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Forensic Engineering Investigation of Vehicle Wheel Separations

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Keywords

Wheel, separation, axle, lugnut, hub, trailer, forensic engineer

Abstract

Vehicle wheel fastenings are deceptively complex systems. Automobile & truck manufacturers closely specify these components and test them extensively. Conversely, trailer manufacturers typically purchase their running gear complete from various suppliers, and commonly perform less in-house engineering. This case involved a trailer wheel separation. Investigation showed a dearth of relevant published research, and the adverse experts made a variety of conflicting assertions. Ultimately, extensive testing was done using computerized biaxial load application with custom stud tension sensing/release apparatus. This paper will discuss the incident-specific nature of wheel separations, the complex interactions between the components, and the legal issues involved in this case.

Background of the incident

A horse breeder purchased a new four-horse trailer from a dealer in August of 2005. The trailer was manufactured in late November 2004 and remained unsold outside on the manufacturer's lot until July 2005. A transportation subcontractor then towed the trailer 1300 miles from the manufacturer to the selling dealer. After the purchase, the trailer was towed 70 miles to the new owner's ranch. The following day, it was loaded with three 1000 pound horses and 400 pounds of supplies, and the owners left for a horse show. That day, the owners had towed the now-loaded trailer approximately 160 miles, 40 of those miles on twisty rural highways, and they were on an interstate highway when they were alerted to stop by a passing car. The owners stopped on the shoulder and found that the left front wheel/tire assembly had separated from the trailer. After exiting the Interstate (with the trailer on three wheels), they drove to the nearby highway patrol post. The highway patrol informed the trailer's owners that the wheel had hit another vehicle going the other direction on the Interstate. The wheel and tire assembly had impacted the Plaintiff's automobile, causing him injury. The hub was found to have four (of eight) broken studs, and lugnuts were missing from the remaining four studs. The lugnuts on the other wheels

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were reported as "loose", though testimony reflects the wheels were torqued by the manufacturer and several other parties prior to the sale.

Facts about the trailer

- Gross Vehicle Weight Rating (GVWR): 12,160 pounds
- Empty weight: 6,160 pounds
- Axle type (two axles): rubber torsional springs, rated load 7,000 pounds each, electric-actuation drum brakes with integral hubs, sealed bearings, eight pressed-in ½-20 studs on a 6.5 inch diameter bolt circle, stud-centric design, effective stretchable stud length 0.9 inches.
- Wheels & tires: cast aluminum wheels with 60 degree conical lugnut seats, mounted with LT 235/85R16 tires. The wheel design was made by a major manufacturer exclusively for a large wholesale distributor.
- Lugnuts: ½-20 chrome plated heat-treated steel acorn nuts with conical seats. The four sets of lugnuts were provided with the four wheels, mounted tires, and hubcaps in a palletized "kit" assembled by the wholesale distributor.

Preliminary Investigation

The author's initial inspection of the trailer occurred in February of 2006, six months after the incident – the investigation was on behalf of the insurance company for the trailer manufacturer. Observations included the following (see Figures 1 – 7):

- The four unbroken studs had accumulations of aluminum pressed into the threads, and the four broken studs had all broken at the hub face surface. Three of the broken studs were adjacent to each other. There were eleven threads of lugnut engagement on the studs.
- The hub face had a scalloped design which appeared to offer reduced and discontinuous support to the wheel, compared to the more typical circular contact surface.
- The wheel's hub mounting face featured a slight recess inboard of the bolt circle centerline. There was evidence of black paint accumulation on the hub contact surface of the wheel. The four lug holes corresponding to the unbroken studs were deformed and abraded.
- Several of the conical nutseats in the subject wheel appeared to show embedment of the lugnut's hex shape in the aluminum. The effective lugnut contact surface (observable in the remaining wheels) seemed minimal. Several of the conical nutseats also showed evidence of contamination with unknown substances.

Figure 1
subject trailer at initial inspection

Figure 2
left front hub and left rear wheel

Figure 3
damaged left front hub

Figure 4
nutseats of separated wheel

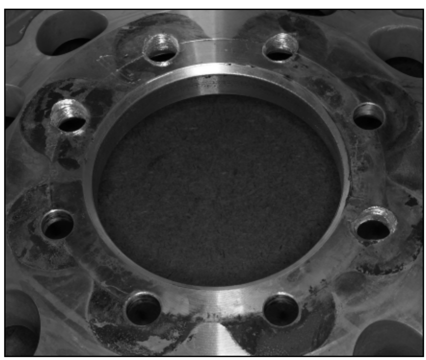
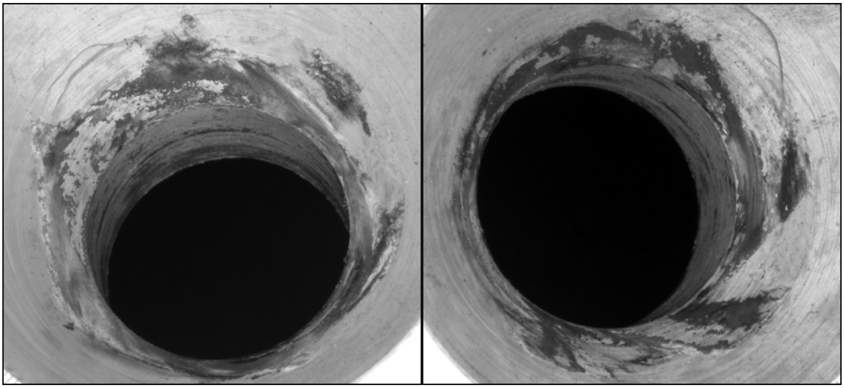


Figure 5
mounting face of separated wheel



Figures 6 & 7
apparent lugnut hex markings in nutseats

Overview of wheel fastening design concepts

- Briefly, wheels for cars, light trucks, and consumer-grade trailers (such as the subject trailer) typically use one of two systems for holding wheels onto axles.
  - Hub-centric: these systems rely on a wheel center hole that fits closely on a cylindrical hub protrusion. This indexing provides radial positioning, and flat-washer-equipped lugnuts press the wheel axially against the hub.
  - Stud-centric: these systems rely on the lugnut/wheel interface to provide both radial and axial positioning. Typically, and consistent with this case, a 60º conical surface is formed on the lugnut and in the wheel's nutseat, and the lugnut installation on the studs provides the wheel positioning. Similarly, wheel bolts may be used which have the 60° bearing surface under the bolt head and which screw into the hub. Wheel bolts are less common; for the purposes of this discussion, we will refer henceforth only to studs when discussing the male threaded portion of the fastening system, with the understanding that wheel bolts will have similar issues.
- Contrary to common perception, a properly designed and tightened wheel fastening system does not rely on the hub-centric cylindrical protrusion or the stud-centric conical nutseats to maintain the radial position of the wheel relative to the hub. Such reliance would impart cyclical bending loads to the cylindrical protrusion or the studs – this can rapidly lead to the types of fatigue failure associated with fully reversed loading. Instead, the goal of both hub-centric and stud-centric wheel fastening systems is to clamp the wheel's mounting face against the hub's mounting face with enough normal force that the radial loads encountered in use do not exceed the limits of the static friction between the clamped mating surfaces. In other words, there

should never be radial (sliding) relative motion between the hub and wheel mounting faces, which would apply cyclical bending loads to the studs. At the same time, lateral wheel loads cause cyclical variance in the axial stud load, so studs must be preloaded (through tightening) to ensure that these cyclical loads are accommodated well into the elastic range of deformation of the studs and the clamped material of the wheel.

- The application of torque to lugnuts is the method used to stretch the stud and apply preload to the joint. However, only about 10% of the applied torque actually goes into stretching the stud. The majority of applied torque goes into overcoming friction in the threads (~40%) and friction between the nut and the object being clamped (~50%)[1]. There are many variables that affect this friction[2, 3] and requirements for lugnut torque delivery are established with the expectation that the majority of torque application will go into overcoming a somewhat unpredictable amount of friction. Another issue is potential lubrication of the wheel fasteners by the consumer, as the preload delivery that can be expected for unlubricated fasteners is lower than can be expected for lubricated fasteners. Fastening system designers (for mass-produced products) must design for the majority of consumers (who don't typically think about the fastener), but must also worry about the minority of consumers, who might think to lubricate the fastener and thereby possibly overstress the fastener. Regardless, the goal is the generation and maintenance (over time) of adequate stud tension. The typical means for facilitating this (torque application), however, is a somewhat indirect and potentially inaccurate method. The technology simply isn't in place in the market to reliably measure stud preload in production vehicles.

- The fastening system designer for a wheel/hub attachment must expect that a variety of torque delivery tools may be used on the lugnuts to provide the fastener preload. Click-type torque wrenches have typical stated torque application accuracies of +/- 6%, while the more expensive dial-type and digital torque wrenches are usually reported as +/- 2% or 1%. Studies of preload scatter (the difference in preload among a number of identical fasteners tightened the same way) have shown that there may be as much as 35% variation in preload for fasteners tightened with click-type torque wrenches[4]. Not only is there uncertainty in the stated calibration, most torque wrenches do not receive periodic re-calibration. Research in this case did not reveal any published studies of uncalibrated torque wrench accuracy variances, but wheel/hub joint designers must expect that torque wrenches of various accuracies will be used on their products. There are a number of sophisticated methods of applying torque more accurately to a joint[5], but they tend to be too novel, complex or expensive to use in settings such as the manufacturing of consumer-grade trailers.

