Steel Support for the Brass Cartridge Case

By James A. Boatright

Introduction

Different types of rifles firing different cartridges demand different levels of support from the steel parts surrounding the rifle's chambered brass cartridge case. In ***Stress in Target Rifles***, *Part II*, we calculated the elastic expansion of the chamber end of a typical stainless-steel target barrel and the strength of the bolt lugs in a blueprinted Remington 700 bolt action in containing a hydrostatic pressure equal to the peak chamber pressure of a typical 308 Winchester target load. Here, we will examine the mechanical effects of a typical pressure-versus-time curve (as illustrated graphically in Figure 1) occurring inside a brass cartridge case contained within the steel firing chamber. In designing a hunting rifle for mass-production, many other criteria are more important than how gently it treats its brass cartridge cases during the firing process.

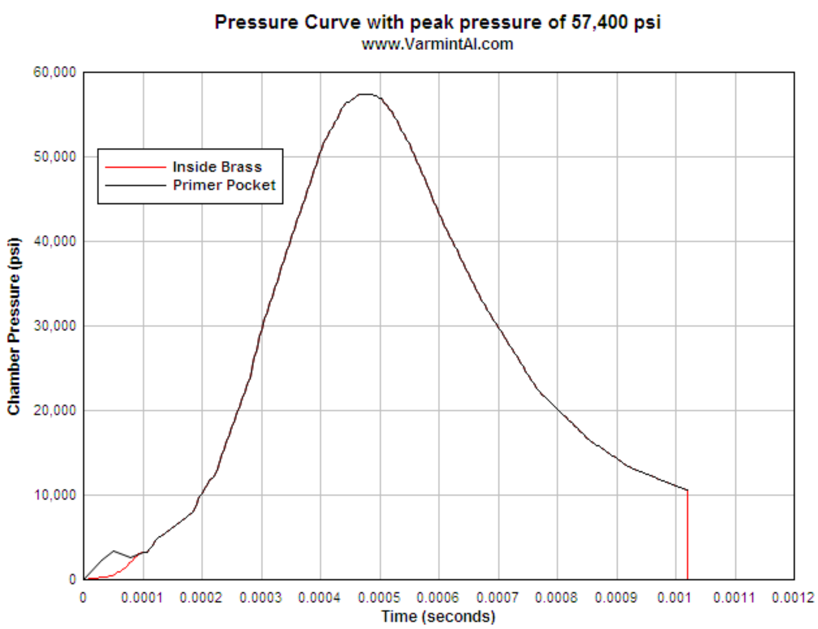

Figure 1. Example Chamber Pressure versus Time.

But a competition benchrest rifle needs only a handful of carefully selected, prepared and fire-formed cases containing precision loads that are matched to that rifle, its chamber and the shooting conditions. This set of cases must produce top accuracy for at least six to eight full-power firings, and we would prefer that they last a bit longer. Since demonstrating the highest possible level of shot-to-shot repeatability is the *raison d'être* for benchrest rifles, we sometimes feel a need to operate them at unusually high chamber pressures in that quest. These are just a couple of the reasons why a 6mm PPC benchrest rifle requires an action that is stronger in some ways than that of a 416 Rigby hunting rifle that will fire only factory-loaded cartridges, just one time each. We will look at the effects of variations in rifle action strength and the effects of various chamber design features on the brass material of a rimless, bottlenecked cartridge case used in our typical modern target rifle.

Bolt Face Setback and Chamber Stretch

Figure 2 shows the distribution of the *effective tri-axial (von Mises) tensile stress* at peak pressure in the chamber area of a rifle similar to our example target rifle. The effective tri-axial stress is a single, scalar value that we calculate for each small element of an object made of a ductile material that is being simultaneously stressed in all three dimensions. When the *local* effective tri-axial stress exceeds the ordinary single-axis tensile "yield strength" for the material, that portion of the material goes into "plastic flow," and suffers permanent distortion. In the von Mises theory of "effective stress," an *equivalent single-axis stress* is calculated that produces the *same distortion energy* in a material element as the three mutually perpendicular "*principal stresses*" acting together. In our cylindrically symmetric problem here, the three principal axes of stress are in the *radial*, *tangential*, and *axial* directions. The von Mises tensile stress then predicts "elastic failure" (that is, the beginning of plastic flow) of the tri-axially stressed material in just the same manner as if it were a uni-axial tensile stress.

