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U. S. ARMY
TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

TRECOM Technical Report 63-29

DEVELOPMENT TESTING OF A TEST-BED RIG ON AIR-SUPPORTED TREADS

FINAL REPORT

Task 1D543006D40406
(Formerly Task 9R47-16-010-06)
Contract DA 44-177-TC-600(T)

June 1963

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prepared by:

WILSON, NUTTALL, RAIMOND
Engineers, Incorporated
Chestertown, Maryland

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HEADQUARTERS
U. S. ARMY TRANSPORTATION RESEARCH COMMAND
Fort Eustis, Virginia

The U. S. Army Transportation Research Command concurs in the conclusions and recommendations expressed by the contractor. Implementation will be subject to availability of funds and appropriate authorization.

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CONTENTS

LIST OF SYMBOLS

AHP	=	adiabatic air horsepower
BHP	=	horsepower input to blower drive
CFM	=	air flow, cubic feet of free air per minute
h	=	manometer reading, in H_2O
ℓ	=	total effective seal length, in.
L	=	length of tread on the ground
m	=	number of treads
n	=	compression exponent
N	=	blower drive input, rpm
p_m	=	manifold pressure, psig
p_n	=	nominal unit ground pressure, psi
p_p	=	pool pressure, psig
p_s	=	stagnation pressure, lb/ft^2
r	=	blower drive ratio
R	=	pressure ratio
T_1	=	inlet temperature, °R
ΔT_a	=	adiabatic temperature rise, °F
ΔT_c	=	calculated polytropic temperature rise, °F
ΔT_m	=	measured temperature rise, °F
v	=	air velocity, fps
V	=	blower impeller tip speed, fps
w	=	tread width, in.
W	=	gross vehicle weight, lb
η	=	adiabatic efficiency
ρ	=	air density, lb sec^2/ft^4

SUMMARY

A test bed for the preliminary evaluation of air-supported treads as possible running gear on off-road transport equipment was constructed earlier in this program. The present report first outlines a modest program of modifications intended to improve performance and/or durability of the original treads, and presents the results of subsequent, brief tests.

The changes resulted in reduced internal resistance, reduced air consumption, improved sheave traction, and extended durability. Considerable practical design experience was gained, and data necessary for estimating air consumption of similar devices of different weights and tread dimensions were developed.

On the basis of this experience and data, a preliminary design for a second-generation, air-supported-tread test vehicle was developed, and is presented herein. This unit is 20 feet 2 inches long and 68 inches wide, is bellyless, steers by articulation, and has a designed net payload of 2500 pounds. It is estimated that its full-load nominal unit ground pressure would be 1.3 psi. The unit would require approximately 700 cfm of air at 1.65psig for full air support. Blowers to supply this would draw some 18 horsepower from the main propulsion engine. Blowers are mounted in the track envelope and are driven at constant speed by a hydrostatic drive system.

It is concluded that the original test rig has served its purpose in demonstrating that the air-supported-tread concept is both feasible and promising, and in providing data for the proper design of a second-generation machine. Such a machine in the 1-1/4-ton net payload range could now be built which would exhibit good performance and durability, and would serve for further study of the merits and limitations of the concept.

It is recommended that the development be continued by the construction and testing of such a second-generation machine.

CONCLUSIONS

1. The concept of air-supported treads for possible use on off-road transport equipment has been shown to be both feasible and promising.

2. Further developments and testing are required to optimize the design of this type of device, particularly in the areas of

   a. side seals
   b. tread guiding.

3. With proper application and detailed design based upon the results of the present performance alone, however, machines of good performance and durability already appear to be entirely feasible.

4. The original test bed is not suitable for further development work.

5. At the present, a near-optimum application of the concept would be to a second-generation, articulated, 1-1/4-ton-net-payload machine, which would immediately serve to demonstrate the concept favorably and which would also be useful as a test bed for further developments and evaluations of the concept which are required.

RECOMMENDATION

Study of the air-supported-tread concept should be continued by the construction, testing, and further development of a second-generation machine in the 1-1/4-ton-payload weight class.

INTRODUCTION

BACKGROUND

In the summer of 1959, WNRE undertook the study, test, and evaluation of the concept of air-lubricated treads through the design, construction, and test of a test-bed rig. After a number of modifications in project scope and research plan details, and after a substantial amount of pioneering work, particularly with new materials, the test-bed rig was completed in March of 1962 (Figure 1). Despite

Figure 1. Configuration of the test rig at the beginning of the development program. Rig was constructed on rear of WW II half-track chassis, and was designed to form the rear unit of a four-tread articulated test bed, if the design proved successful. The expedient air supply mounted on the deck uses an aircraft engine supercharger, driven by a V-8 engine.

excessive air consumption, some sheave slippage within the treads, and a fairly high apparent internal rolling resistance, the first rig was operable after only a few minor changes and showed some promise in brief, preliminary runs. No serious testing was possible at that time, however, due to the exhaustion of contract funds and time (Reference 1).

A modest supplement of funds and time was subsequently allocated so that limited test and evaluation of the rig could be carried out, in view of the promise shown by it on preliminary runs and of the effort already expended in the program. This further work was begun in July 1962.

OBJECT

The object of the original project, of which this is the final phase report, was to design, construct, and evaluate a test-bed rig running on a system of air-supported treads, in order to assess the practical feasibility of such running gear for off-road transport equipment. The specific objects of the concluding development and test program detailed hereinafter were:

1. To obtain air consumption measurements on the test-bed rig under various types of operation, before and after some relatively simple mechanical changes;
2. To reduce its internal rolling resistance as much as possible by simple means;
3. To reduce sheave slippage within the tread belts;
4. To improve air sealing by means of minor mechanical changes;
5. To replace the side skids and the track guiding system originally installed on the test-bed rig with more durable ones; and
6. To assess durability and performance of the complete air-supported-tread system as modified.

THE FINAL PHASE

GENERAL

Both the general and the detailed test objects were achieved in this final phase of the project by making a number of modifications to the original rig, conducting operational and quantitative tests of it, and subsequently making teardown inspections. A number of modification-test-inspection cycles were originally planned; but the success of the first modifications, together with the nature of further changes indicated as desirable by the test results, led to the conclusion that further work with the original test-bed rig would be unprofitable.

In November 1962, the project was redirected towards further evaluation of the concept through the application of project data and experience to the preliminary design of a second-generation test rig.

ALTERATIONS IN TEST-BED RIG

The proposed alterations to be made prior to the first series of serious tests were outlined in the Phase II report under this contract (Reference 1). A sketch from that report is reproduced here, in slightly modified form, illustrating the changes which were actually carried out (Figure 2).

Inner Belt Skids

New inner belt skids were constructed by laminating recessed Teflon strips with natural rubber stock sandwiched into the recesses. (See sections, Figures 3 and 4.) This assembly was hot bonded and vulcanized under pressure into a leaf-spring-like structure. Bolts were also molded into the skids to provide means of attachment to the inner belt. These skids are relatively flexible in the vertical plane for obstacle performance, and relatively rigid laterally for guiding. (See Figure 5.) They present a low friction surface on all working faces. The construction was far more durable than the original, which was the chief object in making this change. These skids are 1-1/4 inches thick (1/4 inch higher than the originals) in order to provide more clearance for the track belt ribs (Figure 6).

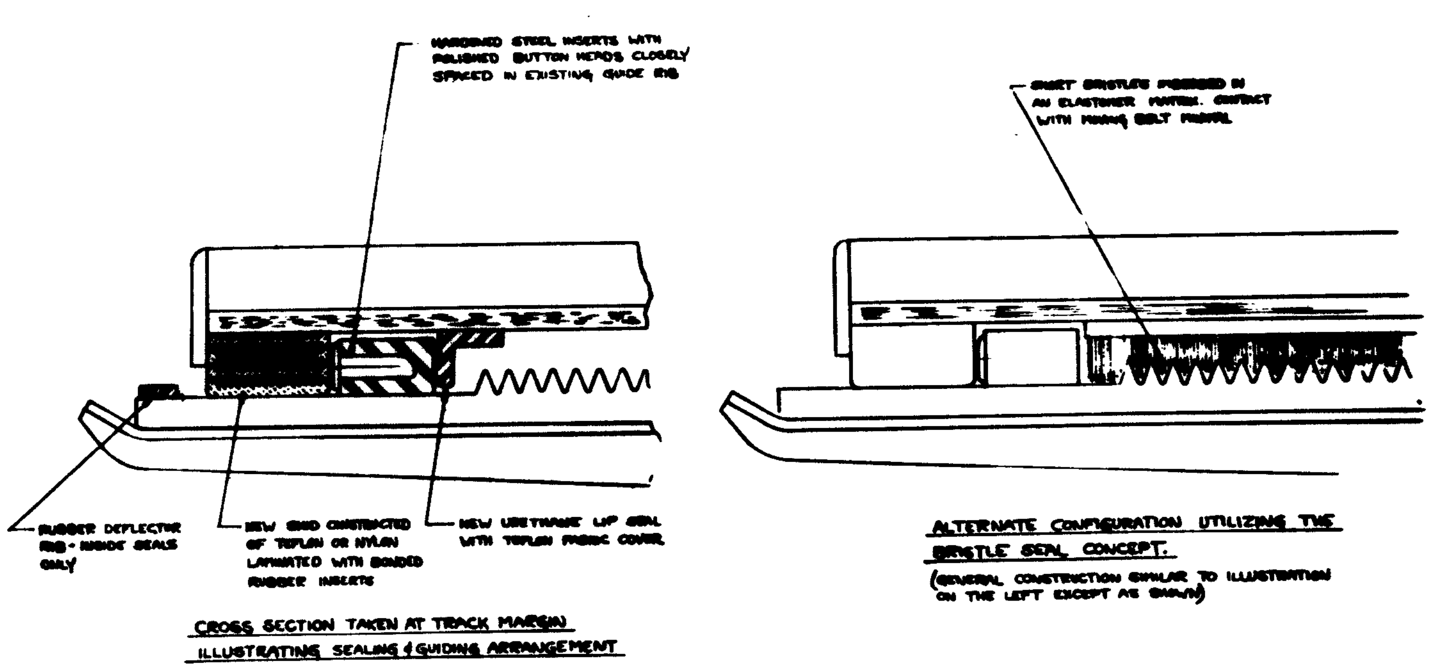
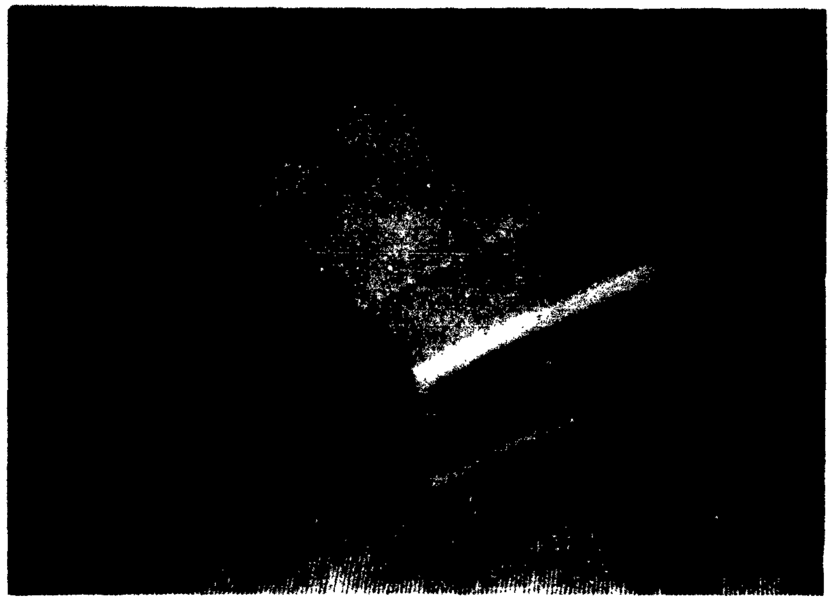
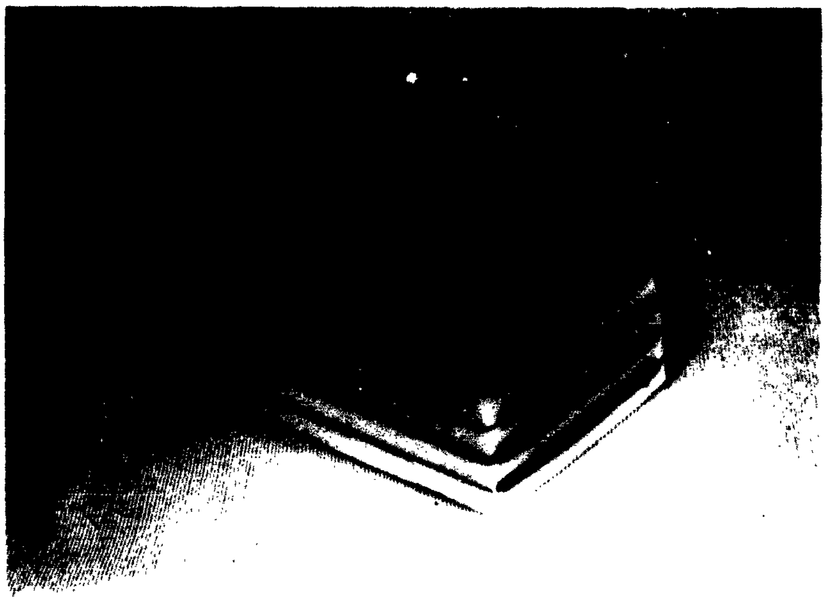


Figure 2. Track Seal and Guide Rib Configurations Used.





Figures 3 and 4. Section of new guide/seal rib made by laminating Teflon strips with natural rubber. Three surfaces were essentially all Teflon. The resulting structure was relatively rigid in the plane parallel to the laminations, flexible along its length in the plane normal to the first.

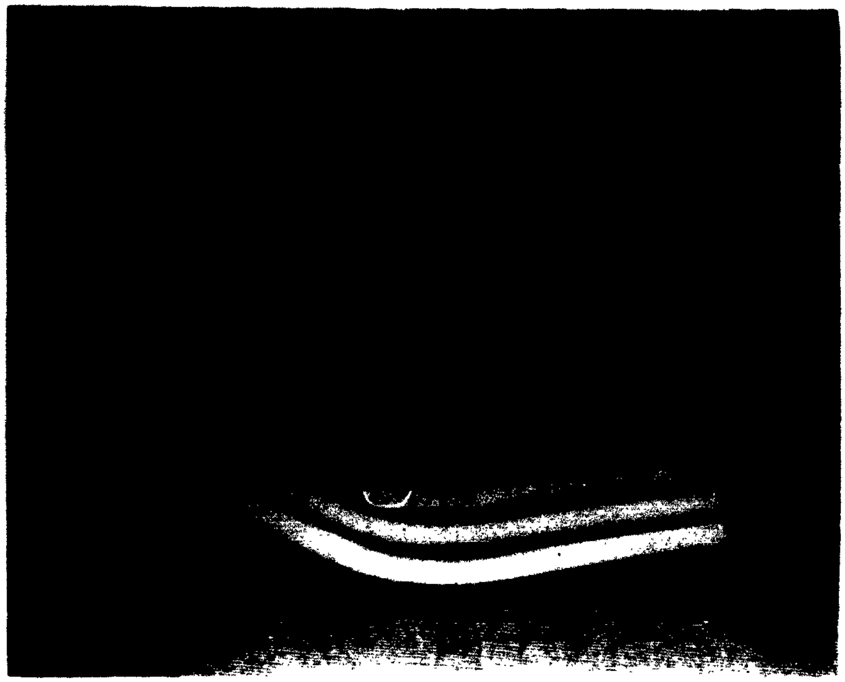


Figure 5. Lamination imparted flexibility in the desired plane by permitting large, nondestructive, horizontal shear deformation in the rubber layers.

On the right inner belt, the original lip seals were replaced by softer urethane lip seals which had Teflon fabric covers. Better sealing with less friction was the aim here.

On the left inner belt, the lip seals were dispensed with altogether. In their place, brush seals, of the type used successfully at the ends of the air pools, were applied as shown in the sketch (Figure 2; also Figure 7). The two different seal systems on the rig were intended to offer a direct comparison of the relative efficacy of the lip seals and brush seals.

Driven Tread

In order to achieve a more durable guiding arrangement with less friction, hardened-steel wear inserts were installed in the existing rubber guide ribs of the treads on the faces which would contact the new laminated Teflon guide ribs of the inner belt (Figures 2 and 8). It was necessary to cut the original rubber rib back approximately 1/4 inch in order to

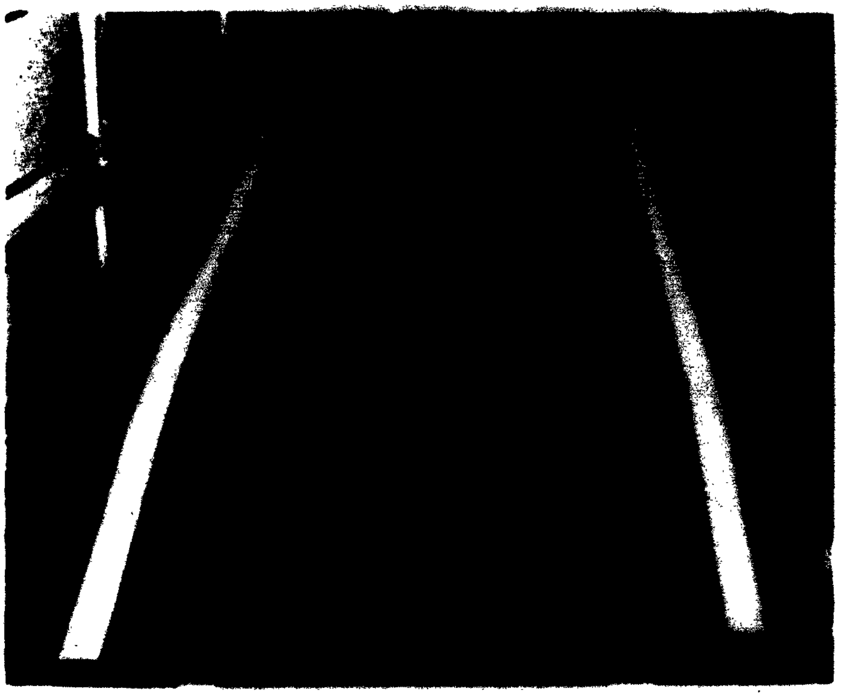


Figure 6. Air pool on right side used a barrier/lip seal at each side to restrict airflow. Air entered the pool through a jet at each end, forming a jet curtain to restrict flow out the ends of the pool. The curtain jet was backed up by a brush seal. The guide/seal rib on each side of the tread runs between the lip seal and the Teflon inner belt rib. The latter was increased in height by 1/4 inch in order to reduce the chance of contact between the inner belt and the tread. Some evidence of earlier contacts is apparent.

provide clearance for the button heads. Some sealing effectiveness in this region was undoubtedly lost in the process.

