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LAUNCHER, GRENADE, XM129: ANALYSIS OF STRESSES DUE TO DYNAMIC LOADS

Technical Report

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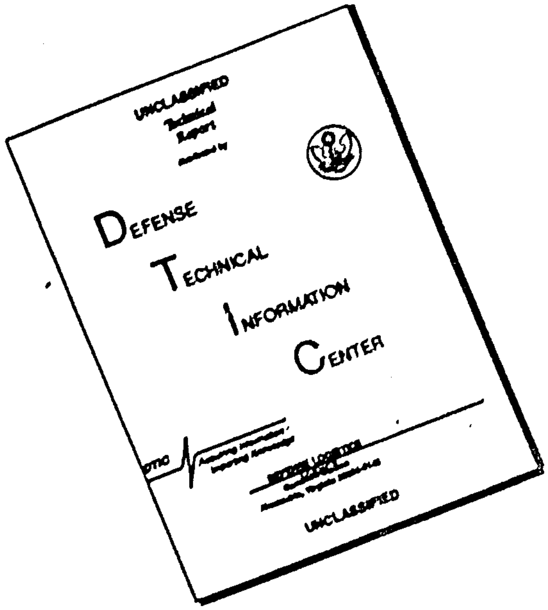

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LAUNCHER, GRENADE, XM129: ANALYSIS OF STRESSES DUE TO DYNAMIC LOADS

Technical Report

McInnis, W. T.

PROJECT TITLE: Aircraft Suppressive Fire Systems

PRON: EJ-2-50007-01-M1-M6

Preparing Agency: Springfield Armory, Springfield, Massachusetts

REPORT
SA-TR20-2741

ABSTRACT

An analysis of the XM129 grenade launcher was conducted to determine the dynamic loads, the resultant stresses, and the operational power requirements. General relationships defining the loads were derived. The equations were evaluated at various points. Power requirements and stresses were determined based upon these equations. The results can be used to facilitate the analysis of any future design changes.

CONTENTS

SUBJECT

A theoretical analysis was made of the XM129 grenade launcher.

OBJECTIVES

1. To determine the load and the stress levels of the various functional components of the weapon under normal firing conditions.

2. To ascertain the operational power requirements.

3. To provide general relationships which would facilitate analysis of the effects of any future design changes.

SUMMARY OF CONCLUSIONS

1. A firing rate of 300 rounds per minute is obtainable with the use of a 3/4 horsepower motor.

2. The stress levels of the critical components are well within the allowable limits.

RECOMMENDATION

A roller device should be incorporated in the follower stud design to reduce friction between the stud and the stud guide.

1. INTRODUCTION

The XM129 is an automatic, 40mm, grenade launcher powered by an electric motor. The weapon has a fixed breech and a reciprocating barrel. The operation is controlled by three cams: the barrel cam, the locking cam, and the feed cam. Chambering is accomplished as the barrel is driven rearward over the round by an internal drum type cam; this action incloses the cartridge and disconnects the link from the adjacent round in the belt. At the same time, the barrel cocks the firing mechanism. When the barrel reaches the rearward position, the locking cam provides translational motion to a sliding barrel lock which engages a recess provided in the barrel. After the firing and dwell portions of the cycle have been completed, the barrel is unlocked and driven forward. The feed cam then functions the feed system, which is a gear and lever arrangement driving a feed slide. The weapon cycle is then completed.

2. PROCEDURE

a. A dynamic force analysis was conducted to determine the stresses and the power requirements. General equations, defining the theoretical loads on the critical components, were derived. The equations were then evaluated and the results tabulated (Appendix A).

b. The results of the force analysis were used to calculate the power requirements (Appendix B).

c. The calculated maximum stresses for the critical components based on the theoretical force analysis are given in Appendix C.

3. CONDITIONS OF ANALYSIS

Analysis of the XM129 was based upon the assumptions that:

a. the rate of fire of the weapon is 300 rounds per minute,

b. total weight of the ammunition belt is 60 pounds,

c. belt pull is constant,

d. coefficients of friction are values believed reasonable with normal surface finishes and lubrication conditions,

e. masses of relatively light components can be ignored,

f. bodies under load are in a biaxial state of stress,

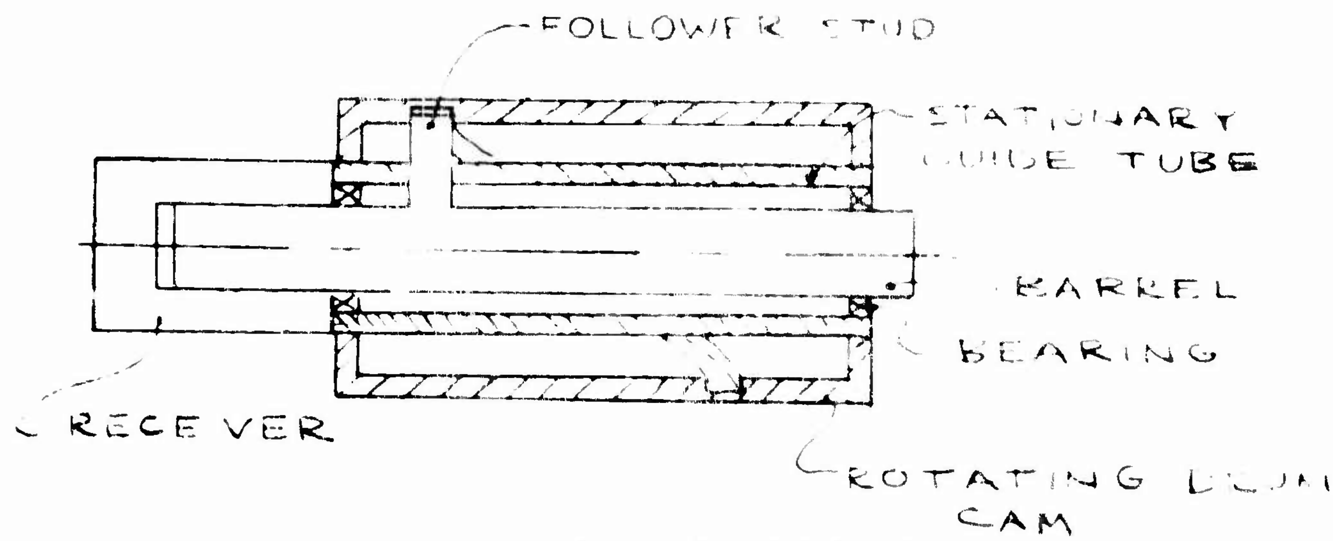
g. frictional forces of small magnitude can be neglected.

4. SYMBOLS

p	Pressure
P_g	Gear diametral pitch
P_f	Cam driving force
P_a	Ammunition belt pitch
P_c	Horizontal force on feed lever
P_t	Feed gear contact force
P_d	Force on cam follower
P_l	Total ammunition belt load
P_b	Ammunition belt inertia load
$F_1 \ldots$	Force
Q	Link stripping force
T	Torque
$R_1 \ldots$	Reaction forces
W	Weight
W_1	Dynamic tooth load
W_a	Ammunition belt weight per linear inch
W_b	Total ammunition belt weight
V_f	Feed velocity
K	Link extension rate
I	Moment of inertia
M	Moment
Y	Gear tooth form factor
X	Displacement
$\dot{X}$	Velocity

4. SYMBOLS - Continued

$\ddot{X}$	Acceleration
m	Mass
θ	Angular displacement
ω	Angular velocity
α	Angular acceleration
$\theta, \Phi, \beta, \alpha$	Angles
μ, α	Coefficients of friction
a,b,....	Dimensional constants



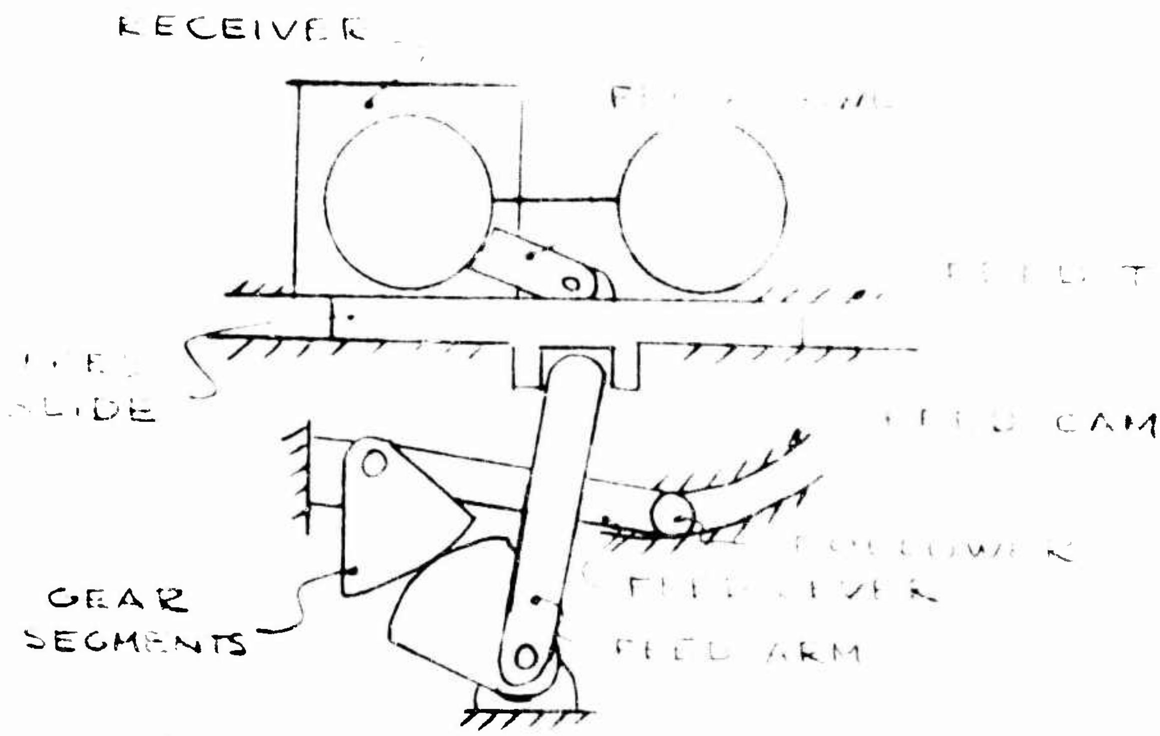
Schematic of Barrel and Drive Cam

5. BARREL AND DRIVE CAM ANALYSIS

The barrel assembly consists primarily of the barrel and a follower stud, which is fastened to the top. The barrel is guided by a bearing at the front and by the receiver at the rear. The reactions at the points are shown on the barrel free bo diagrams (Sheet 2, Appendix A) as R_1, R_2, F_1, and F_2. Rotational mot on is prevented by the follower stud which is guided by a longitudinal slot in a tube encircling the barrel. The reaction at this point is designated by T_1.

