CHAPTER 4

The Mechanics of Intermittent Motion

A SPRING-DRIVEN RATCHET

The Problems of Intermittent Motion Mechanisms

Now that we are experts on graphical and elastic body mechanics, let us take a closer look at a special topic—intermittent motion devices.

It would be nice if all intermittent motion mechanisms were as easy to analyze as a "block-on-the-table" or the "finger ratchet," studied earlier. Unfortunately, however, this is rarely the case. In most machines, the driving and driven linkages are considerably stiffer than fingers. Drive velocities are high, and contact times between parts can be very short and extremely difficult to measure. The result: impact, vibration, chatter, surprisingly high contact forces, large energy losses, instability in the machine, and other horrors. Let us take a look at an actual situation.

Spring-driven Ratchet Mechanism

Figure 4–1 shows progressive stages of a ratchet mechanism in operation and corresponding plots of the net torque on the ratchet wheel as a function of time during a single drive cycle. In the first stage (A), only the drive cam is in motion and the net torque on the wheel is zero. In the second stage (B), the cam has released the arm and the drive spring has moved the arm forward to contact the wheel. All the energy which is stored in the arm during this pre-travel is now dumped rapidly into the wheel and into the wheel bearing. The arm produces a very high force on the wheel for a short period of time. This creates drive torque and a bearing reaction force F_B, which, in turn, produces a friction force F_b. This blow inevitably causes the wheel to jump ahead of the arm so that, for a brief instant, the force exerted by the arm on the wheel disappears.

By the third stage (C), however, the arm (still driven by its spring) has caught up with the wheel (because in the meantime the wheel has been slowed by its load and also by bearing friction) and the arm has delivered another, lesser blow to the wheel. The wheel has jumped ahead again, the arm has caught it again, and struck it a third and a fourth time. Following the fourth collision, the arm and the wheel finally travel along together so that the arm exerts a fairly constant force on the wheel (we are assuming a drive spring with a fairly flat rate). The force levels and times associated with each blow are different than those associated with each previous blow, in general, becoming lesser as the two parts get together. There is some opinion that a "typical" machine situation involves about five separate impacts, each time that two links collide. Theoretically, there are an infinite number of separate impacts in each collision, but our old friend "energy loss" does make it possible for the two parts eventually to get together.

In the fourth stage (D), a so-called non-overthrow tooth on the ratchet arm has engaged the inner row of teeth on the ratchet wheel, exerting a sudden, and very sharp, stopping force on the wheel (and producing bearing reactions as before). The net torque

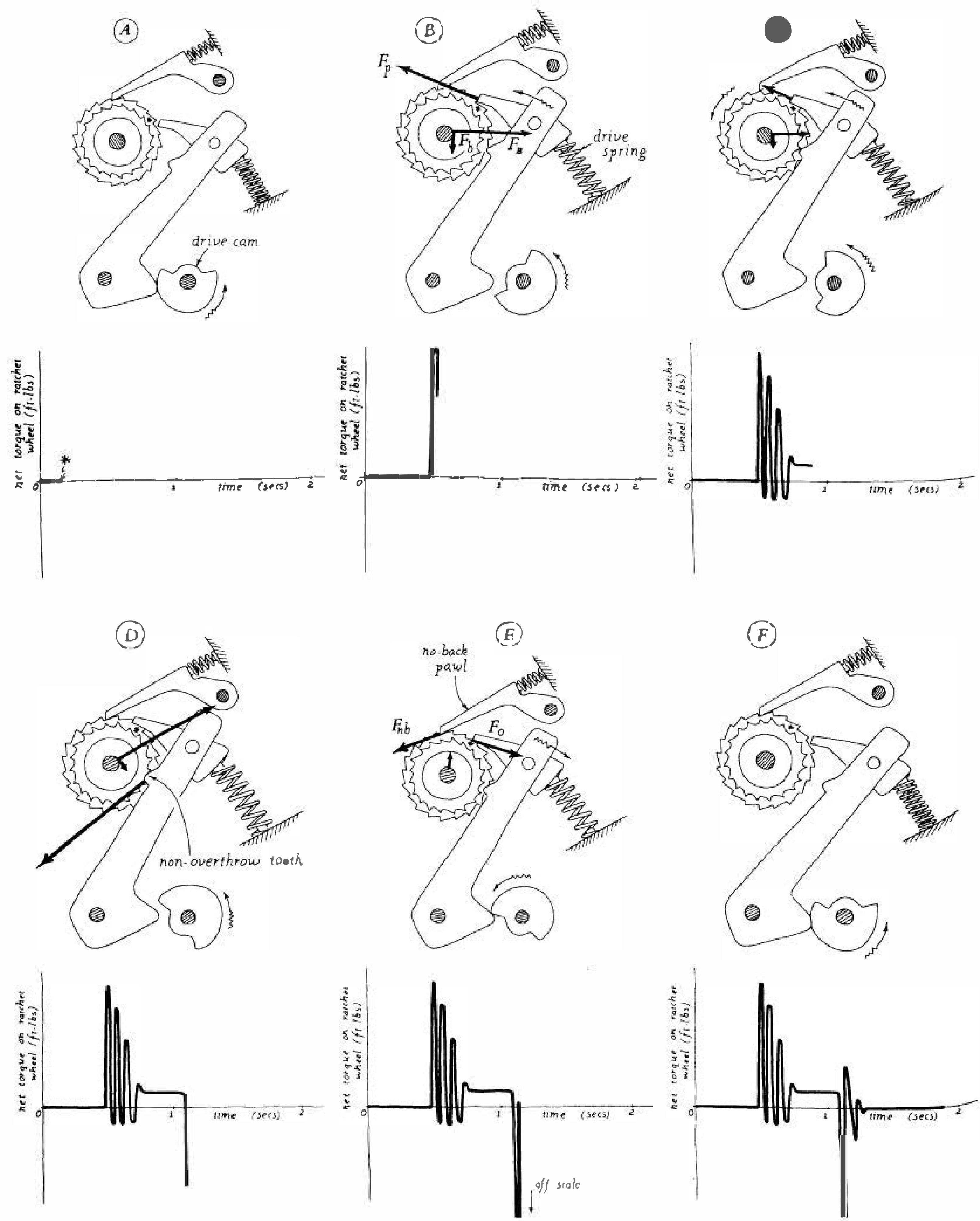


Fig. 4–1. Progressive stages of a ratchet mechanism in operation showing step-by-step development of the torque-versus-time curve for this impacting device.

is negative now and the wheel comes to a stop. The no-back pawl prevents significant rebound but this stop, just as the drive impact, would consist of several short, sharp blows rather than one long one.

As in the simple case of the finger ratchet studied earlier, the positive area under the torque–time curve must be cancelled by an equal negative area if the wheel is to be brought to a complete stop. A little reflection will show that the time required to stop, in this situation, must be considerably shorter than the drive time. In an actual machine the drive time might be a second or more, as shown on the graph, and the stopping time, a small fraction of a second. For the curves to generate equal positive and negative areas, therefore, the stopping torque must be very large, as shown in curve (E).

In stage (E), the ratchet arm has started to retract, driven by the drive cam. The arm, in this condition, exerts a negative drag on the ratchet wheel, but this drag is counteracted by an equal and opposite reaction from the holding pawl, thus the net torque on the wheel is zero and the wheel remains at rest. In stage (F), the net torque on the wheel is still zero. The arm is ready to pick up and drive the next tooth of the wheel.

Calculations—Spring-driven Ratchet

For a real understanding of the ratchet mechanisms, acceleration; velocity; and displacement curves for the wheel as a function of time should be drawn. To draw accurate curves, however, values must be put on the torque curve just sketched in. To do this, the maximum torques, impact and motion times, etc., must be known.

Obviously, these values will be extremely difficult to obtain. Start, perhaps, by drawing a curve of the torque on the drive arm as a function of time. We now have a pretty good understanding of the influence of the drive spring on the arm; since we design the spring (and therefore know just how much force it can produce), and since the spring operates for a fairly long and easily measured time. We know, therefore, what the momentum of the drive arm is when it contacts the ratchet wheel, and can use this to estimate the approximate magnitude of the impact forces when it encounters a stop; similar to the situation illustrated in Figs. 3–1, 3–4, and 3–5.

