The Premier Advanced Recording Technology Forum

# Method to Mechanically Filter Out Longitudinal Tension Spikes Induced in a Flexible Media Tape Unit

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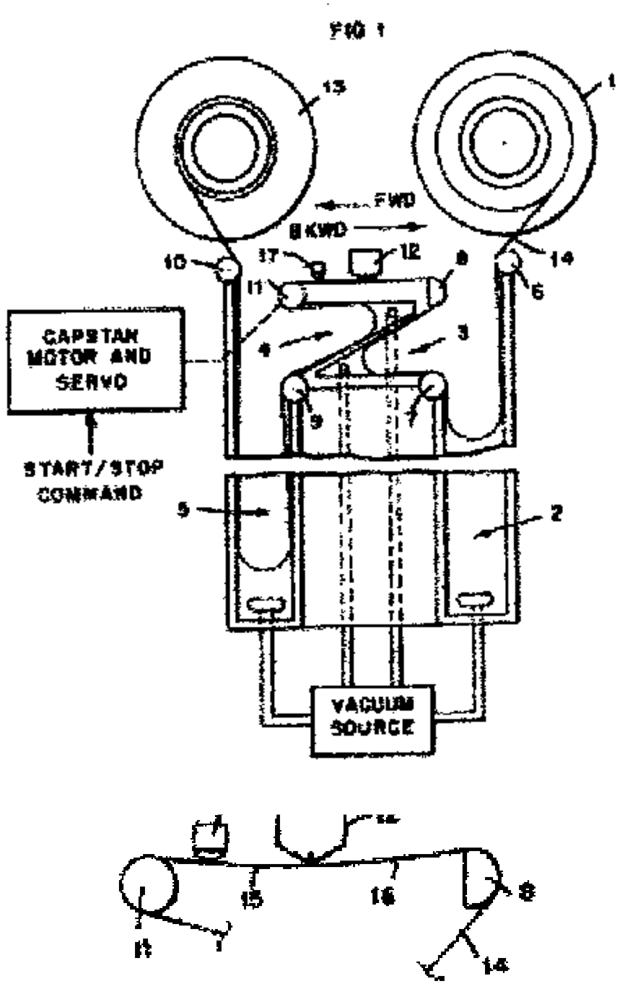
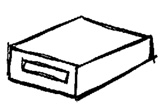
# VACUUM COLUMN TO ROLLER TAPE PATH DESIGN HISTORY

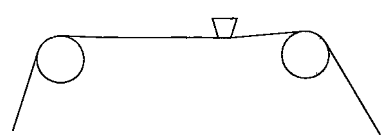



- Large Reel tape drives had expensive pneumatic systems
  - Many components
  - Low reliability
  - Required lot of space
  - Low capacity
    - 6,250 FRPI (246 FRPMM) for 40 MB capacity on 10.5" reel

# VACUUM COLUMN TO ROLLER TAPE PATH DESIGN HISTORY

- Most newer tape drives use rollers
  - DLT, LTO
  - Fewer components
  - Higher reliability
  - Small form factor
  - High capacity
    - 100,000+ FRPI (3940 FRPMM) for 40,000 MB Capacity cartridge
    - LTO at 100 GB





# ROLLER TAPE PATH DESIGN CONSIDERATIONS

- Rollers Introduce High Inertia Components Into Media Paths
- Servo System Designers Have Held Onto Fast Access Time Culture
- High Accelerations Stretch the Media With Fast Starts
- Instant Stretching of the Media  Occurs in the Form of Tension Pulses
- Pulses Travel Down the Media Length-wise at the Speed of Sound
- Rollers Will Reflect These Pulses if NO SLIP is Occurring Between Media and the Roller Surface
- HOW DO THESE WAVES GET GENERATED ?

# THEORY OF LONGITUDINAL VIBRATIONS IN MEDIA

- First known look at impact of longitudinal waves done at IBM-Boulder
- Documented in the IBM Journal of Research and Development May 1972
- IBM interested in vacuum column tape drives---OF COURSE

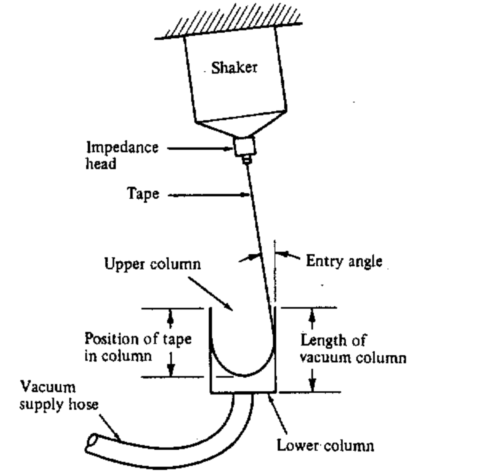
G. W. Baumann

**Viscoelastic Behavior of Computer Tape Subjected to Periodic Motion**

**Abstract:** The purpose of this study was to develop a theoretical means for predicting the longitudinal motion of computer tape in a high-performance tape drive. In particular, this paper treats the motion that is governed by traveling velocity-stress wave reflections, attenuations, and interactions in the length of tape between the tangency point at the capstan and the tangency point at the stubby column in the drive.

The motion of the tape was determined by solving the classical, damped, one-dimensional wave equation subject to the appropriate boundary conditions. J. C. Snowdon's low-damping constitutive model was used to describe the viscoelastic behavior of the tape. The solutions for simple boundary conditions were experimentally verified by mechanical impedance techniques. More complex boundary conditions, such as those for vacuum columns, were experimentally studied to determine the true mathematical boundary conditions.

