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THE DESIGN AND CONSTRUCTION OF A CIRCULAR TRACK PAVEMENT TESTER

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16. Abstract

A test apparatus conforming to the ASTM document entitled "TENTATIVE RECOMMENDED PRACTICE OF THE CIRCULAR TRACK METHOD FOR DETERMINING THE WEAR RESISTANCE OF PAVEMENT SURFACE MATERIALS" was designed, fabricated, and installed in the Arkansas Highway Department's Research Facility in Little Rock.

The apparatus employs a high torque, low speed, hydraulic vane motor as a direct rotating drive for the axle-wheel assemblies. Tire path radius change is provided by a reciprocating vertical hydraulic cylinder connected to a rack and pinion mechanism. The rate of path radius change relative to tire speed is controlled by an adjustable flow divider. A variable displacement axial piston pump, driven by an electric motor permits infinite adjustment of the rotational speed. The mean tire path diameter is fifteen feet, with a radius change capability of six inches.

Features of the apparatus include simplicity of design, high expected reliability, minimal maintenance, protection against overloads during start up and shutdown, and ease of control. A preliminary checkout of the apparatus indicates that it will function as intended.

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THE DESIGN AND CONSTRUCTION OF A CIRCULAR TRACK PAVEMENT TESTER

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SUMMARY

This final report describes the design and analyses of a test apparatus conforming to the ASTM document entitled "TENTATIVE RECOMMENDED PRACTICE OF THE CIRCULAR TRACK METHOD FOR DETERMINING THE WEAR RESISTANCE OF PAVEMENT SURFACE MATERIALS." The body of this report is also presented in a M.S. thesis by Jack E. Helms, Jr. Copies of the thesis have been provided to the Arkansas State Highway Commission.

The apparatus, as described herein, has been fabricated and installed in the Arkansas Highway Department Research Facility in Little Rock, Arkansas. A preliminary checkout of the system indicates that operation is satisfactory.

If the expected performance and reliability are verified by further testing, there is a possibility that this design, or a similar one, will be adopted by other states.

TABLE OF CONTENTS

LIST OF FIGURES

Figure

CHAPTER I

INTRODUCTION

Rising road construction costs, coupled with growing public concern for safer highways, have acted to stimulate a nationwide search for safer, longer lasting pavement surfaces.

Many aggregates are used in pavement surface mixtures. A method for determining the most suitable mixture is desirable. The circular track method of wear resistance testing can be used to assign quantitative ratings to aggregates.

An ASTM recommended practice is being developed to standardize the circular track method of wear resistance testing. The standard method will allow for direct comparison of results between states. The ASTM practice provides the simulation of an average size automobile moving over the surface specimens.

Several state highway departments are currently using circular track pavement testers in their programs. However, none of the existing machines conforms to the ASTM recommended practice. The testers all have fixed tire path radii. A fixed radius limits the polishing action of the tire to a strip the width of the tire tread. The narrow path of wear tends to become a groove, edge deterioration follows, and as a result, highway driving is not simulated well. A test apparatus which continuously varies the tire path radius in a random manner will permit a much greater specimen area to be contacted and will more closely approximate actual highway conditions.

CHAPTER II

OBJECTIVE AND SCOPE OF STUDY

The objective of this project has been to develop a Circular Track Pavement Tester with a mechanism and associated controller for continuously and randomly varying the tire path radius.

The tester designed conforms to the ASTM document entitled "Tentative Recommended Practice of the Circular Track Method for Determining the Wear Resistance of Pavement Surface Materials" and incorporates the following functional subgroups:

1. Mechanism for varying the tire path radius.
2. Controller for the variable radius mechanism.
3. Rotating drive and center support structure.
4. Wheel suspension system with caster and camber adjustments.
5. Wheel weight maintenance method with provision for weight change.
6. Track support and drive foundation structure.

The scope of this design encompasses the synthesis process from initial specifications through concept development to the final, detail design and construction of the tester.

The ASTM recommended practice is included as Appendix B.

CHAPTER III

SYNTHESIS OF CIRCULAR TRACK PAVEMENT TESTER

A systems engineering approach was followed in the design of the pavement tester. This procedure has proven to be the most efficient method available for completion of complex designs.

A systems engineering approach is characterised by the following sequence of events: After careful definition, the problem is broken into its separate basic functions. These functions are then examined for solution possibilities and the most feasible solutions demonstrated by conceptual designs. A trade-off study is then performed on the concepts to see which best satisfies the objectives laid down in the problem definition. The concept and trade-off activities are coordinated to assure acceptable interfacing of the various functions. After a concept is chosen as optimum, it is then developed into a detailed design which can be implemented with off-the-shelf or manufactured hardware.

The pavement tester was considered to have two major functions:

1. Rotary drive
2. Variable radius mechanism

A discussion of the subsequent design effort follows.

Rotary Drive

Mechanical

Since existing circular track wear testers all use mechanical drives for rotation, this method was considered first. However, all systems considered were complex and would require high maintenance effort. Figure 1 shows the schematic of one mechanical drive system

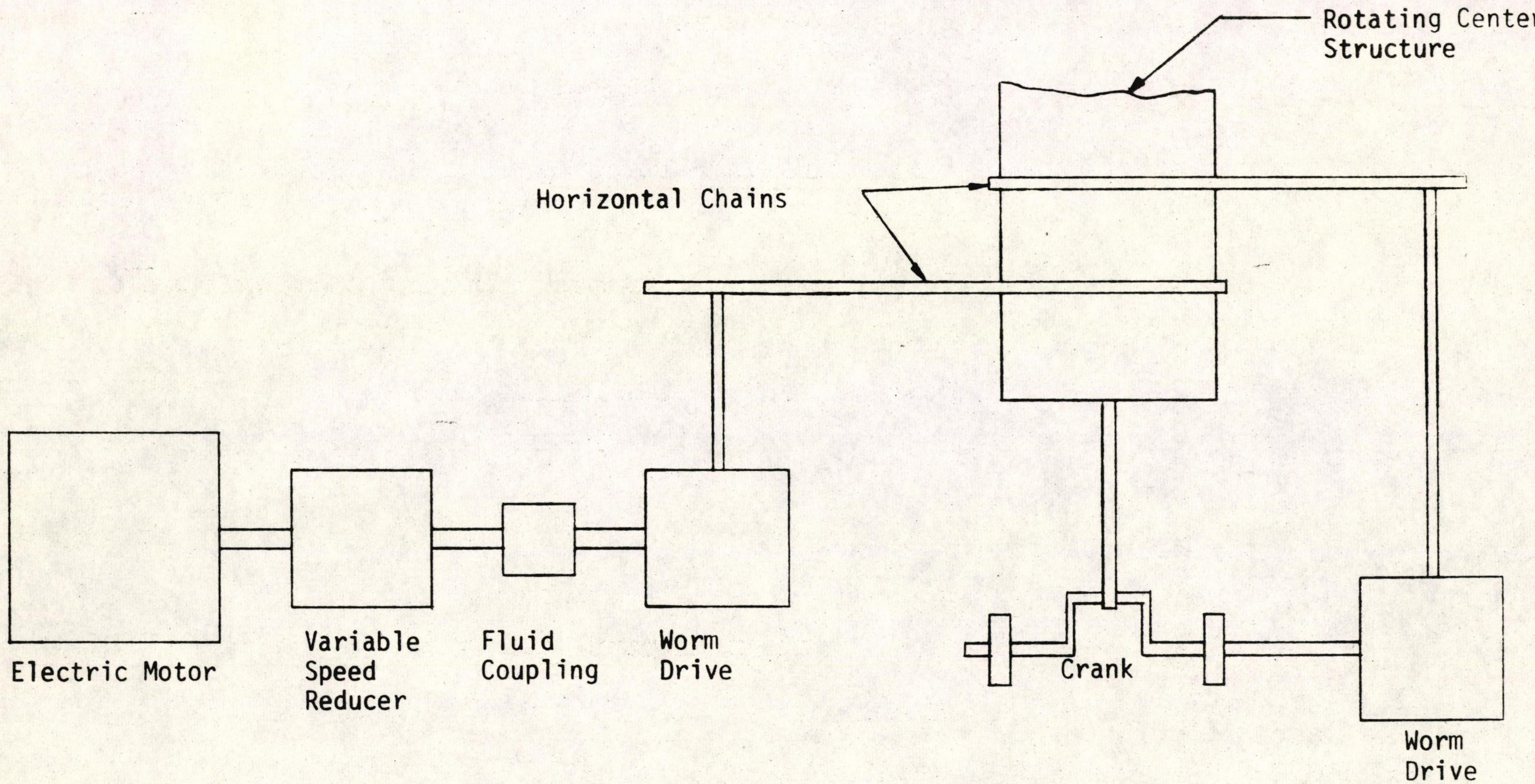


Figure 1. Mechanical Drive Schematic

that was developed to determine its feasibility. Electric motors normally run at higher speeds than the tester (21 rpm maximum). A variable speed reducer is coupled to the electric motor to reduce speed, increase torque, and to permit adjustment in rotational speed. The speed reducer output shaft is coupled to a right angle worm drive through a fluid coupling. Stalling of the electric motor during start-up is prevented by the fluid coupling, which also prevents transmission of shock loads and torsional vibrations back to the speed reducer and electric motor. A horizontal chain, powered by the worm drive, rotates the center column. Another horizontal chain is driven by the center column and powers a second right angle worm drive, which in turn rotates a crank about a horizontal axis. The vertical reciprocating motion of the crank finally drives the variable radius mechanism.

Several problems, in addition to the complexity of configuration are associated with the mechanical system. Alignment of mechanical components is critical, as misalignment can result in vibration, binding and seizing. The high torque, low speed rotary output requires large, expensive reduction units. Since horizontally driven chains are not easily oiled automatically, frequent manual lubrication is required.

Hydraulic

Because of the problems associated with mechanical systems, a decision was made to investigate a high torque, low speed hydraulic drive. Numerous advantages over mechanical systems became quickly obvious, including:

1. Reduced maintenance
2. Simplicity
3. Ease of control
4. Reduced component cost.

Maintenance is reduced because hydraulic fluid acts as a lubricant and increases component life. Internally generated heat is carried away by the flowing fluid, allowing also for the components to be of smaller size.

Simplicity of configuration is possible because components need only be connected by tubing. The noisy components, such as pumps and motors, can be acoustically isolated from the work area.

Control is facilitated by the characteristics of hydraulic components. Actuators have a high speed of response with fast starts, stops, and speed reversals possible. Torque to inertia ratios are large, with resulting high acceleration capability. Hydraulic actuators may be operated under continuous, intermittent, reversing, and stalled conditions without damage.

After appraising the relative merits of the hydraulic and mechanical systems, a high torque, low speed, direct drive was chosen.

Variable Radius Mechanism

The tire path radius must vary in a manner that results in a random coverage of each specimen to insure that a pattern of wear or rutting will not develop. The completed tester should also be aesthetically pleasing.

Two basic types of mechanisms, each with several variations were initially studied. These were the telescoping and pinned linkages. The best of each of these types was then developed into advanced conceptual designs.

Telescoping

The telescoping mechanism incorporates two arms sliding relative to a rotating center member. The telescoping arrangement requires a linkage to synchronize the opposing sliding motions of the rotating arms. Devices considered are listed below, followed by a brief discussion of each.

1. Toggle - external
2. Toggle - internal
3. Crank
4. Rack and Pinion
5. Screw
6. Chain and Sprocket

The internal and external toggle are essentially the same mechanism. The internal toggle facilitates a shorter and possibly stiffer center support structure. Either of the toggles requires a vertical reciprocating motion for actuation. Since the toggle is not a constant force mechanism, the resultant force transmitted is a function of the angle between the horizontal axis and a line along each member of the toggle.

The crank can be used to generate either a horizontal or vertical reciprocating motion.

The rack and pinion arrangement generates a horizontal reciprocating output motion from a vertical reciprocating input motion. Back-to-back racks drive counter rotating pinions. A second pinion, keyed to each shaft, imparts linear motion to the horizontal rack attached to the rotating arm.

