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IBM Copier Scanning System

Abstract: The quality of the copy to be produced by the IBM Copier was of utmost importance in the machine development. This dictated that the amplitude of the vibrations of the document scanning system during scan be limited to a very low level. The design considerations, analysis, simulation, and instrumentation used to assure adequate performance of the system are described.

Introduction

Functional requirements laid down at the inception of the development program for the IBM copier dictated that the image of a document be projected on a continuously moving photoconductor. The electrostatic image thus produced could then be easily developed and transferred to plain bond paper. The means chosen to do this continuous imaging was to scan the document onto a rotating photoconductor or image drum, so that only a small strip of the document would be projected onto the drum at any given time.

Throughout the program the importance of appearance, readability, and general quality of the first copy and of subsequent copies of copies were paramount. For this reason the scanning system development effort was directed toward designing a system from which the copy resolution would be the best economically practicable. There are, of course, many factors that affect copy resolution in an electrostatic copying process. The cost of an optical system increases rapidly with an increase in specified resolution. Additional research and development time is necessary to significantly improve photoconductor, developer, and transfer characteristics that affect resolution; however, one area of the machine in which development is not as economically limited is the document scanning system. Obviously, this is an area that affects resolution as much as any other and also an area where competence and experience in mechanism design can be brought to bear on cost and performance objectives. This paper discusses such an approach.

There are four conventional ways to optically scan a document: 1) move the document; 2) move the lens; 3) move a mirror or mirrors; or 4) any combination of these. The decision was made to move the document for scanning because: 1) mirrors would then not be needed, thus avoiding the loss of resolution and light inherently associated with these optical components; 2) a lens of relatively short focal length could be used to give better resolution and light transmission at a specified cost; 3) the space occupied by optics in the machine could be minimized; and 4) there would be less sensitivity to vibration in a scan motion than in other systems considered, since the vibration would not be amplified optically. This decision was possible in light of a proposed ten-copy-per-minute rate and its resultant 4 in./s scan velocity. The greatest disadvantages in moving the document would be the potential problems associated with document handling: operator inconvenience, difficulty in copying oversize documents, and safety. However, to minimize the potential safety problem the document would be held on a flat glass plane mounted in a moving carriage with the motion isolated from the operator.

System requirements

The initial design objectives established for the moving document scanning system were:

1. to limit acceleration at the document plane to 0.1g to minimize the possibility of a shift in document posi-

tion (the coefficient of friction between glass and paper is 0.15), and to minimize the dynamic forces and resultant deflections in the system;

2. to minimize the distance required for acceleration and deceleration of the carriage to and from the scan velocity, since the overall width of the machine was determined by the total carriage travel;
3. to limit the relative displacement between the image drum and the document image during the time required to move across the scan aperture to ± 0.001 in. which would, in theory, maintain a resolution of about 7.5 lines/mm. or better;
4. to provide accurate registration of the image on the drum for simplicity in timing the various process steps.

Scanning mechanism design

• Evaluation of various design approaches

Several mechanisms were proposed and evaluated as possible ways of simultaneously meeting all four of these requirements. These are listed in Table 1 along with their advantages and disadvantages.

After evaluating the above systems and others involving combinations of these, the decision was made to use a cable and pulley system. It was felt that the accuracy required could be obtained with conventional manufacturing methods, thus eliminating the high cost of manufacturing usually associated with special materials and techniques. The recognized exposure to vibration between the drum and image was a primary concern; therefore, one task was to insure that the means of controlling the acceleration would not induce excessive vibrations.

To meet the objectives set forth, some predictable means of controlling the carriage motion over the complete scan cycle was necessary. The individual portions

of the scan cycle studied were: 1) acceleration to scan velocity; 2) velocity matching at the end of acceleration; 3) constant velocity during imaging; and 4) deceleration to rest at the end of the scan.

One of the direct advantages of the cable and pulley system is that during the scan motion the angular velocity of the carriage drive pulley can be made equal to the angular velocity of the drum by mounting the pulley on the drum shaft and coupling them together with an in-line clutch.

