How to Design Pressure Vessels, Propellant Tanks, and Rocket Motor Casings

Half Cat Rocketry Technical Resources

Authors

Austin Sennott

Charles Sharp

Introduction

Much has been written on the theory of pressure vessels, but very little about their physical implementation on a small, simple scale. This guide will approach the subject from the perspective of an amateur who has reasonable familiarity with high power rocketry and is looking to design a “DIY” propellant tank, combustion chamber, or solid-propellant motor. References to liquid fuels and oxidizers are intended for engines based on nitrous oxide and alcohol or similar fuels. Do not use this document as a reference for cryogenic service or non-standard conditions.

Scope

The first part of this document gives an overview of the first-principles approach to designing a pressure vessel to contain fuel, oxidizer, or combustion in an amateur setting, and will cover bulkheads, O-Rings, thermal liner interfaces, pressure ports, pistons, and concentric tanks.

The second part covers the process of analyzing the stresses present in pressure vessel casings, closures, fasteners, and casing-nozzle interface surfaces. It assumes that you are working from a known pressure and does not include any design information about the internal ballistics of a solid rocket motor itself – i.e., grain geometry, regression rate, propellant flow, or nozzle throat diameter.

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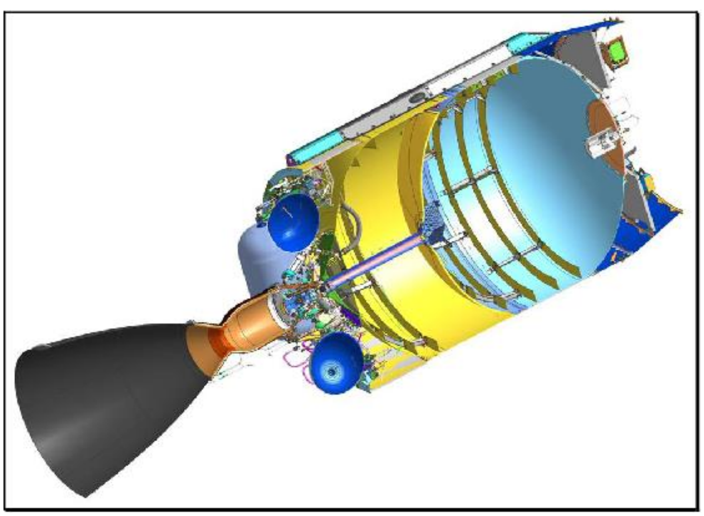
Section 1: Theory and Design

This section explains the principles of creating a pressure vessel for amateur rocketry. As a general safety disclaimer, **people should never be near a container when it is pressurized**. This is the *most important* part of any safety procedure. No matter how well-built it might be, there is no reason to introduce the risk of injury or death. The only exception is for DOT-rated bottles, like the supply tank used to store and transport nitrous oxide.

For more information, see the Safety page on our website.

Pressure Vessel Basics

The purpose of a pressure vessel, whether it be a fuel tank, oxidizer tank, pressurant tank, combustion chamber, or other container, is – as the name implies – to hold and store pressurized fluid. A spherical pressure vessel is structurally the most efficient, able to contain the greatest volume per unit container. However, a spherical vessel is beyond the fabrication ability of most amateurs and not practical for most rocket applications; as the volume requirement increased, a rocket with only spherical tanks would become comically bloated to contain the large diameter.



Source: IEEE

Therefore, most propellant tanks are designed to be cylindrical, the next best option. A cylinder with two hemispherical domes is the most efficient in this case, but again it is not necessarily the most practical option. In the amateur world, where cheap, simple, and easy to make are often the driving factors, structural mass is much less of a concern than it is for a high-performance orbital vehicle.

When starting from scratch, it is best to begin at the very basics of what the pressure vessel must do and then work toward a more refined design, based on what your project requires and allows for.

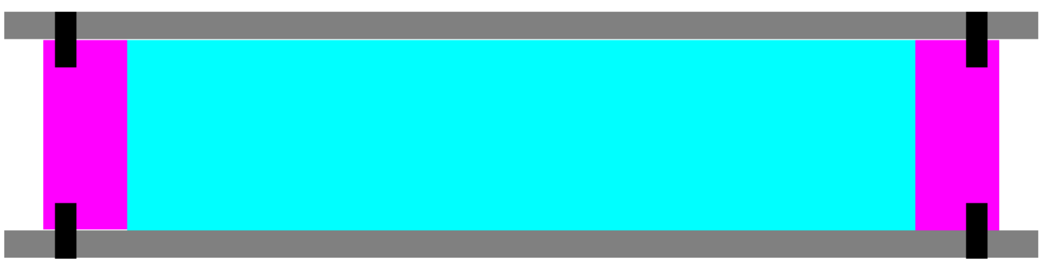
Components

In its absolute most simple form, our pressure vessel has three components: two **bulkheads** and a **casing**. Together, these three pieces form a container to house the **fluid**, which, when pressurized, will push outward in every direction.

Look at the cross-sectional diagram above and imagine what would happen if this vessel were to exist in real life. Remember, there are <u>no other components</u> present right now. There are two main issues.

The first problem is a basic force balance: With nothing resisting the force applied on the internal sides of the bulkheads, they will be ejected from the casing. This leads us to the first necessary feature: **Bulkhead Retention**.

In this scenario, we add **bolts** through the casing wall and into the bulkheads:



Now with that problem solved, what is the second issue? It may not be immediately obvious if you haven't worked with pressure vessels before. Bear in mind that no piece of hardware is ever perfect. The outer diameter of the bulkheads will be slightly less than the inner diameter of the casing so they slide in smoothly; the casing tube itself may be out of round and leave small gaps; and there will be microscopic scratches and imperfections in the surface. The problem, in short, is that fluid will find its way through all those tiny openings.

The solution is to install a material which deforms under stress and squeezes into every microscopic pathway: **O-Rings**.

O-Rings are typically made from an elastomer such a nitrile rubber or silicone. It should be noted that O-Rings stop functioning properly at cryogenic temperatures, but there are other materials and sealing methods for that case. They will also melt at high temperatures, and so must be protected from direct exposure to hot gases.

Let's add **O-Ring glands** to our bulkheads:

(Pictured here is just the gland cut into the bulkhead; the O-Ring gets installed into the gland)

Our simple pressure vessel is now complete. With fasteners and O-Rings installed, it will contain the fluid and hold it in a static condition as long as the pressure stays under the failure point.

<u>Always consider where fluid is trying to escape from.</u> If you reverse the placement of bolts and O-Rings in the above diagram, pressure will flow around the threads and out the side of the casing.

Bulkhead Retention

Snap Ring

There are several ways to retain bulkheads. One of the simplest methods, found in many small solid rocket motors, is a **snap ring**.

A snap ring is compressed using specialized pliers, placed into the groove on the inside of the casing, and relaxed into place. Under pressure, it sits up against the outermost lip of the groove and distributes force roughly even around the circumference.

Some of the *advantages* of snap rings:

- Low part count. Only one needed per bulkhead.
- Simple to calculate safety factor.
- Easy to install at small sizes. (~2 inches or smaller)
- Reusable and relatively cheap.
- Can simplify bulkhead design.

Some of the *disadvantages* of snap rings:

- Cannot be machined very deep into a casing and may be impossible to machine if a casing is too long to put in a lathe or mill.
- Difficult to install at large sizes. Often requires some ability for the bulkhead to be pushed further in, so that the plier tips can fit into the holes on the snap ring, and then pulled back to seat against the snap ring.
- Can cause difficulty while installing a bulkhead with O-Rings due to the O-Ring expanding into the groove then needing to be re-compressed back to the casing inner diameter.
- Groove may tear O-Rings during bulkhead installation.

Bolt Circle

Half Cat Rocketry's preferred retention method is a **bolt circle**. This can either be part of the bulkhead:

Or implemented with a separate **retaining ring**:

Both options achieve the same function, so which is used comes down to the specifics of your hardware design.

Some of the *advantages* of bolt circles:

- Somewhat simple to fabricate. Far fewer limitations on casing size.
- Straightforward to calculate safety factor.
- Easy to install.
- Reusable and relatively cheap.

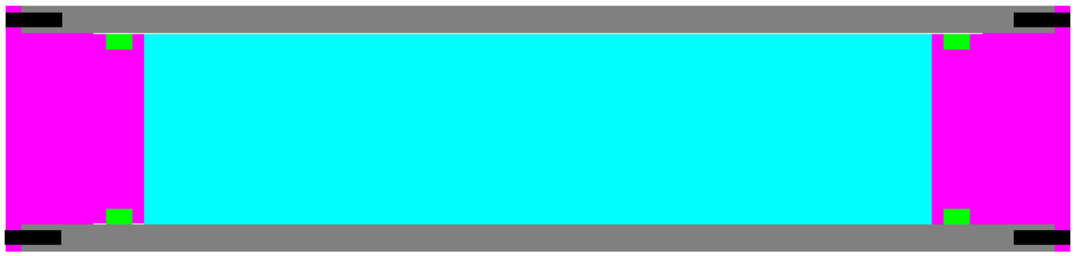
Some of the *disadvantages* of bolt circles:

- Casing holes may require significant widening or elongation from initial size if they are not accurately positioned compared to the holes in the bulkhead.
- Care must be given during bulkhead installation to avoid chewing O-Rings as they slide past the casing holes.
- High part count for large diameters.
- Threaded holes in aluminum bulkheads may be sensitive to cross-threading.

The type of bolt used in a bolt circle matters somewhat. Preload of a button head screw on the outer casing wall is not taken into account in the calculations of this document, but the socket of a set screw needs to be considered. Choosing a type of bolt comes down to the use case; button heads protrude, while set screws are flush. See Fastener Selection for more details.

Bolt Flange

One other style of retention, often found in professional settings with large budgets and greater resources, is the **bolt flange**:



Here the bolts are installed axially into the casing. There are other variations on the flange where the bulkhead and/or casing diameter is larger at the ends. In some cases, threaded rods may be run in tension between two bulkheads for retention.

Some of the *advantages* of bolt flanges:

- Relatively easy to install. The casing has a smooth internal bore with no other features for O-Rings to contend with. The bolts can also be used to pull the bulkhead down evenly and methodically.

Some of the *disadvantages* of bolt flanges:

- Can be hard to fabricate accurately, depending on the available machines.
- Requires a larger diameter of material stock for the bulkhead.
- Requires that the casing have more wall thickness at the ends.
- Not as simple to calculate since the bolts are engaging as threaded fasteners vs. simple pins in bolt circles. See NASA-STD-5020.
- High part count, especially when including washers and lock washers.

Other Methods

There are other ways to retain and seal bulkheads. Some of these include:

- Threaded bulkheads / internally threaded casing
- Threaded retaining cap / externally threaded casing
- Welding

- Face seal O-Rings
- Flat gaskets

Notably, threaded casings are very commonly found in commercial reloadable motors such as Cesaroni brand. However, these features are only present in a minority of amateur projects and as such are left as an exercise for the reader.

Bulkhead Design

Let’s take a closer look at some different bulkhead designs.

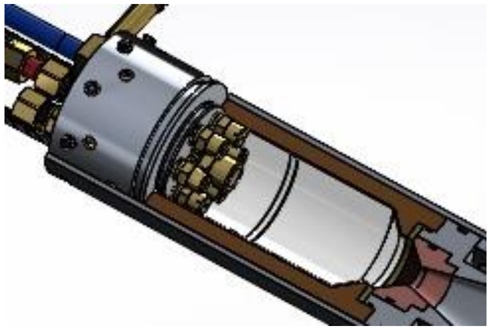
Case #1:

Case #2:

Case #3:

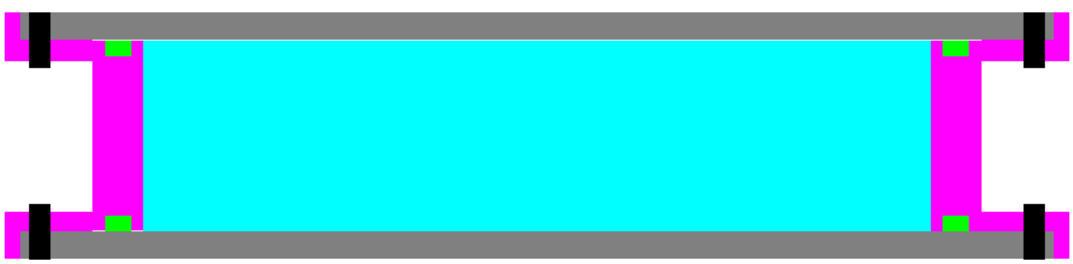
The difference between #1 and #2 is a practical usage which won’t appear until you have manufactured and fully assembled the pressure vessel. After installing bulkheads with O-Rings, and possibly thermal liner adhesives like RTV, the friction of the seals plus any built up soot (if it’s a combustion chamber) can make the disassembly of #1 extraordinarily difficult. The only part to grab onto will be any fittings or protrusion from the exterior side, which may not be enough to grip.