Causes of wheel separations

This paper focuses on wheel separations; hub separations typically also include attached wheels, but hub separations are not discussed in this paper. Also, this case involved a hub with an integral brake drum, but other designs have separate brake drums. Potential causes of wheel separations include (but are not limited to):

- **General design defects:** insufficient engineering factor of safety in the design of component strength for foreseeable loads, tolerance stack-up errors, drawing errors, designed-in stress concentrations, insufficient corrosion prevention or protection, failure to conduct appropriate testing.
- **Material defects:** composition, porosity, voids, grain structure, inclusions, hardness, heat treatment, surface roughness and microfinish, coating formulations and application, lubricant formulations, bonding of dissimilar materials in composites, failure to detect such defects through quality control.
- **Component fabrication defects:** tool chatter, welding defects, stress concentrations due to fabrication, casting/forging flash, as-fabricated surface roughness/microfinish, parts out of dimensional tolerance, positional errors in machining of components, failure to detect such defects through quality control.
- **Wheel / hub / brake drum joint:** insufficient thread engagement of male/female threaded wheel fasteners, incorrect and/or non-uniform torque application, incorrect amount of friction in contacts between lugnuts and stud threads and/or nutseat surfaces, contaminations or excessive coatings between mated axle mounting faces / wheels / brake drums, spinning of pressed-in studs within the hub, incomplete pressing-in of studs into hubs, mechanical interference between components, improper wheel offset, use of inappropriate aftermarket components.
- **Usage conditions:** overloading of axles, harsh usage on atypically rough roadways, usage in unusually abrasive or corrosive environments, inadequate preventive maintenance, failures to inspect.

Summary of joint inspections

The following are summaries of joint inspection results and material analyses of the subject axle and wheel assemblies.

- Axle assemblies
  - Studs: Consistent with Society of Automotive Engineers (SAE) Grade 8, with a knurled shoulder, ½-20 thread, and finished with a black phosphate/oil coating. See Figure 8. Such studs have a recommended proof strength of 19,200 pounds[6]. The broken studs had each fractured at the first thread, at the base of the stronger shoulder section. The fracture

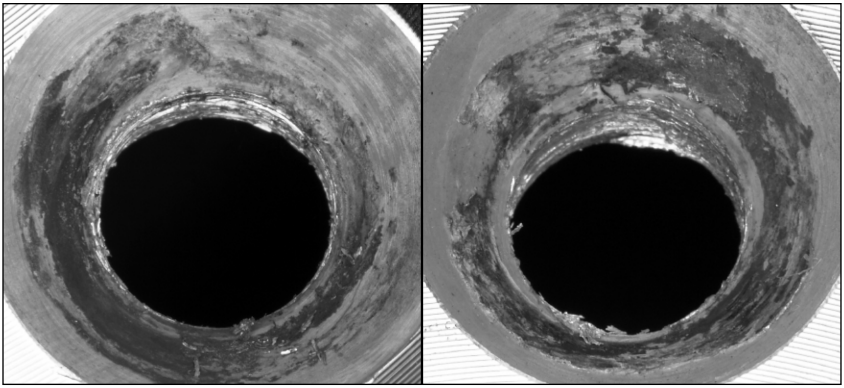
surfaces showed evidence of cyclical reversed bending and the associated fatigue fractures, consistent with a loss of stud clamp force between the wheel and the hub. A sufficient distance was traveled by the loaded trailer that the fatigue fractures could have initiated as early as the beginning of that first loaded trip.

Figure 8
exemplar stud

- Paint: There was a coating of black paint applied to the assembled cast-iron hub and studs by the axle manufacturer. The paint had a measured thickness of approximately 0.001-0.003 inches. This paint showed evidence of having been compressed in the contact area between the hub and the wheel. The studs had enough paint on them that some curls of paint were observed on photographs of the stud threads and inside the lugnuts of the remaining trailer wheels. However, the thickness of the paint on the threads could not be determined, since it had been cleaned off for earlier metallurgical analysis of the studs.
- Hubcaps: There was no evidence of the hubcaps interfering with the wheels or hubs.

- Wheels
  - The castings were aluminum A356, hardness 88.4 HB, comparable with typical 80 HB hardness for T6 temper[7]. The anti-corrosion coating was styrenated acrylic "clearcoat". Galling of lugnut contact surfaces was visible in most wheel nutseats. The roughly hexagonal indentations described earlier were found in the four nutseats corresponding to the broken studs. Many nutseats showed discoloration and contamination with substances that unfortunately were not analyzed for composition during the joint inspections. The nutseat design did not feature the counterbores or steel inserts that are common with aluminum wheels. The 60º countersink of the nutseat continued out to the wheel's exterior face. See Figures 9 – 10.

- Lugnuts
  - Low carbon (AISI 1015) steel, with a nominal 13/16" hex, case hardened to approximately Rockwell C32, and chrome plated. The lugnuts were not of the design typically used with aluminum wheels, known as "bulge" lugnuts. Lugnuts instead were a design we will comparatively refer to as "non-bulge" lugnuts, which are sometimes referred to as a "steel wheel" lugnuts. See Figure 11 for comparative views of these lugnut types.



Figures 9 & 10
discolored and contaminated substances in nutseats of separated wheel

These non-bulge lugnuts have a discontinuous conical surface of contact (with the wheel) due to the cone surface's intersection with the hex flats, and sharp "points" exist at these intersections. The hex points of the subject non-bulge lugnuts (from the remaining wheels) showed galling and accumulations of wheel aluminum. See Figures 12 & 13.

Figure 11
bulge lugnut (left) and non-bulge lugnut

Discussion of initial analysis

- Lugnut/wheel contact surface shape
  - The subject non-bulge 60º conical-seat lugnuts are the type typically used with steel wheels. When used with the aluminum wheel at issue, which features a conical seat approximately 1/2 inch deep with an approximately 5/8 inch diameter stud hole, two concerns arise. First, the effective contact surface area between the lugnut and the wheel is relatively small, leading to high localized stresses in the aluminum and the opportunity for plastic deformation and galling. Secondly, and of greater concern, is that the "points" of the lugnut's hexagonal wrenching flats provide six lobes that will dig in to the aluminum wheel's nut seating surface during torque application. As mentioned, this increases the friction between the lugnut and the wheel, reducing the stud preload in the joint for the given torque

Figure 12
subject lugnuts with aluminum transfer to hex points

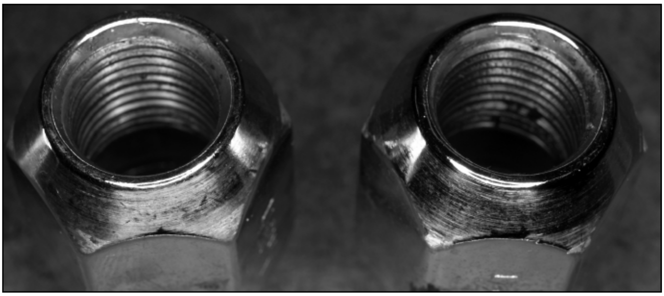


Figure 13
subject lugnuts with aluminum transfer to hex points

application. This will also lead to progressive degradation of the aluminum nut seat surface, with repeated tightening.

- Aluminum wheels are typically used instead with bulge lugnuts that feature, through various means, a separation of the hexagonal "points" from the conical seating surface of the lugnut. Bulge lugnuts also feature an increased major diameter for the conical surface, larger than the hex, which increases the contact surface area with the wheel.
- The SAE standard for the mechanical properties of nuts, J995[8], uses as a reference for nut geometry ASME/ANSI B18.2.2, entitled *Square and Hex Nuts (Inch Sizes)*. Though this standard does not include conical seat nuts, a common design element of the nut geometries detailed in this standard is that the bearing surfaces of the nuts have either a chamfer or a machined 0.016" shoulder feature, which serves to eliminate the contact of the hex points with the mating component[9]. Clearly, this is a known functionality goal in nut design.

- Lugnut/wheel contact surface area
  - As mentioned, bulge lugnuts have an increased conical contact surface area, compared to non-bulge lugnuts. This increased surface area is intentional, due to concerns about exceeding the compressive yield strength of the aluminum nut seat. Stud tension loads and torque loads can be significant, and the increased contact surface area of bulge lugnuts allows the tension and torque loading to be distributed over a larger area of the aluminum. If the compressive yield strength of the aluminum is exceeded, plastic deformation of the nut seat can be expected, with associated residual stud preload relaxation, generation of nut seat surface discontinuities, galling, and potential relative movement of the joint. There is an SAE Recommended Practice, J2315 "Wheel Nut Seat Strength", that provides a method for analyzing the likelihood of such a failure, in a section called "Bearing Surface Recommendation" [10]. Using this recommended bearing surface analysis method to evaluate the pairing of subject wheel and lugnut, the J2315 formula recommended a contact surface area of 0.685 square inches (at a preload of 80% of stud proof strength), when in fact the subject wheel/lugnut surface area is only 0.504 square inches. The subject bearing surface area is 74% of the recommended surface area.
  - Referencing the preceding section, it should be noted that extensive research was conducted to seek out additional published methods of calculating the effects of what appeared to be minimal lugnut/wheel contact surface area – few resources were found. Additionally, among the literature and published standards, any mention of conical contact surfaces proved to be very rare indeed for fasteners, though certainly not for sealing and fluid transfer fittings – which aren't directly comparable. One well-known wheel testing consulting firm suggested that the stud tension divided by the *projected* contact surface area would provide meaningful results for imparted compressive stress. However, due to the 60° cone angle of these fasteners, the actual contact surface area is twice the projected surface area, and it also seemed unlikely that the force vectors associated with such a steep angle would have an insignificant effect on the imparted compressive stress.

- Wheel/hub contact surface shapes
  - The petal-shaped scalloped surface of the cast iron hub provides discontinuous support for the wheel's mounting face, which could lead to "unexpected" stress distributions within the wheel. The 2007 revision of SAE Recommended Practice J694, in fact, introduced a recommendation against the use of scalloped hub faces in commercial truck/bus applications similar (though larger) to this one; reportedly, cyclical fatigue crack-

ing in disc wheels has been attributed to the scalloped surfaces[11]. However, fatigue cracking of the wheel was not the issue in this case; components are smaller, loads are lighter, and there was no correlation established between the scalloped surface and the loss of stud tension.

