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The IBM SELECTRIC Composer

Proportional Escapement Mechanism

Abstract: The IBM SELECTRIC Composer's escapement system employs rotating elements, rather than the conventional rack system, to provide the required displacement. This permits the basic unit of escapement to be varied, and allows the number of units per escapement cycle to vary in proportion to character width. In this paper the authors discuss the machine requirements that led to this approach and describe the elements that have evolved. The analysis used to evaluate the design (and modify it to some extent) is also recorded, in a separate section.

Introduction

To meet the requirements of a true composer, the IBM SELECTRIC Composer had to exhibit the following characteristics:

(1) variable character widths for each selected type size;
(2) variable character sizes for each selected type style;
(3) a large variety of type fonts that could be easily changed; and
(4) printing ease and speed of an electric typewriter.

At the very beginning of Composer development, it was recognized that these requirements could be nearly satisfied by designing a variable pitch, proportional carrier escapement mechanism into the IBM SELECTRIC Typewriter. Requirements (3) and (4) above are met (except for minor modifications) by inherent features of the SELECTRIC Typewriter, and requirements (1) and (2) are essentially escapement functions. Requirement (2), however, does imply a need for variable character height (vertical size). This need has been met by the multiple index system.[1]

The escapement design was accomplished by controlling the horizontal position of the carrier with a leadscrew and by attaching that leadscrew through a changeable gear ratio to a slotted wheel containing movable sliding pins. Proportional escapement which satisfies the requirement (1) of variable character width for a given type size is then realized by selectively sliding pins laterally with respect to the pinwheel. The rotational distance to the selected pin for each cycle is thus variable, and advancement (or escapement) of the mechanism is proportional to this distance. Variable pitch to satisfy requirement (2), variable character size for a given type style, is obtained by varying the ratio of the gear train by which the pinwheel and leadscrew are connected.

A machine speed capability similar to that of the SELECTRIC Typewriter was specified and the machine size was modeled after the 13-inch writing line SELECTRIC Typewriter. A design was selected to provide seven escapement values of 3 through 9 units with three unit, or pitch, sizes of 1/72, 1/84 and 1/96 in., in order to meet the design criteria.

The operation of each escapement element is affected by each of the others; however, in order to describe more clearly the development of this system, most of the elements are discussed individually in the following sections. The original design objectives will be discussed first, as they were applied to the escapement portion of the system; following this will be a brief description of each of the mechanical elements. Finally, some of the analytical considerations will be presented for the interested reader.

Functional requirements

Before describing the mechanical elements, we shall define the design requirements more precisely in the context of escapement operation. As the specifications are explained, it will be seen that the combination of multifont capability *and* proportional spacing creates tasks for the escapement system to which conventional typewriter hardware cannot be adapted. An attempt has been made to define these tasks specifically to indicate the need for new concepts.

1. Multifont capability: variable type size

Composer type fonts had to be available in a number of styles, and in addition (more significantly from the escapement viewpoint) each style was to be available in a number of sizes. For escapement purposes, three size groups were specified. Each group would require a different basic unit* of escapement, and the escapement would be required to selectively translate one invariant unit of displacement at its input (i. e., at the point at which the keyboard signal is received) into any one of three basic units at its output (i.e., at the typehead carrier). In the final specification, the unit widths selected were 1/72, 1/84, and 1/96 inch.

2. Proportional spacing and character width

In addition to variability *among* type fonts with respect to the basic unit width, there was also to be variability *within* a given font with respect to the width of individual characters. Different characters (including spaces and special characters) were to have different escapement values, ranging (in the final specification) from a minimum of three basic units to a maximum of nine. Thus, once it could accommodate each of the three basic unit widths, the escapement would be required further to produce a carrier displacement of any multiple from three through nine of the basic unit, and to do so automatically in response to a keyboard input.

3. Print-point stability

The escapement had to be fast (approximately 0.040 seconds for the maximum displacement), positive and reliable, but the carrier was also required to be completely at rest during print (i.e., there can be no horizontal carrier motion as the typehead prints). This requirement implied a need for constant, instantly available driving force, positive latching, equally positive release, and minimum rebound or oscillation at the end of an escapement cycle.

4. Backspace capability

The backspace capability normally found in typing systems was of course required here, but it was decided that a character-by-character, proportional backspace mode should be available in addition to the more conventional single-unit and simple repeat modes. The action could occur, internally, in single-unit cycles, but some means was needed to repeat the cycle automatically to locate the carrier at the precise position of a previous print operation. Some form of "memory" was thus indicated, so that an operator could return to any of several previous print positions without having to know the unit width of any character, count spaces, or take visual measurements.

* As will be explained, the basic unit is less than the width of any character, and character widths vary in multiples of the basic unit.

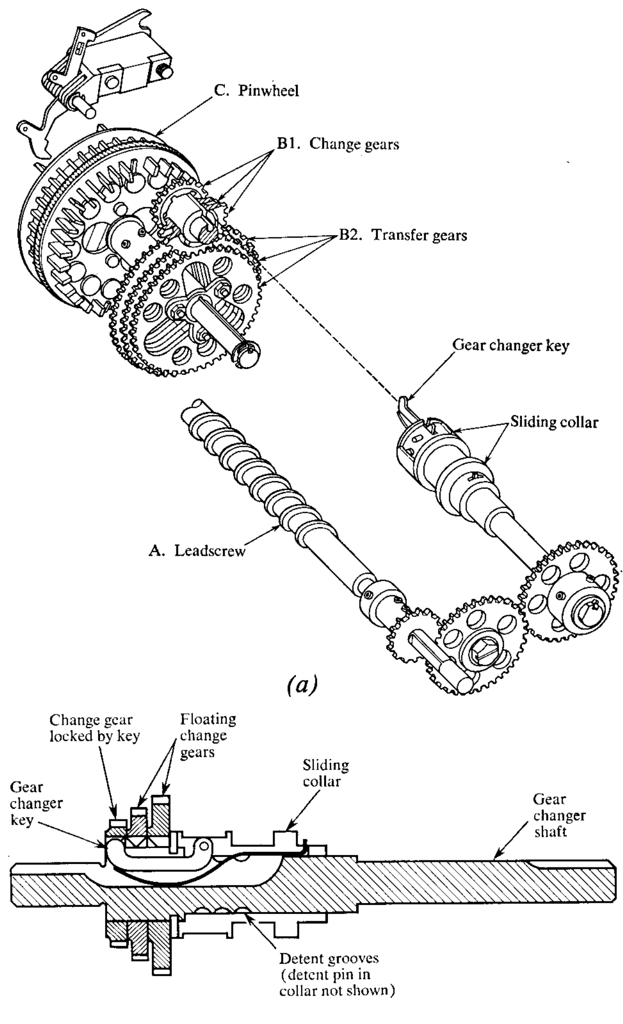


Figure 1 (*a*) Simplified illustration of leadscrew, variable gear train and pinwheel; (*b*) detail of gear changer.

