Miniature Ball Bearings in Clocks

Rex Swensen, September 2004, Updated June 2007

Part 1 - The Case for Miniature Ball Bearings in Clocks

Introduction

The "purist" clockmaker scorns the use of ball bearings in clocks, but even the famous English clockmaker, John Harrison, attempted to make a caged ball bearing. However, in his quest to miniaturize the chronometer, he replaced this early use of anti-friction bearings with tiny jewelled journal bearings.

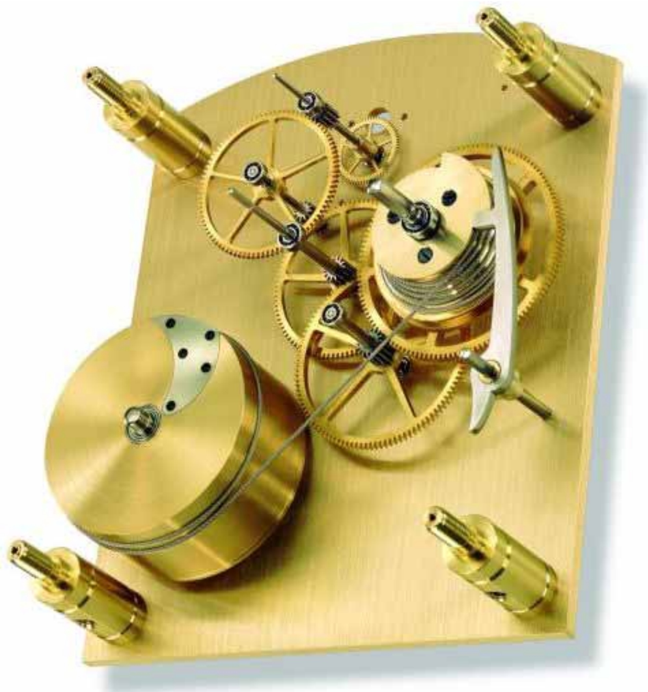
I am sure that if miniature bearings of the quality and price of today had been available in Harrison's time, he would have been the first to use them, probably drooling at the mouth!

Many amateur clockmakers of today do use ball bearings for the great wheel arbor, or if month going, also the intermediate arbor, since these are all heavily loaded. But what about the higher order arbors?

The photo on the right shows a bearing that has a bore of .6 mm and OD of 2 mm – now that IS tiny, also VERY expensive.

Again, the purist will say that a highly polished small diameter pivot of hardened steel, running in a well burnished hole, and well lubricated, cannot be beaten. In this paper I will attempt to show that this is simply not true. The miniature ball races available today are quite economical and win significantly on reduced friction, and probably, longer life, and lower maintenance.

While I had always been interested in this question, the research into the matter was triggered by the discovery that Erwin Sattler in Munich, the eminent modern day maker of high quality mechanical clocks, had moved extensively to the use of tiny ball bearings in their clock trains as shown in one of their photos below. But my curiosity was to determine, if possible, how much better ball bearings would be, particularly in the upper reaches of the train. In fact, should they be used on all arbors? The bearings under consideration here have an ID of 2mm and OD of 5mm, being a popular and inexpensive size.



Erwin Sattler movement showing ball races on every arbor

Evaluation Criteria

The major factors are:

- Friction – coefficient of friction
- Lubrication
- Corrosion
- Bearing Life, Future Availability and Service
- Robustness
- Cost

Methods of mounting ball races will be covered in Part II of this article.

Coefficient of Friction

This is the major criteria. Throughout this investigation, the C of F, commonly referred to as Mu or **μ**, will be expressed in terms of the journal diameter, as if it were a plain pivot. So even if a 2mm bore ball race is compared to a 1mm diameter plain pivot, the **μ** will be calculated at the effective diameter of 1mm. (see Appendix for definition of **μ**).

The challenge was to obtain numerical results for a variety of bearings, considering the stop–start nature of the clock train mechanism.