- The machined 0.010" step on the face of the subject wheel serves two primary functions; one is to move the clamping force profile radially outward, broadening the base of support for operational loads[12], and the other is to attempt to provide a "Belleville washer" effect and enhance joint compliance in the stud. A Belleville washer is an almost-flat washer with a slight conical shape that provides an axial spring tension in a fastener assembly. It should be noted that the "Belleville washer" geometry has been a formed-in feature of steel wheels for many years.

- Wheel/hub contact surface areas – as machined
  - The combination of the scalloped hub face and the machined wheel face indeed led to a contact surface area between the wheel and hub smaller than if the hub and wheel faces had not been so manufactured. However, there was no published research found that studied these issues, and there was no evidence of significant wheel face deformation.
  - Both the wheel face and hub face had microfinish-sized spiral machining grooves on their mating surfaces. These surfaces nominally are the ones that clamp against each other with the static friction that resists relative radial movement when in use. It is noted that each combination of hub face geometry, wheel face geometry, machined surface finishes, and any coatings (such as paint) will affect the coefficient of friction of the clamped joint.

- Wheel/hub contact surfaces and studs – as painted
  - The cast iron hub castings were, as mentioned, painted by the axle manufacturer. There are compelling functional reasons (and customer demand) for using a durable anti-corrosion coating for a ferrous component that will be stored and transported in the presence of humidity and moisture. In the axle manufacturer's repeated testing, paint application had not proven to cause wheel loosening, provided that lugnut retorques were performed on new or replaced wheels at 10, 25, and 50 miles after installation. Clearly, there was awareness that the paint in these hub/wheel interfaces will get redistributed in or extruded from the interface.
  - The valleys of the spiral machining grooves provide a place for the paint to go after wheel assembly, as it is squeezed out of the interfacing groove peaks. This paint migration will cause stud preload relaxation. Once the grooves are full, paint will extrude out from the joint, primarily in a

tangential direction along the grooves. This can be observed on the subject trailer hubs and wheels – see Figure 14.

Figure 14
view showing paint extrusion from hub face

Between the paint extruding into and along the groove valleys and into the general low spots that occur due to lack of planarity, the paint should eventually be squeezed out of the overall hub/wheel joint (or be compressed) to the point that the stud torque and tension will adequately stabilize – provided that appropriate retorques occur. According to the axle manufacturer, this level of torque stabilization is typically reached through the prescribed torque reapplications. If, however, some factor prevents adequate generation of stud tension, this squeezing out of the paint will not occur without increased operational loading or appropriate stud tensioning procedure. A related issue is that the amount of contact surface area and the initial paint thickness will both likely affect the rapidity with which the joint stabilizes.

- In the application of hub paint, because of the conditions outlined above, thickness control is important. Thicker paint will cause more stud tension relaxation and require more retorquing, and (depending upon the paint properties) may take significantly longer to squeeze out of the joint and/or stabilize. There was a NHTSA recall in 2004 for several trailer brands that had problems with wheel loosening; this was due largely to excessive hub paint[13]. In the subject case, there was no evidence that the average thickness of the subject hub's remaining paint was out of the manufacturer's specified maximum of 0.003 inches.

- The axle manufacturer described the paint on the studs as overspray; it was not an intentional application but was nevertheless part of their normal manufacturing process. On this design, with the studs pre-coated with a phosphate/oil finish, paint was not necessary. Prior to the incident, the axle manufacturer had performed quasi-static testing of the torque/tension effects of various thicknesses of paint on studs. The results reportedly showed that the paint acted as a lubricant, increasing stud tension for a given torque application. This testing, however, did not subject the assemblies to in-service loads and temperatures. It would be reasonable to expect progressive extrusion of the paint from the mating thread surfaces, and a corresponding reduction in stud tension.

Additionally, it would have been a simple manufacturing step to place a masking cap on the stud ends before painting.

- Wheel/hub joint design in different markets
  - The most prolific creators of wheel/hub joints (automobile and light-truck manufacturers) have an extensive history of testing these joints. This testing has typically been conducted on test tracks, and has featured a variety of roadway surfaces, roadway sample defects, loading configurations, and specific driver inputs. With advances in computer technology, a shift to computerized loading simulations has occurred in the past few years, most commonly using what are known as biaxial wheel testing machines. These machines primarily allow severe, transitional, and variable duration simulated dynamic loads to be applied to wheels, and (with additional equipment) hub joint and brake components can be tested as well. These machines allow foreseeable loading scenarios over the lifecycle of the wheel/hub joint to be consistently simulated – and not subject to variations in weather or a test driver's own driving style. Review of the SAE recommended practice for biaxial wheel testing [14], and review of the underlying methodology development [15, 16] shows that this recent technology allows designers of wheel/hub joint components to obtain useful durability information relatively easily.
  - In comparison with automobile and light-truck manufacturers, where over a million individual wheel/hub joints may be sold per year for just one model of vehicle, consumer-grade trailer manufacturers tend to be much smaller operations. The core competency of automobile/light-truck manufacturers is creating complete vehicles "from scratch", and this is reasonable given the economies of scale. Trailer manufacturers, conversely, are dealing in quantities of hundreds or thousands, not millions, and there is much more commonality of product offerings across various manufacturers. These simpler products are sold and serviced by dealers that have more basic technical capabilities. Trailer owners may perform their own maintenance or may be located in rural areas far from manufacturer-approved repair facilities. As such, mechanical simplicity and component commonality are market expectations. Given these market elements, and given the price-competitive nature of these somewhat generic products, it is reasonable for trailer manufacturers to use "purchased complete" running gear components. The running gear components are the ones that 1) need the most engineering and testing, 2) require the most specialized manufacturing processes, and 3) are most commonly maintained and repaired. Facilitation and amortization (of the cost) of those first two elements in particular would be beyond the capabilities (and the interest) of most consumer trailer manufacturers,

if for no other reason than economies of scale. And the ability for a smaller repair facility to stock running gear components usable across a variety of trailer brands is beneficial for the trailer manufacturer, the repair facility, and the consumer.

- The manufacturers of trailer running gear components market their components indeed as pre-engineered complete products. There may be mention (in manuals or application guides) of the need for the trailer manufacturer to ensure the compatibility of components from different suppliers, or to verify the suitability of the components for the intended use. However, these pre-engineered purchased products are not trivial in complexity and may have numerous proprietary elements. As such, the trailer manufacturer is not in a position to fully evaluate all engineering details of these components without devoting significant resources to reverse engineering, benchmarking, and analysis. It is reasonable for trailer manufacturers to assume engineering competence on the part of the running gear component manufacturers, and not be required to duplicate that engineering competence within their own companies. At the same time, nominally the manufacturers of motor vehicles (including trailers) are responsible for any safety-related defect in their vehicles or original equipment, per Federal law[17].
- As an aside, it should be noted that commercial heavy truck and trailer manufacturers do also typically use pre-engineered running gear, despite having significant internal engineering resources. However, this is a market commonly featuring large fleet purchases, continuous heavy use of vehicles, in-house maintenance facilities, and significantly more Federal regulation. As such, a comparable discussion of heavy truck and trailer wheel/hub joints is outside the scope of this paper.

Initial opinions and discussion

This case involved complex pre-manufacture interactions between the trailer manufacturer and the subcomponent suppliers, as well as extensive discovery documentation, evolving research & published knowledge within the trailer industry, and a specific sequence of significant events in the time leading up to the production of the trailer. The opinions selected for inclusion forthwith are limited to those that further the technical discussion. These opinions are described as “initial” since court deadlines (and subsequent extensions) led to multiple stages of analysis – as will become apparent.

- This incident occurred due to loss of clamping force in the hub/wheel interface, leading to stud fractures, lugnut loss, and the wheel separation.
- During manufacture and delivery, the system of wheel, lugnuts and axle hub were assembled by the trailer manufacturer using an assembly process func-

tionally in compliance with the recommendations of the axle manufacturer, the wheel/lugnut supplier, and trailer industry associations.

- The application of torque to the lugnuts did not fully seat the wheels onto the axle hub face in the expected manner. This was most likely due to the presence of excessive friction in the conical wheel/lugnut interface caused by the interference and galling of the non-bulge lugnut hex points in the non-counterbored wheel nutseats. This excessive friction led to insufficient generation of stud tension, which in the first case did not allow the paint to be fully extruded from the hub/wheel joint in a timely manner, and which secondly failed to provide adequate clamp to resist the higher shear loading caused when the trailer was loaded with horses and then driven.
- *Discussion:* The core technical aspects of the author's opinions in this case were based on industry practices and extensive research into existing literature on wheel fastening systems. However, very little such information was directly on point with the circumstances of this case. These opinions were revealed in the author's Federal Rule 26 report. Once the report was submitted, and prior to deposition, testing was conducted (on a limited basis due to time constraints) in hopes of better quantifying and documenting the effect of the non-bulge lugnuts in the countersunk nutseats.

First phase of testing

- The testing deemed most relevant would hopefully determine the relationships between lugnut torque and stud tension during installation and dynamic loading of an exemplar wheel/hub assembly. As these performance relationships are in fact quite complex and instance-specific, it would be desirable to dynamically analyze tension loads in multiple studs of the eight-stud hub design. Ideally, it would also be possible to examine the nutseat condition after testing, for evidence of galling, sticking, and embedment (due to the applied torques and dynamic loading). This would need to be done by removing the lugnut (after load testing) without rotating it against the nutseat.
- The main options for conducting such testing would be to fit an exemplar trailer with custom instrumented stud/hub components and drive it the route traveled by the subject trailer, or alternately to conduct the testing using comparable simulated loads on a biaxial wheel testing machine. For liability reasons, it was decided to conduct the load testing using biaxial test machinery, and a highly qualified biaxial wheel testing firm was located. However, the stud tension sensing & releasing capability would need to be designed & fabricated from scratch. Because of time constraints, a simpler analysis method was chosen.