5. Large-increment displacement

One final requirement, created uniquely by the proportional concept, was for some means to move the carrier through large displacements of indeterminate length (for tab operations and carrier return) without affecting the reliability or accuracy of subsequent normal proportional escapement.

Mechanical design

In this section the mechanisms that comprise the escapement system are briefly described, with emphasis on the primary function of each. Detailed description is employed

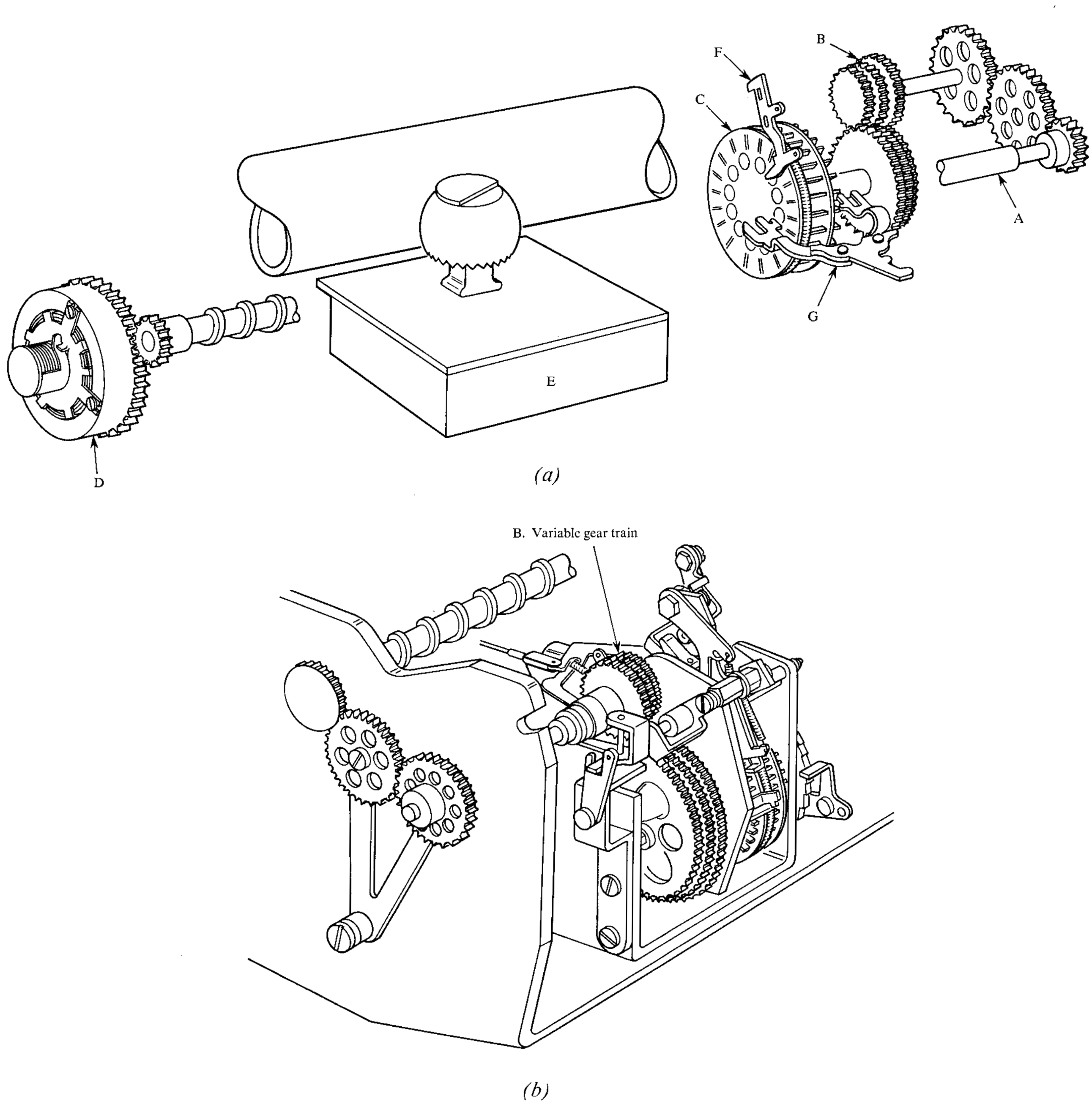


Figure 2 Basic escapement assembly.

when essential to an understanding of primary action, but that which is incidental, potentially confusing, or not wholly concerned with the escapement system proper, has been omitted.

The order of description follows that of the design requirements just described, and the direct relationship of mechanism to requirement is maintained as nearly as possible.

1. Variable type size: rotational elements and gear reduction
The ability to vary the basic unit of escapement over a range of three widths has been realized by connecting the carrier to a leadscrew and connecting the leadscrew to the escapement device by means of a variable gear train, as shown in Fig. 1a, elements A and B, and Fig. 1b. The escapement device itself is a slotted wheel, containing sliding pins and shown as element C in Fig. 1. The leadscrew is driven

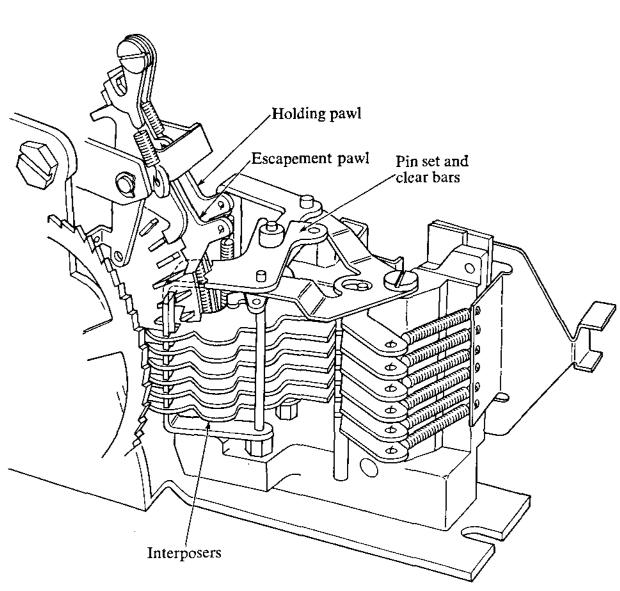


Figure 3 Sketch showing escapement and holding pawls, pin set and clear bars, and pin selection interposers.

