BARREL-WOUND MAINSPRINGS

Part 1: The shape of the torque-turns curve

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Summary

Part 1 of this three-part paper discusses the shape of the torque v turns curves of three barrel-wound clock mainsprings and compares the results with the torque v turns curve of an open (loop-ended) mainspring. The discussion is based on trials conducted by the writer, and concludes that, while the barrel hooking affects the near fully-wound shape of the torque v turns curves (torque 'kick-up'), altering the type of hooking does little to eliminate the kick-up torque during this phase.

However, the outer end hooking has a significant effect on mainspring life and can lead to fatigue damage and ultimate fracture, which is discussed in Part 2. The results lead to the conclusion that traditional peg-and-hole hooking is little better than a failure waiting to happen.

To conclude this trilogy, Part 3 discusses energy storage density which, compared to other energy storage devices, is not very good. It goes on to suggest a possible statistical-based methodology for a preliminary estimate of the size of a mainspring for new design clocks based on chapter ring diameter.

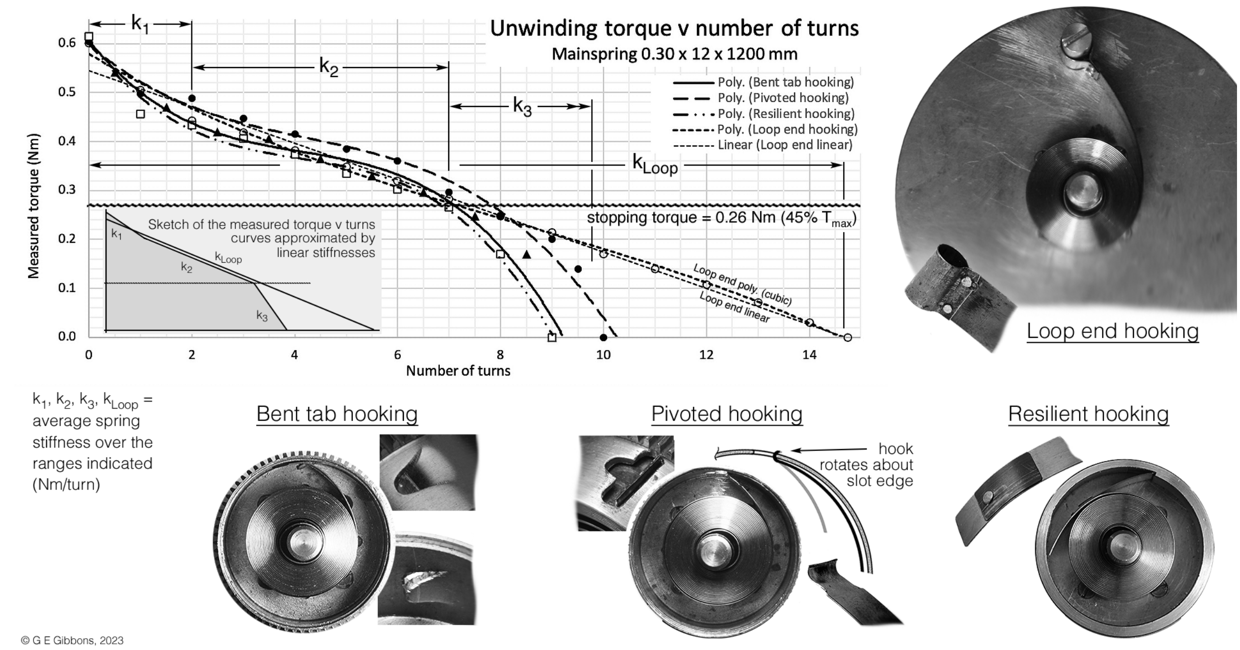

Figure 1: The trialled mainsprings, the pivoted hooking and resilient hooking of the writer's own design

Introduction

Not something that is seemingly discussed in the literature is an explanation of the difference between the shape of the torque-turns curves of a barrel-wound mainspring compared to the simpler open-wound mainspring. So to try to get a better understanding, I decided it was time to conduct some trials.

But before going any further, let me comment on my presentation of torque v turns curves, which are at variance with standard horological practise. Writing in a western language, the convention is that one reads from left to right be it text or image. Moreover, unless one is perhaps considering winding (for example, Figure 3 in Annexe A) or designing automatic winding gear), the primary interest is in the delivered (unwinding) torque, which suggests the x-axis (number of unwinding turns) should be read from left to right.

