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ENGINEERING ANALYSIS AND DESIGN OF A MECHANISM TO SIMULATE A SONIC BOOM

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SUMMARY

A mechanism to simulate the vibrational and acoustical properties of a sonic boom is designed. The simulator reproduces the effects of sonic booms having N-wave shape with rise times as low as ten milliseconds, durations as short as 161 milliseconds, and peak overpressures as high as three pounds per square foot.

A systematic engineering analysis is performed to establish the best simulator design. Numerous different mechanisms are carefully studied. Each design is examined to ascertain its ability to generate the properties of a sonic boom, its ease of adjustment, and its lack of background noise.

The final design chosen employs a moving circular diaphram which creates the required pressure variations by altering the volume of an air tight chamber connected to an acoustical testing room. The diaphram is designed such that a minimum of force is required for its movement. The movement of the diaphram is controlled by the rotation of a specially designed adjustable cam, and a constant force air cylinder.

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INTRODUCTION

With the advent of commercial supersonic transports rapidly approaching, the question is being raised, "What will be the effect on the population of supersonic aircraft flying repeatedly overhead?" That is, will the noise and subsequent vibrations caused by sonic booms have any physiological effect on man?

Even though there is much research being done on the effect of noise and vibration on individuals, there exists only limited useable data on the effects of sonic booms on man. This absence is the result of insufficient amounts of data being recorded in a controlled acoustical testing environment, together with an adequate, controllable sonic boom simulator. There exists at North Carolina State University one of the finest acoustical testing rooms available in this area for noise and vibration studies. However, a machine is needed that can adequately simulate both the acoustical properties, but more importantly, the vibrational properties of a sonic boom. The purpose of this research was to design such a sonic boom simulator.

REVIEW OF LITERATURE

There have been several sonic boom study programs conducted throughout the past decade. The sonic booms for these studies were generated by aircraft in supersonic flight. The most recent studies were conducted at Oklahoma City (1965) and Edwards Air Force Base (1967). (3,4) Studies of this nature contributed valuable data about the reaction of people to sonic booms. However, they were very costly and the sonic booms were not easily controlled.

These shortcomings gave rise to laboratory simulation of sonic booms. In the laboratory there are generally three basic approaches used for controllable sonic boom simulators. They are the headset method, the progressive wave method, and the chamber vibration method. Initial sonic boom simulators used the headset method. These machines used electronic speakers mounted inside an acoustical chamber. Studies like those of Pearson and Kryter (7) in 1965 gave realistic indoor acoustic simulation but lacked vibration stimulus. Similarly, the progressive wave method, which incorporates an explosive source and a magnifying expansion tube, gives a realistic simulation of an outdoor sonic boom. However, studies of this type, like those of Dahlke, et al., (1) in 1968, create only ground shock waves, and therefore also fail to create the disturbing vibrations so often associated with sonic booms. The final method, chamber vibration, is the most exact method of simulating a sonic boom. This creates indoor acoustical and vibrational simulation through the use of two separate vibrating devices or through a single complex machine. The latter type of sonic boom simulator has been constructed (1968) at Stanford Research Institute under a grant from the National

Aeronautics and Space Administration. (5,6) This project has contributed much information about the design, construction, and use of such a sonic boom simulator.

THE DESIGN PROCESS

General Discussion

Before beginning the presentation of the sonic boom simulator design, a review of the design process will be helpful to the reader. The design process is not a rigid or fixed series of steps which are always followed, but varies with individual requirements. However, a logical sequence of steps, usually common to all design projects, does exist. These steps, in chronological order, are listed below:

a. problem definition

b. feasibility study

c. preliminary design

d. detail design

e. development.

These five steps are interrelated and interdependent, each reflecting and effecting all the others.

During the first step, the particular need is recognized and a clear, but general, statement is made of the mechanism function. This process defines the problem and states assumptions, economic and technological constraints, parameters, and design criteria. Next, numerous different schemes are considered for solving the problem. This process discards the inadequate designs, leaving an elite group of designs which satisfy the need and are feasible. During the preliminary design step, each of these feasible designs are considered in more depth. Broad overall figures are established and initial specifications are made for the major components of each design. An analysis is made of the strength and deformation of the major components. Advantages and disadvantages of each feasible design are determined. Each design is then weighed, one against the other, until a preliminary design is chosen.

Once this design is chosen, the fourth step of the design process begins. During the detail design step, a complete engineering description is provided for every component of the mechanism. Tolerances are stated, assembly drawings are completed, and the machine is constructed. Finally, in the development step of the design process, the machine is tested and evaluated. If specifications are not met, the machine is redesigned and improved.

This general design process is followed throughout this thesis. The last step presented is the detailed design of the simulator. Construction, testing, and development is left to further work on the project.

Problem Definition

The need for an adequate sonic boom simulator has been acknowledged in the introduction. The simulator must reproduce a pressure wave that is similar to the type wave that would be recorded at ground level, if created by a supersonic aircraft.

Understanding the basic phenomenon of a sonic boom is essential before attempting to design a mechanism that will create the same results. The occurrence of sonic booms can be described with the aid of Figure 1 as shown (8).

Sound is a pressure disturbance in the air which is heard by the ear. A stationary point soarce emits sound waves, alternating compressions and refractions moving at the speed of sound, in the form of concentric spheres. However, when a soarce is moving, these spheres are no longer concentric. The waves are closer together ahead of the soarce and farther apart to the rear. When this source, or airplane in this case, is moving at or faster than the speed of sound, the waves coalesce to form a shock wave. The sonic boom disturbance generated by a supersonic aircraft is in the form of a traveling shock wave, moving at the speed of the airplane. Generally speaking, this boom is in the form of a sudden increase of pressure, P_o, which slowly decays linearly to a value of negative P_o during a period e, then followed by a similar pressure increase of equal magnitude P_o. So, under ideal conditions the wave can be described by an N-shaped impulse. If the shock wave pattern generated by an airplane could be made visible, it might appear as shown in Figure 1.

The N-wave representation of a sonic boom is an idealized one. Many variables can, and usually do, distort the N-shape wave. The actual

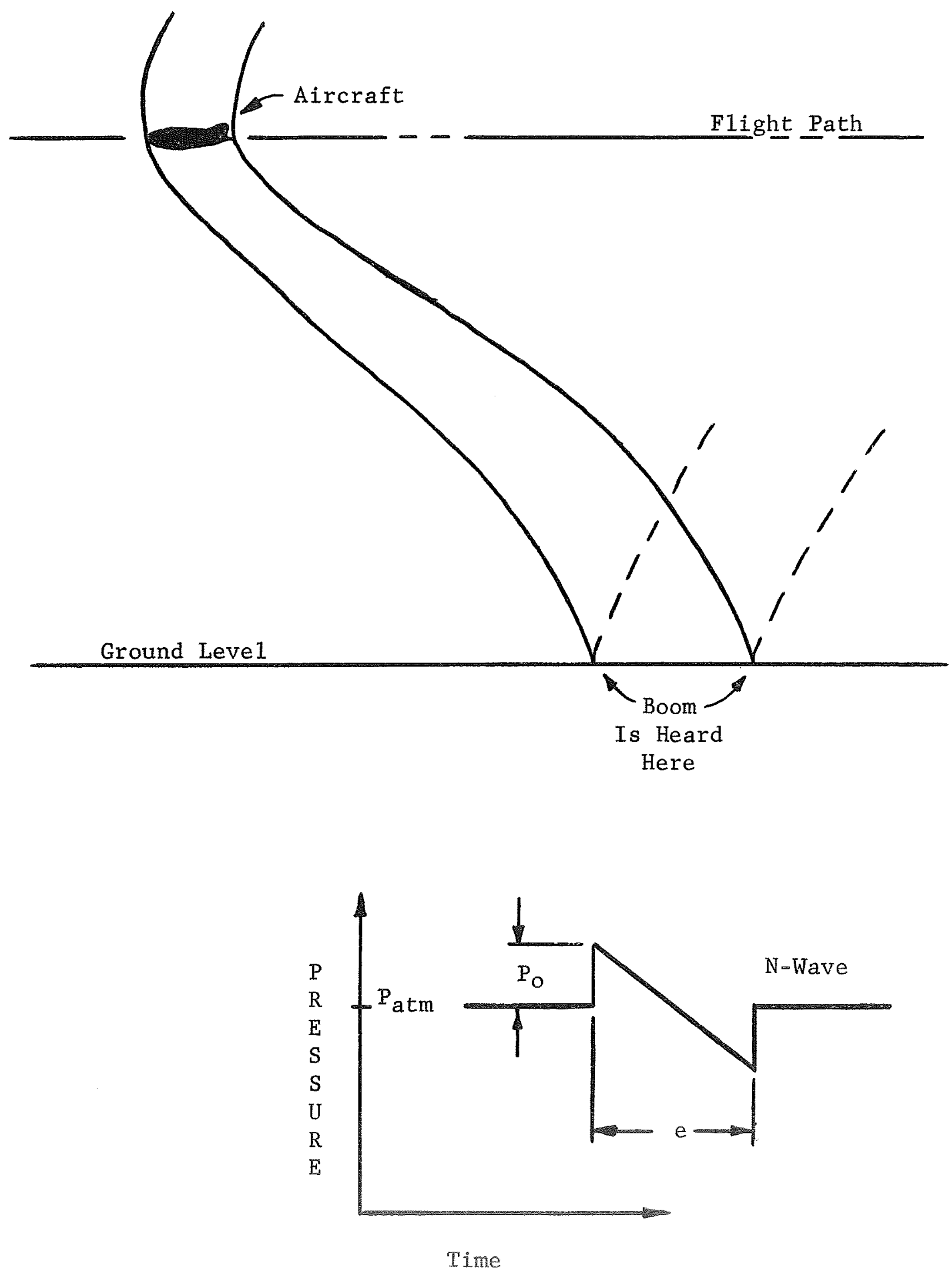


Figure 1. Sonic boom phenomenon

shape of a sonic boom generally depends on three factors, atmospheric conditions, aircraft design, and ground terrain. Figure 2 shows several examples of actual sonic boom shapes.

No matter what the actual shape of the sonic boom pressure wave, it can be described by three parameters, rise time, duration, and peak overpressure. The value of each of these parameters can vary from one boom to another over a fairly large range. The most recent sonic boom data recorded using people and dwellings was done at Edwards Air Force Base. (4) The range of their data was approximately as shown:

rise time	.005 - .010	seconds
duration	.079 - .277	seconds
peak overpressure	.75 - 2.8	pounds per square foot

These values are typical of supersonic aircraft and encompass the predicted values for supersonic transports. The sonic boom simulator, therefore, is designed with the capability to reconstruct booms with parameters approximating this range.

The sonic boom simulator also has to satisfy several additional constraints and design criteria. Firstly, and very importantly, the simulator must be easily adjustable in all three parametric areas. Projected testing procedures indicate the necessity for the simulator to have the capability of quick and easy adjustments. Similarly, the simulator must be capable of repeating itself in a short-time interval. That is, the machine must be able to repeat the same sonic boom effect, over and over again, with a minimum of difficulty. Another very important design criteria is that the simulator must be as quiet as possible. Initial use of the simulator is projected to be sleep-startle studies. Therefore, the simulator should generate as little background noise as possible and

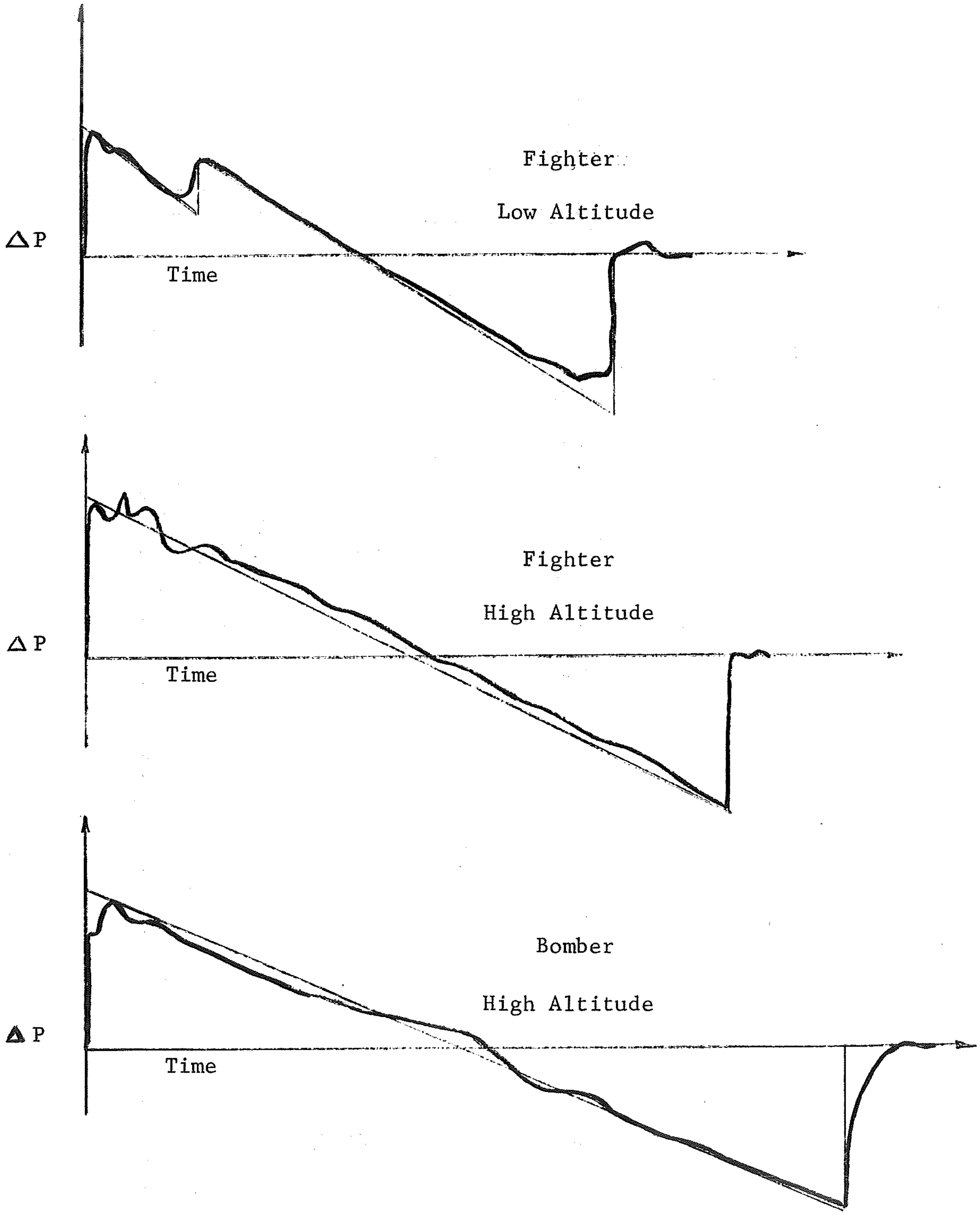


Figure 2. Actual sonic boom shapes

should be designed such that adjustments can be made with a minimum of noise.