Dampers in Ducts

Butterfly dampers were installed in each of the air ducts leading to the ends of the air pools (four dampers in all) so that flow could be balanced for the various tests (Figure 9).

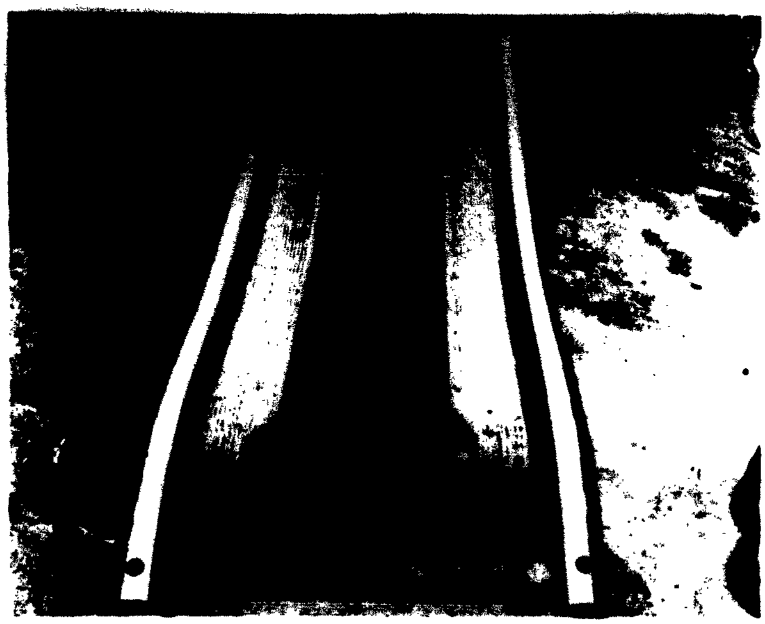


Figure 7. Inner belt on left side utilized brush seals at the sides of the air pool as well as at the ends.

Figure 8. Hardened-steel "buttons" were installed in the tread guide rib to limit guiding contact to steel-on-Teflon. Some sealing effectiveness was undoubtedly sacrificed.

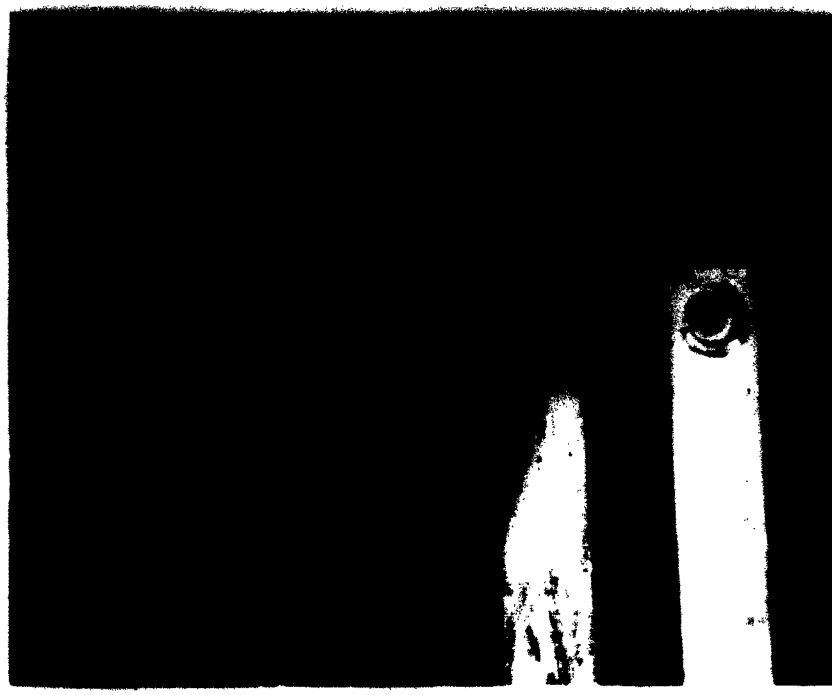


Figure 9. It was anticipated that some flow imbalance between the right and left side might result from the use of different side seals. Control dampers were accordingly installed in all ducts to permit flow adjustment if needed.

Vents in Drive Sheave Housings

Adjustable vents were installed on the drive sheave housings so that air pressure in the housings could be dropped to any desired level. It was thought at the time that air lubrication between the sheave and the track belt in the traction area might be responsible for a substantial amount of the belt slippage (Figure 10).

INSTRUMENTATION

In order to determine air flow rates and losses as well as power consumption, a system of instrumentation was installed on the test-bed rig as follows:

1. Air flow: A venturi sleeve having a Pitot tube and static pressure holes was placed in each of the four air ducts from the blower manifold to the air inlet

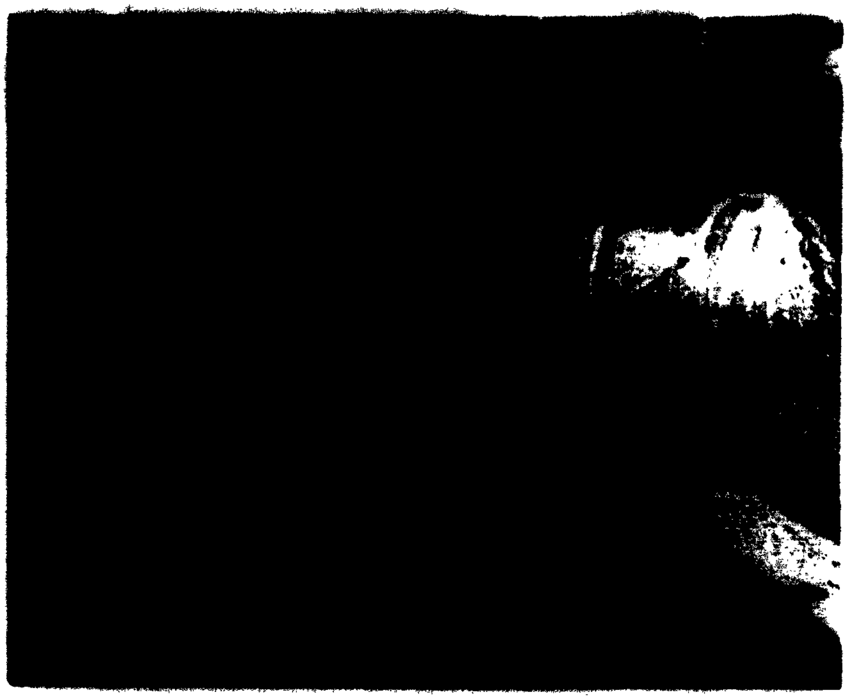


Figure 10. Vents were installed in sheave housings to insure that air pressure did not, in effect, lubricate the driving sheave-belt contacts.

jets at each end of each tread air pool (Figures 11, 12, and 13). The pressures were read by a system of four water and four mercury columns arranged to read air flow head directly in inches of water and static pressure in the duct in inches of mercury in all four ducts simultaneously at a single location (Figure 14). A vacuum gage was installed in the blower inlet to monitor any substantial drop in pressure at this point.

2. <u>Air temperature</u>: Two remote-reading bulb thermometers were installed to measure air temperature rise created by the blower. One bulb was installed at the blower inlet; the other, in one blower discharge duct near the outlet manifold.

3. <u>Blower operation</u>: A tachometer was installed to read transmission output speed (= blower input speed). (Due to the automatic transmission on the engine, blower impeller speed could not be determined simply from engine speed.)

Figure 11. Venturi sleeve with Pitot tube and static pressure tap. One such sleeve was installed in each duct (four in all, one to each end of each air pool).

4. Engine operation: A second tachometer was installed to read engine speed, and a vacuum gage was installed to read engine intake manifold vacuum.

5. Rig speed: The original half-track speedometer (driven from the front wheels) was fitted with 2:1 step-up gears in the drive to increase low speed sensitivity. The revised system was recalibrated to correct for this step-up and also for the oversize (12.00x20) front tires fitted.

All gages except the speedometer were grouped on a single panel (Figure 14). In view of the number of readings to be taken, records were taken by photographing the board when steady-state conditions were thought to have been obtained. Readings and other on-the-spot notes and observations were keyed through photograph numbers, and the whole was consolidated on data sheets after the photos were read (Data Sheets 1-3, Appendix III).

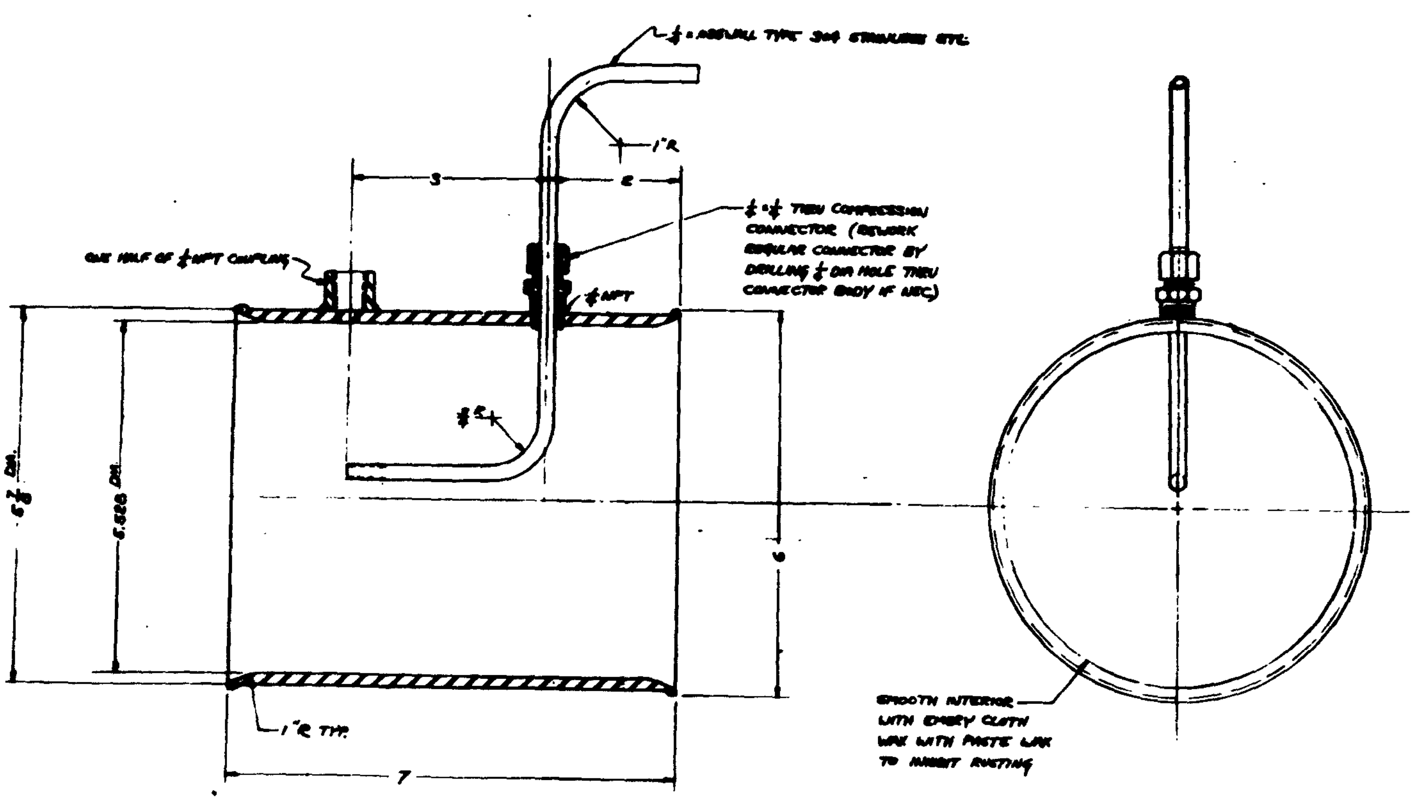


Figure 12. Details of Venturi Installed in Each Duct.

Figure 13. Two ducts from blower to rear of air pools and one from the blower to the front of the left air pool are visible here, with the instrumented sections installed. Also visible is the bulb-type outlet temperature pickup mounted in the left rear duct.

TEST PROGRAM

A simple test program was organized to accomplish the following objectives:

1. To record completely all operating data of interest;
2. To provide a general shakedown run, in order to gain some general operating experience and to take movies;
3. To weigh the entire vehicle and to obtain the load distribution between the treads and the front tires;
4. To check blower performance against data available on its performance in conjunction with an aircraft engine, as a supercharger, in order to determine the extent to which these data were valid (and hence usable) for the present system;
5. To obtain some operating data of direct use in the design of a similar but much lighter vehicle by

lightening the load carried on tracks during tests;

6. To obtain rolling resistance data at various track loadings;
7. To obtain drawbar pull at normal weight;
8. To run the vehicle for some miles in order to obtain some indication of wear points, primary durability, and mechanical adequacy of the system and its parts; and finally
9. To dismantle and inspect the track assemblies for wear and damage.

This program was completed on 2 November 1962.

Figure 14. Central gage panel showed all readings except speedometer. Across the top of the panel are twelve 0-5 psi pressure gages connected to various taps in the tread air system. On the lower left are four pairs of water and mercury manometers, each set connected to one of the instrumented duct sections. The water column gave static head; the mercury column, dynamic head. On the right are temperature readings across the blower, blower engine manifold pressure (vacuum), blower drive and engine tachometers, and a vacuum gage at the blower inlet.

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TEST RESULTS

GENERAL

The test rig was run on five days between 15 October 1962 and 2 November 1962, under a variety of speed and loading conditions, accumulating a total of approximately 20 miles. Several sequences of static testing were included. A compilation of quantitative test data on air consumption, taken from photographs and notes (see earlier), is given on Data Sheets 1-3, Appendix III. Test numbers are keyed to photo numbers. Some of the photos taken were not of the instrument panel, leaving the apparent gaps in recorded test numbers.

OVERALL PERFORMANCE

Except for the failure of the new, softer lip seals on the right side tread, there were no outright failures of parts or materials during the present series of tests. While the rig still has no "durability," in comparison with existing, developed equipment, it can now be run for reasonable distances, under reasonable load and speed conditions, for test and demonstration purposes without excessive "babying" and with a reasonable assurance that it will return to the shop in one piece.

The rig moves much more easily since the modifications than it did earlier. However, its behavior when shifting gears is still that of a "stiff" vehicle (such as a tank), as is confirmed by the towing resistance measurements.

Its "ride," as before, is good, free of track-induced vibrations and of all influence of minor terrain roughness. It does not absorb larger obstacles (over 3-4 inches high) as well as it might, due perhaps to inner and tread-belt stiffness and/or too close control by the foamed urethane bump stops incorporated within the suspension air bags.

RIG WEIGHT

The rig in its normal condition at the beginning of this test series (Figure 1) was transported to a public scale in Chestertown, Maryland, and carefully weighed. The results were as follows:

Front axle	5,200	lb.
Rear treads	12,200	lb.
Total	17,400	lb.

Inasmuch as the nominal effective air-pool area supporting each tread is approximately 2000 square inches, this indicated that a pool pressure of approximately 3.1 psi would be necessary to completely "float" the rig on the air pools. Accordingly, means were immediately sought to reduce tread loadings readily, and the front boom arrangement shown in Figure 15 was devised. By placing 1000 pounds of ballast on the boom end at the front, loading on the tread is reduced a like amount (and front axle loading is increased by 2000 pounds). Although somewhat cumbersome, this rig was used to run tests at tread loadings of 10,000 and 9,000 pounds to supplement the data at the 12,000-pound loading (discussed later).

Figure 15. In order to reduce the tread loadings for test purposes, the boom arrangement shown was installed. Carriage of 1000 pounds of steel-plate ballast forward reduces tread loading a like amount (and increases front axle loading by 2000 pounds).

GENERAL OBSERVATIONS ON THE AIR MEASUREMENTS

The air supply, as recounted in a previous report (Reference 1), was made up from a surplus Curtiss-Wright aircraft engine supercharger, driven by a 354-cubic-inch Chrysler industrial engine through a Torqueflite automotive automatic torque-converter transmission. The system was put together from available, low-cost bits and pieces after the original 40-horsepower system (having a capacity of only 600 cfm) proved, in first trials in the spring of 1962, to be inadequate. Characteristic curves for the Curtiss-Wright blower when working with an aircraft engine (in which combination flow was governed by virtue of the engine displacement and the mechanical coupling between engine and blower) were available. These gave air flow as a function of pressure rise and blower rpm.

