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LVTPX11 BASIC ENGINEERING STUDY

Final Engineering Report

Prepared by
Ingersoll Kalamazoo Division
Borg-Warner Corporation

in response to
Article 1, Item 1, Contract NObs 4561

29 November 1962

Volume I of IV

Republished 28 October 1963

FOREWORD

This basic Engineering Study is submitted in response to Item I of contract NObs 4561 and thus constitutes the preliminary design work for the LVTPX11 amphibian cargo and personnel carrier.

A classified report that outlines ballistic and atomic protection afforded by the LVTPX11 will be submitted under separate cover.

The initial series of one-fourth (1/4) scale self-propelled model tests, conducted by the University of Michigan, indicated a need to increase the propulsion efficiency of the track and to reduce hull drag. To this end, the necessary changes were made and a new series of tests was scheduled at the University of Michigan on 23 November 1962. The additional tests have been scheduled so that a final report will be published and delivered to the Bureau of Ships by 1 January 1963. For these reasons, Section 2 (Model Tests) has been intentionally omitted from this Basic Engineering Study.

Perusal of section 1 (Hull) discloses that the LVTPX11 is capable of mounting the 7.62 mm machine gun, the 50 caliber machine gun, and the Interim Vehicle Rapid Fire Weapons System. It should be noted that sources at ATAC express doubt that the Interim Vehicle Rapid Fire Weapons System will be developed. These sources believe that due to development lead time and other considerations, the Successor Vehicle Rapid Fire Weapons System will be developed and the Interim System dropped. To date, the Ingersoll Kalamazoo Division has been unable to obtain any information concerning the Successor Vehicle Rapid Fire Weapons System.

Attention is invited to the estimated cost of Item 2 contained in section 15. These figures are unchanged from the previous submission. Since detailed design of the LVTPX11 has not been completed, it is not possible to obtain firm price quotes from vendors, hence the original cost estimate is valid until detailed design is completed and firm price quotes are received from vendors.

To assist the reader in his perusal of this study, the appendices containing the preliminary design calculations are included immediately following the section to which they pertain. In addition, the full sized drawings are sequence-numbered (in addition to the identifying SK number).

To further assist the reader, all Ingersoll Kalamazoo Division recommendations are listed below under three categories: (1) recommendations for approval of items; (2) recommendations for changes to the specifications; and (3) recommended changes to the Government-Furnished Materials List. In each instance the appropriate paragraph of the text or specifications is referenced.

RECOMMENDATIONS FOR APPROVAL OF ITEMS

The Ingersoll Kalamazoo Division recommends that:

No further consideration be given to the incorporation of the Interim Vehicle Rapid Fire Weapons System into the LVTPX11 until a determination is made by ATAC as to whether the Interim or the Successor Vehicle Rapid Fire Weapons System is to be developed.

The molded-in-place rubber pad be approved for use in the LVTPX11 track (paragraph 5.5.2).

The combined unit (rim with two molded-in-place tires) be approved for use in the LVTPX11 (paragraph 5.2.2.7).

The riveted-in-place stainless steel wear strips be approved for use on the LVTPX11 roadwheels. (paragraph 5.2.3.3.).

The piston-type hydraulic motor, as depicted in drawing SK-5214, be approved for LVTPX11 bilge pump use (paragraph 8.5.3).

As the Bureau of Ships finishes engineering checks of components or systems, the Contractor be given approval to proceed into Item 2 for that component or system.

RECOMMENDATIONS FOR CHANGES TO THE SPECIFICATIONS

The Ingersoll Kalamazoo Division recommends that the below listed changes be incorporated in the BuShips Contract Specification SHIPS-A-4159 for the reasons stated:

Paragraph 3.12.1.5.1 (a) Towing Devices - be changed to read:

> "(a) Tow hitches are required fore and aft. The after hitch to be of the quick release type, operable by a crewman on the topside of the vehicle."

COMMENT. Paragraph 1.7 of the text lists hull attachments planned for the LVTPX11. In studying preliminary designs of a quick release bow towing hitch the contractor concluded that complexity, cost, and weight make this item impractical to incorporate.

Paragraph 3.12.2.5 Bilge System - be changed to read:

> "A Bilge system with two hydraulic motor-driven pumps of about 300 gpm capacity each in consonance with good marine practice shall be incorporated" (balance of paragraph to remain the same).

COMMENT. Paragraph 8.5.9 of the text contains the reasons for this recommendation.

Paragraph 3.12.2.9 <u>Fire Extinguishing System</u> - be changed to read:

> "Provision shall be made for a fixed system utilizing a five-pound CO_2 bottle and a portable system using a five-pound CO_2 bottle. The fixed system shall be operable from inside and outside the amphibian."

COMMENT. The reasons for this recommendation are stated in paragraphs 11.2.1.2 and 11.2.2.2. of the text.

Paragraph 3.12.1.8.1 <u>Tracks</u> - be changed to read:

> "the tracks shall be steel hinge - pin type of a design that minimizes road damage"

COMMENT. The reasons for this recommendation are stated in paragraph 5.5.1.4 of the text.

RECOMMENDED CHANGES TO THE "GOVERNMENT FURNISHED MATERIAL" LIST (CONTRACT NObs 4561, ARTICLE IX)

The Ingersoll Kalamazoo Division recommends the Government-Furnished Material List for the LVTPX11 be changed as follows:

Delete the words "with ground mount" from the second item.

COMMENT. The Marine Corps no longer requires that a ground mount be carried for the machine gun mounted on tactical vehicles.

Add the following:

To the first line item, the words "with antenna".

To the list, " Intercommunications Amplifier, AM-65/GRC quantity 6".

COMMENT. The radios provided should be complete with antennas and intercommunications amplifier. It is believed that these items were inadvertently omitted from the original GFM list.

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1.0. HULL

1.1. General Description.

1.1.1. An all-welded, aluminum hull has been designed for the LVTPX11 to provide a lightweight, rigid structure with watertight integrity. Hull plating thicknesses have been chosen to afford the required armor protection. The protection afforded with the armor selected for this vehicle has been previously submitted by letter.

1.1.2. The LVTPX11 is fitted with a bow ramp. A cargo hatch is provided in the top deck and crew escape hatches are positioned to facilitate emergency exit by personnel.

1.1.3. The hull envelope (see drawing SK-5180 for lines drawing) conforms to the dimensions established by the specifications, and the bow profile is shaped to provide a smooth transition into the bottom lines to minimize hydrodynamic resistance. A simple hull form is employed to avoid structural complexity, to minimize welding, and to generally facilitate construction.

1.1.4. The protection provided by the LVTPX11 against blast, thermal, and radiological effects of nuclear explosions will be forwarded under separate cover.

1.2. Material.

1.2.1. Aluminum has been selected as the primary structural material for the

LVTPX11 because of its light specific weight, its corrosion-resistant qualities, and its cutting and forming properties which facilitate fabrication.

1.2.2. Sheets, plates, shapes, and special extrusions used in the structure of the vehicle will be of aluminum alloy 5456, (MIL-A-19842B-1 and MIL-A-21170B). This alloy has been chosen because of its high strength, superior weldability, and resistance to corrosion. Structural and other castings will be aluminum alloy A356-T6, (MIL-C-21180B). Where the use of metal other than aluminum is unavoidable, stainless steel will be used to minimize galvanic action between the dissimilar materials.

1.3. Structural Geometry.

The vehicle shell plating thicknesses have been determined by armor protection requirements; consequently, the plating is considerably heavier than would otherwise be required to satisfy structural requirements. This condition makes feasible the use of a monocoque type of structural arrangement similar to aircraft, in which the shell plating provides the primary structural strength and a minimum amount of framing is added for the purpose of withstanding racking loads and restraining the plating to the intended hull shape. This structural philosophy has been chosen for the LVTPX11, because it offers several important features as follows:

A. Minimum hull weight for a given armor requirement.

B. Greater utilization of internal space due to a minimum of frame and gusset protrusions.

C. Minimum welding and consequently less heat distortion.

For structural design calculations see Appendix I immediately following this section. The structural arrangement is shown on drawings SK-5176, SK-5177, SK-5178, and SK-5179.

1.4. Hull Plating.

Aluminum hull plating thicknesses are shown on drawing SK-5194. A special extrusion is used along the sheer to join the top and side plates, and a similar extrusion is used to join the bow and side plates. This extrusion provides the necessary radius at these corners and facilitates alignment during assembly. Forged or cast spherical corner segments are used at the bow where the extruded members intersect. The cargo deck plating is intended to be removable in sections for access to the bilge. The deck plating is secured with flush-mounted, quick-acting fasteners. All joints are detailed in accordance with MIL-STD-21 Welded-Joint Designs, Armored Tank Type.

1.5. Framing.

1.5.1. Transverse Framing.

Over-all transverse rigidity and resistance to racking loads are provided by the following members:

A. Stern Plate.

B. Engine Compartment Bulkhead.

C. Cargo Hatch Aft End Beam.

D. Partial Bulkhead behind Driver's Seat and Cargo Hatch Forward End Beam.

E. Aft Bulkhead of Fuel Cells.

F. Ramp Opening Framing.

In addition, transverse support is provided to the hull bottom and cargo deck by tee members located between these surfaces. The track channel side plates are reinforced locally to support the roadwheel suspension arms.

1.5.2. Longitudinal Framing.

The longitudinal girder strength of this vehicle is provided by the shell plating. The heavy plate thicknesses and the high depth per length ratio of the hull insure adequate girder strength. The track channel side plates act as deep beams to carry the loads imposed by the roadwheels. Longitudinal framing is provided at the cargo hatch and engine compartment hatch sides. Hull openings are shown on drawing SK-5181.

1.6. Structural Details.

1.6.1. Ramp.

1.6.1.1. The bow ramp (see SK-5183) is an all-welded, watertight structure, that pivots on trunnion hinges, and is raised by cables attached to hydraulic cylinders. The clear opening width equals the cargo deck unobstructed width. The inner and outer ramp plates combine to provide the required armor protection. Framing consists of the side plates, two inside stringers, and a transverse member to prevent racking should it be necessary to close the ramp with one cable. Lightening holes will be cut in the inside framing to reduce weight, and the void spaces will be filled with polyurethane foam of low density. The ramp will be assembled by continuous welding of the end closure and side plates and intermittent welding of the inside framing to the inner plate; the outer plate will then be continuous fillet welded to the closure plates and plug welded to the inside framing.

1.6.1.2. A hinge arrangement has been developed to permit a close fitting joint between the ramp and hull bottom, in order to minimize hydrodynamic resistance. A watertight seal is provided all around the ramp by seating the inner plate against a compression type seal. Manually operated latches with a remote quick-release feature will be used to assure positive securing of the ramp. The aluminum hinge will be welded integral with the ramp and hull for maximum strength and rigidity. Nonlubricated, plastic-type bearing inserts will be fitted and rubber alignment bushings will be installed to simplify initial alignment and prevent subsequent binding.

1.6.2. Cupolas.

1.6.2.1. Cast aluminum, spherical cupolas are located over the driver's and troop commander's stations (see drawing SK-5184). Cupola details are similar to the LVTP5 design, with five removable vision blocks, a hinged cover with torsilastic counterbalance, and a locking device to secure the cover in an open position.

1.6.2.2. A cast aluminum cupola (see drawing SK-5185) is located on the vehicle centerline to accommodate the NATO 7.62 millimeter machine gun. Except for the adaptation to aluminum, this cupola is similar to its counterpart on the LVTP5. Provisions for mounting the .50 caliber machine gun as well as the interim vehicle rapid fire weapons system are shown on drawings SK-5187 and SK-5246, respectively. The possibility of fabricating these cupolas from a ballistic-type, transparent plastic is being investigated.

1.6.3. Hatches.

1.6.3.1. Cargo Hatch.

The top deck cargo hatch has a clear opening width equal to the cargo deck width and a length of 85 inches (see drawing SK-5182). A 3/4-inch high coaming is located around the opening to provide a knife edge for sealing. Folding hatch covers, counterbalanced with torsion bars are used. When open, the covers stand vertically outboard of the opening, so that the full hatch opening width is unobstructed. The inboard hatch cover corners are supported by rollers, retained in channel tracks, and either cover may be opened or closed independently of the other. Seals will be fitted to the hatch covers and will seal against the coaming edge to provide a watertight seal. Manually operated, vibrationproof, wedge-action latches, operable from inside and outside the vehicle, are used to secure the hatch covers.

1.6.3.2. Escape Hatches.

1.6.3.2.1. A crew access hatch is located in the top deck with an 18-inch by 24-inch clear opening. The hinged hatch cover is fitted with a wedge-action latch that secures the cover against a watertight seal. Steps fitted on the engine compartment bulkhead will facilitate access through this hatch.

1.6.3.2.2. An escape hatch is located on each side of the vehicle for emergency exit by embarked personnel (see drawing SK-5192). These hatches have a designed clear opening width of 26 inches square, and the covers will weigh approximately 110 pounds each, with hardware. The geometry of the latching mechanism and seal on the LVTP5 side escape hatches has proved satisfactory; hence, this arrangement will be adapted to aluminum for use on the LVTPX11.

1.6.3.3. Engine Hatch.

A hatch is provided in the top deck (see drawing SK-5188) for access to the engine compartment, and it is sized to permit removal of the engine assembly, complete with drop gearbox and transmission. A 3/4-inch high coaming is fitted around the opening to provide a knife edge for the sealing. The hatch covers seal together at the centerline and hinge outboard. The sealing and securing features are similar to the cargo hatch.

1.7. Hull Attachments.

The following hull attachments are shown on drawing SK-5192:

A. Lifting eyes (compatible with 17-ton lifting sling as shown on drawing SK-5070).

B. Mooring bitts.

C. Towing bridle for water tow (nylon).

D. Cargo holddowns (flush with cargo deck and rings welded to track channel sides).

E. Rear tow device with quick release from the top deck; capable of towing direct support artillery.

F. Front tow device.

G. Steel tow cable,one-inch diameter.

H. Boarding steps, self-cleaning and are located on both the port and starboard sides.

I. Safety rails and grab handles.

J. Crash pads.

K. Permanently attached portions of the litter kit.

1.8. Vision Devices.

The driver's cupola and crew chief/troop commander's cupola are provided with periscopes resulting in a total of 360-degree visibility. Vision blocks in the machine gun turret provide 360 degrees of visibility. Vision blocks are also placed in the side of the hull over each escape hatch. The combined field of vision is shown on drawing SK-5190.

1.9. Seating.

Twenty-nine fully equipped Marines and two crewmen can be seated in the personnel compartment as shown on drawing SK-5186. Quick folding, bench-type seats will be furnished that can be easily installed for troop seating when required.

1.10. On Equipment Material.

Brackets and mounting devices will be furnished for storage of OEM. Brackets will be furnished for mounting the radiac set, AN/PDR-27 and the M8A2 filter, unit, gas particulate (CBR) for operational use.

1.11. Weight, Trim, and Stability.

1.11.1. An extensive weight study and trim calculations have been made for the vehicle in the light condition, full troop load, and full cargo load. For these three conditions the weights, freeboard, and trims are:

	Weight Pounds	Freeboard Inches	Trim Degrees
Vehicle in the light condition	27,836	41.16	2°27' By the stern
Vehicle with full troop load	33,926	36.00	1°56' By the stern
Vehicle with an 8,000-pound load	35,836	34.20	0°46' By the stern

1.11.2. A breakdown of the weight, trim, and stability calculations are included in Appendix II immediately following this section.

1.11.3. The curves of form and cross curves are shown on drawing SK-5193. The vehicle has the capability of righting from a 110-degree roll port or starboard and is shown with the land stability on drawing SK-5191. The trim study is shown on drawing SK-5189.

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SUBJECT LVTPXII
Structural Calculations
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-ABBREVIATIONS & SYMBOLS-

C.G. = Center of Gravity

℄ = Center line

d = Distance - inches

E = Modulus of Elasticity - p.s.i.

fb = Flexural unit stress - p.s.i.

fc = Compression Unit Stress - p.s.i.

fs = Tensile Unit Stress - p.s.i.

K = Factor depending on the method of fixing the ends of member

$\frac{KL}{r}$ = Slenderness ratio

L = Length-inches

M = Bending Moment -- inch. lbs.

P = Concentrated load --- lbs.

PL = Plate

r = Least radius of gyration - inches

S.F. = Safety factor

w = Unit load --- p.s.i.

W = Total load ... lbs.

$\bar{y}$ = Distance from center of gravity of an element of section to axis of reference ---- inches

$\bar{Y}$ = Distance from center of gravity of a section to axis of reference --- inches.

Y = Distance from center of gravity of an element to center of gravity of a section --- inches.

Z = Section Modulus = $\frac{M}{fs}$ - in.^{3}

I_o = Moment of Inertia of a section about its neutral axis - in.^{4}

I_T = Total moment of inertia ---- in.^{4}

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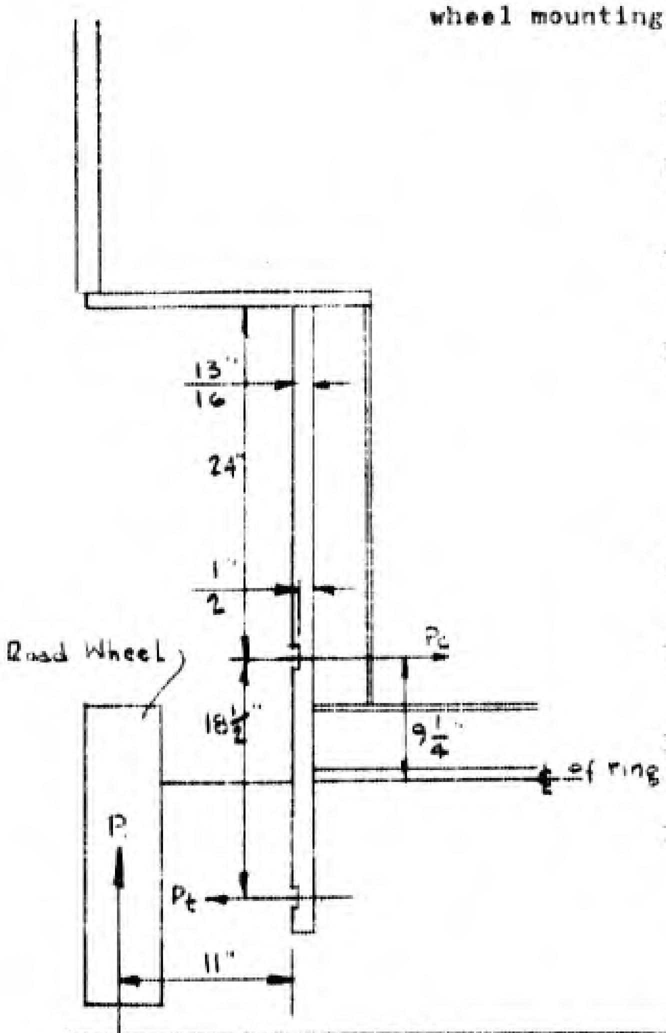
Struct. Calc.

1) **Track Channel Side Plates**

a) Material = $\frac{13}{16}$ Aluminum plate

b) Load = 35,000 lbs. applied at road wheel mounted on plate - Plate will be machined down to $\frac{1}{2}$" to accomodate wheel mounting ring.



$P = 35{,}000\#$ $M_t = 35{,}000 \times 11" = 385{,}000"\#$

$P_c = P_t = \frac{385{,}000}{18.5} = 20{,}750\#$

$M = \frac{Pab^2}{L^2}$ (Alcoa Structural Handbook Page 204 - Case 11)

$M_b = \frac{20{,}750 \times 9.25 \times (24-9.25)^2}{(24)^2}$

$= \frac{20{,}750 \times 9.25 \times 14.75^2}{(24)^2} = 72{,}600"\#$

$Z \text{ req'd} = \frac{72{,}600}{51{,}000} = 1.42 \text{ in.}^3$

Try $4\frac{1}{2} \times 3 \times \frac{5}{16}$ "T"

H. thickness = $\frac{1}{2}$

Effect. Width = $30 \times \frac{1}{2}$ (Roark's page 245-Art.64)

= 15.0 in.

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
"T"	4-1/2x3x5/16	2.52	2.79	7.03	1.89	3.57	9.00	1.78
	15.0 x 1/2	7.50	0.25	2.12	0.65	0.42	3.18	0.16
		10.02		9.15			12.18	1.94

$I_t = 14.12 \text{ in}^4$

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Structural Calculations

$\bar{Y} = \frac{9.15}{10.02} = .90$ in. Z prov. $= \frac{14.12}{(3.62-.90)} = \frac{14.12}{2.72} = 5.20$ in.^{3}

This member is in addition subjected to a vertical axial load = 35,000#, in compression.

$$\frac{fc}{fC} + \frac{fb}{fB\left(1-\frac{fc}{fCE}\right)} \leq 1$$

(Page 62 Reynolds' structural aluminum design handbook)

where

fc = applied axial stress = $\frac{P}{A}$ ................P.S.I.

fC = ultimate strength under axial load alone....P.S.I.

fb = applied bending stress = $\frac{M}{Z}$ ................P.S.I.

fB = ultimate strength under bending alone.......P.S.I.

fCE = ultimate Euler load for buckling in direction of the applied bending moment...............................P.S.I.

fc = $\frac{35,000}{10.02}$ = 3,330#/in^2

$\frac{KL}{r}$ = slenderness ratio (Page 32 - Reynolds' Structural Aluminum Design Handbook)

where

K = factor depending on method of fixing of ends of member= .5

L = Length of Compression member

r = least radius of gyrotion

$\frac{KL}{r} = \frac{.5 \times 24}{\sqrt{\frac{14.12}{10.02}}} = \frac{12}{1.18} = 10.25 \quad 22 \therefore$ Compact Member (Page 105- Reynolds' Structural Aluminum Design Handbook)

fc = 27,000 #/in^2

fb = $\frac{72,600}{5.20}$ = 14,000 #/in^2

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Structural Calculations

$fB = 51{,}000\ \#/in^2$

$fCE = \frac{102{,}000{,}000}{(10.25)^2} = 970{,}000\ \#/in^2$

$$\frac{fc}{fC} + \frac{fb}{fB(1-\frac{fc}{fCE})} = \frac{3{,}330}{27{,}000} + \frac{14{,}000}{51{,}000\ (1-\frac{3330}{970{,}000})}$$

$$= .12 + \frac{14{,}000}{51{,}000} = .12 + .28 = \underline{0.40<1}$$

2) <u>Hull Bottom Tranverse Member</u>

Assume hull bottom PL is $\frac{1}{2}$ " thick.

Try a $4"\times4"\times\frac{3}{8}"$ "T" as main structural support

Eff. Width of PL $= 30 \times t$ (Roark's Formulas for Stress and Strain Page 245 = Art 64)

$= 30 \times \frac{1}{2}$

$= 15$

$\bar{Y} = \frac{35.43}{10.68} = 3.32$ in.

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
T	4"x4"x3/8"	3.18	1.11	3.53	2.21	4.89	15.55	4.56
	15"x1/2"	7.50	4.25	31.90	0.93	0.86	6.48	0.16
		10.68		35.43			22.03	4.72

$I_T = 26.75\ in.^4$

$Z\ prov. = \frac{26.75}{3.32} = \underline{8.06}\ in.^3$

Allow M. = 8.06 x 23,500 = 189,500 in.#

$M = \frac{WL^2}{10}$ (semi-fixed ends - Case 13, page 102 and Case 33, page 108)

$M = \frac{WL^2}{10}$ $\therefore$ $W = \frac{10M}{L^2} = \frac{10 \times 189,500}{(78.5)^2} = 308$ #/in.

Unit load $w = \frac{W}{\text{Spacing}} = \frac{308}{44} = \underline{\underline{7.00}}$ #/in^2 (Allowable load on this member)

7.00 #/in^2 = $\underline{\underline{14.00}}$ feet of water (This corresponds to a static head)

Try a 3" x 3" x 5/16" "T" in engine compartment

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
T	3 x 3 x 5/16	1.92	2.65	5.09	1.91	3.65	7.00	1.58
▭	15 x 1/2	7.50	0.25	1.88	0.49	0.24	1.80	0.16
		9.42		6.97			8.80	1.74

$I_T = 10.54$ in^4

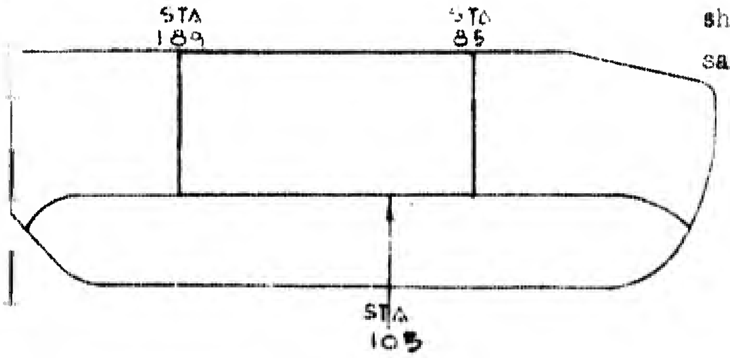
$\bar{Y} = \frac{6.97}{9.42} = .74$ in. Z prov. $= \frac{10.54}{2.76} = \underline{\underline{3.82}}$

Allow. M= 3.82 x 23,500 = 90,000 "#

$W = \frac{10 \times 90,000}{(82)^2} = 134$#/in $W = \frac{134}{21} = 6.38$ #/in^2 (This corresponds to a static head)

3. Track Channel Framing

Assume the track channel side plate as being fixed @ Sta's. 85 and 189



The load P, from one wheel is applied as shown. Local stresses are analyzed and satisfied as shown in Sht.

BY R.B. DATE SUBJECT LVTPX11 Struct. Calc. SHEET NO. 5 OF 28
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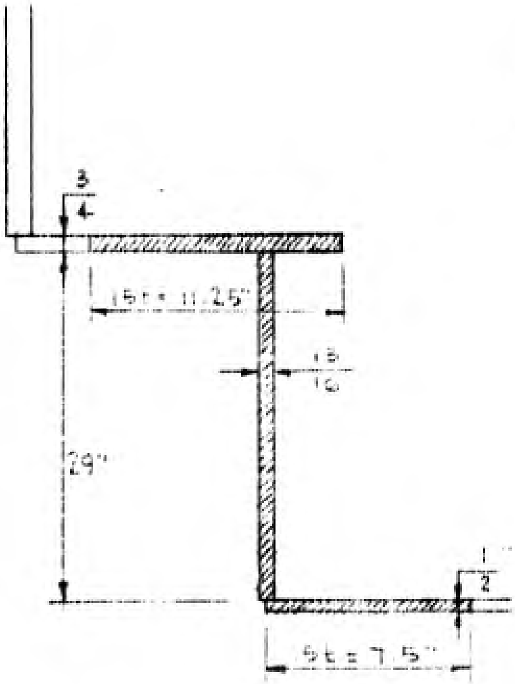

The track channel can be analyzed, as a whole, as a "Z" member - Size of elements is shown @ left.

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
▭	11.25x3/4	8.45	29.88	252.50	12.83	164.50	1390.00	1.58
▯	29x13/16	23.50	15.00	353.00	2.05	4.02	98.90	1650.00
▭	7.5x1/2	3.75	0.25	1.88	16.80	282.10	1060.00	0.08
		35.70		607.38			2548.90	1651.66

$\bar{Y} = \frac{607.38}{35.70} = 17.05$ in

$I_T = 4,300.56\ in^4$

$Z\ act. = \frac{4300.56}{17.05} = 253.0\ in^3$

$M\ act. = \frac{Pa^2 b}{L^2} = \frac{35,000 \times 84^2 \times 20}{105^2}$ (Alcoa Structural Handbook page 204 - Case 11)

$= 447,000$ in.#

$Z\ rq'd = \frac{447,000}{51,000} = \underline{8.78\ in^3}$

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4) Forward Side Plate

Plate thickness = 1-3/16"
Plate height = 46" = a
Plate length = 104 (between sta. 85 & 189) = b

$b/a = 104/46 = 2+$

from Roark's Case 70, page 213

b/a	2.0
B_1	.1908
B_2	.0356
B_3	.3000
B_4	.1709
B_5	.1148

$$\text{Max } S_a = B_3 \frac{wa^2}{t^2} \text{ @ } x = 0;\ y = a$$

$$= \frac{.300 \times 1.84 \times 46^2}{\overline{1.12}^2} = \underline{\underline{925}} \ \#/in^2$$

$$\text{Max } S_a = B_4 \frac{wa^2}{t^2} \text{ @ } x = 0;\ y = 0$$

$$= \frac{.1709 \times 1.84 \times 46^2}{\overline{1.12}^2} = \underline{\underline{526}} \ \#/in^2$$

$$\text{Max } S_a = B_5 \frac{wa^2}{t^2} \text{ @ } x = 0;\ y = .6a$$

$$= \frac{.1148 \times 1.84 \times 46^2}{\overline{1.12}^2} = \underline{\underline{252}} \ \#/in^2$$

w = Unit load

$$w = 46" \times \frac{.04\#}{in^3} = 1.84 \ \#/in^2$$

$$\text{Max } S_b = B_1 \frac{wa^2}{t^2} \text{ @ } x = 1/2\ b$$

$$y = .55a$$

$$= \frac{.1908 \times 1.84 \times \overline{46}^2}{\overline{1.12}^2}$$

$$= \underline{\underline{590 \ \#/in^2}}$$

$$\text{Max } S_b = B_2 \frac{wa^2}{t^2} \text{ @ } x = 0;\ y = .6a$$

$$= \frac{.0356 \times 1.84 \times \overline{46}^2}{\overline{1.12}^2}$$

$$= \underline{\underline{110 \ \#/in^2}}$$

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Max. Allowable Unif. load

From Roark's Case 41, page 205

$$\text{Max } S_b \text{ (@ center of long span)} = \frac{.5wb^2}{t^2(1 + .623\alpha^6)} \qquad \alpha = \frac{a}{b} = \frac{46}{104}$$

$$\alpha = .44$$

$$\therefore\ w = \frac{S \times t^2(1 + .623\alpha^6)}{.5b^2}$$

$$= \frac{23{,}500\ (1.18)^2\left[1 + (.623)\ (.44)^6\right]}{.5 \times (104)^2}$$

$$= \frac{32{,}750\ (1 + .021)}{5{,}400} = \underline{6.19\ \#/in^2}$$

$$\text{Max } S_a \text{ (@ center of short edges)} = \frac{.25wb^2}{t^2} \qquad \therefore\ w = \frac{S\ t^2}{.25\ b^2}$$

$$w = \frac{23{,}500 \times \overline{1.18}^2}{.25 \times \overline{104}^2} = \underline{\underline{12.10\ \#/in^2}}$$

$$S_b \text{ (@ center)} = -\frac{.75wb^2}{t^2(3 + 4\alpha^4)} \qquad \therefore\ w = \frac{S_b\left[t^2(3 + 4\alpha^4)\right]}{.75\ b^2}$$

$$w = \frac{23{,}500\ \overline{1.18}^2\ (3 + 4 \times .07)}{.75 \times \overline{104}^2} = \frac{23{,}500\ (4.63)}{8{,}120} = \underline{\underline{13.25\ \#/in^2}}$$

$$S_a \text{ (@ center)} = .054\ wb^2\ \frac{(1 + 2\alpha^2 - \alpha^4)}{t^2} \qquad \therefore\ w = \frac{S_a\ t^2}{.054b^2\ (1 + 2\alpha^2 - \alpha^4)}$$

$$w = \frac{23{,}500\ (1.18)^2}{.054 \times \overline{104}^2\left[1 + 2\ (.44)^2 - (.44)^4\right]}$$

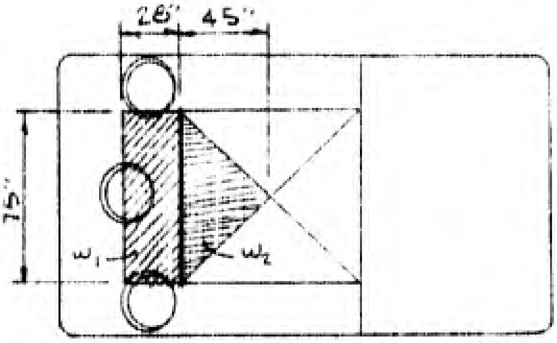
$$= \frac{32{,}750}{585\ (1 + .52 - .07)} = \underline{\underline{38.6\ \#/in^2}}$$

5) Top Member

This member will be designed to support the top plate as well as the cargo hatch.

$Mw_1 = \frac{WL^2}{12}$ (Roark's Case 33, page 108)

$Mw_1 = \frac{wL^2}{12} = \frac{140 \times 75^2}{12} = 65{,}550$ "#

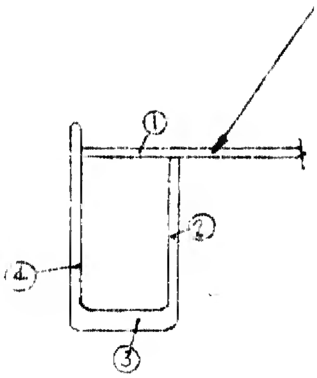


$Mw_2 = \frac{WL}{10} = \frac{8440 \times 75}{10} = \underline{63{,}200}$ "#

$M_T = 128{,}750$ "#

$Z \text{ rq'd} = \frac{128{,}750}{23{,}500} = \underline{5.47} \text{ in}^3$

Eff. width = 15 x 3/4 = 11.25 in



$w_1 = 5 \times 28$

$= 140$ #/in

$W_2 = \frac{5 \times 75 \times 45}{2}$

$= 8440$#

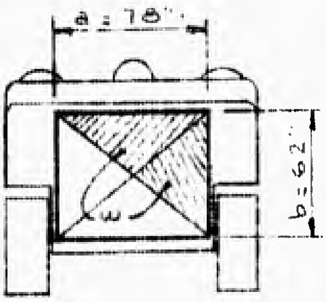
$Y = \frac{36.25}{10.99} = 3.30$ in

$Z \text{ prov.} = \frac{17.66}{3.30} = \underline{\underline{5.35}} \text{ in}^3$

Elem.	Dim	Area	y	$\bar{Ay}$	Y	Y^2	AY^2	Io
(1)	11-1/4x3/4	8.44	3.88	32.70	0.58	0.34	2.84	0.53
(2)	3-1/8x3/16	0.59	1.94	1.14	1.36	1.85	1.09	0.48
(3)	3x3/8	1.12	0.19	0.21	3.11	9.70	10.90	0.01
(4)	4-1/2x3/16	0.84	2.62	2.20	0.68	0.46	0.39	1.42
		10.99		36.25			15.22	2.44

$I_T = \underline{\underline{17.66}} \text{ in}^4$

6) Ramp Framing



w = 5 p.s.i.

Member a

$$M = \frac{WL}{8} = \frac{5 \times 78 \times 31 \times}{2}\frac{78}{8} = 58{,}750 \text{ "\#}$$

$$Z \text{ rq'd} = \frac{58{,}750}{23{,}500} = 2.50 \text{ in}^3$$

Member b

$$M = \frac{5 \times 62 \times 39}{2} \times \frac{62}{8} = 46{,}800 \text{ "\#}$$

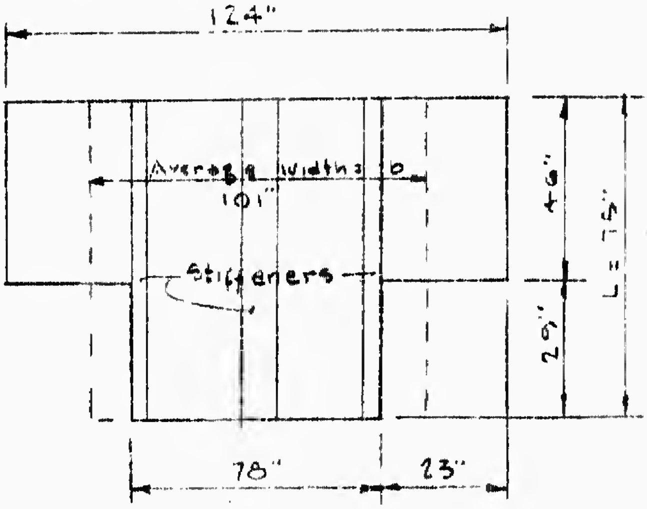
Use 4 x 2-1/2 x .318 Channel - Z = 3.42

7) Transverse Bulkhead @ Engine Compartment

Material = 1/2" Aluminum Plate

This plate is subjected to a load = 5 x 34 = 170 #/in and will be loaded in edge compression

Assume average width b = 101"



$$\frac{KL}{r} = k\frac{L}{t}$$ (Alcoa structural Handbook page 131)

$$\frac{L}{b} = \frac{75}{101} = .74$$

∴ k = 1.82 (Interpolating from ALCOA Structural Handbook page 132, Table 19a)

$$\frac{KL}{r} = 1.82 \times \frac{75}{.5} = 273$$

∴ This plate is a long column

$$fc.\ allow = \frac{102{,}000{,}000}{(KL/r)^2}$$ (ALCOA Structural Handbook page 111)

$$fc.\ allow = \frac{102{,}000{,}000}{(273)^2} = 1358 \text{ \#/in}^2$$

$$fc.\ act. = \frac{170 \times 101}{101 \times .5} = \underline{\underline{340 \text{ \#/in}^2}}$$

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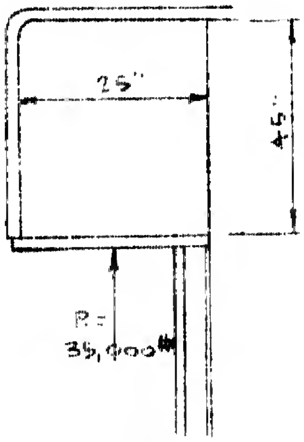
Even though this figure is low enough as compared w/the allowable stress, and provides a safety factor of $\frac{1358}{340} = 4$, stiffeners will be added @ the inner corners of the plates, primarily to support the access hatches.

8) <u>Transverse Bulkhead Aft. of Driver</u>

Assume this plate under an edge compression of 35,000 lbs. from road wheels.
Assume plate thickness = 3/16"



From ALCOA Structural Handbook pages 131 and 132

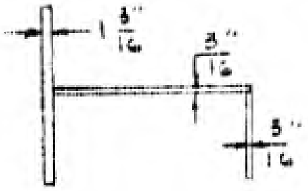
$L/b = \frac{45}{25} = 1.8$ $\qquad KL/r = k^1\ b/t\ ;\ k^1 = 1.16$

$kb/t = \frac{1.16 \times 25}{.188} = 155 > 79$ ∴ Long Column

$fc\ (\text{allow}) = \frac{102,000,000}{(155)^2} = 4,250\ \#/in^2$

$fc\ (\text{act}) = \frac{35,000}{25 \times .188} = 7,450\ \#/in^2$

Analyze ℞ as shown in figure below



30t = 30 x 1-3/16 = 35.6 (Roark's Formulas for stress and strain, page 245 - Art 64)

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
▯	35.6x1-3/16	42.30	0.56	21.30	1.52	2.31	27.20	4460.0
▬	25x3/16	4.70	13.62	64.00	11.54	133.00	625.94	-------
⌊	2.83x3/16	0.53	26.28	13.95	24.20	585.50	311.00	0.35
		47.53		99.25			1033.90	4460.35

$\bar{Y} = \frac{99.25}{47.53} = 2.08$ in

$I_T = 5,494.25$ in^4

$r = \sqrt{\frac{5494.25}{47.53}} = 10.75$ in

$K = .5 \quad \frac{KL}{r} = .5 \times 45 = 2.09 < 20$ $\therefore$ Compact member

fc. act. $= \frac{35,000}{47.53} = 736$ #/in^2

9) Engine Bulkhead Stiffeners

See Sht. # 5 for sketch

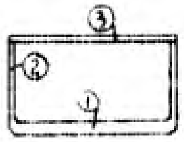
Vertical Members

Side Members

$P =$ Load $= \frac{38 \times (27 + 19)}{2} \times 5 = 4360$# (Contributory load from top deck)

Length = 39 in

A rq'd $= \frac{4360}{20,000} = 0.22$ in^2



Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
(1)	3 x 3/8	1.12	0.19	0.21	0.75	0.56	0.63	0.01
(2)x2	1-1/8x3/16	0.42	0.94	0.39	----	----	----	0.04
(3)	3x3/8	1.12	1.69	1.90	0.75	0.56	0.63	0.01
		2.66		2.50			1.26	0.06

$I_T = 1.32$ in^4

$\bar{Y} = \frac{2.50}{2.66} = 0.94$ in

$r = \sqrt{\frac{1.32}{2.66}} = 0.70$ in

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$\frac{KL}{r} = \frac{.75 \times 39}{.70} = 41.7 > 22$ $\therefore$ Short Column

fc. allow = 25.3 -.153 (41.7) (Reynold's Design Handbook page 105)

$= 25{,}300 - 6400$

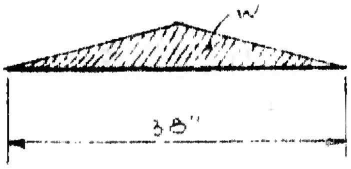
$= \underline{\underline{18{,}900 \text{ P.S.I.}}}$

fc. act. = P/A

fc. act. $= \frac{4360}{2.66} = \underline{\underline{1{,}640 \text{ P.S.I.}}}$

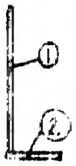
Top Member

$w = \frac{38 \times (27 + 19)}{2} (5) = 4{,}360\#$ (Contributory load from top deck)



$M = \frac{WL}{8}$

Z rq'd $= \frac{20{,}700}{23{,}500} = \underline{\underline{0.88 \text{ in}^3}}$



Elem.	Dim.	Area	y	Ay	Y	Y^2	AY^2	Io
(1)	$6\frac{1}{2} \times \frac{1}{2}$	3.25	3.44	11.18	0.49	0.24	0.78	22.90
(2)	3x3/16	0.56	0.09	00.05	2.86	8.19	4.58	-----
		3.81		11.23			5.36	22.90

$I_T = 28.26 \text{ in}^4$

$\overline{Y} = \frac{11.23}{3.81} = 2.95$ in

Z act. $= \frac{28.26}{(6.69 - 2.95)} = \frac{28.26}{3.74}$

$= \underline{\underline{7.56 \text{ in}^3}}$

Center Member

Load = 5 (38 x 39 x .75 + 53 x 52 x .75 + 12 x 20 x .75) = 3.75 (1482 + 2760 + 240)

= 3.75 (4482)

= 16,820#

Length 39 in.

A rq'd $= \frac{16{,}820}{20{,}000} = .84 \text{ in}^2$ $\frac{KL}{r} = \frac{.75 \times 39}{.83} = 35.4$

fc. actual $= \frac{16{,}820}{3.01} = \underline{\underline{5{,}595 \text{ p.s.i.}}}$

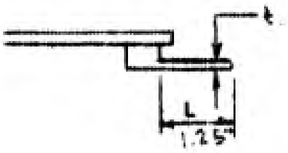
The same section as in the side member could be used here, however, this member will be made 6" wide to accommodate access steps and the two door latches.

10) Personnel Escape Hatch

Coaming

Assuming 5 p.s.i. and a hatch dimension of 30"

load/in = 15 x 5 = 75 #/in



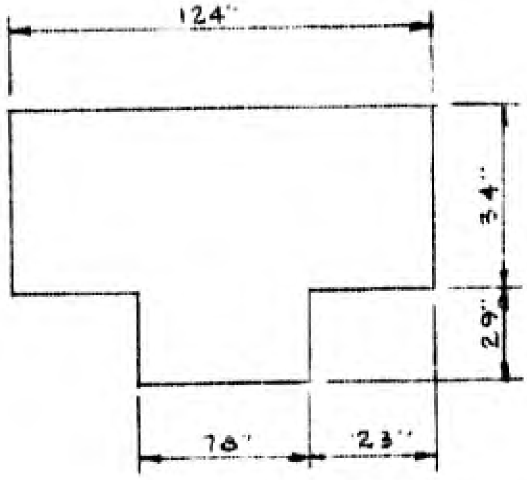
$M = \frac{wl^2}{2} = \frac{75 \times \overline{1.25}^2}{2} = 58.6$ in # (Roark's Case #3 - page 100)

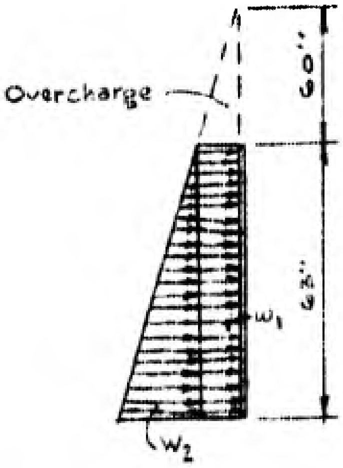
Z rq'd $= \frac{58.6}{23{,}500} = .002 = \frac{bt^2}{6}$ where $b = 1"$

t rq'd $= \sqrt{\frac{6 \times .002}{1}} = \quad .012 \quad = \underline{.11 \text{ inch}}$

11) Stern Sheet

Assume plate thickness = 11/16"





Assuming an overcharge = 5' of water to allow for increase of pressure due to backing speed in water or submersion in water due to waves action, this will produce a maximum pressure @ bottom of plate = 5.25 p.s.i. and @ top of plate = 2.5 p.s.i.

Use for plate width the average = $\frac{124 + 78}{2}$ = 101 in.

$\alpha = \frac{63}{101} = .61$

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$$f_{Sw_1} = \frac{-.75\ wb^2}{t^2\ (3 + 4\ \alpha^4)} = \frac{.75 \times 2.5 \times \overline{63}^2}{.47\ (3.56)} = 4{,}450 \text{ p.s.i.}$$ (Roark's Formulas for Stress and Strain - Case #41 - page 205)

$$f_{Sw_2} = \frac{B_3\ wb^2}{t^2}$$ (Roark's Case 70 - page 213)

$$b/a = \frac{101}{63} = 1.64 \quad \therefore \quad B_3 = .309$$

$$f_{Su_2} = \frac{.309 \times 2.75 \times \overline{63}^2}{.47} = \underline{\underline{7{,}160 \text{ p.s.i.}}}$$ (Simple Supp.)

$$f_S \text{ Total} = 4{,}450 + 7{,}160 \quad \underline{11{,}610 \text{ p.s.i.}}$$

This unit stress is well within allowable value, however, a stiffener will be added @ track channel height, mainly to support the engine.

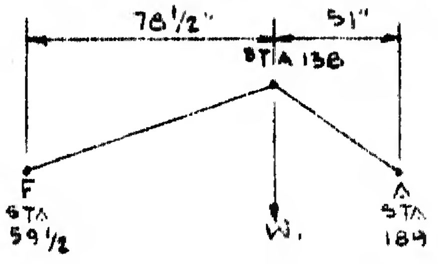
12) **Lifting Eyes**

Assume C. G. @ Sta. 138.

Lifting Eyes will be located @ Sta. 59-1/2 & 189.

Wt. = 35,000# S.F. = 6.

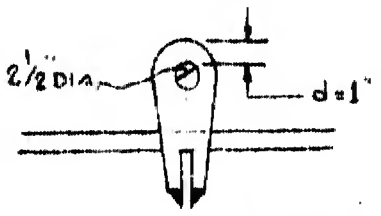
Tot. Wt. = 35,000 x 6 = 210,000#



$$R_F = \frac{51}{129.5} \times 210{,}000 = 82{,}600\#$$

$$P_A = \frac{78.5}{129.5} \times 210{,}000 = \underline{127{,}400\#}$$

210,000#



2 Lifting Eyes/side

$$A \text{ rq'd } 1 \text{ Lifting Eye} = \frac{P}{f_s} = \frac{127{,}400}{2 \times 30{,}000} = 2.12 \text{ in}^2$$

$$\text{Thickness rq'd} = \frac{A}{d} = \frac{2.12}{2 \times 1} = 1.06 \text{ in}$$

$$\text{Length of } 3/4'' \text{ weld rq'd} = \frac{127{,}000}{2 \times 11{,}000} = \underline{5.77} \text{ in}$$

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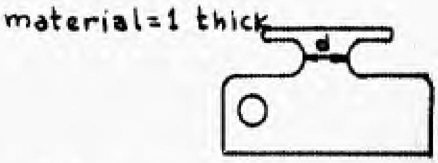
13) Mooring Bits



Assume a load = 40,000#

$A \text{ rq'd} = \frac{P}{fs} = \frac{40,000}{15,400} = 2.6 \text{ in}^2$

$d = \frac{2.6}{t} = \frac{2.6}{1} = 2.6 \text{ in}$

14) Stern Towing Hook

Assume 1" dia. rope - 6 x 37 - Breaking Strength = 100,000#

Using a safety factor of 3.5 against ultimate strength, the load to be used for hook and structure design will be equal = 125,000#

Length of 1/2" weld rq'd = $\frac{125,000}{7,600} = \underline{16.45}$ in.

Size of Pin rq'd = $\frac{P}{fs} = \frac{125,000}{2 \times 30,000} = 2.08 \text{ in}^2$

Dia. rq'd = $\sqrt{\frac{4 \times 2.08}{\pi}} = \sqrt{2.65} = \underline{1.63}$ in

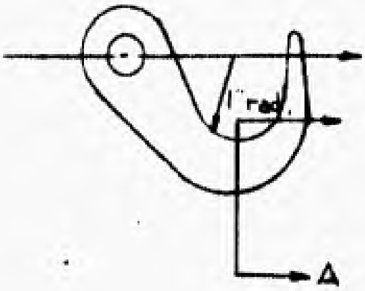
Use 1-3/4"

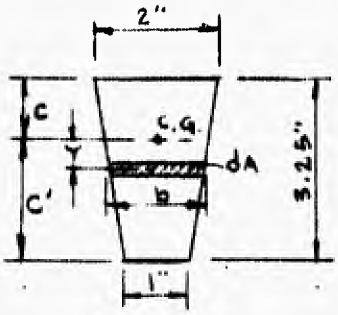
Use 1" thick ℞ for mount.

Edge distance rq'd = $\frac{125,000}{2 \times 30,000 \times 1 \times 2} = 1.04$ in.

Hook

Design as outlined by Max M. Frocht on strength of materials, page 370





Section A

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$$c^1 = \frac{3.25}{3}\left(\frac{2 \times 2 + 1}{2+1}\right) = \frac{3.25}{3}\left(\frac{5}{3}\right) = 1.81 \text{ in.}$$

$$C = \underline{1.44} \text{ in.} \qquad \therefore R = 1 + 1.44 = 2.44$$

$$b = 2 - \frac{(y + 1.44)}{3.25} = \frac{6.50 - y - 1.44}{3.25} = \frac{5.06 - y}{3.25}$$

$$\int \frac{da}{R+y} = \int \frac{b\,dy}{R+y} = \frac{1}{3.25} \int \frac{(5.06-y)\,dy}{2.44+y}$$

$$= \frac{1}{3.25}\left[\int_{-1.44}^{+1.81} \frac{5.06\,dy}{2.44+y} - \int_{-1.44}^{+1.81} \frac{y\,dy}{2.44+y}\right] = \frac{1}{3.25}\left\{\Big[5.06 Pn(2.44+y)\Big]_{-1.44}^{+1.81} - \left[\frac{y}{1} - \frac{2.44}{1}\ln(2.44+y)\right]_{-1.44}^{+1.81}\right\}$$

$$= \frac{1}{3.25}\left(\left\{5.06\left[\ln(2.44+1.81) - \ln(2.44-1.44)\right]\right\} - \left\{\left[1.81 - 2.44\ln(2.44+1.81)\right] - \left[-1.44 - 2.44\ln(2.44-1.44)\right]\right\}\right)$$

$$= \frac{1}{3.25}\left(\left\{5.06\left[\ln(4.25) - \ln(1)\right]\right\} - \left\{\left[1.81 - 2.44\ln(4.25)\right] - \left[-1.44 - 2.44\ln(1)\right]\right\}\right)$$

$$\int \frac{da}{R+y} = \frac{1}{3.25}\left[\left\{5.06\left[1.45 - 0\right]\right\} - \left\{\left[1.81 - 2.44(1.45)\right] - \left[-1.44 - 2.44(0)\right]\right\}\right]$$

$$= \frac{1}{3.25}\left[\left\{7.33\right\} - \left\{\left[1.81 - 3.52\right] - \left[-1.44\right]\right\}\right]$$

$$= \frac{1}{3.25}(7.33 + 1.71 + 1.44)$$

$$= \frac{10.48}{3.25} = \underline{3.22}$$

$$A = \frac{1}{2}\, 3.25\,(2+1) = \frac{1}{2} \times 9.75 = \underline{4.88} \text{ in}^2$$

$$P = 125{,}000\#$$

$$M = -125{,}000 \times 2.44 = \underline{-305{,}000''\#}$$

$$M = h\left[A - R\int \frac{dA}{R+y}\right]$$

$$h = \frac{M}{A - R\int \frac{da}{R+y}} = \frac{-305{,}000}{4.88 - 2.44(3.22)} = \frac{-305{,}000}{4.88 - 7.85} = \frac{-305{,}000}{-2.96}$$

$$= 102{,}800$$

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$$P = RA + h \int \frac{dA}{R+y} \qquad \therefore \frac{1}{A}\left(P - h \int \frac{dA}{R+y}\right) = R$$

$$R = \frac{1}{4.88}\left[125{,}000 - 102{,}800\ (3.22)\right] = \frac{1}{4.88}\ (125{,}000 - 331{,}000)$$

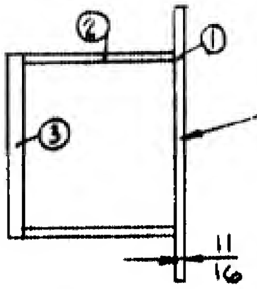
$$= \frac{-206{,}000}{4.88} = -42{,}200$$

$$fs = R + \frac{h}{R+y} = -42{,}200 + \frac{102{,}800}{2.44 - 1.44} = -42{,}200 + \frac{102{,}800}{1}$$

$$= -42{,}000 + 102{,}800 = \underline{\underline{60{,}600}}\ \text{p.s.i. (@ inner surface)}$$

$$fs = -42{,}200 + \frac{102{,}800}{2.44 + 1.81} = \underline{\underline{-18{,}000}}\ \text{p.s.i. (@ outer surface)}$$

Structure



Width = 30 x .69 = 20.7 (Roark's page 245, Art. 64)

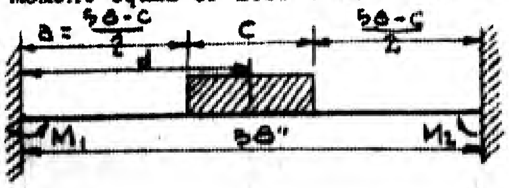
$$\bar{Y} = \frac{26.90}{15.84} = 1.70\ \text{in.}$$

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
(1)	20.7 x .69	14.30	1.84	26.30	0.14	0.02	0.28	0.57
(2) x 2	1.12 x 3/16	0.42	0.94	0.39	0.76	0.58	0.24	0.03
(3)	3 x 3/8	1.12	0.19	0.21	1.51	2.28	2.56	0.01
		15.84		26.90			3.08	0.61

$$Z = \frac{3.69}{1.70} = 2.17\ \text{in}^3 \qquad I_t = 3.69\ \text{in}^4$$

M allow = 2.17 x 51,000 = 111,000 in.#

The load must be distributed over a length "C" as shown in figure below to create a maximum moment equal or less than allowable



$$M_1 = M_2 = \frac{W}{24L}\left(\frac{24d^3}{L} - 6\frac{bc^2}{L} + 3\frac{c^3}{L} + 4c^2 - 24d^2\right)$$

(From Roark's Case 34, page 108)

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$b = a + c$

$d = L - a/2 - b/2$ $\qquad d = 58 - \frac{58-c}{4} - \frac{58+c}{4}$

$b = \frac{58-c}{2} + c$ $\qquad d = \frac{232-58-c-58+c}{4}$

$b = \frac{58+c}{2}$ $\qquad d = 29$

$$M_1 = \frac{W}{24x58}\left[24 \times \frac{\overline{29}^{3}}{58} - 6x\left(\frac{58+c)c^2}{2 \times 58}\right. + \frac{3c^3}{58} + 4c^2 - 24(29)^2\right]$$

$$= \frac{125,000}{1368}\left[10,100 - 3c^2 - \frac{3c^3}{58} + \frac{3c^3}{58} + 4c^2 - 20,200\right]$$

$$= 91.5\,(-10,100 + c^2)$$

$$-111,100 = -925,000 + 91.5c^2$$

$c^2 = \frac{814,000}{91.5} = \therefore c = \underline{\underline{94.3\text{in}}}$. (too long)

Change dimensions as shown.

