Clock and Watch Lubrication

By considering the distribution of frictional losses in a timepiece, the horologist could determine where a high viscosity lubricant is appropriate, and where thinner lubricants should be used because of low power.

In order to find the forces exerted onto each pivot in the gear train of a watch, very sensitive measuring equipment would be needed, much more sensitive than what I have available to me. By creating a mathematical model of the forces in a watch, however, I could get an idea of what happens.

I will reverse the normal order of this project by presenting the conclusion first, in order not to deter those who are intimidated by the math. You would, however, benefit much more by reading the entire project.

Many watchmakers and clockmakers make the mistake of using the same thin lubricant for the first and second wheels as they do for the escapement. When lubricating a watch or clock, consider using the following rule of thumb:
1) use a heavy lubricant for high-torque, low-speed applications (mainspring, 1st and 2nd wheel pivots), and
2) use a light lubricant for low-torque, high-speed applications (3rd, 4th, escape wheel pivots, balance pivots, escape wheel teeth, clock strike governor pivots, etc.).

Also consider climate: Swiss watch oils are formulated in the Swiss Alps, and even most clock oils are almost as thin as water, at least in warm climates. While clock and watch oils have some thickness in colder climate conditions, they are very thin and less effective in warmer climates. We are not told to use different oils for timepieces intended for use in the cold Alps than for use in the hot desert.

Most if not all of my customers live in homes that today have climate control, so clocks in these homes do not get cold in the winter. Clocks that are not intended for use outside should be lubricated with thicker lubricants than those you might use for a clock that does not benefit from climate control. Watches used under normal conditions and worn on the wrist should not be lubricated with ultra-thin lubricants that you might use to lubricate a watch for use in extremely cold conditions, climbing high mountains or diving in cold waters. Timepieces for expeditions to the North and South Poles are usually not lubricated at all! When considering how thick a lubricant to use, you need to determine the thickness of that lubricant at the coldest temperature that the timepiece would be subjected to.

Why use a thicker lubricant if the timepiece were to be used in conditions that would not require a thinner lubricant? Thick lubricants have higher boiling points and they vaporize less easily. Thin lubricants vaporize easily: I have experienced problems with thin oils that dry up prematurely. I have seen several different clock oils dry up after only three years, and particularly one very expensive synthetic clock oil that dried up after only two years. A lubricant that fails after only two years not only fails to lubricate (reduce friction), but also fails to protect against oxidation. Thicker lubricants also have better cohesive and adhesive properties: in plain English, this means that a thicker lubricant stays in its place better when applied to a bushing without spreading or running off. If you consider the main purpose of lubricants, the benefit of a thicker lubricant becomes obvious: to reduce friction by providing a film between the two rubbing metal surfaces that keeps the metals apart. A good lubricant adheres to each metal surface to create this film and is not squeezed out under pressure. A good lubricant is cohesive so that it will exhibit good capillary action that will keep it in its place. A thicker lubricant has larger molecules (a longer carbon chain in the case of petroleum-based lubricants) and therefore provides a thicker film that keeps the metals further apart and resists higher pressures. As lubricant molecules slide over one another, some resistance to movement is caused, which we call 'drag' and which increases with the thickness of the lubricant. In low-torque applications, drag can impair the performance of the timepiece if the lubricant is too thick.

A couple of years ago, I lubricated a 16-size Elgin watch with a tower-clock oil, since I am not afraid to experiment with my own timepieces. It worked very well indeed and kept very good time. When I lowered the temperature by 40º Fahrenheit (in the refrigerator), this watch kept much better time than I was expecting (about thirty seconds slower per day), and there was a small decrease in the amplitude of oscillation of its balance wheel. At about the same time, I overhauled a Swiss watch of similar size and lubricated it with a very expensive synthetic watch oil: two years later, the watch hardly runs because the lubricant appears to have dried up.

