AURORA BEARING COMPANY

ROD ENDS

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Introduction

This publication consists of reprints of articles regarding Rod Ends and Spherical Bearings that originally appeared in the publications TRACKSIDE, Racecar Engineering, GRASSROOTS Motorsports, RACE TECH, Victory Lane, Professional Motorsports World, HISTORIC RACING Technology, as well as one by independent technical writer Wayne Scraba. There is also an excerpted article on anti friction and plane bearings that originally appeared in Race Engine Technology, and an article on rolling element bearings that appeared in Late Model Racer. There is also an excerpted article on fasteners that originally appeared in GRASSROOTS Motorsports. In addition, it also includes select general engineering and technical information. Aurora offers this for informational purposes, and suggests that the reader use these as just one tool in gaining knowledge on the subject.

We suggest consultation with rod end manufacturers and their engineering departments, along with their appropriate product literature. Engineering information should be verified via appropriate material, product, and engineering reference material. We also feel it appropriate to consult recognized car and component manufacturers, racers, and related professionals, as well as independently produced technical literature as part of the knowledge building process.

About Us

In 1971 a new company entered the rod end and spherical bearing marketplace. Founded by a group of bearing professionals with backgrounds in all aspects of the rod end and spherical bearing industry, this new firm, Aurora Bearing Company, soon became a major force in the rod end industry.
The 2-piece design CM/CW series rod ends introduced Aurora to the U.S. market. The design was not a new idea, but it took the engineering and manufacturing expertise of the new company to make it a commercially available and economical product. Now offered by rod end manufacturer's world wide, the 2-piece, all steel rod end is now the standard economy/commercial bearing in the U.S. market.

As Aurora quickly became known for a high quality engineered product and a strong commitment to customer service, the firm dramatically increased its product range and market coverage and now serves nearly every industrial and aerospace market. These markets include among others: textile and packaging machinery, machine tools, business machines, recreation and exercise equipment, agricultural and off highway vehicles, commercial transportation and high performance racing vehicles as well as military equipment and commercial air and space craft.

Over the years, Aurora Bearing has retained its original business philosophy of furnishing a high quality product at competitive prices. In addition, the company's initial goals of providing prompt delivery and furnishing service with a personal touch have been rigidly maintained.

Aurora Bearing offers a complete line of standard rod end and spherical bearings. We also design and manufacture spherical bearings to meet a variety of applications that require custom engineered units or special materials.

Now marketing products worldwide, Aurora Bearing fields a very competent sales force that is available to assist and provide you with a practical and sound solution to rod end and spherical bearing application problems and challenges.

Rod Ends

John McCrory of Aurora Bearings Tells You What You Need To Know....

PHOTOS BY BOB PESHIA

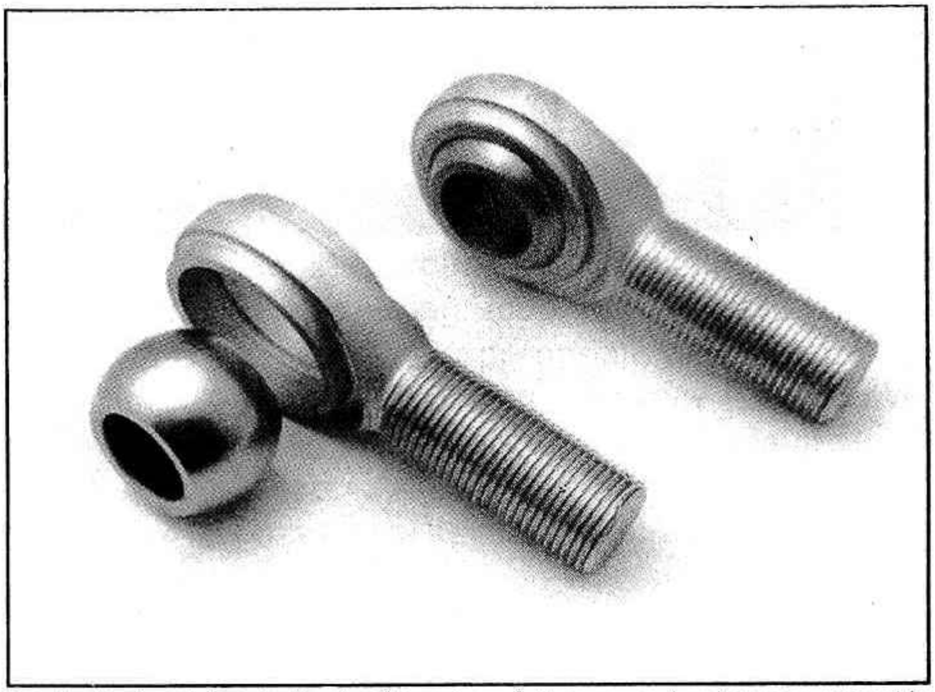
Rod end bearings are an important item that the average racer often takes for granted. Too often they incorrectly assume that one rod end is the same as another. Yet those critical components are what connects the suspension to the frame or connects the wheels to the steering. If your car has the correct rod ends on it, you'll never know they're there. Suffer a failure from an incorrectly used rod end and you'll suddenly be real concerned about your joints.

Following are some questions commonly asked by racers concerning rod end bearings that should help you understand more about those important race car components.

What are the basic types of rod ends used for racing applications?

Rod ends for race cars can be divided into two basic classes. First is the commercial or economy grade rod ends. While there are many configurations of economy rod ends, the only type that should be used for racing application is the fully swaged two-piece design. On these rod ends the body is formed or swaged around the ball so that the race the ball rides on is actually part of the body. This is the only type of economy rod end that has good radial or pull strength along with good axial strength (resistance to the ball being pushed out of the side of the body).

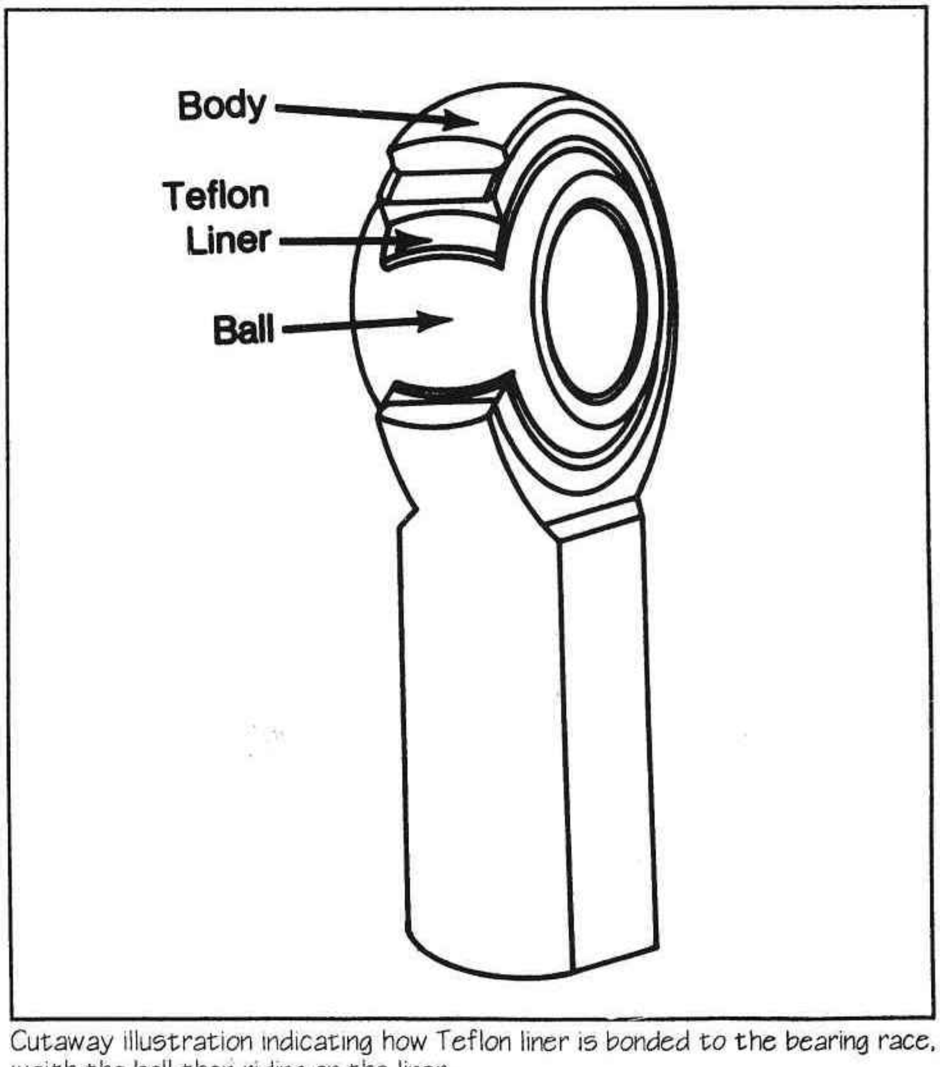
The second type is three-piece precision rod ends. With these rod ends a race is formed around the ball, this ball race insert then being staked into a body. The advantage of this type of construction is that a closer fit and a higher degree of precision is achieved between the ball and race. This is the type of rod end that is also referred to as "aircraft style." The three-piece design allows different materials to be used in the construction of the part to best match it to its application. Races can be made of mild, alloy or stainless steel (brass or aluminum bronze is sometimes used but should be avoided because of its low strength) and bodies are made in mild, alloy, stainless steels, aluminum or even titanium.


Finished ball with finished body of two piece full-swage rod end prior to assembly and finished unit.

What is a teflon-lined bearing?

A teflon (a trademarked product of Dupont) liner in a rod end is a component that allows the unit to be self-lubricating. Any metal on metal bearing needs to be lubricated. It is difficult to oil or grease a unit on a car. Grease fittings should be avoided as they can weaken the part. In addition, the grease on the ball can attract dirt and grit which works its way between the ball and race actually accelerating wear.

A teflon liner eliminates these problems. A liner consists of a carrier component, usually a fabric, which gives compressive strength, a teflon component for lubricity and various bonding resins. The teflon liner is bonded to the race so that the ball actually rides on the liner. The movement of the ball rubs teflon on the ball providing lubrication. Teflon liners are available on both two and three-piece rod ends. These liners should not be confused with virgin teflon which is relatively soft (approximately 10,000 lbs. psi compressive strength). A good composite teflon liner will have a compressive strength of between 40,000 and 60,000 psi. In addition to providing lubrication, the liner also eliminates clearance between the ball and race making for a tighter fit.



Cutaway illustration indicating how Teflon liner is bonded to the bearing race, woith the ball then riding on the liner.

I've heard that teflon liners "beat out."

When people refer to a liner "beating

Damaged rod ends. Note insert on far left rod end is partially pushed out. Unit on far right shows dent in race from severe misalignment. Don't do anything with these rod ends except throw them out!

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P. 53 / Bruce Bennett Photo

On The Cover

Photographer and sometimes-writer Bruce Bennett captured Modified mainstay Frank Cozze doing his thing on the dirt at Orange County. Frank is one intense customer, and we sent a wanderlust-filled Kevin Kovac off to chat with Cozze about what makes him tick. Check it out beginning on page53.

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Columns

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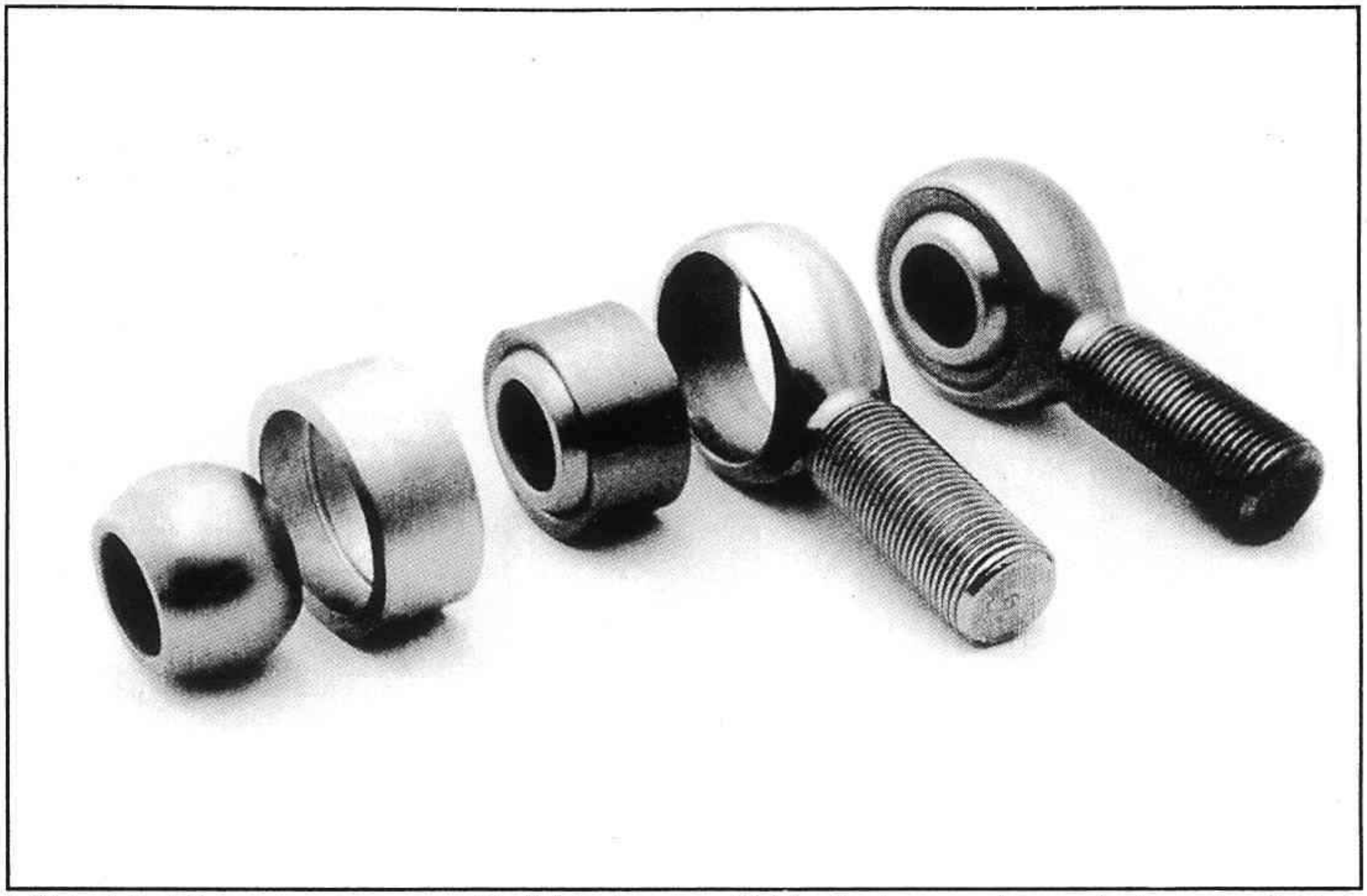
out" they are usually referring to one of two problems. First is the deformation of low strength "self lubricating" liners. Some economy grade rod ends have races that are made of molded plastic, sometimes with a fiberglass filler. Teflon may also be added for lubricity. These units have a compressive strength of no more that 15,000 psi, causing the race to deform way before there is any damage to the body.

The second type of liner "beating out" is failure of the bond between the liner and the race. The self-lubricating nature of the liner makes it difficult to bond to the race. If this bond is not strong enough, the liner will become detached from the race, pieces being spit out with each mis-alignment until there is no liner between the ball and race leaving a rod end with excessive clearance. About the only standard for bond strength is the one included in the government's mil specs for the teflon-lined bearings. It's reasonable to assume if a manufacturer can meet this standard and has a line of military approved bearings, its teflon lined bearings are not likely to suffer this type of "beat out."

Can anything be done about worn out bearings?

Yes...throw them away. There is no safe way to tighten up a worn bearing. Any bearing that is bent or dented in the race should also be thrown out. This also goes for units that show stretching in the threads or head. If a unit is on a component that has been in a hard accident, it should be visually checked for bends or deformation and it would be wise to have it crack-tested as well before re-using it.



Left to right: ball and race prior to assembly, finished insert and finished body prior to being staked together, finished three-piece "aircraft" rod end.

Is buying used rod ends a good way to save some money?

This is one of the more foolish things you can do. Like any piece of hardware a rod end has a finite mechanical life. When you buy a used rod end you don't know if it is at the end of its life or not.

How strong are aluminum rod ends?

7075-T6 aluminum is one of the strongest grades of aluminum and has a tensile strength slightly greater than mild steel. Therefore, two rod ends of similar design, one made of mild steel and one of 7075-T6, would have similar design, one made of mild steel and one of 7075-T6, would have similar strengths. The drawback of aluminum is it is not as forgiving... in other words, it will not stretch or bend as much as mild steel will before breaking. Aluminum rod ends made of grades of aluminum weaker that 70075-T6 will obviously be weaker than the same configuration unit made out of 7075-T6. Also, while there are grades of aluminum that are comparable in strength to certain stainless steels, these grades of stainless are of the low strength types and not comparable to the high strength grades of stainless such as 17-4PH or alloy steels like 4130 and 4340.

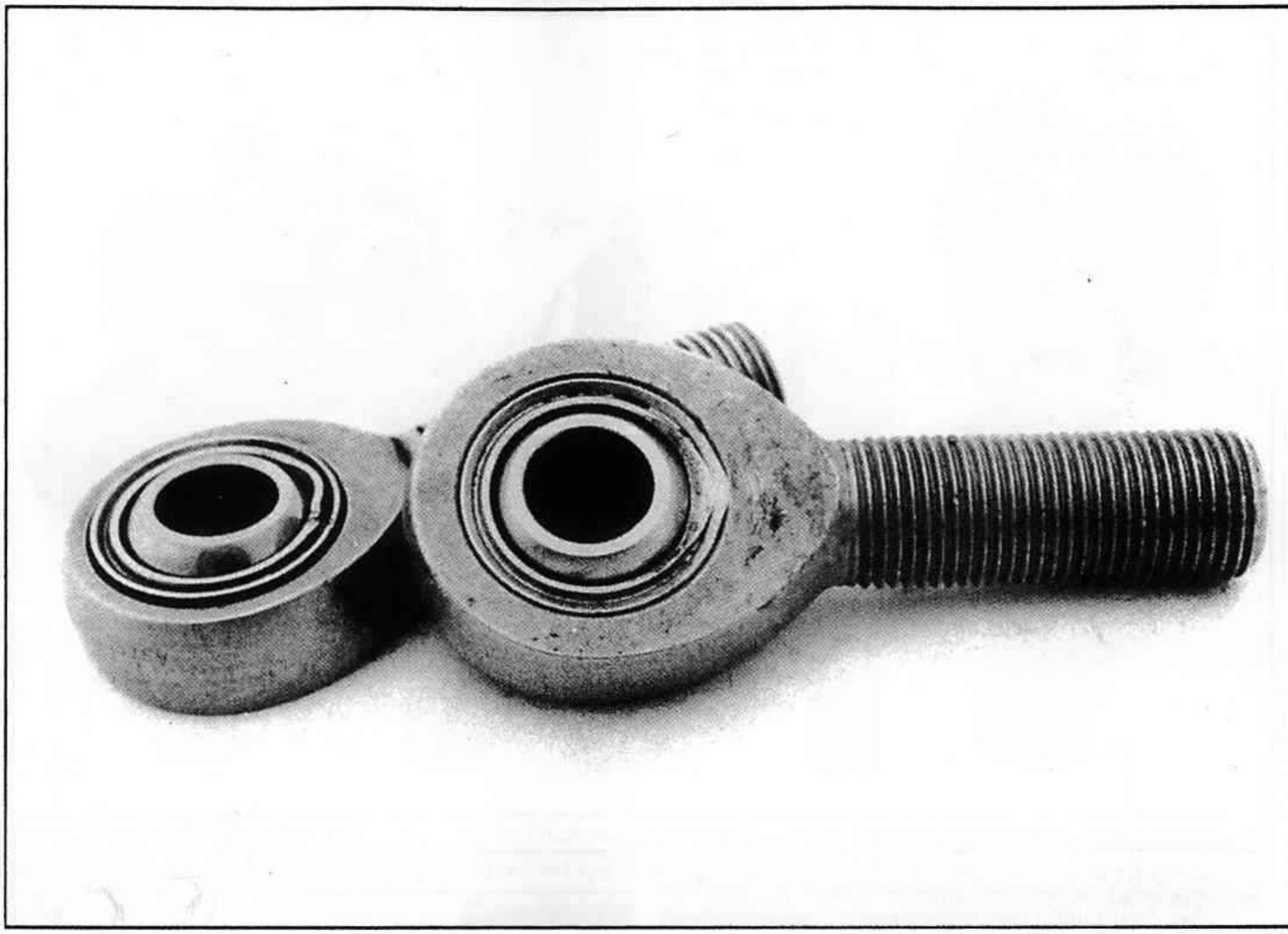
What are the advantages of oversize shank rod ends?

Rod ends are generally made two ways dimensionally. First is with a shank or threaded portion that is the same diameter as the hole in the ball, i.e. 1/2-inch bore x 1/2-inch shank. Second is with a shank one size bigger than the bore, i.e. 1/2-inch x 5/8-inch. The advantages are this: In an application where the unit is subjected to a bending load such as on the rear torsion bar arms on a Sprint Car, the larger shank gives more strength and reserve capacity. also, an oversize shank generally is made by putting an insert one size smaller in the body of the part with the larger shank. Therefore, a 1/2-inch x 5/8-inch part will have a higher load capacity than a 5/8-inch x 5/8-inch part made of the same materials. This is because there is more body material around the insert. This makes, for example, a 5/8-inch x 3/4-inch rod end a better choice than taking a 3/4-inch x 3/4-inch rod end and bushing the hole to 5/8-inch.

How can you tell a high quality joint from a low quality joint?

An obvious way to determine the quality of a joint is to inspect the machine work as you would on any other precision component. Is the race surface smooth or rough? Does the ball have a smooth or rough surface finish? on a non-teflon lined unit, does the ball have a precise fit in the body or is it loose and "rattly", or worse, does it bind up? On a teflon-lined unit, is the liner one continuous tightly bonded piece without gaps or are there gaps and areas where the liner is loose. The first characteristic in each sentence is expected in a quality bearing.

