A DYNAMIC LOADING DEVICE FOR AIRCRAFT SEATS

By

JOSEPH ROBERT LOMBRANO

Bachelor of Science

University of Kansas

Lawrence, Kansas

1948

1949

Submitted to the faculty of the Graduate School of
the Oklahoma State University
in partial fulfillment of the requirements
for the degree of
MASTER OF SCIENCE
August 1961

A DYNAMIC LOADING DEVICE FOR AIRCRAFT SEATS

Thesis Approved:

R. E. Chapel
Thesis Adviser

W. H. Easton

[illegible signature]
Dean of the Graduate School

PREFACE

The rising trend of aircraft accidents has necessitated a closer look at all aspects of flight safety. One such aspect is the realistic testing of passenger seats and tie-downs.

Before arriving at any type of solution, it is necessary to study carefully all dynamic effects acting on the seat and its subject. The initial portion of this study summarizes the results of numerous crash-impact investigations. The second portion covers the design of a loading device for dynamic testing of seats.

Indebtedness is acknowledged to Professor Raymond Chapel for his assistance, guidance and encouragement during the study, and to Douglas Aircraft Company, Inc., whose scholarship assisted materially in the M. S. program. I am grateful to my wife whose moral support and sacrifice made my advanced studies possible.

TABLE OF CONTENTS

LIST OF FIGURES

CHAPTER I

INTRODUCTION

Commercial air transportation is being used more and more by the traveling public for a mode of conveyance. Concurrently, there is a rising trend in airline fatality rate. Official statistics list the passenger fatality rate for 1959 at 0.68 for each 100 million passenger miles, almost twice the 1958 rate of 0.38 and three times the 1957 rate of 0.20.

The objective of aviation safety is twofold. (1)[1] The first is the preservation of human life; the second is the preservation of property. It should be noted that despite the success in operational safety that has been achieved through accident prevention efforts, the fact remains that no one can prevent all accidents. Therefore, accepting the inevitability of accidents, an effort to improve passenger survivability is in order. One approach to this general problem is passenger seat improvements, since the seat is the restraint in the event of crash impact. Typical aircraft seats are shown in Figures 1 and 2.

Aircraft passenger seats are presently required to withstand various test loads per military or commercial specifications. (2)(3). These loads are applied under static test conditions, but the static

---

[1]Note: ( ) refers to Selected Bibliography.

Figure 1. High Density Seat

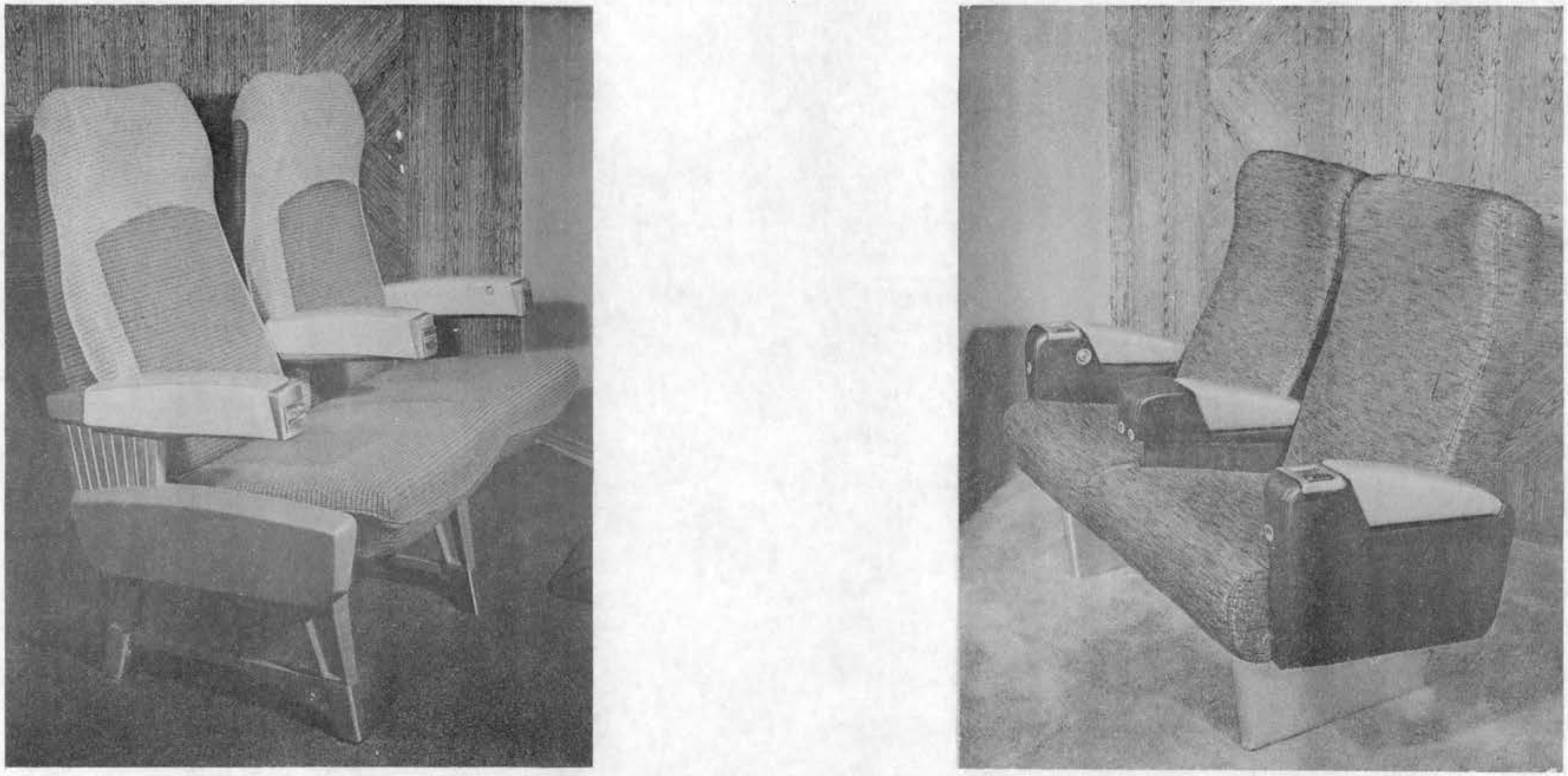


Figure 2. Typical Aircraft Seats

strength of a seat may give a significantly erroneous indication of its ability to resist dynamic loads. There is a great need for the dynamic testing of seats and aircraft structures to provide a maximum of protection and to permit a minimum deterrent to profitable enterprise in civil aviation.

Until recent years, there was no indication of a realistic criterion for a test basis. At least two groups have been actively making crash impact studies. One is the NACA Lewis Flight Propulsion Laboratory, and the other is the Aviation Crash Injury Research Group at Cornell University.

The crash of an aircraft results in loads on the seats due to the deceleration of the aircraft relative to the "seat-man". To define the loads which will probably occur on the seat, the following factors must be considered:

1. Pulse magnitude.
2. Pulse rate of onset.
3. Pulse duration.
4. Pulse direction.

It is essential to realize that not only the crash load varies in magnitude, rate of onset, duration and direction in each accident, but time also must be considered as a variable. (4).

CHAPTER II

DEFINING THE LOADS

A literature survey was made to define the loads which will occur on airplane seats. Relationships were needed to show the response of the aircraft seats to the dynamic loads which are imposed on the aircraft.

According to Preston and Moser (4), simulated crashes of fighter and transport aircraft resulted in floor loads, for fighters crashed at 112 mph at 27° angle of impact, of 40 g's longitudinally and 60 g's normal to the longitudinal axis. Unpressurized transport tests resulted in 8 to 16 g's longitudinally and 9 g's normal, with tests at 16° angle of impact and speeds to 109 mph. Pressurized transport tests resulted in 22 g's longitudinally and 25 to 31 g's normal loads at 27° impact angle.

From simulated airplane crashes, Preston and Pesman (5), show that peak maximum longitudinal loads are 20 to 25 g's at crash impact angle of 29°.

Pinkel and Rosenberg (6) suggest an impulse deceleration curve using the value of 20 g's for transports having a landing speed of 180 mph.

Acker, et al., (7), in an investigation of accelerations in fighter-airplane crashes, show that resultant loads of 60 g's were obtained at the airplane C.G. in simulated crashes at an impact angle of 27°. The normal loads exceeded 20 g's.

Eiband, et al. (8), show that in light-airplane simulated crashes at speeds of 42, 47 and 60 mph, the maximum longitudinal decelerations were 26 to 33 g's and restraint forces in seat-belt-shoulder harness combination exceeded 5800 lbs. (29 g's for a 200-lb seat occupant).

A load direction spectrum is defined by Hasbrook (9). The principal crash force zones are described as within a horizontal arc whose sides are 30° to either side of the aircraft's longitudinal axis and from 30° above to 45° below, the longitudinal axis. It is pointed out that loads in excess of 20 g's may be expected in portions of this spectrum in survivable type accidents. Hasbrook considers that the increase of static strength requirements for seats to 9 g's is not adequate. (3).

A dynamic test of seats with longitudinal load only is presented by Sorin (10). A hydraulic catapult with a nine-foot stroke propels a cart along tracks 200 feet in length. The results show that:

1. The seats which were mounted facing forward failed completely when an average acceleration of 20.2 g's lasting 0.161 seconds was applied.
2. The seat which was mounted in an aft-facing position showed serious failure of the back rest when an average acceleration of 10 g's lasting 0.222 seconds was applied.
3. The seat back was found to deflect from 3.2 to 8.9 inches, and the anthropomorphic dummy's buttocks moved horizontally 2.0 to 4.5 inches.

Hawkes (11) suggests that the key to improving survival chances is in designing the tie-down of passengers and loose equipment up to the ultimate load factor of the airframe, since the anatomy of man can withstand impacts greater than the airplane structure. Hawkes (11) also points out that unrealistic testing procedures have sometimes made it impossible to capitalize on the advantages of a strong floor.

The simulated crashes, as described in the literature, could not be considered as representative of the conditions that would prevail with present day airplanes. In most cases, the airplanes were unloaded and the landing speeds were quite low. These conditions would certainly tend to produce conservative crash loads. But even the tests as they were recorded show that the forward area of the passenger compartment sustained loads in excess of 20 g's.

From a standpoint of maximum passenger safety, the seat test loads should be specified higher than the 20 g's recommended by Pinkel and Rosenberg (6) and Military Specification (2). There is no way of assuring that any given seat will not be located in the area of high g's. An increase in seat strength may perhaps necessitate a slight weight increase for the entire aircraft, but in light of achieving greater passenger safety, any dissentient would have to answer the question: What is human life worth?

It is extremely dubious that there need be any weight penalty at all, because the ultimate in design efficiency has not been reached even with the latest seats.

With a realistic test, the loads should have a different distribution than for a static test. A significantly different seat response should be attained. There should be little or no necessity to increase weight but merely redistribute the present material for a better efficiency.

Both of the seat specifications, one by the FAA and the other by the Military, are considered inadequate. Hasbrook (9) states that the survivable crash condition parameters found in most survivable transport crashes investigated by AV-CIR are 150 knot impact speed and a resultant crash force angle within an arc extending from 15° above, to 45° below, the longitudinal aircraft axis. Referring to Figure 3, if an angle of 25° is assumed, the maximum longitudinal acceleration is 16 g's for an impact speed corrected to 95 mph.

For simplicity, assume the maximum acceleration resulting from the first impact of the airplane to vary directly with the initial momentum and thus, with the initial velocity. Based on this assumption, the acceleration would be approximately 30 g's for an impact speed corrected to 180 mph. More realistic seat tests would be obtained by using an ultimate load corresponding to 30 g's in a direction along the longitudinal axis of the aircraft and 20 g's normal to the longitudinal axis of the aircraft.