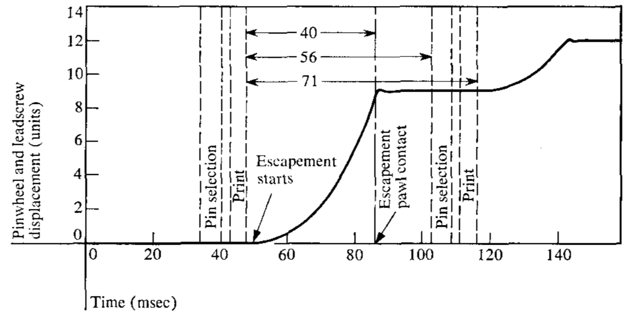


Figure 4 Timing of critical events in two consecutive printing cycles.

by a clockspring, which is encased in a housing as shown in Fig. 2, element D. The carrier *E* thus moves toward the right along the leadscrew, under control of the pinwheel and the holding-pawl mechanism *F*. When the holding-pawl device is disengaged, the leadscrew assembly will rotate; the rotation is stopped by re-engaging the pawl. One unit of displacement at the input is just the rotational distance between two adjacent pins on the pinwheel (refer again to Fig. 1); the pinwheel displacement can be coupled to the leadscrew through any one of three ratios in the gear train B. The output displacement at the carrier per unit of input can thus be changed at will by changing the gear ratio to correspond to changes in type size. The ratios are chosen such that the linear displacement of the carrier per unit of rotation at the pinwheel can be 1/72, 1/84, or 1/96 inch.

2. Proportional spacing: variable pinwheel rotation and pin selection

As discussed above, the escapement system is required to provide a carrier displacement of any multiple from three through nine of the basic unit in automatic response to a keyboard input. This has been accomplished in the design of the pinwheel and the mechanism that positions the sliding pins. The pinwheel is held at rest by a dual pawl* system, shown as element F in Fig. 2 and in more detail in Fig. 3, which contacts a set pin until released during the operating sequence. The rotation increment through which the pinwheel travels after release is proportional to the number of basic units assigned to the character being printed, and is established during each cycle by selectively fixing the distance between set pins. The fixing, or pin selection, mechanism consists of a pair of bars (Fig. 3) called pin set and clear bars, and a set of six selectable interposers (Element G in Fig. 2). When the pinwheel is at rest, tabs on the bars and interposers align with the nine adjacent pins nearest (but not including) the pin being held by the pawls. A solid tab on the pin clear bar aligns with pins 1 and 2 (pin 0 is the one held by the pawls) and will insure that these pins are in the "clear" state each cycle by moving either or both to the right if either (or both) is in the "set" state. The six interposers will clear any of the pins 3 through 8 that are in the "set" state for a nine-unit escapement. If the escapement required is less than nine units, one of the interposers will be selected (unlatched by the keyboard); the position of its tabs will be reversed and it will either set the pin with which it is aligned or maintain it in the "set" state. Thus, when a character is selected requiring n units of escapement ($3 \leq n \leq 9$), the nth pin is set (or maintained in the set state) and all others between 0 and the set pin are moved to (or maintained in) the clear state. After the character is printed, the pawls are withdrawn and the wheel is allowed to rotate. As soon as the 0'th pin passes under the pawls, the escape pawl re-engages in the latched position, thus catching the nth pin and bringing the system to rest. The rotation is just the number of units from three through nine that corresponds to n. The sequence is repeated for the next character of space, and so on consecutively until the carrier reaches the right margin* or until a special operation (such as tab or backspace) is initiated by the operator. Figure 4 shows the timing of two consecutive cycles, a 9-unit escapement followed by one of 3 units. Table 1 gives a character example for each escapement value in the 1/72 inch size group.

* The dual pawl system is required in order to control the escapement mechanism properly during operations other than normal escapement, e.g., backspace, tab and carrier return operations, etc. This design will be discussed later.

* If the copy being printed is later to be justified, the carrier is not allowed to advance to the right margin stop. Instead, the typing line is ended when the carrier enters a 3/4 inch region at the end of the line called the justification zone. The point at which the carrier actually stops corresponds to the first word ending or hyphenation point that lies within the zone. This is discussed more completely in another paper.[2]

Table 1 Examples of proportional character widths.

Units	*Carrier escapement (inches)*	*Example of character*
3	3/72	i
4	4/72	t
5	5/72	a
6	6/72	8
7	7/72	E
8	8/72	w
9	9/72	m

3. Print-point stability

The need for both high speed and print-point accuracy and stability was the most critical factor in the entire escapement design scheme. The successful employment of the rotating-element concept depended completely upon the assumption that problems in this area could be solved. If, for example, the inertia and friction within the system were too great, an acceptable repetition rate would not be possible. If the driving force could not be kept both constant and easily controlled (i.e., easy to "turn on and off"), the timing of the escapement and print cycles would either be inaccurate or require too much power to control. If latching and release were not positive, start-up might occur too soon in a cycle and position accuracy at the latch point would be doubtful.

To meet these objectives, both mechanical innovation and analytical study were required. The mechanical elements that were required are described below, and later sections will treat the problems of work and inertia, timing, control, and analytical design of the escapement pawls.

Low inertia and constant force—To minimize the inertial loads in the system, the leadscrew was designed as a hollow tube and the pinwheel and gear train were made as light as possible within the wear and strength requirements. The clockspring, shown in Fig. 5, was chosen as the source of leadscrew driving torque because a fully-wound clockspring offers a nearly constant torque output. With the constant torque input, the leadscrew system has a constantly accelerated motion until it is stopped by the pawls.

Latching and release—The system is latched and released at the pinwheel by a unique, dual-pawl control mechanism, shown in Fig. 3. The closer pawl in Fig. 3, called the escapement pawl, senses and latches only set pins, controls the pinwheel during normal escapement operations, and is the primary load-carrying pawl. The other pawl, called the backspace holding pawl, senses and latches both cleared and set pins, but bears a load only when a set pin is not present in the 0'th position. (This loading occurs after single-unit backspace operations and may occur at the end of a homing cycle. Both backspace and homing are discussed later.)

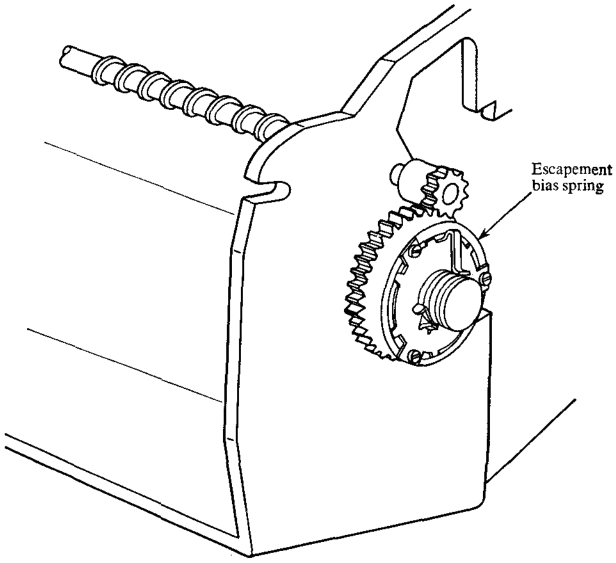


Figure 5 Escapement bias mechanism.