Features that are not so obvious on casual observation but that are found on high quality bearings included precision ground steel balls, use of high quality materials, and thorough

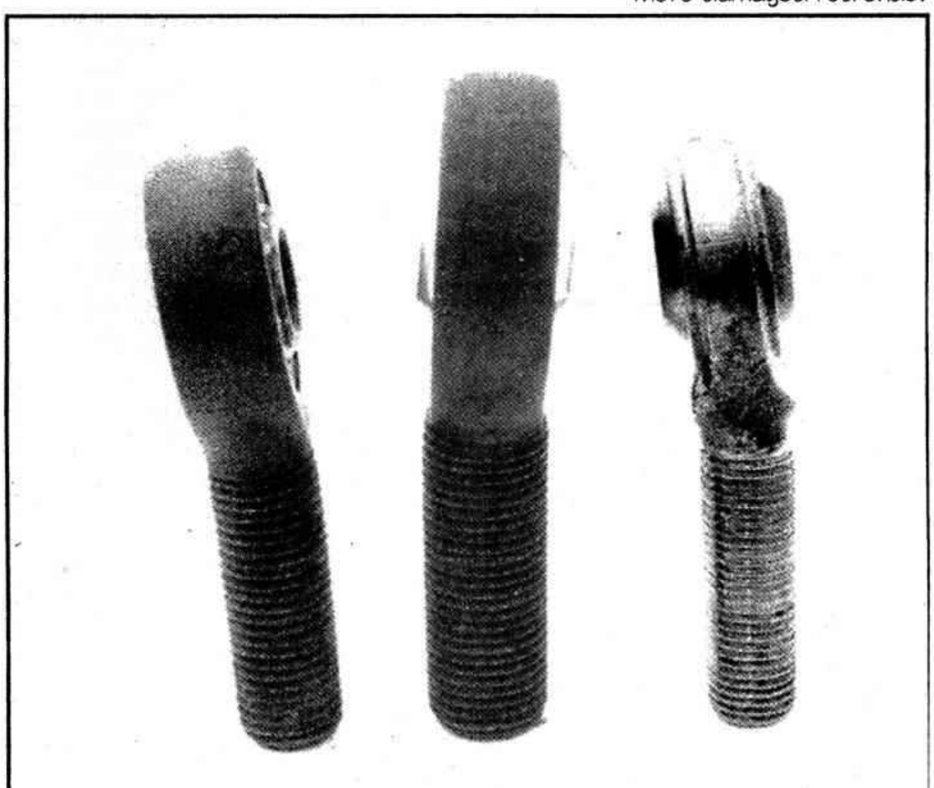


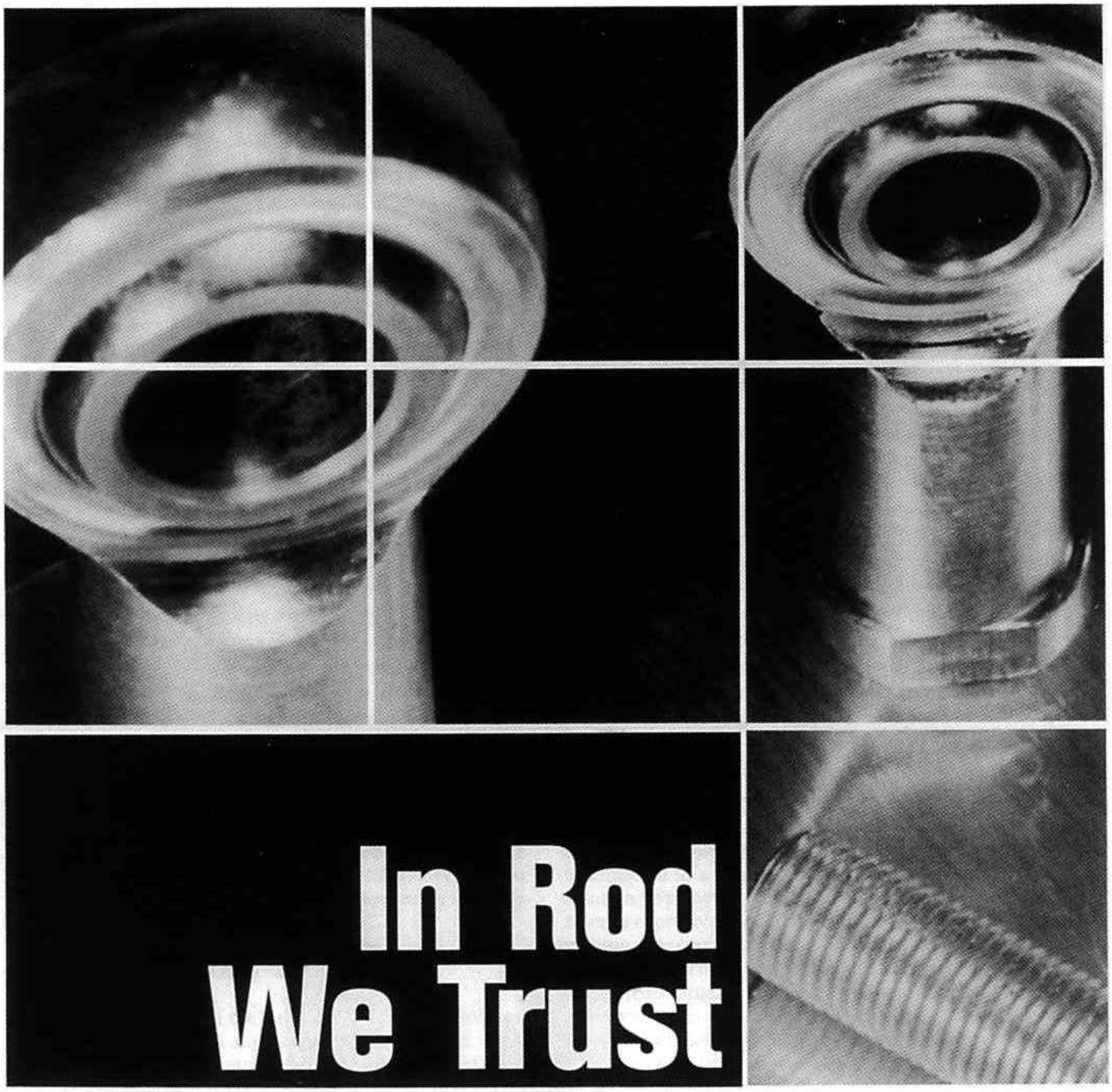
engineering, research and development. Generally though if you ask front running racers and car builders, you can get a good idea who makes a high quality joint and who makes an "almost as good" joint. And like any other race car component in the long run a high quality joint will be less expensive than the low quality type.

Hopefully you now know a little more about race car rod ends. If you have any further questions, call your rod end manufacturer. While it would be irresponsible for a manufacturer to recommend a specific type and grade for your application (that information should come from the car builder or designer), any manufacturer should be able to answer your technical questions regarding their product. By understanding a little more about these critical components, and choosing to use the correct joints in the first place they can become one less area of concern on your race car. ✖

NOTE: "Teflon" is a trade name of E.I. Dupont de Nemours & Co., Inc.

More damaged rod ends.





In Rod We Trust

We look at rod end bearings and ask just why it's so important not to overlook the part they play

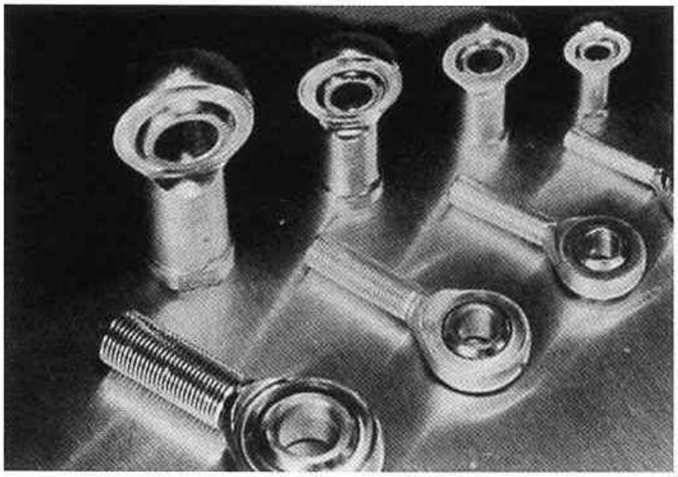
Words	Tim Beynon

Taking anything for granted in motorsport is a mistake, cut corners and you will, inevitably, be caught out in the end. You wouldn't risk the life of a driver by fitting a bargain basement harness, even if it looks as good as a top of the range one, so why then do so many 'professionals' seem to overlook, or underestimate, the importance of rod end bearings? The truth of the matter is that these vital components connect the suspension to the frame, or the wheels to the steering, and a failure in one could have catastrophic consequences. Can you really afford to cut this particular corner?

Rod ends – a definition

In short, rod ends are bearings. They are spherical plain bearings that are installed into female or male rod connectors, made up of a ball machined with a through hole that rotates within a race.

There are two basic categories of rod ends for racecars, commercial, or economy grade rod ends, and three-piece precision →



Care should be taken when choosing male or female rod ends; take a good look at the machining and shake to test for rattles – the fit should be snug

rod ends. The economy versions come in many different guises but as a rule you should only be using fully swaged two-piece rod ends if you go for the economy grade option. These two-piece units have good radial and axial strength as the body is swaged around the ball so that the race that the ball rides on is actually part of the steel body. Other types of economy grade rod ends that can be found in racing applications include those made with injection moulded plastic races, or with races made from one or two pieces of brass. These raceways have very low compressive strength relative to even the basic steel alloys, and hence poor durability.

The three-piece precision rod ends – or 'aircraft style' rod ends as they are sometimes called – are much more intricate. With these a race insert is separately formed around the ball and this insert is then staked into a body. This allows for a closer and more precise fit between ball and race.

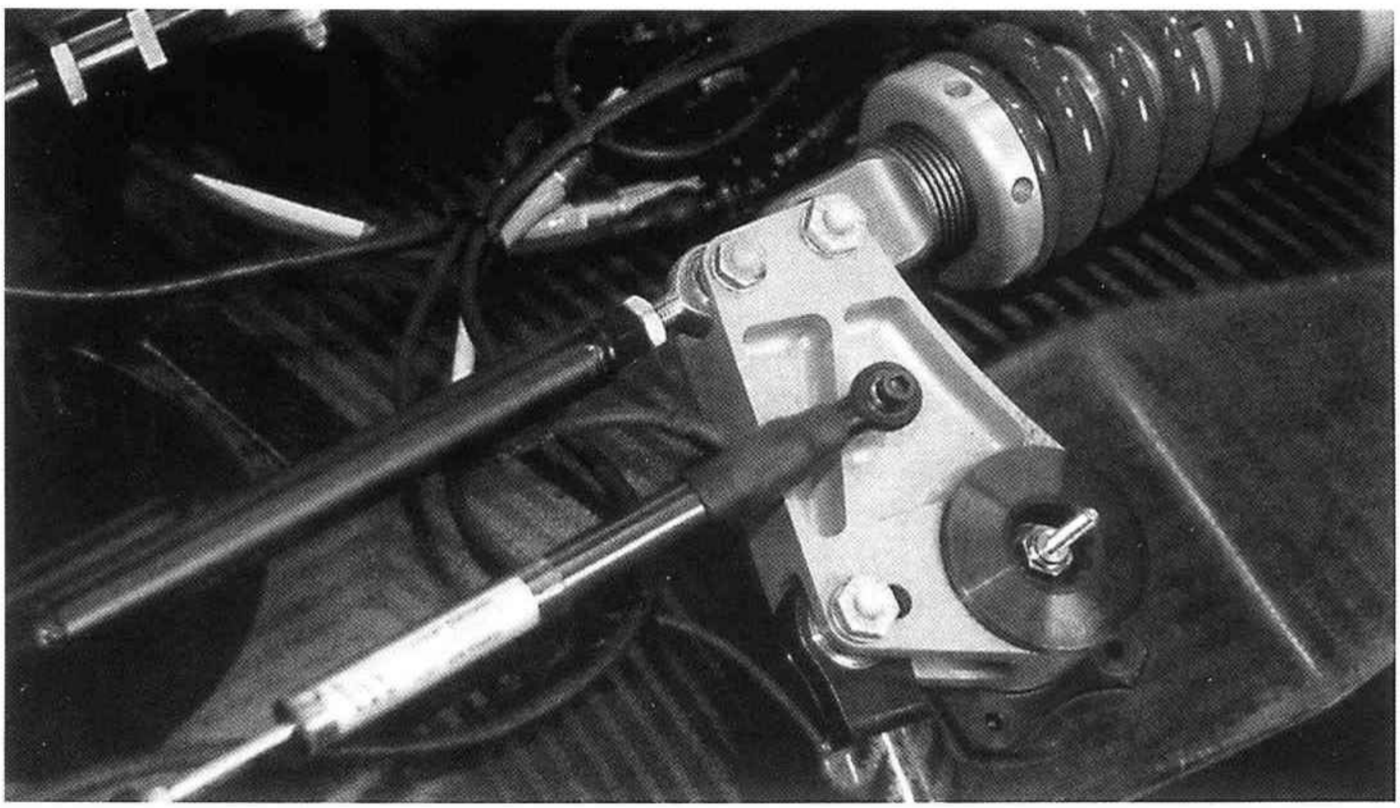
Different materials can be used in the construction of these precision bearings in order to best suit the particular application for which they are to be used. The races can be made from mild, alloy or stainless steel while the bodies can also be constructed from aluminium or even titanium.

Some rod ends come with a teflon lining which allows them to be self-lubricating, this is beneficial as oiling or greasing can increase the amount of wear due to dirt being attracted to the lubricant and working its way between the ball and the race. A teflon liner does not suffer from the same problem and so wears to a lesser extent, it consists of a fabric carrier component (for compressive strength), a teflon component for lubricity and various bonding resins. Lubrication is provided through the transfer of Teflon from the liner onto the ball as the ball moves in operation. Incidentally a good teflon liner will have a compressive strength of up to 60,000psi and should not be confused with lesser virgin teflon which can only withstand 10,000psi. Finally, the liner eliminates the clearance between the ball and race, giving a snug, tight bearing fit.

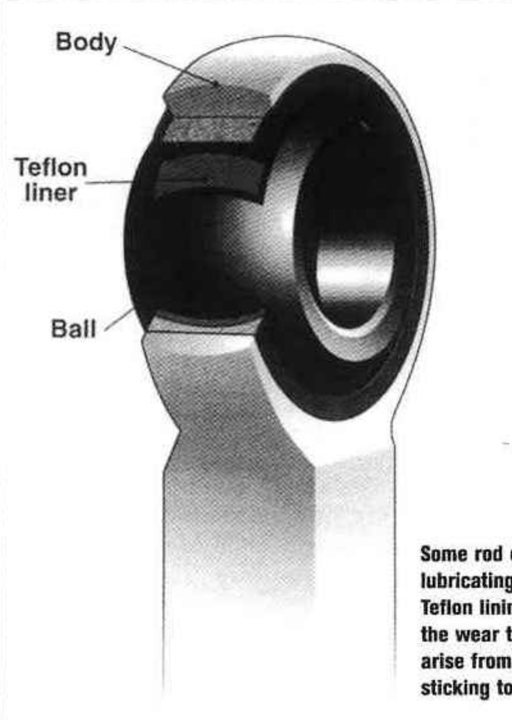
Choosing rod ends

Rod ends are limited to how much each can be misaligned before the sphere binds in the housing. It is therefore critical that the angle of misalignment is carefully considered when choosing rod ends as all are different and, if you were to choose any that exceed the recommended angles of misalignment, then they could wear down prematurely. It is therefore well worth measuring the angles on your car for which the rod end will be required and cross referring this figure with those quoted in a rod end catalogue. You may well find that the most expensive rod end is not actually the best one for your particular needs, in fact the opposite could be the case, it all depends on your requirements.

A further point to consider is whether or not the rod end you choose will be strong enough for your application. However, the whole issue of strength is not a straightforward one with rod ends as the different manufacturers have different methods of measuring strength in radial static load capacity and axial static load capacity. In general, the radial static load capacity listed in catalogues refers to a load that applied once will cause failure. Operating loads should have a factor of safety applied



Rod ends are used in a variety of applications and they are worked extremely hard, so if you are buying used be sure to check the second hand bearing against a new example and try to find out where it was used previously. That said, with components as vital as these it is best to buy new and buy the best



Some rod ends are self lubricating thanks to a Teflon lining. This stops the wear that tends to arise from dirt and debris sticking to the lubricant

to them relative to the catalogue load ratings. It is therefore important that you double check your own figures and those quoted by the manufacturer when making your choice. Note that, in certain instances, you might find the more expensive three-piece rod ends can actually be weaker than their two-piece equivalents.

The uncertainty surrounding the quoted static figures, however, makes it even more important to calculate the dynamic rating required in order to ensure that the rod end will withstand the conditions within which it will be operating. This figure is based on the PV value (the load in psi multiplied by the surface speed in fpm). It can be calculated using the following formula:

PV = WN/12L

This is where W = load in lb; N = rotational speed, RPM and L = bearing length, in

With the figure gained from running this equation and, depending on the material they are made from, rod ends can be separated into three ranges of PV values. Your calculated figure should therefore not exceed the following limits:

- Hard steel on bearing bronze: 50,000psi-fpm
- Hard steel on soft steel: 10,000 - 15,000psi-fpm
- Bronze or steel on filled nylon: 8000psi-fpm

It is worth noting that, as well as the PV limit, the strength of the oil lubricating film also limits the maximum dynamic load and the compressive strength of the bearing elements. In general, the maximum load on the projected bearing area of a metal-to-metal bearing should be no more than 4000psi. Finally, you should also remember to measure the pressure limit of the bearing.

Judging quality

Judging just how good or bad the quality of your rod end bearing is is a relatively simple task. As long as you ask yourself the right sort of questions as you are inspecting the rod end.

Firstly, take a close look at the machine work that has gone into the rod end. Look at and feel of the race, is it smooth or rough? Same again with the ball, smooth or rough? In both cases, the smoother the better. If you are looking at a non-Teflon lined rod end give it a shake to judge the clearance and the preciseness of the fit, does it rattle or does the bearing stick in the race? If it seems to fit snugly and precisely, without sticking, then it should be fine. If a Teflon liner is present look closely, check if the liner is whole or whether there are gaps and pieces missing from it? It should be one continuous, obvious, well fitting liner.

Don't forget to find out what the rod end and the ball bearings have been made of, if they were made from a high-grade precision material or not? Also, were they made by a reputable manufacturer that has invested in its engineering, research and development? As with anything in racing and life, you get what you pay for.

When things go wrong

Failures will happen and when they do many people refer to the rod end 'beating out.' By this they usually mean one of two things. Firstly they could be referring to when low strength 'self lubricating' liners get deformed, notably where the races are made of molded nylon or other plastic, possibly with Teflon added. These rod end races usually have a compressive strength of only 15,000psi making the race susceptible to deformation before any visible damage to the body can be seen. The same is true for brass race bearings.

Secondly 'beating out' could refer to when a proper liner has become detached from the race itself, thus causing a failure. The self-lubricating nature of the liner means that it may not bond strongly with the race. If this happens then, under high load, each time the bearing becomes misaligned, pieces of the liner will be ripped off until, eventually, there is little or nothing left of it. You are then left with a rod end with far too much clearance and it's just a matter of time before it fails.

However, not all rod ends with Teflon liners are destined to suffer this fate. The American government, among its various bearing requirements, sets specifications for bond strength and, if your rod end supplier, meets these mil specs then his/her Teflon rod ends should be fine.

As a rule a faulty or damaged rod end is a useless rod end, it is simply not worth, or indeed safe, to attempt to tighten up a worn rod end or patch up an accident damaged one. If you are unsure, after an accident, as to whether or not a rod end has been damaged always get it checked visually for any deformations, or indeed crack tested.

Buying used

On the whole it is not worth buying used rod ends because, when you do, you will have no idea how they have been used, or indeed for how long they have been used before you came along.

However, judging the amount of wear on a used bearing is certainly possible. One good place to start is by comparing your used rod end with a brand new rod end. Hold the shank of the joint and shake it around, if it rattles then the clearance is probably too great and it's worn out. One further tell-tale sign to look out for with Teflon lined rod ends is an absence of play either axially or radially remembering that a zero clearance fit is one of the benefits of the liner.

It is also, however, worth considering the fatigue life of a rod end. The body of a rod end can fail either due to a massive overload, such as in an accident, or due to natural metal fatigue which is altogether much more difficult to spot. A rod end may look fine but it may still be on the edge of a fatigue related failure, so it is vital that you find out how it has been used in the past and where on the car it had been fitted.

The latter point here is important because rod ends can be loaded in a variety of different ways, all of which have differing effects on its fatigue life. On a suspension push rod, or pull rod, they can be loaded in straight tension or compression, for example. While on a tie or radius rod they could have a straight-line reversing load. When used as a ball joint they can have bending loads applied to the shank or, indeed, a rod end can be subjected to a number of different loads in different directions. So be sure you know just how the bearing was used.

ROD END INSPECTION

A race car's suspension can easily feature a few dozen rod ends, and these joints allow nearly unhindered motion. Over time, however, each one can become a ticking time bomb. Rod ends don't last forever, but knowing how to inspect them can keep a car on track and off the hook. John McCrory is with Aurora Bearing Company, one of the largest rod end suppliers in the free world, and he has some simple advice for racers and crewmembers.

Refurbishing an older race car really requires you to question and evaluate every component. Are parts worn, and how can you tell? Even if various parts do appear to be good, should they be replaced by newer, better-performing parts anyway? Rod ends are one of the many items that can cause headaches for the race car refurbisher.

Evaluation: When evaluating the rod ends on a race car's suspension, the first step is to eliminate joints that don't belong on a racing suspension in the first place.

Joints with brass, bronze, or plastic races should be eliminated right away. Races made of these materials have relatively low compressive strengths and a low tolerance for shock or vibratory loads—they tend to loosen up quite drastically under hard use. These races are sometimes acceptable for secondary linkage applications, but they really aren't up to the demands of a race car suspension.

Two other items to remove immediately: rod ends with grease fittings and rod ends with hollow shanks. While both features allow the rod end to be relubricated, strength is compromised.

Condition: Once you determine that a rod end can be reused, its condition must be examined. Start with the overall condition of the joint.

Is the body bent? Are there signs of stretching on either the shank or the head? Are there marks that indicate the part has ground against something else (maybe the track)? Does the outer face of the race have dents that indicate over-misalignment? Is the race loose in the body or partially pushed out of the body? These are some indications that the joint has been abused, possibly in an accident, and should be replaced. If the joint shows none of these signs of abuse, magnetic particle inspection—like Magnaflux testing—should be done to ensure that the piece is truly free of cracks.

The next thing to do is evaluate the amount of wear on the bearing portion of the joint. Any play in these joints will be more noticeable when the car is together, so shake each corner of the car and try to note any play relative to a joint and its mounting bolt. Touching a finger to both the joint and an adjacent surface should help you detect any relative movement.

Wear can still be evaluated with the parts off the car. On a unit lined with a nonstick material like Teflon (DuPont's brand name for PTFE), low breakaway torque (the force required to move the ball) is not necessarily a sign of a worn-out joint, although it can be. This contradictory statement is rooted in the fact that different bearings manufacturers use different PTFE liner designs. These designs each have different performance characteristics.

One brand of bearing may start out with a very tight fit, then gradually loosen up until it reaches a zero-torque fit and wears out. Others may start out tight, fall off quickly, then maintain a light fit for a long period of time. In either case, the important thing to look for is an absence of play either axially (side to side) or radially (along the direction of the shank).

Judging the wear on a metal-on-metal joint is a little more difficult, as all metal-on-metal joints start life with a small amount of clearance. Comparing your used joints to a new one is a good place to start. An unscientific method is to hold the rod end's shank and give it a good shake. If it rattles, it's worn out.

Looking at it from another perspective, if you car's suspension has metal-on-metal joints, why not replace them with PTFE-lined joints? Remember, lined joints are more precise because of their zero-clearance fit. They're also maintenance free.

One Last Inspection: Before reinstalling an old joint, there's one last factor to consider: the fatigue life of the rod end.

The body of a rod end usually fails for one of two reasons. The first is severe overload. Make a hard enough impact with a curb, wall or another car, and you can overstress a joint to the point of breaking. During your inspection, you may find a joint that's been stressed to, or close to, the breaking point.

