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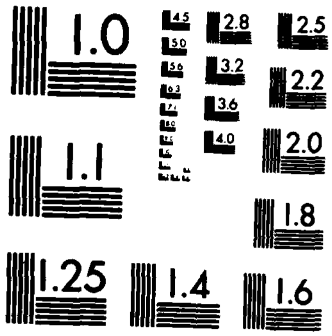

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TECHNICAL REPORT ARLCD-CR-84008

DEVELOPMENT OF LOWER COST TRIGGER SPACER FOR M577 FUZE

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APRIL 1984

U.S. ARMY
ARDC

U.S. ARMY ARMAMENT RESEARCH AND DEVELOPMENT CENTER
LARGE CALIBER WEAPON SYSTEMS LABORATORY
DOVER, NEW JERSEY

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The objective of this task was to reduce the cost of the trigger spacer assembly by changing material and simplifying fabrication processes. A new material, polyphenylene sulfide, 40% glass reinforced, replaced the current die cast aluminum. The trigger spacer assembly was made using the injection molding process with the spacer pins inserted during the molding process. The secondary and assembly operations required on the aluminum spacer were eliminated.

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CONTENTS

I. INTRODUCTION

This report describes the work accomplished by Hamilton Technology, Inc. for ARRADCOM under Task No. 3 of Contract #DAAK10-80-C-0203 from December 1980 through August 1982.

The objective of this task was to reduce the cost of the Trigger Spacer Assembly by changing material and simplifying fabrication processes, and eliminating the need to assemble the locating pins.

The current Trigger Spacer is produced in three processes. First, a blank Trigger Spacer is made as an aluminum die casting. Second, the blank Spacer is machined to produce the axial holes and final configuration. Then the top and bottom Spacer Pins are inserted to the machined Trigger Spacer. Various materials and fabrication processes were considered in this task to reduce the cost without degrading the performance of the Trigger Assembly.

This report contains six sections and appendixes. This Introduction is followed by Summary of Accomplishments (Section II), Conclusions and Recommendations (Section III), Technical Discussion (Section IV), Testing (Section V), Cost and Weight (Section VI), and Appendixes. The theoretical and fabrication aspects of the proposed design are discussed in Section IV, and the descriptions and results of the various tests are shown in Section V. The Appendixes contain the changes and modifications made in the proposed design, drawings of the proposed Trigger Parts, weight and mass of trigger parts, calculations of angular velocity and angular acceleration, and the firing reports.

II. SUMMARY OF ACCOMPLISHMENTS

A new material - polyphenylene sulfide, (PPS, 40% glass reinforced) was selected, and Trigger Spacers were made from the new material by injection molding with Spacer Pins inserted during molding. This plastic, 40% reinforced material was selected because of its high strength and dimensional stability. PPS possesses outstanding chemical resistance and thermal stability because of its crystalline structure.

The fabrication processes have been simplified because the injection molded Trigger Spacers do not require secondary machining operations and the process of assembling Spacer Pins. The new design of attaching the MDF to the three-module assembly (Contract No. DAAK10-79-C-0169, Task No. 2) has been accommodated in this task.

The theoretical aspects and fabrication processes were studied, and PPS Trigger Spacers were assembled. The PPS Trigger Assembly was subjected to a vigorous test program which included successful Ballistic testing in all standard tubes, static compressive loadings, airgun (30,000 g), spin tests (30,000 RPM), and accelerated aging tests such as thermal shock, chemical compatibility and temperature and humidity testing. The theory and the test results indicate that this PPS Trigger Spacer did function properly and safely under the design requirements of 30,000 g acceleration with a 30,000 RPM spin, and will survive long-term storage.

A cost reduction of $0.8799 per fuze has been projected from this task.

III. CONCLUSIONS AND RECOMMENDATIONS

The 40% glass reinforced PPS Trigger Assembly was subjected to a series of ballistic, air-gun and spin tests and various accelerated aging tests with successful results. The theory which was confirmed by the results verifies that the PPS Trigger Assembly will function properly and safely at 30,000-g acceleration, with 30,000-RPM spin design envelope.

Based upon the successful test program and the projected cost savings of $0.8799 per unit, this PPS Trigger Assembly is recommended for incorporation in the M577, MTSQ Fuze Technical Data Package.

IV. TECHNICAL DISCUSSION

A. Introduction

This section describes a feasibility study performed on the proposed material - polyphenylene sulfide (PPS, 40% glass reinforced).

A new material - polyphenylene sulfide, (PPS, 40% glass reinforced) was selected, and Trigger Spacers were made from the new material by injection molding with Spacer Pins inserted during molding. PPS (40% glass reinforced) was selected because of its high strength and dimensional stability. PPS is a high-strength thermo-plastic which has outstanding chemical resistance and thermal stability because of its crystalline structure. PPS requires a processing temperature of 600°F to 675°F for molding, and the resin has no known solvent below 400°F. The resin is inert to most organic solvent and aqueous inorganic salts over a wide temperature range.

Primary engineering applications for the injection molding PPS resins are mechanical and electrical-electronic equipments. The mechanical uses for PPS resins include gears, cams, pistons, bearings and such processing equipment applications as submersible, centrifugal, vane and gear-type pumps, where mechanical strength, high temperature stability and chemical resistance are needed.

There are military (MIL-P-46174) and ASTM (ASTM D-3646) specifications which define the material properties and requirements of the glass-reinforced PPS. In addition to MIL-P-46174, the following is a list of additional military specifications covering various applications of molded polyphenylene sulfide, glass fiber reinforced products:

a. MIL-C-28754

b. MIL-C-55302

c. MIL-C-83513

d. MIL-C-83734

e. MIL-S-83502

f. MIL-S-83505

Currently, the Insulator Spacer (ARRADCOM Dwg. No. 9327315), which was made from PPS (40% glass reinforced) by the injection molding process, is being used in the 105mm M456 Heat Cartridge.

This section contains three subsections. This Introduction is followed by Section IV-B (Theoretical Aspects) which contains discussions concerning mechanical properties, creep, aging, and chemical compatibility characteristics of PPS. The shear stress on the Spacer Pins due to the angular acceleration has been also analyzed in this section. Section IV-C (Fabrication) discusses the fabrication aspect of the proposed Trigger Assembly.

Several minor changes were made to the Trigger Assembly because of the changes in material and fabrication process. The load bearing area of the Trigger Spacer was increased to increase the margin of operability of the proposed Trigger Spacer. The top pivot hole of the Safety Plate Shaft was changed to a clearance hole, and the pivot hole was moved to the Top Plate. Consequently, the length of the Safety Plate Shaft was increased. The Spacer Pins were redesigned to contain grooves so that the insert molded pins can withstand the unseating load requirement, and the lengths of the top and bottom Spacer Pins were changed to the same length. These changes are discussed in Section IV-B.

Additional changes have been made to the Trigger Spacer, and they are discussed in Section V-B-1 (increased Trigger Spacer height) and Appendix C-5 (new method of attaching MDF). The detailed descriptions of changes and modifications are shown in Appendixes A and C, and the drawings are shown in Appendixes B, D, E and F.

B. Theoretical Aspects

The major areas of concern in the Trigger Spacer material substitution to polyphenylene sulfide (PPS, 40% glass reinforced) are strength and rigidity. Other important areas of concern are its long-term mechanical properties (creep, aging, and chemical compatibility) and fabrication properties (insert molding and staking capabilities). These subjects are discussed in this section.

1. Stress of the Trigger Spacer due to 30,000-g Acceleration

The Trigger Spacer is required to withstand the force due to 30,000-g acceleration of the Timing Scroll Assembly (less Mainspring Barrel), Top Plate and Trigger Spacer at the instant the projectile is fired. Since the compressive strength of PPS (21,000 psi)[1] is considerably lower than the die-cast aluminum (46,000 psi)[2], the compressive stress on the Trigger Spacer has been studied.

1. "Property Chart," Modern P ast' s Encyclopedia (McGraw-Hill Publication), Vol. 56, No. 10A (1979-80) . 516.
2. "Note 2," ARRADCOM Drawing No. 9236605, P. 2.

The stress on the Trigger Spacer can be calculated using the following standard stress formula:

$\sigma = P/A$

where σ is the compressive stress on the Trigger Spacer in psi

P is the force due to 30,000-g acceleration of the Timing Scroll Assembly (less Mainspring Barrel), Top Plate and Trigger Spacer in lbs.

A is the area of the Trigger Spacer being compressed against the Bottom Plate in inch^2

The approximate area ($A = 0.320\ \text{inch}^2$) was obtained from a sketch drawing because of the dimensional complexity.

To compare the stress on the aluminum and PPS Trigger Spacer, the compressive stress on the aluminum Trigger Spacer was calculated as follows:

$$
\begin{aligned}
P_a = ma &= (W_a/g)(30{,}000\ g) = 30{,}000\ W_a \\
&= 30{,}000\ (0.0758 + 0.0328 + 0.0625),\ \text{(See Appendix G)} \\
&= 5{,}133\ \text{lbs.}
\end{aligned}
$$

$$
\begin{aligned}
\sigma_a = P_a/A &= 5{,}133/0.320 \\
&= 16{,}000\ \text{psi}
\end{aligned}
$$

Now, the compressive stress on the PPS Trigger Spacer with the same configuration as the aluminum spacer was found as shown below.

$$
\begin{aligned}
P_p = ma &= (W_p/g)(30{,}000\ g) = 30{,}000\ W_p \\
&= 30{,}000\ (0.0758 + 0.0328 + 0.0418),\ \text{(See Appendix G)} \\
&= 4{,}512\ \text{lbs.}
\end{aligned}
$$

2. Deformation of the Trigger Spacer under 1,000 lbs. Compressive Load

The Trigger Assembly is required (in Drawing #9236603) to function properly at 2,000-RPM spin when the Trigger Assembly is loaded with 600 to 1,000 lbs. in compression. For the Trigger Assembly to function properly, the Trigger Spacer should not deform more than the minimum endshake allowed (0.003 inches) between the two trigger plates and the Firing Arm Shaft under 1,000 lbs. compressive loading. Because of the complex Trigger Spacer shape and the lack of Young's Modulus data on PPS, the deformation of the Trigger Spacer under 1,000 lbs. was calculated approximately as shown below.

$$\delta = \varepsilon \cdot \ell$$
$$= (\sigma / \text{Y.M.})$$

where δ = deformation in inches

ε = strain in inch/inch

ℓ = spacer height = 0.540 inches

σ = stress due to 1,000 lbs. in psi

Y.M. = Young's Modulus in psi

Since the Young's Modulus in compression is not known, the Y.M. in tension (1.7×10^6 psi) was used in the following calculation.[1]

1. "Property Chart," Modern Plastics Encyclopedia (McGraw-Hill publication), Vol. 56, No. 10A (1979-80), P. 516.

$$\sigma p = P_p/A = 4{,}512/0.320$$
$$= 14{,}000 \text{ psi}$$

The above calculations show that the margin of operability of the PPS spacer (margin of operability = material strength/ maximum stress expected = 21,000/14,000 = 1.5) is lower than the margin of operability of the aluminum spacer (margin of operability = 46,000/16,000 = 2.9).

Although the expected compressive stress on the PPS Trigger Spacer is lower than the material strength, the design change of increasing the bearing area was made to increase the margin of operability.

By modifying the Trigger Spacer dimensions (see Appendix C-2), the bearing area was increased from 0.320 inch^2 to 0.530 inch^2. The following stress calculation was made, based on this increased area.

$$\sigma p = P_p/A = 4{,}512/0.530$$
$$= 8{,}513 \text{ psi}$$

The margin of operability based on the increased bearing area was calculated to be 2.5.

Now, the expected compressive stress on the PPS Trigger Spacer is about 40% of the material strength, and the margin of operability of the PPS spacer (2.5) is comparable to the margin of operability of the aluminum spacer (2.9).

$$\sigma = P/A$$
$$= 1{,}000/0.530$$
$$= 1{,}900 \text{ psi}$$

$$\delta = (\sigma/\text{Y.M.}) \cdot \ell$$
$$= (1{,}900/1.7 \times 10^6)(0.540)$$
$$= 0.001 \text{ inch}$$

The calculation shows that the expected deformation (0.001 inch) is less than the minimum endshake allowed (0.003 inch), and this indicates that the Trigger Spacer made from PPS will function properly under 1,000 lbs. compressive load.