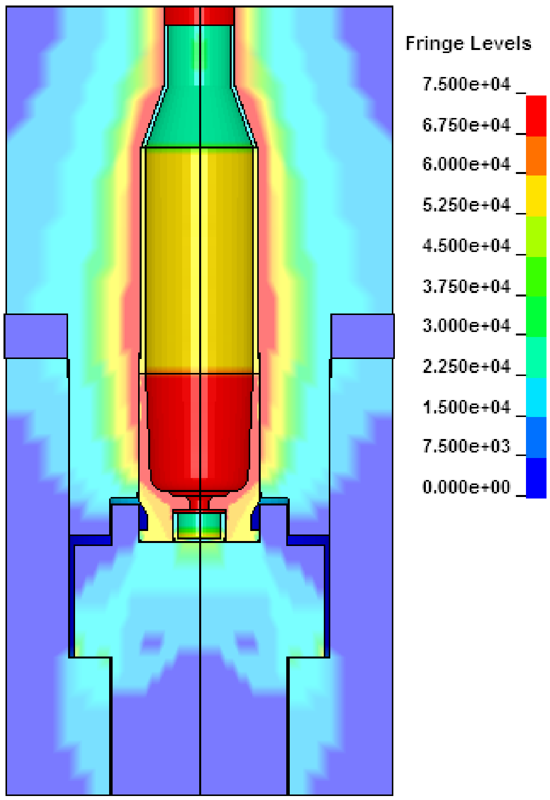

Figure 2. Effective Stress in PSI at Peak Pressure (Made by Al Harral).

As we would expect, no portion of any steel part and only certain portions of the brass cartridge case reach the "yield point" of elastic failure for its constituent material. Those localities where the material does *not* reach the yield point are only elastically strained and return exactly to their original shapes when the stress is removed. Figure 2 illustrates the kinds of powerful results Al Harral is able to produce with Finite Element Analysis (FEA) using the LS-DYNA program from Livermore Software Technology Corporation (www.LSTC.com). The figure shows the stress fringes for a blueprinted Remington Model 7 in 243 Winchester, but the materials data and pressure curve match our example. The entire results can be viewed in full color on this page of his web site, www.VarmintAl.com/amod7.htm . This image shows the peak stress distribution in the chamber area of a rifle far better than it has been previously portrayed.

Perhaps we could calculate the *peak elastic bolt-face setback* in our example Remington 700 action as it fires a *zero headspace* 308 Winchester cartridge at **57,400 psi** (transducer) peak chamber pressure. Using an effective internal piston diameter of **0.400 inches**, we had previously calculated the maximum possible bolt thrust to be **7200 pounds** at this peak internal pressure. We had also calculated the peak shear distortion angle (shear strain) to be **2.5 milliradians** where the shear stress is *concentrated* at the rear edges of each bolt lug as shown in Figure 2. And we can reasonably speculate that the bearing faces of the lug seats in the receiver will also take on a *matching* shear distortion angle at peak shear loading. We can directly estimate one component of the setback of the bolt-face, the setback *in shear* of the two *combined* pairs of **0.146-inch-** high bolt-lugs and lug-seats, **LSB**, to be:

LSB = (0.0025 radians) (0.146 inch)

= 0.00037 inch = 0.37 mils.

Per our discussion in *Part I*, ***Stress in Target Rifles***, we can calculate the *compression,* **LSC**, of these two pairs of lugs and lug seats, having a combined effective cross-sectional area **A** of **0.284 square inches** for both sets of lugs and lug seats working together and an effective working length of **0.880-inch**, to be:

LSC = (0.880 inch) (7200 pounds) / (A E)

= 0.74 mils.

where **E = 30,000,000 psi**, Young's modulus of elasticity for the steels of our bolt and receiver.

The lug seats in the receiver also *flex* out of the way by a small amount **LSF** in addition to *compressing* and distorting backward *in shear* at their bearing faces. Rather than attempting a calculation of this *ledge deflection* here, we will borrow the value of **0.64 mils** scaled from another of Al Harral's FEA studies.

