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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

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Volume II of IV

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5.0. SUSPENSION SYSTEM

5.1. General Description. (Refer to drawing SK-5210).

The LVTPX11 suspension system incorporates an over-the-roadwheel track return as specified in BuShips Contract Specification SHIPS-A-4159. The system is composed of five major component assemblies which are:

A. Support Arm and Roadwheel Assembly.

B. Front Idler Assembly.

C. Sprocket and Support Wheel Assembly.

D. Track Assembly.

E. Shroud and Fender Installation.

Each of these interrelated and matched components, together with vehicle system advantages, will be covered in detail in the remainder of this section.

5.1.2. General Discussion.

5.1.2.1. The use of this type of suspension, commonly called a Flat-Track Suspension System, results in a minimum number of components for utmost simplicity, reduction in maintenance time, and reduction in weight and space requirements. The relatively large roadwheels inherent in the system, together with reduced friction losses afforded by a minimum number of components, results in rolling resistances approximately 25 percent lower than those for support roller systems. The rolling resistances for the LVTPX11 under various conditions are estimated to be:

On hard level surfaces at 40 mph - 70 lb per ton

Average cross-country at 10 to 20 mph - 120 lb per ton

70 percent grade operation at 2 to 4 mph - 75 lb per ton

The above figures compare favorably with the specified rolling resistances of 100 pounds per ton for hard surface operation and 140 pounds per ton for 70 percent grades called for in the contract specification.

5.1.2.2. The estimates noted above are based on curves published by the Detroit Arsenal for similar tracked vehicles. (See figure 5-1).

5.1.2.3. The track ground contact length, centerline of front to rear roadwheel is 135.75 inches, and the tread, centerline to centerline of track, is 102.75 inches for a length per width ratio of 1.32, which should produce very good maneuverability.

5.1.2.4. The 20-inch wide track results in a total ground contact of 6,100 square inches at two inches penetration. At a gross weight of 35,000 pounds, the ground pressure is 5.7 psi which provides the mobility specified in paragraph 3.11.4.1 of BuShips Contract Specification SHIPS-A-4159.

5.2. <u>Support Arm and Roadwheel Assembly</u>. (Refer to drawing SK-5207).

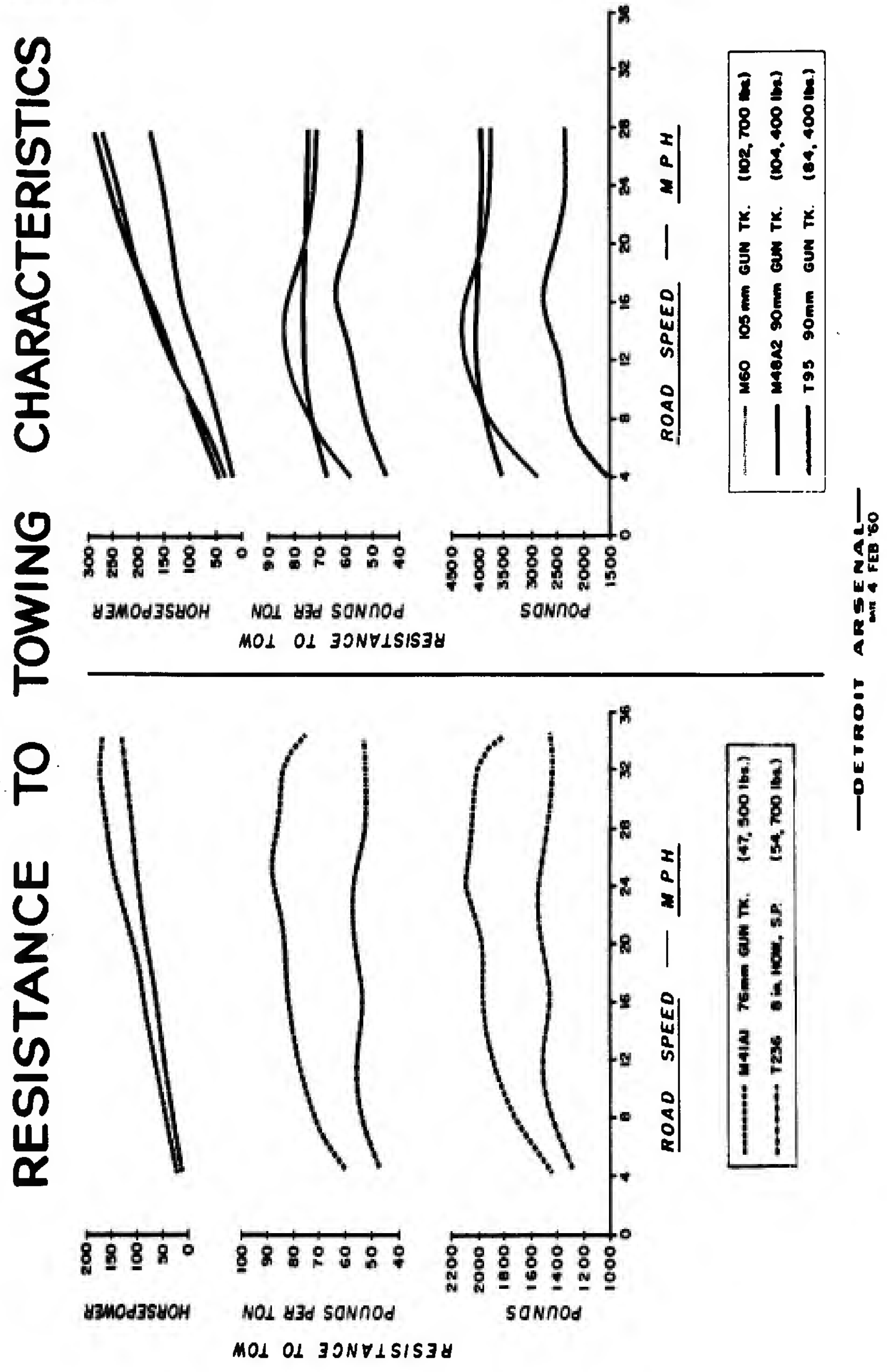

RESISTANCE TO TOWING CHARACTERISTICS
RESISTANCE TO TOW
HORSEPOWER
POUNDS PER TON
POUNDS
ROAD SPEED — M P H
M60 105mm GUN TK. (102,700 lbs.)
M48A2 90mm GUN TK. (104,400 lbs.)
T95 90mm GUN TK. (84,400 lbs.)
M41A1 76mm GUN TK. (47,500 lbs.)
T236 8 in. HOW., S.P. (54,700 lbs.)
DETROIT ARSENAL
4 FEB '60

Figure 5-1.

5.2.1. Function and Description.

The function of the support arm and roadwheel is, as the name implies, to support the vehicle. In addition, it must damp out irregularities in the terrain to insure reasonable comfort for crew and embarked personnel during rough terrain operation. This is accomplished by eight support arm and roadwheel assemblies, four per side, which are cantilever mounted to the hull in accordance with specifications. Rubber torsilastic springs provide the desired spring action by means of trailing arms. The weight of the removable roadwheel is 80.3 pounds and the support arm and hub assembly weighs 152 pounds, for a total assembled weight of 232.3 pounds for each unit.

5.2.2. General Arrangement.

5.2.2.1. The support arm and wheel assembly consists of five major details arranged to produce a package assembly which is easy to service either on or off the vehicle.

5.2.2.2. A hull mounting flange is utilized to attach the assembly to the hull by means of a pilot diameter and bolting ring. The hull mounting has no communication with the interior of the hull, to insure watertight integrity. The opposite end of the flange is arranged to serve as a bearing journal and thrust ring. After the bearings and seals are installed in the support arm, the flange is installed in the arm and retained by two thrust washers and a thrust clamp. The thrust clamp is adjusted by means of shims to insure optimum thrust washer clearance.

5.2.2.3. The support arm and hull mounting flange subassembly are now locked firmly together but free to rotate. To provide the necessary spring action a rubber torsilastic spring is mounted between the flange and the support arm. The outer diameter of the spring is locked to the flange by diametrally opposed keys. The inner diameter of the spring is locked to the support arm by splines. Blind splines, together with offset holes in the hull mounting flanges, will insure proper orientation of the spring and subsequently the proper angle of set when mounted on the hull. Several very distinct advantages occur with this arrangement. First, the bearing arrangement relieves the spring of all thrust, compression, and tilt loads so it can function purely as a torsion member. Second, the spring is totally enclosed and sealed to reduce ozone deterioration. The result will be optimum spring life. An additional attractive feature is that the spring, being a floating member, is easily replaced by sliding the old spring out and the new spring in.

5.2.2.4. The wheel bearing spindle is part of the support arm. This arrangement produces an overlap of the spindle and outer spring and bearing barrel which allows the design of a very compact, lightweight component. The central torque member, which is also integral with the arm, has been sized to accept a lockout system for the special-purpose vehicles which are a continuing part of the program. Specifically the torque member will be splined internally to accept a torque shaft extending through the hull to an internal lockout member.

5.2.2.5. The wheel bearing arrangement consists of a spherical roller bearing and a cylindrical roller bearing. The spherical roller takes all thrust and is clamped in place to eliminate the possibility of failure due to improper adjustment. The cylindrical roller bearing has a separable inner race for ease of disassembly and assembly. An external triple-lip seal with all lips facing outward is utilized so that periodic greasing will tend to flush out any contaminant that may enter the bearing cavity. Replaceable seal races are included to comply with specifications. The arm and wheel hub overlap to provide a seal shedder that will eliminate the possibility of wire or similar stringy materials winding into and destroying the seal. Except for size, the general arrangement is identical to the latest LVTP5 wheels which have proven very satisfactory in service.

5.2.2.6. The wheel hub, which provides the rotating member, contains a pilot diameter and bolting ring for mounting the roadwheels. This component will be a hollow casting with a foam filler to provide maximum buoyancy commensurate with strength requirements. An additional advantage of this hub is that all surfaces are smooth to preclude collection of mud and other debris commonly experienced with externally ribbed or flanged hubs.

5.2.2.7. The original bid proposal recommended separate roadwheels bolted to the hub to form a dual assembly. This arrangement does allow replacement of individual wheels in the event of a failure. Actually the failure is confined to the tire itself because the rim or wheel is usually a very dependable component. Considering failure of the tire, experience has shown that when one tire of a dual set experiences extreme damage or wear the mate is also very close to the failure point. Under this condition a new tire mated with an old tire will seriously affect the life and performance of the new tire because the new assembly has to do twice the work until both reach an equal state of wear. From this it is concluded that tires should be changed as pairs and not as individual units. Further it is known that two separate units performing a single function as a set must be heavier than a single inseparable unit. For this reason the Design Agent recommends the single wheel dual tire arrangement shown on drawing SK-5207. Compared to the dual wheel shown as an alternate, the combined unit will displace 270 pounds more water than will the dual wheels. To summarize, the Design Agent recommends the combined unit because of savings in weight and an increase in displacement. However, either approach is an acceptable design. Therefore the choice must be based on whether the using forces consider weight and displacement, or the ability to change separate wheels as the most important feature.

5.2.2.8. In either case the aluminum wheels with steel wear surfaces and entrapped rubber tire will be used as specified.

5.2.3. Detail Construction.

5.2.3.1. Hull mounting flanges, support arms, wheel hubs and wheels will be cast aluminum. The material will be A356T6 or Tensiloy depending on the casting source. All prototype components will be sand castings and will be x-rayed 100 percent to insure sound castings. Production castings would be produced in permanent molds for an even greater increase in physicals and soundness.

5.2.3.2. The hub and wheel castings will require internal cores and must therefore have core support and vent holes in the side walls. Welding filler plates in these holes to produce watertightness is costly and time consuming plus the fact that 100 percent pressure testing must be done to insure watertightness. Therefore it is planned to fill the cavity with foamed-in-place plastics. Koppers Dyalite which is an expandable bead material applied by steam heat has proved very satisfactory in LVTP5 experimental wheels. Therefore this material will be specified for this application.

5.2.3.3. Sprayed-on stainless steel has been tested on Army Ordnance tank wheels and does provide reasonable resistance to wear. However, it has not been accepted for production because the time required to spray on an acceptable thickness results in excessive cost. It is also noted that repair of the wheels preparatory to molding new tires has not proven satisfactory because too much of the base metal has to be removed. Results are not considered conclusive. Therefore the Design Agent will continue to pursue this approach during detail design. However, pending more definite methods of application and cost data, the Design Agent is proceeding with hardened wear strips riveted to the wheels. This approach will allow repair of used wheels. The worn wheels can be remachined on the mold seal areas to fit in repair molds. New wear strips can then be riveted in place and new tires applied.

5.2.3.4. Wheel bearings are standard bearings and will be specified by MS numbers on production drawings. The support arm thrust washers and radial sleeve bearings will be fabric-laminated Phenolic bearings to reduce wear of the aluminum parts. The bearings will be packed with grease at assembly but no periodic greasing is necessary. Bearings of this type, produced by the Gatke Corporation of Chicago, Illinois, are being used in the suspension arms of the M113 Ordnance vehicle. This application, which is very similar to the installation proposed for the LVTPX11, have proven very satisfactory on production vehicles as well as on test units.

5.2.3.5. The triple-lip seals used to seal the wheels and those used to seal the support arm bearings are special sizes. However, the processing and materials will be identical to the LVTP5 triple-lip seals which have been very satisfactory in the final drive and in the full width rubber tire wheel application. Experience has definitely established that rubber torsilastic springs have the best life potential if split outer shells are used in construction. The use of the split shells during molding allows the rubber to contract naturally and then when the spring is pressed into an outer tube the rubber is put under compression. When the rubber is molded to a solid outer cylinder and a solid inner shaft, the contraction of the rubber produces tension which is undesirable. Apparently, keeping the rubber under compression increases the fatigue life in the same manner that shot peening increases fatigue life in ferrous metals. For this reason the LVTPX11 spring will be furnished as an assembly. The assembly will be fabricated by first molding together a pair of outer half cylinders and a shaft which is splined internally. This spring subassembly will then be pressed into the outer cylinder. Failed springs could be returned to the Vendor who would be able to salvage the metal parts to prepare new springs.

5.2.3.6. All bolts, nuts and other ferrous parts such as wheel wear strips will be suitably plated to inhibit electrolytic corrosion.

5.3. Front Idler Assembly. (Refer to drawing SK-5208).

5.3.1. Function and Description.

The only function that the front idler assembly performs on the LVTPX11 is to return and guide the track to the ground to complete the track laying cycle. This is accomplished by two assemblies, one on each side of the vehicle. Each assembly has a total weight of 149.6 pounds. Each of the removable wheels weighs 36.9 pounds.

5.3.2. General Arrangement.

5.3.2.1. The front idler assembly consists of three major details arranged in a very simple manner to insure ease of maintenance.

5.3.2.2. An idler stand serves to mount the assembly to the hull by means of a pilot diameter and bolting circle. The opposite end of the stand functions as the wheel bearing spindle.

5.3.2.3. The wheel bearing arrangement is composed of spherical roller bearings and cylindrical roller bearings protected by external triple-lip seals. Except for a difference in size to agree with different loading, this arrangement is identical to, and functions the same as, the system described for the roadwheels under paragraph 5.2.2. above.

5.3.2.4. The idler hub is a simple casting that contains a pilot and a bolting circle for mounting the idler wheels.

5.3.2.5. Two idler wheels separated by a spacer are utilized in each assembly. They may be replaced without removing the assembly, and the bearings are not exposed during this operation. The dual wheels are necessary in this case to make the wheels interchangeable with the sprocket support wheels, as will be discussed subsequently in this section.

5.3.3. Detail Construction.

The mounting stand, wheel hub and wheels will be sand castings. Materials and methods of fabrication will be identical to the support arm and wheel details discussed under paragraph 5.2.3 above. Application of wheel wear rings and rubber tires will also be identical to the methods described for the roadwheels.

5.4. Sprocket and Support Wheel Assembly. (Refer to drawing SK-5209).

5.4.1. Function and Description.

The purpose of the sprocket and support wheel assembly is to transmit power from the final drive to the track as well as to guide and direct the track to the front of the vehicle. This is accomplished by two assemblies, one on each side of the vehicle. Ordinarily, two sprockets and a hub are used for this assembly. However, the selection of a center-drive side-guide track allows the use of a very simple drive system with several distinct advantages. The use of the single central sprocket eliminates the problem of precision indexing of sprocket pairs which has proved to be troublesome on many vehicles. In turn the support tires act to set the pitch circle and to guide the track. This allows the sprocket to be relieved at the root circle for good mud clearance, which reduces the tendency to climb or jump the teeth. In addition, all sprocket side loading is eliminated which still further reduces the tendency to climb the teeth. The weight of the sprocket is 35.2 pounds and each support wheel weighs 36.9 pounds for a total assembled weight of 109 pounds.

5.4.2. General Arrangement.

5.4.2.1. The sprocket and support wheels bolt directly to the final drive output shaft on the LVTPX11. Therefore, the arrangement consists simply of a sprocket and two support wheels. To assemble, one support wheel is fed on the pilot ring followed by the sprocket and the second support wheel. Servicing is thus accomplished without affecting the final drive in any way.

5.4.2.2. The tooth sprocket will be steel for optimum wear life. The only deviation from standard sprocket procedure is that the root diameter will be relieved for mud clearance as mentioned earlier in this section.

5.4.2.3. The support wheel must set the pitch diameter of the sprocket; therefore, it has been sized accordingly. The configuration must agree with track guide width and at the same time allow space for the center sprocket mounting.

Once this configuration and size is set, it is then used as the front idler wheel to reduce the number of separate parts that must be stocked.

5.4.3. Detail Construction.

5.4.3.1. The prototype sprockets will be sand castings. Material will be QQ-S-681 b 4C3. The tooth contours will be machined and locally flame hardened at the high wear areas. The possibility of using forged steel sprockets with unmachined tooth contours will be investigated for production.

5.4.3.2. Except for size, the support wheels will be identical to the roadwheels described under paragraph 5.2.3 above.

5.5. Track Assembly. (Refer to drawing SK-5211).

5.5.1. Discussion.

5.5.1.1. BuShips Contract Specification SHIPS-A-4159 paragraph 3.12.1.8.1 states that "the tracks shall be steel hinge-pin type with preferably threaded pin elements and of a type which minimizes road damage". As will be described later the Design Agent has complied with the specification except for the "threaded pin elements".

5.5.1.2. One of the basic problems with a lubricated hinge-pin track is clevis wear. Wear in the clevis areas destroys the seal shedder effect of the mating parts. The exposed seal then becomes vulnerable to the jamming action produced by dirt and other abrasive material during operation of the vehicle. The utilization of a threaded pin would eliminate this wear because all thrust is absorbed internally in a normally lubricated and sealed area. For this reason the Design Agent proposed that a Task be established to test the threaded pin in a modified LVTP5 track. Subsequently, Task 3855-61-244 was established and two tracks were modified and forwarded to the Marine Corps Test and Experimental Unit at Camp Pendleton for evaluation. One of the tracks contained unsealed threaded pins and the remaining track incorporated end-compression type seals. After approximately 80 hours of operation the unsealed track had elongated approximately 20 inches due to wear of the threaded pin area. The sealed track had worn approximately 15 inches in the same period of operation. The wear in this case was due to very inefficient sealing which allowed dirt and salt water to enter the threaded bearing areas.

5.5.1.3. A compression seal of the type used in this track will function well with proper installation. However the efficiency of the seal is dependent on the amount of end compression applied at initial installation. The LVTP5 track or any track with set clevis openings cannot be machined closely enough with practical tolerances to insure optimum end compression of the seal. Another deficiency noted during evaluation at the Test and Experimental Unit is that field assembly of the threaded pins is extremely difficult for two reasons.

One, the clevis joints cannot be interlocked without destroying the seals even when shims are used as specified. Second, the male clevis arms are not stable in the LVTP5 track. Therefore, it is very difficult to start the thread in the inner clevis after passing through the outer and center clevis joints. Another detail that caused considerable trouble during evaluation was the roll pin used to prevent rotation of the connecting pin in the outer clevis areas. The tendency of the pin to rotate in this area was greatly exaggerated by friction of the unsealed threads and could be corrected in future tracks.

5.5.1.4. To summarize, the design agent firmly believes that the threaded pin concept offers a definite solution to a basic track problem. Therefore it should be pursued on an experimental basis, possibly in a double-pin track configuration so that threaded pins and seals could be organized as a package before joining links are installed. However there is no known practical solution to the sealing problem in a threaded hinge-pin track. Therefore, the Design Agent recommends that the threaded pin be abandoned in favor of a self-lubricated bearing-and-pin arrangement similar to the LVTP5 track but incorporating several distinct improvements.

5.5.1.5. The service life of the LVTP5 track is directly dependent on the seal life at the present time. Laboratory testing of the torsion seals has shown that the life of the seal is adequate for the intended service if not acted on by destructive external forces. In turn, the factors which cause premature malfunction or total failure of the seals are definitely known to be:

A. The seal is displaced or cut during installation of the pin. Cutting occurs because the roll pin notch which passes through the seal has not been properly filled. It has also been noted that the taper lead on the connecting pin is often very rough. This condition will extrude and cut the second seal as the pin passes through. Even if the second seal is not cut it is displaced out of the counterbore to the limit of the clevis opening and thus becomes vulnerable.

B. The clevis wears and exposes the seal. Once the clevis has worn, the outer edges of the seal are worn off by the action of the clevis as well as by dirt.

C. The outer diameter of the seal in many cases does not seal effectively. This condition is due to the fact that the counter-bore which contains the outer diameter of the seal is so close to the surface of the track that no appreciable shedder action is provided. For this reason dirt is forced through this surface and enters the bearing area.

D. The oilite bearings, contained in the male clevis, migrate and destroy the seals. Movement of the bearings is caused by lack of stability in the outer clevis. The clevis arms in the LVTP5 track are not well supported. They will move out of line very easily unless they are supported by the pins. The result is that the pins continually move up and down in an arc set by the clevis arm during operation. The end movement of the pins will alternately load the bearings axially and cause movement. It is also believed that this torsion force applied to the pin, when combined with shear and bending forces which are always present, occasionally causes pin failure.

5.5.1.6. The track proposed for the LVTPX11 will reduce or eliminate the malfunctions just described as follows:

A. The roll pin notch will be used only in the outer end of the pin so that no notch will pass through the seal during assembly. The end taper will be proportioned to expand the seal gradually and a polished surface will be specified. The seal will be contained in a shedder so that it cannot be displaced axially during assembly. That is, as the pin passes through the first seal it will be backed up by the counterbore wall. As it passes through the second seal it will be backed up by the flange of the shedder which will have relatively close clearance at the pin diameter.

B. Clevis wear will not expose the seal because the shedder is now used for protection instead of the inner surface of the clevis. The shedder will be hardened to act as a thrust washer. The combination of a hardened surface against a medium hard surface will reduce total clevis wear considerably.

C. The outer surface of the seal will not be exposed to constant jamming by dirt because it will be protected by the shedder. Furthermore, the outer surface of the seals will be cemented to the shedder prior to installation in the track block.

D. The configuration of the track block will produce stability in the clevis area because the projection of the female clevis is held to a minimum. This condition will not only reduce the tendency of the bearing to move but will also result in easier insertion of the connecting pin, because the female clevis pin holes will remain in line as originally machined. To further reduce the possibility of bearing movement it is presently planned to grind the outside diameter of the bearing. The closer tolerances and unlubricated surface will effect increased holding power in the block. Grinding will close the pores of bearings but no flow through of grease is necessary in this application.

5.5.2. Function and Description.

5.5.2.1. The function of a track is to lay down a path for the roadwheels. To perform in a satisfactory manner the track must provide an aggressive bite in the soil so that it will not slip as the power which drives the vehicle is transmitted by means of track tension. The track should be able to proceed over improved surfaces without damage thereto. In addition, an amphibious track must include buckets or vanes for water propulsion.

5.5.2.2. The LVTPX11 track is of the center drive type. Buckets or vanes at the outer ends serve as water propulsion members. The buckets do double duty because the inner flanges serve as roadwheel guides. The general arrangement provides for a smooth broad surface on each side of the center drive lug which serves as a wheel path. The path is paved with rubber to provide a self-cleaning surface which will result in increased tire life. On soft terrain any penetration will cause the side buckets to bite into the soil for aggressive action.

5.5.2.3. A rubber pad is utilized on the ground side of the track to prevent damage to paved surfaces. Both a replaceable ground pad and a molded-in-place pad are shown on drawing SK-5211. Experience has shown that wear of a rubber pad is in proportion to wear of the metal surrounding the pad. Therefore service life of the molded-in-place pad should be satisfactory. Also design studies have shown that utilization of a replaceable pad will increase vehicle weight by 351 pounds and increase cost by approximately $598.00. (A table of weights is shown on drawing SK-5211 for ready reference). For these reasons the Design Agent strongly recommends the use of the molded-in-place ground pad. It should be noted that if the life of the molded rubber pads does not prove satisfactory during prototype evaluation, the bolted-on pad could then be incorporated with very minor changes to the basic track shoe.

5.5.3. General Arrangement.

5.5.3.1. The track assembly will consist of four major details. The basic block will have the road and ground pads molded in place to form a single stocked item. The seal and shedder will also be assembled in all shoes before delivery by the Vendor. However, it must be noted that the seal or the seal and shedder assembly will be replaceable; therefore, a special tool will be designed so that this repair may be readily accomplished in the field. The two remaining details are the connecting pin and a rubber-filled roll pin to lock the connecting pin in place.

5.5.3.2. Detail Construction. The track shoe will be sand cast steel. Material will be QQ-S-681 b 4C3 which is commonly used on all shoes of this type. Sprocket drive lugs, side guides, and ground contact areas will be locally flame hardened to reduce wear. Rubber compounding will be in accordance with MIL SPEC-T-11891B Type I. The shedder will be direct hardening alloy or plain carbon steel. Processing will be forging or direct screw machine operation whichever proves most economical during detail studies. The roll pins will be case hardened alloy steel as specified for the present LVTP5 tracks. Roll pins will also be the same rubber-filled standard pins used in the LVTP5 track.

5.6. Shrouds, Fenders and Turning Vanes. (Refer to drawing SK-5225).

5.6.1. Discussion.

5.6.1.1. For the purpose of this discussion "shroud" will be used to designate members which extend longitudinally along the sides of the hull. "Fender" will be used to designate transverse members at either end of the vehicle extending across the space between the hull and shrouds.

5.6.1.2. Shrouds are used to form a closed track return channel. During water operation they block off water from the sides so that the track runs in dead or aeriated water to produce the lowest possible track efficiency during track return. During land operation the shrouds act as dust shields, feeding dust out the rear of the vehicle rather than allowing it to billow over the entire vehicle. The majority of Ordnance Vehicles utilize removable or hinged metal shrouds. However several modern Ordnance Tracklayers have incorporated shrouds fabricated from rubberized duck belting. The object of the belt shrouds was to produce a flexible member that would readily deflect to allow passage of foreign objects which are occasionally dragged through the return channel. The belting would also absorb external contact without permanent distortion.

5.6.1.3. Investigation of the acceptability of rubber shrouds was directed towards experience with the M113 Ordnance Vehicle, and the LVTP6 Marine Corps Vehicle. Personnel contacted at the Detroit Tank Arsenal advised that the rubber shrouds on the M113 functioned as intended, and that service life was entirely satisfactory.

5.6.1.4. Conversely personnel from the Marine Corps Test and Experimental Unit at Camp Pendleton advised that the rubber shrouds were not satisfactory on the LVTP6. It was agreed that for land operation the LVTP6 shrouds were serviceable, which is in accordance with the information received from the Detroit Arsenal on the M113 which is predominantly a land vehicle. However, water operation with the LVTP6 has shown that the shrouds are not stable enough to confine the water flow to an acceptable degree without excessive metal bracing of the shrouds. Furthermore, it was necessary to add steel wear strips to the shrouds to prevent destruction by the tracks as they returned through the channel formed by the shrouds and the hull proper. For these reasons the Design Agent recommends metal shields for the LVTPX11 as will be discussed later in this section.

5.6.1.5. Fenders are usually curved to follow the track line as it passes over the sprocket and/or the idler. Rear fenders are relatively unimportant. However, the front fenders extend to some degree down over the front end of the channel formed by the shroud and hull. Thus during water operation they block off the water that tends to flow into the channel due to forward motion of the vehicle. The return flow is undesirable because it bucks the returning track and reduces overall track propulsion efficiency. Previous tests have shown that if the fenders are extended around the front idlers and/or sprockets downward to approximately 60 degrees forward from a vertical line projected through the front idler that, added track efficiency will result. Fenders are usually made a permanent part of the hull. However if the extension must completely wrap the idlers they must be retractable for land operation because they are destroyed by contact with stumps, rocks, or banks which the front idler contacts during normal operation.

5.6.1.6. The size, shape, or requirements for fenders and shrouds must be determined by scale model or full scale vehicle tests, since optimum size, configuration, or necessity will vary with hull configuration, type of track, and track return systems. In accordance model tests have been set up to determine the most efficient system for the LVTPX11. The results of model evaluation which appear in another section of this report will dictate final configuration of the shrouds and fenders.

5.6.1.7. At the present time model tests have not determined whether fenders will have any appreciable effect on speed or efficiency. In accordance detail design has been delayed pending final results. It should be noted that gains in this area must be appreciable to warrant the application of movable fenders which will be a continuing source of trouble because of their vulnerable location.

5.6.1.8. Paragraph 3.11.1 (c) of BuShips Contract Specification SHIPS-A-4159 states that "turning vanes" shall be considered as well as fenders and shrouds in endeavoring to meet the target speeds. Investigation of turning vanes has revealed that the vanes must be approximately five inches in depth to have any measurable effect. With a set overall vehicle width, the 10 inches required must be taken out of the center of the hull, or by reduced track widths. The Design Agent did not consider either of these reductions practical. Therefore no further investigation or evaluation was conducted on this subject.

5.6.2. General Arrangement - Shrouds.

The shroud consists of five major details arranged as shown on drawing SK-5225. The details consist of an idler cover which may be removed for servicing the idler assembly. Unbolting the forward shroud will expose the number 1 and number 2 suspension assemblies for servicing. Removal of the after shroud exposes the number 3 and number 4 suspension assemblies. The sprocket cover is removed to service the sprocket assembly. A transverse tube assembly is provided at the center and at each end of the shroud assembly to add stability to the installation. The tubes are bolted to the hull and to the outer plates for ease of replacement in case of damage.

5.6.3. Detail Construction - Shrouds.

All plates and cross tubes will be fabricated of aluminum. Material will be 5456 H 321 for ballistics purposes and for compatibility with hull materials. The forward and after shrouds will be turned in 45 degrees at the bottom edge to impart additional stiffness. All plates are overlapped at the vertical joints and are attached to each other by straps and bolts near the bottom edge, which is the only area not vulnerable to damage by track action. Attaching holes in the plates will be elongated to preclude the need for close tolerance spacing of the tapped holes in the hull.

5.7. Suspension System Computations and Stress Analysis.

Self-explanatory calculations covering all major components and details previously discussed are included in Appendices I and II immediately following this section.

APPENDIX I To Section 5

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 10/4/62 SUBJECT LVT PX11 SHEET NO. I OF
CHKD. BY DATE Index JOB NO. Nobs
Suspension Group 4561-3

Phase I

Preliminary Design Calculations

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/6/62 SUBJECT LVT PX11 Road Wheel SHEET NO. 11 OF
CHKD. BY DATE Design Parameters JOB NO. Nebs
Suspension Group 4561-3-3

A. Vehicle Characteristics

1. GVW = 35,000 lb.
2. Maximum speed = 40 mph
3. Maximum side slope = 60 percent
4. Minimum life = 1,000 hours
5. climatic operation = minus 40°F To plus 125°F
6. Maximum impact load = 8 G's*
7. Roadwheel diameter = 32 inches
8. Static road wheel load = 4375 lb

B. Operating Conditions (Road Wheel)

1. 2 hours at 5 MPH (53 RPM) at 8G* (bump loading)
2. 595 hours at 20 MPH (213 RPM) at static load plus 20 percent (normal cross country operation)
3. 198 hours at 12 MPH (128 RPM) at static load plus 50 percent (severe cross country operation)
4. 190 hours at 40 MPH (426 RPM) at static load plus 20 percent (highway operation).
5. 10 hours at 5 MPH (53 RPM) at static load plus 20 percent (60 percent side slope operation).
6. 5 hours at 5 MPH (53 RPM) at steer condition, static load plus 20 percent (level ground).

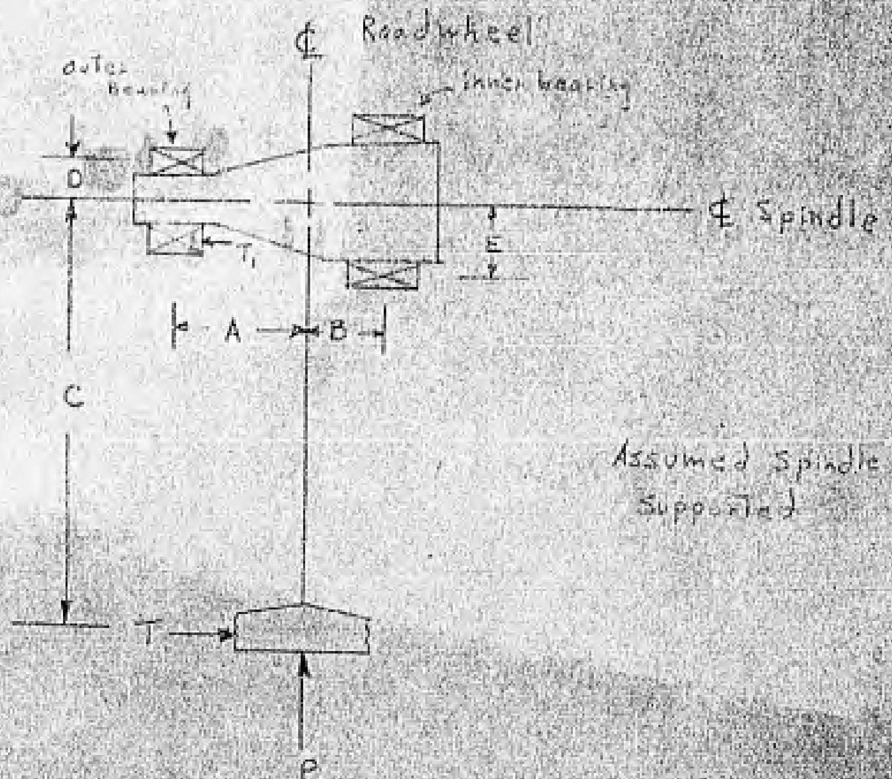
*(G = GVW = 35,000#)

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY J. Nelson DATE 9-6-62 SUBJECT LVT PX11 Roadwheel Bearing Analysis Suspension Group SHEET NO. 1 OF 69
CHKD. BY TH DATE JOB NO. Navy 1561-3-3

Roadwheel Bearing Analysis

Static load Per Roadwheel = 4375#



Assumed Spindle Simply Supported

A = 2 9/16" = 2.5625
B = 1 1/4" = 1.2500
C = 15 1/8" = 15.1250
P = Vertical load on Road wheel
T = Thrust load on Road wheel
R_1 = Radial Bearing Reaction
R_2 = " " "
E = 2 5/32 = 2.1563
D = 1 5/16 = 1.3125
T_1 = Thrust Reaction on Bearing

Assumed Slope [illegible] Roadwheel [illegible]
[illegible] Roadwheel.

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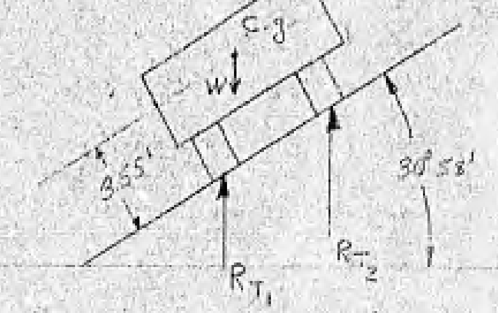
KALAMAZOO, MICHIGAN

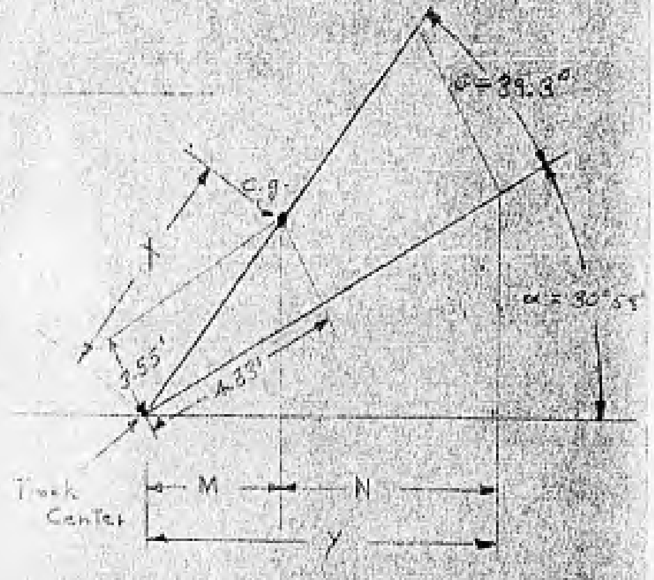
BY G Nelson DATE 9-6-62 SUBJECT LVTPX11 Roadwheel Bearing Analysis Suspension Group SHEET NO. 2 OF 60
CHKD. BY JH DATE JOB NO. Nobs 4561-3-3

Roadwheel Bearing Analysis Cont.

Roadwheel Forces on 60% Side Slope



Assumed Track Reactions $R_{T_1}+R_{T_2}$ act at center of Track



$W = 35{,}000^{\#}$

$\theta = \tan^{-1}\frac{3.55}{4.33} = \tan^{-1}(.819)$

$\theta = 39.3°$

$X(\cos\theta) = 4.33$

$X(\cos 39.3°) = 4.33'$

$X = \frac{4.33'}{\cos 39.3°}$

$X = \frac{4.33}{.775}$

$X = 5.60'$

$y = 2(4.33)(\cos\alpha)$

$y = 8.66(\cos 30°58')$

$y = 8.66(.858)$

$y = 7.43'$

$M = X(\cos\theta+\alpha) = 5.60(\cos 39°18'+30°58')$

$M = 5.60(\cos 70.27°) = 5.60(.338)$

$M = \underline{1.895}$

$N = y - M = 7.43' - 1.895 = \underline{5.535'}$

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BY G. Nelson	DATE 1-6-62	SUBJECT LVT PX11 Roadwheel Bearing Analysis Suspension Group	SHEET NO. 3 OF 68
CHKD. BY J.H.	DATE		JOB NO. Nebs 4561-3-3

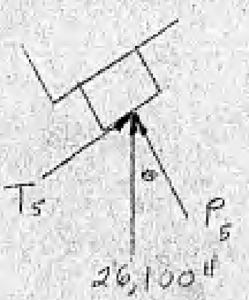
Roadwheel Bearing Analysis Cont.

Roadwheel Forces on 60% Side Slope (Cont.)

$$\circlearrowleft \Sigma M_{R_{T_2}} = 0 = R_{T_1}(M+N) - W(N)$$

$$R_{T_1} = \frac{W(N)}{M+N} = \frac{35{,}000(5.535)}{7.43}$$

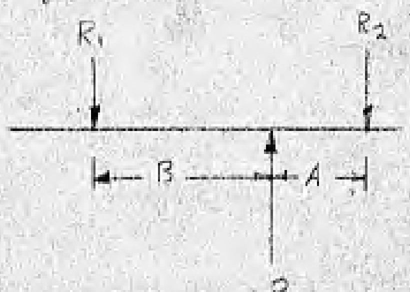
$$R_{T_1} = \underline{26{,}100^{\#}}$$



$$P_5 = 26{,}100^{\#}(\cos\theta) = 26{,}100^{\#}(\cos 30°58')$$
$$P_5 = 26{,}100^{\#}(.858)$$
$$P_5 = 22{,}400^{\#}$$ (Per Roadwheel = 22400/4 = $\underline{5600^{\#}}$)

$$T_5 = 26{,}100^{\#}(\sin\theta) = 26{,}100^{\#}(\sin 30°58')$$
$$T_5 = 26{,}100^{\#}(.514)$$
$$T_5 = 13{,}400^{\#}$$ (Per Roadwheel = 13,400/4 = $\underline{3350^{\#}}$)

Spindle Bearings; Radial Loads



$$\circlearrowleft \Sigma M_{R_2} = 0 = P(A) - R_1(A+B) \qquad \circlearrowleft \Sigma M_{R_1} = P(B) - R_2(A+B)$$

$$R_1 = \frac{P(A)}{(A+B)} = \frac{P(1.2500)}{3.8125} \qquad R_2 = \frac{P(B)}{(A+B)} = \frac{P(2.5625)}{3.8125}$$

$$R_1 = P(.328) \qquad R_2 = P(.673)$$

Spindle Bearings, Radial loads due To vehicle weight.

Cond No.	P – lbs.		R_2 – lbs	R_1 – lbs
B_1	P = 8 · 4375	= 35,000	23,560	11,440
B_2	P = 4375 (1.2)	= 5,250	3,540	1,710
B_3	P = 4375 (1.5)	= 6,560	4,420	2,140
B_4	P = 4375 (1.2)	= 5,250	3,540	1,710
B_5	P = 5600 (1.2)	= 6,725	4,525	2,200

Spindle Bearings, Thrust loads

T_s = Thrust on Track from steer

$$T_s = \frac{\text{GVW} \times \text{Coefficient of Shear}}{8}$$

Assume Coefficient of shear (Track against ground) To be .8

$$T_s = \frac{35,000\,(.8)}{8} = \underline{3,500^{\#}}$$

On 60% side slope under steer (B5)

$$T_{60} = T_5 + (P_5 \times .8)$$

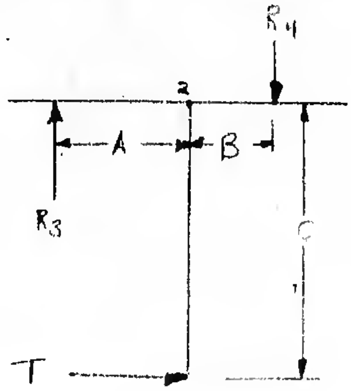
T_5 = Static side thrust on track due to 60% slope
P_5 = Normal weight on road wheel due to 60% slope

$$T_{60} = 3350 + (5600 \times .8) = 3350 + 4480.0$$

$$T_{60} = \underline{7830.0^{\#}}$$

Spindle Bearings

Radial loads due to Thrust loads



Assumed side Thrust (T) to be acting at a point 7/8" above bottom of roadwheel.

Steer Condition (level ground) (B6)

$$\circlearrowleft + \quad \Sigma M_a = 0 = T_s(C) - R_4(B) - R_3(A)$$

$$+\uparrow \quad \Sigma F_y = 0 = R_3 - R_4$$

$$\therefore R_3 = R_4$$

$$T_s(C) = R_4(B) + R_4(A)$$

$$R_4 = \frac{T_s(C)}{(A+B)}$$

$$R_4 = \frac{3{,}500\,(15.125)}{3.81}$$

$$R_4 = 13{,}900^{\#}$$

$$R_4 = R_3 = \underline{\underline{13{,}900^{\#}}}$$

Steer Condition (60% side slope) (B5)

$$\circlearrowleft + \quad \Sigma M_a = 0 = T_{60}(C) - R_4(B) - R_3(A)$$

$$\Sigma F_y = 0 = R_4 - R_3$$

$$R_4 = \frac{7830\,(15.12)}{3.81} = \underline{31{,}100^{\#}} = R_3$$

These Reactions obtained under 60% side slope steer condition (B5), are the maximum values obtained assuming no track slide in the downhill direction. However, with the vehicle on a 60% side slope, the vehicle will be approaching a sliding condition and it is therefore doubtful that the Max. value of thrust (T_{60}) will be obtained. Therefore an assumed value for T_{60} of 5000# will be used for the purpose of calculations. It must also be realized that this Maximum thrust force will only be present for an extremely short period of time and will fall off rapidly after reaching the Maximum value.