It is well known that the static **μ** for normal sliding friction is higher than dynamic friction, and this is true for ball races as well, according to bearing manufacturers. It is also recognised that ball races are at their best when loads and speed are higher, neither of which apply in a clock.

In spite of these apparent disadvantages, the challenge was to actually measure **μ** for ball bearings in a "clock like" environment and see how they compared with traditional pivots.

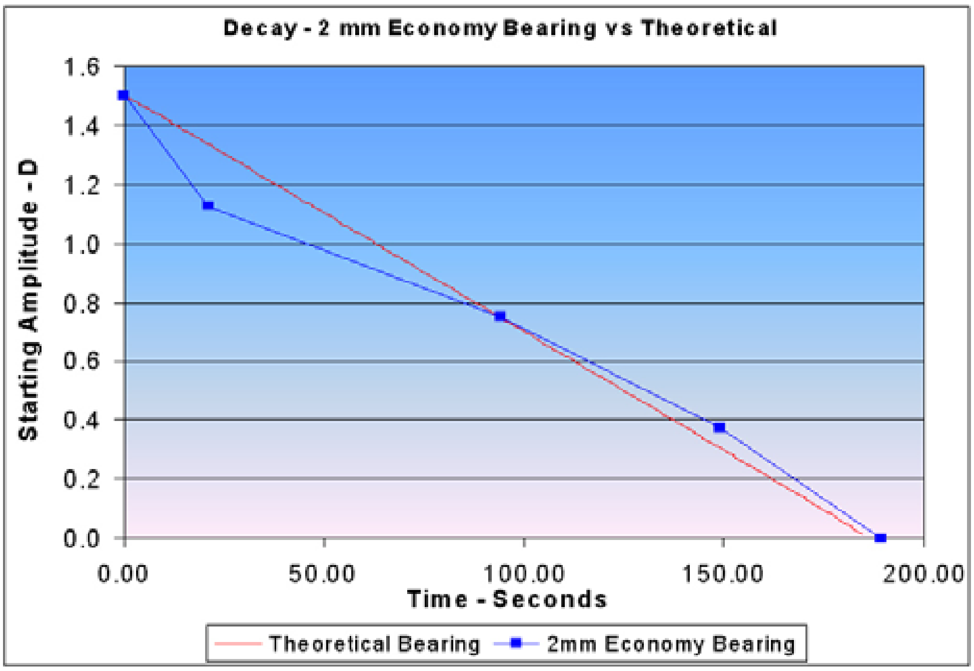
Methodology

It was felt that a swinging pendulum, with the pivot provided by the bearing under test, would provide a simple and easily constructed test rig, provided that the amplitude was kept small. In this way the predominant damping force would be that of friction in the bearing, and not air friction on the pendulum bob. This type of damping is known as Coulomb damping, and should result in a constant rate of decay. It would also provide a combination of static and dynamic friction similar to that found in the stop-start nature of a clock. The assumption of linear decay should also be verified if possible.

Test Rig

The test rig is shown at the right. The bearing under test is inserted into a mounting hole at the top of the pendulum shaft and a pivot held in the chuck. While this is only a single bearing rather than the two as in conventional clock plates, great pains were taken to ensure that the pendulum bob hung directly below the bearing and the plane of oscillation was at right angles to the pivot. The angle bracket on the lathe cross slide has a line inscribed to show bottom dead centre. The cross slide was used to accurately move the datum for the requisite starting amplitude for each test. The mass of the bob approximates the typical bearing load.

An Excel spreadsheet was used to store results and perform all calculations. The pendulum was started at the desired amplitude and the number of cycles were either counted until the oscillations ceased, or the time measured by stop watch and the cycles computed which was necessary with the ball races.