Many types of clutches were considered for this task: magnetic, spring, roller, dog, sprag, disk, and others. In addition, elastomeric shock absorbers, springs, and other auxiliary means of controlling the output acceleration of the clutch were explored. Initially a roller clutch was chosen; however, because of pick variations and the possibility of large deflections, a special dog clutch with a single tooth was designed (see Fig. 1). The single-tooth clutch was applicable since the clutch pickup had to occur at the same point in each cycle. The carriage acceleration is controlled through an accelerator cam that is used to accelerate the drive pulley to the same angular velocity as the clutch. At that point the carriage drive pulley and the clutch are coupled by a wedging lock pawl. This lock pawl and a wedging back check pawl were provided to eliminate backlash in the clutch. Thus, when locked together, the pulley and the drum rotate as one with the same angular velocity, making the only image-to-drum velocity error that error introduced by the dimensional variations in the drum and pulley radii and in the cable's neutral axis. All these variations could be held to less than 0.1 percent with a corresponding total velocity error of less than 0.1 percent.

• System simulation and analysis

With the preliminary design completed, there was still some concern over the dynamic motion of the document

Table 1 Comparison of design approaches for scanning mechanism

<i>Mechanism</i>	<i>Advantages</i>	<i>Disadvantages</i>
Cam and linkage	1) Few moving parts 2) Accelerations, velocities and displacements easily determined	1) Manufacture of precise cam difficult
Gearing	1) Flexible configuration	1) Ultraprecision gearing required 2) Tooth-excited vibrations likely
Dual motor with speed control	1) Precise control possible	1) High cost
Belts	1) Low cost	1) Nonpositive drive
Cable and pulley	1) Low cost 2) Predictable accuracy 3) System easily manufactured	1) Vibrations likely

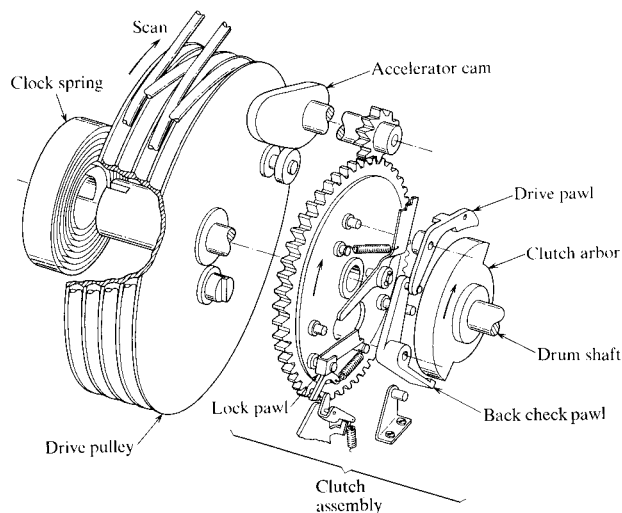


Figure 1 Drive mechanism and clutch assembly for scanning system.

image relative to the photoconductor drum at the start of the scan. As mentioned before, that motion, if greater than 0.002 in., would in theory decrease the copy resolution to less than the 7.5-lines/mm. objective. To evaluate this phenomenon the system as first envisioned was simulated on an analog computer, which was used because the only alternative available at the time, an analog simulator on a small digital computer, required enormous amounts of time for the numerous runs necessary to evaluate the scanning system variations. The lumped-mass model of the system drawn in Fig. 2 is shown in Fig. 3. The applicable differential equations for the acceleration portion of the document carriage's motion are:

$$m_D \ddot{x}_D = k_{MD} (x_M - x_D) + c_{MD} (\dot{x}_M - \dot{x}_D) - c_D \dot{x}_D - k_{DC} (x_D - x_C); \quad (1)$$

$$m_P \ddot{x}_P = F_P - k_{PB} (x_P - x_B) - c_{PB} (\dot{x}_P - \dot{x}_B); \quad (2)$$

$$m_B \ddot{x}_B = k_{PB} (x_P - x_B) + c_{PB} (\dot{x}_P - \dot{x}_B), \quad (3)$$

where m is the mass (or equivalent mass), c is the coefficient of viscous damping, k the spring rate, x the displacement, F the applied force, and the subscripts M, D, C, P, and B refer to the motor, drum, cam, pulley, and document carriage respectively.