Air consumption in the present tests was determined by means of four Pitot tube readings, one taken in each of four ducts between the blower discharge manifold and the air inlet jets at each end of each tread air pool. Calculations* were made on the crude assumption that the velocity profile at

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*The method used to calculate air flow from the Pitot tube readings was essentially to calculate air velocity from the manometer and temperature readings, and to multiply the result by the duct throat area. For the narrow range of air pressures and temperatures involved, it proved to be sufficiently accurate to assume a nominal average condition of temperature and pressure, reducing the calculation formula simply to

$$CFM_{\text{per duct}} = 630\sqrt{h} \tag{1}$$

where h = manometer reading in inches of water.

The value of the constant was assigned as follows:

Velocity of air at the Pitot tube

$$v = \sqrt{2\,p_s/\rho}\ , \text{ ft/sec}$$

where p_s = stagnation pressure, lb/sq ft

ρ = air density, lb sec^2/ft^4

= 0.00245 at 130° F, 2.5 psig .

Substituting for p_s in terms of h (inches of water), etc., each test section was uniform. That this was not so, is illustrated by two typical explorations made with the tube across the test section on one line during a previous brief test sequence (Figure 16). From these profiles, it is apparent that the single readings taken can in fact be interpreted only as indices, directly comparable only to themselves. Actual flows in any duct could conceivably vary from perhaps +10 percent to -50 percent of the values calculated on the assumption of a constant velocity all across the section.

Air consumption during the present test program was also calculated from the available blower curves (despite their limited validity in the present instance) as a rough check on the indications from the Pitot tube readings. Comparison of values calculated from the blower curves and from the Pitot readings showed that the former were only 50-70 percent of the latter. In view of the uncertain validity of the blower curves in the present use, however, the equally uncertain Pitot indications were selected for use in further discussions. These appear to be definitely on the high side. Until better figures are forthcoming, it is proposed to use the same figures for future design. They should be well on the conservative side, insofar as the sizing of any new air supply is concerned. [Inasmuch as the object of measuring flows in the separate ducts was defeated by the uncertainty of the results, it is apparent that the more intelligent arrangement would have been to use an instrumented inlet venturi instead. Such a unit could have been more properly formed and proportioned than present sections and hence would

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$$v = 65 \sqrt{h} \text{ ft/sec} .$$

Air flow through each duct of 1/6 sq ft, at the assumed nominal density, is (assuming uniform velocity across the throat) thus:

$$\text{Flow} = v \times 1/6 \times 60 \text{ cu ft/min}$$
$$= 650 \sqrt{h} .$$

Finally, correcting to standard free air conditions (60° F, 14.7 psig),

$$\text{CFM} = 630 \sqrt{h} \text{ cu ft/min.} \qquad (1)$$

Total flow was obtained by adding flows calculated individually for each duct. (The errors arising from variations in velocity distribution from the simple model assumed were not the same in each duct, of course.)

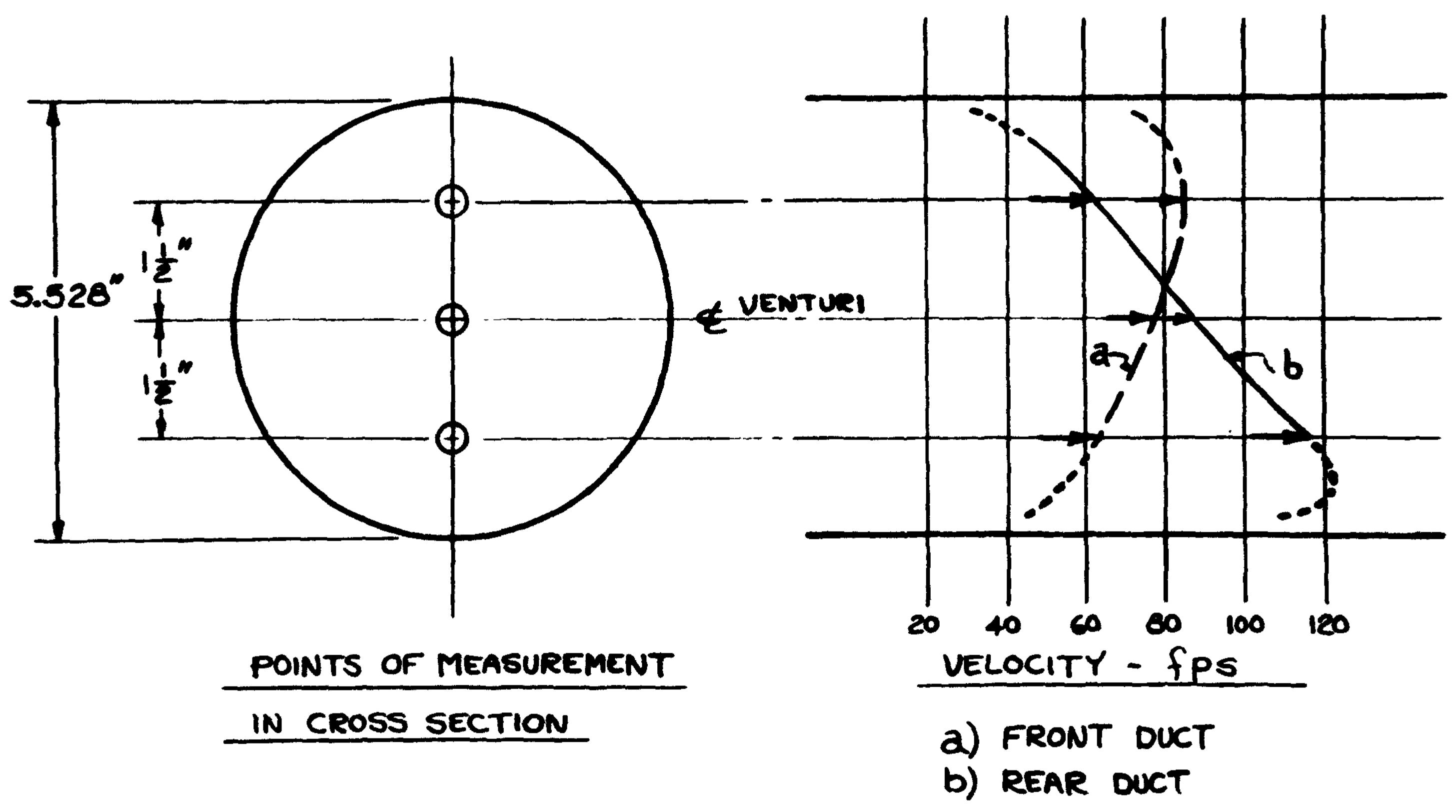

Figure 16. Velocity Distribution Across Instrumented Section.

have been inherently subject to less error. Moreover, the four Pitot tube/manometers used in the separate ducts could have been arranged in the new, larger, single throat to indicate velocity distribution across the section during testing.]

SPEED EFFECTS (Tests 79-86, 116-121, 161-166, 238-243)

Course tests at speeds from 0 to 12 mph, at a total load on the two test treads of 12,000 pounds and a pool pressure of 2.7 psig, indicated (via the Pitot readings) a total air consumption for the two tracks of 2100 to 2300 cfm (free air) at an initial manifold pressure of about 2.8 psig. Over the range tested, air consumption did not vary measurably with speed. Calculated adiabatic air horsepower was, correspondingly, about 31 horsepower.

It was observed during these tests that after about one mile of sustained running at 10-12 mph, there was localized heating of the Teflon rubbing surfaces on the inner and tread belts, resulting in some areas in a slight melting of the Teflon, which was subsequently blown out in fine threads by the escaping air (Figure 17). This, of course, reflects both high local friction forces and the low heat conductance of the Teflon. It demonstrates the necessity, in this type of device, to maintain some air flow in the seal areas for cooling.

Course tests at 0-10 mph at a load on the two treads of 10,000 pounds (using the unloading rigging described earlier) also showed no measurable change in air flow with speed.

TREAD LOADING EFFECTS, RUNNING TESTS (Tests 79-86, 116-121, 161-166, 238-243)

Course tests were run at normal tread loading (12,000 pounds) and at a reduced loading of 10,000 pounds, using the unloading rigging. In both cases the maximum available output of the air system was used. The results are tabulated briefly as follows:

Loading on treads	10,000 lb	12,000 lb
Suspension bag pressure	2.7 psig	2.7 psig
Pool pressure, brush seal side*	2.4 psig	2.6 psig
Pool pressure to "float"**	2.5 psig	3.0 psig
Total air flow	2200 cfm	2200 cfm

Manifold pressure	2.7 psig	2.7 psig
Air hp	30 hp	30 hp

*"Brush seal side" used because the seals were constant throughout the period. The lip seals on the other tread were changed twice during the test period due to damage. Note that the brush seals, to whatever extent they are effective, attenuate pool pressure from its maximum at the center to a lower value at the seal edge, through a distance (in this instance) of 6 inches, so that the required pool pressure must be a little higher than calculated as below.

**Based on 2000-square-inch effective pool area per tread.

From these data, it appears that as the system approaches a full "floating" condition, it merely spills additional air rather than actually floating freely. Thus, intermittent contact continues, and appears necessary at all times to stabilize the system.

Figure 17. Local heating of the Teflon surface on the inside of the treads, due to insufficient air support and resulting high contact pressures combined with high rubbing velocities, melted some of the Teflon. The melted plastic was blown out in threads, which cooled, leaving the evidence shown. This occurred after some 30 minutes of continuous operation at 10-12 mph.

TREAD LOADING EFFECTS, TOWING RESISTANCE (Tests 195, 196, 225-234)

Slow-speed towing resistance (Figure 18) of the complete rig was measured at three tread loadings, using full available air:

Loading on treads	9,000 lb	10,000 lb	12,000 lb
Front axle loading	11,600 lb	9,600 lb	5,600 lb
Gross rig weight	20,600 lb	19,600 lb	17,600 lb
Bag pressure	2.7 psig	2.7 psig	2.7 psig
Pool pressure (brush side)	2.3 psig	2.4 psig	2.3 psig
Pool pressure to "float"	2.3 psig	2.5 psig	3.0 psig
Total air flow	2000 cfm	2300 cfm	2100 cfm
Manifold pressure	2.8 psig	2.6 psig	2.6 psig
Air hp	28 hp	30 hp	27 hp
Total towing resistance*	3150 lb	3060 lb	3160 lb**

*Front axle plus treads plus driveline losses (transmission in neutral).

**Measured drag at this same load before the present modifications was 4600 pounds. (March 1962)

Figure 18. The rig was towed at three tread loadings, using the unloading boom, and total rolling resistance was measured with full air supplied. The same basic setup was used to measure drawbar output.

As might be expected from the air consumption recorded during course running, there is no sudden, downward "break" in motion resistance at the point where full flotation becomes possible (approximately at the 9000-pound load condition).

If it be assumed that the rolling resistance coefficients of the treads and the tires are each constant over their respective test load ranges, the above data might be taken to indicate that

$$R/W_{\text{treads}} = 0.212$$

$$R/W_{\text{front axle}} = 0.106$$

or

Calculated towing resistance	3140 lb	3140 lb	3140 lb

On this basis, the estimated tread resistance is high, but not discouragingly so. The corresponding front axle resistance is also high, which may reflect axle overloading, fairly high driveline resistance, and/or the behavior of the large front tires at modest inflations (20 psi) on a hard surface.

This may also indicate that the foregoing analysis is not valid. Assuming, instead, a constant total rolling resistance coefficient of 0.02 for the front axle, the resistance of the treads would, by differences, be as follows:

Tread load:	9,000 lb	Tread drag:	2920 lb	R/W:	0.32
	10,000 lb		2870 lb		0.29
	12,000 lb		3050 lb		0.25

Such a pattern would indicate the existence of high internal losses in the rig drive, perhaps including large belt hysteresis and/or sheave-belt engagement losses, and/or some constant, unavoidable rubbing losses in the seals which do not decrease substantially with increasing apparent air support. However, the latter could not by themselves account for anywhere near the total drag recorded. The basic frictional coefficient of Teflon-on-Teflon is of the order of only 0.05, and of polished steel on Teflon is less than 0.10, so that all air support would have to be totally discounted in order to even approach the measured values. The true origin of these high resistances merits further, detailed study.

TREAD LOADING EFFECTS, STATIC TESTS (Tests 170-190, 219-223)

Several static air flow tests were run to explore air flow over a still wider range of conditions. In one series, with the rig at a total loading on the two test treads of 10,000 pounds and with 2.7 psig in the suspension bags, the air flow was determined for several blower input speeds. In Figure 19, the measured flow and air horsepower (calculated from cfm and measured manifold pressure) are plotted on the basis of resulting manifold pressure. These curves give a clear indication of the possibilities to reduce air requirements by accepting less air flotation; or, conversely (since excess pool pressure does not appear materially to affect overall performance), it indicates the potentially high cost of providing too much air.

In a second series of tests, the weight of the rig was transferred to jacks, and load was reapplied to the treads and the air pools by adjusting suspension air bag pressure. The blower was run at a speed adjusted to give a pool pressure proportional to the bag pressure, taking 9000 pounds (on the two treads) on 2.7 psig bag pressure as the baseline loading. The results are shown in Figure 20.

These data permit quantitative discussion of the air requirements for other configurations having similar seals and basically similar tread layouts (discussed later).

DRAWBAR PULL (Test 216)

In the initial tests of the rig (spring of 1962), the drive sheaves slipped within the poly-vee tread belt under net drawbar loads of only 2000-3000 pounds. It was conjectured that some air pressure support might be acting to prevent proper engagement of the belts on the drive sheaves. Vents were accordingly installed to permit testing of the effects of a radical release of air pressure in way of the sheaves. One test of the modified rig was performed in which drawbar pull was measured at a 12,000-pound tread loading, 2.7-psi bag pressure, and full air to the treads. The rig was able to slip its tracks on a hard-packed gravel road, developing 8030 pounds of pull, or a net traction coefficient of 0.67. Opening or closing the sheave vents had no apparent influence, however. In this light, examination of other factors in the belt-sheave relationship indicated that the great improvement was probably due to having increased the stationary inner belt length by a small

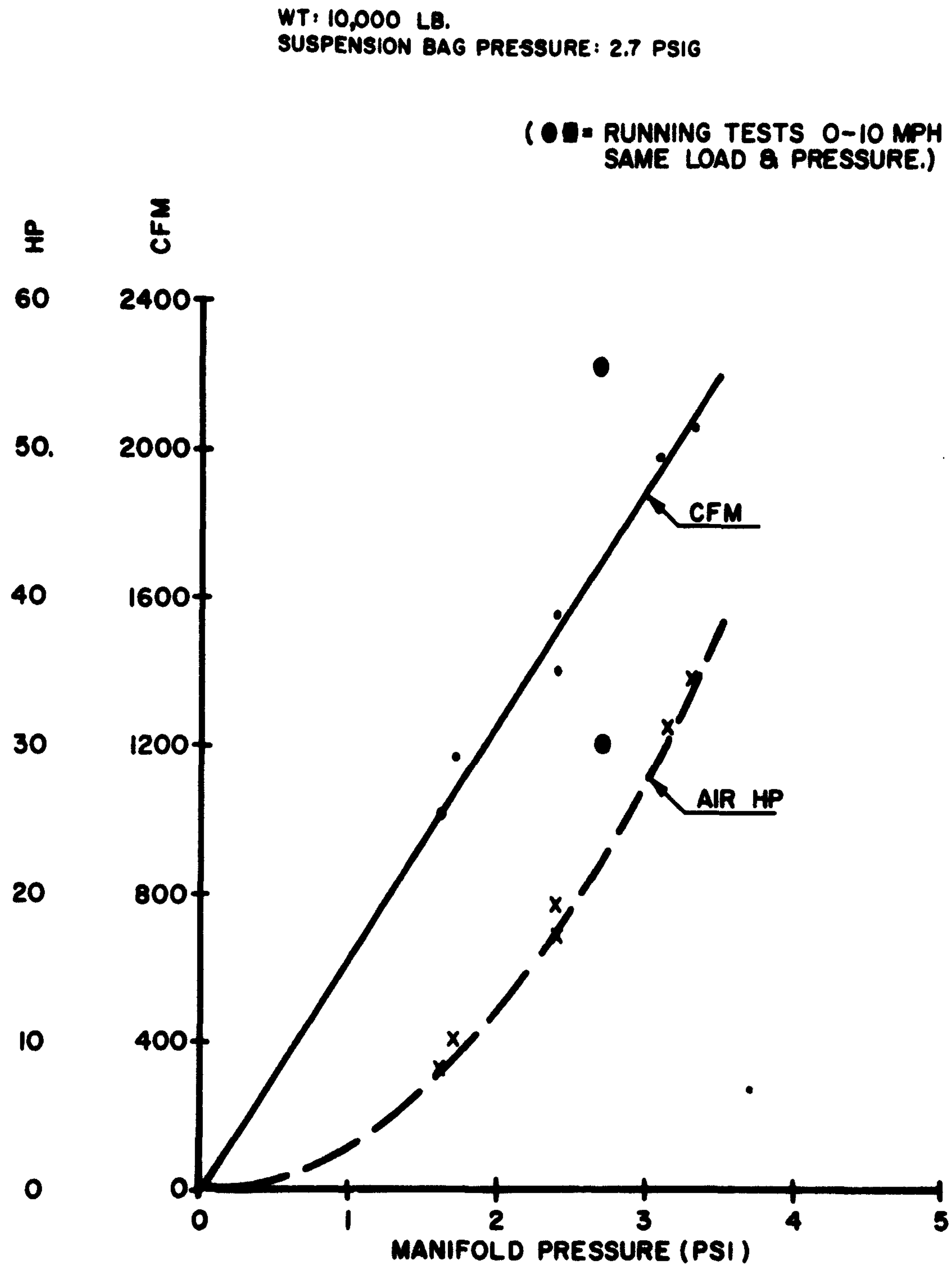

Figure 19. Variation in Air Flow with Manifold Pressure at Constant Loading -- Static Tests.