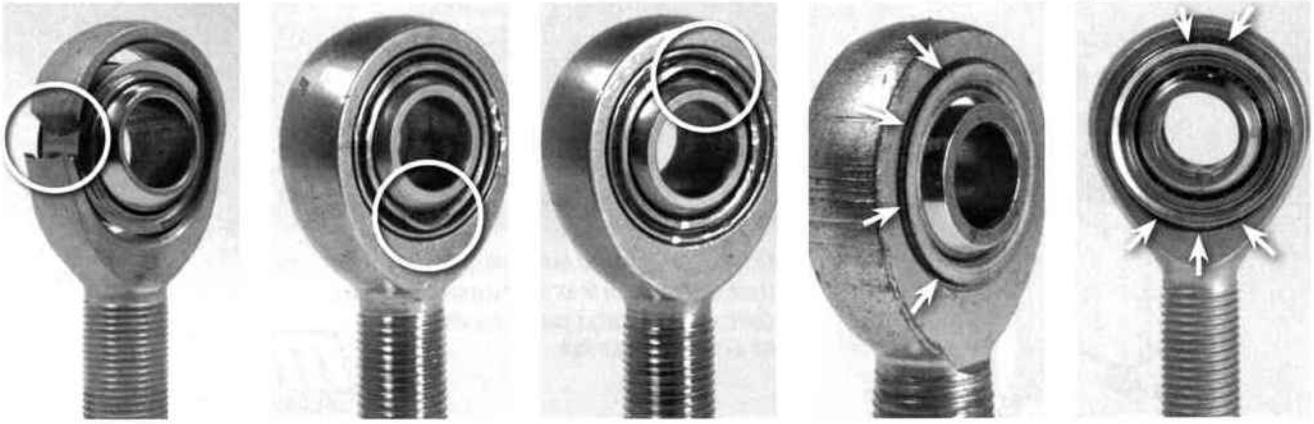
The second reason for failure is fatigue. Like any other metal component, rod ends are subject to wear. Unfortunately, there's no way to tell if one is too fatigued to pass inspection.

Rod ends can be loaded in many different ways, and each use affects the fatigue life of the joint. They can be loaded in straight tension or compression along the direction of the shank, as on a suspension pull rod or push rod. They can have a straight-line reversing load, as on a tie rod or radius rod. They can also have bending loads applied to the shank, as when a rod end is used as a ball joint. Or there can be a combination of loads in various directions.

So what do you do? The safest thing is to automatically replace any old joints with appropriate new joints. The most realistic strategy is to evaluate the physical condition of each joint, consider the stress it endures in its particular application, and figure out what your budget will allow. That should help you draw up a plan of attack.

Common Rod End Failure Modes

photos courtesy aurora bearing



Broken | Misalignment | Misalignment | Pushout | Stretched

More Online

Look for current project updates and race results at grassrootsmotorsports.com.

LEFT: During a wintertime work session, we found that most of our car's suspension joints were rod ends of various ages and conditions. We junked them all and ordered replacements. RIGHT: While our suspension was disassembled, we repainted the trailing links and control arms.

ABOVE: We replaced our old rod ends with brand-new hardware sourced from Aurora. BELOW: The LeGrand also uses a pair of rod ends to support the steering rack. In this case, we used lightweight aluminum pieces from Aurora. We also took a look at our gauges—or, rather, we tried to take a look at them, but they were too hard to read. A new Stack ST 1800 display unit was the solution. It's a customizable setup, and we specced our tachometer with a 15,000 rpm rev range. Our old muffler was too loud, so we swapped it with a new two-stage unit from Burns. This should keep us from raising the ire of the sound police.

Grassroots Motorsports
THE HARDCORE SPORTS CAR MAGAZINE

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Grassroots Motorsports is more than just a magazine, it is a true multimedia publisher producing print magazines, digital and email content, as well as events to bring motorsport to your door.

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From: http://grassrootsmotorsports.com/project-cars/1976-legrand-mk-18/
LeGrand Mk 18: Replacing Rod Ends Oct 27, 2010

After we had to replace a rod end at the Solo National Championships, we took a good long look at all of the rod ends in our LeGrand's suspension. We decided to replace every single one; that was the only way we could ensure we'd have a solid foundation going into our 2011 racing season.

We chose an old favorite of ours, Aurora Bearing Company's alloy-steel PTFE-lined rod ends, for most of the suspension joints. The car is mainly held together by the 3/8-inch versions, but ther are several larger and smaller sizes. We counted 22 joints in the rear suspension alone.

One we have the car back together, we'll need to do a full alignment and "squaring" of the car.

The old rod ends wer worn, and we could feel that several had some play in them. Not good for a suspension!

“CRITICAL COMPONENTS”

“What You Need To Know About Rod Ends & Spherical Bearings”

Wayne Scraba

Photos: Wayne Scraba

Peer inside a modern racecar and you’ll find all sorts of different rod ends and/or spherical bearings. Rod ends are critical components in any racecar and they also prove be rather practical problem solvers in plenty of hot rod and custom car applications too. Rod ends and spherical bearings can be used in any number of locations aside from common suspension and steering components (case-in point: shifter linkage, carb linkage, mechanical brake linkage and so on). Most often though, the suspension and steering systems are where you’ll find rod ends as well as spherical bearings. Here, they’re regularly charged with handling critical loads (for example, a rear suspension link). If a rod end in such a location breaks, then the car gets out of control. It’s that simple.

At first glance, all rods ends look alike. And that’s not good news. Until you dig a wee bit, you can easily pick up a cheap knockoff built in an off-shore sweat shop instead of a high quality aircraft spec job built in America. The reality is, that cheap off-shore built rod end is most likely junk and you shouldn’t put your neck on the line using one.

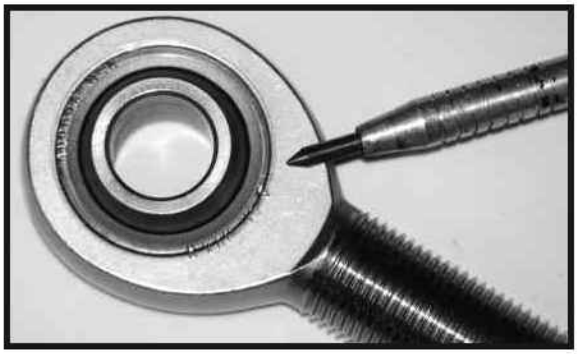
Knowledge is the key. Basically, a rod end consists of a spherical ball, which is engineered to rotate inside a housing. This ball is the bearing and the housing it’s contained in is the race. Each side of the spherical ball is machined flat. The modified “sphere” has a hole bored through the center.

MONEY TALKS...

When purchasing rod ends for your project, you’ll regularly find “economy” or “commercial” configurations. While there are plenty of various economy rod ends on the market (that’s where most of the off-shore imports live and play), the only type you should even begin to think about for a high performance application are two-piece, fully swaged models. In the two-piece configurations, the body is formed (or “swaged”) around the ball so that the race the ball rides on is actually part of the body. When considering inexpensive rod ends, this is the only less costly style that offers reasonable pull strength (radial strength) along with adequate axial strength. In case you’re wondering, axial strength is the resistance of the ball being forced out of the side of the body.

This is a three-piece (“aircraft”) rod end. With this configuration, the race is formed around the ball, and then the race insert is staked into the body.

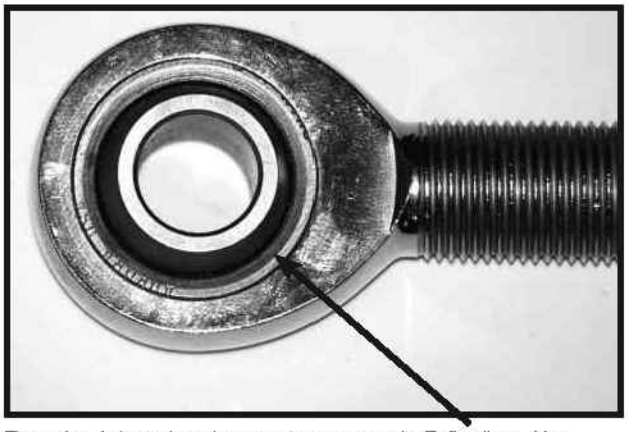
But we're certainly not done: When it comes to rod ends, you definitely get what you pay for. Better quality rod ends are most often based upon a precision three-piece configuration. This three-piece design is regularly referred to as an "aircraft" rod end. Here the race is formed around the ball. Next the race insert is staked into the rod end body. How important is this? Very. With this layout, the result is a much closer component fit (there's much more precision between the ball and the race). The three-piece design allows specific materials to be included while the rod end is manufactured. As a result, the manufacturer can now build a given rod end to match a given application. Additionally, the rod end race can be manufactured from mild, alloy or stainless steel, with bodies manufactured from mild, alloy or stainless steels, aluminum or even titanium. You might find rod ends with races built from brass or aluminum bronze, but due to their low strength, it's a good idea to avoid them at all costs.



Because of the three-piece configuration, various materials can be used in the construction of the rod end. This process allows the rod end to best match the application (which often boils down to strength versus load).

LUBRICATION MAKES A DIFFERENCE...

The Teflon liner option is something you'll soon discover when researching rod ends or spherical bearings. It's important because the liner allows the rod end to be self-lubricating. Just how critical is this? Give this some consideration: In almost any motor vehicle (car, truck, bike, airplane, boat) there are plenty of places where you simply cannot lubricate a rod end or places where you just don't want to. Yes, there are certain rod ends out there that are built with integral grease fittings, but you have to keep in mind that grease fittings will physically weaken the rod end (in essence adding a grease fitting mandates drilling a hole right through a critical part of the rod end - and as you can well imagine, that's not good). One more concern is the problem of dirt being attracted to the grease. Grit eventually finds its way in between the ball and the race and then wear escalates - sometimes rapidly.



The rod end shown here incorporates a composite Teflon liner. You can purchase two or three piece rod ends with Teflon liners. The Teflon liner is bonded to the race so that the ball actually rides on the liner. The movement of the ball rubs Teflon on the ball, which in turn provides the necessary lubrication.

The use of a Teflon (a trade name of DuPont) liner eliminates most, if not all of the issues associated with grit and premature wear. Teflon liners are made up with a carrier component, most often a fabric that provides compressive strength, a Teflon component providing lubricity as well as a collection of bonding resins. The Teflon liner and the race are bonded together. Because of this, the ball actually rides on the liner. As the ball moves, Teflon is rubbed on it. That's where the lubrication comes from. Two or three piece rod ends are commonly available with Teflon liners. When talking Teflon, consider that if virgin Teflon was used, you'd find the material proves to be relatively soft (approximately 10,000 pounds PSI compressive strength). On the other hand, a high quality composite Teflon liner (where "components" are added to the Teflon mix in order to increase strength) will have a compressive strength of somewhere between 40,000 and 60,000 PSI. A rod end with a quality Teflon liner will have a tighter fit. That's because a good Teflon liner eliminates clearance between the ball and race.

Of course, the simple addition of a Teflon liner to a rod end doesn't insure precision. It is certainly no guarantee of performance capability either. A big consideration is the term "beating out". What's that? Sounds simple enough, but it's actually two separate maladies. The first one is the deformation of low strength "self lubricating" liners. Many cheap economy rod ends are built with races constructed from molded plastic (and in some cases, the plastic is mixed with a fiberglass filler). To provide some lubrication, a bit of Teflon might be added. Given the mix of ingredients, these easy-on-the-pocket rod ends typically have a compressive strength of no more than 15,000 PSI. Given the poor compression strength, the race will deform long before the body sees any damage.

Another form of "beating out" failure involves the physical bond between the liner and the race. Since the liners are built with a self-lubricating material, it's sometimes tough for inexperienced (or poorly equipped) rod end manufacturers to bond the liner to the race. If this bond isn't strong enough, then the liner can become detached from the race. The problem is compounded if there is a mis-alignment of the rod end (more on alignment later). Bits of the poorly bonded liner tend to disintegrate. When this happens, internal clearances increase, eventually becoming excessive. Aside from MIL Specs for Teflon-lined bearings, there are no standards set for liner bond strength.

MATERIALS MATTER...

Teflon isn't the only material you have to concern yourself with. The base materials used in the construction of the rod end play very critical roles too. Earlier, we mentioned some of the materials rod ends can be built with. The actual spheres or balls are most often subject to the highest loads the rod end sees. Because of the high loading, the balls mandate the greatest hardness along with the greatest ultimate strength. Certain commercial rod end balls can be manufactured from bronze or even sintered steel materials. For the most part these materials aren't the greatest when it comes to strength, however you might find some sintered steels are fully up to the task. Provided a proper heat treat, sintered steels can be made to work in a light to medium duty rod end ball application. For the most part though, quality rod ends are manufactured with heat-treated steel balls (including balls made from stainless, chrome moly and 52100 bearing steels). The actual balls must be extremely hard in order to remain round (these balls are often chrome plated to provide a smooth bearing surface). The hardness of the ball coupled with the capability to remain round is absolutely critical in use.

What do you look for in a race? It too must be hard, but not to the level of the ball. The majority of three-piece rod ends incorporate a race manufactured from through-hardened steel alloy or from a stainless steel that can be hardened. In either case, the outer races are heat treated for both strength and wear resistance.

The bodies of economy or commercial rod ends regularly have bodies manufactured from low carbon mild steels. It is not possible to through-harden this material. Although a less costly material such as low carbon steel might work in a lightly loaded application, you'll find that a rod end body built from chrome moly steel or heat treated stainless steel is much more satisfactory for severe duty applications. There's something else to ponder too: When a rod end is built with a chrome moly or stainless body, then the size of the rod end can actually be reduced. The reason is (obviously) due to the fact the material it's made out of is significantly stronger. Some rod end bodies are also manufactured from 7075-T6 aluminum. If you do a bit of homework on materials, you'll find that 7075-T6 aluminum proves to be one of the strongest grades of aluminum available. It actually has a tensile strength slightly greater than mild steel. The truth is, if you compare the strength of two similar rod ends - one manufactured from 7075-T6 and the other from mild steel, you'll find they're similar. The trouble is, aluminum won't stretch or bend as much as mild steel before it breaks or bends. Factor a good quality heat-treated chrome moly or stainless rod end into the comparison and you'll soon see that the expensive rod ends are almost twice as strong as the aluminum counterparts. It's very difficult to beat a high quality heat-treated steel body rod end when it comes to ultimate material strength.

MULTIPLE DIMENSIONS...

Dimensionally, a manufacturer can build a specific rod end two different ways. In one, the shank (the threaded part) is built with a diameter that matches the hole in the sphere. As an example, a rod end with a 5/8-inch bore will have a shank with 5/8-inch threads. The other format has a shank diameter one [fractional] size larger than the bore. In this case, an example might be a rod end with a 5/8-inch bore coupled to a 3/4-inch shank. The big shank, small-bore rod end is stronger in applications where bending loads are (or could be) present. A good example is a trailing arm arrangement used on a racecar four link. Here, we have tubular bars acting as levers, transmitting considerable forces, and in turn often accepting equally considerable forces. In this type of application, a larger shank rod end design provides more strength along with a sizable amount of reserve strength capacity. Keep in mind, however, that some smaller size push-pull rod applications mandate the use of female, not male rod ends.

The rod end on the left is a standard configuration model, while the rod end on the right is a heavy-duty model. The special HD model has a 1/2-inch bore while the standard version has a 3/4-inch bore. More in the text.

A rod end with an oversize shank is generally made by installing an insert one size smaller in the body of the part with the larger shank. In some extreme race car applications (drag race 4-link specials), the body is actually two sizes smaller than the shank. Because of this, (again using the big rod ends used for suspension pieces as examples) a 5/8-inch X 3/4-inch rod end exhibits higher load capability than a similar 3/4-inch X 3/4-inch rod end. Of course the caveat would be that each of these rod ends are manufactured from similar materials. The reason for this is because the 5/8-inch X ¾-inch rod end has more body material around the insert. Another bonus is the fact asymmetrical rod ends such as this provide superior wrench access in many applications. The reason for this is simple: It's due to the fact the fastener that passes through the ball bore is smaller.

This a close up look at a heavy-duty rod end. The oversize rod end is generally made by installing an insert one size smaller in the body of the part with the larger shank.

There is one exception to the above though: Some companies offer a rod end where a larger shank is added to a smaller body. Although on the surface this practice seems to serve the same purpose as the oversize shank rod end, it definitely provides less meat around the rod end ball, and that's something to ponder.

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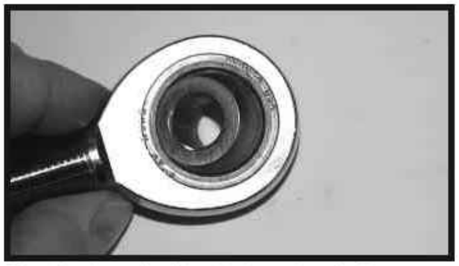
MISALIGNMENT PROPERTIES...

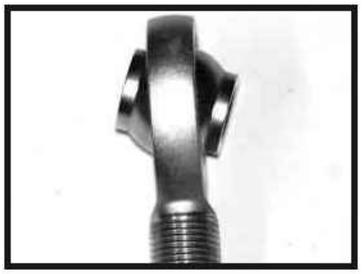
You'll often hear the term "misalignment" when folks discuss rod ends. What's that? Recall when the basics of rod end design (a modified ball or "sphere" inside a race) were detailed? In order to mount the rod end to something, then a fastener of some sort (most often it's a bolt) passes through a hole bored in the center of the sphere. With the bolt in place, then there's no way the sphere can rotate a full 360 degrees. Because of this, all rod ends have specific limitations regarding how far they can be misaligned before the sphere binds in the housing. The angle of misalignment is very important when choosing rod ends for specific applications. Not all rod ends can accept the same degree of misalignment. Most manufacturers publish a maximum recommended angle for a given rod end (specs are usually in the manufacturer's catalog).



Examine the rod end in this photo closely. See how the ball isn't centered? This is essentially the "misalignment" that the rod end design can handle.

If you exceed it, you'll get anything from premature rod end wear to outright rod end failure.

So far so good, but how do you determine what the angle of misalignment really is? It's not rocket science: Simply use a conventional protractor to check the geometry. Compare the measured angles you get with a protractor to the manufacturer's specifications. By the way, buying a bigger rod end to make up for misalignment won't help. Fixing the misalignment or using high misalignment rod ends is the answer (the accompanying photos show some high misalignment rod end options).



Misalignment is the degree of angular movement that a ball or sphere can accommodate without interference. This is a special "high misalignment" bearing (note the larger shoulder).

APPROPRIATE ORIENTATION...

When a pair of rod ends is used in a single component (an example is a single four link bar), the orientation of the rod ends on either end is rather important. This is most often referred to as "clocking". But before examining clocking, we should point out that even very small adjustments in any suspension link that sees pre-load could make a large difference in the way the car handles. In some cars, one-sixth of a turn at a time is sufficient to see a change in the behavior of the chassis. Because of this, it's a good idea to use the "flats" (flat sides) of the jam nuts as a reference point for adjustment.

This is how the small adjustment process works: One side of something like a suspension link is equipped with right hand threads while the other end of the link is fitted with left hand threads. If the jam nuts are loosened, then can lengthen or shorten the entire link, often by simply turning it. Essentially, this works like a factory tie rod adjustment.

So far so good, but where does the "clocking" come into play? Simple. When the rod ends are properly "clocked", that means they're physically aligned. This prevents binding of the suspension, and makes it easy to determine if the link is under tension by "feel". If you grasp the link by hand, and rotate it back and forth, you can tell if the link is "neutral" or under strain.

LIFE CYCLES...

Hands up! Have you ever attended a big swap meet? Odds are pretty good you'll eventually come across a box or two of "lightly used" rod ends. And the price will be right. Bargain! Maybe not. You see, just like any other piece of hardware, a rod end actually has a finite mechanical life. There is no way of knowing if a used or surplus rod end has reached the end of its life cycle. Another important consideration is that there is no safe way to repair or "tighten" a worn bearing. You simply cannot peen them to make them "tight". Additionally, any rod end that demonstrates any amount of stretching in the threads or in the head should be discarded. The same applies to any rod end bearing that has been dented in the race area or is bent. Rod ends such as these have definitely met their end.

Honestly, we're back to square one: At first glance it's nearly impossible to tell a junk rod end from a high quality piece. The only thing you can do is to examine each rod end in the same manner, as you'd look at any precision piece of hardware. Examine the machining. Examine the race. Examine the ball (sphere). In each instance, they should prove well machined and smooth. If the rod end is a non-Teflon configuration, does the ball fit inside the body precisely? Rotate the ball. Is there any bind or is it so loose that it rattles? Examine Teflon liners carefully. Look for areas where the liner is loose (the liner should be one continuous tightly bonded piece). Examine the threads. They should be high quality, rolled threads, just like a good bolt or capscrew. The quality of the rod end is right there in front of you. Use it to help you decide what's hot and what's not.

In the end, we're sure you'll discover (hopefully sooner than later) that good quality rod ends cost good money. Quality rod ends are those that are designed and built by way of proper engineering and then backed by rigorous research, development and testing programs. Remember you're buying a precision mechanical component, and it happens to be a piece you have to put your trust and safety into. These are definitely critical components.

Spherical bearings are actually close relatives of rod ends. They have countless uses in cars. They're often used on custom suspension components - particularly when composite materials are used for construction. Spherical bearings are available in a wide array of bore sizes, usually ranging from 0.1900 to 1.00-inch. In the case of quality components, they're manufactured under the same quality as military approved bearings.

ROD ENDS

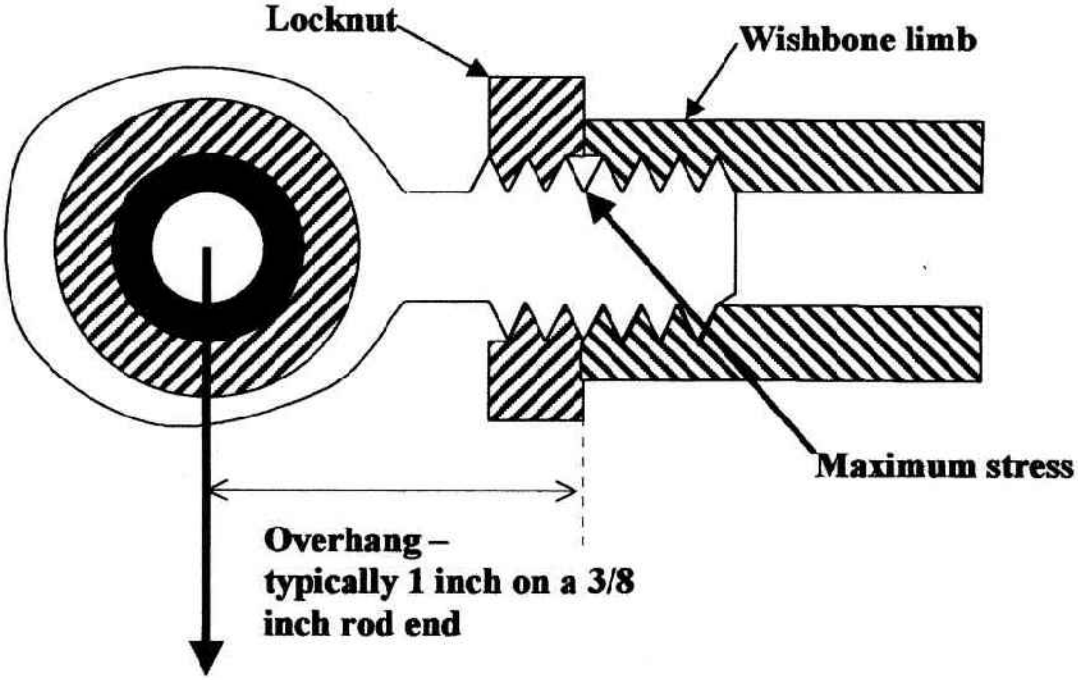


Figure 1 - Bending loads on a rod end

Each weekend in July, during the brief English summer, I went to a motor sport event of one sort or another, writes Mike McDermott. These ranged from a local speed hill-climb to the Grand Prix at Silverstone. With one exception, guess which, at each event I saw that depressingly common motor-racing happening, a failed rod-end, with a damaging trip to the barriers the inevitable result.

Why do rod-ends fail so often on racing cars? And why do two companies, Rose in England and Heim in the USA, always seem to be blamed by the embittered driver when interviewed by the commentator? In the cases I saw, the joint certainly had failed - I went to the paddock and had a look, just to make sure.

Since rod-ends, including those manufactured by the two fore-mentioned companies, were initially developed for aviation, where they rarely fail, the unfortunate drivers are undoubtedly blaming the wrong people. Properly used, rod-ends, including those ubiquitous 'Rose joint' or 'Heim joint', are among the most reliable components ever fitted to a racing car.

