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Title DESIGN AND CONSTRUCTION OF AN IMPROVED CONDUCTOR VIBRATION RECORDER

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Since the very early days of electric power distribution by overhead conductors, one of the causes for line failure has been aeolian vibration of the span. This phenomenon has been understood for many years, but it is only within the last twenty years or so that economic motives prompted efforts at control. The present extensive development of distant power sources, and increasing demand for economical electric power distribution over wide areas, have reemphasized the need for means to study and control this cause of conductor failure.

Early in 1950, the officials of Bonneville Power Administration approached those of the Oregon State College Engineering Experiment Station with the proposal that an instrument for recording this vibration be designed and constructed. Their need was for a compact instrument which would make a continuous record of all conductor vibration between ten and fifty cycles per second, and having a double amplitude of from 0.025 to 1.000 inch. Both frequency and amplitude of the vibration were to be recorded simultaneously on the same chart. Additional specifications included the range of operating temperatures, unattended clock performance time, etc. An agreement to conduct the necessary research and construct the instrument at Oregon State College was reached in September 1950.

The writer became engaged upon the project in January 1951. Preliminary investigation of adaptable commercial instruments had been made and an Esterline Angus recording DC milliammeter had been procured. It had already been decided to utilize the frequency linear characteristic of alternating current flow through a series RC network. This meter was suitable for such a circuit. A Metron DPDT commutation switch had also been selected to reverse current flow, but no method of compensating for sway of the vibrating conductor as it drove the switch had been devised. This compensating action was obviously necessary since a positive linkage between conductor and instrument needed to be used, and vibratory motion transmitted to the switch at extremes of line sway produced no commutation.

Study of this problem indicated that some form of pendulum might be utilized to establish automatically the needed point of reference for commutation switch motion. Subsequent research showed that a damped seismic pendulum accomplished the desired action. Information obtained through this research permitted incorporation of this device in the final instrument.

During the course of work on the project, several other original devices which contributed to operation of the instrument were developed. Examples of these are: A unidirectional hydraulic meter damper to permit recording of peak frequencies; a

method of treating already ruled chart paper to permit recording with a metallic stylus; and a means of securing glass twine so as to prevent slippage and develop the full strength of the fibers.

Performance of the instrument proves the basic design to be sound. Frequency sensing exceeds specifications, and although the upper frequency limit is slightly less than that specified, the usefulness of the instrument is not thereby impaired. In use, the chart shows a magnified conductor motion trace from which the amplitude of vibration may be read. Frequency of vibration is indicated as a continuous trace of peak frequency of vibration. Recording is simultaneous for the two values, and all other requirements have been met.

DESIGN AND CONSTRUCTION OF AN IMPROVED
CONDUCTOR VIBRATION RECORDER

by

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A THESIS

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I. INTRODUCTION

Wind actuated or aeolian vibration of overhead conductors has been known to electrical engineers since the very early days of high voltage electric power utilization. It has only been since World War I, however, that the damaging effects of this phenomenon have been known. During the period to the present day, a great deal of literature on the subject has been published, and much progress has been made toward a better understanding of the problem and control of the damage.

As in so many other cases, however, where evolution of engineering practice is concerned, development of a knowledge of conductor vibration has been strongly influenced by changing economic and operating conditions. The result is that, although established practice has reached a high state of development over the years of study and research, demands for wider power distribution and greater load carrying ability with economy and reliability have presented further problems. Thus the disclosure of new phases of conductor vibration has kept pace with engineering advancement; constantly challenging the ingenuity of engineers.

Chief among current problems is the selection of means of damage control most suitable for a particular situation. The magnitude of this problem will be appreciated if it is realized that transmission lines traverse miles of infinitely varying country, and each line is specifically designed to perform its individual load-carrying function. Each span of each line is therefore individual to the engineer concerned with vibration damage control. Furthermore, as in other such technical work, the desired end results must be achieved with the minimum expense and engineering effort.

It has long been the policy of leading power companies, and specifically of the Bonneville Power Administration (1) to maintain continuous study and correction of vibration as it occurs upon their transmission system. To further this policy the Bonneville Power Administration developed and has used, since 1941, a recording meter by means of which the field vibration of the conductors may be studied. The recording unit of this meter is shown in Figure 1.

In use, this recording unit is mounted upon the tower, as shown in Figures 2, 3, and 4, along with the equipment required for operating the instrument. This required equipment consists of a storage battery, an anemometer, and other items necessary to properly connect these components, as shown in Figure 3. These additional items are required in the determination of the frequency of vibration by calculation from wind velocity.

Using this instrument and a technique (2) developed for interpretation of recorded vibrations, the Bonneville Power Administration has achieved some degree of success in the control of conductor damage and prevention of service interruption from this cause. As the average age of conductors became greater, however,

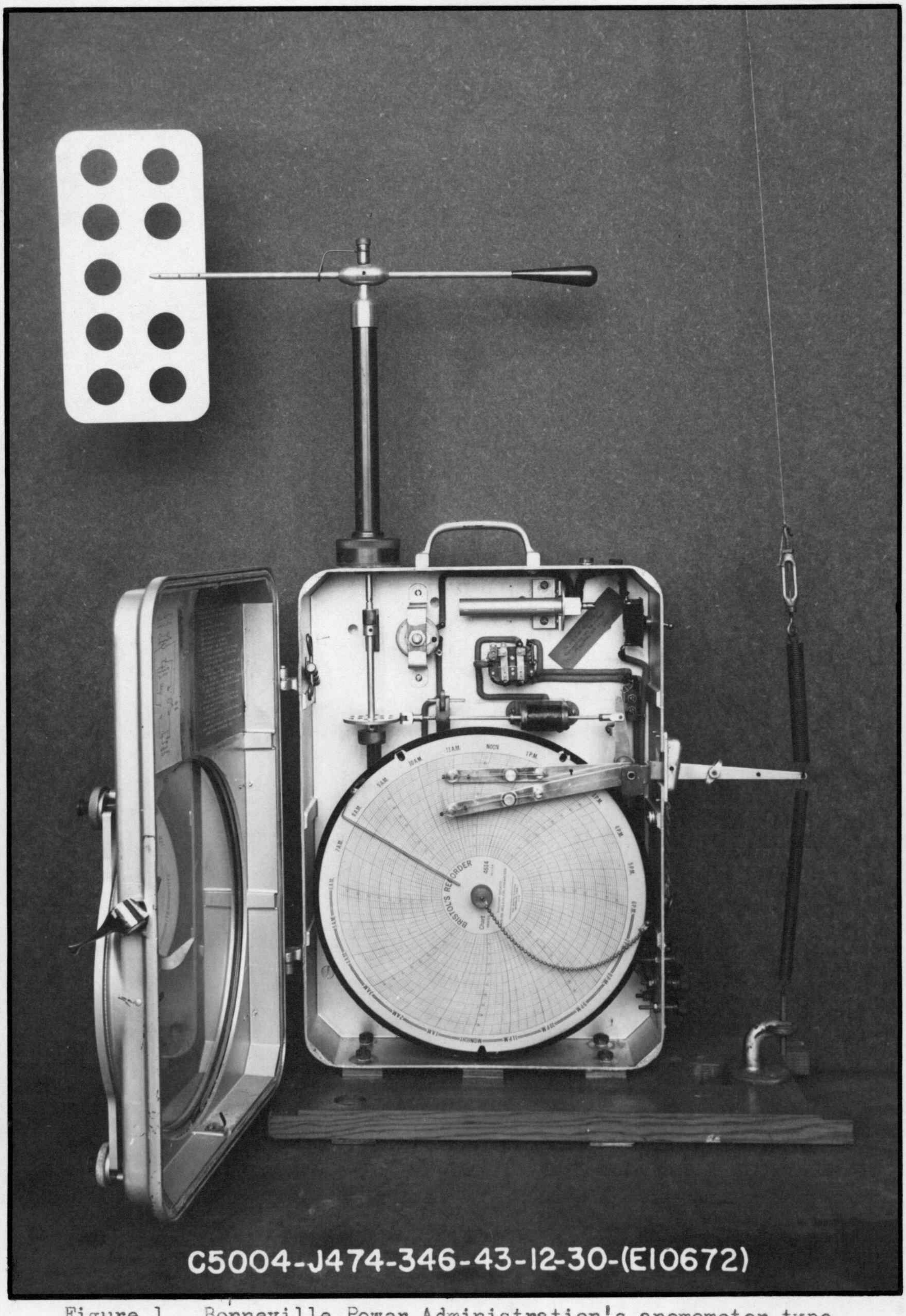


Figure 1. Bonneville Power Administration's anemometer type vibration recording unit.

Figure 2. Installing a vibration recording unit.

