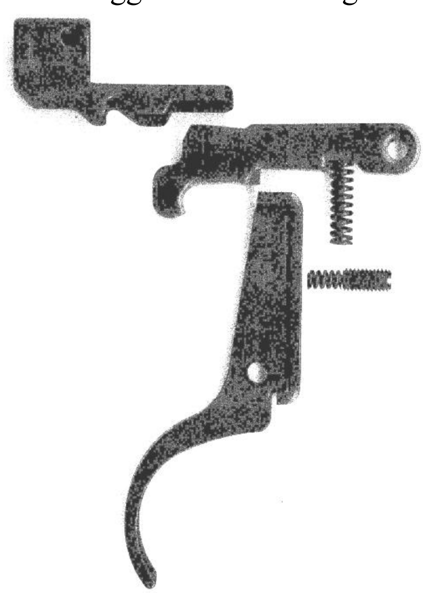
Trigger Force Calculations

By James A. Boatright

Introduction

Much has been made in print lately of possible rifle accuracy problems being traceable to the net upward force on the rear portion of the bolt in many bolt-action designs. This up-force is caused by the trigger mechanism pushing upward on the cocking piece holding back the compressed force of the firing pin spring by bearing against a forward-angled sear face in the trigger unit. The main reason for the popularity of bolt-action fire-control designs utilizing angled contact faces on the cocking-piece-to-sear interface is so that the rather large firing pin spring force of about 25 pounds can quickly and efficiently push the sear out of the way of the cocking piece after the trigger breaks and suddenly removes its support from beneath the sear. This design uses what is termed an "over-riding sear" in contrast, for example, to the "direct pull" vertical sear used in the design of the Model 1898 Mauser. As an engineer, my instinctive reaction to reading of this up-force causing a problem is to ask: "Just how much force are we talking about here?"

Figure 1 is an illustration of the cocking piece, sear and trigger geometry for a "two-lever" trigger of the Remington 700 pattern. In most of our benchrest competition rifles,

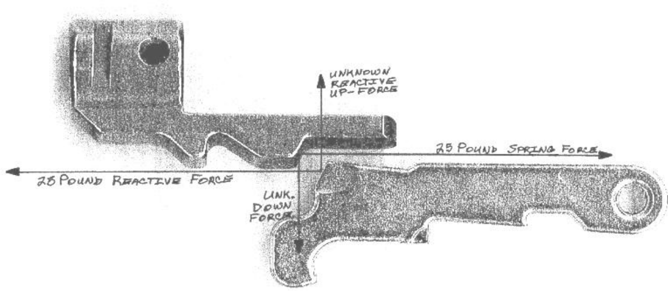


we use an aftermarket "three-lever" match-type trigger having the same Remington 700-type two-pin mounting system, the same dimensional specifications for the trigger unit interfaces with the action, and the same angled sear-to-cocking-piece engagement. For all of these triggers the sear face angle **A** seems to measure within about a half degree either way of **26.5-degrees** off the perpendicular to the firing pin axis. Any analysis of the mechanics of a trigger of any type should start with the force required to restrain the cocked firing pin spring (or the hammer spring in other designs) and then proceed in order through analysis of the sear lever (if present), analysis of any intermediate lever (if present) and, finally, a calculation of the amount of trigger pull required to fire the weapon.