As we computed for the lug seats, we find the compression of the bolt-head **BHC** (being **0.445 inches** deep and having **0.354 square inches** of cross-sectional area) to be **0.30 mils**. And, we can calculate the working length of the pressure-stretched portion of the front receiver ring to be **0.994 inch**, and its cross-sectional area to be **0.541 square inches**. So, we can similarly estimate the stretch of the front ring **FRS** to be **0.44 mils**.

So, our estimate of the peak elastic setback of the bolt-face when we fire a precision reloaded, *zero headspace* 308 cartridge would be the ***sum*** of these five separate "three-letter" elastic setbacks, or **2.49 mils**. This type of analysis usually tends to *over-estimate the summed aggregate* in so far as the pieces of the problem analyzed individually do not separate cleanly from each other, but *cross-couple*, and thus *duplicate* each other to some extent. On the other hand, each of these separate simplified calculations *under-estimates the desired peak effect* by calculating an *average effect* instead, and by not properly considering stress concentrators. Fortunately, our combined estimate of bolt-face setback agrees closely with the **2.5 mils** result of Al Harral's earlier FEA study of the (structurally similar) steel insert and bolt of a Stolle Panda action for **7200 pounds** of peak static bolt thrust. Our estimation errors must have offset each other almost perfectly here in this case. The beauty of knowing this value for our example blueprinted action is that, since each component of the bolt face setback is *elastic (and linear),* as soon as we know the amount of setback for *one* typical level of bolt thrust, we also know it for *all* reasonable amounts of bolt thrust. We need simply to *scale* the total setback *linearly* for the amount of bolt thrust produced by any size cartridge fitting with *zero headspace* into its chamber and operating at any reasonable pressure as compared to the **7200 pounds** of bolt thrust used in each of the studies mentioned above.

This same rate of bolt-face setback, or "*stiffness modulus*" of **2900 pounds per mil** (or **2.9 million pounds per inch** of setback), would have been calculated for this blueprinted Remington 700 action, regardless of what cartridge it had been chambered for and whether our example target rifle had used either the *long* or *short* version of the action. However, had our example action ***not*** been blueprinted, we would have found a smaller *initial* bolt stiffness value, say about **1450 pounds per mil**, applying for the **first mil**, **or two**, of bolt setback. Then, *with both lugs bearing*, we would have to "shift gears" to the larger stiffness modulus of **2900 pounds per mil** to find the setback caused by the remainder of the bolt thrust, in which case our example type of Remington 700 action might have produced as much as **3.0** to **3.5 mils** of total setback at **7200 pounds** of bolt thrust, for a stiffness modulus of only **2.1 mpi**.

Since we have previously calculated (in *Stress in Target Rifles, Part II*) that the inside walls of our example chamber elastically expand *radially* by a peak of **0.75 mils** at this same peak internal hydrostatic pressure, all we are lacking to complete the picture of *how the steel rifle parts support the brass cartridge cases in firing* is a calculation of the *peak forward stretch* of the steel parts surrounding our bottlenecked chamber. The length of the barrel tenon being stretched forward from the threaded receiver is **0.677-inch**, and the cross-sectional area of this **0.010-inch** oversize tenon is **0.726 square inches**. Its forward stretching force has to match the **7200-pound** peak bolt thrust, minus the **4300-pound** peak force accelerating the bullet (calculated in Part I of *Stress in Target Rifles*), plus **2150 pounds** of peak inertial force pulling the **5.0-pound** barrel rearward in a **10-pound** rifle at the peak of free recoil (per an equation given in *Part II*). In addition to this **5050-pound** force stretching the barrel tenon, the **2150-pound** inertial force is also forward-stretching the front ring of the receiver behind the **2-thread-depth** effective attachment point of the barrel tenon. [The **0.44 mils** of pressure-stretching in this portion of the receiver was included in the bolt-face setback calculations above.] The total amount of chamber-lengthening forward stretch **CLS** is found from:

CLS = (0.677 inch)(5050 pounds)/[(0.726 sq. in.)E]

+ (0.994 inch)(2150 pounds)/[(0.541 sq. in.)E]

= 0.157 mils + 0.132 mils

= 0.29 mils, which is quite small.

These "back of an envelope" manual calculations are given here for the benefit of any readers wishing to find similar values for their rifles, but not having access to FEA capabilities.