Rebound governor—If, after being stopped abruptly by the action of the pawls, the rotational elements could rebound far enough to cause one or more pins to slide under the backspace pawl as the pinwheel moves in the reverse direction, a misalignment would result. The pins would simply ratchet under the pawls during rebound, but once the rebound was dissipated and the system became biased once again by the clockspring torque, the forward-direction latching action of the pawls could latch the pinwheel out of position. The clockspring torque, which is opposite to the rebound, is not high enough by itself to hold the rebound below a one-unit (one-pin displacement) minimum. To increase the torque that opposes rebound, the rebound governor[3] shown in Fig. 6 was added to the system. This device consists of a special clutch attached to the leadscrew shaft that couples the inertia of another mass to the leadscrew. The coupling action is effected only when the velocity of the leadscrew is in the reverse, or rebound, direction, because the clutch is unidirectional. The leadscrew thus "feels" the extra inertia and transmits it by the gear train to the pinwheel, only when it is being driven in reverse. The extra inertia is sufficient to minimize rebound, but it can easily be overcome by the power-driven backspace mechanism.

4. Backspace

The backspace function[4] is provided by the mechanism shown in Fig. 7. It consists of a ratchet wheel A mounted on the same shaft as the pinwheel, a driving mechanism B that is connected to the backspace controls on the keyboard, and the backspace holding pawl C. The teeth on the wheel serve to rotate the pinwheel in the reverse direction. A signal from the keyboard causes the drive mechanism to move just

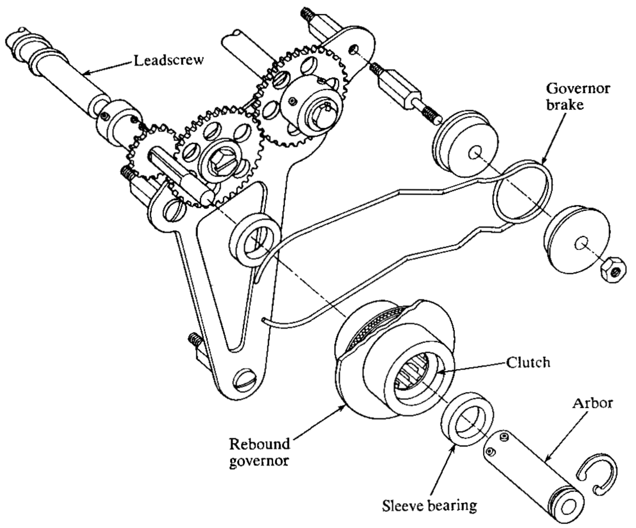
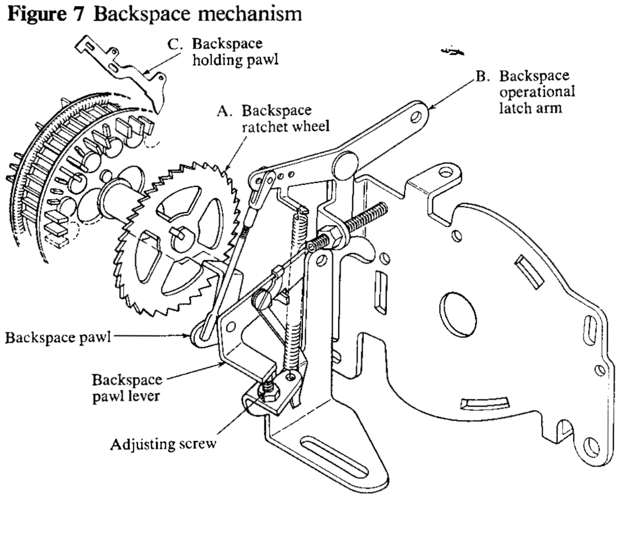


Figure 6 Rebound governor.

Figure 7 Backspace mechanism



enough to rotate the pinwheel shaft one unit per cycle in the reverse direction. As the cycle continues, one pin passes under the holding pawl, which rises against the spring tension and then drops back into the pinwheel to hold it in the new position. At the end of the cycle the drive lever is reset. Extra pressure on the backspace key sends an additional signal to the drive mechanism that causes it to repeat the cycle continuously until the pressure is released, thus providing a repeat mode for the system. Finally, a character-by-character proportional backspace, or "memory" mode, is provided. One input from the memory backspace key causes the system to repeat backspace until the pawls sense and latch upon the next *set* pin in the backspace direction. The reader will recall that the distance between set pins is equal to the unit length of the corresponding character. The backspace is thus exactly proportional, a "memory" function is provided by the pinwheel (the memory "length" is obviously one revolution of the pinwheel, or 60 units), and backspace can continue in a character-by-character mode until the pinwheel completes one revolution.

5. Large carrier displacement: homing and the homing mechanism.
The use of a leadscrew, which provided proportional escapement capability, required an additional mechanism to return the leadscrew to a specific rotational orientation, or home position, to establish even margins after carrier return or tab operations.

Returning the leadscrew to the home position is accomplished by a linkage that lifts both pawls out of the pinwheel and simultaneously rotates a pawl into the path of a stop on the gear changer shaft that corresponds to home position on the leadscrew. As the homing pawl is contacted by the stop, the linkage is unlatched and the escape pawls reenter the pinwheel as the homing pawl releases the homing stop.

If the leadscrew is at home when a carrier return or tabulation operation is signaled from the keyboard, the homing pawl will come down on top of the stop and a spring relief will prevent the pawls from being lifted out of the pinwheel. Repeat homing cycles, which could run down the escapement bias spring (which receives input only during escapement cycles), are thus prevented.

