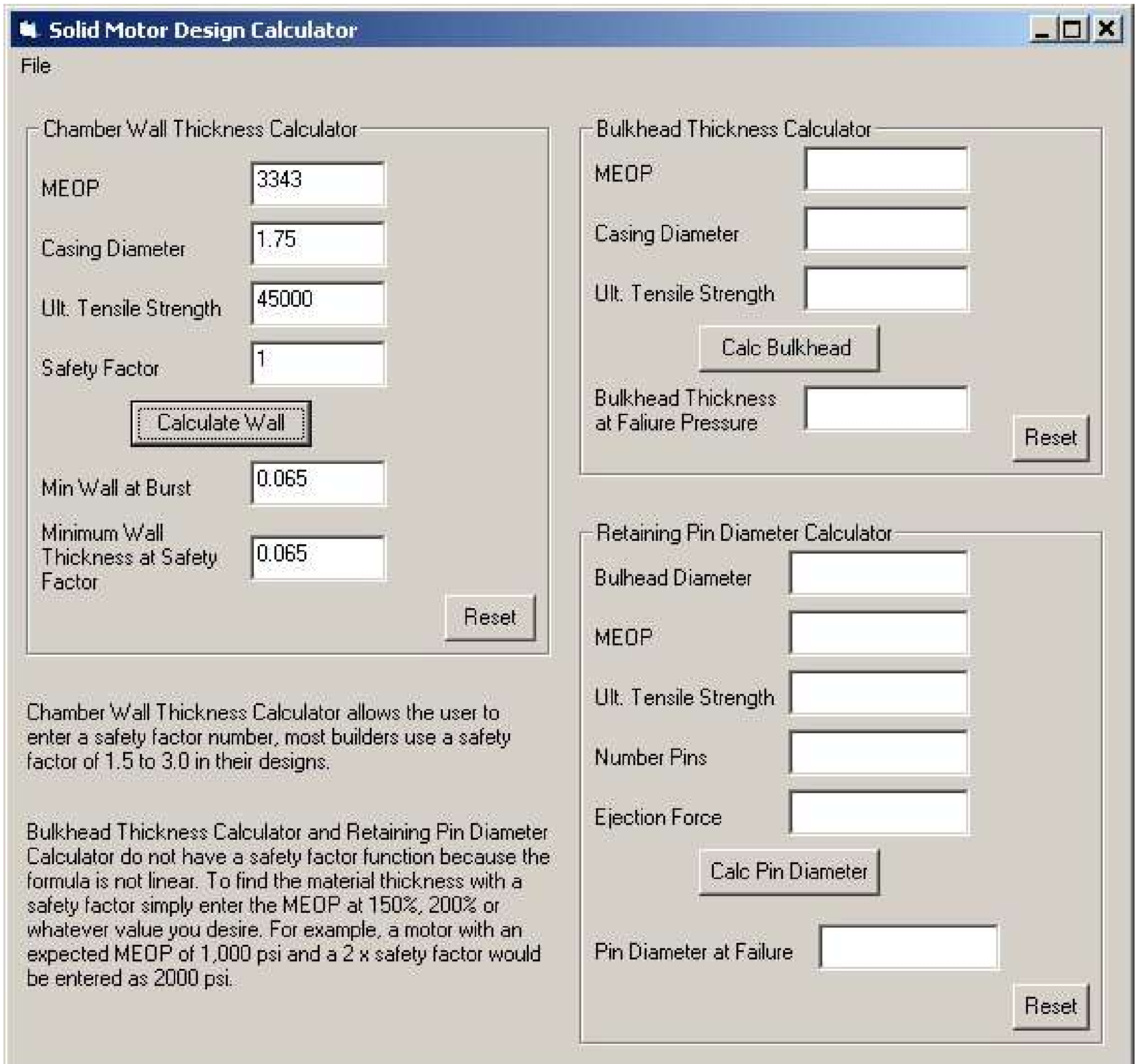
Motor Casings

When I first started making sugar propellant motors, there was a certain allure to steel casings in that steel was fairly resistant to heat. The motor casings didn't need extra thermal protection for short burn KNSU propellant. The casing really is the foundation of motor, essentially just a cylindrical tube, but it must possess great strength, be light in weight and not turn into shrapnel if it bursts. Steel was inexpensive and easy to work with, but it did leave a bit to be desired in terms of weight, while the mild steel I used wasn't prone to fragmenting if it burst, most steel alloys will fragment. In the end, I chose aluminum, as most motor builders do, as the casing material of choice.

The particular alloy is of utmost importance when dealing with aluminum, as poor alloys don't have the required strength. Either 6061-T6 alloy with an UTS (ultimate tensile strength) of 45,000 psi; or 2014-T6 with an UTS of 70,000 psi are the most popular choices. 6061-T6 is widely available, inexpensive and suitable for the vast majority of motor applications. I'll get into thermal issues more later, but I need to bring up a point now. The UTS of aluminum is at room temperature, at elevated temperatures the metal quickly loses strength. It is also easy to anneal aluminum. (Annealing is a process of heating and cooling a metal that decreases its strength, or removes its temper.) When manufactured, aluminum is tempered to increase its strength, at about 350 F. aluminum begins to permanently lose its temper. It's unlikely the amateur would have the abilities to retemper the aluminum once its temper is lost. So keep aluminum cool!