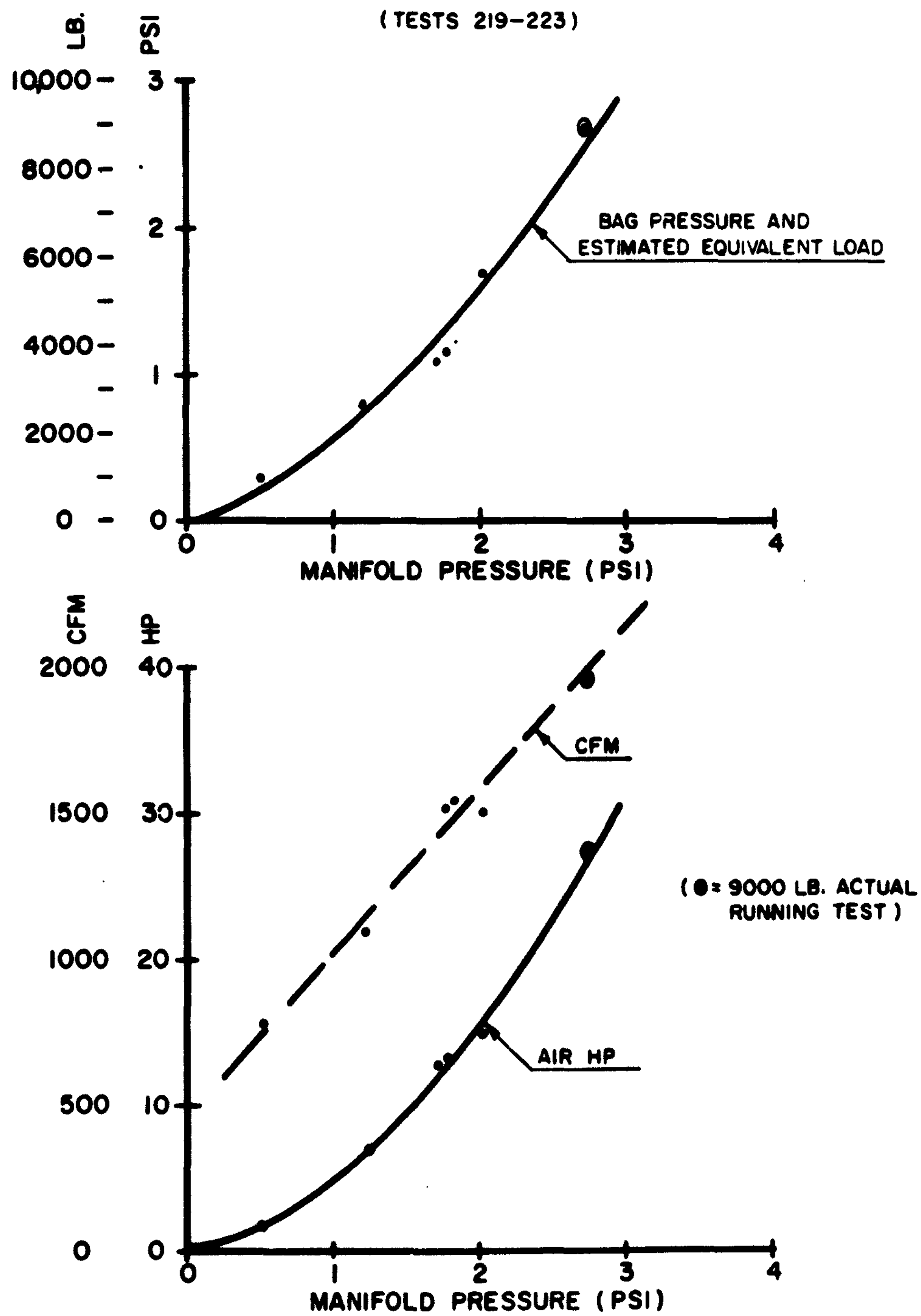

Figure 20. Static Tests at Variable Loadings.

amount (1/4 to 1/2 inch) in making final reassembly adjustments. This permitted more of the tensioning forces from the idler adjustment to be carried by the driven belt, and less by the inner belt. It was not realized initially that this adjustment was so critical.

This conclusion was indirectly confirmed by test operations over a large sand pile which considerably deformed the tread configuration over an appreciable area (Figure 21). In these maneuvers, sheave slippage within the belt again occurred. The most satisfactory explanation, especially in view of the drawbar test results, was that the large deflections tended to pull the drive sheave and idler together through the inner belt, slacking the tension carried directly by the driven outer belt. This indicates the need for a dual take-up arrangement to maintain tread and inner belt tension independently. The present idler take-up appears, in effect, to maintain tension only on the inner belt when crossing obstacles. (The upper track return

Figure 21. Operation over a sand pile deflected system sufficiently to reintroduce drive-sheave slippage within the tread belts.

section was originally intended to operate under a small pressure for the purpose of independently maintaining drive belt tension. However, at pressures suitable to provide tensioning, so much air was lost that additional seals were early added at the sheaves. These reduced the flow to the upper track satisfactorily but nullified the original intention.)

MISCELLANEOUS OBSERVATIONS

It was observed that air consumption increased slightly when the test rig was maneuvering in relatively tight patterns, when reversing, when pulling hard, and when negotiating obstacles. In every case, this appeared to be due to the development of localized gaps between inner and driving belts. The increase was in most cases less than 10 percent.

Although the instrumentation was too crude to permit a clear separation, it appeared that the tread equipped with brush seals drew consistently less air (by a small margin) and maintained approximately 0.1 to 0.2 psig higher pool pressure than the tread equipped with the lip seals. The softer lip seals appeared to retain air better than the original hard ones, but were blown out early in the program. The flexibility needed to improve sealing by this means is not compatible with the rigidity necessary to prevent their blowing into a momentarily enlarged gap and subsequently being destroyed.

TEARDOWN INSPECTION RESULTS

A major teardown inspection was undertaken at the conclusion of all the testing discussed herein. During the test program, however, it was necessary to partially dismentle the tread on the lip seal side on two occasions in order to repair those seals. This provided an opportunity for some interim inspection of the one side (Figure 22).

The final teardown revealed no damage (beyond that to the soft lip seals, repaired earlier by replacing them by the original, harder seals). The rate of wear of all the Teflon parts still appeared to be quite high. Total mileage on the new elements is only about 20 miles. However, the Teflon on the driven tread belts, which is the original material, at that time had a total of nearly

Figure 22. Damage to softer lip seals installed for test. Seals appear to have been blown into temporarily enlarged gaps. Harder seals used originally did not blow out but, by the same token, functioned primarily as a simple barrier rather than as a true lip seal.

50 miles on it, including both the presently reported tests and the earlier preliminary running, plus numerous demonstration runs. It was still in good condition (Figure 23). These parts, in particular, cannot yet be said to have "durability" in comparison with existing developed equipment, but the rig can now be run for moderate distances for test and demonstration purposes without serious risk of outright failures. Wear on the inner-belt Teflon/rubber skid is most serious at the ends of the air pool, where the curvature of treads up to the sheaves and idlers requires higher pool pressures than elsewhere to prevent any rubbing (Figure 24). These increased pressures are not available, so that considerable rubbing must in fact take place. Guiding forces arising during a turn are also a maximum in this same area. Relaxation of the rigid air-support concept to permit use of some mechanical support at these points, still within the "clean" atmosphere of the air-support envelope, would be clearly desirable.

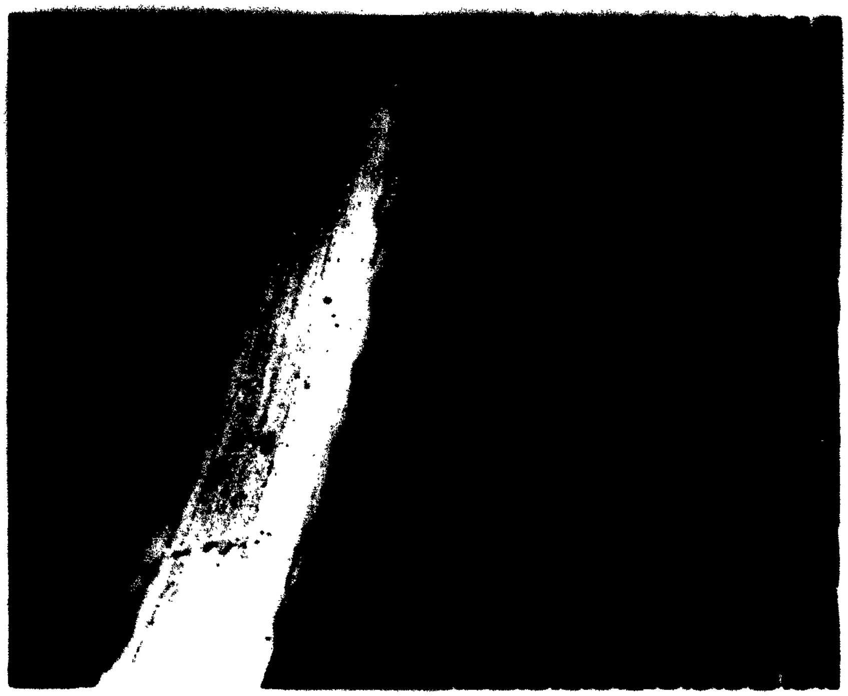

Figure 23. The Teflon rubbing surface on the inside of the tread belt shows definite signs of usage, but appears still to be entirely serviceable after a total of about 50 miles of operation, most of it at pool pressures inadequate to carry the full tread loading solely on the air pool.

Figure 24. Wear on ends of Teflon-rubber laminated guide-seal rib on inner belt. Both guiding forces and tread turning or "pulley" forces are highest at this point.

DISCUSSION

GENERAL OBSERVATIONS ON POWER REQUIREMENTS FOR AIR-SUPPORTED TREADS

As can be seen from the data presented in previous sections, maximum adiabatic air horsepower developed at the blower manifold was of the order of only 30 horsepower. The actual power input to the blower, estimated from the characteristics of the engine and its operation during tests, appears to have been of the order of 120 horsepower. This indicates an overall efficiency of only 25 percent.

The blower adiabatic efficiency may be calculated from the characteristic curves of the blower and, more directly, from measured temperature rise through the blower. Because of lag in the response of the bulb-type thermometer used, temperature data taken at the conclusion of a relatively long, sustained test run are considered to be the most reliable.

Adiabatic blower efficiency is given by the ratio

$$\eta = \frac{\Delta T_{adiabatic}}{\Delta T_{measured\ (or\ calculated)}} \qquad (2)$$

and $\Delta T_{adiabatic} = T_1\ [R^{(n-1)/n} - 1]$

where R = pressure ratio

$= 1 + \frac{p_m}{14.7}$

p_m = manifold pressure, psig

T_1 = inlet temperature, °R

$= 520°$ R

and $n = 1.4$ for isentropic air compression.

Taking $\left[R^{(n-1)/n} - 1\right] \approx \frac{n - 1}{n} (R - 1)$

$$\Delta T_{adiabatic} = 10\ p_m,\ ^\circ F\ . \tag{3}$$

Calculated from the blower curves (Reference 2),

$$\Delta T_{calculated} = \frac{V^2_{imp}}{6750} \tag{4}$$

where V_{imp} = tip speed of impeller, ft/sec

= 0.05 N r

N = drive input, rpm

r = blower drive step-up ratio

= 7.21 for tests through No. 186

= 10.14 for tests after No. 186.

The pertinent measured data from several full-power tests are compared to calculated data in the following tabulation:

Test No.	p_m	Temperature Rise, °F			Adiabatic Efficiency	
		Meas-ured	Calcu-lated	adia-batic	Measured	Calculated
	psig	ΔT_m	ΔT_c (eq. 4)	ΔT_a (eq. 3)	$\Delta T_a/\Delta T_m$	$\Delta T_a/\Delta T_c$
86	2.7	82	96	27	0.33	0.28
121	2.7	74	88	27	0.36	0.31
216	2.7	80	88	27	0.34	0.31
243	2.7	81	85	27	0.33	0.32

From these, an adiabatic efficiency of about 33 percent may reasonably be assigned to the blower. Overall losses may then be summarized approximately as follows:

	Efficiency	Power Loss	Net Power
Engine output			120 hp
Transmission	90%	12 hp	108 hp
Blower mechanical drive	90%	11 hp	97 hp
Blower performance	33%	65 hp	32 hp

A proper blower for this pressure-flow service, designed for efficiency rather than minimum weight, and driven directly, should give an overall efficiency (drive and adiabatic) of the order of 60 percent. Such a blower would reduce engine power required on the present rig to only about 50 horsepower.

The basic power level needed for the air system is established by the pressure rise and the free air flow required. For the limited range of pressures of interest in this type of system, the adiabatic air horsepower is given approximately by

$$AHP \approx 0.0050\ p_m\ (CFM) \tag{5}$$

or, assuming an overall drive and blower efficiency of about 60 percent, engine power required for the system is

$$BHP = 0.0083\ p_m\ (CFM). \tag{6}$$

Thus, power requirements are set equally by the pressure required by the design and by the effectiveness of the seals, etc. From Figure 20 it appears that, for the rig as last tested, and fully "floated" on an air pool, the air flow to the two treads was

$$CFM \approx 500 + 600\ p_m\ . \tag{7}$$

It is reasonable to assume that, with a given general and seal configuration, air flow varies directly with effective total length of seals (ℓ) under primary pressure. This may be taken, for purposes of further discussion, to be

$$\ell = 2 \times 1.2 \times L \times m = 2.4\ L\ m\ ,\ \text{in.} \tag{8}$$

where

L = length of tread on the ground, in.

m = number of treads.

For the present rig

$$\ell = 300 \text{ in.}$$

so that from (7),

$$CFM = 2\,\ell(1 + p_m). \tag{9}$$

Combining with (6),

$$BHP = 0.0166\, p_m\, \ell(1 + p_m). \tag{10}$$

From this it is clear that power required for the air system increases essentially as the square of operating pressure, and linearly as the effective seal length.

Operating pressure (p_m) will be determined by pool pressure (p_p) necessary to just "float" the machine. The relationship between p_m and p_p for the present rig, in terms of net pressure drop, is shown in Figure 25. Despite considerable crudity in the ducting, and the line restriction introduced by the use of curtain jets at each end of the air pools, the pressure drop is only of the order of 10 percent, i.e.,

$$p_m = 1.1\, p_p. \tag{11}$$

Pool pressure, in turn, will be determined by tread dimensions (L, w), number of treads (m), and gross vehicle weight in pounds (W). Using the proportions of the current rig as a guide and assuming equal distribution of the load among the treads, this relationship is given approximately by

$$p_p = \frac{W}{1.1\, L\, m\, (w - 6)} \tag{12}$$

where

$$w = \text{overall tread width, in.,}$$

and the constants reflect the proportions of the supporting air pool in relation to nominal tread contact dimensions as dictated by overall design considerations. Combining (11) and (12),

$$p_m = \frac{W}{m\, L\, (w - 6)} \tag{13}$$

Combining this with (8) and (10) gives, finally,

$$\frac{BHP}{W} = \frac{0.040}{w - 6}\left[1 + \frac{W}{m\, L\, (w - 6)}\right]. \tag{14}$$

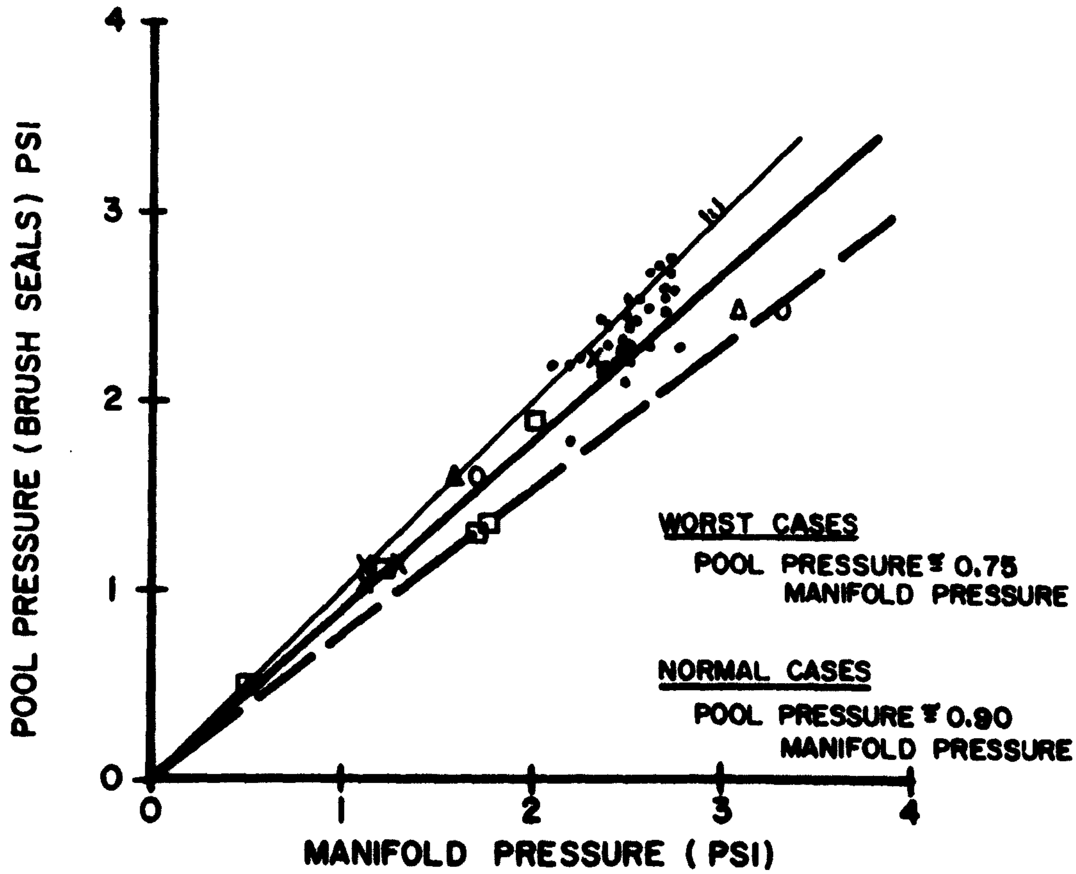

Figure 25. Pressure Drop in Ducts.

