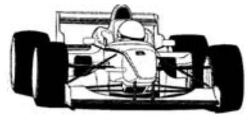
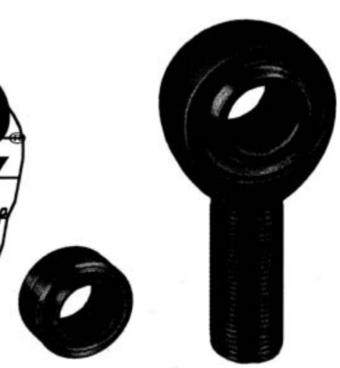




PRESENTS

A,B,C's OF SPHERICAL ROD ENDS

The traditional ball and roller bearings used by industry are anti-friction bearings designed to reduce friction and provide support in all kinds of rotating assemblies. Aurora industrial rod end and spherical bearings, in contrast, are friction-type bearings designed to provide more precise control, greater reliability, and greater wear resistance in oscillating assemblies. They are used in some slowly rotating applications, but their design characteristics are best utilized in back-and-forth, oscillating movements. Aurora rod end bearings are self-aligning. They have the ability to *oscillate radially* and *misalign axially*, qualities which permit sophisticated mechanisms to operate efficiently under conditions of extreme cycling, vibration, stress, load, speed and temperature.

NOMENCLATURE

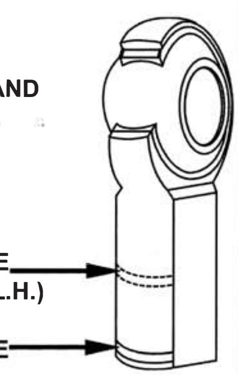
C=COMMERCIAL SERIES
M=MACHINE SERIES
A=ALLOY SERIES

M=MALE THREAD R.H.
B=MALE THREAD L.H.
W=FEMALE THREAD R.H.
G=FEMALE THREAD L.H.

ADULTS=RIGHT HAND
KIDS=LEFT HAND

Nomenclature of Aurora rod end bearings is relatively simple to understand. The first letter identifies the construction and material used in the bearing. There is a wide variety of standard Aurora rod end series. The three most widely used series are C, Commercial Grade, least expensive, M, Machine Grade, medium-priced, and A, Alloy Grade, Premium-priced materials. There are many other standard series as well; all are construction and material variations of the first three series. The HB series is a specially designed ball bearing unit.

The second letter in Aurora rod end nomenclature indicates the type of threads. The letter M, or Man, means Male external right-hand threads. B, or Boy, means Male external left-hand threads. W, or Woman, means Female internal right-hand threads. G, or girl, means Female internal left-hand threads. Just remember that the adults, M, Man and W, Woman are always right and that the kids, B, Boy and G, Girl are always left.



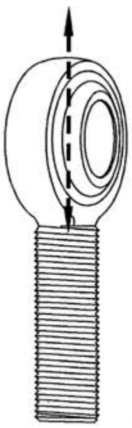
Right-hand and left-hand indicate the slant of the threads. Male right-hand threads slant upward to the right, left-hand threads upward to the left. Aurora CG and MG Series female rod ends; which have internal left-hand threads, have a circular groove near the bottom of the shank to differentiate them from the CW and MW Series female rod ends with internal right-hand threads.

The number after the grade and thread letters in Aurora rod end nomenclature denotes the bore diameter in sixteenths of an inch. Number 3, for example, means three-sixteenths inch; number 20 is twenty-sixteenths or an inch and a quarter. Metric bore diameters are in millimeters. Metric female rod ends have a circular groove in the middle of the shank to differentiate them from inch-series female rod ends.

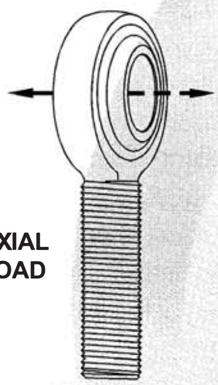
NUMBERS = 16TH OF INCH
3 = 3/16”
20 = 1-1/4”

SUFFIXES

Z = GREASE ZERK
S = THREADED STUD
T = TEFLON LINER



RADIAL LOAD



AXIAL LOAD

ROD END RADIAL STATIC LOAD CAPACITY = LOAD TO BREAK BODY

<u>RECOMMENDED</u> RADIAL OPERATING LOAD = 35% RADIAL STATIC RATING

<u>RECOMMENDED</u> AXIAL STATIC LOAD:
2 PIECE ROD END 15%
3 PIECE ROD END 10%
OF RADIAL STATIC RATING

SPHERICAL BEARING LOADS

<u>RECOMMENDED</u> RADIAL OPERATING LOAD: 40% OF RADIAL STATIC RATING

<u>RECOMMENDED</u> AXIAL OPERATING LOAD: 20% OF RADIAL STATIC RATING

STANDARD BORE TOLERANCES
C = +.0025 -.0005 M = +.0015 -.0005
SPECIAL BORE TOLERANCES
+.0000
-.0005

Standard accessories and modifications are identified by suffixes after the bore number. The letter T indicates the steel race has a PTFE liner, a woven fabric chemically bonded to the inner diameter of the race. A PTFE liner provides permanent lubrication and tight internal fit of the ball to the race, sometimes known as zero backlash or preloaded fit. Also available is a composite PTFE liner in our economical V series, designated by the prefix V. Aurora Bearing also produces a PTFE liner fully qualified to MIL-B-81820.

The letter Z means the rod end has a Zerk-type grease fitting, which is inserted in the shank of female bearings, or in the head of male bearings. F means a Flush-type grease fitting has been inserted in the bearing. The letter S indicates a right-hand threaded stud is affixed to the ball, a feature which provides easier installation for the user and permits greater misalignment of the bearing in a small space.

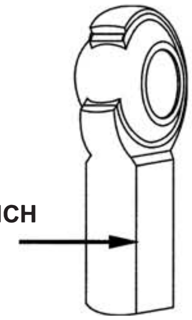
The Operating Load Capacity of Aurora rod ends is based on Ultimate Radial Static Load Rating, in simple terms, the number of pounds of pull necessary to break the rod end body. Operating Load is recommended at 35 percent of Ultimate Radial Static Load for slowly oscillating applications, 10 percent of Ultimate Radial Static Load for full rotation assemblies up to 100 rpm. Axial Static Load Capacity is recommended at 15 percent of Ultimate Radial Static Load for Aurora two-piece rod ends, 10 percent for three-piece rod ends. For PTFE-lined, heavy-duty shank units and studded units, consult with Aurora Engineering Department.