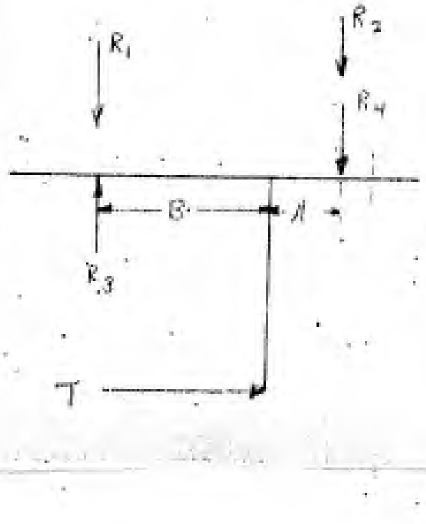
Steer Condition (60% Side Slope) Revised Calculations

$$\Sigma M_a = 0 = T_{60}(C) - R_4(B) - R_3(A)$$
$$\Sigma F_y = 0 = R_4 - R_3$$

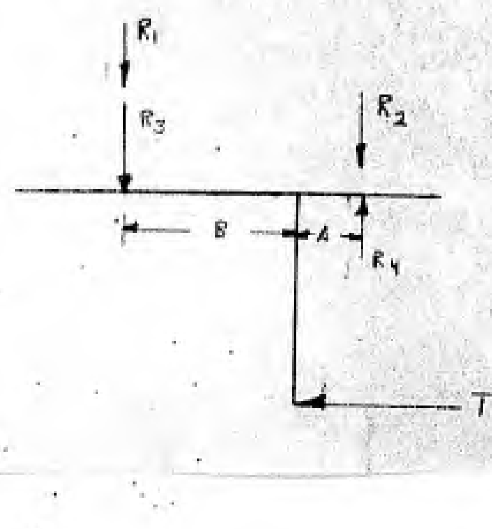
$$R_4 = \frac{5000(15.135)}{3.81} = \underline{\underline{19{,}800^{\#}}} = R_3$$

Radial Reactions on Bearings

Thrust Acting Inward



Thrust Acting Outward



Summary of Bearing loads

Thrust Acting Inward

	Inner Roller Bearing			Outer Thrust Bearing			
Cond No.	Radial (R_2) load-wt.	Radial (R_4) load-Thrust	Radial load Total - lbs	Radial (R_1) load-wt.	Radial (R_3) load-Thrust	Radial load Total-lbs.	Thrust load
B_1	23,560		23,560	11,440		11,440	
B_2	3,540		3,540	1,710		1,710	
B_3	4,420		4,420	2,140		2,140	
B_4	3,540		3,540	1,710		1,710	
B_5	4525	19,800	24,325	2200	-19,800	-12,600	5000#
B_6	3,540	13,900	17,440	1,710	-13,900	-12,190	3,500#

Thrust Acting Outward

	Inner Roller Bearing			Outer Thrust Bearing			
Cond No.	Radial (R_2) load-wt.	Radial (R_4) load-Thrust	Radial load Total - lbs	Radial (R_1) load-WT	Radial (R_3) load-Thrust	Radial load Total-lbs.	Thrust load
B_1	23,560		23,560	11,440		11,440	
B_2	3,540		3,540	1,710		1,710	
B_3	4,420		4,420	2,140		2,140	
B_4	3,540		3,540	1,710		1,710	
B_5	4,525	-19,800	-15,275	2,200	19,800	22,000	5000#
B_6	3,540	-13,900	-10,360	1,710	13,900	15,610	3,500#

Outer Thrust Bearing life Calculations

Bearing Make + No. - SKF 22308C
Bore - 1.5748
O.D. - 3.5433
Width - 1.2992

From SKF Catalog:

Life is determined from table after calculating Load Ratio

Load Ratio $\frac{C}{P} = \frac{\text{basic dynamic load rating (from Cat. table)}}{\text{equivalent load (From formula below)}}$

$P = XVF_r + YF_a$

where $X =$ a radial factor, X_1 or X_2 (from Cat. Table)
$V =$ rotation factor $= 1.2$
$Y =$ a thrust factor, Y_1 or Y_2 (from Cat. Table)
$F_r =$ Calculated radial load
$F_a =$ calculated thrust load

Condition B6, Critical (Thrust Acting Outward)

$\frac{F_a}{VF_r} = \frac{3{,}500}{1(15{,}610)} = .222 \qquad e = .38$

$\therefore$ Use $X_1 = 1$ + $Y_1 = 1.78$

$P = XVF_r + YF_a = 1(1.2)(15{,}610) + 1.78(3{,}500)$
$P = 15{,}610 + 6{,}230$
$P = 21{,}840^{\#}$
$C = 20{,}400^{\#}$

$C/P = 20{,}400 / 21{,}840 = .934$
R.P.M. = 53
Life from table on page 7 of SKF Catalog = 200 hours

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G.Nelson DATE 9/10/62 SUBJECT LVT PX11 Roadwheel Bearing Analysis Suspension Group SHEET NO. 9 OF 69
CHKD. BY TH DATE ____ JOB NO. Nobs 4561-3-3

Cond No.	P	C/P	RPM	Life - hrs.
B_1	11,440 (1.2)	1.485	53	1100
B_2	1,710 (1.2)	9.94	213	60,000
B_3	2,140 (1.2)	2.94	128	130,000
B_4	1,710 (1.2)	9.94	426	80,000
B_5	32,630	.625	53	100
B_6	21,840	.934	53	200

Weighted Life

$$L_t = \frac{1}{\frac{S_1}{LP_1} + \frac{S_2}{LP_2} + \frac{S_3}{LP_3} + \frac{S_4}{LP_4} + \frac{S_5}{LP_5} + \frac{S_6}{LP_6}}$$

Where L_t = Expected life
S_i = Required life in hours expressed as a percentage of total required time.
LP_i = Calculated life in hours for each condition

$$L_t = \frac{1}{\frac{.002}{1100} + \frac{.595}{60,000} + \frac{.198}{130,000} + \frac{.190}{80,000} + \frac{.01}{100} + \frac{.005}{200}}$$

$$L_t = \frac{1}{(1.82 + 9.94 + 1.52 + 2.37 + 100.0 + 25.0) \times 10^{-6}}$$

$$L_t = \frac{10^6}{140.65} = \underline{6,840 \text{ hours}}$$

Inner Roller Bearing

Following Data is from RBC Catalog.

$$\frac{C'}{P'} = \frac{C \times a \times b}{P \times c \times d} =$$

Where C' = modified basic dynamic load rating
P' modified imposed load

C = published dynamic load rating = 22,900*
P = calculated imposed load
a = shaft hardness factor = 1.0
b = race rotation factor = .25
c = type of load factor = 1.0
d = design practice factor = 1.0

Bearing Make + No. - RBC - SJ9717
Bore - 3.5
O.D. - 5.125
Width - 1.750

Condition No. B-5
Thrust Acting Inward

$$\frac{C'}{P'} = \frac{22{,}900(1)(.25)}{24{,}325(1)(1)} = \frac{12{,}150}{24{,}325} = .706$$

Cond No.	P	C'/P'	RPM	Life - hours
B_1	23,560	.728	53	120
B_2	3,540	4.84	213	14,000
B_3	4,420	3.89	128	12,500
B_4	3,540	4.84	426	2,400
B_5	24,325	.706	53	100
B_6	17,410	.983	53	250

$$L_T = \frac{1}{\frac{.001}{120} + \frac{.595}{14{,}000} + \frac{.198}{12{,}500} + \frac{.190}{2{,}400} + \frac{.01}{100} + \frac{.005}{250}}$$

$$L_T = \frac{1}{(8.33 + 42.4 + 15.82 + 25.7 + 100 + 20) \times 10^{-6}}$$

$$L_T = \frac{10^6}{212.25} = \underline{4{,}710 \text{ hours}}$$

Corrections for Redesign

Dimensions $A = 2.5''$
$B = 1.0''$

Corrections for Radial loads due to Vehicle weight

$R_1 = P\left(\frac{1.0}{3.5}\right) = P(.286)$

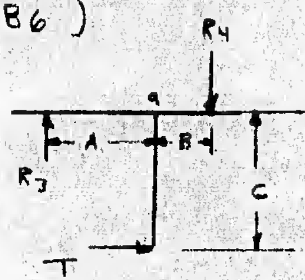
For values of P see sheet 4

$R_2 = P\left(\frac{2.5}{3.5}\right) = P(.715)$

Cond No.	R_2 - lbs	R_1 - lbs
B_1	25,000	10,000
B_2	3,760	1,490
B_3	4,690	1,870
B_4	3,760	1490
B_5	4,810	1915

Corrections for Radial loads due to Thrust loads

Steer condition level ground (Cond B6)



$\Sigma M_a = 0 = T_s(C) - R_4(B) - R_3(A)$

$(+\uparrow)\ \Sigma F_y = 0 = R_3 - R_4$

$\therefore R_3 = R_4$

$T_s(C) = R_4(B) + R_4(A)$

$T_s(C) = R_4(A+B)$

$R_4 = \frac{T_s(C)}{(A+B)}$

$T_s = 3,500^{\#}$
$A = 2.5''$
$B = 1.0''$
$C = 15.125''$

$R_4 = \frac{3500(15.125)}{(3.5)}$

$R_4 = \underline{15,125}^{\#}$

$R_3 = R_4 = \underline{15,125}^{\#}$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY C. Nelson DATE ____ SUBJECT LVT PXII Roadwheel Bearing Analysis Suspension Group SHEET NO. 12 OF 69
CHKD. BY TH DATE ____ JOB NO. Nebs — 4521-3-3

Steer Condition (60% side slope) (Cond B5)

5000# Thrust

$$R_1 = \frac{T_5(c)}{(A+B)} = \frac{5000(15.125)}{3.5} = \underline{\underline{21,600^{\#}}}$$

$$R_3 = R_4 = \underline{\underline{21,600^{\#}}}$$

Summary of Corrected Bearing loads

Thrust Acting Inward

Cond No.	Inner Roller Bearing			Outer Thrust Bearing			
	R2 - lbs	R4 - lbs	Radial Total - lbs	R1 - lbs	R3 - lbs	Radial Total - lbs	Thrust load #
B1	25,000	0	25,000	10,000		10,000	
B2	3760	0	3,760	1,490		1,490	
B3	4690	0	4,690	1,870		1,870	
B4	3760	0	3,760	1,490		1,490	
B5	4810	21,600	26,410	1,915	-21,600	-19,685	5000
B6	3760	15,125	18,885	1,490	-15,125	-13,635	3,500

Thrust Acting Outward

B5	4,810	-21,600	-16,790	1,915	21,600	23,515	5000
B6	3,760	-15,125	-11,365	1,490	15,125	16,615	3,500

Outer Thrust Bearing life Calculations

Bearing Make + No - SKF 22211C
Bore - 2.1654
O.D. - 3.9370
Width - .9843
Fillet Radius - .059

Condition B-5 (Thrust Acting Outward) Sample Calculation

$$\frac{F_a}{VF_r} = \frac{5000}{23{,}515} = .213 \qquad e = .24$$

∴ Use $X_1 = 1.0$ + $Y_1 = 2.87$
Race Rotation Factor $V = 1.2$
$P = XVF_r + YF_a = (1)(1.2)(23515) + 2.87(5000)$
$P = 28{,}200 + 14{,}350$
$P = 42{,}550$
$C = 18{,}300$
$C/P = 18{,}300/42{,}550 = .43$
R.P.M. = 53
Life = 50 hours (Approximate)

Cond No.	P	C/P	R.P.M.	Life hrs
B_1	10,000(1.2)	1.5	53	1050
B_2	1,190(1.2)	10.02	213	75,000
B_3	1,770(1.2)	8.15	128	70,000
B_4	1,490(1.2)	10.02	426	39,000
B_5	42,550	.43	53	50
B_6	34,300	.53	53	70

$$L_t = \frac{1}{\frac{.001}{1050} + \frac{.895}{75{,}000} + \frac{.118}{70{,}000} + \frac{.190}{39{,}000} + \frac{.01}{.50} \; \frac{.005}{70}}$$

$$L_t = \frac{1}{(.952 + 2.94 + 2.83 + 4.88 + 200.0 + 71.4)\,10^{-6}}$$

$$L_t = \frac{10^6}{\quad} = 3490 \text{ hours}$$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 10/3/62 SUBJECT LVT PXII Roadwheel Bearing Analysis Suspension Group SHEET NO. 14 OF 69
CHKD. BY TH DATE JOB NO. Nebs 4561-3-3

Inner Roller Bearing Life Calculations

Bearing Make + No. – RBC – SJ6769
Bore – 3.8750
O.D. – 6.00
Width – 2.25
$c' = 35{,}450^{\#}$ Race Rotation Factor = .75
$c' = 35{,}450(.75) = 26{,}600^{\#}$

Cond No.	P	c'/p'	R.P.M.	Life-hours
B_1	25,000	1.06	53	360
B_2	3,760	7.07	213	28,000
B_3	4,690	5.67	128	24,000
B_4	3,760	7.07	426	14,000
B_5	26,410	1.005	53	310
B_6	18,885	1.41	53	870

For Sample Calculations see sheet No. 10

Weighted life

$$L_t = \frac{1}{\frac{.02}{360} + \frac{.595}{28{,}000} + \frac{.11}{24{,}000} + \frac{.190}{14{,}000} + \frac{.01}{310} + \frac{.005}{870}}$$

$$L_t = \frac{1}{(5.56 + 21.22 + 8.25 + 13.55 + 32.3 + 5.75)\times 10^{-6}}$$

$$L_t = \frac{10^6}{86.63}$$

$L_t =$ 11,520 hrs.

The above calculations show that the outer thrust bearing and inner roller bearing will be adequate for the worst conditions encountered under steering operation, and all other established design conditions.

INGERSOLL KALAMAZOO DIVISION
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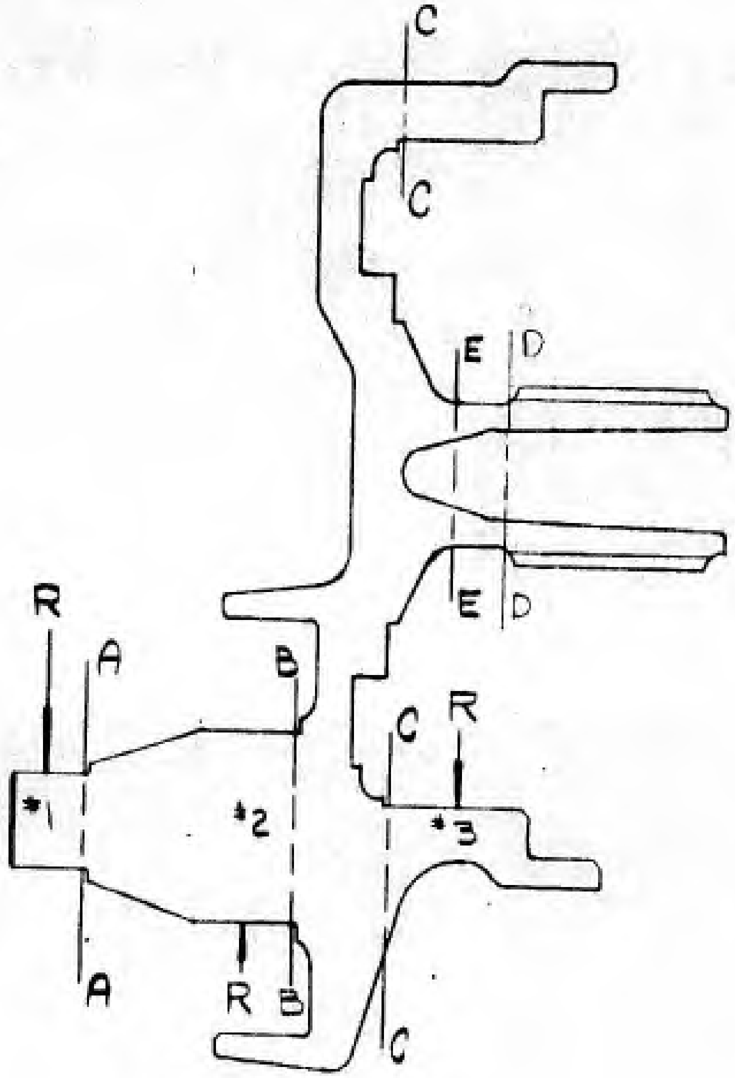
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/20/62 SUBJECT LVT PXII Road wheel Spindle - Diagram Suspension Group SHEET NO. 16 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4561-3-3



INGERSOLL KALAMAZOO DIVISION
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KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/11/62 SUBJECT LVT PX II Spindle Stress Analysis Suspension Group
CHKD. BY TH DATE
SHEET NO. 17 OF 63
JOB NO. Nobs HS61-3-3

Section A-A Initial Design

Assume yield stress of 27,000 psi equal to Max Bending stress, also assume stress concentration factor of 1.5

$$S_B = \frac{S_{B(Max)}}{1.5} = \frac{27,000}{1.5} = 18,000 \text{ psi}$$

$$S_B = \frac{MC}{I} \text{ where } C = \frac{D}{2} + I = \frac{\pi D^4}{64}$$

$$\therefore S_B = \frac{MD}{\frac{\pi D^4 (2)}{64}} = \frac{M64}{\pi D^3 (2)} = \frac{32M}{\pi D^3} = \frac{32(22000)(.6563)}{\pi D^3}$$

$$D^3 = \frac{32(22000)(.6563)}{\pi(18,000)} = 8.17$$

D = 2.18 in. = (Required Diameter)

∴ Use SKF Bearing 22311C

Bore = 2.1654
OD = 4.7244
Width = 1.6929
fillet radius = .079 in.

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson	DATE 9/10/62	SUBJECT LVT PX II Spindle	SHEET NO. 18 OF 69
CHKD. BY	DATE	Stress Analysis	JOB NO. Nebs
		Suspension Group	4561-3-3

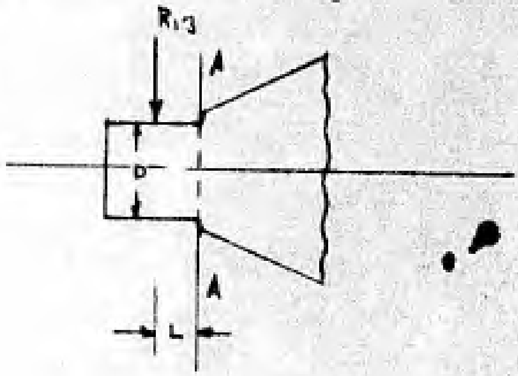
Section A-A Initial Design

(Assume Pure Bending)

$R_{13} = 22{,}000^{\#}$

$D = 2.1654$

$L = .846$ in.



$$S_B = \frac{Mc}{I} = \frac{M}{Z} = \frac{32M}{\pi (D^3)}$$

$$S_B = \frac{22{,}000(.846)(32)}{\pi(2.1654)^3}$$

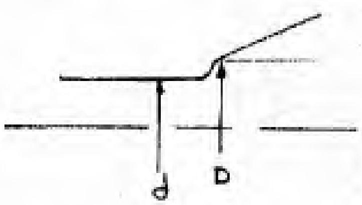
$$S_B = \frac{22{,}000(.846)(32)}{\pi(8.4)} = 22{,}600 \text{ psi}$$

Stress Concentration Factor - Ductile Mat'l, Variable load

(Reference - R.E. Peterson "Stress Concentration Design Factors")

$d = 2.1654$

$D = 2.56$



$$\frac{R}{d} = \frac{.079}{2.1654} = .0364 \qquad \frac{D}{d} = \frac{2.56}{2.165} = 1.18$$

$\therefore k_t = 2.05$

$S_{B(Max)} = S_B(k_t) = 22{,}000(2.05)$

$S_{B(Max)} =$ <u>45,000 psi</u>

Stress is too high, therefore larger section will be needed.

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 10/3/62 SUBJECT LVTPX11 Spindle Stress Analysis Suspension Group
CHKD. BY TH DATE
SHEET NO. 19 OF 69
JOB NO. Nabs 4561-3-3

Revised Design - Section A-A

Material Properties - Aluminum 356-T6

$S_{ult} = 38{,}000$ psi
$S_y = 22{,}000$ psi
$S_{ys} = .6(22{,}000) = 16{,}200$ psi
(Used Through out spindle)

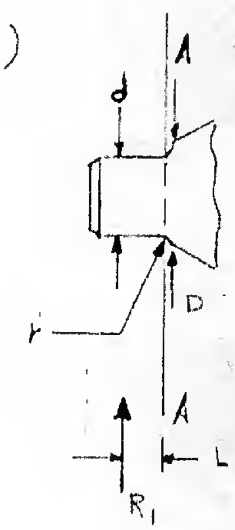
Section A-A (Revised Section)

$S_B = \frac{M}{Z} = \frac{R_1 L}{Z}$

$S_B = \frac{R_1 (L)(32)}{\pi D^3}$

$S_B = \frac{23{,}515(.492)(32)}{\pi(15.6)}$

$S_B = \underline{7{,}540 \text{ psi}}$



$D^3 = 15.6"$
$D = 2.5"$
$d = 2.16$
$r = .059$
$R_1 = 23{,}515$#
$L = .492"$

Cond B-5
Total Radial load $= 23{,}515 = R_1$

$\frac{r}{d} = \frac{.059}{2.16} = .027 \qquad \frac{D}{d} = \frac{2.5}{2.16} = 1.158$

K_t (Bending) $= 2.2$ (Stress Concentration Factor - Ductile Mat'l Variable load)

S_B (Max) $= 7{,}540(2.2) = \underline{16{,}600 \text{ psi}}$ (Bending)

$S_s = \frac{P}{A} = \frac{16{,}410(4)}{\pi D^2} = \frac{16{,}410(4)}{\pi(6.25)} = \underline{3450 \text{ psi}}$ (Shearing)

S.F. (Simple Tension) $= \frac{22{,}000}{16{,}600} = 1.62$ (S.F. in Tension)

S.F. $= \frac{16{,}200}{3450} = 4.71$ (S.F. in Shear)

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BY G. Nelson	DATE 9/10/62	SUBJECT LVT PX II Spindle	SHEET NO. 20 OF 69
CHKD. BY TH	DATE	Stress Analysis	JOB NO. Nobs
		Suspension Group	4561-3-3

Section B-B, Initial Design

Assume Thrust Force acting on center line of spindle as an axial load of 5000#
Therefore Assume Spindle end as a Stiff Bar where Resultant Stress = Direct Stress + Bending Stress

$$S_{R(Max)} = \frac{P}{A} + \frac{Mc}{I}$$

(This will be a compressive stress, for the Max. Stress condition)

$$\frac{P}{A} = \frac{5000^{\#}}{\frac{\pi(3.5)^2}{4}} = \frac{20,000}{\pi(12.25)} = 521 \text{ psi}$$

$$\frac{Mc}{I} = \frac{31,760(2.125)(1.75)}{7.36}$$

$$I = \frac{\pi D^4}{64} = \frac{\pi(3.5)^4}{64}$$

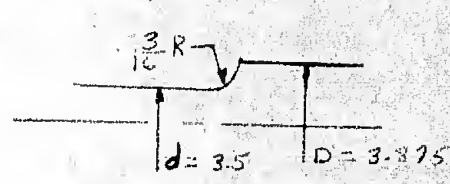
$$I = \frac{\pi(150)}{64} = 7.36 \text{ in}^4$$

$$\frac{Mc}{I} = 16,150 \text{ psi}$$

Stress Concentration Factor
Ductile Mat'l, Variable load

$$\frac{R}{d} = \frac{.100}{3.5} = .0286$$

$$\frac{D}{d} = \frac{3.875}{3.5} = 1.105$$



$$k_t(\text{Bending}) = 2.15$$

$$S_{B(Max)} = S_B(k_t)\ 16,150(2.15) = 34,700 \text{ psi}$$

$$S_{R(Max)} = 521 \text{ psi} + 34,700 \text{ psi} = \underline{35,221 \text{ psi}} \text{ (Compression)}$$

Revised Design

Section B-B (Revised Section)

$$S_y = k_t \frac{M}{Z}$$

$S_y = 27,000$ assume $k_t = 2$ $M = 35,000(2.125)$

$$27,000 = \frac{2(35,000)(2.125)}{Z}$$

$$Z = 5.51$$

$$Z = \frac{\pi D^3}{32}$$

$$D^3 = \frac{32 Z}{\pi} = \frac{32(5.51)}{\pi}$$

$$D^3 = 56.2$$
$$D = 3.84$$

Use Bearing SJ 6769 ID = 3.875 R = .10

$$S_B = \frac{M}{Z} = \frac{35,000(2.125)(32)}{\pi(D^3)} = \frac{35,000(2.125)(32)}{\pi(58)}$$

$S_B = 13,050$ psi

$$\frac{R}{d} = \frac{.100}{3.875} = .0258 \qquad \frac{D}{d} = \frac{4.1875}{3.875} = 1.08$$

$k_t = 2.17$ (Stress Concentration factor)

$$S_{B(Max)} = 13,050(2.17) = \underline{28,300} \text{ psi}$$
$$S_{R(Max)} = S_{B(Max)} + P/A = 28,300 + 575 = \underline{28,875 \text{ psi}}$$

This resultant stress ($S_{R(Max)}$) is slightly above the 27,000 psi yield stress but because of the infrequency of this load application it is considered satisfactory.

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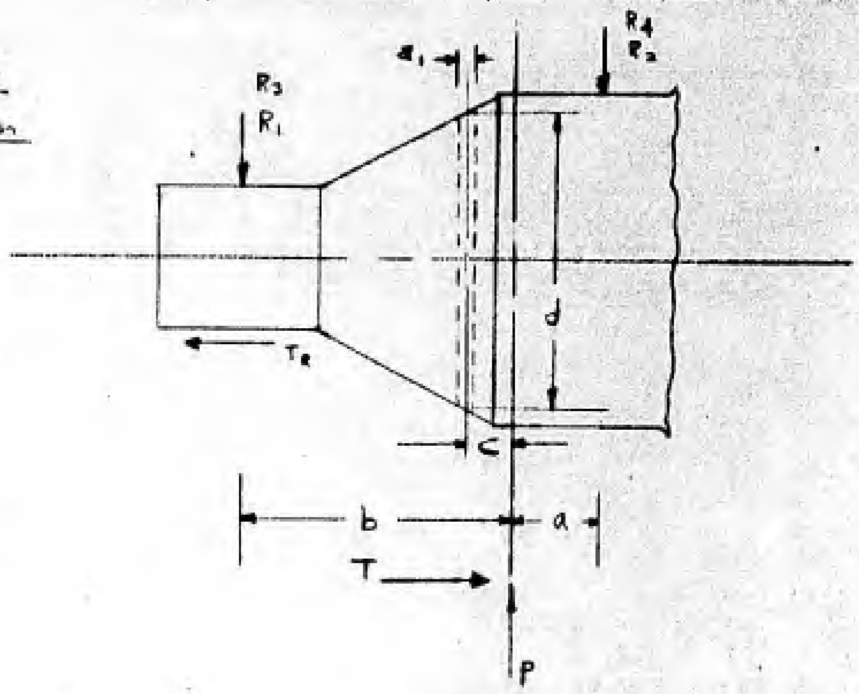
BY G. Nelson DATE 10/26/62 SUBJECT LVTPXII Spindle Stress Analysis Suspension Group SHEET NO. 22 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4561-3-3

Spindle stresses at Hole section



$a = 1"$
$b = 2.5"$
$C = .375'$
$P = 35,000^{\#}$
$T = 5,000^{\#}$
$R_{13}(Max) = 23,515^{\#}$
$R_{24}(Max) = 26,410^{\#}$
$d = 3.75"$

$$M_{(Max)} = R_1(b-c)$$
$$= 23,515\,(2.5-.375)$$
$$= 23,515\,(2.125)$$
$$= 50,000 \text{ in.-lbs.}$$

$$z = \frac{\pi d^3}{32} = \frac{\pi(3.75)^3}{32} = 5.176 \text{ in}^3$$

$$S = \frac{Mc}{I} = \frac{M}{z} = \frac{50,000}{5.176} = 9,680 \text{ psi}$$

$$\frac{a_1}{d} = \frac{.25}{3.75} = .067$$

$$S_{Max} = k_T S + S_D$$

$\therefore k_t = 2.38$
(Stress concentration Factor)

Ref. R.E. Peterson "Stress Concentration Design Factors"

$$S_{max} = 2.38\,(9,680) + \frac{T}{A}$$

$$S_{max} = 23,000 + \frac{5000(4)}{\pi(d)^2}$$

$$S_{max} = 23,000 + 451$$

$S_{max} = \underline{23,451 \text{ psi}}$ (Bending)

$$S_s = \frac{P}{A}(k_t) = \frac{35,000}{11.2}(2.5)$$

$S_s = \underline{7890 \text{ psi}}$ (Shear)

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BY G. Nelson DATE 9/11/62 SUBJECT LVT PX11 Spindle Stress Analysis Suspension Group SHEET NO 23 OF 69
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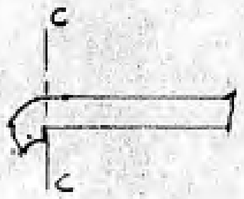
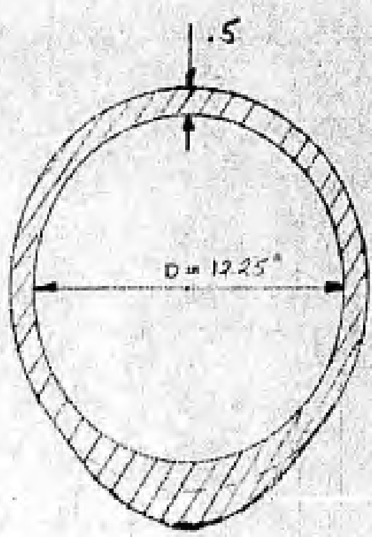
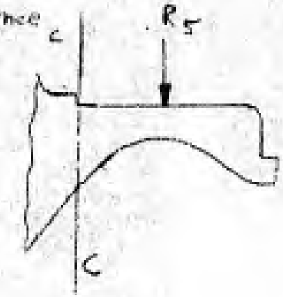
Section C-C





Assume Shear at Section C-C. Assume an effective area provided by 1/3 of the circumference to be taking the load.



D - To Center of Area
$D = 12.25 + \frac{.5}{2} = 12.5$

Condition B_1 Critical

R_5 (Acting Down) = 35,000 lb

$S_s = \frac{F}{A}$ where $A = \left(\frac{\pi D}{3}\right) t = \frac{\pi (12.50)(.5)}{3} = 6.55 \text{ in}^2$

Mat'l Properties
$S_{su} = 26,000$
$S_{s_{All}} = 26,000(.6) = 15,600$

Assume irregular area adds one (1) square inch to effective area.

$A = 7.55 \text{ in}^2$

$S_s = \frac{35,000}{7.55} = \underline{4,630 \text{ psi}}$

$S.F. = \frac{15,600}{4630} = \underline{3.37}$

Section D-D

Assume Pure Torsion, occurring only during lockup, at section D-D

$$S_s = \frac{Tr}{J} \qquad Z = \frac{J}{r} = \frac{\pi(D^4 - d^4)}{16D}$$

$d = 2.75''$
$D = 3.875''$
$T = 48{,}200\#''$
(Preliminary Torque Value)

$$Z = \frac{\pi(3.875^{4} - 2.75^{4})}{16(3.875)}$$

$$Z = \frac{\pi(225 - 57.2)}{16(3.875)} = \frac{\pi(167.8)}{16(3.875)} = 8.5 \text{ in}^3$$

$$S_s = \frac{T}{Z} = \frac{48{,}200\#''}{8.5 \text{ in}^3} = \underline{5670 \text{ psi}}$$

$$S_F = \frac{15{,}600}{5{,}670} = \underline{2.75}$$

Section E-E

Assume Pure Torsion, occurring during vehicle operation.

$T = 84{,}300\#''$
$d = 1\frac{5}{8}''$
$D = 3\frac{13}{16}''$

$$S_s = \frac{T}{Z} \qquad Z = \frac{\pi(D^4 - d^4)}{16D}$$

$$Z = \frac{\pi(3.875^{4} - 1.625^{4})}{16(3.875)}$$

$$Z = \frac{\pi(225 - 6.96)}{16(3.875)} = \frac{\pi(218.04)}{16(3.875)} = 11.05 \text{ in}^3$$

$$S_s = \frac{84{,}300}{11.05} = \underline{7{,}630 \text{ psi}}$$

$$S_F = \frac{15{,}600}{7{,}630} = \underline{2.04}$$

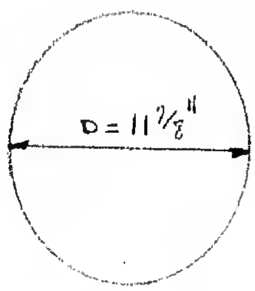
Plastic Bearing (Radial load only)

Reaction R_5, Condition B_1 Critical

$R_5 = 35{,}000^{\#}$

B.P = Bearing Pressure

$B.P._{(all)} = 3000^{\#}$



Length $= 2\frac{1}{16}" = 2.0625$

Projected Area $= DL = 11.875(2.0625)$

$A = 24.5 \text{ in}^2$

$$B.P. = \frac{R_5}{A} = \frac{35{,}000}{24.5} = \underline{1430 \text{ psi}}$$

$$S.F = \frac{3000}{1430} = \underline{2.10}$$

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KALAMAZOO, MICHIGAN

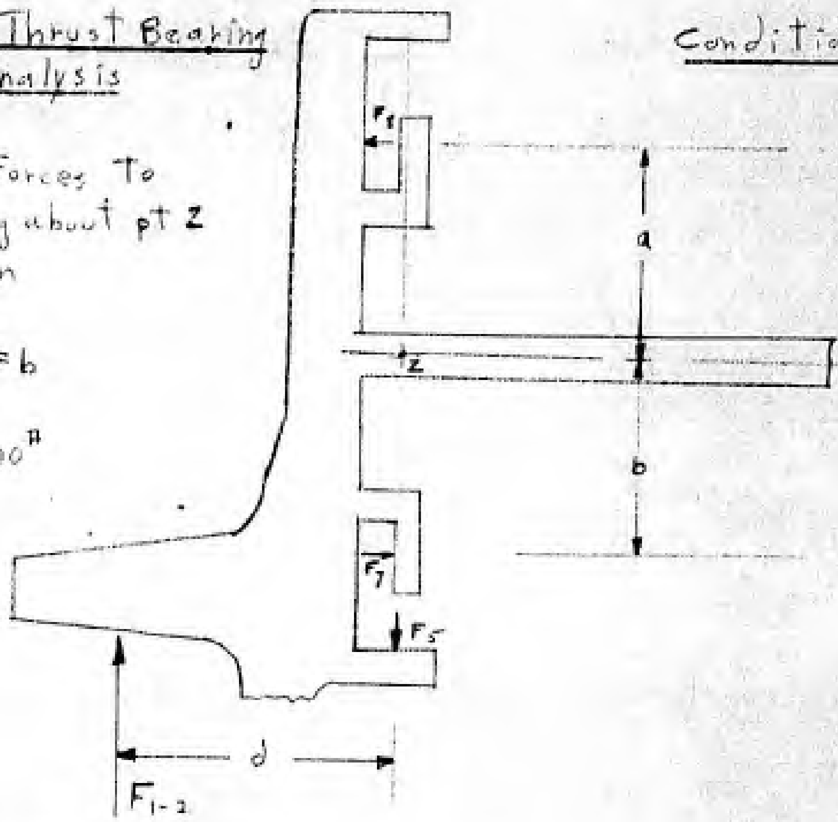
BY G. Nelson DATE 9/14/62 SUBJECT LVT PXII Roadwheel Bearing Analysis Suspension Group SHEET NO. 26 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4561-3-3

Plastic Thrust Bearing
Force Analysis

Condition B_1

Assume Forces To be acting about pt Z as shown

$a = 4.75'' = b$
$d = 4.125''$
$F_{1-2} = 35{,}000^{\#}$



$\xrightarrow{+} \sum F_x = 0 = F_7 - F_8$
$\therefore \quad F_7 = F_8$

$+\uparrow \sum F_y = 0 = F_{1-2} - F_5$
$F_{12} = F_5 = 35{,}000^{\#}$

$\curvearrowright \sum M_Z = 0 = F_{12}(d) - F_7(b) - F_8(a) = F_{12}(d) - 2F_7(a)$

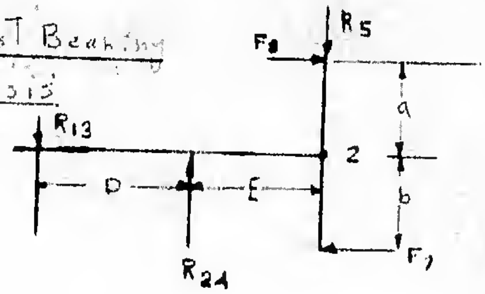
$$F_7 = \frac{F_{12}(d)}{(2a)} = \frac{35{,}000(4.125)}{9.5}$$

$F_7 = 15{,}200^{\#}$

$F_7 = F_8 = 15{,}200^{\#}$

Greatest Reactions occur under This condition, (B_1).

Plastic Thrust Bearing Force Analysis



Condition B5
Thrust Acting Outward

$R_{24} = 21,600$
$R_{13} = 23,515$
$D = 3\frac{3}{4}"$
$E = 2\frac{7}{8}"$
$a = b = 4.75"$

$+\uparrow \Sigma F_y = 0 = R_{13} - R_{24} + R_5$

$R_5 = R_{13} - R_{24} = 23,515 - 21,600 = 1915.0^{\#}$

$\Sigma M_z = 0 = R_{13}(D+E) - R_{24}(E) - F_7(b) - F_8(a)$

$$F_7 = \frac{23,515(6.625) - 21,600(2.875)}{9.5}$$

$F_7 = 16,400^{\#} - 6,540^{\#}$

$F_7 = 9,860^{\#}$

$F_7 = F_8$ $\underline{9,860^{\#}}$

(This condition is not critical)

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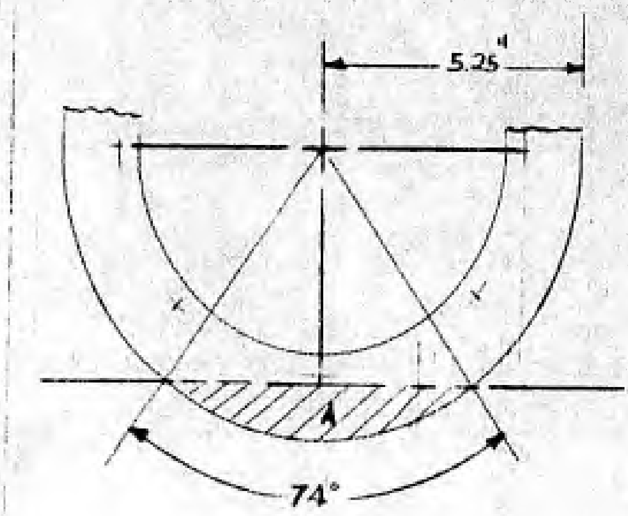
KALAMAZOO, MICHIGAN

BY E. Nelson DATE 9/19/62 SUBJECT LVT PX 11 Roadwheel Bearing Analysis Suspension Group SHEET NO. 28 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4560-3-3

Bearing Pressure - Plastic Thrust Bearing



Area A is assumed to be taking bearing load of 15,200#

From "Machinery's Handbook" The area of a circular segment with an arc of 74° and radius of 1" is .16514 in.^{2}

Area of this segment is equal to .16514 (r^2)

$A = .16514 (5.25)^2$

$A = .16514 (27.3)$

$A = 4.52$

$$B.P. = \frac{P}{A} = \frac{15,200^{\#}}{4.52} = 3370 \text{ psi}$$

These assumptions and calculations show the bearing to be marginal under this maximum loading condition. Allowable pressures for the bearing material should not greatly exceed 3000 psi, Therefore an increase in bearing width of .250" would be advisable. This width increase provides an arc of 84° and area constant of .23578 (from "Machinery's Handbook")

$$A = .23578 (5.25)^2 = .23578 (27.3) = 6.43 \text{ in.}^2$$

$$B.P. = \frac{P}{A} = \frac{15,200^{\#}}{6.57 \text{ in.}^2} = 2310 \text{ psi}$$

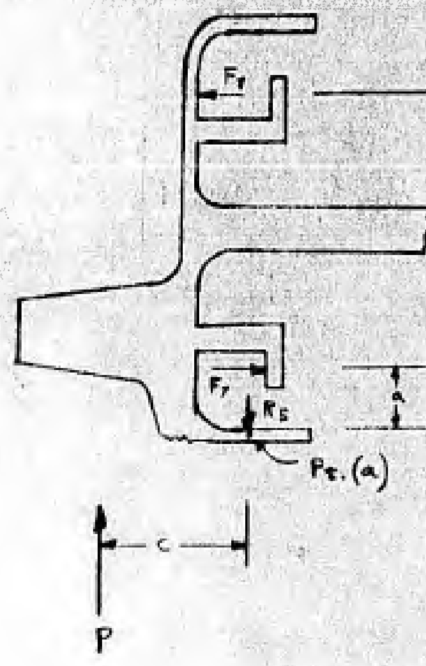
This bearing pressure will then be well within the allowable pressure of 3000 psi.

Thrust Bearing Retainer, Bolted Ring

Analysis of the forces acting in this region will now be approached in a manner differing from the analysis used on the thrust bearing; the object being, to determine if any appreciable difference exists in the magnitude of the reacting forces.

First Cond, Vertical Bump load.

$P = 35{,}000^{\#}$
$a = 1.25''$
$b = 10.75''$
$c = 4.125''$



Spindle is assumed to Rotate about Pt. a.

$\Sigma F_y = 0 = P - R_5$
$P = 35{,}000^{\#} = R_5$

$\Sigma F_x = 0 = F_7 - F_8$
$\therefore\ F_7 = F_8$

$(+)\ \Sigma M_a = P(c) - F_8(b) + F_7(a) = 0$

$F_7 = F_8$

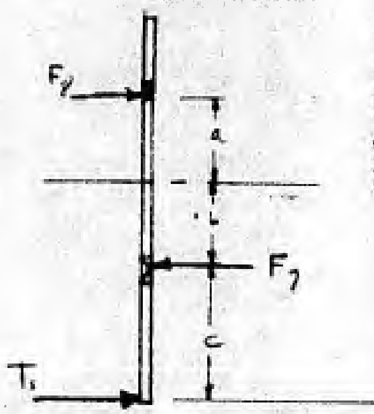
$\therefore\ P(c) = F_7(b) - F_7(a)$

$$F_7 = \frac{P(c)}{(b-a)} = \frac{35{,}000(4.125)}{9.5} = \underline{\underline{15{,}200^{\#}}}$$

This is the same reaction as obtained previously, which is correct, since the moment of a couple (F_7-F_8) should remain the same, regardless about which point it is taken if the distance between the forces remains the same.

Second Cond., Side Thrust load

This condition may be considered analogous to a shaft supported by two bearings, for the purpose of determining reactions.



$T_s = 5000^{\#}$
$a = 4.75''$
$b = 4.75''$
$c = 16.375''$

$\Sigma F_x = 0 = T_s - F_7 + F_8$

$\Sigma M_{F_7} = 0 = F_8(a+b) - T_s(c)$

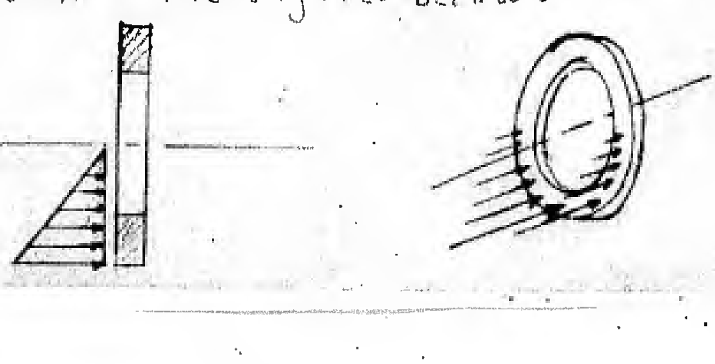
$F_8 = \dfrac{T_s(c)}{(a+b)} = \dfrac{5000(16.375)}{9.5}$

$F_8 = \underline{8610^{\#}}$

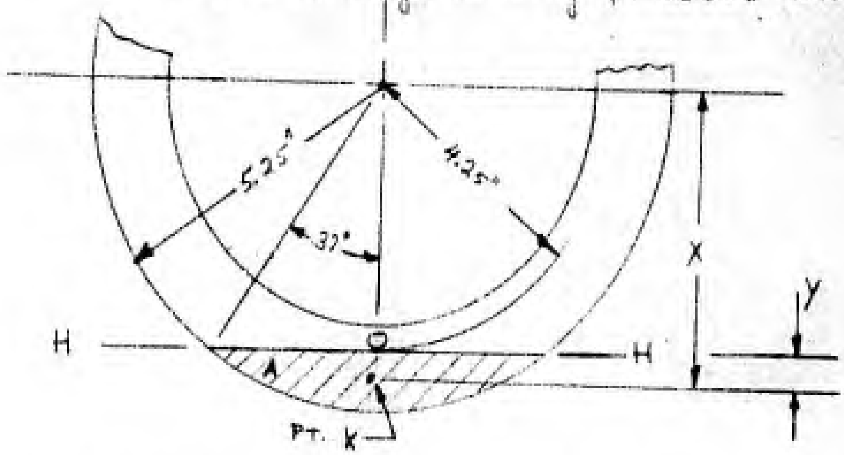
$T_s = F_7 - F_8$
$F_7 = T_s + F_8 = 5000 + 8610^{\#}$
$F_7 = \underline{\underline{13,610^{\#}}}$

Therefore the first condition provides the greatest reaction and will be used for determining the stresses in this region.

From these reactions the observation can be made, that the bearing thrust ring is loaded either on the top or bottom but not both. This load will probably be distributed over the ring in the form of a triangle as shown in the figures below.



For purposes of calculations the 15,200# reaction will be considered acting on the same area as used for determining bearing pressure on sheet 27.



The 15,200# reaction will be assumed acting on the area A at pt. K a distance y from section H-H, at which the section modulus (H-H) will be determined.

$X = \frac{2}{3} r^3 \frac{\sin^3\alpha}{A}$ where $A = \frac{1}{2} r^2 (2\alpha - \sin 2\alpha)$ (The first α being in radians)

and $\sin^3\alpha = 1/4 (3\sin\alpha - \sin 3\alpha)$

Reference - "The Engineer's Manual" by Ralph G. Hudson

$\alpha = 37^\circ = \frac{2\pi(37)}{360} = \underline{.646 \text{ radians}}$

$A = 1/2 (5.25)^2 \left[(2)(.646) - \sin 2(37^\circ)\right] = 1/2 (27.3) \left[1.292 - \sin 74^\circ\right]$

$A = 1/2 (27.3) \left[1.292 - .96\right] = 13.65 \left[.332\right] = \underline{4.58}$

$\sin^3\alpha = 1/4 \left[3 \sin 37^\circ - \sin 3(37^\circ)\right] = 1/4 \left[3(.603) - \sin 111^\circ\right]$

$\sin^3\alpha = 1/4 \left[1.809 - .934\right] = 1/4 \left[.875\right] = \underline{.218}$

$\therefore X = 2/3\, r^3 \frac{\sin^3\alpha}{A} = 2/3 (5.25)^3 \frac{.218}{4.58} = 2/3 (144)(.046) = \underline{\underline{4.42 \text{ inches}}}$

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KALAMAZOO, MICHIGAN

BY G. Nelson DATE 11/5/62 SUBJECT LVT PX 11 Roadwheel Stress Analysis Suspension Group SHEET NO. 32 OF 69
CHKD. BY ZH DATE JOB NO. Nabs 4561-3-3

Section Modulus at section H-H

$$Z = \frac{bh^2}{6}$$

$b_1 = \sin 37°(5.25)$
$b_1 = .602(5.25)$
$b_1 = 3.16$

$$Z = \frac{6.32(.375)^2}{6}$$

$$Z = \frac{6.32(.141)}{6}$$

$$Z = .1485$$

$b = 2b_1 = 2(3.16) = 6.32''$
$h = .375'' =$ Thickness

$$S_B = \frac{M}{Z} = \frac{P(y)}{Z}$$

where $y = X - 4.25 = 4.42 - 4.25$
$y = .17$

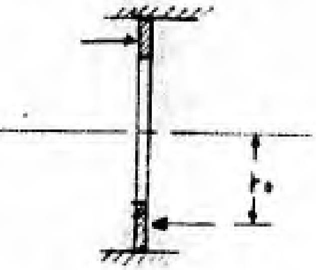
$$S_B = \frac{15{,}200(.17)}{.1485} = \underline{\underline{17{,}400 \text{ psi}}}$$

$S_{All} = 27{,}000$ psi $\quad \therefore$ Safety Factor $= \dfrac{27{,}000}{17{,}400} = \underline{1.52}$

Support Arm Thrust Bearing Supports

These supports can be said to be equivalent to a flat circular plate with a concentric circular hole, The plate outer edge fixed and the inner edge free.

Case 60 from Roark's "Formulas for stress + strain" will be used.



This plate is loaded with forces in the opposite direction rather than the same direction as in case 60. Therefore instead of finding an equivalent load applied at radius r_o, the maximum load of 15,200# will be used to account for any increased stress in the plate because of this loading.

Poisson's Ratio for Aluminum = .36 = ν

$$m = \frac{1}{\nu} = \frac{1}{.36} = 2.78$$

S_t = Stress at inner edge

S_r = Stress at outer edge

W = 15,200#

a =	5.31	a^2 =	28.2
b =	4.31	b^2 =	18.6
r_o =	4.81	r_o^2 =	23.2
t =	.375	t^2 =	.141

$$S_t = \frac{-3W}{4\pi m t^2}\left[(m+1)\left(2\log\frac{a}{r_o} + \frac{r_o^2}{a^2} - 1\right)\right] - \frac{6M}{t^2}\left[\frac{a^2(m-1)-b^2(m+1)}{a^2(m-1)+b^2(m+1)}\right]$$

$$M = \frac{W}{8\pi m}\left[(m+1)\left(2\log\frac{a}{r_o} + \frac{r_o^2}{a^2} - 1\right)\right]$$

$$S_r = \frac{3W}{2\pi t^2}\left[1 - \frac{r_o^2}{a^2}\right] + \frac{6mM}{t^2}\left[\frac{2b^2}{a^2(m-1)+b^2(m+1)}\right]$$

$$M = \frac{15,200}{8\pi(2.78)}\left[(2.78+1)\left(2\log\frac{5.31}{4.81} + \frac{23.2}{28.2} - 1\right)\right]$$

$$M = 218.0\left[3.78\left(2\log 1.105 + .824 - 1\right)\right]$$

$$M = 218.0\left[3.78\left(2(.095) + .824 - 1\right)\right] = 218.0\left[3.78(.19 + .824 - 1)\right]$$

$$M = 218.0\left[3.78(1.014 - 1)\right] = 218.0\left[3.78(.014)\right] = 218.0(.053)$$

$$M = 11.55$$

$$S_t = \frac{-3(15,200)}{4\pi(2.78)(.141)}\left[.053\right] - \frac{6(11.55)}{(.141)}\left[\frac{28.2(1.78) - 18.6(3.78)}{28.2(1.78) + 18.6(3.78)}\right]$$

$$S_t = -9280\left[.053\right] - 492\left[\frac{50.2 - 70.2}{50.2 + 70.2}\right]$$

$$S_t = -492 - 492\left[\frac{-20}{120.2}\right] = -492 - 492\left[-.167\right]$$

$$S_t = -492 + 82 = -410 \text{ psi}$$

$$S_t = \underline{410 \text{ psi}}$$

$$S_r = \frac{3(15,200)}{2\pi(.141)}\left[1 - .824\right] + \frac{6(2.78)(11.55)}{(.141)}\left[\frac{2(18.6)}{28.2(1.78) + 18.6(3.78)}\right]$$

$$S_r = 51,400\left[.175\right] + 1368\left[\frac{37.2}{50.2 + 70.2}\right]$$

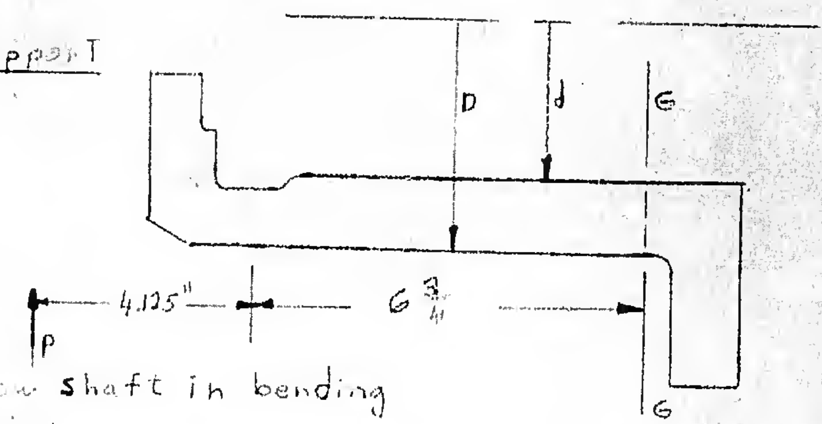
$$S_r = 9000 + 1368\left[\frac{37.2}{120.4}\right]$$

$$S_r = 9000 + 1368\left[.309\right] = 9000 + 422$$

$$S_r = \underline{9,422 \text{ psi}}$$

∴ This piece is satisfactory.

