POWERS and SONS LLC

Se Habla Steering?

Many of us who have been in the business for a number of years take too much for granted in our field of expertise. I walked into the wrong meeting once. (I'm sure I was in the right place, the meeting was wrong.) I sat through a design review of seat mechanisms. I felt like a fish on a bicycle. I didn't know the terms. I couldn't follow the presentation. I didn't have a basic understanding of the system. If I ever went to work for Lear, I'd be lost.

With this in mind we thought about the newcomer to steering linkage systems. It is in our best interest to get this person conversant in steering linkage as quickly as possible. We have prepared the following information to help someone unfamiliar with steering systems get a quick, broad overview of how steering linkage works, some terminology and some rough idea of the design guidelines. We gathered this information from our internal design guide, industry papers and interviews with new employees to see what would be relevant.

This information was compiled, to a large degree, by employees who were newcomers to Powers and Sons LLC. Who better to determine what fresh eyes needed to see than someone with fresh eyes? Much of the information contained here may insult the intelligence of the old salt. This isn't written for you. Pass it along to someone who might benefit from it.

To anyone with comments on how to make this a more effective tool, please call me. To anyone who wants to pick nits over details or gripe about anything here, write your own damn book.

Gene Messenger

Customer Liaison Manager
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-Gene

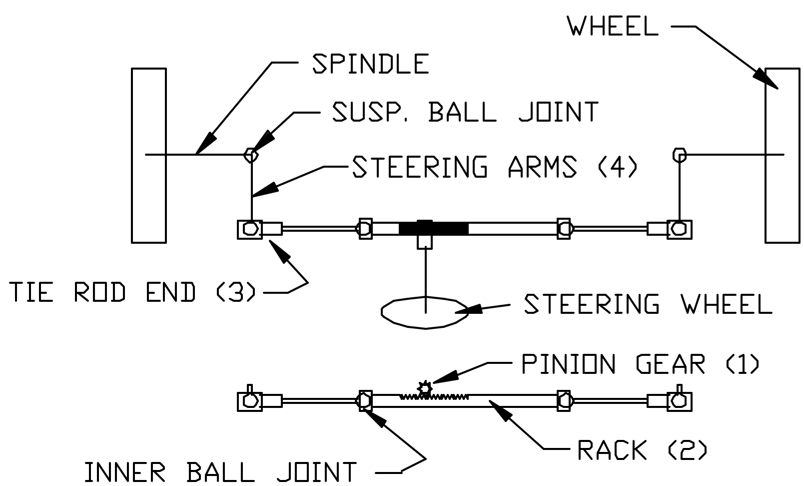
STEERING LINKAGE TYPES

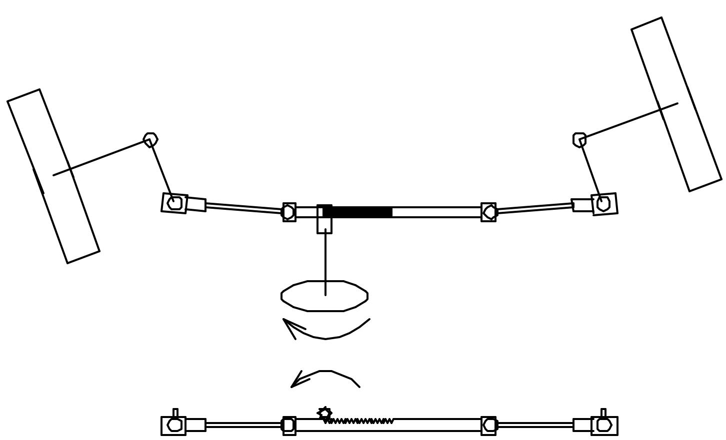
There are three basic types of steering linkage used in passenger cars and light trucks. These are Rack and Pinion, Parallel Linkages and Haltenberger Linkages. There are other variations on those designs described in this section, but these are the three fundamental types. Heavy truck and certain non-highway vehicles utilize significantly different types of steering linkage.

Rack and Pinion Linkage

As the driver turns the steering wheel, the shaft that runs from the steering wheel to a pinion gear (1) causes the pinion gear to turn. As the pinion gear turns it causes the rack (2) to move side to side. As the rack moves side to side it will push the tie rod ends (3) and the steering arms (4) in the desired direction thus turning the wheels.

Advantages of this system include precise geometry, a relatively low weight, and a relatively low cost. The main disadvantages or rack and pinion linkage are a relatively low carrying capacity and inflexibility when it comes to packaging and space within the car.



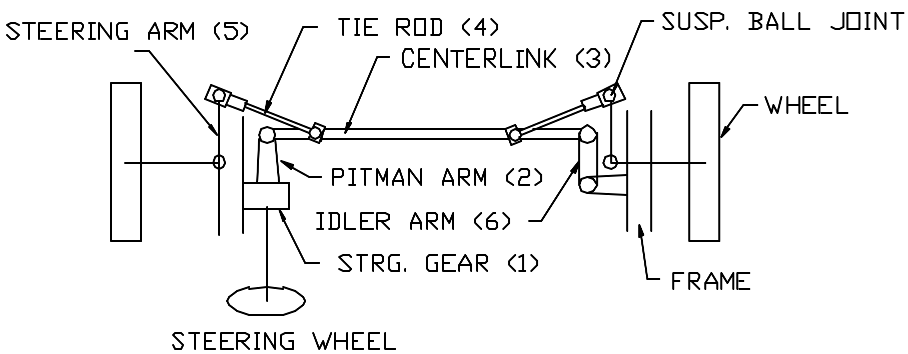


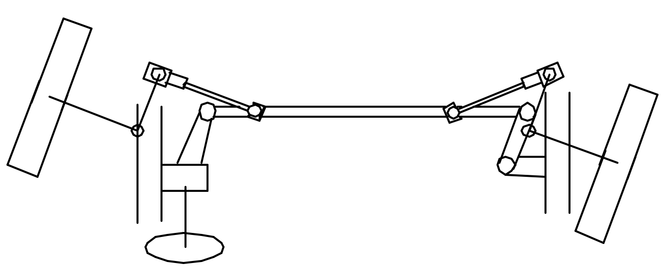
Parallel Linkage

As the driver turns the steering wheel the shaft which runs from the steering wheel to a steering gear (1) rotates causing the pitman arm (2) to rotate. The pitman arm is connected to a center link (3), which will move side to side. This side to side motion will cause the tie rod end/sleeve assembly (4) and steering arms (5) to move thus turning the wheels in appropriate direction. The idler arm (6) connects the steering linkage to the frame and keeps everything parallel and aligned properly.

Advantages of this system include precise geometry, high load carry capability, and package friendly; meaning it can be designed easily to work around obstacles such as an oil pan.

The main disadvantage of parallel linkage is a relatively higher weight and cost.





