DEVELOPMENT AND FABRICATION OF THE VIRGINIA SKID-RESISTANCE MEASUREMENT VEHICLE (MODEL 2)

by

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(The opinions, findings, and conclusions expressed in this report are those of the authors and not necessarily those of the sponsoring agencies.)

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SUMMARY

The inefficiency of the Virginia Highway Research Council, Model 1, skid measurement trailer, and the increasing effort expended by the American Society for Testing and Materials toward the development of more stringent specifications for pavement skid-resistance measurement vehicles made it imperative that the Virginia Highway Research Council obtain a more reliable skid-measurement vehicle if it was to maintain its level of excellence in skid-resistant highway surface development.

The Virginia Highway Research Council, Model 2, skid-resistance measurement vehicle developed in the work reported here consists of a modified Ford F-600 truck chassis, a locally designed and fabricated truck body, the severely modified and rebuilt VHRC, Model 1, skid-resistance measurement trailer, and a completely new instrumentation and recording system whose components are divided between the towing vehicle and the skid-resistance measurement trailer.

This report is devoted to a limited discussion of some of the parameters which were considered in the development of the VHRC, Model 2, skid-resistance measurement vehicle, a description of the resulting vehicle, and a brief description of the operation and limitations of the various systems which make up the vehicle.

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INTRODUCTION

The coefficient of friction existing between highway surfaces and the periphery of rubber tires has been a matter of concern to vehicle operators ever since the introduction of the first motor vehicles. Although this earlier interest may have evolved only about the possibility of starting and stopping the vehicle, the advent of more powerful, faster, and heavier motor vehicles has increased the significance of this frictional force, for it is this force which limits the starting and stopping ability and the degree of directional control that is available for maneuvering all ground contacting, wheel driven motor vehicles.

There are many parameters which influence the coefficient of friction between highway surfaces and various combinations of tires and wheels. These can be divided into three groups: road surface factors, tire factors, and vehicle operation factors. Some of these parameters are as follows:

(1) Road Surface Factors include the type, condition, and texture of the aggregate, the type and quantity of the aggregate binder, surface temperature, surface contamination, surface modification due to weathering and polishing from use, and surface geometry factors relating to elevation, crowning, inclination, curvature, and all possible combinations of these parameters.

(2) Vehicle Tire Factors include tire size, tire load, tire tread pattern, inflation pressure, contact area, tread and sidewall rubber composition, tire body cord composition and geometry, and all possible combinations of these factors.

(3) Vehicle Operation Factors include vehicle size, weight and weight distribution, vehicle speed, rate of acceleration or deceleration, and rate of change of direction of travel.

Members of the Virginia Highway Research Council have been actively interested in the problem of controlling pavement slipperiness since the early days of the Council.[1] Several types of pavement slipperiness measurement methods have been investigated by

the Council over the years of its existence[2,3,4]; but early in the program, it was realized that the most desirable type of slipperiness measurement instrument would be one which was capable of accurately measuring the coefficient of friction of pavement surfaces while operating at, or near, normal traffic speeds, in normal traffic patterns, and with a minimum of disruption of normal traffic flow. The instrument should be self-contained, and should be easily operated by reasonably skilled technicians. And, since pavement surfaces are generally more slippery when wet, the instrument should measure the coefficient of friction of uniformly wet areas of the pavement surface.

The General Motors Corporation published a description of a trailer type vehicle for measuring tire-road coefficient of friction in 1957.[5] The Virginia Highway Research Council used this design, with some modifications, as a pattern for the construction of the Virginia Highway Research Council, Model 1[6], skid-measurement trailer.

The Virginia skid unit did not correlate satisfactorily with the stopping-distance car or with any other skid-test trailer. The force measuring system used in the Virginia unit consisted of a 1949 Buick differential and torque tube assembly mounted under a specially designed trailer body using selected components of the Buick suspension system, a Brush Development Corporation, Model BL-202, dual channel recording oscillograph, and a Brush Development Corporation, Model RD-561200 strain gage amplifier. These last two components were mounted in the towing vehicle. Strain gage elements mounted on the torque tube near the differential sensed the bending moment developed in the tube caused by the application of the trailer brakes, and transmitted this force signal to one channel of the recording oscillograph. This type of force measuring system made it possible to obtain, using a single recorder channel, an indication of the frictional force developed between sliding trailer tires and the pavement surface when the wheels were braked either individually or simultaneously; but unfortunately, this measurement system geometry also was sensitive to the rolling friction of the wheels, axles, bearings, and differential components, and was also sensitive to vertical excursions of the axle and wheel assembly. The resulting oscillograph trace was difficult to evaluate quantitatively with any degree of confidence.

The measurement system difficulties just described together with other clearly identifiable, but less severe, difficulties inherent in the VHRC, Model 1, skid-measurement vehicle, and the increasing effort expended by the American Society for Testing and Materials toward the development of more stringent specifications for pavement skid-resistance measurement vehicles made it imperative that the Virginia Highway Research Council obtain a more reliable skid-measurement vehicle if it was to maintain its level of excellence in skid-resistant highway surface development. The insistence of some members of the Virginia Department of Highways that the VHRC, Model 1, skid-resistance measurement vehicle be rebuilt to meet the new specifications rather than to permit the design of a completely new vehicle dictated, to some extent, the geometry and performance of the VHRC, Model 2, skid-resistance measurement vehicle.

This report is devoted to a limited discussion of some of the parameters which were considered in the development of the VHRC, Model 2, skid-resistance measurement vehicle, to a description of the resulting vehicle, and to a brief description of the operatic

and limitations of the various systems which make up the vehicle. This effort has been the joint responsibility of members of both the Virginia Highway Research Council and the Instrumentation Development Group, Research Laboratories for the Engineering Sciences, School of Engineering and Applied Science, University of Virginia[7,8].

GENERAL CONSIDERATIONS

The Virginia Highway Research Council, Model 2, skid-resistance measurement vehicle has been developed as a composite unit consisting of two mutually dependent components, (1) a towing vehicle and (2) an instrumented trailer. Both of these components were designed to work together in complete harmony, and both were also designed to satisfy, as nearly as possible, the following general specifications.

A. Safety of Both Operating Personnel and Normal Highway Traffic

It was intended that the vehicle be capable of operation at maximum legal passenger car speeds. This is 65 mph in Virginia. Safe operation at these speeds while performing skid-resistance measurements demands that both the towing vehicle operator and the instrument operator sit high enough to have an unobstructed view of the roadway, both in front of and behind the test vehicle. The vehicle must be instantly visible to motorists but not be so outstandingly different from standard vehicles as to invoke unwarranted curiosity and thus provide a distraction for motorists. And certainly, the composite vehicle must, at all times, be capable of passing all applicable Interstate Commerce Commission, state, and local safety regulations.

B. Measurement Performance

The VHRC, Model 2, unit was to be designed as a dual purpose vehicle. It was intended that the vehicle be capable of reasonably efficient application to routine skid-resistance measurement problems and that it could also be applied, without extensive systems modifications, to unique surface studies. The instrumentation should be designed so that a reasonably competent technician could safely perform routine, low speed skid-resistance measurements without assistance. The vehicle should be designed to be self contained so that in excess of 200 skid measurements could be completed between stops. And certainly, the vehicle must be capable of performing skid-resistance measurements which conform to the recommendations established by the American Society for Testing and Materials.[9]

C. Maintainability

The vehicle should be designed to make use of readily available, standard components where such use does not degrade the response or reliability of the vehicle. Especially designed and fabricated parts and components should be reduced to an absolute

minimum. Since the skid unit is likely to be away from its home base for periods of up to two weeks, the vehicle must be designed so that the necessary additional equipment such as tires, tools, and spare parts that are required to maintain the unit may be safely carried on it.

The Virginia Highway Research Council, Model 2, skid-resistance measurement vehicle, shown in Figure 1, conforms to most of the general specifications that have just been discussed. It consists of a modified Ford F-600 truck chassis, a locally designed and fabricated truck body, the severely modified and rebuilt VHRC, Model 1, skid-resistance measurement trailer, and a completely new instrumentation and recording system whose components are divided between the towing vehicle and the skid-resistance measurement trailer.

Figure 1. The VHRC, Model 2, skid-resistance measurement vehicle.

THE TOWING VEHICLE

A. Selection Factors

Considerable analytical effort was devoted to a study of desirable operating parameters prior to the preparation of the technical specifications that were used to obtain the towing vehicle that is a part of the VHRC, Model 2, skid-resistance measurement vehicle. Previous experience with the VHRC, Model 1, vehicle had indicated that the overall length of the new unit should not exceed 33 feet if adequate maneuverability were to be assured. Since the trailer portion of the Model 1 unit could not be readily shortened to less than about 14 feet, the overall length of the towing vehicle could not exceed 19 feet. Maneuverability considerations also indicated that the wheel base of the new towing vehicle should not exceed 11 feet. The existence of numerous bridges and underpasses in Virginia having vertical clearances of 10 feet or less dictated that the maximum height of the towing unit be held to $9\frac{1}{2}$ feet and the existence of narrow automobile-traffic channels led to a 90 inch maximum width limitation.

It was arbitrarily decided that the towing vehicle should be capable of carrying 300 gallons of water to be used for testing. Estimated truck chassis, body, and load weights indicated that the towing vehicle should have a gross vehicle weight of about 13,000 pounds. The ASTM publication cited in reference 9 specifies that the optimum weight of the towed vehicle be 2320 pounds and that this weight remain fixed for all standard skid-resistance measurements.

A limited knowledge of the maximum grades on various Virginia highways and of the maximum surface friction values that are likely to be encountered led to the general requirement that the towing vehicle should be capable of negotiating a 6% grade at a speed of 55 mph while carrying a gross vehicle weight of 10,000 pounds (almost empty water tank) and pulling the 2320-pound trailer with both wheels locked and sliding over a pavement surface having a coefficient of friction of 0.75.

The towing vehicle should be capable of rapidly accelerating up to a speed of 60 mph. The transmission and drive train gearing should be sufficiently flexible so that the vehicle could be operated for short periods of time at maximum engine output power over a wide range of vehicle speeds when measurement conditions demanded; and also, this flexibility should permit the vehicle to be operated over long periods of time at high vehicle speeds without unduly shortening the engine life.

Selection of the best possible combination of towing vehicle engine, transmission, differential ratio, and tire size for a given application does not necessarily follow a unique pattern. For example, the vehicle engine can not be selected on the basis of total horsepower alone. The total horsepower output of an automotive engine is an indication of the maximum power that can be developed only under unique test conditions and few engines will develop maximum power outputs for more than a few minutes without disintegrating.

The net output torque rather than the horsepower is the controlling factor in predicting the capability of a vehicle to carry or pull a load. This ability may be expressed in foot-pounds for the engine or in pounds thrust if the rolling resistance of the vehicle, the rolling radius of the vehicle tires, and the gear train ratio are known.

The shape or progressive slope of the engine torque vs.speed curve, the width of the speed range over which high torques are developed, and the flatness of the high torque portion of the curve are of maximum importance in selecting the proper engine for a specific job. Generally, engines which develop high torques at low speeds will give more usable tractive effort and will also give increased engine life.

The Society of Automotive Engineers Handbook, Supplement 82[10], has been of considerable assistance in the analytical matching of the desired vehicle operating parameters to vehicle manufacturers' published data. An outline of one method used at the VHRC to study this problem is as follows:

1. Using the tables and equations from the SAE Handbook, estimate the power required to overcome the air resistance, rolling resistance, and the chassis friction resistance of a towing vehicle of the required size and weight.

2. Select the appropriate tire size for the vehicle load and service. Convert the total power indicated in step 1 above to tractive effort (force).

3. Add the tow bar pull imposed by the towed vehicle when it is operated in the desired measurement configuration.

4. Convert the net output torque of the engine under consideration to tractive effort using the available drive train gear ratios. Select an engine which gives the most desirable margin of excess available tractive effort over the required tractive effort. Study the entire range of engine and vehicle speeds.

