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HOLLOW ROADWHEEL DESIGN TO INCREASE PERFORMANCE AND REDUCE WEIGHT

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ABSTRACT

Reducing roadwheel weight by replacing legacy steel designs with hollow, aluminum roadwheels could save 690 lb per vehicle (a 34% reduction). Hollow roadwheels offer greater load-carrying capacity, higher stiffness, improved corrosion resistance and no debris entrapment. Two hollow roadwheel configurations, bolt-together (HB) and welded (HW), were physically tested and exceeded the radial and lateral stiffness of the legacy steel (LS) roadwheel. The HW radial stiffness was 32% higher than LS and lateral was 65% higher. The HB radial stiffness was 14% higher than LS and lateral was 73% higher. Both hollow configurations offer significant weight reduction, better performance and can be implemented in the near-term based on their ease of manufacturing and high TRL. The HB configuration has already passed testing at discrete loads of 5G radial and 4G lateral (per roadwheel) with no cracks and acceptable levels of permanent deformation. The hollow roadwheel in its current configuration is designed for vehicles with 40-50 ton GVW, but with slight modification and increased weight could be used for vehicles exceeding 50 ton.

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1. INTRODUCTION

Roadwheel weight reduction is crucial considering the legacy steel roadwheels on the M2/M3 Bradley Fighting Vehicle account for 2,011 lb [1]. Replacing them with hollow, aluminum roadwheels could save 690 lb per vehicle (a 34% reduction), while offering greater load-carrying capacity, higher stiffness, improved corrosion resistance and no debris entrapment. Additionally, the M1 Abrams main battle tank - expected to remain in service until 2050 - has already increased in weight by 23% since inception [2]. It uses the original running gear designed for a much lighter vehicle. Recent vehicle acquisitions [3] and upcoming M1 SEPv4 enhancements planned for the mid-2020s continue to push vehicle weight higher, posing serious reliability concerns. Overloaded roadwheels lead to premature failures, reduced fatigue life and increased costs for component replacement and vehicle downtime.

This paper presents the viable solution of lighter, hollow roadwheels that offer better performance than legacy designs. Evidence is presented demonstrating how hollow roadwheels are superior at resisting typical roadwheel loading. Two distinct uniaxial load cases were selected to assess the stiffness response of the legacy roadwheel versus

Distribution Statement A. Approved for public release the proposed solution. The performance benefits were initially quantified by conducting an analytical examination for each roadwheel design. Two versions of a hollow roadwheel suitable for manufacturing are examined, each consisting of two high-strength, aluminum rim halves either fastened or welded together. The first is a bolt-together roadwheel using a combination of the lug fasteners and wear ring fasteners to secure the rim halves together. One rim half is vulcanized with the roadwheel elastomeric layer. The hollow cavity is enclosed, preventing debris entrapment and can be sealed from liquid ingress / egress by incorporating elastomeric sealing rings inside the cavity around the inner and outer perimeters if so desired (e.g. for amphibious applications). The second is a welded version with sealed hollow cavity. The welded version uses a very low-temperature, solid state process to join the rim halves, resulting in low distortion and shrinkage while yielding excellent mechanical properties.

Following the analytical examination, physical test results for each hollow version are presented to demonstrate correlation between predicted results and actual test performance. These results are contrasted with those for a legacy roadwheel as a baseline of the incumbent product performance.

Last, the industrialization of hollow roadwheels is described based on their current technology readiness level (TRL). By applying mature wheel fabrication processes and joining technology to a novel design approach, this solution requires a shorter development cycle and results in a low-risk, effective solution for the end user. A life-cycle cost analysis is also included to highlight the cost per pound saved for hollow roadwheels based on durability and potential savings based on increased longevity and performance gains.

2. PROBLEM

The vast majority of roadwheels used to-date are overweight because they have a cantilever cross-section design, with outer diameter supported solely on one side - nearest the centerguide - but unsupported on the opposite side. While convenient to manufacture from a single piece of material, this geometry is inherently heavier, requiring additional material to withstand radial loading at the unsupported side. Stiffening features like a thicker cross-section, rolled flange (Figure 1) or separate welded ring on the inside diameter are employed to overcome this weakness. Other designs use a T-shaped cross-section with improved symmetrical radial support, but still require reinforcement at the edges. Both cantilever and T-shaped cross-sections have relatively low lateral stiffness and inferior ability to withstand radial edge loading. A hollow design, such as shown in Figure 1, is superior because it distributes radial loads through both vertical members which serve as load paths.

