CHAPTER 5

Stability and Wear in Intermittent Motion Mechanisms

Types of Wear

The service life of any machine is determined by its stability in use. As long as the shapes of the various parts of the machine are not altered too much by wear or deformation, the principal parts of the machine do not break, and the various loads experienced by the machine (including the friction load), as well as the input power to the machine remain stable, the machine will continue to function as designed. Unfortunately, all operating machines experience gradual changes in all of these factors. The geometry of parts changes as they wear. Loads frequently increase as wear particles accumulate and increase frictional resistance to motion. Drivers (solenoids, motors, etc.) decay as they age magnetically, as brushes and bearings wear, as insulation evaporates, etc. The designer cannot prevent wear and the ultimate "death" of his machine; but by understanding the various types of wear, by being able to predict where it will occur, and by designing the various parts of his machine so the whole will continue to function correctly even when worn, he can extend the life of his machine to some degree.

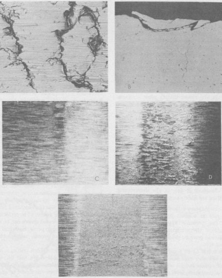
Intermittent motion mechanisms wear just as everything else, and are perhaps, as a class, more affected by wear than other types of machine devices since they are exercises-in-instability to begin with.

Experts differ somewhat in defining and classifying types of wear. But, generally speaking, four types are identified:

1. Plastic flow or ploughing, where the surfaces of one or both of the bodies in contact (sliding contact) deform, but particles are not immediately removed.
2. Adhesion or transfer wear, where particles from one body in sliding contact with another, adhere to the other body and are wrenched free from the first. Sometimes, such particles remain on the second body, or they can fall free as wear particles. At times they later transfer back to the first body.
3. Surface fatigue caused by cyclic stresses produced in surfaces as bodies periodically impact or roll across each other.
4. Abrasive wear or fretting, a grinding action in which wear particles resulting from other types of wear act as abrasives to permit one mating part to grind away the other.

These are the four mechanical types of wear. There are also chemical and corrosive types which are factors in the ultimate death of many machines, but these are not commonly cured by changes in the configuration of the design and thus are not pertinent to this discussion.

The wear in a given design can usually be classified as adhesive, abrasive, etc., only by microscopic examination. And then only by people who know what they are doing. Figure 5–1 shows some microphotographs of different types of wear. Figures 5-1A and 5–1B show surface cracking. The first photo-



Photographs courtesy of Massachusetts Materials Research Inc.

Fig. 5–1. Microphotographs [illegible] worn metallic surfaces; (A) surface cracking; (B) surface cracking; (C) [illegible]ar; (D) adhesive wear; (E) [illegible]

graph is a plastic replica of the worn surface; incidentally, an interesting new process in which a sheet of softened acetate is used to record the features of a surface. Figure 5–1C shows abrasive wear; Figs. 5–1D and 5–1E show adhesive wear.

Wear [illegible]

In general, the rate at which parts wear will be reduced if the parts are

1. Made harder by chemical or heat treatment
2. Operated at lower stress levels
3. Operated at lower speeds
4. Operated at lower temperatures
5. [illegible]
6. Are properly lubricated

Watch out for generalities of this kind when dealing with wear, however. There are many exceptions to the rule. For example, my company discovered years ago that interrupted gears made of plastic would outlast steel or brass gears in a given counter design. The plastic was much "softer" than the metal, yet gave much longer service in the same design. There are probably several reasons for this. First: the plastic must distort under impact loads and so act to dampen shocks. Second: the impacting parts have lower mass, thus reducing impulse. Both factors reduce peak stresses. Third: as the plastic distorts it brings larger surface areas into contact at load points, reducing contact pressures. The result: the softer part lasts longer than the hard! But hard steel usually wears better than soft steel.

Wear experts also report that in some instances an increase in sliding velocity or contact temperature can actually reduce wear rates; in other cases, increasing these factors can drastically increase wear. And some materials will wear less rapidly when unlubricated.

One thing, however, is certain, that the wear rates of machine parts will be hard to predict. Small variations in such things as sliding velocity, operating temperature, alloy content, surface finish, lubrication, etc., can have a drastic, though mysterious, effect on the life and performance of machine parts. One reason for this is that wear rate is usually not a linear function of any of the determining parameters. Here are some examples:

1. Experiments show little variation in adhesive wear rate as contact pressure is increased until the pressure approaches the yield strength of the material. Then very small increases in pressure produce drastic changes in wear rate.
2. There seems to be some stress limit below which adhesive wear *never* occurs. This is not true of surface fatigue which will eventually occur at any stress level.
3. Surface fatigue, however, usually produces no visible wear particles, or the like, for many thousands or even millions of operating cycles. Then the surfaces will start to deteriorate very rapidly. When stress levels are low, however, this delay in the onset of visible wear can result in what amounts to zero wear for the life of the product. (See Fig. 5-2.)
4. Some engineers feel that surface-fatigue wear rates are proportional to the ninth power of applied stress. This means that if the load at a given wear point doubles, the fatigue wear rate will increase by a factor of 512! Or, more realistically, a ten percent increase in stress level can more than double fatigue wear rate.

One result of all of this is that it is virtually impossible to determine the rate at which a given machine will wear except by actual tests of that machine. Bench-testing sample materials can show RELATIVE wear rates of different material combinations, but cannot yield information on the actual wear rates of any machine using those combinations, unless the bench test duplicates exactly the complete operating environment of the ultimate machine: stress levels, contact geometry, temperature, lubrication, contaminants, etc.

