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LABORATORY TESTING OF TRACKED VEHICLE SUSPENSIONS

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ABSTRACT

A unique laboratory suspension testing capability has been developed which, for the first time, enables rapid evaluation of tracked vehicle suspension components. The testing capability was stood up in the Durability Test Lab (DTL) in conjunction with the materials division, both organizations within GVSC. Testing has been ongoing, and the results of that testing are presented, current to the time of publication. Historically, laboratory component testing has been very limited due to the lack of a capability to provide relevant loading conditions. Previous testing capabilities not only were deficient in their vertical speed capability, but more importantly, lacked the ability to apply the corning forces. Further reasoning and details associated with the development of this test system are presented. This capability was developed as part of an ongoing campaign in the materials division of GVSC. The purpose of this campaign is to demonstrate and establish design standards, and develop an optimization process for vehicle components for future vehicles as well as improvements to the current fleet. The goal of this paper is to inform the military community of the existence, progress, and possibilities of this novel testing capability.

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1. MOTIVATION

Reducing the operating and support (O&S) costs of heavy vehicle suspension systems is becoming more important as the US Army shifts towards readiness as a top priority.

When parts require replacement not only should the cost of the individual part to be replaced be considered, but also the cost and limited availability of labor to install them. In some cases, like upper spindle thrust bearings on tracked vehicles, the cost of labor can easily be an order of magnitude greater than the part cost due to the significant work required.

In addition to the financial impact of wear out and replacement of parts, substantial downtime of important assets is incurred, including the vehicle and the people, which negatively affects readiness.

While solutions such as predictive maintenance do have great potential for reducing costs and improvement of readiness by optimization of the maintenance approach, it is not a means to eliminate the underlying reliability issue. Long-term fiscal and personnel constraints are unlikely to subside, particularly if the thrust to improve readiness remains. Therefore, feasibly improving reliability, by addressing root causes of the relevant parts, should also be undertaken. [1]

While weight reduction was not the primary focus of the prototype examples given in this paper, this testing capability is also essential to ensure any weight reduction and optimization efforts do not degrade reliability.

1.1. BACKGROUND

The US Army, based on various considerations, has long expressed a preference for tracked vehicles over wheeled vehicles in combat roles once gross vehicle weight exceeds 10 tons [2]. The unique benefits that tracked vehicles provide are offset by the large O&S costs and logistical challenges. This paper is focused narrowly on a major challenge in improving the reliability of military unique combat tracked vehicle suspension systems.

The major challenge historically has been the lack of a laboratory capable of providing the relevant loading conditions to elicit failure modes seen on actual vehicle components.

This paper addresses the development of a laboratory testing capability to fill the gap. This capability is a first-of-its-kind test system that can produce vertical motion at relevant loading rates while simultaneously providing cornering forces. These cornering forces have been proven necessary to excite failure modes of concern seen in actual vehicle use.

Prototype parts are mentioned as these are being tested by the described capability, but they are not the focus of this paper. For reference the prototype parts are drop in replacements for the current arms, but have been geometry optimized and are intended to eliminate field failures currently experienced. See Figure 1 for a view of the prototype arm, and the parts needed to assemble into the rest of the tank suspension.

After discussing the limitations of analytical methods on their own and the impracticality of design iteration by vehicle testing, the newly developed test capability is introduced. Some initial results in exciting structural failure modes of Abrams road arm assemblies and thrust bearing wear are described herein. In closing, a vision for future use-cases of this new and unique capability are proposed.

1.2. FAILURE MODES & PARTS OF INTEREST

There are many components involved in the Abrams suspension sub-assembly. This assembly includes road arms, bearings, seals, dampers, and springs that comprise of various metals, plastics, and rubber materials. Interactions include heavy press fits, splined interfaces, metal-to-metal wear, static O-rings, etc. Many of these interactions and components degrade or fail via highly complex phenomena, limiting the predictive nature of purely analytical Modeling and Simulation (M&S) approaches.