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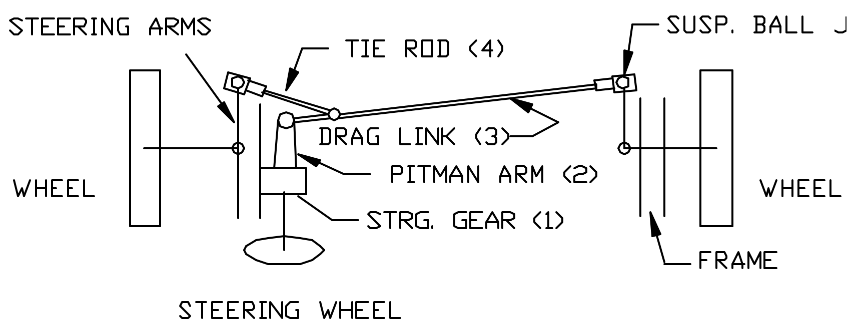
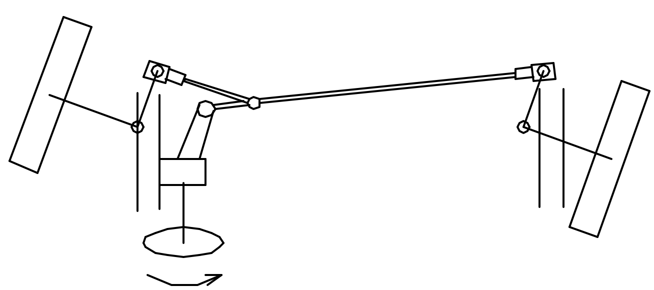
Haltenberger Linkage

As the driver turns the steering wheel the shaft which runs from the steering wheel to a steering gear (1) rotates causing the pitman arm (2) to rotate. The pitman arm is connected to a drag link (3), which will move side to side. This side to side motion (or dragging) will cause the tie rod end (4) and steering arms (5) to move thus turning the wheels in the appropriate direction.

Advantages of this system include a high strength capability; it is less expensive than parallel linkage; and it is package friendly meaning it can be designed to fit around obstacles such as an oil pan.

The main disadvantage of Haltenberger linkage is that it does not provide for precise geometry.





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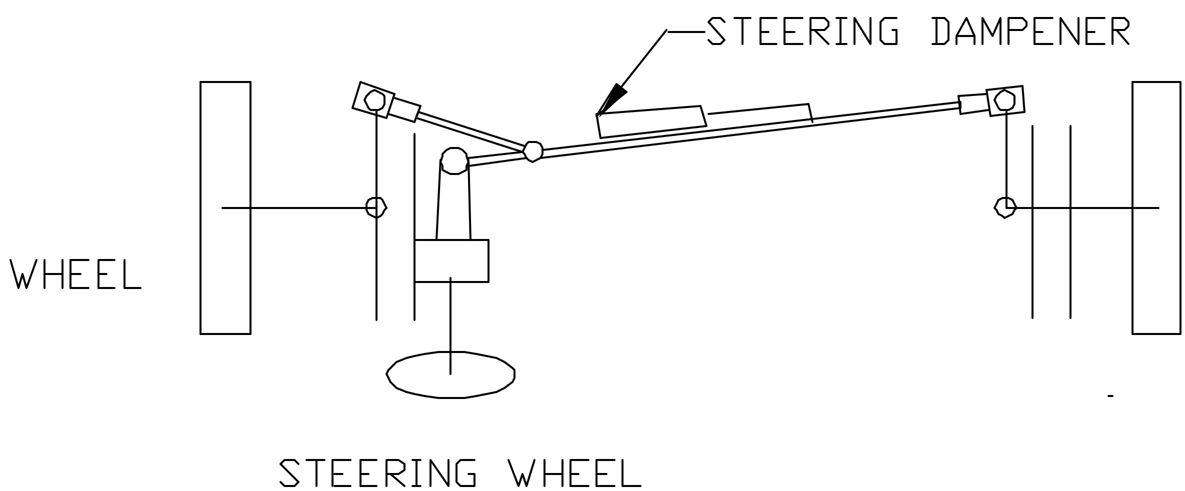
STEERING DAMPER

Some vehicles are prone to shimmy and/or wheel fight. Shimmy is a condition similar to resonance in the suspension and steering system. With minor road inputs, the road wheel can begin to shake at the wheel rotation frequency. It can continue to amplify in intensity without any additional road inputs. Speeding up or slowing down can improve or eliminate the condition. The causes of shimmy are many and include tire construction, suspension geometry, frame flex, excess compliance and system resonant frequency. Wheel fight is the condition where a road irregularity tries to twist the steering wheel out of the driver's hands. Both of these conditions are more prevalent in truck and off road vehicles.

In some vehicles it is necessary to include a damper on the steering linkage to combat shimmy and wheel fight. This damper is a shock absorber mounted between the steering linkage and a fixed member, typically the frame. This shock absorber is designed to have very little resistance during relatively slow, high travel conditions (such as steering cycles). It has maximum dampening resistance during high frequency inputs. It can combat shimmy (repetitive high frequency inputs) and wheel fight (single occurrence high frequency input). The schematic below shows a typical steering damper installation on a haltenberger linkage. The damper must be packaged to insure no interference conditions exist throughout the range of steering travel, jounce or rebound travel. The system must also be designed to insure there is no binding of the damper during steering travel.



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Tie Rod End Ball Sizing

Tie rod end ball sizing is a function of numerous factors. All must be considered.

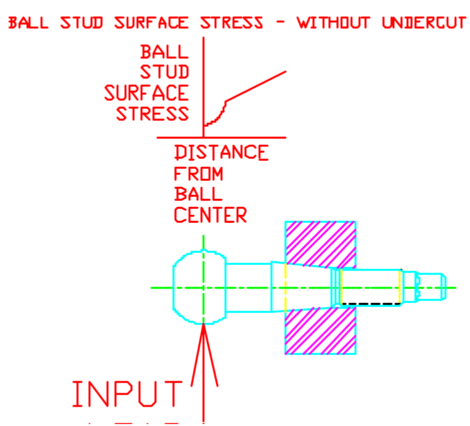
- Pull out strength requirements
- Bearing Material requirements
- Articulation requirements
- Lubrication requirements
- Benchmark Vehicles

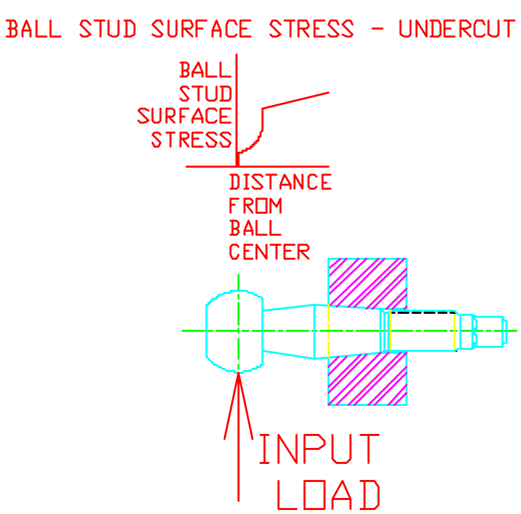
The minimum tie rod end ball diameter must meet all of the above criteria.