In this form, it is clear that power required per pound of gross vehicle weight, which may be taken as an effectiveness number, increases almost linearly with vehicle weight, all other factors remaining constant.

Also, since

$$p_n = \frac{W}{m\ L\ w} \tag{15}$$

= nominal unit ground pressure of the vehicle, psi,

equation (14) may be written

$$\frac{BHP}{W} = \frac{0.04}{w - 6} \left[1 + \frac{w}{w - 6} p_n \right] . \tag{16}$$

This form indicates that it is advantageous from an air system power viewpoint to reduce the nominal unit ground pressure as much as practical, and to do so with as wide treads as practicable.

As a check on the cumulative effect of the several assumptions made in reaching this point, the engine power required to "float" the present rig if the blower and its drive were more efficient may be recomputed.

For the present test-bed rig:

$W = 9000$ lb (the test weight at which pool pressures sufficient for full air flotation were actually achieved)

$w = 35$ in.

$p_p = 900/2 \times 62.5 \times 35 = 2.1$ psi.

$$BHP = \frac{0.04 \times 9000}{29} \left[1 + \frac{35}{29} \times 2.1 \right] = 44 \quad ,$$

which compares with 47 horsepower which would be estimated (28/0.60) from the 9000-pound test data presented previously in connection with the towing resistance tests. Note that since the air flow data used in developing equation (16) are considered conservative (see earlier) and refer, moreover, to the properties of the seal and overall rig configuration at this early stage of development, this equation should be safe for future design estimating purposes.

INTERNAL RESISTANCE AND WEAR

Internal drag and the wear of the moving parts are intimately associated. Anything which reduces one will, in general, reduce the other also.

The sources of the high internal resistances measured are still somewhat obscure. Quite possibly some minor adjustments in inner belt positioning, track tension, etc., could make big differences here, as they have already done in regard to belt/sheave slippage. It is also probable that drive line losses are quite high. As already noted, it is difficult to visualize such levels of resistance arising entirely from the drag of Teflon sliding on Teflon. However, there definitely is drag in these areas, as evidenced by the wear patterns. The apparent rate of wear indicates further that the drag from this source may be substantial.

The areas where wear of the Teflon parts is greatest are at either end of the air pools, and arise both from guiding forces during steering and from "pulley" forces where the tread must be continually pulled around corners. This suggests that the judicious use of resilient rollers at each end of the air pool designed for both guiding and turning the belt, and for sealing as well (see later), would make a big difference in both drag and wear at these critical points.

Some of the wear on the Teflon on the inboard edges of the tread units appears to be attributable to debris being blown into one seal area by air escape from the other. Inasmuch as these inner seals also account for some 50 percent of the air loss (see eq. 16), serious consideration should be given to the use (in an articulated configuration) of single, wide treads under the vehicle units. (The single treads would have to be stabilized in roll by the use of dual, side-by-side suspension units, of course.)

It is evident from the experience to date that proper rubbing surfaces (or other mechanical load-bearing means) must be provided to stabilize the air pool and to accommodate local pressure concentrations which constantly arise. In the air pool configuration tested on the rig, these surfaces double as the final air seals. Separating these functions, so that the load-carrying surfaces are essentially within the sealed area and hence better protected from the elements, should also increase their life.* This implies that the seals be so designed that their contact pressures are limited to the minimum necessary for their function. This would help to increase wear life of these parts substantially. If the seals in addition are sufficiently flexible to follow the deflections of the tread belt more closely, they will not only seal better but, in doing so, will still further inhibit the inflow of abrasive foreign materials and hence increase their own life further. The prospects for major improvements in the life of these parts of the system, and for improved sealing (see below), appear excellent.

EVALUATION OF THE FIRST TEST-BED RIG

The modest development and test program on the original test-bed rig was successfully completed. Crude but useful air-flow measurements were made over a sufficiently wide range of conditions to permit some generalization of the results so that the requirements of similar machines following may be estimated on a reasonable basis. Internal motion resistance was reduced approximately 30 percent, and sheave slippage within the tread belts was greatly reduced. Sealing was marginally improved, and durability was noticeably improved. Although none of these problems was solved completely, the experience suggested, in each case, further means and approaches of great promise. Unfortunately, these in general involved changes to the overall rig and tread design which were so basic and so extensive as to be impractical on the original machine.

The original rig has served its purpose. It has demonstrated that air-supported treads are mechanically feasible; it has indicated further that, properly applied and designed in detail, they have considerable potential as running gear on high-mobility, off-road transport vehicles. The initial promises of improved mobility through near-ideal load distribution between the tread and the ground in the contact area, and of improved "ride" through good suspension action, both remain valid, although demonstrations of these facets were limited. The hoped-for reductions in overall power

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*Care must be taken to keep Teflon wear products out of the drive system, however, unless positive tooth-type drive is provided.

consumption, in running gear weight, and in maintenance requirements were not directly demonstrated by the first test-bed rig. However, the improvements shown in the first cycle of development, and analysis of the measured data and general experience both show that those originally anticipated gains are still quite possible.

The program just concluded also shows that more development work is required. However, the original test-bed rig is not suited either for the further detailed development indicated or for the favorable demonstration in an operating machine of such advantages and potential of the air-supported tread concept as are even now possible.

A suitable "second-generation" machine to continue the development and evaluation of the concept should at this moment have the following features:

a) Be designed for a lower air-support pool pressure than that on the first rig;

b) Incorporate a more flexible sealing system;

c) Utilize a more efficient tread layout; and

d) Incorporate some mechanical guiding means at the ends of the air pools.

These, and a number of more detailed design objectives, are integrated in a preliminary design for a 1-1/4-ton-payload test rig presented in the following section.

PRELIMINARY DESIGN OF A "SECOND-GENERATION" TEST BED

In order to demonstrate and to examine further the stage of development of the air-supported tread concept reached at this point in the program, a "second-generation" test-bed rig was designed (Figure 26). The proposed new test bed is an articulated vehicle with a roll-free, pitch-damped, hydraulic-servo steering joint between units. Each unit rides on a single 60-inch-wide tread to reduce sealing length. Each tread is supported on two independent air-bag suspensions and two independent air pools in order to maintain necessary roll stiffness in the complete vehicle.

The new test-bed vehicle is 20 feet 2 inches in overall length and 68 inches in overall width. It is calculated that it will weigh 5500 pounds light and will gross 8700 pounds

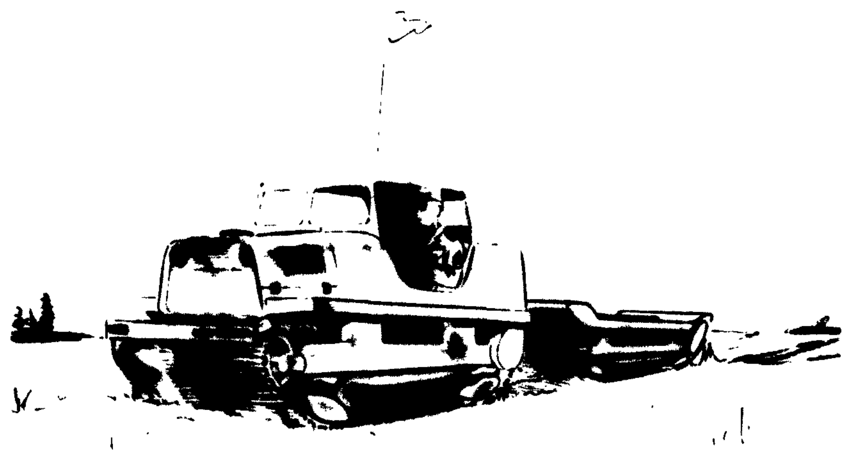


Figure 26. Artist's concept of a "second-generation" test rig designed to carry a net payload of 1-1/4 tons.

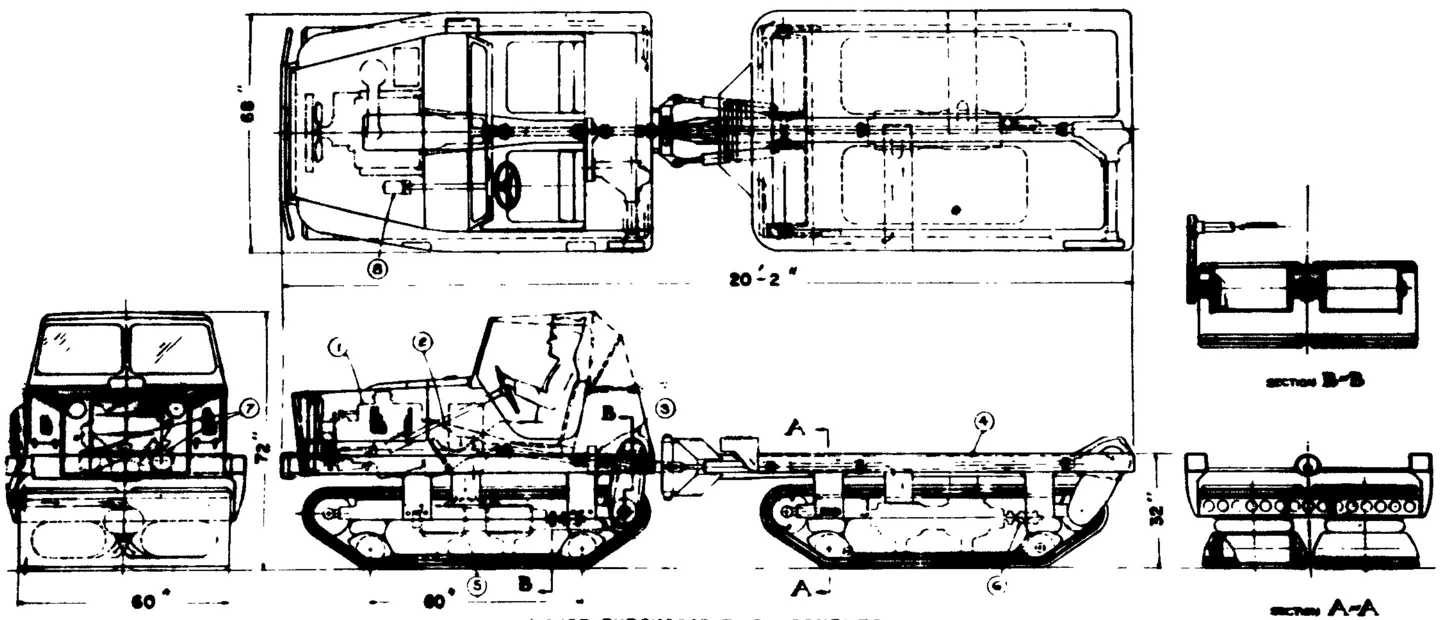
when carrying a net payload of 1-1/4 tons on the rear unit. Under these conditions, nominal unit ground pressure at maximum weight will be 1.3 psi, and total maximum air requirements are calculated to be approximately 700 cubic feet at 1.65 psig at the blower manifold. Blowers to supply this air are located within the tread envelopes, are hydrostatically driven at a constant speed, and draw a total of approximately 18 horsepower from the main propulsion engines at all times. The overall layout of the proposed rig is shown in Figure 27, and some preliminary details of critical design areas are shown in Figures 28, 29, and 30. Preliminary specifications are summarized in Appendix I.

The air-supported treads (Figure 28): Single 60-inch-wide treads support each unit of the vehicle. This reduces air losses by reducing sealing length, but increases steering moments somewhat. (The square contact patch adopted, however, gives the lowest moment for a given single total area.) The single tread also complicates the drive system by truly eliminating the usual, mechanically useful "belly."

Each tread is supported by two independent air-bag suspension units and air-support pools. This insures roll stiffness for each unit of the vehicle as a whole, and also imparts some roll stability to each tread on its inner supporting structure. The use of individual Rootes-type blowers for each air pool within each tread, driven at a common speed, is also designed to improve stability of the treads on their air-support pools.

The treads are built upon spliced, 1/4-inch, smooth fabric-rubber belts, with swage-bolted aluminum tread bars and urethane road pads. Inside each tread belt near the center are two rows of closely spaced nylon guide blocks, similar in function to the outer ribs on the belts of the original test-bed rig. Inside each belt also are Teflon surfaces along each edge in way of the outer seals and outboard inner-belt "bump" ribs, and in the center, between the guide blocks, in way of the center inner-belt "bump" and guide rib. Sealing is provided by means of an inflated seal along each edge, carried on the inner belt.

The inner belt (Figure 28), which forms the top of the air-support pool, is of the same general construction as the tread belt. It is reinforced against transverse bending by regularly spaced rectangular aluminum tubes. These, and the analogous tread bars on the outer tread belt, are interrupted



MAJOR PURCHASABLE COMPONENTS

1. BUICK SPECIAL ALUMINUM BLOCK ENGINE
2. BUICK SPECIAL DUAL-PATH TORQUE-CONVERTER TRANSMISSION
3. SPICER SERIES 53 DIFFERENTIALS (REWORKED)
4. SPICER SERIES 1310 DRIVE SHAFTS
5. MIEHLE-DEXTER TYPE 4012 ALUMINUM BLOWERS IN TANDEM HOOKUP
6. SUNDSTRAND 7M CSR-402 HYDRAULIC MOTORS
7. SUNDSTRAND 32 PVS 12 R-400 HYDRAULIC PUMPS (CONSTANT DELIVERY TYPE)
8. CHAR-LYNN ORBITROL HYDRAULIC STEERING GEAR

Figure 27. Overall Layout of Proposed Rig.

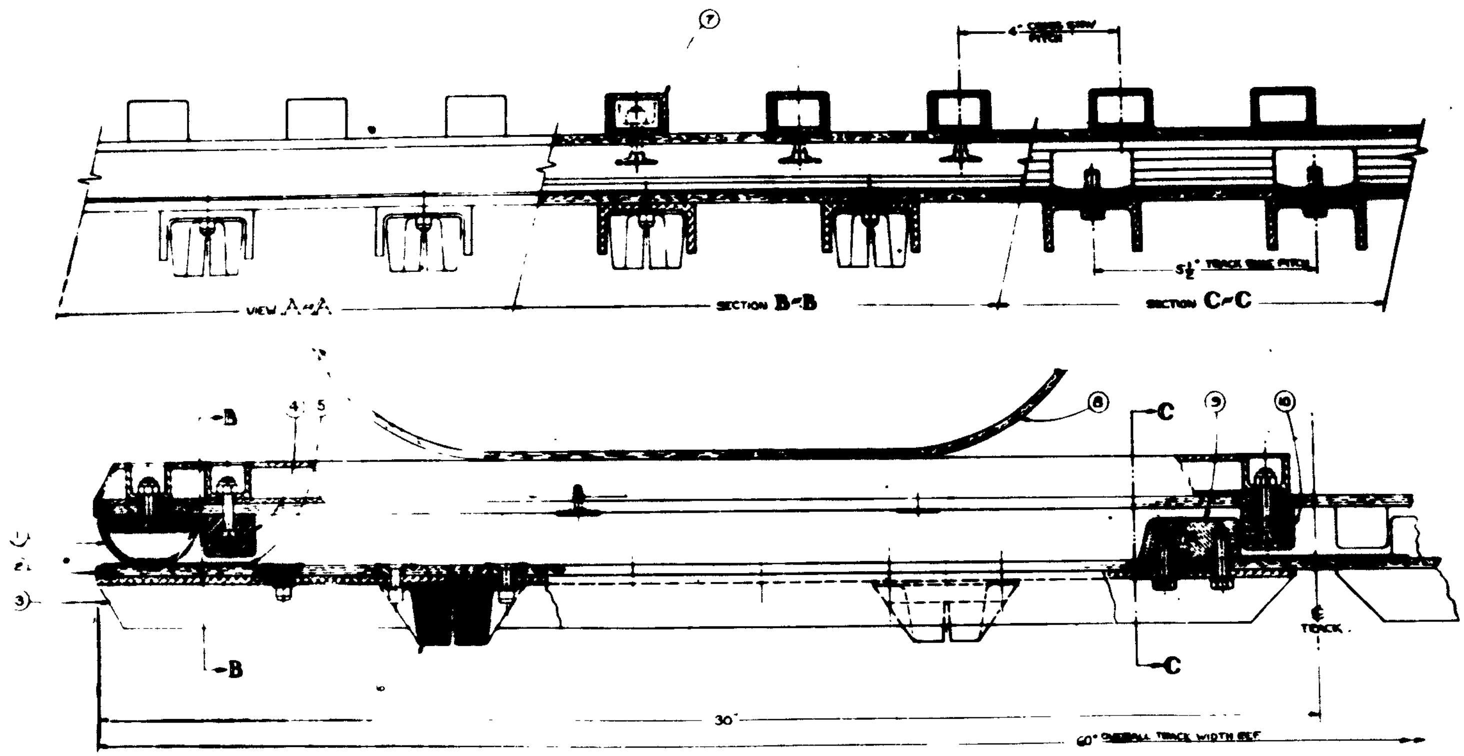
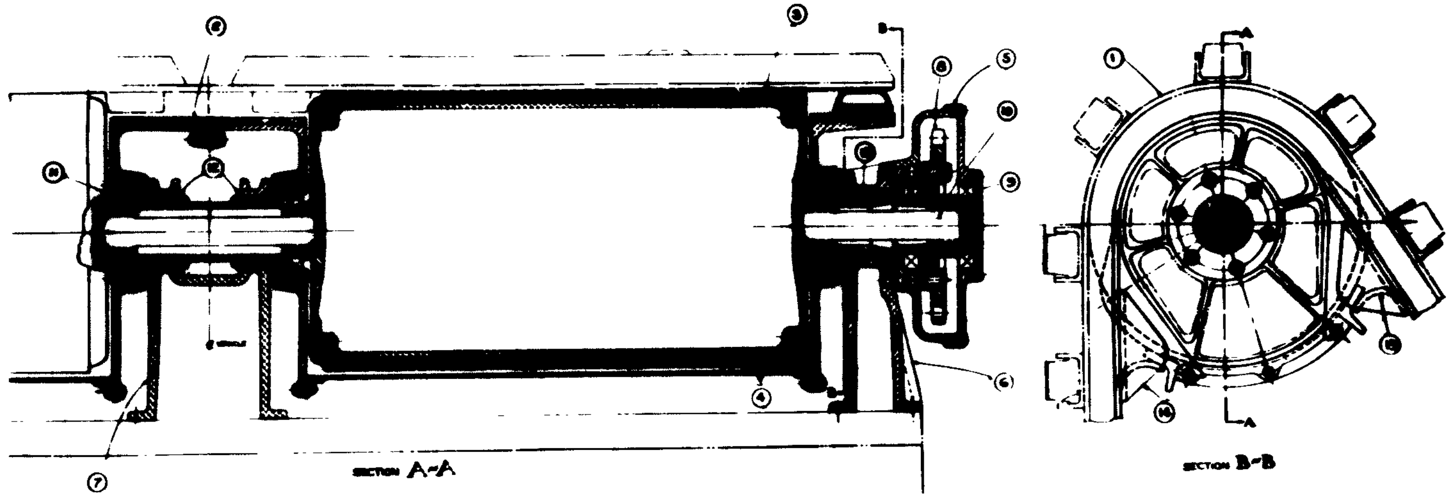