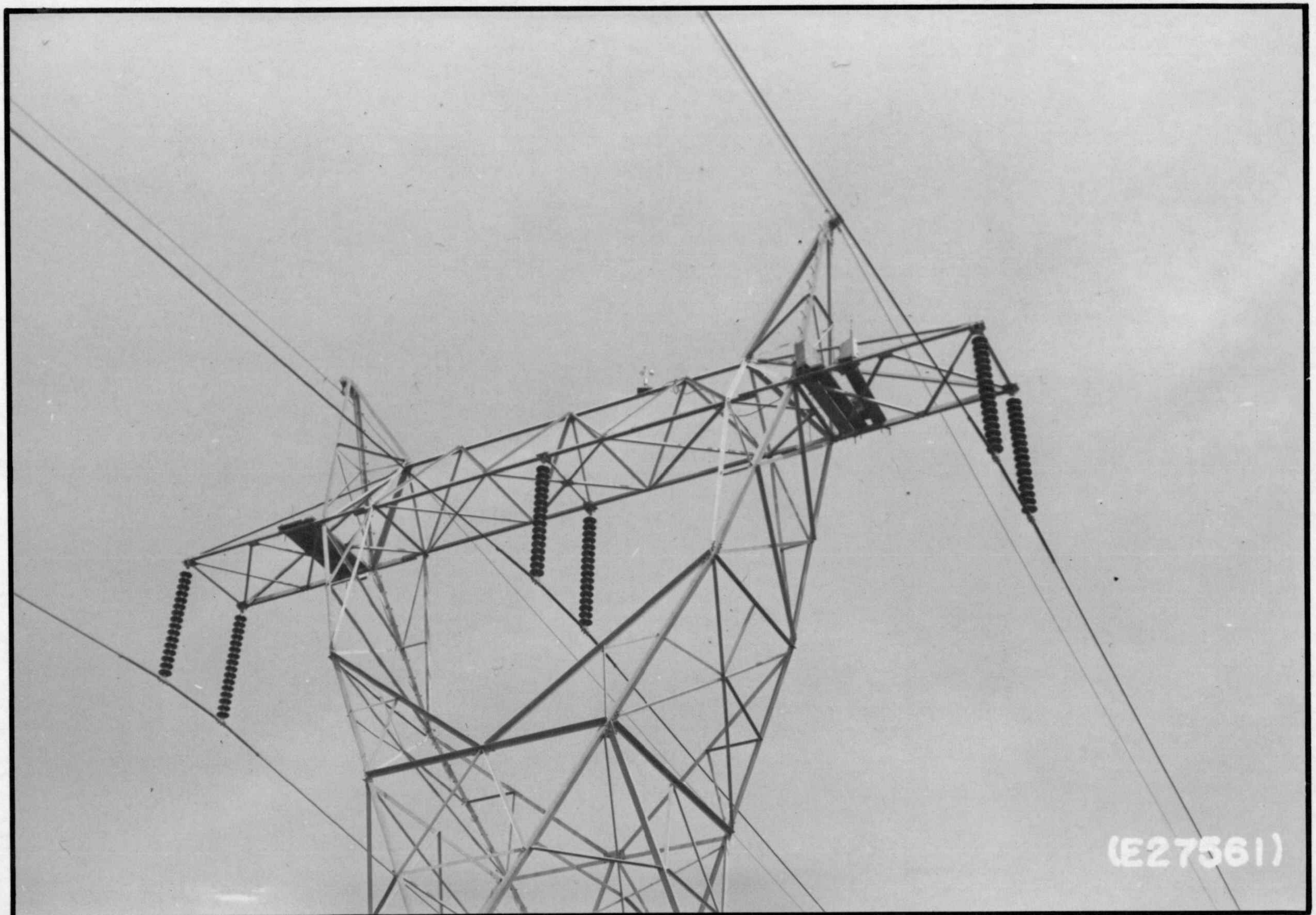


Figure 3. Tower installation of vibration recorder as seen from the ground.

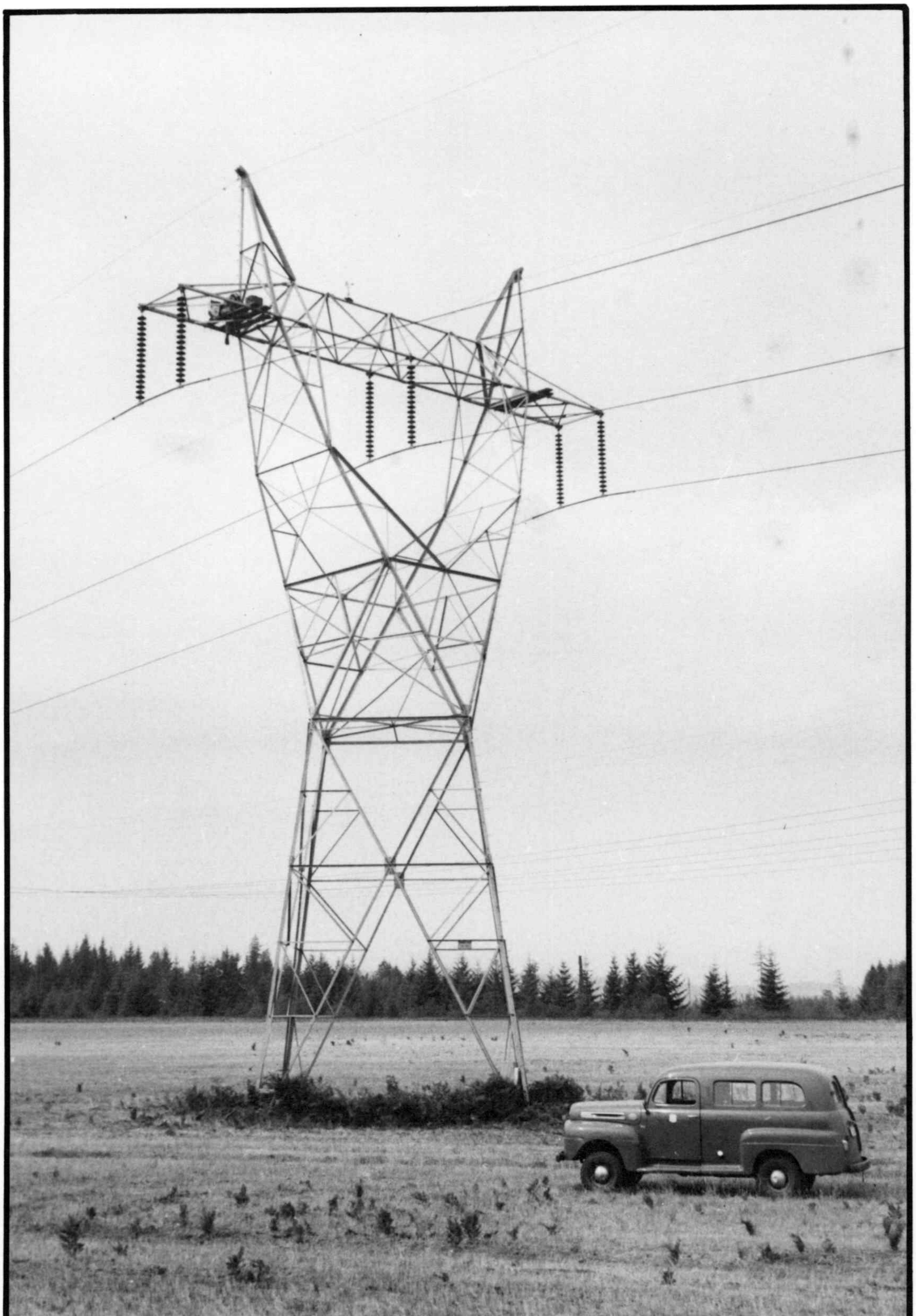


Figure 4. Test tower with vibration recorders installed.

fatigue failures caused by vibration began to be noted. The observation of outer strand failure such as that shown in Figure 5 stressed the need for intensification of the vibration control program and possible modification of methods to allow greater field coverage.

In the course of the resulting review of methods the recorder used for obtaining vibration data came under scrutiny and it was decided that simplification of this instrument could be made to advantage. In collaboration with the Southern California Edison Company of Los Angeles, Bonneville Power Administration officials approached the General Electric Company with the proposal that a new recorder be designed and built which would serve as its predecessor had, but in addition would be very much more compact, and would allow direct reading of frequency without recourse to bothersome calculations. Other desirable features such as unattended operation for a period of several weeks, satisfactory operation in extremes of temperature, and portability were also specified to correct specific weaknesses in the then used methods. As then conceived, the entire control program could be realigned to advantage upon acquisition of a suitably redesigned recorder, and the General Electric Company was therefore urged to proceed with the research and design.