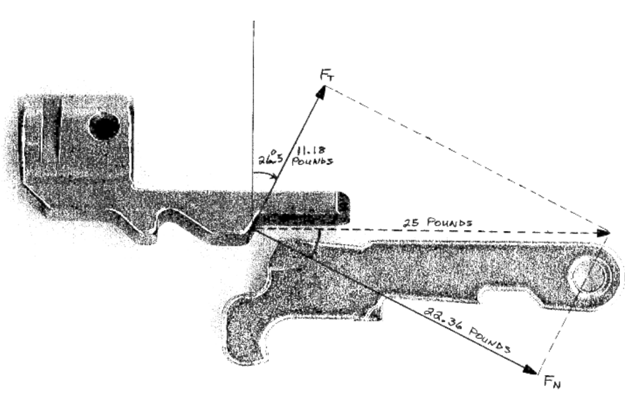
In the following article we set out to find the amount of up-force on the cocking piece, and end by presenting a detailed, concise and accurate "first level" analysis of the static forces acting on all parts of a Remington-pattern two-lever trigger mechanism *as it sits cocked and ready to fire*. I have not seen this type of analysis published lately in the popular literature, and the best available reference contains a slight error. As interesting as this basic level study may be, it does *not* always represent the actual situation at the moment of firing the rifle. This simple, first level analysis is all we need to do if we are working with a target rifle having a 2-ounce, match-type trigger that lacks a designed-in capability to reset itself if the trigger could somehow be only partially pulled without firing. For liability reasons, a small positive release angle on the trigger or sear is a key design feature of all commercially made hunting rifle trigger designs. This trigger release angle (of about 1.5 degrees in the Remington trigger) allows a portion of the load from the cocked firing pin spring to assist in overcoming friction and to help in resetting the partially pulled trigger. The (adjustable) trigger-pull spring force also assists directly in this trigger resetting operation. Unfortunately, both for the analytical complexity involved, and for the hunter wishing for a more sensitive trigger action, this positive trigger release angle also means that the pulling of the trigger lever works backwards through all of the fire-control mechanism to retract the firing pin slightly just before firing. Hence, the heavy 5-to-7-pound trigger pulls on most factory bolt-action hunting rifles. The sear-lifting safety incorporated into the Remington unit retracts the firing pin in a similar manner as it is being engaged. A much more complex "second level" force analysis is required to understand fully the situation just at the moment of releasing the firing pin with the factory unit. I will only outline what would be involved in that type of analysis at the end of this article.

Analyzing the Forces on the Cocking Piece

Let us start by stipulating that, at the bearing face of the cocking piece, the firing pin spring produces a nominal value of **25 pounds** of horizontal compression force **F** while being held at full cock. [The results of this analysis are readily scalable for any other particular value of the spring force **F**.] Figure 2 shows a diagram of the cocking piece bearing upon the sear face with this active horizontal 25-pound force and the supported sear pushing back against the cocking piece with an *exactly matching*, passively generated, 25-pound horizontal "reaction force." The diagram also shows the unknown size "active" down-force on the sear (to be calculated) being exactly matched by a passive "reactive" force **Fv** of this same unknown size, but pushing upward on the cocking piece. These exactly matching reaction forces are mechanically generated by the action parts so as to keep everything *stationary* within the action while we are preparing to fire.



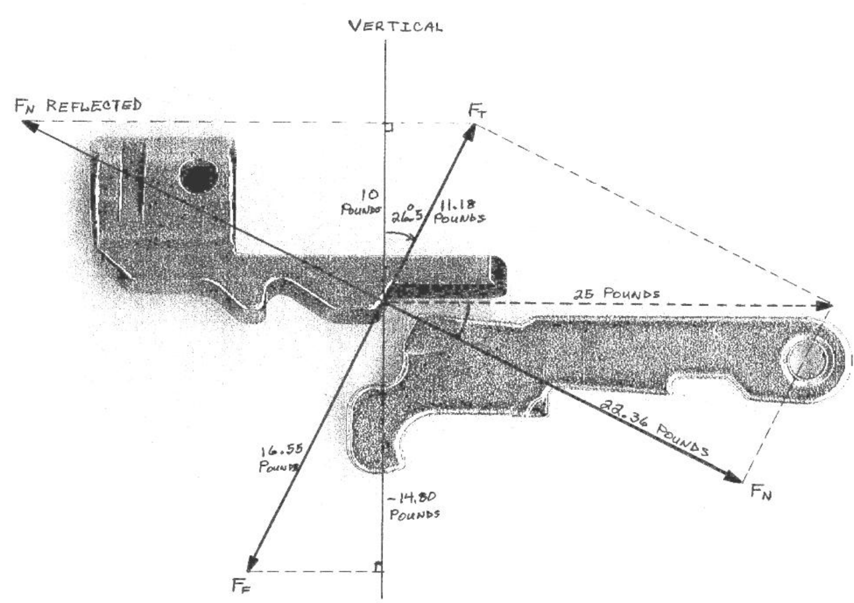
The compressive firing-pin spring force **F** bearing on the angled sear face can be *resolved* into rectangular components: **Fn** normal to the sear contact surface at its angle **A = 26.5 degrees**, and **Ft** tangent to the sear contact surface, as diagrammed in Figure 3. As the diagram shows, these two rectangular force components (**Fn** and **Ft**) would *vector sum* to equal exactly the original imposed **25-pound** spring force **F**. Just as the spring force **F** can be *replaced* by its component forces **Fn** and **Ft**, we can also replace the reaction force (**minus F**) of the



