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The IBM SELECTRIC Composer

Multiple Index Mechanism

Abstract: When high-quality printed copy is desired, the ability to use variable styles of type in one printing system is a fundamental requirement. This feature has existed to a limited extent for some time in low-cost, cold-type printing equipment and typewriters, and has been notably realized in the changeable typehead of the IBM SELECTRIC Typewriter. The ability to vary type *size*, however, is rarely found in such systems and has been restricted for the most part to commercial hot-type equipment or to manual methods. To remove this restriction in the IBM SELECTRIC Composer, it was necessary to design an index mechanism which would provide variable line-to-line increments. Increment selection was to be flexible enough to accommodate several sizes of type, but it had also to be reliable and easily controlled by the operator. The design which has resulted and is described in this paper is a dual-ratchet, planetary gear system. Platen increments of from 5 to 20 "points" can be selected, and the platen can be indexed manually one point at a time. Other features are also provided, including a "carriage-return, no-index" which permits changing styles on the same line of type without manual "rollback."

Introduction

Because the character size in the IBM SELECTRIC Composer can be changed both horizontally and vertically by changing type elements, the horizontal escapement and vertical line spacing must be changeable to correctly space a given type size. The second of these functions, changeable vertical spacing, required a system that would permit the platen index (increment of rotation that is taken with each carriage return) to be selectively variable.

This paper describes that system, beginning with a definition of the mechanical specifications which became the design requirements. The design itself is then discussed in terms of each specification; alternatives that were considered are reviewed, and the present system is explained.

Design requirements

The mechanical requirements for variable index were new, and before work could begin on the design itself it was necessary to define these more specifically. Eight principal objectives were identified in all, and these are listed below.

Indexing must be "automatic" so that the operation will be familiar to electric typewriter operators and necessary training will be minimized.

The line-space increments obtainable with this system must include all standard spacings used in IBM typewriters, plus any special sizes required by the printing industry.

The tolerance between lines of type must be no more than ± 0.003 *inch* or the variation becomes obvious.

The system must be manually controllable in one-point (1/72 inch) increments for special applications, such as subscripts and superscripts.

The system must have a platen variable (i.e., a means of controllably turning the platen and paper through a small increment) comparable to, or better than the 0.025 inch paper adjustment obtained with an IBM SELECTRIC Typewriter.

The index increment selection must be simple so that the operator can be easily trained in its use, and the function of selection does not become a time-consuming task.

Rollback of the paper must be simple and repeatable, mainly in making corrections on the line being typed. Every attempt must be made to attain accurate rollback alignment to previous lines; however, printers, in general, do not use this technique but strip out and insert entire lines in order to retain high quality character alignment.

The system must incorporate some simplified means for an automatic carriage return operation without indexing the platen so that the operator can return to areas of the line just typed for corrections, insertions, or mixing type styles within the same line.

Because the design described here provides a large number of index increments with considerably fewer parts than might normally be used, the design philosophy developed may be of use in other applications where one input selec-

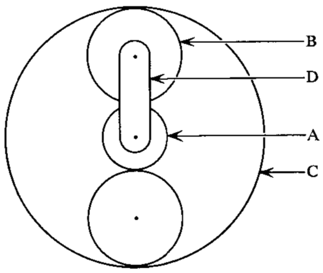
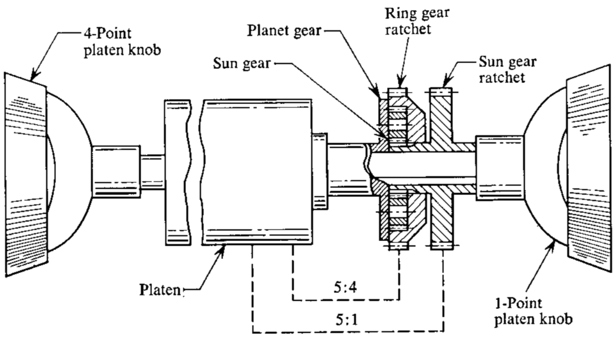


Figure 1 Schematic of planetary gear system.

Figure 2 Schematic of indexing system showing division of manual input function into 4-point (left knob) and 1 point (right knob) increments.



tively provides a multiplicity of outputs. More detailed considerations in this design are examined in the following sections in the sequence of design requirements described above.

Automatic indexing

The very nature of the IBM SELECTRIC Composer made "automatic" indexing a prime requirement because all operational keys on this machine follow the "automatic" philosophy of its predecessor, the SELECTRIC Typewriter. The "automatic" index feature is the same as that in all IBM electric typewriters, in that depresssion of the "Index" or the "Carriage Return" keybutton produces a vertical indexing of the paper.

Line-spacing variation

The original specification for line-space variation called for line spacing of 4, 6, 7.2, 8, 9, and 12 lines per inch (i.e., increments of 0.25, 0.167, 0.139, 0.125, 0.111, and 0.0833 inch, which included the majority of IBM typewriter index increments) with the capability of manually "leading" (adding small increments to the basic increment) in increment sizes of one "point" (1/72 inch or 0.0139 inch), to as much as 3-point "leading" per line for the convenience of printers.

Table 1 Rotation analysis for platen; ring gear rotated with sun gear fixed.

	A	B	C	D
Gears locked and system rotated clockwise one rev.	$+1$	$+1$	$+1$	$+1$
D fixed and A rotated one rev. counterclockwise	-1	$(-1)\dfrac{(-A)}{(+B)}$	$(-1)\dfrac{(-A)(+B)}{(+B)(+C)}$	0
Summation	0	$1 + A/B$	$1 + A/C$	1

Table 2 Rotation analysis for platen; sun gear rotated with ring gear fixed.