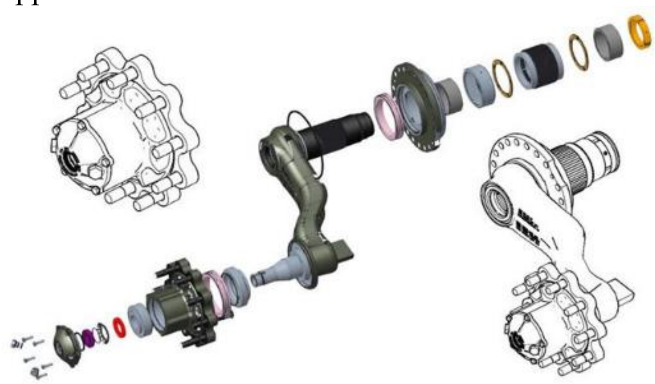

Figure 1: Exploded View of Prototype Arm and Associated Parts

An example of this is within the current road arm: there is a sharp, small radius immediately adjacent to a heavy press fit on a part with a highly complex residual stress state due to induction hardening operations.

An additional problematic area is the connection between the upper spindle to road arm which has a splined joint with a heavy level of interference immediately adjacent to previously mentioned welded joint. Because the parts are comprised of high strength steels, the weld geometry is non-traditional, and the splined joint has unknown interface conditions; welds like this are not susceptible to weld fatigue analysis (e.g., the structural stress method). Similarly, the remainder of the joint, due to large unknown assembly residual stresses, is not susceptible to fatigue analysis. To be predictive, a joint of this complexity requires calibration by component fatigue testing.

Because of the limited predictive capability of pure M&S approaches, some physical experimentation is needed, whether it be laboratory or on vehicle at proving grounds.

1.3. VEHICLE TESTING CHALLENGES

Due to past laboratory capability gaps, evaluation has typically been conducted on vehicles at the US Army`s primary proving grounds: Aberdeen, MD and Yuma, AZ. These vehicles are operated over various courses including pavement, secondary roads, and cross-country terrain.

Vehicle testing has the benefit of accurately reflecting the full physics of vehicle usage obviously. However, conducting such tests is both lengthy and costly; one vehicle evaluation typically takes several years and over a million dollars to complete.

In the case of a prototype part being found to be deficient, a revised prototype may be developed and resubmitted for vehicle testing with further delays due to gaps in test asset availability that can be several months or even years. For these reasons just one iteration of a prototype part by vehicle evaluation, can take multiple years and millions of dollars.

An additional confounding factor with vehicle evaluations is despite great lengths in standardizing testing, large variations in component durability are typical. Due to this large component life variation, unless a dramatic improvement over the baseline exists, in order to make statistically valid comparisons to current production components, multiple simultaneous vehicles must be tested over several years.

Understandably, this activity is rare and typically only conducted as part of major Reliability, Availability, and Maintainability (RAM) evaluations primarily as an assessment of the current design.

Because the high cost of performing on-vehicle testing, with continued improvement desired, many evaluations should be lumped into a vehicle test. An example of this approach is the ongoing Abrams Product Improvement Verification Testing. However, the number of components is limited, as well as the number of test vehicles, so there is preference to only conduct on-vehicle testing after analysis or laboratory testing indicates merit.

These issues make incremental improvements and iterating component designs through vehicle testing impractical.

Obviously, the vehicle itself is required in these evaluations. This is a problem for both from-scratch design prototype vehicles and even current fleet vehicles where weight is increasing to add other capabilities, because of the typically non-linear nature of failure mechanisms.

Because vehicle evaluation may not always be possible until very late in development and iterating component designs through vehicle testing is impractical due to cost and time, laboratory component level testing is generally preferred prior to any vehicle evaluations.

2. TESTING APPROACH

By establishing a test baseline with respect to failure modes of concern, prototypes can be evaluated and due to the quick turnaround and relatively low cost, conducting prototype design iterations should be practical. In addition, it is feasible to test many replicates in highly controlled loading conditions making statistically valid comparisons possible in a short amount of time.

This testing strategy is almost a foreign concept to the military. It is exploratory and iterative to excite failure modes and create a basis for comparison of alternatives rather than the typical "does it meet the requirement" testing. For the issues and components of concern no such standards exist, although the materials group is moving in that direction as previously mentioned.

2.1. TESTING SETUP

Figure 2: Test Setup in Use in the Lab

The test system which was developed to fill the capability gap detailed resides at GVSC`s DTL in Warren, MI and is called the Suspension Test Analysis Rig (STAR). It is comprised of a vertical actuator, lateral actuator, a reconfigurable t-slotted tabletop, a main fixture, and a slip table. Integrated into the main fixture is optional torsional actuator for simulating the torsion springs that are typically used in tracked vehicle suspensions.