This simulator design is constrained in two areas: economics and space. Only a limited amount of money is available for the construction of the sonic boom simulator. Therefore, the design should be as simple and as inexpensive as possible. In addition, the simulator must fit in the area designated for this project. The simulator will be used in conjunction with an acoustical testing room already constructed. This testing facility is situated in a location such that the only available space for the sonic boom simulator to be situated in is a 12-foot section of one side wall as shown in Figure 5.

Ideally, the sonic boom simulator should effect all walls of the testing room. However, the constraints indicate that the 12-foot section of wall is the maximum part of the testing room that can be subjected to a simulated sonic boom. Studies conducted by Lukas (6) indicate that pressure loading only a <u>single</u> wall of a testing room adequately approximates a room in a dwelling under the influence of a sonic boom. The sonic boom simulator is therefore designed to incorporate an air-tight chamber, similar to the one used at Stanford Research Institute, to uniformly load the specified wall section.

Feasibility Study

The constraint dictating the use of an air-tight chamber, or control volume, is a restriction that is helpful to the feasibility study step of the design process. The control volume is constructed so that one side of the chamber is the exterior of the specified testing room wall. The other chamber walls are rigidly constructed. Therefore, any pressure disturbance created within the control volume will effect only the testing room wall. To create the desired N-wave, a pressure change will obviously have to occur within this control volume. While a pressure change can be caused by numerous methods if air is assumed to be an ideal gas, the ideal gas law

$$PV = nR_oT$$

will be applicable. From this equation it is immediately evident that to alter the pressure of this system one must change either the volume, the temperature, or the number or character of the molecules of the system. Each of these three properties of air is examined to determine its feasibility as a single variable to be used to create the desired pressure change.

Temperature is considered first. Using temperature as a variable, it is found that the rise time requirement alone eliminates all temperature-based designs. It is virtually impossible to change the temperature in a linear manner in a time interval of .015 seconds.

Changing the number or character of the molecules is considered next. It is determined that such a change is conceivable by using a chemical reaction, an explosive charge, or an addition of air to the control volume. A chemical reaction poses problems in the areas of linearity, repeatability, and response times. On the basis of these problems, a chemical reaction is eliminated. Using an explosive charge to create pressure changes also creates problems. This type process alters the character of the control volume air. Such changes are not usually linear acting, are not easily controlled, and are difficult to adjust. For these reasons, the explosive charge method is discarded. Lastly, the possibility of adding and extracting air, to cause the N-shaped pressure pulse, is examined. This method shows encouraging possibilities. Through the use of choked-flow nozzles, sufficient air can be added linearly to the system in the required time intervals. However, extracting the air possesses problems. The size of the vaccum chamber required is much too large for the space available. Vaccum pumps could possibly be a solution, but they are unavailable. These facts, together with the possibility of serious valving problems, result in the idea being eliminated from consideration.

By elimination of all designs incorporating either a temperature change or a change in the number of molecules, it is concluded that if an adequate design exists, it must incorporate a volume change to cause the required pressure wave. Any change in volume will have to be caused by moving a section of the control volume's wall. A forcing device of some sort is therefore needed to move this wall section in the prescribed manner. This fact immediately complicates the feasibility study, as there are countless types of forcing devices available. However, it is noted that sonic boom simulators which incorporate volume changes require two rates of wall movement: one rate for the rise time period and one rate for the duration period. This fact is used to remove some of the difficulty by dividing the forcing devices into two groups:

a. A single device, capable of creating both rates of movement

b. Dual devices, one creating the rise time rate and the other the duration rate of movement.

A single forcing device creates and controls a force during both periods. A power screw is such a device. It creates a linear moving force and its rotation controls the force's application. However, a power screw is not applicable for a sonic boom simulator. Adjustability and direction changes are insurmountable problems for a power screw. Moving belts also create and control a force. This type of device has excellent adjustment and constant velocity characteristics. However, a simple constant velocity belt capable of handling the required loads is too large and cannot change directions in the linear manner required. A combination of belts running in opposite directions is also inadequate because of shock loading problems when switching from one belt to another. A rotating device incorporating clutches is considered next. This type of force-producing device is inadequate for a sonic boom simulator because of reaction times and force problems. A cam with positive action in both directions is examined. Such a cam is found to possess inadequate adjustability and excessive acceleration forces. Lastly, a hydraulic device is considered. A dual-action cylinder has adequate force and adjustability characteristics, but valving problems cause the device to be eliminated. It is now evident that if an adequate forcing device exists it must incorporate dual devices. No single device can solve all the design requirements.

Any one of the five types of devices previously mentioned can be combined with each other or additional devices to form the required dual forcing device. Other devices available for only dual systems are springs, electromagnets, and solenoids. Because of force requirements and activation times, only springs are worth considering. Springs have good force and adjustability characteristics, but they are generally not linear acting. Table 1 briefly lists the advantages and disadvantages of all of these devices, including a comparison of how well each device can be activated. The pressure cylinder and the spring both exhibit excellent activation characteristics. During the rise time periods of the N-wave, the ability of start and stop almost instantaneously is necessary. Therefore, one of these two devices should be used for the rise time rate of wall movement. One half of the dual forcing device is now determined.

The main shortcoming of all the single forcing devices is lack of ability to change the direction of wall movement in the prescribed manner. Likewise, many dual systems will encounter the problem of switching from one device to the other. When a spring or pressure cylinder is used, this problem exists to some extent with all the devices except the simple cam. The cam is the only device that lends itself to switching from the spring (cylinder) to the cam, and then back to the spring (cylinder). However, a cam lacks the adjustability required. In order to obtain an adequate system, the cam must be made adjustable. This can be accomplished by incorporating a special adjustable type cam, or an adjustable lever to alter the output of a non-adjustable cam.

All techniques feasible for use in a sonic boom simulator are now identified except for the manner in which the cam will be controlled. The design requirements of repeatability and simplicity, in effect, specify the use of a circular cam. Such a cam is rotation controlled. Therefore, either an A-C or D-C adjustable output motor will be used.

Table 1. Comparison of forcing devices

Device	Main Advantages	Main Disadvantages	Activation Characteristics
Power Screw	Linear Action Forcing Power	Adjustment	Adequate
Moving Belt	Adjustable Linear Action	Switching	Poor
Clutch	Linear Action	Reaction Time	Poor
Cam	Linear Action Forcing Power	Adjustment	Adequate
Pressure Cylinder	Adjustable Forcing Power Linear Action	Valving	Excellent
Spring	Adjustable Forcing Power Linear Action	Non-Linear	Excellent

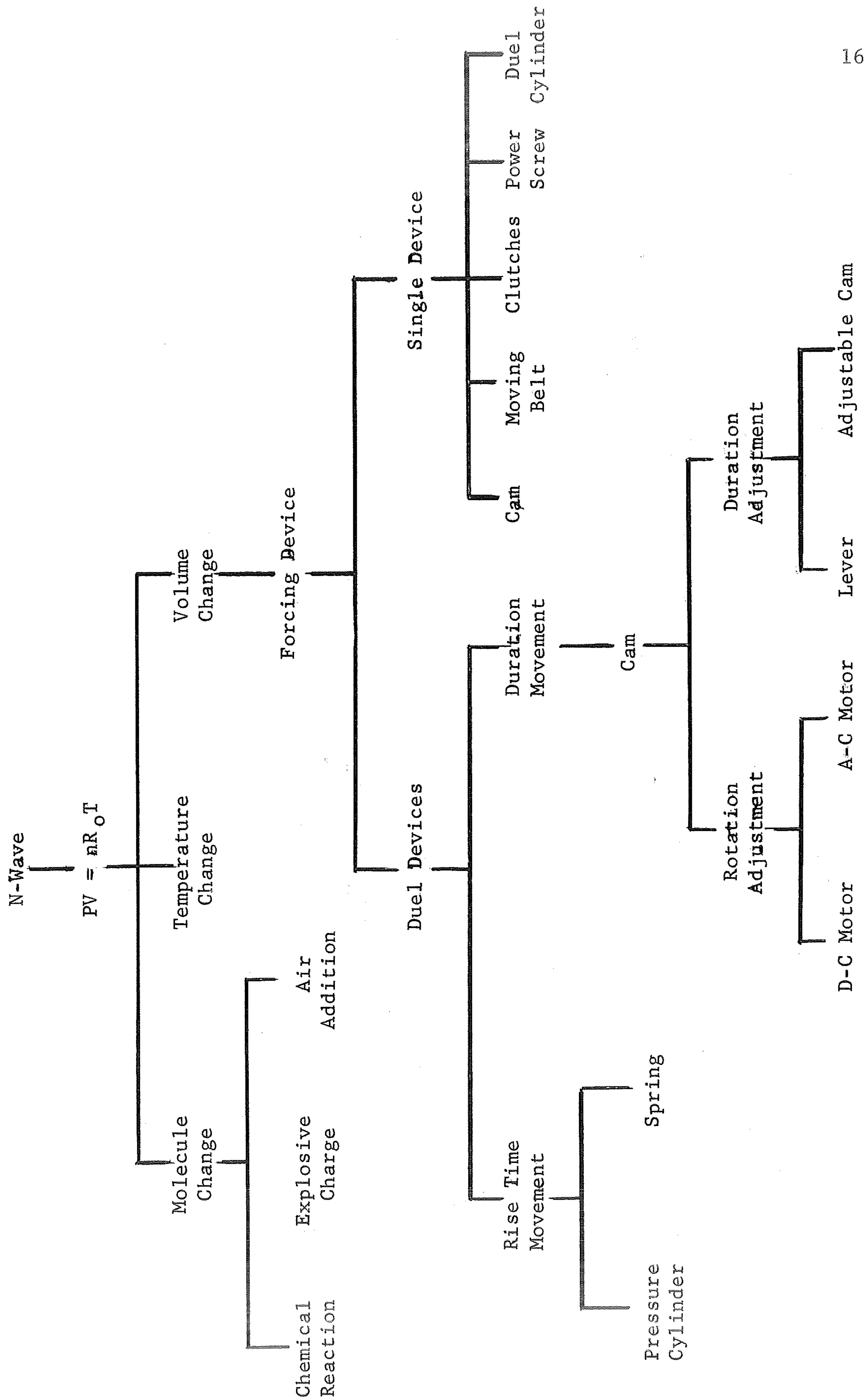


Figure 3. Feasibility study outline

Preliminary Design

The feasibility study determined all the general aspects of the sonic boom simulator design except for three problem areas. Two feasible solutions were presented for each of these problems. The preliminary design phase of the design process analyzes these three pairs of solution and determines the best of each pair. However, before a determination of the best design is begun, certain preliminary dimensions are obtained.

The volume changes required for the simulator takes place in a relatively short interval of time. For this reason, the initial pressure rise process is assumed to be adiabatic. The equation for the resulting pressure rise is

$$\frac{V}{V_{cv}} = {\frac{P_o}{P_{atm}}}^{\frac{1}{k}} \tag{1}$$

where $k \approx 1.4$ for air and $P_{atm} = 2116.2$ p.s.f. The change in volume, ΔV, is caused by the movement of a section of the control volume's wall. This gives

$$\Delta V = A_{ms} \times d \tag{2}$$

substituting into (1) and transposing

$$A_{ms} \times d = V_{cv} \times {\frac{P_o}{P_{atm}}}^{\frac{1}{k}} \tag{3}$$

For simplicity, it is assumed that the moving wall section is a circle of radius R, or

$$A_{ms} = \pi R^2 \tag{4}$$

substituting (4) into (3) and solving for distance gives

$$d = \frac{V_{cv}}{\pi R^2} \times \frac{P_o}{P_{atm}}^{\frac{1}{k}} \qquad (5)$$

For a particular overpressure, all the factors in equation (5) that are constants are grouped and designated K_1. Therefore

$$d = \frac{K_1}{R^2} \qquad (6)$$

Neglecting the force due the pressure increase, the amount of force required to move the circular wall section during the rise time period is determined by Newton's Second Law

$$F = ma \qquad (7)$$

where m = mass of circular wall section and all connected moving parts, and a = acceleration of system. Both a spring and a pressure cylinder are assumed to have constant force characteristics. This is accomplished by using a long, soft spring, or a large reservoir connected to the cylinder. From equation (7) it is easily recognized that for constant force systems, there will be a constant acceleration. A system having constant acceleration cannot have the constant velocity required for an N-wave. However, any error that is present during the rise time periods, when the spring or cylinder are acting, is considered to be insignificant when compared to the entire N-shaped wave. A body with constant acceleration will travel a distance, d, as shown below

$$d = \frac{1}{2}at^2 \qquad (8)$$

where t = rise time of N-wave. For a given rise time, all the constant factors can be grouped and designated $\frac{1}{K_2}$. Solving for acceleration gives

$$a = K_2d \qquad (9)$$

Substituting (6) into (9) gives

$$a = \frac{K_1K_2}{R_2} \qquad (10)$$

The mass of the circular wall section, or circular diaphram, is expressed as

$$m = ZR^2 \qquad (11)$$

The mass of the moving parts associated with the circular diaphram is assumed to be a direct function of the diaphram's mass. Therefore, the total mass that will be moved is

$$m = K_1ZR^2 \qquad (12)$$

where K_3 is a constant.

