ENGINEERING
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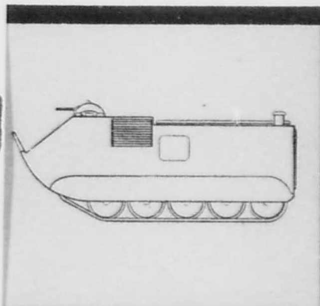
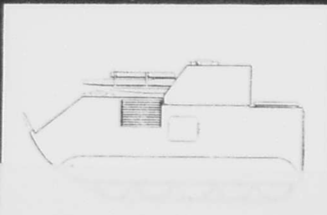
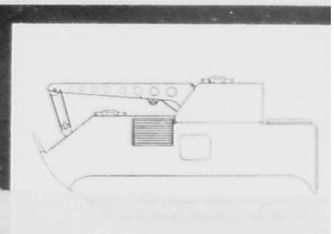
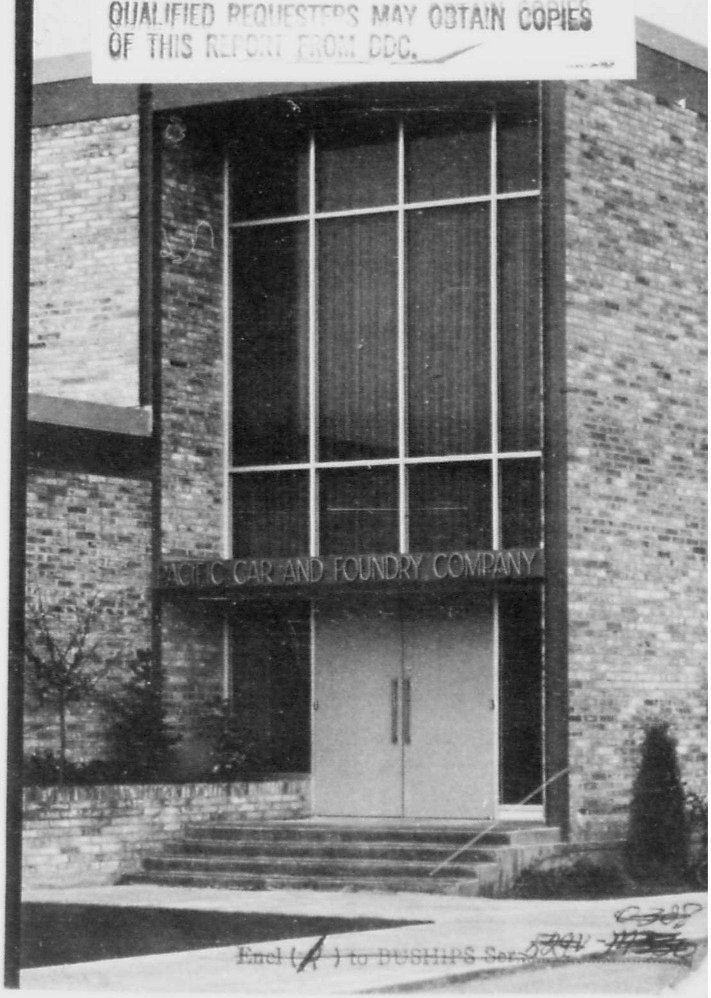






Concept Study Tracked Amphibian Personnel and Cargo Carrier LVTPX II



PACIFIC CAR AND FOUNDRY COMPANY

CONCEPT STUDY

DEVELOPMENT CHARACTERISTICS
FOR
TRACKED AMPHIBIAN PERSONNEL AND CARGO CARRIER

PREPARED FOR

BUREAU OF SHIPS DEPARTMENT OF NAVY

UNDER

CONTRACT NObs 4465

NOVEMBER 1961

PACIFIC CAR AND FOUNDRY COMPANY
RENTON WASHINGTON

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ABSTRACT

The fundamentally rectangular shape of the proposed vehicle is 22'-0" long x 10'-0" wide and 8'-0" high. Crew stations and power train are situated in the forward part of the vehicle. A ramp in the stern provides easy access for loading and unloading the cargo compartment while on shore. Large hatches in the top deck facilitate this operation while afloat.

Large diameter wheels with solid rubber tires run on light sectionalized band tracks, supporting the vehicle through torsion bar springs.

Motive power is supplied to the tracks by a multi-fueled compression ignition engine through a torque converter and a four-speed mechanical transmission and steering unit.

The indications are that the vehicle described in this concept would be superior in all respects to any existing amphibian. Careful and conservative analysis of data developed during the study shows that the desired land performance characteristics can be met and that the water speed will be in excess of 6.5 MPH.

The design is based upon the best available components that, in our judgement, will be available for prototype construction in 1963. In the event of unforeseen developments in the component field, it is felt that changes necessary to incorporate these items could be accomplished without compromising the over-all design.

INTRODUCTION

This engineering concept study is submitted in response to BuShips Contract NObs 4465.

Pacific Car and Foundry Company is aware of the importance of this development, particularly the need for a more compact, rugged, and mobile vehicle to serve the modern concepts of warfare. The Company has considerable background in the development of successful tracked and amphibious vehicles dating back to World War II and continuously improving manufacturing resources and engineering talent, assure successful and timely development of the LVTPX11.

This study is broken down into three major sections for easier perusal and understanding. Part I includes the general philosophy used in determining the over-all configuration, a general description of the design features, dimensions and characteristics, and an analysis of performance. Part II goes into the detailed design of the major components. Part III covers other components and concepts considered in the development of the detailed concept but not adopted, and the growth potential of advanced component concepts which might be incorporated as they are perfected.

To prevent the report from becoming cumbersome, presentation of detailed information has been made concise. All drawings and layouts pertinent to the concept study have been reduced in size and are included in the report. The original drawings or full size prints can be provided on request. The Company will also gladly provide additional discussion if necessary.

HISTORY AND PHILOSOPHY OF DESIGN

In determining the configuration of a vehicle that will meet the requirements set forth in the "Development Characteristics for Tracked Amphibian Personnel and Cargo Carrier", the following criteria were included in the development of the design:

1. Priority of land mobility over water performance.
2. Lightest weight and smallest size vehicle which will perform the required function.
3. Simplicity of design and best utilization of interior space.
4. Easy adaption to any of several special purpose vehicle configurations.
5. Utilization of advanced component concepts.
6. Easy inclusion of advanced developments of engines and power trains as they mature.

The minimum size of the vehicle was determined by summing up the interior volume requirements of components, crew, and required cargo space. Initial studies showed that a satisfactory vehicle would not exceed the gross weight limit of 35,000 pounds. Further studies showed how the vehicle could be made even more compact. The desirability of compactness was then assessed. The tactical situations of modern warfare demand maximum mobility, and experience indicates that mobility is best obtained with the lightest and smallest equipment commensurate with the tasks to be performed. This combination also produces better ballistic protection because, for a given weight, the smallest envelope will provide the thickest armor and smallest target. From the logistics standpoint, the smallest vehicle is always desirable. Finally, it was found that care must be exercised in developing a vehicle of this limited weight with relatively large cargo volume requirements, for it could suffer from a problem not usually associated with armored amphibious vehicles, that of excessive reserve buoyancy. This condition could seriously affect the capabilities of the vehicle when negotiating surf. Compact design minimizes this tendency. Therefore, mobility, armor, logistics, and surfing capability all required the most compact vehicle design.

With regard to hull form and stability, our consultants, Hydronautics, Inc., have developed some novel devices which hold promise of improved marine propulsion characteristics.

In determining the configuration of a vehicle that will provide improved land mobility without sacrificing water performance, a number of hull and running gear arrangements were investigated. Many forms of track, both conventional and unconventional, were scrutinized to make sure that the proposed track would perform satisfactorily in all media and provide a useful service life with minimum maintenance. Recently proven developments have been incorporated in the design that will increase land mobility over previous LVT's, as well as improving water propelling effectiveness.

The internal arrangement of the vehicle was developed concurrently with the determination of the hull configuration. Components location, in order to best utilize the space available, was largely governed by:

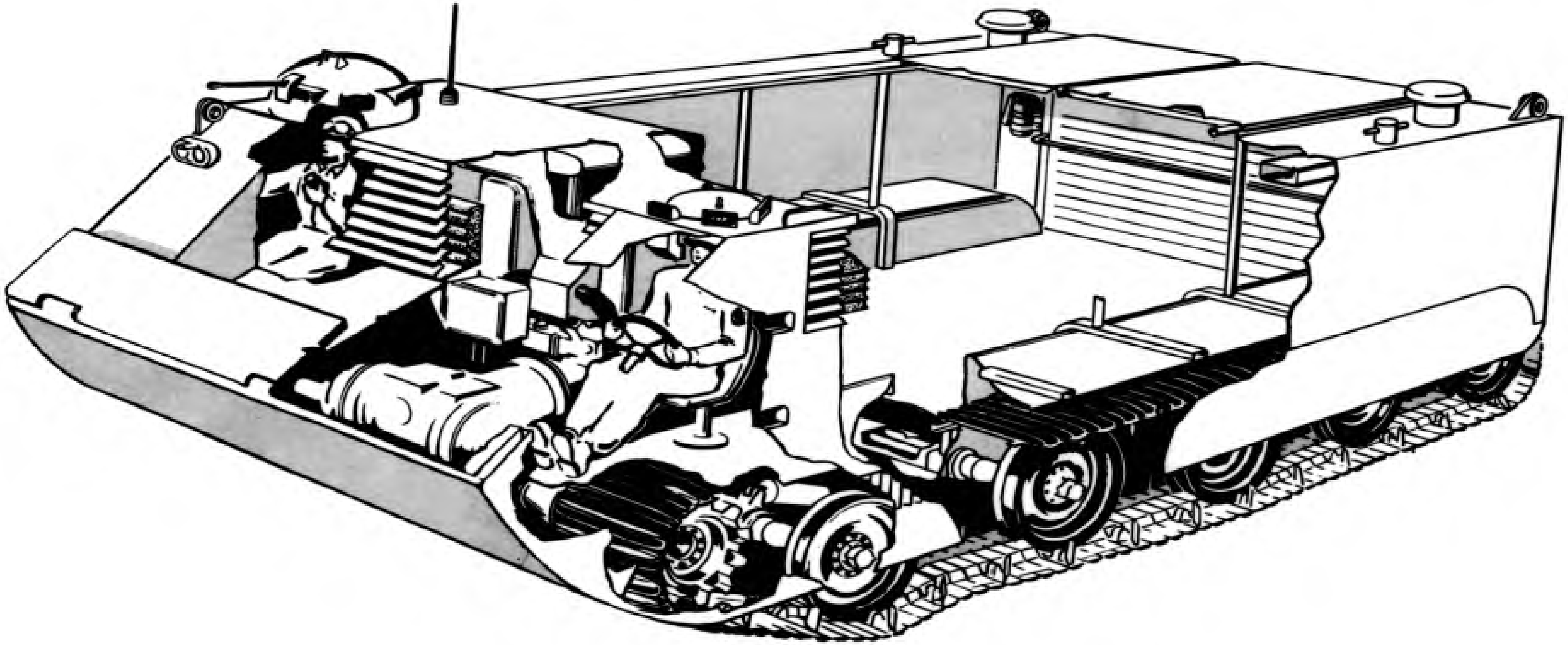
1. Situating the power package to provide the simplest and most direct power path to the tracks with sufficient conservatism to permit rearrangement for accommodating other equipment of comparable power that may be developed.
2. The known requirements of adaptions to produce special purpose vehicles.

The optimum internal arrangement produced certain weight and balance problems. These were, by careful consideration, overcome by slight adjustment of the arrangement without compromising the vehicle operating characteristics.

Two major areas of development in tracked amphibious vehicles are the power plant and the suspension and tracks. The power plant selected uses the most advanced components presently under development that should be available within the allowable time schedule. Advanced planning of this nature demands careful evaluation of all equipment that will be introduced in the future. From these must be selected the items which show the most promise of being satisfactory hardware at the required time. Gas turbines, electric drives, and hydrostatic drives were among other schemes that were scrutinized during this investigation. In all cases, their rejection was based upon examining their current development status and projecting this forward to the time when production vehicles will be required. This is not to say that these can be dismissed; to the contrary, all developments must be followed closely. Developments which prove concurrent with the proposed configuration should be considered.

It has been the Company's experience that considerable operational testing of a vehicle is necessary to obtain an optimum suspension and track system of the required durability. That is to say, the vehicle must not be a test bed for an untried engine, while only the vehicle can be used to test the track-suspension system. The proposed engine and power train are in a state of development commensurate with this necessary track testing.

PART I



PROPOSAL			
LVTPXII TRACKED AMPHIBIAN PERSONNEL AND CARGO CARRIER			
11/8/61	WHB		
	6496 - 39		
PACIFIC CAR AND FOUNDRY COMPANY			
RENTON, WASHINGTON			

GENERAL DESCRIPTION OF DESIGN

Configuration

In keeping with the intent of making the vehicle as compact as possible, over-all dimensions have been limited to 22' long, 10' wide, and 8' high. The configuration is not unlike other tracked vehicles; the general rectangular shape is modified to suit its purpose and operational environment. The driver, gunner, and all machinery are centralized in the bow, providing the most uncomplicated and direct means of controls for operating the vehicle. This arrangement leaves the entire stern portion for carrying space and provides easy adaption to special purpose vehicle configurations.

Hull

The bow shape of the vehicle resulted in a compromise of the configuration desired for best land visibility and water performance requirements. The underside of the bow starts up in a "scow bow" configuration that has proven to have the least resistance through the water within the limits of the confined hull shapes possible. Forward of the driver's compartment, the hull has negative sheer which improves visibility for land operation, provides a convenient and protected area for the engine cooling air intake, and improves ballistic protection because of the obliquity of the plate. A vane is attached across the entire width of the vehicle at the corner formed by the junction of the "scow bow" and the front sheer plate. This is hinged forward to increase performance during water operation and is pulled back against the front sheer plate to improve forward visibility during land operation.

The all-welded hull will be fabricated of aluminum armor (MIL-A-46027), which provides the best ballistic protection per pound of weight for practical hull construction. In line with current practice, mechanically supported joints will be used where design is governed by ballistic protection rather than by mechanical strength.

The top of the vehicle has cargo hatches which provide an unobstructed opening of 6 ft. wide by 10 ft. long.

A hydraulically-operated ramp 6 ft. wide by 4 ft. high which can be lowered to the ground is located across the stern. The ramp is hinged for uninterrupted entrance into the cargo deck to facilitate cargo loading and unloading.

A single plastic fuel tank of 250 gallons capacity will be installed below the cargo deck. This will supply fuel for approximately ten hours of full throttle operation.

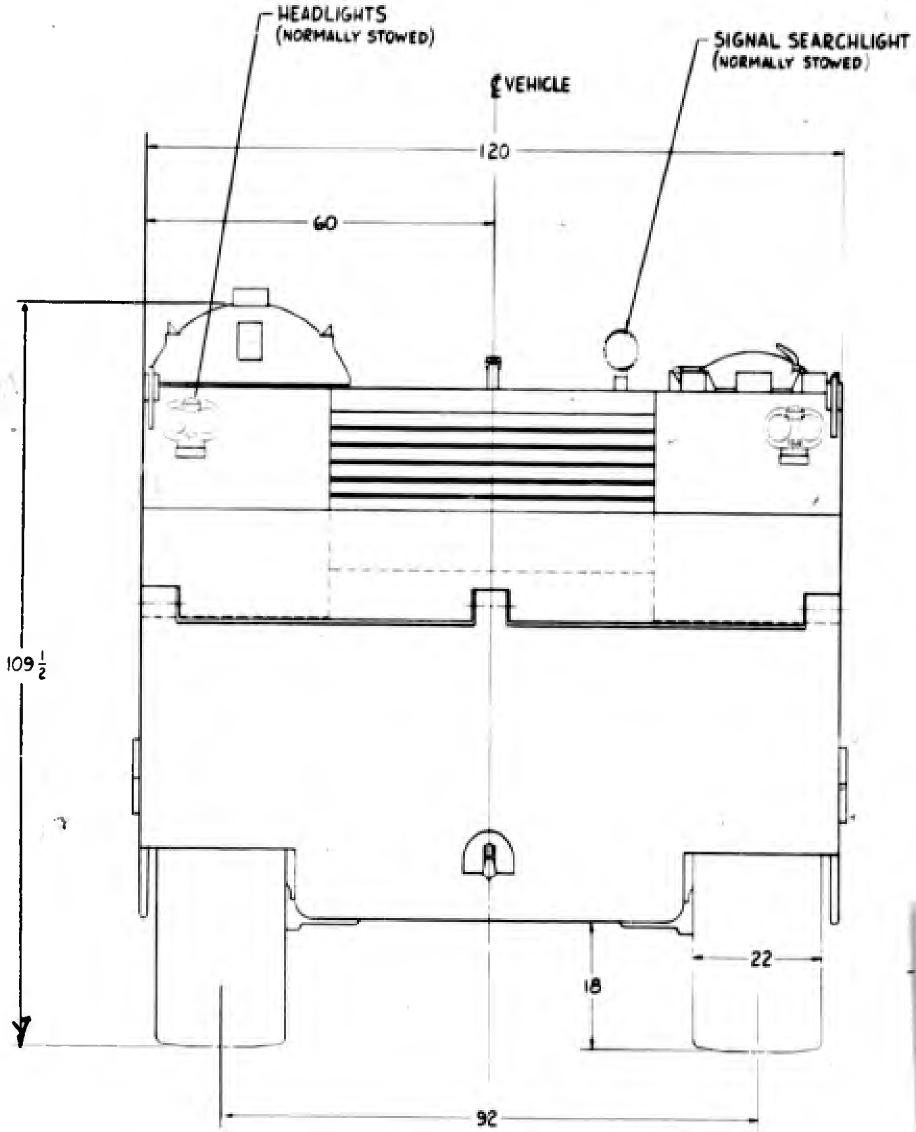
Crew Compartment and Controls

The driver is located in the bow in an area formed by the front sheer plate, the side plate, and the engine compartment panel. A conventional cupola with a top opening hatch and periscope is used. Controls for operating the vehicle are conventional for a tracked vehicle and are used in the same manner, without change, for both land and water operation.

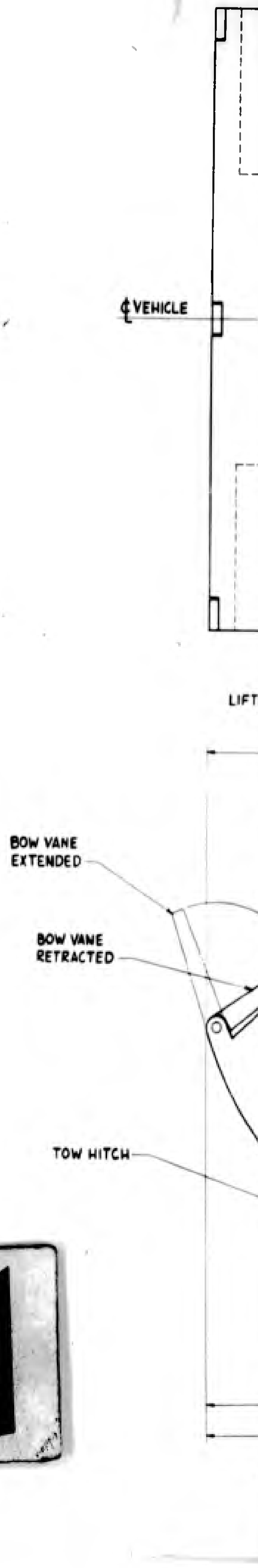
A machine gun turret similar to the turret on the LVTP5 can be accommodated in the opposite side of the machinery compartment in an area similar to the driver's area.

Suspension, Tracks, and Drive

A front drive sprocket has inherent advantages over a sprocket in the rear, apart from providing better interior utility. The track is free

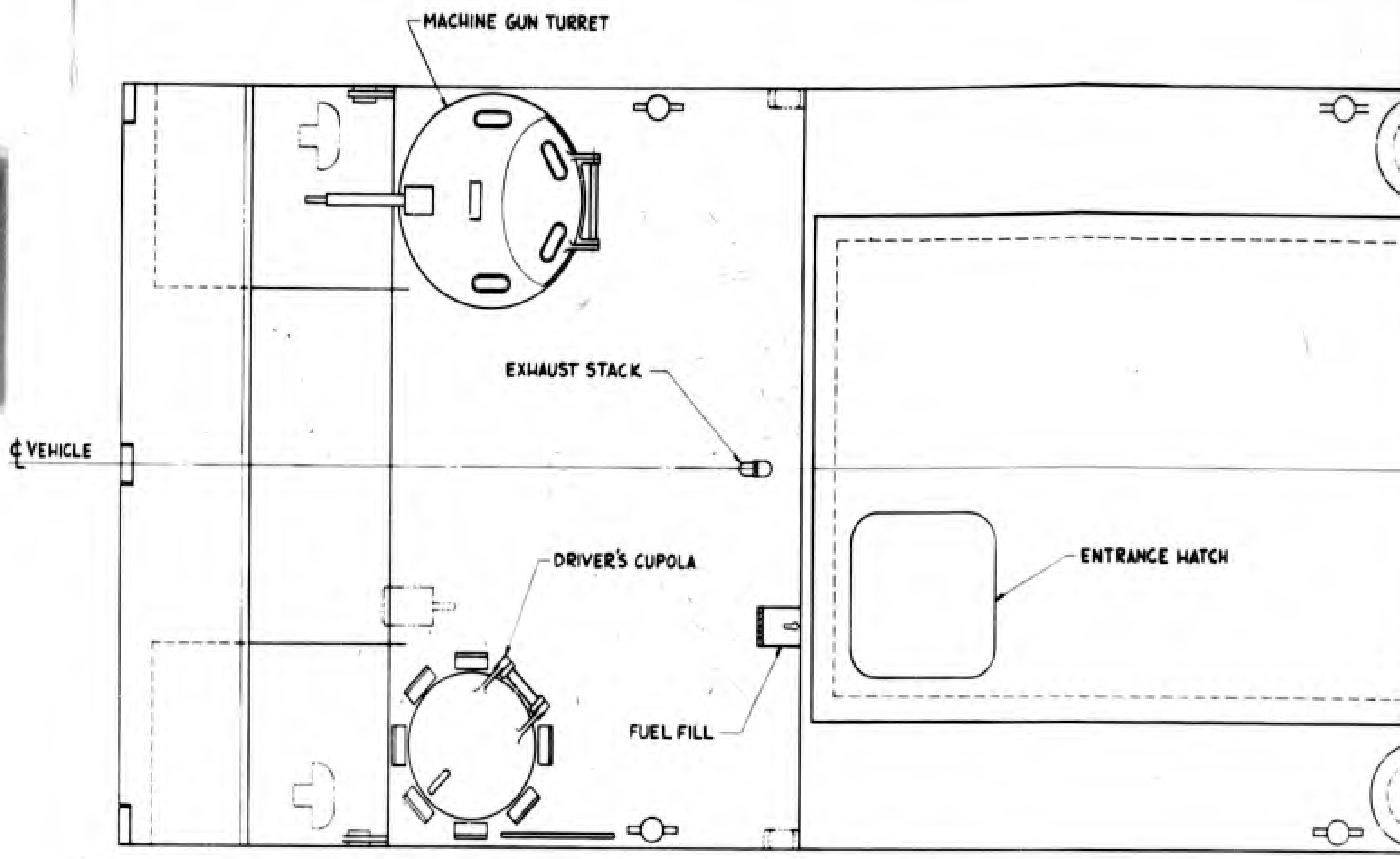

HEADLIGHTS
(NORMALLY STOWED)
SIGNAL SEARCHLIGHT
(NORMALLY STOWED)
℄ VEHICLE
120
60
109 1/2
18
22
92

FRONT VIEW

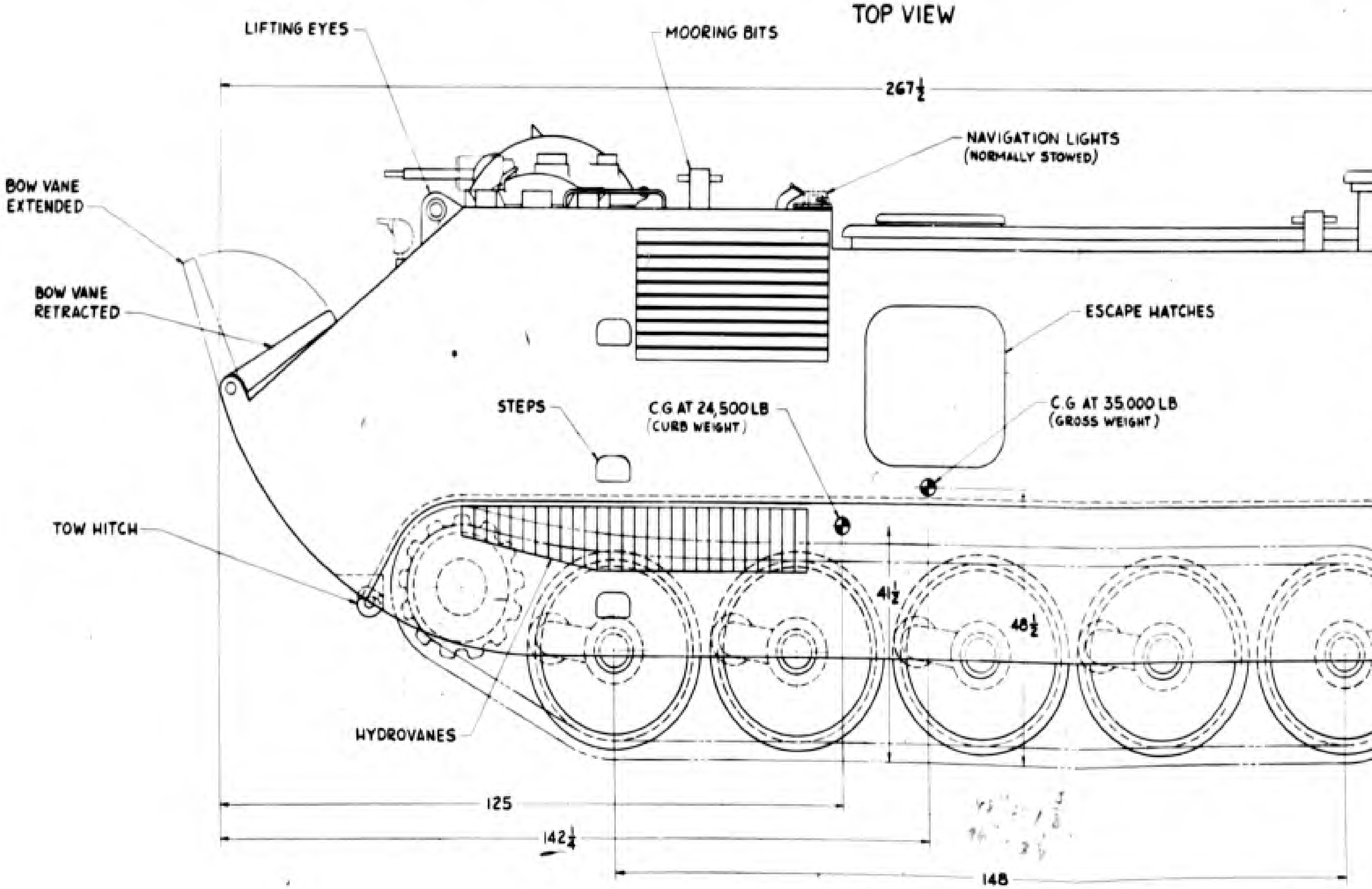

℄ VEHICLE
BOW VANE
EXTENDED
BOW VANE
RETRACTED
TOW HITCH

1

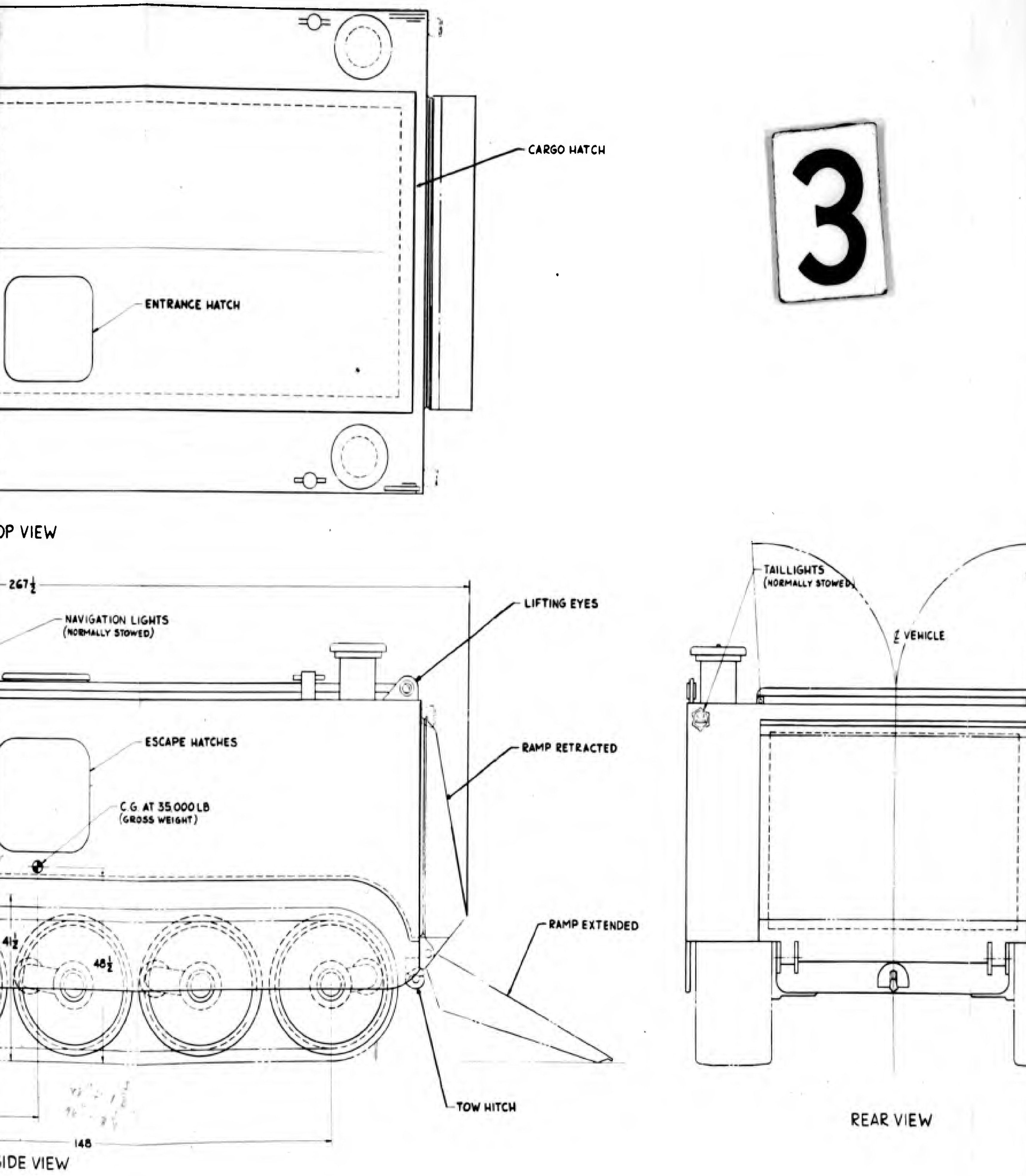
2



TOP VIEW

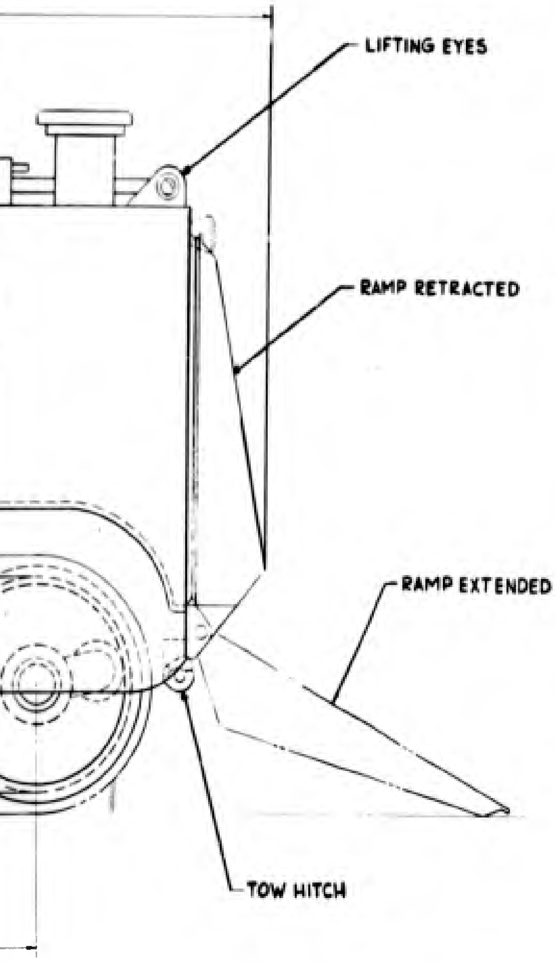


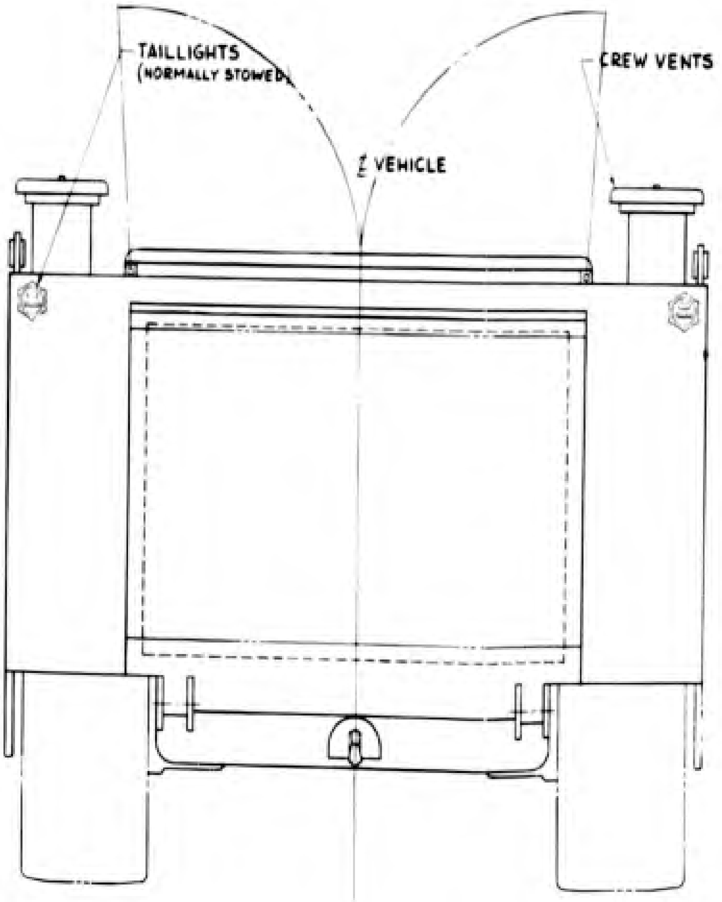
LEFT SIDE VIEW


CARGO HATCH
ENTRANCE HATCH
OP VIEW
267½
NAVIGATION LIGHTS
(NORMALLY STOWED)
LIFTING EYES
ESCAPE HATCHES
RAMP RETRACTED
C.G. AT 35,000 LB
(GROSS WEIGHT)
41½
48½
RAMP EXTENDED
TOW HITCH
148
SIDE VIEW
TAILLIGHTS
(NORMALLY STOWED)
℄ VEHICLE
REAR VIEW

3

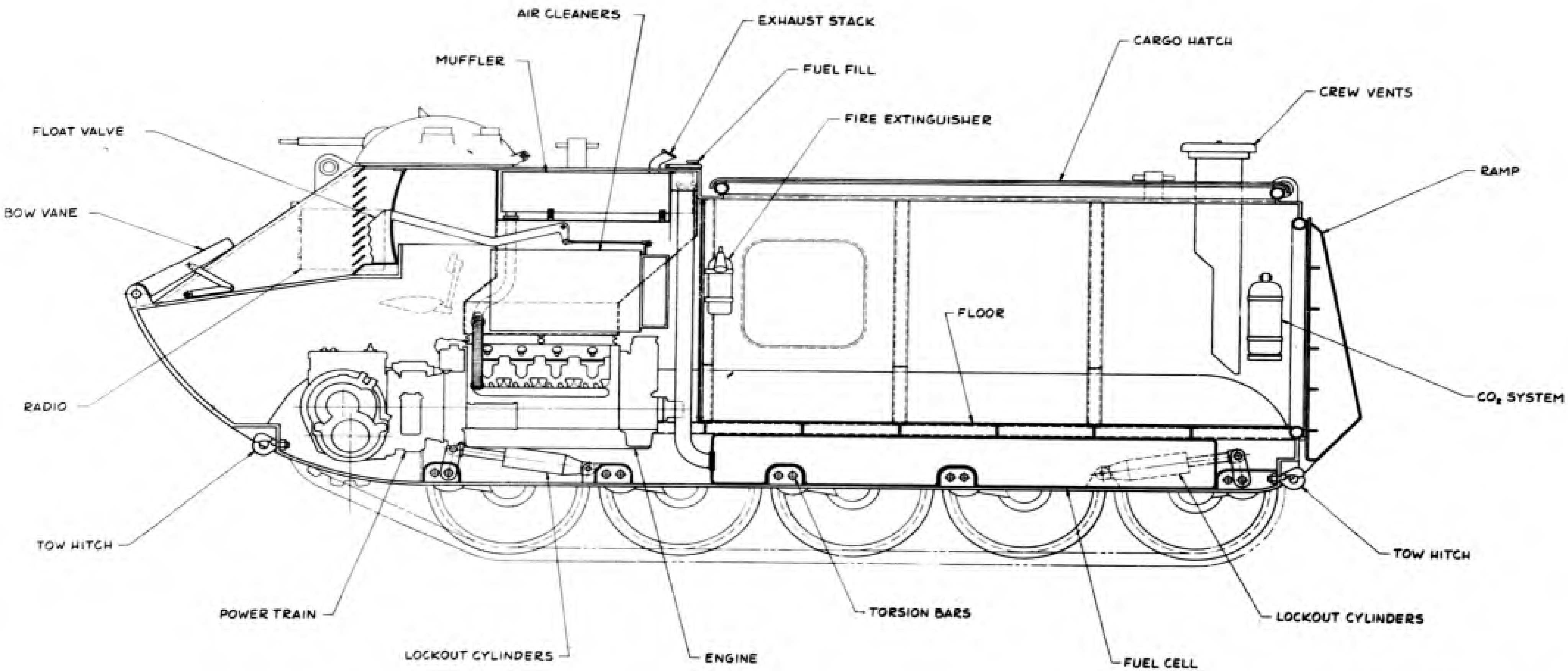
4





REAR VIEW

PROPOSAL	
GENERAL ARRANGEMENT LVTPX11	
3 NOV 1961	
1/8	6496-28
PACIFIC CAR AND FOUNDRY COMPANY RENTON WASHINGTON	



LONGITUDINAL SECTION

PROPOSAL
GENERAL ARRANGEMENT LVTPXII (LONGITUDINAL SECTION)
9 NOV 1961 · 1/8 · 6496-42
PACIFIC CAR AND FOUNDRY COMPANY

of dirt and rocks as it enters the sprocket and its weight helps seat it in the sprocket, eliminating the tendency of the track to skip over the sprocket teeth. With the front drive sprocket, large (32 in. dia.) road wheels are used which: (1) reduce the number of road wheels - in this case to five on each side, (2) eliminate the rear idler and return rollers by having the track wrap around the rear road wheel and return over the tops of the road wheels, and (3) inherently improve land mobility.

Of the types of suspension investigated, the road arm cantilevered from the hull and sprung on a steel torsion bar proved most suitable for energy absorption, physical arrangement, and minimum weight. Both front and rear road arms have an additional lever secured, on the inside of the hull, to provide a means of attaching a Company-developed Suspension Lockout System. This arrangement performs two important functions and is controlled by one simple valve that is operated by the driver. Under normal circumstances, the lockout cylinders operate in the same manner as a shock absorber with an integral bump stop. When necessary, opening the control valve causes a spool to close the ports in the main shock absorber piston, which effectively restrains movement of the piston to the point that the road arm acts as a solid axle. In this condition, the vehicle becomes a very stable, yet mobile, platform for weapons and cranes. The Lockout System is a successful component on both M107 (175 MM Gun) and M110 (8" Howitzer) tracked vehicles now entering full production. It is also incorporated in the T120E1 (Recovery Vehicle) and is responsible for increasing the lifting capacity by almost 30%.

Before selecting the track for this concept, three general types were scrutinized to determine which would be most suitable for this particular vehicle. Basically, the types can be described as:

1. Sectionalized open band track similar to that used on the M116.
2. Rubber bushed link and pin track similar to a track designated T132 by Ordnance Tank Automotive Command and used on the M107 and M110.
3. Sectionalized closed band track ("Speed-Trac"). This is a recent development.

The design of each of the tracks incorporates different principles; each has its advantages and disadvantages. The three tracks are interchangeable with minor suspension system modifications, and a description of the three designs is included for further consideration in the development of the vehicle.

Engine and Power Train

The Lycoming AVM-625 V-8 air-cooled multi-fuel compression ignition engine, currently under Ordnance-sponsored development, has been selected as the best power source for the LVTPX11.

The engine operates on diesel, compression ignition, JP4, JP5, and combat gasoline.

Lycoming has been developing a smaller version of this engine for seven years and will have the 8 cylinder engine operating on a test stand before the end of 1962. Being a 2 cycle engine, it is basically very simple and operates without valves and their associated parts. Considerable effort has been concentrated on ease of maintenance and repair. The fuel injection system will accept the various fuels without adjustment or modification. The engine's integral cooling fans have additional capacity for cooling the transmission and are thermostatically controlled to provide fast warm-up and maintain normal engine temperatures during cold weather operation. A glow plug in each cylinder has been successful in starting the engine at temperatures considerably below the minus 25° requirement.

The gas turbine is slightly lighter than the Lycoming engine, and occupies less space, but has the serious disadvantage of requiring large amounts of combustion air. The Bureau-sponsored installation of a gas turbine in an LVTP5 should be followed closely, and consideration should be given to the future adoption of this unit as the prime mover for the LVTPX11. This can be readily accomplished because none of the framing in the engine compartment forms part of the major load carrying members of the hull.

Consideration has been given to alternate power train components. The Allison XTG-250 power train is utilized, which has a parallel development, the XTG-300, an interchangeable unit for turbine drive. The XTG-250 is well suited for this application and, without modification, gives performance and control that are required for land and water operation. The physical arrangement of the XTG-250 adapts very well to the vehicle; it is cradled in the sprocket drive housings, thereby requiring no special mounts. The unit includes a torque converter and mechanical transmission with four forward and two reverse gear speed ranges with converter lockout in each forward range. It features three types of steering: (1) gear steer in the upper ranges where one track is slowed down to steer, (2) clutch brake in the lower ranges where one track is stopped to steer, and (3) pivot steer where one track runs forward, the other in reverse.

The multi-speed ranges and converter lockout feature promise to make this the most efficient and best performing unit of this type ever developed.

Cooling Air System

Effective means of cooling the engine and power train during the water operations have, in the past, involved numerous and relatively complex components to provide adequate cooling, yet protect the vehicles from swamping during surf operations. Considerable study has been given to providing a simple system that is commensurate with the very short time it will be used during the life of a vehicle. Such a system using a simple buoyant member to limit entrance of water into the air inlet and outlet has been developed. Essentially, ballistic louvers protect both the inlet and outlet cooling air ducts which terminate in the upper portion of a chamber immediately behind the louvers. The chambers contain floats that are the full width of the chamber and are mounted on long arms. Water entering the chamber through the louvers raises the float to close the duct opening the chamber. The side of the float facing the louver is shaped to form a reaction member so that the velocity of the water striking the louvers is used to accelerate the float towards its closed position. The design is such that water alone can maintain the float in its closed position and at all other times cooling air can be circulated freely. The short periods (5 to 10 seconds) of complete submergence with consequent loss of cooling air will not cause the engine to overheat. Incidental water taken inboard is dumped from the chamber into the engine compartment where it is adequately handled by the bilge pumps. The scheme is simple and has the further advantage that it requires no attention from the crew when going from one media to another.

Accessories

A single accessory drive gear box is mounted on the end of the engine opposite the transmission and provides drives for a bilge pump, a hydraulic pump, and a power take-off for operating any equipment required for the special purpose adaptions.

Bilge pumps will be located in the machinery and crew compartments. Since the machinery compartment is vulnerable to entrance of some water, it contains two pumps - one mechanically driven and the other hydraulically driven. Two hydraulically driven pumps are located in the crew compartment, one in each end, to ensure removal of water regardless of vehicle attitude.

The crew compartment is ventilated by a fan drawing air in through a vent located near the stern over the right sponson. The vent is extended to limit entrance of water during waterborne operations and means are provided to prevent ingress of water when the vehicle is submerged. Heaters are adjacent to the inlet so air can be heated as it is drawn in.

A suitable fixed fire extinguishing system is installed with nozzles in the machinery compartment and CO_2 bottles on the left sponson near the stern. A portable extinguisher is mounted on the machinery compartment panel bulkhead conveniently accessible to crew members.

The radio is enclosed in a waterproof box located on the front plate in the gunner's area.

The wide sponsons provide adequate depth for OVE stowage against the vehicle's sides without interfering with troops or cargo carried.

CAPABILITY AND PERFORMANCE

Land Performance

In this vehicle concept, priority has been given to land mobility although care has been taken to assure satisfactory water performance. This compact vehicle will be more mobile than any other tracked amphibious vehicle that has been used by the Marine Corps. It will meet all operating requirements set forth in the Development Characteristics.

The large wheels inherently reduce rolling resistance to 60-70 lbs. per ton over all speed ranges in contrast with 90-100 lbs. per ton for conventional LVT wheels. The larger wheels also afford easier and faster travel over rough ground and terrain.