An example of #1 is Half Cat’s injector bulkhead:



Oftentimes this kind of bulkhead needs to be rammed out from the other side, which can present quite a challenge in trying to avoid damage. The remedy is to use #2, where the bulkhead includes a wider diameter lip of sufficient thickness that a screwdriver or other tool can be used to pry it out. This can be found on 1Cat/3's tank bulkheads:

The main disadvantage to #1 and #2 is that they include a lot of extra material, although with some extra design and manufacturing time it can be made to look like this mass-optimized design:



Here we see why one might choose a snap ring or case #3 from above: They allow the greatest amount of material to be removed with simple machining operations on a lathe. If mass reduction is indeed a priority, #3 can be finely tuned to allow for a conical or hemispherical dome retained by a separate bolted ring. An example of this is 2Cat/3's tank bulkheads:

In general we prefer using bolt circles over flanges and snap rings unless one of those has a specific benefit that makes it outright superior for a particular design. Examples of both can be found in the engines of 1Cat/3A and 1Cat/3C, respectively:

(Snap ring is not installed on 1Cat/3C (right) but you can see the groove for it)

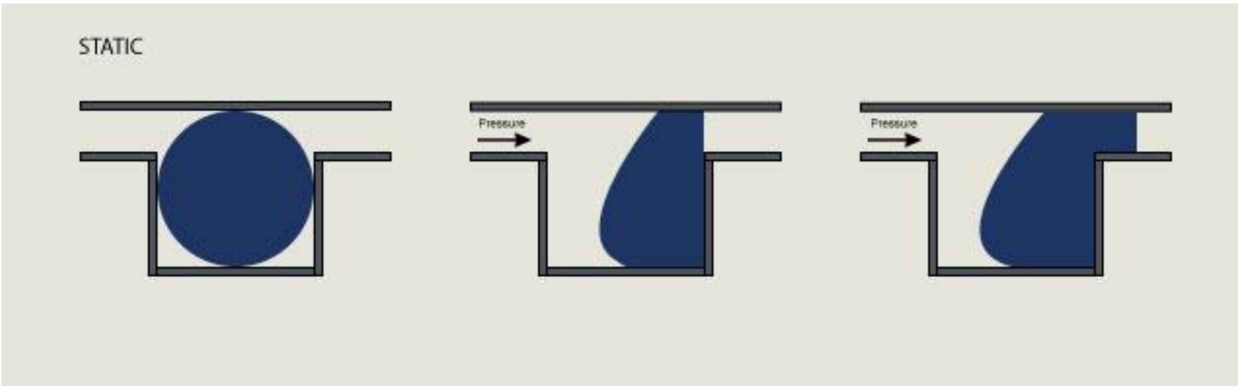
The pressure that every design can handle is dependent on the numbers of bolts, bolt size, edge distance, and material strength. For equations and more information on how to calculate the stresses, see the second part of this guide.

There are of course other variations which you may be able to derive from the basic architecture. In addition, there are some further considerations which will be covered in the following sections. Once you understand all the principles that go into making a bulkhead, you will have a large catalog of options to choose the one that best suits the needs of your pressure vessel.

O-Rings

Basics

How does an O-Ring work to seal a pressure vessel? Take a look at this graphic.



Source: Barnwell

Under pressure, the elastomer is deformed into the gap between bulkhead and casing. As more pressure is applied the seal becomes even tighter, which is why O-Rings work so well. Too much pressure can cause the O-Ring to extrude into the gap and fail, but this is unlikely to be seen for an amateur project.

Achieving a proper seal relies on the O-Ring being intact, in good condition, and free of debris. However, they are very forgiving and even O-Rings with some damage and debris can work perfectly well, as will casings, bulkheads, and glands that are out of tolerance specification.

Redundancy

Oftentimes you will see two or more O-Rings used to seal a bulkhead; this is done so that if the first (innermost) seal were to fail, there is a backup to prevent leakage. It is not strictly necessary to include redundant O-Rings – many of Half Cat Rocketry's designs only include a single O-Ring per sealing point. It is a matter of risk vs. burden: If the chance of unknowingly compromising a single seal is low, and the added bulkhead material, O-Ring cost, or installation difficulty is great, then it is perfectly acceptable to use only a single seal with no redundancy. The location and use case also matter greatly. Redundant O-Rings are more important in combustion chambers and other high-temperatures applications where the seal may be degraded or melted over time, in which case the backup is used to extend the life of the seal.

Material Selection

We must remember that this is amateur application, and the stringent rules set forth by the aerospace industry do not always apply. There are recommendations for material compatibility, but unless people will be in close proximity while your pressure vessel is loaded (this should **never** be the case), more lenient amateur conventions can be employed. Half Cat Rocketry has never experienced a failure resulting from O-Ring material incompatibility – but even if this were to occur, it would simply be considered a known risk to hardware.

Our two main choices for O-Ring material are Buna-N (a.k.a. nitrile, NBR) and silicone. All fluids that we utilize (nitrous oxide, carbon dioxide, isopropyl alcohol) do not exhibit any detrimental effects in contact with these two rubbers. Which you choose depends on the specific application; standard silicone is softer than Buna-N, which make it easier to compress and install but more vulnerable to chewing on bolt holes; Buna-N is by far the cheapest material, and so may be more economical in designs which have large numbers of O-Rings.

One other common material is Viton (fluorocarbon). It feels and behaves very similar to Buna-N, but has much better material compatibility especially with regard to oxidizers. The price is comparatively steep, but not absurd (see: Kalrez).

Although this guide is not intended to be applied to cryogenic fluids, it is worth giving a brief mention of the suitable material. PTFE (Teflon) O-Rings are usually the best choice as they are non-reactive and non-elastomeric, although the design of O-Ring gland may be different since they cannot be stretched and relaxed into place the same way as a normal O-Ring.

Lubrication

A lubricant is needed during installation to help prevent chewing or tearing as the O-Ring slides past internal features, and it's just really difficult to install a bulkhead without lubricating. The exact lubricant is not particularly critical as it has no substantial effect on sealing properties for our purposes.

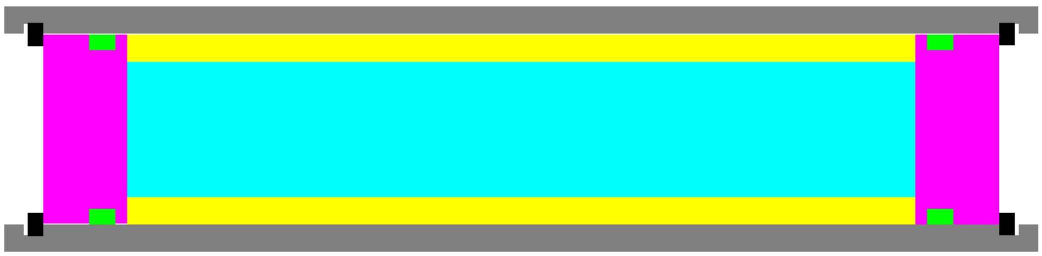
For Buna-N, silicone grease is a very common and available option.

The open question of what silicone grease potentially does to silicone O-Rings led to choosing Red "N" Tacky #2 lithium grease for Half Cat, an absurdly cheap albeit messy lubricant. We cannot officially *recommend* that you use it for vessels containing nitrous oxide, but thus far it has not resulted in any issues for our hardware (although it does seem to slightly dissolve in alcohol, which isn't an issue for our engines).

The proper (but higher priced) grease to use in oxidizer tanks is Krytox. It is non-flammable and small tubes can be found online for a decent price.

Thermal Liner Interfaces

Thermal liners are a necessary part of ablative combustion chambers. The most often overlooked yet crucial feature of them is the interface where they seal to a bulkhead or casing to prevent hot gas from bypassing the liner and attacking a structural component.

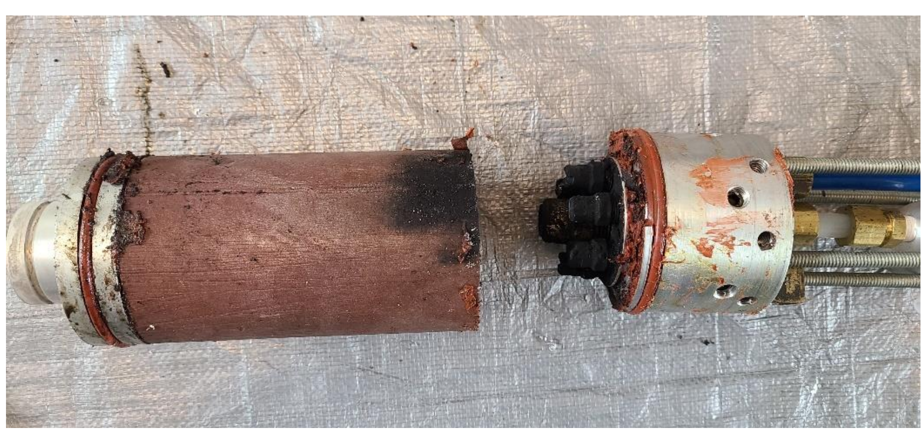


Shown here is a combustion chamber with a **Thermal Liner**.

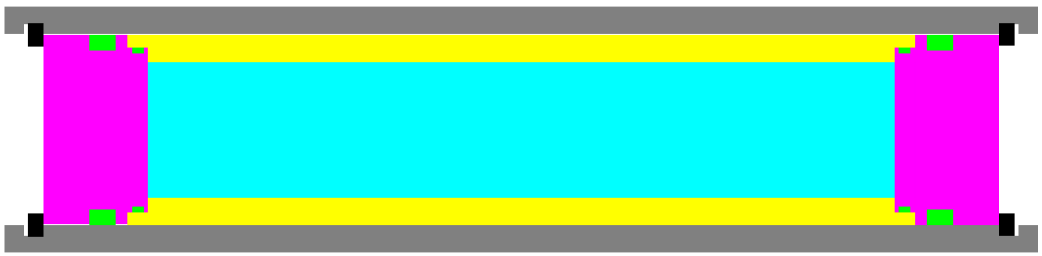
(The nozzle is assumed to be inside or in place of one of the bulkheads)

As it is now, it will not protect the casing wall for very long. Hot, high pressure gas will find it way through the interface between liner and bulkhead and rapidly heat up the wall until it fails. This is especially the case in a liquid engine where the exhaust products are purely gaseous and temperatures are usually higher than a solid or hybrid motor. In a single-use chamber the liner could be epoxied to the bulkheads as a seal, but this makes it difficult or impossible to remove and re-use any engine components.

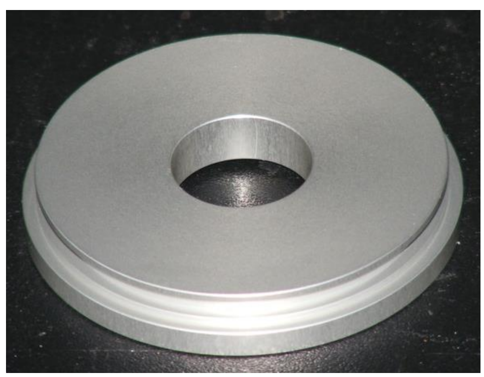
A less-permanent way to adhere and seal these together is by using room temperature vulcanizing (RTV) gasket maker. It cures to a silicone rubber that provides a decent seal, although its usability depends on the conditions and runtime of the engine. RTV was used in Half Cat's chamber, eventually replacing thermal liner O-Rings entirely.



In this post-firing picture you can see the cured RTV on the injector and liner, and a scorch mark from where some combustion gas made its way past the seal and began burning the exterior of the liner. Fortunately, the low temperature and pressure, combined with a limited runtime, meant that no damage was done to the casing wall. You may notice that there is actually an O-Ring groove on the bulkhead; this is because it was not originally designed for RTV but was instead intended to use a silicone O-Ring on a smaller diameter feature which fits into the thermal liner.



Now, with a slight modification, the bulkheads neatly fit into the liner with a smaller cross-section O-Ring. The liner does not have to be stepped; it works just the same to have the bulkhead reduce all the way down to the inner diameter of the liner. Such an interface can be found on larger diameter Aerotech motors' forward seal disks:



Source: Sirius Rocketry

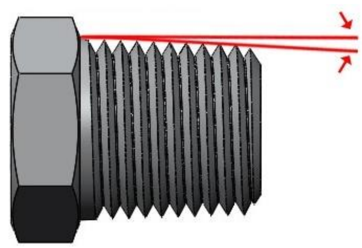
Sealing to the thermal liner does mean that it will effectively become the pressure vessel and make the bulkhead O-Rings obsolete; however, the casing is still a necessary structural component, and, as the Half Cat liner image shows us, the seal may still have small leakage which must be stopped from exiting the casing.

Pressure Ports

Most of the time, your pressure vessel will need a way to pipe fluid into and out of itself – this is where the pressure port comes in. In our application, they're fittings that seal into the vessel. To avoid a lengthy discussion on the pros, cons, and cult-like obsessions that come with each type of fitting, this section will give a brief overview of what you need to know when designing and building an engine.

NPT

National Pipe Taper threads are the simplest and probably most common high-pressure fitting. The thread profile is conical and designed with an interference fit so that as you tighten it down, the threads deform and create a metal-to-metal seal.



Source: Sanitary Fittings

NPT requires the use of PTFE (Teflon) tape, which is wrapped around the threads. National Pipe Taper Fuel (NPTF) threads are made to tighter tolerance and do not require PTFE tape, but it is still a good idea. We make no claims of being an authority on fittings, but the following is knowledge gained by experience. If you do research on the subject of NPT you will find many cautions, such as these threads being difficult to tap deep enough, or what products to apply to avoid galling, or that you cannot uninstall and reinstall these fittings multiple times. Some of this may be true for applications for quality, consistency, and customers matter, but they are not hard and fast rules.