If it is assumed that energy is conserved by the cam and its follower, then

$$F_P dx_P = F_C dx_C, \quad (4)$$

where dx is a differential displacement. Substitution of the spring force on the cam gives

$$F_P = k_{DC} (x_D - x_C) / (dx_P/dx_C). \quad (5)$$

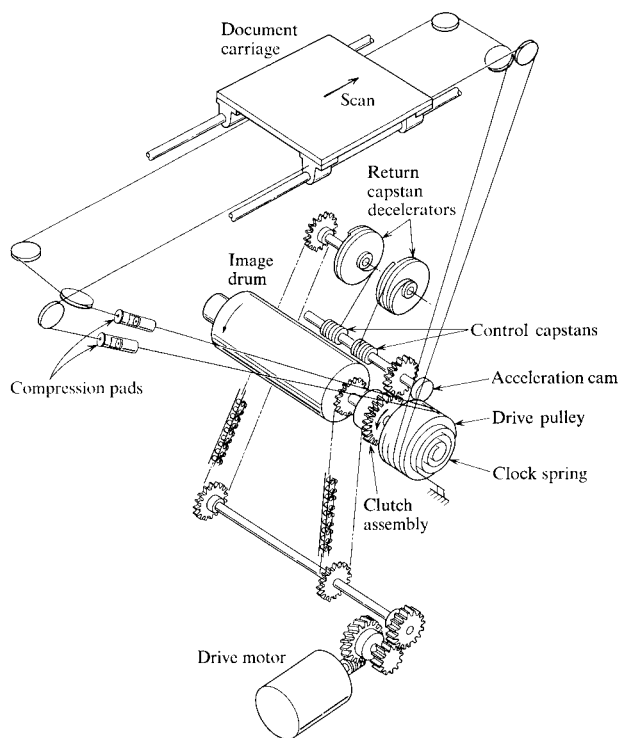
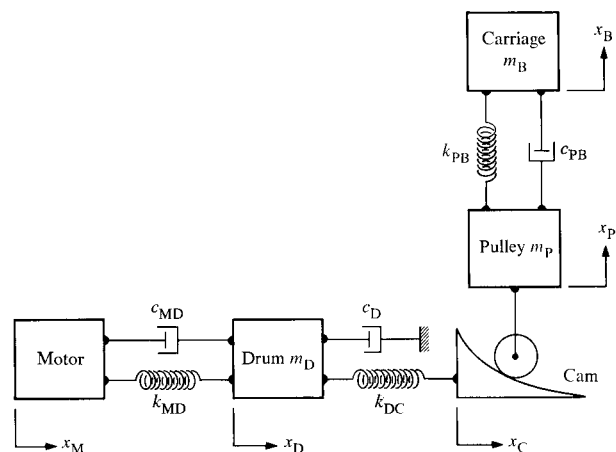


Figure 2 IBM Copier scanning system.

Figure 3 Lumped-mass approximation of scanning mechanism.



The pulley displacement x_P is a function of the cam displacement x_C . Once this relationship is defined by choosing a cam profile, x_C in Eqs. (1) and (4) and the derivative in Eq. (5) are determined and the set of ordinary differential equations, two of which are nonlinear, can be solved for various values of the system parameters.