Tie Rod End Ball Sizing – Pull Out Strength Requirement

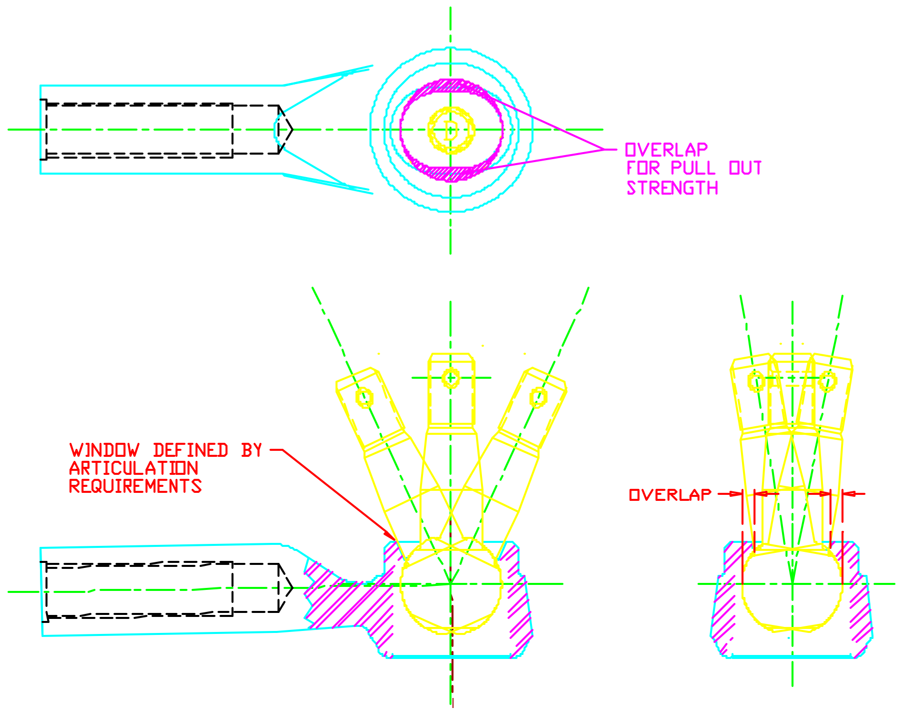
In any ball joint, the shank of the ball stud passes through one side of the housing. The area of the housing where the stud extends through is referred to as the window. The path established by the ball stud during articulation defines this window. Full metal-to-metal jounce and rebound travel + 2 degrees (factor of safety) is used for articulation travel.

Larger ball stud shank diameters (required for impact strength) generally require larger windows. The ball stud shank can be undercut near the ball to minimize the window requirement. Undercutting the ball stud is an added operation. Undercutting the stud in a taper from the ball to the gage line does not reduce the strength of the stud as bending stresses decrease linearly as you approach the ball center.



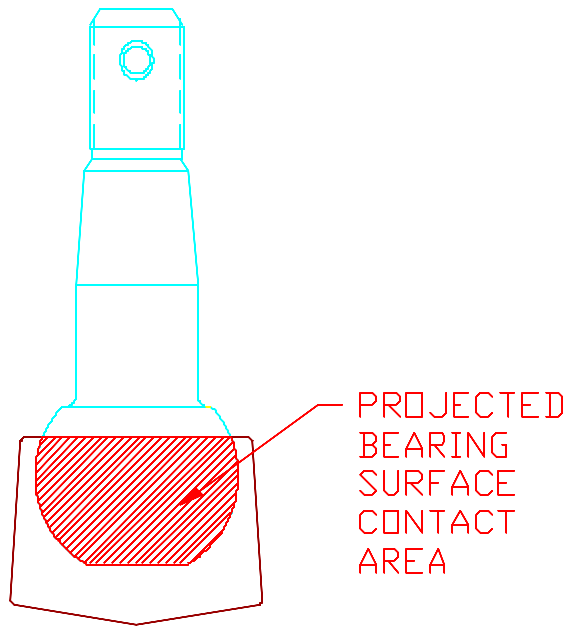


Once the window is defined and the housing thickness is established, a stress analysis can be conducted. Load to be used for this analysis is the maximum impact load multiplied by the sine of the maximum articulation angle + 2 degrees. For non-ferrous bearings, the stress analysis assumes no bearing. Maximum stress on the housing is to be 50% of the yield strength of the housing material. For designs using ferrous bearings, the same loading analysis is conducted, using the metal bearing. For metal bearings maximum stress must be 50% of the housing or bearing material (whichever is lower).



Tie Rod End Ball Sizing – Bearing Stresses

The ball size must be large enough to insure the bearing is not overstressed. Projected bearing-to-ball stud projected surface area is calculated.



Bearing surface contact area must be sufficient to insure projected surface area does not produce stresses beyond the capabilities of the bearing material.

Ball Joint Sizing Based Upon Lubrication Requirements

There are many considerations for the proper sizing of a ball used in tie rod end applications. One of these is the unit pressure exterted on the grease in the socket. This pressure typically should not the pressure carrying capacity of the lubricant selected.

The unit pressure is the sum of the static pressure inherent in the joint and the dynamic pressure due to loading. This is calculated through the following steps. An example is shown *in italics* for a typical 22mm wedge bearing part.

- Static Load
    1. Measure ball joint rotating torque *(.56N-M)*
    2. Calculate ball to bearing surface contact area *(937.9 sq. mm.)*
    3. Calculate the centroidal radius for the diameter of the ball size being evaluated *(9.44mm)*
    4. Calculate drag = torque/radius *(.56 N-M/.00944M = 59N)*
    5. Coefficient of friction for joint lubricant *(0.02)*
    6. Normal Load = drag / coefficient of friction *(59N / .02 = 2950 N)*
    7. Static internal joint pressure = normal load / ball to bearing surface contact area *(2950N / 937.9 sq. mm = 3.15 N/sq. mm)*
- Dynamic Load
    1. Establish working radial load from dry park steering cycle data *(5100 N)*
    2. Calculate projected ball-to-bearing area *(270 sq. mm)*
    3. Working pressure = load/area *(5100N / 270 sq. mm = 18.9N/ sq. mm)*
- Internal Joint Pressure
    1. Sum Static and Dynamic loads *(3.15+18.9 = 22.05 N / sq. mm)*

Converting the above example to a 26mm ball provides the following

- Static Load
    1. Measure ball joint rotating torque *(.56N-M)*
    2. Calculate ball to bearing surface contact area *(1310 sq. mm.)*
    3. Calculate the centroidal radius for the diameter of the ball size being evaluated *(11.16mm)*
    4. Calculate drag = torque/radius *(.56 N-M/.01116M = 50N)*
    5. Coefficient of friction for joint lubricant *(0.02)*
    6. Normal Load = drag / coefficient of friction *(50N / .02 = 2000 N)*
    7. Static internal joint pressure = normal load / ball to bearing surface contact area *(2000N / 1310 sq. mm = 1.53 N/sq. mm)*
- Dynamic Load
    1. Establish working radial load from dry park steering cycle data *(5100 N)*
    2. Calculate projected ball-to-bearing area *(377 sq. mm)*
    3. Working pressure = load/area *(5100N / 377 sq. mm = 13.5N/ sq. mm)*
- Internal Joint Pressure
    1. Sum Static and Dynamic loads *(1.53+13.5 = 15.0 N / sq. mm)*

Converting from a 22mm to a 26mm ball reduced the internal joint pressure in steering cycle conditions from 22 N/sq. to 15 N/sq.mm.