	A	B	C	D
Gears locked and system rotated clockwise one rev.	1	1	1	1
D fixed and C rotated one rev. counterclockwise	$(-1)\dfrac{(-C)(-B)}{(-B)(+A)}$	$(-1)\dfrac{(-C)}{(-B)}$	-1	0
Summation	$1 + C/A$	$1 - C/B$	0	1

Present system: A planetary gear system coupled to two ratchets and automatically indexed—Once the line-space philosophy was oriented in terms of the printer's "point," the most promising solution appeared to be the use of a planetary gear system coupled to two ratchets through the ring gear and sun gear, and indexed simultaneously by two index pawls driven by a pawl carrier mechanism identical to that of the SELECTRIC Typewriter. The planetary system is shown schematically in Fig. 1; a physical representation is shown in Fig. 2. A is a sun gear attached to a ratchet, B is one of the two planet gears, C is a ring gear attached to a ratchet, and D is the platen or driven member.

- *Analysis*

Table 1 shows the results of a rotational analysis of the platen (D) for the case in which the ring gear (C) is rotated and the sun gear (A) is fixed. The summation shows that when the sun gear (A) is fixed, the angular rotation θ_{DC} of the platen is

$$\theta_{DC} = \frac{\theta_C}{1 + (d_A/d_C)},$$

where

θ_C is the angular rotation of the ring gear,
d_A is the diameter of the sun gear, and
d_C is the diameter of the ring gear.

Similarly, Table 2 shows the rotation θ_{DA} of platen (D) when the sun gear (A) is rotated and the ring gear (C) is fixed.

Table 3 Possible index combinations for a planetary system

System no.	1		2		3		4		5		6		7		8	
½ pts./rev.	None		None		None		660		None		660		630		None	
Pts./rev.	330		328		330		330		330		330		315		315	
Gear ratio	3 : 1		4 : 1		5 : 1		5 : 1		6 : 1		6 : 1		7 : 1		7 : 1	
Ratchet teeth	110		82		66		132		55		110		90		45	
	L.H.	R.H.	L.H.	R.H.	L.H.	R.H.	L.H.	R.H.	L.H.	R.H.	L.H.	R.H.	L.H.	R.H.	L.H.	R.H.
Pts. tooth	2	1	3	1	4	1	2	½	5	1	2½	½	3	½		
Point size	*Number of teeth indexed with each ratchet*															
3	1	1					1	2			1	1				
3½							1	3			1	2	1	1		
4	1	2	1	1			1	4			1	3	1	2		
4½							2	1			1	4	1	3		
5	2	1	1	2	1	1	2	2			1	5	1	4		
5½							2	3			2	1	1	5		
6	2	2	1	3	1	2	2	4	1	1	2	2	1	6		
6½							3	1			2	3	2	1		
7	3	1	2	1	1	3	3	2	1	2	2	4	2	2	1	1
7½							3	3			2	5	2	3		
8	3	2	2	2	1	4	3	4	1	3	3	1	2	4	1	2
8½							4	1			3	2	2	5		
9	4	1	2	3	2	1	4	2	1	4	3	3	2	6	1	3
9½							4	3			3	4	3	1		
10	4	2	3	1	2	2	4	4	1	5	3	5	3	2	1	4
10½							5	1			4	1	3	3		
11	5	1	3	2	2	3	5	2	2	1	4	2	3	4	1	5
11½							5	3			4	3	3	5		
12	5	2	3	3	2	4	5	4	2	2	4	4	3	6	1	6
12½							6	1			4	5	4	1		
13	5	3	4	1	3	1	6	2	2	3	5	1	4	2	2	1
13½							6	3			5	2	4	3		
14	5	4	4	2	3	2	6	4	2	4	5	3	4	4	2	2
14½							7	1			5	4	4	5		
15	5	5	4	3	3	3	7	2	2	5	5	5	4	6	2	3
15½							7	3					5	1		
16			5	1	3	4	7	4	3	1			5	2	2	4
16½							8	1					5	3		
17			5	2	4	1	8	2	3	2			5	4	2	5
17½							8	3					5	5		
18			5	3	4	2	8	4	3	3			5	6	2	6
18½							9	1					6	1		
19			5	4	4	3	9	2	3	4			6	2	3	1
19½							9	3					6	3		
20			5	5	4	4	9	4	3	5			6	4	3	2
20½													6	5		
21									4	1			6	6	3	3
21½																
22									4	2					3	4
22½																
23									4	3					3	5

When the ring gear (C) is fixed, the angular rotation, θ_{DA}, of the platen (D) is

$$\theta_{DA} = \frac{\theta_A}{1 + (d_C/d_A)}.$$

Thus, if both the ring gear (C) and sun gear (A) are rotated θ_C and θ_A, respectively, the total rotation of the platen is

$$\theta_{DT} = \theta_{DA} + \theta_{DC} = \frac{\theta_A}{1 + (d_A/d_C)} + \frac{\theta_C}{1 + (d_A/d_C)}.$$

After careful study of various gear ratios (shown in Table 3) a 5:1 gear ratio between the sun gear and platen appeared to give the best coverage of the point size range of interest (between 5 and 20 points) with the smallest number of ratchet teeth and only four-tooth selectivity in indexing. This also allows for a one-point index manually through the sun gear. It was desired to keep both ratchets the same diameter as in the SELECTRIC Typewriter, so the same cam follower mechanism could be used. Thus, the number of teeth on both ratchets was set at 66, and the rotation per tooth

Table 4 Spacing in lines per inch and ratchet combinations for various point sizes.

Point size	Number of teeth indexed: Ring gear ratchet	Number of teeth indexed: Sun gear ratchet	Lines/inch
5	1	1	14.4
6	1	2	12.0
7	1	3	10.3
8	1	4	9.0
9	2	1	8.0
10	2	2	7.2
11	2	3	6.54
12	2	4	6.0
13	3	1	5.54
14	3	2	5.15
15	3	3	4.8
16	3	4	4.5
17	4	1	4.24
18	4	2	4.0
19	4	3	3.79
20	4	4	3.6

of the sun gear and ring gear were equal, or

$$\theta_C = \theta_A .$$

Then the gear ratio between the sun gear and the ring gear is

$$d_C = 4d_A .$$

From Figure 3, $d_B = r_C - r_A$, or $2r_B = r_C - r_A$, and $d_C = 2d_B + d_A$. Also $r_D = r_B + r_A$, or $d_D = d_B + d_A$. During this phase it was necessary to study the platen circumference in order to arrive at a platen periphery which would contain a number of "points" divisible by the number of teeth chosen for the ratchets.