sear upon the cocking piece with its (equivalent, but opposite) normal and tangential component reactive forces. The magnitude of each of the two normal components **Fn** (of the spring force and of the reaction force) is given by **F*Cos(A)**, or **22.36 pounds** in this example. And the two tangential forces **Ft** are each equal to **F*Sin(A)**, or **11.18 pounds**. The asterisks (*****) just mean "multiplied by." [Notice that the normal forces seem to be exactly twice the size of the tangential forces. More on this later.]

One of the reasons why we replaced the incident force vector **F** with its two rectangular components (**Fn** and **Ft**) was so that we could find the size of the normal force **Fn** that is needed to calculate yet another (third) force—the force of friction **Ff** that each of our two steel parts is exerting upon the other. The friction force **Ff** lies in the plane of the sear-to-cocking-piece interface (as does the tangential force component **Ft**), and its size is calculated as the product of the normal force **Fn** multiplied by a suitable coefficient of friction. The coefficient of friction **Cf** is a dimensionless numerical value between a minimum of **0.00** and a (usual) maximum of **1.00**. The particular value to use in any given situation depends completely upon the circumstances involved. We must know the two types of materials rubbing upon each other, whether or not any lubrication or friction modifying treatment is involved, and whether the parts are stationary with respect to each other or already sliding against each other. Here we will use the coefficient of *static friction* for dry (non-lubricated) steel on steel, or **Cf = 0.74** from the handbooks. [The coefficient of *static* friction, or "stiction," is always *larger* than the coefficient of *sliding* friction (or **0.57**) between the same two materials under the same un-lubricated conditions.] Doing the arithmetic yields a friction force **Ff** of **16.55 pounds** always in the opposite direction (resisting motion) from its associated tangential force **Ft** of **11.18 pounds** (trying to induce motion). These friction forces are acting to *raise the sear* and to *drag down the contact face of the cocking piece*. [Note that the friction force **Ff** overpowers the tangential force **Ft** on each part by **5.37 pounds** at this **26.5-degree** sear angle. While we could permanently difference these two opposite-acting, co-linear forces at this point, instead we will wait and algebraically sum them all later.]

The second reason why we resolved the applied force **F** into its **Fn** and **Ft** components is because *only the normal force* component **Fn**, and *neither* of the two tangential forces **Ft** or **Ff**, is "reflected" back into the cocking piece by the angled contact face of the fixed (supported) sear.



So, now we have *three forces* acting on the contact patch of the cocking piece as shown in Figure

4. One way to calculate the net upward force **Fv** on the cocking piece is to *project* each of these three forces separately onto the vertical axis and then to *sum* them algebraically (using plus and minus signs). If we separately evaluate the vertical components of each of the three forces on the cocking piece:

> 1) The vertical component of the tangential force is just **Ft*Cos(A)**, or **10.00 pounds** acting *upward* on the cocking piece, and
>
> 2) The vertical component of the friction force is found from **Ff*Cos(A)**, or **minus 14.80 pounds** acting *downward* on the cocking piece.