In the cam development (Sheet 1 Appendix A), X is the longitudinal displacement, R is the cam radius, and θ is angular displacement. From the development, it can be seen that the barrel is accelerated or decelerated in four areas. In these areas, the displacement equation is parabolic, which results in constant acceleration (Sheet 1, Appendix A). This means that, at any particular speed, the only variables in the analysis result from the geometric configuration. Maximum forces were determined by use of dynamic equilibrium equations derived at the eight points indicated on the cam layout, at the beginning and end of each parabolic segment. The equations were evaluated at these points with the use of the dimensions obtained from drawings and layouts. The results are tabulated on Sheet 6, Appendix A.

In the preliminary calculations, an assumed coefficient of sliding friction of 0.2 was used. This resulted in longitudinal friction forces higher than the driving force; in which case, the mechanism would be self-locking. A coefficient of 0.15, however, would give satisfactory results. In the field, the presence of dirt, wear, etc., could cause the coefficient to reach the 0.2 value, resulting in stalling. Since the major portion of the friction occurs at the interface between the follower stud and the guide tube, a roller device should be incorporated at this point to insure that the friction remains low.



Schematic of Feed System

6. FEED SYSTEM ANALYSIS

The feed cam is a face cam secured to the rear surface of the cam drum. The cam imparts motion to a follower which rotates the feed lever. The rotary motion is transferred through gear segments to the feed arm which in turn drives the feed slide. The feed slide is guided in the feed tray and, by means of a pawl, feeds the ammunition belt.

The belt load was calculated based on a constant feed velocity (Sheets 10, 11, 12; Appendix A). An expression was derived (Sheet 13,Appendix A) by equating the input and output torques in terms of the cam contact force. By use of this equation, the point at which the maximum cam force occurred was determined. Free body diagrams were then drawn for the individual components, and equations relating the forces were derived. The dimensions corresponding to the point of maximum cam contact force were substituted, and the forces on each component calculated (Sheets 14 through 17, Appendix A).

7. POWER ANALYSIS

The power required to operate the gun at a constant velocity is composed of two components: the power required to drive the barrel and the power required to function the feed system.

The power is the product of the average torque and the angular velocity.

7. POWER ANALYSIS - Continued

The torque curves are plotted on Sheets 2 and 3, Appendix B. The average torque is the area under the curve divided by 360 degrees. By use of the average torque, the power requirement was calculated (Sheet 4, Appendix B). The operating power is 0.70 HP.

The power required to accelerate the gun to operating speed in 1.25 revolutions can be approximated (Sheet 5, Appendix B). The moment of inertia of the rotating parts was calculated and the accelerating horsepower was calculated (Sheet 6, Appendix B). The accelerating power requirement is 0.49 HP.

8. STRESS ANALYSIS

a. The maximum forces (Appendix A) were used to calculate the stresses on the feed system components (Sheet 1-7, Appendix C). The maximum stress developed, which is less than 60,000 psi, is within the allowable limits.

b. The forces determined (Appendix A) were used to calculate the stresses on the barrel lock and on the barrel cam follower (Sheets 8 and 9, Appendix C). The stresses here are only 5000 psi (shear) in the lock and 4,400 psi in the follower.

c. The stresses in most of the components are of low magnitude. This indicates that the use of either low strength steels to decrease cost or lightweight alloys to reduce weight is possible.

APPENDICES

A - Barrel Force Analysis

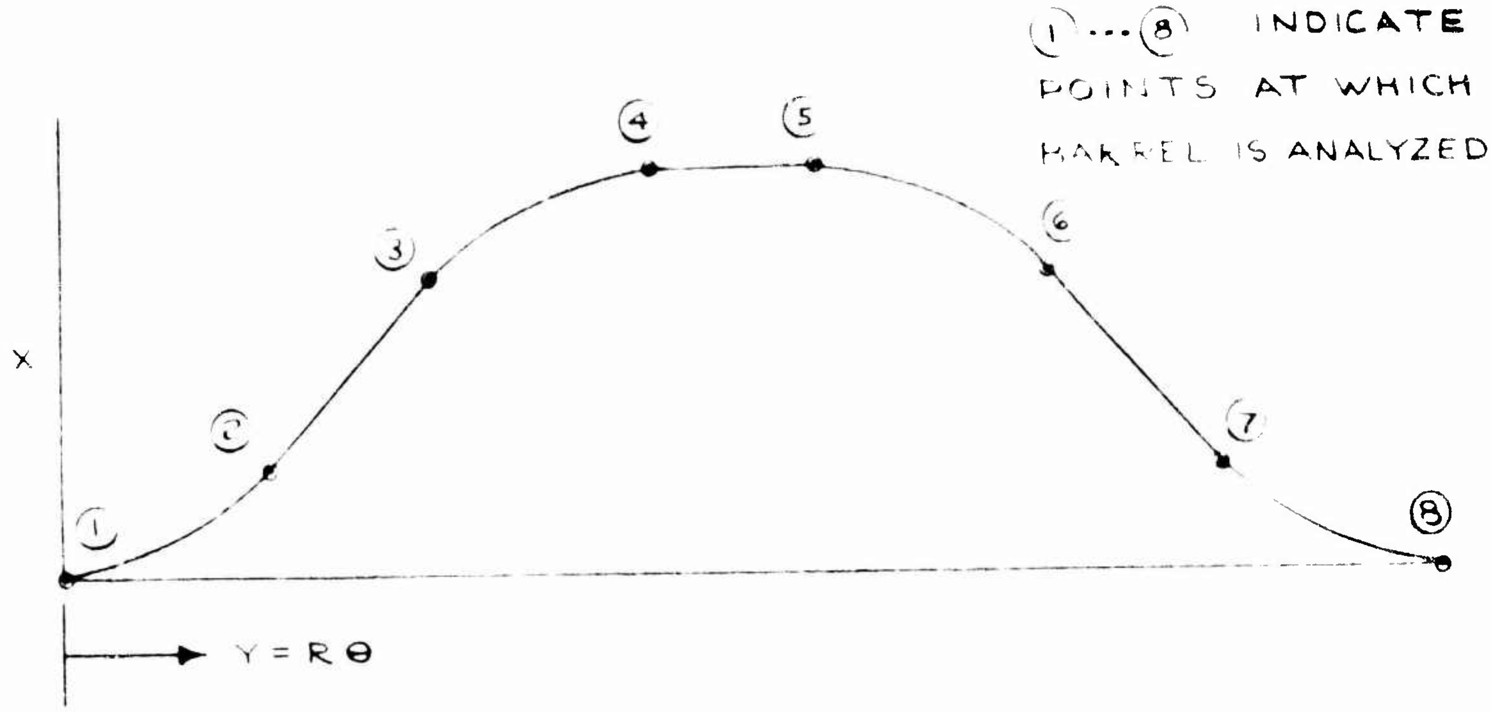
B - Feed System Force Analysis

C - Stress Analysis

D - Distribution

Barrel Force Analysis

BARREL FORCE ANALYSIS



DEVELOPMENT OF CAM

$X = KY^2$ FOR PARABOLIC SEGMENTS OF CAM

$Y = R\theta$

$X = KR^2\theta^2$

$\theta = \omega t$

$X = KR^2\omega^2 t^2$

$$\dot{X} = \frac{dX}{dt} = 2KR^2\omega^2 t$$

$$\ddot{X} = \frac{d^2X}{dt^2} = 2KR^2\omega^2$$

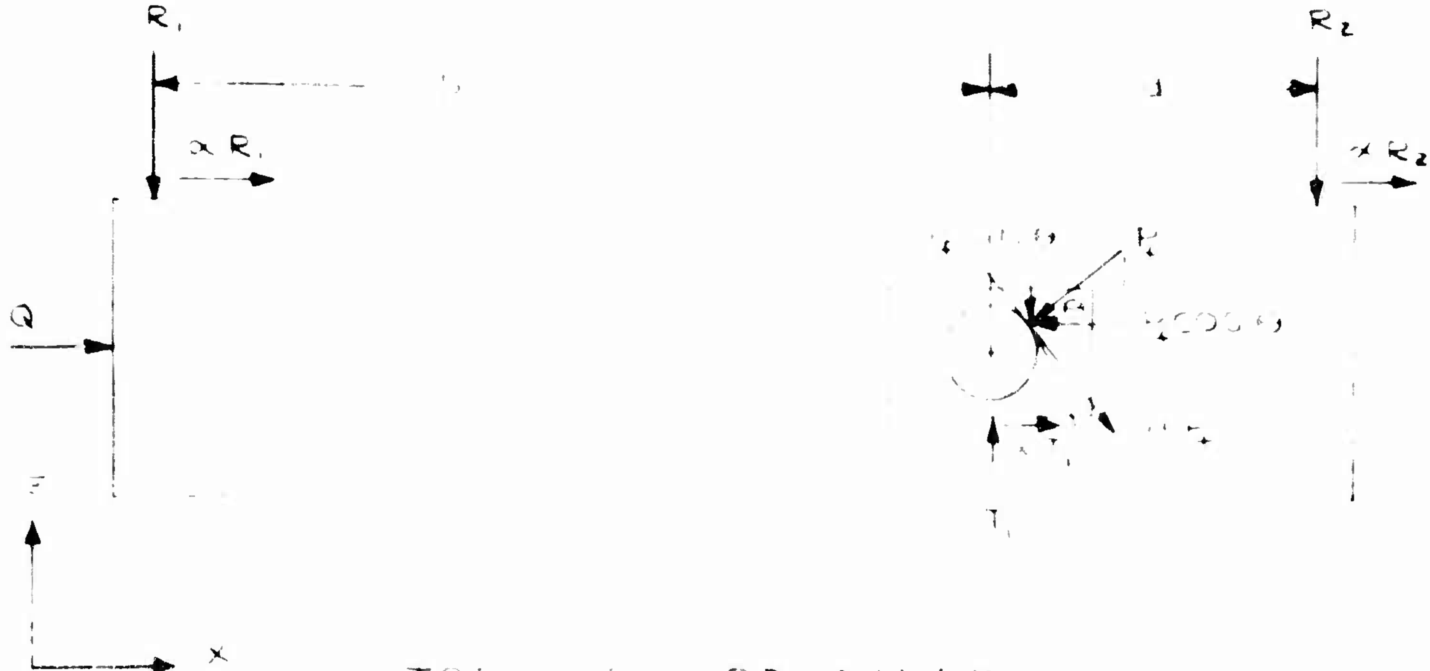
FROM DWG. SA-F42774

$K = .216$ $R = 2.6$ $\omega = 31.4$ RAD./SEC.

