

A Cylindrical Print Element System for a Serial Impact Printer

For an impact serial printer, like a typewriter terminal, it is desirable to print fast and to print with a minimum of maintenance. For printing to take place, the carrier containing the print element must be in the correct position along the print line and the character selection must be made by the proper positioning of the print element. This paper discusses a cylindrical thin-walled print element containing an entire character array on its outside surface. The print element is rotated and translated independently by two prime movers (motors). The cylindrical print element geometry and the prime movers are optimized to minimize the time required for character selection. Surface area, stress, inertia, torque, and motor requirements are considered. Selection times and printing rates are computed for simulated printing conditions.

Introduction

A serial impact engraved character printer as shown in Fig. 1 consists basically of five mechanical subsystems. They are the traverse system, the print-element mechanism, the hammer unit, the paper system, and the ribbon-incrementing unit. The traverse system moves the print head (print-element mechanism and hammer unit) across the paper (print line). The print element contains all the engraved characters and is positioned for the appropriate character to be printed by some prime mover. Then the hammer (actuator) causes the print element to impact the ribbon and paper, causing the image of the appropriate character to appear on the paper.

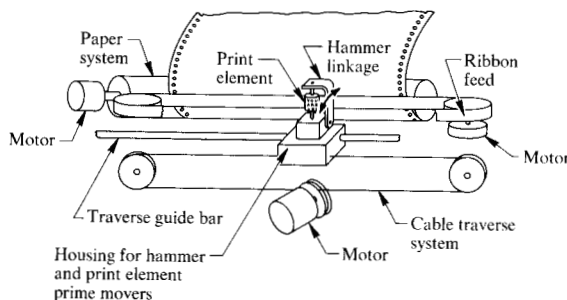


Figure 1 Serial impact engraved character printer.

An impact serial printer should print fast and with a minimum of maintenance. In order to accomplish this, it is desirable to incorporate the following design concepts:

1. Use of fewer parts by controlling the positioning of the print element (character selection) through a prime mover(s) such as an appropriate motor(s).
2. Use of optimum character placement on the print element.
3. Use of a low inertia print element.
4. Removal of elasticity between prime mover and the print element (*i.e.*, no cables, etc.)
5. Overlapping of the actual printing part of the total print cycle with the character-selection operation.
6. Printing while the print head is still in motion (printing on the fly).
7. Use of an inexpensive, replaceable print element.

A critical part of the entire print cycle, as shown in Fig. 2, is the positioning of the print element. Indeed, most of the ideas mentioned above relate directly to reducing the time required to position the print element. Print impact occurs after the print element is stationary. Various investigators [1, 2] have studied the optimal incrementing of an inertial load that is coupled to a drive motor. In those studies optimization is based upon incrementing time and/or motor size. This study involves the determi-

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nation of the incrementing time for a print element when the design of the entire print element system is constrained by such factors as print element surface area for character placement, impact stresses developed in the print element, torque capabilities of available motors, and their respective power dissipation limitations.

There are various possibilities for the shape of the print element: spherical (IBM ball), cylindrical, disk, etc. It is the purpose of this paper to investigate the optimum size of a cylindrical print element and to determine the resulting positioning times associated with this element when driven by an appropriate existing prime mover or motor.

A schematic diagram of the assumed print element positioning system, using a cylindrical print element, is shown in Fig. 3. As can be seen, two directions must be controlled independently for proper character selection. These two directions, rotational and vertical translation, can be controlled independently through the arrangement shown in Fig. 3. Vertical translation is accomplished by a rack-and-pinion type of arrangement with the pinion being driven by a motor. A square segment of the print element shaft is free to move vertically through the rotation gear. Rotation is achieved by an independent set of gears and drive motor. The upper part of the print element shaft is free to rotate in a bearing while the lower part of the shaft is constrained in rotation.

General requirements

It is desired to determine the optimum size of the cylindrical print element for the general configuration shown in Fig. 3. For this to be accomplished, certain design criteria or requirements must be established. The criteria established for the cylinder design are as follows:

1. The surface area should be large enough so that at least 96 characters can be accommodated and each character can be printed independently (no partial printing of adjacent characters).
2. The cylinder should be thick enough to prevent stress failures.
3. The motor size required for rotation should be comparable to the one required for vertical translation.
4. The incrementing time for rotation and translation should be comparable.
5. The positioning should be accomplished in a minimum of time. In addition, the consequences of various assumptions regarding optimum character placement must be investigated.