The following criteria were used in order to aid in the selection of the number of ratchet teeth and platen diameter:

(1) The platen diameter must not be changed from the 1.432 diameter used in the SELECTRIC Typewriter by more than ±0.040 inch. This is because a broader change would cause a major redesign in the paper-feed system. It was felt that the SELECTRIC Typewriter paper feed would be adequate, and that major changes in this area would not be feasible.

(2) The ratchet diameters were to be the same as the SELECTRIC Typewriter ratchet diameter, which is 1.187 inch. This permitted the use of the same pawl, pawl carrier, and operational cam geometry. It was felt that the dynamics of the new system could be predicted more accurately if the standard operational cam were retained.

Because of these factors, and with the use of Table 3, it was decided that:

(1) Both the left and right ratchets would have 66 teeth.

(2) For an output of 4 points from the ring gear-ratchet, and a 1-point output from the sun gear-ratchet, the platen periphery must contain 330 points. This dictated a platen

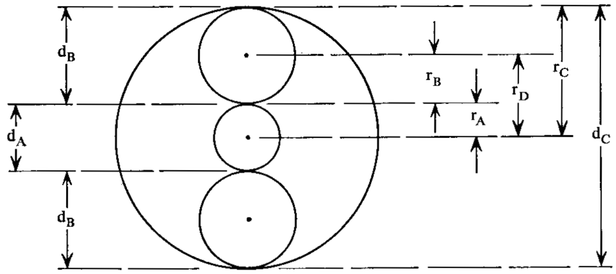
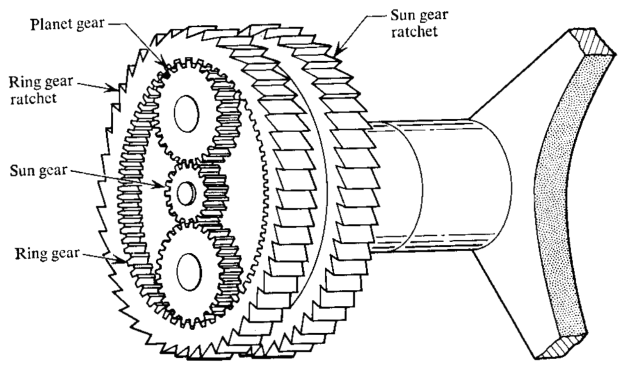


Figure 3 Relative dimensions used in rotational analysis of gear system.

Figure 4 Sketch of complete dual-ratchet planetary system.



circumference of 4.580 inch and a platen diameter of 1.458 inch; however, a standard paper thickness of 0.004 inch was later defined, and this reduced the platen radius by 0.004 inch, making the platen diameter 1.450 inch.

The system finally adopted has a maximum index ratchet rotation of 4 teeth on either ratchet, and a gear ratio of 4:1 between ratchets.

Note from Table 4 that for increments of less than 5 points, one ratchet will provide 4 points per tooth rotated, and the other will provide one point per tooth rotated. The platen diameter was changed from 1.432 ± 0.002 to 1.450 ± 0.002 for multiple indexing. From the platen centerline this amounts to only a 0.009 inch change of the platen radius, which causes only minor positioning adjustments from the platen configuration of the SELECTRIC Typewriter. It will be noted that in the sequence of index combinations, each ratchet is always indexed at least one tooth. This further limitation was set so that no "masking" device would be needed to keep the index pawls from entering the ratchet. Later in the development it was found that certain applications required a carriage return operation and *no* index. The masking idea was utilized successfully in this instance. (This will be discussed in more detail in another section of this paper.)

The planetary system is described as shown in Fig. 4 and contains a 56-tooth ring gear attached to a 66 tooth ratchet,

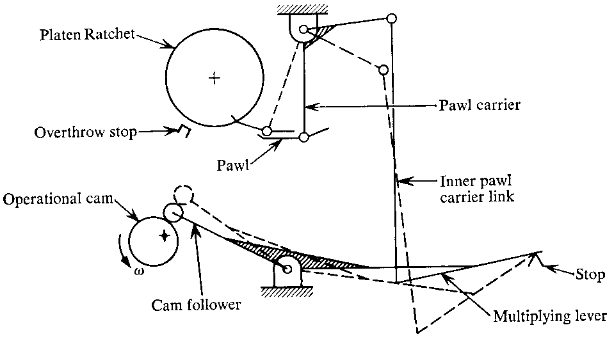
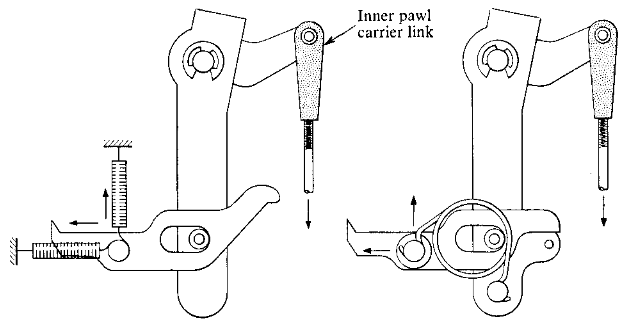