Use a heavier lubricant for high-torque, low-speed applications. Clockmakers should consider using a heavy oil or grease to lubricate the mainspring, the pivots of the great wheel and second wheel. When applying a grease, you should apply a film over the entire area of the pivot and also over the entire area inside the bushing before assembly, because a thick lubricant will not spread by itself. Watchmakers should consider using a heavier lubricant to lubricate the mainspring, the pivots of the barrel and centershaft and third wheel. The watch's winding parts should be lubricated with a heavier lubricant. One problem frequently encountered in automatic watches is wear in the rotor arbor and its bushing: a heavier lubricant should be used here because the rotor is relatively heavy.

Use a thinner lubricant for low-torque, high-speed applications. This includes the pivots of the fourth and fifth wheels and of the pallet arbor in watches and clocks, and also the impulse faces of the pallets and escape teeth in clocks.

Use a light lubricant on the balance wheel pivots and the impulse faces of the pallets and escape teeth in low grade watches, such as a 7-jewel pocketwatch, where a small sacrifice in timekeeping could reasonably be made in order to use a lubricant of longer durability. Use an ultra-light lubricant in these areas in high-grade watches, such as a 23 jewel Railroad watch, where precision timekeeping is a necessity, but be aware that ultra-light lubricants have a tendency to dry up in the short term. Use ultra-light lubricants in watches that are subjected to extreme conditions of low temperature.

Take a moment to consider over-powered clocks with recoil escapements (which require plenty of excess power, as much as 50% more than a Graham Escapement), versus a low power clock with a more efficient Graham Escapement, such as a Vienna Regulator clock. An over-powered recoil escapement should be lubricated with a heavier lubricant, particularly because of the enormous power losses in recoil. A low-power clock should be lubricated with a lighter lubricant in the escapement.

Consider a typical watch, in this case, a 16-size, 17-jewel pocket watch. Assuming that torque is proportional to the normal force on the gear's pivot, where a normal force is the force pushing the pivot against the bushing or jewel, in the direction of the force that acts upon that gear's pinion: if the torque that acts upon the 2nd wheel pivots were taken as 100%, the torque that acts upon the 3rd wheel pivots would be smaller by the ratio of the number of teeth on the 2nd wheel pinion and the number of teeth on the 2nd wheel, or 10/80 x 100% = 12.5%. Using ratios, a chart (see end) of percentage torque values could be created to show the relative torque values that act upon the pivots of each gear in the train, and also the pallet arbor pivots and the balance staff pivots. Using ratios, another chart showing the revolutions per minute (RPM) that each gear would make could also be made.

Using typical pivot diameter sizes for a 16-size watch, the circumference of each pivot could be calculated, each of which could be multiplied by the respective RPM value for that gear to find the amount of sliding that each pivot makes inside each jewel in one hour of operation, sliding that I refer to as 'displacement' in the chart (see end), as if each pivot were traveling a certain distance. A figure for each relative frictional loss could be obtained by multiplying each value of torque with each value of displacement or sliding. Each relative frictional loss could then be expressed as a percentage to reveal where the frictional losses really take place.

To find the RPM value of the pallet arbor pivots, take the degrees of rotation per beat, assumed here to be 18° (a high value), divide by 360° and multiply by 300 beats per hour to get 15 revolutions per minute. To find the RPM value of the balance pivots, multiply 300 beats per hour by the amplitude of oscillation of the balance wheel, assumed here to be 1.5 turns, to get 450 RPM.

If one third the torque of the escape wheel reaches the balance wheel, multiply the torque value for the torque that acts upon the pallets by 0.33.

Finding the frictional loss during impulse is more complicated, since this requires us to find the circumference of the escape wheel that is in contact with the pallet during impulse, neglecting the power lost during drop. Take the diameter of the escape wheel and multiply its value by pi to find its circumference. Find the circumference per beat by multiplying this by the degrees during impulse (10°) and divide by 360°. This value must be divided by the cosine of the impulse angle, which I have assumed to be 45°, to find the length of impulse, or 'distance traveled' during each impulse. Multiply this by 300 beats per minute to find the total length of impulse in one minute (the 'displacement per minute'), which you multiply by the torque acting upon the pallet to find a friction value.