Most transports are capable of withstanding 30-40 g's fuselage loading and still maintain an intact cabin, but the human body can withstand 45 g's for several hundredths of a second and can withstand 20 g's vertically. Human tolerance to acceleration loads for various conditions are shown in Figures 4, 5, and 6. (4).

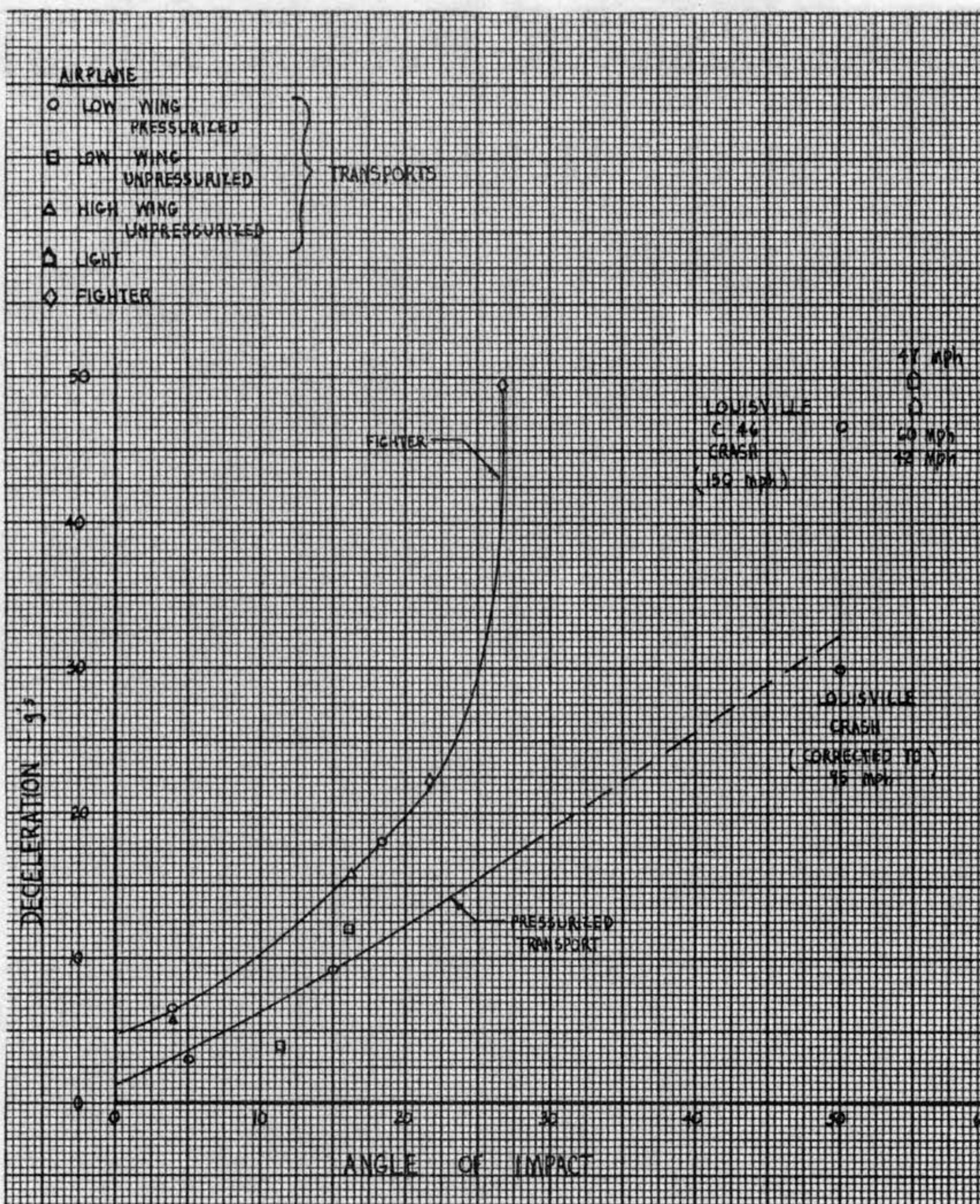


Figure 3. Crash Deceleration vs. Angle of Impact

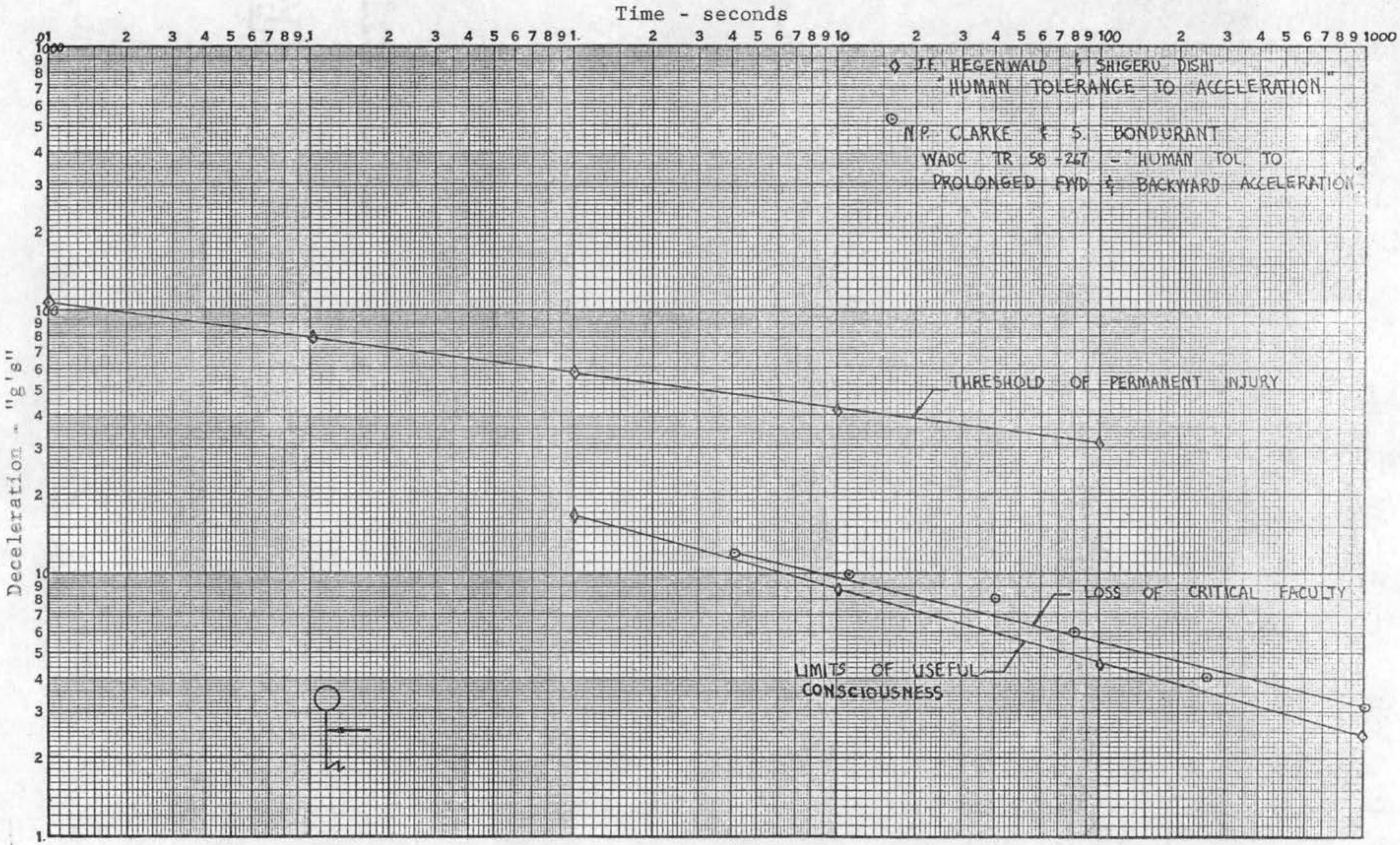


Figure 4. Human Tolerance to (Supine) Transverse Acceleration

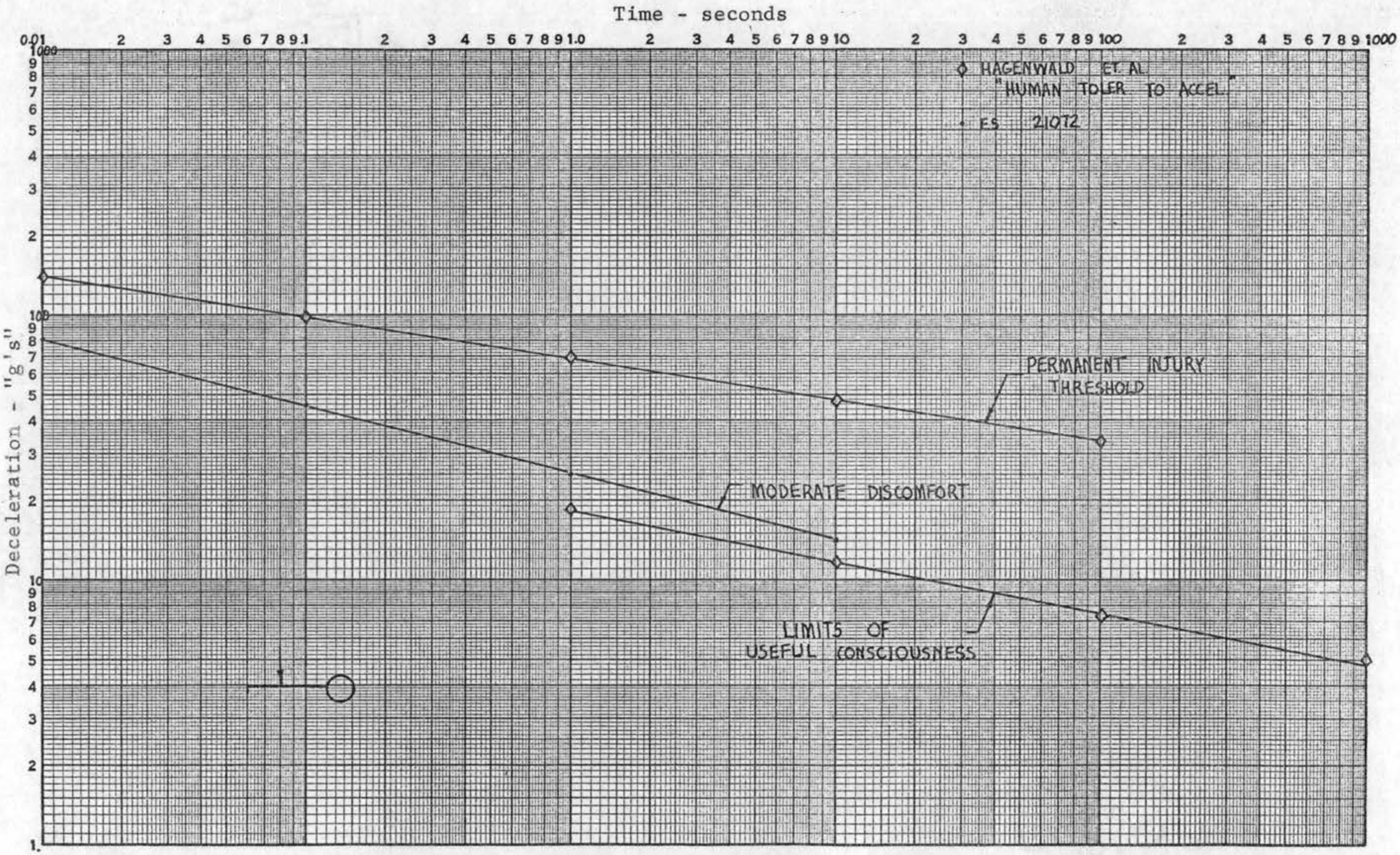


Figure 5. Human Tolerance to (Prone) Transverse Acceleration

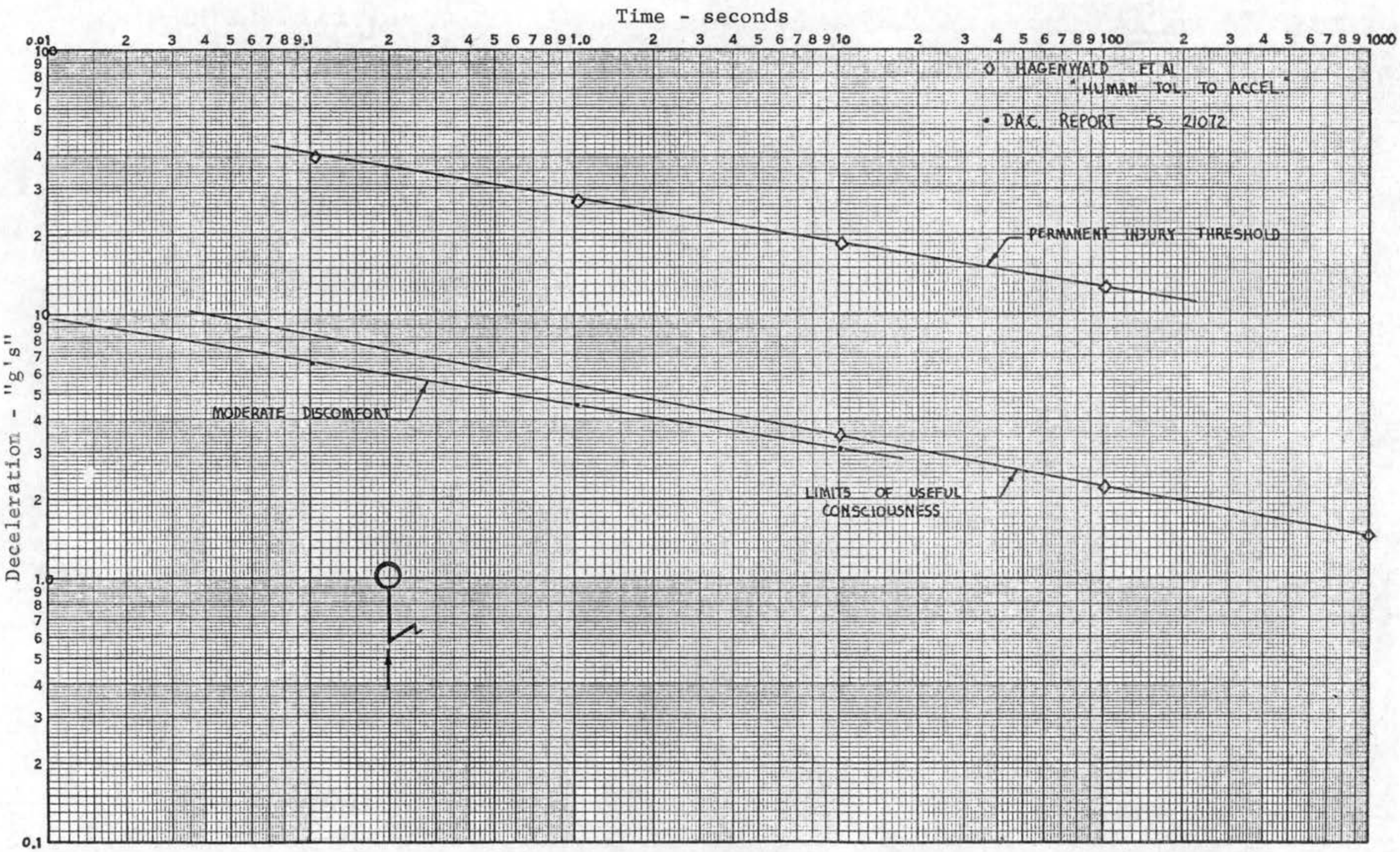


Figure 6. Human Tolerance to Vertical Negative Acceleration

CHAPTER III

APPLICATION OF LOADS

The requirements for seat strength as found in TSO-C39 and related military specifications, consider the horizontal and vertical seat loads to be applied separately.

From a study of the crash data and deceleration rates, it was found that the vertical deceleration acts in some combination with the longitudinal deceleration. The decelerations may not peak in phase, but the in-phase components may be additive.