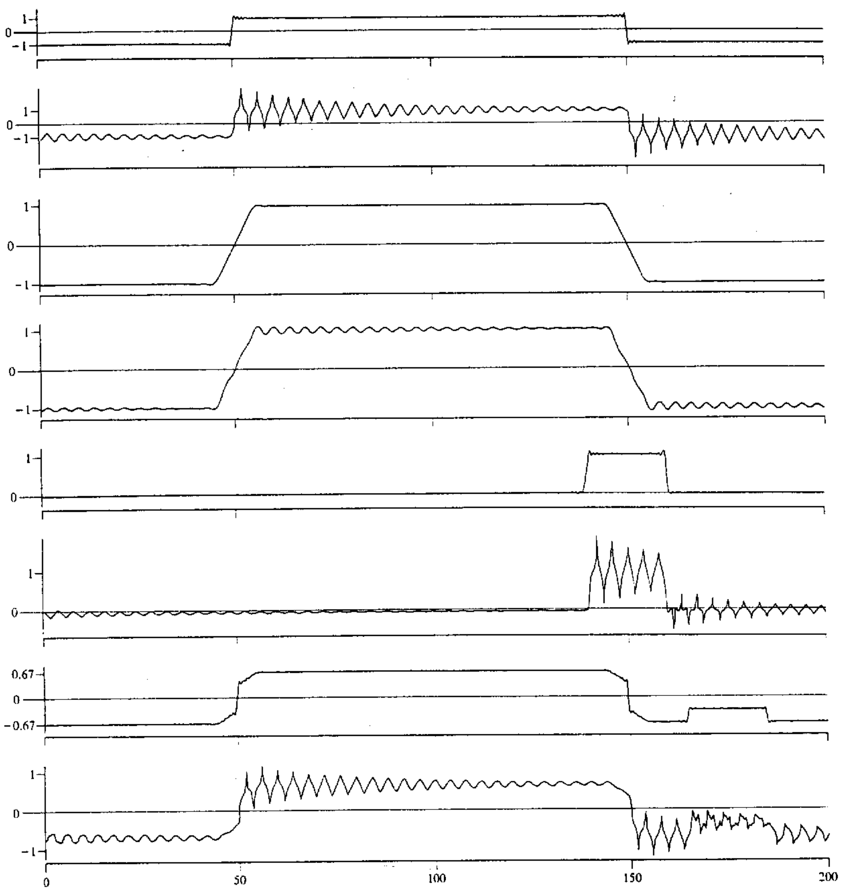
This paper also discusses simple unreflected harmonic waves, simple reflected harmonic waves, and general periodic reflected waves as examples. The significance of the wave interactions in the design of tape drives is considered.




*General Products Division, International Business Machines Corporation, Boulder, Colorado, 80302.

** The length of the resonator is the length of the upper column, or the sum of the lengths of the lower column, the hose, and the plenum.

# MEASURED RESULTS FROM VACUUM COLUMN TESTER




- Observations
  - Waves are generated because of the mass of the media being moved
  - Some Coulomb friction was noted at the interface of the media loop and column side walls
  - Steep input generates more response amplitude
  - There are no reflections involved due to vacuum column termination

# MEASURED RESULTS FROM VACUUM COLUMN TESTER

- Observations (continued)
  - Softer input generates less response amplitude
  - Vacuum column dampens oscillations

- WHAT HAPPENS WHEN ROLLERS ARE INTRODUCED INTO THE PATH ?

# EFFECTS OF ROLLERS IN TAPE PATH

- Complete Handbook of Magnetic Recording by Finn Jorgensen 4th Edition, and A. Walraven IEEE Transaction Sep 1969 both agree on waves' existence
- Both describe the existence of longitudinal waves and how to determine values
- Associate these waves with playback read FLUTTER conditions

# EFFECTS OF ROLLERS IN TAPE PATH

**The Analysis of Longitudinal Tape Vibrations**

A. WALRAVEN

INTRODUCTION

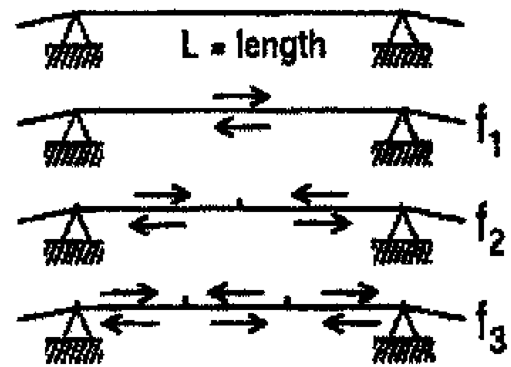
An analysis of the longitudinal tape vibrations in a magnetic recorder as caused by random head-to-tape friction is presented. The occurrence of these vibrations is known to be a major cause of high-frequency flutter (scrape flutter or FM noise), as has been reported by several authors [1]–[3].

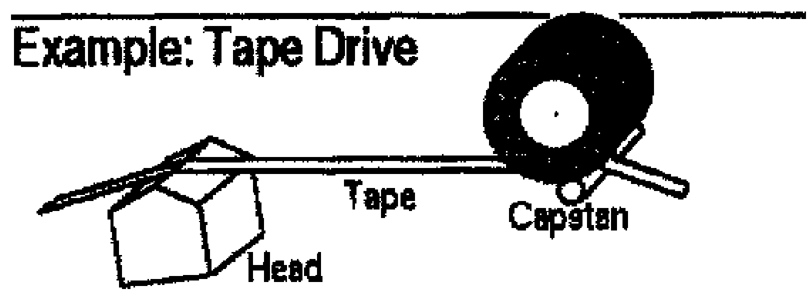
These vibrations are generally found to be of random character and to have a spectrum dominated by the fundamental longitudinal resonant frequency

$$\omega_0 = (\pi/l)\sqrt{E_d/\rho} \qquad (1)$$

where $l$ is the length of the vibrating tape span, $\rho$ its mass per unit length, and $E_d$ the dynamic value of Young's modulus. As a rule, $E_d$ is taken equal to the static modulus multiplied by an appropriate constant depending on the tape base material [2], [5].

The present paper assumes a known friction force spectrum and gives a calculation for the resulting velocity flutter spectrum for various head positions along the tape span. This calculation takes account of the frequency dispersion and the mechanical dissipation in the tape by the use of the complex modulus $\bar{E}(j\omega)$.