The same method is used to calculate a value for the frictional loss in the fork horn and the roller jewel. The frictional loss is determined by the amount of sliding that takes place, which is assumed to be equal to the amount of depthing. I have assumed the depthing to be 0.07 mm. in this example.

The same method is used to calculate a value for the frictional loss during draw. The relative value of this loss is very small. The loss of efficiency caused by draw is mainly caused by angle, which we do not consider here, and not by friction. The loss by friction is relatively small and less than expected because it does not consider the total loss caused by draw.

The friction % values in the chart reveal nothing by themselves, but their relative values can be compared. The frictional losses in the pivots of the gear train are very small. The frictional losses in the pivots of the balance wheel are considerably more. Most significantly, the frictional losses in the pallets during impulse are very large because of two factors:

1) the 'displacement' during impulse is 85% more than the 'displacement' during the rotation of the balance wheel.

2) the torque during impulse is 2.75 times that which is exerted onto the balance wheel.

The results might tempt the watchmaker to lubricate the balance jewels and the escape wheel teeth only, leaving all other jeweled bearings dry (not a good idea!).

The values of torque show how much torque is exerted on the pivots of the first three gears in the train. It would make sense to consider lubricating these pivots with a heavy lubricant or even a grease as these high-torque, low-speed bearings would not suffer the effects of a heavy lubricant. The remaining pivots and friction surfaces are very low torque (less than 2% each). It would make sense to consider lubricating these pivots with a light lubricant as these low-torque, high-speed bearings would certainly suffer the effects of a heavy lubricant, which would cause drag.

The most important point to consider is that more power is lost in the transfer of power from the escape wheel to the pallets in a Swiss Lever type escapement. There are two power losses:

1) Power losses caused by angle. When the direction of the force exerted by the escape wheel onto the pallet is different to the direction of movement of the pallet (i.e. the direction in which the pallet receives the power), there is a loss of efficiency: in a correctly designed Lever Escapement the angle between the directions of the forces is 90° and the maximum achievable efficiency is only 50%. This power loss considers only angle and nothing else.

2) Power losses caused by friction. When the power is transferred in a 'rolling' action, as happens when the escape tooth rotates and provides an impulse to the impulse pallet of the balance wheel in a Chronometer Escapement, the transferor (escape tooth) and the transferee (impulse pallet of balance) appear to roll together, which results in an almost frictionless transfer of power. In the Lever Escapement, however, the escape tooth slides across the pallet's impulse face, causing a frictional loss that is determined by the magnitude of the impulse, the coefficient of friction of the two sliding surfaces and the displacement (or the amount of sliding that takes place during impulse).

By the way, note that in the Chronometer Escapement the direction of the forces is the same at the mid-point of the impulse, which means that this design experiences almost no power loss as a result of angle, in addition to almost no power loss as a result of friction (because there is almost no sliding taking place during impulse). The escape teeth of the Chronometer Escapement should not be lubricated.

The escape teeth of the Lever Escapement should be lubricated most carefully. Each tooth should be lubricated with a minute amount of lubricant because it should not be assumed that the lubricant would spread evenly over all the teeth otherwise. Since a light lubricant must be used, the watchmaker must be very careful not to lubricate in excess, as this might cause the lubricant to run, or be drawn away from the intended area.

As in any simplified hypothetical scenario, assumptions are made to simplify the problem and to overcome otherwise insurmountable or even unquantifiable problems.

This scenario does not consider the efficiency loss caused by the angle of the pallet impulse face because the loss caused by angle is not reduced with lubrication. The only efficiency loss that is reduced with lubrication is friction, and it is therefore only the frictional losses that are considered here.

It is assumed that the coefficient of friction is the same for all the friction surfaces, that all the pivots are in the same condition and that all the jewel surfaces are in the same condition (with no variations).

It is assumed that there is no frictional loss when the pivot shoulder rubs against the jewel, as if each jewel in the train were capped.