Let's return to the motor we designed in the last chapter, we'll use the KNSU version of the motor using the 4) 1.5" diameter by 2.25" long grains. We will have to find a casing with a suitable diameter, making sure we leave some free space in the diameter for an inhibitor layer and for combustion gases to flow behind the grains. A 1.75" outside diameter tube with a wall thickness of .065" should be about right. Let's check it out. 1.75" - (.065 * 2) = 1.62" available inside diameter. With a grain diameter of 1.5", that leaves .12" for an inhibitor, insulation and some free space. I'll get into the details of the inhibitor layer later, but what we will use is a layer of 100 pound card stock and 1) layer of foil tape, these 2 layers have a combined thickness of about .015". Which leaves .105" for insulation, and free space for combustion gas behind the grains, which is adequate.

Now it's time to open another one of my software applications called Solid Motor Design Calculator. In the upper left is the Chamber Wall Thickness Calculator. MEOP stands for maximum expected operating pressure. To find out how much pressure a given casing will handle, enter in a casing diameter of 1.75", the UTS of 6061-T6 at 45000, enter a safety factor of 1, then enter a random value in the MEOP box and click on Calculate Wall. We know we can get the 1.75" diameter tube in a wall thickness of .065", so change the MEOP a few times to zero on .065 in the Min Wall at Burst box.