First efforts to obtain the recorder were not encouraging. General Electric estimated that $50,000.00 would be required to carry on the research, that two years' time would be required for completion, and no assurance was given that a successful instrument could be produced. Bonneville Power Administration and Southern California

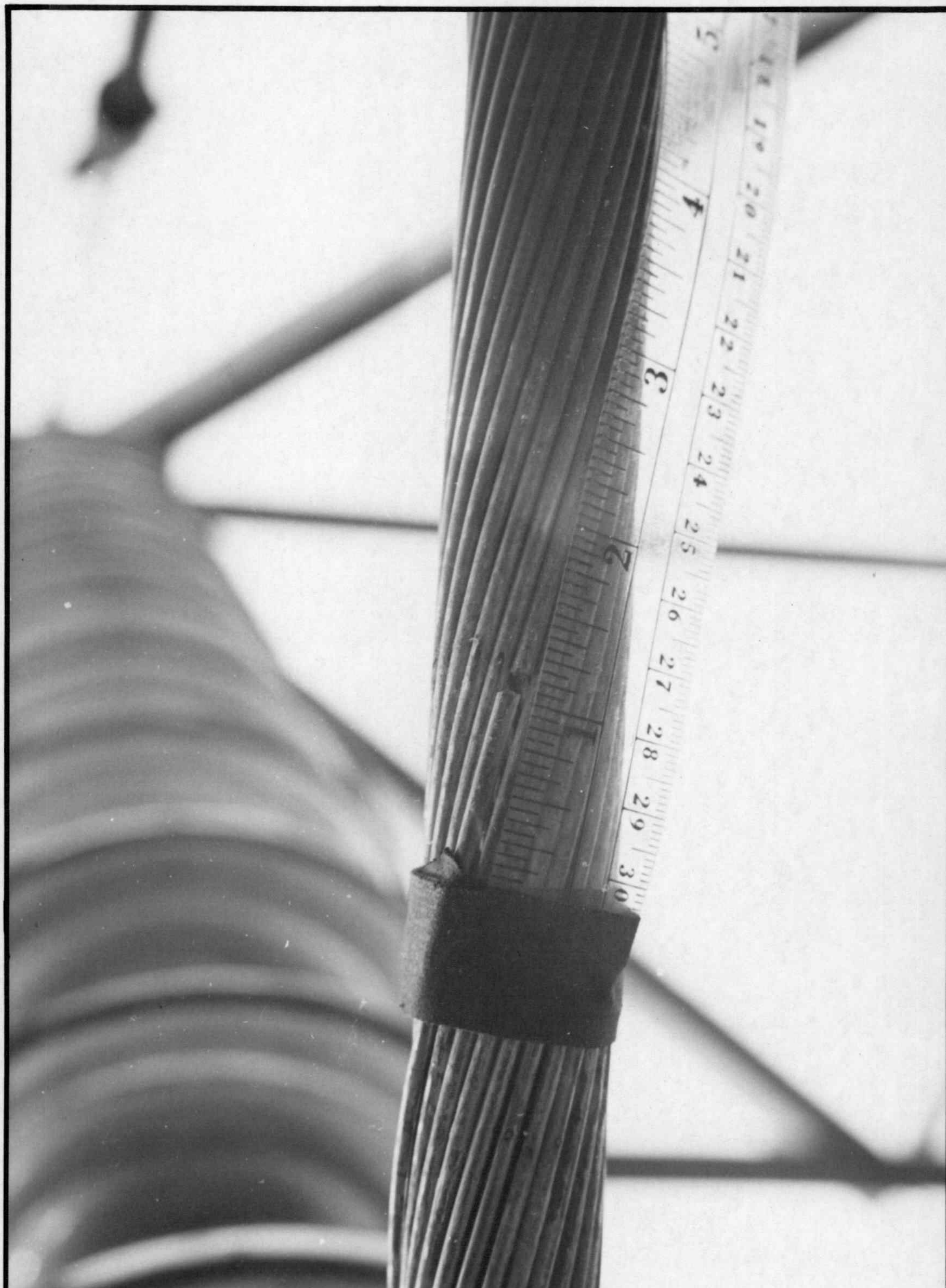


Figure 5. Typical fatigue failure of conductor strand.

Edison officials agreed that these terms were not acceptable.

In June of 1950, the Bonneville Power Administration officials approached those of the Oregon State College Engineering Experiment Station with the same proposal formerly extended to the General Electric Company; to carry out the necessary research, design, and construct the desired conductor vibration recorder. After consideration of the basic problems it was agreed that the work could and would be undertaken at Oregon State College.

This thesis describes the research performed, presents the design and explains the construction and operation of the vibration recorder.

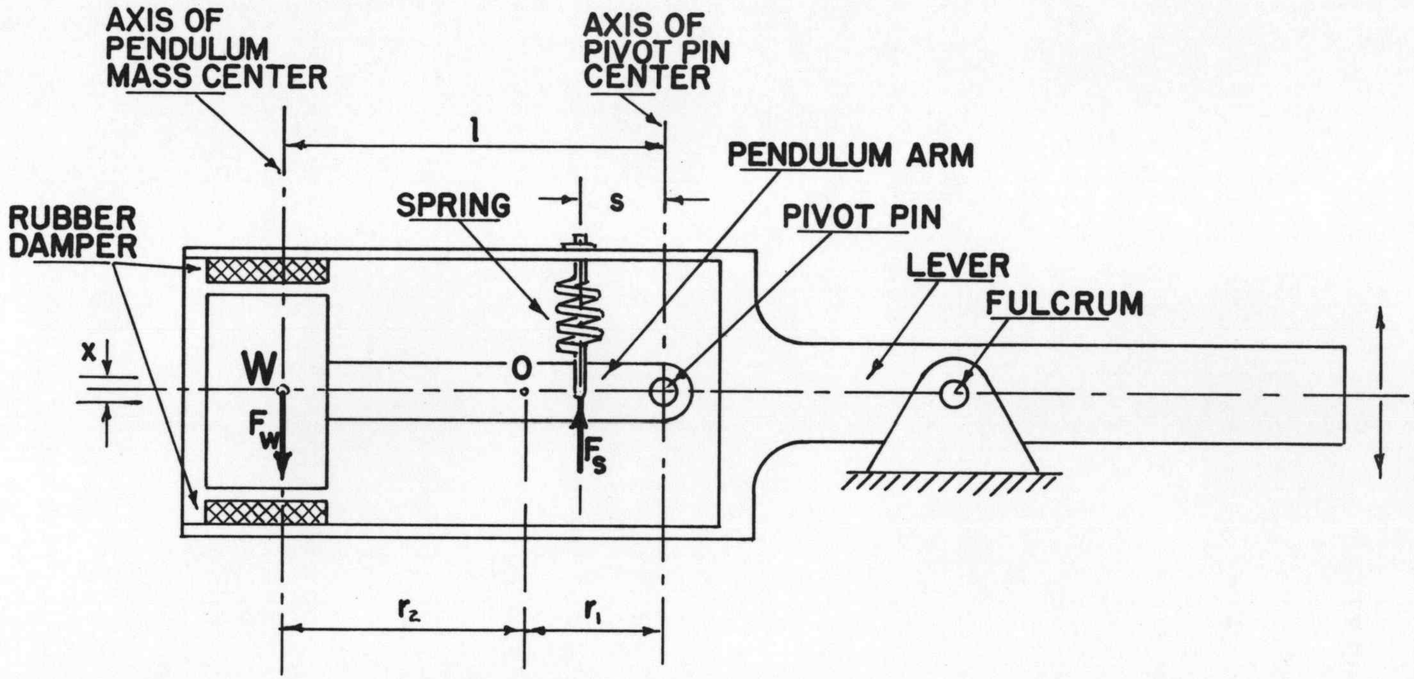
II. THEORY

As originally conceived (3, p10,12,13) the application of the proposed recorder in making field measurements of conductor vibration could be very similar in type to its predecessor, shown in Figure 1. Tower mounting of the recorder was considered feasible, and use of glass twine connection from conductor to recorder was thought to be advisable. Modification was desired however in the use made of motion at the instrument. In addition to producing an amplitude record as before, this motion was to be applied in the direct recording of vibrational frequency. This latter was to be accomplished by mechanical actuation of a DPDT commutation switch in a capacitive reactance circuit containing a series source of electrical potential and a recording milliammeter. The schematic diagram of Appendix Drawing No. E-1-6a shows this circuit. The frequency-linear characteristic of this circuit has been used successfully in tachometer devices in the past, and is currently used by the Metron Instrument Company in their electrical tachometers. The principle was thought to be applicable in this use as well.

Early developmental work based upon the above concept disclosed that although the proposed circuit was applicable, direct actuation of the commutation switch by string attachment was not. This was due to the fact that motion at the instrument was not pure. It consisted of the motion of conductor vibration superimposed on the larger movements caused by swinging of the conductor about insulator-tower tie points, the shortening of the glass twine under wind action, etc. With these larger motions continuously changing the mid-point of switch throw (or point of zero amplitude), proper actuation of the frequency determining circuit was impossible. It was obvious that an additional link was required in the frequency determining chain to automatically compensate for this continuous fluctuation.

Of the various means for accomplishing this compensating action, that considered most applicable was to establish an artificial point of zero motion as in the various "seismic" instruments. This principle utilizes a mass suspended in such a manner that relative motion between it and the vibrating object approaches the actual amplitude of vibration. However, it was not any of the usual applications of this principle which was considered. These had all been rejected in principle in previous consideration (3, p8,9,10). For this application it was necessary to employ relatively smaller mass to avoid loading the conductor excessively and damping out vibration. In any of the usual applications this meant lack of sensitivity to lower frequencies. In that considered, it was believed this limitation would not apply. For the basis of this belief, consider the following analysis:

Figure 6 shows a sketch of the type pendulum proposed for use. It consists of a weight W carried upon one end of an arm, the other end of which is supported by a pin connection, a distance L from the mass center of the weight. The system is held in horizontal equilibrium by action of a spring fastened to the arm a distance s



$$x_S = \frac{s}{l} x_W \qquad F_S = \frac{l}{s} F_W = k x_S = k \frac{s}{l} x_W$$

$$F_W = k \left(\frac{s}{l}\right)^2 x_W$$

Figure 6. Details of pendulum.

from the vertical axis of the arm pin connection. The support for both spring and arm is seen to be upon a lever which rotates through a small angle about a fulcrum. The distance x_W through which the weight W may move is also seen to be limited by action of rubber dampers.