For aluminum-based parts (which should be the vast majority of your machined parts), brass and stainless NPT fittings can be tightened down, taken out, and reinstalled many times with PTFE tape re-applied each time. The tap depth of the female thread can be inconsistent and still seal fine – it is technically possible, given enough torque and willpower, to seal with just 1-2 threads tapped and engaged.

We use NPT heavily for a few reasons:

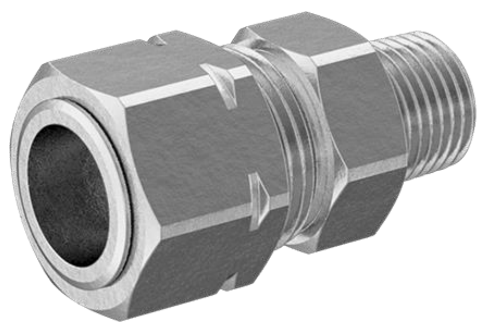
- It is a simple and sure-fire way to seal into a machined part (like a bulkhead).

- Most fittings have NPT on at least one side.
- Our engines are small enough that hand-tapping NPT is feasible.
- NPT fittings are the most common online.

The short of it is that when you have a block of metal, and need to put a fitting in it, you really only have a choice between NPT and a straight thread with O-Ring to achieve retention and sealing – and NPT is what you'll find on sites like McMaster-Carr 99% of the time.

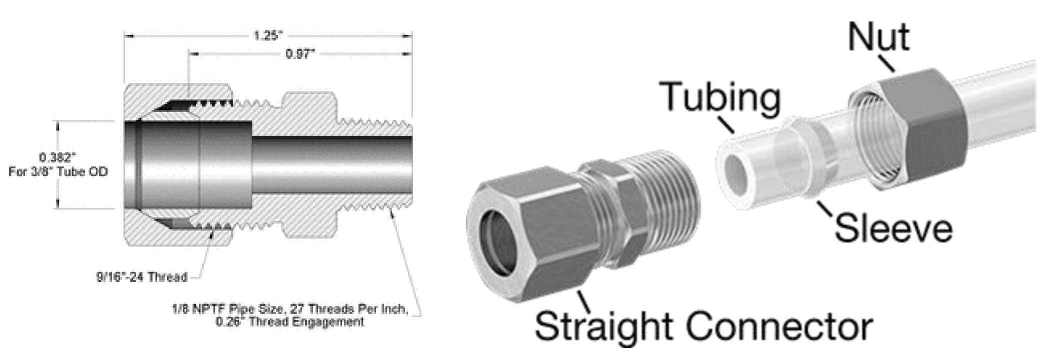
Tapping NPT can be a pain, especially at large sizes, but we stick to the rule "Just do your best, it's gonna be ok." The key to not getting worked up over a fitting is to understand that an amateur project is never going to be perfect, and plan accordingly.

Compression



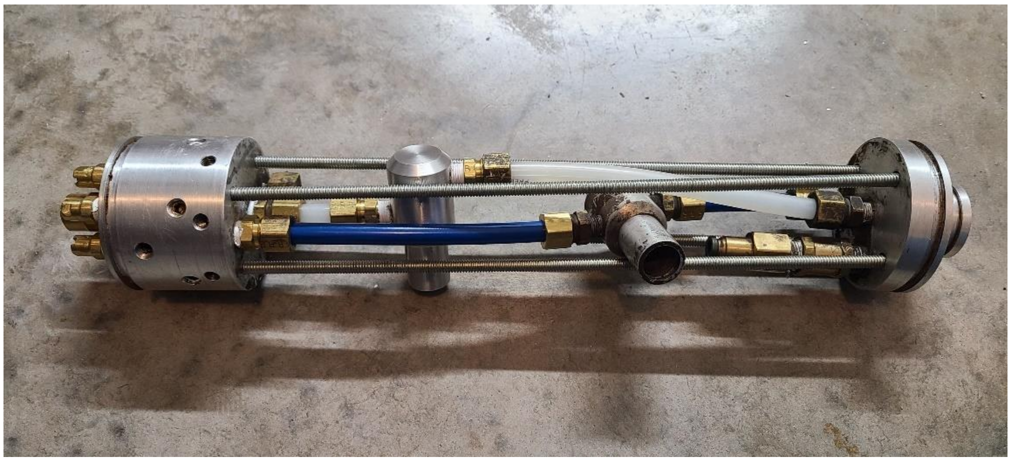
Source: McMaster-Carr

Compression fittings work by compressing a ferrule onto a tube using a special nut; this provides enough pressure to seal, and enough friction to hold the tube in place.

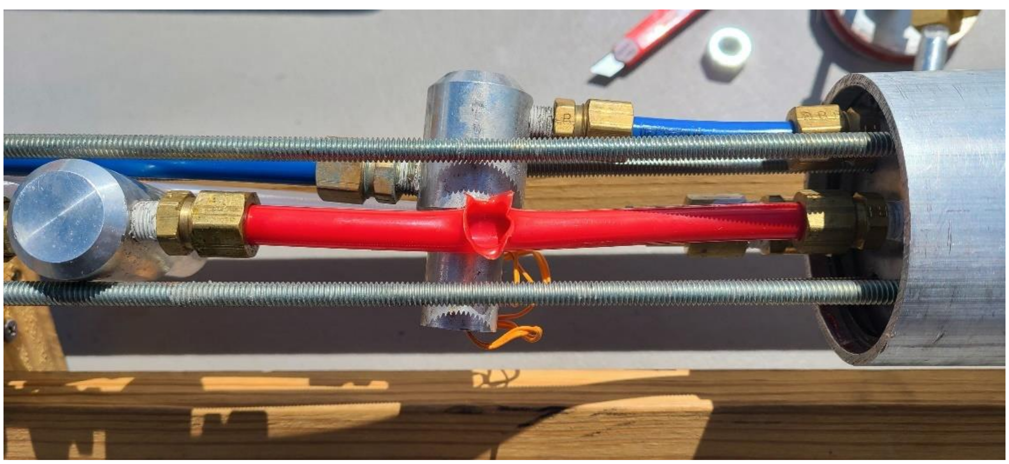


Source: McMaster-Carr

Since it allows you to put in your own metal or nylon tube, flex- and hard-lines can be cheaply built and customized using compression fittings. Unlike NPT, the tube can be positioned rotationally before the nut is tightened down, so that it's not constrained to a particular rotational position. Also, the nut can be loosened and the tube repositioned or replaced without taking the fitting out of its base part.



One example is in Half Cat's plumbing, where nylon lines allowed for flexibility in a very space-constrained assembly. It also allowed tubes to be replaced with higher-pressure versions when some burst.



It is important to know that metal tubing (like copper) must go straight into a compression fitting to seal. If it is bent or deformed just before entering the fitting, its whole circumference may not be in contact with the ferrule and cause leaks. This was the problem with Half Cat's original copper plumbing.

Also note that there are different ferrules for metal tubing vs. plastic tubing. Brass (left) must be used for metal, and Delrin (right) for plastic. There are other types and brands of fittings and ferrules out there, but these are the ones you will find from McMaster-Carr:

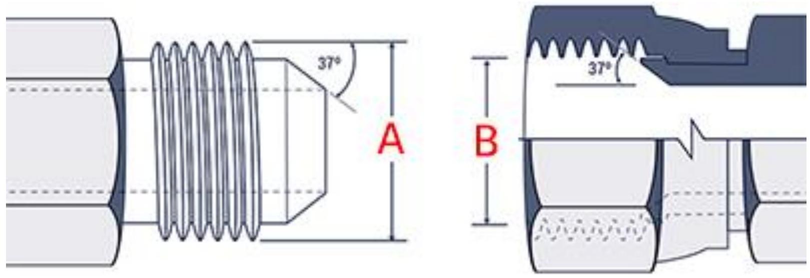
Swagelok

Swagelok is a brand of compression fittings. They work well, they're expensive, they do just about the same thing as similar products, but the company wants you to use theirs exclusively. Swagelok is basically the Apple of tube fittings.

[Source: Swagelok]

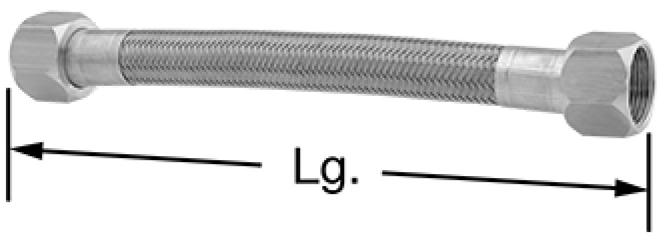
JIC / AN / 37° Flare

This is the kind that you will see on a lot of rocket engine plumbing. The straight threads compress a conical metal seal, so it is reusable and rotationally positionable, as with compression fittings.



Source: Tameson

Flared fittings are most commonly found on stainless steel-braided PTFE flex hoses:



Source: McMaster-Carr

These hoses are the gold standard for flex-lines, although they usually don't work for our designs because we tend to pack a lot of plumbing into very tight spaces where they simply would not fit or be able to bend tightly enough.

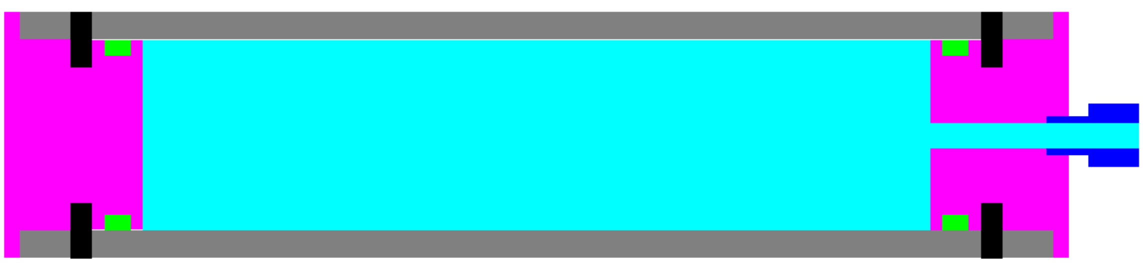
Ad-Hoc Fittings

There are plenty of other ways to fasten, seal, and join to parts together. One non-standard case was in Half Cat's forward oxidizer tank bulkhead, which needed two threaded rods to protrude up and form an attachment point for the load cell (in the test stand) and the airframe (in the vehicle). The simplest way to achieve this was to drill and tap two ¼-20 holes in the bulkhead, then thread in long hex bolts with an O-Ring under the heads.

Once screwed all the way to the end the O-Rings were compressed, forming a kind of face seal which never showed any sign of leakage. With some imagination, you can extrapolate how one might create a custom fitting of arbitrary size using standard straight threads and O-Rings.

Pressure Ports in Bulkheads

The diagram below shows one way you might install a pressure port. This example is common in Half Cat Rocketry designs: It is a **compression fitting** threaded into a tapped NPT hole, which can be connected to a metal or plastic tube and routed to valves or other pressure vessels.

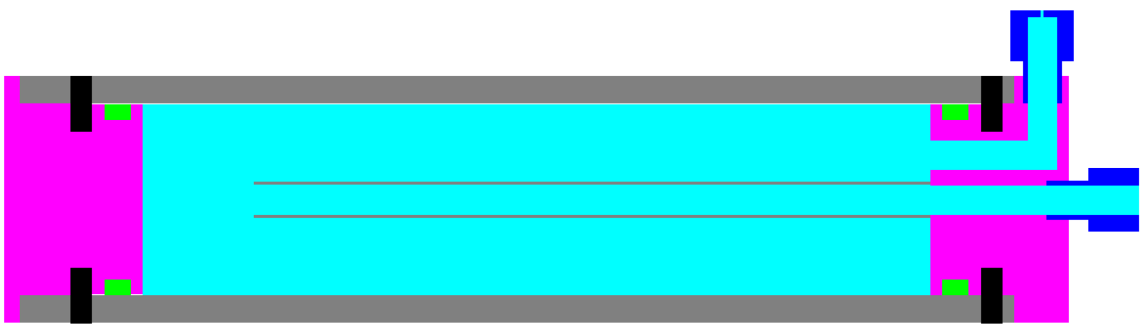


The port does not have to be in the center of the bulkhead, although that is the easiest location to put one since it can be easily done while the part is in a lathe. Nor does it have to enter axially at all, injectors often require fittings installed on the side for fuel or oxidizer.

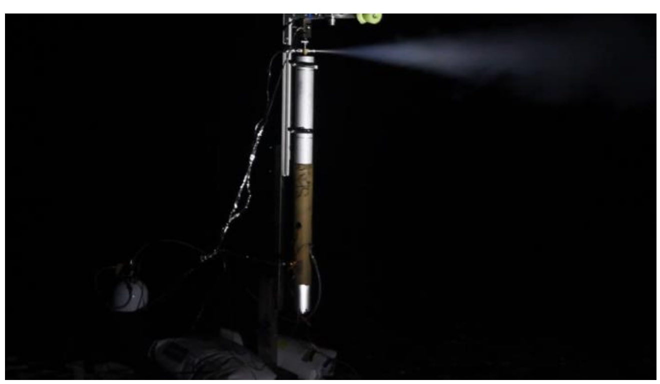
Static Vents

This concept is extremely important to understand when designing a pressure vessel for a fluid like nitrous oxide. It is not only necessary to fill a tank properly, but also a vital safety feature to prevent trapped pressure in the system.

