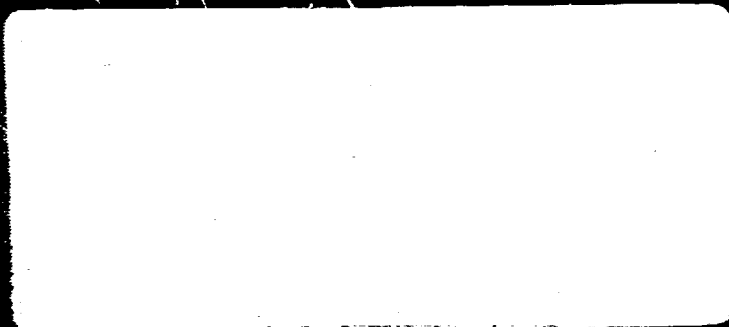


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FACILITY FORM 602

V70-29387	
(ACCESSION NUMBER)	(THRU)
37	1
(PAGES)	(CODE)
CR-110242	14
(NASA CR OR TMX OR AD NUMBER)	(CATEGORY)

750-12

PHOTOHELIOGRAPH
FILM CAMERA DESIGN

August 12, 1968

G. Bastien

Approved by:


Denton Allen, Task Leader,
Photoheliograph Task

JET PROPULSION LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
PASADENA, CALIFORNIA

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PHOTOHELIOGRAPH
FILM CAMERA DESIGN

August 17, 1968

G. Bastien

Approved by:


J. Denton Allen, Task Leader,
Photoheliograph Task

JET PROPULSION LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
PASADENA, CALIFORNIA

FOREWORD

This report covers work on one phase of the photoheliograph development task, NASA Code 945-84-00-01-00, for the period November 1967 through June 1968. The photoheliograph has been proposed to NASA for the Apollo telescope mount (ATM) by Caltech, with Professor Harold Zirin as the principal investigator and Dr. Robert Howard of Mt. Wilson and Palomar Observatories the co-investigator (see TM 33-369, November 1967). The objective of the investigation is to obtain high resolution cinematographs in white light near ultraviolet and narrow band hydrogen alpha. Because of the ATM program uncertainties, emphasis has been placed on areas of technology that are somewhat mission-independent, but the ATM spacecraft has been used to establish design constraints.

ABSTRACT

Image recording in the photoheliograph may be either by TV or photographic film. Light energy histograms in UV (1500 to 3000Å), white light (4000 to 6100Å) and H α (6563Å) have been prepared which show that images can be recorded with reasonable exposure times (less than 1/20 sec.) and on commercially available film (i. e. Kodak SO-375). The advantages over TV are fast frame rates if required, finer resolution, and permanent record. The disadvantages are the limited total number of frames available, the need for Astronaut EVA to retrieve the film, possible radiation fogging, and the greater weight of the cameras. In an attempt to minimize these disadvantages as compared to presently available cameras, JPL undertook to design a light weight, sealed camera with the following features: maximum film load (limited by weight), 70mm film for dual track 35mm recording to reduce spool size, automatically reversible film advance mechanism.

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PHOTOHELIOGRAPH FILM CAMERA DESIGN

REQUIREMENTS

The functional and mission requirements and the ATM constraints together determined the following specifications for the film camera system:

- (1) Full frame, 35mm format (for resolution, field of view, and for compatibility with usual editing and projection techniques).
- (2) 2000 feet, (minimum) of 35mm film - or the equivalent - per load (P. I. requirement. This is with or without intermediate EVA).
- (3) Film cassette to be sealed against evaporation of emulsion moisture in a prolonged (60 day) vacuum environment.
- (4) Film cassette to be removable by an Astronaut during EVA for film retrieval at the conclusion of observation time. Also, mechanisms should be designed to permit replacement of cassette or film with a fresh load.
- (5) Cassette weight, with film, to be less than 50 lbs. (Astronaut manipulation limitations)
- (6) Three camera systems ($H\alpha$, U. V. and white light) need not be identical or interchangeable because different types of film may be used in each camera.
- (7) Each of the three cameras is to be complete and able to operate independently of the others. Each is to have its own shutter, exposure control, film advance, frame identification device, motors, and monitoring vidicon.
- (8) All motors, relays, solenoids (if any) must operate on unregulated 24 to 28 volt D. C. power.
- (9) All motors and controls will be operated remotely by the Astronaut in the LM.

- (10) Frame rates variable from 1 frame per 20 seconds to 20 frames per second. Timing pulses will be generated in the LM console.
- (11) When film is not being exposed (between frames), the light beam will be reflected to the faceplate of an adjacent vidicon. This can be done by placing a diagonal mirror on the outside of the shutter.
- (12) The cameras will be located on a movable platform (for focus adjustment) along with the necessary beam splitters, filters, field stops, heat dumps, etc. in the ATM quadrant adjacent to the telescope.
- (13) Cooling may be accomplished by extending the telescope liquid thermal control system to the camera cluster, or by direct radiation from the cluster to the 50° F wall of the ATM canister.

ATM CAMERA DESIGN CONSIDERATIONS

The following is a preliminary investigation of a camera which may be used on the solar telescope, and of the parameters affecting its design. Several conditions and limitations, including size, shape and weight are placed on the design which has resulted in specialized features unique to this application.

The approach has been to select the simplest system which could accomplish the mission and maintain a high degree of reliability inasmuch as the cameras will have to perform intermittently for an extended period of time in a space environment.

The three cameras will be operated independently in either single frame or cine mode which dictates a variable pulse type. Each camera will be synchronized to a separated shutter assembly and operated from the console within the spacecraft. However, the Astronaut will be required to make at least one EVA, to remove the film cassettes for return to earth. The filters and shutter assemblies will be permanently mounted on the base. The shutter itself will be a variable opening type which will allow control of exposure time.

The white light, ultra violet, and hydrogen alpha cameras are assumed to be alike for the purposes of internal mechanism design with exception of the filters and shutters. The cassette configuration may be such as to allow the

reels to be placed side by side i. e. , with a line from center to center perpendicular to the input light beam, or a line from center to center parallel to the input light for space conservation.

Areas of future effort will include:

- (1) Further study and optimization of a film transport system.
- (2) Structural design of the cassette.
- (3) Latching and interfacing with the telescope.
- (4) Material selection.
- (5) Thermal analysis i. e. , a study of the cassette heating and cooling requirements including insulatory and surface treatment requirements.
- (6) A further study of motors, solenoids, clutches, and brakes which may be used in the system.
- (7) A weight analysis.

DESIGN OBJECTIVES

Several factors are considered in this preliminary design. These include size of cassette, power requirements, and heat dissipation requirements. In each analysis, ballpark numbers were found for a particular configuration which do not necessarily represent the optimum or final configuration.

One particular aspect of investigation was to compare the feasibilities of two systems, i. e. a camera using 1000 feet of 70mm film with reversible feed direction and a camera using 2000 feet of 35mm film with single feed direction. The choice of either system affects the design of the upstream parts of the camera. As an example, using a bi-directional 70mm film camera necessitates a complex reversing mechanism in addition to an extra mirror which must be automatically controlled to shift the image to the other half of the film when film reverses direction. On the other hand, using a single feed direction 35mm film camera eliminates the need for a reversing mechanism and re-imaging mirror, but the film transport and reel drive power requirements and the consequential heat dissipation problems are greater in addition to the considerably larger space required for the film cassettes on the camera platform. This latter

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is an important consideration because of the small packaging space available in the ATM canister. An effort has been made to reduce the number of moving parts such as gears, linkages, motors, bearings, and sprockets to conserve space and weight. Many standard cameras incorporate complex claw pull down mechanisms which necessitate a large number of moving parts. There are, however, other types of pull down mechanisms which operate on a different principle and require fewer moving parts. This is the type considered here.