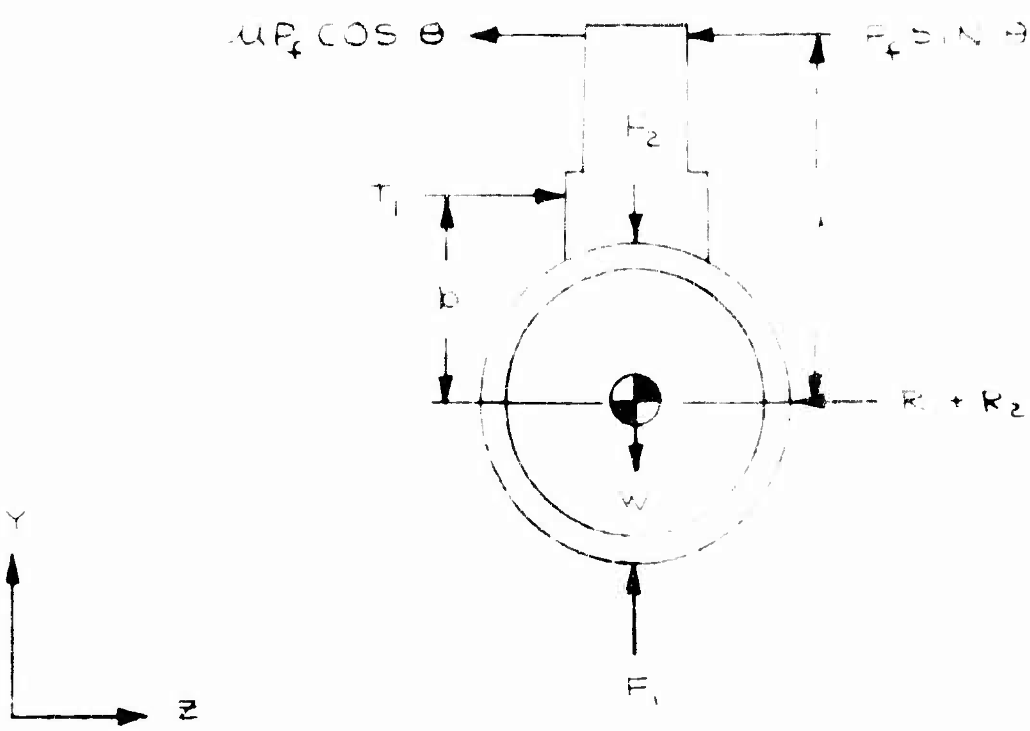
SUBSTITUTING IN ABOVE EQUATIONS

$\ddot{X} = 2866$ IN/SEC.^{2}

BARREL FORCE ANALYSIS - Continued



TOP VIEW OF BARREL
AT POSITIONS 1 2 3 4



END VIEW OF BARREL
AT POSITIONS 1 2 3 4

BARREL FORCE ANALYSIS - Continued

$$\underline{\Sigma \overset{\frown}{M}_{CG}(YZ) = 0}$$

$$bT_1 - a\,\mu\,P_f \cos\theta - a\,P_f \sin\theta$$

$$T_1 = \frac{a}{b}\left(P_f \sin\theta + \mu P_f \cos\theta\right) \qquad \underline{\text{EQ. 1}}$$

$$\underline{\Sigma F(YZ) = 0}$$

$$T_1 - \mu P_f \cos\theta - P_f \sin\theta - (R_1 + R_2) = 0$$

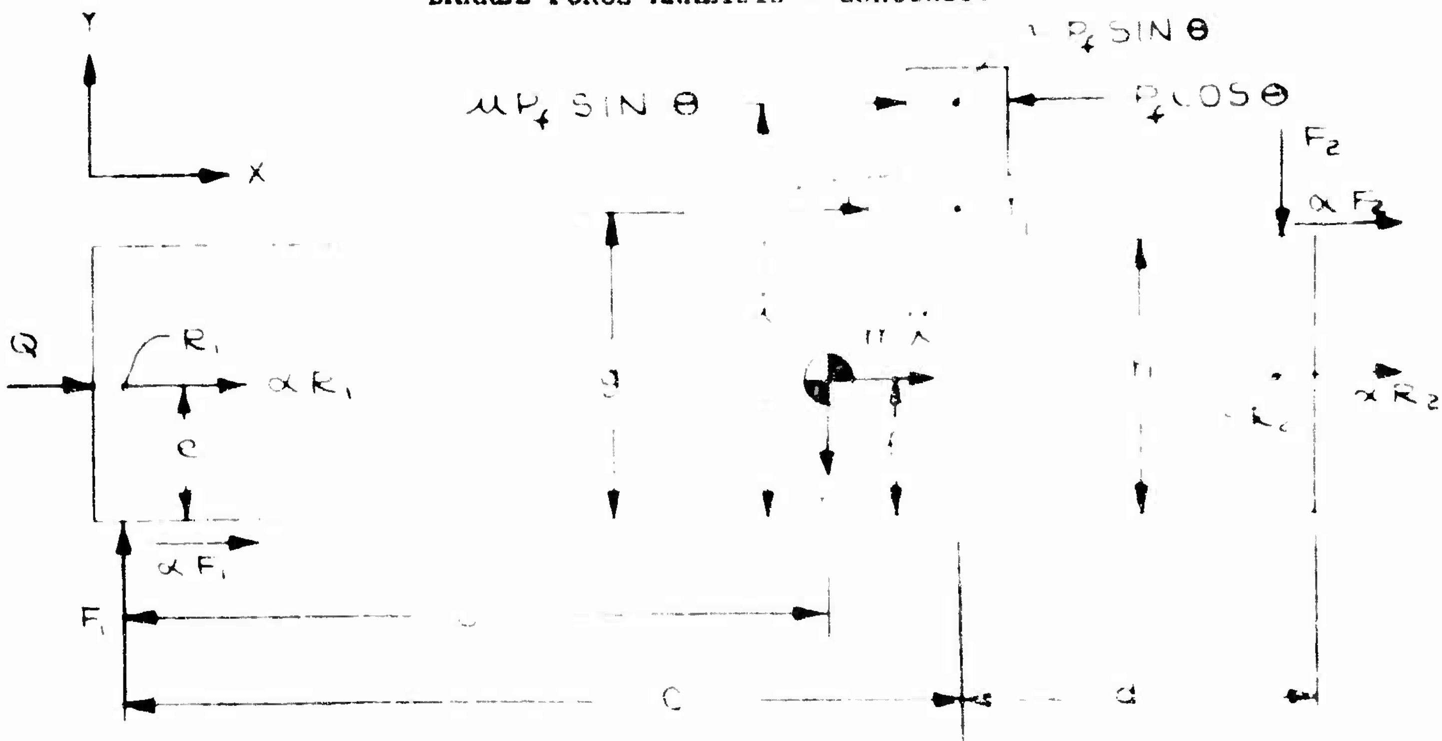
$$R_1 + R_2 = \frac{a}{b}\left(P_f \sin\theta + \mu P_f \cos\theta\right) - P_f \sin\theta - \mu P_f \cos\theta$$

$$R_1 + R_2 = \left(\frac{a}{b} - 1\right)\left(P_f \sin\theta + \mu P_f \cos\theta\right) \qquad \underline{\text{EQ. 2}}$$

$$R_1 = \frac{a + \alpha h/2}{s + d}\left(\frac{a}{b} - 1\right)\left(P_f \sin\theta + \mu P_f \cos\theta\right) \qquad \underline{\text{EQ. 3}}$$

$$R_2 = \frac{s - \alpha h/2}{s + a}\left(\frac{a}{b} - 1\right)\left(P_f \sin\theta + \mu P_f \cos\theta\right) \qquad \underline{\text{EQ. 4}}$$

BARREL FORCE ANALYSIS - Continued



SIDE VIEW OF BARREL
AT POSITIONS 1 2 3 4

$$\Sigma F\,(YX) = 0$$

$$F_1 = F_2 + W \qquad \text{EQ. 5}$$

$$\Sigma M_{F_1}\,(X,Y) = 0$$

$$F_2 = \frac{i P_f \cos\theta - i \mu P_f \sin\theta - \alpha T_1 g - \alpha (R_1 + R_2) x - WJ + m\ddot{x} - eQ}{c + d + \alpha + h} \qquad \text{EQ. 6}$$

$$\Sigma F_x\,(XY) = 0$$

$$Q + \alpha(R_1 + R_2) + \alpha(2F_2 + W) + m\ddot{x} + \alpha T_1 + \mu P_f \sin\theta - P_f \cos\theta = 0 \qquad \text{EQ. 7}$$