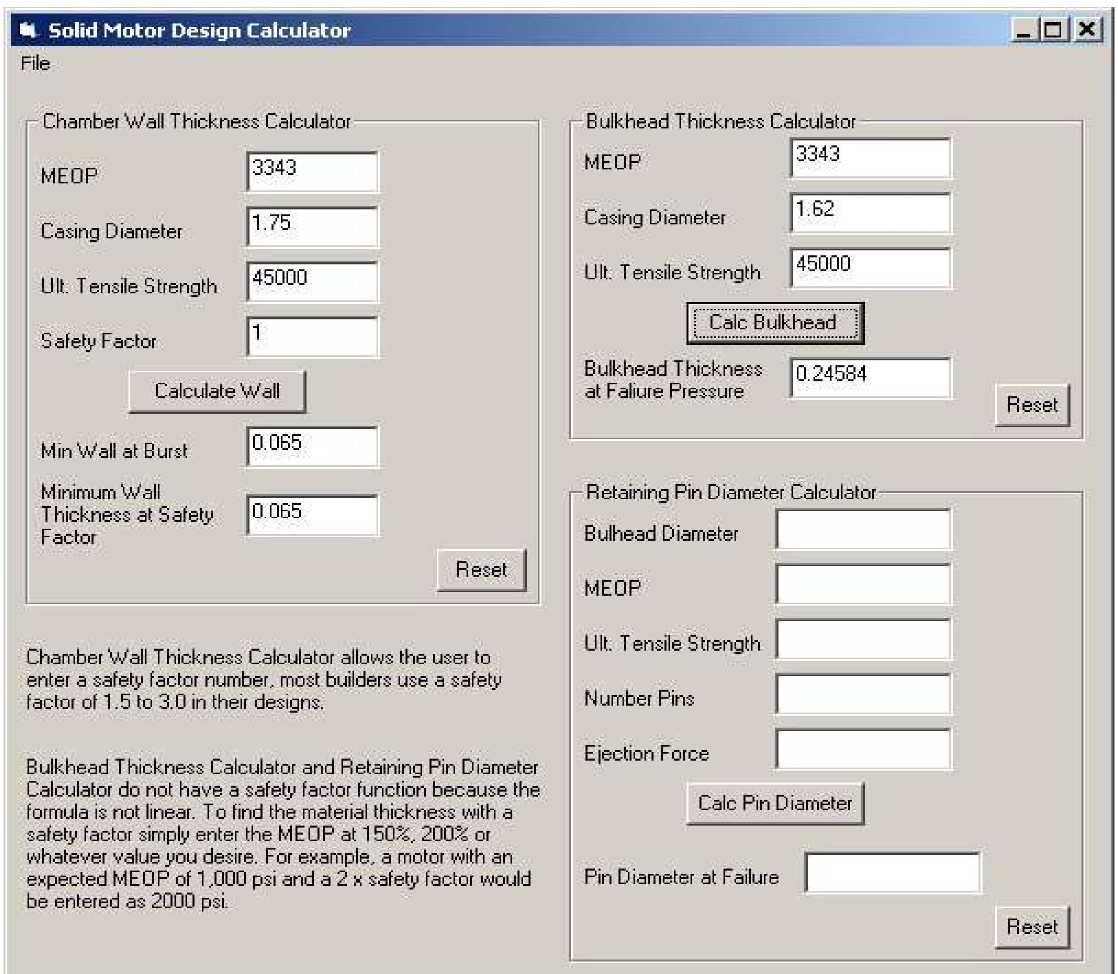
A screen capture from Solid Motor Design Calculator while calculating chamber wall thickness.

As you will soon discover, you can quickly zero in on the casing wall thickness with just a few guesses on the chamber pressure. In this case, our aluminum tube has a burst pressure of about 3,343 psi. It may be surprising to learn the material can handle so much pressure, small diameter tubes handle a lot of pressure, but as the diameter increases you will need to use thicker walled tubing. Keep in mind too, there will be some imperfections in the material that will reduce the strength. There is also thermal capacity of the casing to consider, a thicker casing will have the ability to absorb more heat, and be less prone to heat damage. Our quick burn KNSU should be just fine with this material.

I like to see a design that uses at least a safety margin of 2 on chamber wall thickness. So if you plan to run a motor at 1,200 psi, the casing should be able to handle at least 2,400 psi. There is also a train of thought that says you should only use a safety margin of 1.5 to a maximum of 2. The theory is that if there is a problem with the motor burn, the casing will burst at a lower pressure if the margin is 2 or under, causing less damage due to lower pressure at burst. I understand the theory, I'm just not sure I buy into it. Having worked with high pressure cylinders for many years, and reading numerous reports on accidents, I'm not sure a tube bursting at 4,000 psi would present a significantly larger risk than one bursting at 2,000 psi. I'll leave that up to you to decide, but in my book a larger safety margin seems safer as it is much less likely to burst in the first place.