Tie Rod End Ball Sizing - Benchmarking

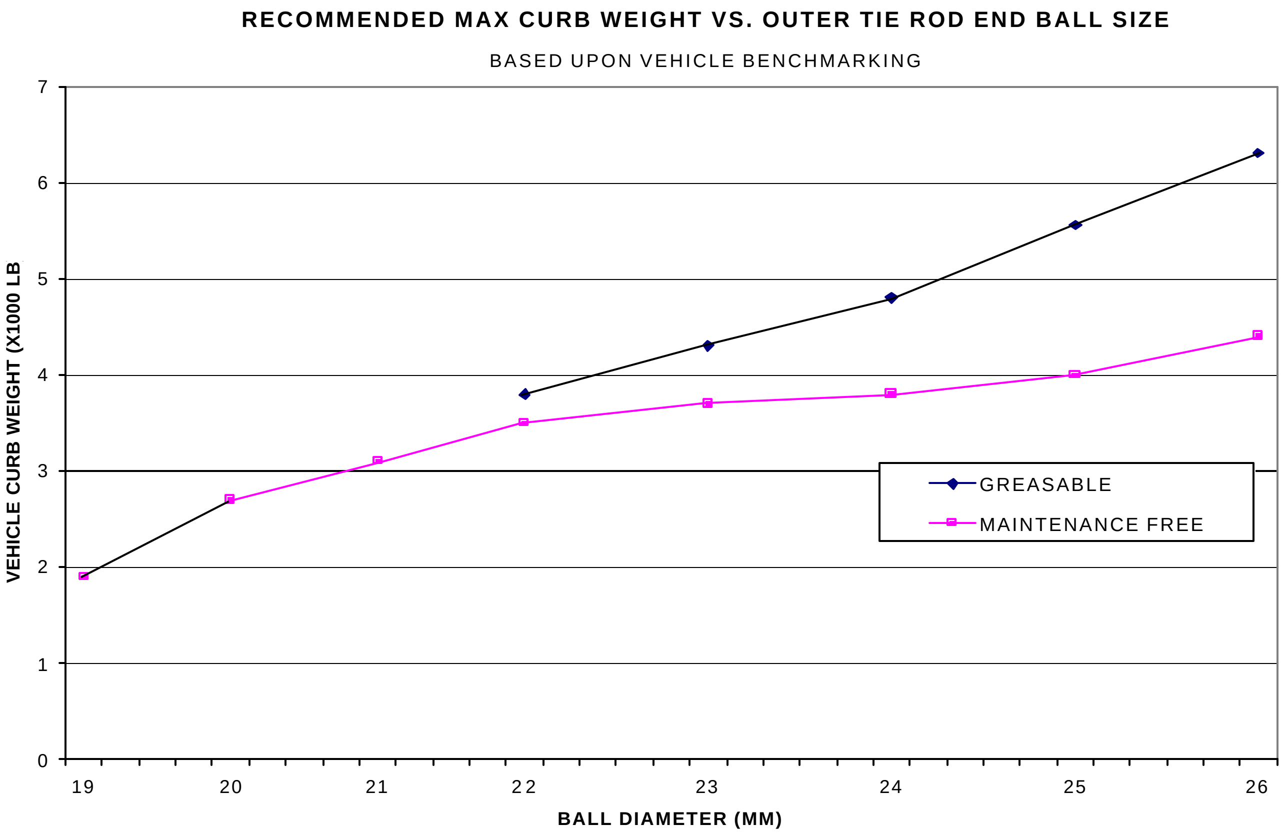
There is a dangerous temptation to size the ball in an outer tie rod end based upon vehicle gross weight or front axle weight. This is dangerous because two vehicles of comparable weights may have significantly different loading or articulation requirements. Loading is a function of many factors, including the following:

- Tire size/pressure. Outer tie rod end peak loading is typically in impact or pothole events. As a wheel passes through a pothole, the suspension begins to go into rebound when the far end of the pothole is struck. A larger wheel/tire combination will strike the edge of the pothole more tangential to the diameter of the tire, reducing the impact to the vehicle suspension. A larger cushion of air due to a larger or high profile tire reduces the impact transmitted through the suspension.
- Suspension tuning/shock absorber valving – A vehicle with stiff rebound control will prevent the tire from dropping into the pothole as deeply as a vehicle with softer rebound control. Therefore, the tire will strike the edge of the pothole more tangential to the diameter of the tire, reducing the impact to the vehicle suspension.
- Roll stiffness / Stabilizer Bars – Like shock absorber valving, a stiff stabilizer bar will prevent the wheel from dropping as deeply into a pothole, reducing the angle of impact.
- Suspension compliance – Rubber bushings at control arm inner pivots can absorb more of the total system impact energy, reducing the amount to be reacted by the tie rod outer end.

It is, however, reasonable to use ball size vs. curb weight for benchmarked vehicles as a “reality check” for a proposed design. If a proposed design is seriously beyond the limits of vehicles with comparable front-end weights, there should be cause for an additional review of the design inputs chosen for that design.

As shown below Powers & Sons has benchmarked numerous vehicles for outer tie rod end ball size vs. front-end weight.

Curves are presented for greasable and maintenance free designs. The “greasable” curve has a distinct “S” shape. This is because the surface area increases as the square of the ball diameter. The greasable joints shown typically used a steel bearing, while the maintenance free designs used a plastic bearing. No greasable designs were observed with ball diameters under 22 mm.



Chassis Alignment Toe

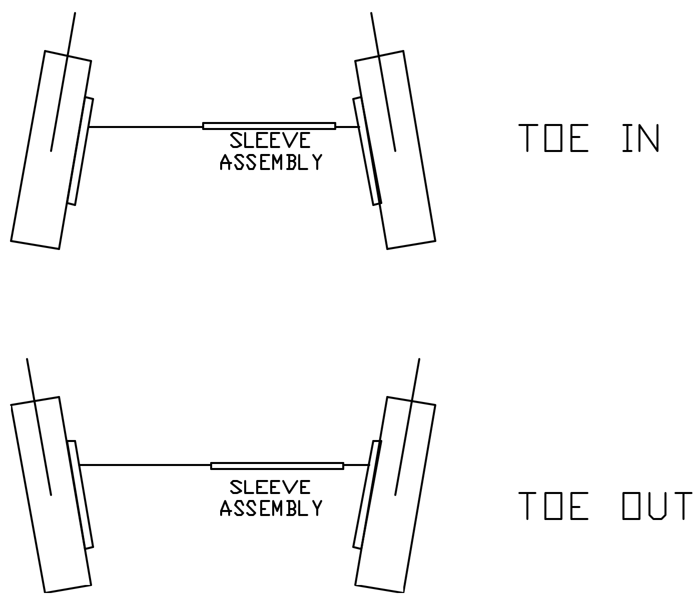
Chassis Alignment toe is defined as the angle between the tires. Toe in is when the front of the tires angle towards the center of the vehicle. Toe out is when the front of the tires angle away from the vehicle centerline.

Toe settings affect vehicle handling, tire wear and fuel economy.

Steering linkage length is adjustable to vary toe settings. This adjustability is necessary on a new vehicle to account for tolerance stack-ups in the frame and suspension components. Toe must be adjustable in service to account for changes due to wear and impact damage (such as potholes).

Typically, steering linkage length adjustment is accomplished in one of two ways. On a rack and pinion vehicle the inner tie rod will have a male thread, while the outer tie rod end will have a female thread. Turning the inner end clockwise will make the tie rod assembly shorter. Turning the inner end counter clockwise will make the tie rod assembly longer. After the length is set, a jam nut on the inner rod is torqued against the outer end to the lock the length

On other vehicles, where the inner and outer ends are anchored, a turnbuckle adjustment is typically provided. A right hand male thread is put on the outer end. A left-hand male thread input on the inner end. A tube with left-hand threads on one-end and right hand threads on the other connects the inner and outer ends. Rotating the tube in one direction lengthens the steering linkage. Rotating the tube in the other direction shortens the steering linkage. Clamps on the outside of the tube are tightened after the toe is set to lock the length.



Camber

Camber is an angle formed by a line drawn through wheel center measured to a vertical line. It is the leaning in or out of the tire at the top. Improper camber may cause abnormal tire wear. The wear will be a smooth uniform wear pattern around the inside or outside edge of the tire, depending if camber is negative or positive. The reason this wear pattern develops is due to a difference between the rolling radii of the inside and outside edge of the tire, or having a difference in the inside and outside diameter or the shoulder areas.

The greater the camber, the greater the difference in shoulder diameter, which not only can cause tire wear, it can also cause a vehicle to pull or lead. This effect is quite similar to riding a bicycle. When leaned a bicycle will turn in the direction it is leaned. Therefore, if a vehicle has a difference in the camber settings side to side, the vehicle will pull or lead to the wheel with the most positive camber setting.

The preferred settings of camber or alignment in general are set to maximize tire life, isolate road shock, enhance stability, maintain bearing load and reduce stress on suspension components.