Talking to engineers specialising in rod-ends, it is clear that they never cease to try to educate racing-car designers to use their products more effectively, but frequently find their advice scorned. Worse still, they see initially excellent designs being progressively degraded as successive generations of engineers get their hands on the original design and find ways to 'improve' it.

The problem lies with the designer and his reluctance to understand or to embrace the theory of bending. So, at the risk of insulting the intelligence of some readers, I'd like to take some time to go through the basics of this tedious but crucial subject.

A typical rod end installation is shown in Figure 1. The manufacturer will rate a typical medium-specification version of a 3/8 x 3/8 inch joint at 4000 lbf radial load - the rod end industry retains very strong ties to the familiar pre-metric dimensions. The cross-sectional area of the thread core, based on a 5/16 inch thread root diameter, is 0.077 square inches. At 4000 lbf load, the tensile stress is 4000/0.077 = 52200 psi = 23 Tsi (tons force per square inch - Imperial tons, that is, equal to 2240 lbf). A medium high-tensile steel with an ultimate tensile strength (UTS) of 40 Tsi will often be tempered to have a yield stress of about 30 Tsi so, in the direction of the threaded shank, the thread is comfortably capable of supporting more than the manufacturer's rated load for the joint. The story is completely different when side loadings are brought into the picture.

In Figure 1, the side load is F and the overhang from its bush is D. A nominal 3/8 inch rod-end can easily have a one inch overhang. This produces a bending moment at the point where it enters the bush of M = F x D.

In bending, the important characteristic is Ixx, the second moment of area about the major diameter, where

$$I_{xx} = \pi \times \text{diameter}^4 / 64.$$

The maximum stress S resulting from the bending moment occurs at the root diameter of the thread, and a standard undergraduate mechanical engineering textbook such as Joseph Shigley's "Mechanical Engineering Design" (McGraw-Hill, 1986, ISBN 0 07 100292 8) will confirm that

$$S = M \times \text{diameter} / 2\, I_{xx}$$

or $$M = S \times 2\, I_{xx} / \text{diameter}$$

For a 3/8 inch round joint shank with 5/16 inch thread root diameter,

$I_{xx} = 0.000468$ inch^4

so the maximum stress S will equal the 30 Tsi yield stress of this medium strength steel when

$$M = 30 \times 2240 \times 0.000468 \times 2 / 0.3125 = 201 \text{ lbf.in}$$

Since the overhang, D, is estimated at 1 inch, this means that a side force F of about 200 lbf on this joint is capable of bending the joint permanently, and 300 lbf generates a stress larger than the UTS, easily breaking the shank at the point where it enters the bush.

Looks familiar? This is a far cry from the typical 4000 lbf joint capacity, comfortably supported axially by the shank of the joint. Going to a 1/2 inch joint raises the critical side load to 555 lbf - better, but still well below the joint's even greater rated capacity. Even the most expensive joints have housings made from steels no stronger than 55 Tsi UTS, so there's no point hoping for salvation that way.

If any doubt remains, then just think about the forces arising in a typical suspension component such as the apex of a wishbone. Figure 2 shows the forces developed by a braking wheel. On a small formula car, typical values are A = 5 inch, B = 12 inch. Decelerating a typical 300 lb (per corner) mass at 1.5 g generates 450 lbf so the lower wishbone sees 12 x 450 / 7 = 771 lbf, and the upper sees 321 lbf. Under normal racing braking, the shank of a typical 3/8 inch joint is overloaded in either location, and even a 1/2 inch joint is overloaded in the lower wishbone. I have lost count of the number of wishbones I have seen incorporating 3/8 inch joints. There are even some optimists who not content with bending

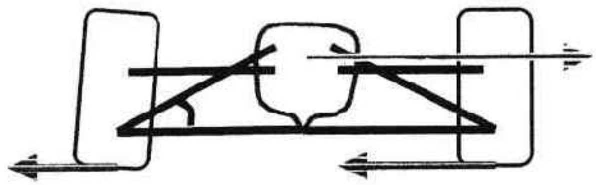
TO PUSH OR TO PULL

The forces displayed in Figure 2 raise an interesting point. The force on the joint at apex of the lower wishbone is always greater than the force generated by the tyre. This is true for longitudinal (braking) forces and lateral (cornering) forces alike. Given the relative powers of engines and brakes, the lower front ball joint usually has to handle the highest loads - in hard braking in a corner at high speed, when downforce allows maximum braking forces - of all the joints on a race car.

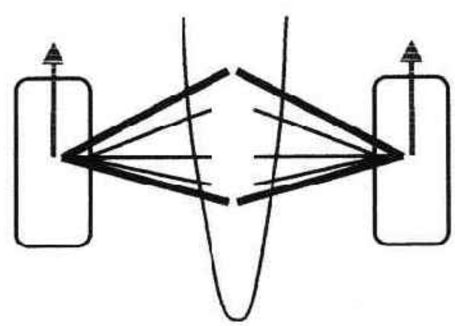
With outboard spring/dampers, or with push-rods, this joint also has to support the weight of the car and any downforce, together with any load transfer. So it has to cope with major loads in all three directions. Proper design of the wishbone, by minimising bending loads, can maximise the load capacity in two of the three directions. But every joint has a much reduced capacity 'out-of-plane', where the load is pushing the ball out of its race. Aurora recommends that this is never more than 15% of the joint's radial capacity, even for the best 2-piece joints; for the popular 3-piece joint, the limit is 10%.

Observers have suggested that the push-rod in a Formula One suspension is an indispensable supporter of the limbs of wishbone under these heavy loads. Given the relatively shallow angle the push-rod makes with the horizontal, this is believable for lateral loads - even if the angle is 30 degrees, cos 30° is 0.87 - see Figure A. On the other hand, it is scarcely credible for the even greater braking loads because of the angle, more-or-less a right angle, that the push-rod makes with the car's lengthwise axis, and cos 90° = 0, as in Figure B.

In Formula One, the currently universal high nose, with its under-slung wing, makes push-rods virtually inevitable, with the need to compromise the lower joint in one of the directions, because there is no space, low down, to put the spring/dampers. But in other series, where the high nose is outlawed or irrelevant, as in CART or in most sports/GT cars except the Toyota GT-One using major internal air flows, a pull-rod transmitting the vertical loads, in tension, from the top wishbone to a low-placed spring/damper is far more elegant. Besides the smaller loads the top joint has to handle, so that its size can be reduced, the lowest possible location of the spring/dampers is possible. The convenience of the high-mounted damper is lost, but much of this has already gone, with many dampers already buried deep in tubs, albeit sometimes with remote valve blocks.



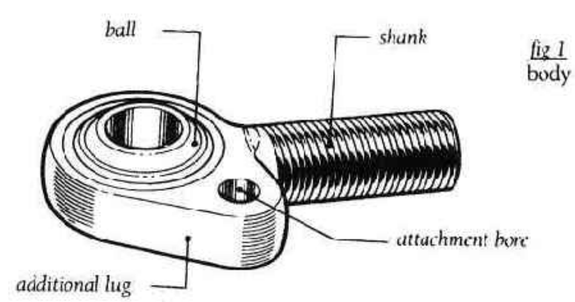
A **Small angle between the pushrod and the horizontal allows wishbones AND pushrods to transmit lateral (cornering) forces from tyres to chassis**



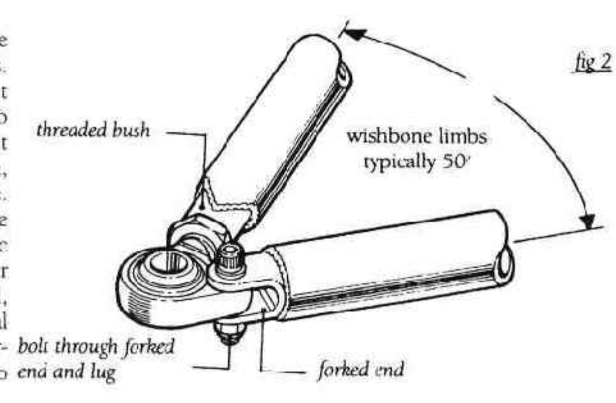
B **Pushrods virtually perpendicular to braking force – unable to transmit load to chassis**

WHATEVER HAPPENED TO...?

There could be a 'half-way house' to taking the bending loads out of the joints for the wishbone apex, which avoids going to the trouble of staking them individually. A few years ago, in RaceTech, I proposed a design for a modified rod-end, with an extra lug on one side of the housing, which allowed it to be used at the apex of a wishbone while still avoiding bending loads - here is the Tony Matthews drawing.



A number of readers have asked what became of this. No manufacturer has yet judged it worthwhile to go ahead. I did file a patent application for the design, but I have now let it lapse. My original RaceTech article put it firmly in the public domain so, taken together with the prior art in the field, there is now no intellectual property reason for an enterprising manufacturer not to develop it.



the shank, arrange the loading so that it's pushing the ball out of its race, and the race out of its housing. To add insult to injury, it's not unknown for the ball to be secured, to an upright for example, in single shear.

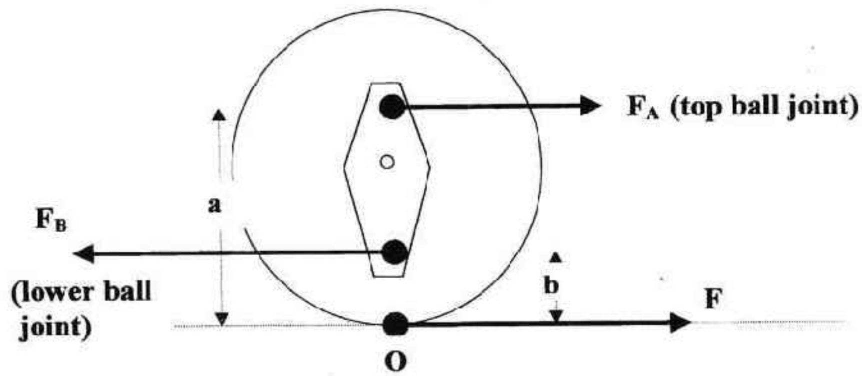
An elegant solution is to use staked joints. It requires a bit more thought, planning and effort than simply screwing a rod-end into a threaded bush in a wishbone, but it allows a lighter, more reliable, longer-living, more confidence-inspiring and, ultimately, cheaper solution to this perennial problem.

Although the highest specification rod-ends are two-piece, the outer race being forged integral with the housing, many rod-ends are staked joints - it's just that the joint manufacturers have already done the staking for you, fixing a ball and race, the bearing, into a pre-prepared housing with a thread attached. You can avoid the compromises inherent in the conventional rod-end housing, if you can make your own custom-designed housing for your wishbone, and then you use simple and cheap tools to do the staking.

The correct procedure is provided by the rod-end manufacturers - indeed many of them freely publish fully dimensioned design drawings of the tools and the steps you must follow to get reliable results. What follows is a guide to the techniques. If you decide to stake your own bearings you should get, and follow, the detailed instructions from your chosen bearing supplier.

The key to the staked joint is the groove, shown clearly in Figure 3, used by kind permission of Aurora Bearings, Aurora, Illinois, USA (aurora_rodends@ibm.net). This groove format is known as the Grumman groove. The simplest, the 'anvil', staking process involves permanently bending, or swaging, the outer rim of the joint's race into a chamfer in the face of the bore of the housing. Doing this on both sides of the joint locks it positively and permanently into the housing. Other techniques using roller swaging tools are used in aerospace applications. In some of the murkier depths of motor racing, it is not unknown for the swaging to be done by carefully using a hammer and a small punch around the circumferences of the groove to peen the lip into the chamfer but, needless to say, this is not approved by the bearing manufacturers. The approved method is quite straightforward and requires only simple tooling, so ad hoc techniques are not really necessary.

The first step is to prepare the housing. Popular "-5 size" 5/16 inch bore bearings have load ratings roughly comparable with a 3/8 inch rod-



$F_A + F = F_B$ (for equilibrium)

$F_A .a = F_B.b$ (moments about O)

So:

$F_A = F . a / (a-b)$

$F_B = F . b / (a-b)$

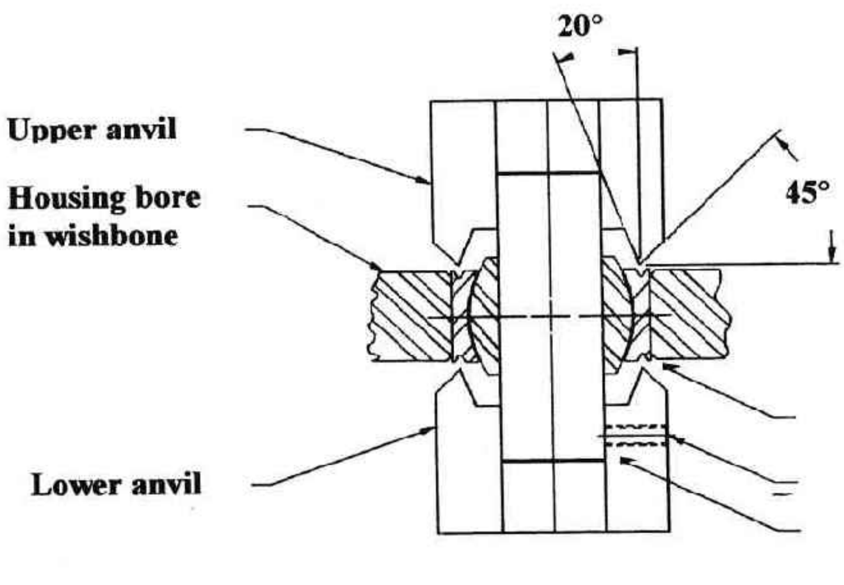


Figure 5 - General arrangement of staking tools for swaging a grooved bearing
(based on a graphic kindly supplied by Aurora Bearings Inc)

end. They have an outer diameter of 11/16 inch, and I have made custom housings from 5/16 x 2 inch bright mild steel strip, and after brazing it into the wishbone leg, machining the bore for the bearing with an 11/16 inch milling cutter (actually a slot drill). The thickness of the strip is conveniently within the permitted 0.005" of the height of the bearing. The bore diameter is critical and must be the bearing nominal OD + 0.0012 "/ - 0.0002 ". Engineer John McCrory of Aurora Bearings recommends final reaming to the size of the housing after the housing has been welded or brazed into the wishbone assembly to avoid ovality. The chamfer is then cut with a 45 degree countersink using the depth stop on a drill press to control its depth and width to the 0.030 " specified for this bearing size. Degrease and then apply Loctite 270 or a similar fitting compound to the mating surfaces for corrosion protection - but ensure that the adhesive doesn't contact the ball or race. The bearing is then pressed into place - this requires only a light force on the outer race.

The swaging is carried out, using a hydraulic press and specially profiled staking tools, the 'anvils', on one side at a time or both together - different national or industry engineering codes have different preferences. The press I use is essentially a hydraulic car-jack in a strong but simple heavy steel frame. A standard workshop press is of course easily capable of doing the job. The dimensions and angles of the tooling are provided freely by the bearing manufacturers in their staked joint catalogues the arrangement in Figure 5, based on the Aurora tooling, is typical.

I machined mine up from standard 'silver steel' bar - 1 inch diameter for the 5/16 inch bearing - then heat-treated to full hardness, around 60 Rockwell C, simply by heating to bright red heat with an oxy-acetylene flame and quenching into water. For staking force, for the 11/16 inch OD bearing, the US National Aerospace Standard (NAS) 0331 specifies 18,000 ± 3,000 lbf per inch of outside diameter, so 5.5 ± 1 tonnes is needed. Goldline Bearings Ltd's rule-of-thumb, which gives the same answer, is ten tonnes per inch of groove diameter. A pressure gauge on the jack, or a calculation of the jack's mechanical advantage used together with a spring balance, will confirm the load to be applied to the jack handle.

Apart from close visual inspection, Goldline recommends a neat and effective way of testing the quality of the job you have done. This involves the breakaway torque of the joint. After installation, this should have increased to up to twice the maximum specified for an un-installed bearing. A simple light spring balance and a 5 or 6 inch long bar through the bore of the ball can be used to check this - for a -5 bearing, typical breakaway torque is 5 lbf.inch as received, up to 30 lbf.inch after installation. If it's too low after swaging, put the job back in the press and repeat the process with a higher load; if it's too stiff, then you have just learned a moderately expensive lesson - you have to scrap the bearing, and possibly the housing as well unless either it is very hard or you can partially machine away the swaged lip, and start all over again. With a new design, or a new operator, the bearing suppliers recommend that you progressively build up the staking force, measuring the breakaway torque each time, until you get the right result. If my experience is anything to go by, you rapidly acquire a 'feel' for the process and soon regularly get satisfactory results.

An alternative or additional test is to apply a 'proof' load to the swaged outer race. Under this proof load the swaged bearing must not be pushed out of its housing. This load is specified for an 11/16 inch bearing in the NAS 0331 as 1000 lbf per inch OD, so it should withstand a load of 688 lbf. The bare bearing has an axial load capacity over twice this.

Grooved bearings are rather cheaper than a corresponding finished complete rod-end, so the overall cost of a suspension part is unlikely to be more than one with a threaded rod-end. Just in case you were wondering, the bearing makers strongly recommend against attempting to give a new lease on life to rod-ends by removing a worn ball and race and replacing it with a new set. They caution that the remaining fatigue life of a reclaimed housing is unknown.

BAD DESIGN

'Some optimists arrange the loading so that it's pushing the ball out of its race, and the race out of its housing. To add insult to injury, it's not unknown for the ball to be secured in single shear. '

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Why a lined bearing has torque

Regarded by many in the motorsport industry as the guru on all rod end matters, **John McCrory** of Aurora Bearings explains the principle behind the self-lubricating PTFE lined bearing

SELF lubricating rod ends and spherical bearings first appeared on race cars in the early 1960s. Their zero clearance fit and maintenance free nature have resulted in the type becoming universal in most forms of motorsport. However, the preload fit can be an impediment to smooth repeatable linkage movement, but hopefully this will help explain why a lined bearing has torque and how it may be managed.

The torque fit of a lined bearing basically results from two things – the thickness of the liner being greater than the radial clearance that exists between the race ID and ball OD ie the compression of the liner, and the co-efficient of friction at the ball/liner interface. While the liner thickness and compression are a result of manufacturing, and generally not user-controlled, the co-efficient of friction can be user-managed.

People often refer to self-lubricating PTFE lined bearings as "Teflon coated". The liner is not a coating as one would find on cookware but rather a PTFE component for lubricity, combined with a carrier, typically a fabric, which gives it load bearing strength.

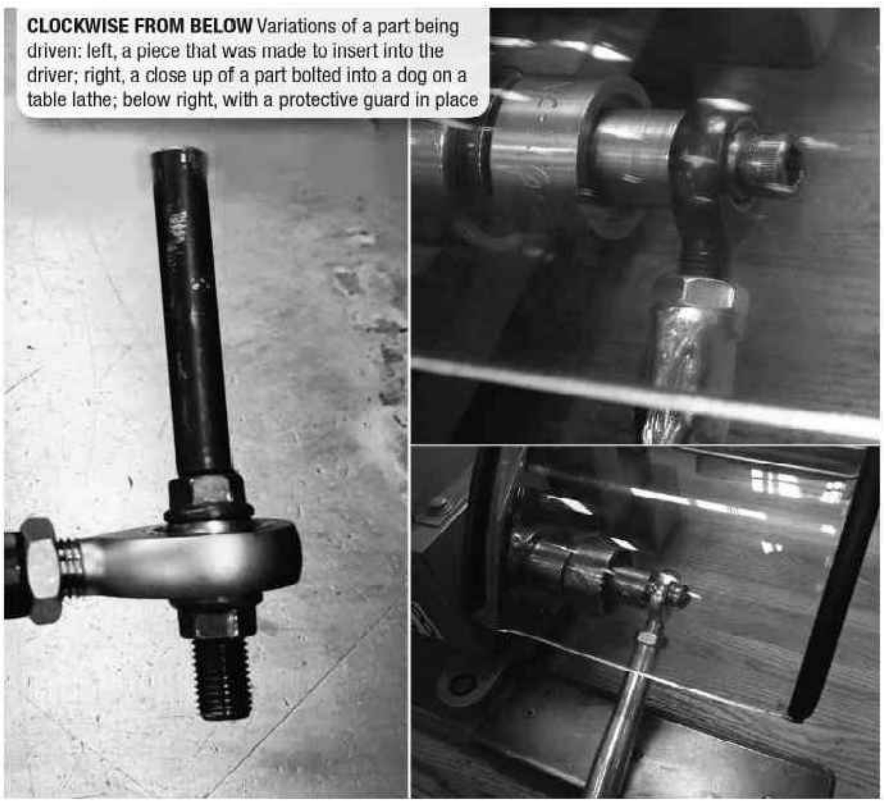
At this point, we will break for some legalese. PTFE is the commonly used abbreviation for PolyTetraFluoroEthylene, a synthetic fluoropolymer of tetrafluoroetheylene. This was discovered by DuPont in 1938 which aggressively and proudly protected their trademark for PTFE, Teflon®. Basically, it should be understood that despite the vernacular use of "teflon", Teflon® is PTFE, but not all PTFE is Teflon®. Use of the Teflon® brand requires express permission and license from Dupont.

PTFE itself is a poor bearing – its compressive strength is only on the order of 10-15,000 psi, but it is an excellent lubricant. However, combined with a fabric , the resulting liner system can have a compressive strength of 50,000 psi or more.

When a bearing is brand new, the liner is in its thickest state, and the co-efficient of friction is highest as no PTFE transfer has occurred; the liner surface upon which the ball rides is not homogenous PTFE, but rather portions of it suspended in a resin, or interwoven with the carrier fabric.

However, as the surface of the ball picks up PTFE from the liner under pressure and movement, it is transferred back to what were initially non-PTFE liner surfaces. This continues where sufficient PTFE becomes burnished onto the ball surface as well as across the liner to the point where no more is transferred as the co-efficient of friction between ball and liner is low enough.

This transfer rate is governed by variables such as load, amount and speed of movement. In instances where ▶



CLOCKWISE FROM BELOW Variations of a part being driven: left, a piece that was made to insert into the driver; right, a close up of a part bolted into a dog on a table lathe; below right, with a protective guard in place

ABOVE Three different components being driven in three different ways. The cutaway (bottom right) is a cross-section of an insert for a rod end bearing showing the outer raceway and the inner ball, with the dark line between the two being the PTFE liner

there are many rod ends in a group or system, such as on the suspension of a race car, this PTFE transfer or "bedding in" can occur at different rates in different locations. However, it is commonly seen as desirable to have all bearings in their optimum state so their torque and resistance to movement are consistent, leading to the action of the suspension, steering, and related control systems being consistent.

In applications where it is desired that all bearings in the system are at their optimum friction level as soon as possible and all at the same time, operations are carried out to "bed in" parts prior to actual use. As a generalisation, a method is found to drive or rotate the ball at a constant speed. For instance, a rod end is fixed in a mating part such as a tube and a bolt is located in the bore of the ball and held in place by a nut. The bolt is then driven, for example by a lathe, at a relatively low constant speed, something less than 100 rpm. Technically, most manufacturers recommend a maximum surface speed for the ball of less than 20 feet/minute, but with no loads 100 rpm is a broad, consistent and safe speed. In addition, the part may also be oscillated.

At a certain point, a full PTFE transfer will be achieved and the torque felt on the mating part will have decreased to a lower, consistent value as compared to the initial condition. If the part gets noticeably hot or begins to smoke, the process must be interrupted until the part cools to ambient temperature. No supplemental lubricant should be used as these are definitely not recommended, either in this process or with PTFE liners in general.

In all cases, not just with the running-in process, lubricants present a unique problem when used with a lined bearing. The introduction of supplemental lubricants such as oil or grease or of moisture/corrosion inhibitors can in fact have a detrimental effect on the performance of PTFE liners.