In the following diagram, let's say that the right side is the top (relative to gravity). We are filling nitrous oxide through it using a dip tube to the bottom. If the tank is completely sealed to the outside world, how will it become full? The answer is that it won't. It will compress the air inside until it reaches equilibrium pressure with the nitrous oxide, at which point no more liquid can enter the tank.



What we need is to let the air escape out the top so that the liquid level will rise until the tank is full. Operationally, this is marked by a visible white cloud suddenly appearing from the static vent as liquid nitrous oxide sprays out.



The static vent, physically, is a small orifice at the top of the oxidizer tank. It can either be drilled directly into the casing wall or into a fitting which comes from the top bulkhead. This image shows the location of Half Cat's vent – it's an NPT plug with a 0.8mm hole, threaded into the tee-fitting which comes from the forward bulkhead and connects to the CO_2 purge line and pressure transducer.

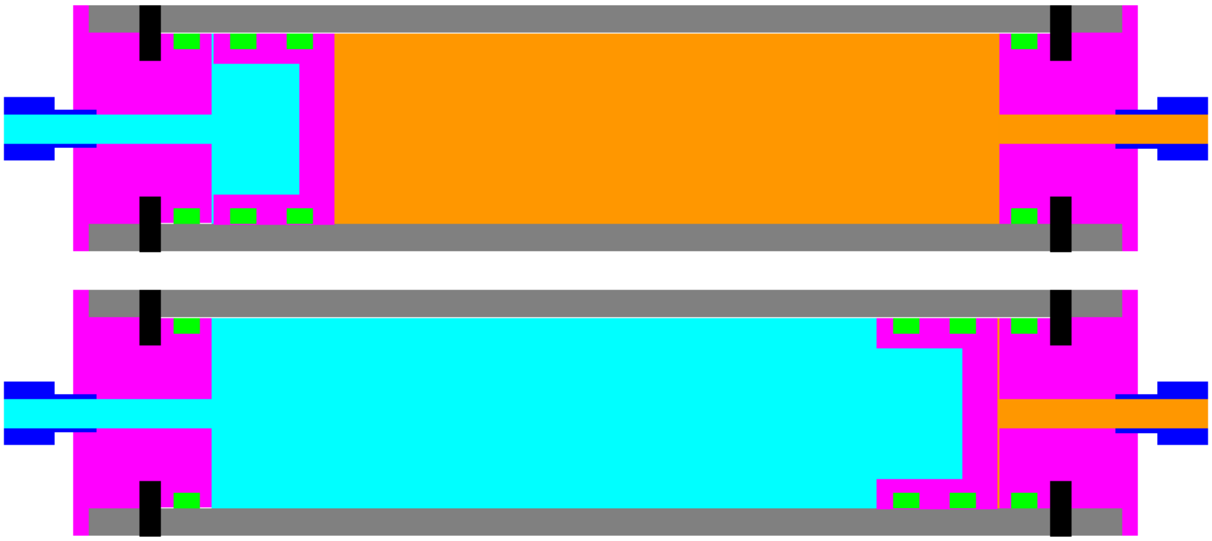
The same process can be done with a controlled valve, but the reason to use a static vent is that in case all power fails or control is lost (which always happens at the worst times), the tank will eventually empty itself by continuously bleeding pressure until all liquid has been expelled. This is an excruciatingly slow process (40+ minutes for Half Cat to vent a full tank) but it *will* fail safe because the static vent cannot be closed. In case of an abort after filling, we recommend that the fill system also include a drain valve to quickly empty the tank.

Contrail hybrid motors, the only remaining brand of commercially available hybrids at the time of writing, uses this concept in its motors for the reasons described above. Famously, and somewhat hilariously (as hilarious as a potential safety hazard can be), at the 2018 Spaceport America Cup the Texas A&M hybrid rocket – which did not have a static vent – failed into a closed state with the nitrous oxide tank full. Without connection and unable to approach, the range had no choice but to remotely puncture it using a rifle. Since we would like to not include firearms in our list of safety equipment, we always use a static vent on nitrous tanks.

A fuel tank (for alcohol or similar fuels) does not need a vent since it is non-pressurizing on its own. The static vent is a designed leak which is used to prevent an unsafe situation where pressure becomes trapped with no relief.

Pistons

One of the clever ways to avoid needing a separate inert pressurizing gas for **fuel**, is to apply pressure from the **oxidizer** tank using a propellant **piston**.



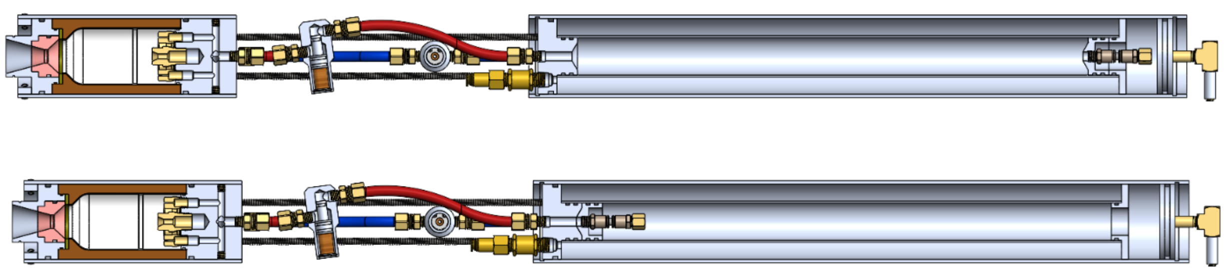
The piston is simply an unrestrained bulkhead which slides down the length of the fuel tank. It is important to include at least two O-Rings to minimize the chance that fuel and oxidizer mix in the tank, and we recommend having three for redundancy since it's a rather critical seal. Nonetheless, it may be inevitable that a tiny amount of fluid from one side may get into the other; if no ignition source is present, then there's no issue other than a slightly elevated chance of explosion while people are far away. Post-firing inspection of Half Cat indicated that there was almost certainly a small bit of fuel leaking into the nitrous oxide tank, although this was attributed to an NPT fitting which had been taken on and off so many times that it could no longer form a complete seal.

Intuition says that the friction of the O-Rings will greatly reduce the pressure on the fuel side, however this is not the case. Pressing down on Half Cat's bulkhead while the tank sat on a scale until the piston moved showed that it only took about 11 lbf, or around 15 psi. When pressurized the piston typically experiences anywhere from 600-1100 psi, which translates to hundreds or thousands of pounds of force.

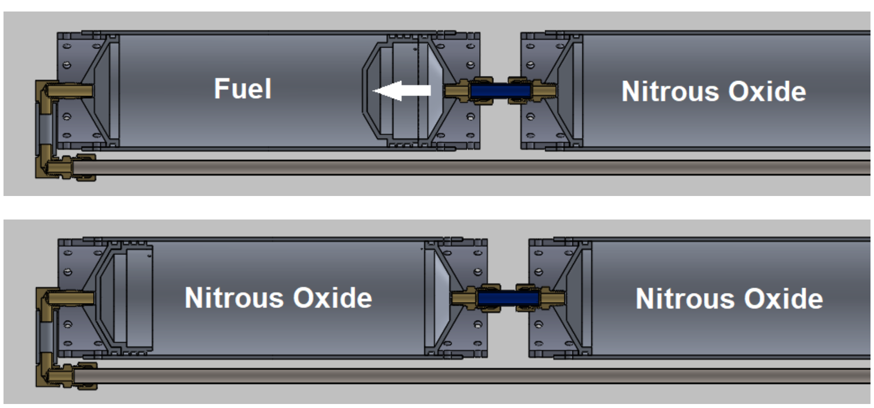
Fuel loading operations need to be considered in the design phase. You will need a way to pour in fuel, then seal off the tank – that could be accomplished by pouring fuel in, pushing the piston down, then capping off a fitting on the piston, as in Half

Cat (left), or by starting with the piston in position, loading fuel through some other fitting, then sealing it up, as in 2Cat/3 and 1Cat/3 (right).

Below is a cross-sectional view of Half Cat before and after operation. You can see how the piston moves down to expel fuel. You can also see the cap on the piston; it is removed in order to push the piston down into the fuel, then replaced to seal the tank.



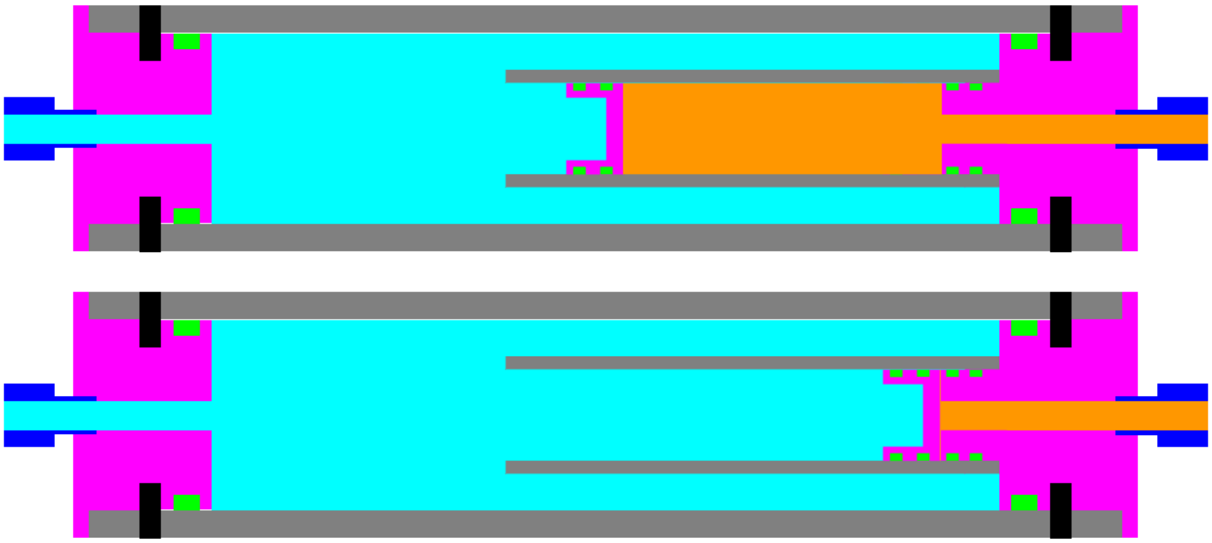
As an additional demonstration, below is a diagram of how the propellant piston works in 2Cat/3.



Notice that in all versions of the piston, it mates seamlessly with a bulkhead at the end of operation. Flat to flat, cone to cone, etc. This is to minimize unused fuel and prevent it from sustaining a fire that could lead to unintended overheating.

Concentric Tanks

Nesting a fuel tank inside of the oxidizer tank is a rather unique way to simplify design and reduce part count. Here, we see what that might look like in our system. Needless to say, this is not to scale and for visual aid purposes only.



The static vent, oxidizer inlet/outlet, piston plug, etc. are omitted here – for practice, picture what components you would have and where they would go (remember, this is a cross-section and some parts may be out of plane).

What this design achieves is a “dumb and simple” method of storing fuel with minimal machined parts and fittings. It is likely to come out lighter than separate tanks, but it does carry some minor downsides. The oxidizer outlet is required to be off-center, unless you get creative with drilling/plugging holes to route fluids inside the common bulkhead. The oxidizer tank also must be quite long to accommodate the entire propellant load, making for an unwieldy single casing if the burn time is long and diameter not very high. Additionally, there is a minimum required diameter of the inner tank so that its length will fit within the outer tank.

If the inner tank has a high fineness ratio (length / diameter), you may want to include a centering baffle, although it is not particularly necessary. Half Cat’s was obsolete after seeing how well it supported itself on its own.

A note on the diagrams: You will see that there are two O-Rings glands where the fuel tank mates to the common bulkhead. This has the same reasoning as for the piston – each O-Ring only seals against one fluid to minimize the risk of premature mixing.

The inner tank does not have to be retained on the common bulkhead. If you think about the force balance, it is always in equilibrium (because the fuel pressure is less than or equal to the oxidizer pressure) and cannot be forced off the common bulkhead. However, you may still choose to fasten the fuel tank on to ensure that it remains rigid at all times and prevent it coming off from handling.

Section 2: Mathematical Analysis

This section was originally written as a stand-alone guide predating Half Cat Rocketry and has been re-formatted to match our document style. It was written in the context of solid motors and hybrid combustion chambers, but the principles and equations can be generalized to all pressure vessels, closures, and bulkheads.

A calculator for many of the equations given here is included in the "Casing Design" tab of HalfCatSim, available for free on our website.

Regarding the term *Safety Factor*: Here it is defined as [Failure Strength] / [Stress]. This means that at FS = 1, the material will be at its failure point (this can be yield or ultimate, depending on which strength is used). Therefore, any value less than 1 is a failure. A safety factor of 2 means that the design can survive twice the expected pressure, at 3 it can survive three times, and so forth.