It would have been desirable to record the actual decay, cycle by cycle, but this was beyond my instrumentation capability. However, some approximation of the decay envelope could be achieved by starting the oscillation from a variety of initial deflections and overlaying the results. The outcome is shown in the Excel spread sheet graph. Theoretically, for Coulomb damping, this should be a straight line showing linear decay as in the red line. The actual results confirm that Coulomb damping is a reasonable assumption. The pendulum has a full cycle period of .68 seconds.
As mentioned earlier, the down side to this rig is that the pendulum swings from only a single bearing, where as the usual configuration is a bearing at either end of an arbor. However the pendulum rod was bent such that the C of G is directly under the bearing and no side thrust is applied. The swing was also initiated so that it did not wobble from side to side.

Ball Bearings Under Test
The bearings tested were procured from MBA via their web site: (www.smallparts.com.au). They are all *shielded* to exclude dust. There is no contact between the shield and the inner ring since it is a labyrinth seal. It is interesting that Sattler use open races. A bearing with 2mm bore was chosen for the main testing since it is small enough for upper arbors, yet has a reasonable price. Note that *sealed* bearings should NOT be used since many seals make positive contact with the inner race, hence higher friction.

Shielded ball races under test

Bearings Tested:

- 2 x 5 x 2.3mm (Bore, OD, Width) economy chrome steel - suitable for higher arbors
- 2 x 5 x 2.3mm high quality stainless steel. - suitable for higher arbors
- 5 x 11 x 5mm economy chrome steel - suitable for great wheel arbors and weight pulleys
- Plain 1mm traditional pivot as used in higher arbors

It was expected that the lubrication in the ball bearings, as received, would have a significant effect on results, and this proved to be the case. Tests were conducted “as received”, after soaking in kerosene, and after soaking in acetone. White spirit may have been better. It is known that a completely dry bearing would yield the lowest **μ** and this is recommended by bearing makers for applications where minimum friction is required. Potential life is somewhat shorter however.

Friction Test Results

1) Economy 2mm Bearing - Safe static load 2.2 Kg. Price ~ $0.85

This bearing received the most detailed testing.

Starting Amplitude - deg	1.500	1.125	0.750	0.375
Calculated **μ** as received	0.0053	0.0044	0.0052	0.0062
Calculated **μ** after soaking in kero		0.0023		

Ten of these bearings were purchased in a packet, and four were tested with the 1.125 deg swing. Results were substantially the same and are within the realm of experimental error.

2) Stainless Steel 2mm Bearing - Safe static load 1.8 Kg. Price ranges from $1.85 to $8.00

Starting Amplitude – deg	1.1250
Calculated **μ** as received	0.0190
Calculated **μ** after kero soaking	0.0010

3) Economy 5mm Bearing - Safe static load 11.8 Kg

This bearing was obviously well packed with grease, and fairly stiff, so it was soaked in kerosene before testing.

Starting Amplitude – deg	1.125
Calculated **μ** after kero soaking	0.0078
Calculated **μ** after soaking in acetone	0.0014

4) Plain 1mm Pivot – polished pivot steel in brass hole

This proved to be the most difficult to test on the pendulum rig. With such small starting amplitudes, the decay appeared to go through two distinct phases. Initially rapid decay, presumably while the rotating friction was in effect, then an extremely long period of very slow decay commencing at a very small amplitude. This latter phase was thought to be when the bearing was rolling within the radial clearance of the bearing, almost like a knife edge. The results were therefore quite meaningless.

An attempt was made to measure the number of cycles for the decay to reach the half amplitude point, presumably while sliding friction was still in control, but this gave a **μ** of 0.16 dry and 0.22 lubricated, which seem rather odd. Perhaps the single pivot method of suspension, rather than an arbor with pivot at either end, had an influence on this outcome and also the drag effect of the lubricant which was normal clock oil.

I therefore think it is safe to assume that the effective **μ** for a hardened and polished steel pivot in brass, lubricated with clock oil, will be about 0.20 considering that the effect of combined static and dynamic friction is present.