- ROLLERS WILL REFLECT THE WAVES IF NO SLIP CONDITION EXISTS. THEN WHAT ?

# DESIGN FOR LOW FLUTTER per Jorgensen

- Design tape path with short spans between components
- Add high inertia rollers to absorb vibrations
- Place head at point where tape speed Is smoothest- normally as close to capstan as possible
- Rubbing against head and fixed guides induces longitudinal vibrations - so keep surfaces as smooth as possible
- Minimize wrap around head gaps
- <u>A ROTATING HIGH INERTIA ROLLER WILL SERVE AS A GROUNDING POINT</u>

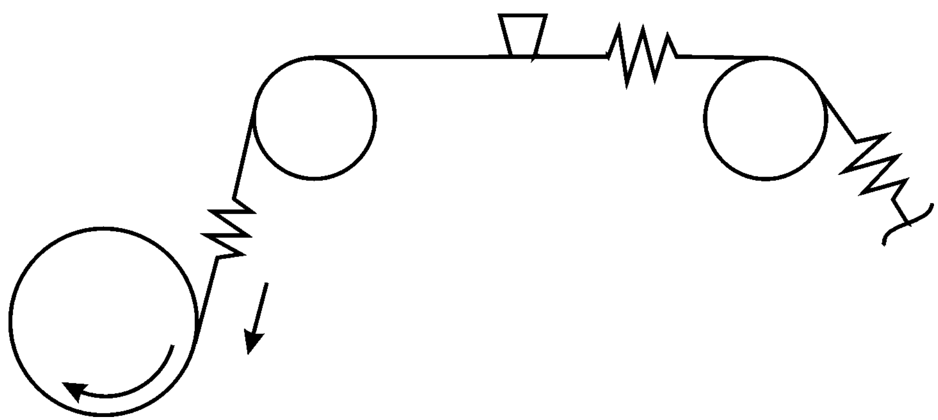
# DESIGN FOR LOW FLUTTER per Cope

- Rollers that do not slip the moving media will reflect the wave back
- Today's heads and guide surfaces are smooth enough
- Longitudinal waves are generated by 'hard' starting reel motors
- Use a <u>Non-Slipping Roller</u> to isolate waves from ever reaching the head
  - Use roller with porous surface
  - Increase roller to media friction coefficient
  - Increase media wrap angle

Porous surface

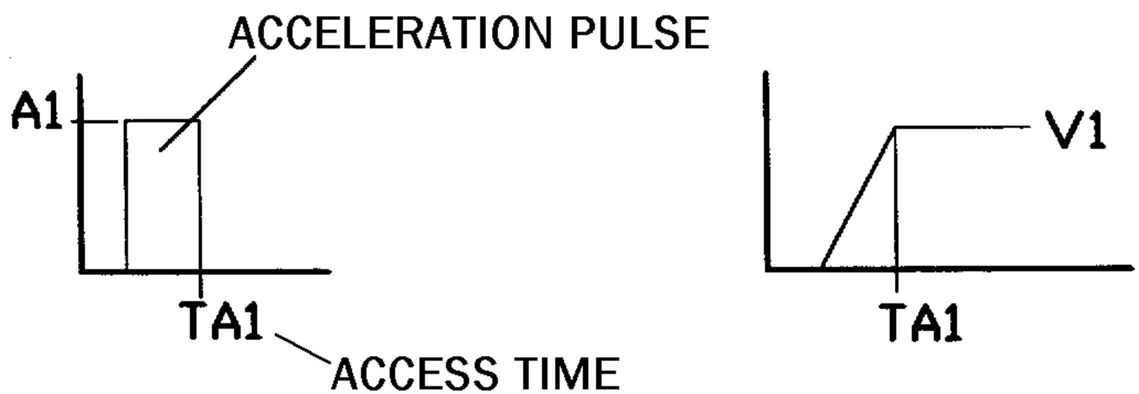
# REFLECTED LONGITUDINAL WAVES IN ROLLER TAPE PATHS

- Today's reel to reel servo control systems start HARD
- Examples are the Quantum DLT family and Benchmark's DLT1
- Tape segments have effective spring constants as shown



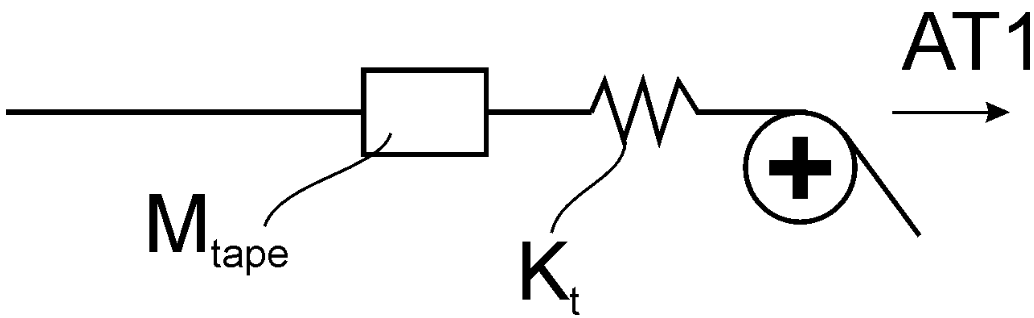
# MECHANCAL EFFECTS OF A TENSION SPIKE

- EXAMPLE
  - Acceleration of start of reel to reel servo system is:



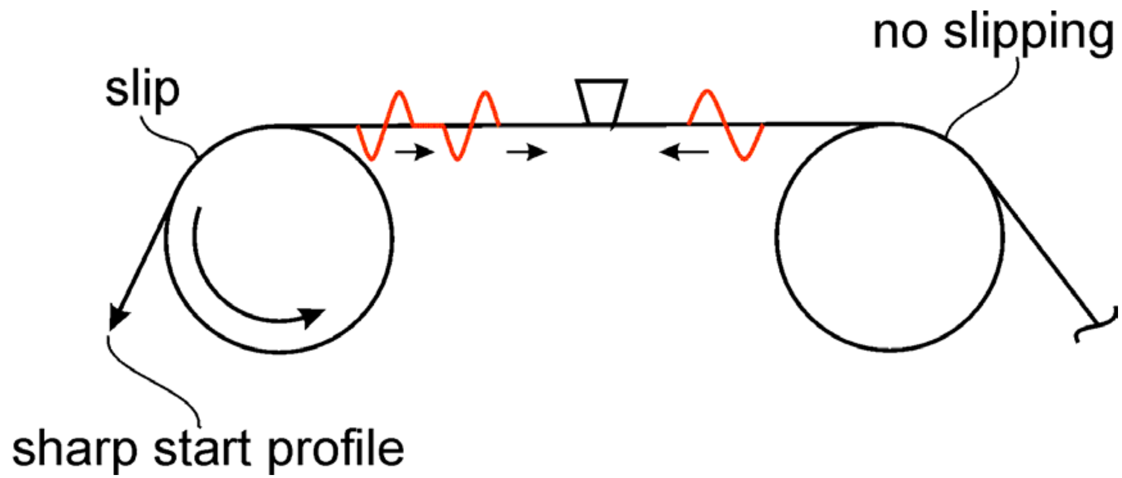

- If the TIME TO APPLY reel motor current in order to reach velocity of 120 IPS (3 m/s) is 1 ms (ala DLT1 with their PWM control scheme*)
- Acceleration level (A1)has to be 120,000 IPSS (3,000 MPSS) or 310 g
- *Personal Observation of Motor Current Envelopes were in this range.

# MECHANICAL EFFECTS-Continued




- WHAT HAPPENS WHEN THE PULSES TRAVEL DOWN THE MEDIA PATH ? . . .

# REFLECTED WAVES PHENOMENA




- A sharp start of media motion will cause a tension pulse
- This pulse is caused by stretching of the media according to the spring constant

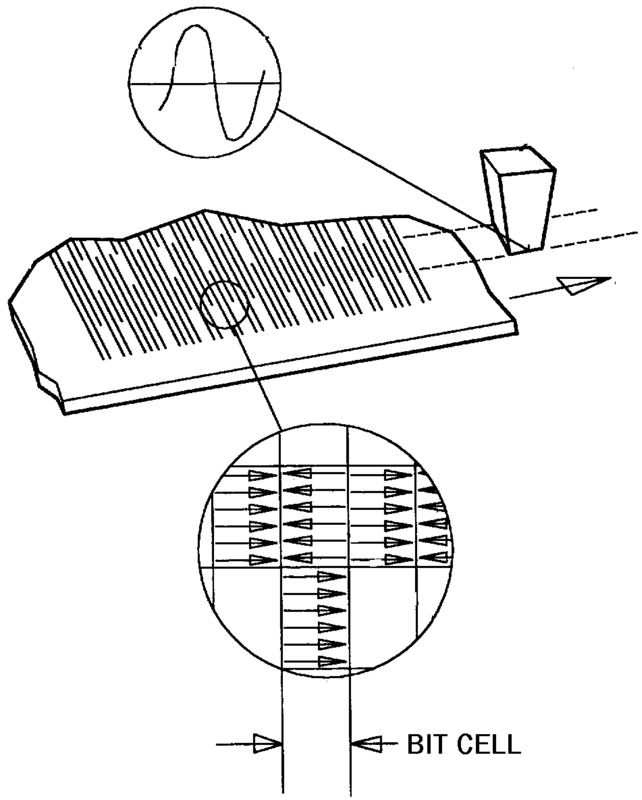
# REFLECTED WAVES PHENOMENA

- If the media slips over the roller during the start it will travel down the path at the speed of sound through the media material
- If the media does not slip over the roller, the pulse will be reflected
- The inertia of the roller acts like a wall bouncing back a tennis ball

.

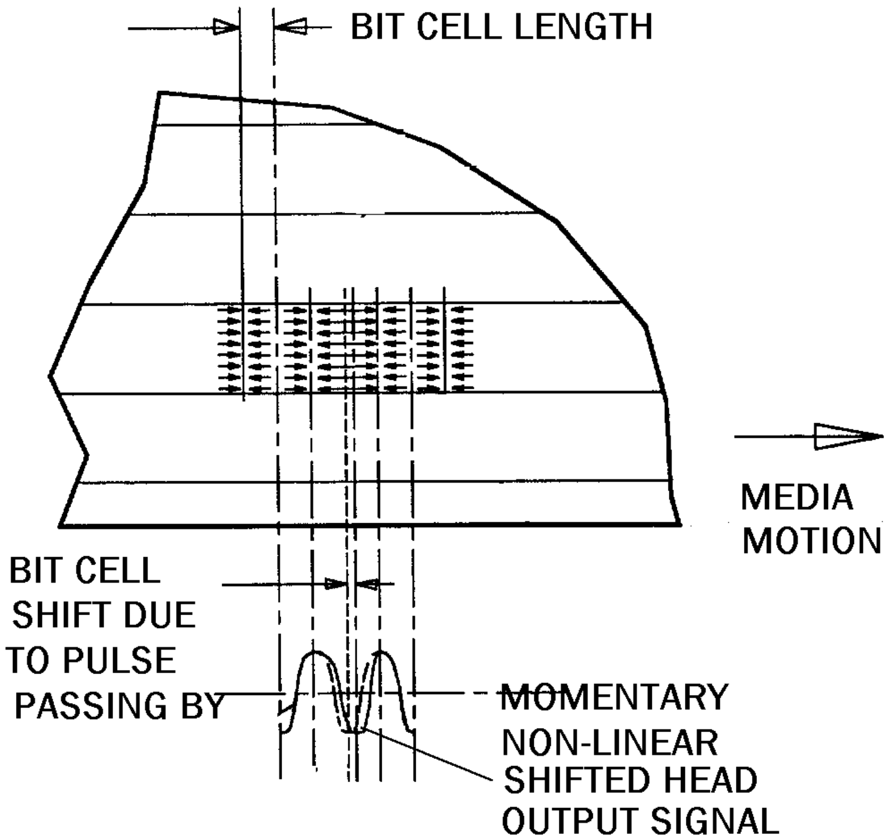
- HOW DOES THE WAVE PULSE IMPACT THE PLAYBACK SIGNALS ? . . .