- The simpler method used a torque-tension tester, which tracks the torque application to a lugnut versus the stud tension. Such testing is done without the application of any radial or lateral loads to the wheel. Because of this, embedment and clamp load relaxation (due to loading) cannot be evaluated. However, in the absence of any other available data on the relative performance of non-bulge lugnuts versus bulge lugnuts in a subject style wheel, testing these types of lugnuts/wheels on a torque-tension machine seemed a reasonable step.
- The test apparatus (see Figure 15) used an electronically-controlled and instrumented wheel gun to tighten the lugnut at a prescribed rate and to a prescribed torque application limit, while a load cell measured the increase in stud tension relative to torque application. Exemplar wheels, studs, and both non-bulge and bulge lugnuts were obtained.


Figure 15
torque/tension test apparatus

The testing apparatus and exemplars were set up and the test cycle initiated – tightening of the lugnut at 20rpm with tension values sampled at 85, 100, 115, and 130 ft*lbf of torque. A squeaking sound was heard as the tightening commenced, and after disassembly, an unexpected yet significant coating of the wheel's clearcoat was found in the lugnut contact area of the wheel's nutseat. The lugnut achieved the desired torque, with a high amount of stud tension, yet the steel lugnut and aluminum wheel nutseat were not actually touching – the clearcoat kept them from making the desired metal-to-metal contact. This was found with both bulge and non-bulge lugnuts. See Figures 16 and 17, and note the powdery clearcoat residue.

The biaxial wheel testing expert, a wheel industry veteran, observed that these surfaces were typically masked to prevent introduction of the soft coating between the wheel and lugnut. This finding highlighted the need to do more research, particularly into establishing whether the subject wheels also had clearcoat in the nutseats. It was also decided that the simple torque-tension tests were not going to provide adequate information, especially if clearcoated nutseats were a factor. The decision was made to construct the custom apparatus previously discussed in order to conduct testing using a biaxial wheel test machine. A draft inspection protocol was created.

Figures 16 & 17
clearcoated nutseats after installation of bulge (left) and non-bulge lugnuts

Discussion of first deposition

The first phase of testing was completed just prior to the author's first deposition in this case. During that deposition, the author revealed that opinions held in this case (and documented in the Rule 26 report) would be better supported by testing, and that testing had begun but had necessarily been found inconclusive and cut short due to the unexpected complications of the clearcoated nutseats. After reviewing the draft protocol for the more extensive second phase of testing, the adverse parties agreed to postpone the completion of the deposition until after that testing.

Second phase of testing – planning

The stated test objectives were to attempt to quantify the effect of lugnut type and nutseat finish on the torque-tension relationship in the presence of lugnut torquing, time and varying cyclic loading.

- The biaxial test machine to be used was of SAE J2562[18] Type "B" configuration, with a rotating metal cylinder that the wheel/tire rides within, driven by the inner surface of the metal cylinder against the outside of the tire. The radial loading is facilitated through the use of hydraulics in pressing the tire against the cylinder interior with more or less force. There is a "curb" or shoulder at each end of the cylinder, which facilitates the lateral loading by (through hydraulics) pressing against the tire sidewall. The "axle" on which the wheel/tire rotates is fitted with data acquisition sensors that track radial and lateral loading. This particular machine also features an inductive slip-ring array that allows sensors rotating with the wheel/tire to pass data out through the fixed portions of the machinery to data recording computers. It was through this slip-ring array that the stud tension data would pass.

- The radial and lateral loads to be applied in this testing required careful consideration. Biaxial wheel test machines are typically used to evaluate the lifetime durability of vehicle wheels, and the typical loading is much more severe than would have been experienced by the subject trailer[19, 20]. The choices regarding loading determination were to instrument and ballast an exemplar trailer and drive it the ~1500 miles, or estimate loading based on vehicle static and dynamic loads. There wasn't sufficient time to instrument, ballast and drive an example trailer, plus it would be impossible to confirm whether the "testing trip" was more or less severe than the actual trip. It was decided that loading estimations would be used. As mentioned, common biaxial loading profiles are meant for lifecycle testing under the harshest foreseeable conditions, not for simulating loads encountered by a low-mileage trailer loaded only during its final 160 miles. Since overly harsh loading would favor the trailer manufacturer's defense, and wishing to avoid this conflict, it was decided to ignore braking and "special event" loading such as driving on washboard roads, striking potholes, or hitting curbs. The biaxial wheel test loading would be conservatively based on no greater than 0.3*g* turns (a level similar to no-braking highway turns), used with basic vehicle dynamics/statics calculations.
- The prescribed retorquings were set to be performed at 10, 25, and 50 miles into the testing cycle. However, due to the accuracy and repeatability limitations of a torque wrench and operator, it was decided that if measured stud tension had decreased less than 5% over these initial testing durations, no retorque would be performed.

- Design of custom stud tension tracking apparatus
  - **Goal: stud tension measurement**: Two common options for stud tension tracking were evaluated – ultrasonic sensors and strain gauges. Ultrasonic sensors are known to have more variability[21], and the biaxial testing expert had more familiarity with strain gauges, so strain gauges were selected as the tension measuring means.
  - **Goal: embedment evaluation**: This goal would require (as mentioned) releasing the tension from the lugnut/stud/nutseat combination without rotating the lugnut within the nutseat. Ordinarily, the act of removing the lugnut for nutseat inspection will further damage the galling & embedment-damaged surfaces. With this design, once the stud tension was released, the nut could be pulled axially away from the nutseat and embedment/galling effects would be largely undisturbed.

- ◦ **Goal: comparable hub and wheel geometry**: Due to the specific "lobed" design of the subject hub casting, it was decided that the hub face of the custom apparatus would have a comparable design. It was decided that the axle manufacturer's hub paint would not be applied to the test hub face, since it would be difficult to duplicate accurately.
- ◦ **Design:** The primary design challenge with the custom apparatus was in determining how to allow embedment evaluation through releasing stud tension independent of lugnut rotation against the nutseat. The author created a design concept using a 3D solid modeling CAD software application. The design focused on applying strain gauges to modified exemplar studs, which in turn were pressed into cylindrical "studholders" that can be rotated to unwind the stud from the lugnut. The apparatus components are shown in Figure 18, a top-level assembly drawing.

- Creation of the loading profile or "load file"
  - ◦ The "load file" is input to the biaxial wheel testing machine, and it comprises the chart of radial loads, axial loads, and their respective load durations, which were meant to conservatively simulate the operational history of the subject trailer.

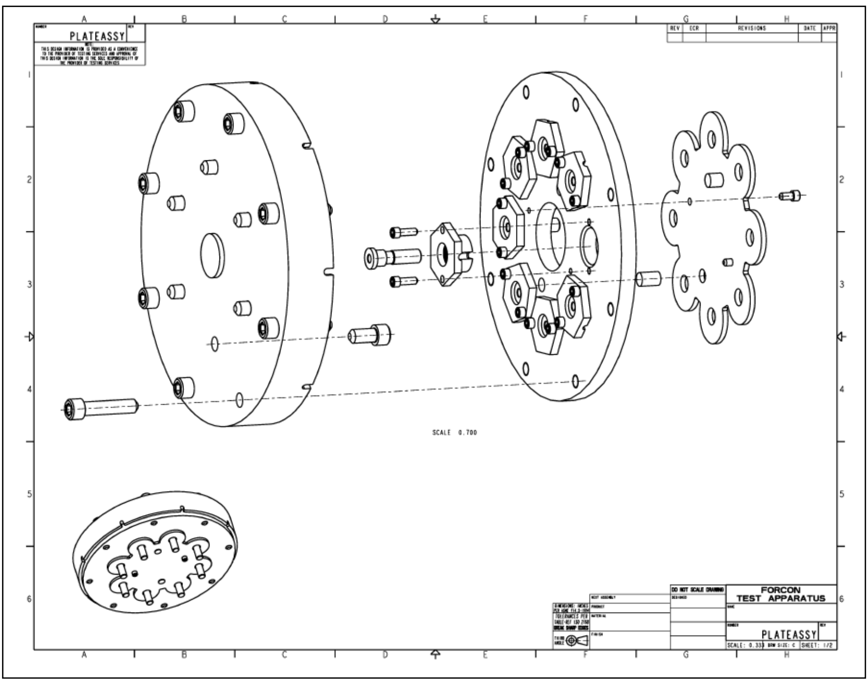


Figure 18
custom apparatus assembly

- The trailer manufacturer provided axle and coupler weights of an empty exemplar trailer, measured at the author's request. As part of determining the load transfer during the 0.3*g* turns to be modeled, the author worked with the trailer manufacturer's existing 3D CAD models (of a largely complete trailer assembly) to determine the center of gravity (CG) position. The loaded CG position (accommodating the 3 horses) would have been difficult to determine without the use of CAD analysis. After some correction of the CAD models, the check of accuracy was to compare the CAD-calculated empty trailer weight with the actual empty weight as previously weighed by the trailer manufacturer – the weights were within about 5 percent. Livestock will exhibit fairly stable mass properties in highway trailer operation[22], so three horses were modeled in CAD as 1000 lb cylinders oriented horizontally, centered 48" above the trailer floor, along with a model simulating 400 pounds of equipment resting in the front floor area of the trailer – see an image of the loaded-trailer CAD model in Figure 19. In the image, the roof skin is hidden for visual clarity.