Figure 2 shows the total rolling vehicle losses as a function of vehicle speed for the VHRC towing unit. Figure 3 shows the total tractive effort required for the vehicle as a function of vehicle speed and the total tractive effort that is available from the engine.
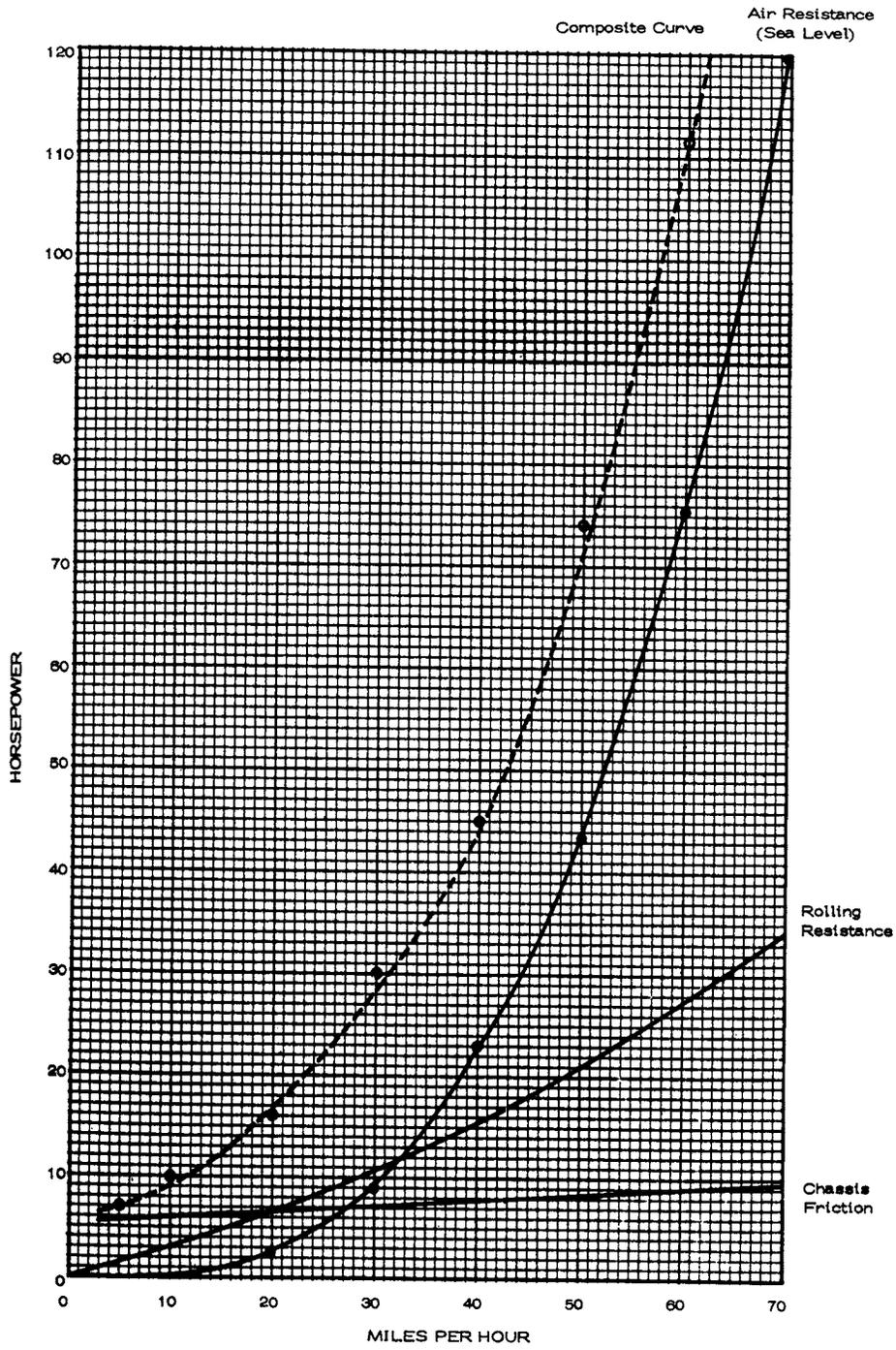
Figure 2. Total towing vehicle losses vs. vehicle speed.
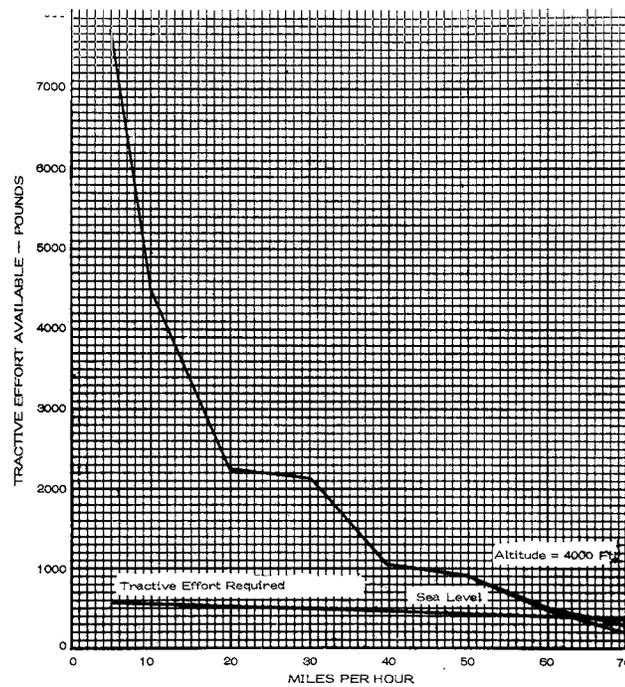
Figure 3. Total tractive effort required (towing unit and trailer) vs. vehicle speed and total tractive effort available vs. vehicle speed.

B. Vehicle Specifications

The more important vehicle specifications used in obtaining the VHRC towing vehicle are listed below:

1. Ford F-600 cab and chassis with 132" wheelbase.

2. Gross vehicle weight 17,000 pounds.

3. Engine specifications:

   a. 428 cubic inch displacement, industrial quality.
   b. Gross horsepower — 345 at 4600 rpm.
   c. Net horsepower — 218 at 3900 rpm.
   d. Gross torque — 462 at 2800 rpm.
   e. Net torque — 358 at 2500 rpm.

4. Front axle capacity — 5000 pounds.

5. Rear axle — Rockwell H-140, single speed, 17,000 pounds capacity and equipped with No-Spin differential gearing having a 5.83 to 1.00 ratio.

6. Transmission — Clark 264VO with synchromesh gearing and the following ratios:

   a. 1st — 6.06   d. 4th — 1.00
   b. 2nd — 3.50   e. 5th — 0.799
   c. 3rd — 1.80   f. Reverse — 6.00

7. Split shaft power takeoff capable of delivering 5 hp at test speeds and with speed ratio with respect to ground speed of vehicle constant and independent of transmission shift level position.

8. Dual rear wheels.

9. Tires — 7.50 x 20 8PR, tube type.

10. Alternator — 60 amps, 900 watts.

11. Air compressor mounted on engine and controlled by electromagnetic clutch.

12. ICC turn signals, emergency flasher signals, and cab corner clearance lights.

C. Vehicle Fabrication

The Richmond Motor Company, a Ford dealer located in Richmond, Virginia, was awarded a contract to supply the chassis and cab. In order that the 428 CID engine might be installed with a minimum of difficulty, the Richmond Motor Company ordered a Model F-600 truck equipped with a 330 CID Medium Duty engine. This engine provided a maximum number of components which could be used with the 428 CID engine. The portion of the 330 CID engine between the front cover plate and bell housing was removed and replaced with the corresponding portion of the 428 CID engine. The water pump, starter, alternator, fan, and fan belt which came with the 330 CID engine were used with the 428 CID engine. It was necessary to use a flywheel and pilot bearing from a 330 CID Heavy Duty engine in order to permit the installation of a heavy duty truck clutch. The connections used between the exhaust manifolds and the two heavy duty, free flow mufflers were locally fabricated. Both sections of the vehicle drive shaft were modified and a Chelsea Model No. 4714A split shaft power takeoff gear box equipped with a Chelsea Model No. 82 JHC power takeoff unit were mounted just behind the truck transmission. Figure 4 shows the essential dimensions of the completed vehicle chassis and cab.

The body for the towing vehicle was designed to complement the dimensions and weight distribution of the vehicle chassis. From considerations of stability, driver control, and weight distribution, it was desirable to have the water tank mounted as low as possible on the vehicle chassis. This arrangement lowers the center of gravity, increases the breakaway force required to initiate spin-out with the result that the vehicle should be capable of safely negotiating a large fraction of the existing highway curves at the posted maximum safe speeds. These considerations led to the conclusion that the water tank should be placed directly on the towing vehicle frame, and that the top of the water tank could then be used as the floor for the occupied portion of the vehicle body. This configuration gives an advantage of placing the instrumentation operator high enough so that he may see over the cab of the towing vehicle and thus, may more readily identify specific portions of the pavement surface that are to be investigated. This vantage point also permits him to see over the tops of the first four or five automobiles which may be following the measurement vehicle and thus he can avoid initiation of the measurement cycle when this action may endanger adjacent motor vehicle traffic.

DIMENSIONS			BASE CURB WEIGHT		
WB	=	132.0"	FRONT	=	3383#
CA	=	60.0"	REAR	=	2512#
AF	=	39.0"	TOTAL	=	5895#
OAL	=	204.5"			
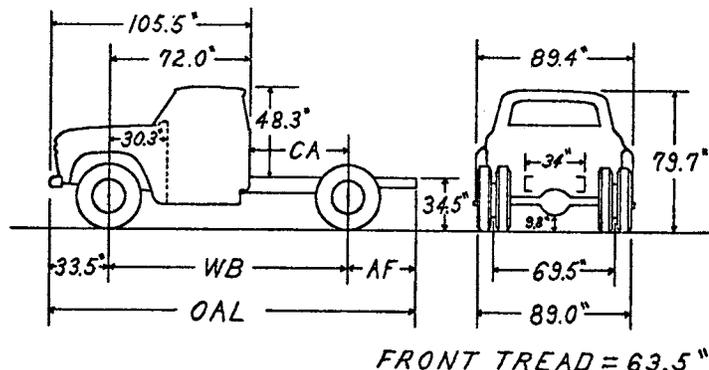
FORD F-600 (Modified)

Figure 4. Towing vehicle chassis dimensions and weights.

A 15-inch high water tank having a capacity of 300 gallons was designed in the form of three connected rectangular parallelepipeds. This geometry was chosen for ease of fabrication and to provide wells to accommodate the truck wheels. The three major sections communicate with each other by means of a "V" shaped trough that is formed along the center of the bottom plate and through holes which were installed in the tank baffles. The tank was designed with sufficient venting to permit safe filling from fire hydrants using a 2" fire hose. In an emergency, it can be filled from the top with a garden hose or other water source. Replaceable anode rods were installed to minimize rusting and the bottom drain opening was designed to permit complete drainage. The entire tank was fabricated from 7-gage sheet steel. Figure 5 shows a plan view of the tank and Figure 6 shows the completed tank installed on the towing vehicle chassis.

The towing vehicle body was designed to fit over the top of and rest on the water tank. Some of the major considerations in the design of the body were as follows:

1. The weight distribution of the body and tank should preserve the optimum distribution of 25% of the load carried by the front axle with 75% being carried by the rear axle.

2. The operator should have 360° visibility.

3. The interior illumination system should provide adequate illumination for work on dark days or at night.

4. The vehicle body should provide adequate protected space for operation and storage of supplies under adverse weather conditions.