The longitudinal force of the passenger is transmitted to the seat through the safety belt. The belt is generally inclined upward from the horizontal, thus the load transmitted to the seat has a vertical component. The vertical load component of the crash impact may, therefore, increase the total vertical load on the seat. In view of the fact that the human tolerance to vertical acceleration is approximately 20 g's, there is little need to test for loads greater than 20 g's normal to the longitudinal axis of the aircraft.

From anthropometric seating studies, as shown in Figures 7 and 8, the load application points as specified in MIL-S-7877 and FAA TSO-C39 appear to agree closely. These locations are as follows:

1. A 30-g load acting rearward uniformly distributed and applied over an area corresponding to 16 inches of width of the seat back with the load C.G. at a point 10.5 inches up from the

Configuration 1

5'9" male passenger (shoes on)
170# assumed weight
Comfortable sitting position
Arm on arm rest

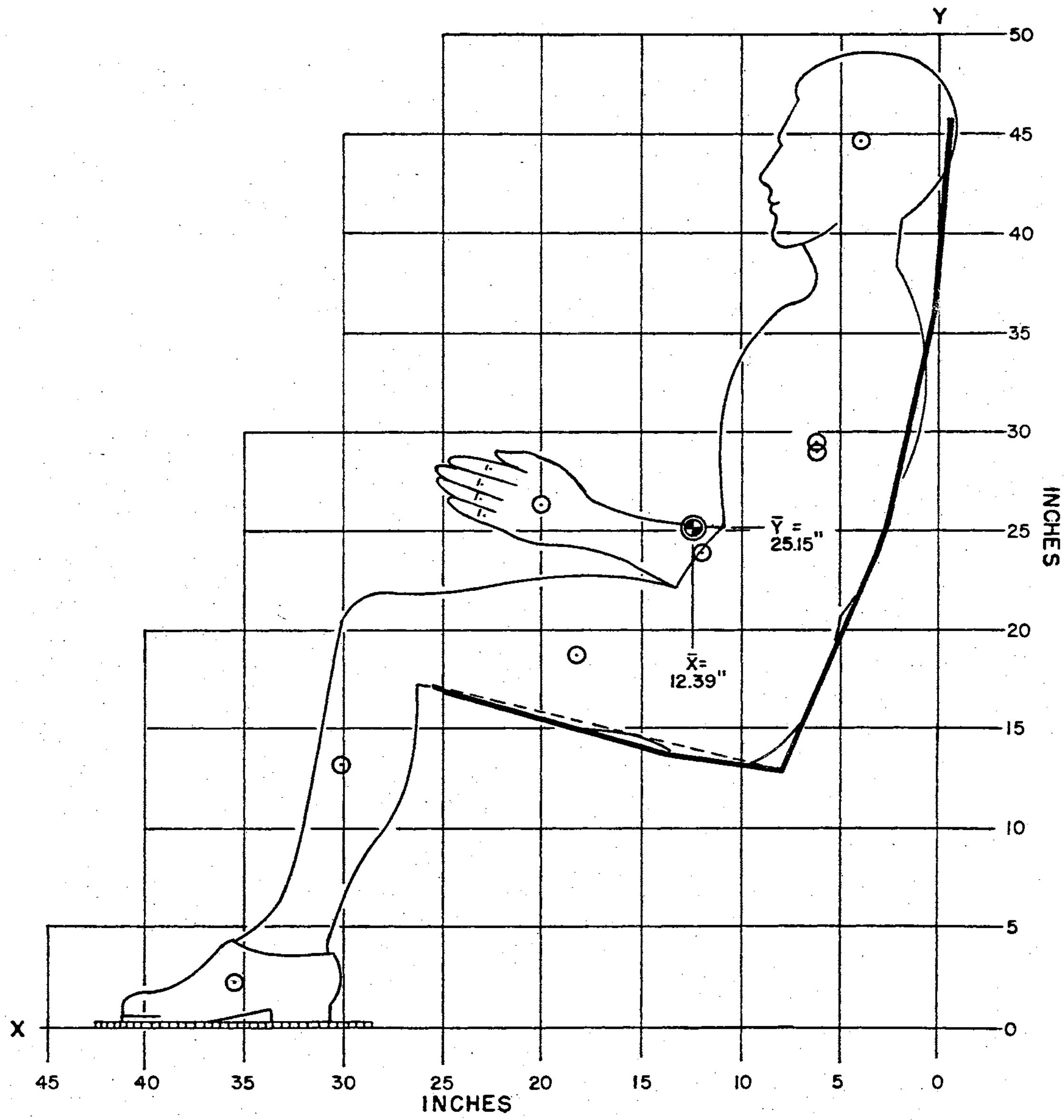


Figure 7. Center of Gravity Standardization Configuration 1

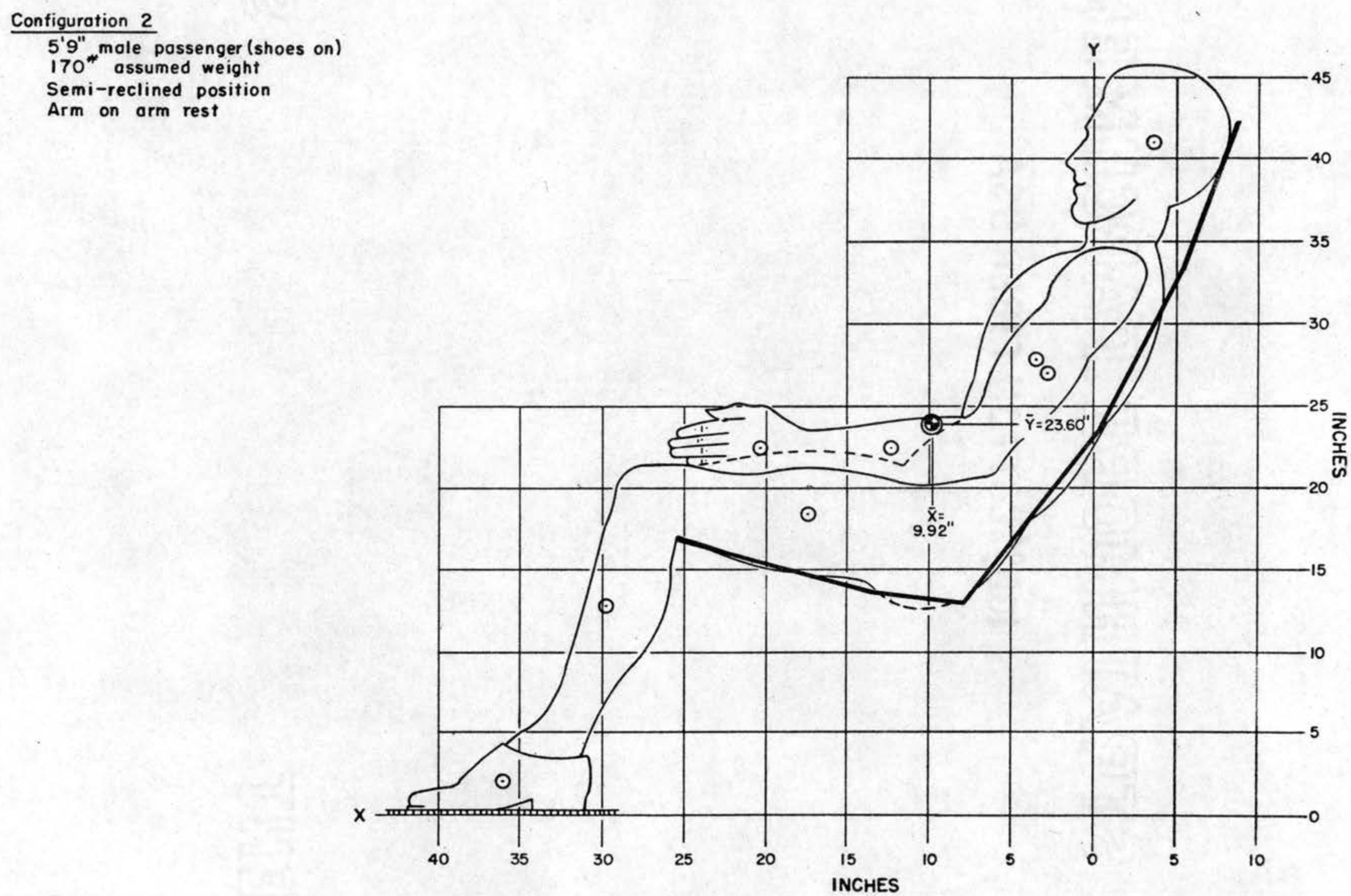


Figure 8. Center of Gravity Standardization Configuration 2

base of the seat back. The load could occur under three separate conditions as follows:

a. Directly to the rear;

b. Thirty degrees to the right;

c. Thirty degrees to the left.