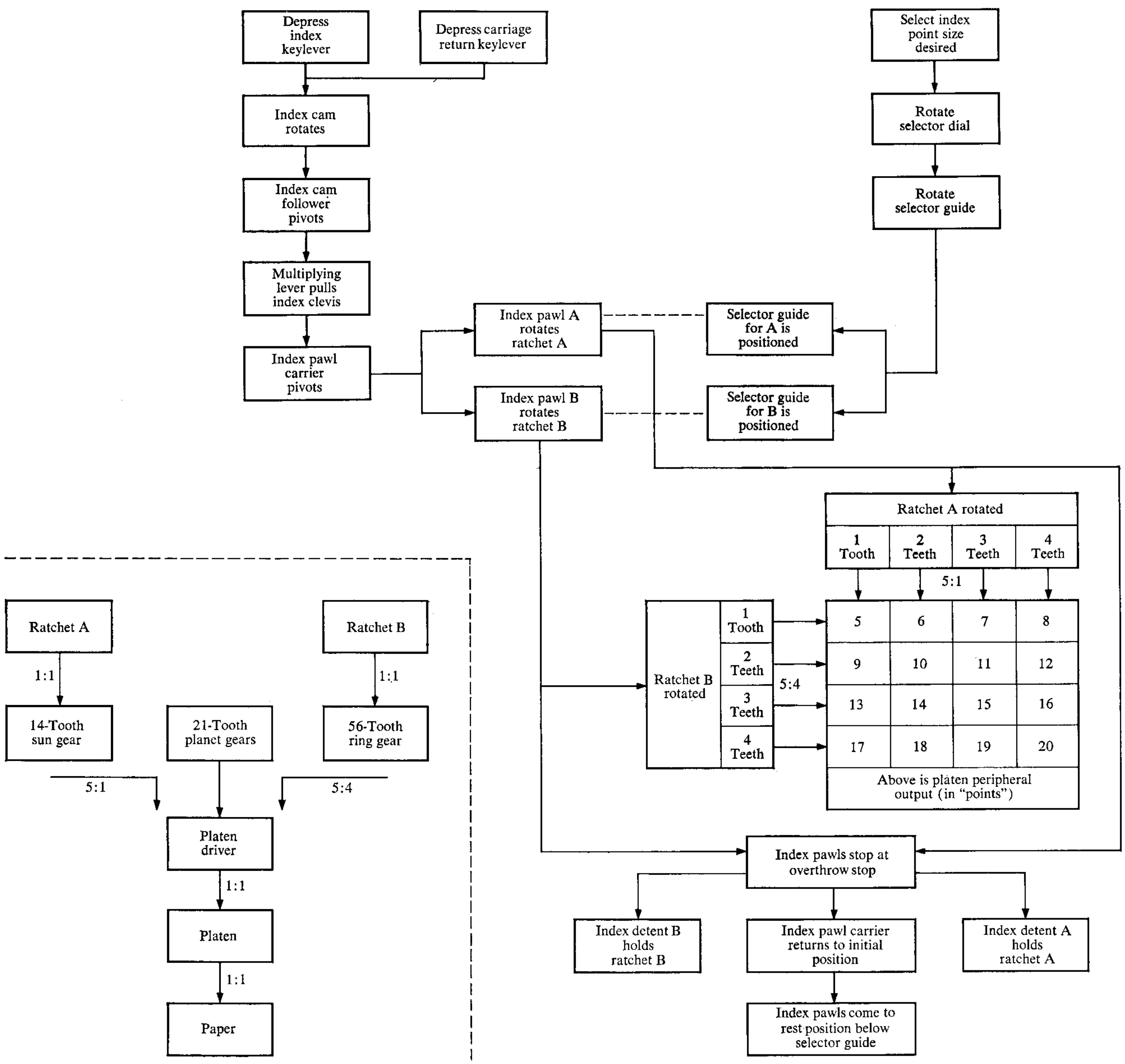
Figure 5 Schematic of actuating mechanism.



(*a*) SELECTRIC Typewriter (*b*) SELECTRIC Composer

Figure 6 Spring-loading designs for (*a*) SELECTRIC Typewriter and (*b*) SELECTRIC Composer index pawls

Figure 7 Operating sequence for multiple index system.



a 14-tooth sun gear attached to a 66-tooth ratchet, and two 21-tooth planet gears to drive the platen from either ratchet motion. This fixes the gear ratio at 4:1 between the ring gear and sun gear; 5:1 between the sun gear and platen; and 5:4 between the ring gear and platen. Notice that some means of selection is needed in order to permit each ratchet to index specific numbers of teeth in a particular mode. (The selection mechanism will be discussed.)

The number of teeth on the ratchet becomes an important parameter with respect to indexing adjustment and repeatability. As the number of teeth on the ratchet increases, the tolerances on the driving mechanism decrease proportionately. Also, the selector system for guiding the index pawls into the proper teeth becomes a more critical mechanism to adjust. The size of the ratchet teeth dictated the size of the detent radius used, and even with 66 teeth this detent radius is so small that a roller-type detent could not be easily designed. Thus a cam-type tip evolved, and widths of ratchets and detents became important in controlling contact stresses.

The dynamics of the multiple index system are quite similar to the SELECTRIC Typewriter index system. As in the SELECTRIC Typewriter, the index pawl mechanism is actuated from an operational cam follower (Fig. 5) by means of the index pawl carrier link connected to the front of the index multiplying lever. The rear of the multiplying lever is always in contact with the multiplying lever stop, which is adjustably attached to the power frame. As the operational cam is cycled, the multiplying lever pivots and pulls down on the index pawl carrier link. The index pawl carrier link always receives the same amount of motion each time it operates, regardless of the index mode selection. The amount of travel is sufficient to allow the index pawls to engage and rotate the ratchets a distance of up to four teeth.

Because the two index pawls are mounted on one index pawl carrier, a space problem developed early in the program. The SELECTRIC Typewriter uses two springs to load the index pawl; a vertical spring to force the pawl to engage the single-double throw cam in the rear of the pawl so that it guides smoothly into the correct ratchet tooth; and a horizontal spring to load the pawl forward on the pawl carrier, so that the pawl will follow the ratchet in case the ratchet tries to get ahead. This is shown in Fig. 6a.