For the same reason, wear tests cannot be accelerated. Remember that wear rates are not linear functions of stress, temperature, or sliding velocity. Accelerating a machine, running it faster to speed up wear tests, will usually increase stress levels drastically. Parts that might never wear in normal operation may now go to pieces very rapidly.

There is another factor that becomes important when accelerated tests are made on intermittent motion devices. In the section on the problems of control, in Chapter 4, it was seen that high-speed stop-and-go mechanisms almost inherently have control problems. As operating speeds increase, the control problems also increase, until finally, the machine becomes too "rubbery" to perform as intended. Wear modes at elevated speeds can be very different from those at design speeds. So actual designs must be tested at actual speeds, under actual conditions to get valid results.

A word of advice: When testing a new design to see what its probable life will be, test as many models as is economically feasible. Performance can vary drastically from one model to another because of the factors already discussed, even though the models appear to be geometrically identical. This nonuniformity of result is a very common problem with intermittent motion mechanisms and to overlook this factor can prove to be costly.

Wear in Intermittent Motion Mechanisms

The potential wear points in intermittent motion mechanisms can usually be determined by inspection if the concept of degrees of freedom is kept in mind. Any machine body can translate in three directions, or rotate in three directions (or a combination of

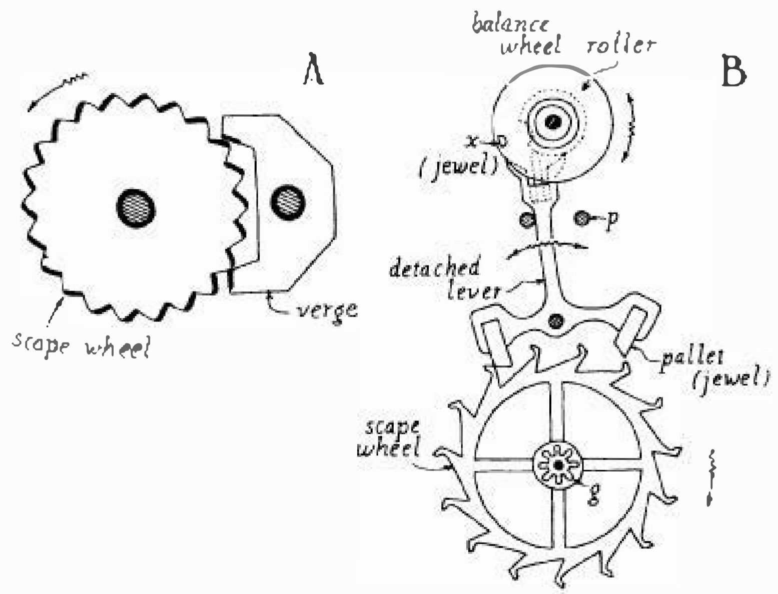


Fig. 5-2. Typical wear points in two timer escapements. Wear points in the "runaway" escapement (*Left*), are indicated by heavy lines. This particular escapement is an "untuned" escapement (see Chapter 11), and is subject to high impact stress. By contrast, the "tuned" escapement (*Right*), will run more than a lifetime if kept clean and oiled. Here the principal impacts occur between jewels (ruby or sapphire) and highly polished or burnished hardened steel. Some tuned chronometer escapements have been operating for over 100 years with no signs of wear.

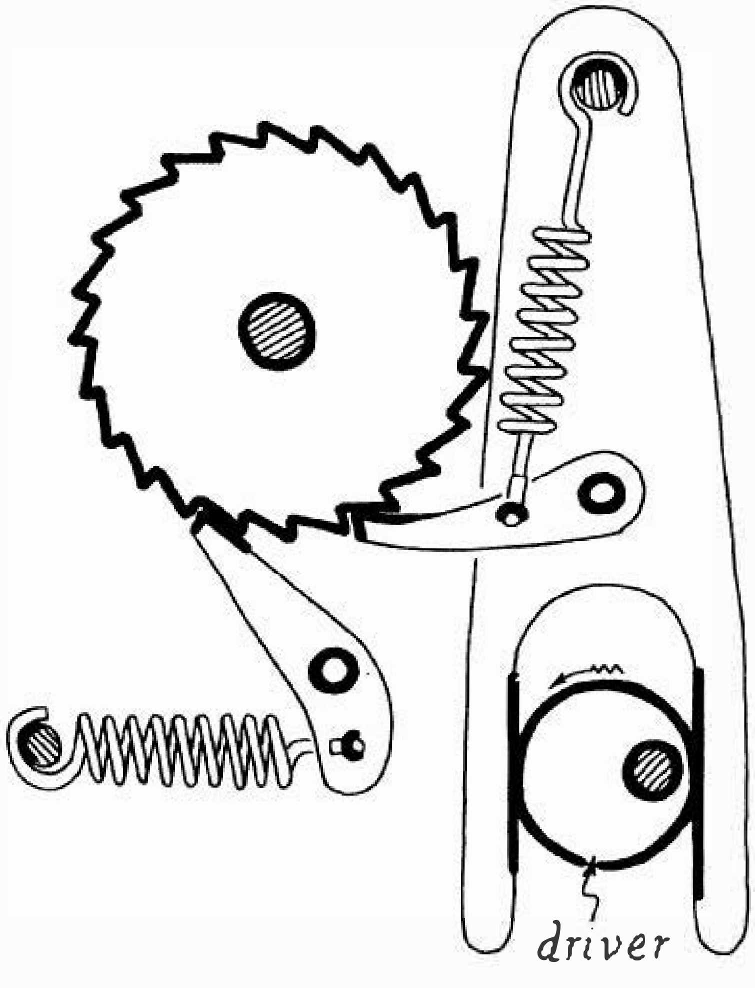


Fig. 5-3. Typical wear points in a cam-driven ratchet driven mechanism. Again, wear points are shown by heavy lines.