5) Jewel Pivots
A jewelled pivot has not been tested, but according to a supplier of these in the USA, the **μ** value is 0 .15. The big advantage of jewels appears to be in their wear resistance, and hence a much longer life than plain pivots. (www.swissjewels.com) Also, as Dave Micklethwaite pointed out after the meeting, they look pretty!

6) Anti-Friction Wheels
At the meeting Dave Micklethwaite showed an “anti-friction wheel” which he had made. It was a finely crossed out wheel some 55mm in diameter made from .63 mm brass. The concept here provides three of these wheels to support and contain each pivot as a rolling contact bearing with each anti-friction wheel running on a jewelled arbor. If the pivot in question was say 1.5 mm

diameter, the effective **µ** for the pivot "bearing" would be approximately .15 x (1.5/55) = .004 which is similar to that of a ball race. This is the type of bearing that Harrison experimented with. However the complexity of wheels and arbors becomes quite daunting. Dave saw this type of bearing in a clock made by a professional clockmaker in Mossvale who had recently immigrated from England. It is believed he was part of Buchanan Clocks. See HJ April 2006 , or http://www.bhi.co.uk/hj/April%2006%20AOM.pdf which describes one of the clocks he made. A truly magnificent achievement.

Pivot Friction Report by Dick Stephen – HJ Sept 2005

Subsequent to my studies, Dick Stephen conducted similar experiments and reported them in the HJ. Dick has access to much more elaborate electronic test equipment than I do and is able to chart the pendulum amplitude as it decays, cycle by cycle. He compared a 1 mm ball race with jewel bearings, both parallel bore and olive shaped holes, with minimum clearance and with .008mm clearance. His results were very similar to mine, but unfortunately he did not pursue the physics and the maths to reach any conclusions on coefficients of friction. So I scanned a few of his amplitude plots and loaded them into AutoCad, picked up the key values and analysed them in my spread sheet. If I assume that the results of his testing on an olive jewelled bearing represented a **µ** of .15, then the **µ** for his 1 mm ball race comes out at .004 which is similar to my findings. But then he did use a 1 mm race whereas mine are 2 mm ID. I reported my findings in a letter to HJ November, 2005.