Chamber Expansions

Figure 3 shows a sketch of our example 308 Winchester chamber with all of our calculated peak chamber expansions in each direction given in mils (thousandths of an inch). Each of these chamber expansions is *elastic* and *scales directly* with hydrostatic chamber pressure. These modest dimensional increases should be easy on the brass cartridge cases and even remain within the purely elastic "spring-back" range for fire-formed, neck-turned brass cases fired in a tight-necked chamber with essentially *zero headspace*.

While the stainless barrel steel surrounding the chamber is the part of the rifle undergoing the *highest stress in firing* (as was shown in Figure2), the largest and most important to the brass cartridge case of these *chamber expansions* is the *elastic setback of the bolt face*. For example, if we were to fire a zero headspace *338 Lapua Magnum* round in a longer version of this same blueprinted action, the bolt-face could be subjected to about **13,000 pounds** of bolt thrust with each shot. The effective piston diameter inside this much larger case head is **0.504 inch**, and the peak chamber pressure could be as much as **65,000 psi**. After *scaling* our previous result for these two values, we find that the similarly blueprinted bolt of that longer hypothetical action would be set back by **4.5 mils** at its face in firing this much larger, more powerful cartridge.

Let us caution here that with clean, dry cartridges fired in a clean, dry chamber, the amount of bolt-thrust *scales linearly* with effective internal piston area and chamber pressure *only* so long as the cases fit the chamber with *zero headspace in a very strong action*. Otherwise, with *any non-zero* headspace, or even any *easy* bolt-face setback, the *stretching of the brass case* adhering to the chamber walls *absorbs* much of the potential peak bolt thrust. ***Whatever chamber pressure is needed to force the brass case head into first contact with the bolt face is usually just stored as elastic stress in the case walls and is never subsequently felt by the bolt-face even after the brass case walls might have yielded into plastic flow.*** Delaying, or time-spreading, the peak pressure load on the brass until after peak chamber pressure has occurred could be another way of reducing peak bolt thrust. With **2.1 mils** of initial headspace, our example case head ***does not contact*** the bolt face until the chamber pressure has risen to **25,000 psi** in a chamber polished with crocus cloth and having a *coefficient of sliding friction* of **0.27** with dry cartridge brass. And, our example case produces only **3950 pounds** of peak ***dynamic*** bolt thrust when fired with this much headspace in this chamber according to our fully dynamic FEA study. One other thing that these FEA studies have revealed is that we should be using **0.3733 inches** as the effective piston diameter inside the 308 case head instead of my too-conservative first estimate of **0.400 inches**. The peak static bolt thrust is then **6282 pounds** in our example.

Oiling a rimless cartridge case fitting our chamber with the usual **1 to 6 mils** of total headspace should produce about the same **6282 pounds** of peak bolt thrust as firing our example cartridge with *zero headspace* in a properly clean and dry chamber in our strong example rifle. Oiling the case or oiling the chamber does ***not*** increase the amount of potential bolt thrust *per se*—it simply interferes with the brass case absorbing much of it, instead of transferring all available thrust to the bolt face. ***It seems that we have come to regard as "normal" the situation where we are stretching our brass cartridge cases to control bolt thrust.***

As soon as the *peak* in chamber pressure has passed, each of these chamber expansions *reverses direction* and starts contracting toward its original chamber dimension. We can appreciate that if this same 308 barrel had been installed into a less strongly-designed or into a poorly-fitted bolt action, the bolt-face would be returning with a *much longer stroke*. In that case, the returning bolt might be able to *jam* the expanded rimless brass case very tightly ***both radially and axially*** into the tapered walls of our bottlenecked chamber. In jamming the case forward into the chamber, just as in full-length resizing, the *radial clearance* behind the case shoulder and the *axial clearance* ahead of the shoulder (as in headspace) are *essentially interchangeable*. In *both* instances, any real movement of the brass awaits the "zeroing-out" of *both* clearances. Just imagine "full-length resizing" our fired 308 case *with no lubrication* to get an idea of what could happen in the chamber of such a rifle with a long bolt-return stroke length. A "jammed case" of this sort can cause extremely "hard extraction."

What Happens to the Brass in Firing?