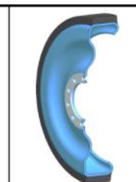
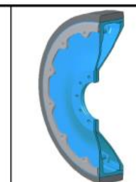
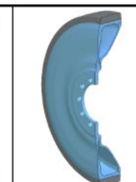
				
Description	Legacy Steel Bradley	Hollow Bolted, with wear ring	Hollow Welded	Hollow Welded, with wear ring
Abbrev.	LS	HB	HW	HW-R
Material	Steel	Aluminum	Aluminum	Aluminum
Dia. x width	24x3.8	24x3.8	24x3.8	24x3.8
Wear ring	Not required	Steel	No (metal spray)	Steel
Theo. wt. (lb)	83.8	56.9	47.4	55-57
Act. Wt. (lb)	86.2	58.2	47.7	(not available)

Figure 1: Legacy vs. hollow roadwheels.

The centerguide imparts massive loads to the roadwheel near the outer diameter generating a very large moment about the lug pattern. Cantilever geometry is unable to resist such large moments without permanent deformation and large deflection. The hollow structure in contrast is far superior at resisting lateral loads due to the inherent structural rigidity of an enclosed cavity.

3. SIMULATION

The Amphibious Assault Vehicle (AAV) Family of Vehicles uses the Bradley roadwheel 12358464 [1]. This wheel was used as the reference legacy design [denoted LS, for Legacy Steel] for comparison with two hollow designs (Figure 1). The Hutchinson hollow roadwheel designs used for comparison were a bolt-together roadwheel with wear ring [denoted HB, for Hollow Bolted] and a welded roadwheel with metal spray [denoted HW, for Hollow Welded]. All roadwheels are 24” OD with a 1” thick rubber tire. The HW roadwheel was designed for a rubber band track application and utilized metal spray in place of a steel wear ring for optimal weight savings.

A version of HW modified to accept a removeable steel wear ring is the third hollow roadwheel shown in Figure 1 [denoted HW-R, for Hollow Welded, with wear ring]. The HW-R configuration was excluded from the analysis since it is still early in development and has not yet been prototyped or tested. It is the lightest hollow roadwheel design *with removeable wear ring* at 55 lb, or 28.8 lb/roadwheel saved compared to LS.

For the purposes of this study the discrete radial and lateral load cases were selected from [1] to analyze the total deformation of each roadwheel structure. The radial and lateral stiffness response of each roadwheel were compared as an indicator of the ability of each design to withstand impact loading.

A linear elastic isotropic finite element analysis model was used to assess the structural behavior and therefore the rubber tire was omitted. A discrete radial (vertical) and lateral static load was applied to each roadwheel in separate simulations. The nominal vertical load was specified as 1G = 8,000 lbf [1]. Both roadwheels in dual configuration were included in the simulation, including a hub adapter with simplified lug nuts and studs with corresponding pretension applied. The hub adapter was fixed in all 6 degrees of freedom. The simulation configuration for each load case is shown in Figure 2. Half-model symmetry was used to reduce solve time by defining a vertical symmetry plane through the wheel axis, necessitating the radial and lateral load magnitudes to be halved.

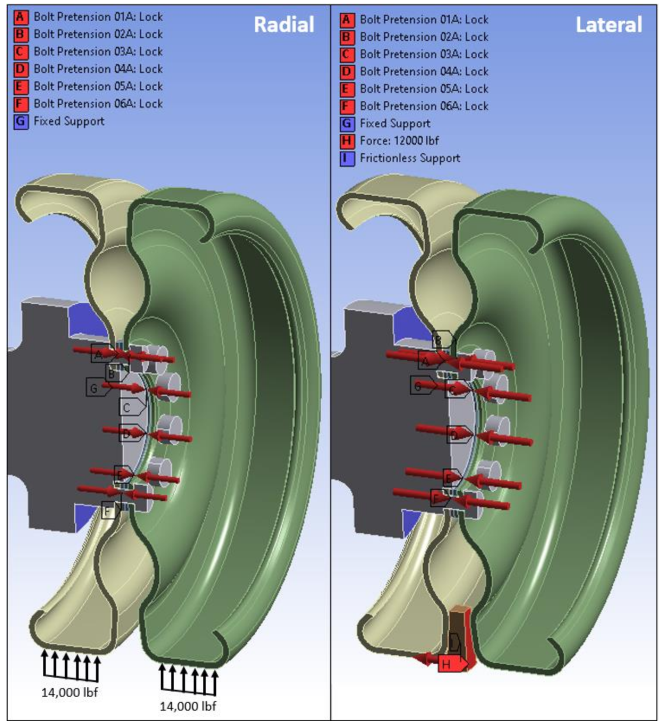


Figure 2: Simulation Configuration. Radial on left, Lateral on right. E_{STEEL} = 2.90e+07 psi, ν_{STEEL} = 0.3. E_{ALUM} = 1.03e+07 psi, ν_{ALUM} = 0.33. (ANSYS)