REFERENCES

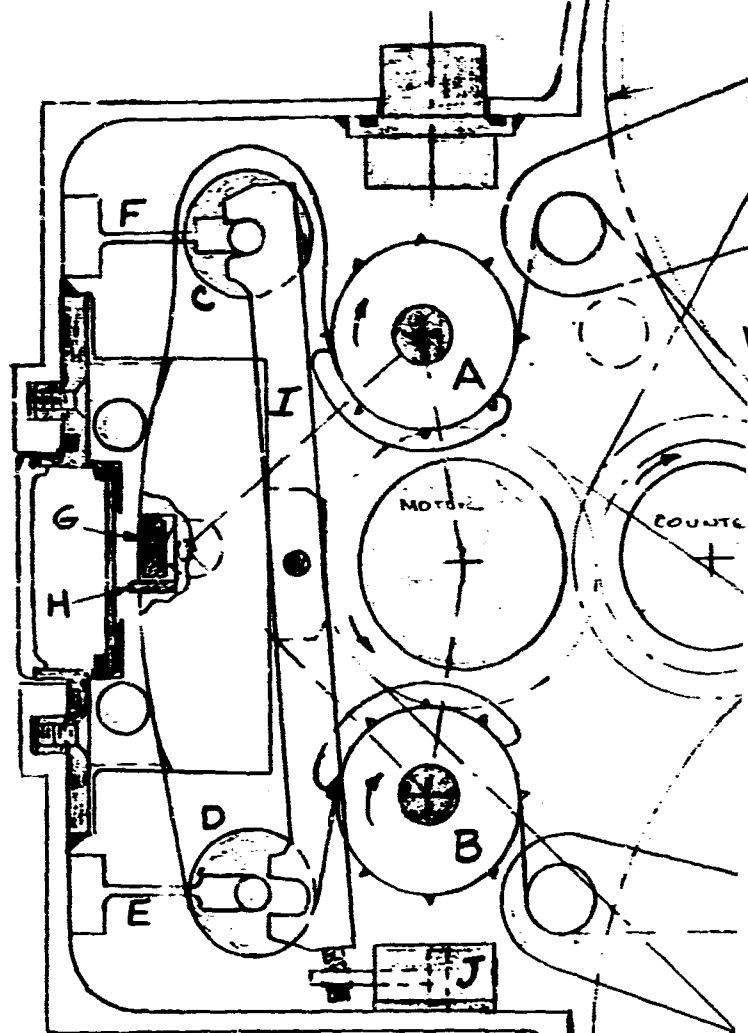
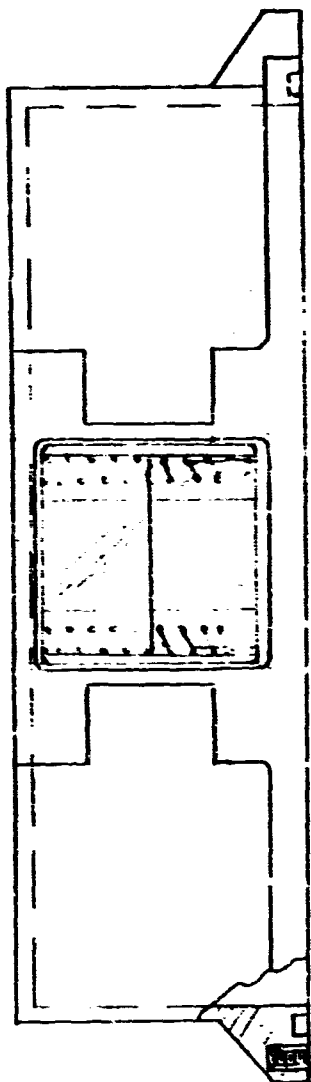
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APPENDIX A
EXPLANATION OF CAMERA MECHANISM

The film is held during exposure by small rollers against a slightly curved shoe in which there is a fixed register pin (H). Sprocket (A) and (B) continuously pay out and take up film, and as sprocket (B) takes up, cantilever spring (E) moves up thus storing spring energy and putting tension in film between (D) and pin (H). Shoe (F) is pushed outwardly by a cam (gang geared to both drive sprockets) until film disengages pin (H). The stored spring energy is at this time released causing sprocket (D) to snap down until film is stopped by sprocket (C), thus placing a new frame into position ready for exposure. If camera is in cine mode, the process recycles.

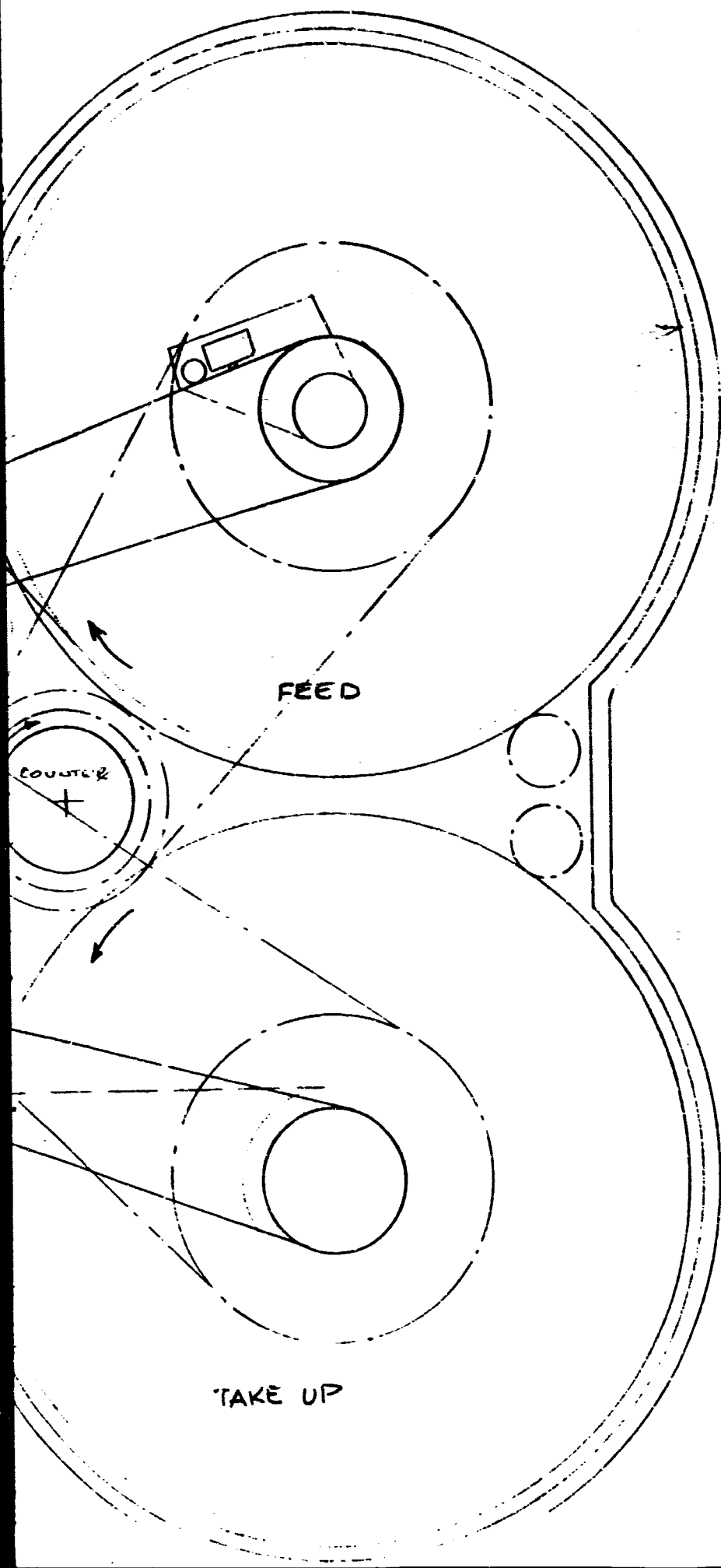
The film can be exposed on its other half at the end of the reel by re-focusing the image on the other half and reversing the drive direction. This is easily accomplished by activating an actuator (J) which causes the lever (I) to engage spring (E) and disengage spring (F). The drive motor direction and the brake-clutch action of the reels are reversed, thus the take up reel becomes the feed reel and vice-versa.