The following narrative describes a series of *typical events* (in approximate time-sequence) that will likely occur when a precision reloaded match round is fired in our example 308 target rifle. These events are described from the perspective of the brass material of the cartridge case and its interactions with the steel parts surrounding the rifle's chamber. Ideally, this cartridge brass is an alloy composed of **70-percent** copper (Cu) and **30-percent** zinc (Zn), varying in hardness all the way from the annealed neck and shoulder area of the case to the full-hard brass, or perhaps even extra-hard brass, of the case head and web. We are assuming a peak dynamic chamber pressure of **57,400 psi** in a cartridge case fitting our *polished chamber* with **2.1 mils** of headspace. The resulting **3950 pounds** of peak dynamic bolt thrust produces only **0.92 mils** of peak dynamic elastic setback of the bolt-face, for a *dynamic* stiffness modulus of **4300 pounds per mil**. Merely having a brass cartridge case present in the chamber reduces the *diametral expansion* of the steel chamber walls at this same peak chamber pressure from the previously computed **1.5 mils** (for hydrostatic pressure in a bare chamber) down to **1.1 mils** as shown in Figure 4. Otherwise, the chamber dimensions are accurately cut to 308 Winchester *SAAMI minimums* and the barrel is installed with a minimum **1.6300-inch** chamber headspace dimension. In this narrative, we are assuming that the chamber walls have been polished with crocus cloth (or "rouge cloth") in accordance with our recommended rifle-building practices (see below). The rear of the stainless steel chamber could provide *radial support* for the brass case walls, anywhere forward of a starting point located **0.156 inches** from the face of the locked bolt.

The following typical sequence of quasi-static events affect our example *precision reloaded*, *fire-formed* brass cartridge case during firing, as best they can be determined by observing and measuring the changes in fired cases, by analysis, and from the FEA studies done by Al Harral:

> The brass case is driven forward (by the **2.1 mils** headspace clearance in this example) and held in "shoulder contact" by four successive, relatively small forces, each more-or-less overlapping the next in time:
>
> a) First the *firing pin impact* against the anvil of the primer drives the entire cartridge forward, then
> b) The internal pressure in the primer cup due to *primer detonation* holds the case forward by pushing forward against the bottom of the primer pocket in the case head and by pushing the unseated primer cup back against the bolt face (and against the tip of the firing pin), while
> c) The primer cup is also forced back against the bolt face as a reaction to the force of projecting the *jet of hot primer gasses and particles* forward into the powder chamber through the flash hole, while the jet pushes forward on the inside of the case, until

d) Finally, the *backflow* of hot gasses from the powder chamber through the flash hole keeps the primer cup back in contact with the bolt face, with a force equal to the pressure inside the primer pocket multiplied by the area of the primer cup, and pushes forward on the cartridge case strictly as an "equal and opposite" *reaction force* (per Newton's Third Law of Motion).

2) As the chamber pressure rises to about **1,800 psi** (**1.8 ksi**), the case neck expands enough to release the bullet and contact the inside of the chamber neck, which also provides the initial obturation of the chamber. If not already in contact with the rifling in the throat of the barrel, the released bullet *moves forward,* in response to **130 pounds** of initial force on its base at this low pressure, by the necessary few mils and *stops* in contact with the origin of the rifling. The tail of the bullet still normally helps to seal the expanded neck of the cartridge case against gas leakage while the bullet rests in the throat of the rifling. [Later, when chamber pressure reaches about **10 ksi**, the main body of the bullet will start forward again, engraving the rifling into its own full-diameter obturating surface. This sudden, great acceleration of the bullet will always "slug up" the soft lead core and thin jacket of a match bullet so as to seal the grooves of the rifling very nicely.]

3) At chamber pressures starting as low as **2.8 ksi**, the quarter-hard, **0.015-inch** thick, brass of the case walls just behind its shoulders expands by **0.5 percent** into a *ring of contact* with the front end of the inside walls of the chamber. This **0.5-percent** tangential (and radial) expansion of this brass ring-element comprises about **0.33 percent** permanent *plastic strain* and about **0.17 percent** stored *elastic strain*. It is important to understand that with "strain hardening" materials like this, *increasing amounts of elastic strain are being stored in the material even while it is undergoing plastic deformation*. This *elasto-plastic* situation is illustrated in the sketch shown in Figure 5, for annealed cartridge brass. Subsequently higher internal pressures will greatly increase the *force of contact* between this ring of brass and the steel chamber walls. At any reasonable coefficient of friction between the brass case and the chamber walls, the rapidly increasing chamber pressure soon causes this front portion of the case walls to "lock onto" the chamber walls. The case shoulders remain in contact with the chamber shoulders until the base of the moving bullet clears the case mouth. The case neck and the front portion of its shoulders then *contract radially*, or "spring back," by the amount of elastic strain stored in them. The outside surfaces of the neck and shoulders are "soot stained" by smoke from the burning, compacted powder mass that is usually attached to the base of the moving bullet. The initial "contact ring" behind the shoulders has now become the gas seal for the rear of the chamber.