3. Stress of Pivot Holes Due to 30,000-RPM Spin

The Trigger Spacer shall not deform at the pivot holes of the trigger shafts under 30,000-RPM spin. The compressive loads will be applied to the pivot holes due to the tangential and normal forces of the trigger shaft assemblies when the projectile is fired.

The compressive stress in the Release Lever Pivot hole, which is the most severely stressed area, has been calculated as shown below. Note that the pivot holes of the Firing Arm Shaft and Safety Plate Shaft (see Section IV-A-5-h) are in the plates.

$$\sigma = F/A = (Fn + Ft)/A$$

where σ = stress in psi

F = combined force in lbs.

A = bearing area in inch^2

Fn = normal force due to 30,000-RPM spin in lbs.

Ft = tangential force due to angular acceleration in lbs.

The normal force (Fn) and the tangential force (Ft) can be found as follows:

$$Fn = mr\omega^2$$

$$Ft = mr\alpha$$

where m = the mass of the Release Assembly (0.0000497 slug)

r = the center distance between the pivot hole and the spin axis (0.039 ft.)

ω = the angular velocity (3141.6 rad/sec) - See Appendix H

α = the angular acceleration (1,545,600 rad/sec^2) - See Appendix H

Assuming the worst case, which is the maximum angular acceleration occurs at the maximum angular velocity, the normal and tangential forces can be added as shown below.

$$\vec{F} = \vec{Fn} + \vec{Ft}$$

$$F = \sqrt{Fn^2 + Ft^2}$$

$$= \sqrt{(mr\omega^2)^2 + (mr\alpha)^2}$$

$$= mr\sqrt{\omega^4 + \alpha^2}$$

Now, substituting the values into the equation,

$$F = mr \sqrt{\omega^4 + \alpha^2}$$

$$= (0.0000497)(0.039) \quad (3141.6)^4 + (1545600)^2$$

$$= 19.4 \text{ lbs.}$$

The bearing area (A) can be found from the minimum shaft pivot diameter (D) and the minimum contact length (L).

$$A = D \cdot L = D(\text{min. pivot length} - \text{max. chamfer length} - \text{max. endshake})$$

$$= (0.060)(0.087 - 0.042 - 0.009)$$

$$= 0.00216 \text{ inch}^2$$

Then, the maximum expected stress at the Release Shaft Pivot hole can be found.

$$\sigma = F/A$$

$$= 19.4/0.00216$$

$$= 9{,}000 \text{ psi}$$

The calculated stress (σ = 9,000 psi) is about 43% of the material strength, and the margin of operability is about 2.3. Note that the actual stress will be less than the calculated stress because the force (F = 19.4 lbs.) will act at the bottom pivot hole as well as at the top pivot hole.

4. Stress of Spacer Pins Due to the Angular Acceleration

The top and bottom Locating Spacer Pins will be loaded in shear due to the angular acceleration when the projectile is fired. Some of the shear load may be borne by the screws holding the Trigger Assembly to the sleeve; however, it is assumed that the entire shear load is on the Spacer pins. Only the stress of the top Locating Spacer Pin was calculated because the higher load will be applied to the top Locating Spacer Pin.

$T = FK$

where T = the torque in inch · lbs.

F = the force created due to the angular acceleration of the Trigger and SSD Assembly in lbs., and

$F = m(K/12)\,\alpha$

K = the radius of gyration in inches, and

$K = \sqrt{(\int r^2 dm)/(\int dm)}$

m = the mass of the combined Trigger and SSD Assembly in slugs

α = the angular acceleration in rad/sec^2, and

$\alpha = 1{,}545{,}600$ (See Appendix H)

r = the distance of dm from the center of rotation in inches

dm = the infinitesimal mass in the combined Trigger and SSD Assembly in slugs

and

$T = Rd$

where R = the reaction force of the top Spacer Pins in lbs.

d = the distance of the top Spacer Pins from the center of rotation in inches, and d = 0.725 (See Appendix B)

Now, the reaction force (R) can be found as follows:

$$T = FK = Rd$$

or

$$R = (FK)/d$$
$$= ((m(K/12)\alpha)K)/d$$
$$= (mK^2\alpha)/(12d)$$

From Appendix G,

mass of PPS Trigger Assembly = 0.0034938 slugs

mass of SSD Assembly = 0.0023168 slugs

and the total mass (m) = 0.0034938 + 0.0023168

= 0.0058106 slugs

Substituting values,

$$R = (mK^2\alpha)/(12d)$$
$$= ((0.0058106)K^2(1{,}545{,}600))/((12)(0.725))$$
$$= 1032K^2$$

Now, the value of the radius of gyration should be found from the equation $K = \sqrt{(\int r^2 dm)/(\int dm)}$ to find the reaction (R).

The approximate value of K has been calculated, because of the complexity, as shown below.

$$K = \sqrt{(\int r^2 dm)/(\int dm)}$$

assuming uniform thickness (t) and density (ρ)

$$= \sqrt{(\int r^2 \rho t dA)/(\int \rho t dA)}$$

$$= \sqrt{(\rho t \int r^2 dA)/(\rho t \int dA)}$$

$$= \sqrt{(\int r^2 dA)/(\int dA)}$$

$$= \sqrt{J/A}$$

where J = the polar moment of inertia of the combined Trigger and SSD Assembly in inch^4

A = the cross-sectional area of the combined Trigger and SSD Assembly in inch^2

Assuming the Trigger and SSD Assembly is a cylinder,

$$J \simeq \pi D^4/32$$

$$A \simeq \pi D^2/4$$

where D = the diameter of the Trigger Spacer in inches, and D = 1.645 (See Appendix B).

Then

$$K \simeq \sqrt{J/A}$$

$$\simeq \sqrt{(\pi D^4/32)/(\pi D^2/4)} = \sqrt{D^2/8}$$

$$= \sqrt{(1.645)^2/8}$$

$$= 0.582 \text{ inch}$$

Now, the reaction force (R) can be found with this approximate value of radius of gyration ($K = 0.582$).

$$R = 1032K^2$$
$$= (1032)(0.582)^2$$
$$= 350 \text{ lbs.}$$

And the stress of the top Spacer Pin can be calculated as shown below.

$$\sigma = R/(2Ap)$$

where σ = the shear stress of the top Spacer Pin in psi

R = the reaction force in lbs., and $R = 350$

Ap = the cross-sectional area of the Spacer Pin in inch^2, and $Ap = \pi\ Dp^2/4 = \pi\ (0.0932)^2/4$

$= 0.0068$ (See Appendix D).

Substituting the values,

$$\sigma = R/(2Ap)$$
$$= 350/((2)(0.0068))$$
$$= 26{,}000 \text{ psi}$$

This calculated stress ($\sigma = 26{,}000$ psi) is about 39% of the minimum material strength specified on the Spacer Pin drawing - Appendix D - based on the heat treating of 313 Knoop Hardness ($\approx$ 120,000 psi tensile $\approx$ 66,000 psi shear strength). The margin of operability is about 2.6.

5. Creep

It is known that plastics creep under a continuous sustained load. There are three places in the Trigger Spacer where a continuous sustained load is applied. They are the Safety Plate top pivot hole, Firing Pin side wall, and Torsion Spring arm slot.

a. Force in the Safety Plate Top Pivot Hole

The top pivot hole of the Safety Plate Shaft is loaded in compression due to the load exerted from the Firing Pin Spring.

From the "spring data - 9.0 lbs. max. at 0.36 inch and the free length (0.64 inch)" shown in Drawing #9236624 and from the calculated minimum spring length (0.290 inch) when compressed, the maximum spring load can be found as shown below.

$$K = P/\delta$$

where K = spring constant in lbs./inch

P = spring load in lbs.

δ = deformation in inches

Then,

$$K = P/\delta = P_1/(L - L_1) = Pm/(L - Lm)$$

where P_1 = 9 lbs.

L = 0.64 inch

L_1 = 0.36 inch

Pm = maximum spring load

Lm = 0.29 inch

Substituting the corresponding values, the maximum spring load (Pm) can be found as follows:

$$P_1/(L - L_1) = Pm/(L - Lm)$$

or

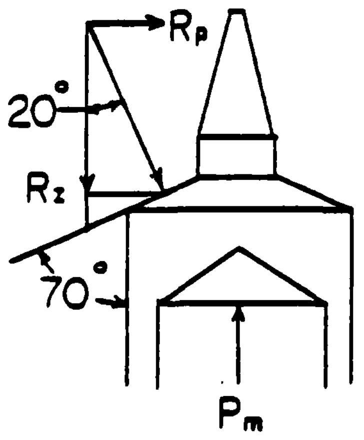
$$Pm = P_1\ (L - Lm)/(L - L_1)$$

$$= 9(0.64 - 0.29)/(0.64 - 0.36)$$

$$= 11.25 \text{ lbs.}$$

Now the compressive load at the top pivot hole can be found as follows.

From the force diagram shown below, the force, Rp, which induces the compressive load at the top pivot hole can be found.



$$Rz = Pm = 11.25 \text{ lbs.}$$

$$Rp = Rz \tan 20$$

$$= Pm \tan 20$$

$$= (11.25)(0.364)$$

$$= 4.1 \text{ lbs.}$$

The force, Rp, is applied to the Safety Plate Assembly, causing reactions at the top and bottom pivot holes and at the point where the Safety Plate contacts the Firing Arm Shaft. The force, Rz, is applied to the Safety Plate Assembly, causing reactions at the top and bottom pivot holes and at the shoulder of the Safety Plate Shaft (see Figures 1, 2 and 3).

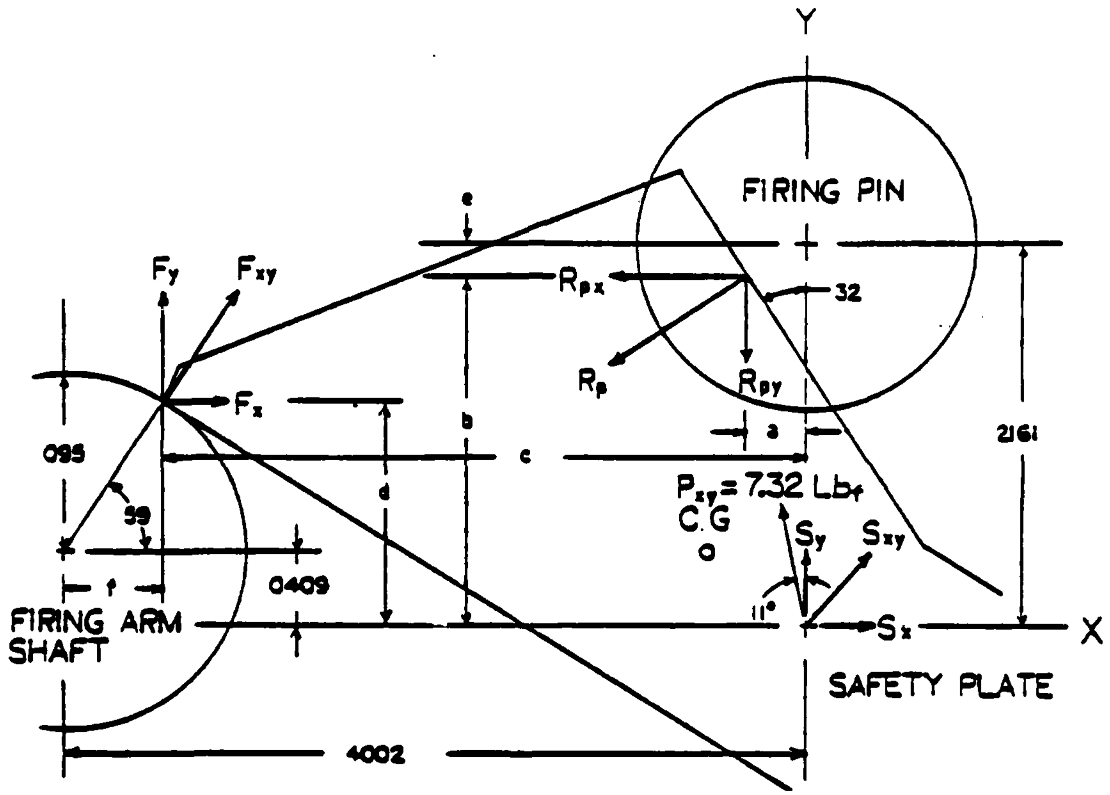


FIGURE 1. Force Analysis Diagram (Due to Rp)

Where a = 0.031, measured

$b = 0.2161 - e = 0.2161 - a(\tan 32) = 0.2161 - 0.031(\tan 32)$

$= 0.1967$

$c = 0.4002 - f = 0.4002 - 0.095(\cos 59)$

$= 0.3513$

$d = 0.0409 + 0.095(\sin 59)$

$= 0.1223$

NOTE: Two angles (32° and 59°) and the dimension "a" were measured from a 12x layout drawing.