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
(1)	20.7 x .69	14.30	2.84	40.55	0.49	0.24	3.44	0.57
(2) x 2	2 x 3/16	0.75	1.50	1.13	0.85	0.72	0.54	0.25
(3)	6 x 1/2	3.00	0.25	0.75	2.10	4.42	13.25	0.06
		18.05		42.43			17.23	0.88

$I_T = 18.11 \text{ in}^4$

$\bar{Y} = \frac{42.43}{18.05} = \underline{\underline{2.35}}$ in. $\qquad Z = \frac{18.11}{2.35} = 7.7$

Mallow = 7.7 x 51,000 = 393,000 in#

$$-393,000 = -925,000 + 91.5c^2$$

$\therefore c^2 = \frac{532,000}{91.5} = 5800$ $\qquad \therefore c = \underline{76 \text{ in}}$

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Change dimensions as shown

$\bar{Y} = \frac{97.70}{18.36} = 5.32$ in.

$Z = \frac{81.06}{5.32} = 15.25 \text{ in}^3$

Elem.	Dim.	Area	$\bar{Y}$	$A\bar{Y}$	Y	Y^2	AY^2	Io
(1)	20.7 x .69	14.30	6.34	90.50	1.02	1.04	14.90	0.57
(2) x 2	5.5 x 3/16	2.06	3.25	6.70	2.07	4.28	8.85	5.20
(3)	4 x 1/2	2.00	0.25	0.50	5.07	25.70	51.50	0.04
		18.36		97.70			75.25	5.81

$I_T = 81.06 \text{ in}^4$

M allow = 15.25 x 51,000 = 780,000 "#

$$C = \sqrt{\frac{925,000 - 780,000}{91.5}} = 39.8 \text{ in}$$

Reduce span from 58" to 54"   d = 27

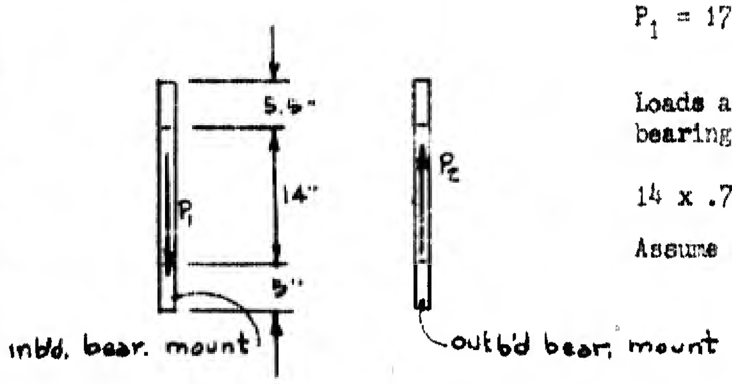
$$M = \frac{W}{24 \times 54} \left(24 \times \frac{(27)^3}{54} - 24\,(27)^2 + C^2\right)$$

$$= 96.5\,(-8750 + C^2)$$

$$= -845,000 + 96.5C^2$$

$$C = \sqrt{\frac{845,000 - 780,000}{96.5}} = \underline{\underline{25.9 \text{ in}}}$$

15) <u>FINAL DRIVE MOUNTS</u>



$P_1 = 175,675\#$; $P_2 = 201,923\#$ (Figures provided by Power Group)

Loads are assumed to be distributed over 75% of bearing diameter.

14 x .75 = 10.5 in.

Assume ℞ thickness = 11/16 in

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Outboard Bearing Mount

$\frac{KL}{r} = \frac{kl}{t}$ (From Alcoa Structural Handbook, page 131)

$L/b = \frac{5.5}{10.5} = .52$

$\therefore$ k = 2.10 (From Alcoa Structural Handbook, page 132)

$\frac{kl}{t} = 2.10 \times \frac{5.5}{.69} = 16.75$

$\therefore$ is compact member

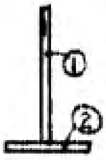
fc allow = 51,000 p.s.i.

fc act. $= \frac{201,923}{10.5 \times .69} = 27,800$ p.s.i.

Inboard Bearing Mount

Acting as beam of span = 10.5 in.

$M = \frac{WL}{12}$ (From Roark's Case 33, page 108)



$M = \frac{175,675 \times 10.5}{12} = 154,000$ "#

$Z \text{ rq'd} = \frac{154,000}{51,000} = 3.02 \text{ in}^3$

Elem.	Dim.	Area	$\overline{Y}$	$A\overline{Y}$	Y	Y^2	AY^2	Io
(1)	5 x 11/16	3.44	2.88	9.90	0.66	0.44	1.50	7.16
(2)	3 x 3/8	1.12	0.19	0.21	2.03	4.13	4.64	0.01
		4.56		10.11			6.14	7.17

$I_T = 13.31 \text{ in}^4$

$\overline{Y} = \frac{10.11}{4.56} = \underline{\underline{2.22}}$ in.

$Z = \frac{13.31}{(5.38 - 2.22)} = \underline{\underline{4.22}} \text{ in}^3$

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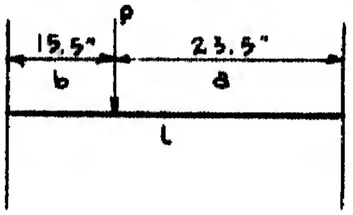
16) ENGINE MOUNTS

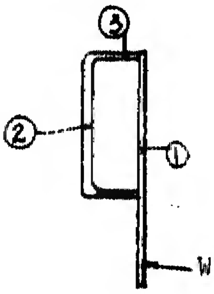
The engine and power package will be supported on flexible mounts which will be carried by the horizontal channel welded to the engine compartment bulkhead at track channel height and by the channel welded on the stern sheet.



P = 5650# (furnished by Power Group)

$$M = \frac{Pa^2b}{L^2}$$ (Roark's Case 32, page 108)

$$M = \frac{P \times \overline{23.5}^2 \times 15.5}{39^2} = 31{,}800 \text{ "\#}$$



$$\bar{Y} = \frac{13.70}{9.04} = 1.52"$$

$I_T = 2.66 \text{ in}^4$

$$Z \text{ rq'd} = \frac{31{,}800}{23{,}500} = \underline{1.35} \text{ in}^3$$

$$Z \text{ prov.} = \frac{2.66}{1.52} = \underline{1.75} \text{ in}^3$$

W = 30 x 1/2 = 15

Elem.	Dim.	Area	$\bar{Y}$	$A\bar{Y}$	Y	Y^2	AY^2	Io
(1)	15 x 1/2	7.50	1.75	13.10	0.23	0.05	0.40	0.16
(2)	3 x 3/8	1.12	0.19	0.21	1.33	1.77	1.99	0.01
(3) x 2	1-1/8 x 3/16	0.42	0.94	0.39	0.46	0.21	0.09	0.02
		9.04		13.70			2.48	0.18

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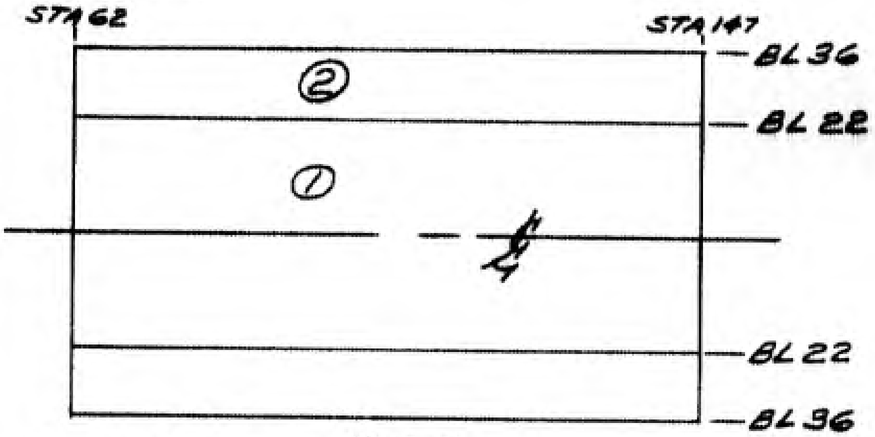
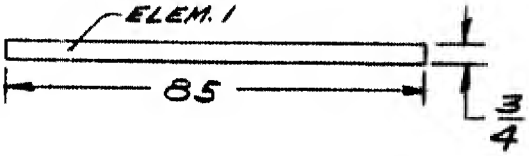
17) CARGO HATCH PLATE COVER



Material is .75 Aluminum

FIG I

Cargo Hatch Cover is to be split at the ℄ and each half is to fold at BL22. This cover has to support a uniform load of 5 psi, caused by a static head of 10 ft. of water. The load on Elem. 1 in Fig. 1 is 22 x 85 x 5 = 9,360#.



$\text{Max } M = \frac{WL}{8}$ (From Roark's Case 13 Page 102)

$\text{Max } M = \frac{WL}{8} = \frac{9{,}360 \times 85}{8} = 99{,}300$ in.lbs.

$Z \text{ req'd} = \frac{M}{f_s}$

$\frac{M}{f_s} = \frac{99{,}300}{26{,}000} = 3.83 \text{ in.}^3$

$Z \text{ prov.} = \frac{bd^2}{6} = \frac{22 \times .75^2}{6} = 2.06 \text{ in.}^3$

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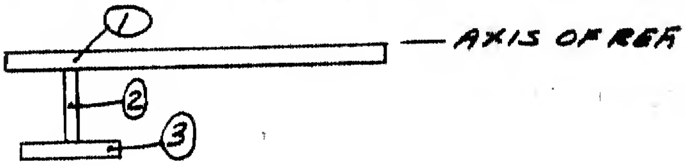
A check now must be made on Elem. 1 for deflection. Assuming that three sides will be supported and the fourth side (along the ℄) will be free, the deflection from Roark's Case 50, Page 206 will be -

$$\text{Max } \Delta = \alpha \frac{wa^4}{Et^3} = .06 \times \frac{5 \times 85^4}{10{,}300{,}000 \times .75^3}$$

$$\text{Max } \Delta = 3.6 \text{ in.}$$

So a member must be provided for bending strength and to reduce the deflection under 1/8 for positive sealing of the covers.



Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
1	3/4x14-7/8	11.12	.37	4.12	.33	1.09	11.21	.53
2	1/4x2-1/2	.63	2.00	1.26	1.30	1.69	1.06	.33
3	1/2x2	1.00	3.50	3.50	2.80	7.85	7.85	.33
		12.75		8.88			20.12	1.19

$$\bar{Y} = \frac{888}{12.75} = .696 \qquad I_t = 20.12 + 1.19 = 21.31 \text{ in.}^4$$

$$Z \text{ actual} = \frac{I_t}{C} = \frac{21.31}{3.75 - .696} = \frac{21.31}{3.054} = 7.00 \text{ in.}^3$$

$$\text{Max}\Delta = \frac{5WL^3}{384EI}$$ (From Roark's Case 13, Page 102)

W will be 1/4 of the total load on Element 1 because of being supported on all sides.

$$\text{Max}\Delta = \frac{5 \times 2{,}340 \times 85^3}{384 \times 10{,}300{,}000 \times 21.31} = .085 \text{ in.} \qquad Z \text{ actual} = \frac{21.31}{3.054} = 7.00 \text{ in}^3$$

A similar member (3x2 tee) shall be used to stiffen Elem. 2 in Fig. 1 and also to carry Element 3.

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The deflection will now be checked on the cover plates.

Element 1 in Fig. 1

$$\text{Max } y = \frac{.1422\ Wb^4}{Et^3(1+2.21\alpha^3)}$$ From Roark's Case 36, Page 203

$$\alpha = \frac{b}{a} = \frac{22}{85} = .26$$

$$\text{Max } y = \frac{.1422 \times 5 \times 22^4}{10{,}300{,}000 \times .75^3(1+2.21 \times .26^3)}$$

$$\text{Max } y = \frac{166{,}000}{4{,}360{,}000(1.0388)}$$

$$\text{Max } y = .0367 \text{ in.}$$

Since the deflection in Element 2 (in Figure 1) is less than in Element 1, the deflection does not have to be checked.

The unit stress in the cover plates will now be checked.

Element 1 in Fig. 1

$$\text{Max } f_s = \frac{.75 \times w \times b^2}{t^2(1+1.61\alpha^3)}$$ From Roark's Case 36, Page 203

$$\text{Max } f_s = \frac{.75 \times 5 \times 22^2}{.75^2(1+1.61 \times .26^3)}$$

$$\text{Max } f_s = \frac{1810}{.562\ (1+.0284)}$$

$$\text{Max } f_s = \frac{1810}{.562 \times 1.0284} = 3{,}120 \text{ PSI}$$

Since the unit stress in Element 2 (in Fig. 1) is less than in Element 1, the unit stress does not have to be checked.

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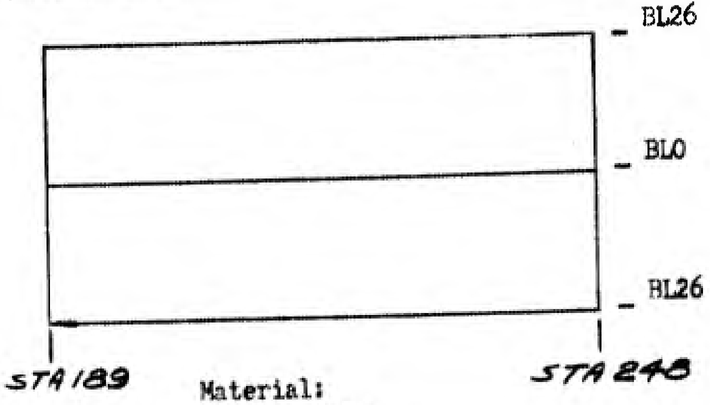
18) ENGINE ACCESS HATCH PLATE COVER



Material:
1/4 Aluminum

Engine hatch cover is to open at the center and has to support a uniform load of 5 PSI caused by a static head of 10 ft. of water

A member will be provided at the ℄ to support both plates

$$W = \frac{5 \times 52 \times 59}{4} = 3{,}830\#$$

$$\text{Max } M = \frac{WL}{8}$$ From Roark's Case 13, page 102

$$\text{Max } M = \frac{3{,}830 \times 59}{8} = 28{,}300 \text{ in. } \#$$

$$Z_{\text{Req'd}} = \frac{M}{f_s} = \frac{28{,}300}{26{,}000} = 1.09 \text{ in.}^3$$

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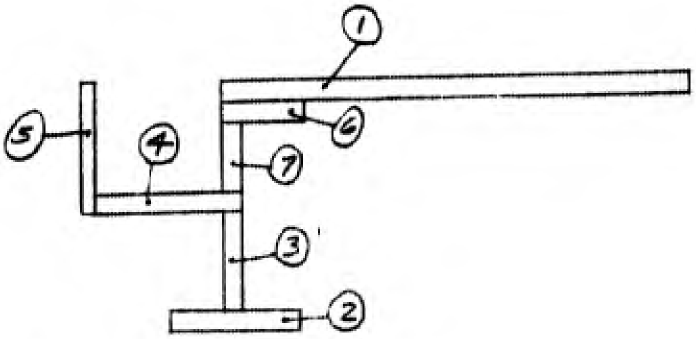
19) This **member** will **be used** to support the engine access hatch **cover plate**



Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	Y	Y^2	AY^2	Io
1	3.75 x .25	.94	.12	.11	1.25	1.56	1.47	.01
2	2-1/4 x.37	.83	3.06	2.54	1.69	2.86	2.37	.01
3	.31 x 1.62	.50	2.06	1.03	0.69	0.48	0.24	.11
4	1.25 x 1/4	.31	1.12	.35	0.25	0.06	0.02	---
5	1/4 x 1	.25	.75	.19	0.62	0.38	0.09	.02
6	1 x .25	.25	.37	.09	1.00	1.00	0.25	---
7	.5 x .25	.12	.75	.09	0.62	0.38	0.05	---
		3.20		4.40			4.49	0.15

$I_t = 4.65 \text{ in}^4$

$\bar{Y} = \frac{4.40}{3.20} = 1.37$ $\qquad Z = \frac{4.65}{(3.25-1.37)} = 2.48 \text{ in}^3$

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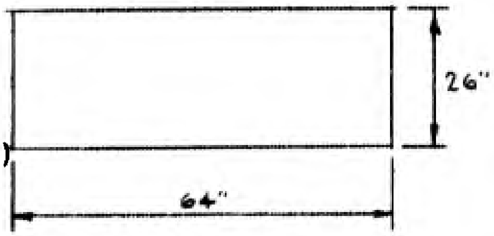
20) **Ramp Plate** This plate has to take a uniform load of 10 PSI

t = .44 in

w = 10 PSI

$$\text{Max} = \frac{.0284\ Wb^4}{Et^3(1+1.056\alpha^5)}$$

From Roark's Case 41 Page 205



$\alpha = \frac{26}{64}$

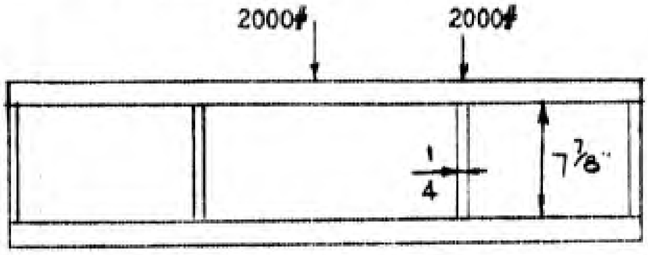
$$\text{Max } \Delta = -\frac{.0284 \times 10 \times 26^4}{10{,}300{,}000 \times .44^3 \left[1 + 1.056 \left(\frac{26}{64}\right)^5\right]}$$

Max Δ = .14

$$\text{Max } f_s = \frac{.5Wb^2}{t^2(1+.623\alpha^6)}$$

From Roark's Case 41, Page 205

$$f_{s_{max}} = \frac{.5 \times 10 \times 24^2}{.44^2 \left(1 + .623 \left[\frac{26}{64}\right]^6\right)} = 14{,}900 \text{ PSI}$$



$$\text{Slenderness Ratio} = \frac{1.63 \times 7.875}{.25} = 51.3$$ ∴ This Member is a short column

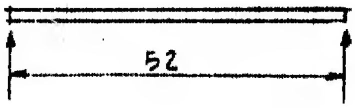
f_c = 45,400 - 368 x 51.3 = 26,500 PSI (Reynolds Structural Handbook, Page 105)

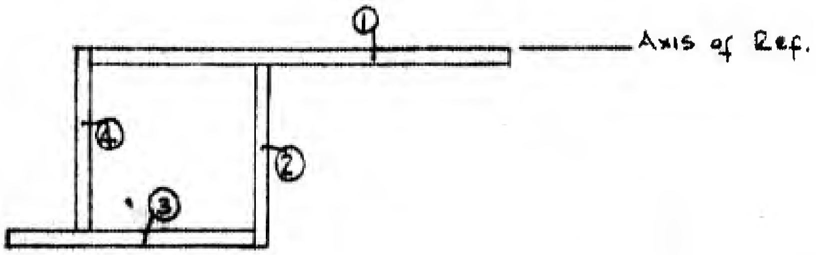
10 x .25 = 2.5 $f_c = \frac{2000}{2.5} = 800\#/in^2$

2000# concentrated between members is assumed

$$S = \frac{3 \times 2000}{2 \times 3.14 \times 3 \times .75^2}\left[(3+1)\ 1.114 + 5\left(1 - \frac{b}{a}\right)\right] = 4237 \text{ PSI}$$ (Roark's Case 42, Page 205)

21) <u>This Member is Prov. to Stiffen the Stern Plate</u>





Elem.	Dim.	Area	$\bar{\bar{y}}$	$A\bar{\bar{y}}$	$\underline{Y}$	$\underline{Y}^2$	$A\underline{Y}^2$	Io
1	11/16 x 10.3	7.08	.34	2.40	.51	.26	1.84	.28
2	2 x 3/8	.75	1.68	1.26	.83	.69	.52	.25
3	3-5/8 x 3/8	1.36	2.50	3.40	1.65	2.72	3.70	.02
4	2 x 1/4	.50	2.34	1.17	1.49	2.22	1.11	.17
		9.69		8.23			7.17	.72

$$\bar{\bar{Y}} = \frac{8.23}{9.69} = .85$$

$$I_t = 7.17 + .72 = 7.89 \text{ in}^4$$

$$M = \frac{7.89 \times 26{,}000}{1.83} = 112{,}000 \text{ in \#}$$

$$M = \frac{PL}{8} \qquad 112{,}000 = \frac{P \times 52}{8}$$

(From Roark's Case 13, Page 102)

$$P = \frac{112{,}000 \times 8}{52} = 17{,}200\# \quad = \quad \text{Max. Bump Load that this member can carry.}$$

IGERSOLL KALAMAZOO DIVISI
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

APPENDIX II OF SECTION 1

CONTRACT NO.
NObs 4561

VEHICLE DESCRIPTION
LANDING VEHICLE (TRACKED) LVTPX11

CALCULATION OF WEIGHTS, TRIM AND STABILITY

DATE
October 1962

Prepared by

H. Streb

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO: NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

GROUP WEIGHT SUMMARY

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
1.	Hull	10,911.9	9.40	102,583.7	3.46	37,741.7
2.	OEM	827.5	5.35	4,427.5	3.50	2,896.2
3.	Power Train	5,607	18.13	101,647.7	2.69	15,097.8
4.	Suspension	6,758	10.66	72,023.0	.26	1,768.1
5.	Hydraulics	1,180.0	16.51	19,478.1	3.47	4,408.0
6.	Electrical	752.3	11.95	8,988.1	3.80	2,859.9
7.	CO_2 System	52.5	10.35	543.6	4.48	235.5
8.	Fuel System	94.1	10.18	957.7	4.61	433.60
	Total Group Weights	26,183.3	11.86	310,649.4	2.50	65,440.8

GERSOLL KALAMAZOO DIVISI
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

HULL GROUP

HULL WEIGHTS

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
1.	Frame No. 1 (Sta. 59 9/16)	86.2	4.96	427.6	2.70	232.7
2.	Frame No. 2 (Sta. 105-1/16)	37.3	8.75	326.4	.60	22.4
3.	Frame No. 3 (Sta. 150 9/16)	37.3	12.55	468.1	.60	22.4
4.	Frame No. 4 (Sta. 189)	291.5	15.75	4,591.1	3.10	903.7
5.	Frame No. 5 (Sta. 208)	18.2	17.33	315.4	.22	4.0
6.	Fwd. Top Plate - 3/4 in. Plate - Sta. 0-Sta. 54	248.0	2.25	558.0	5.85	1,450.8
7.	Intermediate Top Plate 3/4 in. Plate - Sta. 54 - Sta. 189.	652.2	11.00	7,174.2	6.22	4,056.6
8.	Aft Top Plate - 1/4 in. Plate - Sta. 189 - Sta. 250.8	44.00	17.6	774.4	6.2	272.8
9.	Bow Plates - 1 3/16 in. Plate (2)	182.00	.15	27.3	3.65	664.3
10.	Fwd. Side Plate - 1-3/16 in. Plate - Sta. 0 - Sta. 189 (2).	1,601.5	7.95	12,731.9	4.18	6,694.3
11.	Aft Side Plate - 3/8 in. Plate - Sta. 189 - Sta. 244	225.00	18.4	4,140.0	4.46	1,003.5
12.	Track Channel Side - 13/16 in. Plate (2)	911.6	9.7	8,837.2	1.23	1,120.5

IGERSOLL KALAMAZOO DIVISI
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATIONS OF WEIGHTS

CONTRACT NO. NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

HULL GROUP

Hull Group

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity: Station Line: Aft of Sta 0 (Ft.)	Station Line: Moments (Ft.Lbs.)	Water Line: Above Lowest Point of Keel (Ft.)	Water Line: Moments (Ft.Lbs.)
13.	Track Channel Top - 3/4 in. Plate	950.9	11.5	10,935.3	2.48	2,358.2
14.	Stern Plate - 11/16 in. Plate	335.0	20.9	7,001.5	3.38	1,132.3
15.	Stern Framing	18.8	20.41	383.7	2.30	43.2
16.	Hull Bottom Plate ½ in. Plate	765.2	10.62	8,126.4	.02	15.3
17.	Cargo Deck Plate - ¼ in. Plate	306.0	9.00	2,754.0	.38	116.2
18.	Cargo Hatch & Fittings	625.0	8.75	5,468.7	6.40	4,000.0
19.	Cargo Hatch Framing	91.35	9.2	840.4	6.12	559.0
20.	Ramp - Complete	814.1	1.70	1,384.0	2.62	2,132.9
21.	Bow Ramp Framing	112.2	1.70	190.7	2.62	294.0
22.	Engine Hatch - Framing & Fittings	33.0	19.2	633.6	6.1	201.3
23.	Access Hatch - Framing & Fittings	47.7	14.97	714.1	6.25	298.1
24.	Escape Hatches - Framing and Fittings	231.7	9.92	2,298.5	4.08	945.3
25.	Troop Seating Plates and Framing	125.4	9.89	1,240.2	2.73	342.3

IGERSOLL KALAMAZOO DIVISI
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO: NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

HULL WEIGHTS

HULL GROUP

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
26.	Driver & Assist. Drivers Cupola	200.0	3.50	700.0	6.25	1,250.00
27.	Machine Gun Cupola	150.0	2.87	430.5	6.25	937.5
28.	Mooring Cleats (Fwd.)	10.0	.70	7.0	5.90	59.0
29.	Mooring Cleats (Aft)	10.0	20.00	200.0	5.80	58.0
30.	Lifting Eyes	40.0	10.8	432.00	6.40	256.0
31.	Aft Towing Device	24.0	20.60	494.4	1.0	24.0
32.	Fwd. Towing Device	24.0	1.50	36.0	.60	14.4
33.	Misc. Hinges, Brackets & Foundations	550.0	10.30	5,665.0	3.00	1,650.0
34.	Fasteners, Welding Wire & Paint	350.0	8.95	3,132.5	3.60	1,260.0
35.	Corner Extrusion	364.0	5.5	2,002.0	5.00	1,820.0
36.	Corner Caps (2)	20.0	.25	5.0	5.30	106.0
37.	Plates - Final Drive Support	131.8	18.8	2,477.8	1.25	164.8
38.	Rad. Comp. Framing	200.0	18.5	3,700.0	4.94	988.0
39.	Muffler Framing	47.0	20.4	958.8	5.7	267.9
	Total	10,911.9		102,583.7		37,741.7
			9.40		3.46	

IGERSOLL KALAMAZOO DIVIS
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO. NObs 4561 DATE October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPXII

HULL OUTFITTINGS & EQUIPMENT (O&M)

GROUP WEIGHTS

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
1	Winterization Kit	35				
2	Driver's Seat	35				
3	Troop Seats	100				
4	Tow Cable (AFT)	25				
5	Towing Bridle	15				
6	Medical Chest	10				
7	Flashlights (2)	2				
8	Grease Gun	2				
9	Boat Hooks (2)	8				
10	Pick	8				
11	Shovel	4				
12	Axe	5				
13	Sledge	8				
14	Crow Bar	7				
15	Hand Oiler	1				
16	Signal Light	12				
17	Fuel Fill Extension	10				
18	Gun Kit	10				

GERSOLL KALAMAZOO DIVISI
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO. NObs 4561 DATE October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPXII

HULL OUTFITTINGS & EQUIPMENT (OFM)

GROUP WEIGHTS

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
19	Lamp Box w/Spare Lamps	1				
20	Spare Gun Parts	10				
21	Bowline and Sternfast	15				
22	Track Fixture	35				
23	2500 Round (7.62 mm) Ammunition	164				
24	Tow Cable Shackle (2) (Spare)	20				
25	Track Blocks (Spare)(2)	60				
26	Pyrotechnic Box	3				
27	Manuals	20				
28	Tool Roll	50				
29	Electrical Cable	25				
30	Water and Gas Cans (2)	60				
31	M8A2 Gas-Particulate Unit	60				
32	7.62 mm Machine Gun Mount	7.5				
	Total	827.5		4,427.5		2896.25
			5.35		3.50	

IGERSOLL KALAMAZOO DIVIS
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO: NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

POWER TRAIN GROUP

POWER TRAIN WEIGHTS

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
1.	Engine W/Accessories	1,800.0	18.2	32,760.0	3.9	7,020.0
2.	Engine Oil (6.5 Gal.)	49.0	18.2	891.8	3.9	191.1
3.	Transmission (Dry)	1,350.0	18.7	25,245.0	1.2	1,620.0
4.	Transmission Oil (17 Gal.)	127.0	18.7	2,374.9	1.2	152.4
5.	Transfer Drop Box	280.0	16.3	4,564.0	2.6	728.0
6.	Final Drives (2)	800.0	19.2	15,360.0	1.0	800.0
7.	Drive Shaft (Drop-Trans.)	24.0	16.7	400.8	1.3	31.2
8.	Drive Shaft (Trans. Final Dr.	74.0	19.2	1,420.8	1.3	96.2
9.	Mount Assembly (Power Pack)	189.0	18.8	3,553.2	2.2	415.8
10.	Eng. Exh. System	70.0	20.5	1,435.0	5.7	399.0
11.	Aspiration Air Cleaner	14.0	17.6	246.4	3.3	46.2
12.	Radiators (2)	274.0	18.9	5,178.6	4.8	1,315.2
13.	Radiator Coolant (14 Gal.)	122.0	18.9	2,305.8	4.8	585.6
14.	Cooling Water Lines & Contents	36.0	19.2	691.2	4.7	169.2
15.	Radiator Air Inlet Grilles (2)	70.0	18.6	1,302.0	6.2	434.0
16.	Radiator Air Outlet Grilles (2)	45.0	20.9	940.5	4.8	216.0

CALCULATIONS OF WEIGHT

CONTRACT NO: NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

POWER TRAIN GROUP

POWER TRAIN WEIGHTS

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Station 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
17.	Cooling Fans & Mtgs. (2)	15.0	18.4	276.0	5.3	79.5
18.	Controls (Eng. Comp't.)	15.5	18.3	283.7	2.5	38.75
19.	Controls (Crew Comp't.)	28.0	7.5	210.0	2.5	70.0
20.	Control Tower	15.0	2.2	33.0	3.7	55.5
21.	Brake Pedal	12.0	.8	9.6	3.1	37.2
22.	Hand Throttle	1.0	2.5	2.5	4.4	4.4
23.	Accelerator Pedal	1.0	.8	.8	2.7	2.7
24.	Misc. Control Items	30.0	7.5	225.0	2.5	75.0
25.	Engine Water (3 Gal.)	25.0	18.2	455.0	3.9	97.5
26.	Oil Reservoir (Transf.)	12.5	16.3	203.8	0.4	5.0
27.	Oil Res. Contents (8 Gal.)	60.0	16.3	978.0	0.4	24.0
28.	Fan Shroud (2)	10.0	18.6	186.0	5.2	52.0
29.	Surge Tank	58.0	19.7	114.3	5.8	336.4
	Total	5,607.0		101,647.7		15,097.85
			18.13		2.69	

GERSOLL KALAMAZOO DIVIS
BORG WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO. 4561 **DATE** October 26, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

SUSPENSION GROUP

SUSPENSION WEIGHTS

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft. Lbs)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
1	Shroud Assembly (2)	528.1	10.75	5,677	1.625	858.1
2	Suspension Arm & Road Wheel Assembly (8)	1,859	10.8	20,077	.13	242
3	Track (Complete)	3,854	10.75	41,431	.06	231
4	Idler Assembly (2)	299	2.00	598	1.06	317
5	Drive Sprocket Assembly (2)	218	19.45	4,240	.55	120
	Total	6,758.1	10.66	72,023	.262	1,768.1

ERSOLL KALAMAZOO DIVISIC
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO. NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

HYDRAULICS GROUP

HYDRAULIC COMPONENTS & SYSTEM

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft. Lbs)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs)
1.	Hyd. Oil Reservoir	45	17.5	787.5	5.0	225.0
2.	Main Pump Filter	35	16.2	567.0	3.0	105.0
3.	Main Hydraulic Pump	56	17.3	968.8	4.7	263.2
4.	Bilge Pump (Aft.) (2)	40	18.5	740.0	1.7	68.0
5.	Fan Motor (2)	16	18.3	292.8	6.8	108.8
6.	Track Comp. Cyl. (2)	110	18.0	1,980.0	2.3	253.0
7.	Comp. Cyl. Acc. (2)	100	17.0	1,700.0	3.0	300.0
8.	Bilge Pump (Fwd) (2)	40	15.2	608.0	1.7	68.0
9.	Ramp Cylinder (2)	56	2.5	140.0	1.5	84.0
10.	Air Damper Cyl.	2	17.0	34.0	6.5	13.0
11.	Hydraulic Oil:					
	A. Reservoir	200	17.5	3,500.0	5.0	1,000.0
	B. System	80	17.0	1,360.0	4.0	320.0
12.	Piping and Components	400	17.0	6,800.0	4.0	1,600.0
	Total	1,180		19,478.1		4,408.0
			16.51		3.74	

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO: NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

ELECTRICAL GROUP

ELECTRICAL WEIGHTS

No.	Description	Weights (Lb.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Station 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lb.)
1.	Radio and Amplifier	48.0	1.58	75.8	5.37	257.8
2.	Radio Frame	5.6	1.58	8.8	5.00	28.0
3.	Control Box	7.0	4.33	30.3	5.25	36.7
4.	Control Box	3.5	15.20	53.2	5.30	18.55
5.	Antenna	2.0	2.08	4.2	6.25	12.5
6.	Batteries	284.0	16.30	4,629.2	3.22	914.5
7.	Battery Frames	33.9	16.30	552.6	2.75	93.2
8.	Instrument Panel	24.1	1.33	32.1	5.00	120.5
9.	Compass	7.7	2.69	20.7	3.19	24.6
10.	Radio Set	20.0	.75	15.0	4.16	83.2
11.	I-R Viewer	10.7	4.20	44.9	2.60	27.8
12.	I-R Amplifier	9.9	.66	6.5	5.00	49.5
13.	I-R Stowage Box	9.6	4.20	40.3	2.60	24.9
14.	Mount Plates - Amplifier	.2	.66	.1	5.58	1.1
15.	Ext. Mt. Viewer	.9	2.40	2.2	6.42	5.8
16.	Int. Mt. & Bracket	2.3	2.40	5.5	5.83	13.4
17.	Bilge Pump (Aux.)	32.0	16.60	531.2	.66	21.1
18.	Lights - Driving	16.0	1.66	26.6	6.16	98.6

IGERSOLL KALAMAZOO DIVISI
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO: NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

ELECTRICAL GROUP

ELECTRICAL WEIGHTS

No.	Description	Weight (Lb.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
19.	Headlight Mts. & Guards	2.9	1.40	4.1	5.10	14.8
20.	Stop & Tail Lights	6.0	20.8	124.8	4.29	25.7
21.	Signal Light	10.3	1.89	19.5	2.92	30.1
22.	Signal Search Brkt.	.6	1.89	1.1	5.84	3.5
23.	Bow Light	1.5	5.00	7.5	7.00	10.5
24.	Stern Light	1.0	20.00	20.0	6.6	6.6
25.	Port Light	2.2	12.00	26.4	6.50	14.3
26.	Starboard Light	2.2	12.00	26.4	6.50	14.3
27.	Compartment Lights	4.0	1.79	7.2	5.71	22.8
28.	Compartment Light	2.0	13.59	27.2	6.17	12.3
29.	Compartment Lights	4.0	16.2	64.8	6.0	24.0
30.	Distribution Box	10.0	18.0	180.0	3.0	30.0
31.	Alternator	33.0	19.30	636.9	4.30	141.9
32.	Elect. Cable & Clamps	50.0	10.20	510.0	2.50	125.0
33.	Voltage Regulator	8.5	15.80	134.3	3.00	25.5
34.	Horn	2.8	2.08	5.8	6.00	16.8
35.	Engine Comp. Scavenge	28.3	16.58	469.2	5.33	150.8
36.	Engine Air Inlet	21.4	16.54	353.9	4.58	98.0

GERSOLL KALAMAZOO DIVISIC
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NO: NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

ELECTRICAL GROUP

ELECTRICAL WEIGHTS

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
37.	Crew Vent	43.5	6.62	288.0	5.91	257.1
38.	Warning Light	.7	2.6	1.82	6.0	4.2
	Total	752.3		8,988.12		2,859.95
			11.95		3.80	

GERSOLL KALAMAZOO DIVISI
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT: NObs 4561

DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

HYDRAULICS GROUP

CO_2 SYSTEM

No.	Description	Weight (Lbs.)	Moments & Center of Gravity: Station Line: Aft of Sta 0 (Ft.)	Station Line: Moments (Ft.Lbs.)	Water Line: Above Lowest Point of Keel (Ft.)	Water Line: Moments (Ft.Lbs.)
1.	Portable CO_2	21	4.66	97.86	3.16	66.36
2.	CO_2 Tube, Cables & Etc.	7.25	9.87	71.55	6.12	44.37
3.	CO_2 5 lb. Bottle	23.0	15.31	352.13	5.29	121.67
4.	Discharge Nozzle (1)	1.28	17.25	22.08	2.45	3.13
	Total	52.53		543.62		235.53
			10.35		4.48	

GERSOLL KALAMAZOO DIVISI
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT: NObs 4561 DATE: October 30, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

HYDRAULICS GROUP

FUEL SYSTEM

No.	Description	Weight (Lbs.)	Moment & Center of Gravity			
			Station Line		Water Line	
			Aft of Station 0 (Ft)	Moments (Ft.Lbs)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs)
1.	Fuel Fill (2)	39.04	6.26	244.39	6.25	244.00
2.	Fuel Cell (2)	16.38	6.02	98.61	4.25	69.62
3.	Fuel Filter (1)	2.5	17.75	44.38	3.38	8.45
4.	Water Separator (1)	7.5	18.26	136.95	3.38	25.35
5.	Transfer Tank (1)	11.7	16.91	197.85	4.33	50.66
6.	Fuel Pump (2)	2.0	6.02	12.04	2.75	5.50
7.	Fuel System	15.0	14.9	323.5	2.0	30.00
	Total	94.12		957.72		433.58
			10.18		4.61	

DEFINITION OF SYMBOLS AND ABBREVIATIONS

LCG - Longitudinal Center of Gravity

LCB - Longitudinal Center of Buoyancy

KG - Vertical Center of Gravity above Keel

KL - Keel Line

KB - Vertical Center of Buoyancy above Keel

BM - Transverse Metacentric Radius

KM - Height of Transverse Metacenter above Keel

GM - Transverse Metacentric Height

MT In. - Moment to Change Trim 1 in.

CF - Center of Floatation

WP - Water Plane

L_c - Chine Length

GZ - Righting Arm

IGERSOLL KALAMAZOO DIVISI
BORG WARNER CORPORATION
KALAMAZOO, MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NObs 4561 DATE October 31, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

GROUP SUMMARY
CONDITION

LIGHT LOAD

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
1.	Total Group Weights	26,183	11.86	310,649.4	2.50	65,440.8
2.	Fuel	1,253	6.02	7,543.1	4.25	5,325.3
3.	Crew	400	3.50	1,400.0	4.30	1,720.0
4.	Total Weights	27,836		319,592.5		72,486.1
5.	LCG		11.48			
6.	KG Above KL				2.60	

7.	Mean Draft = 2.82 Ft. (Calc.)	KB = 1.43 Ft. Above KL
8.	LCG = 11.48 Ft. Aft of Sta. 0 (5)	BM = 4.45 Ft.
9.	LCB = 10.78 Ft. Aft of Sta. 0 (Calc.)	KM = 5.88 Ft. Above KL
10.	Trim Lever = .70 Ft. (Diff. between 8 & 9)	KG - 2.60 Ft. Above KL
11.	Trim Moment = 19,485 Lbs. Ft. (10 x 4)	GM = 3.28 Ft.
12.	MT In. = 1,845 Lbs. Ft. (Calc.)	
13.	Trim (In.) = 10.56 (11 ÷ 12)	
14.	Trim (Deg.) 2° 27' (13 ÷ 19) = Tan. of trim angle = .04266	
15.	CF = 10.64 Ft. Aft of Sta. 0 (Calc.)	
16.	Drafts:	
	a. Fwd. = 28.38 In.	
	b. Aft = 38.94 In.	
	c. Mean = 33.84 In.	
17.	Centroid of projected chine area aft of sta. 0 = -- Ft. (Calc.)	
18.	LCB = --- Ft.---------------------of centroid (% L_c) (Calc.)	
19.	Length WP = 247.5 In. (Calc.)	

IGERSOLL KALAMAZOO DIVISI
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NObs 4561 DATE October 31, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

GROUP SUMMARY

CONDITION

FULL LOAD (Troops)

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
1.	Total Group Weights	26,183	11.86	310,649.4	2.50	65,440.8
2.	Fuel	1,253	6.02	7,543.1	4.25	5,325.3
3.	Crew	400	3.50	1,400.0	4.30	1,720.0
4.	Full Cargo - 29 Troops At 210# each	6,090	10.00	60,900.0	2.80	17,052.0
5.	Total Weights	33,926		380,492.5		89,538.1
6.	LCG		11.22			
7.	KG Above KL				2.64	

8.	Mean Draft = 3.25 Ft. (Calc.)	KB = 1.75 Ft. Above KL
9.	LCG = 11.22 Ft. Aft of Sta. 0 (6)	BM = 3.65 Ft.
10.	LCB = 10.75 Ft. Aft of Sta. 0 (Calc.)	KM = 5.40 Ft. Above KL
11.	Trim Lever = .47 Ft. (Diff. between 9 & 10)	KG = 2.64 Ft. Above KL
12.	Trim Moment = 15,945 Lbs. Ft. (11 x 5)	GM = 2.76 Ft.
13.	MT In. = 1,890 Lbs. Ft. (Calc.)	
14.	Trim (In.) = 8.44 (12 ÷ 13)	
15.	Trim (Deg.) = 1° 56' (14 ÷ 20) = Tan. of trim angle = .03387	
16.	CF = 10.60 Ft. Aft of Sta. 0 (Calc.)	
17.	Drafts:	
	a. Fwd. = 34.69 In.	
	b. Aft = 43.13 In.	
	C. Mean = 39.00 In.	
18.	Centroid of projected chine area aft of sta. 0 = --- Ft. (Calc.)	
19.	LCB = --- Ft. ------------------------- of centroid (% Lc) (Calc.)	
20.	Length WP = 249.12 In. (Calc.)	

GERSOLL KALAMAZOO DIVIS.
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

CALCULATION OF WEIGHTS

CONTRACT NObs 4561 DATE October 31, 1962

VEHICLE DESCRIPTION

LANDING VEHICLE (TRACKED) LVTPX11

GROUP SUMMARY

CONDITION

FULL LOAD

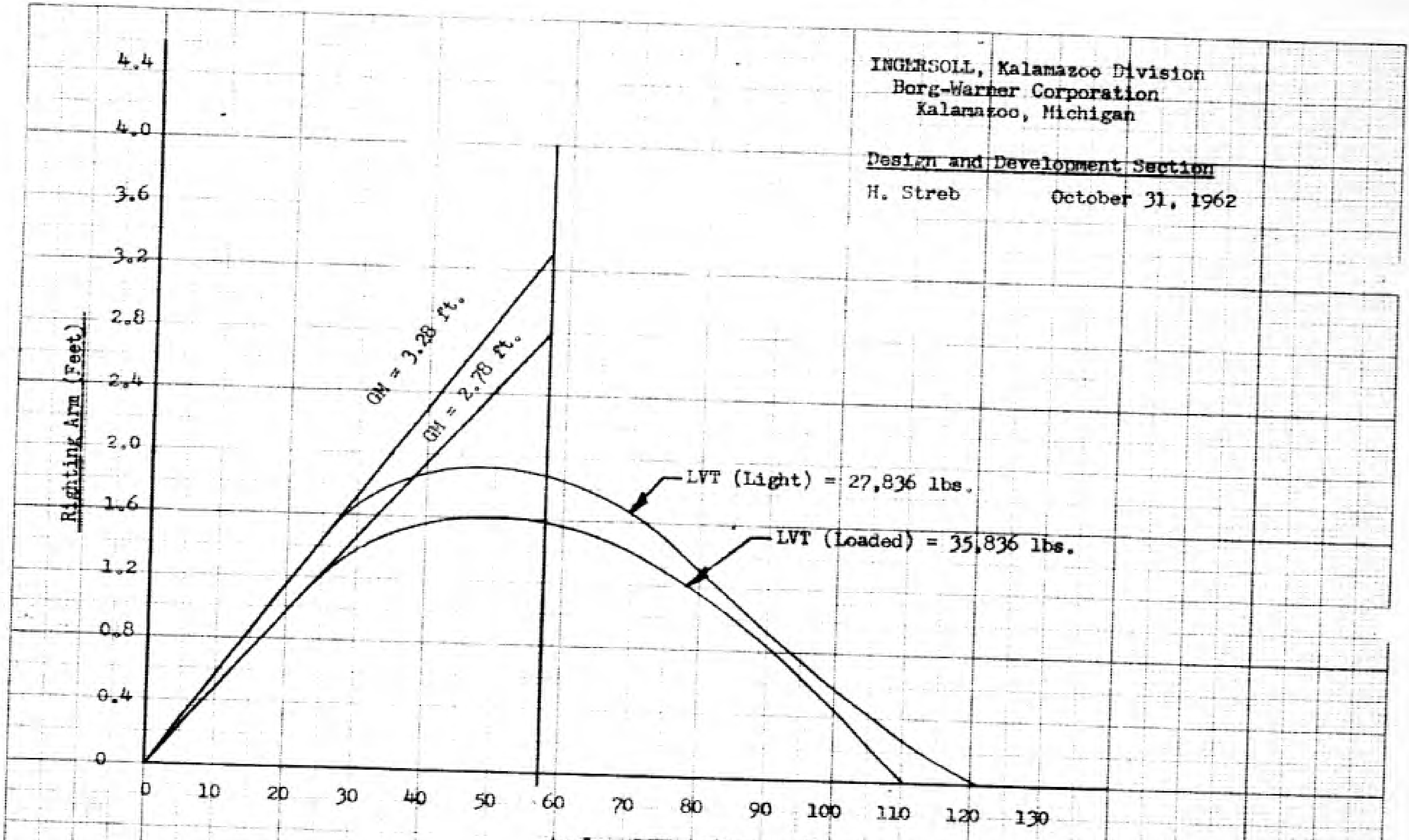
No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
1.	Total Group Weights	26,183	11.86	310,649.4	2.50	65,440.8
2.	Fuel	1,253	6.02	7,543.1	4.25	5,325.3
3.	Crew	400	3.50	1,400.0	4.30	1,720.0
4.	Full Cargo	8,000	9.00	72,000.0	2.44	19,520.0
5.	Total Weights	35,836		391,592.5		92,006.1
6.	LCG		10.93			
7.	KG Above KL				2.57	

8. Mean Draft = 3.40 Ft. (Calc.) KB = 1.83 Ft. Above KL
9. LCG = 10.93 Ft. Aft of Sta. 0 (6) BM = 3.52 Ft.
10. LCB = 10.75 Ft. Aft of Sta. 0 (Calc.) KM = 5.35 Ft. Above KL
11. Trim Lever = .18 Ft. (Diff. between 9 & 10) KG = 2.57 Ft. Above KL
12. Trim Moment = 6,450 Lbs. Ft. (11 x 5) GM = 2.78 Ft.
13. MT In. = 1,900 Lbs. Ft. (Calc.)
14. Trim (In.) = 3.39 (12 ÷ 13)
15. Trim (Deg.) = 0° 46' (14 ÷ 20) = Tan. of trim angle = .01347
16. CF = 10.54 Ft. Aft of Sta. 0 (Calc.)
17. Drafts:
    a. Fwd. = 39.09 In.
    b. Aft = 42.48 In.
    C. Mean = 40.80 In.
18. Centroid of projected chine area aft of sta. 0 = --- Ft. (Calc.)
19. LCB = --- Ft.-----------------------of centroid (% Lc) (Calc.)
20. Length WP = 251.5 In. (Calc.)

INGERSOLL, Kalamazoo Division
Borg-Warner Corporation
Kalamazoo, Michigan

Design and Development Section

H. Streb October 31, 1962



Angle of Heel (Degrees)

FIGURE TRANSVERSE STATIC STABILITY

LVTPX11

2.0 MODEL TESTS

Initial tests of the one-fourth (1/4) scale self-propelled model indicated a need to increase the propulsion efficiency of the track and to reduce hull drag. Necessary changes were made and a new series of tests was scheduled for 23 November 1962.

A final report covering the tests will be prepared and delivered by 1 January 1963.

3.0. PERFORMANCE ANALYSIS

3.1. Performance Specifications. (Ref. BuShips Contract Specification SHIPS-A-4159).

Design Maximum Gross Vehicle Weight	35,000 Lb
Design Unit Rolling Resistance:	
Cross-Country and All Grades	140 Lb/Ton
Hard Level Surface	100 Lb/Ton
Design Speeds:	
Land Forward:	
Hard Level Surface	40 MPH
Cross-Country	20 MPH
Minimum Sustained	2.5-5 MPH
Land Reverse: Maximum not less than	5 MPH
Water Forward:	
Maximum Attainable with Land Locomotion System	7 MPH (Target)
Water Astern	4 MPH
Braking Ability:	
Stop and Hold On	70% Grade

Gradability:	
Maximum Slope Forward or Reverse (From Standing Start to two MPH in Forward Gear)	70%
Maximum Side Slope	60%
Stability:	
Water - Fully Loaded	100° Roll to P/S
Surfability - Fully Loaded	10 Ft Plunging Surf
Land - Fully Loaded	90° Turn On 60% Side Slope
Trench - Crossing Ability	8 Ft Wide and 4 Ft Deep
Vertical - Obstacle Ability	3 Ft High
Endurance Limit:	
Land	300 Miles at 25 MPH
Water	6 Hr at Maximum Speed (whichever is greater)
Design Ambient Temperature Range	-40°F to 125°F
Design Performance Temperature	80°F
Design Operating Life:	
Land Operation (80%)	800 Hr
Water Operation (20%)	200 Hr
Total Life Between Overhauls	1,000 Hr

3.2. Power Requirements.

3.2.1. Power Requirements - Land Operation.

3.2.1.1. Required Tractive Effort. In order to establish vehicle power and torque levels required to provide satisfactory performance, the vehicle rolling resistance and required tractive effort are analyzed. To insure satisfactory performance under all operating conditions, and establish conservative design stress levels throughout the power train, a maximum tractive effort to weight ratio of one (TE/W = 1.0) is used in preliminary determination of required ratio coverage. Also, in estimating vehicle performance, a unit rolling resistance of 140 pounds per ton is used for gradability calculations, and an average unit rolling resistance of 100 pounds per ton in all general performance calculations. It is assumed that the unit rolling resistance is constant at all speeds and includes final drive and track losses.

The required tractive effort, TE, for a vehicle is the summation of its rolling resistance (F_{RR}) plus grade resistance (F_{GR}).

$$TE = F_{RR} + F_{GR}$$

The rolling resistance is the product of the gross vehicle weight (W) times the coefficient of unit rolling resistance (f) times Cos θ, where θ is the grade angle in degrees.

$$F_{RR} = W \cdot f \cdot \cos \theta$$

As a simplification in calculations, it is general practice to neglect the effect of the grade angle. The equation then becomes:

$$F_{RR} = W \cdot f$$

The grade resistance is the product of gross vehicle weight times Sin θ, where θ is the grade angle in degrees.

$$F_{GR} = W \cdot \sin\theta$$

Then, total required tractive effort for any condition is found by the formula,

$$TE = Wf + W \sin\theta = W (F + \sin\theta)$$

For level ground, or zero slope, operation, the equation is simply:

$$TE = Wf$$

When the coefficient of rolling resistance is,

$$f = 100 \frac{\text{lb}}{\text{ton}} = \frac{100}{2{,}000} = 0.050 \text{ (Hard level surface)}$$

$$TE = 35{,}000 \times 0.050 = 1{,}750 \text{ lb}$$

The required tractive effort to negotiate a 70 percent forward slope with unit rolling resistance of 140 pounds per ton is calculated as:

$$\text{Grade Angle, } \theta = 35^{\circ};\ \sin\theta = 0.5736;\ f = 140 \frac{\text{lb}}{\text{ton}} \quad \frac{140}{2{,}000} = 0.07$$

$$TE = (35{,}000 \times 0.07) + (35{,}000 \times 0.5736) = 22{,}550 \text{ lb}$$

In the tabulation below the required tractive effort for other grades is listed.