The power of this vehicle can be compared to the LVTP5 on the basis of horsepower per ton of vehicle weight. The LVTPX11 has 18.6 HP/ton while the LVTP5 has 18.2. Because of improved efficiency in the LVTPX11 power train, which is obtained by multiple speed ranges and the converter lock-up feature, horsepower delivered to tracks for the LVTPX11 is 14.6 HP/ton, while for the LVTP5 it is 10.8 HP/ton. This additional power combined with the increase potential agility of smaller vehicles and lower rolling resistance will produce a very mobile vehicle.

Performance of the LVTPX11 is depicted in Fig. 2 where tractive effort is plotted against vehicle speed. Superimposed are percent grade lines that include a rolling resistance of 65 lbs. per ton of vehicle weight. On level ground, the vehicle will travel 40 MPH at approximately 70% full power; or, it has the power to negotiate more than a 3% grade without decelerating. A 70% grade can be negotiated at 3 MPH. With four speed ranges and the torque converter, the power train can be matched to all types of terrain conditions to ensure maximum horsepower output and adequate performance.

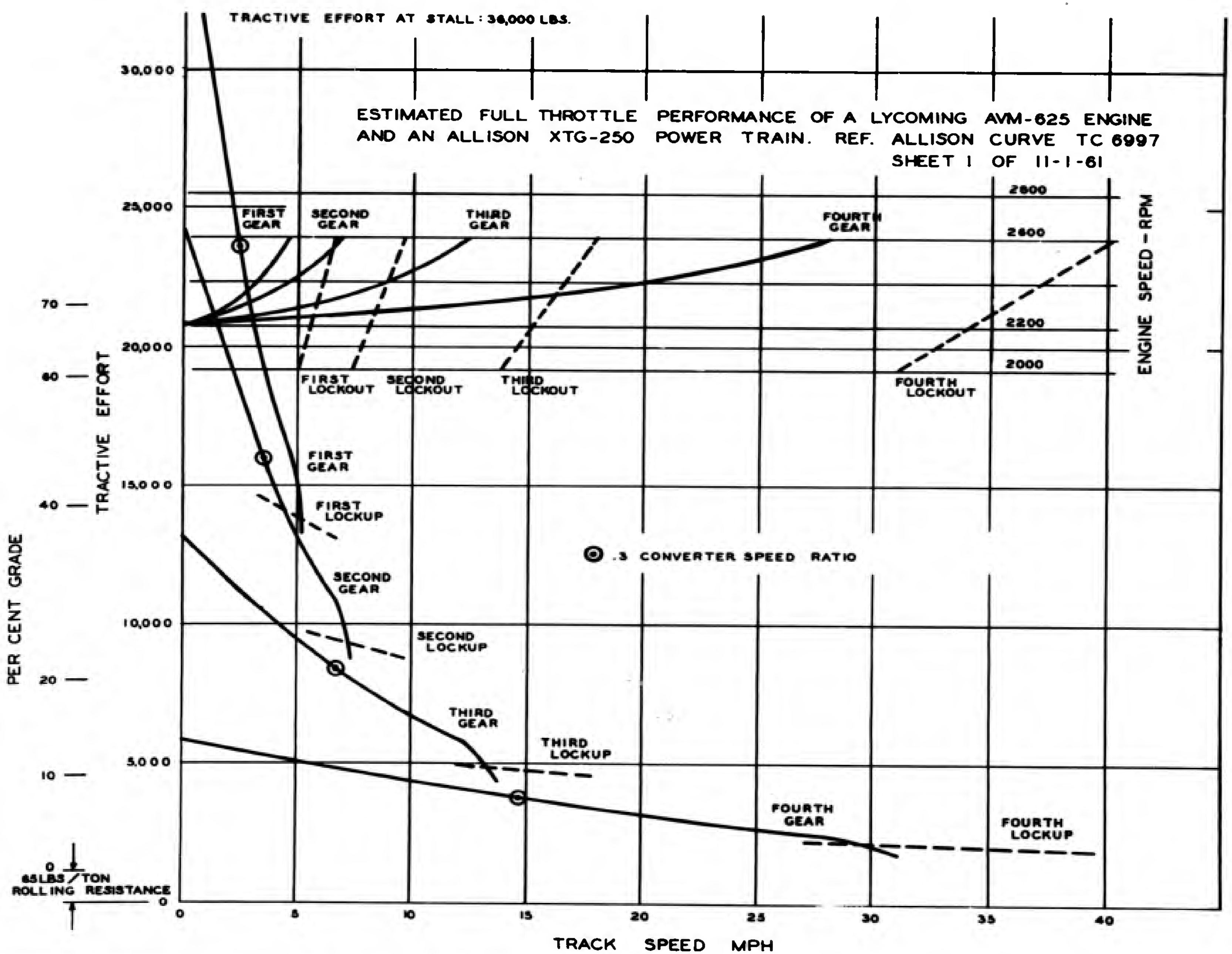

TRACTIVE EFFORT AT STALL: 36,000 LBS.
ESTIMATED FULL THROTTLE PERFORMANCE OF A LYCOMING AVM-625 ENGINE AND AN ALLISON XTG-250 POWER TRAIN. REF. ALLISON CURVE TC 6997 SHEET 1 OF 11-1-61
2800
2600
2200
2000
ENGINE SPEED - RPM
FIRST GEAR
SECOND GEAR
THIRD GEAR
FOURTH GEAR
FIRST LOCKOUT
SECOND LOCKOUT
THIRD LOCKOUT
FOURTH LOCKOUT
FIRST GEAR
FIRST LOCKUP
SECOND GEAR
SECOND LOCKUP
THIRD GEAR
THIRD LOCKUP
FOURTH GEAR
FOURTH LOCKUP
.3 CONVERTER SPEED RATIO
30,000
25,000
20,000
15,000
10,000
5,000
0
TRACTIVE EFFORT
70
60
40
20
10
0
PER CENT GRADE
65 LBS/TON ROLLING RESISTANCE
0
5
10
15
20
25
30
35
40
TRACK SPEED MPH

FIGURE 2

The power train includes provisions for three steering combinations:

1. Geared Steer - The speed of one track is reduced. The 1.475:1 speed ratio difference gives a turning radius of 21 feet.
2. Clutch Brake Steer - The track on the inside of the turn is stopped and the vehicle pivots arpund this track.
3. Pivot Steer - One track operates in reverse and the other in forward.

The power train goes automatically into geared steer in the higher ranges (3rd and 4th). In the lower ranges, clutch brake or pivot steer are selected by the driver. However, since the power train will be 3rd range during most water operation, the power train valve body will be modified to have clutch brake steer in 3rd range to have improved control capability.

Drawing 6496-28 shows the location of the center of gravity of the vehicle in both the light (25,000 lb) and loaded (35,000 lb) conditions. The center of gravity in the loaded condition is slightly ahead of the center road wheel. This favors land mobility. The center of gravity is even more forward in the light condition. Although the vehicle may seem somewhat nose heavy, land mobility should not be impaired. The forward center of gravity location accommodates special adaptions such as the gun turret and the planned arrangement for a recovery vehicle extremely well.

To ensure the vehicle of having the capability of negotiating all types of terrain including swamps, muskeg, and rice paddies, 22 inch wide tracks have been incorporated to provide adequate distribution of vehicle weight. Group pressure with a gross vehicle weight of 35,000 lbs.

is 5.4 PSI. Under difficult muskeg or swamp conditions, the vehicle will undoubtedly sink to the point where it will be supported by the underside of the hull. These circumstances need not, however, immobilize the vehicle providing that the underside of the hull is cleanly designed and without protrusions. Tests of similar vehicles in muskeg at Fort Greely, Alaska, show this to be true even with vehicles having a somewhat higher ground pressure.

Cargo Capability

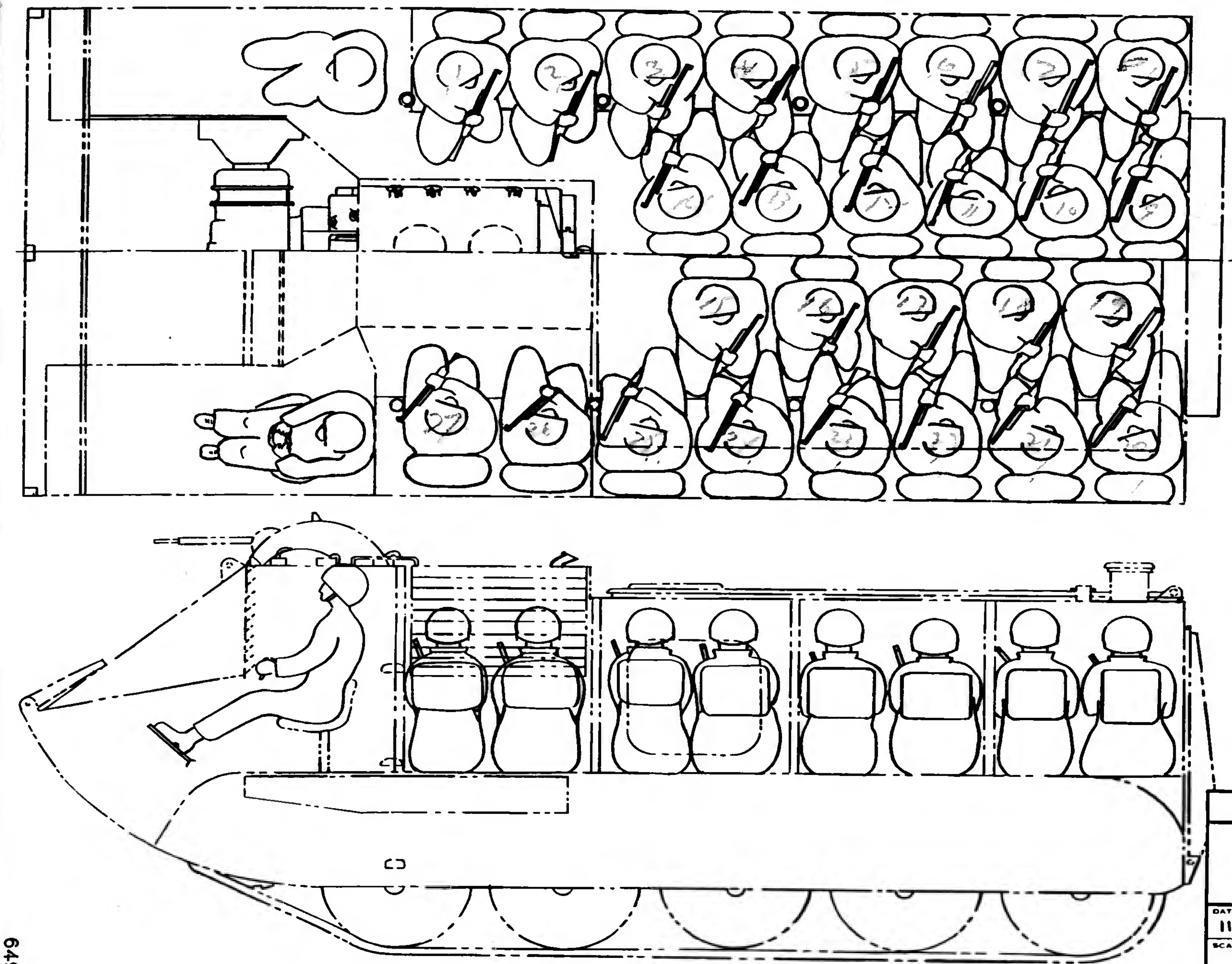
10,000 pounds of cargo can be carried in the cargo compartment which is 10'-6" long, 5'-3" wide between sponsons, 9'-10½" wide over sponsons, and 4'-6" high. Cargo can be loaded and unloaded through either the hatches in the top deck or the rear ramp. Three standard size pallets (40" wide x 48" ling x 48" high) can be stowed.

In addition to the driver and gunner, 27 fully equipped Marines can be carried as shown on drawing 6496-38.

In keeping with the necessity of making the vehicle as compact as practicable, it should be pointed out that, although space is sufficient to accommodate this number of troops, it is a minimum and crowded conditions will exist.

Air Transportability

The proposed vehicle can be transported by the C-133 airplane and the C-141 airplane that is presently being developed. By reducing the over-all width of the top deck 6 inches by sloping the sides, the vehicle would be within the configuration limitations of the C-124 aircraft.



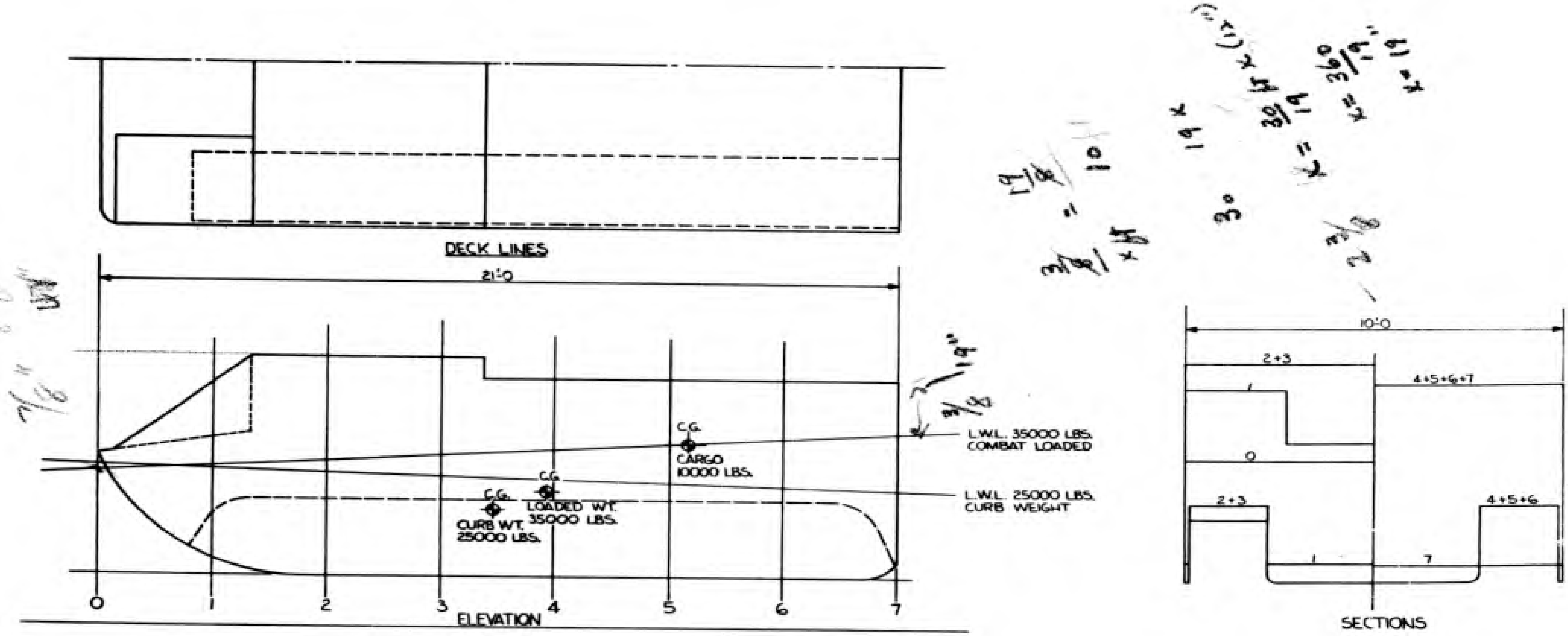
PROPOSAL

LVTPX 11
TROOP CARRYING
ARRANGEMENT

DATE	DRAWN BY	ENGINEER	APPROVED
11-8-61	PEQ	[illegible]	
SCALE	DRAWING NUMBER 6496-38		

PACIFIC CAR AND FOUNDRY COMPANY

6496-3



6496-19
FIGURE 3

PROPOSAL

HULL LINES

LVTPX11

10-25-61

6496-19

PACIFIC CAR AND FOUNDRY COMPANY

RENTON WASHINGTON

Water Performance

Water performance has been carefully studied by the Company and its consultants, Hydronautics, Inc. New developments are incorporated into the design which assure a water speed comparable to the LVTP5, even though the LVTPX11 is considerably smaller. Additional improvement suggested by Hydronautics, as explained in Part III, will be studied.

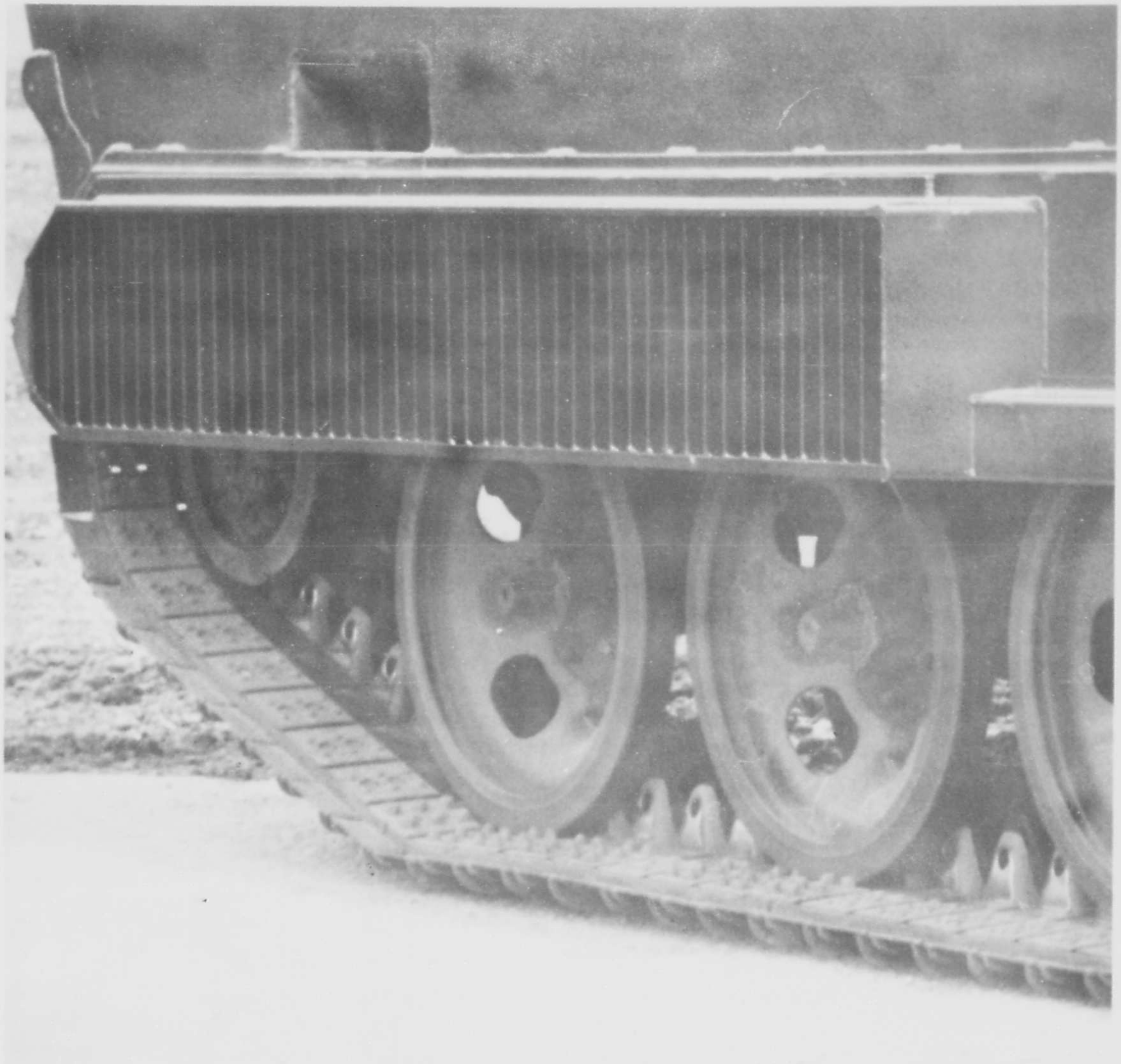
Initial trim and buoyancy studies showed that the vehicle tended to have more reserve buoyancy than was desirable for good surfing characteristics. (The more the reserve buoyancy, the more the vehicle is adversely affected by surf). This dictated a vehicle configuration as compact as possible.

The location of the mechanical components in the bow, for space utilization and land performance considerations, caused difficulties in providing a hull configuration with the center of buoyancy far enough forward to establish acceptable trim. Additional buoyancy was required in the bow, and was attained without increasing the length of the vehicle or shifting the center of gravity forward by extending only the portion of the bow below the normal waterline, putting negative sheer on the fore deck, and using a bow vane to extend the hull to an acceptable shape when waterborne. Fig. 3 shows an outline of the vehicle with waterlines in the loaded and unloaded conditions. Almost level trim has been attained in the unloaded condition with a desirable trim down by the stern in the loaded condition. See Fig. 3 for hull lines, center of gravity, and trim lines.

Several water propulsion methods were considered. Although the propulsion efficiency of propellers or water jets can be made greater than track propulsion, propulsion by tracks is considered most appropriate for this application for the following reasons:

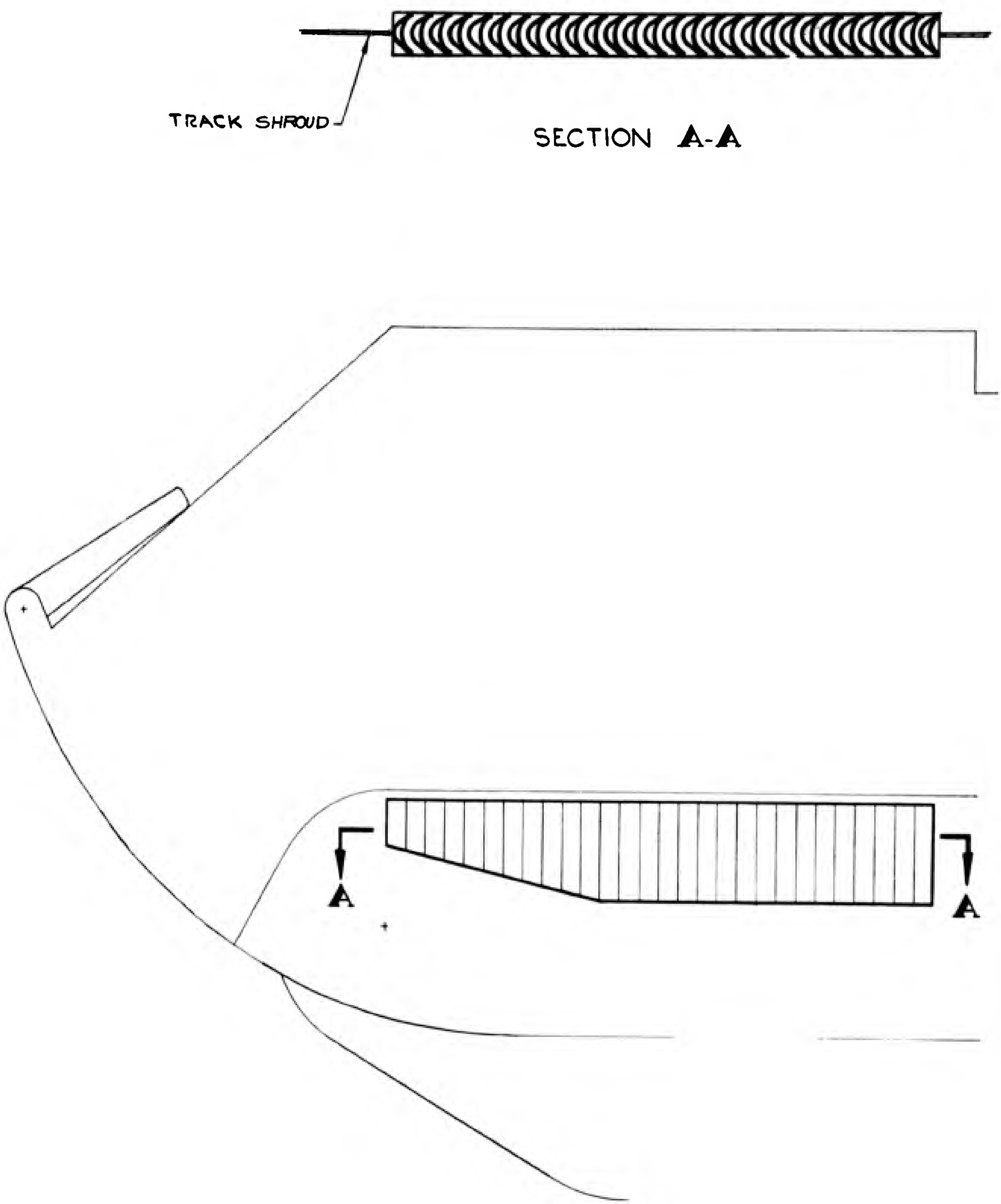
1. Due to the hull displacement and restricted shape, vehicle water speed is limited.
2. No additional equipment is required when propelling with tracks.
3. The major portion of vehicle operation is on land.
4. Improvements have been made to make track propulsion more effective.

The primary advancement that the Company has made to improve track propulsion is the development of "Hydrovanes". This development resulted from a study to improve the propulsion effect of a fully submerged track. Although the tracks form an effective propulsion means, mainly because they act on a great amount of water, they have one disadvantage; the top of the track pushes water in the wrong direction, cancelling out some of the thrust produced by the bottom of the track. Track shrouds are sometimes employed to minimize the backward pumping tendency but are not effective because clearances around the track must be large for practical reasons. To utilize the water entrained around the top of the track, a series of vanes are mounted vertically in the forward end of the track shrouds as shown in Fig. 4 . Some of the water being pushed forward by the top of the track is reversed in direction by the vanes and exhausted backwards. The reaction to this change of direction produces a forward thrust, augmenting the thrust of the bottom of the track. In tests conducted on a T116 Amphibious Tracked Cargo Carrier, water speed was increased from 3.93 to 4.46 MPH. This represents an increase in thrust of about 40%. Because the water is exhausted at an angle to the side of the vehicle, steering and control were improved and the turning radius was decreased 10-15%. Similar improvements in performance are expected on the LVTPX11.

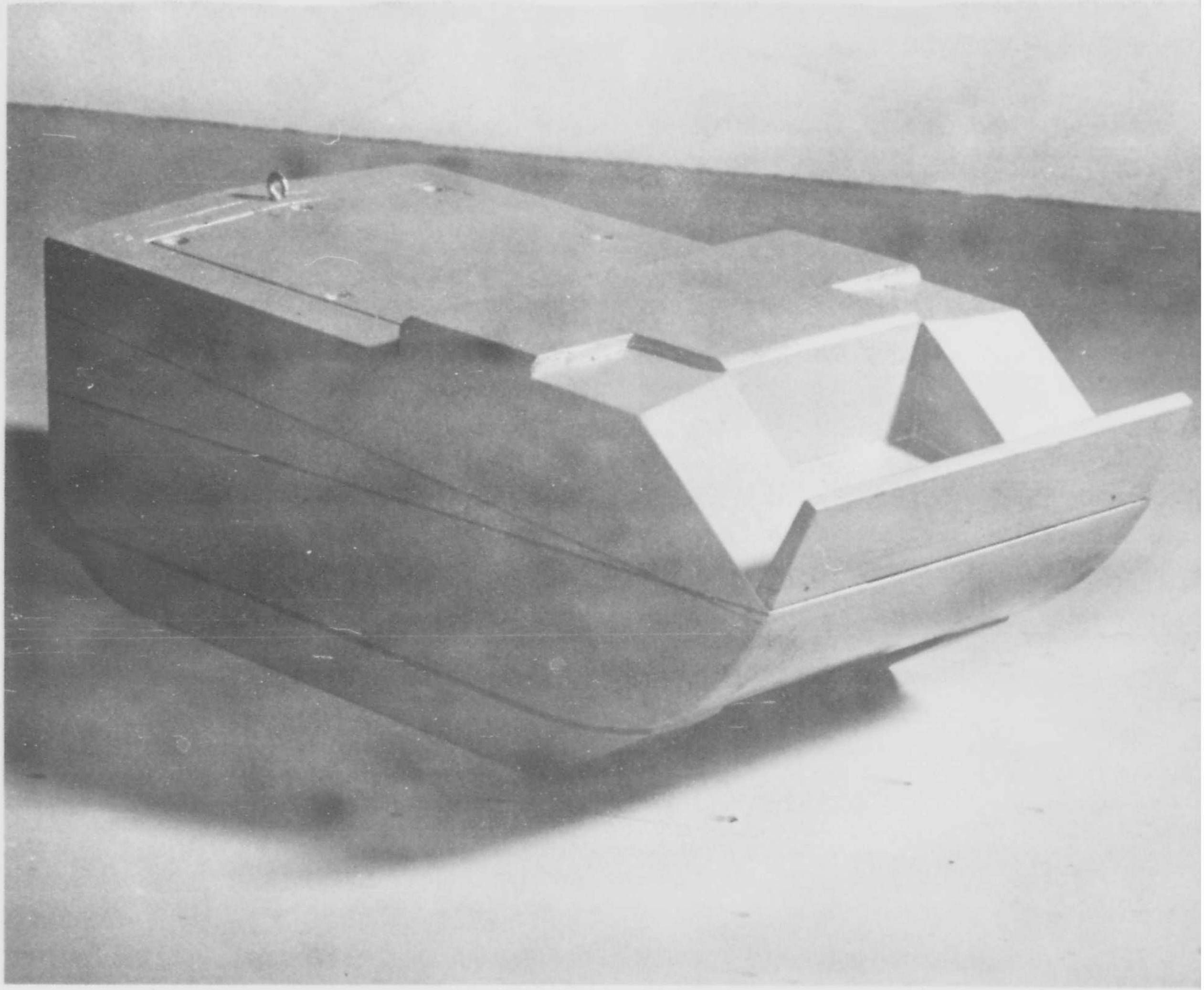


HYDROVANE INSTALLATION M 116 VEHICLE

FIGURE 4

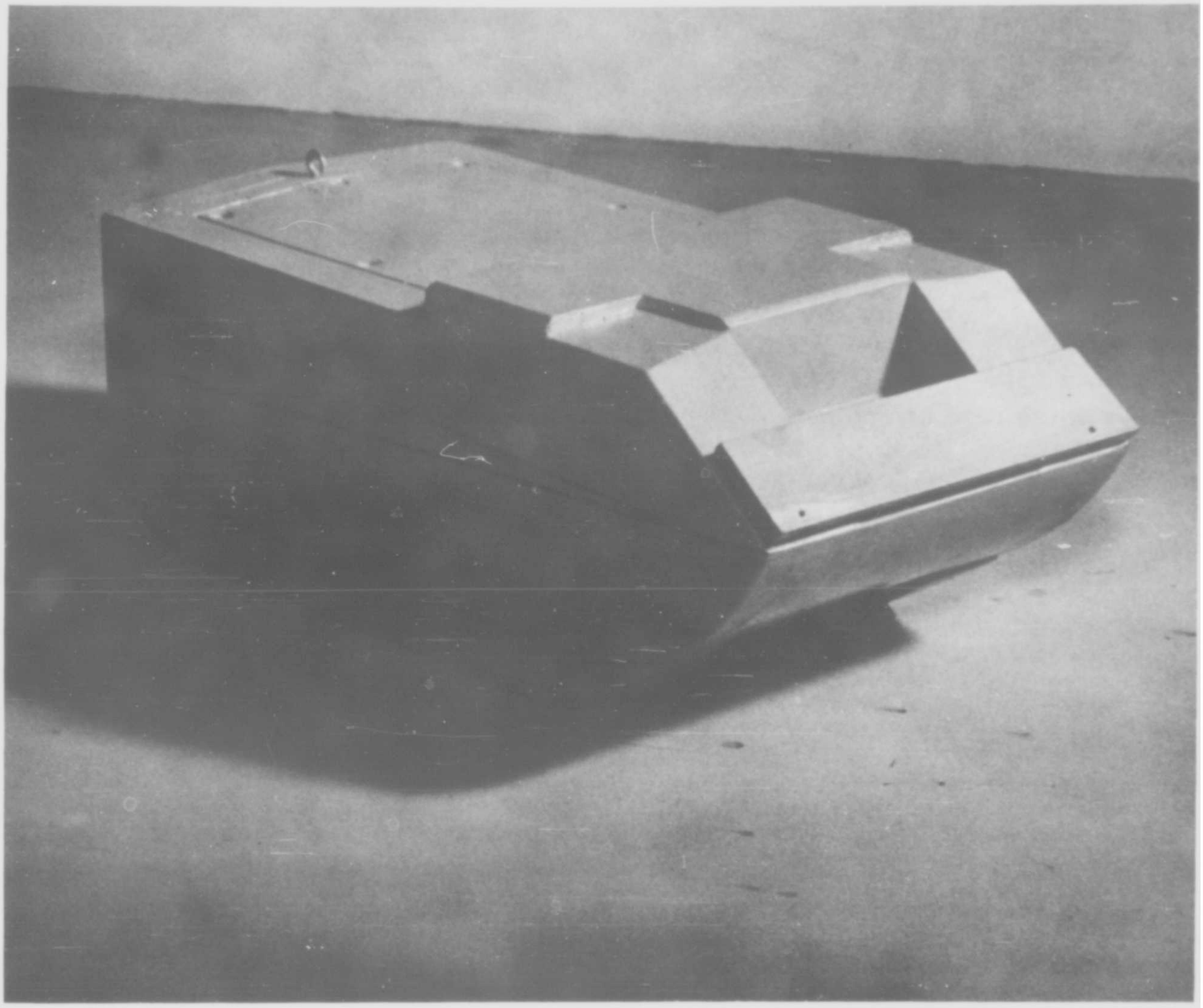


PROPOSAL			
HYDRO VANES LVT PX 11			
DATE 11-7-61	DRAWN BY LeN	ENGINEER	APPROVED
SCALE 1/8	DRAWING NUMBER 6496-29		
PACIFIC CAR AND FOUNDRY COMPANY RENTON WASHINGTON			



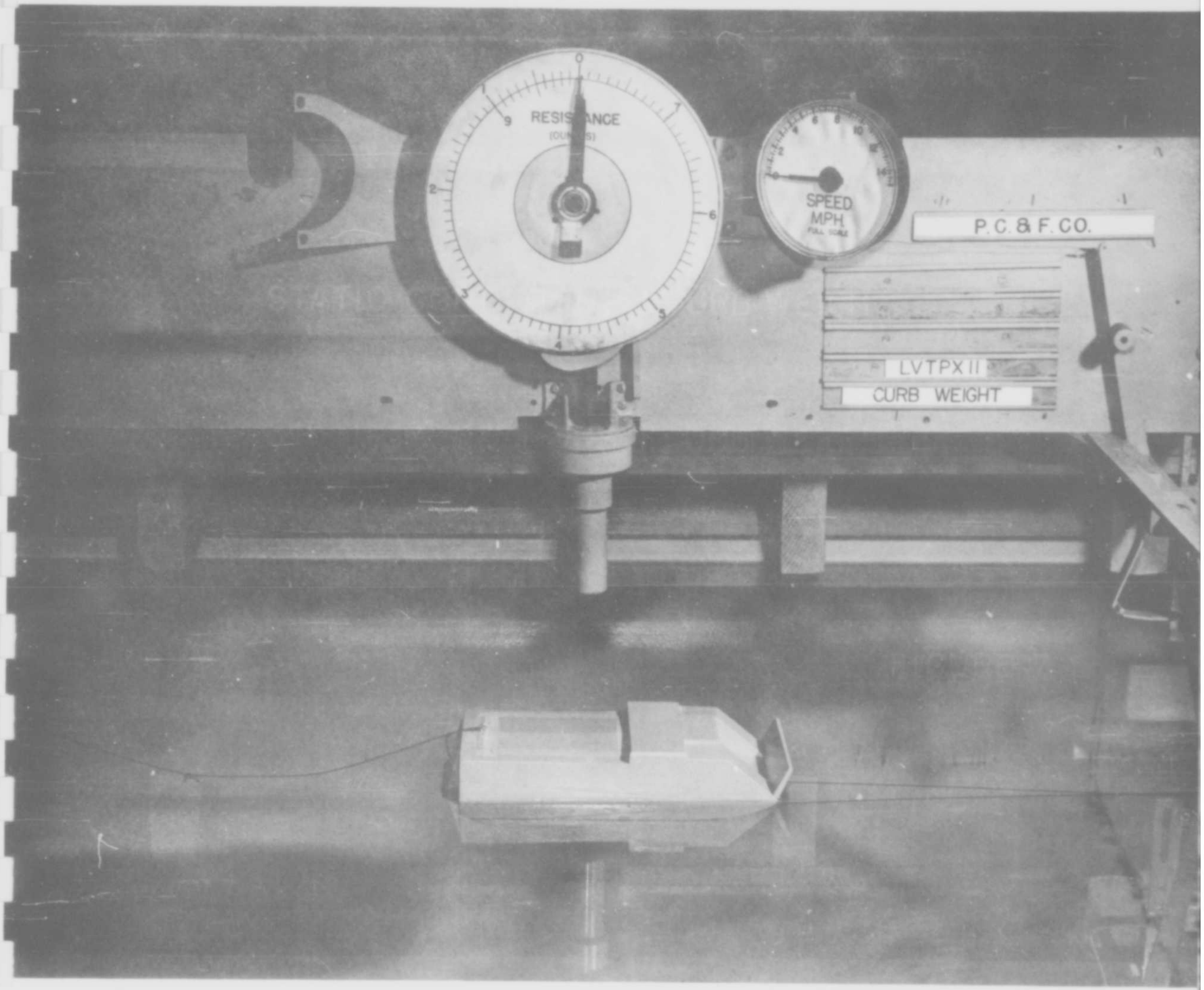
LVTP XII MODEL, WATER TRAVEL

FIGURE 5



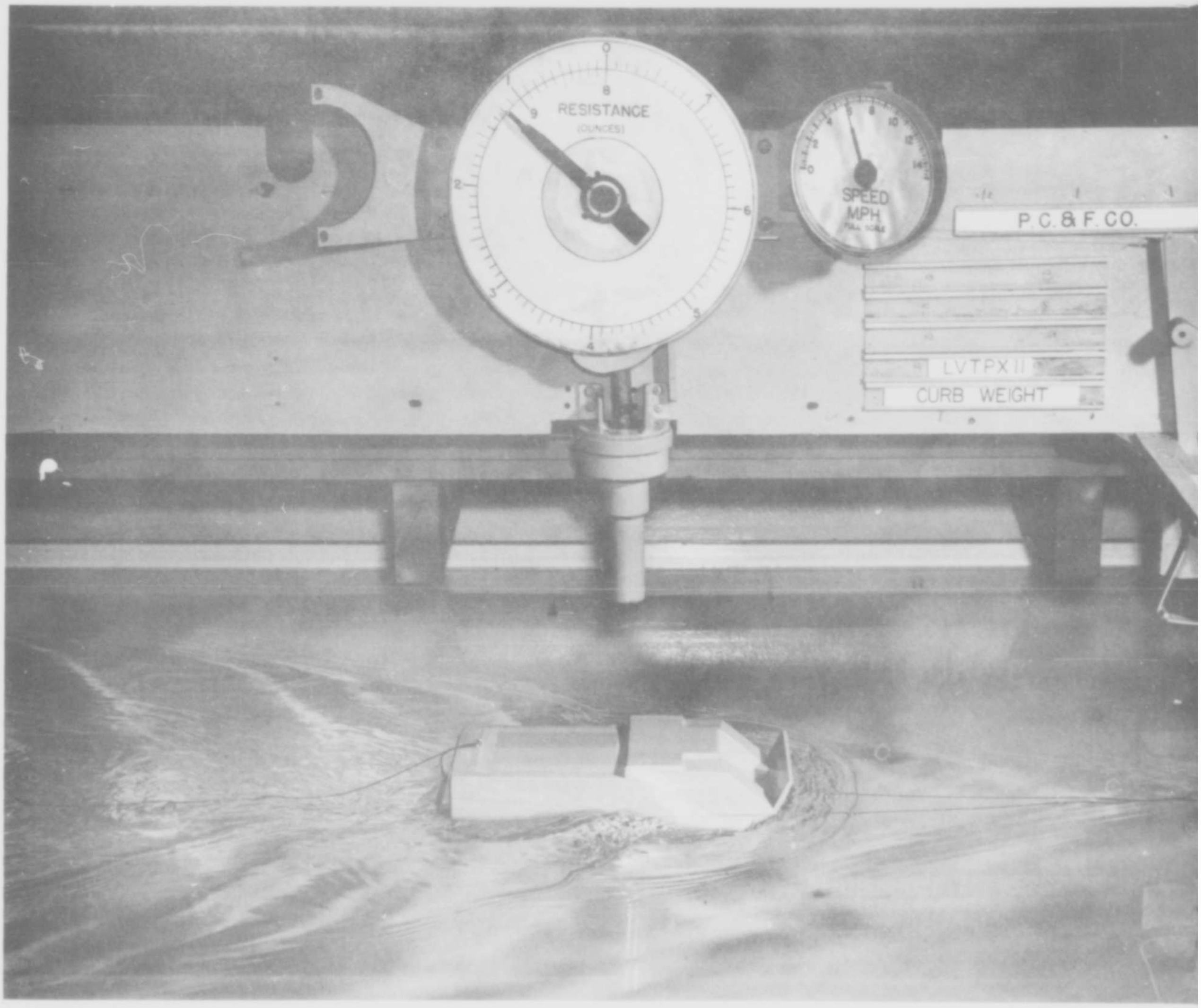
LVTP XII MODEL, LAND TRAVEL

FIGURE 6



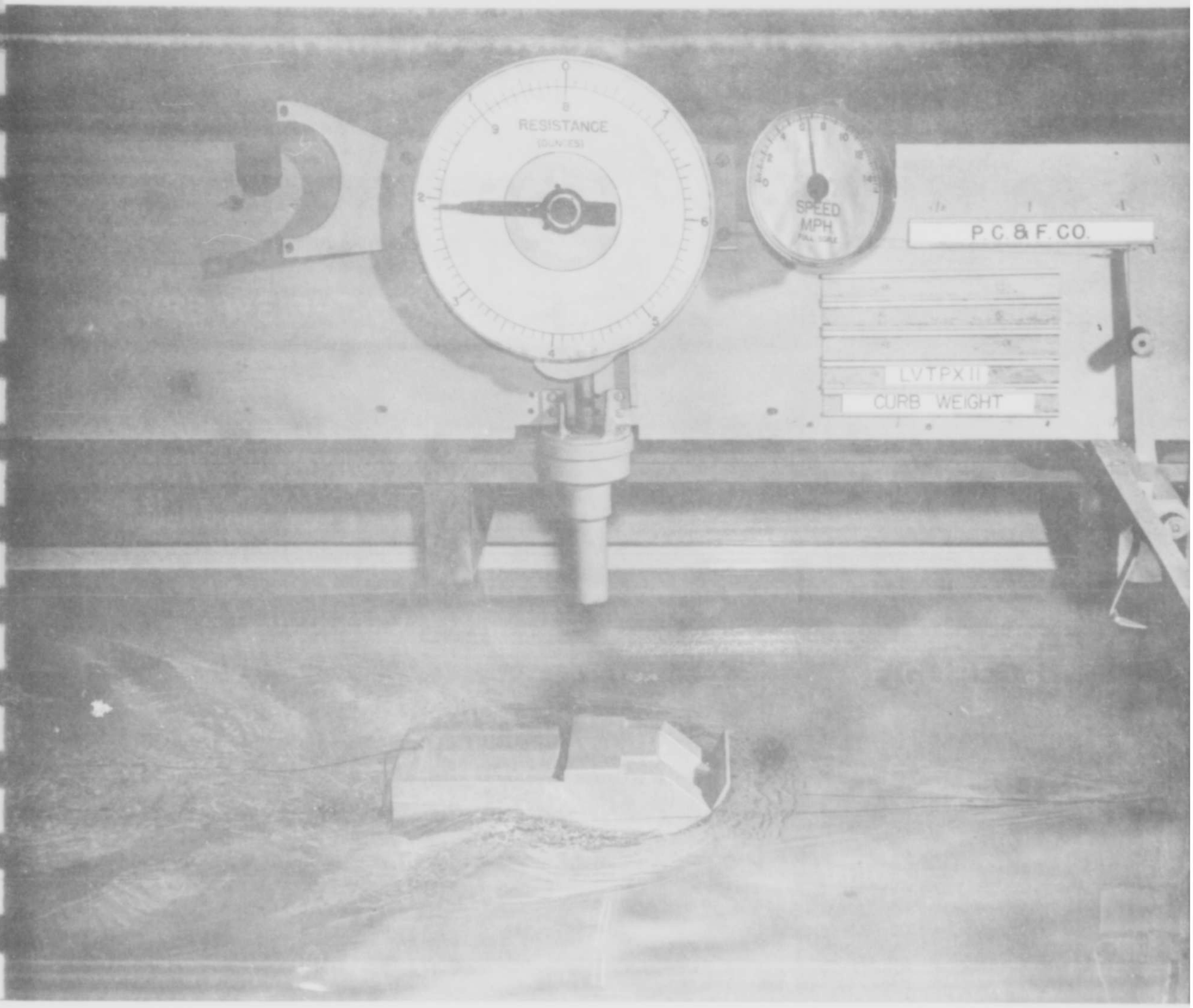
STATIC CONDITION AT CURB WEIGHT

FIGURE 7



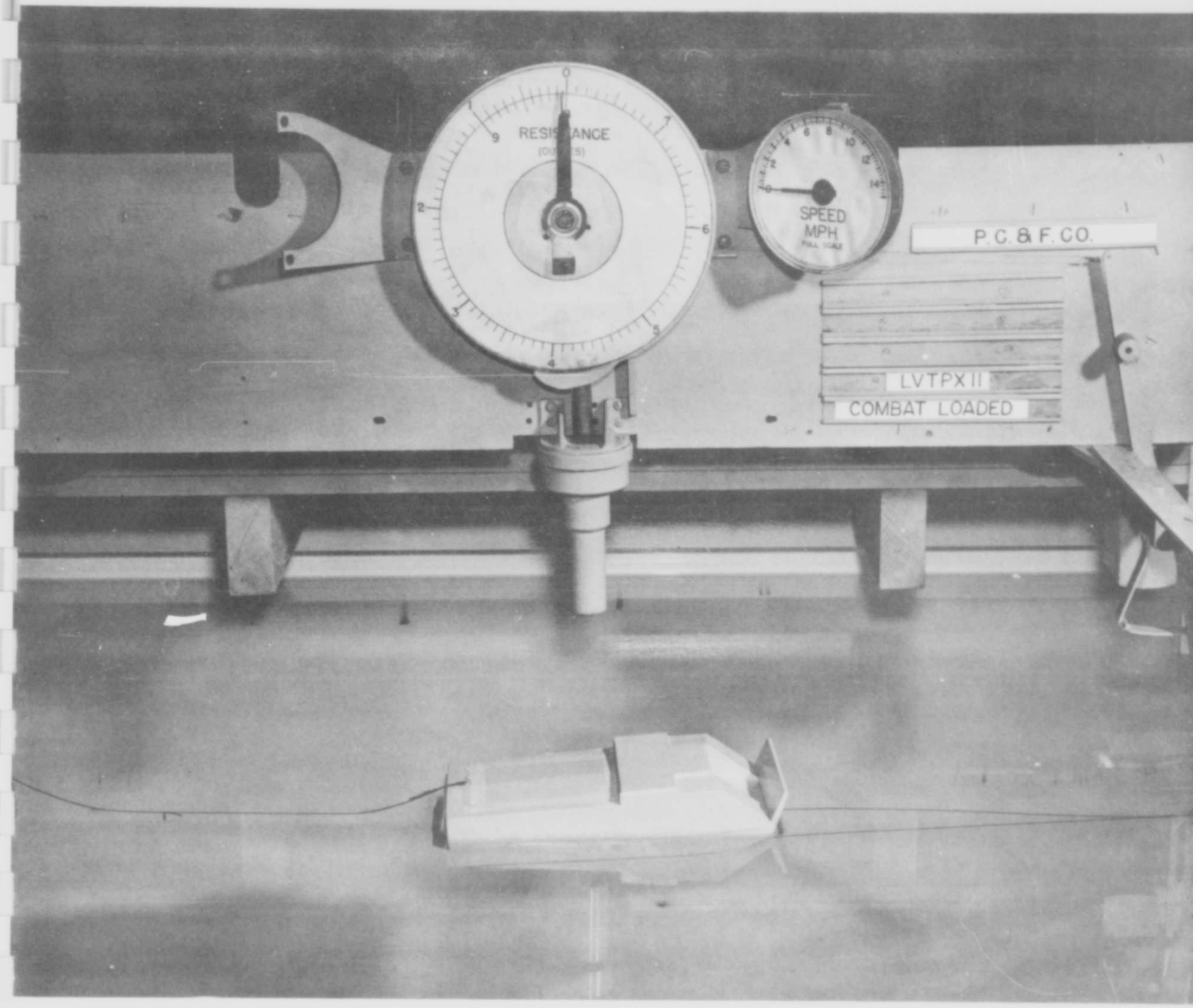
CURB WEIGHT CONDITION SIMULATED SPEED 6 M.P.H.