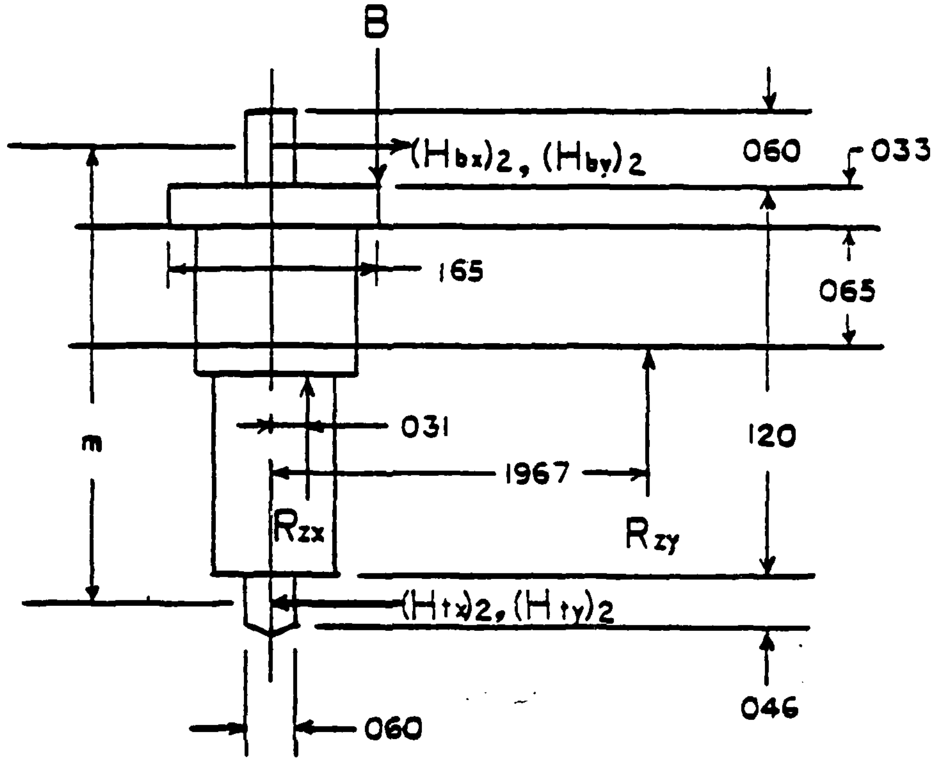


Figure 2. Force Analysis Diagram (Due to Rz)

$$m = 0.120 + (0.060 + 0.046)/2 = 0.173$$

Where m = 0.120 + (0.060 + 0.046)/2 = 0.173

NOTE: Rzx = Rzy = Rz, and B = Rz

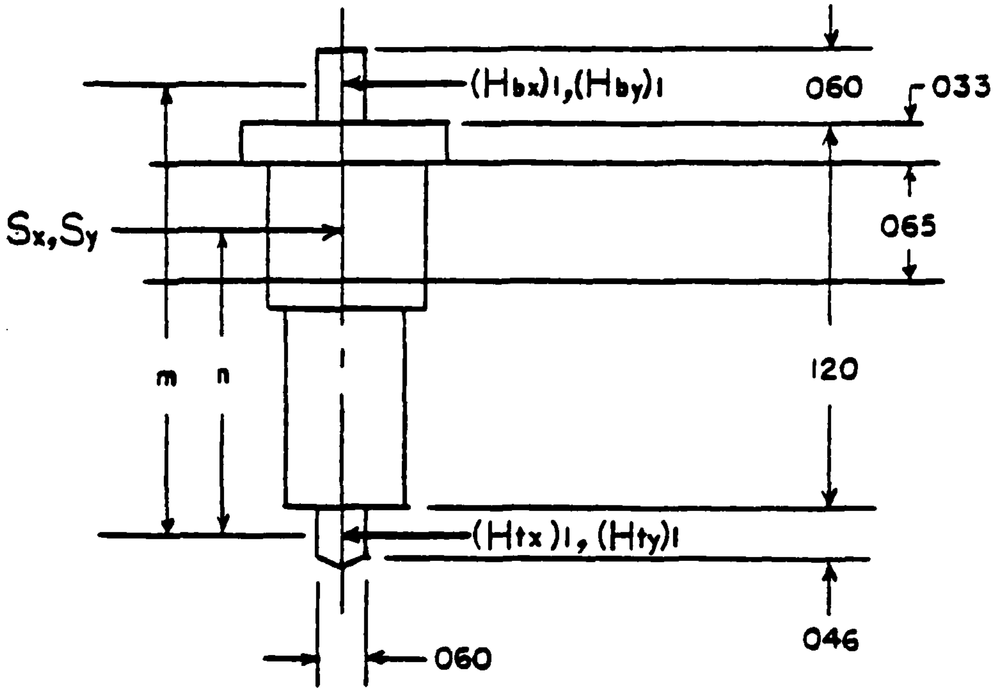


Figure 3. Force Analysis Diagram (Due to Sx and Sy)

Where $m = 0.120 + (0.060 + 0.046)/2$

$= 0.173$

$n = m - 0.033 - (0.060 + 0.065)/2$

$= 0.0775$

From the force diagram shown in Figure 1, the reaction forces of the Safety Plate Shaft due to Rp can be found.

$$\overset{+}{\underset{\rightarrow}{\Sigma}}\ Fx = 0 = -Rpx + Fx + Sx$$

$$= -Rp(\cos 32) + Fx + Sx$$

$$= -4.1(\cos 32) + Fx + Sx$$

$$= -3.48 + Fx + Sx$$

$$\overset{\uparrow +}{\Sigma}\ Fy = 0 = -Rpy + Fy + Sy$$

$$= -Rp(\sin 32) + Fy + Sy$$

$$= -4.1(\sin 32) + Fy + Sy$$

$$= -2.17 + Fy + Sy$$

$$\overset{+}{\Sigma}\ Ms = 0 = -Rpx(b) - Rpy(a) + Fx(d) + Fy(c)$$

$$= -3.48(0.1967) - 2.17(0.031) + Fx(0.1223)$$

$$+ Fy(0.3513)$$

$$= -0.685 - 0.067 + 0.1223Fx + 0.3513Fy$$

From the geometry of Fx and Fy,

$$Fx/Fy = \tan 59 = 1.6643$$

or

$$Fy = 1.6643Fx$$

Substituting 1.6643Fx for Fy in the moment equation,

$$\overset{+}{\Sigma}\ Ms = 0 = -0.685 + 0.067 + 0.1223Fx + 0.3513Fy$$

$$= -0.752 + 0.1223Fx + 0.3513(1.6643Fx)$$

$$= -0.752 + 0.1223Fx + 0.5847Fx$$

$$= -0.752 + 0.707Fx$$

Therefore,

$$F_x = 0.752/0.707$$
$$= 1.06 \text{ lbs.}$$

$$F_y = 1.6643F_x = 1.6643(1.06)$$
$$= 1.76 \text{ lbs.}$$

$$S_x = 3.48 - F_x = 3.48 - 1.06$$
$$= 2.42 \text{ lbs.}$$

$$S_y = 2.17 - F_y = 2.17 - 1.76$$
$$= 0.41 \text{ lbs.}$$

The forces Sx and Sy are applied to the Safety Plate Shaft as shown in Figure 3, and the reactions at the top pivot hole, due to the forces Sx and Sy, can be found as follows:

From the diagram shown in Figure 3,

$$\overset{+}{\Sigma} M_{bx} = 0 = -S_x(m - n) + (H_{tx})_1(m)$$
$$= -2.42(0.173 - 0.0775) + (H_{tx})_1(0.173)$$
$$= -0.231 + 0.173(H_{tx})_1$$

or $(H_{tx})_1 = 1.34$ lbs.

$$\overset{+}{\Sigma} M_{by} = 0 = -S_y(m - n) + (H_{ty})_1(m)$$
$$= -0.41(0.173 - 0.0775) + (H_{ty})_1(0.173)$$
$$= -0.039 + 0.173(H_{ty})_1$$

or $(H_{ty})_1 = 0.23$ lbs.

Therefore, 1.34 lbs. (in X-axis) and 0.23 lbs. (in Y-axis) are acting at the top pivot hole with the minus (-) direction due to the reaction force Rp.

Now, the force acting at the top pivot hole, which is created due to the reaction force Rz, can be found as shown below.

As shown in Figure 2, since the moment arm of Rzy (0.31) is smaller than the moment arm of B (0.165/2 = 0.0825), no force is expected to be applied to the top pivot hole in X-axis due to the reaction force Rz.

Then, $(Hty)_2 = 0$ lbs.

$$\overset{+}{\curvearrowright}\ \Sigma\ Mby = 0 = -Rzy(0.1967) + B(0.0825) + (Hty)_2(m)$$

$$= -11.25(0.1967) + 11.25(0.0825) + (Hty)_2(0.173)$$

$$= -1.285 + (0.173)(Hty)_2$$

or $(Hty)_2 = 7.43$ lbs.

Therefore, only the 7.43 lbs. is acting at the top pivot hole in Y-axis with the plus (+) direction due to the reaction force Rz.

To find the total force acting at the top pivot hole, the force components acting at the top pivot hole due to Rp and Rz have been added as shown below.

$$\overset{+}{\rightarrow}\ \Sigma\ Px = (Htx)_1 + (Htx)_2$$

$$= -1.34 + 0$$

$$= -1.34 \text{ lbs.}$$

$$\overset{\uparrow+}{\Sigma} Py = (Hty)_1 + (Hty)_2$$

$$= -0.23 + 7.43$$

$$= 7.20$$

$$Pxy = \sqrt{Px^2 + Py^2}$$

$$= \sqrt{(-1.34)^2 + (7.20)^2}$$

$$= 7.32 \text{ lbs.}$$

$$\theta = \text{arc tan } (Py/Px)$$

$$= \text{arc tan } (7.32/-1.34)$$

$$= -79^\circ \text{ or } 101^\circ$$

Therefore, the 7.32 lbs. is acting at the top pivot hole with 101° angle (or 11° from Y-axis) in compression as shown in Figure 1.

b. Stress in the Top Pivot Hole

The compressive stress in the top pivot hole of the Safety Plate Shaft can be found from the force Pxy (7.32 lbs.) and the bearing area in the hole.

$$\sigma = P/A$$

where σ = compressive stress in the top pivot hole in psi

P = Pxy = 7.20 lbs.

A = bearing area

= (min. shaft pivot dia.)(min. contact lgth.)

= (0.0597)(0.046)

= 0.0027 inch^2

Substituting the values into the equation,

$$\sigma = P/A$$
$$= 7.20/0.0027$$
$$= 2{,}700 \text{ psi}$$

c. Force in the Torsion Spring Arm Slot

The Torsion Spring arm slot in the Trigger Spacer is loaded in compression due to the torque exerted from the Torsion Spring.

From the "spring data - 0.15 inch · pound max. torque at the final position when unloa^d^ing" and from the spring arm length shown in Drawing 9236610, the force applied to the slot can be found as shown below.

$$F = T/\ell$$

where F = force applied to the slot in lbs.