Spindle Support



$d = 10\frac{3}{4}$

$D = 12\frac{1}{8}$

$P = 35{,}000$

Assume hollow shaft in bending and Torsion

$$S_B = \frac{Mc}{I} \qquad \frac{I}{c} = Z = \frac{\pi(D^4 - d^4)}{32D} = \frac{\pi(12.125^4 - 10.75^4)}{32(12.125)}$$

$$Z = \frac{\pi(21{,}600 - 13{,}350)}{32(12.125)}$$

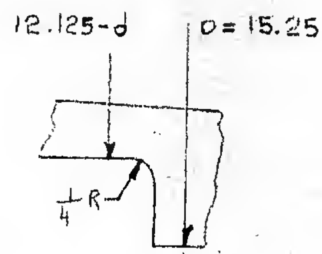
Section G-G

$$Z = \frac{\pi(8{,}250)}{32(12.125)} = 66.70\ in^3$$

$$S_B = \frac{M}{Z} = \frac{35{,}000(10.875)}{66.70} = \underline{5700\ psi}$$

$$S_s = \frac{Tr}{J} = \frac{T}{\frac{\pi(D^4 - d^4)}{16D}} = \frac{T}{132.4} = \frac{84{,}300}{132.4} = \underline{637\ psi}$$

$S_s = \underline{637\ psi}$ (Shearing Due to Torque)



Stress Concentration Factors

$$\frac{r}{d} = \frac{.25}{12.125} = .0206 \qquad \frac{D}{d} = \frac{15.25}{12.125} = 1.26$$

K_t (Bending) = 2.5

K_t (Torsion) = 2.10

S_B (Max) = 5700 (2.50) = $\underline{14{,}250\ psi}$

S_s (Max) = 637 (2.10) = $\underline{1338\ psi}$

Combined Stress

$$S_t = \frac{S_{B_M}}{2} + \sqrt{\left(\frac{S_{B_M}}{2}\right)^2 + \left(S_{S_M}\right)^2} \qquad S_s = \sqrt{\left(\frac{S_{B(M)}}{2}\right)^2 + \left(S_{S(M)}\right)^2}$$

$$S_t = \frac{14,250}{2} + \sqrt{\left(\frac{14,250}{2}\right)^2 + (1338)^2}$$

$$S_t = 7,125 + \sqrt{(7,125)^2 + (1338)^2}$$

$$S_t = 7,125 + \sqrt{(7.125)^2 + (1.338)^2} \times 10^3$$

$$S_t = 7,125 + \sqrt{50.8 + 1.79} \times 10^3$$

$$S_t = 7,125 + \sqrt{52.59} \times 10^3$$

$$S_t = 7,125 + 7.26 \times 10^3 = 7,125 + 7,260$$

$S_t =$ 14,385 psi (Tension)

$S_s =$ 7,260 (shearing)

Mat'l Properties

$S_y = 27,000$ psi
$S_{su} = 26,000$ psi
$S_{s_{All}} = 15,600$ psi

$$SF(t) = \frac{27,000}{14,385} = 1.87 \quad \text{(Safety Factor in Tension)}$$

$$SF(s) = \frac{15,600}{7,260} = 2.15 \quad \text{(Safety Factor in Shearing)}$$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/11/62 SUBJECT LVT PX II Roadwheel Bolt Analysis Suspension Group SHEET NO. 37 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4561-3-3

Bolts - Spindle Support To hull

Assume Failure of bolts to occur by shearing due to Torque on spindle support

$S_s = \frac{P}{A}$ $P = \frac{T2}{D} = \frac{84,300(2)}{13.75}$

T = 84,300 lbs.-in.
Bolt Circle Dia = $13\frac{3}{4}$
Bolt Diameter = $\frac{5}{8}$
Bolt Area = .3068

$P = 12,280$

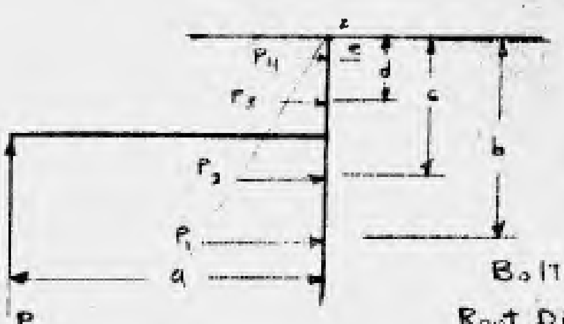
$S_s = \frac{12,280}{.3068} = 39,980$ psi

Use 10 bolts $S_s = \frac{39980}{10} = 3,998$ psi (Stress per bolt)

$S.F. = \frac{15,600}{3998} = 3.91$

Bolt Tension and Compression Force Analysis

P = 35,000
a = 11.875
b = 13.25"
c = 9.5
d = 4.25
e = .5



Bolts - 5/8 - 18 UNF
Root Dia = d_r = .556 in.
Root Area = A_r = .240 in.^{2}
Thread Length = h = .997

$\sum M_{z=0} = P(a) - P_1(b) - P_2(c) - P_3(d) - P_4(e)$

$\tan\theta = \frac{P_1}{b} = \frac{P_2}{c} = \frac{P_3}{d} = \frac{P_4}{e}$

Material Properties (Bolt)
$S_y = 150,000$ psi

$\therefore P_2 = \frac{c}{b} P_1, \; P_3 = \frac{d}{b} P_1, \; P_4 = \frac{e}{b} P_1$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/19/62 SUBJECT LVTPX11 Roadwheel Bolt Analysis Suspension Group SHEET NO. 38 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4561-3-3

$$\Sigma M_z = 0 = P(a) - P_1(b) - \left(\frac{c^2}{b}\right)P_1 - \left(\frac{d^2}{b}\right)P_1 - \left(\frac{e^2}{b}\right)P_1$$

$$P_1(13.25) + \left(\frac{9.5^2}{13.25}\right)P_1 + \left(\frac{4.25^2}{13.25}\right)P_1 + \left(\frac{.5^2}{13.25}\right)P_1 = 35{,}000(11.875)$$

$$P_1(13.25) + 6.81\ P_1 + 1.36\ P_1 + .018\ P_1 = 35{,}000(11.875)$$

$$21.44P_1 = 35{,}000\ (11.875)$$

$$P_1 = \frac{35{,}000\ (11{,}875)}{21.44}$$

$$P_1 = 19{,}350^{\#}$$

Two bolts take load at each reaction $(P_1, P_2, P_3 + P_4)$

Worst load on one bolt equals $\frac{P_1}{2} = \frac{19{,}350}{2} = \underline{\underline{9{,}675}}^{\#}$

Tensile Stress in Bolt (Assuming one bolt taking load)

$$S_t = \frac{P}{A} = \frac{P}{.24} = \frac{9{,}675}{.24} = \underline{40{,}300 \text{ psi}}$$

$$S.F. = \frac{S_y}{S_t} = \frac{150{,}000 \text{ psi}}{40{,}300 \text{ psi}} = \underline{3.72}$$

Shear Stress in Bolt Threads

$$S_s = \frac{P}{\pi D_r h} = \frac{9{,}675}{\pi(.556)(.997)} = \underline{5{,}560 \text{ psi}}$$

$$S.F. = \frac{S_{sy}}{S_s} = \frac{(.6)(150{,}000)}{5{,}560} = \underline{16.2}$$

Aluminum Threads (Holding insert)

Aluminum Thread $d_r = .75$

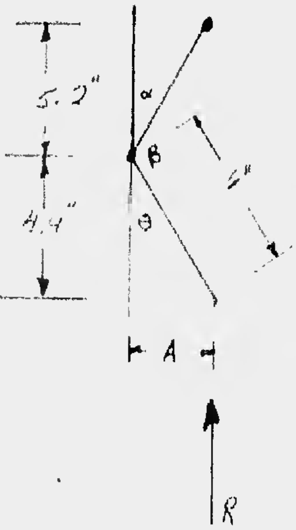
$$S_s = \frac{P}{\pi d_r h} = \frac{9{,}675}{\pi(.75)(.997)}$$

$$S_s = \underline{4{,}120 \text{ Psi}}$$

$$S.F = \frac{S_{sy}}{S_s} = \frac{(.6)(150{,}000)}{4{,}120} = \underline{21.9}$$

Bolts and inserts are very adequate under these static conditions, one bolt carrying the load.

Spline Analysis (Spring To Spindle)



Static Position $\theta = 35°$
Bump Position $\alpha = 30°$
$\beta = 115°$

Static Reaction = $3960^{\#} = R$
Effective Arm = $3.442" = A$
Static Torque = RA
= 3960(3.442)
= $\underline{13630^{\#}\text{-in}}$

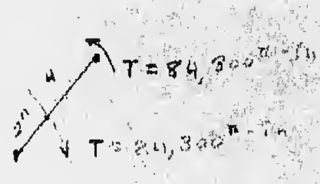
Dynamic Rate = $616^{\#}$-in/deg
Dynamic Windup = 115°
Dynamic Torque = 616(115)
= $\underline{71,000^{\#}\text{-in}}$

Spline Data
L = 5.25"
P.D = 4"

Total Bump Torque = Static + Dynamic
= 13,630 + 71,000
= $\underline{84,630^{\#}\text{-in}}$ (Approx.)

Shear Area of spline $(A) = \frac{CL}{2}$ where $C = \pi(P.D.)$
L = Length of Spline teeth

$S_s = \frac{F}{A}$ $F = \frac{2T}{D}$



$$S_s = \frac{\frac{2T}{D}}{\frac{CL}{2}} = \frac{4T}{\pi D^2 L}$$

Assume half of teeth in contact

Material Properties
$S_y = 27,000$ psi
$S_{s\,all} = 27,000(.6) = 16,200$ psi

$$S_s = \frac{8(84,300)}{\pi(16)(5.25)} = 2,560 \text{ psi}$$

$$SF = \frac{S_{s\,all}}{S_s} = \frac{16,200}{2,560} = \underline{6.33}$$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY C. Nelson DATE 9/13/62 SUBJECT LVT PX II Roadwheel Stress Analysis Suspension Group SHEET NO. 41 OF 69
CHKD. BY TH DATE JOB NO. Nebs 4561-3-3

Key Analysis

Four keys $\frac{1}{4} \times \frac{1}{4}$

Assume All in contact

$$S_s = \frac{F}{A}$$

$$F = \frac{T2}{D} = \frac{24,630(2)}{10.75} = 16,950^{\#}$$

$$A = 4 \times L \times .25 = 4(6.5)(.25) = 6.5 \text{ in}^2$$

$$S_s = \frac{16,950}{6.5} = 2,610 \text{ psi}$$

$$SF = \frac{S_{All}}{S_s} = \frac{15,600}{2,610} = 5.97$$

Lockup Spline Analysis

Assume lockup position as shown below

L = 2.5"
P.D. = 2

Following Calculations based on Reaction R = 14,000# (Preliminary Value)

$$T = R(3.44) = 14,000(3.44) = 48,200^{\#}\text{-in}$$

$$S_s = \frac{8T}{\pi D^2 L}$$

(Assume half of P.D. Area in Shear and half of teeth in contact.)

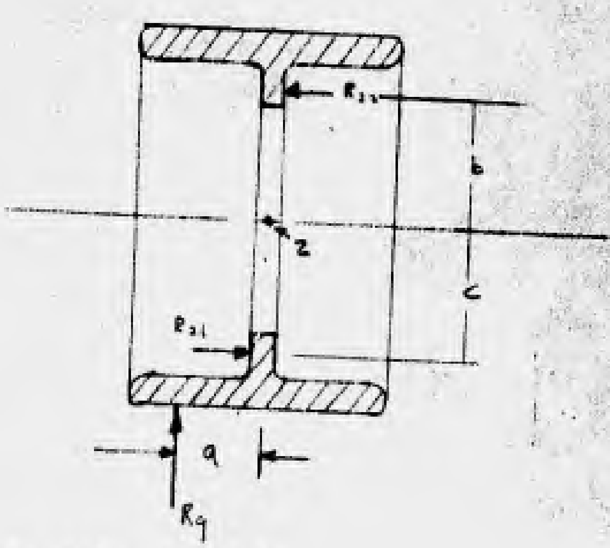
$$S_s = \frac{8(48,200)}{\pi(6.25)(2.5)} = 7,850 \text{ psi} \qquad S_F = \frac{15,600}{7,850} = 1.98$$

INGERSOLL-KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/19/62 SUBJECT LVT PXII Roadwheel SHEET NO. 42 OF 69
CHKD. BY TH DATE ___ Stress Analysis JOB NO. Nobs
Suspension Group 4561-3-3

Roadwheel

$a = 4.25''$
$b = c = 10\ 7/8''$
$R_q = 24{,}500^{\#}$
$R_{22} = R_{21}$



$\circlearrowleft \Sigma M_z = 0 = R_q(a) - R_{21}(c) - R_{22}(b)$

$R_{21} = R_{22}$

$\therefore\ R_q(a) - R_{21}(c) - R_{21}(b) = 0$

$R_{21}(c+b) = R_q(a)$

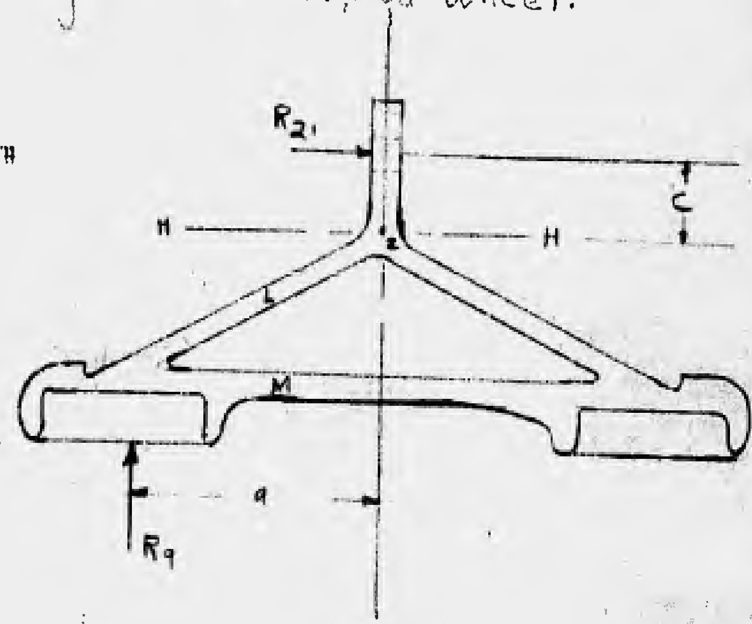
$$R_{21} = \frac{R_q(a)}{(c+b)} = \frac{24{,}500(4.25)}{21.75} = 4780^{\#}$$

$\rightarrow \Sigma F_y = 0 = R_{21} - R_{22}$

$R_{21} = R_{22} = \underline{\underline{4780^{\#}}}$

Consider roadwheel rigid with 70% of 8G shock load occuring on one side of wheel.

$R_q = 24{,}500^{\#}$
$C = 1''$
$a = 4.25''$
$W = .5''$
$r = 10.875$



Check critical sections of wheel for stresses

The critical area of the Roadwheel appears to be section H-H, at which Bending would occur. To fail the L-M portions of the wheel would require the L portion to buckle while at the same time failing portion M in Tension. Therefore section H-H will be assumed critical.

Allowable Stress = 22,000 psi
Assume Bending of rectangular plate

$S_B = \frac{M}{Z}$ $\qquad Z = \frac{M}{S_B} = \frac{4280}{22{,}000} = .177 \text{ in}^3$

$Z = \frac{I}{C} = \frac{LW^2}{6} = \frac{L(.5)^2}{6} = \frac{L(.25)}{6}$

$.177 = \frac{L(.25)}{6}$

$L = \frac{6(.177)}{.25} = 4.24''$

$S = L\theta \qquad \theta = \frac{S}{r} = \frac{4.24}{10.875}$

$\theta = .390 \text{ rad}$

$\theta^\circ = \frac{.390(360)}{2\pi} = 22.4^\circ$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G Nelson DATE 9/19/62	SUBJECT LVTPXII Roadwheel	SHEET NO. 44 OF 69
CHKD. BY TH DATE	Stress Analysis	JOB NO. Nobs
	Suspension Group	4561-3-3

The previous calculations show that section H-H is satisfactory since an arc of only 22.4° is required to give the necessary area needed to withstand bending. And it seems resonable to assume that more area than that found in the 22.4° arc will actually be carrying the load

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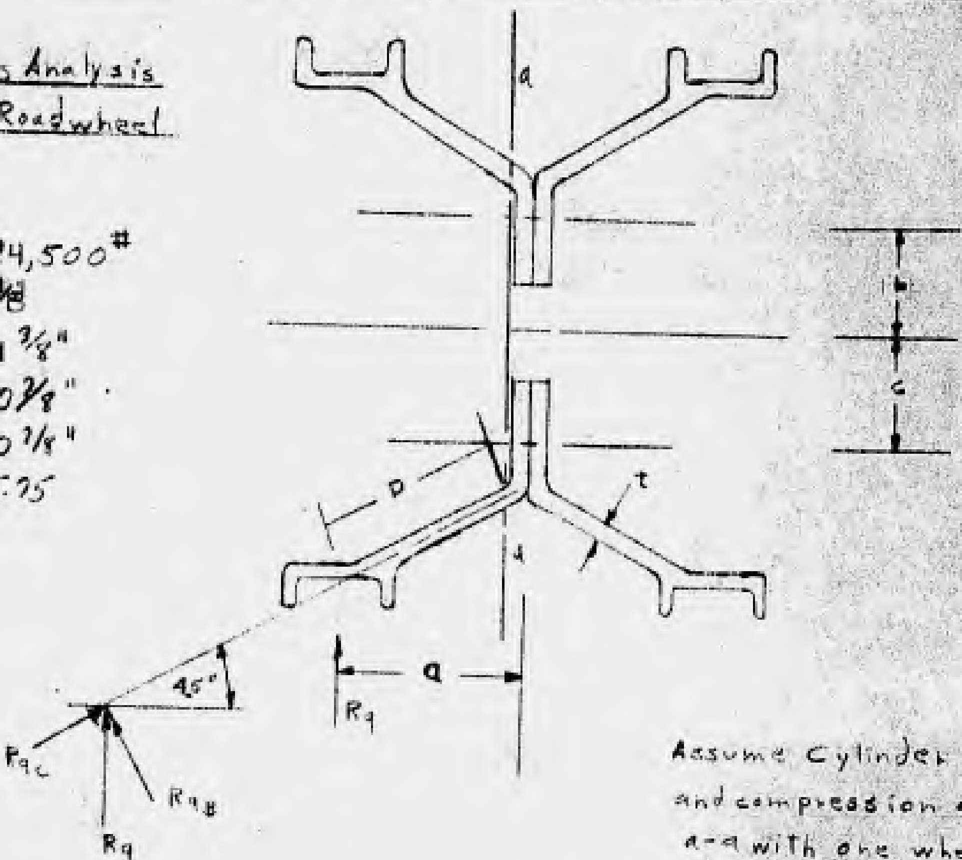
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/20/62 SUBJECT LVT-PYII Roadwheel Two Wheel Version Suspension Group SHEET NO. 45 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4561-3-3

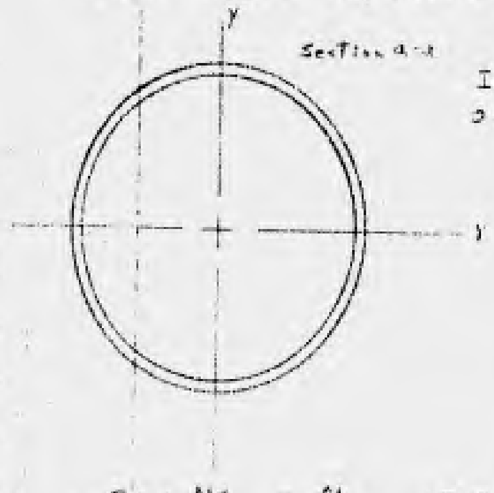
Stress Analysis of Roadwheel

$R_q = 24{,}500^{\#}$
$t = 3/8$
$a = 4\,3/8"$
$b = 10\,7/8"$
$c = 10\,7/8"$
$D = 5.75$



Assume cylinder in bending and compression at section a-a with one wheel taking 70% of an 8G load.

$R_{qB} = R_{qC} = R_q(\cos 45°) = 24{,}500^{\#}(.707) = 17{,}350^{\#}$



Section a-a

ID = 24" = d
OD = $24\tfrac{3}{4}"$ = D

$$Z = \frac{I}{c} = \frac{\pi(D^4 - d^4)}{32D} = \frac{\pi(24.75^4 - 24^4)}{32(24.75)}$$

$$Z = \frac{\pi(377{,}000 - 331{,}000)}{32(24.75)}$$

$$Z = \frac{\pi(46{,}000)}{32(24.75)} = 182.4 \text{ in}^3$$

$$S_B = \frac{Mc}{I} = \frac{M}{Z} = \frac{17{,}350(5.75)}{182.4} = 546 \text{ psi}$$

$$S_c = \frac{P}{A} = \frac{17{,}350}{\frac{\pi(D^2-d^2)}{4}} = \frac{69{,}400}{\pi(37)} = 597 \text{ psi}$$

Resultant Stress $S_R = S_B + S_c = 546 + 597 = 1143$ psi

KALAMAZOO DIVISION
[illegible]NER CORPORATION
[illegible]MAZOO, MICHIGAN

[illegible] Roadwheel	SHEET NO. 46 OF 69
[illegible] Version	JOB NO. Nobs
[illegible] Group	4561-3-3

[illegible] ble stress of 15,000 psi

[illegible] $\frac{[illegible]00}{[illegible]} = 13.1$

[illegible] ve calculations and assumed conditions [illegible] a safety factor of 13.1. To clarify the calculations an explanation should be noted; that being that the loads were assumed distributed over the entire section. This, of course, is not exactly true but does give an indication of strength, and with the large safety factor under these conditions it seems safe to assume the wheel will be satisfactory.

Idler Design Conditions:

C-1 4G Vehicle (2G per idler) center loaded

C-2 600 hours at 20 MPH (358 rpm) 4000# Track Tension

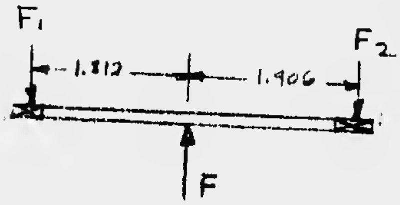
C-3 800 hours at 12 MPH (216 rpm) 6000# Track Tension

C-4 190 hours at 40 MPH (712 rpm) 2000# Track Tension

C-5 10 hours at 5 MPH (89 rpm) 17,500# Track Tension
(Continuous Steer)

Track Tensions are on top half of track and therefore must be doubled to obtain Total load on idler.

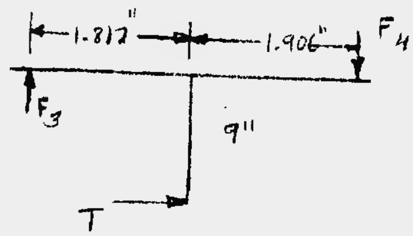
Bearing loads



Radial Force Reactions on idler bearings ($F_1 + F_2$)
F = Force on idler due To Track Tension

$$F_1 = \frac{1.906}{3.712}(F) = .514(F)$$

$$F_2 = \frac{1.812}{3.712}(F) = .489(F)$$



$F_3 + F_4$ are radial loads due to side thrust

For Thrust in opposite direction, Forces are reversed.

$$F_4 = \frac{1.812}{1.906} F_1 = .952 F_3$$

$$T(9) = F_4(1.906) + F_3(1.812) = \frac{1.812}{1.906}(1.906) F_3 + F_3(1.812)$$

$$T(9) = 3.624 F_3 \qquad F_3 = \frac{9}{3.624}(T) = 2.48 T$$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/24/62 SUBJECT LVTPX11 Idler SHEET NO. 48 OF 69
CHKD. BY TH DATE Bearing Analysis JOB NO. Nobs
Suspension Group 4561-3-4

Summary of Bearing loads

Thrust Acting Inward

Assume thrust of 1000#

	Inner Roller Bearing			Outer Thrust Bearing			
Cond #	Radial load - F_2	Radial load - F_4	Radial load Total - lb.	Radial load - F_1	Radial load - F_3	Radial load Total - lbs	Thrust load-lbs
C-2	3880	0	3880	4120	0	4120	
C-3	5840	0	5840	6160	0	6160	
C-4	1940	0	1940	2060	0	2060	
C-5_1	17,000	2360	19,360	18,000	-2480	15,520	1000

Thrust Acting Outward (C-2 Thru C-4 The same as above)

C-5_2	17,000	-2360	14,640	18,000	2480	20,480	1000

For life calculations use C-5_1 for roller Bearing
For life calculations use C-5_2 for Thrust Bearing

Outer Thrust Bearing Life Calculations

Bearing Make + No. - SKF 22214C

Bore - 2.7559

O.D. - 4.9213

Width - 1.2205

fillet radius - .059

From SKF Catalog:

Load Ratio = $\frac{C}{P}$

$P = XVF_r + YF_a$

where
$X = X_1$ or X_2 from Cat. Table
$V = 1.0$
$Y = Y_1$ or Y_2 from Cat. table
F_r = Calculated radial load
F_a = Calculated thrust load

Condition C-5₂

$\frac{F_a}{VF_r} = \frac{1000}{20,480} = .048 \qquad e = .23 \quad \therefore$ use $X_1 + Y_1$

$X_1 = 1.0$

$Y_1 = 2.92$

$P = XVF_r + YF_a$

$P = .67(1)(20,480) + 2.92(1000)$

$P = 13,750 + 2920$

$P = 16,670$

$C = 27,200$

$C/P = 27,200/16,670 = 1.635$

RPM = 89

Life from Catalog = 980 hrs.

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BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/24/62 SUBJECT LVT PX11 - Idler SHEET NO. 50 OF 69
CHKD. BY TH DATE Bearing Analysis JOB NO. Nobs
Suspension Group 4561-3-4

Thrust Bearing

Cond. No.	P	C/P	RPM	Life-hours
C-2	4120	6.62	358	21,000
C-3	6160	4.42	216	10,500
C-4	2060	13.2	717	130,000
C-5	16,670	1.635	89	980

Weighted life = $L_t = \dfrac{1}{\frac{S_2}{LP_2} + \frac{S_3}{LP_3} + \frac{S_4}{LP_4} + \frac{S_5}{LP_5}}$

$$L_t = \frac{1}{\frac{.6}{21,000} + \frac{.2}{10,500} + \frac{.19}{130,000} \ \frac{.01}{980}} = \frac{1}{(28.6 + 19.05 + 1.46 + 10.4)\times 10^{-6}}$$

$$L_t = \frac{10^6}{(59.51)} = \underline{\underline{16,800 \text{ hours}}}$$

Roller Bearing RBC- SJ9688 (Refer To sheet No. for Sample Calculation)

Cond. No.	P	C'/P'	RPM	Life hours
C-2	3880	6.68	356	14,000
C-3	5840	4.43	216	6,800
C-4	1940	13.35	717	56,000
C-5	19,360	1.33	89	460

$C' = 25,900$

Weighted life $= L_t = \dfrac{1}{\frac{.6}{14,000} + \frac{.2}{6,800} + \frac{.19}{56,000} + \frac{.01}{460}}$

$$L_t = \frac{1}{(42.8 + 29.4 + 3.39 + 21.7)\times 10^{-6}}$$

$$L_t = \frac{10^6}{97.29} = \underline{\underline{10,280 \text{ hours}}}$$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 10/3/62 SUBJECT LVT-3PXII Idler Bearing Analysis Suspension Group SHEET NO. 51 OF 69
CHKD. BY TH DATE ______ JOB NO. Nobs 4561-3-A

Roller Bearing life Calculations (Revision No. 1)

Bearing Make & No. - RBC - SJ 6925
Bore - 5.00
O.D. - 7.50
Width - 2.5
C' = 49,000#
fillet radius = .120

Cond. No.	P	C'/P'	RPM	Life-hours
C-2	3,880	12.62	358	90,000
C-3	5,840	8.39	216	44,000
C-4	1,940	25.24	717	800,000
C-5	19,360	2.53	79	3200

Weighted life

$$L_t = \frac{1}{\frac{.6}{9[illegible]} + \frac{.2}{44,[illegible]} + \frac{.19}{[illegible]} + \frac{.01}{3200}}$$

$$L_1 = \frac{1}{(6.[illegible] + 4.[illegible] + .0[illegible]3 + 3.13) \times 10^{-6}}$$

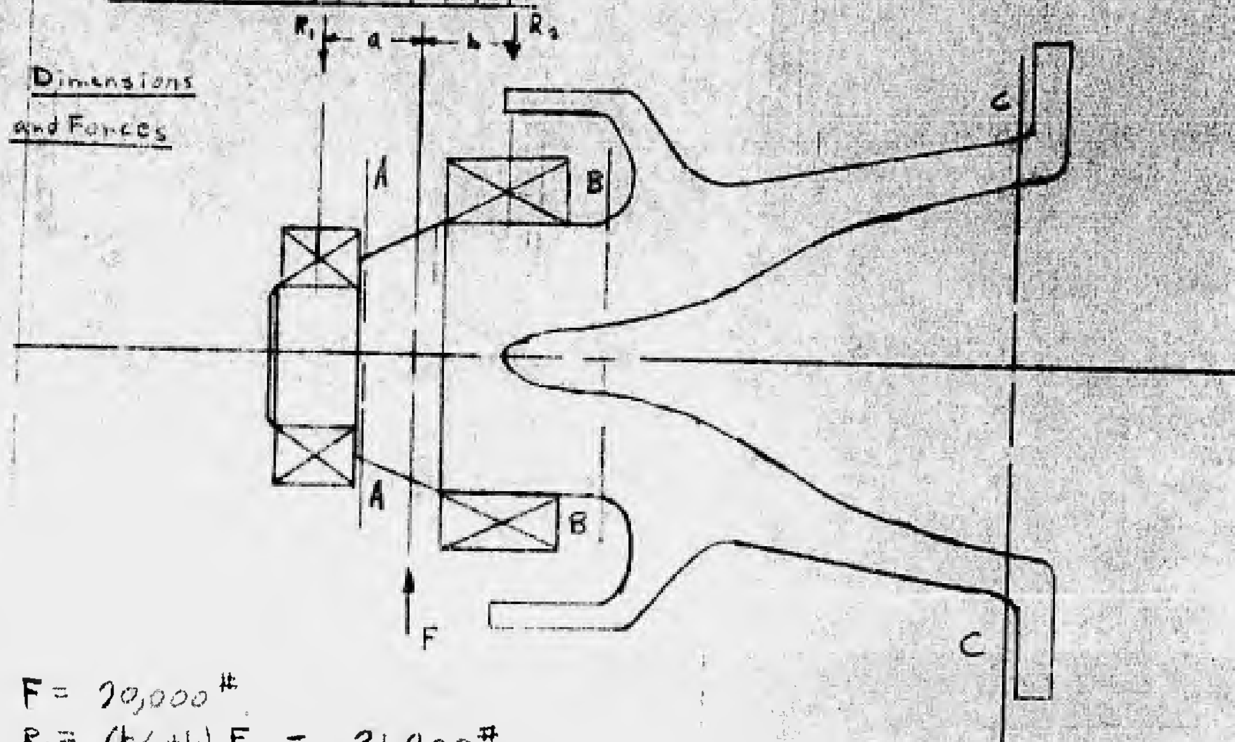
$$L_1 = \frac{10^6}{(14.973)} = 66,800$$

L_1 = 66,800 hour

The above calculations show that both the thrust bearing and the roller bearing are very adequate.

Idler Support Arm

Dimensions and Forces



F = 70,000#

$R_1 = (b/a+b)\ F = 31,700$#

$R_2 = (a/a+b)\ F = 38,300$#

$a = 1\frac{13}{16} = 1.8125$

$b = 1\frac{1}{2} = 1.5$

Section A-A

D = 2.79

D' = 3.125

fillet radius = .059'

Moment Arm = .6102 = L

Section B-B

d = 1.375

D = 5.0

fillet radius = 1.0

Moment Arm = 2.875 = L

D' = 7.0

Section C-C

$d = 7\frac{7}{8}$

$D = 9\frac{3}{8}$

$D' = 9\frac{3}{4}$

fillet radius = 1/2"

Moment Arm = $10\frac{3}{4}$" = L

Section A-A

$$S_B = \frac{M}{Z} = \frac{R_1(L)(32)}{\pi D^3} = \frac{31,200(.6102)(32)}{\pi(2.79)^3}$$

$$S_B = \frac{31,200(.6102)(32)}{\pi(21.6)} = 9140 \text{ psi}$$

$$S_{B(Max)} = S_B(k_t)$$

$$\frac{r}{D} = \frac{.059}{2.79} = .021 \qquad \frac{D'}{D} = \frac{3.125}{2.79} = 1.12$$

$$k_t \text{ (Bending)} = 2.2$$

$$S_{B(Max)} = 9140(2.2) = \underline{\underline{20,100 \text{ psi}}}$$

Section B-B

$$S_B = \frac{M}{Z} = \frac{F(L)\,32D}{\pi(D^4-d^4)} = \frac{70,000(2.875)(32)(5.0)}{\pi(5^4-1.375^4)}$$

$$(D^4-d) = (625-3.57) = 621.43$$

$$S_B = \frac{70,000(2.875)(32)(5.0)}{\pi(621.43)} = 16,500 \text{ psi}$$

$$S_{B(Max)} = S_B(k_t)$$

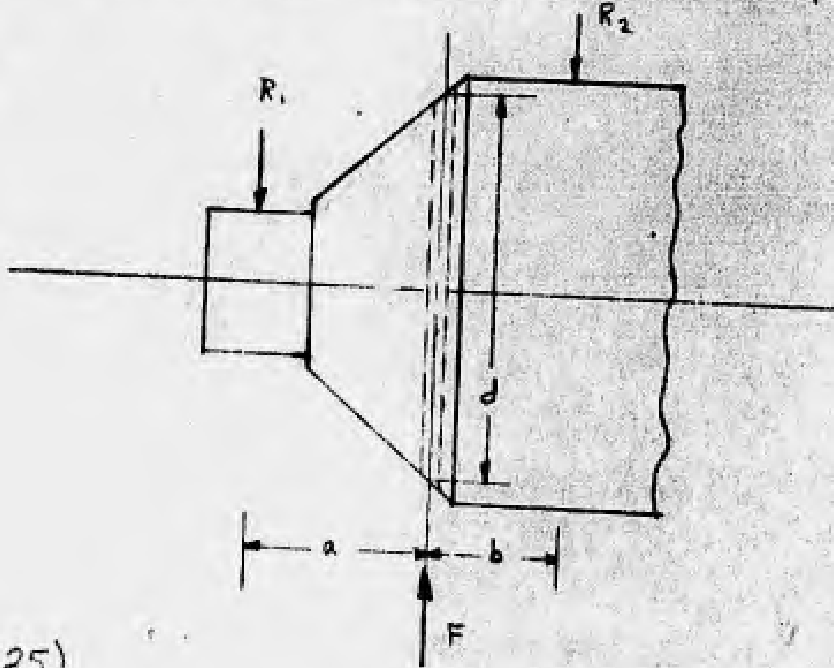
$$\frac{r}{D} = \frac{1.0}{5.0} = .2 \qquad \frac{D'}{D} = \frac{7.0}{5.0} = 1.4$$

$$k_t = 1.41$$

$$S_{B(Max)} = 16,500(1.41) = \underline{\underline{23,300 \text{ psi}}}$$

Spindle Stresses at Hole Section

$a = 1.8125$
$b = 1.5$
$F = 70,000^{\#}$
$R_1 = 31,700^{\#}$
$R_2 = 38,300^{\#}$
$d = 4.75$



$M_{(max)} = R_1(a)$
$M_{(max)} = (31,700)(1.8125)$
$M_{(max)} = 57,500''^{\#}$

$$z = \frac{\pi d^3}{32} = \frac{\pi(4.75)^3}{32} = 10.52$$

$$S = \frac{Mc}{I} = \frac{M}{Z} = \frac{57,500}{10.52} = 5480 \text{ psi}$$

$$\frac{r_1}{d} = \frac{.25}{4.75} = .052$$

$\therefore k_t = 2.45$

$S_{(max)} = k_t S$

$S_{(max)} = 2.45(5480) = \underline{\underline{13,400 \text{ psi}}}$

$$S_s = \frac{F}{A} = \frac{70,000(4)}{\pi(d^2)} =$$

$$S_s = \frac{280,000}{\pi(22.5)} = \underline{3970 \text{ psi}}$$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 10/3/62 SUBJECT LVT PX II Idler Wheel Stress Analysis Suspension Group SHEET NO. 55 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4561-3-4

Section C-C

$$S_B = \frac{M}{Z} = \frac{F(L)\,32\,D}{\pi(D^4-d^4)}$$

$$S_B = \frac{20{,}000(10.75)(32)(9.325)}{\pi(9.325^4 - 7.325^4)}$$

$$(D^4 - d^4) = (7710 - 2960) = 4750$$

$$S_{B(Max)} = S_B(K_t)$$

$$\frac{r}{D} = \frac{.5}{9.325} = .0534 \qquad \frac{D'}{D} = \frac{9.75}{9.325} = 1.04$$

$$K_t\,(\text{Bending}) = 1.75$$

$$S_B = \frac{20{,}000(10.75)(32)(9.325)}{\pi(4750)}$$

$$S_B = 15{,}150 \text{ psi}$$

$$S_{B(Max)} = 15{,}150\,(1.75) = \underline{\underline{26{,}500 \text{ psi}}}$$

The Calculations show all sections to be at a stress level compatible with the design conditions. The most highly stressed region of the idler support arm was found to be next to the hull (Section C-C), indicating that initial failure would occur in that region.

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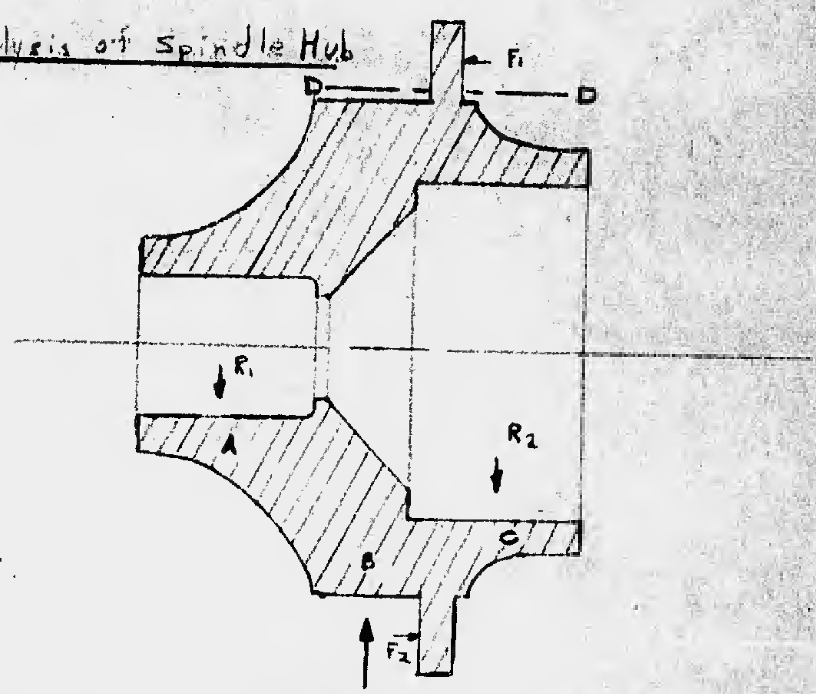
KALAMAZOO, MICHIGAN

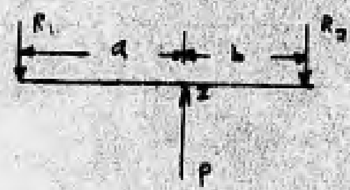
BY G Nelson	DATE 9/27/62	SUBJECT LVT PX II - Idler wheel	SHEET NO. 56 OF 69
CHKD. BY TH	DATE	Stress Analysis	JOB NO. Nobs
		Suspension Group	4561-3-1

Stress Analysis of Spindle Hub



Condition C-1 (2g load)
Centrally Applied load

$P = 70{,}000^{\#}$
$a = 1.81''$
$b = 1.5''$



$\circlearrowleft + \quad \sum M_z = 0 = P(b) - R_1(a+b)$

$$R_1 = \frac{P(b)}{a+b} = \frac{P(1.5)}{3.31} = .453P = .453(70{,}000)$$

$\underline{R_1 = 31{,}700 \text{ lb}}$

$+\uparrow \quad \sum F_y = 0 = P - R_1 - R_2$

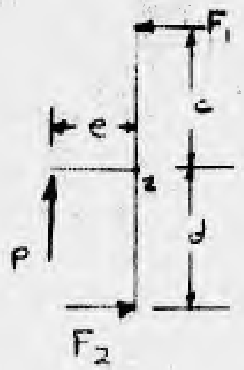
$R_2 = P - R_1 = 70{,}000 - 31{,}700$
$\underline{R_2 = 38{,}300^{\#}}$

70% of load on one Idler Wheel, occuring during normal vehicle operation thus 35,000 lb will be load. (This would be due To foreign matter passing between wheel and track)

Assume all of Turning moment being Taken by bolt Ring (see next sheet)

70% of load on one Idler wheel (cont)

$P = 70\%(35{,}000)$
$P = 23{,}500^{\#}$
$d = c = 5.25''$
$e = 4.125''$
$F_1 = F_2$



$$\circlearrowright\ \Sigma M_z = 0 = P(e) - F_1(c) - F_2(d) \quad = P(e) - F_1(c+d)$$

$$F_1 = \frac{P(e)}{c+d} = \frac{23{,}500^{\#}(4.125)}{10.5} = 9250^{\#}$$

$$F_1 = F_2 = \underline{\underline{9250}}^{\#}$$

Assume Bending at Section D-D

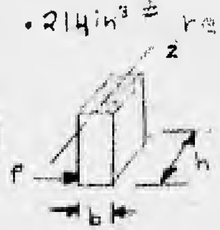
$S = \frac{M}{z}$ Assume all of Turning moment Taken by bolt flange

$S_y = 23000\,psi$

$z = \frac{M}{S}$ Moment Arm = .625''

$$z = \frac{(9250^{\#})(.625'')}{23{,}000} = .214\,in^3 = \text{required section modulus}$$

For a Plate in bending



$I_z = \frac{hb^3}{12}$ $z = \frac{hb^3}{12} \cdot \frac{2}{b} = \frac{hb^2}{6}$

$b = .5$

$\frac{hb^2}{6} = z = .214\,in^3$

$$h = \frac{6(z)}{b^2} = \frac{6(.214)}{.25} = 24(.214) = 5.14''$$

In Terms of degrees needed to obtain the above distance as arc length. $(s = r\theta)$ (where $s = 5.14''$)

$$\theta = \frac{s}{r} = \frac{5.14}{4.62} = 1.11\ \text{rad.} \quad \theta = \frac{1.11(180)}{\pi} = \underline{\underline{63.8^{\circ}}}$$

This shows that an arc of approximately 63.8° will be necessary to provide the effective area required to limit the bending stress to an allowable value. This is only a rough approximation and it must be remembered that the turning moment will also be resisted by the hub at an area (B) to the left of section D-D. Therefore Section D-D should be adequate for an even more severe condition.

Compressive Stresses

Area A

Use Projected Area = D(L)

$$S_c = \frac{P}{A} = \frac{31,700}{(2.75)(1.25)}$$

$$S_c = \underline{9,220 \text{ psi}}$$

Area B

$$S_c = \frac{P}{A} = \frac{70,000}{(10.25)(1.25)} = \underline{5460 \text{ psi}}$$

Area C

$$S_c = \frac{P}{A} = \frac{38,300^{\#}}{(5)(2.5)} = \underline{3,060 \text{ psi}}$$

INGERSOLL KALAMAZOO DIVISION
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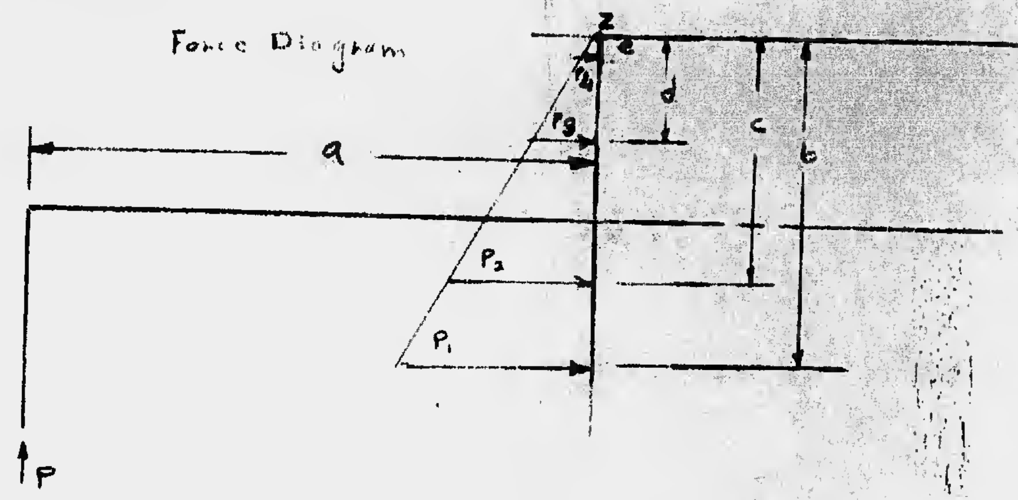
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/27/62	SUBJECT LVT PX11 Idler Wheel	SHEET NO. 59 OF 69
CHKD. BY TH DATE	Bolt Analysis	JOB NO. Nobs
	Suspension Group	4561-3-4

Bolts, Idler to hull

Force Diagram

$P = 20{,}000\#$
$a = 10.625$
$b = 11.00$
$c = 7.815$
$d = 3.625$
$e = .5$



$$\circlearrowright + \quad \Sigma M_Z = 0 = P(a) - P_1(b) - P_2(c) - P_3(d) - P_4(e) - P_5(e)$$

$$\tan\theta = \frac{P_1}{b} = \frac{P_2}{c} = \frac{P_3}{d} = \frac{P_4}{e}$$

$$\therefore P_2 = \frac{c}{b}P_1, \quad P_3 = \frac{d}{b}P_1, \quad P_4 = \frac{e}{b}P_1$$

$$\Sigma M_Z = 0 = Pa - P_1(b) - \left(\frac{c^2}{b}\right)P_1 - \left(\frac{d^2}{b}\right)P_1 - \left(\frac{e^2}{b}\right)P_1$$

$$P_1(11.00) + \left(\frac{7.875^2}{11.00}\right)P_1 + \left(\frac{3.625^2}{11.00}\right)P_1 + \left(\frac{.5^2}{11.00}\right)P_1 = P(a)$$

$$11.0\,P_1 + 5.62\,P_1 + 1.95\,P_1 + .022\,P_1 = 20{,}000(10.625)$$

$$18.57\,P_1 = 20{,}000(10.625)$$

$$P_1 = \frac{20{,}000(10.625)}{18.57}$$

$$P_1 = 40{,}100\#$$

Each reaction ($P_1, P_2, P_3, + P_4$) taken by two bolts.

Reaction on one bolt $= \frac{40{,}100}{2} = \underline{\underline{20{,}050\#}}$

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KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/27/67 SUBJECT LVTPX11 Idler Wheel SHEET NO. 60 OF 69
CHKD. BY TH DATE Bolt Analysis JOB NO. Nebs
Suspension Group 4561-3-1

Bolts - 5/8 - 18UNF
$S_y = 150{,}000$ psi
Root Dia = .556
Root Area = .2400
Thread Length = .997 in.

Tensile Stress in bolts

$$S_T = \frac{P}{A_r} = \frac{20{,}050}{.24} = \underline{\underline{85{,}400 \text{ psi}}}$$

$$S.F. = \frac{S_y}{S_T} = \frac{150{,}000 \text{ psi}}{85{,}400 \text{ psi}} = \underline{1.76}$$

Assume failure by shearing of Aluminum Threads

Insert in hull
Hull Threads
h = length of hull threads
d_r = root dia

$$S_s = \frac{P}{\pi d_r h} = \frac{20{,}050}{\pi(.25)(.997)}$$

$$\underline{S_s = 8740 \text{ psi}}$$

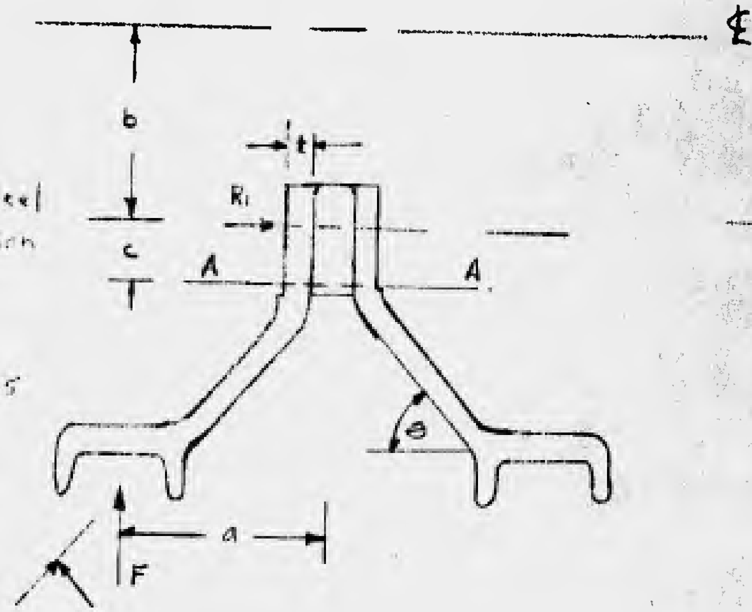
$$S_F = \frac{S_{sy}}{S_s} = \frac{27{,}000(.6)}{8{,}740} = \underline{1.85}$$

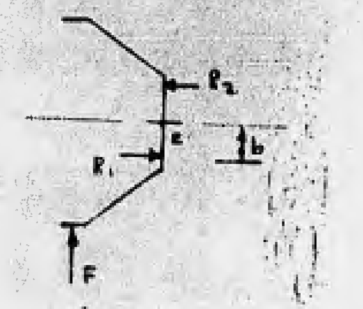
Bolt Threads

$$S_s = \frac{P}{\pi d_r h} = \frac{20{,}050}{\pi(.556)(.997)} = \underline{\underline{11{,}750 \text{ psi}}}$$

$$S.F. = \frac{S_{sy}}{S_s} = \frac{(.6)(150{,}000)}{11{,}750} = \underline{7.65}$$

$\theta = 40°$
F = Force on wheel
R_1 = Bolt Reaction
$a = 4\tfrac{1}{4}''$
$b = 5\tfrac{1}{4}''$
$c = 13/16'' = .8125$
$t = 3/8 = .375$





$R_1(2b)$ = Couple about Pt. Z

$\circlearrowleft \Sigma M_z = 0 \quad F(a) = (R_1)(2b)$

$$R_1 = \frac{F(a)}{2b} = \frac{35{,}000(4.25)}{10.5} = 14{,}150^{\#}$$

$R_1 = R_2 = \underline{14{,}150^{\#}}$

$$S_{B(A\text{-}A)} = \frac{M}{Z} = \frac{R_1(c)}{\frac{bt^2}{6}}$$

For Base assume length of S where $S = r\theta$ and $\theta = 180°$

$S = 6.06 \; \frac{\pi \text{ rad } 180°}{180°}$

$S = 6.06(\pi) = 19.0$

$$S_{B(A\text{-}A)} = \frac{(14{,}150)(.8125)(6)}{19.0\,(.375)^2}$$

$$S_{B(A\text{-}A)} = \frac{14{,}150(.8125)(6)}{(19.0)(.14)} = \underline{25{,}900 \text{ psi}}$$

This approximate stress shows the wheel to be adequate under a 70,000 lb. load assuming each wheel taking half the load. Also no gussets were considered in these calculations however it would be advisable to retain them because of the difficulty in actually determining the stresses in these areas.

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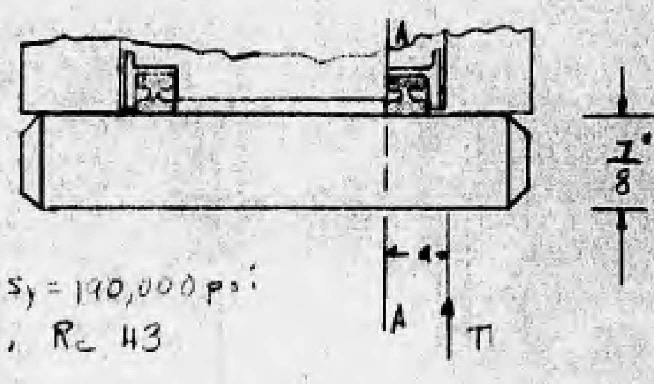
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 9/28/62 SUBJECT LVT PXII - Track Pin, Stress Analysis Suspension Group SHEET NO. 68 OF 69
CHKD. BY TH DATE JOB NO. Nobs 45-61-3-6

Stresses at Section A-A



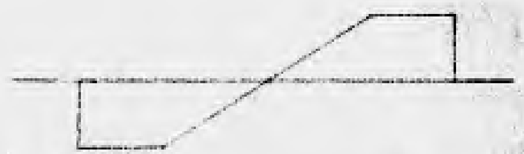
Material
AISI 4340, S_y = 190,000 psi
Temper 850°F, R_c 43

Assume Bending at section A-A

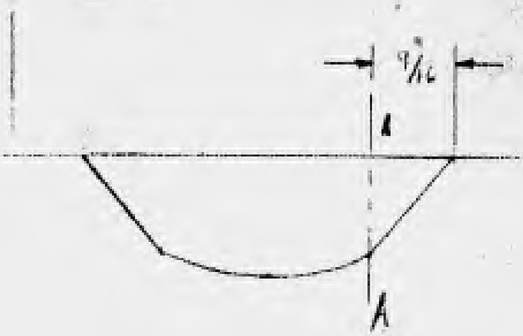
Assumed Acting on Inner edge of Clevis

Known Values
$T = 17{,}500^{\#}/4$
$a = 1.125''$
$D = .875''$
$D^3 = .67 \text{ in}^3$

Shear



Moment



$M = T(a) = \frac{17{,}500^{\#}}{4}(.5625'') = 4375^{\#}(.5625'')$

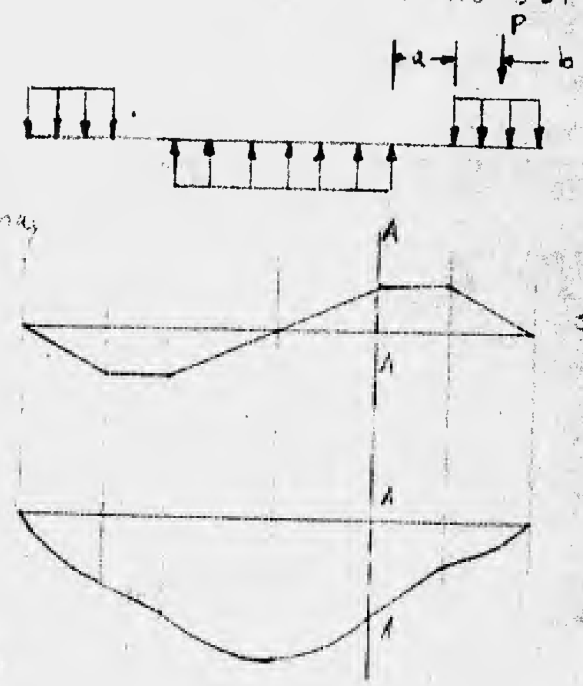
$S_B = \frac{M}{Z} = \frac{M\,32}{\pi D^3} = \frac{32(4375).5625}{\pi(.67)} = \underline{\underline{37{,}400}}$ psi

$S.F. = \frac{S_y}{S_B} = \frac{190{,}000}{37{,}400} = \underline{5.08}$ (Safety Factor in Bending)

$S_s = P/A = 4375/\pi(.437)^2 = 4375/\pi(.192) = \underline{7260 \text{psi}}$ $S.F. = \frac{190{,}000(.6)}{7260} = \underline{15.7}$ (Shearing)

The previous calculations were based on the assumption that the fit between the female clevis and the pin was very tight. Also that bending in the pin would occur along a distance from the edge of the female clevis to the beginning edge in the male clevis. These calculations show a safety factor in bending of 5.08:1 and 15.7:1 in shearing.

