Stress in Target Rifles

Part II

James A. Boatright

Introduction

As a bolt-action target rifle is fired, its metal parts must withstand several kinds of stress. In *Part I* of this article we presented a brief, elementary-level tutorial on the physics of assessing the strength of gun-making materials, especially steel alloys. With that background established, we can begin *Part II* by calculating the strength of our example target rifle in withstanding the ***primary stresses*** of firing modern high-pressure cartridges. We will show how to calculate recoil stresses and then present sample calculations of the strength and the elastic expansion of the chamber end of the barrel, followed by a calculation of the strength of the front locking lugs of the bolt body. While these example calculations serve to illustrate the basic concepts correctly, they are not intended to predict actual failure points. We will end with some pointed comments on residual stress in our target barrels.

Tension and Shear Stresses Resisting Recoil

Figure 4 shows a longitudinal sectional view of a scope-mounting base being held to the tubular front receiver ring of an example bolt-action target rifle with two **6-48** socket-head capscrews. As we installed these two high-quality mounting screws, we might have torqued them to as much as **25 inch-pounds**, so that they each clamp the scope base to the receiver with up to **900 pounds** of tensional *preload force*. Based on a cross-section area of **0.0109 square inches**, the "all-thread" shank of each screw would be under **83,000 psi** of initial ***tension stress***. But notice the *very short* "working length" of each of these screws—not even a length of one *shank diameter* of the **6-48** screw (nominally **0.1380 inch**) is under ***tension loading***. This common design flaw causes these assembled mounting screws to relax their preload force *quite readily* by the microscopic yielding of the receiver threads due to load cycling, to temperature cycling, or simply to mechanical vibration over time. Recall from *Part I* that we can calculate the ***strain ratio*** for this mounting screw (**0.28 percent)** by dividing its **83,000 psi** tension stress by Young's Modulus of Elasticity (**E = 30,000,000 psi** for steel). Then we can multiply this strain ratio by the working length (**0.100 inch**) of the screw to find its ***elongation*** of only **0.28 mils** (thousandths of an inch) for the stressed portion of the screw shank under this absolute maximum tightening torque. A working length of *at least* ***four*** *shank diameters* is generally recommended for threaded fasteners so that they can stretch enough with proper assembly torque to maintain most of their preload over

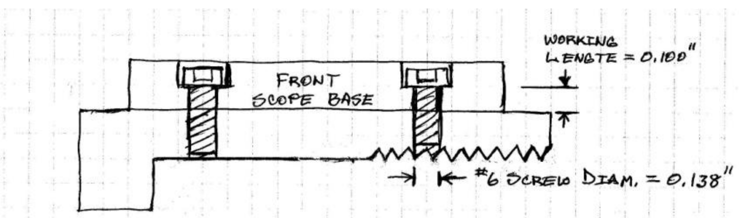

FIGURE 4. SECTION VIEW OF FRONT BASE WITH TWO 6-48 SOCKET HEAD CAPSCREWS.

the long term. As the clamping force of a scope-base mounting screw relaxes, the (proportional) *friction force* between the base and the receiver (that should be helping to handle the forces of recoil) also decreases. After our example rifle has been fired a few times, it is safe to assume that its scope base is handling virtually the entire force of recoil by loading the shanks of its mounting-screws *in shear*. The rapid loss of preload with these too-short screws explains why thread-locking compound is required for proper assembly of these screws.

When a rifle is allowed to recoil freely while it is being fired, the ***peak recoil force*** on its scope base (**Fs**, in pounds), necessary to accelerate the mass of the scope rearward along with the recoiling receiver, can be calculated from the following handy relationship based on Newton's Third Law of Motion ("equal and opposite forces"):

$$Fs = (Ws / Wr)\ Ab\ Pb$$

where

Ws = Weight in pounds of the scope and its rings and bases,

Wr = Weight in pounds of the whole rifle including its scope and mounts,

Ab = Cross-sectional area in square inches of the barrel's rifled bore,

Pb = Peak pressure in pounds per square inch (psi) on the base of the bullet, which is practically the same as the *peak* chamber pressure, and

Fs = Peak free-recoil force in pounds on the scope base attaching screws.

While it might seem counter-intuitive, the peak recoil force on the scope-mounting base of a free-recoiling rifle does ***not*** directly depend upon the weight of the bullet being fired! One could substitute for **Ws** the weight of a bipod, or some other discrete rifle component, and then **Fs** would be the peak recoil force on its attaching system.