BARREL FORCE ANALYSIS - Continued

IF THE SAME ANALYSIS IS USED AT POINTS 5 6 7 8 THE FOLLOWING EQUATIONS RESULT

POSITIONS 5 & 6

$$T_1 = \frac{a}{b}\left(P_f \text{SIN}\,\theta + \mu P_f \text{COS}\,\theta\right) \qquad \text{EQ. 8}$$

$$R_1 + R_2 = \left(\frac{a}{b} - 1\right)\left(P_f \text{SIN}\,\theta + \mu P_f \text{COS}\,\theta\right) \qquad \text{EQ. 9}$$

$$R_1 = \frac{d - \alpha h/2}{c + d}\left(\frac{a}{b} - 1\right)\left(P_f \text{SIN}\,\theta + \mu P_f \text{COS}\,\theta\right) \qquad \text{EQ. 10}$$

$$R_2 = \frac{c + \alpha h/2}{c + d}\left(\frac{a}{b} - 1\right)\left(P_f \text{SIN}\,\theta + \mu P_f \text{COS}\,\theta\right) \qquad \text{EQ. 11}$$

$$F_1 = \frac{\lambda P \text{COS}\,\theta - \lambda \mu P \text{SIN}\,\theta - \alpha T g - \alpha(R_1 + R_2)\ell + f m\ddot{x} - dW}{c + d + \alpha h} \qquad \text{EQ. 12}$$

$$P_f \text{COS}\,\theta - \mu P_f \text{SIN}\,\theta + m\ddot{x} - \alpha T_1 - \alpha(R_1 + R_2) - \alpha(2F_1 + W) = 0 \qquad \text{EQ. 13}$$

POSITIONS 7 & 8

$$T_1 = \frac{a}{b}\left(P_f \text{SIN}\,\theta - \mu P_f \text{COS}\,\theta\right) \qquad \text{EQ. 14}$$

$$R_1 + R_2 = \left(\frac{a}{b} - 1\right)\left(P_f \text{SIN}\,\theta - \mu P_f \text{COS}\,\theta\right) \qquad \text{EQ. 15}$$

$$R_1 = \frac{d - \alpha h/2}{c + d}\left(\frac{a}{b} - 1\right)\left(P_f \text{SIN}\,\theta - \mu P_f \text{COS}\,\theta\right) \qquad \text{EQ. 16}$$

$$R_2 = \frac{c + \alpha h/2}{c + d}\left(\frac{a}{b} - 1\right)\left(P_f \text{SIN}\,\theta - \mu P_f \text{COS}\,\theta\right) \qquad \text{EQ. 17}$$

$$F_2 = \frac{\alpha(R_1 + R_2)\ell + \lambda \mu P_f \text{SIN}\,\theta + \lambda P_f \text{COS}\,\theta + m\ddot{x} - Wd + \alpha T_1 g}{c + d + \alpha h} \qquad \text{EQ. 18}$$

$$m\ddot{x} - \mu P_f \text{SIN}\,\theta - P_f \text{COS}\,\theta - \alpha T_1 - \alpha(2F_2 + W) - \alpha(R_1 + R_2) = 0 \qquad \text{EQ. 19}$$

BARREL FORCE ANALYSIS - Continued

VALUES FOR CONSTANTS

a = 2.50	g = 2.20	k = 4.00
b = 1.00	h = 1.87	α = [illegible]
e = .935	i = 2.44	μ = 0.10
f = 0.94	J = 4.00	Q = 30.0

POS	1	2	3	4	5	6	7	8
θ	0°	45°	45°	0°	0°	45°	45°	0°
c	4"	4"	4"	4"	4"	4"	4"	4"
d	1.5"	2.5"	5"	6"	6	5	2.5	1.5
S	6.5"	5.5"	3"	2"	2	3	5.5'	6.5"
Q	0	0	30#	30#	0	0	0	0
$\ddot{X}$	2866	2866	-2866	-2866	2866	2866	2866	2866

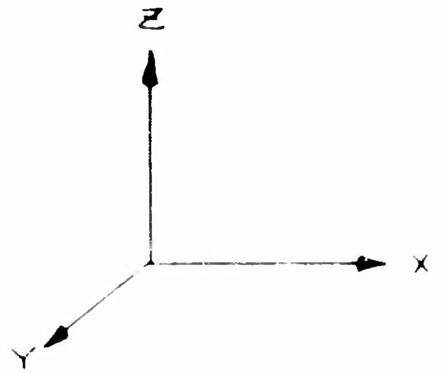
TABLE 1

SOLVING EQUATIONS (1) THROUGH (13) FOR P YIELDS

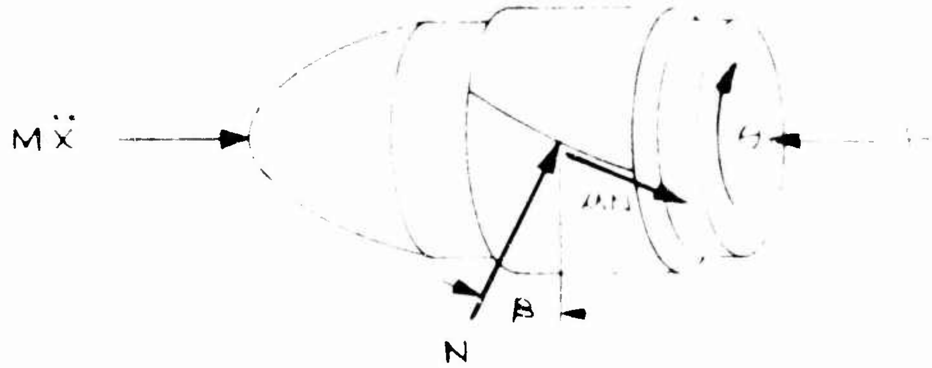
POS	1	2	3	4	5	6	7	8
P	25.1	100.1	34.1	9.0	25.1	99.0	-19.0	-19.2

SHEET 6 of 17

BARREL FORCE ANALYSIS - Continued



FREE BODY DIAGRAM OF PROJECTILE IN BORE



N=LAND NORMAL FORCE

M=PROJECTILE MASS

μ=COEFFICIENT OF FRICTION

$\ddot{X}$=AXIAL ACCELERATION

$\ddot{\theta}$=ANGULAR ACCELERATION

β=LAND ANGLE

I=PROJECTILE MOMENT OF INERTIA

R=PROJECTILE RADIUS

Y=TANGENTIAL DISPLACEMENT

θ= ANGULAR DISPLACEMENT

X=AXIAL DISPLACEMENT

$\Sigma F_x = 0$

$M\ddot{X} + N\sin\beta + \mu N\cos\beta - F = 0$ EQ. 1.

$\Sigma M_x = 0$

$RN\cos\beta - R\mu N\sin\beta - I\ddot{\theta}$ EQ. 2.

$Y = \tan\beta\, X$

$R\theta = \tan\beta\, X$

$R\dot{\theta} = \tan\beta\, \dot{X}$

$R\ddot{\theta} = \tan\beta\, \ddot{X}$

$$\ddot{X} = \frac{R\ddot{\theta}}{\tan\beta} \qquad \text{EQ. 3}$$

COMBINING EQUATIONS ① & ③

$$\ddot{\theta} = (F - N\sin\beta - \mu N\cos\beta)\frac{\tan\beta}{RM}$$

COMBINING WITH EQUATION ②

$$R(N\cos\beta - \mu N\sin\beta) - \frac{I}{RM}\tan\beta(F - N\sin\beta - \mu N\cos\beta) = 0$$

$M = .00097$ $\mu = .3$

$I = .0003$ $F = pA = 17,350$

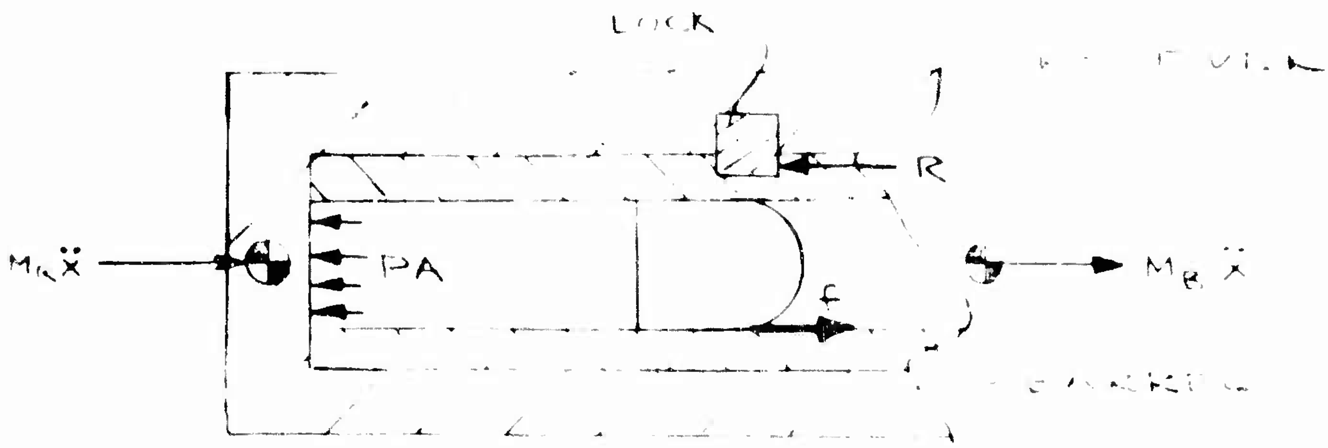
$R = .785$ $\beta = 6^\circ$

SOLVING FOR N

$N = 310$ #

$\text{TORQUE} = (N\cos\beta - \mu N\sin\beta)R$

TORQUE = 705 # IN.