Now safety factors will be calculated, assuming uniformly distributed loads on the outer ends of the pins



$a = .5625$
$b = .5$

These uniformly distributed loads may be replaced by a concentrated load acting at its center of gravity. Therefore the moment about section A-A will be equal to $P(a+b)$

$$S_B = \frac{M}{Z} = \frac{32(4375)(1.0625)}{\pi(.67)} = 70,800 \text{ psi}$$

$$S.F. = \frac{190,000}{70,800} = \underline{2.68}$$

This shows a design safety factor below the required design parameter of 4.0:1, however the actual safety factor is assumed to lie between the 5.08 S.F. of the first case and the 2.68 S.F. of the second case.

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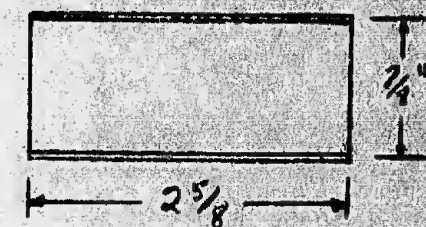
BY G. Nelson DATE 10/1/62 SUBJECT LVTPX11 - Track Pin Bearing Suspension Group SHEET NO. 64 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4561-3-6

Bearing Pressure



$$BP = \frac{\text{Load}}{\text{Projected Area}}$$

$$\text{Load} = \frac{12{,}500}{2} = \underline{8750^{\#}}$$

$$\text{Proj. Area} = DL = \tfrac{7}{8}\left(2\tfrac{5}{8}\right) = .875(2.625) = 2.298$$

$$BP = \frac{8750}{2.298} = \underline{3810\ psi} \quad \text{(static load)}$$

Allowable Bearing Pressure = 25,000 psi

$$S.F. = \frac{P_{All}}{BP} = \frac{25{,}000}{3810} = \underline{6.56}$$

Track Block Clevises

Male Clevis

$S_y = 100{,}000$

$$S_T = \frac{P}{A} = \frac{8750}{2(t)(L)} = \frac{8750}{2(.3125)(2.625)} = \underline{5320\ psi}$$

$$S.F. = \frac{S_{All}}{S_T} = \frac{100{,}000\ psi}{5320} = \underline{18.8}$$

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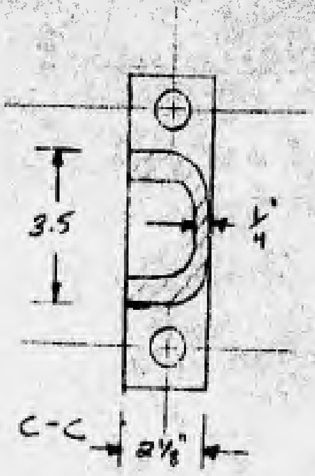
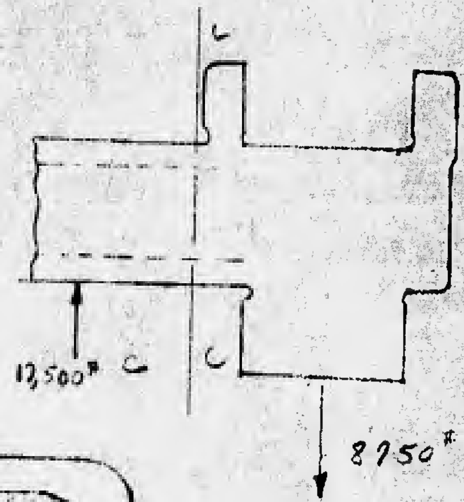
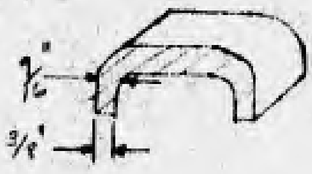
BY G. Nelson DATE 10/2/62 SUBJECT LVT PX11 Track Block Stress Analysis Suspension Group SHEET NO. 65 OF 69
CHKD. BY TH DATE JOB NO. Nobs 4561-3-6



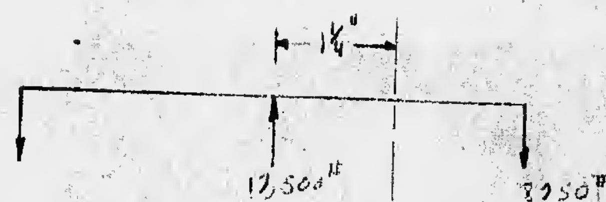


Material Properties

$S_y = 100,000$ psi



Track Block Reactions due to Maximum Track Tension.



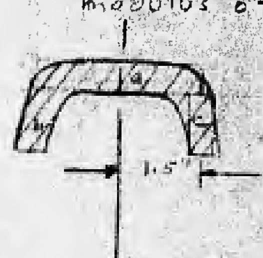
Stresses at Section C-C

$$S_s = \frac{P}{A} = \frac{17,500}{(\frac{3}{8}\times 2\frac{1}{8}) + \frac{1}{4}(\times 2\frac{3}{4})} = \frac{17,500^{\#}}{.798 + .688} = \frac{17,500^{\#}}{1.486} = 11,800 \text{ psi}$$

$$SF = \frac{100,000(.6)}{11,800} = 5.08$$

Find Approximate Section modulus of section C-C

$$S_B = \frac{M}{Z}$$



$$I_{y_a} = \frac{bh^3}{12} = \frac{(\frac{1}{4})(3)^3}{12} = \frac{.25(27)}{12} = .562$$

$$+ I_{y_{b+c}} = \left(\frac{bh^3}{12} + Ad^2\right)2 = \left(\frac{(1.875)(.326)^3}{12} + 1.875(.326)(1.5)^2\right)2$$

$$I_{y_{b+c}} = (.00516 + 1.372)\,2 = (1.37716)\,2 = 2.75432$$

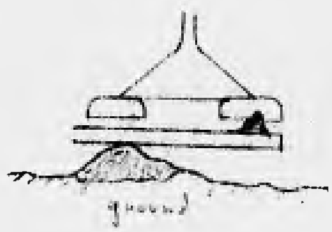
$$I_y = .562 + 2.754 = 3.316 \qquad Z = \frac{I}{c} = \frac{3.316}{1.5} = 2.21$$

$$S_B = \frac{M}{Z} = \frac{12{,}500(1.25)}{2.21} = \underline{9{,}900 \text{ psi}}$$

$$S.F. = \frac{100{,}000}{9{,}900} = \underline{10.1}$$

Bending of Block due to vehicle weight and ground.

Track in normal operation



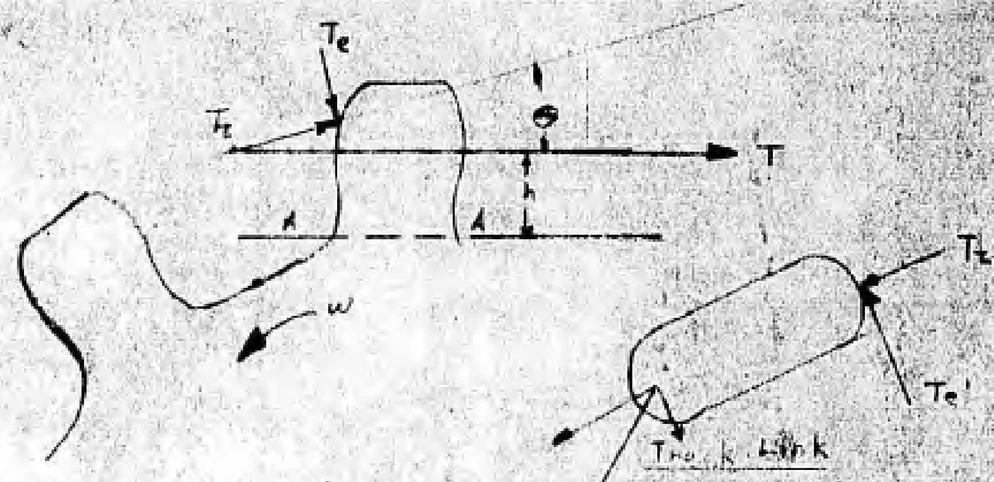
Rock between Road wheel and Track.
Rough ground.
Force Reactions due to condition shown

Because of the Tightly pinned Track, the reaction R_w will Tend To cause bending stresses in the Track block similar to a cantilever beam. However these reactions form a couple which is trying To Twist The Track and Therefore The pins will produce a couple in The opposite direction of $(R_w + R_g)$. Therefore this condition will cause The greatest stresses to occur in the pins and clevises rather than in the main part (Sect. C-C) of the Track block.

Calculation of stresses in The pins and clevises under this condition would be difficult to determine but it seems evident That This condition would be most apt to cause bending or failure. However The safety factor present in these parts under static load represents a sufficient design parameter.

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson	DATE 9/27/62	SUBJECT LVT P7 II - Sprocket	SHEET NO. 67 OF 69
CHKD. BY TH	DATE	Tooth Stresses	JOB NO. N-bs
		Suspension Group	4561-8-5



T = Track Tension
T_e = Track ejection force on tooth
T_t = Tooth force
θ = Pressure Angle = 15°

T_{max} = 17,500#
S.F = 2
S_y = 100,000 psi

h = .75
k_t = 1.3 (Stress concentration Factor)

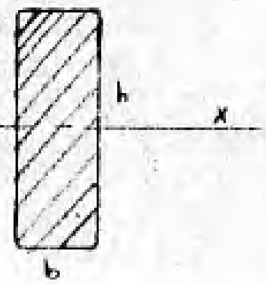
Assume one tooth to be carrying load, (17,500 lb.). For purposes of calculations the tooth will be assumed to be a cantilever beam in bending with the greatest stresses occuring at the root of the tooth. Also the greatest force will occur at low speeds therefore reducing the significance of a velocity factor.

A section Modulus will be determined first

$$S_B = \frac{M}{Z} \qquad Z = \frac{M}{S_B} = \frac{T(h)(S.F)(k_t)}{S_y}$$

$$Z = \frac{17,500(.75)(2)(1.3)}{100,000} = .341 \text{ in}^3$$

Section A-A
(Preliminary)



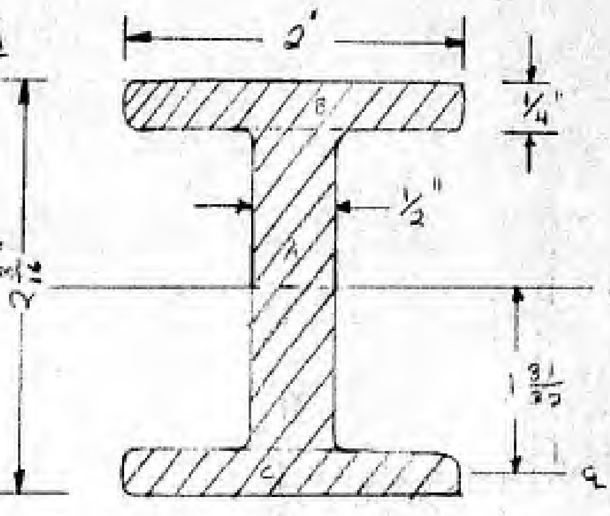
$b = .5$
$h = 2.1875$

$$I_x = \frac{bh^3}{12} = \frac{.5(2.1875)^3}{12} = \frac{.5(10.42)}{12} = \underline{.434 \text{ in}^4}$$

$$Z = \frac{I}{c} = \frac{.434}{1.09} = .396 \text{ in}^3$$

$$S = \frac{M}{Z} = \frac{(35{,}000)(.75)(1.3)}{.396} = \frac{34{,}100}{.396} = \underline{86{,}200 \text{ psi}}$$

Section A-A
(Actual)



$$I_{x(A)} = \frac{bh^3}{12} = \frac{.5(1\frac{11}{16})^3}{12} = \frac{.5(1.6875)^3}{12} = \frac{.5(4.8)}{12} = .2 \text{ in}^4$$

$$I_{x(B)} = I_c + Ad^2 = \frac{bh^3}{12} + Ad^2 = \frac{2(.25)^3}{12} + 2(.25)(1.9688)$$

$$I_{x(B)} = \frac{2(.0156)}{12} + 2(.25)(1.9688) = .0026 + .984 = .9866 \text{ in}^4$$

$$I_x = .2 + 2(.9866) = .2 + 1.9732 = \underline{2.1732 \text{ in}^4}$$

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson	DATE 9/27/62	SUBJECT LVTPX II Sprocket Tooth Stress Analysis Suspension Group	SHEET NO. 69 OF 69
CHKD. BY TH	DATE		JOB NO. Nobs 4561-3-5

$$I_x = 2.1732 \qquad \frac{I_c}{c} = z = \frac{2.1732}{1.9688} = 1.103$$

$$S_B = \frac{M}{z} = \frac{35{,}000(.75)}{1.103} = \underline{\underline{23{,}800 \text{ psi}}}$$

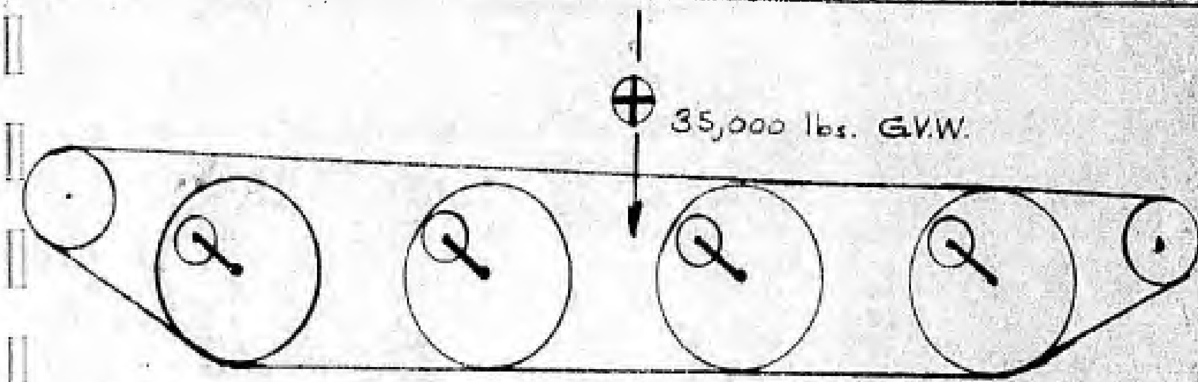
$$S.F. = \frac{S_{All}}{S_B} = \frac{100{,}000}{23{,}800} = \underline{4.2}$$

This safety factor is greater than required, but the tooth design can not be greatly changed to reduce this factor because of the desired width.

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 11/14/62 SUBJECT LVTPXII SHEET NO. 1 OF 10
CHKD. BY ______ DATE ______ Torsilastic Springs JOB NO. Nobs
Suspension Group 4561-3

Ref. B.F. Goodrich Co., Sketches and Calculations.



Estimated Unsprung wt:

Tracks 4000 ÷ 2 = 2000 lbs.
Wheel Assemblies 8X154 = 1232 lbs
Support Arms 8X10 = 80 lbs
3312 lbs

Sprung Weight:

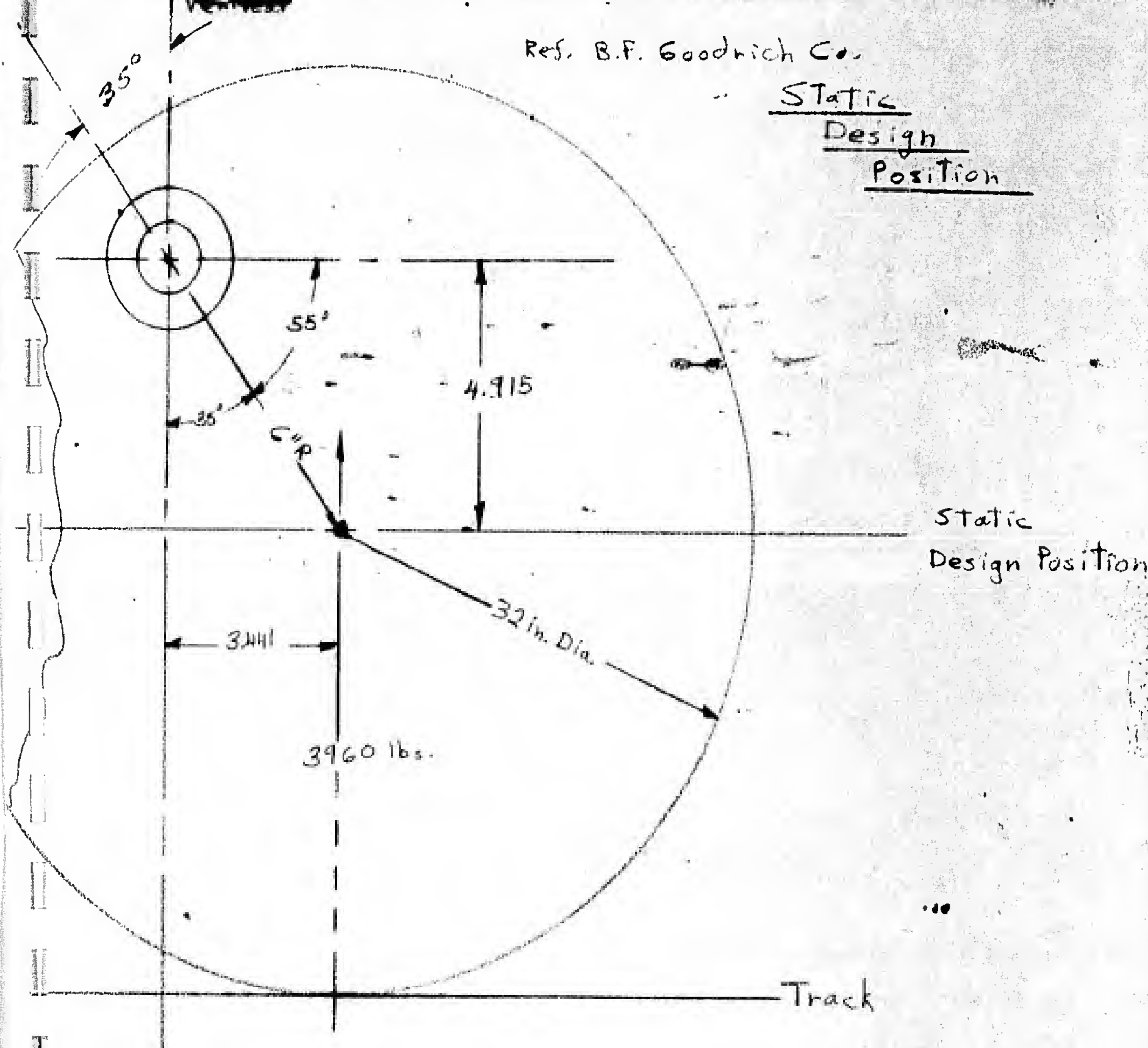
Gross Loaded wt. = 35,000 lbs
Unsprung wt. = -3312
31,688 lbs.

Weight per Road Wheel = 31,688/8 = 3960 lbs.

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 11/14/63 SUBJECT LVT PX11 SHEET NO. 2 OF 10
CHKD. BY ______ DATE ______ Torsilastic Springs [redacted] JOB NO. [redacted]
Suspension [redacted]

Ref. B.F. Goodrich Co.



Sine 55° = .81915 Elevation = 6 x .81915 = 4.915 in.
Cosine 55° = .57358 Effective Arm = 6 x .57358 = 3.441 in.

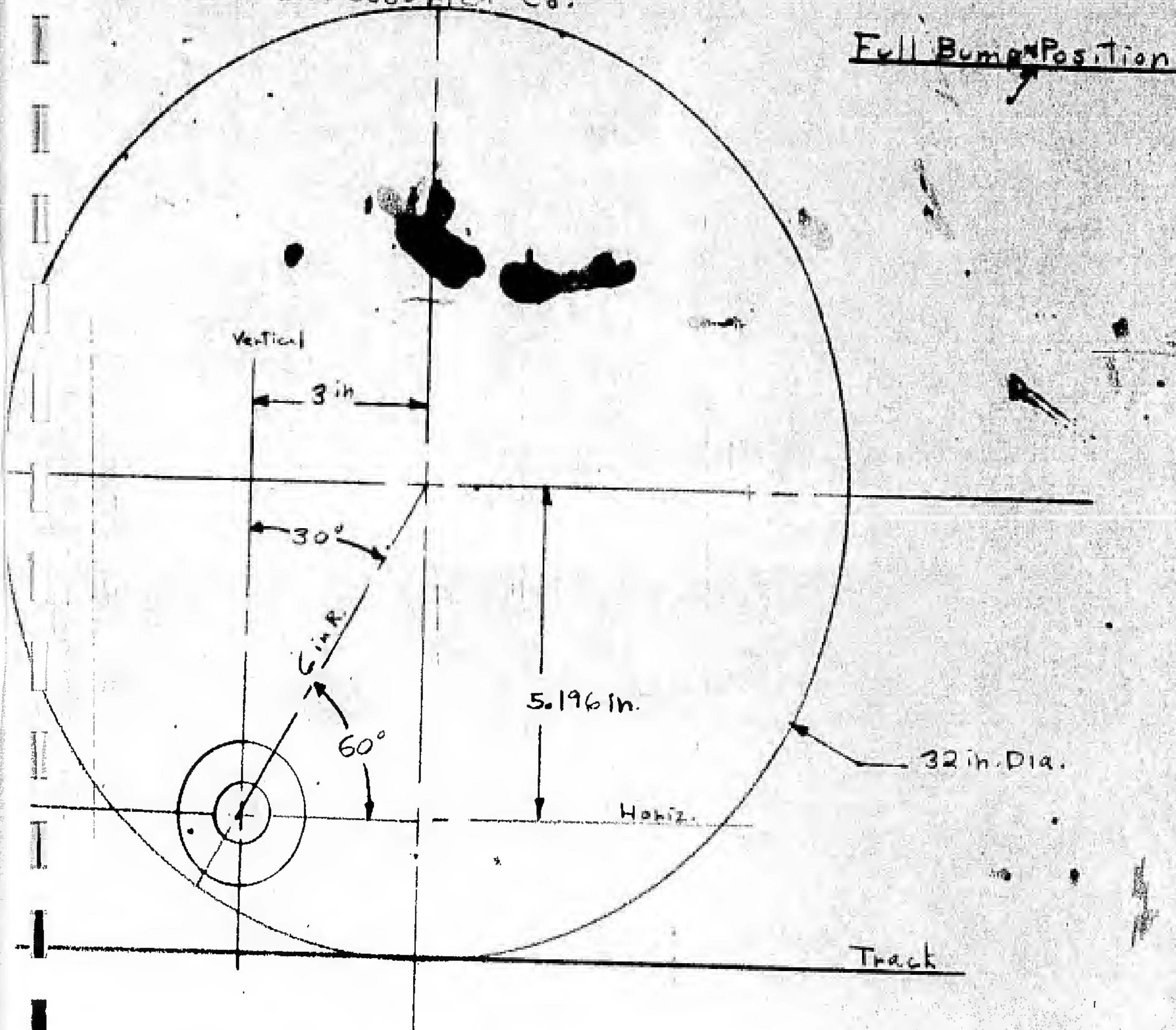
Static Spring Torque = (3.441) (3960) = 13,630 lb-in.

(From Page 7) Static windup = 23°

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO MICHIGAN

DATE 11/14 ... DATE ... SHEET ... OF 10.

Ref.- B.F. Goodrich Co.



Cosine 60° = 0.500 Effective Arm 6 X .500 = 3.00 in.
Sine 60° = 0.866 Elevation = 6 X .8660 = 5.196 in.

Windup at Static position (Page 71 + Page 76) = 23°
Angular Stroke Static To Bump = 60 + 55 = 115°
Therefore Windup at Bump = 138°

INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

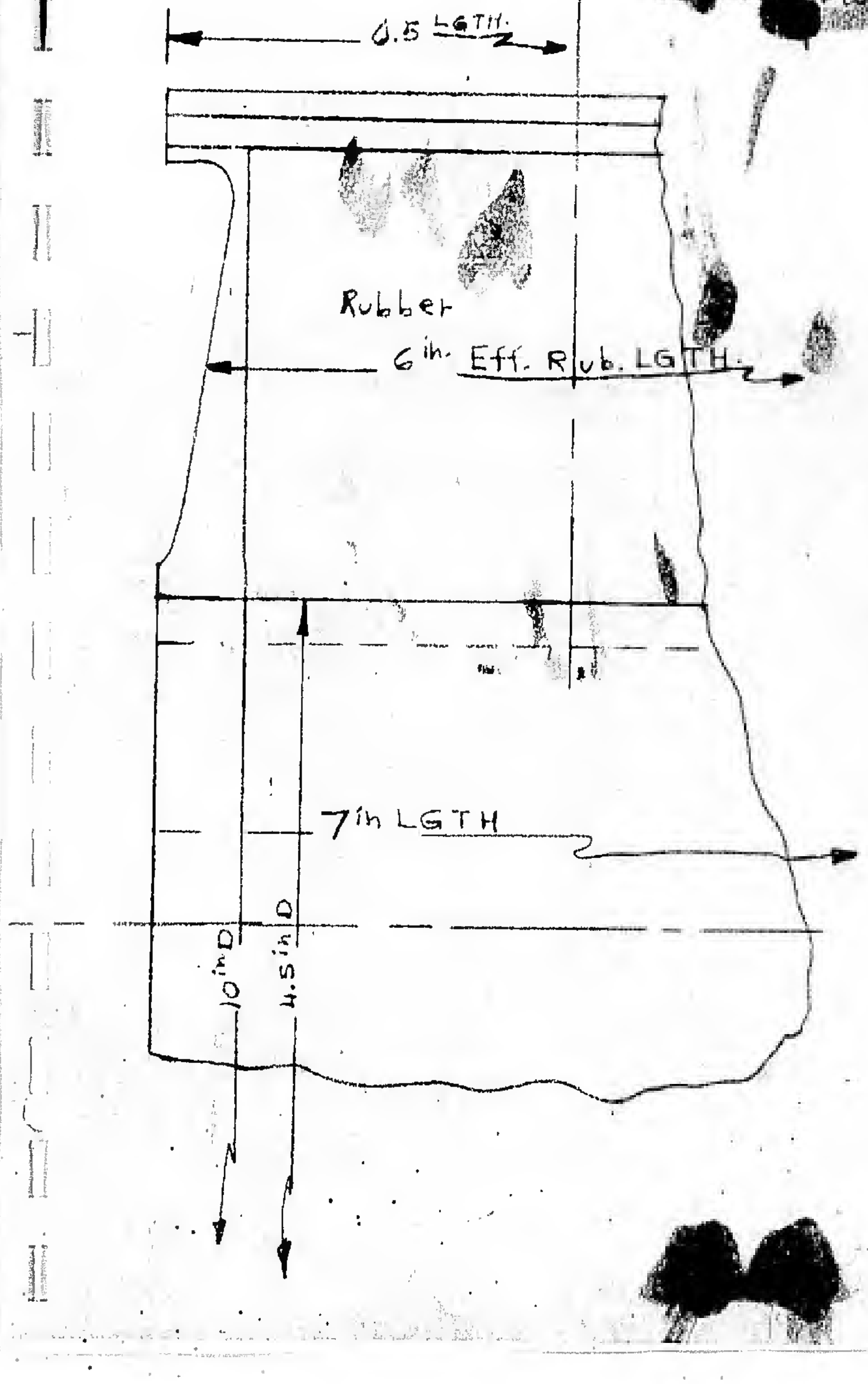
BY G. Nelson SUBJECT LVTPX11 SHEET NO. 4 OF 10
CHKD. BY Tors[illegible] [illegible]ings JOB NO. Nobs
[illegible] 4561-3

Torsilastic Spring:
(Ref.- SK-5207)

Ref. B.F. Goodrich Co.

Assembled Spring O.D. = 10.250 in
2 - 1/8 in half shells = 0.250 in
Rubber O.D. = 10.000 in.
Rubber I.D. = -4.500 in. (4.5 in Shaft Dia.)
5.500
Rub[illegible] wall = 5.500/2 = 2.750

[2.75/4.5 x 100 = 61% Wall] (Ref.)



INGERSOLL-KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

BY G. Nelson DATE 11/15/62 SUBJECT LVTPX11 SHEET NO. 5 OF 10
CHKD. BY ___ DATE ___ Torsilastic Spring Suspension Group JOB NO. Nobs 45613

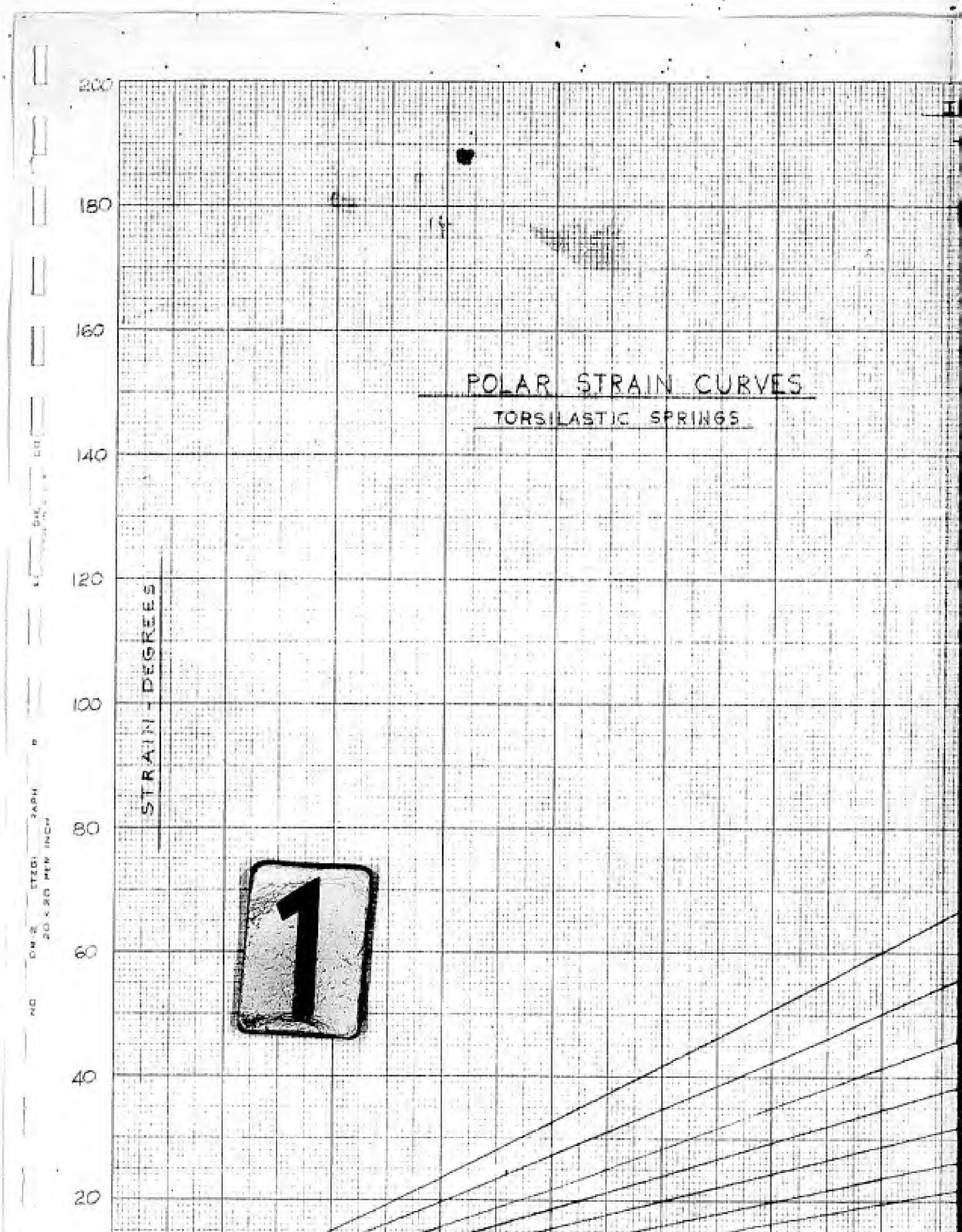
Torque vs. Windup Curve Data Ref B.F. Goodrich Co.

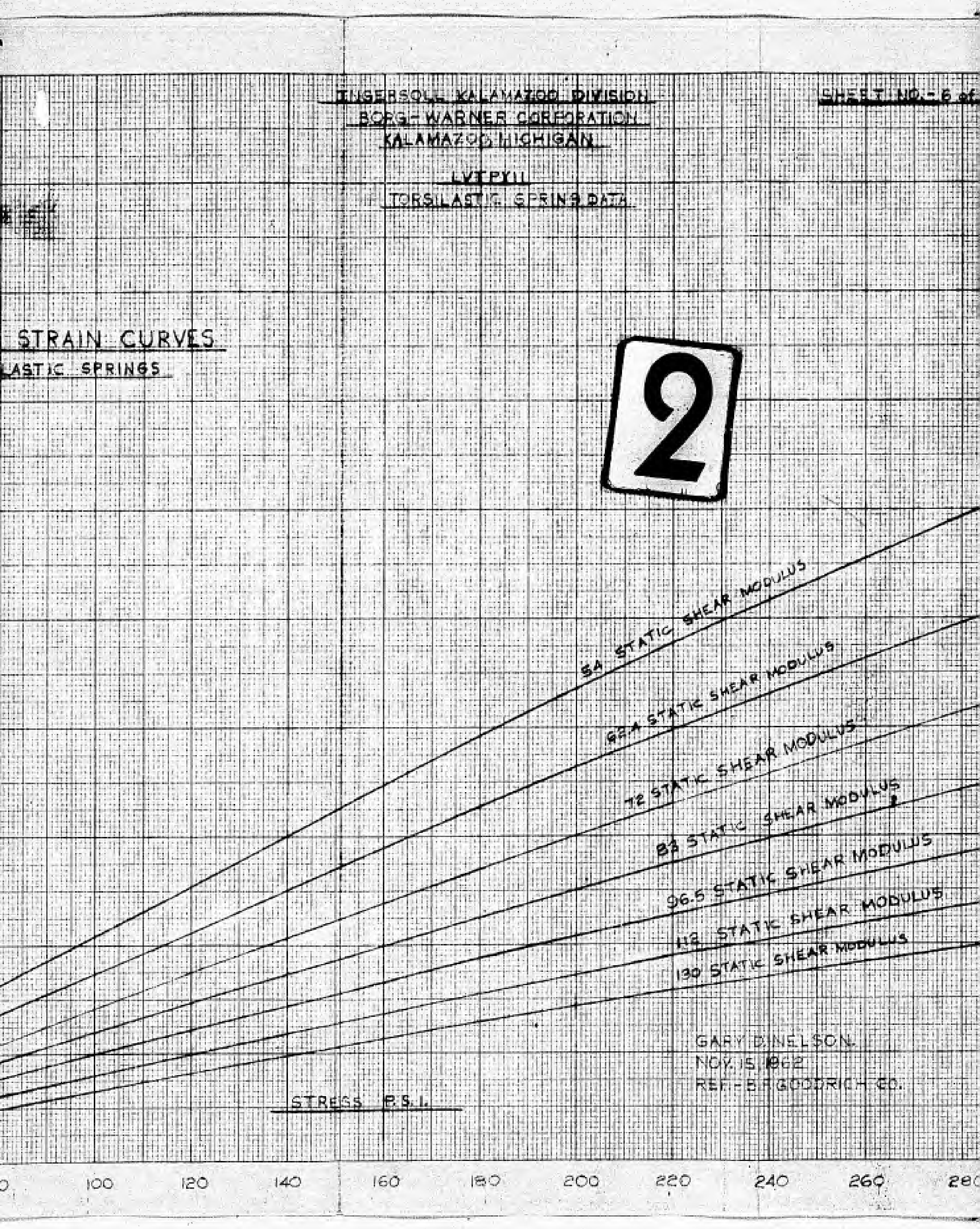
$$S_{ID} = \frac{\text{Torque}}{\frac{4.5}{2} \times 4.5\pi \times 6} = \frac{T}{190.85}$$

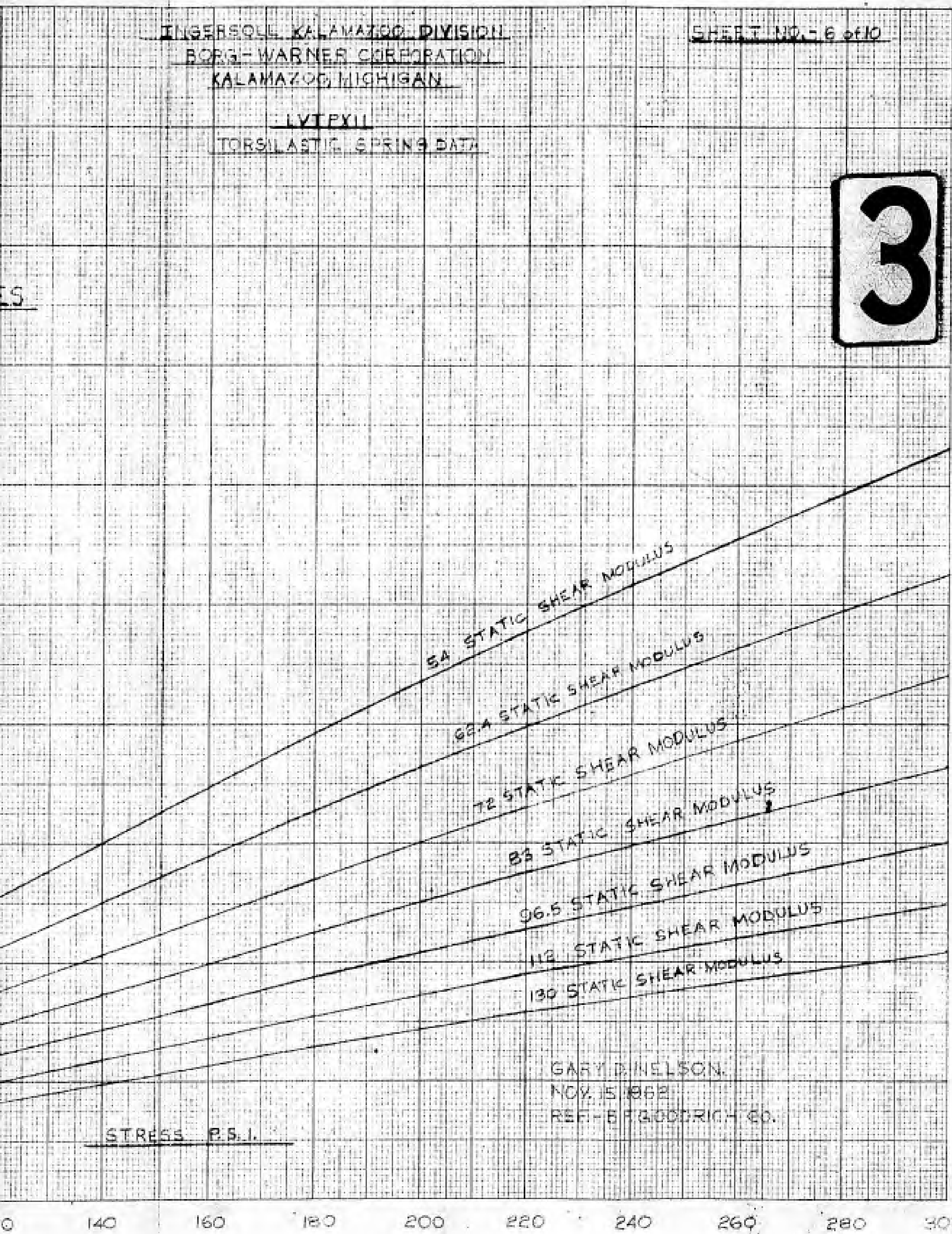
$$S_{OD} = \left(\frac{4.5}{10.00}\right)^2 \times S_{I.D.} = .2025 \times S_{ID}$$

Shear Modulus "G" = 54 (P.G)

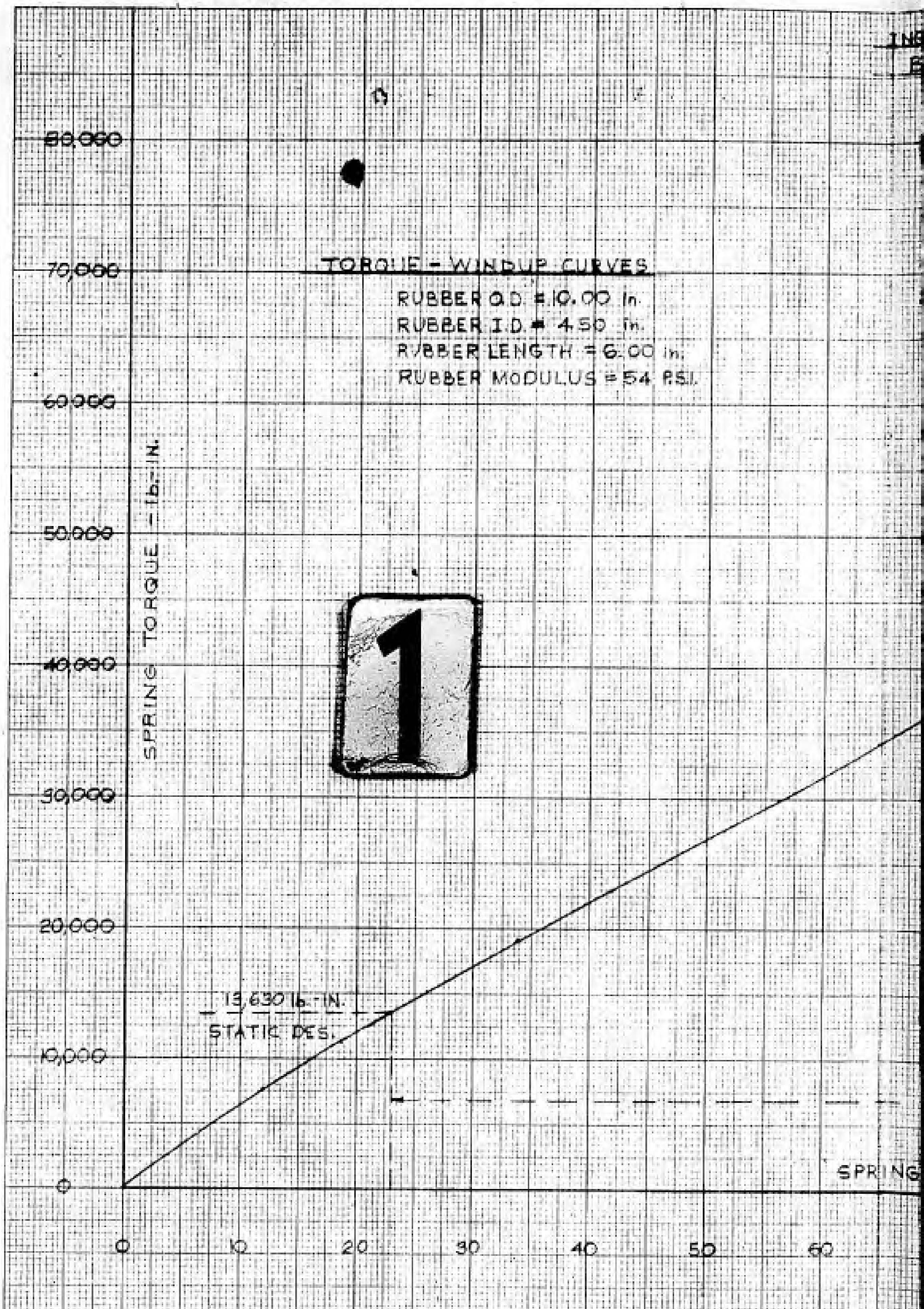
ASSIGN S_{ID} (P.S.I)	S_{OD} (P.S.I.)	TORQUE (lb-in)	α -	β =	θ
20	4.0	3,817	7.2°	1.6°	5.6°
40	8.1	7,634	15.0°	3.0°	12.0°
60	12.2	11,450	23.2°	4.4°	18.8°
80	16.2	15,270	32.0°	6.0°	26.0°
100	20.2	19,080	41.2°	7.3°	33.9°
120	24.3	22,900	50.9°	9.0°	41.9°
140	28.4	26,720	60.2°	10.6°	49.6°
160	32.4	30,540	69.6°	12.0°	57.6°
180	36.4	34,350	78.6°	13.8°	64.8°
200	40.5	38,170	87.3°	15.2°	72.1°
250	50.6	47,710	107.8°	19.1°	88.7°
300	60.8	57,260	126.8°	23.8°	103.0°
400	81.0	76,340	157.0°	32.6°	124.4°

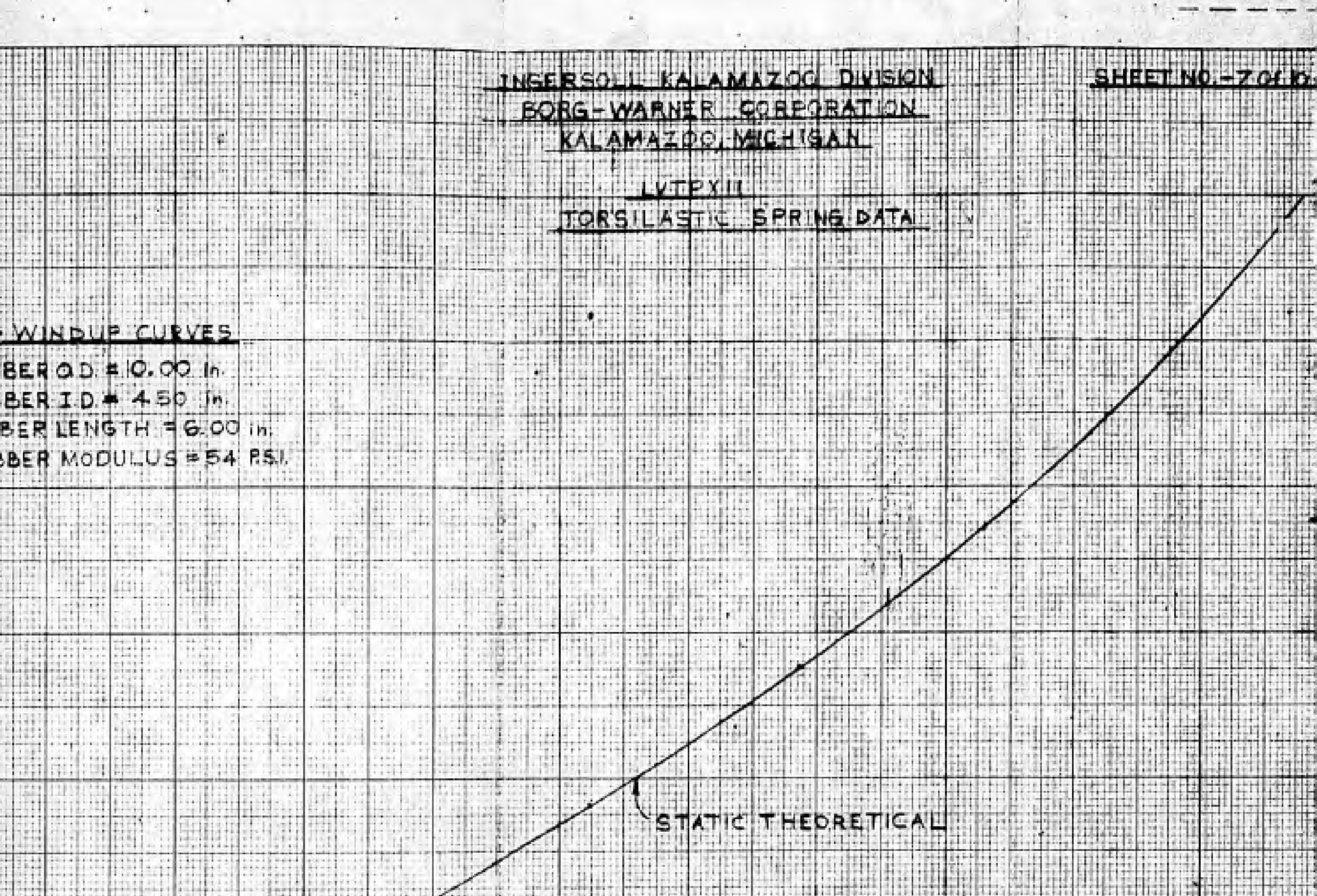

POLAR STRAIN CURVES
TORSILASTIC SPRINGS
STRAIN - DEGREES
STRES
200
180
160
140
120
100
80
60
40
20
0
20
40
60
80
100
120
140
1


INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN
LVTPX11
TORSILASTIC SPRING DATA
SHEET NO. - 6 of
STRAIN CURVES
LASTIC SPRINGS
54 STATIC SHEAR MODULUS
62.4 STATIC SHEAR MODULUS
72 STATIC SHEAR MODULUS
83 STATIC SHEAR MODULUS
96.5 STATIC SHEAR MODULUS
112 STATIC SHEAR MODULUS
130 STATIC SHEAR MODULUS
GARY D. NELSON
NOV. 15, 1962
REF. - B.F. GOODRICH CO.
STRESS P.S.I.
100
120
140
160
180
200
220
240
260
280

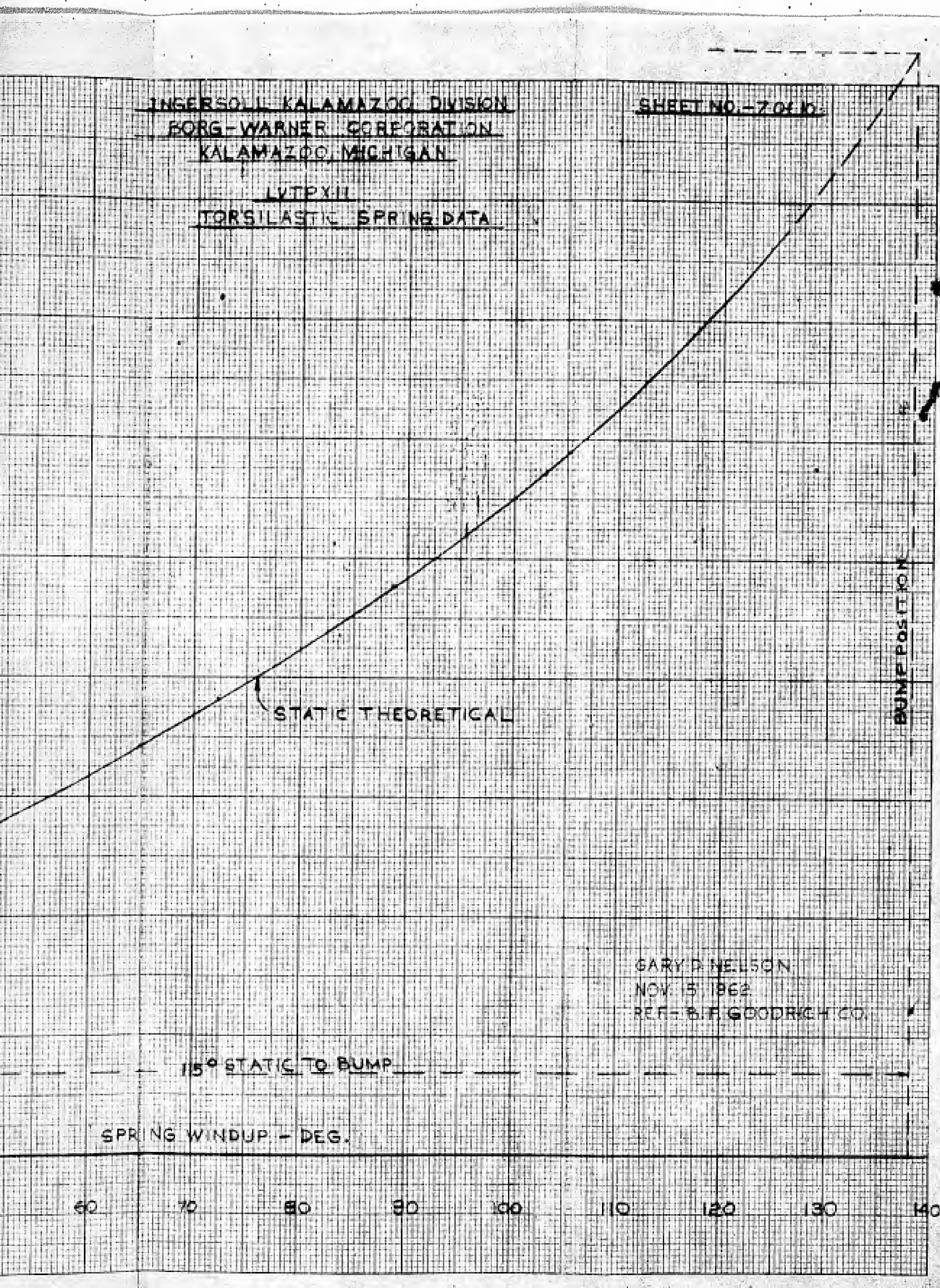


INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN
LVTPXII
TORSILASTIC SPRING DATA
SHEET NO. - 6 of 10
3
ES
54 STATIC SHEAR MODULUS
62.4 STATIC SHEAR MODULUS
72 STATIC SHEAR MODULUS
83 STATIC SHEAR MODULUS
96.5 STATIC SHEAR MODULUS
112 STATIC SHEAR MODULUS
130 STATIC SHEAR MODULUS
GARY D. NELSON
NOV. 15 1962
REF - B.F. GOODRICH Co.
STRESS P.S.I.
0
140
160
180
200
220
240
260
280
300


TORQUE - WINDUP CURVES
RUBBER O.D. = 10.00 In.
RUBBER I.D. = 4.50 In.
RUBBER LENGTH = 6.00 In.
RUBBER MODULUS = 54 P.S.I.
SPRING TORQUE - lb.-IN.
80,000
70,000
60,000
50,000
40,000
30,000
20,000
10,000
0
15,630 lb.-IN.
STATIC DES.
1
0
10
20
30
40
50
60



INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN
LVTPX11
TORSILASTIC SPRING DATA
SHEET NO. - 7 of 10
WINDUP CURVES
BER O.D. = 10.00 in.
BER I.D. = 4.50 in.
BER LENGTH = 6.00 in.
BBER MODULUS = 54 P.S.I.
STATIC THEORETICAL
2
GARY D. NELSON
NOV. 15, 1962
REF - B. F. GOODRIC
115° STATIC TO BUMP
SPRING WINDUP - DEG.
30
40
50
60
70
80
90
100
110
120


INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN
LVTP X-II
TORSILASTIC SPRING DATA
SHEET NO. - 7 of 10
STATIC THEORETICAL
BUMP POSITION
GARY D. NELSON
NOV. 15, 1962
REF.- B.F. GOODRICH CO.
115° STATIC TO BUMP
SPRING WINDUP - DEG.
60
70
80
90
100
110
120
130
140

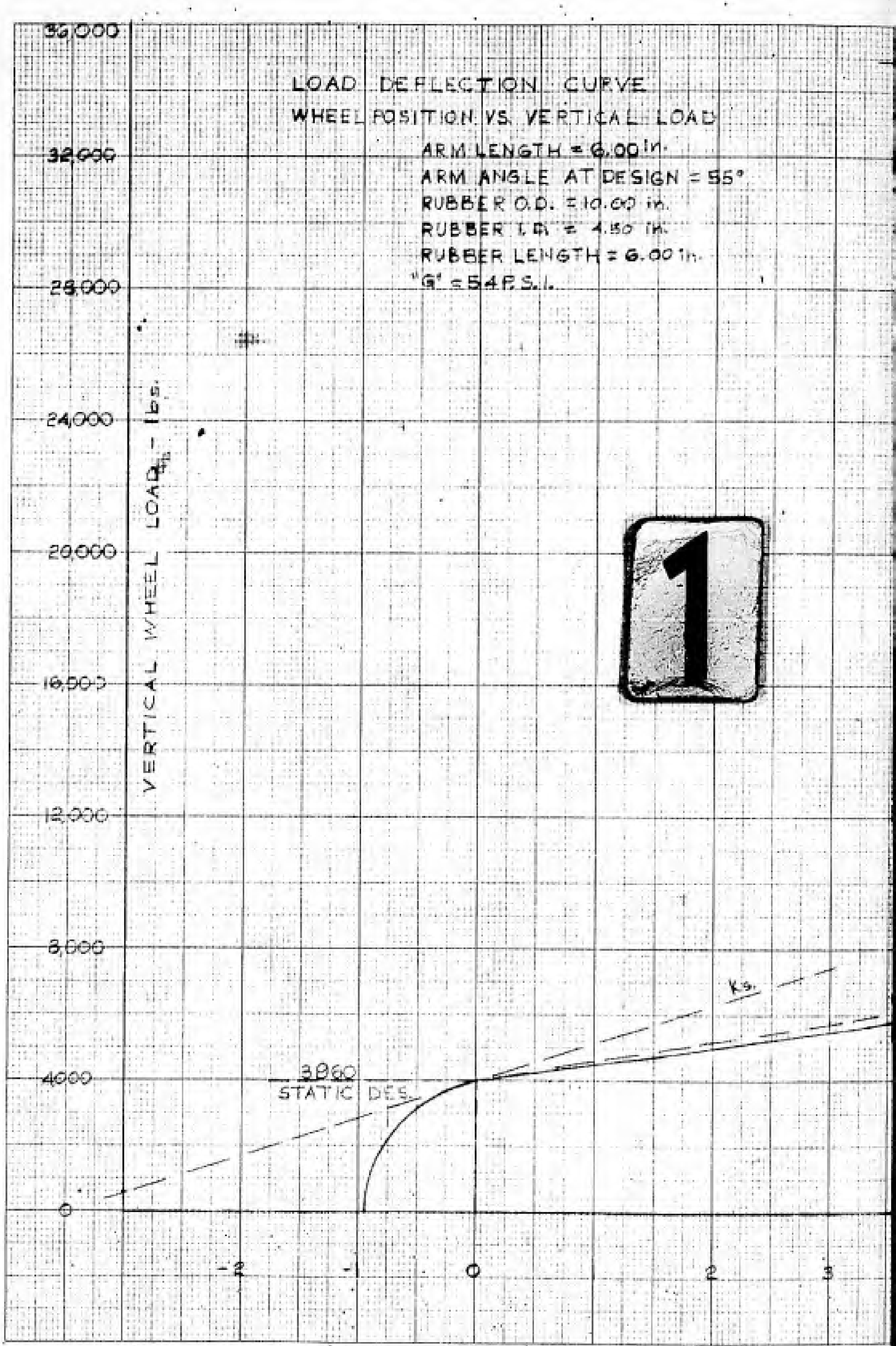
3

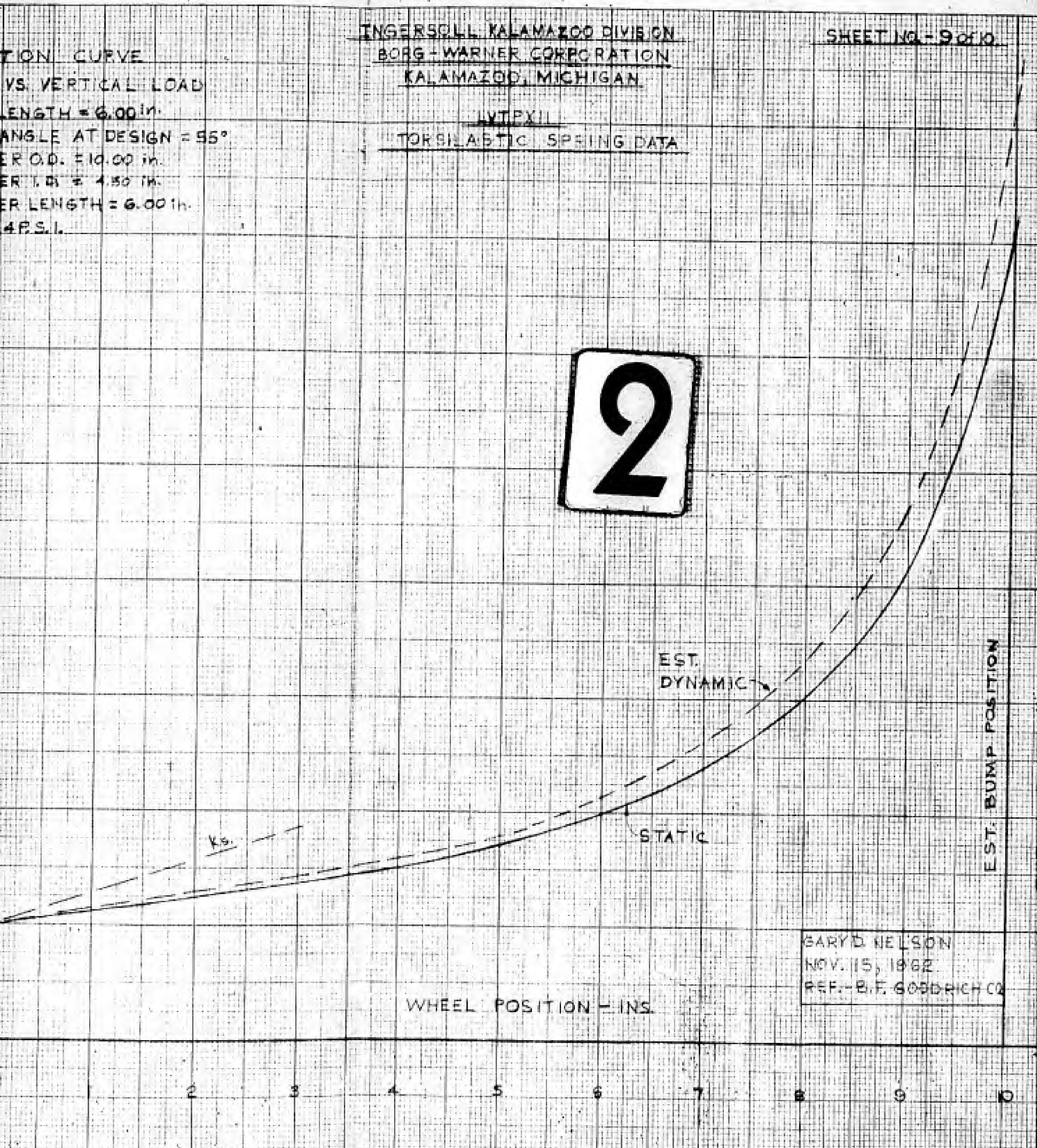
INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN

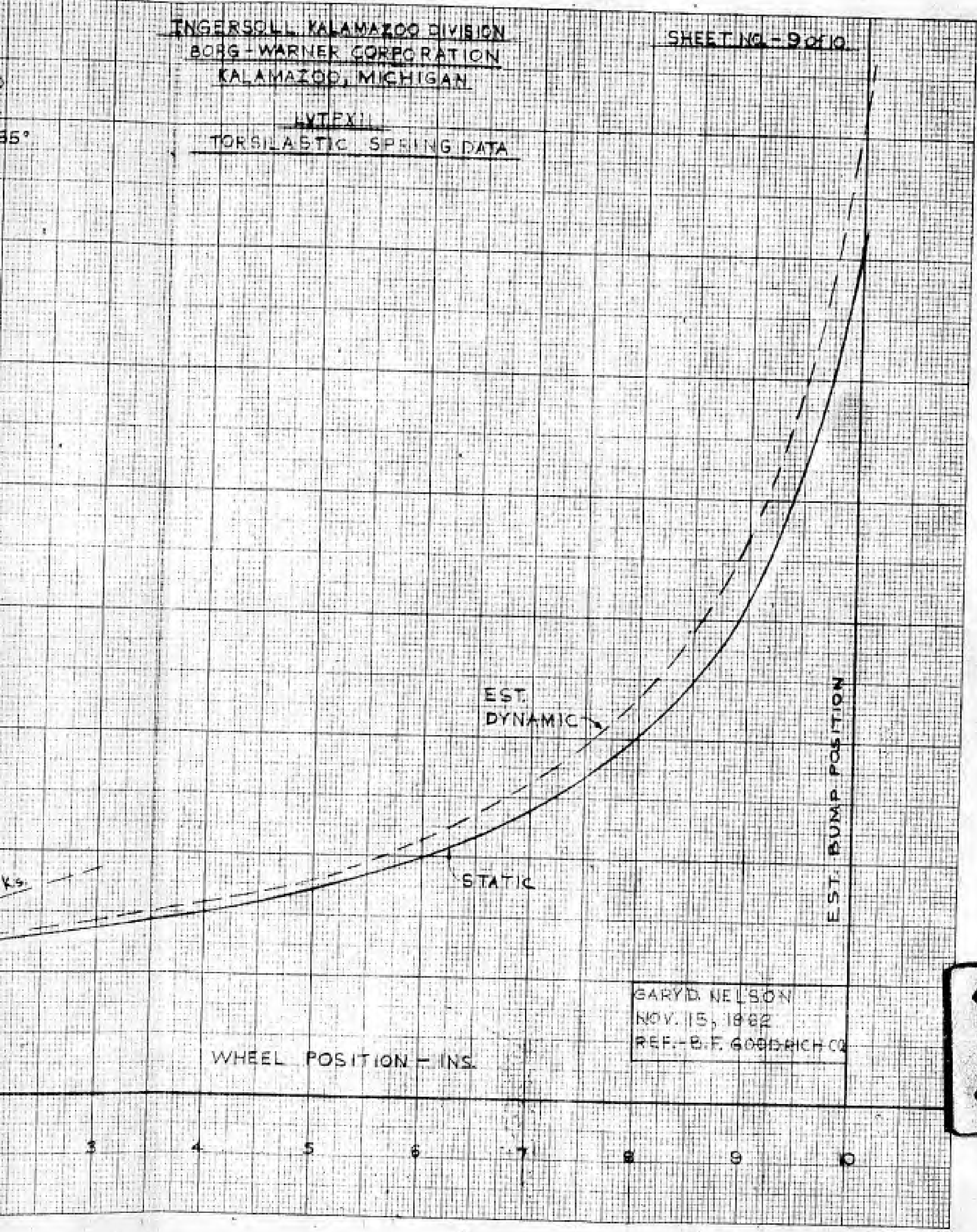
BY G. Nelson DATE 11/15/62 SUBJECT LVT PX II SHEET NO. 8 OF 10
CHKD. BY ______ DATE ______ Torsilastic Spring JOB NO. Nebs
Suspension Group 4561-3

Ref. B.F. Goodrich Co.

Wheel Position	Elevation inches	Sine Arm Angle	Arm Angle	Cosine Arm Angle	Eff Arm Length (in.)	Spring Torque (lb-in)	Spring Windup	Static Vert. Force (lbs)	
+10	5.085	.84750	57.95° (57')	.53066	9.184	90,000	133.95°	28,270	10
9	4.085	.68083	42.92° (55')	.73234	4.394	73,000	120.92°	16,610	9
8	3.085	.51417	30.93° (56')	.85777	5.147	62,000	108.93°	12,030	8
7	2.085	.34750	20.33° (20')	.93769	5.626	53,600	98.33°	9,530	7
6	1.085	.18083	10.42° (25')	.98352	5.901	47,500	88.42°	8,050	6
5	0.085	.01417	0.82° (49')	.99990	5.999	41,600	78.82°	6,930	5
4	-0.915	.15250	-8.77° (46')	.98832	5.930	36,500	69.23°	6,160	4
3	-1.915	.31917	-18.62° (37')	.94768	5.686	31,500	59.38°	5,540	3
2	-2.915	.48583	-29.01° (.6')	.87406	5.244	26,000	48.99°	4,960	2
1	-3.915	.65250	-40.73° (44')	.75775	4.546	20,700	37.27°	4,550	1
0	-4.915	.81915	-55.0°	.57358	3.441	13,630	23.00°	3,960	0
-1/2	-5.415	.90250	-64.48° (29')	.43077	2.585	8,400	13.52°	3250	
-3/4	-5.665	.94417	-70.46° (28')	.32942	1.977	4,800	7.23	2430	
-1	-5.915	.98583	-80.35° (21')	.16763	1.006	-1,600	-2.35°	-1600	


LOAD DEFLECTION CURVE
WHEEL POSITION VS. VERTICAL LOAD
ARM LENGTH = 6.00 in.
ARM ANGLE AT DESIGN = 55°
RUBBER O.D. = 10.00 in.
RUBBER I.D. = 4.50 in.
RUBBER LENGTH = 6.00 in.
"G" = 54 P.S.I.
VERTICAL WHEEL LOAD - lbs.
36,000
32,000
28,000
24,000
20,000
16,000
12,000
8,000
4000
0
3860
STATIC DES.
Ks.
-2
-1
0
2
3


TION CURVE
VS. VERTICAL LOAD
LENGTH = 6.00 in.
ANGLE AT DESIGN = 55°
ER O.D. = 10.00 in.
ER I.D. = 4.50 in.
ER LENGTH = 6.00 in.
4 P.S.I.
INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN
LVTPXII
TORSILASTIC SPRING DATA
SHEET NO. - 9 of 10
EST. DYNAMIC
STATIC
Ks.
EST. BUMP POSITION
GARY D. NELSON
NOV. 15, 1962
REF. - B.F. GOODRICH CO.
WHEEL POSITION - INS.
1
2
3
4
5
6
7
8
9
10


INGERSOLL KALAMAZOO DIVISION
BORG-WARNER CORPORATION
KALAMAZOO, MICHIGAN
TORSILASTIC SPRING DATA
SHEET NO. - 9 of 10
EST. DYNAMIC
STATIC
EST. BUMP POSITION
GARY D. NELSON
NOV. 15, 1962
REF.-B.F. GOODRICH CO.
WHEEL POSITION - INS.
3
4
5
6
7
8
9
10

3

Ref. B.F. Goodrich Co.

Static Vertical Rate at Design = k_s = 1140 lb./in. (Page 9)

with ratio $\frac{\text{Dynamic}}{\text{Static}} = 1.2$

k_D (Dynamic Vertical Rate) = 1.2 X 1140
k_D = 1368 lb/in

Nominal Dynamic Deflection = $\frac{3960}{1368}$ = 2.9 in.

$$\boxed{\frac{188}{\sqrt{2.9}} = 110.5 \text{ cpm Nat. Vert. Freq.}}$$

Static Vertical at Bump = 28,080 lbs.
Static Design = -3960 lbs.
24,120 lbs.

Dynamic Increment = (24,120)(Ratio)
= (24,120)(1.2) = 28,940 lbs.

Vertical Dynamic at Bump = 28,940 + 3960 = 32,900 lbs.

"G" load at 10 inch bump = $\frac{32,900}{3960}$ = 8.3 "G"s

Torque = 32900 lb. (3.184 in) = 104,750 lb.-in.
(P 8)

(P 5)

Rubber $S_{10} = \frac{104,750}{190.85}$ = 548 psi

S_{0D} = 548 X .2025 = 111.0 psi
(P 5)

6.0. FINAL DRIVE AND TRACK COMPENSATING SYSTEM

6.1. Hydraulic Track Compensator.

6.1.1. In the operation of track-laying vehicles, a basic problem has been inefficient track operation resulting from incorrect track tensioning. Correct track tensioning is a matter of control. Methods of controlling track tension under varying vehicle operating conditions have been developed, but no ideal track tension control has been developed to date.

6.1.2. In compliance with Specification SHIPS-A-4561 Ingersoll Kalamazoo Division proposed a dual force hydraulic track compensator which was intended to approach the desired ideal compensator. The proposed compensator design incorporated the following features not found on manual track tensioning devices:

A. Eliminates constant manual adjustment to compensator for track wear.

B. Facilitates track maintenance.

C. Provides for automatically eliminating slack track due to track envelope changes resulting from varying vehicle and track loads.

D. Holds to a minimum slack track produced by vehicle acceleration.

E. Holds sprocket climb or jump to an absolute minimum during steer by preventing slack from occurring at the sprocket.

F. Permits the track to climb the sprocket in an emergency without producing dangerously high track tensions.

6.1.3. The proposed track compensator design would have accomplished all of these design objectives. There was one important design feature which the proposed compensator lacked. Because the compensator was a dual force device, it operated at two force levels. One force level was used to maintain normal operating track tension. The second force level was used to control track tension during vehicle steering. Because only one force level was used for normal track tensioning, the track tension used had to be high enough to meet the severest operating requirements, excluding steering. This meant that for all vehicle operation except the maximum torque range, the track was running under excessive tension and was therefore much less efficient than desired. A study of the proposed track compensator was made during Phase I of contract SHIPS-A-4561 to see if the proposed compensator design could be improved. The study was made treating the entire final drive, track compensator, and tracks and suspension system as a whole. This study revealed that increased track efficiency was a matter of compensator control and that the necessary control features could be designed into the proposed track compensator system while maintaining all of its original design features. The original hydraulic ram compensator could be used with little modification. The remaining changes required were primarily in the area of controls.

6.2. General Description of Variable Force Track Compensator.

6.2.1. The variable force track compensator performs the following functions:

A. Automatically provides track tension proportional to vehicle drive load.

B. Maintains a minimum initial track tension to prevent slack track due to track envelope changes resulting from varying vehicle and track loads.

C. Automatically reduces track compensating forces to prevent excessive track tension due to roadwheel geometry variations.

D. Reduces track slack caused by vehicle acceleration to a minimum.

E. Holds sprocket climb or jump to a minimum during steer by preventing slack track from occurring at sprocket.

F. Prevents sprocket climb during reverse operation or vehicle braking.

G. Provides optimum track tension during water operation.

H. Automatically limits maximum track tension during emergency conditions during all modes of operation.

I. Automatically compensates for track stretch and/or wear.

J. Makes track maintenance easier.

K. Provides for emergency tensioning of track in event of vehicle power failure.

6.2.2. The variable force hydraulic track compensator is composed of three basic units. There are:

A. The compensator cylinder.

B. The compensator control valve assembly.

C. The control valve positioner assembly.

6.2.3. These three basic units are assembled together into one easily handled assembly. This assembly is the track compensator assembly and is shown in drawing SK-5244.

6.2.4. There are two track compensator assemblies used on the LVTPX11 vehicle, one for the port track and one for the starboard track. One common track compensator assembly is used for both the port and starboard tracks, there being no difference in physical arrangement between the two. Each track compensator is mounted to a hull connection on one end and to an arm on the final drive gearbox at the other. This arrangement is shown on drawing SK-5223. Each end of the track compensator is mounted on a ball bushing to eliminate any bending forces from being transmitted from or to the compensator assembly.

6.3. Method of Operation.

6.3.1. In operation, the track compensator controls track tension by applying a reaction torque to the final drive gearbox. Figure 6-1 shows the forces acting on the final drive gearbox due to track tension. These forces are:

F_1 = Force due to tension in the upper track.

F_2 = Force due to tension in the lower track.

6.3.2. These forces act on the final drive through the track drive sprocket. The final drive is mounted on bearings in the hull and is therefore free to rotate in its mounting. The center of the track drive sprocket is mounted on an arm with a radius of eight inches from the centerline of the final drive mounting bearings. This arrangement puts the centerline of the sprocket eccentric with respect to the centerline of the final drive mounting bearings. Thus, the forces created by track tension act at the centerline of the sprocket and generate turning moments about the centerline of the final drive gearbox mounting bearings. These turning moments tend to rotate the final drive gearbox around an axis through its mounting bearings.

6.3.3. In addition to the turning moments created by the track tension, the torque input to the final drive also creates a turning moment on the final drive gearbox..

6.3.4. In order to maintain or increase the tension in the tracks and to balance

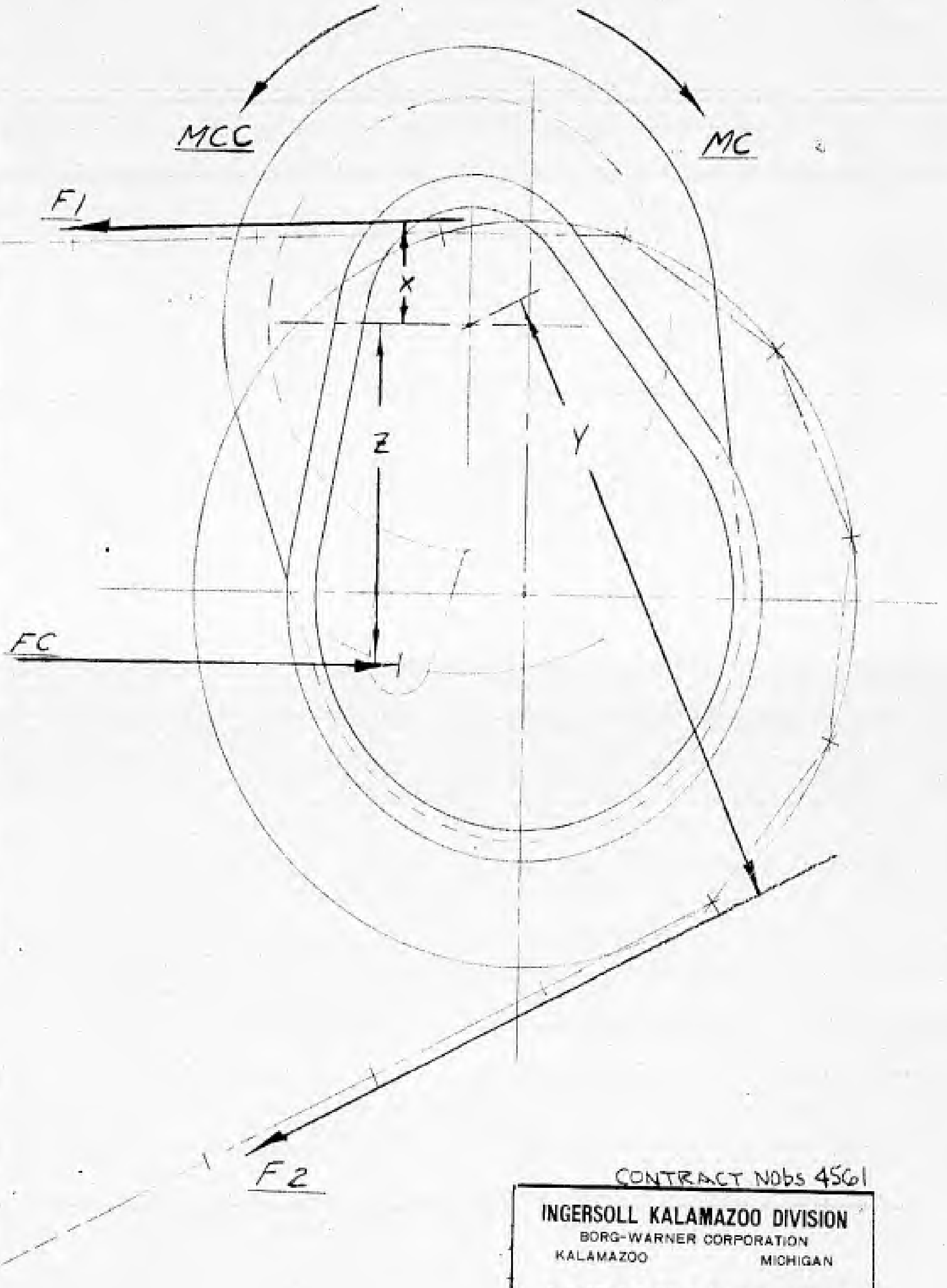


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FIGURE 6-1

the turning moment on final drive gearbox, a reaction torque must be applied to the final drive. This reaction torque is provided by the track compensator. Except for track tensioning during steering, the track compensator functions automatically to balance variations in input torque, to maintain a minimum track tension, to relieve excessive track tension, and to compensate for track wear and stretch.

6.3.5. All of the automatic functions of the track tensioner depend on control of hydraulic fluid under pressure into or out of the track compensator cylinder to maintain a pressure within the cylinder which will provide the desired reaction torque on the final drive gearbox.

6.3.6. Control of the track compensator is established in the following manner: First a reaction torque is applied to the final drive gearbox to provide an initial track tension. The initital track tension is determined by the reaction torque acting on the final drive gearbox, which in turn is determined by the force generated by the track compensator cylinder, which in turn is determined by the actuating pressure applied to the compensator cylinder. The initial track tension is a function of the pressure applied to the track compensator and is controlled by controlling the initial pressure.

6.3.7. Initial track tensioning pressure is applied to the track compensator when the vehicle is standing still (static) and the tracks are slack. The application of the initial tensioning pressure to the compensator generates a force (F_c figure 6-1) which acts on the final drive gearbox creating a turning moment about the axis of the final drive gearbox mounting bearings. This turning moment rotates the final drive gearbox and the track drive sprocket until the track becomes tight enough to generate tension forces (F_1 and F_2 figure 6-1) which will act on the final drive gearbox in such a manner as to create a net turning moment about the axis of the final drive gearbox mounting bearings which is equal to the turning moment generated by the hydraulic compensator.

6.3.8. When the track tension forces are equal to reaction force of the hydraulic compensator, the turning moments about the axis of the final drive gear-box mounting bearings are balanced and the final drive gearbox is in static equilibrium. This condition (See figure 6-1) is expressed by the equation:

$$\sum M_B = -F_1X + F_2Y - F_cZ = 0$$

X, Y, and Z are effective lever arms

F_1 = Upper track tension

F_2 = Lower track tension

F_c = Compensator force

Clockwise moments are positive (+)

6.3.9. When the compensator force is applied to the final drive gearbox and the gearbox is in static equilibrium, two conditions exist. These are:

A. The tracks are under tension and this tension is equal in both tracks.

B. The final drive gearbox has assumed a definite position with respect to the vehicle hull.

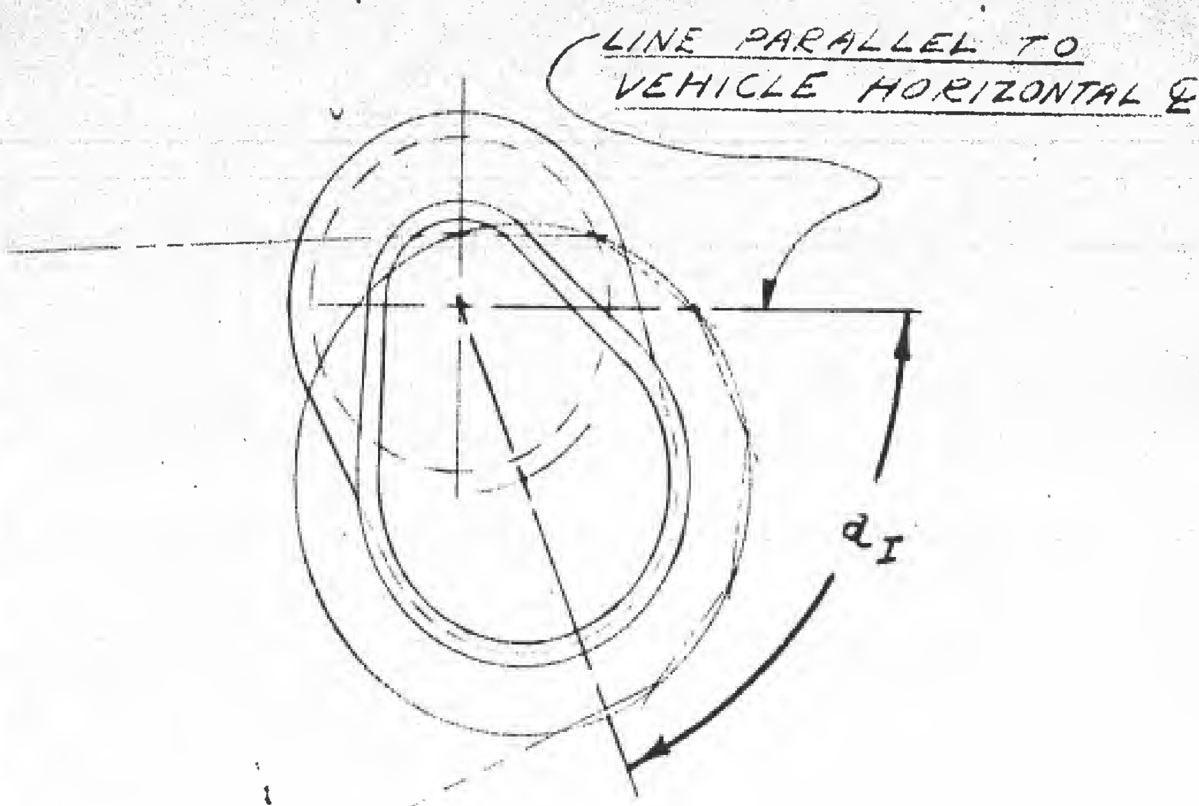
6.3.10. The control of the track compensator is based upon the positional relationship between the hull and the final drive gearbox. This is accomplished in the following manner:

6.3.11. A line passed through the center of the track sprocket and the axis of the final drive gearbox mounting bearings (which will be referred to as the final drive reference line) will form an angle (α) with a line parallel to the true horizontal centerline of the vehicle. As the final drive gearbox rotates about its mounting, axis angle (α) will increase or decrease depending upon the direction of rotation.

6.3.12. For any given initial tensioning pressure (P_{CI}) applied to the compensator, an initial track tension (F_{1I} for the upper track and F_{2I} in the lower track) will be created and the final drive gearbox will be in equilibrium. Under this equilibrium condition some angle (α_I) due to the initial compensator force (P_{CI}) will exist between the final drive reference line and the horizontal vehicle centerline. This angle is called the "set" angle and the position of the final drive gearbox is referred to as the "set" position of the gearbox. (See figure 6-2). It is readily apparent that there can be any number of set angles and positions. This is due to the fact that the set angle depends on:

A. Desired initial track tension - If the initial track tensioning pressure in the track compensator is increased, the final drive gearbox will rotate counterclockwise decreasing the track envelope until the deflection of the roadwheel springs due to the increased track envelope creates sufficient track tension to balance the increased compensator force. In this case rotation of the final drive gearbox counterclockwise will decrease the set angle I_I and create a new set angle (α_{II}) at the new final drive gearbox equilibrium point.

B. Track wear and stretch - As the track wears or stretches, the effective pitch length of the track will increase. If the track is under initial tension and the pitch length of the track is increased due to wear or track stretch, and the track sprocket remains in the position of initial track tension, that is α_I does not change, then the increased track pitch length will permit the track to go slack thereby reducing the track tension. In order to reestablish the initial track tension lost due to track wear or stretch, it will be necessary to assume a new set angle (α_{II}). Refer to figure 6-2.



A

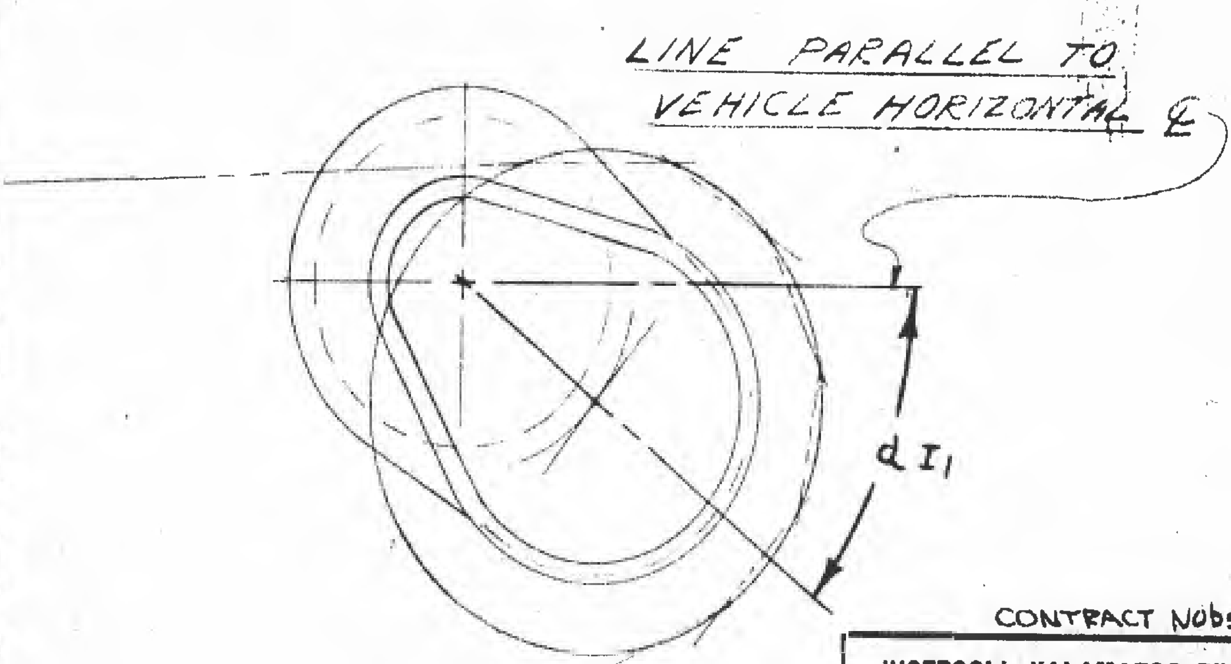


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FIGURE 6-2

6.3.13. The variable force track compensator is controlled in the following manner: With the vehicle in the static (stopped) position, initial track tension pressure is applied to the track compensator cylinder. (Note: This pressure may be set to give any desired initial track tension from zero to the maximum the track compensator will develop.) The application of initial track tensioning pressure will cause the final drive gearbox to assume some set angle (α_I) with the vehicle horizontal centerline. The track compensator will automatically sense this set angle and will always try to maintain this set angle regardless of changes in vehicle loading or track tension due to increased vehicle tractive effort. If track wear or stretch should increase the pitch length of the track after the initial tensioning, the compensator will reset the final drive gearbox to maintain the initial track tension. This action will create a new set angle and position which the compensator control will sense and try to maintain until further track wear or stretch reduces the initial track tension thus requiring the establishment of a new set angle. This action will continue until the compensator has reached its limit of operation.

6.4. Track Compensator and Vehicle Operating Conditions.

6.4.1. A description of the hydraulic track compensator action for various vehicle operating conditions is given below:

A. <u>Establishing Initial Track Tension</u> - This condition exists whenever it is necessary to establish or reestablish the initial track tension. The compensator and compensator circuit work as follows:

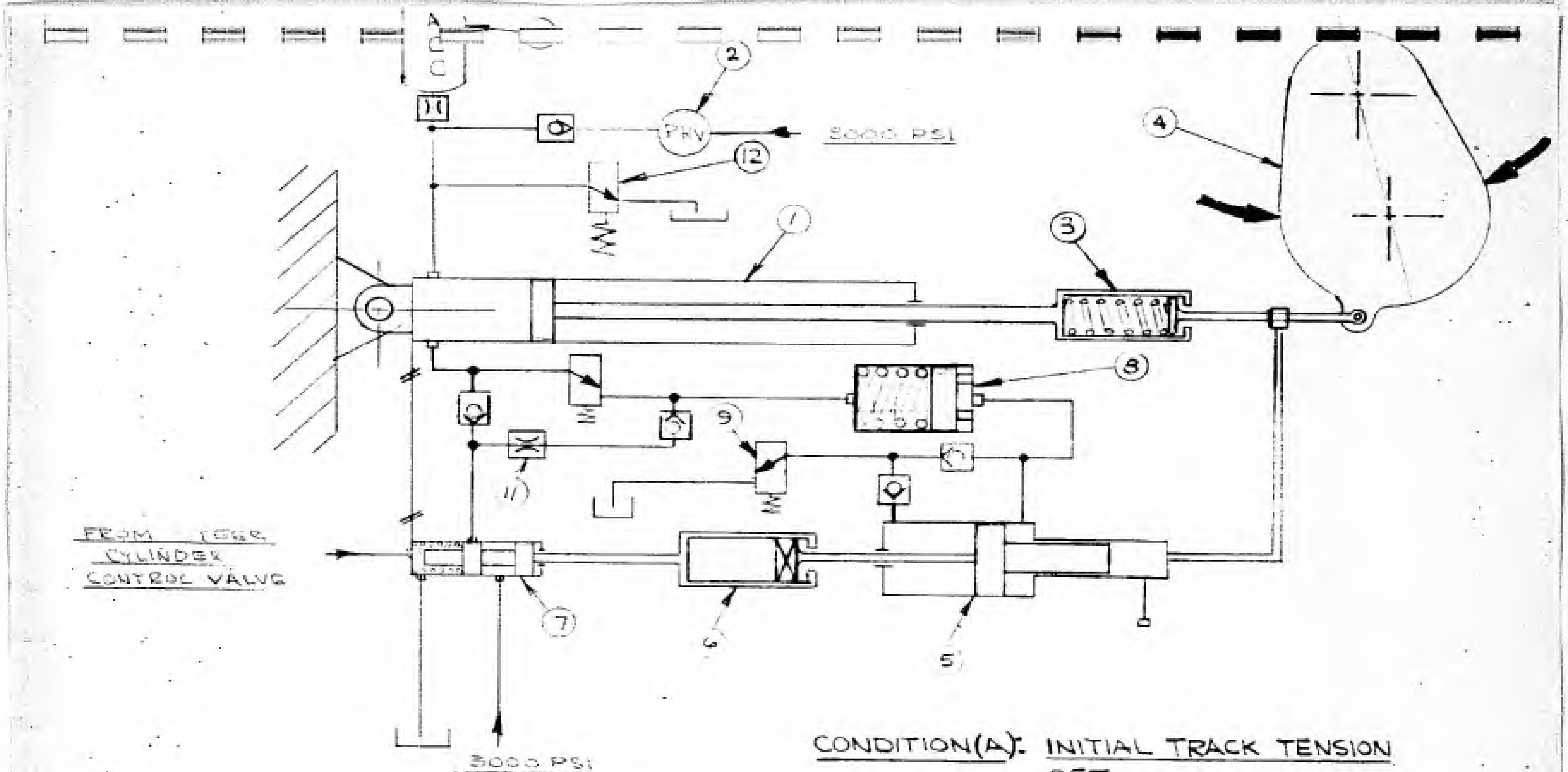
1. Hydraulic fluid at a pressure which will give the desired initial track tension is applied to the track compensator cylinder (1) figure 6-3, from the vehicle hydraulic system through the pressure reducing valve (2). The initial tensioning pressure moves the compensator cylinder piston outward (extends) loading the spring cage (3) and applying force to the final drive gearbox (4), rotating the final drive gearbox counterclockwise, moving the track sprocket against the track, and thereby tightening the track.

2. As the final drive gearbox rotates counterclockwise the following control valve position sensor action takes place:

   a. The position sensor cylinder (5), friction slide (6), and control valve spool (7) move to the right until the valve spool is stopped by the valve housing.

   b. The valve positioner continues to move to the right with the friction slide slipping until the friction slide reaches its right hand stop.

   c. The valve position sensor continues to move to the right. The valve housing and friction slide stop make



CONDITION(A): INITIAL TRACK TENSION SET.

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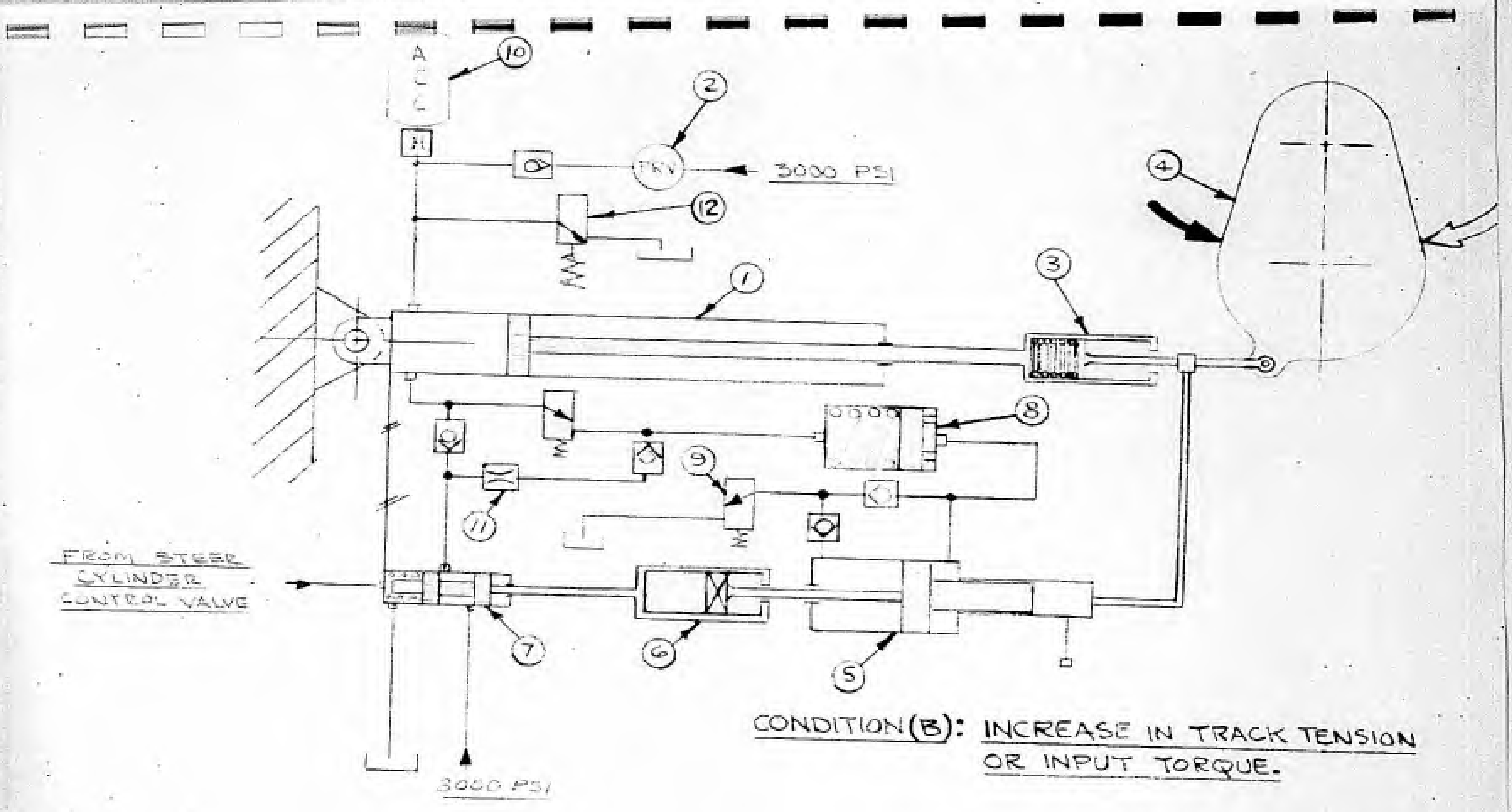
FIGURE 6-3A



CONDITION (B): INCREASE IN TRACK TENSION OR INPUT TORQUE.