# NORMAL SETUP TO READ THE PLAYBACK SIGNAL




- THEN WHEN A TENSION PULSE ROLLS THROUGH THE PATH  WHAT HAPPENS ? . . .

# REFLECTED WAVE IMPACT-Continued




The result is a large number of non-linear timing variations that the channel must decode into data with waves coming from both directions

# DESIGN RECOMMENDATIONS

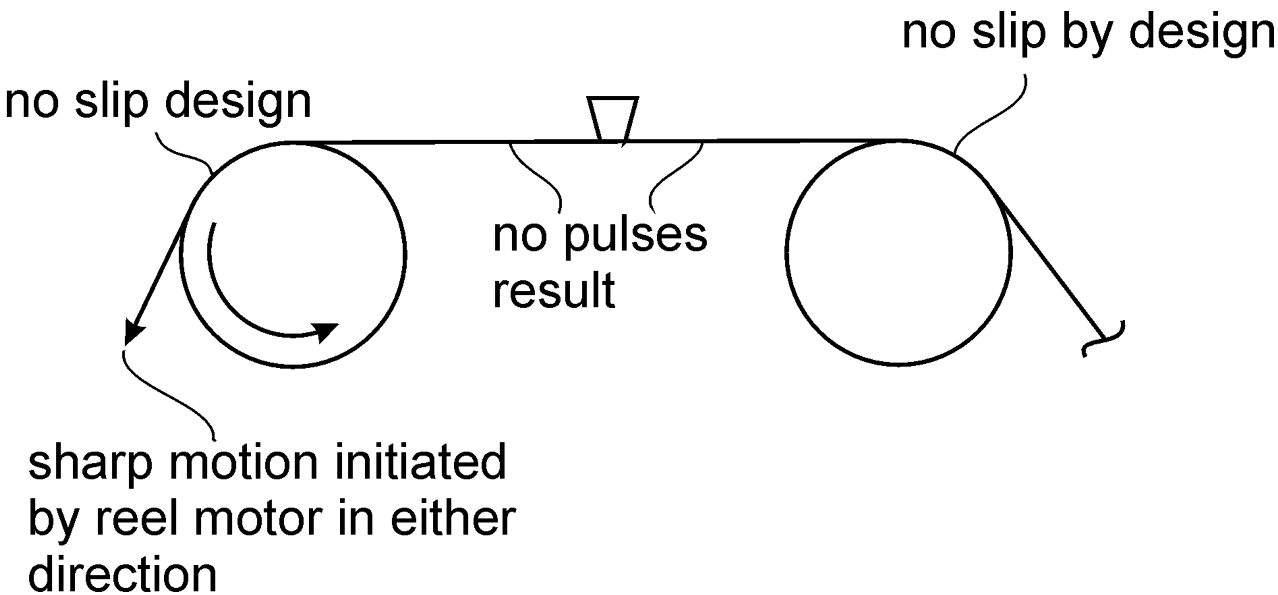
- Design the Tape Unit's Tape Path According to the Recommended Guidelines

OR:

- Limit the Acceleration of the Media so NO SLIP occurs on at least one Roller on either side of the Read/Write Sensor

- HOW IS THE SECOND DESIGN APPROACH DONE ?

# METHOD TO PROTECT R/W HEAD FROM TENSION SPIKES




- HOW CAN A ROLLER BE MADE TO NOT SLIP? . . .

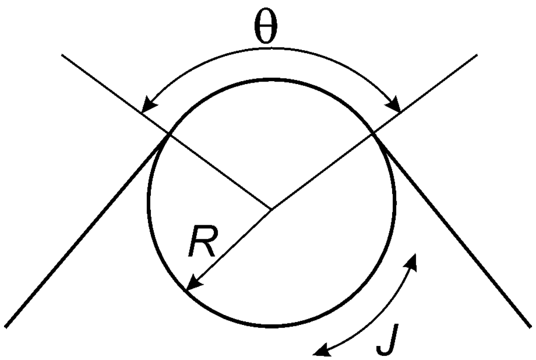
# DESIGN METHOD-Continued

- Lower the acceleration level of the reel-to-reel servo system so the media will not slip on the roller
  - Do the above for at least one roller on each side of the head
  - This will block longitudinal waves from passing the roller to the head, independent of media direction
  - Insuring that none of the rollers in the tape path slips, increases the design margins

- SO WHAT IS THE DESIGN METHOD TO BE FOLLOWED ? . . .

# DESIGN METHOD TO BLOCK TENSION SPIKES IN PATH

1. Determine wrap angles ($\theta$ of all rollers in tape path (radians))
2. Measure the radius (*R*) of each roller (mm)
3. Measure the rotational inertia (*J*) of each roller in path (mm-Kg-s$^2$)




- WHAT DOES TEST FIXTURE LOOK LIKE FOR INERTIA MEASUREMENTS ? . . .

# DESIGN METHOD - Continued

**1. VIBRATION-FREQUENCY CALCULATION**

The following simple cases of calculation of natural vibration frequency are given for reference. Two basic formulas are:

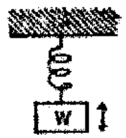


FIG. 1. Linear vibration.