It is assumed that the masses of the gears, pallets and balance wheel have no impact upon friction. While the masses of the first four gears are of negligible influence upon the results, the masses of the escape wheel, the pallets and the balance wheel are significant. Their influence on the results is, however, ignored because their relative effects are unquantifiable. Of particularly significant impact is the mass of the balance wheel, which in a 16-size watch is relatively large, increasing the friction considerably as the balance pivots rotate in the jewels. In the hypothetical scenario, it is assumed that the balance wheel's torque percentage is equal to the impulse received by the roller jewel, which is probably an underestimation for a watch with a heavy balance wheel. Despite this problem, the results are still useful because its percentage of the total frictional loss in the watch would not reach the level of friction in the pallet impulse faces unless it were increased by six times. Increasing the value of friction of the balance wheel pivots would decrease the relative friction percentage values of the other friction surfaces but it does not change the conclusion (instead, it serves to reinforce the conclusion):

The two areas where the most power is lost in a watch are the balance pivots (in their jewels) and the pallet impulse surfaces and the escape teeth during impulse.

I believe that much more power is lost during impulse because of friction (in addition to losses caused by angle) than anywhere else in the watch. The watchmaker should pay most attention here.

I am including torque calculation charts for two very common clocks so that clockmakers could see the relationships between the torque and RPM values for a typical American clock (a Seth Thomas 89) and a typical German chiming clock with a floating-balance (a Hermle 340-020). These differ from the watch chart in order to show how the friction is distributed in the gear train itself without considering the escapement: this information would be more useful to clockmakers. Notice how similar the friction percentage values are for the Seth Thomas clock. In the Hermle, the torque values have been adjusted to account for the stronger chime mainspring. These charts have two RPM columns to show that in the left column calculations were made up the column, and that in the right column calculations were made down the column. Observe how little torque the escape wheel receives: in the Seth Thomas, 289 times more torque acts upon the second wheel's pivots than upon the escape wheel's pivots. In the Hermle, 1650 times more torque acts upon the second wheel's pivots than upon the escape wheel's pivots.

Since thicker lubricants cause more drag, consider the bushings: a pivot turning in a longer bushing will be more affected by lubricant drag than one turning in a shorter bushing because of a larger area of lubricant film. However, using a longer bushing reduces the pressure on the bushing and the pivot because pressure is force divided by area. Most American clocks have very long pivots that protrude well beyond the bushing: in high-torque, low-speed applications, you might consider installing longer bushings that protrude beyond the plates, since longer bushings would be more durable, if the clock being repaired were not a high-grade work of engineering art and repair-as-art would not be called for. A mass-produced clock of lower grade, such as the Seth Thomas 89, would be well served by 3 mm bushings in the second wheel bushings. The bushings must not be longer than the pivots, however, so this could not be done to the Hermle clock's second wheels, for example.

Last, but not least, consider the difference in the rate of movement of the three trains of the Hermle movement. Most of the time, the clock neither chimes or strikes, but when it does, the gears move much faster. Looking at the chart below, you could see the effect this has on friction in the chime and strike trains. While looking at this chart, you will also see that there is more friction in the 3rd wheel pivots of the chime and time trains than in the 2nd wheel pivots: you would expect the reverse since more power acts upon the 2nd wheel pivots and since more wear takes place in the 2nd wheel bushings. If less friction takes place in the 2nd wheel bushings, you would expect less wear. This suggests that the increase in wear takes place not because of friction but because of something else: the yield pressure of the metal is exceeded, causing premature failure which we see as small pits in the pivot. The pressure on the pivot can be decreased by:

1) using a thinner mainspring (which we do not want to do in this case),
2) increasing the diameter of the pivot (the new Hermle 2nd wheels have slightly larger pivots),
3) replacing the pivots with a different metal that has a high yield pressure,
4) lubricating these pivots with a lubricant that has a very high yield pressure (that is, a very thick lubricant, a grease), keeping the metals apart so that this failure does not occur.