REQUIRED TRACTIVE EFFORT

Rolling Resistance Lb/Ton	Grade - %	Tractive Effort - Lb
100	0	1,750
140	0	2,450
140	3	3,500
140	10	5,940
140	20	9,320
140	30	12,510
140	40	15,450
140	50	18,100
140	60	20,460
140	70	22,550

During acceleration the tractive effort required has to overcome not only the rolling resistance F_{RR} and grade resistance F_{GR} but also the inertia resistance of the vehicle F_{IR}. Assuming that the acceleration, a, is constant, the inertia resistance is:

$$F_{IR} = m \times a = \frac{W}{g} a$$

and the total tractive effort, TE_A, required during acceleration is then:

$$TE_A = F_{RR} + F_{GR} + F_{IR}$$

$$= Wf + W \sin\theta + \frac{W}{g} a$$

$$TE_A = TE + \frac{W}{g} a$$

Where: m is the mass of the vehicle and g the gravity acceleration.

To accelerate the vehicle on a 70 percent grade from zero to two mph within a distance of 15 feet an acceleration $a = 0.29\ \text{ft/sec}^2$ is required, as proved in the calculations below:

$$s = \frac{V^2}{2a}$$

From which follows:

$$a = \frac{V^2}{2s}$$

Where s = 15 feet is the distance through which the vehicle is accelerated to the velocity V = 2 mph = 2.94 ft/sec.

$$a = \frac{2.94^2}{2 \times 15} = \frac{8.64}{30}$$

$$a = 0.29\ \text{ft/sec}^2$$

The time for accelerating the vehicle is:

$$t = \frac{V}{a} \quad \frac{2.94}{.29}$$

$$t = 10.14\ \text{sec}$$

The required tractive effort is then:

$$TE_A = 22{,}550 + 35{,}000\left(\frac{0.29}{32.2}\right)$$

$$TE_A = 22{,}550 + 315$$

$$TE_A = 22{,}865\ \text{Lb}$$

3.2.1.2. Required Tractive Horsepower.

The tractive horsepower required at speed V in mph on any grade can be calculated as:

$$HP = \frac{V \times TE}{375}$$

Where TE is the tractive effort required in pounds.

The required tractive horsepower has to be calculated for the different operating conditions, so that the proper engine selection can be made later on.

3.2.1.2.1. 40 MPH on Hard Level Surface.

Assuming a 100 pounds per ton rolling resistance, the tractive effort for zero slope is:

$$TE_o = 1{,}750 \text{ Lb}$$

The required tractive horsepower for V = 40 mph is then:

$$HP_{40} = \frac{40 \times 1{,}750}{375}$$

$$HP_{40} = 186.7$$

3.2.1.2.2. 20 MPH - Cross-Country Operation.

It is assumed that the required average tractive effort for cross-country operation is equivalent to the tractive effort required for a three percent grade:

$$TE_3 = 3{,}500 \text{ Lb}$$

The required tractive horsepower is:

$$HP_{20} = \frac{20 \times 3{,}500}{375}$$

$$HP_{20} = 186.7$$

3.2.1.2.3. Two MPH on 70 Percent Grade.

To maintain a vehicle speed of two mph on a 70 percent grade, TE_{70} = 22,550 Lb, the tractive horsepower required is:

$$HP_{2.0} = \frac{2.0 \times 22,550}{375}$$

$$HP_{2.0} = 120$$

3.2.1.3. Power Train Efficiencies.

In order to preliminarily establish required engine ratings and transmission ratios, an over-all power train efficiency has to be estimated. A preliminary power train for the LVTPX11 is considered to consist, basically, of the following units:

A. Compression-Ignition Engine
B. Transfer Gearbox
C. Steering Transmission
D. Final Drives
E. Track and Sprocket System

For preliminary calculations, the losses due to these components are estimated below. It is assumed here that the over-all efficiency includes engine accessory and auxiliary accessory losses.

3.2.1.3.1. Engine and Accessory Losses.

Here, the standard engine accessory losses are considered to include water pump, generator, air cleaner and muffler losses. Experience has shown that these losses average from six to ten percent of engine gross horsepower. The value of eight percent is assumed here for all preliminary performance estimates.

3.2.1.3.2. Hydraulic Accessory Loss.

It is assumed that for the preliminary power level and performance calculations, the hydraulic accessory loss can be expressed in percent of gross engine rating and is estimated to be 12 percent.

3.2.1.3.3. Transfer Gearbox Efficiency.

The efficiency of the transfer gearbox is estimated to be 95.5 percent.

3.2.1.3.4. Transmission Efficiency.

Based on experience with transmissions similar to the one required for the LVTPX11 vehicle, the expected transmission over-all mechanical efficiency is estimated at 90 percent.

3.2.1.3.5. Final Drive Efficiency.

The expected efficiency of a typical final drive of the type required is estimated to be 96 percent.

3.2.1.3.6. Track and Sprocket Efficiency.

The efficiency of the track and sprocket combination is estimated at 95 percent.

3.2.1.3.7. Over-all Power Train Efficiency.

The final drive efficiency and the track and sprocket efficiency are included in the rolling resistance of 100 pounds per ton and 140 pounds per ton respectively and, therefore, they are not considered for establishing the over-all efficiency. For preliminary estimates of vehicle power requirements the over-all train efficiency is 69.6 percent, as the following calculation shows:

Engine	92%
Hydraulic Accessories	88%
Transfer Gearbox	95.5%
Transmission	90%
Over-all Efficiency:	69.6%

$(.92)\ (.88)\ (.955)\ (.90) = .696$

This is straight mechanical efficiency and does not include torque converter efficiency, which varies with the converter speed ratio.

3.2.1.4. Required Prime Mover Output Power.

On the basis of the foregoing values, the required engine gross output brake horsepower is calculated for the same operating conditions as in 3.2.1.2.

3.2.1.4.1. 40 MPH on Hard Level Surface.

For the maximum speed condition of 40 mph it is assumed that the torque converter is in coupling phase or lock-up and the engine is running close to or at governor speed. The gross engine brake horsepower required is then:

$$HP_{EN} = \frac{186.7}{.696}$$

$$HP_{EN} = 268.3 \text{ BHP}$$

3.2.1.4.2. 20 MPH - Cross-Country Operation.

For the calculation of the brake horsepower to meet the 20 mph condition it is assumed that the torque converter speed ratio is .7 and the converter efficiency is 85 percent. The required gross brake horsepower is:

$$HP_{EN} = \frac{186.7}{.696 \times .85}$$

$$HP_{EN} = 315.6 \text{ BHP}$$

Assuming that the engine is running at 90 percent governor speed, an engine rated at 320 to 330 brake horsepower is necessary to meet this condition.

3.2.1.4.3. Two MPH on 70 Percent Grade.

The required engine gross output brake horsepower to provide a vehicle speed of two mph on a 70 percent grade is estimated at:

$$HP_{EN} = \frac{120}{.696 \times (.70)}$$

$$HP_{EN} = 246.3 \text{ BHP}$$

In the above formula, the value of .70 is the torque converter efficiency. The engine is pulled down by the load and running at approximately 75 to 80 percent of its rated speed. An engine with a rated output of approximately

300 to 310 brake horsepower would be capable of delivering the required 246.3 brake horsepower. From a review of available engines in the nominal 300 brake horsepower class, it is seen that the characteristics of an engine such as the Cummins V8-300 or V8-315 may be utilized for further analysis.

3.2.1.5. Power Train Ratios.

To establish a basis for evaluating vehicle performance and the suitability of possible transmissions, the optimum required over-all mechanical gearing ratios are calculated using a tentative engine selection and estimated power train efficiency.

Assumed Engine Characteristics: (See Par. 3.2.1.4.)

Gross Horsepower, Maximum	310 HP
Rated Engine Speed	3000 RPM
Torque at Rated Speed	542 Lb-Ft
Maximum Torque Gross (Assuming 12 percent torque backup)	607 Lb-Ft
Maximum Torque Net	510 Lb-Ft
Sprocket Pitch Diameter:	21.3 In.
Over-all Efficiency (See 3.2.1.3)	69.6%

3.2.1.5.1. High Range Over-all Ratio.

The minimum required over-all mechanical gearing ratio, engine to sprocket, is found to meet the maximum speed specification of 40 mph.

$$\text{Sprocket RPM at 40 MPH} = \frac{(5{,}280 \times 12)(40)}{(\pi \times 21.3)(60)} = 630 \text{ RPM}$$

$$\text{Over-all High Gear Ratio} = \frac{3000}{630} = 4.76{:}1.0 \text{ Reduction}$$

Assuming a torque converter speed ratio of 0.96 (4 percent slip), $\text{Over-all Ratio} = \frac{3000\,(0.96)}{630} = 4.58{:}1.0$ Reduction

3.2.1.5.2. Low Range Over-all Ratio.

The maximum required over-all mechanical gearing ratio, engine to sprocket is found on the basis of a tractive effort to weight ratio of one (TE/W = 1.0).

Maximum Tractive Effort Required = 35,000 Pounds

$$\text{Maximum Total Sprocket Torque} = \frac{(35{,}000)(21.3)}{12\,(2)} = 31{,}100$$

Lb-Ft

$$\text{Over-all Low Gear Ratio} = \frac{\text{Maximum Sprocket Torque}}{(\text{Maximum Engine Torque})(\text{Efficiency})}$$

$$= \frac{31{,}100}{607\,(0.696)} = 73.61{:}1.0 \text{ Reduction}$$

3.2.1.5.3. Final Drive Ratio.

Preliminary design study indicates that a final drive ratio of approximately 3.0:1.0 reduction may be used.

3.2.1.5.4. Transmission Ratios.

High Range:

Over-all Ratio = 4.58:1.0

Final Drive Ratio = 3.0:1.0

High Gear Ratio $= \frac{4.58}{3.0} = 1.53:1.0$

Low Range:

Over-all Ratio = 73.61:1.0

Final Drive Ratio = 3.0:1.0

With Assumed Torque Converter Stall Ratio = 3.5:1.0,

$$\text{Low Gear Ratio} = \frac{73.61}{(3.0)\ (3.5)} = 7.0:1.0$$

With Torque Converter Stall Ratio = 2.8:1.0,

$$\text{Low Gear Ratio} = \frac{73.61}{(3.0)\ (2.8)} = 8.76:1.0$$

With a Torque Converter Stall Ratio = 2.5:1.0,

$$\text{Low Gear Ratio} = \frac{73.61}{(3.0)\ (2.5)} = 9.82:1.0$$

3.2.2. Power Requirements - Water.

Information on power requirements for water operation will be issued as an addendum to this report on or about 1 January 1963.

3.3. Available Power.

With the foregoing calculations as a basis, a firm selection of actual power train hardware and its arrangement is made. The final power train components are:

Prime Mover	One Cummins V8-315 Compression-Ignition Engine 309 BHP at 3,000 RPM; 593 Lb-Ft at 2,150 RPM (80°F, Sea Level)
Transfer Gearbox	Borg-Warner Spur Gear Ratio 1:1
Transmission	Allison XTG-250-2 with Remote Input and Concentric Output

Over-All Mechanical Gear Ratios

Range	XTG-250 Transmission	Power Train
1st	4.18 x 1.478 x 1.44 = 8.921	29.44
2nd	4.18 x 1.00 x 1.44 = 6.036	19.92
3rd	2.24 x 1.00 x 1.44 = 3.234	10.67
4th	1.00 x 1.00 x 1.44 = 1.44	4.75
Reverse	2.24 x 2.095 x 1.44 = 6.776	22.36

Torque Converter	TC-370
Stall Torque Ratio	2.55:1

Final Drive	Two Borg-Warner Spur Gears Ratio 3.3:1 (Reduction)
Sprocket	Rear Drive Pitch Dia - 21.3 inches

3.3.1. Prime Mover Performance.

The powerplant with characteristics and performance indicated to be most favorable for powering the LVTPX11 is the Cummins Model VINE V8-315 Compression-Ignition Engine, manufactured by the Cummins Engine Company, Inc., Columbus, Indiana. The gross performance, corrected to 80°F and sea level, of the Cummins V8-315 engine is shown in figure 3-1. In establishing these performance curves the following losses were considered:

Air Cleaner Loss

Water Pump Loss

Oil Pump Loss

1-inch Hg Exhaust Loss

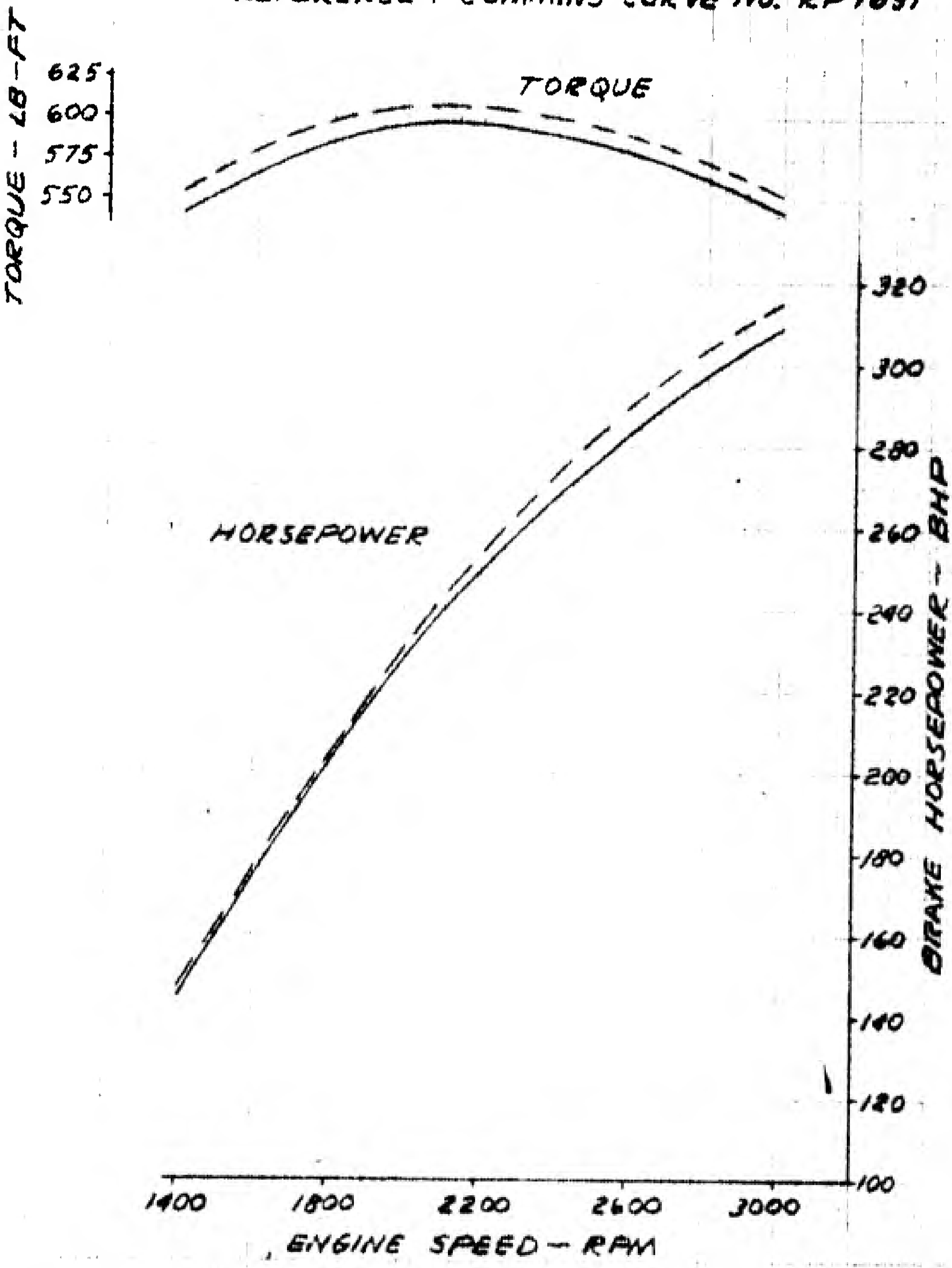
10-inch H_2O Inlet Loss

In order to properly evaluate the Cummins V8-315 engine for the LVTPX11 installation, the additional auxiliary and accessory losses must be considered in determining the available transmission input power.

INGERSOLL KALAMAZOO DIVISION BORG-WARNER CORP
CONTRACT NObs 4561 LVTPX11

ENGINE GROSS PERFORMANCE

ENGINE : CUMMINS V8-315
REFERENCE : CUMMINS CURVE NO. RP 1691



DOTTED LINE CORRECTED TO 60°F INTAKE AIR TEMPERATURE AND 29.92 IN. HG. BAROMETRIC PRESSURE (SEA LEVEL)

SOLID LINE CORRECTED TO 80°F INTAKE AIR TEMPERATURE AND 29.92 IN. HG. BAROMETRIC PRESSURE (SEA LEVEL)

9-19-62

FIGURE 3-1 CUMMINS V8-315 ENGINE GROSS PERFORMANCE

3.3.1.1. Accessory Losses.

The LVTPX11 is equipped with three accessory systems:

A. The Electrical System.

The electrical system is powered by 24-volt batteries. The charging of the batteries is done by an alternator which is driven off the engine using a speed-up-drive of 3.333 to 1 ratio. For average operating conditions, the load on the alternator is estimated to be 64 percent (equal to 80 amperes) of its full load capacity of 125 amperes. The power absorption of the alternator for this load condition is shown in the tabulation below. The values shown were obtained from the curve in figure 3-2.

Engine	Alternator	
RPM	RPM	HP-Input
1400	4668	6.1
1800	6000	6.5
2200	7334	7.1
2600	8667	7.8
3000	10000	8.6

B. The Hydraulic Accessory System.

The hydraulic components are fed by the hydraulic pump which is driven off the transfer gearbox. The ratio between pump rpm and engine rpm is a 1.25 to 1 reduction. The maximum horsepower input to the hydraulic pump is calculated below for the flow requirements of the components:

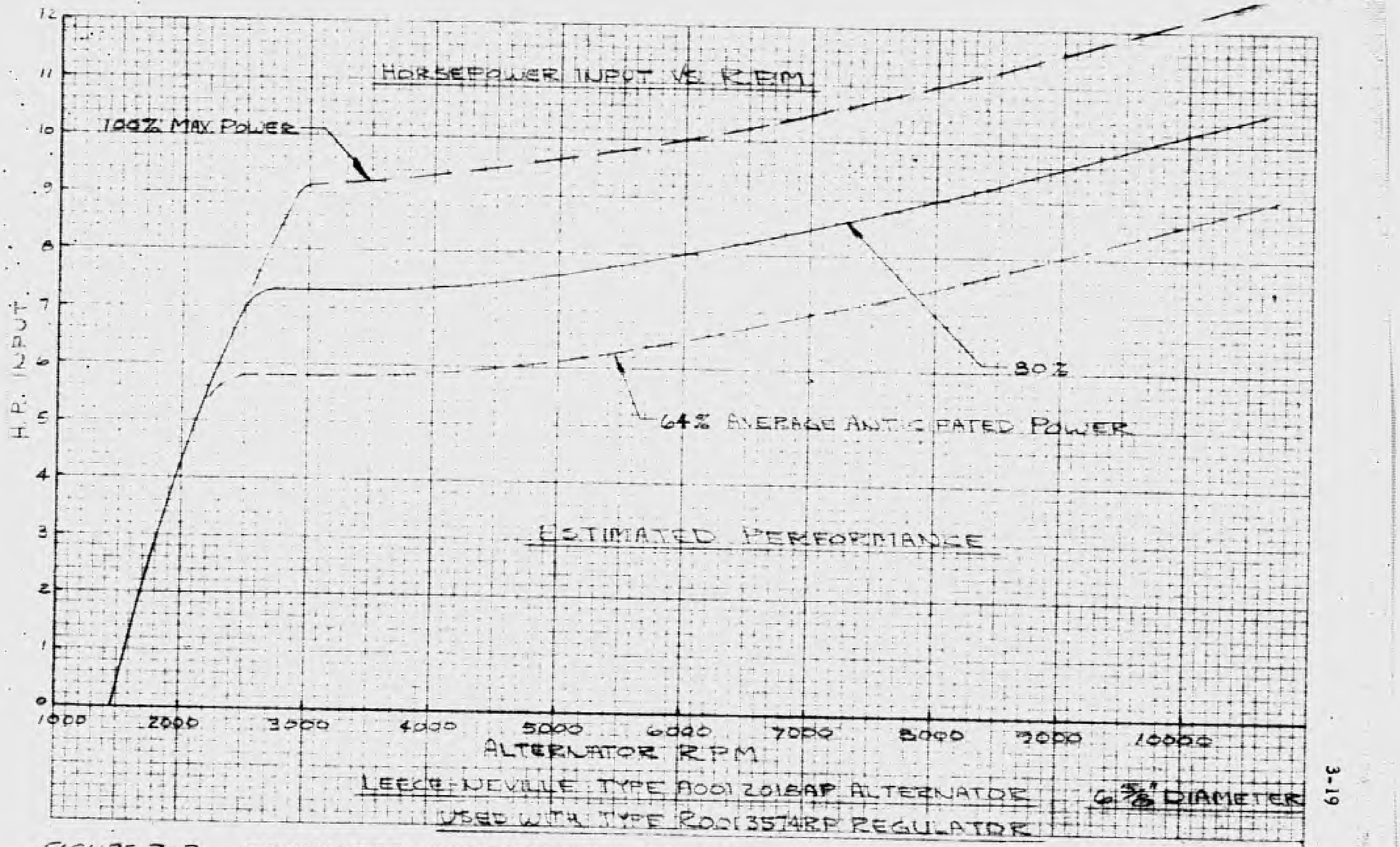


FIGURE 3-2 ESTIMATED PERFORMANCE OF LEECE – NEVILLE TYPE A0012018AP ALTERNATOR

1. Cooling Fan Motor.

The installation has two cooling fans. Each fan absorbs eight horsepower according to estimates of the cooling requirements. Each hydraulic motor requires a flow of 5.62 gpm to produce eight horsepower at 2100 rpm. These data were obtained from manufacturer's graphs.

2. Bilge Pump Motors.

The vehicle is equipped with four bilge pumps, each driven by a separate motor. They are equipped with a load-sensing device, which will throttle back the flow under no-load or land operation.

3. Bilge Pump Requirements:

No-Load Condition (Land Operation)	0.6 GPM
Full Load Condition (Water Operation)	2.6 GPM

4. Total Flow Requirements of Pump:

a. Land Operation

GPM = 2 x 5.62 + 4 x 0.6

GPM = 13.64

b. Water Operation

GPM = 2 x 5.62 + 4 x 2.6

GPM = 21.64

5. Maximum HP-Input to Main Hydraulic Pump:

$$HP = \frac{GPM \times PSI}{1714} \times \frac{1}{\eta}$$

System Pressure: 3000 PSI

Pump Efficiency: $\eta = 95\%$

a. Land Operation

$$HP = \frac{13.60 \times 3000}{1714 \times .95}$$

$$HP = 25$$

b. Water Operation

$$HP = \frac{21.64 \times 3000}{1714 \times .95}$$

$$HP = 40$$

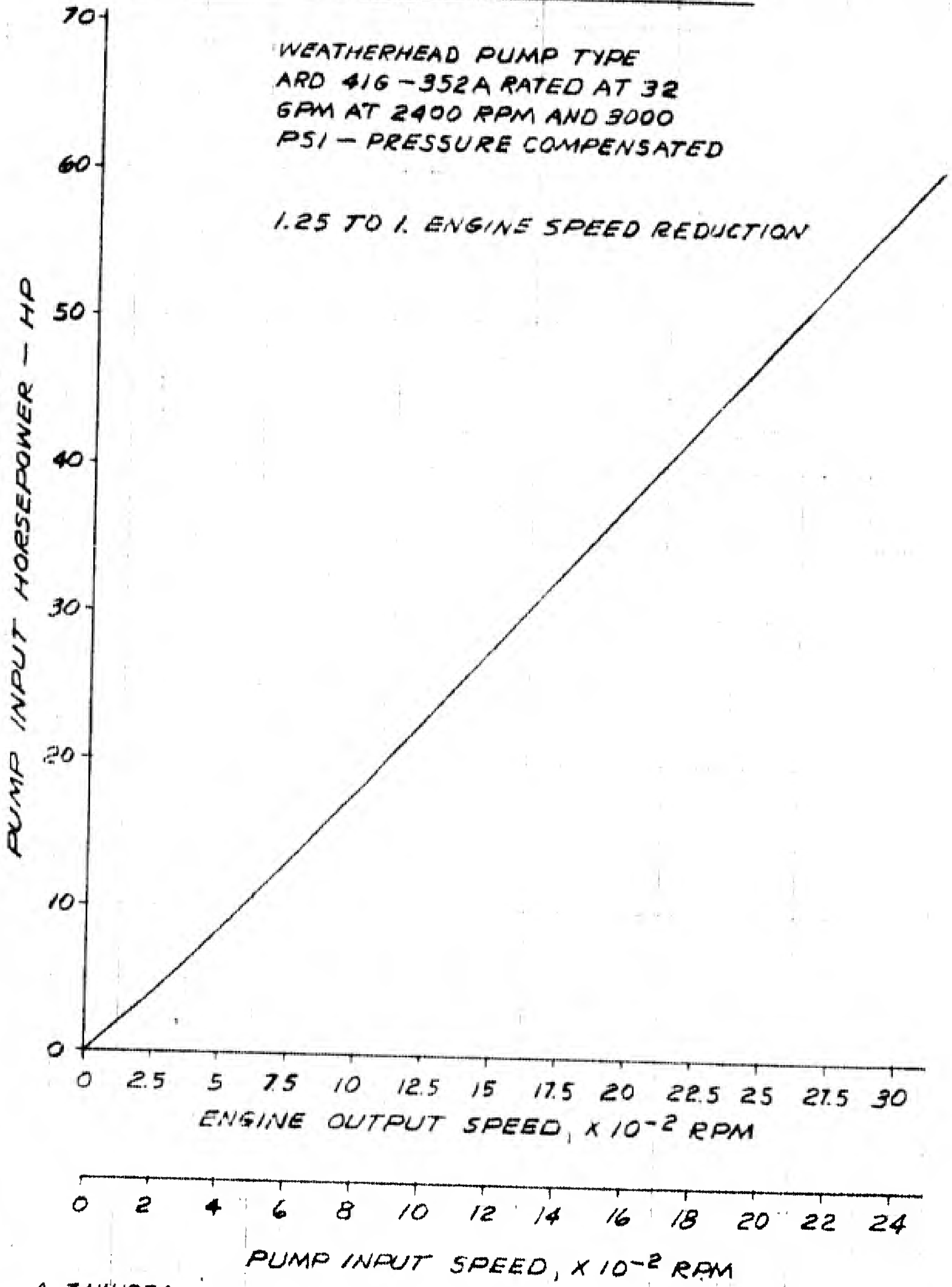
6. The hydraulic pump is pressure compensated. Its maximum horsepower input at different engine speeds is plotted in figure 3-3. From the curve in figure 3-3 it can be seen that:

   a. For land operation the horsepower input of 25 is obtained at an engine speed of 1350 rpm and is then constant up to an engine speed of 3000 rpm.

   b. For water operation the horsepower input varies with engine speed and reaches the required input of 40 horsepower at an engine speed of 2100 rpm and is then constant up to an engine speed of 3000 rpm.

INGERSOLL KALAMAZOO DIVISION BORG-WARNER CORP,
CONTRACT NO DS 4561 LVTPX11

PUMP HP vs PUMP & ENGINE RPM



A. ZAWISZA
9-17-62

FIGURE 3-3 MAIN HYDRAULIC PUMP INPUT POWER CHARACTERISTICS

The horsepower values for other engine speeds are listed below.

Engine RPM	Main Pump RPM	HP-Input
1400	1120	27
1800	1440	34
2100	1680	40
2200	1760	40
2600	2080	40
3000	2400	40

C. Lube System.

The transmission lube system is part of the transmission installation. The horsepower absorbed by this system was considered by Allison in preparing the transmission output curves. The lube system for the transfer gearbox is fed by a pump which is driven off the transfer gearbox. This pump absorbs a constant two horsepower over the whole speed range of the engine.

3.3.1.2. Available Horsepower and Torque at Transmission Input.

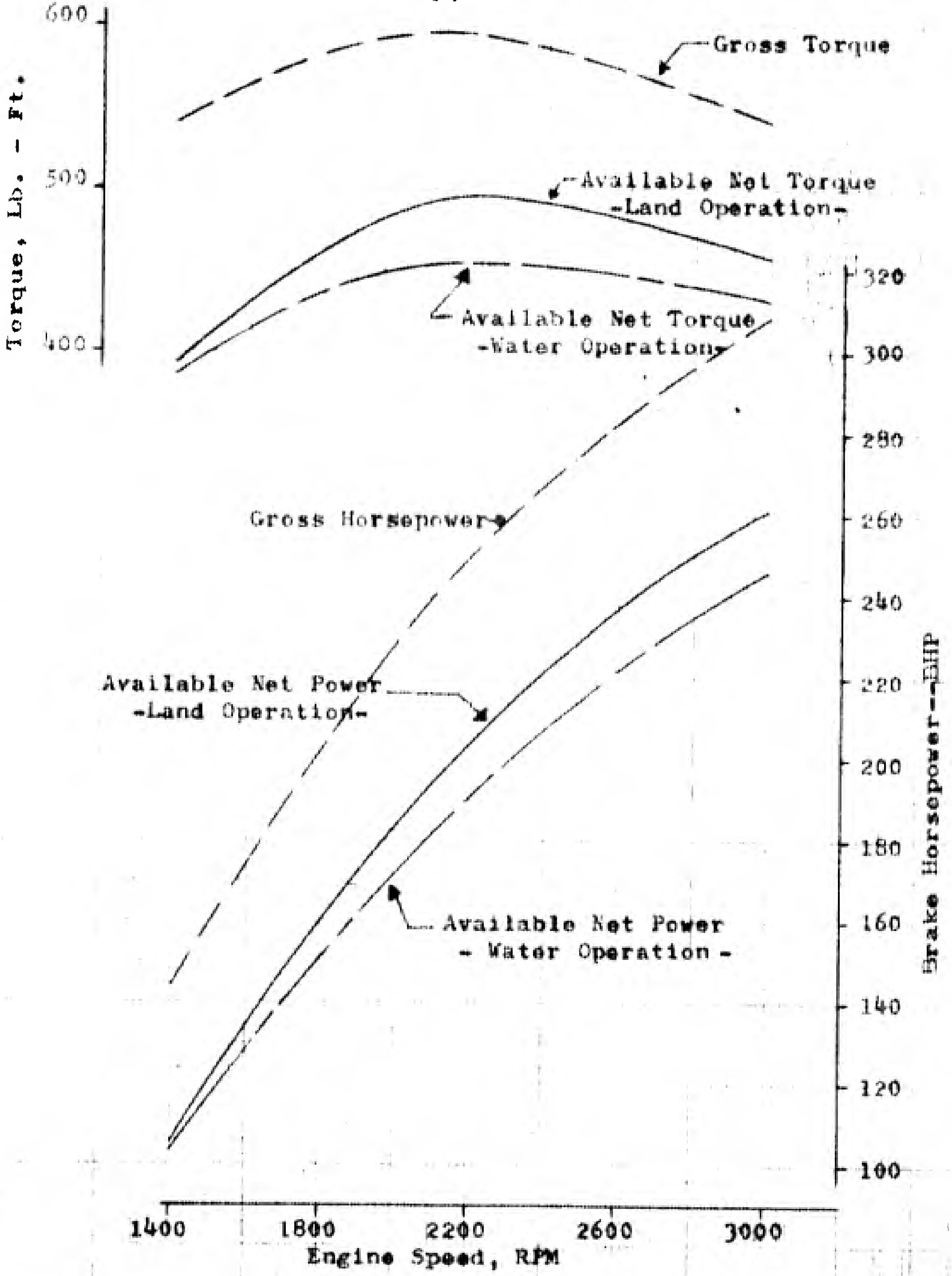
To obtain the available horsepower at the transmission input, the above determined accessory losses and the transfer gearbox loss of four percent must be subtracted from the engine gross performance. This is done in table 3-1. The resulting curves are plotted in figure 3-4.

Table 3-1. Accessory Loss Summary

Engine		Accessory Losses			Drop Gear Loss HP	Sum of All Losses HP	Transmission Input Available	
RPM	Net HP	Alternator HP	Main Pump HP	Booster Pump HP			HP	Torque Lb-Ft
LAND OPERATION								
1400	144	6.1	25	2	5.7	38.8	105.2	394
1800	200	6.5	25	2	8.0	41.5	158.5	463
2200	248	7.1	25	2	9.9	44.0	204.0	497
2600	283	7.8	25	2	11.3	46.1	236.9	480
3000	309	8.6	25	2	12.4	48.0	261.0	457
WATER OPERATION								
1400	144	6.1	27	2	5.7	40.8	103.2	387
1800	200	6.5	34	2	8.0	50.5	149.5	437
2100	240	7.0	40	2	9.6	58.6	181.4	454
2200	248	7.1	40	2	9.9	59.0	189.0	451
2600	283	7.8	40	2	11.3	61.1	221.9	448
3000	309	8.6	40	2	12.4	63.0	246.0	431

ENGINE NET PERFORMANCE AT TRANSMISSION INPUT

Engine	:	Cummins V8-315
Reference	:	Cummins Curve No. RP-1691
Conditions	:	80°F Inlet Air, Sea Level
Auxiliaries	:	Alternator, Main Hydraulic Pump, Lube Pump, Transfer Gearbox



Zawisza
9/19/62

FIGURE 3-4 ENGINE NET PERFORMANCE AT TRANSMISSION INPUT

3.3.2. Available Transmission Output.

The actual numerical efficiencies of the XTG-250 transmission are not disclosed by Allison. Rather, given a net engine performance curve, they prepare transmission output performance curves including all transmission losses and giving net output performance. By using values obtained from figure 3-4, horsepower and torque available at transmission input, the transmission performance curves, figures 3-5 and 3-6 were prepared by Allison.

3.4. Vehicle Performance Prediction.

3.4.1. Vehicle Land Performance.

As is general practice, for all final calculations of land performance, the track and sprocketplus final drive losses are considered to be included in the rolling resistance values of 100 pounds per ton on hard, level surface and 140 pounds per ton on all grades, respectively. The precise final drive ratio of 3.3 to 1, to be used with the XTG-250 transmission is established so as to meet the maximum speed specification of 40 mph, as calculated below:

Sprocket RPM (N_{SPR}) at 40 MPH:

$$N_{SPR} = \frac{(5280 \times 12)\ 40}{(\pi \times 21.3)\ 60}$$

$$N_{SPR} = 630 \text{ RPM}$$

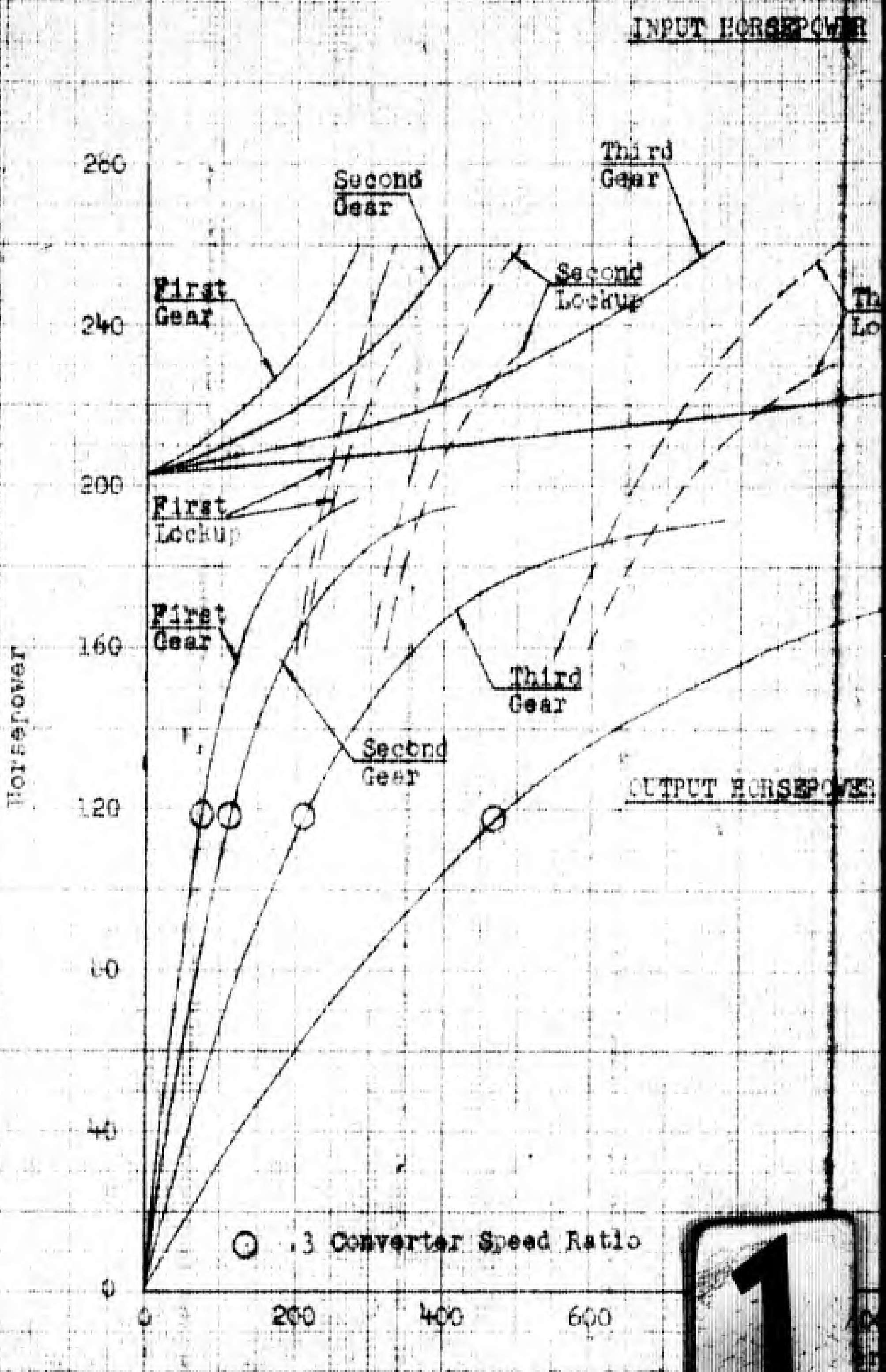
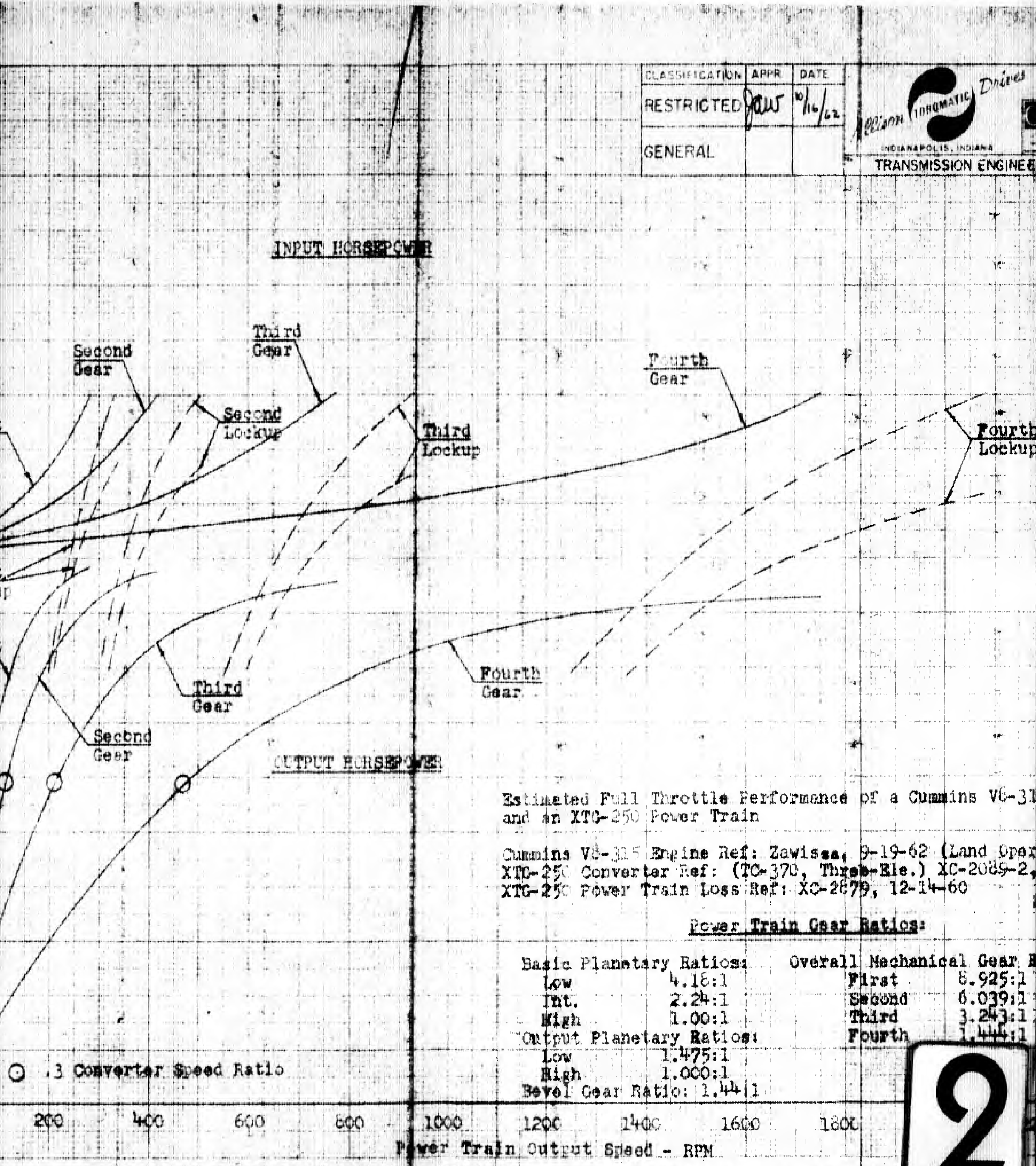


FIGURE 3-6. ESTIMATED FULL-THROTTLE PER

CLASSIFICATION	APPR	DATE
RESTRICTED	Jaw	10/16/62
GENERAL		

Allison Torqmatic Drives
INDIANAPOLIS, INDIANA
TRANSMISSION ENGINEE



⊙ .3 Converter Speed Ratio

Estimated Full Throttle Performance of a Cummins V8-31
and an XTG-250 Power Train

Cummins V8-315 Engine Ref: Zawissa, 9-19-62 (Land Oper
XTG-250 Converter Ref: (TC-370, Three-Ele.) XC-2089-2,
XTG-250 Power Train Loss Ref: XC-2879, 12-14-60

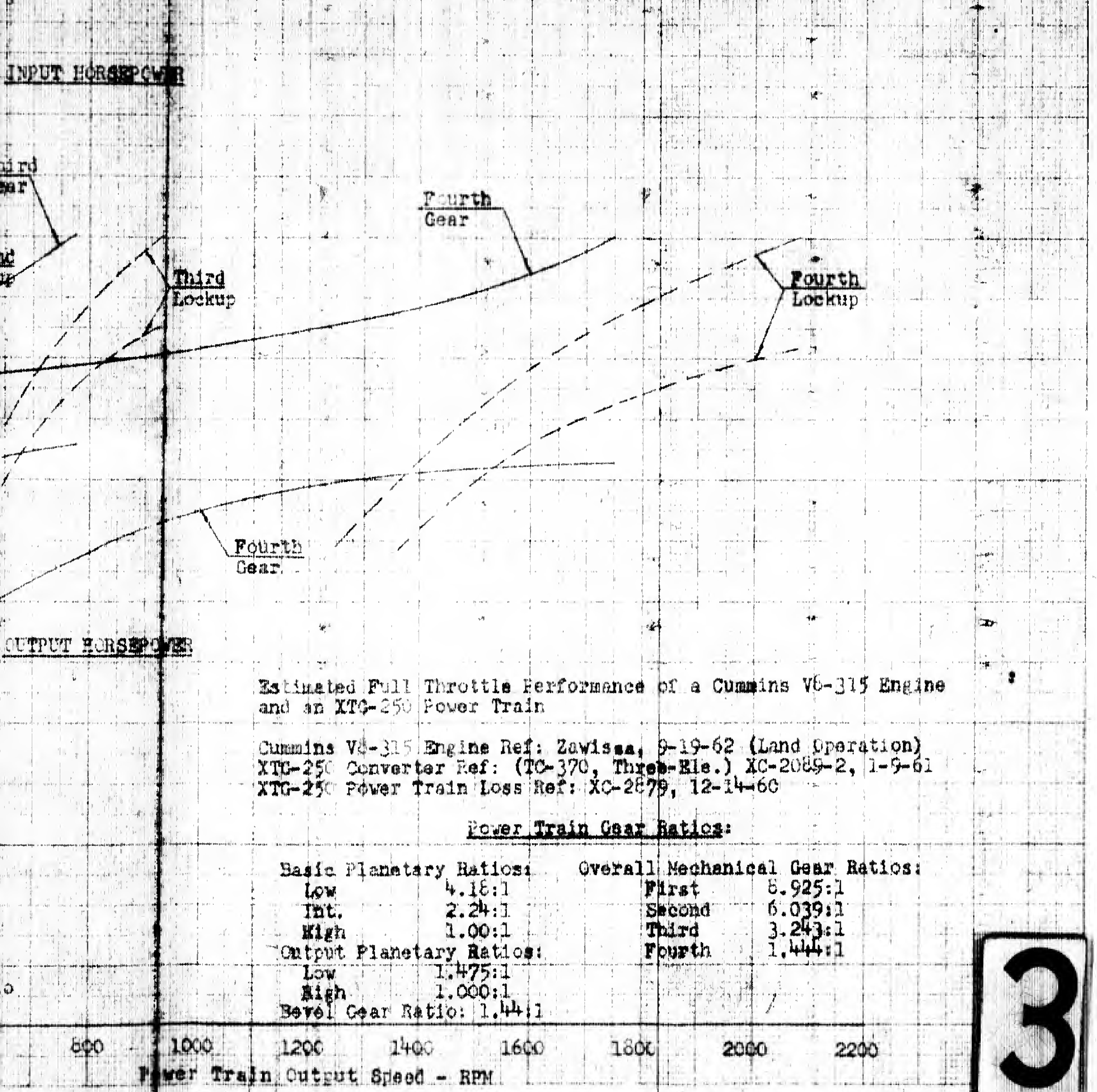
Power Train Gear Ratios:

Basic Planetary Ratios:
Low 4.18:1
Int. 2.24:1
High 1.00:1
Output Planetary Ratios:
Low 1.475:1
High 1.000:1
Bevel Gear Ratio: 1.44:1

Overall Mechanical Gear
First 8.925:1
Second 6.039:1
Third 3.243:1
Fourth 1.444:1

URE 3-6. ESTIMATED FULL-THROTTLE PERFORMANCE (HORSEPOWER) OF A CUMMINS V8-315 ENGINE AND AN ALLISON XTG-250

CLASSIFICATION	APPR	DATE	Allison Torqmatic Drives INDIANAPOLIS, INDIANA	DATE 10-15-62 BY Nigh	TC 7490 Sht.2 of 2
RESTRICTED	Jaw	10/16/62	TRANSMISSION ENGINEERING DEPARTMENT		
GENERAL					



Estimated Full Throttle Performance of a Cummins V8-315 Engine and an XTG-250 Power Train

Cummins V8-315 Engine Ref: Zawissa, 9-19-62 (Land Operation)
XTG-250 Converter Ref: (TC-370, Three-Ele.) XC-2089-2, 1-9-61
XTG-250 Power Train Loss Ref: XC-2879, 12-14-60

Power Train Gear Ratios:

Basic Planetary Ratios:		Overall Mechanical Gear Ratios:	
Low	4.18:1	First	8.925:1
Int.	2.24:1	Second	6.039:1
High	1.00:1	Third	3.243:1
Output Planetary Ratios:		Fourth	1.44:1
Low	1.475:1		
High	1.000:1		
Bevel Gear Ratio:	1.44:1		

E PERFORMANCE (HORSEPOWER) OF A CUMMINS V8-315 ENGINE AND AN ALLISON XTG-250 TRANSMISSION

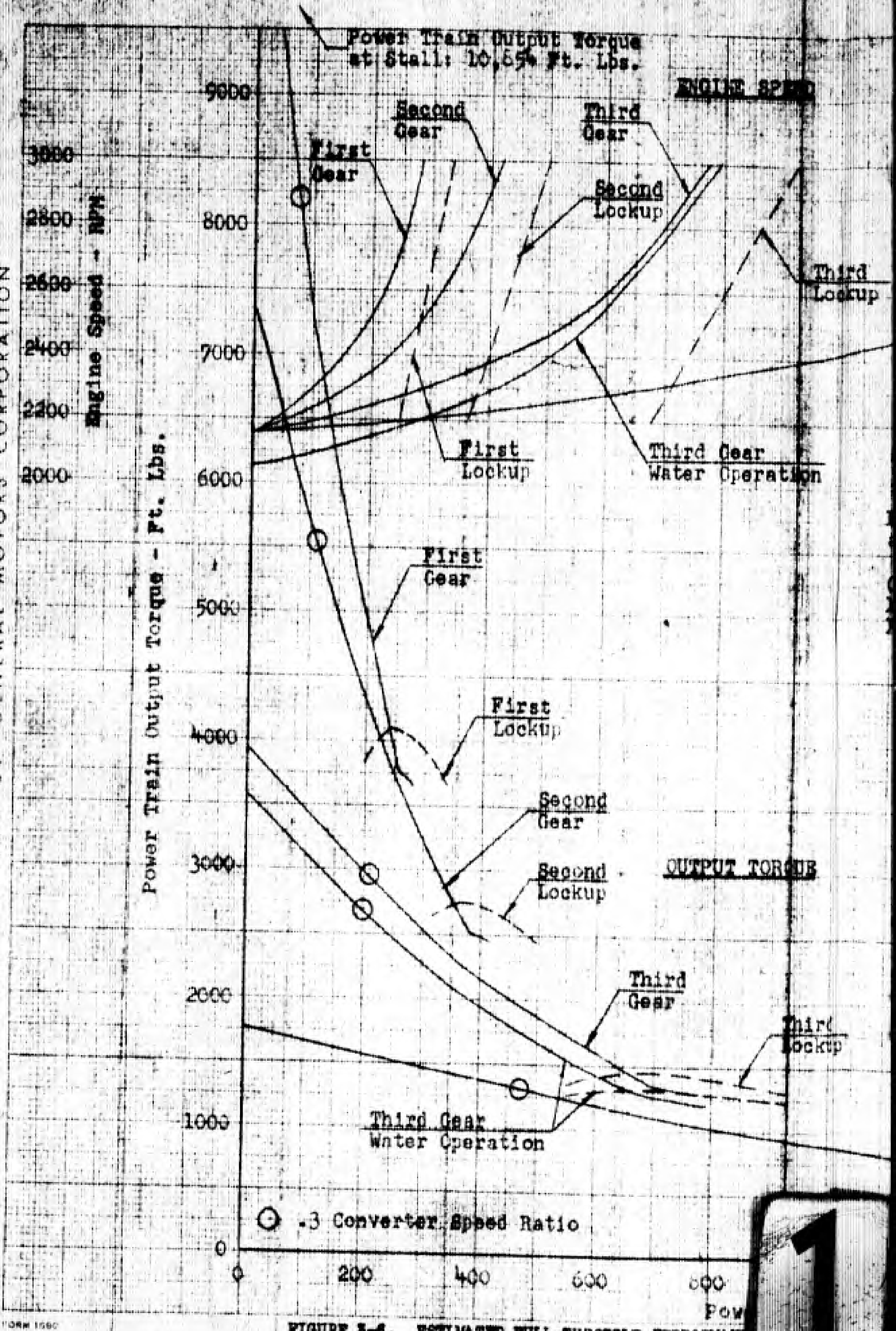


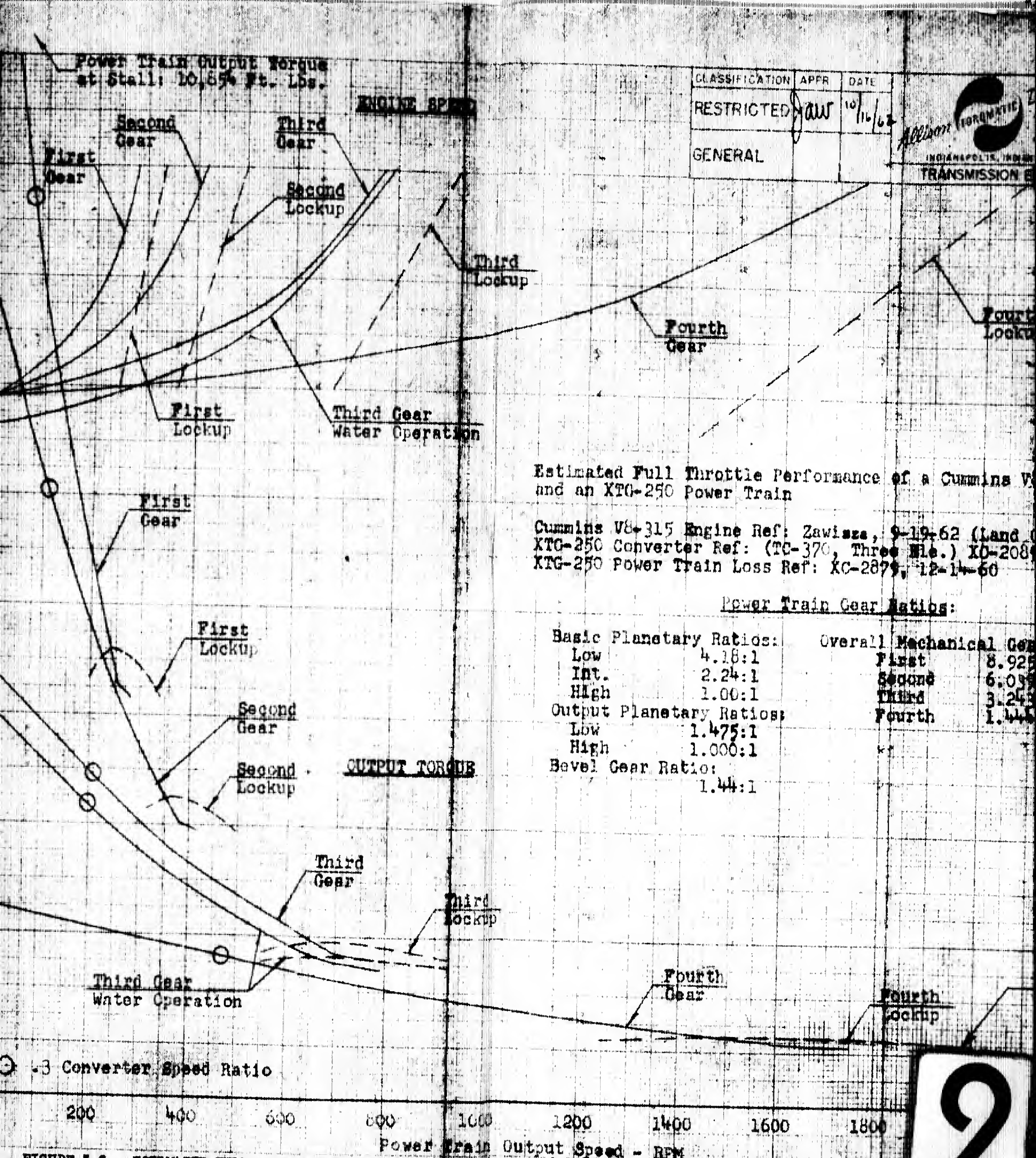
FIGURE 3-6. ESTIMATED FULL-THROTTLE PERFORMAN

CLASSIFICATION	APPR	DATE
RESTRICTED	jaw	10/16/62
GENERAL		

Allison TORQMATIC

INDIANAPOLIS, INDI

TRANSMISSION



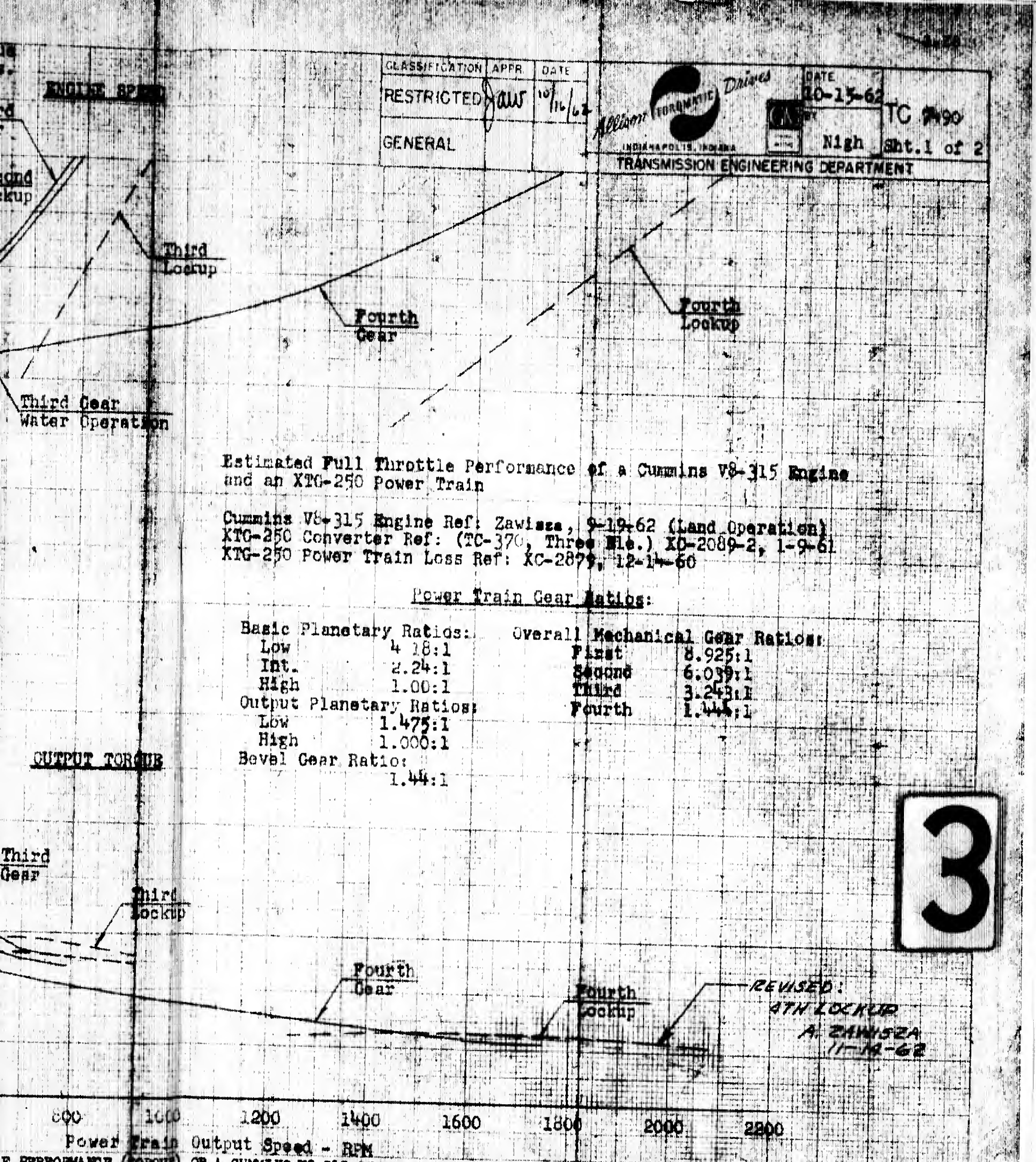
Estimated Full Throttle Performance of a Cummins V
and an XTG-250 Power Train

Cummins V8-315 Engine Ref: Zawisza, 9-19-62 (Land
XTG-250 Converter Ref: (TC-370, Three Ele.) XO-208
XTG-250 Power Train Loss Ref: XC-2879, 12-14-60

Power Train Gear Ratios:

Basic Planetary Ratios:		Overall Mechanical Ge	
Low	4.18:1	First	8.92
Int.	2.24:1	Second	6.09
High	1.00:1	Third	3.24
Output Planetary Ratios:		Fourth	1.44
Low	1.475:1		
High	1.000:1		
Bevel Gear Ratio:			
	1.44:1		

FIGURE 5-6. ESTIMATED FULL-THROTTLE PERFORMANCE (TORQUE) OF A CUMMINS V8-315 ENGINE AND AN ALLISON XTG-2



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GENERAL		

Allison Torqmatic Drives
INDIANAPOLIS, INDIANA
DATE 10-15-62
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Estimated Full Throttle Performance of a Cummins V8-315 Engine and an XTG-250 Power Train

Cummins V8-315 Engine Ref: Zawisza, 9-19-62 (Land Operation)
XTG-250 Converter Ref: (TC-370, Three Ele.) XD-2089-2, 1-9-61
XTG-250 Power Train Loss Ref: XC-2879, 12-14-60

Power Train Gear Ratios:

Basic Planetary Ratios:		Overall Mechanical Gear Ratios:	
Low	4.18:1	First	8.925:1
Int.	2.24:1	Second	6.039:1
High	1.00:1	Third	3.243:1
Output Planetary Ratios:		Fourth	1.444:1
Low	1.475:1		
High	1.000:1		
Bevel Gear Ratio:	1.44:1		

REVISED: 4TH LOCKUP
A. ZAWISZA
11-14-62

3

E PERFORMANCE (TORQUE) OF A CUMMINS V8-315 ENGINE AND AN ALLISON XTG-250 TRANSMISSION

Maximum Transmission Output Speed (N_{TR}), with Converter Locked:

$$N_{TR} = 2080 \text{ RPM}$$

Final Drive Ratio (M):

$$M = \frac{N_{TR}}{N_{SPR}} = \frac{2080}{630}$$

$$M = 3.30:1$$

3.4.1.1. Tractive Effort and Gradability.

3.4.1.1.1. The land performance is calculated from the Allsion power train output curves noted in paragraph 3.3.2. above. The tractive effort is obtained with the following formula:

$$TE = \frac{T \times M \times 24}{D}$$

Where T is the output torque, in pound-feet, M is the final drive ratio, and D is the pitch diameter of the sprocket, in inches.

$$M = 3.30:1$$

$$D = 21.3 \text{ inches}$$

$$TE = \frac{T\ 3.3 \times 24}{21.3}$$

$$TE = 3.72\ T \text{ pounds}$$

3.4.1.1.2. Vehicle speed is obtained from the formula:

$$V = \frac{N \times \pi \times D \times 60}{M \times 5{,}280 \times 12}$$

Where N is the power train output speed.

$$V = N \frac{\pi \times 2.13 \times 60}{5{,}280 \times 12 \times 3.3}$$

$$V = \frac{N}{52.10} = .019\ N\ \text{MPH}$$

3.4.1.1.3. The result of these computations (table 3-2) is shown in figure 3-7. This is the full throttle performance with unit rolling resistance of 100 pounds per ton for hard level terrain and 140 pounds per ton for all grades.