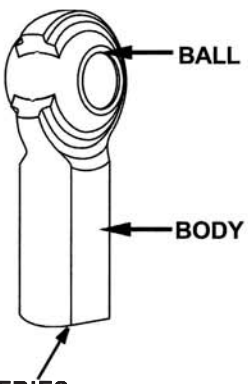
A rod end bearing's ability to misalign is measured by the degree of angle the ball can accommodate without interference. The misalignment angle is dependent upon the shaft-mounting procedure and the ball and body width.

Aurora makes many standard spherical bearing series, such as COM, meaning Commercial, HCOM, Heavy Duty Commercial, MIB, Machine Insert Bearing and AIB, Alloy Insert Bearing. They are identified by the letters and bore numbers. PTFE liners are available on all spherical bearings.

Maximum Radial Static Load Capacity of Aurora spherical bearings is based on a permanent set in the race of two-tenths of 1 percent of the ball diameter. Axial Load is recommended at 20 percent of the Radial Static Load, when the load-bearing surfaces are properly contained by the housing. Operating Load is recommended at 40 percent of Radial Static Load.

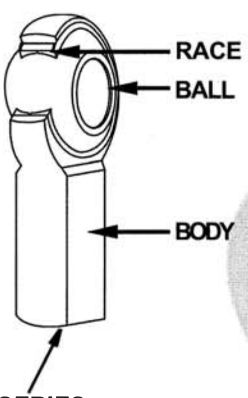
All production stages at Aurora Bearing Company undergo strict quality control inspection, from incoming materials through processing to final shipment. Our quality system is based on a simple philosophy - Do It Right The First Time. Today, we use many tools such as SPC control charting, parts analysis, flow charting, statistical sampling, gage variation studies, team problem solving, vendor quality rating, and many more.





C-SERIES

2 PIECE CONSTRUCTION
3/16 TO 3/4 INCH BORES



/I-SERIES

PIECE CONSTRUCTION
/8 TO 2 INCH BORES

-SERIES

PIECE CONSTRUCTION
LL ALLOY MATERIAL
/16 TO 2 INCH BORES

The manufacturing of Aurora Bearings starts with the balls. Aurora C Series rod ends, which use the body as the race, have sintered steel balls. Aurora sintered steel balls have identifying lines on the face, or ends, of the ball. Other Series rod ends have solid steel balls that are through hardened. The balls are made to a standard bore tolerance of three-thousandths of an inch for C Series rod ends, two-thousandths of an inch for other standard bearings. Bore tolerances can be maintained up to five ten-thousandths of an inch for special user requirements.

Aurora races are produced on screw machines, then drilled for oil or grease lubrication, tumbled, plated for corrosion resistance, and dipped in chromate for added protection.

Aurora rod end bodies are under careful quality control beginning with the screw machine operation, and continuing through every stage of construction.Aurora rod end bodies are plated, and dipped in chromate or dichromate. The plating and dipping of bodies and races is a standard Aurora construction feature that increases corrosion protection 80 to 100 percent over regularly-plated bearings. Special plating is applied on S Series rod ends for use in excessively corrosive atmospheres. All Aurora rod ends have precision UNF, UNJF, or ISO metric threads. Female rod ends are machined to incorporate full-length wrench flats for easier assembly by the user. Aurora uses only full swage-type assembly, a procedure which assures conformance of parts and careful compliance with specifications. The ball is always precision fit after assembly to provide closely controlled radial clearance.

Aurora C Series economy rod ends are of a two-piece design consisting of a low carbon steel, 75,000 PSI body and a heat-treated, sintered steel ball, with the body serving as the race. By eliminating a separate race, a smaller bore is required in the rod head, providing greater head strength. The ball is inserted in the body, which is swaged around the ball. C Series bearings offer precision service at lower cost for operations requiring strong, economical bearings. They are available in eight bore sizes from 3 to 12, male and female.

Aurora M Series rod ends have a three-piece construction for applications requiring high precision and wear resistance. A one piece, low carbon steel race is swaged around a through-hardened steel ball, forming a spherical bearing which is then staked into a low carbon steel 75,000 PSI body. The standard bore sizes range from 2 to 32, male and female. Male sizes 6 through 12 are available with a lubrication hole up the center of the shank as an optional feature. If a solid shank is required, use the suffix Y.

Aurora A Series rod ends are also of a three-piece construction. They have a heat-treated alloy steel, one-piece race swaged around a through hardened alloy steel ball. This is staked in an alloy steel 175,000 PSI body. The A Series bearings are for applications requiring the ultimate in strength, precision, and wear resistance. They are available in standard bore sizes from 3 to 32, male and female. Both M and A Series rod ends frequently are used to replace other makes of bearings which have failed to maintain specified work requirements.

"-T" SUFFIX

3 PIECE CONSTRUCTION
3/16 TO 2 INCH BORES

Aurora "-T" Suffix rod ends have three-piece construction with a PTFE-lined race, the finest in pre-lubed, friction-type bearings. They are offered in bore sizes 3 through 32, male and female, and are available in all three-piece rod end variants.

K-SERIES

3 PIECE CONSTRUCTION
ALLOY HEAT TREAT RACE
3/16 TO 2 INCH BORES

K Series rod ends have the M Series ball and body with an A Series heat-treated alloy steel race. They are especially reliable for jobs requiring greater wear resistance under high vibration or high frequency reversing loads. K Series rod ends are available in bore sizes from 3 to 32, male and female.

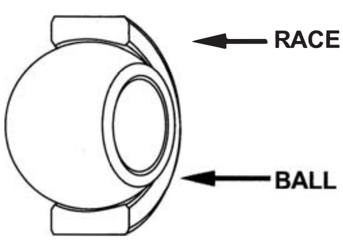
X-SERIES

3 PIECE CONSTRUCTION
HEAVY DUTY
MALE THREADS
1/8 TO 1-1/2 INCH BORES

Aurora X Series rod ends feature an extra heavy duty shank and a larger thread diameter to provide greater load carrying capacity. They are available only with male external threads in bore sizes from 2 through 24, and either low-carbon, or alloy steel bodies.

Where corrosion is a problem, Aurora offers stainless steel parts in both two and three piece designs, with or without PTFE liners. They are available in most bore sizes, male and female. Aurora HB Series rod ends are self-aligning ball bearings with ductile iron housings. They are designed for high-rotation operations and are available in three bore sizes: 8, 10, and 12.