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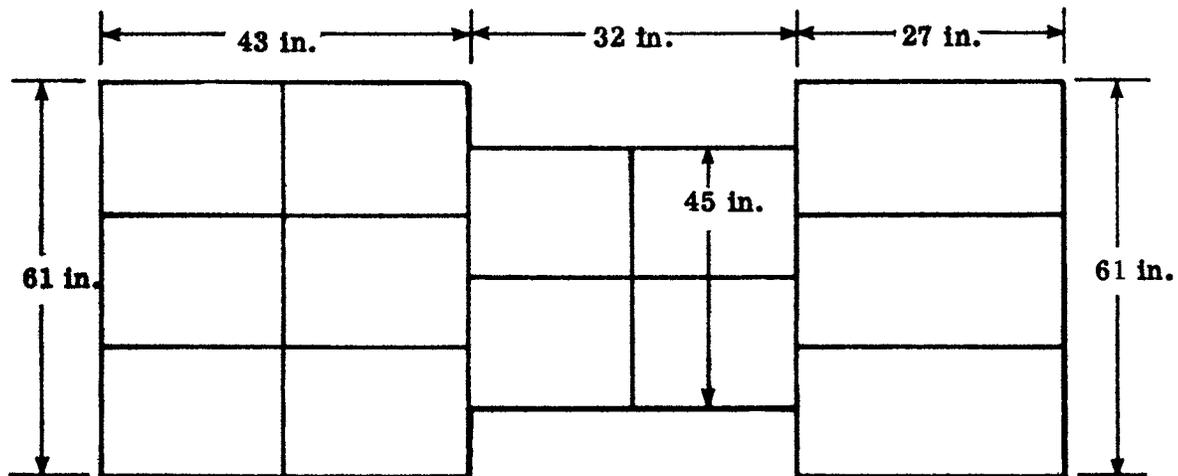
Figure 5. Plan view of water tank.
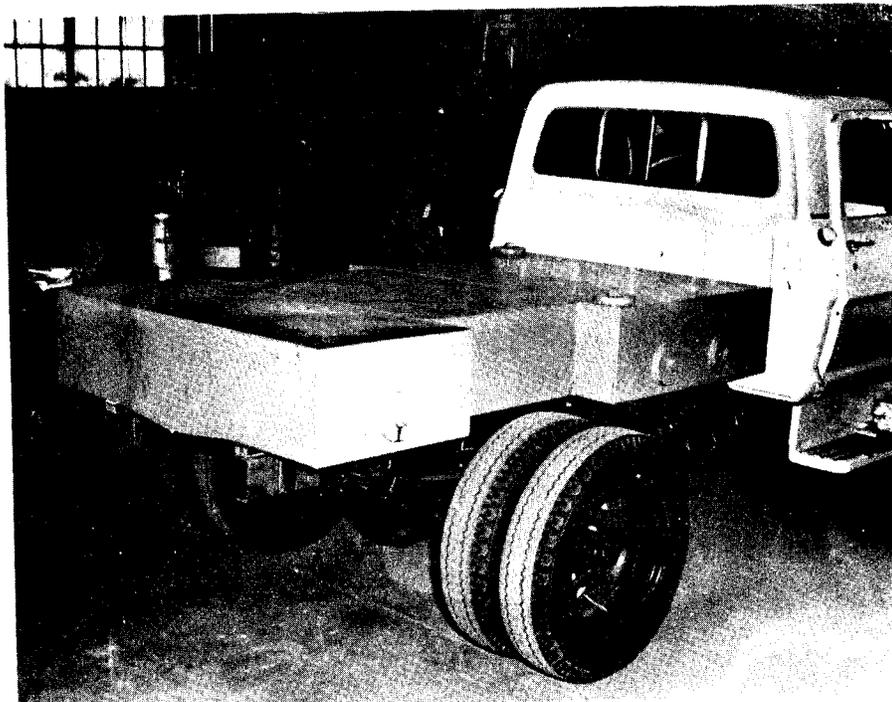
Figure 6. Water tank installed on chassis.

5. The operator's compartment should be heated for winter operation.

6. The rear of the vehicle body should provide adequate space of sufficient strength to permit the mounting of a trailer hitch and for the mounting of the connector panels and manifolds that are essential to the operation of the trailer.

In the interest of economy and ease of fabrication, the vehicle body was designed to make the maximum use of standard, readily available utility truck body components. The Baker Equipment Engineering Company of Richmond, Virginia, completed the fabrication of the body and installed the body and water tank on the truck chassis. Figures 7, 8, and 9 are plan views of the outside left, right, and rear body panels. Figure 10 is a photograph of the interior of the hose cabinet and Figure 11 is a plan view of the stationary cabinets that are built into the inside of the body on the left side of the operator's position. There are no cabinets opening into the body on the right side of the operator. This space was reserved for instrument installation.

The completed towing vehicle with the trailer attached, with two operators and all standard equipment aboard weighs a total of 12,540 pounds with the water tank empty. The front axle carries 3,740 pounds of this weight while the rear axle carries the remaining 8,840 pounds. With the water tank full, these weights increase to 15,460, 3,920 and 11,540 pounds respectively. The vehicle has exhibited the ability to accelerate the skid-resistance measurement trailer from 0 to 60 mph in 36.5 seconds with a full tank of water aboard. It is regularly driven on interstate highways at 65 mph and it has been used to measure the surface friction of the center portions of mile-long aircraft runways at speeds up to 75 mph. The average gasoline mileage obtained with the unit has been about 6.2 miles per gallon when the average is taken over a long period of time.
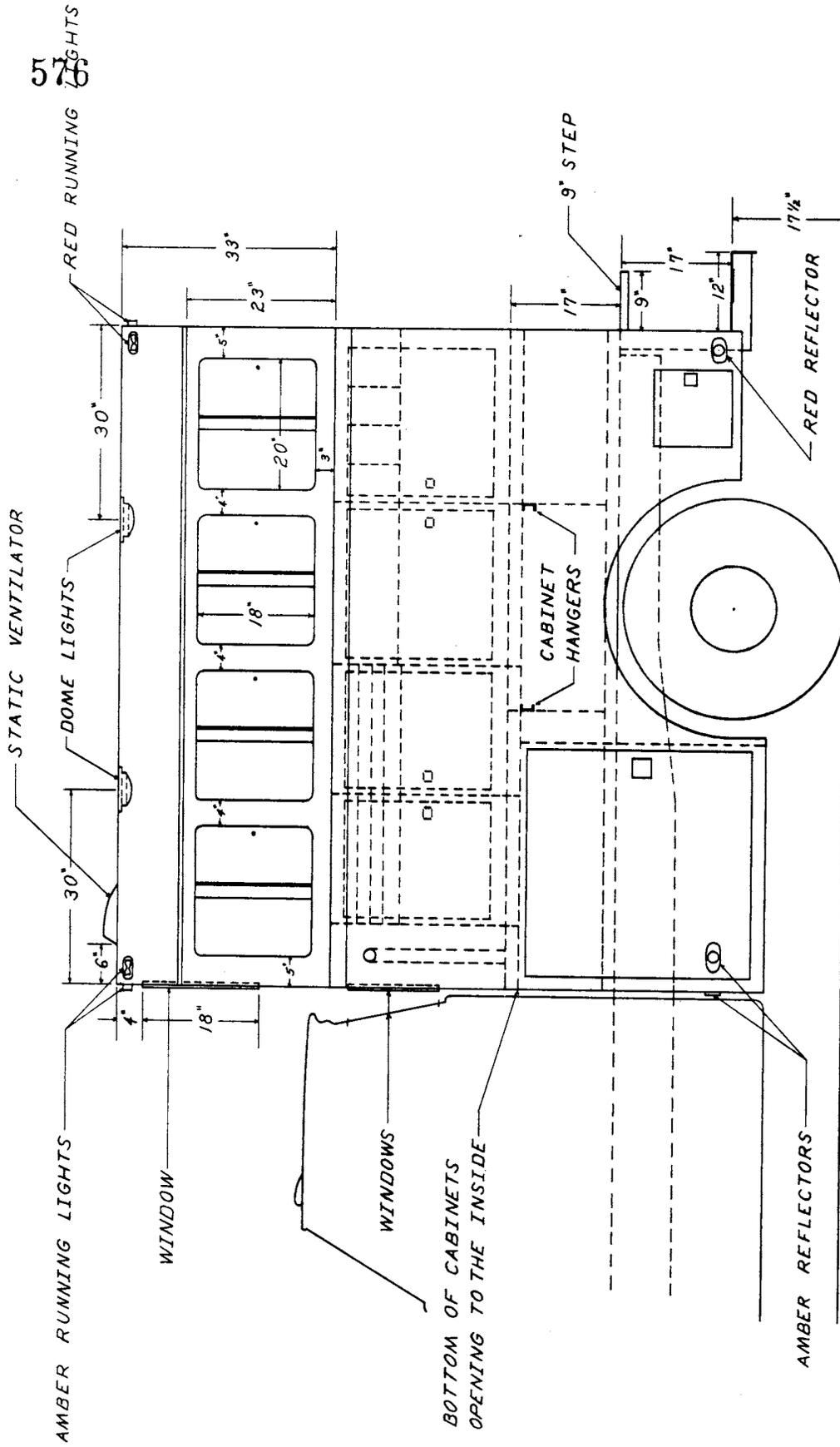
Figure 7. Body, plan view, left exterior.

577
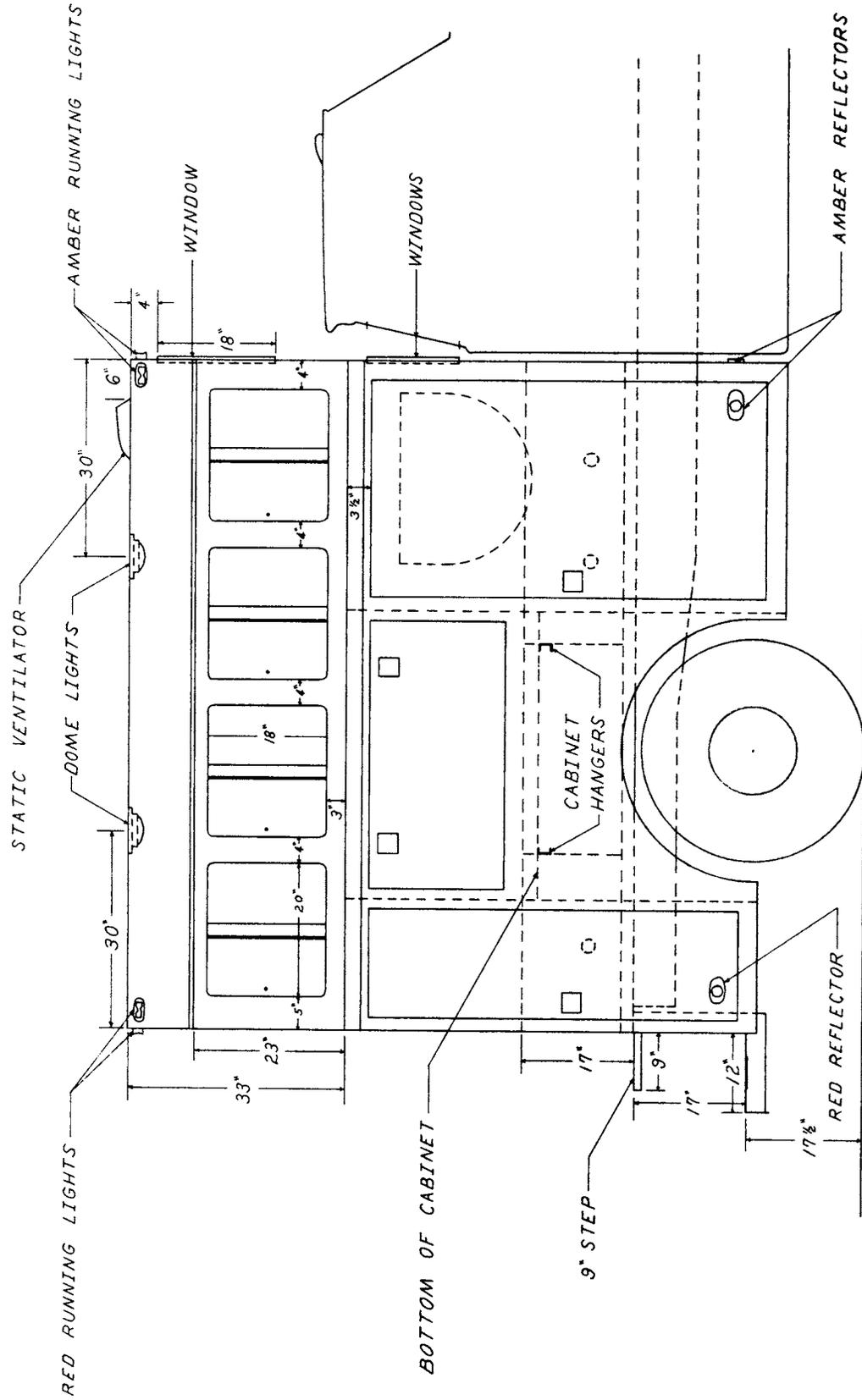
Figure 8. Body, plan view, right exterior.