The circular diaphram will have to be supported by a set of ribs. When the diaphram is suddenly accelerated, the resulting force will cause the diaphram's surface to deflect, and the ribs to bend. Any deflections of this nature will cause error in the amount of volume change. The maximum deflection of such a ribbed circular diaphram is expressed in the form

$$Y_{max} = \frac{qwR^4}{EZ^3} \qquad (13)$$

where q is a constant and w is the force per unit area on each "pie section." The value of w is some fraction of the acceleration force of equation (7). The maximum deflection is set at two per cent of the distance moved by the diaphram. Substituting this restriction in (13) and solving for Z gives

$$Z = R^2 \left(\frac{\rho}{E} \times K_2 \times K_4\right)^{\frac{1}{2}} \qquad (14)$$

where K_4 is a constant. Substituting (14) into (13) gives

$$m = R^4 \times \left(\frac{\rho^3}{E}\right)^{\frac{1}{2}} \times K_3 \times K_2^{\frac{1}{2}} \times K_4^{\frac{1}{2}} \qquad (15)$$

substituting (15) and (10) into (7) gives

$$F = K_1 \times K_2^{\frac{3}{2}} \times K_3 \times K_4^{\frac{1}{2}} \times \left(\frac{\rho^3}{E}\right)^{\frac{1}{2}} \times R^2 \qquad (16)$$

This equation shows that the force required for diaphram movement during the rise time period is a function of only the diaphram's radius and a ratio of the density and Young's Modulus of the construction material.

The best construction material is chosen by examining the factor $(\rho^3/E)^{\frac{1}{2}}$, which can also be written $\rho(\rho/E)^{\frac{1}{2}}$. Therefore, the best material is one with an adequate (ρ/E) ratio and a small density. Aluminum has the same (ρ/E) ratio as steel, but is less dense. Likewise, it is easily available and relatively inexpensive. The aluminum used for the circular diaphram will be approximately .050 inches. This is the minimum value that lends itself to easy construction techniques.

Using .050 inch aluminum, the diaphram and associated parts weigh approximately eight pounds and require about 450 pounds of acceleration force.

The spring or pressure cylinder must deliver 450 pounds of force. This value is possible using either device. Therefore, if one system is better, its advantages must be in its adjustability technique, its quietness, or its simplicity. The spring is adjusted by changing its length. This procedure is relatively easy, but has the possibility of being noisy. Also the spring, by necessity, is long, which would take up valuable limited space. Similarly, the pressure cylinder will consume space. The interior volume of the cylinder is large to insure reasonably constant force output. Cylinder adjustability is obtained by changing only a regulator value. This method is easy and makes little noise. The adjustment technique of the cylinder is superior. Therefore, the pressure cylinder is chosen as the most desirable forcing device for the rise time periods of diaphram movement.

The simple cam, which is to be used with the air cylinder, must be amplitude adjustable. Either a special adjustable cam or a level will be used. Any adjustable cam must have the required shape at all of its possible adjusting positions. This requirement is near impossible to attain for the shape cam that is required. Various adjustable cams are possible, but they all create errors of as large as 30% from the desired N-shape. Similarly, a lever will introduce non-linearity into the system. However, a lever will cause maximum deviations of only 5% from the N-shape desired. Both devices can be noisy during the adjustement, and both could be expensive. The lever device will add some additional weight, since it is a moving part. The adjustable cam adds no weight. The lever will necessitate space beside the diaphram and the adjustable cam will necessitate about the same area behind the cam. Examining the stresses present in the lever device reveals that excessively high impact stresses exist. The only method of alleviating this stress state is to vastly increase the size of the lever which, in effect, increases the mass that is to be accelerated. Such mass additions are not feasible. The impact stresses in the adjustable cam device are not excessive.

Therefore, the adjustable cam method is the only feasible means of amplitude adjustment.

The cam must have rotational adjustment to alter the duration of the simulated boom. This can be achieved by using an adjustable output A-C or D-C motor. Initial estimates of cam size indicate the need for a 3/4 hp motor. The motor will be used in conjunction with a flywheel to reduce or eliminate any speed variations due to shock loading. An adjustable D-C motor is a relatively common piece of machinery. A solid-state control unit controls the speed while keeping the torque constant. The control unit makes little acoustical noise, but the D-C motor does create a fair amount of background noise. This motor noise is the result of the contact of the motor brushes on the armature.

An adjustable A-C motor is a large, complicated piece of machinery. An A-C motor has no brushes and therefore does not produce as much noise as its D-C counterpart. However, adjustable A-C motors are very expensive. The small noise reduction does not offset the large price increase. Even though adjustable A-C motor units cannot be used, there is the possibility of using the A-C motor together with a mechanical, variable speed reducer. All such mechanical devices are inherently very noisy, except for a variable pitch pulley arrangement. This type of variable speed device, like the D-C unit, will deliver variable speed and constant torque. Little or no data is available on the noise produced by variable pitch pulleys. However, it appears that such pulleys would have the capability of producing noticeable noise. Slipping belts can also cause a squealing noise.

Both devices create background noise. However, the D-C motor is chosen as the best device becasue it has less sources of possible noise and it is simplier.

Detail Design

Preliminary design has established the major components of the sonic boom simulator. In this section, detail design, a complete engineering description is presented for every subsystem of the mechanism. To simplify presentation, the simulator is divided into five subsystems, control volume, diaphram, pressure cylinder, adjustable cam, and drive system.

Control Volume

The control volume, Figures 4,5, is in the form of a trunkated pyramid. The base of the pyramid is the eight-foot high by twelve-foot wide section of the testing room wall, which is similar to an exterior wall of a typical residential dwelling. Fundamental considerations indicate that the smaller the control volume, the less energy the sonic boom simulator has to supply. The maximum interior dimension of eight inches is the smallest possible distance that gives the pressure wave time to reach the outer regions of the control volume before it has reached the room wall, at the closest point, and risen to its peak value of pressure. Therefore, the 28 cubic foot volume of this control volume is the smallest possible for an 8 by 12 wall.

The side walls of the control volume are stiffened extensively to minimize any bowing that would reduce the pressure load on the testing room wall. The stiffeners are placed on the exterior of the chamber to insure that the pressure wall is not obstructed.

All joints and seams of the control volume are sealed air tight.

Diaphram

The diaphram is constructed as shown in Figures 6 through 8. This subsystem is designed for minimum weight and maximum strength. Four main components compose the diaphram subsystems, a circular plate, twelve ribs, a shaft, and a cam follower.

The aluminum circular plate is two feet in diameter and .050 inches thick. Around the perimeter of the circle there is a one-half inch lip perpendicular to the plate. This lip gives additional rigidity to the plate. A sheet of neophrene covers the entire surface of the plate and extends across the clearance area to the control volume top. The neophrene is secured to the diaphram and control volume, creating a seal to keep the chamber air tight.

The twelve ribs, which support the circular plate, are also constructed of .050 inch aluminum. The ribs are equally spaced around the circumference of the plate, as shown in Figure 8. All three edges of each rib are folded 90° to give a one-half inch lip. This process increases the rigidity of each rib and simplifies construction. Two holes are bored in each rib at points of low stress. This procedure decreases the weight of the ribs, but has an insignificant effect on the strength of each rib. The ribs are secured to the shaft and plate with epoxy glue. An auxiliary support ring is connected to each rib to insure the ribs do not buckle.

The aluminum circular shaft is 22 inches long and .125 inches thick. A circular plate of the same thickness is welded to the rear end of the shaft. This plate acts as the piston head of the pressure cylinder.

The aluminum cam follower is clamped to the diaphram shaft. A nylon pad is joined to the follower at the point of cam contact. Nylon is used because it has a low coefficient of friction and will not create significant noise as it rubs against the cam surface.

The cam follower, together with the shaft, ribs, and plate of the diaphram, weigh approximately eight pounds. This is the total weight that has to be accelerated by the pressure cylinder.

Information from Roark (9) indicates that the maximum deflection of the circular plate, as designed, does not exceed .024 inches and the maximum impact stress in the plate does not exceed 15.5 kpsi. Similarly, information from Shigley (10) shows that the maximum impact stress in the shaft, which occurs at the cam follower, is found not to exceed 24 kpsi. A minimum safety factor of 1.7 exists for all stresses when compared to the fatigue strength of machined aluminum with a 2.5×10^4 cycle life, which is the equivalent of five years of daily use. A typical stress analysis calculation is presented in the appendix.

A rubber bearing is used for bearing support of the moving diaphram. This type bearing effectively dampens unwanted noise and vibration. Rear bearing support is supplied by the pressure cylinder.

Pressure Cylinder

The pressure cylinder, Figures 9, 10, is placed directly behind the diaphram shaft. This type of cylinder is used because the mass of the piston head and piston rod of a commercial air cylinder has to be accelerated just as if they are part of the diaphram-shaft assembly. The additional weight is eliminated by constructing an air cylinder with the diaphram shaft acting as the piston.

The cylinder is connected to a 100 psi pressure line, utilizing a regulator value to control the magnitude of the forcing pressure. The maximum pressure of 100 psi will create 707 pounds of force. However, the simulator, as designed, requires only 415 pounds of force. The pressure cylinder is designed so that the maximum shaft displacement causes less than a 4.6 per cent deviation from the constant force requirement.

The air cylinder device also acts as a guide and, therefore, eliminates the necessity of rear bearing support.

As the amplitude of the simulated boom is altered, the air cylinder backpressure will also have to be changed to achieve the same rise time. A pressure gage and calibration chart are supplied to assure easy adjustment. The cylinder is designed to create forcing pressures capable of creating simulated sonic booms with rise times of .01 seconds, throughout the entire amplitude range. Of course, higher rise times are possible by using a lower backpressure.

Adjustable Cam

The adjustable cam, Figures 11, 12, consists of two half-cylinders which slide to adjust the boom amplitude. Only one-half of the cam profile is used to create the N-shape pressure wave required. The other half of the profile is used for starting and stopping and has no effect on the diaphram's movement. The cam is adjusted by turning an adjustment screw to reach the desired amplitude, and then tightening a nut to insure the profile does not change during operation.

This adjustable cam does not create a linear pressure decay like the idealized case of an N-shaped pressure wave. The per cent error of the cam is defined as the difference between the pressures of a simulated sonic boom and an idealized N-shaped sonic boom of the same amplitude, compared to that amplitude. Using this definition, the adjustable cam has a maximum error of 26% at the maximum boom amplitude, and a minimum error of 23% at the lowest amplitude.

Drive System

The simulator drive system, Figures 13, 14, 18, consists of four main parts, a variable speed motor, a flywheel, a speed reducer, and a single revolution clutch. The entire drive system is enclosed within an acoustical chamber, Figure 17, to reduce the background noise that is transmitted to the testing room.

A variable speed, 3/4 hp D-C motor and solid-state control unit is used to drive the adjustable cam. The motor has the capability of running at full torque over a range of 785-1725 revolutions per minute. Sleeve bearings, instead of ball bearings, are used on the motor armature to reduce running noise. The motor is equipped with dual shafts, one shaft connects to the speed reducer and the other shaft connects to the flywheel.

A 24-pound flywheel is mounted on one of the motor shafts. The flywheel is a solid steel cylinder of five-inch radius. Fundamental calculations show that the flywheel has a moment of inertia of 300-pound square inches. This magnitude of inertia, running on the high speed maximum speed reduction of less than one per cent when the single revolution clutch is suddenly engaged.

The speed recuder is connected to the other motor shaft. Two sets of V-belts and V-pulleys are used to obtain a total speed reduction of 9.16. This reduction gives a maximum cam rotation of 187 revolutions per minute, or the speed necessary for a simulated sonic boom of .161 millisecond duration. By adjusting the motor control, boom durations as high as 350 milliseconds can be obtained. V-belts are used for speed reduction because they are relatively quiet and they can easily transmit the power required for this mechanism. To insure against any belt slippage, special notched V-belts are used. Sleeve bearings are incorporated throughout the speed reducer subsystem, as well as the entire drive system, to reduce background noise.

A single revolution clutch is incorporated within the drive system to insure that only a single simulated boom is produced at a time. This clutch is positioned between the speed reducer and the adjustment cam, as shown in Figure 18. When the cam is activated, the reduced motor rotation is transmitted directly to the cam for a single revolution. The clutch is activated by a solenoid device. This solenoid device lends itself well to computer triggering of the sonic boom simulator.

The drive system, as well as the entire sonic boom simulator, is supported on a heavy steel mounting table. This mounting table is secured to the floor as shown in Figure 15. This technique reduces the amount of unwanted noise and vibration that is transmitted to the testing room.