M_R = MASS OF GUN - MASS OF BARREL

M_B = MASS OF BARREL

PA = GAS PRESSURE FORCE

f = TANGENTIAL RIFLING REACTION

R = FORCE ON LOCK

EQUILIBRIUM EQUATION OF ENTIRE GUN

$$\ddot{x} = \frac{PA - f}{M_R + M_B} \qquad \text{EQ.1}$$

EQUILIBRIUM EQUATION OF BARREL

$$R = f + M_B \ddot{x} \qquad \text{EQ.2}$$

EQ.1 + 2

$$R = \frac{M_B}{M_R + M_B}(PA) + f\left(1 - \frac{M_B}{M_R + M_B}\right) \qquad \text{EQ.3}$$

$$R = \frac{2.5}{42}(19{,}000) + 232\left(1 - \frac{2.5}{4.2}\right)$$

R = 1350 #

CALCULATION OF BELT LOAD

$$P_L = P_b + \alpha W_b \qquad \text{EQ. A}$$

$$P_b = \frac{V_F}{19.65}\left(\frac{P_a W}{K}\right)^{\frac{1}{2}} \qquad \text{EQ. B}$$

WHERE:

P_L = TOTAL BELT LOAD

P_b = BELT INERTIA LOAD

W_b = TOTAL BELT WEIGHT

V_F = FEED VELOCITY

P_a = PITCH

W = BELT WEIGHT PER LINEAR INCH

K = LINK EXTENSION RATE

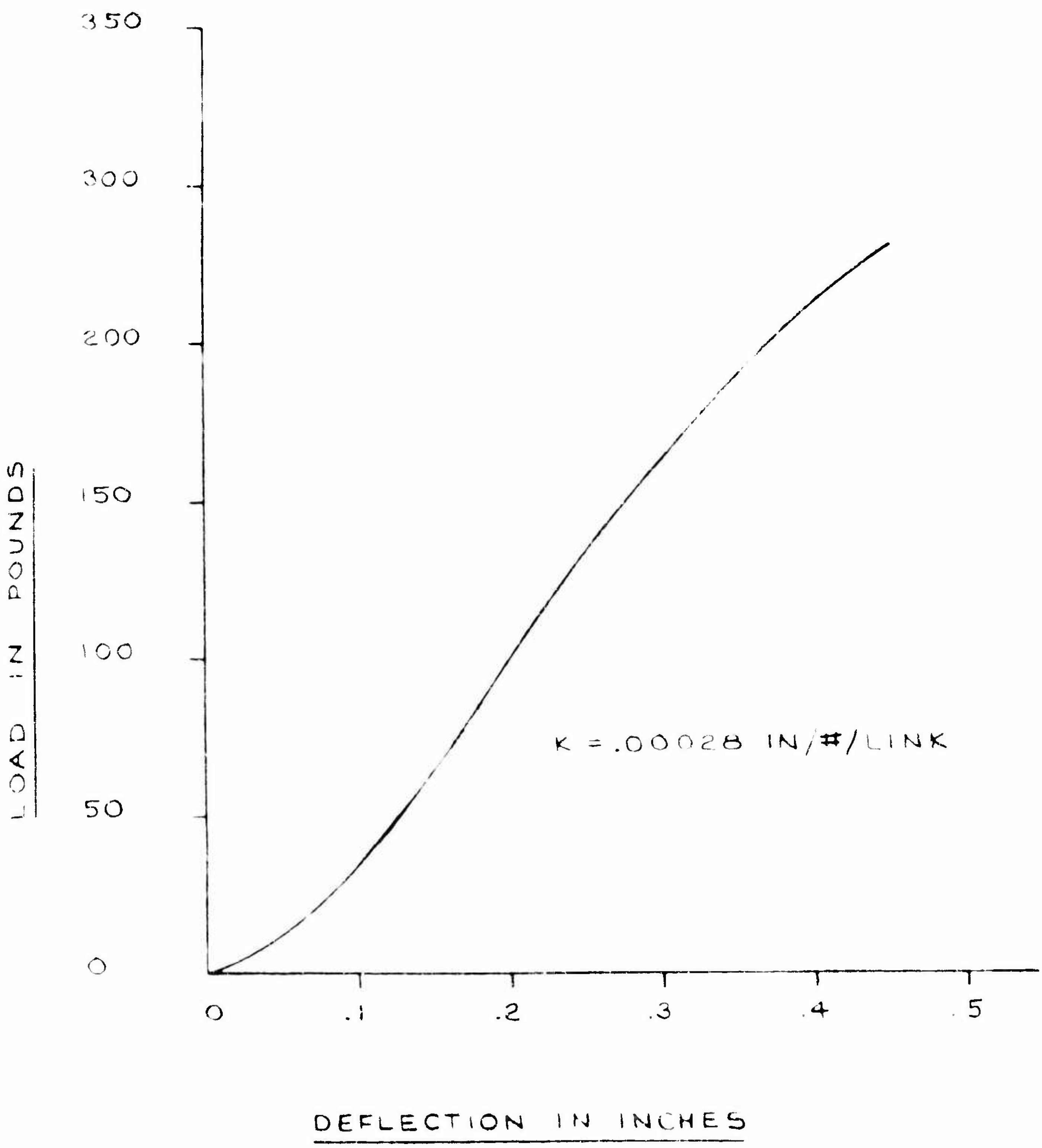
$$V_F = \frac{\omega P \, 360^\circ}{\Delta \theta} \qquad \text{EQ. C}$$

WHERE:

ω = ANGULAR VELOCITY

$\Delta\theta$ = ANGULAR DISPLACEMENT DURING FEEDING

LOAD DEFLECTION CURVE
OF FIVE POUND BELT



K = .00028 IN/#/LINK

W= .342 #/IN

P_d = 1.9 IN

ω= 5 REV/SEC

$\Delta\theta = 44^\circ$

FROM EQUATION C

V_F = 77 IN/SEC

FROM EQUATION B

$$P_b = \frac{77}{19.65}\left(\frac{1.9 \times .342}{.00028}\right)^{\frac{1}{2}}$$

$\alpha \approx .15$

$W_b \approx 60\#$

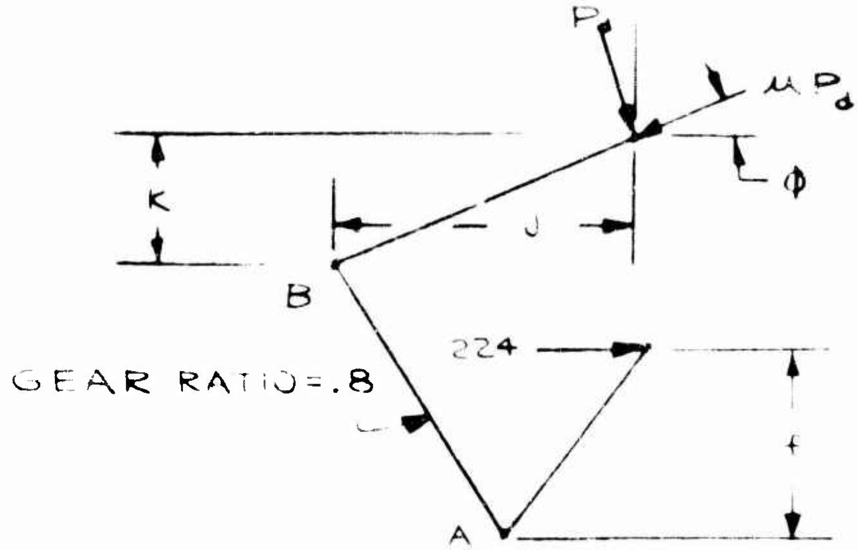
FROM EQUATION A

P_L = 200 #

$P = P_L / \cos\theta$

P = 230 #

BARREL FORCE ANALYSIS - Continued



FREE BODY DIAGRAM OF FEED ASSEMBLY

$$(.8)\sum \overset{\frown}{M_A} = \sum \overset{\frown}{M_B}$$

$$(.8)224\,f = P_d J(\cos\phi + \mu \sin\phi) + P_d K(\sin\phi - \mu\cos\phi) \qquad \text{EQ. D}$$

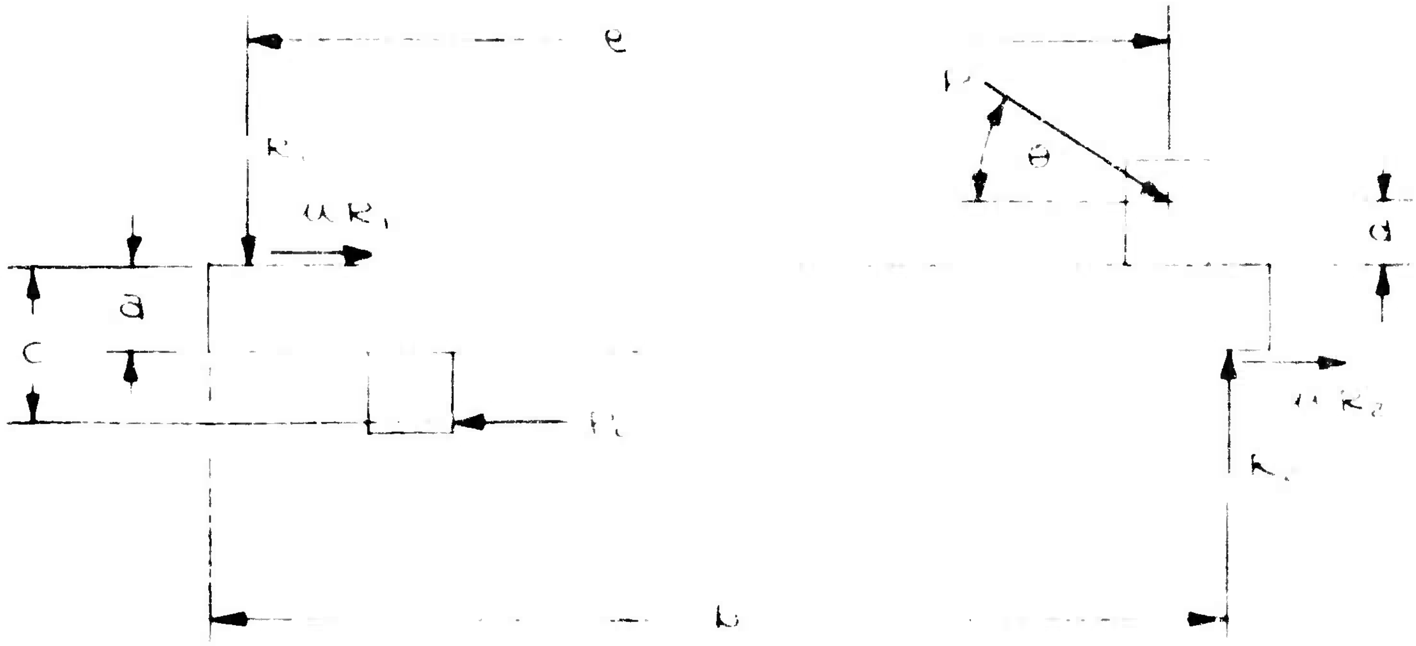
EQUATION D IS SOLVED AT 5 POINTS

FROM LAYOUT

POS	1	2	3	4	5
μ	.08	.08	.08	.08	.08
ϕ	0°	38°	52°	50°	0°
f	1.85	2.05	2.12	2.0	1.8
J	1.10	1.20	1.22	1.19	1.07
K	.55	.30	0	-.28	-.50
P	313	313	460	580	290