3.1. Radial Load Case Simulation

The rubber was replaced by an equivalent load of magnitude 28,000 lbf (3.5G) per wheel applied directly to the outer diameter of the metal wheel, shown simplified in Figure 2 as a distributed load beneath each roadwheel of magnitude 14,000 lbf. The load area width (parallel to wheel axis) was the rubber nominal contact width projected onto the metal wheel. The deformation results of the radial load simulation are shown in Figure 3 with undeformed wireframe shown in black.

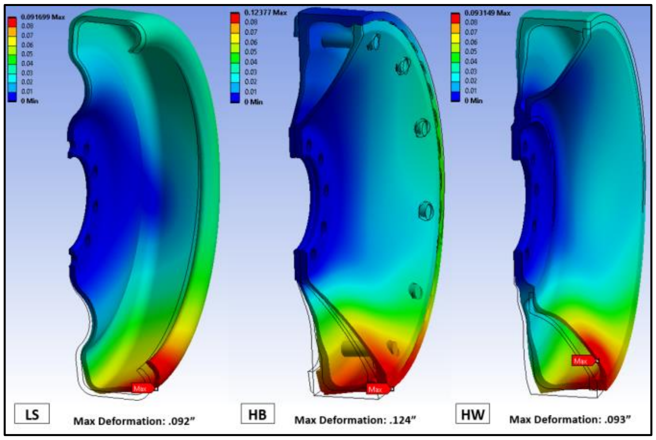


Figure 3: Radial Load Case - Deformation (outboard dual roadwheel shown)

The analysis predicted a maximum elastic deformation at the outer diameter for both LS (.092") and HB (.124"). However, HW (.093") maximum deformation was located approx. 1.75" away from the outer diameter on the conical member. HB and HW had 35% and 1% *higher* total deformation, respectively, than LS. The LS design had approximately equivalent deformation compared to HW but at a weight penalty of 36.4 lb/roadwheel. For reference, the HW-R hollow welded roadwheel with removeable steel wear ring (instead of metal spray) shown in Figure 1 weighs approximately 55-57 lb (CAD estimate), yielding a minimum weight savings of 26.8 lb/roadwheel (643 lb/vehicle).

3.2. Lateral Load Case Simulation

A lateral load of 24,000 lbf (3G) was applied to a representative steel centerguide acting on the inboard dual roadwheel only. The Lateral image in Figure 2 represents the load as 12,000 lbf (half-model symmetry). The centerguide was constrained to translation in the axial direction only. The deformation results of the lateral load simulation are shown in Figure 4, with undeformed wireframe shown in black and approximate dial indicator location as noted.