FIGURE 8



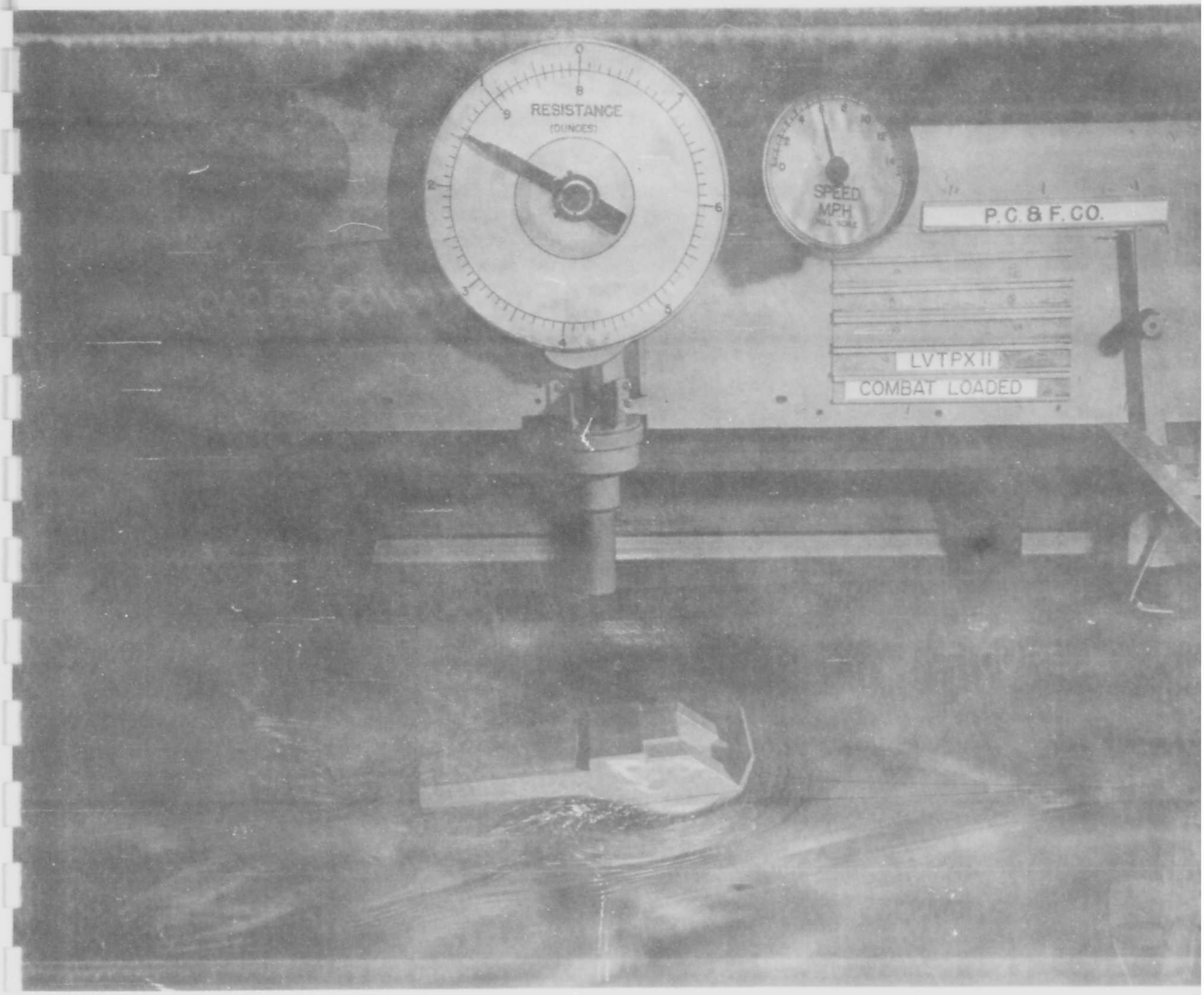
CURB WEIGHT CONDITION SIMULATED SPEED 7 M.P.H.

FIGURE 9



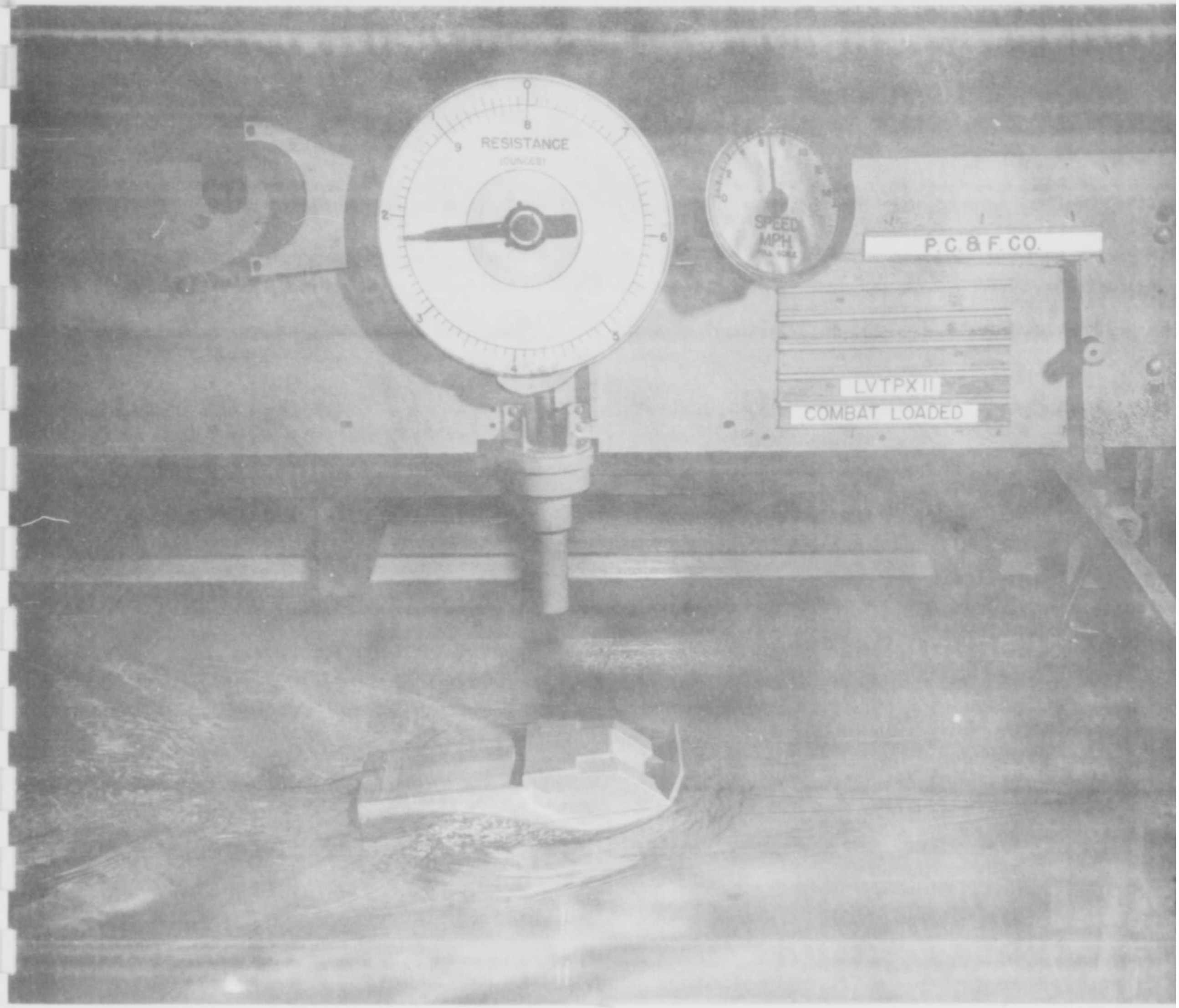
STATIC CONDITION WHEN LOADED

FIGURE 10



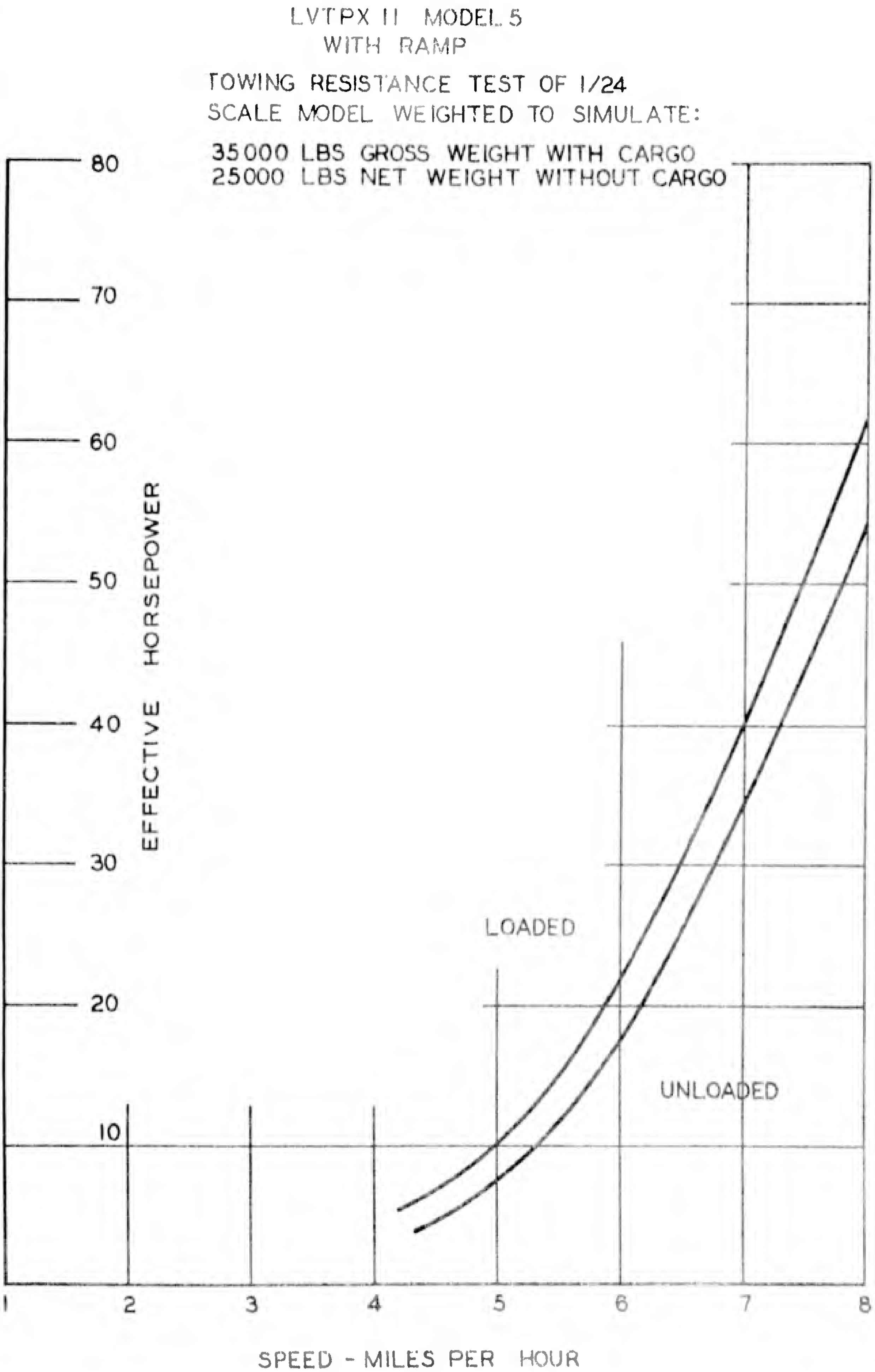
LOADED CONDITION SIMULATED SPEED 6 M.P.H.

FIGURE 11



LOADED CONDITION SIMULATED SPEED 7 M.P.H.

FIGURE 12



LVTPX II MODEL 5
WITH RAMP
TOWING RESISTANCE TEST OF 1/24
SCALE MODEL WEIGHTED TO SIMULATE:
35000 LBS GROSS WEIGHT WITH CARGO
25000 LBS NET WEIGHT WITHOUT CARGO
EFFECTIVE HORSEPOWER
80
70
60
50
40
30
20
10
LOADED
UNLOADED
1
2
3
4
5
6
7
8
SPEED - MILES PER HOUR

FIGURE 13

Models to 1/24 scale were built of several hull configurations to test water performance. These were tested in the Company's model tow tank. Figures 5 through 12 show the model with the final hull configuration being towed at various speeds. The model tests indicate that good performance may be anticipated. The graph of test results is plotted showing effective horsepower at various speeds (Fig. 13). It is estimated that track speed in water will be 12 to 14 MPH with a propulsion efficiency of approximately 12%. The effective horsepower transmitted to the water will be 27 giving a water speed of 6.3 MPH in full loaded condition. However, with the benefit of the hydrovanes, water speed is estimated to be 6.5 - 7.0 MPH.

Effective reverse operation should result since the high ratio of the two reverse speeds incorporated in the XTG-250 Power Train makes possible a track speed of 18 MPH.

3. Performance

Gross HP to Weight Ratio	18.6 HP/ton
Net HP to Weight Ratio	14.6 HP/ton
Cruising Range, Land	250 miles
Max. Land Speed, Forward	40 MPH
Max. Land Speed, Reverse	18 MPH
Max. Water Speed, Forward	6.5-7 MPH
Max. Grade	70%
Max. Side Slope	60%
Max. Trench	8 feet
Max. Wall	3 feet
Min. Turning Radius, Geared Steer	21 feet
Min. Turning Radius, Clutch Brake Steer	About Inside Track
Min. Turning Radius, Pivot Steer	About Vehicle Center

4. Engine

Number	1
Make	Lycoming
Model	AVM-625
Number of Cylinders	8
Bore	4.25 inches
Stroke	5.5 inches
Type	2 Cycle, Vee, Air-Cooled
Displacement	625 cu.in.
Fuel	Diesel, Compression Ignition, JP4 and JP5, Combat Gasoline
Fuel Capacity	250 gallons

4. Engine (continued)

Gross HP	325 @ 2600 RPM
Net HP	250 @ 2600 RPM
Max. Net Torque	580 ft.lbs. @ 1900 RPM
Cooling System	Air
Weight, Dry	1000 pounds
Wet Weight/Net BHP	4 lbs/BHP

5. Power Train

Transmission	Allison XTG-250
Differential	In Transmission
Torque Converter	In Transmission
Other Units	Accessory Drive
Over-all Ratio, Engine to Sprocket	105:1
Steering Control	In Transmission
Brakes	Mechanical, in Transmission
Oil Cooling System	Radiator in Air Stream

6. Final Reduction — In Transmission

7. Running Gear

Type of Suspension	Steel Torsion Bars
Number of Suspension Units	5 per side
Road Wheel Size	32 inches diameter
Track Width	22 inches

8. Fire Extinguishers

Fixed	2 on rear, port sponson
Capacity	10 pounds each
Portable	1 in crew compartment, on engine compartment bulkhead - 5 pounds.

9. Electrical

Nominal Voltage	24 volts
Alternator	1
Amperage	100 amperes
Batteries	2
Starting Motor	1

10. Communications

Radio Set	Collins 618T or AN/PRC 47
Location	Bow, in Gunner's Area
Intercom	In Radio
Number of Outlets	3

11. Armament

Primary Caliber	.30
Mount	Turreted
Traverse Right and Left	360°
Elevation	60°
Depression	15°

12. Armor

Permanent Hull, Aluminum	Front - 1-1/8 Sides - 3/4 Bottom - 3/4 Stern - 3/4 Top - 1/2
Top Hatches, Aluminum	1/2
Ramp, Aluminum	3/4

13. Optical Devices

Driver's Vision	Periscopes, M17
Gunner's Vision	Vision Blocks

14. Cargo Compartment

Length	130 inches
Width, Between Sponsons	63 inches
Width, Over-all	118½ inches
Ramp Location	Stern
Cargo Hatch Opening	6 feet x 10 feet
Personnel	27 Combat Equipped

Adaptions to Special Purpose Vehicles

The concept of the LVTPX11 as a basic vehicle adaptable to special configurations for logistic, engineering, and weapon support of assault troops is attained by the concentration of all major components and crew in the front portion of the vehicle. The cargo carrying portion is free for modification to suit the required adaption.

To provide additional versatility, the deck and transom can be made removable in the area of the cargo compartment. The LVTPX11 hatch frame, combing, and stanchions all can bolt to the deck margin and sponsons. The ramp frame and associated ramp acutators would secure to the transom frame sides and bottom and to the hatch frame at the top. The heavier deck beams and stanchions supporting a 105 MM howitzer turret bearing can be substituted readily for the hatch deck for an amphibious field artillery option shown in drawing 6496-34 and recovery vehicle shown in drawing 6496-33.

To enclose the Mauler Air-Defense System and provide the added buoyancy necessitated by its overweight, the ramp is replaced by an extended transom. The latter increases the hull length approximately 15 inches. The Mauler package mounts on three ball supports at cargo deck level and is sealed into the hull by rubber moldings secured to a bolt-on frame. Elastic seating at deck level is desirable to mitigate straining the Mauler housing by supports redundant to the aforementioned ball sockets. Specifically, the hull's width between sponsons, and its depth are dimensioned to accommodate the Mauler system.

The concept of the basic cargo compartment as a deep-walled pick-up truck bed is as fundamental to its adaptability as is the concept of the packaged weapon or utility systems.

The figure depicting an amphibious howitzer motor carriage indicates that little of the nature of a space problem exists to complicate this adaption. Preliminary stability figures concur that a 3/4 inch average thickness, aluminum turret enclosure of a 105 MM howitzer of the M2A2 type will be feasible from a weight distribution standpoint. Further investigation of this installation should be predicated on a new 105 MM howitzer being developed for the XM104 light weight motor carriage, as well as a careful evaluation of the field of use of the weapon. The sophistication of the gun-laying system can vary from the simple manual control and sighting equipment of the field mount to systems capable of accurate fire while underway on land or sea. A system embodying high reliability with rapid and accurate gun positioning is the hydraulic-manual arrangement of the M110 Howitzer motor carriage This is a manual system with auxiliary hydraulic powering means which is rate controllable in a slow-speed range for tracking moving targets or accuracy in positioning plus a high-speed slewing ability.

It is considered unwarranted to place the fire control capability of this field artillery piece in the class of a tank by use of a stabilization system.

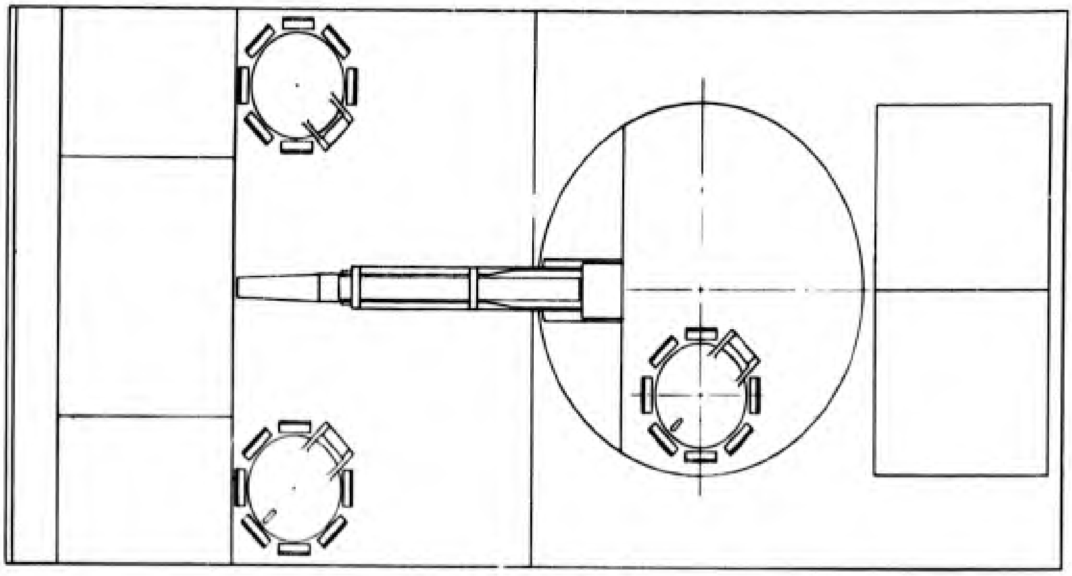
The loading and recoil space in the turret is preserved by trunnioning the tube near its breech, necessitating, in turn, a vertical equilibrator. Such forward placement of the weapon inevitably leads to a horizontal unbalance or slewing moment on side slopes. This can be partially compensated by rearward placement of ready ammunition and ballast at the loader's platform level.

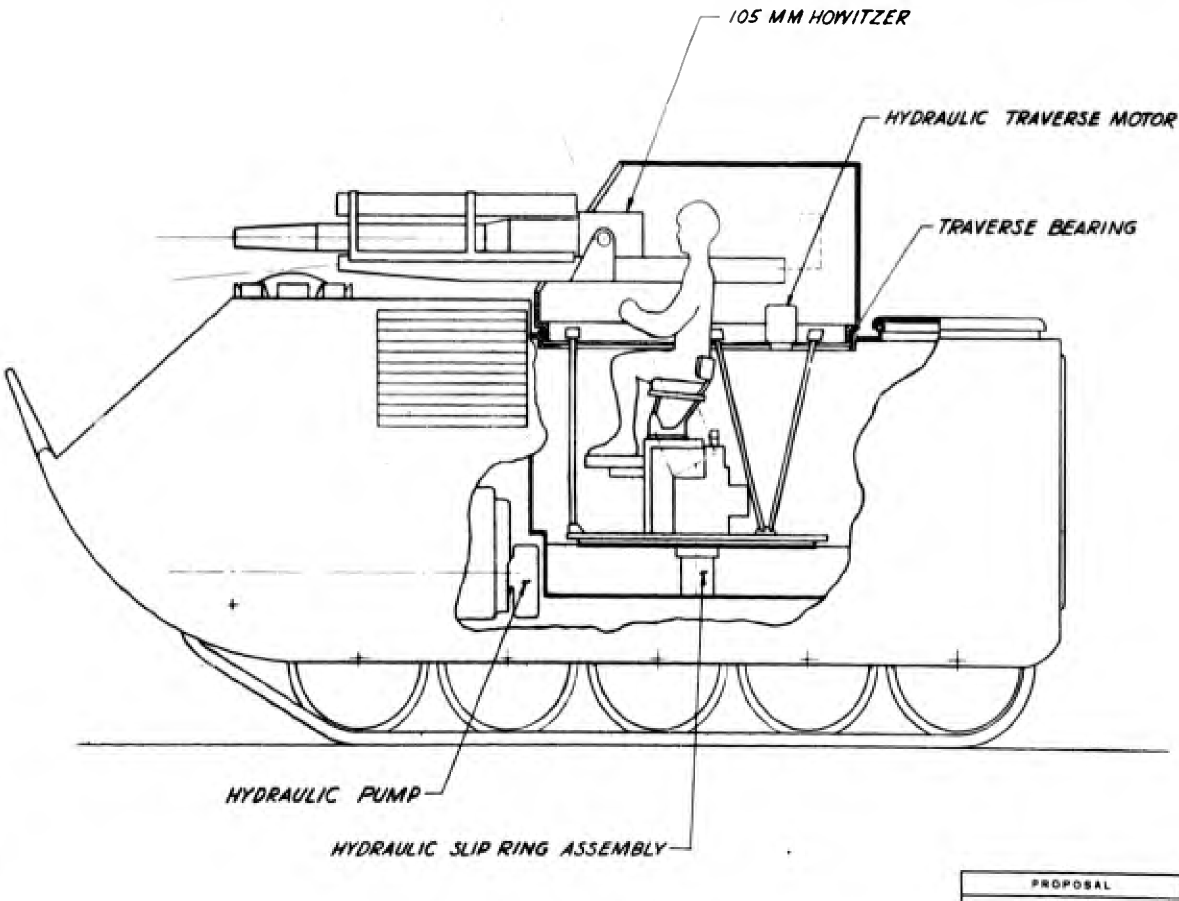
A rigid firing plane is established by the lockout suspension elements of the front and rear road wheels, to increase accuracy of gun laying and reduce jump under recoil forces.

As a recovery vehicle, the howitzer turret is replaced by a self-contained crane cab as shown in drawing 6496-33 . Hydraulically powered, it retains the traversing motor common to the turret, trunnions a crane boom with luffing cylinders and a boom-mounted 5 ton hoisting winch, and supports a 15 ton towing winch. The towing winch fairlead sheave mounted under the boom traverses to guide the towline and elevates to clear hull obstructions or to apply the lifting component desirable in breaking out mired, disabled equipment. The lockout suspension increases the capabilities in this respect. Winches incorporated are the P-10 (5 ton) and P-30 (15 ton) planetary types in current production at Pacific Car and Foundry Company. Their small space requirement is the result of enclosing all gearing and brake elements within the hub of the cable drum. Control and power are applied hydraulically. The entire wrecker design is a reduced scale version of the Ordnance T120 Vehicle now in successful prototype operation and nearing type-classification as a standard military wrecker. Lifting capacity at 90 inch maximum boom reach over the vehicle side is 17,000 pounds with the two-part hoist as limited by vehicle stability.

The towing capability from the boom-mounted fairlead is 24,000 pounds with suspension locked-out and based on complete unloading of the front road wheels. A towline paid through a fairlead on the opened ramp door can develop the 30,000 pound winch pull, 40 inches above ground, on surfaces capable of .85 tractive friction.

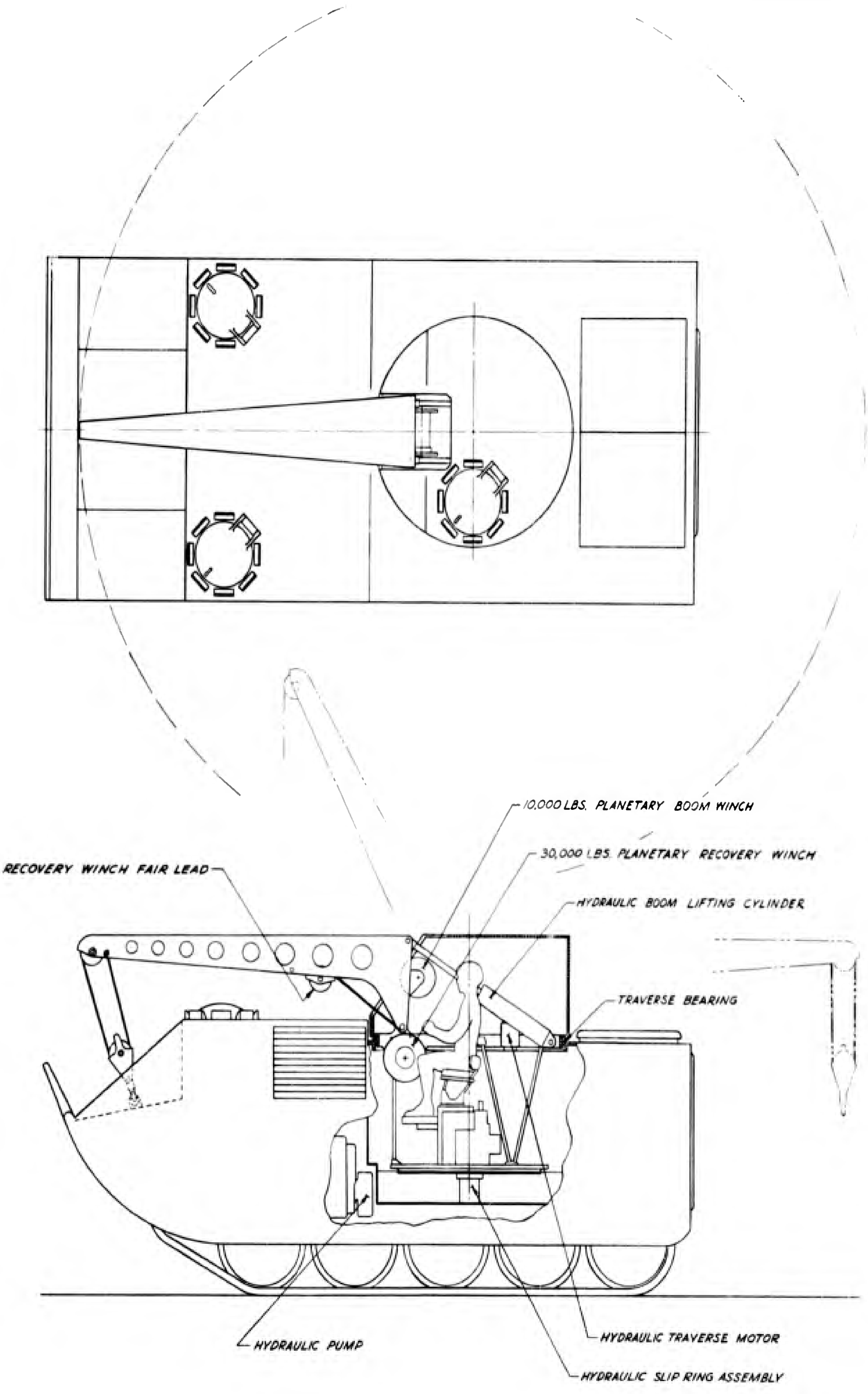
For an engineer mine clearance vehicle, an additional 6 inch width allowance is available within the 10'6" specification limit for mounting arms and trunnions. Other than the above, the characteristics of the proposed layout are antagonistic to its use as a mine clearance vehicle. This drawback stems from a forward location of the center of gravity of the empty vehicle. Correction of trim by stern ballast, coupled with a light and buoyant clearing rake would produce a single purpose vehicle without load capacity for any but the lowest density cargo. It would also be difficult to lift the clearing blade when buried in heavy soils, so the above "jury rig" provisions could only be justified by the expendable nature of the service.





6496 - 34

PROPOSAL
LVT PXII AMPHIBIAN FIELD ARTILLERY WEAPON VEHICLE
6496-34
PACIFIC CAR AND FOUNDRY COMPANY



6496-33

PROPOSAL

LVTPX11
RECOVERY VEHICLE

DATE	DRAWN BY	ENGINEER	APPROVED
11-7-61	LIJ	[illegible]	

SCALE	DRAWING NUMBER
3/8" = 1 FT	6496-33

PACIFIC CAR AND FOUNDRY COMPANY
RENTON WASHINGTON

PART 2

HULL STRUCTURE

In the selection of the hull material, consideration was given steel armor, aluminum armor, and fiberglass laminate. The selection in favor of aluminum armor of 5083 alloy was based on its generally superior ballistic properties, the ease of fabrication by recent welding developments, and the simplicity of design that it engenders. An evaluation of the degree of ballistic protection provided by the proposed hull is appended as classified material to this report.

Pacific Car and Foundry Company has had sufficient experience in the fabrication of the M116 Amphibious Cargo Carrier aluminum hull as to regard such weldments as conventional.

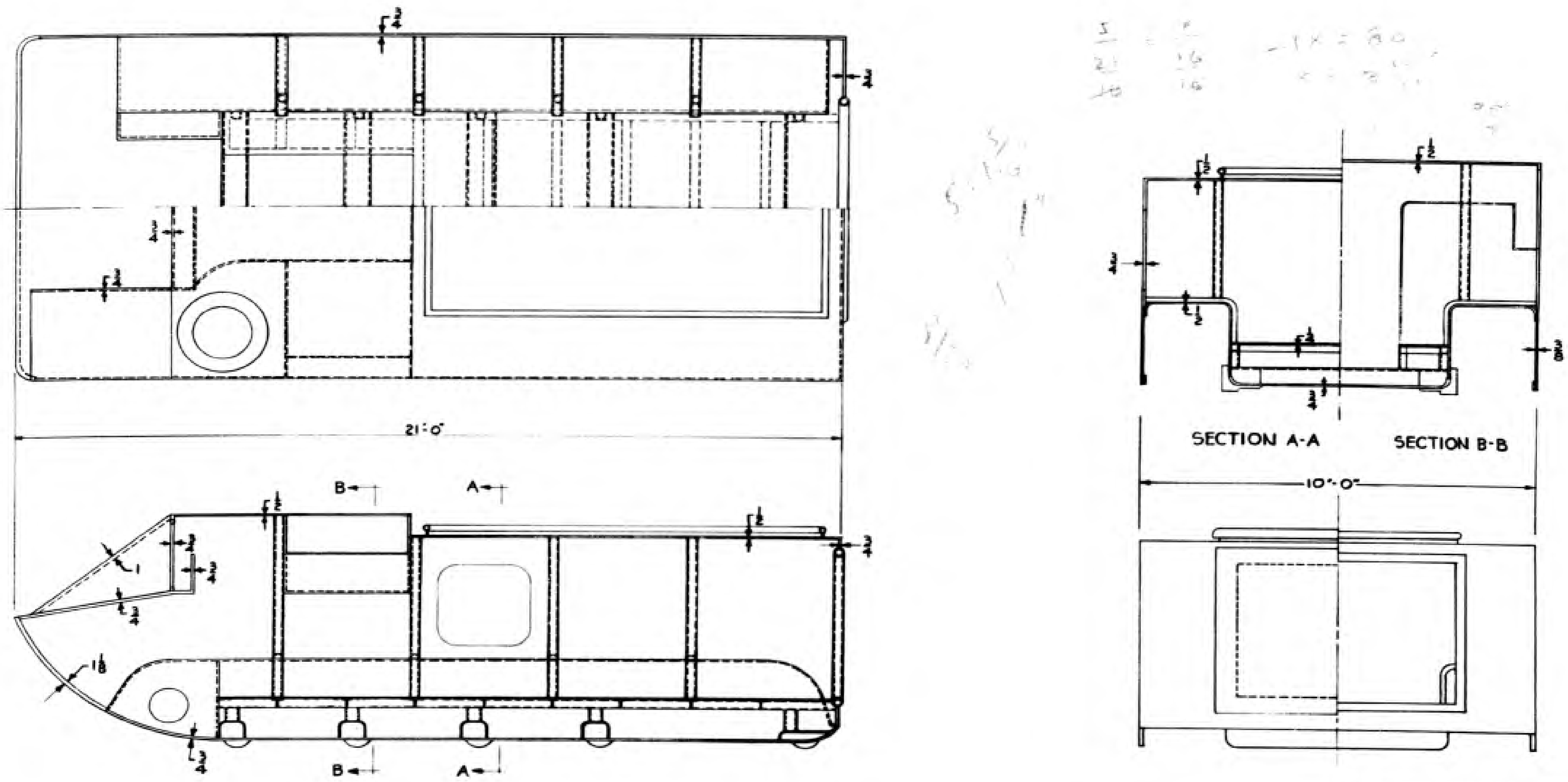
The advantage of aluminum over steel construction derives from its seven-fold increase in bending stiffness in the same weight of skin plate. This increase makes possible a monocoque design free from the stiffening members required of thinner stressed-skin steel structures at points of concentrated loading by the suspension elements. The requirement corollary to ballistic protection is uniformity in skin thickness which prohibits lightweight space frame construction.

Plastic laminate armor is attractive ballistically, expecially in light sections. To achieve these properties demands high pressure curing, economical only in simple forms. This weighs against its selection for structural parts of any complexity. Its heterogeneous strength properties also impose severe design and production limitations.

The following lists define the constructional thicknesses of the LVTPX11 hull as shown in drawing 6496-18.

Bow Plate	1-1/8 inch
Front Deck	1 inch
Air Intake Structure	3/4 inch
Hull Sides	3/4 inch
Deck Plates	1/2 inch
Hull Bottom	3/4 inch
Sponsons	1/2 inch
Stern Plates	3/4 inch
Fenders	3/8 inch
Sprocket Supports	3/4 inch
Bottom Framing over Torsion Bars	3/4 inch
Floor Framing	1/4 inch
Floor Beams	3 inches x 1/4 Tee
Floor	1/4 inch
Stanchions	3 inches x 1/4 Tubing
Hatch Framing	2 inches x 1/4 Tubing
Ramp Framing	3 inches x 1/4 Tubine
Hatch Skin	1/4 inch, top and bottom
Ramp Skin	3/8 inch, inside and outside

The hull is 21 feet long and 10 feet wide. Because of the relative large plating thickness, little framing is required. Framing in the bottom and on the sponsons is designed to transfer floor and sponson loads directly to the suspension. The stanchions support the deck and any deck loads that might be carried. Total estimated bare weight of the hull, including hatches and ramp, is 9500 pounds. The center of



PROPOSAL

HULL ASSEMBLY

LVT P X 11

6496 - 18

PACIFIC CAR AND FOUNDRY COMPANY

gravity of the bare hull is located 128.9 inches from the bow and 36.2 inches above the bottom.

Ramp

The ramp configuration, reference drawing 6496-22, was determined by hydrodynamic requirements. To reduce the pressure drag and the effective transom submergence, the lower portion was extended to continue the lift in the bottom lines at the rear of the vehicle.

The ramp is fabricated of aluminum with ballistic resistance properties equivalent to the 3/4 inch thickness of the surrounding hull rear plate.

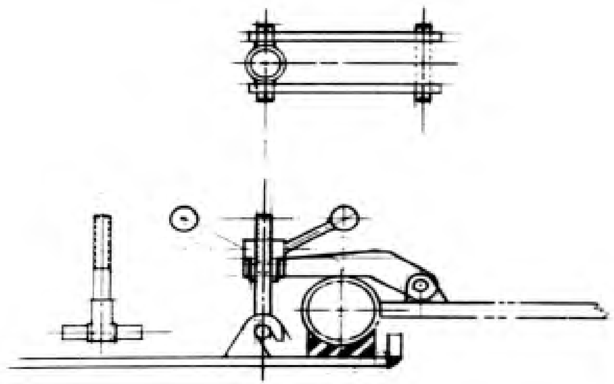
Raising and lowering is accomplished with single acting hydraulic cylinders connected into the hydraulic system, drawing 6496-40. Actuation of the ramp is controlled by the driver. However, a mechanical lock which must be manually actuated at the rear of the vehicle secures the ramp. It would be possible to include a linkage operable by the driver to trip the lock or provide a hydraulically operated lock which would be controlled in the same manner as the hydraulic lifting cylinders.

The ramp is hinged on rubber torsion springs which are torsionally neutral when the ramp is partially open. When it is closed, the springs assure the ramp will travel over center and open. They also assist in raising the ramp from its fully lowered position.

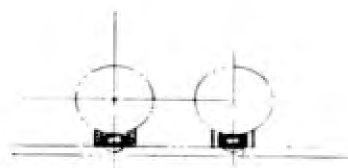
Covers and Panels

Aluminum is used to frame the machinery compartment. Panels are removable around the engine compartment to facilitate servicing and maintenance. The material planned for the panels is a vinyl plastic comprising two 1/16 inch thicknesses of homogeneous vinyl separated by a 3/8 inch layer of expanded unicellular vinyl. This material has excellent insulating and noise suppression qualities, and is commercially available.

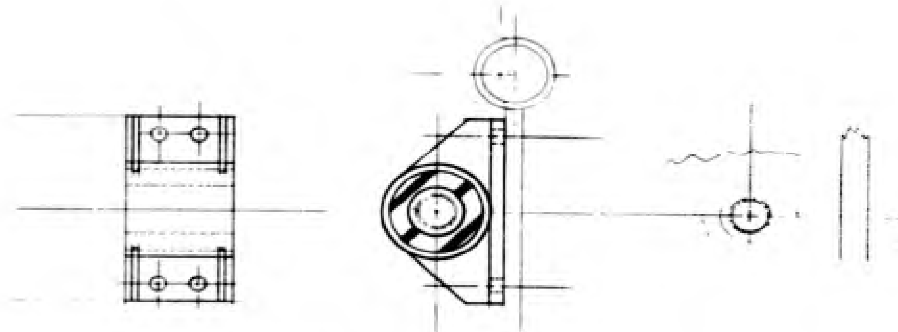
Three removable portions of the armored top deck allow engine and power train removal. These are (1) the raised deck over the engine, (2) air inlet louvers, and (3) the deck just forward of the louvers.



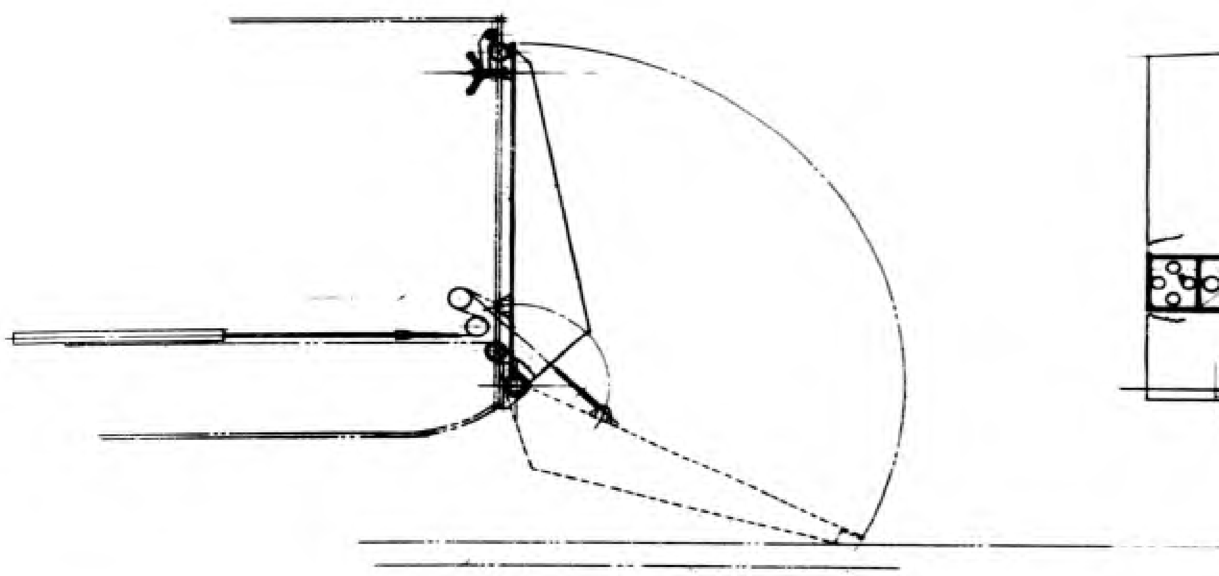
LOCK

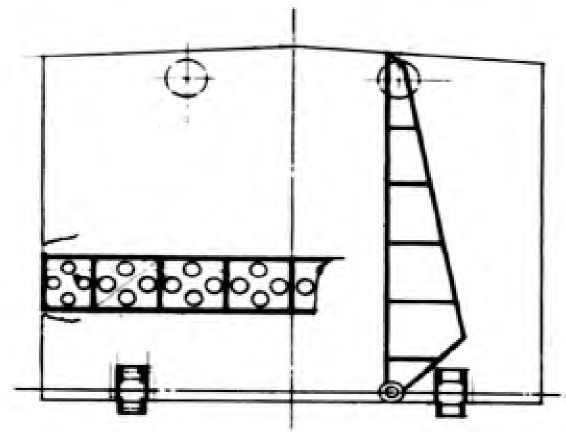


SEAL (ALTERNATES)



TORSION SPRING





PROPOSAL

RAMP ASSY

NOV 3 - 1961 | TWB

1/8

6496-22

PACIFIC CAR AND FOUNDRY COMPANY

CONTROLS SYSTEM

The manual controls, reference Drawing 6496-35, include steering wheel and linkage, hand and foot brake, hand and foot throttle controls, shift lever and linkage, pivot steer selector, engine shut-off control, and bow vane actuating mechanism. Two instrument panels are also shown in position, but not detailed.

Three types of controls are used. Cables and pulleys are used on the brakes. Flexible push-pull cables are used on the hand throttle, engine shut-off, and engine-transmission throttle control linkages. Bell-crank and rod arrangements are used elsewhere.