Figure 28. Tread and Inner Belt Assembly.



1. TRACK BELT
2. INNER BELT
3. RUBBER-COVERED TRACK DRIVE SHEAVE
4. SHEAVE SHIELD ASSEMBLY
5. CHAIN CASE
6. SHEAVE ASSEMBLY SUPPORT-OUTBOARD
7. SHEAVE ASSEMBLY SUPPORT-INBOARD
8. SHEAVE DRIVE SPROCKET
9. SPROCKET CARRIER BEARINGS
10. SPLINED, FLOATING DRIVE AXLE
11. SELF-ALIGNING SPHERICAL ROLLER BEARINGS
12. TRUNNION BEARINGS PERMITTING SHIELD ASSEMBLY TO MOVE WITH INNER BELT
13. FLEXIBLE INNER BELT FLAP ATTACHED TO SHIELD
14. INNER BELT FLAP END COVERS

Figure 29. Drive Sheave Assembly.

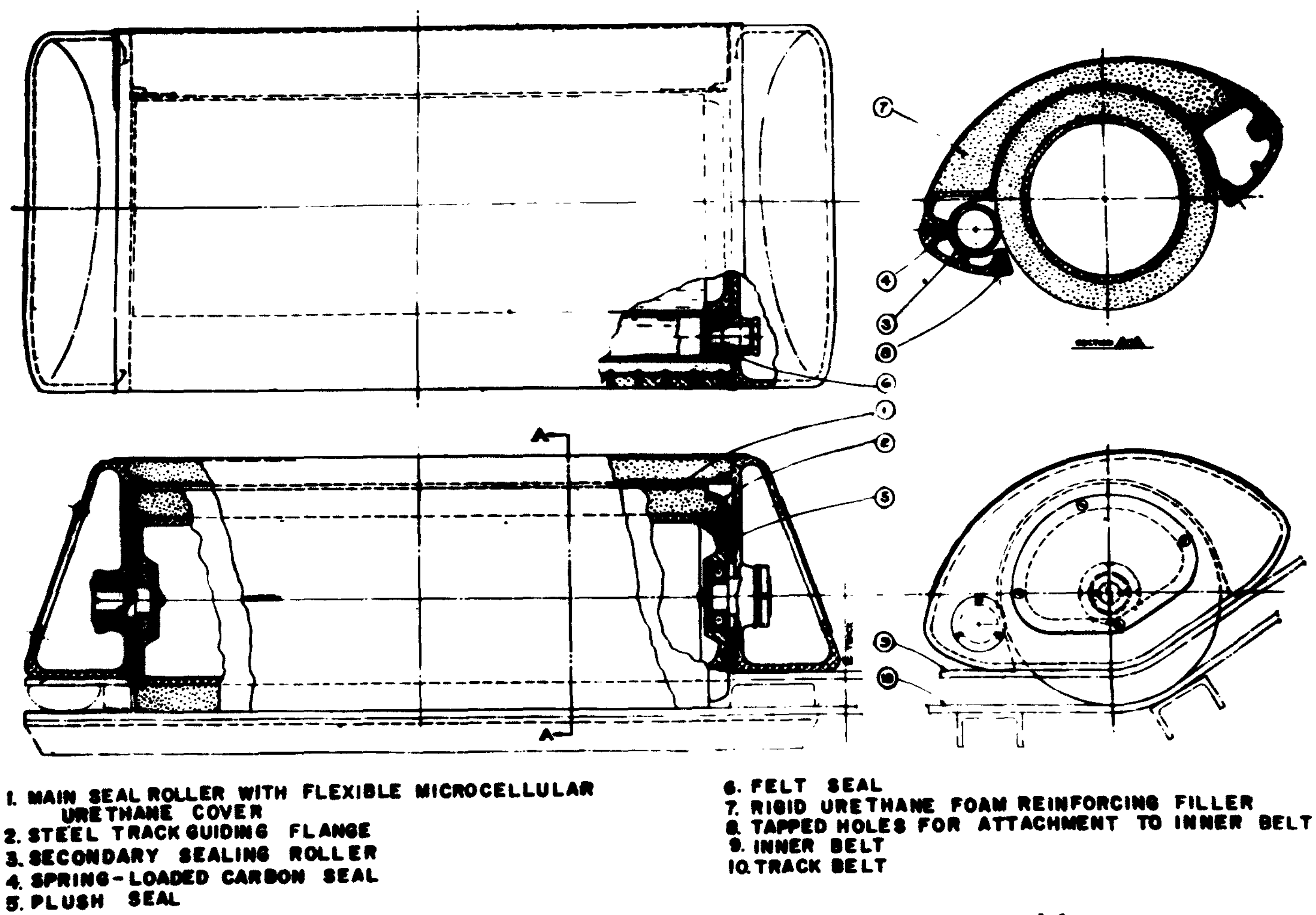


Figure 30. Tread Guide and End Seal Roller Assembly.

for several inches across the vehicle centerline to permit some transverse conformance of the tread width to large transverse ground irregularities.

The inner belt is essentially continuous around the complete tread circuit and is provided with its own tensioning system independent of the main tread tensioning system. It has an inflated seal along each edge. These seals are Teflon-covered and run on the matching Teflon-covered sections on the tread belts already referred to. Seal inflation pressure will be maintained a fraction of a pound-per-square-inch higher than pool pressure by tapping it from a simple Pitot tube in the blower outlet duct.

The inner belt has a continuous, Teflon-faced rubber "bump" rib just inside each seal to prevent complete collapse of the air pool under high, short-duration, local loads. It should normally run clear of the tread belt. When it does contact the moving tread, it will be on another Teflon surface. Finally, the inner belt will have a multipurpose bump, guide, and transverse air barrier rib along the center, laminated of Teflon and rubber in similar fashion to the inner-belt ribs successfully demonstrated on the first test-bed rig after the last period of modifications.

Drive for each tread is by means of two 21-inch-wide by 12-inch-diameter, smooth, rubber-covered drive sheaves in the rear idler position (Figure 29). These operate wholly within the pressurized envelope and, hence, in an essentially clean atmosphere. The drive sheaves and front idler sheaves are the same except that the idlers are mounted to permit fore-and-aft movement on adjustable tread tensioning springs. Both idlers and drive sheaves have shaped steel guide plates which engage the nylon guide blocks on the tread belt center for lateral control of the treads at these points.

A guide-seal roller, rubber covered, 21 inches wide by 8 inches in diameter, is fitted through the inner belt and rides on the tread belt at each end of each air pool (Figure 30). These rollers turn freely in close-fitting, cast-aluminum housings which are structurally integrated with the inner belts. The rollers perform three functions. First, they form a positive rolling seal at each end of each air pool. Air passage around a roller is prevented by seals between the roller surfaces and its housing. Second, the rollers incorporate shaped steel guide plates which engage the nylon guide blocks of the moving tread belt and thereby provide, at each end of the contact patch, positive lateral guiding independent of the Teflon ribs. Finally, the rollers provide rolling contact at the points where the treads continually change direction and, hence, exert higher normal loadings on the supporting structure than in the normal flat contact patch.

The suspension of each tread, and of its matching inner belt with the end roller housings, is by means of three fabric-rubber pneumatic bags on each side (Figure 27). One long central bag on each side supports the main tread contact area at an inflation of 4 to 5 psi. It is shaped so that vertical deflection at any point along its length produces increasing resistance primarily by virtue of increasing contact area between the bag underside and the inner belt assembly. This will give each unit some pitch stability.

A smaller bag is fitted over each air pool end roller. These bags will be inflated independently to somewhat higher pressures than the main bags, to keep the end rollers normally in light contact with the ground. These small independent bags will also contribute to the pitch stability of the individual units of the vehicle.

The complete system is designed to have an effective full bump to rebound "wheel travel" of about 6 inches. Foamed polyurethane bump stops will be fitted as on the first rig. They will be neither as firm nor as restrictive (relatively) as those on the original machine, however. They will be outside of the suspension bags so that adjustments can be readily made in their properties if necessary.

Tread air supply: Calculations of the air flow required for the new design were made in accordance with equations (9) and (10). Because of space, drive, and air-pool stability considerations, it was decided to provide an individual blower for each of the four air support pools on the vehicle (one each side, each unit). The nominal air flow required from each, at a pool pressure of 1.5 psi, or a manifold pressure of about 1.65 psi, was accordingly (eq. 9):

$$\text{CFM} = 0.5 \times 2 \times 60\ (1 + 1.65) = 160\ \text{CFM}$$

and power required (eq. 10):

$$\text{HP} = 0.5 \times 0.0166 \times 60 \times 1.65\ (1 + 1.65) = 2.2\ \text{hp.}$$

A slightly oversized unit was selected, the Miehle-

Dexter 4006 three-lobe Rootes-type blower. The manufacturer's information on this unit shows that it delivers 200 cfm and 1.65 psig at 2500 rpm, while absorbing 3.0 horsepower at the input shaft. The aluminum version weighs 70 pounds (Reference 3).

Two of these are mounted within the track envelope of each unit. Each pair draws its intake air from a common duct and filter system, through intake grills located above deck level and, insofar as possible, in relatively dust-free areas. The frame and deck side supports of lightweight welded-aluminum box sections serve as the necessary ducts in most places.

The blowers on each unit are driven by a common hydraulic motor. Power for each hydraulic motor is furnished by its own servo pump, belt-driven from the main propulsion engine. Each pump-motor system is designed to maintain a constant output speed of 2500 rpm at all engine speeds from 1000 rpm to 4000 rpm. Rated output power of each motor is 6 horsepower, which matches the requirements of the two blowers to be driven by each. This complete system (9 gpm, 1500 psi) is commercially available from Sundstrand. It was developed for use in driving refrigerator equipment on trucks, where a similar requirement exists for the auxiliary machinery to run at substantially constant speed regardless of main engine speed.

The power train (Figure 27) consists of a Buick Skylark aluminum V-8 passenger-car engine which develops 150 gross horsepower at 4000 rpm. This is used with the Buick Dual-Path torque converter two-speed transmission. These elements were selected for their very low weight, the engine-transmission package weighing only 450 pounds complete. The torque-converter transmission is desirable also. The engine must be brought up to 1000 rpm or more to insure air support, and the characteristics of the torque converter insure that the engine and air system will at all times cooperate.

Drive is taken from the transmission to a light chain case incorporated with the input to a modified Spicer Series 53 axle on the front unit. The chain case provides a through-drive to a similar axle (without chain case) on the rear unit. Drive passes to the rear via a Spicer Series 1310 drive line through the interunit joint, in the same manner as on the WNRE POLECAT, MUSK-OX, etc. (Reference 4).

Final drive from the axles is via chain cases over one side. These incorporate a final reduction sufficient to insure 70-percent gradeability of the vehicle at maximum gross weight. These, in turn, drive the rear-mounted, rubber-covered dual drive sheaves. Overall drive ratios have been selected to insure the quoted performance even at the same levels of internal losses and air horsepower requirements found on the first rig. The final drive chain cases will be designed to permit ready adjustment of the ratio, however, so that all improvements which may be effected, whether in the initial design or during subsequent development, may be more-or-less immediately reflected in terms of maximum speed.

The structure will be of magnesium and welded aluminum. Box sections will be used for maximum rigidity and minimum weight. In many places they will double as air ducting.

Special efforts will be taken in the design of all special parts, and in the selection of all components, to hold the vehicle weight light to the level shown. A detailed breakdown of the estimated weight is given in Appendix II. It will be seen that, although the requirements are stringent, they should prove feasible with careful design.

FLOATER CONFIGURATIONS

The configuration illustrated and discussed thus far is not a "floater" or amphibian. However, a light fiberglass/plastic laminate hull could readily be added to each unit, as shown in Figure 31a. This, plus the addition of necessary waterproofing in some places and of such accessories as bilge pumps, side skirts, and end deflectors on the treads, would add approximately 150 pounds to the weight of each unit. The height of the side above ground would be 45 inches and full-load freeboard would be about 8 inches.

In this configuration, the rear unit when light would float with the hull bottom just immersed. This is essential to insure its static stability in this condition. At any time that the waterline lies between the upper tread and the sponson bottom, stability will be greatly reduced and the machine will list to an appreciable angle under the influence of minor shifts in weight, light breezes, small waves, etc. Fully closing the sides around the return treads with a continuous 4-inch-wide buoyant section on each side will improve matters somewhat, but will aggravate the problem of "clogging" the return tread with mud, vegetation, etc., in some environments.

An alternate configuration, in which the lower tread section was designed to provide the total buoyancy required, was also briefly investigated. This is shown in Figure 31b. Comparative figures for the two schemes for floating are as follows:

	Hull Flotation	Ponton Flotation
Length overall	21 ft. 0 in.	24 ft. 0 in.
Width overall	68 in.	84 in.
Height overall (windshield and top down)	56 in.	62 in.
Curb weight	6300 lb	7200 lb
Draft	37 in.	25 in.
Tread width	60 in.	76 in.
Length of tread on ground	60 in.	76 in.
Nominal unit ground pres- ure at max. GVW	1.3 psi	0.9 psi
Required manifold pressure	1.5 psi	1.0 psi
Required total airflow	720 cfm	730 cfm
Air horsepower	5.4	3.7
Maximum speed (governed)	15 mph	13.5 mph
Maximum gradeability at GVW	70%	60%
Estimated maximum water speed	4-5 mph	6-7 mph

Both use the same basic power train, air system, and layout as the nonfloating version.

The overall cost in weight, size, and land performance necessary to float the machine on its treads appears to be too great for the minor gain in water speed which might be achieved as a result.