T = 0.15 inch pounds

ℓ = min. Torsion Spring arm length

$= 0.14 + (0.42 - 0.14)/2$

$= 0.28$ inch

Substituting the values into the equation,

$$F = T/\ell$$
$$= 0.15/0.28$$
$$= 0.54 \text{ lbs.}$$

d. Stress in the Torsion Spring Arm Slot

The compressive stress in the Torsion Spring arm slot can be found from the force (0.54 lbs.) and from the bearing area in the slot.

$$\sigma = F/A$$

where $F = 0.54$ lbs.

A = bearing area in the slot

= (spring diameter)(contact length)

= (0.019)(0.44 - 0.16)

= 0.0053 inch^2

Substituting the values into the equation,

$$\sigma = F/A$$

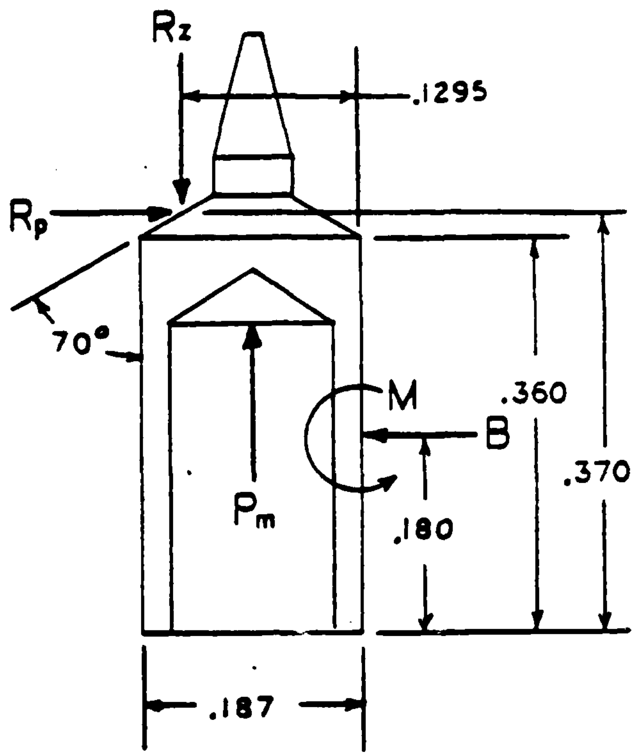
$$= 0.54/0.0053$$

$$= 102 \text{ psi}$$

e. Force on the Side Wall of the Firing Pin Hole

The side wall of the Firing Pin hole is loaded in compression due to the load exerted from the Firing Pin Spring and the reaction of the Safety Plate.

The force diagram has been constructed to find the magnitude and the mode of the reaction (B) at the side wall as shown on the following page.



From the force diagram shown above,

$$\overset{+}{\vec{\Sigma}}\ Fy = 0 = Pm - Rz \quad \text{and} \quad \overset{+}{\vec{\Sigma}}\ Fx = 0 = Rp - B$$

or Pm = Rz = 11.25 (see Section IV-5)

Rp = B = 4.1 (see Section IV-5)

$$\overset{\curvearrowright +}{\Sigma}\ M_B = 0 = -M + Pm\ (0.187/2) - Rz\ (0.1295) + Rp\ (0.370 - 0.180)$$

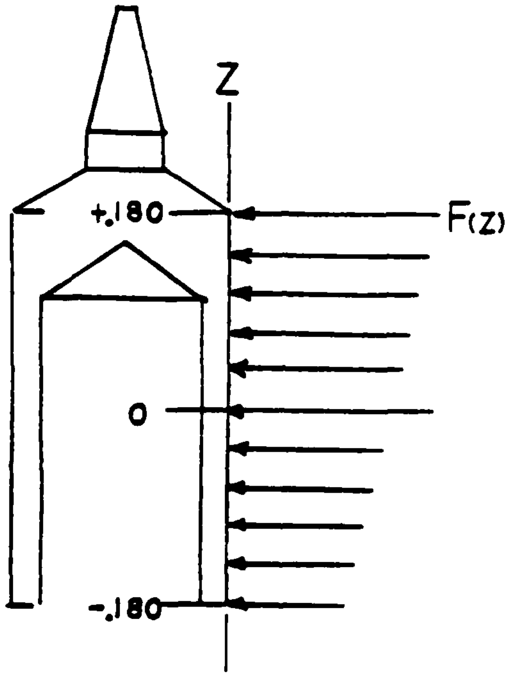
$$= -M + (11.25)(0.187/2) - (11.25)(0.1295) + 4.1(0.370 - 0.180)$$

$$= -M + 0.374$$

or

M = 0.374 inch · pounds

Since there is a moment ($M = 0.374$ inch · lbs.) with the direction as shown in the above diagram, the load distribution along the side of the Firing Pin varies, and the maximum load occurs at the point end of the Firing Pin as shown in the diagram below.



Now, the maximum force (F(z) max.) can be found as follows:

Since the force F(z) is the dependent of Z as shown in the above diagram,

$F(z) = aZ + b$

where "a" and "b" are constants, and "a" is the slope and "b" is the force at Z = 0.

and

$$\overset{\curvearrowleft +}{\Sigma} Mo = \int_{-0.180}^{+0.180} F(z) \cdot Z \cdot dz = M$$

$$\overset{\overset{+}{\leftarrow}}{\Sigma} F = \int_{-0.180}^{+0.180} F(z) \cdot dZ = B$$

where $F(z)$ = the force per unit length in lbs./inch.

Z = the position along the side of the Firing Pin in inches

$M = 0.374$ inch $\cdot$ lbs.

$B = 4.1$ lbs.

Substituting "$aZ + b$" for $F(z)$ into the moment and the force equations,

$$\overset{\curvearrowleft +}{\Sigma} Mo = \int_{-0.180}^{+0.180} F(z) \cdot Z \cdot dz$$

$$= \int (aZ + b)Z \cdot dz$$

$$= \int (aZ^2 + bZ)dz$$

$$= a \int Z^2 dz + b \int Zdz$$

$$= (a/3)Z^3 + (b/2)Z^2$$

$$= (a/3)((0.180)^3 - (-0.180)^3) + (b/2)((0.180)^2 - (0.180)^2)$$

$= (a/3)(0.011664)$

$= 0.003888a$

$= 0.374$

Therefore, $a = (0.374)/(0.003888)$

$= 96.2$

$$\overset{+}{\underset{\Sigma}{\uparrow}} F = \int_{-0.180}^{+0.180} F(z)dz$$

$= \int (aZ + b)dz$

$= a \int Zdz + b \int dz$

$= (a/2)Z^2 + bZ$

$= (a/2)((0.180)^2 - (-0.180)^2) + b((0.180) - (-0.180))$

$= b(0.360)$

$= 4.1$

Therefore, $b = (4.1)/(0.360)$

$= 11.4$

Now, the maximum force occurs at $z = 0.180$,

$F(z)$ max. $= F(z = 0.180) = aZ + b$

$= (96.2)(0.180) + 11.4$

$= 28.72$ lbs./inch

f. Stress on the Side Wall of the Firing Pin Hole

The maximum force ($F(z)$ max. = 28.72 lbs./inch) will be applied to the side wall of the Firing Pin hole at the point where the pin-point end of the Firing Pin contacts the side wall. Then, the maximum bearing stress at the side wall can be calculated based on this maximum force and the diameter of the Firing Pin as follows:

$$\sigma \text{ max.} = F(z) \text{ max.}/D$$

where σ max. = the maximum bearing stress in psi

F(z) max. = 28.72 lbs./inch

D = the minimum Firing Pin diameter in inches

and D = 0.187 (Dwg. #9236623)

Substituting the values,

$$\sigma \text{ max.} = F(z) \text{ max.}/D$$

$$= 28.72/0.187$$

$$= 154 \text{ psi}$$

g. Creep Deformation

Since data concerning the creep deformation of PPS is not available, the creep deformation of another high strength plastic, Acetal (40% glass reinforced) was calculated, based on the study results published in the "Plastics Engineering" magazine shown as follows.[1]

1. J. R. Louglin, "A New Creep Law for Plastics," Plastics Engineering, Feb. 1968, P. 97-103.

The article "A New Creep Law for Plastics" in the magazine shows that the flexural creep of Acetal (40% glass reinforced, tested with 4660 psi initial stress at 25°C) has the following creep characteristics:

$$\text{Log}(6.1075 - \text{Log } E) = -1.13812 + 0.196236 \text{ Log } t$$

where t = time in hours

E = pseudo - modulus of elasticity at time "t" in psi

By substituting 175,200 hrs. (20 yrs.) for time "t" in the above equation, the pseudo-modulus of elasticity after 20 years can be found as shown below.

$$\text{Log }(6.1075 - \text{Log } E) = -1.13812 + 0.196236 \text{ Log } 175200$$

$$\text{Log }(6.1075 - \text{Log } E) = -0.109149842$$

or

$$\text{Log } E = 6.1075 - 10^{-0.109149842}$$

$$= 5.329732$$

Therefore,

$$E = 10^{5.329732}$$

$$= 214{,}000 \text{ psi}$$

Then, the creep deformation of the Safety Plate top pivot hole, Firing Pin sidewall, and Torsion Spring arm slot can be found from this pseudo-modulus of elasticity

($E = 214{,}000$ psi) and the calculated stresses - 2,700 psi (for the top pivot hole), 102 psi (for the Torsion Spring arm slot), and 154 psi (for the sidewall) as shown below.

$$E = \sigma/\varepsilon = \sigma/(\delta/\ell) = (\sigma \cdot \ell)/\delta$$

or

$$\delta = (\sigma \cdot \ell)/E$$

where δ = creep deformation for 20 years in inches

σ = stress at the area of concern in psi

ℓ = effective length in inches

E = 214,000 psi

To find the creep deformation, the effective length (ℓ) should be further obtained. However, the effective length (ℓ), which will deform under the creep loading, is not readily obtainable because of the dimensional complexity. Therefore, the approximate effective lengths (0.133 inch for the Top Pivot hole, 0.070 inch for the Spring Arm Slot, and 0.8225 inch for the side wall of the Firing Pin hole) have been used for the following creep calculations.

Creep Deformation at the Top Pivot Hole of the Safety Plate Shaft

$$\delta = (\sigma \cdot \ell)/E$$
$$= (2{,}700)(0.133)/214{,}000$$
$$= 0.0017 \text{ inch}$$

Creep Deformation at the Arm Slot of the Torsion Spring

$$\delta = (\sigma \cdot \ell)/E$$
$$= (102)(0.070)/214{,}000$$
$$= 0.00003 \text{ inch}$$

Creep Deformation at the Side Wall of the Firing Pin Hole

$$\delta = (\sigma \cdot \ell)/E$$
$$= (154)(0.8225)/214{,}000$$
$$= 0.0006 \text{ inch}$$

h. Discussions of Creep Deformation

As noted, the above creep deformations were calculated based on the creep characteristics of Acetal. Although the creep characteristics of PPS may be somewhat different from Acetal, the calculated creep deformations were used as a guideline for the PPS Trigger Spacer design modification.

Since the creep deformations at the arm slot of the Torsion Spring and the side wall of the Firing Pin hole are negligibly small (0.00003 and 0.0006 inch) and the positions are not functionally critical, the creep deformations have been accepted as tolerable.

Since the creep deformation at the Top Pivot hole (0.0017 inch) is, however, larger than the pivot hole tolerance (0.001 inch) and the pivot hole is functionally critical, the design change to avoid creep has been made. By providing the Top Pivot hole at the Top Plate instead of in the Trigger Spacer, the creep problem in the Top Pivot hole has been avoided.

Because of this design change, the pivot hole in the Trigger Spacer has been changed to a shaft clearance hole, and the pivot hole has been added to the Top Plate. Also, the Safety Plate Shaft length has been increased.

The new drawings and change notes are shown in Appendixes A-1, A-3, A-4, B, C-4, E and F.

6. Aging

Since PPS (40% glass reinforced) has been in existence for less than ten (10) years, there is no actual data available concerning the natural aging properties of PPS for twenty

(20) years. The natural aging properties were, therefore, estimated from the thermal aging properties of PPS.