Rotary motion about a horizontal axis is produced from a rotary input motion about a vertical axis by a right angle drive in the screw mechanism. The horizontally rotating screw imparts a linear motion to a drive-nut that is rigidly attached to the sliding member.

The chain and sprocket differ from the other variations, in that force is available only during the return stroke. A chain will not act as a compression member unless it is rigidly constrained. This arrangement would depend on the centrifugal force of the rotating members to provide outward motion.

A trade-off study established the double rack and pinion arrangement as the best telescoping mechanism. Its primary advantage is its ability to provide precise positioning and synchronization in a linear manner. Positive, precise positioning is necessary to minimize dynamic imbalance in the large rotating masses.

Pinned Linkage

The chief feature of a pinned linkage mechanism is that it eliminates sliding contact. However, precisely sized members are required to generate a horizontal reciprocating motion. Drive mechanisms considered were:

1. Crank
2. Rack and Pinion
3. Push-Pull

As in the telescoping mechanism the crank can be rotated either about the horizontal or vertical axis. When rotating about the vertical axis the crank acts as a direct drive. Force is transmitted directly along each arm of the tester. The crank acts indirectly when rotated about the horizontal axis. A vertical reciprocating motion is produced which must be changed to a horizontal motion of the linkage.

A single rack and pinion arrangement transduces a rotary input motion to a vertical reciprocating motion. A linkage is required to change the vertical motion to horizontal motion.

The push-pull mechanism can use the crank, rack and pinion or a hydraulic cylinder for its input motion. This mechanism is very similar to the external toggle described above.

The push-pull drive mechanism was chosen for advanced development. Elimination of sliding contact was the primary reason.

The two advanced concepts were carefully compared to determine which one better met the design requirements. The rack and pinion variation of the telescoping mechanism was chosen for the following reasons:

1. The displacement is linear throughout the radius change.
2. The linkage would require high precision in the location of pinned connections.
3. Considerable torque would have to be transmitted through the pinned joints.
4. The rack and pinion arrangement provides much better arm length adjustment.

Final Design

Figure 2 is a plan view of the circular track pavement tester installation. The track is enclosed within a heavy chain link fence for safety reasons. The power pack is isolated in a small acoustical enclosure. A control panel is shown next to the power pack enclosure; however, its position is arbitrary.

Section AA, Figure 3, is a frontal view of the tester, concrete supports and track.

Detail B, Figure 4, is a cutaway of the center support structure. The heavy walled tube (outer member) is stationary and houses the bearings on which the smaller center tube turns. The center tube