4) When the chamber pressure reaches **13 ksi**, the *last* ring-shaped element of the case walls has just yielded into contact with the inside walls of the chamber. The *extent of case wall contact* with the interior walls of the chamber has progressed rapidly rearward back to this "**0.280-inch** point" (a distance measured from the case head along the outside of the case walls) of our example 308 Winchester case. The brass case walls can be described as progressively "laying down" rearward into inside contact with the steel chamber walls, without slipping, like a tiny "track-laying vehicle." During this process, our example case head has moved back through **<u>1.5 mils</u>** of the original **2.1 mils** of headspace. Our example brass 308 Winchester case

prefers to stretch *elastically* by this amount of **1.5 mils** over the **1.274-inch-**long contact region of its case sidewalls during this "yielding into contact" process, providing that the case head does not first come into contact with the bolt face. ***The same chamber pressure increases that cause the ring-shaped brass elements to yield in succession also cause certain amounts of axial elongation to be "locked into" each ring of the case walls as elastic tensile strain***. The rearward progress of the cylindrical case walls yielding tangentially and transitioning into elasto-plastic flow is *reliably stopped* at this **13 ksi** pressure level, and at the "**0.280-inch** point" (plus or minus about **0.010-inch**), and is prevented from progressing farther back along the sidewalls of the case, by the "end effect" of *support* from the very strong case head and its internal web. Examination of many once-fired cases and many multiply-reloaded cases, all with "**308-size**" case heads, confirms that the thicker (more than **0.045 inches** thick), stronger (possibly extra-hard) brass case walls behind this **0.280-inch** point ***never*** expand into contact with the chamber walls, even after many normal firings.

5) Between the chamber pressures of **13 ksi** and **25 ksi** in this example, ***a complicated combination of elastic elongation and differential sliding along the chamber walls takes place in the brass case walls***. This is the interval during which the *chamber friction* characteristics are *most important*. In our example here, an additional **0.8 mils** of *elastic elongation* is stored into the brass case walls, including the last remaining **0.6 mils** of the original headspace and the first **0.2 mils** of bolt-face setback. *I cannot model how this occurs, so I do not know the maximum elastic and plastic distortions that could occur in this interval.* By the time chamber pressure has reached **25 ksi**, the case walls are *tightly* "locked onto" the chamber walls over the entire contact region, stopping the distributed sliding of the case walls. The ring-element at the **0.280-inch** point also starts to *yield plastically* in an axial direction as chamber pressure reaches **25 ksi**.
6) It just so happens in this example that, as chamber pressure reaches the mid-range value of **25 ksi**, not only does the brass in a narrow region around the "**0.280-inch** point" finally *yield to axial stress*, but also the brass case head itself *finally comes into contact with the bolt-face*. The protruding primer cup gets seated once again into its pocket, but this time only *flush* against the bolt face. At this **25 ksi** chamber pressure, the stainless steel chamber walls themselves have expanded elastically by half of their peak dynamic expansion amounts in all radial directions away from the axis of the chamber. For the remainder of the chamber pressure excursion above this **25 ksi** point, *any remaining headspace* (none in this example) and *all remaining bolt-face setback* (**0.7 mils** that is yet to occur in this example) *must cause some permanent plastic strain* within a region about **40** to **50 mils** in width, centered upon the "**0.280 inch**" point.
7) The case head and bolt face *move rearward together* during the remaining upper half of the chamber pressure excursion, maintaining contact with each other for the further **0.7 mils** of bolt-face setback that eventually will occur in this example at pressures above **25 ksi**. At the peak chamber pressure of **57.4 ksi**, the contacting walls of the brass case have lengthened by a total of **3.0 mils** (**2.3 mils** of well-distributed elastic stretching and **0.7 mils** of mostly plastic strain concentrated in the area of the **0.280-inch** point). Only some *minor yielding* should occur in the extra-hard brass of the

case head and web. On *first firing* at full pressure, the case head should "squish out" (just once) to a larger diameter by *no more than* **0.5 mils** (or a permanent expansion of **0.10 percent**, based on measuring the expansions of factory-loaded cases and similarly-performing handloads in virgin brass). Also, the primer pocket will likely become a fraction of a mil shallower *with each firing*. I measure case head expansion with a good Mitutoyo electronic micrometer just above the extraction groove of the case head, and I always clean out and "re-deepen" the primer pockets after each firing with the same solid carbide, end-cutting Whitetail Engineering primer-pocket uniforming tool that I used in initial case preparation. (Available from Russ Haydon, a ***PS*** advertiser.)