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578
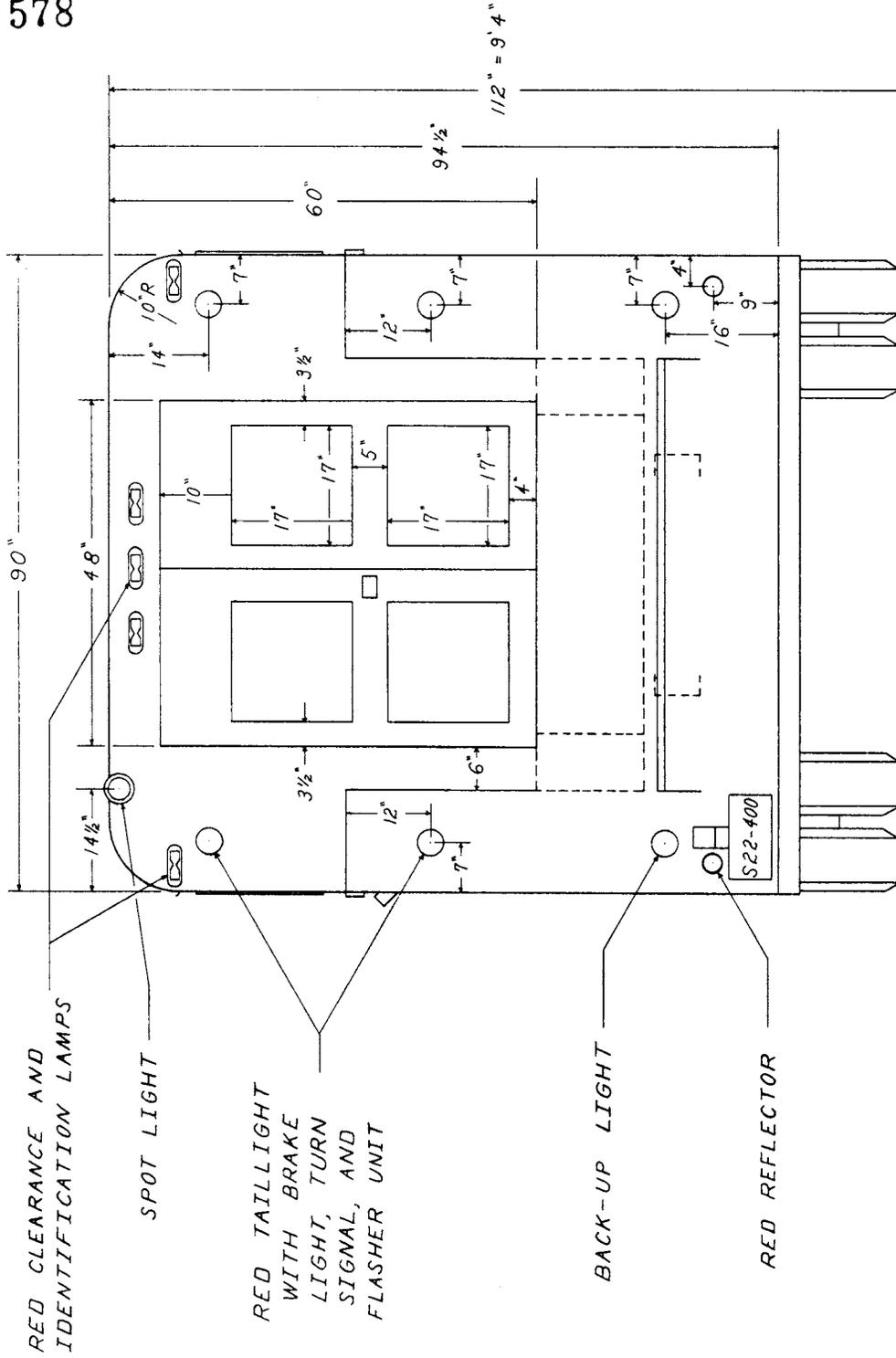
Figure 9. Body, plan view, rear.

RED CLEARANCE AND
IDENTIFICATION LAMPS

SPOT LIGHT

RED TAILLIGHT
WITH BRAKE
LIGHT, TURN
SIGNAL, AND
FLASHER UNIT

BACK-UP LIGHT

RED REFLECTOR

– 16 –
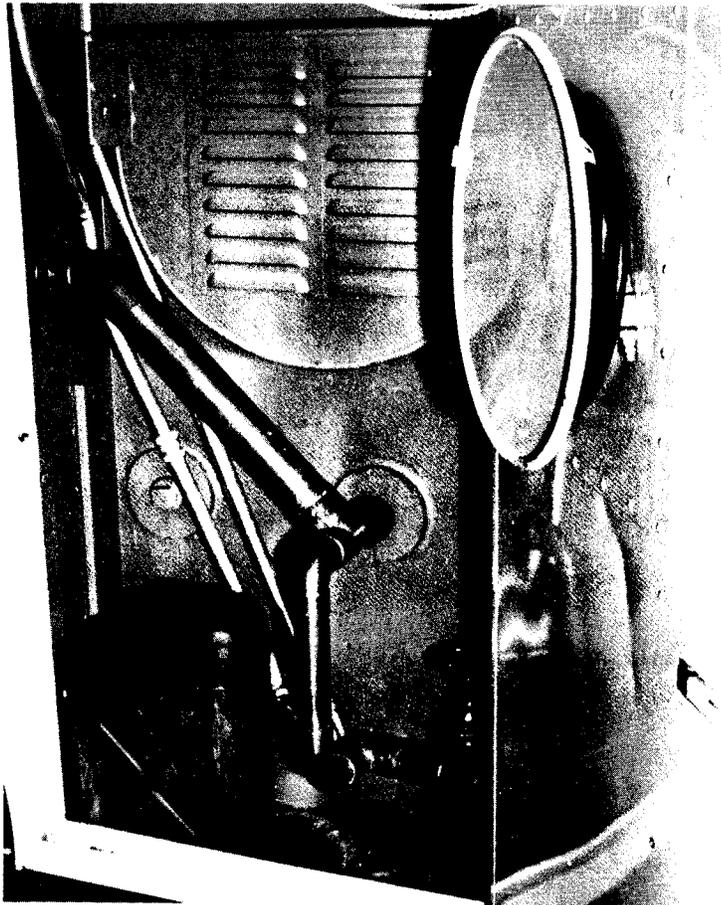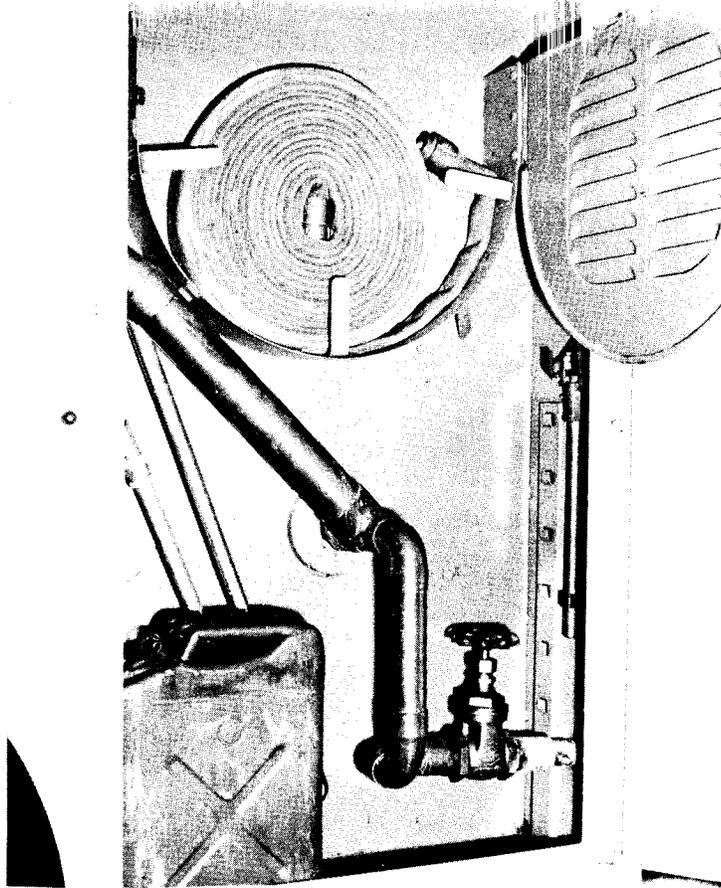
Figure 10. Hose cabinet interior.

580

Figure 11. Stationary cabinets, plan view.

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THE SKID TRAILER

The VHRC, Model-1, skid-resistance measurement trailer which was to be modified and rebuilt to conform to the more recent specifications was completely disassembled and all major parts were examined to determine which were suitable for use in the newly designed unit. The results of this examination were that the only really useful components were the trailer body and frame, the Buick differential torque tube and wish-bone assembly, the chassis springs, and the engine driven generator set which supplied 115 Volt, 60 Hz power to the recording instruments.

One of the goals of the rebuilding task was to develop a vehicle which could make repeated skid-resistance measurements at a rate of 5 measurements per mile while the test vehicle was travelling at 60 mph over a pavement surface having a coefficient of friction of 0.5. If one assumes that the test vehicle is equipped with shoe and drum brakes, that the drum is made of cast iron and weighs about 30 pounds, and that the test wheel requires 0.5 seconds from brake application to lockup, mathematical analysis shows that the rate of temperature rise of the brake drum, in the absence of cooling, will be about 45°F per minute. This would limit the measurement sequence to about 35 determinations or about 7 miles of pavement before brake overheating would terminate the sequence.

A. Trailer Frame and Wheels

A somewhat abnormal wheel and brake assembly geometry was adopted for the new vehicle in order to decrease the rate of brake drum temperature rise by making use of the stream of air which flows by the trailer body when the vehicle is in motion. The brake system was designed to place the brake drum outside of the test wheel. A Prior No. 3560-8, rectangular steel axle having a 4,000 pound load rating and a 60 inch normal tread width was used for this application.

The Buick torque tube used with the VHRC, Model 1, trailer was modified, lengthened, and fastened to an adapter flange which had been welded to the center of the new axle. This torque tube, with its associated wish-bone assembly, was attached to the trailer frame by means of a ball-and-socket joint located in line with and as near to the trailer hitch as the geometry of the trailer would permit. The torque tube thus serves to restrain the rotational tendencies of the trailer axle when the trailer brakes are applied; to provide a firm coupling to transmit the longitudinal shock imposed on the trailer axle during brake application directly to the trailer hitch instead of to the trailer frame; and, because of its long length, to force the trailer axle to execute nearly straight line vertical motion when accommodating pavement unevenness. An anti-sway bar fastened to the trailer axle at one end and to the trailer frame at the other end and running approximately parallel to the trailer axle is used to control lateral motion of the trailer body with respect to the axle. With this geometry, the position of the trailer wheels with respect to the trailer hitch is accurately known at all times. This is not the case for axles restrained by leaf springs alone.

The original Buick coil springs were used to support the trailer body. New, heavy-duty, direct-acting, linear shock absorbers having closed and open lengths of

9-7/8 and 16-1/8 inches respectively were installed on pads located just in front of the trailer axle. Figure 12 shows the completed trailer axle assembly, Figure 13 is a photograph of the torque tube ball joint together with its preload spring, and Figure 14 shows the torque tube attached to the trailer frame.
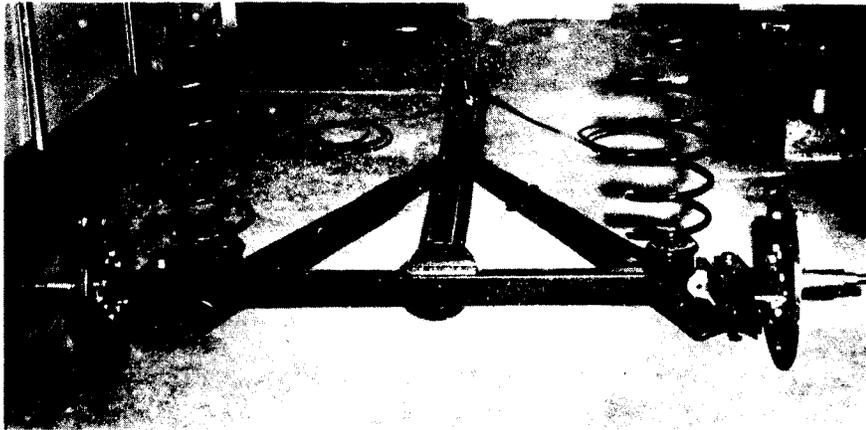
Figure 12. Trailer axle assembly.

The trailer wheels were fabricated from Budd No. M35210 wheels by cutting the hub from one wheel and the rim from a second. These were welded together with the hub reversed from its initial position. The resulting wheel and axle combination gives a center-to-center tread width of 64-3/4 inches when two size 7.50 x 14 ASTM standard test tires are mounted.