The transfer of PTFE from the liner to the ball and back is the basis of the self-lubricating nature of a lined bearing. When a lubricant is introduced, the lubricant film that develops between the ball and liner acts as a barrier that impedes the burnishing action. The movement between the two surfaces can remove the exposed PTFE from the liner face, but the material will not adhere to or burnish on to the ball surface. Eventually, the wear surface of the liner may have all the exposed PTFE material removed at which point the coefficient of friction between the uncoated ball and the wear face of the liner will drastically increase.

In the normal cycle of PTFE transfer, as it is no longer exposed above the liner surface, localised portions of the liner are removed to expose additional PTFE and continue the self-lubricating transfer cycle. In the case of a lined bearing that has been lubricated, little or no PTFE is exposed above the liner face, not just locally but around the entire ball surface, and degradation of the wear face to expose sufficient PTFE cannot occur fast enough. When this occurs, a used bearing will actually become or be tighter than when new. "Running-in" will jump start and accelerate the PTFE transfer process, ideally recreating a PTFE on PTFE interface.

Whether it is to establish the optimum friction level on new, unused bearings, or recreate a PTFE on a wear surface after it has been compromised, running in is an effective and established procedure and can be a benefit when done correctly. HRT

Broken Rod End at BRIC

-by John McCrory
Aurora Bearing Company

At the end of this Summer's Brian Redman International Challenge weekend, a friend of mine flagged me down, and tossed me a metallic object. "Here," he said. "They found this as they were cleaning up the mess." The mess he referred to was the unfortunate pile up at the start of the Sunday Group 6 race. The object was the head of a rod end. To most people, it would be considered a piece of race track refuse. To me, it's a lot more. It tells me a lot of things that would concern me if I found this part on a car, and I'd like to share these things with you.

Portion of the rod end found at the track.

Before going any further, I will be up front that the part, or more correctly, the part that this piece was off of was not an Aurora part, and I work for the Aurora Bearing Company. The manufacturer of this part is unimportant, but I want to be honest about what I know. Also, I do not know anything as far as a direct fact as to what car it was off of, or where it was used. However, my experience with rode ends, race cars, and the types of cars involved in the incident, lead me down a certain path of reasoning.

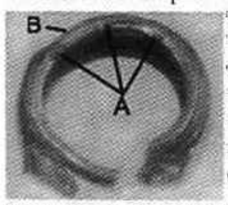
The part was a two-piece, commercial grade, female rod end, 5/8" bore. The profile of the head, with the spherical inner area is the obvious indication of the two-piece commercial design. The female conclusion is drawn from the geometry on either side of the break that indicates a transition to a wide, barrel shape female body as opposed to the narrower cross section of a male body at the junction of the head and threaded shank. From this, I would suspect it was used as a tie rod end joint. The tie rods on most late model Detroit cars as in the field, i.e. Camaros, Mustangs and derivatives, Corvettes, etc. have a 5/8" male thread, so this is a logical place to see a joint as described used.

Rod end piece found, along with a complete unit.

I don't think this was the first incident that this joint, or car, was involved in. There are two big flat spots on the outside diameter of the piece. These marks occurred long enough ago that a coating of light rust has appeared. There are also a number of smaller gouges and dents that have discolored. There are also a few gouges that are shiny. These more likely occurred in the incident.

This part was not well maintained. If it had, the inner race would have had a coating of fresh grease. This part has a thin film of almost dry lubricant combined with dirt. There are a few areas that show through and are highly polished from the ball running dry on the race area. I would suspect that if you looked at the ball from this part, it would show discoloration or black scorch marks from running dry under load. The way the part shows the shank being torn from the head, and the ball being ripped out leads me to believe that the spindle/upright/wheel was torn from the control arm, with the rod end ball left on the steering arm, and the rod end body left on the tie rod.

The fact that the area where this piece separated from the rest of the part shows no evidence of rust, combined with the fact that you would not expect a piece of debris this large to have laid undetected on an active race track makes me conclude it was generated in the incident.



Gouges and dents from previous incidents (A), and previously ground down area (B).

So beyond the fact that I've proven I can make a lot out of a little, what have I shown that might be reason for concern?

This part had been in incidents before, had visible signs of damage, and therefore its integrity had been compromised. It probably had a noticeable amount of play in it. I would not want to see this on a race car, especially in the application presumed.

It was not well maintained, again as evidenced by the lack of grease. This would accelerate wear, and therefore development of play between ball and race.

I don't know if it would be the place of a tech official to object to this part being on the car. The car was certainly operating with the joint on it, and the damage to the joint (i.e. the body ripped in two) was most certainly an effect of the wreck.

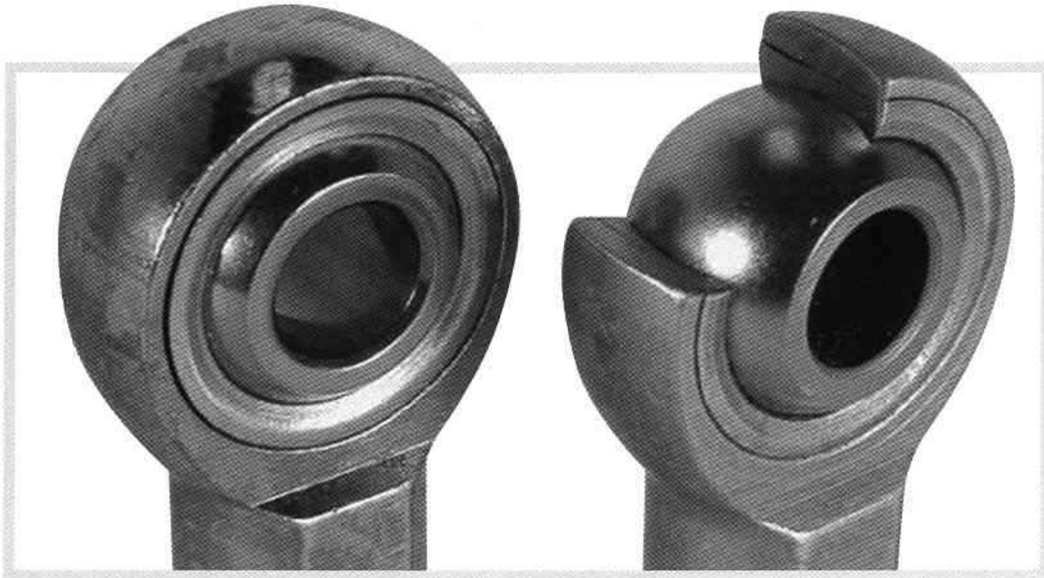
However, I would argue that running the joint on the car was a questionable choice. Again, the joint appears to have been previously damaged, it had also been poorly maintained. It is reasonable to say that it no longer could be relied on to provide the same level of performance, structural integrity and safety it did when new.

2 piece commercial grade rod end. Low carbon steel body, 9500 lb. load capacity.

This type of joint retails for about $15. It is made from low carbon steel, and has a load capacity of about 9500 lbs. It requires maintenance in the form of maintaining a coating of clean grease on the ball and race. For about $45 retail, this joint could be replaced by a precision grade, self-lubricating, heat-treated alloy bodied rod end. This type of joint has a load capacity of over 17,000 lbs, and is maintenance free. While it's easy for the sales guy to spend the customer's money, the long-term reliability and performance of the higher-grade joint should be well worth the cost.

3 piece high strength precision grade rod end. Heat treated alloy body, 17,000 lb. load capacity.

If there is one thing racers should take away from this, it's that a closer look at the condition of the rod ends on your car, and consultation with the manufacturer of the joints, and car builder if available, about the application, along with replacement when appropriate, will enhance the long-term reliability of your race car. Again this joint appears to have failed because of severe stress caused by an accident. However, it was much closer to the end of its life than the beginning. For the sake of reliability and long-term safety, it probably should have been replaced a while ago. ■



ROD END INTEGRITY

ACCORDING TO **AURORA** THE STRUCTURAL INTEGRITY OF STEEL RACE ROD ENDS DESERVES CONSIDERATION IN NON-SUSPENSION APPLICATIONS

At the highest levels of motorsport, the general specifications of rod ends used in suspension linkage applications are fairly universal. The component parts will all be constructed of heat-treated steel, and the part will include a liner combining a carrier fabric for strength with a PTFE component for lubricity. A PTFE-lined rod end offers optimal stiffness, strength and frictional characteristics cost-effectively.

Surprisingly, the same care in choosing rod ends for suspension applications is often not exercised when choosing joints for control linkages in areas such as throttle control or clutch linkages. These are often low-specification, industrial- or commercial-grade rod ends. This class of bearings is geared toward applications with low loads, without shock or high-frequency oscillation.

The most common commercial joint has the raceway formed by two pieces of brass or similar material wedged into body cavities. Also common are molded race bearings where the body is machined with a through bore larger than the ball diameter, and the area between filled with an injected plastic, typically a nylon-based compound. Many include channels to achieve mechanical locking of the race. With both configurations, the race material is relatively soft at no more than 2,070bar, and not tolerant of high-frequency, vibratory loads.

In the US market, a common alternative is the two-piece rod end, where the raceway is machined as part of the rod end body, and cold formed around the ball. This one-piece steel raceway offers greater strength and vibration resistance. It has greater integrity and reliability due to the full wrap around the ball.

Three-piece, commercial-grade rod ends use a one-piece steel raceway, swaged around the ball. The insert is then staked into the housing, positively locking it, giving high dynamic integrity. This is the same basic design used on high-performance rod ends.

PTFE-lined joints can be seen in control linkages, although their use requires understanding of their characteristics.

A lined bearing requires a force to oscillate the ball, often defined as no load breakaway torque – the amount of force required to initiate and maintain ball rotation with no load on a part. Care must be taken to determine if this torque will be an impediment to proper action of the assembly. Typically this is a concern in throttle linkages, where the system forces are light.

Also of concern is temperature. A liner is bonded to the raceway using a thermosetting glue. The temperature limit of these glues is typically no more than 163°C. Temperature is also a concern with molded race parts, with many manufacturers recommending a maximum temperature of 120°C.

In applications with very low loads combined with minimal movement, it may be tolerable to not lubricate a metal race bearing. However if the ball discolors then lubrication is required. As loads increase, the ability of greases to maintain a lubricant film decreases. With increasing loads, the viability of the PTFE-lined alternative becomes greater. A non-lined part exhibits no breakaway torque at rest. As loads overcome the strength of the applied lubrication film, the breakaway torque becomes much greater. At this point, the lined part that had a torque at no load will actually have a lower torque than the unlined metal race part with minimal or compromised lubrication.

TOP: A THREE-PIECE MALE ROD END ON THE LEFT AND A CUTAWAY VIEW (RIGHT)

ABOVE RIGHT: AN EXPLODED VIEW OF A THREE-PIECE MALE ROD END

LEFT: CUTAWAY VIEW OF A FOUR-PIECE BRASS RACE MALE ROD END

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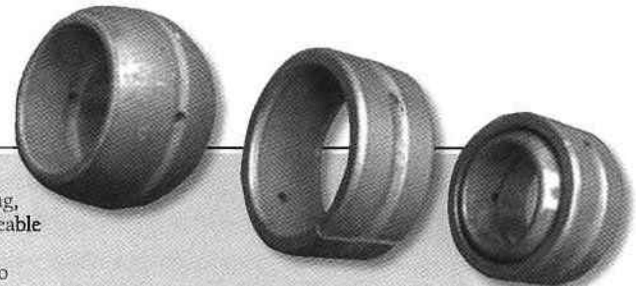
READER ENQUIRY REFERENCE
006

ALTERNATIVES TO FRACTURED RACE BEARINGS

READER ENQUIRY REFERENCE **020**



Most spherical bearings used in motorsport applications incorporate single-piece race construction, with the raceway formed around the ball. Swaged bearings provide close tolerance fits, allowing precise motion control.

Dampers and strut mounts (camber plates) often incorporate fractured race bearings, which use raceways manufactured separately from the ball, fractured across the circumference, and spread open to be assembled around the ball. Because the components are finished separately, these bearings do not have the ball to race conformity or close tolerance fit of swaged bearings. Industrially, they are intended primarily for static situations or slow dynamic applications, with incidental misalignment.

They are generally the least expensive form of spherical bearing, and not dimensionally interchangeable with common swaged bearings.

Swaged bearings are designed to accommodate regular misalignment. Materials are chosen to provide dynamic load-bearing capacity, as well as durability. Used in dynamic motorsport applications, users often find fractured race bearings deliver less than the desired precision and durability.

Alternatives to the metal-on-metal fractured race bearing are available, but not readily known. As with swaged bearings, PTFE lined alternatives are available from many bearing manufacturers. Fractured race bearing part numbers generally carry a suffix 'ES'.

Replacing this with a 'C' or 'ET-2RS' will specify the dimensional equivalent with a self-lubricating, clearance diminishing or eliminating PTFE liner. Certain manufacturers also offer full swage equivalents. Ultimately, manufacturer consultation will prove useful in determining beneficial specification upgrades.

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The Busted Knuckle

By Art Elm

Get Your Bearings

While most people are familiar with bearings to the extent of having them on their car for the wheels, they do not realize the car has a great number of many different types of bearings.

What is this great little machine called a bearing? How do we use them in a racecar, and how can we make the best use of them?

The concept behind a bearing is the simple fact that things roll better than they slide. The tires on a car are like bearings, but if they were like skis, it would be a lot more difficult to push down the road. This is because when things slide, the friction between them tends to slow them down. But if the two surfaces can roll over each other, then the friction is reduced by a great amount. Bearings reduce the friction because they are made with smooth metal balls, or rollers, and a smooth inner and outer metal surface for the balls to roll on. The balls or rollers hold the weight or load, and allow it to spin or move smoothly.

A load is a force that is applied to the bearing. The application and the load applied to it determine the type of bearing used. Bearings generally have to deal with two different kinds of loading called radial and thrust. Depending on what the bearing is being used for, it may see all radial loading, all thrust loading or a combination of both.
A radial load is applied perpendicular to the shaft axis. The water pump and the alternator are examples of components that use bearings that are subject to only radial loads. In these examples, most of the load comes from the tension in the belt that connects them with the crankshaft. That is the reason for setting proper tension on the belt. Excessive belt tension can lead to bearing failure.

A thrust, or axial, load is applied parallel to the shaft axis. Picture a bar stool or a Lazy Susan turntable. The bearing underneath is subject purely to thrust, and the entire load comes from the weight of the person or thing sitting on it. These types of loads are found in transmissions or steering boxes where helical gears try to push the shafts out of their housings.

A combination of the two loads can be found in the hub of the racecar wheels. This bearing has to support a radial load from the weight of the car as well as a thrust load that comes from the cornering forces in the turn.

There are many types of bearings that are used for a variety of different purposes. They include ball bearings, roller bearings, ball thrust bearings, roller thrust bearings, and tapered roller thrust bearings.

The most common type of bearing is the ball bearing. They can be found in everything from skateboard wheels to computer hard drives. These bearings can handle both radial and thrust loads and are usually found in applications where the load is small. In a ball bearing, the load is transferred from the outer race to the ball, and then from the ball to the inner race. Since the ball is round, it only contacts the races at a very small point. This means that the balls must be larger to carry a load equal to that of a roller bearing that distributes the load over the length of the roller. They are used mostly where size and load capacity is not important, but where ease of assembly and low cost are.

This is a ball bearing that has been taken apart. The manufacturing process makes sure that the balls are precisely ground for minimum friction. It also uses a metal cage to hold the balls in place on the inner race.

Roller bearings are used in applications such as conveyer belt rollers, where they must hold heavy loads. In these bearings, the roller is a cylinder instead of a ball, so the contact point becomes a line. This spreads the load out over a larger area and allows it to handle much greater loads than a simple ball. It is not designed to handle much thrust loading, although they are used as rear axle bearings in a number of vehicles. A variation of this type is called a needle bearing, because of the use of small diameter cylinders. This allows them to be used in tight places and is found under the gears in transmissions to keep them spinning.

Thrust bearings are flat and are generally used in transmission gear sets

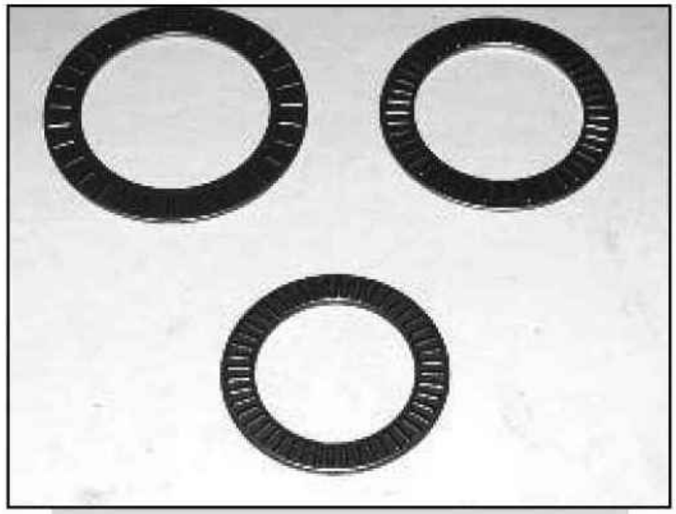
An example of a caged needle bearing. This particular bearing is used as a pilot bearing for a clutch assembly. It is used to keep the transmission input shaft spinning free in the end of the crankshaft. It is a radial load type bearing.

and between gears to keep them apart and free. They are sometimes referred to as Torrington bearings. The tapered roller bearing combines the benefits of all the other bearing types as well as some additional advantages. They combine radial and thrust load capacity and a longer relative life or reduced bearing size for a given load capacity. They are less sensitive to contamination due to the natural pumping action, which forces any particle contaminants out of the roller/race contact area. They have a lower friction coefficient and higher speed capabilities due to a truer rolling motion, when compared to other bearing types. For optimum performance either endplay or preload values can be specified, making them more adjustable.

Because bearings must meet precise constraints regarding load and minimum friction, the manufacturing process is as precise as any high-tech machine. The raw material is first shaped to within several 100ths of an inch. This is done by cold pressing, forging or hot rolling of bars of Cr6 type steel to form the balls or rollers. The bearing races are first shaped from tubes of the same material on a spindle turning machine. They are then heat treated to 850 degrees C. This is called Austenitizing, which alters the structure of the material. It is then rapidly cooled to 40 degrees C. to fix the structure of the material and thus increase hardness. Then it is tempered, or reheated, to 170 degrees C. to reduce the effects of thermal shock inside the structure and stabilize the material. It is then ground to the finished shape to within 1000th of an inch. Finally the bearing balls, rollers and races are super-finished by polishing with a grinding rod to a finish of .06 micron.

This is a cut-apart view of a tapered roller bearing commonly used as a car type wheel bearing. Notice the inner race where the rollers ride and you can see that it needs to be perfectly smooth. Also of note is the metal cage that holds the rollers together and separates them. This also helps to make the bearing free, with less drag.

Each bearing is designed to live in a particular environment. The bearing has a life expectancy because it is a wear part, and because of this it is destined to die a natural death. The bearing's service life is a function of its capacity to withstand fatigue. When rolling bodies, the balls or rollers, roll on the raceway, or races, they create significant cycles of compression and shear stress loading. Just like potholes in roads are caused by the constant passage and pounding of cars and trucks, bearing races crack under the repeated passage of the balls or rollers. Even under standard conditions, the bearing will eventually be damaged. The contact between the rolling bodies



This is a picture of thrust bearings or Torrington bearings that are used in transmission gear sets. Also used at the front of engine camshafts to keep the cam from walking forward or back due to the force of the helical cut distributor gear.

and the race give rise to extremely high loads, both compression stress loading at the surface and shearing stress loading in the sub-layers of the material. The bearing's service life is predicted by its ability to withstand this stress loading.

Premature damage of a bearing can cause an early death of the bearing and a component failure. Incorrect lubrication, such as too little or the wrong type, accounts for approximately 70 percent of bearing failures. In about 20 percent of failure cases, the penetration of liquid or solid foreign particles caused damage. This of course shows the importance of proper and good condition seals. The incorrect assembly that can cause excessive heat causes about 10 percent of failures. Over-tightening of the wheel-bearing retainer nuts will definitely burn out a bearing, even with the best type of lubrication. A small percentage of failures are generally caused by the use of bearings too small for the normal load, corrosion, the passage of an electrical current or some other strange occurrence.

These two bearings are heavy duty ball bearings used for the input and output shafts of a manual transmission. The inner race, the balls, the cage and the outer race are all made together as to ensure maximum load and minimum friction.

For racecars all this means that you should repack your wheel bearings often with the correct type of high-temperature grease, maintain the seals in good condition, and adjust them properly, with the correct amount of preload. Check and maintain the proper belt tension on the water pump, alternator, and power steering pump. Replace any and all bearings at the first sign of roughness when they are turned by hand. They will only get worse and can cause a component failure at the worst time.

Get your bearings and they won't get you.

On a roll

David Cooper examines the rolling-element bearing – the different types, materials, applications and how it compares with other bearing technologies

Bearings are possibly one of the more innocuous components of a racing powertrain, but without effective and reliable bearings, any motorsport powertrain would rapidly grind to a disastrous halt. This article is about current trends in powertrain bearing technology; in particular there have been a lot of recent developments in rolling-element bearings, as we identify here.

Bearings can be divided into two broad types – rolling element or contact, and journal or fluid film. As its name suggests, a rolling-contact bearing is one in which the main load is transferred through elements in rolling contact between an outer and inner surface or 'race', as opposed to a journal or fluid-film bearing, where the load is transferred by the relative sliding of a lubricated shaft and bushing (where the lubricant may be fluid or solid film).

While journal bearings find extensive applications in an engine, the wider powertrain is the domain of the rolling-element bearing. Journal bearings are particularly suited to high-speed, high-temperature applications, where space is at a premium, such as for crankshaft main bearings and con rod big- and small-end bearings, where they are used almost exclusively. Rolling-element bearings find applications in areas where the reduction of friction is of paramount importance, although they tend to be more complex to manufacture. They also need far more space, although needle roller bearings can be more competitive in this regard.

One-piece journal bearings, often used in four-stroke con rod small ends and NASCAR camshafts (Courtesy of Dura-Bond)

While the overall design concept for rolling bearings remains fundamentally unchanged, the subtleties of their materials and design have advanced significantly to provide optimum performance and reliability. Rolling-element bearings can take many forms depending on the application, ranging from bearings capable of supporting high radial loads with minimal friction to tapered or axial contact bearings capable of resisting axial loads, so they are a particularly versatile solution.

The mechanisms of bearing failure are a key issue, as it is these mechanisms (alongside the search for reduced friction) that drive material selection and development, and the overall performance of bearings.

Bearing failure

Rolling-element bearings are traditionally made from hardened steel alloys, providing the minimum surface friction and greatest wear resistance possible, while remaining cost-effective to manufacture to high tolerances. With the move to increasing service life and reducing maintenance costs at all levels – from Formula One's mandatory engine and gearbox lifetimes to life expectancies well in excess of 150,000 miles for domestic automotive vehicles – the design, manufacture and installation of highly reliable long-life bearings is vital.

Bearing failure is generally a materials problem, with failure occurring through several mechanisms or combinations of operating conditions. Assuming that a bearing remains correctly lubricated and sealed against the ingress of debris, the primary mechanism is materials fatigue, generally termed rolling contact fatigue (RCF).

Rolling contact fatigue

The ultimate limit on bearing life is failure by RCF, which induces spalling of the material's surface. To combat RCF in steel bearings, it is important to understand the material's tribology – the mechanisms by which the surface of the bearing wears with friction and lubrication.