8) As the chamber pressure starts decreasing from its peak value, the bolt face and case head begin to return forward elastically together with a stroke length of **0.92 mils** toward its pre-firing neutral position at the rear of the chamber. The force available for re-compressing the brass case walls at any time during this pressure reduction phase comes from the *elastic stress* remaining stored in both the steel of the barreled action and in the brass of the case walls, minus the retarding force of the gas pressure remaining inside the case.
9) Finally, as chamber pressure falls quite low once again after the bullet has cleared the muzzle, the brass case walls *elastically retract* (or "spring back") from radial contact with the internal chamber walls starting at the rearmost contact (at the "**0.280-inch** point") and rapidly progressing forward to the case shoulders. If the brass just behind the case shoulders is *at least "quarter-hard,"* the case walls will retract by *more* than the steel chamber walls are contracting in this example, and thus provide some *radial clearance* allowing for easy extraction. The case also *elastically contracts in length* over this same **1.274-inch**-long *contact region* of the case walls by whatever amount of *elastic elongation* (**2.3 mils** in this example) had earlier been stored in these case walls.
10) The *post-firing headspace* of this example cartridge case should be the original **2.1 mils** of headspace ***minus*** the **0.7 mils** of *plastic stretching* that occurred at pressures above **25 ksi** in the region around the "**0.280-inch"** point, for a net of about **1.4 mils** of final headspace, but the details of the *contracting phase* of the firing cycle have not yet been studied.

We should point out that these "case events" *can vary tremendously* with different chamber, cartridge and firing conditions.

Ackley Improved Chambers

I believe that Parker O. Ackley was *correct* in claiming that a cartridge fired in an "Ackley Improved" (**AI**) chamber suffers *less case stretching* than would its parent cartridge fired similarly in its respective SAAMI-specified chamber. The difference in *case lengthening* can be attributed primarily to his use of *sharper* (**40-degree**) *shoulder angles* and, at least somewhat, to his use of a *sharper* (**0.060-inch**) *turning radius* at the junction of the case shoulders and neck walls. Mr. Ackley was convinced empirically that the "minimum wall taper" design feature of his **AI**-modified chamber also played an important role in preventing case stretching. Minimum taper cases "grip" the chamber walls better and absorb a maximum amount of potential bolt thrust, especially if the chamber walls are roughened to have higher friction with the brass. Ackley needed to *prevent as much bolt thrust as he could* because many of the actions with which he worked set back their bolt faces easily and excessively. No harm comes to the brass cartridge case as long as it is only being stressed elastically to minimize bolt thrust. Otherwise, in the absence of this control of bolt thrust, the long and powerful forward "return stroke" of the bolt face in such a *weaker action* would compress the brass case from its head forward to its shoulders *regardless of chamber wall taper*. Here, the term "weaker action" refers to one having smaller elastic "stiffness modulus" in its handling of bolt thrust. [The topic of friction between the chamber walls and the case walls is discussed below.]

In such a weaker action, the forward compression of the case walls by the returning bolt-face would *reliably stop* within the steeply angled **40-degree** "Ackley shoulders." *Lines of axial compression stress* can flow through the **17.3-degree** shoulders and onward into the neck of a standard **30-06 Government** case, for example, much more efficiently than they can make the **two 40-degree** turns in a fire-formed "**30-06 AI**" case. Shallower-angled shoulders would continue to conduct compressive wall stress on forward, right through the shoulders, which would lead to some of the *shoulder brass* being *extruded* into the case neck. This would *lengthen* and *thicken* the walls at the back of the neck before the chamber pressure had dropped enough to halt all brass flow.