If we are firing a **308 Winchester** target rifle, weighing **10.0 pounds** (including **1.5 pounds** for its scope, rings and bases), with a typical peak chamber pressure of **57,400 psi** (as measured in a pressure barrel using a fast-responding, conformal, piezo-electric pressure transducer), the peak recoil force **Fs** accelerating the scope rearward during firing calculates to be **642 pounds**. The peak chamber pressure occurs reasonably early in the firing process, shortly after the bullet has finished engraving the rifling and started moving down the barrel, and, if the rifle is being held normally, it will still be in essentially free-recoil when the peak recoil force occurs.

This tangentially-acting **642-pound** peak recoil force is stress-loading the scope-base mounting screws *in shear*. If for some reason only one of the **6-48** scope-base mounting screws (with a cross-sectional shear area of **0.0109 square inches**) had to bear this entire shear load, its ***peak shear stress*** would divide out to be **59,000 psi** (neglecting frictional forces from any remaining base-mounting preload). These mounting screws are subject to failure ***either in tension or in shear*** and the ***combined stress*** is more difficult for the mounting screws to withstand than the separate stress loads would be. Let us put these *stress levels* into perspective by noting that each of these high-grade **6-48 capscrews** probably has a ***tensile strength*** of at least **150,000 psi**, indicating a ***yield strength*** of about **120,000 psi *in tension,*** and a corresponding ***shear strength*** of about **112,500 psi**. We should also point out that the use of an efficient muzzle brake on the rifle would result in the scope mounts also undergoing a significant ***reverse*** (counter-recoil) ***inertial force*** at the end of each firing cycle. By using high-quality **8-40** base-mounting screws instead, with their **41 percent** larger cross-sectional shear and tension bearing areas, we could reduce the peak shear stresses proportionately while accommodating an increased initial tensile preload force. But, short of ***silver-soldering*** the steel scope base onto the receiver, we cannot solve the "too-short mounting-screw" design problem with this type of plain tubular receiver.

Chamber Radial Strength and Expansion

Now, we will examine the radial-direction strength and elastic expansion of the chamber end of the stainless steel barrel used in our example rifle. We can continue to use a custom-barreled, blueprinted target rifle chambered in 308 Winchester as our illustrative example. The barrel of our example rifle was made from AISI Type 416R stainless steel alloy, heat-treated throughout, and with a Rockwell C-scale surface hardness of **Rc28**. [The "R" in "416R" stands for "Re-melted," a manufacturing process used to improve the quality of the steel ingot.] This steel would have been tempered at about **1100 degrees Fahrenheit** after hardening and would retain an ultimate tensile strength (**STR**) of about **130,000 psi**.

We know how to calculate the pressure holding capability of an *idealized cylinder* made of this same stainless steel and having its inner diameter **d** and outer diameter **D** matching a portion of the chamber end of our example rifle barrel. This target barrel has a **1.250-inch** outside diameter ($\mathbf{D = 2\,r_o}$) all along its chamber swell. For the inside diameter ($\mathbf{d = 2\,r_i}$) of our cylinder, we can use the **0.455-inch** shoulder diameter inside the front part of the bottlenecked chamber to represent the high-pressure, chamber portion of the barrel out ahead of the encircling recoil lug and front ring of the receiver.

In Machinery's Handbook (and in many others) we find a simplified form of Lamé's [La-MAY's] equation which is often employed for calculating the material tensile strength required (**STR**) to hold a given internal hydrostatic pressure ($\mathbf{P_i}$) safely within a long, thick-walled cylinder made of a brittle material (think cast iron):

$$\mathbf{STR = P_i\,(1 + R)/(1 - R)}$$

$$= \mathbf{75{,}000\ psi}\ \text{(Tensile strength required)}$$

where

$\mathbf{P_i}$ = Peak hydrostatic internal pressure in psi

= **57,400 psi** for our example 308 ammunition

$\mathbf{r_i}$ = Inside radius of cylinder in inches = **d/2**

$\mathbf{r_o}$ = Outside radius of cylinder in inches = **D/2**

$\mathbf{R} = \mathbf{(d/D)^2 = (r_i/r_o)^2 = 0.1325}$ in this example (a dimensionless intermediate value found by squaring the ratio of the cylinder's inside to outside radii or diameters).

