Spring Buzz and Failure in Spring Piston Airguns

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1.0 Summary

The tendency of the main spring of an airgun to break close to its rear end coils and the buzzing sound often felt by the shooter immediately after firing the gun are two related issues. Spring failure commonly results from fatigue stress, which this study shows can be significantly more intense near the rear end of the spring. The spring dynamics have significant impact on the maximum stresses the spring experiences during the firing cycle and must be included in fatigue life calculations. Most of the dynamic activity in the spring happens after the projectile has left the barrel and is therefore not detrimental to accuracy. To the extent that the buzzing sound originates in the spring longitudinal dynamics, spring buzz may be annoying but doesn't have a negative impact on accuracy. Computations show that tight rear guides are effective in reducing buzz and potentially extending the spring fatigue life.

2.0 Introduction

This work builds up on my previous research on spring piston airguns. I will assume you are familiar with the workings of a spring piston air gun. If you are not familiar with the internal components of this type of airgun, please read at least the introduction in Ref. 1, available at Researchgate.net.

As was the case with my previous work on airguns, the primary audience for this article are advanced undergraduate students in engineering or physics, as well as airgun tuners and aficionados with a quantitative interest.

When you release the trigger of a cocked and loaded airgun[1], the compressed main spring, which is the source of power for the gun, propels the piston forward, compressing the air in the compression chamber behind the projectile, which is seated in the breech. When the air reaches a pressure that overcomes the breaking force needed to start the movement of the projectile into the bore, the projectile quickly accelerates and exists the gun. By the time the projectile begins to move into the bore, the piston has already advanced almost to the end of the compression chamber. Typically, after the pellet has moved a short distance within the barrel, the piston reaches the end of the chamber and bounced backwards, compressing the spring (the piston may or may not impact the end of the chamber before

1. Unlike firearms, which can be cocked and dry fired (trigger release with the gun unloaded), spring piston airguns must be loaded when fired lest damage to the internal components results.

bouncing, depending on the power of the spring and other parameters - for details, please see Ref. 1. After the pellet leaves the barrel, inertia causes the piston to continue to compress the spring, which then propels the piston forward yet again. In theory, more than one bounce may occur, depending on friction between the piston seal and the chamber walls, and friction between the spring and its guide or guides, and between the spring and the inner surface of the piston.

It often happens that the shooter hears a loud buzzing sound when the gun is fired. This sound is commonly viewed as undesirable and annoying, its origin is usually attributed to the spring, and there isn't a consensus as to whether the vibrations that translate into this buzzing sound are detrimental to accuracy. As a result, a sizeable service industry has emerged whose business it is to make the firing cycle of spring airguns smoother, getting rid of the vibrations linked to this buzzing sound. The individuals engaged in this line of service are air gunsmiths known as "tuners".

Another source of distress to spring airgun aficionados is that airgun springs often break. The breaking of the spring in an airgun is not a catastrophic event in that, in most cases, the gun continues to function at a somewhat reduced power level, and in many cases the shooter doesn't notice his gun as a broken spring. Springs usually break near their rear ends, causing a collapse of one or two rear coils. Cursory inspection of the broken coil reveals what is clearly identifiable as fatigue failure, as illustrated in Figure 1.

FIGURE 1. Fatigue failure of an airgun spring - failure happened in the third rear coil.

A peculiarity of fatigue failure of main springs is that it happens with a frequency that is inconsistent with standard fatigue design practice of mechanical components. Off-the-shelf airgun springs tend to break after thousands of firing cycles, while mechanical components, such as valve springs in internal combustion engines or orthopedic metal implants, are designed to last millions of loading cycles. The question emerges as to whether the dynamic response of the spring during firing, which is related to the buzzing sound that characterizes off-the-shelf airguns, does effectively increase the number and perhaps the intensity of the loading cycles the spring is subjected to.

These considerations lead to the following questions.

1. Does the fatigue behavior of the spring respond to an effective number of loading cycles much greater than the cocking/firing cycles, due to the dynamic response of the spring during firing?
2. If the buzzing sound originates in the dynamic response of the spring, does this mean that by tuning the gun so as to suppress the buzzing sound we are also reducing the effective number of spring loading cycles, thereby extending its useful life?

As will emerge from the analysis that follows, the answer to these two questions is yes.

Since the dynamic behavior of the spring during firing is central to the issues of fatigue and buzzing, this investigation will focus on the motion of the spring from the moment the trigger is released.

Firing a spring airgun causes the sudden release of the cocking rod, as Figure 2 illustrates. To visualize what happens when the trigger is released, let’s consider two situations; one where the spring is idealized as an elastic energy storage device without mass of its own, and a more realistic situation where the spring stores elastic energy but has non-negligible mass. A spring whose mass is not negligible is referred to as a heavy spring[1].

In the case where the spring has no mass of its own, there is no dynamic response by the spring itself after the trigger is released. Assuming there is no friction between the spring and its guide, the spring delivers its elastic energy by expanding uniformly, with all its coils separating from each other at the same rate. In other words, an ideal mass-less spring doesn’t form any waves propagating up and down its coils. Such a spring would not produce any buzzing.

Real-life springs are not massless, however. In fact, a typical airgun spring has a mass that cannot be neglected in comparison with the mass of the piston. In the case analyzed here (the same air rifle considered in Ref. 1), the spring mass is 0.105 kg, while the mass of the piston (including seal and cocking rod) is 0.298 kg. In other words, in this case the spring mass is about 30% that of the piston assembly.

There are two issues related to the spring mass. One issue is how the spring mass affects the movement of the piston, another is how the spring mass affects the spring dynamics. An ad-hoc way of accounting for the mass of the spring on the piston movement, at least to some extent, is to assume that a fraction of the mass of the spring is added to the mass of the piston (Galloni, Ref. 2) and proceed with the internal ballistics computations ignoring the spring mass, as discussed in Ref. 1. This procedure works for harmonic oscillators[2], but in our case it must be viewed as a rough ad-hoc correction.

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1. Heavy doesn’t mean the spring weighs a lot; it means the spring mass is taken into account in computing its dynamic properties.
2. In the case of an object suspended from a spring which is compressed and then released, the fraction of spring mass to be added to the object to get the correct period depends on the relative magnitude of both masses.

When the trigger is released and the spring is free to start moving forward overcoming its own inertia and the pneumatic pressure of the air in the chamber, the coils nearest the front end of the spring are the ones that move first, while the ones behind tend to lag due to their own inertia. The coils begin to separate from each other starting at the front of the spring, in a wave-like configuration that propagates backwards. As the spring expands, an assortment of waves are formed, of which the most important are longitudinal compression and expansion waves moving up and down the spring. These waves are stimulated by the piston movement and bounce. This backward and forward wave propagation intensifies the spring compression at its ends, and it is this difference in the level of coil deflection due to the dynamic response of the spring that is responsible for the shortened fatigue life of coils near the ends.

It appears natural to assume that the dynamic response of the spring is also responsible for the buzzing sound we hear when a shot is fired. The computations show that the vast majority of this dynamic activity takes place after the projectile has left the muzzle, and is therefore not directly detrimental to accuracy. Tightening the rear guide can significantly dampen spring oscillations, eliminating much of the buzz and presumably prolonging the spring life (here I say presumably because to make a positive statement this assumption needs to be verified empirically in a statistically significant manner).

A rigorous analysis of spring dynamics is a complex undertaking with a long history, dating back to the 19th century (Love, Ref. 3). Spring vibration is much more complicated than what everyday experience may suggest. Due to the combination of stresses the spring is subjected to, the familiar longitudinal form of vibration is only one aspect of the spring dynamic response (Wittrick, Ref. 4).