The trialled mainsprings

Four modern mainsprings[1] all of identical length, width and thickness made of the same alloy steel were trialled, each fully wound mainspring being photographed to scale in Figure 1. The measured torque-turns curves are plotted in the upper chart, and polynomial cubic equations fitted to all curves[2]. In addition a straight line (linear) equation was fitted to the open-wound (loop-end) mainspring. All springs use conventional peg hooking at the 7mm diameter arbor, the barrel diameter having a 34 mm internal diameter.

The torque-turns curves of the barrel-wound springs are more complex than that of the loop-end mainspring, and to a first approximation the shape of each of the barrel wound curves can be re-drawn as three straight lines over the k stiffness ranges k_1, k_2 and k_3, these ranges being annotated in the main chart in

Figure 1. Along with k_{Loop}, these straight line approximations are illustrated in the thumbnail sketch to the left in Figure 1.

The average gradient of each of these lines is effectively a measure of the spring's stiffness, k, this being defined as torque/number of turns (N), which is a variation of the more mathematical $k = T/\theta$ where T = torque and θ = the angle of rotation in radians (N being equal to $\theta/2\pi$).

Spring stiffness ranges k_{Loop}, k_1, k_2, k_3

k_{Loop} (approx. -0.035 Nm/turn) is the average unwinding stiffness over the entire unwinding cycle of the loop-end mainspring.

k_1 ($-0.09 > k_1 > -0.07$ Nm/turn) is the average unwinding stiffness of the barrel-wound springs over the first two unwinding turns. The springs have an apparent high stiffness and the average delivered torque is high. No up-stopwork was fitted in any of the trials.

k_2 ($-0.03 > k_2 > -0.02$ Nm/turn) is the average unwinding stiffness from turn 2 to turn 7 (5 full turns). Although a little more uniform, the lower spring stiffness is, perhaps not surprisingly, much the same as that from the open-wound spring, k_{Loop}.

k_3 ($-0.13 > k_3 > -0.10$ Nm/turn) is the average unwinding stiffness from turn 7 to approximately turn 9.5 (2.5 full turns to fully unwound). The spring has a high average stiffness.

Outer end hooking

Photographs of the trial mainsprings outer hooking are included in Figure 1:

- The outer loop-end of the open-wound spring is free to rotate around a post (typically a frame pillar in a real clock),
- The conventional 'peg and hole' barrel wall hooking is a bent tab as found in mass-produced movements from the second half of the 20th Century,
- The pivoted (rotating) barrel hooking is based on engineering practise for many spiral-wound springs, and is an attempt to emulate the freedom to rotate possessed by the open-wound loop-end mainspring. It is also more compact than the hooking of the other barrel-wound springs, which explains the greater number of unwinding turns (10.2 turns cf. 9.2 turns)[3],
- Based on an examination of two 1960's 'schoolboy' wrist watches, two types of resilient hooking of my own devising were trialled, but only the one giving the better concentricity of development during unwinding is presented[4].

Loop-end mainspring – stiffness k_{Loop}

Essentially the reference spring, the stiffness k_{Loop} at -0.035 Nm/turn is an almost perfectly linear straight line from fully wound to fully unwound. The literature also suggests that the loop-end spring should be a straight line, but detailed examination suggests it is not totally free from k_1 'kick-up' when nearing fully wound.

Barrel mainsprings – stiffness k_2

I shall return to stiffness k_1 in a moment, but during the k_2 phase the barrel-wound mainsprings are essentially free from the barrel wall for their full 1200 mm length and so behave in a similar manner during the k_2 phase to the open mainspring (k_2 and k_{Loop} have a similar stiffness and offer a similar deliverable torque). For good isochronism, the k_2 length between unwinding turns two and seven is effectively the useable length of the barrel-wound mainspring.

Barrel mainsprings – stiffness k_3

After about seven unwinding turns, the outer coils of the mainsprings start to 'pile-up' against the barrel wall preventing these turns from delivering any torque (if there can be no change in the radius of these coils, there can be no change in the bending strain and therefore no delivered torque). These piled-up coils essentially reduce the mainspring's length and, as every student of spiral springs knows, a reduced length increases the spring's stiffness. The average value of k_3 is in excess of -0.10 Nm/turn, this stiffness increasing as the springs continues to unwind and more turns pile-up against the barrel wall.