• Surface area

In order to avoid shadow printing (imaging of neighboring characters), certain minimum distances must exist be-

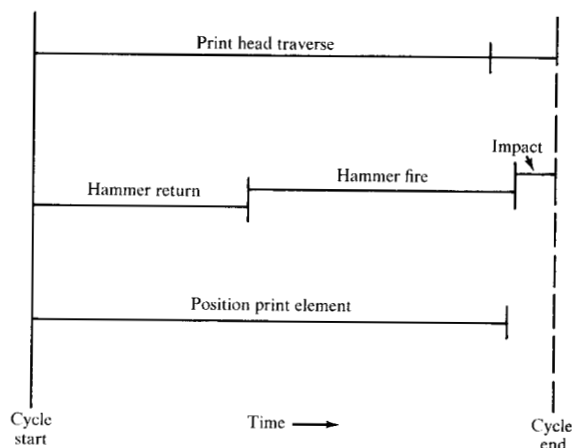


Figure 2 Overlapping operations for a serial printer.

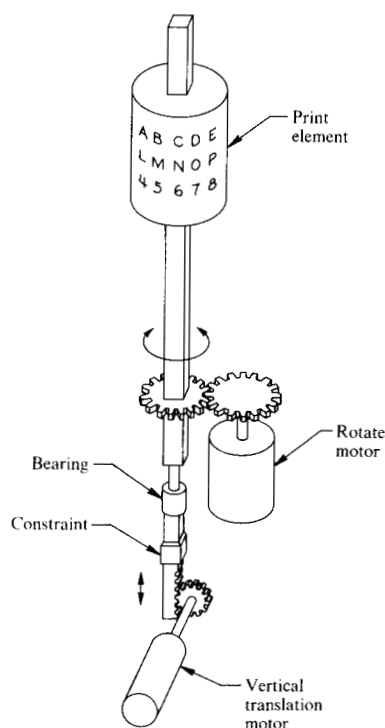


Figure 3 Print selection schematic.

tween adjacent characters both vertically and horizontally. Thus, there is a given amount of surface area of the cylindrical print element associated with each character. From previous experience it has been shown that the center to center distance between adjacent horizontal characters is approximately 0.28 cm or greater for cylinders of

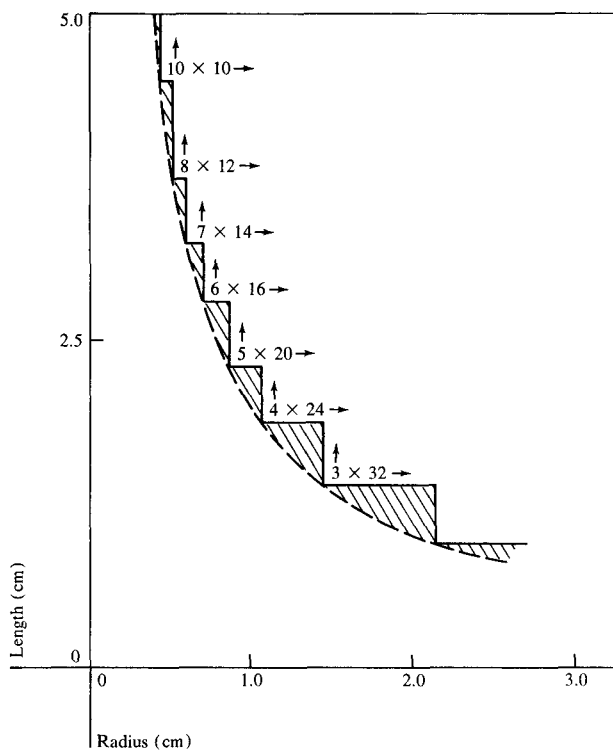


Figure 4 Surface distance requirements.