FIGURE 6-3B

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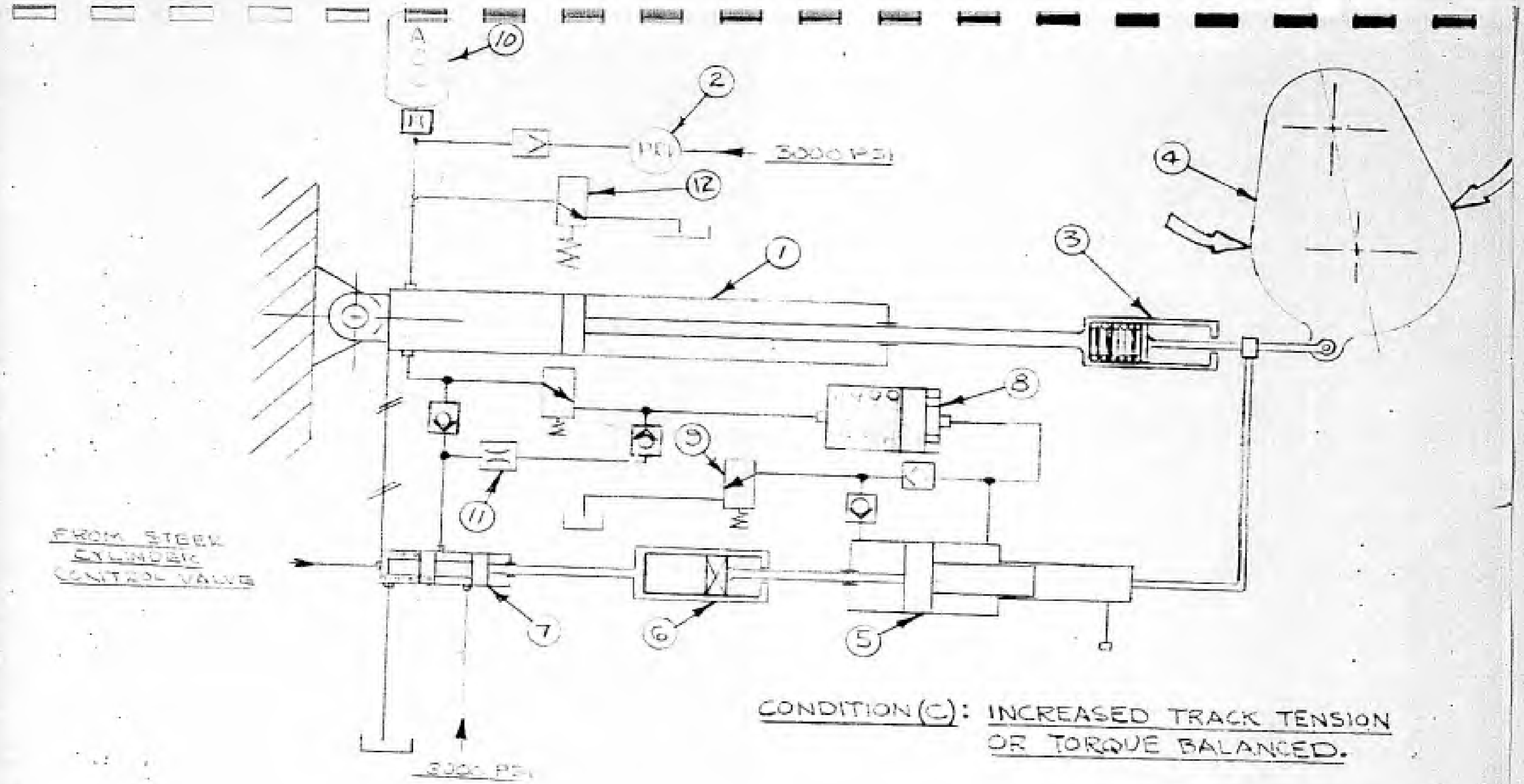

CONDITION (C): INCREASED TRACK TENSION OR TORQUE BALANCED.

FIGURE 6-3C

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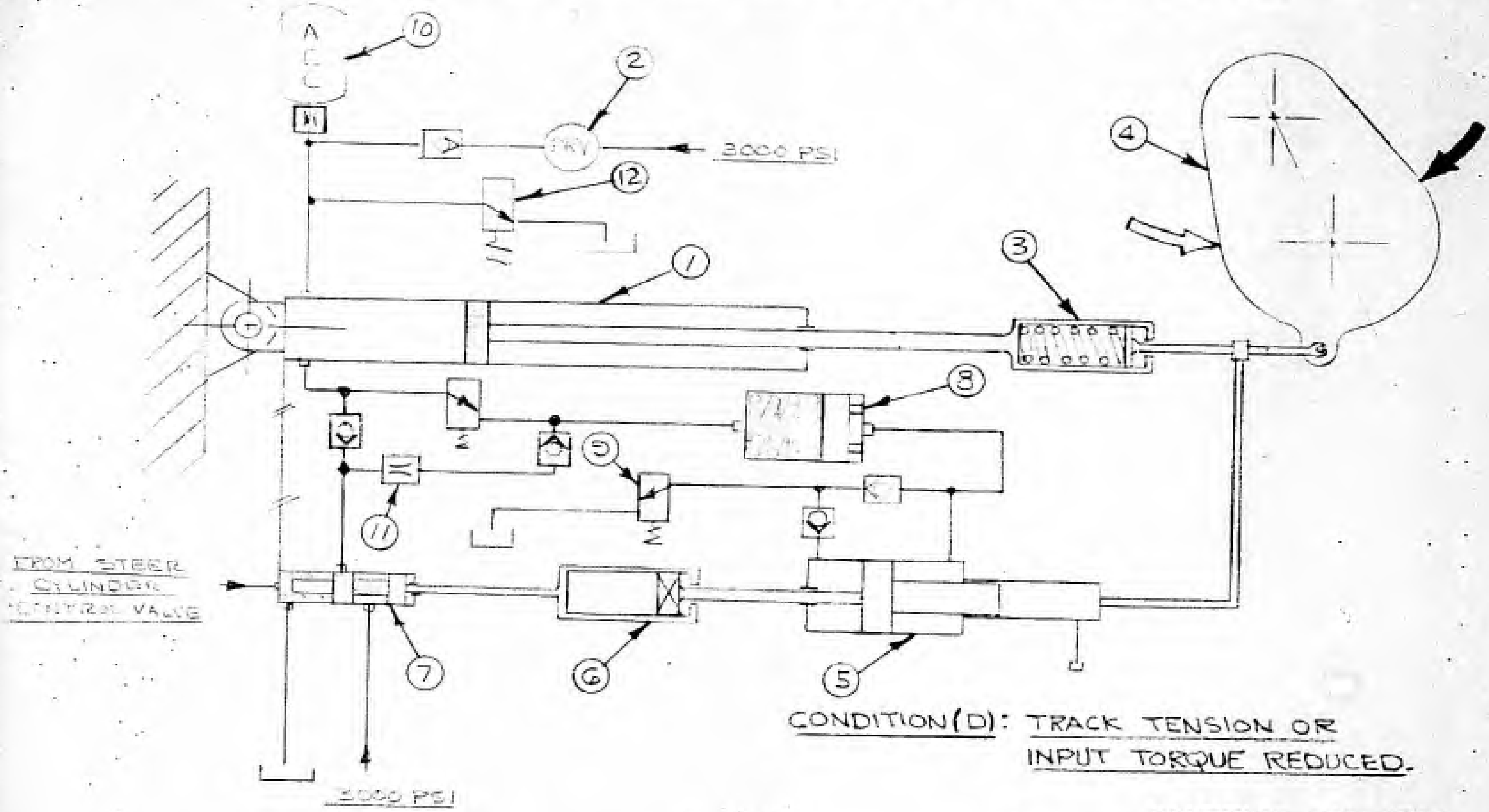

CONDITION (D): TRACK TENSION OR INPUT TORQUE REDUCED.

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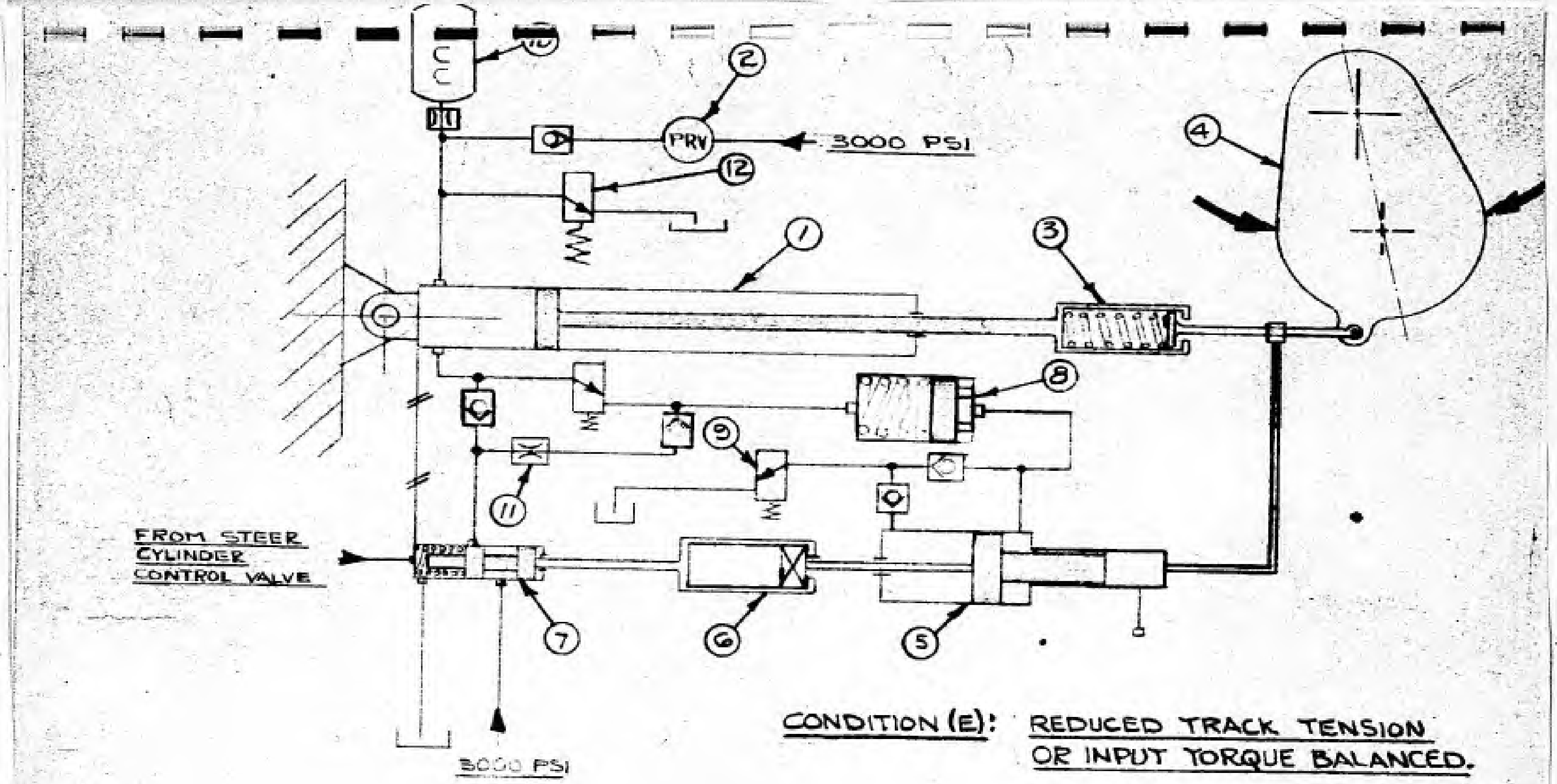
FIGURE 6-3D



CONDITION (E): REDUCED TRACK TENSION OR INPUT TORQUE BALANCED.

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FIGURE 6-3E

the control valve linkage go solid. When the valve linkage is solid, further movement of the final drive gearbox moves the cylinder of the control valve position sensor to the right. This action causes fluid to be pressurized in the left hand chamber of the sensor cylinder. The position sensor cylinder assembly is designed so that piston displacement is greater in the left hand chamber than in the right hand chamber. As the sensor cylinder is moved to the right, fluid is displaced into the right hand chamber and into the reset cylinder (8). The reset cylinder piston moves against a stop. Excess fluid from the sensor cylinder and reset cylinder is dumped over a relief valve (9). This action continues until the initial tensioning pressure is reached at which time sequence valve (10) opens allowing fluid to flow to the left hand side of the reset cylinder. A restrictor (11) in the sequence valve discharge line creates a pressure drop which causes the reset cylinder piston to move right, displacing the fluid in its right hand chamber into the right hand chamber of the position sensor cylinder. This action kicks the position sensor piston and the valve spool to the left, thereby shutting off fluid flow from the sequence valve and the track compensator cylinder. The control valve is now set and further track compensating action is controlled by the compensator control valve.

B. Controlling Track Tension During Vehicle Operation - Once initial track tension has been established and the track compensator control valve position sensor set, further track compensation is controlled by the track compensator control valve as long as total track tension forces do not drop below the initial track tension. The track compensator control valve controls track tension in the following manner:

1. An increase in input torque to the final drive or an increase in track tension due to a change in the track envelope will increase the clockwise turning moments on the final drive gearbox. This action will compress the spring cage (3) and move the valve position sensor and valve spool to the left. Movement of the valve spool to the left allows fluid under pressure from the hydraulic system to flow to the track compensator cylinder, increasing the force on the spring cage, which in turn reacts to overcome the turning moment on the final drive gearbox, rotating it counterclockwise and moving the control valve spool to the right until flow is cut

off to compensator cylinder.

2. A reduction in the final drive input torque or a decrease in track tension due to a change in track envelope will cause a reduction in the clockwise turning moments acting on the final drive gearbox. Because the spring cage (3) is under compression, a reduction in clockwise turning moments will cause the final drive gearbox to rotate couterclockwise. This action will cause the inner compensator control valve spool to move to the right against its restoring spring, opening the discharge side of the sequence valve to the tank. If the fluid pressure in the compensator cylinder is higher than the sequence valve setting, fluid will flow from the compensator under pressure created by the spring cage force. As the fluid is displaced from the compensator cylinder, the loading on the spring cage will drop and the counterclockwise turning moment on the final drive gearbox will be reduced until the clockwise turning moment acting on the final drive gearbox rotates it clockwise to return it to the set position, at which point the compensator control valve inner spool has moved to the left and shut off the fluid flow from the sequence valve, thereby holding the compensator cylinder from moving

and further reducing the spring cage loading and the reaction moment on the final drive gearbox.

C. Track Excess Tension Protection - In the event that any vehicle operating condition could create a track loading condition where track tension could be raised to a level where damage to the track or drive system components might take place, it is desirable that provision be made to relieve this condition. Such a provision has been made in the variable force track compensator design. This is how it works:

1. Any condition which will create an excessive track tension (for example a rock between the track and the sprocket) will increase the clockwise turning moment on the final drive gear-box greater than the maximum counterclockwise moment that the compensator can generate. Under this condition the spring cage will go solid, applying the full force developed by the excessive clockwise moment to the compensator cylinder. This action will cause the control valve spool to move left, applying full system pressure to the track compensator cylinder. If the counterclockwise reaction moment created by the full system pressure acting on the compensator piston is not great enough to overcome the counterclockwise moment acting on the final drive gearbox, the pressure in the cylinder

will rise until the relief valve (12) opens relieving any excess cylinder pressure. This action allows the compensator to maintain a constant counterclockwise reaction turning moment on the final drive gearbox while at the same time permitting the compensator cylinder to collapse, allowing any obstruction in the track system to clear itself. Whenever the obstruction or track overload condition is relieved, the accumulator (10) will apply a restoring pressure along with the system pressure applied to the compensator through the control valve override spool.

2. This pressure will return the compensator to the original control valve position sensor set position at which time the control valve override spool will become ineffective and the regular control valve spool will resume control of the hydraulic compensator.

D. Controlling Track Tension During Vehicle Steering and Backing - During vehicle steering, maximum reaction force is applied by the track compensator to the outside vehicle track. This is accomplished by applying full hydraulic system pressure to the track compensator circuit from the vehicle steering valve. The steering pressure is applied to compensator control valve in such a manner that the control valve and circuitry is locked out or ineffective during steering. Track overload and excess tension relief is still maintained and effective during steer. The application of steer pressure to the compensator cylinder will cause the final drive gearbox to rotate counterclockwise (unless an overload condition exists) which will move the control valve inner spool to the right in an attempt to reduce the compensator cylinder pressure, but during steer this valve is ineffective. However, as soon as steer pressure is released, the inner control valve spool will relieve pressure in the compensator cylinder until the control valve resumes its original set position.

6.4.2. Shock Loading and Shock Absorbing.

6.4.2.1. During vehicle operation, the track compensating system will be subjected to high shock loads. These loads must be absorbed or dissipated in the system. Absorbing or dissipating the shock loads imposed on the system will reduce high stress-producing loads on the components of the system and will tend to stabilize the system by providing a dampening action. Shock absorption is accomplished by an accumulator built into each track compensator system. Dampening action is provided by a flow control device which is sized to provide the desired dampening rates.

6.4.2.2. The accumulators used to provide shock absorbing action also act as pressure sources to provide compensator cylinder restoring force when the compensator must relieve track overloads.

6.4.3. Emergency Operation.

In the event of failure of the source of hydraulic pressure, due to a loss of prime mover or other malfunction, provision is made to tension the track for vehicle recovery. Emergency track tension is accomplished by isolating the track compensator cylinder (by means of a valve provided for this purpose) from the remainder of the track compensator system, and pressurizing the track compensator cylinder with fluid pressure from an emergency pressure source. This emergency pressure source will be a manually operated device with its own fluid supply.

6.4.4. General Design Considerations.

6.4.4.1. The design of the variable force track compensator incorporates components utilizing design features which have been proven in service.

6.4.4.2. The track compensator system consists of a group of individual components arranged in a manner so that under given operating conditions the individual components will operate together to provide a desired action. In the case of the track compensator, the desired action is to control the vehicle track tension. The results of the interaction of the individual components is an over-all action which is the system action. The way in which the individual components interact depends to a large extent on the adjustment and balance between individual components. Because the track compensator is installed in the vehicle as a single unit ready for operation, all adjustments and inter-related functions between components must be made and tested prior to installation of the compensator in the vehicle. This approach will enable all initial control relationships to be made on a bench test setup. After installation in the vehicle the compensator will operate automatically; however, proper automatic operation is dependent solely on the initial component adjustments.

7.0. ELECTRICAL SYSTEM

7.1. General System Description.

7.1.1. The electrical system in the LVTPX11 furnishes power for engine starting and operation, indicating system, warning system, distribution system, accessories, communications, lighting, and controls. Refer to drawing SK-5161 Electrical Schematic Diagram. The basic system consists of:

A. Batteries that furnish power during periods in which the engine is not in operation.

B. An alternator and voltage regulator that furnish regulated power during periods when the engine is operating.

C. An indicating system that provides accurate monitoring of the engine, fuel, battery, alternator, and transmission through instrumentation.

D. A warning system that provides monitoring of the engine and transmission, utilizing warning lights.

E. A communications system that includes radios, intercom, and signal light for complete communication coverage.

F. An interior and exterior lighting system that furnishes lighting during service or blackout conditions on land or while waterborne.

G. A control system that provides remote control of all electrical and electromechanical functions.

H. A distribution system that transmits the power from the sources to the points of utilization.

7.1.2. These systems are designed, utilizing proven principles as well as new developments in these fields, to provide maximum reliability consistent with long service life, low weight, and maximum economy. A minimum number of components are used to do the job effectively. Sufficient power is available for all present and anticipated future needs.

7.1.3. Waterproofing of all individual components, not sealed in waterproof assemblies, is maintained throughout the vehicle. Corrosion resistant assemblies and hermetically sealed components are used where necessary to complete the waterproofing system. Solid permanent mountings are provided on main and exterior assemblies. Ease of servicing is exemplified by the use of quick-disconnect fasteners on the instrument panel for convenient removal and repairs. The shortest possible lead and cable lengths are provided to keep power losses low. All circuits in the vehicle are protected by automatic-reset-type circuit breakers to provide maximum dependability.

7.1.4. The basic electrical system is a 24 volt dc system as utilized in the majority of military vehicles today. The voltage and distribution system utilized provides the greatest degree of flexibility and compatibility with other military vehicles. A single-wire system is utilized to provide the lightest weight and simplicity. Bonded structural members complete the low power loss circuit to the electrical components.

7.2. Batteries.

The static source of power for the vehicle is obtained from four 12-volt, lead-acid, MS35000-3 storage batteries. Refer to drawing SK-5171, Battery Installation. These batteries are the standard Ordnance type 6TN, which may be shipped charged and dry to reduce deterioration. At installation in the vehicle an electrolyte solution of sulphuric acid 1.28 specific gravity at 77°F is added to the batteries for instant available power. The batteries are connected in a series parallel system to obtain 24 volts at a 200 ampere hour rating. This 200 ampere hour capacity provides the heavy starting current required by the compression ignition engine and provides the normal standby power to operate the electrical equipment such as the electric bilge pump, radio, scavenger blower, crew heater, and searchlight. Since the batteries are standard Ordnance equipment, they can be easily replaced at any repair depot. Proven reliability, economy, and serviceability make them the logical design choice. The batteries are located on top of the track channel next to the bulkhead in the engine compartment, two on the port and two on the starboard side. At these locations the batteries can be inspected and serviced from the crew compartment by utilizing the access doors provided in the bulkhead between the crew compartment and engine compartment. Also, the wiring lengths and associated losses are reduced and ventilation of the batteries is assured since they are located in front of the bulkhead air intakes. The batteries are mounted on a half-inch neoprene pad in a drip pan to reduce shock, provide an effective stable mount, and protect the vehicle from the corrosive effects of the battery acid. To reduce acid corrosion, the holddown frames are covered with 1/32-inch thick neoprene per MIL-R-003065A (ORD) and the drip pan is sprayed with black lacquer per Federal Specification TT-L-54. A single-wire system with a negative ground return provides a low-loss, simple, and effective ground return system. Heavy current grounds are made on the major structural members to provide the shortest return length with the lowest possible loss.

7.3. Alternator System.

7.3.1. An alternator is the dynamic power source selected for this vehicle. This machine will supply a current of 125 amperes at a voltage of 28 volts. At the engine maximum speed of 3,000 revolutions per minute, the alternator rotates at 10,000 revolutions per minute. The alternator is an alternating current generator with the output rectified to provide the direct current necessary for compatibility with the vehicle electrical system. The rectifiers are an integral part of the alternator which provides for better reliability as well as ease of cooling, since the cooling fan of the alternator is utilized for cooling both the machine and the rectifier. Use of this type of power source in lieu of a direct current generator is advantageous because power may be obtained during engine idling conditions, reducing the load on vehicle batteries. This alternator, as compared to a standard direct current generator, will reduce maintenance problems and improve reliability, since the commutation problem is eliminated. Longer brush life may be expected with slip rings rather than a commutator. The rectifiers are the small semi-conductor variety which are inherently reliable.

7.3.2. The power analysis for this vehicle, as portrayed in the accompanying table 7-1, shows that the maximum possible load for continuous plus intermittent operation is 148.20 amperes. This condition occurs when the vehicle is operating at night under service conditions, in the water. The alternator minimum capacity required is then determined as follows:

From table 7-1:

Anticipated continuous current = 38.44 amperes maximum

Anticipated intermittent current = 109.76 amperes maximum

Assume 85 percent probability factor for the continuous requirement.

Probable continuous current = (38.44) (0.85)

= 32.67 amperes

Assume 50 percent probability factor for the intermittent requirement.

Probable intermittent current = (109.76) (0.5)

= 54.88 amperes

Table 7-1. LVTPX11 Electrical Current Requirements

Description	Current Required	Quantity	Water Service Continuous	Water Service Intermittent	Water Blackout Continuous	Water Blackout Intermittent	Land Service Continuous	Land Service Intermittent	Land Blackout Continuous	Land Blackout Intermittent
Port & Starboard Lights	.66	2	1.32							
Infrared Light Amplifier	5.50	1			5.50				5.50	
Headlights	2.86	2	5.72				5.72			
Service Stop Light	1.15	1		1.15				1.15		
Service Tail Light	.23	1	.23				.23			
Stern Light	.42	1	.42							
Masthead Light	.42	1	.42							
Compartment Lights, Service	.75	5	3.75				3.75			
Panel Lights	.17	6	1.02		1.02		1.02		1.02	
Compartment Lights, Blackout	.19	5			.95				.95	
Signal Light	1.95	1	1.95				1.95			
Horn	1.10	1		1.10				1.10		
Warning Lights	.29	2		.58		.58		.58		.58
Voltmeter	.04	1	.04		.04		.04		.04	
Fuel Gauge	.01	2	.02		.02		.02		.02	
Engine Oil Pressure Gauge	.01	1	.01		.01		.01		.01	
Transmission Oil Pressure Gauge	.01	1	.01		.01		.01		.01	
Engine Water Temperature Gauge	.01	1	.01		.01		.01		.01	
Transmission Oil Temperature Gauge	.01	1	.01		.01		.01		.01	
Speedometer	2.50	1	2.50		2.50		2.50		2.50	
Tachometer	2.50	1	2.50		2.50		2.50		2.50	
Fuel Pump	.80	1	.80		.80		.80		.80	
Winterization Kit	6.00 Run, 15.00 Start	1	6.00	15.00	6.00	15.00	6.00	15.00	6.00	15.00
Scavenge Blower	8.70	1	8.70		8.70		8.70		8.70	
Bilge Pump	55.00	1		55.00		55.00				
Radio	6.75	1	.75	12.00	.75	12.00	.75	12.00	.75	12.00
Infrared Headlights	Ave. 1.19	2							2.38	
Master Contactor	.38	1	.38		.38		.38		.38	
Starter Contactor	.38	1		.38		.38		.38		.38
Auxiliary Starter Contactor	20.00	1		20.00		20.00		20.00		20.00
Bilge Pump Contactor	.38	1	.38		.38					
Voltage Regulator	.50	1	.50		.50		.50		.50	
Relay, CO_2	.38	1		.38		.38		.38		.38
Relay, Radio	.38	1		.38		.38		.38		.38
Relay, Scavenge	.38	1		.38		.38		.38		.38
Air Diverter Solenoid	.31	1		.31		.31		.31		.31
Fuel Shutoff Solenoid	.50	1		.50		.50		.50		.50
CO_2 Plug	2.00	1		2.00		2.00		2.00		2.00
Track Solenoids	.60	2		.60		.60		.60		.60
Starter	650.00*	1		650.00*		650.00*		650.00*		650.00*
Blackout Tail Lights	.19	2			.38				.38	
Blackout Stop Light	.19	1				.19				.19
Blackout Marker Lights	.19	2			.38				.38	
Subtotals			38.44	109.76	30.84	107.70	34.90	54.76	32.84	52.70
TOTALS			148.20		138.54		89.66		85.54	

*This figure is not included in the totals because this is supplied by batteries only.

Maximum total probable (continuous + intermittent) current

A_t = 32.67 + 54.88

A_t = 87.55 amperes

7.3.3. Based on the above, the capacity of the alternator plus the capacity of the batteries will satisfy the complete power requirements of the vehicle under emergency conditions, and will supply sufficient charging current to quickly recharge the batteries. At an engine idle speed of 600 revolutions per minute, the alternator will supply 45 amperes at 28 volts which will supply the continuous current demand of 38.44 amperes plus a normal battery charging current. Full capacity of the alternator is obtained at an engine speed of 900 engine revolutions per minute. Four horsepower is required to drive the alternator at engine idle, and 12.2 horsepower is required at maximum engine speed.

7.3.4. A transistorized voltage and current limiting regulator rated at 28 volts, 125 amperes is provided This regulator is a compatible unit and has been designed to operate in conjunction with the alternator. The regulator is shock mounted, waterproofed, corrosion and fungus proofed, and radio suppressed. A single external voltage control potentiometer is utilized to set the alternator output voltage. The regulator will control up to five amperes of alternator field current, and incorporates a load relay which is energized when the ignition switch is in the START or RUN position. This removes the regulator load from the main power bus when the engine is not operating, thus reducing current drain from the batteries.

7.4. Instrument System.

7.4.1. The instrument panel is designed to allow the operator to control and monitor the engine, transmission, and accessory equipment. Refer to drawing SK-5168. This panel incorporates all the gauges, warning and panel lights, and switches, resulting in a simple yet complete control system. The panel is constructed of .125 inch aluminum sheet utilizing a wraparound design for rigidity. Instruments mounted on this panel are easily removable thus providing for a minimum of maintenance time. Black stencil paint conforming to Federal Specification TT-P-98 Type 1 is silk-screened on the panel for all lettering. The panel is anodized for a permanent finish, and the contrast between the anodized aluminum panel and the black lettering makes the panel easily readable. A subpanel is welded behind the main panel to provide the flexibility of design required for simplified wiring and a more compact main panel. Shorter lead lengths and less panel space is required by mounting the associated electrical components on the subpanel. These components include molded junctions, electrical connectors, and circuit breakers.

7.4.2. The circuit breakers conform to the MS39062 waterproof series in the range of 15 to 30 amperes. They were chosen because of their small size, automatic resetting ability, and quick-disconnect feature. Waterproof connectors ORD 8338565 are incorporated into the circuit breaker to provide the quick-disconnect feature. Molded rubber junctions are used conforming to synthetic class SC515AB Military Specification MIL-R-003065A. The use of this type of molded waterproof junction permits considerable saving in space, weight, and complex wiring over standard single or double connector methods. AN type electrical connectors were chosen because of their reliability, long life, and water resistance. These connectors are Military Standard and can be replaced quickly from stock at most repair depots.

7.4.3. The instrument panel is mounted in a plane ten degrees from vertical so that standard Ordnance gauges may be used. The panel is shock mounted with multiplane shock mounts which are selected for their long service life, high strength, economy, and wide multiplane frequency range. These shock mounts are capable of providing isolation at vibrating frequency range. These shock mounts are capable of providing isolation at vibrating frequencies as low as 600 cycles per minute.

7.4.4. The arrangment of the gauges, switches, and controls on the instrument panel assembly is designed to provide the most advantageous placement of components for operator accessibility and vision. The switches are located at the hand level position where they may be easily reached. The gauges and other components can be easily observed while operating the controls. A meter scale of approximately 330 degrees is used on all the gauges where applicable. The greater accuracy and the smaller current required per degree of movement makes these gauges a decided improvement. The meter scale background is flat black with white graduations and pointer for reflection free readings under normal or adverse lighting conditions. Luminous dial markings and pointer permit easy reading of the gauges even under adverse lighting conditions. The pointers on the gauges will point straight down if a loss of power to the gauge system occurs. This feature aids the operator in knowing whether the gauge is at fault or if a malfunction in some other system has actually occurred.

7.4.5. Two identical temperature gauges are used to monitor the transmission oil and engine water temperature. The 60° to 260° F meter scale covers the range required with the normal operating range near midscale. The twenty-degree graduations permit the operator to observe normal readings more closely and therefore he is better prepared to note abnormal conditions.

7.4.6. A transmission oil pressure gauge and an engine oil pressure gauge are used. The transmission oil pressure gauge indicates from 0 to 250 pounds per square inch covering the pressure range required with normal pressure reading near midrange. The engine oil pressure gauge indicates from 0 to 120 pounds per square inch with the normal pressure reading near midrange.

7.4.7. Two fuel level gauges monitor the port and starboard fuel tanks. Both the gauges and transmitter have been hermetically sealed for system reliability and include waterproof connectors. The gauges are easily read and employ a nonlinear scale which emphasizes the fuel graduations from empty to 1/4, and from 3/4 to full, where accuracy is most needed. The battery-generator gauge ORD 8380686, indicates the vehicle electrical system voltage by an easily discernable three-color meter scale which permits quick electrical system analysis. The red section indicates low battery voltage only, the yellow section indicates normal voltage, and the green section indicates generator voltage. Also furnished on the instrument panel is the speedometer, ORD 7527357, which covers the speed range of 0 to 60 miles per hour. The speedometer is larger than the other gauges and is located near the center of the panel in conjunction with the engine tachometer. This speedometer has been chosen because of its ruggedness, reliability, and accuracy, and includes an odometer which registers statute miles traveled. The engine tachometer, ORD 7527356, covers the speed range of 0 to 4000 revolutions per minute. Engine hours are recorded on a time totalizer located in the tachometer. A single warning light, ORD 8729063, is located in the center of the panel to quickly indicate malfunctioning of the engine or transmission.

Refer to paragraph 7.5. An indirect lighting system is provided by six panel lights, ORD 8376500, which very effectively light the panel gauges. The lights are arranged so that there are two illuminating each instrument. Sufficient light is provided for both normal night and blackout driving conditions. The main light switch, MS51113, incorporates panel light positions OFF, DIM or BRIGHT to provide adequate light intensity and control, and is located at the lower left hand corner of the panel. This switch also allows the operator to control the blackout and service lights. The lights activated in the various positions are as follows:

Position 1. BO DRIVE

A. Blackout tail - Port

B. Blackout tail - Starboard

C. Blackout marker - Port

D. Blackout marker - Starboard

E. Blackout stoplight - Starboard when stoplight switch is energized.

Position 2. BO MARKER

A. Blackout tail - Port

B. Blackout tail - Starboard

C. Blackout marker - Port

D. Blackout marker - Starboard

Position 3. OFF

A. All lights and markers OFF except panel lights.

Position 4. STOPLIGHT

A. Service stoplight - Port when stoplight switch is energized.

Position 5. SER DRIVE

A. Service headlights - Port
B. Service headlights - Starboard

These are dual element sealed beam units which are selected by the dimmer switch.

C. Service stoplight - Port when stoplight switch is energized.

D. Service tail light - Port

7.4.8. This main light switch was chosen because of its compact multiple circuit design, ruggedness, proven reliability, and complete waterproofing. A locking lever must be pressed before the selector switch can be placed in any position other than blackout marker.

7.4.9. The Engine Control Switch, ORD 8360081, is a three-position (OFF-RUN-START) rotary type switch that is located in the lower right hand corner of the panel. A momentary (START) position is included to actuate the engine cranking motor. In the RUN position the gauges, warning lights, and electric blower are energized.

7.4.10. Seven rugged waterproof toggle switches, similar to ORD 8376806, are located across the lower half of the panel. The switch contact ratings are conservative. They will handle six amperes with a lamp load, 15 amperes with an inductive load and 25 amperes with a resistive load. ORD 8338565 connectors are utilized to provide the quick disconnect features, reduction of bulky wiring, and ease of switch replacement. A toggle allows the switch to be easily operated. These switches are used to control the accessory electrical equipment furnished on the vehicle. A three-position switch located near the main light switch actuates the infrared beam selector. The other two switches located in the lighting control group of the panel operate the infrared viewer and the navigation lights. The emergency electric bilge pump is controlled by a toggle switch located in the remote equipment control group of the panel. A spare switch is provided for any future improvements or modifications. A switch is provided to control the electric blower which scavenges the bilge. The master power switch is located in the lower right hand corner of the panel and is so placed that it is easily accessible for normal use and also in the event of an emergency. The horn switch, ORD 7354622, is located in the lower left hand corner of the panel. This is a single pole, momentary on, pushbutton type, waterproof switch which incorporates a snap action switch assuring fast make and break.

7.5. Warning System.

7.5.1. The warning system is composed of two warning lights, ORD 8729063, two engine monitoring switches, two transmission monitoring switches and a flasher unit. Reference drawings pertaining to the warning system are:

A. Engine and transmission exterior warning light installation, drawing SK-5170.

B. Instrument panel assembly, drawing SK-5168.

C. Electrical schematic drawing SK-5161.

7.5.2. One warning light is placed in the center of the instrument panel where it can be easily seen. The other light is located on top of the driver's cupola approximately 30 degrees starboard off the center line and between the periscopes. This light can be easily seen by the driver when he is operating the vehicle with his head above the cupola. The light is recessed in a tubular assembly and welded to the cupola or it may be recessed into a housing cast into the cupola. Recessing the light concentrates the light beam over a small area and thus detection by anyone other than the driver is eliminated. The lights operate in a series parallel circuit which is energized when any sending unit switch closes. This action takes place under low oil pressure or high oil or water temperature conditions of the engine or transmission. Provisions are included for any additional warning functions which may be added in the future. The warning light attracts the immediate attention of the driver who may then determine the malfunction by observing the gauges on the instrument panel. A No. 313, .17 ampere bulb is used to provide a high light intensity. A red lens covering the light combined with a thermal flashing unit effectively attracts the driver's attention. He can then concentrate on driving and is relieved of the duty of continually checking the instrument panel gauges for malfunctions. The use of this simple two-light warning system gives effective protection yet reduces the number of lights normally required.

7.6. Auxiliary Electric Power.

The electrical system is provided with a readily accessible outlet (slave receptacle) of sufficient power capacity to provide for the transfer of power from one vehicle to another, or from the vehicle to an external source other than another vehicle. This permits operation of vehicle accessories and communication equipment from other power sources for emergency or convenience purposes.

7.7. Compartment Lights.

Lighting of the crew compartment and engine compartment is accomplished through the use of five compartment lights, ORD 7064671, placed at strategic points throughout the interior of the vehicle. Refer to drawing SK-5175, Compartment Lighting. These lights provide interior lighting during both service and blackout conditions and are completely waterproof. Two lamps are incorporated in each assembly, the service lamp rated at 15 candlepower and the blackout lamp rated at six candlepower. An Ordnance waterproof female connector is included in the assembly for convenience in wiring and compatibility. The assembly includes a rotary switch to provide the following positions: Center - Off, Clockwise - Blackout lamp on, Counterclockwise - Service lamp on. A safety lock on the switch insures against accidental illumination of the service lights during blackout conditions. The light assemblies are mounted on four weld studs installed underneath the upper deck. Sufficient illumination is produced under service conditions for normal operation in the vehicle and to permit inspection of the engine compartment. Two easily accessible lights are installed immediately forward of the driver's and commander's positions. An additional light is located on the centerline of the vehicle ahead of the engine compartment bulkhead to cover the lighting requirements of the central portion of the vehicle. The remaining two lights are positioned in the engine compartment on either side of the engine, but clear of the engine access hatch.

7.8. Navigation Lights.

For operation on waterways during nontactical conditions, navigation lights are included as set forth by the U. S. Coast Guard International Regulations for Class I craft. Refer to drawing SK-5206, Navigation Lights Installation. Control of these lights is accomplished by means of a toggle switch located on the instrument panel. Two side lights, one masthead light, and one stern light comprise the lights required for operation on international waters. The side lights are located forward, on the outboard sides of the cupolas. On the starboard side a green side light is installed which gives an unbroken arc of light through ten points of the compass, starting directly forward and ending two points abaft the beam. The same is true of the port side light, only the color is changed to red. Both lights can be seen for a distance of two miles. A masthead light is located behind and on the port side of the Commander's cupola above the side lights. This is a white light that covers an unbroken arc of 20 points facing the bow of the vehicle, ten points on each side. The light is visible for a distance of three miles. The masthead light may be swung down to the deck into a protective box and locked in place if desired to reduce the over-all height of the vehicle. A white stern light is located on the stern deck at approximately the same level as the side lights. This light covers an unbroken arc of 12 points, six points either side of center facing aft. The light is visible for a distance of three miles. The lamps provided in the lights are double contact bayonet base and six candlepower. This allows simplified stocking and ready interchangeability. All the navigation lights are watertight and made of cast bronze, to reduce corrosion.

7.9. Service and Blackout Lights.

7.9.1. Two service headlamps, ORD 8735874, and two infrared headlamps, ORD 8735875, have been selected and are located on the top deck forward and between the cupolas out of the driver's and commander's line of vision. Refer to drawing SK-5205, Headlight and Taillight Installation. The lamps are sealed beam, dual element units that supply sufficient illumination for all normal service and blackout driving. The headlamp assembly rests on a multidirectional mount which is welded to the deck. Headlamps can be easily replaced or adjusted using this type mounting. A heavy aluminum guard surrounds the headlamps on three sides giving adequate protection. A foot operated dimmer switch, ORD 8396186, is provided in the vehicle to select either the HI-beam or LO-beam. The switch is single-pole double-throw and will handle a resistive load of 15 amperes, an inductive load of ten amperes and a lamp load of eight amperes. This is a compact, waterproof switch that can be operated over a temperature range of -65°F to +165°F for a minimum of 50,000 cycles. A blackout marker light MS51303 is included in the service-infrared light assembly and is protected by the same guard utilized to protect the headlights. A welded angle bracket securely mounts the light assembly to the deck. The lamp rating and lens configuration of the blackout marker lights provide the required illumination necessary for blackout lighting conditions. The two taillight-stoplights used are MS51329 (port side) and MS51330 (starboard side). They are installed on the stern of the vehicle, above the waterline as far outboard as possible and are recessed into the hull for protection.

7.9.2. In addition to the blackout marker capabilities, provisions are also included in these lights to provide a stoplight indication for both service and blackout conditions.

7.9.3. All lights are completely waterproof to provide reliable operation on both land and water.

7.10 Infrared Viewer.

7.10.1. An infrared viewer, BuShips 1940882, is installed on this vehicle to aid the driver in night driving. See reference drawing SK-5198, Infrared Viewer and Stowage Box Installation. Included in this installation is a high-voltage power supply, an internal and external mount, an internal stowage box, and the infrared viewer assembly. The power supply is located behind the instrument panel underneath the top deck to provide a short lead length and low voltage loss. The stowage box is located on the track channel below and aft of the driver's seat for easy accessibility. The external mounting of the viewer assembly bracket is on the cupola directly in front of the driver. The viewer assembly is mounted horizontally with limited horizontal and vertical motion. The horizontal swing is 23 degrees either side of center. The internal mounting for the viewer assembly bracket is located in front of the driver, inside of the cupola, so that the lenses of the viewer are located directly in front of the periscope with the viewer in the operating position. When not needed, the viewer is swung down and locked. The viewer is normally stowed inside the vehicle and can be installed in front of the periscope for viewing when the hatch is closed, or installed on the external bracket for operation when the hatch is open. The infrared equipment can be utilized for driving under complete blackout conditions or for observation purposes to detect infrared radiation sources.

7.10.2. The infrared viewing system includes two infrared projectors (Infrared Headlights ORD 8735875) a 16,000 volt direct current power supply and a lens system image converter type infrared viewer. When in operation as an active system, infrared rays are projected to the object from the headlights and are reflected back to the lens system in the viewer. The photosensitive cathode of the image converter emits photoelectrons that are electrostatically focused onto a fluorescent screen where the image is viewed through field lenses and eyepiece assemblies. The high-voltage power supply provides the necessary voltage required to focus and accelerate the electrons to the screen. When the system is utilized passively, the projectors are not turned on, and the viewer will then detect objects illuminated by ambient infrared light or an infrared source.

7.10.3. The viewer assembly is 11-1/2 inches long, 8-1/4 inches wide, 5-5/8 inches high, and weighs 10 pounds 11 ounces. Its performance specifications are as follows:

Operating voltage	16,000 volts
Magnification	1 power
Field of view, horizontal	45 degrees
Field of view, vertical	19 degrees
Focal point	18-20 yards
Visibility (in-line viewer mounted externally)	45-401 feet
Visibility (in-line viewer mounted internally)	25-240 feet

7.11. <u>Signaling Searchlight</u>.

7.11.1. A portable signal searchlight, BuShips 9903-S6600-600497 has been incorporated to provide increased external lighting and signaling flexibility. This searchlight has been modified to include a 24-volt sealed beam lamp to permit compatability with the voltage supplied by the electrical system. It has a pistol type grip incorporating a momentary switch for signaling and an ON-OFF switch on the rear of the light for continuous use. This light is adaptable for use with a detachable color lens. The searchlight also can be locked in any desired position. A steel wire grille protects the face of the light from damage. Long life, high power, resistance to salt spray, and reliability were the main factors considered in selecting this searchlight. The light is mounted forward and to the starboard side of the commander's cupola. This position was chosen because of operator accessibility, 360-degree usable horizontal arc and over 180-degree vertical arc, with no interference to the commander's vision when inside the vehicle using the periscopes. The searchlight mount is provided with a quick fastening clamp that securely fastens the searchlight to the hull. This mount is permanently welded to the hull and is provided with a water drainage hole to reduce corrosion.

7.11.2. Since it is not desirable to leave the light mounted outside the vehicle when not in use, an interior mount has been provided for stowage. It is located underneath the commander's seat on the track channel. This point was chosen because it is close enough to the commander to be easily reached yet is is protected and out of the way below and aft of the seat. A simple canvas strap clamps the light in the mount but can be unbuckled quickly when needed.

7.12. Radio Equipment.

7.12.1. The government furnished AN/PRC-38, or the alternate AN/PRC-47 single side band radio is located near the upper deck in front of the seat on the starboard side. Refer to drawing SK-5199, Radio Installation. The vehicle's electrical system provides power for the radio equipment which is also capable of utilizing an external 115-volt, single-phase, 400 CPS power source. The antenna is located directly above the radio on the upper deck of the vehicle so that vision through the periscope is not impaired and to keep the antenna lead wire length at a minimum. Space is provided under the seat for stowage of an AN/PRC-54 portable radio.

7.12.2. The intercommunications amplifier AM-65 /GRC is located adjacent to the AN/PRC-38 radio. Operating controls for the radio and the intercom amplifier are located on the front of the respective equipment. Three C-375/VRC remote control stations are connected to the radio and intercom amplifier to provide remote operation of this equipment from the driver's station, the embarked troop commander's station, and from a station at the aft end of the troop compartment.

7.12.3. Provision is made for the stowage of the radiac set AN/PDR-27 immediately forward of the radio and amplifier.

7.13. Compass.

A magnetic, corrected compass, FSN6605-255-0238 (BuShips 1756482) is included in the vehicle for navigational purposes. Refer to drawing SK-5200, Compass Installation. The compass includes a radium dial and a three-inch top reading card which has five-degree graduations. The compass is located on the port track channel below and outboard of the front of the driver's seat. The compass can be readily seen by the driver from the low and high seat positions. A simple U-shaped mount welded on the track channel provides sufficient support and allows ease of installation or replacement. A guard consisting of a half-inch round rod welded to the hull provides protection to the compass without interfering with the visibility of the face.

7.14. Emergency Bilge Pump.

An electrically driven bilge pump similar to BuShips 1290136 is installed on the deck of the engine compartment near the engine compartment bulkhead. Refer to drawing SK-5172, Emergency Bilge Pump Installation. The emergency bilge pump provides a supplemental pump capacity of approximately 125 gallons per minute. In an emergency when the hydraulic bilge pumps are not operating, the emergency bilge pump will effectively remove any infiltration of water from the bilges. An aluminum wire mesh screen attached to the pump intake effectively filters the water so the pump will not become clogged, thus retarding the water flow. The screen opening area is 28.687 square inches which allows adequate water passage even when 50 percent of the total screen area is clogged. A flexible rubber hose directs the water to the outlet which is well above the waterline inside of the radiator compartment. The pump is rated to deliver a free flow rate of 130 gallons per minute, with a six-foot head 125 gallons per minute, and a stall head of 30 feet. Corrosion resistant clamps are used to install the hose. A hinged outlet cover retards water from entering the vehicle through the discharge hose due to normal wave action. The water from the outlet feeds into the radiator compartment which is open to the outside. The pump and motor are effectively sealed to allow continuous operation under completely submerged conditions while pumping water, or running dry. At 27.5 volts direct current the motor draws approximately 55 amperes at the rated six-foot head. This is a shunt wound motor which exhibits a very high starting torque and fairly constant speed. If the load on the motor increases, a greater torque is developed, keeping the speed relatively constant. This is a heavy duty motor that is designed to take overloads and still retain a long service life. The pump is a centrifugal type capable of relatively high pressure against the dynamic heads encountered in the vehicle. Supplementary power can be supplied to the vehicle for emergency bilge pump operation by means of the slave receptacle as discussed in paragraph 7.6.

8.0. HYDRAULIC SYSTEM

8.1. General.

8.1.1. Design Requirements.

8.1.1.1. This section outlines the basic design, functions, and components of the LVTPX11 hydraulic system.

8.1.1.2. The determination of the design parameters for the hydraulic system to be incorporated into the LVTPX11 vehicle depends on several prime requirements. These requirements are:

A. Reliability

B. Weight

C. Cost

D. Ease of maintenance

E. Flexibility

8.1.1.3. In designing a system which provides optimum fulfillment of the prime requirements, many other secondary requirements must be considered. Among these requirements are (not necessarily in order of importance):

A. Operating Conditions

1. Temperature Limits

2. Environment

B. Functions

1. Loads
2. Speeds
3. Response

C. Method of Control

1. Manual
2. Mechanical
3. Electrical
4. Hydraulic

D. Availability of Material

1. Standard Items
2. Delivery
3. Materials

E. Compatibility of Components with Existing Hardware

1. Are required components available as MS, AN, BuShips, or ORD standard items?
2. Can available components meet all required specifications?

F. Experience

1. What operational data (if any) is available to substantiate component design?

8.1.1.4. All of the prime and secondary requirements have been taken into consideration in the design of the LVTPX11 hydraulic system.

8.1.2. Hydraulic System Functions.

8.1.2.1. The design of any hydraulic system begins with the determination of what functions are to be powered or actuated by the hydraulic system. In the LVTPX11 vehicle the hydraulic system will furnish power to perform the following functions:

A. Power the hydraulic track compensator.

B. Drive the engine coolant cooling fan.

C. Drive the transmission lubricating oil cooling fan.

D. Raise (close) the bow ramp.

E. Control the rate of lowering (opening) of the bow ramp.

F. Drive the vehicle bilge pumps.

8.1.2.2. These functions and their duty cycles and time rates are shown in table 8-1.

8.1.2.3. After the functions to be powered or actuated by the hydraulic system have been determined it is necessary to ascertain what priority, if any, the functions are to be given. The establishment of functional priority depends primarily on how critical each function is to the sustained operation and performance of the vehicle. Functional priority generally affects only the

Table 8-1 HYDRAULIC SYSTEM LOADS

No.	FUNCTION	DUTY CYCLE	TIME RATE	INPUT HP REQUIRED	
				MINIMUM	MAXIMUM
1.	Track Compensator	Intermittent	See Note 1	See Note 1	See Note 1
2.	Engine Coolant Cooling Fan	Continuous	Continuous	0	9.52
3.	Bow Ramp (Raise Only)	Intermittent	4 Seconds	0	6.2
4.	Transmission Lubricating Oil Cooling Fan	Continuous	Continuous	0	9.52
5.	Bilge Pumps	Intermittent	Continuous		
	Forward Starboard	See Note 2	See Note 2	.25	2.7
	Forward Port	See Note 2	See Note 2	.25	2.7
	Aft Starboard	See Note 2	See Note 2	.25	2.7
	Aft Port	See Note 2	See Note 2	.25	2.7

Note 1: This horsepower requirement will vary depending on the vehicle operational profile but is not considered high enough to affect the total system load appreciably because the system is essentially closed end.

Note 2: Refer to section 8.1.2.1, item 5.

hydraulic system circuitry and does not appreciably affect the total loads imposed upon the hydraulic system.