For purposes of analysis the angularity both of lever motion about the fulcrum and arm motion about its supporting pin can be disregarded since angles concerned are very small. Motion of the mass W/g is therefore assumed linear with respect to the lever. In static equilibrium the expressions presented in Figure 6 are seen to apply.

Summation of dynamic forces in terms of those applied at the mass M = W/g produces

$$F_W = Ma = -k\left(\frac{s}{L}\right)^2 x_W = M\ddot{x}_W$$

From which $\omega_n = \frac{s}{L}\sqrt{\frac{k}{M}}$

or $f_n = \frac{1}{2\pi}\frac{s}{L}\sqrt{\frac{k}{M}}$

where ω_n = the rotational natural frequency of free vibration for the isolated spring and mass system, in radians per second.

f_n = natural frequency, in cycles per second, corresponding to

k = spring constant of the spring used, in pounds per inch deflection.

These equations are characteristic of simple spring and mass systems and would in many cases establish the frequency at which system resonance would occur. In the system under analysis, however, this is not the case since for any appreciable motion at a finite value of frequency the resultant motion (4, p7) of the arm pivot point and spring support relative to that of the pendulous mass center is about some instantaneous center O. r_1 and r_2 are the distances from this instantaneous center to the pivot and pendulous mass centers respectively.

From geometry of the system, if motion of the mass is x, then motion of the spring tie point is

$$\left(\frac{s-r_1}{r_2} \right) x$$

Assuming motion of the support to be sinusoidal, spring action is

$$a_o \sin \omega t - \left(\frac{s-r_1}{r_2} \right) x$$

where a_o is the maximum amplitude of the impressed vibration and is the impressed rotational frequency.

Spring force is then

$$k \left[a_o \sin \omega t - \left(\frac{s-r_1}{r_2} \right) x \right]$$

and similarly the force at the center of pendulous mass is

$$\left(\frac{s-r_1}{r_2} \right) k \left[a_o \sin \omega t - \left(\frac{s-r_1}{r_2} \right) x \right]$$

Assuming simple harmonic motion, by Newton's Law

$$M \ddot{x} + \left(\frac{s-r_1}{r_2} \right)^2 kx = \left(\frac{s-r_1}{r_2} \right) k \, a_o \sin \omega t$$

$$x = x_o \sin \omega t$$

$$\ddot{x} = -x_o \, \omega^2 \sin \omega t$$

Substitution of these latter values in the differential equation and simplification results in an expression for the maximum amplitude of the pendulum.

$$x_o = \frac{\left(\frac{s-r_1}{r_2} \right) \frac{k}{M} a_o}{\left(\frac{s-r_1}{r_2} \right) \frac{k}{M} - \omega^2}$$

But at the instantaneous center

$$a_o = \frac{r_1}{r_2} x_o$$

and substituting $\frac{s}{r_2} = \rho$ and $\frac{r_1}{r_2} = \alpha$

$$x_o = \frac{(\rho-\alpha)\frac{k}{M}}{(\rho-\alpha)^2 \frac{k}{M} - \omega^2}$$

but since $\rho = \frac{s}{L}(\alpha + 1)$, amplitude ratio is

$$\frac{x_o}{a_o} = \frac{1}{\alpha} = \frac{(\rho-\alpha)}{(\rho-\alpha)^2 - (\frac{\omega}{\omega_n})^2 (\frac{s}{L})^2}$$

This equation can be written in the form of a quadratic in α, thus

$$\alpha^2 \left[2-3\frac{s}{L} + \frac{s^2}{L^2}\right] + \alpha \left[2\frac{s^2}{L^2} - 3\frac{s}{L}\right] + \frac{s^2}{L^2}\left[1 - (\frac{\omega}{\omega_n})^2\right] = 0$$

Figure 7 shows solutions of this equation for various values of s/L upon coordinates familiar in vibration analysis as those in terms of which resonance characteristics will appear. Note the absence of the usual resonance peak where $\omega = \omega_n$, and also that values of amplitude ratio for practical lower values of frequency ratio are such as to allow damping as proposed.

The analysis and curves of Figure 7 also reflect good instrument sensitivity at low frequencies with reasonable values of pendulous mass. This characteristic is shown by the curves, in the good motion amplification at low frequencies for all values of s/L illustrated. This reflects the domination of acceleration upon performance of this device. Note also that the factor s/L is of significance as a multiplier in establishing natural frequency in terms of pendulum mass. Low values of this factor therefore mean

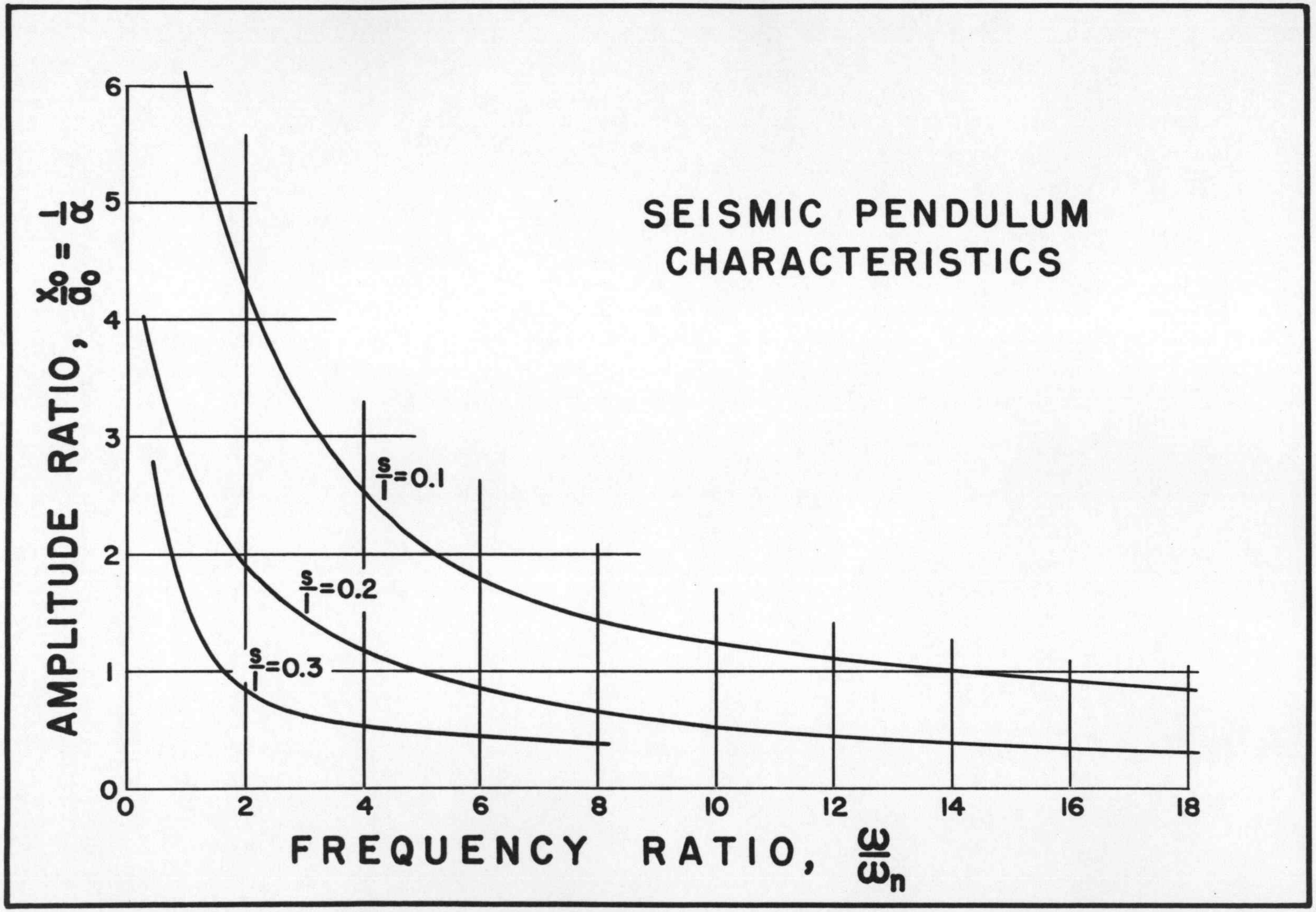


Figure 7. Seismic pendulum characteristics.

that for a given mass the natural frequency of this system will be many times lower than that of a simply supported system.

Use of this elastic system as a means of discriminating against the extraneous motion described as disrupting switch commutation is to mount the switch upon the lever and drive it with the pendulum. By this means, random conductor motion or other nonsignificant motion transmitted to the lever (Figure 6) at A will only change inclination of the lever but not affect operation of either the pendulum or the commutation switch.