these six), unless constrained by shafts, bearings, slides, etc. (See Figs. 1-14 through 1-16.) Wear points always occur at the points at which the body contacts such constraints, as well as at the points where the input forces are applied to the body, and the points at which the body exerts force on its load. Typical wear surfaces for three intermittent mechanisms are shown as heavy black lines in Figs. 5-2 through 5-4. In contrast with the untuned mechanism shown in Fig. 5-2A, Fig. 5-2B shows a "Tuned

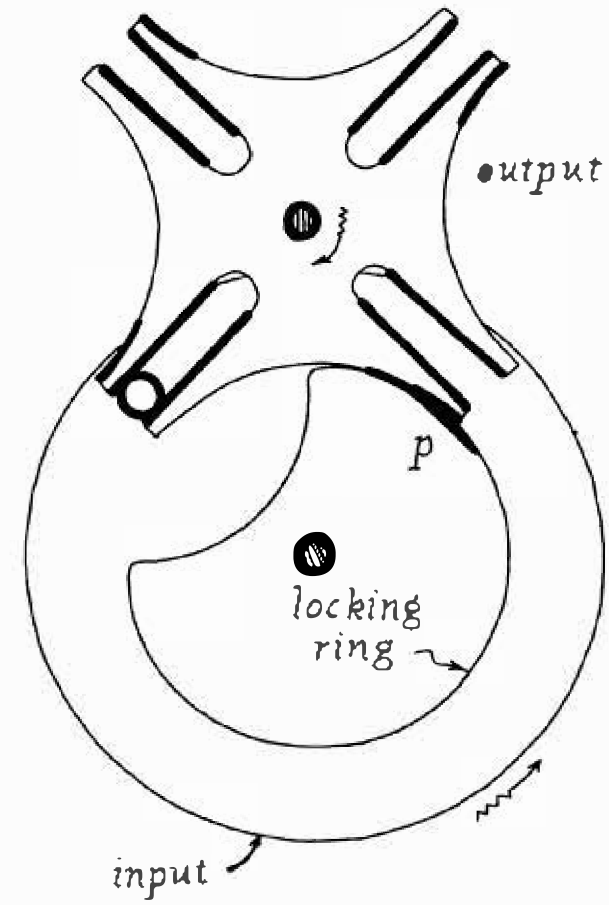


Fig. 5-4. Heavy black lines indicate the typical wear points in a Geneva mechanism.

Escapement" which, if kept cleaned and well oiled, will have a long life expectancy.

Intermittent motion devices are also subjected to high stress levels because of the forces generated during impact or during sudden changes in velocity. These forces, however, are usually of very short duration and scientists have found that materials can withstand high forces more readily for short periods of time than they can for long periods. This is not to say that intermittent motion devices can tolerate high stress levels indefinitely without wearing, just because such stresses are of short duration, but it does say that they sometimes operate at stress levels that would wear out continuous mechanisms very rapidl[illegible] As a result, they sometimes [illegible] ife ~~characterist~~[illegible] that textbook and handbook data would [illegible]. Most books, however, have not [illegible], come to [illegible] the world of [illegible].

Mos[illegible] intermittent motion [illegible] are subjected to cyclic stresses because impacts or cyclic rolling occurs someplace in the mechanism train. As a result, such mechanisms experience surface-fat[illegible] wear more commonly, perhaps, than do other [illegible].

As suggested ea[illegible] [illegible]urface-fatigue wear differs from other [illegible] of wear [illegible] several ways. For examp[illegible], it [illegible]as learned that fatigue [illegible]ear rates may ~~be proporti~~onal to the [illegible]nth power of stress levels If this were true, a ten percent reduct[illegible]on [illegible] operating stress level would increase the life of a [illegible] part by 258 percent [illegible] a 30 percent reduction would [illegible] the life by 2,500 percent. By [illegible] the stress wear-rate relationship of a[illegible] wear shows that a 50 percent reduction [illegible] stress level wi[illegible] only double the life of the [illegible] part

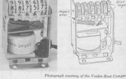
These st[illegible] of th[illegible] [illegible] intermit[illegible]t mot[illegible] showed us that stress [illegible] are extremely difficult to control where impact [illegible] [illegible]volved If the duration of an [illegible] [illegible] decreased from 10 [illegible] seconds to 9 [illegible], [illegible]he stress level [illegible] the surfaces [illegible], 10 percent. This means that life [illegible] reduced by a factor of 2.58. [illegible] differences make a *lot* of difference [illegible] performance and [illegible] a frequent theme when dealing with [illegible].

Surface-fat[illegible] wear rate[illegible], [illegible] [illegible]me other types of wear, also vary [illegible] [illegible]it[illegible] [illegible]perating tem[illegible]. Mentioned prev[illegible], [illegible]t[illegible] known that surface fatigue charact[illegible] produces [illegible] wear for a long period -and then produces rapid [illegible].