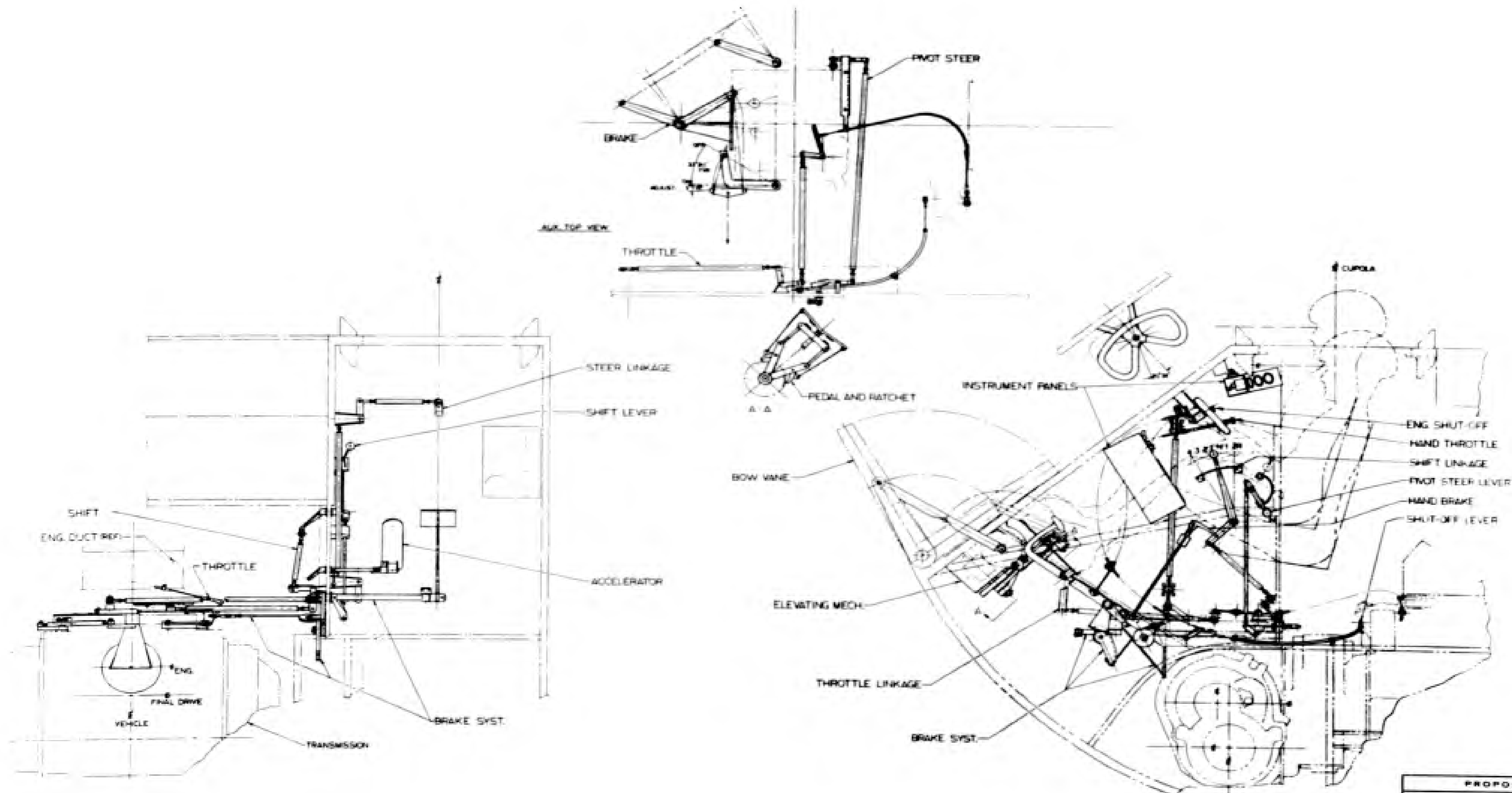
A special feature in the braking system is the elliptical cam-pulley which enables the operator to obtain a 100 lb.ft. torque on each transmission brake shaft with but a 20 lb. pedal load. Adjustments in the brake system can be made by two turnbuckles in the cable system or by changing the mating positions of the gear and sector. The maximum tension load in the 5/32 diameter stainless steel cable is 270 lbs., approximately. The hand brake is an automotive, 90 degree, turn-to-lock type.

The flexible controls shown are similar to the Arens Button Lock, Model 21-001. The engine compartment section of the flexible engine shut-off control is a flexible control inside a formed, rigid, metal tube.

All push rods shown are of welded tube and end stud construction. The rod ends shown are single-ball, commercial types similar to that shown on Ordnance Dept. Drawing 7536600, (Heim Co. Brg. No. HF-4, or equivalent).

The bow vane actuator control elevates and retracts the bow vane by pumping a foot pedal similar to the brake pedal. The foot pedal is linked to a ratchet wheel and mechanism which turns an irreversible worm and wheel set which, in turn, elevates the vane by means of a torque rod and arm and link system.

All loads, torques, and displacements have been calculated. The controls have been positioned to allow operation by both short and tall drivers in the two positions shown. Seat adjustment will be on a line 30° to the vertical, up and forward to down and backward.

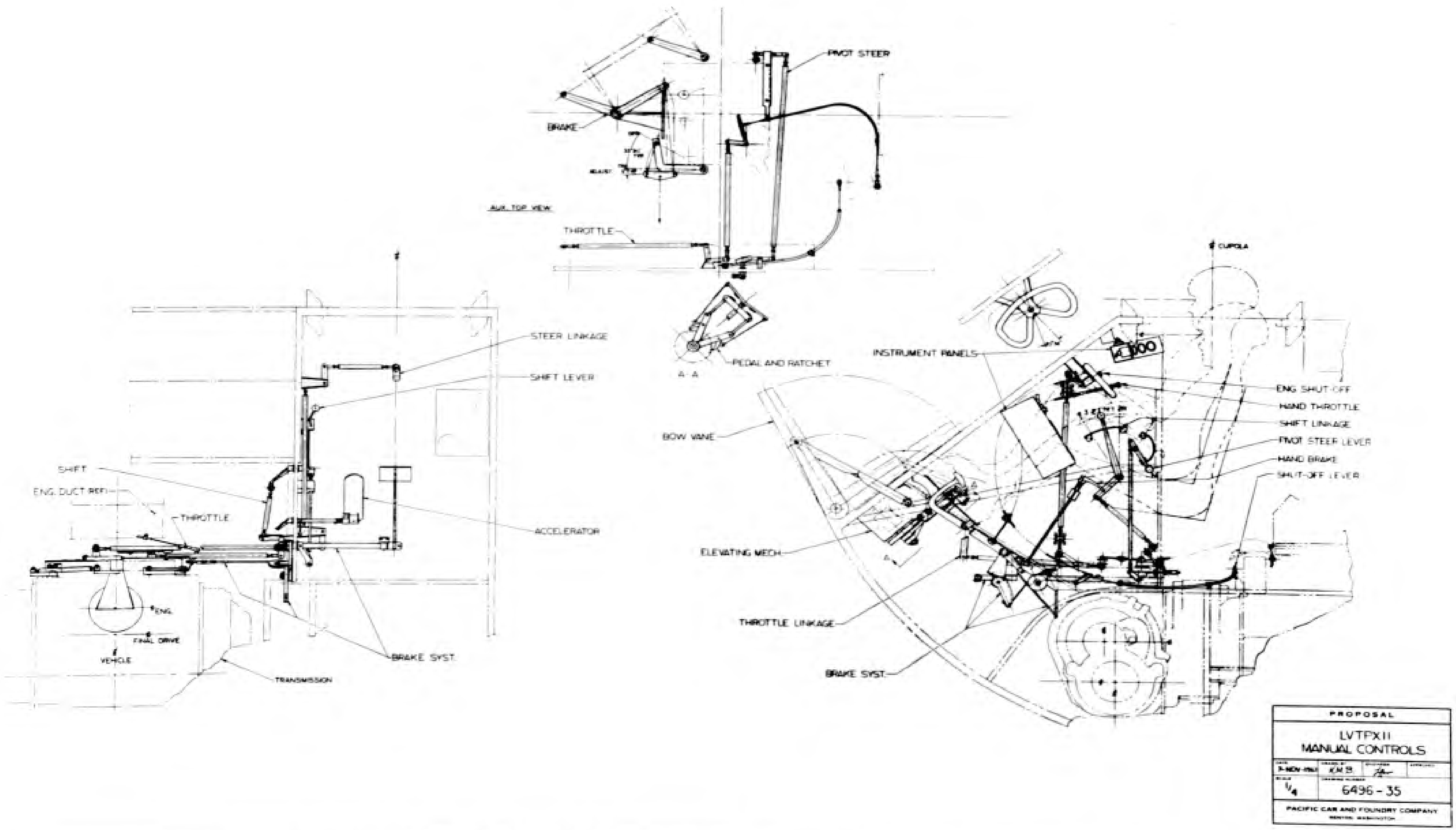


PROPOSAL

LVTPX11
MANUAL CONTROLS

7-NOV-1961 | KMB
1/4 | 6496-35

PACIFIC CAR AND FOUNDRY COMPANY


PIVOT STEER
BRAKE
AUX. TOP VIEW
THROTTLE
PEDAL AND RATCHET
A - A
STEER LINKAGE
SHIFT LEVER
SHIFT
ENG. DUCT REF.
THROTTLE
ACCELERATOR
ENG.
FINAL DRIVE
VEHICLE
BRAKE SYST.
TRANSMISSION
CUPOLA
INSTRUMENT PANELS
ENG SHUT-OFF
HAND THROTTLE
SHIFT LINKAGE
PIVOT STEER LEVER
HAND BRAKE
SHUT-OFF LEVER
BOW VANE
ELEVATING MECH.
THROTTLE LINKAGE
BRAKE SYST.
PROPOSAL
LVTPX11
MANUAL CONTROLS
1/4
6496 - 35
PACIFIC CAR AND FOUNDRY COMPANY

ENGINE AND POWER TRAIN

The engine and power train, reference drawing 6496-37, Sheets 1 and 2, form a single integral unit located in the front of the vehicle.

Engine

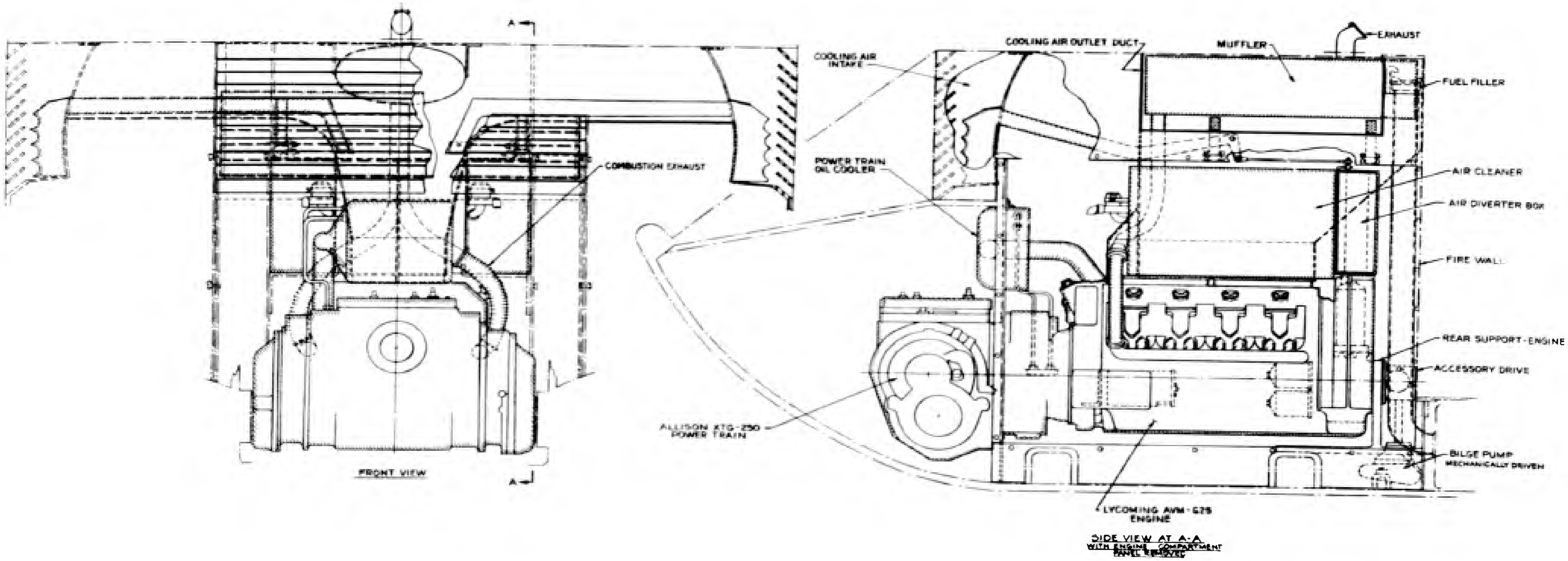
The Lycoming engine is an air-cooled, multi-fuel, compression ignition, loop scavenged engine with the following characteristics:

Cylinder Arrangement	90°V
Number of Cylinders	8
Gross Power	310 HP
Net Power	250 HP
Peak Torque	655 lb.ft.
Maximum Speed	2600 RPM
Minimum Fuel Consumption	.43 lbs/BHP/hr
Cycle	2
Displacement	625 cu.in.
Compression Ratio	18.7 to 1
Bore and Stroke	4.25 x 5.5 inches
Aspiration Air Required	1320 cfm @ 60°F
Maximum Inlet Aspiration Air Pressure Drop (Air Cleaner, Etc.)	20 in. H_2O
Exhaust Temperature	650°F
Maximum Exhaust Aspiration Air Back Pressure	2 in. Hg
Cooling Air Flow	11,260 cfm @ 100°F
Cylinders	5,730 cfm @ 100°F
Engine Oil Cooler	4,010 cfm @ 100°F
Transmission Oil Cooler	1,520 cfm @ 100°F

Maximum Cooling Air Pressure Drop	9 in. H_2O
Internal	5 in. H_2O
Louvers and Ducts	4 in. H_2O
Heat Rejection	16,280 BTU's/min
Cylinders	11,700 BTU's/min
Engine Oil Cooler	2,580 BTU's/min
Transmission	2,000 BTU's/min
Temperature, Cooling Fan Out	185°F
Length	46 inches
Width	32 inches
Height	29 inches
Weight	1000 pounds

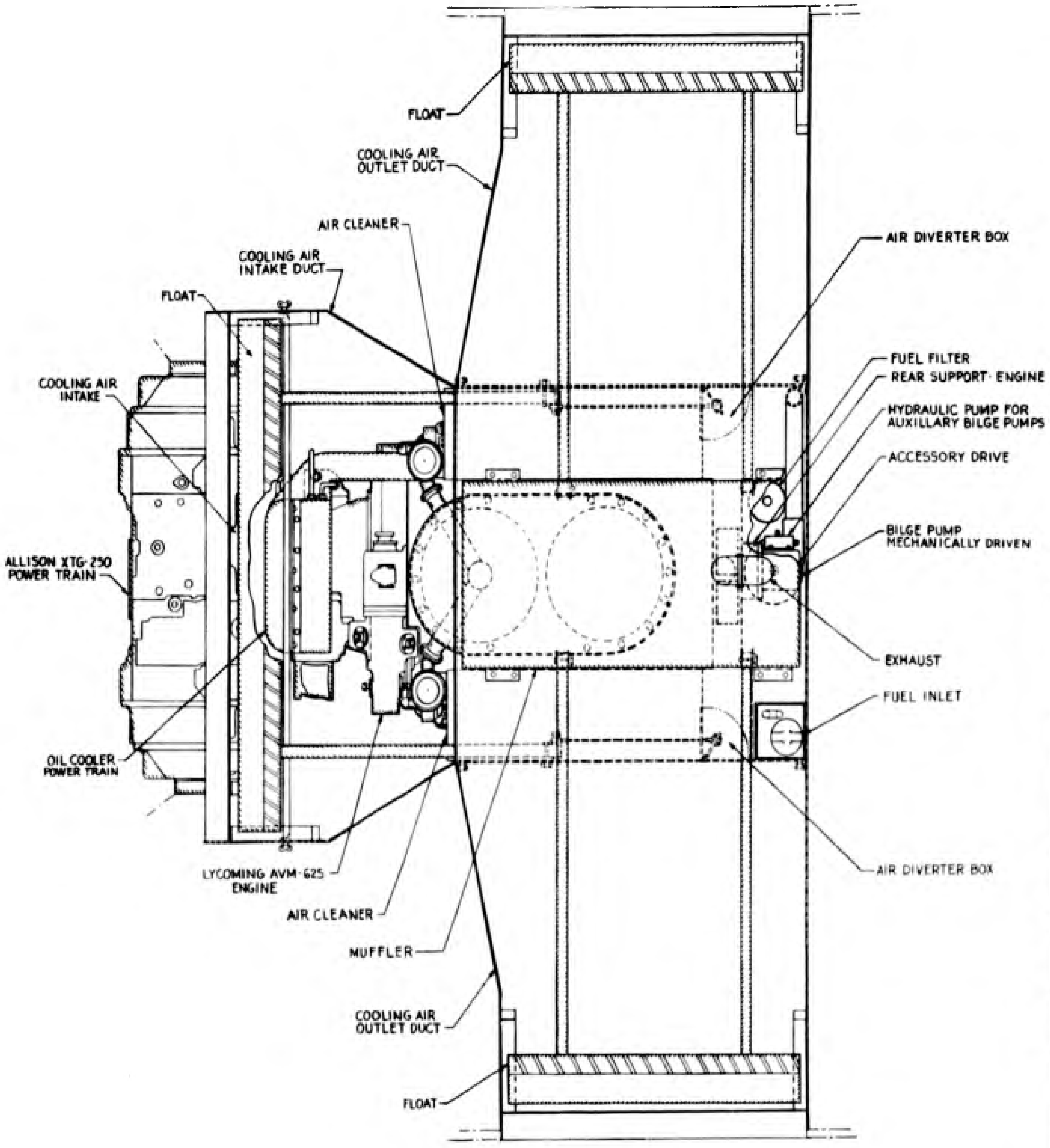
The basic arrangement and over-all dimensions of this engine are shown on Lycoming Drawing No. 63169. Two cooling air fans are mounted between the cylinder banks and draw air through the engine and exhaust it upwards. Provisions are included for cooling the power train. The fuel injection pump, starter, and alternator are mounted on the flywheel housing. The scavenge air blower, oil cooler, and power take-off are located at the accessory end.

Gross and net engine performance curves are shown on Figs. 14 through 17. These curves are self-explanatory, however, it should be noted that since their publication, information has been received that engine speed will be limited to a maximum of 2600 RPM. The AVM-625 has the capability of a 30% horsepower increase by use of a turbocharger.



PROPOSAL		
LVTP XII MACHINERY COMPARTMENT		
DATE 11-6-1961	DRAWN BY PIQ	ENGINEER HH
SCALE	DRAWING NUMBER 6496-37 SH 1	
PACIFIC CAR AND FOUNDRY COMPANY RENTON, WASHINGTON		

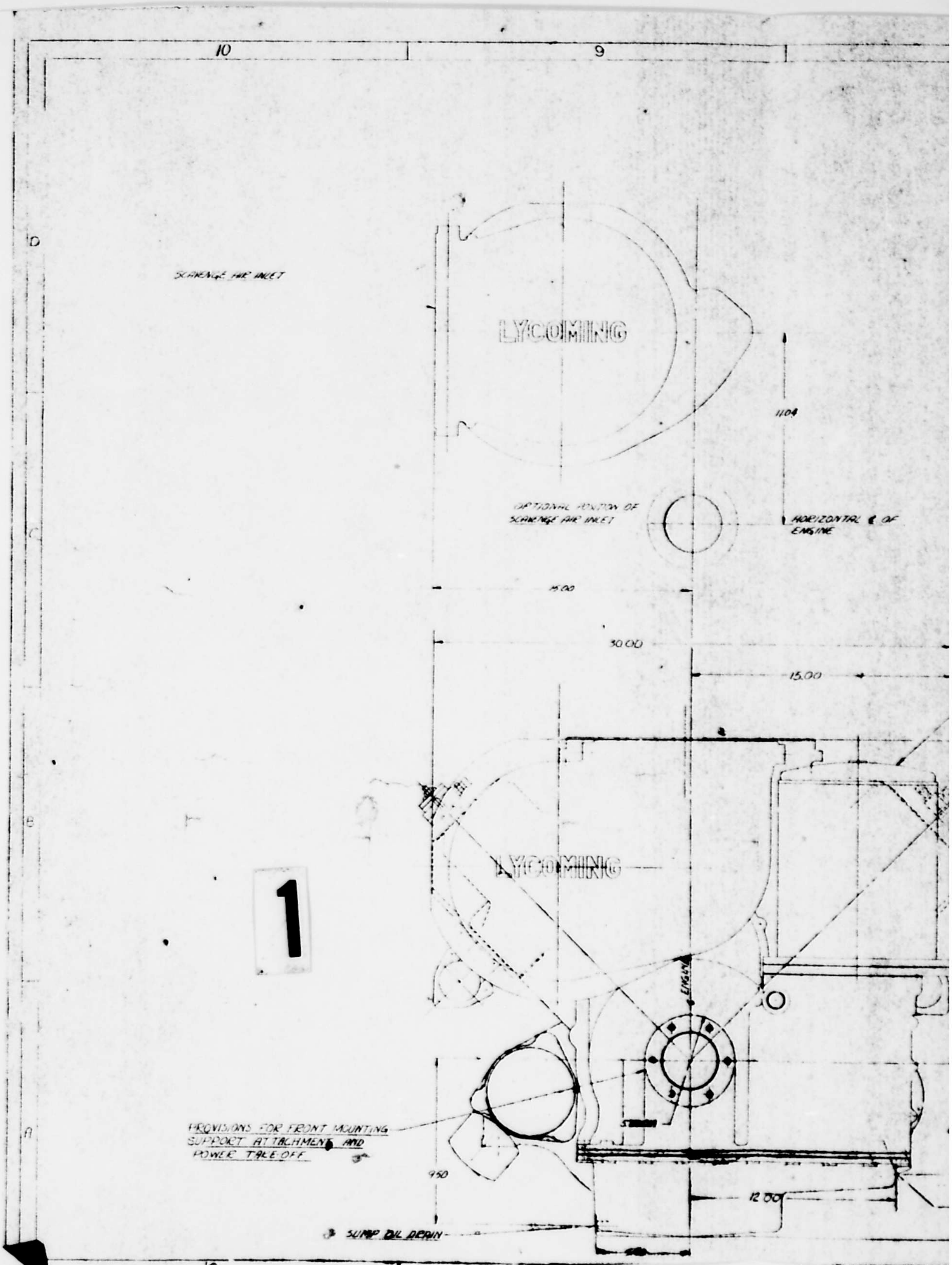
6496-37 SHEET 1



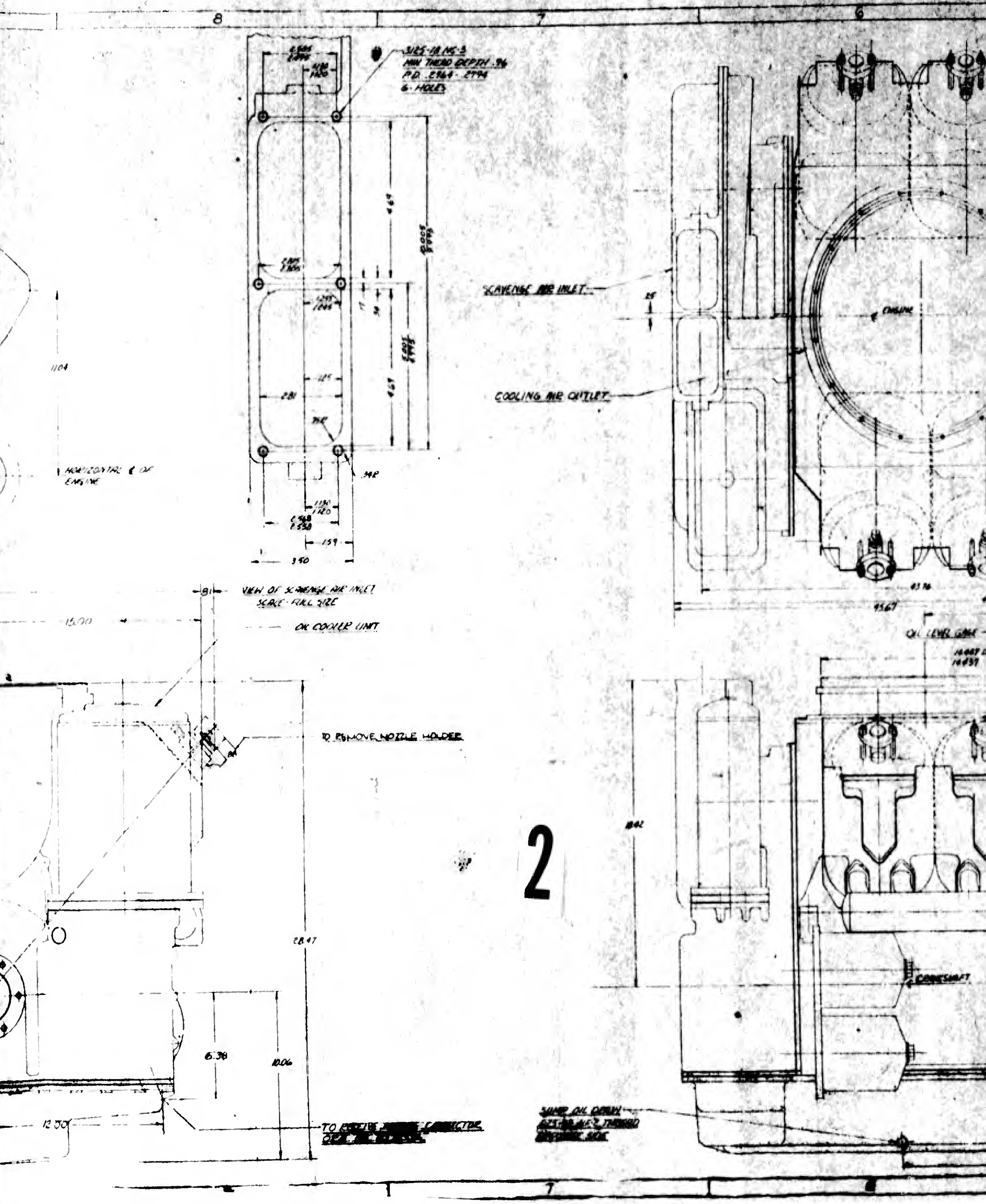
PLAN VIEW
WITH TOP PLATE REMOVED

6496-37 SHEET 2

PROPOSAL
LVTP XII
MACHINERY COMPARTMENT
11-6-1961
6496-37 Sht 2
PACIFIC CAR AND FOUNDRY COMPANY
RENTON, WASHINGTON

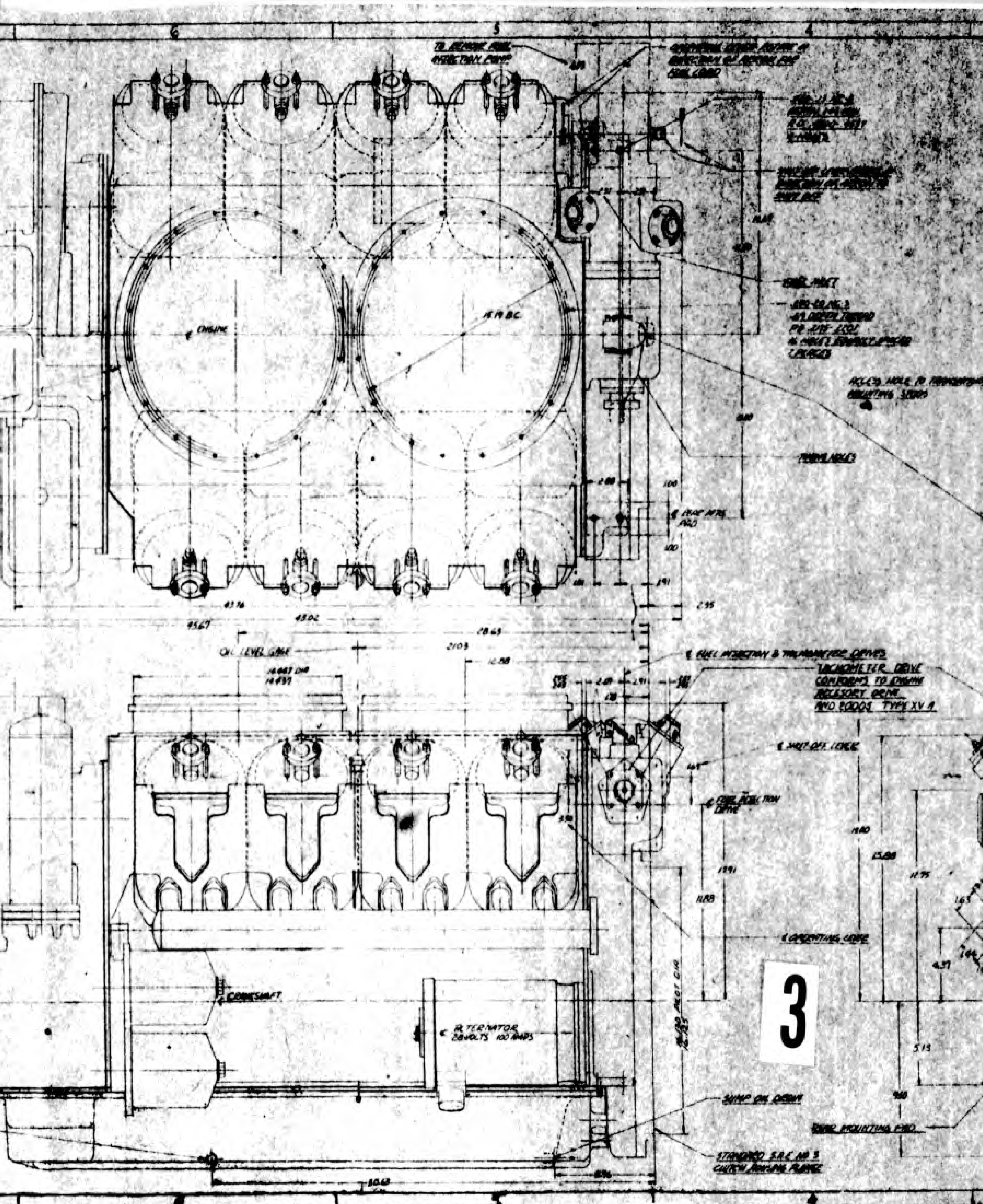


10
9
D
SCAVENGE AIR INLET
LYCOMING
11.04
OPTIONAL POSITION OF SCAVENGE AIR INLET
HORIZONTAL ℄ OF ENGINE
15.00
30.00
15.00
C
B
LYCOMING
1
A
PROVISIONS FOR FRONT MOUNTING SUPPORT ATTACHMENT AND POWER TAKE-OFF
9.50
12.00
SUMP OIL DRAIN
10

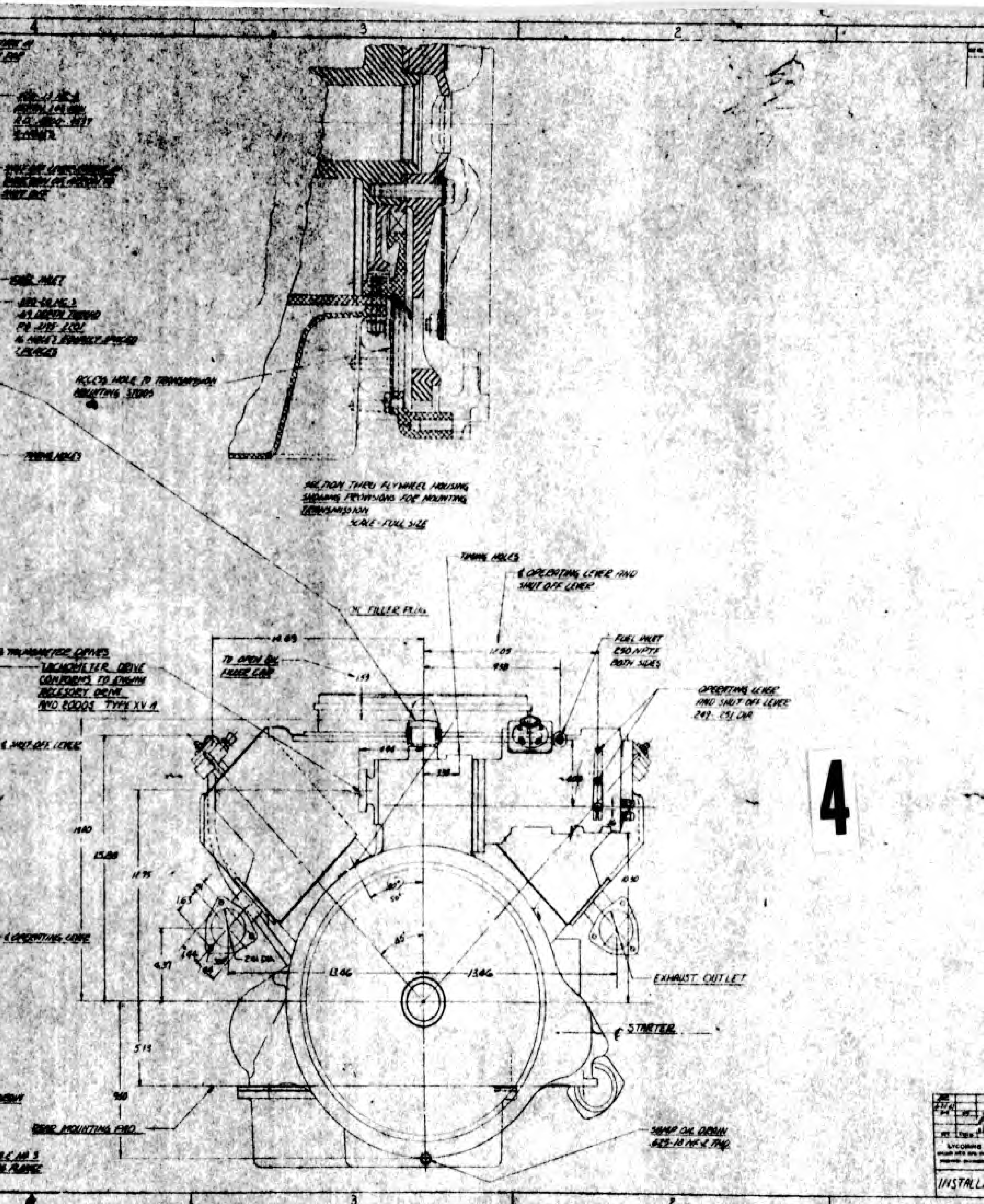


SCAVENGE AIR INLET
COOLING AIR OUTLET
HORIZONTAL ℄ OF ENGINE
VIEW OF SCAVENGE AIR INLET
SCALE - FULL SIZE
OIL COOLER UNIT
TO REMOVE NOZZLE HOLDER
SUMP OIL DRAIN
CAMSHAFT
OIL LEVEL GAGE
ENGINE

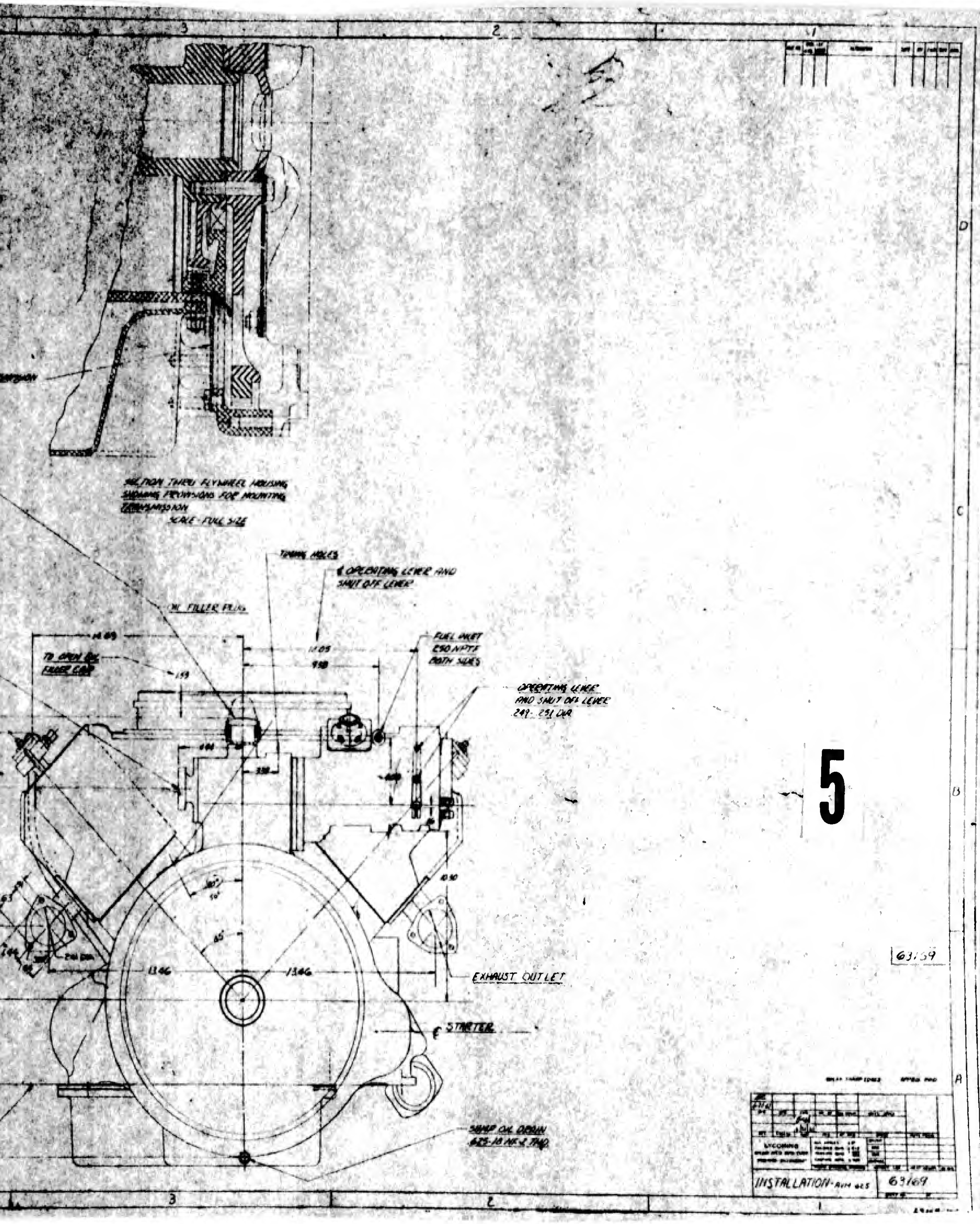
2



TO REMOVE FUEL INJECTION PUMP
ENGINE
OIL LEVEL GAGE
FUEL INJECTION & TACHOMETER DRIVES
TACHOMETER DRIVE CONFORMS TO ENGINE ACCESORY DRIVE AND 20005 TYPE XV A
SHUT-OFF LEVER
OPERATING LEVER
CRANKSHAFT
ALTERNATOR 28 VOLTS 100 AMPS
SUMP OIL DRAIN
REAR MOUNTING PAD
STANDARD S.A.E. NO 3 CLUTCH HOUSING FLANGE
3



ACCESS HOLE TO TRANSMISSION MOUNTING STUDS
SECTION THRU FLYWHEEL HOUSING SHOWING PROVISIONS FOR MOUNTING TRANSMISSION
SCALE - FULL SIZE
TIMING HOLES
OPERATING LEVER AND SHUT OFF LEVER
OIL FILLER PLUG
FUEL INLET
.250 NPTF
BOTH SIDES
TO OPEN OIL FILLER CAP
TACHOMETER DRIVE
OPERATING LEVER AND SHUT OFF LEVER
18.69
12.05
9.18
13.46
13.46
EXHAUST OUTLET
STARTER
REAR MOUNTING PAD
SUMP OIL DRAIN
4
LYCOMING
INSTALL

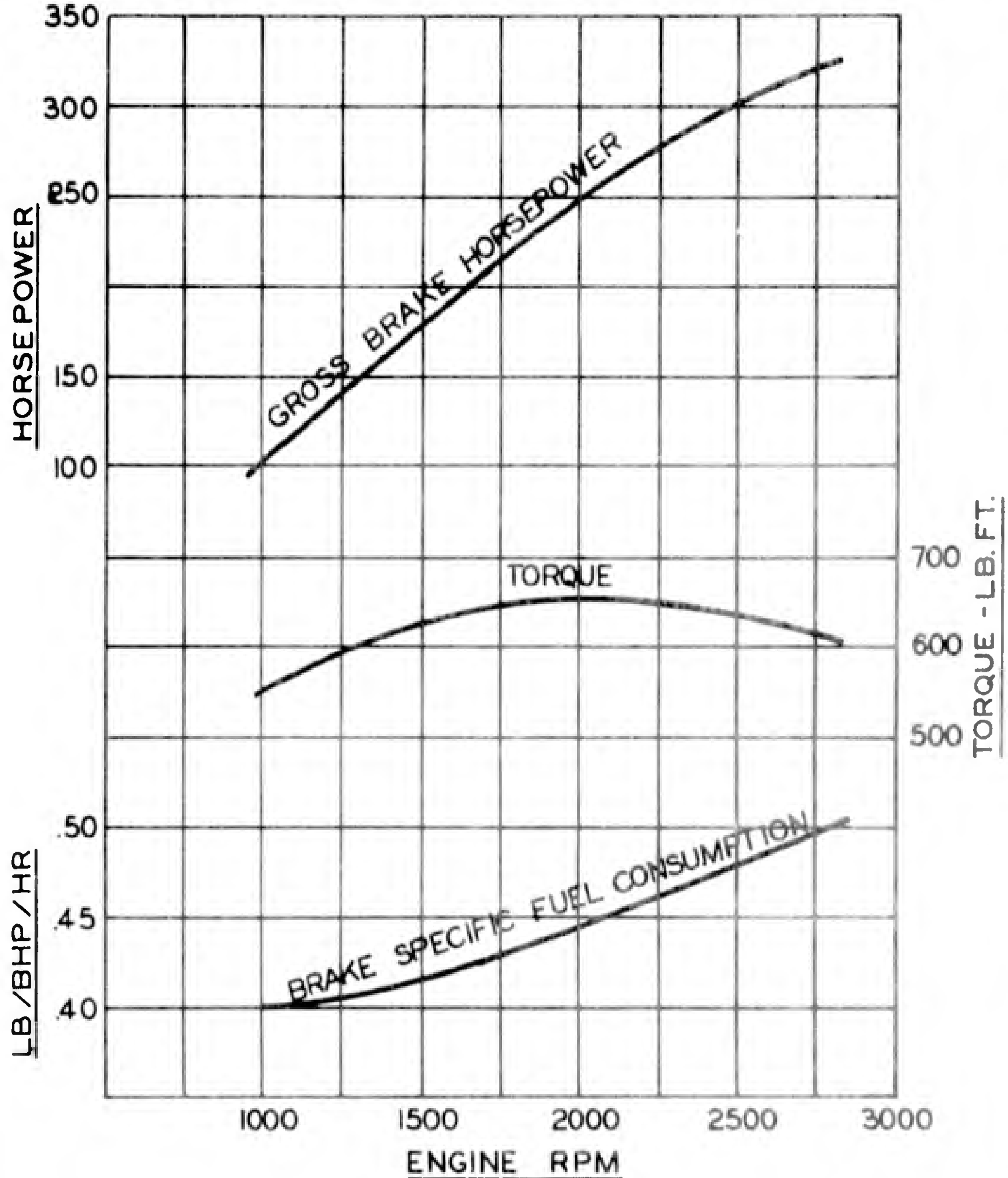


SECTION THRU FLYWHEEL HOUSING SHOWING PROVISIONS FOR MOUNTING TRANSMISSION
SCALE - FULL SIZE
TIMING HOLES
℄ OPERATING LEVER AND SHUT OFF LEVER
OIL FILLER PLUG
TO OPEN OIL FILLER CAP
FUEL INLET .250 NPTF BOTH SIDES
OPERATING LEVER AND SHUT OFF LEVER .249-.251 DIA
EXHAUST OUTLET
℄ STARTER
SUMP OIL DRAIN .625-18 NF-2 THD
13.46
13.46
INSTALLATION - AVM 625
63169