For this calculation, being a "thick-walled" cylinder requires that its wall thickness exceed **ten percent** of its inside diameter. By saying "a long cylinder," we mean that we are *not* calculating "end effects," nor are we allowing the cylinder to *stretch lengthwise* under pressurization.

The **130,000 psi** tensile strength rating of our example barrel steel provides a ***safety factor*** of **1.73** times the "required tensile strength" of **75,000 psi** found from this version of Lamé's equation. But the short time-duration of the dynamic chamber-pressure peak in the firing of a rifle makes this instantaneous peak pressure level ***far easier*** for the barrel to withstand than would have been the case with a steady pressure of that same amount, as Lamé's equation calculates. And the surrounding receiver ring and recoil lug strengthen the area just aft of the point being studied, while the rapidly tapering shoulders of the 308 Winchester chamber strengthen the cylinder just ahead. Moreover, as we will soon explain, *exceeding the elastic limit* for the ductile steel immediately surrounding the chamber (as calculated by Lamé's equation) is not really such a bad thing.

However, any plaintiff's attorney would probably seek a *minimum safety factor* of **2.0** from this traditional calculation using this quite conservative, "handbook" form of Lamé's equation. We would need our 416R barrel steel to test at **Rc32** (having been tempered at **1000 degrees Fahrenheit**) to meet a lawyer-proof tensile strength rating of **150,000 psi**.

But, because our barrel is made of steel that is quite ***ductile*** (non-brittle), we should examine other, more complete versions of Lamé's equation which will allow us to incorporate the value of Poisson's (Pwa-SOHN's) ratio **μ** into our barrel strength calculations. [**"μ"** is the lower case of the twelfth Greek letter "mu" from which we get our letter "m," and it is traditionally used to represent this ratio.] Poisson's ratio **μ** expresses just how much a test block of a given material will "shrink" laterally compared to its elongation under tension. [**μ = 0.30** for steel, as in our example here; **μ = 0.26** for cast iron; and **μ = 0.34** for aluminum and brass] From ***Advanced Strength of Materials*** by Professor J. P. den Hartog of MIT (McGraw-Hill 1952 and Dover 1987), we have a more detailed form of Lamé's equations for the two principal components of the stress **S** in our cylinder's walls, and we have the corresponding expression for the resulting small radial displacement **U** of any point at radius **r,** within the thick steel walls of our ideal pressurized cylinder:

$$S_r(r) = P_i*[r_i^2/(r_o^2 - r_i^2)]*[1 - r_o^2/r^2]$$

$$S_t(r) = P_i*[r_i^2/(r_o^2 - r_i^2)]*[1 + r_o^2/r^2]$$

$$U(r) = P_i*(r / E)*[r_i^2/(r_o^2 - r_i^2)]*[(1 - \mu) + (1 + \mu)*(r_o^2/r^2)]$$

Where

$S_r(r)$ = Radial component of stress (positive outward) in the cylinder walls at radius r from the center, with r ranging from r_i to r_o

$S_t(r)$ = Tangential Stress, or "Hoop" stress, in the cylinder at radius r

E = Young's Modulus of Elasticity for our example barrel steel, and

U(r) = Radial displacement in inches (positive outward) for an element of the steel at radius **r** within the cylinder, from Hooke's Law:

$$\mathbf{U(r) = (r/E)*[S_t(r) - \mu*S_r(r)].}$$

We should note here that both Hooke's Law and Lamé's equations (above) hold true only for the barrel steel that remains ***elastic*** (i.e., not stressed into the ***plastic strain*** region).

As we sweep the radius **r** from $\mathbf{r_i}$ out to $\mathbf{r_o}$, the ***radial stress*** component $\mathbf{S_r(r)}$ goes from:

$\mathbf{S_r(r_i) = -P_i = -57{,}400\ psi}$ (acting *inward* at $\mathbf{r = r_i}$, the inside surface)

up to:

$\mathbf{S_r(r_o) = 0.0\ psi}$ (at $\mathbf{r = r_o}$; i.e., at the outside surface of our cylinder).