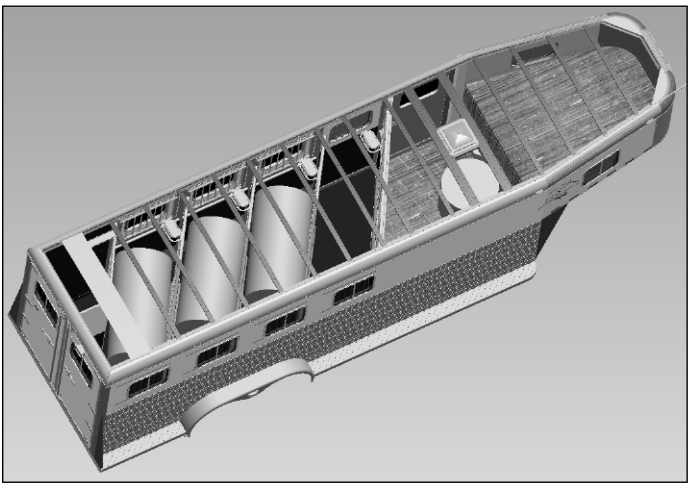


Figure 19
shaded view of 3D CAD model of trailer

The loaded CAD model displayed a weight of 9380 pounds, and the trailer manufacturer's predicted loaded weight was 9560 pounds – within 2 percent of the modified CAD model. The model's coupler weight was similarly comparable to the exemplar's coupler weight. It seemed reasonable, therefore, to use the center of gravity positions obtained from the CAD data.

- Again with the goal of creating a conservative loading scenario that would not unreasonably favor the trailer manufacturer, it was decided that the only lateral loads to be simulated would correlate to the 40 miles of travel on rural highways encountered by the loaded trailer on the trip from the horse breeder's farm to the Interstate. The rest of the mostly-freeway trip from the trailer manufacturer's lot to the incident scene was approximated as radial loading only with no turns. In the interests of time, it was decided to abbreviate the 1370 miles of actual unladen travel (from the manufacturer to the owner's farm) to 300 miles of biaxial testing. Therefore, the three phases of testing were "low load, high speed" unladen with radial loading only for 300 miles, "high load, low speed" laden with lateral & radial loading for 40 miles, and "high load, high speed" laden with radial loading only for 120 miles.

◦ The lateral loads for the 40 mile rural highway portion were approximated with the help of satellite imagery. Turn radiuses and lengths for the 15 significant turns were estimated and evaluated at 55mph (with 5 tighter turns as the exception) and no more than 0.3 lateral *g*. The intermediate straight sections were distributed evenly between the 15 turns. This data is shown in Figure 20.

Route analysis-300 miles unloaded travel: manufacturer-farm					est speed					duration	duration
low load high speed - radial load only					(mph)					(feet)	(wheel revs)
10mi cumulative to first retorque					55					52800	7052
25mi cumulative to second retorque					55					79200	10578
50mi cumulative to third retorque					55					132000	17629
300mi cumulative to biaxial testing					55					1320000	176295
Route analysis - 40 miles loaded travel: farm to interstate											
high load low speed	right turn radius	length	left turn radius	length	est speed (mph)	est speed (feet/sec)	lateral g (v^2/r)	lateral g (group)	duration (sec)	duration (feet)	duration (wheel revs)
Highway A											
turn 1	1000	1200			55	81	0.20	0.20	14.9	1200	160
turn 2	700	1000			55	81	0.29	0.30	12.4	1000	134
turn 3	3000	2000			55	81	0.07	0.10	24.8	2000	267
turn 4			1500	1500	55	81	0.13	-0.10	18.6	1500	200
turn 5			1000	1200	55	81	0.20	-0.20	14.9	1200	160
turn 6	3000	1500			55	81	0.07	0.10	18.6	1500	200
Highway B											
turn 7			50	80	(.3g)	22	0.30	-0.30	3.6	80	11
Highway C											
turn 8	40	60			(.3g)	20	0.30	0.30	3.1	60	8
turn 9			1200	1100	45	66	0.11	-0.10	16.7	1100	147
turn 10	3000	1200			55	81	0.07	0.10	14.9	1200	160
turn 11			2200	1500	55	81	0.09	-0.10	18.6	1500	200
turn 12			1600	1500	55	81	0.13	-0.10	18.6	1500	200
turn 13	2500	2000			55	81	0.08	0.10	24.8	2000	267
Highway D											
turn 14	80	125			(.3g)	28	0.30	0.30	4.5	125	17
Interstate											
turn 15			50	80	(.3g)	22	0.30	-0.30	3.6	80	11
STRAIGHT SECTIONS										195155	26064
Route analysis - 120 miles loaded travel: Interstate to incident					est speed					duration	duration
high load high speed - radial load only					(mph)					(feet)	(wheel revs)
460mi cumulative					55					633600	84621

Figure 20
route analysis worksheet

Calculated lateral g forces were rounded to the nearest $0.1g$ and used with the mass and center of gravity position data to determine the radial and lateral load transfer in turns, using free-body diagrams and force/moment equations. The resulting load file was provided to the biaxial testing expert for programming into the biaxial test machine. The radial loading values ranged between 1175 and 2666 pounds, and the lateral (axial) loading values ranged between 0 and 723 pounds.

  - Following the creation of the test apparatus, protocol, and load file, these elements (and their supporting documentation) were presented to an SAE-published wheel testing expert for his review. After review of the test methodology, he did not find any elements he thought should be changed in order to meet the test objectives.

Biaxial wheel testing final setup

Following the fabrication of the custom stud-tension measurement apparatus, the testing was commenced as follows with a selection of exemplar wheels and lugnuts to be tested. Counsel for the wheel manufacturer would not allow testing of the subject wheels to see if the observable discoloration and contamination in the subject wheel nutseats was indeed clearcoat – unfortunately, this had not previously been tested for by any of the various parties. Fortunately, exemplar wheels were discovered both with and without clearcoat in the nutseats.

- The exemplar wheels were unused old stock found in the warehouse of the trailer manufacturer. Wheels B, D, and E had date-stamps of 11/2004, and wheels A, G, and H had date-stamps of 4/2006. A coating thickness tester brought for testing of nutseat clearcoat thickness provided inconsistent readings, due to the conical surface of the nutseats. The continuity testing feature of an electronic multimeter was used instead simply to determine whether clearcoat (an insulator) was present in the nutseats at all. Wheels B, D, and E had no clearcoat in the nutseats, and wheels A, G, and H had clearcoat in the nutseats.
- For each test, the wheels were mounted as follows: A vibrating/beeping digital torque wrench (with stated accuracy of +/- 1% and peak torque application recording capability) was set first to 85 ft*lbf and the lugnuts were torqued in a pattern of 1-5-7-3-8-4-6-2. Following this, the torque wrench was set to 120 ft*lbf and the lugnuts were torqued again using the same pattern. The recorded peak torques were noted for each lugnut, as were the resulting stud tensions – statistical results will be shown below. See Figures 21 – 23 for images of the test apparatus.

Biaxial wheel testing

It should be noted that the strain gauges on studs 4 and 5 (of the 8 total) did not function properly and were ignored for the following testing. Cyclical loading data was sampled for stud tension at 150Hz.

- **Test 1**: Wheel G (clearcoated nutseats), bulge lugnuts. The wheel was mounted and the 460-mile dynamic testing was begun. Early in the dynamic testing it was determined that the range setting (-10K lbf to +10K lbf) for the cyclical tracking of stud tension needed to be increased, and the testing of Wheel G was cancelled. The initial stud torque/tension data was valid for Wheel G, however, and it is included in the results below. See Figures 24 and 25 showing the condition of the Wheel G nutseats after test cancellation – the lugnuts damaged the clearcoat at the periphery of the contact area.

- **Test 2**: Wheel D (bare nutseats), non-bulge lugnuts. No retorquings were necessary, as the stud tension didn't drop over 5% in the first 50 miles. After testing, the apparatus functionality successfully allowed stud tension to be released from each lugnut without rotating the lugnut in the nutseat. See Figure 26.

  There was significant embedment and mechanical bonding of the lugnuts to the nutseat. After unwinding the stud/studholder assemblies 3 turns from the lugnuts, it was necessary to strike the stud/studholder assem-

Figure 21
biaxial wheel test machine and controls

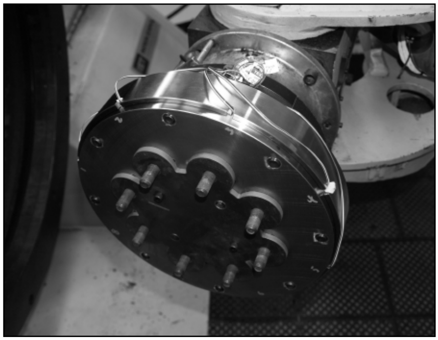


Figure 22
custom test apparatus mounted to biaxial wheel test machine

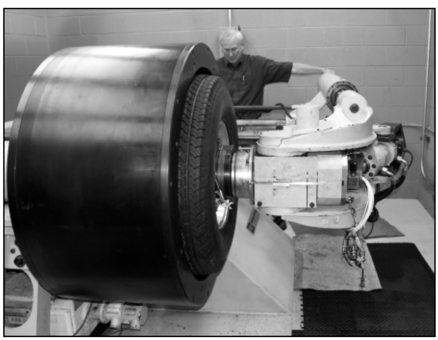
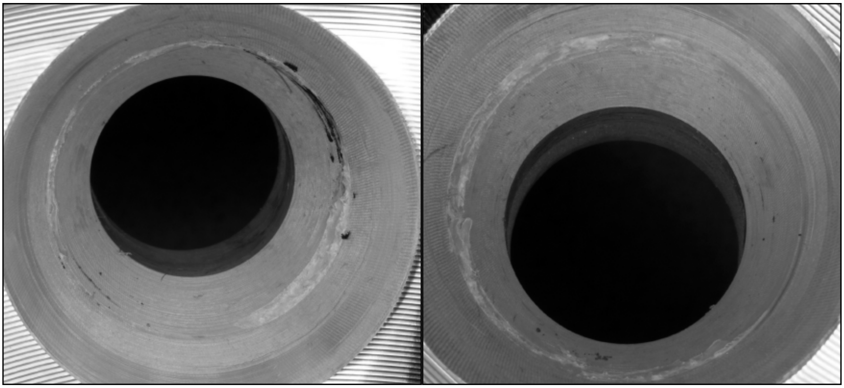


