ROD ENDS

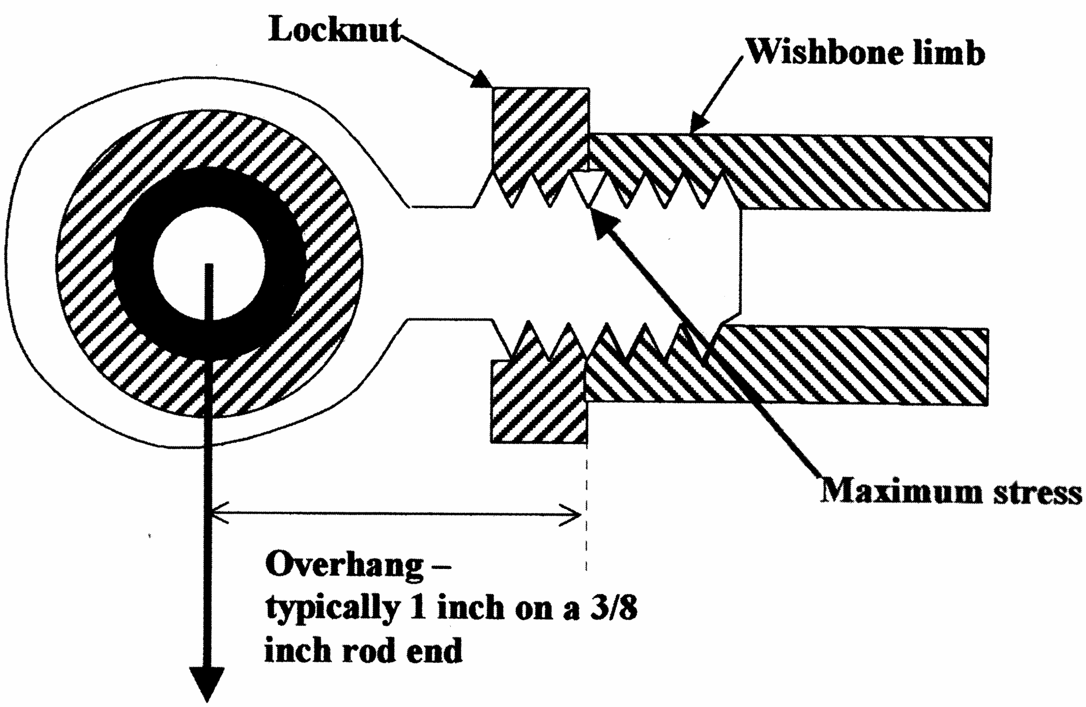


Figure 1 - Bending loads on a rod end

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

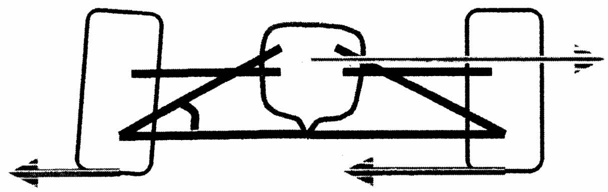
$$M = 30 \times 2240 \times 0.000468 \times 2 / 0.3125 = 201\ \text{lbf.in}$$

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.

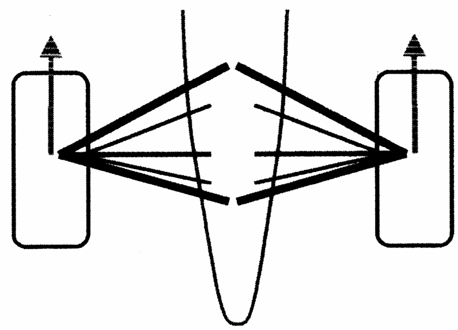
Looks familiar? This is a far cry from the typical 4000 lbf joint capacity, comfortably supported axially by the shank of the joint. Going to a 1/2 inch joint raises the critical side load to 555 lbf - better, but still well below the joint's even greater rated capacity. Even the most expensive joints 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



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

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.

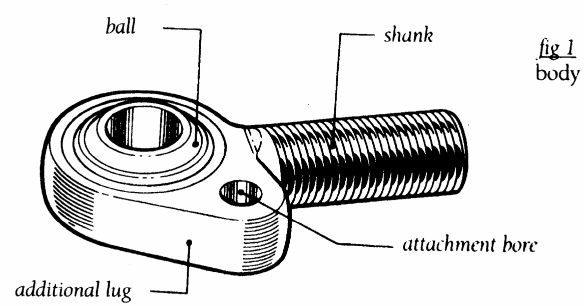
With outboard spring/dampers, or with push-rods, this joint also has to support the weight of the car and any downforce, together with any load transfer. So it has to cope with major loads in all three directions. Proper design of the wishbone, by minimising bending loads, can maximise the load capacity 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.