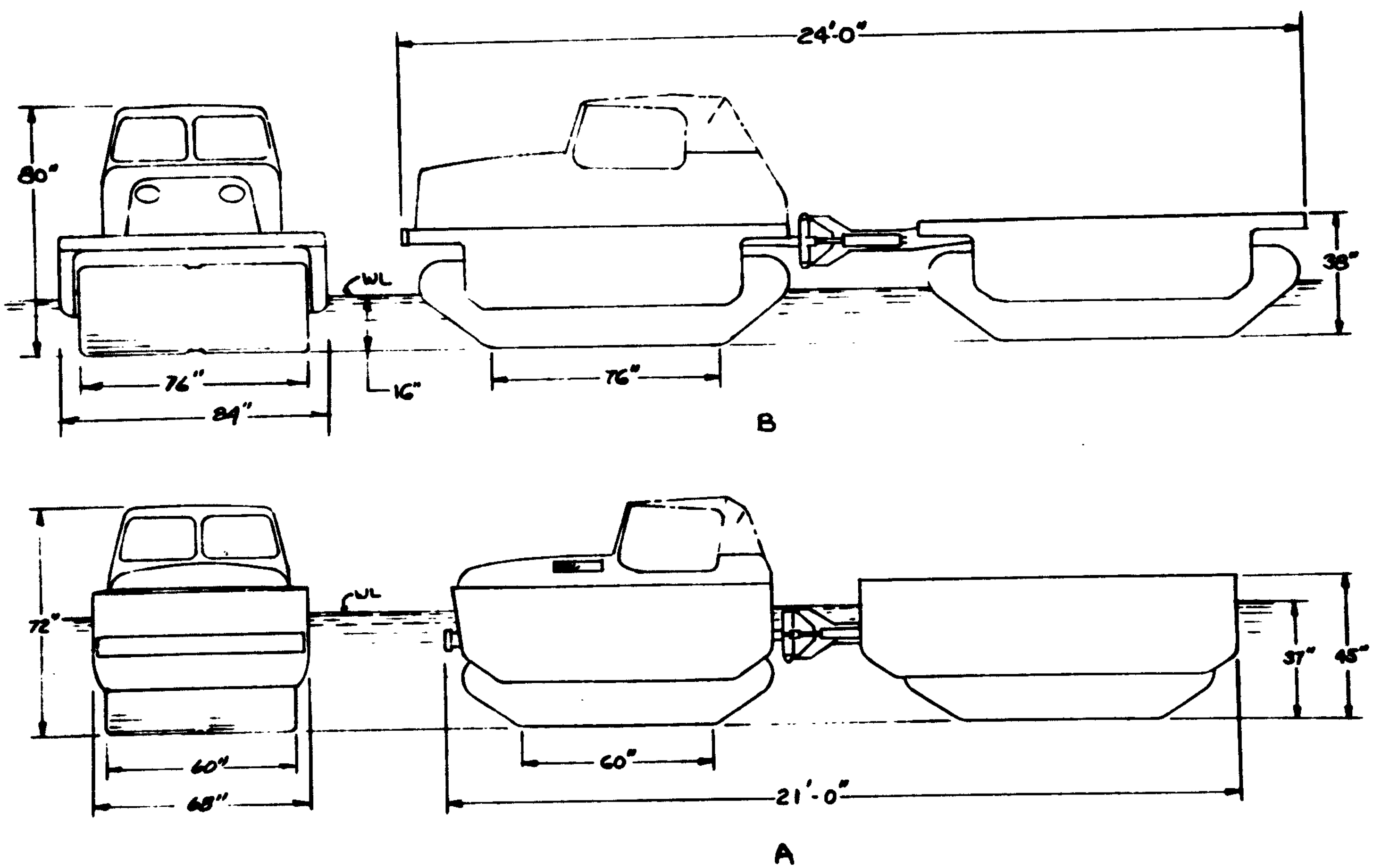


Figure 31. Alternate Floating Configurations.

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2. "Wright Aircraft Engine Performance," SP-976, Engine Model R1820-82, 15 December 1952.

3. M-D Blowers, Inc., Racine, Wisconsin, "Rotary Blower Performance Curve, Model 4006," No. C1-406, 11 November 1958.

4. Nuttall, C. J., Jr., "Steering of Tracked Vehicles by Articulation," First International Conference on the Mechanics of Soil-Vehicle Systems, Preprint No. 10, June 1961.

APPENDIX I

PRELIMINARY SPECIFICATIONS

RESEARCH BED, AIR-SUPPORTED TREAD, ARTICULATED, 1-1/4-TON NET CARGO

DIMENSIONS

	Front Unit	Rear Unit	O.A.
Length	9 ft. 4 in.	11 ft. 2 in.	20 ft. 2 in.
Width	68 in.	68 in.	68 in.
Height-over top	72 in.	--	72 in.
-to deck	32 in.	32 in.	32 in.
Belly clearance	Bellyless	Bellyless	Bellyless
Tread width	60 in.	60 in.	60 in.

WEIGHTS

	Front Unit	Rear Unit	O.A.
Light	3400 lb	2200 lb	5600 lb
Gas, oil, driver	400 lb	--	400 lb
Curb weight	3800 lb	2200 lb	6000 lb
Net payload	300 lb*	2500 lb	2800 lb
Gross weight	4100 lb	4700 lb	8800 lb

(*passenger, gear)

CHASSIS

Magnesium and welded aluminum construction
Side slope ability: fully controllable on 40 percent side slopes
Verticle obstacle climbing: 16 inches
Trench crossing: 6 feet (approximate)
Angle of approach: 60 degrees
Angle of departure: 60 degrees

POWER TRAIN

Engine:	Buick Skylark V-8, 215 cu. in., 8.8:1 compression ratio. 150 GHP at 4000 rpm, 220 lb.-ft. gross torque at 2400 rpm. Governed at 4000 rpm.
Cooling:	Full load at ambient 100° F.
Transmission:	Buick Dual-Path torque converter, two-speed. Stall ratio 2.50:1; high, 1:1; low, 1.58:1.
Drive Line:	Spicer Series 1310.
Axles:	Modified Spicer Series 53. Ratio 4.88:1.
Final Drive:	Enclosed, lubricated chain case, 1-inch single chain; 1.85:1 reduction.
Drive Pulleys:	Rubber covered, 12-inch diameter x 21-inch wide.
Brakes:	8-inch single caliper hydraulic disc brake incorporated in drive line at front unit drive-split chain case; foot operated.
Fuel:	30-gallon tank on front unit. Range: 50-100 miles.

AIR SYSTEM

Two Miehle-Dexter 4006 aluminum Rootes-type blowers driven in tandem, operating in parallel, in belly of each unit. One blower supplies air to each side of tread air pool.

Drive:	Front and rear blower pairs each driven by separate hydrostatic system: Sundstrand 32 PVS 12R-400 constant delivery pump and 7MCSR-402 motor. 6 horsepower at 2500 rpm output available at blowers for all engine speeds from 1000 rpm to 4000 rpm. (9 gpm at 1500 psi)
Blowers:	M-D 4006 - at 1.65 psig, 2500 rpm, horsepower input is 3 hp each; air delivery 200 cfm each.

PERFORMANCE

Speed:	Maximum at 4000 rpm, 15 mph.
Gradeability:	70 percent at maximum GVW, at converter stall.

TREADS

60-inch-wide, single tread under each unit. Aluminum channel tread bars and polyurethane road pads, swage bolted to 1/4-inch continuous rayon/nylon-fabric-based rubber belt. Rubber-to-rubber friction drive from rear drive sheave in each unit. Each tread runs on dual air pools, with pneumatic inflated side seals, dual barrier center separators. Each air pool supplied by separate blower. End sealing via resilient roller. Guiding primarily by means of steel guide plates on seal rollers, working on nylon guide blocks on tread at center.

	Front	Rear	Total
Tread area:	3600 sq.in.	3600 sq.in.	7200 sq. in.
Nominal unit ground pressure			
at curb weight:	1.1 psi	0.6 psi	0.9 psi
at GVW:	1.2 psi	1.3 psi	1.3 psi
Air pool area:	3200 sq.in.	3200 sq.in.	
Pool pressure			
at curb weight:	1.2 psig	0.7 psig	
at GVW:	1.3 psig	1.5 psig	

Tread tensioning via coil spring loaded and screw-adjustable plunger-mounted front idler supports.

SUSPENSION

Two independent main pneumatic bags under each unit at about 4 psi. Smaller independent pneumatic bags over each rolling air pool-end seal, at about 6 psi (four of these per unit). Inner pool belt, of same material type as tread belt, separately spring-tensioned. Inner belt transverse supports of rectangular aluminum tubing. Foamed polyurethane full-bump stops. Maximum effective travel, 4-inch bump, 2-inch rebound.

STEERING

Full hydraulic actuation of 14-inch, 35° POLECAT articulating joint between units. Hydraulic follow-up.

Pump:	6 gpm, 1500 psi, Vickers VTM27-50/40
Valve:	Orbitrol U203
Cylinders:	two 2-1/2-inch diameter x 13-inch stroke
Turning diameter:	47 feet curb to curb

CAB

Seating for driver plus one passenger in open fiberglass cab. Folding windshield, soft top. Two windshield wipers.

Controls: 16-inch steering wheel, transmission range selector, foot and hand throttle, foot brake.

Horn, light switches, full engine and air system instruments.

ELECTRICAL

12 V electric system. Flood-type headlights, tail lights, 45-ampere-hour battery.

CARGO PROVISIONS

1-1/4-ton cargo carried on rear unit deck, 56 inches x 92 inches. Loading height: 32 inches. Corrugated magnesium decking.

APPENDIX II

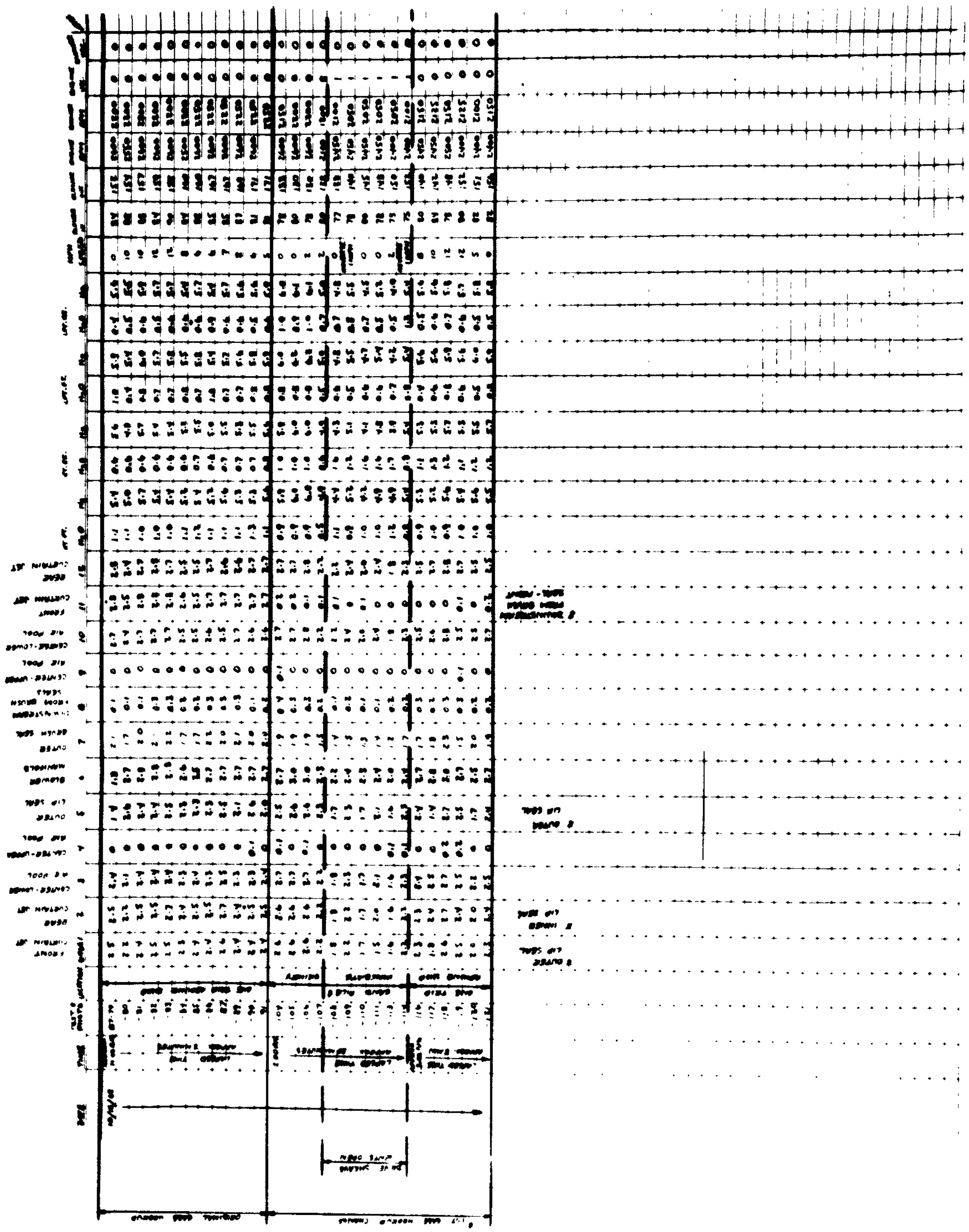
WEIGHT ESTIMATE

	Front Unit (lb.)	Rear Unit (lb.)
POWER TRAIN		
Engine	340	-
Transmission	110	-
Drive line	20	30
Axle	60	60
Chain case	40	40
Radiator, etc.	50	-
Battery, etc.	50	-
Brake system	20	-
Fuel tanks, system	30	-
AIR SYSTEM		
Drive pumps, belts, etc.	60	-
Blowers	140	140
Motors	30	30
Piping, couplings, etc.	10	10
Ducts	20	20
Air cleaners	10	10
STEERING SYSTEM		
Pump, belts, etc.	20	-
Joint	60	20
Cylinders	30	10
Wheel, valve	20	-
Piping	10	-
TREAD SYSTEM		
Treads	390	390
Inner belts, tensioning, etc.	240	240
Suspension bags	60	60
Guide/seal rollers, housings, etc.	180	180
Drive sheave system	170	170
Idlers, tensioning, etc.	190	190
Bump stops	30	30

<u>STRUCTURE</u>		
Lower, deck, frames	330	340
<u>MISCELLANEOUS</u>		
Bumper	30	-
Pintle	-	10
Instruments, controls, panel	30	-
Seats	40	-
Hood, body, top, windshield	160	-
Electrical	30	10
Miscellaneous piping, fluids	80	10
	3090	2000
Margin 10 percent	310	200
	3400 lb.	2200 lb.

APPENDIX III

DATA SHEETS



DATA SHEET NO. 1.

DATE	TIME	TEST NO.	CONDITION	GAGE 1	2	3	4	5	6	7	8	9	10	11	12	GR. PKT. H_2O	Hg	GR. ?? H_2O	Hg	UPR. ?? H_2O	Hg	UPR. ?? H_2O	Hg	MPH SPEED	GAGE IN.	GAGE WT.	ENGINE RPM	BLOWER RPM	ENGINE VAC.	BLOWER VAC.	NOTES
10/27/62	[illegible]	ST-158	[illegible]	2.2	2.2	2.3	2.3	2.2	2.5	1.5	0.3	0	2.6	0	3.4	1.1	5.0	1.5	5.0	0.4	5.6	0.2	5.5	0	70	98	2500	2400	0	0	1
		-159		2.2	2.1	2.2	0	2.2	2.5	1.5	0	0	3.4	0	3.4	1.2	5.0	1.7	4.9	0.5	5.4	0.2	5.3	0	70	131	2500	2150	0	0	1
10/31/62	9:30	-160		2.5	3.6	3.5	0.6	3.4	3.0	1.7	0	0	3.7	3.8	3.7	0.9	5.5	1.3	5.6	0.4	6.0	0.6	5.9	0	70	83	2500	2450	0	0	2
	9:35	-161		2.5	3.7	3.6	0.3	3.5	3.0	1.7	0	0.1	3.7	3.8	3.7	1.0	5.7	1.1	5.7	0.5	6.0	0.4	5.9	0	71	116	2500	2150	0	0	2
	9:36	-162		1.2	1.3	1.1	0	0.8	1.5	0.8	0.4	0	1.4	1.5	1.4	1.3	3.9	1.2	3.8	0.3	3.5	0.4	3.5	[illegible]	55	136	2400	2050	0	0	2
	9:38	-163	[illegible]	1.4	1.5	1.4	0	1.3	1.9	0.9	0.6	0	1.7	1.8	1.7	1.3	3.4	1.4	3.3	0.8	3.9	0.4	3.7	[illegible]	52	138	2400	2050	0	0	2
	9:40	-165		1.8	1.9	1.8	0	1.8	3.3	1.4	0.6	0	2.6	2.6	2.6	1.2	4.3	1.7	4.3	0.4	5.8	0.4	4.8	2	50	135	2400	2050	1	0	2
	9:46	-166		2.3	2.3	2.3	0	2.3	2.4	0.4	0.4	0	3.4	3.4	3.4	1.0	5.4	1.2	5.5	0.5	5.7	0.6	5.7	5	45	132	2400	2050	0	0	2
	10:10	-170	[illegible]	2.2	3.4	2.3	0	2.2	3.4	1.5	0	0	2.6	2.7	2.5	1.1	5.2	1.6	5.2	0.5	5.6	0.5	5.6	0	78	138	2500	2150	0	0	3
	10:22	-171		2.3	2.3	2.3	0	2.5	3.6	1.5	0	0	2.5	2.7	2.5	1.1	5.2	1.6	5.2	0.5	5.6	0.5	5.5		78	128	2650	2150	0	0	3
	10:25	-172		1.9	2.0	1.9	0.1	1.7	2.3	1.0	0	0	2.2	2.3	2.2	1.1	4.5	1.6	4.4	0.6	4.9	0.6	5.9		73	108	2600	2150	0	0	3
	10:30	-173		1.9	2.0	1.9	0	1.7	2.3	1.9	0	0	2.2	2.3	2.2	1.2	4.5	1.6	4.5	0.6	4.9	0.6	4.8		78	122	2600	2150	0	0	3
	10:37	-174		1.5	1.5	1.4	0	1.2	3.0	0.5	0	0	1.8	1.9	1.8	1.2	3.6	1.2	3.5	0.5	4.1	0.5	4.0		70	120	2500	2150	0	0	3
	10:50	-175		1.5	1.5	1.5	0	1.2	1.8	0.5	0	0	1.8	1.8	1.8	1.3	3.6	1.7	3.6	0.6	4.1	0.5	4.0		70	124	2500	2150	0	0	3
	10:58	-176		1.0	1.1	1.2	0	1.3	1.1	0.8	0	0	1.1	1.1	1.0	0.3	2.4	0.4	2.4	0.2	2.5	0.2	2.4		75	130	1250	1175	14	0	3
	11:10	-177		1.0	1.1	1.0	0	0.8	1.5	0.9	0	0	1.2	1.4	1.2	1.2	3.7	1.6	2.6	0.6	3.1	0.8	3.0		75	110	1400	2150	9	0	3
	11:18	-178		0.7	0.7	0.8	0	0.8	0.7	0.8	0	0	0.8	0.8	0.8	0.2	1.6	0.3	1.6	0.1	1.6	0.1	1.5		67	90	1000	1000	15	0	3
*	11:19	-179		0.9	0.9	0.9	0	0.8	1.3	0.2	0	0	1.1	1.2	1.1	0.6	2.6	0.5	2.5	0.2	2.7	0.4	2.6		68	95	2000	1675	7	0	3
*	11:20	-180		1.8	1.9	1.9	0	1.9	2.1	1.5	0	0	2.1	2.2	2.0	0.4	4.5	0.5	4.5	0.3	4.6	0.4	4.6		73	108	2000	1575	2	0	3

SINGLE VALVES

VARIOUS VALVES WITH JACK

NOTE NO. 1 TEST NOS. -158 & 159 INDICATED LIP SEAL FAILURE. TESTING WAS HALTED UNTIL INSPECTION & REPAIRS WERE COMPLETED.