Most vehicle manufacturers will specify a slight amount of positive camber on the front wheels. By leaning the tire out at the top it is possible to project more of the vehicle load toward the larger inner wheel bearing, thus aiding in reducing stress on other suspension components. Another advantage of positive camber is that it may assist in driving stability. On vehicles with independent suspension, the rear wheel negative camber may be used again to aid the vehicle stability.

When positive camber is used on each front wheel the two front wheels are pulling against one another. This sets up two opposing forces with each wheel trying to roll outward away from the center of the vehicle, and the end result is improved straight line stability.

Caster

Caster is an angle formed by a line drawn through the steering axis, measured to vertical, as viewed from the side. When the top of the steering axis is tilted towards the rear of the vehicle it is known as positive, and when tilted towards the front it is known as negative.

The primary function of caster is to project vehicle load. Most manufacturers today will specify a positive caster setting for a variety of reasons, however a positive setting is mainly specified to improve handling. When a vehicle incorporates positive caster, the load (vehicle weight) is projected through the steering axis out in front of the tire point on contact at road surface.

When the load is projected in front of the tire, the tire will want to follow the load down the road. This condition is sometimes called caster lead, as the load is proceeding the tire. As caster is moved more positive the more stability the vehicle will have. However, if caster is excessively positive it may cause additional problems; such as hard

steering, and a high-speed shimmy condition. The increased effort in steering is due to the load projection effects of positive caster. With positive caster as the wheels are turned the weight is picked up, and the farther out the load is projected in front of the tire the more difficult the vehicle becomes to steer or turn the wheels. Because of this, some manufactures may recommend a negative caster setting in order to reduce steering effort especially on those vehicles, which do not have power steering.

The high speed shimmy can be brought about by much the same reason, again the farther out the load is projected, the more determined the wheels are to follow. If, while in motion, the wheels hit bumps or dips in the road, they will deflect slightly from a straight-ahead position. When caster is excessively positive, as the wheels deflect, and return. They return so fast and hard, that they overshoot the straight-ahead position. Once the wheels overshoot, they now must try again to correct, or return and the end result is a constant correct and overshoot condition, which causes a shimmy to develop. Caster normally isn't directly associated with tire wear. It can cause camber to change.

Steering Axis Inclination

Steering axis inclination, sometimes referred to as S.A.I. or K.P.I. (king pin inclination), is an angle that is formed by a line drawn through the steering axis and measured to the same vertical reference point from which camber is measured. This angle is viewed from the front of the vehicle.

As previously stated SAI is the leaning in of the steering axis, and by doing this the spindle of the vehicle will be forced to travel in an up and down arcing motion. The high point of travel is when the wheel is in a straight-ahead position.

Whenever the front wheels are turned from straight ahead, the spindles are arcing down trying to force the wheel through road surface; since this is impossible. Therefore, the vehicle raises up. In motion when the wheels are turned, the weight of the vehicle is actually trying to force the wheels back to straight-ahead position. The greater amount of SAI used, the more determined the wheels would be to remain or return to a straight-ahead position. In recent years vehicle manufactures have been using higher amounts of SAI especially on those vehicles equipped with a strut suspensions design. On a vehicle equipped with the short arm long arm (SLA) the average SAI maybe is 7 to 10 degrees, and a strut suspension may sue as much as 10 to 17 degrees of SAIO. This may also explain why some vehicle can exhibit a large difference in the side to side camber or caster and still not have a pulling or handling problems when driving.

SAI is designed into the vehicle to keep the wheels in a straight-ahead position. Help the wheels automatically return to straight ahead after making a turn, improve directional stability thus reducing the need for additional positive caster, aid in placing more load on the larger inner wheel bearing, and assist in maintaining straight line control of the vehicle when braking.

Scrub Radius

Scrub Radius s the Distance between the steering axis pivot point and the center of the tire. When the steering axis and the wheel center line are both extended down to an intersecting point it may be noticed that these lines do not intersect. When the wheels are turned the wheel's center line pivots around the steering axis at this intersecting point.

When these lines intersect below the road surface, it is known as positive scrub radius, and when the lines intersect below the road surface, it is known as a positive scrub radius, and when the lines intersect above the road surface, it is known as negative scrub radius.

Although scrub radius is not an angle, it is a very important part of the suspension geometry. There are four factors that determine a vehicle's scrub radius, camber and SAI are two, and the other two are tire size and wheel offset.

Application of scrub radius may vary due to suspension design. In general, most rear wheel drive vehicles will use a positive scrub radius. Vehicle load is projected on the inboard side of wheel centerline, a vehicle that incorporated a positive scrub radius will have a tendency to force the wheels outward, away from vehicle center, when in motion. This is one of the primary reasons that most rear wheel drive vehicles will specify a static toe setting or toe in.

Vehicle manufactures design the amount and type of scrub radius for several factors. First, to provide the driver a feel for the road. Project vehicle load to the larger inner wheel bearings. Assist in providing stability under adverse road conditions. Work in conjunction with static toe settings to help bring about a running toe of 0. In some cases assist in maintaining control in the event of a blow out. Provide the necessary resistance to help the tire maintain control, or contact the to the road surface.

Turning Angle

The turning angle, often referred to as toe out on turns, is an angle that is determined by the positioning of the steering arms. When a vehicle is involved in a turn the inside wheel of the vehicle will be turning on a smaller radius than the outside wheel.

If the front wheels turned at exactly the same amount in degrees, there would be some side scuffing and abnormal tire wear. But the wheels tow out as the vehicle turns, which is a function of the steering arms.

Several factors are considered when a manufacturer design the toe out or turns of a vehicle, including wheel base, tread width and tire design. A basic theory relating to the design of the steering arms is to extend a line from each front wheel steering axis to the center of the rear axle, and the steering arms should be made to the same angle. This would be known as an Ackerman steering system and may be the most common design in use today.

Turning angle readings are normally only checked on vehicles where some type of collision damage is suspected. If these readings do not fall within the manufacturer's specifications, normally at 1 degree to 1.5 degree spread the most probable cause is a bent steering arm that should be replaced.

Geometry Pt 14-Pt 12

Suspension and steering linkage geometry points are numbered for kinematic analysis. For our purposes, 2 points are key, point 12 and point 14. Point 12 is the center of the outer end ball stud sphere. Point 14 is the center of the inner end ball stud sphere. When geometry points are provided for jounce, rebound, inside turn and outside turn, <u>points 12 and 14 are what are significant to us.</u> Regardless of whether it is left side or right side of the vehicle, 12 is outer and 14 is inner.

5 Geometry conditions

There are numerous suspension geometry conditions, which are of interest to us. They are:

- Design height/straight ahead – Design height is curb weight plus two passengers& fuel.

- Design/inside turn – Inside turn refers to a plan view. In right turn, the right wheel is the inside wheel. In left turn, the left wheel is the inside wheel. For rack and pinion steering geometry, it is assumed the geometry is symmetric left to right. Left turn and right turn is not used. Inside turn and outside turn is used.
- Design/outside turn – Outside turn refers to a plan view. In right turn, the left wheel is the outside wheel. In left turn, the right wheel is the outside wheel. During turn, the inner end rod will articulate in the plan view and the outer end ball stud will rotate.
- Full jounce/straight ahead – Full Jounce is the condition where the road wheel is up in the furthest position possible, limited by the jounce bumper. Occasionally, the term modified jounce will be used. Modified jounce assumes that the jounce bumper has been removed and travel is limited by metal-to-metal contact. In the front view, jounce will determine articulation angles in one direction of the inner end and outer end.
- Rebound/straight ahead – Rebound is the condition where the road wheel is down in the furthest position possible. In the front view, rebound will determine articulation angles of the inner end and outer end in the opposite direction of jounce travel.
- Jounce/inside turn, Jounce/outside turn, Rebound/inside turn, Rebound/outside turn – These conditions may define slightly larger articulation and rotation angles than the constraints described above.

