided on the stiffeners of the main cargo hatch and on the overhead beam of the bow portal structure.

Anti-skid material is provided on the top deck in all areas where personnel will be required to walk. Stowable benches are provided for the transportation of personnel.

4.5.2 STRUCTURAL DESIGN

5456-H343 aluminum armor plate has been selected for the external hull structure for the following reasons:

(1) Although equal weights of aluminum and steel armor plate provide roughly equivalent ballistic protection, the aluminum plate, which is approximately three times as thick (if equal armor protection is provided), has more bending strength and much more rigidity. Consequently, the aluminum structure requires fewer supporting members and framing.

(2) The welding properties of aluminum are well known and the processes involved in welding armor plate are firmly established.

(3) Aluminum armor plate has several current applications where ballistic protection from small arms fire is required.

(4) The properties of magnesium for ballistic protection are not as well established.

(5) The cost of titanium, as well as the relative lack of experience in its use for ballistic protection, argues against its use.

The hull is primarily a shell or monocoque-type structure deriving most of its strength from its external skin. The skin thickness are:

Bottom	1/2 inch
Top	3/8 inch
Front	1-1/8 inch
Sides (above sponson)	5/8 inch
Sides (below sponson)	1/2 inch
Track skirts	3/8 inch
Sponsons	1/2 inch
Rear	3/8 inch

Prior to fabrication, self-positioning (self-jigging) features would be worked into joint designs wherever feasible to reduce the cost of fabrication.

The hull has one major transverse bulkhead, between the personnel-cargo and engine compartments, and a partial frame between the turret and the cargo hatch. The bulkhead consists of 3/16 inch 5456-H343 sheet with one main lateral zee (stiffener) on the forward side of the skin running from the top of one sponson to the top of the other and five secondary vertical channel stiffeners on the aft side of the skin. The bulkhead is attached to the external hull skin by means of angles on the top, the sides above the sponson, and the bottom. It is attached with the transverse zee (with one leg removed) over the sponsons and with the vertical channels on the sides below the sponsons. These members also provide support for the outer shell. The stowage cabinet and engine compartment access hatch openings are framed on the forward side with angles which provide a seating for the hatch seals. A lateral angle on the forward side supports the floor panels. The partial frame consists of a transverse beam on the top running from one side of the hull to the other with vertical tee-section struts providing intermediate supports just inside the sponsons and transferring loads from the top of the vehicle to the hull side below the sponsons. The transverse beam consists primarily of the rear surface of the raised section of the hull top but with the section between the longitudinal stringers enclosed into a box section to resist torsional loadings.

Longitudinal tee-section beam-stringers run the entire length of the top of the vehicle transferring normal loads on the top of the hull fore and aft to the front portal structure, the partial frame, the main bulkhead and the rear vertical surface of the hull. Another longitudinal stringer is located on the vehicle centerline running forward from the main bulkhead to the transverse box beam which forms the threshold of the front ramp opening. Loads from the cargo floor and from the bottom skin of the hull are carried fore and aft to the main bulkhead, the threshold beam and an intermediate transverse bulkhead below the floor which also forms the forward end of the fuel compartment. Heavy longitudinal extrusions are located at the lowest points of the hull to prevent the bottom from wearing through in these areas.

The front ramp opening is framed with the threshold box beam on the bottom as well as box beams on the top and sides, with welded corner joints to form a strong, continuous portal structure partially compensating for the large door opening. The door, when closed, will, however, prevent excessive parallelogramming of this portal structure under extreme loadings. The threshold box beam also provides support for the aluminum forging which, in turn, supports the front pintle hook and the inboard ends of the compensating idler assemblies.

The hull wall below the sponson and the track skirt provides support for all of the suspension components. A skid rail extrusion on the outboard side of the skirts near its bottom strengthens it against accidental damage and also protects the bolt heads in the torsilastic spring support brackets. The hull sides in the areas of the compensating idlers and the final drive are locally thicker to support the high local loadings.

The critical design loadings which were considered in the hull structural design are:

(1) 1000 pounds per square foot water pressure on the top of the hull.

(2) 500 pounds per square foot water pressure on the bow, stern, sides and bottom.

(3) 490 pounds per square foot inertial loading downward, due to a vertical load factor of 4 on the cargo, acting on the cargo floor.

(4) 5 pounds per square inch upward from pressure testing of fuel cells acting on the cargo floor above them.

(5) A moving wheel load of 750 pounds on the cargo floor from the Mighty Mite when it is being driven in and an inertial load of 3000 pounds when it is in position.

(6) 644 pounds per square foot water pressure on the main bulkhead, which is sufficient to float the vehicle if the engine compartment is completely flooded.

(7) 52,500 pounds on the compensating idler (1.5 times vehicle weight).

(8) 10,000 pounds per pair of roadwheels.

(9) 35,000 pounds on the sprocket.

(10) 15,000 pounds on the shock absorber support beams.

(11) 35,000 pounds on the towing hitches.

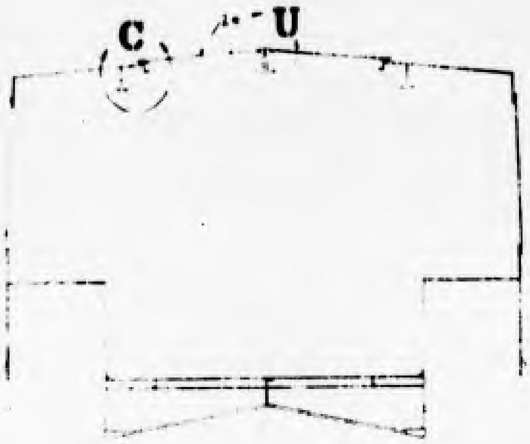
(12) 35,000 pounds at any point on the lowest part of the hull bottom which occurs when the bottom of hull hits obstacles.

(13) 35,000 pounds on each hoisting eye.

(14) 10,000 pounds on each mooring post.

The design stress for the aluminum components ranges from 25,000 psi to 30,000 psi, depending on the anticipated frequency of the loading, the reserve strength of the component after reaching the yield point, and the importance of the component with respect to vehicle survival.

C



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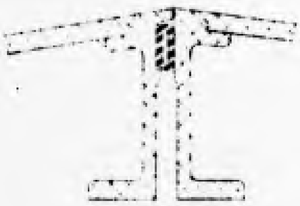

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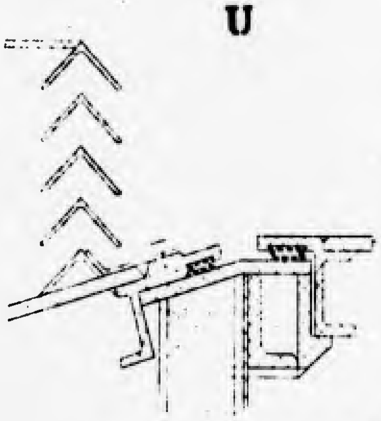

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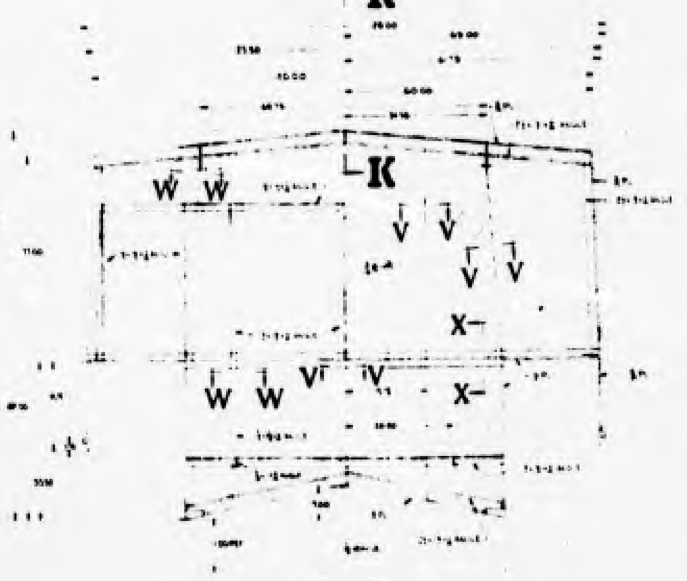

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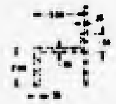

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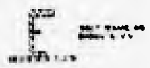

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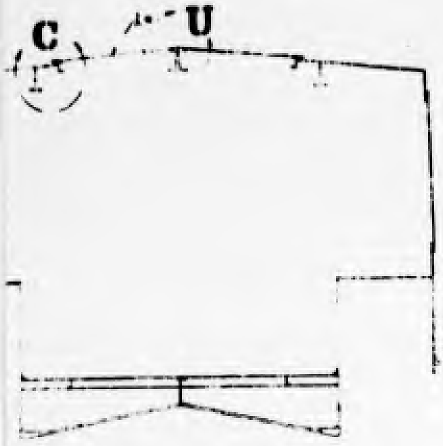

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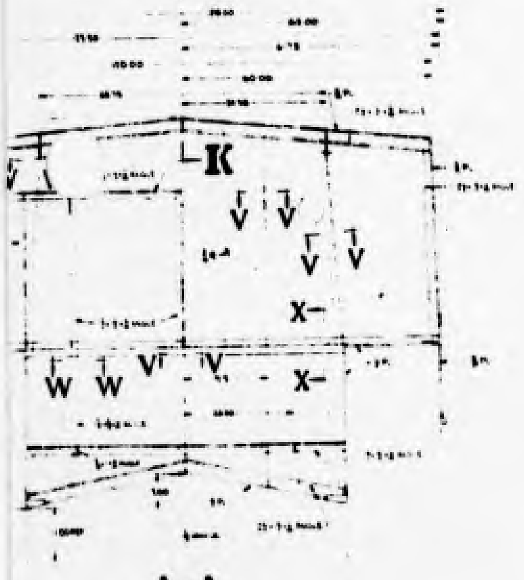

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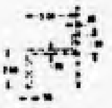

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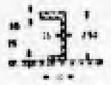

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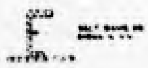

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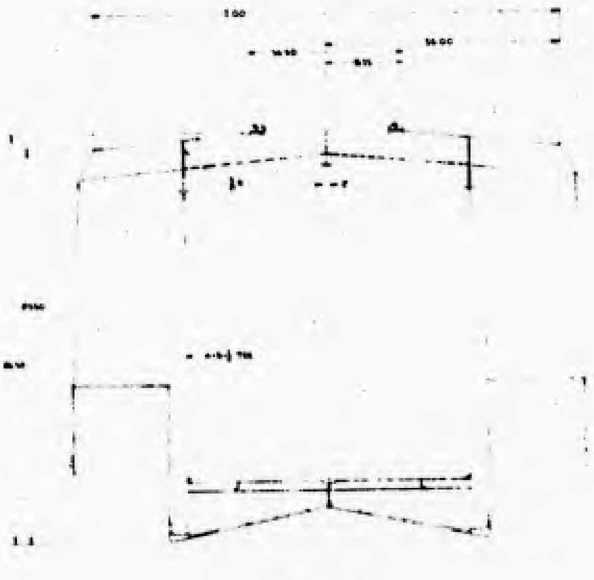

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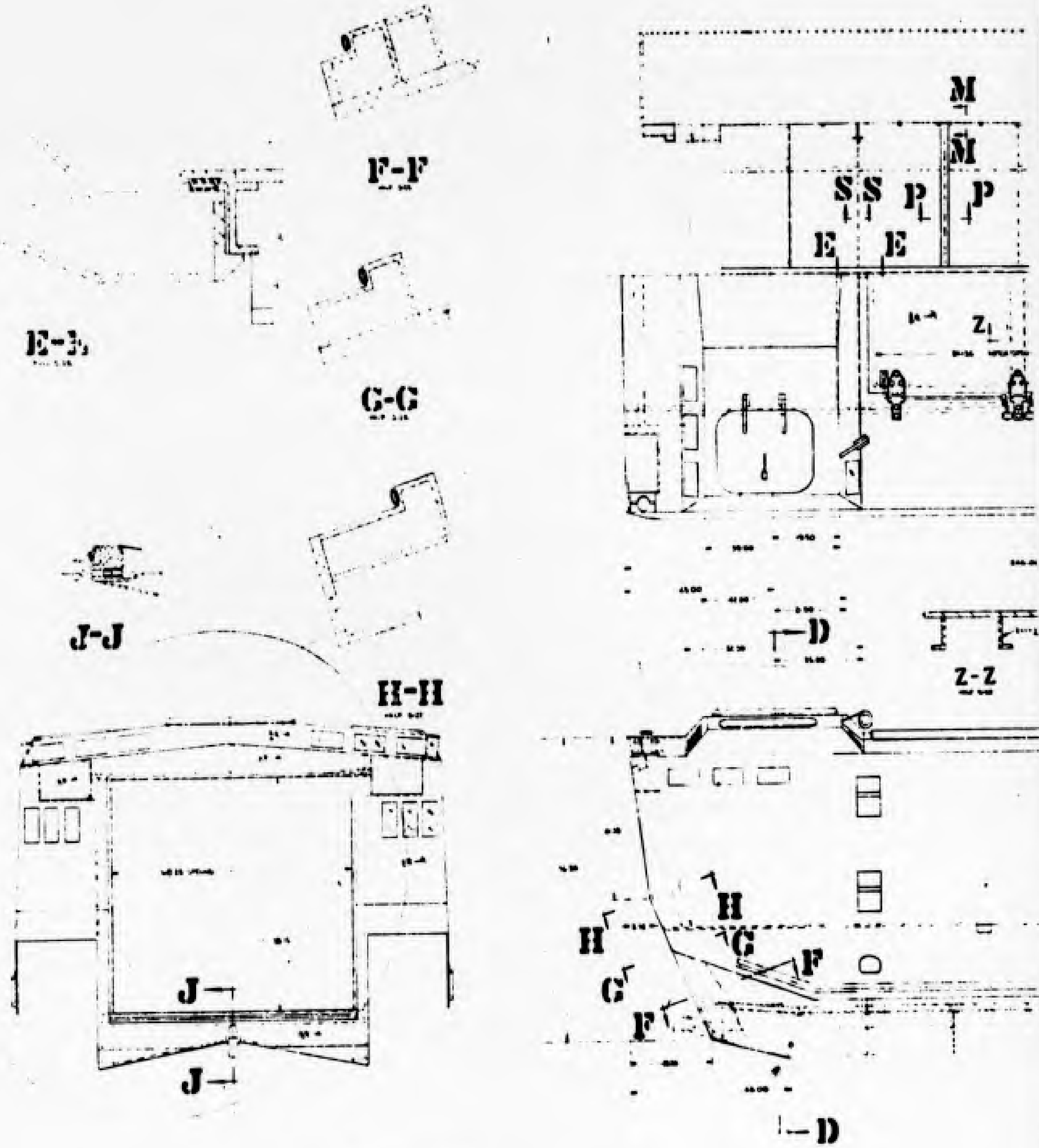

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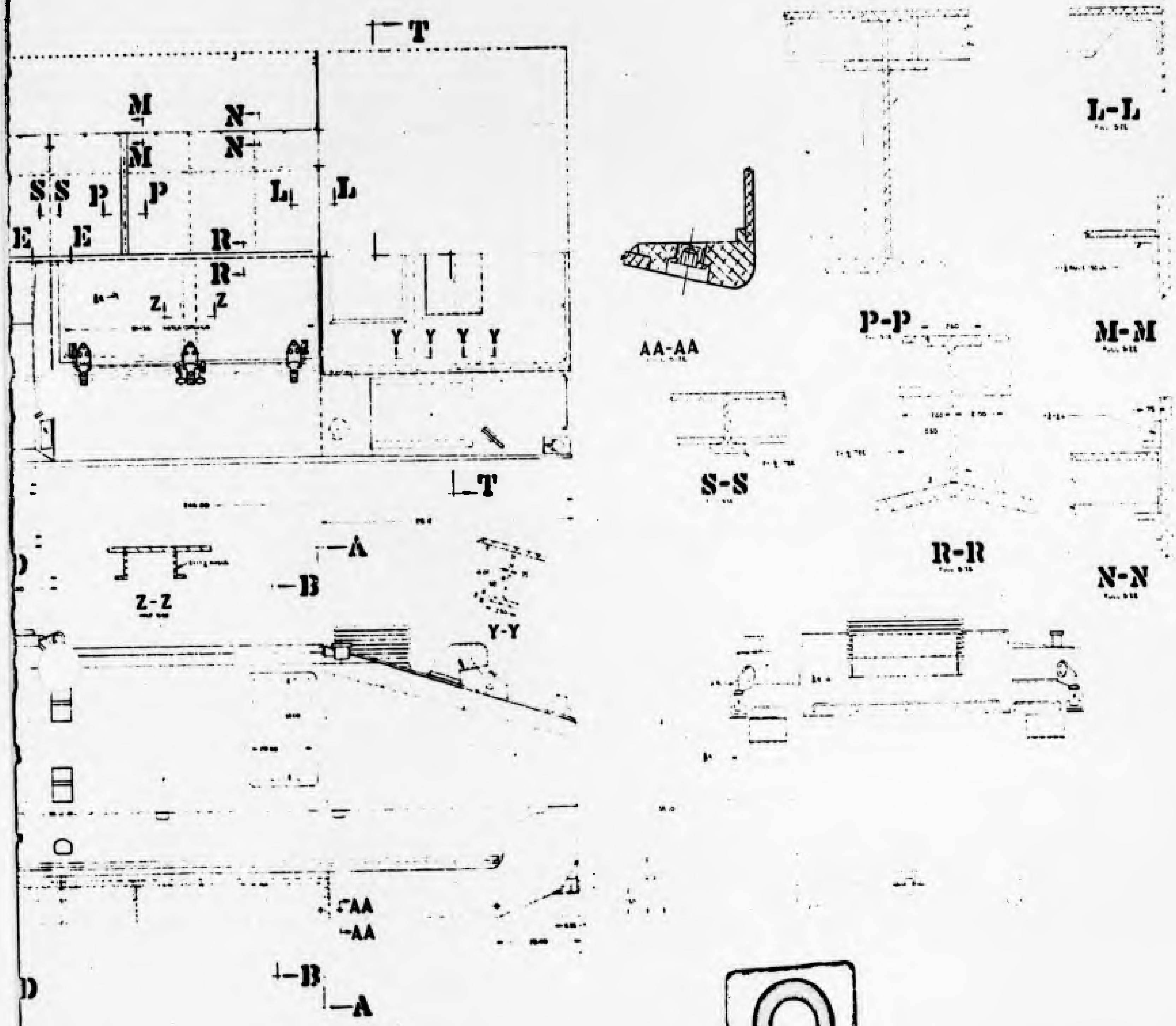

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HULL STRUCTURE TD 106532

4.5.3 RAMP (TD 106533)

The ramp is an all welded aluminum box-type structure 62-5/8 inches by 70-1/4 inches varying in the thickness from 3-3/16 inches to 8-3/16 inches. The outer (1/2 inch) and inner (3/4 inch) skins are of 5456-H343 aluminum armor plate and the lateral bulkhead (1/4 inch) and framing extrusions (3/8 inch) are of 5456-H311 aluminum. Transverse strips of aluminum abrasive tread plate two inches wide are welded to the inner skin to provide a skid-resistant surface on the ramp.

A pair of aluminum hinges with a one inch steel pin are located at either side of the bottom of the ramp. Two manually operated latch mechanisms are located at the top of the ramp. These latches pull the ramp up against its rubber seal by means of handwheels. Both latches are released simultaneously by means of trigger type devices actuated by a lever in the driver's area.

A tubular rubber seal which is keyed into and bonded to the portal structure provides a watertight seal between the ramp and the hull.

A hinged 1/4 inch plate, the full width of the ramp, bridges the gap between the ramp and the vehicle threshold when the ramp is extended.

The ramp is actuated by means of a single hydraulic cylinder, pulley, and cable arrangement on the starboard side of the ramp opening. (Refer to Section 4.7 for a discussion of the hydraulic circuit).

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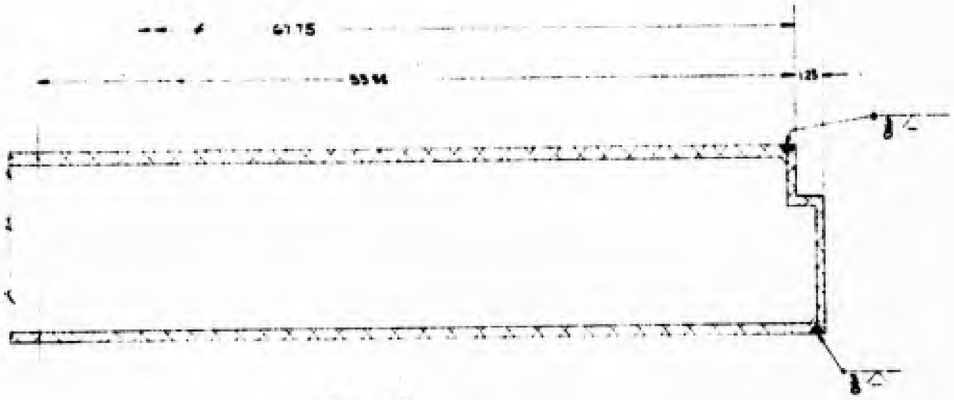

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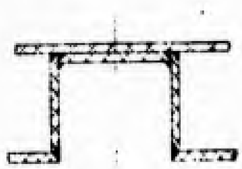

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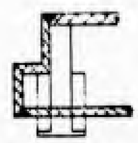

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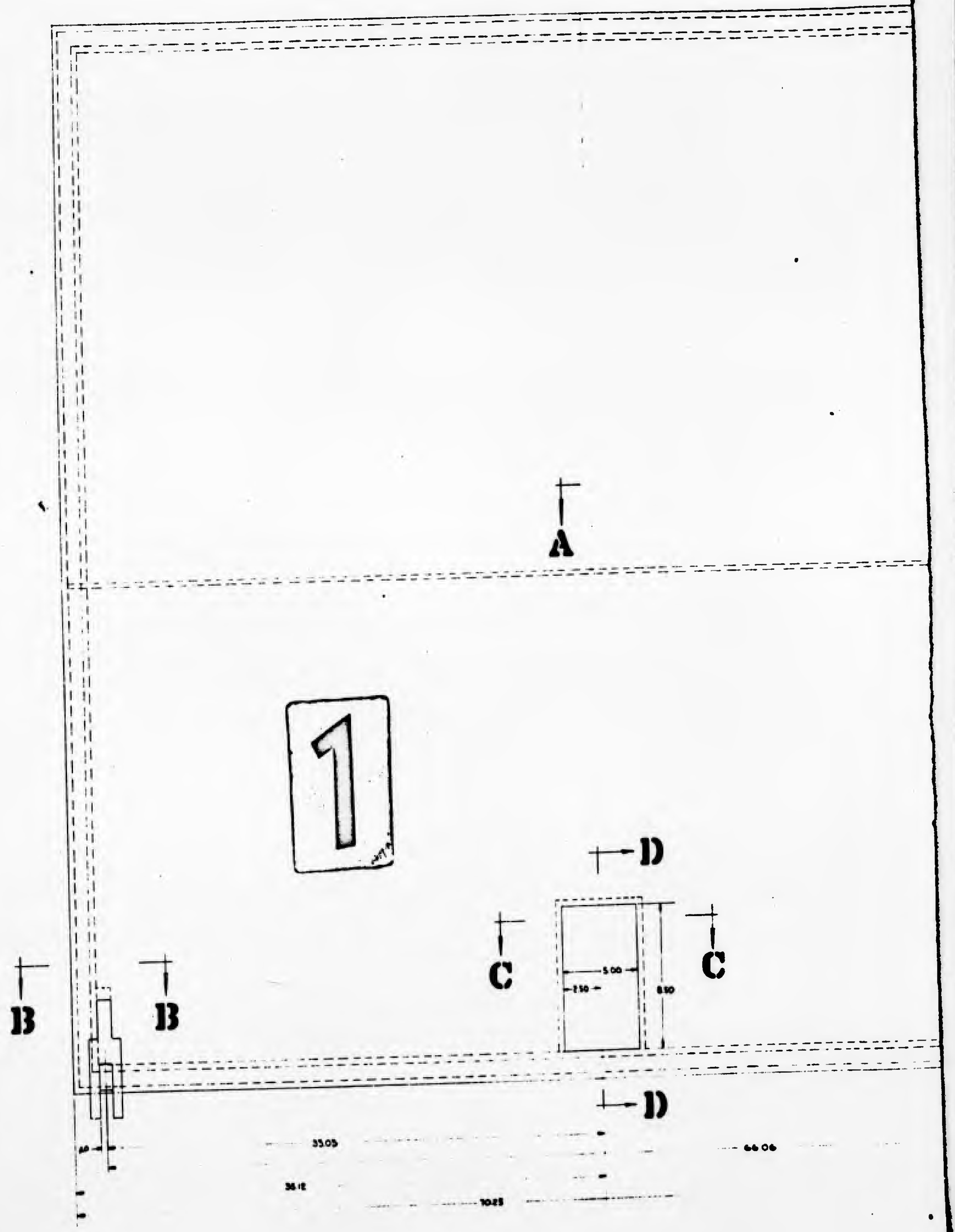
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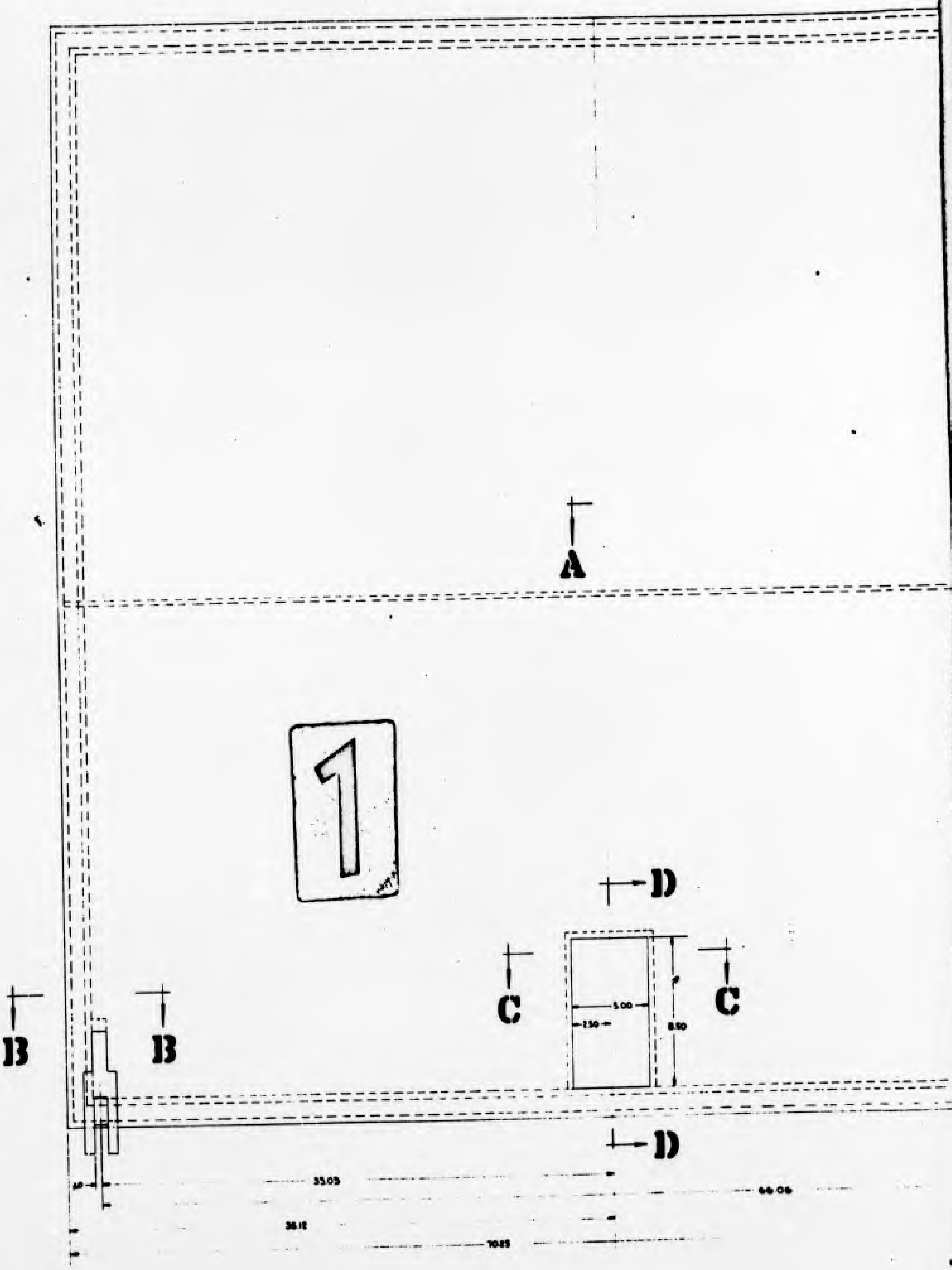
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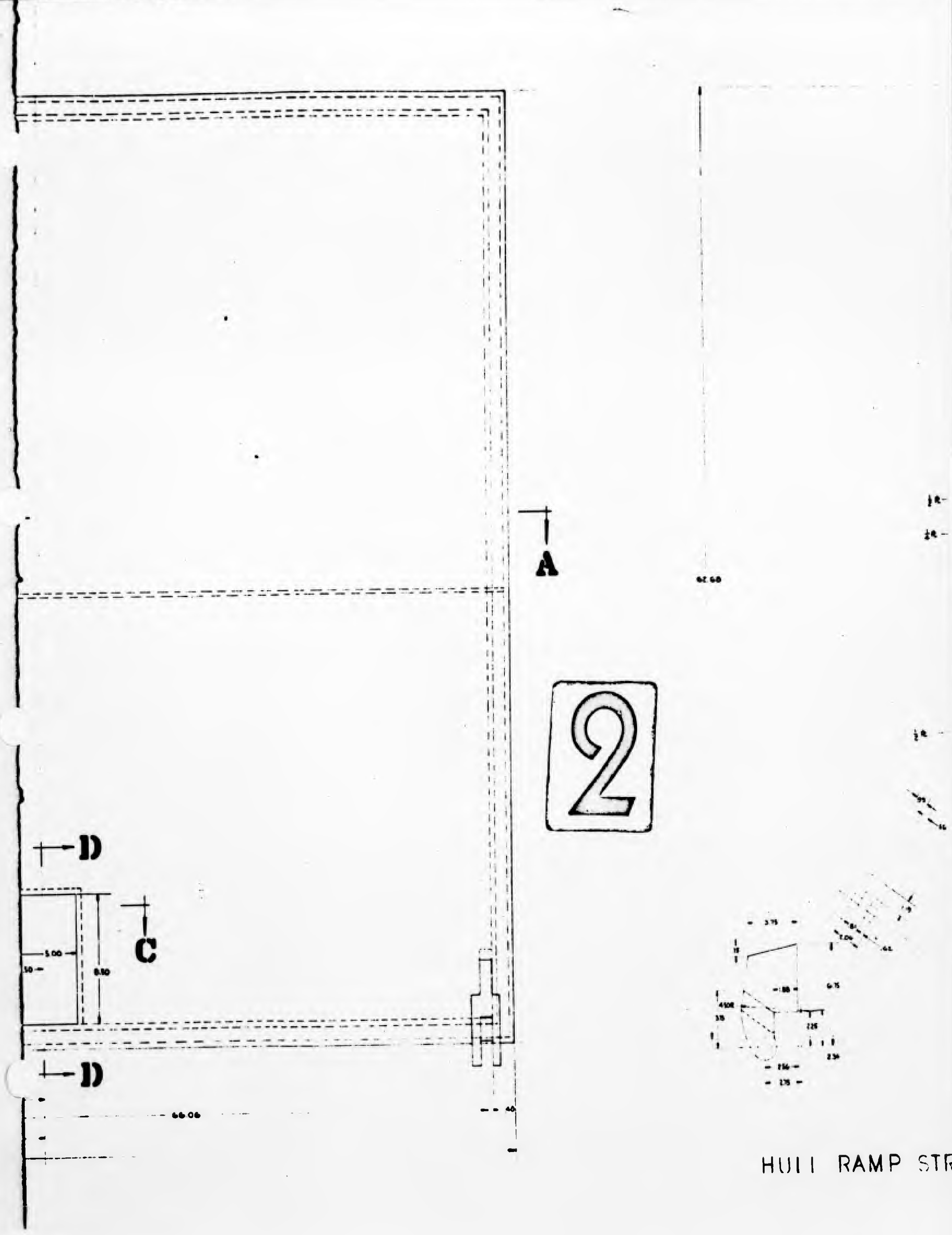
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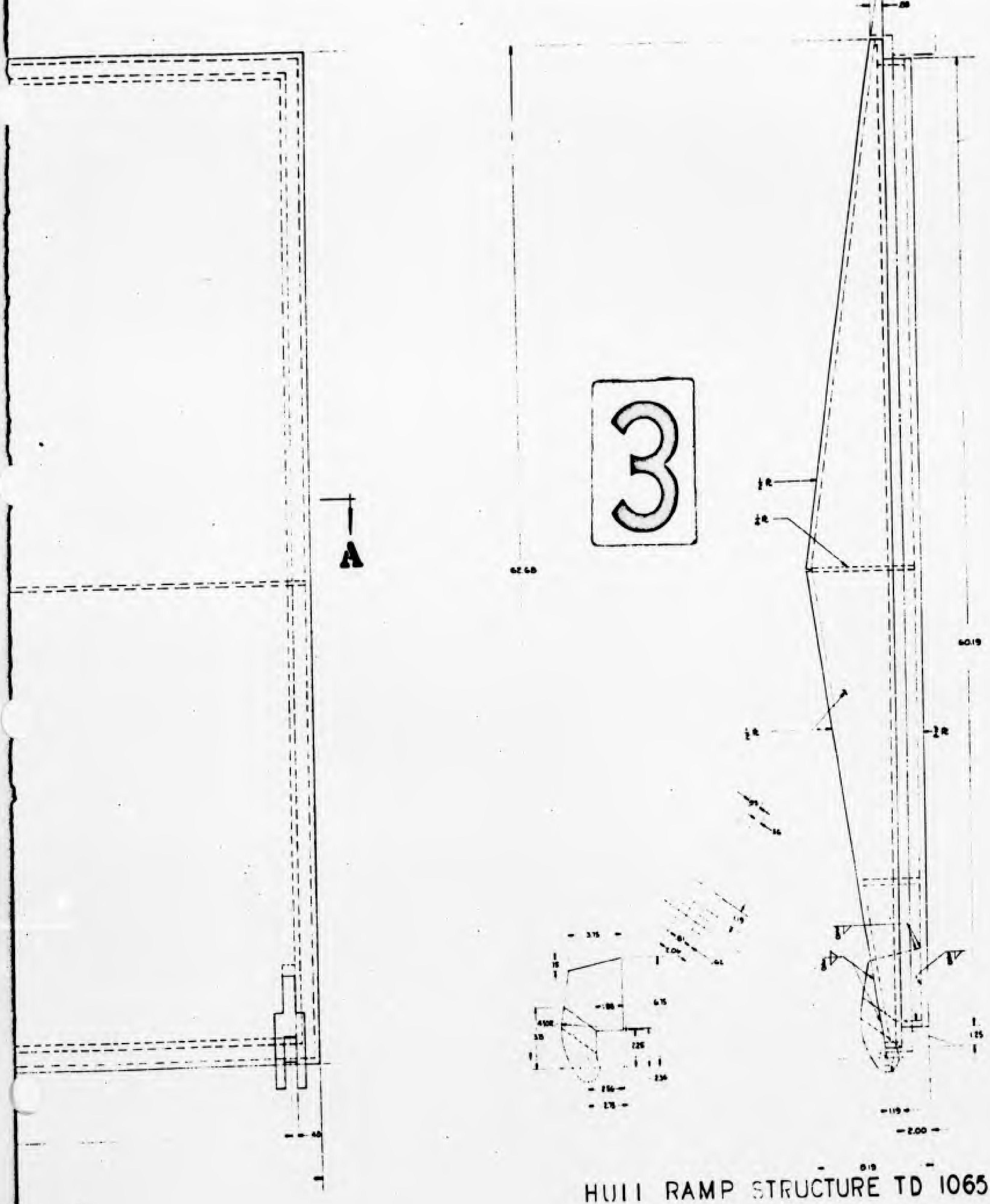


HULL RAMP STRUCTURE TD 106533

4.5.4 <u>HATCHES AND COVERS</u> (TD 106534 following page 4-58)

All of the various hatches and covers, as enumerated in Section 4.5.2, are provided with tubular rubber compression seals which are keyed into and bonded to either the hull or the hatch or cover. When in the closed position, the seal is compressed a nominal 1/4 inch to compensate for tolerances in the clearance between the hatch or cover and the hull and will thus have sufficient compression to provide a waterproof seal.

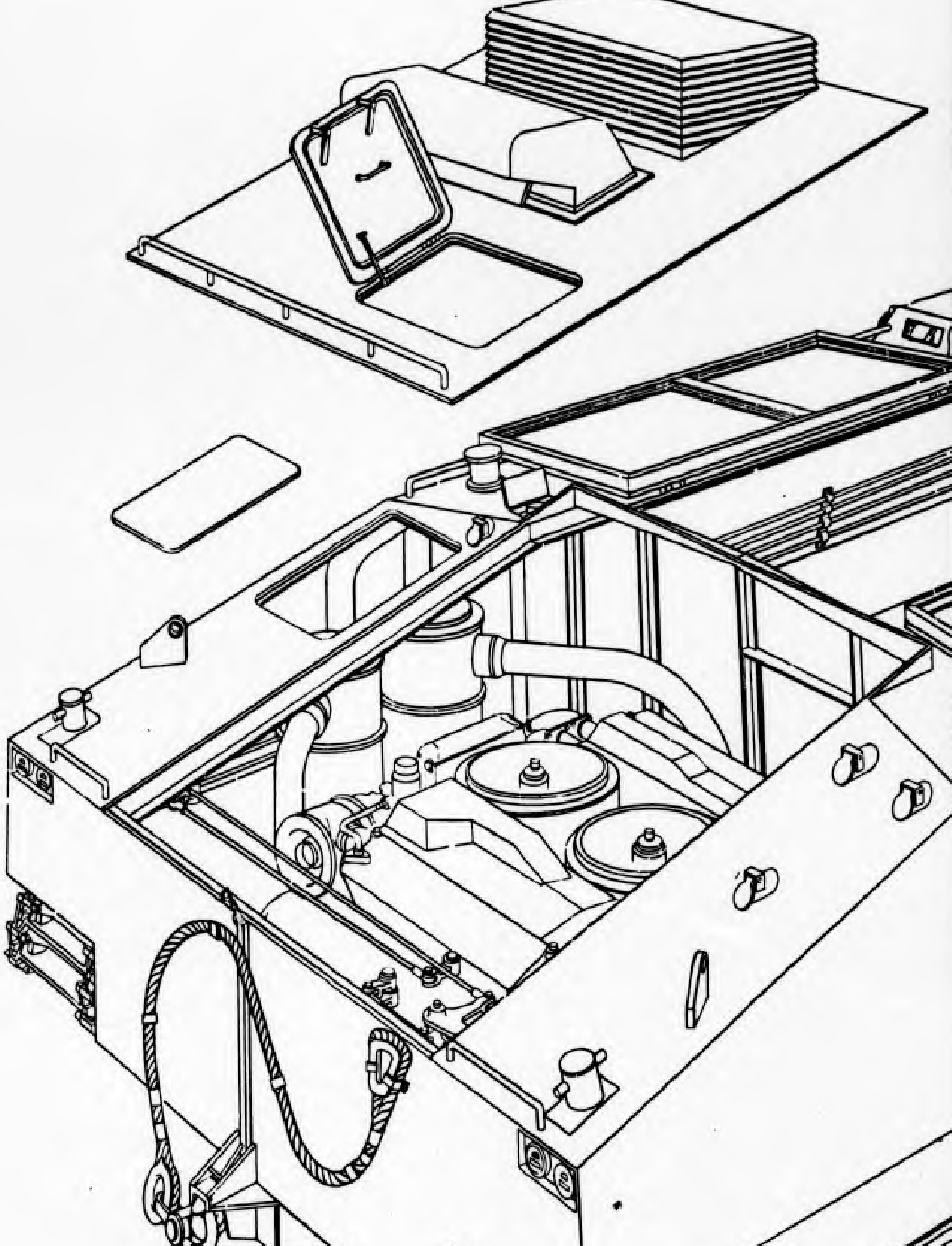
The main cargo hatch opening in the top of the vehicle has two 3/8 inch thick aluminum doors, 87 inches by 29-1/2 inches each. The edges of the door at the hull centerline are supported by 5 inch deep special aluminum extrusions, one of which has an indentation for mechanical retention of a rubber seal. The other three edges are stiffened with 2-5/8 inch deep extrusions which also retain rubber seals. This extrusion overlaps the edge of the hatch opening and the seal compresses against the top of the hull. Three hinges are provided on the outboard edge of each door, two near the corners and one at the center. Two torsion bar springs are located near the hinge line, which counterbalance the weight of the door when it is closed or open 180 degrees. Two cam-type latches, operable from either the outside or the inside, secure each of the doors in the closed position. Locking slide bolts are provided on the inside. Stays to hold the doors in the open position are located on the top deck.