But the wheel is not a stop. It is free to move and, in fact, is intended to move under the influence of the blow exerted on it by the drive arm. As a result, the arm will lose some, but not all, of its energy when it strikes the wheel--some of this energy going into useful work to drive the wheel; some being lost in bearing and internal friction. The arm will then gain more energy from its drive spring before striking the wheel a second time; and will then let loose a second bundle of energy. It is doubtful that there is a man alive who could calculate these energy losses and transfers.

If the time involved in the multiple impacts was known, on the other hand, impact forces could presumably be computed since we know that the area under the stopping portion of the torque curve must cancel the area under the drive portion of the curve. But again, we hit a snag. We could estimate the time involved in a total collision process, but to measure peak forces, the time for each and every one of the four or five little separate impacts that actually occur during one collision must be known.

Of course, if we could experimentally measure or estimate the maximum forces involved impact times could then be estimated, but this is very difficult also. Impact specialists tell us that the forces generated at the surfaces of colliding bodies are not transmitted equally to all other parts of the body; because we are dealing with elastic bodies, not rigid bodies. What happens is that stress waves move through the body, reflect from bearings or free surfaces, return toward the source of impact, magnifying or cancelling subsequent stress waves (from other individual impacts) that have been generated in the meantime and are themselves surging through the body. This situation makes it virtually impossible to measure impact force levels, even if strain gages were mounted on the colliding bodies.

In fact, there appears to be little that can be done to put actual values on the graph of torque as a function of time which has been drawn for this simple ratchet situation. Mathematically inclined designers are in the same boat, however. They can write equations for this situation and can produce numerical answers, but only by *assuming* impact times or maximum force durations, or spring rates, etc., for the colliding bodies. We can do this too. If you are designing the ratchet, you will know what its inertia is and how stiff the drive and pawl springs are. If you assume that each impact involves a force–time relationship that is sinusoidal (remember the quasi-static model), and that total time duration is on the order of magnitude of a few milliseconds (a good assumption in many machines), also that

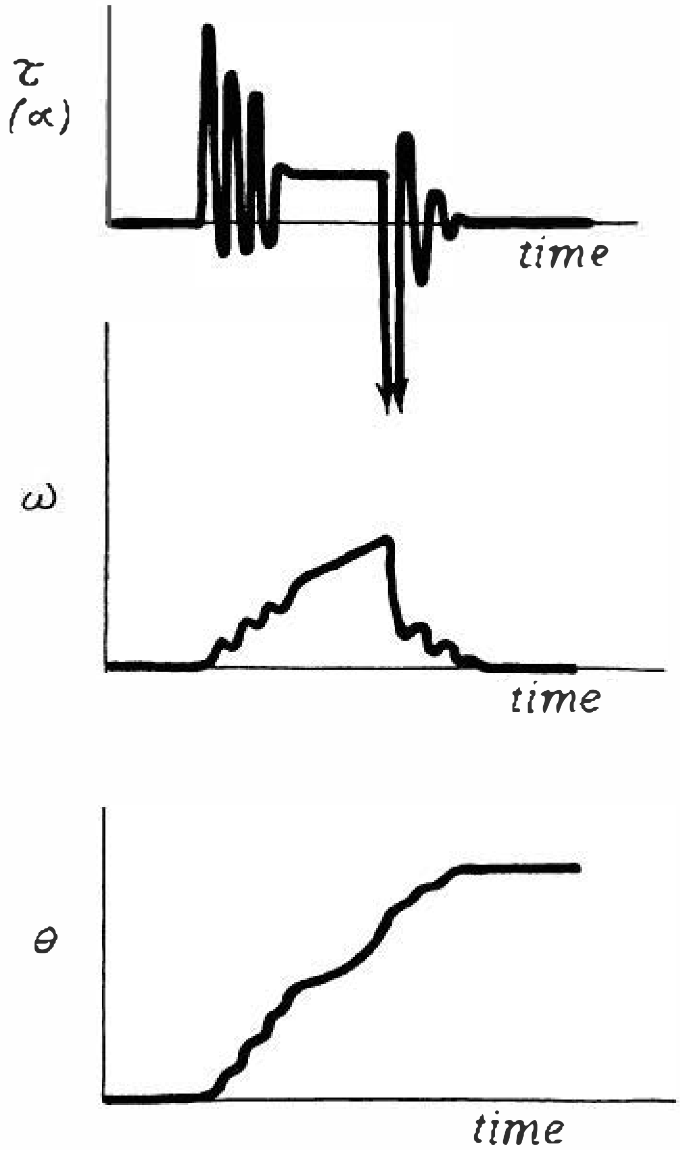


Fig. 4-2. Complete motion curves for the ratchet mechanism of Fig. 4-1.

there will be approximately four or five impacts during each collision, you will be able to rough-in a torque-time curve that is as good an approximation to the actual as the world's finest "calculator" could achieve.

Using graphical techniques we will now approximate the acceleration, velocity, and displacement curves for the ratchet wheel as well. The resulting curves are shown in Fig. 4-2.

THE PROBLEM OF FORCE MAGNIFICATION

We have run into a serious problem in trying to calculate the exact force, acceleration, velocity, and displacements of an intermittent motion device when collisions are involved. The analysis, however, has revealed other problem areas that are perhaps more important to us as machine designers than the purely theoretical problem of being able to put exact numbers on the various functions involved. For example, we have again exposed the problem of force magnification first considered in Chapter 3. Even though we cannot calculate exactly what the peak impact forces are, we can see that they can be many, many times greater than the drive forces which we have thoughtfully provided for our machines—structures and linkages intended to survive only under drive-force levels would fail very rapidly in many intermittent motion mechanisms. Increased stiffness and strength is most easily accomplished by an increase in the mass of the parts and structure involved. But we have seen that increases in mass tend to increase force levels during impact, defeating the purpose of the mass increase. It is necessary, therefore, to do everything possible to provide increased strength with a minimum increase in mass. Machine design experience shows that this can be done in most situations so that the various parts are capable of withstanding the forces involved.

But the problem goes beyond that of designing parts that are strong enough not to break down if subjected to operating forces. Usually the concern is about the cost of the machines; and infinite strength and stiffness cost money, especially if mass must be minimized. Furthermore, even if machine members are strong enough not to fail catastrophically, magnified forces can lead to serious problems of wear and instability, as will be seen in the next chapter. Higher forces should not just be compensated for then, we should try to avoid them. And the only way to do this is to avoid sudden changes in velocity.

The ratchet just studied has built-in sudden-changes-in-velocity, of course. The wheel was at rest when struck by a rapidly moving drive arm. Now the wheel must suddenly start to move and the arm suddenly slow down when they collide. This is inherently a questionable design; this type of mechanism should only be used where loads are very light and where low cost is very important. Under these conditions, of course, spring or coil-driven ratchets can give excellent service; witness the highly reliable stepping switch (Fig. P-2 in the Preface, for example). But whenever possible, avoid impact! Or at the very least, impact velocities should be kept as low as possible.

Reducing Forces by Reducing Impact Velocity

One way to reduce impact velocity is to design the ratchet in such a fashion that the driver acquires a high velocity as soon as possible during the pre-travel, but has a relatively low final velocity, as shown in Fig. 4-3. This can be done by proper design of a cam, perhaps, or a drive solenoid or spring system. We are usually interested in a high *average* velocity in an intermittent motion drive system so that a large number of operating cycles can be produced in a given time. A high average can be achieved by using a high initial acceleration and a low final or impact velocity will still remain, as shown in 4-3A. On reflection, this is not a common approach, but it is an excellent way to improve the performance of a high-speed intermittent motion machine.

Impacts generated in a driver are also transmitted through an entire machine. They will be magnified further wherever there is clearance, backlash, pre-travel, etc., involved. Keep the whole machine as "tight" as possible; nothing is more effective in reducing wear in an intermittent motion situation.

Another method of reducing impact is to use a system in which the driving member approaches the driven member at an angle, when first making contact. This results in a low RELATIVE velocity between colliding surfaces at the time of impact, even if the driver is moving at a high rate of speed. And it is relative velocity, not absolute velocity that determines the magnitude of impact forces, as was seen in the previous chapter.