All of these factors make the life of an int[illegible]r[illegible] [illegible]otion device extremely difficult to predict It also makes the life of such a [illegible] extremely difficult to det[illegible] experiment[illegible]. One of the characterist[illegible] of t[illegible] class [illegible] [illegible] that they [illegible] [illegible]fference [illegible] life from one model ~~of a given design to another mo~~del of the same design. It [illegible] not at [illegible] unusual to get a [illegible]tion [illegible] lifet[illegible] of 50- or 100-to-1 in a large sample of a given design What appear to be small differences [illegible] the shapes material compositions, driver forces, backlash and pre-tra[illegible], [illegible] voltage, operat[illegible] temperatures, inertias and densities of parts, etc., result [illegible] small changes in stress level which are magnified by the nonlinearity of surface fatigue to produce large differences in wear rates.

It should be mentioned, too, that surface-fatigue wear is found not only at those points in the machine where impact has been designed in; for example, in a ratchet-wheel tooth or drive pawl, but at all other points in a machine, that are connected to an impacting or cyclic-rolling pair. As noted in the coupled oscillator model for an elastic-body machine, we realize that any sudden disturbance at one point in the machine will be transmitted as a serious of shocks and vibrations throughout the machine. Therefore, surface-fatigue wear occurs in the bearings of the ratchet wheel just as it does in the ratchet teeth; and even more mysteriously, surface fatigue will appear in the bearing surfaces in the frame that supports the shaft on which the ratchet wheel is mounted. Generally speaking, wear will be most rapid at those points in the mechanism train where clearances and/or backlash are greatest. This can sometimes result in some very unpredictable failures.

The counter illustrated in Fig. 5–5, for example, usually wears most rapidly at the input end, where a solenoid-driven inverse escapement drives the first count wheel. Here, where there are repeated, direct blows of the driver on the load, we would expect this to be the ultimate wear point, and it usually is. But I recall one test in which one counter failed when the long shaft on which the transfer pinions (intermittent gears between count wheels) are mounted, became so loose at the far end (most



Photograph courtesy of the Veeder-Root Company

Fig. 5–5. Wear occurs at the point of greatest clearance in many situations, not always at the expected point. In a counter shown in photo (*Left*) and detailed drawing (*Right*), for example, wear can sometimes occur on the left-hand end of the pinion shaft even though the drive impacts are introduced on [illegible] remote from the driver), that the pinions jammed the count wheels. The pinion shaft in this design is normally fixed to the frame. It is not intended to rotate. However, the pinion shaft was slightly loose in the frame on the particular counter tested. This small clearance allowed the shaft to impact the frame every time a load was put on the first or right-hand pinion. The repeated impacts gradually opened an ever larger hole in the frame until the shaft misalignment became so great that the mechanism was finally jammed.

Figure 5-6 also illustrates the fact that shocks and impacts are transmitted throughout an entire machine and produce unexpected wear phenomena. The illustration shows two systems used at Veeder-Root for life testing small ratchet-driven counters. In system (A), a four-bar linkage is used to drive the counter; the input motion, therefore, is roughly sinusoidal. At (B), a large star wheel periodically strikes the input lever to the counter, driving it in that way. Here the input is a series of blows. The counters being tested by both machines are identical; both are full of intermittent motion devices (ratchet inputs, mutilated pinion transfer mechanisms, etc.). But the life test results obtained on these two testing machines differ by a factor of nearly 30-to-1. Counters driven by the four-bar crank system operate much longer than those driven by the star wheel. And the modes of failure do not differ much in the

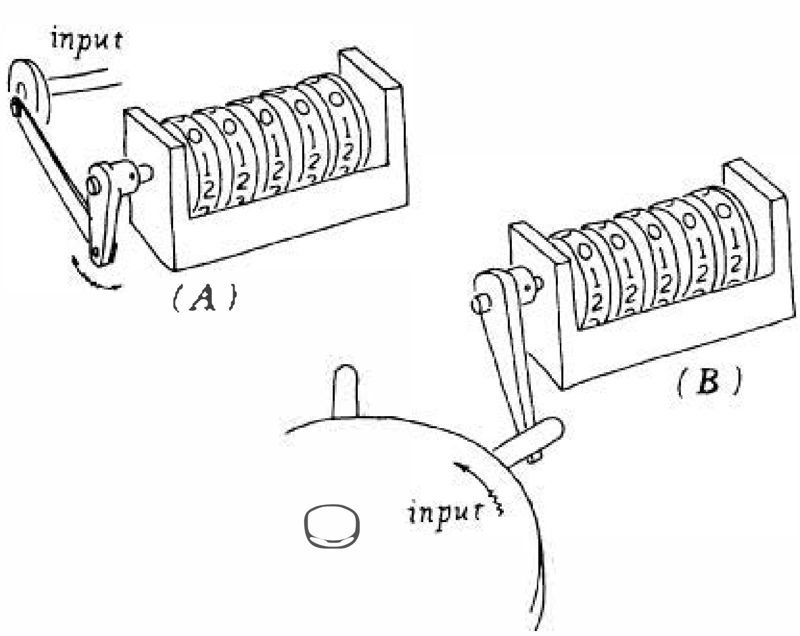


Fig. 5-6. Impact at one point in the machine will be transmitted throughout a machine and can cause wear at a number of different points. Typically, the useful life of a counter will be ten times greater when driven by the crank mechanism of (A); than when driven by the spoke wheel (impact) system, shown in (B).

two tests. The same failures occur in the same places in each, but at greatly different rates.