The driver's and assistant driver's hatches are hinged on their inboard edge and are secured with dog-type latches, operable from either the inside or the outside of the vehicle. The assistant's hatch has a locking slide bolt on the inside, while the driver's hatch can be locked with a padlock on the outside.

The escape hatches, located on the sides of the vehicle, have quick release, toggle-type latches which, when released, allow the hatch covers to fall away until restrained by chains provided to prevent their loss.

The engine compartment access hatches, both on the top of the vehicle and in the bulkhead between the personnel cargo compartment and the engine compartments, are hinged and are secured by dog-type latches from either inside or outside. The engine compartment access hatch on the top can be locked with a padlock on the outside.

The removable panel on the top of the engine compartment is secured in position with 12 bolts. When removed, the engine and transmission assembly can be removed as a unit. This panel is of 3/8 inch aluminum and is framed with an aluminum stiffener to assure a uniform fit around the edges. Two lateral stiffeners provide intermediate support for the panel. The air intake tower, the exhaust tower and one of the engine compartment access hatches are located on this panel.



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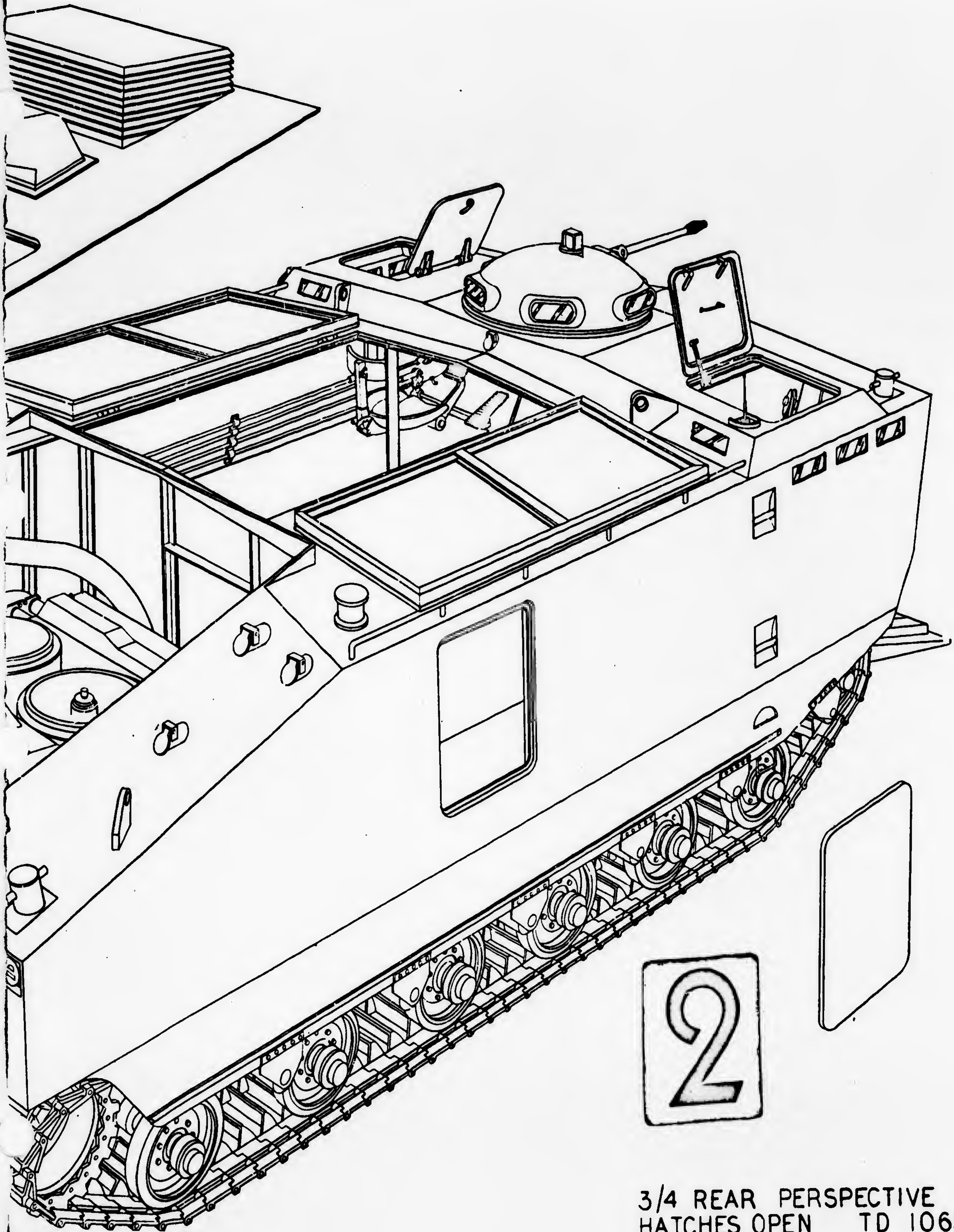

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3/4 REAR PERSPECTIVE
HATCHES OPEN TD 106534

4.5.5 VISION BLOCKS

The driver's station and crew chief station are equipped with ten vision blocks each. The vision blocks have been located to afford maximum visibility from each station without sacrificing ballistic protection of the personnel. The vision blocks located at the driver's station are arranged as follows: three facing forward on the front vertical face of the superstructure, three on the front surface of the hull below the headlights, outboard of the ramp, three on the top front vertical surface of the hull side, and one on the outboard rear vertical face of the superstructure.

Each vision block is nine inches wide and four inches high tapering back on each side fourteen degrees. The top horizontal surface slopes downward from the front face thirty degrees, while the bottom surface slope is twenty degrees downward. The depth of the vision block is five and one-half inches, while the depth normal to the front face is two inches. The rear face dimensions are nine and one-quarter inches wide and four inches deep.

The vision block will consist of a minimum of seven glass plies conforming to government specification DD-G-451, type 1. The glass utilized in fabrication of the vision blocks will be bonded together with commercial polyvinyl butyral having a minimum thickness of twenty-thousandths of an inch. The glass quality and uniformity of the finished vision will conform to government specification MIL-G-54858. The adhesive sealing compound will conform to government specification MIL-S-11031.

The luminous transmittance will be a minimum of sixty-five per cent when tested in accordance with government specification MIL-B-11352.

4.5.5.1 DRIVER'S VISION CAPABILITIES

The vehicle driver will be provided with a series of vision blocks to give the following instantaneous fields of view. The forward horizontal field of view is 98 degrees. The forward vertical field of view is 8 degrees throughout the 98 degree horizontal plane and 20 degrees throughout a 12 degree horizontal plane. The driver's side vision is 72 degrees horizontally and 20 degrees vertically.

Night vision will be accomplished by the use of the T6A helmet-mounted infrared binoculars. A 5 per cent loss of infrared vision will be experienced through the vision blocks. The selection of the helmet-mounted infrared binoculars is intended to provide the driver the maximum possible field of view.

4.5.6 SEATS

The vehicle is equipped with two seats, mounted over the tracks, at the forward end of the troop compartment. The driver's seat is on the port side, and the crew chief's seat is located symmetrically opposite on the starboard side.

The seat is the padded bucket type. The seat is adjustable upward, downward, forward and rearward. The seat back angle is also adjustable. Both seats are equipped with web type safety belts.

The seat operation requires body weight to control action while the seat is being adjusted. To raise or lower the seat, the vertical adjustment handle must be pulled, and the seat moved to the desired height, and the release of the adjustment handle locks the seat in the new vertical location. The horizontal adjustment handle must be depressed to move the seat forward or rearward, and release of the adjustment handle will lock the seat in its new position. The seat back angle is set by pulling up on the adjustment lock, and locating the desired position, then engaging the lock. The seat back proper may be re-adjusted by loosening the wing nut on the back side, resetting the seat back and tightening the wing nut.

The safety belt is the web type with a quick release buckle, and the belt is adjusted by pulling the free end of the belt through the buckle to the desired tightness.

The cargo compartment can be converted into a troop compartment by seating twenty-seven men on eight benches and a gunner's stool. The seats are of the bench type in order to accommodate all body types and sizes without adjustment. The bench type is inexpensive to construct, and adopts most easily to this arrangement. This type of seat is readily stowable against the outer walls inside of the vehicle behind the driver and crew chief seats, forward of the escape hatches. When the benches are in position they afford generous aisle spaces. The benches can be repositioned to serve as temporary shelving or stowage surfaces.

4.5.7 CREW COMPARTMENT VENTILATION & HEATING

4.5.7.1 VENTILATION

Aside from providing adequate spatial accommodation, the troop compartment must also provide a comfortable environment. The physical variables which have an influence on troop comfort and must be controlled within permissible limits are ventilation, temperature and humidity. All three of these factors are a function of the outside natural environment.

Ventilation requirements are dictated by the necessity to remove from the compartment the carbon dioxide, heat and moisture generated by the body and to provide oxygen for breathing. Of secondary importance is the removal of or reduction of the intensity of body odors generated in order to forestall motion sickness. Carbon monoxide, normally an important criteria, will not affect ventilation requirements due to its low yield in diesel exhaust and the presence of a watertight bulkhead separating engine and troop compartments. The ventilation rates required to maintain carbon dioxide concentration 0.5 per cent vary up to approximately 2 cfm for men at rest. Normal practice is to provide 10 cfm per person, which would necessitate 290 cfm total for the 29 man vehicle capacity.

To provide good ventilation for people closely confined, an air change once every five minutes is recommended. Then to provide ventilation and cooling, which means exhaust ventilation with a rate of air change high enough to provide a combination of good ventilation, refreshing air movement and effective heat exhaust, the recommended air change is once every two minutes. Since the effective volume in the LVTPX 11 troop compartment is approximately 550 cu. ft., this results in an air flow of 110 cfm for a five minute air change, or 275 cfm for a two minute air change. Since the two figures of 290 and 275 agree very closely, Chrysler elected to provide 300 cfm of ventilating air. To increase the flexibility of the system for cold weather operation, two speed motors on the two blowers would be incorporated. The slower speed would provide a reduced rate of 150 cfm, which is less than the approximate 200 cfm of the selected heater, thus allowing for some heated air recirculation. The reduced flow still provides over 5 cfm/man which is in excess of the minimum requirement for preventing an excessive CO_2 concentration.

To provide for ventilating air, an intake vent is placed in the vehicle top, over the starboard stowage compartment. A trap arrangement will allow any water which enters the vent to be piped to the bilge area. A spring-loaded flap valve has been placed in this pipe to prevent air from being sucked from the bilge area (Drawing TD106535 following page 4-64). A wire-wound fabric duct routes the air through the watertight bulkhead into the engine compartment area. The duct connects into two centrifugal blowers mounted on the bulkhead wall. Grille openings are provided, one on either side of the vehicle centerline, for exhausting air into the troop compartment between the rows of seated troops at about head level. All equipment is water tight to maintain the watertight bulkhead concept. Grille outlet area will be such as to keep the air velocity consistent with that recommended for persons seated. To allow for proper air flow, an exhaust vent from the troop compartment will be placed to the rear of the cupola in the raised superstructure.

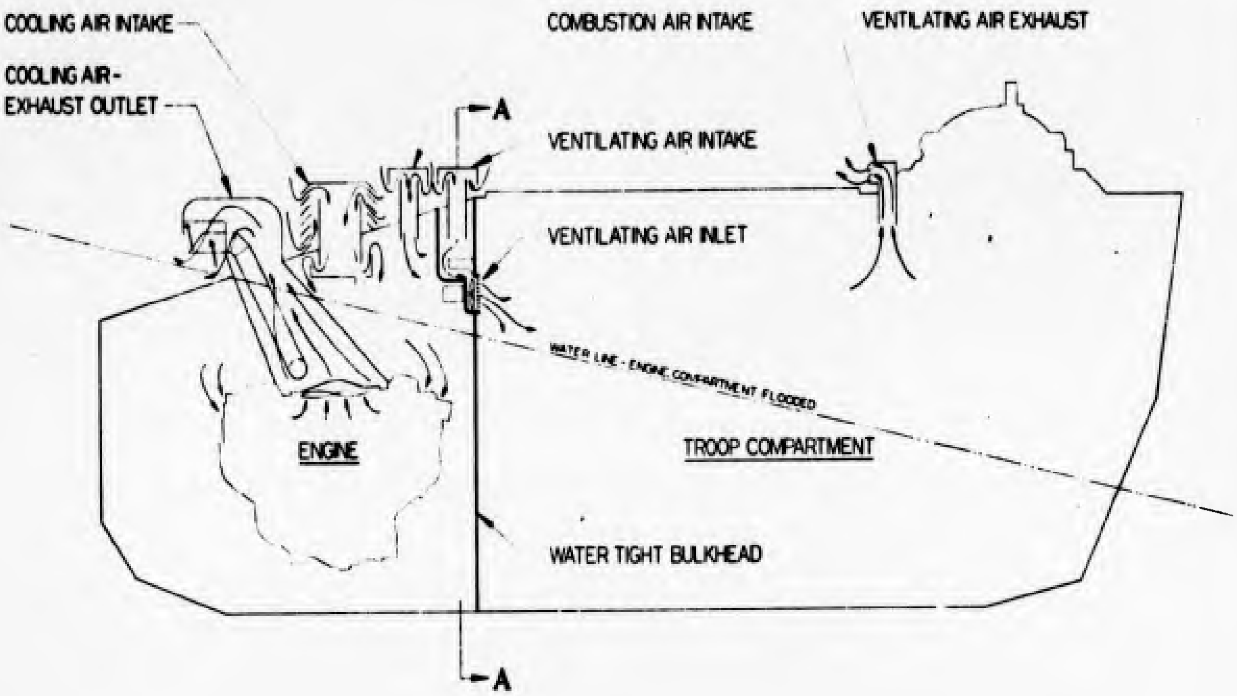
4.5.7.2 HEATING

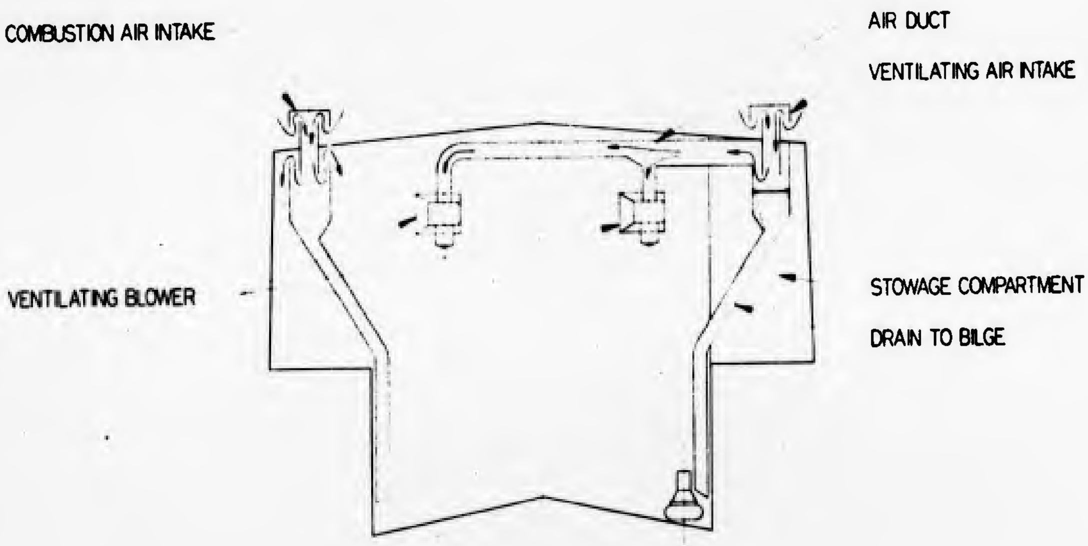
The vehicle's interior space is approximately 720 cu. ft. and it has been specified that the ambient design temperature would be -25 degrees F. Based on personnel normally being clothed for protection against the ambient temperatures and with no actual personnel contact with the outside skin of the vehicle due to a seating arrangement with a back rest and raised floor, it is concluded that one 60,000 BTU/hour heater would adequately warm the troop compartment to provide warmed ventilating air. Provisions for mounting a single 60,000 BTU/hour Stewart-Warner Model 8460-C24 "All Purpose Multifuel Heater", or its equivalent, would be made in the troop compartment near the driver's position.

A winterization kit would provide this heater and necessary fuel line, exhaust line and electrical harness.

The fuel line going to the primer pump has a plugged tee which would be utilized to provide fuel to the heater. An opening near the port bilge outlet would be provided to route an exhaust tube from the heater to the outside. A heater control switch and indicator lamp are mounted on the instrument panel.

The Continental AVDS 750-1 engine chosen does not require any special type of starting aid for ambient temperatures down to -25 degrees F.





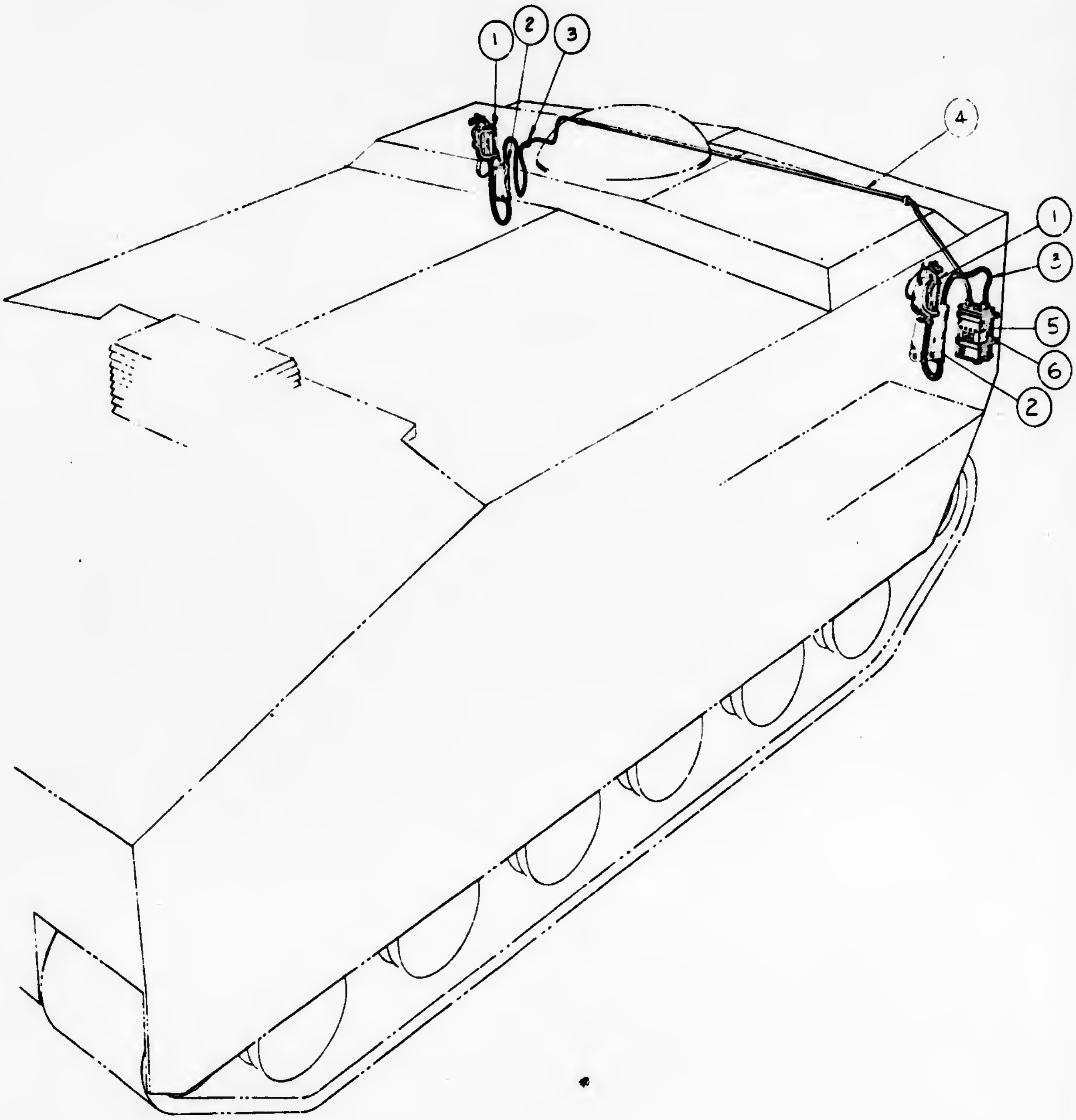
SECTION A-A

VENTILATION SCHEMATIC TD 106535

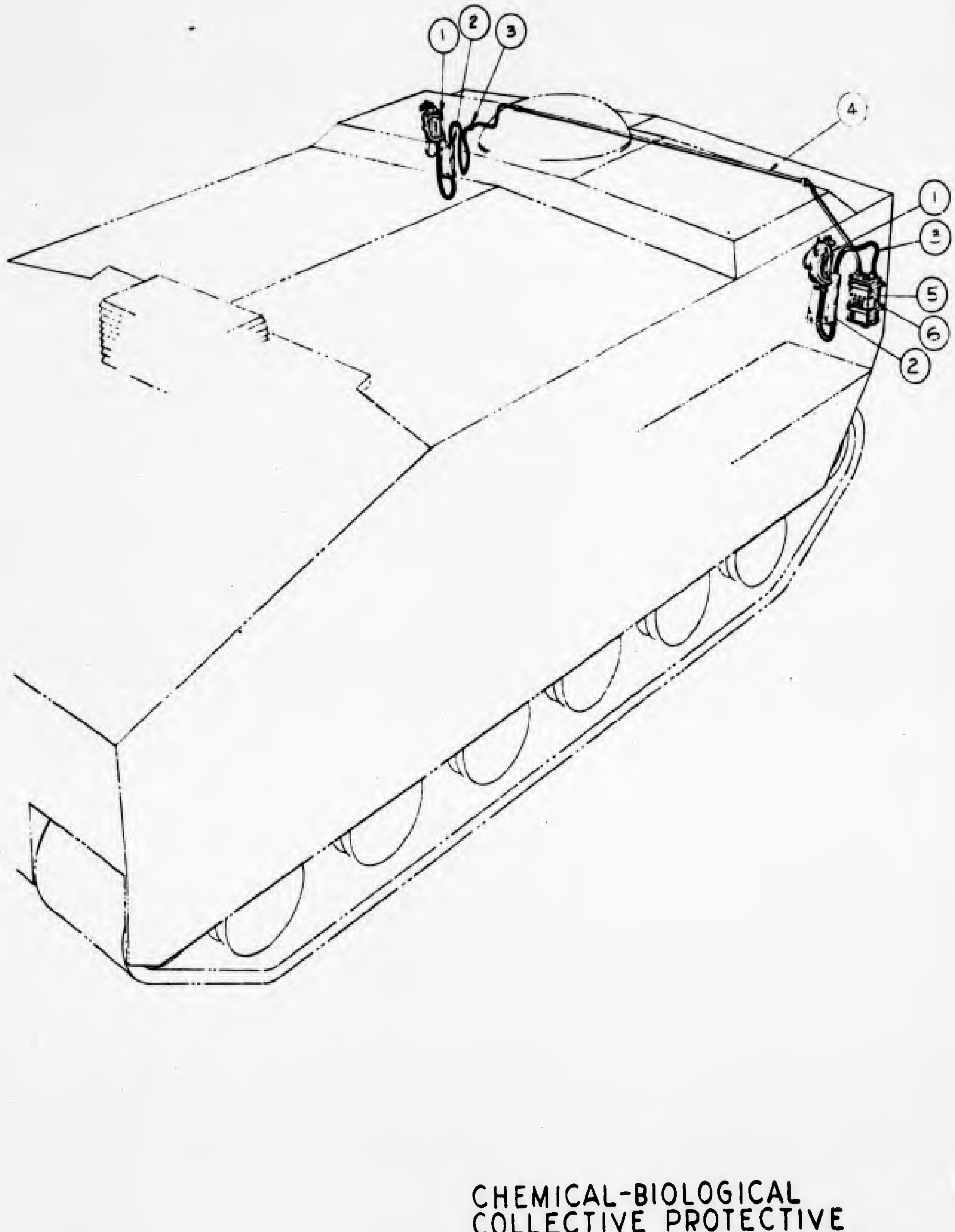
4.5.8 CHEMICAL-BIOLOGICAL PROTECTOR COLLECTIVE UNIT

To provide Chemical-Biological protection for the crew members, "Protector Collective Tank, Three Man M8A1" and "Filter Unit Gas Particulate M8A2", were reviewed. They are identical except that the M8A2 has an installation group consisting of cable assembly, frame assembly and lamp switch assembly. Since this group can be utilized in this installation, we have selected the M8A2 filter unit. The frame assembly is mounted on the vehicle structure and utilized to retain the air purifier assembly while the lamp switch assembly is incorporated into the instrument panel for unit operation and the cable assembly would be adaptable into the vehicle electrical harness.

To review the system, reference should be made to Drawing TD106536. Snap clips have been provided beside both the driver and assistant driver to hang the carrier assembly and protective mask. Flexible hose connects the mask to the carrier assembly. Hose and tubing in the case of the driver's equipment and hose only for the assistant driver's equipment, connects the carrier assembly to the air purifier assembly. Consideration in locating the system was made to allow free entry of air and exhaust of foreign matter to and from the purifier assembly. Hose and tube lengths utilized are well within the footage provided in the kit. The tube running across the front of the vehicle to service the driver's unit would be made of thin wall aluminum tubing and would be placed inside the box frame member of the vehicle structure. The lamp switch assembly mounted on the instrument panel will be actuated to turn the unit on or off.



CHEMICAL-BIOLOGICAL
COLLECTIVE PROTECTIVE
DEVICE TD-106536



CHEMICAL-BIOLOGICAL
COLLECTIVE PROTECTIVE
DEVICE TD-106536

4.5.9 BILGE PUMP SYSTEM

The vehicle is provided with four (4) hydraulically driven bilge pumps and two electrically driven pumps, as shown on Drawing TD106537 (following page 4- 67). The four hydraulic bilge pumps are interconnected with the engine of the vehicle and are engaged and operated continuously with the power plant. The electrically driven pumps are intended for emegency usage when the main engine may be inoperative. The bilging capacity of the hydraulically driven pumps is 300 gpm each and the auxiliary electrical units have a capacity of 125 gpm each. The vehicle, then, has a total bilging capacity of 1,450 gpm. The pumps are arranged in the vehicle so that there is bilging capability during the periods the vehicle is listing severely to either side or the vehicle is trimmed down severely at the bow or stern. The hydraulically driven bilge pumps are driven continuously with the engine in order to provide the vehicle with bilging immediately upon water entry without any attention from the driver.

The bilge pump units are centrifugal pumps with a radial-type impeller, having an inlet diameter of three inches and an outlet diameter of six inches. The centrifugal pump was chosen because:

(1) It has no reciprocating parts; the units are inherently balanced.

(2) Clearances are relatively large. There are no internal rubbing parts. Consequently, there is less wear due to passage of grit and sand.

(3) The pumps are self-priming.

(4) Because of the relatively high operating speed, the units can be designed smaller and lighter

than comparable units having the same capacity and head, thereby reducing the required space, weight, installation, and initial cost of the unit.

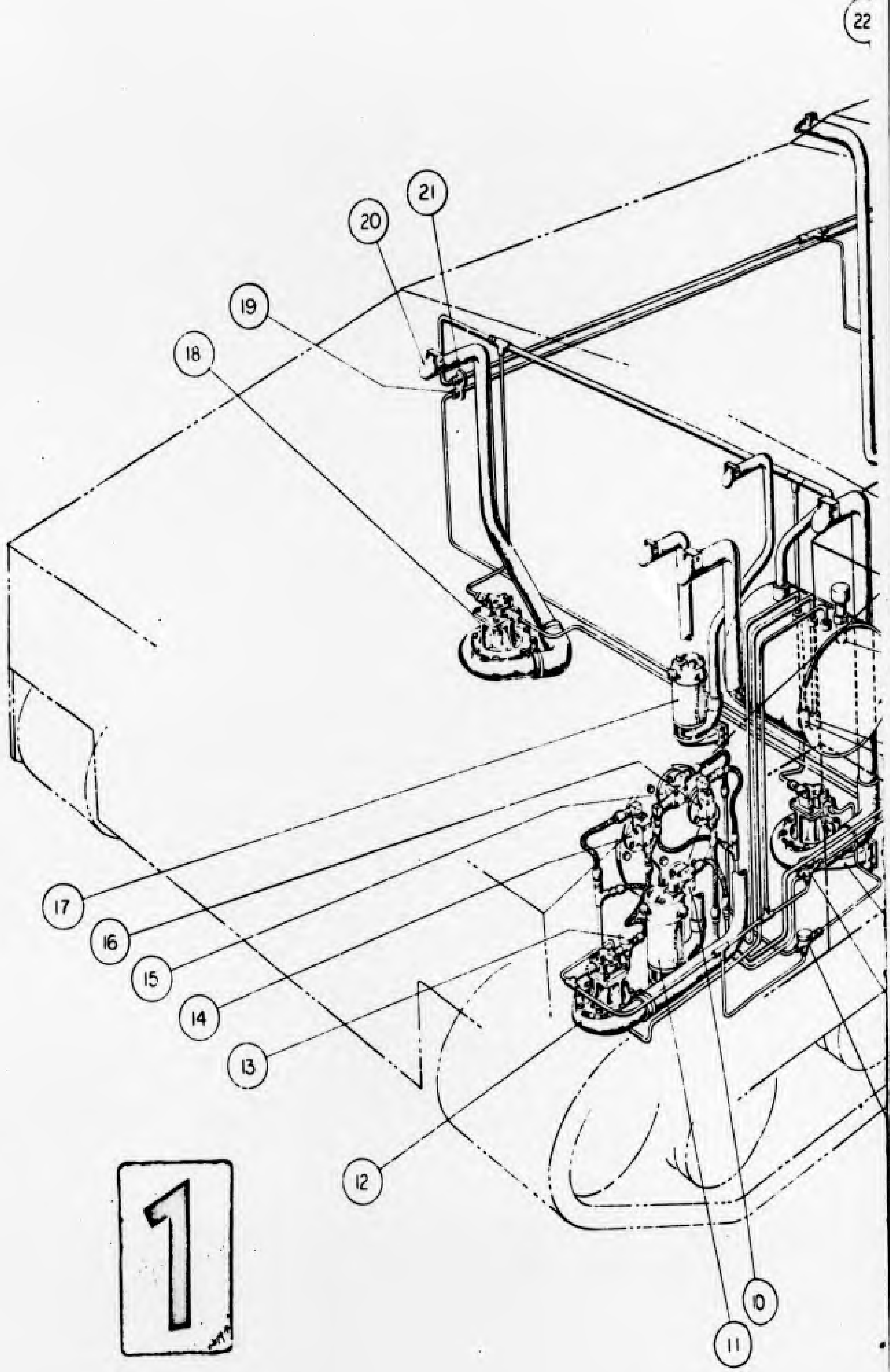
(5) There are no internal valves or receivers to cause operating difficulties.

Furthermore, the bearings are located outside of the pump casing and are tightly sealed keeping the water from coming in contact with the bearing lubricant. The units also lend themselves readily for direct coupling to drive motors so that transmission losses are kept small.

The bilge pump is driven by a hydraulic motor at speeds directly proportional to engine speeds. (The centrifugal pump and motor will be operating between 800 and 3,000 rpm.)

Hydraulic fluid is supplied the motors by individual hydraulic pumps that are driven off a common gear box, as shown on Drawing TD106523 (following page 4-15). The gear box is in turn driven by a power take-off directly from the transmission.

The auxiliary electrical bilge pumps are discussed in further detail in section 4.8.5.



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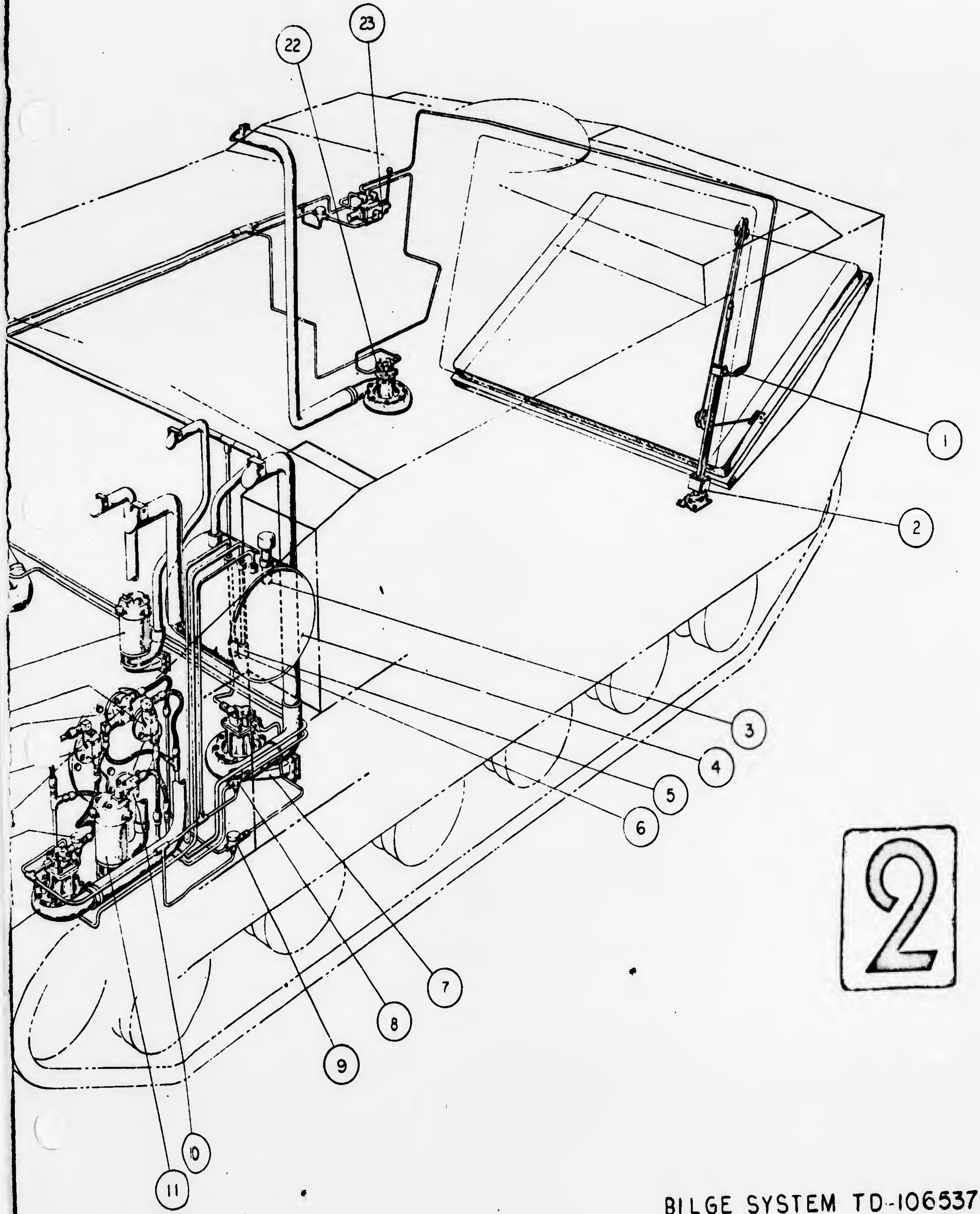


BILGE SYSTEM TD-106537

4.5.10 LIFTING AND TOWING DEVICES

4.5.10.1 LIFTING EYES

Four lifting eyes are welded to the top surface of the hull with reinforcing members inside of the hull. These provide a means of attaching a lifting sling, capable of supporting the completely loaded vehicle.

4.5.10.2 TOWING DEVICES

The vehicle shall be equipped with a wire rope assembly and two towing eyes. The forward towing eye is located under the ramp opening, at the vehicle centerline, attached to the forging that also anchors the ends of the compensating idler. The second towing hook assembly is located at the bottom center of the stern sheet. Therefore, the vehicle may be towed either backward or forward, as required.

A wire tow rope assembly is held on the stern sheet by clips. A link on one end of the cable is attached to the tow hook. The link is quickly released from the vehicle by the unlatching hook.

The hinged towing hook may be unlatched by pulling on the hook lock. The hook then falls open by its own weight. The hook may also be unlatched by pressing a foot pedal on the top deck, which is connected to the tow hook release by a wire cable assembly. The release mechanism on the bow is routed around the ramp opening.

4.5.11 DRAIN PLUGS

Eight drain plugs are located symmetrically on both sides of the hull bottom. Drain plugs are placed at the front and rear of the two watertight compartments. This allows the vehicle hull to be drained in any position. The plugs are one and one quarter inches in diameter and are positioned in the heavy longitudinal wear extrusions at the sides of the hull bottom. They are recessed into the extrusion to prevent their being damaged. Two of the eight plugs are located under the fuel tanks, and also serve for fuel tank drain access. The aft plugs are access plugs for the hydraulic reservoir, engine, and transmission drains.

How do you get to the plugs?

4.6 MECHANICAL CONTROLS

4.6.1 GENERAL

A perspective of the mechanical control system is shown on TD 106538, following Page 4-77. The choice of a mechanical rod and lever linkage arrangement for the brake, steering, throttle, and transmission range controls was made based on consideration of the vehicle design objectives of simplicity, ease of maintenance, durability, reliability, and minimum cost. The relatively direct routing of controls between the driver station and transmission made this type of system the most desirable due to the minimum number of joints and pivots required.

The longitudinal linkage between the driver station and engine compartment is routed along the left side of the vehicle between the cargo compartment and the outer skin. The control rods are retained by supports (6) * at frequent intervals along their length to prevent bending and interference. The transverse linkage to the transmission will be insensitive to driveline torsional oscillation since the rods are parallel to the rotational axes of both the engine and transmission. The rod ends employ sealed spherical bearings to maintain minimum backlash and friction. The relatively high location of control rods and levers minimize the possibility of a malfunction due to ice accumulation from intermittent water immersion.

To facilitate adjustment of rod lengths during installation, each lever will be provided with an alignment hole. The lever will be accurately positioned when the centerline of

* (6) numerals refer to Callouts on TD 106538

this hole coincides with a corresponding hole in the lever mounting bracket. In this manner, optimum adjustment is maintained for efficient control operation. Oilite bushings will be used at all pivot points to eliminate a lubrication requirement. Watertight boots (8) are mounted on the control rods at the bulkhead between the engine and cargo compartments.