2. A 20-g down load applied evenly over the seat bottom.
3. A 30-g load applied to the belt. This load to be applied upward and forward in a single plane which makes an angle of 40° with the horizontal.
4. A 3-g side load applied to the armrest outward or inward perpendicular to the armrest in a horizontal plane. The load to be applied midway fore and aft on the armrest.

CHAPTER IV

DESIGN DEVELOPMENT OF A TESTING UNIT TO APPLY LOADS TO THE SEAT

Several possibilities of a motivation source for applying the dynamic loads to a test seat were considered. The motivation sources that were investigated are as follows:

1. Hydraulic Piston
2. Chemical Catapult
3. Dropping weight with cables
4. Electric motor and clutch arrangement
5. Pneumatic Piston

The chemical (explosive) catapult has serious disadvantages in that control of the pressure pulse and piston travel are difficult, and development cost is likely to be too high.

The dropping weight with cable system and the electric motor and clutch arrangement were eliminated because of possible adverse effects due to cable elongation, control of pressure pulse and probable high development cost.

The systems that were considered to have the required capacity with a low initial cost are the hydraulic (oil) piston and the pneumatic (gas) piston. The pneumatic piston system was chosen to take advantage of the compressibility relationships for the charging gas.

Some of the requirements that need to be met are summarized as follows:

1. Pulse magnitude should be a maximum of 30 g's.
2. Pulse should have a sharp front (500 g's per second).
3. Pulse should be controllable as to magnitude and duration.
4. Pulse shape should be triangular.

These requirements are shown by the various curves in Figure 9.

Use of a pneumatic system is considered to give relatively lower cost due to the lack of complexity in design. The pulse magnitude, duration, and shape can be controlled by control of the gas expansion in the cylinder.

For applications on both forward-facing seats and aft-facing seats, the piston will need to be operable in a push-pull configuration; and in order to get a triangular pulse shape, a quick opening valve will be needed. Also the maximum piston travel will have to be approximately 9.0 inches. (10).

There are two ways of introducing gas pressure on the piston. One way is to use a quick opening valve of proper size to allow rapid entrance of the gas into the cylinder. The other way is to allow the gas pressure to build up in the cylinder directly on the piston by locking the piston rod.

In the latter method, a definite requirement would be a quick release for unlocking the piston.

For either method, a quick opening exhaust valve is required in order to obtain the proper pulse shape. It is planned to have the action required to open the exhaust valve conveniently timed to the piston travel. This would insure the proper valve opening sequence.

The mass of the piston and rod may or may not be of consequence.

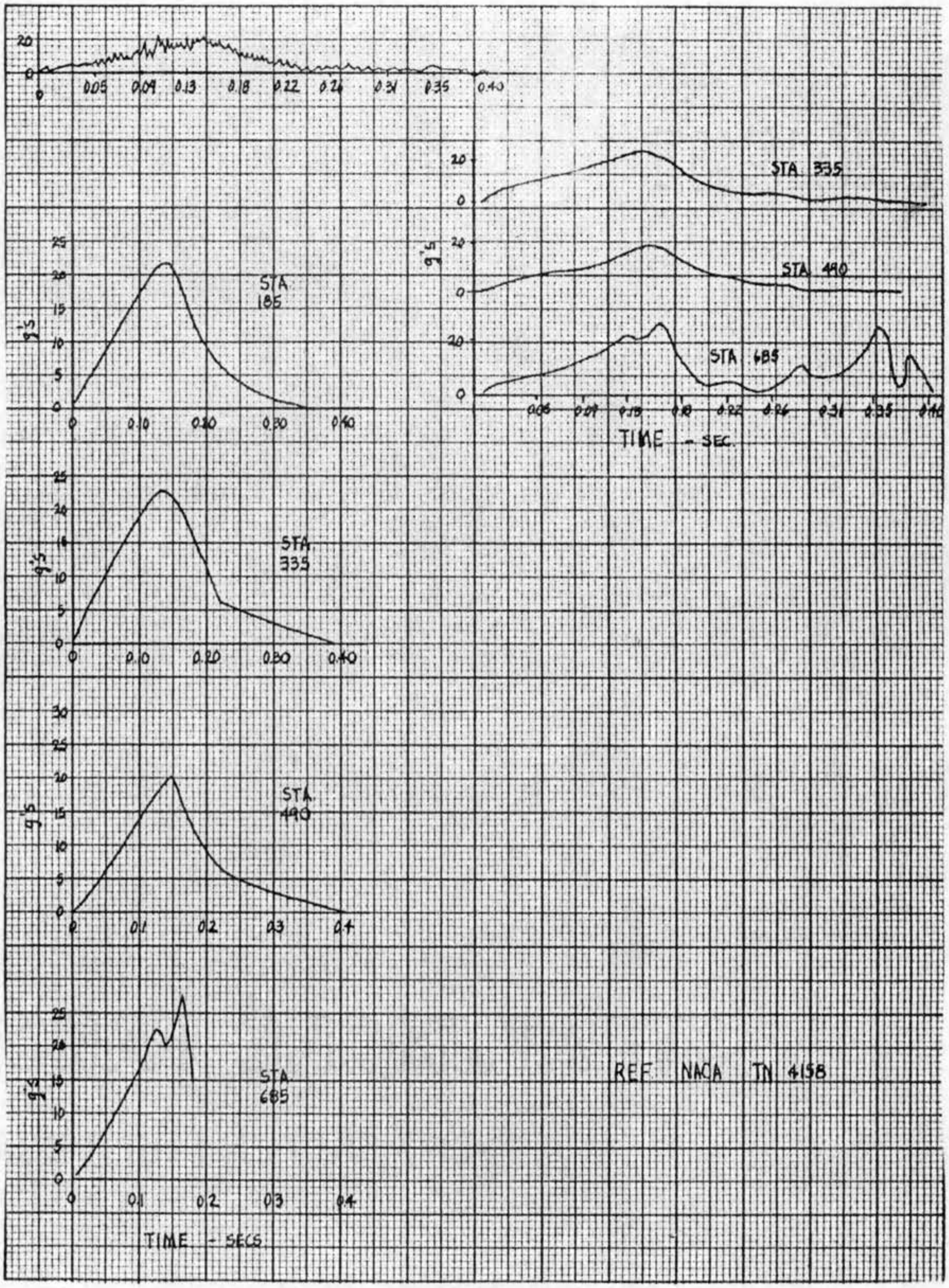


Figure 9. Crash Impulses From NACA Tests

Consideration will have to be made of this mass and its effect on the pulse shape or on the inertia effect on the seat. The overall size of the cylinder and piston should be the determining factor.

Likewise, the frame of the testing unit will have to be considered from the standpoint of deflections as well as stress. The deflections of the frame must be kept small in order not to affect the pulse shape.

The basic components of the testing unit are therefore:

1. A suitable frame
2. A set of pneumatic cylinders
3. A properly timed system to actuate the desired sequences
4. A set of appropriate tie-downs
5. A section of typical aircraft type flooring

The proposed complete test facility is shown in Figure 10.

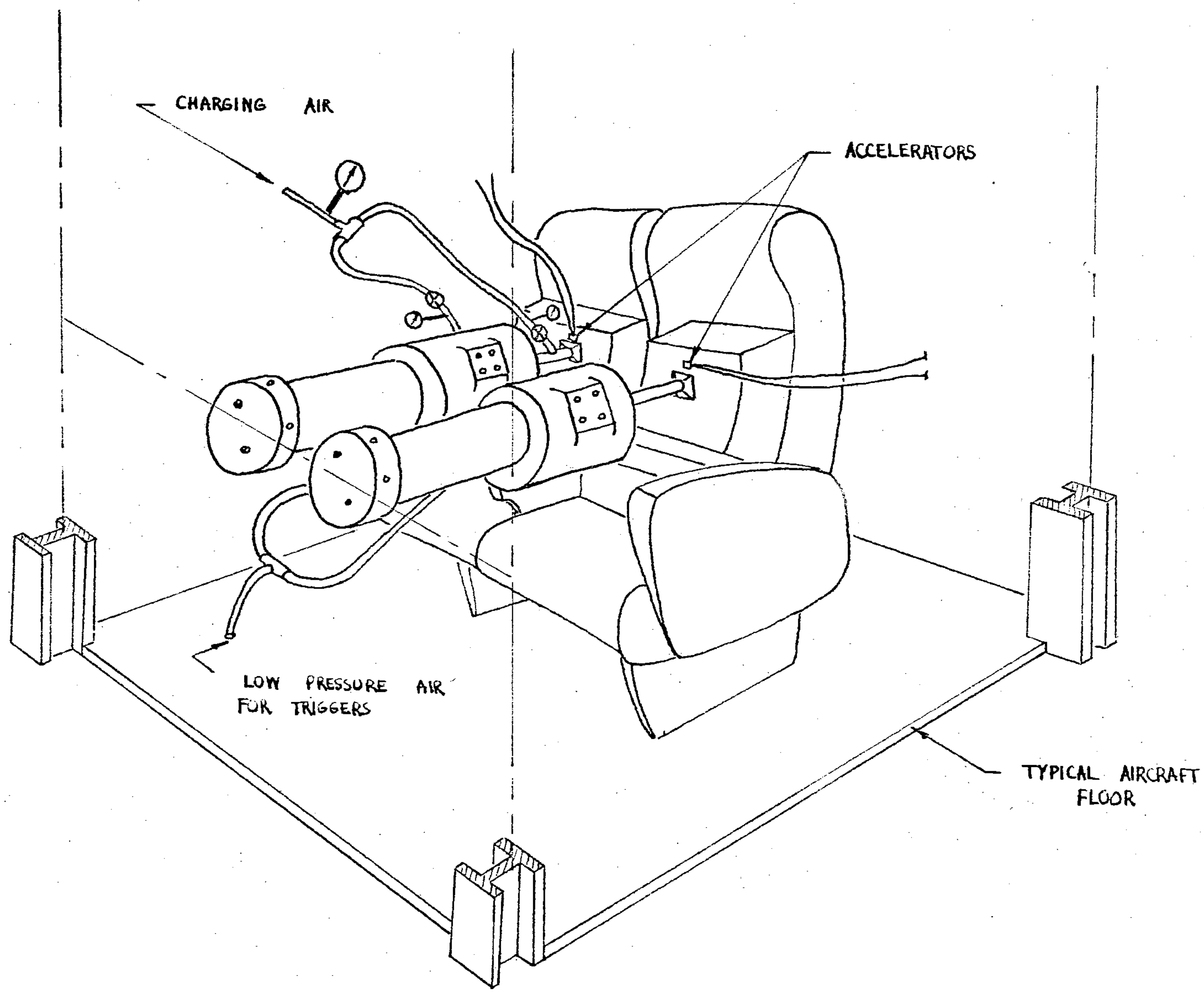


Figure 10. Proposed Test Facility

CHAPTER V

DESIGN OF PROPOSED LOADING DEVICE

The design parameters for the Dynamic Load Device are as follows:

1. Pulse magnitude of 30 g's.
2. Pulse onset of 500 g's per second.
3. Pulse duration of 60 milliseconds.
4. Pulse shape to be triangular.