The third force is the reflected normal component **Fn** of the sear reaction force, acting *generally upward* upon the cocking piece. Then,

> 3) The vertical component of this reflected force is calculated as **Fn*Sin(A)**, or **10.00 pounds** acting *upward* on the cocking piece.

Since this third vertical force component will ***always exactly match*** the size of the vertical part of the tangential force that we first calculated (because they are *mathematically identical*), we can simply *double* that first calculated vertical force.

Finally, algebraically summing the three vertical force components, we find that **the net upward force on the cocking piece is 5.2 pounds when the firing pin spring force is 25 pounds**. Perhaps some reader can devise a technique for accurately measuring this force in his rifle. Just remember that since the force of friction **Ff** acts downward along the face of the cocking piece, lubricating these parts so as to reduce friction will only ***increase*** the net upward vertical force **Fv** on the cocking piece.

Deriving an Equation

Even though we have now answered the question that I had first posed as to just how large is this upward force on the cocking piece, we might learn a bit more by examining a symbolic expression for this net upward force **Fv**. We can derive just such a suitable symbolic expression (or equation) for calculating the net vertically upward force **Fv** by repeating the steps we just verbally described above, but this time using our symbols instead of numerical values. Then, we will express each of the three forces on the cocking piece in terms of the fully cocked firing pin spring force **F**—just as we had earlier derived each of those forces. After collecting terms and factoring, we have the expression:

$$\mathbf{Fv = [F*Cos(A)]*[2*Sin(A) - Cf*Cos(A)]}$$

First, notice that the firing-pin spring force **F** only appears as a *common factor* on the right-hand side of this equation. This is what we meant earlier by saying that the result **Fv** would be "readily scalable" for different values of **F**. Also, notice the *factor of two* in this equation. If you should be fortunate enough to have a copy of Stuart Otteson's excellent but long out of print book, ***The Bolt Action, Volume I***, you may wish to pencil in the *missing factor of two* in front of the first term in his otherwise similar equation shown in Figure 9 (on page 275 of my edition) in his ***Appendix on Triggers***.

Now let us use a couple of tricks to examine the above equation more thoroughly. First, let us separately consider just the expression in the right-hand brackets of the above equation. If this part of the expression, involving only the coefficient of static friction **Cf** and the sear angle **A**, should go to **zero** separately, the net up-force **Fv** on the cocking piece would also vanish and the cocking piece would become *unable to over-ride the sear* and the trigger would then no longer be able to release the firing pin reliably. If we wish to find out just when this condition would occur with different values for our two variables (**Cf** and **A**), then we should set that part of the expression separately *equal to zero* and sort out the variables:

If $[\mathbf{2*Sin(A) – Cf*Cos(A)] = 0}$

then $\mathbf{A_{Critical} = Arctan(Cf/2)}$

For **Cf = 0.74**, our handbook value of the coefficient of static friction for dry steel on steel as discussed above, this *Critical Sear Angle* $\mathbf{A_{Critical}}$ calculates to be **20.3 degrees**. That is, the sear angle **A** must safely exceed this *Critical Sear Angle* for the rifle to be able to fire reliably, and our actual sear angle *does* measure well above this critical angle. If we *misguidedly* were to polish and super-lube the cocking piece and sear interfaces so that the coefficient of static friction **Cf** were reduced to, say, a slippery value of **0.40**, the *Critical Sear Angle* would reduce to only **11.3 degrees**, but the net up-force **Fv** on the bolt would increase to **12.00 pounds**. And probably that is not what we might have been hoping to achieve. So, then we might decide to get *really clever* and rough up the cocking piece and sear interfaces to an extreme degree by filing interlocking, very fine, 60-degree serrations across each of them, so that **Cf** goes up to its nominally maximum value of **1.00**. Then the *Critical Sear Angle* would become **26.57 degrees**. But, even though Remington's engineers had years ago arrived at essentially this same value for the sear angle **A** in their 700-series actions, your modified parts with their max-ed out frictional forces would probably no longer allow the rifle to fire reliably. So, don't serrate the sear and cocking piece interfaces either. At this point, we can only surmise that Remington's designers probably arrived at this sear angle of **26.5 degrees** after considerable real-life testing. Perhaps this angle might have been chosen to work reliably in some particularly difficult field conditions.