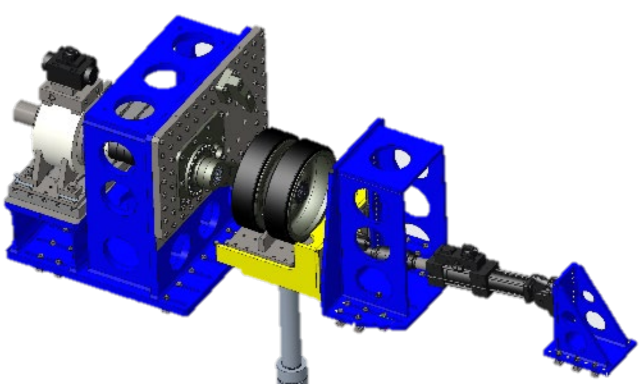

Figure 3: STAR at GVSC in Abrams configuration

The vertical actuator in this test setup is a custom actuator specifically developed to have the capability of simulating the ground forces and motion relative to the chassis experienced by heavy tracked vehicle suspensions. It is a low mass double acting actuator with 18.5 inches of dynamic travel, three valves rated at 400 gallons per minute each, and has 80 gallons of close-coupled hydraulic accumulation. It is capable of developing a maximum static force of 72kip, and a peak velocity of 11 m/s. Figure 4 shows the actuator without any fixtures above it.

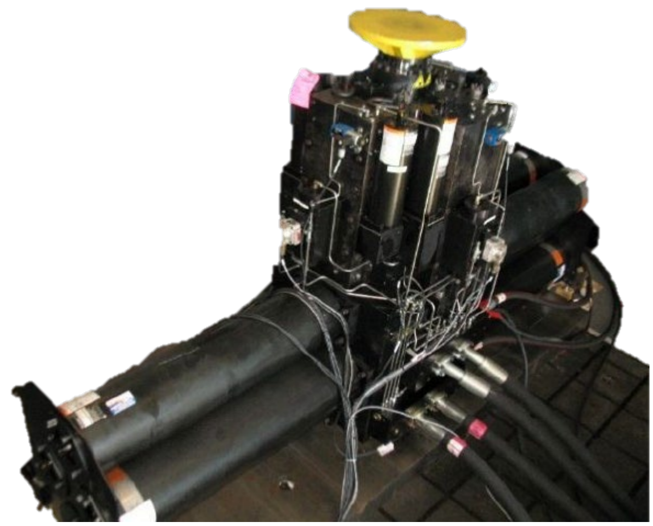

Figure 4: STAR Vertical Actuator

Mounted to the tabletop is a re-configurable main fixture that can simulate various vehicle hulls with minimal fixture costs by swapping a mounting plate and center guide simulant. This system is not only capable of testing traditional torsion bar based suspension systems but also external suspension systems.

To simulate the Abrams hull, this mounting plate replicates the hole pattern and cut out in the vehicle hull for the road arm housing as shown in Figure 5. Replicating the Bradley suspension requires both a mounting location for the road arm housing as well as the anchor for the damper as shown in Figure 6.