4.6.2 STEERING CONTROL

The vehicle is steered through a linkage system (1) between the steering wheel and the steering control lever (11) on the transmission. Clutch-brake steering is employed in the two lower gear ranges and reverse while geared steering is provided in the two higher ranges. Full steering is accomplished by rotating the steering wheel 24 degrees in either direction. For this reason, and to provide better driver visibility, the abbreviated form of wheel are shown on TD 106539, (following Page 4-78).

4.6.3 ENGINE THROTTLE CONTROLS

The engine throttle lever (7) is operated through linkage system (2), which is actuated by an accelerator pedal through a shaft and lever arrangement. This lever is also operated from a hand throttle through a push-pull cable. A tension spring on the lever returns the linkage to the engine idle position when the pedal is released. An additional rod is mounted between the transmission throttle valve (9) and the transmission-mounted throttle pivot lever. This actuates the throttle valve in conjunction with the engine throttle. Through this arrangement, the speed at which the torque converter goes into lock-up is varied relative to vehicle power requirement.

4.6.4 SERVICE BRAKE CONTROL

Control of vehicle braking is accomplished through the service brake linkage (4). The shaft-mounted brake pedal actuates the linkage through a lever at the outer end of the brake shaft. Sufficient lever ratio is provided at the pedal to permit the driver to develop adequate braking torque to hold the vehicle on a 70% grade. Separate levers (12) and (14) are provided on the transmission to apply the left and right brakes respectively. These levers are actuated through an equalizer bar (13) which applies equal forces to both brakes independent of brake adjustment.

4.6.5 TRANSMISSION SHIFT AND PARKING BRAKE CONTROL

The transmission ranges are manually selected through a hand-operated lever and control linkage (3) which actuates the transmission shift lever (10). Both driver station and transmission levers are detented to provide positive engagement and proper feel. In addition, the driver's lever is gated to prevent inadvertent shifting between forward and reverse ranges.

A lateral movement of the shift lever from the neutral position engages the parking brake pawl on the bottom end of the lever with a ratchet (5) on the service brake shaft. If the service brake is now applied or has been previously applied and not released, the ratchet will prevent releasing of the brakes until the shift lever is returned to the neutral position. This arrangement simplifies the control system by eliminating a separate parking brake lever and prevents the vehicle from being driven with the parking brake applied.

4.6.6 DRIVER STATION CONTROLS (TD 106539 following page 4-78)

The general arrangement of the controls and instruments is shown on the perspective drawing, TD 106539. Every effort has been made to provide the driver with maximum control accessibility in both the upper and lower driving positions. Controls and instruments have been grouped in logical clusters to minimize the time required for a new operator to become familiarized with the vehicle. For example, the engine controls are located to the right side of the instrument panel, while the various lighting controls are on the left.

In the following section, a brief functional description of each control and instrument is given. (The numbers refer to callouts on Drawing TD 106539).

4.6.6.1 MECHANICAL AND HYDRAULIC CONTROLS

(2) Service Brake Pedal -- The service brake pedal is mounted on the floor ahead of the driver's right foot. Depressing the pedal applies the transmission brakes through a mechanical linkage.

(11) Primer Pump Lever -- The primer pump lever is located on the left wall of the driver station, behind the transmission range selector. A push button on the end of the lever opens the manifold heater fuel solenoid valves and energizes the ignition units. Lever actuation, with the button depressed, supplies fuel to the manifold heaters for cold weather starting by means of a double-acting primer pump. Lever actuation without depressing the button purges the fuel system.

(12) Transmission Range Selector and Parking Brake Lever -- Located along the left wall of the driver station, this lever selects the transmission driving range with a fore and aft movement through a mechanical linkage. Moving the control to the right from the neutral position prevents the release of the service brake linkage by means of a pawl and ratchet arrangement.

(13) Ramp Operating Lever -- This control, located ahead of the range selector, operates a 4-way hydraulic valve used to control the ramp actuating cylinder.

(22) Steering Control -- The steering control actuates the transmission steering shaft through a mechanical linkage, providing clutch brake steer in the two lower gear ranges and geared steer in the two higher ranges. Steering rate is proportional to control movement, with maximum steering attained at 24 degrees rotation in either direction.

(35) Hand Throttle -- The hand throttle is a push-pull control on the right side of the instrument panel. It is used to set the engine at a fixed speed for bilge pump operation, battery charging, etc.

(46) Accelerator Pedal -- The accelerator pedal is located on the floor, to the right of the brake pedal. Depressing the pedal advances the engine throttle.

4.6.6.2 INSTRUMENTATION

(18) Battery Generator Indicator -- The gage is on the left side of the instrument panel. It indicates the condition of battery charge.

(20) Engine Tachometer and Hour Meter -- The tachometer, located near the center of the instrument panel, is an electrical unit indicating engine speed in rpm. Scale range is from 0 to 4000 rpm. The hour meter records effective engine hours at a continuous speed of 2200 rpm.

(21) Engine Oil Pressure Gage -- Located at the upper left corner of the instrument panel, it reads engine oil pressure in pounds per square inch. Scale range is 0 - 120 psi.

(24) Low Oil Pressure Warning Lamp -- Located between engine and transmission oil pressure gages, the warning lamp glows when either engine oil pressure drops below 15 psi or transmission lube oil pressure drops below 9 psi.

(25) Transmission Lube Oil Pressure Gage -- Located to the right of the engine oil pressure gage, it reads transmission lubrication pressure in pounds per square inch.

(28) Speedometer and Odometer -- The speedometer, mounted to the right of the tachometer, indicates vehicle land speed in miles per hour. The odometer records accumulated mileage. Identical rotary type electrical transmitters are used for both speedometer and tachometer, being driven by the engine and transmission output respectively.

(29) Engine Oil Temperature Gage -- The gage is located to the right of the steering column and indicates engine oil sump temperature. Scale range is 120 - 240°F.

(30) Compass -- The compass AN5733 is located above the right side of the instrument panel so as to be easily visible to the driver. The compass is compensated by two adjusting screws for the north-south and east-west directions.

(31) Transmission Oil Temperature Gage -- Located in the upper right corner of the panel, the gage reads torque converter oil temperature with a scale range of 160 - 320°F.

(32) Oil Temperature Warning Lamp -- The warning lamp, located between the two oil temperature gages, glows when the engine oil temperature exceeds 245°F or the transmission oil temperature exceeds 285°F.

(34) Fuel Level Gage -- The fuel gage indicates the fuel level in either of the fuel tanks. The gage is located to the right of the speedometer.

4.6.6.3 ELECTRICAL SWITCHES AND INDICATORS

(3 45) Fire Extinguisher Switches -- The fire extinguisher switches, located in the center of the instrument panel, energize the first and second shot solenoid valves of the fire extinguisher system. The first shot switch also energizes the fuel shut-off solenoid to stall the engine. Guards over the switches prevent accidental actuation.

(4) Headlamp Dimmer Switch and (26) High Beam Indicator -- The dimmer switch, operated by the driver's left foot, selects high or low beam for both service and infrared headlamps.

(7) Master Heater Switch, (5) Hi-Lo Switch, (6) Heater Push-to-Test Lamp, and (1) Instruction Plate -- The master heater switch prevents accidentally shutting off the heater without permitting it to purge. The heater is started by putting the hi-lo switch in either the high or the low position. The lamp indicates that the heater is on. The instruction plate provides instructions for proper operation of the heater. The heater controls are located in the lower center of the instrument panel.

(8) Blackout Selector Switch -- The blackout selector switch, to the right of the lighting switch, selects either infrared or blackout drive lighting for operation under blackout conditions.

(9) Navigation Light Switch -- The navigation light switch, to the right of the lighting switch energizes the bow and stern navigation lights.

(10, 15) Auxiliary Bilge Pump Switches and (16, 17) Indicators -- The switches and indicators for the personnel and engine compartment auxiliary electric bilge pumps are located on the left side of the instrument panel. The indicators warn the driver that the pumps are on to avoid unnecessary discharging of the batteries.

(14) Lighting Control Switch -- The lighting control switch operates the service and blackout driving lights, blackout marker, and instrument panel iights. It is located at the lower left corner of the panel.

(23) Horn Button -- This momentary on switch in the center of the steering column actuates the horn.

(36) Fuel Sending Unit Switch -- This switch is mounted below the fuel level gage and enables the gage to read the fuel level in either fuel tank depending on switch position.

(37) Fuel Pump Switch -- This switch energizes the two submerged electric fuel pumps. A guard is employed to keep the switch in the on position under normal operation, in which case the fuel pumps are controlled by the master switch. The switch is turned off only when the master switch is to be left on for a prolonged period without running the engine.

(38) Engine Starter Button -- The starter button is a momentary on switch which activates the starter relay. It is located near the lower right corner of the instrument panel so as to be convenient to the driver's right hand, leaving the left free to operate the primer pump. The switch is protected by a cylindrical shield.

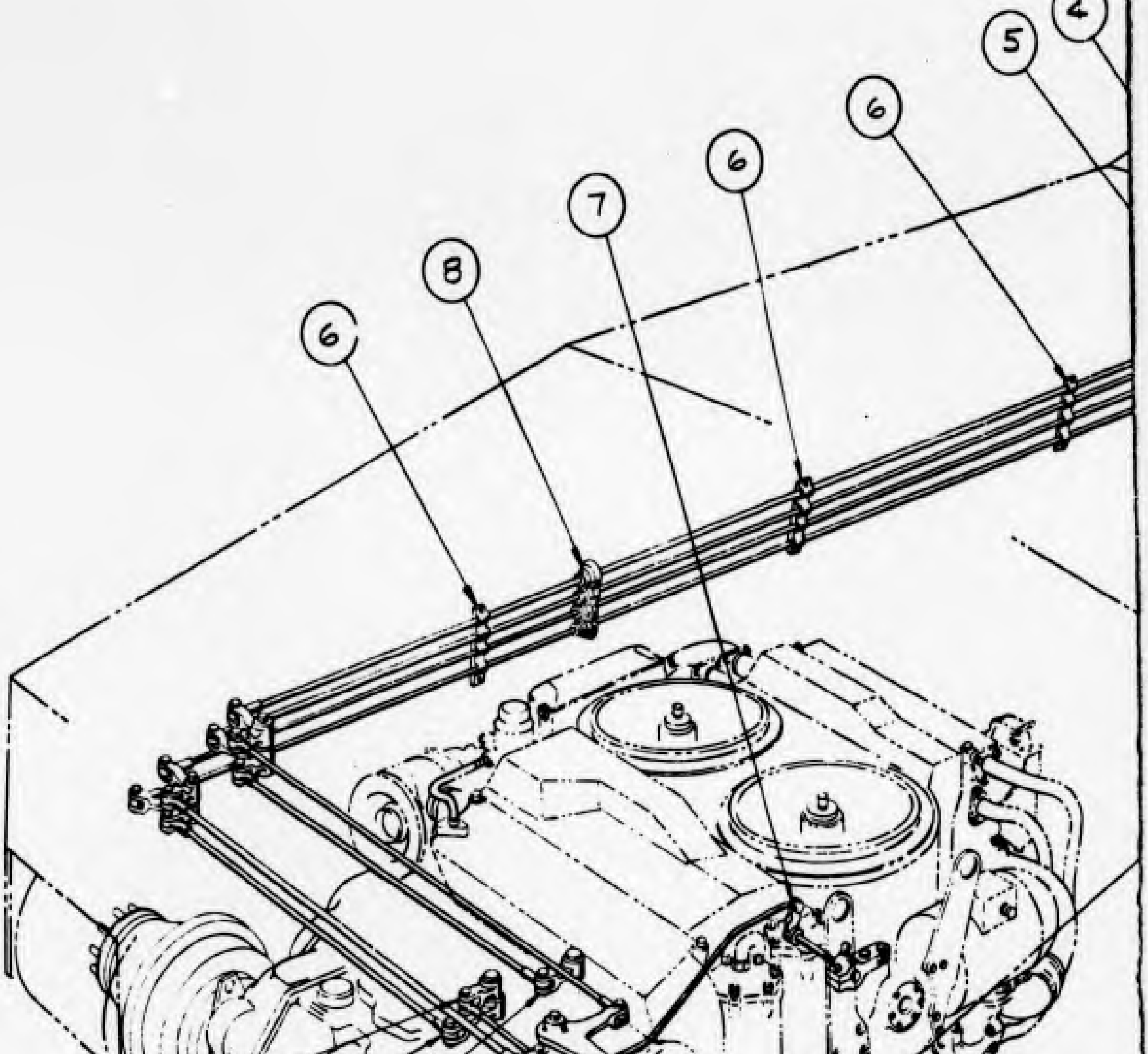
(39) Fuel Shut Off Switch -- This switch, below the starter button, stalls the engine by closing the fuel shut off solenoid valve.

(40) Personnel Ventilation Blower Switch -- This switch, on the lower right side of the panel, energizes the two blowers on the engine compartment bulkhead which ventilate the personnel and crew compartments.

(42) Master Switch and (41) Indicator -- The master switch energizes the entire vehicle electrical system, except for the heater which has its own master switch. The indicator glows when the master switch is on.

(44) <u>Gas-Particulate Filter Unit and (43) Indicator</u> -- The switch and indicator are located on the right side of the instrument panel. The switch energizes the filter unit blower and indicator lamp.

1. Steering Linkage
2. Throttle Linkage
3. Transmission Shift Linkage
4. Service Brake Linkage
5. Parking Brake Ratchet
6. Control Rod Supports
7. Engine Throttle Lever
8. Control Rod Boots
9. Transmission Throttle Valve
10. Transmission Shift Liver
11. Steering Control Lever
12. Left Brake Lever
13. Equalizer Bar
14. Right Brake Lever



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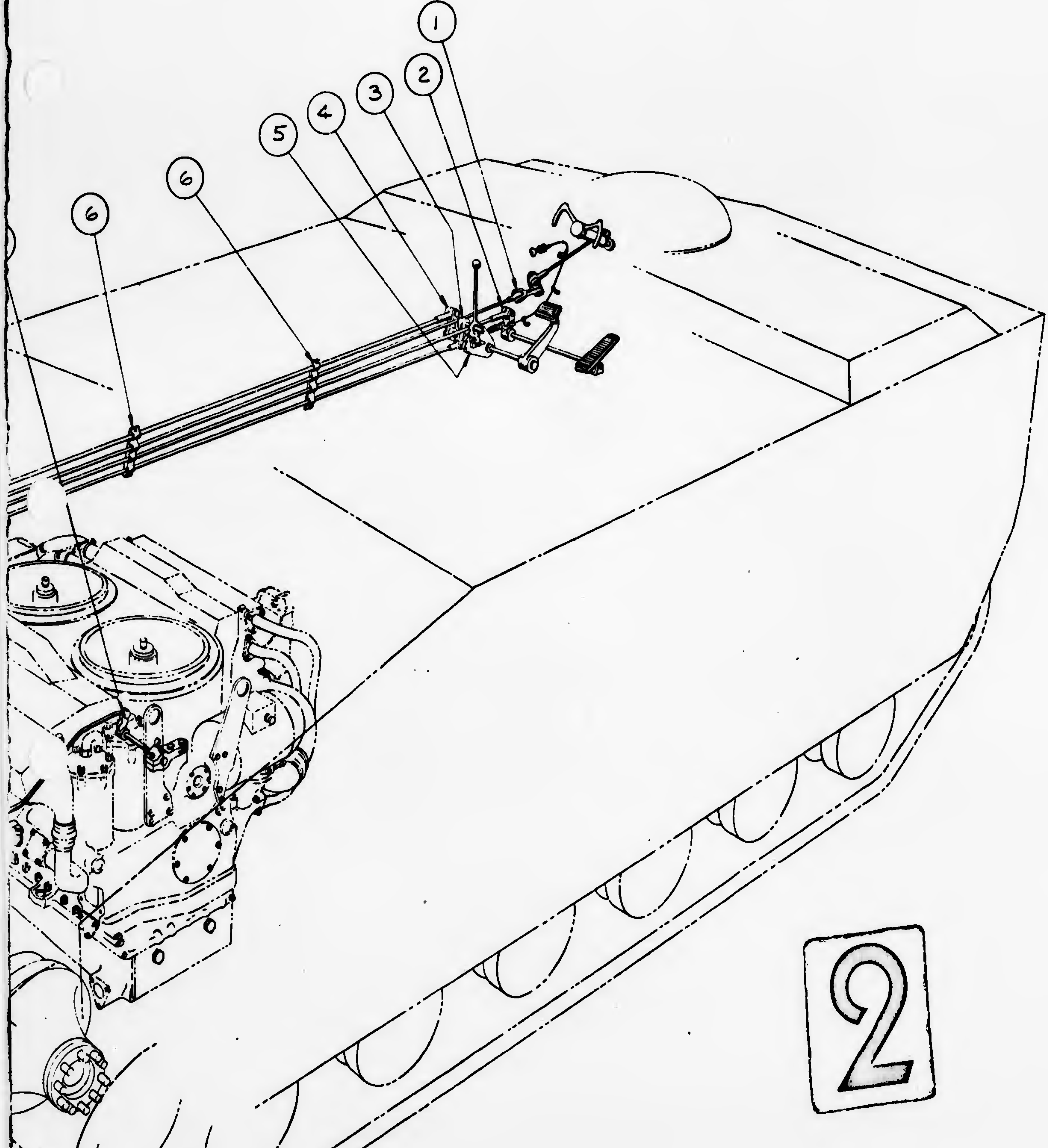

VEHICLE CONTROLS
TD-106538

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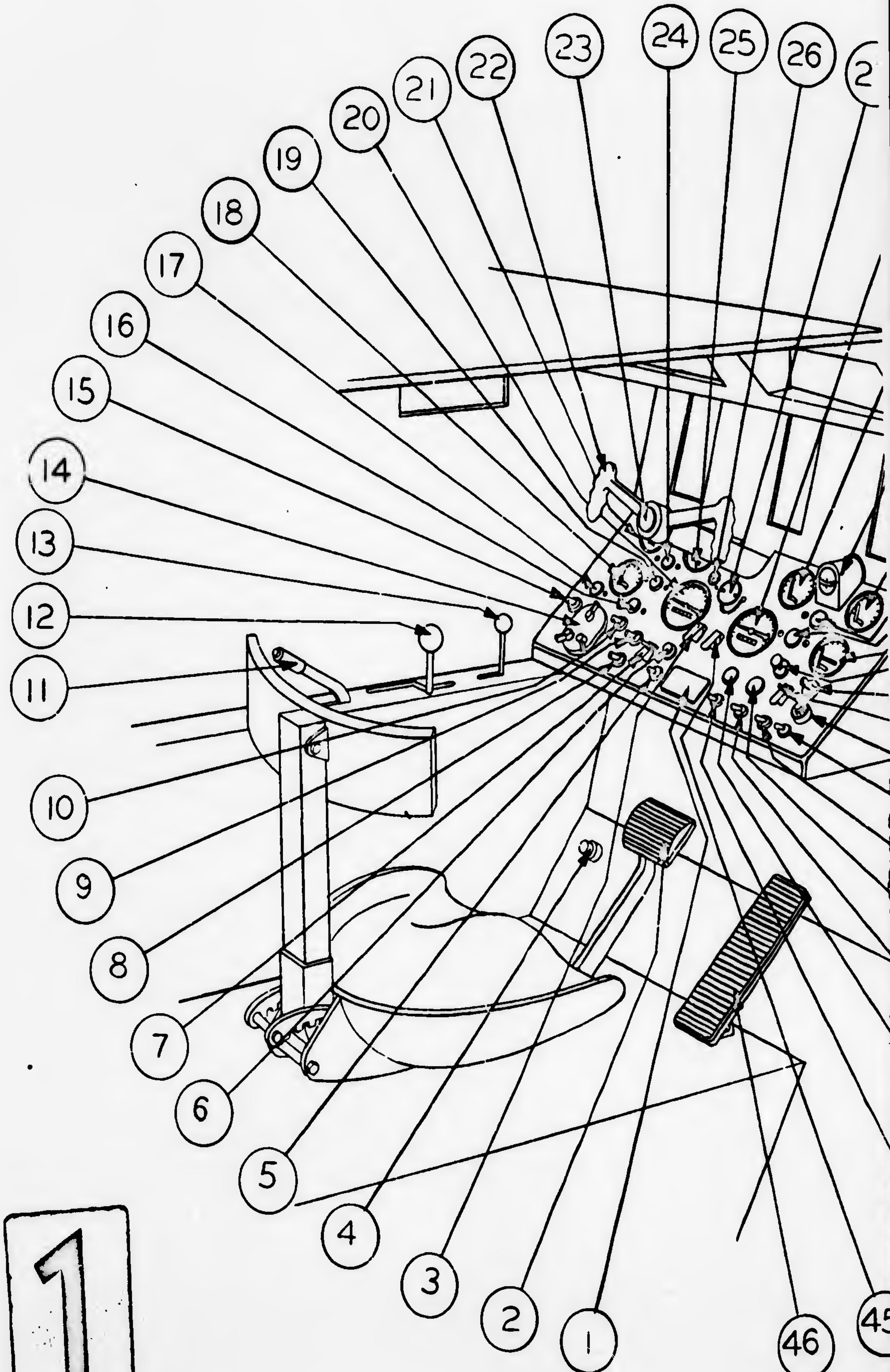
1. Heater Instruction Plate
2. Service Brake Pedal
3. Fire Extinguisher Switch (First Slot)
4. Headlamps Dimmer Switch
5. Heater Hi-Lo Switch
6. Heater Push to Test Lamp
7. Master Heater Switch
8. Blackout Selector Switch
9. Navigation Light Switch
10. Aux. Bilge Pump Switch (Personnel Compartment)
11. Primer Pump Lever
12. Transmission Range Selector and Parking Brake Lever
13. Ramp Operating Lever
14. Lighting Control Switch
15. Aux. Bilge Pump Switch (Engine Compartment)
16. Aux. Bilge Pump Indicator (Personnel Compartment)
17. Aux. Bilge Pump Indicator (Engine Compartment)
18. Battery Generator Indicator
19. Panel Light
20. Engine Tachometer and Hour Meter
21. Engine Oil Pressure Gage
22. Steering Control
23. Horn Button
24. Low Oil Pressure Warning Lamp
25. Transmission Lub. Oil Pressure Gage
26. Hi Beam Indicator
27. Utility Outlet
28. Speedometer and Odometer
29. Engine Oil Temp. Gage
30. Compass
31. Transmission Oil Temp. Gage
32. Oil Temp. Warning Lamp
33. Panel Light
34. Fuel Level Gage
35. Hand Throttle
36. Fuel Sending Unit Switch
37. Fuel Pump Switch
38. Engine Starter Button
39. Fuel Shut-Off Switch
40. Personnel Ventilation Blower Switch
41. Master Switch Indicator
42. Master Switch
43. Gas-Particulate Filter Unit Indicator
44. Gas-Particulate Filter Unit Switch
45. Fire Extinguisher Switch (Second Slot)
46. Accelerator Pedal



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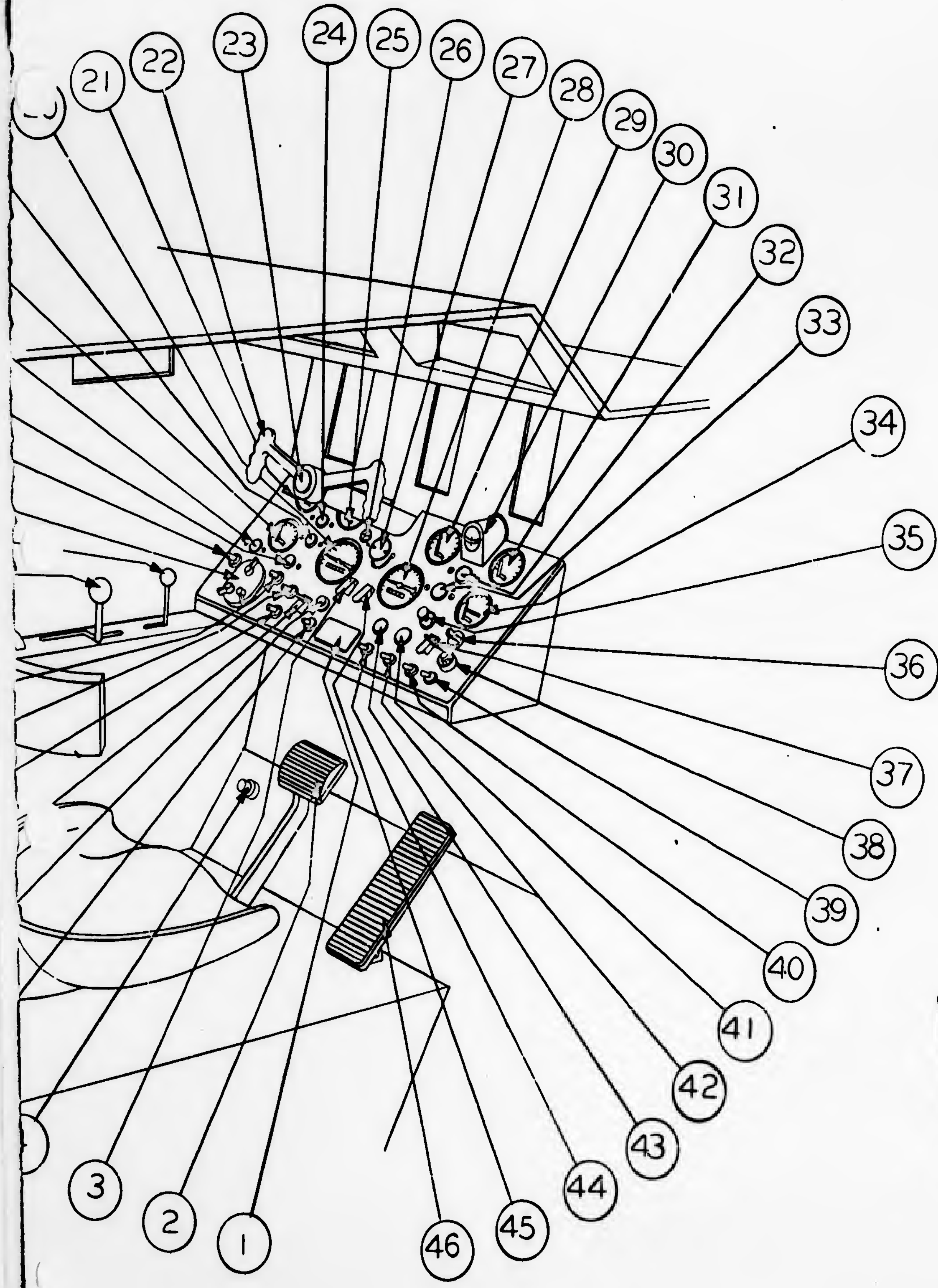

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DRIVERS STATION
TD 106539

4.7 HYDRAULIC CIRCUIT

TD-106540 (following Page 4-80) shows the schematic of components and circuitry of the vehicle hydraulic system. A 20 gallon cylindrical container located in the engine compartment is the reservoir for the system. Four hydraulic pumps mounted on a gear box attached to the transmission supply fluid directly to hydraulic motors for driving the bilge pumps. Each gear-type pump displaces approximately 0.7 cu. in. per revolution directly to a gear type fluid motor. The units are capable of operating for extended periods without malfunctioning. The units are continuously in operation whether the vehicle is on land or water.

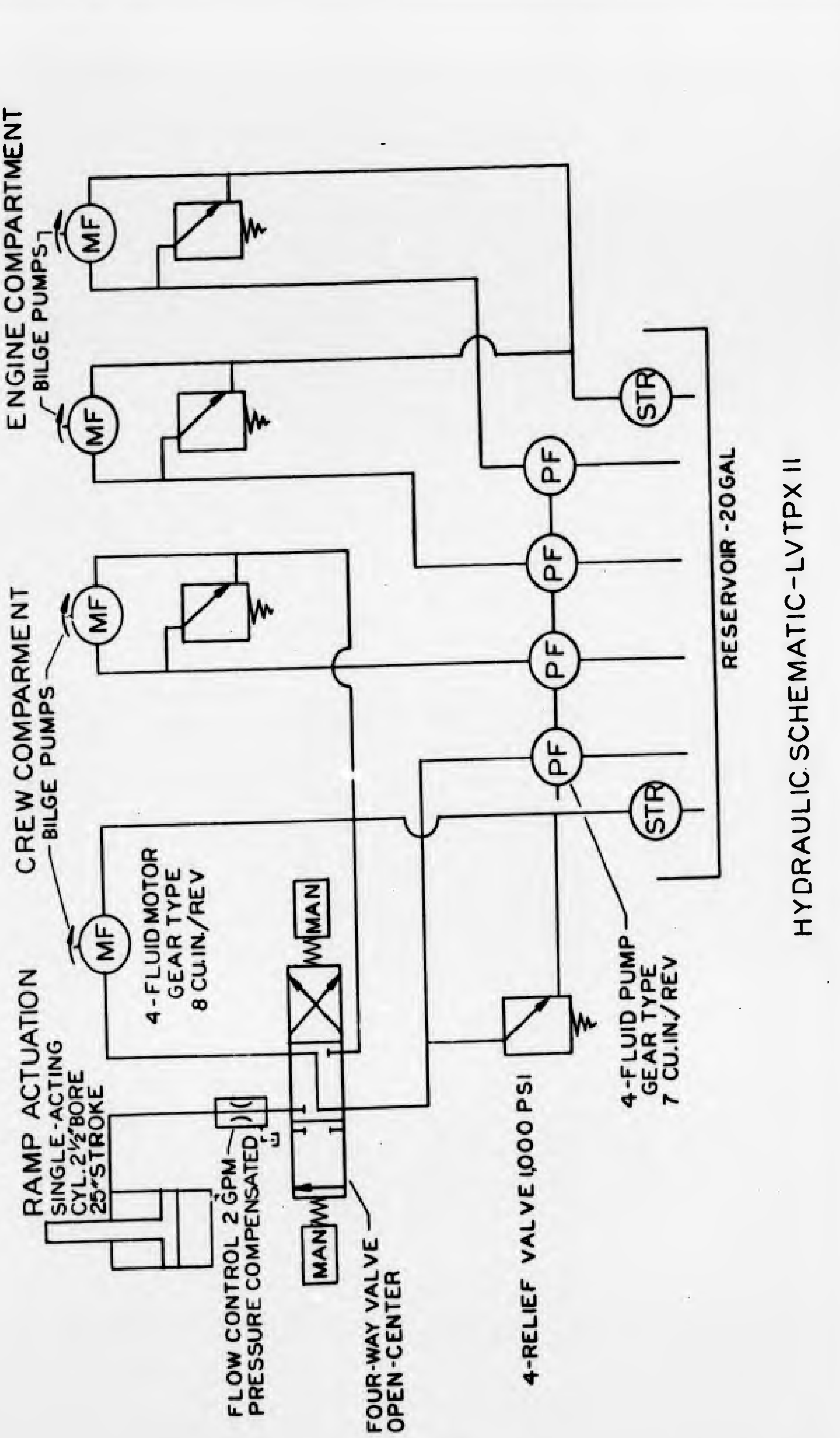
Each circuit is protected with a relief valve against overloading and excessive pressure build-up.

A set of sheaves, a cable and a single-acting cylinder comprise the ramp actuating system as shown on TD-106537 (following Page 4-67). The single-acting cylinder is employed to raise the ramp into its stowed position. Attached to the rod of the cylinder is a cable that is routed around a pair of hull mounted sheaves and anchored to the ramp. The flow of fluid into the cylinder is regulated by a flow control valve to limit the speed of actuation to 14 seconds from the open to the stowed position.

A manually-operated directional control valve, located at the driver's side is employed to control the ramp actuation. The valve is normally in the open-center position with hydraulic fluid passing directly to the bilge pump drive motor in the forward section of the cargo compartment. To raise the ramp, the valve is shifted in one direction. The fluid is then directed from the bilge motor into the flow control valve and into the ramp cylinder.

When the cylinder is retracted the attached cable raises the ramp. To lower the ramp, the valve is shifted manually in the opposite direction. The hydraulic fluid is directed to the bilge motor while the fluid in the retracted cylinder is allowed to return to the drain line. The force for lowering the ramp is provided by its own weight.

The bilge pump system has been previously discussed in Section 4.5.9.



HYDRAULIC
SYSTEM SCHEMATIC TD 106540

4.8 ELECTRICAL SYSTEM

4.8.1 DESCRIPTION

The electrical system is a 24-volt d.c. waterproof single wire, common ground type.

A schematic of the system appears on TD 106541, following Page 4-85 . The latest Military Standard Ordnance electrical components have been selected wherever possible to provide maximum compatibility with other current military vehicles. A vehicle electrical system perspective is shown on TD 106542, following Page 4-86 .

4.8.2 BATTERIES

Electrical power is provided by four MS-35000-3 dry-charged 100 amp hour 12-volt batteries connected in series parallel. They are located in a waterproof box mounted on the starboard sponson under the hydraulic reservoir. Servicing of the batteries is accomplished by means of a side opening in the box which allows the batteries to be extracted on a sliding tray. The box is vented by a tube to the forward upper area of the engine compartment.

4.8.3 ALTERNATOR

A waterproof 100 amp alternator, Ordnance Part No. 7954722 provides adequate electrical power to supply the normal vehicle electrical requirements and maintain the batteries in a charged condition. For maximum reliability the alternator is spline driven by the engine. The alternator rectifier and relay are mounted on an electrical panel adjacent to the right side of the engine.

4.8.4 INSTRUMENT PANEL

The driver's instrument panel contains all the instrumentation and electrical controls necessary to operate the vehicle. The panel is shock mounted in rubber to prevent instrument damage from track vibration and road shock. A detailed description of the instruments and controls appears in the section on the driver's station. (Section 4.6.6). Removal of the complete panel assembly for servicing is facilitated by a waterproof disconnect in the wiring harness behind the panel.

4.8.5 BLOWER AND AUXILIARY BILGE PUMP MOTORS

Two dual speed waterproof blower motors are mounted on the engine compartment bulkhead to provide ventilation for the personnel and crew compartments. The motors are energized by a double throw switch on the instrument panel.

Two electrical auxiliary bilge pumps, Ordnance Part No. 1940284, are incorporated in the vehicle to provide an emergency bilge system for the personnel and engine compartments in the event of an engine failure. Each pump has a capacity of 125 gpm and a full load current draw of 55 amp. They are controlled by individual switches with indicating lamps on the instrument panel.

4.8.6 LIGHTING

The headlamp assemblies, Ordnance Part No. 7972325, are mounted in cavities along the upper edge of the front surface of the hull. Each headlamp assembly contains a service hi-low beam, an infra-red hi-low beam, a blackout drive lamp, and a blackout marker lamp. Only the left blackout drive lamp is used, however. The headlamp assemblies are protected by brush guards extending across the cavities along the forward edge of the vehicle.

The left headlamp cavity also contains the horn MS51074.

The tail lamp assemblies are mounted in cavities on the rear surface of the hull. The left tail lamp. Ordnance Part No. 8378785, contains the service tail and stop lamps and a blackout tail lamp. The right tail lamp, Ordnance Part No. 8378786, contains a blackout tail and stop lamp. The headlamps and tail lamps are operated from a lighting control switch MS51113-1 and blackout selector switch MS39061-7 on the left side of the instrument panel. Selection of high or low beam is made by a foot operated dimmer switch. Headlamp and tail lamp assemblies are demountable and are stowed in a compartment ahead of the crew chief.

Navigation lights are provided to meet the requirements of the Motor Boat Act of 1940 for a Class 1 motor boat (16 to 26 feet in length). The stern white light is pole mounted to provide visibility in all directions for at least two miles. The combination bow light which is lower than the stern light shows green to starboard and red to port and is shielded so as to throw the light from right ahead to two points abaft the beam on their respective sides. Both lights are demountable and are stowed with the headlights when not in use.

A portable signal searchlight 80064-C3602 with a detachable color lens is provided for mounting on the top of the hull ahead of the crew chief's hatch. The searchlight is normally stowed in the compartment ahead of the crew chief. Inside dome lamps MS51073-1 are provided at the driver and assistant driver stations, two on each side of the cargo compartment, one on each side of the engine compartment, and one forward of the cupola. Each lamp assembly contains a 15 c/p white lamp, a 6 c/p red blackout lamp, and a selector switch.

The lamp assemblies are waterproof and are shock mounted to prevent damage from vibration.

4.8.7 HULL WIRING

The hull wiring consists of several taped harness assemblies providing the circuitry to the various electrical components throughout the vehicle. These can be seen on the electrical system perspective, TD 106542. All electrical connections to components and between harnesses are made with waterproof Ordnance connectors.

An overhead harness assembly supplies power to the headlights, personnel and crew compartment dome lights, bow light, searchlight, gas-particulate filter unit, and horn. The fuel pump and sending unit harness is routed beneath the flooring. The harness between the instrument panel and engine compartment is routed along the left side of the vehicle above the control rods. A disconnect is provided at the engine compartment bulkhead to facilitate harness installation and provide a water tight seal. All engine and transmission wiring passes through the engine disconnect which allows these units to be removed from the vehicle without disturbing power train or hull wiring.

Auxiliary electrical outlets are provided on the instrument panel and on both sides of the engine compartment near the dome lamps. A slave receptacle is located near the batteries to permit starting the vehicle from a remote power source or starting a disabled vehicle from the batteries. A slave cable is stowed in the stowage compartment ahead of the batteries.

4.8.8 RADIO

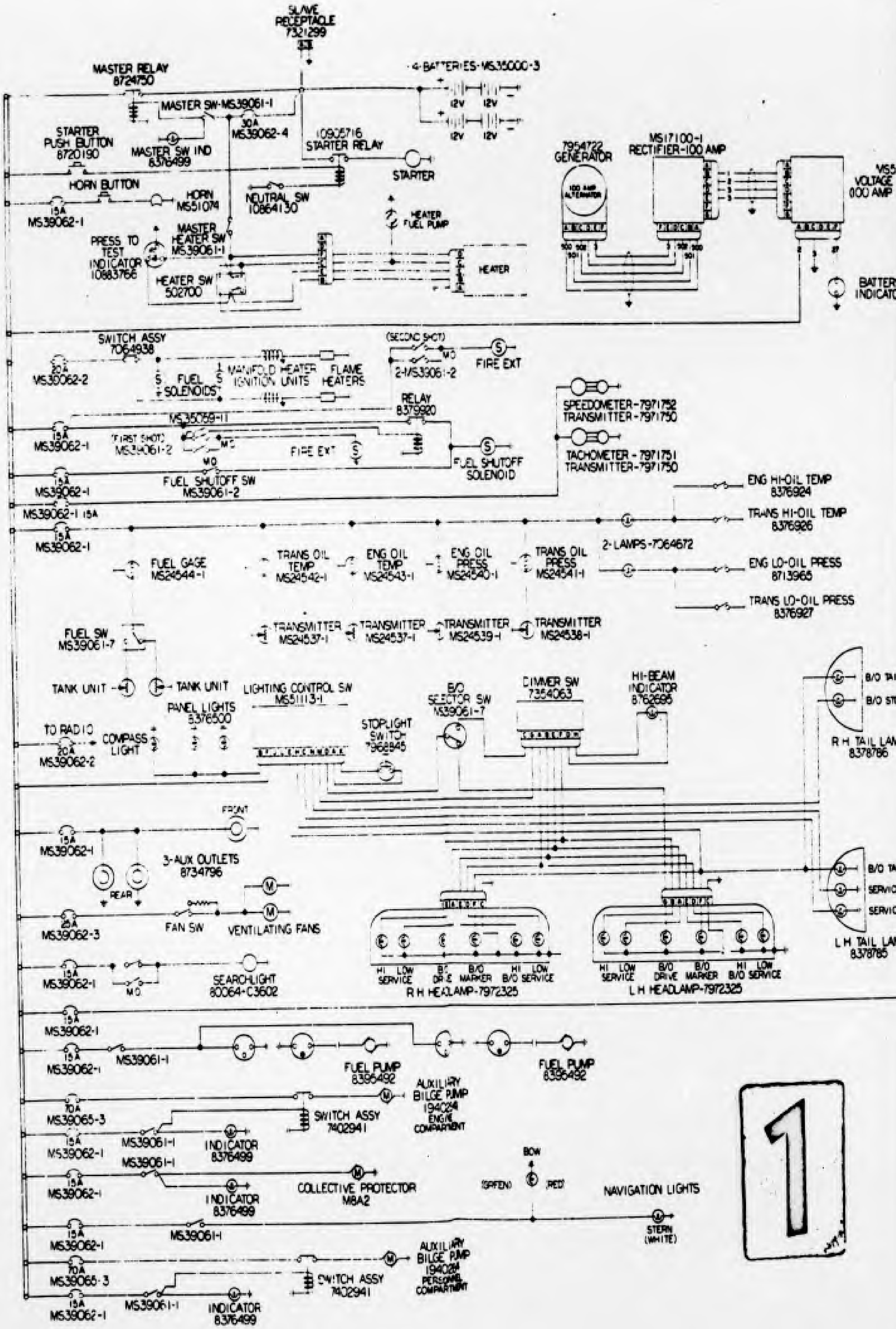
Provisions have been made for permanent installation, in a waterproof box, of radio receiver-transmitter AN/PRC 47. The radio installed in the box is good for ambient humidity range of zero to one hundred per cent relative and an altitude range of from sea level to 10,000 feet. The exterior dimensions of the waterproof box are 21-11/16 inches long, 13-1/4 inches high, and 7-7/16 inches wide. The weight of the receiver-transmitter radio is 42.5 pounds.