Forward Closure

Now that we have a motor casing, we will need to plug the forward end of the motor, of course the aft end will hold the nozzle. The most common approach is to use a plug made from 6061-T6 aluminum with o-rings to seal it in the casing. We need to determine how thick to make the closure, so let's go back to the Solid Motor Design Calculator, the top right function labeled Bulkhead Thickness Calculator. The first entry box is labeled MEOP, we may as well use the same value as our casing strength of 3,343 psi. Next, enter the casing diameter in the input box, for more safety you could enter the actual casing diameter of 1.75", but the inside diameter is actually only 1.62" so we'll use that value. Next enter the UTS for the material we are using, which again is 6061-T6 aluminum rated at 45,000 psi. Now click the Calc Bulkhead button.



After pressing the calculate button, we find the minimum bulkhead thickness is about .25 inches. This is for a bulkhead with no holes drilled through it. Drilling out a hole for a pressure port or delay grain would weaken the bulkhead, requiring a thicker bulkhead. The bulkhead will need to be longer than the .25" required, as we need room for o-ring grooves on the bulkhead. Using two o-rings is good insurance against leakage, I have got by with just one, but if you have blow by it will most certainly destroy the motor casing.

When you turn a bulkhead on the lathe, it will of course, have to fit inside the motor casing. While our casing should have an inside diameter of 1.62", it won't. There are always minor imperfections in the material. You can use a cylinder hone to true up the inside of a casing, but I think it's easier and safer to just turn the bulkhead down to the point is slides into the casing without force. I like to keep the bulkhead within a few thousandths (.003" - .005") of the casing internal diameter. If you turn the bulkhead too small, you risk o-ring extrusion while under pressure. What is critical, is making the o-ring grooves the proper depth in the bulkhead.

For o-rings to function, they need about 10% compression when in place. To get the proper compression, we need to do a little math. If we are using an O-ring with a diameter of .103" and 10% compression, that means the distance between the casing wall and the bottom of the o-ring groove needs to be .103 * .9 = .0927". The .9 is 90% of the original o-ring diameter, or 10% compression. Now double that because we're working on a cylinder, i.e., one cut of .1" on the lathe cuts .2" off the diameter. So, .0927 * 2 = .1854". .1854" is how much deeper the o-ring groove must be than the inside diameter of the casing.

What that means is we have to cut our bulkhead o-ring groove down to 1.62" - .1854" = 1.4346". The outside diameter of the o-ring grooves need to be turned down to 1.435" for the o-rings to have 10% compression.

Here's the formula in one line: Bulkhead/Nozzle O-ring Groove Outside Diameter = Casing Inside Diameter - ((O-ring Diameter *.9) *2)

As for the width of the groove, I usually cut them about 150% wider than the o-ring, as the o-ring needs room to flatten out when pressure is applied.

Here is a bulkhead I turned for a 2.25" diameter motor.

Speaking of O-rings, I generally use standard buna-N o-rings, I make them myself from o-ring cord from McMaster-Carr. While it is cheaper to make your own o-rings, the reason I do it is so I always have an o-ring of the correct size available. The original o-ring making kit had an assortment of cord diameters, and a small tool to make a nice cut in the cord, and then a little "V" groove at each end to align the two ends when you glue them together. While I have seen special o-ring glue available, I can't tell a bit of difference between it and regular super glue. In fact, I spoke to the owner of an o-ring manufacturer once, and when they had to make special size o-rings they did it the same way I did with off the shelf super glue.

I lube the o-rings with silicone grease prior to assembly. The grease helps the o-rings seal, and makes inserting a nozzle or bulkhead into a casing much easier. It's really easy to tear (or cut) an o-ring when sliding the nozzle or bulkhead past bolt holes or a snap ring groove. I use a counter sink tool with finger force only to take the sharp edge off holes in the casing, you can also taper the back side of a snap ring groove slightly to help prevent tearing. In my experience, the first o-ring to go into a casing I can twist and push, getting the first o-ring in without damage, but the second o-ring invariably does get slightly damaged. Bulkheads are the worst because there isn't much to hold onto and push them in, so a rubber mallet is often used to "assist the insertion".