Counters driven by electrical stepping motors also last much longer than the same counters driven by impacting mechanisms. Stepping motors, as will be seen in a later chapter, involve no impacts or metal-to-metal contact of any kind. Counter life can be increased by a factor of 100, or more, by switching to this sort of drive.

Another thing which is characteristic of intermittent motion devices is that their functional life is often strongly dependent upon the cyclic rate at which they are operated. By cyclic rate is meant the number of cycles per minute; not velocity. In a ratchet mechanism, for example, the drive forces and load forces (and, therefore, the impact and operating velocities) are the same in a given cycle whether or not the machine is being operated at a rate of 10 cycles per minute, or is only operated once a day. Why, then, should the ratchet mechanism fail to function properly after only 1,000,000 cycles when operated at a high cyclic rate, when it will go 200,000,000 cycles when operated at a lower rate? Measurements of wear, furthermore, show that wear rates per cycle are the same in both cases, as would be expected. Yet almost always, a strong correlation between performing life and operating cycle rates is found.

The answer lies in the fact that intermittent motion devices are always struggling on the verge of instability. Internal vibrations, distortions, discontinuities, etc., can limit the operating speed even if force magnification does not produce destructive stresses. Although identical ratchets operated at two different speeds can exhibit identical wear rates, the machine being driven at the higher speed will be more sensitive to small amounts of wear; and its performance, therefore, will be degraded more rapidly than will that of the machine operating at the lower speed. At some speeds the vibrations and distortions caused by a given impact will not have "settled out" before the next impact arrives. Small increases in clearance, etc., will cause jamming or skipping that would not occur if there were more time between operating cycles.

Performance of Worn Intermittent Motion Mechanisms

Different types of intermittent motion mechanisms respond to wear in different ways. Some continue to function, but with errors in positional control,

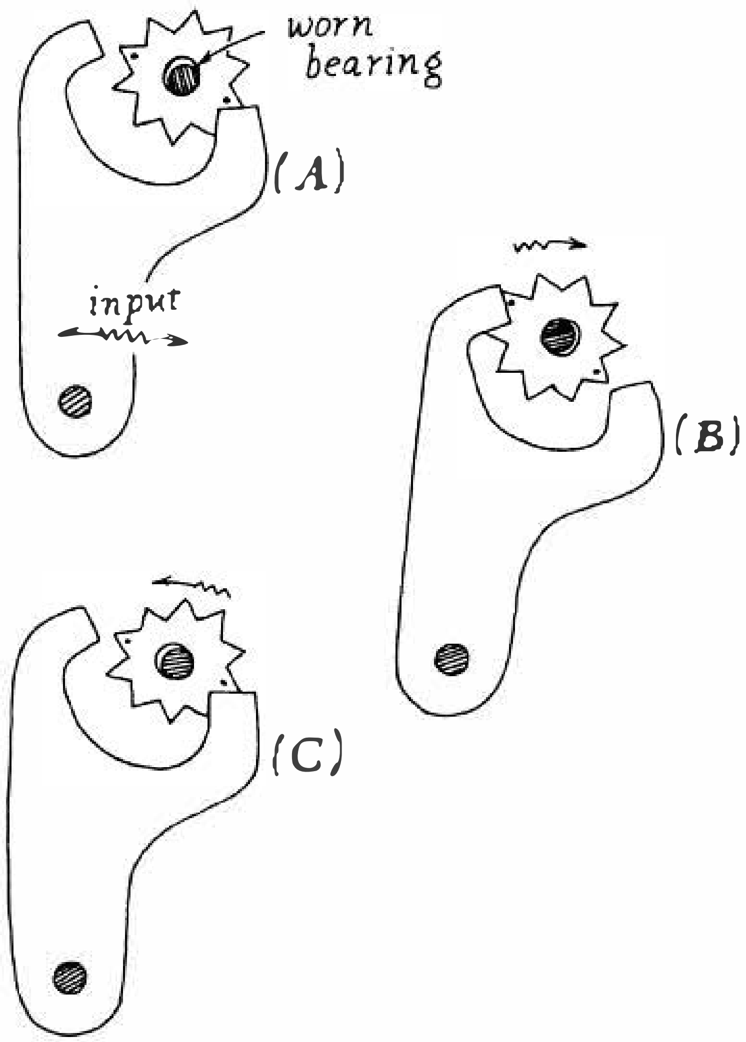


Fig. 5–7. Worn inverse escapement. The output wheel will "slump" (A) in its loose bearing between strokes of the driver. As a result, the wheel will often oscillate (B and C) instead of indexing. The wheel should index in a clockwise direction.

cyclic rate, or the like. Some continue to function but produce violent distortions in the system. Others just simply cease to function.

In Fig. 5–7, for example, an inverse escapement mechanism is seen, in which the bearing hole on the output wheel has been worn by repeated impacts of the driver against the wheel. On the first half of the stroke (part A), when the right-hand drive tooth comes in to move the wheel forward, it merely moves the wheel up from a slumped-down position due to the looseness in the bearing. Then, when the left-hand tooth engages the wheel (part B), it returns it to a slumped-down position. Part C is a repetition of the starting position (A) of the cycle. Thus the mechanism will no longer work—it just oscillates back and forth around a single drive tooth.