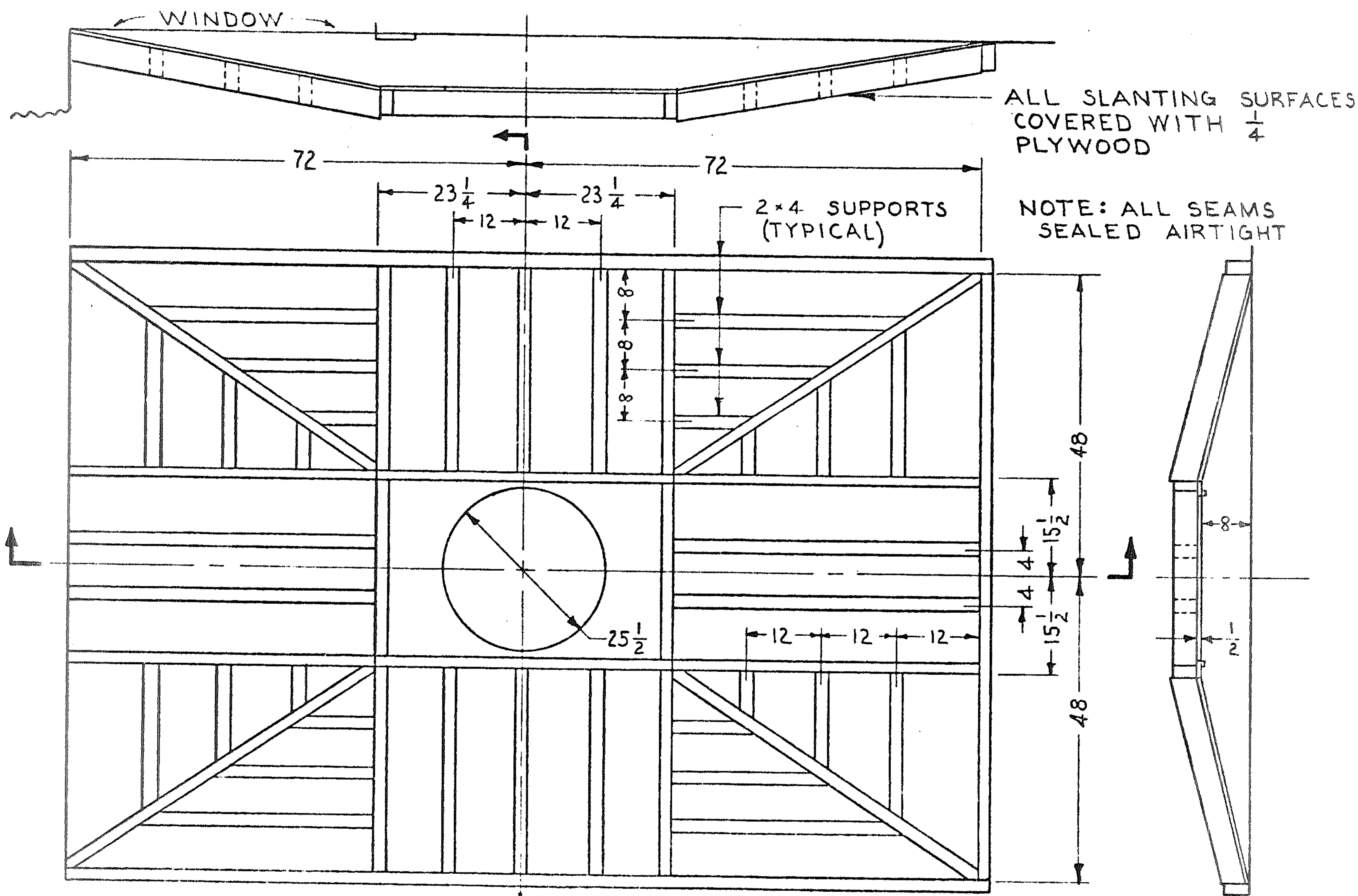


FIGURE 4 CONTROL VOLUME DETAIL

SCALE: 1"=24"

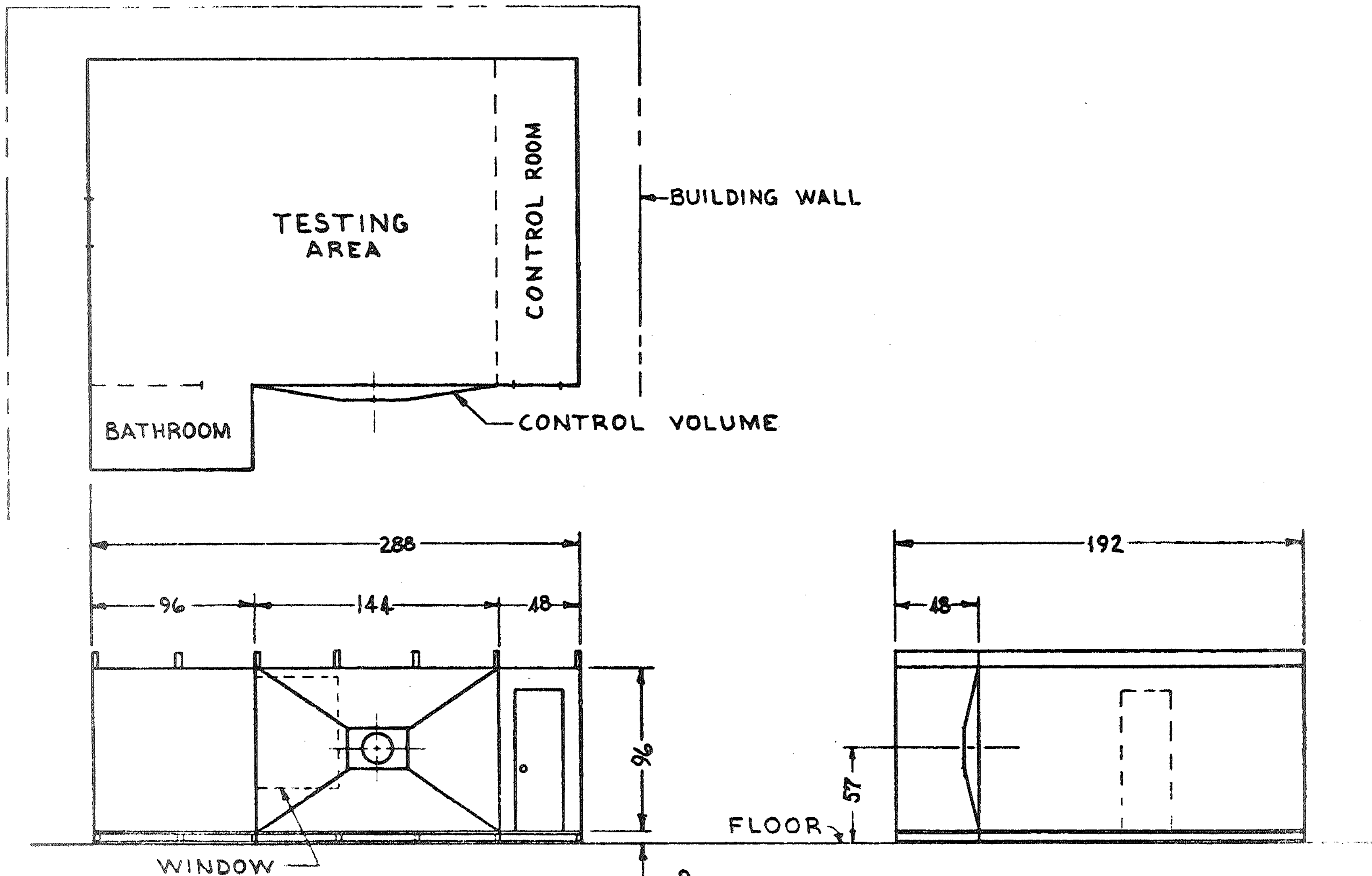


FIGURE 5 CONTROL VOLUME SUBASSEMBLY SCALE: 1"=96"

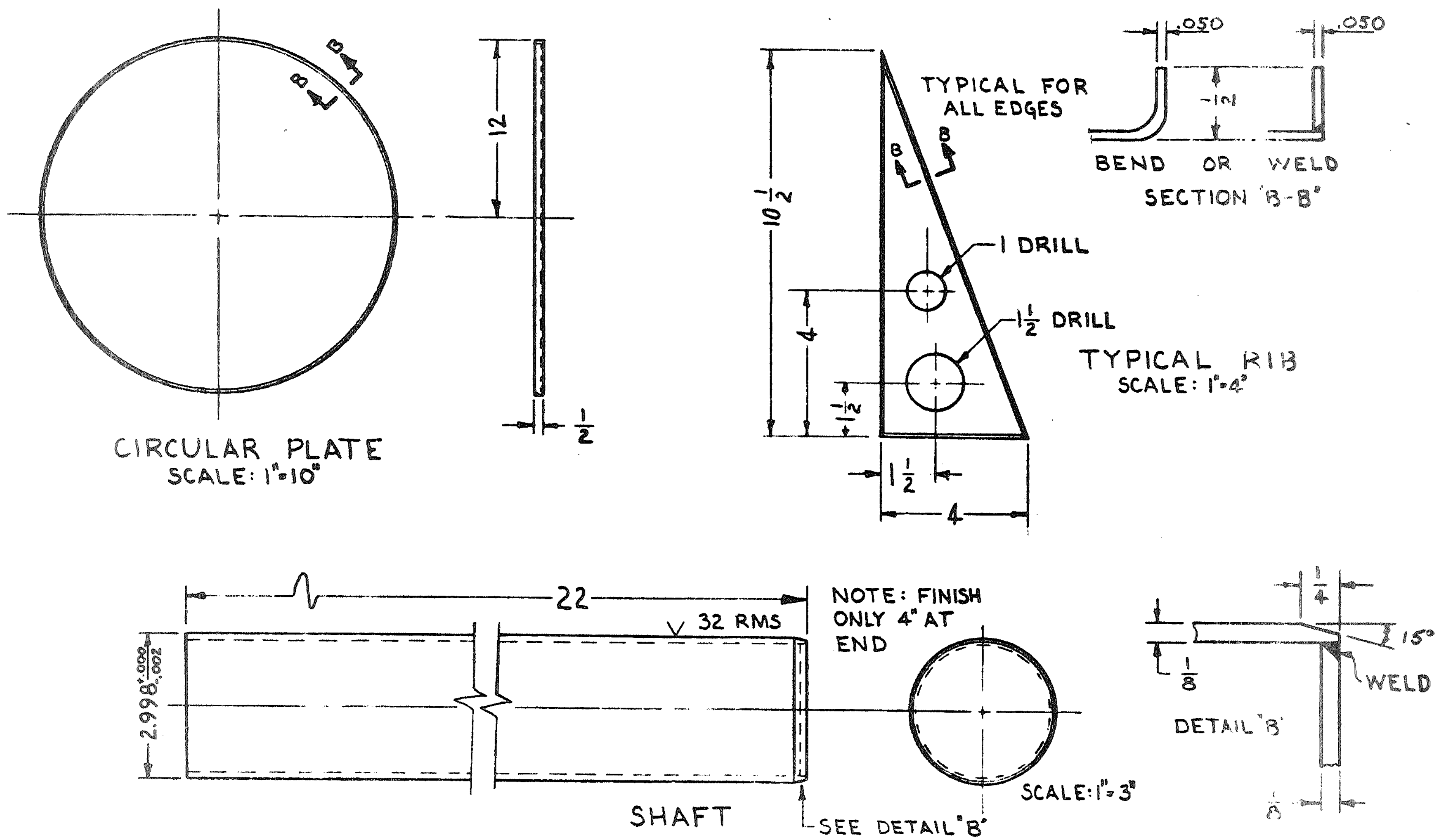


FIGURE 6 DIAPHRAM DETAIL NO. 1

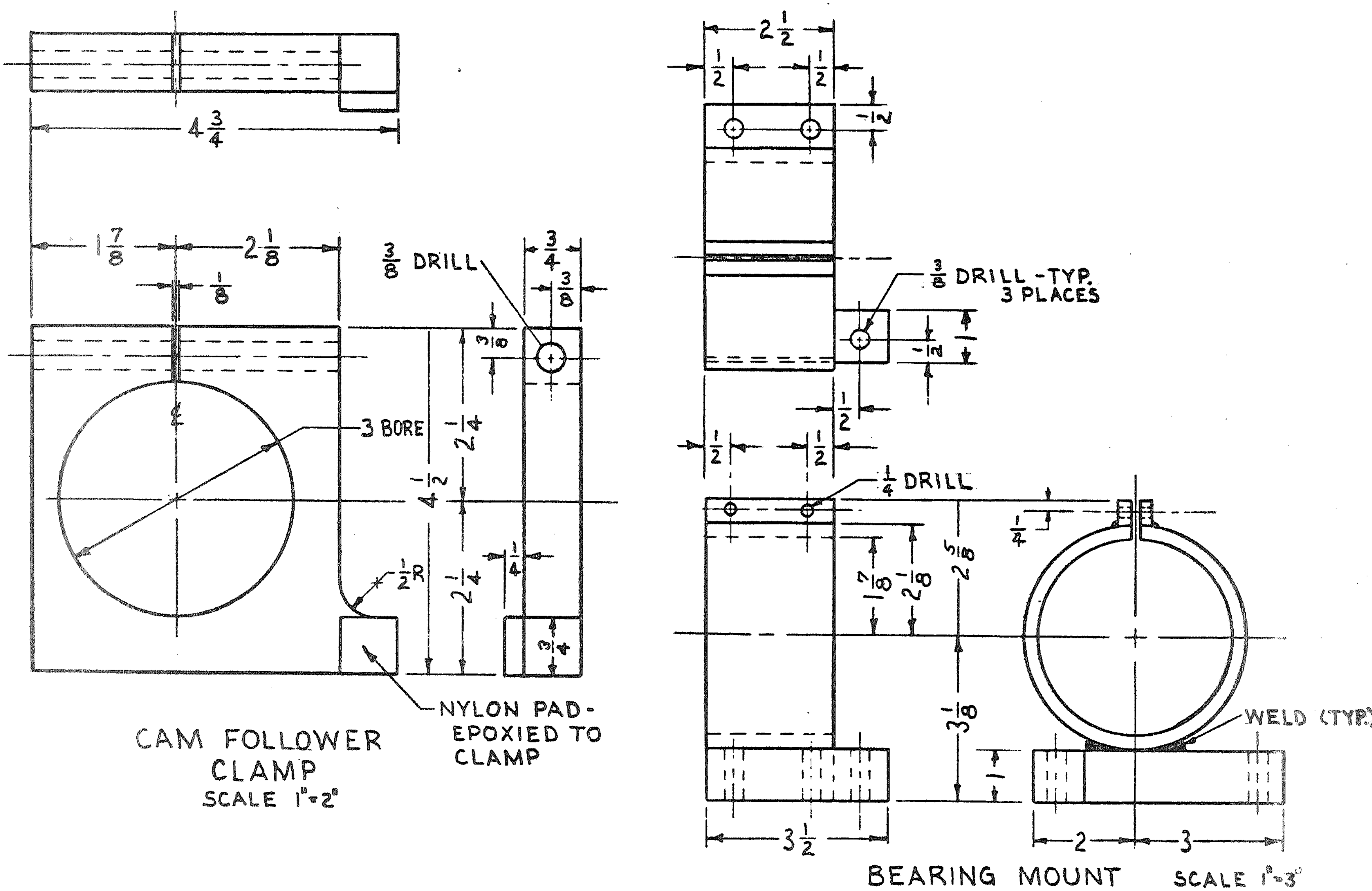
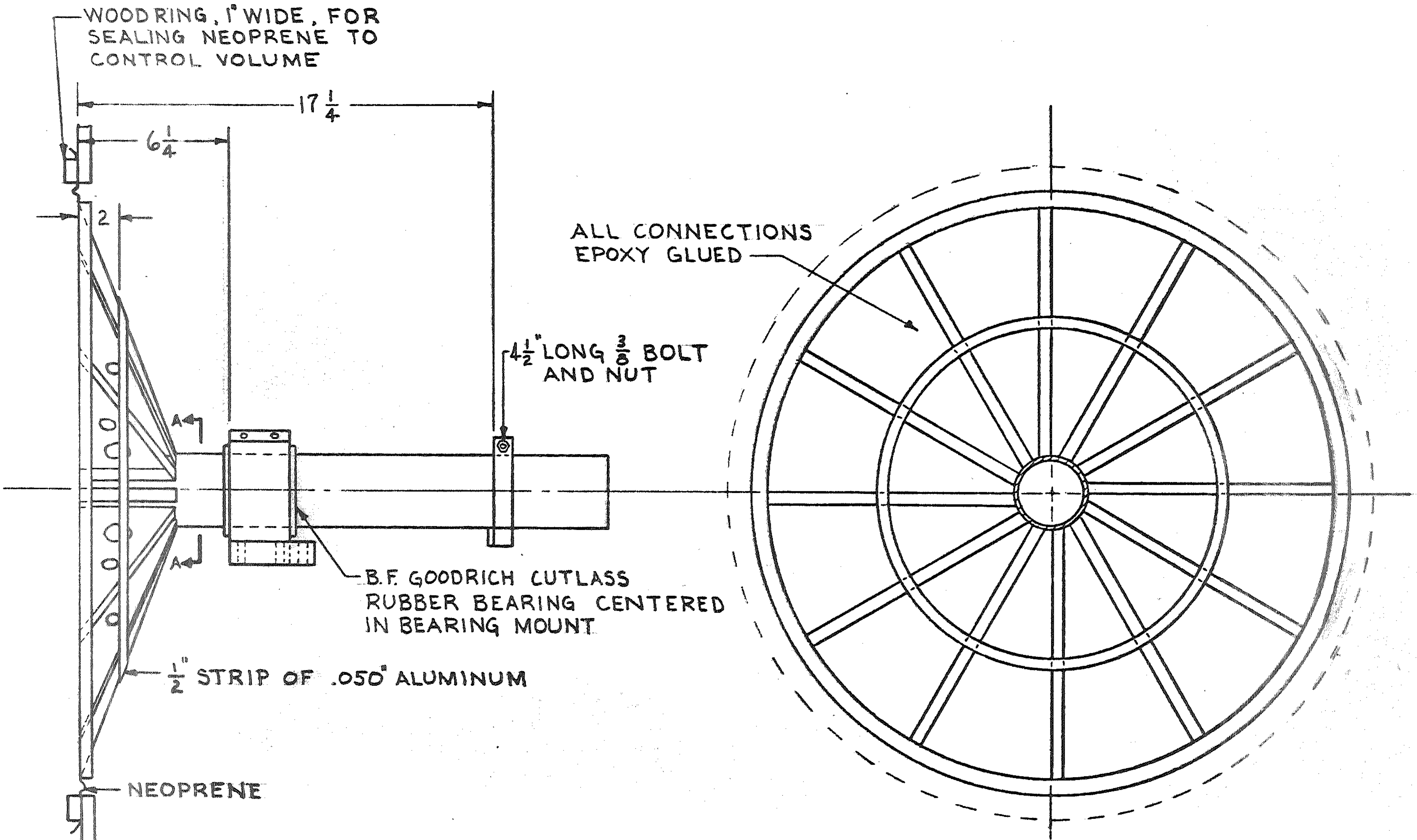