The vehicular installation of this receiver-transmitter radio uses the vehicle battery (26.5 volts) for primary power to the set. This set is also capable of operation from an external 115 volt, single phase, 400 cps power source.

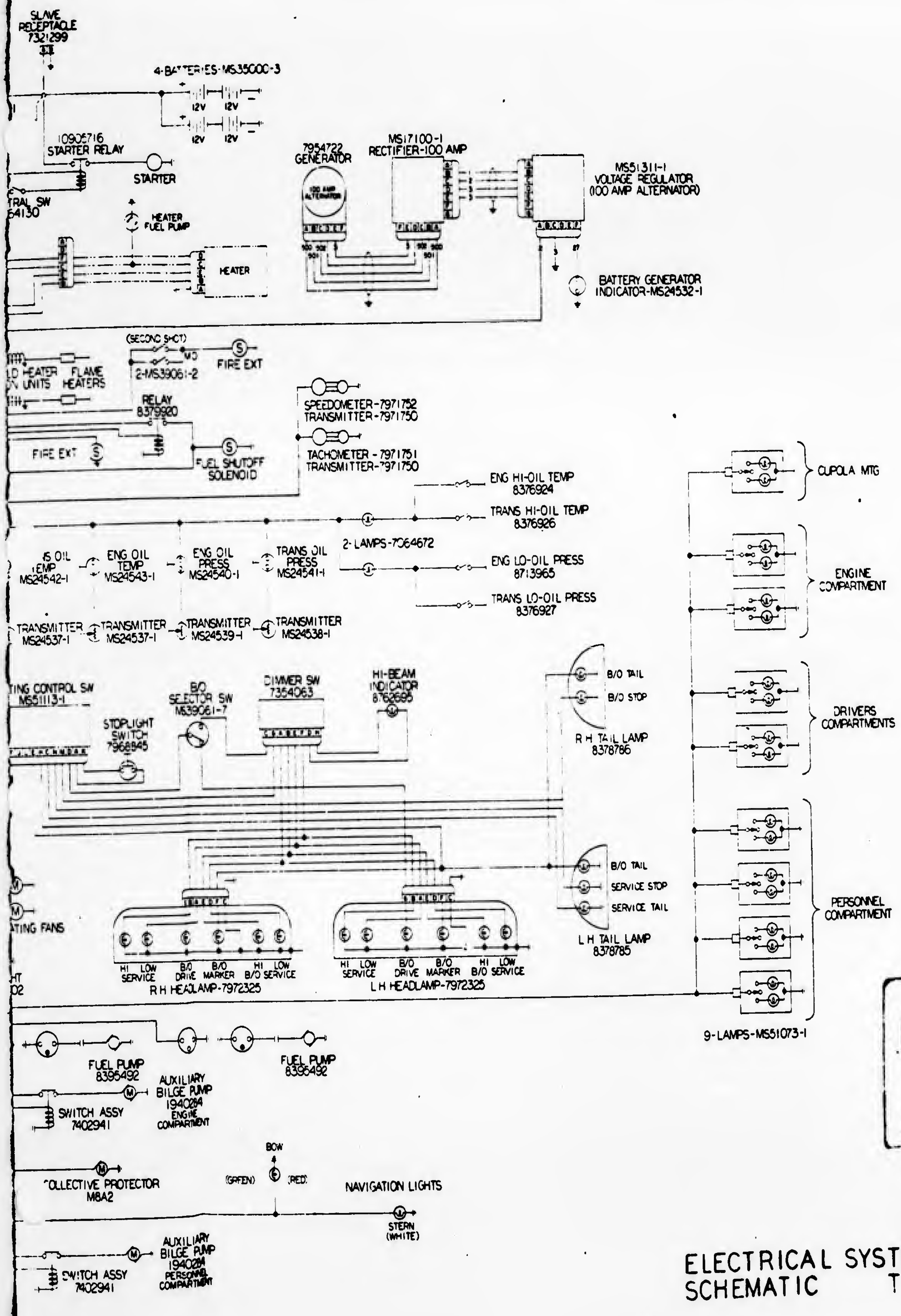
The installations that may require voice, CW, or FSK operation will use the following items:

Voice Operation	Handset H-33E/PT, or a combination of Headset Microphone H-63/U and Chest Set Group AN/GSA-6 can be attached to the receiver-transmitter.
CW Operation	A combination of Headset H-70C (modified) and Telegraph Key J-45 attached to audio cable, which, in turn, is connected to the audio connector.
Remote Operation	Control Group AN/GRA-6 connected to the audio connector. The small metal panel protecting the FSK terminals on the control panel of the receiver-transmitter is removed for FSK operation, and FSK Unit CV-768 is connected to the terminals. Control Group AN/GRA-6 and FSK Unit CV-786 are not supplied with the system.

The radio set may be operated with three different antennas, whip antenna, long wire antenna or vehicular antenna. The whip antenna has a height of 180 inches.



SLAVE RECEPTACLE 7321299
MASTER RELAY 8724750
4-BATTERIES-MS35000-3
12V
MASTER SW-MS39061-1
30A MS39062-4
STARTER PUSH BUTTON 8720190
MASTER SW IND 8376499
10905716 STARTER RELAY
STARTER
7954722 GENERATOR
MS17100-1 RECTIFIER-100 AMP
VOLTAGE
100 AMP
HORN BUTTON
HORN MS51074
15A MS39062-1
NEUTRAL SW 10864130
HEATER FUEL PUMP
PRESS TO TEST INDICATOR 10883766
MASTER HEATER SW MS39061-1
HEATER SW 502700
HEATER
BATTERY INDICATOR
SWITCH ASSY 7064938
20A MS39062-2
FUEL SOLENOIDS
MANIFOLD HEATER IGNITION UNITS
FLAME HEATERS
(SECOND SHOT)
2-MS39061-2
FIRE EXT
RELAY 8379920
MS35059-11
(FIRST SHOT) MS39061-2
MO
FUEL SHUTOFF SOLENOID
FUEL SHUTOFF SW MS39061-2
SPEEDOMETER-7971752 TRANSMITTER-7971750
TACHOMETER-7971751 TRANSMITTER-7971750
ENG HI-OIL TEMP 8376924
TRANS HI-OIL TEMP 8376926
MS39062-1 15A
2-LAMPS-7064672
FUEL GAGE MS24544-1
TRANS OIL TEMP MS24542-1
ENG OIL TEMP MS24543-1
ENG OIL PRESS MS24540-1
TRANS OIL PRESS MS24541-1
ENG LO-OIL PRESS 8713965
TRANS LO-OIL PRESS 8376927
TRANSMITTER MS24537-1
TRANSMITTER MS24537-1
TRANSMITTER MS24539-1
TRANSMITTER MS24538-1
FUEL SW MS39061-7
TANK UNIT
TANK UNIT
LIGHTING CONTROL SW MS51113-1
B/O SELECTOR SW MS39061-7
DIMMER SW 7354063
HI-BEAM INDICATOR 8762695
PANEL LIGHTS 8376500
TO RADIO
COMPASS LIGHT
20A MS39062-2
STOPLIGHT SWITCH 7968845
R H TAIL LAMP 8378786
FRONT
3-AUX OUTLETS 8734796
REAR
25A MS39062-3
FAN SW
VENTILATING FANS
HI LOW SERVICE
B/O DRIVE
B/O MARKER
HI B/O
LOW SERVICE
R H HEADLAMP-7972325
L H HEADLAMP-7972325
L H TAIL LAMP 8378785
SEARCHLIGHT 80064-C3602
MS39061-1
FUEL PUMP 8396492
FUEL PUMP 8396492
70A MS39065-3
SWITCH ASSY 7402941
AUXILIARY BILGE PUMP 194024 ENGINE COMPARTMENT
INDICATOR 8376499
COLLECTIVE PROTECTOR M8A2
BOW
(GREEN)
(RED)
NAVIGATION LIGHTS
STERN (WHITE)
AUXILIARY BILGE PUMP 194024 PERSONNEL COMPARTMENT
1



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ELECTRICAL SYSTEM SCHEMATIC TD 106541

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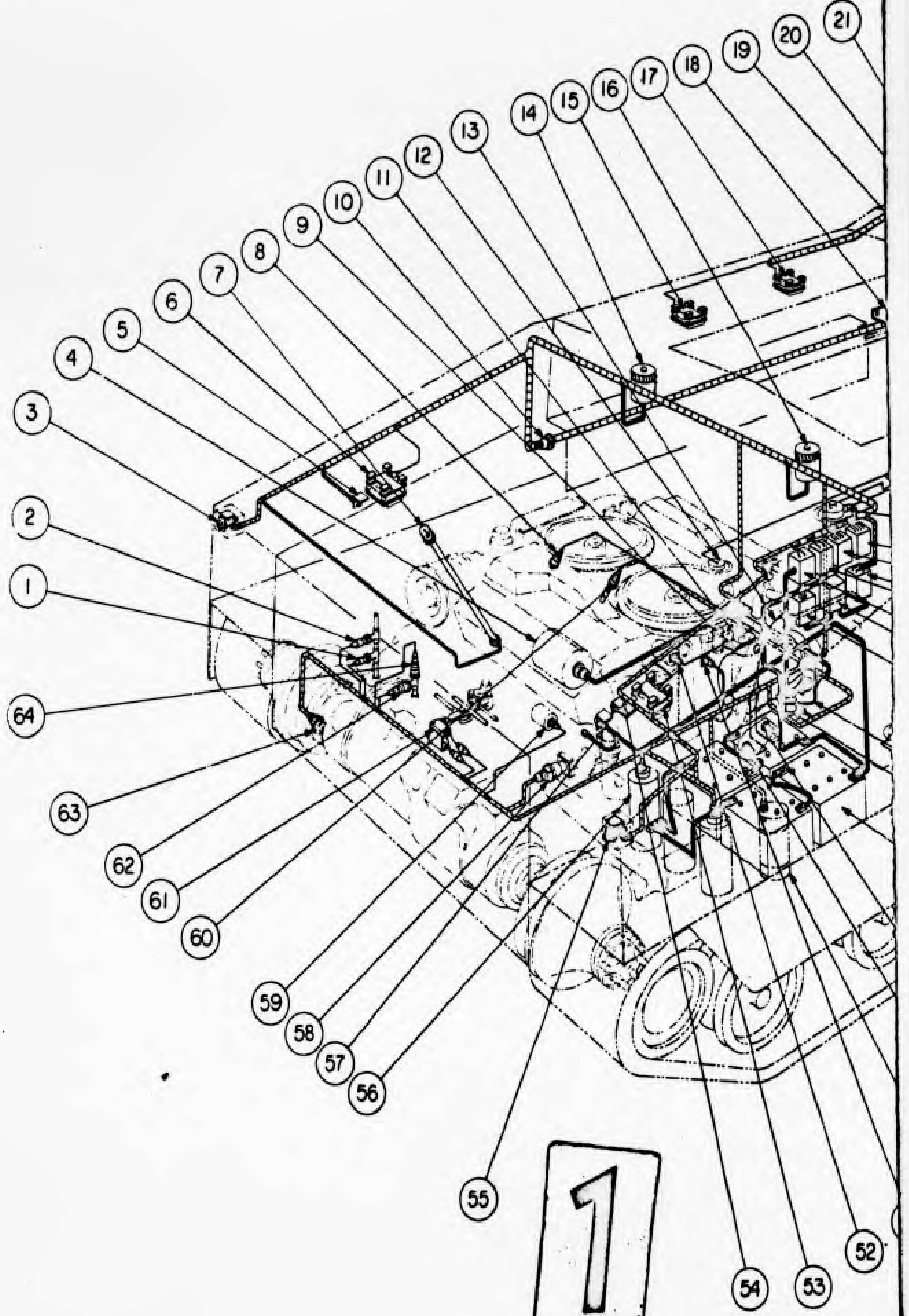
1. Transmission Oil Temp. Transmitter
2. Transmission Hi-Oil Temp. Switch
3. Blackout Tail, Service Stop and Service Tail Lamps
4. Starter Motor and Solenoid
5. Utility Outlet
6. Dome Lamp (Engine Compartment)
7. Stern Light
8. Engine Oil Press Transmitter
9. Fuel Solenoid Manifold Heater
10. Bulkhead Disconnect
11. Auxiliary Bilge Pump - Personnel Compartment
12. Fuel Tank Sending Unit (L.H.)
13. 100 Amp. Alternator
14. Ventilating Blower Motor
15. Dome Lamp (Personnel Compartment)
16. Ventilating Blower Motor
17. Dome Lamp (Personnel Compartment)
18. Stoplight Switch and Mounting Bracket
19. Primer Pump
20. Headlamp Dimmer Switch
21. Dome Lamp (Driver's Compartment)
22. Horn
23. Hi-Low Service, Blackout Drive, Blackout Marker and Hi-Low Infrared Headlamp L.H.
24. External (First and Second Shot) Fire Extinguisher Switches
25. Horn Button
26. Instrument Panel (See TD 106539)
27. Dome Lamp (Cupola Mtg.)
28. Bow Light
29. Dome Lamp (Personnel Compartment)
30. Searchlight
31. Dome Lamp (Assistant Driver's Compartment)
32. Hi-Low Service Blackout Marker and Hi-Low Infrared Headlamp (R.H.)
33. Gas-Particulate Filter Unit (Collective Protector)
34. Dome Lamp (Personnel Compartment)
35. Fuel Pump (L.H. Tank)
36. Bilge Pump Relay (Personnel Compartment)
37. Bilge Pump Relay (Engine Compartment)
38. Fire Extinguisher Relay
39. Starter Relay
40. Fuel Tank Sending Unit (R.H.)
41. Master Relay
42. Rectifier (100 Amp)
43. Voltage Regulator
44. Fuel Pump (R.H.Tank)
45. Engine Disconnect
46. Fuel Solenoid, Manifold Heater
47. Batteries
48. Engine Hi-Oil Temp. Switch
49. Engine Oil Temp. Transmitter
50. Auxiliary Bilge Pump (Engine Compartment)
51. Fuel Shutoff Connector
52. Ignition Unit, Manifold Heater
53. Dome Lamp (Engine Compartment)
54. Utility Outlet
55. Blackout Stop & Blackout Tail Lamp (R.H.)
56. Fire Extinguishers
57. Slave Receptacle
58. Tachometer Transmitter
59. Ignition Unit, Manifold Heater
60. Engine Lo-Oil Pressure Switch
61. Neutral Switch
62. Transmission Lo-Oil Pressure Switch
63. Speedometer Transmitter
64. Transmission Oil Pressure Transmitter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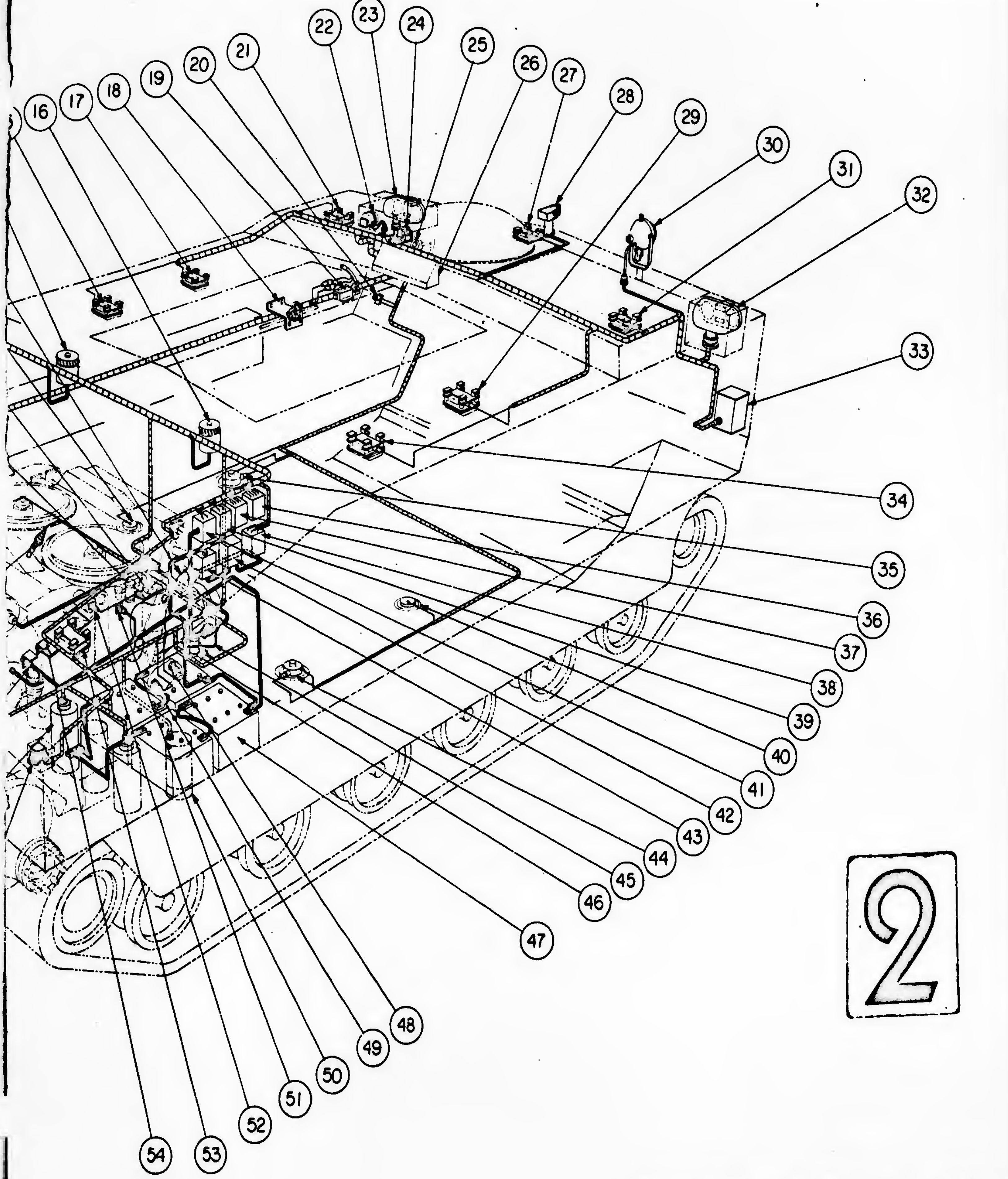


ELECTRICAL SYSTEM TD 10654Z

4.9 FIRE EXTINGUISHING SYSTEM

With the incorporation of a multifuel type engine for the LVTPX 11 vehicle it was necessary to provide adequate fire protection for the most volatile of the usable fuels, gasoline. The system proposed is therefore appropriate for all usable fuels and is based on previous combat vehicle installations, tests, and actual use with air cooled type engines.

4.9.1 FIXED FIRE EXTINGUISHER SYSTEM

A fixed fire extinguisher system is incorporated into the engine compartment, as shown in Drawing No. TD 106543, following Page 4-39. This system contains three 10 lb. bottles mounted above the starboard sponson and are connected into a so-called "two shot system." That is, if a fire occurs, the first shot is fired which releases one bottle. If the fire is not extinguished by the one bottle or flashback occurs, then the second shot is fired. This releases the remaining two bottles. Also, if the fire is extinguished by the first shot, then the second shot provides a reserve. This type of system has proven to provide the maximum effectiveness for extinguishing all types of engine compartment fires while remaining relatively simple.

A description of the complete two shot system notes that the three 10 lb. bottles are the standard military version meeting all safety and performance specifications. Each bottle is fitted with a flood valve which when actuated opens to allow discharge of the bottle into the discharge line. The flood valves of the first shot bottle and one of the second shot bottles are fitted with solenoid type control heads. These solenoid heads allow actuation of the flood valves electrically.

The second shot bottles are tied together so that release of the first bottle actuates the release of the second at the same time. This is accomplished by placing an "interconnector" on the first bottle between the solenoid valve and flood valve with a line connection between this interconnector and the pressure head of the second bottle.

A momentarily-on switch with guard assembly is provided for actuating each solenoid control head. (Reference Drawing No. TD 106541 Electrical Schematic). That is, two switches, one for each shot, are located on the instrument panel. Two more switches which serve the same purpose are located on the vehicle outside in the port headlight well. The first shot switch on the instrument panel and only this switch isalso connected to an engine fuel shut-off solenoid valve which shuts down the engine. The purpose of shutting down the engine is to cut out forced air flow into the engine compartment. This action is not provided in the first shot switch on the outside of the vehicle as it would present an easy method for the enemy to disable the vehicle. The second shot switches, also do not have the engine shut down provision, so they are wired in parallel.

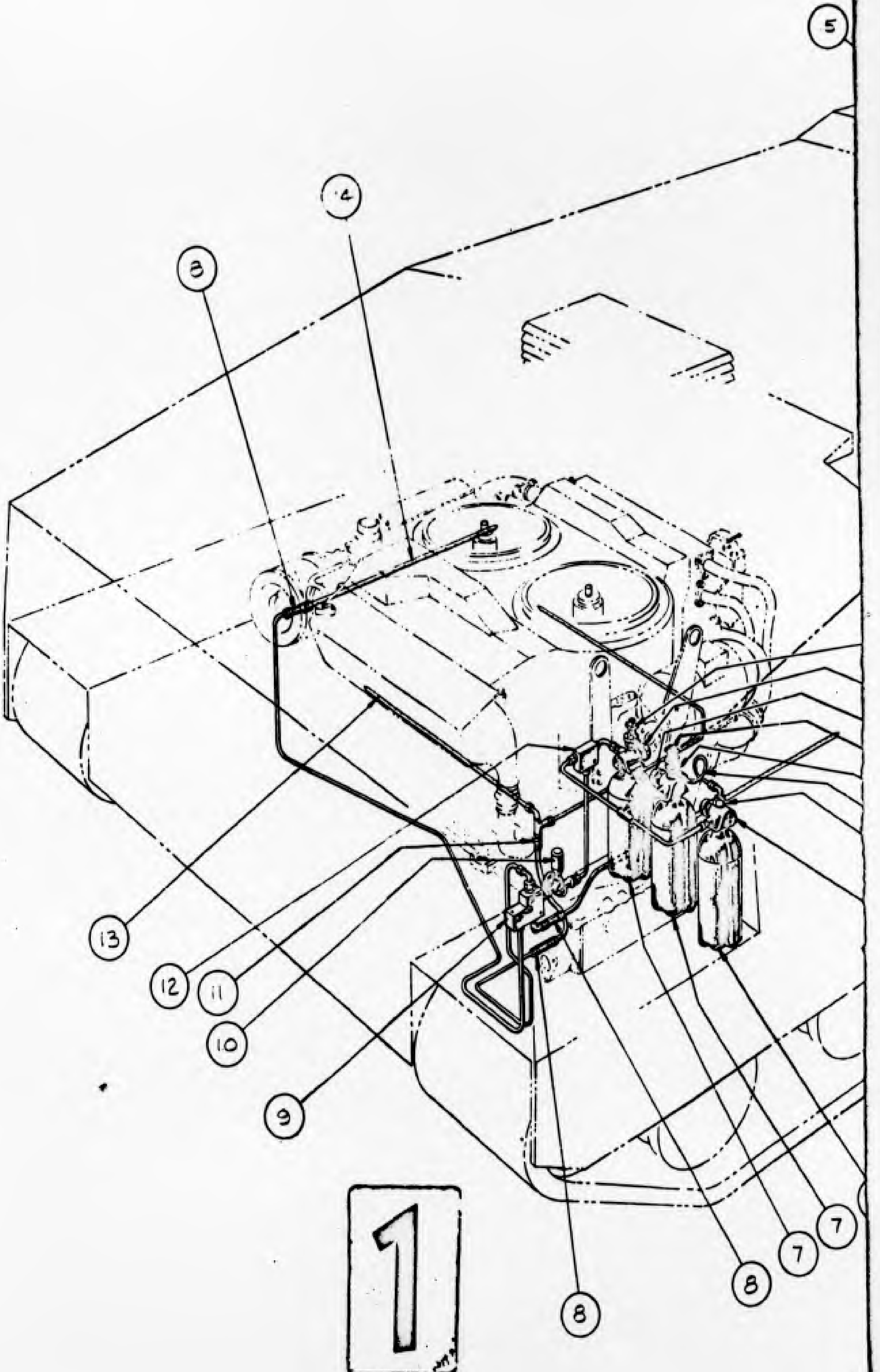
Actuation of either first shot switch results in CO_2 discharge from the respective bottle. The CO_2 first travels to a double check valve which serves the purpose of allowing CO_2 discharge from both the first and second shots to be further routed in the same lines. Discharge lines from the two second shot bottles are brought together into a tee and then a single line continues on to the same double check valve. From the double check valve the CO_2 passes on to a "discharge delay assembly; then to a manifold assembly and then into four perforated discharge line which releases the CO_2 into the engine compartment.

The discharge delay assembly is a pressure actuated mechanism which delays further routing of the CO_2 for approximately 7 seconds. This delay action allows the engine to die down before delivery of the CO_2 to the engine compartment. Normal procedure when fire occurs is for the operator to first cut off the engine and then actuate the fire extinguisher system. This delay mechanism allows the operator to do both in quick sequence. The manifold provides a means of distributing the CO_2 equally to the four discharge lines. The four perforated discharge lines are located as shown on the drawing with one on either sponson and one on either side of the engine. Design of the perforated tubing would be in accordance with the extensive test report PP1 649 "Study of Flow Phenomena of Carbon Dioxide through Distribution Systems."

Another safety feature which can be incorporated is a mechanical actuation mechanism located at the bottles. Its purpose would be to allow immediate actuation of the fire extinguishers during maintenance within the engine compartment.

4.9.2 PORTABLE FIRE EXTINGUISHER

The troop compartment will contain one portable 5 lb. CO_2 fire extinguisher. It is of standard military type and is to be mounted on the wall to the right of the assistant driver's position for maximum accessibility.



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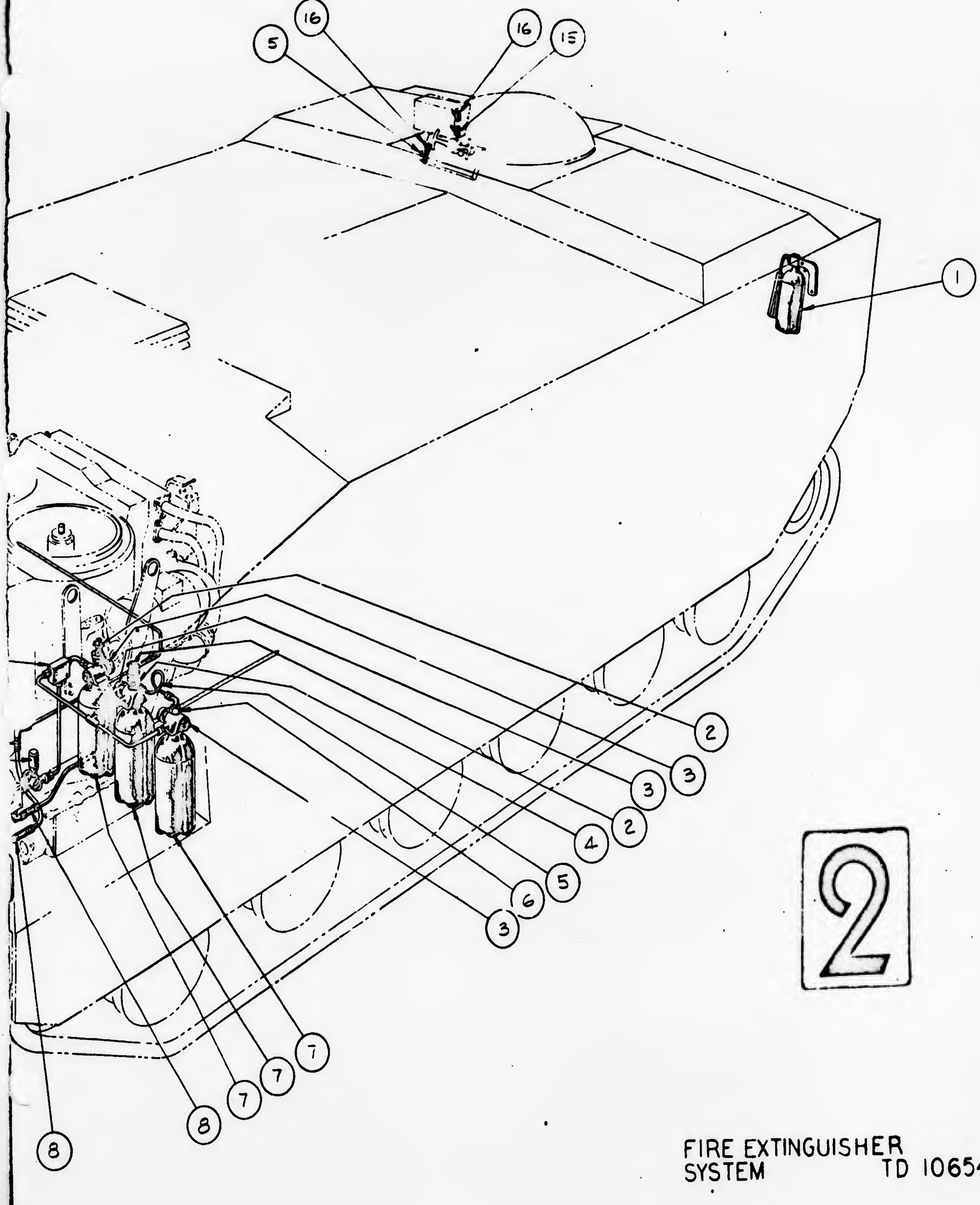

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FIRE EXTINGUISHER SYSTEM TD 106543

4.10 ARMAMENT

The Tracked Amphibian personnel and cargo carrier will mount a G-1 type cupola without a hatch in the forward section of the vehicle. The cupola is light armor steel, equipped with wide-angle vision blocks to provide the gunner with a 360° field of vision capability.

The modified G-1 cupola mount will carry a 7.62 mm machine gun, located to the right of the cupola centerline. An M28C Periscope will be used to sight the machine gun on target. The M28C Periscope is a 1-1/2 power, forty-eight degree field-of-view instrument, capable of directing the line of sight from fifteen degrees depression to sixty degrees elevation. The periscope is mechanically linked to the machine gun.

The G-1 cupola mount is presently being modified by a Bureau of Ships contractor to receive the 7.62 mm machine gun and this mount is a self-contained watertight unit. There are currently two versions of this mount. They carry a standard 250-round ammunition box, or a curved 500-round fabricated box, and also are available with and without an armored hatch cover.

A new cupola of lightweight armor could be designed, but if small arms protection is required, the weight saving would be insignificant and the volume inside the cupola would be reduced if the exterior dimensions were retained.

5. CENTER OF GRAVITY

The total vehicle weight and location of the center of gravity were determined for both the loaded and unloaded condition, (see Figure 5-1). The calculations were based on a tabulation of the weight and coordinates of the center of gravity of each component in the vehicle. Wherever possible, actual weights have been made. If actual weights were not available, an accurate calculation or estimate was made. In placing components in the vehicle, care was taken to keep the transverse center of gravity on the centerline of the vehicle.

A summary of the weight and C.G. analysis appears on the following pages. The following coordinate system was employed:

X = Longitudinal distance from front of vehicle.

Y = Transverse distance from extreme left sides of vehicle.

Z = Vertical distance from ground line of loaded vehicle.

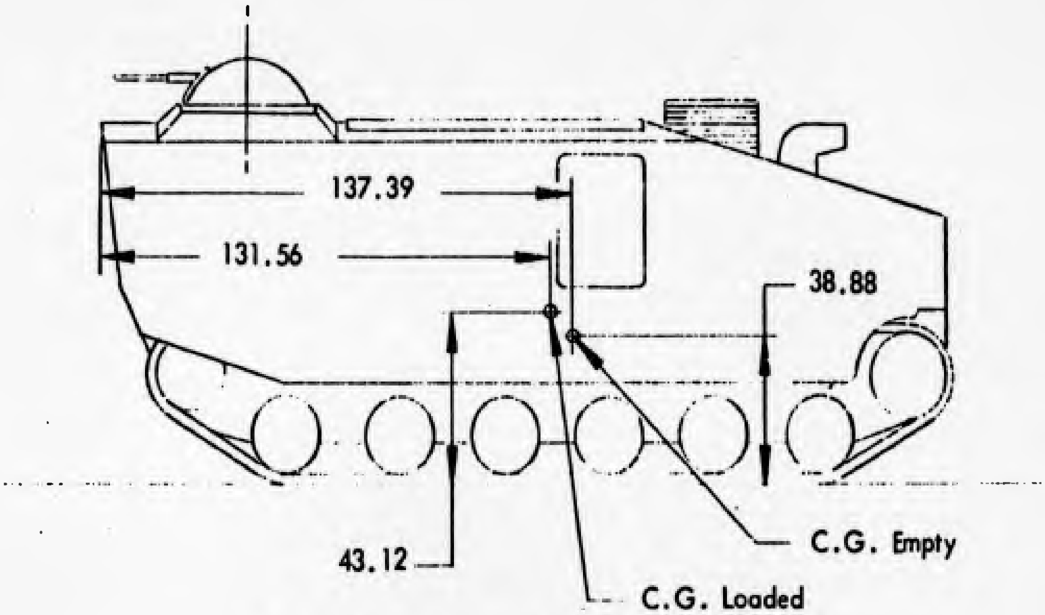


Figure 5-1 Center of Gravity Location

WEIGHT & C.G. DATA LVTPXII

SUMMARY

DATE 10-12-61
PREPARED BY J. Stickney

PART NO.	DESCRIPTION	UNIT WT. CALC.	UNIT WT. ACT.	QUAN. REQ'D.	TOTAL WT.	$\bar{X}$ -IN.-	W $\bar{X}$	$\bar{Y}$ -IN.-	W $\bar{Y}$	$\bar{Z}$ -IN.-	W $\bar{Z}$
	Hull				8630.1		944,950		549,038		467,667
	Power Train				5848.3		1,245,973		326,742		149,326
	Suspension				5618.0		615,108		353,934		121,065
	Fuel System and Fuel				1454.3		182,985		89,870		29,749
	Turret				441.0		18,963		27,783		45,423
	Hydraulic System				715.8		103,726		56,026		33,063
	Electrical System				653.6		97,252		57,709		34,150
	Fire Extinguisher System				172.1		32,329		18,518		8,820
	Air Induction System				368.0		69,906		15,939		22,862
	Mechanical Controls				104.0		11,246		2,141		5,217
	Gas Particulate Filter Unit				47.8		623		4,813		3,256
	Crew				350.0		12,600		22,400		24,500
	Ammo-1000 Rounds Belted				59.0		10,148		6,726		2,891
	Ground Mount for Gun				20.0		3,440		2,280		980
	Towing Cable				50.0		12,300		4,200		2,200
	TOTALS										

CONTINUED

WEIGHT & C.G. DATA LVTPXII

SUMMARY (Cont'd)

DATE 10-12-61
PREPARED BY J. Stickney

PART NO.	DESCRIPTION	UNIT WT. CALC.	UNIT WT. ACT.	QUAN. REQ'D.	TOTAL WT.	$\bar{X}$ -IN.-	W $\bar{X}$	$\bar{Y}$ -IN.-	W $\bar{Y}$	$\bar{Z}$ -IN.-	W$\bar{Z}$
	Stowage				300.0		51,600		34,200		14,700
	Paint				75.0		9,225		4,725		3,975
	Suspension Lube				75.0		9,900		4,725		1,500
	Total - Unloaded				24,982	137.39	3,432,274	63.32	1,581,769	38.88	971,344
	Cargo				10,000		1,170,000		630,000		537,000
	TOTALS Loaded				34,982	131.56	4,602,274	63.23	2,211,769	43.12	1,508,344

6. VEHICLE PERFORMANCE

6.1 LAND PERFORMANCE

6.1.1 SPEED, GRADEABILITY AND RANGE

The LVTPX-11 vehicle performance requirement data has been presented in tabulated form in Part 4, which resulted in the Continental AVDS 750-1 Engine and Allison XTG 411-7 Transmission selection. To meet the 40 mph speed and 70% slope operation, tractive effort was calculated for various speeds and plotted on an estimated power package output torque curve (Figure 6-1). This tractive effort curve notes that adequate power is available to meet the 40 mph speed requirement while excess power is available at the lower speed ranges.

A second plot of estimated full-throttle horsepower curves is presented showing vehicle rolling resistance and gradeability performance (Figure 6-2). This data, shows that a 35,000 lb. LVTPX-11 vehicle, using No. 2 diesel fuel will meet all the performance specifications.