The ***tangential (hoop) stress*** $\mathbf{S_t(r)}$ goes from the "required strength" expression of the "handbook" version of Lamé's equation at $\mathbf{r = r_i}$ (the inside wall of the cylinder):

$\mathbf{S_t(r_i) = P_i*(1 + R)/(1 - R)}$

$\mathbf{= 75{,}000\ psi}$ (tensile strength required, as before)

down to:

$\mathbf{S_t(r_o) = P_i*(2R)/(1 - R) = 17{,}500\ psi}$ (of tension stress at $\mathbf{r = r_o}$, in the outside walls)

where $\mathbf{R = (d/D)^2 = (r_i/r_o)^2 = 0.1325,}$ as before in this example.

And the ***total strain ratio*** $\mathbf{T_i}$ for the "inner-most fiber" (at $\mathbf{r = r_i}$) of the steel immediately surrounding the shoulder of our example chamber is:

$$\mathbf{T_i = U(r_i)/r_i = [S_t(r_i) - \mu*S_r(r_i)]/E = 0.32\ percent}.$$

Recalling from *Part I* that we had estimated the ***maximum elastic strain*** for this same type of barrel steel to be **0.35 percent**, we see that our *peak inside total strain* $\mathbf{T_i}$ is already **92 percent** of that *elastic strain limit*. Alternatively, we could say that a *peak chamber pressure* of **62,300 psi** would just reach the point of *elastic failure* for the inner-most fiber of steel surrounding the chamber shoulder in this example barrel.

Since the gas pressure on the outside of the chamber can be assumed to be essentially **zero**, the peak ***outside total strain*** $\mathbf{T_o}$ is given by the peak *outside hoop strain* and can be calculated from:

$$\mathbf{T_o = U(r_o)/r_o = S_t(r_o)/E = 0.046\ percent.}$$

If we multiply these two calculated *peak total strain ratios* ($\mathbf{T_i}$ and $\mathbf{T_o}$) by their respective *inner and outer chamber diameters* (**d** and **D** in this example), we can calculate that our inside chamber walls briefly expand by **0.0015-inch** in diameter at the shoulder of the chamber, while the outside walls of our example chamber increase by only **0.0006-inch** in diameter at this location over the shoulder of the chamber. Using suitable values, we could calculate in similar fashion the pressure expansion of the bore farther down the barrel. Just remember that in firing a high-pressure *bottlenecked* cartridge, the full *dynamic chamber pressure* does not make it far down the smaller diameter bore behind the bullet.

By examining these relationships for peak inner and outer total strains ($\mathbf{T_i}$ and $\mathbf{T_o}$), we can appreciate that when a thick-walled cylinder, such as our example barrel chamber, begins to fail from over-pressuring, the cylinder will fail *first* in its *inside wall* where the strain is greatest. But, the actual situation is neither so dire nor as simple as these expressions might seem to suggest because this "first failure" does *not* mean the rupture or bursting of the barrel. While the steel immediately surrounding the chamber of our newly installed barrel might "fail" on its *first firing*, our example barrel can contain over ***twice*** (actually **2.28 times,** per relationships developed in Professor den Hartog's textbook) the **62,300 psi** internal pressure corresponding to the initial ***yielding*** of the inner-most steel fiber surrounding the chamber. So, our example barrel should be able to contain a hydrostatic chamber pressure of **143,000 psi** before blowing up.

When the internal pressure first exceeds the "yield pressure" for the inside walls of this chamber, an ***inner cylindrical core*** of the barrel steel surrounding the high-pressure portion of the chamber goes into "***plastic deformation***," and is permanently expanded by a small amount, but also is safely contained by the outer portion of the barrel steel that has not yet reached its elastic limit. Note the small amount (one twentieth of one percent) of total strain $\mathbf{T_o}$ in the outer fiber of steel surrounding the chamber at our nominal peak chamber pressure of **57,400 psi**. Figure 5 shows a cross-sectional diagram of this situation as seen from the chamber end of the barrel.