SPHERICAL BEARINGS



COM-SERIES
HCOM-SERIES
MIB-SERIES
AIB-SERIES
SIB-SERIES

1/8 TO 2 INCH BORES

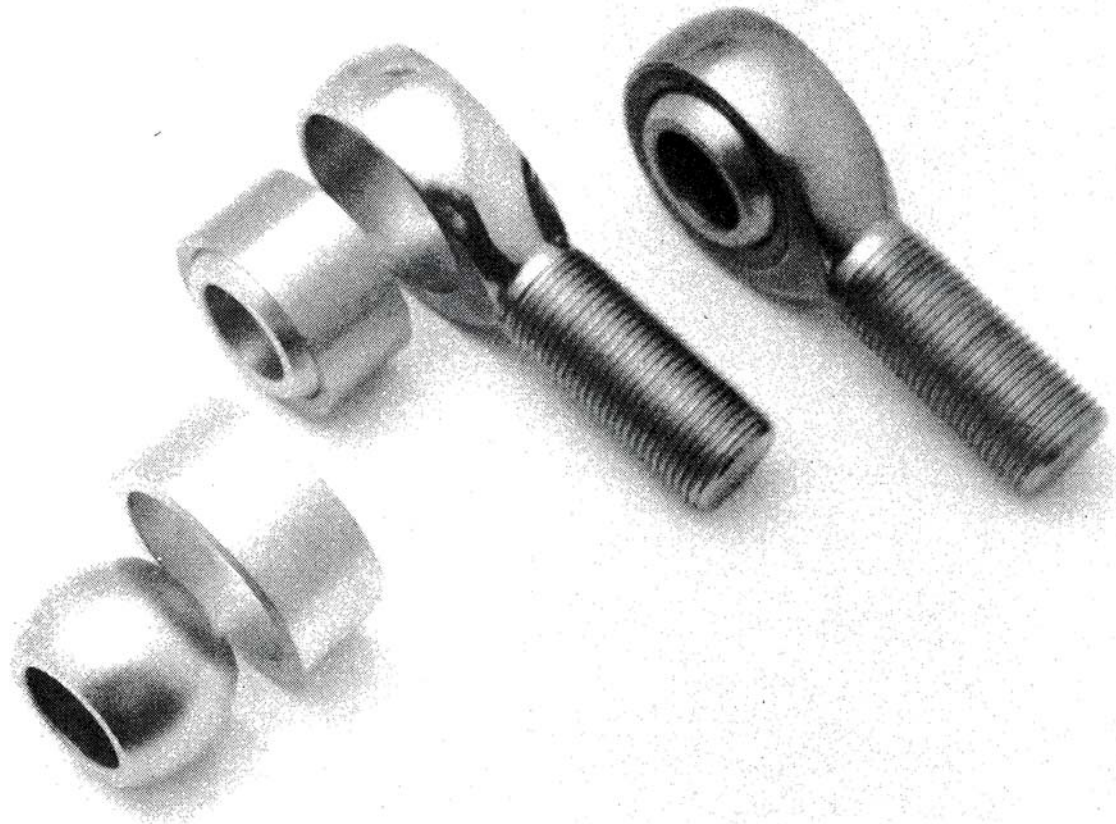
The many Aurora spherical bearing series, made for insertion into user housings, have a one-piece, oil-coated, steel race swaged around a through-hardened ball. The spherical bearings are available in standard bore sizes from 2 to 32. Larger sizes can be made to order. They may be ordered with or without PTFE liners or heat-treated alloy, or stainless steel races.

Aurora Bearing Company has developed and received approval of a PTFE Liner System to MIL-B-81820, spherical bearings to MS-14101 through MS-14104, rod ends to MIL-B-81935, and PTFE lined bushings to MIL-B-81934.

AURORA®

Virtually everywhere you look, there are products which use Aurora bearings, or which would function more precisely and durably if they were using Aurora bearings! Aurora is highly experienced in designing and manufacturing rod end and spherical bearings for highly unique requirements in new and redesigned equipment. The variety of unusual bearings made by Aurora includes rod ends that are bent to clear other components, bearings with special materials such as high nickel cobalt steel, to withstand turbine temperatures in excess of 14 hundred degrees Fahrenheit! Bearings in virtually any shape, size, and material, according to the dictates of customer engineering needs! Whenever precision industrial rod end and spherical bearings are involved, you should call on Aurora Bearing Company. The people who devote all their time to precision rod end and spherical bearings. The Motion-Transfer Specialists.

Rod Ends



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

PHOTOS BY BOB PESHIA

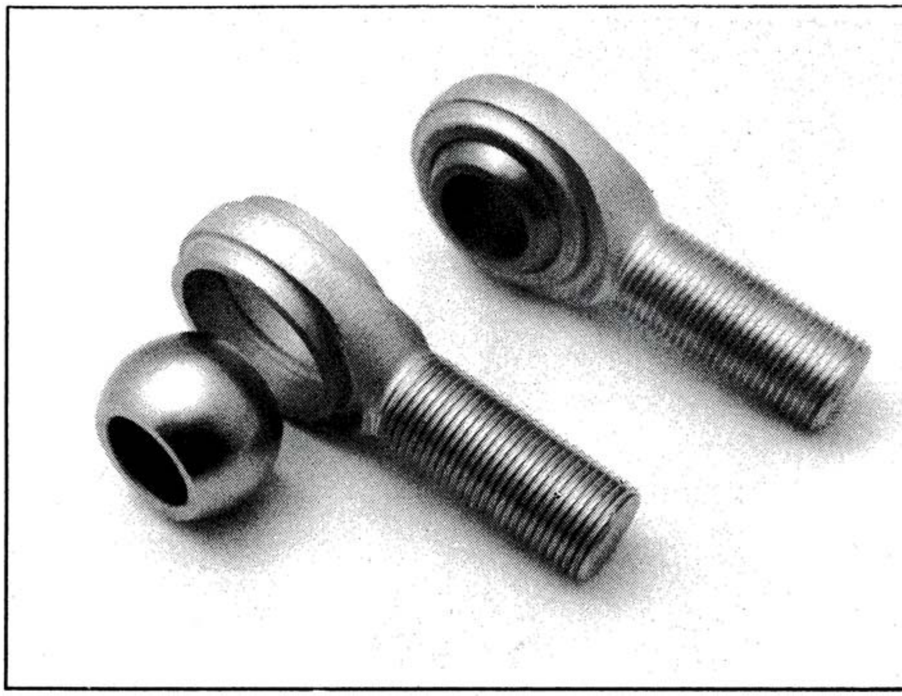
Rod end bearings are an important item that the average racer often takes for granted. Too often they incorrectly assume that one rod end is the same as another. Yet those critical components are what connects the suspension to the frame or connects the wheels to the steering. If your car has the correct 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.