A point of significance is the effect fuel density has on engine performance, since compression ignition injection systems supply fuel on a volume basis. This pertains to fuel heat content, injection output and thermal efficiency. Using No. 2 diesel as a standard, there would be a reduction of fuel heat content of approximately 9%, a reduction of injector output of approximately 5% and reduced thermal efficiency of approximately 3% when going to regular combat gasoline in a multifuel engine. Use of any of the other fuels between Diesel No. 2 and gasoline results in a proportional

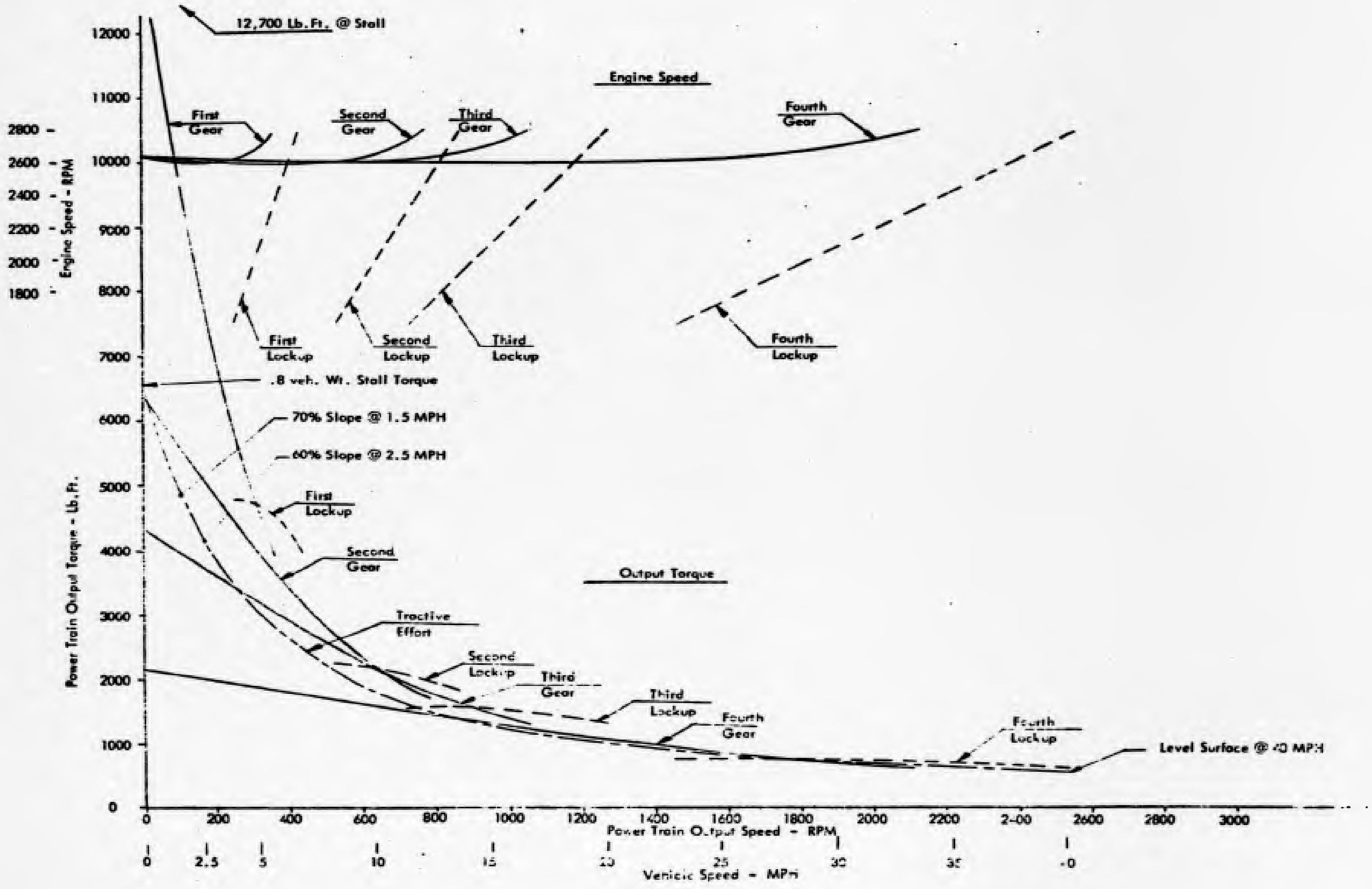


Figure 6-1 Estimated Full Throttle Performance of a Continental AVDS-750 Engine and an Allison XTB-411-7 Power Train and LVTPX-11 Tractive Effort Curve

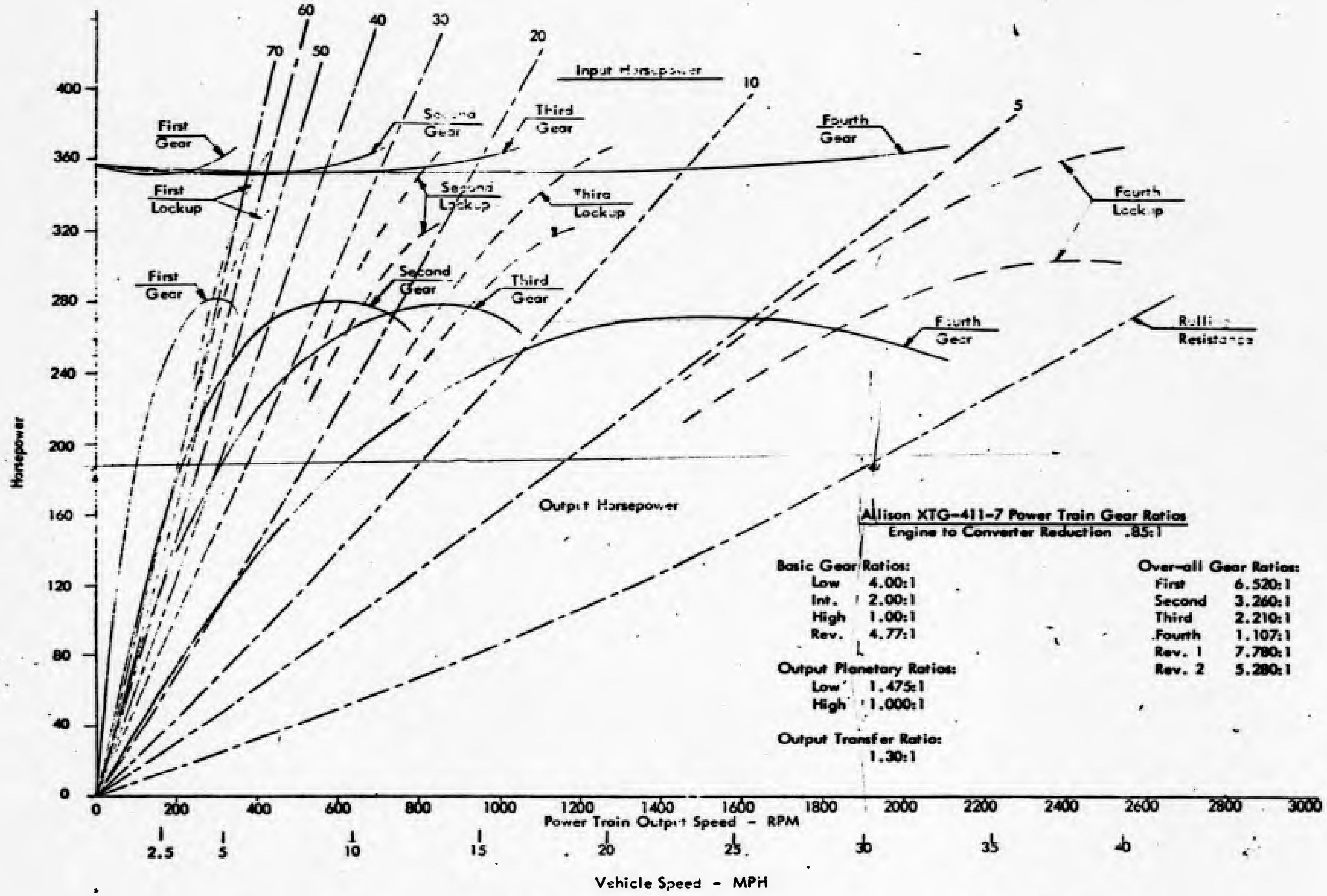


Figure 6-2 Estimated Full Throttle Performance of a Continental ADVS-750 Engine and an Allison XTG-411-7 Power Train and Gradeability Curves

lesser reduction in the three above items. For this reason it is necessary that an engine be selected which provides more than marginal power output.

Preliminary calculations of land and water range fuel requirements were performed which showed that the maximum amount of fuel was required to meet the 50 miles of water operation at the maximum water speed. Reference to Section 6.2 on water operation has shown this in number of hours at maximum engine output, along with 3/4 and 1/2 engine power ratings. The 180 gallons of usable fuel will permit 7.7 hours of vehicle operation at 6.5 miles per hour. This meets the specified 50 mile minimum range at maximum vehicle speed in calm water. If the same amount of fuel is utilized at 1/2 engine power rating, the vehicle speed would drop to approximately 5 mph with a duration of 20 hours, which results in a range of over 100 miles.

The specification requests 250 miles of land operation at 25 mph. The rolling resistance at 25 mph on hard level surface is 146.0 hp at the sprocket.

$$\text{Transmission hp} = \frac{146 \text{ sprocket hp}}{.96 \text{ final drive-off}} = 152.0$$

At 25 mph, transmission speed = 1560 rpm.

Transmission efficiency at 1560 output rpm and 4th gear lock-up

$$= \frac{228 \text{ hp output}}{254 \text{ hp input}} = 89.8\%$$

$$\text{Engine hp} = \frac{152}{.90} = 169.5$$

Engine speed would be 1720 rpm. The Brake Specific Fuel Consumption at 1720 engine rpm and this load is 0.44.

$$169.5 \text{ HP} \times .44 \text{ lb/BHP-Hr} \times \frac{1}{7 \text{ lb/gal}} \times \frac{250 \text{ miles}}{25 \text{ miles/hr}} = 107.2 \text{ gals.}$$

Since this fuel quantity is less than the 180 gallons available, a new land cruising range was calculated based on the 180 usable gallons. This results in a land cruising range at 25 mph of 420 miles.

6.1.2 VEHICLE MOBILITY

This tracked amphibian must be capable of operating over various types of terrain, such as beach and desert sand, coral reefs, shoals, gravel, rocks, rice paddies, snow, ice, swamps, tunda, muskeg, and through forested areas, with trees up to three inches in diameter.

The mobility of the LVTPX -11 was computed based on the method of the Land Locomotion Research Branch, Research and Development Division, Ordnance Tank Automotive Command, Detroit, Michigan. The vehicles chosen for comparison purposes were the LVT 3, LVT 4, LVTP 5 and LVTP 6. The soil used for the mobility comparison calculations was Michigan farm soil with moisture contents of 14, 16, 18, 20, 22, and 24 per cent.

The values used for each of the vehicles are shown in the following table:

	LVT 3	LVT 4	LVTP 5	LVTP 6	LVTPX 11
Weight	45,290 lbs	32,100 lbs	87,780 lbs.	50,600 lbs.	35,000 lbs.
Length of Track	128 in.	156 in.	232 in.	140 in.	169 in.
Width of Track	12 in.	14.25 in.	20.75 in.	21 in.	20.75 in.
Ground Pressure	14.8 psi	8.5 psi	9.22 psi	7.9 psi	5.0 psi

The final design of LVTPX-11 has a track length of 154 inches which results in a ground pressure of 5.46 psi at zero penetration. This difference in values would not change the mobility relationship, since 5 psi ground is obtained with 2 1/2 inches sinkage.

The results of the mobility calculations are shown summarized on Figure 6-3. The graph shows the mobility relationship of the different vehicles. The vehicle with the greatest drawbar pull per ton of weight (DP/W) is calculated to be the most mobile of the group. The results of the calculations made, based on the LLRB method, indicate the LVTPX 11 to be the most mobile over the range of soil conditions included in this calculation.

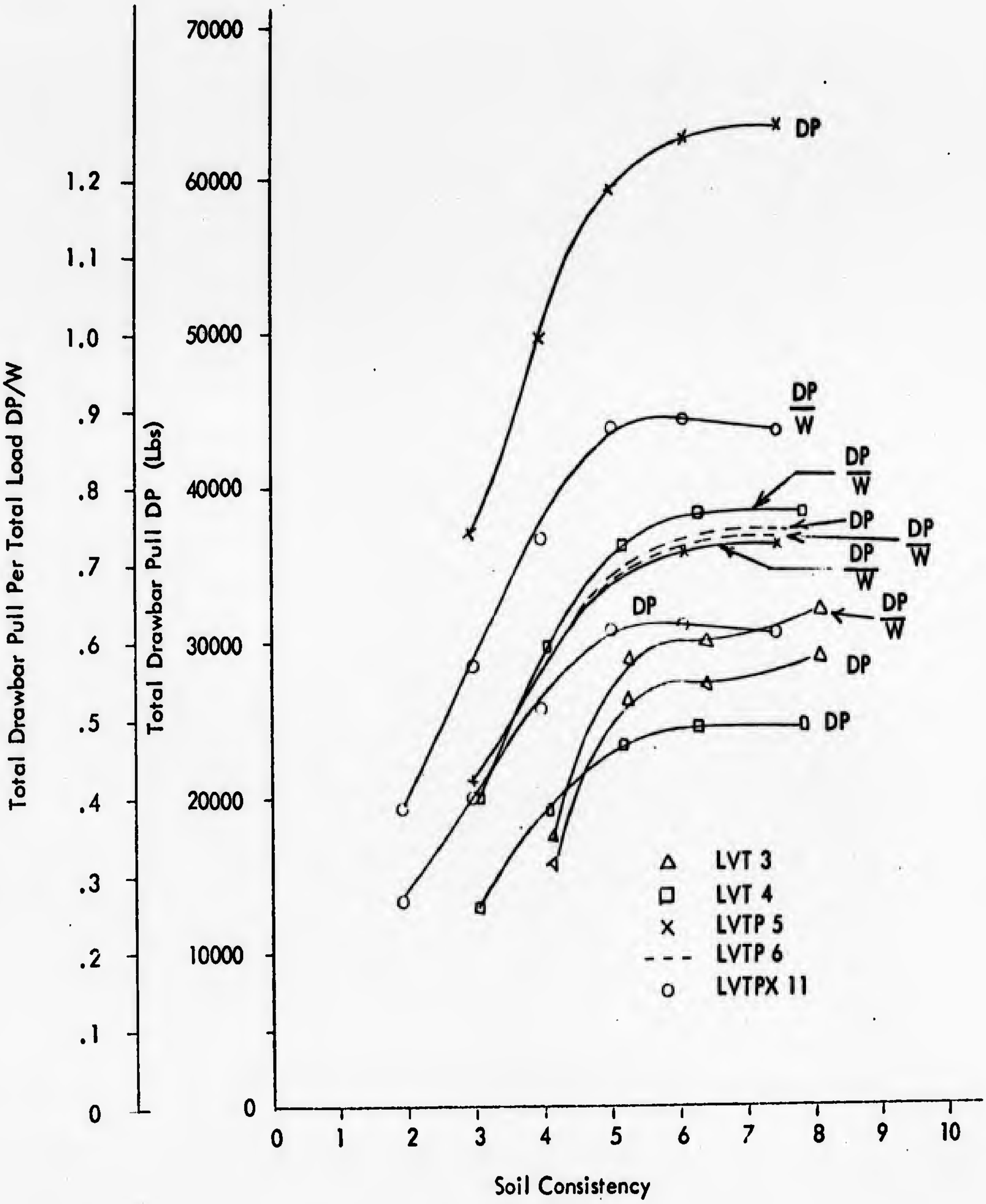
The mobility of the LVTPX 11 was further compared to the LVTP 6 for the following additional soil conditions, the values for which were obtainable from LLRB.

(1) Dry "Valclay," a pure dry bentonite clay.

(2) Dry pumice powder.

(3) Dry uniform quartz sand, smooth spherical particles.

(4) "Mazon" sand, Detroit, Michigan area; dry non-uniformly distributed rough non-spherical sand.



DP AND DP/W OF AMPHIBIANS IN MICHIGAN "FARM" SOIL

Figure 6-3 Vehicle Mobility -- Michigan "Farm" Soil

(5) Silt sand mixture from Vicksburg, Mississippi area.

(6) A disturbed loam in its natural surroundings, Ft. Knox, Kentucky.

(7) Snow; Houghton, Michigan, 8 February 1957

The mobility of the two vehicles on the first four types of soil condition show similar mobility, while calculations indicate the LVTP 11 to be the more mobile of the two under the last three sets of soil conditions. (See Figure 6-4).

The Land Locomotion Laboratory of Detroit Arsenal at this time have no soil values that can be used to compare vehicle mobility on coral reefs, shoals, rocks, rice paddies, ice, swamps, tunda, or muskeg.

6.1.3 OBSTACLE CROSSING (Figure 6-5)

The LVTPX 11 land performance characteristics stated that this vehicle must be capable of spanning an eight-foot trench and climbing a 36 inch vertical wall. Of these two special evaluations, only the trench crossing can be evaluated without very complex mathematics and uncertain assumptions.

6.1.3.1 TRENCH

The trench crossing solution was solved graphically for the center of gravity location listed on the vehicle characteristic sheet. The center of gravity location is aft of the mid-point of the vehicle, meaning that the worst condition is the passage of the rear of the vehicle over the trench. A graphical solution of this condition indicates that a maximum trench width of 98 inches can be crossed.

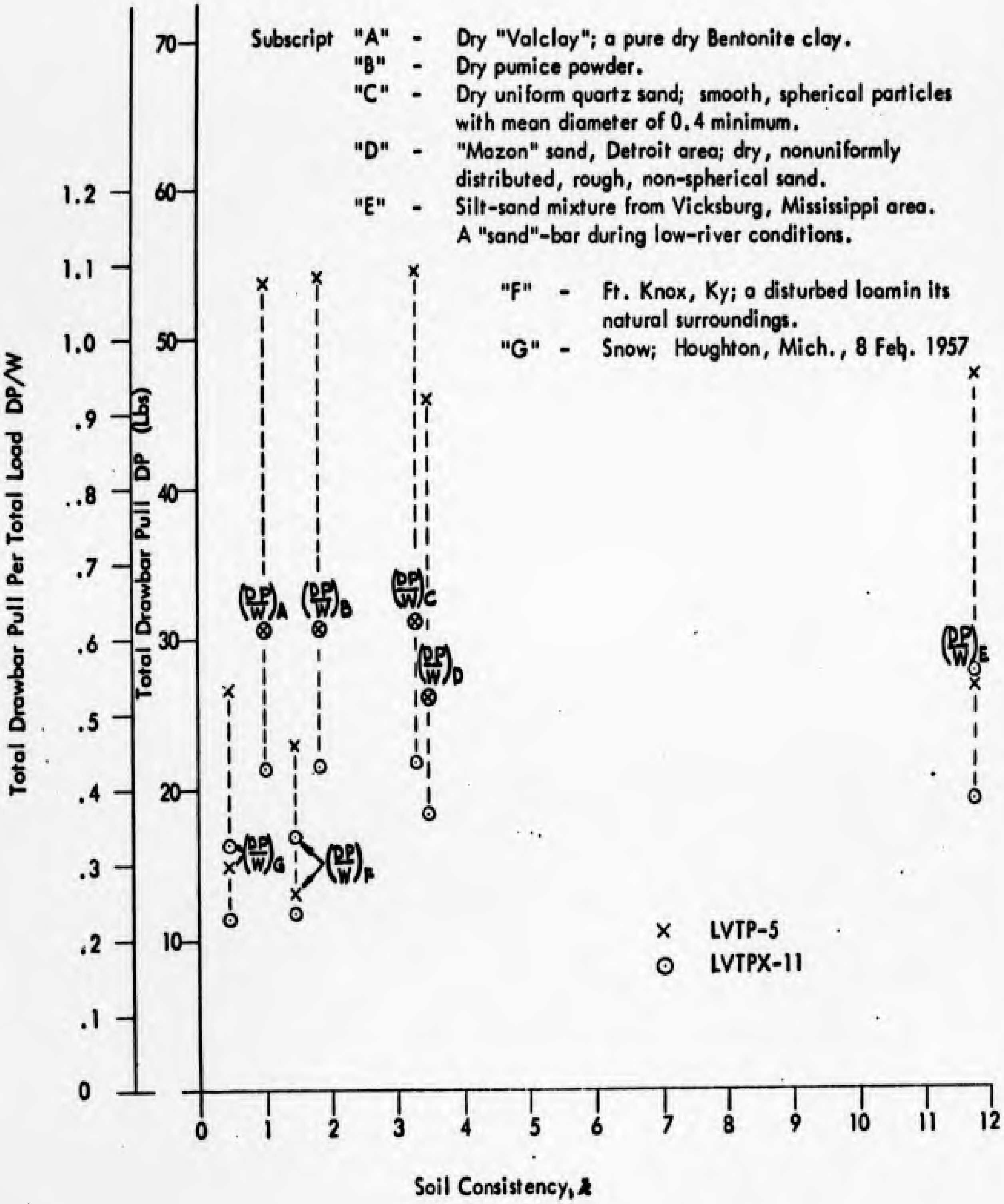
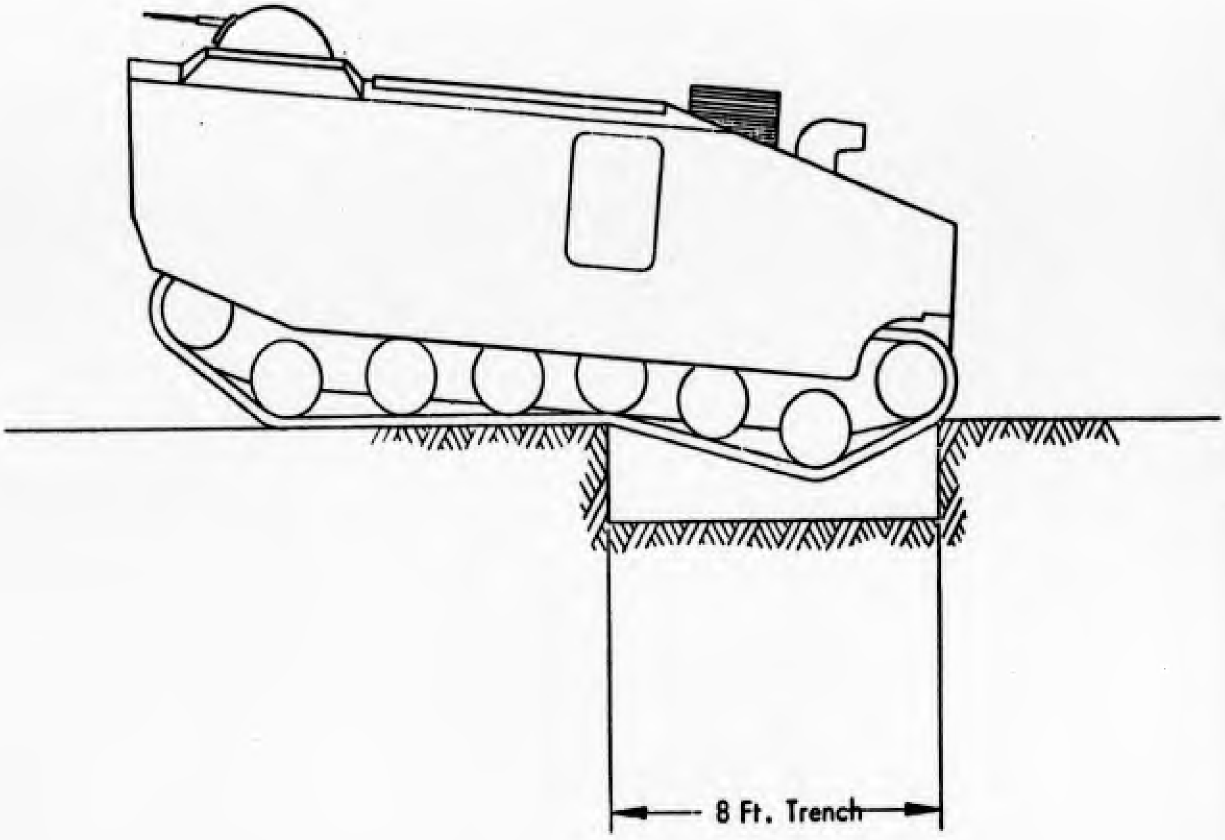
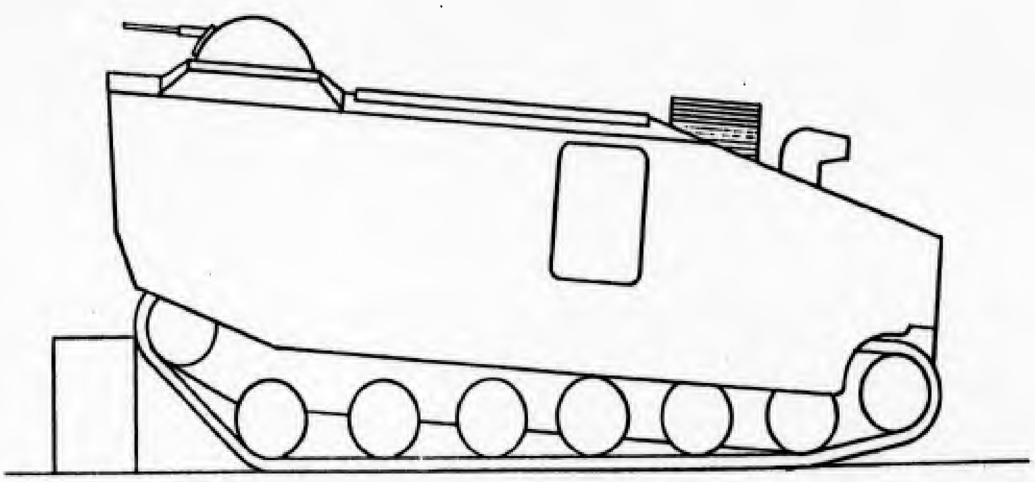


Figure 6-4 Vehicle Mobility -- Various Soils





36 Inch Vertical Wall

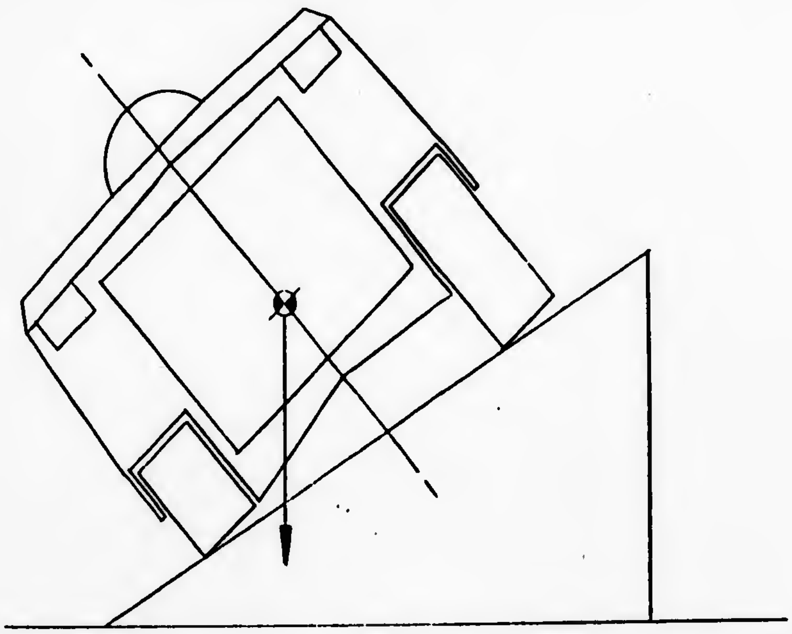
Figure 6-5 Obstacle Crossing

6.1.3.2 VERTICAL WALL

The height of the vertical wall that the LVTPX 11 must be able to climb is 36 inches. The usual check for vertical wall climbing is the height of the centerline of the sprocket or idler wheel above the ground, and if this is of greater height than the wall to be climbed, it is then assumed that the vehicle will experience no difficulty in climbing over the obstacle. However, if the vertical wall is above the sprocket or idler wheel center, the vehicle upon contact with the wall will climb, with the rear of the vehicle tending to squat, lifting the front sprocket or idler wheel center. The maximum height of the sprocket or idler wheel under this condition, without the vehicle actually climbing the wall, was plotted graphically and measured 42 inches. The LVTPX 11 will negotiate a 36-inch vertical wall, based upon this graphical solution.

6.1.4 SIDE SLOPE STABILITY (Figure 6-6)

The LVTPX-11 Tracked Amphibian must be capable of executing a 90 degree turn with a full payload on a sixty per cent side slope. The most severe condition of this requirement is when the longitudinal centerline of the vehicle is normal to the sixty per cent slope. A vertical through the vehicle's center of gravity intersects the ground between the two tracks, assuring a righting moment during such a maneuver. Assume the very worst condition that could happen, such as full road wheel deflection on the downhill side, and full rebound on the uphill side. This would allow as much as an additional fifteen degrees tilt of the vehicle's vertical axis. This would give the vehicle's vertical axis a total tilt close to forty-six degrees and reduce the vehicle's righting moment to approximately twenty per cent of the value the vehicle has when in the normal position.



Vehicle positioned on a 60 per cent side slope with downhill track in full jounce.

Figure 6-6 Side Slope Stability

6.2 WATER-PERFORMANCE

The performance of the vehicle in water is dependent on the installed power, hull hydrodynamic resistance, and the method of propulsion. The weight of the vehicle and its distribution throughout the hull affect the hydrodynamic resistance and the stability of the vessel. All of these design criteria have been considered in the selection of components and a system to satisfy the operational requirements of the vehicle.

As previously stated in Section 2.3, track propulsion was chosen for water operation as it provides the vehicle with a relatively simple system that in our opinion best meets performance requirements. The control of the vehicle on land or in water remains the same and the transition from land to water can be accomplished quickly and safely without the need of specialized procedures.

6.2.1 SPEED AND RANGE

The speed of the vehicle was determined from the resistance data obtained from model tests at the University of Michigan Towing Tank (See 6.2.3), the power plant chosen for the vehicle and an assumed propulsive efficiency. The maximum water speed thus determined will be approximately 6.5 miles per hour.

The propulsive efficiency assumption was based on a survey of test results of both model and full scale vehicles having a configuration in hull and track detail similar to those chosen for this vehicle. The track chosen for the vehicle was designed to match the details of the most efficient water propulsion track that is known to date.

To provide the track with the required slip-speed in water (an equivalent 13.6 to 14.2

land miles per hour), a 3.26 to 1 ratio was chosen for the transmission second gear forward range.

The reverse gear train ratio and the track propulsive efficiency in that direction limit the vehicle speed in water to 2.2 miles per hour in reverse.

Sufficient fuel tank capacity will be provided to enable the vehicle to meet the specified range of 50 miles at the maximum speed (6.5 miles per hour) in still water. The vehicle in water operation will be capable, thus, to operate:

(1) 7.7 hours at full power;
(2) 11.7 hours at three-fourths power; and
(3) 20 hours at one-half power.

6.2.2 HYDROSTATIC CALCULATIONS

To determine the buoyancy and attitude of the floating vehicle, a displacement and center of buoyancy curve, (Figure 6-7), was plotted from data calculated, as shown on Calculation Sheet 1, Page 6-18. Cross-sectional areas of equally-spaced stations were determined with a planimeter for several waterlines. Using the appropriate Simpson's Rule multipliers, the volume and location of the center of buoyancy was determined.

The draft has been calculated as follows:

(1) loaded vehicle of 35,000 pounds is 44 inches
(2) unloaded vehicle of 25,000 pounds is 34 inches.

The metacentric heights in the longitudinal and transverse planes were calculated as shown on Calculation Sheets 2, 3, and 4 Pages 6-19, 20 & 21 to determine the static stability of the vehicle in the unloaded and loaded condition.

The empty vehicle (25,000 pounds) will trim normally down at the stern approximately

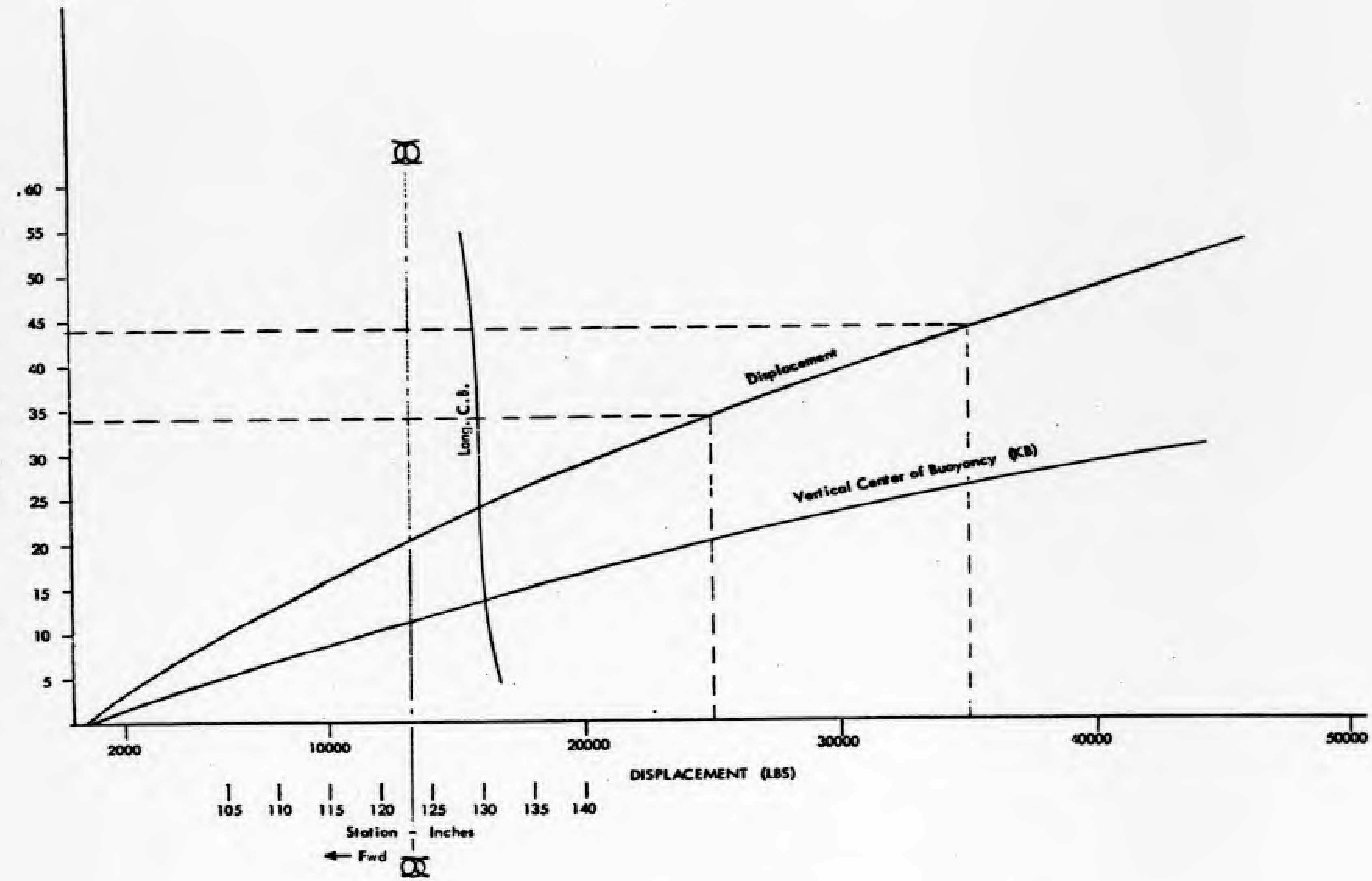


Figure 6–7 Displacement and Center of Buoyancy Curve

4.875 inches and will require a moment of 1,700 foot pounds to trim one inch and a moment of 1,770 foot pounds to heel one degree.

The payload of 10,000 pounds of uniform density was located centrally with respect to the cargo hatches for determining the center of gravity location of the loaded vehicle. The loaded vehicle of 35,000 pounds thus has an attitude down at the stern approximately 2.24 inches (See Figure 6-12, Page 6-34) and will require a moment of 1,755 foot pounds to trim one inch and a moment of 1,732 foot pounds to heel one degree.

To determine the stability for large angles of heel a curve of transverse static stability, Figure 6-8, was plotted from data calculated as shown on the sample Calculation Sheet 5, Page 6-22. The 35,000 pound (loaded) vehicle was chosen for this calculation. The vehicle's range of heeling stability extends to an inclined angle of 126.5 degrees. The righting moment of 60 degrees of roll is 56,700 foot pounds.

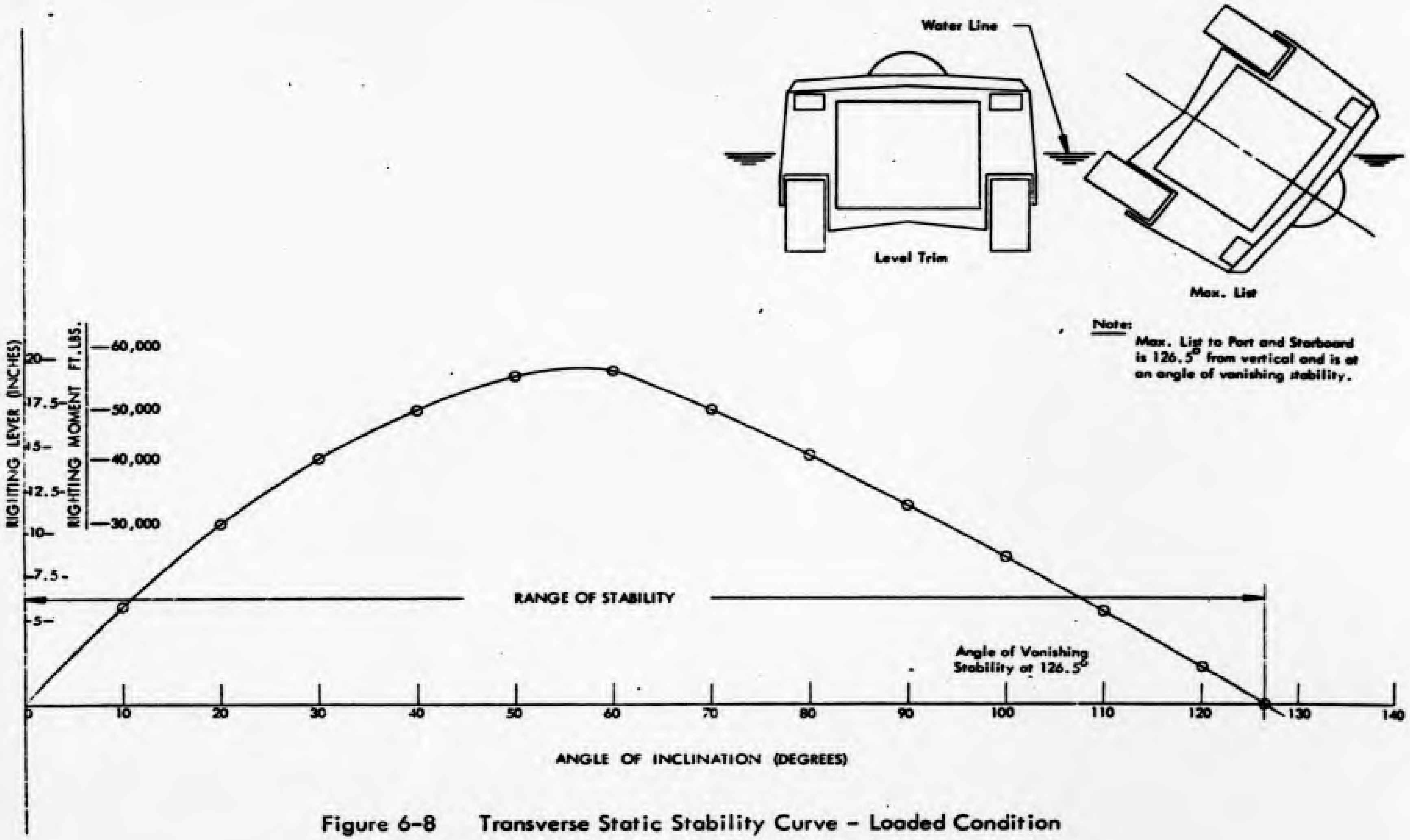


Figure 6-8 Transverse Static Stability Curve - Loaded Condition

DISPLACEMENT & CENTER OF BUOYANCY LOCATION

Draft 36.5 inches

Water Length 19.804 feet

Longitudinal Interval (L.I) = $\frac{W.L.}{20}$ = .990 feet (Station Spacing)

M_1 = 4/3 (L.I) = 1.333 (.990) = 1.3197

Station	1/2 Area (Ft^2)	SM	f (Areas)	Arm	f Mom Areas
0	0	1/2	0.	10	0.
1	8.7	2	17.4	9	156.6
2	11.	1	11.	8	88.
3	11.4	2	22.8	7	159.6
4	11.52	1	11.52	6	69.2
5	10.70	2	21.40	5	107.
6	11.95	1	11.95	4	47.8
7	11.40	2	22.80	3	68.4
8	11.60	1	11.60	2	23.2
9	11.60	2	23.20	1	23.2
10	10.70	1	10.70	0	+ 743.0
11	11.95	2	23.90	1	23.90
12	11.30	1	11.30	2	22.60
13	11.10	2	22.20	3	66.60
14	11.50	1	11.50	4	46.00
15	10.50	2	21.00	5	105.00
16	11.20	1	11.20	6	67.20
17	10.40	2	20.80	7	145.60
18	10.50	1	10.50	8	84.00
19	11.80	2	23.60	9	212.40
20	7.35	1/2	3.17	10	30.17
		f (Area) =	323.54		- 803.47
					+ 743.00
				Difference	- 60.47

Volume Displaced (V) = f (Areas) x M_1 = 323.54 x 1.3197 = 426.97 ft^3

Pounds Displaced (Δ) = V x s.w. $\frac{(lbs)}{ft^3}$ = 426.97 x 64 = 27,336 lbs.