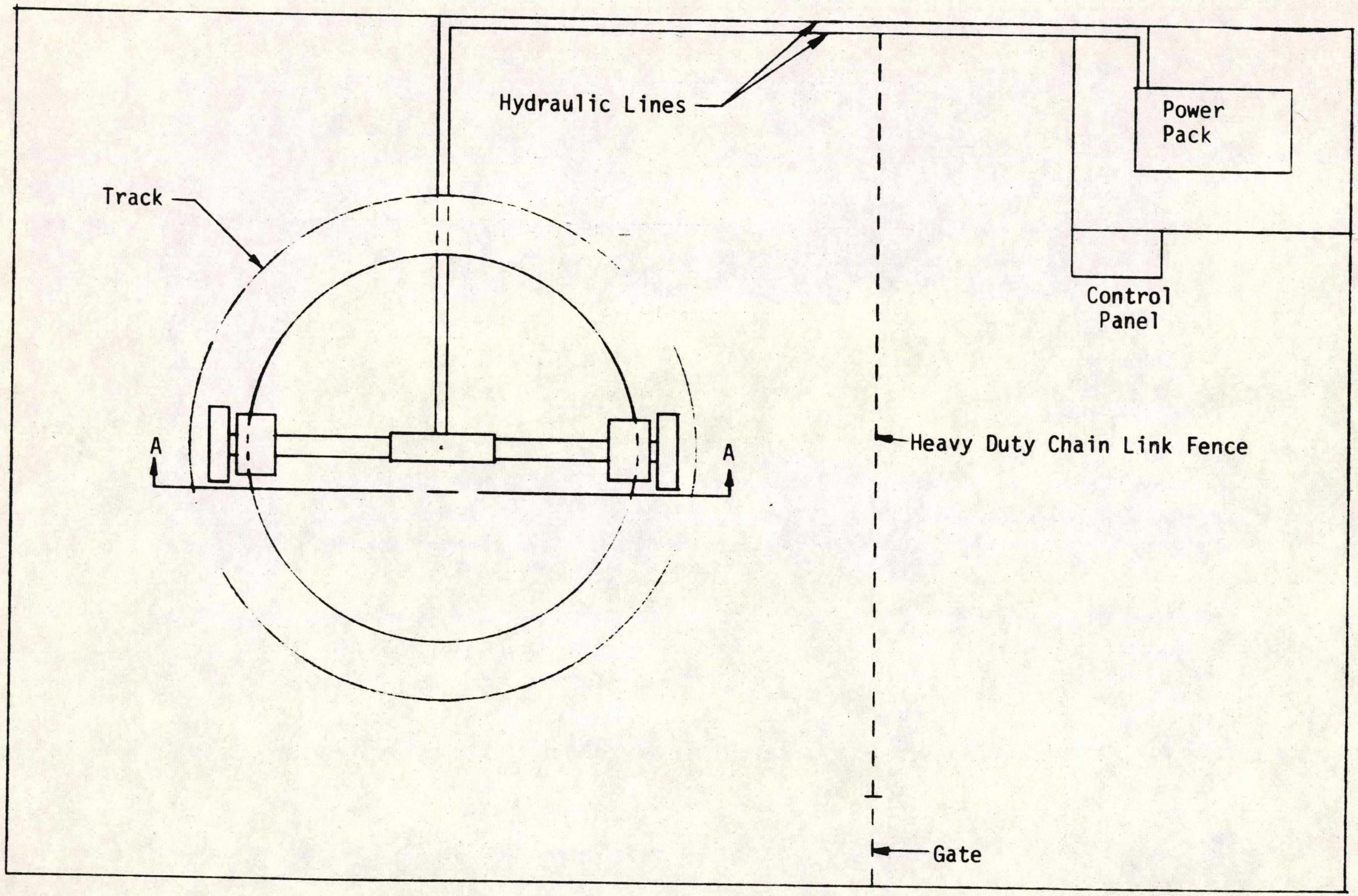


Figure 2. Plan View of Tester Installation

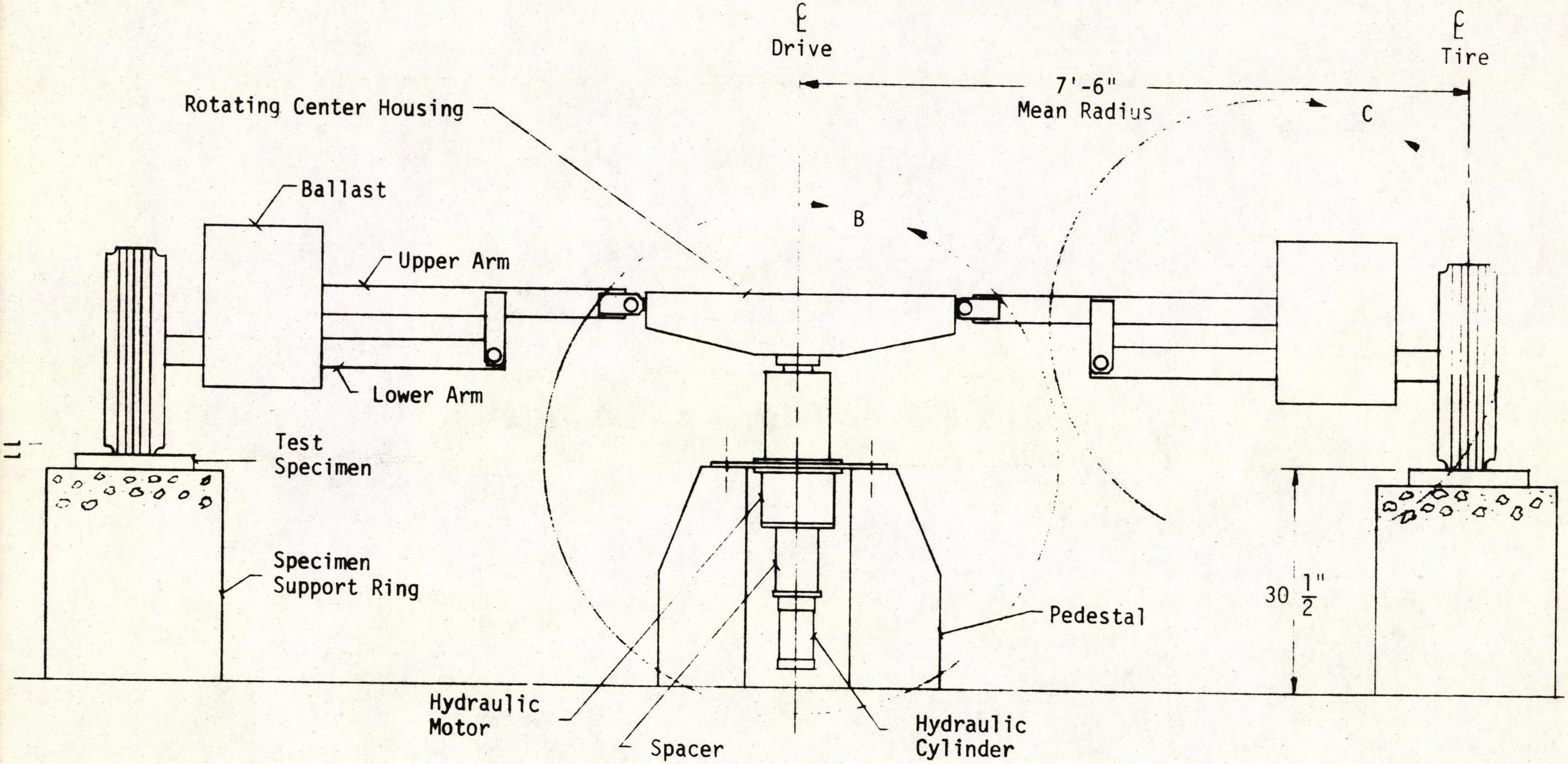


Figure 3. Section AA, Frontal View of Tester

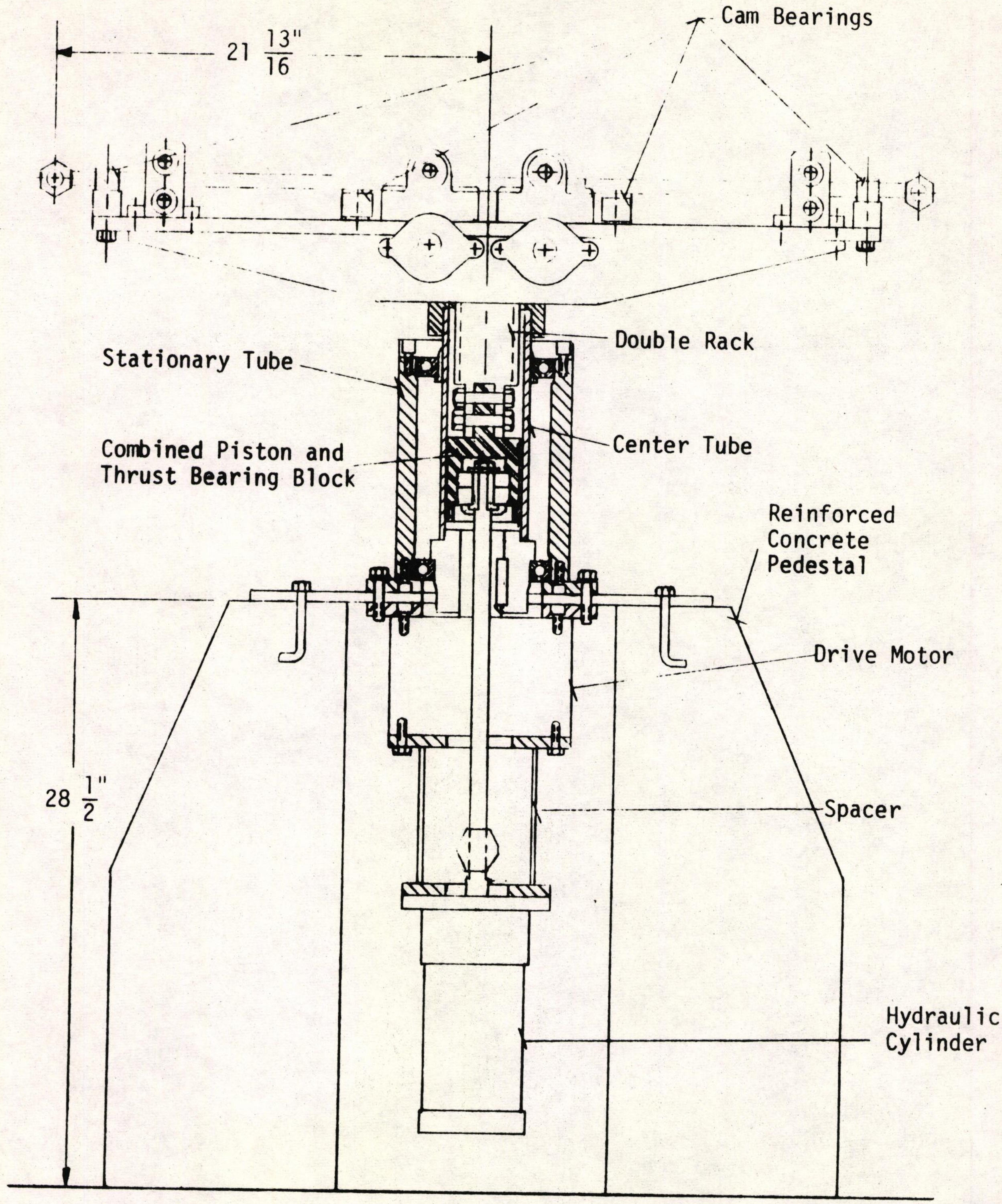


Figure 4a. Detail B, Center Support Structure

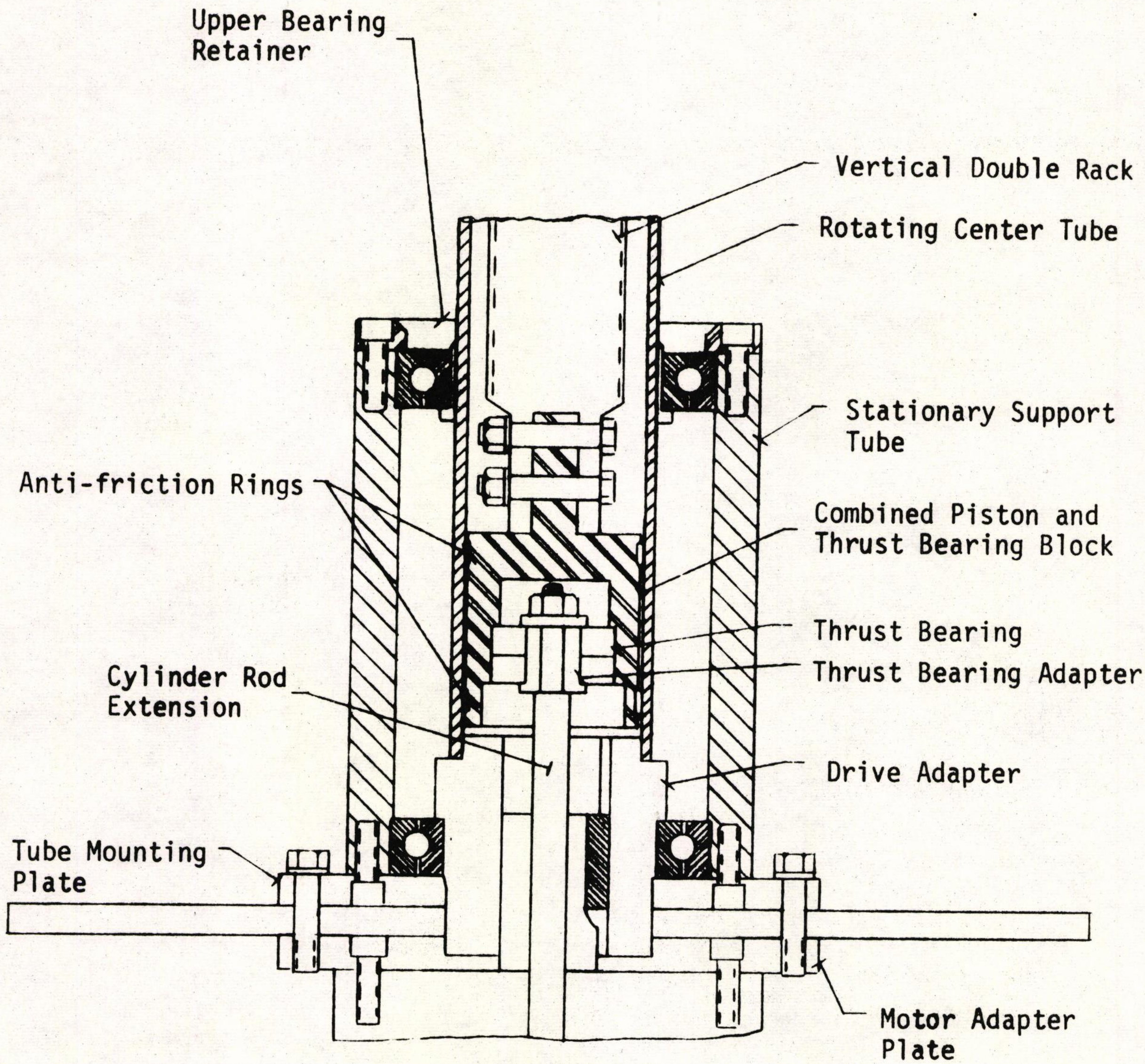


Figure 4b. Stationary Tube and Contents

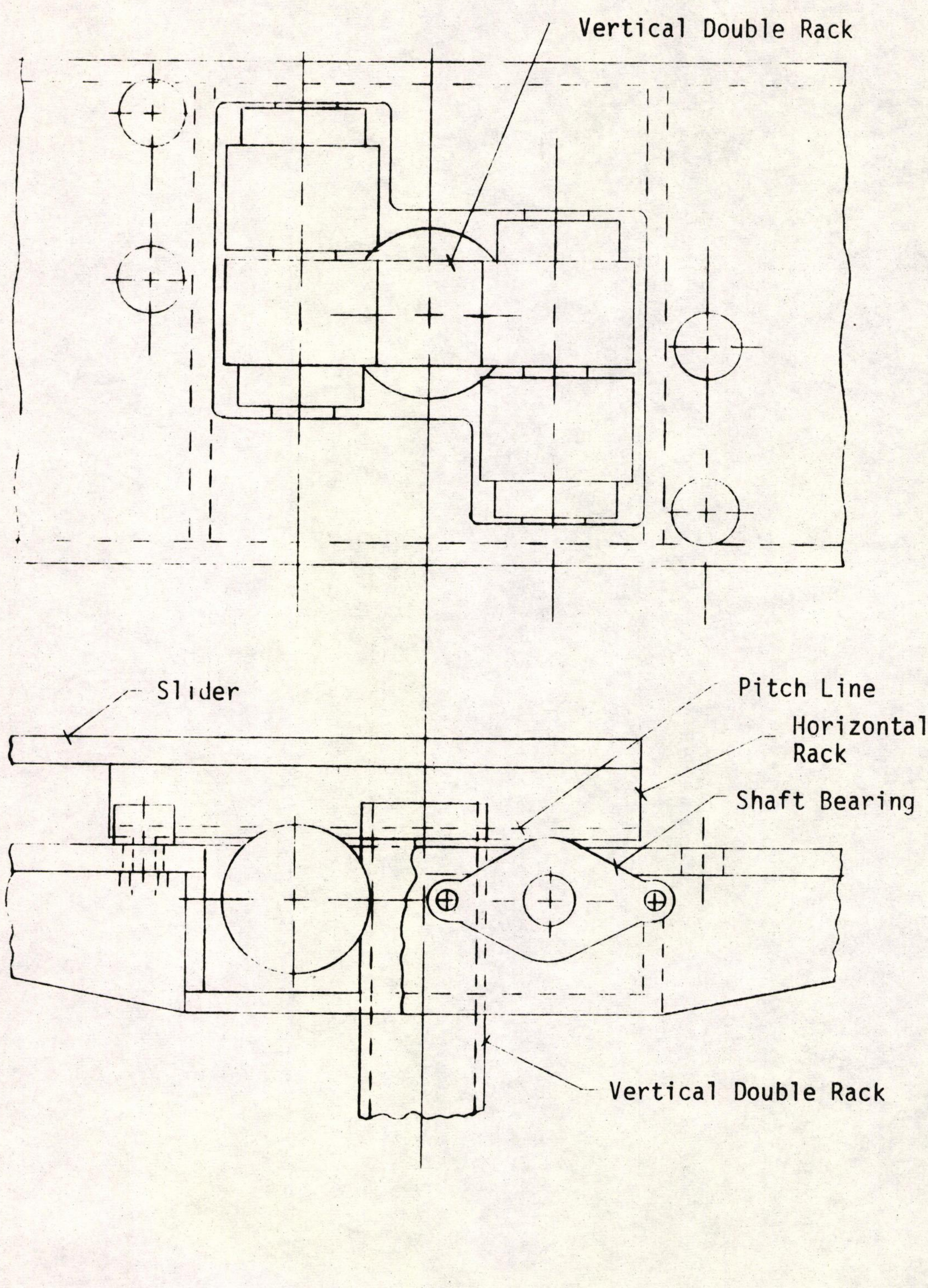


Figure 4c. Detail of Slider Drive

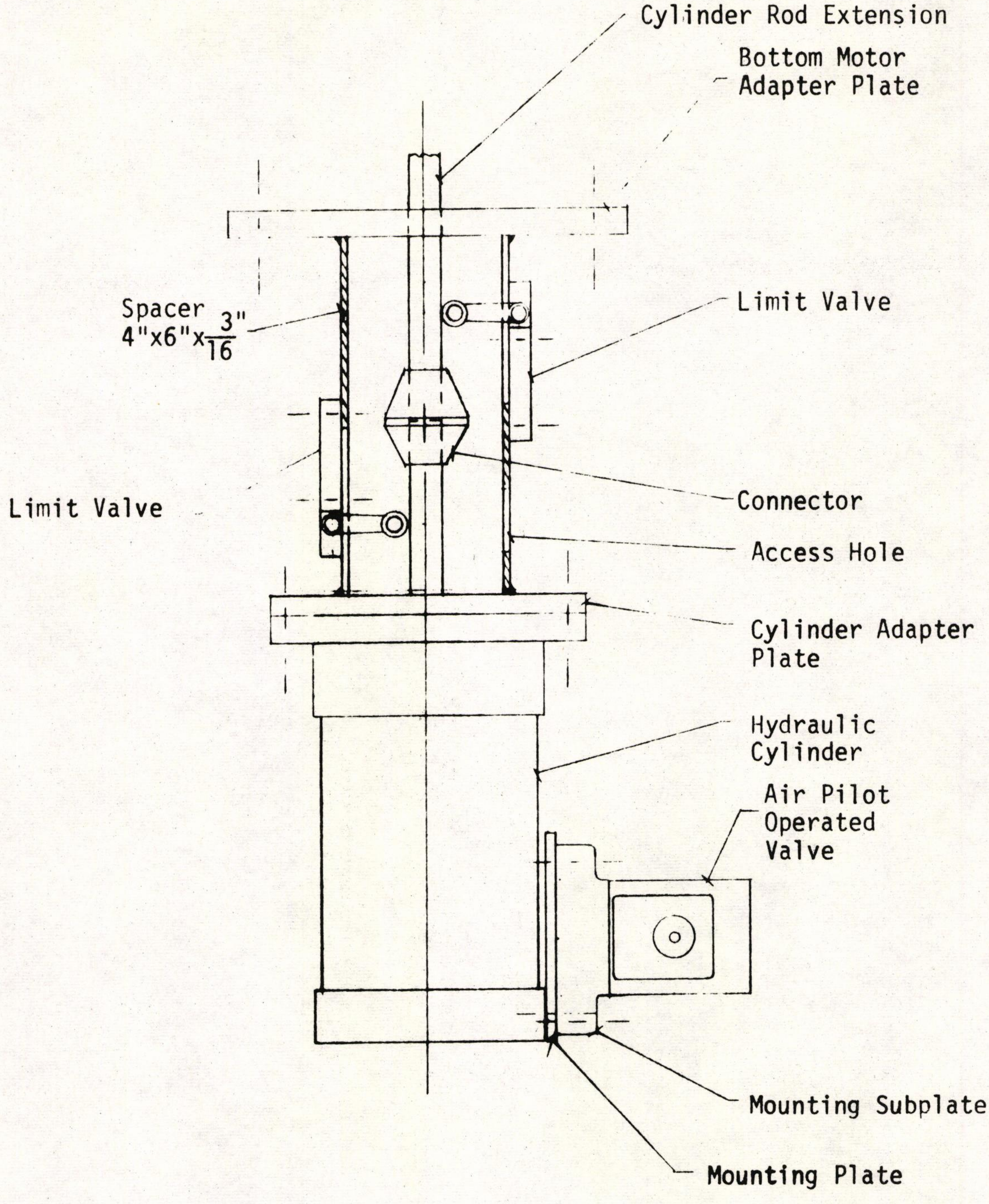


Figure 4d. Detail of Cylinder and Spacer

is a piece of honed hydraulic cylinder tubing connected to the shaft of the main motor and the rotating arm assembly. The cylinder rod extension passes through a hole bored in the drive motor shaft and connects to the vertical double rack through a combined piston and thrust bearing block. Anti-friction rings fit in grooves on the piston and contact the honed bore of the rotating tube. The vertical rack is reciprocated by the hydraulic cylinder and imparts counter rotary motion to two sets of pinion gears. Two pinion gears are keyed to a common horizontal shaft on each side of the centerline of the vertical rack. The inner pinions are driven by the vertical rack, while the outer pinions drive horizontal racks attached to the sliding members. This provides the motion necessary to vary the tire path radius. The hydraulic cylinder is attached to the bottom motor adaptor plate with a spacer which serves as a mounting for the two pneumatically operated switches that limit travel of the cylinder. The spacer also provides access to the grease fitting on the bottom face of the motor shaft. The four-way, two direction, pilot operated power valve is mounted on the lower end of the cylinder on a mounting plate.

Detail C, Figure 5, is a partial cutaway of one wheel assembly and rotating arm. The threaded hub shaft is mounted in a spherical bushing which is in turn captured in an adapter plate at the end of the lower arm. Four bolts, with self aligning pads at the ends, contact machined flats on the hub shaft. They hold the hub in place and provide a method for adjusting both caster and camber. The ballast box of steel shot provides the additional weight required to generate a wheel loading of 1085± 25 lbs. The weight can be adjusted by adding or removing shot. Partitions in the ballast compartments also permit