Stresses in Thin-Walled Pressure Vessels

The first step in designing a rocket motor casing is to ensure that it will withstand the pressure that it will experience. The hoop stress in a cylindrical thin-walled pressure vessel is

$$\sigma_h = \frac{P \times r}{t}$$

Where P is the pressure inside the casing, r is the mean radius of the casing, and t is the wall thickness. Note that this thin-walled approximation will be applicable to virtually any reasonable rocket motor casing design. The formulation for axial stress in the casing is

$$\sigma_a = \frac{P \times D}{4t}$$

Here D is the pressure vessel diameter; taking D to be the inside diameter of the casing yields a conservative approximation.

For general design purposes, stress and safety factor calculations are performed using the motor's *MEOP*, or Maximum Expected Operating Pressure. This is the highest pressure you expect the inside of the casing to reach under the normal operating conditions of the motor, without anything going wrong. For solids, this is the peak chamber pressure. For hybrids and liquids, it is typically the pressure of the nitrous oxide tank at the maximum temperature that it will reach when filled.

However, the MEOP is not necessarily the highest pressure the motor will be subjected to, even if nothing goes wrong. For large experimental rocket motor hardware, proof pressure testing may be required to verify the robustness and safety of the design. The proof pressure testing standard for Spaceport America Cup as of 2019 is 1.5 times the MEOP for no less than twice the maximum expected system working time. As the stresses in the casing should be below the yield threshold, the amount of time for which the casing is pressurized should not have any measurable effect, however it is important that it be able to maintain a reasonable safety factor when subjected to the proof pressure. This should be taken into account when determining the minimum safety factor in the design stage; for example, a safety factor of 3 at MEOP is a safety factor of 2 at proof pressure. Proof pressure safety factors below 1.5 should be avoided if possible.

Additionally, the material properties should ideally be evaluated at the motor hardware's maximum expected operating *temperature*. As most rocket motors become quite hot during operation, it may be necessary to take into account the reduction in the strength of aluminum that occurs as temperature increases. For 6061-T6, this can mean reduction in yield strength of as much as 20%. If the properties of the casing material at higher temperatures are not readily available, this can be accounted for with an increased safety factor. A minimum safety factor at MEOP of 3 is recommended if it can be achieved without making significant compromises, and a safety factor of 2.25 at MEOP is the minimum to maintain a safety factor of 1.5 at Spaceport America Cup proof pressure (1.5x).

Bolted Closure Design

SUMMARY:

There are four failure main modes that can occur in bolted rocket motor casings: bolt shear failure, bolt tear-out failure, tensile failure, and bearing stress failure. Formulae for calculating each type of stress and the corresponding safety factor is presented in this section.

KEY TERMINOLOGY:

- **Casing**: this is the tube that contains the pressure of the rocket motor. In a solid motor, it is the combustion chamber, but in a monotube hybrid motor the casing can include both the combustion chamber and oxidizer tank.
- **Closure**: a rocket motor has two closures, one forward (at the top) and one aft (at the nozzle end). The closures are what holds everything inside the casing. They typically include features such as sealing bulkheads (forward), nozzle retainers and tail-cones (aft), but here the term refers only to the portion that contains fastener holes which secure the component to the casing.
- **Bolts**: the threaded fasteners that hold the closures into the casing against the internal pressure. Technically they might be screws rather than bolts depending on the type of head, but functionally there is no meaningful distinction in this context. The term *fastener* is also used interchangeably.
- **MEOP**: Maximum Expected Operating Pressure. This is the highest pressure you expect the inside of the casing to reach under the normal operating conditions of the motor, without anything going wrong. For solids, this is the peak chamber pressure. For hybrids, it is typically the pressure of the nitrous oxide tank at the maximum temperature that it will reach when filled.
- **Edge distance**: denoted by E, this is the distance from the end of the casing to the *center* of the fastener hole.

Bolt Shear Stress

This is the failure mode that occurs when the bolts used to hold the closure into the casing break due to the force applied perpendicular to the axis of the fastener. It is most likely to occur in designs with few fasteners of relatively small diameter, and is the only failure mode not affected by casing wall thickness. The break occurs between the closure and the inside wall of the casing, where the bolt is under the most stress. In a rocket motor, bolts are typically in single shear (rather than double shear), as they usually only go through two layers: the wall of the casing, and the closure.

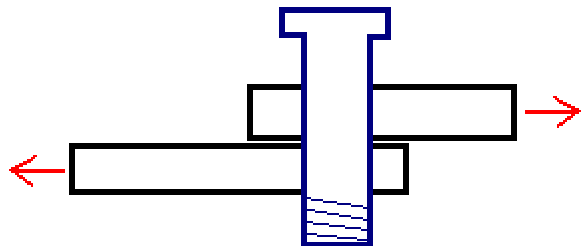


Figure 1: A basic diagram of a bolt in single shear

The shear stress on each bolt is simple to calculate. First, find the area of the inside diameter of the casing, and multiply it by the MEOP to find the force from pressure acting on the closure. Then, find the effective shear area of each bolt[1], and multiply it by the number of bolts in your design. Next, divide the force by the total area of the bolts to find the shear stress in each bolt. The equation below combines this into a single calculation.

$$\sigma_{bolt\ shear} = (\frac{\pi}{4} D_{i,casing}{}^2 \times MEOP)/(N \times \frac{\pi}{4} d_{bolt,minor}{}^2)$$

Where $D_{i,casing} \equiv casing\ inside\ diameter$

$d_{bolt,minor} \equiv minor\ diameter\ of\ the\ bolt$

$N \equiv number\ of\ bolts$

[1] Note that if the fasteners being used are fully threaded, you cannot use the outer diameter to find area, but must use the minor diameter. The minor diameter is the diameter of the smallest cross-section of the screw, while the major diameter is the outer diameter of the threads. For example, a standard ¼-20 bolt has a major diameter of 0.2500" but a minor diameter of only 0.1887". There is a nearly 50% decrease in area when calculating with the minor diameter.

Once a value for shear stress is obtained, it can be compared to the material properties of the fastener. For most steels, the shear strength can be estimated as $0.75 \times UTS$(Ultimate Tensile Strength). The safety factor is then determined by:

$$FS_{bolt\ shear} = \frac{0.75\ \times\ UTS}{\sigma_{bolt\ shear}}$$

Bolt Tear-Out Stress

Bolt tear-out is the failure in which the bolts tear through the end of the casing. It is most likely to occur in designs in which the fastener holes are very close to the edge of the casing, or the casing wall is relatively thin. This type of failure is the result of shear stresses in the aluminum casing wall. The area on which this shear stress acts can be found by multiplying the distance between the edge of the fastener hole and the edge of the casing by the thickness of the casing, then multiplying by two for each bolt.

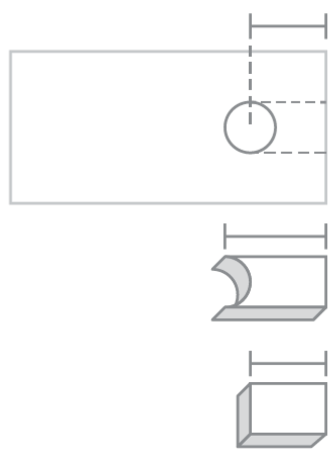


Figure 2: Bolt tear-out shear areas

The figure above shows the area of the casing that the bolt is pushing against, as well as two different sections with different shear areas. The lower section is the conservative shear area, which it is recommended to use here. The upper area is the conventional shear area, which represents a best-case scenario. For the purposes of simplifying this and other calculations, it is useful to determine the force acting on each bolt, which we will denote F_{bolt}.

$$F_{bolt} = \frac{\frac{\pi}{4} D_{i,casing}{}^2 \times MEOP}{N}$$

The equation for calculating the tear-out shear stress is then:

$$\sigma_{tear-out} = \frac{F_{bolt}}{E_{min} \times 2t}$$

Where $E_{min} \equiv E - \frac{d_{bolt,major}}{2}$, and t = casing wall thickness. Note that $E \geq 2d_{bolt,major}$ is highly recommended, however edge distances as small as 1.5 times the hole diameter can be used if great care is taken in the design, and a higher safety factor is allowed for bearing stress. The holes in Figure 3 below are at an edge distance of 1.75 times the hole diameter and did not experience any deformation.

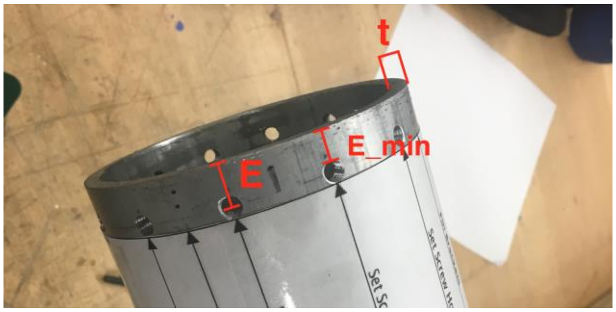


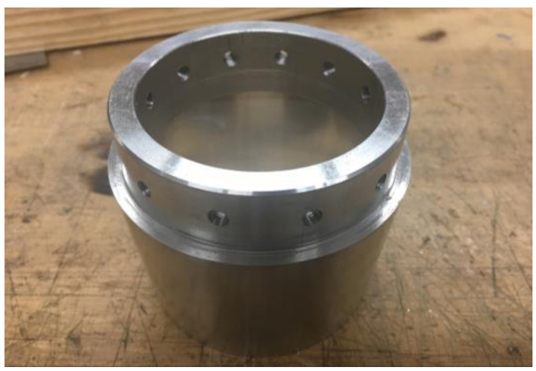
Figure 3: Edge distances and casing thickness on a real-world example

The safety factor is calculated by:

$$FS_{tear-out} = \frac{shear\ strength}{\sigma_{tear-out}}$$

The shear strength value for aluminum alloys is readily obtained from materials properties sheets and is equal to 30 ksi for 6061-T6.

It is important to note that the closure itself experiences the same forces as the casing and must also be evaluated to ensure bolt tear-out will not occur. In general, the closure is made thicker than the casing and so will not fail before the casing does, however the calculation above can be performed with the closure thickness substituted for the value of t to determine the closure safety factor.



A bolted aft closure displaying a thickness significantly greater than that of the casing shown in Figure 3. Note the integrated tail-cone.

Casing Tensile Stress

Casing tensile failure occurs when the portion of the aluminum casing between the fastener holes is stretched beyond its breaking point. It is most likely to occur in designs with a large number of relatively large bolts, or when the casing wall is relatively thin. To find the maximum tensile stress in the casing, you will need to divide the force from pressure acting on the closure by the minimum cross-sectional area of the casing, which occurs between the centers of the fastener holes.

$$\sigma_{tensile} = \frac{\frac{\pi}{4} {D_{i,casing}}^2 \times MEOP}{[(D_{o,casing} - t)\pi - N \times d_{bolt,major}] \times t}$$

Where $D_{o,casing} \equiv casing\ outer\ diameter$. Note that here $d_{bolt,major}$ is assumed to be equal to the major diameter of the female thread, which is not exact, but a sufficiently close approximation. If only the closure holes are threaded and through-holes are used in the casing, the clearance diameter may be substituted here. Similarly to the previous cases, safety factor is:

$$FS_{tensile\ yield} = \frac{YTS}{\sigma_{tensile}}$$

Where YTS is the Yield Tensile Strength of the casing material. This is the point at which the casing will plastically deform and be permanently damaged. To find the point at which the casing will fail completely, substitute ultimate tensile strength. For 6061-T6, YTS = 38 ksi and UTS = 42 ksi, both taken at 200˚F.

As with bolt tear-out, the closure is subjected to the same forces in tension as the casing, although if the closure is thicker than the casing it will not be the first point of failure. To perform the tensile stress calculations for the closure, replace D_o with the outer diameter of the closure (this can be taken to be equal to D_i assuming a very close fit tolerance), and replace t with the closure thickness.

Bearing Stress

Bearing failure occurs when the force of the bolts pushing against the edges of their holes causes the casing material to fail in compression. It is most likely to occur in designs with a small number of bolts of relatively small diameter, or when the casing wall is relatively thin. To find bearing stress, divide the force acting on each bolt by the product of the bolt diameter and casing thickness:

$$\sigma_{bearing} = \frac{F_{bolt}}{d_{bolt,major} \times\ t}$$

Once again, the safety factor is:

$$FS_{bearing} = \frac{BYS}{\sigma_{bearing}}$$

Where BYS is bearing yield strength, which is given as 56 ksi for 6061-T6. As mentioned previously, the bearing safety factor should be significantly increased if the design requires a bolt hole edge distance of less than twice the hole diameter. Additionally, the bearing stress calculations here assume complete contact between the fastener and the inside of the hole, as is effectively the case when the hole is tapped. If a through-hole is used, there may be some minor deformation caused by stress concentrations at the peaks of the fastener thread, or if there is excessive clearance between the fastener and the hole. Any through-holes in the casing should be a close locational fit.

See **Appendix A** for a worked example of the calculations in this section.

Fastener Selection for Bolted Closures

When designing a motor casing with bolted closures, it is important to select appropriate fasteners to hold the closures in place. The main characteristics to consider are length, threads-per-inch (TPI), and head-type. The diameter is driven by the bolt shear calculation previously discussed.