The trailer axle was supplied with two Prior No. 30071 bearing, hub, and drum assemblies. These drums were designed to accept 2 x 12 inch brake shoes. A 6-inch diameter steel flange was welded near each end of the trailer axle to position a set of 2 x 12 inch Bendix brake shoes mounted on a Bendix backing plate in the center of the brake drum contact area. The backing plates and brake shoes were designed for use in the 1960-63 series of 3/4 ton GMC trucks. The spot welds which normally hold the brake anchor pin bearing blocks to the backing plates were reinforced by running a single pass welding bead around the periphery of each bearing block. The remainder of the brake components used in this assembly were standard Bendix parts except for the brake anchor pins.
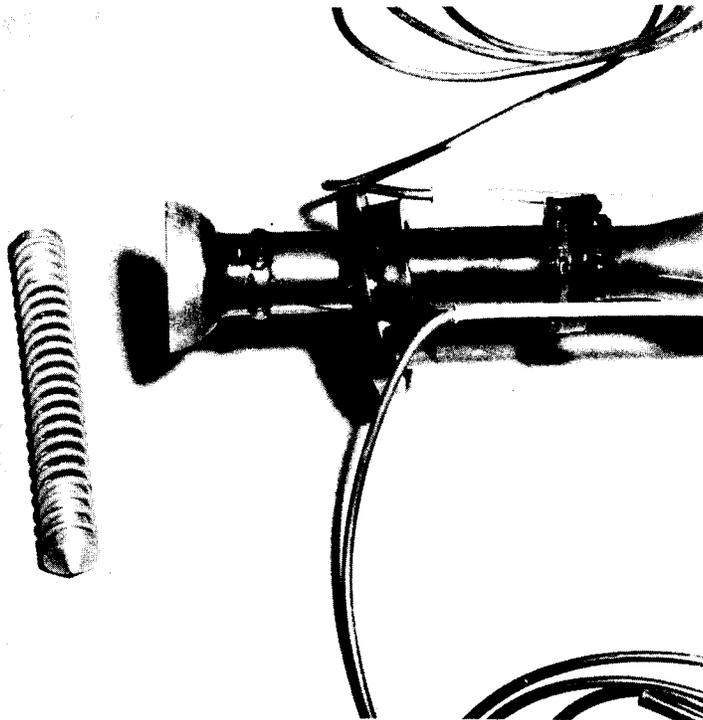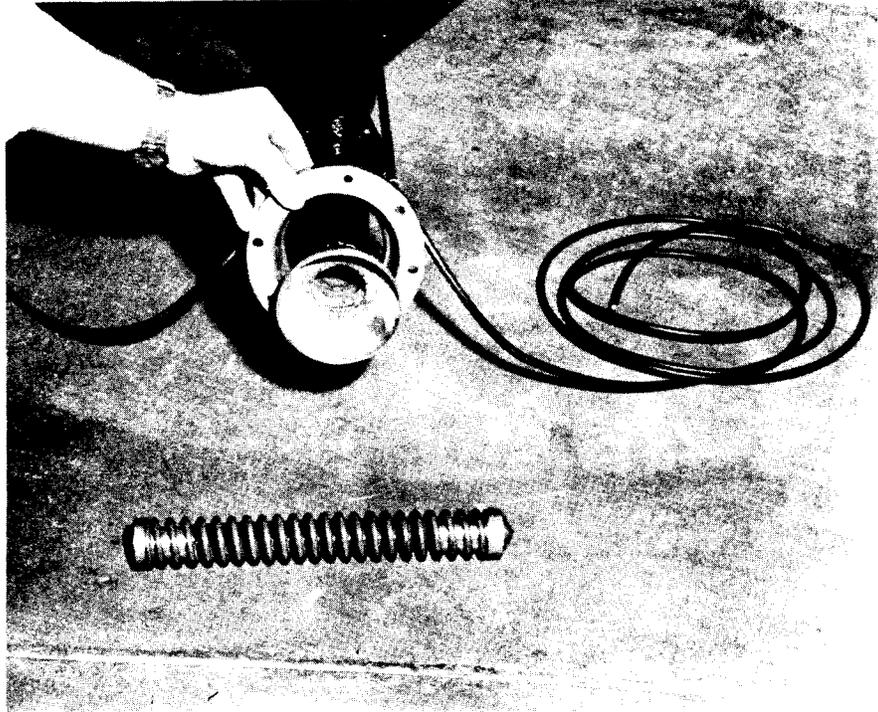
Figure 13. Torque tube with ball joint and preload spring.

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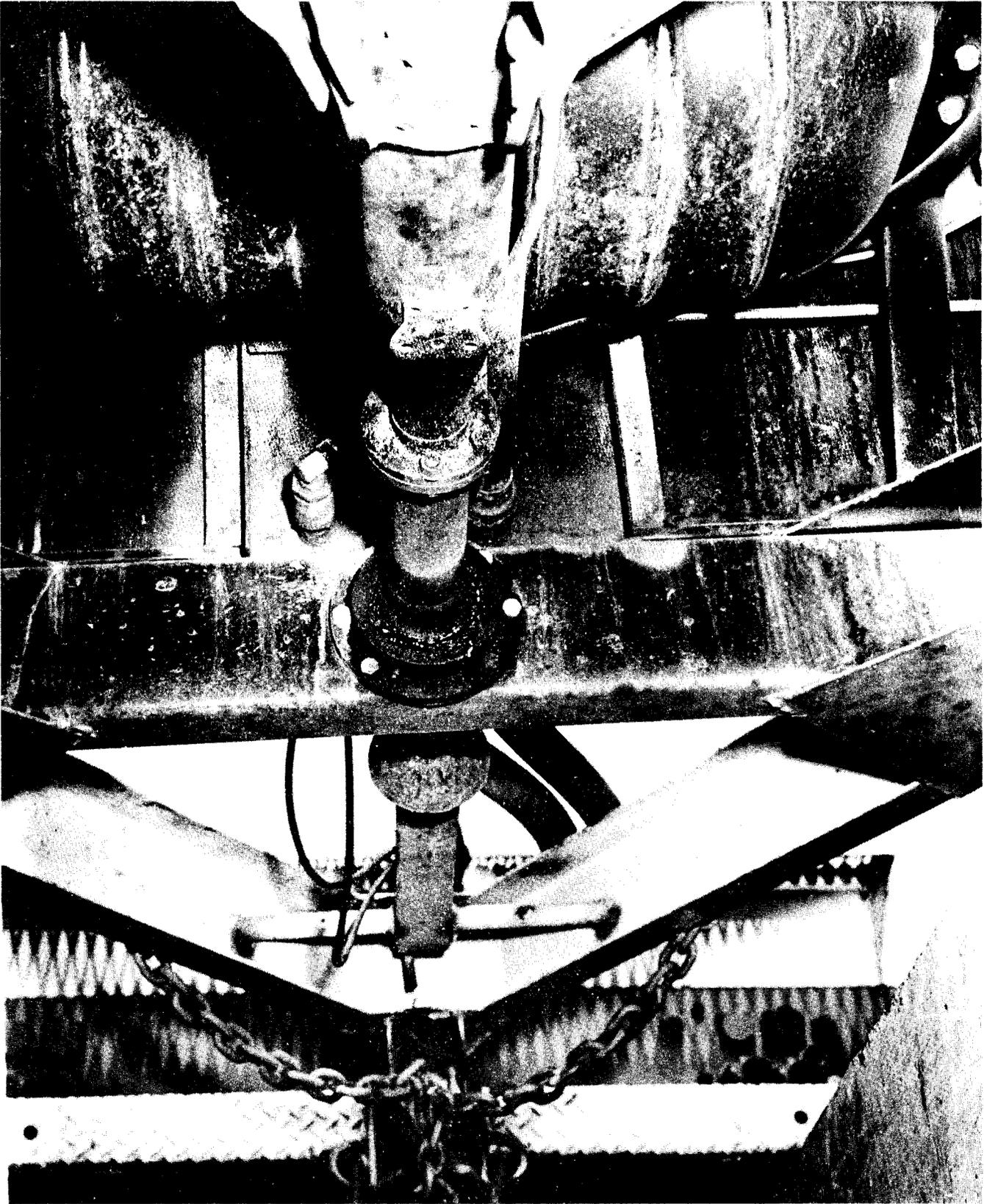
Figure 14.   Torque tube attached to trailer frame.

B. Brake Anchor Pins

The brake anchor pins were fabricated from Aircraft Quality AISI-4140 steel which had been oil quenched from a temperature of 1600°F and tempered at a temperature of 800°F. Figure 15 shows the dimensions of the anchor pins. Baldwin-Lima-Hamilton No. AB-11 strain gages were fastened to the flat sides of the anchor pins using BLH-EPY-500 high temperature epoxy. The gages on each pin were connected to form a complete Wheatstone bridge, the gages were waterproofed and the gages and connecting leads were coated with RTV silicone rubber to provide a degree of mechanical protection. Each Wheatstone bridge was connected to shielded, flexible, rubber covered cable which carried the electrical signals to the front of the trailer.

GAGE B ——
GAGE A ——

GAGE D ——
GAGE C ——

LEFT SIDE   RIGHT SIDE
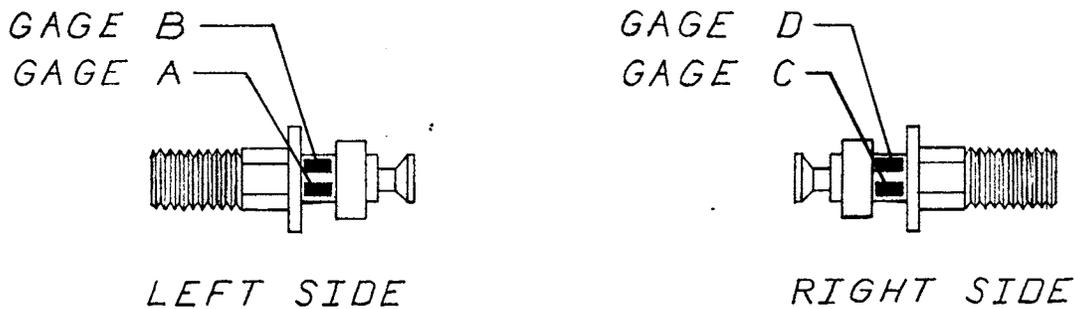
A AND B = TENSION GAGES
C AND D = COMPRESSION GAGES
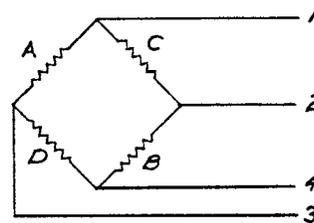
WHEATSTONE BRIDGE

RIGHT WHEEL STRAIN GAGE INSTALLATION

Figure 15. Instrumented brake anchor pin.

C. Brake Control System

The brake shoes are hydraulically activated by means of a modified Bendix Hydrovac unit which can, in turn, be driven by either the skid-resistance measurement system or by the vacuum developed by the towing vehicle engine intake manifold. The brake control system has been designed to permit individual control of each trailer wheel, to provide automatic override of the skid-measurement cycle under the control of the towing vehicle operator, and to provide automatic trailer brake application in the event of a towing hitch failure.

The active elements of the brake control system are shown in Figure 16. The major components are as follows:

(1) Check Valves **A**, **B**, and **C**:
Nupro No. B-6C4-1/3, 3/8", 1/3 psi.

(2) Truck Cab Brake Valve: Bendix Vacuum Brake Control
(removed from old skid vehicle).

(3) Vacuum Valve A: Hannifin No. CJ1-37, 3/8", three-way, 115 Volt, 60 Hz.

(4) Vacuum Valve B: Hannifin No. CJ1-37, 3/8", three-way, 12 Volt, DC (modified).

(5) Hydraulic Valves C and D: Skinner No. V51-DA1-150, 115 Volt, 60 Hz. (modified).

(6) Hydrovac: Bendix Vacuum Brake Booster modified to function as direct acting unit giving 1200 psi hydraulic pressure when subjected to 20" Hg. vacuum.

(7) Trailer Brake Master Cylinder: Bendix master cylinder which serves as brake fluid reservoir.

(8) Vacuum Accumulator Tank, 12 gallon capacity.

(9) Vacuum Relief Valve: Nupro No. B-6R-4M-10, 1/4", 10 psi.

(10) Vacuum Pump: Bell and Gossett, BLV, 1/6 hp, 115 volt, 60 Hz.

Figure 16 shows all of the valves in the non-energized position. In this condition, the entire system is under vacuum including chambers 1 and 2 of the Hydrovac when either the truck engine, the trailer vacuum pump or both are in operation. Both trailer wheels may roll free.