Strain relief arms are provided to limit the force on the film to a designated maximum. Limit switches are also provided to clutch and brake automatically.



CAMERA - REVERSIBLE DRIVE .
 CONFIGURATION I
 MJB.

FOLDOUT FRAME 1



FOLDOUT FRAME 2

8 C - 10025004

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APPENDIX B
ANALYSIS OF MECHANISM

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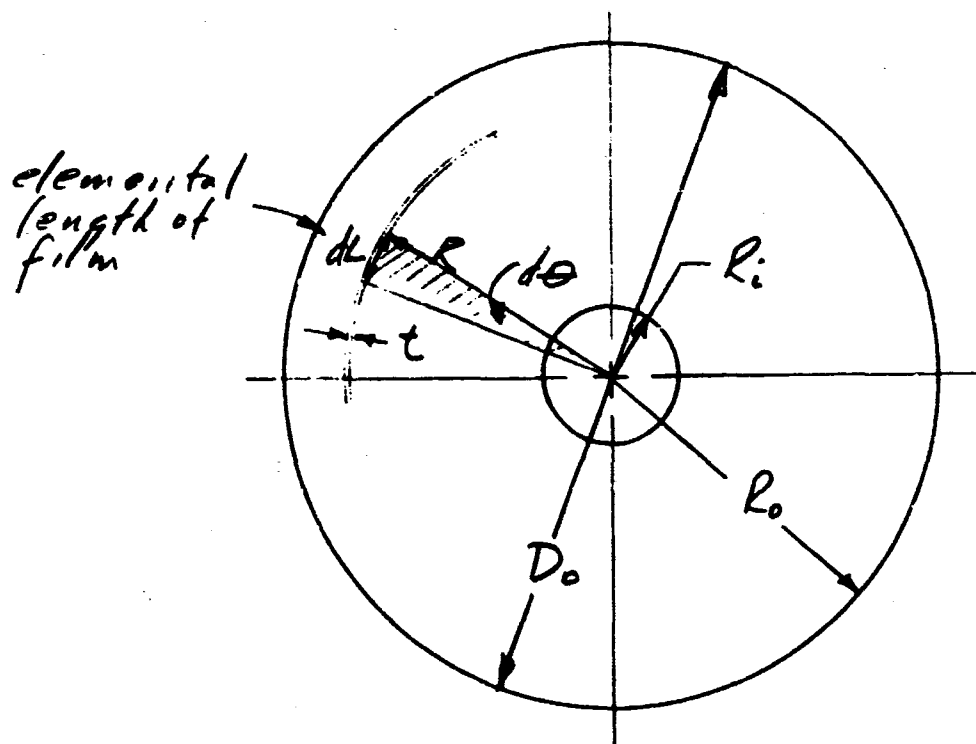
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I. DETERMINATION OF OUTSIDE RADIUS OF REEL



$$L = R\theta \quad dL = R d\theta$$

$$R = R_i + \frac{\theta}{2\pi} t$$

$$\begin{aligned} L_{\text{TOTAL}} &= \int_0^{\theta_r} R d\theta = \int_0^{\theta_r} \left(R_i + \frac{\theta}{2\pi} t \right) d\theta \\ &= R_i \theta_r + \frac{t}{4\pi} \theta_r^2 \end{aligned}$$

$$\theta_r^2 + \left(\frac{R_i 4\pi}{t} \right) \theta_r - \left(\frac{L 4\pi}{t} \right) = 0$$

$$\theta_r = \frac{-\left(\frac{R_i 4\pi}{t} \right) \pm \sqrt{\left(\frac{R_i 4\pi}{t} \right)^2 + 4 \left(\frac{L 4\pi}{t} \right)}}{2}$$

$$R_o = R_i + \frac{\theta_r}{2\pi} t, \quad D_o = 2R_o$$

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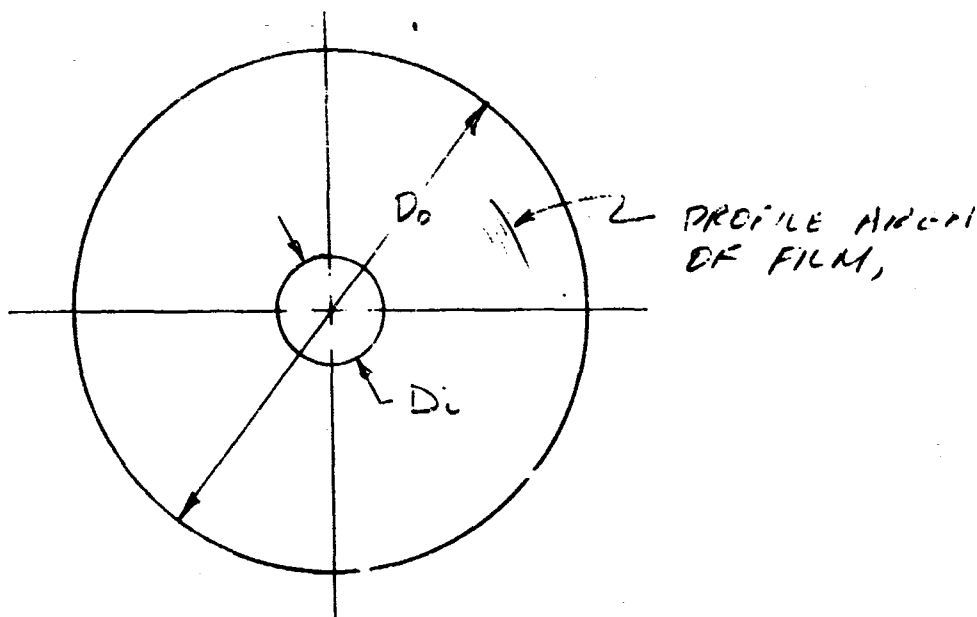
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ALTERNATE FORMULATION



Length of film = L

thickness of film = t

Profile Area of straight length = Lt

Profile Area of Spool = $\frac{\pi}{4} (D_o^2 - D_i^2)$

$$Lt = \frac{\pi}{4} (D_o^2 - D_i^2)$$

$D_i = 2$ in

Stock thickness of film : 2.5, 4.0, 5.2 mils

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70 mm film - 2000 ft REEL (35mm);

DOUBLE SIDED FILM, 1000 ft EACH SIDE.