RCF is a failure mode by which a surface in rolling contact with another begins to deteriorate after the cyclic loading experienced in operation. This failure is linked directly to the stress field within the material in the region just below its surface, generally referred to as the Hertzian contact stresses. Hertzian stresses are proportional to the normal load, and inversely proportional to the contact area, so they are particularly high for ball bearings where there is a minimal contact area. This stress distribution, having its maximum stress at a point below the material's surface, leads to the formation of subsurface cracks as a result of cyclic loading. Once a crack has been initiated below the surface, a flake or chip eventually works loose, and the bearing's performance begins to degrade from this point.

Measures against RCF

To improve the RCF life of steels, various coatings and surface heat treatments can be applied, either by thermal spraying or vapour deposition to apply a harder-wearing coating. Surface coatings of ceramic or harder metals can improve RCF properties, but applying such coatings can be tricky, as thermal mismatch between the coating material and its substrate can lead to residual stresses for example. Coating parameters must also be precisely controlled to ensure consistent coating thicknesses and quality.

Steels may also be case-hardened, or otherwise surface treated to produce a refined surface microstructure that can better resist the loads required.

RCF can be accelerated by problems outside of the bearing itself, such as excessive loading or off-axis loading created by a distorted shaft or misaligned bearings, which can increase the probability of a surface fatigue failure

Possibly the greatest advance though in reducing failure by RCF has been in the ever-improving quality and cleanliness of the steels used. Contaminants within the material can act as stress raisers or nucleation sites for cracking to occur, but by better quality control of the raw materials the likelihood of an early fatigue failure is reduced.

Fatigue is not a standalone problem though; it is likely to be accompanied or accelerated by additional damage from the presence of debris or other abrasive particles, or excessive heat caused by improper or insufficient lubrication.

Lubrication failure

If there is insufficient lubrication then overheating can occur. This overheating, or a breakdown in the lubrication itself, can have various effects on a bearing, any of which alone or in combination can lead to its failure.

Excess heat, caused either by running the bearing beyond its operating speed or load, or by poor lubrication, can affect the performance of lubricants. Heat can ultimately degrade and melt the grease-type lubricants commonly used in sealed bearings, leading to further overheating and a degrading spiral of cause and effect.

Extreme surface temperatures can then alter the material's microstructure within that critical subsurface region, effectively annealing the hardened steels and reducing their mechanical properties once the bearing cools and the microstructure re-crystallises, so reducing the surface hardness and strength. Even if no microstructural change takes place, the mechanical properties of steels degrade quickly with temperature, moving the materials out of their safe operating window with respect to the loads experienced, increasing the chance for failure and accelerating the effect of material fatigue.

In the event of a lubrication failure or reduction, the heat increase in the bearing can also cause micro-weld adhesion. The materials of the rolling element and the race do not have perfectly flat surfaces – as with any machined component, microscopic peaks and valleys are actually present. The lubricant serves to fill the valleys and form a film between the peaks, distributing the load over a contact area and preventing any real surface-to-surface contact.

If this lubrication film breaks down, the peaks may make contact, creating an exceptionally high pressure. These pressures lead to temperature rises, and momentarily friction-weld the tiny contact areas together, before inertia overcomes the micro-weld and the rolling element breaks away. This generates surface damage and debris, accelerating bearing failure. The primary method to combat micro-weld adhesion is therefore to ensure adequate lubrication, and so maintain the bearings at their operating temperature.

Corrosion and debris

The presence of corrosion or rust on the surface of a raceway or rolling element can form surface pitting, which accelerates fatigue failure. Although bearings are stored and packaged with a film of lubricant to keep oxygen from the surface, partial removal of the film during installation, or the presence of water – perhaps via a minor seal failure on a water pump, or through condensation – can provide the opportunity for corrosion to develop.

While journal bearings typically include a soft metal coating designed to capture hard foreign particles via embedding, rolling-element bearings are not so forgiving. The presence of debris can be highly damaging to the surfaces of the rolling elements or races, leading to vibration, increased wear and premature failure. Debris may enter the bearing, or may be generated within the bearing itself via spalling. As the bearing rotates, the particles may then be indented into the raceway, or may gouge it. This surface deformation then leads to increased localised surface stresses that in turn accelerate a fatigue failure, generating more debris and so on.

Brinelling

Brinelling occurs when an excessive (usually one-off) radial loading forces the rolling element to indent the surface of the race, plastically deforming it. This reduces the effectiveness of the bearing, as it will no longer rotate smoothly. False brinelling, which is similar in appearance to brinelling, can also occur if a bearing experiences severe vibration while remaining relatively stationary. Micro-cracking can occur where the rolling elements contact the race, which then leads to premature spalling; however, such a scenario is unlikely to occur within the ever-turning bearings of a powertrain.

Applications

The biggest factor affecting the lifetime of a bearing is its application. Very similar bearings may be used in a circuit race engine and a drag car engine. However, the expectations differ; for many applications, con rod bearings will be expected to last a season before servicing and replacement. The same bearing, in a very similar engine architecture ▸

but within an 8000 hp nitromethane dragster, will be replaced or at least inspected after a bare 4 s or so of full-power running.

Bearing materials

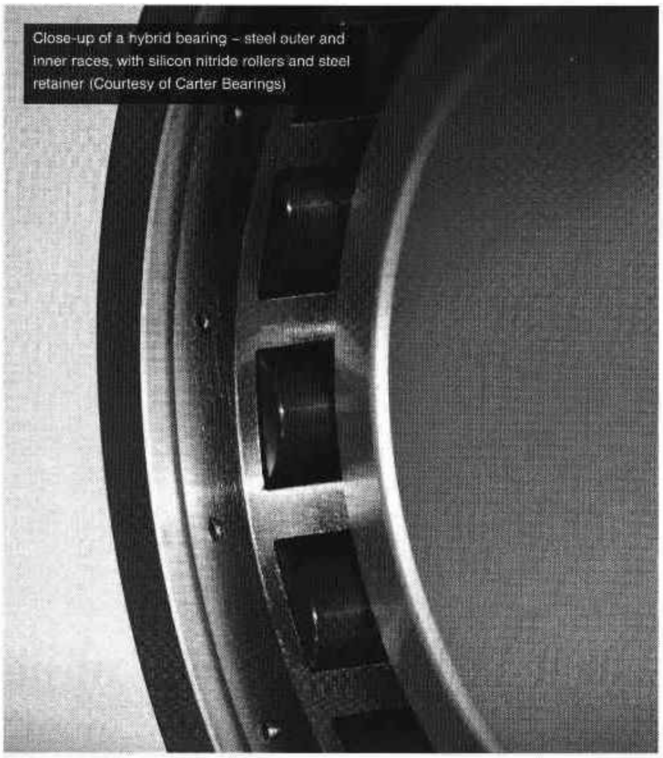
Bearings are normally made from steel alloys, either stainless chromium-based or high-alloy steels for enhanced corrosion resistance or surface hardness. Typically races are made from alloys such as 51200 tool steel, while 440C stainless steel can be used if the application calls for it.

Despite the perceived excellence and growing use of materials such as titanium and high-strength aluminium alloys – and they are undoubtedly excellent in their applications, having fantastic specific (per kg) strength and stiffness – few materials can beat the ultimate properties of steel, particularly given its low cost and high availability, so steel remains the mainstay bearing material. Apart from very high-end or specialist applications, steel has been the only sensible, reliable bearing material choice for rolling bearings throughout the driveline, at least until recently.

Ceramic bearings are the new technology on the block. Although not a recent advance, the use of ceramic materials for bearings has grown significantly, and they are becoming increasingly cost-effective. Bearings can be completely ceramic (rollers and races) but a good combination for performance and cost-effective use appears to be hybrid ceramic bearings.

As they sound, hybrid bearings are a combination of a steel and ceramic bearing, generally a chromium steel race coupled with ceramic ball bearings. The use of high-quality steel and machining processing for the races allows a high surface finish to be achieved, resulting in an exceptionally low-friction bearing.

The rolling elements, having a relatively simple geometry, are presumably more economical to manufacture from ceramic than would be the entire bearing, and the most popular ceramic in use appears to be high-grade silicon nitride, which possesses excellent mechanical properties, with much greater hardness (1500-1600 HV) than can be achieved with steel (up to 650 HV). Silicon nitride rolling elements have a much lower density – $3.2 g/cm^3$ – making them 60% lighter than steel, which provides benefits not only for static weight but also in reducing inertial loads and parasitic losses.



Close-up of a hybrid bearing – steel outer and inner races, with silicon nitride rollers and steel retainer (Courtesy of Carter Bearings)

Silicon nitride rolling elements begin life as a raw powder material, which is then sintered under extreme temperature and pressure to consolidate a fully dense blank. The blanks can then be machined, typically via a diamond grinding-type process capable of achieving the high dimensional tolerances, along with exceptional surface finish.

Although the performance benefits (low rolling resistance) are available by using a hybrid bearing, the area where hybrids really win out is in reliability and service life. Comments from several bearings manufacturers during the research for this article suggest a general four- to fivefold increase in bearing life for a hybrid bearing over a traditional steel bearing. Depending on the application, this may translate into replacing a bearing after perhaps two or three racing seasons compared to every season with a steel bearing. This increase in service life not only saves the cost of replacing bearings but also the attendant time in replacing or servicing them.

Hybrid bearings can also be more forgiving than steel, as they can survive with less lubrication than an equivalent steel bearing. Ceramic rolling elements have a much lower friction coefficient (about 10% that of steel) so lubrication is not as essential for friction/rolling, while the difference in material – particularly the ceramic's high melting point – removes the potential for failure by micro-weld adhesion. The use of steel races, however, indicates that lubrication will certainly remain a necessity, to prevent contact damage, corrosion and overheating of the steel race.

Retainers

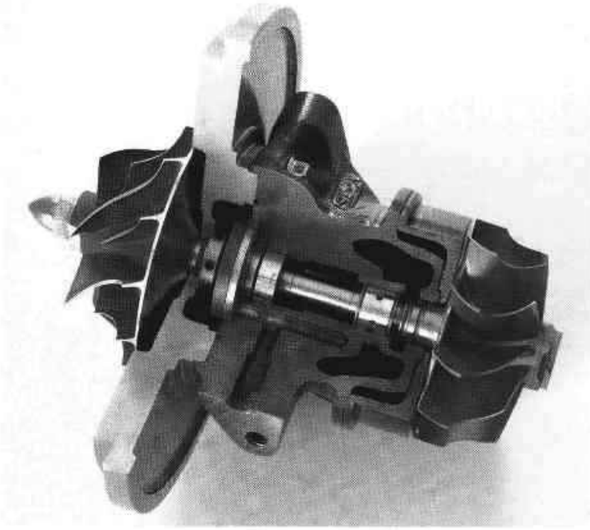
Bearing retainers are required in many forms of rolling-element bearing, as they serve to keep the rolling elements from contacting each other. Depending on the application, retainers can be made from steel or polymers. Retainers need to be stiff enough to maintain the elements' relative positions, although any improvement in weight can only be beneficial.

While machined or pressed steel retainers are common, a trend towards polymers for lower weight and cost is evident, although they are limited by service temperature. PEEK (polyether ether ketone) is an alternative where temperatures and corrosive combustion by-products exceed the abilities of common polymers; the cost trade-off though means steel retainers are likely to remain.

Seals

The modern ball bearing is more often than not a sealed unit, pre-lubricated and sealed at the factory. This guarantees the correct quantity and type of lubrication for the rolling elements while also preventing the ingress of debris – wheel bearings are an ▸

Hybrid turbocharger bearing system, showing thrust bearing and ceramic bearing at left-hand end, journal bearing at right (Courtesy of Turbonetics)



obvious example here. Sealing the bearing also separates it from its environment, allowing the optimum lubricants to be used, for example in the gearbox, and the optimal lubricant for the bearing, rather than requiring the gearbox lubricant to serve two masters.

Seals also protect against the ingress of moisture. While this can cause corrosion of steel raceways, accelerating failure, moisture can also emulsify with the lubricants present, reducing their performance and again leading to failure.

Bearing applications

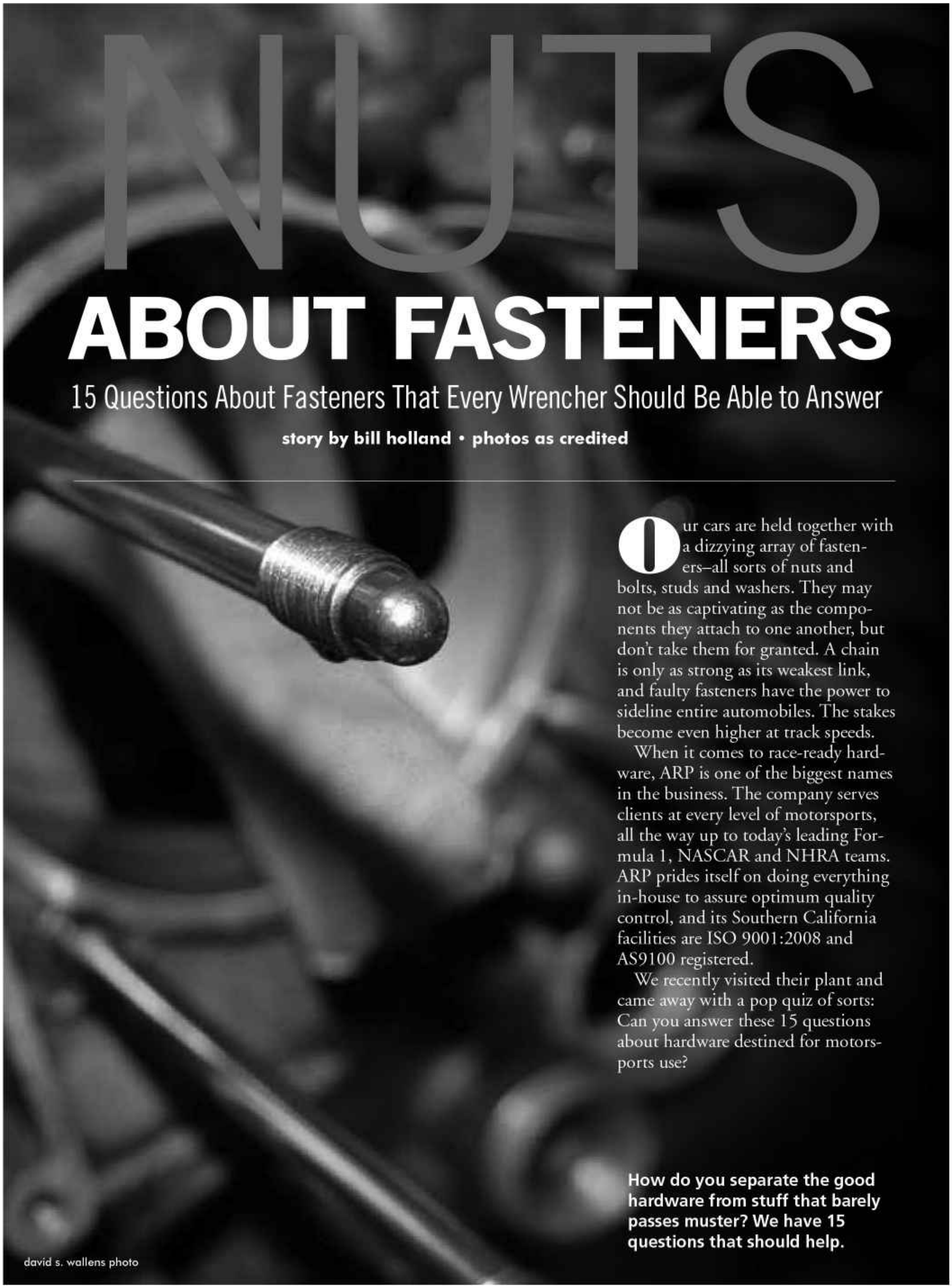
The applications for steel rolling-element bearings, and now more commonly hybrid bearings, are wide-ranging in the racecar powertrain. Looking deeper into the anatomy of the drivetrain, the bearings are present in a variety of forms. Steel needle rollers, larger roller/barrel bearings and ball bearings all find applications in common types of constant velocity joints; sealed ball-bearing races remain the mainstay for the support of gearbox shafts, while engine ancillaries use many forms of rolling-element bearings to support rotating shafts in water pumps, oil pumps, alternators or belt tensioners.

Although relatively mundane, the consistent and reliable performance of all these bearings remains critical to the smooth running of any racecar. Even if no failure occurs, longer service lives translate to lower maintenance costs. It appears, however, that rolling-element bearings do have a part to play in some of the more exotic future powertrain applications, finding uses in turbochargers and mechanical (flywheel) energy recovery systems – although not without competition from other bearing technologies.

Race Engine Technology

Established in 2003, Race Engine Technology is a unique, high quality review of contemporary racing powertrain technology. It is widely read by design and development engineers and others involved professionally in this worldwide industry and just as avidly by those interested in a subject that has a huge ‘enthusiast’ following. it reaches the majority of all competition engine builders right across the globe - the heart of its readership. It is published by High Power Media which is let by specialist publisher Simon Moss and renowned motorsport editor, Ian Bamsey. HPM also publishes annual technology reports focused on specific forms of motorsports, along with two free-for-life online technical resources, RET-Monitor and F1-Monitor.

Nuts About Fasteners originally appeared in the May 2016 issue of Grassroots Motorsports.
This is an excerpt from the original 15 point article.



NUTS ABOUT FASTENERS

15 Questions About Fasteners That Every Wrencher Should Be Able to Answer

story by bill holland • photos as credited

Our cars are held together with a dizzying array of fasteners–all sorts of nuts and bolts, studs and washers. They may not be as captivating as the components they attach to one another, but don't take them for granted. A chain is only as strong as its weakest link, and faulty fasteners have the power to sideline entire automobiles. The stakes become even higher at track speeds.

When it comes to race-ready hardware, ARP is one of the biggest names in the business. The company serves clients at every level of motorsports, all the way up to today's leading Formula 1, NASCAR and NHRA teams. ARP prides itself on doing everything in-house to assure optimum quality control, and its Southern California facilities are ISO 9001:2008 and AS9100 registered.

We recently visited their plant and came away with a pop quiz of sorts: Can you answer these 15 questions about hardware destined for motorsports use?

How do you separate the good hardware from stuff that barely passes muster? We have 15 questions that should help.

david s. wallens photo

1. In addition to fatiguing from the cyclical loading and offloading, ferrous materials can deteriorate over time due to stress corrosion cracking, rust, permanent deformation and/or galling. It's good practice to not reuse old fasteners for any critical application without carefully and thoroughly inspecting them.

2. Most quality commercial fasteners have markings on the head. Three lines on an SAE bolt indicate Grade 5, or a tensile strength of 120,000 psi. Some racing associations require at least Grade 5 bolts for attaching roll bars and the like. The metric equivalent is stamped "8.8."

Six lines denote Grade 8 hardware, which has a nominal rating of 150,000 psi. Class 10.9 is the metric equivalent.

ARP stamps its initials on its bolt heads, and sometimes also places identifiers regarding the materials and/or tensile strength. The company's cylinder head, main bearing, connecting rod and driveline fasteners are rated at a minimum 200,000 psi–some are even rated as high as 280,000 psi.

Bottom line: If the head of a bolt is blank, beware.

3 COARSE OR FINE THREAD?

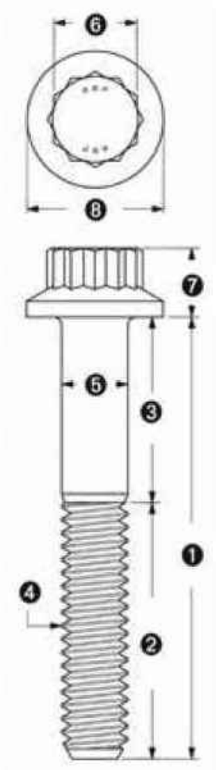
3. There are several reasons why fine-threaded fasteners are stronger than coarse, not the least of which is the larger minor thread diameter (better shear strength) and more threads (improved tension load).

Fine threads also have less of a tendency to loosen because the thread incline is less steep. While fine threads are more easily tapped into hard materials and thin-walled tubes, coarse threads are better suited to softer materials, like aluminum and cast iron.

4 WHAT DO FASTENERS AND SPRINGS HAVE IN COMMON?

4. To achieve preload or clamping force, a fastener should be stretched a measured amount. A properly installed fastener works like a spring: The resulting "rebound" applies clamping force.

A typical 3/8-inch-diameter rod bolt made of 8740 chrome moly will need to be stretched about 0.006 inch to achieve a 10,000-pound clamping force. And, like a spring, if you don't pull it very far, there's little rebound; if you pull it too much, it will not return to its original length and shape–and will fail in service.



5 HOW DO YOU SELECT THE PROPER BOLT?

5. If you want to order the correct fastener for a given application, you'll need to consider at least eight dimensions. They include the (1) underhead length, (2) thread length, (3) grip length, (4) thread type, (5) grip diameter, (6) wrenching, (7) head height and (8) collar diameter You'll also need to decide whether you want standard hex or 12-point heads.

7 WHAT'S PRELOAD SCATTER AND HOW CAN YOU PREVENT IT?

7. When you apply torque to a fastener, much of the energy is expended on overcoming friction in the threads, any load-bearing surface (the underside of the bolt head or nut and washer against whatever surface it's tightened against) and, most important, the lubricant itself. (Note that moly, oil, diesel lube and the like all put up varying degrees of resistance).

As a result, the torque wrench may "click" at the desired setting–but it doesn't mean the desired preload has been achieved. This is called preload scatter. The difference between actual and desired preload can be dramatic–as much as 30 percent.

Now picture uneven preload placed on adjacent bolts or studs in a cylinder head. Result: distortion of the cylinder bore and a negative impact on piston ring seal.

For decades, the only proven method for assuring consistent preloading was to cycle the fastener–torque, loosen, re-torque–up to a half-dozen times to mitigate the friction. Now there's an easy way. Following extensive testing, ARP introduced their Ultra-Torque Fastener Assembly Lubricant, and they say it delivers 95 to 100 percent of the desired preload on the first–and any subsequent–pull of the torque wrench.

10 WHAT'S THE BEST METHOD FOR USING ANAEROBIC GLUES AND FASTENER LUBES?

10. For most high-preload applications like cylinder heads, main caps and connecting rods, fastener lube by itself is sufficient. If the bolt or stud is "wet"–meaning that it protrudes into a water passage–then you'll want to use a sealer.

And for some applications, like flywheels, clutches and ring gears, most racers prefer to use Loc-Tite or a similar anaerobic glue. If you wish to follow suit, the tech team at ARP strongly recommends the following method:

Sequentially secure all fasteners by torquing to the required level first, using oil as the lubricant. Next, remove and clean the first fastener and thread. Then, apply Loc-Tite and promptly re-torque the bolt before moving on to the next one.

Why this procedure? Because while a group of fasteners is tightened, the anaerobic glue sets up quickly and can start to harden before the desired torque is applied to the last ones–which then throws off everything.

Grassroots Motorsports
THE HARDCORE SPORTS CAR MAGAZINE

This article originally appeared, in its full form, in the May 2016 issue of Grassroots Motorsports

Grassroots Motorsports is more than just a magazine, it is a true multimedia publisher producing print magazines, digital and email content, as well as events to bring motorsport to your door.

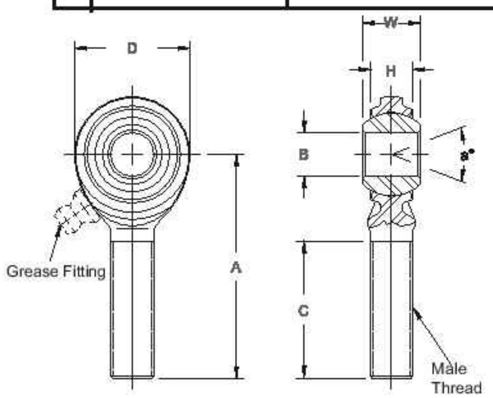
See: **http://grassrootsmotorsports.com/** to explore the world of Grassroots Motorsports.