The thermal aging properties of PPS (40% glass reinforced) were studied by Phillips Petroleum Company, and the results of the studies are contained in "Report on Component - Plastics."[1] The report shows that the PPS is expected to retain at least one-half of its original tensile strength, with the following temperature and time relation:

$$\text{Log } Y = 5320.6861\ X - 5.4801723$$

where the test sample thickness = 0.060 inch

Y = time in hours

X = $1/^{\circ}$Kelvin

Solving the above regression line equation for the points with which we are concerned, the following results can be obtained:

1. At 165°F, PPS takes 735,600 years to loose one-half of its original tensile strength.

2. For twenty (20) years, PPS retains one-half of its original strength at 430°F.

Extrapolating the 95% confidence lines, which are also shown in the same report, PPS takes 4,500 years to lose one-half

1. "Report on Component - Plastics," Philips Petroleum Co., File E54700, Project 73NK6336, December 11, 1973, P. 2 and P. I11-1B.

of its original tensile strength at 165°F. Note that the deviation gets larger as the extrapolating values get further away from the actual test points, and that is why the 95% confidence value (4,500 years) is significantly lower than the regression line value (735,600 years).

Although the aging values (4,500 years and 735,600 years) were not obtained from the test conducted at 165°F, the estimated values indicate that the decrease in strength of PPS by natural aging is not perceivable in twenty years.

7. Chemical Compatibility

The PPS is known to have a hydrolytic stability at elevated temperatures, and also known to be chemically compatible with a broad range of solvents, minerals, and organic acids and alkalies.

The Military Specification (MIL-P-46174) notes that "the glass-reinforced polyphenylene sulfide is intended for use in anti-friction gears, pistons, corrosion-resistant coating, electronic components and bearing materials." The specification also notes that "there are no known solvents for PPS below 400°F, and PPS has excellent chemical resistance and thermal stability."

Currently, there are six lubricants which have been approved for use in the Trigger Assembly. They are Dow Corning 55M

Grease (Part No. 9236521), mixture of Astro Oil and Molybdenum Disulfide (Part No. 9236485), Instrument Grease 812 (Part No. 9236557), Nyogel 843 (Part No. 9236606), Nyefilm 545 (Part No. 9236492) and Nye Astro Oil (Part No. 9236482).

PPS Trigger Spacers have been tested with satisfactory results in Astro Oil, which is the main component of all approved lubricants except Dow Corning 55M Grease.

C. Fabrication

The PPS Trigger Spacer Assemblies were made complete from the injection molding process, with the Spacer Pins inserted during molding. No secondary machining operation or Spacer Pins assembly process was needed.

However, the redesign of the Spacer Pins has been made to contain grooves, so that the insert molded pins can withstand the axial unseating load requirement (min. 25 lbs.). The new Spacer Pin drawing is shown in Appendix D.

During assembly, the PPS Trigger Spacers required no special steps. The PPS Trigger Spacers were assembled with the trigger parts according to the standard assembly procedure, and the staking and riveting operations were performed in the usual manner on the current M577 Fuze production line.

V. TESTING

A. Introduction

This section contains the descriptions and results of various tests performed on the proposed Trigger Assembly.

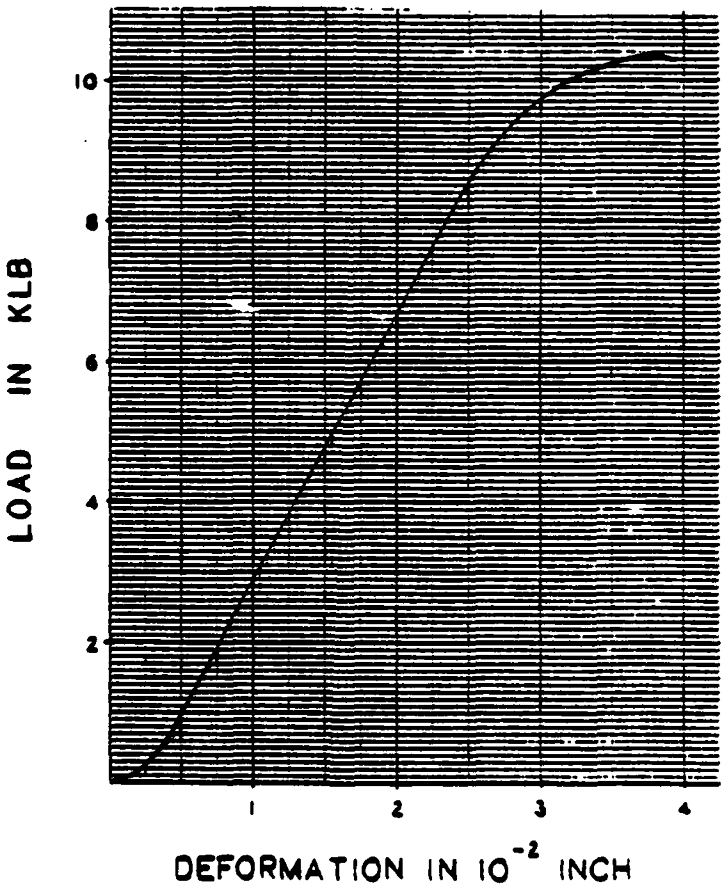
Section B (Static Test) is consisted of four subsections - Static Compression Test, Aging Test, Chemical Compatibility Test, and Temperature and Humidity Test. The Functional, Spin, and Air-gun tests are shown in Section C, D, and E accordingly. The Ballistic Test is shown in Section F, and Section F also contains the Jolt and Jumble, Rough Handling, and Transportation Vibration tests.

The height of the proposed Trigger Spacer was modified after the Static Compression Test (Section V-B-1) to provide the required end shake between the trigger shafts and the two plates. The detailed description of this modification is shown in Appendix C-3.

B. Static Test

1. Strength and Rigidity

A static compression test was performed on the PPS Trigger Assembly in a Tinius Olsen Electmatic Universal Testing Machine with a recorder and deflectometer. The load was applied to the Trigger Assembly with the deflection rate of less than 0.100 inch per minute at the ambient temperature. The "load vs deformation" curve of the PPS Trigger Assembly was obtained from the test, and a typical "load vs deformation" curve is shown below.



The load vs deformation curve shows that the ultimate compressive strength of the assembled PPS Trigger Spacer is over 10,000 lbs. As shown in Section III-A-1, the force induced from 30,000 g acceleration is 4,512 lbs., which is about 45% of the ultimate compressive strength of the assembled PPS Trigger Spacer. This agrees with the theoretical study result, which showed that the expected stress due to 30,000 g acceleration is about 40% of the material strength.

The load vs deformation curve also shows that the assembled PPS Trigger Spacer deforms about 0.005 inches under 1,000 lbs. compressive load. This test result does not agree with the theoretical study result, which showed that the PPS Trigger Assembly is expected to deform less than 0.001" under 1,000 lbs. compressive load.

The difference between the test result (0.005 inches deformation) and the study result (0.001 inches deformation) may have been caused by the non-uniform deformation of the PPS spacer because of its uneven cross-sectional area.

Because of this excessive deformation, a design modification was made to provide the minimum of 0.005 inches end shake between the Firing Arm Shaft and the two trigger plates. (See Appendix C-3.)

2. Aging

a. Natural Aging

Two test samples made from PPS - 40% glass reinforced material - (which was molded at least before 1975 with the published material compressive strength of 21,000 psi) were tested in compression, and the results showed an average compressive strength of 23,700 psi. We have also received a compression test result showing an average strength of 20,200 psi for the same material from Phillips Chemical Company, who performed independent testing recently.

The difference in the two test results may have been caused by the difference in sample preparation method. Both samples were made from a slab which contained a textured surface, which had to be machined to avoid stress concentrations. At Hamilton, only the textured surface was machined and the other surface was not machined to maintain the molded skin. At Phillips, however, both surfaces were machined. It is, therefore, understandable that Phillips' test result was lower than Hamilton's test result.

As expected, comparing the two test results (23,700 psi and 20,200 psi) with the published compressive strength (21,000 psi), the decrease in compressive strength of PPS material by the natural aging for seven years is not evident.

b. Thermal Aging

Two PPS Spacers were submerged into water in a closed jar and kept for 170 hours at 165°F. The weight and the major dimensions specified in MIL-F-58983 (Rev. A) were measured before and after the test. The test results showed no discernable changes. The test results are shown below.

Weight in Lbs.

Sample No.	Sample 1			Sample 2		
	Before	After	Change	Before	After	Change
Weight	0.0418	0.0420	+0.48%	0.0418	0.0420	+0.48%

Height in Inches

Sample No.	Sample 1			Sample 2		
	Before	After	Change	Before	After	Change
Height	0.537	0.537	None	0.537	0.537	None

True Position in Inches: measured center-to-center distance between Datum -D- and the hole

Sample No.	Sample 1			Sample 2		
	Before	After	Change	Before	After	Change
Safety Hole	0.716	0.716	None	0.717	0.717	None
Release Hole	0.959	0.959	None	0.959	0.959	None

Diameter in Inches

Sample No.	Sample 1			Sample 2		
	Before	After	Change	Before	After	Change
Safety Hole	0.092	0.092	None	0.092	0.092	None
Release Hole	0.063	0.063	None	0.063	0.063	None

3. Chemical Compatibility

A limited amount of chemical compatibility testing was performed on the PPS Trigger Spacers. The PPS Spacers were tested in the Astro Oil, which is the main component of all approved lubricants except Dow Corning 55M Grease. After the compatibility test, the strength, rigidity, weight, dimensional variation, and the appearance of the tested PPS Spacers were checked.

The test results showed that no discernable changes in the PPS Spacers occurred during testing. The test results are shown below.

a. Strength and Rigidity

Two PPS Spacers were hung over the Astro Oil in a closed jar and kept for 340 hours at 165°F. One of the samples tested was then submerged into the Astro Oil and kept for an additional 340 hours at 165°F.

The appearance, strength, and the rigidity of the tested samples were checked.

Sample No.	Before Testing	1	2
Test Time, Hrs.		340	680
Strength, Lbs.	10,350	10,100	10,100
Rigidity, Lbs/Inch	2.5×10^5	2.5×10^5	2.2×10^5
Appearance	Black	No Change	No Change

b. **Weight and Height**

Two PPS Spacers were submerged into the Astro Oil in a closed jar and kept for 170 hours at 165°F. The weight and the height of the PPS Spacers were checked before and after the test.

Sample No.	Sample 1			Sample 2		
	Before	After	Change	Before	After	Change
Spacer Wt. in Lbs.	0.418	0.419	+0.24%	0.418	0.419	+0.24%
Spacer Ht. in Inches	0.537	0.536	-0.001	0.537	0.536	-0.001

c. **True Position and Diameter**

The true position and the diameter of the Release Lever and Safety Plate Shaft holes were checked using the same samples tested in the previous section - Weight and Height.

The true position was measured by the center-to-center distance between the shaft hole and the Datum -D- of the Drawing 9236605.

True Position in Inches	Sample 1			Sample 2		
	Before	After	Change	Before	After	Change
Safety Hole	0.718	0.717	-0.001	0.717	0.717	None
Release Hole	0.960	0.959	-0.001	0.960	0.959	-0.001

Diameter in Inches	Sample 1			Sample 2		
	Before	After	Change	Before	After	Change
Safety Hole	0.092	0.092	None	0.092	0.092	None
Release Hole	0.063	0.063	None	0.063	0.063	None

4. Temperature and Humidity Test

Five PPS Trigger Spacers were subjected to the Temperature and Humidity Test per MIL-STD-331 (Test 105.1). The weight and height of the spacers and the diameter and true position of the Release Shaft pivot holes were measured before and after the test. The test results showed no weight change and a small dimensional change (about 0.001"), which is considered negligible. The test results are shown below.

Weight in Lbs.