Bulkhead/Nozzle Retention

Now that we have a casing and a bulkhead, we need a way to secure the bulkhead into the casing. I've used two methods, snap rings and bolt/pin retention. Both work well, but there are pros and cons of each method.

Snap Rings

Snap rings are probably the mainstay of retention methods, they are inexpensive, quick and easy to install and have very good strength. The down side to snap rings is that a lathe is required to cut the groove inside the casing where the snap ring rests. Not only is a lathe required, but you will need a lathe with a long bed, and a steady rest to hold the free end of the casing. The accepted practice is to cut the groove 1/2 (or just less) of the casing wall thickness. A first thought, would be that a groove cut to 1/2 of the casing thickness would reduce the pressure limit of the casing by 1/2. This isn't the case though. A cylindrical casing has about twice its strength in length as it does in hoop strength. So hoop stress is our limiting factor, and our snap ring groove is cut after the o-rings on the outside end of the bulkhead. So there is little hoop stress in the area of the snap ring groove.

To start cutting a snap ring groove, the groove should have a minimum margin (distance between groove and end of casing) of 3 times the groove depth. In our case, the snap ring groove will be 1/2 of the casing thickness, or, .065 / 2 = .0325" snap ring depth. So the groove should have a distance of .0325 * 3 = .0975" or about 1/10" space between the end of the casing and the start of the groove.

To calculate the stresses, I'm afraid we'll have to use a couple of formulas. To make sure the snap ring is strong enough, we'll have to calculate snap ring stress. Snap Ring Stress= (Bulkhead ejection Force)/(3.14 x [Snap Ring Thickness] x [Chamber Inside Diameter])

So in our case, using 3,343 psi chamber pressure, to find the bulkhead ejection force: Bulkhead Ejection Force = (3.14 * R squared) * Chamber Pressure

For our casing: (3.14x(.81x.81)) x 3,343= 6,887.1 pounds ejection force

Snap Ring Stress= 6,887.1 / (3.14 * .07 * 1.62)

Snap Ring Stress= 6,887.1 / .356076

Snap Ring Stress= 19,341.6 psi

The snap ring I choose is .07" thick, and is hardened steel with an UTS of 90,000 psi. So our snap ring stress is well below the limit of our snap ring and should work fine.

Next, we need to look at the stress the snap ring creates in the chamber. Here's the formula for that.

Snap Ring Stress in Chamber= Bulkhead Ejection Force / {(3.14/4) x (Chamber Outside Diameter(squared) - Groove Diameter (squared))}

And in our case.

Snap Ring Stress= 6,887.1 / (.785 x (3.0625 - 2.418025))

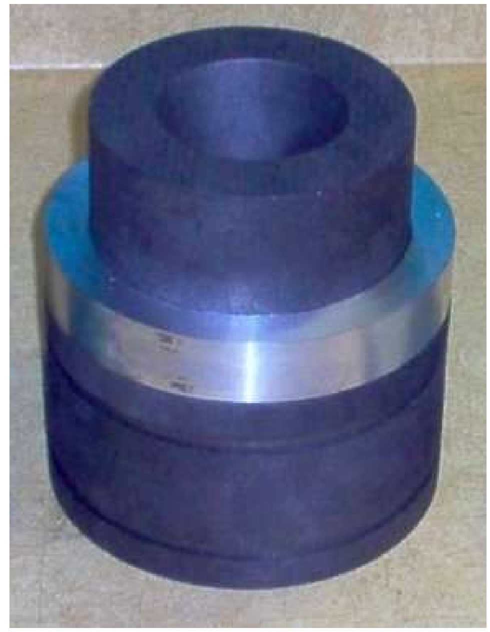
Snap Ring Stress= 6,887.1 / (.785 x .644475)

Snap Ring Stress= 6,887.1 / .505913

Snap Ring Stress= 13,613.21 psi

So the stress created by a chamber pressure of 3,343 psi on the casing material is 13,613.21 psi. We are using 6061 aluminum, with an ultimate tensile strength of 45,000 psi. So we're well within the limits for this metal and pressure.