12-14 length

The distance between points 12 and 14 is significant for two reasons; articulation angle and buckling strength.

The vertical coordinate of point 14 does not change, it is attached to the rack, which does not have vertical movement. Point 12 at the wheel does move vertically with jounce and rebound. For a given amount of wheel travel (typically 125mm jounce/ 150mm rebound in trucks), a greater 12/14 length will require higher articulation angles. Two vehicle trends are at work here. First, the outer end (12) wants to be outboard as far as possible to yield an optimum toe curve. Second, the gear/frame mounting points should be outboard as far as possible to give rack stability. When these points move out, point 14 needs to move outboard accordingly. Short 12-14 length results in high articulation angles. Articulation angles exceeding 30 degrees require extreme design measures (a.k.a. expensive) to maintain joint strength.

Buckling strength is reduced as 12-14 length increases. A note regarding jam nut location. The highest buckling stresses are at the center of the loaded bar. The section necks down in the threaded area where the inner rod attaches to the outer end, making this area weakened for buckling. For this reason the outer end should be as short as possible. It may be necessary to lengthen the outer end to accommodate operator toe setting in the assembly plant.

Looking for loads

Given the kinematics "stick geometry" of point 12-14, we can begin to design an inner and outer tie rod. From here on, it's simple calculations and packaging. How big a ball socket do we need for loading? How big a rod do we need for the loads? Do we have to package around anything? How to we attach to mating parts?

Outer end (point 12) design is driven by load data. There are two primary load conditions for high loads on a rack and pinion tie rod. The P3-26D pothole course provides one of these conditions. P3-26D is approximately ¼ mile long and includes 3" deep steel rimmed potholes at 90 degrees to vehicle travel and 73 degrees to vehicle travel. Some potholes are wide and hit with both wheels, others are narrow and only hit with one wheel. The most severe single pothole event in P3-26D has one wheel going in a pothole while the other side is going up a ramp. This pitches the pothole side more severely into the pothole, increasing the load. Ford requires 1000 circuits of P3-26D for truck as part of the Silver Creek test. Data for this is usually not available until late in the program.

For preliminary design we must use curb push-away loading. In this event, a vehicle is parallel parked with the right side wheels contacting a curb. The driver conducts a lock-to-lock steering cycle, pushing the vehicle away from the curb with the leverage of the

front wheels. Obviously, the loads input to the tie rod are limited by the output rack force of the steering gear. This calculated rack load output capacity is used for sizing the ball joints and rod of the inner/outer tie rod assembly.

We now have enough data to design an outer end ball stud. This is an iterative process. First we assume a ball diameter. For UPN254, we may begin at 30mm. If we have the liberty of dictating the gage length (center of ball point 12 to the knuckle face at the large end of the taper), we want the shortest length we can package with a seal. Seal package for articulation requirements without pinching the seal is the limiting factor in how short a ball stud gage length can be.

If we can reduce the ball diameter we can shorten the gage length. The highest stress on the ball stud is at the knuckle interface. The greater distance from this point to the ball center (load input point), the higher the bending moment. The higher the bending moment, the larger shank diameter required for the stud. As the shank diameter increases, the less forging window is available for articulation. The larger the forging window the less pull out strength available.

To get the required pull out strength back, two design options exist. First is to increase ball diameter. Looking at the socket down the axis of the ball stud, we see an elliptical forging window. The major axis of this ellipse is determined by articulation requirements and ball shank diameter. On the minor axis of this ellipse, there is overlap of the forging window and stud ball. This overlap is what gives pull out strength. Increasing ball diameter increases pull out capability by increasing the stressed surface area.

A second alternative is to use a hardened steel upper bearing in the socket. This allows increased pull out loads without increasing ball diameter by allowing higher pull out working stresses. Typically 30,000 PSI can be used for non heat-treated parts. For heat-treated parts, this increases to approximately 50,000 PSI. A steel upper bearing can be used in a lube-for-life application. It is being used in 26.3mm sockets in B-Van tie rods and 2004 Durango outer tie rods. For aluminum WK upper control arms, a 30mm hardened dual seat bearing was required for high articulation with good pull out in an aluminum socket.

Customer input is required for greasable/maintenance free outer ends and rotating torque targets. These will dictate joint design type.

Knowing the ball stud shank and ball diameter drives the rest of the outer end socket. The taper length should be a minimum of 2 times the diameter of the stud at the large end of the taper. Taper angle can be 1:8 for a forged steel knuckle. Taper angle must be 1:6 for cast or aluminum knuckles. The taper diameter at the large end, the taper angle and taper length dictate the thread size on the stud. If the knuckle is made from aluminum the ball stud must be coated to prevent galvanic corrosion.

While the outer tie rod end is radially loaded, the inner is axially loaded. Again, the articulation angles and loads dictate ball size. The inner end will have to be maintenance free because the rack boot seal will enclose it.

Typically the inner end is attached to the rack by an external thread formed on the inner end housing. We will need to know the thread details on the end of the rack. Typically the face on the inner end of the inner end housing acts as the travel stop. We will need to know details of diameter and location of this stop from the gear supplier.

Because the rack boot is sealed to the gear housing and the inner tie rod, the boot expands and contracts during steering travel. This causes a significant pressure build up during contraction and a vacuum during extension. Historically, the rack and pinion gear has used a cross tube to vent from one rack boot to the other. There is a school of design, which uses a hollow rack with a venting mechanism through the inner end-to-rack attachment to eliminate this external tube. We will need to know if <u>the gear will be internally or externally vented.</u>

If the wheel turn angle (also called wheel cut) is severe, there may need to be an <u>offset in the outer end forging</u> to prevent the wheel from contacting the outer end forging during full turn. This will dictate the shape of the outer end forging and affect buckling requirements of the inner/outer assembly. System stiffness will also be affected.

System stiffness will be determined by the rod offsets and bar diameter. The thread size of the inner/outer joint is much less affected by strength requirements than by stiffness requirements.

The boot will attach to the inner rod by seating in a radial groove in the rod. <u>The boot supplier typically determines details of this groove</u>. For ease of toe set, <u>knurls or an extruded hex</u> is typically put on the inner rod just inboard of the threads.

Bearings

Steel Dual Bearing Type-

The steel dual bearing uses to concentric half spheres. The ball stud rides on a bearing that handles rotational movement. While the bearing fits in a machined socket to control vertical forces. Distributing forces over two surfaces reduces wear. The preloaded spring also assists in compensating wear.

Nylon Wedge Bearing Type-

This type of bearing has full ball contact. It uses a tapered forging bore and a spring to compensate for wear. A grease fitting can be used to purge contaminates from the system.

Pre-Cap Bearing Type-

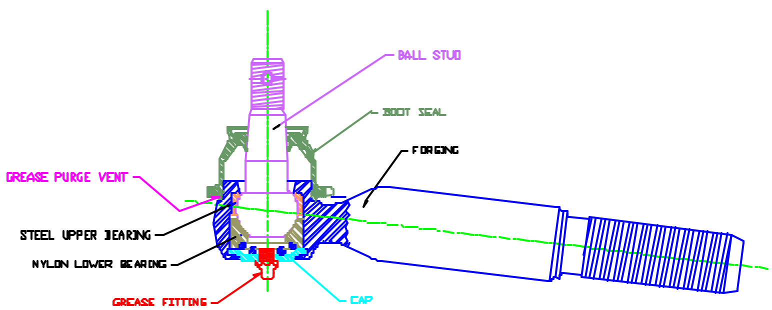
With this type of bearing a plastic bearing surrounds the ball stud. Wear is normally compensated by the initial preloaded conditions set at the factor. This is normally a lubed for life application.