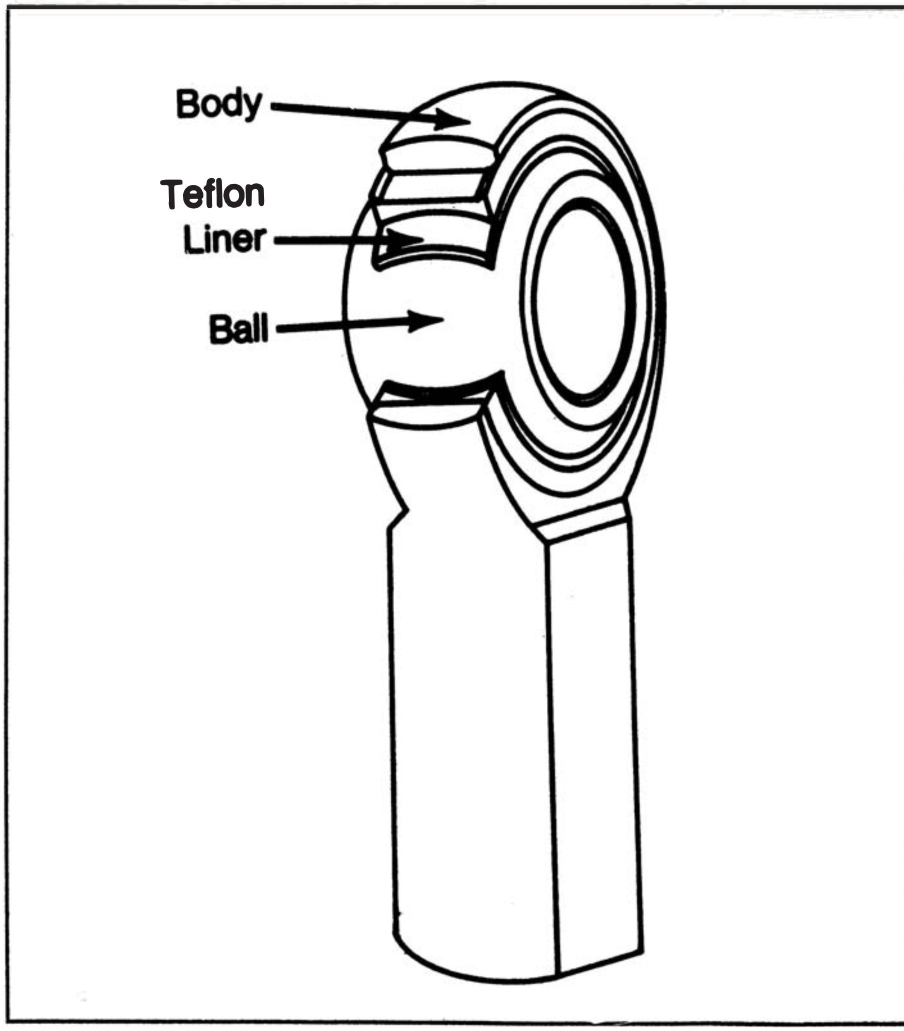


Finished ball with finished body of two piece full-swage rod end prior to assembly and finished unit.

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

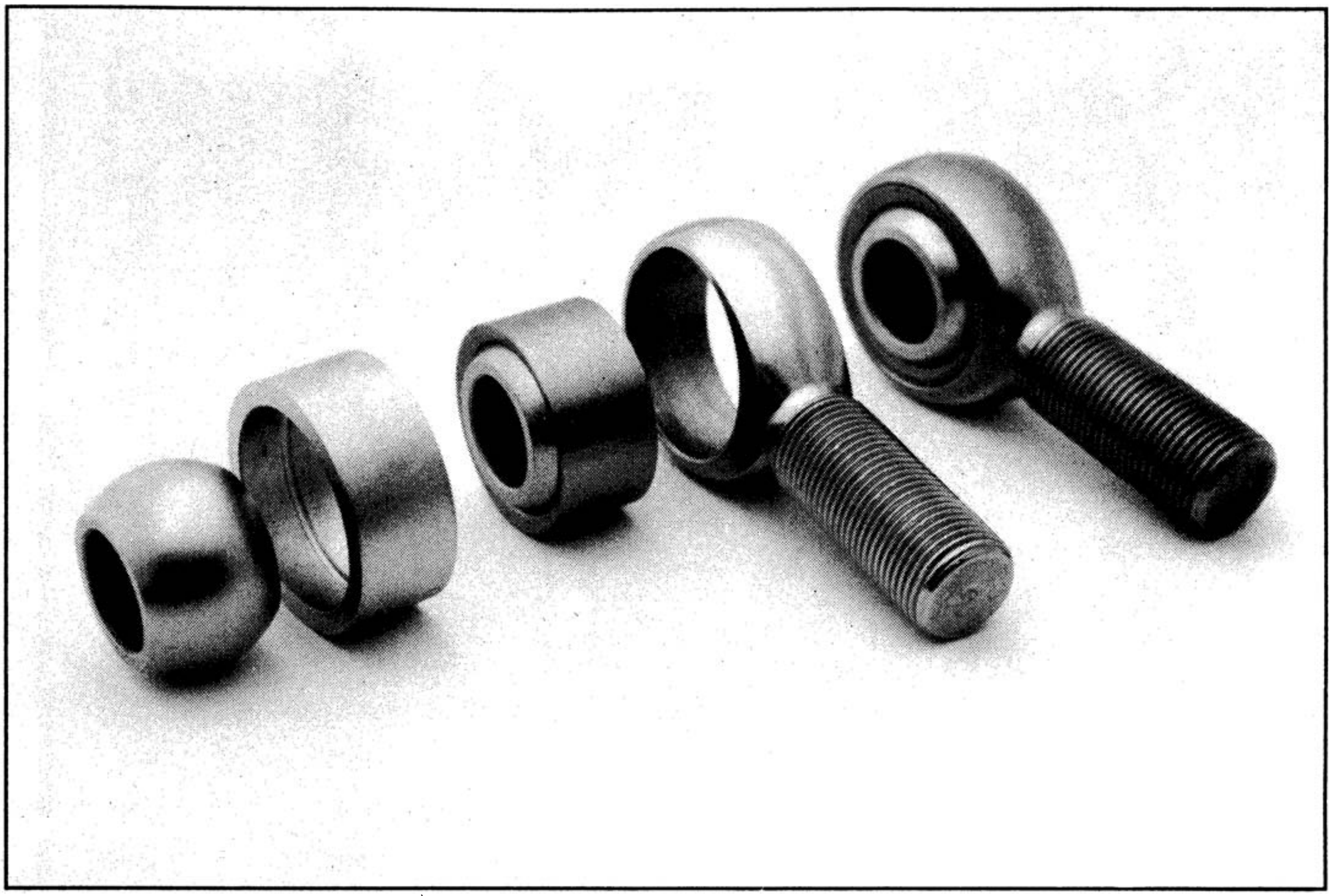
Damaged rod ends. Note insert on far left rod end is partially pushed out. Unit on far right shows dent in race from severe misalignment. Don't do anything with these rod ends except throw them out!

out" they are usually referring to one of two problems. First is the deformation of low strength "sel lubricating" liners. ome 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.