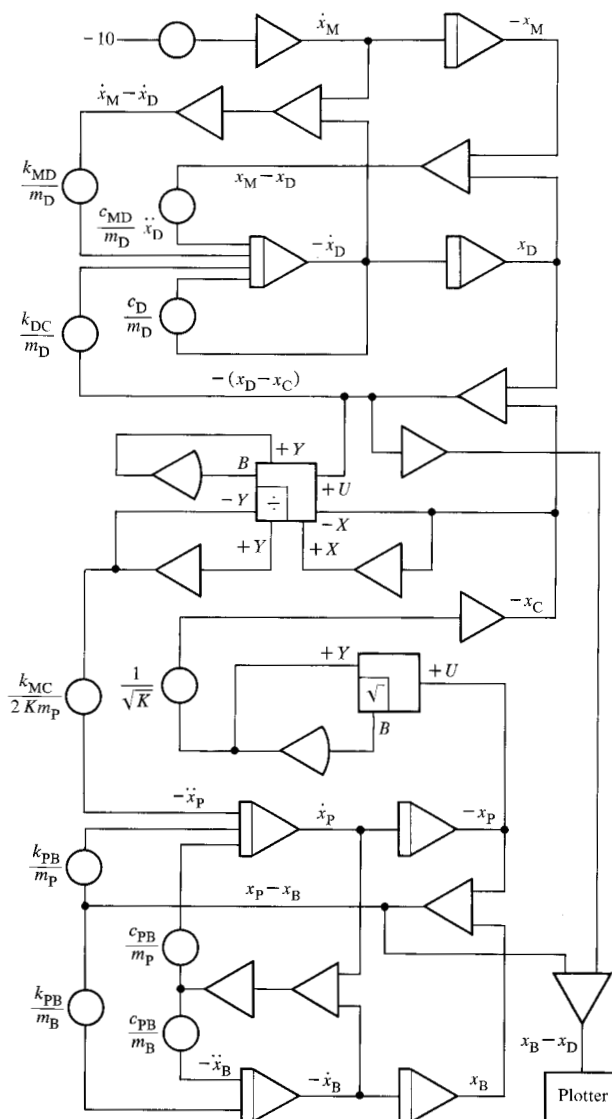
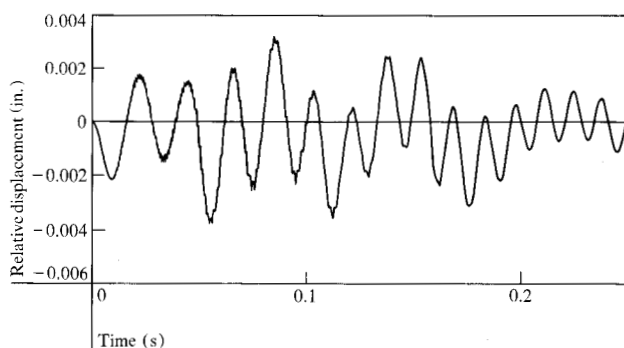


Figure 4 Analog circuit used to simulate system at start of scan.

Figure 5 Image-to-drum relative displacement vs time from start of scan acceleration, plotted from the analog simulation.



The choice of the cam profile determines the nature of the nonlinearity and consequently the complexity of the analog computer circuit. This was the key to the simulation. To solve the equations on the available computer and at the same time approximate the dynamically desirable modified trapezoidal profile, a constant acceleration cam was chosen, i.e.,

$$x_P = Kx_C^2, \quad (6)$$

where K is the constant that determines the acceleration of the document carriage from rest. The motor's equivalent mass was over 100 times that of the rest of the system, and the motor was therefore assumed to have a constant velocity. An unscaled analog computer circuit diagram for the simulation is shown in Fig. 4 and a sample simulation plot in Fig. 5.

• Simulation results and design changes

The document carriage acceleration cam initially selected used only half the displacement allotted for carriage acceleration. The remaining half was intended as a buffer for the decay of any oscillation induced during the acceleration period. The results from the simulation of a design using that cam showed that the cam buffer was ineffective because of the minimal system damping and, in addition, that the image-to-drum relative displacement of 0.0043 in. exceeded the 0.002-in. specification. An investigation was therefore begun to determine the changes to the system that would provide the most effective improvement in performance. An alternate cam, producing one-half the acceleration of the original but requiring the entire displacement allotted for carriage acceleration, was then simulated. With that, the carriage and image drum would be coupled immediately following the carriage acceleration. With the alternate cam, the induced carriage vibration at the time of coupling was 0.0039 in., an improvement of 0.0004 in. over the original cam. Although the cam modification improved the system performance, additional changes were required for the scanning system to meet the design objectives.

By investigating a number of combinations of the various system parameters, a set of realistic changes to the design was developed to provide the prescribed system dynamics. The changes included stiffening the scanning system drive between the motor and image drum and adding damping between the pulley and the document carriage. Damping between the pulley and document carriage was added by changing their drive connection from steel tapes to stranded wire cables. Additional damping was obtained by coupling an elastomer compression pad in series with the cables, as shown in Fig. 2. The stiffening of the system between the motor and image drum was accomplished by removing a jackshaft, replacing two belts with a chain, and increasing two crit-

ical shaft diameters. With these changes and the low acceleration cam, the simulated maximum relative displacement between the image and drum was 0.0003 in.