Dual Seat Bearing Type-

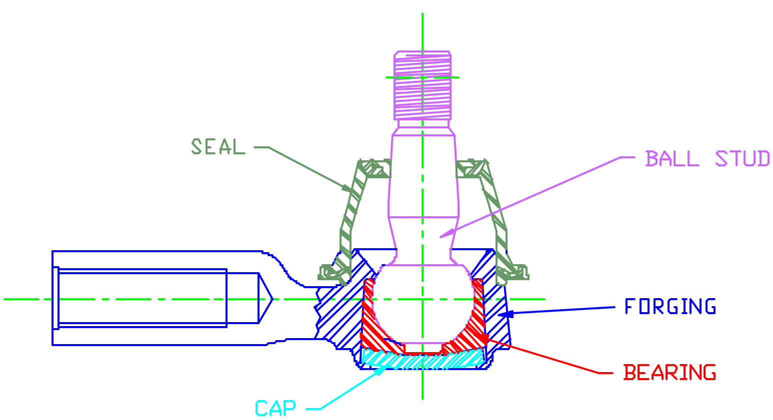
This bearing uses a metal upper bearing and a plastic upper bearing. The upper bearing has a spring to reduce the effects of wear. When the bearing wears, the spring forces the bearing down on the ball reducing lash.

Grease-able Vs. Lubed for Life Joints

The simplest way to explain the difference between a grease-able and a lubed for life joint is in what their respective names implies. A grease-able joint receives new grease injected into it via a grease fitting and grease gun at regular maintenance intervals. The lubed for life joint only uses the grease it had when it was first made. The reasons why one is used instead of the other are based on several engineering, marketing and cost factors.



GREASABLE TIE ROD END



LUBED-FOR-LIFE SOCKET ASSEMBLY

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A primary failure mode of ball joints is the degradation of the grease. Once the grease is contaminated, washed out or oxidized the wear of the internal parts is amplified and it is only a short time until joint failure occurs. With a lubed for life joint, once the "seal" is compromised it allows for contaminants (such as dirt, water and salt) to degrade the grease. A grease-able joint has the same problem but receives new grease on a regular basis. The greasing of the joint helps to purge out the contaminated grease with new fresh grease. A grease purge feature must be included in the greasable joint. If the purge vent allows grease to leave the joint, it may also allow contamination to enter the joint. Without regular maintenance the greasable joint could fail quicker then the lubed for life joint. In a way the lubed for life joint is a "safety factor" for negligence of regular maintenance.

Marketing-

In the auto industry there has been a push for maintenance free cars and trucks for its average consumer (you and me). It has gotten to the point where the owner only has to put gas in the car, take it to the mechanic for an oil change and put air in the tires (even putting air in the tires is becoming optional). Ball joints are not an exception. A lubed for life joint can be installed at the assembly plant and forgotten for the life of the part. It is important to note that the life of the part is not necessarily the life of the car. The life of the part varies based on design, manufacturing capabilities, in-vehicle location, type of vehicle, etc. In comparison the grease-able joint requires regular attention.

Higher load applications tend to use grease-able joints because they use steel-on-steel bearing surfaces. Commercial and fleet customers believe the greasable design is more robust and demand greasable joints. The reason is that a properly greased, grease-able joint will operate far longer then lubed for life which makes them more appealing than a lube for life joint in some situations.

Cost-

There are several cost variables that can be looked at when choosing the best joint for the job. First there is the cost of the different parts in the different joints. For example a grease-able joint uses a grease fitting (approximately $.01-. 03 per joint) vs. a lubed for life joint that requires a better seal (approximately $.25 per a joint). In general, a lubed for life joint costs more to manufacture than a grease-able joint.

The life span of each part makes a difference also, with regular maintenance a grease-able will likely out last a lubed for life joint. It's bearing design and fresh grease allows it to last longer then the lubed for life design. Thus having to replace the lubed for life joint more often is a cost consideration.

The Maintenance Free-Greasable Joint -

Automotive marketing has blurred the distinction between consumer and commercial vehicle applications. Consumers who do not want to be bothered with higher maintenance products are purchasing vehicles that were designed for a commercial maintenance schedule. To bridge the gap, greasable joints are now being manufactured with boot seals with one-way valves to allow grease to purge out but prevent contaminants from entering. Premium greases with high molybdenum content are being used. If the joint does not get field lubricated, the premium grease will provide good, high mileage wear characteristics. If the joint does get field lubricated, the premium grease will be purged out.

In conclusion grease-able vs. lubed for life joint style selection should to be based on the application of the joint, intended life and cost. Neither type of joint is necessarily better then the other as long as the design meets the necessary requirements.

BALL STUD HEAT TREATMENT

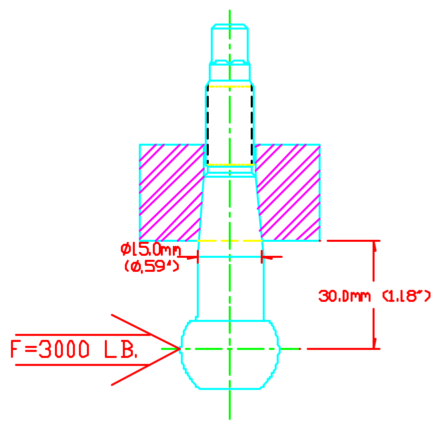
Ball stud heat treatment is critical to the performance of a suspension or steering system. Load carrying capacity and wear characteristics are affected by the ball stud heat treatment.

Load Carrying Capacity:

A typical tie rod pothole load may be in the range of 4000 lbs. Maximum stress on the ball stud is defined by the equation:

σ = M*c/I + 4/3 F/A Where Mc/I is the bending stress and 4/3 F/A is the sheer stress.

Calculating the ball stud surface stress (in English units) for the example below,



M= 3000*1.18= 3540 in-lb
C= 0.59/2 = 0.295 in
I = $\pi d^4/64$ = .00595 in^4
R = 0.59/2 = 0.295 in
A = $\pi * R^2$ = .273 in^2

σ = 3590*.295/.00595 + 4/3*3000/.273
= 177991+ 14652 = 192643 psi

In this example, if the yield strength of the ball stud at the surface exceeds 192643 psi the part will never go into yield and theoretical infinite life can be assumed.

Yield strength of non-heat treated cold drawn steel is 60-100 KSI. Additional strength is required to meet the combined bending and shear stress of our model. The field of steel heat treatment is too vast and complex to cover in detail here. However, there are two typical methods of heat treatment used for ball studs, through hardening and case hardening.

Through hardening of the stud can raise the yield strength to the 200KSI range. However, the resulting part will not be very ductile. Ductility is a significant factor in fatigue life.

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Case hardening results in a part with a hard shell at the surface, but a relatively soft, ductile core (rather like an M&M). This results in high bending strength with good ductility. The case hardening can leave the part with a tensile strength of approximately 130KSI (measured on a classic tensile bar). At the case, hover, there are residual compressive stresses which have been measured in the 110KSI range. In bending, it is necessary to overcome this residual compressive stress before even observing positive surface strains. Case hardening the ball stud in the above example results in a bending yield strength of 240KSI (110+130). This results in a 24% margin of safety.