The Geneva mechanism, shown in Fig. 4-4, works

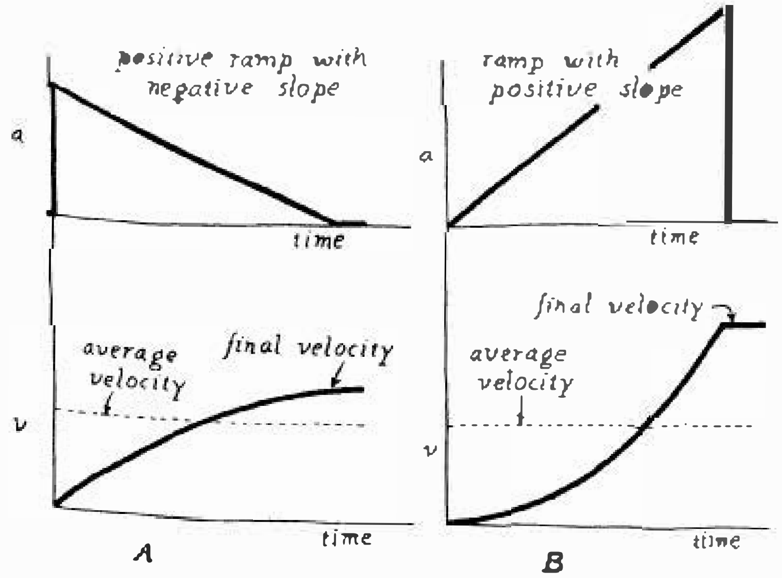


Fig. 4-3. If the acceleration curve is a positive ramp with a negative slope it will produce a lower final velocity than would an acceleration curve with a positive slope, even though both produce the same average velocity and, therefore, the same cycle time. A negative slope torque (acceleration) applied to a ratchet arm, for example, will produce a lower impact velocity without increasing operating time.

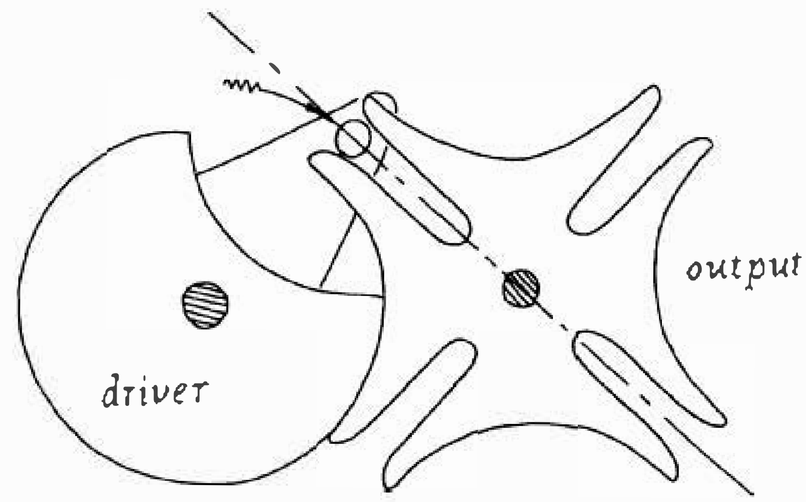


Fig. 4-4. The drive pin in a Geneva mechanism enters the drive slot parallel to the wall of the slot to reduce impact velocity.

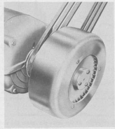
on this principle. As the drive pin enters the slot in the stationary output member the pin is moving along, tangent to the centerline of the slot, with zero relative velocity between the pin and the wall of the slot. As the driver continues to rotate, the pin moves against the wall of the slot and picks up and indexes the output member. The change in velocity in the output is fairly gradual. True impact is not involved if the parts are made correctly. Genevas will be studied more thoroughly in a later chapter.

This type of mechanism, incidentally, where the driver approaches the load at an angle, produces less output motion for a given input stroke than does a direct push device, such as a ratchet. But again, this is an excellent way to reduce impact stress levels.

Programming Drive Forces to Reduce Stress

Another method of reducing stresses caused by collisions is to design the power plant so that it pushes the driver link gently until contact has been made with the output link; and then pushes hard. This will probably increase the time required for the driver to move from its cocked position to drive position (the pre-travel time) but could have significant benefits in the life of the machine. Again, this is not a very common approach, but it certainly has a lot to recommend it if this complexity can be afforded. There is one rotary solenoid manufacturer who has manufactured solenoids that produce a "soft" followed by a "hard" force of this type. Cam systems can also be used effectively to change the leverage of a driver, for example, during the stroke.

Pneumatic and hydraulic drivers can be designed



Photograph courtesy of Fairchild Hiller, Industrial Products Division

Fig. 4-5. Eddy current coupling used to program the torque delivered to a load and thus reduce the rate-of-change of acceleration when starting and stopping. Also see Fig. 4-6.

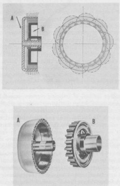


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Fig. 4-6. Further details of the eddy current coupling used to reduce accelerations in a drive system. Also see Fig. 4-5.

to produce tailored forces, also. In systems involving clutches, brakes, electric motors, and the like (Chapters 13 and 15, for example), we can reduce peak torques applied to a load by using devices such as the eddy current coupling shown in Figs. 4-5 and 4-6. This coupling will slip until sufficient velocity is attained by the driver to generate a drive torque in excess of the load torque. By programming the driver speed, the rate at which torque is applied to the load can be programmed.

Programming the Velocity of the Driver

A system can also be designed in which the velocity of the driver is programmed to reduce impact

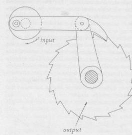
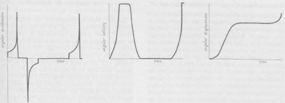


Fig. 4-7. Ratchet mechanism driven by a four-bar linkage. Also see Fig. 4-8.

velocity. An example is shown in Fig. 4-7, where a ratchet is driven by a cam rather than by a spring, as in the first ratchet example. The resulting acceleration, velocity, and displacement of the ratchet wheel is shown in Fig. 4-8; this is quite an improvement over that in Fig. 4-2.

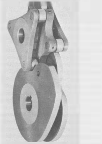
Cam Systems—Programmed Displacement

This idea of programming velocity is applied to many intermittent motion mechanisms, even to those that do not normally involve impacts. Cam systems, for example, are carefully designed to avoid sudden changes in velocity of the driven member, to reduce



[illegible] angular [illegible], and angular [illegible] shown in Fig. 4–7. A [illegible] is type o[illegible] smoothly [illegible] hat [illegible] Fig. 4–1 because impacts have been reduced or eliminated by the "programmed" drive system.

inertial loads, and hence, to reduce force magnification. Generations of cam designers have struggled to find the "perfect" cam profile for an indexing mechanism, and still argue heatedly in support of their favorit[illegible] the driver usually [illegible] consta[illegible]y but [illegible] shaped in such a way as to



Photograph courtesy of [illegible] Cam and Machine Company

Fig. 4–[illegible] impacts [illegible] cam system such as this. See Chapter 8 for further details.

program the displacement of the follower with which it [illegible] always [illegible] contact. In a well-designed system the follower starts at zero, [illegible] accelerated gradually, and is decelerated gradually. Ideally, initial and final acceleration curves have zero slopes to [illegible]te jerk. A typical cam indexer is shown in Fig. 4–9. Three popular motion curves for cam systems are shown in Fig. 4–10.

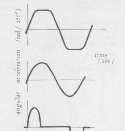


Fig. 4–10. Examples of the type of motion curves which [illegible] be obtained with well-designed cam systems.

Reducing Forces through Absorption of Energy

Another method of reducing forces in intermittent motion devices is to use a damper of some sort to absorb energy during impacts. In general, there are two kinds of dampers: those which absorb energy and retain it; and those which absorb energy and return it to the system at some later time. If a marshmallow was fastened to the tip of the drive pawl on a ratchet it would absorb and retain impact energy to some extent, and would thereby reduce the magnitude of stress, etc., during impact. Unfortunately, the marshmallow would not last very long in this application nor are there many materials or devices which would. Sometimes, friction can be deliberately built into a mechanism train to absorb energy in this manner, but friction is very unpredictable. Besides, any sort of frictional energy absorption can lead to wear. Energy absorbers of the "permanent" deformation type are used for structure isolation only and are not used in power transmission systems.