The length of the fastener should be sufficient to extend fully through both the casing and the closure (or to the specified depth if using blind holes in the closure), without interfering with any other components located inside the closure, such as the nozzle in certain designs.

The TPI is, as implied, the number of threads per inch of screw length. For example, a half-inch-long 1/4-28 screw has 14 threads. Any given diameter of screw usually has two available TPI values, one coarse and one fine. For rocket motor casings, fine-thread fasteners are preferred for multiple reasons. First, fine thread screws have a larger minor diameter than coarse ones, so the bolt shear safety factor will be higher for a given diameter. Second, fine-thread screws are less likely to back out under vibration. If backing out is a concern, removable thread-locking compound may be used, however this is usually unnecessary when using fine-threaded fasteners with heads. The downside to fine threads is that they cross-thread easier than coarse threads – this means that while installing, you may accidentally start threading them improperly, causing damage to the tapped hole. Oftentimes, fine threads are unnecessary for small fastener sizes. Coarse threads are acceptable at larger diameters as well, as long as you have considered the safety factors appropriately.

There are several types of screw heads that may be used with bolted closures, but the two most common are socket heads and rounded heads. The only significant difference between the two is that rounded heads will incur less drag when on the exterior of a sub-minimum diameter rocket, and should be used where high performance is desired. Other head types such as hex head may be used, but do not offer any tangible benefit. Regardless of head type, there is an advantage inherent to any screw with a head: when the screw is tightened against the casing, it creates preload in the joint. Bolt preload can significantly reduce the stress on the boltes themselves, and make the joint stronger by causing friction between the closure and casing, when tolerances are small enough. However, the effects of bolt preload are difficult to accurately calculate in the context of a rocket motor casing and are therefore neglected when performing design calculations. Because it can only increase safety factors beyond what is predicted, this is considered acceptable.

In some designs, set screws may be used to retain bolted closures. This can further reduce drag or allow the casing to be inserted into a motor tube. Unlike other screw types, set screws do not have a head, but rather a hex socket recessed into the shaft. This means that the cross-sectional area of the portion of the screw with the socket is much smaller than that of headed screws, making set screws much weaker in shear. This is especially true when the depth of the socket is greater than the thickness of the casing, which is very common. **If set screws are to be used in a bolted closure design, the shear stress calculation must account for the decrease in cross-sectional area due to the socket.** Most designs using set screws require them to be a much larger diameter than traditional screws, which increases the required length of the closure and decreases usable chamber volume. Set screws should therefore be used only when absolutely necessary.

Multiple Bolt Circles

The previous section assumes that only a single radial bolt pattern is necessary, however this may not always be the case. When designing hardware with a relatively thin casing wall and/or much larger diameter (>6 inches), multiple radial bolt patterns may be needed to reach the desired safety factors. It is assumed here that both bolt patterns will have the same number and size of fasteners.

Fortunately, the equations for a single bolt pattern are readily converted for use with a double bolt pattern, and all safety factors are calculated in exactly the same manner once the stress values are obtained. It is assumed here that more than two bolt patterns will not be used, but it is possible to extrapolate these equations for use with a larger number.

Bolt Shear

The bolt shear stress formula remains the same regardless of the spatial distribution of bolts. The equation is reproduced below for convenience. Note that here *N* is the *total* number of bolts, not the number per bolt pattern.

$$\sigma_{bolt\ shear} = (\frac{\pi}{4} D_i{}^2 \times MEOP)/(N \times \frac{\pi}{4} d_{bolt,minor}{}^2)$$

Bolt Tear-Out

For calculating bolt tear-out, there are two cases that must be considered. The first and most common case is that of non-overlapping bolt patterns; that is, where the edges of the projected areas of the inner ring of bolt holes do not intersect with the edges of the bolt holes in the outer ring. In this case, the tear-out force of the inner bolts will be much higher due to their greater edge distance. This can be averaged with the tear-out force of the outer bolt pattern, as the two bolt patterns are rigidly connected by the closure.

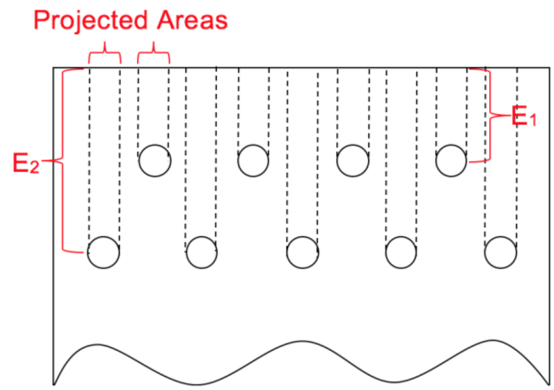


Figure 4: Non-overlapping bolt patterns with projected areas and edge distances indicated.

The tear-out stress for the non-overlapping case can then be calculated by:

$$\sigma_{tear-out} = \frac{F_{bolt}}{\frac{E_{min,1} + E_{min,2}}{2} \times 2t}$$

Where

$$E_{min,i} \equiv E_i - \frac{d_{bolt,major}}{2}$$

In the second case, the edges of the inner ring's projected areas and the outer bolt holes are overlapping, which greatly reduces the effective edge distance of the inner ring of bolts.

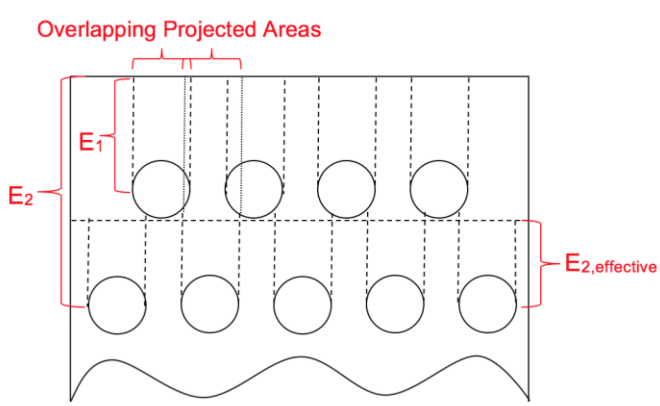


Figure 5: Overlapping bolt patterns with projected areas and edge distances indicated.

In the overlapping case, the tear-out force calculation will require the effective edge distance of the inner bolt pattern,

$$E_{2,effective} = E_2 - (E_1 + \frac{d_{bolt,major}}{2})$$

Note that in reality the edge distance of the inner bolts could be measured to the point at which the edges of the projected areas actually intersect the outer bolt holes, but this distance would be much more complex to calculate, so a more conservative and simplified approach is taken. In general, $E_{2,effective}$ should be similar to, and not less than, EE_1. In the case of $E_{2,effective} = E_1$, the formula for a single bolt pattern may be used. The tear-out stress for the overlapping case when $E_{2,effective} \neq E_1$ is therefore,

$$\sigma_{tear-out} = \frac{F_{bolt}}{\frac{E_{min,1} + (E_{2,effective} - \frac{d_{bolt,major}}{2})}{2} \times 2t}$$

Note that $E_{2,effective} - \frac{d_{bolt,major}}{2}$ corresponds to the $E_{min,2}$ term in the previous case, and is calculated using the effective edge distance. It was not substituted by an E_{min}term in this equation to avoid excessive subscripts.

Casing Tensile Stress

The casing tensile stress for multiple bolt circles is calculated nearly identically to the single bolt circle case. The equation

$$\sigma_{tensile} = \frac{\frac{\pi}{4} {D_{i,casing}}^2 \times MEOP}{[(D_{o,casing} - t)\pi - \frac{N}{2} \times d_{bolt,major}] \times t}$$

may be used where N is the total number of bolts in both circles combined, or it may be written as

$$\sigma_{tensile} = \frac{\frac{\pi}{4} D_{i,casing}{}^{2} \times MEOP}{[(D_{o,casing} - t)\pi - N_{circle} \times d_{bolt,major}] \times t}$$

where N_{circle} is the number of bolts per bolt circle, rather than the combined total.

Bearing Stress

The bearing stress calculation for multiple bolt patterns is identical to that for a single bolt pattern, provided that the edge distance guidelines previously set forth are followed. Thus the equation is once again

$$\sigma_{bearing} = \frac{F_{bolt}}{d_{bolt,major} \times t}$$

Nozzle Interface Design

When designing rocket motor hardware, it is essential to consider how the nozzle will be retained within the casing, and to ensure that the nozzle itself is able to withstand the forces acting on it without succumbing to material failure. There are several different types of external nozzle geometry that will be discussed here. For each type of interface, the bearing area and, where applicable, shear area are discussed. The stresses in these areas can then be evaluated against the nozzle's material properties and a safety factor calculated. O-Ring groove locations and sealing surfaces are considered as well. This section pertains only to the parts of the nozzle that contact the casing and aft closure, and does not deal with any calculations for the design of the converging cone, throat, or diverging cone.

Submerged Nozzle

A submerged nozzle is perhaps the simplest type of nozzle interface, in which the nozzle is completely contained, or "submerged," within the casing, and has a cylindrical outer shape. This type of interface is most commonly seen in small to medium sized motors that use snap rings as a means of retention rather than a bolted closure, but can also be used with a bolted "retaining collar" closure in some designs.

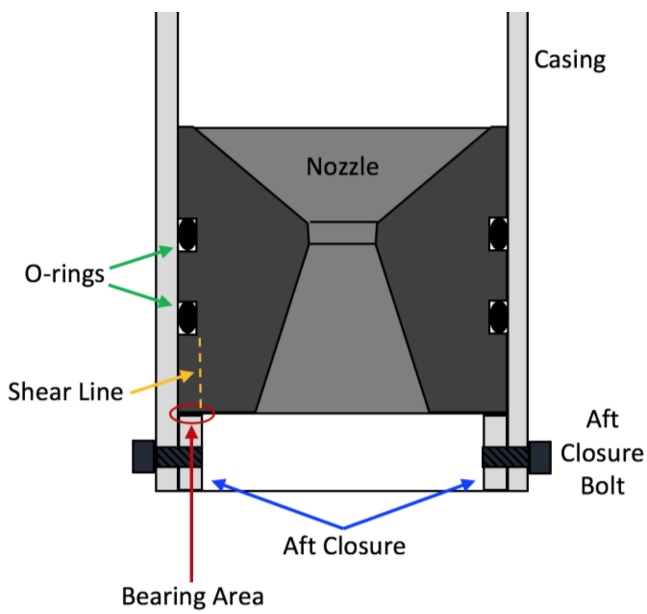


Figure 6: Cross-section of a submerged nozzle with a bolted aft closure, showing the bearing area and shear line.

In a submerged nozzle design, the dominant stress type is bearing stress, which occurs where the aft surface of the nozzle is pressed against the aft closure. The cross-section of the bearing area is indicated in the figure above, but the complete bearing area is a ring-shaped surface whose outer diameter and thickness are equal to that of the aft closure. This area can be calculated as

$$A_{bearing} = \pi(D_{o,closure} - t_{closure}) \times t_{closure}$$

Where $D_{o,closure}$ is the outer diameter of the closure and $t_{closure}$ is the thickness of the closure. The bearing stress on the nozzle is then

$$\sigma_{bearing} = \frac{\frac{\pi}{4} D_{i,casing}{}^{2} \times MEOP}{A_{bearing}}$$

And the safety factor

$$FS_{bearing} = \frac{nozzle\ material\ bearing\ strength}{\sigma_{bearing}}$$

Submerged nozzles are also subject to shear stress, which acts along the shear line shown in Figure 6 above. However, the shear line in submerged nozzles is usually quite long, resulting in a large shear area and a low shear stress. For most submerged nozzles, the shear line can be taken to extend from the inner diameter of the aft closure to the lower O-Ring groove, as the closure thickness and O-Ring groove are usually roughly equal to one another. The shear area can then be found by:

$$A_{shear} = \pi \times D_{i,closure} \times L$$

Where L is the length of the shear line described above. The shear stress is therefore

$$\sigma_{shear} = \frac{\frac{\pi}{4} D_{i,casing}{}^2 \times MEOP}{A_{shear}}$$

And the safety factor

$$FS_{shear} = \frac{nozzle\ material\ shear\ strength}{\sigma_{shear}}$$

The O-Ring grooves (also called glands) in a submerged nozzle design are typically located on the nozzle itself, as shown in the previous figure. This creates a radial seal between the nozzle and the inside of the casing wall, referred to in some O-Ring design manuals as an O.D. seal, where the casing is the bore. The length of a submerged nozzle typically allows for two O-Ring glands to provide redundancy.

There are two major disadvantages of a submerged nozzle in a bolted closure design like the one shown in Figure 6. The first is that the nozzle takes up a relatively large volume inside the casing, reducing the space available for propellant. The second is that great care must be taken to ensure that the rocket exhaust will not impinge on the interior surface of the closure. This can sometimes be achieved by decreasing the half-angle of the diverging cone and/or decreasing the diameter of the exit plane; however, this will cause the exhaust to expand more rapidly as it exits the nozzle, rather than continuing along the angle of the diverging cone. The most effective way to avoid impingement is to minimize the distance by which the nozzle exit plane is recessed inside the casing. In smaller motors, this is easily achieved by using snap-ring retention, but a bolted closure is necessarily much longer. For this reason, submerged nozzles are not generally recommended for bolted closure designs, except in cases where the target chamber pressure is relatively low. This allows smaller-diameter fasteners to be used, which reduces the length of the retaining collar and results in a lower exhaust expansion angle.