The ratchet of the system in Fig. 5–3, on the other hand, will continue to function even after it has worn badly; but the ratchet wheel will be indexed to different positions than it was when new. This may make the machine driven by the ratchet unusable, but since the ratchet is still capable of stepping the wheel, it may still have useful life in some situations.

The Geneva mechanism of Fig. 5-4 falls into a different category. Here the stroke is so long that it takes a great deal of wear in the drive pin, or slot, before the positional accuracy of the device is affected. But a little wear can drastically change the acceleration and velocity pattern of the mechanism. A "good" Geneva has a drive pin that slides in but fills the drive slot with no clearance or backlash. Such a Geneva exhibits the displacement-versus-time curve shown as a solid line in Fig. 5–8. A Geneva with a loose slot might exhibit the displacement curve shown as a dotted line in the same illustration: if it were driving a heavy, rigid, well-damped load.

If the system is partially elastic, however, and/or is operating at a high rate-of-speed, then it is possible for the worn Geneva to produce a displacement-time curve such as that shown in Fig. 5-9. As the load hesitates and jumps ahead of the driver, the pin contacts first one wall of the slot and then the other. So far, this does not seem to be too much of a problem. The mechanism is still functioning: the input pin is indexing the output slot with little positional error.

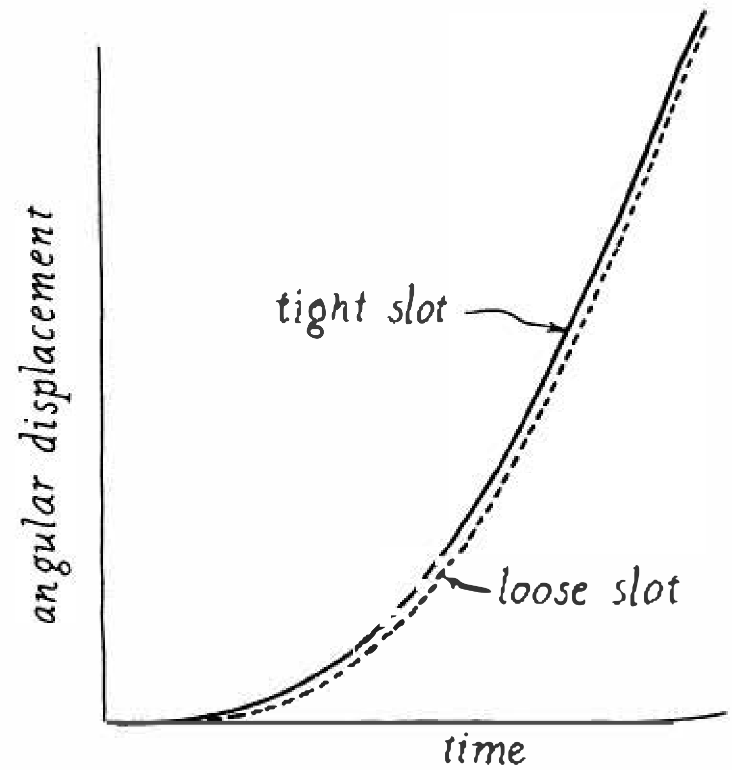


Fig. 5-8. Angular displacement-versus-time curve for a Geneva mechanism in which the roller is a snug running fit in the slot (heavy line); and for a Geneva mechanism with a loose roller-slot combination (dotted line); with a heavy, well-damped load and no elastic body reactions.

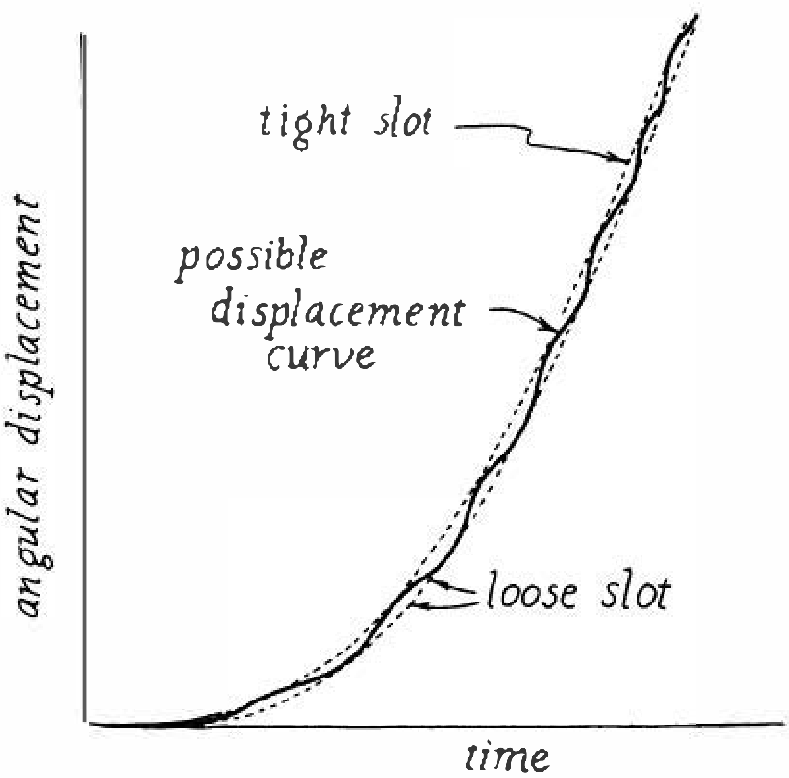


Fig. 5-9. Angular displacement-versus-time curve for a loose roller-slot Geneva driving an elastic load. Displacement of the output is shown as a heavy line. Compare with Fig. 5-8.