No lubricant lasts forever, so lubricating these pivots with a heavy lubricant would one day end in failure unless the clock were maintained before the lubricant fails. Since almost everybody uses their clocks until they will not run at all before they bring them in for repair, clockmakers should consider using a grease on the 2nd wheel pivots that has graphite: the magic of graphite is that it continues to work after the lubricant has failed! Either buy a lubricant with graphite added, or buy the graphite powder and mix some into the lubricant before applying it to the pivots and bushings. Many clockmakers do not like graphite because they think it is messy and makes the clock look ugly. The choice is yours, but I must say that I do not care much for a beautiful clock that does not work.

I have very carefully avoided recommending any particular lubricant or any particular brand of lubricant, as this is a very volatile issue. I have, however, had very unsatisfactory results with synthetic lubricants, and will never use them on my clocks or watches.

I would like to thank Dan Henderson for his suggestions. Dan is a mechanical engineer at 3M. He collects and repairs mechanical clocks and watches (and does not use synthetic lubricants). Many areas in this website would not have been possible without his suggestions!

A lot of work went into creating the charts below. I hope they are useful to you.

16 SIZE WATCH							53.44	***100***		
				PIV SIZE	PIVOT	DISPLACEMENT	FRICTION	***FRICTION***	***RELATIVE***	***RELATIVE***
	NO. TEETH	***TORQUE %***	RPM	MM	CIRCUM	PER HR: X	:=FX	***%***	***TORQUE***	***SPEED***
BARREL	80									
2ND PINION	10	***100.00***	0.02	0.50	1.57	0.03	2.62	***5***	***HIGH 2***	***LOW***
2ND WHL	80									
3RD PIN	10	***12.50***	0.13	0.32	1.01	0.13	1.68	***3***	***HIGH 3***	***LOW***
3RD WHL	75									
4TH PIN	10	***1.67***	1.00	0.28	0.88	0.88	1.47	***3***	***LOW 4***	***HIGH***
4TH WHL	70									
5TH PIN	7	***0.24***	10.00	0.22	0.69	6.91	1.65	***3***	***LOW 5***	***HIGH***
5TH WHL	15									
PALLETS	18 DEG/BEAT	***0.11***	15.00	0.22	0.69	10.37	1.15	***2***	***LOW***	***HIGH***
BALANCE	33%	***0.04***	450.00	0.12	0.38	169.65	6.28	***12***	***LOW***	***HIGH***
	1.5 TURNS									
DIAMETER										
ESC WHL	8.5	MM								
CIRCUM	26.70	MM								
ANGLE PER										
BEAT	10	DEGREES DURING IMPULSE								
CIRCUM										
PER BEAT	0.74	MM								
LENGTH OF				X	F		FX			
IMPULSE	1.05	MM	x300	314.70	0.11		34.97	***65***		
DEPTHING			BPM							
OF ROLLER										
JEWEL	0.07	MM APPROX	300	21.00	0.04		0.78	***1***		
ANGLE IN										
DRAW	2	DEGREES								
RADIUS OF										
PALLET CIR	2.45	MM								
CIRC OF										
PALLET CIR	15.42	MM								
LENGTH OF										
DRAW	0.09	MM	300	25.70	0.11		2.86	5		