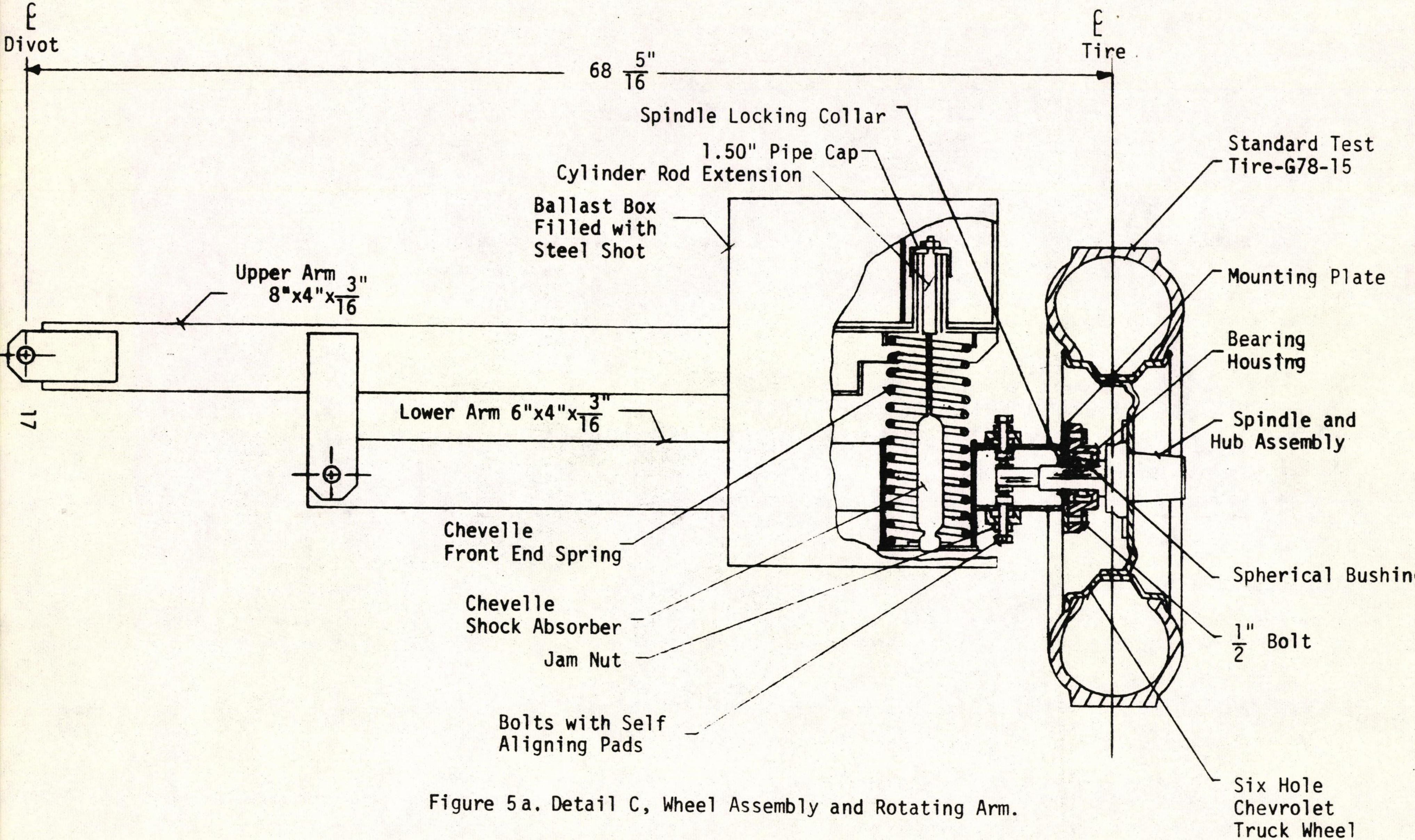


Figure 5a. Detail C, Wheel Assembly and Rotating Arm.

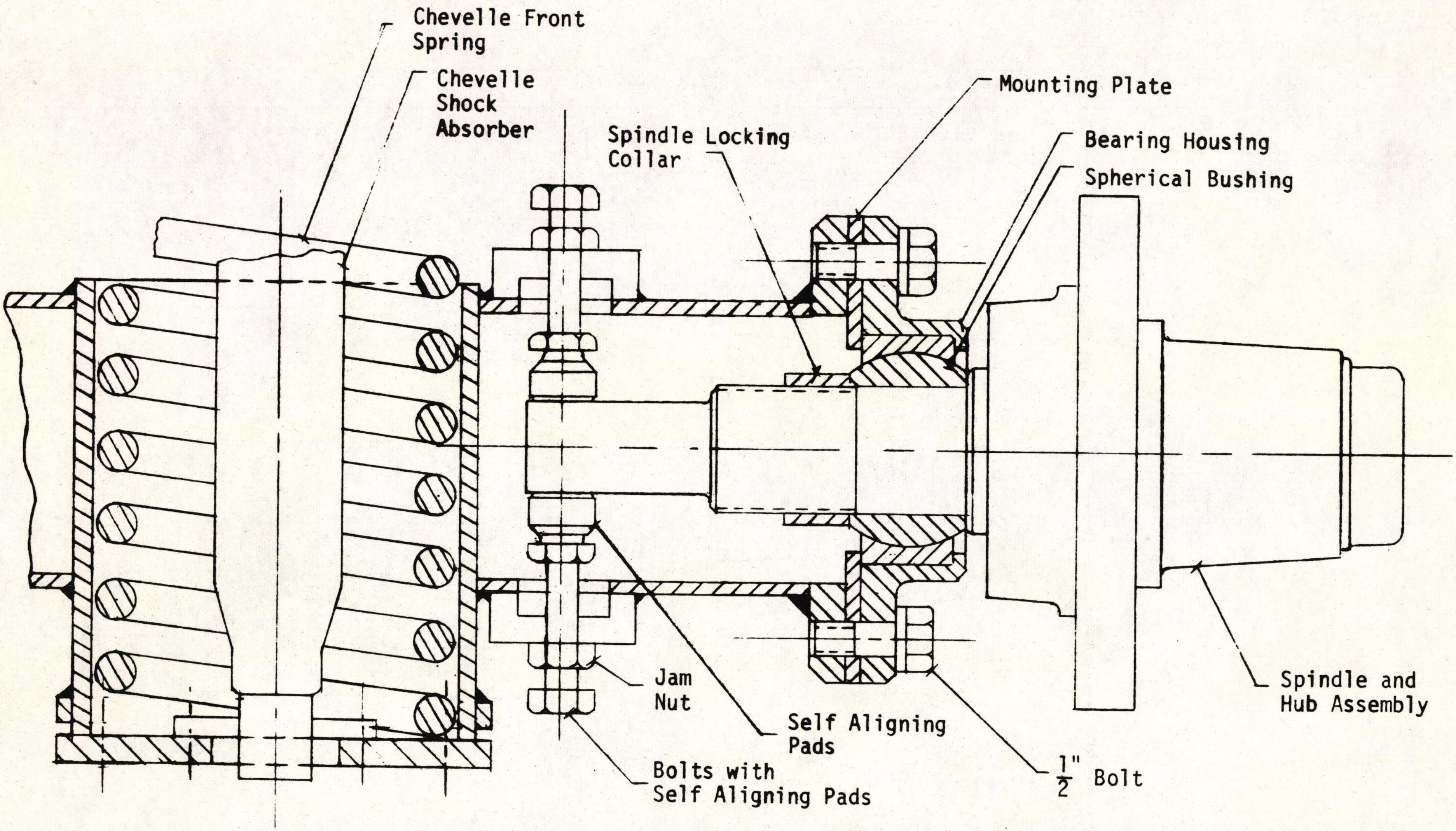


Figure 5b. Detail of Hub Mount

a radial weight shift as needed for dynamic balance. The spring and shock arrangement provide damping to eliminate bouncing and help to simulate the performance of an automobile.

Chapter IV discusses the design of the hydraulic drive and control systems.

CHAPTER IV

DESIGN OF HYDRAULIC CONTROL SYSTEM

While the rotational speed of the drive motor is selected simply by adjusting the displacement of the main pump, the control of the tire path radius variation mechanism requires a more sophisticated system. Not only must the radius change rate relate to the drive motor rotational speed, but the direction must reverse at the limits of travel.

Two basic types of control system were investigated. They were:

1. Electrohydraulic and mechanical servovalves
2. Flow metering and mechanism travel reversal

Servovalves

Hydraulic servo systems are capable of handling large inertia loads with high accuracy and rapid response (1). A sensor transduces the rotational displacement of the drive motor into a mechanical or electrical signal, which acts as an input to the servovalve. The flow through the valve causes the actuator to move and change the tire path radius. A feedback mechanism closes the valve as the actuator moves such that actuator displacement is proportional to angular rotation of the drive motor. At the limits of radius travel the valve shifts to reverse flow to the actuator. Both electrohydraulic and mechanical servovalves and related sensors are relatively expensive. Their cost and complexity led to the consideration of a less sophisticated system.