Oh, by the way, at this *Critical Sear Angle* $\mathbf{A_{Critical}}$ of

$$\mathbf{A_{Critical} = Arctan(1/2) = 26.57\ degrees},$$

the size of the normal force **Fn** on the sear just happens always to be *exactly twice* the size of the tangential force component **Ft** (as we had noted earlier in passing, and as suggested by the above tangent of 1/2 expression). Furthermore, for any given sear angle **A**, the vertical components of each of these two rectangular force-resolution components (**Fn** and **Ft**) must be *exactly equal to each other* in magnitude and *opposite* in direction so that they can (vectorially) sum to **zero**, since the original firing pin spring force was *purely horizontal* in our terminology.

Second, let us briefly examine the idealized case where there is *no friction at all between these two parts*. Then in this case with **Cf = 0.00**, the equation for **Fv** would simplify to:

$$\mathbf{Fv = F*[2*Cos(A)*Sin(A)] = F*[Sin(2A)]}$$

By substituting a double-angle trigonometric identity for the bracketed expression as shown, we can see that, with no friction at all, the up force **Fv** would have a *maximum* value of **Fv** (that is, **Fv = Fv$_{\mathbf{Max}}$**) at a sear angle **A = 45 degrees** and that **Fv** would vanish (i.e., go to **zero**) at **A = 0 or 90 degrees**. This all makes good sense to me. [I always like to check these sorts of things to see if my symbolic equations behave as they should. In this case, the full symbolic expression for **Fv** behaves quite reasonably—so it might even be correct, as far as it goes.] All of our analysis thus far has concerned the cocking piece interacting with the *supported sear*, mechanically held stationary by the other parts of the trigger assembly. That is to say, up to this point in our force analysis, the sear lever might just as well have been clamped in a vise. A more complete insight into the operation of the over-riding sear must await at least the "free body" analysis of the sear lever given below. These types of force analyses for the trigger parts are seldom trivial and can be quite illuminating for the gunsmith or the rifle designer. So, let us look next at the sear lever all by itself.

Calculating the Loads on the Sear

Now we will analyze the forces on a Remington-style sear lever—rotating on its front pin and supported from below by the trigger lever and sear spring. Most of the 25-pound spring force transmitted from the cocking piece into the sear pushes radially into the sear pin and is of no further interest here. But let us look carefully at the torque components about the sear pin axis from the three sear forces corresponding to the three cocking-piece forces that we found in the preceding analysis, plus the torque on the sear due to the force of the compressed sear spring. Each of the three forces that we found to bear upon the cocking piece must now be *reversed in direction* when they are considered from the "sear side" of the interface between these two parts. The net sum of the torques about the sear pin arising from each of the three forces on the sear face, as well as the torque from the sear spring, determines the "reaction torque" (and the force) that must be supplied by the trigger lever to keep the sear lever just exactly stationary while we are waiting to fire. We will now solve for this unknown size of the down-force **Fnt** exerted by the sear upon the top face of the trigger lever.