8.1.2.4. The hydraulic system functions if the LVTPX11 vehicle is arranged so that the functions operate in a cascade fashion; that is, one function is given priority over all other functions. A second function is given priority over all functions except the first, and a third function is given priority over all functions except the first and the second. In the LVTPX11 vehicle the hydraulically actuated functions have been given the following priorities:

1st priority - Power to the hydraulic track compensator.

2nd priority - Engine coolant cooling fan drive motor.
Raise bow ramp.

3rd priority - Transmission lubricating oil cooling fan motor.

4th priority - Bilge pumps.

8.1.2.5. These priorities have been established on the following basis:

8.1.2.5.1. First priority is given to the hydraulic track compensator for two primary reasons:

A. The operational capability and mobility of the vehicle depend upon the tracks and the track operation of the vehicle.

B. The hydraulic track compensator uses essentially a closed end circuit requiring little if any flow, but which requires the maintenance of hydraulic system pressure when the compensator is functioning. This arrangement absorbs little continuous power from the hydraulic system and therefore has but a small effect on subsequent hydraulic functions.

8.1.2.5.2. Second priority on hydraulic power is given to the engine coolant cooling fan drive motor. The reason for this is that the vehicle cannot be operational unless the engine (prime mover) is operating, and whenever the engine is operating it must be cooled. Also given priority along with the engine coolant cooling fan motor is the bow ramp raising function. This function is of such short duration that even if the bow ramp raising functions should completely stop the cooling fan functions, no serious malfunction would result. Further, bow ramp actuation is a relatively infrequently performed function compared to total vehicle operational time.

8.1.2.5.3. Third priority on hydraulic system power is given to the transmission lubricating oil cooling fan. The reason for this is that if the prime mover (engine) is not operating, the transmission will not be operating and therefore requires no cooling. Whenever the engine is in operation, hydraulic cooling fan horsepower is always required in proportion to the horsepower output of the engine and the horsepower input to the transmission. As engine speed and horsepower increase, the amount of hydraulic horsepower available for cooling fan operation becomes greater; however, the actual cooling fan horsepower used is only that amount required to perform the cooling. (See paragraph 8.4.3.).

8.1.2.5.4. Fourth priority on hydraulic power is given to the bilge pumps as a group. This distinction is necessary because each bilge pump is given individual priority depending on the imposed load. This is discussed further in paragraph 8.4.4.2. The reasons for assigning fourth priority to the bilge pumps are:

A. Bilge pump operation is not required for land operation.

B. Bilge pump operation is required during water operation at which time the engine is generally operating at high speed and at which time maximum hydraulic horsepower is available.

C. During water operation coolant cooling fan power requirements are lower than land operation because of the greater heat rejection rates from the engine and transmission radiators due to the greater temperature differences between the coolants and the sea water.

8.1.3. Hydraulic System Loads.

For each of the functions the hydraulic system powers there is a load imposed upon the system. Some of these loads are constant and some are variable. Some loads are continuous and some are intermittent. These loads and their duty cycles are shown in table 8-1.

8.1.3.1. Load Determination.

A. Track Compensator - For track compensator loads see section 6.0.

B. Engine Coolant Cooling Fan Motor Load - The engine coolant cooling fan motor load is determined by configuration of the cooling fan, required air flow, etc., as outlined in paragraph 4.7. The fan motor load is variable and continuous. Fan motor load is proportional to the engine heat rejection rate and varies directly with the temperature of the engine coolant. The fan speed and hydraulic load are determined by a thermally operated control valve.

$$HP_H = \frac{HP_F}{E_m}$$

$$HP_H = \frac{8.1}{.85} = 9.52$$

HP_F = Horsepower required to drive fan

HP_H = Required hydraulic input horsepower

E_m = Hydraulic motor efficiency at 3000 PSI and fluid at 50 SSU

C. Bow Ramp Load - The bow ramp load is determined by the actuator displacement required to raise the ramp and the actuation time to perform the function.

$$D = S\frac{(A_B - A_R)}{231}$$

$$D = \frac{26.375\ (1.76 - .78)}{231}$$

D = .112 gal per cylinder

A_B = Area Bore

A_R = Area Rod

S = Cylinder Stroke

D = Actuator Displacement (gal)

Flow Rate

$Q = \frac{60D \times 2}{4} = 30D$ $\quad$ Q = Flow rate gpm

Q = 3.36 gpm for 2 cylinders

Load = Input horsepower $\quad$ E_c = Efficiency of cylinder (worst condition)

$Hp = \frac{Q(PSI)}{1714\ (E_c)} = \frac{3.36\ (3000)}{1714\ (.95)}$

Hp = 6.2

D. Transmission Lubricating Oil Cooling Fan Motor Load - The transmission lubricating oil cooling fan motor load is determined the same way that the engine coolant fan motor load is determined except that fan motor load and speed are controlled by the transmission lubricating oil temperature.

$HP_H = \frac{HP_F}{E_m}$ $\quad$ HP_H = Required hydraulic input horsepower

$HP_H = \frac{8.1}{.85}$ $\quad$ E_m = Hydraulic motor efficiency at 3000 PSI and fluid at 50 SSU

$HP_H = 9.52$ $\quad$ HP_F = Horsepower required to drive fan

E. Bilge Pump Loads - The bilge pump load is continuous and variable. This is due to the fact that the bilge pumps run continuously as long as the vehicle engine is running and cooling fan requirements are met, and because the bilge pumps run at two speeds, maximum output and idle. Pump load is controlled by the individual bilge pump control valve. The bilge pump control valve is designed to permit the bilge pumps to idle at a given speed (approximately 200 rpm). The valve senses any increase in load on the pump when running at idle and increases the flow to the pump automatically with an increase in pump load. This arrangement reduces the power required to drive the bilge pumps to a minimum during land operation and during periods when the bilges are dry due to the rolling or pitching of the vehicle during sea operation. This arrangement also reduces heat build-up in the bilge pump circuits to a minimum. It is absolutely necessary for maximum efficiency in a closed center hydraulic system to reduce heat loss to a minimum level. The bilge pumps utilize approximately .25 horsepower in order to achieve control through the control valves. This horsepower is approximately one-twelfth the maximum bilge pump demand and is considered acceptable. Every effort will be made to reduce the amount of control horsepower required by the bilge pumps.

F. Maximum Bilge Pump Horsepower:

$$HP_H = \frac{HP_P}{E_m}$$

HP_H = Required hydraulic input horsepower required by bilge pump.

$HP_H = \frac{2.3}{.85}$

E_m = Efficiency of bilge pump drive motor at 3000 PSI and 50 SSU fluid.

HP_H = 2.7 per pump

HP_P = Horsepower required to drive bilge pump.

G. Total Connected Bilge Pump Load:

$HP_{HT} = 2.7(4) = 10.8$ horsepower

8.2. Hydraulic System Design.

The hydraulic system designed for the LVTPX11 vehicle is a high-pressure (3000 psi operating pressure), closed-center, constant-pressure system. This system design was selected because it fulfills both the prime and secondary design requirements. This type of system provides the following design features:

8.2.1. Reliability.

System reliability and endurance will be high. This is due principally to the high reliability of the individual components. Components are of proven military standard design, or incorporate commercial adaptations of designs originally created and tested for high reliability and endurance as lightweight aircraft components. System reliability is enhanced by the quick response of the system and constant availability of hydraulic power whenever the prime mover is operating.

8.2.2. Weight.

The use of a high operating pressure enables the system to provide the greatest amount of power in the smallest package. A high pressure system reduces the required size of all components including line and fittings sizes. The state of the art has progressed extensively, and commercially available lightweight components are readily available for use at higher pressures.

8.2.3. Cost.

The use of a high pressure permits the use of smaller flows, smaller line sizes, smaller fitting sizes, and smaller components, thereby reducing costs as well as weight.

8.2.4. Ease of Maintenance.

Maintenance is improved through the use of lighter, easier-to-handle components. Maintenance is reduced by the improved components inherent in high pressure hydraulic system design.

8.2.5. Flexibility.

8.2.5.1. The use of the closed center system permits the use of separate hydraulic circuits to accomplish individual hydraulically powered functions. With the closed center type system it is not necessary for all of the hydraulic fluid flow to pass through each circuit and each component in each circuit as in an open center type system. Therefore circuits utilizing small flows for actuation do not have to have components sized for higher flows required by other circuits or the attendant power losses associated with excess flow relieving devices.

8.2.5.2. The utilization of a constant pressure system permits the use of a variable-displacement pressure-compensated pump as the main source of hydraulic power. The variable-displacement pressure-compensated pump maintains a constant hydraulic system pressure. With this arrangement, power is required from the prime mover only when demanded by the hydraulically actuated functions. Prime mover power losses into the hydraulic system are reduced to a minimum.

8.2.5.3. Another equally important advantage of the constant pressure system utilizing a variable-displacement pressure-compensated pump is the ability to get high hydraulic system performance over the entire prime mover operating speed range.

8.2.5.4. The hydraulic pump output in a hydraulic system design incorporating a fixed displacement pump will be a function of the pump drive speed. When the pump is being driven by the vehicle prime mover, the pump output will vary with the speed of the prime mover. This means one of two things must take place. They are: (1) if the pump is sized to provide full power at low prime mover speeds, excessive flows and power drain and consequent heat build-up will result at high prime mover speeds; (2) if the pump is sized to prevent excessive power losses at high prime mover operating speeds, the power available for hydraulic functions at low prime mover operating speeds will be insufficient, which will require excessive manipulation on the part of the vehicle operator to match the required hydraulic load to the engine output and at the same time maintain the correct vehicle speed or attitude. The use of the constant-pressure closed-loop system and a variable displacement pump permits the hydraulic system to operate at maximum efficiency through a better load match between the prime mover operating speed turndown ratio and the hydraulic system load demand. This arrangement eliminates the problems of insufficient or excessive hydraulic flows which are created with the use of the constant displacement pump.

8.2.5.5. The use of a constant-pressure closed-loop system with a variable volume pump also provides for flexibility in special vehicle design. The same pump used in the LVTPX11 vehicle can be used to drive larger load hydraulic functions such as winches, turntables, remotely mounted electric generators, air compressors, or pumps whenever sufficient prime mover output can be diverted from the vehicle driving power requirements.

8.3. Hydraulic Component Design.

Hydraulic components are selected to meet all of the prime and secondary requirements set forth in paragraph 8.1 and to meet all of the requirements of BuShips Contract Specification SHIPS-A-4561.

8.3.1. Tubing.

8.3.1.1. Hydraulic tubing is one of the most important elements in the hydraulic system and should be designed to provide optimum system performance. In order to provide optimum performance, tubing selection must take into account flow losses, weight, cost, vulnerability to physical damage, and ability to contain and resist applied hydraulic loads.

8.3.1.2. Tubing sizes are generally determined on the basis of flow velocities. The applicable current specification covering flow velocities is specification MIL-H-5440 which covers the design of high pressure (3000 psi) hydraulic systems for aircraft use. This specification recommends an average flow velocity in pressure and return lines of 15 feet per second. Actually this velocity limit is an arbitrary member and is only to provide a starting point for choice of a proper line size. In order to realize maximum savings in cost and weight, hydraulic line sizes for the LVTPX11 vehicle will be determined on the basis of acceptable pressure drops for each hydraulically operated function. In general, line velocities will fall within the following ranges:

A. Pressure lines

a. Continuous flow functions i.e., hydraulic motors.
(15 to 20 feet per second)

b. Intermittent flow functions, i.e., hydraulic cylinders.
(20 to 30 feet per second)

c. Special applications.
(As required by design)

B. Return lines

a. Continuous flow	10 to 15 feet per second
b. Intermittent flow	15 to 20 feet per second
c. Special applications	As required by design

8.3.1.3. Design effort is always made to keep fluid velocities on the low side of the range whenever possible. Currently available sizes and wall thicknesses sometimes dictate system operation at velocities in the upper portion of the design range.

8.3.1.4. Tubing material selected for the LVTPX11 hydraulic system is stainless steel type 304 meeting the requirements of specifications MIL-T-8504 or MIL-T-6854. Stainless steel tubing was chosen because of (1) corrosion resistance (2) weight to strength ratio (3) availability (4) previous contractor experience. Aluminum tubing is not considered suitable for the LVTPX11 vehicle hydraulic system because of its low fatigue strength and the larger line sizes required for equivalent flow velocities due to lower yield strength (which increases fitting sizes required which in turn increases weight and cost). The safety factor using stainless steel tubing will be four to six.

8.3.2. Fittings.

8.3.2.1. The fittings to be used on the LVTPX11 hydraulic system will be MS standard flareless type meeting requirements of Specification MIL-F-18250 or commercial equivalent wherever possible. In order to facilitate procurement and reduce cost and weight, commercially available fittings which are compatible with the MS standard fittings will also be used.

8.3.2.2. Flange type fittings which use an O-ring gasket for sealing will be commercially available types per Society of Automotive Engineers or J.I.C. standards.

8.3.2.3. All pipe threads whenever used will be S.A.E. standard dryseal threads. Components of the hydraulic system will be provided with ports in bosses which will accept fittings using an O-ring and jam nut seal whenever possible. Component parts which do not have female connections shall be provided with fitting connections meeting the requirements of MS standards MS33514 and MS33515 whenever possible. Both the O-ring type port connection and the MS33514 and MS33515 fitting connections are service proved and have a high reliability factor and are readily available.

8.3.3. Hoses.

8.3.3.1. Hose assemblies to be used in the LVTPX11 hydraulic system will be:

A. High pressure - (above 1500 psi) hose assemblies per MS standard MS28762, MS28759, or specification MIL-H-19606 (SHIPS) type B where pressures are compatible, or commercial grade where no specification applies for sizes in higher pressure range or where fittings are not compatible with MS or MIL Spec. hoses.

B. Medium pressure - (to 1500 psi) commercial grades suitable for the application and with compatible fittings.

C. Suction - (0 psi to 30 inches mercury) best commercial grade wire supported suitable for application.

8.3.3.2. Hose assemblies will be provided with reusable hose ends whenever possible.

8.3.4. Reservoir.

The hydraulic fluid reservoir is designed to have a capacity of 30 gallons. This capacity will provide for approximately a 60-second fluid interchange under maximum hydraulic demand conditions. The reservoir is designed to operate pressurized to approximately 4 to 6 psi and is provided with pressure and vacuum relief valves. The reservoir is provided with a filler neck, filler screen, and pressuretight cap with hydraulic fluid level indicating dip-stick. A suitable drain arrangement with shutoff valve will be provided to permit draining the reservoir to the outside of the vehicle.

8.3.5. Hydraulic Pump & Motors.

8.3.51. Pump.

The pump selected for the hydraulic power source on the LVTPX11 vehicle is a pressure compensated, axial piston type with an operating pressure of 3000 psi. The pump has an output of 3.2 cubic inches per revolution at 3000 psi and a miximum speed of 2400 rpm. The pump is Weatherhead Company model 41G365.

8.3.5.2. Motors.

The LVTPX11 uses hydraulic motors for three functions: driving the bilge pumps, driving the engine coolant cooling fan, and driving the transmission lubricating oil cooling fan. The hydraulic motors used on these applications will be gear type and/or axial piston type. Gear type motors will be of the pressure compensated gear faceplate type. That is, each sealing face against which gears rotate is pressure loaded to maintain an effective seal against the gear face. This type of motor is available for 3000 psi operation and provides high efficiencies. Piston type motors can be more effectively matched to their driven loads than gear type motors and generally have higher operating efficiencies but are also higher in cost. Piston type motors will be suitable for the purpose intended in all respects.

8.3.6. Filters.

8.3.6.1. Filters will be of the permanent cartridge continuously cleanable type or of the replaceable metal cartridge type. All filtration is designed to remove particles in the 10-to 40- micron size range. In order to assure continuous operation under emergency conditions all filters will be provided with by-pass relief valves which will have pressure settings sufficiently high to guarantee against the valve opening under transient pressure conditions.

8.3.6.2. Suction strainers are provided with magnetic particle traps to remove all harmful magnetic particles from the hydraulic fluid.

8.3.7. Valves.

8.3.7.1. All valves used to control hydraulic functions will be of a design which will provide a miximum of reliability with a minimum of maintenance. Valves will incorporate good valve engineering design and any applicable advances in the state of the art.

8.3.7.2. Electrically operated valves will be provided with protection against moisture as specified in specification MIL-E-13856.

8.3.8. Cylinders.

Hydraulic cylinders will be designed and built in accordance with the best commercial practice.

8.3.9. Accumulators.

Hydraulic fluid pressure accumulators will be of the piston type. Gas used to precharge accumulators will be dry nitrogen. The accumulators will be of high strength aluminum or steel construction. Maximum feasible corrosion protection will be provided in all cases.

8.4. Hydraulic Circuitry and Component Installations.

The LVTPX11 hydraulic circuit is shown schematically on drawing SK-5243. Hydraulic component installations are shown on the drawings listed in their applicable paragraphs below.

8.4.1. Ramp Circuit.

8.4.1.1. The ramp circuit provides power for raising the bow ramp (drawing SK-5212) as specified in paragraph 3.12.1.4 of BuShips Contract Specification SHIPS-A-4159. This circuit is controlled by a manual four-way valve with three positions: RAISE, LOWER, and spring-centered HOLD or normal position. This valve actuates the two single-acting ramp cylinders. Time response is controlled by a flow control valve which gives proper speed in raising the ramp. Ramp opening is gravity actuated with hydraulic fluid from the ramp cylinders returned directly to the reservoir.

8.4.1.2. The cylinders actuate cables connected to the ramp for raising and lowering. The cable arrangement will prevent damage to the ramp cylinder mechanism when loading or unloading. Two cylinders are used for reliability and safety. The cylinders are located on each side of the ramp at the bow of the vehicle. The ramp actuating cylinder installation is shown in drawing SK-5212.

8.4.2. Track Compensator Circuit.

Hydraulic fluid under pressure is supplied to the track compensator from the hydraulic pump through a check valve. From the check valve hydraulic fluid is delivered to the remainder of the track compensator circuit. In order to prevent duplication, the remainder of the hydraulic track compensator circuit is described along with the operation of the compensator in section 6.

8.4.3. Engine Coolant and Transmission Lubricating Oil Cooling Fan Motor Drive Circuit.

8.4.3.1. The engine coolant and transmission lubricating oil cooling fan motors are located in their respective cooling ducts in the engine compartment. The two fan drive hydraulic motors (see drawing SK-5165) run continuously. The two cooling fan motors are rated at 6.3 gpm at 3000 psi to drive the cooling fans at 2200 rpm. The speed of each cooling fan motor is controlled with a modulating thermally operated control valve which reduces the fan motor horsepower load on the prime mover during cold weather operation or whenever maximum engine cooling is not required. The engine coolant and transmission lubricating oil cooling fan installations are shown on drawing SK-5165.

8.4.3.2. First priority in the cooling fan motor circuit is given to the engine coolant cooling fan motor (see paragraph 8.1.2.). This is accomplished by a priority (sequence) valve which insures pressure at the engine coolant cooling fan motor inlet before flow to the transmission lubricating oil cooling fan can take place.

8.4.4. Bilge Pump Motor Circuit.

8.4.4.1. The LVTPX11 vehicle is provided with a bilge evacuation system consisting of four hydraulically powered bilge pumps (see paragraph 8.5 for component description).

8.4.4.2. The four bilge pumps are driven by hydraulic motors and run continuously. The flow of hydraulic fluid to the bilge pump motors is automatically controlled by a load sensitive flow control valve. This arrangement insures maximum bilge fluid evacuating capacity during engine idling operations and reduces hydraulic power losses through the bilge pump motors to a minimum during land operation. Refer to paragraph 8.1.2. for a discussion of the bilge pump motor fluid power priority arrangement. The bilge pumps and system are described in paragraph 8.5.

8.4.5. Hydraulic Maintenance Connection.

The hydraulic system has connection points which provide for an external hydraulic power source for vehicle maintenance purposes.

8.5. Bilge Evacuation System.

8.5.1. The LVTPX11 vehicle bilge evacuation system is designed to incorporate four hydraulically driven centrifugal type bilge pumps, the necessary fittings, plumbing, and hardware to insure proper system operation, and meets the requirements set forth in paragraph 3.12.2.5 of BuShips Contract Specification SHIPS-A-4159. The bilge pumps are of all-aluminum construction except for bearings, shafts, and seals. The pump impellers are mounted directly to the hydraulic driving motor shaft.

8.5.2. Two pump - motor arrangements have been considered and are shown on drawings SK-4800 and SK-5214.

8.5.3. The bilge pump assembly shown on drawing SK-4800 consists of two major components; a standard bilge pump and a gear type hydraulic motor. The bilge pump assembly shown on drawing SK-5214 consists of two major components: a standard bilge pump and a piston type hydraulic motor. The contractor, at this time, recommends the bilge pump assembly shown on drawing SK-5214 for performance and reliability.

8.5.4. The four bilge pumps are located below the deck of the vehicle with two pumps in the aft portion of the crew compartment and two pumps in the engine compartment. Bilge fluids are pumped overboard through smooth-wall flexible tubing. Each bilge pump outlet is provided with a flapper type discharge valve to prevent the entry of sea water during rough water operation. The bilge system is shown on drawing SK-5165.

8.5.5. During the contractor's preliminary design phase a comparative study was made of the specified LVTPX11 bilge system and the bilge system of comparative types of vehicles.

8.5.6. This study has established the feasibility of reducing the number of bilge pumps in the LVTPX11 vehicle to two from the four pumps specified in paragraph 3.12.2.5 of BuShips Contract Specification SHIPS-A-4159, without any appreciable loss of vehicle operational capability or safety.

8.5.7. A summary of data resulting from the comparative bilge system study is tabulated in table 8-2. This table lists the vehicle type, number and type

TABLE 8-2, BILGE SYSTEM COMPARATIVE STUDY

VEHICLE TYPE	BILGE PUMPS TYPES & CAPACITY (GPM)	TOTAL BILGE PUMP CAPACITY (GPM) SEE NOTE	TYPE OF VEHICLE	VEHICLE DISPLACEMENT (LBS)	RATIO VEHICLE DISPLACEMENT TO BILGE PUMP CAPACITY LB/GAL/MIN
LVT1	2 Hand Operated 1 Engine Driven	100	Open	16,900	169
LVT2	Same as LVT1	500	Open	28,000	56
LVT3	1 Hand Operated Cap: 40 (GPM) 2 Engine Driven Cent. Cap: 250 (GPM) Each	500	Open	35,000	70
LVT3 (c)	Same as LVT3	500	Closed	39,700	79.4
LVT4	1 Hand Operated Cap: 40 (GPM) 1 Engine Driven Cap: 500 (GPM)	500	Open	33,600	67.2
LVTA4	Same as LVTA4	500	Covered	40,300	80.6
LVTP5 (Also Includes LVTH6, LVTR1, LVTE1	1 Electric Motor Driven Cap: 125 GPM 4 Hydraulically (Engine) Driven Cap: 300 GPM Each	1325	Covered	68,780	51.9
LVTP6	3-Engine Driven Cap: 150 GPM Ea.	450	Covered	40,000	88.8
LVTPX11	1-Electric Motor Driven Cap: 125 GPM 4-Hydraulically (Engine) Driven Cap: 300 GPM Each	1325	Covered	35,000	26.4
LVTPX11 (PROPOSED)	1-Electric Motor Driven Cap: 125 GPM 2-Hydraulically Driven (Engine) Cap: 300 GPM Each	725	Covered	35,000	48.2

NOTE: 1 - Includes Power Driven Pumps Only.

TABLE 8-2, BILGE SYSTEM COMPARATIVE STUDY of bilge pumps installed on each vehicle, individual bilge pump capacities, total bilge pump capacities, vehicle displacement (for size comparison), and whether the vehicle was of the open or covered type. Also listed is the ratio of vehicle displacement to total bilge pump capacity. A graphic display of the comparative bilge system data is tabulated in figure 8-1. This graphic display readily shows the relationship of the LVTPX11 vehicle to other vehicles of the same type.

8.5.8. As may be evidenced from a study of table 8-2 and figure 8-1, the use of only two bilge pumps in the LVTPX11 bilge system will change the displacement to total bilge pump capacity ratio from 26.4 to 48.2 which is more comparable to the same ratios in other types of LVT vehicles. Further benefits resulting from the use of two bilge pumps instead of four are a savings in cost and weight, reduced loads on the hydraulic system and engine, lower fuel consumption, and reduced maintenance.

8.5.9. On the basis of the bilge system comparative study, the contractor recommends reducing the number of bilge pumps required in the LVTPX11 vehicle bilge system from four to two and that paragraph 3.12.2.5 of BuShips Contract Specification SHIPS-A-4561 be modified accordingly.

BY C. Gregory DATE 11-4-62 SUBJECT Bilge System Comparative Study SHEET NO. 1 OF 1
CHKD. BY ______ DATE ______ JOB NO. 4561-4

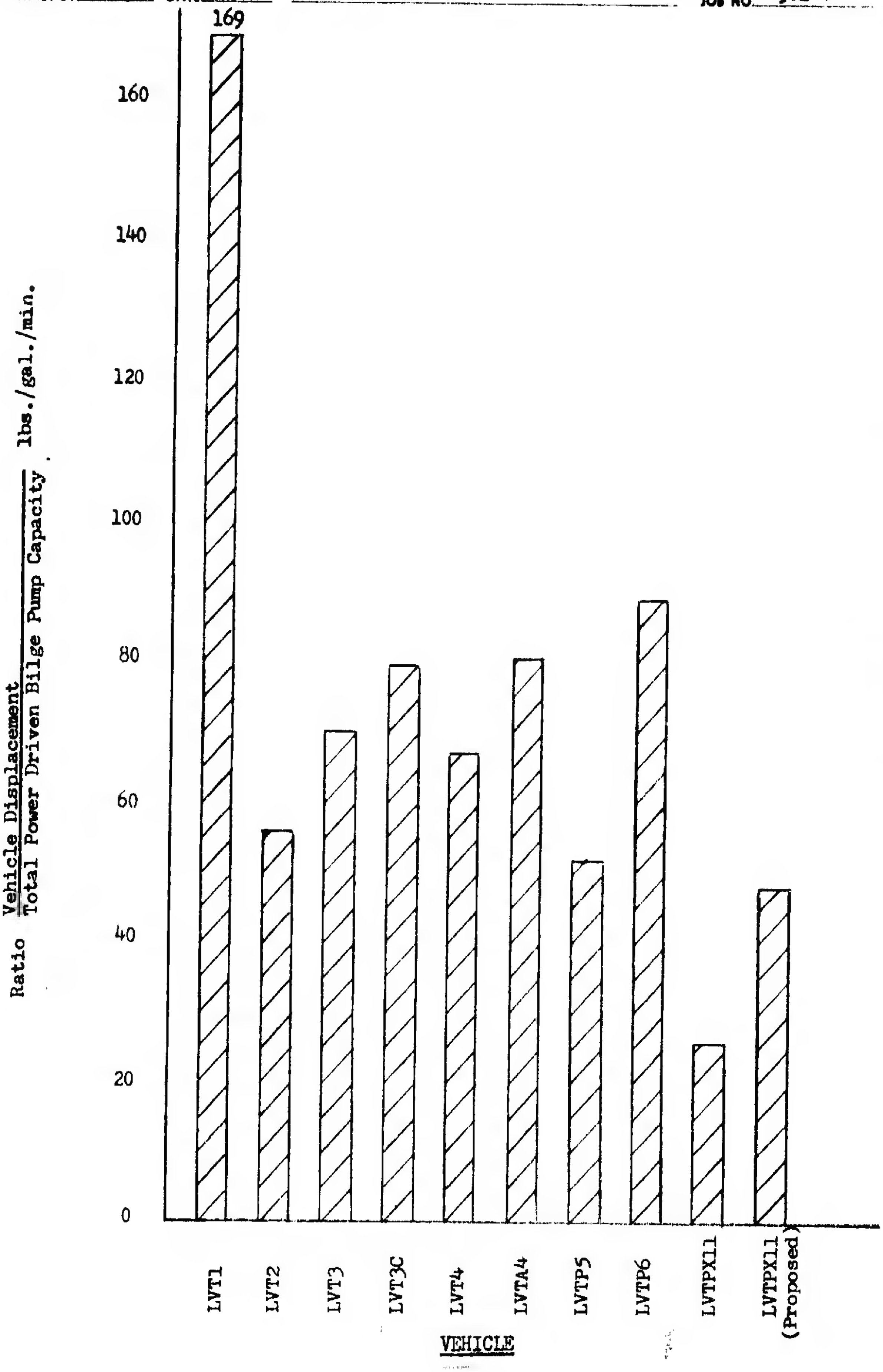


Figure 8-1 BILGE SYSTEM COMPARATIVE STUDY

9.0. FUEL SYSTEM

9.1. General Description.

9.1.1. The fuel system for the LVTPX11 (see drawing SK-5233) complies with BuShips Contract Specification SHIPS-A-4159, paragraph 3.12.2.3. Fuel is gravity fed from the fuel tanks through fuel control valves to fuel booster pumps. The fuel is then pumped through check valves into a fuel transfer tank. Fuel is drawn from the fuel transfer tank through a fuel strainer and fuel water separator to the engine fuel pump by the engine fuel pump. The fuel control valves are located at each fuel tank and have three positions: ON, OFF, and DRAIN. The OFF position allows maintenance of the water separator, booster pumps, and engine fuel filter. The DRAIN position allows fuel to be piped outside the vehicle. Check valves in the connecting piping between fuel tanks prevent interchange of fuel during the various vehicle attitudes. The transfer tank prevents engine flooding when engine is not in operation.

9.1.2. A two-way valve, incorporated in the fuel system between the check valves and transfer tank, provides a connection point for utilizing a portable auxiliary fuel tank for engine operation during vehicle manufacturing and testing. For testing, the auxiliary fuel tank will be used in place of the vehicle fuel tanks to prevent accumulation of volatile gases in the vehicle fuel tanks and system. This practice will insure maximum vehicle safety prior to field operation.

9.1.3. An overboard vent is provided for each tank and is designed to prevent sea water from entering the fuel tanks. The vents are positioned on the tanks to prevent fuel loss while negotiating 60 and 70 percent slopes with the tanks filled to capacity.

9.1.4. Each fuel tank liquid head is self-leveling. This is accomplished by the check valves and interconnecting piping. Each fuel tank has an electrical-type sending unit. (See Electrical System, Section 7.0, for details of the sending unit and fuel gauge.)

9.2. Fuel Specifications.

The Contractor recommends that in arctic operations, the Using Forces use JP-4 or JP-5 fuels, or fuels with similar freeze points, to eliminate the use of additional fuel heaters for proper starting and operation. The reason is as follows: the pour point means that the material will flow at a certain temperature, but it will not exhibit the flow characteristics of normal diesel fuel. However, the usable temperature, to avoid formation of wax, should be at least 15° to 20°F above the pour point. A formation of wax would collect at the filters and prevent the flow of fuel to the engine, should a vehicle be operated with diesel fuels, types I and II, at their pour points. To assure that diesel fuel temperature is 15° to 20°F above the pour point, heaters in the tank, filters, transfer tank and all lines would be required. It is felt that this would be a very costly winterization kit. Listed below in table 9-1 are the pour and freeze points of the various fuels:

Table 9-1. Fuel Specifications

Fuel	Freeze Point Maximum	Pour Point	Usable Ambient Temp. Range
JP-4 MIL-J-5624E	-76°F		-60° to 125°F
JP-5 MIL-J-5624E	-55°F		-40° to 125°F
Diesel, Marine Type I MIL-F-16884C (SHIPS)		0°F	-20° to 125°F
Diesel, Marine Type II MIL-F-16884C (SHIPS)		35°F	-15° to 125°F
Fuel Oil, Diesel, Referee Grade, MIL-F-45121	-76°F		-60° to 125°F

9.3. Fuel Tanks.

The fuel supply is stored in two fuel tanks which have a volume of 96.3 gallons each. Calculated unusable fuel amounts to 7 percent, or 6.3 gallons per tank. This is based on 5 percent, or 4.5 gallons, allowed for thermal expansion and 2 percent, or 1.8 gallons, retained in the tank due to the location of the tank outlet fitting. The net usable fuel capacity of each tank is 90 gallons. Fuel tanks are located forward on each side of the vehicle, one above each track channel. Each fuel tank is supported in the tank cavity by a system of hangers that are vulcanized to the tank, around the top edges at each corner. Metal hangers are located on the top edge at each corner of tank cavity adjacent to the tank hangers. The top surface of the fuel tank is supported by a fuel tank fitting bolted to the tank cavity. The bottom surface is held in place by the outlet fitting. To remove spillage or leakage in the tank cavity, a drain valve is provided. The fuel tank installation is shown in drawing SK-5231.

9.4. Fuel Water Separator.

Fuel water separator consists of two cartridges: a coalescer cartridge and a separator cartridge. Fuel flows through the coalescer cartridge where dirt is removed and water coalesced into large droplets, to the water separator cartridge where the water droplets are separated from the fuel and settle to the bottom of the cartridge housing. The clean, dry fuel flows from the separator cartridge to the engine fuel pump. Separated water in the bottom of the housing is drained through a valve. Water removal is 99.9 percent and dirt removal is 99 percent and sump capacity is 30.5 cubic inches in the water separator unit.

9.5. Fueling at Sea Arrangement.

Fuel fills are provided at each tank and are located on the top deck above each fuel tank. The fill tube is designed to extend above the deck and lock in position to permit refueling at sea. The fill tube is so designed that fueling can be accomplished with the fill tube in either the extended or retracted position. The fuel fill installation is shown in drawing SK-5232.

9.6. Selection of Components.

9.6.1. The fuel tanks selected for this vehicle are of nonmetallic, flexible cell construction per military specification MIL-T-6396B (ASG) Amendment 1, type II, class C. The material used in the construction of these fuel tanks meets the requirements for minimum tear from gunfire as per paragraph 4.5.19 of MIL-T-6396B.

9.6.2. Tubing is 6061-T6 aluminum conforming to specifications MIL-T-7081C (ASG).

9.6.3. Hose is a lightweight type conforming to specification MIL-H-19606A.

9.6.4. Rotary Teflon plug-type valves are used in the system and do not require special maintenance lubrication to keep them operative.

9.6.5. Check valves are low cracking pressure and self-cleaning types conforming to specifications MS28884 or MS28885.

9.6.6. Fuel strainer is of the disc type. The filter element is made up of filter discs. These discs can be cleaned and reused.

9.6.7. Fuel water separator consists of two cartridges, a coalescer and a separator cartridge conforming to MIL-F-15618 (SHIPS) class 2. This separator is used on Coast Guard vessels at the present time.

9.6.8. Tube fittings conform to MIL-F-5509 and MIL-F-18280 specifications. All pipe threads are NPTF (dry seal) pipe threads.

9.6.9. All fasteners conform to MS Specifications.

9.6.10. Quick disconnects conform to specification MS24334.

9.6.11. Fuel booster pumps ORD 7748814 are an auto-pulse type, electrically driven.

10.0. LUBRICATION SYSTEM

10.1. General.

Components for the LVTPX11 vehicle are designed or selected to minimize required manual lubrication. Permanently lubricated components or units requiring minimal manual lubrication are given preference in system designs in order to increase vehicle reliability and reduce vehicle maintenance.

10.2. Lubrication of Engine, Transfer Gearbox, Steering Transmission, and Final Drives.

Self-contained lubrication systems are included in the engine, steering transmission, and final drives. The engine lubrication system is a full pressure type utilizing a gear-type oil pump capable of operation to an engine inclination of 45 degrees. The transfer gearbox has a separate spray lubrication system with remote reservoir which provides an oil sump and effectively augments cooling. The steering transmission includes a pressure lubrication system integral with the hydraulic system through a common valve arrangement. The final drives incorporate a wet sump-and-splash lubrication system.

10.3. Lubrication of Drive Shaft Joint Bearings, Roadwheel Bearings, Supporting Arm Seals, Idler Bearings, Towing Hitch, and Machine Gun Turret.

Manual lubrication through grease fittings is required for the drive shaft joint bearings, the roadwheel bearings and supporting arm seals, the idler bearings, the towing hitch, and the machine gun turret.

10.4. Selection of Bearings.

The use of reinforced plastic bearings and sealed cable units permits maintenance-free components to be utilized in the hatch cover hinges and locks, the ramp hinge and release mechanism, the cupola hinges, and in all components of the control system. These components are designed to give maintenance-free usage throughout their normal operative life cycle.

11.0. FIRE EXTINGUISHING SYSTEM

11.1. General Description.

A simple and reliable CO_2 fire extinguishing system has been selected for the LVTPX11 to assure the maximum fire protection for the vehicle while allowing minimum CO_2 concentration in the personnel compartment (see drawing SK-5234). The extinguishing agent is an inert gas which extinguishes a fire by diluting the oxygen content in a given space to a point where it will not support combustion. This is a clean, dry, noncorrosive system that will not damage or injure anything with which it may come in contact. Liquefied carbon dioxide is stored in a steel cylinder under high pressure. One of the most valuable properties of this gas is its amazingly high expansion ratio of 450 to 1. A cylinder completely empties without any additional pressure.

11.2. System Design Data.

11.2.1. Design Requirement Number 1.

To extinguish a gasoline fire in the engine compartment with the aspiration air fan motor off and the engine running.

11.2.1.1. The following tabulated data pertains to design requirement number 1:

A. Five-pound CO_2 bottle 1

B. Design concentration of CO_2 for a gasoline fire (National Fire Protection Association data) 34%

C. Total unoccupied engine compartment volume. . . . . . . 42 cu ft

D. Openings, not closed, when CO_2 is discharged . . . . . . 1.14 sq ft

1. Between engine compartment and personnel compartment . . . . 1.0 sq ft

2. Scavenge blow pipe . . . . 0.14 sq ft

E. Volume factor of CO_2 for a 34% concentration (National Fire Protection Association data). . . . . 14 cu ft/lb CO_2

F. Maximum effective volume of CO_2 for a 34% concentration Note: 1 lb of CO_2 is required for every square foot of opening, therefore, (5 - 1.14) x 14 = 54.04 . . . . 54.04 cu ft

11.2.1.2. Results of this data indicate that one five-pound CO_2 bottle will discharge 1.28 times the required volume to extinguish a gasoline fire in a 42 cubic foot volume, resulting in a CO_2 concentration of approximately 44 percent and an oxygen content of approximately 11.7 percent.

11.2.2. Design Requirement Number 2.

To extinguish a gasoline fire in the engine compartment when the engine is not running, so that the degree of CO_2 concentration in the personnel compartment at this time is not injurious to human life.

11.2.2.1. The following tabulated data pertains to design requirement number 2:

A. Total unoccupied volume in the engine compartment and personnel compartment . . . . 636 cu ft

B. CO_2 concentration, by volume

1. cu ft/lb CO_2 = 636 cu ft $\div$ 3.86 lbs CO_2 = 165 cu ft/lb

2. % CO_2 = 8.35 cu ft/lb CO_2 $\div$ 165 cu ft/lb = 5 %

11.2.2.2. Results of this data indicate that a five percent carbon dioxide concentration will cause one to breathe rapidly but will otherwise have no important effect for relatively short exposures. At concentrations of about 20 percent, death would follow in approximately 20 to 30 minutes.

11.2.3. Fixed System.

11.2.3.1. The fixed fire extinguishing system consists of one steel cylinder, a simple piping system, and a discharge nozzle. The five-pound cylinder, which is located on the port side of the personnel compartment, forward of the engine compartment bulkhead, has a discharge head for instantaneous release of the carbon dioxide from the cylinder. Simple piping from the cylinder terminates at one discharge nozzle located on the engine compartment bulkhead in the engine compartment. An electrical thermostat and a remote manual control system are the two methods used to release the CO_2.

11.2.3.2. The electrical thermostat system releases the CO_2 at a predetermined temperature or at a fixed rate of rise in temperature. This type of release is a safety feature that protects the vehicle both when in operation and when it is unmanned. The exact location of the thermostat will be determined experimentally with thermocouples after the vehicle has been completed.

11.2.3.3. The remote control system consists of two pull handles. One handle is accessible from outside the vehicle and is located amidships on the port side. The other is located adjacent to the driver's station. These pull handles are connected to the cylinder release mechanism with 1/16-inch diameter, extra-flexible, corrosion-resistant steel cable. The cable is housed in 1/4-inch diameter aluminum tubing and connected to the handles and discharge head with fittings. (The CO_2 installation is shown in drawing SK-5235.)

11.2.3.4. Both the electrical thermostat and the remote manual control systems operate in conjunction with the ventilation system to permit the CO_2 to effectively extinguish a fire. When the CO_2 bottle is discharged either electrically or manually, the CO_2 line pressure switch closes, actuating an indicating fuse located next to the CO_2 bottle. If the ventilating blower is not operating, and the master switch is off, the CO_2 will be discharged into the engine compartment and the discharge indicator will be actuated which indicates that the system has been discharged. If the ventilating blower is operating, and the system discharged either manually or automatically, the air diverter is actuated and the ventilating blower is deenergized. This closes the engine compartment scavenge air inlet, and reduces the loss of CO_2 out of the engine compartment through the air exhaust opening. A time delay relay prevents the fan from operating for a predetermined period of time to assure that the fire is extinguished by permitting the high concentration of CO_2 to remain in the compartment.

11.2.4. Portable Unit. In addition to the integral CO_2 system, a portable five-pound CO_2 cylinder is provided. It is located on the forward port side of the cargo area.

11.3. Selection of Components. Components that are used in the fire extinguishing system conform with Ordnance and Military Specifications, except for electric discharge connector.

11.3.1. Tube fittings conform to MIL-F-5509 and MIL-F-18280 specifications.

11.3.2. Tube clamps conform to MS specifications.

11.3.3. All fasteners conform to MS specifications.

11.3.4. All tubing conforms to MIL-T-7081C (ASG) specifications.

11.3.5. Manual release cable conforms to MIL-C-5424 specifications.

11.3.6. Electric discharge connector (The Fyr-Fyter Co. Number 94255).

12.0. WINTERIZATION KIT

12.1. General.

12.1.1. In accordance with BuShips Contract Specification SHIPS-A-4159, cold weather starting aids and heaters for the embarked personnel are provided for operation in arctic zones in ambient temperatures as low as minus 40°F. In this subzero temperature it is required that the personnel compartment be heated to a temperature of plus 25°F.

12.1.2. To determine the number and size of available military heaters required for this application, a heat loss study was conducted. Assumptions made in performing the necessary calculations included the following:

A. Outside air moving past the vehicle = 40 mph.

B. Uniform air movement inside the vehicle = 100 fpm.

12.1.3. The calculations, given in Appendix I immediately following this section, were performed for the crew compartment only, and the heater for the machinery compartment was selected which would provide the maximum heat output consistent with the space available.

12.2. Heaters.

12.2.1. Based on the calculations, two 30,000 Btu per hour heaters were selected for the crew compartment, and one 60,000 Btu per hour heater was selected for the machinery compartment. The crew compartment heaters are installed in the aft end of the crew compartment, near the upper deck, as illustrated on Sheet 1 of drawing SK-5242. The output of each heater is ducted to the floor of the vehicle and discharged under the troop foot rest. The discharge is dispersed both forward and aft to provide heat as uniformly as possible in this compartment. The vertical duct will cause some inconvenience to two troops on each side of the vehicle since a duct will be between them. The duct size and location were established so as to maintain a free cargo area while providing a minimum amount of seating inconvenience.

12.2.2. The heater in the machinery compartment is located under the starboard radiator compartment as shown on Sheet 2 of drawing SK-5242. It will distribute heated air toward the engine, transmission, and batteries to provide easier starting. This heater is of the 60,000 Btu output range. In both crew and machinery compartment installations, every precaution was taken to safeguard against exhaust gases from the heater entering the vehicle, by exhausting out through the top deck away from the air inlets.

12.2.3. Each heater is a multifuel unit, and is radio-suppressed per Specification MIL-S-10379. Each unit consists of a generator-type vaporizing burner and combustion chamber, which discharges the products of combustion through a heat exchanger surrounding the combustion chamber, and a blower assembly supplying combustion and ventilating air. These heaters are controlled by the use of manual switches in the control box located near the driver.

APPENDIX I OF SECTION 12

BASIC INFORMATION OBTAINED FROM VEHICLE CONFIGURATION

	Material Thickness in Feet	Area in Square Feet
1. Track Channel Side, Port and Starboard	.0679	59.4 total
2. Machine Gun Turret (top)	.0624	8.91
(side)	.099	1.31
3. Hull Bottom	.0416	87.3
Cargo Deck	.0208	87.3
4. Bow, less Ramp	.099	11.5
5. Ramp (inner)	.0625	35.6
(outer)	.0364	37.8
6. Cupolas (2), (top)	.0624	10.55 total
(side)	.099	5.89 total
7. Upper Deck	.0624	142.3
8. Hull Sides, Port and Starboard	.099	86.3 total
9. Track Channel Top, Port and Starboard	.0625	51.8 total

From Tables and Graphs: (See References)

K_m for 5456 Aluminum at 77°F = 67.5.

h_i = 1.7 at 100 fpm inside air velocity.

h_o = 9.7 at 40 mph outside air velocity.

C = 1.02 per four inches of dead air space.

CREW COMPARTMENT HEAT LOSS

Explanation of symbols and formula:

General Formula: (Single Wall)

$$H_L = \frac{A\,(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m1}} + \frac{1}{h_o}}$$

(Double Wall with air space):

$$H_L = \frac{A\,(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m1}} + \frac{1}{C} + \frac{X_2}{K_{m2}} + \frac{1}{h_o}}$$

H_L = Heat loss through parallel layers of material in British thermal units per hour.

$X_1, X_2 \ldots X_n$ = Thickness of conducting layers in feet.

$(t_1 - t_2)$ = Temperature difference between inner and outer surfaces in degrees F.

$K_{m1}, K_{m2} \ldots K_{mn}$ = Mean thermal conductivity of individual layers for the temperature range in Btu/(hr)(sq ft)(deg F/ft)

A = Area in square feet, taken perpendicular to the direction of heat flow.

h_i = Inside surface conductance in Btu/(hr)(sq ft)(deg F).

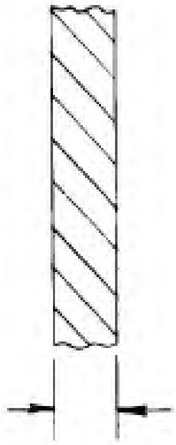
h_o = Outside surface conductance in Btu/(hr)(sq ft)(deg F).

C = Air space conductance in Btu/(hr)(sq ft)(deg F).