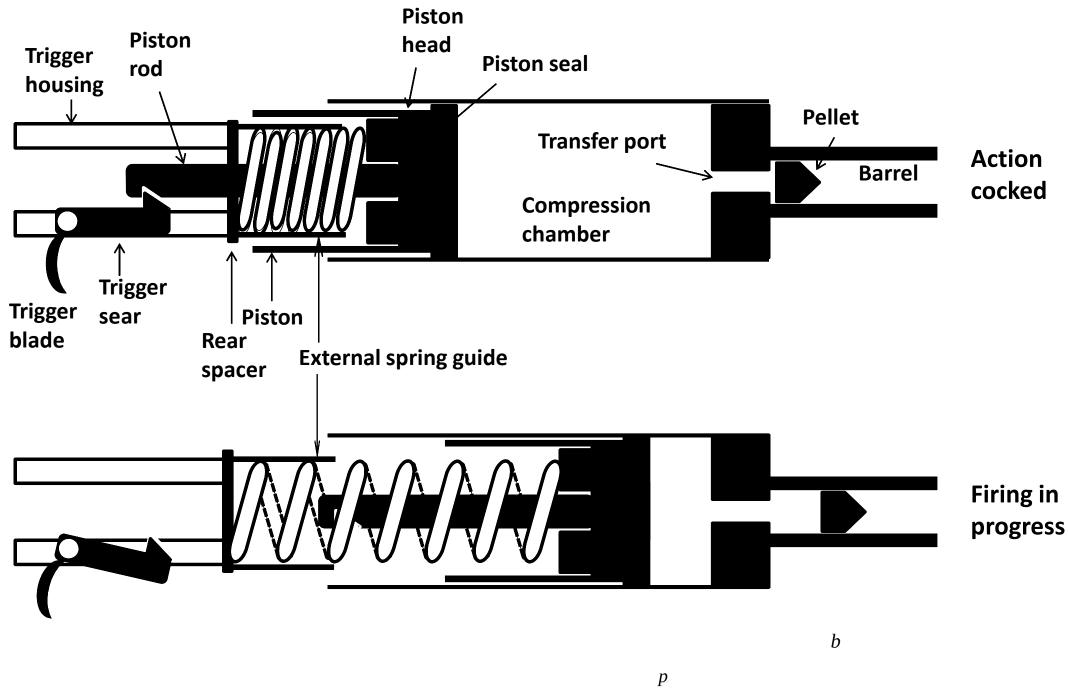
The objective of this work is to assess the airgun spring dynamic response using the simplest possible model with a minimum of mathematics. The simplest model is a one-dimensional idealization of the spring treated with an elementary form of finite element analysis, as described in detail in Ref. 1. A one-dimensional spring model means we treat the spring as a slender rod capable of sustaining longitudinal waves only. Under this assumption, we can use the same simple math of linear wave propagation in a slender rod to come up with a wave speed that applies to the spring[1]. Calculating the wave speed theoretically is an important step in benchmarking the solidity of our computations. It is important to realize that this simplistic model neglects other forms of the spring vibration (see Sorokin, Ref. 5, for an excellent review), which I will assume are not as important as longitudinal vibrations as far as fatigue is concerned. To what extent other forms of spring vibration are important to buzzing, on the other hand, is something that needs experimental verification.

3.0 Analysis

The following figure, a variation of which is explained in greater detail in Ref. 1, shows a conceptual schematic of a spring piston airgun action.

1. If the rod were not slender, the equations of elasticity would introduce radial waves into the picture, which are not relevant to our very simplistic spring model.

FIGURE 2. Conceptual schematic of a spring piston airgun action, fitted with an external guide.



A difference between the action shown in Figure 2 and the same action discussed in Ref. 1 is the presence of a spring guide. Spring guides can be external, internal, or a combination of both. The spring guide in Figure 2 is external and consists of a tube within which the spring coils slide as the spring expands and contracts. An internal guide would be similar, only it would be inside the spring, and the cocking rod (piston rod) would slide inside the guide[1].

Compression ratio. The force acting on a spring of free length L, which at time $t > 0$ has length $L_s(t)$, is

$$F = K(L - L_s(t)) \tag{EQ 1}$$

where K is the spring stiffness factor. If $L_s(t) < L$ there is a compression force.

Defining the spring compression ratio as follows (this definition is intended to associate a positive force with compression),

$$\eta = \frac{L - L_s(t)}{L} \tag{EQ 2}$$

the force acting on the spring is

1. A popular tuning kit consisting of a spring and a tightly fitting external guide made of synthetic material is produced and marketed by Vortek Products Inc.

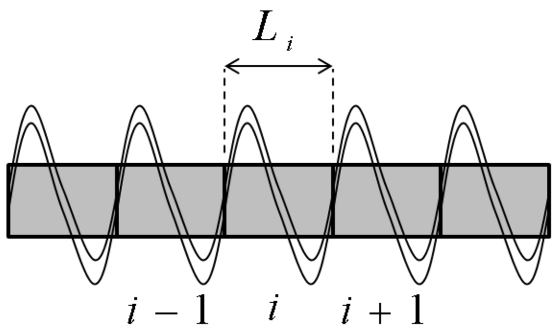
$$F = KL\eta \quad \text{(EQ 3)}$$

If we represent the spring by discrete segments of length L_i, where subscript i identifies the position of the segment along the spring (see Figure 3), the magnitude of the force acting on the i^{th} segment is

$$F_i = K_i L_i \eta_i \quad \text{(EQ 4)}$$

where $\eta_i = \frac{L_i - L_i(t)}{L_i}$ is the compression ratio of the i^{th} segment.

FIGURE 3. Spring represented by continuous segments (piston is on the right, rear end is on the left).



If the spring is initially uniform (meaning all coils are equidistant and coil and wire diameters are uniform), and all segments are of equal initial length,

$$K_i = nK \quad \text{(EQ 5)}$$

where n is the number of segments. The number of segments doesn't have to be the same as the number of coils, but it is intuitively appealing to assume they are equal.

The main spring of an airgun typically has two states of compression. The spring is often, but not always, in a pre-compressed state when the gun is un-cocked, and is further compressed when the gun is cocked. Table 1 shows the lengths and compression ratios relevant to our airgun (see Ref. 1 for additional details about this particular airgun).

TABLE 1. Spring lengths and compression ratios.

	Spring Length (m)	η
Free	0.2857	0
Solid	0.1234	0.5681
Uncocked	0.2330	0.1845
Cocked	0.1330	0.5345

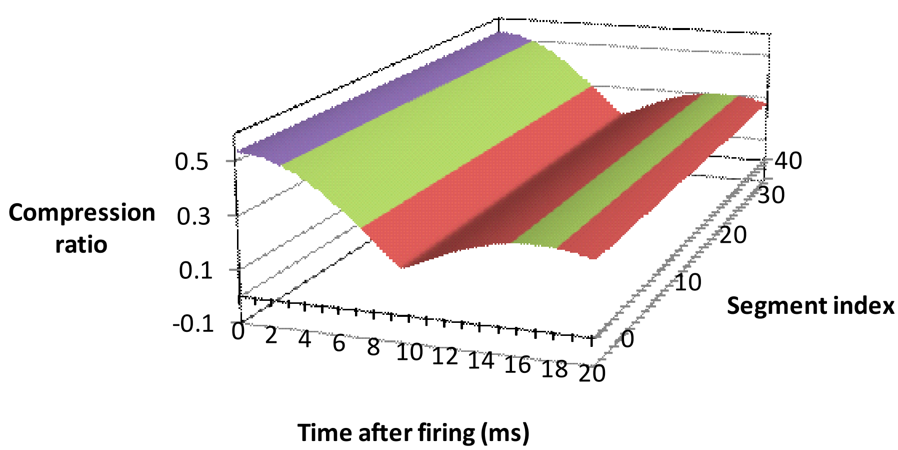
Notice that when cocked, this spring is barely under 10 mm longer than its solid length. The difference between compressed length and solid length is known as the *clash allowance* of the spring. In our case, the clash allowance is less than 4% of the spring free length. This is a very small value when compared with most spring applications, where the recommended clash allowance is significantly greater. The small clash allowance suggests this spring has been preset[1], either before installation, or as a result of compression on cocking. The correct free length to use is the length after presetting. For details on presetting and how it influences fatigue life, see SAE, Ref. 6. As we will see, one of the consequences of the small clash allowance is that, in the absence of enough friction, the spring oscillations cause some of the coils to clash against each other.