FIGURE 7 DIAPHRAM DETAIL NO. 2



SECTION A-A

FIGURE 8 DIAPHRAM SUBASSEMBLY SCALE: 1" = 6"

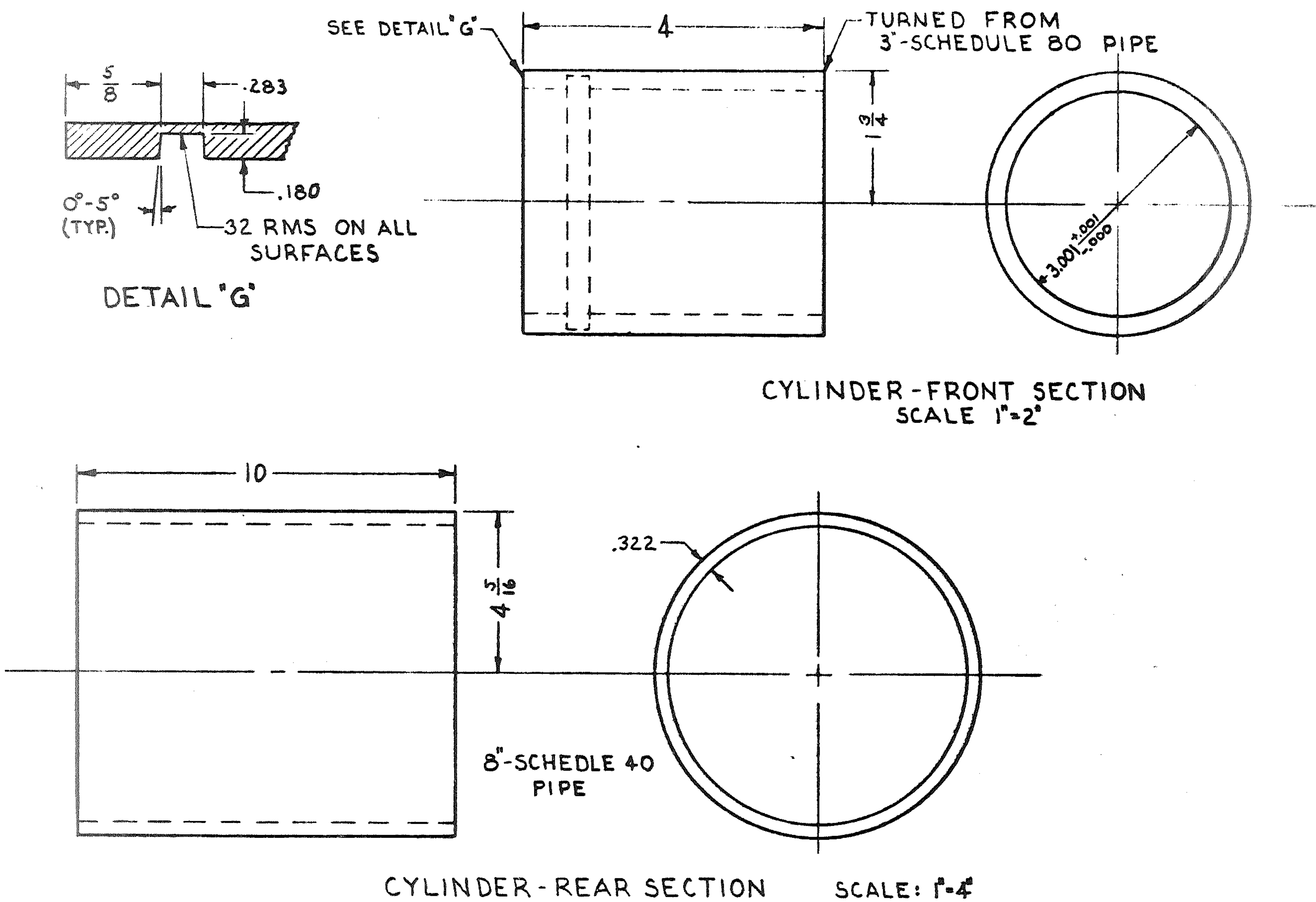


FIGURE 9 PRESSURE CYLINDER DETAIL

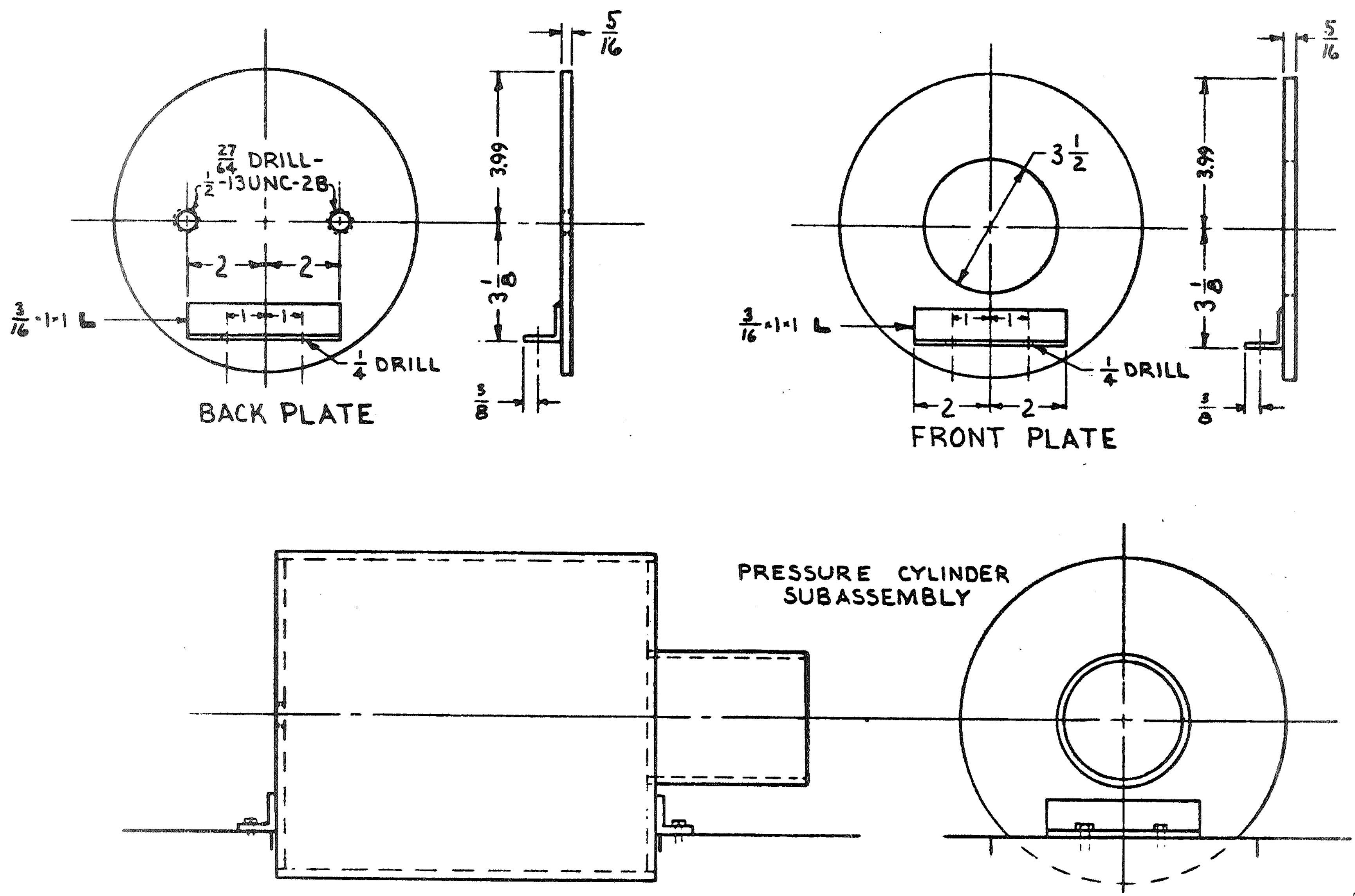


FIGURE 10 PRESSURE CYLINDER DETAIL & SUBASSEMBLY

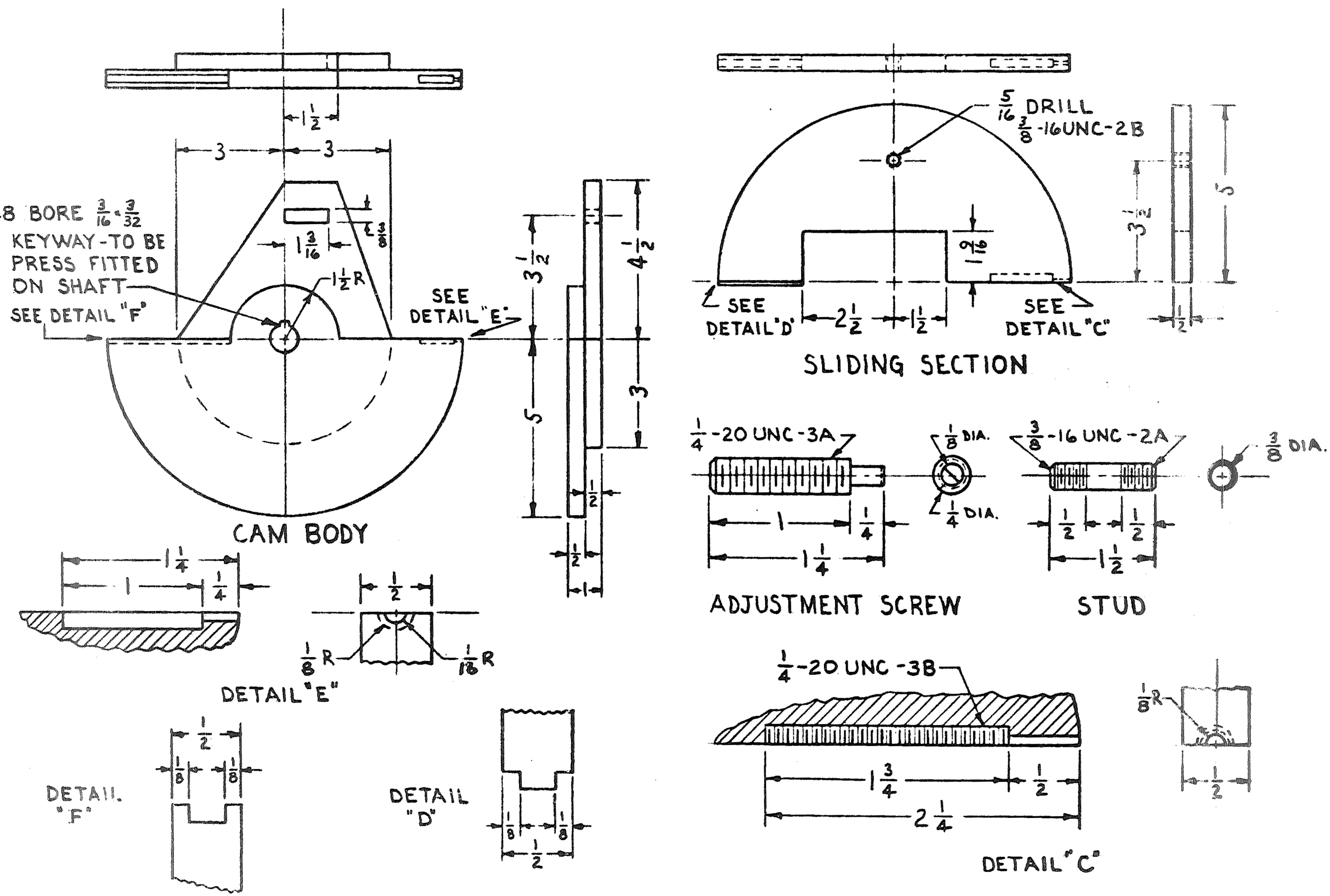


FIGURE 11 ADJUSTABLE CAM DETAIL SCALE: 1"=4"