REEL SHALL THEREFORE HAVE 1000 ft CAPACITY
AND BE REVERSIBLE TO EXPOSE BOTH SIDES

Area of film per foot = .228 ft² LENGTH = 1000'

t mils	w/ft	Wt(1000')	D _o	J(1/2 m R _o ²)
2.5	.00525	5.25 lb	6.5 in	.072 lb sec ² /in
4.0	.00775	7.75	8.1	.165
5.2	.0089	8.90	9.15	.240

TABLE SHOWING WEIGHT, DIA., AND J FOR
VARIOUS FILM THICKNESSES

DENSITY OF FILM, (TYPICAL VALUES)

THICKNESS

DENSITY

2.5 MILS

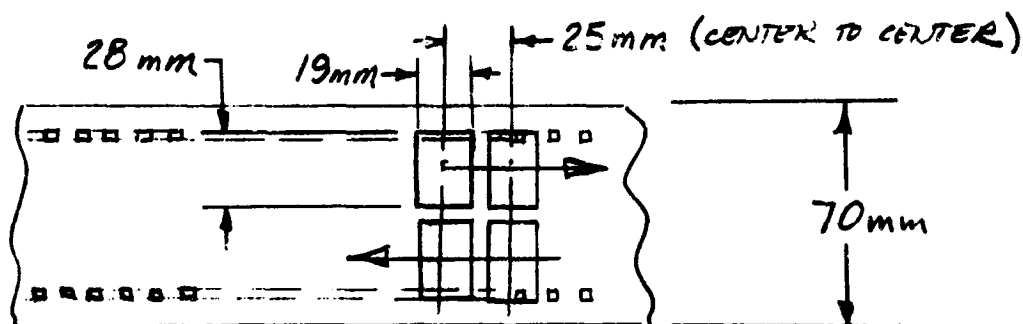
.022-.023 lb/ft²

4.0

.034

5.2

.039



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35 mm film , 2000 ft REQSINGLE SIDED FILM ; THEREFORE, REEL
SHALL HAVE 2000 ft CAPACITYArea of film per foot = $.114 \text{ ft}^2$ LENGTH = 2000 ft

t mils	wt/ft	Wt(2000')	D _o	J($\frac{1}{2} m R_o^3$)
2.5	.00263	5.25 lb	8.75	.130 $\text{lb sec}^2/\text{in}$
4.0	.00387	7.75	11.20	.373
5.2	.00443	8.90	12.80	.473

TABLE SHOWING WEIGHT, DIA., AND J FOR
VARIOUS FILM THICKNESSES

DENSITY OF FILM, (TYPICAL VALUES)

THICKNESS OF FILM

DENSITY

2.5 MILS

.022 - .023 lb/ft^2

4.0

.034

5.2

.039

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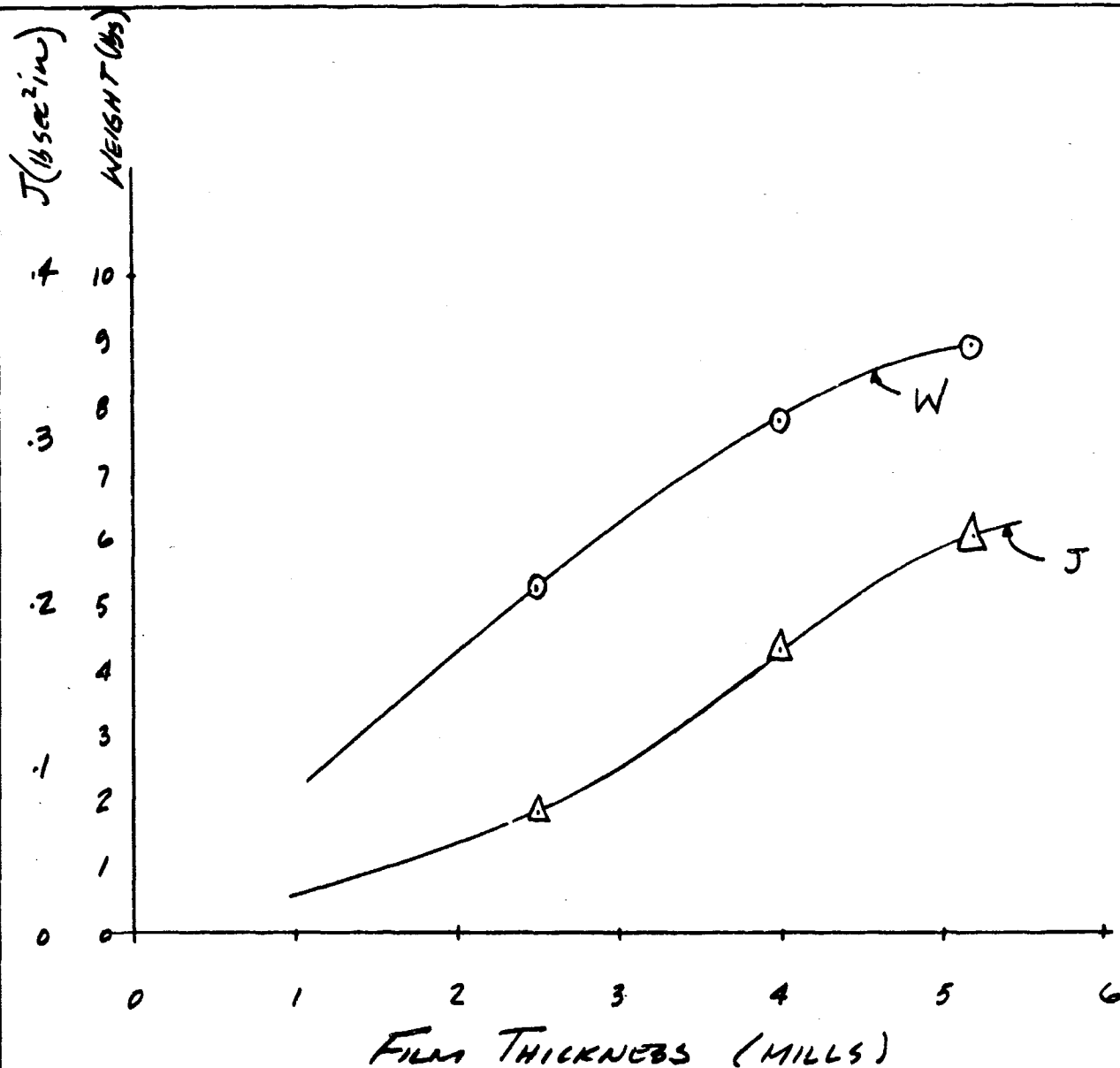
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WEIGHT AND MOMENT OF INERTIA OF
FILM BASED ON 70 mm x 1000 ft.,
VARIATION OF DENSITY INCLUDED.
WEIGHT AND MOMENT OF INERTIA OF
METAL REEL NEGLECTED.

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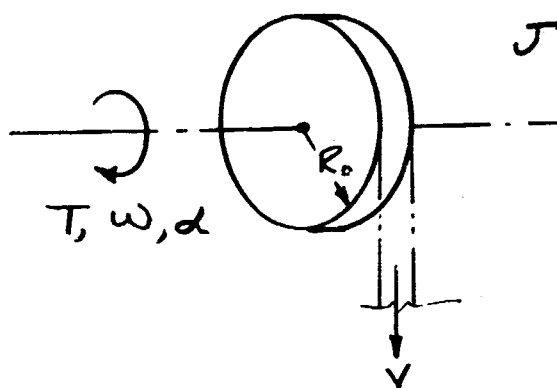
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II

Power required to bring
a full film reel up to speed.