III. DEVELOPMENT

Frequency sensing. As indicated in the previous chapter, the need for automatic compensation for frequency sensing was not at first realized. The first mechanism designed to actuate the switch employed a cantilever leaf spring mounted upon the lever in the approximate location of the pendulum pivot of Figure 6. The spring extended parallel to the lever and was coupled at the free end to the commutation switch. This lever-spring mechanism is that shown at the top of Figure 8.

Performance of the spring drive proved to be satisfactory within limits. The spring was designed to have a natural frequency of 70 cycles per second and so no spring resonances were introduced within the range of frequencies from 10 to 50. Also the spring was sufficiently long so that specified amplitudes were easily handled. Difficulties were encountered, however, in zero adjustment of the commutation switch.

It was first thought that since the switch commutated with but 0.005 inch motion, the better than two to one amplification of the lower-spring combination would be sufficient to assure dependable commutative action. Of course this was contingent upon correct initial placement of the switch so that the point of zero amplitude was within 0.04 inch of mid-throw on the switch. Means of accomplishing this adjustment were incorporated in the switch mounting.

First test results were inconclusive due in part to error introduced by slippage of knots used in securing the glass twine to

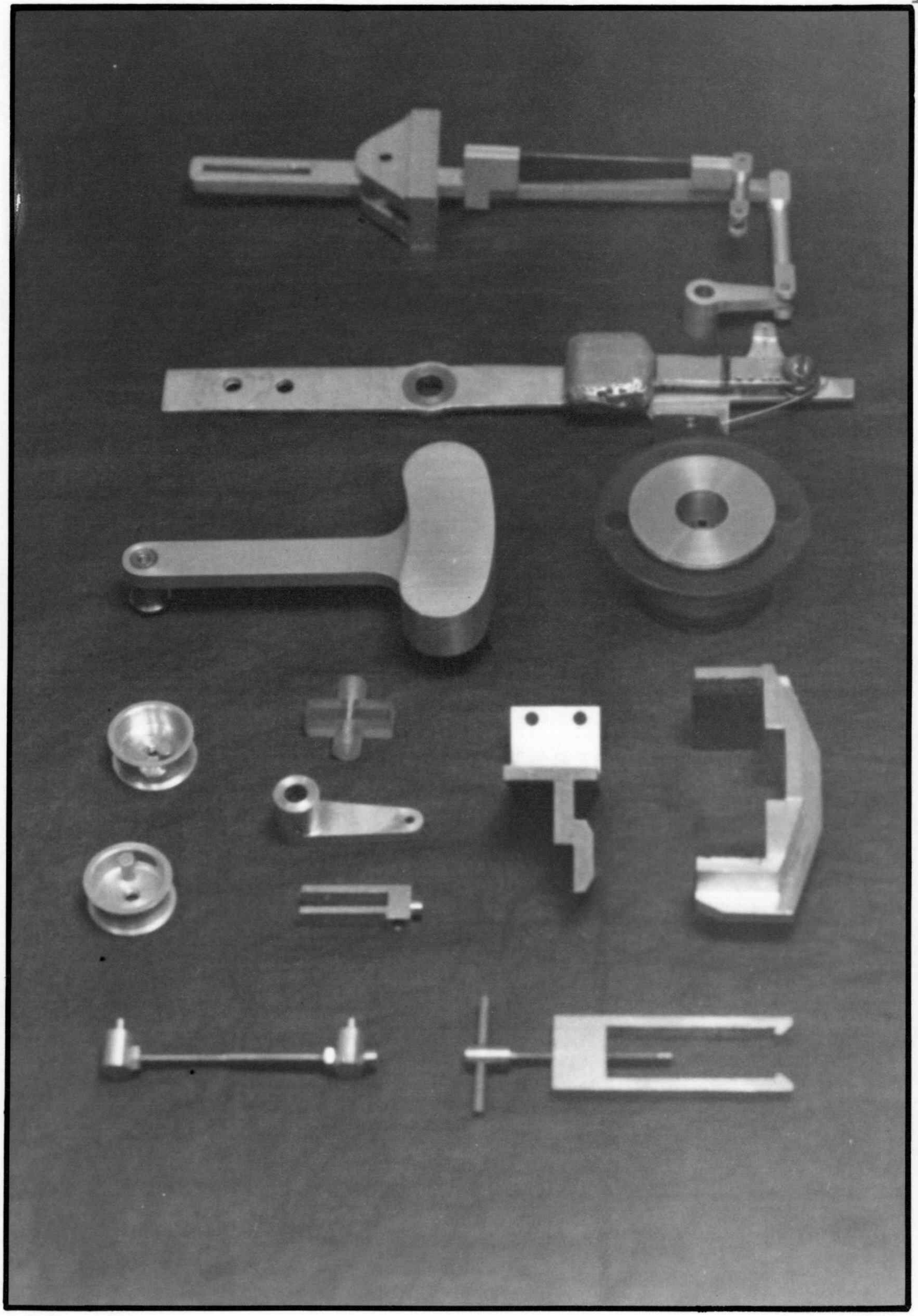


Figure 8. Experimental models of elements and obsolete instrument parts.

both the instrument and the vibration table. At this point in the development it was thought that proper operation would depend upon dimensional constancy of all elements in the vibration transmission system. Time was therefore taken to correct slippage of the fastenings. Several knots known for nonslip characteristics were carefully tested with a Dillon testing machine. Results were as follows:

Knot	Total Slippage in.	Failure Loading lb	Slippage per lb $\times 10^4$ in.
Two Half Hitches	0.19	56.5	33.6
Clove Hitch	0.09	73.0	12.3
Fisherman's Bend	0.07	99.0	7.1

6" Gage Length

Fastenings to 1/8" x 1 3/16" D polished steel ring.

From these data it was concluded that even the best of these antislip knots were not sufficiently reliable for the service intended. Several types of clamping devices were then considered but all were ultimately rejected in favor of a device reminiscent of the familiar principle for holding a bull with a rope by wrapping the rope around a tree. This device is shown in Appendix Drawings No. M-1-1a and M-1-1b, and early versions are shown on the left in Figure 9.

As shown in Drawing No. M-1-1b, a double faced abrasive disc is interposed between the drum and body to which the drum is clamped by screw action. This disc acts as a clutch plate to prevent

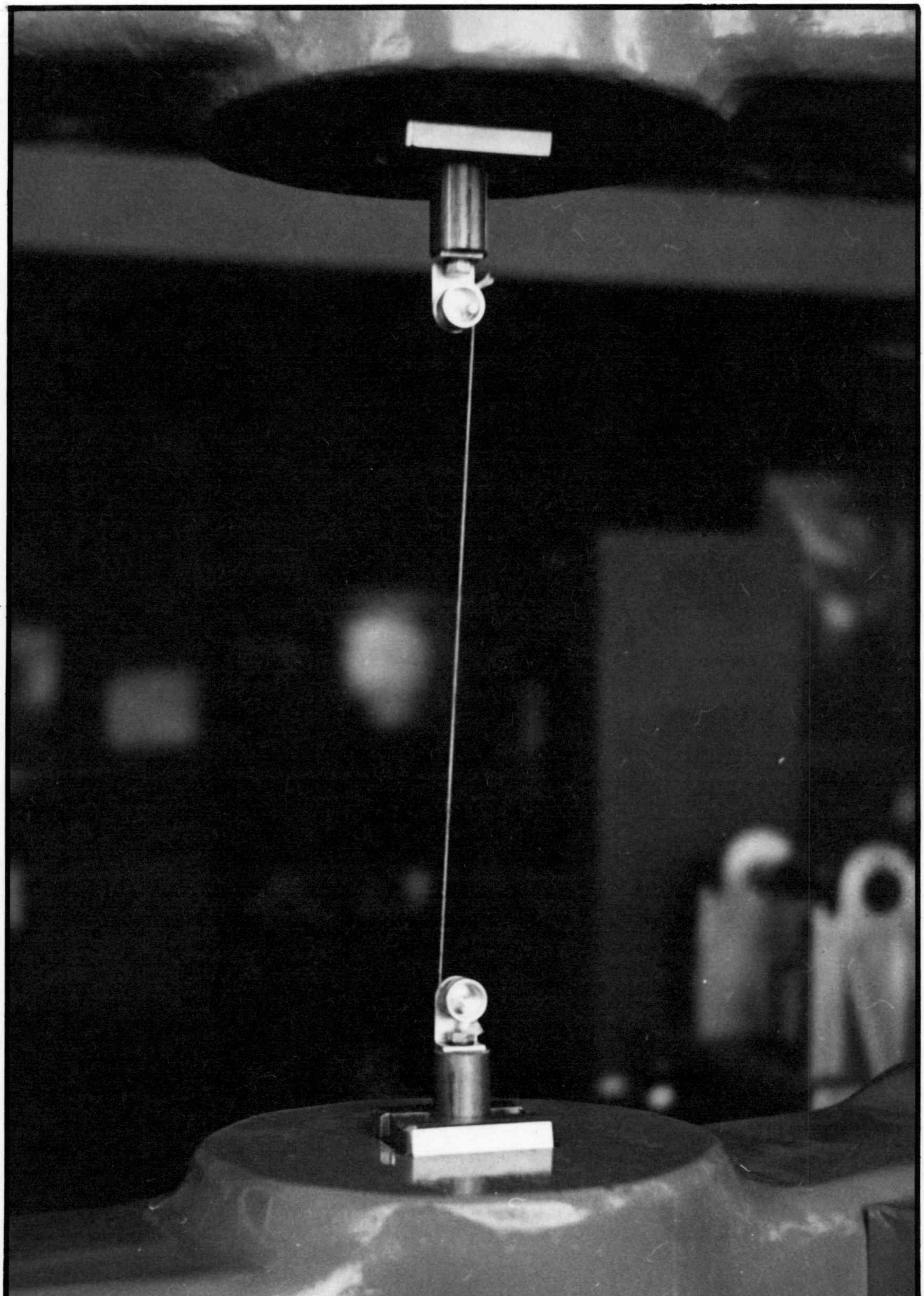


Figure 9. Testing experimental models of the twine connector.

relative motion between drum and body when screw pressure is applied, and yet allows relative positional adjustment when loosened. In use the line is dead-ended by passing a loop through the hole in the surface of the drum and around the pin. Twisting the free end back around the main lead before wrapping the twine upon the drum completes the dead-end. Since security of this type connector is entirely due to friction of twine upon the drum, it is desirable to fill the drum with wrappings as nearly as possible. For obvious reasons it is necessary to place these wrappings upon the drum in a clockwise or screw tightening direction, and the line should lead off the drum as close to the back (friction) face of the drum as possible.