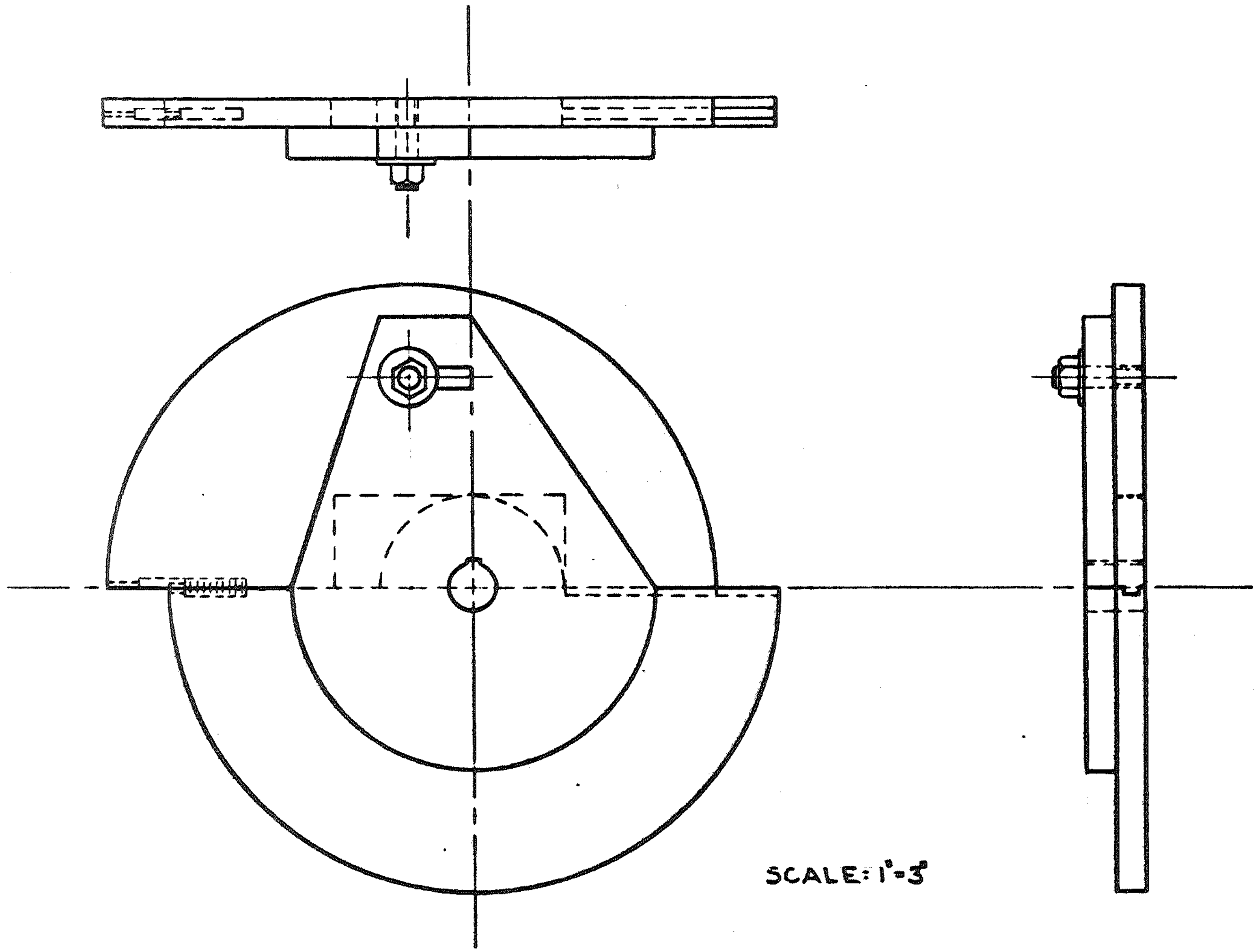


FIGURE 12 ADJUSTABLE CAM SUBASSEMBLY

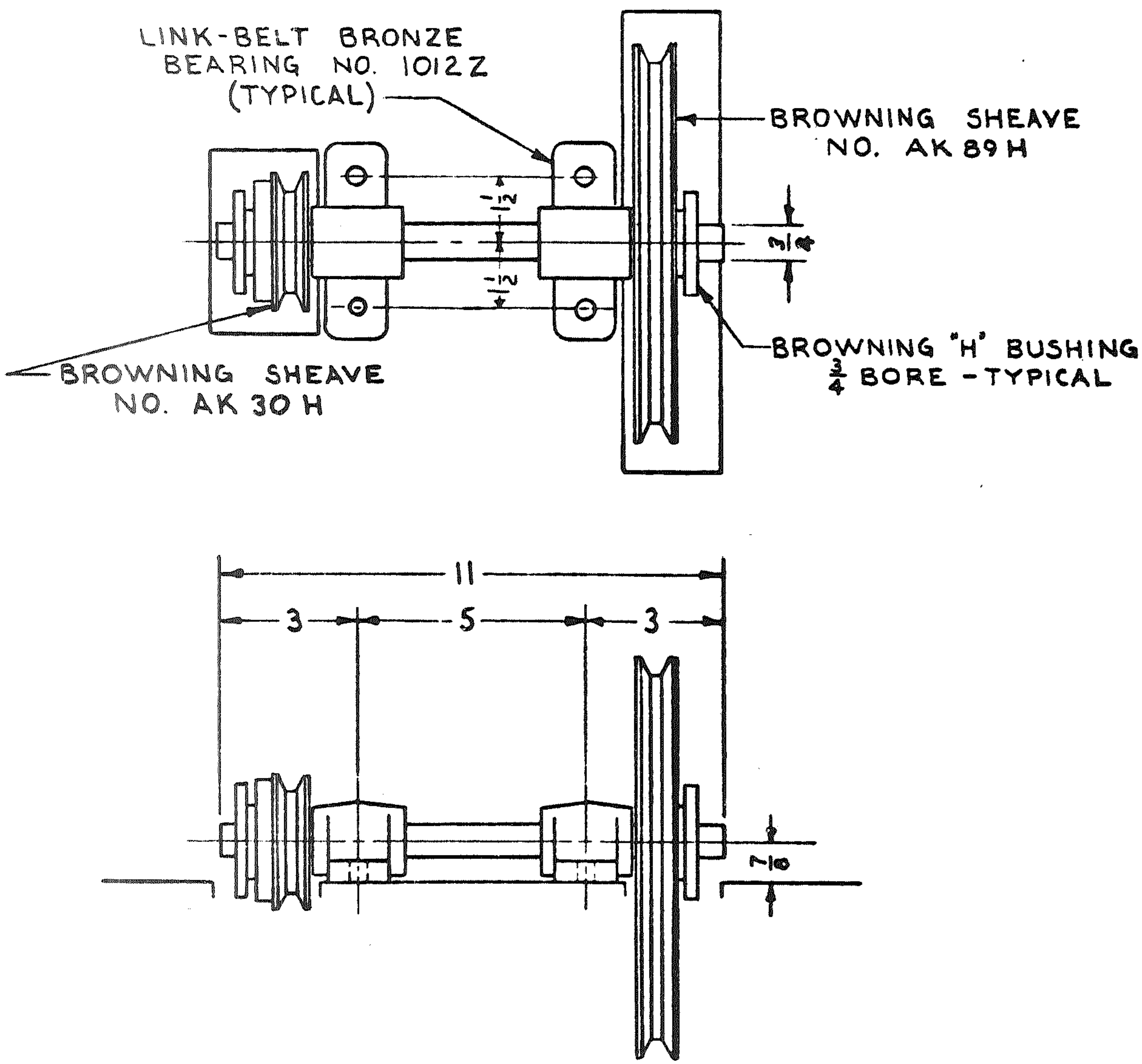


FIGURE 13 INTERMEDIATE SHAFT SUBASSEMBLY
SCALE 1"=4"

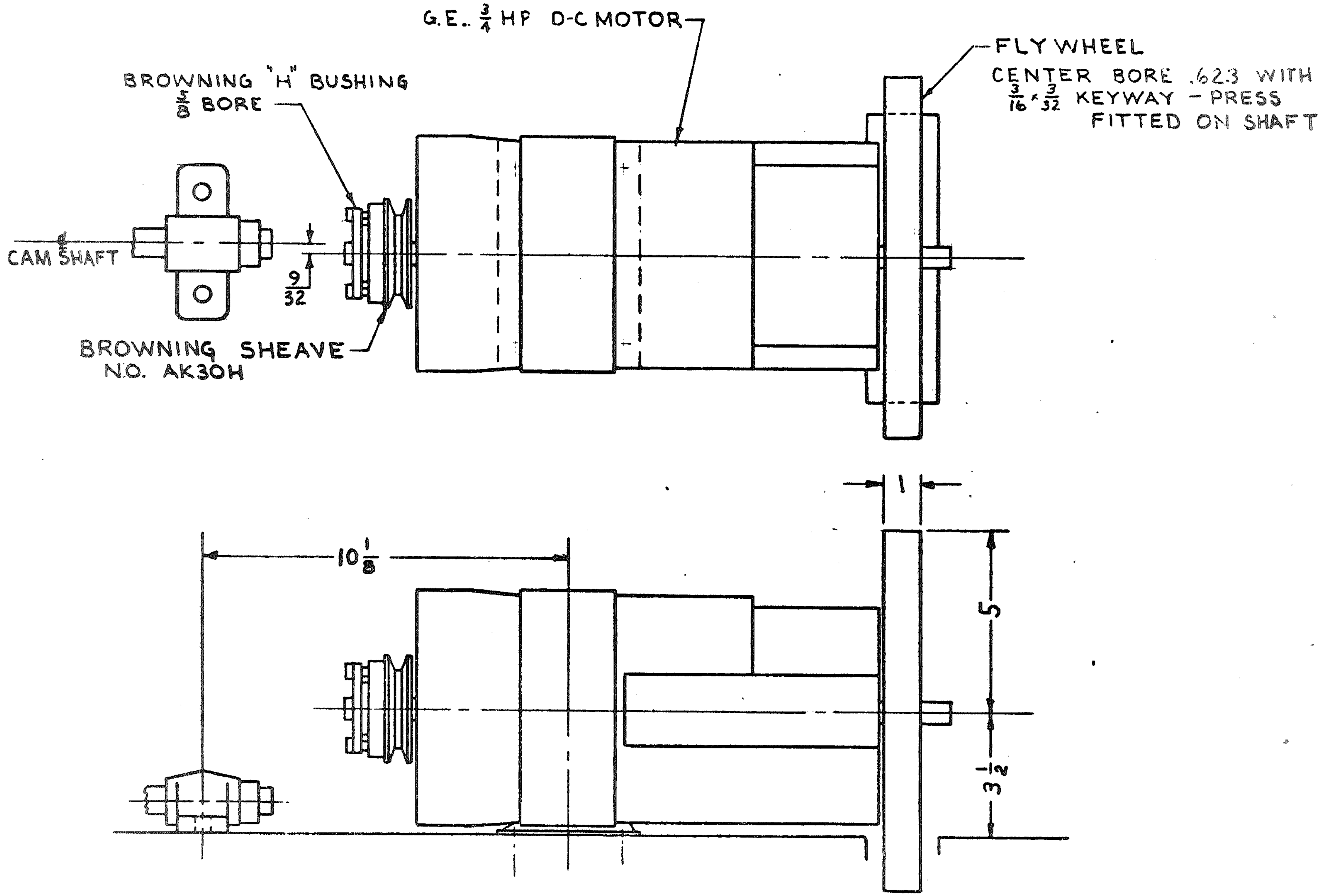


FIGURE 14 MOTOR SHAFT SUBASSEMBLY SCALE 1"=4"

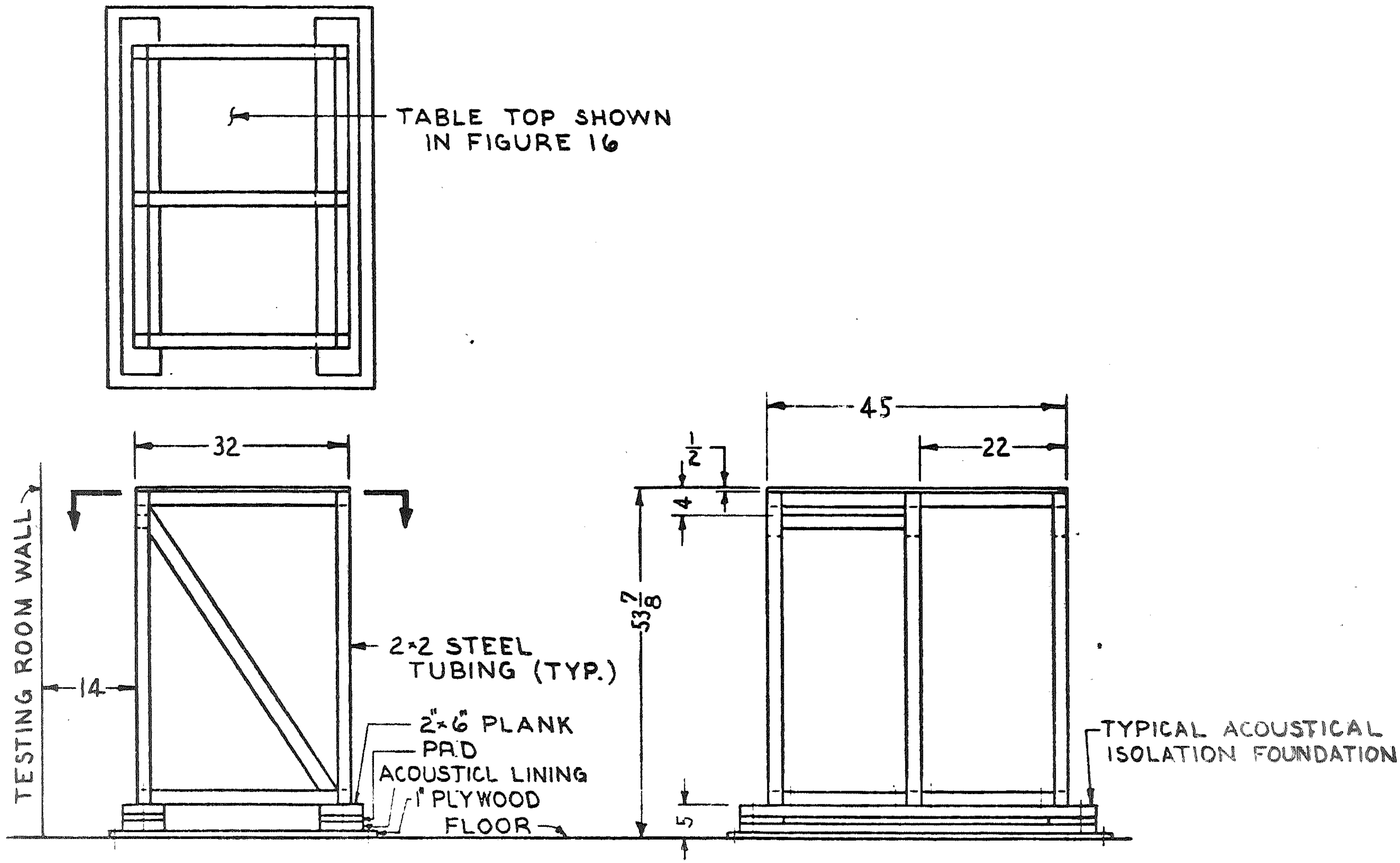


FIGURE 15 MOUNTING TABLE SCALE 1"=12"