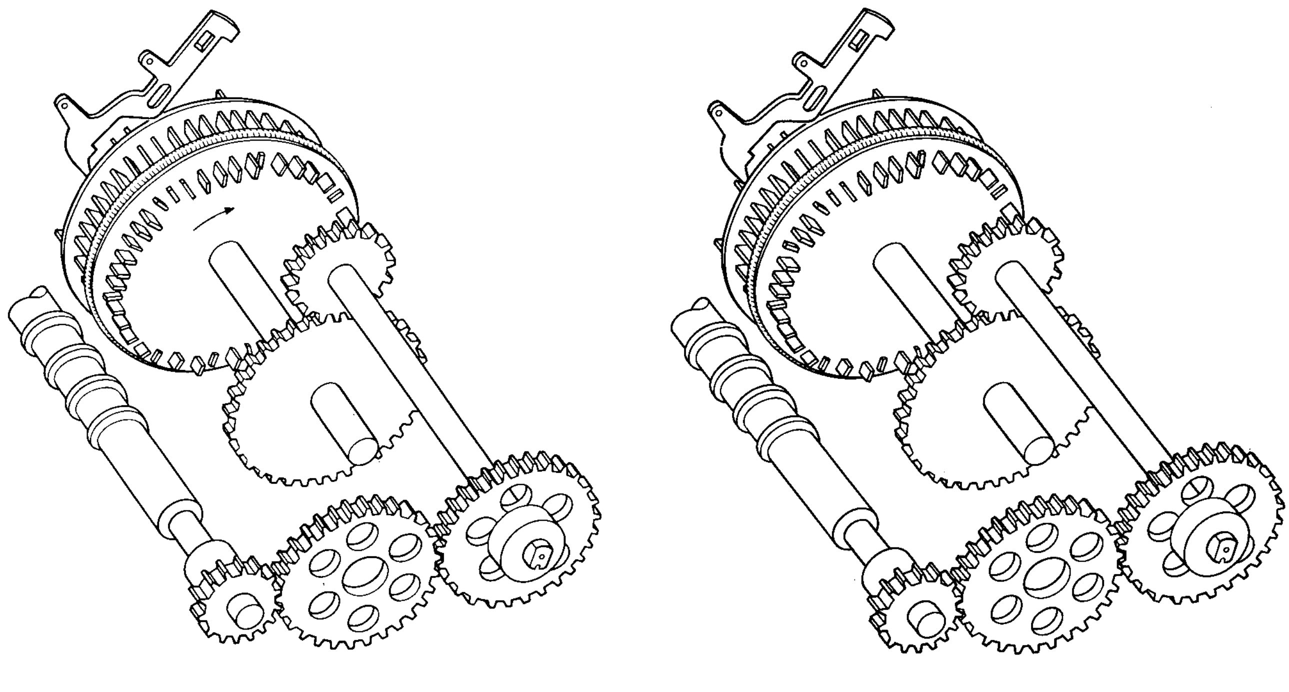
Analytical design

Having qualitatively described the principal escapement components, we present here a summary of the more significant analytical work. This work was begun early in the development program to test the feasibility of the rotating-element approach, confirm certain empirical assumptions, and reduce the need for prototype testing. It became, in effect, the controlling standard for the mechanical design.

Our intent, primarily, is to define the relationship of dynamic parameters for the entire escapement system. There is, however, one area of particular interest: The interaction of the backspace and homing operations with the basic escapement function created a problem of timing and control in the design of the escapement pawls. This led to a derivative study of the pawl mechanism and a subsequent redesign. The study has been included as an appendix to this summary.

- *System Dynamics*

The objectives established as guiding criteria were
- minimum work to wind the clockspring,
- minimum impact force,
- adequate strength of the parts, considering both wear and fatigue,
- a predictable, controlled escapement time, and
- minimum pinwheel rebound. (A one-unit rebound will cause an error.)



(a)

(b)

Figure 8 Schematic of escapement system (*a*) in motion and (*b*) stopped by pawl.

To implement these objectives, with a minimum of cut and try, the controlling parameters of the dynamic system were determined. The time for the leadscrew to escape a given displacement is completely controlled by the inertia of the system and the net torque (spring torque less frictional drag torque). The escapement time is expressed by the following constant acceleration equation:

$$t = \left[\frac{2 I_E \theta}{T_s - T_f}\right]^{1/2}, \tag{1}$$

where

θ = total displacement,
T_s = torque output of the clockspring,
T_f = frictional drag torque referred to the clockspring shaft, and
I_E = the equivalent moment of inertia of the rotating parts, referred to the clockspring shaft.

Examination of Eq. (1) reveals that the objective of minimum work to wind the spring may be accomplished by minimizing the frictional torque T_f and the equivalent inertia I_E of the system. By making I_E and T_f as small as possible, the spring torque T_s may also be held to a minimum, which accomplishes the objective of minimum winding work.

Minimizing the drag torque was accomplished by using permanently lubricated, shielded ball bearings throughout the system. In order to make the leadscrew bearing friction independent of temperature, a synthetic oil of extremely low viscosity was chosen for these small bearings.

Minimizing the total equivalent inertia was done by examining the primary contributors to the total. The total *equivalent* moment of inertia is not equal to the sum of the moments of the individual parts. The parts that rotate at the highest speeds have the largest equivalent moments. (For example, the equivalent leadscrew moment of inertia referred to the pinwheel shaft through one of the three gear ratios is twenty-five times as large as the leadscrew moment of inertia alone because the leadscrew rotates five times as fast as the pinwheel. Therefore, reducing the leadscrew inertia by one unit is twenty-five times as effective in reducing the total equivalent inertia as reducing the pinwheel inertia by one unit (neglecting friction). This factor guided the design of the leadscrew so that it has a minimum tooth profile size (that is, the minimum possible for adequate strength) and is a tube rather than a solid shaft. The use of Eq. (1) also enabled the designers to evaluate the effect of part changes without requiring models to be built to demonstrate the change.

The control of the escapement time is essential to the proper operation of the Composer. Again, Eq. (1) was used to place specifications on the controlling torque parameters so that the range of escapement times would be known. The flat spring-rate characteristic of the clockspring enabled the driving torque to be set within 6% of the nominal value. The ball bearings enable the frictional drag of the rotating system to be so small that unwanted gear binds caused by off specification parts or improper assembly could be detected and eliminated during assembly.

When a machine is assembled within these torque specifications, the escapement timing is within specification because all values of the controlling parameters in Eq. (1) are fixed.

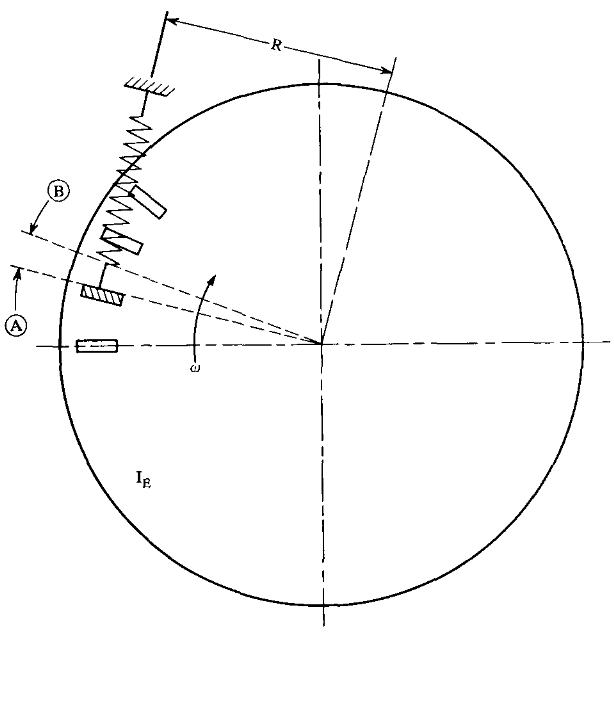
In order to design the parts for adequate strength, the maximum loads had to be determined. An approximate mathematical model was developed to relate the variables. For example, the maximum force on the pins occurs at the end of a 9-unit escapement when a pin impacts the pawl and stops the system. Figure 8a illustrates the position of the system just before impact. Figure 8b shows the pawl just as the system is stopped and the maximum impact force is generated.