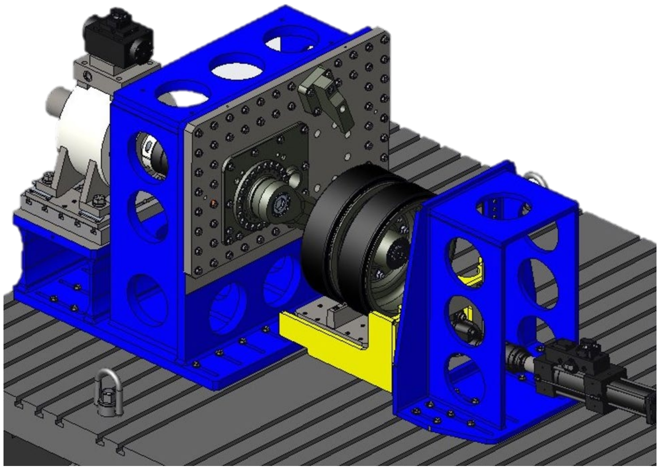

Figure 5: STAR CAD showing Abrams configuration

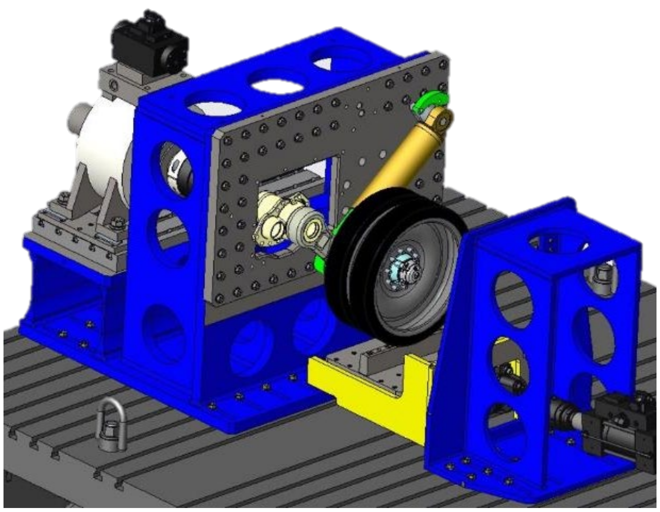

Figure 6: STAR CAD showing Bradley configuration

Rather than integrating a full-length torsion bar into the test setup, the load levels that the torsion bar provides to the road arm is simulated. This approach allows not only a more compact test system but also allows expanded range of evaluation for road arms since the torsion bar used on vehicles typically has an inferior fatigue performance.

To accomplish simulation of the torsion bar, a rotary actuator mounted atop a compliant table reacts load through a torsion bar simulant into the splines on the road arm. The control system is then configured to simulate the response of a full-length torsion bar.

Lateral loading of the trailing arm is accomplished by a slip table and lateral actuator. The slip table is mounted atop of the vertical actuator. The lower part of this slip table connects directly to the vertical actuator. The lower and upper parts of the slip table are connected via large linear bearings, allowing the upper part of the slip table to be driven by the lateral actuator. The lateral actuator is anchored to the table top via a small right-angle fixture, and mounted on spherical joints on both ends. The lateral forces are transmitted into the road wheels via a center guide simulant mounted on top of the upper slip table plate.

Lateral forces and displacements are controlled via continuous calculations using the kinematics of the vertical and lateral actuator geometry. Due to the change in lateral actuator angle relative to the ground, the actual lateral force changes through the vertical travel. Therefore, additional computed channels were created for control of the lateral loads during vertical testing.

Due to the highly non-linear nature of the road wheel to the center guide contact during lateral testing, there was another challenge with controlling the system. During load reversal, there is loss of contact and return of contact. For this reason, a special waveform was developed to conduct the testing quickly but smoothly.

In attempts to maximize the speed of testing, many waveforms were tried including triangle and pure sine waves. Eventually, a stepped slope waveform was developed, which is shown in Figure 7. This special waveform enabled not only a higher cyclic rate but also provided very smooth operation, thereby eliminating unnecessary wear and tear on the test system. This waveform provides a slow rate of loading while transitioning from contact with the inner wheel to contact with the outer wheel and vice-versa. Once contact of the wheel is reached, the loading is rapid. A short hold period at high load is included to have a high level of repeatability in achieving desired peak forces.

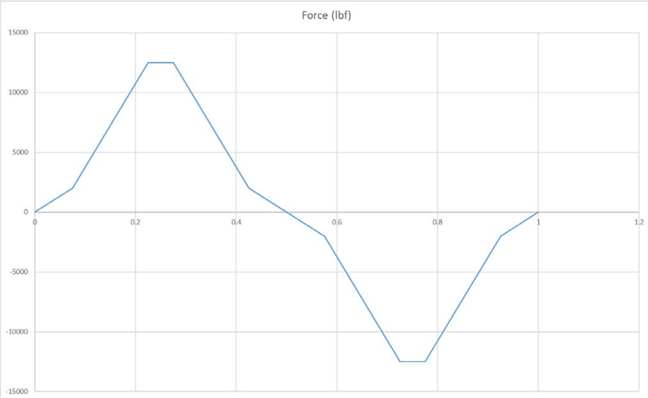

Figure 7: Lateral Loading Profile

This Multi-Degree of Freedom (MDOF) approach simulates the actual loading mechanism on suspension assemblies by loading the arm via the track in both the vertical and lateral directions.