NOTE NO. 2 TEST NOS. -161 THRU -166 WERE CARRIED OUT AS A GENERAL CHECK OF BEHAVIOR CHANGES RESULTING FROM LIP SEAL REPAIRS AND ADDITION OF UNLOADING [illegible]

NOTE NO. 3 TEST NOS. -170 THRU -180 WERE RUN TO ESTABLISH RELATIONSHIP BETWEEN LOWER AIR SUSPENSION BAG PRESSURE AND CORRESPONDING AIR POOL PRESSURE UNDER SIMULATED LOWER WEIGHT CONDITIONS. TESTS WERE RUN AT THE SEVERAL ARBITRARILY CHOSEN BAG PRESSURES AT NORMAL BAG HEIGHT. HEIGHT WAS MAINTAINED BY MEANS OF JACK UNDER VEHICLE ON THE LONGITUDINAL CENTER LINE. BLOWER SPEED WAS ADJUSTED TO PROVIDE APPROX. NORMAL AIR SUPPORT BETWEEN TRACK AND WHEEL BELT SKIDS. SINCE A DIFFERENCE IN AIR GAP EXISTED WITH BALANCING DAMPERS OPEN, R.H. DAMPERS WERE ADJUSTED TO GIVE APPROXIMATELY EQUAL SUPPORT ON BOTH TRACKS.

TEST NO.	(PSIG) BAG	DAMPER
-170	2.0	OPEN
-171	2.0	
-172	1.5	
-173	1.5	
-174	1.0	
-175	1.0	
-176	1.0	
-177	0.5	CLOSED
-178	0.5	
-179	MANUAL	ENGAGE TRACK
-180	MANUAL	ENGAGE TRACK

RUN AT NORMAL ENGINE RPM

* (REMAINS OF 3 FURTHER SPEED TESTS WHICH WERE MADE LATER)

NOTE NO. 4 ORIGINAL GAGE HOOKUP WAS USED ON ALL TESTS ON THIS AND NEXT PAGE.

DATA SHEET NO. 2

TRANSMISSION IN HIGH RANGE BLOWER RATIO 7.3:1

BLOWER RATIO 7.3:1 TRANSMISSION IN LOW RANGE

SHAFT FAILURE

TRANSMISSION IN LOW RANGE BLOWER RATIO [illegible]

DATE	TIME	TEST NO.	[illegible]	1	2	3	4	5	6	7	8	9	10	11	12	RT. FR. H_2O	H_2	RT. RR. H_2O	H_2	LFT. FR. H_2O	H_2	LFT. RR. H_2O	H_2	MPH SPEED	BLOWER IN	BLOWER OUT	RPM ENGINE	RPM BLOWER	CHANGE [illegible]	[illegible]	
11/1/62	1:30 PM	-182	[illegible]	2.5	2.5	2.5	0.1	2.5	2.8	1.9	0	0	2.9	3.0	2.9	0.8	6.3	0.8	6.3	0.5	6.1	0.7	6.4	0	72	103	[illegible]	[illegible]	0	0	1
	1:22	-183		2.2	2.2	2.2	0	2.2	2.2	1.7	0	0	2.4	2.4	2.4	0.5	5.1	0.4	5.2	0.2	5.2	0.4	5.2	0	77	135	[illegible]	1800	5.2	0	1
	1:30	-184		1.5	1.4	1.6	0	1.6	1.7	1.1	0	0	1.7	1.7	1.7	0.2	3.6	0.2	3.6	0.2	3.6	0.3	3.6	0	78	125	[illegible]	[illegible]	0.2	0	1
	2:00	-185		2.5	2.5	2.5	0.1	2.5	2.9	1.9	0	0	1.8	1.9	1.9	0.7	6.0	0.7	6.1	0.5	6.2	0.6	6.2	0	78	150	[illegible]	1550	0	0	2
	2:15	-186		2.7	2.8	2.7	0.1	2.6	2.9	1.9	0	0	2.9	2.9	2.9	1.0	5.8	1.2	5.9	0.5	6.2	0.5	6.2	0	[illegible]	112	[illegible]	[illegible]	0.1	0	2
		-187		2.6	2.7	2.7	0.1	2.9	2.9	1.9	0	0	2.8	2.9	2.8	0.8	5.8	1.1	5.8	0.4	6.0	0.5	6.0	0	88	125	[illegible]	1475	1.2	0	3
		-188		1.5	1.5	1.6	0	1.5	1.6	1.1	0	0	1.6	1.6	1.6	0.1	3.5	0.3	3.5	0.1	3.6	0.2	3.5	0	72	86	1650	975	18.8	0	4
		-189		2.1	2.2	2.2	0	2.3	2.4	1.7	0	0	2.4	2.4	2.4	0.2	5.0	0.5	5.0	0.2	5.1	0.3	5.1	0	75	98	[illegible]	1300	9	0	4
		-190		2.5	2.6	2.5	0	2.5	3.1	1.9	0	0	2.9	2.9	2.9	0.6	6.2	0.2	6.3	0.5	6.4	0.7	6.4	0	80	119	[illegible]	[illegible]	0	0	4
		191	[illegible]	1.7	1.8	1.8	0	1.7	2.7	0.7	0	0	2.1	2.2	2.1	0.7	4.7	0.8	4.8	0.7	4.9	1.0	4.7	2 [illegible]	76	126	[illegible]	1500	0.5	0	5
		-192		2.0	2.1	2.0	0	2.0	2.7	1.2	0	0	2.4	2.5	2.4	0.8	5.0	1.1	5.1	0.6	5.2	0.8	5.2	2 [illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.1	0	5
		-195		2.2	2.3	2.2	0	2.2	2.5	1.1	0	0	2.2	2.3	2.2	0.6	4.9	1.7	5.0	0.7	5.0	1.0	4.9	2	60	116	[illegible]	1525	0	0	6
		-196		2.2	2.2	2.2	0	2.2	2.3	1.9	0	0	2.2	2.3	2.3	0.6	4.8	1.0	5.2	0.7	4.9	1.1	4.9	TOWED 2	60	116	[illegible]	1500	0	0	6
		-199		1.9	1.9	1.8	0	1.8	2.5	1.2	0.1	0	2.2	2.2	2.2	0.6	4.6	1.5	4.3	0.5	5.0	0.8	5.0	STALL	65	105	[illegible]	1550	0	0	6
	3:30	-200		1.0	1.1	1.0	0	1.0	1.3	0.5	0.4	0	0.9	1.1	1.0	0.7	2.6	1.1	2.5	0.6	2.7	1.4	2.4	[illegible]	60	110	[illegible]	1500	0.7	0	6
11/2/62	9:10 AM	-211		2.3	2.4	2.2	0	2.2	2.6	1.2	0	0	2.5	2.5	2.5	1.1	5.2	1.5	5.2	0.1	5.6	0.5	5.7	2 [illegible]	55	96	[illegible]	1550	0	0	6
		-212		2.4	2.5	2.3	0	2.3	2.9	1.2	0	0	1.8	1.9	1.8	1.1	5.5	1.6	5.5	0.3	6.1	0.5	6.0	[illegible]	52	107	[illegible]	1575	0	0	–
		-213		2.3	2.4	2.3	0	2.3	2.7	1.2	0	0	2.5	2.6	2.5	0.9	5.8	1.1	5.7	1.2	6.1	0.5	6.1	0	55	111	[illegible]	1525	0.1	0	6
		-214		2.5	2.5	2.5	0	2.4	2.7	1.4	0	0	2.6	2.7	2.5	0.9	5.5	1.3	5.6	0.1	5.8	0.6	5.8	TOWED [illegible]	58	112	[illegible]	1550	[illegible]	0	6
		-215		2.3	2.5	2.3	0	2.3	2.0	1.9	0	0	2.8	2.9	2.8	1.2	5.3	1.6	5.4	0.3	6.1	0.5	6.0	[illegible] 2	55	128	[illegible]	1550	[illegible]	[illegible]	6
		-216		2.7	2.7	2.5	0	2.5	3.2	1.2	0	0	2.7	2.8	2.8	1.2	5.1	1.6	5.2	0.4	5.2	0.6	5.5	FWD STALL	58	[illegible]	[illegible]	1550	0	0	6
		-219	[illegible]	1.8	1.9	1.9	0	1.9	2.0	1.1	0.4	0	1.9	2.1	1.9	0.5	4.2	0.6	4.2	0.1	4.4	0.4	4.2	0	80	116	[illegible]	[illegible]	[illegible]	0	7
		-220		1.3	1.4	1.3	0	1.2	1.7	0.6	0.3	0	1.5	1.6	1.5	0.7	3.1	0.7	3.1	0.1	3.4	0.2	3.3	0	77	118	[illegible]	1250	7.2	0	7
		-221		1.3	1.3	1.3	0	1.2	1.7	0.6	0.1	0	1.5	1.6	1.5	0.6	3.3	0.7	3.3	0.1	3.5	0.3	3.4	0	78	117	[illegible]	[illegible]	[illegible]	0	7
		-222		1.0	1.1	1.1	0	1.0	1.2	0.5	0.3	0	1.2	1.3	1.1	0.2	2.6	0.2	2.7	0.1	2.7	0.2	2.7	0	74	102	[illegible]	1000	12.0	0	7
		-223		0.5	0.5	0.5	0	0.5	0.5	0.1	0	0	0.5	0.4	0.4	0.1	1.3	0.1	1.3	0.1	1.3	0.1	1.3	0	72	94	[illegible]	[illegible]	[illegible]	0	7
	2:[illegible] PM	-224	[illegible]	2.2	2.2	2.2	0	2.2	2.8	2.5	0.3	0	2.5	2.7	2.7	0.9	5.7	0.4	5.9	0.3	5.9	0.8	5.9	2 [illegible]	[illegible]	117	[illegible]	[illegible]	0	0	8
		-225		2.2	2.2	[illegible]	0	2.1	2.8	1.2	0.4	0	2.5	2.7	2.6	0.9	5.6	0.6	5.6	0.3	5.8	0.8	5.7	2	62	123	[illegible]	1550	0	0	8
		-232		2.2	2.3	2.3	0	2.2	3.1	1.7	0	0	2.4	2.4	2.4	0.9	5.0	1.4	5.1	0.4	5.8	0.9	5.8		61	[illegible]	[illegible]	1550	0	0	[illegible]
		-233		2.3	2.4	2.4	0	2.3	2.7	1.7	0	0	2.5	2.5	2.5	0.7	5.5	1.0	5.4	0.3	5.7	1.0	5.6		62	[illegible]	[illegible]	1500	1.0	0	[illegible]
		-234		2.3	2.2	2.2	0	2.2	2.5	1.2	0	0	2.1	2.2	2.2	0.8	5.0	0.9	5.0	0.1	5.1	0.9	5.1		62	127	[illegible]	1550	0	0	[illegible]
		-238	[illegible]	2.2	2.3	2.2	0	2.2	2.4	1.4	0	0	2.4	2.5	2.4	1.0	5.2	1.6	5.2	0.3	5.6	0.9	5.6	[illegible]	64	[illegible]	[illegible]	1550	0	0	9
		-239		2.3	2.5	2.3	0	2.4	3.0	1.4	0	0	2.5	2.7	2.6	1.0	5.4	1.4	5.3	0.2	5.8	0.7	5.8		63	137	[illegible]	[illegible]	0	0	9
		-240		2.4	2.5	2.3	0	2.2	2.7	1.8	0	0	2.5	2.7	2.5	1.0	5.1	1.4	5.1	0.3	5.5	0.7	5.4		65	145	[illegible]	[illegible]	0	0	9
		-241		2.2	2.2	2.2	0	2.1	2.6	1.5	0	0	2.4	2.4	2.3	0.9	5.2	1.0	5.2	0.3	5.4	0.9	5.4		62	[illegible]	[illegible]	1500	0	0	9
		-242		2.5	2.6	2.5	0	2.5	2.8	[illegible]	0	0	2.5	2.7	2.6	1.0	5.2	1.4	5.3	0.8	6.1	0.7	6.0		65	[illegible]	[illegible]	1550	0	0	9
		-243		2.5	2.5	2.3	0	2.4	2.7	1.5	0	0	2.5	2.6	2.5	1.2	4.8	2.0	4.7	0.3	5.4	0.5	5.4		67	[illegible]	[illegible]	[illegible]	0	0	9

NOTE NO 1. TEST NOS 182, 183, & 184 SERIES WAS RUN AS A CHECK OF BLOWER PERFORMANCE AT 3 DIFFERENT SPEEDS. INTENDED BLOWER RPMS WAS 2500, 1800, & 1400. THIS SERIES WAS REPEATED LATER BECAUSE INSUFFICIENT RUNNING TIME DID NOT ALLOW BLOWER OUTLET TEMP. TO STABILIZE. RH SIDE DAMPERS WERE LEFT CLOSED.

NOTE NO 2. SPARK PLUG HEATING & FOULING CAUSED A DELAY AT THIS POINT. SINCE COLDER SPARK PLUGS WERE NOT LOCALLY AVAILABLE, IT WAS DECIDED THAT OTHER AVAILABLE GEAR RATIOS SHOULD BE [illegible] WITH TO ALLEVIATE THIS CONDITION. TRANSMISSION WAS PLACED IN LOW RANGE AND ENGINE RUN UP. FOULING CEASED AND PERFORMANCE WAS EXCELLENT. IT WAS FELT HOWEVER THAT ENGINE RPM WAS UNNECESSARILY HIGH FOR SUSTAINED RUNNING SO BLOWER WAS SHIFTED BACK TO 10:1 RATIO FOR NEXT TEST.

NOTE NO 9. TRIP AROUND SHOP WITH 2400 LB UNLOAD

NOTE NO 3. -187 WAS A CHECK OF NEW GEAR RATIO

NOTE NO 4. -188, 189, & 190 IS A RETEST OF NOTE NO 1 WITH THE FOLLOWING DIFFERENCES. DUE TO BLOWER RATIO CHANGES SPEEDS CHOSEN WERE 2250 - 1275 - 1580 BLOWER RPM. TEMP DATA SHOULD BE ACCURATE, DUE TO LONGER RUNS AND REVERSING THE PROCEDURE - LOW TEMPS FIRST.

NOTE NO 5. -191 - 192 WERE TAKEN BACKING OUT OF SHOP IN THE DRAWBAR TESTS.

NOTE NO 6. -195 THRU -216 WERE ROLLING RESISTANCE & DRAWBAR TESTS. THESE WHEN FORCES WERE RECORDED SEE TABLE →

NOTE NO 7. RETEST OF -170 THRU -180 EXCEPT CONTINUING [illegible] PRESSURE @ 2, 15, 10, & .5 PSI - [illegible] PSI = 1.75, 1.1, [illegible], .25 [illegible]

NOTE NO 8. -228 THRU -234 ROLLING RESISTANCE AT 3 WEIGHTS

DRAWBAR TESTS

TEST NO.	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
-195	OPEN	2960			
-196	CLOSED	3000			
-199	OPEN				4230
-212			4230		
-214			4160		
-216				8000	
EARLY TESTS				6530	
-224			3720 (3000 [illegible])		
-225		3180			
-226		3630 (FULL [illegible])			
-228	OPEN	4480 (FULL [illegible])			
-232		3340 (2400 LB [illegible])			
-233		3210 ([illegible])			
-234		3190			

DATA SHEET NO. 3.

DISTRIBUTION

U. S. Army Infantry Center	2
U. S. Army Arctic Test Board	1
U. S. Army Armor Board	1
The Research Analysis Corporation	1
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Office of Chief of R&D	2
Environmental Sciences Division, OCRD	1
U. S. Army Tank-Automotive Center	2
U. S. Army Combat Developments Command, Transportation Agency	1
U. S. Army Transportation Research Command	26
Marine Corps Landing Force Development Center	1
Marine Corps Liaison Officer, U. S. Army Transportation School	1
U. S. Army Mobility Command	2
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Defense Documentation Center	10

Wilson, Nuttall, Raimond Engineers, Inc., Chestertown, Md., DEVELOPMENT TESTING OF A TEST-BED RIG ON AIR-SUPPORTED TREADS, TRECOM Technical Rep. 63-29, June 1963, 66 pp. (Contract DA 44-177-TC-600(T). USATRECOM Task 1D543006D40406,

Unclassified Report

A test bed for the preliminary evaluation of the potential for air-supported treads as transport elements, constructed in previous phases of this

(over)

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