Figure 23
biaxial wheel test machine during insertion of tire into drum



Figures 24 & 25
Wheel G holes 1 and 4 after partial Test 1, showing clearcoat damage

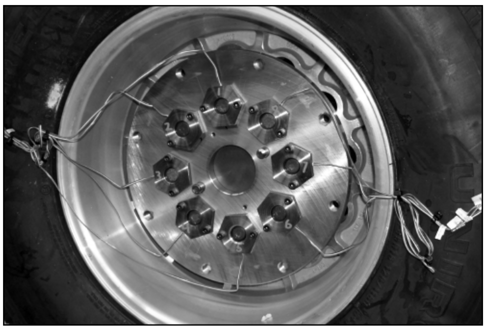
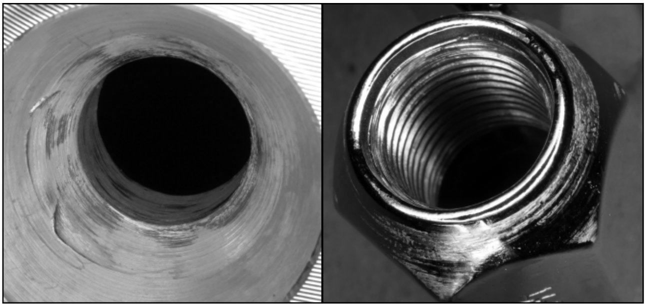


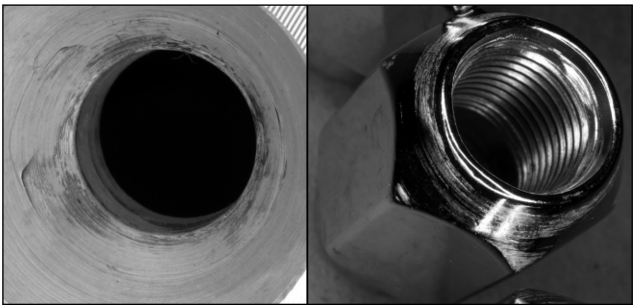
Figure 26
stud/studholder assemblies prior to removal from lugnuts

blies sharply with a mallet, to axially dislodge the lugnuts from the nutseats. Metal transfer was evident on the hex points and narrow end of the conical lugnut surfaces, and embedment was visible in the nutseats – visually similar to the embedment observed on the subject separated wheel. Selected images of Wheel D nutseats and lugnuts are provided as Figures 27 – 30. Test results data is shown in Figure 31. The apparatus was checked and reassembled for the next test. See Figure 32.

- **Test 3:** Wheel H (clearcoated), non-bulge lugnuts. No retorquings were necessary. At the completion of the 340 mile high-load low speed portion, with the wheel temperature measured as 85ºF, it was decided to include hub/wheel heating in the testing. This was because the stud tension hadn't dropped significantly (at 340 miles) under the conservative loading, and the cancelled Wheel G testing had shown minimal flow/extrusion of the clearcoat. It seemed likely that normal braking heat transfer into the hub would affect the hardness of the clearcoat. Heat was applied over about 70 minutes, using an electric heat gun pointed at the wheel spoke area while slowly rotating the wheel; a temperature of ~160ºF was eventually reached at the nutseat area. Clamp load was observed to increase for about 10 minutes during heating, likely due to the higher coefficient of linear thermal



Figures 27 & 28
Wheel D hole 5 galling/embedment damage and metal transfer



Figures 29 & 30
Wheel D hole 7 galling/embedment damage and metal transfer

				STUD TENSION (LBF)					
test sequence	exemplar wheel	clearcoat in nutseats	lugnut style	at 85 ft*lbf applied torque	at 120 ft*lbf applied torque	post low-load, high speed (300mi)	post high-load, low speed (340mi)	post high-load, high speed (460mi)	average tension drop
1	G	yes	bulge	not recorded	7491-10958	cancelled	-	-	-
recorded torque standard deviation				-	0.6				
average resulting stud tension				-	9621				
resulting tension standard deviation				-	1501				
2	D	no	non-bulge	2992-4308	4170-5467	4174-5419	3986-5251	4003-5266	2.9%
recorded torque standard deviation				2.2	0.6				
average resulting stud tension				3700	4856				
resulting tension standard deviation				474	502				
3	H	yes	non-bulge	7309-9684	9854-11847	9745-11717	9600-11625	8698-10702	11.5%
recorded torque standard deviation				0.6	0.8				
average resulting stud tension				8488	10981				
resulting tension standard deviation				923	822				
4	B	no	bulge	2947-4262	3958-5670	3943-5624	3835-5467	3838-5473	3.8%
recorded torque standard deviation				4.6	0.5				
average resulting stud tension				3652	4867				
resulting tension standard deviation				425	565				

Figure 31
test results summary chart

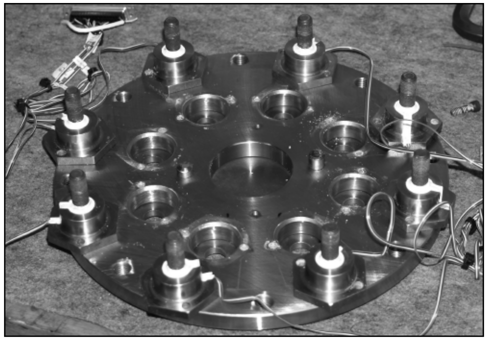
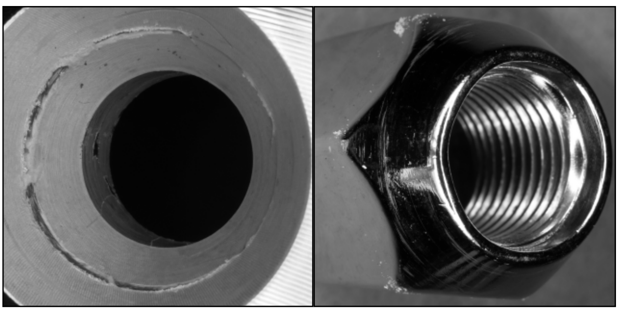
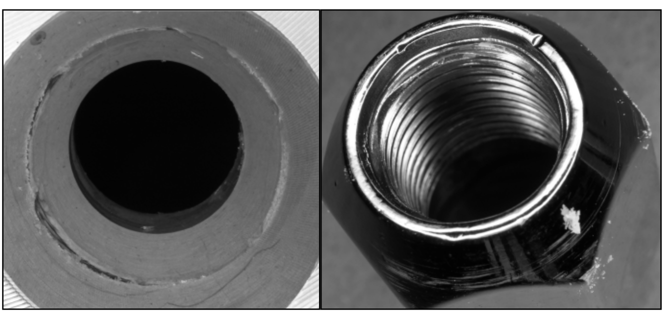


Figure 32
inspection and reassembly of custom apparatus

expansion for the aluminum wheel versus the steel fasteners. After 10 minutes, however, the clamp load started to decrease even though the temperature of the wheel was continuing to increase. It was discussed that this was likely due to the onset of clearcoat relaxation and flow. Following the heating to 160º, which seemed to be the limit of the heat gun, the balance of the testing was commenced with no further heating of the wheel. The non-bulge lugnut hex points cut into the clearcoat surface, and metal-to-metal contact was minimal. Overall test results are shown in Figure 31. It is noted that this Test 3 with clearcoated nutseats shows roughly twice the stud tension of Test 2 with bare nutseats, using identical lugnuts, and independent of the heating. The clearcoat was acting as a lubri-



Figures 33 & 34
Wheel H hole 1 showing hex point contact and clearcoat damage



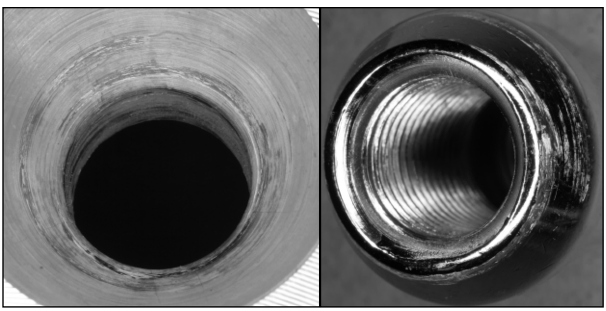
Figures 35 & 36
Wheel H hole 4 showing hex point contact and clearcoat damage

cant between the lugnut and the nutseat. Selected images of Wheel H nutseats and lugnuts are provided as Figures 33 – 36.

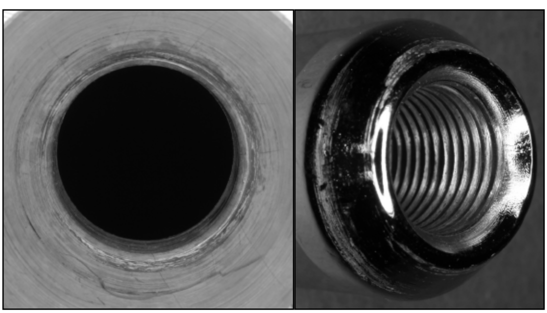
- **Test 4:** Wheel B (bare nutseats), bulge lugnuts. No retorquings were necessary. The nutseats showed fairly uniform metal-to-metal contact with the bulge lugnut contact surfaces. Overall test results are shown in Figure 31. Selected images of Wheel B nutseats and lugnuts are provided as Figures 37 – 40.