Graphical differentiation of this new displacement curve, however, reveals the problem, as shown in Fig. 5-10. The velocity and acceleration patterns are violently distorted from what they should be. (See Fig. 9-4 for comparison.) A little clearance between pin and slot has magnified and distorted the drive forces. New vibration stress and control problems will result. Wear rates throughout the machine will accelerate. Jamming, fatigue failures, or timing problems will finally result and the machine will cease to function.

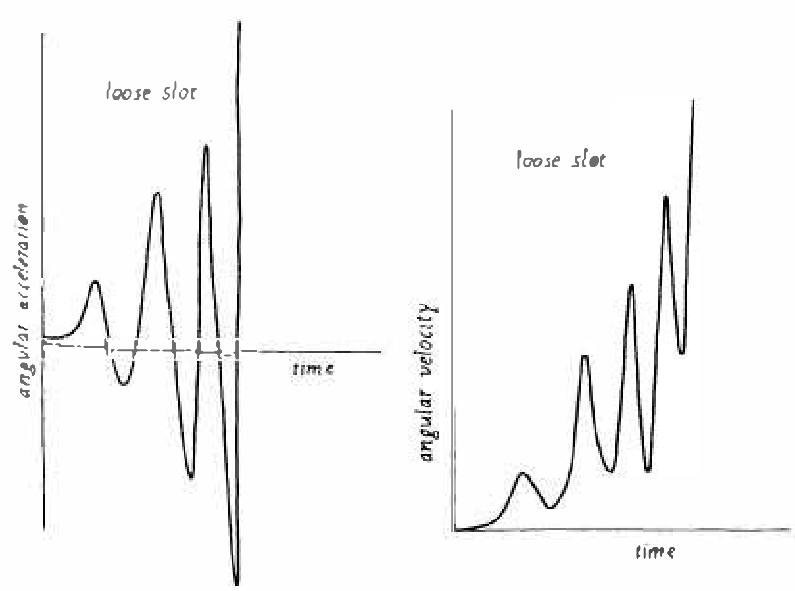


Fig. 5-10. Angular velocity- and acceleration-versus-time curves for the loose-slot Geneva driving an elastic load. These curves were obtained by successive differentiations of the heavy curve in Fig. 5-9.

Design Recommendations to Improve Stability

The importance of minimizing wear or the consequences of wear cannot be overstressed. Machines weighing tons are often reduced to scrap by wear-induced changes in geometry of only a few thousandths-of-an-inch on some critical part. There are several things that can be done to improve the life of intermittent motion machines. All of these things are intended to improve its stability in one or more ways; either to stabilize the geometry by reducing

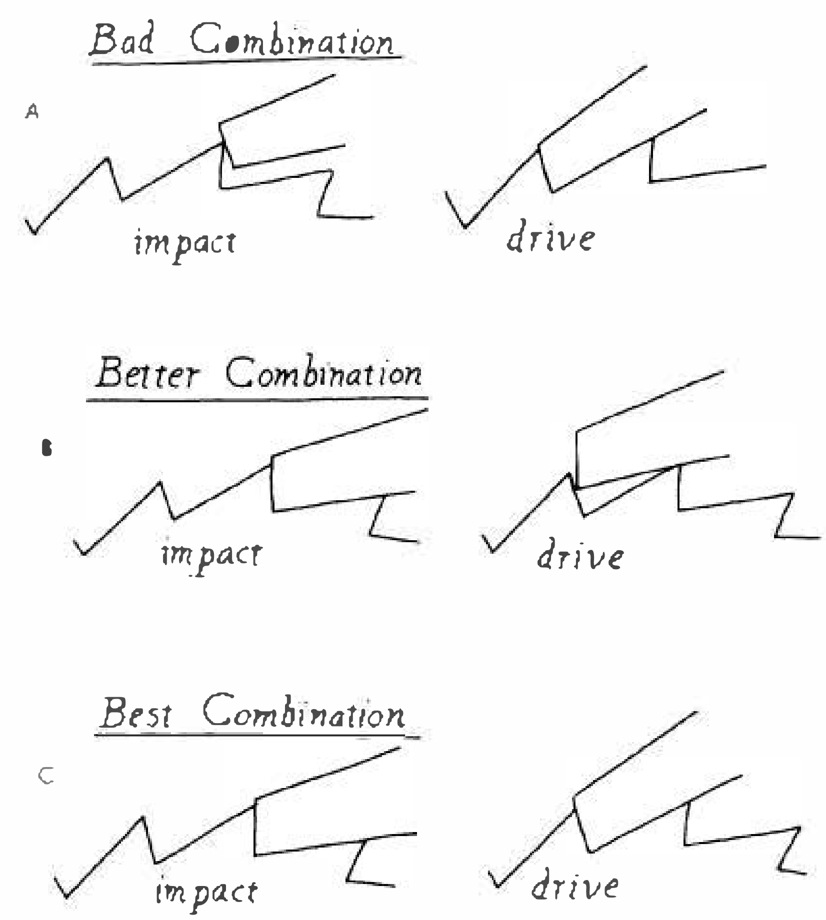


Fig. 5-11. Ratchet-tooth shapes to reduce wear. Ideally, ratchet teeth (and the teeth of other intermittent motion devices, such as gears, escapements, and clutches) should have large bearing surfaces at impact points to minimize impact stresses. It is also desirable to distribute drive forces over large areas, but since impact forces greatly exceed drive forces, large impact areas are more important than large drive areas.

wear, or to stabilize performance by making the design more tolerant of wear, or both.