These requirements are summarized in Figure 11.

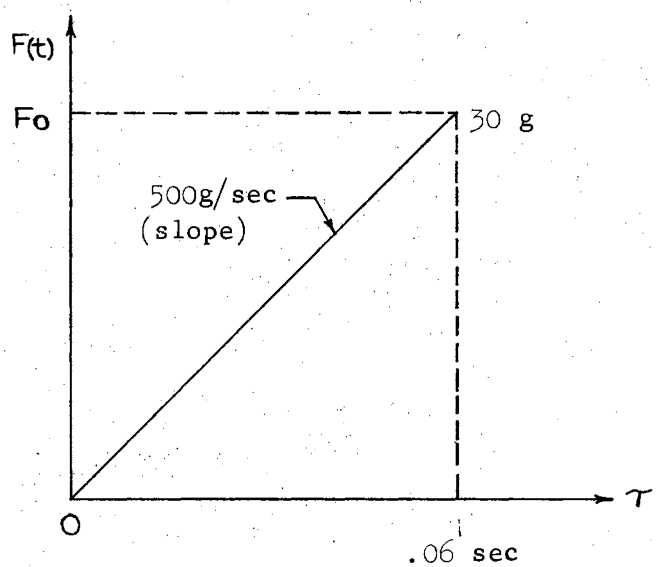


Figure 11. Triangular Impulse

DESIGN LAYOUTS

Figure 12 is a layout of the Dynamic Load Device. The proposed design for the Variable Area Orifice is shown in Figure 13. The loading device is operated by first introducing air into the variable area orifice cylinder and pressurizing so that there will be a force slightly higher than the force acting to open the variable orifice, plus the force acting to open the end orifice. A check valve is used to hold the pressure. Air is then introduced to the front side of the piston through the back pressure charge inlet until the pressure reaches 562 psia. Next, the annular area around the piston is charged to a pressure of 663 psia. The device is thus cocked and ready for firing. (Calculations for the necessary initial pressures are included in later sections of the chapter.)

The Dynamic Load Device is fired by introducing starting air at a higher pressure than the pressure on the front side of the piston. A movement of approximately 1/2 inch of the piston allows the air charge in the annular chamber to act on the piston head thus accelerating the piston. As the piston moves, the air in front of the piston is compressed. The pressure rises to a value such that the force due to the pressure on the front side of the orifice pin piston is greater than the opposing force due to the pressure on the back side of the piston. This piston will move back causing the air to discharge, both from the main cylinder and from the end orifice. The end orifice is to control the rate of movement and serve as a damper for the orifice pin. It is anticipated that the actual design of the end orifice will depend on experimental developments.

The variable orifice pin will thus control the pressure on the front side of the main piston and thereby control the pressure pulse. Once the cylinder piston moves beyond the pressure outlet tube, the cylinder is closed and a back pressure is built up to cushion the piston at the end of its travel.

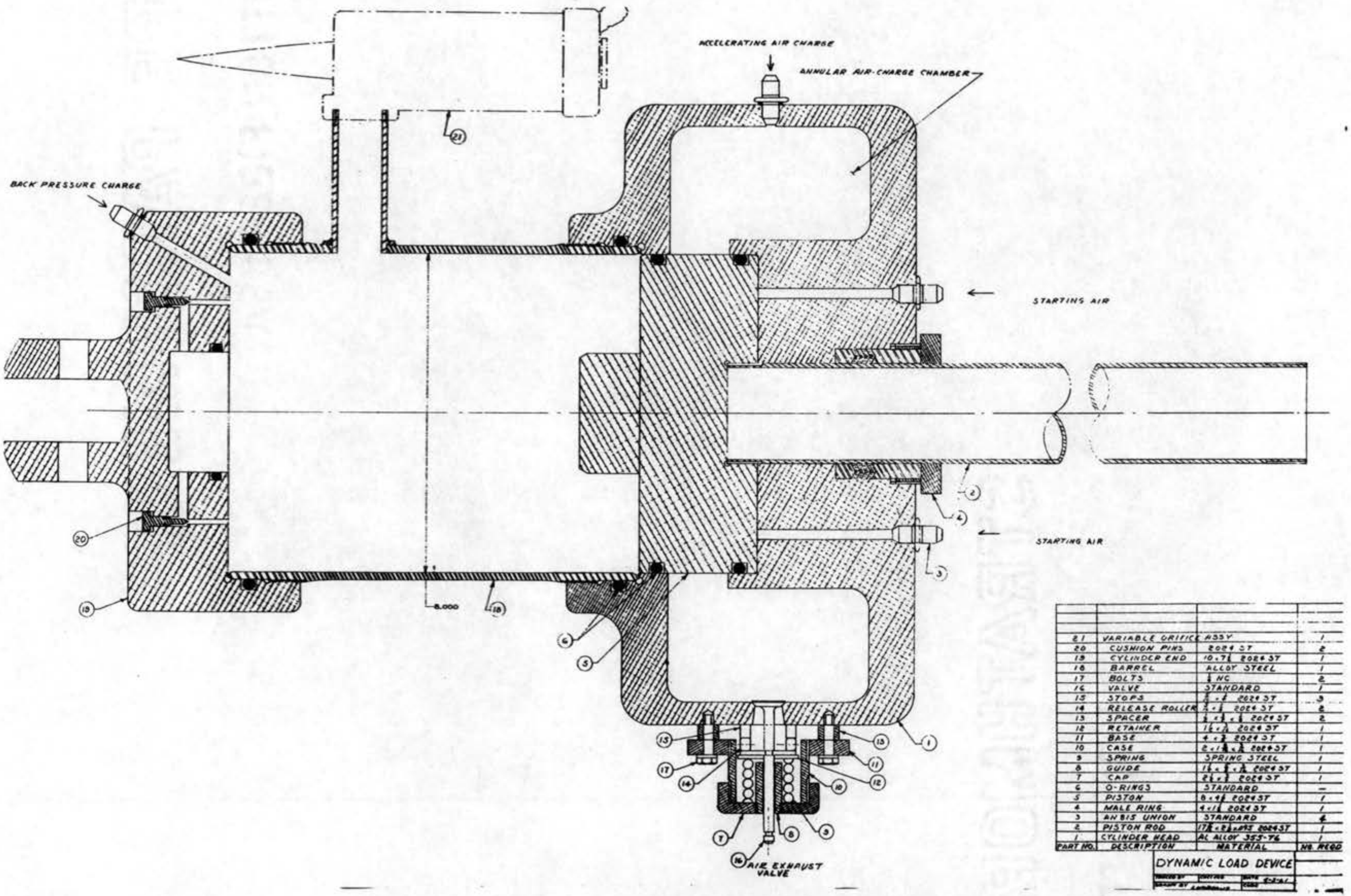
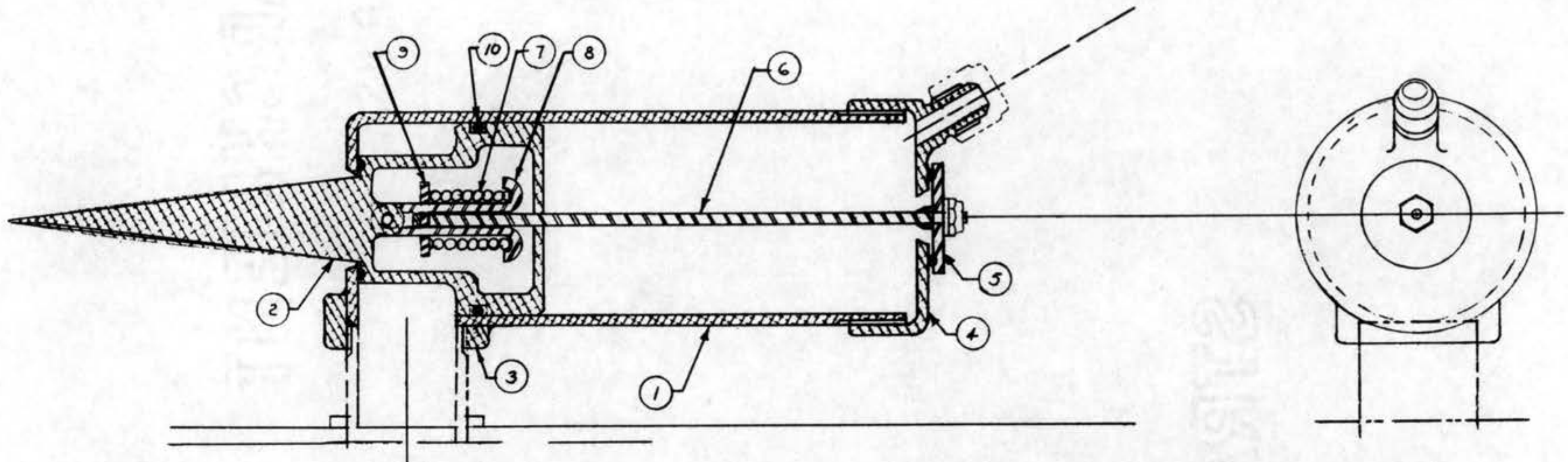


Figure 12. The Dynamic Load Device



PART NO.	DESCRIPTION	MATERIAL	NO. REQD
10	O- RING	STANDARD	3
9	RETAINER	$\frac{3}{8}$ x.10 STEEL	1
8	SPRING CLIP	.095 STEEL WIRE	1
7	SPRING	.109 SPRING STEEL	1
6	ORIFICE SHAFT	.125 STEEL	1
5	END ORIFICE CAP	1.25 x.125 STEEL	1
4	END CAP	2.7 x 1.625 2024 ST	1
3	DOUBLER	.625 x 1.9 6061 ST	1
2	PISTON	6.25 x 2.2 2024 ST	1
1	BARREL	2.5 x .15 6061 ST	1

VARIABLE AREA ORIFICE

TRACED BY	CHECKED	DATE 4-3-61
DRAWN BY LOMBRANO		

Figure 13. Variable Area Orifice

THERMODYNAMIC COMPUTATIONS FOR SIZING REQUIREMENTS

From mechanics, the pressure (P) is

$$P = \frac{F}{A} \tag{5-1}$$

Since the expansion process is quite rapid, the assumption is made that it is an adiabatic process. The compression process on the front side is also assumed to be adiabatic even though there is a mass loss through the orifice.