Discussion of biaxial testing results

- Though the above testing provided some useful information, the magnitudes of radial and lateral loading applied were insufficiently severe to cause damage at the level observed on the subject components. If an instrumented and ballasted exemplar trailer were driven the described route of travel experienced by the subject trailer, it is possible that the actual load magnitudes could have been better approximated, though the actual subject trailer loading simply cannot be duplicated. A different trailer with a different usage history may have been better approximated with the use of established SAE or trailer industry test load profiles.



Figures 37 & 38
Wheel B hole 1 showing uniform metal-to-metal contact



Figures 39 & 40
Wheel B hole 6 showing uniform metal-to-metal contact

- Consistent with the earlier discussion of applied lugnut torque versus resulting stud tension, the test results show that even with indicated torque standard deviations of less than 1 ft*lbf, there were still stud tension standard deviations of 7% – 15% for each of the four wheel/lugnut combinations tested. The variability appeared to be largely independent of the sequence used in lugnut torquing – see Figure 41.
- It was decided to change the focus of testing to try testing various other elements. Suggestions by opposing parties included that the trailer manufacturer had not ever torqued the wheels, or that retorques were not performed, and the following supplemental tests were intended to explore those scenarios.

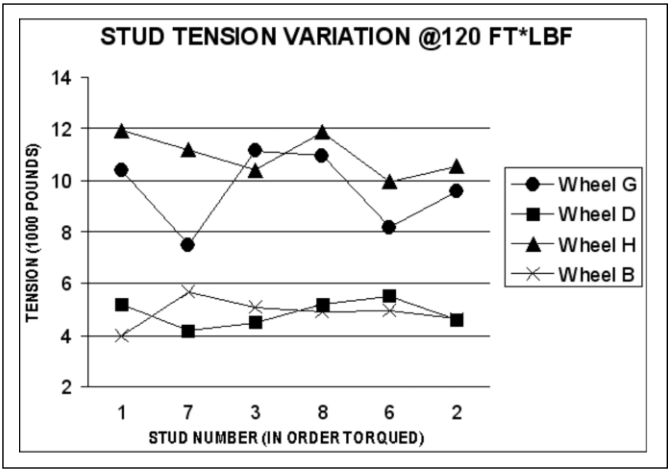


Figure 41
stud tension variation versus tightening order

Supplemental testing – clamp threshold

This test was devised to see what amount of radial/lateral loading could be accommodated by the minimum clamp expected in this joint. For this test, the studs were tightened to 1000 pounds of tension each, corresponding to about 20-25 ft*lbf of torque from the torque-tension relationships previously seen. This represents the minimum setting on common ½ inch drive torque wrenches, and is also comparable to the torque imparted by "spinning on" lugnuts using an impact gun. It was expected that the onset of relative motion (radial slipping) between the hubface plate and wheel would show as a noticeable transition in the stud loading data.

- **Test 5:** Wheel E (bare nutseats), non-bulge lugnuts. Load application profile was a continuous linear increase over 20 minutes from 1000 pounds radial + 40% of radial load for lateral load, to an ending value of 3500 pounds radial + 1400 pounds lateral. Graphical results of this testing (see Figure 42)

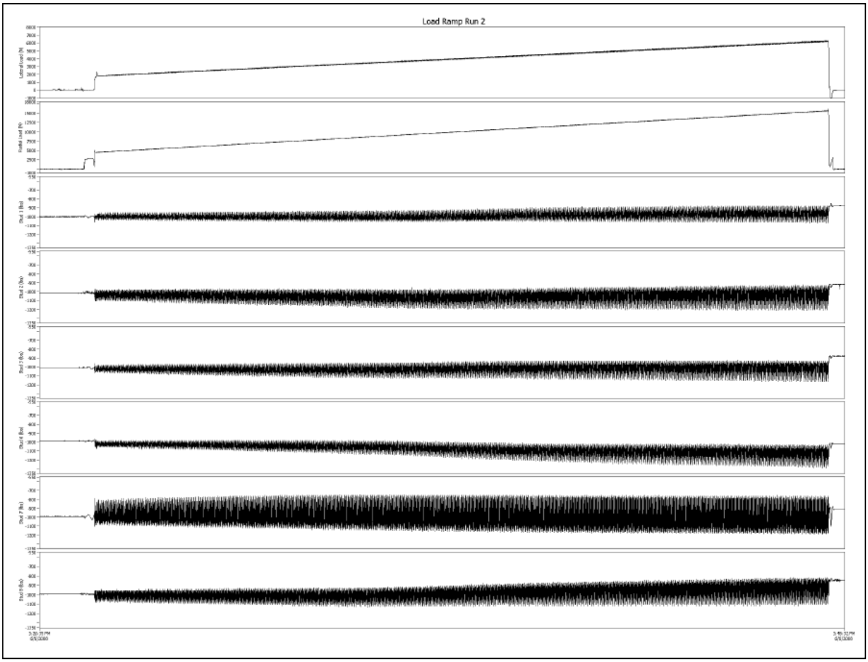


Figure 42
clamp threshold test results for 20-minute test duration

showed no apparent signs of a transition to radial relative motion between the hubface plate and wheel. This is likely due to the number of fasteners (8) and the resulting net clamp force which, through the coefficient of friction, was adequate to resist the combined radial + lateral loading.

Supplemental testing – long term low torque

This test was to see the effects of long term cyclical loading of the wheel/hub at the minimum 1000 pound stud tension described above.

- **Test 6:** Wheel E (bare nutseats), non-bulge lugnuts: This test was to see the effects of long term cyclical loading of the wheel/hub at the minimum 1000 pound stud tension described above. The loading profile applied a continuous 2400 pounds radial load and 960 pounds lateral load, corresponding to about twice the static radial load of the unladen trailer with a 40% added lateral load component. Testing was run at about 15mph, and tension limits were set to prevent damage to the biaxial machine if studs started to break. The test completed 662 miles before the biaxial testing provider experienced a lengthy power failure. There were no component failures.

Supplemental testing – long-term heating

This test was to see the effects of long-term heating (due to operational bearing friction and braking) on the clearcoat and wheel/hub assembly. It was decided that the electric heat gun used previously was not getting the wheels up to a temperature as high as they would see in foreseeable use[23]. Though the referenced study reported brake drum temperature, not aluminum wheel temperature, it was decided that this was reasonable since the wheel nutseats are directly adjacent to and clamped against the brake drum's hub face.

- **Test 7:** Wheel G (clearcoated), non-bulge lugnuts. The initial 460 mile loading profile was applied in conjunction with a propane burner. Wheel temperature tracking was connected in place of sensing stud 8 tension, so tension data was now sampled in five of the eight studs versus the previous six of eight. The wheel temperature profile was as follows:
  - First 300 miles of protocol (low load, high speed): 115-120ºF.
  - Biaxial phase (high load, low speed): 190-200ºF.
  - Post-biaxial phase (high load, high speed): 160-170ºF.
- Results of this testing showed clamp stress relaxation upon the application of heat. One stud (stud 3) showed a significant tension drop at 25 miles, so a retorque was performed. As testing progressed the data provided by studs 2 and 3 began to look suspect, with intermittent wide variations in their values. A retorque of the lugnuts was performed at 300 miles to see if that would stabilize the tension values – it did not. In retrospect, it appears that both retorques were likely stimulated by erroneous tension readings in studs 2 and 3. These retorques made the final tension readings non-comparable to those of the other tests, which did not have 25 mile and 300 mile (or any

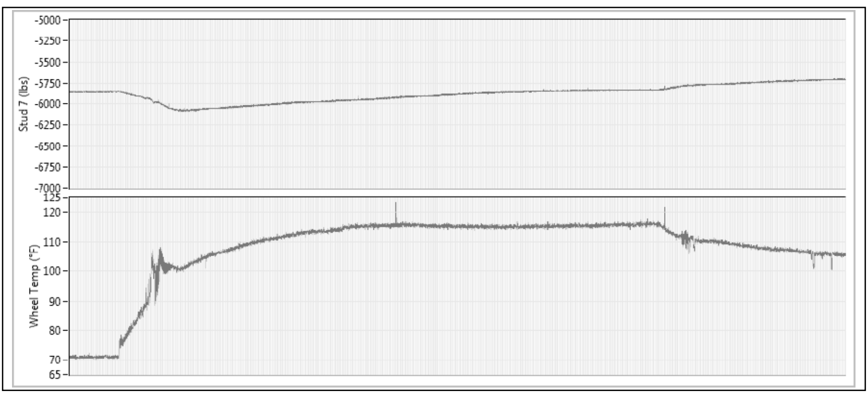


Figure 43

Test 7 stud 7 tension variation with heat application during initial 10 miles of cycle. Note: upper plot tension values increase top-to-bottom, and lower plot temperature values increase bottom-to-top.

other) retorquing applied. Regardless, there were significant and continuous decreases in stud tension with heating (sample data shown in Figure 43), likely due to the softening of the clearcoat. Discounting studs 2 and 3, the data from the remaining three sensored studs showed a an average 16.2% tension drop following the final (300 mile) retorque through the completion of the test. See Figure 44 for the Test 7 data. It should be noted that the above-mentioned power outage at the biaxial test facility ended with just enough time to finish this long-term heating test before a court-specified testing deadline. Ideally, control tests would have also been performed.