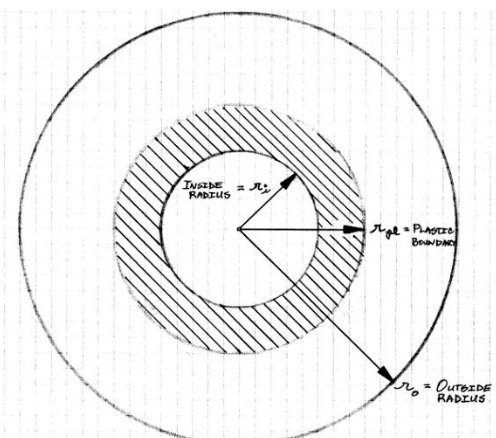

FIGURE 5. PLASTICIZED CORE PHENOMENON

As the internal pressure drops after the first "hot load" is fired, we notice that a couple of "good things" have happened in the steel surrounding the chamber. First, the plastically distorted, inner steel has been significantly strengthened by "***work hardening***," and, second, a significant amount of *compressive* tangential stress has been permanently implanted into this expanded portion of the barrel steel immediately surrounding the chamber. This "no-load" compression in the inner steel acts as a "pre-stress" load that will ***directly cancel*** several thousand psi of the peak chamber pressure during ***all subsequent shots***.

The diameter of the boundary between the "***plastically expanded core***" and the "***hoop-stressed***" outer containment portion of the cylinder *will increase each time an even greater internal pressure is applied*. At least up to a point, this "pre-stressing" is desirable and should be accomplished once and for all by firing a suitable high-pressure "proof load." If the quality of the barrel steel is uniformly good, actual barrel "blow-up" failure occurs only at such high pressures that the boundary of the plasticized region has grown at some point to breach the outer surface of the barrel profile. Given the wall thickness of our example chamber, the internal pressure $\mathbf{P_{po}}$ at which the outside stress reaches the elastic limit for our steel is relatable to the pressure $\mathbf{P_{pi}}$ at which the inside steel wall first went into plastic flow (**62,300 psi** in this example), and can be found from the expression:

$$\mathbf{P_{po} = P_{pi}*[(1 + R)/(r_i/r_o + R)]}$$

$$\mathbf{= 143{,}000\ psi} \quad \text{in this example.}$$

So, the true safety factor is actually 2.5 for this example barrel made of* 130,000 psi *steel. We are indebted to Professor den Hartog for his textbook discussion of the hydraulic upsetting of big gun barrels in which he covers the mathematics of this "plasticized core" phenomenon.

Bolt Action Strength

Before we can calculate the typical ***shear load*** on the locking lugs of our example bolt, we need to estimate the typical bolt-thrust generated in firing. And to do that, we first need to determine the *effective piston diameter* inside the rear portion of the 308 Winchester cartridge case being fired in our example target rifle. For normal firing, we should use the largest internal diameter inside the base of the expanded brass case just ahead of the case web, or about **0.385** to **0.400 inches**. By specifying that we are firing a cartridge having ***minimum headspace*** in our example chamber, we can ignore the *significant*, but *highly variable*, amount of "gripping force" developed due to friction between the inside walls of the chamber and the sidewalls of the pressure-expanded brass cartridge case. We will continue using a peak internal pressure of **57,400 psi** (transducer) to represent a typical peak chamber pressure.

Under these assumptions the typical peak bolt thrust **BT** (in pounds) to which our example 308-size bolt face might be subjected, would be given by:

$$\mathbf{BT = (57{,}400\ psi)*(Pi/4)*(0.400\ inch)^2}$$

$$\mathbf{= 7{,}200\ pounds}$$

This peak bolt thrust of **7200 pounds** is about typical in firing ***minimum headspace*** 308 Winchester cartridges in a minimum SAAMI-specification target rifle chamber. A clean, dry 308 Winchester cartridge, fired in the dry, purposely scratch-roughened chamber of our example bolt-action target rifle, ***could*** grip the chamber walls with enough force to absorb virtually the *entire* potential bolt thrust *if the cartridge had enough* ***excess headspace***. In that situation, only the unseated primer cup would actually bear on the bolt face (with between one and two thousand pounds of force). We have all probably seen a few "backed-out primers" in some of our extracted cases. This phenomenon clearly indicates that, with enough excess headspace, the case-head itself may ***never contact*** the bolt face in firing even full-pressure loads. But firing a cartridge in this manner would stretch its brass case so badly that it could not safely be re-loaded. Benchrest competitors use minimally re-sized, fire-formed cases having less than **0.001-inch** headspace in order to eliminate this otherwise significant variable in the firing process—easier on the brass, but the action has to withstand maximal bolt thrust with every shot.