The implementation of this simple one-dimensional finite elements model, coupled with the thermodynamics of the air in the compression chamber and the dynamics of the piston and the projectile, results in set of ordinary differential equations you can integrate numerically over an arbitrary time horizon. Ref. 1 contains full details of implementation, here I will limit the discussion to the interpretation of results.

3.1 Massless spring response

Lets first consider what happens to the spring is we assume the spring has no mass. In this case, all the coils separate from or approach each other uniformly, and all segments have the same time-varying compression ratio (I will assume there is no friction acting on the massless spring). A practical way to visualize what happens to the spring is to plot the compression ratio of the segments as a function of time and the segment index. For ease of visualization, it is convenient to display a continuous function of both time and segment index, as Figure 4 shows.

FIGURE 4. Compression ratio in the massless spring case in the first 20 ms after firing (piston first bounce at 9.5 ms, pellet exit at 10.9 ms).

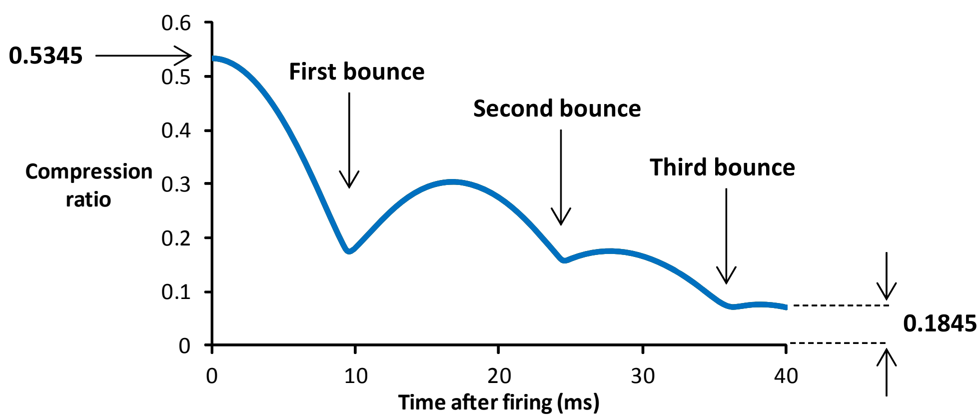


1. Presetting causes permanent internal shear strain in the spring, making it more resistent to fatigue.

In our case we have 41 active coils, and we assume we have 41 segments. As the spring expands after firing, the compression ratio drops. Shortly before 10 ms, the piston bounces backwards and the spring compresses again. The projectile leaves the barrel shortly after 10 ms, before the spring returns to its un-cocked initial state. Figure 4 shows a planar surface because all segments expand or contract uniformly.

Figure 5 shows how the compression ratio of any segment in the ideal spring evolves in time. The piston bounces at least three times before it settles in its un-cocked position. The spring compression ratio evolves from 0.5345 in it cocked configuration, down to 0.1845 in the un-cocked configuration.

FIGURE 5. Individual segment compression ratio for the massless spring - cocked compression ratio is 0.5345, un-cocked 0.1845.



Even in the absence of friction between spring and guide, the spring will eventually settle down in the un-cocked configuration as energy is dissipated by friction between the piston and the chamber walls, as well as by heat formation due to air flowing non-isentropically in and out of the chamber through the breech.

3.2 Heavy spring response

The introduction of spring dynamics dramatically changes the situation illustrated in Figures 4 and 5. The spring dynamics are dominated by wave motion that starts at the forward end of the spring at the time of firing. As mentioned in the introduction, a full description of the spring motion is a very complicated business, but we can gain significant understanding by neglecting the torsional and lateral dynamics, and assuming the spring behaves like an axially elastic rod. For simple cases such as a weight suspended from a spring, this assumption allows us to get analytical solutions for the spring motion (Galloni, Ref. 2). Although analytical treatments such as the one in Ref. 2 could in principle be extended to more complicated situations, given the complexity of our case, it is easier to rely on numerical simulations.

The first step in understanding the spring response is to characterize the wave motion that starts when the trigger is released. Upon release of the trigger, a de-compression wave starting at the forward end of the spring will propagate backwards with a velocity that immediately after onset depends on the spring parameters only.

Spring characteristic speed. If you bang one end of a steel rod with a hammer, a sound wave ensues which propagates down the rod with the speed of sound in steel, which is a function of the steel modulus of elasticity (Young's modulus) and the steel density. If you bang one end of a spring with a hammer, a compression wave results which propagates down the spring with a velocity that depends on the spring stiffness. Since a spring is typically much softer in the axial direction than the steel it is made of, the compression wave in the spring will propagate with a velocity that is much smaller than the speed of sound in steel. You can exploit this parallel between a sound wave and a compression wave to determine the initial wave speed in the spring[1].

The axial stress σ in a thin rod with Young's elastic modulus E and density ρ is related to the rod axial strain, $u(x, t)$, as follows,

$$\sigma = E\frac{\partial u}{\partial x} \qquad \text{(EQ 6)}$$

where x is the axial coordinate, which lines up with the rod axis of symmetry. If there are no external forces acting on the rod, linearized analysis shows that longitudinal waves in the rod propagate with velocity (Landau and Lifshitz, Ref. 7)

$$v = \sqrt{\frac{E}{\rho}} \qquad \text{(EQ 7)}$$

You can now exploit this relationship to derive the initial wave propagation velocity in the gun spring when the shot is fired, assuming there are no friction forces. To do this, you replace E and ρ with quantities relevant to the spring deflection.

Within a linearized framework, the role of $\frac{\partial u}{\partial x}$ in the discretized model of the spring is played by the compression ratio with negative sign, $-\eta_i(t)$, where the subscript identifies the finite elements segment. The role of σ is played by the force per unit of cross-sectional area of the spring (with negative sign),

$$\frac{F_i}{A} = \frac{K_i L_i}{A}\eta_i \qquad \text{(EQ 8)}$$

1. What follows can be done with dimensional analysis alone, but in that case any constants in multiplying a non-dimensional group would remain unknown.