The maximum impact force was predicted by the use of the angular momentum relationship. By making two approximations Fig. 8b was converted to Fig. 9, from which a mathematical model was derived, and I_E was defined as the equivalent moment of inertia of the system referred to the pinwheel shaft. The two simplifying approximations were:

1. All of the inertia of the system may be transferred to the pinwheel shaft, and

2. the force that stops the system is linearly related to the displacement θ of the pinwheel during stopping.

Figure 9 Equivalent dynamic system for prediction of impact force.



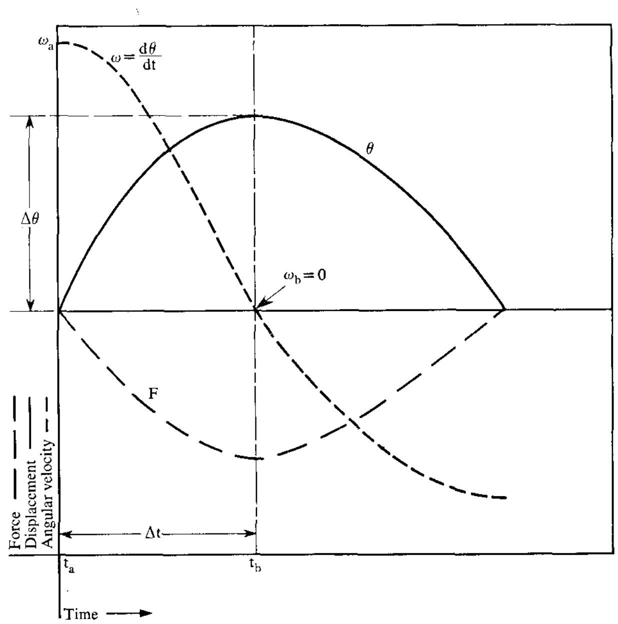


Figure 10 Curves for displacement, velocity and force during impact.

The two assumptions imply the relationships shown in Fig. 10. These relationships were derived by the application of the principle of angular momentum to the system shown in Fig. 8b. The equation form is

$$\left[\Sigma M = \frac{d}{dt}\,(I_E\omega)\right];$$

where $\Sigma M \equiv$ summation of external moments and
$I_E\omega \equiv$ angular momentum of system.

$$-FR = I_E\frac{d\omega}{dt} = I_E\frac{d^2\theta}{dt^2}, \qquad (2)$$

or $$\frac{d^2\theta}{dt^2} + \frac{FR}{I_E} = 0,$$

in which

θ = angular displacement of the pinwheel,
F = impact force,
R = radial distance to the pin,
I_E = total equivalent moment of inertia of the system, and
ω = $d\theta/dt$, the angular velocity.

From assumption (2), the force is proportional to the displacement, i.e.,

$$F = kR\theta. \qquad (3)$$

By combining (2) and (3), the equation below can be derived:

$$\frac{d^2\theta}{dt^2} + \frac{kR^2}{I_E}\theta = 0. \qquad (4)$$

Equation (4) is the equation of motion during impact. The relationships for displacement, velocity, and force shown in Fig. 10 are the solutions to Eqs. (3) and (4). By applying the principle of angular momentum in a different manner, the important relationship between maximum force, maximum velocity and inertia can be derived:

$$\Sigma M = \frac{d}{dt}\,(I_E\omega)\,,$$

$$-FR = I_E\frac{d\omega}{dt}\,, \quad \text{or}$$

$$d\omega = -\frac{FR}{I_E}\,dt\,. \tag{5}$$

Integrating over the time Δt for which the velocity goes to zero,

$$\int_{\omega}^{0} d\omega = -\int_{0}^{\Delta t}\frac{FR}{I_E}\,dt\,, \quad \text{or}$$

$$\omega_a = \frac{R}{I_E}\int_{0}^{\Delta t} F\,dt\,.$$

Combining (5) with

$$F = F_{\max}\sin\omega_n t \tag{6}$$

and integrating yields

$$F_{\max} = \frac{I_E\omega_a}{2R\Delta t}\,. \tag{7}$$

Equation (7) describes the parameters which control the maximum impact force.

The maximum force can be measured indirectly by the use of Eq. (7) simply by attaching a displacement potentiometer (or rotary velocity transducer) to the pinwheel shaft and determining the values of time Δt, and maximum angular velocity ω_a experimentally. By combining the information from Eq. (7) and the constant acceleration relationships of the escapement motion, the impact force for all escapement times or displacements can be determined. Another form of Eq. (7) is

$$F_{\max} = \frac{\omega_a}{\pi}\,(k\,I_E)^{1/2}\,, \tag{8}$$

which shows the effect of stiffness and system inertia on the maximum impact force.

Although Eqs. (7) and (8) are subject to the limitations of the two assumptions previously listed and therefore are not exact, they are accurate enough to serve a useful engineering purpose in relating the variables. Strain gages applied to the pawl were used to measure the force accurately and then Eqs. (7) and (8) were used to relate force and velocity.

With the knowledge of the maximum force as a function of velocity, components such as gears and pins could be designed to have adequate strength in fatigue and resistance to wear. (The gears, for example, were designed for the maximum ratio of strength to inertia. They are made of steel hardened to RC 60–65 with a width which is sufficient for adequate strength and proper mesh.)

It is evident, in conclusion, that the design and operation of the escapement system to meet the previously stated goals was greatly facilitated by the use of some relatively simple mathematical relationships to describe the controlling parameters of the system.

• *Mechanical development of the escapement pawls*

The rotational escapement system designed for the Composer required a pawl mechanism that would allow the rotational elements to advance or "escape" reliably, but would hold the rotational elements in the proper position after each backspace input of one unit. It was also necessary that this pawl mechanism permit the system to advance during a homing operation but hold the rotational elements in the proper position after each homing cycle, without regard to whether a set pin or a cleared pin would be located in the position corresponding to the pawl. This pawl system, then, had to contain a pawl that would hold on any pin whether set or cleared. With this pinwheel-pawl escapement system, pins are moved laterally in the pinwheel and detented in place as required. "Set" pins are defined as those that are positioned to be contacted by the escape pawl. "Cleared" pins are defined as those that are positioned not to be contacted by the escape pawl. The pawl system must therefore also have an actuating linkage that will remove the pawl from the path of the pins during escapement or homing, and a sensing device to allow the pawl to re-engage on the proper pin after an escape or homing operation.