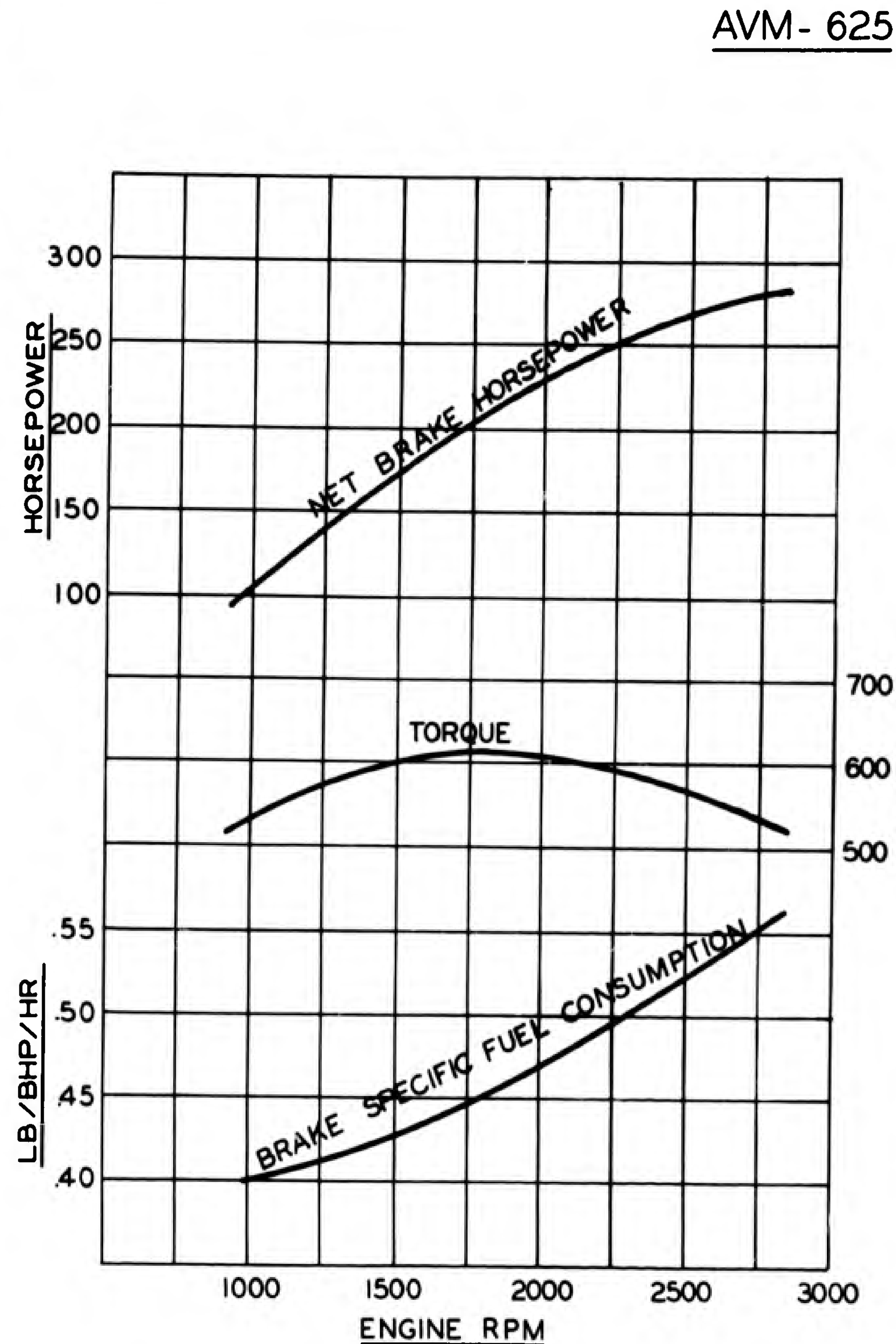
5

63169

AVM-625



ESTIMATED PERFORMANCE
AVM-625 8 CYL. ENGINE



ESTIMATED PERFORMANCE
AVM-625 8 CYL. ENGINE

FIGURE 15

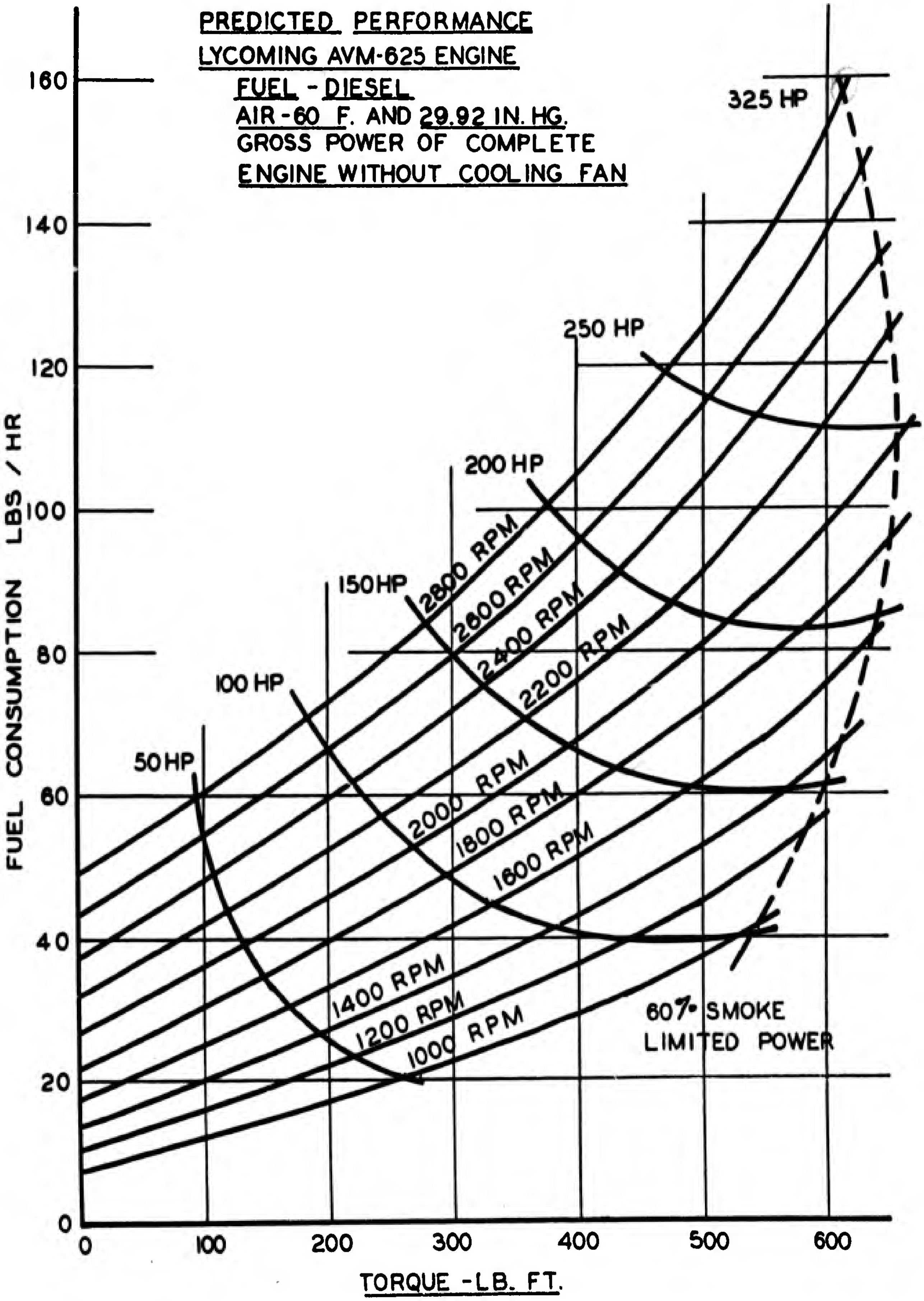

PREDICTED PERFORMANCE
LYCOMING AVM-625 ENGINE
FUEL - DIESEL
AIR - 60 F. AND 29.92 IN. HG.
GROSS POWER OF COMPLETE
ENGINE WITHOUT COOLING FAN
FUEL CONSUMPTION LBS / HR
TORQUE - LB. FT.
0
20
40
60
80
100
120
140
160
0
100
200
300
400
500
600
50 HP
100 HP
150 HP
200 HP
250 HP
325 HP
2800 RPM
2600 RPM
2400 RPM
2200 RPM
2000 RPM
1800 RPM
1600 RPM
1400 RPM
1200 RPM
1000 RPM
60% SMOKE
LIMITED POWER

FIGURE 16

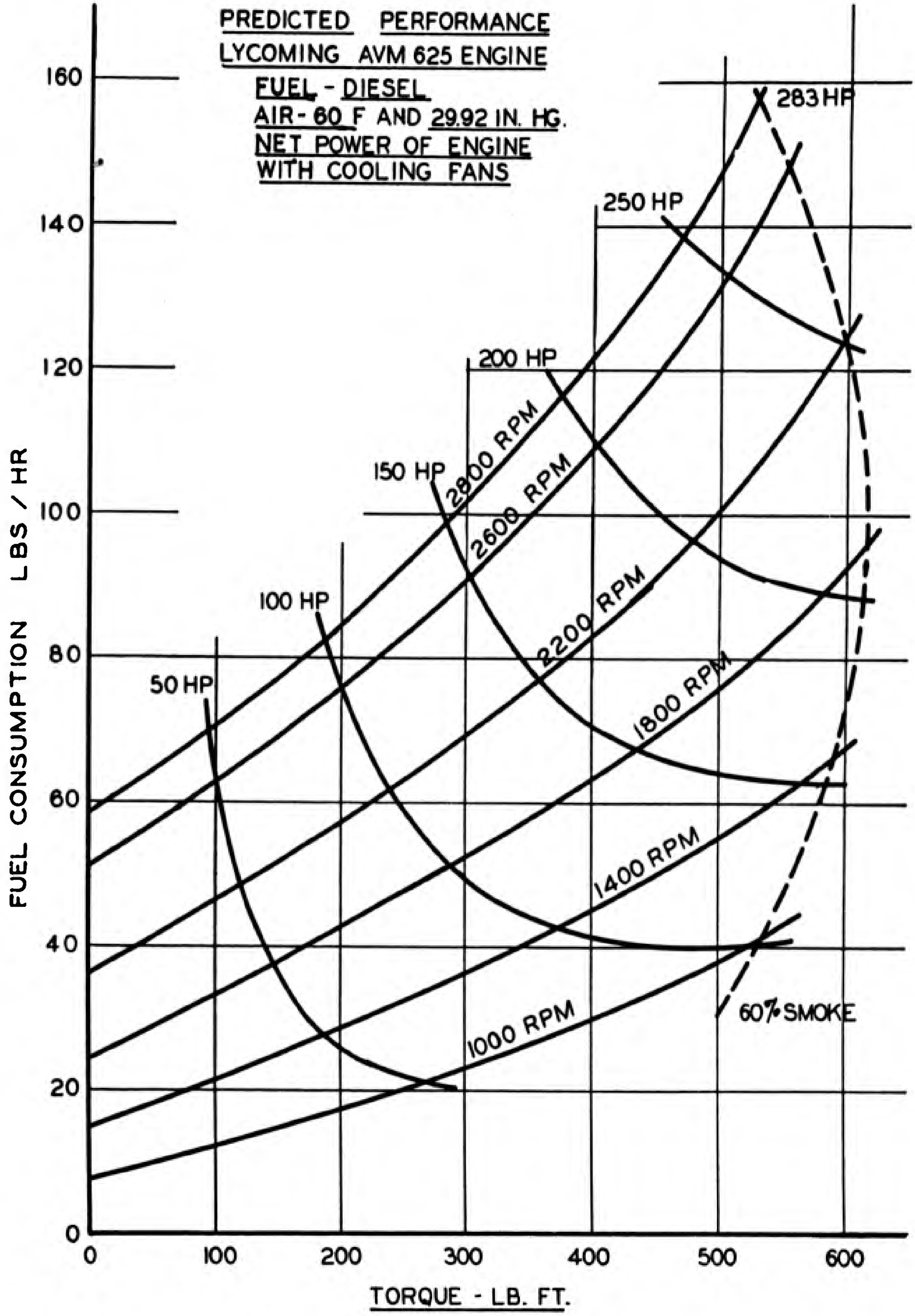

PREDICTED PERFORMANCE
LYCOMING AVM 625 ENGINE
FUEL - DIESEL
AIR - 60 F AND 29.92 IN. HG.
NET POWER OF ENGINE
WITH COOLING FANS
283 HP
250 HP
200 HP
150 HP
100 HP
50 HP
2800 RPM
2600 RPM
2200 RPM
1800 RPM
1400 RPM
1000 RPM
60% SMOKE
FUEL CONSUMPTION LBS / HR
TORQUE - LB. FT.
0
20
40
60
80
100
120
140
160
0
100
200
300
400
500
600

FIGURE 17

Power Train

General operation of the power train was explained in Part I; detailed operation is explained in this section.

Information covering the operation and general arrangement of the XTG-250 power train was obtained from the Allison Division of General Motors Corporation and is included as Figures 18 through 21. This information was based on the XTG-250 matched to the Lycoming AVM-625 power source characteristics. However, at high speeds in the converter ranges, torque is noted to decrease. Because the engine speed is near maximum at these conditions, power drops off due to characteristic droop in the engine overspeed governor. Performance will not be sacrificed since the converter will normally be locked out.

Since the required final output gear ratio varies from that presently being incorporated into the power train (ARAAV has a 13 inch sprocket, the LVTPX11, a 24 inch sprocket) no final reduction has been shown in Fig. 20.

A lower final drive ratio is needed. To accommodate this, the output gears and housings must be modified. The unit can be supplied without a final reduction as shown on Fig. 21. However, integral final reductions have the following advantages over separate final drives:

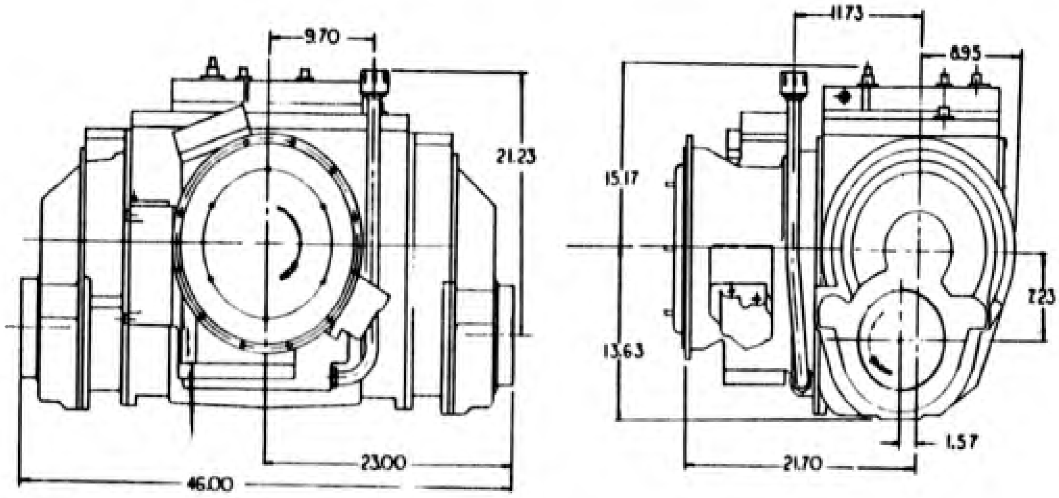
1. No separate lubrication system is required.
2. Cradling the unit in the sprocket housings is an easier method of mounting.
3. Drive shaft universal joints are not needed.
4. Over-all weight of the power train and sprocket assemblies should be less.

An air-to-oil cooler is mounted above the power train to cool power train oil and control oil. Air is drawn into the cooler by the engine cooling fans through a duct running from the cooler to a flange on the flywheel end of the engine.

Sheet 1 of 2
4/14/61

PRELIMINARY
CHARACTERISTIC SHEET

Component: Power Train, X-Drive, Model XTG-250 Hydraulic Torque Converter, Planetary Gear Type, all Torque Shifting.



The XTG-250 includes a hydraulic torque converter with a lockup clutch. The planetary range gearing in combination with the steer and output planetary sets provides four forward and two reverse ranges. The transmission also incorporates geared steer, clutch brake, and pivot steer systems and full vehicle brakes as outlined in the following specifications:

GENERAL SPECIFICATIONS

Rating:
- Max. Input Torque Lbs. Ft. -- 520
- Max. Input Speed RPM --2800
- Max. Input HP -- 250

Manufacturer ------------------------------------Allison Division, GMC
Model ---XTG-250
Drive Ranges--Fourth, Third, Second, First, Neutral, Reverse 1, and Reverse 2
Drive Range, Steering, and Shift Control (External) --------------Mechanical
Shift and Steering Mechanism (Internal Control)-------Hydraulic & Mechanical
Steering Type --Geared Steer
Clutch Brake
Pivot Steer
Power Take-off --Turbine Driven
Rating (Continuous Operation) -----------------------------------470#'
Type of Clutches (All Ranges)------Multiple Plate, Engaged by Oil Pressure
Brake--------Multiple Wet Plate, Steering, Service, and Parking, Mechanical Application
Hydraulic Torque Converter-------Single Stage, Multiple Phase with Lockup Clutch
Maximum Converter Multiplication ----------------------------------2.55:1

FIGURE 18

XTG-250

Range Position	Gear Ratio	Steering Normal	Steering Pivot
R_2	2.095:1	C.B.	Pivot
R_1	4.69:1	C.B.	Pivot
N	-	-	-
1	6.16:1	C.B.	Pivot
2	4.18:1	G.S.	Pivot
3	2.24:1	G.S.	G.S.
4	1.00:1	G.S.	G.S.

Bevel Gear Ratio -- 1.44:1
Output Steer Planetary Ratios:
  Under Drive -- 1.475:1
  Direct -- 1.00:1
Final Drives (Integral) Ratio --- 2.22:1
Total Torque Ratio Coverage:
  First Gear Stall ------------------------- 2.55 x 6.16 x 1.44 x 2.22 = 50.3:1
  Fourth Gear Lockup -------------------- 1.00 x 1.44 x 2.22 = 3.20:1
Oil Specifications ---MIL-L-2104A, Grade 10
Oil Capacity, Gal. -- (Est.) 10
Transmission Dry Weight Lbs. --(Est.) 1290

FIGURE 19

XTG-250 POWER TRAIN SCHEMATIC

(STRAIGHT THRU VERSION)

RANGE SECTION

HIGH CLUTCH	1.00:1
INTERMEDIATE CLUTCH	2.24:1
LOW CLUTCH	4.18:1

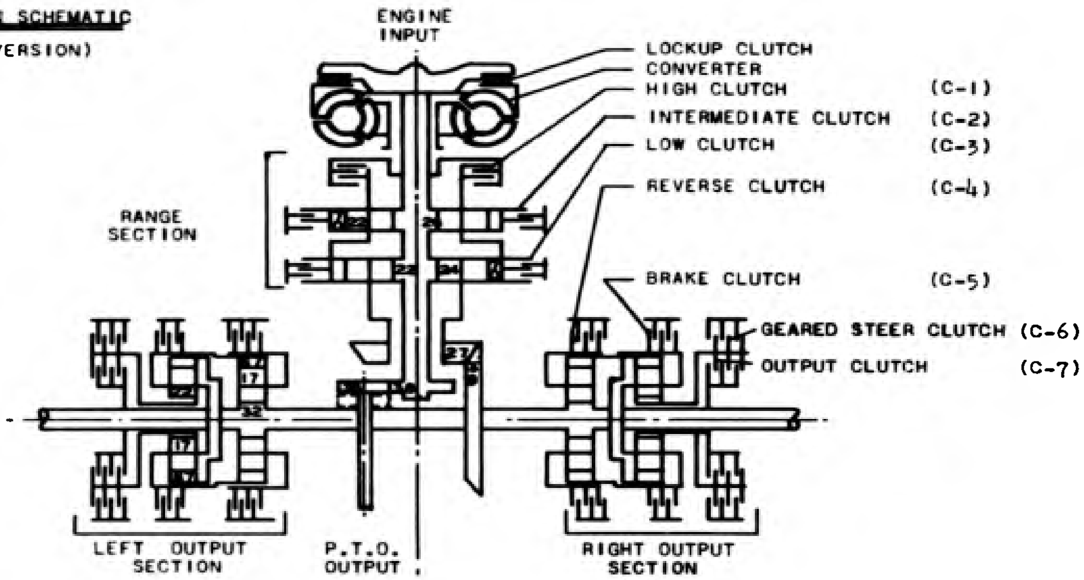
OUTPUT SECTION

REVERSE	2.095:1
GEARED STEER CLUTCH	1.478:1
OUTPUT CLUTCH	1.00:1

BEVEL GEAR RATIO 1.444:1

TRANSMISSION GEAR RATIOS

$REVERSE_1$	2.095:1
$REVERSE_2$	4.69:1
NEUTRAL	-
FIRST	6.178:1
SECOND	4.18:1
THIRD	2.24:1
FOURTH	1.00:1



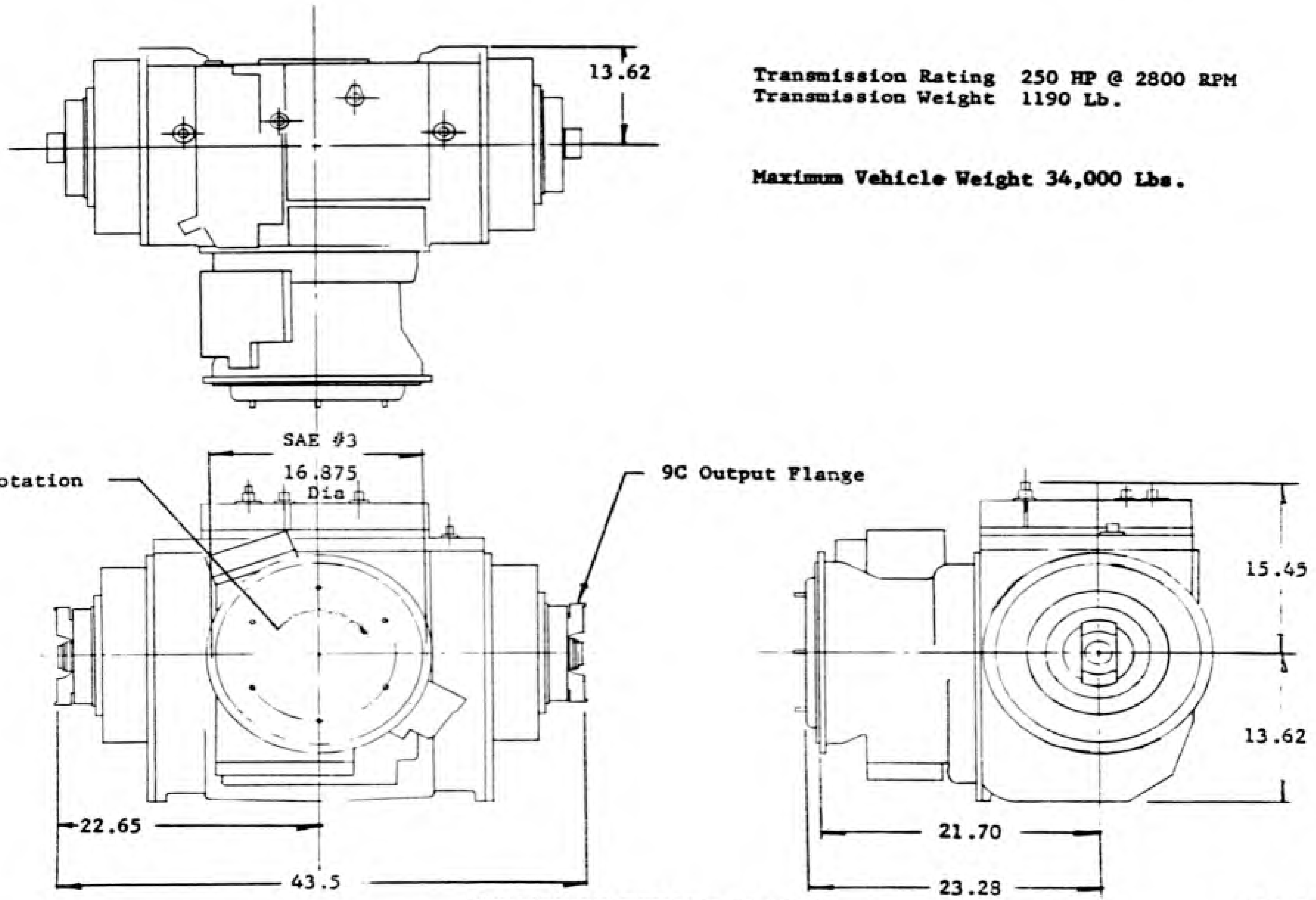
SELECTOR POSITION	RANGE	NORMAL FORWARD OR REVERSE OPERATION	STEERING: NORMAL OPERATION: TYPE	NORMAL OPERATION: CLUTCHES ENGAGED: OUTER TRACK	NORMAL OPERATION: CLUTCHES ENGAGED: INNER TRACK	NORMAL OPERATION: OUTER TRACK OVER-ALL RATIO	PIVOT OPERATION: TYPE	PIVOT OPERATION: CLUTCHES ENGAGED: OUTER TRACK	PIVOT OPERATION: CLUTCHES ENGAGED: INNER TRACK	PIVOT OPERATION: OUTER TRACK OVER-ALL RATIO
R_2	$REVERSE_2$	C-1, C-4	C.B.	C-4	C-5	3.026:1	PIVOT	C-4	C-6	3.025:1
R_1	$REVERSE_1$	C-2, C-4	C.B.	C-4	C-5	6.776:1	PIVOT	C-4	C-6	6.776:1
N	NEUTRAL	-	-	-	-	-	-	-	-	-
1	FIRST	C-3, C-6	C.B.	C-6	C-5	8.921:1	PIVOT	C-6	C-4	8.921:1
2	SECOND	C-3, C-7	G.S.	C-7	C-6	6.036:1	PIVOT	C-6	C-4	8.921:1
3	THIRD	C-2, C-7	G.S.	C-7	C-6	3.235:1	G.S.	C-7	C-6	3.235:1
4	FOURTH	C-1, C-7	G.S.	C-7	C-6	1.444:1	G.S.	C-7	C-6	1.444:1

A.C.H.
3/15/61
REV. 10/31/61

FIGURE 20

Transmission Rating 250 HP @ 2800 RPM
Transmission Weight 1190 Lb.

Maximum Vehicle Weight 34,000 Lbs.



PRELIMINARY ENVELOPE DRAWING
XTG-250 POWER TRAIN (LESS FINAL DRIVES)

XAS 61-001
8-11-61
A.C.H.

FIGURE 21

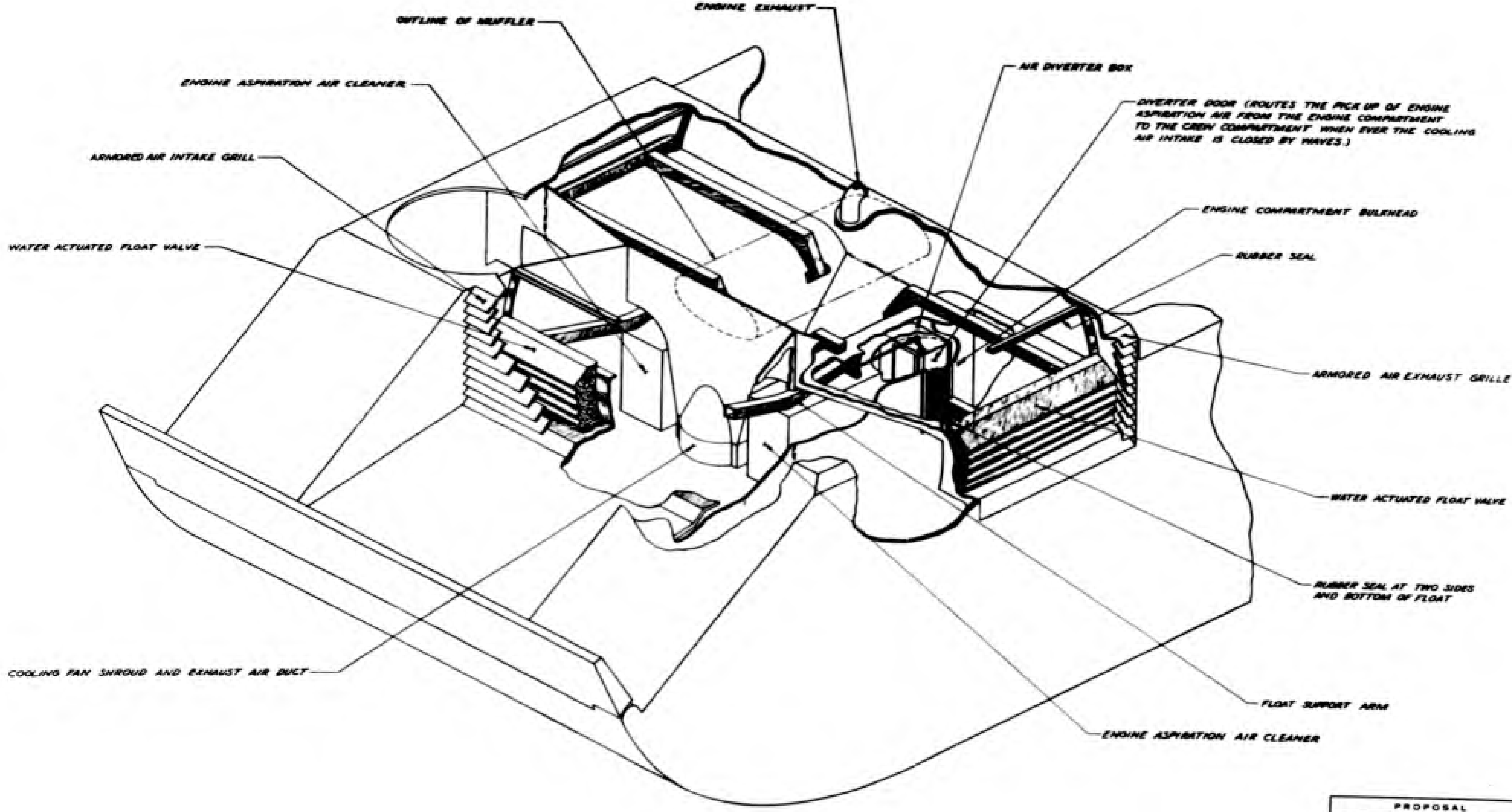
COOLING AIR SYSTEM

A single cooling air inlet is located between the sloping portions of the front of the hull while two cooling air outlets are placed on each side of the vehicle just aft of the driver and gunner. Air enters the vehicle through fixed armored louvers and flows down over the power train, alongside the engine, and is drawn through the engine and oil coolers by the engine cooling fans. Air discharged from the fans passes through a duct above the engine, flows past the muffler and is exhausted through fixed armored louvers in both sides of the hull. A relatively unrestricted flow path results with this arrangement.

Taking the air in at the front and exhausting it out the sides should effectively prevent recirculation of cooling air and consequent overheating of the engine and power train. This location for the air inlet was also selected because it will be relatively well-protected during water operation. The extended bow vane deflects the bow wave around the sides. The depression between the vane and hull should produce rapid draining, aided by the trough of the bow wave being directly under the depression. This desirable condition is shown on page 1.14, Figure 11 of the model tow test.

The louvers are made of steel armor to limit their volume and the consequent air restriction. They are straight slats without ballistic traps. Their purpose is to deflect the particles upward so their remaining energy can be dissipated against the armored sides and top of the air inlet and outlet compartments.

The floats, which were developed to limit entrance of water into the vehicle during rough water operation, are located just inside the louvers. They are supported on long arms to minimize influence of the



PROPOSAL

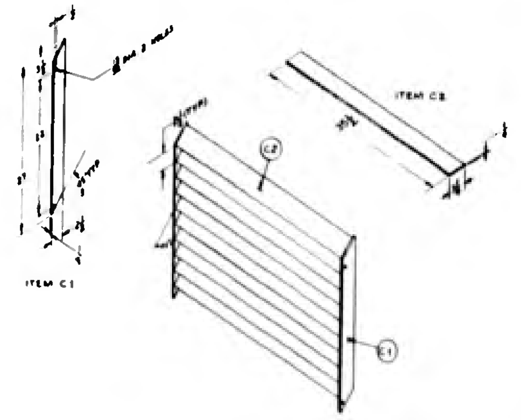
LVTPX11

ENGINE COOLING SYSTEM

6496-41

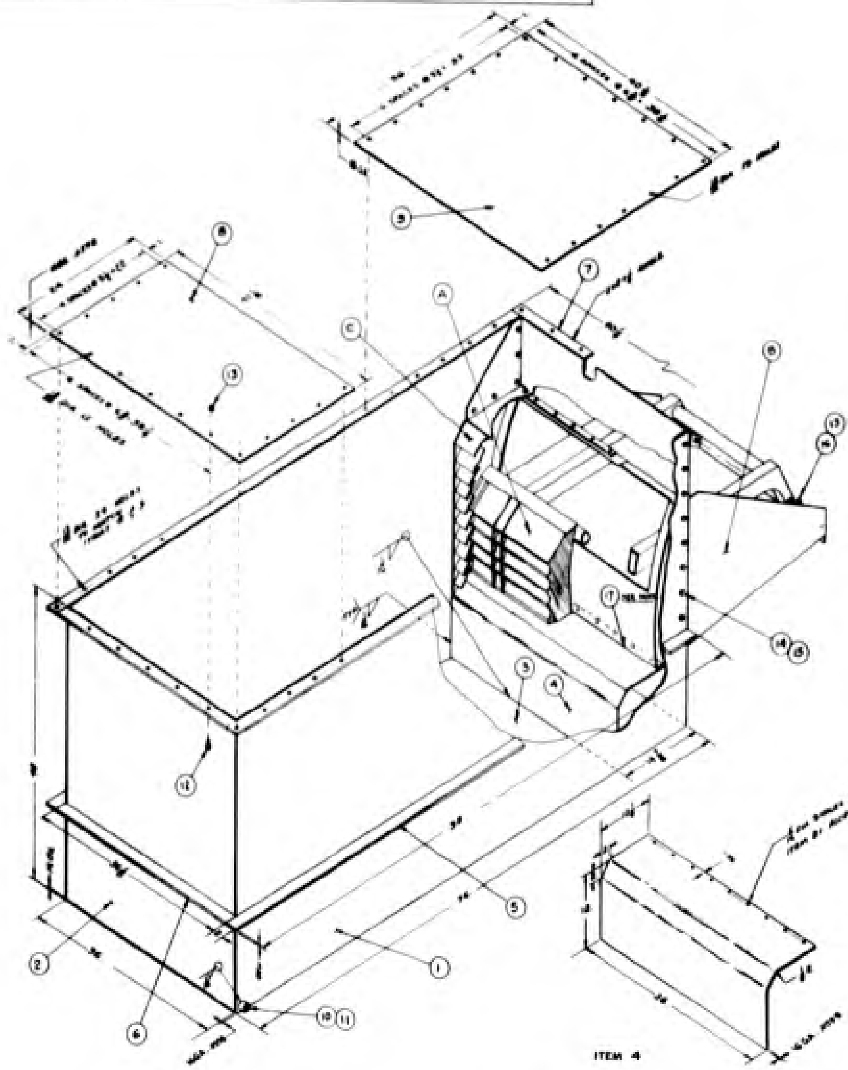
PACIFIC CAR AND FOUNDRY COMPANY

BILL OF MATERIAL			
ITEM NO.	NO. REQ'D	SIZE	MATERIAL
C1	2	[illegible] 2'-5"	MILD STEEL
C2	10	[illegible] 3/16 x 2 - [illegible]	



ASSEMBLY C

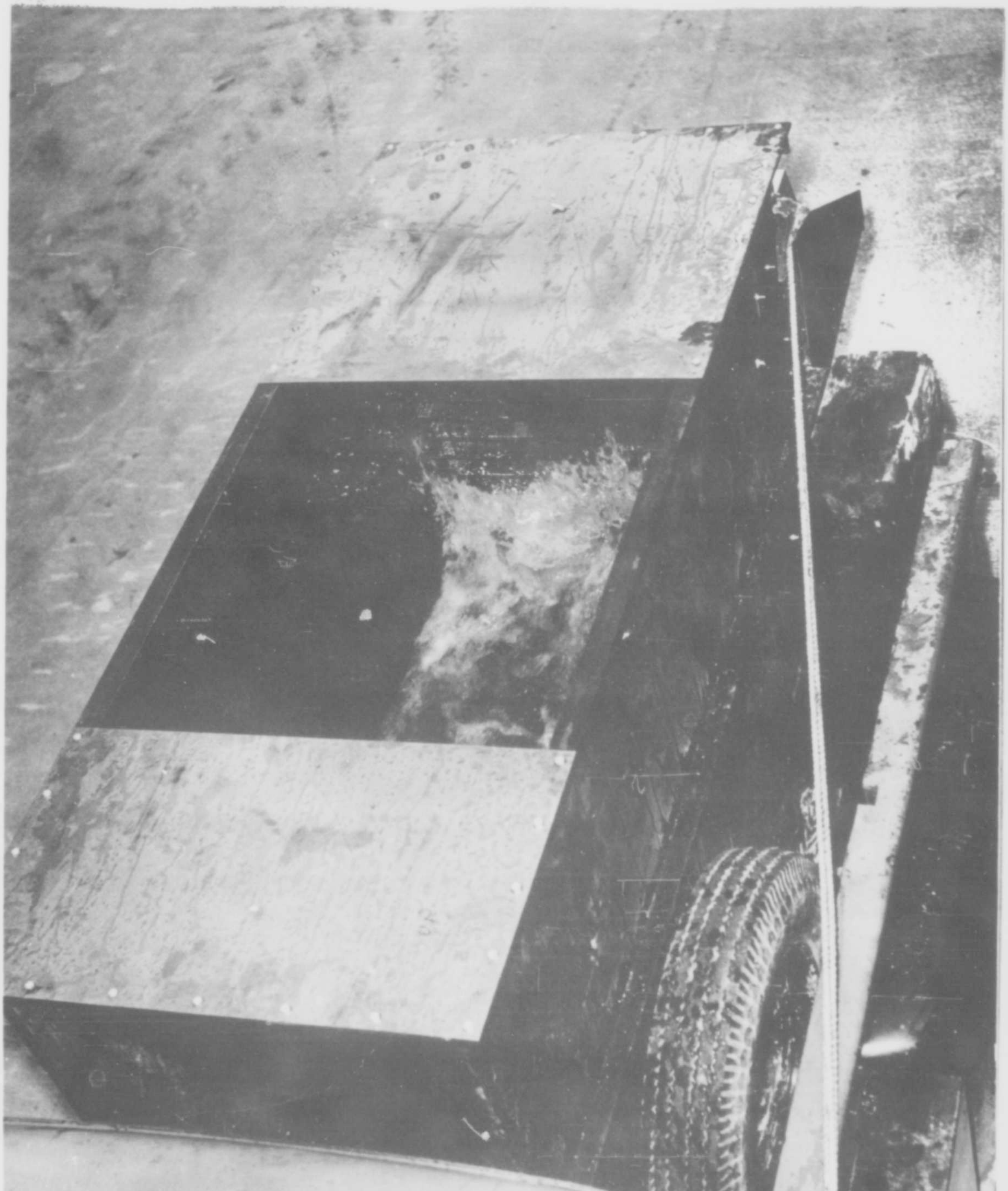
BILL OF MATERIAL			
ITEM NO.	NO. REQ'D.	SIZE	MATERIAL
1	2	[illegible]	MILD STEEL
2	1	[illegible]	" "
3	1	[illegible]	" "
4	1	[illegible]	" "
5	4	[illegible]	" "
6	2	[illegible]	" "
7	1	[illegible] ANGLE	" "
8	1	[illegible]	" "
9	1	[illegible]	" "
10	1	[illegible] PIPE COUPLING	
11	1	[illegible] " " PLUG	
12	24	[illegible]	[illegible] SCREW
13	38	[illegible]	[illegible] NUT
14	2	[illegible]	[illegible] SCREW
15	2	[illegible]	[illegible] NUT
16	4	[illegible]	[illegible] SCREW
17	1 [illegible]		ALUMILASTIC
	1	ASSEMBLY - A	
	1	ASSEMBLY - B	
	1	ASSEMBLY - C	



ITEM 4

NOTE:
FILL BOLTED JOINTS WITH ALUMILASTIC
TO MAKE WATER TITE TANK.

MOCK UP, FLOAT VALVE



TEST UNIT MOUNTED ON PICKUP TRUCK

FIGURE 22

FIGURE 23
WAVE IMPINGING ON LOUVERS

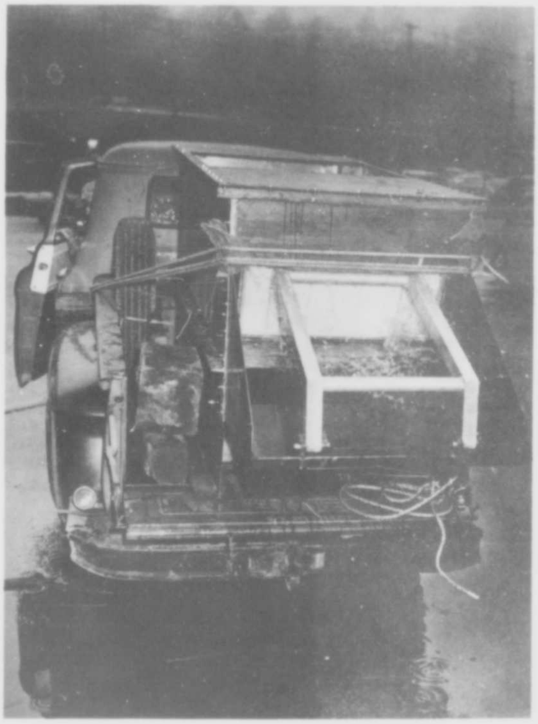


FIGURE 24
OUTSIDE VIEW WITH FLOAT OPERATING

FIGURE 25
OUTSIDE VIEW WITHOUT FLOATS

when the float was blocked and exemplifies the effectiveness of the float system.

It is realized that additional small quantities of water will be carried into the vehicle through the air inlet by the air itself in the form of spray. If the vehicle is expected to remain in the water for extended periods without the engine running the bilge pumps will then be inoperative. If consideration of this condition is necessary, arrangements can be easily made for manually locking the floats in position over the air openings to completely seal them off. But, this introduces the possibility of damaging the engine if it is run with the openings closed. It is suggested that this manual override be considered only if found necessary by actual vehicle test.

The air outlet duct, over the engine, is attached to the engine fan shrouds and to the stationary outlet leading to the sides of the vehicle. The duct is fabricated of sheet aluminum and is removable after the top cover over the engine and power train is removed.

Aspiration Air System

Two air cleaners are mounted on each side of the outlet duct. Aspiration air is normally drawn through the air cleaner from the machinery compartment. However, during water operation when waves cause the intake float to close, the engine cooling fans will draw most of the air out of the compartment, and could starve the engine of combustion air. To prevent this occurrence, a plenum attached to the outlet of each air cleaner incorporates a flapper valve which is actuated through a mechanical linkage by the intake float. When the float closes, the position change of the valve closes off the outlet from the air cleaner and opens a passage directly into the crew compartment. Aspiration air, therefore, is drawn from the crew compartment during the momentary periods that the vehicle is inundated. Since the availability of air for combustion is affected only by the intake float, the air outlet floats are completely independent of this mechanical linkage.

ELECTRICAL SYSTEM

The electrical system is conventional and simple because no major components are electrically operated. All electrical components and connectors are waterproof with the exception of the engine-mounted 100 ampere alternator. Batteries are located in the rear of the left sponson for better weight distribution and the convenience of applying heat with the ventilating heaters during cold climate operation.

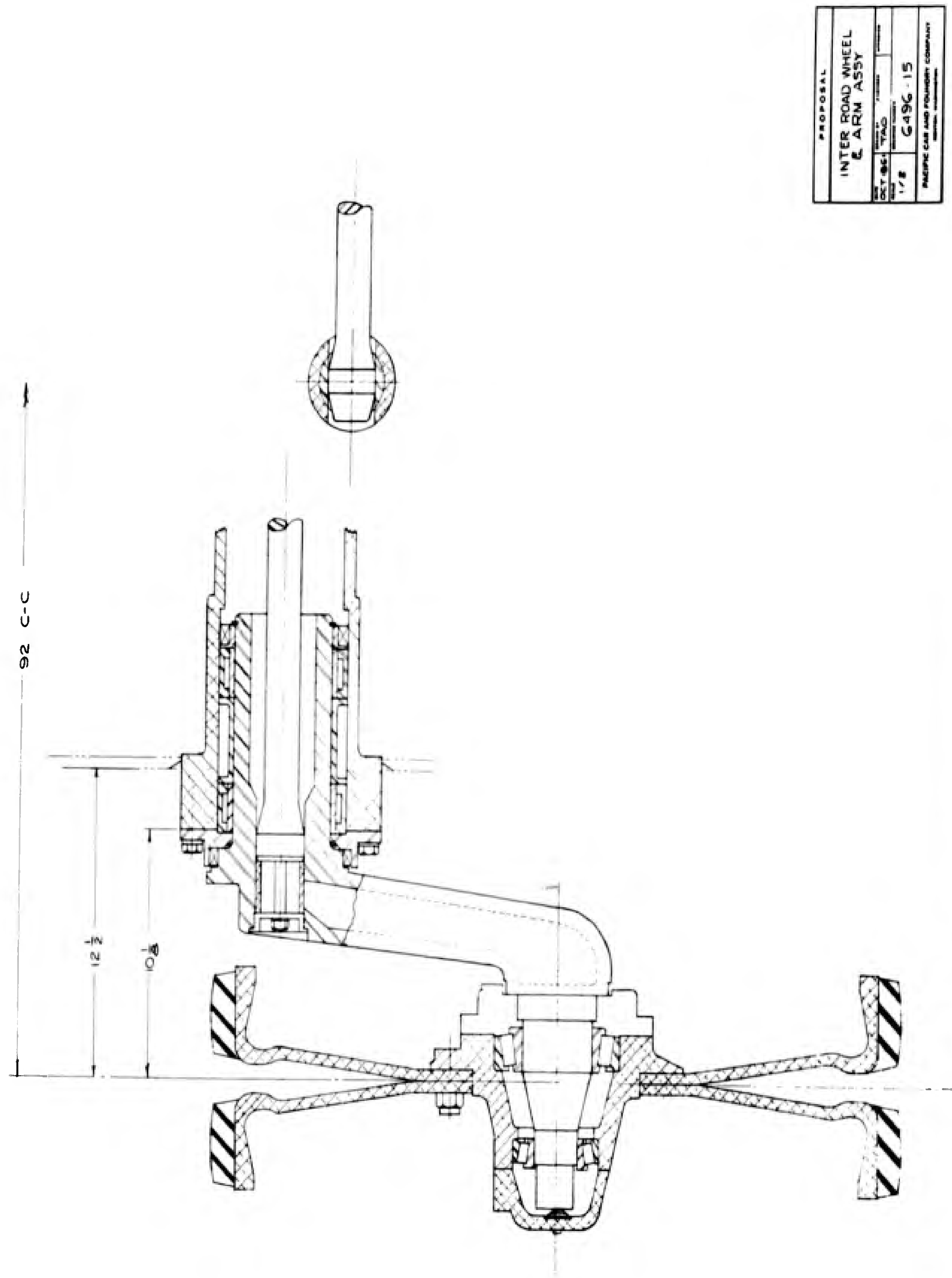
SUSPENSION ASSEMBLY

The suspension assembly comprises tracks, front drive sprockets, ten road wheel assemblies, and ten torsion bars.

A typical road wheel assembly as shown on drawing 6496-15 consists of two road wheel discs, a hub, an arm, a torsion bar, bearings, and fasteners. Drawing 6496-8 shows the housing, welded into the hull, that contains the road arm parts. The 32 inch diameter aluminum road wheel discs are mounted on the hub. These road wheels would be used with either the band or single-pin block tracks. The hub contains two tapered roller bearings and a face type seal secured to the arm. The arm is mounted and retained in the hull on two needle bearings. Lip seals retain the needle bearing lubrication and exclude dirt and water. A torsion bar is installed through the road wheel arm and anchored at the far end to the hull. An internal spline in the arm winds up the torsion spring. An external spline on the arm is keyed into another arm that actuates the lockout cylinder. Similar components are found on existing tracked vehicles.