But there is another factor in play that is never mentioned in the literature: by the time the spring has unwound by seven turns, the delivered torque has dropped to a level that no longer drives the clock, this stopping torque[5] being indicated by the 0.26 Nm horizontal wavy line on the chart in Figure 1. Essentially, k_3 is unused; it is irrelevant, just as is the torque delivered by the open-wound spring after seven unwinding turns.

Barrel mainsprings – stiffness k_1

Perhaps the most difficult to fully comprehend, without any form of up-stopwork, stiffness region k_1 effectively brings the winding process to an abrupt stop. This is bad engineering practise, most engineering uses of 'clock springs' (called by engineers as 'spiral power' or 'motor springs') incorporating some sort of up-stopwork to relieve the spring of the high localised stresses resulting from the sudden stop.

In addition, in a going-barrel clock the winding torque exerted on the key by the person winding the mainspring is increasingly dominated not by the reaction torque from the nearing fully-wound mainspring but by the strength and vigour of the wrist of the person winding the clock as it nears fully wound – see Annexe A. This further exacerbates the abruptness of the stop (and hence mainspring loading) at full winding if no stopwork is fitted. This will be discussed further in Part 2 of this paper.

The recoiling click and slipping bridle

Not up-stopwork, the recoiling click invariably found in modern watches merely relieves the mainspring of the last fraction of a turn when full winding has been completed.

The commonly advanced justification is that it reduces the fully-wound torque that might otherwise cause overbanking of the escapement. This I don't doubt, but superficially, it would seem to be redundant if a slipping bridle with its far greater capability to reduce the k_1 kick-up torque by one or two turns is fitted – Figure 2.

Note: as presented the number of turns associated with the jagged fully-wound slip-stick portion on the chart in Figure 2 can only be the number of turns of the slipping bridle during the trial and not the unwinding of the mainspring. Moreover the peaks will be associated with the notches in the barrel wall that is a feature of many modern automatic watch mainsprings.

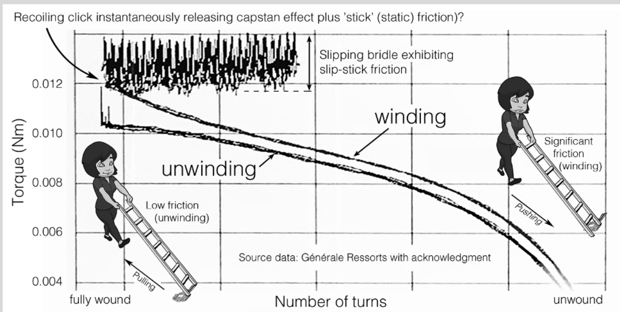

Figure 2: Showing the effect of deliverable k_1 torque by a recoiling click in reducing the fully-wound capstan effect friction and hence torque (top left) on relaxation of the winding button/key. The capstan effect is perhaps analogous to the direction-sensitive friction felt by Betty, which is significantly reduced when she moves the ladder by pulling (= unwinding) rather than by pushing (= winding)

So where does this added k_1 torque come from? As the mainspring coils start to wrap tightly together, the friction between the coils will increase exponentially. This is called the capstan effect[6]. Theoretically, this perhaps can be calculated, but in practise it will not be possible as there are too many unknown variables, the final fully-wound torque on key release also being dependent upon how much recoil (with or without a ‘recoiling click’ – see box) takes place before the ratchet click (pawl) engages with the ratchet wheel.

Once the coils start to become closely wrapped, the capstan effect will gradually decrease the effective length of the spring subjected to bending during winding. Ultimately this length will be little more than the free length between the outer coil and the barrel hook, during which time winding to ‘hard-up’ will generally result in reverse-bending[7] of this short length of spring.

During the trials, this reverse bending was visible in the last quarter of a turn to fully wound, the free length straightening out noticeably with very little rotation of the main body of the tightly-wrapped mainspring. To reverse-bend this short length of spring requires considerable torque supplied by the person winding the clock and, upon release, these bending strains will result in the k_1 torque kick-up.

There will also be similar short-length bending of the resilient hook, though how this manifests itself is more complex to determine.