The motor-to-drum spring rate in the scanning system could have been made larger with an all-gear drive. The simulation results, however, showed the intermediate chain drive rate to produce superior dynamic results. Further investigation indicated that the carriage and drum moved essentially in phase with a low motor-to-drum spring rate and in opposition with a very high-rate system. Thus the drum-to-carriage vibrations associated with those modes were larger than those of the intermediate-rate system. Results of the simulation are shown in Fig. 6.

In addition to indicating the need for the previously mentioned design changes, the scanning system simulation was used to study the effect of a kinematic velocity mismatch between the drum and document image at the time of coupling. The increase in the relative displacement magnitude produced by an assumed one-percent kinematic velocity mismatch was negligible (about 0.00002 in.). The assumption of one percent was based on the accuracy of the drive system components with reasonable manufacturing tolerances.

The 0.0003 in. relative displacement result predicted for the modified system indicates that more changes were made to the original than were necessary to meet the design requirements. This was done because of the realization that it was impossible to predict exactly the dynamic performance of a system without including such factors as backlash, coulomb friction, gear tooth excitations, etc., which were impossible to determine accurately with only a layout of the mechanism.

• Instrumentation

The next step in the development process was to measure the actual machine performance when the prototype was built and make additional changes where necessary. Measurement of the relative displacement between the document image and the photoconductor drum as the document was being scanned was accomplished somewhat indirectly. It was assumed that the motion of the document itself accurately reflected the motion of the image; i.e., it was assumed that the lens would not move in any such manner as to cause the image to displace in the scan direction. Thus, the displacement of the document carriage was measured and compared with the displacement of the drum.

In order to make the measurement, transmission gratings on film with 0.005-in. wide opaque bars 0.005 in. apart were attached to the document carriage and to the image drum (see Fig. 7). Light from microscope lamps was passed through the gratings onto phototransistors, each of which was covered by a mask with a single

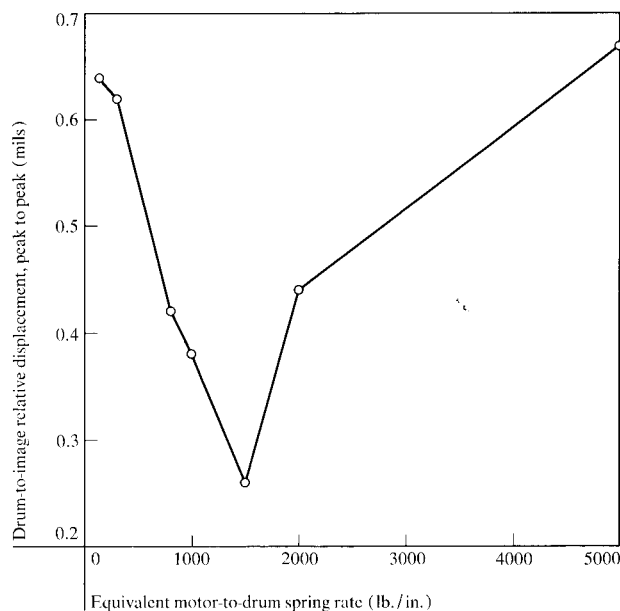
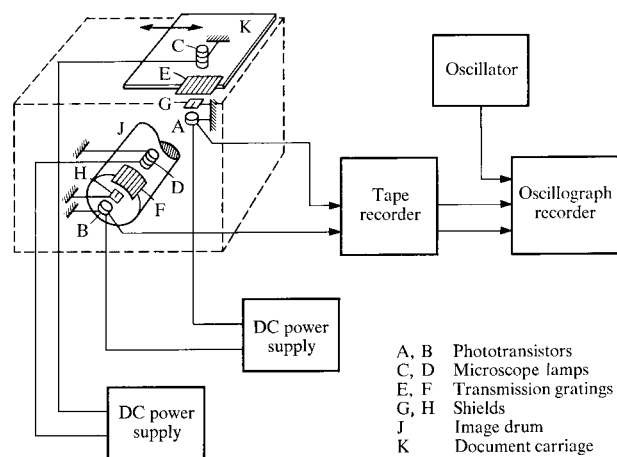


Figure 6 Image-to-drum relative displacement vs motor-to-drum spring rate, from the simulation.