The general "solution" to wear is the same as the "solution" to the problems considered in the last chapter. Keep stress levels low, minimize impact velocities, minimize sudden changes in velocity and acceleration, and the designs will work better and last longer.

More specifically, we can reduce relative velocities between sliding and/or impacting parts. Reducing velocities between sliding parts reduces adhesion and fretting wear rate. [illegible] also reduces temperature generally speaking, and so reduces ploughing effects and the [illegible]her types of wear. Redu[illegible]ing [illegible] velocity under impact [illegible]nditions, of course, reduces stress levels [illegible], as has already been seen.

Stress le[illegible] can also be reduced by increas[illegible] contact areas. This [illegible] especially [illegible]tant for areas of contact during collisions.

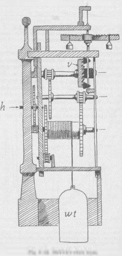
Ratchet teeth should be [illegible] for example, so that they [illegible] be loaded over flat surfaces when they [illegible] means that they subsequent[illegible] [illegible] against sharp or rounded surfaces as [illegible] the [illegible]dle [illegible] (B) of Fig [illegible]. It [illegible] not desirable to have ratchet or clutch teeth impact on sharp surfaces even [illegible] they push agains[illegible] [illegible]aces as shown [illegible] the upper [illegible] (A) of the illust[illegible]. Ideally, of course, they should bot[illegible] and [illegible] on flat surfaces, as shown [illegible] the lower [illegible] (C), but if it [illegible] necessary to make [illegible] choice, the impact [illegible]eometry should be favored [illegible] this is where the maximum stress levels are generated.

Another [illegible] which should be done as often as possible, [illegible] to provide parts th[illegible] are shaped to func[illegible] properly even after they hav[illegible] [illegible]orn. In Fig. 5-7 there [illegible] inverse escapeme[illegible] that had ceased to funct[illegible] after the bearing hole in the output wheel had been slightly enlarged by wear. It [illegible] possible t[illegible] [illegible]sign this mechanism to reduce wear (a better bearing, for example), or to m[illegible] the design more toler[illegible]nt of wear. A center of rotati[illegible] for [illegible] [illegible]rm could be selected, and [illegible] and wheel tooth geomet[illegible], that would reduce [illegible]

The best way to study the wear tolerance of a new design [illegible] to make paper or pla[illegible] cutouts of the [illegible] links of the [illegible], [illegible]ing [illegible] [illegible] parts at all wear [illegible]. Put these pseudo parts through [illegible] p[illegible]ces on a drafting board and look [illegible], [illegible] of control, et[illegible]. [illegible] is [illegible] to a tolerance study, of course, but with [illegible]al wear-induced [illegible] changes accounted for. It is surprising how few de[illegible] study [illegible] effects of wear [illegible]ay, yet very often a design could be improved drastically by a few dimensional changes.

The designer should also consider supplying excess drive torque to accommodate the build-up of friction in a worn [illegible]. He may have provided bearings that never wear, but wear [illegible] "[illegible]portant" parts often end up [illegible]ast d[illegible] [illegible]nd must be accounted for. Extra torque has extended the life of many [illegible]. On the other hand, extra torque can also limit the life of [illegible], because extra input power means more power lost or used somewhere [illegible]. If the load does not absor[illegible] it, the system will, [illegible] friction, impact, distort[illegible], and w[illegible]. Thus the designer is caught in a dilemma. Maybe a [illegible] extra torque to provide for a little fric[illegible] [illegible]

Of course the de[illegible]er should also keep the servi[illegible] man[illegible]nd. If wear cannot be [illegible]ed, provide [illegible] maintenance of the crit[illegible]l elements of the ma-chin[illegible] [illegible] or replacement of parts i[illegible] common. So[illegible] heavy-duty Genevas, for example, are provided with replaceable "wear shoes" on each



side of each slot to prevent development of the motion patterns of Fig. 5–10.

Beyond this: test all designs thoroughly, recognizing and correcting original errors. If there is any company objection as to design time consumed, make it clear that you are now engaged in the "development" phase of research and development! Most managements assume that you (and all other "good" designers) will do everything right the first time and they may object to modifications. But intermittent motion devices are NOT predictable; testing and design development are almost always mandatory.

Historical Note

The amount of energy absorbed by an intermittent motion device determines such things as its efficiency and the rate at which it will wear. In general, of course, the more efficient the machine, the longer it will last and the better it will work. In this regard, it is interesting to see how far we have come in the design of clock and watch mechanisms.

The illustration (Fig. 5–12) shows a clock train built by Henri deVick between 1364 and 1370. The escapement is marked *v*; and *h* indicates the shaft on which the hands were mounted. These hands were about 3 feet long and probably weighed several pounds. The clock was intended for use in a municipal or church tower, as were all such clocks built at that time. The weight which drove this clock, labeled *wt* in the attached sketch, weighed 500 lbs and fell 32 feet in 24 hours. The input to the machine, therefore, was 16,000 ft-lbs every 24 hours! The output work, of course, was the rotation of the clock hands. They must have been cutting the gear teeth as they kept time! Comparing the efficiency of this clock to that of the modern electro-mechanical watch mechanisms that run for a year on a tiny battery shows how far we have come in our understanding of this class of intermittent motion devices.