TABLE 3

BARREL FORCE ANALYSIS - Continued



FREE BODY DIAGRAM OF FEED SLIDE

$\underline{\Sigma F_x = 0}$

$$\mu R_1 + \mu R_2 + P\cos\theta + m\ddot{x} - P_c = 0 \qquad \text{EQ. 1}$$

$\underline{\Sigma F_Y = 0}$

$$R_1 + P\sin\theta - R_2 = 0 \qquad \text{EQ. 2}$$

$\underline{\Sigma M_{R_1}}$

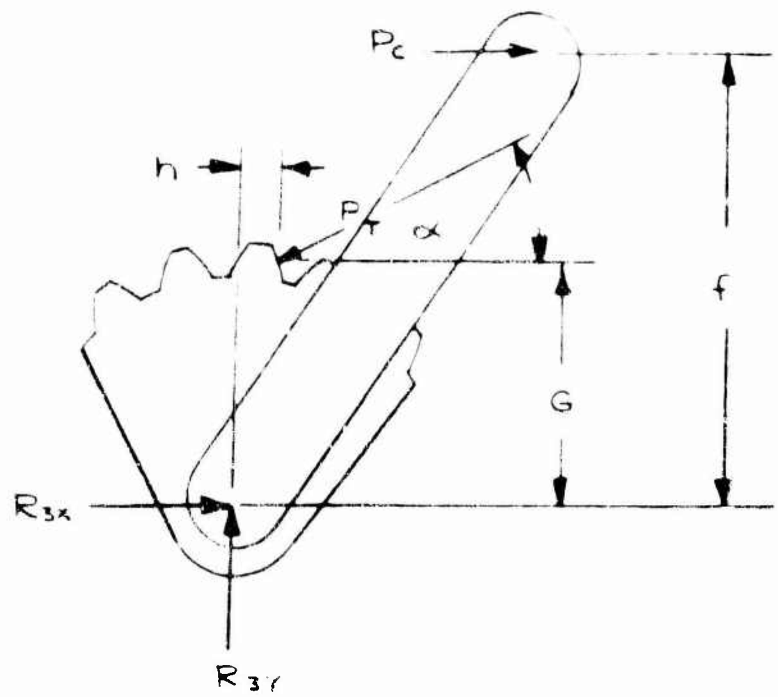
$$bR_2 + a\mu R_2 + d/2\, m\ddot{x} - cP_c - eP\sin\theta - dP\cos\theta \qquad \text{EQ. 3}$$

COMBINING EQUATIONS (1) (2) & (3)

$$(P_c + \mu P\sin\theta - P\cos\theta - m\ddot{x})\left(\frac{b}{2\mu} + \frac{a\mu}{2\mu}\right) + a/2\, m\ddot{x} - cP_c$$

$$- eP\sin\theta - dP\cos\theta = 0 \qquad \text{EQ. 4}$$

BARREL FORCE ANALYSIS - Continued



FREE BODY DIAGRAM OF FEED LEVER

$\Sigma F_x = 0$

$P_T \cos\alpha - R_{3x} - P_c = 0$ EQ. 5

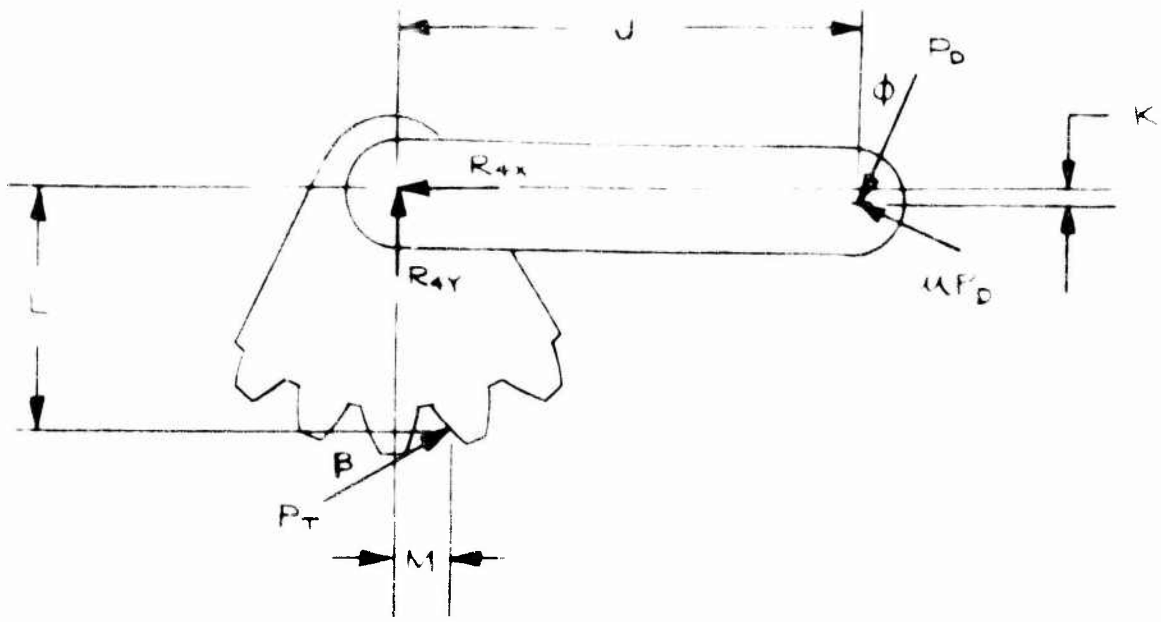
$\Sigma F_Y = 0$

$P_T \sin\alpha - R_{3Y} = 0$ EQ. 6

$\Sigma \overset{\frown}{M_{R3}} = 0$

$f P_c + h P_T \sin\alpha - G P_T \cos\alpha = 0$ EQ. 7

BARREL FORCE ANALYSIS - Continued



FREE BODY DIAGRAM OF CAM FOLLOWER AND GEAR

$\Sigma F_X = 0$

$$P_D \sin\phi + R_{4X} - P_T \cos B + \mu P_D \cos\phi = 0 \qquad \text{EQ. 8}$$

$\Sigma F_Y = 0$

$$P_D \cos\phi - R_{4Y} - P_T \sin B - \mu P_D \sin\phi = 0 \qquad \text{EQ. 9}$$

$\Sigma M_{R4} = 0$

$$L P_T \cos B + M P_T \sin B - K P_D \sin\phi - J P_D \cos\phi - K \mu P_D \cos\phi + J \mu P_D \sin\phi = 0 \qquad \text{EQ. 10}$$

FROM LAYOUT OF ASSEMBLY AT POINT 4

$a = .19$	$e =$ [illegible] 8	$J = 1.19$	$\theta =$ [illegible]$°$	$\phi = 50$
$b = 4.2$	$f =$ [illegible]$.0$	$K =$ [illegible]8	$\alpha =$ [illegible]	
$c = .25$	$g = .7$	$L = .8$	$\beta =$ [illegible]	
$d = .25$	$h =$ [illegible]$.5$	$M = .65$	$\mu = .08, .15$	

FROM EQUATIONS 1 → 4

$P_c = 224$ #

$R_1 = 3.25$ #

$R_2 = 128.2$ #

$m\ddot{x} = 4.0$ #

$\mu R_1 = 1.98$ #

$\mu R_2 = 19.24$ #

FROM EQUATIONS 5 → 7

$P_T = 553$ #

$R_{3Y} = 453$ #

$R_{3X} = 93$ #

FROM EQUATIONS 8 → 10

$P_D = 575$ #

$R_{4X} = 695$ #

$R_{4Y} = -19$ #

Power Analysis

POWER ANALYSIS

TORQUE REQUIRED TO DRIVE BARREL

TORQUE = FR

WHERE:

F = CAM DRIVING FORCE (TANGENTIAL)

R = CAM RADIUS

VALUES FOR F WERE TAKEN FROM TABLE 2

AND THE TORQUE CURVE PLOTTED ON SHEET

AVERAGE TORQUE IS THE AREA UNDER THE

CURVE DIVIDED BY 360°

AVERAGE TORQUE = 70 IN·#

TORQUE REQUIRED TO FEED

TORQUE = SR

WHERE:

S = CAM DRIVING FORCE (TANGENTIAL)