The variability of k_1 with different hooking arrangements

The torque-turns curves in the k_1 region of Figure 1 are all different, but these differences may be rather illusory and should not deflect from the underlying principles just mentioned:

- During the trials, the writer recorded neither the recoil from fully wound before click engagement at full winding (but see Annexe A) nor the maximum torque applied to the winding key,
- At and near the fully-wound condition, the variability in the measured torque data points was high, which the writer ascribes at least partly to the ‘slip-stick’ nature of the inter-coil friction. A supplementary trial measuring the data points at one-quarter turn intervals during the first two unwinding turns did little to improve the variability of the measured data,
- The shape of the k_1 curves derived by the Microsoft *Excel* curve-fit analysis is to some extent affected by the number and quality of the measured data points as well as the chosen curve-fit (in some ways, a logarithmic curve fit was perhaps more appropriate for the k_1 kick-up).

So can one eliminate k_1 stiffness kick-up?

Based on the writer’s trials and thinking, the variability of frictional effects and changing effective (active) mainspring length leave the writer feeling that the k_1 kick-up will not only always occur (see also Annexe B) but cannot be affected significantly by the choice of barrel hooking. But it does suggest a benefit of the loop-ended open mainspring that is often derided by teaching establishments.

For the high-performance timekeeper with a barrel-wound mainspring, the only solution would appear to be to regard the k_1 region as unusable, which can only be achieved by using some form of up-stopwork. If implemented, this will either reduce the going time or require a larger barrel and longer mainspring.

This is what modern watch manufacturers do – they eliminate kick-up by incorporating a slipping bridle[8] into the barrel, this bridle limiting the torque that can be applied to the mainspring. In going-barrel clocks, a few manufacturers incorporate a Geneva mechanism[9] stopwork, but most just don’t bother, accepting that the clock will just have to get along with k_1 kick-up.

However, for other reasons, this is not to say that the design of barrel hooking is unimportant; get it wrong and the result may be premature mainspring failure as I shall suggest in Part 2 of this paper.

Summary – Part 1

At least superficially, this preliminary investigation suggests reasons for the shape of the barrel-wound mainspring torque-turns curves. With perhaps the exception of the loop-end mainspring, the useable torque is far from linear when nearing fully wound, but hopefully these preliminary trials and discussion will offer readers a starting point for further thought.

In Part 2, I shall explore the benefits and disadvantages of each of the trialled barrel hooking arrangements.

Annexe A – the human wrist

From the writer’s review of several research papers (eg. the use of screwdrivers, control knobs/handles, etc.), the maximum torque that can be exerted by a human wrist is of the order of 12 Nm, but in general the maximum convenient torque is about 2 Nm[10]. If one makes the crude approximation that the wings of the clock winding key are about 1.5 times the diameter of the mainspring barrel, the writer’s chart at Figure 3 can perhaps be considered as a guide to the typical maximum torque that an average person winding the clock may apply to the mainspring at full winding.

What it does suggest is that, for the trial springs (Figure 1) wound with a 50 mm winged key, a peak winding torque of three to four times the maximum mainspring torque is entirely possible, this figure being comparable with the maximum torque applied by the writer using a key with 50 mm wings during his trials (up to 2 Nm cf. the 0.60 Nm maximum measured unwinding torque).

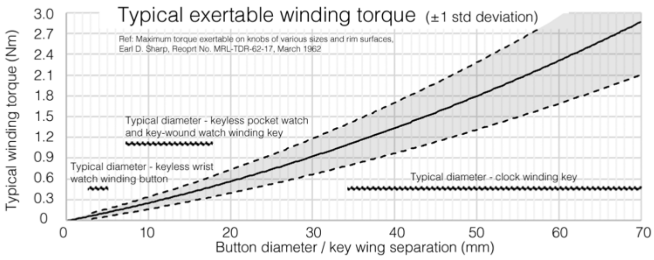

Figure 3: Typical maximum winding torque from the human finger and thumb

The chart is perhaps also applicable for knurled keyless winding crowns where there is no wrist action. Indeed, Figure 3 seems not entirely implausible even if one fails to make a correction for the gearing ratio between winding stem and mainspring.