3.4.1.2. Vehicle Land Stability - Ninety Degree Turn on 60 Percent Slope.

3.4.1.2.1. The stability of a vehicle negotiating a slope is determined by the moments due to height and acceleration around the assumed tipping point. In practice, the moments due to acceleration are neglected and stability is determined for static condition only. Therefore it is only necessary to know the location of the center of gravity of the vehicle to determine the stability margin.

3.4.1.2.2. The relationship of the center of gravity to the centerline of the road-track wheels tipping point is 5.68 feet in the vehicle side elevation, figure 3-8. In the end view of the vehicle, the center of gravity is 5 feet 3 inches to the outermost point of the road track tipping point; therefore, the stability is less in this transverse plane, due to the shorter ground contact base.

Table 3-2. Vehicle Performance - Power Train

Transmission Output		Engine	Final Drive Output				
Speed RPM	Torque Lb-Ft	Speed RPM	Speed RPM	Total Torque Lb-Ft	Total Torque Effort Lbs	Speed MPH	Horsepower
			FIRST GEAR				
0	10,854	2,157	0	35,800	40,400	0	0
54	9,000		16.4	29,650	33,500	1.025	92.5
80	8,000		24.2	26,400	29,800	1.52	121.6
110	7,000	2,320	33.35	23,100	26,000	2.09	146.2
150	6,000		45.5	19,800	22,300	2.85	171.1
197	5,000		59.6	16,500	18,600	3.74	187.2
255	3,770		77.4	12,420	14,000	4.85	182.6
280	3,670	3,000	84.9	12,040	13,650	5.32	196.5
			FIRST GEAR LOCKUP				
200	3,840		60.6	12,680	14,300	3.80	146.2
240	4,100	2,200	72.7	13,500	15,250	4.56	187.2
305	3,900		92.5	12,880	14,500	5.8	227.
340	3,650	3,000	103.	12,040	13,600	6.46	236.
			SECOND GEAR				
0	7,350	2,157	0	24,200	27,360	0	0
70	6,200		21.2	20,450	23,000	1.33	82.6
80	6,000		24.22	19,800	22,300	1.52	91.4
149	5,000		45.1	16,500	18,600	2.83	142.
225	4,000		68.3	13,200	14,900	4.28	171.4
324	3,000		98.3	9,900	11,150	6.16	185.
380	2,530		115.	8,350	9,410	7.23	183.
415	2,470	3,000	126.	8,090	9,190	7.89	195.2
			SECOND GEAR LOCKUP				
318	2,650		96.5	8,420	9,860	6.05	160.4
366	2,750	2,200	111.	9,090	10,250	6.95	191.
435	2,650		132.	8,420	9,860	8.27	219.
495	2,480	3,000	150.	8,090	9,230	9.41	234.

Table 3-2. Vehicle Performance - Power Train (Continued)

Transmission Output		Engine	Final Drive Output				
Speed RPM	Torque Lb-Ft	Speed RPM	Speed RPM	Total Torque Lb-Ft	Total Torque Effort Lbs	Speed MPH	Horse-power
			THIRD GEAR				
0	3,930	2,157	0	12,960	14,600	0	0
196	3,000		59.4	9,900	11,140	3.72	112.
310	2,500	2,320	94.	8,250	9,300	5.9	147.4
360	2,300		109.	7,600	8,550	6.85	157.5
450	2,000		136.3	6,600	7,440	8.55	171.
640	1,500		194.	4,950	5,590	12.15	182.4
700	1,350		212.	4,460	5,020	13.3	179.5
760	1,325	3,000	230.	4,290	4,840	14.41	191.5
			THIRD GEAR LOCKUP				
550	1,350		166.7	4,460	5,020	10.44	141.
660	1,450	2,200	200.	4,790	5,400	12.52	182.
700	1,460		212.	4,820	5,440	13.3	194.
928	1,305	3,000	281.	4,290	4,910	17.6	230.5
			FOURTH GEAR				
0	1,760	2,157	0	5,810	6,550	0	0
200	1,570		60.5	5,190	5,840	3.8	59.8
400	1,380		121.	4,550	5,140	7.6	105.
710	1,100	2,320	212.	3,630	4,100	13.46	148.6
800	1,020		242.5	3,360	3,800	15.2	155.
1200	780		364.	2,575	2,900	22.8	178.
1600	580		485.	1,913	2,160	30.4	176.7
1750	556	3,000	530.	1,815	2,066	33.24	185.1
			FOURTH GEAR LOCKUP				
1245	600		377.	1,980	2,230	23.65	142.1
1550	650	2,200	470.	2,142	2,420	29.45	192.
1800	600		545.	1,980	2,230	34.2	205.6
2078	560	3,000	630.	1,850	2,080	39.9	223.

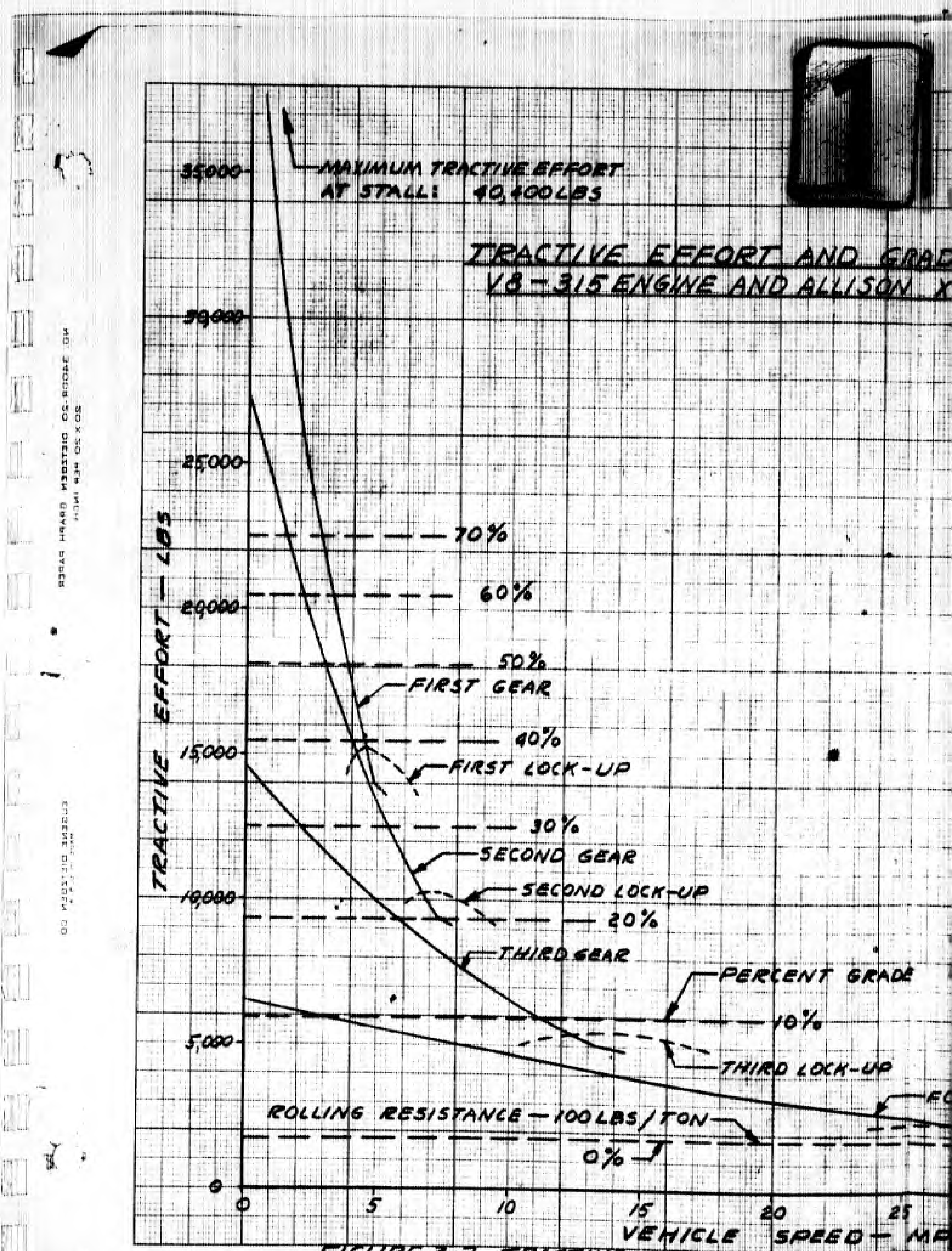
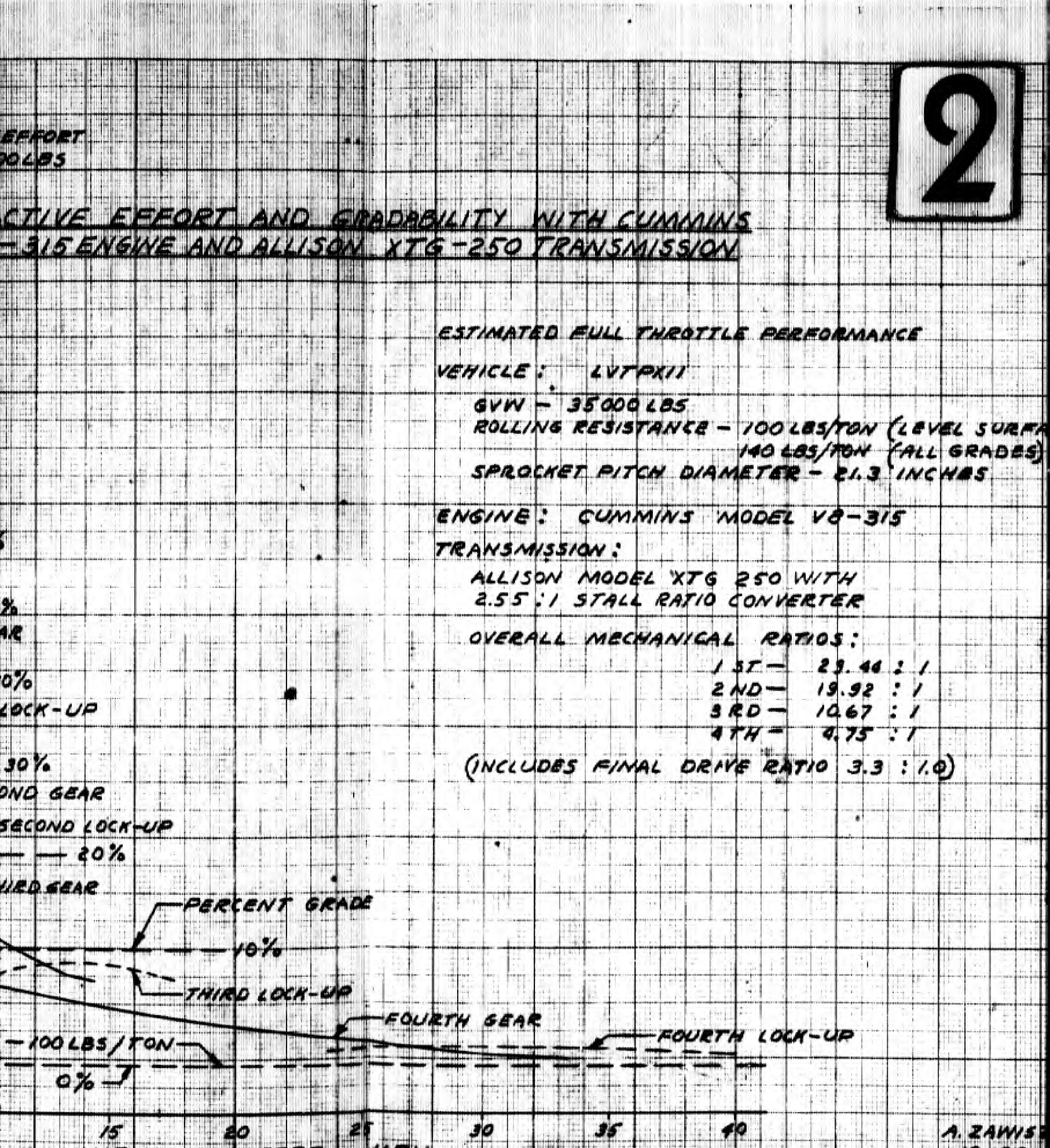


FIGURE 3-7 TRACTIVE EFFORT AND GRADAB
ENGINE AND AN ALLISON XT



ESTIMATED FULL THROTTLE PERFORMANCE

VEHICLE: LVTPXII

GVW – 35000 LBS
ROLLING RESISTANCE – 100 LBS/TON (LEVEL SURF
140 LBS/TON (ALL GRADES)
SPROCKET PITCH DIAMETER – 21.3 INCHES

ENGINE: CUMMINS MODEL V8-315

TRANSMISSION:

ALLISON MODEL XTG 250 WITH
2.55 : 1 STALL RATIO CONVERTER

OVERALL MECHANICAL RATIOS:

1 ST – 23.44 : 1
2 ND – 19.92 : 1
3 RD – 10.67 : 1
4 TH – 4.75 : 1

(INCLUDES FINAL DRIVE RATIO 3.3 : 1.0)

RACTIVE EFFORT AND GRADABILITY WITH A CUMMINS V8-315
GINE AND AN ALLISON XTG-250Z TRANSMISSION

RT AND GRADABILITY WITH CUMMINS
AND ALLISON XTG-250 TRANSMISSION

ESTIMATED FULL THROTTLE PERFORMANCE

VEHICLE: LVTPXII

GVW - 35000 LBS
ROLLING RESISTANCE - 100 LBS/TON (LEVEL SURFACE)
140 LBS/TON (ALL GRADES)
SPROCKET PITCH DIAMETER - 21.3 INCHES

ENGINE: CUMMINS MODEL V8-315

TRANSMISSION:

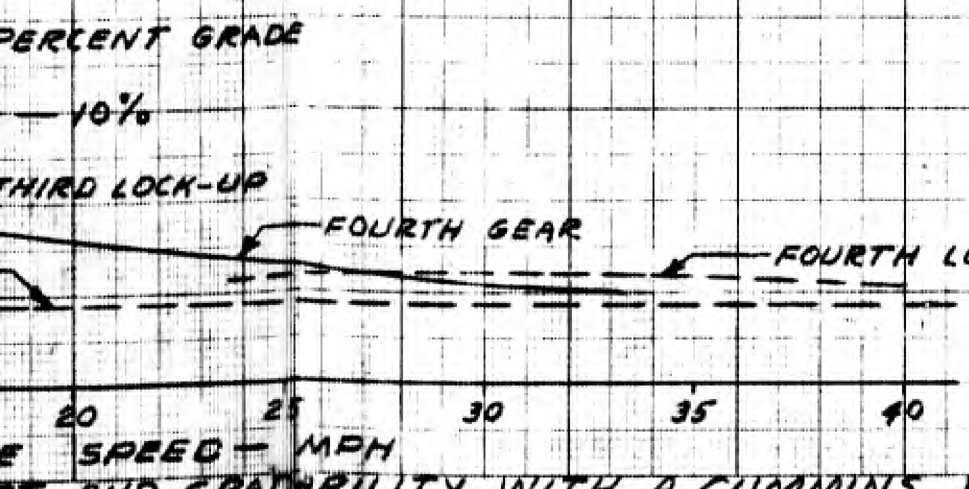
ALLISON MODEL XTG 250 WITH
2.55:1 STALL RATIO CONVERTER

OVERALL MECHANICAL RATIOS:

1ST - 29.44 : 1
2ND - 19.92 : 1
3RD - 10.67 : 1
4TH - 4.75 : 1

(INCLUDES FINAL DRIVE RATIO 3.3 : 1.0)

3



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ET AND GRADABILITY WITH A CUMMINS V8-315
N ALLISON XTG-2502 TRANSMISSION

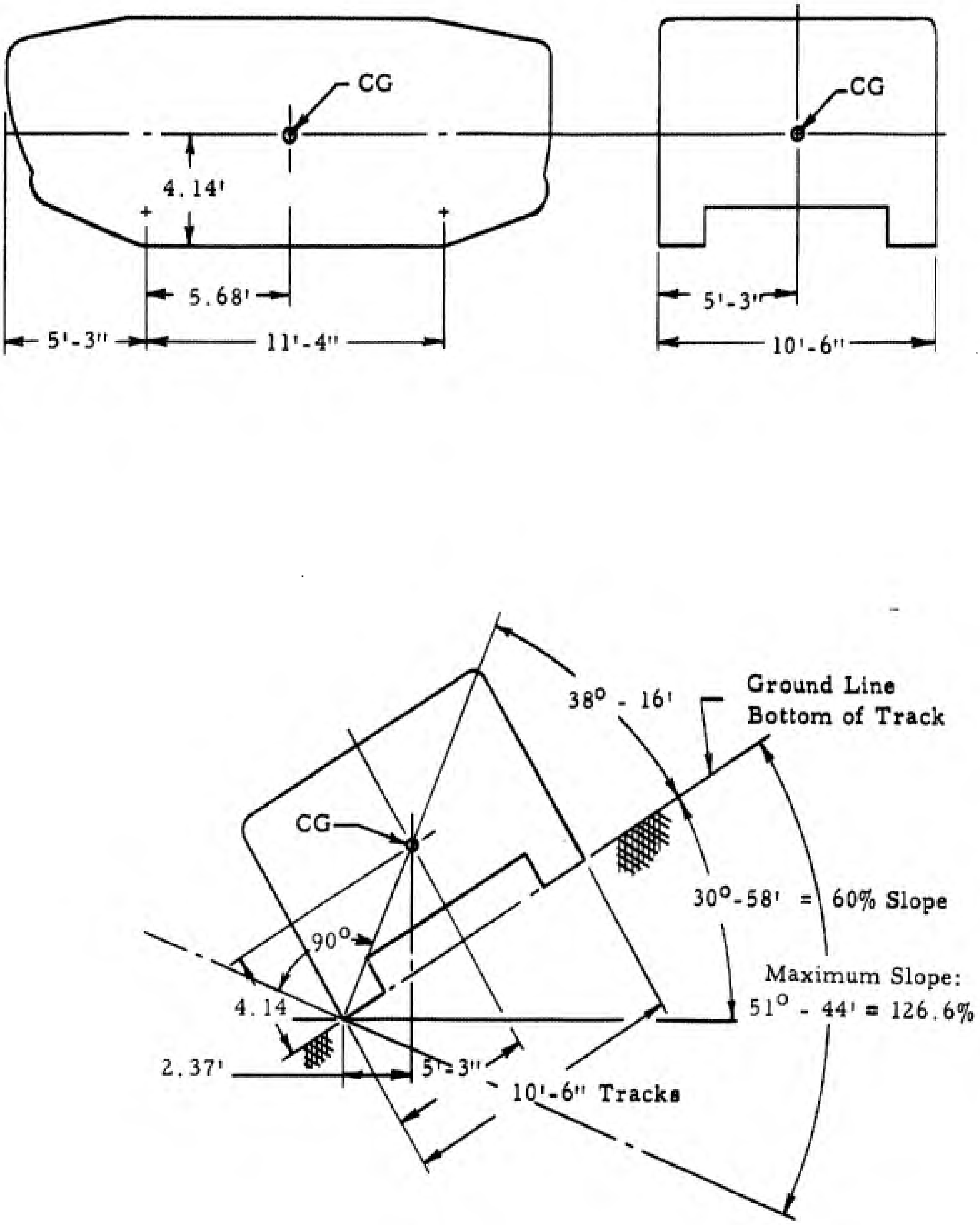


Figure 3-8. Vehicle Stability on Land

3.4.1.2.3. In figure 3-8, with the vehicle shown on a 60 percent grade and the vehicle shown in the less stable plane, it is evident that, as the vehicle turns on the slope, the horizontal relationship of the center of gravity to the lower, extreme point of ground contact will increase as will the vehicle stability. As the slope is increased, with the resulting decrease in the relationship of the center of gravity to the lower, extreme ground contact point, the stability will decrease until the center of gravity is directly above the lower, extreme ground contact point. At this time the vehicle will be in equilibrium and any further increase in slope would result in tipping. As shown in figure 3-8, the angle measured between the slope and a line through the lower, extreme ground contact point and center of gravity is 38 degrees 16 minutes, i.e., theoretically the vehicle can negotiate a 126.6 percent side slope without tipping over. The vehicle then has a stability margin of:

$$\left(1 - \frac{60}{126.6}\right) 100 = 52.6 \text{ percent}$$

3.4.1.2.4. The prevailing ground conditions, however, will be the determining factor as to the maximum slope on which the vehicle may operate without slipping.

3.4.1.3. Vehicle Obstacle Climbing and Trench Crossing Ability.

3.4.1.3.1. Based on past tracked vehicle experience and performance tests, obstacle climbing ability is determined by a combination of conditions, such as sprocket height and projection, location of front roadwheel, and an approximate point of contact of the top of a reasonably solid obstacle such as a concrete block or a wooden log bunker, with the hull. This point of contact is located approximately on a vertical line tangent with the forward-most projecting point of the track on the sprocket to where it contacts the hull. It is also dependent upon the amount of tractive force available. The tractive force and hence obstacle climbing ability is dependent upon prevailing ground conditions, and an increasing loss of track on the ground as the vehicle progresses up the obstacle. With the foregoing conditions in mind, it is estimated that this vehicle will climb a 36-to 40-inch high solid obstacle and a 38-to 44-inch earthen obstacle (figure 3-9), since the vehicle hull will displace and move a certain portion of the upper part of an earthen obstacle.

3.4.1.3.2. The trench-crossing ability due to center of gravity location, at slow speeds in unloaded and loaded conditions will be 8-1/2 to 9 feet and 9 to 9-1/2 feet, respectively. Illustrations of the vehicle trench-crossing ability are shown in figures 3-10 and 3-11.

3.4.2. Vehicle Water Performance.

Information on vehicle water performance will be issued as an addendum to this report on or about 1 January 1963.

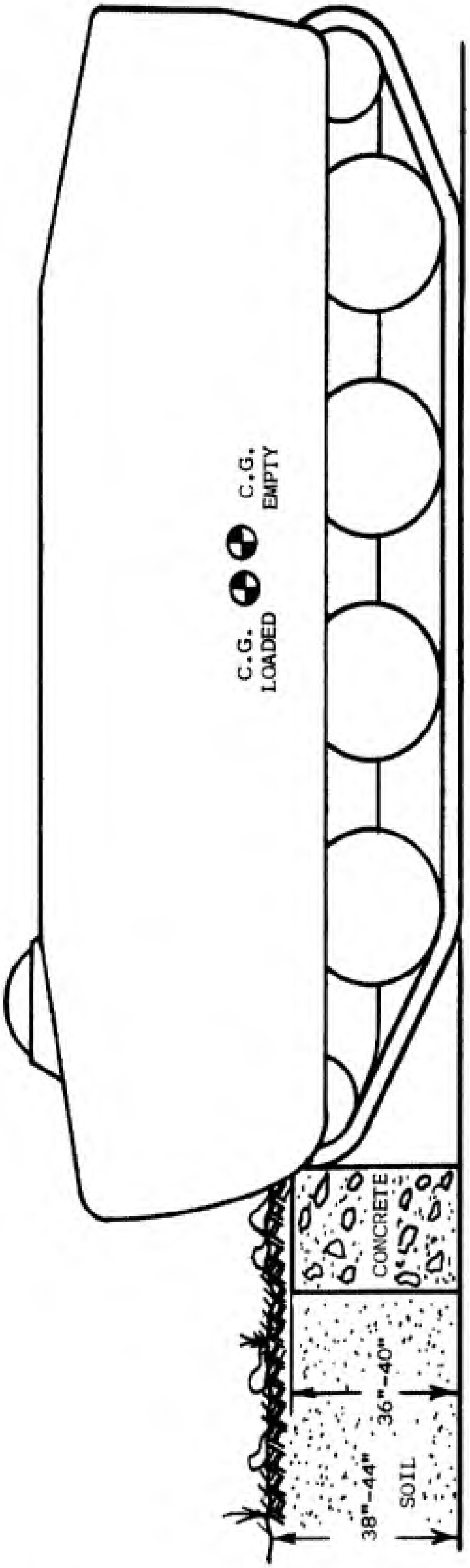


Figure 3-9. Vehicle Obstacle Climbing Ability

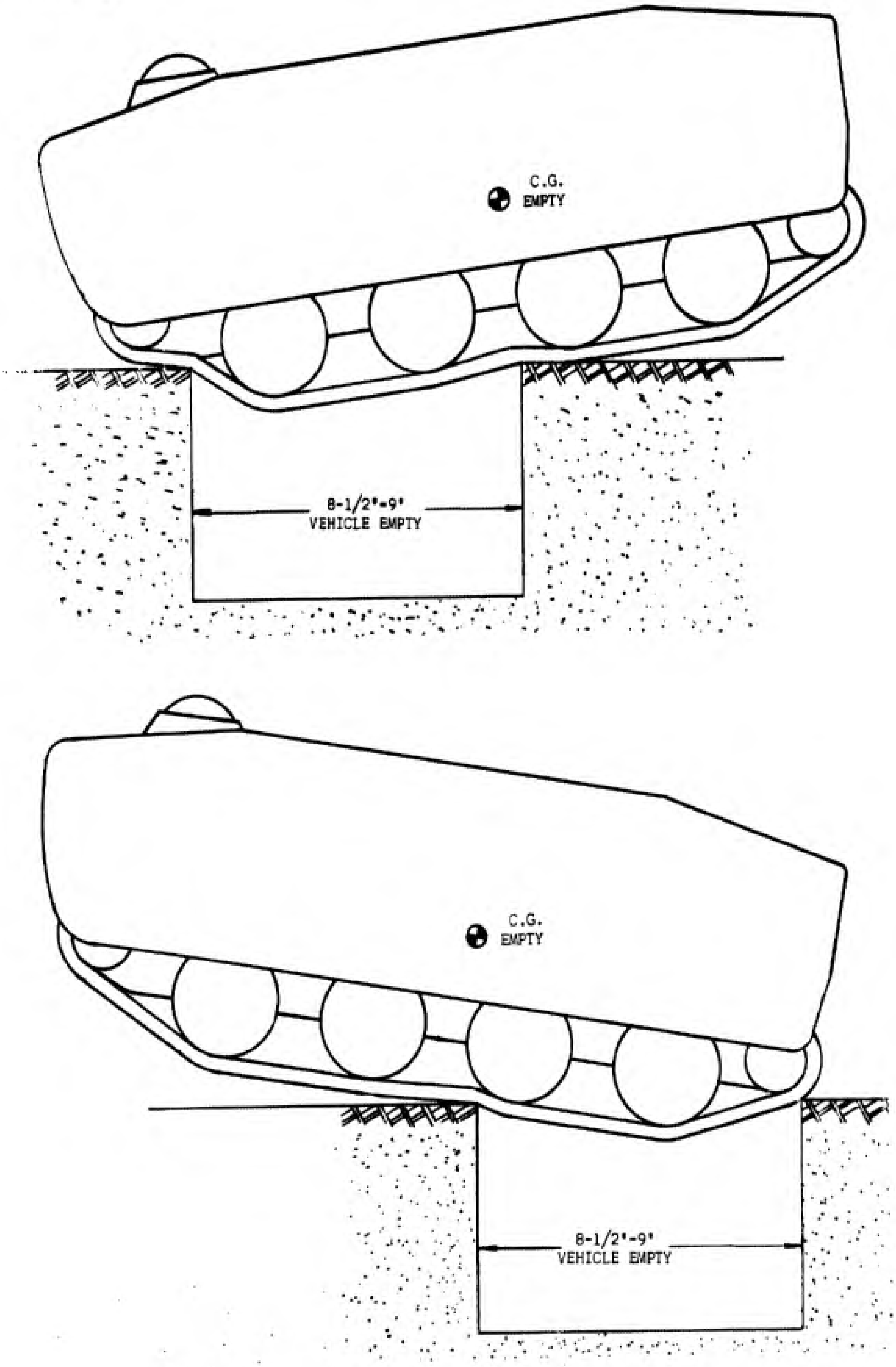


Figure 3-10. Vehicle Trench Crossing Ability - Empty

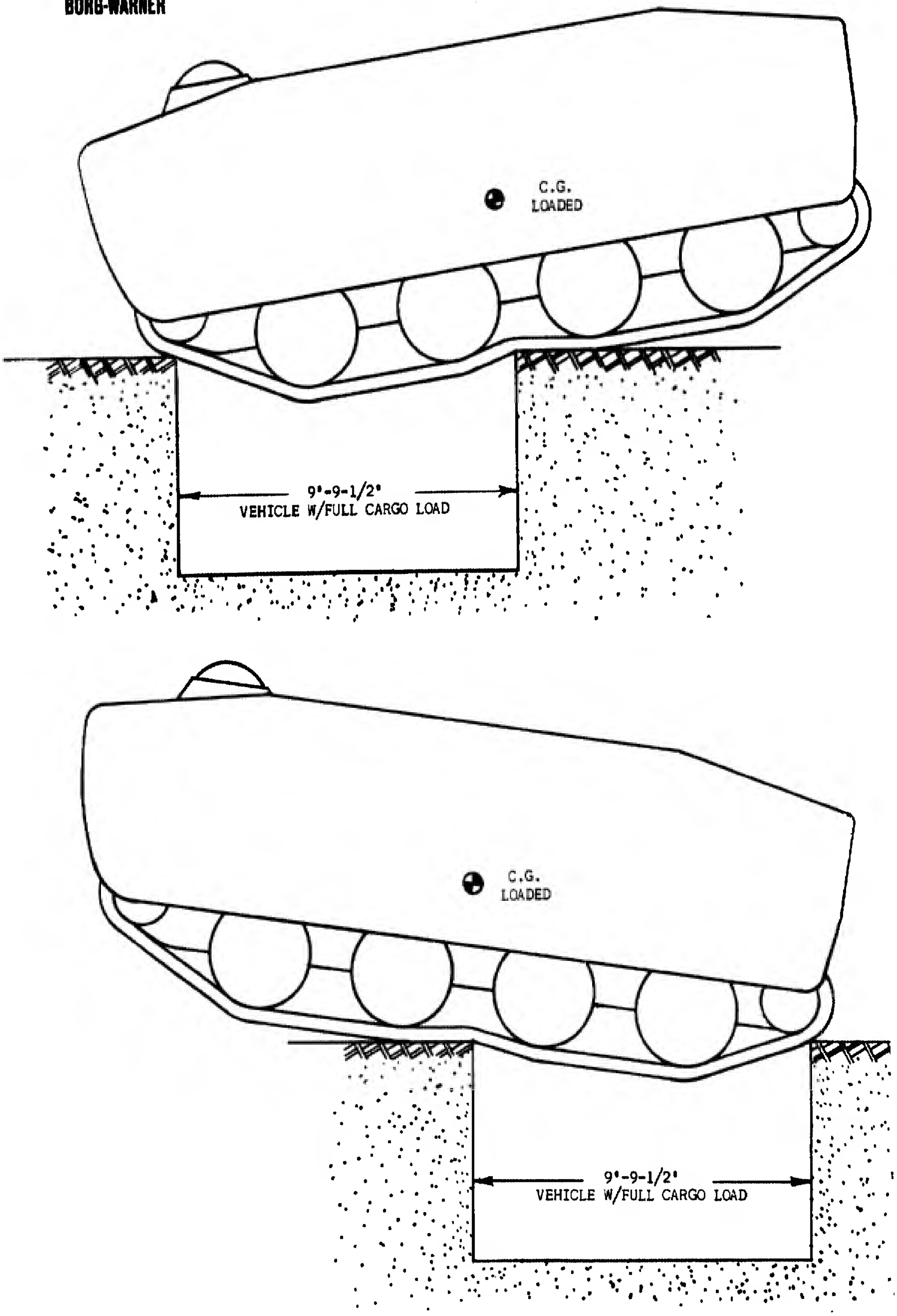


Figure 3-11. Vehicle Trench Crossing Ability - Loaded

3.5. Fuel Consumption Analysis.

Figure 3-12 shows the part-throttle fuel consumption of the Cummins V8-315 engine operated on diesel fuel. This information was furnished by the engine manufacturer. The full-throttle fuel consumption curve, figure 3-13, was derived from these data, and both are used for the calculations below. No fuel consumption data were available for the engine, if operated on other than diesel fuel. The consumption of the other fuels, JP-4, JP-5, and CIE-fuel, was estimated. For this estimate, it was assumed that the Btu per hour delivered to the engine is constant. The fuel consumption is calculated below for the following fuels:

FUEL	LB PER GAL	LOWER HEATING VALUE BTU PER LB
Diesel	6.96	19,300
JP-4	6.60	18,400
JP-5	6.79	18,300
CIE-Fuel	6.65	18,500

3.5.1. Fuel Capacity.

The LVTPX11 has a fuel capacity of 180 gallons (see paragraph 9.3). The fuel capacity, in pounds, is therefore:

A. Diesel Fuel:

180 x 6.96 = 1252.8 Lb

B. JP-4 Fuel:

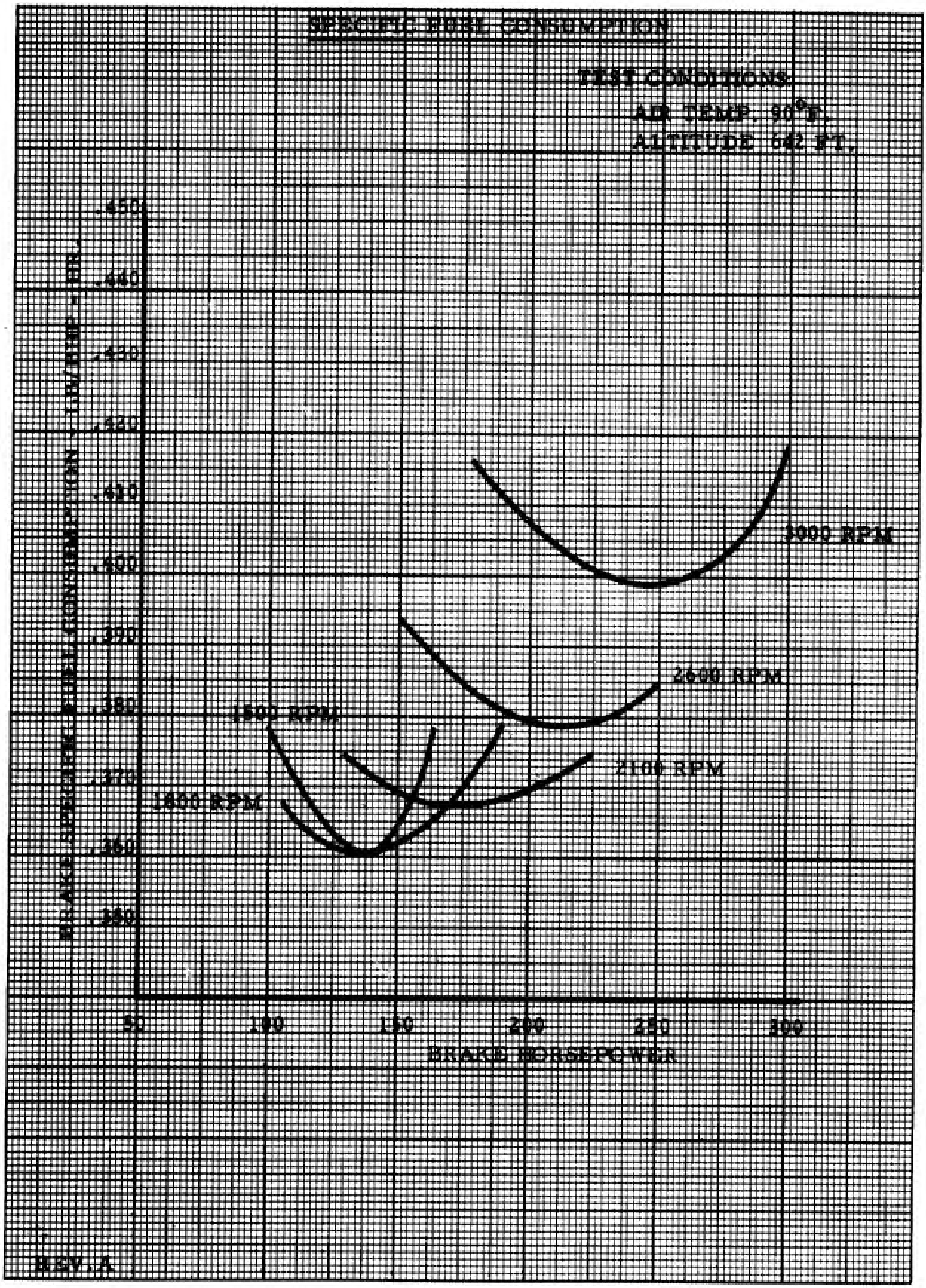
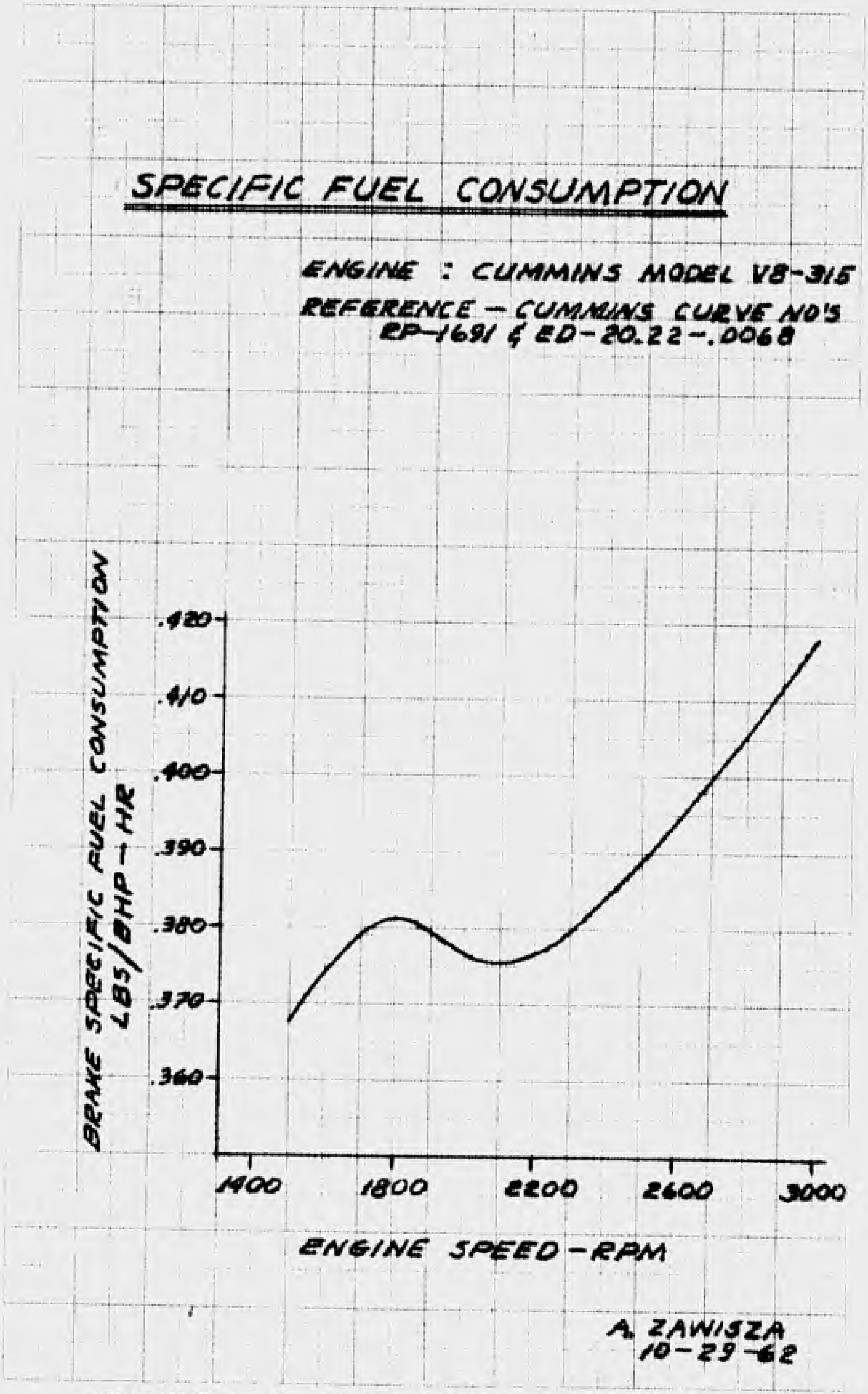


Figure 3-12. Fuel Consumption of Cummins V8-315 Engine Operated on Diesel Fuel (Part Throttle)



SPECIFIC FUEL CONSUMPTION

ENGINE : CUMMINS MODEL V8-315
REFERENCE — CUMMINS CURVE NO'S
RP-1691 & ED-20.22-.0068

A. ZAWISZA
10-29-62

FIGURE 3-13. FUEL CONSUMPTION OF CUMMINS V8-315 ENGINE OPERATED ON DIESEL FUEL (FULL THROTTLE)

180 x 6.6 = 1188 Lb

C. JP-5 Fuel:

180 x 6.79 = 1222.2 Lb

D. CIE-Fuel:

180 x 6.65 = 1197 Lb

3.5.2. Land Endurance.

To study the land endurance the following conditions were analyzed:

3.5.2.1. 40 MPH on Hard Level Ground.

3.5.2.1.1. In this condition the torque converter is locked and the transmission is in fourth gear. The required transmission output horsepower is found with:

$$HP = \frac{V \times TE}{375}$$

Where:

V = 40 MPH (Vehicle Speed)

TE = 1750 Lb (Tractive effort based on 100 Lb/Ton Rolling Resistance)

The output horsepower is then:

$$HP = \frac{40 \times 1750}{375}$$

$$HP = 186.7$$

3.5.2.1.2. Required transmission input horsepower:

$$HP_{TR} = \frac{HP}{\eta}$$

Where η = 85 percent is the transmission efficiency derived from the transmission performance curves.

$$HP_{TR} = \frac{186.7}{.85}$$

$$HP_{TR} = 219.6$$

3.5.2.1.3. Required engine gross horsepower output:

$$HP_E = HP_{TR} + HP_L$$

HP_L = 48 horsepower is the total accessory loss at 3000 rpm engine speed (see table 3-1)

$$HP_E = 219.6 + 48$$

$$HP_E = 267.6$$

3.5.2.1.4. Transmission output speed at 40 mph is:

$$N_{TR} = 52.1 \times V = 52.1 \times 40$$

$$N_{TR} = 2080 \text{ RPM}$$

3.5.2.1.5. The engine output rpm is:

$$N_E = N_{TR} \times 1.44 = 2080 \times 1.44$$

$$N_E = 3000 \text{ RPM}$$

3.5.2.1.6. The specific fuel consumption (BSFC) at 3000 rpm engine speed and 267.6 horsepower engine output is 0.401 pound per brake horsepower per hour and the fuel consumption (FC) per hour is:

$FC = BSFC \times HP_E$

$FC = 0.401 \times 267.6$

$FC = 107.4$ Lb/Hr

Endurance (E):

A. Diesel Fuel

$$E = \frac{1252.8}{107.4}$$

$E = 11.7$ Hr

B. JP-4 Fuel

$$FC = 107.4 \times \frac{19,300}{18,400}$$

$FC = 112.6$ Lb/Hr

$$E = \frac{1188}{112.6}$$

$E = 10.6$ Hr

C. JP-5 Fuel

$$FC = 107.4 \times \frac{19,300}{18,300}$$

$FC = 113.3$ Lb/Hr

$$E = \frac{1222.2}{113.3}$$

$E = 10.8$ Hr

D. CIE-Fuel

$$FC = 107.4 \times \frac{19,300}{18,500}$$

$$FC = 112.0 \text{ Lb/Hr}$$

$$E = \frac{1197}{112.0}$$

$$E = 10.7 \text{ Hr}$$

3.5.2.2. <u>25 MPH on Hard Level Ground.</u>

TE = 1750 pounds, based on 100 pounds per ton rolling resistance.

The transmission is in fourth gear converter drive.

3.5.2.2.1. Required transmission output torque (T) and output speed (N) are:

$$T = \frac{TE}{3.72} = \frac{1750}{3.72}$$

$$T = 470.4 \text{ Lb-Ft}$$

$$N = 52.1 \times V = 52.1 \times 25$$

$$N = 1302.5 \text{ RPM}$$

3.5.2.2.2. Required transmission input torque (T_{TR}) and input speed (N_{TR}) are:

$$T_{TR} = \frac{T}{R_T \times \eta \times M}$$

$$N_{TR} = \frac{N \times M}{R_N}$$

In the above formulae, M is the transmission gear ratio, η is the transmission efficiency, and R_T is the torque ratio and R_N the speed ratio of the torque converter. To find R_N and R_T the following method is used:

Torque converters are rated by using a K-factor, where:

$$K = \frac{\text{Input Speed}}{\sqrt{\text{Input Torque}}}$$

For a certain value of K the torque ratio and speed ratio is constant, therefore, multiplying both sides of the above equation with R_N and $\frac{1}{\sqrt{R_T}}$ results in a new K-factor:

$$K \frac{R_N}{\sqrt{R_T}} = \frac{\text{Input Speed}}{\sqrt{\text{Input Torque}}} \times \frac{R_N}{\sqrt{R_T}}$$

$$K_{out} = \frac{\text{TC Output Speed}}{\sqrt{\text{TC Output Torque}}}$$

In other words, a K-factor derived from torque converter (TC) output was found. In the same manner as above a K-factor based on transmission output can be derived:

$$K_{out} \frac{1}{M\sqrt{M}} = K^1{}_{out} = \frac{\text{TC Output Speed}}{M \sqrt{\text{TC Output Torque x M}}}$$

$$K^1{}_{out} = \frac{N}{\sqrt{T}}$$

Using the equations for required transmission input torque and speed, the following is obtained:

$$K^1{}_{out} = \frac{N_{TR} \times \frac{R_N}{M}}{\sqrt{T_{TR} (R_T \times \eta \times M)}}$$

Simplifying this equation with:

$$k_N = \frac{R_N}{M}$$

$$k_T = R_T \times \eta \times M$$

Results in:

$$K^1_{out} = \frac{N}{\sqrt{T}} = \frac{N_{TR} \times k_N}{\sqrt{T_{TR} \times k_T}}$$

From the 25 mph condition, transmission output torque and speed were found above. The K^1_{out} - factor is then:

$$K^1_{out} = \frac{1302.5}{\sqrt{470.4}}$$

$$K^1_{out} = 60.1$$

From the transmission output curve for fourth-gear converter drive a condition has to be found where $K^1_{out} = 60.1$. For this condition the transmission input torque and speed can be obtained from figures 3-5 and 3-6. Knowing the input and output values of torque and speed for one, i.e., a single condition where $K^1_{out} = 60.1$, the ratios k_N and k_T can be determined for this condition. k_N and k_T are constant for all conditions where $K^1_{out} = 60.1$; therefore, they can be used to find the required input speed and torque for the 25 mph condition.