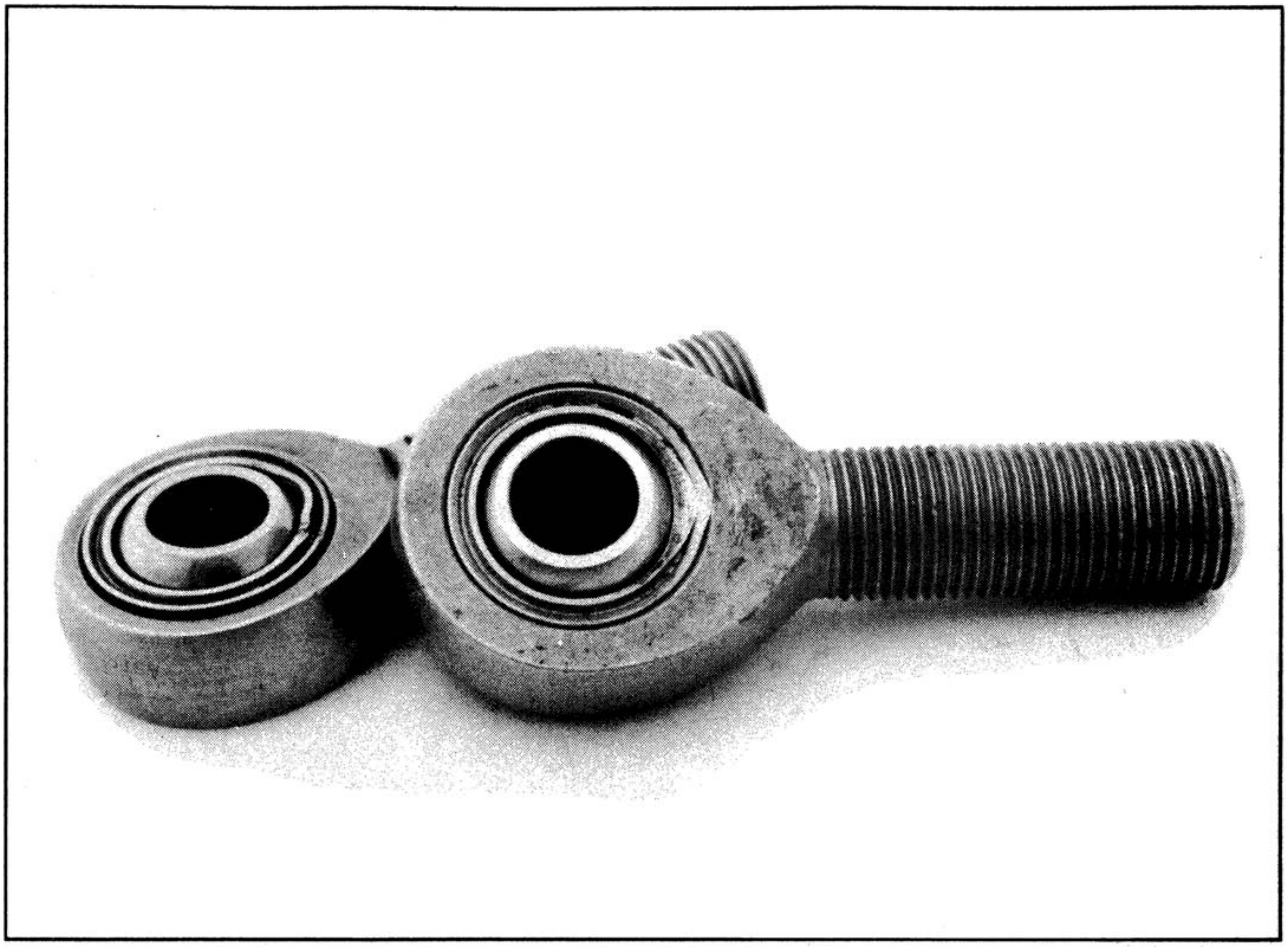
What are the advantages of oversize shank rod ends?

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

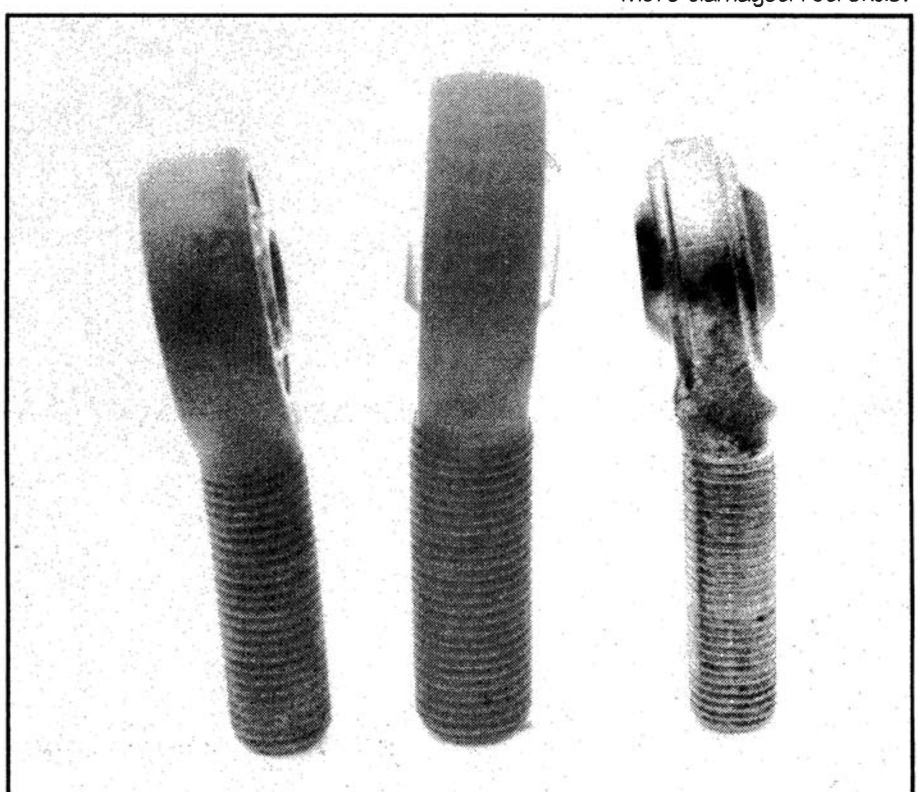


More damaged rod ends.