This tells you that $\frac{K_i L_i}{A}$ plays the role of E in EQ 6. To get the relevant proxy for ρ, we ask what is the density engaged in a longitudinal wave down the spring. Within a one-dimensional analysis, this density is the mass of a single segment divided by the initial volume of that segment (the volume before any waves set in). Assuming the length of the elements is the same as the distance between coils, the density relevant to the spring dynamics is

$$\frac{m_c}{L_i A} \qquad \text{(EQ 9)}$$

where m_c is the mass of a single coil, and L_i is the distance between coils in the spring initial state (right before firing). Replacing in EQ 7, the speed of wave propagation in the spring is

$$v_s = \sqrt{\frac{K_i L_i^2}{m_c}} \qquad \text{(EQ 10)}$$

If the spring is uniform and the segments are all equal in length, EQ 10 can be written in terms of spring quantities, rather than coil quantities,

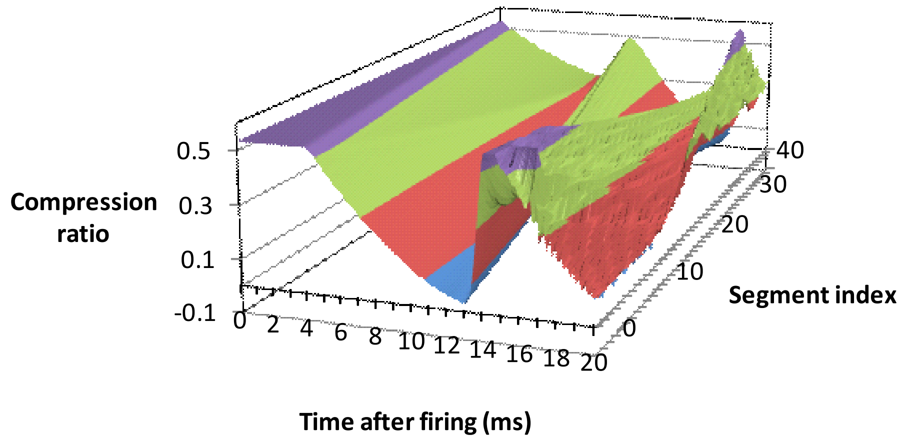
$$v_s = \sqrt{\frac{KL^2}{m}} \qquad \text{(EQ 11)}$$

This particular spring has 41 effective coils, the wire diameter is 3 mm, and the outer diameter is 17.78 mm. With steel modulus of $7.93\ 10^7$ kPa, density of 7750 kg/m3, and cocked length of 0.133 m, EQ 11 gives us an initial wave speed of 32.23 m/s. It is important to realize that EQ 7 holds under assumptions of linearity. This means the density changes very little as a result of the sound wave. This isn't the case in our spring, however. The expression in EQ 9 can change significantly as the spring wave propagates, as will become clear shortly. In addition, in EQ 6 and in the equation of motion that leads to EQ 7 there is an underlying assumption of continuity, which the finite element approximation violates. For these reasons, the calculated wave speed is only an approximate result and the wave velocity we compute with our simple finite elements model will differ from 32.23 m/s.

3.2.1 No-friction (loose guide) case

Let's first look at what happens if we neglect the effect of friction between spring and guide(s). This situation occurs when the spring guide is loose and makes little contact with the spring, as is often the case with a new spring gun. Figure 6 shows the spring compression ratio as a function of time, in milliseconds, after trigger release (as was the case with Figure 4, the plot shows a continuous surface for clarity) assuming there is no friction between the spring and its rear guide. Comparing Figure 6 with Figure 4, it is obvious that the spring mass introduces a great deal of complexity in the dynamics.

FIGURE 6. Compression ratio of 40 spring segments as a function of time (in milliseconds), no guide friction - notice this is a continuous depiction of discrete quantities.



The flat, triangular area starting at time zero in Figure 6 represents the finite element segments not yet reached by the expansion wave originating at the front end of the spring (index 40[1]). From the plot, you can see that the wave front takes between three and four milliseconds to reach the rear of the spring. This is in keeping with the theoretical value of 32.23 m/s

What follows after the first wave arrives at the rear of the spring is a combination of forward and backward waves of increasing complexity. Notice the feature (in purple) at about 15 ms and small segment indexes. There is what looks like a hill with a flat top. This represents coils that have clashed together. Since the solid length compression ratio cannot be crossed, this is a situation where inertia causes some of the coils to come into direct contact with each other.

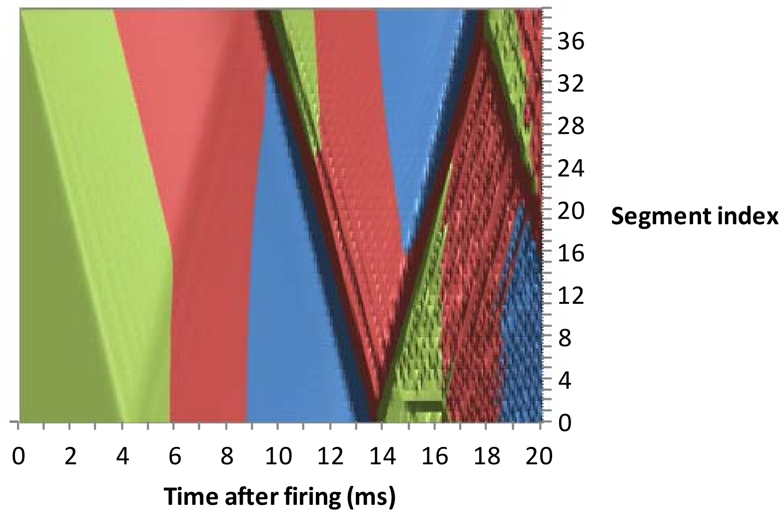
You get another view of the wave pattern by looking at Figure 6 from above. This is what Figure 7 shows. The pattern of straight lines corresponds to relatively constant wave speeds, and the left and right tilting of the lines correspond to backward and forward moving waves. The slope of the lines in Figure 7 are consistent with the theoretical wave speed we calculated earlier.

To visualize the complex pattern of interacting waves it is convenient to plot the time evolution of the compression ratio at three locations in the spring. Figure 8 shows the rear, middle and front segments compression ratio during 40 ms after firing, in the absence of

1. There are 41active coils, but the plot looks better if highest value in the depth axis is 40 rather than 41.

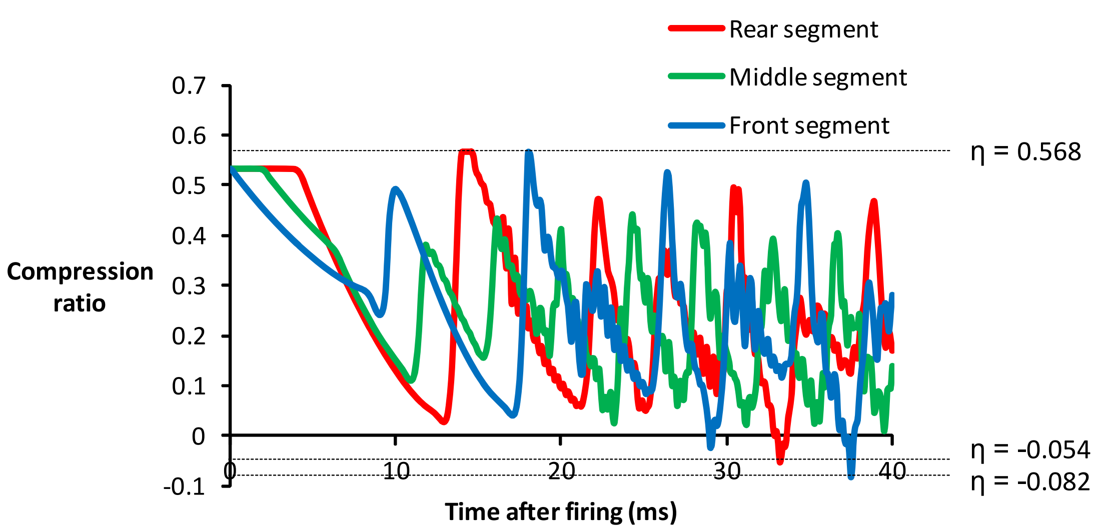
guide-spring friction. Comparison with Figure 5 reveals startling differences introduced by the spring dynamics.