It is possible to provide a damper which absorbs and returns energy as would the spring shown in Fig. 4–11, but the designer should realize that such energy *will* be returned to the system, and that if it is returned in phase with a new load or drive-peak torque, the "damper" would magnify, not reduce, stress levels. See Fig. 3–22 which shows what can happen in a mechanism train when elasticity is uncontrolled.

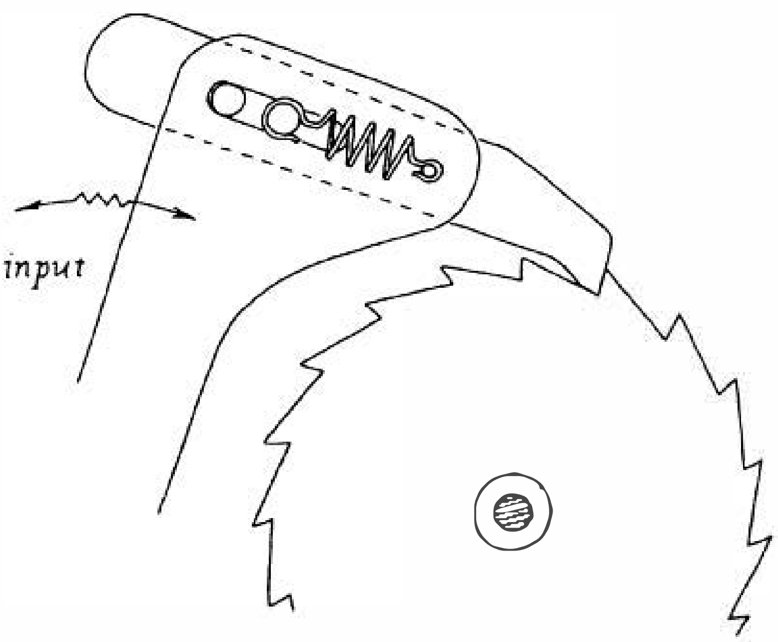


Fig. 4–11. Ratchet mechanism with a spring-loaded drive pawl. The spring would either reduce or increase impact stresses depending upon the rate at which the spring returned energy to the system and the rate at which the device was operating.

THE PROBLEM OF CONTROL

Another problem revealed by the spring-driven ratchet mechanism of Fig. 4–1 is the problem of control. When studying the motion of this ratchet at the beginning of the chapter it was seen that the collision between the ratchet and the ratchet wheel resulted in a series of impacts between the two bodies, with the ratchet wheel jumping ahead of the ratchet arm after each impact; the wheel was then slowed down by frictional drag until the ratchet arm could catch it. Ultimately, thanks to the energy loss during this process, the wheel and the arm moved together. Actually, a ratchet of this type functions more reliably when it has frictional drag in its bearing than it would if it were on a frictionless bearing. The amount of "jump" of the ratchet wheel after each blow is determined by a number of factors including the masses of the arm and wheel, the collision velocity, surface conditions, material spring rates, etc. It is possible to maximize or minimize the follow-through of the arm by using the proper inertia ratio between driver and driven links (and load), as will be seen in a moment.

But first, consider the potential difficulties of such a driver. It is always desirable in a machine for the driver(s) to determine the exact motions of the load(s). Perhaps "always" is too extreme, but certainly in most cases it is desirable. It is only on the golf links or in other game situations, where a driven member (ball) that follows an independent path of motion after being struck by the driver is tolerated. This state of affairs should definitely be avoided in most machines! Yet in the spring-driven ratchet this situation exists. It is only friction that (ultimately) makes the driven wheel stay with the driver. And friction can come when it is least wanted, then vanish when you are depending upon it. What is needed are intermittent motion devices in which the output is always under the control of the input; the position of the output link is determined by the position of the input link.

As will be seen, intermittent motion mechanisms vary widely in the amount of positional control they maintain over the driven members. Escapements control the output only at the beginning and end of motion—and then, only if they are properly designed. Ratchets require no-back pawls, non-overthrow pawls, and sometimes dampers to maintain even this much control. Interrupted gearing does a little better, but needs a locking ring to hold the output during dwell periods. Genevas and star wheels also must incorporate locking rings. Clutches must be accompanied by brakes. Cams can generally take care of both motion and dwell periods, as far as load control is concerned; but again, only if they are properly designed.

Also, in almost every mechanism, there are moments during each operating cycle when the control situation gets a little difficult. Is the brake applied and the clutch then released, or vice versa? Does the locking ring become effective at the exact instant the drive pin leaves the slot in the Geneva? Does the stopping pallet of the escapement reach an interference position just before or just after the escape pallet has released the load, etc.? Because we are dealing with intermittent motion we are dealing with periodic change in the state of the mechanism; clutched–unclutched; motion–no motion; etc. And control during change is very difficult to achieve.

There is another factor that causes designers to be alert when dealing with intermittent motion. The magnified forces studied earlier distort the various links in a machine—elastic bodies—remember. The quasi-static model revealed that each link in a machine could be considered to consist of a rigid body connected to all the other links by springs. And as in Chapter 3, the motion of a mass-spring combination can be very different than the motion of the input member that excites it. How can the output of an intermittent device ever be made to behave?

Maintaining Control by Reducing Forces

One way to reduce the control problem is to minimize drive and stopping forces; using some of the techniques discussed under the third section in this chapter, "The Problem of Force Magnification." This will reduce the effectiveness of the quasi-static springs, resulting in dealing with bodies which appear to be "more rigid." Reducing forces will also reduce wear and, therefore, improve stability in general, as will be seen in the next chapter.

But we cannot use all of the force-reduction techniques and hope to improve control, simultaneously. An eddy current slip coupling must introduce positional uncertainty between driver and load in order to function. Feedback of some kind must be used, or the coupling must be overridden by brakes and other devices, to maintain full control. Dampers also reduce positional control while modifying the influence of the driver on the load.

The Impossibility of Full Control During Impact

Many intermittent motion mechanisms inherently involve impact; they require it, perhaps, to impart motion to the load. Examples include ratchets, escapements, interrupted gearing, some cam systems, some clutch-brake systems, etc. Can the designer avoid loss of control during impact by matching the load and driver inertia or the like? Table 3–1 (in Chapter 3) lists some of the methods of maximizing or minimizing the effects of impact. How can this discussion be related to the spring-driven ratchet control problem? To start, reconsider the linear collision of two elastic blocks, using the quasi-static model. As before, the blocks can be considered to be rigid bodies separated during the collision by a stiff linear spring. In Fig. 4–12A the block on the left, labeled M_1, is approaching the second block with velocity v_1. The second block at the beginning of the action is at rest, so that this is, indeed, a linear model of a ratchet wheel and drive-arm system. Block M_1, furthermore, is moving under the influence of a constant drive force, F.

To start with, assume that the masses of the two blocks are equal. The forces on each body are shown as a function of time in Fig. 4–12B. To begin with, there is a positive small force (F), on block M_1 and no force on block M_2. When the collision starts, the force between the blocks builds up sinusoidally as a function of time. This will be an equal and opposite force and will be negative in direction on M_1, and positive in direction on M_2.