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 racecar 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.



ROD ENDS

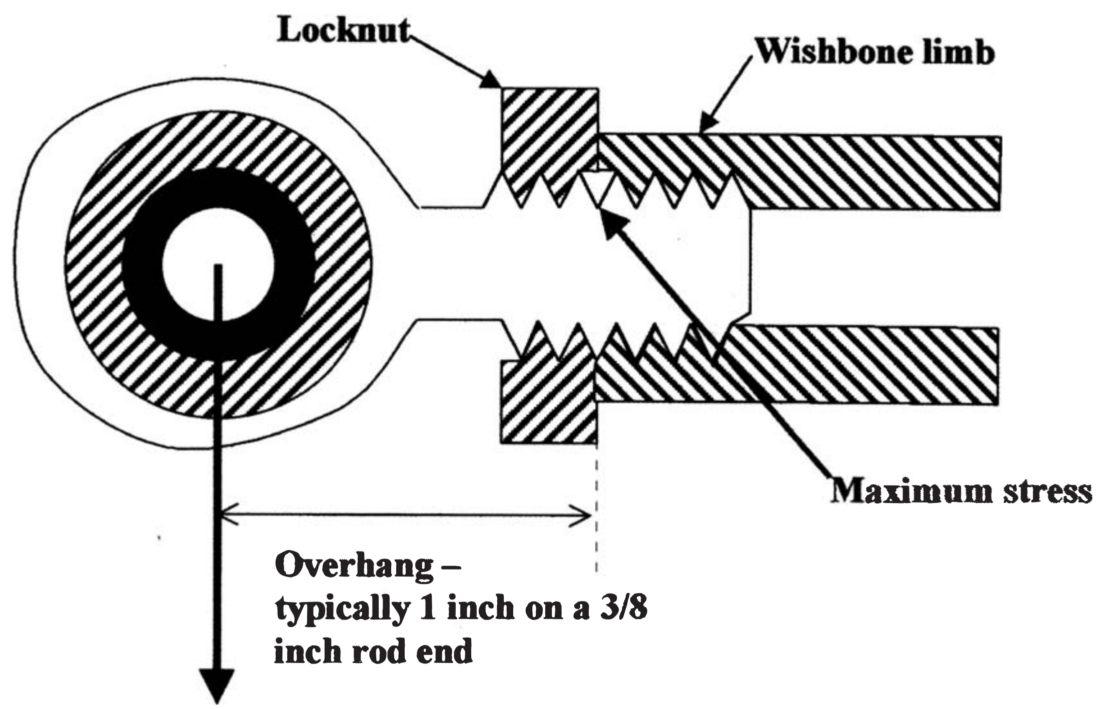


Figure 1 - Bending loads on a rod end

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

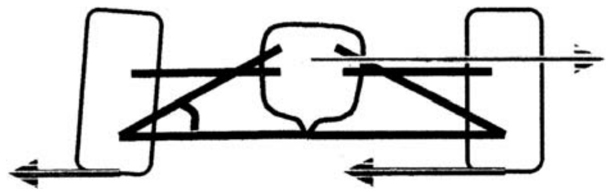
TO PUSH OR TO PULL

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.

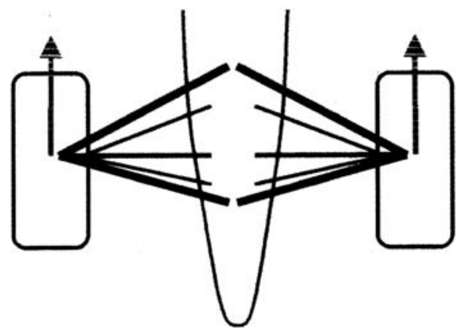
With outboard spring/dampers, or with push rods, this joint also has to support the weight of the car and any downforce, together with any load transfer. So it has to cope with major loads in all three directions. Proper design of the wishbone, by minimising bending loads, can maximise the load capacity 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 30º 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 pushrod makes with the car's lengthwise axis, and cos 90º = 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**

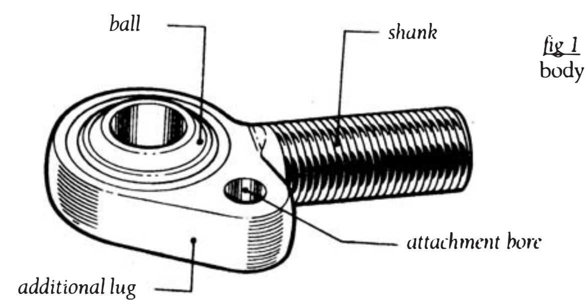


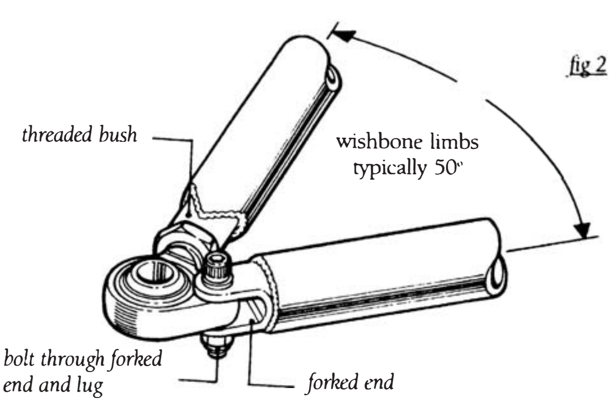
B **Pushrods virtually perpendicular to braking force – unable to transmit load to chassis**

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.