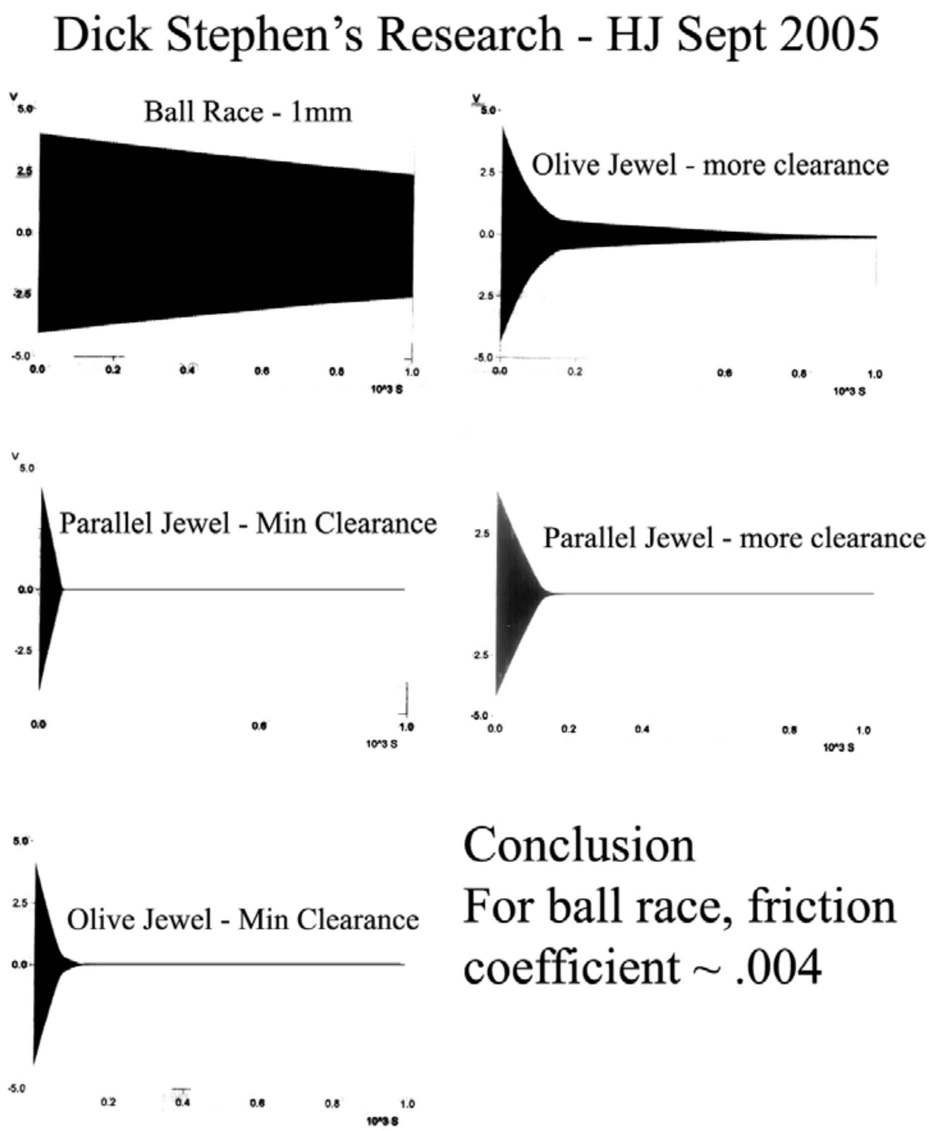
Dick's article may be read in full at: http://www.bhi.co.uk/hj/AoM%20September%2005.pdf

Arbor Shoulder Friction - End Stones

End stones are frequently recommended on high quality clocks to eliminate pivot shoulder friction. It is worth noting that these are totally unnecessary with ball races, since the axial thrust capability provides the same function. But care must be exercised in machining the bearing housing such that there is no contact between the inner race and the housing.

Comparison of Friction Results

From these test results it would appear reasonable to assume a **µ** of .002 for these tiny ball races, if well soaked out in white spirit or similar solvent. For plain pivots, a value of .15 - .20 is appropriate. This means that the plain pivot has at least **70** times as much friction as the ball race.



Lubrication

The type and viscosity of the lubrication in the “as shipped” bearing is highly significant and may represent a considerable drag inside the bearing. I recommend that the grease or oil be soaked out with white spirit or acetone. I expect that the white spirit leaves a thin residue of lubricant, although it is doubtful if this is still there after say twenty years. However manufacturers recommend completely dry bearings for absolutely the lowest friction. This is thought to reduce bearing life.

Corrosion

There are two types of materials in use in the commonly obtainable bearings.

SAE52100 Chrome Steel as used in the economy pack races and others

AIS I 440C Stainless Steel

Naturally the chrome steel is slightly cheaper, but more likely to corrode in the long term when the maximum bearing life is desired. I have now switched to using the hardened stainless steel variety. I prefer shielded races to keep out any dust. It is amazing how dust infiltrates a clock movement. I recently took the hood off my longcase clock and noticed a vale of dust on the seat board. And this was only after a couple of years. And the hood fitted very well with barely a 1 mm air gap.

