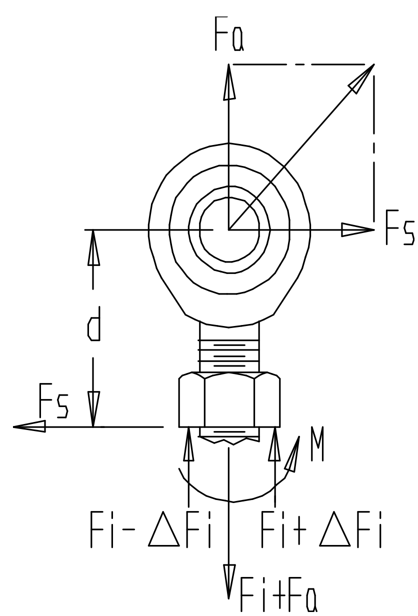
ROD END STRESS CALCULATIONS
By Dick Roberto

Historically, rod ends have been misused by nearly everyone in racing. Solar car builders are no exception. There seems to be a pervasive view that these little devices are bullet proof, idiot proof and should be used indiscriminately anywhere in a vehicle where relative rotation needs to occur between two rigid bodies or not so rigid bodies. As many of you know, rod ends are made for the transmission of loads where the loads are colinear to the axis of the stud, i.e., rod ends are the load points in a 'two-force' body that push and pull along the axis of the rod end stud. Load ratings listed in catalogs reflect this kind of loading. If rod ends are used in any other manner, the load ratings published in catalogs are meaningless.

The following is an attempt to assist solar car designers with estimating the stresses induced in the rod-end stud when both axial and shear loads are present. The sketch below shows a rod end with a typical improper application of loads. Since the rod ends are specifically designed for axial loading, axial stresses, in most cases, are not the critical issue here. The critical issue is the bending stress induced by a shear load (Fs) for which the rod end is not designed. It would be safe to assume that most of the rod end failures are the result of excessive bending stresses in the stud due to the presence of shear loads.

The calculations of stress in a rod end are approximate at best. Jam nuts should be used to stabilize the rod end, but their use complicates the issue of calculating the stress in the stud. A jam nut is a mixed blessing. While it may tend to reduce the bending moment on the stud, the jam nut applies an axial pre-load (Fi) when it is tightened. How much it helps and/or how much it hurts, depends on the amount of preload applied. Since the writer has never witnessed the use of a torque wrench by anybody in tightening a jam nut, it is safe to assume that the pre-load (Fi) is generally an unknown quantity.



Given the uncertainty of the pre-load, one can consider a worst case scenario relative to the bending stress by assuming that the preload is not present ($F_i = 0$). With reference to the sketch, the stress on the stud can then be calculated as:

$$\sigma = Fa/At + 32M/\pi dr^3$$

and, $$\eta = Sy/\sigma$$

Where: Fa = axial force
At = tensile area of thread. (See thread specs attached)
M = bending moment … M = Fs*d
d = moment arm
dr = root diameter of thread. (See thread specs attached)
Sy = yield strength of the rod end…(reference catalog)
η = factor of safety

This calculation reflects a static condition only. For a fatigue design (fluctuating stresses), a stress concentration factor would have to be used. The stress concentration factor for rolled threads is 3.0 for the alternating stresses. This means that the predicted dynamic stress levels are much greater than what would be calculated using the above equation. In order to keep things simple, and since solar cars do not have extended lives, a static design will suffice. An appropriate factor of safety is a choice based on (1) how safe the designer wants to be, (2) how long the designer wants the rod ends to last, (3) how well the loads on the rod ends are defined, etc. The writer believes that a suitable factor of safety should be at least 1.5.

Reference: MACHINE DESIGN, by Robert L Norton, second edition, Prentice Hall, 2000

Table 14-1 Principal Dimensions of Unified National Standard Screw Threads
Data Calculated from Equations 14.1—See Reference 3 for More Information

Size	Major Diameter d (in)	Coarse Threads—UNC Threads per inch	Coarse Threads—UNC Minor Diameter d_r (in)	Coarse Threads—UNC Tensile Stress Area A_t (in^2)	Fine Threads—UNF Threads per inch	Fine Threads—UNF Minor Diameter d_r (in)	Fine Threads—UNF Tensile Stress Area A_t (in^2)
10	0.1900	24	0.1359	0.0175	32	0.1494	0.0200
12	0.2160	24	0.1619	0.0242	28	0.1696	0.0258
1/4	0.2500	20	0.1850	0.0318	28	0.2036	0.0364
5/16	0.3125	18	0.2403	0.0524	24	0.2584	0.0581
3/8	0.3750	16	0.2938	0.0775	24	0.3209	0.0878
7/16	0.4375	14	0.3447	0.1063	20	0.3725	0.1187
1/2	0.5000	13	0.4001	0.1419	20	0.4350	0.1600
9/16	0.5625	12	0.4542	0.1819	18	0.4903	0.2030
5/8	0.6250	11	0.5069	0.2260	18	0.5528	0.2560
3/4	0.7500	10	0.6201	0.3345	16	0.6688	0.3730
7/8	0.8750	9	0.7307	0.4617	14	0.7822	0.5095
1	1.0000	8	0.8376	0.6057	12	0.8917	0.6630

Table 14-2 Principal Dimensions of ISO Metric Standard Screw Threads
Data Calculated from Equations 14.1—See Reference 4 for More Information

Major Diameter d (mm)	Coarse Threads Pitch p mm	Coarse Threads Minor Diameter d_r (mm)	Coarse Threads Tensile Stress Area A_t (mm^2)	Fine Threads Pitch p mm	Fine Threads Minor Diameter d_r (mm)	Fine Threads Tensile Stress Area A_t (mm^2)
5.0	0.80	4.02	14.18			
6.0	1.00	4.77	20.12			
7.0	1.00	5.77	28.86			
8.0	1.25	6.47	36.61	1.00	6.77	39.17
10.0	1.50	8.16	57.99	1.25	8.47	61.20
12.0	1.75	9.85	84.27	1.25	10.47	92.07
14.0	2.00	11.55	115.44	1.50	12.16	124.55
16.0	2.00	13.55	156.67	1.50	14.16	167.25
18.0	2.50	14.93	192.47	1.50	16.16	216.23
20.0	2.50	16.93	244.79	1.50	18.16	271.50
22.0	2.50	18.93	303.40	1.50	20.16	333.06
24.0	3.00	20.32	352.50	2.00	21.55	384.42