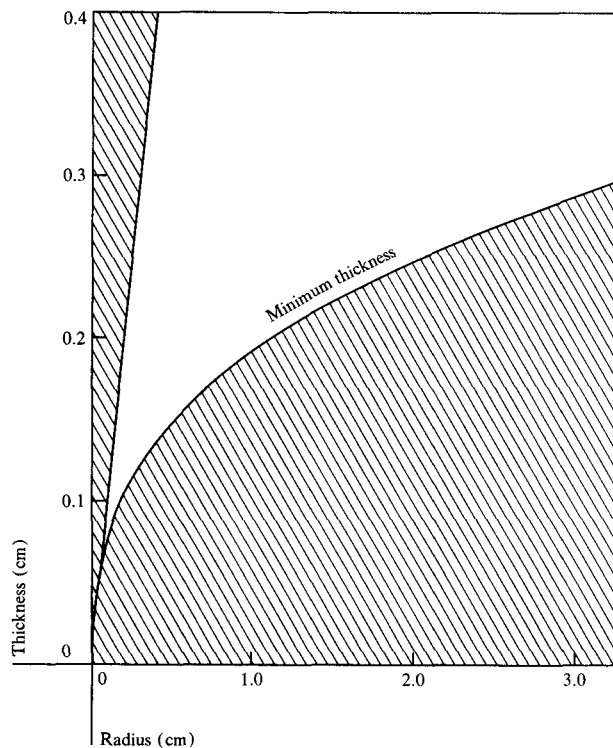


Figure 5 Stress requirements.

1 cm or less in radius. Thus, for a row of N characters,

$$R \geq 0.28(N/2\pi), \quad (1)$$

where R = cylinder radius (cm).

The center to center distance between vertically spaced characters should be greater than 0.49 cm. Thus, for a column of M characters,

$$\ell \geq 0.49M, \quad (2)$$

where ℓ = cylinder length. Since 96 characters are required,

$$NM \geq 96. \quad (3)$$

These relationships are plotted in Fig. 4. Admissible values for the cylinder's radius and length are those values above the curve of Fig. 4. The dotted line in Fig. 4 represents the minimum values for the radius and length of the cylinder, or

$$R\ell \geq 1.9. \quad (3a)$$

• Stress considerations

The stress developed in the cylindrical print element is determined in an approximate manner. The bending stress developed in the cylinder during impact is approximated by a long cylindrical shell loaded uniformly along a circular section. The resulting maximum bending stress [3] is

$$\sigma_{\max} = \frac{1.5F}{\sqrt[4]{3(1-\mu^2)}} \frac{1}{\sqrt{h^3/a}}, \quad (4)$$

where

$$a = \frac{R+r}{2},$$

F = print force/unit length,

r = inside radius of cylinder,

μ = Poisson's ratio,

h = thickness of cylinder.

Thus, the maximum stress, $\sigma_{\max}$, must be less than the endurance stress for the print element material, $\sigma_E \geq \sigma_{\max}$. Typically, the total print force is approximately 90 N for reasonable print quality. Also, the character length is approximately 0.25 cm. For high-strength plastics, typical values for Poisson's ratio and endurance strength are 0.3 and 45 MPa, respectively. Thus, the following relationship must be satisfied for stress considerations:

$$\frac{R - (h/2)}{h^3} \leq 120. \quad (5)$$

In addition, the radius can never exceed the thickness. These relationships are shown in Fig. 5. The cross-hatched areas indicate the inadmissible values for the cylinder radius and thickness.

● Mass and inertia relationships

From Fig. 4, it can be seen that for minimum rotational as well as translational inertia, the combination of length and radius to choose would always be at the intersection of the dashed line with the solid stepped line. This ensures that for a given array of characters (say, 4 rows and 24 columns) the minimum dimensions are chosen.

The mass of the cylinder is

$$m = \frac{\gamma\pi\ell}{g} (R^2 - r^2) = \frac{\gamma\pi\ell}{g} h(2R - h), \quad (6)$$

where

γ = material density,

g = gravitational constant.

For the minimum mass, Eq. (3a) is used, and

$$m_{\min} = \frac{\gamma\pi}{g} 1.9h(2 - h/R). \quad (7)$$

The minimum value for the thickness h is found from the relationship of Eq. (5). Combining this with Eq. (7) gives the data points shown in Fig. 6. The design curve obtained is shown by a dashed line, since the minimum values for mass exist at only the noted discrete points. Thus, for a given value of the cylinder radius, the minimum mass must be above the design curve of Fig. 6.

The relationship for the polar mass moment of inertia for a cylinder about its axis of rotation is

$$J = \frac{m}{2} [R^2 + (R - h)^2]. \quad (8)$$

By combining Eqs. (5), (7) and (8), a relationship between the minimum polar mass moment of inertia and radius is obtained. This is shown as a dashed line in Fig. 7. This relationship is also only true at the noted discrete points in the figure.