In the multiple index system, a single hairpin spring for each pawl was developed to give the same loading as the vertical and horizontal springs on the SELECTRIC Typewriter. This is shown in Fig. 6b.

To halt the ratchets in a detent position and to dissipate some of the platen inertia one overthrow stop is used for both index pawls because of the constant throw of the system.

Two one-piece levers are used for detenting the ratchets; a detent tip is built into each lever. This technique has the advantage of maintaining the alignment between ratchets

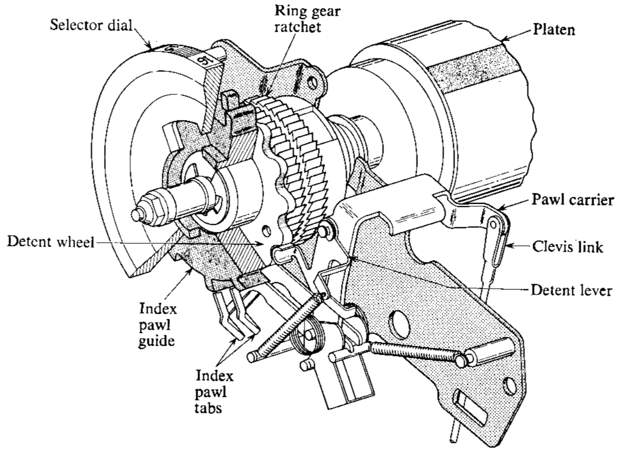


Figure 8 Selection and detent mechanisms.

much better than with other methods. Wear tests have proven these detents capable of many years of life without wear if lubricated.

Figure 7 shows, in schematic form, the index sequence of operation, and the means of obtaining the variable output required. Operating parts are shown in Fig. 8; refer also to Fig. 2.

It is worth noting that the German or British "point" differs from the American "point" (is longer by a factor of 1.062). To make this index mechanism compatible with the European system without a major change in the paper-feed characteristics, it is again required that the platen diameter be changed no more than ± 0.040 inch. However, a study of the system has shown that a simple change from a 66-tooth ratchet to a 62-tooth ratchet, while retaining the present platen diameter, will accomplish this end. The detent radii can be retained as now designed because the tooth-to-tooth span at a ratchet diameter of 1.187 inch is only changed by 0.0036 inch. The same is true for the index pawls and overthrow stop. The total throw (for a maximum four-tooth ratchet rotation) is only changed by 0.014 inch. However, it should be noted that the European platen will now contain only 310 points on the periphery, rather than the 330 obtained with the 66-tooth ratchet.

Line-spacing accuracy

Obviously, in a machine such as the SELECTRIC Composer, the line-space accuracy is of prime importance. The preliminary specifications stated that "line-to-line accuracy must be better than that of the SELECTRIC Typewriter." To be "better" than the SELECTRIC Typewriter it is necessary to know how good the SELECTRIC Typewriter is. Experimental data taken from new, well-adjusted SELECTRIC Typewriters show that line-space variation between two successive lines is ± 0.004 inch. Such a variation with a ratchet that is fixed directly to the platen was not understood at first, but further investigation revealed that the detent roller's "seated" position in a ratchet tooth varied from cycle to cycle, from one side of the tooth to the other. A portion of the variation can be attributed to the eccentricity of the ratchet, which allows the index throw to vary slightly from the overthrow stop to the engagement point. Concentricity of the ratchets with the platen bushing was therefore a major objective for the new system. Later it was found that this eccentricity has a smaller effect on the multiple index output because the total motion is reduced by the gear reduction from each ratchet to the platen. This reduction does, in fact, reduce the variation at the platen to about half, due to the random orientation of the eccentricity inherent in the two individual ratchets.

At this point in the development, gear backlash became the major contributor to line-to-line variation. Backlash in the gears was initially recognized as a problem, but the approach taken was to mount each planet gear driving the platen on a separate plate with a lock-together adjustment between the plates. In this manner, a maximum backlash condition could be found and adjustably locked out by rotating one plate relative to the other, which brought all of the gears into positive contact. This approach was soon abandoned, however, because of the random pattern of backlash. Once locked, it was found that binding occurred where backlash was lower, thus forcing the gears into tight contact. Using this system, it was necessary to allow a certain amount of backlash so that the gears could run freely.

An elastic gear tooth approach was also tried, whereby the planet gears were a molded plastic, the idea being to use the built-in elasticity or springiness of the plastic gears to eliminate binding. The idea did not work out well in application because of the tooth-to-tooth variation between adjacent gear teeth. However, the idea of elasticity or springiness led naturally to the spring-biased gear system now used.

In the gear bias approach, the two planet gears are mounted on separate plates, as before, and are spring-loaded together rotationally. This has the advantage of eliminating backlash during rotation, and the tooth-to-tooth variations in the gears and ratchets become major factors in controlling the magnitude of line-to-line variation in the system. Statistics of these variations were studied to determine the predictable variation (see Appendix). The statistical analysis showed that line-to-line variation should be of the order of ± 0.001 for 1-sigma; ± 0.002 for 2-sigma; and ± 0.003 for 3-sigma, which includes 99.73% of all possible indexes made. These variations were calculated for a commercial Class 3 gear. Obviously, the variation can be spread or narrowed, depending on the quality class of gear used and the tooth-to-tooth variation allowed in the ratchets. Economics will also play a role because the cost of quality gears increases exponentially with quality. Based

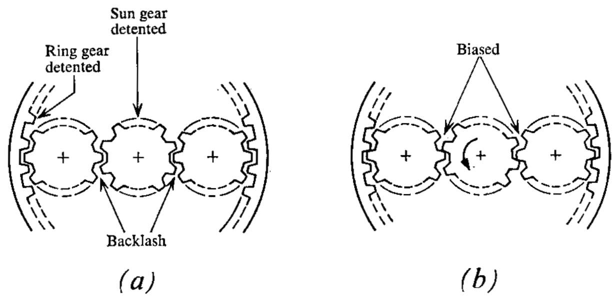


Figure 9 Effect of bias on backlash: (*a*) unbiased system; (*b*) biased system.

on the calculations above, however, it is felt that a commercial Class 3 gear is necessary.