At this point, we will switch from analyzing forces to working with torques about the sear pin axis because the sear lever is constrained to allow only rotary motion about its pin. *Torque is the rotational analog of force in the linear domain*. We must have torque in order to produce torsional (twisting) stress and strain, for example. We all understand that if we pull with a crosswise force of 25 pounds upon the handle end of a one-foot-long wrench, we are exerting 25 foot-pounds of torque on the fastener engaged by the wrench. More generally, there are several ways in which we can calculate the torque about a given axis that is equivalent to a given off-axis force. [For simplicity, we are working here only with forces that are constrained to lie in a single plane that is perpendicular to our torque axis, as is the case with our firing system parts.] One method of finding the torque is to construct a radius from the torque axis to the point of application of the force, and then to project that force onto the perpendicular to that radius. Then the amount of the torque (in inch-pounds) is just the product of the radial distance (in inches) times the normal component of the force (in pounds). A different, but entirely equivalent, method of calculating the torque value for a force would be to project the "line of action" of the force until it reaches the "point of nearest approach" to the axis, and then to multiply the full amount of the force by the radial distance of closest approach (measured perpendicular to the direction of the force).

The horizontal distances from the sear pin axis to the points of application of the two vertical support forces are **0.345-inch** to the sear spring perch and **0.862-inch** to the release edge of the sear ledge where the trigger support releases to initiate the firing process. The radial distance from the sear pin axis to the top edge of the **26.5-degree** inclined cocking piece contact surface is **1.314 inches**. Looking horizontally forward from this sear face contact point, the axis of the sear pin lies at an angle of depression of about **6.5 degrees** below the horizontal.

Referring back to our work on the cocking piece, the force of static friction dominates the two co-linear forces tangent to the sear contact face (and lying in that plane) by a net of **5.37 pounds**, acting generally *upward* along the sear face. [Remember that these forces are reversed now that we are working on the sear lever instead of the cocking piece.] From the geometry of the sear itself, this force along the sear face is acting at an angle of only **20 degrees (26.5 – 6.5 = 20 degrees)** from the normal (perpendicular) to the radius from the sear pin axis to this contact point. [Remember our earlier calculation of the critical angle of **20.3 degrees** for the sear face in order for the sear to be able to over-ride the cocking piece. The sear lever cannot over-ride the cocking piece by moving vertically downward, but is constrained to pivot out of its way by rotating downward about its well forward, but somewhat lower, sear pin. ***Now we finally see why Remington had to use that 26.5-degree sear face angle!***] As this net tangential force is acting generally upward on the rear face of the sear (and is opposing sear over-ride), we will call the torque it produces a *negative* value. This torque component then calculates to **minus 6.63 inch-pounds**. The line of action of the **22.36-pound** normal force component **Fn** bearing on the sear face misses the sear pin axis by **20 degrees** and, thus, produces a **positive** torque calculating to **10.05 inch-pounds**. So, the total torque from the three forces on the sear contact face is a net positive value of **3.42 inch-pounds** (trying to over-ride the sear).

We can use a good quality trigger-pull gauge to measure the vertical up-force of the sear spring back at the top edge of the sear-to- cocking-piece contact face (in the assembled trigger unit). My example trigger used in this study is from an old Remington 40X that had been used in benchrest competition some years before I acquired it. Since the sear spring shows obvious grinding marks on both ends, it would appear that the sear spring in this trigger unit has been “softened” *mistakenly* to **1.0 pounds** of up-force measured at the rear contact point of the sear lever. This force corresponds to **minus 1.30 inch-pounds** of torque about the sear pin, and therefore the net torque acting upon the top of the trigger lever is now **2.12 inch-pounds**. Note that a slightly *stiffer* sear spring (producing, say, about **1.5 pounds** of up-force at the back of the sear) would *cancel out* more of the downward acting net torque (**3.42 inch-pounds**) on the sear and would result in a *lighter trigger pull*.

To calculate the normal force **Fnt** acting vertically downward on the trigger release surface, we must *divide* the downward-acting net torque on the sear (**2.12 inch-pounds**) by the distance from the trigger release edge of the sear to its pin axis (**0.862 inch**). Thus, a normal force **Fnt** of **2.46 pounds** is pushing downward onto the top of the trigger lever. As long as the action remains cocked, the trigger lever will supply an upward acting **2.46-pound** “reaction force” to balance exactly the net down-force applied by the sear lever.