Bearing Life, Future Availability and Service

Will such bearings still be available in a hundred years time when the present ones wear out? Firstly, who says they will wear out in a hundred years. While it is probably unrealistic to use standard bearing life formulae with the ultra light loadings and stop-start motion found in our clocks, the calculations indicate an extremely long life expectancy. For the bearings in my Vienna regulator and the loadings present, the most critical bearing is at one end of the centre arbor, and the calculated life is 11,000 years! The great wheel arbor with its larger bearings, comes out at 31,000 years. It is doubtful if this long life would actually be achieved, but it indicates that very long lives could be expected and a full strip-down service will be very infrequent. It is my belief that the widespread use of these bearings in delicate instrumentation is increasing, rather than decreasing, so I do not expect availability to be a problem. You only have to look at the web site referenced above to see the enormous range of bearings available today. There is some 3200 individual bearing sizes catalogued in the range from 0.6mm ID to 60mm ID.

However teaching clock restorers to understand and respect ball bearings, and to procure replacements, is a different matter altogether. Most seem to live in a world from 100 years ago. So getting adequate servicing in the future may be a problem. So do leave a clear inscription on your clock that it has ball races fitted!

Robustness

One way of reducing the friction with plain pivots is by reducing the pivot diameter. This leads to very fragile pivots which are easily snapped off during assembly if the arbor is allowed to tilt too much. The arbor for the 2 mm ball races is massively robust by comparison. However the race that it is going into must be respected since it now becomes the more fragile component in the assembly process. It will tolerate quite a degree of angular misalignment during assembly, but it should still be respected. Stainless steel then become the material of choice for arbors

Cost

Here the ball race initially falls down. It is clearly more expensive. But over the life of the clock it will probably come out in front due to the reduced service interval.

How to Fit Ball Races

Part II of this paper will explore some techniques for fitting ball races.

Conclusions

The reduction in coefficient of friction, and the assumed reduction in wear and maintenance requirements, makes a very strong case for ball races on all arbors. Many constructors fit them to the great wheel arbor on the basis that it is the more heavily loaded. This is true but it also has the most torque applied. The upper arbors have much less load, but also much less torque. It could be that friction has an equal or greater impact at the top of the train than lower down.

The economy grade races from MBA, at a cost of approximately a dollar each, appear to be quite adequate for clock applications. However corrosion is still an open question, so I have adopted the stainless steel variety for future use.

I have also adopted them for the pallet arbor which is oscillating about 4 degrees. Ball races are not recommended for this mode of use, but the loads on this arbor are the lightest of all, so I have confidence that long, trouble free life will be achieved. It has been suggested that they could be used for pendulum pivots as well, even eliminating the crutch, but here the loads are considerable due to bob mass, so I have not recommended this approach. Perhaps someone can give it a try?

As stated earlier, it would appear reasonable to assume a **μ** of .002 for these tiny races, if well soaked out in white spirit. For plain pivots a value of .15 - .20 is appropriate. This means that the plain pivot has at least 70 times as much friction as the ball race.

Final Thought

It is interesting to note that a ball race with a μ of .002 relative to a 1 mm pivot, has an equivalent friction drag to a plain pivot with a diameter of .013mm (.0005")! Less than a human hair!

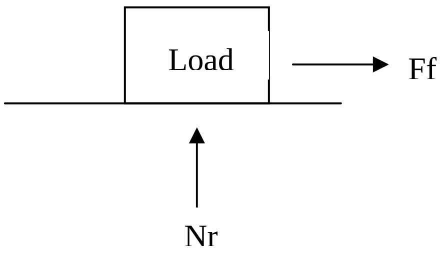
Rex Swensen
September 2004, updated June 2007

APPENDIX

Definition of Coefficient of Friction (μ)

Static **μ** = Ff/Nr at the point when sliding just commences

Dynamic **μ** = Ff/Nr when steady sliding movement is established. It is usually less than Static **μ**



BALL BEARINGS IN CLOCKS

Rex Swensen – December 2005, Updated May 2007

Part II Mounting Ball Races in Clocks

This is a short note to give some thoughts on how ball races may be mounted in clock plates.