The redoubtable Mr. Ackley was also *correct* in that his "minimum wall taper" chamber design solved the "hard extraction" problems due to *case jamming* that were being encountered by gunsmiths experimenting with higher chamber pressures at the time. The returning bolt first compresses the case shoulder hard against the front of the chamber, then converts additional axial compression into radial expansion of the case walls, and finally compresses the case walls elastically and plastically in the region of the **0.280-inch** point (a distance along the case walls from the head). This forward hammering of the case also eventually extrudes brass from the case shoulders into the case necks as discussed above.

I continue to be an admirer of Ackley's chamber design precepts, but perhaps, not for the same reasons that he advocated them. My only reservations about recommending certain **A**ckley **I**mproved (**AI)** chamberings are that (1) some of his **AI** *case capacities* are necessarily a bit "over bore," that (2) his beautifully simple case-forming procedure does nothing to correct the *critically short neck lengths* of most of the parent cartridges, that (3) minimum taper chambers are slightly more difficult to install using conventional chamber reamers, and that (4) they might not feed reliably through some machine guns. That being said, each of Mr. Ackley's chamber modifications is separately a good, sound chamber design feature for use in a precision bolt-action rifle, and they work particularly well together as a system:

1) Minimum wall taper (less than **0.0075 inches in diameter per running inch**, or **0.43 degrees**, included angle),
2) Sharper **40-degree** shoulder angle, and
3) Sharper **0.060-inch** turning radius at neck-to-shoulder junction.

Each of these **AI** design features usually requires *fire-forming* of the cartridge cases for effective implementation.

Purposely Roughing the Chamber Walls

The practice of *purposely roughing-up the steel walls* of the chamber installed in a custom-rebarreled hunting rifle was likely developed originally by gunsmiths working in Ackley's day as an *additional method* of reducing the "hard extraction" problem due to case jamming at higher operating pressures they wished to use. *Roughing-up the chamber walls can greatly reduce the maximum bolt-thrust handled by the action whenever the effective total headspace of the cartridge in the chamber exceeds about* ***one mil***. Generally, the greater the chamber friction and the total headspace, and the less tapered are the chamber walls, the greater will be the *reduction* in bolt thrust. These old-time gunsmiths found that they could preventatively control the *stroke length* of the returning bolt face in these ways so as not to *jam* the expanded case far enough forward to cause *case sticking* in the chamber.

Cases fired at *high chamber pressure* in a weaker or poorly fitted rifle action will usually extract easily enough from the roughened chamber, but they may be too badly *damaged* by *plastic stretching* and *re-compressing* to be safely reloaded. *Purposely roughing-up the chamber walls is probably still a reasonable practice when re-chambering a non-blueprinted hunting rifle that will mostly fire factory-loaded ammunition, but it has little application when building a precision target rifle.* This practice can only militate toward *stretching our brass cases*. Al Harral likens it to protecting the steel front bumper of your automobile by using its brass radiator core to soften impacts. In accordance with his FEA studies of plastic flow in fired cases, I join Mr. Harral in recommending that we *polish-out the newly installed chamber in a precision rifle barrel with crocus cloth, or better, instead of roughing-up its walls with fresh 220 to 400-grit emery paper*. We should be aiming to reduce the *coefficient of sliding friction* between our clean, dry brass cartridge cases and the inside chamber walls from the **0.50-to-0.60 range** of current practice to about the **0.25-to-0.30** range.

Conclusions

Changes in chambering practice can significantly reduce the plastic distortion that typically damages our brass cartridge cases with each high-pressure firing. I can only speculate that some of these improvements are due to subtle changes in the simultaneous case wall stretching and sliding processes occurring as the chamber pressure is rising from **13 ksi** to **25 ksi** in this example. Perhaps the chamber of the future IBS Sporter or Light Varmint Class benchrest rifle will have *highly polished chamber walls* having *minimum wall taper* and **40-degree** *"Ackley shoulders."* Even with polished, minimum-taper chambers, we must remain ever vigilant to guarantee that no more than **one mil** of total headspace sneaks into our precision-reloaded, fire-formed cartridges. Precision-made target rifle actions are strong enough to control bolt-face setback and withstand full bolt-thrust on each firing without help achieved at the cost of purposely stretching our brass cases. I have produced a detailed ***Technical Note*** entitled ***Yielding of Brass Case Walls in the Chamber*** that explains the basis for many of the claims made in this article.