Cotter Pin Vs. Prevailing Torque Nut

A review of recall history or warranty trends will show that one of the biggest problems in the industry today is loose fasteners. In steering linkage, a ball stud with a loose nut will result in separation of the joint and can impair steering function. If the nut falls off, separation will occur. If the nut is just loose, fatigue life of the ball stud is reduced.

Advances in fastener technology have resulted in “smart” nut runners which measure torque vs. angularity as a nut is tightened. SPC audits of nut torque is also in wide use. These help significantly. The design of the nut can also affect this condition.

In the world of retaining nuts for ball joints, there are two types commonly used. . In ball stud fasteners, cotter pins with “castle” nuts or prevailing torque nuts can be used to keep a fastener from loosening. The cotter pin slides through a hole in the threaded portion of the ball stud and across slots on the castle nut. The cotter pin acts like a locking pin that will not allow the nut to loosen or back off. The prevailing torque nut uses increased friction designed into the nut to keep the nut from loosening or backing off. The increase friction is a result from either a nylon washer built into the nut (nylock nut) or the nut inner diameter being oblong (crimp lock nuts).

Castle Nut and Cotter Pin

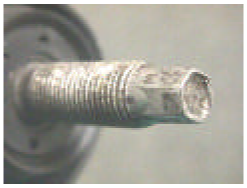
Nylock Nut

Engineering-

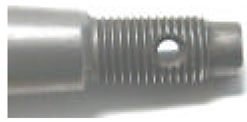
The cotter pin/ castle nut setup works well. When the fastener is properly designed the pin will not fit if the nut is not tightened. The cotter pin acts as a 100% “poke-yoke” inspection method. Additionally, the pin prevents the nut from loosening after installation. On the assembly line the cotter pin arrangement can cause some problems since the assembly worker must first torque the nut, then change the angle of the nut so that the slots on the castle nut align with the hole. This is a labor-intensive installation.

If the ball joint has low rotating torque and utilizes a prevailing torque nut, the stud needs to contain another source for holding the stud from rotating while the nut is being tightened. This is often done by use of either an internal or external hex head. This requires a unique tool on the assembly line for quick and effective assembly. One over-looked aspect of internal vs. external hex head is that the external hex can damage adjacent parts in the shipping container due to it's sharp protruding hex. The internal design doesn't have this problem.

The difference between the nylock nut and crimp lock nut is that the crimp lock nut has significantly more variation in the prevailing torque that it supplies then the nylock nut. This introduces greater variance into the amount of hut torque actually used to secure the joint.



External Hex Ball Stud



Cotter Pin Stud w/dog point

Internal Hex Ball Stud

Cost-

There are several costs to consider. First is the manufacturing costs of the components. The cost of the cotter pin and hole for the castle nut design is less then 1 cent each in high volume production. Nylock nuts are more expensive than castle nuts or crimp nuts. The largest single design cost associated with a prevailing torque nut system is the cost of installing the anti-rotation feature on the ball stud (typically an internal or external hex head formed on the stud during the heading operation). An external hex typically costs 3-5cents. An internal hex can be in excess of 30 cents.

Often an assembly plant will request the hex head design because it is less man power thus cheaper to assemble. The cost of assembly tooling is usually cheaper for an internal hex ball stud than it is for external hex head because of the design of the assembly tool. The design engineers sometime look for the cheapest design with out realization of assembly costs. The breaking point is where the piece cost multiplied by the amount of pieces becomes more then the cost of more expensive assembly.

Both types of nut setups work well. It is important to keep in mind assembly considerations, cost and reliability when choosing the appropriate system.

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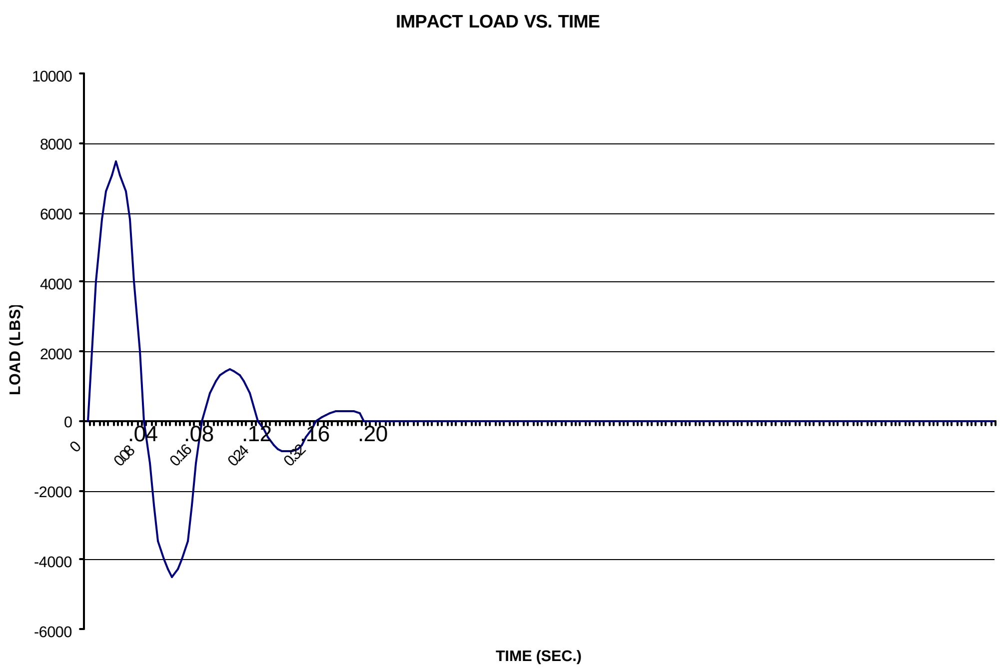
STATIC VS DYNAMIC LOADING

When developing a steering linkage, design consideration must be given to three loading conditions. These are static, fatigue and single event impact. Static and fatigue conditions are fairly well understood. Single event impact loading, however is less well recognized.

The static consideration is relatively easy. Static load inputs are obtained for steering cycles or curb pushaway. These are typically a function of the maximum steering gear output force available from the steering gear. Classic design calculations and tools such as finite element analysis can be used to insure adequate strength exists in each component of the linkage.

Fatigue analysis is also fairly straightforward. Load inputs are taken from the customer provided durability cycle data. It may be in the form of a block-cycle histogram or a single load taken from the S-N curve. Basically, if the fatigue stresses are kept below the tensile yield, infinite life can be attained. Fatigue testing is typically conducted at a frequency of approximately 3 Hz.

Dynamic impact load analysis is more complicated. Below is a typical tie rod load vs. time trace for a pothole impact load:



This data was obtained by attaching strain gages to the tie rod. The gage is, in effect, an electronic device which varies resistance as it is stretched or compressed. A 10 volt input is sent to the gage. The voltage output will be less than 10 V depending on how much it is stretched. The tie rod strain gage is calibrated to a load in a tensile test machine. During road load events, the voltage output is monitored and converted to loads.

Note that, in our example, zero to maximum load occurs in approximately .02 sec. Load duration is very short. Depending on the dynamic system response of the steering linkage involved, much higher loads can be tolerated in dynamic impact than in a static condition.

As an example, consider drag link buckling strength. A long, slender truck drag link may buckle under a 6000 lb. static compressive load. A sinusoidal fatigue load of +/- 3000 lb. at 3 Hz may cause this part to whip elastically over 20mm off the load axis. This same part on a vehicle driven through a pothole may see over 9000 lb. compressive load without plastic deformation. A reasonable analogy is the old magicians trick of forcing a plastic drinking straw through a potato. Done slowly, the straw will buckle under load. Done fast enough, the straw goes through the potato.

This phenomenon defies conventional analysis.

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