Great Wheel Bearings

These have been described in many constructional articles. **Figure 1** shows a typical layout for a 5mm ID by 11mm OD race. The housing is located in the plate, in an accurately bored hole. It is typically attached with three 10BA screws. On the winding side (shown), the hole for the arbor must have clearance around the arbor. The fit of the race in the housing should be an "easy push fit" – not tight, as it is then difficult to remove. The fit on the arbor should be a very easy push fit, with the normal amount of "shake". Note the clearance for the inner race in the housing.

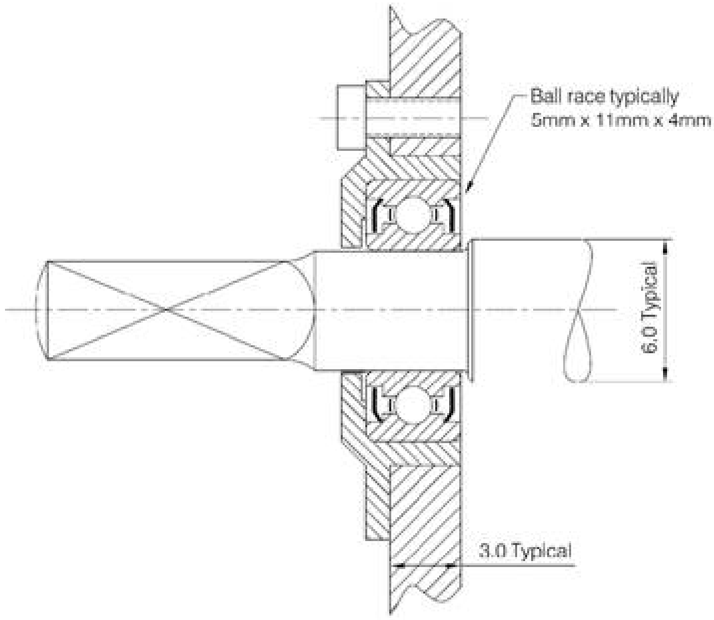


Fig 1 Typical Great Wheel bearing housing

If you are fitting ball races to the great wheel arbor, then it is probably a good idea to also fit one to the weight pulley.

Other Arbors, including the Pallet Arbor

The arrangement I have used recently is shown in **Figure 2**. A special spotface cutter was made as shown in **Photo 1**. It is made from 1/4" silver steel. The four teeth were hand filed, hardened and polished before the spigot was pushed in. The spigot is 3/32" to suite a 3/32" pilot hole. This provides clearance over 2mm if the arbor projects through a bit. Be sure to provide about .005" clearance for the inner race to ensure that the inner race does not rub against the plate. This is highlighted in detail in **Figure 2.**