Annexe B – capstan effect and lubrication

One short series of trials compared the effect of lubrication on an identical barrel-wound mainspring. Three trials were conducted:

- Lubricated with the manufacturer’s preservative oil ‘straight-out-of-the-box’,
- Unlubricated after degreasing (55°C for 2 hours) in an empty domestic dishwasher with a ‘Finish’ brand dishwasher tablet and no rinse aid[11]. On completion, a few patches of surface rusting were apparent,
- Lubricated with a good quality lathe oil (mineral oil).

The results are shown in Figure 4, and superficially very little difference in performance can observed. However, as the capstan effect takes hold in the k_1 stiffness region, the peak kick-up torque is in the region of 8% higher in the unlubricated (degreased) spring. Imparted during winding, this I ascribe to the higher inter-coil friction (the capstan 'holding power') from the capstan effect. It is probably for this reason that many modern watch mainsprings are PTFE ('Teflon') coated to minimise the inter-coil friction.

As an aside, these curves also perhaps suggest that modern, high-polish mainsprings heat treated in an inert atmosphere do not suffer as much from lack of lubrication over their primary working range (k_2) as those fitted in former years with their possible poorer surface finish.

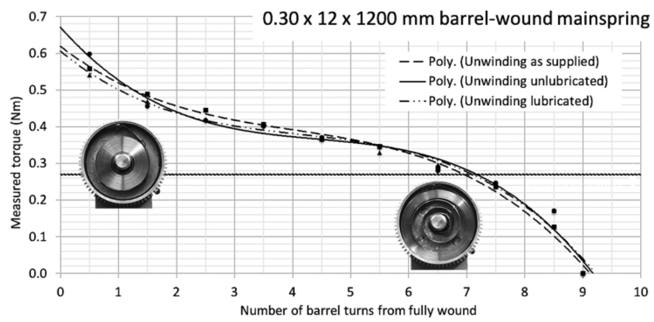

Figure 4: Unwinding torque-turns curves with three different lubrication regimes

Moreover, some authorities say that all traces of preservative oil should be removed before fitting and lubricating a new mainspring, though remaining somewhat silent as to the demonstrable benefits of so doing. As can be seen from Figure 4, my trials curves suggest very little difference in the performance of preservative oil and mineral oil.

And do I really believe that clock and watch manufacturers clean off all traces of the spring makers' preservative before fitting into a barrel? I don't think so.

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1. Mainsprings manufactured Ca. 2000 with an estimated yield strength approaching 2000 MPa. At 1200 mm, the barrel-wound mainspring is at it optimum barrel fill length.

2. Polynomials were fitted using the Microsoft *Excel* trendline function.

3. Slightly reducing its compactness, the trialled pivoted hook would need to be reinforced for a real clock to ensure it stayed hook-shaped for the spring's lifetime.

4. I should mention that the writer's design of resilient hook is very different from that used by modern Swiss watch manufacturers and, consequently, probably not good. For example, Eisenegger (Uhrentechnik, ISBN 978-3-7375-1337-1) suggests the schleppfeder (the riveted-on reverse spring segment) should be 1.5 to 2 times the mainspring thickness and have a wrap angle of 0.9 turns.

5. From the writer's experience of repairing clocks, this stopping torque tends to occur at about 45% of the fully-wound torque.

6. What engineers call the 'capstan effect', the frictional holding power being determined by the exponential equation $T_{load} = T_{hold}e^{\mu\theta}$ where μ = the inter-coil coefficient of friction and θ the wrap angle in radians.

7. Like the reverse-wound (pre-set) watch mainspring, the direction of the bend is largely irrelevant to the energy stored and hence the delivered torque.

8. The slipping bridle is not entirely different in principle from the obsolete Stackfreed mechanism for reducing excessive torque in that it is a way of limiting the applied torque by friction. In both, the frictional slip would seem to be dependent upon accurate design and manufacture, and lubrication unvarying with age.

9. While reducing the k_1 kick-up, the Geneva mechanism lacks resilience and will not necessarily prevent high local stresses reaching the barrel hook during vigorous winding. Indeed, experience of servicing clocks has shown that the Geneva mechanism driving peg will itself not infrequently fail (snap off) due to this abrupt stop.

10. The literature suggests that women have about two-thirds the strength of men, so the quoted figure is very much an overall average figure. But it perhaps explains some post-servicing call-outs from lady customers where the only fault is an inability to wind the clock for a full weekly run.

11. I understand one purpose of rinse aid is to leave a film of 'gleam' on washed dishes.