FIGURE 7. Top view of compression ratio plot of 40 coil segments, same parameters as in Figure 6.



The middle segment (green) shows much less activity than the end segments. Notice that the front segment (blue) begins expanding immediately after firing, while the rear segment (red) begins expanding after about 4 ms (in keeping with the wave speed of 32.23 m/s). The blue line clearly shows the bounce of the spring (at about 9.5 ms), when the front segment begins compressing again, while the rear and middle segments continue to expand. Notice that the red and blue waves are out of phase, with their peaks and valleys close together. This explains why the middle segment is much less active.

FIGURE 8. Time evolution of compression ratio of three segments during the first forty milliseconds after firing. No friction between spring and rear guide. Compare with Figure 5.



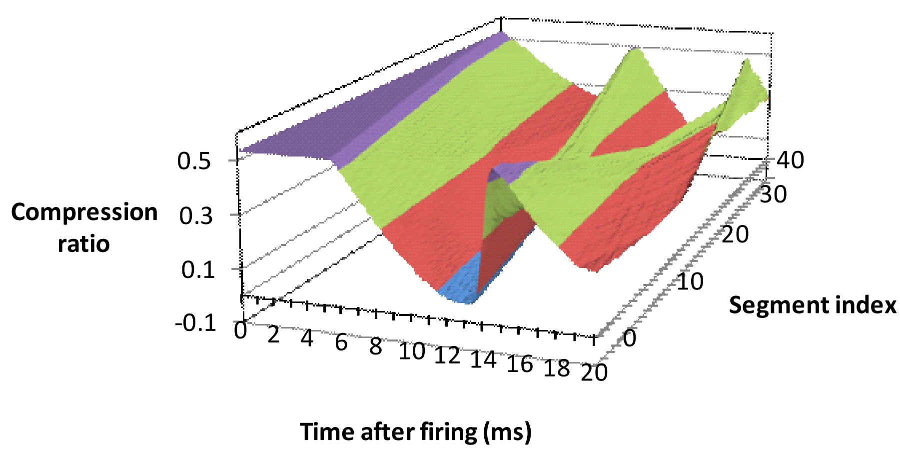
Notice that in some instances the spring segments are, during brief periods, in tension (negative compression ratio). This surprising result adds to the range of dynamic stresses that contribute to fatigue[1]. The red line in Figure 8 clearly shows that the rear segment compression ratio tops off at 0.568 when the coils clash.

If there were no guide friction to dampen the spring oscillations, Figure 8 tells us the front and rear coils of the spring would be highly stressed, with stress oscillations ranging from slightly negative to clash values, with the rear coils especially stressed. In the absence of a tight guide not only are the stresses high, but the effective number of cycles that influence fatigue is much higher than the number of times the gun is cocked and fired.

3.2.2 Effect of guide friction

Guide friction, achieved through tight tolerances between the spring and its guide, can have a dramatic effect in dampening spring oscillations. The next three figures show results for an initial rear guide friction force of 100 N (about 22 lb). This is the force required to cause the spring to slide out of the guide. As the spring expands and slides out of the guide, the net friction force acting on the spring drops, since only the coils in contact with the guide are affected by friction. In the computations, the friction force is divided by the number of coils initially within the guide, and this force per coil is assumed to remain constant for all coils in contact with the guide. The effect of guide friction is independent of whether the guide is external or internal. In the results that follow, the guide is taken to be 100 mm in length, which means the guide initially affects about 75% of the spring coils.

FIGURE 9. Compression ratio of 40 spring segments as a function of time after trigger release (in milliseconds), under initial guide friction of 100 N (compare with Figure 7).



1. Although the results in Figure 8 assume the spring and guide have no friction, there is friction between the piston and the compression chamber wall.

Figure 9 shows the compression ratio in the presence of guide friction. This figure appears to be very similar to the case without friction, but there are significant differences. Notice that, unlike the case of Figure 8, there are no flat-top peaks. This means none of the coils clash. The surface is also smoother, suggesting a lower content of high frequencies. Some of the differences become more clear when you view the plot from the top, as shown in Figure 10.

Although a force of 100 N may sound like a lot of friction between the spring and its rear guide, in this particular gun its effect is to lower the muzzle energy by just two Joules (or about 12 m/s lower velocity for a 0.67 g pellet).

Figure 10 clearly shows that the initial wave moving from the front to the back slows down as a result of friction. This is indicated by the break in the line that separates the dark and light green areas near the left side of the figure.

FIGURE 10. Top view of compression ratio plot of 40 coil segments, same parameters as Figure 9 (compare with Figure 8).

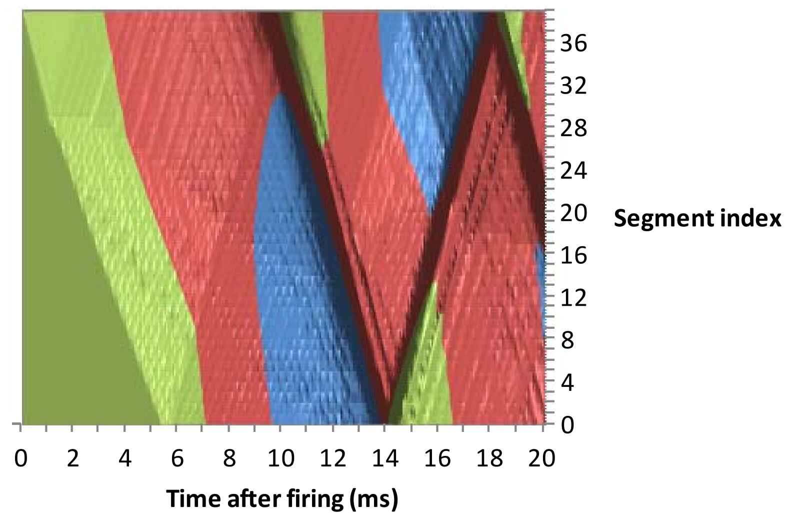
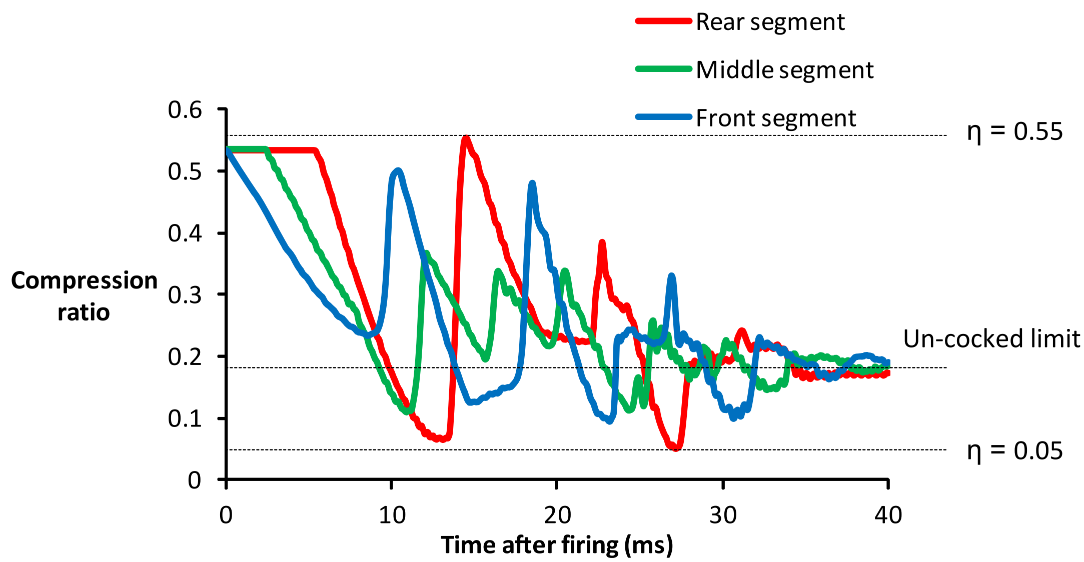


Figure 11 is the most revealing as to the effect of guide friction on the spring dynamics. Friction between spring and guide has a remarkable effect on quieting down the spring oscillations. After about 30 ms, the oscillations practically disappear, and the spring compression settles down to the un-cocked limit of 0.1845 (see Table 1).