Assuming that the diameter of the REEL
does not change appreciably initially,



$J = \text{Moment of inertia about Axis} = \frac{1}{2} m R_0^2$

$$\text{Power} = T \omega$$

Torque required To bring spool up to speed
 $\approx J \alpha$

$$\text{Power} = (J \alpha) \omega$$

$$\text{Power Ratio} = \frac{(J \alpha) \omega \Big|_{70 \text{ mm (1000')}}}{(J \alpha) \omega \Big|_{85 \text{ mm (2000')}}}$$

$$= \frac{\frac{1}{2} m (R_0)_{70}^2}{\frac{1}{2} m (R_0)_{85}^2}, \text{ since masses are equal}$$

$$\& (R_0)_{70} \approx 1.4 (R_0)_{85}$$

$$\therefore \text{P.R.} = \frac{R_0^2}{(1.4 R_0)^2} = \frac{1}{2}$$

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THUS IT CAN BE SEEN THAT THERE IS A DISTINCT ADVANTAGE, FROM A POWER STANDPOINT, IN USING A 70MM - 1000 FT REEL AS COMPARED TO A 35MM - 2000 FT REEL. THIS DOES NOT NECESSARILY RULE OUT THE USE OF THE 35MM REEL, BUT IN A SYSTEM WHERE POWER IS AT A PREMIUM, TWICE THE POWER MUST BE CONSIDERED A HEAVY WEIGHING FACTOR.

A LIST OF ADVANTAGES AND DISADVANTAGES OF USING EITHER SYSTEM IS SHOWN TO INDICATE OTHER WEIGHING FACTORS WHICH SHOULD BE CONSIDERED IN FUTURE EVALUATIONS.

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Advantages of using 35mm - 2000ft REEL (unidirectional)

1. The primary advantage is that only one direction of film feed is required. i.e., film will not be required to feed in reverse direction, thereby eliminating complex reversing mechanism.

Disadvantages (relative to 70mm - 1000 ft)

1. Approximately 40% greater diameter spool required, therefore 40% larger casset required in area where space is a premium.
2. Approximately twice the power required to bring spool up to speed.
3. Approximately 50% greater size of motors, brakes, and clutches required to operate this size spool.
4. Cassett size and weight greater by 40%, therefore 40% greater storage space required.

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Advantages of using 70mm - 1000 ft REEL (Bi directional)

1. The disadvantages of the 35mm - 2000' REEL become the Advantages of the 70mm - 1000' REEL
i.e.,
40% smaller diameter Reel
50% less power required to accelerate Reel
50% Smaller motors, clutches, and brakes to operate this size Reel.
cassett size and weight 40% less.
40% less storage area required.

Disadvantages

1. The primary disadvantage is that film feed direction must be reversed at end of reel to take pictures on other half of film thus necessitating complex reversing mechanisms.

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IV.

DETERMINATION OF FORCE ON
FILM AND TIME FOR REEL TO COME UP TO
SPEED.

A STRAIN RELIEF ARM IS INCORPORATED
TO LIMIT THE AMOUNT OF FORCE ON THE FILM
I.E., A HELICAL SPRING IN THE ARM
GRADUALLY TRANSFERS THE RETARDING FORCE
OF THE INERTIA OF THE REEL ONTO THE FILM.
THIS REDUCES THE SUDDEN FORCE (JERK)
APPLIED TO THE FILM AND BRINGS THE REEL
UP TO SPEED. SINCE ONLY BALLPARK
NUMBERS ARE NEEDED AT THIS POINT, AN
APPROXIMATE ANALYSIS FOLLOWS.

IT IS ASSUMED THAT :

1. THE REEL IS FULL AND THEREFORE
THE RADIUS r , THE MASS m , AND THE
MOMENT OF INERTIA J ARE AT MAX. CONDITIONS
AND DO NOT CHANGE.
2. THE PULL FORCE F DUE TO THE PULL ANGLE
REMAINS CONSTANT.

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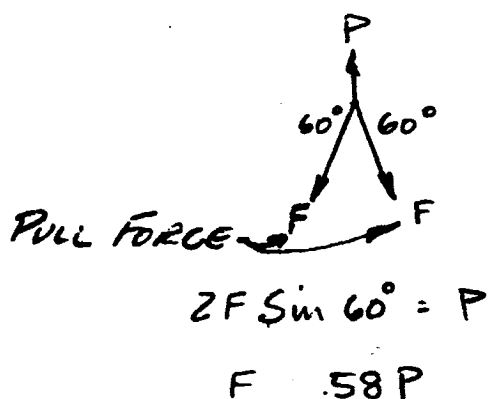
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FIND ω vs t 

$$P = K \theta_1 \quad K = 16/\text{rad}$$

$$\therefore F = .58K \theta_1$$

$$\theta_1 = \left(1 - \frac{r\omega}{V}\right) \theta_{\max}$$

$$F \cdot r = J \frac{d\omega}{dt} \quad \text{Torque Req. to accelerate Reel}$$

$$.58K \theta_m r \left(1 - \frac{r\omega}{V}\right) = J \frac{d\omega}{dt}$$

$$a = .58Kr\theta_m$$

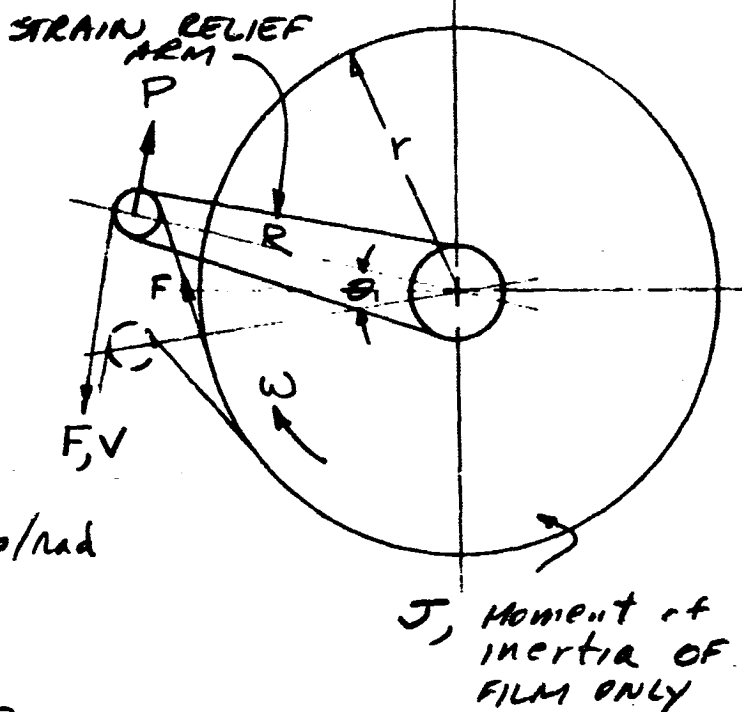
$$b = \frac{r}{V}$$

$$\text{then} \quad \int dt = \int \frac{J d\omega}{a(1-b\omega)}$$

$$t = -\frac{J}{ba} \ln(1-b\omega) + C$$

$$\text{i.e., } t=0, \omega=0 \quad \therefore C=0$$

$$\omega = \frac{1}{b} \left(1 - e^{-ba/s t}\right) \quad \text{Angular Speed vs time}$$



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$$F = \frac{J}{r} \frac{dw}{dt} = \frac{J}{r} \frac{a}{J} e^{-\frac{ba}{J}t} = \frac{a}{r} e^{-\frac{ba}{J}t}$$

$$F_{max} = \frac{a}{r} = \frac{.58 K \times \Theta_m}{r} = \underline{\underline{.58 K \Theta_m}} \text{ @ } t=0$$