In order to operate this MDOF test system smoothly, cascade control has been applied to all control channels to substantially smooth the response, along with the custom loading profiles, like described above. Cascade control removes the inconsistency of loading with a non-homogeneous surface interaction and stiffness [3]. Here, cascade control refers to controlling a linear actuator in displacement control and having an outer loop to adapt to force targets. Similarly for the torsional actuator, the angle and torque are cascaded rather than position and force.

Pressure variations and undesired oscillations were minimized as well via accumulator tuning, valve tuning, and PID loop tuning to conduct the testing smoothly and as fast as possible.

With these control and hydraulic system approaches applied not only was the level of consistency of loading improved but substantial wear and tear was avoided thereby preventing unnecessary cost and down time due to repairs.

In order to minimize wear and tear on the test stand, special consideration was additionally given in mechanical design to minimize uncontrolled reaction loads on the actuators.

The test setup with a general concept of the loading state is shown in Figure 8. External applied vertical and lateral actuator input forces are shown in green, test specimen two wheel vertical reactions and one lateral reaction are shown in red, and unknown bearing internal loads are shown in black. All moments except the torsion bar loading are omitted for clarity.

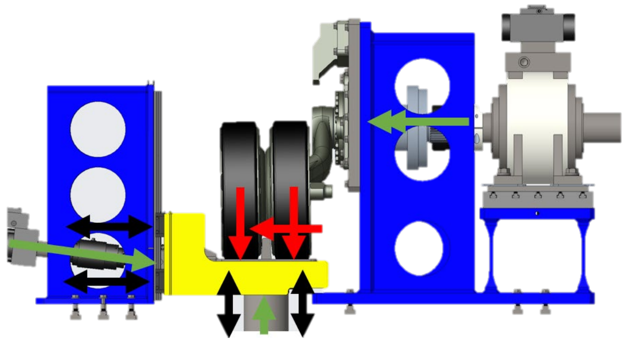
Figure 8: Load directions of test setup

These reaction loads are caused by eccentricity of road wheel loading due to fore-aft movement of the wheel set and unequal outboard-inboard wheel loading. Other loading cases which cause undesired reaction loads include non-horizontal orientation of the cornering actuator, and deflections throughout the system.

To stop side load forces being applied to the vertical actuator the slip table is supported in the vertical direction by large vertical bearing supports. Due to their high stiffness relative to the hollow vertical actuator rod, these bearings prevent eccentric wheel reaction loads from creating vertical actuator side loads and moments.

Forces and off axis moments transmitted into the torsional actuator via the torsion bar simulant must be minimized to prevent failure of the torsion bar simulant and premature wear of the actuator. To achieve this, test stand suspension housing mounting must be very stiff relative to the torsional actuator stand portion, particularly along the actuator shaft axis direction. The torsional actuator stand portion with integrated flexures by inclusion of thin webs in the axial direction of the actuator.

With the approaches described above, GVSC has developed a test stand that can replicate forces and has proven sufficiently durable, to be used to evaluate components, as intended.

3. INITIAL RESULTS

In this section, some preliminary results are presented, comparing current production road arms to prototypes under development by GVSC. These prototype components were optimized by use of FE based fatigue predictions using the same vertical and cornering fatigue conditions that are now being used in lab testing on the baseline components.

Testing began with an exploratory loading of the baseline components. A highly accelerated stepped load case was used to determine relevant single amplitude load levels, and the failure mode was found to be consistent with field failures. The load levels at failure were equivalent to expected field failure levels.

The arm was loaded in the lateral direction with vertical and torsional loads set to equivalent vehicle loads during turning. These loads are calculated from the known force application rate of the torsion bar, and the known ride height angle of the tank.

It was unknown what load levels are required to replicate the field failures, therefore exploratory testing was used with a small number of repeats. The lateral loads were fully reversed, starting at 5,000 lbf, and stopping at 18,000 lbf with each step repeated for 6,000 cycles. Each subsequent step the loading was increased by a factor of 1.2.

While 6,000 repeats is a small number, it is likely that infrequent, high overload conditions are responsible for in-field weld cracking. Therefore, the number of repeats used might not be entirely inappropriate.