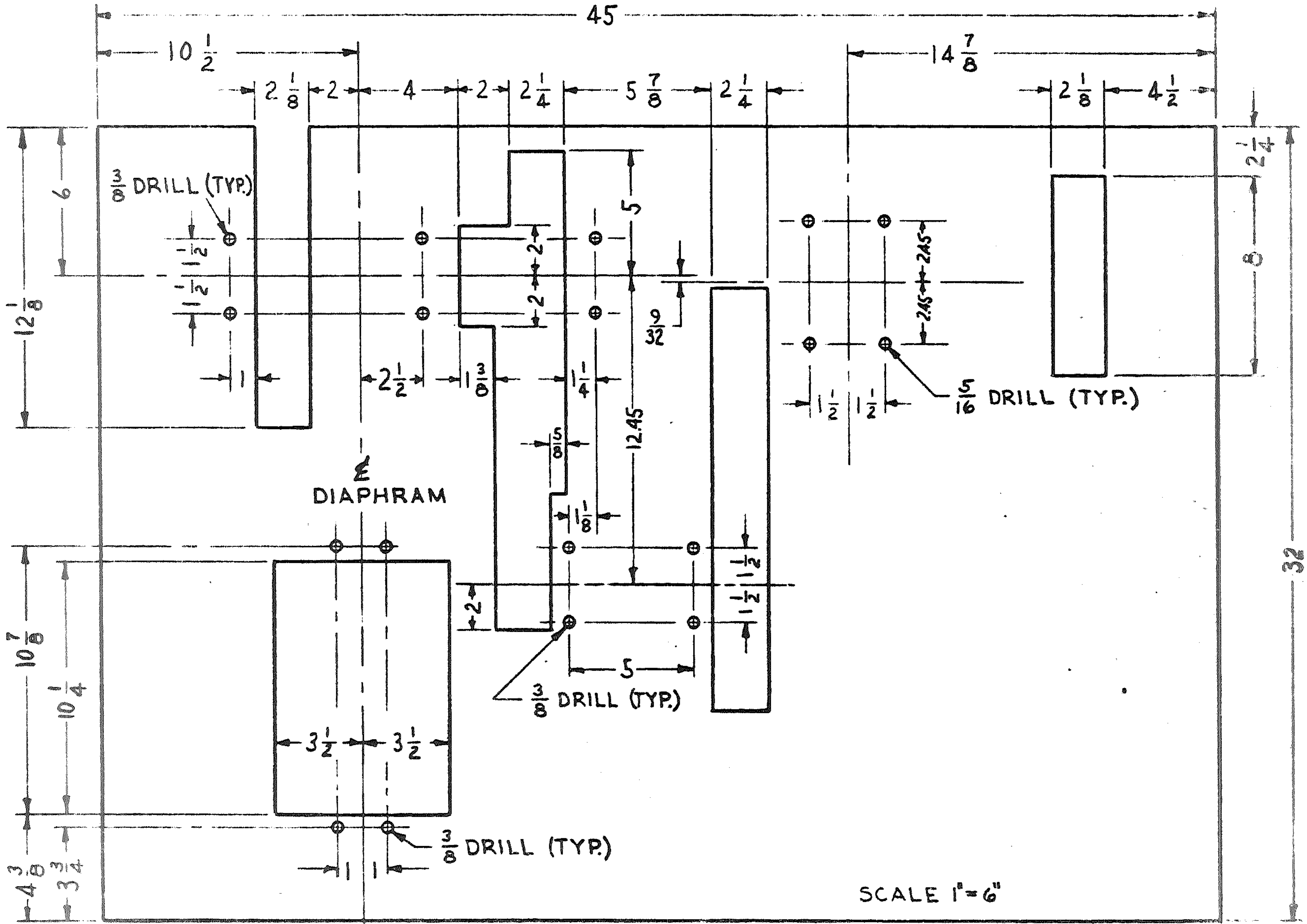


FIGURE 16 MOUNTING TABLE TOP DETAIL

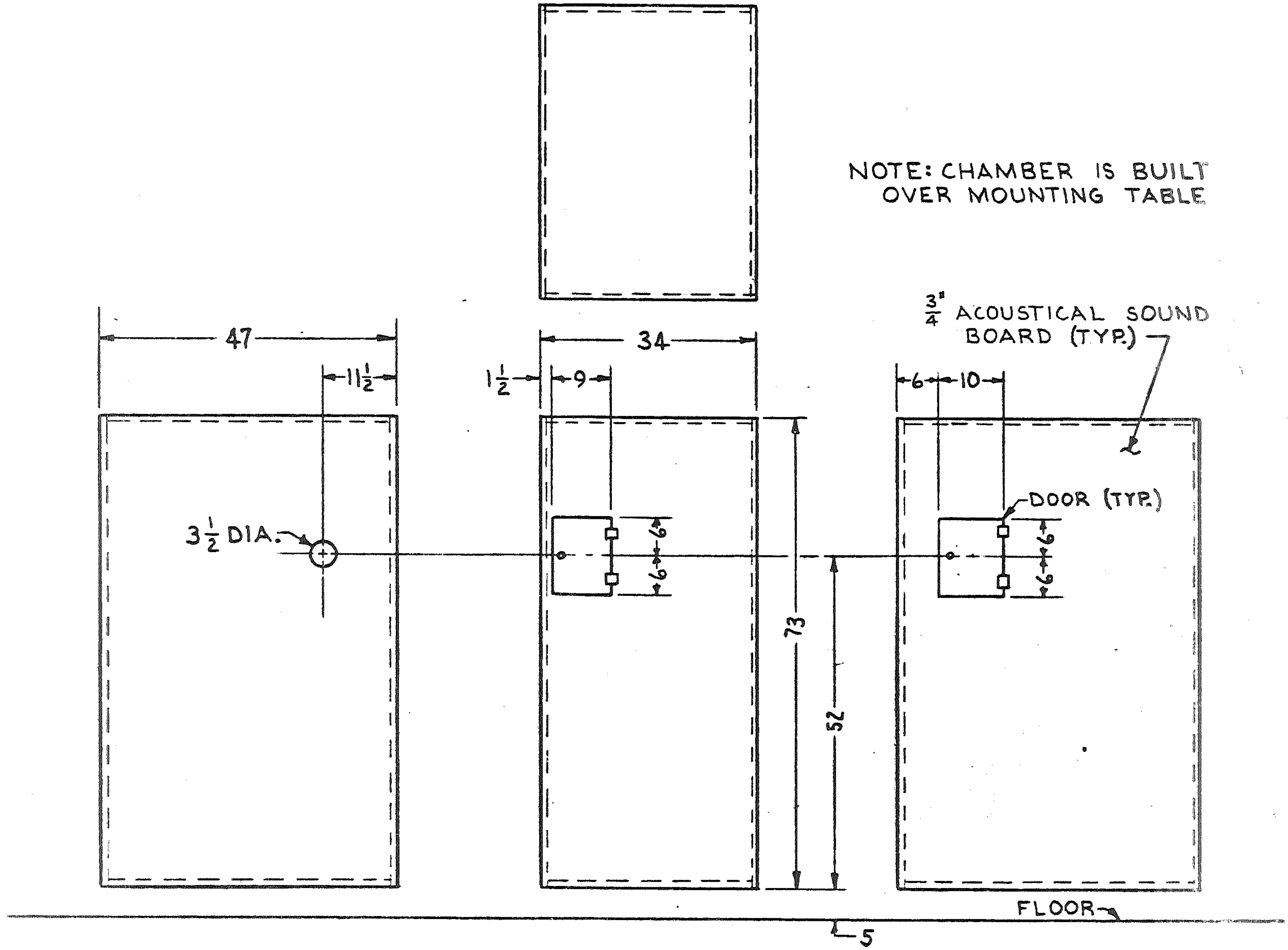


FIGURE 17 ACOUSTICAL CHAMBER
SCALE 1" = 24"

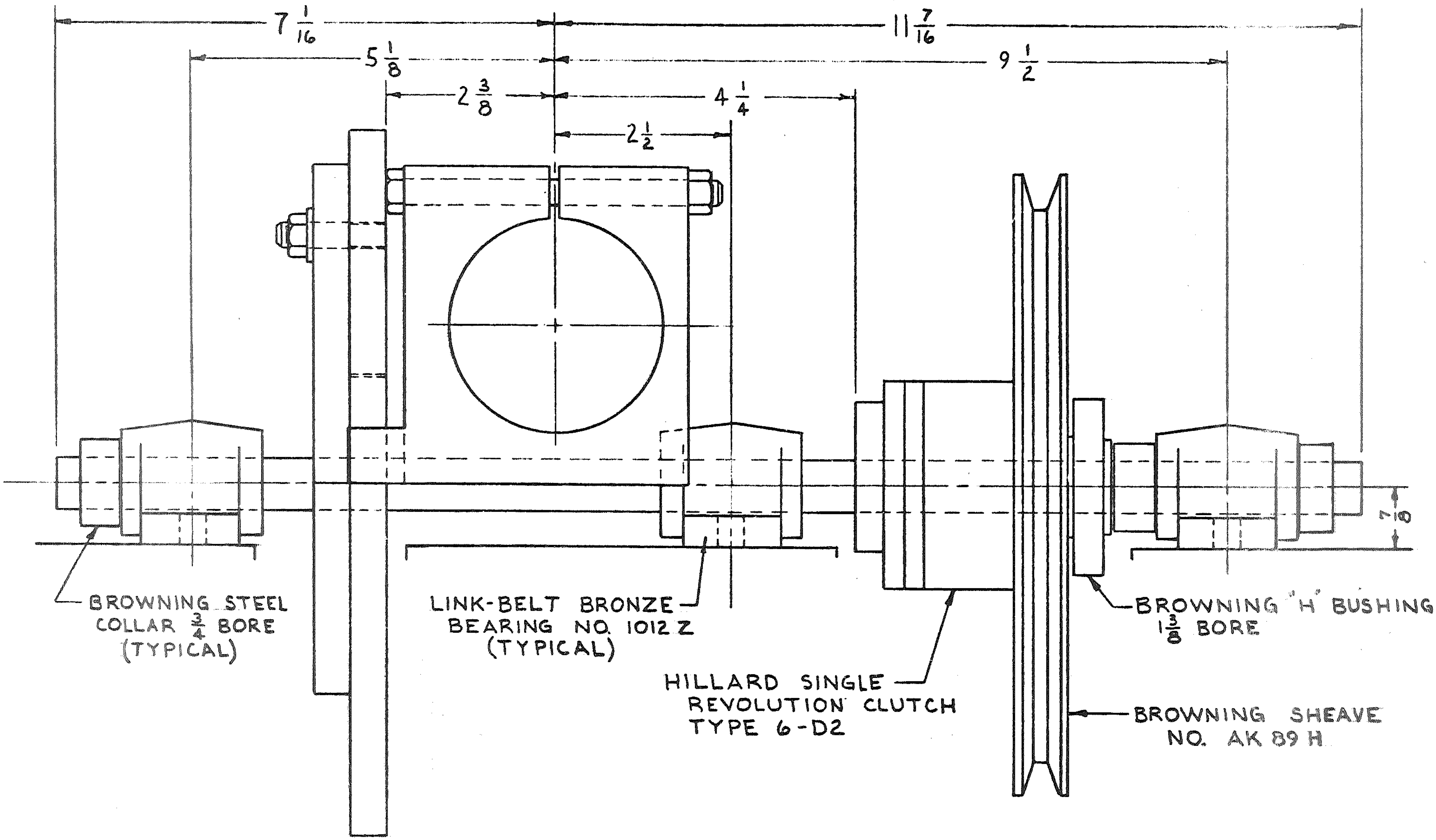
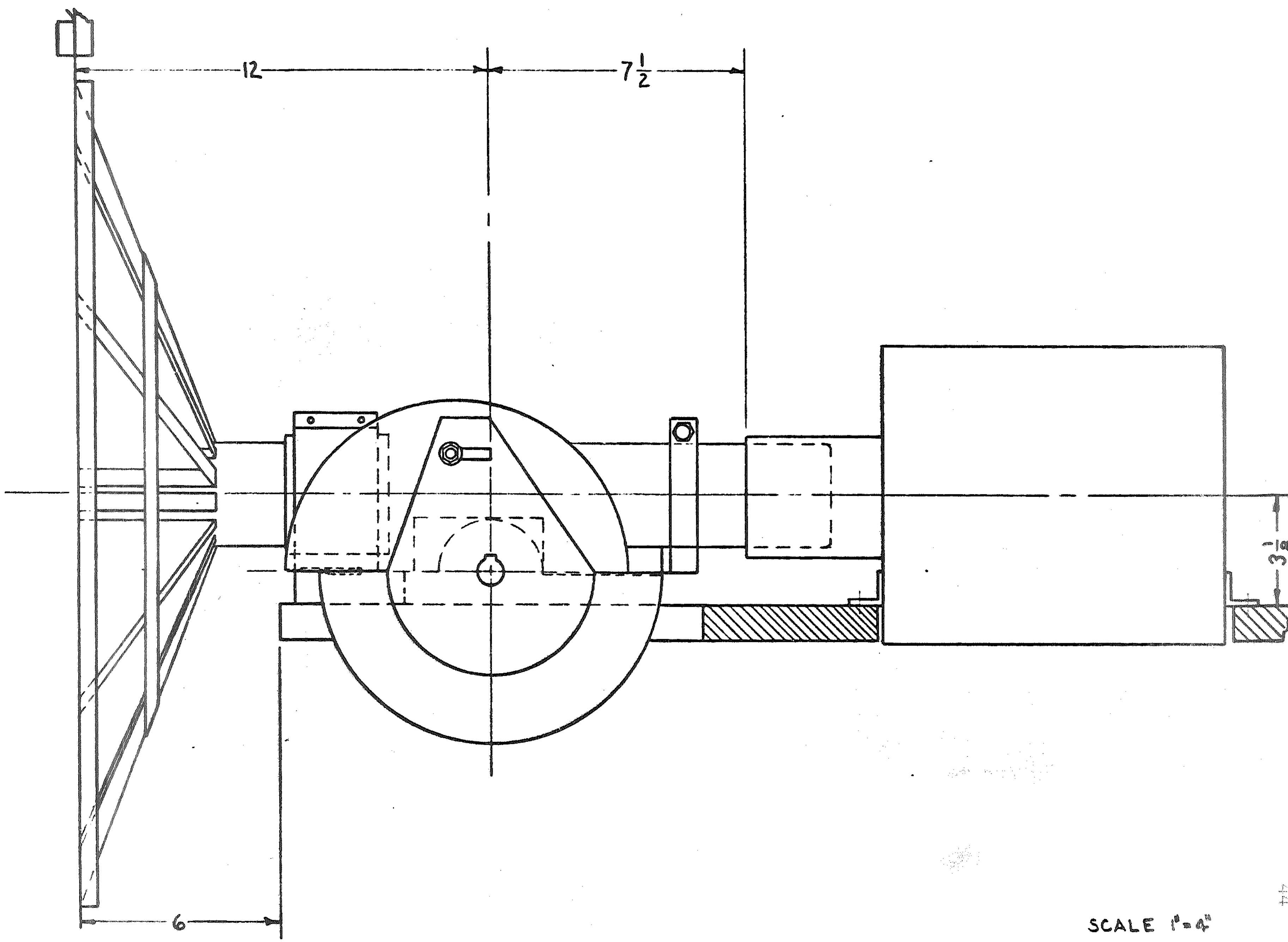