The fundamental thermodynamic relationship is

$$P\ V^{k} = C \tag{5-2}$$

with an assumed value of k = 1.4

then from Equation 5-2

$$P_1 = P_2\left(\frac{V_2}{V_1}\right)^{1.4} \tag{5-3}$$

The initial volume (V_1) is equal to the volume of the air charge chamber plus the volume in the cylinder swept by the piston prior to the instant the piston is subjected to the accelerating air charge. The initial volume will arbitrarily be sized so that the ratio of the initial volume to the final volume is 1:3. If both sides of Equation 5-3 are multiplied by the area (A), and the pressure is expressed in terms of force (F) the relationship of 5-3 reduces to

$$F_1 = F_2\left(\frac{V_2}{V_1}\right)^{1.4} \tag{5-4}$$

The following conditions are further assumed:

1. The weight of a single seat is 20 lbs. and the seat occupant weight is 170 lbs.
2. Friction is negligible.
3. For preliminary sizing computations the gas pressure is

assumed to be 650 psi.

Corresponding to a force factor of 30 g's, the final force should be:

$$F_2 = (30)(190) = 5700 \text{ lbs.}$$

Then by Equation 5-4, the initial thrust force is:

$$F_1 = (5700)(3)^{1.4} = 26{,}500 \text{ lbs.}$$

The necessary piston cross-sectional area is:

$$A = \frac{F_1}{P_1} = \frac{26{,}500}{650} = 40.7 \text{ sq. in.}$$

which corresponds to a piston diameter of 7.2 inches.

If an 8 inch diameter is arbitrarily selected, then the required pressure and force are determined and corrected to account for atmospheric force on the front face of the piston at the end of the effective stroke. The final force is

$$F_2 = 5700 + [(14.7)(50.24)] = 6438 \text{ lbs.}$$

Then the initial force will be.

$$F_1 = (6438)(3)^{1.4} = 30{,}000 \text{ lbs.}$$

For a pull type design, a correction has to be made for the decreased piston area caused by the piston rod. If a 2-1/2 inch diameter rod is assumed, the pressure will be

$$P_2 = \frac{30{,}000}{(.7854)[(8)^2-(2.5)^2]} = 663 \text{ psia}$$

CALCULATIONS FOR VARIABLE AREA ORIFICE

The procedure followed is as follows:

1. Determine the displacement from Equation A-4 in the Appendix.
2. Since the end force and displacement are known, a curve may be drawn to show the relationships between thrust and dis-

placement. The actual curve is plotted by first establishing the final conditions then solving for the preceding positions. This computation is shown in the Appendix. The displacement time relationships are tabulated in Table I. Then the corresponding thrust requirements are listed in Table II.

3. The thrust versus displacement curve is shown on the Operational Envelope as Figure 14. For convenience, the corresponding thrust-time values are tabulated in Table III in the Appendix.

4. From Figure 11, the values of the force factors (g) per time period (τ) are determined. These values are listed in Table IV in the Appendix.

5. The retarding force (F_d) necessary to give the desired deceleration is found from a summation of forces on the piston.

$$F_d = F_b - F_f \tag{5-5}$$

where

F_b = force on back side of the piston

F_f = force on front side of the piston

Substituting for pressure (P) and area (A), Equation 5-5 becomes

$$P_f A_f = P_b A_b - F_d \tag{5-6}$$

The corresponding back pressures are calculated and tabulated in Table V in the Appendix. The back pressures values are shown on Figure 14 to complete the Theoretical Operational Envelope. The resultant load versus time curve is shown as Figure 15. The zero displacement corresponds to the piston position when the charge gas acts on the piston.

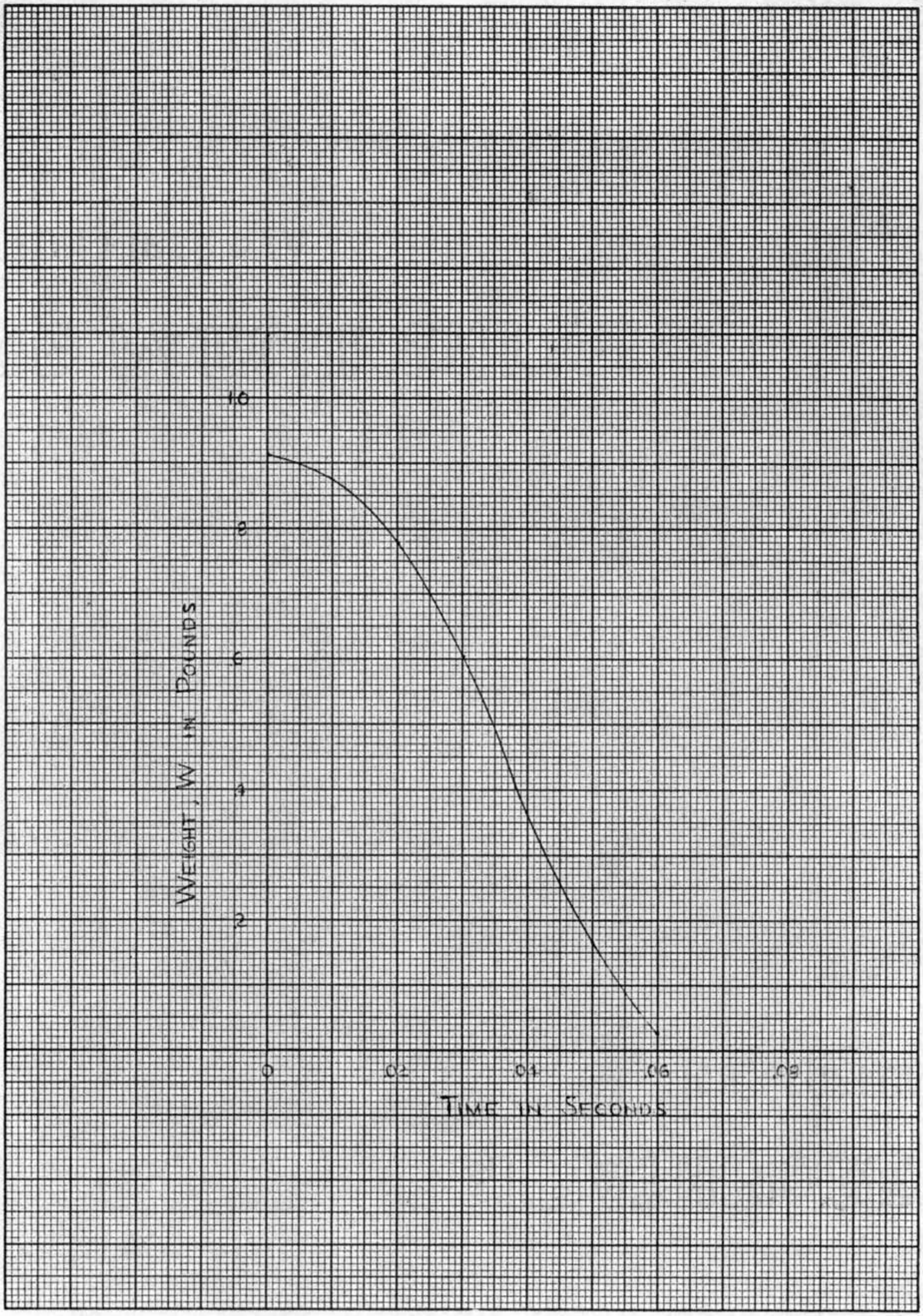


Figure 14. Operational Envelope

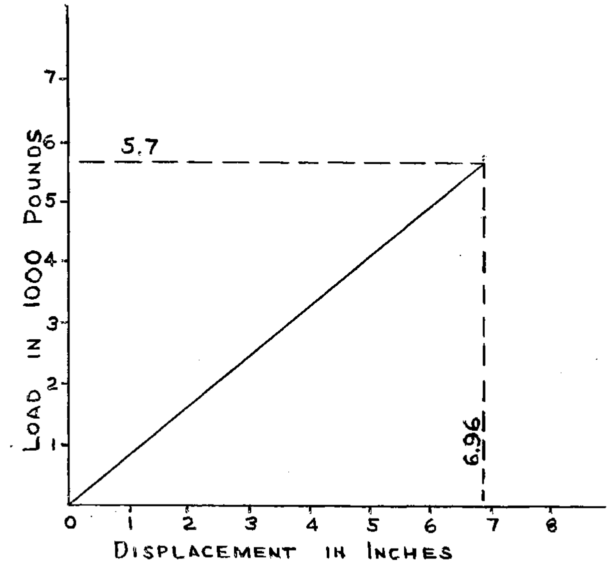


Figure 15. Load-Displacement Curve

6. From the relationship

$$T_{f_2} = T_{f_1} \left(\frac{P_{f_2}}{P_{f_1}}\right)^{\frac{k-1}{k}}$$

and assuming T_{f_1} = 530°R, the corresponding temperature for each pressure (P_f) is determined. These values for pressure and temperature are shown in Table V in the Appendix. The corresponding values for the critical pressure (P_{fc}) and critical temperature (T_{fc}) are also tabulated in Table V.

7. From the thermodynamic relationship for a perfect gas

$$P\ V = WRT \tag{5-7}$$

with R = 53.3

the weight of the air in the front end of the cylinder may be found for each time period (τ). Using the values of pressure and temperature from Table V, the volume may be found as follows:

Assume the maximum stroke of the piston is 10 inches and the piston moves 1/2 inch prior to the instant the air charge acts on the back of the piston. Since the effective piston stroke is 6.96 inches, the volume in the cylinder at the time the piston reaches the end of its stroke is

$$V_c = [10 - (6.96 + .50)](50.24) = 127.61 \text{ cu. in.}$$

Assume the volume of the discharge tube plus the volume in the orifice chamber is equal to 40 cu. in. The volume swept by the piston at any time (τ) is ΔV_f = Area of front of the piston times the stroke. Thus,

$$\Delta V_f = (A_f)(L_\tau) = 50.24\ (L_\tau)$$

The volume (V_f at any time (τ) is therefore

$$V_f = (V_c + 40) + (6.96)(50.24) - \Delta V_f$$

which reduces to

$$V_f = 517.61 - \Delta V_f \qquad (5\text{-}8)$$

Values of V_f are tabulated in Table VI of the Appendix.

Solving for the air weight W from Equation 5-7

$$W = \frac{P_f V_f}{53.3(T_f)} \qquad (5\text{-}9)$$

Values of W are tabulated in Table VII in the Appendix. The corresponding weight versus time curve is shown in Figure 16.

8. The slope $\frac{dw}{dt}$ of the weight versus time curve in Figure 16 is obtained by graphical differentiation at .01 sec time intervals. Values for $\frac{dw}{dt}$ are tabulated in Table VIII in the Appendix.

9. Using the Equation A-9 with an assumed value of C_p = .24, the required orifice area may be determined at any time (τ). The values are tabulated in Table IX in the Appendix.