3.1. WELD CRACKING AND SEAL LEAKS

The first success of this test setup was recreating the main field failure seen on the Abrams road arm sub-assembly, upper spindle weld cracking. This failure has major O&S cost implications because when this weld cracks and oil leaks, the arm must be replaced. Based on FE analysis it was expected that the cause of failure would primarily relate to alternating left and right turns while on pavement, which was borne out in lab testing. This condition was simulated by cycling the lateral actuator in a fully reversed condition with load amplitudes up to 12,490 lbf, while vertical and rotary actuators were held at a constant torque and road arm orientation. See Figure 9 for an example of weld area cracking. The load level of ~12 kip was picked due because it is the most likely worst case on the vehicle. The higher loads would almost never happen in the real world.

Figure 9: Baseline Weld Cracking

An issue with causing weld cracking was found during the baseline testing. It was found to be very inconsistent. There was high variation between the numbers of cycles to failure of the weld on baseline parts, due to inconsistent components and manufacturing issues.

While adjusting pre-loads of the system to find a load case that resulted in a more consistent failure rate, an important factor was found. The torque pre-load on the arm was found to be necessarily induce a crack. When the loads were reduced below 10,000 ft-lbf, cracks wouldn’t occur even after 150,000 cycles. When the torque was increased to 25,750 ft-lbf cracking was induced in only 10`s of thousands of repeats. This higher torque level is consistent with wheel loadings to be expected during operations on pavement for the current vehicle weight. This result provides and experimental basis for declaring that weight is a major factor in part reliability for Abrams tanks.

Both the baseline and prototype arms have been run in this lateral loading condition and for both, leaking of oil at the upper spindle location has been induced in under 24 hours of test time. For the baseline case this was the weld cracking and for the prototype case the leaking of an elastomeric seal.

Once the initial leaking of the prototype was discovered, a quick change of the sealing material to a more compliant type was attempted. This approach also had insufficient durability so a new seal geometry has been developed and will soon be evaluated in the lab. This fast-paced iterative prototyping specifically shows the advantage of lab testing.

3.2. SPLINE BURSTING

After the weld cracking failure mode had been replicated on both the baseline and prototype parts, exciting the next failure mode was desired. For production parts, this next failure mode was unknown, as the parts are replaced in the field after a weld crack occurs.

This evaluation cannot be replicated in full vehicle testing at the proving ground, because when the weld cracks oil leaks into the environment which is unacceptable. In the laboratory, this oil can easily be captured and the exploration is feasible. The merit of this exploration is understanding as vehicle weight increases what new failure modes may be excited and ideally they are eliminated prior to costly field issues occurring.

First the aforementioned “exploratory testing” was conducted: fully reversed lateral loads starting at 5,000 lbf, and stopping at 18,000 lbf with each step repeated for 6,000 cycles and a load factor between cycles of 1.2. Then testing continued at 12,490 lbf load case.

The baseline arm continued to 100,000 cycles of 12,490 lbf, so the load was stepped up. The baseline then continued to ~60,000 cycles of 15,000 lbf. The area around the weld burst, tripping the load limits on the system and stopping the test. Figure 10 shows the result of the fatigue loading.

Figure 10: Baseline Spline Burst

The same load profiles were run on the first prototype iteration. That failed at ~60,000 cycles of the 12,490 lbf load case. Figure 11 shows the prototype spline burst. This information of comparison to the baseline could be used for future design iterations and would be impossible to gain from vehicle evaluation.

Figure 11: Prototype Spline Burst

3.3. BEARING THRUST WASHER WEAR

As stated in the introduction, upper spindle thrust washers, while being low-cost brass components, are incredibly labor intensive to replace. If extending the life of these components is possible, significant savings in labor could be achieved.

Testing will be conducted to compare current production wear rates against prototypes with increased wear area and alternative grooving patterns.

In order to produce wear, the vertical actuator was cycled in a triangle wave 4 inches peak to peak with cyclic frequency of 2 Hz while simultaneously applying a lateral force of 12,490 lbf. The rotary actuator was held at 5,000 ft-lbf to keep contact between the road wheel and the slip table. By applying the lateral load in the outward direction wear is induced in the outboard bearing, and vice-versa for the inward direction and bearing. Cycling was conducted in the outward direction until sufficient wear was induced and then repeated in the inward direction.