**Linear vibration for spring-supported weight (Fig. 1):**

$$f = \frac{1}{2\pi}\sqrt{k/M} \qquad (1)$$

**Torsional vibration for disk on rod (Fig. 2):**

$$f = \frac{1}{2\pi}\sqrt{L_1/I} \qquad (2)$$

where $f$ = frequency, cycles per second; $M$ = mass, pound inch$^{-1}$ second$^2$ = $W/g$; $W$ = weight, pounds; $g$ = acceleration of gravity = 386 inches per second$^2$; $k$ = stiffness constant of spring = force required to stretch it to unit distance, pounds per inch; $L_1$ = same constant in torsion or torque required to twist spring or rod through 1 rad of arc, inch-pounds per radian; $I$ = moment of inertia, pound-inch second$^2$ = $WR^2/g$; $W$ = weight of body, pounds, executing torsional vibrations; $R$ = radius of gyration, inches. Equation 1 is used more often than eq. 2, as most problems involve linear, rather than torsional, vibrations.

By substituting the value of $g$ for the inch-pound-second system, eqs. 1 and 2 reduce to:

**Linear vibrations:**

$$f = 3.13\sqrt{k/W}\text{ cps} \qquad (3)$$

**Torsional vibrations:**

$$f = 3.13\sqrt{L_1/WR^2}\text{ cps} \qquad (4)$$

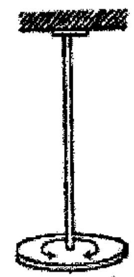


FIG. 2. Torsional vibration.

**GRAVITY DEFLECTION FORMULA.** For a spring carrying a weight (Fig. 1) causing extension $d$ of the spring, elastic constant $k$ must equal $W/d$ = weight per unit deflection. Substituting this in eq. 3,

$$f = 3.13\sqrt{1/d} \qquad (5)$$

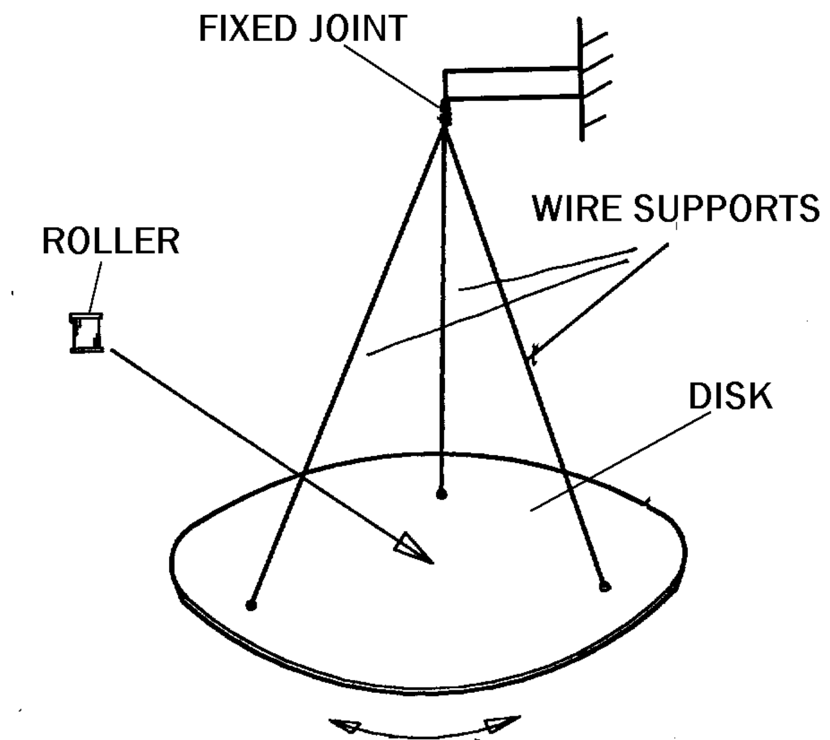
where $f$ = cycles per second and $d$ = inches deflection. Thus, if a weight causes an elastic support of relatively small weight to deflect downward through $d$ inches, eq. 5, at once gives the up-and-down frequency. This formula is useful for quick, approximate

9-02

SOURCE; MACHINERY'S HANDBOOK, 23RD EDITION

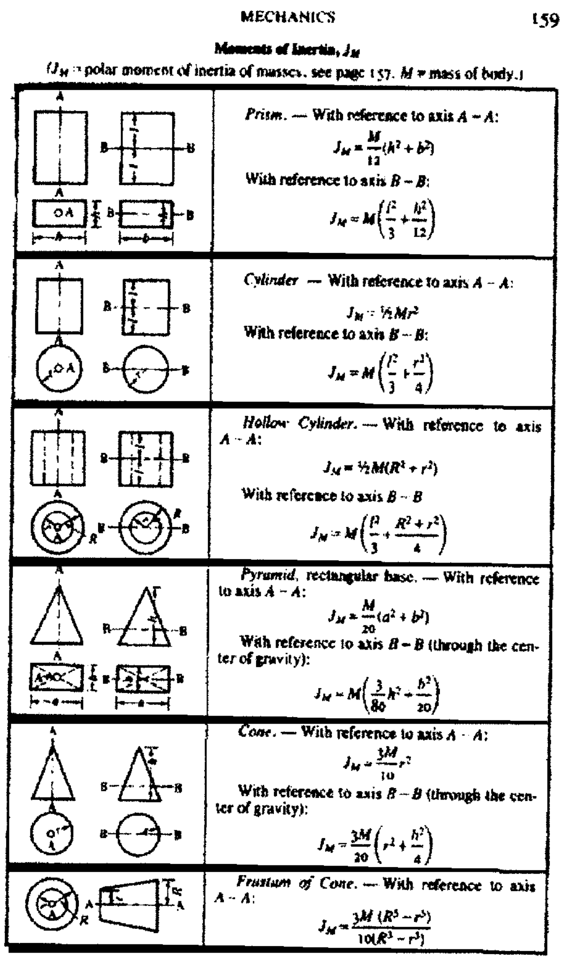
- HOW DO YOU BUILD AN INERTIA FIXTURE ? . . .

# DESIGN METHOD - Continued




- HOW DO YOU DETERMINE THE SPRING CONSTANT OF THE TOOL ABOVE ? . . .

# DESIGN METHOD - Continued



MECHANICS 159

**Moments of Inertia, $J_M$**

($J_M$ = polar moment of inertia of masses, see page 157. $M$ = mass of body.)