As these forces develop in the bodies, there will be a change in velocity in each body, with M_1 tending to come to rest and M_2 starting to move. At the midpoint of the collision process, the velocity

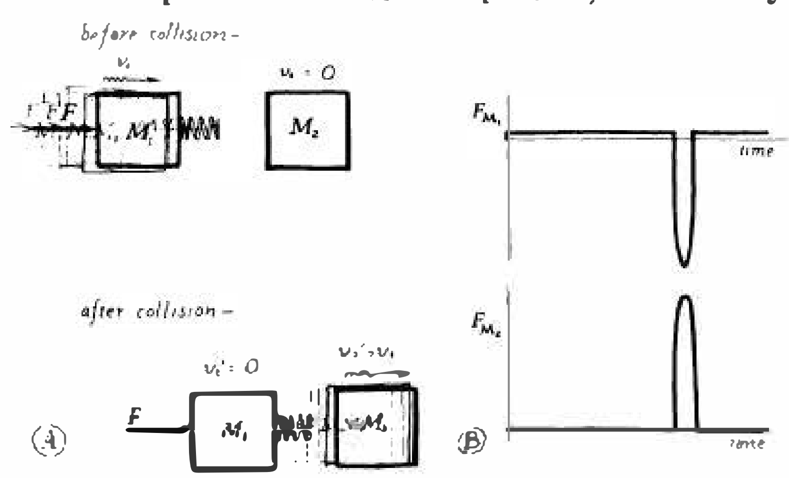


Fig. 4–12. Collision between equal masses. M_1 is the driver. In this situation it stops momentarily after the impact and the load (M_2) moves off with a velocity equal to the initial velocity of the driver. This happens even though there is a continuous drive force (F) on M_1. A very poor control situation.

of the two blocks will be equal to each other, and towards the right. This turns out to be the midpoint of the collision process, or the point at which the two bodies have reached maximum deformation during the collision: the spring, assumed to exist between the two bodies in the quasi-static model, is now fully loaded. If the collision process were to cease at this point, the two bodies would now travel together with a common velocity towards the right and there would be a single-impact collision. If we assume, however, that the bodies are elastic, then the spring between them will unload; at least in part. (In part, because there is energy loss during an actual collision and the spring will not return as much energy to the system as it has stored during the first half of the collision.) As the spring unloads its energy, it will push equally on the two blocks, pushing to the right against M_2 and to the left against M_1. These forces tend to increase the velocity of M_2 and decrease that of M_1. If the blocks are, indeed, equal in mass to start with, and if there is not too much energy loss during the collision process, then the unloading of the spring will produce just enough negative impulse (force × time) to bring block M_1 to a rest, and enough positive impulse to increase the velocity of M_2 to a value approaching the velocity of M_1 at the start of the collision.

It is not possible to determine from FAVD curves why the first block comes to rest and the second acquires the velocity of the first during a collision between bodies having equal mass, or no energy loss. In order to determine why this is so, it would be necessary to solve the equations for the conservation of energy and the conservation of linear momentum, simultaneously. However, for the purpose of acquiring a general feel for intermittent motion mechanics and mechanisms, it will be sufficient at this point, to state that this is indeed true. With equal masses and no energy loss, the first block imparts all of its energy to the second.

Relating this now, to the ratchet situation, if there is a ratchet arm and drive system striking a ratchet wheel and load of nearly equal inertia, then the chances are that the drive arm will come to rest upon striking the wheel and that the wheel will jump ahead with a velocity equal to the impact velocity of the drive arm. This is an efficient transfer of energy from driver to driven link, but probably, it will be difficult for the ratchet arm to keep up with and control the ratchet wheel. The arm will follow the wheel. It just will not follow closely enough to maintain control. This is, in fact, somewhat akin to the golf ball situation.

If the mass of the driver is increased substantially, and/or the mass of the load and driven link reduced, we find that the situation can be improved somewhat. The driven link will move away from the collision at some velocity greater than the collision velocity, also the driver does not change its own motion appreciably during the collision, but follows the load fairly closely. This allows the driver to control the load a little better, as seen in Fig. 4–13.

The larger the driver becomes in relationship to the load, the higher is the velocity imparted to the load during the collision. If the driver had an infinite mass, the load would be driven away at twice the velocity of the driver at the start of the collision as shown in Fig. 4–14. Their relative velocity is now V_1 (since the infinite mass would not slow down at all) and so the lack of control is the same as that of the equal mass system studied earlier. An infinitely large driver is, however, non-existent. Having a "fairly large" driver and a smaller load will usually tend to improve the control situation. It is not common, however, to find ratchets that are larger than their loads!

If the driver is small, compared to the load, then it will rebound or reverse its velocity during the collision and the load will start very slowly. In other words, the load's initial velocity will be significantly less than that of the velocity of the driver, upon

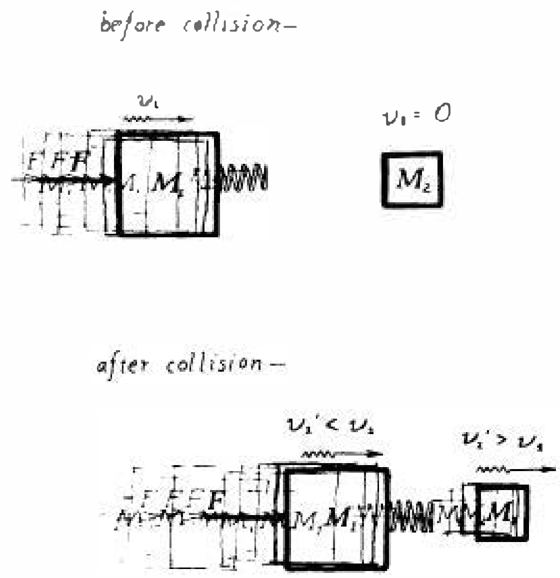


Fig. 4–13. Large drive-mass striking small load. The driver continues to move following impact, though with reduced velocity. This is a better control situation than that shown in Fig. 4–12, and may be the best of all possible choices.

impact. If there is a drive force on the driver, as is common in machines, its reverse will reverse again, and eventually the driver will come back to take another shot at the load; but, in general, this is a very bad arrangement and it takes a long time for driver and load to get together. Control of the load is difficult or even impossible for the driver. (See Fig. 4-15.)

Although the three cases: a very small driver, an equal (to the load) driver, and a very large driver have been discussed, in none of them will the load and driver stay together during and after the first impact. Unfortunately, there are no conditions between these points where the bodies will move together after the impact. Only in a fully plastic collision (all the energy absorbed by the bodies during collision is dissipated within the bodies) will they move together after the impact. In such a collision, however, the bodies are deformed and stay deformed, which obviously cannot be tolerated in an actual machine design situation. Inevitable, therefore, is the conclusion that whenever impact occurs in a machine there is always a control problem to be faced. The driver and load will not, and cannot, stay together until after a considerable amount of energy has been dissipated in the machine by multiple impacts.

The only way out of this dilemma is to try to

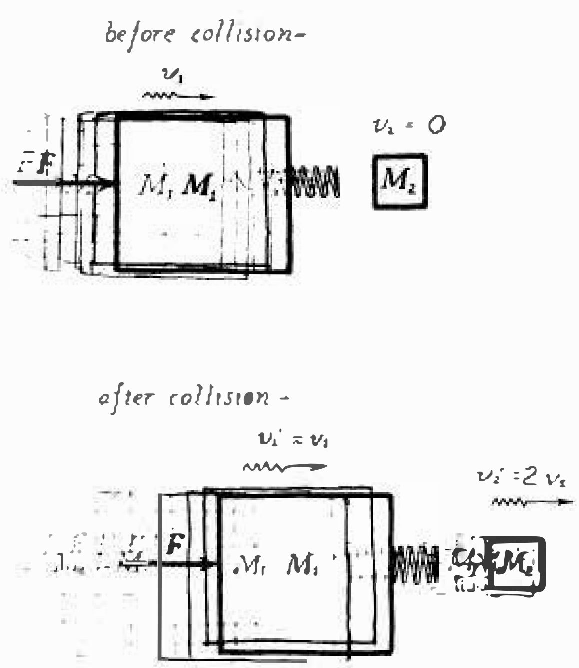


Fig. 4-14. Very large driver striking a very small load. The driver velocity is not affected significantly by the impact, but the load moves off at twice the driver velocity after being struck; thus this system produces the same lack of control as the system of Fig. 4-12.

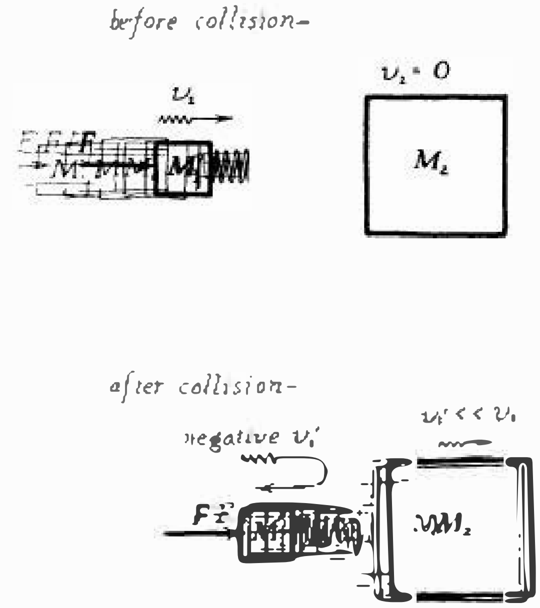


Fig. 4-15. Small driver striking a large load. The driver reverses direction of motion on impact. A very poor control situation.

design a mechanism that avoids impact; or failing that, to try to design one which reduces the magnitude of impacts. Don't "load the spring" and you will retain better control.