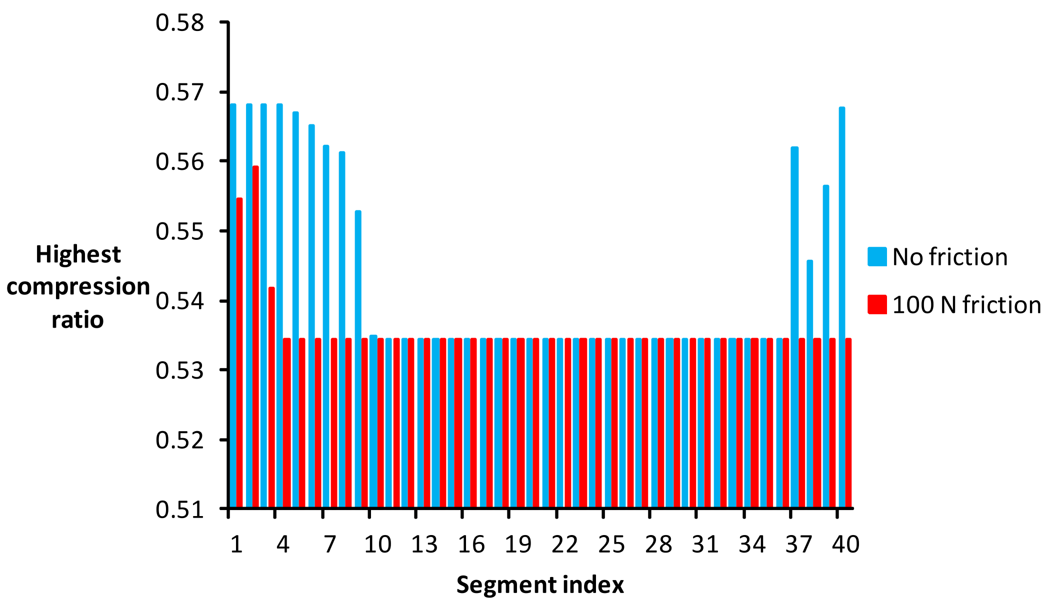
You can now compare what happens to the coils in the rear end, center, and front end of the spring (actually, in the case of guide friction analyzed here, the highest compression occurs inside the guide close to the guide front end, with the rest of the segments ahead of the guide being relatively uniformly stressed - this justifies looking at the front segment as representative of a highly stressed part of the spring). Notice that the red line (rear segment) in Figure 11 shows significantly more dramatic up and down swings than the blue (front segment), and green (middle segment) lines. Notice also that, as expected, negative compression (expansion) of the spring is no longer present.

FIGURE 11. Time evolution of compression ratio of three segments during the first forty milliseconds after firing, with 100 N rear guide friction - compare with Figure 8.



Since different segments of the spring reach their highest level of compression at different times, it is easier to appreciate the differences in peak compression values if we plot the highest compression ratio of each segment over the simulation horizon of 40 ms. This is what Figure 12 shows.

FIGURE 12. Highest compression ratio over 40 ms for a no-friction case and a case with 100 N of total guide initial friction force (guide length 0.1m, compressed spring length 0.133 m).



It is clear from Figure 12 that the effect of friction is to make the highest compression ratio fairly uniform across all spring segments. Intuitively, we expect that the parts of the spring subjected to friction from the guide will see their compression swings ameliorated. What Figure 12 shows is that *all segments*, including the ones that are never in contact with the guide, have their compression peaks reduced. The effect of a tight rear guide is to clip off the compression peaks at *both ends* of the spring, but especially at the front end. If you think about the spring inertia as the reason why compression waves form, then the fact that a tight guide reduces the stress of coils not in contact with the guide becomes clear. The spring region not affected by the tight guide involves a smaller number of coils whose inertia is brought to bear in the formation of compression waves, while at the same time, the coils affected by the guide are inhibited from forming waves travelling to the front of the spring as effectively as in the absence of friction. It therefore makes intuitive sense than a tight rear guide will be beneficial to portions of the spring ahead of the guide.

As we will see in the next section, when comparing the loose guide (no friction) with the tight guide set up, peak compressions occur at very different times and different places.

3.3 Sequence of spring events during firing

We will focus on four events: a) the first piston bounce, which happens about one millisecond before the projectile exists the barrel, b) the first piston re-bounce (or forward bounce), which happens after the pellet has left the barrel, c) the maximum compression in the rear half of the spring, and, d) the highest forward compression in the forward half of the spring. The guide friction strongly influences the points in time of the compression maxima.

In the discussion that follows, I will identify the no-friction case with a loose guide and the friction case with a tight guide. I will associate the loose guide with an internal rod, and the tight guide with an external sleeve. This doesn't mean that a tight guide must be in the shape of a sleeve, and that a rod cannot be a tight guide. It is usually the case, however, that an off-the shelf airgun is fitted with a rather loose internal guide, and that the easiest way to have a tight guide is to install a sleeve external guide.