For a normal skid-resistance measurement, the trailer vacuum pump operates continuously. Electrical signals from the control panel cause hydraulic valves C or D to close in response to the operator's selection of the wheel path to be measured.
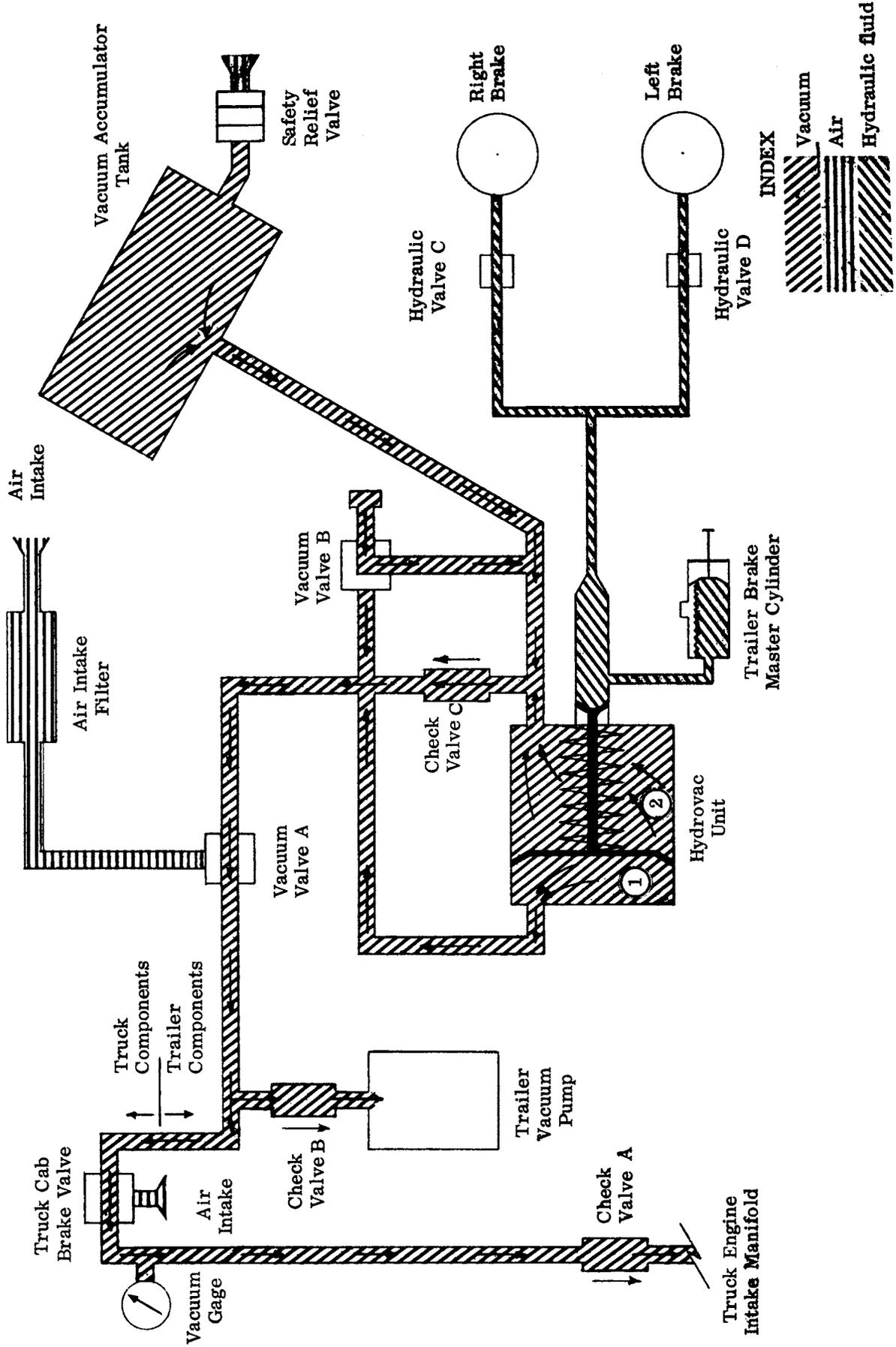
Figure 16. Trailer braking system.

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Vacuum
Air
Hydraulic fluid

Vacuum Accumulator Tank

Safety Relief Valve

Right Brake

Left Brake

Hydraulic Valve C

Hydraulic Valve D

Air Intake

Air Intake Filter

Vacuum Valve B

Vacuum Valve A

Check Valve C

Trailer Brake Master Cylinder

Hydrovac Unit

Truck Components
Trailer Components

Truck Cab Brake Valve

Air Intake

Check Valve B

Trailer Vacuum Pump

Check Valve A

Vacuum Gage

Truck Engine Intake Manifold

After closure of the hydraulic valve, vacuum valve **A** is energized. This disconnects the trailer vacuum system from the truck vacuum system, disconnects the trailer vacuum system from the trailer vacuum pump, and simultaneously admits air at atmospheric pressure to chamber No. 1 of the Hydrovac; air is prevented from entering chamber No. 2 of the Hydrovac by check valve **C**. The difference in air pressures in the two chambers of the Hydrovac causes the large piston to move rapidly to the right until the force developed on the large piston is exactly balanced by the force developed in the hydraulic cylinder. The initial application of hydraulic pressure causes the hydraulic valve, which has been closed electrically, to seal tightly and the entire pressure developed by the Hydrovac is applied to the selected wheel cylinder, which causes rapid application of the brake shoes in that wheel. At the termination of the skid-resistance measurement, vacuum valve **A** and the selected hydraulic valve are de-energized and both the truck vacuum system and the trailer vacuum pump start to remove the air that has been admitted to the system. Vacuum valve **B** is energized for about one second to permit rapid equalization of the pressure in both chambers of the Hydrovac and the return spring in chamber No. 2 rapidly moves the large piston to the left, which returns the hydraulic pressure to near zero and releases the trailer brakes.

Should the towing vehicle driver wish to apply the trailer brakes, he moves the truck cab brake valve. This disconnects the truck engine manifold from the brake system and admits a quantity of air, which is under the control of the operator, into the trailer brake system. A microswitch located in the truck cab brake control also turns the trailer vacuum pump off and returns vacuum valve **A** and the hydraulic valves to their non-energized position if they happen to be energized. The air admitted to the trailer brake system applies the trailer brakes in the manner described in the above paragraph. When the truck operator returns the truck cab brake valve to the off position, the trailer vacuum system is reconnected to the truck vacuum system, the trailer vacuum pump is re-started, and vacuum valve **B** is energized for about one second to permit rapid release of the trailer brakes.

In the event of failure of the trailer coupling and the vacuum line connecting the trailer to the truck is parted, air entering into the open portion of the trailer vacuum system will cause application of the trailer brakes as long as there is a reduced pressure in the vacuum accumulator tank. A signal light mounted on the instrument panel of the truck is illuminated whenever there is any hydraulic pressure in the trailer brake system. This warns the driver that the trailer brakes are on either intentionally or accidentally.

D. Watering System

The watering system developed for the new vehicle was designed to apply a water film in front of a pavement test tire which would conform to the recommendations of the ASTM tentative specification. This specification indicates that the water film should be wider than the test tire, that the water film thickness should be 0.02 ± 0.005 inch, that the water shall be applied to the pavement from 1 to 2 feet ahead of the test tire, and that water should not be splashed on the test tire. This water film thickness specification was assumed to mean that the water applied ahead of the test tire should be sufficient to form a film 0.05 inch thick on an ideal, plane surface. The watering system used with the VHRC, Model-2 vehicle was designed around this assumption and the observation that

a 6 inch wide band of water would exceed the footprint width of the ASTM standard pavement test tire by almost one inch. Since it was anticipated that skid-resistance measurements would not all be conducted at the same vehicle speed, it was decided that it would be most desirable if the water film thickness remained constant, without operator supervision, over a wide range of test speeds, but the system should not be so inflexible as to inhibit the ability to be quickly changed to permit modification of the rate of water application.

Water Nozzles

The water application nozzles used on the new vehicle were designed to apply a uniform film of water 6 inches wide by 0.02 inch thick for test speeds ranging from 10 to 70 mph. This corresponds to water efflux rates ranging from 5.5 to 38.5 gallons per minute for each nozzle. A 5/8 inch diameter orifice plate located in the distributor of each nozzle should be changed to 7/16 inch diameter for best results for water flows lower than 8 gallons per minute, and if this orifice is changed to 3/4 inch diameter, the water flow rate may be increased to 60 gallons per minute.

Each water nozzle is fabricated from a Sporlan Valve Co., type 1125, brass refrigeration distributor having twenty-four 1/4 inch diameter openings equally spaced around the periphery of the distributor; twenty-four 23 inch lengths of 1/4 inch copper tubing; and a 1/4 by 1½ by 8 inch piece of brass. Twenty-four holes 1/4 inch in diameter are drilled in the brass plate to form two rows of 12 holes each. The holes are spaced 1/2 inch apart along the rows, the rows are 3/8 inch apart, and the holes in one row are offset 1/4 inch from the holes in the second row. One length of copper tubing is soldered into each of the 1/4 inch holes in the distributor. The tubes are then formed so that two tubes emanating from holes spaced at 180° with respect to each other in the distributor lead to adjacent holes in the same row in the brass plate. The two tubes which emanate from the holes in the distributor which are at 90° with respect to the first two tubes are formed so that they lead to adjacent holes in the second row in the brass plate. This second pair of holes in the brass plate are displaced along the plate by 1/4 inch from the first pair. This process is continued around the distributor until all of the tubes have been formed, and are positioned in the brass plate. The last 5 inches of each tube is straightened, the top row of tubes is extended 1 inch beyond the brass plate and the bottom row is extended 1/2 inch, the tubes are soldered to the brass plate, and the final shape of the nozzle is determined by bending the collection of tubes all at one time.

This arrangement of holes and tube placement ensures that an equal distribution of water will occur if the distributor is disturbed slightly from the vertical position or if the measurement vehicle is used on a mildly curved section of pavement. The identical tube lengths ensure that the efflux velocities for the water coming from each tube will be identical, and the staggered arrangement of the tube ends ensures that the individual streams of water will not contact each other before they reach the surface of the pavement. Each of these conditions must be satisfied for a uniform water distribution.

The assembly of tubes, brass plate, and distributor are covered with three layers of fiberglass and epoxy to provide a degree of mechanical protection before the nozzle assembly is used on the test vehicle. Figure 17 shows a partially completed nozzle as seen from the efflux end and Figure 18 shows a completed nozzle mounted on the test vehicle.

Figure 17. Partially completed nozzle.
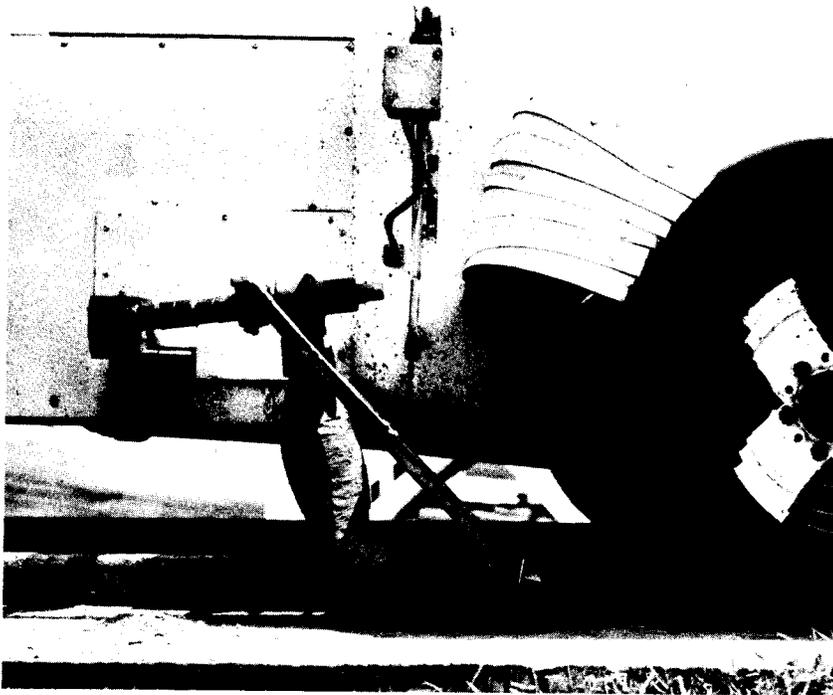
Figure 18. Nozzle mounted on test vehicle.

The water distribution nozzles are fastened to the rigid portion of the plumbing system by means of locally designed watertight rotary joints. The efflux ends of the nozzles are supported by electrically operated linear actuators which permit the nozzles to be remotely lowered to the proximity of the pavement surface when slipperiness measurements are in progress, but permit them to be raised to a height of about 10 inches for normal highway travel. A section of single roller machine chain about 4 inches long is included in the linkage supporting each nozzle. These flexible sections permit pavement surface obstructions to raise the nozzles without causing damage to the nozzles or to the supporting linkage.

Water Pumps

The water furnished to each nozzle is supplied by a separate Worthington Co., Type G5-A gear pump mounted on the towing vehicle under the water tank and between the main frame members. These pumps have been modified so that a Pitts Industries, No. 10332, 12 Volt, DC, electromagnetic clutch may be mounted on each unit. Torque is supplied to each clutch by means of a separate timing belt driven by a countershaft, which in turn is driven by a second timing belt assembly from the output shaft of the truck power takeoff unit. The gear ratio which must exist between the power takeoff unit and the gear pumps to provide the desired water flow at the test speed is established by an installation of belts and pulleys of various pitch diameters. The power takeoff output shaft on the VHRC, Model-2 towing vehicle, with the water tank half full, makes 3442.5 revolutions per mile, or turns at a rate of 3442.5 rpm at a speed of 60 mph. The measured displacement of the Worthington pumps is 33 gallons per minute at 20 psi and at a speed of 1800 rpm. A set of pulleys giving a power takeoff shaft to gear pump speed ratio of 11/21 is used to obtain a water film thickness of 0.02 inch with the new vehicle. Since there is a positive drive connection between the towing vehicle drive shaft and the gear pumps through the timing belts when the electromagnetic clutches are engaged, and since the gear pumps are positive displacement devices over a wide range of operating speeds, the resulting water flow through the wider distribution nozzles is directly proportional to the speed of the towing vehicle and the water film thickness is constant and independent of the speed of the vehicle.