THE PROBLEM OF SPEED

The forces generated during impacts and during sudden changes of velocity in intermittent motion mechanisms eventually limit their operating speed. Remembering the quasi-static model, we see that the mechanism shown in Fig. 4-16, part (a) can be represented by the series of rigid bodies and springs shown at the bottom of that same illustration, part (b). The motor shaft on the left is trying to drive the four-bar linkage, which drives the ratchet pawl, which indexes the ratchet wheel. Because of impacts between the ratchet pawl and wheel, very high forces are generated at that point. These forces are high enough to distort not only the ratchet wheel and pawl, but any other links between that point and the original power source in most designs. The situation is made considerably worse if there is any backlash or clearance in the joints of the four-bar linkage; as there almost always is.

Any success in driving the ratchet wheel will depend upon the impact-produced distortions and vibrations that modify the intended geometry of the elements of the machine. Remember, too, that force is proportional to acceleration which is the rate-of-change of velocity. And velocity is the rate-of-change

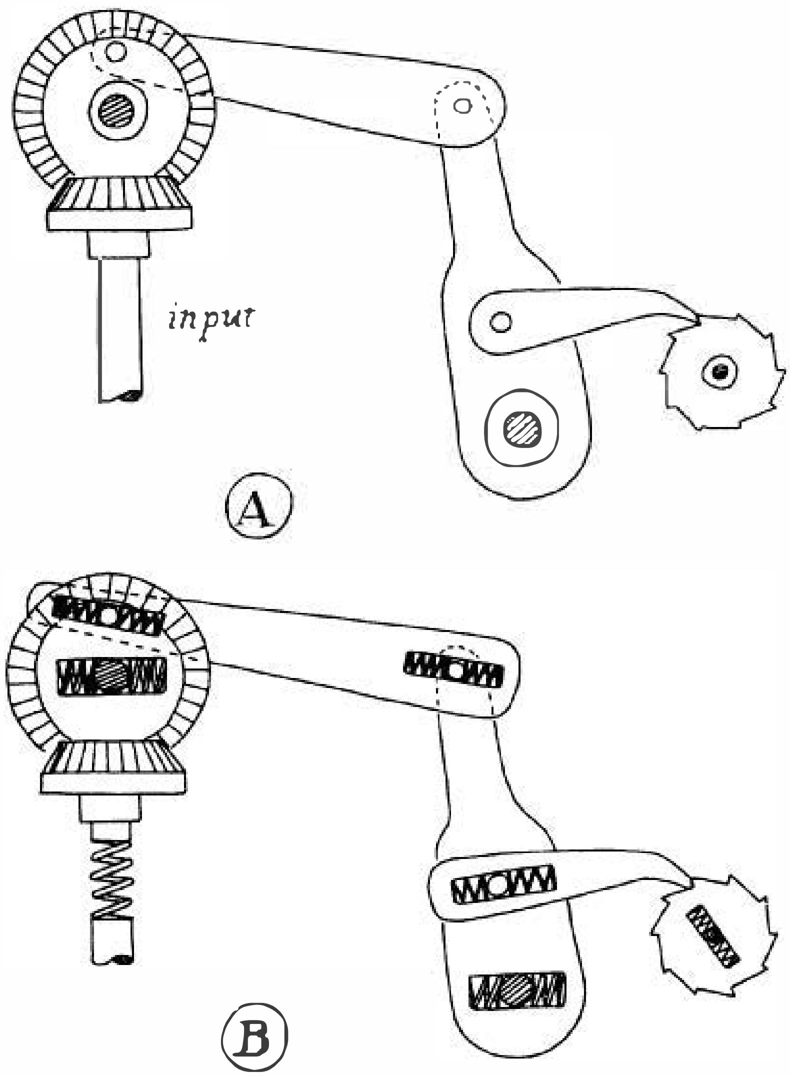


Fig. 4-16. (A) Four-bar linkage ratchet mechanism. If this mechanism is driven at too-high an indexing rate its performance will become elastic and it will behave just as the quasi-static model shown in (B), at the bottom of the illustration. A designer's nightmare!

of displacement. As the operating speed of the machine increases, it requires greater and greater forces to make the various links move through their intended displacement cycles. Drive torques must be increased drastically as the desired speed is increased. As with impact, this results in increased deformation and vibration within the machine. At some input speed, galloping or hesitation would begin to be apparent in the ratchet wheel. The connecting links have become too rubbery, or jammed, to drive the wheel consistently. At a higher speed no motion at all might be seen in the ratchet wheel. Of course, at speeds of this magnitude, the stresses could be large enough to destroy the machine, or the motions of various elements could get so far "out-of-time" that destructive jamming could occur.

Again, we are faced with a situation that will never occur in the world of "rigid body" mechanics where geometry is king, and there is no limit on the theoretical speed at which a four-bar linkage can be put through its paces. But in real life there is a limit, as explained by our quasi-static model and by our study of the vibration of coupled oscillators. The closely related problems of control, force amplification, and speed limitation are very real in the land of intermittent motion.

TORQUE CHARACTERISTICS OF DRIVERS

Most intermittent motion mechanisms must be driven by something else in the system. Input must be provided from a rotating shaft, an electrical motor, an hydraulic motor, a solenoid, a spring, or some other type of power source. Perhaps the only exception would be the stepping motor which is self-powered and which is discussed as an intermittent motion "mechanism" in Chapter 14 of this book. A motor is not really a mechanism, of course, but steppers are such an important way of providing intermittent motion in this day and age that we could not neglect them.

In any event, most of the devices we shall consider will require a driver of some sort; and a few moments should be spent at this time in considering some of the characteristics of typical drive systems, since they will inevitably influence the performance of our mechanisms.

Very commonly we will be driving our mechanisms from a constant velocity source, usually a rotating shaft whose velocity does not change significantly as long as the machine is turned on. Such a shaft, in turn, could be driven by a synchronous motor or by a variable-speed motor whose torque capability is such that the loads imposed on it by the intermittent motion mechanism do not affect its overall velocity. In most cases, we will assume that the constant velocity shaft can also provide considerably more torque than is required. And again, this is not a bad assumption in most machine design situations (as long as we have selected our power source properly). Most of the motion curves you will find in this text are based on the following type of drive input: constant velocity with more than adequate torque. Whatever happens in the output of the intermittent motion device the input will keep turning around at the same speed.

In some situations, however, torque is definitely limited, even though the input runs at a constant velocity. Many systems, for example, include a slip clutch of some sort that will limit the peak torque which can be delivered through an intermittent motion mechanisms to the load. If the clutch were to slip at too low a torque level, the mechanism might not be able to produce the peak accelerations required by its output motion curves. If this were the case, the machine and mechanism might malfunction, and the designer must be sure to avoid this. One example of the use of a slip clutch in an intermittent motion system is given in the design example in Chapter 16, where it is used to provide manual control over an intermittent motion rather than to limit torque. But it will limit torque, too, and the designer must take this into account.

Other types of drivers will run at a velocity that is proportional to the load on the driver. A spring, for example, falls in this category. It is definitely torque limited, but will not slip and so will produce output acceleration proportional to the load. We will see examples of spring drivers in our studies of ratchets and inverse escapements as two examples. Here the output motion curves are strongly influenced by the torque capability of the driver.

Some electrical motors will also run at a velocity that is proportional to the load on the system. Again, the designer must take this factor into account when constructing motion curves for the output. Check your motor supplier for the speed-torque characteristics of his products.

Finally, some drivers will produce torque which is a function of time. Electrical solenoids, for example, produce torque in proportion to the electricity passing through their coils. Since it takes time for this electricity to reach the peak value, it takes time for the solenoid to develop full torque. Again, this sort of driver is frequently used with ratchets and inverse escapements.