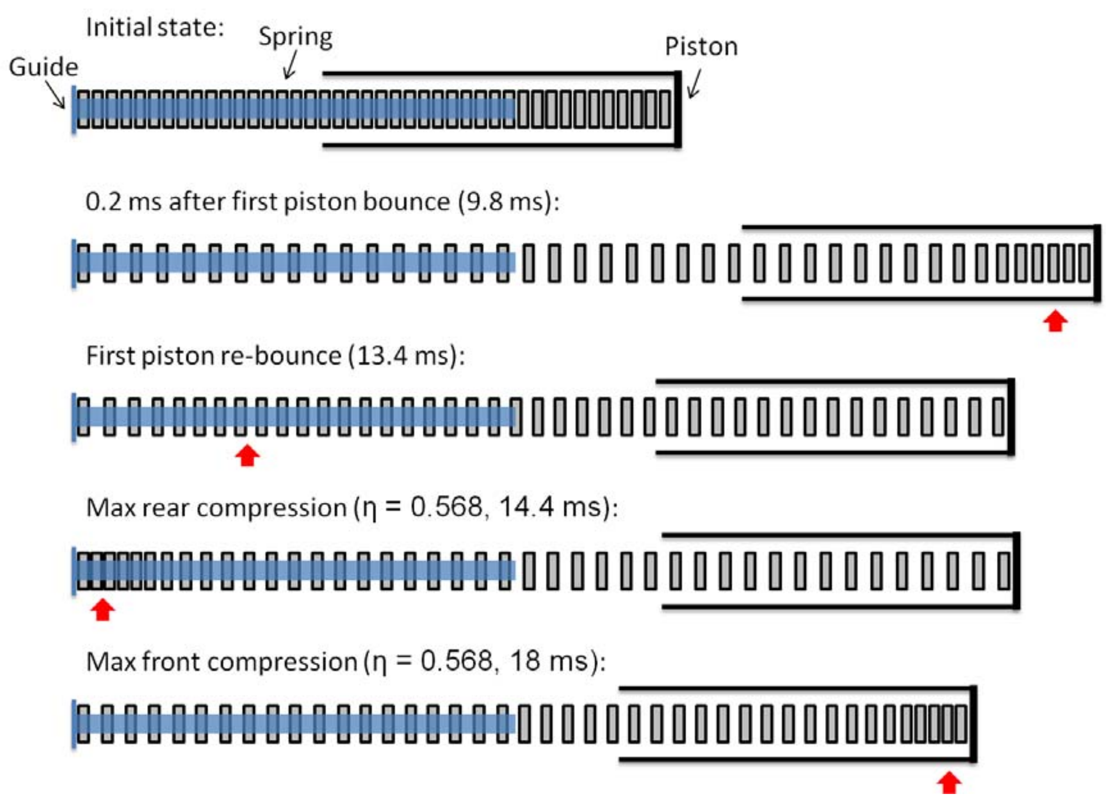
Figure 13 shows the sequence of these events for the case of a loose guide. The figure shows the guide (in blue) for illustrative purposes only, since a no-friction guide has no effect on what happens to the spring. If there is no guide friction, the spring energy is dissipated through the compression chamber and the pellet. This means that after the pellet leaves, any elastic energy left in the spring will be dissipated through entropy formation as a result of air flowing in and out of the compression chamber, and through friction between the piston and the chamber wall. For this reason, the piston may bounce backward and forward several times before coming to a stop.

In the absence of friction, firing first releases an expansion wave that travels backwards, roughly at the speed predicted by EQ 11. This wave reaches the end of the spring in approximately 3 ms. During the next 6.4 ms, the spring expands fairly uniformly, until the piston reaches almost the end of the chamber and bounces backwards[1]. By this time, the projectile has started moving down the barrel. The piston backward bounce triggers a

1. The piston may or may not hit the end of the chamber before bouncing, depending on the volume of air in the transfer channel and other parameters - see Ref. 1 for details.

compression wave that travels backwards down the spring coils - the second image from the top in Figure 13 is a snapshot of the spring two ms after the first bounce, the compression wave, indicated by the red arrow, is clearly visible as it starts its travel towards the rear of the spring. Approximately one ms before this compression wave reaches the rear coils of the spring, the piston bounces forward (first re-bounce). Shortly after that, the compression wave reaches the rear of the spring, and the rear-most coils clash together. The compression wave bounces forward and reaches the front of the spring while the piston is experiencing a second backward bounce. This causes also causes a clash of the forward-most coils, which reach the limiting compression ratio of 0.568. The last two segments to the right in the bottom image in Figure 13 corresponds to the right-most blue bar in Figure 12.

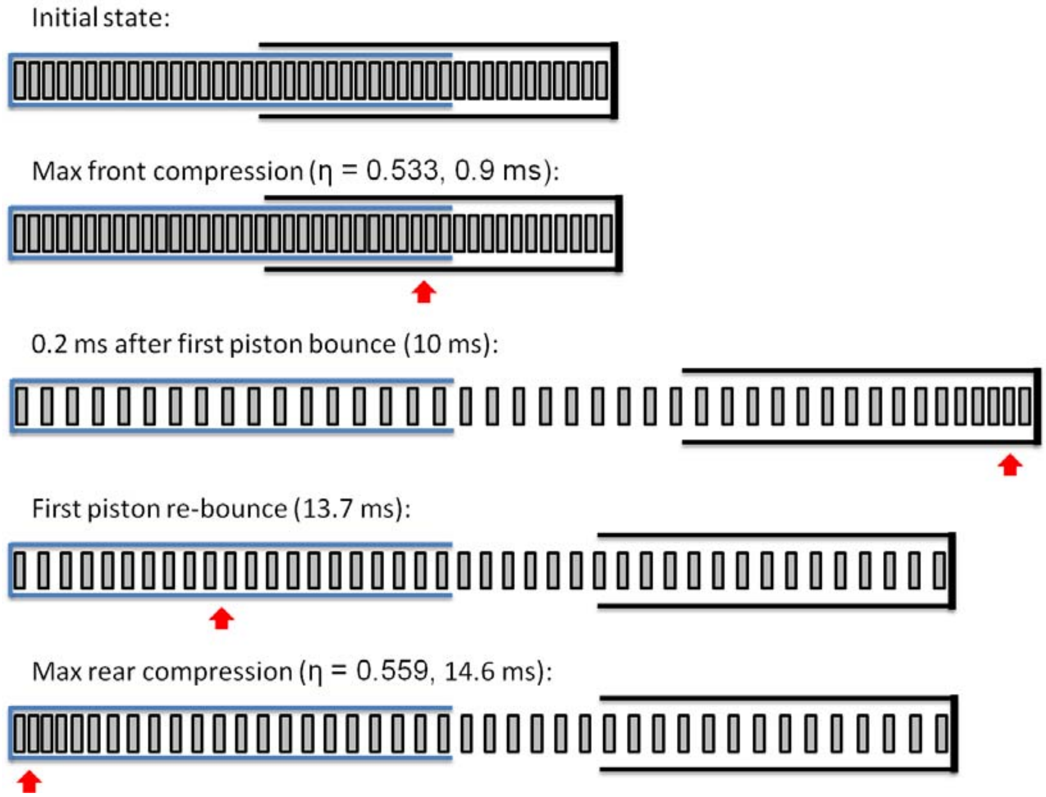
FIGURE 13. Sequence of four events during first 40 ms after trigger release - arrows point to compression wave (no guide friction).



How important are these compression peaks? We will discuss this in detail in the next section, but the answer is they are very important. If the guide is loose, the dynamic impact results in clashing, which implies a rise of about 6.3% in the spring stress above the cocked state. This is not all, however. The fatigue life of a spring depends not only on the maximum stress it experiences, but also on the difference between the maximum and minimum stress. The dynamic impact also lowers the minimum stress, causing a wider swing between highest and smallest stress values for the most affected coils.

A tight guide substantially changes this picture, as Figure 14 illustrates.

FIGURE 14. Sequence of four events during the first 40 ms after firing - arrows point to compression wave (100 mm long rear guide and 100 N friction force).



In the presence of friction forces, EQ 11 is no longer valid to predict the backward elastic wave velocity over the entire length of the spring. EQ 11 is, however, valid (with the caveats mentioned earlier) over the portion of the spring ahead of the rear guide. The portion of the spring that initially protrudes out of the rear guide is approximately 0.033 m. Upon trigger release, a dilation wave travels backwards, and reaches the mouth of the guide, causing the highest compression of the spring segment near the guide mouth. This maximum compression in the forward half of the spring happens 0.9 ms after trigger release. This time corresponds to a wave velocity of about 36 m/s, approximately in line with EQ 11. The maximum compression in this case is approximately 53%, which is close to the cocked state compression and significantly lower than in the loose-guide case.

After the spring reaches maximum compression near the mouth of the rear guide, the spring expands approximately uniformly for about 9 ms, until the piston experiences its first bounce. As was the case without friction, the backward bounce of the piston triggers an elastic compression wave that travels backwards. Also, as in the no-friction case, this wave continues to travel backwards after the piston re-bounces forward and reaches the rear of the spring after the piston experiences a second bounce backwards, causing a maximum compression of about 56% in the rear coils, after the pellet has left the barrel.