FIGURE 18 CAM SHAFT SUBASSEMBLY

SCALE 1"=2"



SCALE 1" = 4"

FIGURE 19 SONIC BOOM SIMULATOR ASSEMBLY NO. 1

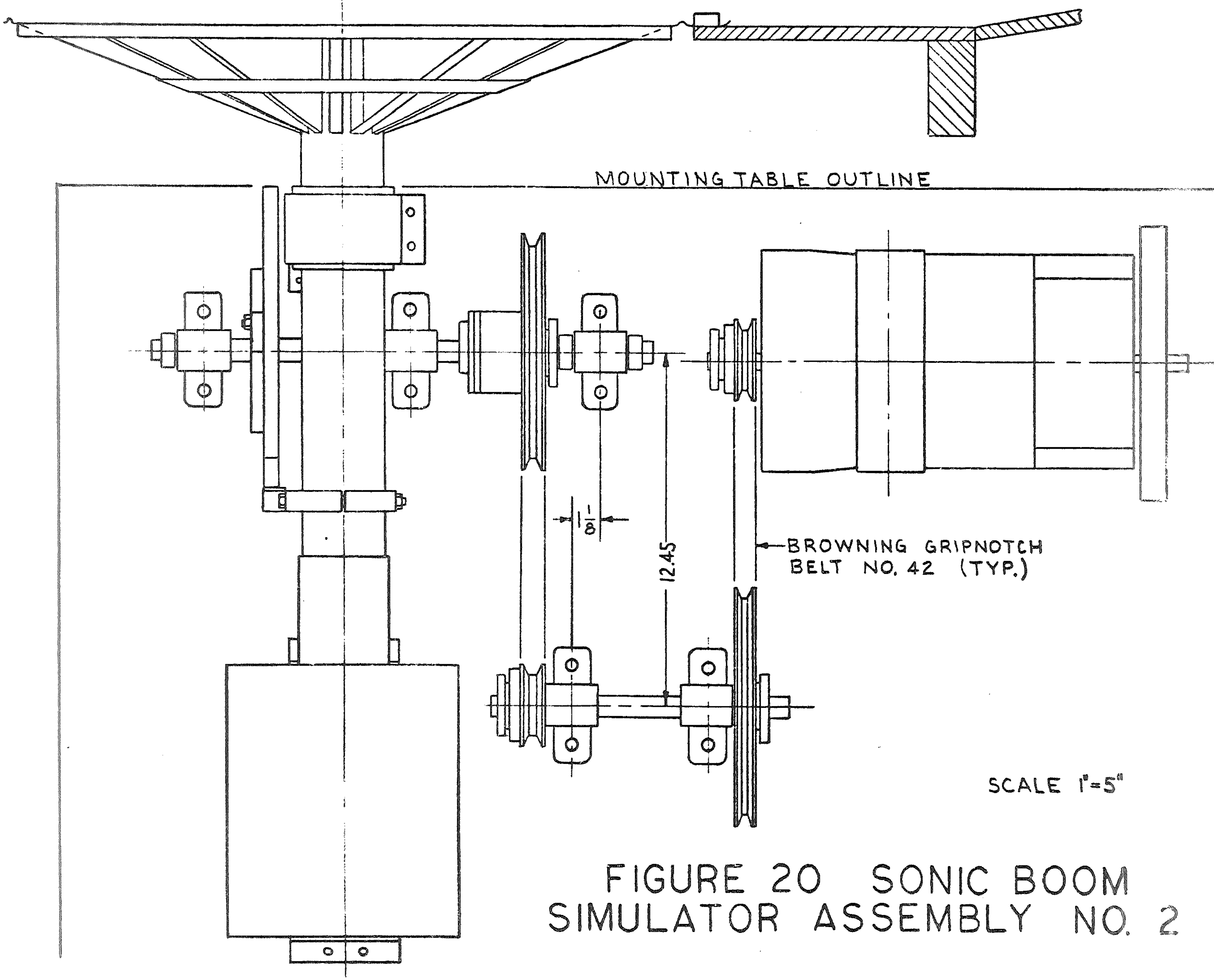


FIGURE 20 SONIC BOOM
SIMULATOR ASSEMBLY NO. 2

Table 2. Major parts list of sonic boom simulator

Name	Quan.	Supplier	Specifications
Motor	1	General Electric	Statotrol SCR Drive System 3/4 hp - 1725 rpm - 230 volt -dripproof motor No. BCD56RA75 3SRA71 Operator's Controls - non-reversing, mounted No. 3SRP10 Special Modifications - double shaft extension sleeve bearings
Single Revolution Clutch	1	Hillard Corporation	Type 6 - Size D-2 clockwise rotation
Bearing	5	FMC Corp. Link-Belt Bearing Div.	Bronze Sleeve Bearing type 1012Z 3/4" shaft
Rubber Bearing	1	L. Q. Moffitt	B. F. Goodrich Cutlass Rubber Bearing code: navy (cut into four 3" pieces)
V-Pulley	2	Browning Mfg. Company	FHP Sheave AK 30 H
V-Pulley	2	Browning Mfg. Company	FHP Sheave AK 98 H
Pulley Bushing	4	Browning Mfg. Company	"H" Bushing 1 3/8" bore (1) 3/4" bore (2) 5/8" bore (1)
V-Belt	2	Browning Mfg. Company	Gripnotch Belt AX 42
Set Collar	3	Browning Mfg. Company	No. SC-3/4

Table 2. continued

Name	Quan.	Supplier	Specifications
O-Ring	1	Parker Seal Company	Size No. 2-337 Compound No. N219-7
Shaft	2	--	3/4" hardened steel 18 1/2" and 11" long 3/16" x 3/32" keyway (entire length)

RECOMMENDATIONS

Even though the design of the mechanism is finalized, it is recognized that certain problems might occur after the sonic boom simulator is constructed and put into operation. In this section, several possible problem areas are identified and recommendations are presented for their solution should they be needed.

Information is unavailable on the amount of noise created by the single revolution clutch. The acoustical chamber surrounding the simulator should effectively absorb all the clutch noise. However, if excessive noise is present as the clutch is activated, a shock absorbing material should be placed at the solenoid release point, or possibly within the clutch assembly.

The amount of noise that is transmitted down the diaphram shaft, across the control volume, and into the testing room is felt at this time to be insignificant. However, if excessive noise is recorded within the testing facility, a Lord Manufacturing Company form mount, or similar noise and vibration reducer, may have to be incorporated within the diaphram shaft. Such a mount would significantly increase the mass of the moving diaphram and consequently necessitate additional forcing power. The additional force requirement will probably restrict the sonic boom simulator to rise times greater than .010 seconds.

The entire sonic boom simulator is enclosed by an acoustical chamber. In addition to absorbing simulator background noise, this chamber will retain much of the heat generated by the motor. The motor and control unit is designed to run under normal temperature conditions and thus may experience overheating. If excessive overheating occurs within the vicinity of the motor, some additional ventilation system will be required. A small fan of special ventilation ducts should provide adequate additional ventilation.

CONCLUDING REMARKS

The sonic boom simulator, together with the acoustical testing room, constitutes the best method for conducting accurate experimentation on the physiological effects of sonic booms. The simulator produces the effects of sonic booms with rise times between .01 and .015 seconds, durations between .161 and .350 seconds, and overpressures between .5 and 3.0 pounds per square foot. Adjustment of each boom parameter can be accomplished independently and with a minimum of effort.

All components of the simulator are designed for maximum strength, with minimum weight and minimum deflection. In addition, the mechanism transmits a minimum of background noise to the testing facility.

Sonic booms usually have rise times less than the .01 second figure of this sonic boom simulator. The amount of force necessary to create such a boom and the magnitude of the impact stresses created, restrict the designed mechanism from attaining simulated sonic booms in this range. The simulator will create booms with rise times of .010 seconds throughout the overpressure range. However, at the lowest overpressure position (.5p.s.f.) rise times of .005 seconds are conceivable. The exact overpressure range of these reduced rise times is left to further study.

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APPENDICES

APPENDIX A

TYPICAL STRESS ANALYSIS CALCULATION

Stress Analysis of the Diaphram Shaft Rear Plate

The diaphram shaft rear plate is exposed to a uniform load over its entire surface, equivalent to the pressure within the air cylinder. The aluminum plate is welded to the diaphram shaft and, therefore, its edges are assumed to be fixed. This flat plate stress state is described by Roark (9), case number six. The maximum static stress is the trangential stress at the plate edge. The value of this stress is determined from the equation presented by Roark (9)

$$S_t = \frac{3P_{max}R^2v}{4\ Z^2}$$

$R = 1.5$ inches

$P_{max} = 100$ pounds per square inch

$v = .334$ (aluminum)

$Z = .125$ inches

Therefore:

$$S_t = \frac{3 \times 100 \times 1.5^2 \times .334}{4 \times .125^2} = 3{,}610 \text{ psi}$$

From information presented in Shigley (10), it is found that the impact stress present, when the shaft and plate is suddenly stopped, is equivalent to exactly twice the static stress. Therefore, the maximum stress experienced by the plate is 7,220 psi.

To determine the maximum allowable stress that the rear plate can withstand, the fatigue life of aluminum has to be calculated. The entire sonic boom simulator is designed to run ten times a night, five nights a week, fifty weeks a year, for five years. Noting that the plate is shocked twice per boom, a life of 2.5×10^4 cycles is, therefore, required.

Data presented in Juvinall (2) indicates that for an endurance limit of 2.5×10^4 cycles, the peak alternating stress of aluminum should not exceed 40 kpsi. However, this stress limit is effected by stress concentrations, size, surface finish, and type of load. Therefore, the actual maximum allowable stress is expressed as

$$S_n = S_n' \times C_f \times C_d \times C_s \times C_l$$

S_n = Maximum allowable strength

S_n' = 2.5×10^4 cycle strength

C_f = Factor due to stress concentration

C_s = Surface factor

C_d = Size factor

C_l = Load factor

For the case of rear plate, there are no points of stress concentration, therefore, $C_f = 1.0$. Similarly for all bending stress problems, $C_l = 1.0$. The exact values of C_s and C_d are difficult to obtain for aluminum. However, examination of these factors for other materials indicate an approximate combined factor of 0.75. Therefore

$$S_n = 40 \times 1.0 \times .75 \times 1.0 = 30 \text{ kpsi}$$

To determine a safety factor, the actual stress is compared to the maximum allowable stress. For the rear plate

$$\text{Safety Factor} = \frac{30,000}{7,220} = 4.15$$

APPENDIX B

LIST OF SYMBOLS

a = Acceleration

A_{ms} = Area of moving section

d = Distance moved by moving section

e = Duration of sonic boom

E = Young's Modulus

F = Force

k = Constant

kpsi = 10^3 pounds per square inch

K_1, K_2, K_3, K_4 = Constants

m = Mass

n = Number of gas molecules

psf = Pounds per square foot

psi = Pounds per square inch

P = Pressure

P_{atm} = Atmospheric pressure

P_{max} = Maximum pressure of air cylinder

P_o = Peak overpressure

q = Constant

R = Radius

R_o = Gas law constant

S_t = Tangential stress

t = Rise time of sonic boom

T = Temperature

v = Poisson's Ratio

V = Volume

V_{cv} = Control volume

Z = Material thickness

ΔV = Change in volume

ρ = Density