From figures 3-5 and 3-6:

$T_i = 635$ Lb-Ft $\quad T_{iTR} = 472$ Lb-Ft

$N_i = 1515$ RPM $\quad N_{iTR} = 2760$ RPM

$$K^1_{out} = \frac{1515}{\sqrt{635}} = 60.1$$

$$k_N = \frac{N_i}{N_{iTR}} = \frac{1515}{2760}$$

$$k_N = .55$$

$$k_T = \frac{T_i}{T_{iTR}} = \frac{635}{472}$$

$$k_T = 1.35$$

Required transmission input for 25 mph condition is then:

$$N_{TR} = \frac{N}{k_N} = \frac{1302.5}{.55}$$

$$N_{TR} = 2368 \text{ RPM}$$

$$T_{TR} \frac{T}{k} = \frac{470.1}{1.35}$$

$$T_{TR} = 348 \text{ Lb-Ft}$$

$$HP_{TR} = \frac{T_{TR} \times N_{TR}}{5252} = \frac{348 \times 2368}{5252}$$

$$HP_{TR} = 157 \text{ Horsepower}$$

3.5.2.2.3. Required engine gross horsepower output:

$$HP_E = HP_{TR} + HP_L$$

HP_L = 46 horsepower is the total accessory loss at 2475 rpm engine speed (see table 3-1 and figure 3-4)

$$HP_E = 157 + 46$$

$$HP_E = 203 \text{ Horsepower}$$

3.5.2.2.4. The specific fuel consumption (BSFC) at 2368 rpm engine speed and 203 horsepower engine output was extrapolated from figure 3-12, and found to be 0.377 pounds per brake horsepower per hour. The fuel consumption (FC) per hour is then:

$$FC = BSFC \times HP_E = .377 \times .203$$

$$FC = 76.5 \text{ Lb/Hr}$$

Endurance (E):

A. Diesel Fuel

$$E = \frac{1252.8}{76.5}$$

$$E = 16.4 \text{ Hr}$$

B. JP-4 Fuel

$$FC = 76.5 \times \frac{19,300}{18,400}$$

$$FC = 80.2 \text{ Lb/Hr}$$

$$E = \frac{1188}{80.2}$$

$$E = 14.8 \text{ Hr}$$

C. JP-5 Fuel

$$FC = 76.5 \times \frac{19,300}{18,300}$$

$$FC = 80.7 \text{ Lb/Hr}$$

$$E = \frac{1222.2}{80.7}$$

$$E = 15.1 \text{ Hr}$$

D. CIE-Fuel

$$FC = 76.5 \times \frac{19,300}{18,500}$$

$$FC = 79.8 \text{ Lb/Hr}$$

$$E = \frac{1197}{79.8}$$

$$E = 15 \text{ Hr}$$

3.5.3. Water Endurance.

Information on water endurance will be issued as an addendum to this report on or about 1 January 1963.

3.5.4. Endurance Range and Fuel Margins.

3.5.4.1. Land Endurance Range:

300 miles at 25 mph or

12 hours at 25 mph

The above calculations show that the LVTPX11 has, at the specified speed of 25 mph and using

A. Diesel fuel, an endurance of 16.4 hours.

B. JP-4 fuel, an endurance of 14.8 hours.

C. JP-5 fuel, an endurance of 15.1 hours.

D. CIE-fuel, an endurance of 15.0 hours.

In the worst case, using JP-4 fuel, the vehicle has a margin of 2.8 hours or 70 miles before refueling is necessary.

3.5.4.2. Water Endurance Range:

Information on the water endurance range will be issued as an addendum to this report on or about 1 January 1963.

4.0. POWER TRAIN

4.1. General Description

4.1.1 The basic power train for the LVTPX11 is comprised of a single compression-ignition engine which supplies power to the tracks through a transfer gearbox and an ordnance-type steering transmission having pivot steer capability, and two final drives which mount the drive sprockets and incorporate an integral hydraulically-actuated variable-force track compensator. The systems supporting the basic power train are the engine aspiration air system, exhaust system, and cooling system.

4.1.2. The prime mover is a Cummins Model VINE V8-315 compression-ignition engine with a gross rating of 315 brake horsepower at 3000 rpm (60 degrees F.). The transmission is an Allison Model XTG-250-2 X-Drive Power Train which incorporates a hydraulic torque converter with a lockup clutch in combination with planetary range gearing providing four speeds forward and one speed in reverse, and including a steer system and full vehicle brakes. The final drives are of new design by the contractor to match the transmission/vehicle requirements, providing an optimum fixed reduction ratio in an assembly which permits sprocket movement to provide optimum track tension.

4.1.3. The power train general arrangement is shown in drawing SK-5165, Preliminary General Arrangement LVTPX11. The engine and transmission are joined into a unit power package by an inter-connecting transfer gearbox and mounting frame structure. The cross-drive transmission is installed with outputs aligning with the final drive inputs, and with the input facing forward. The engine is installed above the transmission with the flywheel end forward over the input to the transmission. The engine centerline lies logitudinally. The transfer gearbox lowers the drive from the engine output to the transmission input. The power package is symmetrically centered on the vehicle centerline in the machinery compartment in the rear of the hull.

4.1.4. The final drives are mounted in hull mounting rings located in the lower rear corners of the hull. The track compensator mechanism is attached at the lower side of the inner housing of the final drive.

4.1.5. The cooling system, exhaust system, and air cleaners are installed in the upper and side areas of the engine compartment. Convenient access to the engine compartment is provided through a top deck hatch and by two removable access panels on the engine compartment bulkhead.

4.2 Engine

4.2.1 General Description:

4.2.1.1. The powerplant used to power the LVTPX11 is the Model VINE V8-315 Compression-Ignition Engine (Figure 4-1), manufactured by the Cummins Engine Company, Inc., Columbus, Indiana.

4.2.1.2. The Cummins V8-315 is a 4-cycle, 90-degree V-type, 8-cylinder engine

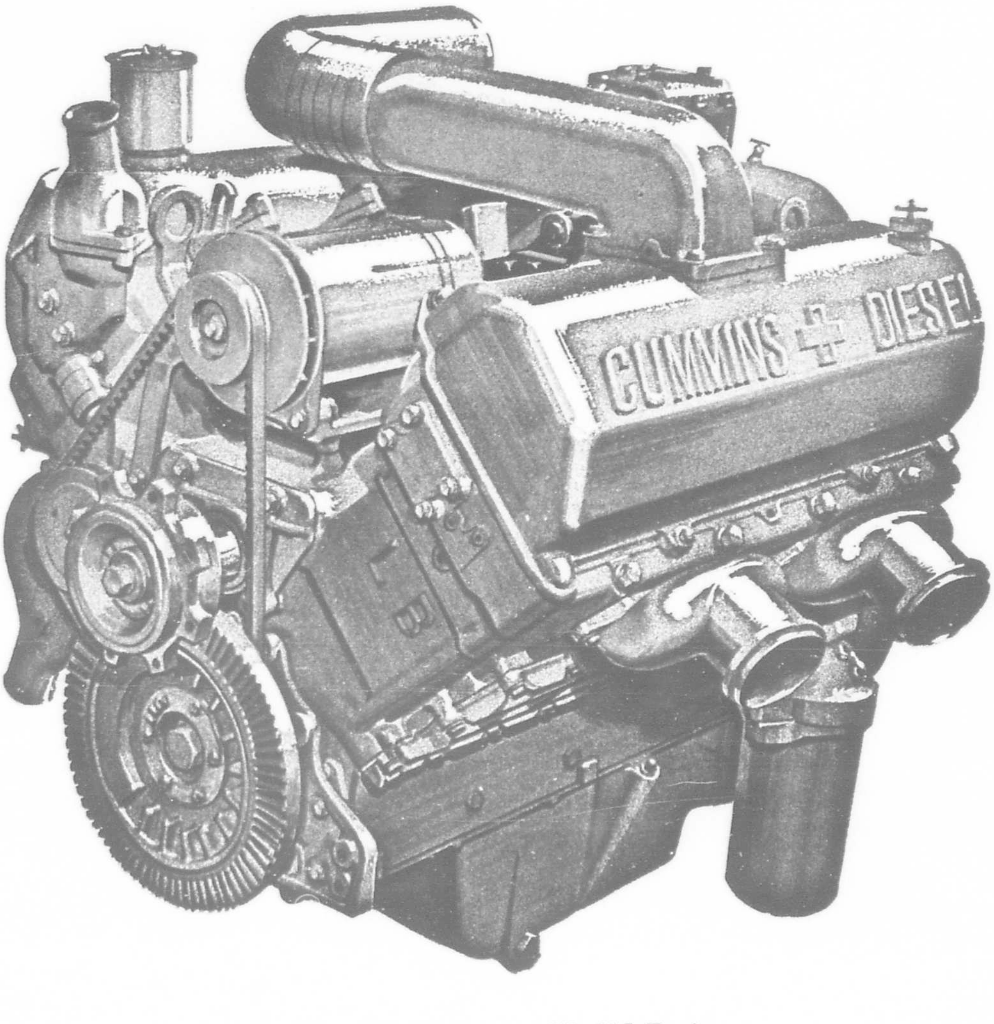


Figure 4-1. Cummins V8-315 Engine

of 785 cubic inches displacement. It is a naturally-aspirated engine rated at 315 brake horsepower at 3000 rpm, and 606 pound-feet maximum torque at 2150 rpm at 60 degrees F corrected conditions. The compression ratio is 15:1. It features an "over-square" design with a bore of 5-1/2 inches and a stroke of 4-1/8 inches. The nominal size dimensions are: length - 43.75 inches, width - 34.18 inches, and height 41.81 inches. The nominal net weight with cast-iron block, oil, and standard accessories is 1849 pounds.

4.2.1.3. The Cummins V8-315 engine is an up-rated version of the standard commercial Model V8-265. The higher rating is obtained through the use of special fuel metering and operation at higher speed. These engines have been designed for applications requiring minimum weight combined with dependability, fuel economy, and long life with low maintenance cost. They have found particular success in the long-haul trucking industry, especially in the demanding usage of western mountain areas. This commercial service experience added to the military experience in the U. S. Transportation Corps LARC 5 and LARC 15 amphibians will serve to establish further the qualifications of this engine.

4.2.1.4. The engine will be manufactured with a cast-iron block for the initial vehicles. However, the manufacturer states that it can be made available in aluminum if the demand warrants aluminum production. In aluminum, the weight would be reduced by approximately 400 pounds.

4.2.1.5. It is qualified for use of Diesel fuels, the JP-Series of fuel, and C.I.E. fuel. Development work is being accomplished on full multi-fuel usage, and the manufacturer states that multi-fuel capability can be added when required.

4.2.1.6 This engine is naturally-aspirated, and offers a considerable growth potential in power rating through turbo-charging should the requirement for more power be established.

4.2.1.7 The higher speed capacity, 3000 rpm, and the V-design reduces the over-all package size of the Cummins V8-315 engine, and coupled with a wide speed range, provides a smooth flow of driving torque and acceleration characteristics. The 90-degree V-design provides complete balance of both primary and secondary forces and, therefore, an inherently smoother running engine.

4.2.1.8. Advantageous components of this engine are; 24-volt immersion-proof starting motor, high-angularity oil pan, front trunnion mount, front outlet exhaust manifolds, full-flow lube filter, and 24-volt fuel shut-off solenoid valve with manual override position.

4.2.2. Design Features.

4.2.2.1. The V8-315 engine is clean and neat in appearance with all fuel and lubricating oil lines internally located. The fuel pump is compactly mounted between banks in the 90-degree V, and is gear driven for maximum dependability. The valve gear of each cylinder head is covered with a single cover that has an integrally designed intake manifold for simplicity of design and appearance.

4.2.2.2. Modern developments incorporated into this new engine by Cummins include the governor-regulated PT (pressure-Time) fuel system, steel rocker lever bushings, and stainless injector cups. In addition, it has proven features such as the open-type combustion chambers with direct injection, four-cycle design, chrome-plated top compression rings, cam ground pistons, gear-driven accessories, and cam shaft actuated injectors with roller type cam followers.

4.2.2.3. The engine block, of cast iron alloy, utilizes replaceable wet cylinder liners. It has a single gear-driven cam shaft, which actuates dual intake and exhaust valves and the fuel injectors. The exhaust manifolds are reversible to facilitate exhaust pipe routing.

4.2.2.4. The main and connecting rod bearings are precision type with steel backed inserts. The crank shaft journals are induction hardened with sufficient material for multiple regrinds. The crankshaft is fully counterweighted. The pistons are aluminum with two compression and one oil ring and are cam ground. They have full floating bearings. There are dual intake and exhaust valves for each cylinder which are actuated through solid push rods and roller type cam followers.

4.2.2.5. Engine lubrication is provided by a full pressure lubrication system. A gear type oil pump is located in the aluminum oil pan, allowing the engine to be tilted to 45 degrees. The pump delivers oil under pressure to the main, connecting rod and cam shaft bearings, to wrist pins, rocker arm shafts, and push rod ends. All oil passages are internal.

4.2.2.6. An integral oil cooler maintains the lubrication oil at a predetermined temperature and viscosity for best lubrication. This unit is a radiation-conduction type of oil cooler which is incorporated as a part of the cylinder block assembly. The absence of O-rings, seals, gaskets, and tube connectors eliminates any possibility of oil to water leaks. A finned aluminum cover, bolted to the cylinder block, facilitates cleaning of the cooler element.

4.2.2.7. When the engine is cold, the oil cooler brings the oil up to operating temperature by utilizing warm water recirculating through the engine. A full-flow oil filter cleans the oil. A large capacity cooling system assures even, adequate coolant flow through large volume internal passages to all areas of the engine. The belt driven water pump is located on the right front of the engine. It delivers 90 gpm at 3000 rpm to the cylinder liners, valve seats, fuel injectors, and oil cooler, which are constantly surrounded by circulating coolant. A radiator by-pass, from the thermostat back to the pump for recirculation, assures fast warm-up when the engine is cold.

4.2.2.8. Accessories may be driven from the front of the engine by use of grooved pulleys on the crankshaft. At the rear of the engine, a one or two groove pulley can be provided from the extension of the standard air compressor drive.

4.2.2.9. The fuel system consists of a Cummins PT (Pressure-Time) wear-compensating system with internal flyweight-type governor. This system utilizes new fuel system features to provide more trouble free metering and injection. The gear-type fuel pump draws fuel from the reservoir, passing it through a filter capable of filtering dust and dirt ten microns in diameter. Fuel is forced through a spigot-type throttle and a solenoid shut-off valve and delivered to the front of the cylinder banks through internal passages to the camshaft-actuated injectors, where the pressure-time principle accurately meters the fuel and delivers it as a fine spray into the combustion chamber. This injection principle eliminates fuel delivery lag, which often presents a problem in high pressure fuel systems. It also assures consistent fuel economy at all loads and speeds. Excess fuel passes from the injectors through internal passages and then to the fuel cell. Internal feed and return lines, which are drilled passages in the cylinder head, eliminate replacement of damaged tubes and permit the use of insert-type injectors. This system simplifies removal and installation of the injectors.

4.2.3. Engine Specifications.

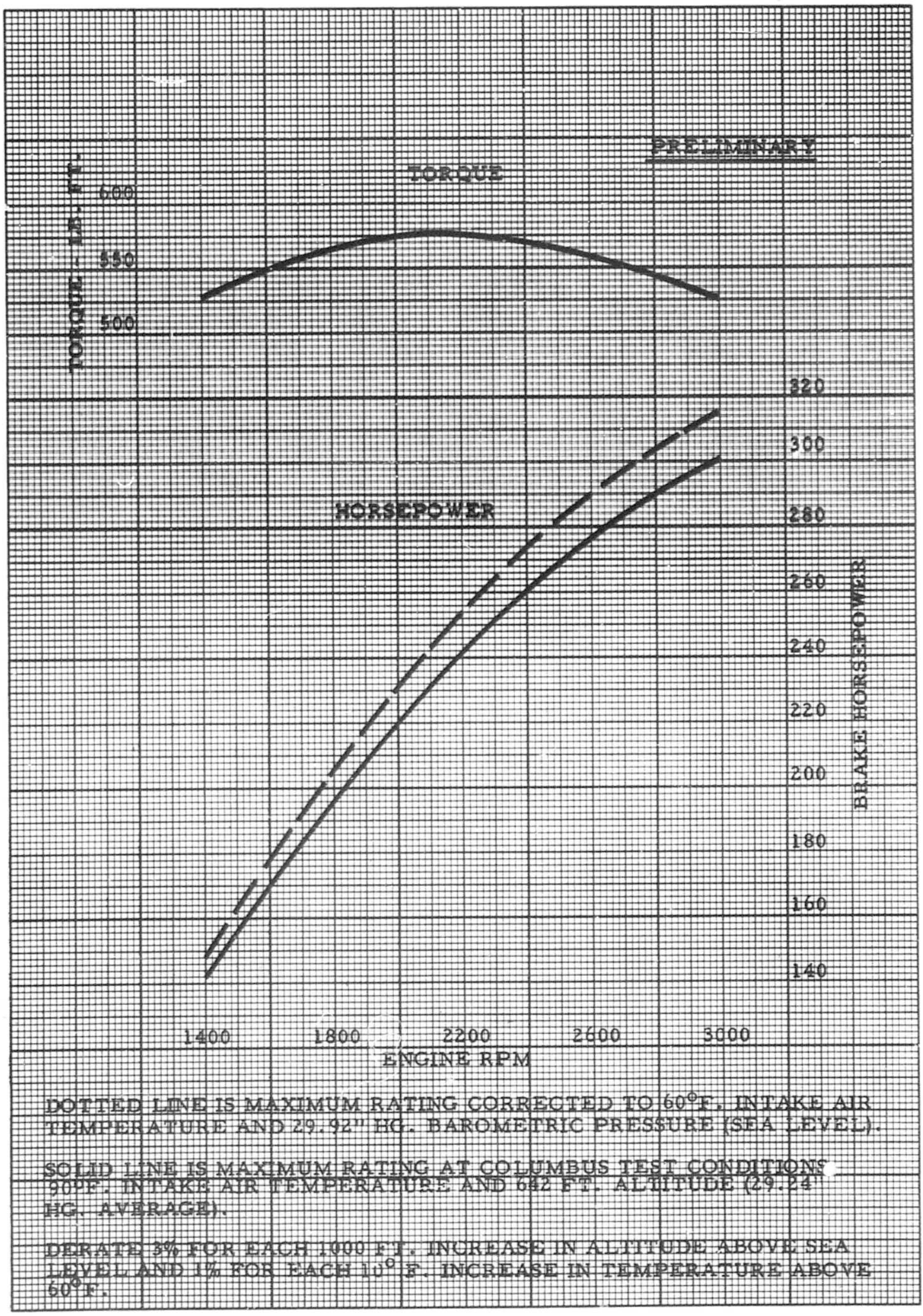
Summary of Cummins V8-315 Engine Specifications

Horsepower (60 degree F and Sea Level)	315

Governed Speed	3000 RPM
Maximum Torque at 2150 RPM	606 Lb-Ft
Number of Cylinders	8
Bore	5-1/2 In.
Stroke	4-1/8 In.
Piston Displacement	785 Cu In.
Angle of V	90°
Operating Cycles	4
Compression Ratio	15:1
New Weight w/Standard Accessories and Oil	1849 Lb.
Net Weight per Horsepower	5.9 Lb.
Dimensions:	
Length	43.75 In.
Width	34.18 In.
Height	41.81 In.
Aspiration Air Flow at 3000 RPM	600 CFM
Heat Rejection	27.7 BTU/HP/Min @ 3000 RPM
Maximum Fuel Filter Restriction	8 In. Hg
Maximum Air Restriction at Air Inlet	25 In. H_2O

4.2.4. Performance Characteristics.

Horsepower and torque performance of the Cummings V8-315 engine are shown by the curves of figure 4-2.



DOTTED LINE IS MAXIMUM RATING CORRECTED TO 60°F. INTAKE AIR TEMPERATURE AND 29.92" HG. BAROMETRIC PRESSURE (SEA LEVEL).

SOLID LINE IS MAXIMUM RATING AT COLUMBUS TEST CONDITIONS 90°F. INTAKE AIR TEMPERATURE AND 642 FT. ALTITUDE (29.24" HG. AVERAGE).

DERATE 3% FOR EACH 1000 FT. INCREASE IN ALTITUDE ABOVE SEA LEVEL AND 1% FOR EACH 10° F. INCREASE IN TEMPERATURE ABOVE 60°F.

Figure 4-2. Horsepower and Torque Performance

4.2.5. Flexible Coupling.

4.2.5.1. A radial-type flexible coupling will be installed in the drive line between the engine and the transfer gearbox. An installation layout of this coupling is shown in drawing SK-5220. The coupling is bolted to the flywheel and splined to the transfer gearbox input shaft.*

4.2.5.2. The coupling functions primarily as a torsional vibration dampener. This function is especially desirable in the LVTPX11 because the steering transmission is remote-mounted. The torque converter in the transmission, and the engine flywheel have relatively large mass at large diameters. These units might be likened to two large rotating masses on opposite ends of a shaft. Such a system has resonant frequencies of vibration with very high amplitudes which can cause shock loads leading to failure. The flexible coupling will dampen these excessive loads to acceptable values. In addition to the loads described, vibratory loads will be caused by the tracks. In this case the coupling will modify the natural frequencies of the entire system so that the vibratory loads do not resonate with the system' s natural frequencies.

4.2.5.3. The coupling itself is composed of two main parts, the case and the hub. The case encloses six neoprene biscuits whose axes are arranged radially about the coupling centerline. The biscuits flex under peak vibration loads

*(The Morse No. 87 size coupling has been recommended by the manufacturer for LVTPX11 application.)

and prevent the vibrations from being transmitted through the coupling. The hub picks up torque through radial hardened pins, one projecting from the center of each biscuit. The hub is internally splined to accommodate the transfer gearbox input shaft.

4.3. Transfer Gearbox.

4.3.1. General Description.

The primary function of the transfer gearbox is to transmit power from the prime mover to the steering transmission. The secondary function is to supply power through drive mounting pads for the main hydraulic pump, transfer gear lube pump, and an auxiliary pad which may be used for a hydraulic system supercharge pump or other accessory. The requirement for this unit arises as a result of the "engine-over-transmission" concept for the LVTPX11 power train. The unit, then, is of such a configuration that it will transmit power through a vertical distance of 27 inches and also supply auxiliary power as described.

4.3.2. Design Selection.

4.3.2.1. Phase I study of the transfer gearbox has considered two drive concepts. The first concept involves the transmission of power by means of a train of four straight-spur gears. The second concept involves the use of a Morse "Hy-Vo" chain. The chain-drive concept requires one set of gears, in addition to the chain, to obtain proper rotation at the gearbox output. Since the chain-drive concept requires inclusion of a gear set, it has been concluded from this evaluation that a transfer gearbox made up of a complete gear-drive unit is a more direct and simplified design approach. Hence, the final concept selected for development has been the gear-drive concept described below and shown schematically in figure 4-3. This gearbox is shown in more detail in drawing SK-5219.

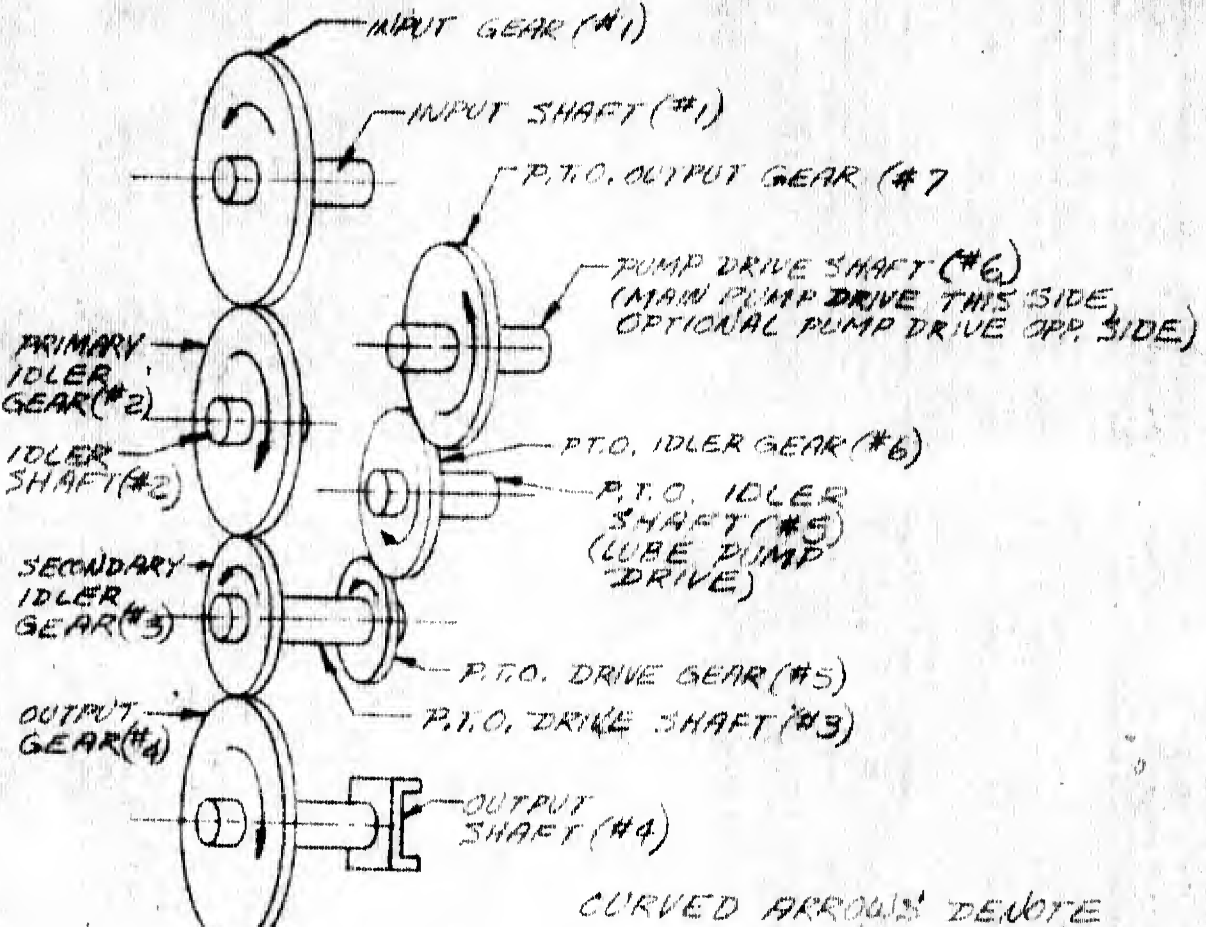
4.3.2.2. This gear-drive unit has been designed to utilize gearing components commensurate with commercial practice. Basically this unit consists of a set of four straight-spur gears, all on the same vertical centerline, housed in a cast-aluminum case. An additional auxiliary set of three straight-spur gears provides power to the three pump-mounting pads. The gearbox is rigidly mounted to the engine flywheel housing and extends downward to a point where it supplies power to the steering transmission through a horizontal, close-coupled, double universal joint.

4.3.3. Design Parameters.

The transfer gearbox has been designed to perform optimumly with the powerplant selection and to include the capability to perform with powerplants of up to 350 gross horsepower. Complete design parameters, including factors for duty cycles, auxiliary power requirements, and lubrication, etc., are cumulative in determining component design to give 1000 hours life. (Refer to Appendix II immediately following this section).

BY D. Percy DATE 10/19/62 SUBJECT LVTPX11
CHKD BY DATE
POWER TRAIN
DESIGN ANALYSIS
SHEET NO OF
JOB NO N0bs 4561-1

TRANSFER GEARBOX



CURVED ARROWS DENOTE ROTATIONAL DIRECTION

FIGURE 4-3 SCHEMATIC SHOWING TRANSFER GEARBOX GEARING AND SHAFTING ARRANGEMENT

4.3.4. Gearing.

The gears in this unit are designed to carry the load requirements for 1000 hours with the most efficient use of material commensurate with reasonable cost. The gears are made of a commercially available high-strength carburizing steel, such as SAE 8620, and the teeth are cut with a high degree of precision and are case-hardened.

4.3.5. Housing.

4.3.5.1. The transfer gearbox housing is made up of three sections as shown on drawing SK-5219. The main (middle) section includes a mounting flange at its upper end designed to mate with an SAE No. 2 engine flywheel housing, and bolting flanges for attachment of the other two housing sections. Accurate alignment of bearing bores are maintained with dowel pins. The bores are machined in assembly and are straight-through bores wherever possible to simplify machining. Simple bearing retainer covers are used to enclose the bores.

4.3.5.2. The housing sections are made from high-strength aluminum alloy castings of material specification Alcoa 356-T6, or equivalent. This material is used extensively for gearbox housings with good results, providing good corrosion resistance, adequate strength, and light weight. Cast-iron inserts are cast into the housings to provide durable bearing seats. A design with bearings pressed directly into the aluminum has been considered; however, heavy interference fits are required which make assembly very difficult, and also, great care is required during disassembly and reassembly for maintenance inspections to prevent scuffing of the bore which would result in loose bearing fits.

4.3.6. Shafts.

4.3.6.1. In general, the transfer gearbox shafts are to be designed with the most economical use of material, using good quality shafting steels falling in the SAE 3000 and 4000 series (for example, SAE 3140, SAE 4140, or SAE 4340). Wherever possible, readily available pre-hardened steels may be specified as alternates. These steels are hard enough so that they require no heat treatment subsequent to the purchase of the raw stock. An example is the LaSalle Steel Company's line of "Stressproof" steels. Final material selection is to be made after detailed stress analysis has been completed. Unless unusual circumstances arise, all shafts are to be made from the same material to simplify procurement and manufacturing procedures.

4.3.6.2. The shafts shown on the preliminary layout, drawing SK-5219, have been designed to a point where they are known to be adequate to carry the loads. Involute splines have been selected for power-transmitting connections because of their reliability. Detail design will be aimed primarily at reduction of weight and further simplification.

4.3.7. Bearings.

4.3.7.1. Standard metric straight-roller bearings are used throughout the transfer gearbox. This bearing type selection was made because of the following:

A. Straight-roller bearings are able to carry higher radial loads, size for size, than ball bearings.

B. Straight-spur gearing transmits little, if any thrust loads to supporting bearings.

C. Straight-roller bearings are generally more economical, size for size, than ball bearings.

4.3.7.2. Note that drawing SK-5219 shows only three different bearing sizes althrough there are a total of twelve bearings required. This is done to provide the maximum possible degree of interchangeability and, hence, simplify ordering and stock-handling requirements.

4.3.8. Seals.

4.3.8.1. The transfer gearbox requires only two oil seals, one at the input and one at the output.

4.3.8.2. The input seal prevents gearbox oil from leaking into the engine flywheel housing by means of a spring-loaded lip, and prevents dust from entering the gearbox by means of an outer dust-lip. This seal can be of standard commercial construction since it is totally enclosed and isolated from salt-atmosphere corrosion.

4.3.8.3. The output seal is of the same type of construction as the input seal. It does, however, have a cadmium-plated case and a stainless steel lip-retaining spring since it is exposed to a corrosive atmosphere during salt-water operation.

4.3.9. Lubrication.

4.3.9.1. Since the maximum gear pitchline velocities are quite high, (up to 8750 fpm), positive lubrication is accomplished by means of an oil spray system to assure adequate lubrication and cooling of the gear teeth. A network of spray nozzles is arranged so that each gear mesh receives a direct oil spray on the entering mesh. The lube lines to all of the nozzles are supplied by a small lube pump, driven by the gearbox itself. A remotely located reservoir provides an oil sump and serves effectively to augment cooling.

4.3.9.2. The gears for the auxiliary drives have not been included in the foregoing discussion. These gears carry relatively light loads and are sufficiently lubricated by the oil "mist" created in the gearbox by the nozzles.

4.4. Transmission.

4.4.1. General Description.

4.4.1.1. The Allison Model XTG-250-2* Transmission, developed by Allison Division of General Motors Corporation, is being used in the LVTPX11 vehicle. It is a manually controlled, full torque shifting, X-Drive power train transmission which includes a hydraulic torque converter with lockup clutch in combination with a four speed forward and one reverse planetary gearset incorporating a steer system and full vehicle brakes. The transmission controls

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*The basic transmission designation is Allison Model XTG-250. The suffix "-2" designates the Allison transmission without final drives as used in the LVTPX11 installation.

include vehicle steer, range selector, pivot steer selector, and single pedal vehicle brakes. External views of the transmission assembly, including the Allison final drives, are shown in figures 4-4 and 4-5.

4.4.1.2 The XTG-250-2 transmission has a maximum input rating of 260 hp, maximum input torque of 520 pound-feet and an input speed of 3000 rpm. It is classed for use in 35000-pound vehicles. It is waterproof and will operate at temperatures to -65 degrees F. An aluminum housing is used for salt-atmosphere corrosion resistance. This transmission incorporates components which provide considerable interchangeability with other commercial and military transmissions manufactured by Allison.

4.4.1.3. The flow of power through the XTG-250-2 transmission is from the engine to the torque converter to the basic planetary gearset and into a bevel gearset. A cross shaft from the bevel gearset drives into the reverse planetary, the output steer planetary, and the brake package at each side of the transmission. See figure 4-6, XTG-250-2 Power Train Schematic, and figure 4-7, Cross Section of XTG-250 Transmission.

4.4.2. Design Features.

4.4.2.1. Converter. The hydraulic torque converter is a single-stage, multiple-phase converter with a lockup clutch. Maximum converter torque multiplication is 2.55:1 at stall. Control of the automatic converter lockup clutch (available in all ranges) is provided by a converter-turbine-driven governor.

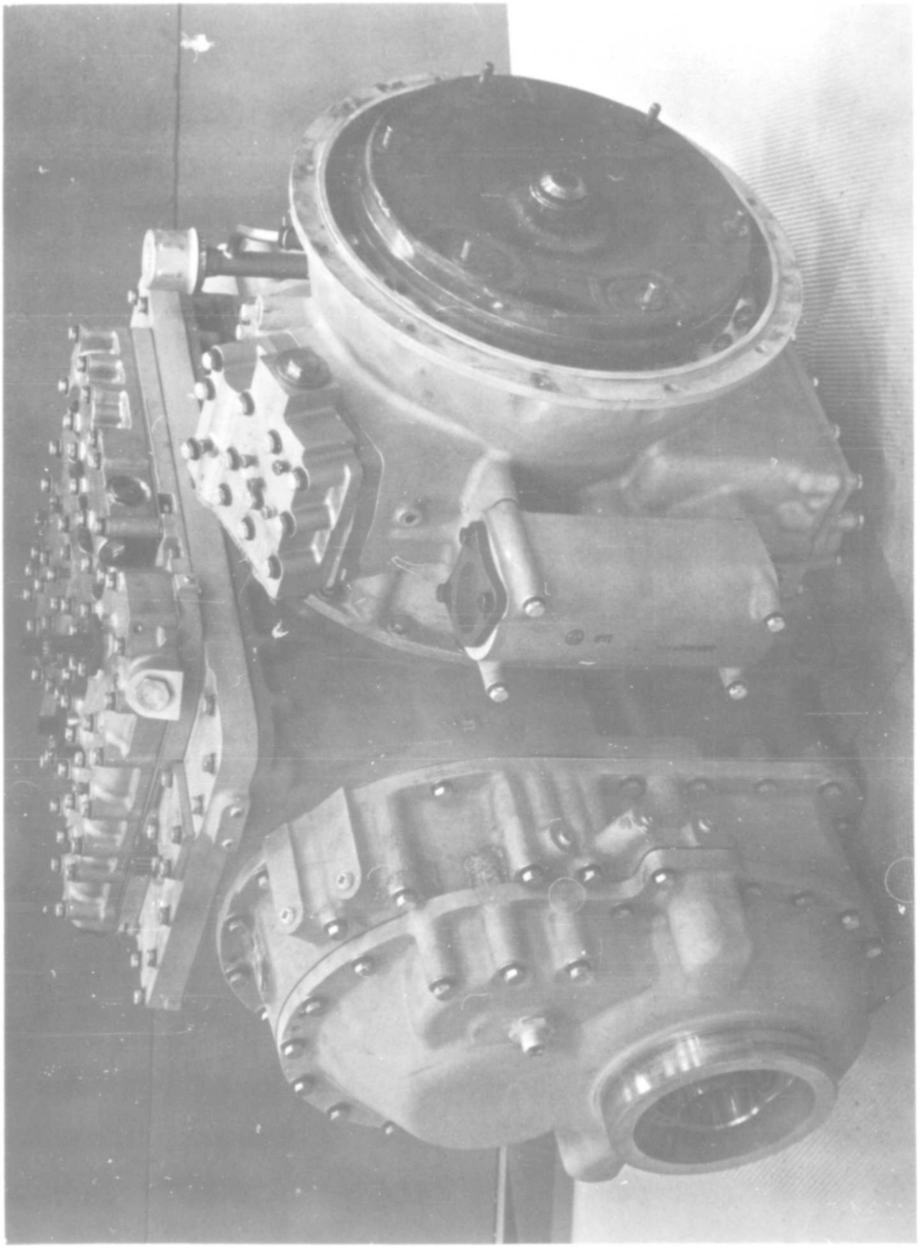
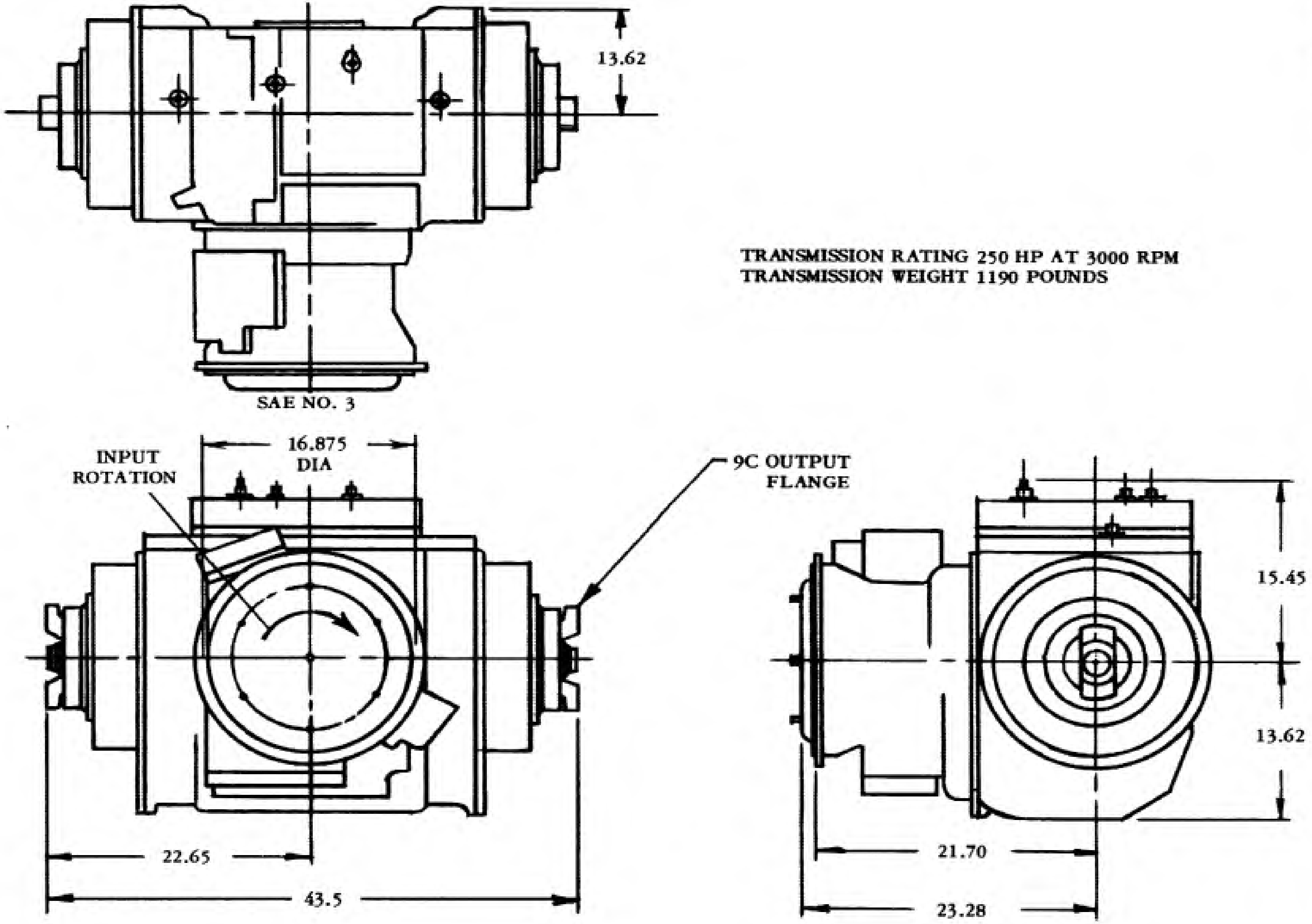


Figure 4-4. Allison XTG-250 Transmission



TRANSMISSION RATING 250 HP AT 3000 RPM
TRANSMISSION WEIGHT 1190 POUNDS

Figure 4-5. Preliminary Outline Drawing - Allison XTG-250 Transmission

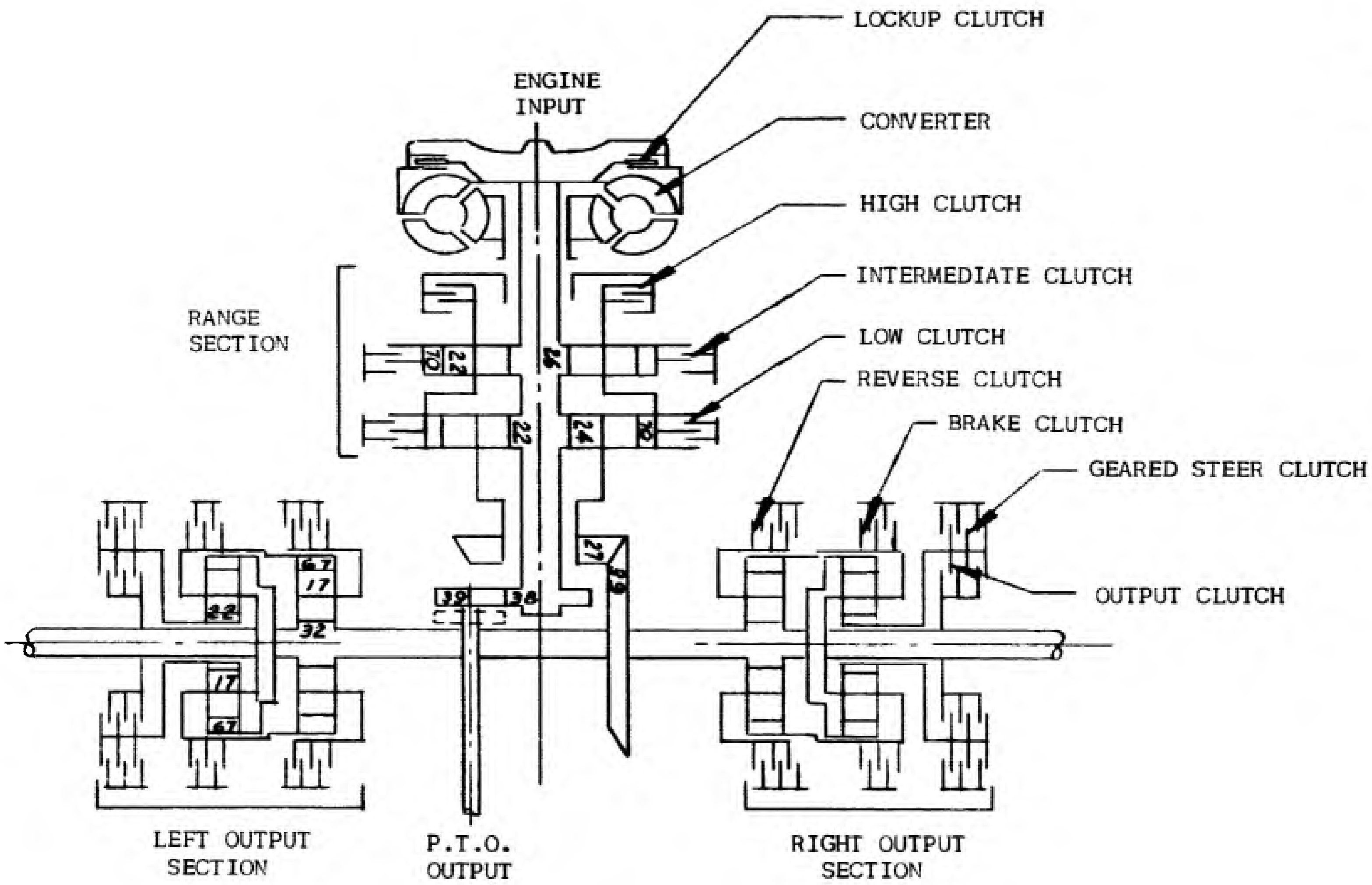


Figure 4-6. Allison XTG-250-2 Power Train Schematic
(Straight Through Version)

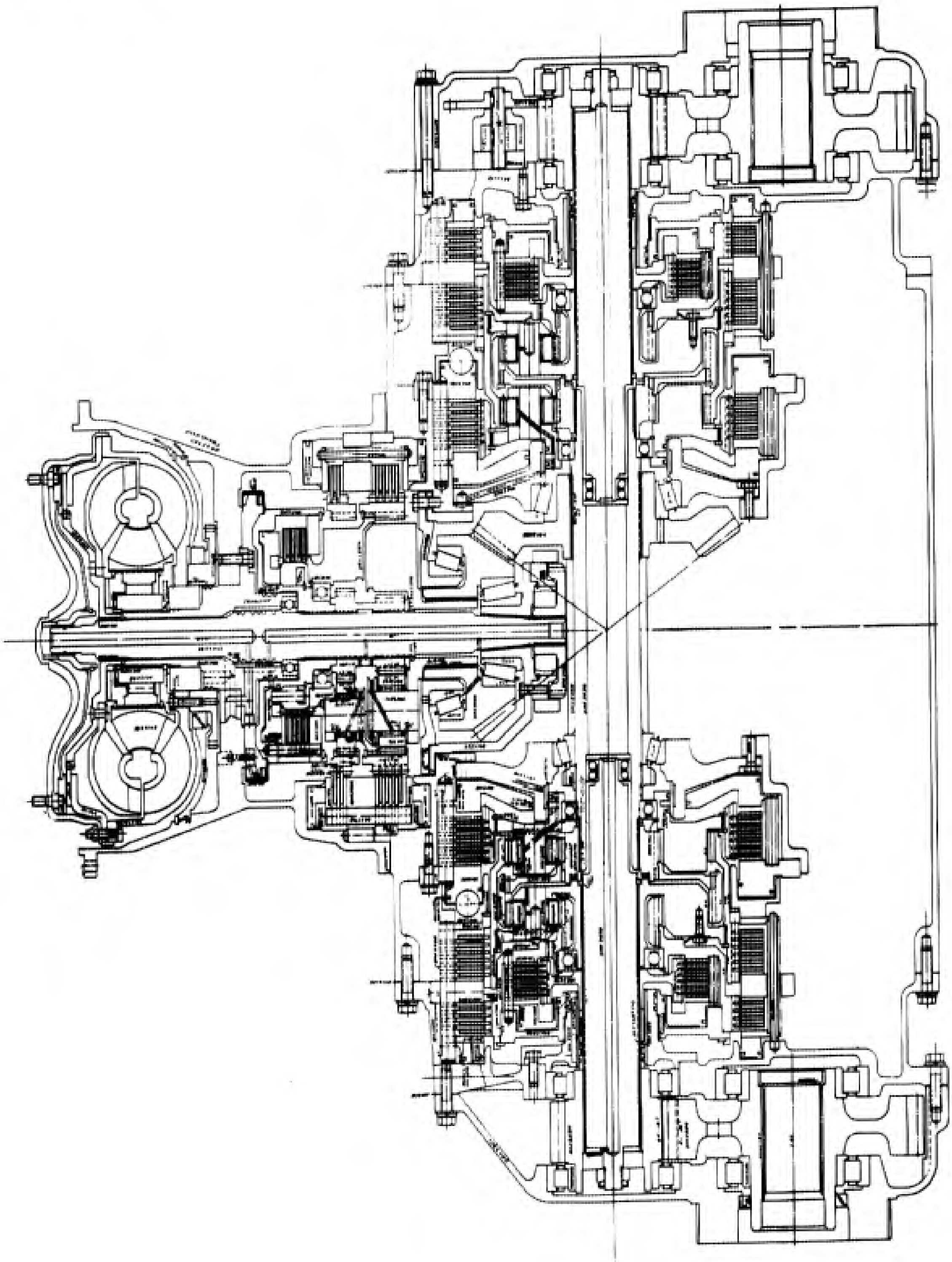


Figure 4-7. Cross Section of Allison XTG-250 Transmission

The governor is throttle-position modulated to permit lockup operation at lower speeds at part throttle for additional economy.

4.4.2.2. Controls.

4.4.2.2.1. During normal steering operation of the XTG-250-2 transmission, a gear ratio ranging from 1:1 to 1.475:1 (in the output planetary steering gear) is available for geared steer operation in fourth, third, and second ranges during land operation. Pivot steering, through counter-rotation of the tracks, is used for steering in first and reverse ranges during land operation, and is used in first, second, third, and reverse for water operation.

4.4.2.2.2. A neutral position, disengagement of the inboard track, is reached before pivot steering is put into effect. This neutral position is used for steering the vehicle during marine operation.

4.4.2.2.3. The steering system is hydraulically activated by a valve which is operated mechanically. The gear selection is through multiple plate clutches which are engaged by oil pressure. The brake system, used for service and parking, also consists of multiple wet plates which are mechanically actuated, oil cooled and oil lubricated.

4.4.2.2.4. The desired ratios and steering controls are as follows:

Range Position	Range Gear Ratio	Steering Normal (Land)	 Pivot (Water)
R	4.69:1	Pivot	Pivot
N	--	--	--
1	6.16:1	Pivot	Pivot
2	4.18:1	G.S. *	Pivot
3	2.24:1	G.S. *	Pivot
4	1:00:1	G.S. *	G.S. *

*G.S. = Gear Steer

4.4.2.2.5. Shifting to third, second, first, and reverse ranges are inhibited at speeds which would be dangerous to vehicle and crew. Vehicles using the XTG-250 transmission can also be push- or tow-started by shifting into second range at speeds of three to six mph, but not over eight mph.

4.4.2.3. Lubrication.

The hydraulic and lubricating systems function under four systems: the oil supply system, the lubricating system, the hydraulic control system, and the converter system. These systems, using the same oil, are interconnected through the valve body which distributes the oil to the various systems under the correct pressures.

4.4.3. Modifications for LVTPX11

Since the XTG-250 transmission was developed under an Ordnance program, it does not, in its present design operating configuration, completely meet the LVTPX11 operational and performance requirements. Certain minor modifications will be made. They include:

A. Omission of integral final drives.

B. Housing material change from magnesium to aluminum.

C. Provision of a new remote input drive and front closure.

The above three modifications are covered in the transmission model designated by addition of the suffix -2, i.e., XTG-250-2.

4.4.4. Preliminary Characteristics for XTG-250-2 Transmission

4.4.4.1. Component.

The selected component is an X-Drive, Model XTG-250-2 Transmission with Hydraulic Torque Converter, Planetary Gear sets and all Torque Shifting. The XTG-250-2 includes a hydraulic torque converter with lockup clutch. The planetary range gearing in combination with the steer and output planetary sets provide four forward ranges and one reverse range. The transmission also incorporates, geared and pivot steer systems, and full vehicle brakes as outlined in the following specifications:

4.4.4.2. General Specifications.

Rating:

Maximum Input Torque, Lb-Ft	520
Maximum Input Speed, RPM	3000
Maximum Input Horsepower	260
Manufacturer	Allison Division - GMC
Model	XTG-250-2
Drive Ranges	Fourth, Third, Second, First, Neutral, Reverse 1
Drive Range, Steering, and Shift Control (External)	Mechanical
Shift and Steering Mechanism (Internal Control)	Hydraulic & Mechanical
Steering Type	Geared Steer, Pivot Steer
Power Take-Off	Converter-Turbine Driven
Rating (Continuous Operation) Lb-Ft	470
Type of Clutches (All Ranges)	Multiple Plate, Engaged by Oil Pressure
Brake	Multiple Wet Plate, Service and Parking, Mechanical Application
Hydraulic Torque Converter, Single Stage, Multiple Phase with Lockup Clutch, Maximum Converter Multiplication	2.55:1

Range Position	Overall Transmission Gear Ratio	Steering Normal (Land)	Steering Pivot (Water)
R_1	6.776:1	Pivot	Pivot
N	-	-	-
1	8.921:1	Pivot	Pivot
2	6.036:1	G.S.*	Pivot
3	3.234:1	G.S.*	Pivot
4	1:44:1	G.S.*	G.S.*

*G.S. = Geared Steer

Bevel Gear Ratio	1:44:1
Output Steer Planetary Ratios:	
Under Drive	1.475:1
Direct	1.00:1
Total Torque Ratio Coverage:	
First Gear Stall	2.55 x 6.16 x 1.44 = 22.62:1
Fourth Gear Lockup	1.00 x 1.44 = 1.44:1
Oil Specifications	MIL-L-2104A, Grade 10
Oil Capacity, Gallons	(Estimated) 14
Lube Pump Output, GPM	20
Transmission Dry Weight, Pounds	(Estimated) 1350

When this unit is coupled with a Borg-Warner final drive, the total torque ratio coverage is:

Borg-Warner Final Drive Ratio	3.3:1
First Gear Stall	2.55 x 6.16 x 1.44 x 3.3 = 74.64:1
Fourth Gear Lockup	1.00 x 1.44 x 3.3 = 4.75:1

4.5. Final Drive.

4.5.1. General Description.

Unlike previous final drive design concepts, the final drive for the LVTPX11 has a dual purpose. It not only serves as a speed reduction unit, but also as a track-tensioning device. As a matter of review, the final drive acts as a gearbox, reducing speed and multiplying torque in amounts great enough to meet vehicle specifications of speed and tractive effort. As the name implies, it contains the final power train gearings, with the output shaft serving as a mounting for the track drive sprocket. When functioning as a track-tensioning device, the final drive housing oscillates in such a way as to move the track drive sprocket so that the track tension is adjusted to meet vehicle operating condition. See Section 5.0, Suspension System, for further discussion on track behavior.