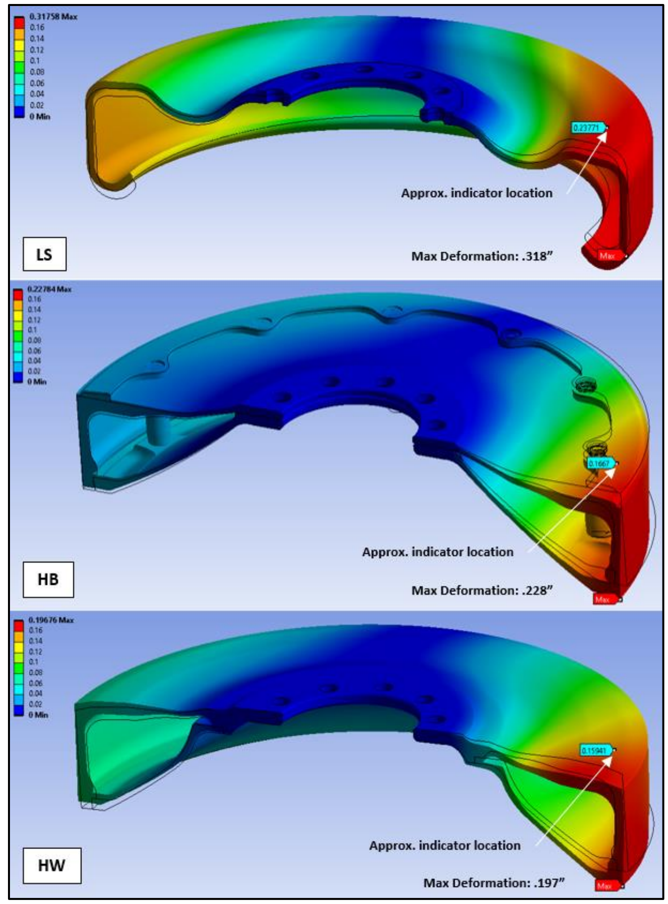


Figure 4: Lateral Load Case - Deformation (inboard dual roadwheel shown)

The maximum deformation for LS (.318"), HB (.228") and HW (.197") were all located at the outer diameter on the opposite side of the wear ring and on the symmetry plane. HB and HW had 28% and 38% less deformation, respectively, vs. LS. Also shown in Figure 4 are the deformations at the approximate dial indicator locations, for comparison with deformations observed during physical testing. These are summarized in Figure 5.

Roadwheel	Lateral FEA Total Deformation (in)		
	Max	At Indicator	Δ
LS	0.318	0.238	0.080
HB	0.228	0.167	0.061
HW	0.197	0.159	0.038

Figure 5: Lateral Deformation Summary

The deformation based on interrogating the model at the approximate indicator location was between 19-27% less than the maximum and was used for comparison to test results, to better correspond to the physical measurement location on each roadwheel.

4. TESTING

4.1. Test Equipment

A two-axis static test machine was used for the physical testing (Figure 6), consisting of a pedestal on which the roadwheel is mounted and loaded either individually in one axis or simultaneously in both axes.

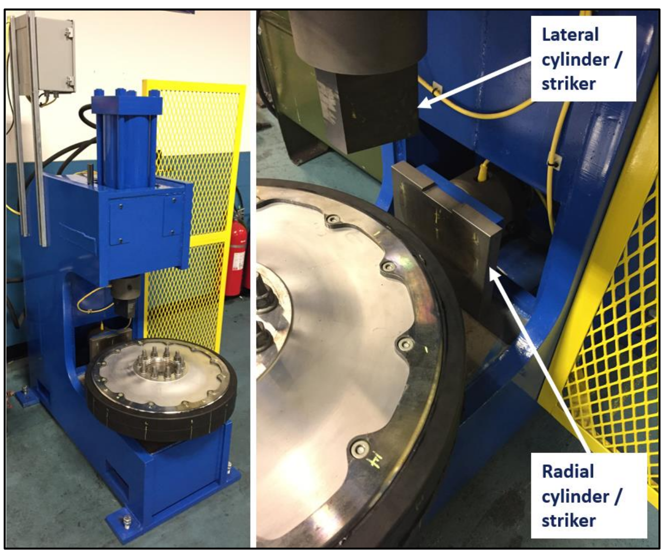


Figure 6: Static test machine

The radial hydraulic cylinder loads the outer diameter against a flat plate, imparting a static load as shown in Figure 7. Similarly, the lateral cylinder loads the wear ring surface using a simulated centerguide striker (trapezoidal shape). The centerguide striker has an inclined contact surface and has rounded edges. The centroid of the centerguide striker is approximately centered on the wear ring width. These simulate - in a static fashion - the magnitude of the dynamic load imparted during a vertical impact event on the vehicle.