*Prism.* — With reference to axis $A-A$:

$$J_M = \frac{M}{12}(h^2 + b^2)$$

With reference to axis $B-B$:

$$J_M = M\left(\frac{l^2}{3} + \frac{h^2}{12}\right)$$

*Cylinder* — With reference to axis $A-A$:

$$J_M = \tfrac{1}{2}Mr^2$$

With reference to axis $B-B$:

$$J_M = M\left(\frac{l^2}{3} + \frac{r^2}{4}\right)$$

*Hollow Cylinder.* — With reference to axis $A-A$:

$$J_M = \tfrac{1}{2}M(R^2 + r^2)$$

With reference to axis $B-B$

$$J_M = M\left(\frac{l^2}{3} + \frac{R^2 + r^2}{4}\right)$$

*Pyramid*, rectangular base. — With reference to axis $A-A$:

$$J_M = \frac{M}{20}(a^2 + b^2)$$

With reference to axis $B-B$ (through the center of gravity):

$$J_M = M\left(\frac{3}{80}h^2 + \frac{b^2}{20}\right)$$

*Cone.* — With reference to axis $A-A$:

$$J_M = \frac{3M}{10}r^2$$

With reference to axis $B-B$ (through the center of gravity):

$$J_M = \frac{3M}{20}\left(r^2 + \frac{h^2}{4}\right)$$

*Frustum of Cone.* — With reference to axis $A-A$:

$$J_M = \frac{3M\,(R^5 - r^5)}{10(R^3 - r^3)}$$

SOURCE; MACHINERY'S HANDBOOK, 23RD EDITION

Process:

1. Measure natural period (T) of oscillation of tool's disk. Invert for frequency (f)

$$f = \frac{1}{T}$$

2. Calculate the rotational inertia of a disk from table left (Jdisk)

$$Jdisk = \frac{1}{2} M_{disk} * radius^2$$

3. Determine the spring constant of the tool (Ktq) from the formula

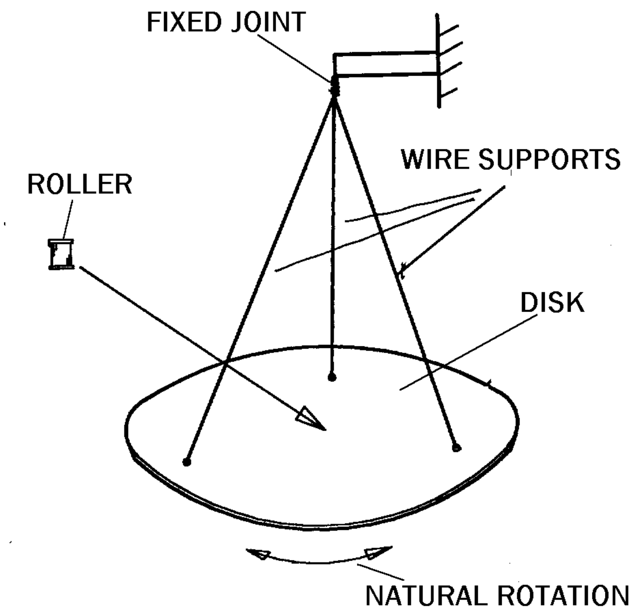
$$f = \left(\frac{1}{2 * \pi}\right) * \sqrt{\frac{Ktq}{J_{disk}}}$$

*Where J, $\pi$, and f are know values*

$$Ktq = \frac{4 * \pi^2 * j_{disk}}{T_{disk}^{\ 2}}$$

4. THEN Ktq is a known value for the tool

# DESIGN METHOD - Continued




5. Place the roller in the center of the tool's disk
6. Re-measure the increased natural rotation ($T_{both}$)
7. Calculate the increased frequency

$$f_{both} = \frac{1}{T_{both}}$$

8. With the known Ktq calculate the total inertia of both pieces

$$J_{both} = Ktq * T_{both}{}^{2}$$

9. Subtract the disk's inertia and you obtain the inertia of the roller

$$J = J_{both} - J_{disk}$$

# DESIGN METHOD - Continued

2228 FRICTION BRAKES

**FRICTION BRAKES**

**Formulas for Band Brakes.** — In any band brake, such as shown in Fig. 1, in the tabulation of formulas, where the brake wheel rotates in a clockwise direction, the tension in that part of the band marked $x$ equals $P\frac{1}{e^{\mu\theta}-1}$

The tension in that part marked $y$ equals $P\frac{e^{\mu\theta}}{e^{\mu\theta}-1}$

$P$ = tangential force in pounds at rim of brake wheel;
$e$ = base of natural logarithms = 2.71828;
$\mu$ = coefficient of friction between the brake band and the brake wheel;
$\theta$ = angle of contact of the brake band with the brake wheel expressed in radians (one radian $= \frac{180 \text{ deg.}}{\pi \text{ radians}} = 57.296 \frac{\text{deg.}}{\text{radian}}$).

For simplicity in the formulas presented, the tensions at $x$ and $y$ (Fig. 1) are denoted by $T_1$ and $T_2$ respectively, for clockwise rotation. When the direction of the rotation is reversed, the tension in $x$ equals $T_2$, and the tension in $y$ equals $T_1$, which is the reverse of the tension in the clockwise direction.

The value of the expression $e^{\mu\theta}$ in these formulas may be most easily found by using a hand-held calculator of the scientific type; that is, one capable of raising 2.71828 to the power $\mu\theta$. The following example outlines the steps in the calculations.