4.5.2. Design Alternates and Final Selection.

4.5.2.1. During the course of Phase I design work, several final drive design concepts have been derived, criticized, and compared in an effort to achieve a final concept which gives an optimum relationship between simplicity, cost, weight, ruggedness, and ease of assembly. Three basic arrangements of gearing have been evaluated: 1) A combination of a set of external spur gears and a gearset consisting of a pinion meshing with an internal gear (figure 4-8.A); 2) A combination of a set of external gears and an output planetary (figure 4-8, B); 3) and a single set of external gears (figure 4-8, C). The relative simplicity of the single gearset arrangement is readily seen in figure 4-8. In addition, three types of mounting arrangements have been studied. These arrangements are shown in figure 4-9. The arrangement in figure 4-9, A, does not interrupt the hull surface, but the mounting bearings need to be very large and costly since at least one of them must encompass the entire gearbox. The arrangement in figure 4-9, B, requires a hull pocket and the outboard mounting bearing must be very small. The arrangement in figure 4-9, C, requires a small pocket but allows complete freedom in selection of mounting bearing size and gives greater strength at less cost.

4.5.2.2. The final concept selected for the LVTPX11 final drive provides a simple, single-reduction gearing arrangement in a two-piece housing which is mounted in sleeve bearings in the hull, as shown in figure 4-9, C. (Refer to drawing SK-5218 for layout of the final drive). The gearing transmits power directly from the cross drive transmission output shafts to sprockets mounted on each final drive housing. This concept also proides a track-tensioning device, by permitting oscillating movement of the sprocket about the input shaft centerline and utilizing a hydraulic cylinder to control this movement. (Described

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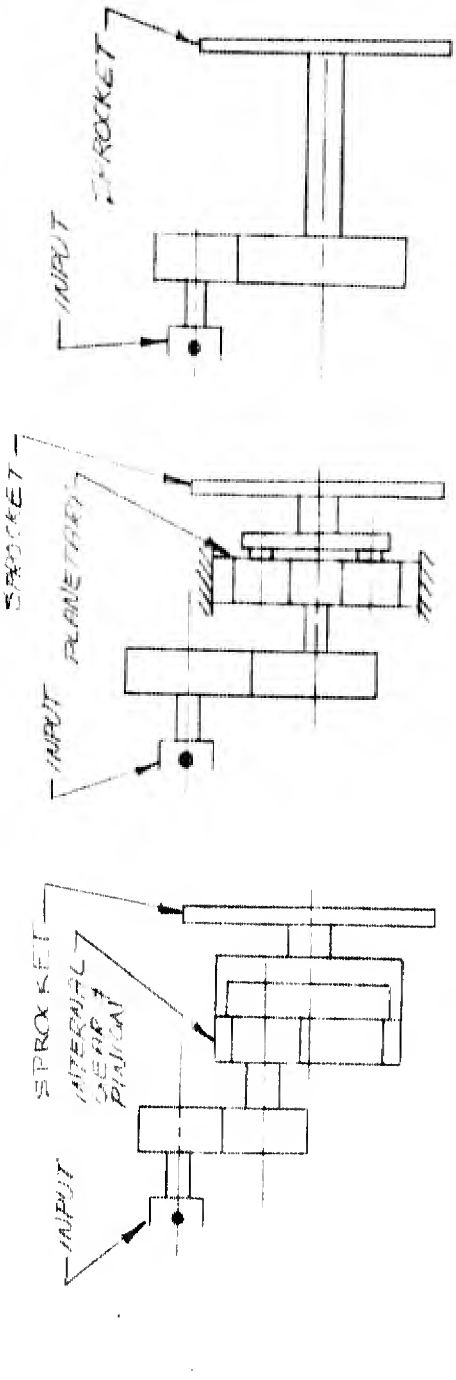

A. TWO-STEP INTERNAL GEAR

B. TWO-STEP PLANETARY

C. SINGLE-STEP (FINAL SELECTION)

FIG. 4-8 GEARING ARRANGEMENTS CONSIDERED IN FINAL DRIVE CONCEPT STUDY

CUSHING & CO.

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

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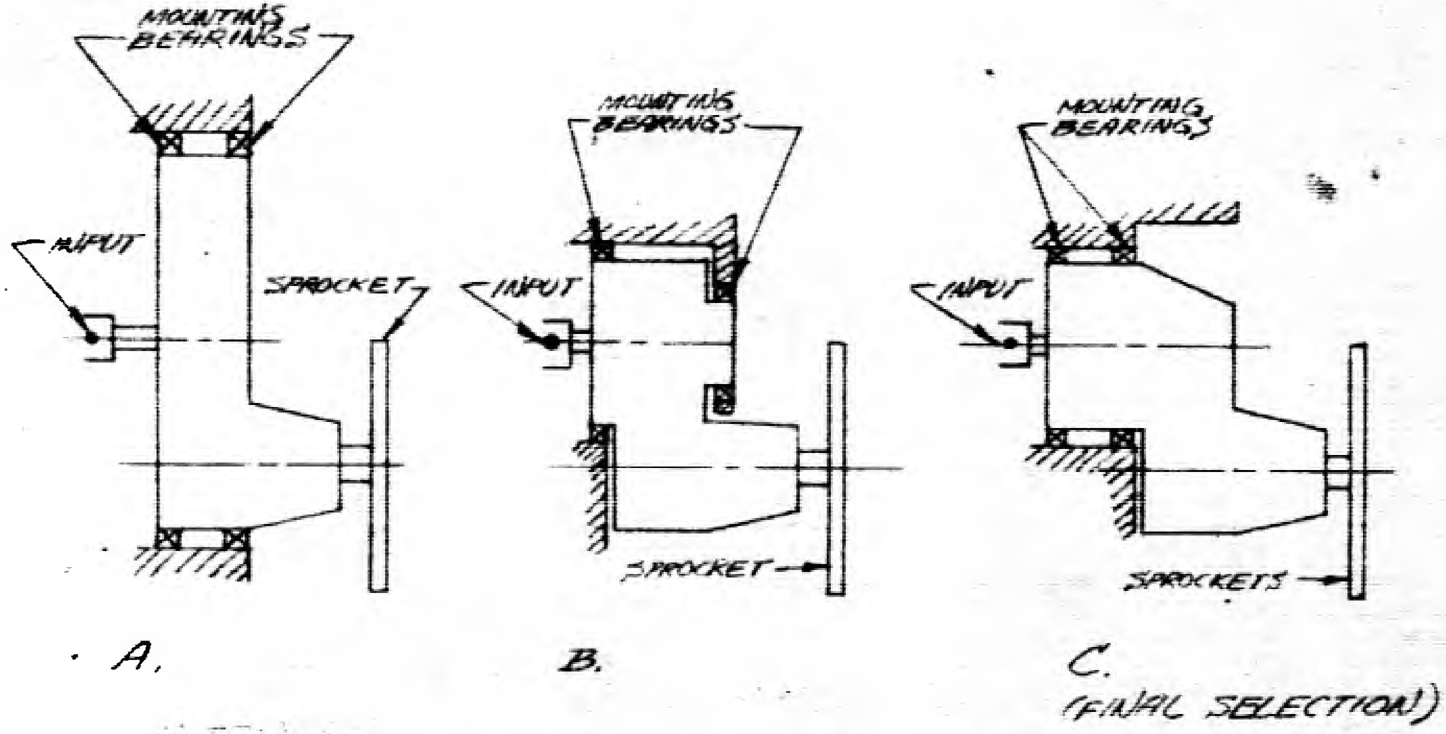

FIG. 4-9 MOUNTING ARRANGEMENTS CONSIDERED IN FINAL DRIVE CONCEPT STUDY

in detail in section 8.0, Hydraulic System.

4.5.3. Design Parameters.

In general, the final drive is to be designed to be compatible with the maximum track capabilities and gross vehicle weight, i.e., the design loads on the final drive are derived from sprocket and track reaction loadings, since these are limiting loads, rather than from possible input loads. The design values of loads, speeds, and maximum torques transmitted by the final drive are derived, generally, from relative values shown to be acceptable in LVTP5 vehicle production and upon vehicle operating characteristics which have been collected by the contractor at his test facilities and various service field test activities over the past decade.

4.5.4. Gearing.

The gears are straight-spur involute type, manufactured from heat treated alloy steel forgings, with case-hardened (carburized) teeth. Calculations for the gearing design, gear geometry, strength and endurance factors, etc., are included in Appendix II immediately following this section. Following the requirements for this type of precision gearing, the gears are finished by shaving and the profiles are altered to eliminate tip interference. All gear tooth modifications are to be discussed with and approved by a qualified gear manufacturer such as National Broach and Machine Company, Detroit, Michigan, who have had wide experience in heavy-duty, precision gears.

Finally, shot peening is specified for the gears to extend the fatigue life if final calculations indicate that it is necessary.

4.5.5. Housing.

The two sections of the final drive housing are made from high strength cast steel. Steel is selected for its superior ability to withstand the high track tensions, shock loads and severe abrasive action from foreign particles picked up by the track chain. The general configurations for the castings are determined from functional requirements, including the oscillatory movement required for track tensioning, and to a large extent upon past experience.

4.5.6. Shafts.

4.5.6.1. The input shaft includes the pinion gear as an integral part. Such an arrangement, dictated by the relatively small diameter of the gear, eliminates the need for machining a means of securing the gear to the shaft. The design of this shaft follows the parameters for gearing.

4.5.6.2. The output shaft is splined into the gear hub at its inboard end and supported by a self-aligning bearing at its outboard end. The shaft also includes a flange at its outboard end for mounting the track drive sprocket and road wheel sections. Note that the design would permit removal of the shaft, in case of failure, without having to dismantle the entire final drive. This type of output shaft design has proven to be very satisfactory on the new spur-gear final drive, now replacing other obsolete drives, on existing LVT's. The output shaft is a steel forging, the exact composition of which will be selected on the basis of strength requirements, weight, and cost.

4.5.7. Bearings.

The pinion and gear bearings are of the standard metric straight roller type. This type has been selected because of its ability to carry higher loads than ball bearings of the same size. The self-aligning, double row, roller bearing at the outboard end of the output shaft is selected, also, because of its high load-to-size capacity and its ability to allow some shaft misalignment.

4.5.8. Seals.

Preliminary selection of seals is as discussed below:

A. Input Seal.

1. Since the final drive is close to the bottom of the hull, the input area may be exposed to bilge water splash, especially during "heavy-sea" operation where listing can occur.
2. A double lip seal is selected to isolate both the oil within the final drive housing and water within the bilge. Since much of the vehicle operation is in salt water, the seal design includes a cadmium-plated casing and stainless steel lip-retaining springs to minimize corrosion. Other than this, the seal is of a standard commercial design.

B. Output Bearing Seals.

1. During this development many types of both internal and external seals have been tested along with several types of shedder arrangements. The result is the proven sealing arrangement used on the new spur-gear final drive used in the LVTP5. The same arrangement will be used in the LVTPX11.

2. The arrangement consists of an external, triple-lip seal protected by a shedder (or labyrinth) on the outer side of the bearing and a double-lip seal on the inner side of the bearing. The bearing is greased through a passage machined in the output shaft.

3. The bearing is isolated from the rest of the gearbox so that contamination of the entire box will not occur in case the bearing fails. The shedder is a "tongue-in-groove" setup machined between the housing and the inner surface of the output shaft flange. The output seal and shedder is designed to function as follows:

   The labyrinth protects the seal from large particles of sand; the first seal lip stops smaller particles of sand and dust; the second lip stops water; and the third lip isolates the grease within the final drive.

   The seal case is cadmium-plated and the lip-retaining springs are made of stainless steel to minimize corrosion damage.

4. The inner bearing seal separates the oil within the gearbox from the grease around the bearing. It also protects the gearbox in the event of output bearing failure. This seal also has a cadmium-plated case and a stainless steel spring for protection against corrosion.

C. Hull Seal.

Since the final drive goes through a small amount of rotation in the hull, a hull seal is provided which can withstand limited rotation. This seal need not be as complicated as a standard shaft lip seal which is designed for relatively high speed rotation. Hence, an "O" ring has been selected for this application. The "O" ring is protected by a machined labyrinth which stops relatively large abrasive particles of sand and dust.

4.5.9. Lubrication.

4.5.9.1. Preliminary investigation has led to the conclusion that a wet-sump, splash type of lubrication is sufficient for the final drives. Oil is contained in the bottom of the housing at a level of sufficient height to wet the two gear supporting bearings. The gear, then runs in the oil and picks up a small amount of oil on each tooth and carries it to the pinion. Since the gear is quite large (12.25 inch pitch diameter by 3.5 inch face width), the total amount of oil picked up is sufficient to lubricate the pinion and its bearings.

4.5.9.2. As previously described, the outer, self-aligning bearing is greased through a lube fitting at the outer end of the output shaft.

4.5.9.3. No special cooling provisions are provided since the final drives are exposed to the outside air during land operation and water during marine operation.

4.6. Drive Shafts.

4.6.1 General Description.

Phase I study has established a requirement for three drive shaft assemblies. The first transmits power from the transfer gearbox output yoke to the steering transmission input yoke. The requirement for this shaft has arisen since the decision to delete rigid mounting between the transfer gearbox and the steering transmission. The other two drive shafts are identical and are required for transfer of power from the port and starboard steering transmission output yokes to the respective final drives.

4.6.2. Design Selection.

4.6.2.1. In both applications, automotive type, Mechanics Division, Borg-Warner Corporation, roller bearing universal joints will be used. Provision for axial slip of the U-joint assemblies to compensate for installation or operating changes in length is through a floating flange on one end of each joint assembly.

Grease fittings are provided at each joint for ease of maintenance. Due to space limitation, close-coupled assemblies will be used. This type of an assembly has a center plate rather than a tubular yoke connector. (See drawing SK-5165 and note drive shaft locations).

4.6.2.2. These universal joints are selected to have ample capacity with respect to vehicle braking forces, maximum power torque output, and loads resulting from pivot steer conditions. Consideration has also been given to critical speeds and the resulting effect on performance.

4.6.3. Disconnect Requirement.

Accessibility is provided to all drive shafts for disconnect in event of power package or component removal from the vehicle compartment.

4.7. Engine Aspiration Air System.

4.7.1. General Description.

The engine aspiration air system consists of an inlet air plenum chamber (in the case of the LVTPX11 the entire interior hull spaces), a means of filtering aspiration air, and a duct to transfer the filtered air to the engine air intake horn. Supply of air to the interior hull spaces is discussed in section 13.0, Vehicle Air Systems.

4.7.2. Design Selection.

4.7.2.1. Due to weight and space limitations, and the improvements made in the field of dry cartridge type air filters, this type has been chosen for use rather than the conventional oil bath type filter.

4.7.2.2. Phase I proposals from the Donaldson Company, United Air Cleaner Company, and the Air-Maze Corporation were evaluated and the Air-Maze filter has been selected as the optimum filter. This selection was concluded by the assembly weight and component material aspect of the proposal evaluation.

4.7.3. Design Features.

4.7.3.1. Through vendor discussions, it became apparent that a further weight and space saving could be realized by eliminating the filter's dirt container, or outer case, and allowing accumulated dirt to collect in the hull bilge areas. This would require then, that the cartridge be supported in the vehicle by brackets attached directly to the cartridge retaining hardware. The Air-Maze cartridge is constructed of an inert unwoven fiber blanket encased between layers of aluminum screen and deeply pleated, with the inner and outer cylinders formed of corrosion proof hardware cloth. Due to the strength of this construction, end mounting brackets provide sufficient support.

4.7.3.2. Aspiration air enters the cleaner around its entire periphery, and leaves the center core through fiberglas flexible ducting to enter the engine air horn.

4.7.4. Maintenance Requirements.

Normal servicing of dry type air cleaners includes removal of dirt from the cleaner container. Due to the omission of the container, for the LVTPX11 application, servicing requirements will occur at longer time intervals.

4.8. Engine Exhaust System.

4.8.1. General Description.

4.8.1.1. The LVTPX11 powerplant, which is located on the centerline in the stern of the vehicle, is equipped with dual exhaust manifolds, the outlets facing the rear of the vehicle. From the manifold outlets, exhaust pipes are routed to two mufflers, each located in the upper rear corner of the vehicle and mounted in a separate muffler compartment. (See drawing SK-5240).

4.8.1.2. Engine location and exhaust manifold outlet direction dictate to some extent the muffler location. The shortest routing of exhaust pipes to mufflers has been chosen to reduce any additional heating of the engine compartment. Furthermore, the design must be such as to eliminate completely the possibility of crew compartment contamination caused by a leaky exhaust system. This is of especially vital importance for a vehicle of this type where mufflers are sometimes submerged during amphibious operation, which causes more back pressure than is normally encountered.

4.8.2. Design Selection.

4.8.2.1. To overcome excessive heating of the engine compartment, the exhaust pipes are covered with a ceramic fiber insulation (Babcock and Wilcox "Kaowool" or equal) capable of a continuous-use limit of 2000°F. This material has a very low thermal conductivity. The insulation is first wrapped with a stainless steel wire mesh screen and then wrap-covered with heavy-duty self-adhering strip aluminum foil. This provides a low-cost, effective and durable method of insulating the exhaust pipes.

4.8.2.2. The design of the exhaust system piping, from engine manifolds to mufflers, utilizes flexible "ball and socket" type joints formed as an integral part of the piping. They are standard type joint assemblies as manufactured by Kay Industries, Ryan Aeronautical Company, and the Hopkins Company. Due to the design of these special joints together with the use of "Marman Clamps", at the connection of manifolds and mufflers to exhaust pipes, the usual problems of misalignments, vibration, heat expansion, and leakage have been eliminated. To avoid "freezing" and leakage of the joints a highly heat resisting and sealing compound is used to keep the joints flexible and tightly sealed at all times.

4.8.2.3. The mufflers are placed horizontally in their compartments. They are mounted solidly at the inlet, and flexibly at the outlet end to allow for heat expansion. Since the muffler compartments are completely separated from the hull interior, the openings into the hull for the exhaust pipes are sealed against water leakage. These seals are incorporated in the muffler inlet mountings. Muffler inlets and outlets are placed in such a manner to minimize the possibility of water entering the exhaust system during amphibious operations. The inlet tube is located parallel to, and vertically above, the muffler body centerline. The outlet tube is at a 60 degree angle, up and aft, perpendicular to the muffler body, and is provided with a weathercap which keeps water from entering the system. The muffler bodies have small holes in the bottom to allow drainage of water which might have collected. Normal exhaust gas pressure will keep water from flooding mufflers when submerged.

4.8.2.4. All components of the exhaust system (pipes, flexible joints, clamps and rings, mufflers, and weathercaps) are manufactured of stainless steel to insure long life and trouble free operation. The system is designed to provide a simple, low cost installation, requiring a minimum of maintenance. All components are designed to provide ease of replacement. Aluminum muffler covers are easily removable, allow quick muffler accessibility, protect against heat, and prevent infrared radiation.

4.9. Cooling System.

4.9.1. General Description.

The power train cooling system shown in drawing SK-5245, is designed to provide optimum cooling for the engine coolant water, engine lubricating oil, and converter-transmission assembly lubricating oil through the use of a dual radiator installation and oil-to-water heat exchangers. Drawing SK-5226 shows a schematic diagram of the engine coolant, engine lubricating oil, and converter-transmission lubricating oil cooling systems. The schematic diagram clearly shows the cooling systems are conventional in design. Simplicity of the system has been retained, without sacrificing the operating efficiency and reliability of the system and the individual components.

4.9.2. Design Features.

4.9.2.1. Two radiators are utilized. The radiators are installed in individual compartments located in the rear of the vehicle and on either side of the machinery compartment. This is shown in drawing SK-5165. The radiators are mounted on a 45 degree angle in the compartments and are forced-draft air-cooled by hydraulically-driven propeller type fans. The air intake is through louvered grille openings on the top side of the vehicle. The hydraulically-driven fans push the air through the radiator cores and louvered grille outlets at the rear of the vehicle. It is possible for the ducts to ship water during surfing operations; however, the exhaust grille is such that the duct will quickly drain to prevent a flooded condition.

4.9.2.2. The fan and radiator cooling system is completely isolated from (and waterproof to) the engine compartment. This arrangement insures that ambient air will be pushed through the radiator cores, giving an efficient cooling system. By pushing the air through the radiator cores, less horsepower is used. This condition occurs because the ambient air volume pushed to the cores is less than the exhausted volume due to the temperature rise through the radiator core. The direction of flow of the exhaust air gases, in a downward direction, tends to hold down any trailing dust. This system meets the requirements of the LVTPX11 design characteristics by not requiring manual operation by the driver to actuate, i.e., open or close any water or land air intake grilles or ducts. This system will eliminate water entry and flooding problems connected with an internal cooling system requiring excessively large cooling air inlets and outlets in the hull. The fan horsepower requirements are also lower due to the elimination of complicated water-expulsion type air inlet and outlet grilles and flapper valves.

4.9.2.3. The radiator construction is typically that of automotive practice. Each radiator has a bottom tank and a top tank; the top tank has adequate capacity for water expansion and water replenishment for the system. A filler neck, an overflow tube (which drains within the radiator compartment), and the pressurized filler cap are mounted on the top tank. The top tank, core, and bottom tank are adequately tied together with side members to sustain operating loads. The bottom tank of each radiator incorporates an integrally-mounted, plate-type oil cooler; one unit for the engine lubricating oil and the other unit for the converter-transmission lubricating oil. The integrally-mounted oil coolers eliminate the need for extra piping and mounting, are more efficient, and weigh less (about one-fourth) than a separate heat exchanger. The cost is also reduced considerably. Access to the filler caps is through an opening in the air inlet grille. The various components of the radiator are constructed of the following materials: top and bottom tanks, tubes, headers, inlet and outlet tubes, and filler necks are brass-solder coated; fins are copper-solder coated; plate type oil coolers are cupra-nickel; side members, mounting angles, gussets, and fan shroud are terneplate.

4.9.2.4. The radiator fans are six-blade, propeller-type, constructed entirely of aluminum alloy. The fans are hydraulically driven at 2000 rpm and have a capacity of 10,500 cfm at a total flow resistance head of 2.5 inches H_2O. They require 8.1 driving horsepower. The fans are mounted on the hydraulic motor and within faired transition shrouds which in turn are mounted to the radiators. These components are readily accessible and detachable from the radiator units. The air flow requirement ratio to radiator size requirement has been properly matched to assure minimum radiator system weight with a minimum of air-flow-horsepower.

4.9.2.5. The engine incorporates a standard dual-thermostat housing with dual parallel-flow water outlets assuring proper flow to each radiator. A thermostat and by-pass system is used on each radiator, to insure a continuous water flow to the engine if the radiator core becomes blocked. This by-pass is also used for engine warm up. The thermal opening of the thermostat starts at 170°F and is fully open at 185°F. The system pressure which is regulated by the pressure cap is seven psi. A temperature warning light and gauge system is used to give the vehicle operator a warning of adverse conditions.

4.9.2.6. A by-pass type corrosion-resistor, which is a standard Cummins Engine Company unit is used in the system. This resistor minimizes corrosion and keeps the cooling system clean by performing the following operations:

A. Treats the coolant through the addition of a chromate which is one of the most effective corrosion inhibitors known.

B. Keeps the PH value of the coolant in the range of 8.5 to 9.5 by means of an alkaline buffering chemical contained in the element.

C. Softens the coolant by means of chemicals to prevent scale formation, which can block the radiator and coolant passages.

D. Filters the coolant to remove dirt, and precipitates foreign matter which may collect on the water passage surfaces.

E. Sets up a reverse voltage by means of two megnesium resistor plates, to counteract electrolysis in the engine.

4.9.2.7. The engine lubricating oil and the converter-transmission oil are cooled by the separate, integral plate-type oil coolers mounted in the bottom tanks of the main radiators. The engine has an integral oil cooler on one side of the block which is connected in series with the bottom tank oil cooler in the starboard radiator to give adequate cooling. The engine oil cooling system incorporates a "Vernatherm" combination temperature by-pass and differential pressure relief valve. This unit by-passes the oil cooler during warm-up operation, closes at operating oil temperatures allowing the oil to circulate through the oil cooler, and allows pressure by-pass of the radiator to prevent rupture of the oil cooler during cold weather cold oil starts or plugged oil cooler conditions. The engine oil cooling system incorporates a full-flow oil filter and a temperature warning light and gauge system to give the vehicle operator a warning of adverse oil temperatures.

4.9.2.8. The converter-transmission lubricating oil is connected to the bottom tank oil cooler in the port radiator. A "Vernatherm" valve is also used in this system and performs the same function as the valve in the engine oil cooling system described in the previous paragraph. The transmission oil cooling system also incorporates a full-flow oil filter and a warning light and gauge system to indicate adverse oil temperatures.

4.9.2.9. The engine oil cooling system design and the converter-transmission oil cooling system design give relatively equal heat rejection to the water in each radiator, thereby giving a well balanced over-all cooling system. The systems have been designed to provide proper systems functioning and cooling at -40°F to +125°F vehicle operating ambients.

4.10. Controls.

4.10.1. General Description.

4.10.1.1. The driver's station includes all controls required to permit vehicle operation for normal and emergency conditions. Manually actuated control systems include: steering, speed range gear selection, water-land operation selection, service and parking braking and engine throttle control. (See drawings SK-5196 and SK-5222).

4.10.1.2. The design objective of this phase is for positive component control combined with the utmost drive efficiency through control simplification.

4.10.2. Design Selection.

In the preliminary designs presented in this Phase I Report, the Controlex

Model 538 MV mechanical flexible push-pull control unit has been specified. The use of this type control (rather than the more conventional rigid rod and bellcrank installation) provides a positive control link, greatly simplifies controls installation procedure, and largely eliminates the periodic field inspection and maintenance procedures required with conventional systems.

4.10.3. Design Features.

4.10.3.1. Control functions are transmitted from the mechanical input levers into, and from the push-pull unit through Teflon lined spherical rod ends manufactured by the Radial Bearing Company and/or the Heim Rod End Company. The vehicle driver control input crank pivots on cylindrical bearings made of special Teflon "alloy" manufactured by Garlock, Incorporated.

4.10.3.2. For complete protection against moisture, corrosive fluids, or gases, the outer case of the Controlex flexible push-pull unit is covered with vinyl plastic. These control assemblies require no lubrication throughout their operative life and will function normally in temperature variations from -85°F to +750°F. (See figure 4-10, LVTPX11 Flexible Push-Pull Control.)

4.10.3.3. It is anticipated that the mechanical controls system described will be relatively maintenance free in that the use of Teflon lined bearings at all friction points will eliminate completely the need for any scheduled maintenance procedures.

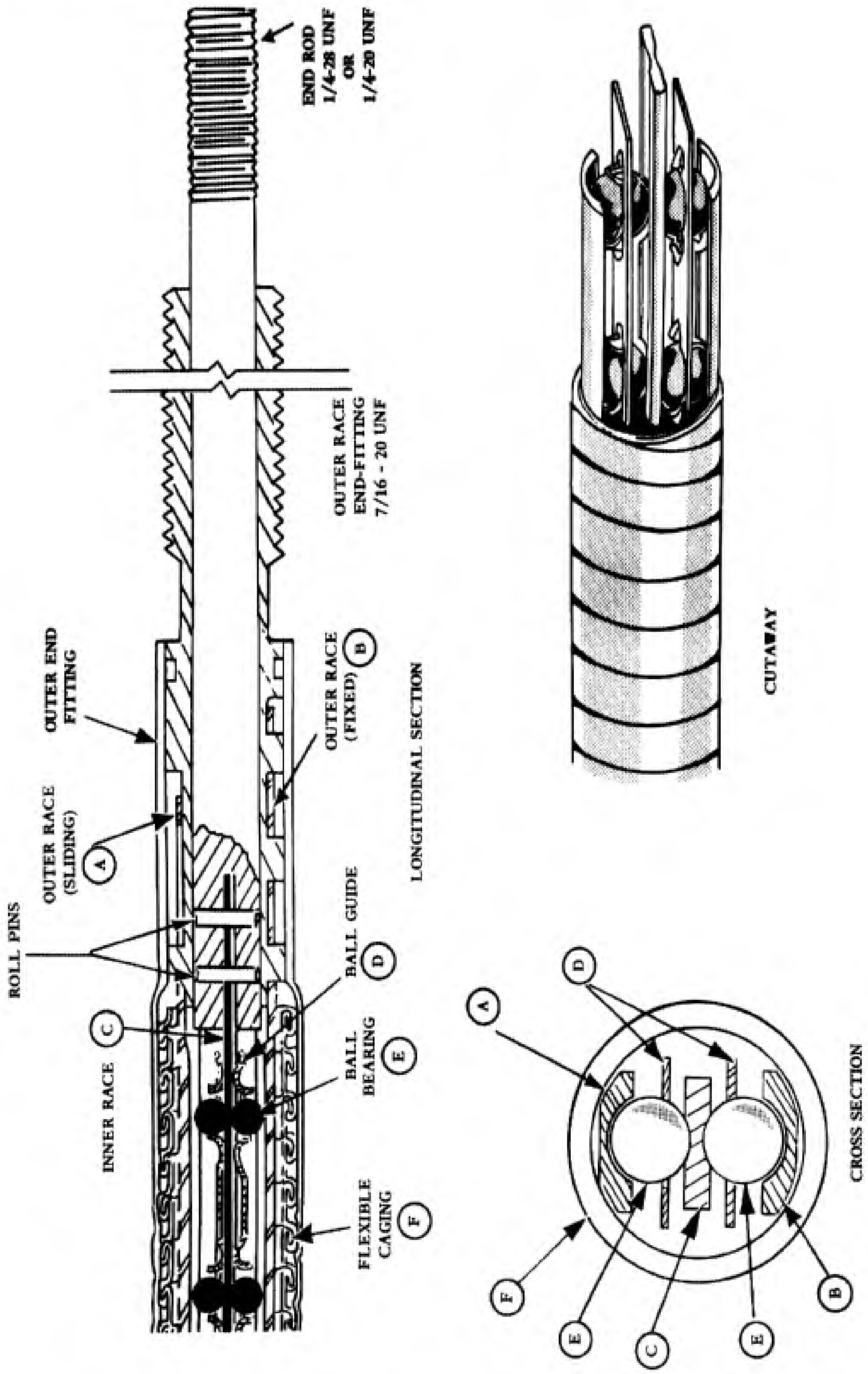

Figure 4-10. LVTPX11 Flexible Push-Pull Control

4.10.3.4. A disconnect feature, slide-out slots in all brackets, is provided for all Controlex connections to the power train components to facilitate engine-transmission removal from the vehicle.

4.10.4. Land-Water Functions.

4.10.4.1. As the transmission provides either a geared steer, normally used for land operations, or at the driver's option, a pivot steer normally used for water steer, the driver has been provided a hand operated lever to permit selection of either steering mode.

4.10.4.2. Control is transmitted through a flexible push-pull directly to a two-position, detent-held valve spool at the transmission. This valve, normally used in Ordnance vehicle applications as a two-position, pivot-steer selector valve, is used in the LVTPX11 as a "Land-Water" steering mode selector valve. When in the "Land" steering position, geared steering will be available for normal land operation. When in "Water" steering position, during normal steering with the steer control stick, a de-clutch steer will be available while in the optimum third speed range. A "hard-over" application of the steer control lever will provide pivot steer when in this water speed range.

4.10.5. Steering and Gear Select Control.

4.10.5.1. Directional and speed range selection is provided by a simple "stick" control unit, cantilever mounted from the base of the driver's instrument panel. The "stick" control is positioned much as in the LVTP5 family of vehicles.

4.10.5.2. This unit consists of the control lever with its integral reverse lockout lever and shift control output crank. This assembly is shaft-mounted to pivot longitudinally to effect the shift control function. The lever pivots in a yoke assembly which is shaft-mounted in the control base to pivot laterally to effect the steering control function. Fixed to this shaft, and reacting against the base, is a torsion-type steer centering spring.

4.10.5.3. The control lever, steering yoke, and mounting yoke are machined of 356T6 aluminum alloy castings, shafting is plated steel, the reverse lever lock and steer centering springs are of stainless spring wire.

4.10.6. Throttle - Hand and Foot.

4.10.6.1. Engine throttle control is offered primarily by an automotive type, foot-controlled accelerator pedal linked directly to the engine throttle through a flexible push-pull unit.

4.10.6.2. A secondary hand-operated throttle control is provided at the driver's left hand. This control provides a positionable "Idle" stop at the foot accelerator pedal, ranging from idle to maximum throttle. A torque-adjusting feature is provided in the hand throttle hub and mounting assembly. It is noted that the hand throttle setting may be overridden at any time by the foot pedal.

4.10.7. Brakes - Service and Parking.

4.10.7.1. The vehicle service and parking brakes are an integral part of the steering transmission and are mechanically actuated through independent control arms.

4.10.7.2. The brake actuating control arms are connected to the driver's foot actuated control pedal through dual flexible push-pull controls, permitting individual adjustment of each brake. Included in this control is a cam device to permit force multiplication as required in the control cycle.

4.10.7.3. Parking brakes are applied by depressing the foot brake pedal and pulling the parking brake "Tee" handle mounted at the lower left hand area of the instrument panel. This will actuate a foot pedal ratchet lock, locking the foot pedal in the brake apply position. To release the parking brake, the foot pedal is momentarily depressed, atuomatically releasing the ratchet lock.

4.11. Power Package Mounting.

4.11.1. The engine-transfer gearbox-transmission package is supported by a "U"-shaped horizontal frame made of square tubular members. The engine is three-point mounted to the frame in a conventional manner with two rigid supports bolted to the flywheel housing and a rigid trunnion mount at the front of the engine. The transfer gearbox is rigidly mounted to the engine flywheel housing. The transmission is hung on two trunnions connected to the output housings of the transmission and rigidly mounted to the box frame, while the front mount is rigidly connected to the input cover of the transmission and the box frame.

4.11.2. The mounting frame is resiliently mounted in the hull by four sets of tube-form mounts. These mounts are of the captive-controlled, combination shear-compression type and will still support the power package if failure occurs. They are used to minimize the vibration and shock loads. (The mountings are manufactured by Lord Manufacturing Company).

4.11.3. The detailed engine-transfer gearbox-transmission mount design is predicated on the necessity of controlling stresses, deflection and weight, as well as assuring the required alignment of the components. The alignment of the components is obtained by shimming of the prime mover. Mounts are designed to withstand acceleration loads of five G's in a positive and negative vertical direction, three G's in a sideways direction and three G's fore and aft. The mounting of the three components to the frame can effectively be done outside the vehicle, and can be installed in the vehicle as one unit or removed as one unit. Also, the engine and transfer gearbox have the capability of being removed as a unit without dismounting the transmission. Refer to drawing SK-5224 for power package mounting.

4.11.4. The power package mounting attachment points and all cooling, exhaust, aspiration, oil, fuel, and electrical connections to the engine, transfer gearbox and transmission are readily accessible and easily disconnected to facilitate removal of the power package assembly.

LVTPX11

PHASE I

APPENDIX I of SECTION 4

The following section includes the analysis of powerplant and transmission selection.

Choice of Powerplant and Transmission

The selection of the power train machinery has been based upon the design requirements for achieving the desired vehicle performance. A study of vehicle performance has been made from theoretical calculations and experience factors, and evaluation of this data has provided a basis for the selection or design of optimum power train components, ratios, and sizes, matched to the requirements. In this manner it is considered that the possible attainment of the performance goals is enhanced by provision of a complete power train which is the most efficient in size, weight, and performance by being matched to the installation.

A comprehensive search for and study of existing power train components, including standard commercial and military engines, transmissions, and related hardware adaptable to the power and performance requirements of this installation, has been made with leading, qualified manufacturers. Performance estimates were made, along with preliminary design layouts to evaluate the vehicle requirements for the installation of various combinations and arrangements of machinery.

Powerplant Selection

The powerplant, or engine, is the primary component of the power train to be considered. The choice of the powerplant must be based on the primary consideration of performance, i.e., power and torque required and the manner in which they are obtained. Secondary to this are considerations of the type of fuel used, powerplant weight, size, availability, cost, and compliance with military specifications. The final selection, between possible engines, is based on the best over-all installation compatibility. A basic criterion for final evaluation has been the desire to provide an engine which meets the requirements while offering the greatest mechanical simplicity and reliability; an engine with design which is free of complex devices which could give trouble and cause failure or malfunction in the field during combat.

Selection of the powerplant has been made from evaluation of a number of possible compression-ignition engine designs. Leading engine manufacturers were canvassed for their recommendations to determine the availability of engines of the required ratings. The engine requirements were established on the basis of a 35,000-pound gross vehicle weight with a single engine. From preliminary design studies, it was determined that for a vehicle of 35,000 pounds gross vehicle weight, the optimum engine installation would place a single power package centrally in the vehicle hull. For complete evaluation of this installation, information on engines in the nominal 300 brake horsepower rating class was obtained.

There are several compression-ignition engines in this class which are extremely similar in physical dimensions and performance. The pertinent characteristics of the several possible engines are summarized in table A, below:

Table A. Comparative Engine Data

Manufacturer	Cummins Engine Company	International Harvester Company	Caterpillar Tractor Company	General Motors Corp.
Model	V8-315	DVT-573	D-333	6V-71T
Cylinder	8-90°V	8-90°V	6	6-63.5°V
Rated Horsepower	315	300	300	332
Rated RPM	3000	3000	2400	2300
Peak Torque, LB-FT	606	565	750	790
Peak Torque, RPM	2150	2200	1600	1750
Type Cycle	4	4	4	2
Multifuel	To C.I.E	To C.I.E	Multi	Multi
BSFC at Rated Power	.42	.413	.40	.40
Supercharged	No	Yes	Yes	Yes
Weight in Pounds	1849	1900	1980	1875
Length in Inches	43.75	42.0	53.40	50.0
Width in Inches	34.18	33.6	31.0	37.3
Height in Inches	41.81	38.5	42.19	57.3
Availability	Production	Production	Production	Production
Cost - Production $	3515-3750	4000-5000	3100-3800	-

The Cummins V8-315 engine is selected as the powerplant for the LVTPX11 on the basis of the following:

1. Rated power and speed. The rating of 315 brake horsepower appears to match vehicle requirements most closely. The maximum speed of 3,000 rpm agrees with the trend to higher speed and lighter components in the power train and meets vehicle maximum speed specifications with existing transmission ratios.

2. Weight. It has the least weight of the presently available fully-developed 300 brake horsepower class engines.

3. Size. Compared with the other engines evaluated, the Cummins engine has the least over-all length and comparable width and height.

4. Compatibility. This engine is a completely satisfactory match to the Allison XTG-250 power train, at the upper limit of torque converter rating.

While this engine is presently being manufactured with a cast-iron block, it will be available in aluminum if the demand warrants aluminum production.

It is presently qualified for use of diesel fuels, the JP-series of fuel, and C.I.E.-fuel. Development work is being accomplished on full multifuel usage, and the manufacturer states that multifuel capability can be added when required.

A factor in the selection of this engine is its projected use in the U. S. Army, Transportation Corps, LARC 5 and LARC 15 vehicles, which will establish it as a military-qualified engine.

Since this engine is presently naturally-aspirated, it offers considerable growth potential in power rating through turbo charging should the requirement for more power be established.

At least two alternate engines should be specified so that, in case of war or any other emergency, powerplants for the LVTPX11 would be available from different sources. This multisource requirement can be met with minimum compromise with the following engines.

The International Harvester DVT-573 engine has characteristics very nearly matching the Cummins V8-315 and must be considered as a prime alternate. Therefore, it also matches well to the Allison XTG-250 power train. It weighs 51 pounds more than the Cummins engine and is slightly longer.

The Caterpillar D-333 engine is another possible alternate powerplant because of its power rating. However, it is heavier and considerably longer, and develops its rated power at a relatively low speed (2,400 rpm). In addition, it develops too much torque (770 lb-ft) to consider matching to the Allison XTG-250 power train without adding a speed-up gear ratio between engine and transmission, which would require redesign of the transfer gear.

The General Motors 6V-71T engine may be considered as a possible alternate powerplant. It is a high production engine and is used in other military vehicles. It is available as a multifuel engine and can be matched to the Allison XTG-250 transmission by using a speed-up ratio in the transfer gearbox. Its height is satisfactory but it is slightly longer and higher than the Cummins engine.

Transmission and Related Powertrain Component Selection.

The selection of the design of the individual units comprising the powertrain, from the engine to the drive sprockets, has been based principally upon the requirements for vehicle performance as reflected in ratios and number of transmission ranges, etc., physical size and space considerations, and compatibility with mating components so as to form a complete and optimum power train with minimum compromise.

Evaluation of Transmission Types.

In meeting the requirements of the vehicle specifications and in the interest of providing the optimum powertrain, investigation and study has been made of the following general transmission types:

1. Mechanical
2. Hydraulic
3. Electrical

Discussion.

The fundamental purpose of a transmission is to match the prime mover power output characteristics to the vehicle requirements. This is accomplished by transmitting the power from the prime mover to the vehicle drive outputs at the desired torque and speed to provide the required vehicle performance. Thus the transmission is a matching device which allows adaption of a prime mover with specific speed and torque characteristics to the speed and torque requirements of the vehicle drive outputs. The transmission must provide ratio coverage for the maximum speed required as well as the maximum tractive effort required. For the amphibian, where water propulsion is required by means of the tracks, an optimum ratio for water operation is desirable.

For the tracked vehicle, it is required that the transmission also provide the function of steering, since steering is accomplished by control of the speed and/or direction of movement of the tracks relative to each other. This is most efficiently done in the main transmission. It is desirable that regenerative steering be possible as well as a pivot steer capability. Transmissions can also provide vehicle braking and thus eliminate the need for separate braking mechanisms.

For purposes of comparing transmission fundamental types, however, there is a natural division of transmissions into three types involving the relation between the input and output characteristics of speed and torque. These three types are:

1. Fixed ratio

2. Limited variable ratio
3. Continuously variable ratio

An example of the fixed ratio type of transmission is the multiratio gearbox with either a friction clutch or a fluid coupling. The output torque is a direct function of the input torque and the gear ratio. The output speed is a function of the input speed and the reciprocal of the gear ratio. The fixed ratio transmission requires power dissipation for stall torque and shift points due to clutching or fluid coupling dissipation, with no output multiplication other than the fixed ratio.

An example of a limited variable ratio transmission is a torque converter, with or without gearbox. The output torque is a function of a combination of the input speed and input torque. The output speed is independently variable up to a limit of less than the input speed for the most common designs. Stall torque is produced by power dissipation because slip is used to generate torque.

An example of a continuously variable ratio transmission is a hydrostatic type. Output speed is independent of input speed, and output torque is independent of input torque. Input power is a function of output power and the input power can be produced at any desirable combination of speed and torque that is compatible with the prime mover characteristics. Stall torque is produced without slip and the transmission does not require power dissipation to produce this torque.

In performance, a transmission must give the vehicle the mobility and flexibility needed to complete its mission. When extreme mobility is required, tracked vehicles are used. Their transmissions must give high torques for quick accelerations, accept high torques for sudden braking, and have these torques infinitely variable in order to perform turns ranging from pivot to infinite radius.

Transmission efficiency affects the mission capabilities, logistics for the vehicle, and may affect the vehicle performance. In judging the efficiency of transmissions, three factors must be considered: (1) the transmission efficiencies for part loads as well as rated loads are needed in order to estimate the efficiency of steady-state speed operation, (2) accelerations, steering, and low speed pulling or pushing must be considered in terms of heat rejection, (3) finally, to obtain the affect on over-all fuel consumption, consideration must be given to the extent that the transmission allows the engine to operate efficiently.

The transmission must be designed for minimum weight and size. The space and weight saved by an optimum-sized transmission may be used for increased vehicle capacity, increased ground clearance, lower silhouette height, or reduced vehicle weight, providing more maneuverability.

It is considered to be beyond the scope of this study to enter into a full description and analysis of the fundamentals of each transmission type, and the related factors regarding vehicle performance.

With the above considerations in mind, evaluation of existing transmission designs has been made. In addition, other considerations offered are newly designed mechanical, hydraulic and electric transmission systems tailored for this installation.

Mechanical Transmissions.

A mechanical transmission has been selected as the primary choice of transmission type because of its proved, reliable, and conventional design, its high efficiency, and low cost.

The approach taken here is to consider the use of an existing, production transmission which is being developed for other military vehicle programs. This would offer a satisfactory power train at minimum cost, current availability, and a high degree of standardization and interchangeability in the military system.

The choice of available, fully-developed transmissions suitable for powering tracked military vehicles is, indeed, limited. The single, established source for power trains for military tracked vehicles is Allison Division of General Motors. Other specialized transmissions have been developed by companies such as FMC Corporation; however, these have limited applicability outside of the special vehicles for which they were developed.

The Allison XTG-250 powertrain is the only available, fully-developed transmission design suitable for installation in the LVTPX11. Other transmission models developed by Allison are not suitable with regard to physical size, ratings or ratios.

The Allison Model XTG-250 powertrain matches fairly well to the LVTPX11 vehicle requirements and is considered to be a marginal but suitable transmission. It matches satisfactorily to several leading, qualified, compression-ignition engines, thereby permitting considerable flexibility in its application. It is a developed unit available with or without integral final drives. The ratio coverage is marginal. It is available at present only with magnesium housings; however, it is felt that aluminum housings are to be developed.

The XTG-250 unit would be utilized for the LVTPX11 installation less the final drives, since the standard final drive ratio is non-optimum. Also, it is advantageous to provide a separate final drive with a geared reduction to obtain the offset necessary to the track compensator design.

The present ratings of the XTG-250 unit are for maximum input horsepower of 250, maximum input torque of 520 pound-feet, and maximum input speed of 2,800 rpm. It is classed for use in a 34,000-pound vehicle. These ratings have been discussed with Allison representatives and they have tentatively approved use of the XTG-250 with the Cummins V8-315 engine and in a vehicle of 35,000 pounds gross weight.

The XTG-250 unit is presently qualified to function on a 60 percent forward slope and 40 percent side slope. The manufacturer foresees no extreme problem in qualifying it for operation on a 70 percent forward slope and 60 percent side slope.

LVTPX11

PHASE I

APPENDIX II of SECTION 4

The following power train design analysis is presented to record preliminary analyses made during Item 1 study, and to illustrate the format for compilation and presentation of design philosophy and data to be developed during Item 2 detail design.

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION

U.S. Navy
Bureau of Ships
Contract NObs 4561

Landing Vehicle Tracked

LVTPX11

POWER TRAIN DESIGN ANALYSIS

1962

PRELIMINARY

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Sheet Number — Title

Sheet Number	Title
See Note	Driveshafts
See Note	Engine Aspiration Air System
59-64	Exhaust System
65-78	Cooling System
See Note	Controls
See Note	Powerpackage Mounting

NOTE: This section to be completed during Item 2 detail design

NObs 4561 LVTPX11
Power Train Design Analysis

VEHICLE BASIC DESIGN PARAMETERS

Summary of Leading Specifications

Physical Characteristics. -

Design Maximum Gross Vehicle Weight		35,000 lbs
Design Maximum Hoisting Weight		27,000 lbs
Design Cargo Capacity (with crew of 2)		8,000 lbs or 29 Fully Equipped Personnel
Length Overall (LOA), Maximum		21'-3"
Beam, Maximum		10'-6"
Height Overall, Maximum		9'-0"
Cargo Space:	Length, Min.	12'-0"
	Width, Min.	6'-0"
	Height, Min.	5'-6"
Ground Clearance		18"
Angle of Approach		65°
Angle of Departure		45°

Performance Characteristics. -

Speed:		
Land - Forward (Hard, Level Surface)		40 mph
Land - Forward (Cross-Country)		20 mph
Land - Forward (Minimum sustained)		2.5-5.0 mph
Land - Reverse	Maximum not less than	5 mph
Water - Forward	Maximum attainable	Target 7 mph
Water - Reverse		Target 4 mph
Braking Ability	Stop and Hold On	70% grade
Gradability: -		
Forward Slope	Forward or Reverse Gear	70%
(From standing start to 2 mph minimum forward)		
Side Slope		60%

NObs 4561 LVTPX11
Power Train Design Analysis

VEHICLE BASIC DESIGN PARAMETERS

Summary of Leading Specifications (Cont'd)

Stability:	
Water - Fully Loaded	100° roll to P/S
Land - Fully Loaded	90° turn on 60% side slope
Trench Crossing Ability	8 ft wide X 4 ft deep
Vertical Obstacle Ability	3 ft high
Surfability - Fully Loaded	10 foot plunging surf
Design Unit Rolling Resistance:	
Off-Highway, Grades	140 lb per ton
Hard Surface	100 lb per ton
Design Ambient Temperature Range	-40° to 125° F.
Design Performance Temperature	80° F.
Design Operating Life - Minimum Overhaul (TBO)	1,000 hours
Land Operation	800 hours
Water Operation	200 hours
Endurance:	
Land - Hard, Level Terrain	300 miles at 25 mph
Water	6 hours at Max Speed (whichever is greater)
Fuel Capability	JP-4, JP-5, Diesel fuel oil, Army CIE -fuel

NObs 4561 LVTPX11
Power Train Design Analysis

PERFORMANCE ANALYSIS

See Section 3.0 of Basic Report

NObs 4561 LVTPX11
Power Train Design Analysis

POWER TRAIN DATA

Power Train Design Parameters

Design Gross Vehicle Weight	35,000 lb
Design Speeds:	
Land - Forward, Maximum (Hard, Level Surface)	40 mph
Land - Reverse - Maximum not less than	5 mph
Water - Forward - Maximum attainable	Target 7 mph
Water - Reverse	Target 4 mph
Braking Ability - Stop and Hold On	70% grade
Gradability - Forward and Reverse Grade (From standing start to 2 mph minimum forward)	70%
-Side Slope	60%
Design Performance Temperature (at Sea Level)	80°F.
Design Power Train Component Life (TBO) (Based on vehicle operation of 800 hr. land and 200 hr. water)	1,000 hours
Design Rolling Resistance:	
Off-Highway, Grades	140 lbs per ton
Hard Surface	100 lbs per ton

POWER TRAIN DATA (Cont'd)

Power Train Components Data

Basic Power Train

Prime Mover	One Cummins V8-315 Compression-Ignition Engine 309 BHP at 3,000 rpm 593 lb-ft at 2,150 rpm (80°F, Sea Level)
Transfer Drop Gear	Borg-Warner Spur Gear Ratio 1:1
Transmission	Allison XTG-250-2 with Remote Input and Concentric Output

Overall Mechanical Gear Ratios:

1st	4.18 x 1.478 x 1.44 = 8.921
2nd	4.18 x 1.00 x 1.44 = 6.036
3rd	2.24 x 1.00 x 1.44 = 3.234
4th	1.00 x 1.00 x 1.44 = 1.44
Reverse	2.24 x 2.095 x 1.44 = 6.776

Converter Stall Torque Ratio 2.55:1

Final Drive	Two Borg-Warner Spur Gear Ratio 3.3:1 (Reduction)
Sprocket	Rear Drive Pitch Dia. - 21.3 inches

NObs 4561 LVTPX11
Power Train Design Analysis

POWER TRAIN DATA

Power Train Components Data (Cont'd)

Auxiliaries and Accessories

Electrical System.-

Alternator	One Leece-Neville 24-volt, 125-amp

Hydraulic System.-

Main Hydraulic Pump	One Weatherhead variable-displacement pressure-compensated piston pump, 2,400 rpm 3.2 cu. inch per rev, 3,000 psi

Supplying power for:

Cooling Fans	Two 23-inch diameter, 6-blade, propeller-type
Bilge Pumps	Four centrifugal 300 gpm each
Auxiliaries	Ramp actuator Track compensator

Lubrication System.-

Transfer Gearbox Lubrication Pump

Design Operating Conditions

The following design operating conditions have been selected as the primary conditions to be utilized in the design analysis of the power train systems.

Drop Gear Transfer Box

The design operating conditions for the drop gear transfer box are based on engine output. The following three engine output points were selected:

Cond.	Engine-Output	Design Life
1	Maximum Torque	20%
2	Average Load	50%
3	Maximum Horsepower	30%
	Total Operating Life:	100% = 1,000 Hours

The average load point is assumed to be at a point 2/3 between maximum torque and maximum horsepower point. The engine output speed "N_A" at average load point is then:

$$N_A = \frac{2N_M + N_T}{3}$$

Where N_M is engine governor speed (maximum horsepower point) and N_T is engine speed at maximum torque output.

XTG-250 Allison Transmission

NObs 4561 LVTPX11
Power Train Design Analysis

The design life calculations for the Allison transmission are based on an estimated duty cycle. This duty cycle for the transmission is derived from the following average vehicle operating conditions:

Land Operation

2-3 MPH at 70% Slope - 5%
5-7 MPH at 20% Slope - 25%
15 MPH at 8% Slope - 35%
Maximum Speed - 10%
Reverse 2 to 3 mph - 5%

Total Land Life: 80% = 800 hours

Water Operation

Maximum Speed - 15%
Reverse - 5%

Total Water Life: 20% = 200 hours

Duty Cycle for XTG-250 Transmission

This duty cycle is for full-throttle operation, total design life is 1,000 hours.

Transmission Ratio	Vehicle Speed MPH	Design Life Percent
1st-Gear		
Converter Drive	2-3	5
Lockup	5-7	10

Duty Cycle for XTG-250 Transmission (Cont'd)

2nd-Gear		
Converter Drive	5-7	15
Lockup	8-10	5
3rd-Gear		
Converter Drive	10-12	20
Lockup	13-15	15
4th-Gear		
Converter Drive	18-20	10
Lockup	35-38	10
Reverse		
Converter Drive	2-3	5
Lockup	5-7	5

Final Drive

The design conditions for the final drive are derived from previous experience.

	Operating Conditions		
Condition	Vehicle Speed MPH	Slope Percent	Design Life Percent
1	5	Steering Condition Based On $f = 1.0$	5
2	14	20	60
3	25	8	35
		Total Life:	100% = 1,000 hours

NObs 4561 LVTPX11
Power Train Design Analysis

ENGINE DATA

Engine Specifications

The power plant with characteristics and performance indicated to be most favorable for powering the LVTPX11 is the Cummins Model VINE V8-315 compression-ignition engine, manufactured by the Cummins Engine Company, Inc., Columbus, Indiana.

The Cummins Model VINE V8-315 is a 4-cycle, 90 degree V-type, 8-cylinder engine of 785 cubic inches displacement. It is a naturally-aspirated engine rated at 315 bhp at 3,000 rpm, and 606 lb-ft maximum torque at 2,150 rpm corrected to 60°F. and sea level conditions. The compression ratio is 15:1. It features an "over-square" design with a bore of 5-1/2 inches and a stroke of 4-1/8 inches. The nominal size dimensions are: length - 42 inches, width - 39 inches, and height - 42.5 inches. The nominal net weight with cast-iron block and standard accessories is 1,775 pounds.