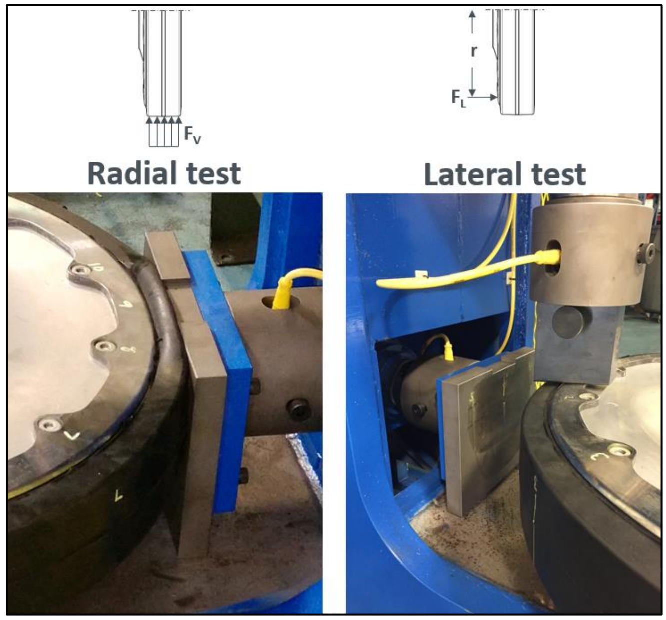


Figure 7: Example Radial and Lateral test

4.2. Test Procedure

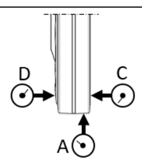
A single wheel of each type, LS, HB and HW were used for all tests. Two radial tests were conducted sequentially, followed by two lateral tests. The first test was loaded at the 6 o'clock position followed by a second test at the 12 o'clock position. Each load was held for 1 minute duration at ±1%, as registered by the load cell readout for each cylinder. After both radial tests, a dye check was performed to detect the presence of cracks prior to proceeding with the lateral tests. The lateral tests were conducted at the same positions as radial. The runout was measured at three locations (wear ring, opposite wear ring, and rubber outer diameter) on each roadwheel before and after each test to detect permanent deformation. After both lateral tests, another dye check was performed. The striker deflection was measured during each test using dial indicators mounted on the striker (radial) or just adjacent to the striker (lateral) as shown in Figure 8. Radial deflection was measured in the radial direction – perpendicular to the wheel axis, while lateral deflection was measured in the lateral direction – parallel to the wheel axis.

Figure 8: Dial Indicator Locations (for deflection measurement)

The test evaluation criteria included magnitude of permanent deformation, magnitude of deflection and the presence of cracks after testing.

4.3. Test Results

The radial and lateral test results are summarized for each roadwheel below in Figure 9, including total deflection, permanent deformation (by location) and the presence of cracks.



	Roadwheel	Deflection (in)[1]	Max. Perm. Deformation (in) Location A	Location C	Location D	Structural Cracks
Radial	LS	0.474	0.014	0.025	0.007	N
	HB	0.414	0.016	0.027	0.005	N
	HW	0.360	0.007	0.014	0.011	N
Lateral	LS	0.352	0.014	0.086	0.081	N
	HB	0.204	0.008	0.025	0.024	N
	HW	0.213	0.013	0.017	0.021	N[2]

1 - Avg deflection both tests (Radial = striker, Lateral = 2.75" from striker CL)
2 - Superficial cracks present in metal spray coating only, non-structural

Figure 9: Radial and Lateral Test Results

For the radial tests, LS had the maximum deflection in the radial direction (.474”) followed by HB (.414”) then HW (.360”). HB and HW had 13% and 24% less deflection, respectively, vs. LS. The maximum permanent deformation was observed at location C (opposite side of wear ring) in the lateral direction for all roadwheels. HB and LS had the highest at .027” and .025” respectively, whereas HW only had .014” (or 48% less than HB and 44% less than LS) at the same location. None of the wheels cracked during these tests.

For the lateral tests, LS had the maximum deflection in the axial direction (.352”) followed by HW (.213”) then HB (.204”). HB and HW had 42% and 39% less deflection, respectively, vs. LS. LS had the highest permanent deformation at .086” (location C), whereas HB had .025” (location C) and HW had .021” (location D), or 71% and 76% less than LS, respectively. None of the wheel parent material cracked during these tests, however the abrasion-resistant metal spray coating on HW had superficial non-structural cracks from both lateral tests, as noted in Figure 9 and seen at 3 and 9 o’clock in Figure 10.