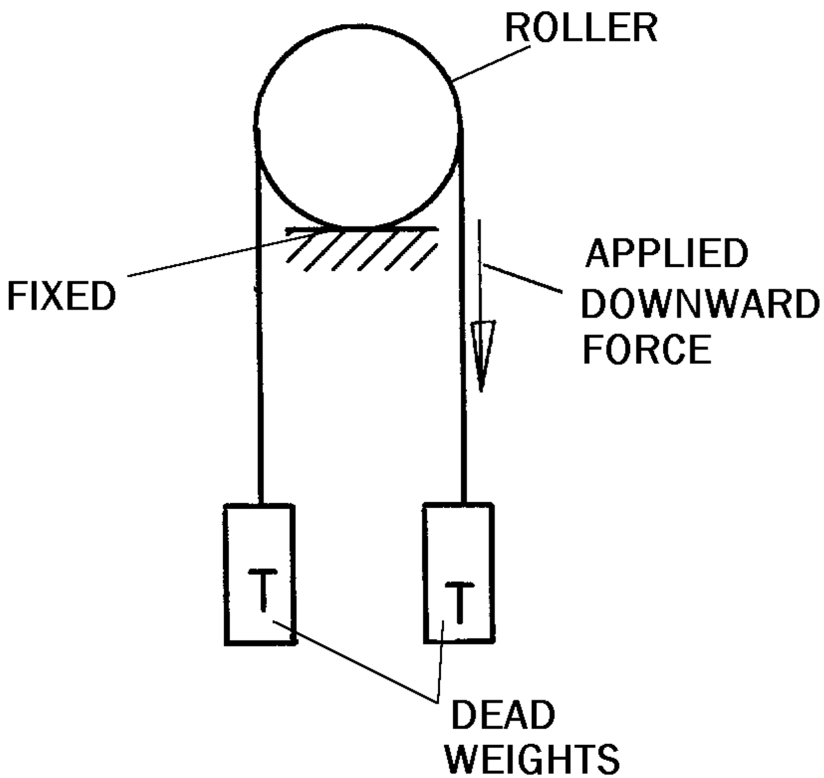
10. Determine the coefficient of friction ($\mu$) between the roller outer surface and the media as follows:
11. Assume a non-slipping roller acts like a band brake
12. Using the relationship well known in the brake industry as shown left, one can calculate the coefficient of friction ($\mu$)

$$F2 = F1 * e^{\mu * \theta}$$

SOURCE; MACHINERY'S HANDBOOK, 23RD EDITION

- What is the process to determine $\mu$ ?....

# DESIGN METHOD - Continued




13. Fix roller as shown left
14. Wrap media over top of roller
15. Apply dead weights (T) to simulate operational media winding tension
16. Note that the wrap angle ($\theta$) is fixed at 180° ($\pi_{rad}$)
17. Slowly apply downward force (*f*) on one side of the media until media slips on the roller surface
18. Calculate the coefficient of friction from the formula

$$\mu = \ln \frac{\frac{F+T}{T}}{\theta}$$

- THEN WHAT IS THE SLIP THRESHOLD ? . . .

# DESIGN METHOD - Continued

19. Since the coefficient of friction ($\mu$) between the operational rollers and media is known. Then the slip threshold can be determined for each roller in the tape path
20. The force (*F*) that induces slip is determined from the 'brake equation' where

$$F2 = F1 * e^{\mu * \theta}$$

Where e is a known constant

$\mu$ is measured friction value for media and roller, and

$\theta$ is the actual wrap angle of the roller of interest

$$F = F2 - F1$$

- THEN WHAT IS THE SLIP TORQUE OF THE ROLLER ? ....

# DESIGN METHOD - Continued

- The remaining calculations are straight forward and not complicated at all.....

  Thus

$$TQ = F * R[Kg - mm]$$

Where *TQ* is the slip torque
*F* is the slip force and
*R* is the roller radius

The slip acceleration

$$\alpha = \frac{TQ}{J}\left[\frac{rad}{\sec^2}\right]$$

And the linear slip acceleration

$$XDD = \frac{\alpha}{R}\left[\frac{mm}{\sec^2}\right]$$

# DESIGN METHOD - Conclusion

- CONCLUSION:
  - IF THE REEL TO REEL SERVO CONTROL SYSTEM KEEPS THE LINEAR ACCELERATION OF THE MEDIA AT THE PERIPHERAL RADIUS OF THE REELS TO THE VALUE (OR LESS) CALCULATED FOR XDD ABOVE, THEN THE ROLLERS WILL NOT SLIP.
  - THUS ANY LONGITUDINAL WAVES (SPIKES) GENERATED (BY THE REEL MOTORS) WILL BE FILTERED OUT MECHANICALLY BY THESE ROLLERS. THE RESULT WILL BE A MORE STABLE TIMING DISTRIBUTION OF DATA FROM THE READ/WRITE SENSOR.
  - THIS APPLIES TO BOTH THE WRITE AND READ FUNCTIONS OF THE MAGNETIC TAPE RECORDER.

# SAMPLE CALCULATION

- Benchmark's DLT1 tape drive with DLT4 media
  - Third roller from supply reel

1. $R$ = 0.249 (IN) 6.325 (mm)
2. $J$ = 0.000003 (in-lb-sec$^2$) 0.0000342 (mm-Kg-sec$^2$) (measured)
3. $\theta$ = 90°(deg) for the third roller from the supply reel
4. $\mu$ = 0.17 (measured)
5. $T$ = 8 ounces (specified) 0.227 Kg
6. $T1 = T$
7. $T2 = T1 * e^{\mu * \theta} = 1.333$
8. $T2 = T1 * 1.333$ = 0.665 lbs 0.302 Kg
9. $F = T2 - T1$ = 0.1653 lbs 0.075 Kg
10. $TQ = F * R$ = 0.0412 in-lbs 0.475 mm-Kg
11. $\alpha = \frac{TQ}{J} = 13{,}720 \ \frac{rad}{\sec^2}$
12. $XDD = \alpha * R = 3{,}416 \ \frac{in}{\sec^2}$ $86{,}766 \ \frac{mm}{\sec^2}$

- Thus the maximum allowable linear acceleration for DLT1 with DLT 4 media is: 3,416 $\frac{in}{\sec^2}$ or 86,766 $\frac{mm}{\sec^2}$

QUESTIONS ?