Because the large amount of labor to disassemble and physically measure thrust washers, an indirect method of thrust washer wear was used. The gap between the arm and the stationary housing varies with loading direction and current thickness of the washer. Testing was periodically interrupted, a constant lateral outward force applied, gap measured, then constant lateral inward force applied, and gap measured. With these values, the change in thickness of the inboard and outboard trust washers can be inferred because the washers are comprised of brass while the mating surfaces are hardened steel.

Figure 12 shows example wear data collected and Figure 13 is an example of a worn thrust washer.

An unexpected result which was only possible to discover through highly controlled laboratory conditions was found. In Figure 12 the rapid wear rate labeled “Development” occurred when the torsional damper was active and the other two conditions when the damper was non-functional.

Note these bearings sit within an oil bath common to the damper on the damped stations. Because of vertical suspension travel, the damper generates significant heat and elevates the temperature of this oil.

Initially, the damper was left unmodified producing oil temperatures in excess of 250 degrees Fahrenheit and producing significant wear grooves in the brass bearings in a matter of hours. Next, the valves were removed from the dampers, eliminating the self-heating. Oil temperatures remained below 110 Fahrenheit and despite over 40 hours of testing almost negligible wear was observed. Because of the dramatic dependence of wear rate on oil temperature, temperature-controlled characterization testing will be conducted by introducing an oil heating system. Understanding this wear temperature relationship is critical to reducing bearing wear, and only possible in a laboratory setting.

Further studies to reduce this wear by alternative suspension housing designs and oil types are planned.

Figure 12: Thrust Washer Wear

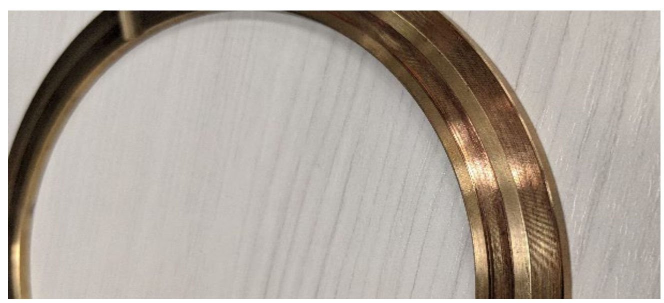

Figure 13: Thrust Washer Wear Patterns

3.4. ROAD ARM COMPLIANCE

Lastly, a result of high importance for modeling and simulation as well as running gear packaging was found. Because the force and displacement responses of the arm were measured, full assembly compliances can be known. Because this assembly is comprised of many complex contacts as well as bearings predicting this compliance accurately with pure M&S is infeasible, physical testing is required.

Compliance was calculated and reported by taking the linear portion of the loading region in both the positive and negative displacement directions and averaging these values. The unloading of the wheel slip regions were not part of the lateral stiffness fitting.

The relative stiffness of the baseline vs. prototype was calculated and graphed. It was expected that the prototype would be less stiff with the removed material, and the testing proved that, quantitatively. See Figure 14.

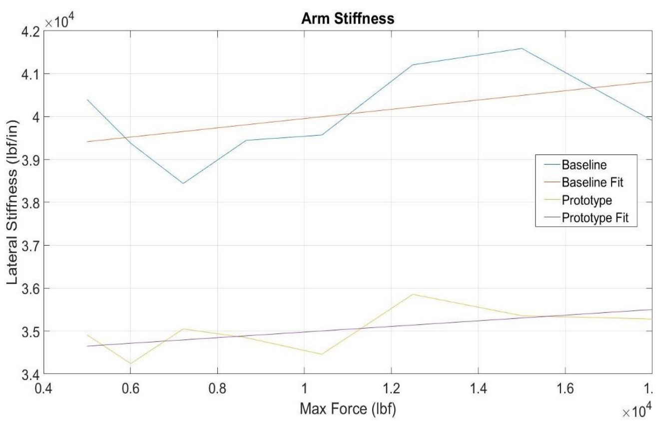

Figure 14: Stiffness of Baseline vs Prototype Arms

Figure 15 shows a full cycle of the lateral loading for the 12,490 lbf load case. The baseline and prototype responses are overlaid. Figures 16 and 17 show all the baseline load cases, and the prototype load cases, respectively.