Flow Metering and Mechanism Travel Reversal

The characteristics of the hydraulic servo system described on the preceding page can be approximated by metering actuator flow proportional to the angular speed of the tester and by reversing flow at the actuator travel limits. Travel reversal can be easily attained by using a four-way, two-position, pilot-operated valve controlled by actuator limit valves. The methods of flow metering and the types of valves investigated are discussed below.

Flow Metering

A specific relationship between rotational speed of the tester and linear speed of radius variation was derived from the specifications. A rotational speed of 2.2 rad/sec (21 rpm) corresponds to a maximum of radius variation of 0.35 in./sec. This relationship requires metering flow to the hydraulic actuator of the variable radius mechanism in proportion to rotational speed.

Three methods for producing the required flow ratio for drive and control are listed and discussed as follows:

1. Flow divider
   a. Manufactured
   b. Two piston motors
2. Mechanically connected pump for variable radius mechanism
3. Use of drive motor exhaust

Commercially available flow dividers consist essentially of two (or more) gear motors coupled by a common shaft and enclosed in one housing. Flow is divided according to the relative displacements of the motors. The metering ratios available are fixed. Flow divider shaft speed is a function of the total amount of flow passing through the device. The resulting speed of the variable radius mechanism actuator will be in a fixed proportion to the drive motor speed for all possible speeds of the tester, except for small variations caused by leakage rates in the drive motor.

Two hydraulic piston motors, with coupled shafts, will act as a flow divider very similarly to the commercial type discussed above. An adjustable flow dividing ratio can be developed by coupling a fixed displacement motor to a variable displacement motor. Theoretically, ratios from one to infinity are possible if the two motors are of the same maximum flow capacity. The chief advantage of the variable displacement motor-fixed displacement motor combination is that it will allow the actuator speed to be adjusted as experience dictates and will permit periodic calibrations needed by changing drive motor leakage.

A pump driven off the shaft of the drive motor will develop a flow proportional to the speed of the drive and can be used to provide metered flow to the actuator of the variable radius mechanism. A variable displacement pump will provide a range of flow ratios. The commercially available pumps are not recommended for vertical mounting and require a chain and sprocket arrangement or a right angle drive.

The entire drive motor exhaust could be used to actuate an eight inch bore cylinder at the required speed of 0.35 in./sec. This would require the drive motor to operate against the fluctuating back pressure developed by the cylinder. While this method would be the least expensive, the fluctuating back pressure could cause unsteady speed of the drive motor. Also, any change in drive motor leakage would alter the speed ratio.

Of the preceding possibilities, the variable displacement motor-fixed displacement motor combination was considered to be the best method of flow division.

Mechanism Travel Reversal

The variable radius mechanism is required to complete a cycle of extension and retraction in ten or more revolutions of the tester. Since the speed of the actuator is limited by the flow divider adjustment, reversal of the actuator flow is the only requirement remaining.

The following mechanical valves were investigated to determine their suitability:

1. Cam operated valve
2. Clevis operated valve
3. Air pilot operated valve

Of the several reversing systems considered, a four-way, two direction, air pilot operated valve was determined to be most suitable. The valve mounts on the cylinder and is connected to the cam operated air limit switches by tubing. Size of the limit switches and provision for their adjustment determined the length of the structure between the actuator and motor.

Final Design

The integrated drive and control system consisted of three sub-groups:

1. Power pack
2. Drive system
3. Controller for variable radius mechanism

Figure 6 depicts the components of the power supply. The relationship of the power pack components to the remaining hydraulic components is shown by the fluid power diagram, Figure 7.

A pallet was designed to support the power pack components.

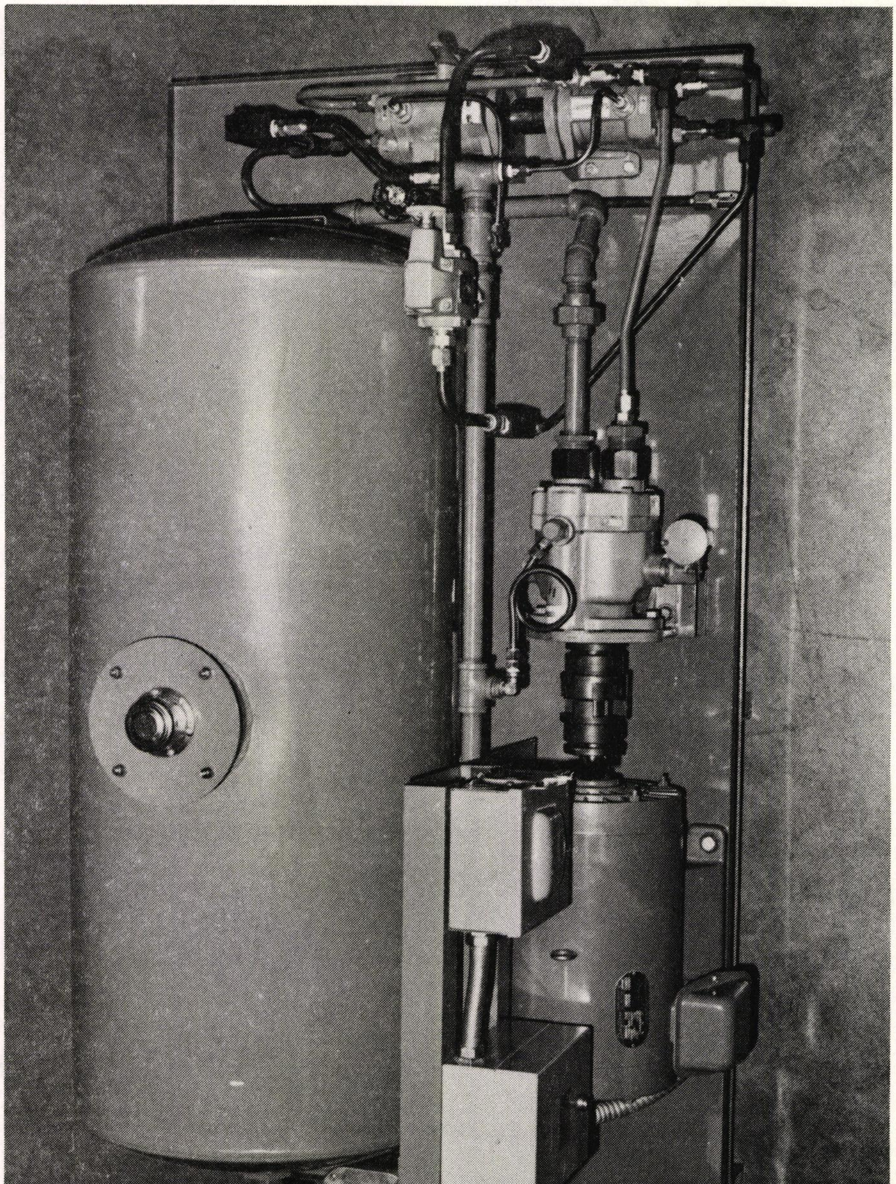


Figure 6. Palletized Power Supply

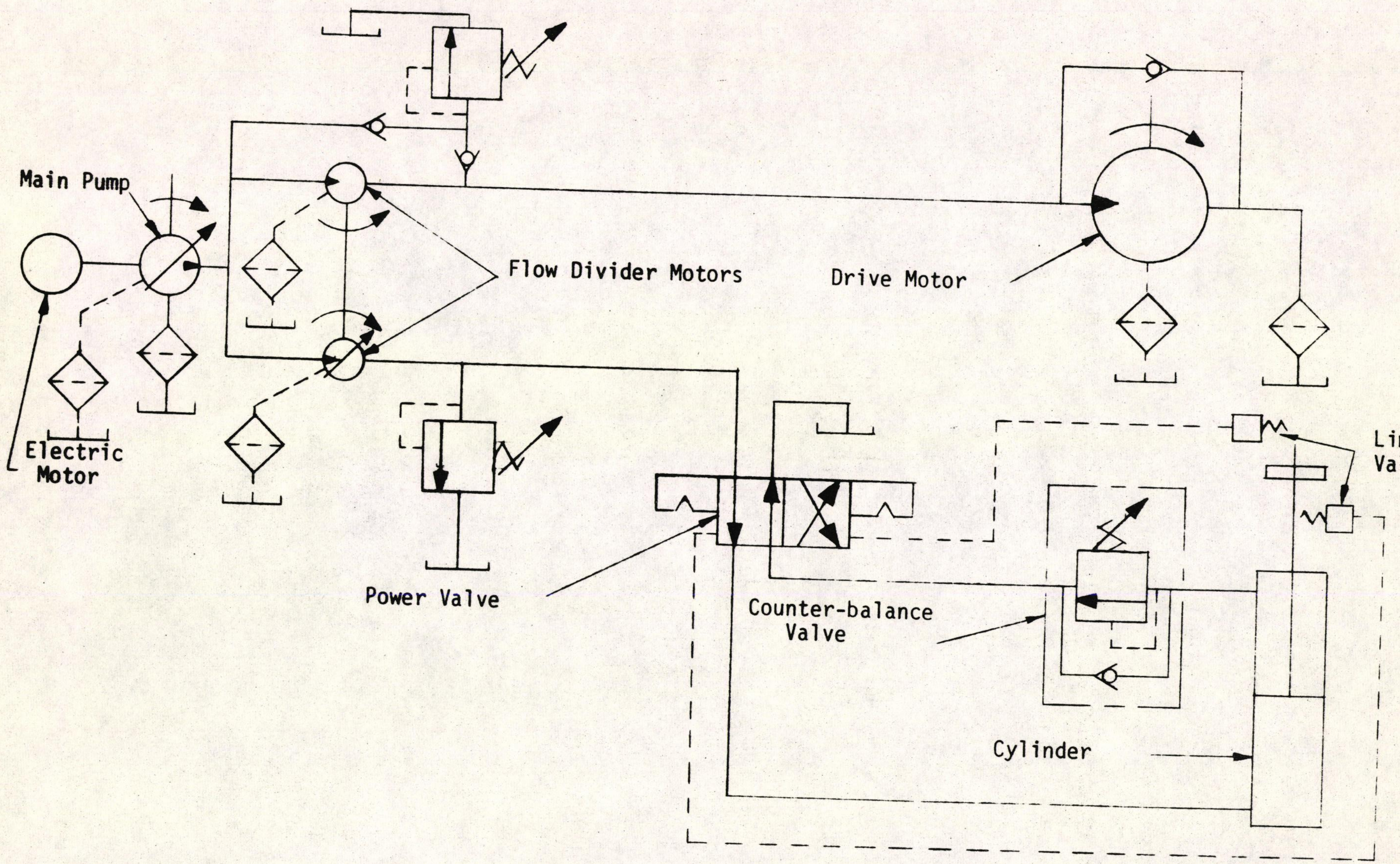


Figure 7. Fluid Power Diagram

Space was provided for fork lift tines by welding three extra channels, Figure 6, to the bottom of the pallet for ease in handling. The pump and motors, are mounted on the pallet which can be isolated for noise reduction and aesthetic reasons.

Dynamic analyses (see Appendix) indicated that a 7.5 hp electric motor would be sufficient as a primary power source. A 10 gpm variable displacement pump (Vickers PVB10) is coupled to the electric motor. Hand wheel position determines pump displacement which regulates the rotational speed of the drive motor and the linear speed of the cylinder.

Two 5 gpm inline piston motors, one constant and one variable displacement (Vickers MFB5 and MVB5 respectively), are coupled to form a flow divider. Adjustment of the variable displacement motor determines the ratio of drive flow to actuator flow and thus the relative speed. The two flow divider motors and main pump are protected by a 3000 psi relief valve.

A sixty gallon tank is used for the hydraulic reservoir. The large volume will help maintain a low oil temperature without use of a heat exchanger. Return line filtration of 10 microns and suction line straining by a 100 mesh strainer protect the hydraulic components against contamination. An oil level sight gauge and a filler-breather cap are also provided for safety reasons. A shut-off valve is provided in the suction line to allow removal of the line without draining the tank. A large, removable cover permits easy access to all areas of the interior.