C.B. aft/fwd of Sta. 10 = $\frac{\text{Difference of f (Mom A) x L.I.}}{\text{f (Areas)}}$ = .185 ft. aft

LONGITUDINAL & TRANSVERSE STATIC STABILITY - 25,000 LBS.

Vessel Wt. (Δ) = 25,000 lbs.
Draft (H) = 34 in.

Water Length 236.4 in. = 19.7 ft.
Water Width 124.8 in. = 10.4 ft.

Area of Waterplane (WA) = 19.7 x 10.4 = 204.83 ft^2

Volume (V) = $\frac{25000}{64}$ = 390.62 ft^3

Vertical Center of Buoyancy above Keel (KB) = $1/3 \; \frac{5H}{2} - \frac{V}{WA}$ H = Draft = 2.833 ft.

$$KB = 1/3 \left(\frac{5\,(2.833)}{2} - \frac{390.62}{204.83} \right) = 1.7277 \text{ ft.}$$

Long. Moment of Inertia (I_L about C.F.) = $\frac{b\,h^3}{12}$ where b = 10.4 ft., h = 19.7 ft.

$$I_L = \frac{10.4 \times (19.7)^3}{12} = 6623 \text{ ft}^4$$

Transverse Moment of Inertia I_T about C_L = $\frac{b\,h^3}{12}$ where b = 19.7 ft., h = 10.4 ft.

$$I_T = \frac{19.7 \times (10.4)^3}{12} = 1840 \text{ ft}^4$$

Metacentric Radius (BM) = $\frac{I}{V}$ $BM_L = \frac{6623}{390.62} = 16.7$ ft.

$BM_T = \frac{1840}{390.62} = 4.65$ ft.

Metacentric Radius	BM_L = 16.7 ft.	BM_T = 4.65 ft.
Vertical Center of Buoyancy	KB_L = 1.7277 ft	KB_T = 1.7277 ft.
Metacenter above Keel	KM_L = 18.4277 ft	KM_T = 6.3777 ft.
Vertical Center of Gravity	KG_L = 2.323 ft	KG_T = 2.323 ft.
Vertical Distance between KG and BM =	GM_L = 16.1047 ft	GM_T = 4.0547 ft.

Moment to trim one inch $(MT_1) = \frac{\Delta\, GM_L}{12\,(W.L.)} = \frac{25000 \times 16.10}{236.4} = 1700$ ft. lbs/in.

Longitudinal distance between CB & CG = BG_L = 7.95" Aft (from displacement curve)

Trim of Vessel $= \frac{BG_L\,\Delta}{MT_1} = \frac{BG_L \times WL}{GM_L} = \frac{7.95 \times 236.4}{12 \times 16.1047} = 9.75$ in. stern heavy

Trim at each end $= \frac{Trim}{2}$ = 4.875 in.

Moment to Heel 1° $= .01745 \times GM_T \times \Delta$ =

$= .01745 \times 4.0547 \times 25{,}000$ - 1,770 ft.lbs/deg.

LONGITUDINAL AND TRANSVERSE STATIC STABILITY - 35,000 lbs.

Vessel Wt. (Δ) = 35,000 lbs. Draft = 44 in. Water length 240.5 in. = 20.04 ft.
Water width 124.8 in. = 10.40 ft.

Area of Waterplane (WA) = 20.04 x 10.4 = 206.16 ft^2

Volume (V) - $\frac{35000}{64} = 546.87\ ft^3$

Vertical Center of Buoyancy above keel (KB) $= 1/3 \left(\frac{5H}{2} - \frac{V}{WA}\right)$ H = draft = 3.66 ft.

$$KB = 1/3 \left(\frac{5\,(3.66)}{2} - \frac{546.87}{206.16}\right) = 2.166 \text{ ft.}$$

Long. Moment of Inertia (I_L about C.F.) $= \frac{b\,h^3}{12}$ where b = 10.40 ft., h = 20.04 ft.

$$I_L = \frac{10.4\,(20.04)^3}{12} = 6903.5 \text{ ft.}^4$$

Transverse Moment of Inertia I_T about C_L $= \frac{b\,h^3}{12}$ where b = 20.04 ft., h = 10.40 ft.

$$I_T = \frac{20.04\,(10.40)^3}{12} = 1870 \text{ ft.}^4$$

Metacentric Radius (BM) = $\frac{I}{V}$ $BM_L = \frac{6903.5}{546.87} = 12.6$ ft.

$BM_T = \frac{1870}{546.87} = 3.35$ ft.

Metacentric Radius $BM_L = 12.6$ ft. $BM_T = 3.35$ ft.

Vertical Center of Buoyancy	KB_L = 2.166	KB_T = 2.166
Metacenter above Keel	KM_L 14.766	KM_T = 5.516
Vertical Center of Gravity	KG_L 2.677	KG_T = 2.677

Vertical Distance between

KG & BM = GM_L = 12.089 ft. GM_T - 2.839 ft.

Moment to trim one inch (MT_1) = $\frac{GM_L}{12\ (W.L.)} = \frac{35000 \times 12.089}{240.5}$ = 1755 ft.lbs/in.

Longitudinal Distance between CB & CG = BG_L = 2.7 aft (from displacement curve)

Trim of Vessel $= \frac{BG_L \Delta}{MT_1} = \frac{BG_L \times WL}{GM_L} = \frac{2.7 \times 240.5}{12 \times 12.089}$ = 4.48 in stern heavy

Trim at each end $= \frac{Trim}{2} = \frac{4.48}{2} = 2.24$ in.

Moment to Heel 1° = $.01745 \times GM_T \times \Delta$ =

= $.01745 \times 2.839 \times 35000$ = 1732 ft. lbs/deg.

RIGHTING LEVER - LARGE ANGLES OF HEEL

Weight = 35,000 lbs. $\Theta = 60^\circ$ Sin Θ = .86603

Draft = 44 inches Water Length = 20.04 feet Displacement (V) 546.87 ft^3

Longitudinal Interval (L.I.) = $\frac{W.L.}{20}$ = 1.002

M_1 = 2/3 (L.I.) = .6675 KB = 2.166 ft. KG = 2.677 ft. VBG = .511 ft.

Righting Lever

Station	Wedge Area $Ft.^2$	SM	f(Areas)	hh_1 (Ft)	hh_1 x Area
0	0	1/2			
1	10.15	2	20.30	4.87	49.187
2	11.90	1	11.90	5.34	63.500
3	11.63	2	23.26	5.20	60.476
4	12.00	1	12.00	5.42	65.040
5	11.51	2	23.02	5.17	59.506
6	12.35	1	12.35	5.27	65.084
7	11.80	2	23.60	5.52	65.130
8	11.51	1	11.51	5.17	59.506
9	12.05	2	24.10	5.44	65.600
10	11.51	1	11.51	5.17	59.506
11	12.03	2	24.06	5.40	65.000
12	11.80	1	11.80	5.37	63.200
13	11.60	2	23.20	5.24	60.700
14	11.65	1	11.65	5.38	62.800
15	10.85	2	21.70	5.35	58.000
16	10.34	1	10.34	4.94	51.200
17	9.90	2	19.80	4.94	48.900
18	8.90	1	8.90	4.50	40.000
19	8.20	2	16.40	4.65	38.100
20	6.48	1/2	3.24	4.02	26.100
Sum Area	218.16	f (Area)	324.64	Sum Mom (A)	1126.53

Wedge Volume (V) = f Area x M_1 = 324.64 x .6675 = 218.0 ft^3

Distance between Wedge Centroids $\overline{hh_1} = \frac{\text{Sum (Mom A)}}{\text{Sum Area (Pop P)}} = \frac{1126.53}{218.16} = 5.16$ ft.

Righting Lever = $\frac{(V)hh_1}{V}$ − VBG Sin Θ = $\frac{218 \times 5.16}{546.87}$ − (.511 x 866.03) = 1.618 ft. = 19.41 in.

6.2.3 MODEL TESTS

A one-fifth scale model of the hull shape designed by Chrysler Defense Engineering for the LVTPX 11 vehicle was prepared by the Ships Hydrodynamic Laboratory of the University of Michigan, under the direction of Professor Richard B. Couch, of the Department of Naval Architecture and Marine Engineering.

Tests were performed on the basic model to determine its resistance. Trim changes were also evaluated. Several changes to the basic hull configuration were made and additional tests were performed to determine their effect on resistance. The actual report of the tests, including results, curves, and recommendations follows:

REPORT OF MODEL RESISTANCE TESTS

OF TRACKED AMPHIBIOUS PERSONNEL AND CARGO CARRIER

(LVTPX II)

U of M Model No. 935

Prepared by Prof. Richard B. Couch
for Chrysler Corporation, Detroit, Michigan

At the request of Chrysler Corporation in September, 1961, a 1/5 scale model (Model No. 935) of a proposed new Tracked Amphibious Personnel Passenger and Cargo Carrier (LVTPX II) for the U. S. Navy was manufactured and resistance tests carried out at The University of Michigan towing tank. The model was manufactured of wood with wheels, suspension, and track simulated. The track itself was made of wood without any attempt to represent the actual details.

The results of the basic resistance tests in the form of EHP curves are given on Figure 6-9. Several additional limited tests were run to determine the effect of trim change and certain hull form changes. The effect of these on resistance are discussed in the following paragraphs.

The method of expanding the data is the same as that normally used for tests of such models, i.e. no friction correction is used and the total resistance of the model is expanded to full scale by Froude's Law. This practice has

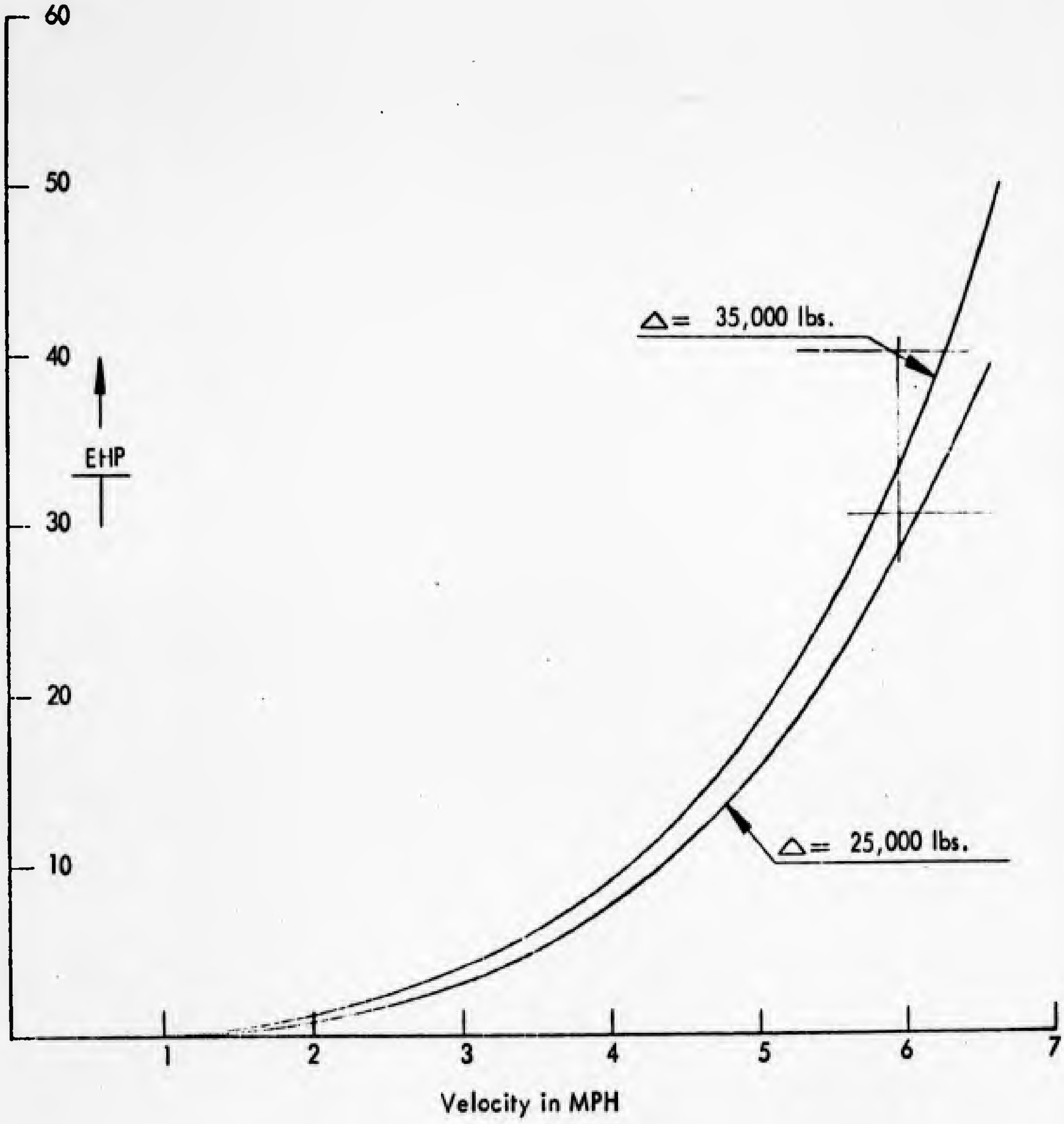
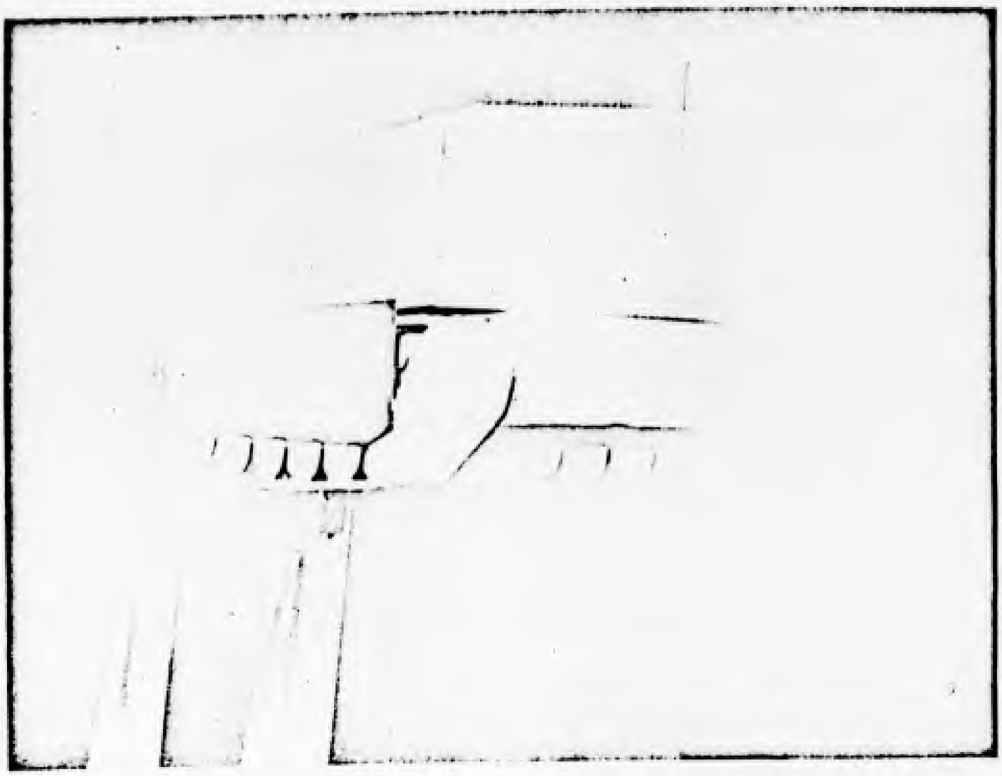


Figure 6-9 EHP vs MPH - Basic Condition

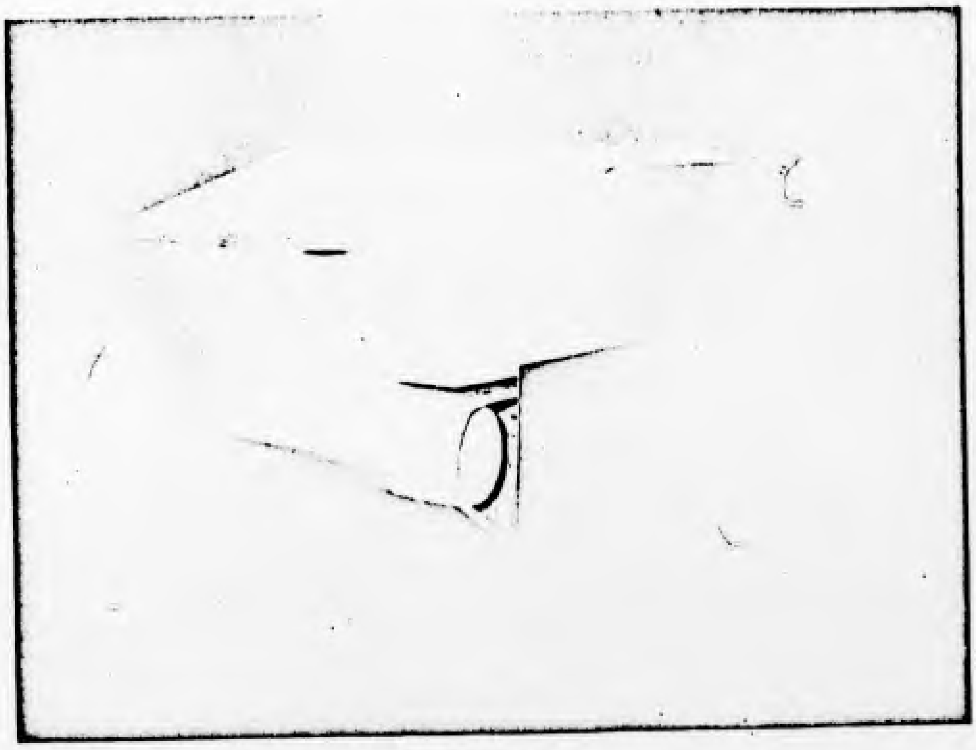
been followed at The David Taylor Model Basin in tests of similar vehicles. The David Taylor Model Basin has also followed the practice of manufacturing a simplified block model and correcting for the increased resistance of the actual track and suspension system. Since the modelling of the track and other details for The University of Michigan tests was more complete than the TMB tests (TMB report no. 1253), it is believed that no correction should be made. In fact, it is considered that the true resistance of the track and suspension is probably actually less than The University of Michigan tests indicate. However, no correction has been made in working up the test data.

The value of EHP (effective horsepower) of such vehicles is somewhat academic since the trim of the craft has an appreciable effect on resistance, and it is not possible to readily determine the trim when propelled by the tracks. The thrust line of the tracks will be somewhat lower than the lowest tow point obtained in the resistance tests. As a result, the model will tend to trim by the bow when towed and will have less trim when propelled. Nevertheless, an attempt to determine the effect of trim was made.

The initial tests of the model were made with the model built exactly to Chrysler Plan CWO No. 400956, Layout No. 4931 of 9/14/61. Tests were run at 35,000 lbs. and 25,000 lbs. displacements with freeboard as indicated in the following table. Subsequently, the model was trimmed 5 inches more by the stern, and the side skirts were deepened to 32 inches below the hull line. (Figure 6-10) The results of the tests in this condition showed considerable



3/4 Front View



3/4 Rear View

Figure 6-10 Towing Tank Model - Basic Condition improvement over the first tests/hence were used to expand to full scale EHP data. The curves included in this report are for this condition, and it is referred to herein as the basic condition.

Many other variations of the hull were tested in a limited way to obtain an indication of the effect on resistance. (Figure 6-11) Since a complete range of speeds was not explored in these latter tests, only the percentage differences from the basic condition are reported. Table 6-1 on following page summarizes the results of all tests.

DISCUSSION OF RESULTS

The hull design of the basic condition is assumed to be representative of the final design. Minor changes in hull will not appreciably affect the speed of the vehicle. On the basis of the EHP values obtained, it is estimated that the full scale vehicle will be able to attain a water speed ahead of about 6 MPH. This assumes a propulsive coefficient, i.e. $\frac{EHP}{SHP}$ of about 13% and a horsepower at the tracks of 260. The efficiency of track propulsion is difficult to assess, hence this can only be approximate.

Most of the hull variations tested produced very minor changes in resistance. One of the most significant gains was realized when the hull was trimmed aft by 5 inches (full scale) more than the original waterline. This new trim is that used in the basic condition. The effect of the increased depth of side skirts was included with the trim change, hence cannot be readily separated but it is believed that approximately half of the reduction in resistance was

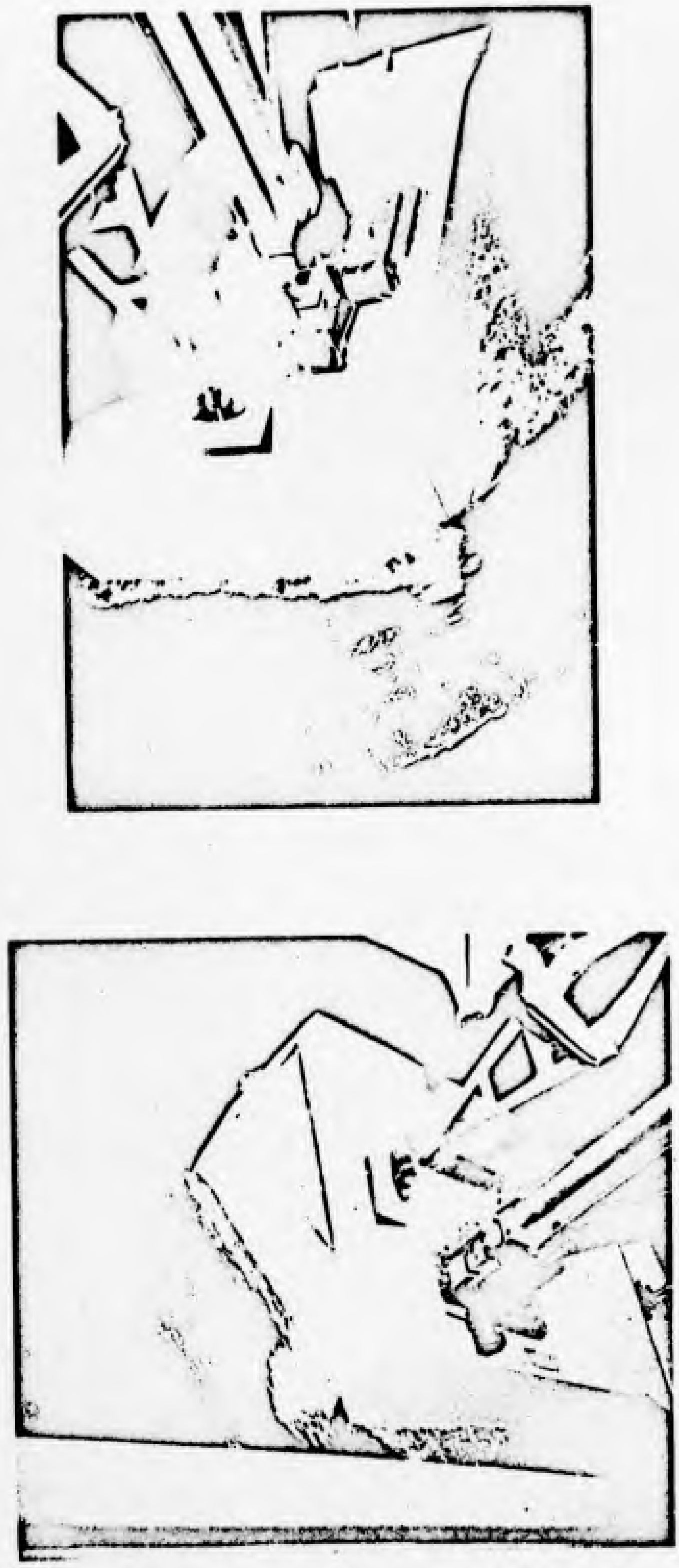


Figure 6-11 Towing Tank Model - Hull Variations

TABLE 6-1

FREEBOARD AT REST FWD.	AFT	DESCRIPTION	RESISTANCE CHANGE FROM BASIC CONDITION
7 3/4"	1 7/8"	Side Skirts Deepened	0%
6 3/4"	2 7/8"	Original Skirts	+26
6 3/4"	2 7/8"	Side Skirts Deepened	+19
7 11/16"	1 7/8"	Trim Changed from Above	+ 5
7 9/16"	1 13/16"	With Bow Plt. & Lg. Skirts	- 1
7 9/16"	1 13/16"	Bow Plt. Position Changed from Above	- 4
8 1/8"	1 1/16"	Trim Changed from Above	- 2
not measured	1 7/8"	Bow Corners Cut Off with Long Skirts	+ 1
"	1 7/8"	With Pointed Bow, Flat Bottom, & Long Skirts	-18
"	2 7/8"	Trim Changed from Above	-16
"	2 7/8"	Removed Pointed Bow	-10
7 3/4"	1 7/8"	Added 29.0 lb. to Model Weight	- 4

due to deeper side skirts and half to the aft trim. Since the track propulsion will tend to cause trim aft, it is recommended that the initial waterline trim be held to in the final design.

The effect of an inclined bow plate was investigated. Only minor improvement was found, hence it was not pursued. After the tests described above were completed, several other changes were investigated. First, the bow waterline corners were cut off at about a 45 degree angle. This produced an insignificant change. Later, the bow was fitted with a wedge-shaped stem by extending the line of the cut-off corners to a pointed stem. At the same time, the bottom was made flat by filling in the inverted "V". The combination of the two changes produced a substantial reduction in resistance, about half of which is credited to each change. In the test, the freeboard aft and the displacement was held to that for the basic condition. The buoyancy of the additions to the hull caused a reduction of draft and a greater aft trim which probably caused part of the reduction in resistance. Removal of the bow addition reduced the changes by about half. Addition of 29 lbs. to the model ballast to compensate for the additional buoyancy of the flat bottom and bring the draft back to the original figures reduced the improvement in resistance to about 4%, as indicated in the table. If the "V" bottom had been changed to a flat bottom by cutting back to the height of the apex of the "V", it is believed the change in resistance would be negligible.

RECOMMENDATIONS

Since so many limited tests were run, it is difficult to readily analyze the data.

However, certain recommendations can be made relative to the original model configuration. These are as follows:

(1) Design for the original trim conditions. The thrust of the tracks will tend to improve the trim conditions when underway.

(2) Deepen the side skirts as much as practicable.

(3) Add a bow wave and spray deflector to minimize wetness forward when underway.

(4) If practicable, develop a portable or folding bow addition similar to that tested.

(5) If No. 4 above is not practicable, at least take off the bow corners as much as practicable.

GENERAL CONCLUSION

The speed obtainable with a vehicle of this type is limited because of the short length, box form, and high drag of the track assembly. When track water propulsion is used, a very low propulsion efficiency results which also limits speed. The estimated speed obtainable ahead in relatively smooth water is about 6 MPH. Assuming that considerably greater horsepower were installed in this vehicle, it is believed that a water speed of about 7 MPH would be the maximum obtainable. The estimated speed astern for the present configuration is estimated to be about 3 to 4 MPH, depending upon the track configuration and available horsepower.

6.2.4 SURFABILITY

The high degree of roll recovery indicated in the aforementioned calculations will make the vehicle ideally adaptable for surfing operations from a stability viewpoint. Water tightness of all openings, the lightweight of the vehicle, the ruggedness of construction as indicated in paragraph 4.5 will provide the vehicle with the required strength and displacement volume. A watertight engine and crew compartment have been provided in the vehicle and the machinery components arranged so that in an emergency, wherein the engine compartment may become completely flooded the vehicle will remain afloat. Although the vehicle in this position may be trimmed severely down at the stern, the engine air-breathing openings remain above the water line as well as all crew compartment hatches.

The engine is thus capable of operating the bilging system to evacuate the water from the hull and righting the vehicle.

In an extreme emergency where sufficient force has been generated (in excess of 58,000 foot pounds) to completely overturn the vehicle, the vessel will remain afloat in this position with a trapped air pocket in the crew compartment. The trapped air pocket is of sufficient volume and at a level to permit the crew and troops to stand with their heads above the water line; prior to escape underwater through the topside hatch openings.

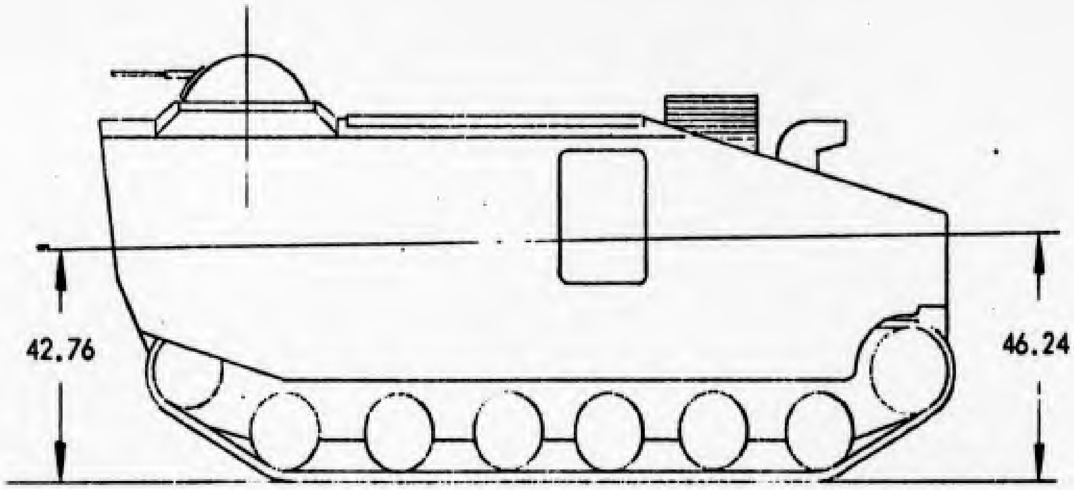


Figure 6-12 LVTPX-11 Loaded Draft Trim Line

7. NUCLEAR BLAST EFFECTS

The capability of the proposed Tracked Amphibious Vehicle LVTPX-11 to resist the blast, thermal radiation, and nuclear radiation effects of a 0.1 MT yield nuclear bomb has been estimated. The 0.1 MT bomb was chosen as it is felt that it is the largest yield nuclear weapon which will be used against this vehicle in a combat situation. The analysis includes studies of overturning, sliding, structural loading, heating, and nuclear radiation.

The analyses, assumptions, and conclusions contained herein constituted a limited analysis of the vehicle. Such considerations as the effects of an underwater blast, larger yield bombs, and special design against thermal and nuclear radiation have not been considered.

7.1 OVERTURNING AND SLIDING

The primary blast effect is overturning. However, if the vehicle is aligned head-on with the blast or offers resistance to overturning, displacement caused by aerodynamic impulse becomes a consideration. The results of the overturning and sliding analyses are shown in Figure 7-1 for broadside, and Figure 7-2 for head-on. Examination of these results shows that the vehicle has an over-all blast resistance to overturning and sliding of 11.5 psi over-pressure. This conclusion is based on the supposition that sliding will not substantially affect a vehicle of this design and mission.

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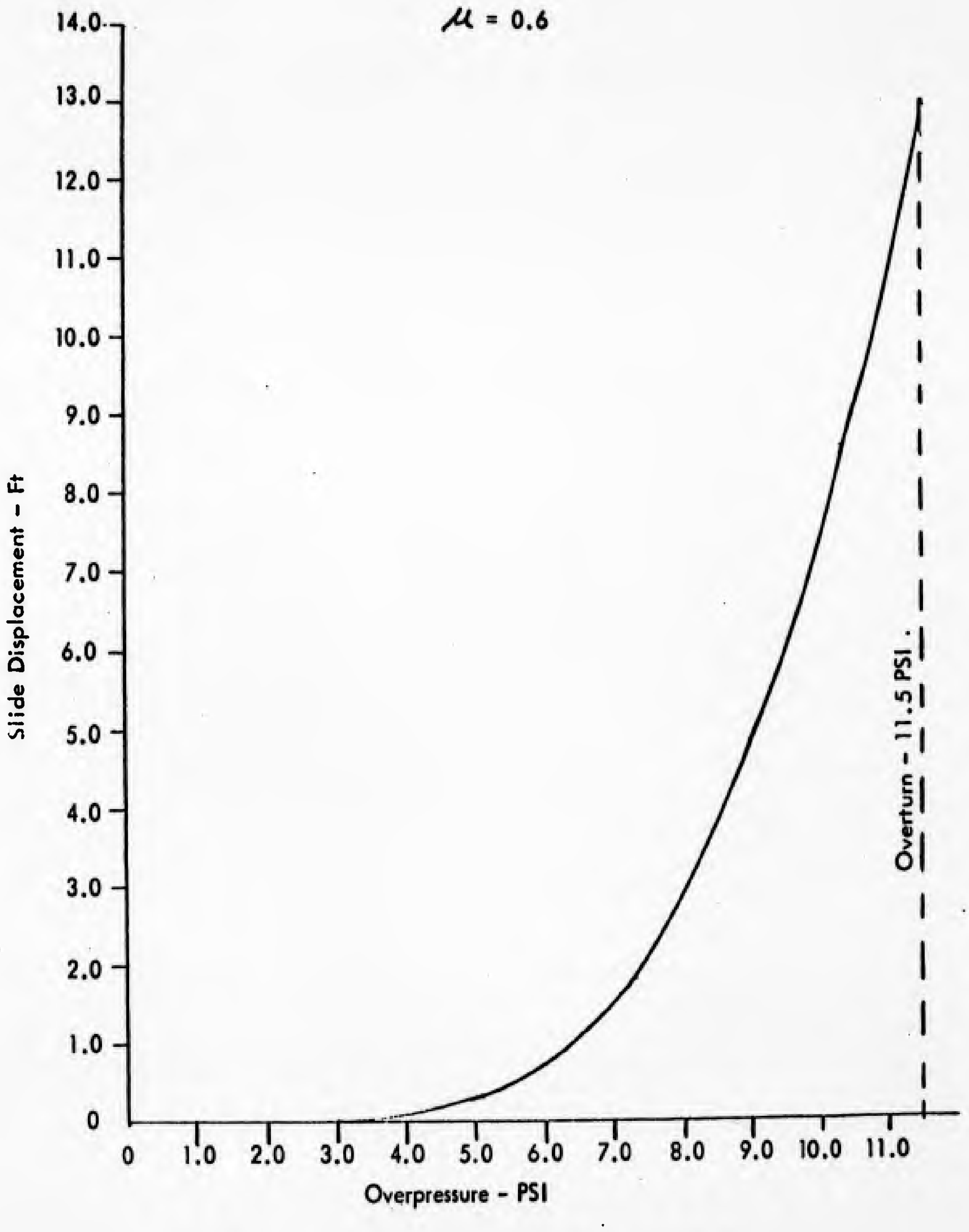


Figure 7-1 Overturning and Sliding Effects - Broadside

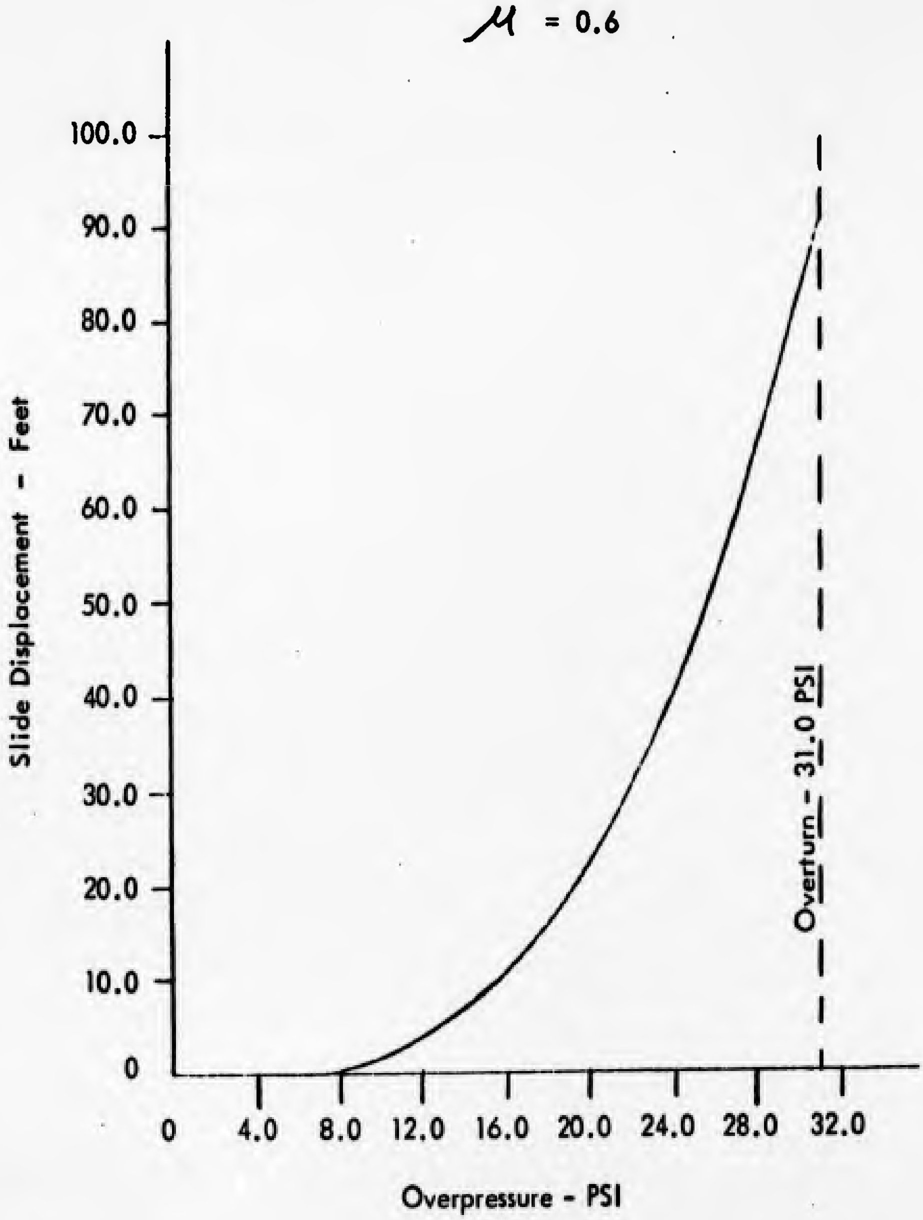


Figure 7-2 Overturning and Sliding Effects - Head-on

7.2 STRUCTURAL LOADING

Structural loading of the sides of the vehicle was considered to be approximated by the case of a square plate with simply-supported sides. The maximum stress is assumed to be obtained at the center of the plate. The results of a structural loading analysis show that maximum allowable over-pressure for plates over 42 inches square and 0.5 inches thick is 9 lbs/in^2. This is based on the assumption that the critical stress value is 60,000 lbs/in^2, which under the conditions imposed is considered to be conservative. Therefore, it is felt that the effects of overturning are more critical, and that the structural loading requirements can be designed into the vehicle.

7.3 THERMAL RADIATION

The thermal radiation from the fireball transmitted to the target varies inversely as the square of the slant range. The heat flux at the target has been calculated based on a value of transmittance determined from Figure 7.118, "The Effects of Nuclear Weapons". This curve represents the total transmittance for a 6000°F black body source under the conditions of 10 mile visibility and 10 gm/cubic meter water vapor concentration.

This heat flux incident to the vehicle causes a temperature gradiant in the walls. This temperature gradiant has been calculated for the case of no heat flow across the inside surface of the wall. Figure 7-3 represents the temperature gradiant through the vehicle wall for surface temperatures of 200, 300, and 400 degrees Fahrenheit. A mean temperature of 400°F is assumed critical since at this temperature the strength of aluminum is reduced by a factor of two.