F_{max} , MAXIMUM Force on film

$$\Theta_1 = \left[1 - \left(\frac{r}{V} \cdot \frac{1}{T} \right) (1 - e^{-\frac{ba}{J}t}) \right] \Theta_m$$

$$\Theta_1 \text{ max @ } t=0 = \Theta_m$$

$$\Theta_1 = \Theta_m e^{-\frac{ba}{J}t}$$

THE REEL WILL COME UP TO 62.3% OF ω_{max} AT ONE TIME CONSTANT WHERE THE TIME CONSTANT IS GIVEN BY,

$$T_C = \frac{J}{a J} = \frac{J V}{.58 K r^2 \Theta_m} \approx \frac{M V}{2 F_m}$$

NOTE V = FILM SPEED ≈ 10 in/sec

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FOR 70 mm FILM,
 t (thickness of film) = 5.2 MILS
 r (RADIUS OF REEL) = 4.55 in
 J (MOMENT OF INERTIA) = .240 lb sec² in

SINCE SPACE IS LIMITED, θ_m SHOULD BE
 RESTRICTED TO $\theta = \frac{x}{R} = \frac{1.0}{6} = .166 \text{ rad.}$
 (9.5°)

THE MAXIMUM FORCE ON THE FILM
 SHOULD NOT EXCEED 10 OR (1 lb).

$$\therefore K = \frac{F_m}{0.58 \theta_m} = \frac{1 \text{ lb.}}{0.58 (.166) \text{ rad.}} \approx 10 \text{ lb/rad.}$$

THE RESULTING TIME CONSTANT IS,

$$T_c = \frac{(.240) 10}{0.58 (10) (4.55)^2 (.166)} = .120 \text{ sec.}$$

UNITS $\frac{\text{lb sec}^2 \text{ in}}{\text{lb in}^2 \text{ sec rad}} \rightarrow \text{sec}$

W WILL BE APPROX 96% OF W_{max}
 IN APPROX $4T_c$ OR .480 SEC.

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SOME TIME CONSTANTS FOR OTHER FILM
THICKNESSES ALL OTHER FACTOR BEING EQUAL.

<u>FILM THICKNESS</u>	<u>T_c</u>
4.0 MILS	.100 SEC
3.5	.092
3.0	.081
2.2	.029

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V. POWER REQUIREMENT FOR MOTOR

 F_1 (FILM FORCE) F_2 (DRAG)

$$F = F_1 + F_2$$

$$F_1 = 1 \text{ lb}$$

$$F_2 = 1 \text{ lb}$$

(ESTIMATED)

 F_1, V

$$V = \frac{10 \text{ ft}}{12} \text{ / sec.}$$

$$\left[\text{Horse Power} \right]_{\text{REQ.}} = \frac{F V}{550} = \frac{(2) \frac{10}{12} \cdot \frac{1}{550}}{1} = .003 \text{ HP}$$

$$= 2.2 \text{ Watts.}$$

$$\left[\text{Torque} \right]_{\text{REQ.}} \approx R \times F = 1 \text{ in} \times 2 \text{ lb} = 2 \text{ in lb}$$

TYPICAL D.C. TORQUE MOTOR CURVES

INDICATE THAT EFFICIENCY VARIES
FROM 20% TO 50% FOR MAX POWER.

$$\text{MOTOR OUTPUT SPEED} = \frac{10 \text{ in}}{\text{sec}} \times \frac{1}{1 \text{ in}} \times \frac{\text{REV}}{2\pi \text{ rad}} \times \frac{60 \text{ sec}}{\text{MIN}}$$

$$= 96 \text{ RPM}$$

OPERATING AT $T_m = 1.5 \text{ in-oz}$ $S = 2000 \text{ RPM}$

GEAR DOWN RATIO = 21

$T_{\text{out}} = 2 \text{ in lb}$ OK

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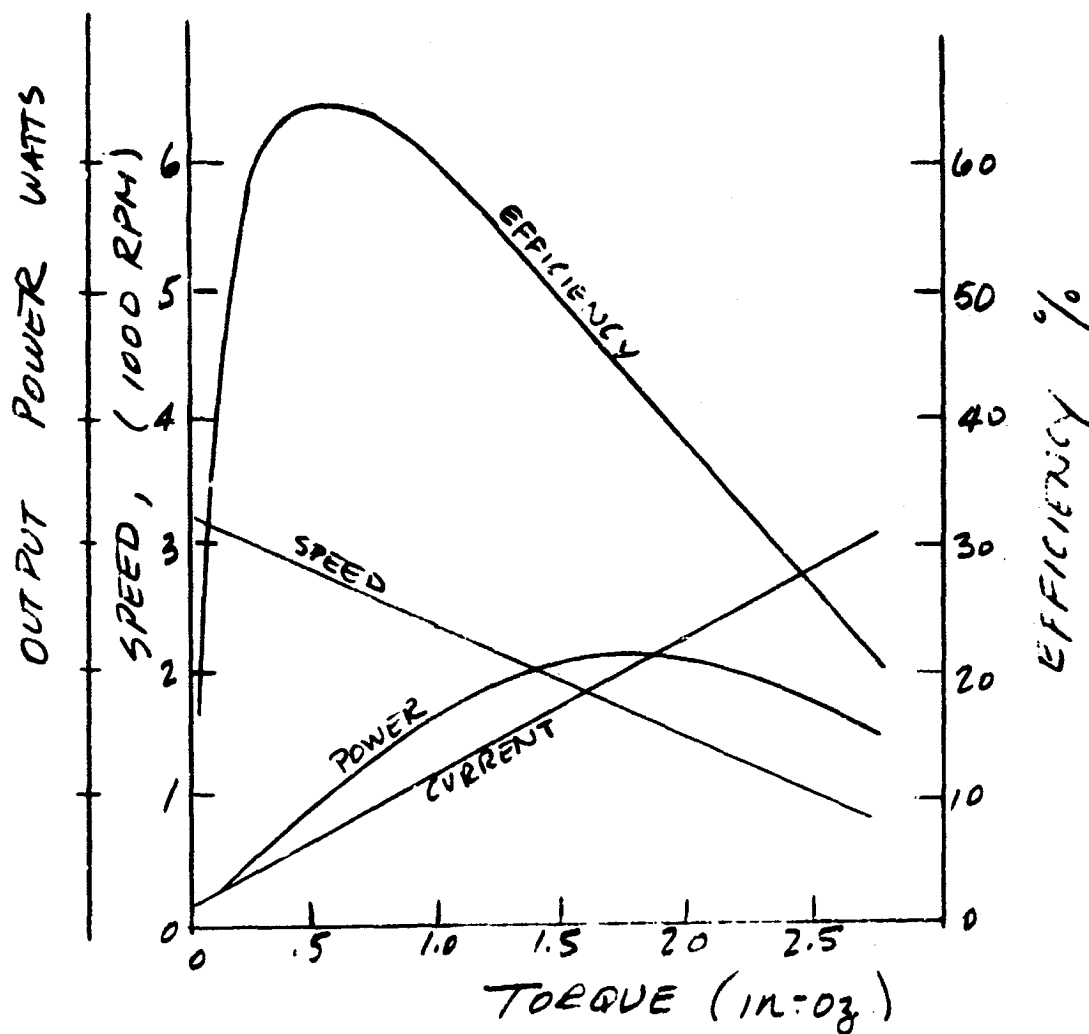
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TYPICAL D.C. TORQUE MOTOR CURVES