● Torque requirements

The cylinder must be rotated and translated vertically for proper character selection. Each of these motions (rotation and translation) is controlled independently. Thus, the cylinder must be accelerated and then decelerated to the new proper position (rotary and vertical). It is assumed that the motor/print element system is coupled in a stiff manner. The idealized acceleration time trace is a constant acceleration a_p to time $t_f/2$ and a constant decel-

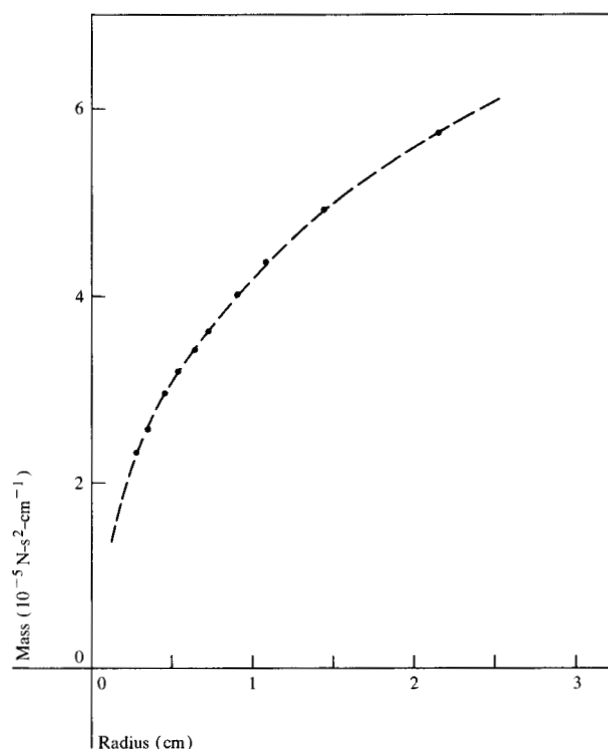


Figure 6 Minimum mass requirement.

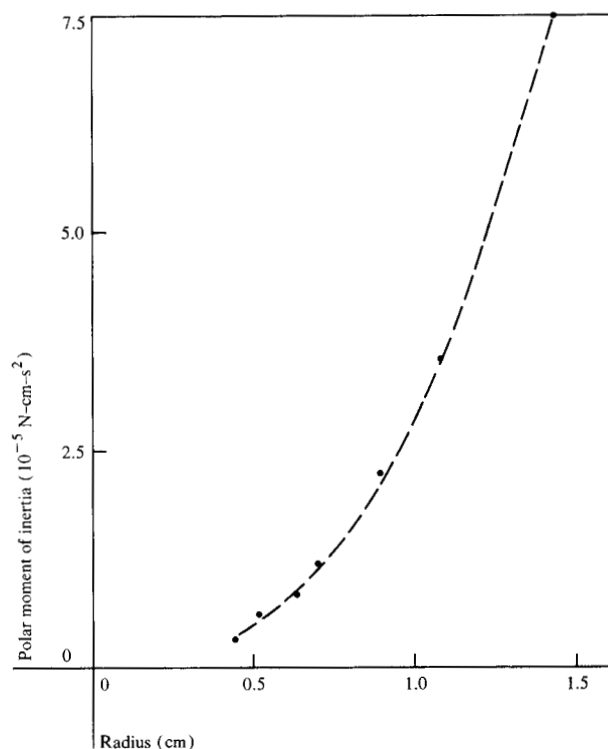


Figure 7 Minimum inertia requirement.