Calculating the Trigger Pull

Now let us analyze the trigger lever itself. First, we can calculate the force of static friction **Fft** between the top face of the trigger lever and its contact patch where it supports the bottom of the sear. Even though this sear support engagement can be adjusted down from a **nominal 0.025-inch** to about **0.010 to 0.012-inch** in width (***for Target rifles only***), we still calculate the force of static friction as the product of the normal force **Fnt** (just calculated above) times a suitable coefficient of static friction. One difference here is that the static friction between these two parts should always be reduced by carefully honing these surfaces and then impregnating them with molybdenum disulfide (moly) by burnishing the trigger and sear release surfaces with high moly-content grease (sometimes called a "trigger job in a can") and then wiping off all of the excess moly grease. We also lightly polish and dry-lube the lateral bearing surfaces (sides) of the sear lever and of the upper portion of the trigger lever that goes up inside the trigger housing, as well as the inside bearing surfaces of the trigger housing itself to minimize the interference of any stray frictional drag forces with the smooth operation of the trigger unit. The coefficient of static friction **Cf** should be about **0.60** for the critically honed and moly-treated trigger-to-sear release surfaces. With a normal force of **2.46 pounds**, the force of friction **Fft** between the trigger lever and the sear is about **1.5 pounds**.

Since the trigger lever has about a **one-to-one mechanical advantage**, depending upon exactly where we pull on the trigger shoe, this friction force **Fft** is directly additive to the trigger-pull force required to overcome the adjustable trigger-pull spring alone. We can again use our accurate trigger-pull gauge to measure the part of the trigger pull due to the (adjustable) trigger-pull spring alone (with sear contact prevented by engaging the safety or by opening or removing the bolt). If this pull had been adjusted down to **1.0 pounds**, then the total trigger pull should be **2.5 pounds**.

In the un-modified "as-designed" trigger geometry, the "line of action" of the sear down-force on the trigger lever seems to pass about **0.0275-inch aft** of the trigger pin axis, located about **1.045 inches** below the sear ledge. Thus, the factory trigger release angle, a ***positive*** **1.5 degrees*****, is an important design consideration in any type of trigger mechanism.*** A close examination of this particular trigger reveals that the contact face at the top of the trigger lever in this competition-modified trigger unit had long ago been honed by someone to a *zero degree* release angle so as to eliminate all sear lift during trigger release. So, in this case, the force of friction **Fft** at the top of the trigger lever plus the trigger-pull spring force adjustment should be the actual trigger release force for this modified trigger. Finally, we can use our trigger-pull gauge just as it was intended and measure the actual force required to release the sear and fire the rifle. How well does this calculated **2.5-pound** trigger pull agree with our measurements of the trigger pull required to release the striker? [Unfortunately, some of these trigger part were used to effect an emergency repair on a customer's rifle before I got around to measuring the actual trigger pull carefully. But, the calculated value of **2.5 pounds** is quite reasonable and agrees with my memory of the "feel" of that trigger.] But, will this modified-geometry trigger reset itself if it is somehow only partially pulled? **No**, not in this reduced-pull, competition-modified trigger, ***but it positively must do so in a hunting rifle***. [The answer to this question should be of more than academic interest to gunsmiths and to their customers.] The benchrest competitor is aware of this and other safety limitations with his competition rifle.

The Effect on Trigger Pull of a Positive Release Angle

Does pulling the trigger (in the factory trigger geometry) initially raise the sear and retract the firing pin (even just a little) before releasing it? **Yes**, the forward-rotating top end of the trigger lever must raise the sear lever very slightly before it clears the front edge of the sear ledge. Just at the moment of release, the sear should have been raised by **about 1.0 thousandths of an inch** (at the cocking piece contact point) if the amount of trigger-to-sear engagement has been left adjusted to a nominal **0.025-inch**. This calculation is based on the height of a **1.5-degree** *wedge* with a base width of **0.025 inches** (the sear engagement), amplified by the mechanical disadvantage (working backwards) of the sear lever itself (**1.515**). In raising the sear lever, the moving trigger lever enjoys a mechanical advantage of **38:1** at this small **1.5-degree** wedge angle. This slight raising of the **26.5-degree** sear contact face will retract the firing pin by **one half of this amount (or 0.0005 inch)** during the pulling of the trigger if no motion is lost in taking up slack or in the bending of any parts.