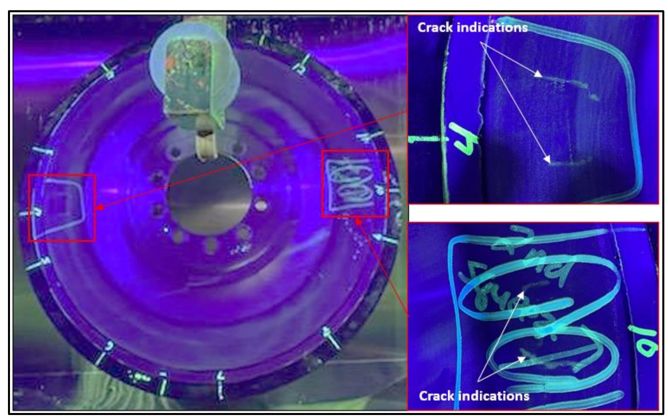


Figure 10: HW Dye Check After Lateral Test

The high hardness of the coating (>905 Vickers [64 Rockwell C]) combined with the very low thickness caused it to crack when compressed directly by the steel striker. The HW lateral test was repeated at an alternate location on the same wheel, except using an intermediate 0.5” thick rubber spacer between the striker and wheel surface. The repeated test with spacer resulted in no cracks after dye check, and only discolored dark streaks / lines from rubber contamination under the striker contact area (Figure 11). Although the HW roadwheel with metal spray was originally designed for use with a rubber band track, the structure was able to withstand the extreme lateral impact load case endured by both the legacy steel roadwheel (LS) and hollow aluminum roadwheel with steel wear ring (HB). Potentially some modifications could be made to the coating, substrate, or both to prevent cracks during this test, which may allow the HW to be used with metal spray for steel track applications.

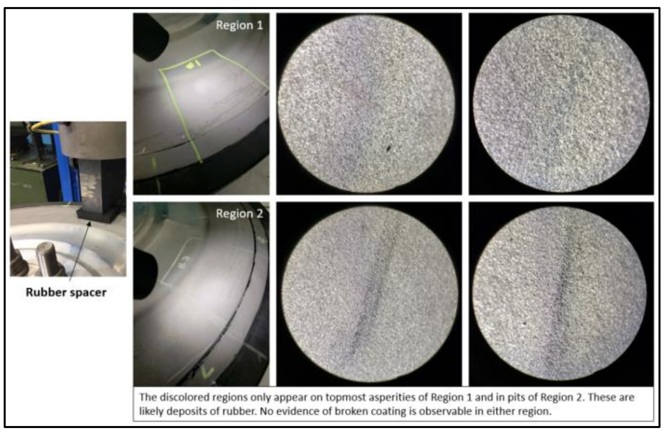


Figure 11: Lateral retest with rubber spacer. Dark lines observed post-test show no indication of cracks or broken coating

In summary, the predicted equivalent stiffness response from FEA is compared to the actual response observed during testing in Figure 12. The equivalent stiffness simply equals the test load divided by the total deflection. This represents the combined response of the roadwheel under load (elastic and plastic deformation of the structure, plus elastomer compression for radial).

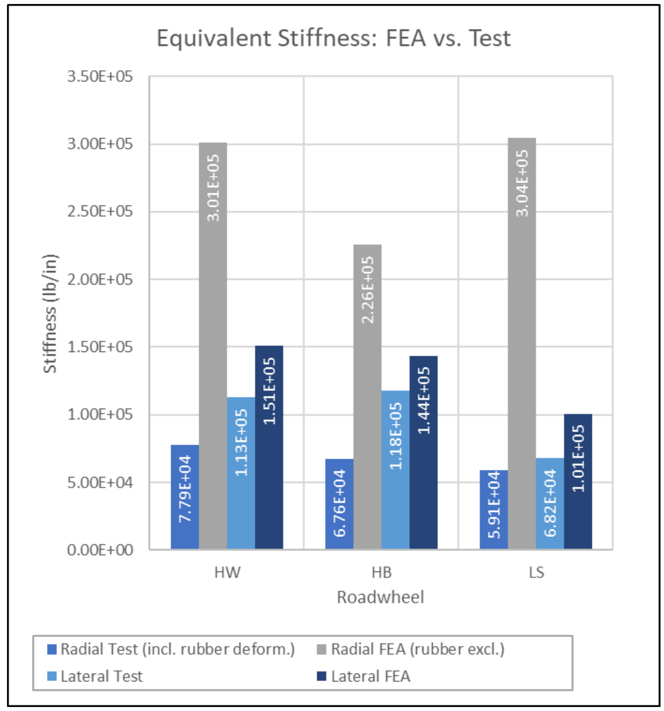


Figure 12: Equivalent Radial and Lateral Stiffness Response

When comparing predicted radial stiffness to actual, omission of the rubber compression resulted in a significant gap between predicted stiffness vs. actual, as expected. This highlights the importance of including the nonlinear elastomer response in the simulation for the radial load case simulation. For our comparison purposes, the FEA correctly predicted HW to have a higher stiffness than HB, confirmed by the test result. However, LS had the highest predicted stiffness but was lowest during testing at 24% lower than HW and 13% lower than HB.

When comparing predicted lateral stiffness to actual, the predicted stiffness was higher than actual for all three roadwheels. The predicted stiffness was also highest for HW, however the test indicated that HB had a slightly higher stiffness than HW. LS had the lowest actual equivalent stiffness response at 39% below HW and 42% below HB.