A steel nozzle may simply be retained by a steel snap ring. But if you plan to use a graphite nozzle, a support of some sort should be used as graphite doesn't have enough strength in most cases and the graphite will crumble at the snap ring. A steel or aluminum support ring is generally required with graphite.



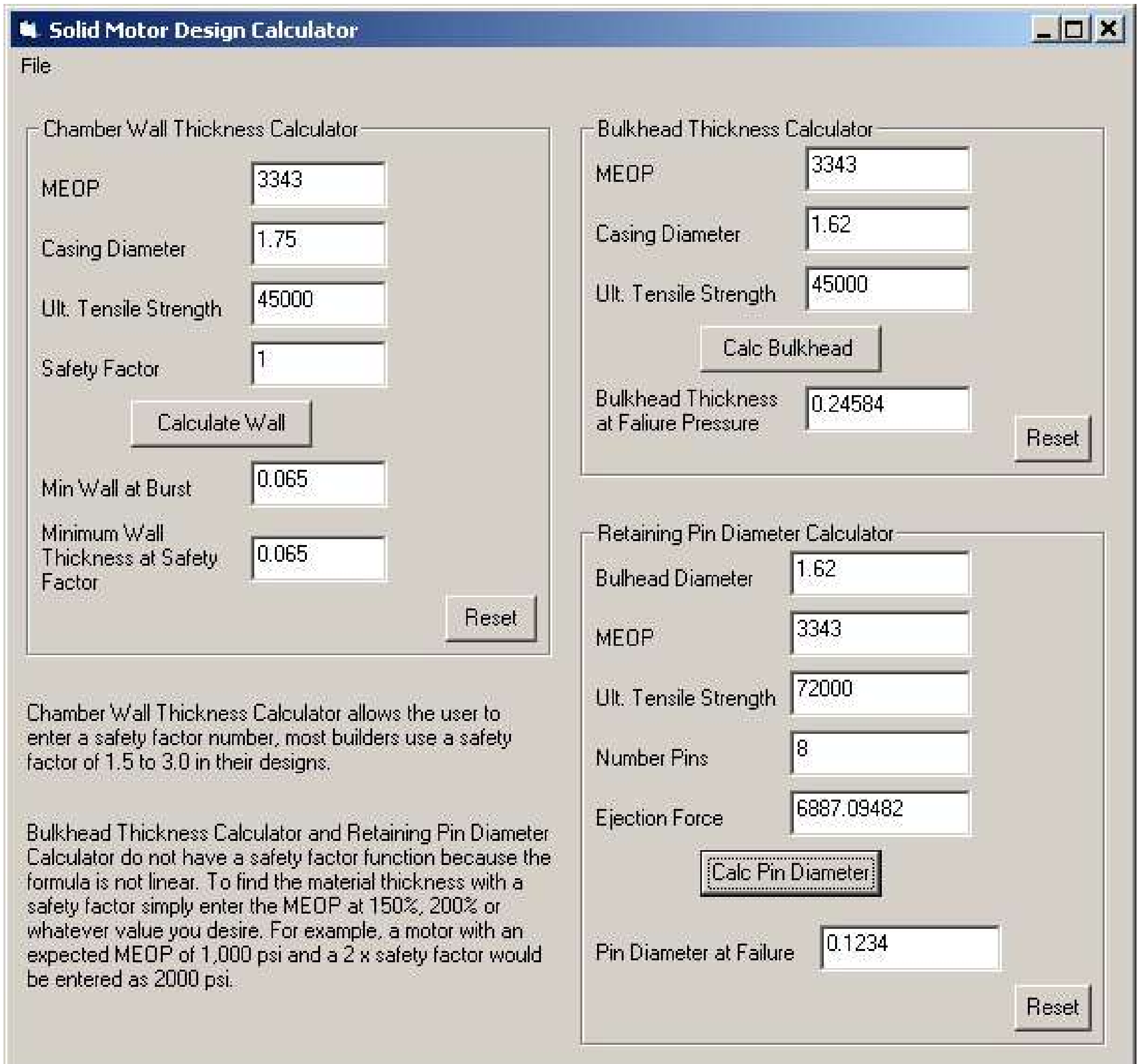
On the left is a 2.25” diameter nozzle that uses a nozzle washer to bear the snap ring load. On the right is a 6” diameter nozzle that use a heavy aluminum ring to bear the load from retaining bolts.