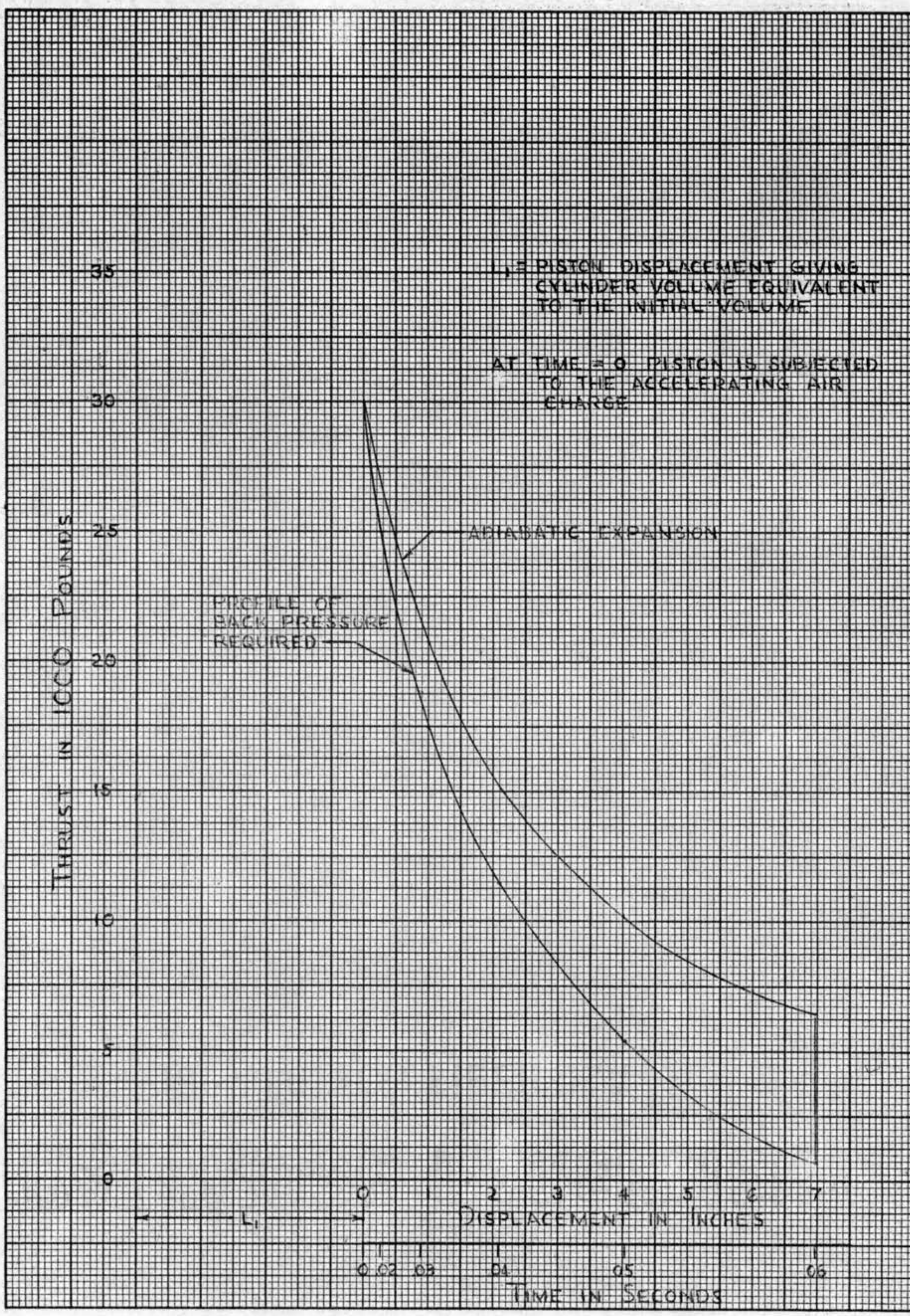


Figure 16. Cylinder Air Weight vs Time

10. From the equation for the area of a circle, the largest diameter required corresponds to the area of 17.15 sq. in. The

$$d = \sqrt{\frac{(4)\ (A)}{\pi}} = \sqrt{\frac{(4)\ (17.15)}{\pi}} = 4.67 \text{ inches}$$

The value of d = 4.67 inches is assumed to be the initial size for the orifice.

11. The pin diameters at positions corresponding to the various time intervals may be computed from

$$d_{pin} = \sqrt{(4.67)^2 - (d_{req'd})^2}$$

Values for d_{pin} are shown in Table X in the Appendix.

CALCULATIONS FOR THE EXHAUST PORT SIZING

The calculations for exhaust port sizing as presented here are in the form of a preliminary analysis since they are based on the initial flow rate. The final design of the exhaust port should be developed in a manner similar to Lichty's (14) analysis for a release (exhaust) process.

The accelerating air charge behind the piston expands from the initial pressure of 663 psia to (P_2) at the end of the effective stroke. The pressure (P_2) may be found from Equation A-5.

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^k = (663)\ (1/3)^{1.4} = 142.5 \text{ psia}$$

Assume the cylinder evacuation time is 10 milliseconds. The volume of the chamber is

$$\text{Vol.} = \text{Vol}_{initial} + \text{Vol}_{swept} = 157.3 + (45.3)(6.96) = 471.5 \text{ in}^3$$

The temperature is:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = (530)\left(\frac{142.5}{663}\right)^{.286} = 341.5^{\circ}R$$

The critical pressure is:

$$P_c = (P_2)\ (.528) = (142.5)\ (.528) = 75.3 \text{ psia}$$

The critical temperature is:

$$T_c = T_2 \left(\frac{P_c}{P_2}\right)^{\frac{k-1}{k}} = (341.5) \left(\frac{75.3}{142.5}\right)^{.286} = 284.5^{\circ}R$$

From the Perfect Gas Equation

$$P\ V = W\ R\ T$$

the specific volume may be found from

$$\frac{V}{W} = \frac{R\ T}{P} = \frac{(53.3)\ (341.5)}{(142.5)\ (144)} = .886 \text{ cu ft per lb}$$

The critical specific volume is

$$V_c = V_2 \left(\frac{P_c}{P_2}\right)^{-\frac{1}{k}} = .886 \left(\frac{75.3}{142.5}\right)^{-\frac{1}{1.4}} = 1.40 \text{ cu ft per lb}$$

The air weight is

$$W = \frac{P\ V}{R\ T} = \frac{(142.5)\ (144)\ 471.5}{(53.3)\ (341.5)\ (1728)} = .308 \text{ lb}$$

The weight flow per unit time is

$$W = \frac{.308}{.01} = 30.8 \text{ lb per sec}$$

By Equation A-9

$$A = 30.8\ (1.4) \left(\frac{1}{\sqrt{2gJC_pT_1 \left[1 - \left(\frac{P_c}{P_1}\right)^{\frac{k-1}{k}}\right]}}\right)$$

$$= (30.8)\ (1.4)\ (.001205) = .052 \text{ sq ft} = 7.48 \text{ sq in}$$

In order to reduce the gas force acting on the valve, four ports will be used, the sum of the areas to be equal to the total area required. The area of each port is thus

$$A = \frac{7.48}{4} = 1.87 \text{ sq in}$$

The force on each valve is then

$$F_v = (142.5)\ (1.87) = 266 \text{ lbs}$$

Solving for the corresponding diameter of the orifice

$$d = \sqrt{\frac{1.87}{\pi/4}} = 2.38 = 1.545 \text{ inches}$$

MACHINE DESIGN COMPUTATIONS

The required wall thickness (t) is determined as follows: Assuming a thin wall cylinder, the hoop stress (s) is given by

$$s = \frac{Pd}{2t} \qquad (5\text{-}10)$$

where

P = pressure in the cylinder, psi

d = internal diameter of the cylinder, in

t = wall thickness of the cylinder, in

For a typical heat treated aircraft quality steel, use a yield strength of 180,000 psi. With a safety of factor of 4, the design stress (S_d) is

$$S_d = \frac{180,000}{4} = 45,000 \text{ psi}$$

Then, solving for the necessary wall thickness by Equation 5-10

$$t = \frac{Pd}{2S_d} = \frac{(663)\ (8)}{(2)\ (45,000)} = .059 \text{ in}$$

To facilitate machining of the "long" bore the wall thickness will be assumed to be 0.25 inches in this preliminary design.

PISTON ROD DESIGN

The piston rod design is based on Figure 5-2 in the ANC 5 (1942 edition) for the following conditions:

1. Piston rod length equal to 12 inches.
2. Design load equal to 26,500 lbs.
3. End fixity coefficient equal to 1.

The solution is by trial and error. After several trials, a positive margin of safety of .045 is given for a 24 ST Tube with a wall thickness of .083 inches and an outside diameter of 2.5 inches. The low margin of safety is used for the piston rod design since the piston rod is a component of the moving system.

CALCULATIONS FOR CYLINDER HEAD THICKNESS

The cylinder head thickness was calculated assuming the head to be a simply supported circular flat plate with a circular hole. The equation for the thickness is given by Roark (13) as

$$t^2 = \left[\frac{3P}{4mS_d(a^2 - b^2)}\right]\left[a^4(3m+1) + b^4(m-1) - 4ma^2b^2 - 4(m+1)a^2b^2 \ln\frac{a}{b}\right]$$

where

t = head thickness, inches

m = reciprocal of Poisson's ratio

P = pressure in the cylinder, psia

S_t = design stress, psi

a = radius of cylinder, inches

b = radius of cut out for piston rod, inches

Therefore

$$t^2 = \frac{(3)\ (1500)\ (0.3)}{(4)\ (25000)\ (16 - 1.268)}\left[(256)\ (10.99) + (1.6)\ (233) - (4)\ (3.33)\ (16)\ (1.268) - (4)\ (4.33)\ (16)\ (1.268)\cdot \ln\frac{4}{1.125}\right]$$

$t = 1.4$ inches

EFFECT OF PARAMETER CHANGES

As may be seen from Figure 14, any change in the pressure on either side of the piston may be made in order to vary the shape of the pulse wave, likewise any change in the accelerated weight will vary the pulse shape. The more pronounced effect is given by a change of the pressure on the front side of the piston. Should the device be used in another application, there may be a need to redesign the variable orifice since this is the controlling device for obtaining the desired pressure form.

The use of several variable orifices was considered, but no thermodynamic analysis was attempted.

From the design standpoint, the main cylinder should have two or three small openings in place of one large opening. The final design of the device should be made on the basis of at least two orifices. It is anticipated that the final design of the variable orifice would need substantial experimental development.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

This report has discussed the dynamic forces acting on an aircraft during a crash impact. The importance of these forces on the aircraft seat is established on the basis of achieving passenger safety through dynamic testing of the seat. It is recommended that more adequate and up-to-date information on crash forces be gathered and presented in a statistical probability display for better usefulness.

The theoretical design of a pneumatic device for performing dynamic seat tests has been presented along with a discussion of the problem of formulating design criteria. It is recommended that a test device be built and tested, for there is certainly adequate justification for testing seats dynamically.

At this time, seats with energy absorbing devices are coming into use but adequate seat testing is still not being used. A device, as presented here, would give a realistic test for all seats and maybe more lives could be saved.

SELECTED BIBLIOGRAPHY

1. Hasbrook, A. H. "How Biomechanical Investigation and Analysis of Survivable Type Aircraft Accidents Can Save Lives", Aero. Eng. Review, V.16, N.6, June 1957.

2. Military Specification MIL-S-7877, dated 20 August 1956, Seats; Passenger, Cabin Folding.

3. Federal Aviation Agency Technical Standard Order TSO C-39.

4. Preston, G. Merritt and Moser, J. C. "Crash Loads", NACA Conference Report, April 17, 1956.

5. Preston, G. Merritt and Pesman, G. J. "Accelerations in Transport Airplane Crashes", NACA TN 4158, February 1958.

6. Pinkel, I. Irving and Rosenberg, E. G. "Seat Design for Crash Worthiness", NACA Conference Report, April 17, 1956.

7. Acker, L. W.; Black, Dugald O. and Moser, Jacob C. "Accelerations in Fighter-Airplane Crashes", U. S. NACA RM E57G11, November 1957.

8. Eiband, A. Martin; Simpkinson, Scott H.; and Black, Dugald O. "Accelerations and Passenger Harness Loads Measured in Full Scale Light Airplane Crashes", US NACA TN 2991, 1953.