The Model VINE V8-315 engine is an up-rated version of the standard commercial Model VINE V8-365. The higher rating is obtained through the use of special fuel metering and operation at higher speed.

Summary of Cummins V8-315 Engine Specifications

Horsepower (at 60°F, Sea Level)		315
Governed RPM		3,000
Maximum Torque at 2150 RPM, Lb-Ft		606
Number of Cylinders		8
Bore, Inches		5-1/2
Stroke, Inches		4-1/8
Piston Displacement, Cubic Inches		785
Angle of V, Degrees		90
Operating Cycles		4
Compression Ratio		15:1
Net Weight, with Standard Accessories (Lb)		1775
Pounds per Horsepower		5.9
Dimensions: (Nominal)		
	Length, Inches	42
	Width, Inches	39
	Height, Inches	42.5

Engine Specifications (Cont'd)

Aspiration Air Flow at 3000 rpm	600 cfm
Heat Rejection	32.5 Btu/HP/min
Maximum Fuel Filter Restriction	8 inch Hg
Maximum Inlet Air Restriction	25 inch H_2O

ENGINE DATA

Engine Performance Characteristics

Engine Gross Rated Performance

Cummins Engine Co., Inc., Columbus, Indiana has established the maximum rating of the basic V8-265 engine at 315 BHP at 3,000 rpm under Columbus test conditions. This is the maximum horsepower available in the naturally-aspirated state with an acceptable smoke rating, and assuring reliability and durability.

Reference: Cummins Performance Curve No. RP-1691, Fig. A

Engine Model: Cummins V8-315 (VINE)

Cummins Test Conditions: (Corrected) 60°F., Sea Level

Engine Gross Performance Includes:
- Air Cleaner
- Generator (1 HP)
- Water Pump
- Oil Pump
- 1 inch Hg Exhaust Loss
- 10 inches H_2O Inlet Loss

Corrected Engine Gross Rated Performance

The LVTPX11 specifications require performance ratings at 80°F., and sea level (29.92 inches Hg).

Corrected Engine Gross Rated Performance (Cont'd)

Cummins Curve No. RP-1691 then must be de-rated as noted, i.e., 1% for each 10°F. increase in temperature above 60°F., or a total of 2% to conform to the vehicle installation design condition of 80°F. ambient temperature.

The following table shows corrected data. Curve as shown in figure A plots this data.

Table A. Cummins V8-315 Performance 80°F., Sea Level

Engine RPM	HP at 60°F	HP at 80°F	Torque, lb-ft at 80°F
1400	147	144	540
1800	204	200	584
2200	253	248	593
2600	289	283	572
3000	315	309	542

Engine Net Rated Performance

In order to properly evaluate the Cummins V8-315 engine for the LVTPX11 installation, the additional auxiliary and accessory losses unique to the vehicle must be considered in determining a net engine curve for use in evaluating transmission performance and predicting vehicle performance.

Essentially, the design performance of the vehicle, and particularly fuel consumption and endurance performance, is considered to be derived optimumly from an average operating condition. Therefore, the following determination of net engine performance is based on average accessory losses derived from assumed normal vehicle operation, as opposed to maximum operation during which certain accessories may be absorbing full load. (Note: The critical minimum performance areas of the vehicle are properly considered at another point.)

Also, it may be seen that typical operation on land and water may utilize different accessories; e.g., bilge pumps during water operation. Therefore, in a refined analysis different accessory loss values may be determined for each operating regime.

Accessory Losses

The LVTPX11 utilizes three principal accessory systems which are driven off the engine:

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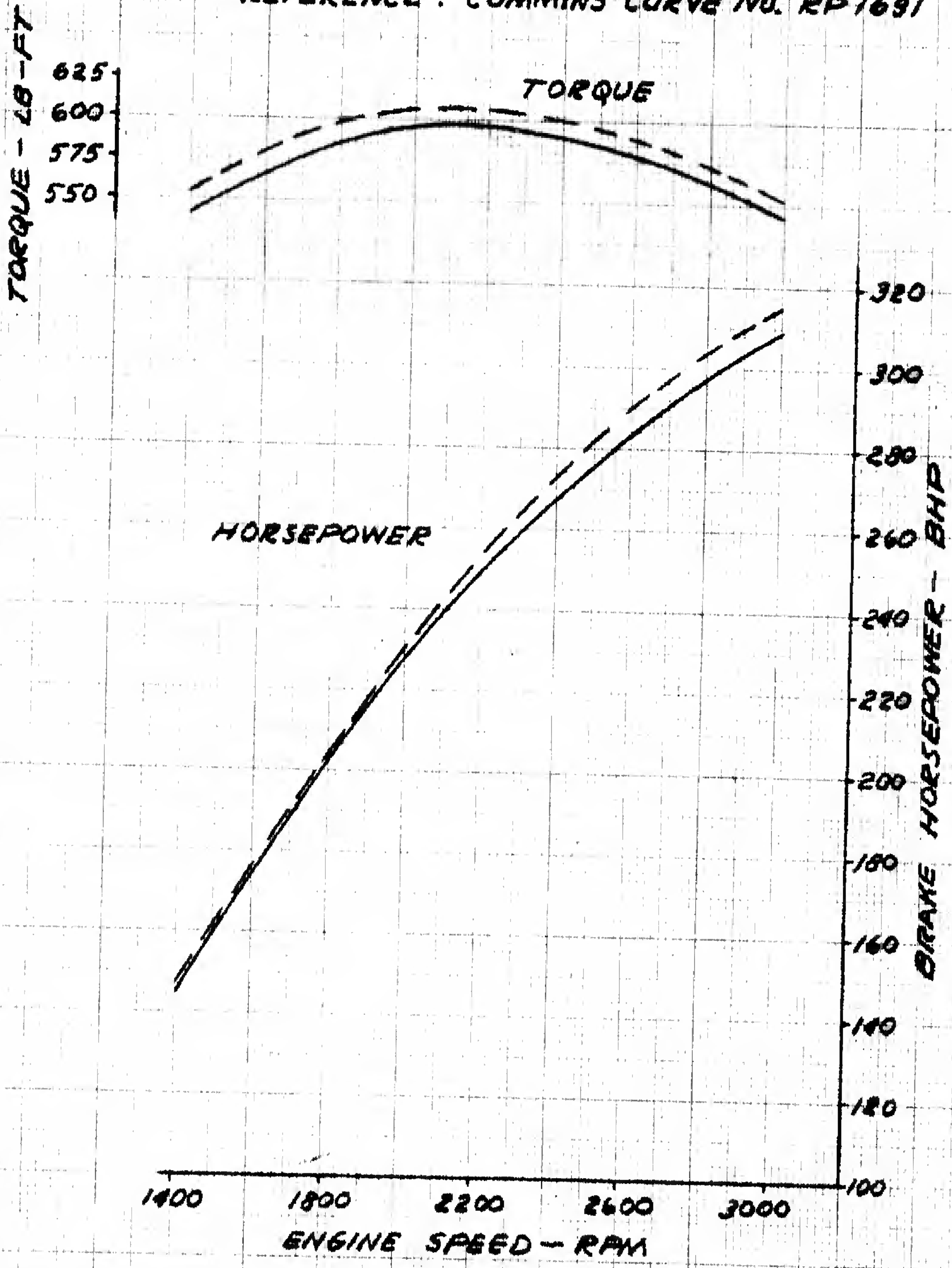
CONTRACT NObs 4561 LVTPX11

ENGINE GROSS PERFORMANCE

ENGINE: CUMMINS V8-315
REFERENCE: CUMMINS CURVE NO. RP 1691



DOTTED LINE CORRECTED TO 60°F INTAKE AIR TEMPERATURE AND 29.92 IN. HG. BAROMETRIC PRESSURE (SEA LEVEL)

SOLID LINE CORRECTED TO 80°F INTAKE AIR TEMPERATURE AND 29.92 IN. HG. BAROMETRIC PRESSURE (SEA LEVEL)

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FIGURE A CUMMINS V8-315 ENGINE GROSS PERFORMANCE

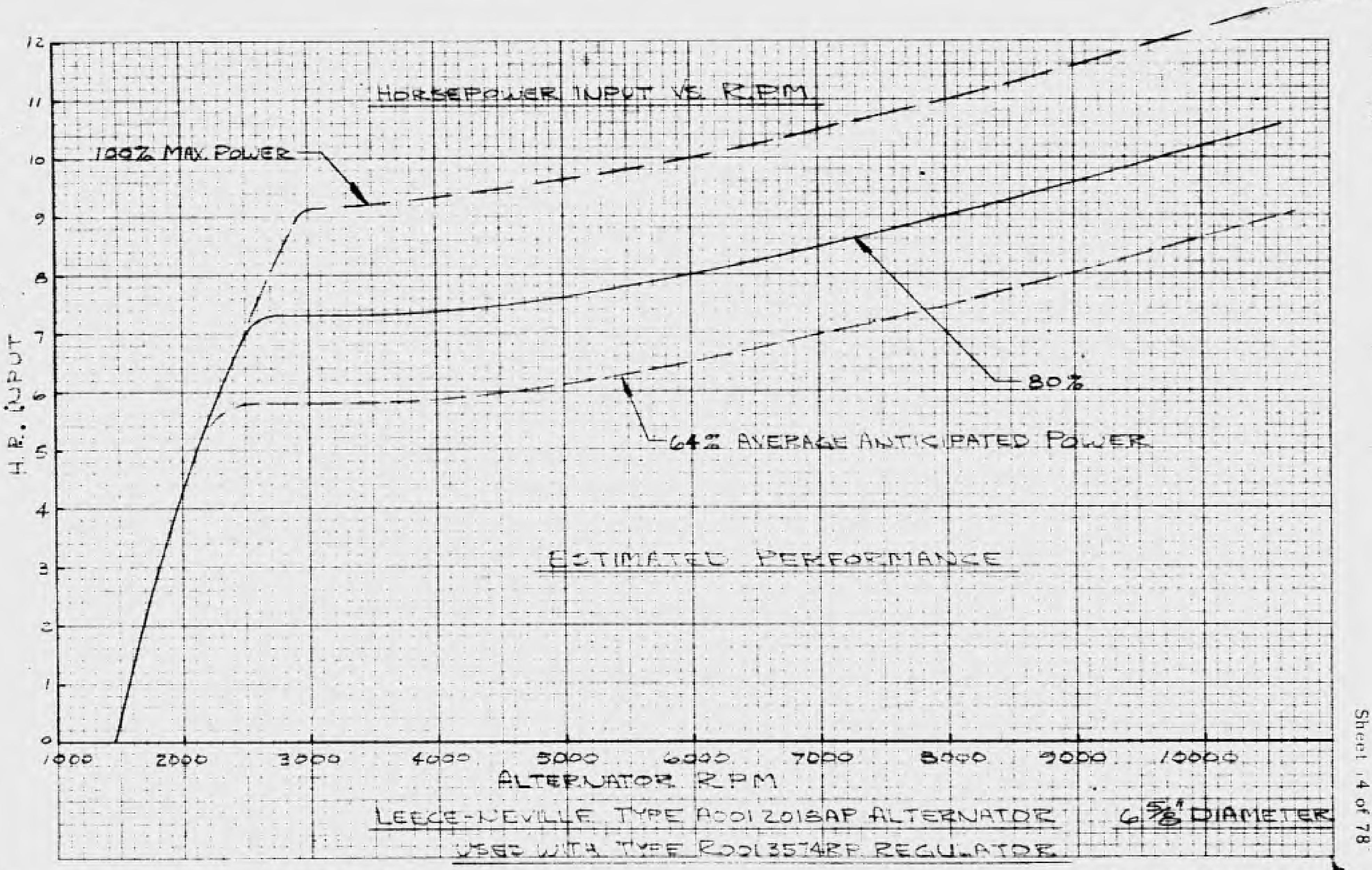

FIGURE 5 ESTIMATED PERFORMANCE OF LEECE-NEVILLE TYPE A0012018AP ALTERNATOR

1. Electrical System

The LVTPX11 electrical system is a 24-volt system powered by lead-acid batteries. The charging of the batteries is done by an alternator which is driven directly off the engine, using a speed-up drive ratio of 3.333 to 1.

For average operating conditions it is estimated the load on the alternator is 64 percent (equal to 80 amperes) of its full load capacity of 125 amperes. The power absorption of the alternator drive for this load condition is obtained from the curve of figure B.

Engine RPM	Alternator	
	RPM	Input HP
1400	4668	6.1
1800	6000	6.5
2200	7334	7.1
2600	8667	7.8
3000	10,000	8.6

2. Hydraulic Accessory System

The hydraulically-driven components are powered by the main hydraulic pump which is driven off the transfer drop-gear box. The drive speed ratiobetween engine rpm and pump rpm is a 1.25 to 1 reduction.

The nominal horsepower input to the main hydraulic pump is calculated below for the flow requirements of the components.

a. Cooling Fan Motors

The installation has two cooling fans driven by hydraulic motors. Each fan absorbs a nominal eight horsepower according to estimates of the cooling requirements.

The hydraulic fan motor requires a flow of 5.62 gpm to produce eight horsepower at 2,100 rpm. These data were obtained from available motor performance characteristics.

b. Bilge Pump Motors

The vehicle is equipped with four bilge pumps which operate continuously, each driven by a separate hydraulic motor. They are equipped with a load-sensing device, which throttles back the flow under no-load conditions, e.g., during land operation.

Bilge pump flow requirements:

No-load condition (land operation)	0.6 gpm
Full-load condition (water operation)	2.6 gpm

c. Total Flow Requirement of Pump

Land Operation: GPM = 2(5.62) + 4(0.6) = 13.64
Water Operation: GPM = 2(5.62) + 4(2.6) = 21.64

d. Maximum HP-Input to Main Pump

$$HP = \frac{GPM \times PSI}{1714} \times \frac{1}{\eta}$$

System Pressure: 3,000 PSI
Pump Efficiency: $\eta = 95\%$

Land Operation:

$$HP = \frac{13.64 \times 3000}{1714 \times .95}$$

$$HP = 25$$

Water Operation:

$$HP = \frac{21.64 \times 3000}{1714 \times .95}$$

$$HP = 40$$

e. HP-Input at Various Engine Speeds . . . . can be determined from the appropriate curve. (See figure C)

During land operation, the main hydraulic pump is absorbing a constant 25 HP.

For water operation, HP-Input to main hydraulic pump is given below:

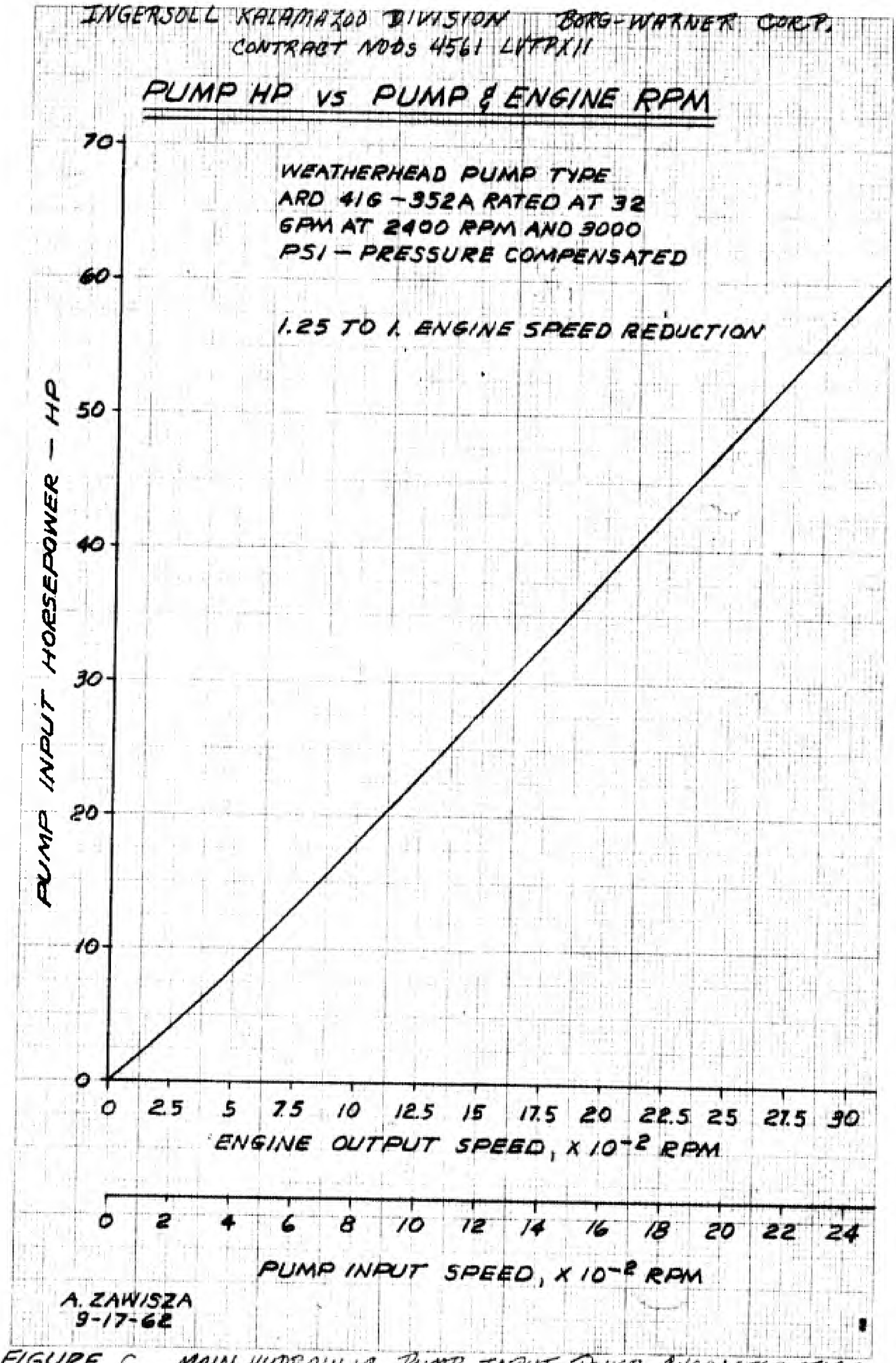

FIGURE C MAIN HYDRAULIC PUMP INPUT POWER CHARACTERISTICS

Engine RPM	RPM	Main Hydraulic Pump HP-Input
1400	1120	27
1800	1440	34
2100	1680	40
2200	1760	40
2600	2080	40
3000	2400	40

The losses due to other auxiliary hydraulic services, on the vehicle, e.g., rampactuation and track compensator, are considered negligible due to their intermittent nature.

3. Lube System

The transfer gearbox lube pump, which is also driven off the transfer gearbox, absorbs approximately two horsepower over the entire speed range.

Transfer Gearbox Loss

For convenience in performance estimates, engine net rated performance for this installation is considered to include the losses in the transfer gearbox.

The over-all efficiency of the transfer gearbox is estimated at 96 percent, giving a straight loss of four percent of transmitted horsepower.

Available HP and Torque at Transmission Input

To obtain the net available HP and torque at the transmission in-

NObs 4561 LVTPX11
Power Train Design Analysis

Table B. Accessory Loss Summary

Engine		Accessory Losses			Transfer Gear Loss HP	Total Losses HP	Net Available Transmission Input	
RPM	Gross HP	Alternator HP	Main Hydraulic Pump HP	Lube Pump HP			HP	Torque Lb-Ft
LAND OPERATION								
1400	144	6.1	25	2	5.7	38.8	105.2	394
1800	200	6.5	25	2	8.0	41.5	158.5	463
2200	248	7.1	25	2	9.9	44.0	204.0	497
2600	283	7.8	25	2	11.3	46.1	236.9	480
3000	309	8.6	25	2	12.4	48.0	261.0	457
WATER OPERATION								
1400	144	6.1	27	2	5.7	40.8	103.2	387
1800	200	6.5	34	2	8.0	50.5	149.5	437
2100	240	7.0	40	2	9.6	58.6	181.4	454
2200	248	7.1	40	2	9.9	59.0	189.0	451
2600	283	7.8	40	2	11.3	61.1	221.9	448
3000	309	8.6	40	2	12.4	63.0	246.0	431

The engine net rated performance for LVTPX11, i.e., net available horsepower and torque at the transmission input, is shown in Figure D.

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ENGINE NET PERFORMANCE AT TRANSMISSION INPUT

Engine	:	Cummins V8-315
Reference	:	Cummins Curve No. RP-1691
Conditions	:	80°F Inlet Air, Sea Level
Auxiliaries	:	Alternator, Main Hydraulic Pump, Lube Pump, Transfer Gearbox

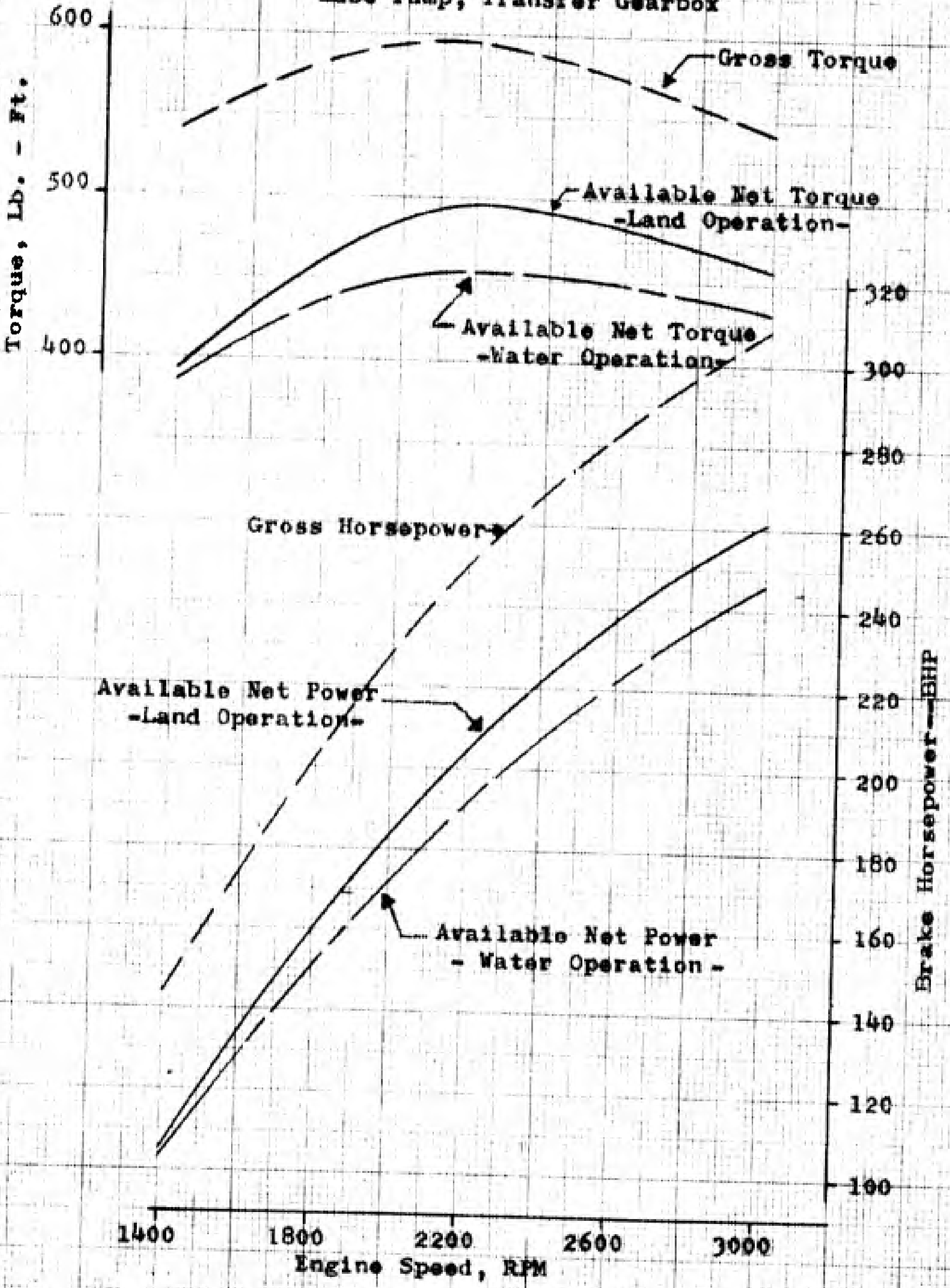

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FIGURE D - ENGINE NET PERFORMANCE AT TRANSMISSION INPUT

put, the above determined accessory losses and the drop gear loss of four percent must be subtracted from the engine gross performance.

TRANSFER GEARBOX

Introduction

Design Approach

Because of the unique arrangement of the engine over the transmission in the basic LVTPX11 design, it is necessary to transfer the line of drive from engine to transmission over a vertical distance of approximately 27 inches between the centerline of the engine crankshaft and the transmission input center. Also, because the engine is oriented 180 degrees from the normal engine relationship to the transmission, it is necessary to provide a drive arrangement which, when viewed from the flywheel end of the engine, gives opposite-hand rotation of the output with respect to the input. In addition, there is a requirement for an auxiliary drive for the vehicle main hydraulic pump.

These requirements are satisfied with a transfer gearbox which is mounted integrally to the engine flywheel housing and has a remote output in line with the transmission input. The transfer of the drive with proper rotation is accomplished with four straight involute spur gears. An auxiliary accessory drive is obtained with an offset set of three spur gears. The gearing is housed in a simple two-piece housing.

During preliminary study of this design, another concept utilizing a chain drive in lieu of gears was evaluated. It was initially considered because of the possibility of reducing weight, and providing greater simplicity and quieter operation at less cost. However, further study disclosed that, in order to obtain proper rotation, a set of gears was also required, and the auxiliary accessory drive was obtained with greater difficulty and complexity. Also, it was considered that the chain drive may require, relatively, a more extensive amount of testing and development time.

The all-gear drive is considered to be most straightforward and simple in construction and, hence, easiest to assemble and maintain, most competitive in cost, and offers greatest possibility of first-time success with minimum testing and development time.

The design of this unit, mounted directly to the engine and with a remote (free) output, provides ease of installation with freedom of alignment with the transmission input. Should the output be also mounted integrally, rigid and inflexible, to the transmission input, considerable difficulty in alignment might be encountered at initial or replacement installation. Also, relatively high stresses could be induced in the housings due to improper alignment of flanges at installation or loads imposed during operation.

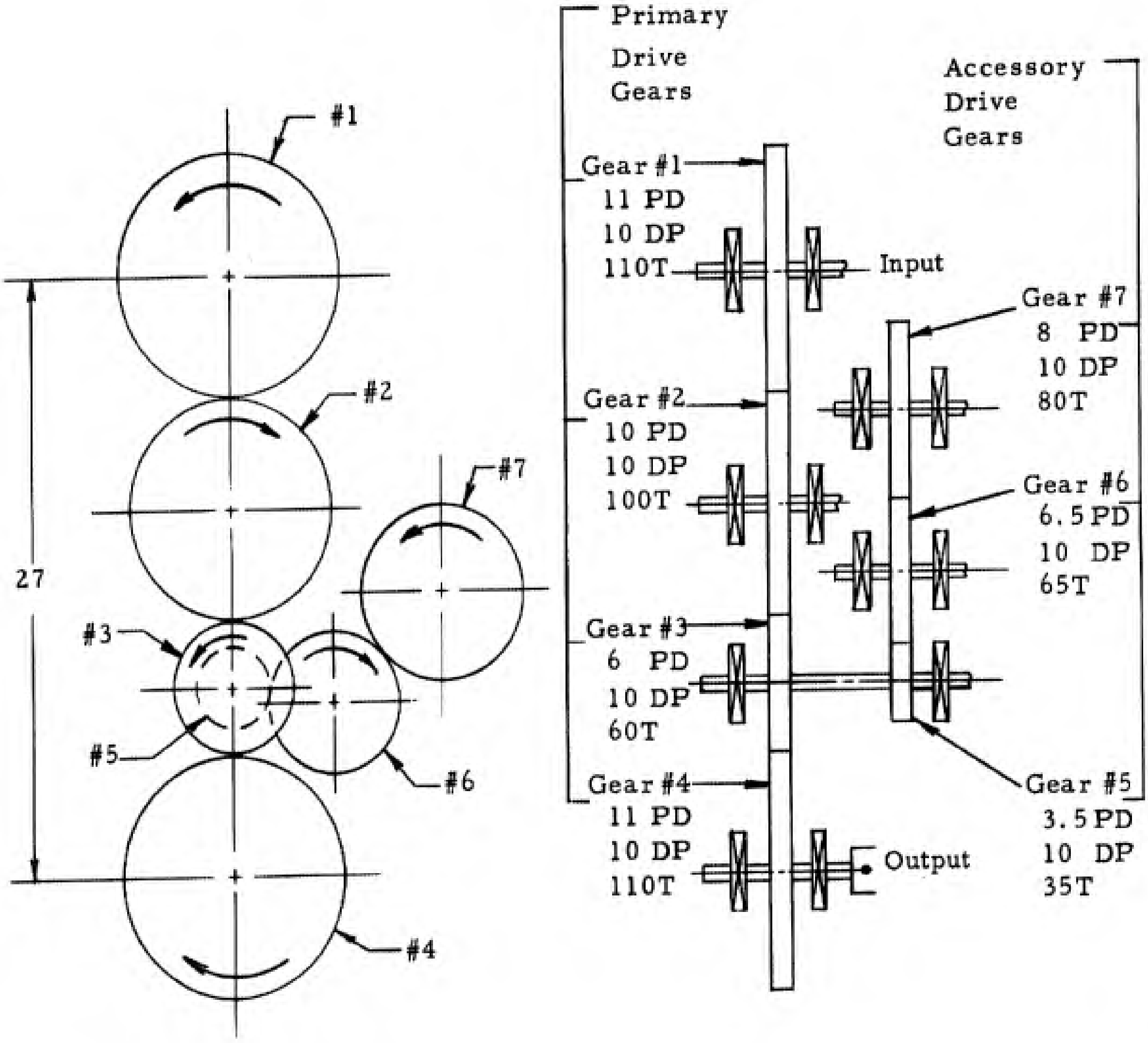
The design has been developed to provide space at the input for a torsional damper mounted to the engine flywheel. An auxiliary accessory drive section provides mounting pads for three accessories; specifically, a drive for the vehicle main hydraulic pump, the lube pump for the transfer gear, and an additional pad suitable for a small hydraulic boost pump are provided.

Positive lubrication is accomplished by means of an oil spray system which assures adequate lubrication and cooling. The lube pump is mounted on and driven by the gearbox. A remote located reservoir provides an oil sump and serves effectively to augment cooling.

A 1:1 speed ratio in the main drive gearing is chosen for optimum matching of the prime choice of powerplant, the Cummins V8-315 engine which is rated at 3,000 rpm, to the Allison XTG-250 transmission, which is also rated at 3,000 rpm maximum input speed. The main hydraulic pump is rated at 2,400 rpm maximum and, thus, a 1.25:1 reduction is provided in the accessory drive section.

Because of the distinct possibility of future utilization of an advance component powerplant providing increased power, the transfer gear is designed to transmit a nominal 350 shaft horsepower with adequate design margins. Full consideration of high strength-to-weight ratios during detail design will provide the higher rating without weight penalty for use with the present powerplant.

TRANSFER GEARBOX



View Looking at Engine Flywheel

Figure. E. Transfer Gearbox Schematic - Preliminary

TRANSFER GEARBOX

Design Data

Preliminary Specifications

Rating

Maximum Input Power - - - - - 350 HP

Maximum Input Torque - - - - 660 Lb-Ft

Maximum Input Speed - - - - - 3,000 RPM

Over-all Gear Ratio - - - - - - - - - - 1:1

Center Distance (Input to Output) - - 27 inches

Mounting:

Flange to mate with SAE No. 2 engine flywheel housing

Accessory Drive Provision

Three pump-mounting pads as follows:

Pad 1 - - SAE type "C" 4-bolt mounting pad

Rated at: 70 HP at 2,400 RPM

Total Reduction (Input to pad) 1.245:1

Pad 2 - - SAE type "A" 2-bolt mounting pad

Rated at: 10 HP at 2,400 RPM

Total Reduction (Input to pad) 1.245:1

TRANSFER GEARBOX

Preliminary Specifications (Cont'd)

Pad 3 - - SAE type "A" 2-bolt mounting pad

Rated at:	5 HP at 2,960 RPM
Total Reduction (Input to pad)	1.012:1

Lube System

Type: Pressurized spray system with remote cooling reservoir.

Flow: 3.6 gpm at 3,000 rpm Engine Speed

Spray System: The main drive gears are lubricated by spray nozzles

Reservoir:

Remotely mounted

Capacity - 8 gallons

Design Temperatures:

For selection of oil flow for cooling -- Maximum Continuous 200°F

For selection of seals and gaskets -- Maximum Intermittent 300°F

Design Data

Design Operating Conditions

Design of the transfer gearbox is based on a minimum life of 1000 hours

TRANSFER GEARBOX

Design Operating Conditions (Cont'd)

between overhauls. In determining design conditions for the transfer gearbox a distinction is made between primary driving components and accessory drive components. The primary driving components constitute gears, shafts and bearings which transmit engine power to the transmission: the accessory drive components include all gears, shafts, and bearings delivering power to the accessory pads.

Primary Driving Components Conditions

The total 1000-hour life rating is broken up into three conditions, since the power transmitted by the gearbox is of a variable nature. These conditions are a conservative prediction of probable engine performance during the entire 1000-hour life.

Condition 1. Maximum engine torque for 20% total life, or 200 hours.

Condition 2. Average full-throttle load for 50% total life, or 500 hours.

Condition 3. Maximum engine power for 30% total life, or 300 hours.

Condition No's. 1 and 3 are self-explanatory.

TRANSFER GEARBOX

Primary Driving Components Conditions (Cont'd)

Condition No 2 calls for "average-load." The average-load is assumed to be the full-throttle power point at an engine speed 2/3 between the speeds corresponding to the maximum torque point and the maximum horsepower point. The "average" speed, N_A is calculated as follows:

$$N_A = N_T + 2/3\ (N_M - N_T),$$

where N_M is the engine governed speed, maximum power point, and N_T is engine speed at maximum engine torque output. The above equation is then simplified to the following:

$$N_A = \frac{2\ N_M + N_T}{3}$$

Specific engine data for conditions described above are based on the Cummins VINE-315 "gross" characteristics at 80°F. These characteristics are described by the torque and horsepower curves shown in Figure A. The following table shows the engine output data to be used for the operating conditions.

Table C. Engine Output Data for Operating Condition

Cond.	Engine Speed RPM	Engine Output Horsepower	Engine Output Torque, Lb-Ft
1	2150(N_T)	242	596
2	2700(N_A)	291	571
3	3000(N_M)	309	543

N_A is calculated as follows:

$$N_A = \frac{2(3000) + 2100}{3} = 2700 \text{ RPM}$$

Accessory Drive Components Conditions

As described in the previous section, there are three SAE standard pump-mounting pads which must receive power through accessory-drive gearing. Operating conditions for the design of these components are established on the basis of maximum accessory power which may be required by "special vehicles," rather than the basic vehicle, to eliminate the need for later redesign. The conditions for each mounting pad are described below:

Pad 1:

This pad is used for the main hydraulic pump and is designed to deliver 70 horsepower at a speed of 2400 rpm for 1000 hours.

TRANSFER GEARBOX

Accessory Drive Components Conditions (Cont'd)

Pad 2:

This pad is provided for the mounting of a hydraulic-system supercharge pump (not required on the basic vehicle). The pad must deliver 10 horsepower at 2400 rpm for 1000 hours.

Pad 3:

This pad is used for mounting the transfer gearbox lube pump. It must deliver 5 horsepower at 2960 rpm for 1000 hours.

In each of the three cases described above, the selected conditions are based on the conservative assumption that each pump will operate at its maximum power requirement during the entire 1000-hour design life.

Design Torques and Speeds

Design torques and speeds, corresponding to the conditions for the primary driving components, are derived from engine output data. Accessory drive torques and speeds are based on the conditions describing the accessory power requirements. For a review of the conditions for both cases, see sheet 7, Design Operating Conditions. Calculations of torques in all cases neglect power losses in gears and bearings. Also, accessory loads are neglected in calculation of primary component torques.

TRANSFER GEARBOX

Primary Drive Gears

The primary drive gears include gears numbered one through four. The torques and speeds on gear Number 1 (the input gear) are identical to the engine output data. Torques and speeds on gear Number 2 are obtained by the following equations:

Torque

$$Q_2 = \frac{Q_1 \times D_{P_2}}{D_{P_1}}$$

Speed

$$N_2 = \frac{N_1 \times D_{P_1}}{D_{P_2}}$$

Where Q is torque, Dp is pitch diameter, and N is speed in rpm. The subscripts "1" and "2" refer to gears "1" and "2".

Torques and speeds for gear Number 3 is found in the same manner. Since the over-all gear ratio, between gears 1 and 4 is 1:1 the torques and speeds for gear Number 4 are the same as those for gear Number 1. Data for the primary drive gears is shown in the following table.

NObs 4561 LVTPX11
Power Train Design Analysis

TRANSFER GEARBOX

Primary Drive Gears (Cont'd)

Table D. Primary Drive Design Torques and Speeds

		Cond. 1		Cond. 2		Cond. 3	
Gear No.	Pitch Diameter inches	Speed RPM	Torque Lb-Ft	Speed RPM	Torque Lb-Ft	Speed RPM	Torque Lb-Ft
1	11	2100	597	2700	571	3000	543
2	10	2310	542.5	2970	518	3300	494
3	6	3850	326	4950	311	5500	296
4	11	2100	597	2700	571	3000	543

Accessory Drive Gears

The accessory drive gears (5, 6, and 7) are in a single gear train and transmit accessory power to the shafts of gears 6 and 7.

The entire train is driven by gear Number 5 which is on the same shaft as gear Number 3. Therefore, both gears must rotate at the same speed. Gear Number 5 transmits the entire accessory power, 70 + 10 + 5 = 85 horsepower. Its maximum speed is 5500 rpm and its maximum torque is:

$$Q_5 = \frac{5252 \times HP_5}{N_5} = \frac{5252 \times 85}{5500} = 81.2 \text{ Lb-Ft}$$

TRANSFER GEARBOX

Accessory Drive Gears (Cont'd)

Gear Number 6 handles the entire accessory load of 85 horsepower; torque and speed are calculated as follows:

$$Q_6 = \frac{Q_5 \times D_{P_6}}{D_{P_5}} = \frac{81.2 \times 6.5}{3.5} = 151 \text{ lb-ft}$$

$$N_6 = \frac{N_5 \times D_{P_5}}{D_{P_6}} = \frac{5500 \times 3.5}{6.5} = 2965 \text{ rpm}$$

Gear Number 7 delivers a total of 80 horsepower to accessory pads Numbers 1 and 2. Its design speed and torque are calculated as follows:

$$N_7 = \frac{N_6 \times D_{P_6}}{D_{P_7}} = \frac{2965 \times 6.5}{8} = 2410 \text{ rpm}$$

$$Q_7 = \frac{5252 \times HP_7}{N_7} = \frac{5252 \times 80}{2410} = 174.5 \text{ lb-ft}$$

Torques and speeds for gears 5, 6, and 7 are shown in the following table.

Table E. Accessory Drive Design Torques and Speeds

Gear No.	Pitch Diameter inches	Speed RPM	Torque Lb-Ft
5	3.5	5500	81.2
6	6.5	2965	151
7	8	2410	174.5

TRANSMISSION ENGINEERING DEPARTMENT

ALLISON DIVISION

GENERAL MOTORS CORPORATION

TRANSMISSION

XTG-250 POWER TRAIN PRELIMINARY NOTES

A. Description

1. The XTG-250 is a manually-controlled, full torque shifting, cross-drive power train which includes a hydraulic torque converter with lockup clutch in combination with a four speed forward and one reverse planetary gear set incorporating a steer system and full vehicle brakes. The power train controls include a steering wheel, or T-bar, single-pedal vehicle brakes, a steer selection mechanism, and a range selection mechanism.

2. The flow of power through the XTG-250 is from the engine to the torque converter to the basic planetary gear set, and into a bevel gear set. A cross shaft from the bevel gear set drives into the reverse planetary, the output steer planetary, and the brake package at each side of the transmission.

3. Control of the automatic converter lockup clutch, available in all ranges, is provided by a converter turbine driven governor and is throttle position modulated to permit lockup operation at lower speeds at part throttle for additional economy.

   Shifts to third, second, first, and reverse ranges are inhibited at speeds which would be dangerous to vehicle and crew.

4. Under normal operation of the XTG-250, the geared steer planetary sets at each output permit controlling the output ratio as desired between 1:1 and a maximum reduction of 1.475:1 during steer when operating in fourth, third, and second ranges. In first and reverse ranges, steering is by the clutch brake system with each brake being hydraulically applied or by pivot steer, one output in reverse with the other remaining in forward.

   The service and parking brakes are integral with the steer system and are mechanically operated, oil cooled and lubricated, multiple-disc-type brakes.

B. Tabulated Data

Manufacturer	Allison Division, G.M.C.
Type	Cross Drive
Model	XTG-250
Rating:	
Maximum Input Torque	520 lb-ft
Maximum Input Speed	2800 rpm
Maximum Input Horsepower	250
Weight, Dry	1350 lbs (Initial Pilots)
Length	46.00 in.
Height	28.80 in.
Width	30.65 in.
Mounting (Saddles on Both Output Housings)	2 Points
Drive Ranges	Fourth, Third, Second, First, and Reverse
Power Take-Off	Turbine Driven
Rating (Continuous Operation)	470 lb-ft
Shift and Steering Control:	
External	Mechanical
Internal	Hydraulic and Mechanical
Clutches:	
Engaged by	Oil Pressure
Released by	Spring Pressure
Brakes:	
Quantity	2
Type	Wet, Multiple Disc
Application	Mechanical
Cooling	Oil
Oil Pumps:	
Quantity	3
Type	Spur Gear
Oil Capacity (Excluding Oil Coolers):	
Initial Fill	14 Gal.
Refill	11 Gal.
Converter Multiplication at Stall	2.55:1

Overall Transmission Ratios:

Range	Normal Steer	Pivot Steer
First	19.841:1 Clutch Brake	19.841:1 Pivot
Second	13.423:1 Geared	19.841:1 Pivot
Third	7.210:1 Geared	7.210:1 Geared
Fourth	3.210:1 Geared	3.210:1 Geared
Reverse	15.096:1 Clutch Brake	15.096:1 Pivot

Total Torque Ratio Coverage:

First Gear Stall	2.55 x 19.841 = 50.60:1
Fourth Gear Lockup	1.00 x 3.210 = 3.21:1

C. Operation

1. Power Shifting. Full power shifting can be made in all forward ranges.

2. Starting Procedure. The starting procedure with respect to the power train is:

   a. Check to see if oil level is at or above the "Safe to Start Engine" mark.

   b. Set brakes.

   c. Place shift control lever in neutral.

   d. Start engine and idle at 1300 rpm.

   NOTE: The lube low pressure warning light should go out within 30 seconds after starting with engine running at 1300 rpm.

3. Tow Start. With power train in neutral range, increase towing speed to 3 to 5 mph. Shift power train into first range.

4. Power Take-Off Engagement. With power train in gear, stop vehicle and engage power take-off. Shift to neutral for power take-off operation with a stationary vehicle.

D. Operating Precautions

1. Do not shift into reverse until vehicle is stopped and engine at idle.

2. Do not attempt to operate power train if oil temperature warning light (set for 300°F.) comes on. Put transmission in neutral and run engine at 1000-1500 rpm to cool oil.

3. If downshifting is attempted at excessive speeds and the shift is prevented by the shift inhibitors, do not continuously force the shift control lever since this may prevent downshifting regardless of speed.

4. Do not attempt to engage power take-off with vehicle moving or with power train in neutral range.

5. Do not start engine if power train oil level is below "Safe to Start Engine" mark.

E. Maintenance

1. Oil Level Check. The oil level check procedure is:

Before Starting Engine

a. Check oil level.
b. Add oil to the "Safe to Start Engine" mark if required.

Before Vehicle Operation

a. Run engine to warm oil 3 to 8 minutes 1200 to 1500 rpm, 4th gear, brakes applied.
b. Check oil level 1200 to 1500 rpm, neutral range, brakes applied.
c. If the measured oil level is below the "Add" mark, add oil to bring the level to the "Add" mark.

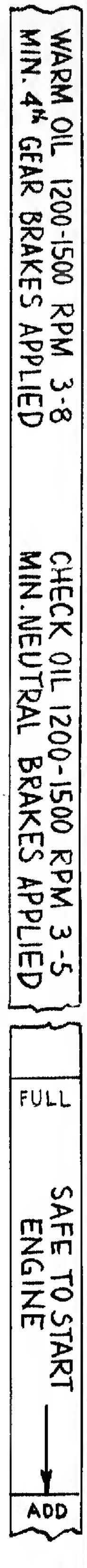
Following Vehicle Operation

a. Run engine 3 to 5 minutes 1200 to 1500 rpm, neutral range, brakes applied.
b. Check oil level 1200 to 1500 rpm, neutral range, brakes applied.
c. Add oil to a level not to exceed the "Full" mark. It is not necessary to add oil if the measured level is above the "Add" mark.
d. Drain excess oil to the "Full" mark if overfull.

Precautions

a. Following initial oil fill or oil change, recheck the "Safe to Start" level after 2 to 3 minutes of running 1200 to 1500 rpm, neutral range (additional oil may be required to supply the converter and the oil cooler circuit).
b. Do not overfill.

2. Oil Change. Change oil every 2000 miles or six months, whichever occurs first. Use MIL-L-2104A, Grade 10 oil for ambient



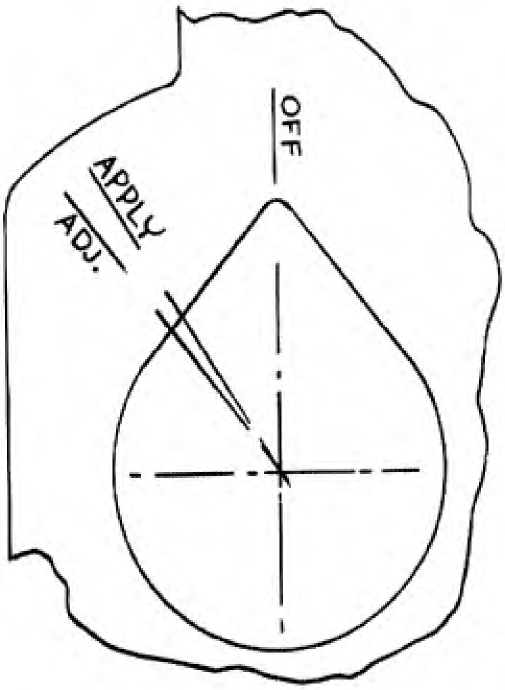
temperatures of from 125°F. to -10°F. and use MIL-O-10295 Arctic oil from -10°F. to -65°F.

Check oil screen at every oil change and clean if necessary.

3. Brake Adjustment

a. The right and left brakes are adjusted separately. The linkage should be disconnected from both during adjustment. Each brake should be adjusted so that, with 100 ft-lb of apply torque, the pointer on the brake indicator aligns with the "Apply" mark on the power train top cover.

b. To adjust the brakes, remove the brake adjustment covers. Loosen the locknut and turn the adjusting screw. To decrease the brake travel (tighten), turn the screw counter-clockwise (viewed from the rear of the power train). To increase the brake travel (loosen), turn the adjusting screw clockwise.



c. The brakes should be readjusted when the brake indicator aligns with the "Adjust" mark. The travel should never be less than the distance between the "Off" and "Apply" marks. Neither should the travel ever be greater than the distance between the "Off" and "Adjust" marks.

d. When the adjustment is completed, tighten the locknut to secure the adjusting screw. When the brakes are adjusted as outlined above, the correct running clearance is maintained between the brake plates.

4. NOTE: Many components of this power train are made of magnesium. Due to this fact, the following precautions should be taken:

a. When mounting dissimilar metals to the magnesium components, make sure that the parts being mounted are cadmium-plated.

b. Scratches in the magnesium components should be coated with dichromate (Dow No. 1) and repainted (per 8351940 - primer per MIL-C-15328, gloss white acti-thane paint (urethane) approved source: Saran Protective Finishes Company).