R = CAM RADIUS

VALUES FOR S WERE TAKEN FROM TABLE 3

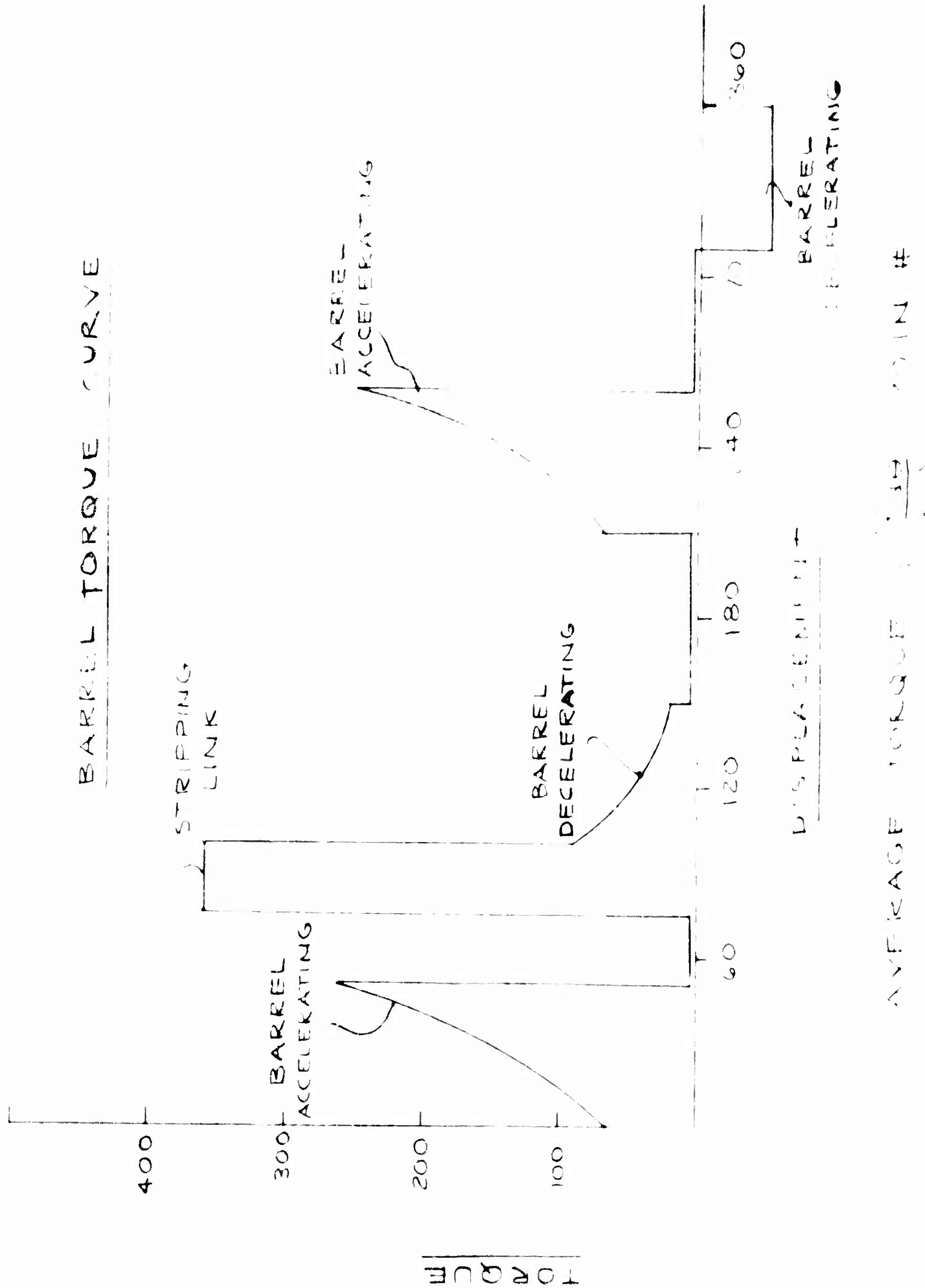
AND VALUES FOR R FROM DWG. SA-D-49291.

TORQUE CURVE IS PLOTTED ON SHEET .

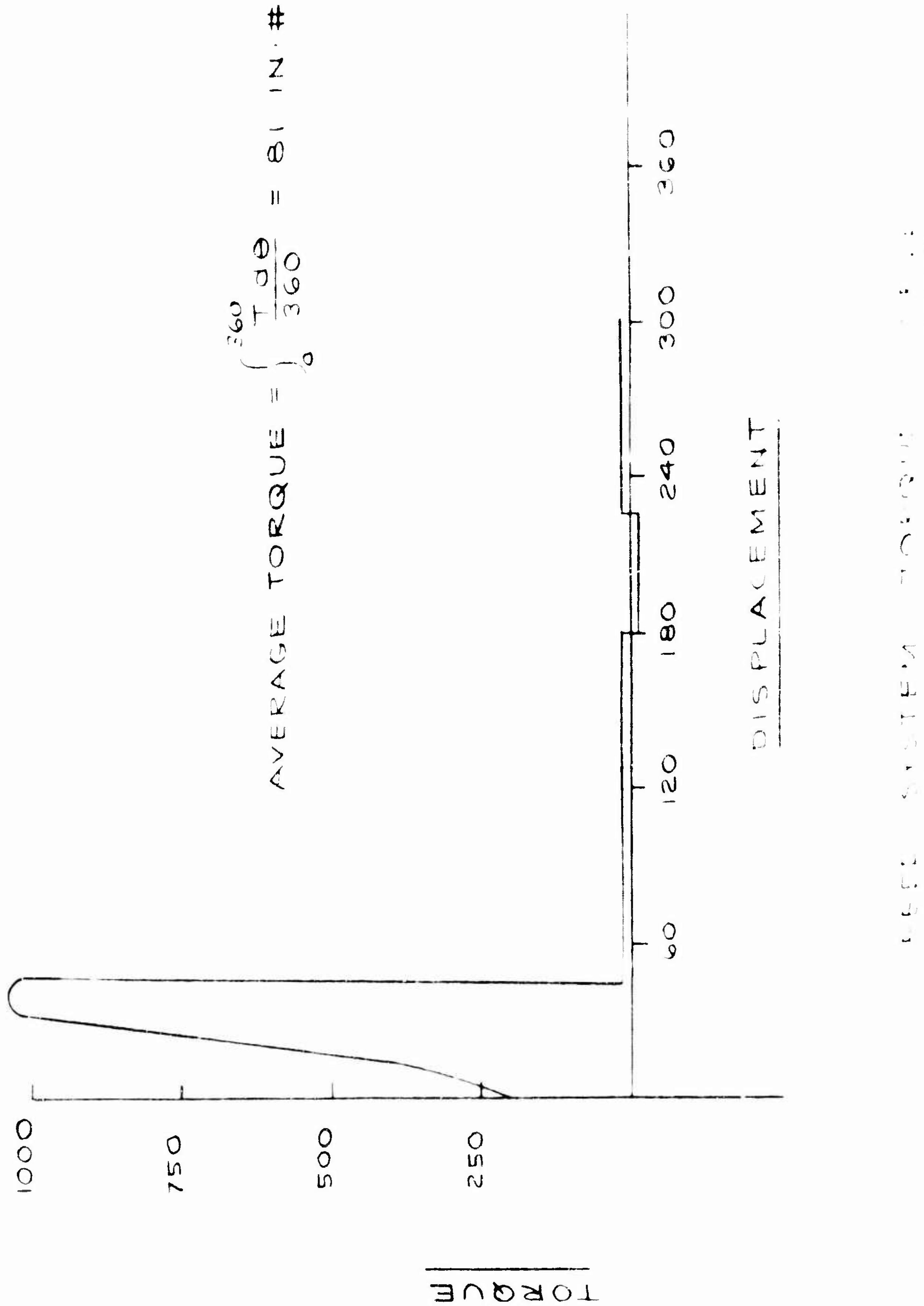
AVERAGE TORQUE = 81 IN·#

TOTAL AVERAGE TORQUE T_{AV} = 151 IN·#

POWER ANALYSIS



SHEET 2 of 6


AVERAGE TORQUE = ∫ T dθ / 360 = 81 IN·#
360
300
240
180
120
60
DISPLACEMENT
1000
750
500
250
TORQUE

POWER ANALYSIS

HORSEPOWER REQUIRED TO OPERATE GUN AT 300 R.P.M.

$$H.P. = \frac{T \times \omega \times 2\pi}{12 \times 550}$$

WHERE:

T = AVERAGE TORQUE IN IN.#

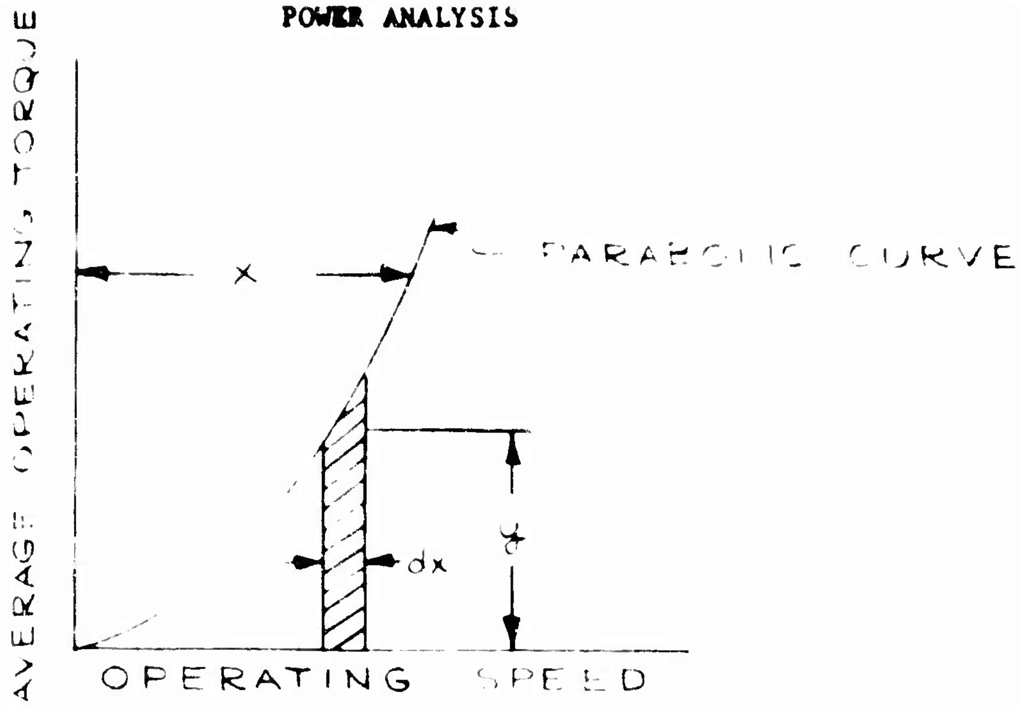
ω = ANGULAR VELOCITY IN REV/SEC

$$H.P. = \frac{151 \times 5 \times 2\pi}{12 \times 550}$$

H.P. = .70

SHEET 4 of 6

POWER ANALYSIS



ACCELERATING TORQUE IS THE TORQUE TO ACCELERATE THE DRUM AND CAMS PLUS THE AVERAGE OPERATING TORQUE, FROM ZERO R.P.M. TO OPERATING SPEED.