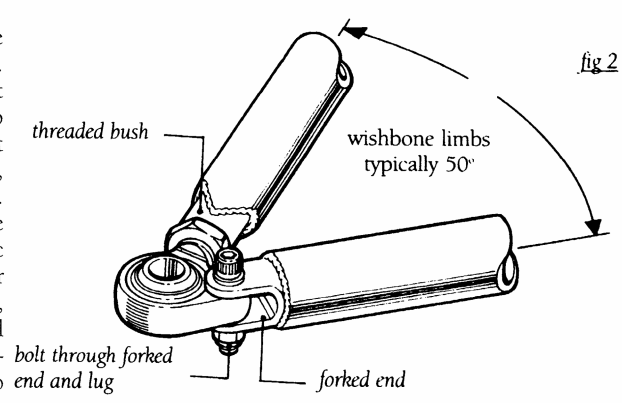
In Formula One, the currently universal high nose, with its under-slung wing, makes push-rods virtually inevitable, with the need to compromise the lower joint in one of the directions, because there is no space, low down, to put the spring/dampers. But in other series, where the high nose is outlawed 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.

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.

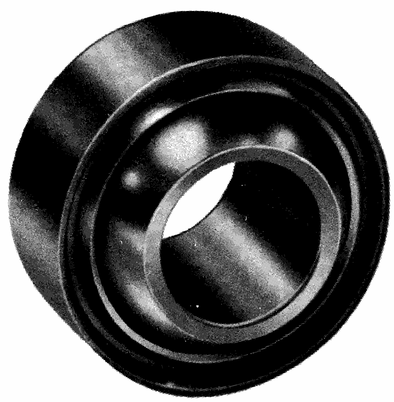


the shank, arrange the loading so that it's pushing the ball out of its race, and the race out of its housing. To add insult to injury, it's not unknown for the ball to be secured, to an upright for example, in single shear.

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.

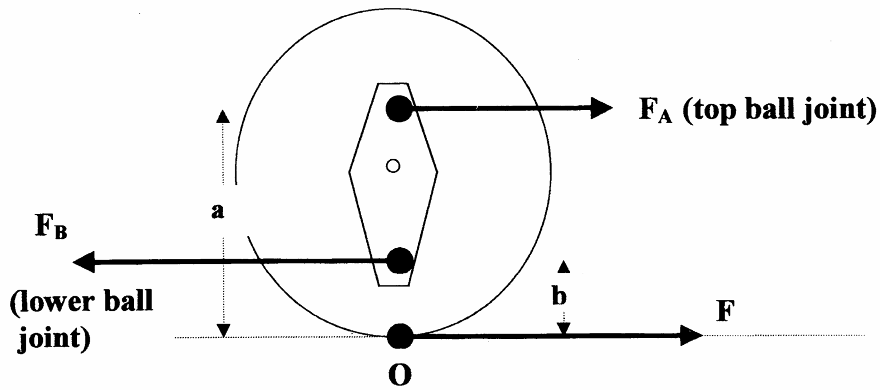
Although the highest specification rod-ends are two-piece, the outer race being forged integral with the housing, many rod-ends are staked joints - it's just that the joint manufacturers have already done the staking for you, fixing a ball and race, the bearing, into a pre-prepared housing with a thread 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:

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

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

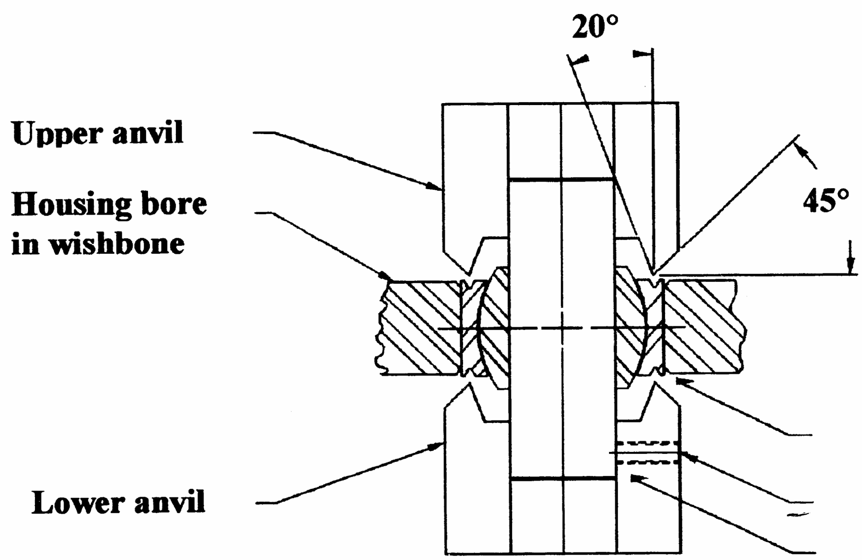


Figure 5 - General arrangement of staking tools for swaging a grooved bearing
(based on a graphic kindly supplied by Aurora Bearings Inc)

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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