Two-pawl arrangement—Early in the development program, a two-pawl system was adopted: one pawl for all pins, set or cleared, called the holding pawl; and one for set pins called the escape pawl, together with the necessary actuating linkages. It had been realized previously that the escape pawl system requirements could be satisfied with a single pawl and a set-pin sensor. The mechanism dynamics, however, made it more feasible to retain the set-pin sensor as the prime load-carrying pawl.

The two-pawl system as originally designed is shown in Fig. 11a and the sequence of operation is described in Figs. 11b–11e. Operations not shown in Fig. 11 are described as follows:

(1) The trip lever (shown in Figure 11) which has a slot similar to that of the escape pawl is spring-biased downward as well as in the restoring direction. This is done so that if the restore motion is not complete when the escape pawl slides back to its original position (Fig. 11e), the escape pawl can lift the trip lever without damage (Fig. 11f). The trip lever can then reset when the restore motion is completed (Fig. 11e).

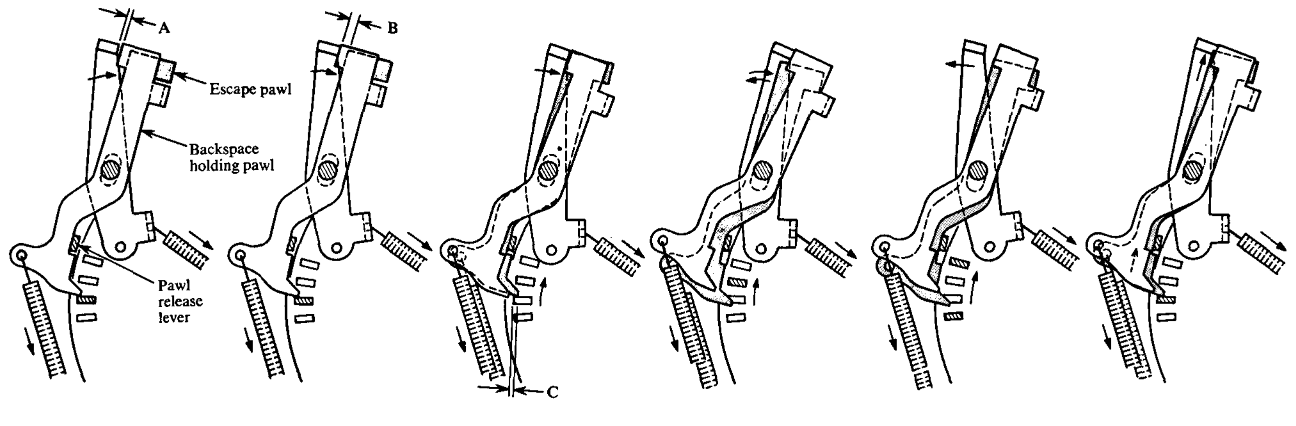


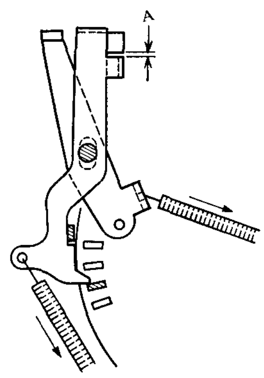
Figure 11 (*a*) Original two-pawl design. In operation, trip lever and holding pawl move (*b*) through distance B; then escape pawl begins to move also. As both pawls lift clear, pinwheel rotates in escape direction (*c*), and escape pawl slides forward and rotates downward into path of next set pin. Trip lever and holding pawl continue to end of linkage motion (*d*), then begin to restore. Holding pawl is stopped and latched out of pinwheel path when latch surface contacts rear of escape pawl (*e*). In (*f*), escape pawl is contacted by set pin and slides backward to limit of pivot slot. Holding pawl is thus released and drops back into pinwheel.

(2) The pawl release lever (Fig. 11a) has two functions:
(a) to act as a down-stop for the pawls, whereby the pawl engagement with the pins can be adjusted; and
(b) to lift both pawls out of the path of the pins to allow the system to advance during homing.

(3) During backspace, the rotational elements are moved one unit each cycle, in the direction opposite to escapement, and the pawls simply ratchet over pins as necessary.

The first problem with two pawls—This initial set of escape pawls was functionally satisfactory. However, from Figs. 11b and 11c, it may be seen that the escapement system is released at different times, depending upon whether it is held by the holding pawl or the escape pawl. In order to insure a stationary escapement system at all times while the machine is printing, the earliest escapement can begin only when print is completed. The other escapements would then begin later and the time for the motion required by the function shown in Fig. 11c is in effect subtracted from the total time available for escapement. Because of the many events in the cycle, escapement time is at a premium. Therefore, to regain the time lost by the motion described in Fig. 11c the pawl geometry was changed as shown in Fig. 12. The sequence of operation is much the same with this configuration except that both pawls begin to move at the same time and release the system at the same time. Note that to operate this set of pawls, it was necessary to increase the clearance at A in Fig. 12 so that after the pawls have released the pin, the escape pawl can move forward sufficiently to be disengaged from the trip lever without interference from the top of the holding pawl latch surface. After disengaging from the trip lever, the escape pawl will

Figure 12 Redesigned two-pawl system.



both rotate back into the path of the pins and continue forward for the length of its slot. This will accomplish the latching function and the cycle will move on to completion.

The second problem with two pawls—As machine development progressed, this set of pawls was found to have two unsatisfactory characteristics. The first was a result of the increased clearance between the latching surfaces (at A in Fig. 12) on the escape and holding pawls. As described in the sequence of operations, the pin driving the escape pawl along its axis would unlatch the backspace holding pawl. The increased clearance resulted in the holding pawl unlatching early, and allowed the holding pawl either (1) to drop into the path of the pins sufficiently to absorb part of the impact, with only a fraction of the designed pin engagement; or (2) to drop on the pin during the time of peak impact loading when the pin was slightly bowed (Figs. 13a and 13b). Both of these conditions placed a high stress on the

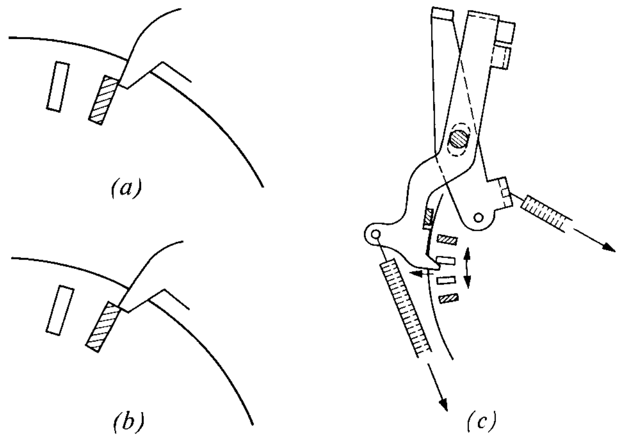


Figure 13 *(a)* and *(b)* Two failure-inducing wear conditions caused by early release of holding pawl. *(c)* Latch-out caused by oscillation.

sharp corners of the pin and the pawl, causing wear sufficient to create operational failures.