Sources:
Automotive Racing Products, Inc.
arp-bolts.com
(800)826-3045

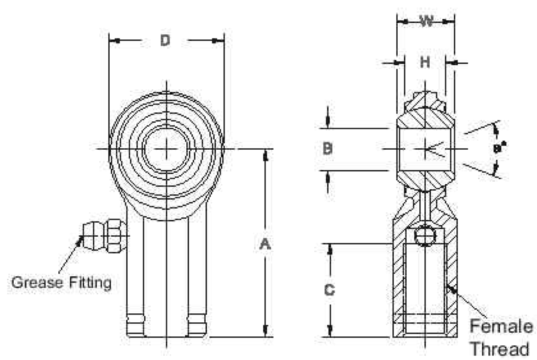
Terms Translations

A guide to translate common rod end industry terms from English to noted language. **This is a guide only.** Consultation with appropriate language refrences is recomended for best accuracy.

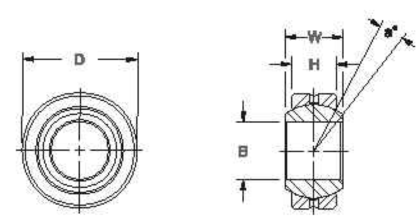
	English	French	German	Italian	Portugese	Spanish
	Aerospace	Aérospatial	Luftfahrt	Aerospaziale	Aeroespacial	Aeroespacial
	Aluminum	Aluminium	Aluminium	Alluminio	Aluminio	Aluminio
	Axial	Axiale	Axial	Assiali	Axial	Axial
	Ball	Bague interieure	Innenringe	Anello Interno	anel intero	Esfera
	Body	Corps	Gehaeuse	Testa	Corpo	cuerpo
A	Base to Center	base pour centrer longueur	Abstand, Unterseite zur Mitte	lunghezza, da base al centro	distância, base ao centro	altura, base a centro
B	Bore	Alesages	Bohrung	foro	Furo	agujero
	Commercial		Kommerziell	uso generico	Comercial	Comercial
	Chrome plate	chromage	verchromt	cromatura	Placa do cromo	cromado
D	Diameter	Diametre	Durchmesser	Diametro	Diametro	Diametro
	Female thread	Filetages Femelle	Innengewinde	Filettatura intera	Rosca Femea	rosca hembra
	Grease Fitting	Graisseur	Schmiernippel	Ingrassatore	lubrificador	lubricador
H	Head or Race Width	largeur de logement	Gehaeusebreite / Lagersschalenbreite	spessore testa / spessore boccola	largura externa do anel	anchura de la pista
	Industrial grade	Industriel	Industrie-Qualitaet	qualità industriale		Industrial
	Keyway	Rainure de Clavette	Keilnute	Scanalatura		cunero
	Left Hand	gauche	Links	sinistra	inversa	Rosca izquierda
	Load capacity	capacite de charges	belastbar	capacità di carico	Capacidade da carga	Capacidad de Carga
	Male thread	Filetages male	Aussengewinde	maschio filettato	Rosca macho	Rosca macho
	Military Specification	Spécifications Militaires	Militär-Vorschrift	Specifica Militare	Especificação Militar	Especificacion Militar
á	Misalignment	angulaires	Kippwinkel	ribaltamento	inclinacao	desalineamiento
	Race	Logement	Lagerschale	boccola / anello esterno	anel externo	Pista
	Radial	Radiale	Radial	Radiali	Radial	Radial
	Right Hand	Normes, Droite	Rechts	a destra	normal	Rosca Derecha
	Radial Static Load Capacity	charge statique	statische Tragzahl	Carico Statico	Capacidades de carga estatica	Carga radial maxima
	Rod end	Embouts	Gelenkkopf, pl. Gelenkkoepfe	Teste a Snodo	Terminais Rotulares	Cabezas de articulación
	Spherical bearing	Rotules	Gelenklager	Snodi Sferici	Rotulas	Rotula Esferica
	Stainless steel	Acier Inoxydable	Rostfrei	Acciaio inox	Aco inox	Acero inoxidable
	Steel	Acier	Stahl	Acciaio	Aco	Acero
	Thread	Filetage	Gewinde	Filettatura	Rosca	Rosca
C	Thread Length	longueur du filetage	Gewindelaenge	lunghezza del filetto	comprimento da rosca	Longitud de Rosca
W	Width of ball	largeur bille	Innenringbreite	spessore anello interno	largura interna do anel	anchura de la esfera



Male Rod End



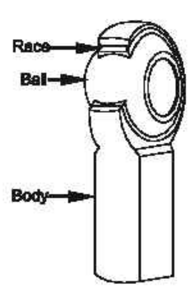
Female Rod End

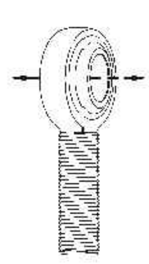


Spherical Bearing

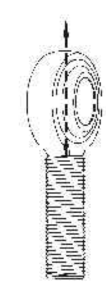
Left Hand Thread

Right Hand Thread





Axial



Radial

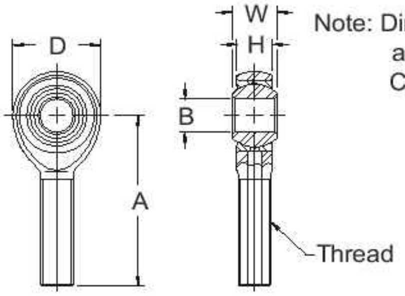
	English	Chinese	Japanese	Korean	Russian
	Aerospace	航天	航空宇宙	항공 우주	авиакосмический
	Aluminum	铝	アルミ	알루미늄	алюминий
	Axial	轴向	軸方向の	축 방향	осевой
	Ball	内圈	玉,内輪	공	шар
	Body	体	体	몸	тело
A	Base to Center	从底部的长度到中心	下から中心部までの長さ	거리,베이스 중심합니다	Расстояние от базы до центра
B	Bore	缸径	ボア	구멍 크기	внутренний диаметр
	Commercial	商业的	商業の	상업적으로	коммерческого класса
	Chrome plate	镀铬板	クロームプレート	크롬 플레이트	хромированная пластина
D	Diameter	轴承公称外径	径	직경	Диаметр
	Female thread	内螺纹	雌ねじ	내부 스레드	внутренняя резьба
	Grease Fitting	润滑脂	グリースニップル	윤활 니플	пресс-маслёнка
H	Head or Race Width	住房的宽度	ハウジングの幅	하우징의 폭	ширина беговой дорожки
	Industrial grade	产业素质	工業用グレード	산업 등급	промышленного класса
	Keyway	键槽	キー溝	키홈	шпоночный паз
	Left Hand	左手螺纹	左ねじ	왼손	левая резьба
	Load capacity	静载荷额定	負荷容量	부하 용량	Грузоподъемность
	Male thread	外螺纹	雄ねじ	외부 스레드	наружная резьба
	Military Specification	军事规格	ミリタリー仕様	군사 사양	Военная спецификация
á	Misalignment	对齐角度	ミスアライメント	오정렬각도	несоосность
	Race	种族	レース	레이스	беговая дорожка
	Radial	径向	ラジアル	방사형	радиальный
	Right Hand	右旋螺纹	右ねじ	오른손	Правая резьба
	Radial Static Load Capacity	径向静负荷能力	ラジアル負荷容量	레이디 얼 정적 부하 용량	Допустимая статическая нагрузка
	Rod end	杆端	ロッドエンド	로드 선단	наконечник
	Spherical bearing	关节轴承	球面軸受	구면 베어링	Сферические подшипники скольжения
	Stainless steel	不锈钢	ステンレス鋼	스테인리스 강	нержавеющая сталь
	Steel	钢	鋼	강철	сталь
	Thread	螺纹	ねじ山	나사산	резьба
C	Thread Length	螺纹长度	ねじ部長さ	나사 길이	Длина резьбы
W	Width of ball	球的宽度	内輪幅	공의 폭	сборочная ширина

	English	Arabic	Hindi
	Aerospace	الطيران	एयरोस्पेस
	Aluminum	الالمونيوم	अल्युमीनियम
	Axial	الاتجاه المحوري	अक्षीय
	Ball	الطوق الداخلي	बॉल
	Body	جسم	काया
A	Base to Center	المسافة من القاعدة إلى المركز	केंद्र के लिए आधार से दूरी
B	Bore	حفرة	छेद
	Commercial	تجاري	व्यावसायिक
	Chrome plate	طلاء الكروم	क्रोम प्लेट
D	Diameter	قطر	व्यास
	Female thread	المسطحات الداخلية	महिला धागा
	Grease Fitting	تركيب الشحوم	तेल निपल
H	Head or Race Width	العرض مسار أو السفن	आवास की चौड़ाई
	Industrial grade	الدرجة الصناعية	औद्योगिक श्रेणी
	Keyway	مفتاح مجرى	कुंजी स्लॉट
	Left Hand	اليد اليسرى	बायां हाथ
	Load capacity	الحد الأقصى للحمولة	लोड क्षमता
	Male thread	مترابطة خارجي	पुरुष धागा
	Military Specification	المواصفات العسكرية	सैन्य विनिर्देशों
	Misalignment	زاوية الاختلال	संरेखण कोण
	Race	الطوق الخارجي	दौड़
	Radial	شعاعي	रेडियल
	Right Hand	اليد اليمنى	दायाँ हाथ
	Radial Static Load Capacity	قدرة تحمل شعاعي ثابت	रेडियल स्थिर भार क्षमता
	Rod end	نهاية قضيب	रॉड समाप्त होता है
	Spherical bearing	محامل كروية	गोलाकार असर
	Stainless steel	الفولاذ المقاوم للصدأ	स्टेनलेस स्टील
	Steel	فولاذ	स्टील
	Thread	خيوط المسمار	धागा
C	Thread Length	طول خيط المسمار	धागा लंबाई
W	Width of ball	عرض الكرة	गेंद की चौड़ाई

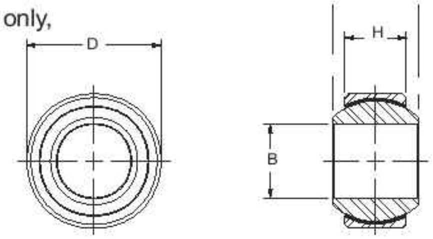
Special Size Race Car Parts

Parts commonly used in the race car industry, in non standard sizes and dimensions, along with common part number references. While not cataloged with standard parts series, many of these are stocked.
Check with dealer or factory for availability

Aurora part number	Common part number	B	H	W	D	A	Thread	Description/note
AM-4T-9	RC04	0.2500	0.281	0.375	0.7500	1.562	5/16-24	
AM-6T-6	2057P	0.3750	0.406	0.500	1.0000	1.938	1/2-20	
AM-6T-7		0.3750	0.500	0.625	1.3120	2.438	5/8-18	
AM-6T-15	1865P	0.3750	0.500	0.625	1.3120	2.438	1/2-20	
AM-12-2		0.7500	0.875	0.875	1.7500	2.875	3/4-16	Solid rod eye
AM-12T-22		0.7500	0.687	0.875	1.7500	2.875	3/4-16	Heat treated race
ATM-8T	MNR08U	0.5000	0.395	0.500	1.4590	2.544	1/2-20	
ATM-10T	MNR10U	0.6250	0.505	0.625	1.7630	2.832	5/8-18	
CM-12-12		0.7570	0.593	0.875	1.7500	2.875	3/4-16	
COM-8T-7	21CNZ081	0.5000	0.375	0.437	0.8750			Heat treated race
COM-8T-18	3087P	0.5000	0.400	0.750	1.1875			
COM-8T-20	1243P	0.5000	0.375	0.437	0.8750			Mild steel race
COM-8T-33	RBJ 205L	0.5000	0.3125	0.437	0.9375			HT race, PTFE lined
COM-8T-45		0.5000	0.390	1.000	1.0000			Drag Shocks
COM-10T-18	2280P	0.6250	0.630	0.875	1.3750			
COM-10T-32		0.6250	0.567	0.875	1.3750			
COM-12T-34		0.7500	0.593	0.750	1.4375			All Aluminum
CW-M2E-1		2.00 MM	2.24 MM	4.5MM	6.9 MM	12.45MM	M3X.5	
HAB-3T	ABYT-3	0.1900	.210	0.500	0.5625			HAB-3TG = ABYT-3V
HAB-5TG-3	ABYT-5V	0.3125	0.255	0.625	0.6875			
HAB-12T-11		0.7500	0.615	1.280	1.5625			
HCOM-24ET-8	AKW24V-4010	1.5000	1.500	1.960	2.9170			286,551 RSLC
HXAM-3T	ARYT-3	0.1900	.220	0.500	0.8060	1.562	5/16-24	
HXAM-6T-4		0.3750	0.355	0.813	1.1500	2.125	3/8-24	High Misalingment
HXAM-8T-8		0.5000	0.411	0.937	1.5250	2.625	1/2-20	High Misalingment
LCOM-8T	LS8, RS8	0.5000	0.531	0.687	1.3125			
LCOM-10T	LS10, RS10	0.6250	0.687	0.875	1.5625			
M-8B-FI-1		0.5000	0.390	0.625	1.0590			
MIB-4T-4		0.2500	0.281	0.375	0.6000			Mt bike shock
PRXM-5T-1		0.3125	0.327	0.437	1.0250	2.687	7/16-20	
PRXM-6T-4		0.3750	0.416	0.500	1.2500	2.750	1/2-20	
PRXM-7T-1		0.4375	0.452	0.562	1.2450	2.500	1/2-20	
PWB-8T-3	RWR08D/E	0.5000	0.500	0.750	1.1250			
PWB-12T-4		0.7500	0.625	0.750	1.3750			
RAM-8T-5		0.5000	0.687	0.750	1.7500	2.875	3/4-16	Ball dia 1.312
RXAM-8T-3		0.5000	0.562	0.750	1.7500	2.875	3/4-16	Ball dia. 1.125
RXAM-12T-1		0.7500	0.687	0.875	2.0000	3.375	7/8-14	
RAM-14T-1		0.8750	0.765	0.875	2.0000	3.375	7/8-14	
RXAM-14T-1		0.8750	0.765	0.875	2.3250	3.500	1-14	84,897 lb. RSLC
RAM-16T-1		1.0000	1.000	1.375	2.7750	4.125	1-14	100,643 lb. RSLC
RAM-16T-3		1.0000	1.000	1.375	2.7750	4.125	1 1/4-12	100,643 lb. RSLC
XAM-6T-1	1560P	0.3750	0.406	0.500	1.3120	2.438	1/2-20	
XAM-8T-2		0.5000	0.562	0.750	1.7500	2.875	3/4-16	
XAM-8T-5	MNR08U	0.5000	0.406	0.500	1.3120	2.438	1/2-20	
XAM-8T-10		0.5000	0.562	0.750	1.7500	2.625	5/8-18	
XAM-8T-11		0.5000	0.562	0.750	1.5000	2.625	5/8-18	
XCM-8		0.5000	0.453	0.625	1.5000	2.625	5/8-18	2 piece. Lined:VXCM-8



Note: Dimensions and descriptions are for reference only, and may not fully reflect part specifications. Check catalog or factory for related material specifications, tolerances, etc.

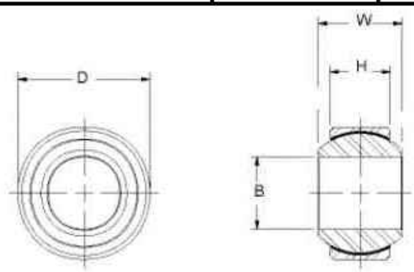


March 8 2016

Metric Race Car PTFE Lined Spherical Bearings

Parts commonly used in the race car industry, along with common part number references. While not cataloged, many of these parts are stocked. Check with dealer or factory for availability.

Aurora Part Number	Common Part Number	B	H	W	D
AIB-M12T-1	G-12D	12	12	16	26
GE12C	GE12E	12	7	10	22
COM-M14T-3		14	10	19	28
HAB-M14TG-1	MBYT-14V	14	10	23.5	29
GE15C	GE15ES	15	9	12	26
COM-M15T-4	GE15ES	15	9	12	26
COM-M15T-8	GEH15ES	15	10	16	30
AIB-M16T-1	G-16D	16	15	21	32
HAB-M16TG-1	MBYT-16V	16	14	30.5	35
GEG17ET-2RS	GEG17ES	17	12	20	35
AIB-M20T-1	G-20D	20	18	25	40
GEG20ET-2RS	GEG20ES	20	16	25	42
GEG25ET-2RS	GEG25ES	25	18	28	47



Note: Dimensions are for reference only, and may not fully reflect actual part specifications. Check catalog or factory for related series material specifications, tolerances, etc.

Fractured Race Bearings Interchange

Manufacturer	Inch	Metric	For seals, add	Inch,PTFE lined	Metric, PTFE lined	Metric PTFE lined
Aurora	GEZ *X* ES	GE *Y* ES	2RS	GEZ *X* ET-2RS	GE *Y* ET-2RS	GE *Y* C
Alinabal	CBB *X* B2					
Askubal		GE *Y*	2RS		GE *X* D2RS	GE *X* D
FAG		GE *Y* ES	2RS			GE *Y* DE.5
Fluro		GE *Y* E	2RS			
Hunger	GE *Y*		H-A		GE Y HW-A	
IKO	SBB *X*	GE *Y* ES	2RS		GE *X* EC-2RS	
INA/Elges	GE *W* ZO	GE Y DO	2RS		GE *X* UK-2RS	GE *X* UK
National	GEZ *W* ES	GE *Y* ES				
NMB	ABB *X* S	SBH *Y* S	ZZ			
NTN	SA2 *X* B	SA1 *Y* B	SS			
RBC	B *X* L	MB *Y*	SS	B *X* FSS	MB Y FSS	
SKF	GEZ *X* ES	GE *Y* ES	2RS	GEZ *X* TE-2RS	GE *X* TE-2RS	GE *X* C
THK		SA1 Y B	UU			
Torrington	*U* SF *X*					

U = Approximate bore in inches. (ex. 5 = .5", 15 = 1.5").
X = Bore in 1/16". (ex. 8 = 1/2", 1/2" = 8/16")
Y = Bore in mm. (ex. 20 = 20mm.)
W = Bore is nearest metric equivalent for inch size. (ex. 12 = .5", .5" = 12.7 mm.)
Table to be used as a reference guide. Ultimate interchangeability should be determined by comparing relevant manufacturers specifications.

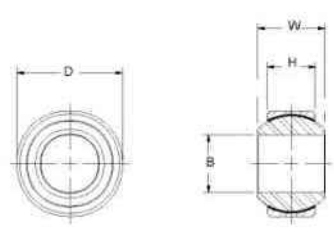
SHOCK ABSORBER BEARINGS

Bearings used in motorsports shock absorbers or dampers, both as standard, and for common special applications

Aurora Part #	Description/Application	B	D	W	H
COM-6T-20	52100 eared ball	0.3750	1.0000	0.9965	0.3900
COM-6T-24	7075 Aluminum eared ball	0.3750	1.0000	0.9965	0.3900
COM-6T-25	7075 Aluminum eared ball	0.3750	1.0000	0.8750	0.3900
GEZ008ES	Fractured race metal on metal	0.5000	0.8750	0.4370	0.3750
GEZ008C	GEZ008ES size "+1 quality" economy PTFE lined	0.5000	0.8750	0.4370	0.3750
COM-8T-20	GEZ008ES "+2", LC swaged race ptfe lined	0.5000	0.8750	0.4370	0.3750
COM-8T-7	Heat treated swaged race ptfe lined	0.5000	0.8750	0.4370	0.3750
COM-8		0.5000	1.0000	0.5000	0.3900
COM-8T		0.5000	1.0000	0.5000	0.3900
COM-8TKH	4130 HT race, AT1400 liner	0.5000	1.0000	0.5000	0.3900
PNB-8T	17-4 Heat treated race, AT3200 liner	0.5000	1.0000	0.5000	0.3900
COM-8-53	COM-8 with .625" W.	0.5000	1.0000	0.6250	0.3900
COM-8T-35	COM-8T with .625" W	0.5000	1.0000	0.6250	0.3900
COM-8T-45	Eared ball, common drag shock bearing	0.5000	1.0000	1.0000	0.3900
COM-8T-48	Eared ball	0.5000	1.0000	0.8750	0.3900
COM-8T-55	US "oval track" shocks	0.5000	1.0600	0.6250	0.3840
COM-8T-63	15MM. shocks to US Drag Race	0.5000	*26.0000*	1.0000	*9.0000*
COM-8T-64	15MM shock to COM-8T application	0.5000	*26.0000*	0.5000	*9.0000*
COM-8T-24	GE15 race dimension, 1/2" bore, .625""W"	0.5000	*26.0000*	0.6250	*9.0000*
COM-8TKH-9	COM-8T-24 Dim. Heat treated race	0.5000	*26.0000*	0.6250	*9.0000*
GE15ES	Metal on metal 15mm Fractured Race	15.0000	*26.0000*	*12.0000*	*9.0000*
GE15C	15mm PTFE lined	15.0000	*26.0000*	*12.0000*	*9.0000*
GE15ET-2RS	Fractured race, ptfe lined, seals	15.0000	*26.0000*	*12.0000*	*9.0000*
COM-M15T-4	Swaged L. C. race, AT1400 liner	15.0000	*26.0000*	*12.0000*	*9.0000*
COM-M15TKH-1	Swaged Heat treated race, AT1400 liner	15.0000	*26.0000*	*12.0000*	*9.0000*
COM-M15T-6	COM-8T w 15mm bore Convert COM-8 shock to 15mm	15.0000	1.0000	*12.0000*	0.3900
M-8B-FI-1	US oval track economy	0.5000	1.0600	0.6250	0.3900
M-10-FI-5	5/8 bore US oval track	0.6250	1.0600	0.6250	0.3900

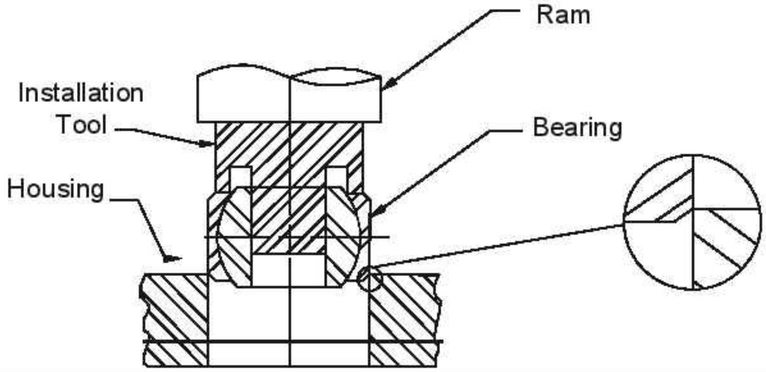
Dimensions nominal, in inches or *metric*. Consult factory for more complete dimensions.

These are more commonly used parts. Contact factory for interchange to parts not listed, or for custom specials



INSTALLATION OF SPHERICAL BEARING

FIGURE 1

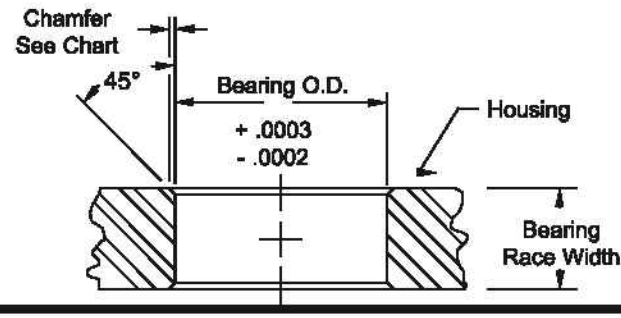


SPHERICAL BEARING INSTALLATION

Proper installation of the bearing is important to prevent bearing failure as well as housing damage. Under no circumstances should a tool that induces shock or impact to the bearing be used. The use of an arbor press or hydraulic press is recommended. A tool as shown above (Figure 1) is advised. All force is to be applied on the bearing race face (not on ball). A lead chamfer or radius on the bearing and/or housing is vital.