The electromagnetic clutches are controlled by signals generated in the test sequence control panel and the pumps are operated only during the test cycle. The connections between the towing vehicle and the trailer are made through quick disconnect fittings fastened to the rear panel of the truck. Pressure relief valves set at 10 psi are included in series with each of the distribution nozzles to ensure that the connecting lines between the pumps and the nozzles stay full of water between skid-resistance measurements. These lines must be full to ensure that water flow from the nozzles begins as soon as the electromagnetic clutches are engaged.

Figure 19 shows the two gear pumps, the two electromagnetic clutches, the driving belts and pulleys, the power takeoff unit, the rear of the split shaft gear box, and a portion of the towing vehicle drive shaft as they appear from the ground. Figure 20 shows the trailer hitch and the connecting cables and hoses which run from the towing vehicle to the trailer.
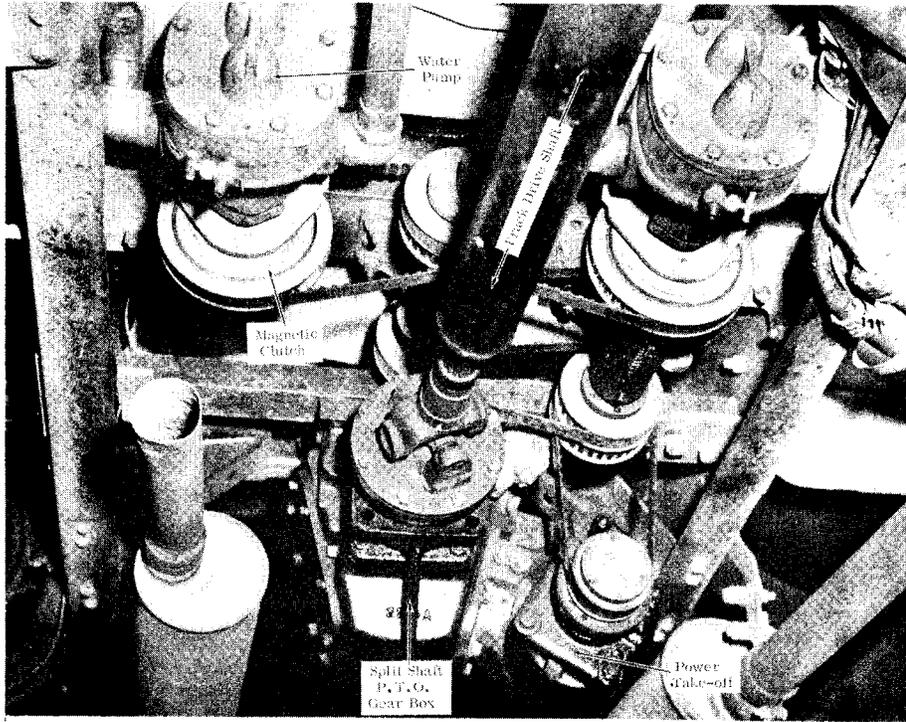
Figure 19. Water pump system.
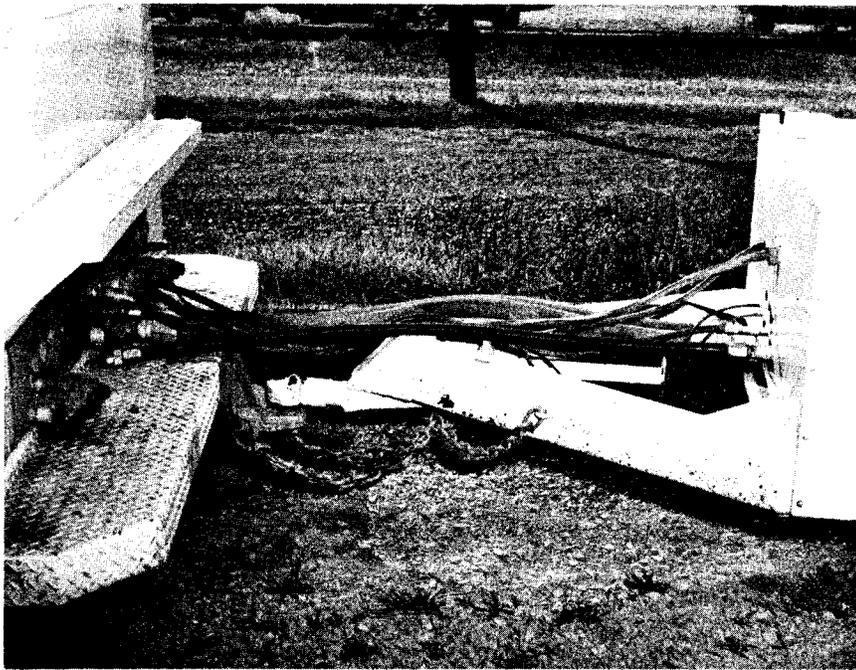
Figure 20. Trailer hitch.

E. Power Supply

Power for the operation of the measurement and recording instrumentation is supplied from a Kohler Model No. 1 5RM21, 1.5 kw, single phase, 115Volt, 60 Hz. electric plant that is mounted in the rear of the skid trailer. The generating plant may be started remotely from either operating position in the towing vehicle using the towing vehicle battery. Gasoline for operation of the generating set is obtained from the truck gas tank by means of an auxillary, electrically operated fuel pump.

F. Trailer Dimensions

The completed trailer has a center of trailer axle to hitch point distance of 119.75 inches, a hitch point height of 12.5 inches, a left wheel weight of 1060 pounds, a right wheel weight of 1020 pounds, a hitch point weight of 192 pounds, and the center of mass lies 4.75 inches above and 12.0 inches in front of the trailer axle.

CONTROL AND RECORDING SYSTEMS

The control and recording systems in the skid-resistance measurement vehicle were selected from considerations of reliability, ease of operation, and simplicity, in that order. The control system was designed to make use of a motor-driven mechanical timer working with standard relays and switches to provide automatic sequence control of the measurement cycle. The recording system was designed around a Brush Development Corporation, Mark 240, 4-channel, strip chart recorder incorporating peripheral equipment selected to make the recording unit self-contained with the exception of the necessary sensors. All components of both the control system and the recording system are located at the instrument operator's position in the towing vehicle except those which, by functional nature, must be mounted at the location of the controlled or measured phenomena. All of the electrical controls essential for the performance of normal skid-resistance measurements are duplicated in a control panel mounted just above the sun visor at the towing vehicle operator's position. A single operator, by making use of this panel, may conduct skid-resistance measurements unassisted.

A. Control System

Figure 21 shows a front view of the main control panel, and Figure 22 shows the companion panel that is mounted in the cab of the towing vehicle. The switches that appear on the main panel and their functions are as follows:

1. Transfer Switch: This switch enables the operator to elect to control the measurement cycle from either the towing vehicle operator's position or from the instrument operator's position. Parallel control of the measurement cycle is not permitted.

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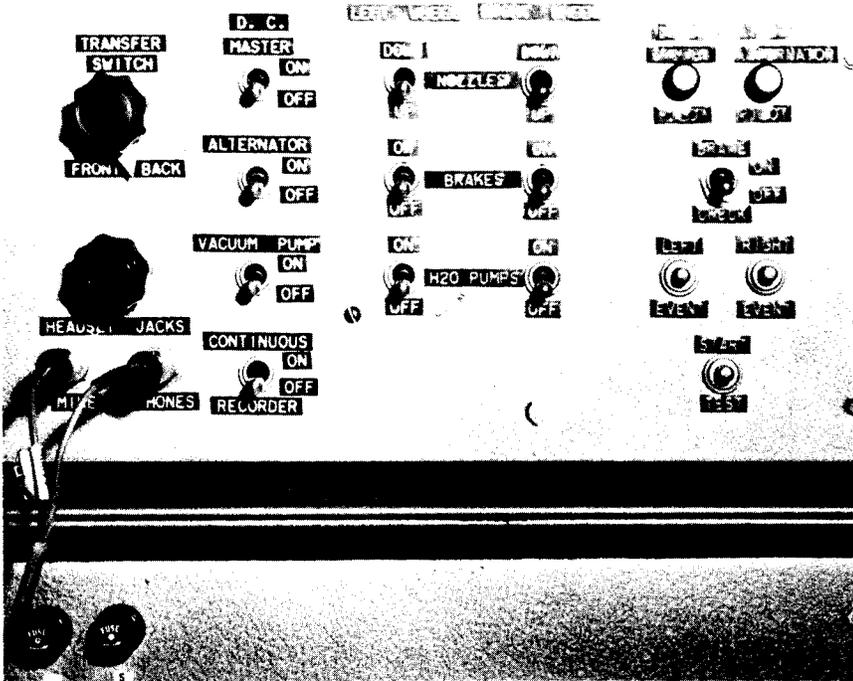
Figure 21. Main control panel.
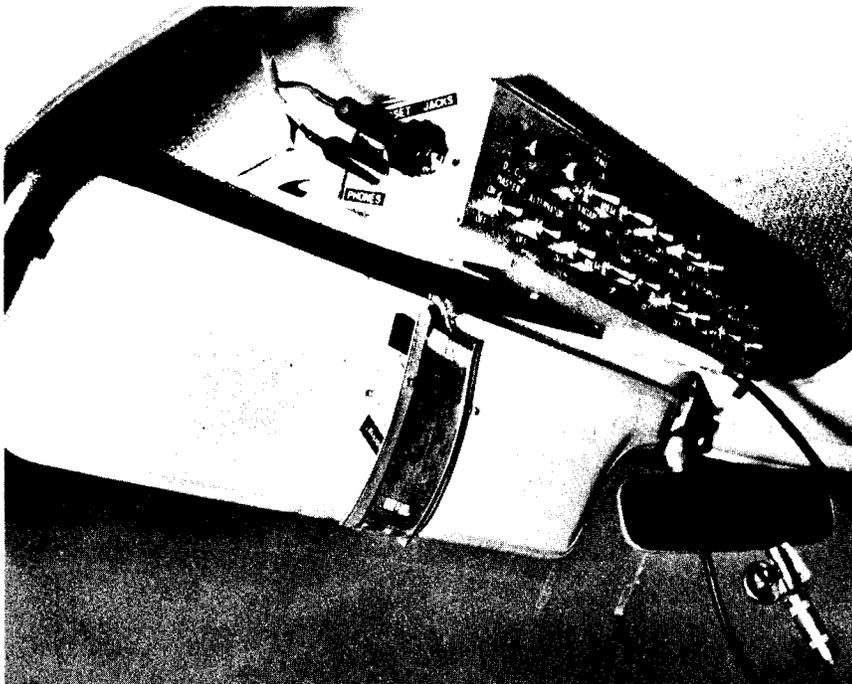
Figure 22. Vehicle driver's control panel.

2. DC Master: This switch connects the 12 Volt, DC operated systems in both the towing vehicle and the trailer to the towing vehicle battery. It also starts the pressure controlled electric fuel pump which supplies gasoline to the engine-driven generator mounted in the rear of the trailer.

3. Alternator: This switch permits the operator to start and stop the engine-driven generator from the active operating position.

4. Vacuum Pump: This switch energizes the vacuum pump mounted in the trailer.

5. Continuous Recorder: In the interest of saving chart paper, the recorder is connected so that it can be operated with all circuits active except for the chart drive motor and the recording pen drive motors. When the continuous recorder switch is in the off position, the chart drive and recording pen motors are controlled by the measurement sequence timer and these motors run for 4.5 seconds during each measurement cycle. When the continuous recorder switch is in the on position, the recorder operates continuously.

6. Nozzles, Left and Right: These switches permit the operator to raise and lower the water distribution nozzles independently. Limit switches located in the linear actuators mounted on the trailer automatically position the nozzles in the extreme up or down position depending on the position of the panel mounted switches.

7. Brakes, Left and Right: These switches complete circuits to hydraulic valves C and D located in the trailer so that, at the proper place in the skid-measurement sequence, as determined by the sequence timer, the appropriate hydraulic valve will be closed to achieve a single-wheel measurement. The brake system has been designed so that both trailer wheels will respond to brake application in the absence of electrical activation of the hydraulic valves.

8. Water Pumps, Left and Right: These switches complete circuits that permit the sequence timer to energize the electromagnetic clutches that are mounted on the water pumps at the proper time in the measurement sequence.