A test was made to determine holding power of these connectors. For this purpose the Baldwin Hydraulic testing machine was adjusted to medium range and set at a loading speed of 300 pounds per minute. The connectors were installed upon the two heads of the machine as shown in Figure 9, and glass twine specimens of 12 inch gage length were used. Precautions cited above were observed, and the following observations were made for each of the two types of glass twine available:

<u>Series A</u>: Tan, heavily waxed twine (thought to be a type no longer used for field application but suitable for laboratory use). Four specimens tested. Breaking load 100 pounds. Failure by strand breakage and unraveling at point of contact with the stationary drum. No slippage was noted, either of the drum or of twine upon the drum, but it was necessary to impregnate the wax

upon the twine quite heavily with powdered rosin to prevent line slippage upon the drum and failure at the dead-end.

Series B: White, lightly waxed twine (this type old and somewhat frayed, but of the type currently used for field application). Four specimens tested; breaking load 200 pounds. Failure the same as that observed in specimen of Series A. No slippage either of drum or of twine upon the drum. It was not necessary to use powdered rosin on twine to prevent slippage upon the drum and failure at the dead-end.

From the results of these tests it was concluded that connector security is good up to the breaking strength of the twine.

With the twine slippage problem no longer of concern, tests were resumed to determine performance of the spring driven commutator switch mechanism. For this test, field conditions were simulated as closely as practicable in the laboratory. The instrument was placed upon a balcony approximately twelve feet above and directly over the vibration table. A suitable length of glass twine was run through a hole drilled in the concrete floor of the balcony and fastened by twine connectors to the vibration table upon the lower end, and the instrument actuation arm upon the upper end. Glass twine vibration dampers of the type developed by R. F. Steidel, and shown in Appendix Drawing No. M-1-2, were placed at the third points to prevent twine resonance. A line-pull balancing spring similar to that shown in Figure 1 was held in place over the instrument by means of a wooden frame, and the spring was adjusted to approximately 40 pounds initial balancing tension.

The instrument was adjusted to mid-amplitude balance prior to start of the test, and sufficient time was allowed between final adjustment and start of the test to assure take-up of initial set in the twine. No appreciable change in adjustment was observed at the end of three hours under a static spring tension of 40 pounds.

During the course of the test, frequency was varied for each of several amplitude settings and performance of the frequency sensing mechanism was noted. Essential agreement with the linear equation $I = 2\pi f CE$ was noted (Standard Electrical Engineering symbols used), but serious discontinuities of frequency trace due to loss of switch commutation were also observed. This was attributed to minor shortening of the glass twine in spite of adequate damping action of the vibration absorbers. Repeated efforts to so adjust the switch placement with respect to the spring drive as to correct this discontinuity failed to produce desired results and it was finally decided that this type mechanism was totally unsatisfactory.

Experimentation was next conducted with a form of seismic pendulum (5, p 75-78) in the continued effort to find some means of reliable commutation switch actuation. The first such model constructed for this purpose is pictured in the next to the top position in Figure 8. It will be noted that this model is similar to the modified seismic pendulum of Figure 6; being merely reversed upon the lever and not provided with rubber dampers. Note also that provision was made for mounting the commutation switch upon the lever in position for driving from a location upon the pendulum arm close to the pivot point. This mounting was made adjustable along the longitudinal axis of the lever so that optimum driving point on the pendulum arm could be determined.

Tests of this device were conducted in the same manner as described above. Results of these tests were encouraging but not conclusive. Commutation was still discontinuous for frequencies above 40 cycles per second at all amplitudes, and behavior was very erratic at all frequencies for amplitudes of 0.025 inch or greater. There was, however, no discontinuity of frequency trace due to lever inclination as occurred formerly. Furthermore, commutative action could be reestablished after cut-out by manual damping of the pendulum motion. This lead to the conclusion that a pendulum of greater mass was required. It was also decided at this time that a torsional spring, similar to that shown in Appendix Drawing M-2-3b, would be more suitable for this application, where exciting motion was itself rotational, than a linear tension or compression spring.

The pendulum utilized in the next model is shown immediately below and to the left of its predecessor in Figure 8. Construction was as a unit from a solid piece of 24 ST aluminum, and it was so proportioned as to place essentially all the pendulum material at a given radius from the pivot point. It was hoped that this would help stabilize forces to the end that need for a bulky pivot bearing would be eliminated. This attempt proved to be unsuccessful due to resultant lack of stiffness in the pendulum arm which made it impossible to maintain desired pendulum alignment. The bearing, a

3/16 inch diameter Oilite bushing 1/4 inch long, failed during the first tests to which the pendulum was subjected.

Performance of this model was encouraging in spite of its early failure. Behavior was good up to a frequency of 40 cycles per second at an amplitude of 0.025 inch, and manual damping produced response beyond this limiting frequency. Bearing failure resulted when amplitude was increased to 0.05 inch.

Up to this stage in the experimentation, significance of the extended response observed to result from manual damping had been misinterpreted. As previously stated, the first observation of response extension prompted design of a pendulum having greater mass. It was thought that increase of mass would lend proportionally greater stability to the system in accordance with the force relationship, $F = Mr\omega^2$, characteristic of such systems. Pendulum damping was not considered seriously until tests of the larger pendulum showed that manual damping was very much more effective in this regard than increase of mass. In attempting to explain this behavior it was theorized that pendulum motion is erratic due to friction of the pivot, dynamic characteristics of the spring, etc. Manual damping of this motion acts to prevent such wandering from theoretical mid displacement position, but in addition the resilience of flesh causes a bouncing action from one finger to the other, and thus restores commutative action. This latter was considered to be of the greatest significance.

A new pendulum was designed, based upon the above considerations, and built to dimensions shown in Appendix Drawing No. M-2-3a.

Also incorporated in this new system was a bracket attachment for the lever which allowed mounting sponge rubber blocks above and below the pendulum. This bracket with rubber blocks attached is shown to the right of center in Figure 8. The pendulum was mounted upon the lever as before, with the pivot pin at the end of the lever and the pendulum mass nearest the lever fulcrum. The commutation switch was mounted upon the lever and held in position below the pendulum mass by means of an attachment shown to the left of the pendulum damper bracket in Figure 8. The switch was driven by means of the elements shown in Appendix Drawing No. M-2-3c. The bent end of the connecting rod was inserted into a hole drilled at the approximate mass center of the pendulum and was fastened at the lower end to the cross-head which drove the switch. The torsion pendulum-balancing spring was adjusted so as to position the statically supported pendulum midway between the two sponge rubber pads when the commutation switch was at mid-throw position.

Testing procedure was modified somewhat for the ensuing series of tests. Since adoption of seismic pendulum means of lever position compensation automatically corrected for loop shortening of vibration transmission string, need for use of this string in simulating field conditions no longer existed. A platform for supporting the instrument immediately over and as close as possible to the vibration table was therefore constructed, and connection was made from table to instrument by a rigid connection. This platform is shown in Figure 23. Whereas former tests required two persons, one to run the vibration table and one to observe instrument performance on the balcony, tests could now be made by one person. This new arrangement also made it possible for the operator to observe the effect of small variations in frequency. This latter permitted close approach to element resonances without the danger of instrument damage which would undoubtedly have resulted from continued use of the former test procedure.

Using the new test procedure, performance of the newly fashioned pendulum was checked. Results obtained confirmed the theory. The pendulum was observed to bounce from upper to lower sponge rubber pad and the range of response was continuous for all frequencies from ten to sixty cycles per second at medium and larger amplitudes. For smaller amplitudes, however, erratic response was observed at all but higher frequencies.