Sample	1	2	3	4	5
Before	0.0416	0.0417	0.0419	0.0416	0.0414
After	0.0416	0.0417	0.0419	0.0416	0.0414
Change	None	None	None	None	None

Height in Inches

Sample	1	2	3	4	5
Before	0.537	0.537	0.536	0.536	0.536
After	0.538	0.538	0.538	0.538	0.537
Change	+0.001	+0.001	+0.002	+0.002	+0.001

Release Hole Diameter in Inches

Sample	1	2	3	4	5
Before	0.063	0.063	0.063	0.063	0.063
After	0.062	0.062	0.062	0.062	0.062
Change	-0.001	-0.001	-0.001	-0.001	-0.001

Release Hole True Position in Inches: measured center-to-center distance between Datum -D- and the hole

Sample	1	2	3	4	5
Before	0.962	0.962	0.960	0.962	0.960
After	0.962	0.961	0.961	0.960	0.959
Change	None	-0.001	+0.001	-0.002	-0.001

5. Thermal Shock Test

Five fuzes with PPS Trigger Assemblies were subjected to the Thermal Shock Test per MIL-STD-331, Test 113.1. After the test, the fuzes were disassembled, and the Trigger Assemblies successfully passed the 1,000 RPM non-function, and 2,000 RPM function spin, loaded with 600 to 1,000 lbs., as per MIL-F-50893 specifications. In addition, the Trigger Assemblies passed the inspection of the Firing Arm Assembly Release of the Safe Separation Release and Firing Safety Plate Assemblies per Notes 7 & 8, Drawing Number 9236603.

The Trigger Assemblies were disassembled and the Trigger Spacers were visually inspected. The Spacers showed no evidence of cracking or material distortion as a result of the test.

C. Functional Test

The PPS Trigger Assemblies with the modifications shown in Appendix A were subjected to the tests specified in Notes 7 & 8 of Drawing No. 9236603 to check the Firing Arm Assembly's releasing and non-releasing performance. The PPS Trigger Assemblies met the requirements satisfactorily.

The PPS Trigger Assemblies also performed satisfactorily at the 1,000 RPM no-fire and 2,000 RPM with load must-fire spin tests before and after the assemblies were subjected to the 5,000 lbs. (33,200 g) static compression.

D. <u>30,000 RPM Spin Test</u>

Ten PPS Trigger Assemblies with the modifications shown in Appendix A (except changes in Appendix C-5) were subjected to the 30,000 RPM Spin Test, and the test results show that the PPS Trigger Spacers assembled with aluminum (2024 T4) Trigger Shafts are satisfactory in strength and rigidity for 30,000 RPM spin.

The test results are shown below.

<u>Spin Test</u>
for
<u>PPS Trigger Spacers Assembled with Aluminum Shafts</u>

Sample No.	Spin RPM	Time Sec.	Result	
			Function	Note
1	30,000	25	Fired properly	
2	"	20	at the RPM	
3	"	30	shown in the	
4	"	11	Spin column	Spun twice - timer was not set
5	"	19	"	
6	"	19	"	
7	27,000	8	"	RPM reduced - timer did not turn
8	30,000	20	"	
9	"	20	"	
10	"	20	"	Spun twice - timer did not turn

E. <u>Air-Gun Test</u>

Ten PPS Trigger Assemblies (with the same configuration as the samples tested for 30,000 RPM spin) were air-gun tested at 27,265 to 32,051 g's. Seven of nine units, including four units air-gun tested at greater than 30,00 g's, functioned satisfactorily during the 2,000 RPM spin test under a 600 to 1,000-lb. load. The sleeve was broken in the two units that would not function. The breaking of the Sleeve causes an additional load on the Trigger Assembly from the Timer Assembly. One unit could not be tested after the air-gun testing because the outside diameter of the Trigger was too large to fit into the test fixture.

Air-Gun Test
for
PPS Trigger Spacers Assembled with Aluminum Shafts

Sample No.	"g" Level	Sleeve	Trigger Assembly	
			1,000 RPM	2,000/Load
1	30059	Broken	No test	No test
2	30193	OK	OK	OK
3	29323	"	"	"
4	30127	Broken	"	DNF
5	31333	"	"	"
6	31254	OK	"	OK
7	28292	"	"	"
8	32051	Broken	"	"
9	27265	OK	"	"
10	30534	"	"	"

F. Ballistic Test

Two hundred and forty-four fuzes were built with the PPS Trigger Assemblies shown in Appendix A for ballistic and laboratory testings. The Ballistic Test was performed at Yuma Proving Grounds.

One hundred fuzes were subjected to a feasibility testing (Supplement 10 to TPR-2594) and one hundred and forty-four fuzes were subjected to the standard acceptance testing (Supplement 11 to TPR-2594).

1. Feasibility

Eighty-five fuzes were subjected to feasibility ballistic testing at Yuma Proving Grounds with satisfactory results. The overall reliability was 98.6%. The one dud, which was non-functional at ground impact, was not recovered.

Ballistic Results of PPS Trigger Assemblies for Feasibility

Gun	Zone	Temp °F	Time Sec.	Mean	Std. Dev.	Function
105mm, M103	7	145	3	3.179	0.048	20/20
105mm, XM204	8	70	45	45.161	0.095	15/15
* 105mm, XM204	8	70	45	45.164	0.204	15/15
8 inch, M2A1	1	70	3	3.078	0.043	14/15
155mm, M198	8	70	3	3.086	0.070	20/20
105mm, M103	7	70	Ship. Set.	-	-	0/15

1. Test Sample Lot Number: HAT 81 H 00E055
*2. Control Group Lot Number: HAT 81 H 00E068

2. Acceptance

The results of the acceptance testing showed that the proposed Trigger Assembly performed satisfactorily.

a. Jolt and Jumble

Examination of twelve units, subjected to Jolt and Jumble test per MIL-STD-331 (Test 101.2 and 102.1), showed that all the units were undamaged and therefore safe to handle.

b. Rough Handling

Ballistic Test was performed on thirty-two units which were subjected to the Sequential Rough Handling test per Figure 3 of MTP 4-2-602 (less Loose Cargo Test). The test results are shown below.

Gun	Zone	Temp °F	Time Sec.	Mean	Std. Dev.	Function
105mm	7	70	50	50.058	0.075	26/32

Notes: 1. Test sample Lot Number: HAT 81 H 00E067.

2. One unit could not be set properly because the Housing Retainer was disengaged after the Rough Handling Test.

3. X rays taken after the Rough Handling process showed the Setback Pin in the timer down in each of the five duds.

c. Transportation Vibration

Thirty-five units were subjected to the Transportation Vibration Test per MIL-STD-331 (Test 401). After the Transportation Vibration testing, ten units were examined and twenty-five units were subjected to the Ballistic Testing. The results are shown below.

Examination on 10 Units

Examination of ten units, subjected to Transportation Vibration Testing, showed that all the units were safe and operable.

Ballistic Testing

Gun	Zone	Temp. °F	Time Sec.	Mean	Std. Dev.	Function
155mm	8	70	50	50.020	0.144	10/10
105mm	7	70	Ship. Set	-	-	* 0/15

Notes: 1. Test Sample Lot No.: HAT 81 H 00E072

* 2. Non-function board test

d. Ballistic Testing

Ballistic Testing was performed on sixty-five units which were not subjected to any laboratory testings. The results are shown below.

Gun	Zone	Temp. °F	Time Sec.	Mean	Std. Dev.	Function	Note
155mm	8	70	75	74.919	0.248	9/10	1
105mm	7	145	50	50.120	0.090	10/10	
105mm	7	70	50	50.137	0.108	10/10	
8 inch	1	-35	25	24.935	0.033	8/10	2
8 inch	1	70	15	15.002	0.067	10/10	
155mm	1	70	Ship. Set.	-	-	15/15	P.D. function test

NOTES:
1. One dud was FGI.
2. Two duds were NFGI, and one unit which was recovered showed the timer did not function properly.
3. Test Sample Lot Number: HAT 81 H 00E072

VI. COST AND WEIGHT

A. Cost

By substituting PPS Trigger Spacer for the current aluminum Trigger Spacer, a cost saving of $0.8799 per fuze is expected. The detailed cost comparison is shown below.

Part No.	Current $/part	Proposed $/part	Savings $
9236604 (Trigger Spacer Assy.)	1.3945	*0.4951	0.8994
9236619, Aluminum 2024-T4 (Safety Plate Shaft)	0.0604	0.0799	-0.0195
Total Savings per Fuze			0.8799

NOTES:

1. The cost saving of $0.8799 per fuze does not include tools and gages, G&A, and profit.
2. The cost of the proposed design is based on the production quantity of 300,000 pcs. per year.
3. The Top Plate (9236608) and the Spacer Pin costs are omitted because there is no cost difference between the current and the proposed parts.
4. No cost which is common to the current and the proposed design is included in the cost comparison.

*5. The proposed Trigger Spacer Assembly (9236604) requires new tools and gages which will cost approximately $37,000.00

B. <u>Weight</u>

By substituting PPS Trigger Assembly for the current aluminum Trigger Assembly, a weight reduction of 0.0206 lbs. per fuze is expected.

	Current Lbs.	Proposed Lbs.	Weight Change
Spacer With 4 Pins	0.0625	0.0418	-0.0207
Safety Plate Assembly (2024 T-4 Shafts)	0.0015	0.0017	+0.0002
Top Plate	0.0328	0.0327	-0.0001
Total			-0.0206 Lbs.

APPENDIX A

CHANGES AND MODIFICATIONS MADE IN
TRIGGER ASSEMBLY (9236603)

The following changes and modifications have been made in Trigger Assembly (9236603) for the proposed PPS Trigger Spacer.

1. <u>Trigger Spacer</u>

The material has been changed and the dimensions have been changed as shown in Appendix B and C.

2. <u>Spacer Pin</u>

The Spacer Pins (9236487-1 and 9236487-2) have been redesigned to contain grooves, and the lengths have been changed. New Spacer Pin is shown in Appendix D.

3. <u>Safety Plate Shaft</u>

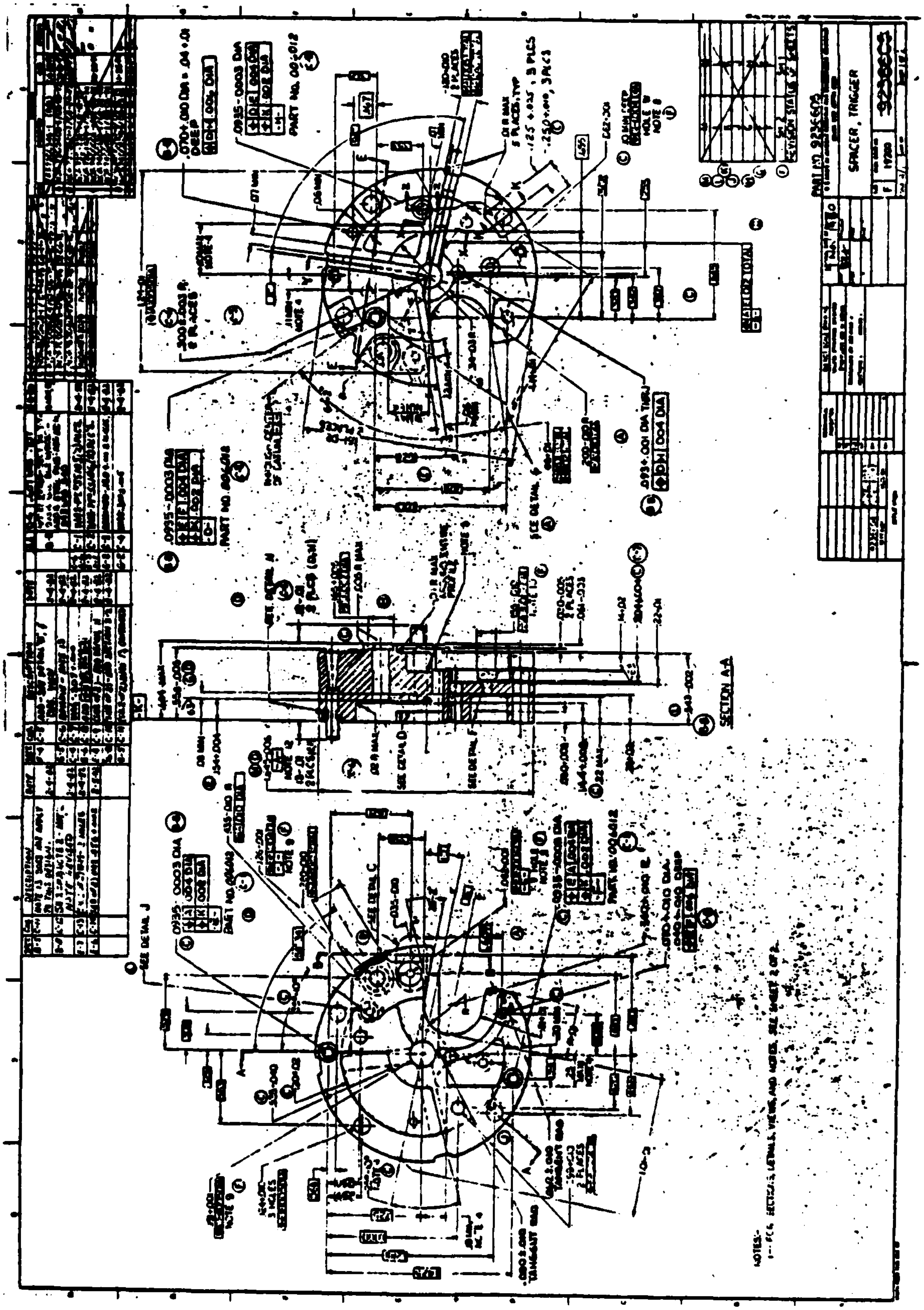
The Safety Plate Shaft length between endshake shoulders has been increased to 0.536 - 0.002 inches from 0.123 - 0.003 inches. The modified Safety Plate Shaft is shown in Appendix E.