Figure 7 Schematic of digital instrumentation used to measure dynamic relative displacement between document image and drum surface.



0.005-in. wide slit. The output from each phototransistor approximated a sawtooth waveform as the machine was operated.

The sawtooth signals from the phototransistors were recorded on magnetic tape at a tape speed of 60 in./s. This information was then recorded on an oscillograph strip chart with the magnetic tape running at 7.5 in./s and the chart at 128 in./s to obtain easily readable and, consequently, more accurate data. At the same time an 8-kHz reference signal from an audio oscillator was recorded on the chart.

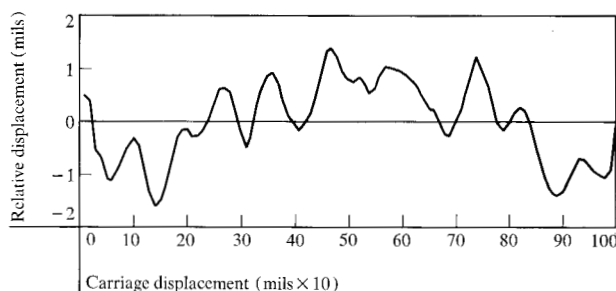


Figure 8 Image-to-drum relative displacement vs carriage displacement, from the instrumented prototype.

From the strip chart were read the times for each 0.010 in. of drum displacement and of document carriage displacement to the nearest thirty-second of a millisecond (quarter-cycle of the reference signal). Since the information required was the difference between the displacements of the drum and the carriage as a function of time, reduction of the data was necessary and a computer was used. Before the displacement difference computation was done, a first-degree least squares approximation relative to three points was made in order to smooth the incremental data.

The final result was a tabulation and plot of the relative displacement between the drum surface and the document carriage vs carriage displacement. A sample of the plotted data is shown in Fig. 8.

• Results

The 0.003-in. peak-to-peak relative displacement value indicated in Fig. 8 was typical of the three prototype machines measured. Although ten times larger than the predicted value of 0.0003 in., the value was only 50 percent larger than the specified maximum of 0.002 in. As is apparent from the nondecaying nature of the plot, excitations other than the cam acceleration introduced vibrations into the system after the acceleration period. These are attributable to a combination of the factors of backlash, etc., mentioned previously. In fact in a later study, several modifications to the simulation brought the predicted and experimental results to within 15-percent agreement. These modifications were 1) the addition of backlash between the drum and motor masses; 2) the addition of a sinusoidal velocity perturbation to the motor velocity to account for gear-tooth excitations; and 3) the addition of a coulomb friction force to the drum mass. The most important of these was the addition of backlash. The 0.002-in. relative displacement objective was later found to be conservative in that the actual hardware, although having a maximum relative displacement of 0.003 in., produced copy resolution of greater than 9 lines/mm. This improved resolution was due to

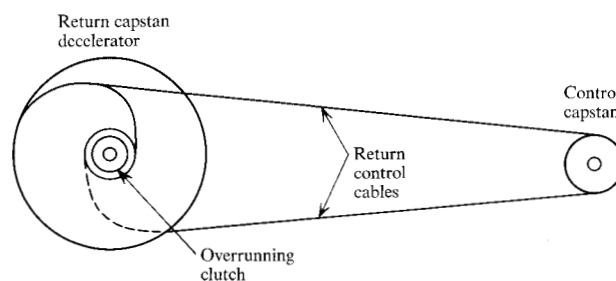


Figure 9 Governor and brake for carriage velocity control during carriage return.

phenomena involved in the rest of the copy process, e.g., the discharge time constant of the photoconductor and the field gradient of the electrostatic image.