F. Vehicle Functional Check

1. Main Pressure at Stall

190 to 220 (at 1500 rpm or above)

2. Lockup Operation

Main Pressure 180 ± 10

	Turbine Speed	Gov. Press. PSI
Engagement (Full TV)	2150 ± 50	50
Disengagement (Full TV)	1850 ± 50	38
Engagement (No TV)	1350 ± 50	19
Disengagement (No TV)	1050 ± 50	12

3. Pressure Under Maximum Steer

Normal Operation

4th, 3rd, and 2nd Ranges

Geared Steer Clutch Pressures 110±10 at 1500 rpm Input Speed

1st and Reverse Ranges

Brake Pressure 110±10 at 1500 rpm Input Speed

Pivot Operation

1st and 2nd position

Geared Steer Clutch Pressure 180±10 at 1500 rpm Input Speed Reverse Clutch Pressure 110±10 at 1500 rpm Input Speed (opposite applied geared steer clutch)

Reverse Range

Geared Steer Clutch Pressure 110±10 at 1500 rpm Input Speed (opposite applied reverse clutch)
Reverse Clutch Pressure 180±10 at 1500 rpm Input Speed

4. Lube Pressure

In Neutral at 1500 rpm	15 psi minimum
In Neutral at 1300 rpm	11 + 2 psi warning light should be out

5. Stall Speed

1900 to 2100 rpm (6V-53T Engine using No. 2 Diesel Fuel)

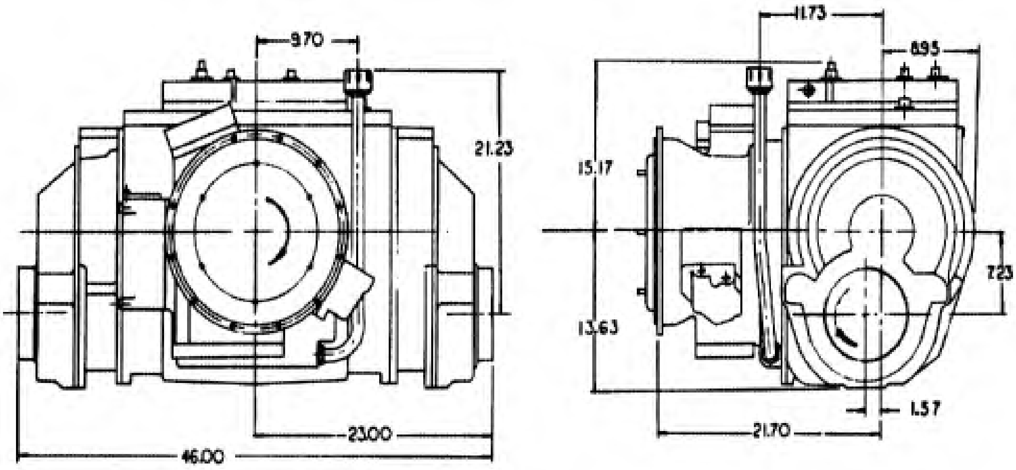
PRELIMINARY XTG-250 PRESSURE SCHEDULE

Functional Check (Free Spin)

Range	Neut.	4th	3rd	2nd	1st	Rev.
Input Speed	1500	1500	1500	1500	1500	1500
Main Pressure	180±10	180±10	180±10	180±10	180±10	180±10
Conv. In Pressure	60±10	60±10	60±10	60±10	60±10	60±10
Conv. Out Pressure	50±10	50±10	50±10	50±10	50±10	50±10
Lube Pressure	25±10	25±10	25±10	25±10	25±10	25±10
Governor Pressure	23± 3	23± 3	23± 3	23± 3	23± 3	23± 3
Lockup Cl. Pressure	0	0	0	0	0	0
High Cl. Pressure	0	180±10	0	0	0	0
Int. Cl. Pressure	0	0	180±10	0	0	180±10
Low Clutch Pressure	0	0	0	180±10	180±10	0
L.H. Reverse Clutch	0	0	0	0	0	180±10
L.H. Ouput Clutch	180±10	180±10	180±10	180±10	0	0
L.H. Geared Steer	0	0	0	0	180±10	0
R.H. Reverse Clutch	0	0	0	0	0	180±10
R.H. Output Clutch	180±10	180±10	180±10	180±10	0	0
R.H. Geared Steer	0	0	0	0	180±10	0
Sump Temp. °F.	185±15	185±15	185±15	185±15	185±15	185±15
Cooler Flow GPM	14± 2	14± 2	14± 2	14± 2	14± 2	14± 2

PRELIMINARY CHARACTERISTIC SHEET

Component: Power Train, X-Drive, Model XTG-250-1A Hydraulic Torque Converter, Planetary Gear Type, All Torque Shifting



The XTG-250-1A includes a hydraulic torque converter with a lockup clutch. The planetary range gearing in combination with the steer and output planetary sets provide four forward ranges and one reverse range. The transmission also incorporates geared steer, pivot steer, 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-250-1A
Power Train Installation & Assembly Part No. -----------------DAX-3469
Drive Ranges--------Fourth, Third, Second, First, Neutral, and Reverse
Drive Range, Steering, and Shift Control (External) ----------Mechanical
Shift & Steering Mechanism (Internal Control) ----Hydraulic & Mechanical
Steering Type ---Geared Steer
Pivot Steer
Power Take-off ---Turbine Driven
Rating (Continuous Operation) ------------------------------470 Lb.Ft.
Type of Clutches (All Ranges)-------Multiple Plate, Engaged by Oil Pressure
Brake----Multiple Wet Plate, Service and Parking, Mechanical Application
Hydraulic Torque Converter--Single Stage, Multiple Phase w/Lockup Clutch
Maximum Converter Multiplication --------------------------------2.55:1

XTG-250-1A

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

Bevel Gear Ratio --1.44:1
Output Steer Planetary Ratios:
  Under Drive --1.475:1
  Direct ---1.000:1
Final Drives (Integral) Ratio -----------------------------------2.220:1
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.) 14
Transmission Dry Weight, Lbs. -------------------------------(Est.)1300

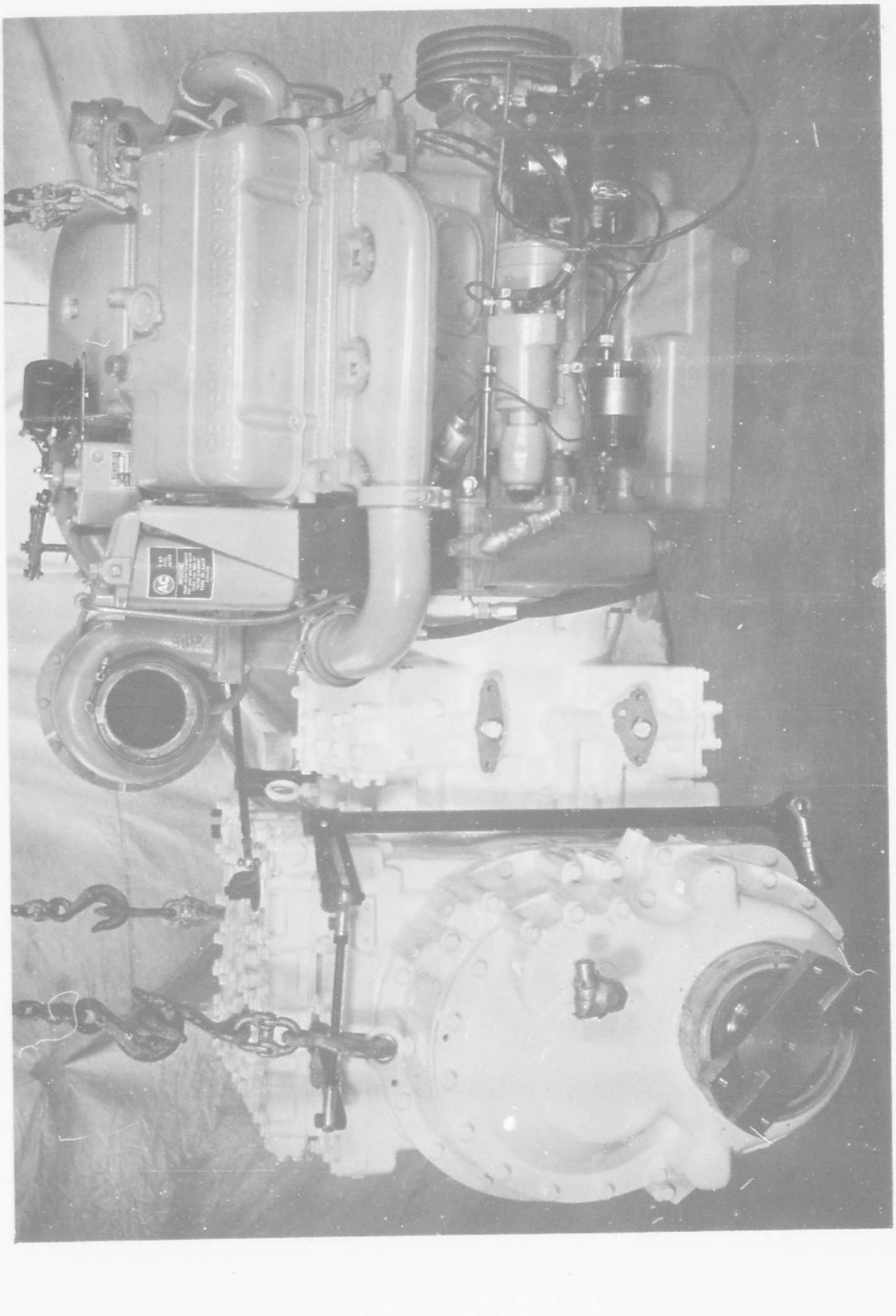


Figure F. Allison XTG-250 Transmission (Side View)

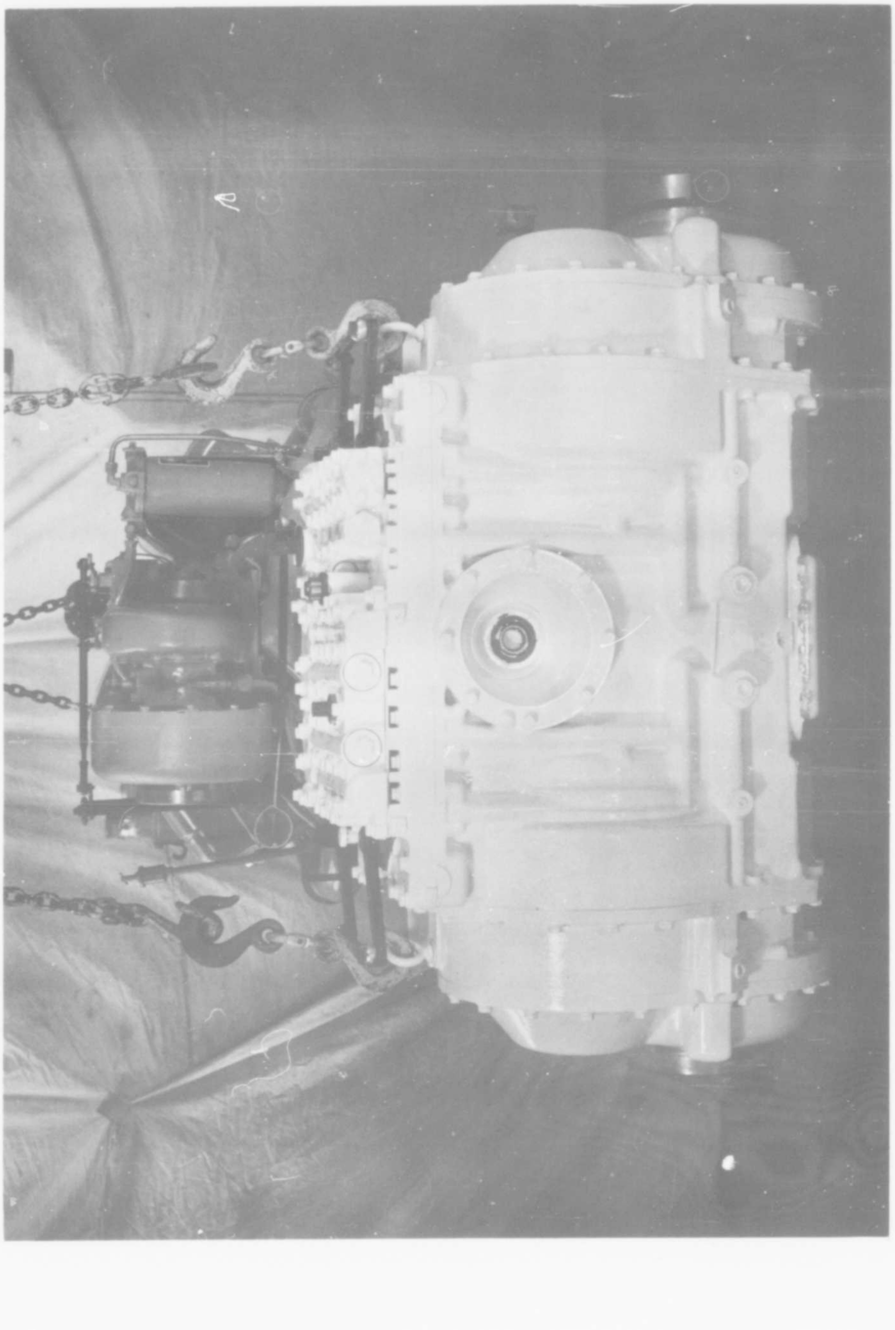


Figure G. Allison XTG-250 Transmission (Rear View)

Range Section

High Clutch	1.00 :1
Intermediate Clutch	2.24 :1
Low Clutch	4.18 :1

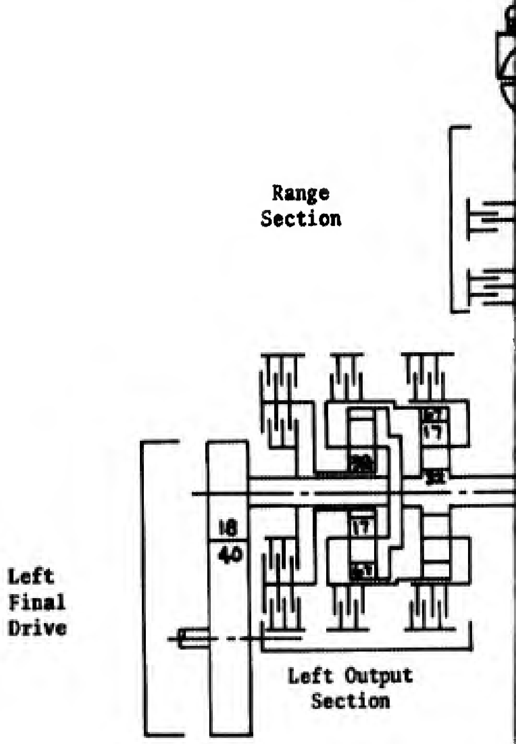
Output Section

Reverse	2.095:1
Geared Steer Clutch	1.478:1
Output Clutch	1.00 :1

Bevel Gear Ratio	1.444:1
Final Drive Ratio	2.222:1

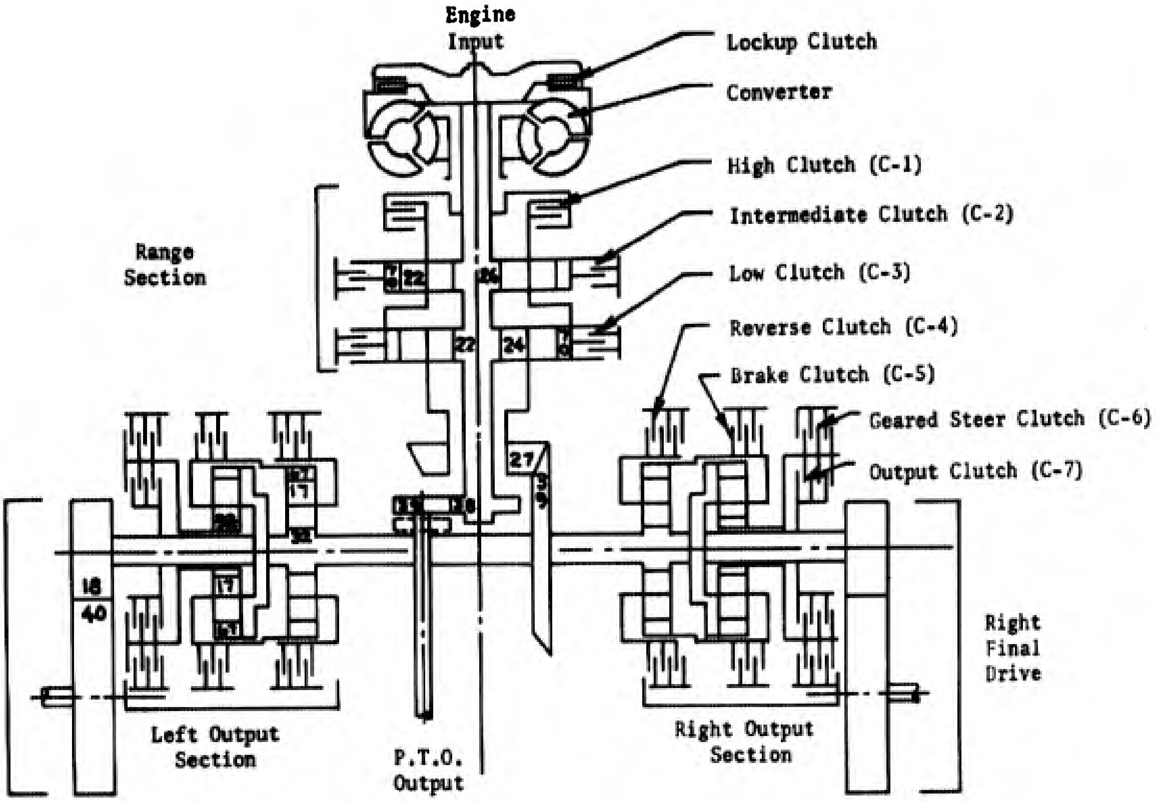
Transmission Gear Ratios

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



			STEERING				
			Normal Operation				
				Clutches Engaged			
Selector Position	Range	Normal Forward or Reverse Operation	Type	Outer Track	Inner Track	(Outer Track) Overall Ratio	Type
R_2	Reverse_2	C-1, C-4	C.B.	C-4	C-5	6.720:1	Pivot
R_1	Reverse_1	C-2, C-4	C.B.	C-4	C-5	15.096:1	Pivot
N	Neutral	-	-	-	-	-	-
1	First	C-3, C-6	C.B.	C-6	C-5	19.841:1	Pivot
2	Second	C-3, C-7	G.S.	C-7	C-6	13.423:1	Pivot
3	Third	C-2, C-7	G.S.	C-7	C-6	7.210:1	G.S.
4	Fourth	CC-1, C-7	G.S.	C-7	C-6	3.210:1	G.S.

Figure H. XTG-250 Power Train Schematic



STEERING							
Normal Operation				Pivot Operation			
	Clutches Engaged		(Outer Track)		Clutches Engaged		(Outer Track)
Type	Outer Track	Inner Track	Overall Ratio	Type	Outer Track	Inner Track	Overall Ratio
C.B.	C-4	C-5	6.720:1	Pivot	C-4	C-6	6.720:1
C.B.	C-4	C-5	15.096:1	Pivot	C-4	C-6	15.096:1
-	-	-	-	-	-	-	-
C.B.	C-6	C-5	19.841:1	Pivot	C-6	C-4	19.841:1
G.S.	C-7	C-6	13.423:1	Pivot	C-6	C-4	19.841:1
G.S.	C-7	C-6	7.210:1	G.S.	C-7	C-6	7.210:1
G.S.	C-7	C-6	3.210:1	G.S.	C-7	C-6	3.210:1

ure H. XTG-250 Power Train Schematic

2

FINAL DRIVE

Introduction

Design Approach

The final drive in a conventional vehicle design has only one function; to deliver power to the sprocket. For the LVTPX11 a new final drive concept is utilized. The final drive for the LVTPX11 has a dual function; it delivers power to the sprocket and serves as track slack compensator.

The final drive assembly is designed as a separate, hull-mounted, drive-sprocket support, incorporating reduction gearing providing a 3.3:1 reduction ratio. It is designed to pivot in the hull mounting with an offset which permits changing of the sprocket center location to adjust track tension. The final drive reduction is designed to match the vehicle requirements to the components selected for the remainder of the power train.

Several final drive gearing arrangements and mountings were investigated in order to achieve simplicity, low cost and ruggedness and at the same time to meet the dual function of the final drive with an optimum design.

The following gear arrangements were considered:

1. A combination of a set of external spur gears and a gearset consisting of a pinion meshing with an internal gear.

FINAL DRIVE

Design Approach (Cont'd)

2. A combination of a set of external gears and an output planetary.
3. A single set of external spur gears.

The single gearset was selected for the final design, because it is the least costly and the simplest arrangement.

The various hull mountings which were investigated are shown on figure J. The arrangement c was selected because it allows complete freedom in the selection of mounting-bearing size and requires only a small hull pocket.

Design Data

Specifications

Rating

Maximum Input Power	240 hp at 2100 rpm
Maximum Input Torque	4750 lb-ft
Maximum Input Speed	2100 rpm

Lubrication System

Wet sump, splash-lubricated gearing

Externally-greased output bearing

CUSHING & CO

DS-17

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

Sheet 50 of 78

BY D. Percy DATE 10/15/62 SUBJECT LVTPX 11 SHEET NO. OF
CHKD. BY ✓ DATE PHASE I REPORT JOB NO. NObs 4561
DATA

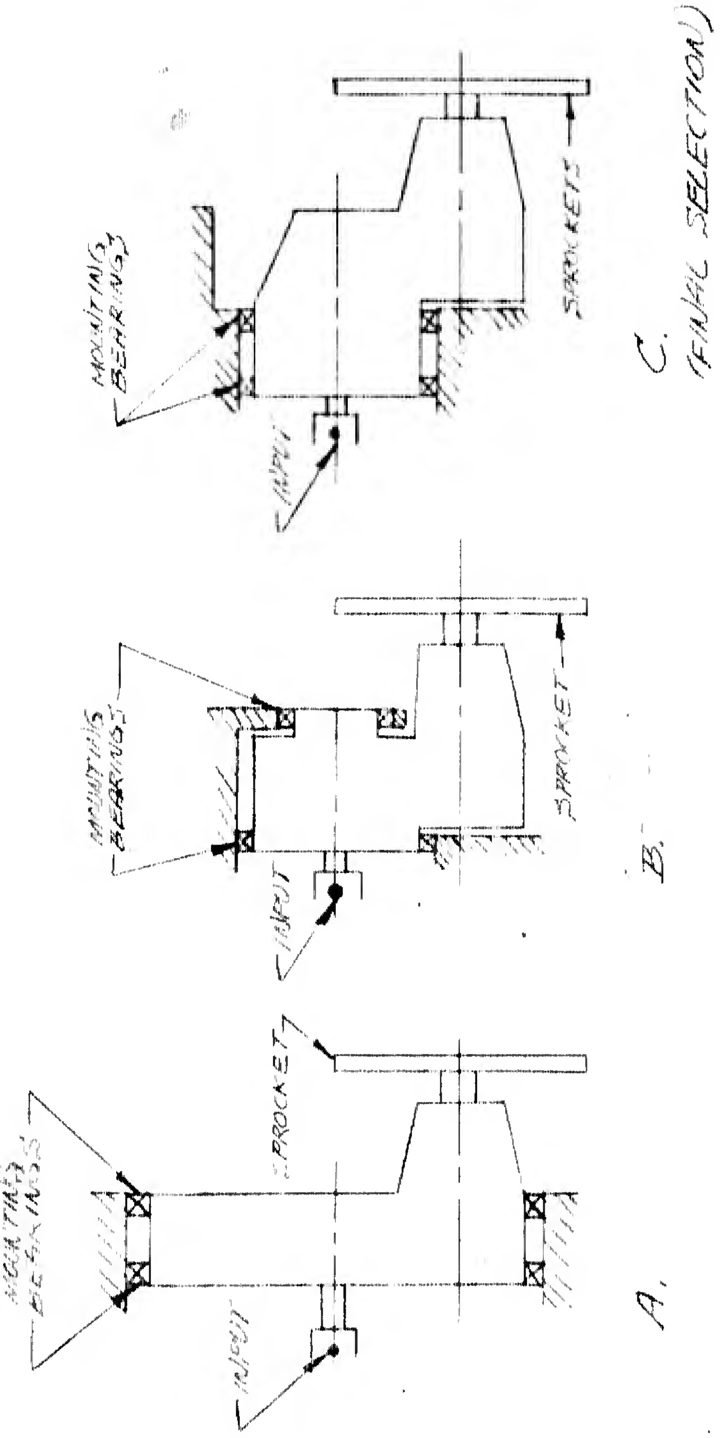


FIG. J MOUNTING ARRANGEMENTS CONSIDERED IN FINAL DRIVE CONCEPT STUDY

FINAL DRIVE

Design Temperatures

Maximum continuous for cooling calculations	200°F
Maximum intermittent for selection of gaskets and seals	300°F

Special Design Feature

The final drive delivering power to the sprocket serves also as track compensator.

Design Operating Conditions

The final drive design operating conditions are divided into three groups; Gearing Design Conditions, Compensator Design Conditions, and Mounting Design Condition. The gearing design conditions are used for design of components which transmit power for vehicular locomotion. The compensator design conditions are used for analysis of track compensator forces. The mounting design condition is used for design of the final drive housing and its mounting rings. They are also used for establishing bearing loads and shaft loads for the final drive output shaft.

Gearing Design Conditions

Design conditions for gearing and related components are based on a total minimum life of 1,000 hours between overhauls. Three conditions are

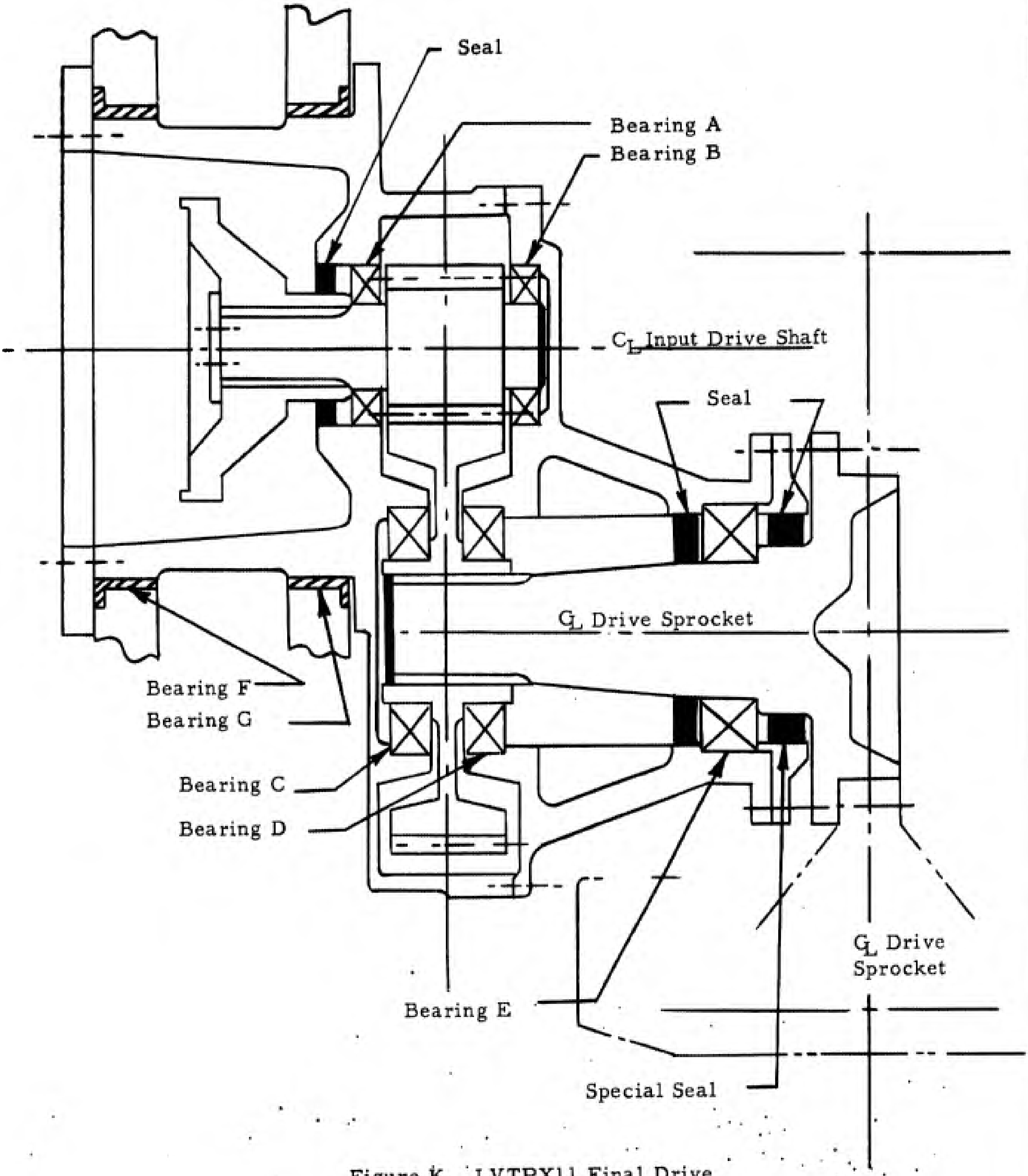


Figure K. LVTPX11 Final Drive

FINAL DRIVE

Gearing Design Conditions (Cont'd)

selected as a conservative estimate of vehicle performance. Each of these conditions is based on gross vehicle weight and track requirements. The conditions follow:

1. Steering for 50 hours at 5 mph, with a tractive effort to weight ratio of 1 (TE/W = 1). Under this condition half the vehicle weight bears on one track.
2. Driving at 14 mph for 600 hours on a 20 percent grade with 140 pounds per ton unit rolling resistance.
3. Driving at 25 mph for 350 hours on an 8 percent grade with 140 pounds per ton unit rolling resistance.

Compensator Design Conditions

The following four conditions have been established for the purpose of investigating the various load conditions occuring when the final drive acts as a track slack compensator. In all the conditions the ground coefficient is assumed to be 1.0 with half the gross vehicle weight on each track.

Compensator Design Conditions:

1. Forward operation

FINAL DRIVE

Compensator Design Conditions (Cont'd)

2. Pivot steering during forward motion
3. Reverse operation
4. Pivot steering during reverse motion

Mounting Design Condition

For the mounting design calculations a shock condition is used. It is assumed that a maximum load of three G's can be encountered on the sprocket when the vehicle dips over an embankment and the full vehicle weight is absorbed at the sprockets. One half of this load is taken by each sprocket and is assumed to act vertically through the center of the sprocket.

Design Loads and Speeds

Gearing Design Torques and Speeds

Final drive gearing design torques and speeds for the conditions listed above are calculated and shown in table F. Output torques and speeds are based on the use of a drive sprocket of 21.3-inch pitch diameter. Input torques and speeds are direct functions of the output data and the 3.3:1 gear ratio.

FINAL DRIVE

Gearing Design Torques and Speeds (Cont'd)

Table F Final Drive Design Torques and Speeds

Cond.	Input Torque Lb-Ft	Input Speed RPM	Output Torque Lb-Ft.	Output Speed RPM
1	4,720	263	15,550	79.7
2	1,255	739	4,135	224
3	707	1,315	2,330	398

Compensator Design Loads

Figure L shows a force diagram of the final drive as track compensator.

Symbols used in the figure are described below:

Q_{IN} = Final Drive Input Torque in Lb-In.

F_U = Upper Track Force in Lb.

F_L = Lower Track Force in Lb.

F_C = Compensating Cylinder Force in Lb.

Using the center of the input shaft as reference point, the following general equation can be written:

$$\Sigma M = 0$$

$$Q_{IN} + 18.375 F_L + 2.625 F_U + 9.75 F_C = 0$$

BY D. Percy DATE 11-7-62 SUBJECT LVT PX11
CHKD. BY DATE FINAL DRIVE
DESIGN ANALYSIS

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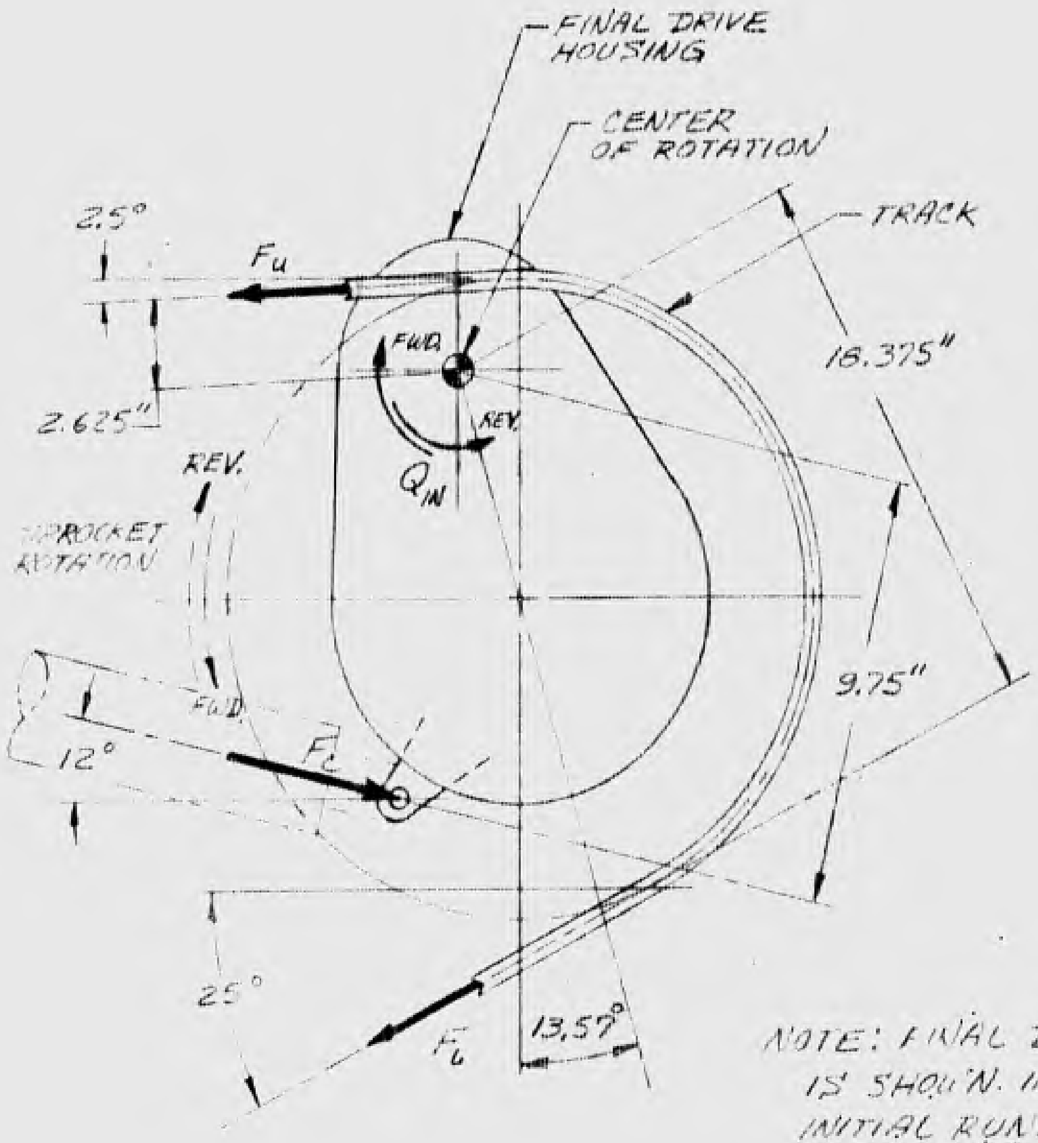

NOTE: FINAL DRIVE IS SHOWN IN INITIAL RUNNING POSITION (NO TRACK WEAR)

FIG. L SCHEMATIC – FINAL DRIVE MOMENT SYSTEM FOR TRACK COMPENSATION

FINAL DRIVE

Compensator Design Loads (Cont'd)

In addition, the relationship between input torque and track force can be expressed as follows:

$$Q_{IN} = \frac{D}{2 \times M} \quad F = \frac{21.3}{2 \times 3.3} \quad F = 3.54\, F$$

where

D = Sprocket Pitch Dia. = 21.3 inches

M = Final Drive Reduction = 3.3:1 ratio

F = Track Force in pounds

The track force F is equal to F_L during forward driving and is equal to F_U during reverse driving.

These two equations are used to determine the forces F_L, F_U and F_C for the design conditions stated previously. In these calculations, clockwise moments were used with a positive sign, counterclockwise moments with a negative sign. The results are listed below.

FINAL DRIVE

Compensator Design Loads (Cont'd)

Table G Compensator Preliminary Design Loads

Cond. No.	Q_{IN} Lb-In.	F_C Pounds	F_L Pounds	F_U Pounds	Remarks
1	-61,970	39,350	17,500	0	Both Tracks
2	+61,970	0	5,875	17,500	Inboard Track
	-61,970	39,350	17,500	0	Outboard Track
3	+61,970	0	5,875	17,500	Both Tracks
4	-61,970	39,350	17,500	0	Inboard Track
	+61,970	0	5,875	17,500	Outboard Track

In table G the "inboard track" is defined as the track on the vehicle side corresponding to the direction of steer. All loads are preliminary and based on the assumption that initial track tension is zero. During "Phase 2" design a detailed force analysis will be made, considering all aspects.

Mounting Design Load

Magnitude of the impact load described in mounting design condition is calculated as follows:

$$F_I = \frac{3\,(GVW)}{2} = \frac{3\,(35,000)}{2} = 52,500 \text{ pounds}$$

where

F_I = impact load

GVW = gross vehicle weight

EXHAUST SYSTEM DESIGN ANALYSIS

INTRODUCTION

The dual exhaust system of the LVTPX11 is connected to the "V" type, 8-cylinder Cummins engine and consists of flexible ball-type joints, exhaust pipes and mufflers. (See drawing SK-5240)

The preliminary analysis of the complete system follows below.

PRIMARY SPECIFICATIONS:

Sound Level Approximately 130 to 150 Sones.

Prevention of crew compartment contamination.

Minimum Infrared radiation.

1000 hours minimum component life.

Minimum engine compartment heating.

DESIGN CONDITION:

To provide an adequate silencing and exhaust system for all operating conditions of engine and vehicle. The maximum horsepower extraction point (maximum governed speed) is used as the design condition.

DESIGN CALCULATIONS:

According to the engine manufacturer's standard conditions, the following values obtained for the exhaust system design are based on a barometric pressure of 29.9 inches of mercury, at sea level, and 60°F intake air temperature.

Total air intake at manifolds	$(2\ V_1)$ 600 CFM
Exhaust Gas temperature at exhaust valve ports	(T_2) 1100°F
Allowable back pressure in exhaust system	1 inch Hg
Dual exhaust system, exhaust pipes	3 inches OD

Intake air pressure at intake valves (P_1) 1 psi below atmospheric pressure.

Exhaust air pressure at exhaust valves (P_2) 4 psi above atmospheric pressure.

Exhaust air volume	(V_2)
Intake air temperature	(T_1) 60°F

$0°K = 273°C = (32 + 273\ 9/5)°F = 523.4°F$

$T_1 = 523.4° + 60° = 583.4°F$

$T_2 = 523.4° + 1100° = 1623.4°F$

Barometric pressure = 29.9 inches Hg

Atmospheric pressure = 1 atm = 14.7 psi

$P_1 = 14.7 - 1 = 13.7$ psi

$P_2 = 14.7 + 4 = 18.7$ psi

To obtain the exhaust gas volume V_2 at the manifold outlet, the equation for an ideal gas is used:

$$\frac{P_1 \times V_1}{T_1} = \frac{P_2 \times V_2}{T_2}$$

and: $$V_2 = \frac{P_1 \times V_1 \times T_2}{P_2 \times T_1} = \frac{13.7 \times 300 \times 1623.4}{18.7 \times 583.4} = 610 \text{ CFM}$$

The engine manufacturer requests a dual exhaust and 3 inch diameter ducting. The design shows two bends and one foot of straight tubing.

Friction Losses In Exhaust Tubing

Bend Radius = 5 inches

Tube Diameter = 3 inches

R/D = 1.666

This value is equivalent to 10 x D = 30 inches of straight duct length.

Two bends = 60 inches = 5 feet of straight ducting.

Friction loss of 3-inch ducting with 600 CFM passing thru = 50 inches of water in 100 feet of tubing.

In 5 feet = $\frac{50}{100} \times 5 = 2.5$ inches water $= 2.5 \times 0.0735 = 0.184$ inches Hg

NObs 4561
Power Train Design Analysis

$$1 \text{ ft of straight ducting} = \frac{50}{100} \times 0.0735 = 0.0367 \text{ in. Hg}$$

$$\text{loss in tubing} = .2207 \text{ in. Hg}$$

$$\text{loss in muffler (from manufacturers data)} = .655 \text{ in. Hg}$$

$$\text{Total loss} = .8757 \text{ in. Hg}$$

$$0.875 \text{ in. Hg} < 1 \text{ in. Hg}$$

$$0.875 \text{ in. Hg} = \frac{14.7}{29.9} \times 0.875 = 0.43 \text{ psi back pressure}$$

It can be seen, with the frictional losses of .875 inches mercury being only 12.5 percent under the maximum allowable, that the choice of the exhaust system design is justified. Stainless steel tubing in conjunction with the "ball-socket" type joints gives a smooth inside surface with a minimum of friction to overcome. The use of flexible tubing, with a rougher inner surface, could increase the frictional losses, and therefore the back pressure, above the maximum allowable.

Exhaust Gas Velocity at Muffler Outlet

It is assumed that the exhaust gas temperature of 1100°F at the exhaust valves will drop 150°F going through the manifold, entering the exhaust pipes with 950°F. The exhaust pipes are insulated with a ceramic fiber of very low thermal conductivity, and the temperature drop between inlet and outlet is only two percent. The gas temperature entering the muffler is therefore, 950°F x .98 = 930°F. A final temperature decrease through the muffler of 280° brings the exhaust gas outlet temperature to 650°F.

Exhaust gas volume at muffler outlet:

$$V_2 = \frac{P_1 \times V_1 \times T_2}{P_2 \times T_1}$$

where: (CFM) V_2 = Exhaust Gas Volume at Muffler Outlet

(CFM) V_1 = Exhaust Gas Volume at Manifold Outlet

(PSI) P_1 = Exhaust Gas Pressure at Manifold Outlet

(PSI) P_2 = Exhaust Gas Pressure at Muffler Outlet

(°F) T_1 = Exhaust Gas Temp. at Manifold Outlet

(°F) T_2 = Exhaust Gas Temp. at Muffler Outlet

$P_1 = 18.7\text{ psi} + 0.43 = 19.13\text{ psi}$

P_2 = Atmospheric Pressure = 14.7 psi

$T_1 = 523.4 + 1100 = 1623.4°F$

$T_2 = 523.4 + 650 = 1173.4°F$

V_1 = 610 CFM at each manifold

$$V_2 = \frac{P_1 \times V_1 \times T_2}{P_2 \times T_1} = \frac{19.13 \times 610 \times 1173.4}{14.7 \times 1623.4}$$

$$\underline{\underline{V_2 = 574\text{ CFM}}}$$

NObs 4561
Power Train Design Analysis

Exhaust Gas Velocity = $\frac{574}{D^2\pi/4}$, D= Muffler Outlet dia. = 3 in. = 0.25 ft

$$= \frac{574 \times 4}{0.25^2\pi} = 11700 \text{ feet per minute}$$

$$= 136 \text{ miles per hour}$$

COMPONENT SELECTION

Selection of components based on this analysis in conjunction with vendor recommendations are as follows:

Mufflers

8.5 in. dia. x 31 in. long

Material: Stainless Steel

Exhaust Pipes

3 in. dia. Stainless Steel Tubing

Connectors

Stainless Steel "Ball-Socket" Type

COOLING SYSTEM

General Description

The cooling system for the LVTPX11, designed to provide optimum cooling for the engine coolant and lubricating oil, and the transmission assembly including converter and lubrication oil, utilizes a dual radiator and combination heat exchanger installation.

Two water-to-air radiators and integral oil-to-water heat exchangers are utilized. The radiator assemblies are installed in individual compartments located at the rear of the vehicle on either side of the machinery compartment. Cooling air flow through each compartment is supplied by a hydraulically-driven propeller-type fan. The fans are mounted ahead of the radiators, i.e., they act to push the air flow through the radiator core. The fans operate continuously during vehicle operation.

The arrangement of the cooling components is designed to provide an over-all balanced system by dividing the heat load and giving relatively equal heat-rejection values to each radiator assembly. The engine coolant flow is divided by connecting a radiator assembly to each engine cylinder bank outlet. The return flow to the engine is collected and enters the engine again at a common line. The heat exchanger in the starboard radiator assembly is in series with, and augments, the integral engine oil cooler. The heat exchanger in the port radiator assembly dissipates the heat rejected from the transmission.

Division of the cooling system into two parts permits optimum sizing of the radiators and fans, and permits arrangement of the compartments to place the fans as well as the radiators above the normal load waterline. An advantageous feature of the hydraulically-driven fans is the ability to stall or "slip" for indefinite periods should the fan be temporarily inundated during surf operation.

Cooling System Design Data

Design Conditions

Preliminary investigation indicates that two design conditions may be utilized in the design and sizing of the cooling system:

(See figure M, LVTPX11 Cooling System Schematic)

CONDITION I:	Continuous operation at maximum vehicle speed (40 mph) and 125°F ambient temperature
CONDITION II:	Continuous operation at 70 percent converter efficiency and 125°F ambient temperature

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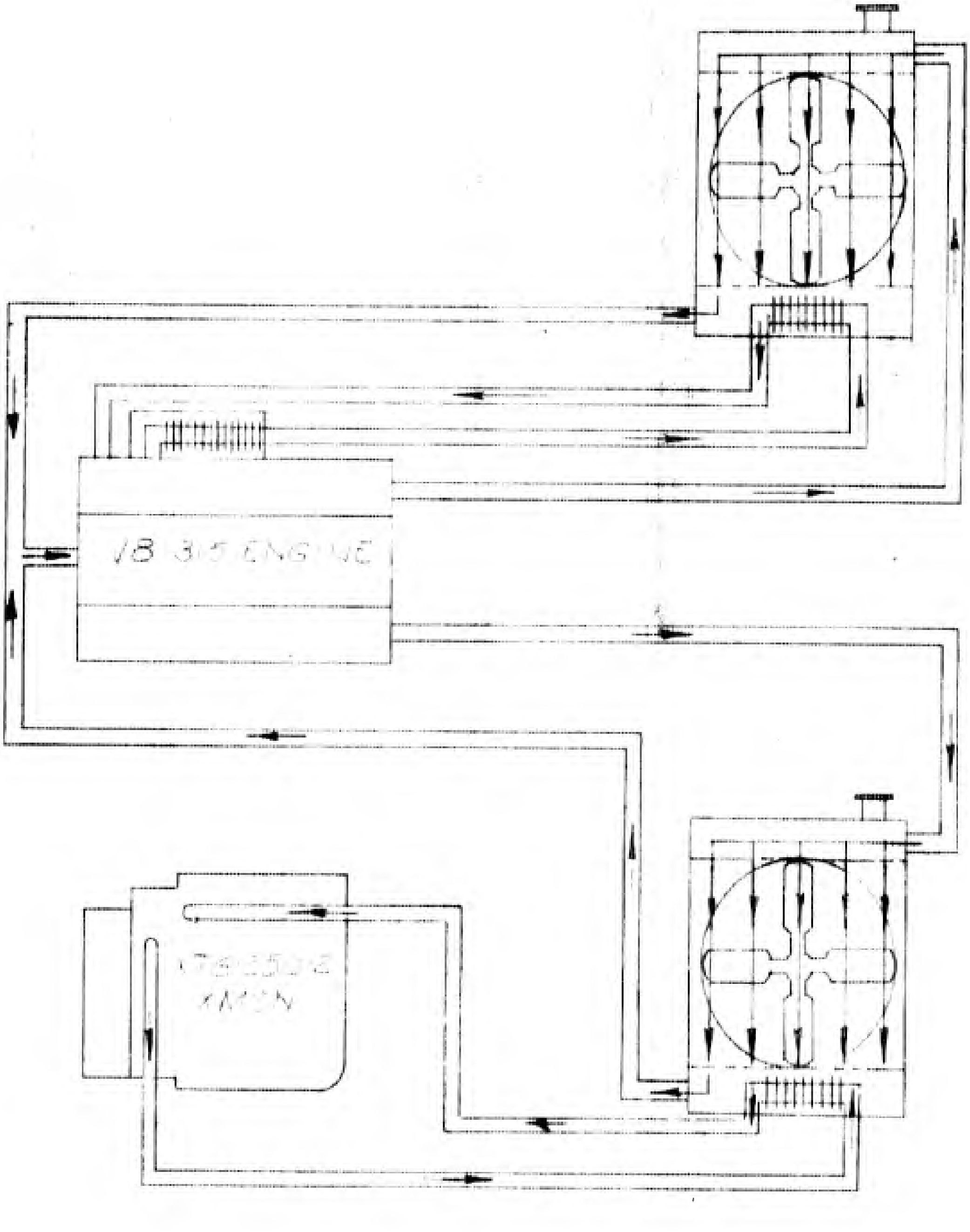

Fig. M. **LVTPX11 Cooling System Schematic**

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Design Temperatures

The nominal design temperatures are established as:

Maximum Ambient Air Temperature---Land	125° F.
Maximum Ambient Air Temperature---Water	90° F.
Maximum Sea Water Temperature	85° F.
Maximum Engine Coolant Temperature	200° F.
Maximum Engine Oil Temperature	250° F.
Maximum Transmission Oil Temperature	300° F.

Arrangement

Radiator "A" (Stb'd): Cooling capacity for 1/2 engine coolant, plus engine oil heat rejection

Radiator "B" (Port): Cooling capacity for 1/2 engine coolant, plus transmission oil heat rejection

Fan Data: 23-inch diameter, 6-blade propeller type, 2,000 rpm

Design Heat Rejection Values

Components: Cummins V8-315 engine, with Allison XTG-250-2 Transmission

<u>CONDITION I:</u> 40 mph, level ground at 125 °F ambient temperature
268 brake horsepower gross at 3,000 rpm engine speed

Heat Rejection Rate-Engine Coolant	26.0 Btu/hp/min
Total Heat Rejection to Engine Coolant 26.0 (268)	6,960 Btu/min
Engine Oil Heat Rejection	2,000 Btu/min
Transmission Heat Rejection at lockup, 12 gpm oil flow	1,600 Btu/min

CONDITION II: Converter at 70 percent efficiency, 125°F ambient temperature
248 brake horsepower gross at 2,310 rpm engine speed

Heat Rejection Rate-Engine Coolant	28.8 Btu/hp/min
Total Heat Rejection to Engine Coolant 28.8 (248)	7,150 Btu/min
Engine Oil Heat Rejection	1,500 Btu/min
Converter-Transmission Heat Rejection 20 gpm oil flow, 42 psi	2,230 Btu/min

Then, the cooling loads on the radiators are as follows:

RADIATOR "A":	Condition I	Condition II
1/2 Engine Coolant Heat	3,480 Btu/min	3,575 Btu/min
Engine Oil Heat	2,000 Btu/min	1,500 Btu/min
Total Head Load -	5,480 Btu/min	5,075 Btu/min
RADIATOR "B":		
1/2 Engine Coolant Heat	3,480 Btu/min	3,575 Btu/min
Converter-Transmission Heat	1,600 Btu/min	2,230 Btu/min
Total Heat Load -	5,080 Btu/min	5,805 Btu/min

It may be noted that the XTG-250 Transmission lubricating circuit bypasses oil flow at the higher speeds, resulting in the respective flows shown above.

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Cooling System Calculations

Based on the foregoing values, a preliminary analysis of the cooling system is made.

It may be seen that Condition II, Radiator "B", is critical for design. Since the radiator assemblies are to be interchangeable, the cooling system components are, consequently, sized and designed for this condition.

Analysis:

A radiator assembly having a core constructed of tubes and fins of the following dimensions may be considered for further evaluation:

Radiator: Young Radiator Type AAH
6-Row, 9 Fins per inch, 24 15/16 inch x 24 inch core opening

Wanted:

1. Water Temperature Drop Through Radiator
2. Oil Temperature Drop Through Heat Exchanger
3. Fan Horsepower and Selection

Given:

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Condition I:

Heat Rejected Radiator A	5480 Btu/min
Heat Rejected Radiator B	5080 Btu/min
Water Flow Engine (Total)	90° gpm
Water Temperature (Engine Outlet)	200 °F
Oil Flow Engine	15 gpm
Oil Temperature Maximum Engine	250 °F
Oil Flow Transmission	12 gpm
Oil Temperature Maximum Transmission	300 °F
Ambient Temperature	125 °F

Temperatures are maximum for normal driving conditions. Temperatures and flows are supplied by engine and transmission manufacturers.

Condition II:

Heat Rejected Radiator A	5075 Btu/min
Heat Rejected Radiator B	5805 Btu/min
Water Flow Engine (Total)	67 gpm
Water Temperature (Engine Outlet)	200° F
Oil Flow Engine	11 gpm
Oil Temperature Maximum Engine	250 °F
Oil Flow Transmission	20 gpm
Oil Temperature Maximum Transmission	300 °F
Ambient Temperature	125 °F

Temperatures are maximum for normal driving conditions. Temperatures and flows are supplied by engine and transmission manufacturers.

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Theory:

1. Find water temperature drop through the radiator to the oil heat exchanger inlet.

ΔT, the water temperature drop $= \dfrac{H}{Qd(SpHt)}$

$$\Delta T = T_1 - T$$

$$T \text{ average} = \frac{T_1 + T}{2}$$

$$\Delta T_A = T_{ave} - T_{amb}$$

where H = heat rejected to radiator Btu/min
Q = flow rate, gpm

d = pounds per gallon, for water d = 8.1 pounds per gallon

SpHt = Specific Heat = 1 for water

T_1 = temperature into radiator
T = temperature out of radiator and into oil heat exchanger
T_{amb} = ambient air temperature
T_{ave} = average water temperature
ΔT_A = average water temperature minus ambient air temperature.

2. Find over-all heat transfer coefficient for radiator, U

$$U = \frac{H}{F_A \, \Delta T_A}$$

F_A = Core Area

3. Correct for flow rate:

   U/K K = Correction Factor

4. From manufacturer obtain air flow required through radiator to dissipate heat

   V_A = Air Velocity

5. Air Flow

   $$CFM = V_A \times F_A \text{ at } 70^{\circ} F$$

   $$CFM_{corrected} = CFM \times \frac{(460 + T_{amb})}{(460 + T_{70^{\circ}})}$$

   $$= CFM \times \frac{(460 + T_{amb})}{530^{\circ}}$$

6. Hp Required

   $$Hp = \frac{CFM_{corrected} \times \text{back pressure (in. } H_2 0)}{6350 \times \text{fan efficiency } (\%)}$$

Solution:

The highest continuous heat load is used in each condition.

Condition I:

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1.

$$\Delta T = \frac{H}{Qd(SpHt)}$$

$$\Delta T = \frac{5480}{45 \times 8.1 \times 1}$$

$$= 15^\circ F$$

$$T = T_1 - \Delta T = 200 - 15$$

$$= 185^\circ F$$

$$T_{ave} = \frac{T_1 + T}{2} = \frac{200 + 185}{2}$$

$$= 192.5^\circ F$$

$$\Delta T_A = T_{ave} - T_{amb} = 192.5 - 125 = 67.5^\circ F$$

$H = 5480$ Btu/min
Rad. A
$Q = 90/2 = 45$ gpm
to each radiator
$d = 8.1$ pounds per gallon
$SpHt = 1$
$T_1 = 200^\circ F$
$T_{amb} = 125^\circ F$
T_2 = temperature out

Elevated water temperature out of radiator due to heat dissipated by oil heat exchanger.

$T_{in} = T_1$ to $T_{out} = T_2$

$$T = \frac{H_{water}}{Qd(SpHt)} = T_1 - T_2$$

Condition I

$$T = \frac{3480}{45 \times 8.1 \times 1} = 9.55^\circ F$$

$$T_2 = 200 - 9.55 = 190.5^\circ F$$

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2. $U = \dfrac{H}{F_A \, \Delta T_A}$

$= \dfrac{5480}{4.15 \times 67.5}$

$= 19.5$ Btu/min/Ft^2/100°F ΔT

$F_A = 24 \times 24.9375/144$

$= 4.15 \; F_t^2$ Core Area

3. $U_{corrected} = U/K$

$= 19.5/.96 = 20.3$

$K = .96$ (from manufacturer for flow of 45 gpm)

4. $V_A = 1750$ FPM (from manufacturer)

at 70°F

with 1.3" back pressure

5. $CFM = V_A \times F_A = 1750 \times 4.15 = 7260$ CFM at 70°F

$CFM_{corrected} = 7260 \times \dfrac{585}{530} = 8000$ CFM at 125°F

6. $Hp = \dfrac{CFM_{corr} \times \text{back pressure}}{6350 \times \text{Eff.}}$

$= \dfrac{8000 \times 1.9}{6350 \times .41} = 5.9$ Hp

Back pressure

$= 1.3 + .6 = 1.9$ in. Hg.

(.6 in Hg duct loss)

Eff = 41%

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Condition II

1. $\Delta T = \frac{5805}{33.5 \times 8.1 \times 1} = 21.4^{\circ}F$

H = 5805 Btu/min (Rad. B)
Q = 67/2 = 33.5 GPM
d = 8.1 pounds per gallon
SpHt = 1
$T_1 = 200^{\circ}F$
$T_{amb} = 125^{\circ}F$
T_2 = Temp. Out

$T = 200 - 21.4 = 178.6^{\circ}F$

$\Delta T_{ave} = \frac{200 + 178.6}{2} = 189.3^{\circ}F$

$\Delta T_A = 189.3 - 125 = 64.3^{\circ}F$

Elevated water temperature out of radiator due to heat dissipated by oil heat exchanger.

$\Delta T = \frac{3575}{33.5 \times 8.1 \times 1} = 13.2^{\circ}F$

$T_2 = 200 - 13.2 = 186.8^{\circ}F$

2. $U = \frac{5805}{4.15 \times 64.3} = 21.8$

3. $U_{corr} = \frac{21.8}{.95} = 23$

4. V_A = 2050 ft/min at 70°F (from manufacturers data back pressure = 1.6 + .6)
Rad. ducts & grilles

5. $\text{CFM} = 2050 \times 4.15 = 8500$ CFM at 70° F

$\text{CFM}_{corr} = 8500 \times \frac{585}{530} = 9400$ CFM at 125°F

6. $\text{Fan Horsepower} = \frac{\text{CFM} \times \text{pressure drop inches } H_2O}{6350 \times \text{fan eff. } \%}$

$\text{F.H.P.} = \frac{9400 \times 2.2}{6350 \times .41} = 8 \text{ hp}$

Hence the radiator and fan selection have been sized to condition II, which is the severest continuous duty.

Air Temperature Change Thru Radiator

$$\Delta T = T_{out} - T_{in} = \frac{H}{Q_1 d_1 SpHt}$$

H = Heat Rejected
Q_1 = Air Flow CFM
d_1 = pounds per cubic foot of air = .0679 pounds per foot at 125°F
$SpHt$ = .24 Btu per pound°R for air

T_{out} = Air Temp Out
T_{in} = $T_{ambient}$ = 125°F
Q_I = 8000 CFM at 125°F
Q_{II} = 9600 at 125°F

Condition I

$$\Delta T = \frac{5480}{(8000)\ (.0679)\ (.24)} = 42^\circ F$$

$$T_{out} = 125^\circ F + 42 = 167^\circ F$$

Condition II

$$\Delta T = \frac{5805}{(9600)\ (.0679)\ (.24)} = 37^\circ F$$

$$T_{out} = 125 + 37 = 162^\circ F$$

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Oil Temperature Change Through Radiator

$$\Delta T = T_{in} - T_{out} = \frac{H}{QdSpHt}$$

H = Heat Rejected
G = Flow Rate, GPM
d = Pounds per gallon = 7.7 Pounds per gallon of oil (average)
SpHt = .5 for oil specific heat

Transmission Oil: Condition II

$$T = \frac{2230}{20 \times 7.7 \times .5} = 29\,^{\circ}F$$

$$T_{out} = 300\,^{\circ}F - 29 = 271\,^{\circ}F$$

Q = 20 GPM
H = 3090 Btu/min

Engine Oil Change Through Radiator
Condition I

$$\Delta T = \frac{2000}{15 \times 7.7 \times .5}$$

$$= 34.6\,^{\circ}F$$

$$T_{out} = 250 - 34.6 = 215.4\,^{\circ}F$$

Q = 15 GPM
H = 2000

Tip Velocity of Fan:

$$V_T = N \times \frac{\pi D}{12}$$

$$= \frac{2000 \times 23 \times \pi}{12}$$

N = Fan Speed RPM = 2000 RPM

D = Fan Dia. = 23 inches

= 12,000 FPM. This is within a noise limit velocity of 14,000 FPM.