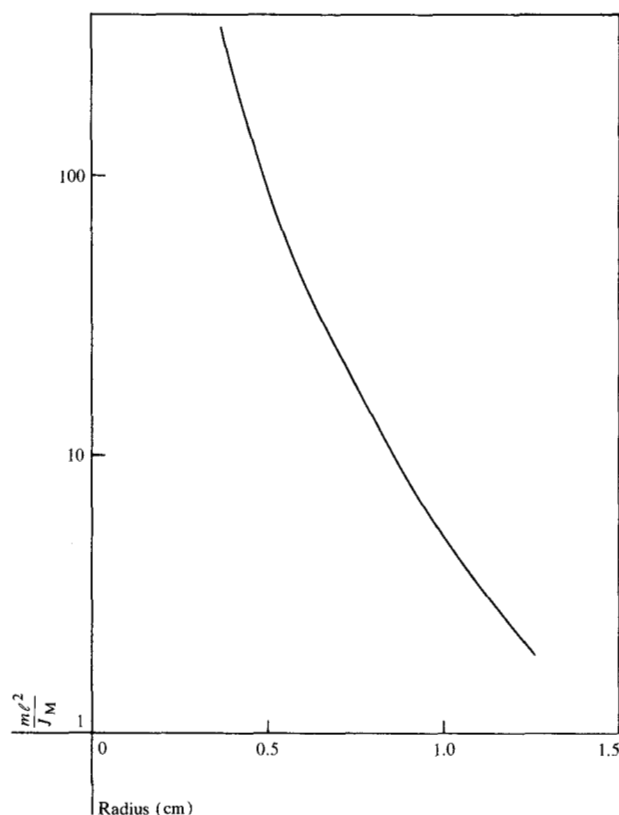


Figure 8 Inertia ratio for radius selection.

eration a_p from $t_f/2$ to t_f . Thus,

$$d = \frac{1}{2} v_{\max} t_f,$$

$$v_{\max} = a_p \frac{t_f}{2},$$

where

d = resultant displacement,

t_f = time for positioning,

$v_{\max}$ = maximum velocity obtained,

a_p = translational acceleration.

Thus,

$$a_p = \frac{2v_{\max}}{t_f} = \frac{4d}{t_f^2}. \quad (9)$$

In rotation,

$$\alpha_p = \frac{4\theta}{t_f^2}, \quad (10)$$

where

α = angular acceleration attained,

θ = resultant rotational displacement.

In rotation the equation of motion for the cylinder is

$$T_M = (J_L + J_M)\alpha_p, \quad (11)$$

where

T_M = torque provided by the motor,

J_L = polar mass moment of inertia of cylinder,

J_M = polar mass moment of inertia of motor.

The load is assumed to be rigidly coupled to the motor. Combining Eqs. (11) and (12),

$$T_M = (J_L + J_M) \frac{4\theta}{t_f^2}. \quad (12)$$

The rotation gears are assumed to be quite small compared to the load motor inertias, and are therefore not included in the calculation of inertia. For vertical translation, the driven mass is reflected back to the gear of the drive motor. Thus,

$$J_{\text{total}} = J_{MV} + r_G^2 m,$$

where

r_G = gear radius,

m = mass of cylinder,

J_{MV} = polar mass moment of inertia of vertical translation motor.

Thus, the governing equation of motion of the vertical translation motor is

$$T_{MV} = (J_{MV} + r_G^2 m) 4 \left(\frac{d}{r_G} \right) / t_f^2. \quad (13)$$

In order to be able to select any character, the cylinder must be able to be rotated $\pm 180^\circ$ and be translated a distance of $\pm \ell$ within the selection time. Thus, the torque supplied by the rotation motor, T_M , is defined as follows:

$$T_M \geq (J_L + J_M) 4\pi / t_f^2. \quad (14)$$

Similarly for the translation motor,

$$T_{MV} \geq (J_{MV} + mr_G^2) 4 \left(\frac{\ell}{r_G} \right) / t_f^2. \quad (15)$$

Differentiating T_{MV} with respect to r_G and setting the result equal to zero, we obtain

$$r_G = \sqrt{J_{MV}/m}. \quad (16)$$

This is the value of r_G such that a minimum torque is required to be produced by the translation motor. It is interesting to note that this thus requires the motor inertia to equal the equivalent load inertia. Combining Eqs. (15) and (16) results in

$$T_{MV} \geq 8\sqrt{mJ_{MV}} \ell / t_f^2. \quad (17)$$

For relatively short increment times [1] it has been shown that, for minimum motor size and input power re-

quired, the effective load inertia and motor inertia should be approximately equal. Therefore, Eq. (14) becomes $T_M \geq 8\pi J_M/t_f^2$.

In addition, the same size motors should be used for translation and rotation. Therefore, $J_M = J_{MV}$, and $T_M = T_{MV}$; thus

$$8\sqrt{mJ_M} \ell t_f^2 = 8\pi J_M/t_f^2,$$

$$\sqrt{m\ell^2/J_M} = \pi,$$

$$m\ell^2/J_M = \pi^2. \quad (18)$$

The ratio $m\ell^2/J_M$ can be plotted as a function of the cylinder radius R by using Eqs. (4), (5), (7), and (8). This is shown in Fig. 8. The ratio $m\ell^2/J_M = \pi^2$ corresponds to a radius R of approximately 0.86 cm. The admissible cylinder would then have a radius of approximately 0.91 cm. This would correspond to a cylinder of five rows of characters, with each row having 20 characters. The inertia of this cylinder is 2.81×10^5 N-cm-s².