Notice that unlike the case of no-friction, in this situation there is no significant compression of the coils near the front end of the spring as a result of the re-bounce.

4.0 Fatigue and buzz implications

The standard procedure for designing a spring to withstand fatigue under fluctuating loads is a two-parameter procedure that uses the maximum amplitude of the fluctuating load, called the range load, and the mean or average load (Deutschman, Ref. 8). To design a spring for fatigue, you would exploit linearity between stress and strain and use the parameters η_{ave} and η_{range} in Table 2.

If you were to design a spring in the absence of dynamics effects, that is, a spring for a gun that gets repeatedly cocked and un-cocked but not fired, you would use the shear stress associated with $\eta_{ave} = 0.36$ as mean stress, and the shear stress associated with $\eta_{range} = 0.35$ as range stress. These values would apply to all coils in equal measure.

In the presence of dynamic effects the situation changes substantially. Peak stresses are now associated with larger compression ratios, and range stresses are enhanced by stronger de-compression, as Table 2 shows[1].

TABLE 2. Compression ratios for fatigue load design under three scenarios.

Fatigue parameters			
Comp Ratio	**No dynamics**	**Loose guide**	**Tight guide**
Maximum	**0.535**	**0.568**	**0.559**
Average (η_{ave})	**0.360**	**0.243**	**0.304**
Minimum	**0.185**	**-0.082**	**0.048**
Range (η_{range})	**0.350**	**0.650**	**0.511**

The standard framework to use the values in Table 2 for fatigue calculations is to associate the mean and range stresses to a fluctuating load configuration as illustrated in Figure 15.

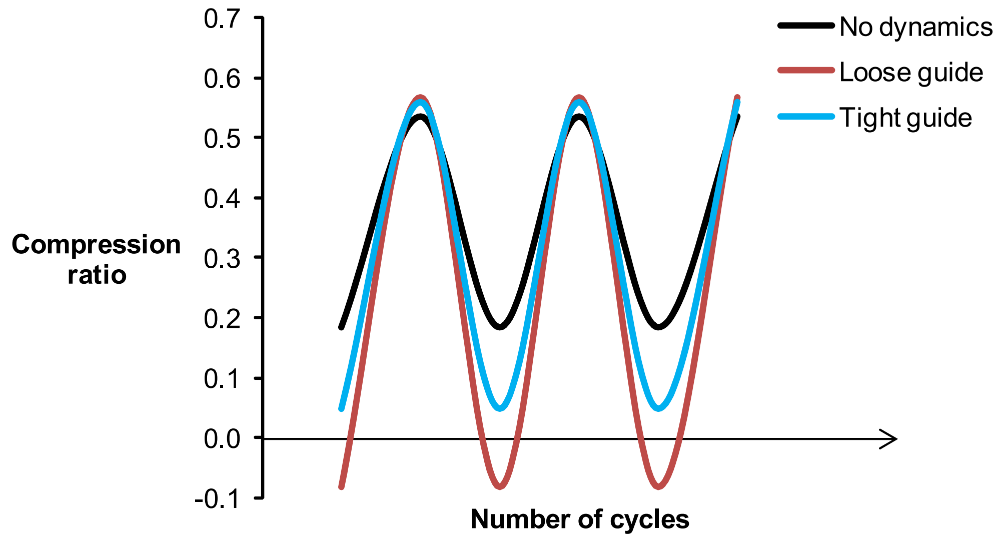
The black line in Figure 15 represents the fluctuating loads you would get by cocking and un-cocking the gun without firing it, the red line represents the strains in the absence of friction, and the blue line is the strain in the presence of a tight guide. With a loose guide, not only is the stress amplitude higher than in the case of a tight guide, but the effective number of cycles that expose the spring to fatigue is also higher, since during each firing cycle the spring experiences multiple episodes of compression and expansion.

In the case analyzed here, this means that the spring dynamics expand the range of stresses for fatigue design by over 85% in the absence of friction, and by about 45% in the presence of a tight guide with 100 N friction force.

1. The negative value in Table 2 refers to a coil in the front of the spring. You can also use the rear coil value -0.054, since, as a group, the rear coils are significantly more exposed to fatigue strain, as Figure 12 clearly shows.

This means that if the spring dynamics are not taken into account, the spring should be expected to fail much sooner than a quasi-static application of fatigue analysis would suggest.

FIGURE 15. Compression ratios for fluctuating loads for spring fatigue assessment (tight guide imparts 100 N initial friction force).



If the buzzing sound we hear when firing a gun with loose guides stems from the spring longitudinal vibrations, shown in Figure 8, then Figure 11 tells us that the use of a tight internal guide suppresses the buzzing sound by dampening oscillations that occur mostly *after* the projectile has left the barrel. This means the buzzing sound may be annoying, but it is not detrimental to accuracy.

There is an important corollary to the effect of friction. Since a tight guide lowers both the intensity of the fatigue stress acting on the spring, as well as the effective number of cycles the spring is subjected to (by mitigating the longitudinal oscillations), with a tight guide you may be able to use a more powerful and more highly-stressed spring without risking premature fatigue failure.

5.0 Conclusions

Using an elementary discretization model of the main spring of an airgun and solving the conservation equations for the spring, the air in the compression chamber, and the projectile, this research shows that in the absence of spring-guide friction the spring dynamics may lead to coil clashing near the rear end of the spring, which is consistent with the common observation that springs usually break at their rear end.

In the absence of friction, significant longitudinal spring oscillations take place which happen mostly after the projectile has left the gun, and are not detrimental to accuracy. The use of a tight guide, external or internal to the spring, mitigates the oscillations, resulting in lower dynamic stresses. A rear guide not only ameliorates oscillations in the rear part of the spring, but also in the front part where the spring makes no contact with the guide.

We can assume (pending empirical verification) that the mitigation of longitudinal vibrations associated with a tight rear guide is also the reason a tight guide eliminates spring buzz. Since the oscillations that are mitigated occur most after the projectile has left the barrel, the silencing effect of spring friction is not relevant to accuracy.

As a corollary to the effect of friction on spring fatigue life, you can also resort to a tight guide to allow for the use of a more powerful and more highly stressed spring.

6.0 References

1. D. Tavella, "Internal ballistics of spring-piston airguns", 2015, available at https://www.researchgate.net/publication 274638905_Internal_Ballistics_of_Spring_Piston_Airguns
2. E.E. Galloni and M. Kohen, "Influence of the mass of the spring on its static and dynamic effects", American Journal of Physics, **47**, 1076, 1979
3. A.E.M. Love, "The propagation of waves of elastic displacement along a helical wire", Trans. Cambl Philo. Soc., **18**, 364-374, 1899
4. W.H. Wittrick, "On elastic wave propagation in helical springs", Int. J. Mech. Sci. **8**, 25-47, 1966
5. S.V. Sorokin, "Linear dynamics of elastic helical springs: asymptotic analysis of wave propagation", Royal Society Proceedings, DOI: 10.1098/rspa.2008.0468, 2009
6. Society of Automotive Engineers, "Spring design manual", Second Edition AE-21, 1996
7. L.D. Landau & E.M. Lifshitz, *Theory of Elasticity*, Pergamon Press, 1970
8. A.D. Deutschman, W.J. Michaels, and C.E. Wilson, *Machine Design, Theory and Practice*, Macmillan Publishing Co., 1975