TRACK CHANNEL SIDES, PORT AND STARBOARD

CREW COMPARTMENT

HEAT LOSS CALCULATIONS



5456 Aluminum

$X_1 = .0679$ ft

$A = 59.4$ sq ft

$K_{m_1} = 67.5$

$h_i = 1.7$

$h_o = 9.7$

$t_1 - t_2 = 25^oF - (-40^oF) = 65^oF$

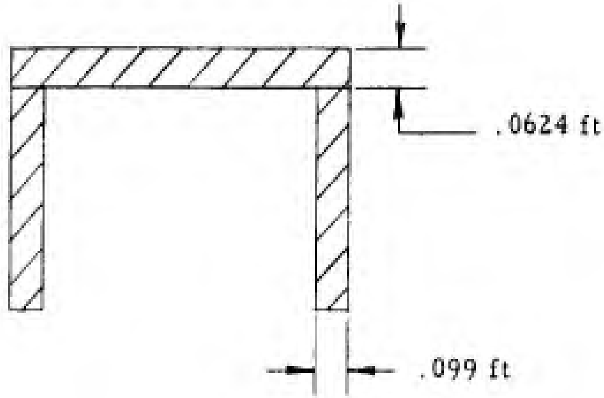
$$H_L = \frac{A\,(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m_1}} + \frac{1}{h_o}}$$

$$H_L = \frac{(59.4)\,(65)}{\frac{1}{1.7} + \frac{.0679}{67.5} + \frac{1}{9.7}}$$

$H_L = 5598$ Btu/hr

MACHINE GUN TURRET

HEAT LOSS CALCULATIONS



5456 Aluminum

$X_1 = .0624$ ft

$X_2 = .099$ ft

$A_1 = 8.91$ sq ft

$A_2 = 1.31$ sq ft

$K_{m_1} = 67.5$

$h_i = 1.7$

$h_o = 9.7$

$t_1 - t_2 = 25^{o}F - (-40^{o}F) = 65^{o}F$

$$H_L = \frac{A_1 (t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m_1}} + \frac{1}{h_o}} + \frac{A_2 (t_1 - t_2)}{\frac{1}{h_i} + \frac{X_2}{K_{m_1}} + \frac{1}{h_o}}$$

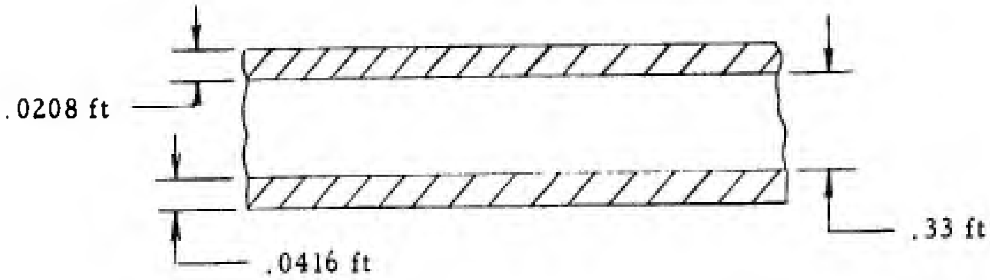
$$H_L = \frac{(8.91)(65)}{\frac{1}{1.7} + \frac{.0624}{67.5} + \frac{1}{9.7}} + \frac{(1.31)(65)}{\frac{1}{1.7} + \frac{.099}{67.5} + \frac{1}{9.7}}$$

$H_L = 925$ Btu/hr

HULL BOTTOM AND CARGO DECK

CREW COMPARTMENT

HEAT LOSS CALCULATIONS



5456 Aluminum

$C = 1.02$

$X_1 = .0208$ ft

$X_3 = .0416$ ft

$A = 87.3$ sq ft

$K_{m_1} = 67.5$

$h_i = 1.7$

$h_o = 9.7$

$t_1 - t_2 = 25^oF - (-40^oF) = 65^oF$

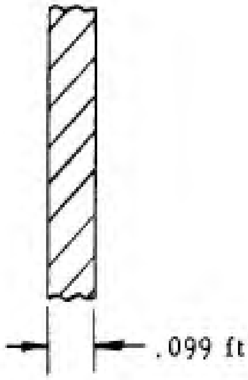
$$H_L = \frac{A(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m_1}} + \frac{1}{C} + \frac{X_3}{K_{m_1}} + \frac{1}{h_o}}$$

$$H_L = \frac{(87.3)(65)}{\frac{1}{1.7} + \frac{.0208}{67.5} + \frac{1}{1.02} + \frac{.0416}{67.5} + \frac{1}{9.7}}$$

$H_L = 3391$ Btu/hr

BOW, LESS RAMP

HEAT LOSS CALCULATIONS



5456 Aluminum

$A = 11.5$ sq ft

$X_1 = .099$ ft

$h_i = 1.7$

$h_o = 9.7$

$K_{m_1} = 67.5$

$t_1 - t_2 = 25^oF - (-40^oF) = 65^oF$

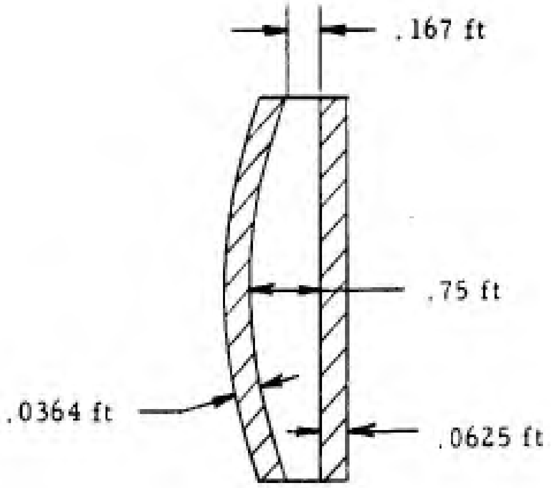
$$H_L = \frac{A\,(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m_1}} + \frac{1}{h_o}}$$

$$H_L = \frac{(11.5)\,(65)}{\frac{1}{1.7} + \frac{.099}{67.5} + \frac{1}{9.7}}$$

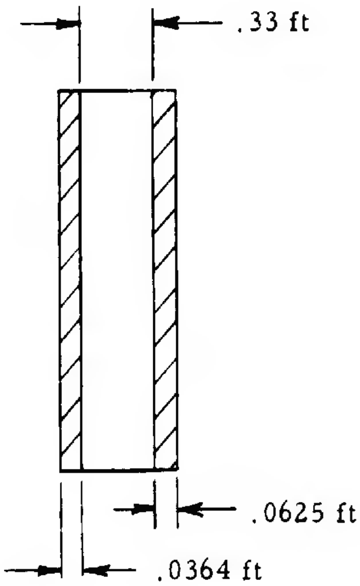
$H_L = 1,084$ Btu/hr

RAMP

HEAT LOSS CALCULATIONS



ACTUAL CROSS SECTION



ASSUMED CROSS SECTION

5456 Aluminum

$C = 1.02$

$X_1 = .0364$ ft

$X_3 = .0625$ ft

$A_1 = 37.8$ sq ft (outer plate)

$A_2 = 35.6$ sq ft (inner plate)

$A = 36.7 \text{ sq ft} = \dfrac{A_1 + A_2}{2}$

$K_{m_1} = 67.5$

$h_i = 1.7$

$h_o = 9.7$

$t_1 - t_2 = 25^oF - (-40^oF) = 65^oF$

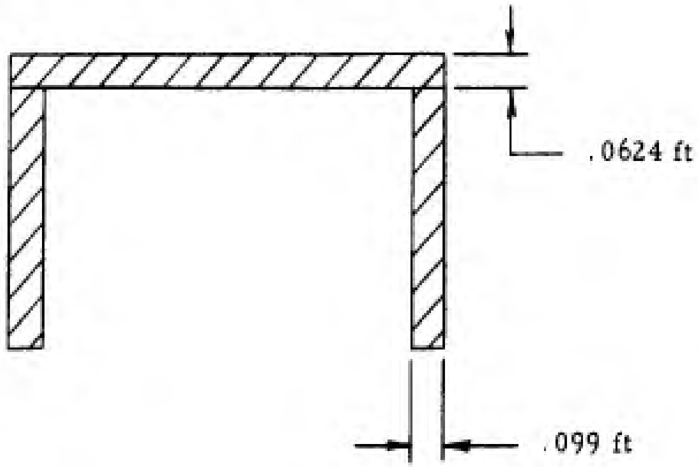
$$H_L = \frac{A(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m_1}} + \frac{1}{C} + \frac{X_3}{K_{m_1}} + \frac{1}{h_o}}$$

$$H_L = \frac{(36.7)(65)}{\frac{1}{1.7} + \frac{.0364}{67.5} + \frac{1}{1.02} + \frac{.0625}{67.5} + \frac{1}{9.7}}$$

$$H_L = 1{,}426 \text{ Btu/hr}$$

CUPOLAS (2)

HEAT LOSS CALCULATIONS



5456 Aluminum

X_1 = .0624 ft

X_2 = .099 ft

A_1 = 10.55 sq ft

A_2 = 5.89 sq ft

h_i = 1.7

h_o = 9.7

K_{m_1} = 67.5

$t_1 - t_2 = 25^oF - (-40^oF) = 65^oF$

$$H_L = \frac{A_1(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m_1}} + \frac{1}{h_o}} + \frac{A_2(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_2}{K_{m_1}} + \frac{1}{h_o}}$$

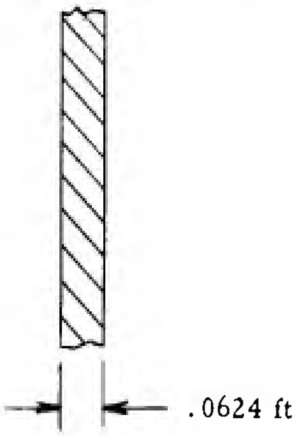
$$H_L = \frac{(10.55)(65)}{\frac{1}{1.7} + \frac{.0624}{67.5} + \frac{1}{9.7}} + \frac{(5.89)(65)}{\frac{1}{1.7} + \frac{.099}{67.5} + \frac{1}{9.7}}$$

H_L = 1,550 Btu/hr

UPPER DECK

CREW COMPARTMENT

HEAT LOSS CALCULATIONS



5456 Aluminum

$X_1 = .0624$ ft

$h_i = 1.7$

$h_o = 9.7$

$A = 142.3$ sq ft

$K_{m_1} = 67.5$

$t_1 - t_2 = 25^\circ F - (-40^\circ F) = 65^\circ F$

$$H_L = \frac{(A)\,(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m_1}} + \frac{1}{h_o}}$$

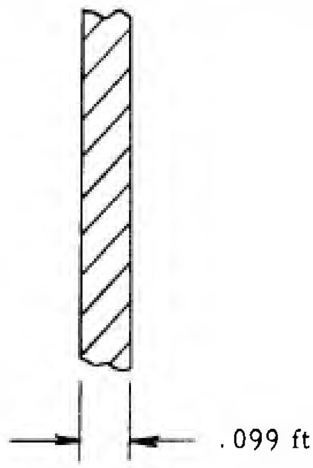
$$H_L = \frac{(142.3)\,(65)}{\frac{1}{1.7} + \frac{.0624}{67.5} + \frac{1}{9.7}}$$

$H_L = 13,413$ Btu/hr

HULL SIDES, PORT AND STARBOARD

CREW COMPARTMENT

HEAT LOSS CALCULATIONS



5456 Aluminum

$X_1 = .099$ ft

$A = 86.3$ sq ft

$h_i = 1.7$

$h_o = 9.7$

$K_{m1} = 67.5$

$t_1 - t_2 = 25^{o}F - (-40^{o}F) = 65^{o}F$

$$H_L = \frac{(A)\,(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m1}} + \frac{1}{h_o}}$$

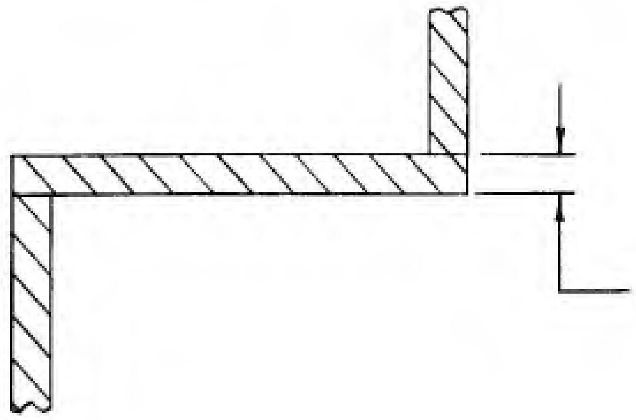
$$H_L = \frac{(86.3)\,(65)}{\frac{1}{1.7} + \frac{.099}{67.5} + \frac{1}{9.7}}$$

$H_L = 8,135$ Btu/hr

TRACK CHANNEL TOP, PORT AND STARBOARD

CREW COMPARTMENT

HEAT LOSS CALCULATIONS



5456 Aluminum

X_1 = .0625 ft

A = 51.8 sq ft

K_{m_1} = 67.5

h_i = 1.7

h_o = 9.7

$t_1 - t_2 = 25^oF-(-40^oF) = 65^oF$

$$H_L = \frac{A\,(t_1 - t_2)}{\frac{1}{h_i} + \frac{X_1}{K_{m_1}} + \frac{1}{h_o}}$$

$$H_L = \frac{(51.8)\,(65)}{\frac{1}{1.7} + \frac{.0625}{67.5} + \frac{1}{9.7}}$$

H_L = 4,882 Btu/hr

SUMMARY

Results of Heat Loss Calculations

	5456 Aluminum in Btu Hr
1. Track Channel Sides, Port and Starboard	5,598
2. Machine Gun Turret	925
3. Hull Bottom and Cargo Deck	3,391
4. Bow, less Ramp	1,840
5. Ramp	1,426
6. Cupolas (2)	1,550
7. Upper Deck	13,413
8. Hull Sides, Port and Starboard	8,135
9. Track Channel Top, Port and Starboard	4,882
Total heat loss	41,160

References: Marks, "Mechanical Engineer's Handbook," 5th edition, McGraw-Hill Book Company, Inc., New York, 1951.

Brown and Marco, "Introduction to Heat Transfer", 2nd edition, McGraw-Hill Book Co., Inc., New York, 1951.

Brochure-"Aluminum Wrought Alloys", form number 780-8-9(25-760), Reynolds Metals Company, Richmond, Va.

13.0. VEHICLE AIR SYSTEMS

13.1. General.

It had been previously determined that four separate air systems, in addition to engine-transmission cooling ducts, were necessary to fulfill the fresh air requirements of the LVTPX11. These systems consist of the crew ventilation system, the engine air aspiration system, the bilge scavenge system, and the engine compartment scavenge system. All four systems have been integrated into a single, interdependent system to provide maximum efficiency, with the lowest weight.

13.2. Description.

13.2.1. The entire system consists of two outside air intake openings in the crew compartment at the forward end of the vehicle behind the cupolas, one outside air intake in the engine compartment, air intake openings in the bulkhead from the crew compartment, and one exhaust opening from the engine compartment to the outside. (See drawing SK-5204, Air System Installation. Sheet 2 of SK-5204 shows the typical design of the covers.) Ballistic protection has been maintained by providing a 3/4 inch thick cover on the crew compartment openings and a 1/4 inch thick cover on the engine compartment openings. The cover is securely welded to a 3/4 inch diameter rod, and the assembly is supported in a spider welded inside of the air opening. A spring holds the cover in the open position normally. However, if water should momentarily envelop the cover to a depth of 0.339 to 0.476 feet due to wave or surfing action, the weight of the water on the cover will overcome the resistance of the spring causing the cover to close, thus preventing the entrance of large amounts of water through the opening. The cover can be closed and locked from outside of the vehicle by exerting a force of approximately 25 to 35 pounds on top of the cover and turning clockwise until the cover engages the locking slot.

13.2.2. The crew compartment air intake openings are provided with a drain duct to permit any water which may enter the opening to drain directly into the bilges.

13.2.3. The engine compartment air intake opening is equipped with a throttling valve which permits a variable amount of air to be taken into the engine compartment through this opening. A lock is provided to hold the valve in any selected position, providing various degrees of opening, or it may be closed completely. The volume of air entering the vehicle through the intake openings in the crew compartment may be controlled by changing the setting on the throttling valve.

13.2.4. The engine compartment air outlet (see drawing SK-5204, Sheet 3) is equipped with an electric blower rated at 600 cubic feet per minute at a static pressure

(Ps) of 0.75 inch W.G. The electric motor meets the requirements of MIL-M-8609A, Class B. The blower is a vane-axial type capable of operating against the pressures encountered in this installation.

13.2.5. An air diverter is incorporated into the blower exhaust duct. This diverter normally permits air to be drawn from the top of the engine compartment into a large opening formed by the sides of the duct. Locating the large inlet opening near the top of the vehicle permits the warmer air from the engine compartment to be removed and exhausted directly to the outside. The air diverter is operated by a solenoid when the bilge scavenge switch is actuated on the instrument panel or when the CO_2 is discharged. The large opening in the inlet duct is then closed and air is drawn through a smaller bilge scavenge duct and exhausted directly to the outside. The bilge scavenge duct is of smaller diameter to increase the intake velocity, and the opening is located near the deck of the vehicle in the engine compartment to efficiently remove any accumulation of volatile fumes from the bilges.

13.2.6. All air will enter the vehicle as a direct result of the electric blowers and the aspiration requirements of the engine. This air may enter the engine compartment directly from the outside through the engine compartment air intake, or it may enter the engine compartment via the bulkhead openings from the crew compartment air intakes first traversing the length of the crew compartment.

13.3. Conditions of Operation.

To determine the fan requirements and the air flow through the crew and engine compartments, the following conditions were considered and analyzed. Refer to drawing SK-5227, air schematic. Note: In all conditions the ventilating blower is operating and all crew access and cargo hatches are closed.

Condition Number 1	Maximum air requirement with full engine rpm and a minimum crew air supply of 31 x 20 = 620 cubic feet per minute with all inlets open.
Condition Number 2	Maximum air flow obtained with full engine rpm and all inlets open.
Condition Number 3A	Maximum air flow through the crew compartment with the aft inlet closed, forward inlets open, and the engine at full rpm.
3B	Maximum air flow through the crew compartment with the aft inlet closed, forward inlets open, and the engine at idle rpm.
Condition Number 4A	Maximum air flow through the engine compartment with the forward inlets closed, aft inlet open, and the engine at full rpm.

4B Maximum air flow through the engine compartment with the forward inlets closed, aft inlet open, and the engine at idle rpm.

Condition Number 5 Maximum air flow with the engine OFF, aft inlet closed, and the forward inlets open.

Condition Number 6 Maximum air flow in the engine compartment during bilge scavenging with the aft inlet open.

13.4. Air Calculations.

13.4.1. The air calculations are based on the following conditions:

A. Air density = 0.075 pounds per cubic foot.

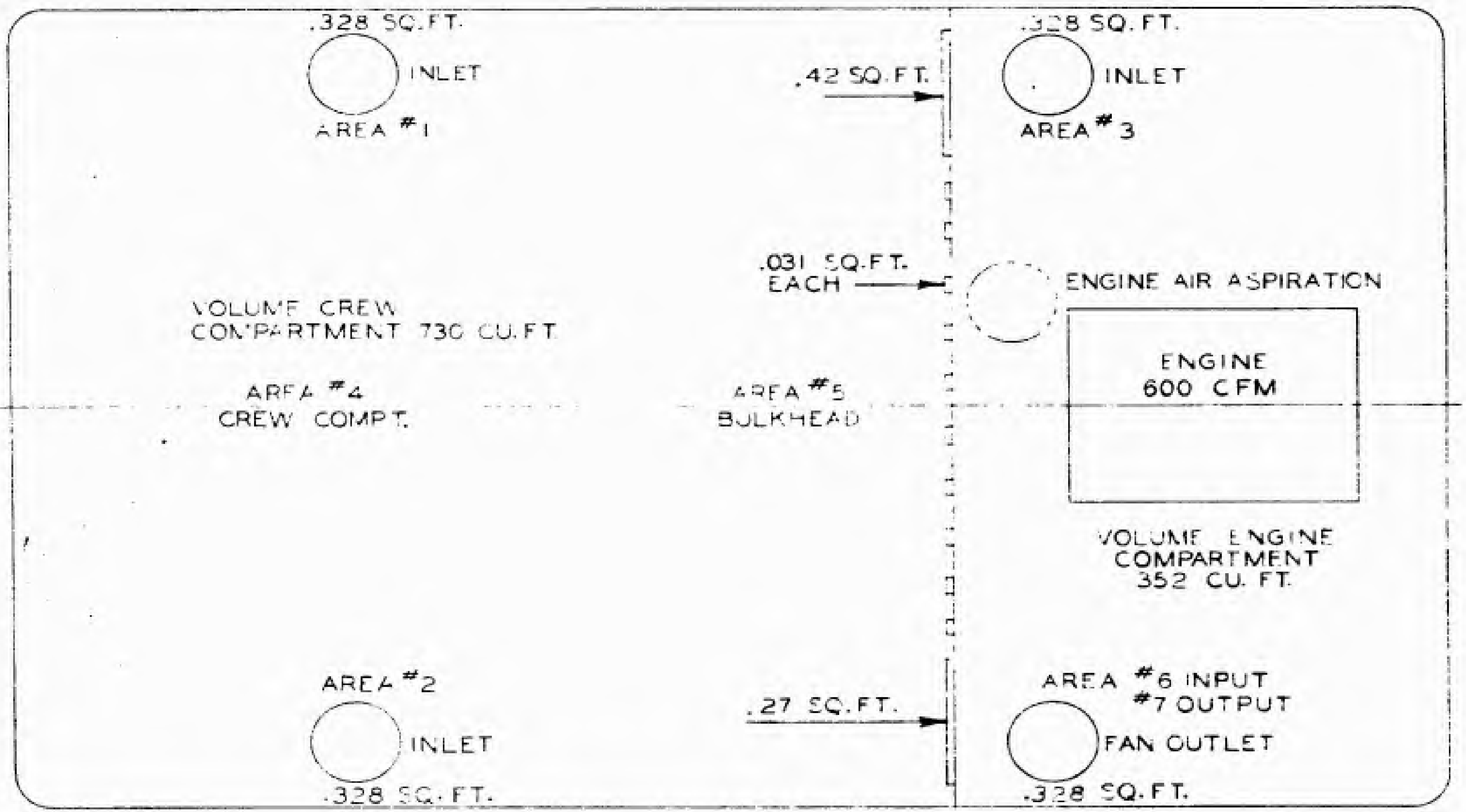
B. Ambient temperature of 70°F.

C. Sea level elevation.

D. Dry air.

13.4.2. To determine the air friction losses through the various openings and ducts in the vehicle, the equivalent duct lengths and air friction tables from the American Society of Heating and Air Conditioning Engineers' Handbook have been utilized.

13.4.3. The various air inlets, associated ducts, bulkhead openings, and air outlet are referred to as areas and are numbered on figure 13-1.



TOTAL AREAS

AIR SYSTEM SCHEMATIC

Figure 13-1

A. The equivalent 8-inch diameter duct length for Area Number 1 (Forward air inlet) is determined as follows:

1. It is estimated that the air in traveling around the cover is turned a total of 125 degrees. This is equivalent to one round section Mitre 90 degree elbow and one round section Mitre 45 degree elbow.

   Equivalent length 90 degree elbow = $53D = \frac{53 \times 8}{12} =$ 35.3 feet.

   Equivalent length 45 degree elbow = $\frac{53D}{2} = \frac{53 \times 8}{2 \times 12} =$ 17.7 feet.

   Total equivalent 8-inch dia duct = 35.3 + 17.7 = 53.0 feet.

2. Straight 8-inch duct length of 4 inches $= \frac{4}{12} = .33$ feet

3. After leaving the straight duct the air is turned 90 degrees and passes through two rectangular openings totaling 48 square inches cross sectional area. This area is approximately equivalent to one 8-inch diameter duct. The equivalent 8-inch duct length for a rectangular section Mitre 90 degree elbow $= 76D = \frac{76 \times 8}{12} = 50.6$ feet.

4. For the purpose of determining air friction losses, the amount of air flowing through the water drain will be neglected at this time.

5. The air leaving the air intake openings is subject to an abrupt expansion.

   a. The velocity V_1 of the air at the opening at 350 cfm $= \frac{350 \times 144}{48} = 1050$ feet per minute.

   b. The velocity V_2 of the air in the crew compartment at (2) (350) $= \frac{700}{51} = 13.7$ feet per minute.

$$\frac{V_1}{V_2} = \frac{1050}{13.7} = 76.6$$

   Since the ratio of V_1/V_2 is greater than one, the equivalent duct length equals $1D = \frac{8}{12} = 0.66$ feet.

6. The total equivalent 8-inch diameter duct length for Area Number 1 (Forward Air Inlet) = 53.00 + 0.33 + 50.60 + 0.66 = 104.59 feet.

B. The equivalent 8-inch diameter duct length for Area Number 2 (Forward Air Inlet) is identical to Area Number 1 = 104.59 feet.

C. The equivalent 8-inch diameter duct length for Area Number 3 (Aft Air Inlet) is determined as follows:

1. The air entering through Area Number 3 is subject to the same loss as paragraph 13.4.3. A1 above:

   Equivalent length = 53.0 feet.

2. Straight length of 8-inch diameter duct, 54 inches long = $\frac{54}{12} = 4.5$ feet.

3. The abrupt expansion of the air leaving the duct is equal to paragraph 13.4.3 A5 above:

   Equivalent length = .66 feet.

   Total equivalent 8-inch diameter duct length through Area Number 3 aft air intake = 53.00 + 4.50 + 0.66 = 58.16 feet.

D. The air friction losses through Area Number 4 (Cargo and Personnel Area) is negligible due to the large cross sectional area of 51 square feet.

E. The equivalent 8-inch diameter duct length for Area Number 5 (Engine Bulkhead) is determined by considering the openings through the engine bulkhead individually. These openings total one square foot of cross sectional area and consist of ten 2.4-inch diameter holes and two rectangular holes of 60 square inches each. The rectangular hole on the port side is partially filled with control and electrical cables. Therefore, the cross sectional area is assumed to be 38.7 square inches.

   Total cross sectional area = 38.7 + 60.0 + (10) (4.53) ≈ 144 sq in = 1 sq ft.

   The friction loss of air passing through the bulkhead at 700 cfm is .0236 inches of water. This value is equivalent to 3.03 feet of 8-inch duct and is determined as follows:

1. Round hole of 2.4-inch diameter = 4.53 sq in.

   Assume air flow equals 21 cfm through each round hole.

   Equivalent duct length for sharp-edge round orifice = 30D=

   $\frac{30 \times 2.4}{12} = 5.75$ feet.

   Friction loss for 21 cfm and 2.4-inch diameter tube = 0.41

   inch per 100 feet = $\frac{0.41 \times 5.75}{100} = \underline{\underline{0.0236 \text{ inch Water Gauge}}}$.

2. Rectangular opening on starboard side of 60 square inches:

   Assume air flow = 305 cfm

   Equivalent round tube diameter for a rectangular hole of

   60 square inches = 8.75 inches diameter.

   Equivalent duct length for abrupt entrance = 30D = $\frac{30 \times 8.75}{12}$ =

   21.84 feet.

   Friction loss for 305 cfm and 8.75 diameter duct = 0.1075

   inch per 100 feet = $\frac{0.1075 \times 21.84}{100} = \underline{\underline{0.0236 \text{ inch WG}}}$

3. Rectangular opening on port side of 38.7 square inches:

   Then air flow = 700 - (10x21) - 305 = 185 cfm

   Equivalent tube diameter for a rectangular hole of 38.7 square

   inches = 7-inches diameter.

   Equivalent duct length for abrupt entrance = 30D = $\frac{30 \times 7}{12}$ =

   17.5 feet.

   Friction loss for 185 cfm and 7-inch diameter duct = 0.135

   inch per 100 feet = $\frac{0.135 \times 17.5}{100} = \underline{\underline{0.0236 \text{ inch WG}}}$

4. The friction loss in inches of water must be identical for all inlet holes in the bulkhead. Since the friction losses are equal for the areas described in paragraphs 13.4.3. E1, E2, and E3, the assumed values for air flow through these holes are correct.

5. To determine the equivalent duct length of an 8-inch diameter duct for Area Number 5 at 700 cfm, the friction loss in inches of water = 0.78 per 100 feet.

$$\text{Length} = \frac{0.0236 \times 100}{0.78} = 3.03 \text{ feet.}$$

F. The equivalent 8-inch diameter duct length for Area Number 6 (Fan Inlet) is determined as follows:

1. The air entering the input side of the exhaust fan is subjected to an abrupt entrance = $30D = \frac{30 \times 8}{12} = 20.0$ feet.

2. The air after entering the duct is turned 90 degrees, which is equivalent to a round-section Mitre 90 degree elbow = $53D = \frac{53 \times 8}{12} = 35.3$ feet.

3. Total input equivalent 8-inch diameter duct = 20.0 + 35.3 = 55.3 feet.

G. The equivalent 8-inch diameter duct length for Area Number 7 (Fan Outlet) is determined as follows:

1. The air leaving the output side of the fan is forced around the sides of the cover and turned 125 degrees, which change of

direction is identical to Area Number 1. The equivalent duct length is therefore 53.0 feet as calculated in paragraph 13.4.3. A1 above.

13.4.4. Fan Requirements and Air Flow.

13.4.4.1. Condition 1 Maximum Air Requirements:

A. Crew ventilation requirements of 20 cfm per man = $20 \times 31 = 620$ cfm

B. Engine aspiration at full rpm = 600 cfm

Total requirement = 620 + 600

= 1220 cfm

A figure of 700 cfm through the crew compartment was selected to provide the crew ventilating requirements of 620 cfm with 80 cfm safety factor.

The friction loss in inches of water can now be determined. (Unit losses obtained from standard ASHAE tables).

C. Area Number 1

$$\text{Volume} = \frac{700}{2} = 350 \text{ cfm}$$

Equivalent length 8-inch diameter duct = 104.59 feet

Friction loss from table = 0.22 inch per 100 feet.

$$\text{Friction loss} = \frac{0.22 \times 104.59}{100} = 0.23 \text{ inch WG}$$

D. Area Number 2

Area Number 2 = Area Number 1 = 0.23 inch WG

E. Area Number 5

Volume = 2 x 350 = 700 cfm

Equivalent length 8-inch diameter duct = 3.03 feet

Friction loss from table = 0.80 inch per 100 feet

$$\text{Friction loss} = \frac{0.80 \times 3.03}{100} = 0.02424 \text{ inch WG}$$

F. Area Number 3

Volume Area Number 3 = 1220-700 = 520 cfm

Equivalent length 8-inch diameter duct = 58.16 feet

Friction loss from table = 0.426 inch per 100 feet

$$\text{Friction loss} = \frac{0.426 \times 58.16}{100} = 0.248 \text{ inch WG}$$

Friction loss of Area Number 1 + Area Number 5 =

0.23 + 0.02424 = 0.25424 inch WG

Since the friction loss of Area Number 3 is less than the combined friction loss of Area Number 1 plus Area Number 5, the required volume of air passing through the crew compartment is assured by restricting the air flow through Area Number 3 by adjusting the throttling valve.

G. Area Number 6

Maximum volume to be exhausted by fan = volume 1B-volume engine aspiration = 1220-600 = 620 cfm

Equivalent length 8-inch diameter duct = 55.3 feet

Friction loss from table = 0.80 inch per 100 feet.

Friction loss = $\frac{0.62 \times 55.3}{100}$ = 0.342 inch WG

H. Area Number 7

(Exhaust Side of Fan)

Volume Area Number 7 = Volume Area Number 6 = 620 cfm

Equivalent length of 8-inch diameter duct = 53.0 feet

Friction loss from table = 0.62 inch per 100 feet

Friction loss = $\frac{0.62 \times 53.0}{100}$ = 0.328 inch WG

Total Friction Loss = Area Number 1 + Area Number 5 + Area Number 6 + Area Number 7 = 0.23 + 0.02424 + 0.331 + 0.328 = 0.91324 inch WG

Based on these losses, a fan is selected which is capable of exhausting a minimum of 620 cfm when installed in the vehicle.

I. The air in the engine compartment is changed at the rate of 1220 cfm due to fan operation and engine aspiration. The air changes per minute equal the air flow volume divided by the engine compartment net volume of 211.2 determined previously, $\frac{1220}{211.2}$ = 5.78 changes per minute maximum.

13.4.4.2. Condition Number 2 Maximum air flow obtained with the fan selected for this application

A. Area Number 1

Assume volume = 378 cfm

Equivalent length 8-inch diameter duct = 104.59 feet

Friction loss from table = 0.245 inch per 100 feet

$$\text{Friction loss} = \frac{0.245 \times 104.59}{100} = 0.256 \text{ inch WG}$$

B. Area Number 2

Area Number 2 = Area Number 1 = 0.256 inch WG

C. Area Number 5

Area Number 5 Volume = 2 x 378 = 756 cfm

Equivalent length 8-inch diameter duct = 3.03 feet

Friction loss from table = 0.905 inch per 100 feet

$$\text{Friction loss} = \frac{0.905 \times 3.03}{100} = 0.0274 \text{ inch WG}$$

D. Area Number 3

Friction loss of Area Number 3 = the combined friction loss of Area Number 5 and Area Number 1 = 0.256 + 0.0274 = 0.2834 inch WG

$$\text{Friction loss} = \frac{0.2834 \times 100}{58.16} = 0.488 \text{ inch WG}$$

With a friction loss of 0.488 inch WG per 100 feet for 8-inch diameter duct, the air flow is 544 cfm through the opening.

E. Area Number 6

Volume exhausted by fan = Volume Area Number 5 + Volume Area Number 3 - Volume engine aspiration = 756 + 544 - 600 = 700 cfm

Equivalent length 8-inch diameter duct = 55.3 feet

Friction loss from table = 0.777 inch per 100 feet

Friction loss = $\frac{0.777 \times 55.3}{100}$ = 0.428 inch WG

F. Area Number 7

Volume Area Number 7 = Volume Area Number 6 = 700 cfm

Equivalent length 8-inch diameter duct = 53.0 feet

Friction loss from table = 0.777 inch per 100 feet

Friction loss = $\frac{0.777 \times 53.0}{100}$ = 0.411 inch WG

Total static pressure (P_S) = Area Number 1 + Area Number 5 + Area Number 6 + Area Number 7 = 0.256 + 0.0274 + 0.428 + 0.411 = 1.1224 inches WG

13.4.4.3. Condition 3A.

A. Area Number 1

Assume volume = 612 cfm

Equivalent length 8-inch diameter duct = 104.59 feet

Friction loss from table = 0.61 inch per 100 feet

Friction loss $\frac{0.61 \times 104.59}{100}$ = 0.638 inch WG

B. Area Number 2

Area Number 2 = Area Number 1 = 0.638 inch WG

C. Area Number 5

Area Number 5 Volume = 2 x 612 = 1224 cfm

Equivalent length 8-inch diameter duct = 3.03 feet

Friction loss from table = 2.35 inches per 100 feet

Friction loss = $\frac{2.35 \times 3.03}{100}$ = 0.0712 inch WG

D. Area Number 6

Volume to be exhausted by fan = Volume Area Number 5 -

Volume engine aspiration = 1224-600 = 624 cfm

Equivalent length 8-inch diameter duct = 55.3 feet

Friction loss from table = 0.628 inch per 100 feet

Friction loss = $\frac{0.628 \times 55.3}{100}$ = 0.347 inch WG

E. Area Number 7

Volume Area Number 7 = Volume Area Number 6 = 624 cfm

Equivalent length 8-inch diameter duct = 53.0 feet

Friction loss from table = 0.628 inch per 100 feet

Friction loss = 0.628 x 53.0 feet = 0.333 inch WG

Total static pressure (P_S) = Area Number 1 + Area Number 5 + Area Number 6 + Area Number 7 = 0.638 + 0.0712 + 0.347 + 0.333 = 1.3892 inch WG

13.4.4.4. Condition 3B

A. Area Number 1

Assume Volume = 404 cfm

Equivalent length 8-inch diameter duct = 104.59 feet

Friction loss from table = 0.278 inch per 100 feet

$$\text{Friction loss} = \frac{0.278 \times 104.59}{100} = 0.291 \text{ inch WG}$$

B. Area Number 2

Area Number 2 = Area Number 1 = 0.291 inch WG

C. Area Number 5

Area Number 5 Volume = 2 x 404 = 808 cfm

Equivalent length 8-inch diameter duct = 3.03 feet

Friction loss from table = 1.04 inch per 100 feet

$$\text{Friction loss} = \frac{1.04 \times 3.03 \text{ ft}}{100} = 0.0316 \text{ inch WG}$$

D. Area Number 6

Volume to be exhausted by fan = Volume Area Number 5 -

Volume Engine = 808-122 = 686 cfm

Equivalent length 8-inch diameter duct = 55.3 feet

Friction loss from table = 0.758 inch per 100 feet

$$\text{Friction loss} = \frac{0.758 \times 55.3 \text{ ft}}{100} = 0.419 \text{ inch WG}$$

E. Area Number 7

Volume Area Number 7 = Volume Area Number 6 = 686 cfm

Equivalent length 8-inch diameter duct = 53.0 feet

Friction loss from table = 0.758 inch per 100 feet

Friction loss = $\frac{0.758 \times 53.0}{100}$ = 0.402 inch WG

Total static pressure (P_S) = Area Number 1 + Area Number 5 + Area Number 6 + Area Number 7 = 0.291 + 0.0316 + 0.419 + 0.402 = 1.144 inch WG

13.4.4.5. Condition 4A.

A. Area Number 3

Assume volume = 1140 cfm

Equivalent length 8-inch diameter duct = 58.16 feet

Friction loss from table = $\frac{2.0 \times 58.16}{100}$ = 1.161 inches WG

B. Area Number 6

Volume to be exhausted by fan = Volume Area Number 3 - Volume engine = 1140-600 = 540 cfm

Equivalent length 8-inch diameter duct = 55.3 feet

Friction loss from table = 0.47 inch per 100 feet

Friction loss = $\frac{0.47 \times 55.3}{100}$ = 0.260 inch WG

C. Area Number 7

Volume Area Number 7 = Volume Area Number 6 = 540 cfm

Equivalent length 8-inch diameter duct = 53.0 feet

Friction loss from table = 0.47 inch per feet

Friction loss = $\frac{0.47 \times 53.0}{100}$ = 0.248 inch WG

Total static pressure (P_S) = Area Number 3 + Area Number 6 +

Area Number 7 = 1.161 + 0.248 = 1.669 inch WG

13.4.4.6. Condition 4B.

A. Area Number 3

Assume volume = 775 cfm

Equivalent length 8-inch diameter duct = 58.16 feet

Friction loss from table = 0.93 inch per feet

Friction loss = $\frac{0.93 \times 58.16}{100}$ = 0.541 inch WG

B. Area Number 6

Volume to be exhausted by fan = Volume Area Number 3 -

Volume engine = 775-122 = 653 cfm

Equivalent length 8-inch diameter duct = 55.3 feet

Friction loss from table = 0.69 inch per feet

Friction loss = $\frac{0.69 \times 55.3}{100}$ = 0.381 inch WG

C. Area Number 7

Volume Area Number = Volume Area Number 6 = 653 cfm

Equivalent length 8-inch diameter duct = 53.0 feet

Friction loss from table = 0.69 inch per feet

Friction loss = $\frac{0.69 \times 53.0}{100}$ = 0.366 inch WG

Total static pressure (P_S) = Area Number 3 + Area Number 6 + Area Number 7 = 0.541 + 0.381 + 0.366 = 1.288 inches WG

13.4.4.7. Conditon 5.

A. Area Number 1

Assume volume = 350 cfm

Equivalent length 8-inch diameter duct = 104.59 feet

Friction loss from table = 0.21 inch per 100 feet

Friction loss = $\frac{0.21 \times 104.59}{100}$ = 0.22 inch WG

B. Area Number 2

Area Number 2 = Area Number 1 = 0.22 inch WG

C. Area Number 5

Volume = 2 x 350 = 700 cfm

Equivalent length 8-inch diameter duct = 3.03 feet

Friction loss from table = 0.78 inch per 100 feet

Friction loss = $\frac{0.78 \times 3.03}{100}$ = 0.0236 inch WG

D. Area Number 6

Volume = 700 cfm

Equivalent length 8-inch diameter duct = 55.3 feet

Friction loss from table = 0.78 inch per 100 feet

Friction loss = $\frac{0.78 \times 55.3}{100}$ = 0.428 inch WG

E. Area Number 7

Volume Area Number 7 = Volume Area Number 6 = 700 cfm

Equivalent length 8-inch diameter duct = 53.0 feet

Friction loss from table = 0.78 inch per 100 feet

Friction loss = $\frac{0.78 \times 53.0}{100}$ = .411 inch WG

Total static pressure (P_S) = Area Number 1 + Area Number 5 + Area Number 6 + Area Number 7 = 0.22 + 0.0236 + 0.428 + 0.411 = 1.0826 inches WG

13.4.4.8. Condition 6.

To determine the maximum amount of scavening air that can be exhausted by the fan when operated as a bilge scavenger blower, it is assumed that the scavenge air will be drawn into the vehicle through the aft inlet Area Number 3.

A. Area Number 3

Assume volume = 573 cfm

Equivalent length 8-inch diameter duct = 58.16 feet

Friction loss from table = 0.53 inch per 100 feet

Friction loss = $\frac{0.53 \times 58.16}{100}$ = 0.308 inch WG

B. Volume Area Number 6 = Volume Area Number 3 = 573 cfm

NOTE: When operating as a bilge scavenging blower the diverter valve is operated. Therefore, the intake air to the fan is obtained through the 4-inch bilge scavenge pipe which is 4.67 feet long, plus the length of the diverter housing.

Friction loss from table for 8-inch diameter duct = 0.53 inch per 100 feet

1. Friction loss = 0.53 x 0.91 = 0.005 inch WG

2. Friction loss from table for 4-inch diameter duct = 18.6 inch per 100 feet

   $$\text{Friction loss} = \frac{18.6 \times 4.67}{100} = 0.868 \text{ inch WG}$$

3. Friction loss due to abrupt expansion

   $$\text{Equivalent length} = 22D = \frac{(22)\ (8)}{12} = 14.65 \text{ ft}$$

   $$\text{Friction loss} = \frac{(14.65) \cdot (0.53)}{100} = 0.777 \text{ inch WG}$$

   Total friction loss Area Number 6 - $B_1 + B_2 + B_3$

   = 0.005 + 0.868 + 0.777 = 1.65 inches WG

C. <u>Area Number 7</u>

Volume Area Number 7 = Volume Area Number 6 = 573 cfm

Equivalent length 8-inch diameter duct = 53.0 feet

Friction loss from table = 0.53 inch per 100 feet

Friction loss = $\frac{0.53 \times 53}{100}$ = 0.286 inch WG

Total static pressure (P_S) = Area Number 3 + Area Number 6 + Area Number 7 = 0.308 + 1.65 + 0.286 = 2.244 inches WG

13.5. Conclusions.

The air flows obtained in the vehicle under the foregoing conditions of operation, are summarized in the following chart:

Conditions	Areas							
	1	2	3	4	5	6	7	Engine Aspiration
Condition 1	350	350	520	700	700	620	620	600
Condition 2	378	378	544	756	756	700	700	600
Condition 3A	612	612	0	1224	1224	624	624	600
Condition 3B	404	404	0	808	808	686	686	122
Condition 4A	0	0	1140	0	0	540	540	600
Condition 4B	0	0	775	0	0	653	653	122
Condition 5	350	350	0	700	700	700	700	0
Condition 6	0	0	573	0	0	573	573	0

It may be seen that the air flows induced by the combination of the fan and engine aspiration will satisfy the requirements of the vehicle.

Figure 13-2 illustrates the system equivalent orifice lines applied to the output curve of the selected fan.

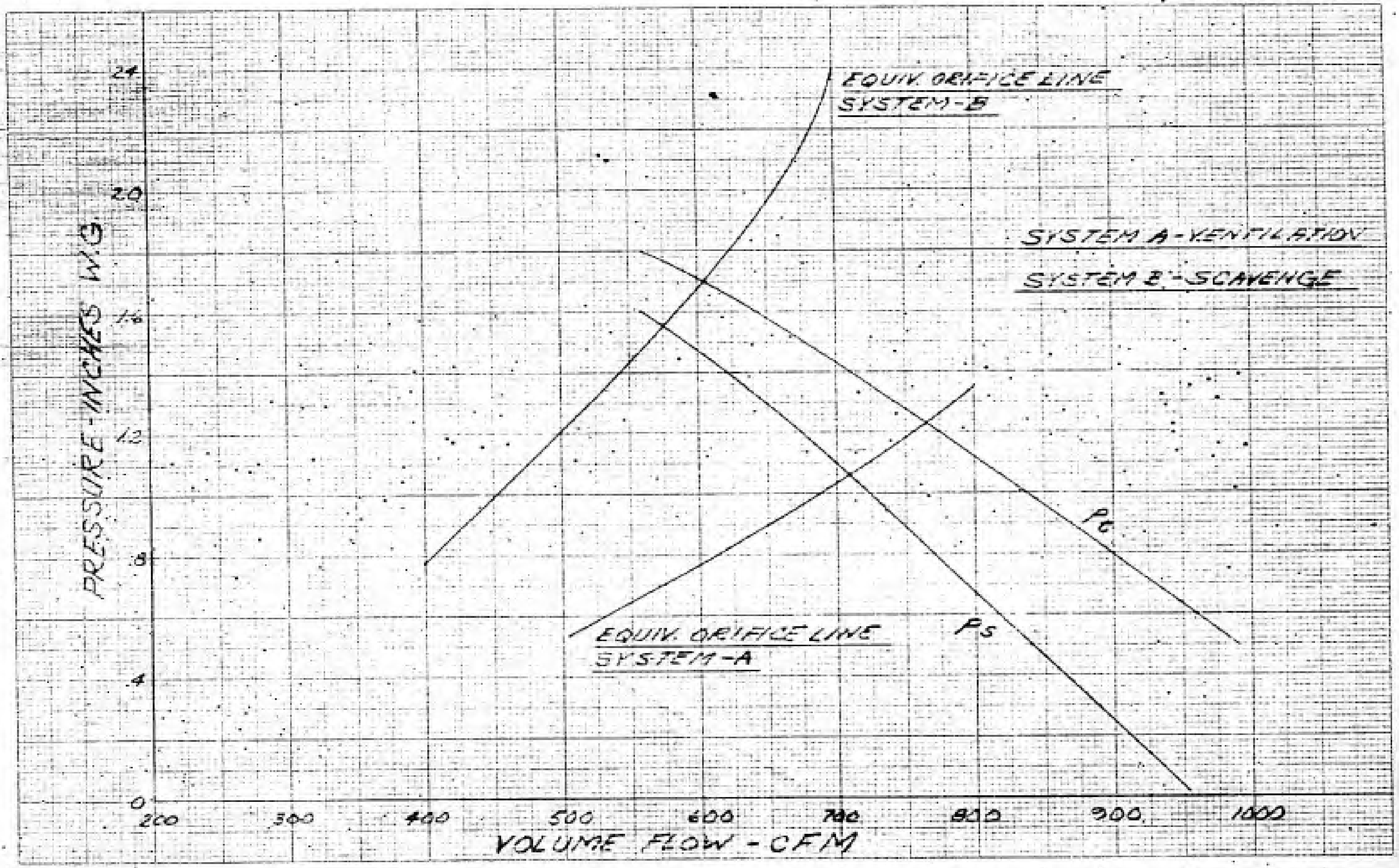
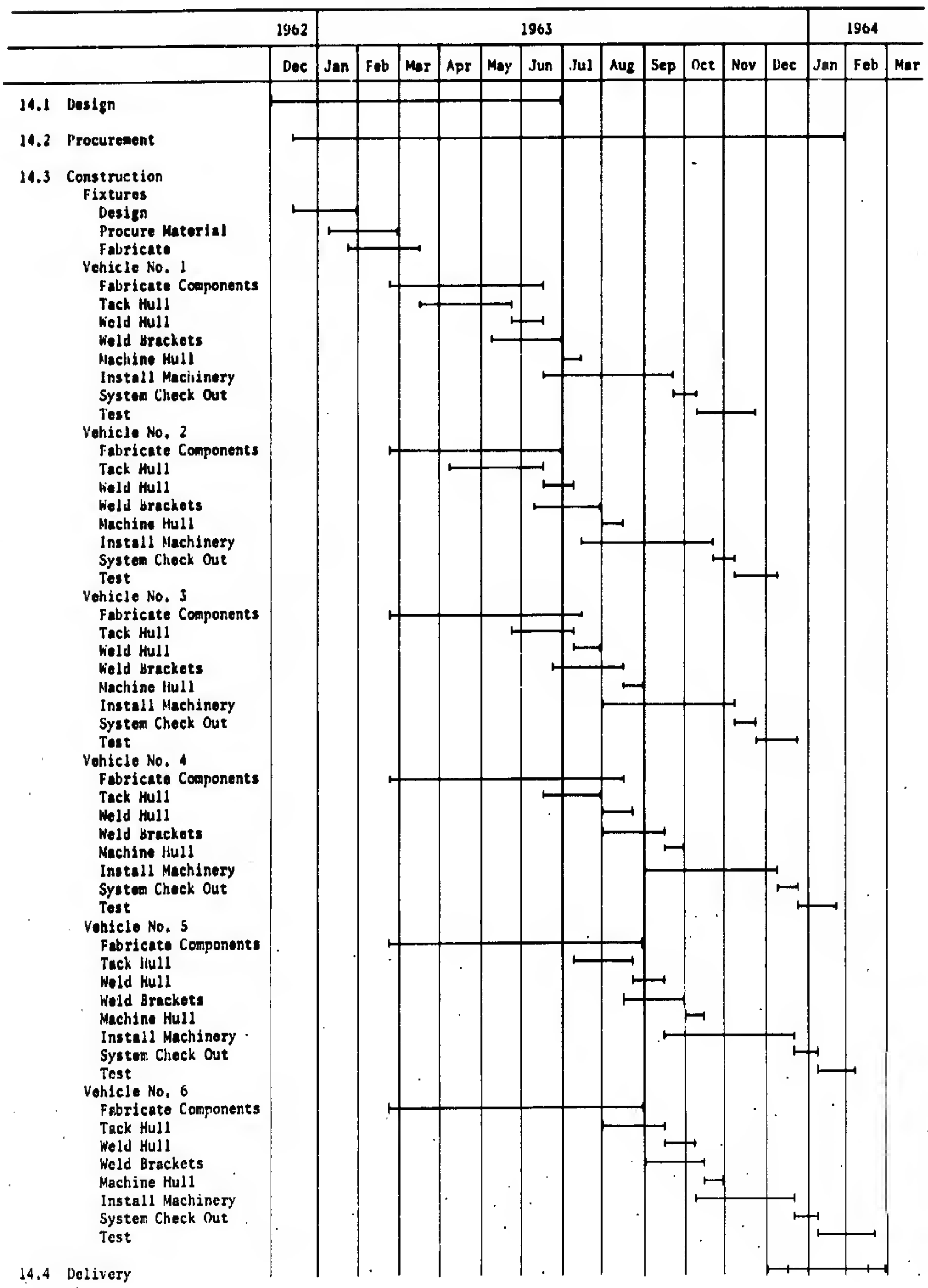


FIGURE. 13-2. FAN PERFORMANCE CURVE

14.0. ESTIMATED TIME SCHEDULE

TIME SCHEDULE
DESIGN - CONSTRUCTION - TEST



15.0. ESTIMATED COST B
NObs 4561 Item 2
LVTPX11

	Total Hours	Total Amount	Desig[n] Hours
DIRECT LABOR			
Engineering			
Project Engineers @ $3.95/hr.	11,824	$ 46,705	9,840
Draftsmen @ $3.06/hr.	38,605	118,131	37,799
Technical Publications @ $2.89/hr.	228	659	46
Engineering Records @ $2.44/hr.	788	1,922	788
Total Engineering Direct Labor	51,445	$ 167,417	48,473
Fabrication			
Prototype Construction @ $2.82/hr.	37,234	105,000	
Total Direct Labor	88,679	$ 272,417	48,473
BURDEN			
Engineering @ 82% of Direct Labor		$ 137,282	
Prototype Construction @ 130% of Direct Labor		136,500	
Blue Prints @ $.037 per Sq. Ft.	460,000(A)	17,019	460,000(A)
Testing (Offeror's Activity) @ $6.80/hr.	956	6,501	
Total Burden		$ 297,302	
DIRECT MATERIAL		983,492	
SPECIAL TOOLING		9,000	
TRAVEL AND SUBSISTENCE		6,500	
OTHER DIRECT COSTS (plant rearrangement)		2,000	
Sub-Total		$1,570,711	
GENERAL AND ADMINISTRATIVE		78,535	
Sub-Total		$1,649,246	
TARGET FEE @ 7%		115,447	
TOTAL	(B)	$1,764,695	

(A) Square feet of Blue Prints.
(B) Subject to negotiation downward since transmissions for three (3) vehicles are to be furnis

15.0. ESTIMATED COST BREAKDOWN
NObs 4561 Item 2
LVTPX11

	Total		Design		Construction		Acceptance Test	
	Hours	Amount	Hours	Amount	Hours	Amount	Hours	Amount
.	11,824	$ 46,705	9,840	$ 38,868	1,765	$ 6,972	219	$ 865
9/hr.	38,605	118,131	37,799	115,665	806	2,466		-0-
hr.	228	659	46	133	46	133	136	393
Labor	788	1,922	788	1,922		-0-		-0-
	51,445	$ 167,417	48,473	$ 156,588	2,617	$ 9,571	355	$ 1,258
2/hr.	37,234	105,000		-0-	37,234	105,000		-0-
	88,679	$ 272,417	48,473	$ 156,588	39,851	$ 114,571	355	$ 1,258
r		$ 137,282		$ 128,402		7,848		$ 1,032
Direct Labor		136,500		-0-		136,500		-0-
	460,000(A)	17,019	460,000(A)	17,019		-0-		-0-
80/hr.	956	6,501		-0-		-0-	956	6,501
		$ 297,302		$ 145,421		$ 144,348		$ 7,533
		983,492		-0-		983,492		-0-
		9,000		-0-		9,000		-0-
		6,500		4,000		2,000		500
ngement)		2,000		-0-		2,000		-0-
		$1,570,711		$ 306,009		$1,255,411		$ 9,291
		78,535		15,300		62,771		464
		$1,649,246		$ 321,309		$1,318,182		$ 9,755
		115,447		22,491		92,273		683
	(B)	$1,764,695		$ 343,800		$1,410,455		$ 10,438

ince transmissions for three (3) vehicles are to be furnished by the Government

2