Flow from the constant displacement flow divider motor is ported to the high torque, low speed hydraulic motor (Vickers MHT32) used for the rotary drive. The motor is protected by the same high pressure relief valve used to protect the flow divider motors through a tee and check valve arrangement, Figure 7. A check valve is also provided to port exhaust flow back into the motor in the event of cavitation at the motor inlet during shutdown or sudden loss of electrical power.

The variable radius mechanism is actuated by a 5 inch bore, 6 inch stroke, double acting hydraulic cylinder with a single 1 inch rod. Flow from the variable displacement flow divider motor is ported to an air pilot operated, four-way, two direction detent valve. The head end of the cylinder connects directly to the detent valve, while rod end flow passes through a counter balance valve before reaching the detent valve. The counter balance valve permits free flow into the cylinder but limits flow out of the cylinder to prevent runaway of the load on the extension stroke. (The rotating masses produce a large centrifugal force in the direction of extension travel.) The two air operated limit switches generate signals to shift the valve and reverse actuator flow. A relief valve prevents pressure from exceeding 170 psi in the actuator control circuit.

Schedule 40 pipe in the suction line facilitates the use of a gate valve. The remaining full size lines are 0.5 inch O.D. 3000 psi hydraulic tubing. The smaller drain lines are 0.25 inch O.D copper tubing.

CHAPTER V

SAFETY

A number of safety precautions are included to protect the system against potentially dangerous situations that could occur. Each of these is briefly discussed below.

The entire tester will be enclosed in a heavy duty chain link fence to prevent someone from being struck by the massive rotating members. The fence will also help to contain anything that might shake loose during operation. Large warning signs will be placed inside and outside of the fence.

Electrical power failure during operation of the tester would result in a stoppage of pump output and block the hydraulic supply and return. Inertia would cause the tester to continue rotation, which would temporarily convert the drive motor to a pump with a blocked inlet and outlet. A check valve is included in a bypass line to route exhaust flow back to the motor inlet to prevent cavitation. The check valve will operate when the pressure at the motor inlet becomes 5 psi below the exhaust pressure. The system will then gradually coast to a stop.

A high pressure (3000 psi) relief valve is provided to port pump output to tank in the event of seizure of the drive motor or one of the flow divider motors. A system of tees and check valves is used to facilitate the use of only one relief valve as shown in Figure 7.

The power valve, which reverses cylinder travel, is air operated. A failure of the air supply would prevent shifting of the valve when the cylinder reaches its limit of travel. Since the valve is of the detent type, the cylinder would continue to travel in the same direction until the piston bottomed out or the sliders hit their stops. The pressure would then build up quickly and flow would be ported to tank by the low pressure relief valve.

The low pressure relief valve will also port flow to tank in the event of seizure of the variable radius mechanism.

Stops are provided to prevent the rotating arms from coming off of the rotating center structure, should a failure occur in the rack and pinion drive.

Cam type idler-roller bearings are included on the slider top to prevent overturning of the rotating arms relative to the rotating center structure in the event of wheel seizure.

An oil level sight gauge is provided on the reservoir. It must be checked periodically to insure that a sufficient quantity of oil is available. An insufficient oil level would result in heating of the oil, cavitation and excessive wear.

The start-up controls are located outside of the safety fence for operator safety.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

This design effort has resulted in the development and construtuion of a circular track pavement tester with a mechanism and associated controller for continously and randomly varying the tire path radius.

The tester designed conforms to the mechanical portion of the ASTM recommended practice (Appendix B) concerning circular track wear testing. Instrumentation is required for the tester to meet all of the specifications of the recommended practice.

The integrated hydraulic drive and controller will require minimum maintenance and has a long life expectancy. Since the mechanical parts of the tester will move at slow speeds, only limited periodic maintenance will be required.

Recommendations

In the interest of safety, an oil level sensor could be included in the reservoir to switch off the electric motor in the event excessively low oil level.

A linear speed sensor on the cylinder rod and a rotational speed sensor on the rotating center structure would be helpful in calibrating the flow divider.

This pavement tester is the only one existing that meets the specifications of the ASTM recommended practice. It is therefore recommended that this machine be considered for adoption as an ASTM standard test apparatus after appropriate testing.

Limit valves should be used to sense excessive downward deflection of the rotating arms. The valves would operate cut-off switches to stop tester rotation in the event of a flat tire.

A vibration sensor and cut-off switch should be included to protect this tester in the event that a dynamic imbalance develops.

Potential alternate uses for the tester should be investigated. Tire wear vs. pavement surface type is suggested as one possibility.

BIBLIOGRAPHY

Referenced:

1. Merrit, Herbert E., Hydraulic Control Systems. John Wiley & Sons, Inc., 1967.

2. Shigley, Joseph E., Mechanical Engineering Design. McGraw-Hill Book Company, Inc., 1972.

3. Browning, Power Transmission Equipment, 1968.

4. Taborek, Jaroslav J., "Motion-Resisting Forces." Machine Design (July 25, 1957).

General:

5. Ernst, Walter, Oil Hydraulic Power and Its Applications. McGraw-Hill Book Company, Inc., 1949.

6. Paul-Monroe Hydraulics Inc., Fluid Power Designers Lightning Reference Handbook. (1967).

7. Parker-Hannifin, Fluid Power Design Engineers Handbook. (1973).

8. Machine Design, Fluid Power Reference Issue (1966).

9. Henke, Russ, Hydraulic Systems. The Penton Publishing Company. 1964.

10. Kauffman, Jack, "Hydraulic Systems Design." Machine Design (Sept. 4, 1969).

11. Keller, George R., Hydraulic System Analysis. Hydraulics and Pneumatics. 1969.

12. Brown and Sharpe Manufacturing Company, Formulas in Gearing (1933).

13. Phelan, Richard M., Dynamics of Machinery. McGraw-Hill Book Company, Inc., 1967.

14. Higdon, A., Ohlsen, E. H., Stiles, W. B., Weese, J. A. Mechanics of Materials. John Wiley & Sons, Inc., 1967.

15. Rothbart, Harold A., Ed., Mechanical Design and Systems Handbook. McGraw-Hill Book Company, Inc., 1964.

APPENDICES

APPENDIX A

CALCULATIONS

Load Analysis

A load analysis was conducted to determine the amount of ballast required to achieve a wheel loading of 1085$\pm$25 lb.

Lower Arm

A = 70 lb., Tire, wheel and spindle - C. G. at center line of tire.

B = 2 lb., Bushing.

C = 10.9 lb., Bushing mount.

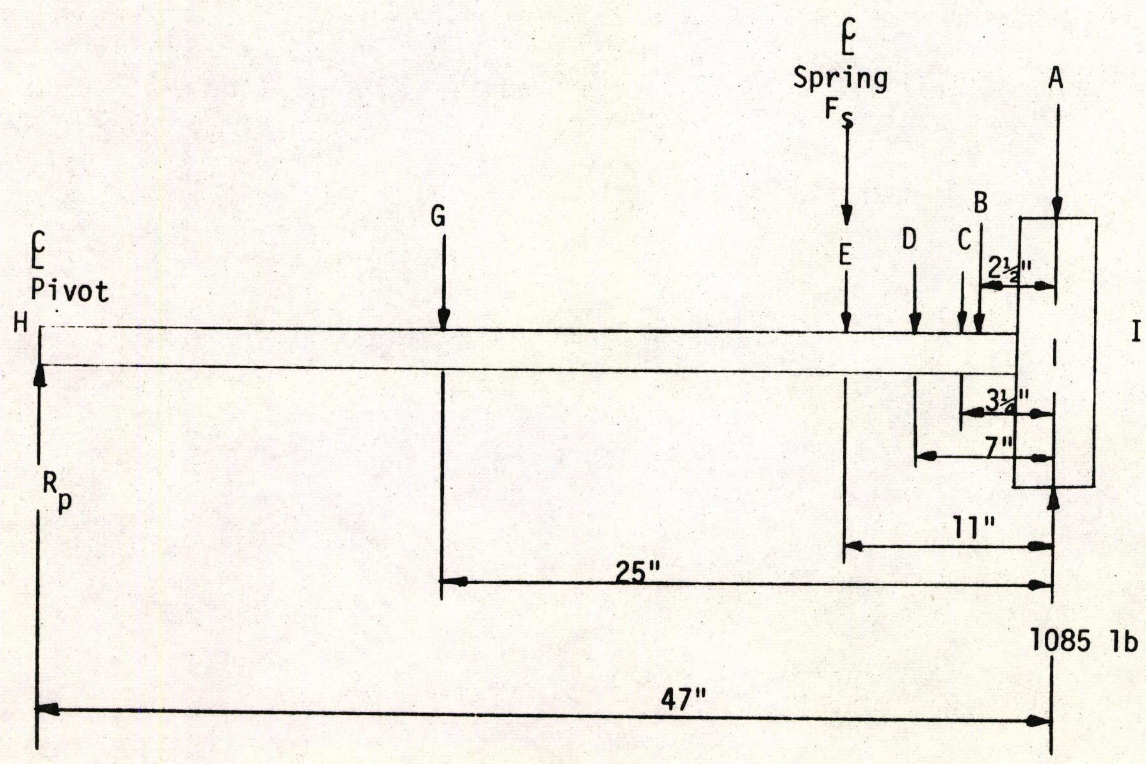
D = 5 lb., Camber adjust plate.

E = 24 lb., Spring, shock, tube, mounting plates.

F_s = Spring Force.

R_p = Pivot Reaction

G = 43.5 lb., 6 x 4 tube.



$\Sigma M_H = 0$ (+)

$$(43.5)(22) + (24)(36) + F_s(36) + 5(40) + (10.9)(43.75) + (2)(44.5) + (70)(47) - (1085)(47) = 0$$

$$F_s = 1253.3 \text{ lb} \downarrow$$

$\Sigma M_I = 0$ (+)

$$(2)(2.5) + (10.9)(3.25) + (5)(7) + (24)(11) + (1253.3)(11) + (43.5)(25) - R_p(47) = 0$$

$$R_p = 323.7 \text{ lb} \uparrow$$

Check $\Sigma Fy = 0$ (+↑)

$$323.7 - 43.5 - 1253.3 - 24 - 5 - 10.9 - 2 - 70 + 1085 = 0$$

$$\text{Spring deflection} = \frac{1253.3 \text{ lb.}}{313 \text{ lb./in.}} = 4.0 \text{ in.}$$

Upper Arm

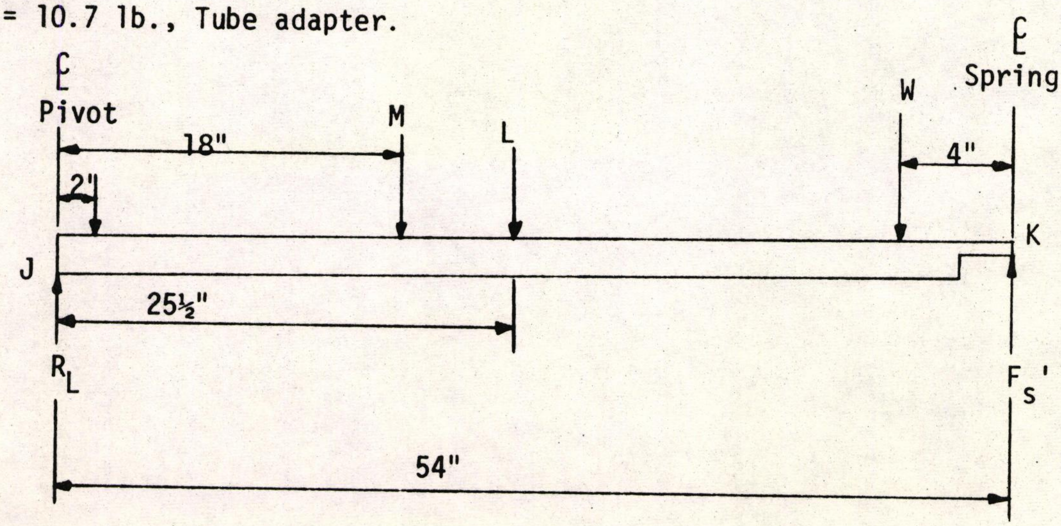
F_s' = 1253.3 lb., Spring reaction.