1983 HERMLE 340-020								952.16	100		
	NUMBER				PIVOT	PIVOT	PIVOT	TOTAL	%		
CHIMES	OF TEETH	RPM	RPM	TORQUE	DIAMETER	CIRCUM	DISPLACEMENT	FRICTION	FRICTION	TORQUE	SPEED
2ND WHL	60	0.20		144.08	1.8	5.65	1.13	234.78	25	HIGH	LOW
3RD WHL PINION	10										
3RD WHL	80	1.20	1.20	24.01	2.5	7.85	9.42	326.08	34	HIGH	LOW
4TH WHL PINION	8										
4TH WHL	72		12.00	2.40	1.4	4.40	52.78	182.61	19	LOW	HIGH
5TH WHL PINION	8										
5TH WHL	93		108.00	0.27	0.8	2.51	271.43	104.35	11	LOW	HIGH
GOVERNOR PINION	7		1434.86	0.02	0.8	2.51	3606.19	104.35	11	LOW	HIGH
								6.14	100		
	NUMBER				PIVOT	PIVOT	PIVOT	TOTAL	%		
TIME	OF TEETH	RPM	RPM	TORQUE	DIAMETER	CIRCUM	DISPLACEMENT	FRICTION	FRICTION	TORQUE	SPEED
2ND WHL	55	0.00		100.00	1.8	5.65	0.02	1.71	28	HIGH	LOW
3RD WHL PINION	10										
3RD WHL	64	0.02	0.02	18.18	2.25	7.07	0.12	2.14	35	HIGH	LOW
4TH WHL PINION	8										
4TH WHL	48		0.13	2.27	0.8	2.51	0.34	0.76	12	LOW	HIGH
5TH WHL PINION	8										
5TH WHL	50		0.80	0.38	0.8	2.51	2.01	0.76	12	LOW	HIGH
ESCAPE WHL PINION	8										
ESCAPE WHL	15		5.00	0.06	0.8	2.51	12.57	0.76	12	LOW	HIGH
								715.89	100		
	NUMBER				PIVOT	PIVOT	PIVOT	TOTAL	%		
STRIKE	OF TEETH	RPM	RPM	TORQUE	DIAMETER	CIRCUM	DISPLACEMENT	FRICTION	FRICTION	TORQUE	SPEED
2ND WHL	52	0.58		100.00	1.8	5.65	3.26	326.24	46	HIGH	LOW
3RD WHL PINION	9										
3RD WHL	63	3.33		17.31	1	3.14	10.47	181.23	25	HIGH	LOW
4TH WHL PINION	7										
4TH WHL	63	30.00	30.00	1.92	1.15	3.61	108.38	208.42	29	LOW	HIGH
5TH WHL PINION	7										
5TH WHL	63		270.00	0.21	0.8	2.51	678.58	144.99	20	LOW	HIGH
GOVERNOR PINION	7		2430.00	0.02	0.8	2.51	6107.26	144.99	20	LOW	HIGH

SETH THOMAS 89								19.16	100		
	NUMBER				PIVOT	PIVOT	PIVOT	TOTAL	%		
	OF TEETH	RPM	RPM	TORQUE %	DIAMETER	CIRCUM	DISPLACEMENT	FRICTION	FRICTION	TORQUE	SPEED
2ND WHL	60	0.01	0.01	100	1.6	5.03	0.05	5.03	26	HIGH	LOW
CENTERSHAFT	36	0.02									
3RD WHL PINION	8										
3RD WHL	42		0.08	13.33	1.5	4.71	0.35	4.71	25	HIGH	LOW
4TH WHL PINION	7										
4TH WHL	45		0.45	2.22	1.5	4.71	2.12	4.71	25	LOW	HIGH
ESCAPE WHL PINION	7										
ESCAPE WHL	39		2.89	0.35	1.5	4.71	13.63	4.71	25	LOW	HIGH
								4422.40	100		
STRIKE	NUMBER				PIVOT	PIVOT	PIVOT	TOTAL	%		
2ND WHL PINION	OF TEETH	RPM	RPM	TORQUE %	DIAMETER	CIRCUM	DISPLACEMENT	FRICTION	FRICTION	TORQUE	SPEED
2ND WHL	52	2.31		100	1.6	5.03	11.60	1159.97	26	HIGH	LOW
3RD WHL PINION	8										
3RD WHL	60	15.00	15.00	15.38	1.5	4.71	70.69	1087.47	25	HIGH	LOW
4TH WHL PINION	6										
4TH WHL	70		150.00	1.54	1.5	4.71	706.86	1087.47	25	LOW	HIGH
GOVERNOR PINION	7		1500.00	0.15	1.5	4.71	7068.58	1087.47	25	LOW	HIGH
striking once every two seconds.											

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