A more complex “second level” analysis is required to compute the trigger-pull at the instant at which the factory trigger releases. In this situation, the trigger lever train is now ***actively loaded at both ends***. The motions of the sear and cocking piece are now “backwards,” so that the frictional forces between them are now reversed in direction from what they had been in earlier analyses. Also, we are now dealing with sliding friction rather than with static friction. And, there is no longer any single place where we can start our analysis so that we can proceed straight through to final results.

If we were planning to go forward with this analysis, we should pause and write a little computer program at this point to model the entire fire-control system mathematically, complete with all its spring forces and all the friction forces, plus the trigger-pull force acting on each of its parts. We must closely examine the friction within the bolt shroud caused by retracting the cocking piece **0.0005-inch** via a sliding **26.5-degree** contact with the sear lever. More up-force on the cocking piece causes more friction inside the bolt shroud, which requires more up-force to overcome it, et cetera. As these computed loads on the cocking piece increase, the loads on the sear and trigger levers also increase. This program would have to *iterate* its computations back and forth, from one end to the other, through the train of the firing system levers until only negligible changes were being computed during each iteration. [One difficulty to be expected is the creation of a model, and its associated iteration control strategy, that converges, rather than oscillating or wildly diverging (as in extrapolating with an earth climate model, for instance).] I will hazard a guess that such a procedure would finally yield a trigger-pull increase of about **2.5 to 3.5 pounds** greater than what we had calculated above for the “zero-release-angle” trigger. This would indicate that a minimum safe trigger pull would be about **5 to 6 pounds** for the factory trigger geometry, which is in line with my experience with these triggers.

Another value that we could determine as a result of this computerized analysis would be the up-force on the cocking piece at the moment of trigger release. I suspect that this actual vertical force **Fv** might be closer to **50 pounds** (with a factory-geometry trigger) than to our initially calculated value of **5.2 pounds** (if we are employing a competition benchrest trigger).

Summary

We have completed a simplified analysis of the forces affecting the primary working parts in the popular Remington 700 style bolt-action cocking piece and trigger unit. We have demonstrated all of the calculations in the order in which they must be done. These analytical techniques are generally applicable to other trigger designs. We started out merely to calculate the static up-force on the cocking piece at the rear of a Remington-style bolt. We found this up-force to be **5.2 pounds** when using a nominal **25-pound** firing pin spring with a non-resetting trigger. Then we developed a symbolic equation to allow the calculation of this up-force and uncovered a small oversight in the most authoritative reference on the subject. Eventually, realizing that we were already half way through the complete "first level" analysis, we decided to go on and analyze the sear and trigger levers in turn. With one simplification (a non-resetting trigger release), we were finally able to calculate the trigger pull for an old, benchrest-modified trigger unit. Someone with better measuring equipment, or with access to factory blueprints, could refine the accuracy of these calculations. Then, we briefly indicated some of the complexity involved in considering the self-resetting trigger with its positive release angle.

Along the way, we pointed out a couple of popular trigger modifications that are actually *counter-productive* in attempting to reduce the trigger pull:

1) Do not reduce the friction of the sear-to-cocking-piece contact faces, and

2) Do not lighten the sear spring if you want a reliable, lightweight trigger pull.

And, for safety and liability reasons, do ***not*** modify the positive **1.5-degree** trigger-to-sear release angle designed into these triggers for any reason. Nowadays, you should obtain any of several excellent aftermarket "three lever" triggers for use in competition.