W = Ballast.

L = 58.8 lb., 8 x 4 tube.

M = 340.7 lb., Pivot force, mount.

N = 10.7 lb., Tube adapter.



$\Sigma M_J = 0$

$$(10.7)(2) + (340.70)(18) + (58.8)(25.5) + W(50) - (1253.3)(54) = 0$$

$$W = 1200.5 \text{ lb.} \downarrow$$

$\Sigma M_K = 0$

$$(1200.5)(4) + (58.8)(28.5) + (340.7)(36) + (10.7)(52) - R_L(54) = 0$$

$$R_L = 357.4 \text{ lb.} \uparrow$$

Check $\Sigma Fy = 0$

$$357.4 - 10.7 - 340.7 - 58.8 - 1200.5 + 1253.3 = 0$$

Ballast

End Plates	2(16 x 22 + 8 x 8) =	832
Bottom	(23 x 15.5) +(1.75 x 7)=	368.75
Sides	(15.5 x 22)=	341
Partition	4(20.5 x 7.5) + 2(8 x 6.5)=	719
Covers	15.5(15 + 8)=	356.5
	Total	2617.25 in^2

Angles 1/8 x 2 [93 + 22 + 12] = 31.75 in^3

Weld = 3 lb

Volume of 1/4" plate = $\frac{2617.25}{4}$ = 654.3 in^3

Weight of plate = 654.31 x .283 =	185 lb
Weight of angle = 31.75 x .283 =	9 lb
Weight of weld =	3 lb
Weight of container	197 lb

Volume of container

Sides 15.5 x 15 x 21.25 = 4940.6

Middle 8 x 15.5 x 7.25 = 899

Total = 5839.6 in^3

Weight of shot = 5839.6 x .1699 lb/in^3 = 992 lb

Volume in upper arm

$$3.62 \times 7.62 \times 8.0 = 220.7 \text{ in}^3$$

Weight of shot = 220.7 x .1699 = 37.5 lb

Total ballast available 1226.5 lb

Void Space in Shot

$$100 \left(1 - \frac{.1699}{.283}\right) = 40\%$$

Dynamic Analysis

From specifications, maximum wheel velocity:

$$V = 1.5 \ \frac{\text{mph}}{\text{ft}} \times R$$

R = 7.5 ft., tester radius.

Substitution into above equation yields:

$$V = 11.25 \text{ mph}$$

$$\omega = \frac{V}{R}$$

ω = circular frequency, rad/sec

Substitution yields:

$$\omega = 2.20 \text{ rad/sec}$$

Track circumference, C:

$$C = 2\pi R = 47.1 \text{ ft}$$

Distance traveled in 5 revolutions:

$$S = 5C = 235.6 \text{ ft} = Vt$$

S = distance traveled, ft.

t = time required, sec.

Solving for time yields

$$t = 14.28 \text{ sec.}$$

Actuator speed

$$v_a = \frac{5.0 \text{ in.}}{14.28 \text{ sec}} = 0.35 \text{ in/sec}$$

Shaking Force

$$F = \frac{\omega^2}{g} \sum_{i=1}^{10} W_i r_i$$

F = shaking force, lb

ω = 2.20 rad/sec, circular frequency.

g = 386 in/sec^2, acceleration due to gravity.

W_i = Weight of component pieces, lb

r_i = radial distance to center of gravity of component pieces, in.

Substituting values from upper and lower arm load calculations yields:

$$F = \frac{(2.20)^2}{386} [70 \times 90 + 2 \times 87.5 + 10.9 \times 86.75 + 5 \times 83 + 24 \times 79$$
$$+ 43.5 \times 65 + 1200.5 \times 75 + 58.8 \times 50.5 + 340.7 \times 43 + 10.7 \times 27]$$
$$= 1511.0 \text{ lb}$$

This is the maximum outward radial load for each arm.

Gear Strength Analysis

P = 5 in^{-1}, diametral pitch

ϕ = 20 , pressure angle

d = 3.5 in., pitch diameter

N = 18, number of teeth

F = 2.5 in., face width of gear

Lewis' equation must be modified due to:

1. Stress concentration at the tooth fillet
2. The effect of contact ratio on division of tooth load
3. The velocity, or dynamic effect, on tooth load

With these modifications, Lewis' equation becomes:

$$\sigma = \frac{W_t P}{K_v F J} \quad (2)$$

σ = modified bending stress

W_t = 1511 lb., transmitted load

K_v = 1.0, velocity or dynamic factor

J = 0.37, geometry factor

Substitution into this equation yields:

$$\sigma = \frac{1511.0 \times 5}{1.0 \times 2.5 \times 0.37}$$

$$= 8167.6 \text{ lb./in.}^2$$

From Gear Specifications:

Material stress for steel gear = 20,000 psi. This value is based on beam strength but not wear. Service factors must be used to correct material stress for type of loading and lubrication (3).

Service factor for no shock = 1.0

Service factor for intermittent lubrication = 0.7

$$\text{Corrected material stress} = \frac{20{,}000 \times 0.7}{1.0}$$

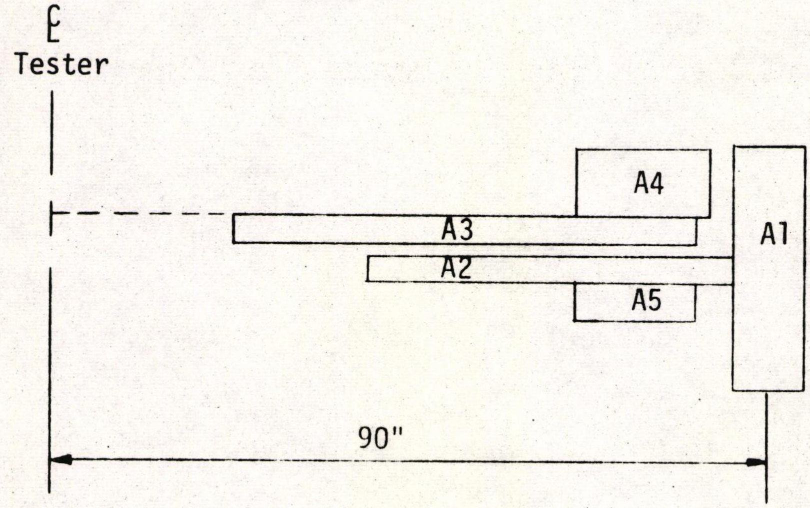
$$= 14{,}000 \text{ psi}$$

$$\text{Factor of safety} = \frac{\text{corrected material stress}}{\text{modified bending stress}} = \frac{14{,}000}{8167.6} = 1.71$$

Power Requirements

Wind Drag

The model shown below was used to determine frontal area of the rotating arms.



	Area (in.^{2})	Moment Arm (in.)	Moment (in.^{3})
A_1	240	90	21600
A_2	180	63.5	11430
A_3	232	52.5	12180
A_4	128	81.2	10388
A_5	60	74	4440
Total	840		60038

$$\bar{X} = \frac{60038}{840} = 71.5 \text{ in.}$$

Speed at centroid

$$V = 71.5 \text{ in. x } 2.20 \text{ rad/sec}$$

$$V = 157.24 \text{ in/sec}$$

$$= 13.10 \text{ ft/sec}$$

Drag force

$$D = 1.28 \frac{\rho}{2g_c} A V^2$$

D = drag force, lbf

ρ = 0.076 lbm/ft.^{3}, density of air.

A = 840 in.^{2}, total projected area.

V = 13.10 ft./sec, speed of centroid.

g_c = 32.17 lbm ft./lbf sec ^{2}, constant.

Substitution into the equation yields

$$D = 1.51 \text{ lbf (per arm)}$$

Total Drag

$$D_T = 2 \times D = 3.03 \text{ lbf (at centroid)}$$

Horsepower required to overcome wind drag

$$P_{WD} = \frac{71.5}{12} \text{ ft.} \times 3.03 \text{ lbf} \times 2.20 \text{ rad/sec} \times \frac{1}{550} \frac{\text{ft.-lbf/sec}}{\text{hp}}$$

$$P_{WD} = 0.072 \text{ hp}$$

<u>Rolling Resistance and Inertia</u>

$$f = 0.01 \left(1 + \frac{V}{100}\right) \quad (4)$$

f = rolling resistance coefficient, dimensionless.

V = 11.25 mph, wheel speed.

Rolling resistance equals f multiplied by wheel loading.

$$F = f \times 1085 \text{ lbf/tire} \times 2 \text{ tires}$$

The value of f varies little and it is convenient to define an average value.

$$f_o = 0.01$$

$$f_v = 0.011$$

$$f_{ave} = \frac{f_o + f_v}{2} = 0.01$$

$\bar{F} = f_{ave} \times 1085$ lbf/tire x 2 tires

$\bar{F} = 22.92$ lbf

Summing torques about the axis of rotation of the tester:

$$T = J\ddot{\theta} + \bar{T}$$

T = torque required, ft.-lbf

J = mass moment of inertia about axis of rotation, ft.-lbf-sec^2.

$\ddot{\theta}$ = angular acceleration, rad/sec^2.

$\bar{T} = 7.5$ ft. x $\bar{F} = 1.719$ ft.-lbf, average torque due to rolling resistance.

Rearranging the torque equation:

$$T - \bar{T} = J\ddot{\theta}$$

Integrating once and solving for start-up time yields:

$$t = \frac{J\ddot{\theta}}{T-\bar{T}}$$

The mass moment of inertia can be calculated from following relation:

$$J = \frac{2}{g} \sum_{i=1}^{10} W_i r_i^2$$

$$= \frac{2}{32.17 \times 144} [70\times(90)^2+2\times(87.5)^2+10.9\times(86.75)^2+5\times(83)^2+24\times(79)^2$$

$$+ 43.5\times(65)^2+1200.5\times(75)^2+58.8\times(50.5)^2+340.7\times(43)^2$$

$$+ 10.7\times(27)^2]$$

$$= 3701.2 \text{ ft.-lbf-sec}^2$$

For a start-up time of 30 sec:

$$T = \frac{J\dot{\theta}}{30} + \bar{T}$$

$$= \frac{37\ 01.2(2.20)}{30} + 171.9$$

$$= 433.3 \text{ ft.-lbf}$$

Power required for a 30 sec. start-up:

$$P = T\omega = \frac{433.3 \times 2.20}{550}$$

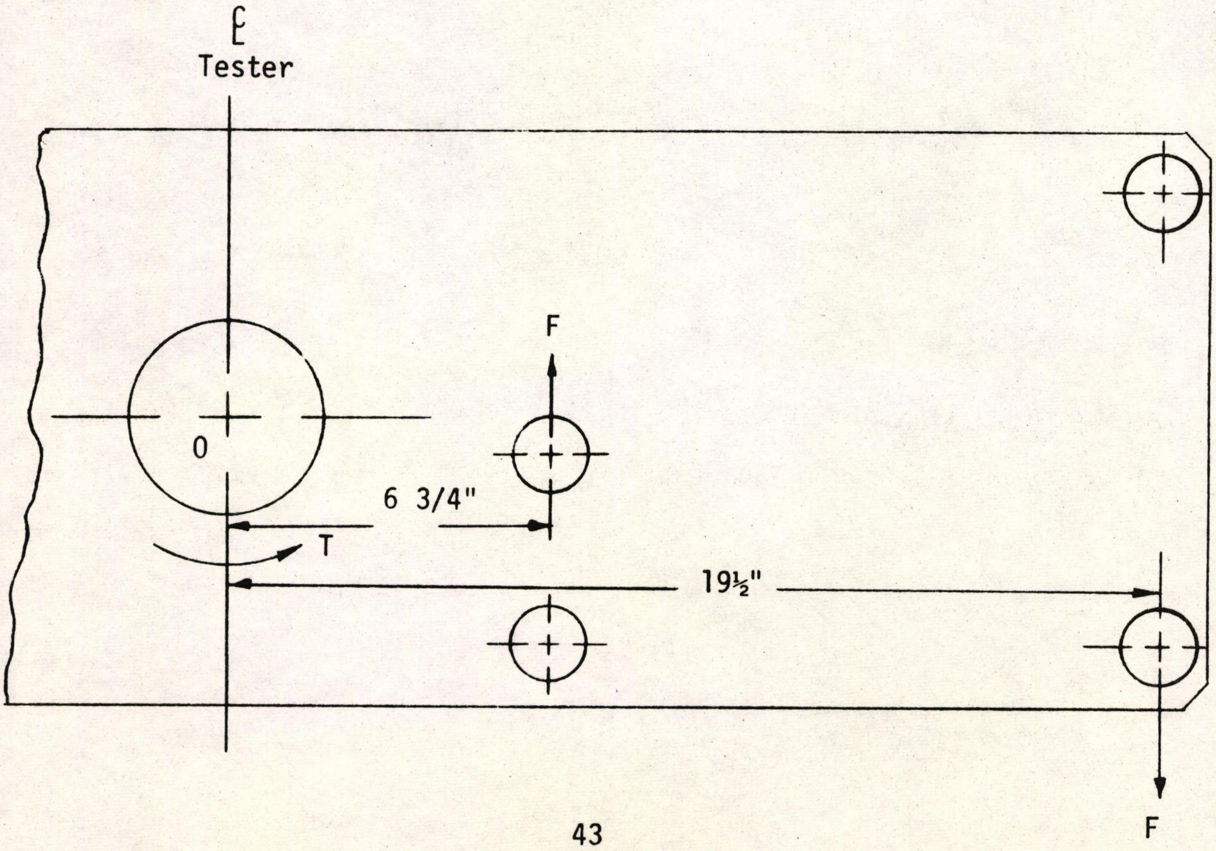
$$= 1.77 \text{ hp}$$

Total hp required to overcome wind resistance, rolling resistance and inertia for a 30 sec start-up:

$$P_T = P_{wd} + P = 1.85 \text{ hp}$$

Cam Type Bearings

A free body diagram of half of the rotating center structure was drawn to determine if the bearings would support the loads generated by the drive motor.



At Stall

$$\Sigma M_o = 0$$

$$T = 19.5F - 6.75F$$

F = maximum radial load, lb

T = one half of motor torque, in.-lb

From specifications (Osborn Load Runners), the maximum radial load that the bearings will support equals 640 lb. Substitution yields:

$$T = 19.5 \times 640 - 6.75 \times 640$$

$$= 8160 \text{ in.-lb}$$

$$= 680 \text{ ft.-lb}$$

The two bearings shown will handle this torque.

From the motor specifications (Vickers Industrial Products Catalog) maximum motor torque equals 580 ft.-lb. The two bearings shown must handle one half of this torque or 290 ft.-lb. Therefore:

$$\text{factor of safety} = \frac{680}{290} = 2.34$$

Hydraulic Analysis

For a 5 in. bore cylinder with a 1in. rod.

$$A_{HE} = \frac{\pi 5^2}{4} = 19.63 \text{ in.}^2$$

$$A_{RE} = \frac{\pi(5^2-1^2)}{4} = 18.85 \text{ in.}^2$$

$$Q_{HE} = V_a A_{HE} = 0.35 \times 19.63 = 6.87 \text{ in.}^3\text{/sec.}$$

$$= 1.79 \text{ gpm}$$

Flow through drive motor:

$$Q_m = D_m \omega$$

D_m = 24.0 in.^{3}/rev, motor displacement

Substitution of Dm yields:

$$Q_m = 24.0 \times 21.01$$

$$= 504.2 \text{ in.}^3/\text{min}$$

$$= 2.18 \text{ gpm}$$

Total flow required:

$$Q_T = Q_m + Q_{HE} + Q_L$$

Q_L = leakage flow, gpm

Substitution yields:

$$Q_T = 3.97 + Q_L$$

Assuming a leakage flow of 1 gpm yields:

$$Q_T = 4.97 \text{ gpm}$$

In the line between the pump and the flow divider assembly, maximum flow occurs.

at 100°F:

$\mu \approx 2.0 \times 10^{-6}$ lb.-sec/in.^{2} , viscosity of hydraulic oil.

$\rho = 0.78 \times 10^{-4}$ lb.-sec^2/in.^{4}, density of hydraulic oil.

The Reynolds number:

$$N_{RE} = \frac{4 \rho Q_T}{\pi \mu D}$$

D = 0.402 in., inside diameter of hydraulic tubing.

Substitution into the above formula yields:

$$N_{RE} = \frac{4 \times 0.78 \times 10^{-4} \times 4.97}{\pi \; 2.0 \times 10^{-6} \times 0.402} \; \frac{231}{60}$$

$$= 2364 > 2000$$

The Reynolds number is in the transition range and the flow may possibly be turbulent.

The drive motor develops 32 ft.-lbs. of torque per 100 psi of pressure drop across motor.

$$\text{Torque} = \frac{1.85 \times 550}{2.20} = 462.5 \text{ ft.-lb}$$

$$\text{Torque} = 32 \times \frac{\Delta\rho}{100}$$

Solving for $\Delta\rho$ and substituting values yields:

$$\Delta\rho = \frac{462.5 \times 100}{300} = 1445.3 \text{ lb/in.}^2$$

Track Coverage

Track coverage is required to be random. The frequency of the circular motion was compared to the frequency of radius variation and the ratio of the two frequencies was set equal to an integral multiple of an irrational number to insure that the two motions coincide only once.

$$\frac{f_T}{f_a} = a\sqrt{2}$$

f_T = 2.20/2π cycle/sec, frequency of tester rotation.

f_a = frequency of radius variation, cycle/sec

a = integer

$$f_a = \frac{2.20/2\pi}{a\sqrt{2}}$$

$$= \frac{0.248}{a} \text{ cycle/sec}$$

From initial specifications

$$f_a \leq \frac{1 \text{ cycle}}{10 \text{ rev}} \times \frac{1 \text{ rev}}{2 \text{ rad}} \times 2.20 \text{ rad/sec}$$

$$\leq 0.0350 \text{ cycle/sec}$$

For a = 8

$$f_a = 0.0309 \text{ cycle/sec} < 0.0350 \text{ cycle/sec}$$

For this frequency the speed of radius variation

$$v_a = 0.0309 \text{ cycle/sec} \times 10\text{in./cycle}$$

$$= 0.309 \text{ in./sec}$$

This speed is below the maximum allowable value of 0.35 in./sec.

APPENDIX B

First Draft 11-7-70
Revised to Recommended
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Revised 5-1-71
Revised 8-24-71

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Subcommittee E 17.22

TENTATIVE RECOMMENDED PRACTICE OF THE CIRCULAR TRACK METHOD FOR DETERMINING THE WEAR RESISTANCE OF PAVEMENT SURFACE MATERIALS

1. Scope

1.1 This recommended practice is a guide for determining pavement slipperiness characteristics through the medium of polishing or wearing specimens of pavement surface or laboratory mixed specimens which simulate the pavement surface using a circular track.

1.2 The circular track will be utilized to hold the specimens in order that the specimen surface may receive the polishing action of full size tire(s) as the tire(s) rotate about the axle and around the track.

2. Apparatus

2.1 Track Dimensions - The minimum radius distance from the center of the track to any part of the tire shall not be less than 2 feet (61.0 cm). The circular track pad width should be sufficient to contain the polish specimen without causing excessive edge stresses and deterioration to the specimen.

2.2 Axle-Wheel Assembly - The Axle-Wheel Assembly shall be of rugged construction and should conform to the following requirements:

2.2.1 Suspension - The wheel suspension should include both springing and shock-strut damping.

2.2.2 Wheel Load - The axle-wheel assembly design shall be such as to provide a loading of 1085 ± 25 lb. (492.2 ± 11.3 kg) to each wheel at all times during polishing.

2.2.3 Rim Size - The rims shall be capable of mounting a 7.50 -14 track test tire.

2.2.4 Adjustment - The axle-wheel assembly design should be such as to provide adjustment to both the toe and camber of the wheel to a plane perpendicular with the centerline of the axle.

2.2.5 Wheel Applications - The axle-wheel assembly should be designed such as to provide a continuously varing radius as measured from the center of the track to the center of the tire tread. The assembly will allow the wheel to be slowly forced to change in radial positions, causing a variable radius of at least 5 inches (12.7 cm). The radial positioning should be randomized with time and/or circumferential travel to prevent tire tracking. The rate of change in the variable radius should be such that the full radial change be experienced in not less than five revolutions of the axle(s). The centerline of the axle shall be parallel to an imaginary horizontal plane formed through the center of the polish path of the specimens. The wheel should be positioned perpendicular to the axle centerline.

2.2.6 Instrumentation - an instrumentation system will be utilized whereby a continuing record of wheel applications and angular velocity will be readily available.

2.3 Power requirements - The source of rotational power to the axle(s) should be sufficient to maintain a selected angular velocity. The linear wheel velocity of the test tire should not exceed 1.5 times the minimum radius distance where the dimension of the linear wheel velocity is in miles per hour and the dimension of the minimum radius is in feet. In no event should the linear wheel velocity exceed 20 mph (32.2 km/hr.).

2.4 Test Tire - The polishing effort shall be applied through a tire(s) conforming to the requirements of ASTM Specifications E 249, Standard Tire for Pavement Tests. The Test Tire shall be inflated in accordance with ASTM Specification E 249, Standard Tire for Pavement Tests.

3. Test Specimen

3.1 Specimen Shape and Dimensions - Test Specimen(s) may vary in shape and dimension but the specimen(s) shall be aligned on the circular track pad in such a manner as to allow a continuous circle of material to be polished. The minimum specimen dimension perpendicular to the direction of the tire travel shall be such that at no time during polishing shall the distance measured radially from the edge of the specimen, to any part of the tire be less than the thickness dimension of the specimen unless the specimen is restrained along the edges. If the specimen is restrained along the edges, the minimum specimen dimension perpendicular to the direction of tire travel shall be such as to allow 1 inch (2.5 cm) additional distance on each side of that portion of the specimen which receives the polish effort of the tire. The arc length of the specimen measured at the center of the polish path of the test tire shall be not less than 8 inches (20.3 cm).

3.2 Specimen Preparation - Test Specimens may be obtained from the roadway or prepared in the laboratory. Laboratory specimens shall be prepared in a manner to closely simulate the surface characteristics of the pavement type being studied through out the entire testing period.

3.3 Method of Retaining Specimens - Specimens shall be retained on the track pad in a manner to eliminate movement of the specimens on the surface of the track pad. Specimen(s) shall be prepared and placed in such a manner as to minimize wheel bounce. If more than one specimen is to be placed on the track pad, joint openings between specimens shall not be greater than 1/8 inch (3.2 mm) and thickness differentials between specimens shall not exceed 1/16 inch (1.6 mm).

4. Environment

4.1 The specimen shall be dry during polishing. The ambient temperature during polishing should not be less than 60° F (15.6° C) or more than 80° F (26.7° C).

5. Friction Determinations

5.1 Friction Measurement Intervals - Friction values shall be obtained after the specimen(s) have been placed on the track and prior to any polish effort; at periodic intervals of sufficient number to clearly define the rate of polish; and at the completion of polishing before the specimen(s) are removed from the track pad. The "completion of the test" shall be defined as the point at which there is no significant change in friction, either an increase in friction value or a decrease in friction value, after repeated polish applications of the tire(s).

5.2 Number and Position of Friction Measurement - Reported friction measurement shall be obtained in the direction of tire travel and at the center of the polish path. Sufficient measurements shall be collected to reflect the friction value of the specimen and unless the friction measurement is obtained along the entire arc length of the specimen, at least two "spot" friction values should be reported.

6. Report

6.1 The report should include such data as listed in 6.1.1 through 6.1.7 along with other pertinent data as seemed necessary, especially those items which do not conform to the subject recommended practice:

6.1.1 Pavement type, mix composition and material type.

6.1.2 Surface characteristics.

6.1.3 Type of Friction Tester Utilized.

6.1.4 Lineal wheel velocity when polishing calculated when the wheel is positioned at the center of the polish path.

6.1.5 Minimum and Maximum Radius Dimensions

6.1.6 Date of friction test, number of applications at time of testing, and friction value.

6.1.7 Remarks.