Trailing Idler

The trailing idler road wheels are placed opposite each other so that the same length track can be used on either side of the vehicle. One road wheel arm is longer to allow this. Drawing 6496-17 shows the arrangement. Leading road arms are used to accommodate track tension forces in a simple fashion. The road wheel arm spindle is offset inboard to allow clearance for the track. The track tightening mechanism is located on the trailing idler arm road wheel spindle. An eccentric allows an adjustment of 3-3/4 inches. The eccentric is locked by the clamping action of two large bolts.



PROPOSAL
INTER ROAD WHEEL
& ARM ASSY
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6496-15
PACIFIC CAR AND FOUNDRY COMPANY
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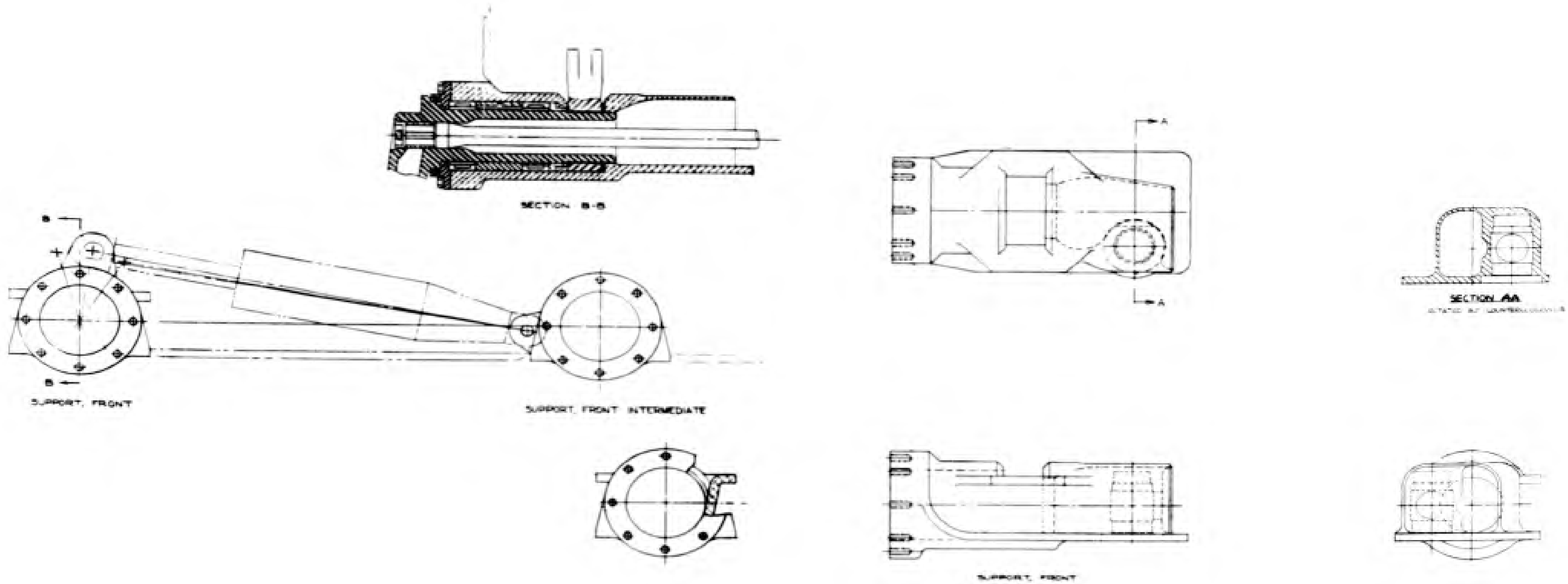

PROPOSAL

SUPPORT

ROAD ARM FRONT

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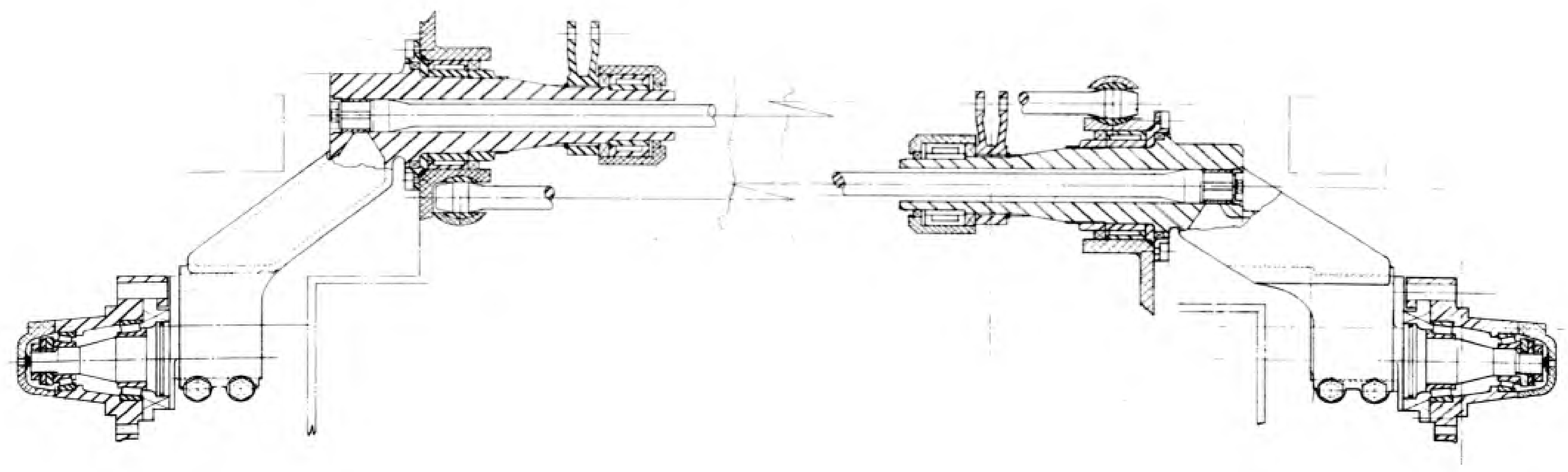

PROPOSAL

TRAILING IDLER
ASSY

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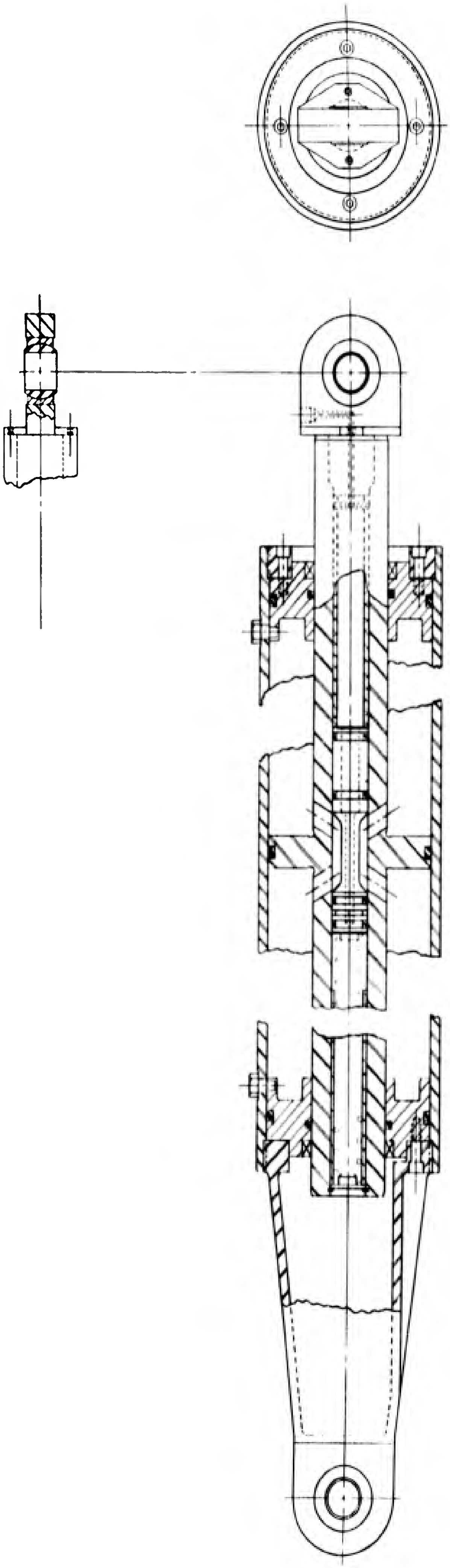
PACIFIC CAR AND FOUNDRY COMPANY

6496-17



6496-16
FIGURE 1

PROPOSAL

LOCKOUT CYLINDER

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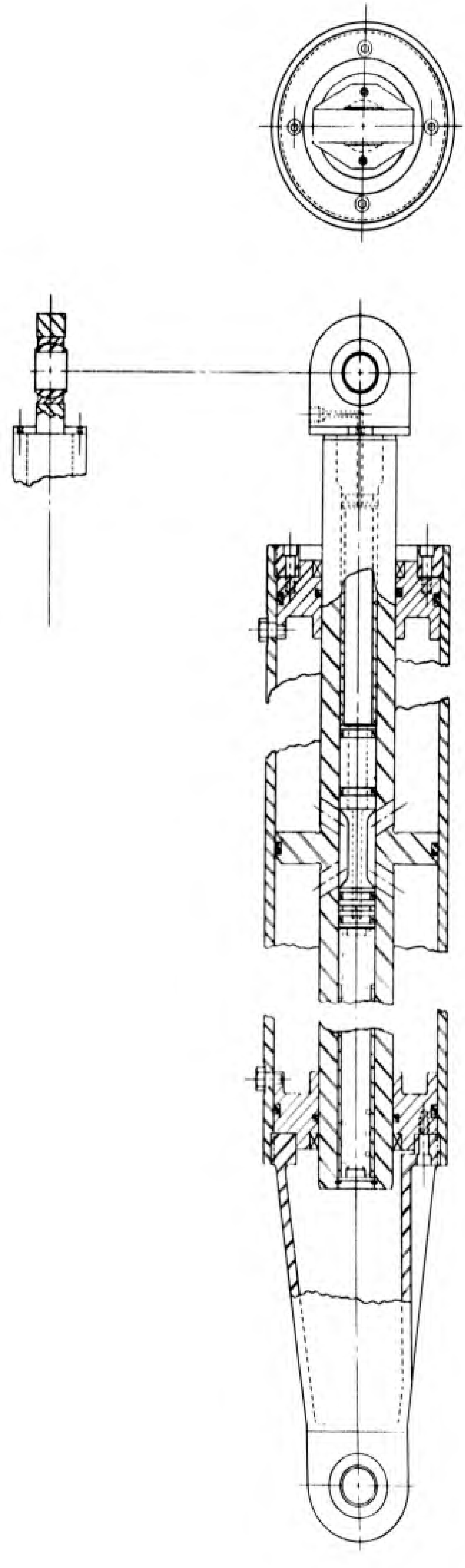
PACIFIC CAR AND FOUNDRY COMPANY
RENTON, WASHINGTON



6496 - 16
FIGURE 1

PROPOSAL

LOCKOUT CYLINDER

OCT 18,61	TAO
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PACIFIC CAR AND FOUNDRY COMPANY

Lockout Cylinders

Provisions have been made for the use of a lockout cylinder type of shock absorber. The lockout cylinder, pictured in drawing 6496-16, is an effective shock absorber that includes a hydraulic cushion at each end of its stroke to function as a bump stop. The long piston in the center of the cylinder can be displaced by low pressure oil to block off the flow of aoil from one side of the main piston to the other to prevent movement of the road arms. This very desirable feature would be used for the Artillery Vehicle, Recovery, and other special purpose adaptions of the basic vehicle where firm and stable platform is desired. In the personnel carrier, however, it will function as a shock absorber only. When actuated by road arm movement, oil is forced through an orifice from one end of the cylinder to the other through an orifice which restricts the flow sufficiently to dampen vehicle pitch.

This type of suspension, with the track returning over large road wheels, has a rolling resistance of 60 pounds per ton at 13 MPH on the M110 Self-Propelled 8-Inch Howitzer.

Standard Ordnance torsion bars are used throughout except in the trailing idler. Vertical spring rate is 960 pounds per inch and a force of 10,500 pounds is required to bottom out each road wheel.

TRACKS

To maintain a vehicle weight within the 35,000 pound limit, particular care was given to limiting the weight of the track. It was determined that in no instance should the track weight exceed 4,000 pounds. To do this, special consideration was given to light, high strength materials and to developing a configuration that would impose the lowest stress for a given load.

During the course of this investigation, the preliminary designs of three different types of tracks were developed. They are:

1. A band track comprising bands made of of cable tension members bonded into a rubber belt to which suitable track shoes are riveted.
2. A single-pin rubber-bushed block type track with certain unconventional features necessitated by the forged aluminum track blocks.
3. A track called the "Speed Trac", which has cable tension members but utilizes inter-locking track bars of an unconventional configuration.

All of the above listed concepts have advantages and limitations; no perfect track has, or will be, built. Each has its indeterminate areas which cannot be justified without further investigation and testing. Although the "Speed Trac" was developed by an employee of Pacific Car and Foundry Company, patented rights for development and manufacture have been turned over to the Caterpillar Tractor Company. Clearance for this military development must be obtained.

For these reasons, definite recommendations for a specific track cannot be made at this time. It is our recommendation that more study be given to the track problem before a decision is reached. Each of the three concepts is explained as follows.

<u>Single Pin, Rubber-Bushed Track</u>

A single pin, rubber-bushed track, drawing 6496-27, was designed to conform to conventional practice on moderate sized vehicles. Preliminary investigations, however, showed that the amount of material and consequent weight is determined by the minimum section that can be forged and not by strength requirements. Therefore, a forged aluminum body was used to minimize the weight of the shoe. The strength to weight ratio and fatigue limits of aluminum are sufficient to ensure adequate shoe strength at considerably less weight than the lightest steel configuration. To account for the lack of hardness and abrasion resistance of aluminum, the shoe contains an imbedded steel member that forms the guide for the wheels and a drive lug for an internal sprocket. In this manner, the shoe is retained and driven by a hardened steel member that has the necessary wearing qualities.

A smooth rolling surface for the wheel has been provided by filling in the depression in the top of the shoe with rubber. The underside, which contacts the ground, has a molded rubber chevron grouser. By this proper utilization of rubber, none of the aluminum is subject to wear or abrasion.

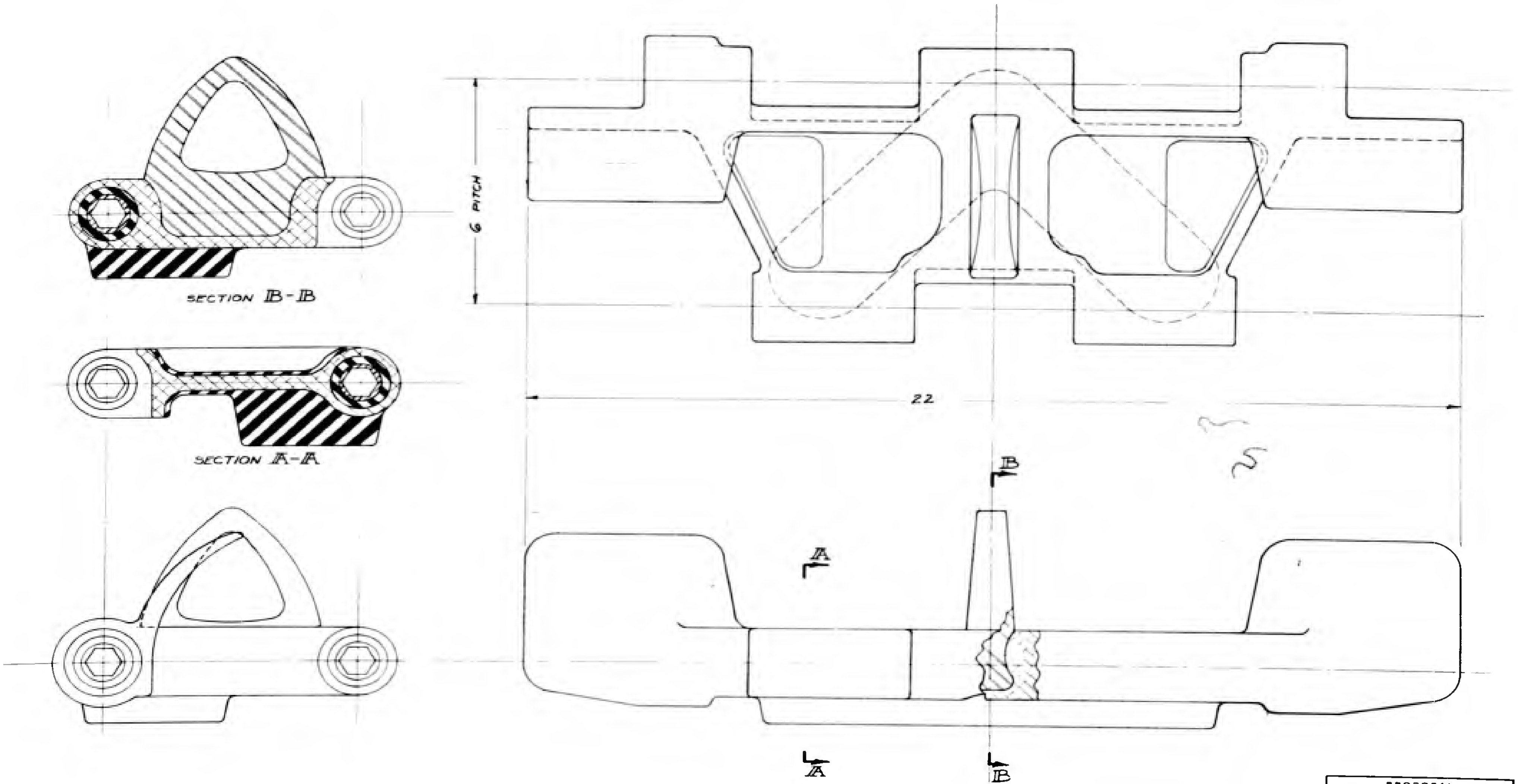
The hinged joint is a conventional rubber-bushed single-pin type. That is, steel bushings with hexagonal holes and rubber molded to the outside diameter are pressed into the machined hole in each hinge lug. A pin pushed through the hexagonal hole of adjoining shoes makes the final connection. Hinge action is taken torsionally in the rubber.

Paddles for propulsion in the water are extended from the end of the shoe.

The major area of question in this track is with the steel drive-guide lug. Consultants at Kaiser Aluminum and Chemical Sales, Inc. believe that the lug can be successfully imbedded in the aluminum forging. However, the magnitude of the impact loads that the lug could be subjected to are indeterminate and consequently adequacy of the lug retention over the life span of the track cannot be assured without further testing.

Driving the track from a point so remote from the center of pivot or hinge point may also cause engagement problems with the sprocket. Simple analysis of the track engaging a sprocket shows that the amount of sliding during engagement is proportional to the distance away from the hinge point.

Army Ordnance has sponsored a similar track development on a vehicle weighing 14,000 pounds. This track has the same general configuration, however, the body of the shoe was cast of aluminum rather than forged as proposed. The vehicle with this track has operated only about 400 miles to date. During that time, the lugs have shown no signs of loosening. Wear on the sprocket and lugs appears to be excessive and reasons for the wear have not yet been determined. However, it is assumed that it is aggravated by the large amount of sliding occurring on engagement. Whether or not a modified tooth form will overcome this problem requires further study.



PROPOSAL

LVTPX11

RUBBER BUSHED SINGLE PIN BLOCK TRACK

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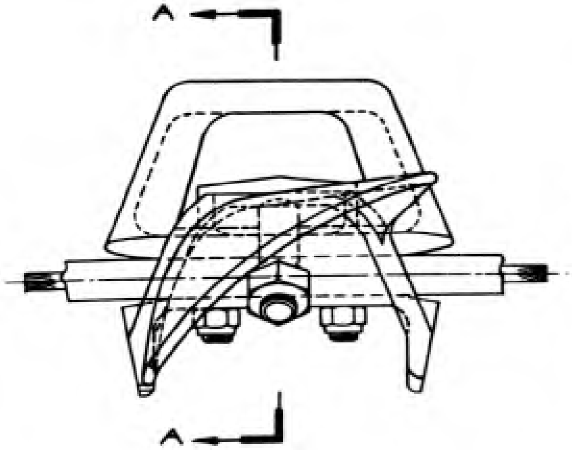
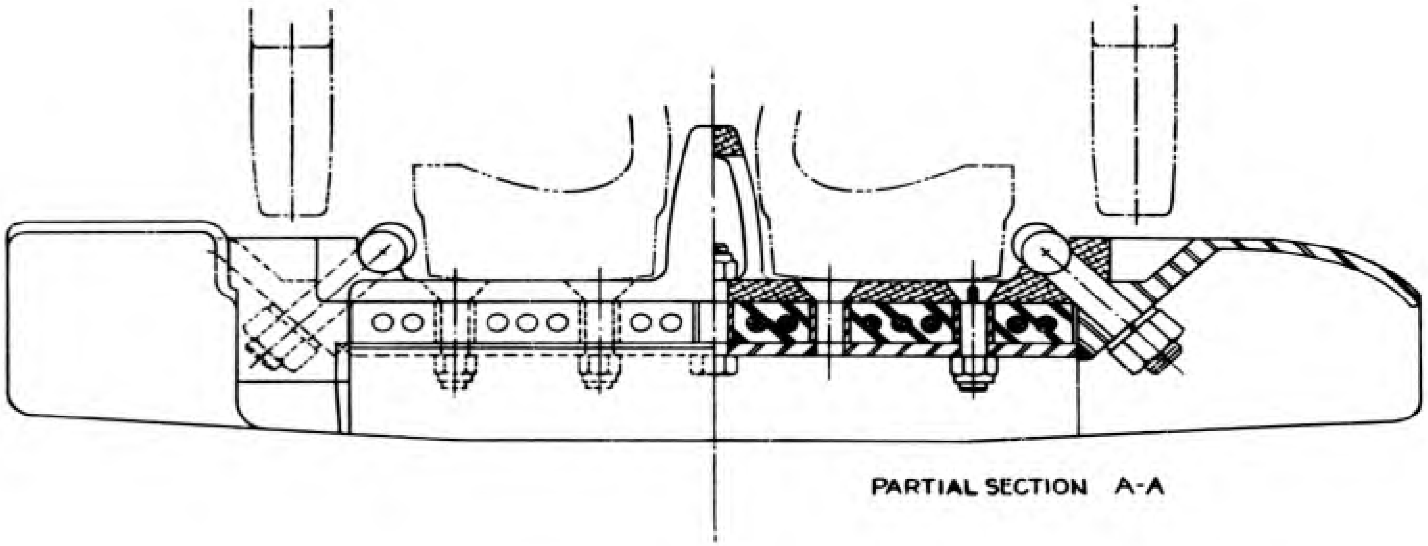
SCALE	DRAWING NUMBER
FULL	6496-27

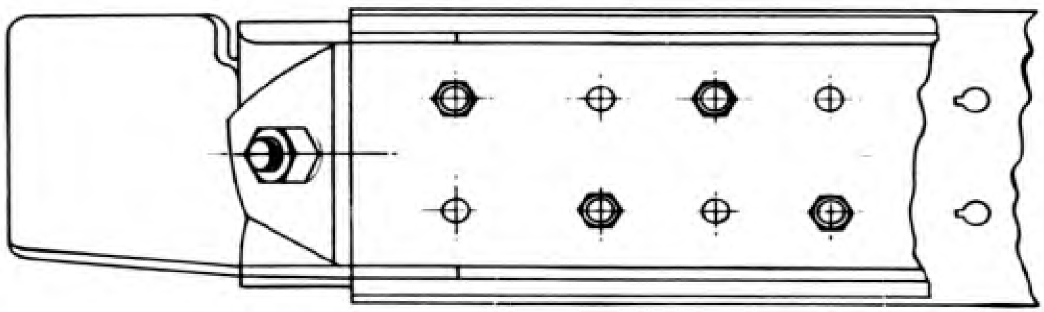
PACIFIC CAR AND FOUNDRY COMPANY

RENTON, WASHINGTON





PARTIAL SECTION A-A



SPECIFICATIONS

PITCH $6\frac{5}{8}$ IN
WIDTH 22 IN
WEIGHT 45.34 LBS PER FT.
BAND SIZE 11.25 X .714 IN
BAND CABLES NO 14
CABLE SIZE $\frac{5}{8}$ DIA 7X19
BAND STRENGTH 140,000 LBS

6496-36

PROPOSAL			
BAND TRACK 22 INCH LVTP-X11			
NOV 1 1961	JCW		
FULL	6496-36		
PACIFIC CAR AND FOUNDRY COMPANY			

The proposed track has a feature that is not inherent in the other two concepts. The use of rubber bushings at the hinge points allows the track, when under load, to stretch by displacement of the rubber in the bushings. The advantage of this feature is that the track can yield, when obstructions pass through the track and suspension, and reduce overloads that could result in damage had there been no yielding. The disadvantage is that the track is less rigid longitudinally, laterally, and torsionally, which can aggravate track throwing. This problem can be overcome by reasonable design considerations.

Characteristics of the single-pin rubber-bushed track are:

Over-all width	22 inches
Pitch	6 inches
Weight Per Pitch	21 pounds
Weight Per Foot	42 pounds
Weight Per Vehicle	3300 pounds

<u>Band Track</u>

The sectionalized open-band track, drawing 6496-26, was designed in accordance with the parameters set down for Ordnance vehicles by the Detroit Arsenal. This reference is the MANUAL ON DESIGN OF TRACK BAND ASSEMBLIES, prepared under Contract No. DA-20-018-ORD-14511, Detroit Ordnance District, by the General Motors Engineering Staff. In preparing this manual, a number of tracks were made and tested on a dynamometer vehicle of the 18 to 25 ton class.

The principle advantage cited in this reference for a band track as tested on an 18 ton M-59 amphibious personnel carrier is a reduction of rolling resistance from 83 to 70 pounds per ton, in a comparison with a block and pin track. The reduction, moreover, appears in the speed range from 20 to 30 MPH. While the cable reinforced belt itself provides the essentials of a track, namely, flexibility with high longitudinal tensile strength, its inherent transverse flexibility introduces problems in retention. Track throwing occurs when the track is deflected by steering side forces especially during side slope operation. For example, consider the outside track on a vehicle turning on a gravel-surfaced hill. The forward portion of the track on the ground is slack because of track stretch under the high driving forces caused by the grade, and because of changes of geometric circumference of the suspension, deflected by the assymetric loading of the tilted vehicle. On turning, the ends of the track collect gravel over the track between the road wheel and its normal path on the track, lifing the front wheel above the track guides. At the same time, the track is deflected edgewise by the turning force so that it misaligns with the rotating plane of the wheel. The wheel rolls through the gap between track guides, at an angle to the track, meeting the next guide on its rolling surface instead of its guiding flange, and rolls over the top of the guide. This process is repeated with the following wheels and results in loss of the track. If the ground is uneven, torsional flexing of the track can augment this process. This example illustrates most of the elements that contribute to track throwing.

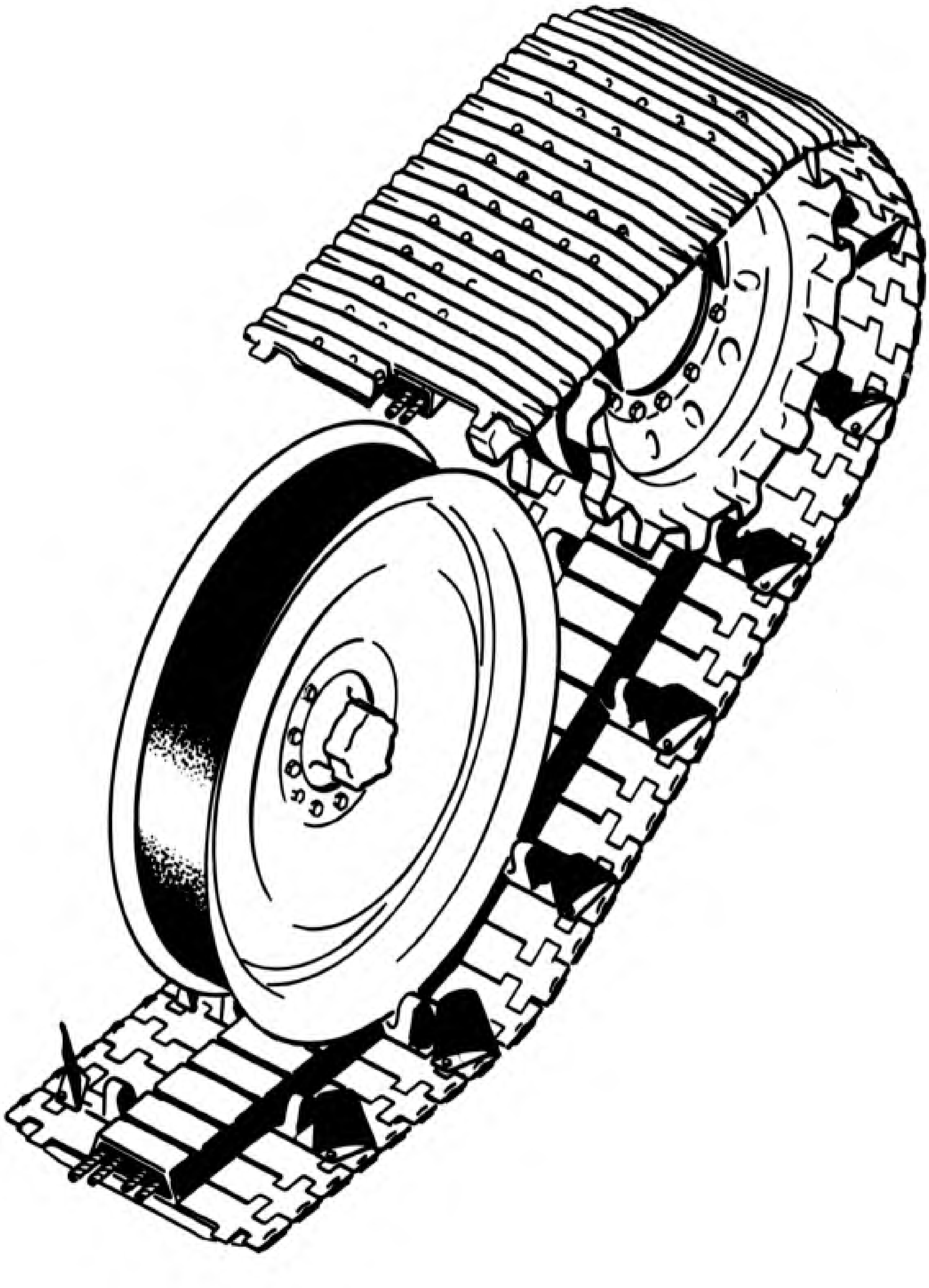
Detailed attention has been given the band track to minimize its track-throwing potential. First, the multiple cable construction is very rigid in tension which limits the slack developed at high track tensions. Second, edgewise deflection is minimized. The mechanism of track-throwing functions with an edgewise curvature of the track. This curvature, in turn, depends inversely upon the edgewise bending and sheer stiffness. Great stiffness in the proposed track is obtained by means of a single wide belt with multiple closely spaced cables, the width contributing edgewise moment of inertia, high ratio of steel to rubber minimizing sheer deflection. Third, the disadvantage of high torsional flexibility of the band track is minimized by detail attention to the track guides and their relation to the road wheels. The track guide is as long as possible at the top to reduce the gap between guides. It is flat-topped to eliminate damage to the rubber of the wheel, should it ride on the guides momentarily. All track laying vehicles experience this partial throwing more frequently than is suspected by the driver, as evidenced by damage to the tires by the sharp-topped guides of most tracks. Dual wheels and center guiding has proved most effective in Ordnance vehicles and it is considered essential for this torsionally flexible band track as the arrangement is effective through the widest range of track tilt. However, the guiding surfaces must be contoured to avoid binding between the wheels. This binding can force the wheels to climb incorrectly designed guides of greater height.

The track engages dual sprockets straddling the track band to provide initial alignment with the first road wheel. The debris piled on the track during its sidewise motion in a turn is important if it is of a dense nature that resists penetration or packs under the road wheel. Narrow tires assist penetration and it has been found that even modest amounts of rise designed into the ends of the track bars or grousers cause them to "ski" over firm soils when skidding in the turn. Such a ramp is a structural member of the proposed track bar.

The above features, together with the inherently stable characteristics of the short Christie suspension will provide adequate track retention. To provide adequate life, the belt has a tensile strength of four (4) times the vehicle weight. To ensure the longest fatigue life of its cables, it is desirable that a minimum number of flexings of the longest radius be imposed on them. To this end, the track bars are broad-based relative to the pitch, and the track plates are curved on both wheel and belt sides, so that the shoe does not tilt as the tire rolls to the next plate, and any bending of the belt is limited as to sharpness of bend.

The next strength consideration is that of transverse bending of the track bar. To sustain the bridging loads when the bar spans two obstructions at its ends, or the radial component of band tension when supported and driven by the sprocket, the track bar and track plate sandwiching the band are mechanically joined to co-act as a beam of such dimensions that their respective materials are stressed in proportion to their strengths. The sprocket engaging teeth are an integral part of the steel castings which incorporate the track tip foils and are welded to the central, rolled channel, track bar. If increased grouser life is desired, rubber blocks can be bolted within the channel, however, a penalty in weight and aggressiveness will be incurred.

For water propulsion, and for increased track area in soils of high sinkage, the track bar tips are formed as a cascade of low-aspect-ratio foils curving into reaction paddles. These are a development of the best similar type tested at the David Taylor Model Basin. Their effectiveness is due to their action in increasing the mass flow through the propelling elements from a region of high wake, in turn, caused by the drag of suspension elements and the returning top track. The action of a low aspect ratio foil at high angle of attack is one of high loss by



PROPOSAL

"SPEED TRAC"
PAT. PEND.

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— | 6496-24

PACIFIC CAR AND FOUNDRY COMPANY

circulation about the tip. The circulation induces the flow into the following foil. In this case, the loss, or drag, is the propulsion reaction. The slight sweep forward is used to bias the flow toward the track centerline to utilize some track bar area.

Characteristics of the band track are:

Over-all Width	22 inches
Pitch	6-5/8 inches
Weight Per Pitch	25 pounds
Weight Per Foot	45.34 pounds
Weight Per Vehicle	3540 pounds

Speed-Trac

Speed-Trac is a name given to a patented track construction currently being developed by an engineer at Pacific Car and Foundry Company, Mr. T. Fikse, and which has been turned over to the Caterpillar Tractor Company for adaption to crawler tractors. The structure, shown on drawing 6496-24 opposite, consists of cable tension members surrounded by rubber. The rubber is, in turn, almost entirely surrounded by the track segment structure. Adjacent track bars have registering means that prevent sheer motion between them. The track segment caps are drawn down with bolts to the track bar and in so doing compress the rubber around the cables. The rubber is not vulcanized to the cables. The segments are spaced closely enough to effectively pressurize the rubber. This is a distinct difference between the Speed-Trac and the open-band type track.

The registry tabs provide the Speed-Trac with a torsional stiffness and shear strength equivalent to most link and pin tracks.

The Speed-Trac design shown on drawing 6496-23 enables a steel surface to be presented to the sprocket and road wheels. A lighter track structure could be achieved if the bars were constructed of aluminum.

The most practical deterrant to the electrolytic corrosion problem of this steel and aluminum construction seems to be to use cadmium plated bolts and caps. The use of cadmium reduces the potential difference between metals to a minimum. Schemes that utilize an insulating material between the steel and aluminum appear impractical because impact forces would damage the insulators.

Experience with Speed-Trac to date has suggested that aluminum grouser edge wear would be acceptable because the short pitch of the Speed-Trac design reduces grouser scuffing by a factor of 4 and hence grouser wear by a like amount. The end of the track bars would receive a water propulsion paddle made of the same material as the track bars, and would be replaceable. The replaceable feature of the paddles would allow the achievement of an optimum design more easily since the basic track structure would not have to be altered for paddle shape experiments.

The cables currently used in Speed-Trac are of the endless type. However, for use on the LVTPX11 we propose a cable coupler that would be an outgrowth of the structure shown in Figure 26. The track could then be made in convenient sections and attached for final assembly. Cables required for the Speed-Trac are (4) 5/8" 6 x 37 IWRG. The strands of this cable would turn out radially in the coupler and be swaged to a coupler block in this radial zone. Preliminary successful tests of the coupler have been made. This method would provide a satisfactorily short (2 inches) coupling.

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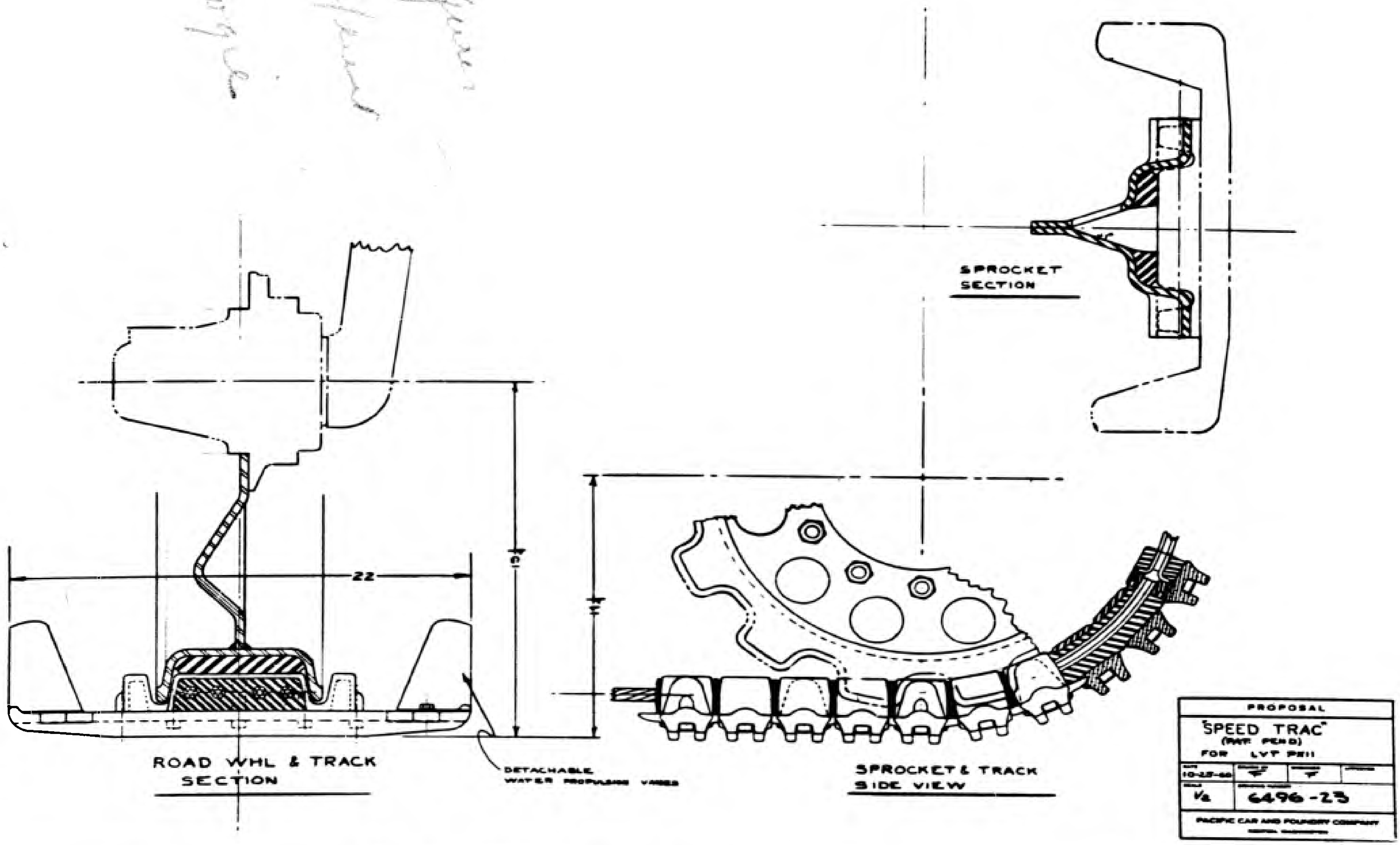

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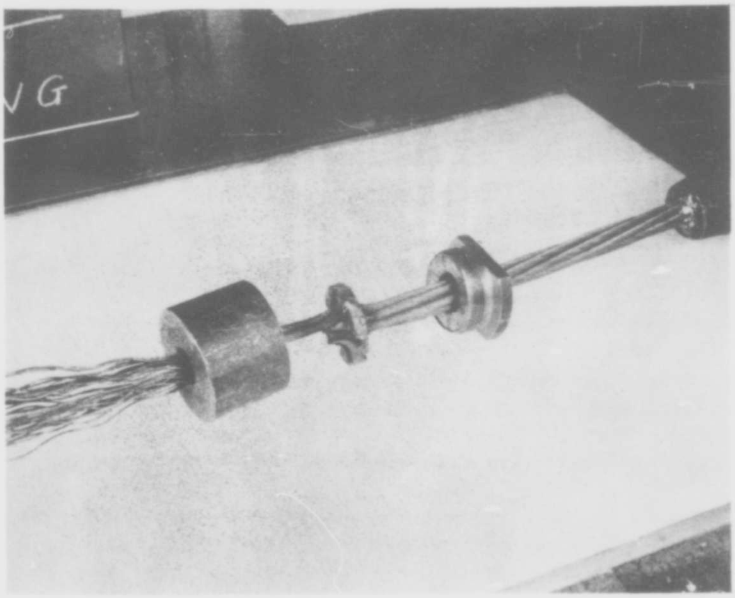

TENSION TEST OF SPEED-TRAC COUPLER SHOWING TWO STRANDS THAT WITHSTOOD 90% PULL

FIGURE 26

DRIVE SPROCKET ASSEMBLY

The drive sprocket assemblies, drawing 6496-12, through which power is transmitted to the tracks, are located in the front of the vehicle. The XTG-250 power train is mounted from the sprocket housing directly inboard of the sprocket assemblies. Since the power train contains the final gear reduction, no additional final drive or drop gear boxes are required. Power is transmitted directly to the sprockets.

Double sprockets drive the track and are attached to a shaft which runs on a single self-aligning roller bearing. An aluminum housing mounted into the hull supports the sprockets, shaft, and bearing. The housing is not cantilevered from the outside plate but is supported by the outside plate and an integral hull plate approximately 5 inches inboard. This type of support eliminates the tensile load on the bolts attaching the housing to the hull and distributes loads imposed by the sprocket into the hull over a considerably larger area. The inner portion of the housing includes a cradle and cap into which the power train is mounted and retained. This arrangement affords a direct and positive means of mounting the power train and provides accurate alignment of the power train and sprocket.

This arrangement makes possible the use of a single bearing supporting the outer end of the shaft since it permits the inner end to be radially restrained by the spline in the power train output gear.

A spherical bearing was considered necessary because of deflection that will occur in the shaft under high track tension and impact loads. However, bearing life will be more than adequate since its size was governed by the shaft size required to resist these loads.

A face type seal that has proven successful in this type of application will be used for the outer seal; a synthetic rubber lip type will be used for the inner seal. The assembly is lubricated with grease through a fitting located near the bolts which hold the housing to the hull.