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POWER INPUT REQUIRED FOR MOTOR IS
THEREFORE,

$$P_{in} = \frac{2.2 \text{ watts}}{\text{Eff. NOM.}} = \frac{2.2}{.40} = 5.5 \text{ watts}$$

$$\text{SURGE POWER} \approx 0.4 \text{ AMPS} \times 27 \text{ VOLTS} \approx 11 \text{ WATTS}$$

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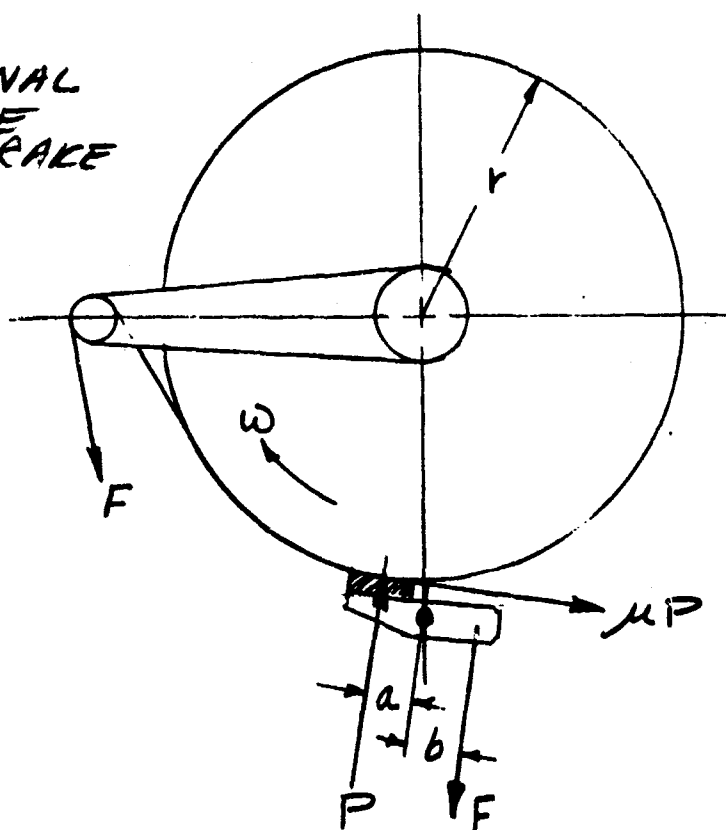
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VI.

BRAKE DESIGN OF REEL

1. EXTERNAL
SHOE
BRAKE



IMPULSE MOMENTUM EQUATION

$$\int M dt = J \omega$$

ENERGY EQUATION

$$T = \int M d\theta = \int J \ddot{\theta} \frac{d\theta}{dt} dt = \frac{1}{2} J \omega^2$$

IT IS ASSUMED THAT ;

1. REEL IS FULL I.E., r , J , AND MASS ARE MAXIMUM AND CONSTANT.

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FROM IMPULSE EQN.

$$r_{M P} t = J (w_1 - w_2)$$

$$t_s = \frac{J (w_1 - w_2)}{r_{M P}}, \text{ STOP TIME}$$

THE ANGLE θ THROUGH WHICH THE REEL
ROTATES IN COMING TO A STOP CAN BE FOUND
FROM THE ENERGY EQN.,

$$\theta_s = \frac{1}{2} \frac{J (w_1^2 - w_2^2)}{r_{M P}}, \text{ COASTING ANGLE}$$

APPROX. μ FOR BRAKE SHOE MATERIAL ; $\mu = .45$

STOP TIME, BASED ON 70mm-1000 ft REEL

FILM THICKNESS = 5.2 MILS

$J = .240 \text{ lb sec}^2 \text{ in}$

$r = 4.55 \text{ in}$

$$t_s = \frac{.240 (2 - 0)}{4.55 (.45) P} = \frac{0.23}{P}$$

COASTING ANGLE,

$$\theta_s = \frac{.240 (2^2 - 0)}{2 (4.55) (.45) P} = \frac{.23}{P}$$

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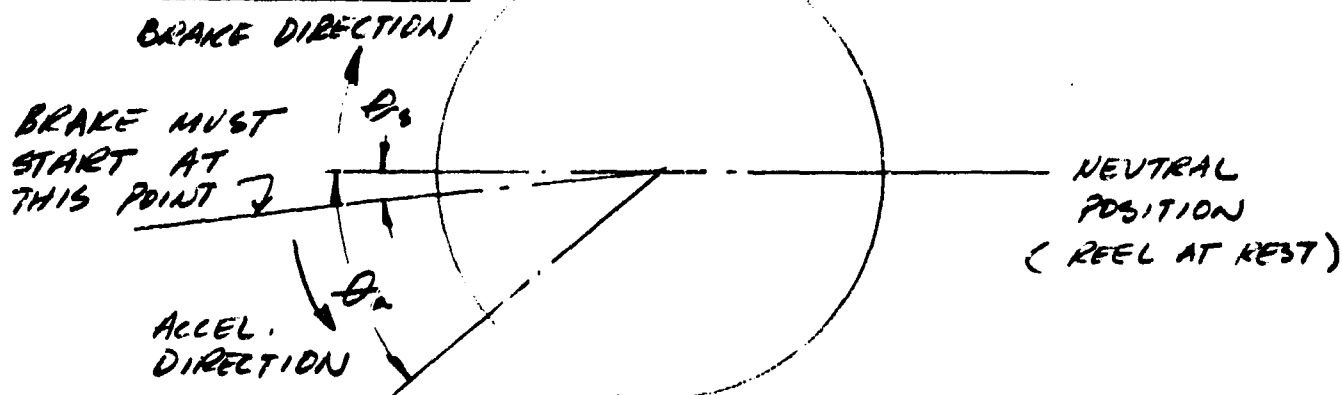
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ASSUME BRAKE FORCE P IS IN THE ORDER
OF 10 lb i.e., $P = 10 \text{ lb}$

$$t_s = \frac{0.23}{P} = .023 \text{ SEC} \quad (23 \text{ ms})$$

$$\theta_s = \frac{0.23}{P} = .023 \text{ rad} \quad (1.3^\circ)$$

BRAKE TIMING



$$\theta_a = \text{ACCELERATION TIME} = 9.5^\circ$$

$$\theta_s = \text{STOP TIME} = 1.3^\circ$$

$$\frac{\theta_s}{\theta_a} = \frac{1.3^\circ}{9.5^\circ} = .137$$

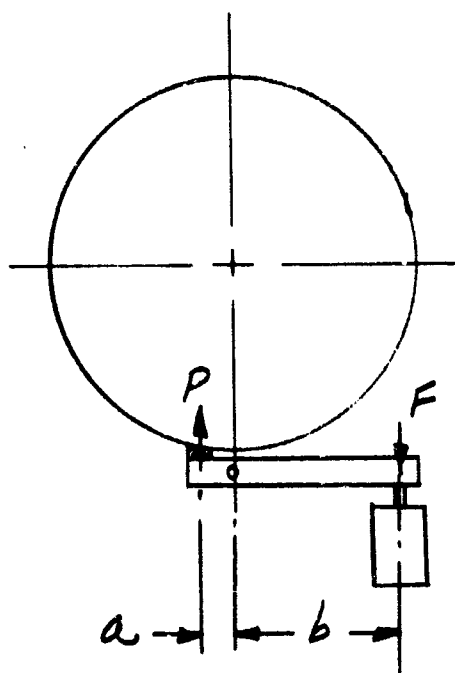
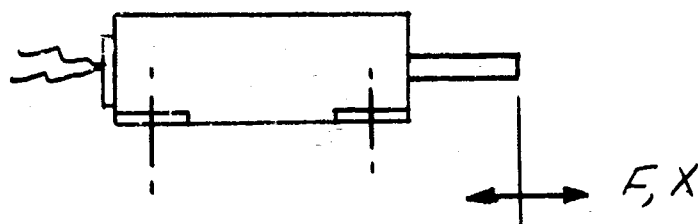
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SOLENOID BRAKING POWER LOAD