Calculating Drive Torque Requirements

As mentioned above, the motion curves you will find throughout this text describe the motion of the *output* of various intermittent mechanisms. These curves usually represent output displacement, velocity and acceleration as functions of time or of input displacement. By multiplying the output acceleration by the inertia of the load, we could, of course, calculate the torque required to accelerate the load in accordance with that acceleration curve; and if we are not careful, we might assume that the resulting torque curve would define the torque which the input driver must produce to move the load, but

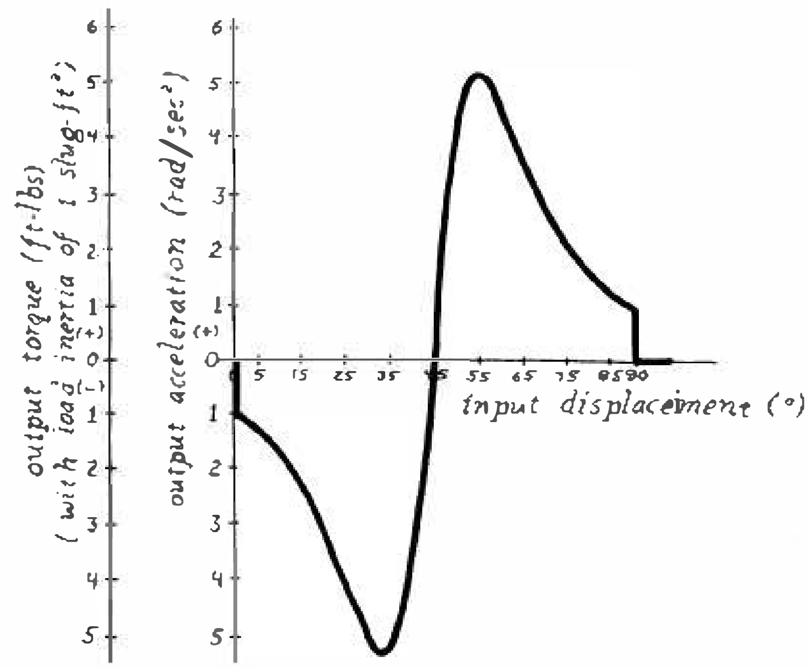


Fig. 4-17. Acceleration of the output member of a four-slot Geneva mechanism as a function of input displacement when the input is rotating at a constant angular velocity of one radian per second. The same curve also shows the torque the driver must exert on the output member to produce this acceleration pattern in a load having an inertia of one slug-ft². (See the second vertical scale.)

this is not the case. As we saw in Figs. 1-9 through 1-12, the torque produced on a load is not necessarily equal to the torque required of the driver. In fact, it is probably rare that we will find a situation in which input and output torques are equal to each other. Let us consider an example to be sure we understand this very important point.

Figure 4-17 shows an acceleration curve for the output of a 4-slot Geneva mechanism that is rotating in a counterclockwise direction with an input velocity of 1 rad/sec. (See Chapter 9 for a complete discussion of Genevas). Notice that the peak acceleration produced by this mechanism has a magnitude of approximately 5.2 rad/sec². (The acceleration is negative because rotation is counterclockwise.) If we want this Geneva to drive a load having an inertia of 1 slug/ft² (or 1 ft-lb-sec²) then there will be a one-to-one relationship between the acceleration and torque scales. An acceleration of 5.2 rad/sec² will require a torque on the load of 5.2 ft-lbs. Let us make this load assumption, therefore, and add a torque scale to the illustration, as shown. We now have a graph of the acceleration of the output member of the Geneva—and of the torque exerted on the output by the input to achieve this acceleration.

We have defined the amount of torque required to drive the load. What drive torque must the input member now provide to produce this amount of output torque? To answer this question, we must make geometrical studies of the Geneva mechanism.

Figure 4-18 shows the 4-slot Geneva after the input has completed five degrees of motion. The graph in Fig. 4-17 would indicate that the load torque requirement at 5 degrees is 1.2 ft-lbs. With the aid of the drawing we can determine the line of action of the force transmitted from the driver to the output member of the Geneva. We are assuming, in this analysis, a frictionless system to clarify the calculations and are following, therefore, the example illustrated in Fig. 1-10. The force between input and output must be perpendicular to the surface of the drive slot. In "real life" we would undoubtedly have to assume that friction were present to some extent, would have to estimate its magnitude, and would follow the procedure illustrated in Figs. 1-11 and 1-12; but for the moment, we assume no friction.

To return to Fig. 4-18: as mentioned above, the drawing shows the line of action of the force exerted by the driver on the output member (or vice versa). Once this line of action is known, we can draw perpendiculars from it to the centers of rotation of

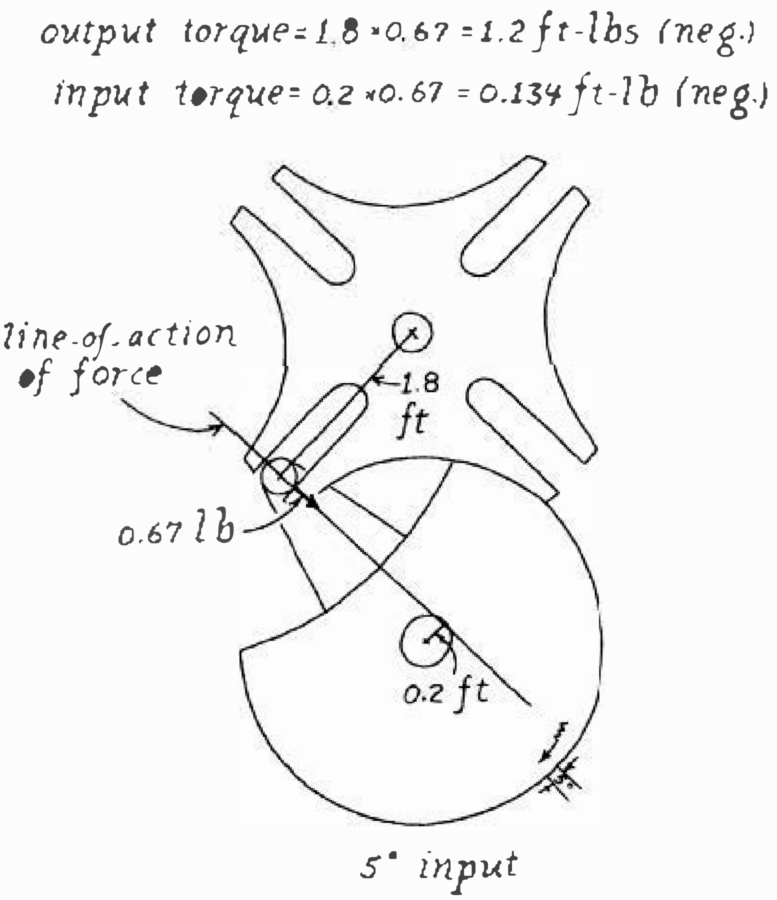


Fig. 4-18. Construction for determining the force generated by the input member on the output member (and vice versa) after five degrees of rotation of the four-slot Geneva. Input and output "lever arms" for this force are also shown. Resulting input and output torques are noted. They are both negative because both are counterclockwise

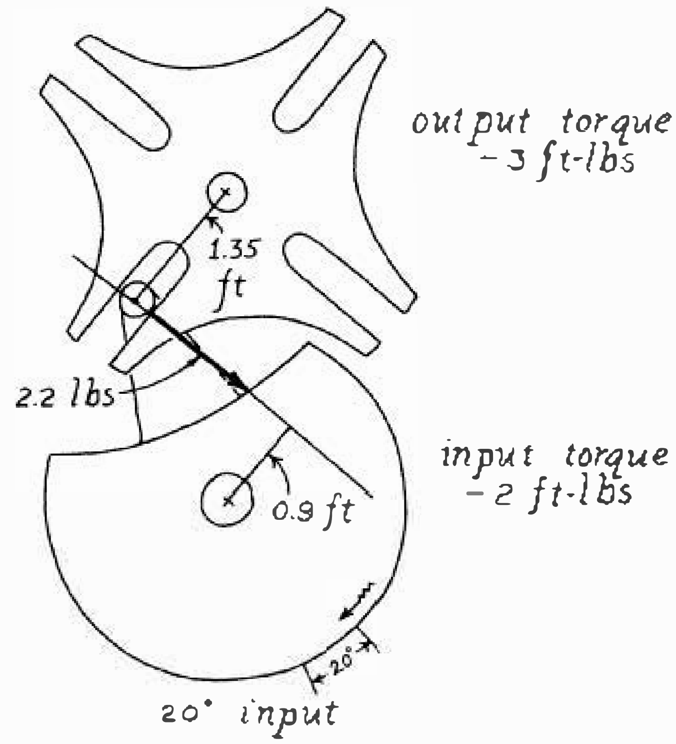


Fig. 4-19. A force and torque construction for the same four-slot Geneva after 20 degrees of input motion.