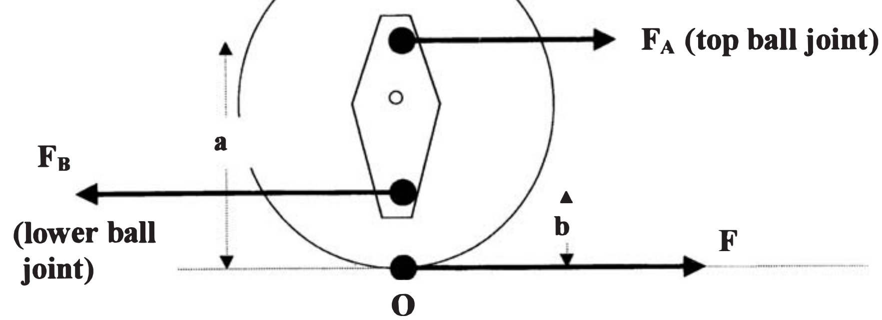


the shank, arrange the loading so that it's pushing the ball out of its race, and the race out of its housing. To add insult to injury, it's not unknown for the ball to be secured, to an upright for example, in single shear.

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.



$$F_A + F = F_B \quad \text{(for equilibrium)}$$

$$F_A .a = F_B.b \quad \text{(moments about O)}$$

So:

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

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

The key to the staked joint is the groove, shown clearly in Figure 3. 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 hous ing. 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 careful ly 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 tech niques are not really necessary.

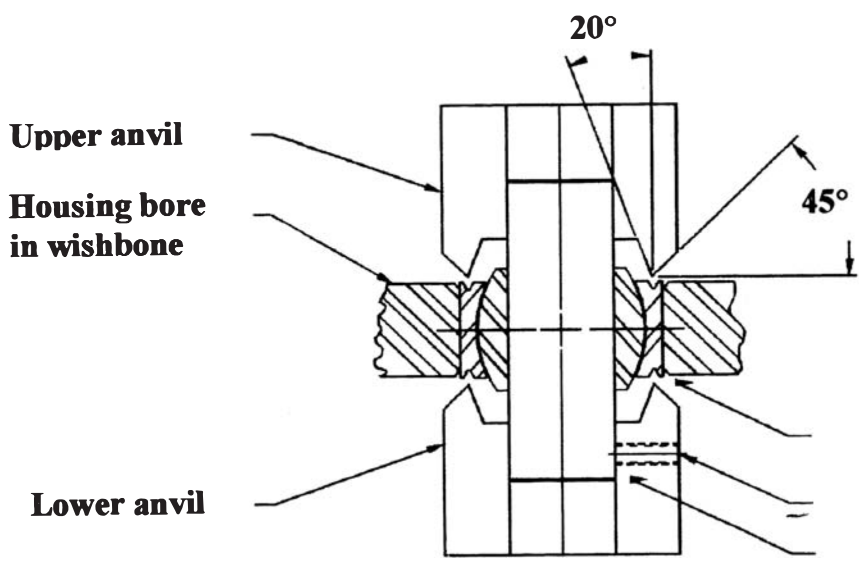


Figure 5 - General arrangement of staking tools for swaging a grooved bearing
(based on a graphic kindly supplied by Aurora Bearings Inc)

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 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 (actu ally a slot drill). The thickness of the strip is con veniently within the permitted 0.005" of the height of the bearing. The bore diameter is criti cal 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.

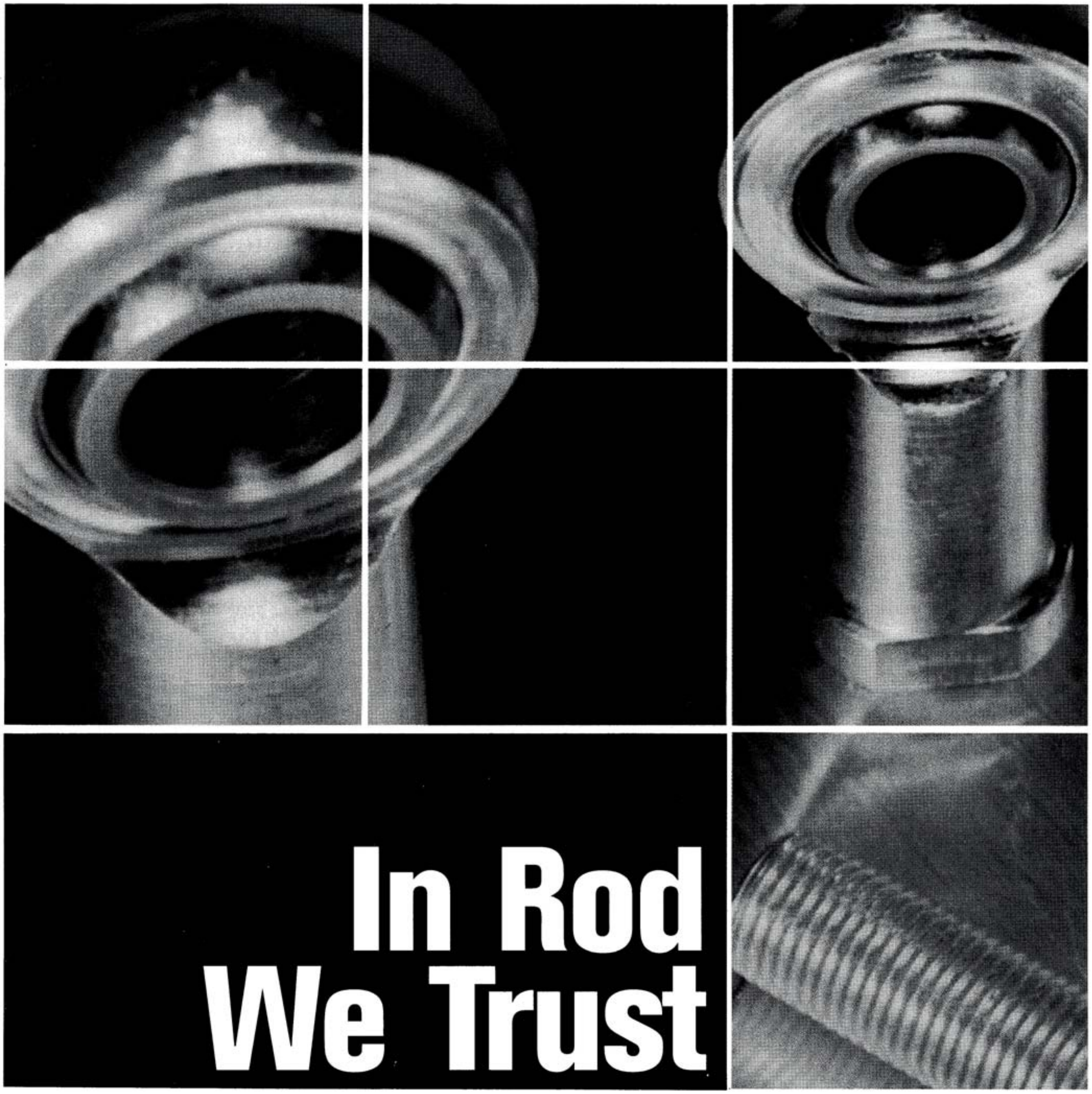
The swaging is carried out, using a hydraulic press and specially profiled staking tools, the 'anvils', on one side at a time or both together different national or industry engineering codes have different preferences. The press I use is essentially a hydraulic car jack in a strong but simple heavy steel frame. A standard workshop press is of course easily capable of doing the job. The dimensions and angles of the tooling are pro vided 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 installa tion, this should have increased to up to twice the maximum specified for an un installed bear ing. 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 expen sive 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 regular ly get satisfactory results.