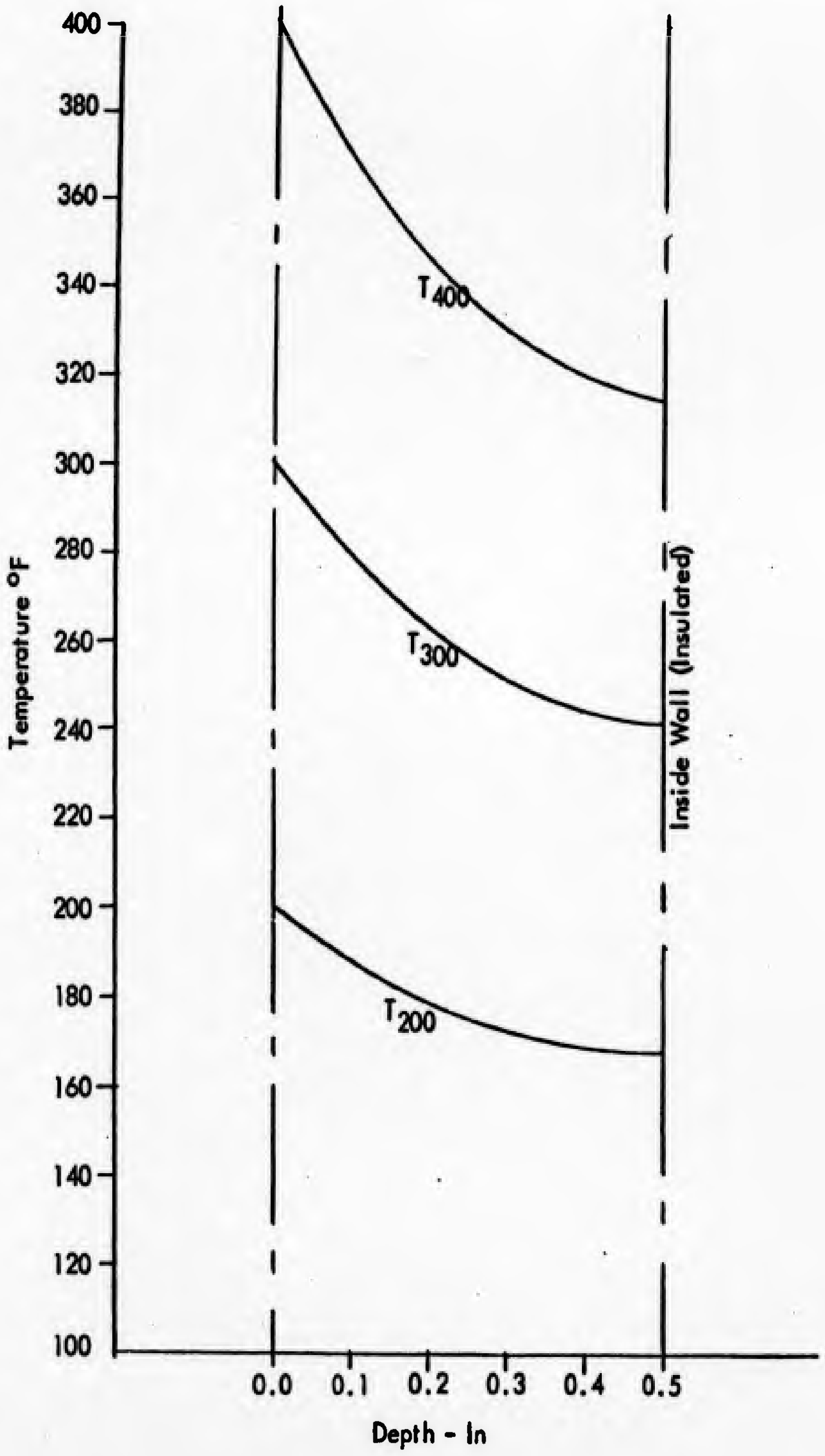


Figure 7-3 Vehicle Wall Temperature Gradient

7.4 NUCLEAR RADIATION

The transmission of nuclear radiation to the target area is governed by a relationship similar to that for thermal radiation, the two important factors being slant range and transmittance. The lethal biological dose due to gamma radiation and neutrons is 650 roentgens. A dose of 450 roentgens will prove fatal to 50% of the exposed personnel.

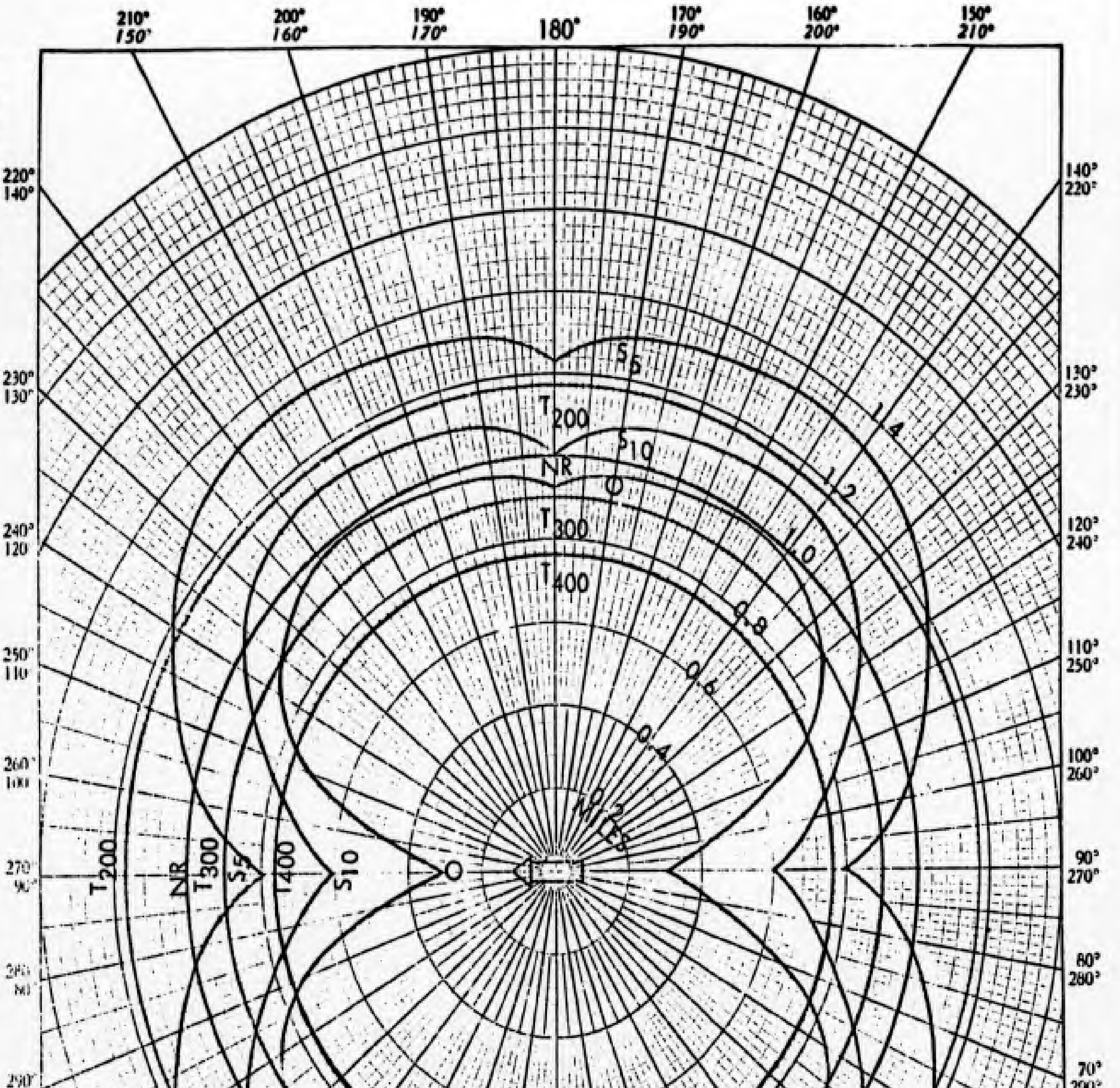
The total attenuation factor of one-half inch aluminum is approximately 1.05. Therefore, the lethal dose of combined nuclear radiation for personnel inside the vehicle is approximately 680 roentgens. The effects of residual radiation were not considered, since it was felt that it would not affect the primary mission of the vehicle.

There are some detrimental effects of nuclear radiation on electronic components and circuitry, but the state-of-the-art in this respect is not advanced to the point where an analysis can be made.

7.5 NUCLEAR EFFECTS RADII

Figure 7-4 shows the radii of the various effects of a 0.1 MT nuclear blast. This curve summarizes the previously discussed results. The various designations are:

S_5	Slide displacement of five feet
S_{10}	Slide displacement of ten feet
O	Overturn
T_{200}	Wall surface temperature of 200°F
T_{300}	Wall surface temperature of 300°F
T_{400}	Wall surface temperature of 400°F
NR	Nuclear radiation lethal dosage (680 roentgens)



NUCLEAR BLAST EFFECT RADII
FOR
PROPOSED TRACKED AMPHIBIOUS VEHICLE, LVTPX-11

Figure 7-4

Examination of the curves shows that a 0.1 MT bomb falling within the nuclear radiation radius will effectively kill the vehicle. This radius is one mile. Since sliding is not considered deterimental, one mile is considered to be the kill radius of the vehicle and 8.8 psi is the correspondingly required vehicle hardness.

8. DESIGN ALTERNATIVES

In Paragraph 2.1 of this report, a comment was made that certain trade-offs had occurred to us during the concept design program. In each such case, where a trade-off would cause an increase in the gross weight of the vehicle, we chose to follow the path that would lead to a 35,000 pound vehicle, regardless of the apparent attractiveness of the design alternative.

We believe, however, that we would not be living up to our responsibilities fully if we failed to point out those areas where, in our opinion, an opportunity exists to improve the usefulness or effectiveness of the vehicle if the specifications and guidance which has been provided by the Bureau of Ships could be altered to accommodate the change.

Accordingly, we will discuss four areas of possible trade-offs as we view them from the designer's point of view and suggest that the Bureau of Ships and the Marine Corps exercise the judgment as to whether or not the alternatives should be brought into the vehicle design during the detailed design and prototype fabrication phase of this program.

8.1 ARMOR PROTECTION

It is Chrysler's opinion that this combat vehicle should be capable of transporting personnel and cargo to a point approximately 200 to 300 yards from an enemy armed with shoulder-fired small arms. We don't believe that the armor protection which we have been able to provide within the weight limitation which was placed upon us is really adequate for this vehicle. We would like to suggest that the Government consider permitting the gross vehicle weight to increase so as to provide 1-1/4 inches of armor on the sides of the hull and 3/4 inch on the top and bottom. If this were to be done, the weight of the additional armor would add approximately 5,000 pounds to the gross weight of the vehicle.

The performance of the vehicle in a 40,000 pound configuration has been investigated to be certain that the mission capability would not be unreasonably depreciated. The results are as follows:

(1) With respect to mobility, the ground pressure of the 35,000 pound vehicle is 5.46 psi; that of the 40,000 pound vehicle would be 6.35 psi.

The drawbar pull to gross vehicle weight ratio in one soil condition considered in the mobility analysis of the 35,000 pound vehicle is 0.873; that of a 40,000 pound vehicle would be 0.77. (The DBP/W ratio of the LVTP-5 is 0.725).

From this we would conclude that while the mobility of the heavier vehicle would be somewhat less than that of the lighter one, the mission capability would remain essentially undiminished.

(2) The side slope stability of the 40,000 pound configuration would be approximately equal to that of the 35,000 pound vehicle since the height of the center of gravity above the ground would be relatively unchanged.

(3) The gradeability of the 40,000 pound vehicle would be 70% at a speed of 3.2 mph. The gradeability of the 35,000 pound configuration is 70% at 5 mph.

(4) The maximum speed of the 40,000 pound vehicle would be 38.7 mph; that of the 35,000 pound vehicle is 40 mph.

(5) The range (on land) of the 40,000 pound vehicle would be 370 miles; that of the 35,000 pound vehicle is 420 miles.

(6) The static stability of the 40,000 pound vehicle at small angles of heel will be somewhat better than that of the 35,000 pound vehicle.

The range of stability is 126 degrees for the 35,000 pound configuration; the range of stability of the 40,000 pound vehicle would be approximately 120 degrees.

(7) The loaded draft of the 35,000 pound vehicle is 44 inches; that of the 40,000 pound vehicle would be 48.5 inches.

(8) The water speed of the 40,000 pound vehicle would be reduced by approximately 1/2 mph with respect to that of the 35,000 pound vehicle (from 6-1/2 to 6 mph). The range in miles at top speed would be reduced (if the fuel supply were not increased), but the endurance in hours would remain the same.

In summarizing this point on armor protection, Chrysler would strongly recommend that consideration be given by the Government to the 40,000 pound vehicle configuration and the added combat effectiveness which we believe that it would provide. We feel certain that the heavier vehicle could be designed so as to insure that performance on land and in the water would not in any way be compromised. The advantages to be gained from the improved ballistic protection more than offset, in our opinion, the marginal sacrifices which would be made in terms of performance, and that, therefore, this particular trade-off deserves serious study.

8.2 CARGO-CARRYING CAPACITY

The Development Characteristics provided Chrysler by the Government state that a payload of 10,000 pounds should be provided for. The configuration of the payload was not specifically defined. Chrysler, therefore, began an investigation of Conex containers, vehicles, and weapons which might be carried in a vehicle such as the LVTPX-11. The cargo area was proportioned initially in the concept study to accommodate the 27 combat-equipped Marines and the 40 x 48 inch palletized container.

We discovered early in the program that the cargo area within the hull would not accommodate either the 105 mm towed Howitzer or the Mighty Mite vehicle.

With respect to the Mighty Mite, we found that by increasing the over-all width of the vehicle from the 9 feet 11 inches, which would otherwise be adequate, to the full 10 feet 6 inches, we could accommodate that vehicle in the cargo compartment of the LVTPX-11. The increase in width has had, we believe, an adverse effect on water resistance. We would suggest, therefore, that the Bureau of Ships and the Marine Corps review the matter to determine just how real is the need to transport the Mighty Mite in the LVTPX-11. If the need does not, in fact, exist, then the width of the vehicle described in this report can be decreased by 7 inches. If the width were to be so decreased, the weight of the hull would be reduced by 300 pounds, which could then be reinvested in armor or fuel.

8.3 PERSONNEL CARRYING CAPACITY

The volume of the cargo area has been determined principally by the requirement to carry the specified number of personnel. The total weight of the personnel to be carried (plus their equipment) will not equal 10,000 pounds. The volume provided in the cargo compartment is probably larger than is required to carry 10,000 pounds of high density cargo.

Chrysler would recommend, therefore, that the organization of the Marine Corps Squad and Platoon be carefully studied, in their present and proposed (for the future) configurations. If, for example, the Marine Corps were to adopt a Platoon of four 10-man squads, then perhaps this vehicle should be designed to carry two 10-man squads, plus a crew of two. If such an arrangement should prove feasible, and Chrysler has very little information to go on in this area, it might be possible to design a vehicle having a better length-to-width ratio (for water operation) than does the vehicle described in this report.

8.4 ADDITIONAL FUEL CAPACITY

We would like to call attention to the fact that there is space available under the cargo floor, forward of the fuel tank area, which could be used for additional fuel tank capacity. The space available would provide for an additional 85 gallons of fuel. The additional fuel would add approximately 595 pounds to the gross weight of the vehicle and the fuel cells, lines and fittings associated with the added fuel capacity would add another 60 pounds. Thus, for an increase of about 655 pounds of weight, the range of the vehicle could be extended from 420 to 600 miles.

Chrysler would recommend that this fact be kept in mind as a possible kit adaptation to be installed in Personnel Carrier vehicles which might be fitted for special missions. By so doing, the gross weight of the vehicle (when carrying personnel) would remain below the 35,000 pound figure.

9. SPECIAL VEHICLE CONFIGURATIONS

9.1 COMMAND VEHICLE (See Figure 9-1)

The Command Vehicle is an LVTPX 11 vehicle, designed to be used as an amphibious mobile field headquarters communication center. The standard radio equipment installed in the LVTPX 11 would be used in conjunction with the additional radio sets to provide a multi-channel communication system.

All the basic features of the LVTPX 11 would be retained in the Command Vehicle. The basic hull structure would be modified to accommodate the additional antenna bases on the top deck. The tapped blocks necessary to mount the additional equipment and various small supports and studs necessary for cable routing can be welded to the hull.

A radio rack would be added to provide mounting for the additional radio equipment. The work tables will be provided for the radio operators, together with seating and safety belt facilities.

The additional radio equipment that would be added is a TCS radio set, an AN/JRC-22 set and an AN/ARC-27. Mounting facilities would also be provided in the main radio rack for troop-carried AN/PRC-8, 9, 10, or 47 radio sets. Additional equipment, such as control and junction boxes, can be added to make the complete communication system. Power and antenna lead cables would also be added.

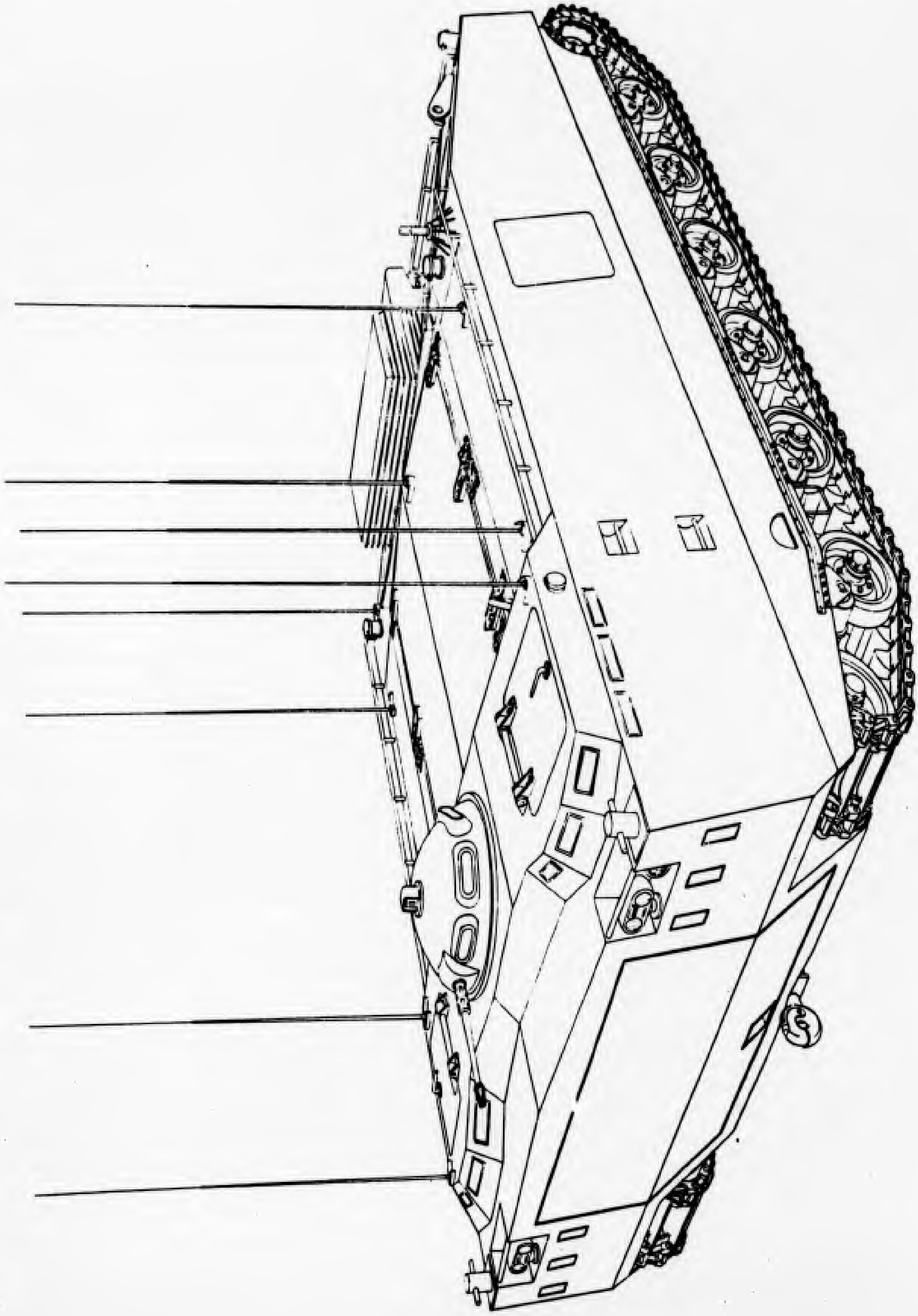


Figure 9-1 Command Vehicle

9.2 RECOVERY VEHICLE (Figure 9-2)

The recovery vehicle will be a lightly armored, full-track laying, low-silhouette vehicle, built on the tracked amphibian LVTPX 11 chassis. It will be equipped with a boom and auxiliary winch, a boom or hoist winch, and a tow winch for recovering disabled vehicles, and assisting in repairs of the vehicles under typical field conditions.

The cargo compartment will contain recovery equipment, and modifications will be made to the vehicle hull to insure an adequate structure for the installation of the boom and winch. The recovery vehicle will incorporate all the basic components of the tracked amphibian LVTPX 11.

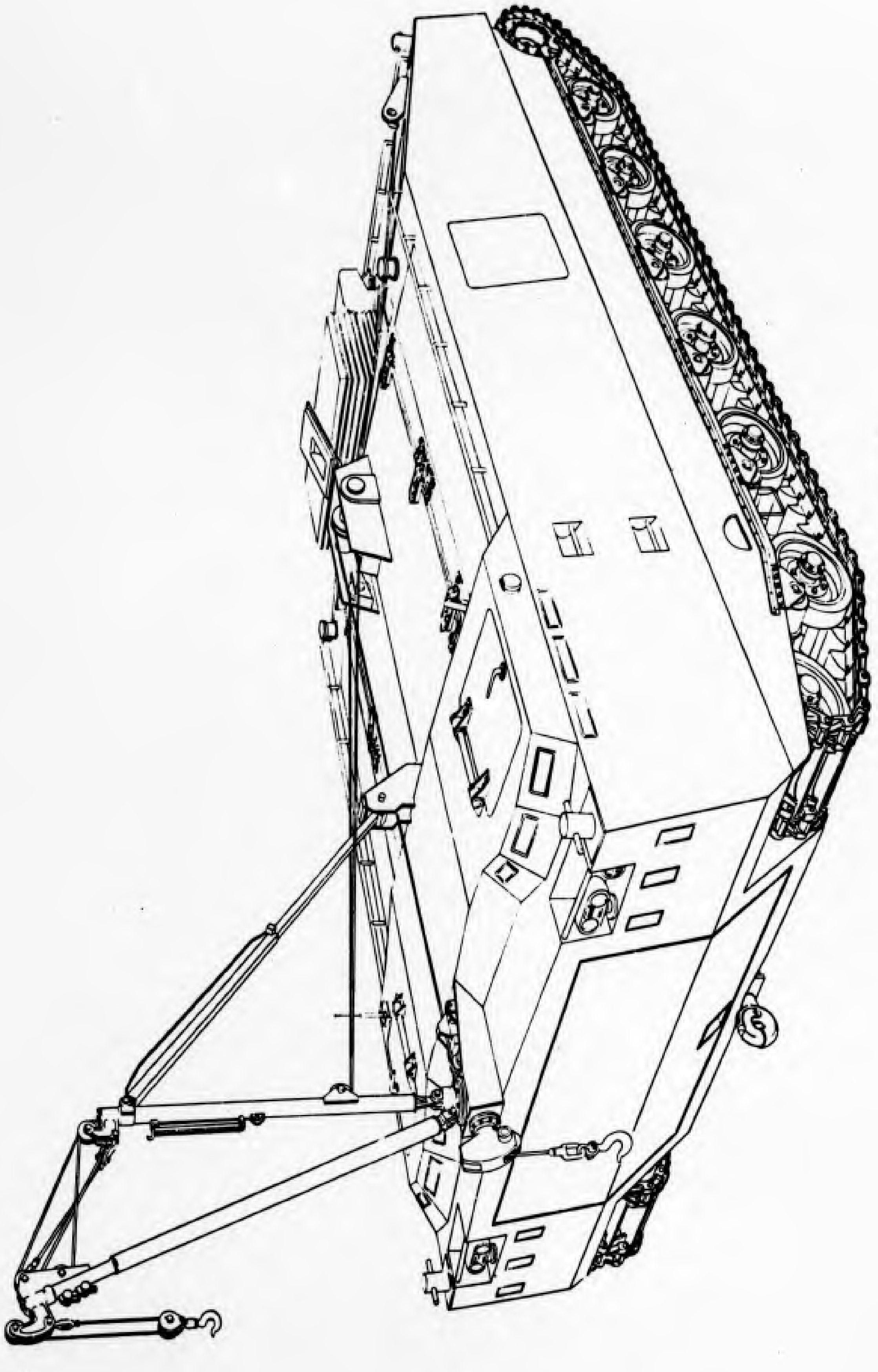


Figure 9-2 Recovery Vehicle

9.3 AMPHIBIAN FIELD ARTILLERY WEAPON VEHICLE (Figure 9-3)

The Tracked Amphibian converted to an Amphibian Field Artillery Weapon Vehicle would use a modified form of the XM102, 105 mm Howitzer as the main armament.

An analysis was made of the available weapons systems and the modified form of the XM102, 105 mm Howitzer was selected, based on the following advantages:

(1) Lightweight (1700 pounds, excluding mount) of the gun aids in vehicle floatation.

(2) Low trunnion reaction force (22,000 pounds with a 50-inch recoil), with the new recoil mechanism employed in this gun makes it suitable for this lightweight vehicle.

(3) While the optimum operation of the weapon requires a two-man firing crew, the gun may be fired by the gunner only.

A new mount must be designed for installation in a light armor turret. Provision has been made in this turret for 52 inches of gun recoil, as well as approximately 40 ready rounds in the turret, with an additional 30 rounds stowed in the hull of the vehicle. The necessary armor required to protect the extended recoil mechanism presents a frontal attack angle of extreme obliquity, thus providing the gun crew with maximum protection in the frontal area with a minimum of armor. The gun will be capable of firing 360 degrees in deflection and from 15 degrees depression to 60 degrees elevation. The turret race ring diameter is 88 Inches. While the extreme recoil distance of the weapon dictates a forward tunnion location, and a resulting out-of-balance condition, the weapon will be equilibrated to balance the elevation loads.

The commander-gunner will be stationed to the right of the 105 mm gun in an individually rotatable cupola. The cupola will be provided with 360 degree vision capability through newly designed wide-angle vision blocks, 4 inches thick. These blocks will provide protection against 30 caliber projectiles, fired at an angle of 90 degrees to the surface of the block. A 7.62 mm machine gun is mounted to the right of the cupola center and is utilized with an M28C Periscope.

The XM39 - XM40 Periscopes will be used to lay the Howitzer for direct fire. The XM39 Periscope will provide the gunner with binocular vision covering a field of view of 10 degrees at 7 power. The XM40 Periscope is interchangeable, with the XM39 instrument to provide the gunner with infrared vision for night firing and a visible light channel for use when conditions permit.

Indirect fire, employing a one-man system, may be accomplished by the gunner by establishing the deflection lay through the XM39 or XM40 Periscope and an elevation quadrant mounted to the gun. Employing a two-man system, the gunner will complete the deflection lay through the periscope and the loader will set the range elevation with the elevation quadrant.

The 50 caliber machine gun, mounted coaxially with the main gun, will be aimed with the aid of the XM39 or XM40 Periscope.

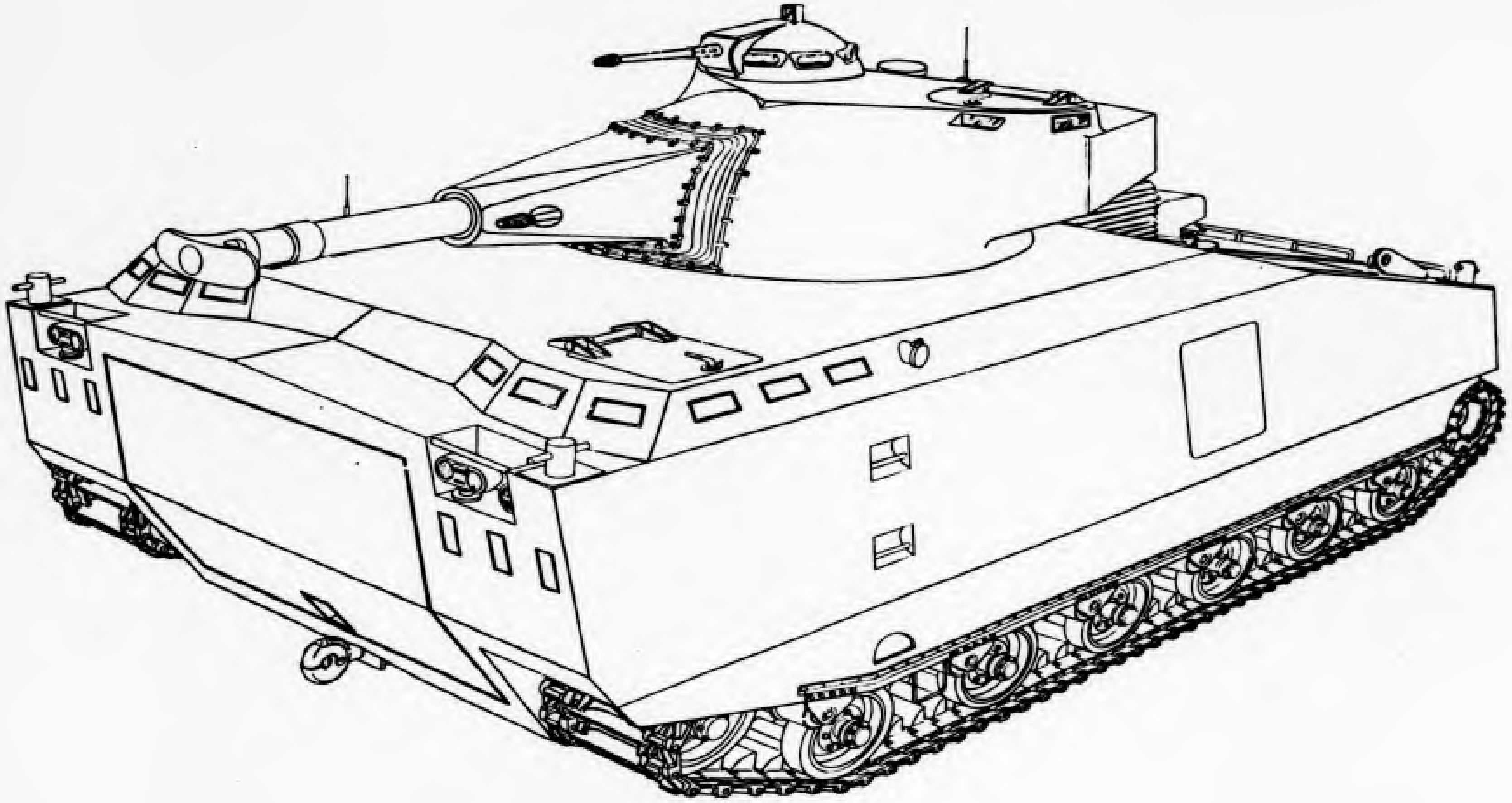


Figure 9-3 Amphibian Field Artillery Weapon Vehicle

9.4 AMPHIBIAN ASSAULT/ANTI-MECHANIZED WEAPON VEHICLE (Figure 9-4)

An Amphibian Assault/Anti-Mechanized Weapon Vehicle would carry two SS-11 or ENTAC wire-guided missile launchers mounted in the forward cargo section of the LVTPX 11, to be used against mechanized targets.

An analysis of the available weapon systems led to the selection of the wire-guided missiles to provide the vehicle with a hard target kill capability. The conclusion was based on the following advantages:

(1) Armor penetration of the shaped charge is superior to light artillery.

(2) First round kill probability, in the hands of a trained gunner, is higher than that of non-guided projectiles fired by trained gunners.

(3) Relative lightweight and small size of the missile enable the vehicle to stow a minimum of 48 ENTAC missiles.

(4) The missile and launcher may be mounted to the vehicle in such a manner that they could be removed from the vehicle for troop field use or fired remotely.

The missiles will be fired from elevated launchers concealed within the vehicle when not in use. The launcher will present the missile at a 10 degree elevation fire angle, with no provision for elevation or deflection, as the path of the missile is controlled by the gunner. When the missile is placed in the launcher rack by the loader and elevated to the ready fire position, the rack elevation mechanism will open two light armored hinged doors through a linkage system. These doors will protect the missile on the exposed flanks from small arms fire and fragmentation. At the same time the missile is elevated, an inter-lock will insure that the forward personnel hatch doors are secured. A blast deflector will be an integral part of the launch rack to protect the air intake housing, located aft of the missile launchers, from missile blast.

The missiles will be hand-elevated, one at a time, to increase reliability at a minimum cost. This is the most practical method of operation, as determined from previous installations. The loader will have ample time to load and ready a second missile while the gunner is firing the first.

Two sights were considered for gunner operation on guiding the missile -- the French APX Binocular Periscope and the U.S. Ordnance Corps M28C Monocular Periscope. The M28C Periscope was selected for its wider field of view, larger exit pupil, and increased elevation and depression of the line of sight. (The APX sight cannot be elevated or depressed in its present form.) The M28C Periscope will be a standard item on the basic vehicle for machine gun fire control, thus only one instrument will be necessary. The APX Periscope has a dual magnification of 1.6 power; however, it is not felt that this advantage makes up for the shortcomings of the instrument.

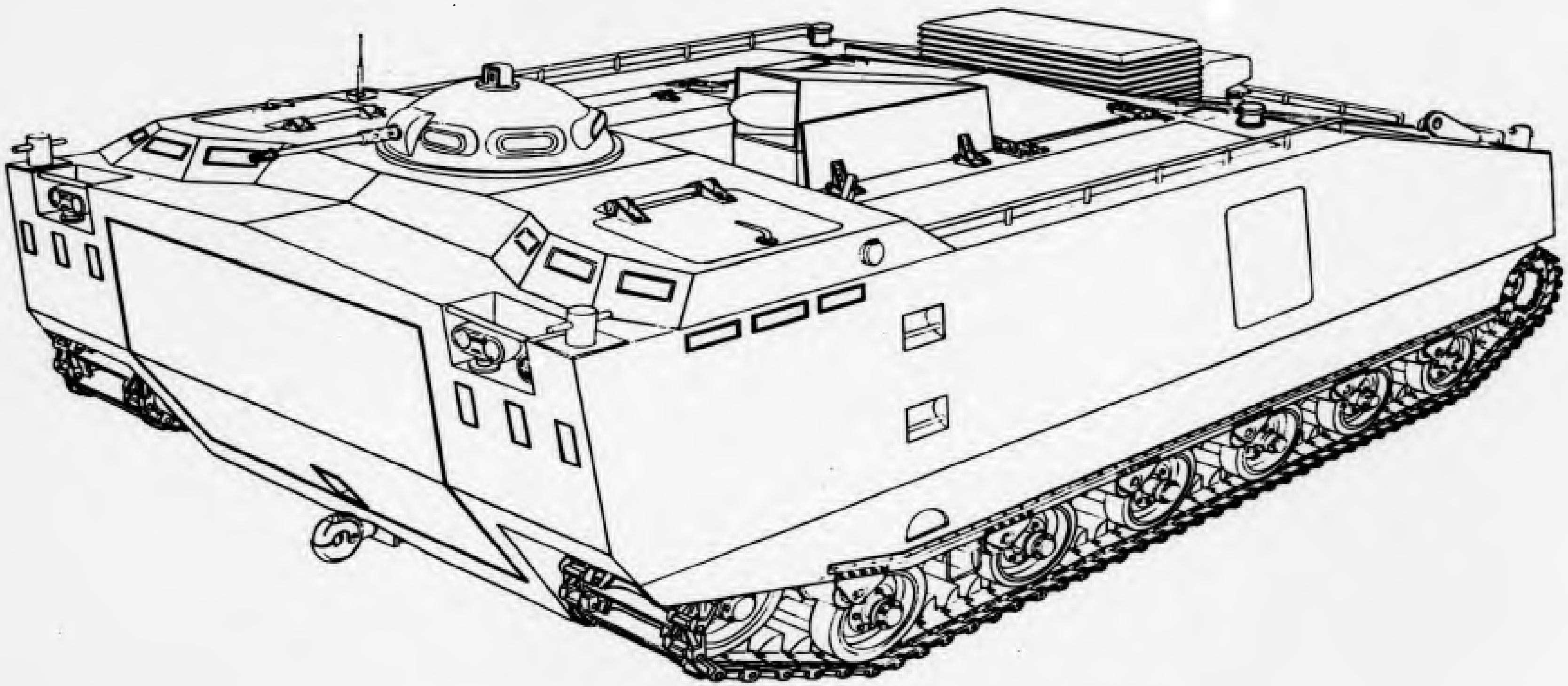


Figure 9–4 Amphibian Assault/Anti-Mechanized Weapon

9.5 AMPHIBIAN LIGHT AIR DEFENSE WEAPON VEHICLE (Figure 9-5)

The Tracked Amphibian can be converted to an Amphibian Light Air Defense Weapon Vehicle, using the Mauler Guided Missile System.

The Mauler System was selected, based on the following advantages:

(1) The Mauler is an available packaged, self-contained weapons system of a small enough size to be incorporated in this vehicle without extensive modification.

(2) The weapons system is of proven effectiveness, as determined by preliminary testing.

(3) The first round kill probability of this weapon exceeds that of any other weapon investigated for this type of aircraft defense.

(4) The unit may be removed from the vehicle or used when the vehicle itself is out of commission because the Mauler is a self-supporting system.

The basic vehicle hull design must be modified to accommodate the extreme width of the Mauler pad. This modification will be limited to increasing the hatch opening. The 12,000 pound weight of the Mauler System is the maximum cargo load of the vehicle. A further redesign of the Mauler System in this installation would result in advantageous weight saving and provide space for additional rounds. In addition to the 12 ready rounds of missiles, additional rounds may be stowed in the vehicle cargo area. The exact quantity of rounds that may be stowed is unknown, due to the classified size of the missile. This vehicle will also carry a 7.62 mm machine gun in the gunner's cupola for vehicle protection. The cupola will provide the gunner with 360 degree vision capability through the use of the newly designed wide angle, 4-inch thick vision blocks. The vision blocks and the light armor cupola will protect the gunner against fragmentation and small arms fire.

No special optical fire control is required for the Mauler System. Troop-carried hand-held binoculars may be of limited use. The 7.62 mm machine gun will be aimed and fired with the aid of an M28C Periscope.