9. Brake Check: This switch, which is spring loaded to return to the off position, permits the operator to momentarily apply brakes to either or both trailer wheels as selected by the brake switch without having to go through the complete skid-measurement sequence.

10. Event Markers, Left and Right: The Brush Instrument Corporation recorder has been equipped with event marker pens which place identifying marks on the right and left edges of the chart paper. These push-button switches control the two event marker pen motors.

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11. Start Test: This push-button switch initiates the start of a skid-measurement sequence. If the button is pushed and held down for 0.25 second, the sequence timer will carry out the test sequence using the elements selected by the positions of the various switches on the control panel. A normal measurement sequence lasts for 6 seconds. Figure 23 illustrates the normal skid-resistance measurement sequence. If the test start push-button is held down, the test sequence is arrested 3.0 seconds after the initiation of the test and the skid-resistance measurement will continue until 0.75 second, following the release of the test start button. The measurement will then be terminated following the normal cycle sequence. A second test start push-button has been provided at the end of a 15 foot rubber covered cord to permit the instrument operator to initiate measurements from locations remote from the normal operator's position.
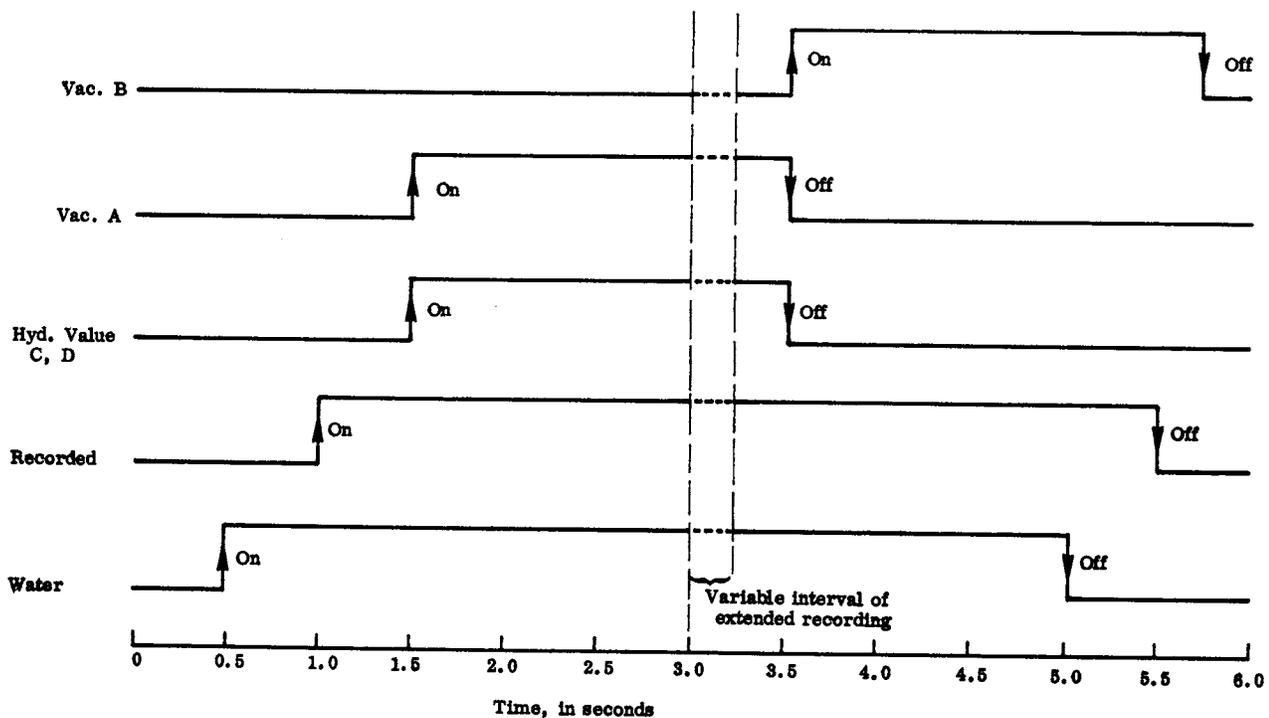
Figure 23. Test sequence.

The other items that appear in the photograph of the main control panel are pilot lights to indicate the status of the AC and DC power sources, headset and microphone jacks for an intercom set, a volume control for the intercom, and fuses for the protection of the electrical circuits.

The heart of the measurement sequence control system is an ATC mechanical timer that was salvaged from the VHRC, Model 1, vehicle. This unit is a 6-cam, motor driven, repeating timer that makes one cycle each 6 seconds. The cams are

adjustable as to opening and closing times, and each cam operates a single SPDT microswitch. The timer has been modified so that it executes a single 6-second sequence each time that it is started. The cams have been adjusted to execute the program shown in Figure 23.

<u>B. Recording System</u>

The recording system is a Brush Instruments Corporation, Mark 240, 4-channel strip chart recorder. The recorder has been equipped with two Brush No. RB-4212-00 carrier type preamplifiers, two Brush No. 13-4211-20 DC preamplifiers, two Brush No. 15-5221-11 dual pen motor driver amplifiers, and two event markers. The recorder and the amplifiers are capable of following sinusoidal signals ranging from 0 to 55 Hz. The two carrier type preamplifiers provide the excitation voltage for the two strain gage bridges that are mounted on the brake anchor pins. They contain bridge balance controls, sensitivity adjustment controls, they provide electrical signals for verifying system calibration, and they supply electrical signals to the pen motor driver amplifiers that are proportional to the strain developed in the brake anchor pins. This strain results from the torque applied to the brake shoes when the locked wheel is caused to slide over the pavement surface. Essential details of the measurement circuit are shown in Figure 24. Both the left and right wheel measuring circuits are identical and interchangeable, which arrangement can sometimes be of assistance in the temporary correction of equipment failure when the vehicle is away from its garage area.
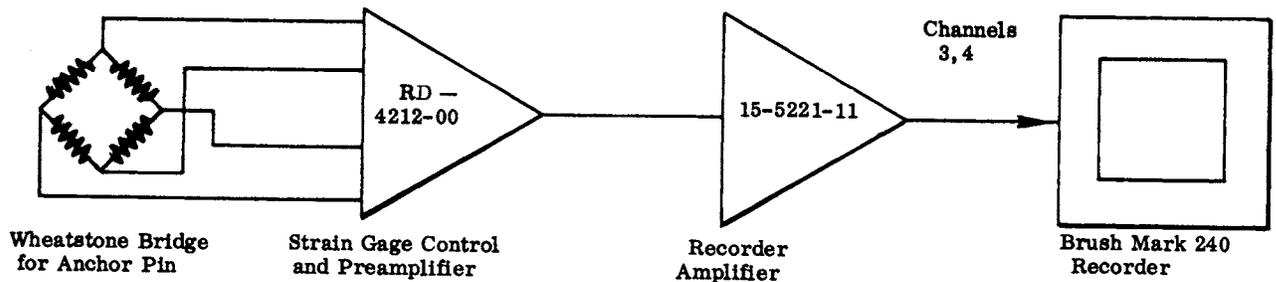
Figure 24. Skid-force recording system.

Vehicle speed is recorded simultaneously with skid-resistance by one channel of the Brush recorder. An electrical signal that is proportional to the speed of the vehicle is generated by a Weston Instruments Company, Model 750, DC, tachometer generator mounted on a tachometer pad that was purchased as a part of the split shaft gear box. This electrical signal is processed by one of the Brush DC preamplifiers and is recorded on the strip chart. Operation of the speed recording system as it was originally installed revealed that the tachometer generator and recorder combination were capable of responding to the small angular accelerations that are inherent in most universal joint drive shaft systems and that the voltage excursions caused by these vibrations made the task of reading the strip chart somewhat difficult.

A small low-pass filter installed between the tachometer generator and the input to the DC preamplifier eliminated this difficulty. Figure 25 gives the functional details of the speed recording system. Figure 26 shows the strip chart recorder mounted in the operating position in the skid-resistance measurement vehicle, and Figure 27 shows a typical skid-resistance measurement.
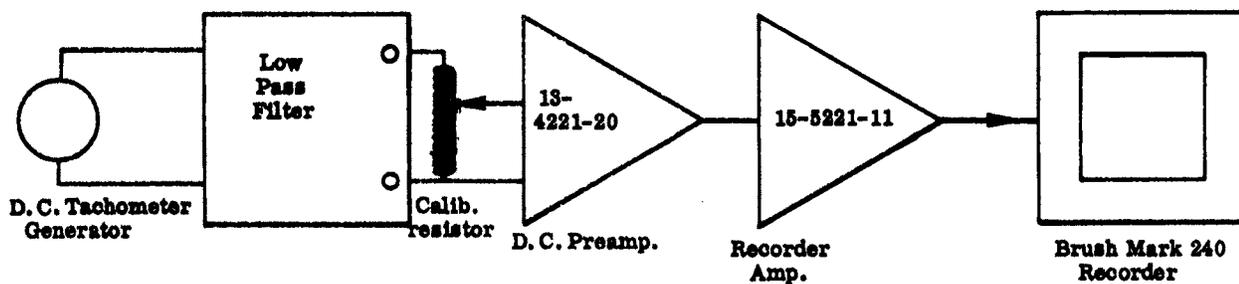
Figure 25. Speed recording system.

Figure 26. Strip chart recorder.
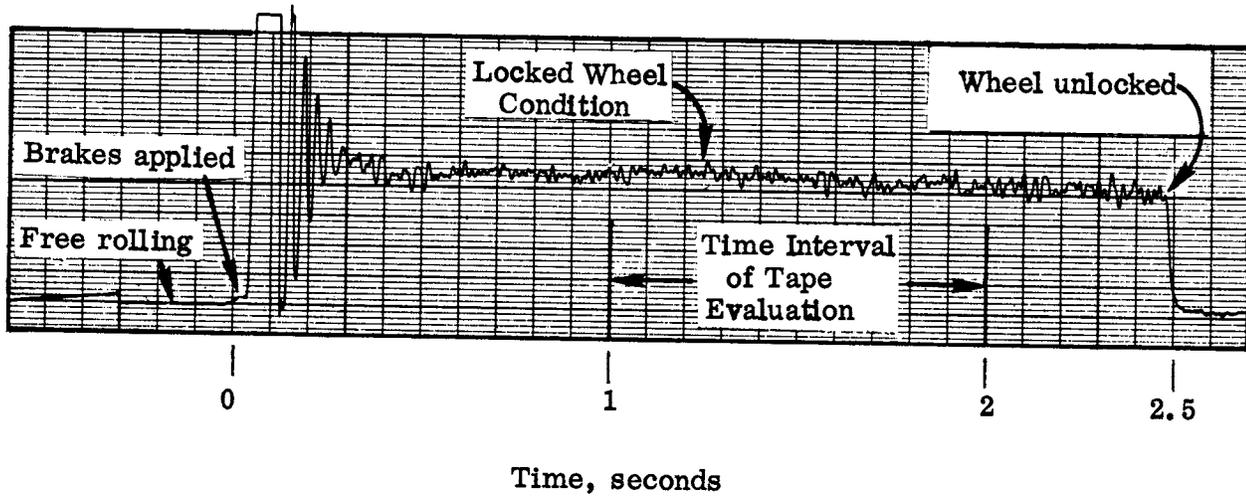
Figure 27. Typical skid-resistance measurement record.

CONCLUSIONS AND RECOMMENDATIONS

A. Conclusions

The Virginia Highway Research Council, Model-2, skid-resistance measurement vehicle has now been in use for a number of months and, for the most part, the performance of the vehicle has been as anticipated. There have been some maintenance difficulties but these have been, for the most part, the result of failures involving commercially available components. Highway experience with the vehicle has indicated that the proposed addition of a digital readout system must be carried to a satisfactory conclusion and that both the cab and the operator's space in the towing vehicle should be air conditioned in order to furnish a more satisfactory environment for the operation of the electronic equipment, and to provide a more comfortable atmosphere for the equipment operators.

The skid-resistance measurements made with the new vehicle still do not correlate as closely as one would wish with data obtained on the same surfaces using the Virginia stopping-distance car. One observation which may be of significance in this problem is that high speed motion pictures made of the skidding wheel of the skid-resistance measurement vehicle and of a skidding wheel on the stopping-distance car on the same piece of pavement while the vehicles were traveling at approximately the same speed show that the skid-resistance measurement vehicle wheel slides smoothly without bouncing, while the wheel on the stopping-distance car may execute severe vertical excursions and, at times, may completely loose contact with the pavement surface.

B. Recommendations

In view of the above observations, it is strongly recommended that a detailed study be made of the performance of several types of skid-resistance measurement vehicles with the study being designed to provide an understanding of the sources of the phenomena which lead to this lack of correlation.

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