In an attempt to understand this performance so that needed correction could be made, both the softness of the rubber and the amount of free space between pendulum and rubber pad was varied. It was found that both were effective in establishing the low frequency cut-off point, or the lowest frequency at which response occurs. Since study of this phenomenon was obviously not pertinent to the problem at hand further investigation was not undertaken.*

In further attempting to explain and correct erratic performance at small input amplitudes and low frequencies, it was rationalized

*Contemporary with this investigation, R. F. Steidel made use of this principle and explains it more in detail in reference No. 4.

that under these conditions relative pendulum motion would not be sufficient to actuate the commutation switch properly unless pendulum bounce occurred. This could not occur, however, if accelerations involved were too low. Since acceleration varies directly with the amplitude and the square of the frequency, pendulum amplitude needed to be a magnification of input amplitude for adequate low frequency response. Space limitation within the instrument prevented extending the lever on the pendulum side of the fulcrum or use of a lever system to magnify this motion. It was accordingly decided to turn the entire pendulum mechanism end for end on the lever. This resulted in a ratio increase of 3.4:1 for a net amplification of 1.66:1. The arrangement was then that shown in Figure 6, as can be seen in the photograph of the complete lever and frequency sensing element per Figure 10.

Tests of the reversed pendulum showed characteristics essentially as shown in Figure 11 for the final instrument. Some further adjustment of pendulum natural frequency was required to produce the response shown in the vicinity of the knee of the curve, but this was routine and minor. The response region is the entire area above and to the right of the curve. The 0.239 G acceleration curve has also been shown, as a dotted line, to illustrate how closely the performance curve matches that of constant acceleration. This is due to the fact that the mass-damper system maintains a constant level of kinetic energy, as implied in discussion of theory. Only the lower end of both amplitude and frequency range has been

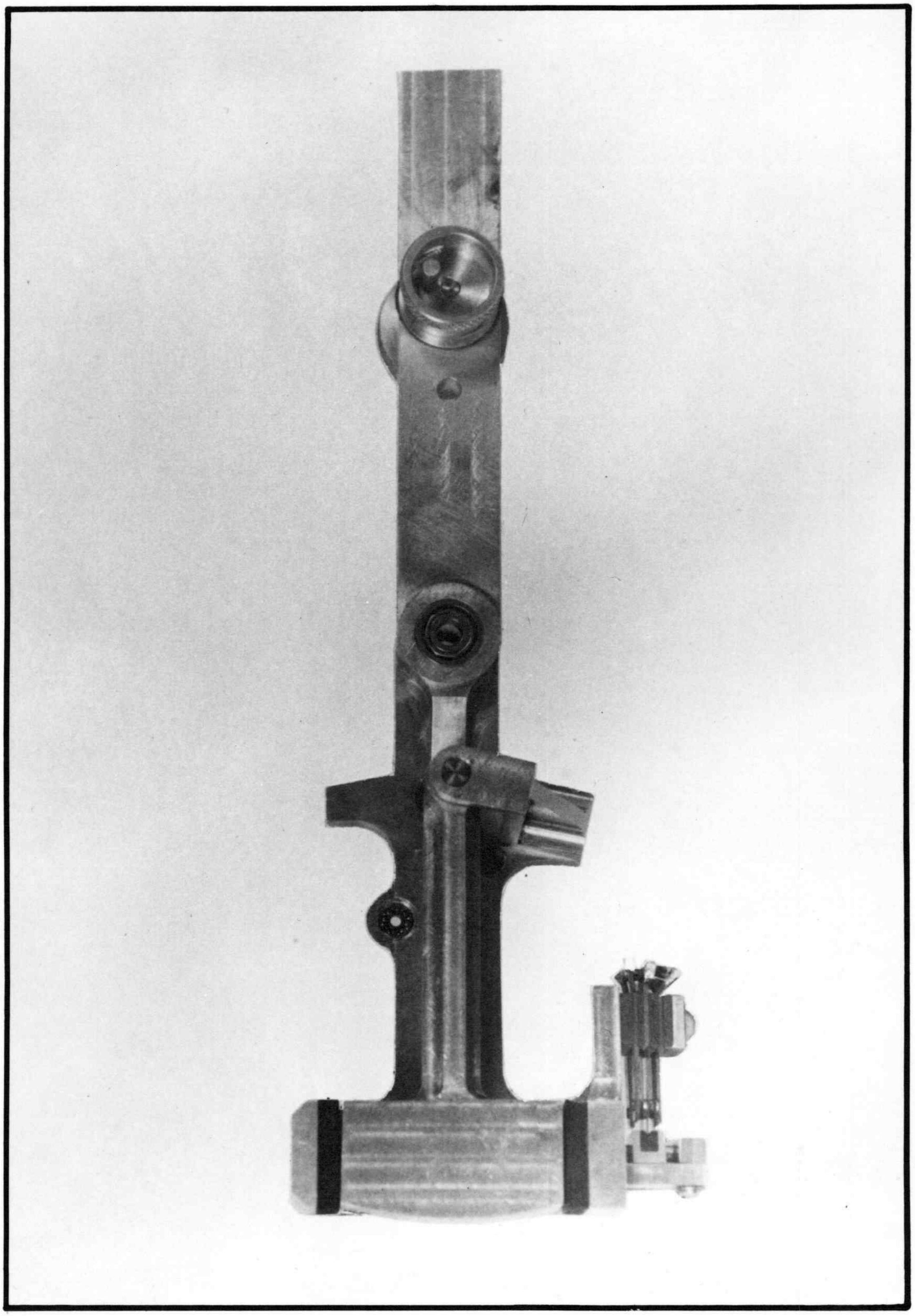


Figure 10. Actuation lever and frequency sensing element.

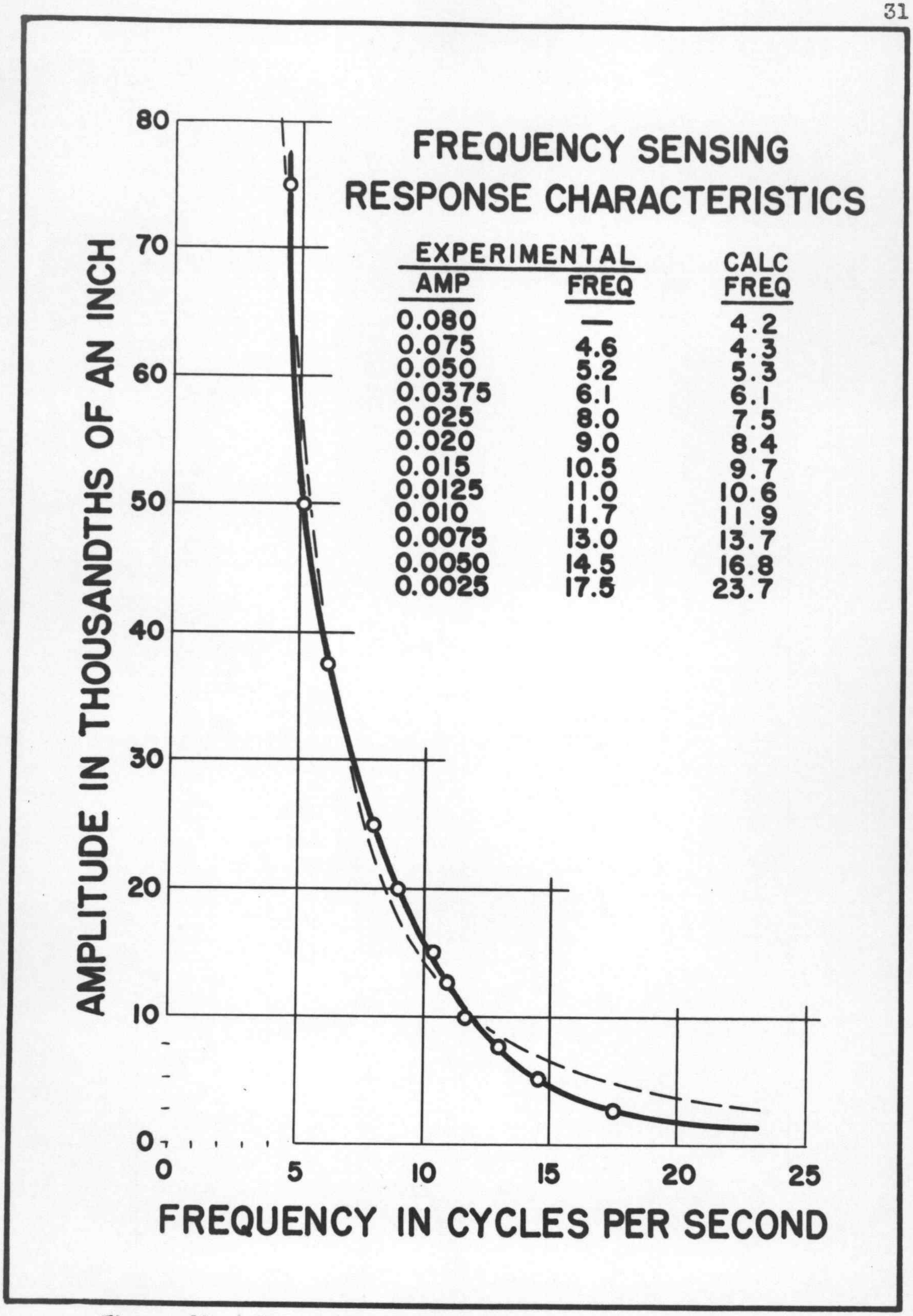


Figure 11. Frequency sensing response characteristics.

shown in Figure 11 for better curve definition. The curve is asymptotic to four cycles per second and 0.0005 inch of amplitude for values of frequency and amplitude beyond the range shown.