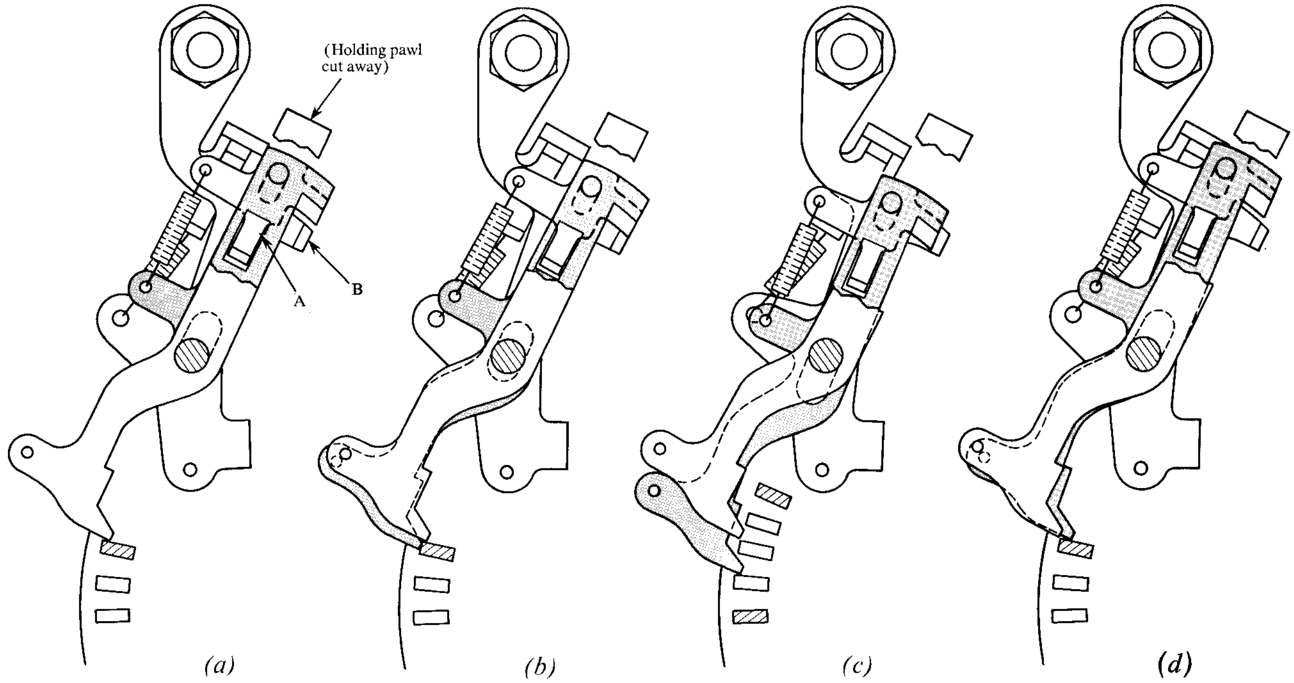
The second unsatisfactory characteristic was also inherent in previous designs of the pawls but the dynamics of the rotational elements of the escape system had not been sufficiently refined to demonstrate the problem. At the end of a homing operation the system is stopped initially by the homing pawl. As the homing pawl releases the system, the escape pawls are lowered back into the path of the pins. Depending on the displacement and the impact at the end of the homing cycle, the pinwheel could oscillate through mechanical clearances at the time the pawls were reentering the pinwheel to reengage the pins. The timing of this action occasionally allowed the backspace holding pawl to be struck from behind by a pin, thereby rotating it outward relative to the escape pawl. This would result in a latched-out condition equivalent to escapement. The system would not stop on the proper pin, then, but would advance in an escape mode to the next set pin.

- *The final design*

The final escape pawl configuration eliminated the two unsatisfactory characteristics discussed above. From Fig. 14a it may be seen that a small sliding latch A has been added to the escape pawl. This latch will collapse upward with rerespect to the escape pawl when the escape pawl moves forward after releasing the pinwheel. This action will allow the escape pawl to disengage from the trip lever without interfering with the holding pawl latch surface (Fig. 14b). After it is disengaged, it will rotate back into the path of the pins and continue forward to the end of its slot. At the same time, the spring on the sliding latch will restore the latch forward with respect to the escape pawl. When the escape pawl is moved upward by the advancing set pin, the holding pawl unlatching will now be delayed sufficiently to avoid impact damage because the latch surface clearance (at B in Fig. 14a) can be kept very small.

Again referring to Fig. 14b we observe that the holding pawl latch surface has been moved from the holding pawl to the trip lever. With this design the trip lever is latched in a partially actuated position by the escape pawl and thereby holds the holding pawl out of the path of the pins until both

Figure 14 Final design showing *(a)* sliding latch on escapement pawl and *(b)*–*(d)* operating sequence.



are released by the escape pawl (Fig. 14c). Since the holding pawl has no latch surface, it will quickly return to its proper position if it is bumped on the reverse side by an oscillating pin at the end of a homing operation.

Latching out the trip lever as mentioned above, however, does mean that the escape pawl will restore under the trip lever actuating surface on all escapements (Fig. 14d). Previously, this was accommodated without damage by a slot in the trip lever which allowed it to move upward until restored to the rest position. Since this would now require the latch surface on the trip lever also to move upward, the slot was removed and the trip lever actuating arm was hinged to the trip lever with a downward spring bias. This allows the actuating arm to swing upward until the trip lever and the latch surface restore, at which time the arm will reset to the actuating position.

This set of pawls now satisfies the original design requirements. That is, when cycled by the trip lever, they will allow the rotational elements of the escape system to advance from any pin to the next set pin. When the system is rotated backward, one pin at a time, the holding pawl will ratchet over each pin and hold the system at that point. Finally, when the pawls are actuated by the homing mechanism through the release lever, they will allow the system to advance to home and will reenter the pinwheel and hold the system in the proper position when the operation is completed.

Conclusions

The rotating-element escapement has made it possible to add reliable proportional-character capability to the multifont SELECTRIC Typewriter printing system. The original requirements for both variable type style and variable type size have thus been realized without sacrificing speed and reliability.

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