While supporting the bolt head in resisting rearward bolt thrust, the bolt-locking lugs are loaded *in shear*; that is, the bolt thrust, together with its reaction forces, are trying to "shear off" the lugs from the body of the bolt. After our example bolt and receiver have been trued, each of the two locking lugs measures **0.4425 inch** wide by **0.4415 inch** in fore-and-aft length. The body of the bolt measures **0.695 inches** in diameter where the two lugs attach, and the outside diameter over the lugs is **0.987 inch**.

> ***Important note****: In truing the rear faces of these bolt-locking lugs, (1) the factory-made radiused fillet at the lug-to-bolt-body junction must not be disturbed, (2) a lathe tool bit with a large tip radius should be selected, and (3) the minimum facing cut that trues up both lugs should be faired into the factory radius. Carelessly facing all the way down to the junction with the bolt body using a sharp-pointed tool bit would build in a* ***severe local stress concentrator*** *in the resulting short-radius inside corner at the rear of each locking lug. And this very slight extra machining cut* ***would weaken the bolt head by about 25 percent****. The lesson here is to radius* ***all*** *inside corners to avoid creating stress concentrators.*

A little arithmetic shows that each lug attaches to the cylindrical bolt body over a curved *shear-loading area* of **0.2118 square inches**. Be careful not to confuse these shear-loaded, curved, lug-to-bolt-body "attachment areas" with the compression-loaded, rear-facing, trued "bearing faces" of the two bolt-locking lugs. Dividing one half of the typical bolt thrust *distributed* over each of the two bearing faces, or **3,600 pounds,** by the ***shear area*** of each load-bearing lug yields a typical ***shear stress*** loading of **17,000 psi** on each of the lug-to-bolt-body junction areas with both lugs working together. Actually, this value is the ***average*** shear stress over the attachment areas of the lugs. The unavoidably more concentrated peak shear stress near the rear edges of the two bolt-lugs rises locally to about **1.67 times** this average amount, or to about **28,300 psi**.

For illustrative purposes, we will conservatively estimate an ultimate tensile strength rating of **150,000 psi** for the heat-treated steel of our example bolt head (probably SAE 4140, 4340 or similar chrome-moly or chrome-vanadium steel). As another rule of thumb, the *ultimate shear strength* for this class of heat-treated steel is about **75 percent** of its ultimate tensile strength, or at least **112,500 psi** in this case. Thus, the estimated *shear strength* for each of the bolt lugs is about **4.0 times** the short-duration, maximum peak *shear stress* caused by the typical bolt thrust and concentrated at the rear edges of the bolt locking lugs. Without considering the effect of shear stress concentration within the lug attachment areas, this **safety factor** would have been cited as **6.6**, a value in the range of the safety factors more typically quoted for this stout two-lugged bolt head *when it is used with a 308 bolt face*. The safety factor will always be lower with larger diameter or higher pressure cartridges. At the normal peak bolt-thrust of **7200 pounds** for our example 308 rifle, this concentrated peak shear stress of about **28,300 psi** causes a peak shear distortion angle of only **2.5 milliradians** both in the steel bolt lugs and in the matching steel lug seats in the receiver at their contacting faces.

Residual Stresses in Barrels

Button-rifling is currently the most widely used method employed for rifling match-grade barrel blanks. As a side effect of this high-stress approach to installing the rifling, a full-length central "core" of the barrel steel surrounding the newly-rifled bore is ***permanently***

expanded so that the un-expanded outer layers of barrel steel retain quite a lot of *axially symmetric hoop stress* distributed along the entire length of the barrel blank. In fact, the same stress patterns previously shown in Figure 5 for an over-pressurized barrel chamber apply all up and down the newly button-rifled barrel blank, as well. If we were to perform any machining, or any other type of *stress relieving* operation, on the outer portions of the barrel blank containing this stress pattern, we would necessarily relax some of this hoop stress. The plastically expanded inner-barrel core, including the bore, would then expand *radially outward* as we relax the compression stress that had been containing it.