Line-to-line variation is random due to the "closed" nature of this system. Accumulated error is not expected to be greater than ± 0.003 inch. That is, *any* two lines of type should be within 0.006 inch of the theoretical line-space accuracy of the particular index mode used.

There are only two factors observed to which accumulated line-space variation may be attributed:

(1) The platen diameter tolerance on the ground rubber surface. This tolerance is ± 0.002 inch and, once ground, the diameter is a constant.

(2) The thickness of the paper used with the platen system is also a constant for a particular paper. Actual paper-feed tests showed a variation very similar to that predicted. Occasionally, a 0.006 inch line-to-line variation was found, but the average variation appeared to be more in the 2-sigma or ± 0.002 category. Paper slippage is the only other factor involved which can affect the line-to-line variation. This is an all-inclusive term, dependent on such variables as coefficient of friction of platen, paper, and feed rolls; normal forces applied to paper; skew effects of rollers, etc. These factors are not discussed here. The indexing system can logically be separated from the paper-feed system, and the variation of line spacing is controlled separately, so that the problem of line-space variation may be isolated.

The bias spring system was designed to produce a torque capable of turning the platen against its drag (feed rollers and bushings) when the ratchets are detented. This torque must turn the platen until the planet gear teeth come positively into contact with both the ring gear and sun gear teeth (Fig. 9).

During the course of development of the bias spring system, a "cocking" condition was found in the ring gear-ratchet. A binding condition resulted, causing the ratchets to "break loose" from the detents when they were supposed to be holding.

The following modifications were made to the system:

(1) A two-spring bias system was designed; each spring producing half the torque required. By mounting the springs opposite each other, it was theoretically possible to eliminate bearing reactions, except the one caused by the reaction force to the load through the planet gear shaft.

(2) A reaction surface was added to the platen bushing to aid in reducing the cocking tendencies of the ring gear-ratchet due to the detents.

(3) A ball bearing was added between the sun gear and ring gear to reduce friction losses.

A second study investigated the holding characteristics of the index detents with a "pure" detent wheel designed for this purpose. The conclusion drawn from that analysis was that a sizable gain (52%) in holding torque could be attained by a redesign of the ratchet and detent tip geometry. Maximum contact stresses were of the order of 164,000 psi for the old system tested, so this value was used in the design of a new system. The major gain is in the holding angle, which is 55° in either direction, compared to the old angle of 39°. The contact force, F_d, was actually decreased for the new configuration from 4.0 lb. to 3.5 lb. This is due to a reduction in the index detent spring force, F_s, from 1.3 to 1.14 lb. at the operating length. The actual holding torque gain over the previous system is 70%, without considering the sliding friction of the detents, which boosts this gain to 136%, or a 2.36:1 gain over the old system.

Manual control to one-point accuracy

With the planetary gear system (Fig. 2), it is a relatively easy matter to divide the input function so that the ring gear-ratchet is driven by the left-hand knob (giving a 5:4 rotational motion between the ring gear and platen), and the sun gear-ratchet is driven by the right-hand knob (giving a 5:1 rotational motion between the sun gear and platen). By coupling the system as described, it is possible to obtain a manual 4-point index per tooth rotated on the ring gear-ratchet (1/66 x 4/5 x 330 points). Notice that the left-hand platen knob turns the platen 1:1, but feeds through a 4:5 increase to turn the ring gear-ratchet while the sun gear-ratchet remains at rest. Also, it is possible to obtain a manual 1-point index of the platen per tooth rotated on the sun gear-ratchet (1/66 x 1/5 x 330 points). Notice that the right-hand platen knob turns the sun gear-ratchet 1:1, but feeds through a 5:1 reduction to turn the platen while the ring gear-ratchet remains at rest.

It is worthy of note that the platen output in this design *cannot* be interchanged from the left to the right platen knob. There are crossovers in the gear system and ratchets that make this approach impossible; however, the right-hand knob *can* be designed to fasten directly to the platen shaft. This would make both knobs a 4-point output, but the 1-point manual capability would require the addition of a third knob coupled to the sun gear-ratchet.

Platen variable

In the SELECTRIC Typewriter platen variable the increment indexed between serrations is 0.025 inch. With the present system the increment indexed through the right-hand knob is 0.0139 inch. Because of the fine control achieved with the right-hand platen knob, the SELECTRIC Typewriter platen variable was discarded early in the present development program.

Ease of index selection

This selection philosophy was considered separately as the multiple index system developed. The selection method (Fig. 8) is a 2-guide cam mounted directly on the platen. As the index pawl swings in a fixed circumferential arc, tabs on the pawls follow the two guide surfaces on the cam, guiding the index pawls into the proper teeth. It was found that by eccentrically mounting the index detents, the entry of the pawl tips into the ratchets could be precisely controlled by a simple eccentric adjustment. The throw and initial position of the pawls are controllably adjusted by the multiplying lever stop and index clevis link, respectively. Thus, only one selector cam adjustment is necessary and all other modes are automatically set by this one adjustment. The guide is detented in position by a detent wheel and detent lever. The index dial is fastened to the guide with index mode numbers on the periphery. The dial protrudes through a slot in the cover for easy selection.

Rollback

In the marketing philosophy for the SELECTRIC Composer, there is little need for page rollback; however, such capability must be present to accommodate exceptions to that rule. As designed, the system has this capability. Some hypothetical examples will illustrate.