AVERAGE $Y = \dfrac{\int_0^x y\,dx}{x}$

$$y = Kx^2$$

$$Y_{AV} = \frac{Kx^2}{3}$$

ACCELERATING TORQUE $= I\alpha + \dfrac{T_0}{3}$

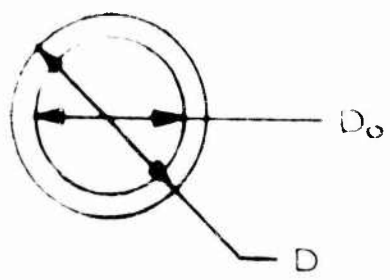
WHERE:

I = MOMENT OF INERTIA OF DRUM AND CAMS

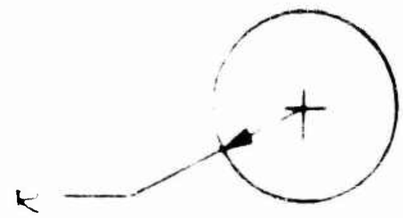
α = ANGULAR ACCELERATION

T = OPERATING TORQUE

POWER ANALYSIS



DRUM AND MAIN CAM



GEAR, FEED, AND LOCK CAMS

$$I_D = m\pi\ell\frac{\left(\left[\frac{D}{2}\right]^4 - \left[\frac{D_o}{2}\right]^4\right)}{2}$$

WHERE:

ℓ = LENGTH

m = MASS DENSITY

$$I_D \approx .205$$

$$I_G = \frac{MR^2}{2}$$

$$I_G \approx .36$$

$$I_T = I_D + I_G$$

$$I_T = .565$$

$$T_a = T\alpha + \frac{T_o}{3}$$

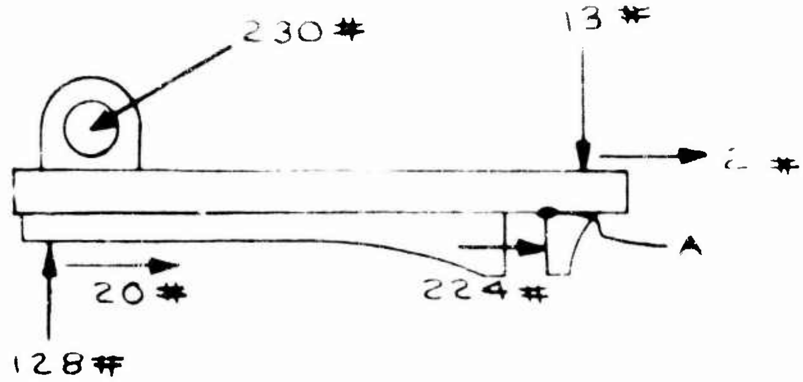
ω IS $0°/SEC \rightarrow 5°/SEC = 2.5°/SEC$ = 1.25 IN .5 SEC

$$T_a = .565\,(62.8) + \frac{151}{3}(1.25)$$

T_a = 98 IN·# (FOR 1.25 REV. OR .5 SEC)

H.P. = .47

STRESS ANALYSIS



FEED SLIDE

SHEAR IN PIN

$$S = \frac{P}{A}$$

$$S = \frac{230 \times 4}{2 \times .7854 \times .25^2}$$

S = 2400 P.S.I.

BEARING IN HOLE

$$S = \frac{P}{A}$$

$$S = \frac{230}{.484 \times .25}$$

S = 1900 P.S.I.

COMBINED BENDING AND COMPRESSION IN LUG

$$S = \frac{MC}{I} + \frac{P}{A}$$

$$S = \frac{230 \times .866 \times .26 \times .28 \times 12}{.48 \times .55^3} + \frac{230 \times .5}{.48 \times .55}$$

S = 2600 P.S.I.

BENDING IN FEED SLIDE PLATE

$$S = \frac{MC}{I} + \frac{P}{A}$$

$$S = \frac{224 \times .5 \times .094 \times 12}{.75 \times .188^3} + \frac{224}{.75 \times .188}$$

$S = 27{,}000$ P.S.I.

BENDING IN BOTTOM LUGS

$$S = \frac{MC}{I}$$

$$S = \frac{224 \times .5 \times .15 \times 12}{.5 \times .3^3}$$

$S = 14{,}900$ P.S.I.

SHEAR IN FEED SLIDE AT POINT A (TORSION)

$$S_s = \frac{T}{\alpha a b^2}$$ (FROM SPOTTS' "MACHINE DESIGN")

$$S_s = \frac{.5(112)}{.235 \times .31 \times .188^2}$$

$S_s = 21{,}600$ P.S.I.

SHEET 2 of 9

COMBINED BENDING AND SHEAR AT POINT A.

BY MOHR'S CIRCLE

$$(S_S)_{MAX} = (21{,}600^2 + 7{,}450^2)^{\frac{1}{2}}$$

$S_S = 23{,}000$ P.S.I.

SHEAR STRESS FROM FEED ARM CONTACT

$$S_C = 3190\left(\frac{P}{D}\right)^{\frac{1}{2}}$$

$S_C = 76{,}500$ P.S.I.

$S_S = .304\ S_C$

$S_S = 23{,}300$ P.S.I.

BENDING STRESS

$$S = \frac{MC}{I} = \frac{4P}{\pi r^3}$$

$$S = \frac{580 \times 4 \times .25}{3.14 \times .15^3}$$

$$S = 57000 \text{ P.S.I.}$$

CAM FOLLOWER CONTACT STRESS

$$S_c = 5190 \left(\frac{580}{.311}\right)^{\frac{1}{2}}$$

$$S_c = 137{,}000 \text{ P.S.I.}$$

$$S_s = .304 \; S_c$$

$$S_s = 41{,}600 \text{ P.S.I.}$$

FEED ARM AND GEAR ANALYSIS

SHEAR STRESS IN FOLLOWER PIN

$$S = \frac{P}{A} = \frac{224}{.109}$$

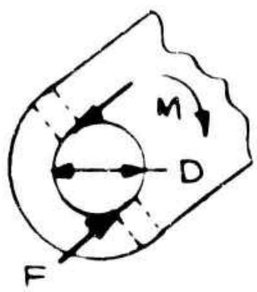
$$S = 2055 \text{ P.S.I.}$$

BEARING STRESS IN PIN HOLE

$$S = \frac{P}{A} = \frac{224}{.112}$$

$$S = 2000 \text{ P.S.I.}$$

SHEAR STRESS IN SHAFT PIN



$$M = 224 \times 2.1 = 470$$

$$F = \frac{M}{D} = \frac{470}{.5} = 940$$

$$S = \frac{P}{A} = \frac{940}{.0188}$$

$$S = 50{,}000 \text{ P.S.I.}$$

STRESS IN GEAR KEY

SHEAR ACROSS KEY WIDTH

$$S = \frac{P}{A} = \frac{1880}{.0625}$$

$S = 30,000$ P.S.I.

CRUSHING OF KEY

$$S = \frac{P}{A} = \frac{1880}{.03175}$$

$S = 59,200$ P.S.I.

GEAR TOOTH BENDING STRESS

$$S = \frac{WP}{fY}$$

WHERE:

Y = FORM FACTOR

W = DYNAMIC TOOTH LOAD

f = TOOTH WIDTH

P_d = DIAMETRAL PITCH

$$S = \frac{553 \times 12}{.5 \times 322}$$

S = 41,100 P.S.I.

SINCE ꜰ FOR MATING GEAR IS LARGER
THE STRESS WILL BE LOWER

SHEAR STRESS IN FEED GEAR PIN

$M = 1.125 \times 553 = 622$

$$F = \frac{622}{.6} = 1036$$

$$S = \frac{F}{A} = \frac{1036}{.027}$$

S = 38,400 P.S.I.

STRESS IN FOLLOWER ARM

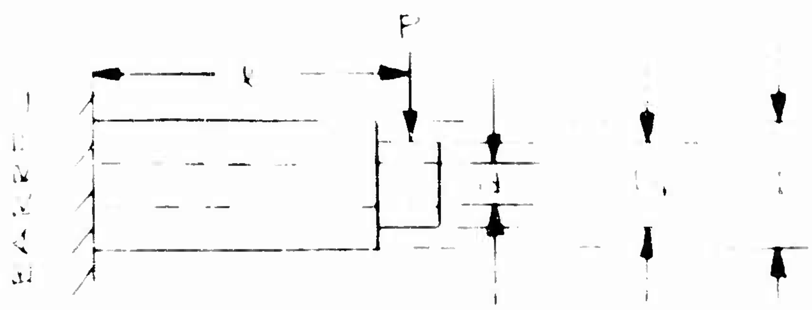
SHEAR STRESS

$$S = \frac{P}{A}$$

$$S = \frac{580}{.07}$$

S = 8,300 P.S.I.

STRESS IN [illegible] (BARREL)



P = 160 #   D = [illegible]   d = [illegible]

ℓ = 1 D_F = .4[illegible]6

SHEAR AT SHOULDER

$$S = \frac{F}{A}$$

$$S = \frac{P\,4}{\pi(D_F^2 - d^2)}$$

S = 1,350 P.S.I.

BENDING AT BARREL

$$S = \frac{M}{Z} = \frac{4MR}{\pi(R^4 - r^4)}$$

S = 4,400 P.S.I.

BEARING STRESS ON BARREL CAM FOLLOWER DURING FIRING

TORQUE = 705 IN #

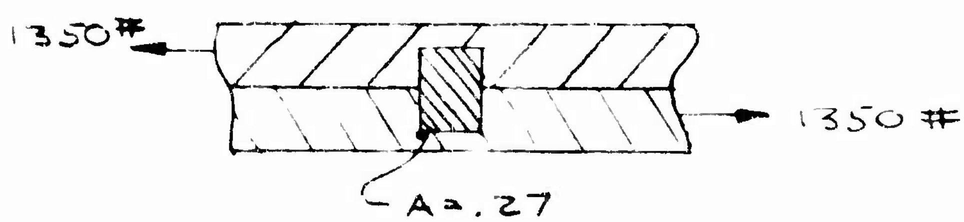
$$F = \frac{T}{R} = \frac{705}{1.2} = 590^{\#}$$

$$S = \frac{P}{A}$$

$$S = \frac{590}{.25}$$

S = 2,360 P.S.I.

SHEAR STRESS IN BARREL LOCK



$$S = \frac{P}{A}$$

$$S = \frac{1350}{.27}$$

S = 5000 P.S.I.