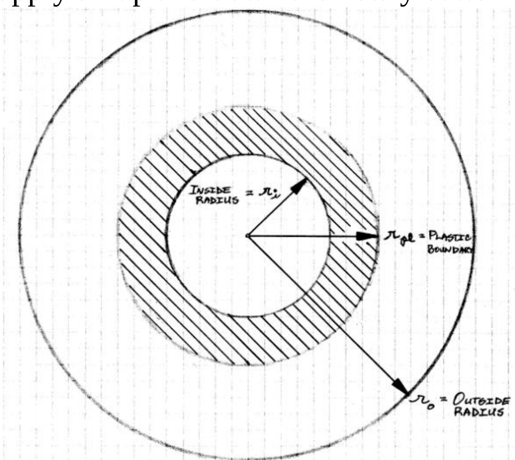

FIGURE 5. PLASTICIZED CORE PHENOMENON

Because of the prevalence of this stress condition and because of the accuracy loss that any bore expansion could cause, especially as it usually creates a "reverse taper" condition (an enlarging of the bore toward the muzzle anywhere along the barrel), we recommend that one ***never re-profile nor flute any match barrel blank***. The required barrel profile and any desired barrel fluting can be ordered from the custom barrel maker who can inspect for, and attempt to lap out, any consequent bore problems before shipment. When these rifle-building caveats are observed, button-rifled barrels are capable of producing the *finest possible rifle accuracy*—thereby demonstrating that axially symmetric hoop stress in the fitted barrel need ***not*** be detrimental to rifle accuracy.

One other type of residual stress has to be completely *unacceptable* in any match barrel blank—the internal stresses that *must remain* in any barrel that has been ***straightened by counter-bending***. While a noticeably bowed barrel blank is not acceptable for use in building a target rifle, a straightened barrel is actually much worse. The over-bending stresses involved in barrel straightening must *exceed the elastic limits* of the barrel steel in order permanently to deform it, even slightly. In barrel straightening, *residual stresses* are carefully *implanted* into the barrel steel by manually controlled deformations. These implanted residual stresses are *necessary* to hold the barrel straight after removal of the counter-bending stresses. Figure 6 shows diagrams of how barrel straightening is done and how the residual stresses are implanted into the straightened barrel. But these same residual stresses will cause the point of impact to "walk" away from the bull's-eye as the barrel warms during firing. The barrel seems to attempt *resuming its original curvature* (before straightening) as

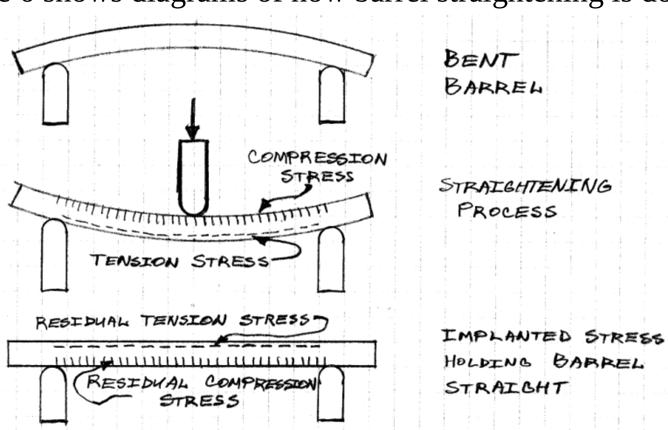

FIGURE 6. BARREL STRAIGHTENING

it warms up, even by just a few degrees, above the ambient temperature at which it was "straightened." Newly received barrel blanks could be inspected for straightness both at **room temperature** and at about **200 degrees Fahrenheit** to weed-out these bad apples.

Summary

In this wide-ranging two-part article, we have at least mentioned most of the more important types of stress in target rifles except for any "match jitters" induced by the operator. The concepts of mechanical force and of material stress, strain, elasticity, and strength were introduced and briefly explained in *Part I* of this article so that we would all be "speaking the same language." Then we dived right into calculating the strengths and elastic expansions of the metal parts of a bolt-action rifle in withstanding the *primary stresses* of firing. The problems that might arise from residual stress in match barrel blanks were discussed. Paragraphs containing discussions of the deleterious effects of stress due to ***differential thermal expansion*** in the scope-mounting systems used for typical "sniper-style, tactical rifles," and due to employing ***too much tightening torque during barrel installation***, have been pulled out and expanded into short, free-standing articles. We also plan a separate article continuing the investigation of elastic chamber expansion to include bolt-face setback in order to consider ***all aspects of the steel support for the brass cartridge case*** being fired in a bolt-action target rifle. Along the way, we endeavor to slay a few more myths about bolt-action rifles and to shed more light on the science behind the designing and building of accurate target rifles.