4. <u>Top Plate</u>

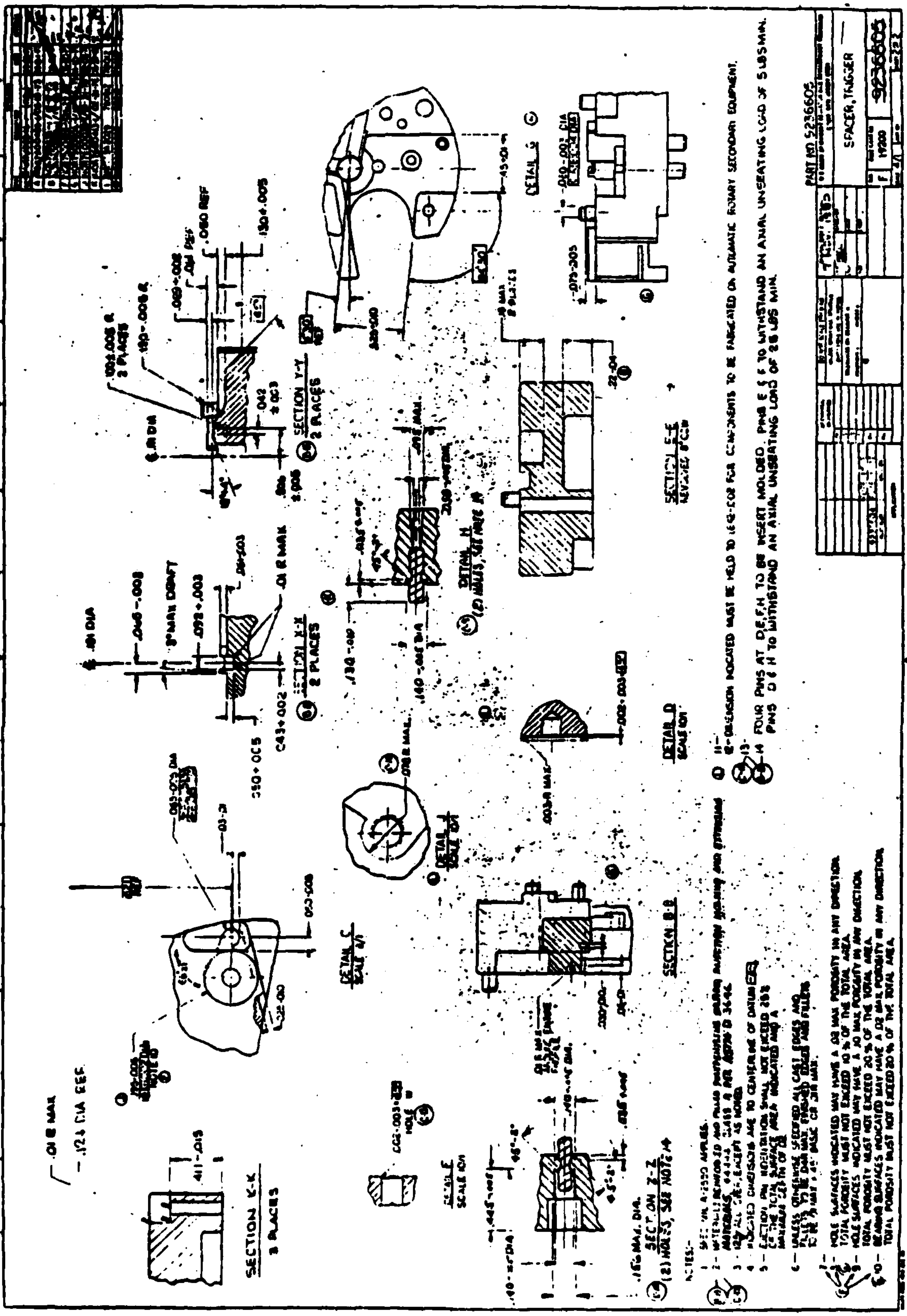
The Safety Plate Shaft top pivot hole (0.062 + 0.001 DIA, through hole) has been added in the Top Plate (0.022 from the center line and 0.940 from the Datum -A-). The modified Top Plate is shown in Appendix F.

APPENDIX B

PROPOSED TRIGGER SPACER DRAWING



SK5672


SECTION K-K
3 PLACES
SECTION B-B
SECTION E-E
DETAIL C
DETAIL D
DETAIL G
SECTION Y-Y
SPACER, TRIGGER

SK5672

APPENDIX C

CHANGES AND MODIFICATIONS MADE IN
TRIGGER SPACER (9236605)

The following changes and modifications have been made in the proposed Trigger Spacer.

1. <u>Material Change</u>

The Trigger Spacer material has been changed

to: "Reinforced and Filled Polyphenylene Sulfide - injection molding and extrusion material - 44444, Class A per ASTM D 3646"

from: "Aluminum Alloy - Die Casting, SC-84B or SG-100A per ASTM B 85".

2. <u>The following Trigger Spacer dimensions have been changed to increase the load bearing area of the proposed Trigger Spacer</u>. (See Section III-A-1.)

A. The chamfer of the Firing Pin hole at the top plane has been removed. Consequently, the "Section H-H" drawing has been removed.

B. The Firing Arm rotation clearance has been decreased to 0.340 + 0.010 R. It was 0.210 + 0.010 from the Firing Arm Assembly slot, which is equivalent to 0.550 + 0.010 R. Consequently, tangent radii 0.090 ± 0.010 R (at the juncture of the clearance radius and the Firing Pin hole) and 0.062 ± 0.010 R (at the juncture of the clearance radius and the spacer outside the diameter) have been added.

The "Section A-A" drawing reveals this change at the Safety Plate Shaft hole, which shows the solid section extended to the top plane.

C. The circular recess area at the edge of the bottom plane has been changed to a solid section except at the three (3) screw hole slots. The dimensions and the position of three (3) screw hole slots are shown in the "Bottom View" drawing. The depth of these slots is shown in "Section K-K" drawing.

The "Section A-A" and "Section E-E" drawings have been changed accordingly to reveal this change by showing the solid section, which is extended to the bottom plane at the outside diameter.

D. Two holes (0.70 + 0.01 DIA x 0.04 + 0.01 Depth) have been added to the Trigger Spacer to provide clearance for the half shear protrusion (Section E-E in 9236608) of the Top and Bottom Plates. One hole is located at 0.607 and 0.308 inches from the center line on the top plane, and the other hole is located at 0.167 and 0.655 inches from the Datum -D- on the bottom plane.

This change is shown in the Top and Bottom views of the Trigger Spacer drawing.

3. <u>The following Trigger Spacer dimensions have been modified to increase the endshake between the Firing Arm Shaft and the Top and Bottom Plates</u>. (See Section V-A-1)

A. The spacer height from the top plane to the bottom plane has been increased to 0.543 - 0.002 inches from 0.540 - 0.002 inches (or 0.543 - 0.005 inches because of Note 11 in 9236605). Consequently, the overall maximum spacer height has been increased to 0.604 inches from 0.601 inches.

Also, the note concerning the spacer height in "Note 11" has been eliminated. The calculation shows that the requirements of the Firing Arm protrusion (0.065 - 0.029 inches) and the overall height (0.670 inches max.) should be met if the trigger parts meet the respected requirements.

B. In "Detail Drawing D," the recess depth requirement has been eliminated. Also, the note concerning this requirement in "Note 11" has been eliminated. These requirements are not necessary because the dimensions in "Section A-A" drawing defines this recess depth requirement.

4. The design change made to avoid the creep deformation of the Safety Plate Shaft top pivot hole. (See Section III-A-3-f.)

The Safety Plate Shaft top pivot hole (0.062 + 0.001 Dia. x 0.100 min. Depth) has been changed to a Safety Plate Shaft clearance hole (0.093 + 0.001 Dia. - thru hole), and the pivot hole is provided in the Top Plate (see Appendix A-4 and Appendix F). Consequently, the requirement "2 holes W" has been removed from the "Bottom View" and the "Detail F" drawing.

This change is shown in the "Bottom View," "Section A-A," "Top View," and "Detail G" drawings.

5. The design change made to accommodate the new design of attaching the MDF to the three module assembly (Contract No. DAAK10-79-C-0169, Task No. 2).

A. The MDF channel side wall (2 places) has been added, and the dimension is shown in the "Bottom View" and the "Section X-X" drawings.

B. The MDF channel width has been decreased to 0.092 + 0.003 inches from 0.120 + 0.010 inches, and this change is shown in the "Bottom View" and "Section X-X" drawings. Because of this change, the distance of the void's side position has been increased to 0.070 inches min. from 0.060 inches min. as shown in the "Bottom View" drawing.

C. The bottom contour of the MDF channel has been modified, and the juncture of the outside diameter and the MDF channel has been chamfered. These modifications are shown in the "Section Y-Y" drawing.

6. <u>The design change made to show the insert molded Locating Spacer Pins</u>. (See Section III-B.)

A. The dimensional requirements of the insert molded Locating Spacer Pins (E, F, D, and H) are shown in the "Top View," "Section A-A," "Bottom View," "Section Z-Z," and "Detail H" drawings. Consequently, the radius in the "Detail J" drawing has been changed to 0.078 inches R. max. from 0.056 + 0.010 inches Radius.

Also, the "Section B-B," "Section E-E," "Section Y-Y," and "Detail G" drawings show the inserted molded Locating Spacer Pins.

B. The "Note 14" has been added to define the unseating load requirements of four (4) Locating Spacer Pins.

C. The Datum -K- has been assigned to the top plane in the "Section A-A" drawing to define the perpendicularity of the Locating Spacer Pins.

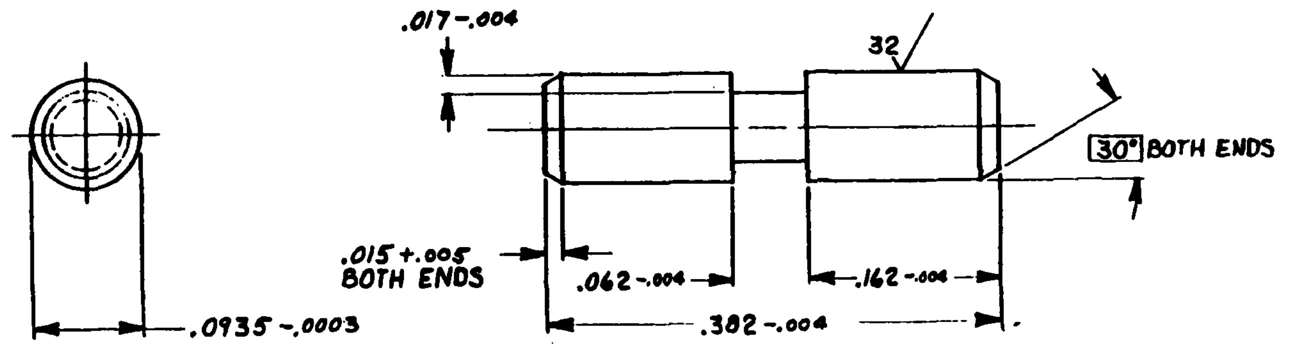
7. <u>The "Note 7" has been eliminated</u>.