Return mechanism

• Design approach

The scan operation takes approximately 270° of drum rotation; the remaining 90° must therefore be used to return the carriage reliably while minimizing the accelerations. To meet the design objective of a 0.1-g maximum cycle acceleration, constant acceleration devices (a clock spring and a magnetic brake) were initially selected to control the carriage motion during return. These were chosen because they would provide constant torque and thus produce minimum accelerating and decelerating forces on the carriage. In addition to the above devices the same cam used for carriage acceleration during the scan cycle was used to control the carriage motion during the last half-inch of return.

A further consideration in choosing a spring-driven return was the power required. Since the carriage return takes place in approximately one-third the scan time, a significantly larger motor would have been required if used to drive the carriage during return as well as during scan.

• Analysis and results

To determine the feasibility of using a constant acceleration and deceleration in the return cycle, a functional tolerance analysis was made in which the effects of small variations in the component properties were evaluated. By simple application of constant-acceleration equations, the analysis indicated that torque variations as small as 5 percent in either the clock spring or brake would lead to failure in meeting the time and acceleration criteria. With that information, alternate schemes for the return cycle were examined. Although the constant acceleration-deceleration return cycle was not unique within the specification constraints, there was lit-

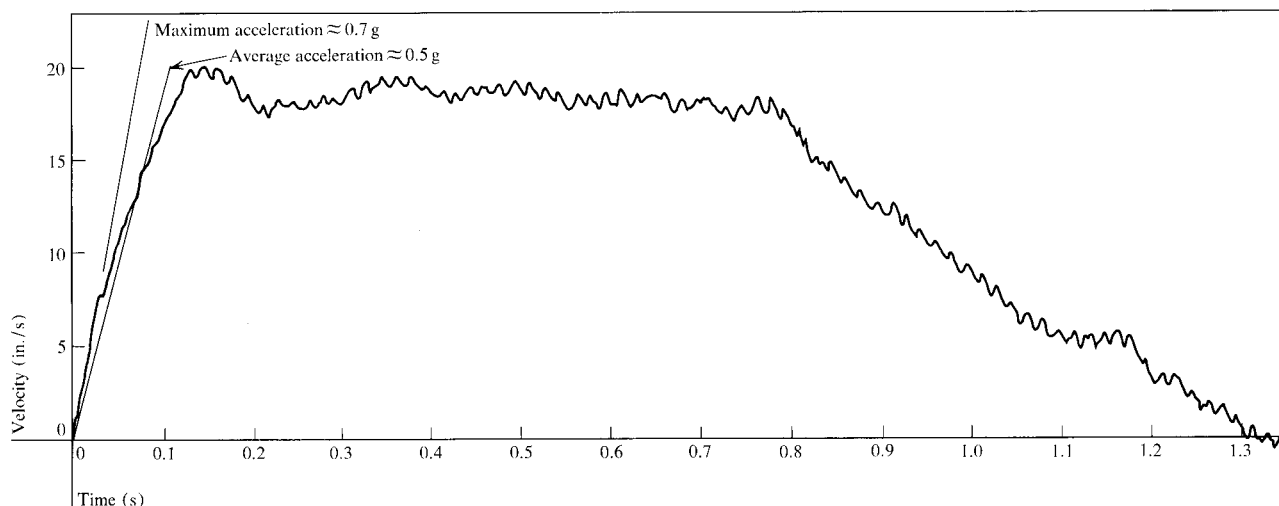
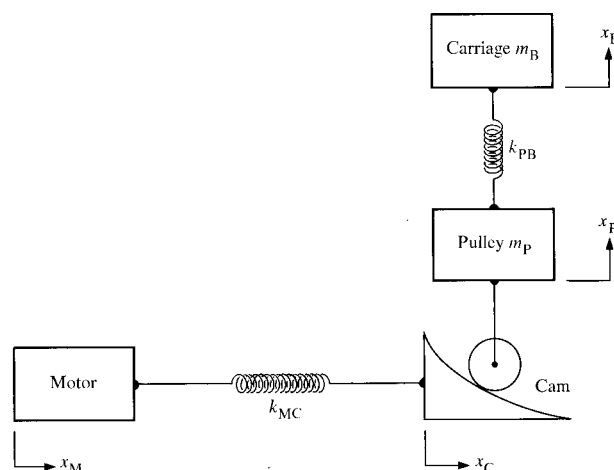