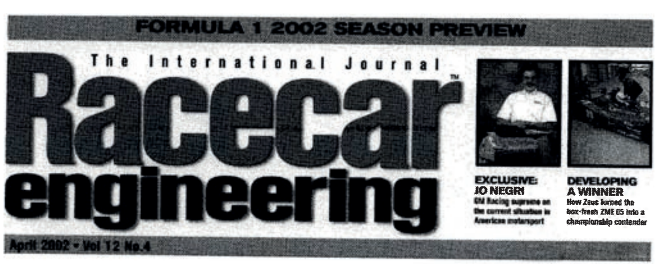
An alternative or additional test is to apply a 'proof' load to the swaged outer race. Under this proof load the swaged bearing must not be pushed out of its housing. This load is specified for an 11/16 inch bearing in the NAS 0331 as 1000 lbf per inch OD, so it should withstand a load of 688 lbf. The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

Grooved bearings are rather cheaper than a cor responding 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.



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 the bodies can also be constructed from aluminium or even titanium.

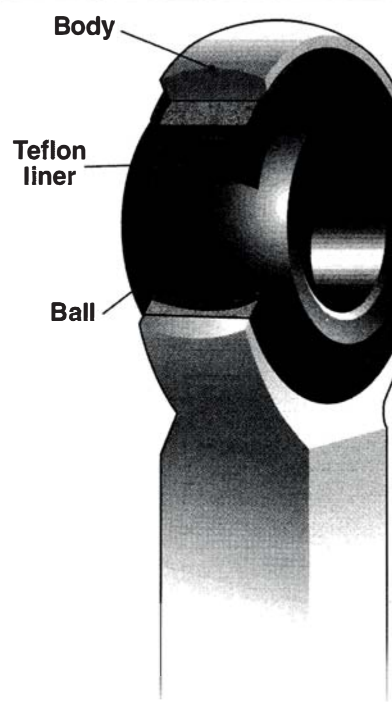
Some rod ends come with a teflon lining which allows them to be self lubricating, this is beneficial as oiling or greasing can increase the amount of wear due to dirt being attracted to the lubricant and working its way between the ball and the race. A teflon liner does not suffer from the same problem 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.

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,

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

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.

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

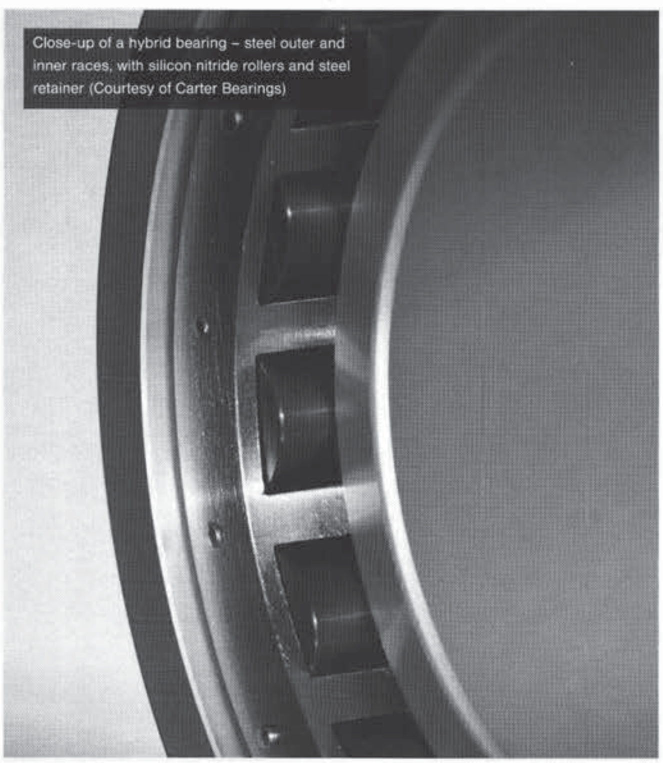
Bearings are normally made from steel alloys, either stainless chromium-based or high-alloy steels for enhanced corrosion resistance or surface hardness. Typically races are made from alloys such as 51200 tool steel, while 440C stainless steel can be used if the application calls for it.

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



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

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 – 3.2g/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.

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

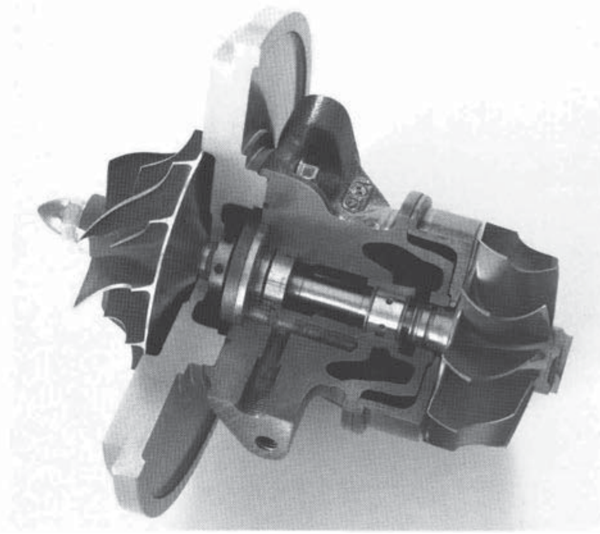
Bearing retainers are required in many forms of rolling-element bearing, as they serve to keep the rolling elements from contacting each other. Depending on the application, retainers can be made from steel or polymers. Retainers need to be stiff enough to maintain the elements' relative positions, although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