9. Hasbrook, A. Howard. "Design of Passenger Tie-Down", CSDM No. 1, 1956, AV-CIR-44-0-66. Aviation Crash Injury Research of Cornell University.

10. Sorin, A. Benjamin. "The Relative Crash Protection Qualities and Deficiencies of Mil-S-7877 Passenger Seats in Forward and Aft Facing Positions", Paper presented at the Flight Safety Foundation Air Safety Seminar, Taxaco, Mexico November, 1955.

11. Hawkes, Russell. "Air Crash Death or Injury May Be Prevented by Sound Detail Design", Aviation Week, November 5, 1956.

12. Everett, H. A. "Thermodynamics", 2nd Edition, D. Van Nostrand Company, Inc., 1941.

13. Roark, Raymond J. "Formulas For Stress and Strain", 3rd Edition, McGraw-Hill, 1954.

14. Lichty, L. C. "Internal Combustion Engines", McGraw-Hill, 1951.

APPENDIX

DERIVATION OF DISPLACEMENT-TIME RELATIONSHIP

By definition, the velocity (U) is

$$dU = (a)\,dt \tag{A-1}$$

with

a = acceleration

Based on Figure 11, the desired acceleration is

$$a = Kt$$

Substituting in Equation A-1 and integrating

$$U = K\int_0^t t\,dt = \frac{Kt^2}{2} \tag{A-2}$$

By definition the displacement (L) is

$$dL = (U)\,dt \tag{A-3}$$

Substituting for the velocity from Equation A-2 and integrating

$$L = \frac{K}{2}\int_0^t t^2\,dt = \frac{Kt^3}{6} \tag{A-4}$$

Values for piston displacement versus time are shown in Table I. The computations were based on a value for K = 500 g.

TABLE I

DISPLACEMENT VERSUS TIME

L ft	L in	τ sec
0.0027	0.032	0.01
0.0215	0.258	0.02
0.072	0.869	0.03
0.172	2.067	0.04
0.336	4.03	0.05
0.58	6.96	0.06

THERMODYNAMIC RELATIONSHIPS

The applicable basic thermodynamic relationships for a Perfect Gas are

$$P_1V_1^k = P_2V_2^k \qquad \text{(A-5)}$$

$$P_1 = P_2\left(\frac{V_2}{V_1}\right)^k \qquad \text{(A-6)}$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \qquad \text{(A-7)}$$

For a volume ratio $(V_2/V_1) = 3)$ and with a stroke $(L = 6.96)$ the volume ratio in terms of piston stroke is

$$\frac{V_2}{V_1} = \frac{X + 6.96}{X} = 3$$

where

X = piston stroke corresponding to equivalent initial volume.

Solving

$X = 3.48$ inches

The volume for unit length of stroke is

$$V = \frac{(.7854)\ (D^2-d^2)}{1} = \frac{(.7854)\left[(8)^2 - (2.5)^2\right]}{1} = 45.3\ \text{in}^3$$

The general expression for volume is given by

$$V = V_1 + (45.3)L \qquad \text{(A-8)}$$

The initial volume, which must be sized into the design, is

$$V_1 = (45.3)(3.48) = 157.5\ \text{in}^3$$

Solving Equation A-8 for various displacements

$L = 1.00$, $V = 202.8\ \text{in}^3$

$L = 2.00$, $V = 248.1\ \text{in}^3$

$L = 3.00$, $V = 293.4\ \text{in}^3$

$L = 4.00$, $V = 338.2\ in^3$

$L = 5.00$, $V = 384.0\ in^3$

$L = 6.00$, $V = 429.3\ in^3$

$L = 6.96$, $V = 471.5\ in^3$

Other computations based on the basic thermodynamic relationships are shown in Tables II, III, IV, V, VI, VII, VIII.

TABLE II

PRESSURE AND THRUST VERSUS DISPLACEMENT BY EQUATION A-6

L in	P_b psi	F_b lb
1.00	462.0	20,900
2.00	349.0	15,800
3.00	276.0	12,500
4.00	226.5	10,250
5.00	189.5	8,580
6.00	161.8	7,320
6.96	142.5	6,438

TABLE III

THRUST FOR VARIOUS TIMES (τ)

τ sec	F_b lb
0.00	30,000
0.01	29,800
0.02	26,800
0.03	21,700
0.04	15,250
0.05	10,200
0.06	6,438

TABLE IV

FORCE FACTOR (g) REQUIRED PER TIME PERIOD (τ)

τ,sec	F (g)	$F_d = [F\ (g)] \times [\text{weight } (190 \text{ lbs})]$
0.00	0	0
0.01	5	950
0.02	10	1900
0.03	15	2850
0.04	20	3800
0.05	25	4750
0.06	30	5700

TABLE V

THRUST (F_f) AND PRESSURE (P_f) FROM EQUATION 5-5 AND 5-6

τ sec	F_f lb	P_f psia	P_{fc}	T_f °R	T_{fc} °R
0.00	30,000	598.0	316.0	530	520.0
0.01	28,850	574.0	303.0	524	514.0
0.02	24,900	496.0	262.0	503	493.0
0.03	18,850	375.5	198.5	464	456.0
0.04	11,450	228.0	120.5	403	396.0
0.05	5,450	108.5	57.4	326	320.0
0.06	738	14.7	14.7	184	180.5

TABLE VI

CYLINDER VOLUME IN FRONT OF PISTON AT TIME (τ)
BY EQUATION 5-8

τ sec	L in	ΔV_f in^3	V_f in^3	V_f in^3
0.00	0.000	0.00	517.61	.2995
0.01	0.032	1.61	516.00	.2950
0.02	0.258	12.98	504.63	.2920
0.03	0.869	43.60	474.01	.2750
0.04	2.067	104.00	413.61	.2390
0.05	4.030	202.50	315.11	.1825
0.06	6.960	350.00	167.61	.0969

TABLE VII

AIR WEIGHT AT TIME (τ) BY EQUATION 5-9

τ sec	W lb	V_1 ft^3/lb
0.00	0.913	0.329
0.01	0.875	0.337
0.02	0.779	0.375
0.03	0.601	0.457
0.04	0.366	0.654
0.05	0.164	1.114
0.06	0.021	4.600

TABLE VIII

WEIGHT FLOW RATE AT TIME (τ)

τ sec	$\frac{dW}{dt} = \frac{lb}{sec}$
0.01	7.33
0.02	15.00
0.03	23.10
0.04	25.90
0.05	18.30
0.06	10.00

DERIVATION OF THE EQUATION FOR THE VARIABLE AREA ORIFICE

From the general energy equation, the velocity in the orifice is given by

$$U^2 = 2gJ\ (h_1 - h_2) + U_1^2$$

Assuming the air corresponds to a Perfect Gas the change in enthalpy is

$$h_1 - h_2 = C_p(\Delta T)$$

Since the initial velocity is small it may be neglected. Then the equation for the velocity reduces to

$$U = \sqrt{2gJC_p(T_1 - T_c)}$$

Which can be expressed in terms of pressure as

$$U = \sqrt{2gJC_pT_1\left(1 - \frac{P_c}{P_1}\right)^{\frac{k-1}{k}}}$$

The weight rate of flow $\frac{dW}{dt}$ is

$$\frac{dW}{dt} = \frac{UA}{v_c} \quad \text{with } v = \text{specific volume cu ft per lb}$$

Substituting for v_c from the pressure relationship

$$v_c = v_1\left(\frac{P_c}{P_1}\right)^{-\frac{1}{k}}$$

and solving for the area A

$$A = \frac{\left(\frac{dW}{dt}\right) v_1 \left(\frac{P_c}{P_1}\right)^{-\frac{1}{k}}}{U}$$

Then substituting for the velocity, the expression for area becomes

$$A = \frac{dW}{dt}\, v_1\left(\frac{P_c}{P_1}\right)^{-\frac{1}{k}}\left[\frac{1}{\sqrt{2gJC_pT_1\left[1-\left(\frac{P_c}{P_1}\right)^{\frac{k-1}{k}}\right]}}\right] \qquad \text{(A-9)}$$

Computations for the required orifice area at various time intervals are shown in Table IX. The values for the diameter of the pin are shown in Table X.

TABLE IX

REQUIRED ORIFICE AREA

τ sec	$\frac{dW}{dt}$ lb/sec	T_1 ^oR	$\left(\frac{P_c}{P_1}\right)^{-1/k}$	$\left(\frac{P_c}{P_1}\right)^{\frac{k-1}{k}}$	V_1 ft^3/sec	A ft^2
0.01	7.33	524	1.579	-.833	0.337	0.00381
0.02	15.00	503	"	"	0.375	0.00880
0.03	23.10	464	"	"	0.457	0.01725
0.04	25.90	403	"	"	0.654	0.02970
0.05	18.30	326	"	"	1.114	0.03980
0.06	10.00	184	1.579	-.833	4.600	0.11950

TABLE X

CROSS-SECTIONAL DIA. OF PIN FOR VARIABLE AREA ORIFICE

τ sec	$(d_{reqd})^2$ in^4	$21.85-(d_{reqd})^2$ in^4	d_{pin} in
0.01	0.70	21.15	4.60
0.02	1.62	20.23	4.49
0.03	3.19	18.66	4.40
0.04	5.42	16.43	4.06
0.05	7.31	14.54	3.81
0.06	21.85	0.00	0.00

DETERMINATION OF AIR CHARGE IN FRONT OF PISTON

Since the piston moves 1/2 in before being subjected to the accelerating air charge, the determination of the initial pressure in front of the piston must consider the slight pressure rise due to the piston travel. From Figure 15 at $\tau = 0$, $F_d = 0$.

Equation (5-6) becomes

$$P_f \, A_f = P_b \, A_b$$

then

$$P_f = P_b \, \frac{A_b}{A_f} = (663) \left(\frac{45.3}{50.24}\right) = 598 \text{ psia}$$

Initially with the piston in the full back position the pressure (P_{f1}) is

$$P_{f1} = P_f \left(\frac{V_f}{V_{f1}}\right)^k$$

From Table VI at $\tau = 0$, $V_f = 517.61$ in^3. The volume swept by the piston moving 1/2 in is $V = 50.24 \left(\frac{1}{2}\right) = 25.12$ in^3. Then

$$V_{f1} = 517.61 + 25.12 - 542.73 \text{ in}^3$$

therefore

$$P_{f1} = (598) \left(\frac{517.61}{542.73}\right)^{1.4} = 562 \text{ psia}$$

VITA

JOSEPH ROBERT LOMBRANO

Candidate for the Degree of

Master of Science

Thesis: A DYNAMIC LOADING DEVICE FOR AIRCRAFT SEATS

Major Field: Mechanical Engineering

Biographical:

Personal Data: Born in Guerrero, Coah., Mexico, November 1, 1923, the son of Nieves and Mary F. Lombrano.

Education: Attended grade school in Kansas City, Kansas; graduated from Wyandotte High School in 1941; attended two years of pre-engineering at Kansas City Kansas Junior College; received the Bachelor of Science degree from the University of Kansas, with a major in Aeronautical Engineering, in June 1948; received the Bachelor of Science degree from the University of Kansas, with a major in Mechanical Engineering, in June 1949; completed the requirements for the Master of Science degree in August 1961.

Experience: Entered the United States Navy in 1943 and was commissioned in 1944 and served as general deck officer, assistant gunnery officer and communications officer. Employed by Wyandotte County Engineers office as assistant engineer. Employed by Universal Construction Co., Inc. of Kansas City, Kansas as layout and field engineer. Presently employed by Douglas Aircraft Company as a design engineer.

Professional Organizations: Member of American Society of Mechanical Engineers. Member of American Society of Professional Engineers. Licensed Engineer in State of Kansas.