Protruding Nozzle

A protruding nozzle is one that extends beyond the aft end of the casing, and may also be called a “hanging nozzle,” as it “hangs” from the inside edge of the closure.

This is a common design for larger rocket motors, particularly those that use commercially-made molded phenolic nozzles.

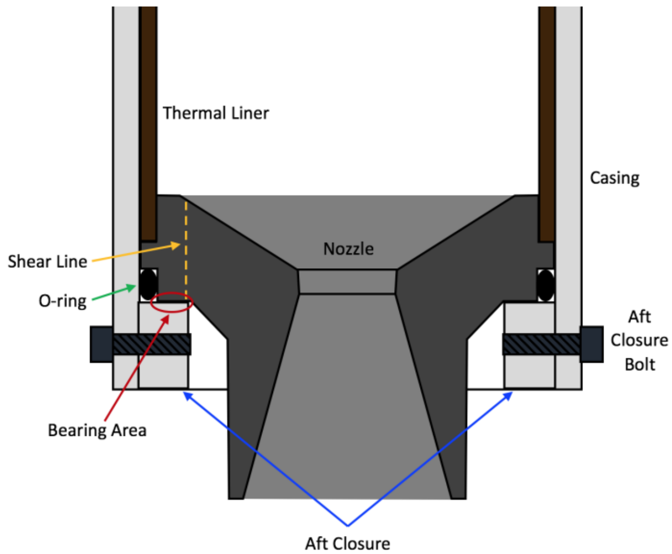


Figure 7: Cross-section of a protruding nozzle with a bolted aft closure, showing the bearing area and shear line.

With a protruding nozzle design, the O-Ring gland is comprised of the nozzle, the casing wall, and the aft closure. The O-Ring creates a radial seal between the nozzle and the inside wall of the casing. The location of the O-Ring gland immediately adjacent to the aft closure has the consequence of reducing the bearing area and increasing stress, making it better suited for stronger nozzle materials such as molded phenolic or machined steel. Because it is originally intended for screw-on closures, this type of O-Ring gland can make it difficult to properly seat the O-Ring in a bolted casing, as it will usually not stay put on just the nozzle. In this case a large amount of force must be applied from the top to squeeze the O-Ring into place during assembly. This issue could likely be mitigated by increasing the width of the portion of the gland on the nozzle beyond the standard gland dimensions. It is also possible to design a protruding nozzle with the O-Ring gland located entirely on the nozzle (similarly to the submerged nozzle), at the expense of slightly increasing the overall nozzle length. To keep the nozzle as short as possible, protruding nozzles typically only have a single O-Ring. Provided that the O-Ring is properly lubricated and inspected for damage prior to installation, there is little increased risk associated with the lack of a redundant O-Ring.

As can be seen in the figure, this type of nozzle design also interfaces with the thermal liner, which sits between the upper part of the nozzle and the casing wall. At the aft end of the motor, this interface between the nozzle and thermal liner does not need to be carefully sealed (and thus has no O-Ring) but is convenient for affixing the nozzle into the thermal liner with epoxy, making a propellant load easier to handle as a single piece.

Once again, bearing stress is dominant, as the bearing area is smaller than the shear area. As the bearing area calculation must take the O-Ring gland into account, it can be found by:

$$A_{bearing} = \frac{D_{i,closure} + D_{i,gland}}{2} \times \pi \times (D_{i,gland} - D_{i,closure})$$

As with the submerged nozzle, the bearing stress is then

$$\sigma_{bearing} = \frac{\frac{\pi}{4} D_{i,casing}{}^{2} \times MEOP}{A_{bearing}}$$

And the safety factor

$$FS_{bearing} = \frac{nozzle\ material\ bearing\ strength}{\sigma_{bearing}}$$

As before, the shear area is

$$A_{shear} = \pi \times D_{i,closure} \times L$$

With L being the length of the shear line, which extends parallel to the long axis of the motor from the inner edge of the closure to the point at which it intersects the outside surface of the nozzle. As shown in the figure, the shear line will likely intersect with the converging cone rather than the top surface of the nozzle, depending on the nozzle entry plane diameter. The shear line length L must be found from the specific nozzle geometry.

Again, shear stress and safety factor are

$$\sigma_{shear} = \frac{\frac{\pi}{4} {D_{i,casing}}^2 \times MEOP}{A_{shear}}$$

$$FS_{shear} = \frac{nozzle\ material\ shear\ strength}{\sigma_{shear}}$$

The protruding nozzle has two significant advantages in comparison to the submerged nozzle. First, it takes up much less space inside the casing, allowing for more propellant or lower casing mass. Second, because it extends beyond the aft end of the casing, there is no risk of exhaust impingement against the hardware. The disadvantage of a protruding nozzle is that it may be at greater risk of damage on landing, as it will likely be the rocket's first point of impact with the ground. For single-use nozzles this is not necessarily a problem, but may still be undesirable. The issue can be mitigated while also reducing aerodynamic drag by the addition of an integrated tail-cone to the aft closure, which extends from the aft end of the closure to (or slightly past) the end of the nozzle. Care should be taken not to extend the tail-cone too far past the nozzle exit plane, lest the risk of impingement be re-introduced. However, as the tail-cone is not a structural load-bearing component of the casing, impingement against its interior surface will not cause any real harm, though the resulting erosion can be unsightly.

Top: A protruding nozzle recessed inside an integrated tail-cone, immediately after a static test fire at 16' ASL. There is no erosion of the tail-cone, indicating little or no impingement.

Bottom: the same tail-cone after launching from 2700' ASL on an identical motor configuration, showing the effects of impingement from the increased exhaust expansion angle at lower ambient pressure

Fully-Supported Nozzle Insert

A fully-supported nozzle insert is similar in appearance to a protruding nozzle, with the main difference being that the nozzle itself is smaller in diameter, and the entire outer surface of the nozzle is in contact with the aft closure. This significantly reduces the bearing stress on the nozzle by distributing the force across multiple surfaces, and effectively eliminates shear stress as a concern. This design also has the advantage of fully protecting the nozzle, which is desirable for less durable materials such as graphite. Although it requires a more complex and substantial aft closure, it can allow the nozzle itself to be manufactured from smaller-diameter stock material, significantly reducing the cost. The nozzle insert can also be shortened when paired with an integrated nozzle extension (Shown in Figure 9). Because this design relies on the nozzle contacting the closure in multiple planes, it is more sensitive to manufacturing tolerances than other designs.

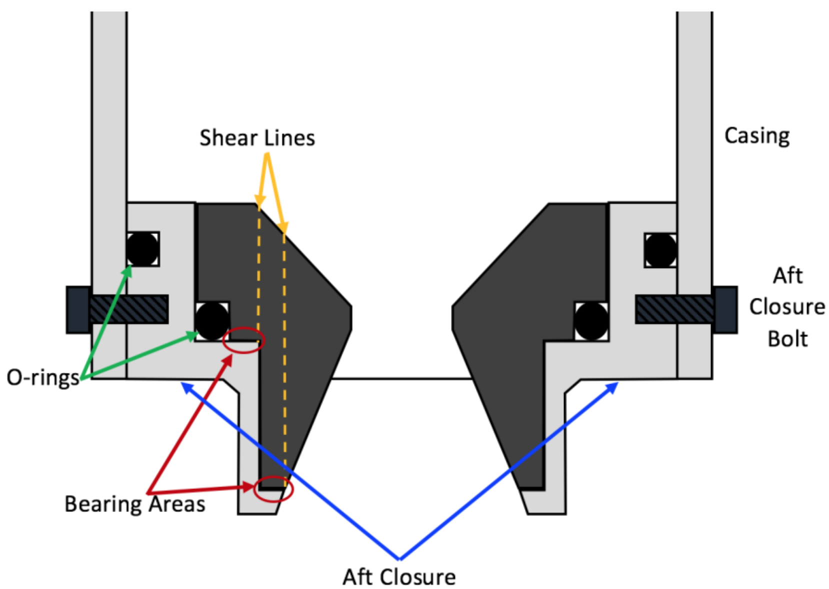


Figure 8: Cross-section of a typical fully-supported nozzle insert design using a bolted aft closure.

As with other nozzle interface types, the bearing stress and safety factor can be calculated by

$$\sigma_{bearing} = \frac{\frac{\pi}{4} D_{i,casing}{}^{2} \times MEOP}{A_{bearing}}$$

$$FS_{bearing} = \frac{nozzle\ material\ bearing\ strength}{\sigma_{bearing}}$$

A formula for $A_{bearing}$ is not given here because it is too specific to the design of the closure, but it is easily determined.

Unlike other nozzle interfaces, a fully-supported nozzle insert like the one shown in Figure 8 requires two O-Rings. One sits in a gland on the aft closure itself, creating a radial seal between the closure and casing above the bolt circle. The fastener holes must either be blind holes, or positioned in such a way that the inner O-Ring prevents the combustion gases from reaching them. The second O-Ring sits in a gland comprised of the nozzle insert and the aft closure, similarly to the protruding nozzle in Figure 7. It is also possible to design a fully supported nozzle insert with a single O-Ring seal, however it would require the nozzle insert to be made from larger-diameter stock, negating the main benefit of this design.

As can be seen in Figure 8, there will be exhaust impingement on part of the aft closure near the nozzle exit plane. As such, this design works best in motors with short burn times or relatively cool-burning propellants such as sugar/potassium nitrate mixtures. The closure can be designed to minimize or eliminate impingement if necessary, as the portion of the closure aft of the nozzle exit plane is very small. If used in a motor where exhaust impingement will not damage the aluminum (i.e., short-burning sugar motors), the closure can be designed with an integrated nozzle extension to improve the expansion ratio.

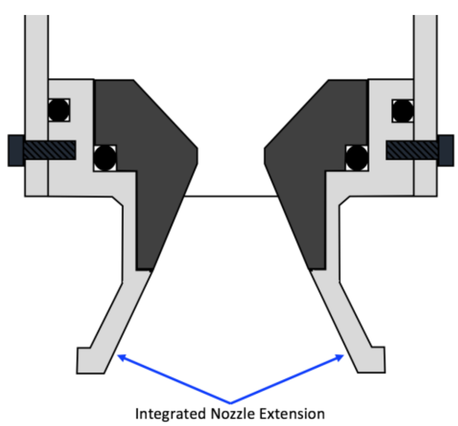


Figure 9: Cross-section view of a fully-supported nozzle insert with a bolted after closure and integrated nozzle extension.

This type of closure is also well-suited to use in snap ring casings, as long as the maximum outer diameter of the nozzle extension provides sufficient clearance for the snap ring and pliers.

The nozzle interface types presented above should be no means be taken as a comprehensive list. There are an infinite number of possible variations on – and combinations of – the designs presented here. The goal of this section is simply to introduce the principles of nozzle interface design while providing some reference points.

Forward Bulkhead Design

The forward bulkhead is the component that seals off the forward end of the motor casing. It is often also the forward closure, but not always; in some designs, the forward bulkhead is retained by a separate closure.

A forward bulkhead from a 98mm motor with double O-Ring grooves and threaded eye-bolt mounting post.

The most important feature of a forward bulkhead is the O-Ring groove(s). Forward bulkheads frequently include two O-Rings for redundancy, as shown in the image above, however a single O-Ring may be used oftentimes. Designing for a single O-Ring will make the bulkhead much easier to insert into the casing, as will selecting a smaller O-Ring cross-section, both at the cost of a slightly increased risk of failure.

The O-Ring(s) and groove dimensions should be designed according to [an O-Ring manufacturer's recommendations to create a static radial seal](), also called a piston-in-cylinder OD seal where the casing is the bore. If a single O-Ring is used, the groove should be designed to the standard dimensions (within the given tolerances); if two O-Rings are used, the groove(s) can be made several thousandths of an inch deeper and/or wider than the specified dimensions (beyond the given tolerances) without causing problems. In either case, the width of a gland is less critical than the depth – and even depth can be made deeper than specified, within reason. Additionally, the larger the cross-section of the O-Ring, the less critical the tolerances become. This is due to the short service life of O-Rings in rocket motors compared to normal industrial applications. O-Rings should be replaced after every firing, and ideally O-Rings used when firing the motor should not be inserted into the casing more than a few times, as they can be damaged over time by sliding past bolt holes or snap ring grooves.

Stresses in Forward Bulkheads

The force acting on a forward bulkhead is equal to the product of the pressure inside the casing and the inside diameter of the casing. This is the same

$$\frac{\pi}{4}{D_{i,casing}}^2 \times MEOP$$

term that appears in many of the bolted closure calculations in previous sections.

A forward bulkhead may be modeled as a uniformly loaded circular disk with clamped edges, with the maximum stress calculated by

$$\sigma_{bulkhead} = \frac{3\mathrm{P}{r_{rdisk}}^2}{4t^2}$$

Where P is the pressure inside the casing, r_{disk} is the radius of the bulkhead disk (see Figure 10; the inside radius of the casing may also be used for a more conservative approximation), and t is the thickness of the forward bulkhead. This equation is used to find the minimum bulkhead thickness. The bulkhead safety factor is

$$FS_{bulkhead} = \frac{YTS}{\sigma_{bulkhead}}$$

Where YTS is yield tensile strength of the bulkhead material.