The "Note 7" which defines the maximum porosity at the unmachined area has been eliminated because no machining is necessary on PPS Trigger Spacer.

APPENDIX D

PROPOSED SPACER PIN DRAWING

REVISIONS							
REV	ZONE	BY	DESCRIPTION	ECO	DATE	CHKD	APPD
a		JCY	Hardness requirement added		Apr 6, 82	JCY	JCY



REF. (58105 / 58106)

PART NO. 9236487-1 & 2

NOTES:

1 - SPEC MIL-A-2550 APPLIES.
2 - MATERIAL :- FREE-MACHINING STAINLESS AND HEAT-RESISTING STEEL WIRE. TYPE 416 OR 416Se, ASTM A581. KHN 313-380 (500 gr Load)
3 - 125√ ALL OVER EXCEPT AS NOTED.

DRAFTSMAN D.S.	DATE 1-29-82	CUSTOMER NO.	REV.
CHECKER	ENGR. J.Y.	PIN, SPACER, TRIGGER	
APPROVAL	PROJ. ENGR		
UNLESS OTHERWISE SPECIFIED TOLERANCES ANGLES ± DECIMALS ± FRACTIONS ±	SURFACE FINISH √	SCALE 10:1	UNIT WT. / SHEET OF
NEXT ASSY. / USED ON — APPLICATION			
DO NOT SCALE THIS DRAWING			
DIMENSIONS ARE IN INCHES			
SYSTEMS DIVISION HAMILTON WATCH CO., LANCASTER, PA 17604	CODE IDENT NO. 26992	SIZE B	DRAWING NO. 006012

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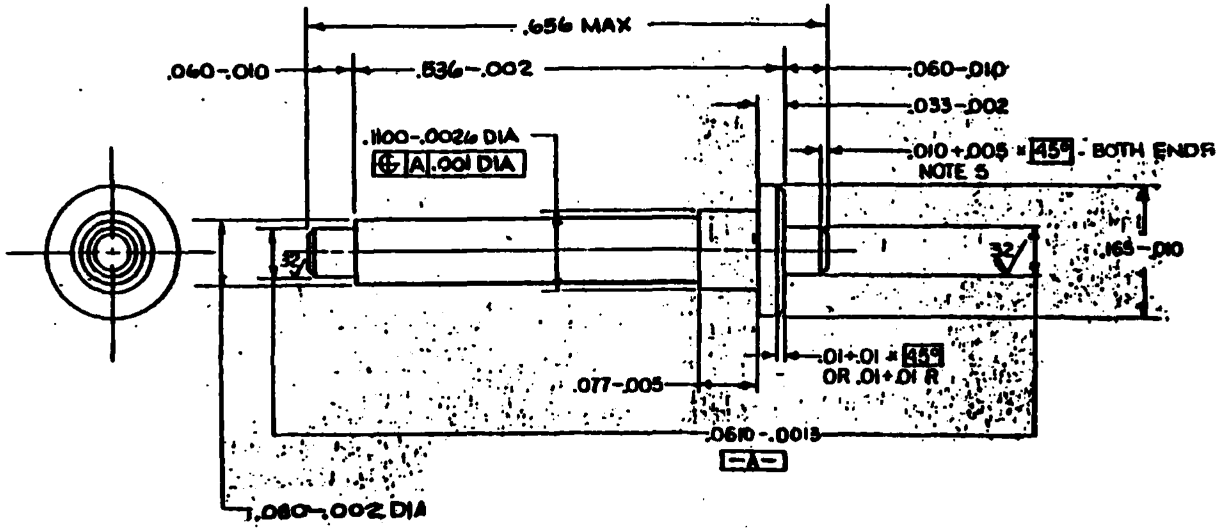
APPENDIX E

MODIFIED SAFETY PLATE SHAFT DRAWING



NOTES:-
1-SPEC MIL-A-2550 APPLIES.
2-MATERIAL:-ALUMINUM ALLOY BAR, ROD, AND WIRE; ROLLED, OR COLD FINISHED, 2024 TEMPER T4, ASTM B 211
3-125/ALL OVER EXCEPT AS NOTED.
4-UNLESS OTHERWISE SPECIFIED ALL FILLETS WILL BE .005 R MAX AND ALL CORNERS WILL BE .005 R MAX OR .005 MAX × 45° BASIC.
5-KEY AUTOMATIC ASSEMBLY DIMENSIONS.

REVISIONS

SYM	DESCRIPTION	DATE	APPROVAL
a	Changed material spec.	Apr ...	...

PART NO. 9236619

9-3 667	FUZE, M582 / FUZE, M577	MECHANICAL PROPERTIES	DO NOT SCALE DRAWING. UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. TOLERANCES ON DECIMALS ± FRACTIONS ± ANGLES ±
NEXT ASSY	USED ON		
APPLICATION			

ORIGINAL DATE OF DRAWING 12 OCT 81

U.S. ARMY ARMAMENT COMMAND
PICATINNY ARSENAL, DOVER, NEW JERSEY 07801

SHAFT, SAFETY
PLATE, TRIGGER

SIZE	CODE IDENT NO.	
C	19203	005526
SCALE 10/1	UNIT WT.	SHEET

(REF 5817)

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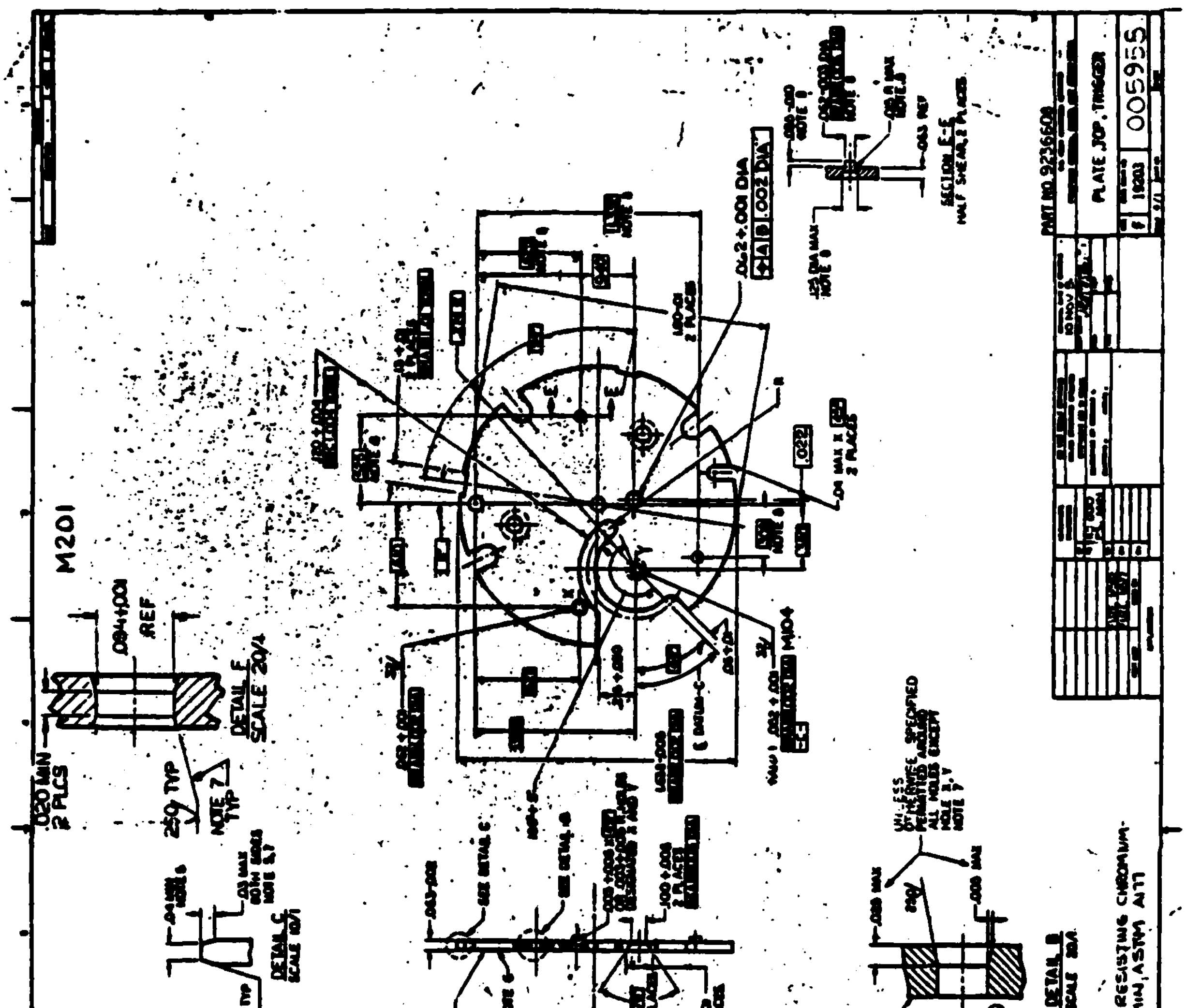
APPENDIX F

MODIFIED TOP PLATE DRAWING



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APPENDIX G

WEIGHT AND MASS OF TRIGGER PARTS

Part or Assembly	Material and Design	Weight Lbs.	Mass Slug
Spacer with 4 Pins	Aluminum	0.0625	
	PPS	0.0418	
Firing Arm Assembly	S/S	0.0048	0.0001419
Rotor Detent Assembly	S/S	0.0010	0.0000311
Release Lever Assembly	S/S	0.0016	0.0000497
Safety Plate Assembly	S/S (current)	0.0018	0.0000560
	S/S (proposed)	0.0023	
Top Plate	S/S (current)	0.0328	
	S/S (proposed)	0.0327	
Bottom Plate	S/S	0.0230	
Two Rivets	Aluminum	0.0026	
Setback (Pin, Spring and Retainer)	S/S	0.0007	
Firing Pin, Spring and Insert	S/S	0.0020	
Total	Aluminum	0.1328	
	PPS	0.1125	0.0034938
Timing Scroll Assy. less Mainspring Barrel		0.0758	
SSD Assembly	Current	0.0746	0.0023168

APPENDIX H

CALCULATION OF ω AND α

1. Calculation of ω

The value of ω (angular velocity in radians per second) may be found from 30,000-RPM spin as follows:

$$\omega = 30{,}000\ \frac{\text{revolutions}}{\text{minute}} = 30{,}000\ \frac{2\pi\ \text{radians}}{\text{minute}} = 30{,}000\ \frac{2\pi\ \text{radians}}{60\ \text{seconds}}$$

$$= (30{,}000)(2\pi/60)(\text{radians/sec}) = 3141.6\ (\text{rad/sec})$$

2. Calculation of α

The value of α (angular acceleration in radians per second squared) may be found as follows:

$$\alpha = \frac{d\omega}{dt} = \frac{d}{dt}(KV) = K\frac{dV}{dt} = K \cdot a$$

where K is a constant and $K = \frac{12(2\pi)}{ND}$, Radians/ft

V is the Muzzle Velocity in ft/sec

N is a constant and N = 1/twist

D is the Land Diameter in inches

a is the Linear Acceleration in ft/sec^2

Now the maximum linear acceleration (a) is 30,000 g, and the maximum K results when the value of N times D is the smallest. From the cannon data in Report No. APG-MT-4503 (Methodology Investigation on Setback and Spin by Heppner), the minimum value of N times D is 47.19 for the 40mm gun.

therefore, $\alpha = K \cdot a = [12(2\pi)/ND]\ a = 1.60(a) = 1.60(30{,}000\ g)$

$$= 1.60(30{,}000)(32.2) = 1{,}545{,}600\ (\text{rad/sec}^2)$$

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