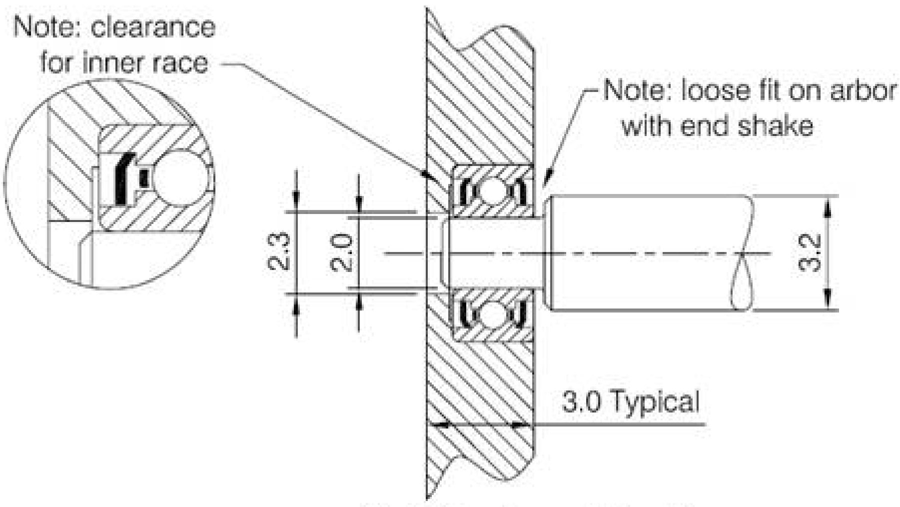


Fig 2 Bearing sunk in plate

Bearing - 2mm ID x 5mm OD x 2.3mm W

I made my cutter slightly oversize on the 5mm OD, then stoned it down between test cuts to aim for the somewhat elusive "easy push fit". On no account should the bearing be a tight fit, mainly to ensure that it will come out readily. The fit on the arbor may be fairly loose to ensure end float.

It would be better if the cutter flutes were somewhat deeper to aid chip clearance.

You might think a slot drill or end mill would be better, but mine did not cut truly to size, and I wanted to have a pilot spigot to guide the cutter. All the arbor holes were drilled at

Photo 1 Special spotface cutter

3/32" in one setting in both plates, then opened out from the inside with the spotface cutter to leave a shouldered hole.

Another approach is to bore the hole very slightly under size then push in a hardened and polished arbor to burnish the hole to size. Do not use a rotating arbor, just a stationary arbor in the drill press for good alignment.

Figures 3 and **4** show several variations for mounting bearings.

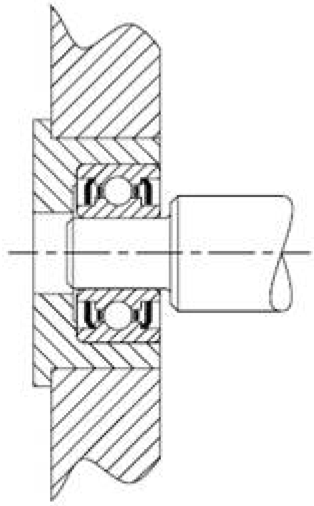


Fig 3 Bushing mount - loctited in plate

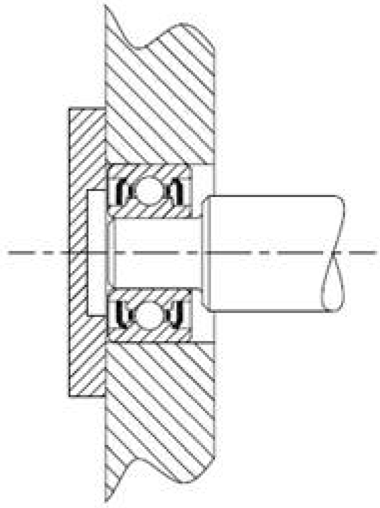


Fig 4 Reamed hole with Cover Plate
Note - clearance for arbor and inner race

Miniature bearings MUST be handled with care. The depth of material in the ring groove is tiny and may be damaged with excessive force when fitting, especially misalignment. See more details at www.machinedesign.com/ASP/viewSelectedArticle.asp?strArticleId=56385&strSite=MDSite&Screen=CURRENTISSUE

Minimum Friction

The bearings should have the lubricant washed out with a solvent such as white spirit, or procured dry for minimum friction. Friction will then be about 1/70th the friction of a plain pivot.

Bearing materials

The preferred material is **AIS I 440C Stainless Steel** which is fully hardened stainless steel, but usually more expensive. The alternative is **SAE52100 Chrome Steel** which is cheaper but open to corrosion in the long term, similar to any un-blued hardened steel.

Procurement Source

These bearings may be procured from Miniature Bearings Australia via their web site: http://www.smallparts.com.au/cgi-bin/store/product.pl?alpha=B&opencat=EX5#EX5. Look for Bearings, Single row, Radial, Unflanged. The catalogue is huge, but not all bearings are carried as stock items. Prices vary considerably from time to time so keep an eye on them. Note also, that the minimum freight charge is $8.25, even if you buy just one tiny bearing. So look ahead and buy up big, or pool your order with others.

Remember, ball races have only 1/70th the friction of plain pivots!