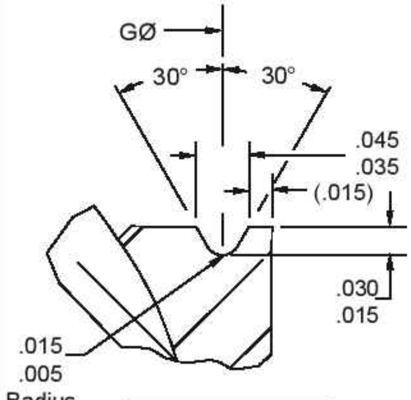
HOUSING CHAMFER - GROOVED BEARINGS

CHAMFER FOR GROOVE TYPES	
GROOVE TYPE A	.020 ± .005
GROOVE TYPE B	.030 ± .005
GROOVE TYPE C	.050 ± .005



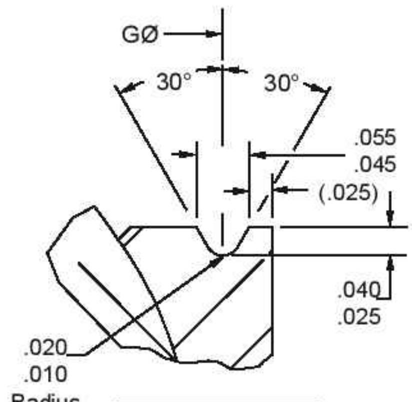
STAKING GROOVE TYPES

GROOVE TYPE A



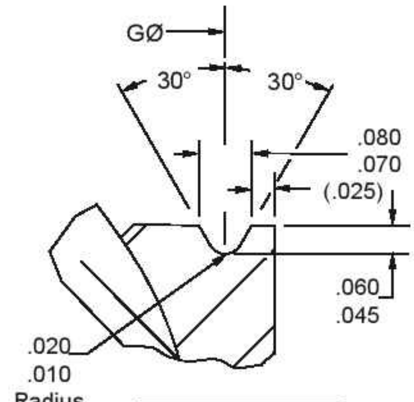
Groove Type A
PNB-3TG
PNB-4TG
PWB-3TG
PWB-4TG
PWB-5TG
HAB-3TG
HAB-4TG
HAB-5TG
HAB-6TG
HAB-7TG
HAB-8TG

GROOVE TYPE B



Groove Type B
PNB-5TG
PNB-6TG
PNB-7TG
PWB-6TG
PWB-7TG
PWB-8TG
PWB-9TG
PWB-10TG
HAB-10TG
HAB-12TG
HAB-14TG

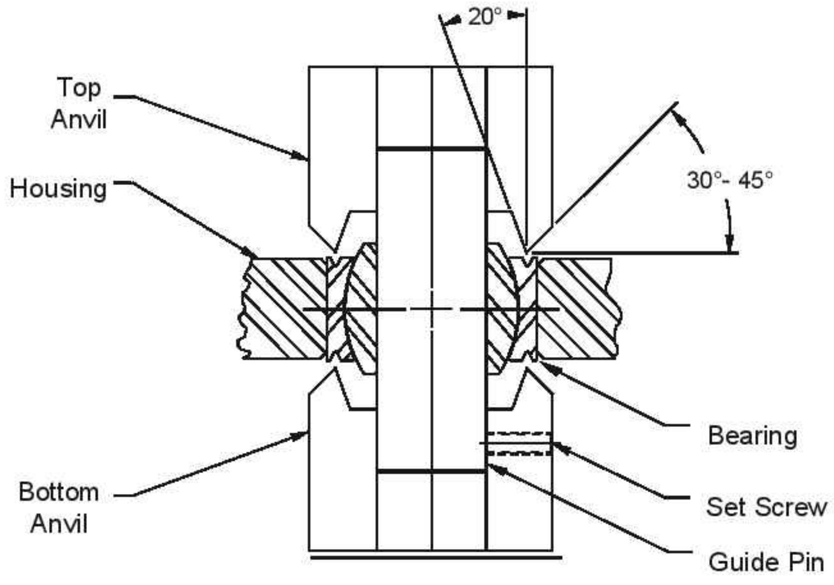
GROOVE TYPE C



Groove Type C
PNB-8TG
PNB-9TG
PNB-10TG
PNB-12TG
PWB-12TG

INSTALLATION OF SPHERICAL BEARING WITH STAKING GROOVES
TOOLS AND STAKING METHODS

FIGURE 2



SPHERICAL BEARING INSTALLATION

The bearings have grooves in each side of the bearing race face, leaving a small lip. Staking tools (as shown above in Figure 2) are then used to stake the lip over the chamfer edges of the housing. A typical arrangement consists of two identical anvils and one guide pin which is secured by a set screw in the bottom anvil.

PROCEDURES

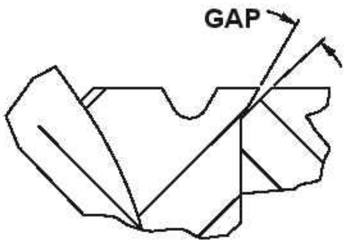
1. Install bearing into housing as shown in Figure 1 (pg. 34) and position bearing symmetrical about housing centerline.

2. Align bearing with staking tool and guide pin as shown in Figure 2.

3. A trial stake assembly should be made to determine staking force necessary to meet thrust load requirements. Proper staking force is required because excessive pressure could result in bearing distortion along with life.

4. Pressure established by trial assembly is to be applied. After first stake is completed rotate assembly 90° and re-apply. Repeat operation through a minimum of three rotations to insure 360° uniformity of stake.

5. After staking, a slight gap may exist between race lip and housing chamfer. This slight gap (shown below) may not be cause for rejection if bearing meets or exceeds thrust loads.



ENGINEERING INFORMATION

TAP DRILL SIZES FOR INCH THREADS			
Screw Thread		Commercial Tap Drills	
Thread size	Root Dia.	Size or Number	Decimal Equiv.
6-32	0.0834	36	0.1065
10-32	0.1469	22	0.1570
1/4-28	0.2036	3	0.2130
5/16-24	0.2584	I	0.2720
3/8-24	0.3209	Q	0.3320
7/16-20	0.3726	25/64	0.3906
1/2-13	0.4001	27/64	0.4219
1/2-20	0.4351	29/64	0.4531
5/8-11	0.5069	17/32	0.5312
5/8-18	0.5528	37/64	0.5781
3/4-16	0.6688	11/16	0.6875
7/8-14	0.7822	13/16	0.8125
1-12	0.8918	59/64	0.9219
1-14	0.9072	15/16	0.9375
1 1/4-12	1.1418	1 11/64	1.1719
1 1/2-12	1.3918	1 27/64	1.4219
1 3/4-12	1.6050	1 21/32	1.6563
2-12	1.8557	1 29/32	1.9063

TAP DRILL SIZE METRIC THREADS	
THREAD SIZE/TAP	METRIC DRILL SIZE
M3 X 0.5	2.50
M5 X 0.8	4.20
M6 X 1.0	5.00
M8 X 1.0	7.00
M8 X 1.25	6.80
M10 X 1.25	8.80
M10 X 1.5	8.50
M12 X 1.25	10.80
M12 X 1.75	10.20
M14 X 1.5	12.50
M14 X 2.0	12.00
M16 X 1.5	14.50
M16 X 2.0	14.00
M18 X 1.5	16.50
M20 X 1.5	18.50
M20 X 2.5	17.50
M22 X 1.5	20.50
M24 X 2.0	22.00
M30 X 2.0	28.0

These tables above are to be used as a guides only. Consult the appropriate reference to determine best size based on fit requirements, materials used, etc.

ALUMINUM DESIGNATION CROSS REFERENCE TABLE

USA	BRITAIN	EU	CHINA	GERMANY	ITALY	JAPAN
AA	B.S.	DIN 17007	GB	DIN 1700	UNI	JIS
2014	H15	3.1255	LD20	AlCuSiMn	"P-AlCu4,4SiMnMg"	A2014
2024	2L97/98	3.1355	LY12	AlCuMg2	"P-AlCu4,5MgMn"	A2024
6061	H20	3.3211	LD30	AlMg1SiCu	P-AlMg1SiCu	A6061
7075	2L95/96	3.4365	LC9	AlZnMgCu1.5	"P-AlZn5,8MgCu"	A7075

STEEL DESIGNATION CROSS REFERENCE TABLE

USA	BRITAIN	EU	CHINA	GERMANY	ITALY	JAPAN
AISI	B.S. 970	EN	GB	DIN	UNI	JIS
1015	040A15	32C	15	Ck15	C15	S15C
1018	080A15	32B	20Mn	C16.8	1C15	S18C
1022	120M19			20Mn5	G22Mn3	S20C
1045	080A47	43B	45	C45	C45	S45C
1144	212M44		Y40Mn	45S20	CF44SMn28	SUM43
1215	230M07	1B	Y13	9SMn36		SUM23
4130	708A30		25CrMo4	25CrMo4	25CrMo4	SCM420
4140	708M40	19	42CrMo	41CrMo4	41CrMo4	SCM440
4340	817M40	24	40CrNiMoA	34CrNiMo6	35NiCrMo6 KB	SNCM447
52100	534A99	31	GCr15	100Cr6	100Cr6	SUJ2
303	303S21	X8CrNiS18-9	Y1Cr18Ni9	X10CrNiS18-9	X10CrNiS 18 09	SUS303
410	410S21	56A	1Cr13	X15Cr13		SUS410
440C	A-1b	9Cr18	X105CrMo17		SUS440C	
17-4				X5CrNiCuNb1714		SUS80

These tables are to be used as a guides to assist in finding comparable metal designations only. True interchange can be determined only by comparing chemical composition, mechanical properties, and manufacturing technologies.

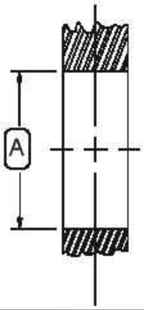
Most commonly used conversions in **Bold**.

INCH/METRIC CONVERSION TABLE

INCH		MM.	INCH		MM.	INCH		MM.	INCH		MM.
FRACT.	DEC.		FRACT.	DEC.		FRACT.	DEC.		FRACT.	DEC.	
	0.00004	0.001	17/64	0.2656	6.746		0.6693	17.0		1.3780	35.0
	0.00039	0.01		0.2756	7.0	43/64	0.6719	17.066		1.4173	36.0
	0.0010	0.025	9/32	0.2812	7.1437	11/16	0.6875	17.4625	1 1/2	1.5000	38.1
	0.0020	0.051	19/64	0.2969	7.5406	45/64	0.7031	17.859		1.5354	39.0
	0.0030	0.0762	**5/16**	**0.3125**	**7.9375**		0.7086	18.0		1.5748	40.0
	0.00394	0.1		0.3150	8.0	23/32	0.7187	18.256		1.6535	42.0
	0.0050	0.1270	21/64	0.3281	8.334	47/64	0.7334	18.653	1 3/4	1.7500	44.45
	0.00984	0.25	11/32	0.3437	8.731		0.7480	19.0		1.7717	45.0
	0.0100	0.254		0.3543	9.0	**3/4**	**0.7500**	**19.05**		1.8898	48.0
1/64	0.0156	0.396	23/64	0.3594	9.1281	49/64	0.7656	19.446		1.9685	50.0
1/32	0.0312	0.793	**3/8**	**0.3750**	**9.525**	25/32	0.7815	19.843	2	2.0000	50.8
	0.03937	1.0	25/64	0.3906	9.9219		0.7874	20.0		2.0472	52.0
3/64	0.0469	1.191		0.3937	10.0	51/64	0.7969	20.240		2.1654	55.0
	0.0591	1.5	13/32	0.4062	10.318	13/16	0.8125	20.6375		2.2047	56.0
1/16	0.0625	1.5875	27/64	0.4219	10.716		0.8268	21.0	2 1/4	2.2500	57.15
5/64	0.0781	1.984		0.4331	11.0	53/64	0.8281	21.034		2.3622	60.0
	0.0787	2.0	**7/16**	**0.4375**	**11.1125**	27/32	0.8437	21.431	2 1/2	2.5000	63.5
3/32	0.0937	2.381	29/64	0.4531	11.509	55/64	0.8594	21.828		2.5197	64.0
	0.0984	2.5	15/32	0.4687	11.906		0.8661	22.0	2 3/4	2.7500	69.85
	0.1000	2.54		0.4724	12.0	**7/8**	**0.8750**	**22.225**		2.8346	72.0
7/64	0.1094	2.778	31/64	0.4844	12.303	57/64	0.8906	22.621		2.9528	75.0
	0.1181	3.0	**1/2**	**0.5000**	**12.7**		0.9055	23.0	3	3.0000	76.2
1/8	**0.1250**	**3.175**		0.5118	13.0	29/32	0.9062	23.018		3.1496	80.0
	0.1387	3.5	33/64	0.5156	13.096	59/64	0.9219	23.416	3 1/4	3.2500	82.55
9/64	0.1406	3.571	17/32	0.5312	13.493	15/16	0.9375	23.8125	3 1/2	3.5000	88.9
5/32	0.1562	3.968	35/64	0.5469	13.891		0.9449	24.0		3.5433	90.0
	0.1575	4.0		0.5512	14.0	61/64	0.9531	24.209	3 3/4	3.7500	95.25
11/64	0.1719	4.366	**9/16**	**0.5625**	**14.2875**	31/32	0.9687	24.606		3.9370	100.0
	0.1772	4.5	37/64	0.5781	14.684		0.9843	25.0	4	4.0000	101.6
3/16	**0.1875**	**4.7625**		0.5906	15.0	63/64	0.9844	25.003	4 1/4	4.2500	107.95
	0.1969	5.0	19/32	0.5937	15.081	**1**	**1.0000**	**25.4**		4.3307	110.0
13/64	0.2031	5.159	39/64	0.6094	15.478		1.0630	27.0	4 1/2	4.5000	114.3
7/32	0.2187	5.556	**5/8**	**0.6250**	**15.875**		1.1024	28.0		4.7244	120.0
15/64	0.2334	5.953		0.6299	16.0		1.1811	30.0	4 3/4	4.7500	120.65
	0.2362	6.0	41/64	0.6406	16.271	1 1/4	1.2500	31.75	5	5.0000	127.0
1/4	**0.2500**	**6.35**	21/32	0.6562	16.668		1.2992	33.0	5 1/2	5.5000	139.7

INCH/METRIC CONVERSION FACTORS

Inches	x 25.4	= Millimeters	Pounds	x .4536	= Kilograms	Pounds(Force)	x 4.448	=Newtons
Millimeters	x .03937	= Inches	Kilograms	x 2.2046	= Pounds	Newtons	x .2248	=Pounds(Force)
Sq. Inches	x 6.4515	= Sq. Centimeters	Lbs. per In.^{2}	x .0703	= kg per cm^2	Temperature Conversion (Approximate)		
Sq. Centimeters	x .155	= Sq. Inches	kg per cm^2	x 14.2231	= Lbs. per In.^{2}	Degrees C = (Degrees F -32)(.5556)		
						Degrees F = (Degrees C)(1.8)+ 32		



Suggested Housing Bores for Aurora Bearing's COM, HCOM, MIB, AIB & SIB Spherical Bearings

COM & HCOM Series

Bearing Series COM HCOM	Bearing Outside Diameter +.0000 -.0007	Steel Housing Max.	Steel Housing Min.	Aluminum Housing Max.	Aluminum Housing Min.
	DIMENSIONS IN INCHES — (A) Suggested Housing Bore For Press Fit of Spherical Bearings				
3	.5625	.5619	.5614	.5618	.5612
4	.6562	.6556	.6551	.6555	.6549
5	.7500	.7494	.7489	.7493	.7487
6	.8125	.8119	.8114	.8118	.8112
7	.9062	.9056	.9051	.9055	.9049
8	1.0000	.9994	.9989	.9993	.9987
9	1.0937	1.0931	1.0925	1.0930	1.0923
10	1.1875	1.1869	1.1863	1.1868	1.1861
12	1.4375	1.4369	1.4363	1.4368	1.4361
14	1.5625	1.5619	1.5613	1.5618	1.5611
16	1.7500	1.7494	1.7486	1.7493	1.7485
16	2.0000	1.9994	1.9986	1.9993	1.9985
19	2.3750	2.3744	2.3736	2.3743	2.3735
20	2.3750	2.3744	2.3736	2.3743	2.3735
24	2.7500	2.7494	2.7486	2.7493	2.7485
28	3.1250	3.1244	3.1236	3.1243	3.1235
32	3.5000	3.4994	3.4986	3.4993	3.4985

Bearing Series COM HCOM	Bearing Outside Diameter +.000 -.018	Steel Housing Max.	Steel Housing Min.	Aluminum Housing Max.	Aluminum Housing Min.
	DIMENSIONS IN MILLIMETERS — (A) Suggested Housing Bore For Press Fit of Spherical Bearings				
3	14.288	14.272	14.260	14.270	14.255
4	16.667	16.652	16.640	16.650	16.634
5	19.050	19.035	19.022	19.032	19.017
6	20.638	20.622	20.610	20.620	20.604
7	23.018	23.002	22.990	23.000	22.985
8	25.400	25.385	25.372	25.382	25.367
9	27.780	27.765	27.750	27.762	27.744
10	30.162	30.147	30.132	30.145	30.127
12	36.512	36.497	36.482	36.495	36.477
14	39.688	39.672	39.657	39.670	39.652
16	44.450	44.435	44.414	44.432	44.412
16	50.800	50.785	50.764	50.782	50.762
19	60.325	60.310	60.289	60.307	60.287
20	60.325	60.310	60.289	60.307	60.287
24	69.850	69.835	69.814	69.832	69.812
28	79.375	79.360	79.339	79.357	79.337
32	88.900	88.885	88.864	88.882	88.862

Dimensions given in the above tables are for bearings fabricated of standard race materials. Should other materials be used, consult our engineering department for modification of these dimensions.

RADIAL STATIC LOAD CAPACITY

These loads are maximum static based on maximum permanent set in the bearing race of 0.2% of the ball diameter. If a greater permanent set can be allowed or if alternate race materials are used consult our engineering department for change factors.

AXIAL STATIC LOAD CAPACITY

These loads are approximately 20% of the radial loads listed when the load bearing surfaces are properly supported.

ALTERNATE RACE AND BALL MATERIALS

Materials other than those listed can be incorporated in bearings manufactured by Aurora Bearing Company. Stainless steels to improve corrosion resistance, heat treated alloy steels to increase wear life are frequently used in special applications.

PTFE lined races are also available in this series to provide service requiring no relubrication and improved frictional characteristics. **Tables are representative of Aurora Bearing's Metal to Metal parts, please consult our engineering department regarding PTFE lined parts.**

MIB, AIB & SIB Series

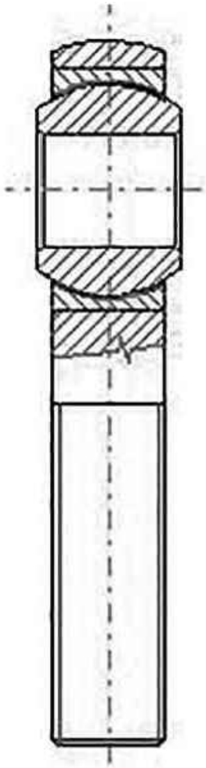
Bearing Series MIB AIB SIB	Bearing Outside Diameter +.0000 -.0007	Steel Housing Max.	Steel Housing Min.	Aluminum Housing Max.	Aluminum Housing Min.
	DIMENSIONS IN INCHES — (A) Suggested Housing Bore For Press Fit of Spherical Bearings				
3	.5312	.5306	.5301	.5305	.5299
4	.6094	.6088	.6083	.6087	.6081
5	.7500	.7494	.7489	.7493	.7487
6	.8437	.8431	.8426	.8430	.8424
7	1.0000	.9994	.9989	.9993	.9987
8	1.0937	1.0931	1.0925	1.0930	1.0923
10	1.3125	1.3119	1.3113	1.3118	1.3111
12	1.5000	1.4994	1.4988	1.4993	1.4986
14	1.6250	1.6244	1.6236	1.6243	1.6235
16	2.1250	2.1244	2.1236	2.1243	2.1235

Bearing Series MIB AIB SIB	Bearing Outside Diameter +.000 -.018	Steel Housing Max.	Steel Housing Min.	Aluminum Housing Max.	Aluminum Housing Min.
	DIMENSIONS IN MILLIMETERS — (A) Suggested Housing Bore For Press Fit of Spherical Bearings				
3	13.492	13.477	13.465	13.475	13.460
4	15.479	15.464	15.451	15.461	15.446
5	19.050	19.035	19.022	19.032	19.017
6	21.430	21.415	21.402	21.412	21.397
7	25.400	25.385	25.372	25.382	25.367
8	27.780	27.765	27.750	27.762	27.744
10	33.338	33.322	33.307	33.320	33.302
12	38.100	38.085	38.070	38.082	38.064
14	41.275	41.259	41.239	41.257	41.237
16	53.975	53.960	53.939	53.957	53.937

Notes

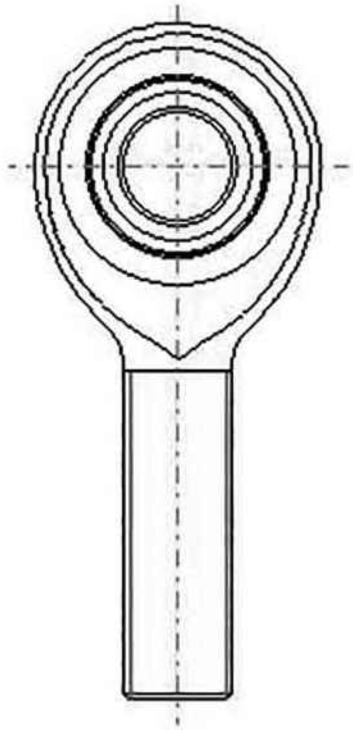
Available Exclusively from The *Motion - Transfer* Specialists: 2D and 3D CAD Drawings of Aurora Bearing Rod End and Spherical Bearings

Aurora Bearing has developed a CAD drawing library of it's entire catalog offering of Rod End and Spherical Bearings, including Mil.spec approved parts. These 2D and 3D images are importable into most major CAD and solid modeling packages, and includes both line drawings and 3D models. Files are accessible at www.aurorabearing.com, or the entire program and library is available on a CD. This CD also includes the Aurora Bearing commercial and aviation products catalogs, and is available free upon request.

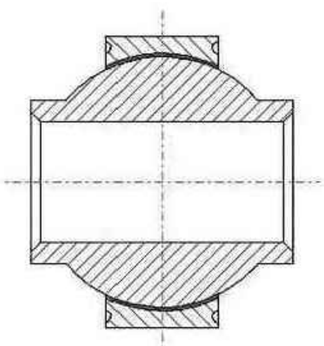


Cross section, MM-8T

3D model, MM-8T

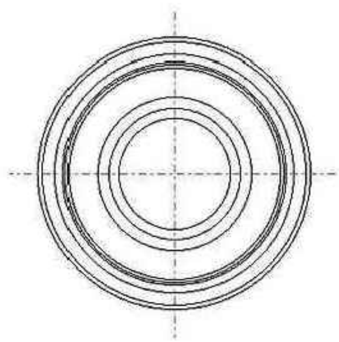


Front view, MM-8T



Cross section, HAB-6TG

3D model, HAB-6TG



Front View, HAB-6TG

AURORA BEARING COMPANY

Geiser Brothers: www.geiserbros.com

Scott Bloomquist: www.bloomquistracing.net

Danny Thompson: www.thompsonlsr.com

John Lawson: John Lawson roadster "Lash"

Trey Hutchens: www.treyhutchensracing.com

"Blowfish": www.radrides.com

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