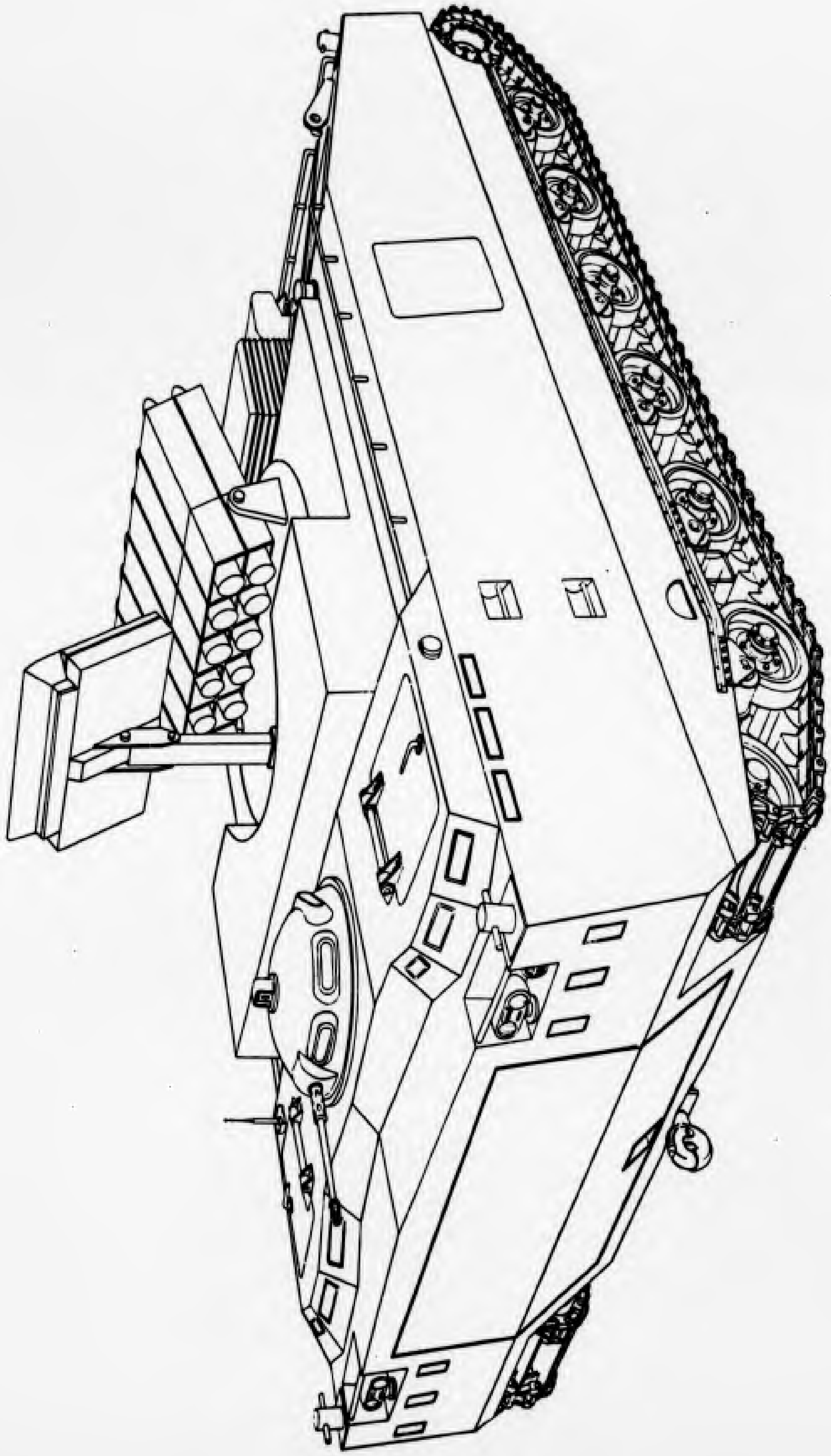


Figure 9-5   Amphibian Light Air Defense Weapon

9.6 ENGINEER MINE CLEARANCE VEHICLE (Figure 9-6)

The Engineer Mine Clearance Vehicle will be basically the tracked amphibian LVTPX 11, modified for use by assault troops in the reduction of beach obstacles and mine fields during amphibious assault operations.

The vehicle will be equipped with these special attachments. A hoist mechanism and carrying rack designed to carry vertically-displaced demolition line changes. A plow-shaped mine excavator with floatation tanks to give the neutral buoyancy in water, with elevation control and special vision devices. The blade controls would be located outboard of the vehicle driver, arranged to permit the operator to use power up or down, hold the blade in any desired position, or to allow the blade to float freely. A hydraulic winch with a reel would also be installed.

The side access and exit doors will be located forward in the hull on each side, replacing the standard escape hatches.

The cargo compartment will provide space for the engineer demolitionists and their equipment.

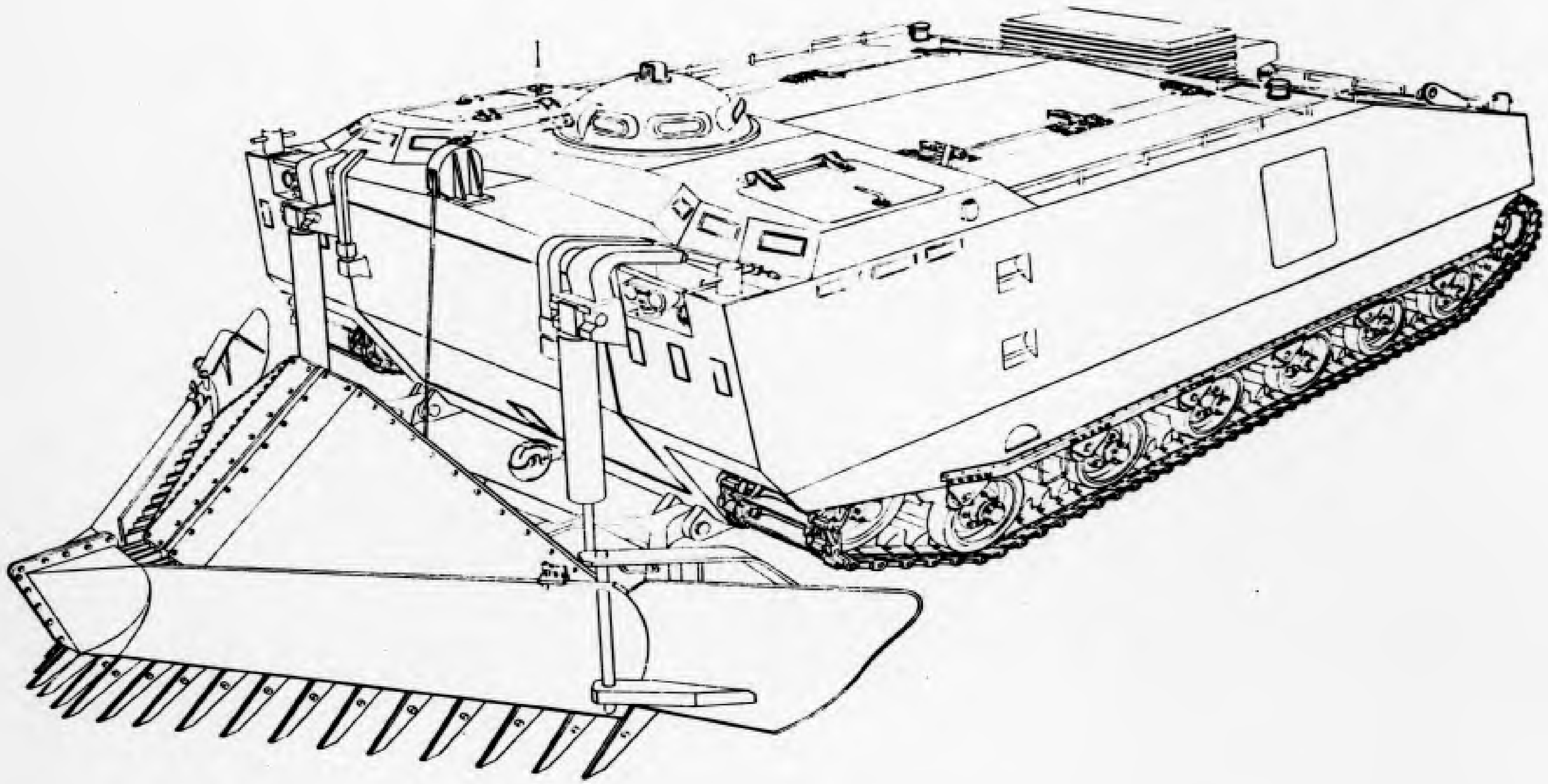


Figure 9-6 Engineer Mine Clearance Vehicle

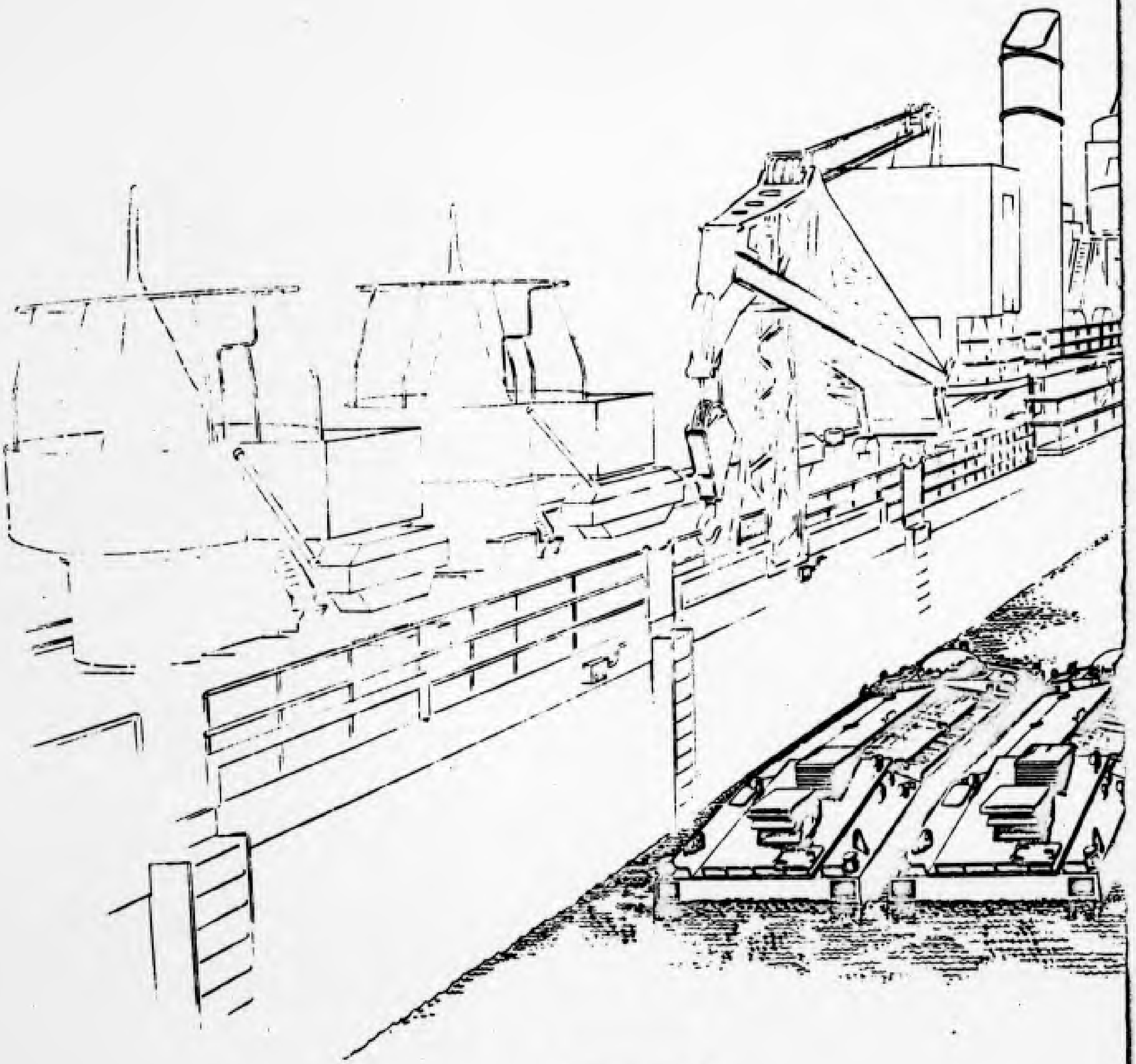
10. COMPATIBILITY WITH AMPHIBIOUS SHIPPING

We understand that the LVTPX-11 must be designed to be capable of being transported within and operated from transport ships, such as the LST, LSD, and LPD.

The Tracked Amphibian LVTPX-11, with a width of 10 ft. 6 in. and height of 9 ft. 8 in., and a gradeability of 70%, will have no problem negotiating the ramps of the following transport vessels:

		Minimum Ramp Clearance	
Type	Class	Width	Height
LST	1-1152	12 ft. 5 in.	11 ft. 11 in.
LST	1153	14 ft. 0 in.	13 ft. 10 in.
LST	1171, 1173-1178	15 ft. 0 in.	12 ft. 2 in.
LSD	1-27	42 ft. 8 in.	24 ft. 9 in.
LSD	28-35	48 ft. 10 in.	26 ft. 0 in.
LPD	1-	49 ft. 6 in.	28 ft. 0 in.

The number of tracked amphibians which can be transported in these special transport vessels is determined by the cargo deck area and vessel's tonnage capacity. The LST Class 1153, with a well deck of 30 ft. 6 in. width and 224 ft. length, is rated to transport fourteen (14) LVTP's. Twenty-one (21) of the LVTPX-11 can be carried in the same well deck area at less tonnage. The LSD, which is rated at transporting thirty-two (32) LVTP-5's could carry forty-four (44) LVTPX-11's and the LSD 28-35 Class, which has well deck that is over six feet wider than LSD 1-27 Class, can transport as many as fifty-nine (59) LVTPX-11's because it allows the LVTPX-11 to be stowed four abreast (see TD 106544). The LSD's referred to have the capacity of 1400 tons, are without modifications, and don't allow provisions for using the mezzanine deck or superdeck for the stowage of additional vehicles. If these items were considered, then possibly the total number of vehicles that could be carried would be about one hundred, but the number of combat-loaded LVTPX-11's which could be carried based on the 1400 ton capacity of the LSD's, would be 80 vehicles.



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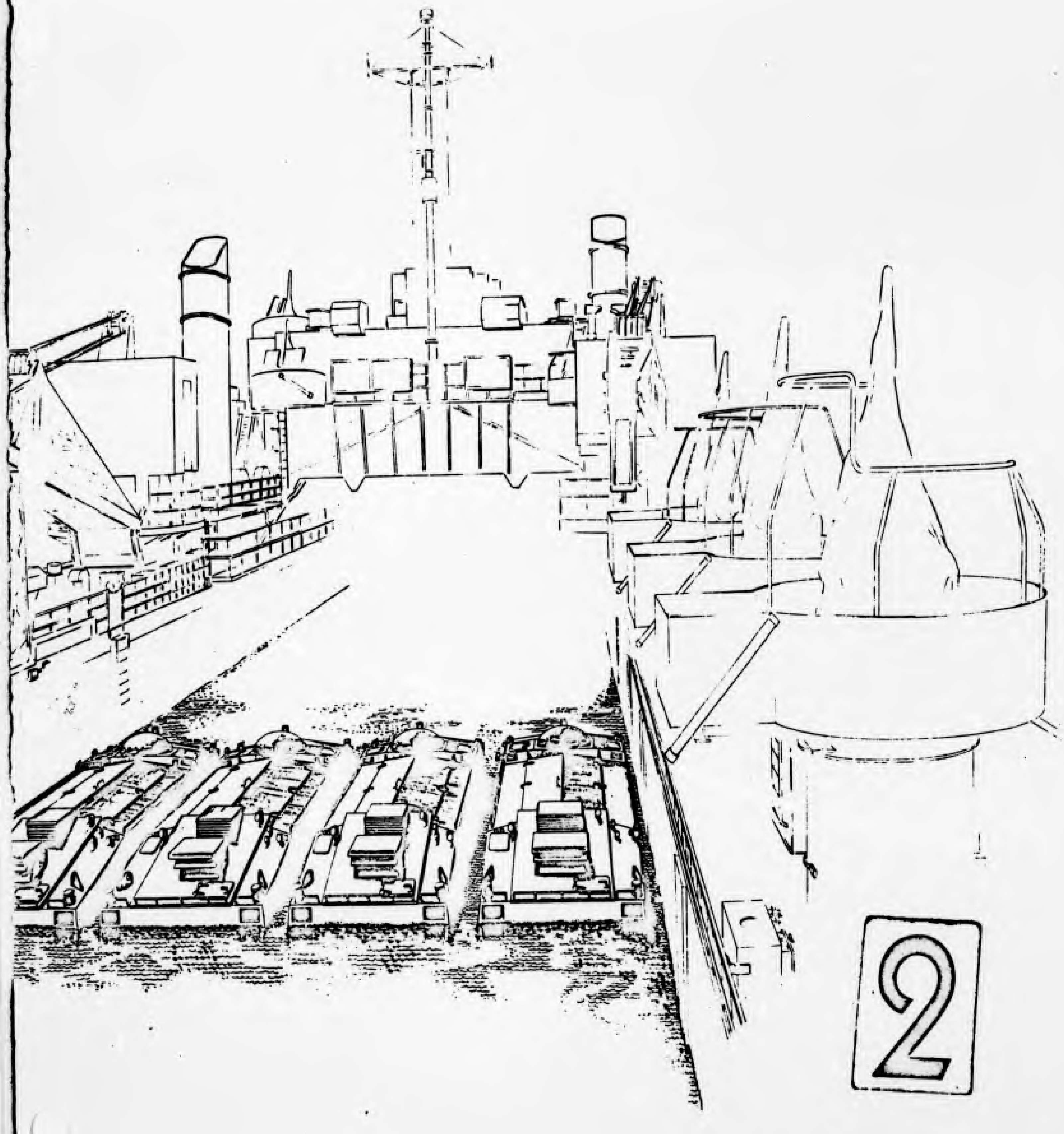


LVTPXII IN LSD
WELL DECK TD 106544

APPENDIX A

DEVELOPMENT CHARACTERISTICS FOR TRACKED AMPHIBIAN PERSONNEL AND CARGO CARRIERS

1. PURPOSE

a. Identification Item: The features, characteristics and capabilities stated herein are established as guides for the development of the Tracked Amphibian Personnel and Cargo Carrier.

b. Operational Employment and Effects on Present Equipment: This vehicle will be used to transport infantry, infantry weapons and cargo from ship to shore and to inland objectives and to provide fire support in amphibious operations. The vehicle is envisioned as the basic configuration for a family of vehicles that would replace the LVTP5, LVTP5 (CMD) (Command), LVTH6 (Amphibian Howitzer), LVTR1 (Recovery, and LVTE1 (Engineer Mine Clearance).

c. Organizational Concept: Units will be organized and equipped with various configurations of this carrier to provide sufficient troop, cargo and artillery lift to support the assault elements of a Marine Division and to provide support, air defense, mine clearance and obstacle clearance during beach assault and subsequent operations ashore.

2. COORDINATION

a. In any instance where attainment of a particular specification contained herein threatens the orderly progress or timely realization of this development, the contractor shall immediately advise the Bureau and shall make appropriate alternative remedial recommendations.

3. CHARACTERISTICS OF SYSTEM, EQUIPMENT, TECHNIQUE AND MATERIAL TO BE DEVELOPED

a. General Design Considerations

(1) Emphasis in design is to be placed on:

(a) Simplicity

(b) Long trouble-free life

(c) Ease of maintenance

(d) Economy of operation

(2) This vehicle will operate ashore 80% of its life and afloat 20% of its life. Optimum land performance is desired; commensurate with the requirements for simplicity, long trouble free life, ease of maintenance and economy of operation. Full consideration in design must be given to compatibility with economic manufacturing procedures for production line manufacture of quantities in excess of 100 vehicles The design should be such that any qualified manufacturer could reasonably be expected to enter into competitive bidding for quantity production The design should not incorporate features which could logically restrict production of the vehicle to the peculiar capabilities of a specific firm. In selecting major components such as power plant, power train, suspension and track special consideration should be given to new developments and materials. The machinery arrangement should provide enough flexibility to take advantage of newly developed items which prove to be desirable for incorporation in the prototype vehicles.

b. Special considerations.

(1) The basic hull, power train and suspension arrangement should be adaptable to other special purpose vehicles such as:

(a) Command vehicle.

(b) Recovery vehicle

(c) Amphibian Assault/Anti-Mechanized Weapon Vehicle.

(d) Amphibian Field Artillery Weapon Vehicle.

(e) Amphibian Light Air Defense Weapon Vehicle.

(f) Engineer mine clearance vehicle.

c. Operational characteristics.

(1) Payload - 10,000 lbs. exclusive of crew, fuel, OVE and OVM.

(2) Gross Vehicle Weight - Minimum practicable, target 35,000 lbs.

(3) Troop Capacity - 27 fully equipped Marines.

(4) Crew - 2 (crew chief - driver, assistant driver).

(5) Water Performance

(a) Highest forward water speed consistent with the power required for the specified land performance.

(b) Highest reverse water speed consistant with simplicity of power train and required maneuverability in the water.

(c) Surfability - capable of negotiating 10 foot plunging breakers, with a full payload, going both seaward and ashore.

(d) Stability - laterally stable under all conditions of loading and capable of righting from a 60 degree roll to port or starboard while fully loaded.

(e) Range - minimum 50 miles acceptable.

(6) Land Performance.

(a) Improved Roads - 40 MPH

(b) Cross Country - 20 MPH

(c) Gradeability - 70% forward slope and 60% side slope.

(d) Stability - must be capable of executing a 90 degree turn with full payload on a 60% side slope.

(e) Range - minimum 250 miles at 25 MPH

(f) Vertical wall - 36 inch

(g) Trench span - 8 foot minimum acceptable.

(7) Terrain and Climatic Limitations - The vehicle shall be capable of operating:

(a) In temperate, tropic and arctic zones.

(b) Over beach and desert sand, coral reefs, shoals, gravel, rocks, rice paddies, snow, ice, swamps, tundra, muskeg and through forested areas with trees up to 3 inches in diameter.

(c) In air temperatures ranging from 125° F. to minus 25° F. with relative humidity as low as 5% at air temperatures of 125° F., as high as 97% at temperatures from 80° to 85° F., and as high as 100% at all temperatures lower than 80° F.

d. Physical Characteristics

(1) Dimensions

(a) Length - Minimum Practicable

(b) Width - Minimum practicable; maximum allowable 10 feet 6 inches.

(c) Height - Minimum practicable

(d) Ground Clearance - 18 inches

(e) Approach and Departure Angles 30°

e. Required Characteristics

(1) Hull to be inherently bouyant

(2) A ramp or other means will be provided for ease in loading troops and cargo. Controls for the ramp to be situated so as to be operable by the driver.

(3) Troop seats desired. If used they shall be of the quick fold away type so that the vehicle can switch from the personnel carrier role to the cargo carrier role with a minimum amount of time and effort.

(4) Infra-Red night driving equipment is to be incorporated in the design.

(5) Headlights and tail lights are required (normal and blackout types). These lights are to be normally carried inside the vehicle and mounted in place when required.

(6) Navigation lights for operating in inland and coastal waters of the U. S. are required. Should be stowed in the vehicle and mounted externally when required.

(7) Means shall be provided to allow cargo and personnel loading alongside ships and docks

(8) Adequate means for escape from the vehicle shall be provided for both the normal and upside down condition

(9) Fuel - The amphibian shall be capable of using at lease one of the following standard fueld, gasoline combat, JP4, JP5, Navy Diesel fuel, Army compression fuel. A multi-fuel capability is desirable

(10) Maximum all around visibility under all driving conditions is required Similar provisions to be incorporated for the assistant driver if such a station is incorporated in design.

(11) Towing devices:

(a) Quick release tow hitches are required fore and aft These hitches should be operable by a crewman on the topside of the vehicle.

(b) A towing cable, mounted externally, will be provided

(12) Adequate bilge pump capacity shall be provided. To be continuous operating when the main engine(s) is operating. Bilge pump suction lines to be arranged so that a sever list to either side or a severe down by the bow (or stern) trim will not adversely affect the operation of the bilge pumps.

(13) Fire extinguishers:

(a) An adequate fire extinguisher system is required. Capacity to be dependent upon the risk (i e., type of fuel used gasoline, diesel, JP4, etc.) System to be operable from inside and outside the vehicle.

(14) Lifting devices. Lifting eyes and a lifting sling are required, capable of lifting the vehicle at gross vehicle weight. (The sling is not to be OVE, stowage provisions not required).

(15) Mooring Devices: Four mooring bits are required topside, one near each corner of the vehicle.

(16) Safety rails and Grab Handles: to be provided inside and topside for the safety of crew and embarked personnel.

(17) Cargo hold downs are to be provided, arranged in such a manner so as to provide a maximum of flexibility in securing various types of cargo.

(18) Ventilation of the Crew/Cargo compartment is required for the comfort of personnel during operation in tropic and temperate zone operation, both afloat and ashore.

(19) If highly volatile fuel is used consideration must be given to the requirement to scavenge those closed spaces where fuel leakage will present a safety hazard.

(20) Signal Searchlight: A portable searchlight with detachable color lens will be provided. This light will normally be carried inside the vehicle with provisions for exterior mounting when required.

(21) Inside lights, both white and blackout are required, to allow suitable illumination of the crew space, cargo/troop space and engine compartment.

(22) Boarding steps on each side of the vehicle are required. These steps to be recessed into the hull and to be of such design that sand, debris and extraneous matter cannot collect in them.

(23) Compass: A low cost, reliable, easily compensated compass will be provided.

(24) Cold weather starting aids and heaters for the comfort of embarked personnel are required but may be provided in kit form.

(25) Stowage brackets, cabinets and boxes are required for crew weapons, tools, equipment, on vehicle spares, armament and ammunition normally carried on the vehicle.

(26) Auxiliary starting aid shall be provided to allow one vehicle to start in similar manner.

(27) Refueling at sea: The vehicle must be capable of being refueled at sea in rough water without shipping water into the fuel fill opening.

(28) Armament: A turret with sighting equipment, mounting a single 7.62 mm machine gun (M73) will be incorporated in the design in such a manner as to give maximum close in support and a 360° field of fire. Provisions for stowage of 1000 rounds of belted 7.62 mm ammunition in boxes, plus a ground mount, for the gun is required.

(29) Armor - Sufficient armor to protect the crew and embarked personnel from small arms fire and shell fragments, commensurate with maximum vehicle weight specified.

(30) Chemical - Biological collective protective device for the crew similar to M8A1 or 2 is required. (Passengers will have their own protective masks).

(31) Passive Protection against the Blast, thermal and radiological effects of atomic explosions is required to the maximum extent possible, commensurate with specified vehicle weights.

(32) Controls. To be the simplest possible.

(33) Instruments - Warning devices as listed below are the minimum required:

(a) Speedometer (record miles also)

(b) Tachometer (record hours also)

(c) Voltmeter

(d) Fuel level gauge.

(e) Warning lights as required for high coolant temperature or low oil pressure.

(34) Non skid deck covering material inside and top side.

(35) Engine aspiration and exhaust system to be of such a design that dangerous amounts of carbon monoxide cannot collect in the crew and passenger spaces during vehicle operation.

(36) All hatches, closures, doors and topside openings will be sealed against the entry of water.

(37) All hatches, closures, doors and openings will be provided with a means of positive locking from the inside, except one that shall have a provision for locking with a padlock from the outside.

f. Characteristics desired but not required

(1) Designer to make maximum use of fiber glass, high impact plastics and other light weight materials for interior covers, separators, boxes, and bulkheads.

(2) Pivot steer.

(3) The transition from water to land (and vice versa) to be made with no requirement for the driver to actuate any special "water steer" switch or open or close any "Water or land air intake ducts" etc.

g. Landing Force Aspects

(1) This vehicle will be transported to the target area aboard amphibious shipping of the LST, LSD, and LPD types hence it must negotiate the ramps of the various ships or any future amphibious ships in both ahead and astern direction. Bureau sketch 022970 LST 1-1152 class vehicle clearance through bow, Bureau sketch 022971 LST 1153 class vehicle clearance through bow, LSD 1-27 class vehicle clearance through stern dated 31 December 1959, dimensions for clearances for tank and cargo stowage LST 1173 class, LSD 28-35 class vehicle clearance through stern dated 31 December 1959, and LPD 1 craft stowage information dated 6 January 1960 give pertinent dimensions.

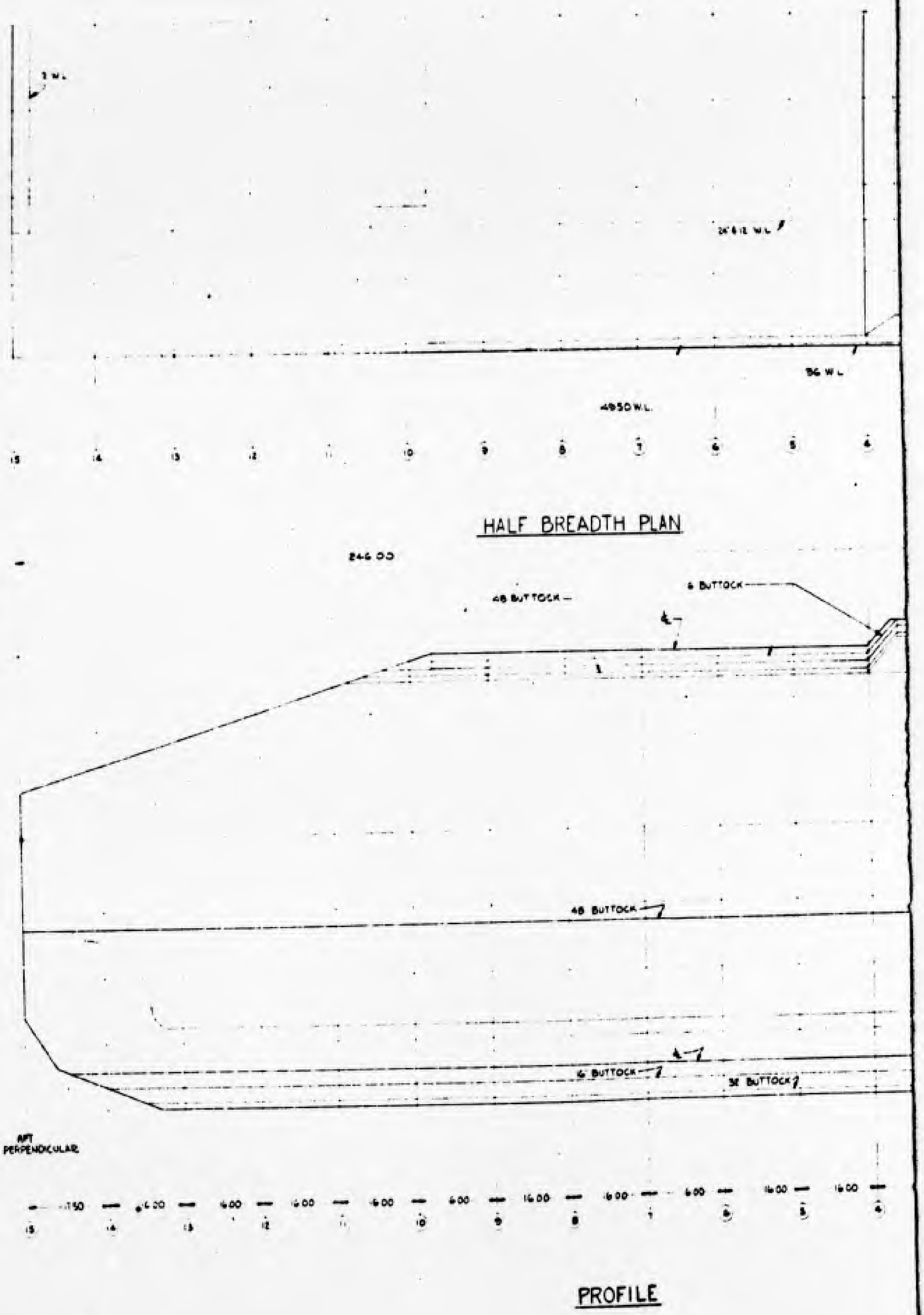
4. RADIO FREQUENCY COMPATIBILITY

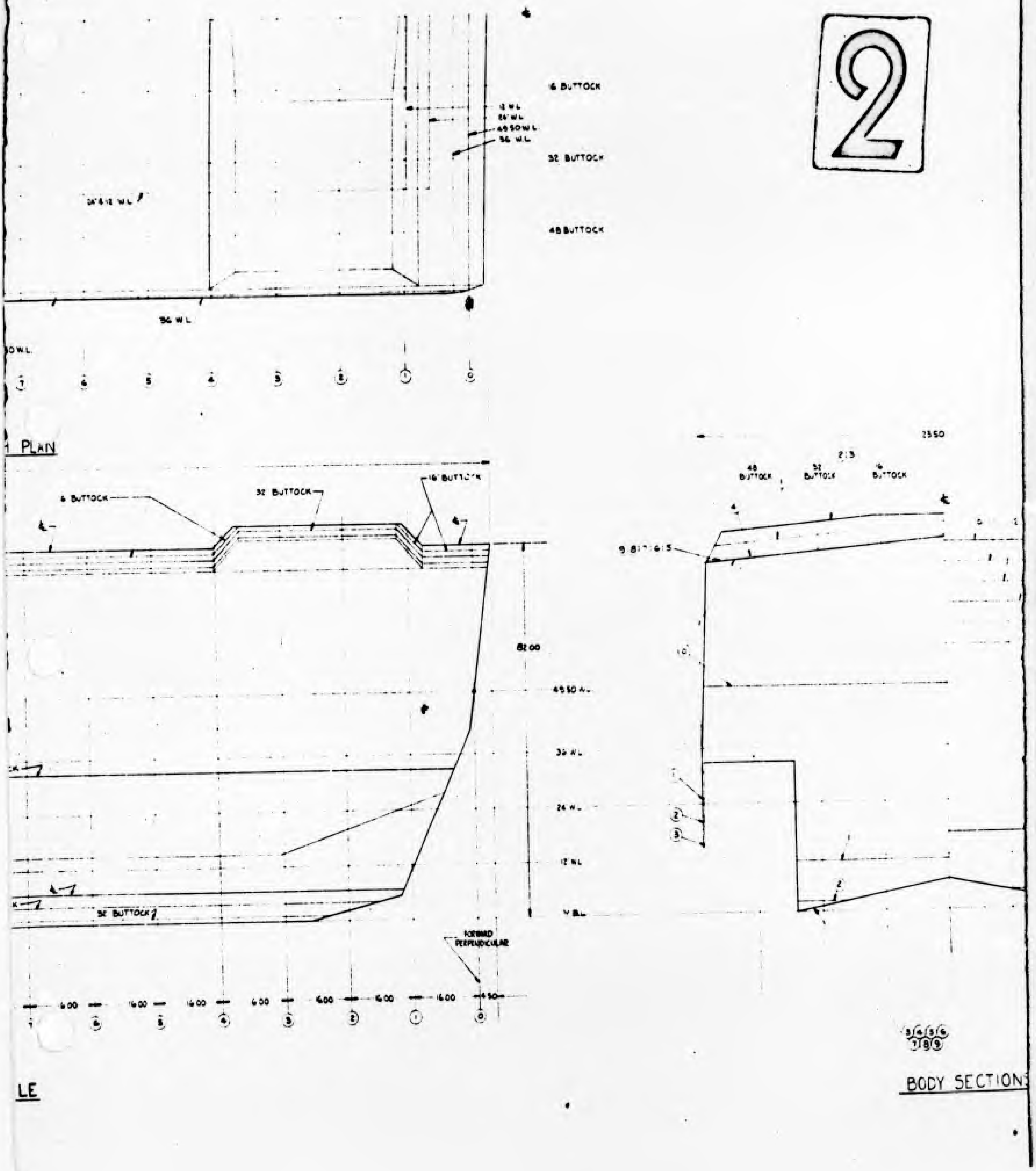
a. Provisions for permanent installation, in a waterproof box, of radio transmitter-receiver equipment must be made.

b. It is anticipated that the landing force radio equipment to be government furnished will be either the AN/PRC 47 or Collins Radio Model 618T. Approximate weight and space requirements are as follows:

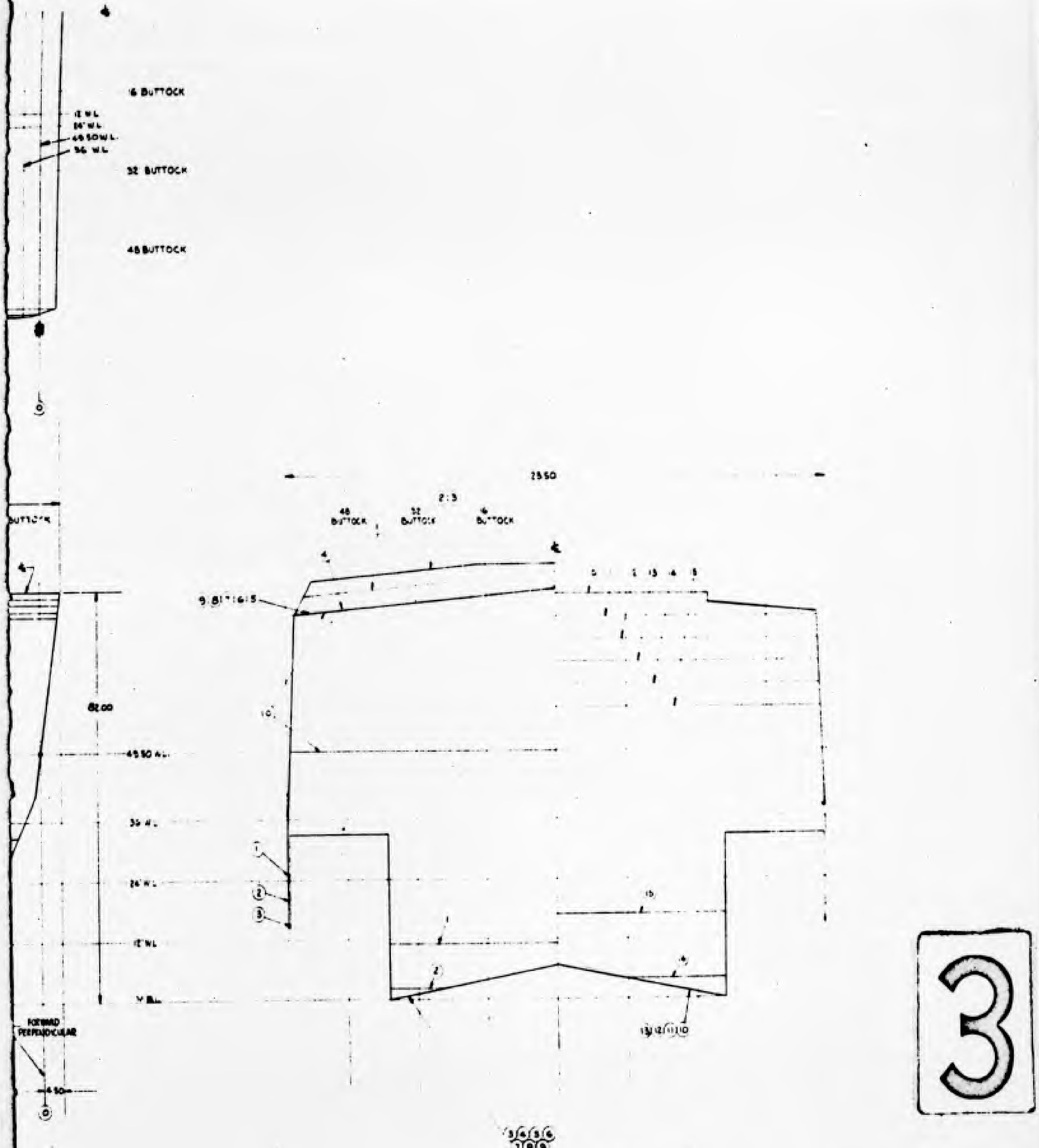
	Collins 618T	AN/PRC 47
weight	55 pounds	less than 55 pounds
height	12 inches	15 inches
width	24 inches	30 inches
depth	26 inches	18 inches

c. Antenna base mounts must be provided.


HALF BREADTH PLAN
48 BUTTOCK
AFT PERPENDICULAR
PROFILE



2
16 BUTTOCK
32 BUTTOCK
48 BUTTOCK
36 W.L.
PLAN
32 BUTTOCK
16 BUTTOCK
82.00
45.50 W.L.
36 W.L.
24 W.L.
12 W.L.
32 BUTTOCK
FORWARD PERPENDICULAR
1600
48 BUTTOCK
32 BUTTOCK
16 BUTTOCK
2350
LE
BODY SECTION



BODY SECTIONS HULL LINES TD 106546