The combined equivalent stiffness response based on test results for each roadwheel (radial and lateral) are shown in Figure 13. Both hollow roadwheels exceeded the threshold radial and lateral stiffness for LS. HW radial stiffness was 32% higher than LS and lateral was 65% higher. HB radial stiffness was 14% higher than LS and lateral was 73% higher.

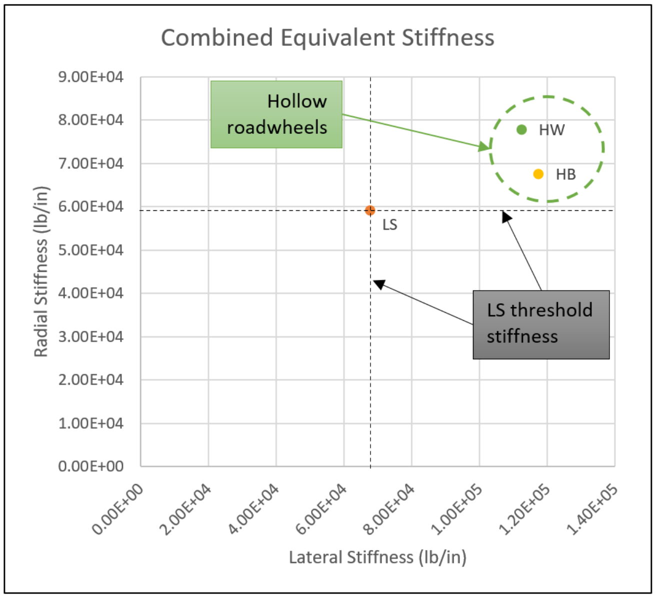


Figure 13: Combined Equivalent Stiffness Response - Radial vs. Lateral (results from test)

5. INDUSTRIALIZATION

5.1. Technology Maturity

Hutchinson Defense and Mobility Systems' core competencies include the design and manufacture of runflats, beadlocks, tire shields, anti-vibration mounts, and vulcanized tracked vehicle components including track pads, track pins and roadwheels. Hutchinson has also been designing and manufacturing lightweight two-piece bolt-together aluminum wheels for use with pneumatic tires in wheel-tire assemblies for over thirty years. Applying this Engineering knowledge and manufacturing capability to roadwheels resulted in the HB and HW novel roadwheel designs.

The HB roadwheel currently has a TRL of 6-7. It has successfully completed lab testing and roadwheels made with production-level tooling are currently undergoing vehicle durability testing.

The HW roadwheel similarly uses two halves joined together, however the halves are first permanently joined using Friction Stir Welding (FSW) creating a sealed cavity before the elastomer is vulcanized to the outer diameter. FSW is a well-known, mature technology invented in 1991 and currently used for numerous applications in aerospace, shipbuilding and automotive [4]. Tier 1 automotive suppliers currently use FSW for production of high-volume passenger car aluminum wheels and seat frames. The HW roadwheel has a slightly lower TRL of 5-6 and is currently undergoing lab testing to optimize the fatigue performance.

Both the HW and HB roadwheels are near-term viable production solutions. Hutchinson currently has the capability and capacity to manufacture these in Trenton, NJ.

5.2. Lifecycle Economic Analysis

A roadwheel lifecycle cost estimate is shown below in Table 1. The table allows for substituting either of the two hollow roadwheel solutions (HB and HW) for the legacy steel (LS) roadwheel. Individual roadwheel weights shown were those measured by Hutchinson (Figure 1) as worst-case, since they exceeded the theoretical roadwheel weights.

The estimated lifecycle cost for each roadwheel is calculated based on the individual component costs (roadwheel, rubberization, wear ring and metal spray) and their respective estimated life. For example, for LS an estimated life of 2,000 miles was used for both roadwheel and rubberization. Therefore two roadwheels with rubberization are required to achieve the 4,000 mile total lifecycle, or twice the component cost.

A lifecycle of 4,000 miles was selected based on the HB roadwheel requirement. The HB roadwheel is currently undergoing vehicle durability testing and surpassed 1,200 miles earlier this year, continuing onto the estimated life of 4,000 miles based on customer requirements. The HW roadwheel vehicle durability testing was planned for 2020 but has been delayed due to vehicle availability and is awaiting rescheduling. Based on analytical and lab test results, the HW estimated life is equal to or greater than the HB roadwheel. Hutchinson previously tested the metal spray used on the HW roadwheel but applied to another aluminum roadwheel. It completed approximately 1,100 miles of vehicle durability testing on a rubber-tracked vehicle and was in almost new condition at the end of test. Based on this, the metal spray estimated life is 4,000 miles, however it can be reapplied if required to extend the service life.