(1) An error has been made on the line being typed. The platen is rolled back with the 4-point knob until the paper is at a height at which the correction is to be made. Once the correction is made, the platen is returned to the writing line easily with the 4-point knob because the 4-point increment is visually easy to align to the correct position. There is no alignment variation tolerance in this application.

(2) All other rollback applications, including mixing type on previous lines, and error correction to previous lines, should be done with the right-hand platen knob. The reason is that alignment to a previously-typed line must normally be done with the 1-point increment in any case, and rolling up and back with this knob produces no alignment variation when returning to the last line typed. However, the alignment variation expected on *any* previous line will be as predicted statistically: ±0.003 inch (or a maximum of 0.006 inch). As long as the system gears are al-

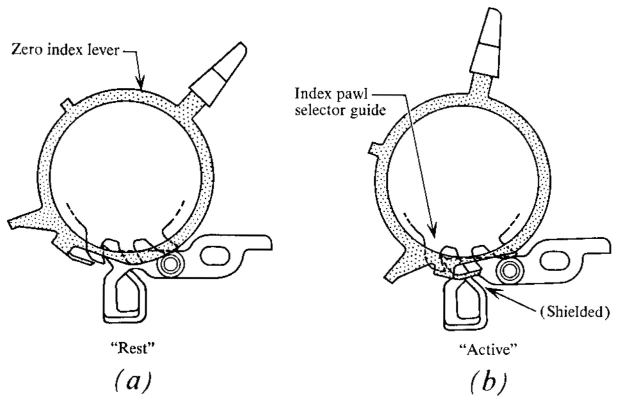


Figure 10 Zero index lever and pawl guide: (*a*) at rest; (*b*) active.

lowed to shift positions from what they were when a particular line was typed, the 3-sigma variation of ± 0.003 inch must be expected.

In line with this thought, there is an obvious solution to line-to-line variation. This variation can be virtually eliminated only by reversibly returning to a previous line with the gears in the same orientation as when the line was typed. This means a *reverse index* feature. Obviously, such a system will become complex when considering the dual ratchets, detents, index pawls, and overthrow stop already in the system. If such accuracy becomes mandatory in the future, however, a reversible index mechanism could be developed.

Carriage-return, no index capability

The usefulness of a "zero index" function becomes obvious when one studies the various applications of the composing machine. It is frequently necessary to return to the left with the carrier, in order to center material which has been trial typed in "no print," to "justify" material, to insert type of a different style or font, and to type material "flush right" after it has been initially typed in "no print." Notice that these applications call for returning to the left of the page (on the same line). Normally, the only methods available to move the carrier to the left on the same line are by pushing the carrier by hand, or depressing the "backspace" keybutton. Both methods have disadvantages; pushing the carrier manually is slow, and the carrier is not easily accessible because of the location of the justifier mechanism. Backspace, even in "repeat," is also slow; it is therefore evident that some new mechanism was necessary.

During the development of the selector system, it was apparent that a "zero index" could be obtained by inserting a special position on the selector cam which guides the pawls. The principle is that the index pawls must be "held out" so that they do not enter the ratchets but are guided under the overthrow stop. This condition was not easily obtained with the original two-cam arrangement behind the index pawls, but with the index pawl guide mounted on the platen, a "mask-out" position could be quite easily placed on the periphery of the guide.

The approach of putting a "zero index" position on the index selector dial was made initially because no additional parts would be required. This method was also found to have utility when used in conjunction with the "carriage return" function. However, later work in human factors engineering showed that, although the function was useful, the method of obtaining it left much to be desired. It was found that to use the function an operator must turn the index selector dial from the index mode being used to the "zero" position, return the carrier and then *remember* to reset the index dial. Turning the dial to "zero" was the first disadvantage because this operation wastes time, but remembering to reset the dial to the previously set mode proved to be the major problem. An operator would often forget to reset the index dial; the platen would therefore not index, and the operator would immediately type *over* the line that had just been typed.

From this study it was obvious that a new definition was needed for the "zero index" function to minimize the possibility of operator error.

The method finally adopted is a lever mounted on the platen, with a guide that can be rotated into position in front of the index pawls and mask them out of engagement with the ratchets. The action of the lever is shown in Fig. 10. The lever is spring-loaded so that when released, it returns to its original position and the normally set index mode is once again operational. This means that the "carriage-return, no index" function is a two-handed operation which an operator can only achieve intentionally. The lever is pulled forward and held until the "carriage return" keybutton is depressed. The lever may then be released and will return to its inactive position. An "exploded" diagram of the complete "zero index" is shown in Fig. 11.

Summary

The development of a multiple index system grew in a logical manner from a recognized need in the specifications for a composing machine. The achievement of the design goals is summarized here.

The "automatic" index feature has been achieved by combining the existing SELECTRIC Typewriter operational mechanism with the new dual-ratchet planetary gear system. Variable vertical spacing has been achieved for increments between 5 and 20 points. It covers nearly all IBM standard index increments and also provides the printer with one-point versatility. Line-space tolerances have been held to within ± 0.003 inch by the proper selection of gear quality and ratchet-tooth tolerances and the addition of a

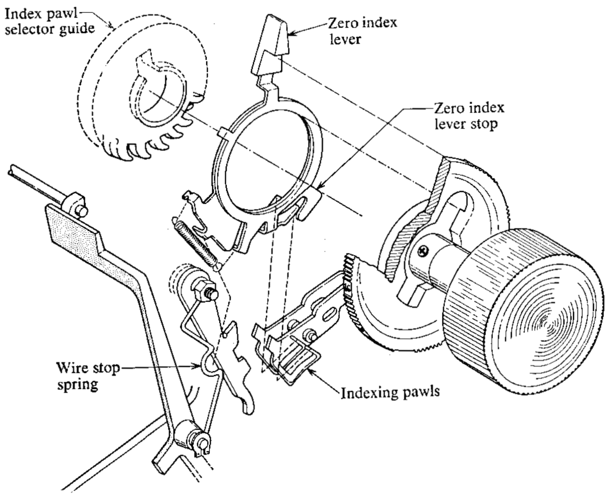


Figure 11 "Exploded" schematic of zero index mechanism.