The selection of this cylinder in part is predicated upon the requirement that character selection be accomplished for any possible combination of two successive characters within the selection time t_f ; thus, rotation of $\pm 180^\circ$ and translation of $\pm \ell$. This, however, may not be a realistic requirement, and in fact is quite stringent. In another study [4], it was shown that approximately one-fourth of the characters are used over 90 percent of the time. Thus, if these high-usage characters are placed near one another on the print element, the selection requirements can be relaxed. For example, if only $\pm 90^\circ$ rotation and $\pm \ell$ translation is required with the same nominal selection time, Eq. (18) changes to

$$8\sqrt{mJ_M} \ell/t_f^2 = 8\left(\frac{\pi}{2}\right)J_M/t_f^2,$$

or

$$\frac{m\ell^2}{J_M} = \frac{\pi^2}{4}. \quad (19)$$

Looking to Fig. 8, this would correspond to a radius of 1.18 cm. The closest admissible cylinder would then have a radius of 1.09 cm with four rows and 24 characters in each row. Thus, for different requirements for rotation and translation, different size print elements would be used.

Motor requirements and motor selection

As seen above, the translational and rotational requirements determine the size of the cylindrical print element. The next step is to select the appropriate drive motors. Since comparable drive motor sizes are desired for each

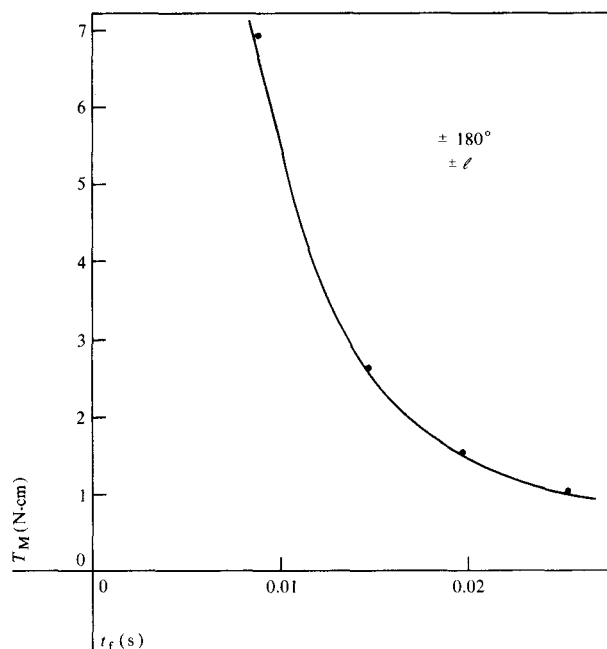


Figure 9 Motor torque requirements.

motion, we need only look at, say, the rotational requirements to select a motor. Since the rotational inertia of the print element has now been determined and the driven and drive inertias are matched, Eq. (12) can be used to determine the relationship between motor size and increment time. A plot of this relationship is shown in Fig. 9, where the rotational requirement is $\pm 180^\circ$ rotation and the translational requirement is $\pm \ell$. The greater the rotation (or translation), the greater the required size (torque capability) of the incrementing motors. Similarly, for faster increment times, larger torque capability motors are needed.

As mentioned previously, the characters should be arranged on the print cylinder such that high-usage characters are located close to one another in order to minimize the increment time of the print element. An APL program was written to simulate the operation of the print element. The characters were arranged on the print element in an optimum manner and various statistical print patterns were simulated.

The motor constants for various appropriate drive motors were read into the computer program and the resulting print element increment times were computed. The torque output of the motors under investigation was limited by the allowable power dissipation of the motors. The average printing rate was calculated for each motor.

From the computer simulations, printing rates as high as 60 characters per second appear to be achievable. In fact, subsequent laboratory experiments performed with a larger equivalent print element inertia and with a proportionately larger drive motor substantiate the computer simulation.

Acknowledgment

The author thanks Charles Ross, who wrote the APL program for simulation of the printer operation.

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