Total Lifecycle: 4,000 miles

Roadwheel	Wt. (lb)	Component	Est. Life (mi)	Est. cost	Est. lifecycle cost[4]
LS	86.2	Roadwheel	2000	$ 315	$ 630
		Rubberization[3]	2000	$ 140	$ 280
		Total		**$ 455**	**$ 910**
HB[1]	58.2	Roadwheel	4000	$ 1,030	$ 1,030
		Wear ring	4000	$ 140	$ 140
		Rubberization[3]	2000	$ 140	$ 280
		Total		**$ 1,310**	**$ 1,450**
HW[2]	47.7	Roadwheel	4000	$ 1,060	$ 1,060
		Metal spray	4000	$ 140	$ 140
		Rubberization[3]	2000	$ 140	$ 280
		Total		**$ 1,340**	**$ 1,480**

1 - Assume HB is used with a steel track.
2 - Assume HW is used with a rubber band track.
3 - Rubberization cost and performance normalized based on common legacy rubber compound.
4 - Total lifecycle cost excludes maintenance and downtime.

Table 1: Lifecycle cost estimate based on 4,000 miles

As an alternate lifecycle analysis reference, Patria in [2] indicates that a 5X roadwheel durability improvement facilitates replacing the incumbent roadwheel with a more expensive roadwheel, while still realizing a lifecycle cost savings using the example of a 5-year analysis.

Not included in the above estimate are tangible improvements in vehicle performance. In [5], Hart and Gerth found through simulation that an M2A3 Bradley *with 15% weight reduction* demonstrated significant improvement in speed on grade and fuel economy. Soft soil traversability improved by as much as 16%. Combat effectiveness improved by up to 63%, while maximum and average speed in kill zone increased by up to 10% and 8%, respectively. Also hits sustained reduced by as much as 18%. To assess operational energy, Hart and Gerth also conducted an analysis using a 10 day Major Combat Operations (MCO) model for Armored Brigade Combat Teams (ABCT). The scenarios included an ABCT with a total of 212 vehicles, 87 M1A2 Abrams and 125 M2A3 Bradley vehicles, each at 85% GVW. They found the primary impact was on overall fuel consumption, with the vehicles at 85% GVW requiring 8,000 fewer gallons of JP-8 diesel. This 1.2% reduction in operational energy for the entire ABCT equated to 6 fewer fuel trucks. In their model, the M2A3 spent only a fraction of the time in motion versus idle, however the 15% lighter M2A3 Bradley corresponded to a 14% reduction in fuel consumption. Replacing 24 legacy steel roadwheels (83.8 lb each) with lighter hollow HW-R roadwheels (55 lb each, est.) reduces the vehicle weight by 0.86% (690 lb for a 40 ton vehicle) - not 15% as in Hart's simulation – but could be combined with other weight-reduction efforts to help realize the aforementioned performance enhancements and reduced fuel consumption.

Alternatively, the 690 lb saved by using lighter hollow roadwheels could be reallocated to combat multipliers, such as additional sensors, munitions or protective systems while maintaining the same vehicle performance.

Additionally, the debris-shedding smooth exterior surfaces of the hollow roadwheel design prevents mud and debris from becoming entrapped, unlike the open cavity of the M1 Abrams roadwheel

[Figure 14]. The added weight of retained mud or clay negates other weight-saving efforts, further reducing fuel economy and vehicle performance while being labor intensive to remove – especially from the obscured inboard dual roadwheel positions.

Figure 14: M1 Abrams with mud lodged inside roadwheels

6. CONCLUSIONS

Test results for the bolt-together (HB) and welded (HW) hollow roadwheels exceeded the threshold radial and lateral stiffness for LS. HW radial stiffness was 32% higher than LS and lateral was 65% higher. HB radial stiffness was 14% higher than LS and lateral was 73% higher. Both configurations offer significant weight reduction with better performance. They can also be implemented in the near-term based on their ease of manufacturing and high TRL. While the load cases presented in this paper were related to the AAV platform [1], the HB configuration has already passed testing at discrete loads of 5G radial and 4G lateral (per roadwheel) with no cracks and acceptable levels of permanent deformation. The hollow roadwheel in its current configuration is designed for vehicles with 40-50 ton GVW, but with slight modification and increased weight could be used for vehicles exceeding 50 ton.

7. ACKNOWLEDGEMENTS

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