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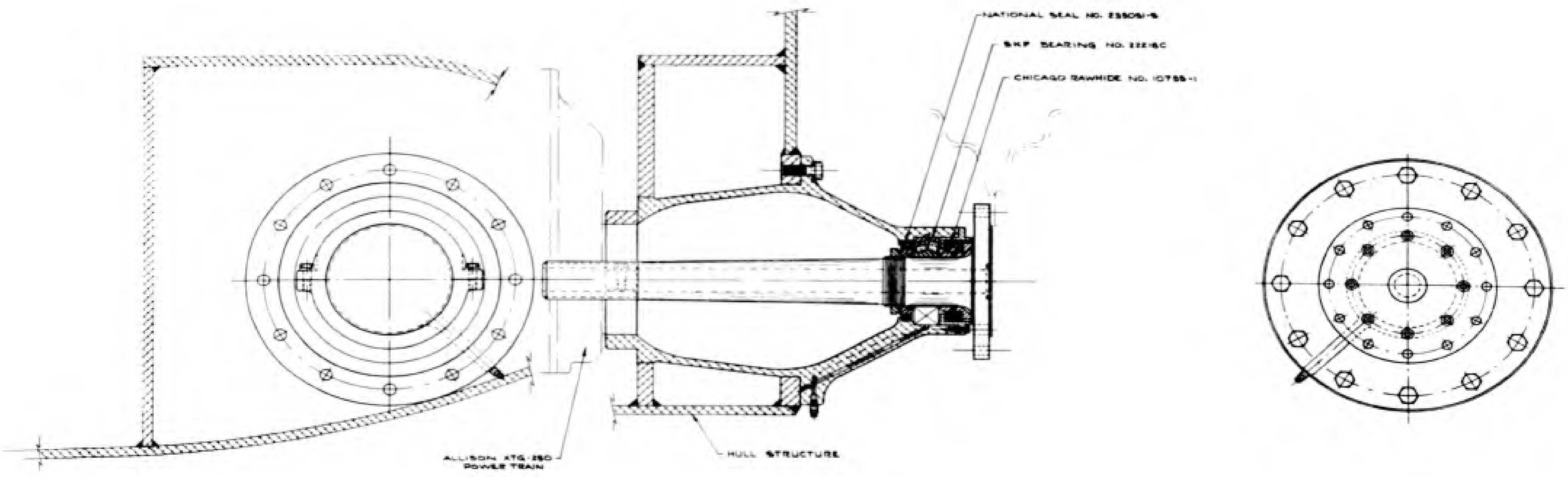

PROPOSAL

DRIVE SPROCKET ASSY.
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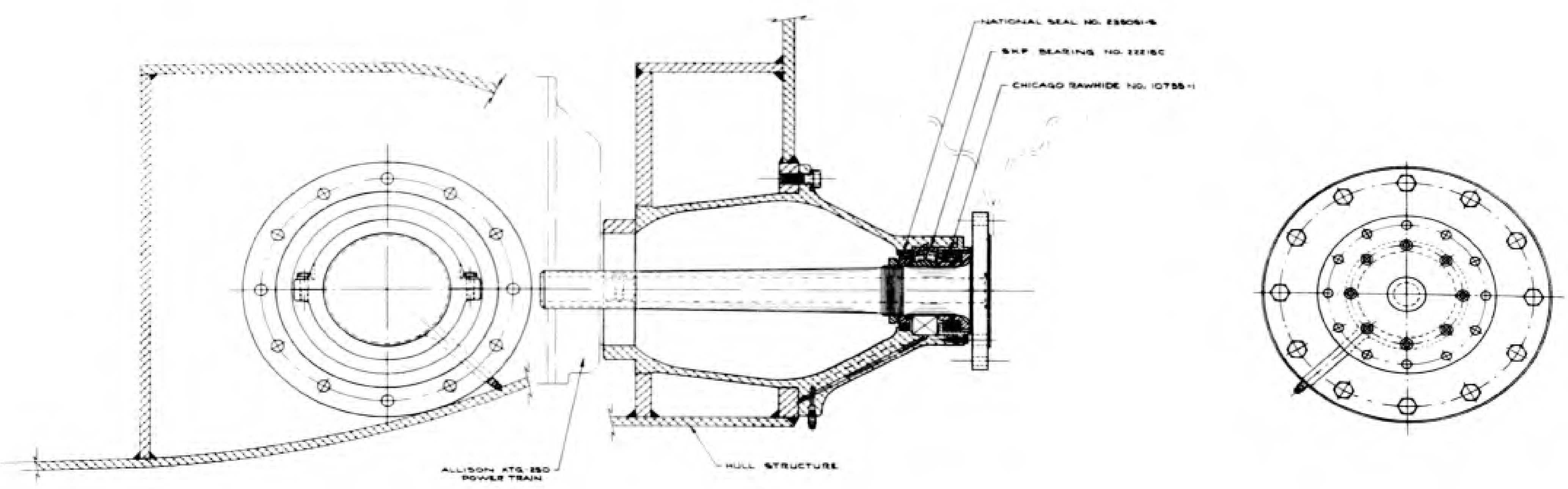
PACIFIC CAR AND FOUNDRY COMPANY
RENTON, WASHINGTON



PROPOSAL

DRIVE SPROCKET ASS'Y.
LVT PX-11

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PACIFIC CAR AND FOUNDRY COMPANY

VENTILATION AND WINTERIZATION

Ventilation air for the crew compartment is drawn in and exhausted through two "mushroom" type vents located on the rear corners of the hull, drawing 6496-30. Air is drawn in through the right vent by an electric axial flow fan with a capacity of 1000 cfm. The fan blows the air through a duct running the length of the cargo compartment; even distribution of the air is accomplished by louvers. Air is exhausted through the vent on the left corner.

The mushroom vents are extended as far above the deck as possible without increasing over-all vehicle height. Each mushroom vent includes a buoyant float that closes the vent to limit entrance of water during rough sea operation. The floats, constructed of resin impregnated fiberglass over a unicellular plastic, fit very loosely to prevent binding. To take care of incidental leakage, a duct within the hull is arranged with baffles to direct water into the bilge where it is discharged overboard by the bilge pumps.

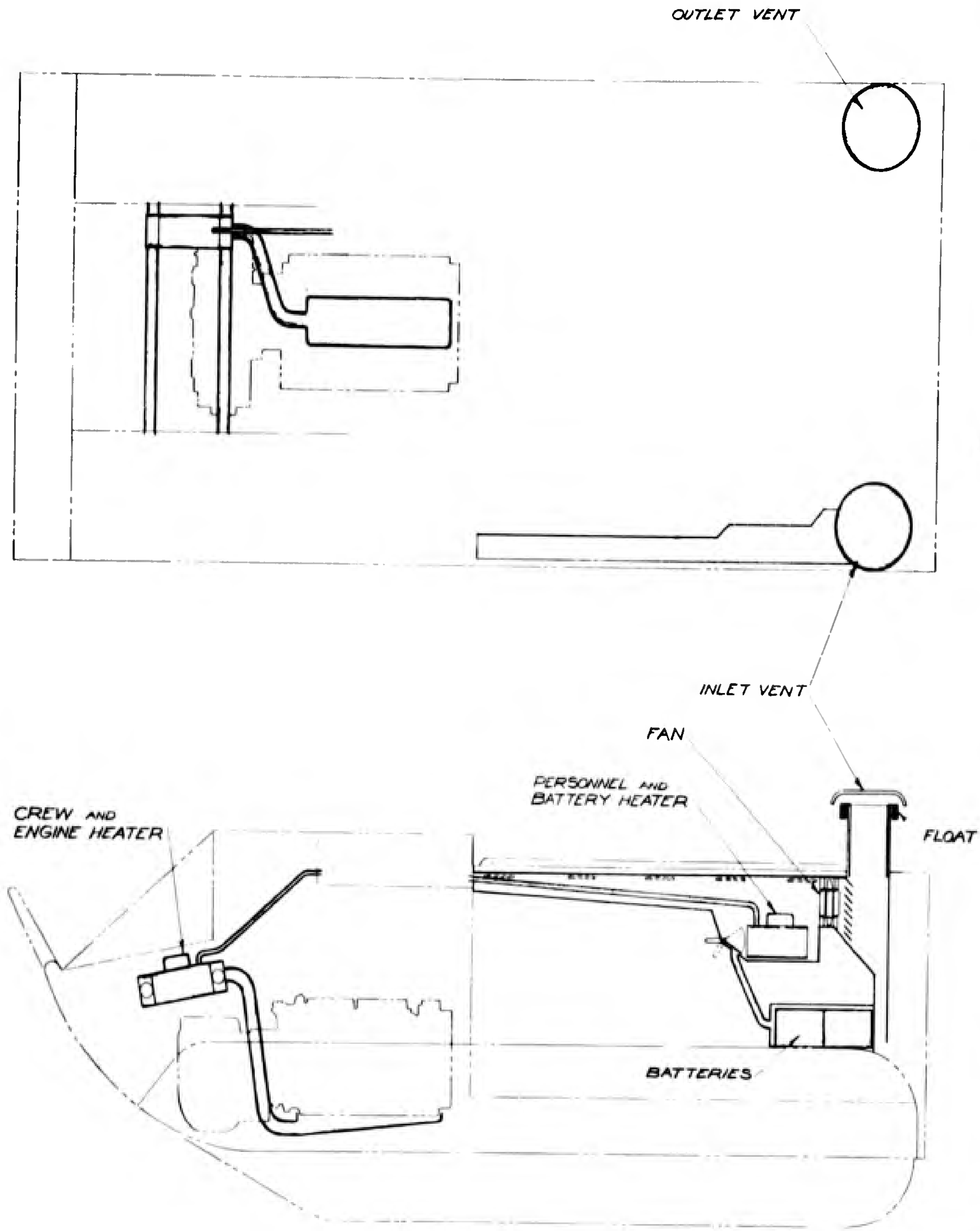
A 60,000 BTU heater that burns engine fuel is located in a protrusion in the duct so that incoming air can be heated for personnel comfort. Heater controls will be arranged to run the vent fan at a moderate air flow rate to ensure that heated air will be exhausted into the crew compartment and not back out the inlet vent. The heater exhaust ducts into the engine cooling air exhaust.

An additional 60,000 BTU heater is located inside the machinery compartment bulkhead next to the gunner to heat the driver's and gunner's area. Air is drawn from the driver's and gunner's area, heated, and returned. Combustion gases from this heater are also exhausted into the engine cooling air exhaust duct. The distance between the inlet vent and combustion exhaust should eliminate any possibility of carbon monoxide contamination of ventilating air.

The heaters probably will not be capable of sustaining combustion during rough sea operation due to air pressure variation caused when the various air inlets and outlets close.

These heaters will also be used for the winterization kit. The battery box, located in the rear of the left sponson directly below the heater has a duct running to it from the heater. A butterfly valve will divert an adequate amount of heated air into the battery box. A duct from the front heater will disperse heated air under the engine.

During water operation, the vent fan should operate at full capacity since engine aspiration air, at times when the cooling air inlet is closed, is drawn from the crew compartment. The vent fan may be operated at variable speeds for land operation.



PROPOSAL			
LVTPXII VENTILATING AND WINTERIZATION SYSTEM			
DATE 11-5-61	DRAWN BY	ENGINEER	APPROVED
SCALE SCHEMATIC	DRAWING NUMBER 6496-30		

PACIFIC CAR AND FOUNDRY COMPANY
RENTON, WASHINGTON

ts 750 miles at 75 mph

FUEL SYSTEM

Wrong

To meet the requirements of ten hours full throttle operation without refueling, 250 gallons of fuel are stored in a flexible light weight plastic fuel tank under the cargo deck.

Fuel is stored in a single cell, drawing 6496-20, that is constructed of plies of nylon or dacron bonded together with a hydro-carbon fuel-resistance synthetic elastomer - in this case a polyurethane plastic. Wall thickness is less than 1/16 inch. Strength and abrasion resistance of this construction is sufficient to require only intermittent support of the cell at convenient support points. It may, therefore, bulge into some depressions and span others. Since the cell is flexible, it is supported by the hull and suspended from hangers bonded to the top periphery of the cell. Metal rings and fittings for attaching the filler pipe and tubing are bonded integrally into the cell.

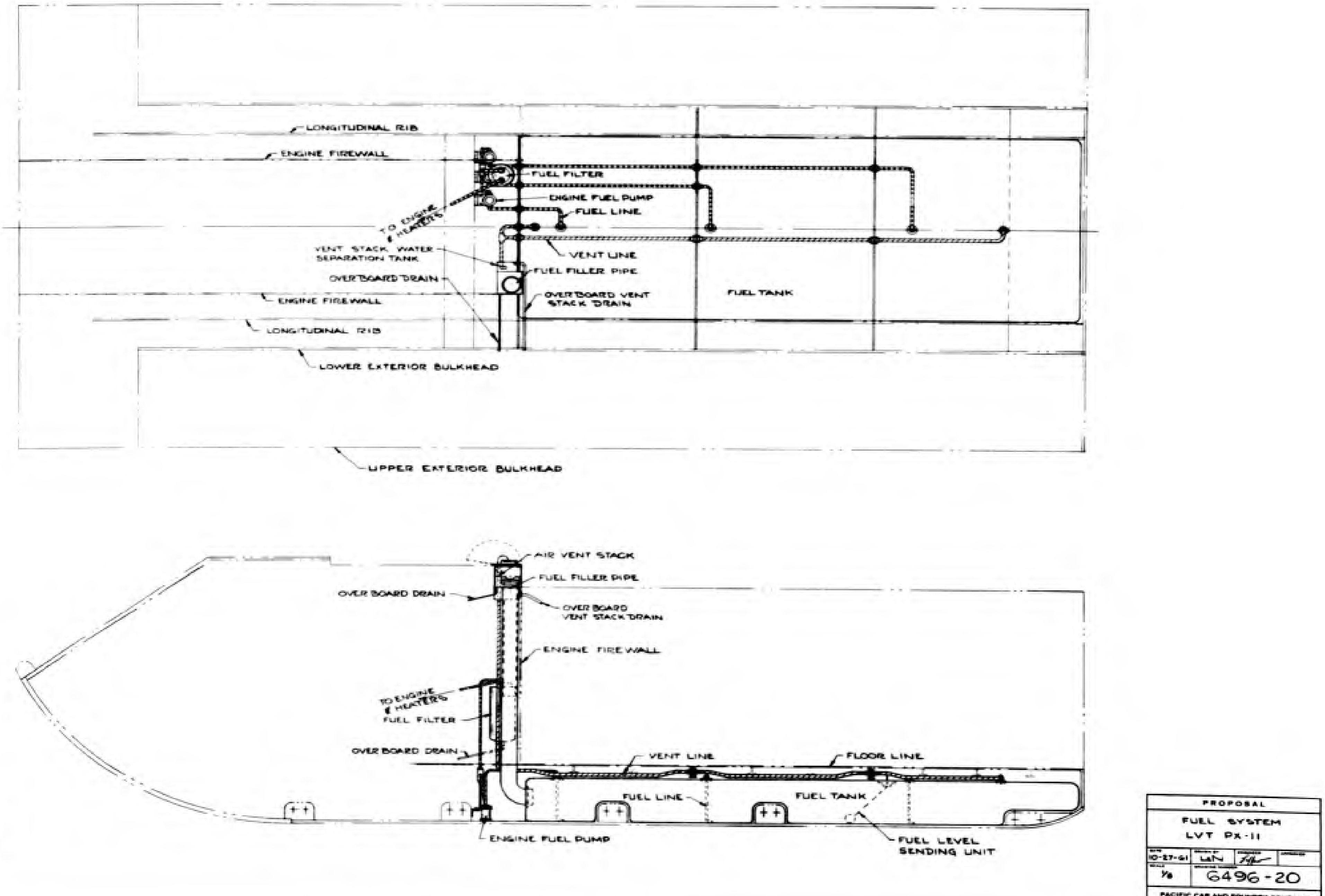
A single electric "float on arm" type fuel level sending unit is mounted in the top of the fuel cell.

The fuel filler pipe, fabricated of aluminum, is located forward of the cell and extends up to an armored waterproof box in the top deck. The filler includes an extensible tube to prevent water from leaking into the system during shipside fueling operations. The filler box has an overboard drain.

The cell is vented at each end by tubing that terminates in a water trap adjacent to the filler. The trap contains an overboard drain which goes to the left side of the vehicle. Since water running back through the drain has to fill the trap completely before it will flow into the vent, the vent will be adequately protected. Except where flexible hose is required, it is planned to use light, corrosion resistant, high impact polyvinylchloride plastic tubing for vent lines.

Because the hull structure is divided locally by the suspension system, the cell contains three pockets. Each pocket is a low point in the cell, so fuel is drawn from each pocket by "Autopulse" type pumps.

In addition to filters mounted on the engine, fuel passes through an additional fuel strainer - water separator. The strainer traps water and contains a multiple disk element that traps particles larger than .003 inches. The strainer reservoir will retain up to 1-1/2 gallons of water. By removing an engine compartment cover, the filter bowl can be drained either into a separate container or into the bilge.


LONGITUDINAL RIB
ENGINE FIREWALL
FUEL FILTER
ENGINE FUEL PUMP
FUEL LINE
TO ENGINE & HEATERS
VENT STACK WATER SEPARATION TANK
VENT LINE
FUEL FILLER PIPE
OVERBOARD DRAIN
OVERBOARD VENT STACK DRAIN
FUEL TANK
ENGINE FIREWALL
LONGITUDINAL RIB
LOWER EXTERIOR BULKHEAD
UPPER EXTERIOR BULKHEAD
AIR VENT STACK
FUEL FILLER PIPE
OVER BOARD DRAIN
OVER BOARD VENT STACK DRAIN
ENGINE FIRE WALL
TO ENGINE & HEATERS
FUEL FILTER
OVER BOARD DRAIN
VENT LINE
FLOOR LINE
FUEL LINE
FUEL TANK
ENGINE FUEL PUMP
FUEL LEVEL SENDING UNIT
PROPOSAL
FUEL SYSTEM
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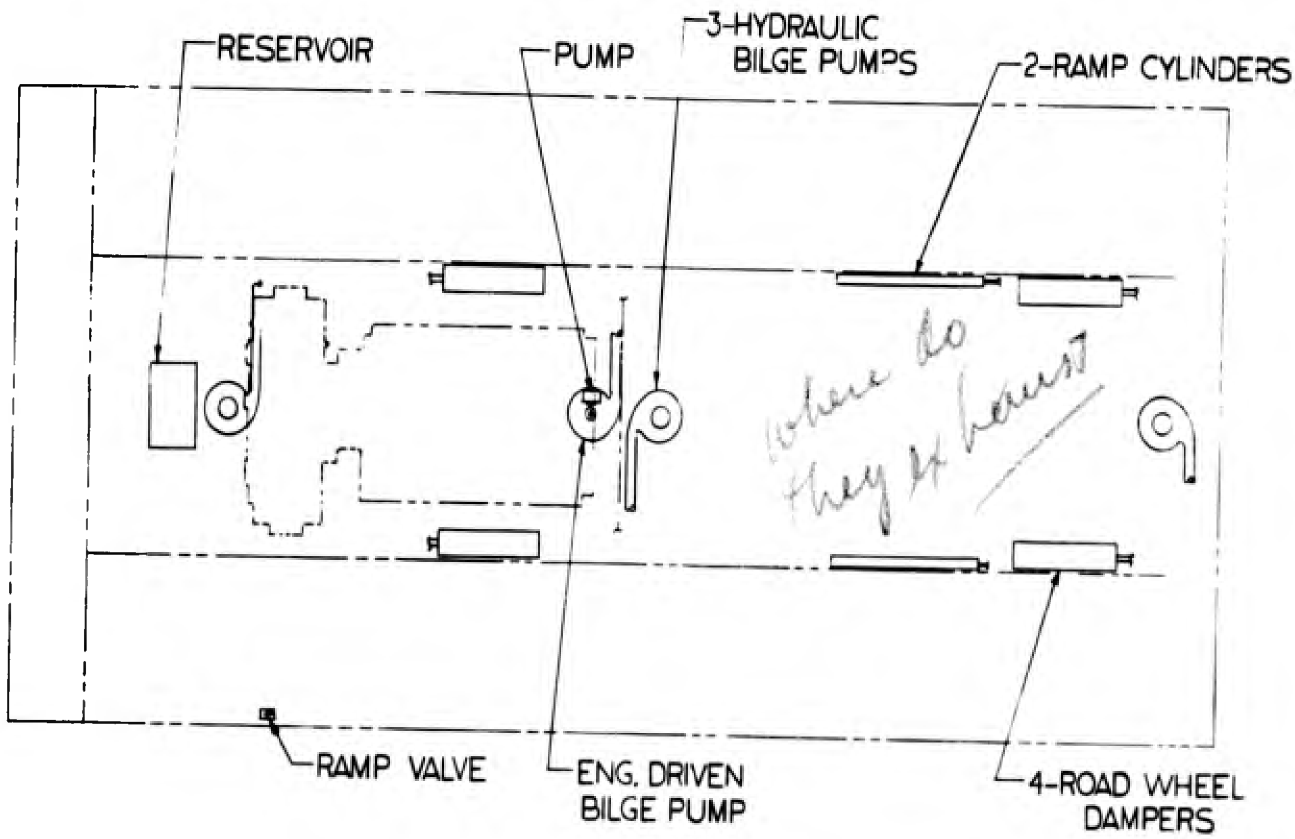
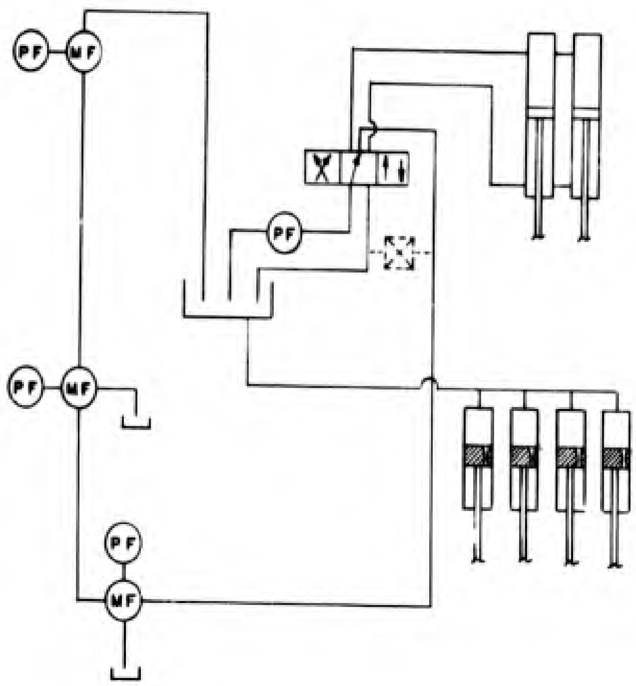
TOP VIEW

FWD.





HYDRAULIC SYSTEM SCHEMATIC

PROPOSAL

LVT PX-11
HYDRAULIC SYSTEM

DATE	DRAWN BY	ENGINEER	APPROVED
8-NOV-1961	K.M.B.	[illegible]	
SCALE 1/16	DRAWING NUMBER 6496-40		

PACIFIC CAR AND FOUNDRY COMPANY
RENTON, WASHINGTON

HYDRAULIC SYSTEM

A single hydraulic system is used to perform the following tasks. See drawing 6496-40.

1. Drive three bilge pumps.
2. Raise and lower the ramp.
3. Provide oil for the suspension lockout cylinders.

The system consists of:

1. A pump driven by the accessory drive box which is mounted on the accessory end of the engine.
2. A mobile equipment type valve that includes the system relief valve which is mounted conveniently to the operator.
3. Three hydraulic motors which drive the bilge pump.
4. Two cylinders that actuate the ramp.
5. A suitable reservoir located in the bow.

The system is arranged so that oil normally has unrestricted flow through the valve and travels to the bilge pump motors, which are in a series arrangement. The bilge pumps are continuously operating whenever the engine is operating.

The valve is designed to direct the oil flow to the bilge pump motors when the ramp valve is in neutral. When the ramp valve is in the "raise" or "lower" position, the bilge pumps will be momentarily inoperative.

Lockout cylinders require a supply of oil to make up any leakage that may occur. They are gravity-fed from the reservoir. For the special purpose vehicles, the feed lines can be pressurized by reduced system pressure.

ESTIMATE OF WEIGHTS

A list of components and their estimated weights are as follows:

Hull, including Hatches and Ramp	9,495 pounds
Cupola	75 pounds
Machine Gun and Turret	400 pounds
Towing Hitches	75 pounds
Seats and Crew	450 pounds
Controls	125 pounds
Instruments	75 pounds
Miscellaneous Deck Fittings and Rails	250 pounds
Suspension	3,060 pounds
Drive Sprocket Assemblies	532 pounds
Tracks	3,560 pounds
Engine	1,000 pounds
Power Train	1,290 pounds
Oil Coolers and Piping	75 pounds
Lube Oil	275 pounds
Cooling System and Ducts	530 pounds
Air Cleaners and Ducts	150 pounds
Muffler and Piping	75 pounds
Engine Mounts	55 pounds
Hydraulic System	175 pounds
Bilge Pumps	140 pounds
Electrical System	125 pounds

Batteries	150 pounds
Fuel System	175 pounds
Fuel	1,500 pounds
Fire Extinguisher System	160 pounds
OVE and Miscellaneous Equipment	1,028 pounds
Curb Weight	25,000 pounds
Cargo	10,000 pounds
Loaded Weight	35,000 pounds

The center of gravity of the entire vehicle at Curb Weight is located at 124.9 inches from the bow and 41.4 inches above ground level. The center of gravity of the entire vehicle at Loaded Weight is located at 142.2 inches from the bow and 48.5 inches above ground level. Center of gravity locations are depicted on drawing 6496-28.

At Curb Weight, the vehicle will float with 57.75 inches draft forward; 29.5 inches aft. At Loaded Weight, the vehicle will float with 55.75 inches draft forward; 70.25 inches aft.

PART 3

ALTERNATE PROPOSAL

DUAL ENGINE
ELECTRO-MECHANICAL DRIVE CONCEPT

In the Company's investigation of power plants and power trains, a number of components now under development showed enough potential to encourage this unique and practical design approach.

The use of these components allows the designer a freedom of choice of their location in the vehicle, resulting in a practical LVT of much smaller size with ideal weight distribution. Although the practicability of the engine and power train has not been fully proven, the advantages of this concept are significant and merit consideration.

Extraordinarily compact, the vehicle is only 19 feet long, with a cargo space 5 feet 8 inches wide by 16 feet long, and will accommodate 27 fully-equipped Marines.

In arranging the power train components, first consideration was given to "fail-safe" operation. All of the vulnerable power train components are dual or multiple and a failure of any component will, at most, only reduce the performance and not disable the vehicle. A vehicle schematic is shown on drawing 6496-21.

General Configuration

The outside appearance, hull, crew stations, tracks, and suspension, are all similar to the foregoing proposal. However, the interior arrangement and power package components differ considerably. Two engines, mounted in the rear of each sponson, each drive an alternator to power the vehicle. AC motors, coupled to a specially designed power train, are all located in the bow. Auxiliary equipment (OVE, ventilating system, etc.) is stowed in the sponsons, leaving the entire area between the sponsons available for cargo.

Engine and Power Train

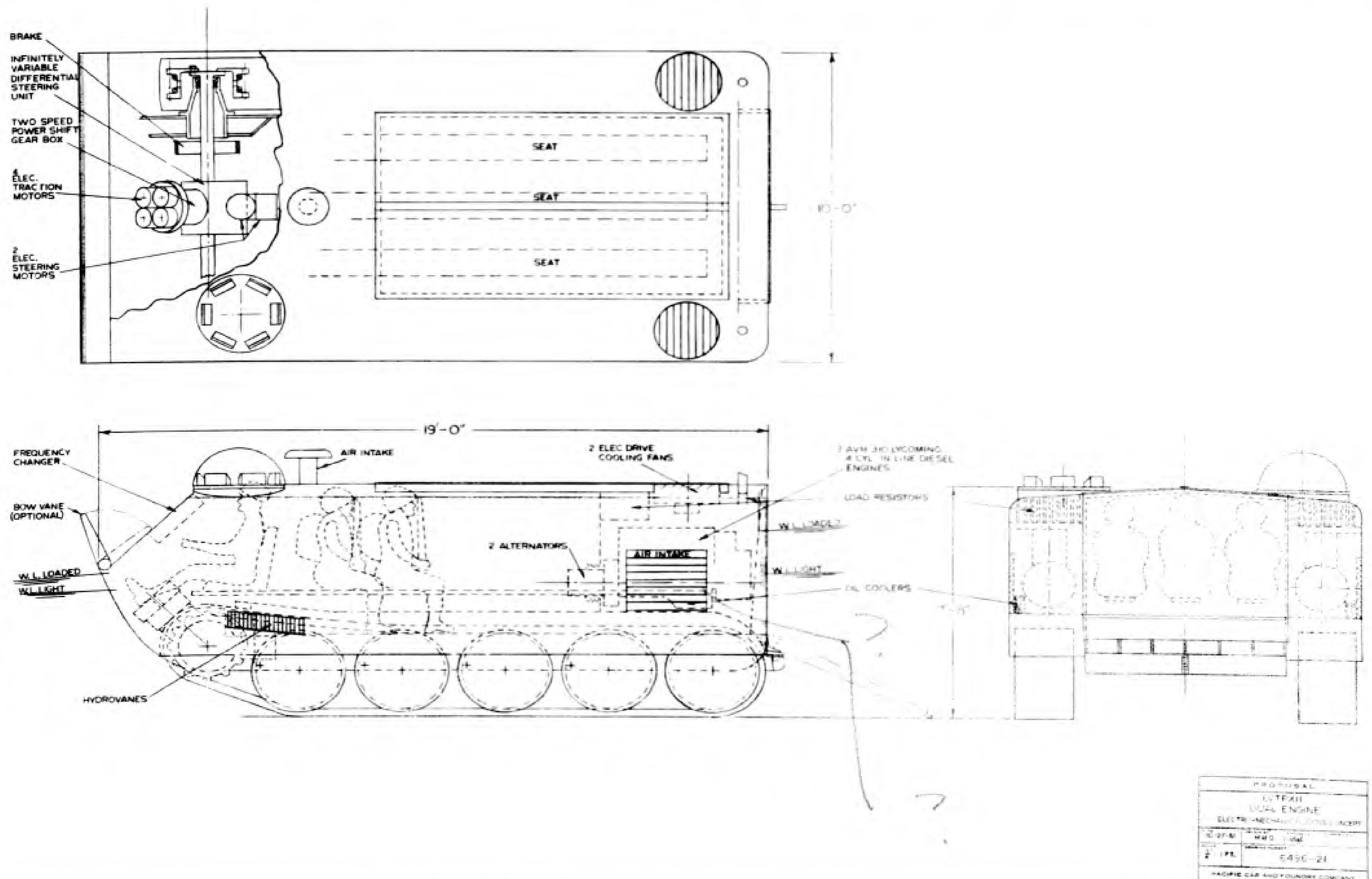
The complete power train can be classified as a diesel-electric differential drive. Two Lycoming 4 cylinder, in-line, 310 cu.in. displacement, air-cooled, multi-fuel, compression ignition, engines are used for the prime movers. Each engine equals one bank of the AVM 625 V-8 engine in the foregoing proposal.

The engines are each connected through a step-up gear box to a 7,500 RPM high frequency alternator. Current from the alternators is fed into dual silicon controlled rectifier frequency changes. The controlled frequency current is then fed to the propulsion motors, or steering motors, or both, by the driver's controls. Speed and steering are controlled by varying the frequency. The electric drive system is discussed further on page 3.20.

The induction motors for propulsion and steering are variable frequency 65 HP, 12,000 RPM, units. These are arranged in a cluster of four for propulsion and two for steering.

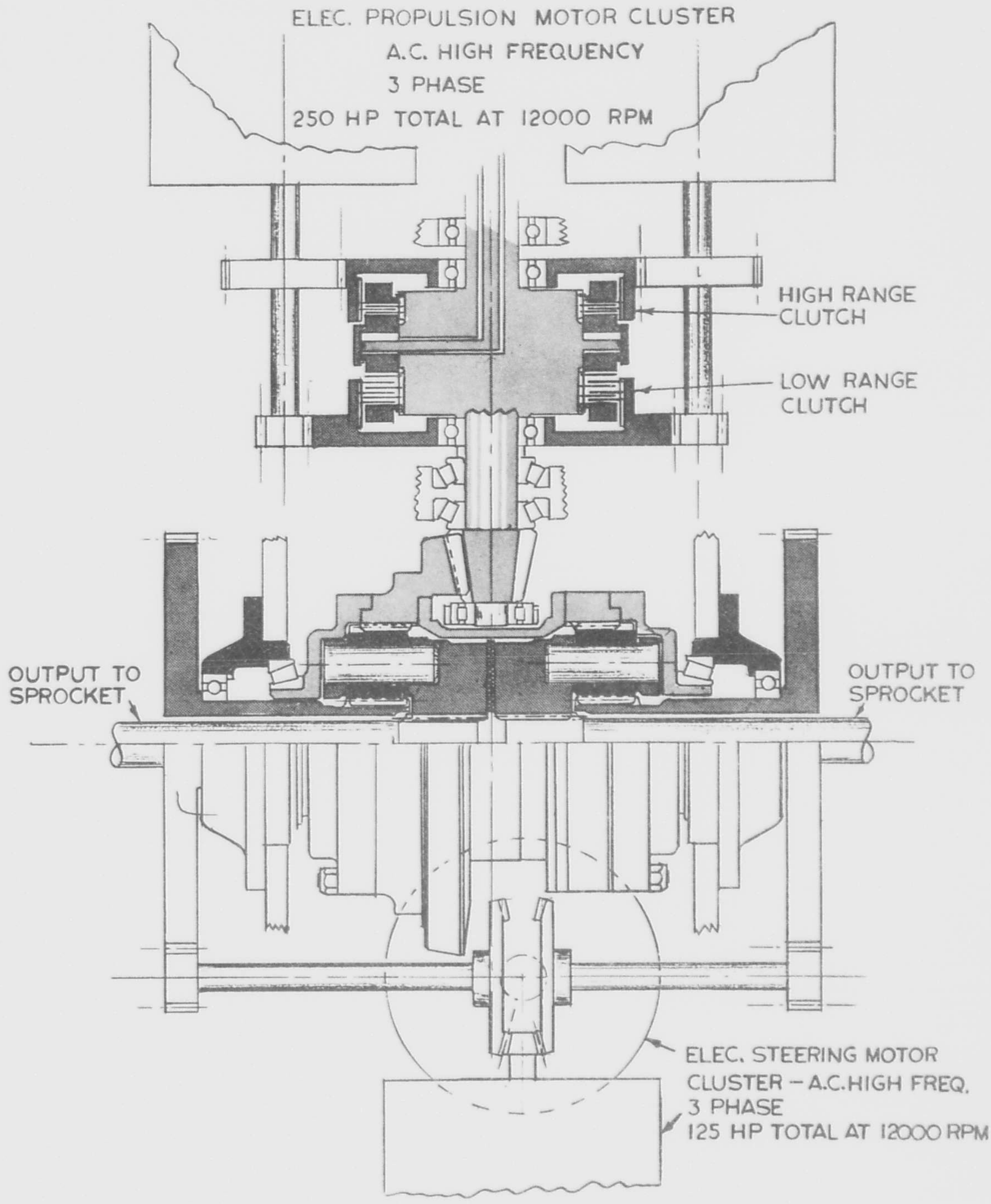
The propulsion motors are connected to a two-speed power shift gear box with a 3 to 1 ratio. The motors, themselves, have a 6 to 1 speed range. This gives the drives an 18 to 1 over-all variable ratio with two speed ranges. This ratio is adequate from 40 MPH on level ground to 2.5 MPH on a 70% grade. A schematic of this power unit is shown on drawing 6496-32.

Power is fed from the two-speed gear box into the common internal ring gears of a differential drive-steer unit. The two sun gears, reacting against the steer motor drive, serve as reaction members, and the power is transmitted through the planet carriers directly to the output shafts, which are connected to the drive sprockets. Unlike the conventional clutch-brake or so-called geared steer, the steering of this



BRAKE
INFINITELY VARIABLE DIFFERENTIAL STEERING UNIT
TWO SPEED POWER SHIFT GEAR BOX
4 ELEC. TRACTION MOTORS
2 ELEC. STEERING MOTORS
SEAT
SEAT
SEAT
10'-0"
19'-0"
FREQUENCY CHANGER
AIR INTAKE
BOW VANE (OPTIONAL)
W.L. LOADED
W.L. LIGHT
HYDROVANES
2 ALTERNATORS
AIR INTAKE
2 ELEC DRIVE COOLING FANS
3 AVM 310 LYCOMING 4 CYL. IN LINE DIESEL ENGINES
LOAD RESISTORS
W.L. LOADED
W.L. LIGHT
OIL COOLERS
PROPOSAL
LVTPXII
DUAL ENGINE
6496-21

6496-21



ELECTRO-MECHANICAL DRIVE-STEER UNIT
INFINITELY VARIABLE STEERING
PACIFIC CAR AND FOUNDRY CO. RENTON WASH.
DWG NO. 6496-32

unit is smooth, continuous, and positive, regardless of load, grade, or speed conditions. When steering, there are no clutch releases or brake engagements.

Steering

The track drive sprockets are always "locked in" at a ratio with each other which is proportional to the steering wheel position and the speed of the vehicle. That is, a given displacement of the steering wheel creates a fixed difference of track speeds. At high vehicle speeds, this will produce a gradual turn; at lower speeds, a sharper turn.

With the steering motors at zero RPM, the vehicle assumes a straight course. When the steering motors are energized with the propulsion motors at zero RPM, the two sun gears of the drive-steer unit are rotated at the same speed in opposite directions. This delivers equal velocity forward to one track and reverse to the other, which results in pivot steer.

If the vehicle is in motion when the steering motors are actuated, then the rotation of the sun gears, in opposite directions, adds to the speed of one track and subtracts from the other, which results in a normal turn, the rate of turn being a function of the vehicle speed and the input speed of the steering motors, which is controlled directly by the driver. This drive-steer unit has equal performance in either forward or reverse, and in all modes of operation, the regenerative steering torque is taken by the gear train, and the steering motors need only supply enough energy to skid the tracks.

The maximum steer rate selected for this vehicle was predicated on water maneuvering requirements. The rate has been arbitrarily set at 14 MPH maximum speed difference between the tracks.

Cooling

The electric motors, alternators, and gear boxes are oil-cooled. Their waste heat is dissipated through oil-to-air heat exchangers located low in the engine compartments. During land operation, a portion of the cooling air is circulated through these heat exchangers. In water operation, they are submerged. The frequency changer is cooled by contact with the hull.

The engines are air-cooled while on land. Cooling air is circulated by thermostatically-controlled electrically-driven fans. Afloat, the engines are either fully or partially submerged depending upon the cargo load and its distribution. With the engines fully submerged, there is no cooling problem. However, when the engines are only partially submerged, some air must be circulated to cool the cylinders. The thermostats located on the cylinder heads will automatically start the fans whenever they are needed. The low mass in their electric drives permits the fans to be designed to be submerged while operating without sustaining damage.

The cooling system is unquestionably the simplest that has been proposed for an air-cooled engine in an amphibious vehicle of this type. It completely eliminates all shutters, radiators, and snorkels.

The Lycoming Company is aware of this unusual cooling system, and they concur in its feasibility since their engines are designed for submersion during deep water fording. Precautions, however, must be taken to prevent deterioration of the engine due to the repeated immersions it will undergo.

Aspiration Air

On land, a portion of the engine aspiration air is taken from the cargo compartment to provide fresh air for the crew and troops.

In water, all of the engine aspiration air is taken from the cargo compartment. This air is admitted to the cargo compartment through the air intake in the forward top portion of the hull. This intake is equipped with an automatically closing valve which is explained in Parts I and II.

Bilge Pumps

Three electrically driven bilge pumps are located in the space below the cargo deck. They are activated automatically when the vehicle is in the water.

Water Performance

This vehicle is designed to have the highest water speed and maneuverability attainable within the hull configuration dictated by other considerations.

"Hydrovanes" are provided in the upper track channel to re-direct the water on the top of track aft to recover as much thrust as possible.

Well-rounded corners fore and aft reduce the drag about the bow and transom.

With equal track speed and steering in either forward or reverse, the water maneuverability is greatly increased.

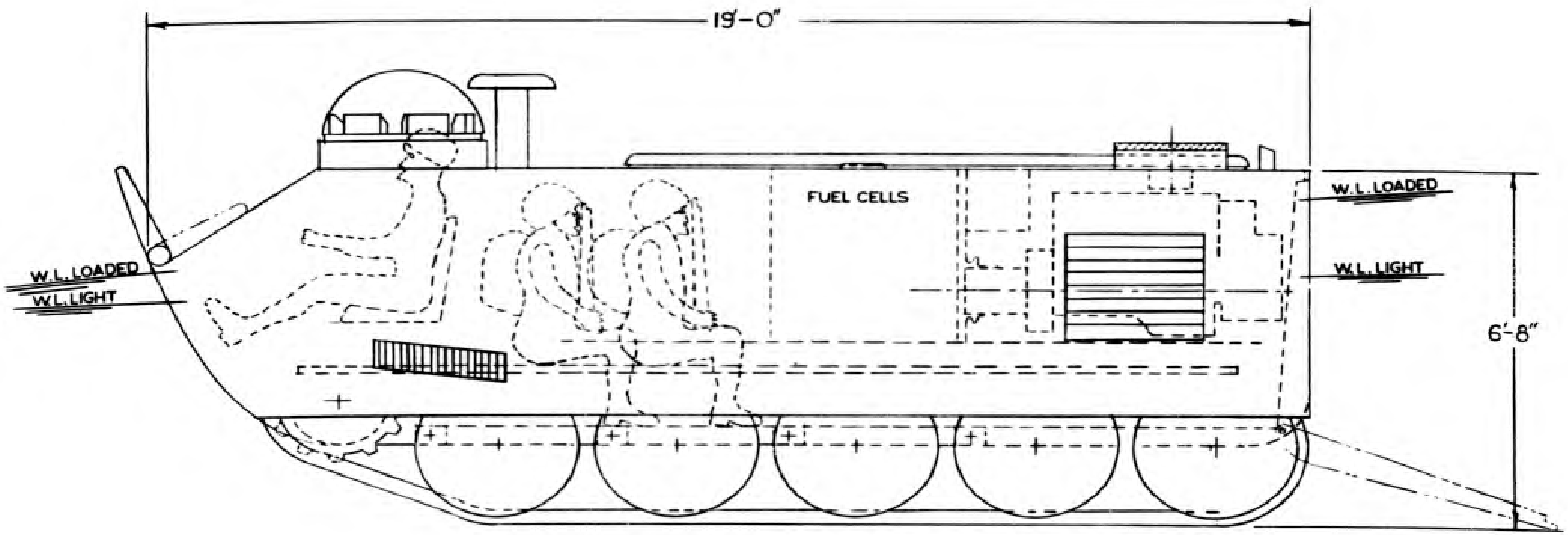
Bonus Features

The vehicle can, without alteration, supply:

1. 15,000 watts of AC power at any frequency from 60 to 400 cps over a wide range of voltages up to 250 volts.
2. 15,000 watts of DC power at voltages up to 250. This power may be used for welding, heating, slave duty, etc.

With the flexibility of component arrangements, it is possible to place the fuel tanks on the sponsons and lower the cargo deck approximately 12 inches. This would lower the height of the vehicle by the same amount and further reduce the tendency toward over-buoyancy.

A schematic of the vehicle is shown on drawing 6496-31.



PROPOSAL

LVTP(X)
DUAL ENGINE
ELECTRO-MECHANICAL DRIVE
LOW SILHOUETTE CONCEPT

HMO

6496 - 31

PACIFIC CAR AND FOUNDRY COMPANY

TRIAL CONCEPT NO. 1

An early concept, later discarded, proposed the use of the Boeing gas turbine, a Jack and Heintz variable frequency electrical drive to each road wheel (no drive sprockets), and pneumatic tires mounted on walking beams (no suspension springs). Drawing 6496-1 shows the arrangement.

Although the turbine is very light and is in production, its high cost ($12,000), high fuel consumption, and the requirement for a large uninterrupted flow of air defeated its further consideration.

Individual wheel drive eliminates drive sprocket and provides mobility even without tracks. The tracks may also be lighter. Traction between the wheels and tracks is not, however, reliable during water or mud operation. The other merits and disadvantages of electric drive are discussed elsewhere.

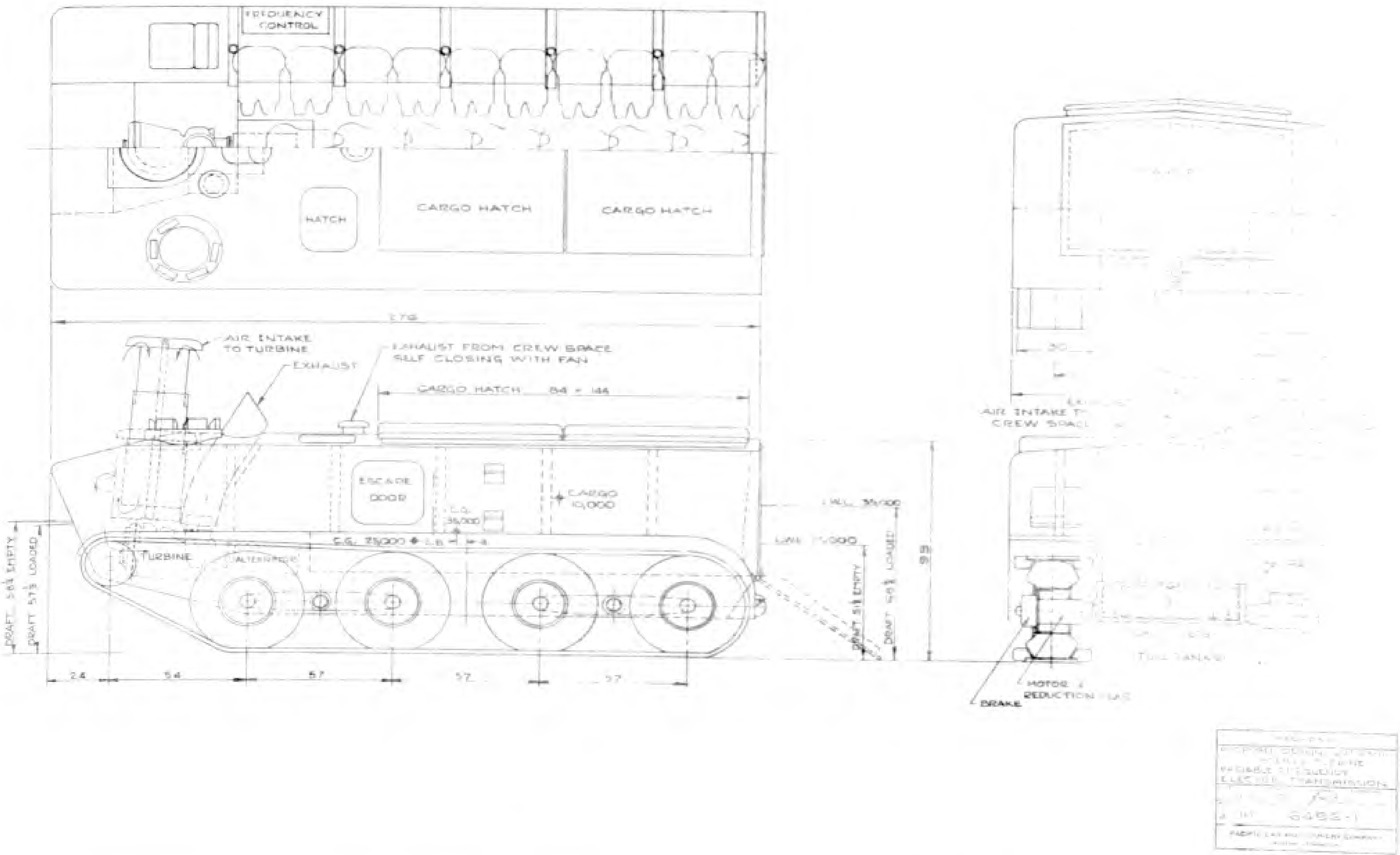
The suspension arrangement eliminates all springs. On the other hand, pneumatic tires are heavy and vulnerable to small arms fire and burst fragments.

In summary, the concept was discarded because of component disadvantages.