input and output members, respectively. We know from Fig. 4-17 that we require a torque of 1.2 ft-lbs in the output at this point (5 degrees of input motion) in the cycle. The force required to produce this amount of torque at a radius of 1.8 ft is found by making a simple calculation:

$$F = \tau/R_1$$

$$F = 1.2/1.8 = 0.67 \text{ lbs}$$

The force, therefore, is 0.67 lbs. The reaction force (force exerted by the output on the input) will also be 0.67 lbs in the opposite direction. The drawing shows that the line of action of this equal and opposite force will pass 0.2 ft away from the center of rotation of the input member or driver. The drive torque, therefore, after 5 degrees of input motion will be:

$$\tau_2 = FR_2$$

$$\tau_2 = 0.67 \times 0.2 = 0.134 \text{ ft-lbs}$$

Since both torques act in a direction to cause counter-clockwise motion, they are both negative. The driver, however, continues to turn in a clockwise direction under the greater influence of the drive motor.

At this point in the cycle, then, an input torque of 0.134 ft-lbs is producing 1.2 ft-lbs of torque on the load. At first glance this seems difficult to believe, but I am sure that a study of the geometry of the mechanism will convince you that this is indeed the case. If you need additional conviction, look at the velocity curve for a four-slot Geneva (this curve is given in Chapter 9). You will see that the velocity of the output is nearly zero at the instant shown in Fig. 4–18. The input, however, is turning at a constant velocity of 1 rad/sec in our present example. This means that we have a significant velocity ratio or "gear ratio" between input and output members at the 5 degree point and would expect, therefore, to find a large difference between input and output torques.

This procedure is repeated in Figs. 4–19 and 4–20, each one representing a different instant of time during the motion cycle of our Geneva. In Fig. 4–19, for example, the input has moved 20 degrees, the output requires 3 ft-lbs, and the input torque is 2 ft-lbs. In Fig. 4–20, input and output torques are equal. There are only two instances during each cycle of the Geneva when this is true; one during the acceleration phase and one (that shown in Fig. 4–20) during the deceleration phase (again, remember our curves are "upside down" because torque is counterclockwise).

If the input torque requirements were calculated in this way for six or eight different positions of the Geneva during one cycle, we would have enough information to draw a complete input torque curve.

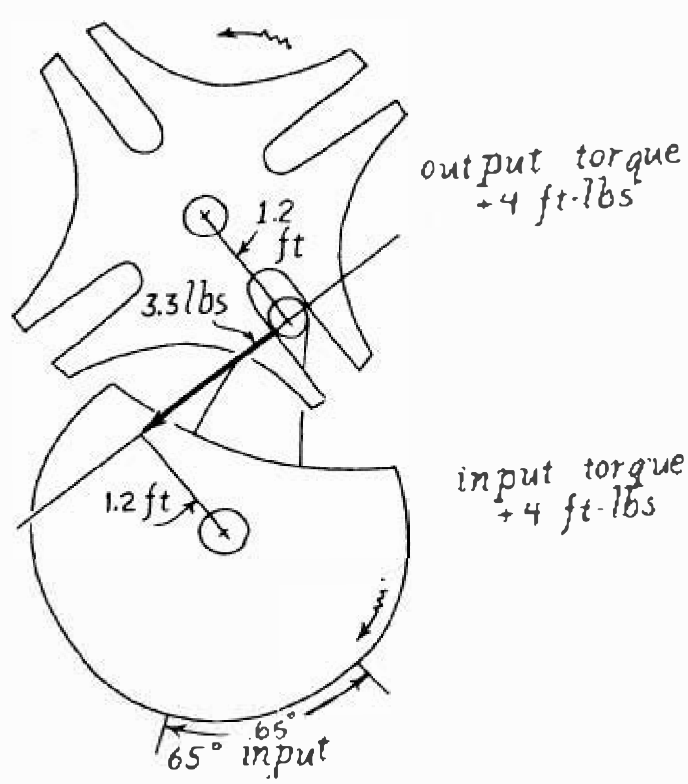


Fig. 4-20. Force and torque calculation for the four-slot Geneva after 65 degrees of input motion.

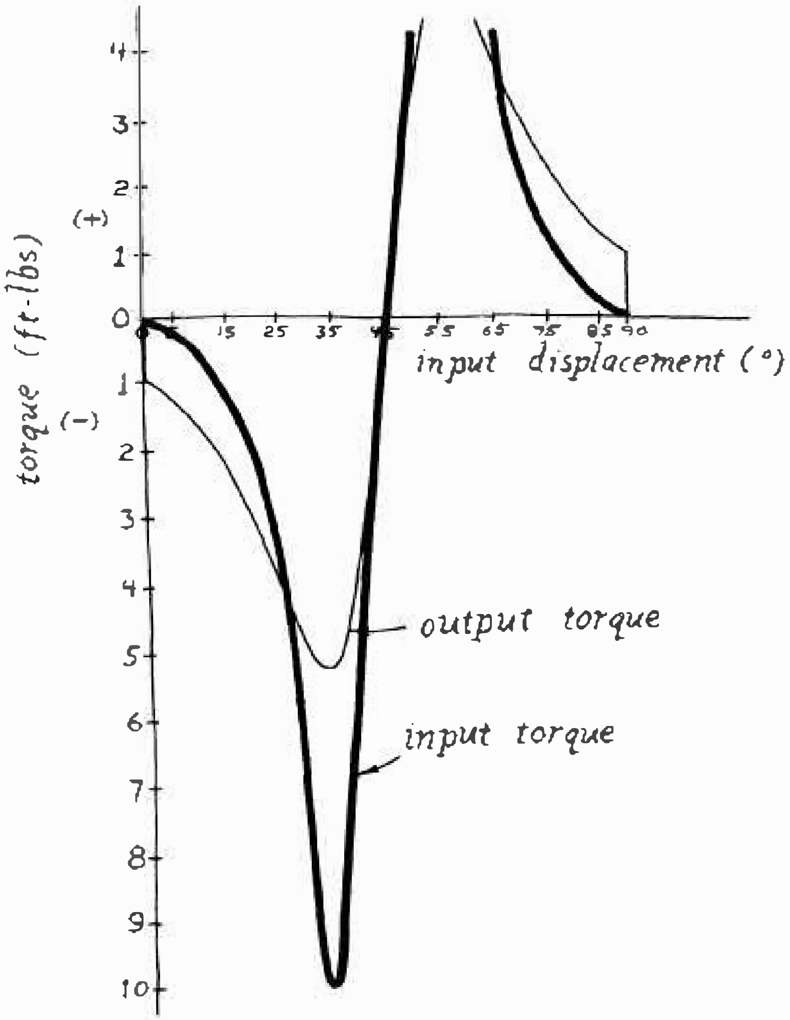


Fig. 4-21. Complete torque curves for the four-slot Geneva for one motion cycle. "Input torque" is the torque exerted on the input by the output member; "output torque" is that exerted on the output, or load, by the input. The two curves are similar in shape, but differ considerably in magnitude. Again, the horizontal axis represents displacement of the input, and output torque requirements are based on a constant input velocity of one radian per second.

This is shown in Fig. 4–21, superimposed on the original output torque curve. You will note that the two curves are similar in shape, but that the peak input torque is nearly double the peak output torque for this particular mechanism. This could be a significant design disadvantage in many situations. Cam designers, especially, worry about input torque curves and try to find cam profiles that will minimize peak input torque requirements.

In some other mechanisms, load-and-fire escapements, for example, input and output torque curves differ drastically in both magnitude and shape. Therefore, in most design situations it is best to consider driver characteristics and requirements before calling your intermittent mechanism design completed.