Bolt or Pin Retention

Using bolts or pins to retain a nozzle or bulkhead has the advantage of ease of manufacture, in that a small drill press and some sort of jig to hold the casing is all that is required. The bolts also have good strength and many different sizes are available for virtually any motor configuration you can imagine. In fact, I used (18) hardened steel 5/16" bolts in my 6" diameter "O" class motor with great success. The main disadvantage to using bolts or pins is that the bolt heads extend outside the casing diameter, which means you have to make the motor mount tube in a rocket larger in diameter. To minimize the motor mount tube diameter while using pins or bolts, I will typically grind the heads down to under .1". This helps, but still doesn't totally alleviate the problem. In some motors, I would use a snap ring at the forward closure, then use a bolt pattern at the nozzle (which is outside the motor tube) to keep the motor mount tube at its minimum diameter.

The holes in the casing should be located at least two times the diameter of the hole size from the end of the casing. If we use .125" bolts, we should leave at least .25" from the hole to the end of the casing. This hole pattern needs to be absolutely perpendicular to the length of the casing. I use jig made from angle iron, which has a flat welded on one end to hold the casing in place while drilling. The jig is clamped to the drill press table to prevent movement.

Let's go back to the Solid Motor Design Calculator one last time.



In the Retaining Pin Diameter Calculator enter the bulkhead diameter of our motor at 1.62". Now enter the MEOP, which earlier we calculated at 3,343 psi, next enter the UTS of the pin or bolt material (you'll have to get that from your source), then enter the number of pins you want to use. Try to use the minimum number of pins while keeping the pin diameter at a reasonable size. If you put in too many pins, you end up having the end of casing looking like Swiss cheese, and reduce the strength of the casing.

If we use 8) pins with a diameter of .125" that will create 1" of holes around the diameter of the case. .125" * 8 = 1". While our casing circumference is 5.495", 3.14 * 1.75" = 5.495". I'd try to keep the hole to circumference ratio at about 1 to 4 or better. In our case the ratio is over 1 to 5, so we should be fine.

You can play with the number of pins, until the number and diameter are suitable to you. Keep in mind if you use bolts, the actual diameter of the bolt is less than it's stated bolt diameter because of the threads, you'll have to reference your supplier for exact bolt dimensions and strength. I usually thread the outside of the casing so I can screw the bolts into the casing, then let the bulkhead butt up against the bolt. This works, because most of the forces acting on the bolt are shear forces only. You certainly may drill and thread into the bulkhead as well, and it likely would be somewhat stronger. But if you want to use a given bulkhead in another motor casing, it's hard to drill out the casing holes in the exact same location as the holes in the bulkhead.

Here is the nozzle and bolt pattern on my 6" diameter motor. The aluminum retainer was 1" thick and 1" deep. This bolt circle held some 14,000 pounds of force.

To make sure my aluminum retainer was strong enough, I calculated the surface area of the retainer in contact with the graphite at 8.143 square inches. With 14,500 pounds of force on the nozzle, that left 14,500 / 8.143 = 1,780.7 psi of force on the graphite. Depending on the grade of graphite you use, it has an UTS of between 2,300 to 4,300 psi.

I didn't take into account the fact that the nozzle has a nice size hole in it (the throat), in reality the actual force would be less than I stated, increasing the safety margin even more.