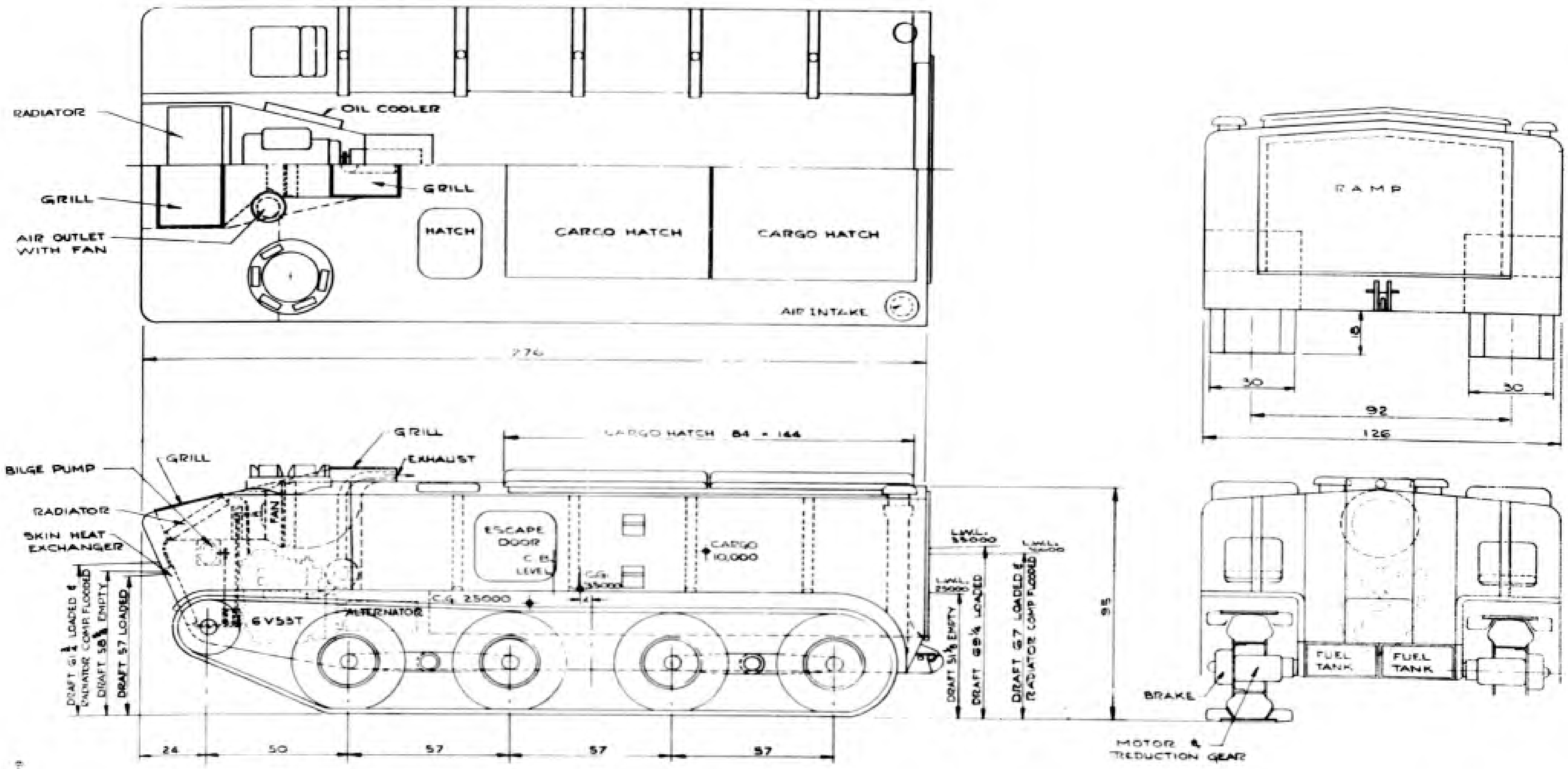
TRIAL CONCEPT NO. 2

Another concept, also discarded, replaced the Boeing turbine with a General Motors 6V53T diesel and substituted a recent small diameter tire design for the large tires of Trial Concept No. 1 to reduce over-all vehicle height.

An evolutionary design, Trial Concept No. 2 differs little from its predecessor. The general arrangement is shown on drawing 6496-9.



FREQUENCY CONTROL
HATCH
CARGO HATCH
CARGO HATCH
176
AIR INTAKE TO TURBINE
EXHAUST
EXHAUST FROM CREW SPACE SELF CLOSING WITH FAN
CARGO HATCH 84 × 144
ESCAPE DOOR
CARGO 10,000
C.G. 35,000
C.G. 25,000
TURBINE
ALTERNATOR
W.L. 35,000
LWL 25,000
DRAFT 58½ EMPTY
DRAFT 57½ LOADED
DRAFT 51½ EMPTY
DRAFT 56½ LOADED
99
24
54
57
57
57
30
AIR INTAKE TO CREW SPACE
MOTOR & REDUCTION GEAR
BRAKE
FUEL TANKS
TRANSMISSION



6496-9

PROPOSAL

REVISED DESIGN LVT PX-11
6V53T DIESEL 15 × 21 TIRES
VARIABLE FREQUENCY
ELECTRIC TRANSMISSION

7-48

6496-9

PACIFIC CAR AND FOUNDRY COMPANY

ENGINES

A number of compression ignition engines were investigated for possible use in the LVTPX11. Only those that have or can incorporate a multi-fuel capability were considered. All, except the Lycoming and Continental, were liquid-cooled.

General Motors Diesels

Both the 71 cu.in. and 53 cu.in. series engine are within the possible power range of this vehicle. Pertinent features of the 671T engine are as follows:

Cylinder Arrangement	In Line
Number of Cylinders	6
Gross Power	310 HP
Speed	2300 RPM
Fuel Consumption	.39 lbs/BHP/hr
Cycle	2
Displacement	426 cu.in.
Aspiration	Turbocharged
Length	55-1/2 inches
Width	32-1/2 inches
Height	53-3/4 inches
Weight (w/ aluminum block)	1,820 pounds

The 6V53T engine will be used on an interim basis in the 34,000 pound ARAAV vehicle being developed by Army Ordnance. Ordnance plans to change the ARAAV engine shortly because of the low power output of the 6V53T. The engine has the following features:

Cylinder Arrangement	90°V
Number of Cylinders	6
Gross Power	255 HP
Speed	2800 RPM
Peak Torque	518 lb.ft.
Fuel Consumption	.390 lbs/HP/hr
Cycle	2
Displacement	319 cu.in.
Aspiration	Turbocharged
Length	38 inches
Width	34 inches
Height	44 inches
Weight	1,410 pounds

Although the power of the 671T engine is adequate and should provide reliable operation, the horsepower-to-weight ratio is high. The engine does not incorporate recent advances in engine development that reduce weight. The excessive length of the engine shortens the cargo space.

The 6V53T engine, being quite short, fits well. However, gross horsepower is 255. Deducting power to drive the cooling fan and accessory drives, horsepower to the transmission is 209; this is not considered adequate to provide desired vehicle performance.

Caterpillar Engine

The Caterpillar Tractor Company is developing a line of light-weight engines that use aluminum extensively. Their D333 engine is within the power range requirements of the LVTPX11, and has the following characteristics:

Cylinder Arrangement	In Line
Number of Cylinders	6
Gross Power	320 HP
Speed	2400 RPM *
Peak Torque	770 lb.ft.
Fuel Consumption	.41 lbs/BHP/hr
Cycle	4
Displacement	525 cu.in.
Aspiration	Turbocharged
Length (approx.)	50 inches
Width (approx.)	30 inches
Height (approx.)	45 inches
Weight (approx.)	1,600 pounds

* Engine RPM can be increased to 2800 if desired.

This engine offers no weight advantage over others considered. Its durability has yet to be proven. Under Ordnance sponsorship, the Company is installing the somewhat larger LDS-750 Caterpillar engine in a T236 Vehicle. Testing will begine the latter part of 1961.

Cummins Engine

The Cummins Engine Company has recently developed advanced, compact 6 and 8 cylinder engines. The 6 cylinder engine could be adapted to the LVTPX11. At the preset, it is rated at 200 HP without turbocharger.

Cummins would not furnish performance data for the engine with turbo-charger until it has been tested and proven. However, they estimate that 300 horsepower can be obtained.

Present characteristics are:

Cylinder Arrangement	90°V
Number of Cylinders	6
Power	200 HP
Speed	2600 RPM
Peak Torque	444 lb.ft.
Fuel Consumption	.37 lb/BHP/hr
Cycle	4
Displacement	588 cu.in.
Aspiration	Natural
Length	36 + inches
Width	33 inches
Height	39 inches
Weight	1,475 pounds *

* An aluminum version is planned for release in the latter part of 1963. Weight is estimated to be 1,200 pounds.

Continental Engine

The Continental Aviation and Engineering Corporation is developing an air-cooled multi-fuel engine for military applications. This engine is too large for the LVTPX11 (425 BHP gross and 2,592 pounds).

Characteristics of the Lycoming AVM-625 engine were given in Part I. After considering the additional weight of radiators, fans, and coolant for each engine, the weight of the Lycoming installation was found to be 700 pounds less than the lightest liquid-cooled installation (GMC 6V53T engine). This weight reduction, coupled with the advanced design of the Lycoming and simplified float system for preventing entrance of water during water operation, led to the selection of the Lycoming engine over others considered.

HYDROSTATIC DRIVES

Several hydrostatic transmissions using piston pumps and motors were investigated.

Some are completely hydraulic drives while others incorporate combinations of hydraulic and mechanical drives. All, except the Gar Wood concept, require the power source to be closely coupled to power train. Such close coupling eliminates "freedom of design" for remotely placing the power source for best weight distribution and space advantage.

The main advantage of the hydrostatic drives is their ability to provide smooth stepless ratio changes which can be governed to operate the engine at its most economical speed, and positive steering that is infinitely variable over its operating limits. It is doubtful that the limited advantage gained warrants the cost.

These drives fell into three categories which may be described as:

Type 1 Fixed displacement motors with variable displacement reversible pumps (Gar Wood)

Type 2 Variable displacement non-reversing motors with variable displacement non-reversing pumps (Sundstrand)

Type 3 Combination gear transmission with parallel hydrostatic unit to smooth steps (Vickers and General Electric)

The Type 1 System is being developed by Gar Wood Industries, Inc. It consists of a variable displacement reversible pump driving fixed displacement wheel motors with two-speed gear boxes. Steering details were not furnished by the developer. The transmission does not offer any weight or efficiency advantages over the Allison XTG-250. Its main advantage is that it permits a choice of location for the power plant. Some hardware is being tested.

<u>The Type 2 System</u> is a different approach to the hydrostatic drive problem under development at Sundstrand Avaiation. This system combines a variable displacement non-reversing pump with two variable displacement non-reversing motors, with valving and lines, into a single package. The motors and pump are of similar design and embody a cross-head type of piston action which greatly increases the variable range. The increased variable range makes it possible to couple the motors directly to the final drives without the use of a shifting gear box between the motor and the final drive. Since neither the pump nor motors will reverse without reversing the hydraulic flow, steering is somewhat more complicated than with some of the other arrangements. When negative track torque is required for steering, reversing valves are actuated by the steering mechanism, which reverses the hydraulic flow to the inside track; braking the track and thereby providing the necessary steering effort. A block diagram, Figure 27, on the following page, shows the basic arrangements.

The advantages of this system are:

1. No long high pressure lines.
2. Compact design. Unit can be substituted for conventional cross drive transmission without major hull alterations.
3. Engine speed can be programmed for most efficient operation.

Disadvantages:

1. Steering is complicated.
2. Hydraulic motors must be four times the size needed for propulsion in order to handle the torques developed in steering.
3. Engine, transmission, and final drive weight are all concentrated in one area on the vehicle.
4. Over-all efficiency is no better than conventional cross drive transmission.
5. Weight is greater than a mechanical transmission; 1500 to 1600 pounds. (200 to 300 pounds more than the XTG-250).

In addition to the transmission described in the foregoing paragraphs, the Sundstrand Company (at the suggestion of the Company) also proposes a differential hydrostatic drive. The hydraulic motors each drive the sun gear of the planetary final drives. The internal ring gears are geared to a steering hydraulic motor and the planet carriers are coupled directly to their respective sprockets as shown in Figure 27.

Steering is accomplished by rotating the internal ring gears in opposite directions. Steering is continuously variable. Regenerative power is not taken through the hydrostatic components.

We believe that this is the most practical approach to the hydrostatic transmission. Its outside configuration would permit it to be easily substitued for the Allison XTG-250 transmission for testing.

A schematic is shown on Sundstrand's sketch, Figure 28.

BLOCK DIAGRAM
1 PUMP - 2 MOTORS

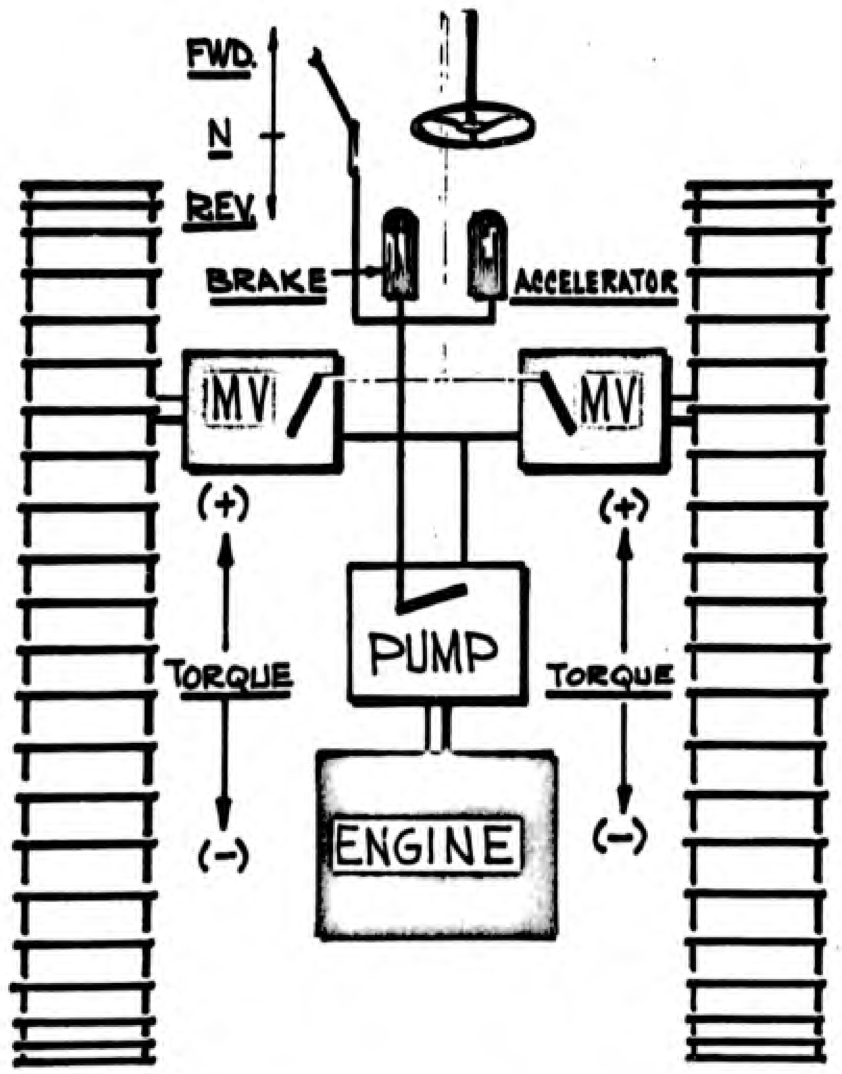
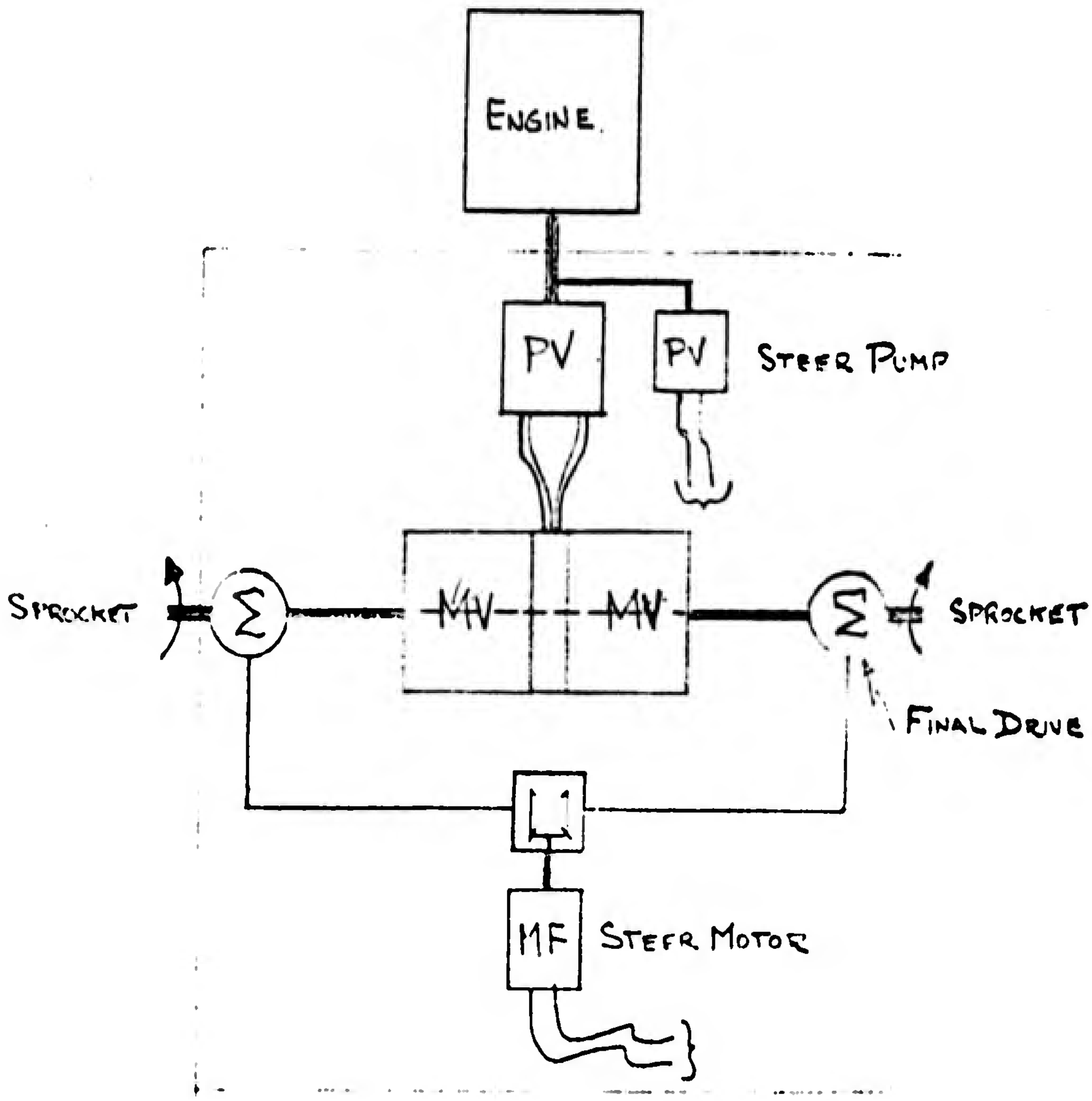


FIGURE 27

DIFFERENTIAL STEERING HST



CONTROLS

A. MODE SELECTOR ~ FWD, NEUTRAL, REVERSE

B. ACCELERATOR ~ TORQUE & ENGINE SPEED CONTROL

C. BRAKE ~ DYNAMIC BRAKE

D PARKING BRAKE ~

E. STEERING ~ DIFFERENTIAL TRACK SPEED

FIGURE 28

<u>The Type 3 System</u>. Two hydrostatic drive concepts were received with split power paths, in which power is transmitted through parallel fixed ratio (mechanical) and variable ratio (hydrostatic) paths. The intent is to combine the high efficiency of a mechanical drive with the varaible output of a hydrostatic drive.

The General Electric Company proposed a system with two pumps and motors mounted on an engine-driven cross-shaft. Each end of the cross-shaft drives the sun gear of a planetary gear set. A fixed displacement hydraulic motor drives each of the ring gears in the planetary. The planetary carrier connects to the output to each track.

Output speed is controlled by varying the hydraulic motor speed. A reversible, variable displacement pump allows the motor speed to be gradually varied from a maximum forward to the same maximum in reverse. The motor, therefore, can either add or subtract to the output. In this case, the planetary ratio is such that the vehicle speed is approximately 15 MPH when the ring gear is stationary (motor stopped). For speeds greater than 15 MPH, the motor runs in a forward direction, adding to the output speed. As the motor is driven in the reverse direction, output speed is reduced, halted, or reversed.

Steering is simply accomplished with a speed difference between the motors.

Operating efficiency is said to be about 85% in the moderate speed ranges, peaking out when power is entirely transmitted mechanically (zero motor speed). It drops off moderately at higher speeds and significantly at lower speeds because of the high recirculation of power through the hydrostatic components.

Estimated weight of this power train is comparable to straight mechanical power trains.

General Electric recommends their ball piston pumps and motors for this power train. They are currently used in aircraft constant speed alternator drives. A straight hydrostatic drive without the split mechanical path and steering is being developed for a wheeled vehicle. Operational testing has not been started. The method of varying the ratio is interesting and results in a relatively uncomplicated drive.

Vickers, Incorporated, proposes a hydrostatic system which divides the power into three parallel paths - two power flow paths, to permit the clutch in the idler path to shift without load, and a steering path. The power train basically is a 4-speed mechanical transmission which utilizes hydrostatic components to provide smooth stepless ratio changes and variable steering ratios.

Approximately 20% of the power passes through the hydrostatic path and 80% through the mechanical path.

Vickers has provided a schematic block diagram, Figure 29, shown on the following page. The input shaft simultaneously drives Gear Set #1 and a variable delivery steering pump. One portion of the output of Gear Set #1 drives Gear Set #2 through the Ratio Hydraulic System. The other portion of the output of Gear Set #1 drives Gear Set #3 through either the Low Range or High Range Clutch. All clutches engage and disengage when there is no power flow through them. Therefore, shifting is smooth and long clutch life is assured.

Gear Set #2 drives Gear Set #4 through either the High Range, Low Range, or Reverse Clutch. Again, clutch engagement occurs at no load.

SK-3372

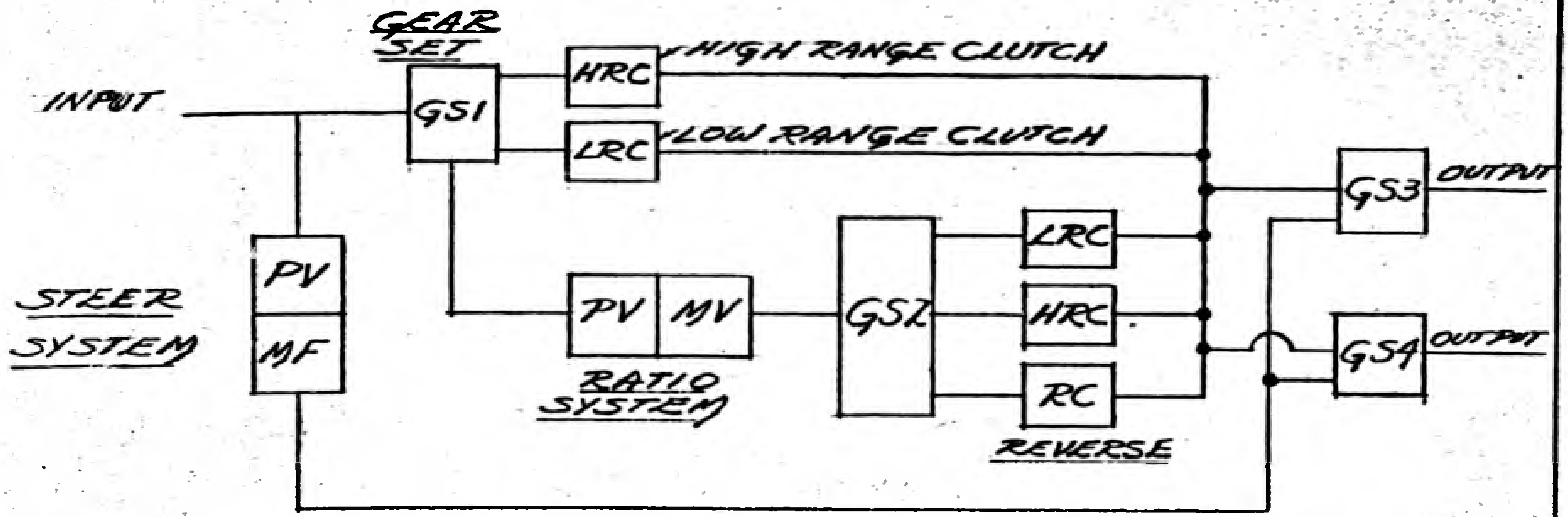


FIGURE 29

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	CARDWALE	10-27-61
CHECKED		
METHODS		
TOOL-ENG.		
EXAM.-ENG.		10-27-61
APPR.-ENG.		10/27/61
REL. FOR PROD.		

UNLESS OTHERWISE SPECIFIED
MACHINED DIMENSIONS
DEC. ± .007 FRAC. ± 1/64
ANGULAR ± 0°-30'
HOLES TO BE LOCATED WITHIN .007 OF THEIR TRUE LOCATION
SURFACE FINISH ✓
CONCENTRICITY TO BE HELD WITHIN ______ TOTAL INDICATOR READING
MACH. SURFACES INTERSECTING AT RIGHT ANGLES SHALL BE SQUARE WITHIN ______ PER INCH

MATERIAL

HEAT TREATMENT

UNIT SIZE-

FIRST USED ON- NEG 2352

SIMILAR TO-

PART OF SUB. ASSEMBLY

VICKERS INCORPORATED
WATERBURY PLANT
WATERBURY, CONN., U.S.A.

TITLE- SCHEMATIC BLOCK DIAGRAM-LVT

SCALE-

SPECIAL FINISH-

PATT. OR FORG. NO.-

B SK-3372 ISSUE

AERO 120

The output of both sets of clutches is actually a common cross-shaft. The cross-shaft powers Gear Sets #3 and #4, which reduce speed to the desired output.

Steering is accomplished by positioning the yoke of the steering pump at some positive or negative angle, depending on the desired direction. This ports flow to the fixed displacement motor. The motor output shaft adds speed to one track and subtracts the same amount from the other track through an idler gear.

Efficiency of the unit is approximately equal to that of the Allison XTG-250 Power Train. Weight and configuration of the unit are also comparable to the XTG-250.

Development has not reached the hardware stage. Vickers intends to use "off the shelf" hydraulic pumps, motors, and accessories. The gear train is typical for a mechanical transmission and poses no particular problems.

The advantages of this unit lie in the stepless ratio changes and variable, but positive, steering.

MECHANICAL DRIVE

The Curtiss-Wright Corporation is developing a traction drive that uses rollers running between two toroidal discs to provide a stepless variable ratio in a mechanical transmission. Although the principle of a friction drive is not new and units of limited horsepower have been used, Curtiss-Wright states that recent developments have eliminated problems that formerly resulted in poor efficiency and durability characteristics.

The transmission uses a toroidal drive unit with conventional multiple disc clutches and brakes. Steering is accomplished by differentially mounting another toroidal drive unit in the output shaft to reduce the speed of one output shaft and increase the speed of the other. Clutch-brake steer is made possible by the inclusion of (1) a lockup clutch on the steering differential, (2) a disengage clutch on the steering cross-shaft, and (3) individual engagement of service brakes on each output. A schematic of this arrangement and a preliminary installation drawing are included as Figures 30 and 31 on the following pages.

The efficiency of the power train is estimated to be approximately 85% over most of its operating range.

Weight of the power train would probable be about 1500 pounds.

Additional testing will be required before a specific statement can be made regarding the reliability of the toroidal drive.

As with the hydrostatic drives, the main advantage of this drive is its ability to provide smooth speed changes and steering variations.

TOROIDAL TRACTION DRIVE TRANSMISSION SCHEMATIC

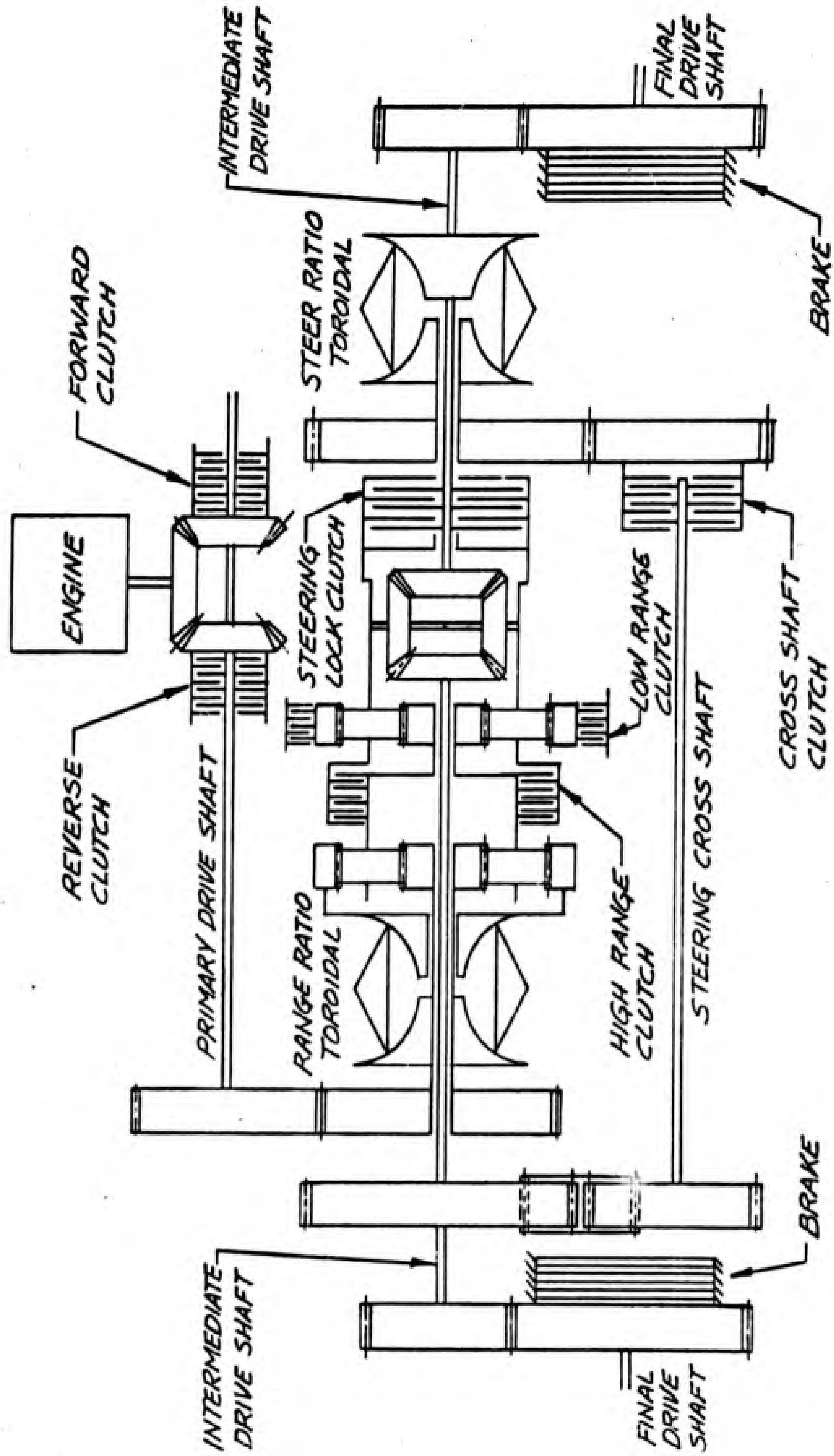


Figure 30

PRELIMINARY INSTALLATION DRAWING
TOROIDAL TRACTION DRIVE - RANGE AND STEERING TRANSMISSION

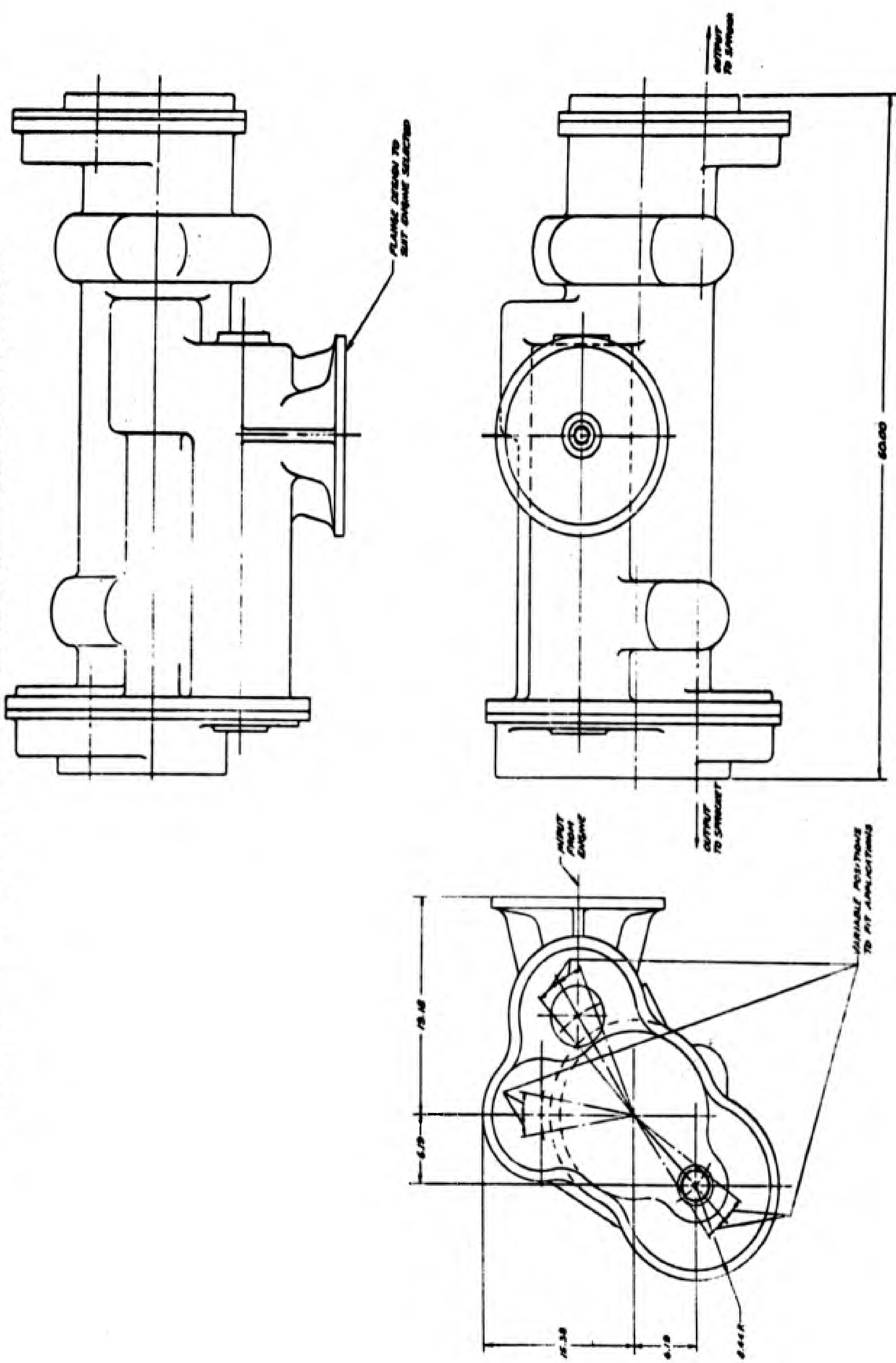


Figure 31

ELECTRIC DRIVES

Several "electric drives" were investigated. They all convert mechanical energy to electrical energy and electrical energy back to mechanical energy to drive the vehicle. Their fundamental advantages are:

1. Electrical energy can be indirectly transmitted allowing design freedom in vehicle configuration and arrangement.
2. Within limits, output torques and speeds can be smoothly graduated electrically thus simplifying or eliminating variable mechanical drives.

Electrical transmission can be grouped in three classes:

1. All DC Drives.
2. AC-DC Drives.
3. All AC Drives.

1. <u>All DC</u> The Westinghouse Electric Corporation suggested a direct current system which would be quite cumbersome in a vehicle. The drive comprises a DC generator and one or more DC motors, usually a series-wound type. This system has infinitely variable speed and torque and would be ideal except for the large size and weight of the components. By increasing the speed of the components, the size and weight can be reduced, however, with speed increase, commutation becomes difficult. Motors and generators requring commutators and brushes should be avoided from a reliability standpoint; sea water corrosion could be a serious problem.

2. <u>AC-DC Drives</u> The General Electric Company recommended a modified DC drive that makes use of the recently developed solid state rectifier to utilize the desirable features of both the AC and DC systems.

The system consists of an AC high speed, high frequency alternator, a solid state rectifier and DC motors driving the tracks. The use of the alternator presents a considerable saving in weight and size and also eliminates the commutator and brushes. The DC motors, even in the high speed version, are large and heavy. The two motors required to drive the LVTPX11 sprockets would weight 1,600 pounds each. This weight makes the system impractical.

3. <u>All AC Drives</u> The most promising system considered for the LVTPX11 consists of a high speed alternator, a solid state frequency changer using silicon-controlled rectifiers, and one or more high speed AC squirrel cage motors. The alternator and motors are of high speed, high frequency design using solid rotors without rotating electrical windings or slip rings. Their size and weight are considerably less than any DC units. The frequency changer varies the frequency and voltage at the command of the operator. This system is under development by several manufacturers, however, none of these systems has been advanced to the hardware stage. The Jack and Heintz Division of Siegler Corporation has contracts to apply their system to military vehicles and until some of these contracts are completed, no detailed performance data will be available.

General information, however, concerning the physical and estimated performance characteristics was obtained. Over-all efficiency of the drive from the engine output to the tracks is 65%. This is generally lower than the mechanical transmissions presently in use; however, this efficiency is fairly constant over all load and speed ranges while the efficiency of the mechanical transmission decreases at partial load and high torque multiplications.

Because of the large motors needed in all electric drives to produce the high tractive effort required for negotiating grades and for turning, weight of the power train will exceed 2,000 pounds, which is considerably more than the mechanical transmission weighs. Weight of the motors, however, can be reduced by combining them with a mechanical drive to multiply motor torque. The solution to the weight problem is not yet available from Jack and Heintz.

<u>Application of Variable Frequency AC Drives to an LVT</u>

Assuming that the inherent difficulties of this system can be solved, its application opens up interesting possibilities. The main power generating unit location is no longer governed by location of the sprockets, wheels, and transmission; it can be placed anywhere in the vehicle for best weight distribution. Several feasible engine-alternator combinations are:

1. Single turbine and alternator placed on the floor between the two drivers.
2. Single diesel and alternator placed in the same location as 1.
3. Two turbines and alternators, each unit placed on the sponson anywhere behind the drivers. This configuration leaves the center of the hull clear for cargo and personnel over the full length of the vehicle.
4. Two diesel engines and alternators placed in the same manner as 3.

These systems enable the tractive force to be applied in different ways:

a. By placing motors in all road wheels, preferably pneumatic-tired wheels, a vehicle exists that is not dependent on the track for mobility. As the track no longer transmits all the tractive force, it can be much lighter. With one or both tracks broken the vehicle can proceed as usual on all but extremely rough terrain. There is, however, some possibility that under conditions requiring high tractive effort or in water where the coefficient of friction between the tracks and wheels is reduced, the wheels will slip inside the track.

b. By placing motors in the front and rear wheels only, using larger motors, the danger of slipping might be eliminated.

c. By placing motors in the front idlers and in the rear wheels, the danger of slipping would be eliminated completely. This system would, however, require sprockets and a different type track. Some mobility, without the tracks, would be maintained with the driven rear wheels.

d. If a suspension were selected using small road wheels, it would then not be possible to place motors in the wheels and all tractive effort would have to be applied at the sprockets. This can be accomplished by grouping several motors around a large gear which, in turn, would drive the sprockets. Mobility without tracks, however, would be lost.

e. To eliminate the difficulty of feeding electrical energy produced by the regenerative torque through the frequency converter, it is suggested that a mechanical means be used to transfer the regenerative torque by the use of a controlled differential, and the electrical system be used to apply the tractive force with one set of motors and the steering with a second set of motors. This system would simplify the electrical system by reducing the number of motors and reducing the complexity of the frequency changer and controls. This system is incorporated into a vehicle concept as shown on drawing 6496-21 and 6496-92.

SUSPENSION

Pneumatic Tires

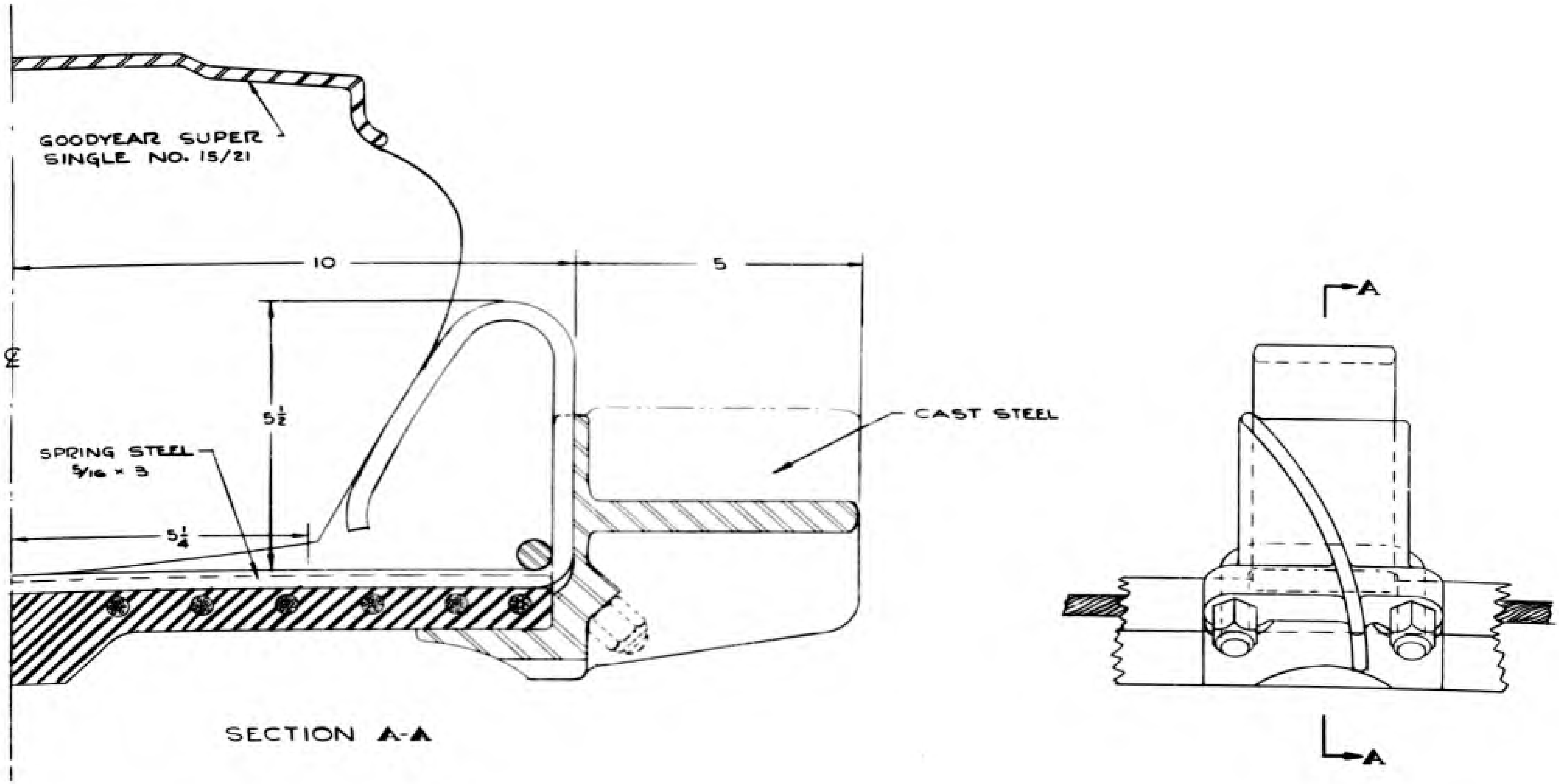
Initially, the use of pneumatic tires was considered because of the following advantages:

1. Inherent energy absorbing ability, possibly to the exclusion of mechanical springs.
2. Suitable vehicle mobility for limited use without tracks.
3. The possibility of an extremely light flexible band track running on large low pressure tires.

A first design of a tire and track is depicted in drawing 6496-25. The tire selected is a recently-developed design that would be made in steel cord construction. For reasonable track support, four tires per side were selected, giving a total tire and wheel weight of 2,008 pounds.

To utilize the energy absorbing ability of the tires, a beam suspension was used (see drawing 6496-1). A vertical shock at any wheel causes a reaction at the opposite end of the beam, hence both tires absorb the energy of the shock. Probable tire pressures and deflection would be 45 PSI and 3 inches, respectively. This arrangement of beam and tires would provide a dynamic deflection rate of 1,500 pounds per inch per tire.

Various schemes have been used in the past to provide some armor protection for tires, but all are unacceptable for either of two reasons; excessive tire heat build-up, or excessive weight.



PROPOSAL
BELT TRACK
37.5 #/FT.

9-19-61	F	F	
FULL	6496-25		

PACIFIC CAR AND FOUNDRY COMPANY

6496-25

A possible band track design for use on the pneumatic tires is shown on drawing 6496-25. This track embodies laterally flexible consturction in an effort to reduce track weight. A laterally rigid track would weigh more because as the lateral members are stiffened, their encountered loads increase, requiring weight and strength increases. The band track shown would weigh 34 pounds per foot.

Springs

The following analysis shows a logical selection of spring material, based on the lightest energy absorber criteria.

Aluminum, steel, and rubber were considered for the energy absorbing means in the suspension. Air springs were not seriously considered because of their inherent vulnerability.

	Aluminum	Steel	Rubber
Design Stress	23,000 PSI	140,000 *	200 PSI
Shear Modulus	3.9×10^6	12×10^6	250
Deflection of one cubic inch	.0059"	.0116"	.8"
Shear Force on one cubic inch @ Design Stress	23,000 PSI	140,000 PSI	200 PSI
Energy/in^3	136"#	1620"#	160"#
Wt/in^3	.10#/in^3	.284#/in^3	.06#/in^3
Energy/lb	1360"#/lb	5710"#/lb	2670"#/lb

* This value of shear stress is justified for torsion bars that have been pre-set

Steel is the best present material for a spring.

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