VOLTAGE INPUT = 27 VDC

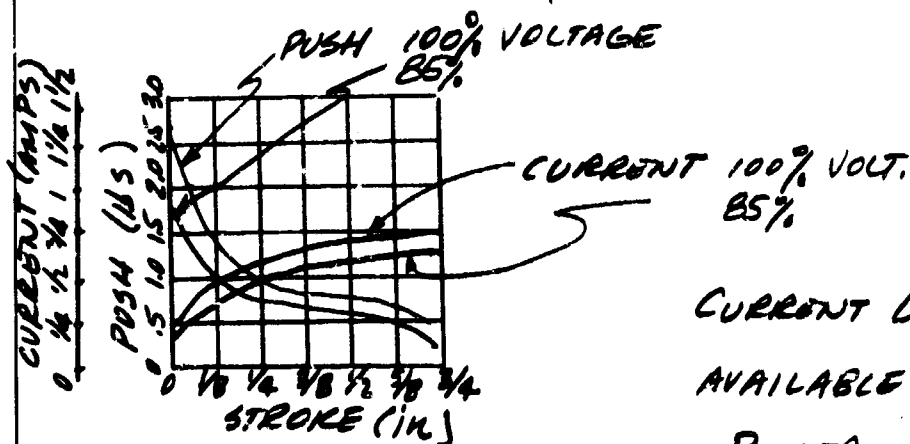


LET: $P = 10 \text{ lb}$
 $b = 6 \text{ in}$
 $a = 0.5 \text{ in}$

$$F \cdot b = P \cdot a$$

$$F = \frac{P \cdot a}{b} = \frac{10(0.5)}{6} = 0.83 \text{ lb}$$

ASSUME 85% VOLTAGE
 i.e., $V = 23 \text{ VOLTS}$



TYPICAL DC SOLENOID

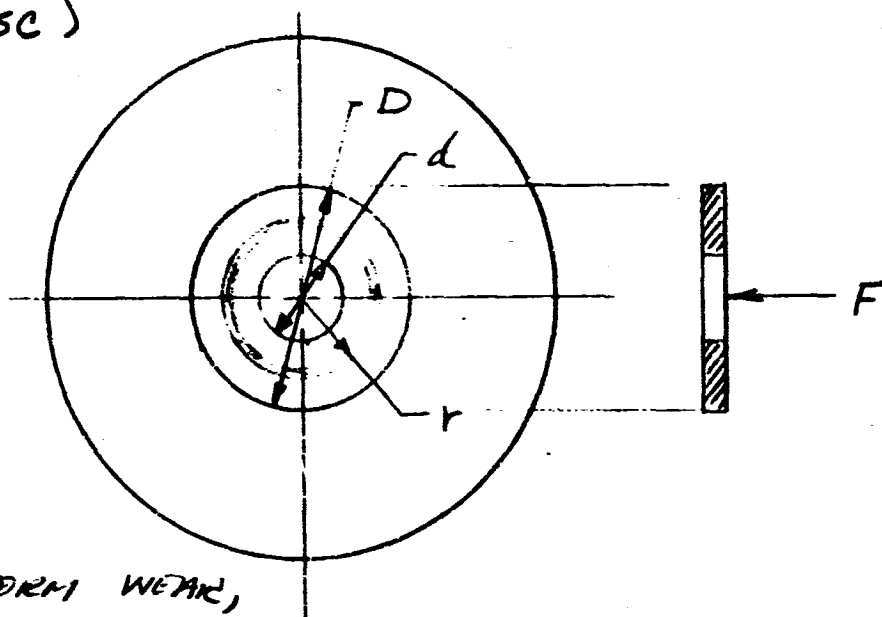
CURRENT LOAD = .37 AMPS

AVAILABLE STROKE = 1/8 in

POWER LOAD = 8.5 WATTS

2. TASK BRAKE

(FOR ONE DISC)



BASED ON UNIFORM WEAR,

(GREATEST PRESS. AT $r = d/2$ FOR
UNIFORM WEAR, $P_a = \text{MAX. PRESSURE}$)

$$Pr = P_a \frac{d}{2} \quad \text{or} \quad P = P_a \frac{d}{2r}$$

$$F = \int_{d/2}^{D/2} 2\pi P r dr = \pi P_a d \int_{d/2}^{D/2} dr = \frac{\pi P_a d}{2} (D - d)$$

$$T = \int_{d/2}^{D/2} 2\pi \mu P r^2 dr = \frac{\pi \mu P_a d}{8} (D^2 - d^2)$$

$$\text{OR } T = \frac{\mu F (D + d)}{4} / \text{DISC}$$

FROM IMPULSE EQN.,

$$t_s = \frac{4J(W_1 - W_2)}{\mu F(D+d)}, \text{ STOP TIME}$$

$$\theta_s = \frac{2J(W_1^2 - W_2^2)}{\mu F(D+d)}, \text{ COASTING TIME}$$

ASSUME,

1. $D = 4 \text{ in}, d = 2 \text{ in}$
2. $\mu = .45$
3. $J = .240$
4. $W_1 = 2 \text{ rad/sec}$

$$t_s = \frac{4(.240)(2-0)}{(.45)(4+2)F} = \frac{.71}{F} \left(\frac{1}{N} \right)$$

$$\theta_s = \frac{2(.240)(2^2-0)}{(.45)(4+2)} = \frac{.71}{F} \left(\frac{1}{N} \right)$$

WHERE $N = \text{No. OF DISCS}$

THUS 3 DISCS ARE REQUIRED FOR
EFFECT COMPARABLE TO EXTERNAL SHOE BRAKE.

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VII DETERMINATION OF HEAT GENERATION OF BRAKE

IN BOTH CASES OF BRAKING

$$J = .240 \text{ lb sec}^2 \text{ in}$$

$$T = J\alpha$$

$$= .02 \text{ slug} \cdot \text{ft}^2$$

$$T = \frac{12 J 2\pi n}{60 t}$$

$$n = \text{RPM}$$

$$t = \text{BRAKE TIME (SEC)}$$

BRAKE ENERGY E_K ,

$$E_K = \left(\frac{2\pi T}{12} \right) \left(\frac{1}{2} \frac{n}{60} t \right) = \frac{2\pi T n t}{1440} = \frac{\pi^2 n^2 J}{1800}$$

$$n = 2 \frac{\text{rad}}{\text{SEC}} \times \frac{\text{REV}}{2\pi \text{ rad}} \times 60 \frac{\text{SEC}}{\text{MIN}} = 120 \text{ RPM}$$

$$E_K = \frac{\pi^2 (120)^2 (.02)}{1800} = 1.53 \text{ ft} \cdot \text{lb}$$

$$H_{\text{Power}} = \frac{1.53 \text{ ft} \cdot \text{lb}}{.02 \text{ SEC}} \cdot \frac{1}{550} = .14 \text{ HP}$$

$$.14 \text{ HP} = 102 \text{ WATTS}$$

HEAT GENERATED IN FRICTION PER CYCLE (H),

$$H = .102 \text{ KW} \times 56.9 = 5.69 \text{ BTU/MIN}$$