Figure 10 Carriage velocity vs time for return cycle (measured).

tle latitude for alternate approaches. It was soon realized that any noncritical return cycle would require violation of the 0.1g acceleration criterion. Several types of return cycles were considered: 1) a 0.2-g constant acceleration-deceleration cycle; 2) one with an initial and final 0.2-g constant acceleration with a constant velocity portion between; and 3) one with an initial 0.5-g constant acceleration, a final 0.1-g constant acceleration, and a constant velocity portion between. The last was selected because there was concern that carriage oscillation induced by an increase in the acceleration level at the end of the return cycle would carry over into the succeeding scan cycle. In addition, the acceleration criterion at the beginning of the return cycle was critical only relative to document slippage, which could easily be controlled by a cover on the carriage. With the basic return scheme using a constant velocity region determined as the most satisfactory, components were examined to achieve this reliably. When properly designed, a clock spring exhibits a relatively small variation in output torque and was considered an acceptable accelerating device. For the constant velocity control, the torque variation in conventional centrifugal governors and magnetic brakes was too large because of factors such as atmospheric conditions, foreign material, and wear. To overcome the problem, a special velocity control device was designed, viz., a cable and variable-radius capstan used in conjunction with an overrunning clutch mounted on a shaft, the speed of which is controlled by the drive motor. A diagram of the device is shown in Fig. 9. Return-cycle carriage velocity results for a machine equipped with the clockspring kinematic governor system are shown in Fig. 10. (The high-frequency ripple on the signal is 60-Hz power line hum.)

Figure 11 Lumped-mass approximation of return mechanism.



• Simulation

During all of the simulation work on the scan portion of the copy cycle it was assumed that all parts of the system were at rest at the beginning of the acceleration of the document carriage. The accuracy of this assumption was investigated early in the program by simulating the end of the return portion of the copy cycle, where the carrier and pulley are decelerated by the cam. The drum is decoupled at that time and the lumped-mass approximation of the system is as shown in Fig. 11. Once again, a constant acceleration cam was specified; the differential equations are

$$m_B \ddot{x}_B = k_{PB} (x_P - x_B); \quad (7)$$

$$m_P \ddot{x}_P = F_P - k_{PB} (x_P - x_B) \quad (8)$$

where, as before

$$F_P = k_{MC} (x_M - x_C) / (dx_P/dx_C) \quad (9)$$

and

$$x_P = K x_C^2. \quad (10)$$

The amplitude of vibration of the document carriage relative to the motor, assumed to be a constant-velocity device, was investigated for varying values of the return velocity and for different cam accelerations. The simulation results showed the amplitude of the vibration induced in the system during the return cycle to be insignificant at the start of the succeeding scan cycle. The two factors that were instrumental in producing that result were the selection of a 0.1-g maximum carriage deceleration at the end of the return cycle and a nominal dwell time between the return and scan cycles of 0.3 s. The large dwell time was more than sufficient for the decay of vibration induced by the return motion.

Summary

The basic design requirements, which were an image resolution of at least 7.5 lines/mm and carriage accelerations to and from scanning velocity over the smallest practicable displacement, have been met or exceeded by the cable-and-pulley design used. The corollary criteria

for relative displacement between document carriage and drum and for acceleration during carriage return, initially assumed to be 0.002 in. and 0.1g respectively, were found to be conservative. A prototype, which had been modified as a result of simulation, was instrumented and found to produce image resolution of better than 9 lines/mm with a carriage-to-drum displacement of 0.003 in. Within the time required for the drum to rotate the 90° not used for scan, a clock-spring mechanism controls the return motion of the carriage through a three-part cycle (0.5g acceleration, constant velocity, and 0.1g deceleration) with enough time to spare for a nominal 0.3-s dwell before the following scan portion of the copy cycle. Although the initial return acceleration is higher than originally proposed, the complete return cycle is such that the induced vibration is damped out during the dwell interval.

The mechanism is compatible with requirements for safety and convenience, and has permitted the use of stationary (and hence efficient and economical) optical components in the machine.

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