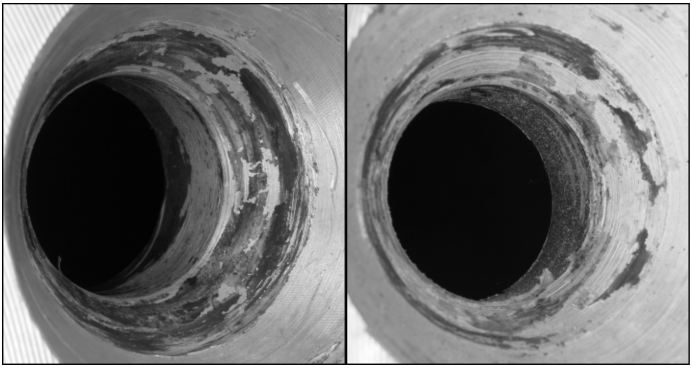
				STUD TENSION (LBF)					
test sequence	exemplar wheel	clearcoat in nutseats	lugnut style	at 85 ft*lbf applied torque	at 120 ft*lbf applied torque	post low-load, high speed (300mi)	post high-load, low speed (340mi)	post high-load, high speed (460mi)	average tension drop
7	G	yes	non-bulge	3808-4720	5655-6206	6608-7216	5867-6732	5492-6054	16.2%
recorded torque standard deviation (n=5)				1.3	0.9	(retorque @ 25mi)	(retorque @ 300mi)		(n=3)
average resulting stud tension				4333	5976				(see notes)
resulting tension standard deviation (n=5)				332	215				

Figure 44
Test 7 results summary

Post-testing inspection of components

As previously mentioned, the subject wheels had not been evaluated for the presence of clearcoat during several joint inspections; the primary focus had been on hub paint and metallurgy. Following the exemplar testing above, permission was received for the author to visually re-inspect the subject wheels, hubs, and fasteners – but no material characterization or other destructive testing was allowed, per the wheel manufacturer.

- Closer study showed what visually appeared to be clearcoat in the majority of the nutseats (see Figures 45 & 46); one wheel had an area where apparent clearcoat was peeling away from the nutseat – see Figure 47. Also in this inspection, based on observations in the exemplar testing, more attention was paid to the significant galling of the nutseats and material transfer to the lugnut hex points; many wheels showed evidence of galling and apparent clearcoat – see Figure 48. The apparent clearcoat appeared to be broken up into "islands" of material, which was similar to the appearance of the nutseat clearcoat in Wheel G following the long-term heating Test 7. The nutseats on one subject wheel showed evidence of a poor machining surface finish in the nutseats, which would also increase the lugnut/nutseat friction.
- Based on the significant effect of clearcoated nutseats on stud clamp retention, it was necessary to quantify the thickness of this clearcoat. As such, exemplar Wheel A was sectioned to allow perpendicular microscopic measurement of the clearcoat. See Figure 49.



Figures 45 & 46
apparent clearcoat in nutseats of subject wheels

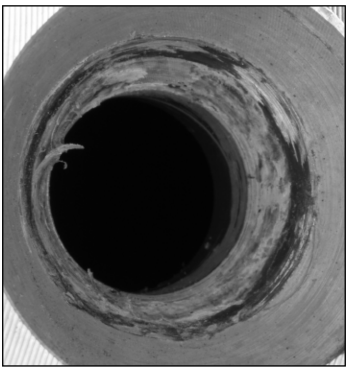


Figure 47
curled portion of apparent clearcoat in separated wheel

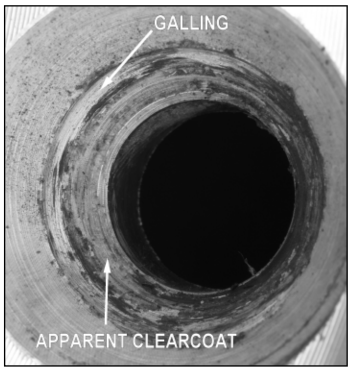


Figure 48
subject wheel nutseat with galling and apparent clearcoat

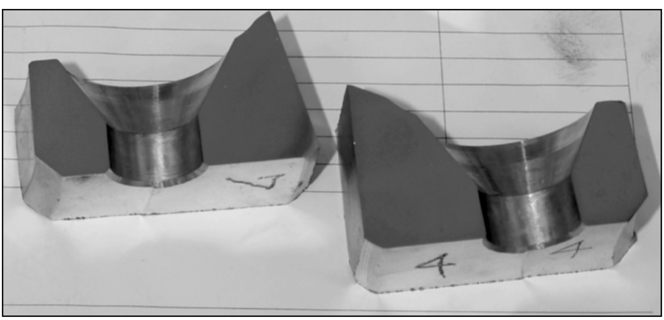


Figure 49
nutseat cross-sections for clearcoat thickness measurement

Of the sections tested, the average thickness was 0.0012 inches. Though this may seem minimal, this coating on the conical nutseat geometry affords the opportunity for 0.002 inches of axial lugnut travel [24], which (when combined with 0.003 inches of hub paint) could lead to 0.005 inches of reduction in stud stretch. Assuming full extrusion of these soft materials and excluding the deformation contributions of the other components, this correlates to over 17,000 pounds of possible reduction in stud tension [25].

Final opinions from the author's Supplemental Report

- The properties of the wheel/hub joint components, such as material strengths, stiffnesses, thermal expansion rates, finishes, and coatings (among other things) all affect the frictional interactions between the components. In turn, the varying effects on the joint of temperature, static and dynamic forces, cargo load and distribution, braking, driver skill, and roadway condition will make each trip unique, in terms of demands on the clamp joint integrity. Ultimately, in efforts to determine the cause of this accident, it would not be possible to accurately simulate all that the subject trailer's wheel joints experienced in the trip from the manufacturer to the incident site.
- If the lugnuts had only been installed "finger tight" by the trailer manufacturer, one or more wheels would have noticeably loosened or fallen off before reaching the reported 10 mile retorquing stop.
- Production personnel would be unlikely to apply less than 25 ft*lbf of torque to the lugnuts using the available impact and torque wrenches. Testing showed that uniform torquing to this level may resist radial hub/wheel joint movement (on an unloaded trailer) for many miles. However, it is reasonable to assume that based on the available tools, expectable skills, established assembly processes, and the eventual extrusion of the subject wheel's (apparent) nutseat clearcoat and the subject hub's paint, an appropriate amount of torque was applied by the trailer manufacturer.
- It would be difficult to determine the hub/wheel temperatures that existed on the trip from the manufacturer to the selling dealer, but it is likely that once the trailer was loaded with horses and driven (with braking) on twisty roads and through towns to the Interstate, the wheel temperatures would be significantly higher than on the initial (unloaded) trip to the dealer along flat Interstate highways. With higher loads & stud tension demands, combined with elevated temperatures, it is expectable (and testing shows) that the wheel's clearcoat (if present) would soften and deform, as would the hub and stud paint. It is known also that drum brakes, once hot, do not cool quickly. This wheel clearcoat / hub & stud paint deformation would cause

progressive stud clamp relaxation and the potential for radial sliding motion between the wheel and hub. With this motion comes (for right-hand threads on left-side wheels) self-loosening[26] of lugnuts, cyclical bending stresses, and eventual stud fracture. The evidence in the subject incident is consistent with such a failure.

- The apparent presence of clearcoat in the nutseats of the subject wheels compromised the wheel/hub joint integrity. The heightened lubricity and cold stiffness of clearcoat, combined with its flow and deformation at foreseeable hub temperatures, would provide a non-durable and inconsistent surface for the lugnut to seat against.
- The wheel/lugnut supplier providing non-bulge lugnuts for the fully-countersunk aluminum wheels compromised the wheel/hub joint integrity. As mentioned, it is true that the conservative loading in the exemplar testing was not severe enough to result in significant lugnut embedment clamp loss or need for retorquing of the non-bulge lugnuts in exemplar Wheel D, as the initial analysis indicated. Retorquing of this wheel may have provided more quantifiable documentation of the observed sticking and embedment of the lugnuts to the nutseats. However, the balance of findings that can be made about the various subject and exemplar wheel/lugnut interactions supports that the non-bulge lugnuts were likely a contributing factor to this incident. The hex points of the non-bulge lugnuts consistently provided sharper points of higher pressure (relative to the lugnut's uniform conical surface) against the nutseats, clearcoated or not, and this served to cause galling and degradation of the surface they were against. Galling and degradation of the nutseats can be expected to cause increased friction between the lugnut and nutseat, leading to reduced and inconsistent development and maintenance of stud tension. The bulge lugnuts tested, in comparison, showed more uniform load distribution and there were no hex points to dig into and gall against the nutseat.
- We can consider the test results in analysis of this incident, with the subject combination of 1370 miles of unloaded use of the trailer, apparently clearcoated nutseats and hub/stud paint, and the loaded travel on twisty roads. With these elements, a reasonable scenario (consistent with the evidence) is that the wheel installation and retorques were done adequately but the apparent clearcoat, hub paint and stud paint did not significantly deform until the trailer was loaded and operated (with braking) on twisty roads.

Third party defendant *Motion To Exclude*

Following the submittal to the court of the author's supplemental report, the wheel manufacturer submitted a *Motion To Exclude* the report and opinions derived from the second phase of testing. The substance of the motion was that the trailer manufacturer didn't notify the wheel manufacturer that the author's supplemental testing and report were going to develop new opinions or change prior testimony. The court denied the motion, with findings summarized as follows:

- When an expert learns of new information after their report has been submitted, the court may exclude new opinions if a supplemental report (that analyzes the new information) is not provided.
- The opinions in the supplemental report appeared to support and refine the opinions in the initial report.
- The inconclusive first phase of exemplar testing highlighted the need for additional testing of, and opinions on, the effects of the clearcoat.
- The supplemental report contained many other opinions that did not pertain to the wheel manufacturer.
- The supplemental report did not cause unfair prejudice to the wheel manufacturer, and they had the opportunity to rebut it.

Conclusion

This case settled before trial, following the author's supplemental deposition.

Many forensic engineering cases rely on well-established and widely known facts and procedures. In this case, there was a notable lack of relevant published information. Exhaustive research, extensive testing, and significant financial expenditure went into the analysis of this incident, yet the broad variety of variables (primarily related to friction) ultimately led to a necessary reliance on engineering opinion, which was supported by the research and testing.

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