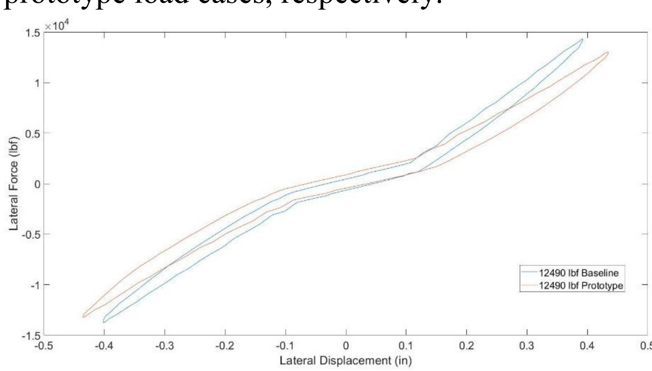

Figure 15: Baseline & Prototype Force vs. Displacement

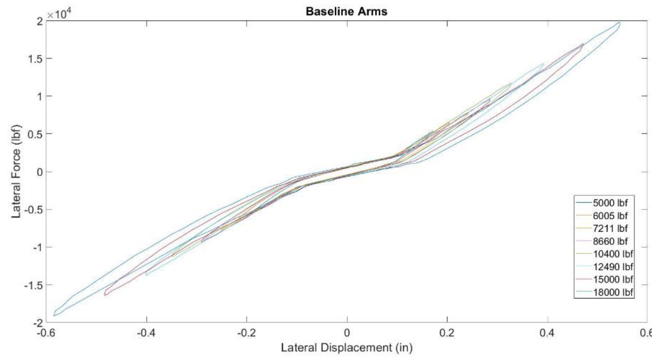

Figure 16: Baseline Load Characterization

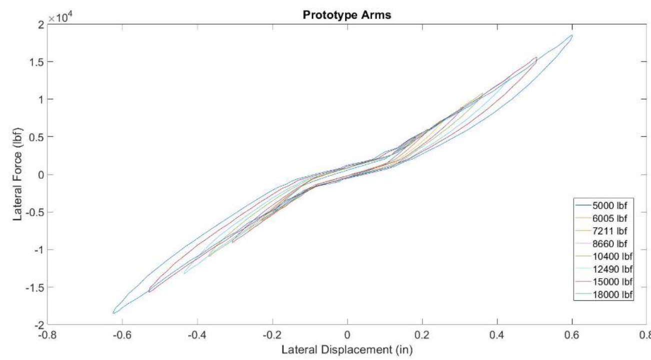

Figure 17: Prototype Load Characterization

4. FUTURE USE CASES

The fixturing and equipment were purposefully built to be cross platform compatible. Therefore, it inherently could be used for other tracked vehicles with very low fixture costs. While not the primary intent of the tester, minor fixture changes similarly permit testing of wheeled suspension systems.

While this testing may be common for light vehicles, previous trailing arm testing for heavy wheeled vehicles has been primarily limited to vertical loading conditions. Cornering forces being included in a wheeled suspension test could help target other failure modes other than just from vertical travel.

The compliance of the lateral and vertical loading has already been proven out on the Abrams setup. With additional load cells and some additional fixture modifications, a kinematics and compliance rig could be commissioned. Because this capability is applicable to tracked vehicles, this would introduce another novel capability.

Another application with minimal effort to implement is field-loading replication. This is where suspensions are instrumented, operated on vehicle test courses, and returned to the lab to simulate vehicle-loading conditions in a rapid, controlled, and cost-effective manner.

Lastly this field loading replication effort can also be used to establish the forces transmitted to vehicle hulls during usage to improve hull designs.

5. CONCLUSIONS

The testing is still ongoing, but the intent of the testing capability has been met. The re-configurability, and robustness of the test rig design have been verified.

The speed at which parts can be tested to failure, removed, new parts setup, and tests continuing, has exceeded expectations on reducing the amount of time, cost, and effort needed to perform this kind of testing. The future of component level testing vs. field testing for prototype efforts is clearly one with many opportunities.

Not only is iterative design now open to tracked suspension systems, the potential for M&S efforts to be expanded has been shown. Data from this capability, as it is now, can be used to improve models, both for simulation and for design work. As stated above, more instrumentation and fixture changes could widen the value of the output from this test capability to the M&S community, and drive improvements into the fleet.

6. REFERENCES

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