In additional to the stress calculated above, there will also be shear stress in one or more parts of the forward bulkhead. At a minimum, shear stress will occur at the edges of the bulkhead disk. Other areas may also be in shear, particularly if the closure is a separate component like in Figure 10, rather than integrated into the bulkhead. The specific geometry of a bulkhead must be inspected to determine where shear occurs, as this will vary from one design to another.

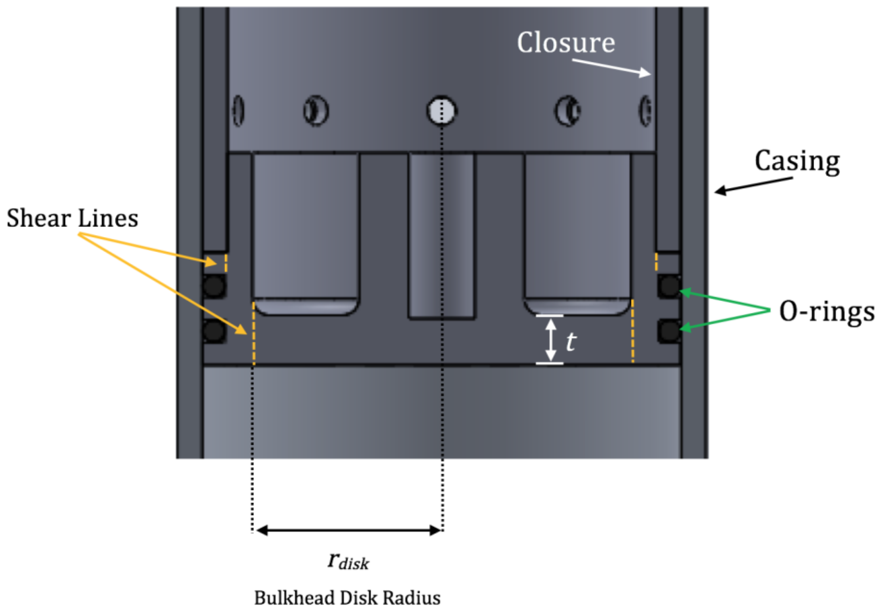


Figure 10: Cross-section view of a forward bulkhead with separate closure, showing disk radius and shear line locations.

In Figure 10 above, shear occurs in two places: at the edge of the bulkhead disk (as in all bulkhead designs), and above the upper O-Ring gland, where the bulkhead is pushed up against the forward closure. This second shear line is much shorter than the one in the bulkhead disk and will therefore experience a much higher shear stress. The shear stress at any point in any bulkhead design can be evaluated using the generalized equation

$$\sigma_{shear} = \frac{\frac{\pi}{4} D_{i,casing}{}^{2} \times MEOP}{2\,\pi\, r_{shear}\, L}$$

Where r_{shear} is the radial distance from the center of the bulkhead to the shear line, and L is the shear line length.

Pressure Transducer Taps

A commonly-desired feature of a forward bulkhead is the ability to connect a pressure transducer, to monitor the pressure of either the combustion chamber (in solids) or the oxidizer tank (in hybrids & liquids). This usually takes the form of a 1/8-27 NPT-tapped hole in the center of the bulkhead. A two- to four-inch long section of ⅛ inch pipe, called a manifold tube, is used to connect the pressure transducer to the forward bulkhead. For most transducers, a female-female NPT coupler will also be needed. The manifold tube is usually filled with grease to protect the transducer's delicate ceramic chip from hot or cold gases or corrosive chemical products of combustion. If the pressure transducer is not used in flight, the tap is sealed with a 1/8-27 NPT plug. Teflon tape should always be used with NPT threads to ensure a proper seal.

A 75mm single O-Ring bulkhead with manifold tube and pressure transducer installed.

Eye-Bolt Mounting Posts

The forward bulkhead can also be a convenient place to anchor the rocket's recovery system, especially in minimum-diameter and sub-minimum-diameter rockets. This is accomplished by integrating an internally threaded post into the bulkhead, into which a forged eye-bolt can be installed. The tensile stress in the post under recovery system loading (conservatively, a maximum of 50 times the rocket's weight at motor burnout) should be evaluated to ensure a sufficient safety factor. Any other stresses induced in the bulkhead by recovery loads will generally be much lower than the stresses caused by the internal pressure of the motor. If a large enough eye-bolt is used, a pressure transducer tap may be located at the bottom the eye-bolt mounting post, with the manifold tube extending out through the threaded hole for the eye-bolt. Note that this precludes the pressure transducer and recovery mounting point from being used simultaneously.

Tracking Smoke Wells

For some rockets, it is desirable to create a trail of smoke that persists after the motor's propellant has burned out, to aid in visual tracking of the rocket. This is achieved by designing the forward bulkhead of a solid motor with an integrated compartment for a slow-burning smoke grain, called a tracking smoke well or smoke charge well. The stresses in the wall of the smoke charge well must be evaluated similarly to those in the casing and forward bulkhead.

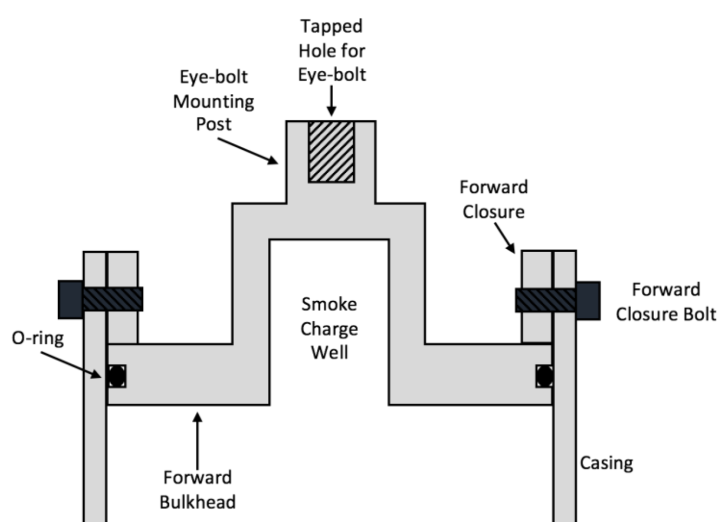


Figure 11: Cross-section view of a typical forward bulkhead with smoke charge well.

Snap Ring Groove Design

Snap rings offer a simpler alternative to bolted closures for retaining nozzles and bulkheads in a rocket motor casing, though they have their limitations. Cutting snap ring grooves requires turning the entire casing in a lathe, and they are therefore only suited to relatively short casings, or ones whose diameter is less than the lathe's spindle through-hole (typically 38mm casings and smaller). Longer casings may be placed in a lathe using a steady-rest, but the casing must still be sufficiently shorter than the bed of the lathe to allow positioning the tool holder past the end of the casing.

Designing a snap ring casing is relatively quite simple. The axial and hoop stresses must be evaluated in the same manner as for a bolted casing, and the distance from the end of the casing to the snap ring groove must be calculated by

$$E_{min} = \frac{P \times D_i}{\tau}$$

Where E_{min} is the distance from the end of the casing to the snap ring groove, P is the pressure in the casing, D_i is the inside diameter of the casing, and τ is the shear strength of the casing material. This equation is provided by Richard Nakka's Experimental Rocketry Website, and is stated to include a generous safety factor. However, you may also apply a safety factor to the pressure term if desired, for additional assurance. In small casings, this formula often returns a very small edge distance, so multiplying the pressure term by a factor of 2-3 is recommended in this case and has been empirically found to be quite reliable.

The tensile stress in the casing wall where the snap ring groove has been cut may be found by

$$\sigma_{tensile} = \frac{{D_{i,casing}}^2 \times MEOP}{({D_{o,casing}}^2 - {D_{groove}}^2)}$$

Where D_{groove} is the diameter of the groove. The snap ring groove depth should be no greater than half the wall thickness of the casing, so $D_{groove} \leq D_{o,casing} - t$

The snap ring size should be selected so that the manufacturer-recommended groove diameter is as close to this calculated D_{groove} as possible, although it is acceptable to deviate from this considerably. When D_{groove} does not exactly match a standard groove diameter, the snap ring with the closest specified groove diameter larger than D_{groove} should be selected.

Appendix A: Bolted Closure Worked Example

The calculations described above are performed here for a casing of the following specifications:

- D_o= 4 inches
- D_i= 3.624 inches
- t = 0.188 inches
- MEOP: 1400 psi
- Material: 6061-T6 aluminum

Bolts: single pattern, 12 x 1/4-28 screws

- $d_{bolt,major}$= 0.2500 inches
- $d_{bolt,minor}$= 0.2052 inches
- Material: Alloy steel, UTS = 120 ksi
- Smallest Edge Distance E = 0.4375 inches (1.75 times the hole diameter) at aft end

Bolt Shear

$$\sigma_{bolt\ shear} = (\frac{\pi}{4} \times 3.624^2 \times 1400)/(12 \times \frac{\pi}{4} \times 0.2052^2) = 36{,}388.86\ psi$$

$$FS_{bolt\ shear} = \frac{0.75\ \times\ 120000}{36388.86} = 2.47$$

Bolt Tear-out:

$$F_{bolt} = \frac{\frac{\pi}{4} \times 3.624^2\ \times\ 1400}{12} = 1{,}203.41\ lbf$$

$$\sigma_{tear-out} = \frac{1203.41}{0.3125 \times 2 \times 0.188} = 10{,}241.79\ psi$$

$$FS_{tear-out} = \frac{30000}{10{,}241.79} = 2.93$$

Tensile Stress:

$$\sigma_{tensile} = \frac{\frac{\pi}{4} \times 3.624^2 \times 1400}{[(4 - 0.188)\pi - 12 \times 0.25] \times 0.188} = 8{,}557.87\ psi$$

$$FS_{tensile\ yield} = \frac{38000}{8557.87} = 4.44$$

Bearing Stress:

$$\sigma_{bearing} = \frac{1203.41}{0.25 \times 0.188} = 25{,}604.47\ psi$$

$$FS_{bearing} = \frac{56000}{26504.47} = 2.19$$

As can be seen above, the casing presented in this analysis has a safety factor of greater than 2 in all failure modes, and the first failure mode would be bearing failure at the aft end of the casing.

In general, it is desirable for the first failure mode to result in the ejection of the aft closure, in order to prevent shrapnel/debris from projecting laterally away from the rocket. Additionally, if a failure of the aft end of the motor were to occur in flight at a sufficient altitude, the rocket may be able to deploy its recovery systems and return its payload to the ground undamaged.

A note on material properties at elevated temperatures: As rocket motors typically become quite hot during operation, it may be necessary to take into account the reduction in the strength of aluminum that occurs at elevated temperature. For 6061-T6, a typical reduction in yield strength is approximately 20%. This has been taken into account in the tensile stress calculations above.

Appendix B: Additional Photographs

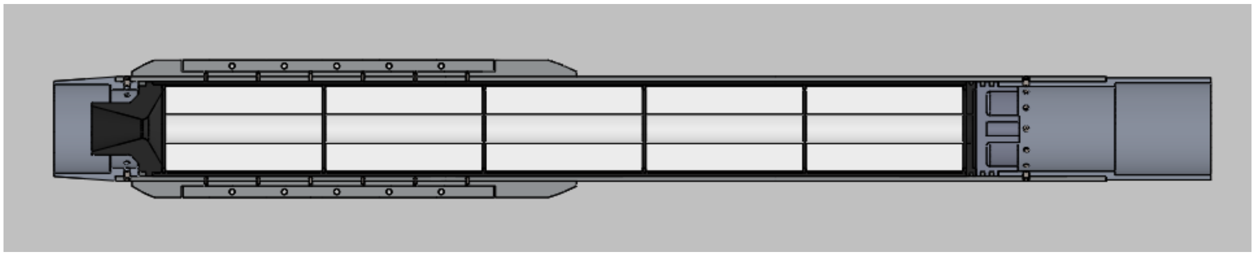
The aft end of the 4-inch bolted casing with screws installed.

The forward end of the casing; the extremely large edge distance is due to the structural airframe coupler extending 4 inches into the casing, and allows the tear-out calculations to be neglected for this set of fasteners

The holes were tapped through both the casing and the closure, however for fasteners with a larger diameter head, through-holes would be acceptable on the casing with sufficiently tight tolerances.

A view of the inside of the aft closure, without bolts installed. The aluminum plate inside is to seal the aft end of the casing for hydrostatic proof pressure testing, and takes the place against the aft closure that is occupied by the nozzle when the motor is assembled for firing.

The forward bulkhead of the motor, which seals the combustion chamber and provides an interface for a pressure transducer in ground testing and a recovery mounting eye bolt in flight. This component is technically excluded from this document's definition of a closure as it does not have any fastener holes. Instead, it is retained by the structural airframe coupler, which is bolted to the casing.



A section view of the fully loaded motor assembly in Solidworks, showing the configuration of the closures and bolt holes, as well as the fin-mounting brackets, forward seal disk, propellant grains, thermal liner, and nozzle. O-Rings are omitted in this image.