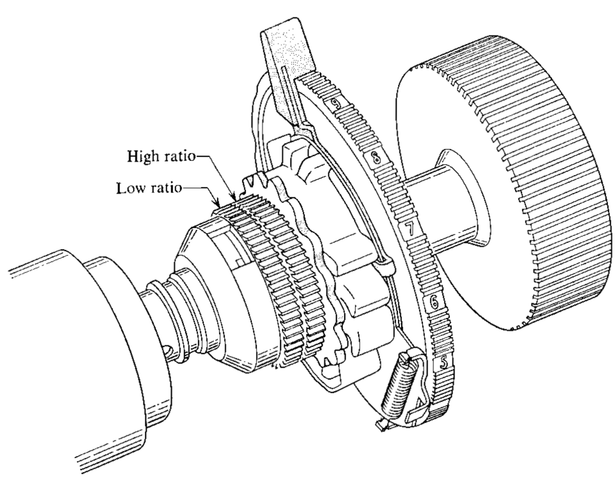


Figure 12 Complete system (except for actuating mechanism and detent lever) as installed in Composer.

gear bias. Manual control in one-point (0.0139 inch) increments is available with the use of one platen knob. Because of the built-in one-point increment, the need for a separate "platen variable" function has been eliminated. Index increments are selected with an operator-controlled dial which detents a circular cam to guide the index pawls into the proper tooth combinations. Rollback of the paper for correction and return to the previous line has been accurately achieved by the use of the 4-point platen knob. The "carriage-return, no index" function has been achieved by adding a "masking" lever which prevents the index pawls from indexing the ratchets.

In Fig. 12, the complete mechanism is shown as it appears in the composer system. At the left is a part of the platen; other visible components are, from left to right, the two ratchets, the detent wheel, the index pawl selector guide, the "zero index" lever and spring, and the selection dial.

Acknowledgments

Acknowledgment and thanks are due to W. O. Cralle, who participated in the design of this mechanism, and a special note of appreciation to C. C. Wilson, whose analytical work was of great value in achieving the final design.

Appendix: line-to-line variation in multiple indexing

The total error in a line space due to the gears and ratchets is:

Total error = LH ratchet error + RH ratchet error + sun gear error + ring gear error + planet gear error,

or

$$E_T = \left(\frac{D_P}{D_R}\right)\left(\frac{P}{RG}\right)(E_{RL}) + \left(\frac{D_P}{D_R}\right)\left(\frac{P}{SG}\right)(E_{RR}) + \left(\frac{P}{SG}\right)(E_{SG}) + \left(\frac{P}{RG}\right)(E_{RG}) + 2\left(\frac{P}{PG}\right)(E_{PG}),$$

where

E_T	= Total possible error	
D_P/D_R	= Ratio of platen diameter to ratchet diameter	= 1.450/1.187,
P/RG	= Speed ratio or gear ratio from LH ratchet (ring gear) to platen	= 4/5,
E_{RL}	= Maximum tooth-to-tooth error of LH ratchet teeth	= 0.004 inch,
P/SG	= Speed ratio from RH ratchet (sun gear) to platen	= 1/5,
E_{RR}	= Maximum tooth-to-tooth error of RH ratchet teeth	= 0.004 inch,
E_{SG}	= Maximum circumferential error of sun gear	= 0.001 inch,
E_{RG}	= Maximum circumferential error of ring gear	= 0.001 inch,
P/PG	= Speed ratio of pinion gear to platen (when ring gear turns, worst case)	= 4/3, and
E_{PG}	= Maximum circumferential error of planet gear	= 0.001 inch.

Then

$$E_T = \left(\frac{1.450}{1.187}\right)\left(\frac{4}{5}\right)(0.004) + \left(\frac{1.450}{1.187}\right)\left(\frac{1}{5}\right)(0.004)$$
$$+ \left(\frac{1}{5}\right)(0.001) + \left(\frac{4}{5}\right)(0.001)$$
$$+ 2\left(\frac{4}{3}\right)(0.001)$$
$$= \left(\frac{1.450}{1.187}\right)(0.004) + \left(\frac{10}{3}\right)(0.001)$$
$$= (0.00488) + (0.00332)\,.$$

$$E_T = 0.00820 \text{ maximum.}$$

This represents the maximum possible error which could ever be measured between two line spaces. This variation is not cumulative. To determine the normally expected variation in line spacing, these factors must be treated statistically. The variance may be expressed as

$$\sigma_{\text{Total}} = [(\sigma_{\text{LH ratchet}})^2 + (\sigma_{\text{RH ratchet}})^2 + (\sigma_{\text{sun gear}})^2 + (\sigma_{\text{ring gear}})^2 + (\sigma_{\text{planet gear}_1})^2 + (\sigma_{\text{planet gear}_2})^2]^{1/2}\,;$$

$$\sigma_T = \left\{\left[\left(\frac{1.450}{1.187}\right)\left(\frac{4}{5}\right)(0.002)\right]^2 + \left[\left(\frac{1.450}{1.187}\right)\left(\frac{1}{5}\right) \times (0.002)\right]^2 + \left[\left(\frac{1}{5}\right)(0.0005)\right]^2 + \left[\left(\frac{4}{5}\right) \times (0.0005)\right]^2 + \left[(2)\left(\frac{4}{3}\right)(0.0005)\right]^2\right\}^{1/2}.$$

$$\sigma_T = 0.00104 \text{ inch}\,.$$

For a 99.73% confidence limit,

$$3\sigma_T = 0.00312 \text{ inch}\,.$$

Thus, only 0.27% of any line space measurements would fall beyond the tolerance of ±0.003 inch.

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