The group of photographs comprising Figures 12, 13, 14, and 15 show the integration of the actuation lever and frequency sensing element into the assembled instrument. Note that assembly has been made within a lucite case to make possible inspection of interior parts. The service case is of course of more durable material.

Amplitude train. Original instrument specifications called for an amplitude train magnification of between 15 and 20. The first lever constructed, that shown at the top of Figure 8, was accordingly designed for an amplitude magnification of 17.9, based upon a maximum input amplitude of 0.125 inch. It was subsequently decided that input amplitudes up to 0.500 inch would need be recorded, and motion magnification of 4.0 to 4.5 would suffice. The reduction in magnification was necessary because the six inch Esterline Angus chart had but 4.5 inches of usable chart width. This was an improvement over the 1:1 ratio of the older instruments, however, and specifications were changed accordingly.

The basic requirements of the amplitude transmission mechanism were to properly magnify conductor motion imparted to it and to change the plane of motion from the vertical to the horizontal. This latter was of course necessary in adapting the Esterline Angus instrument for use, as the recording plan of this instrument is horizontal. Of

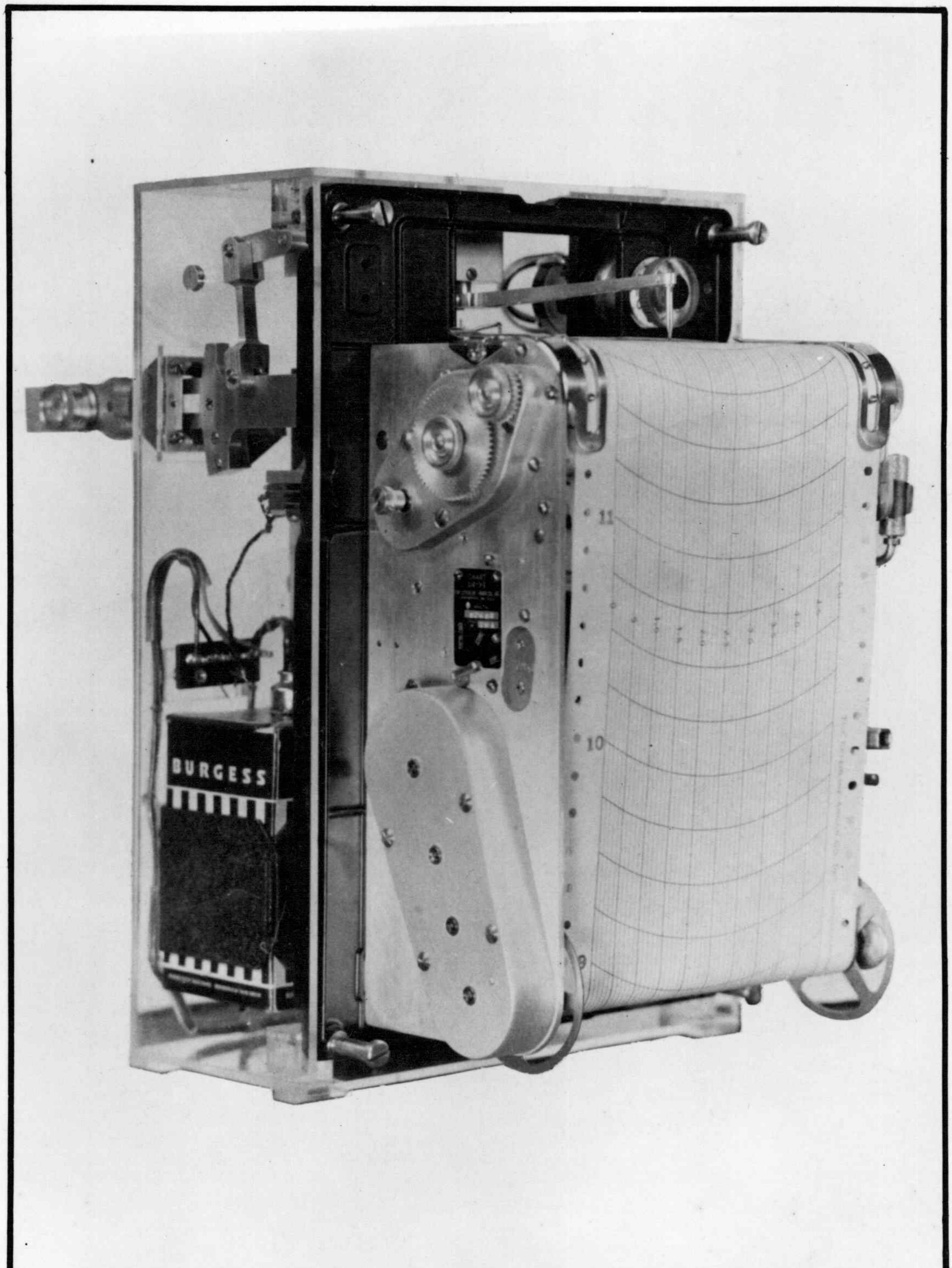


Figure 12. Left front view of assembled instrument.

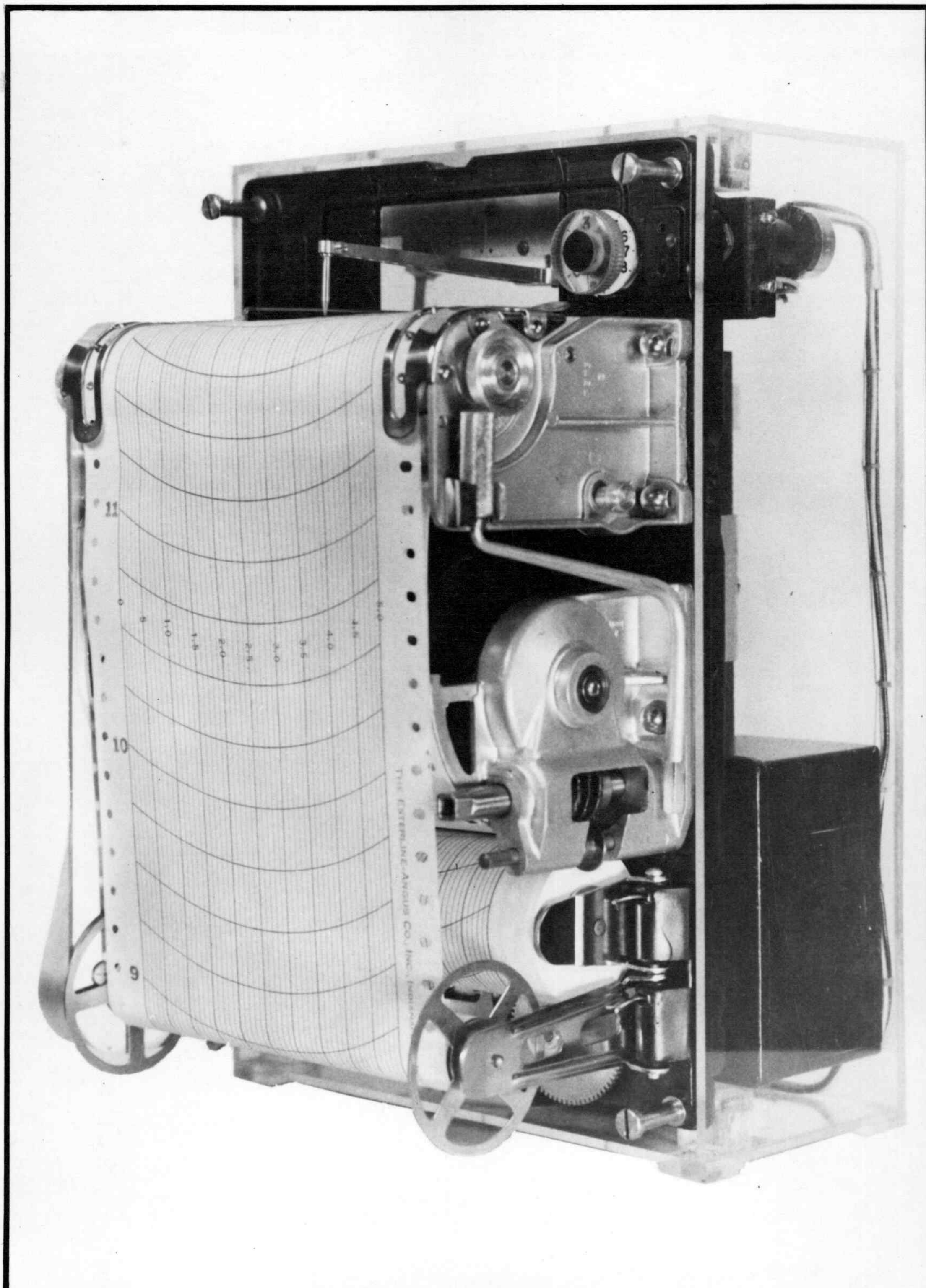


Figure 13. Right front view of assembled instrument.

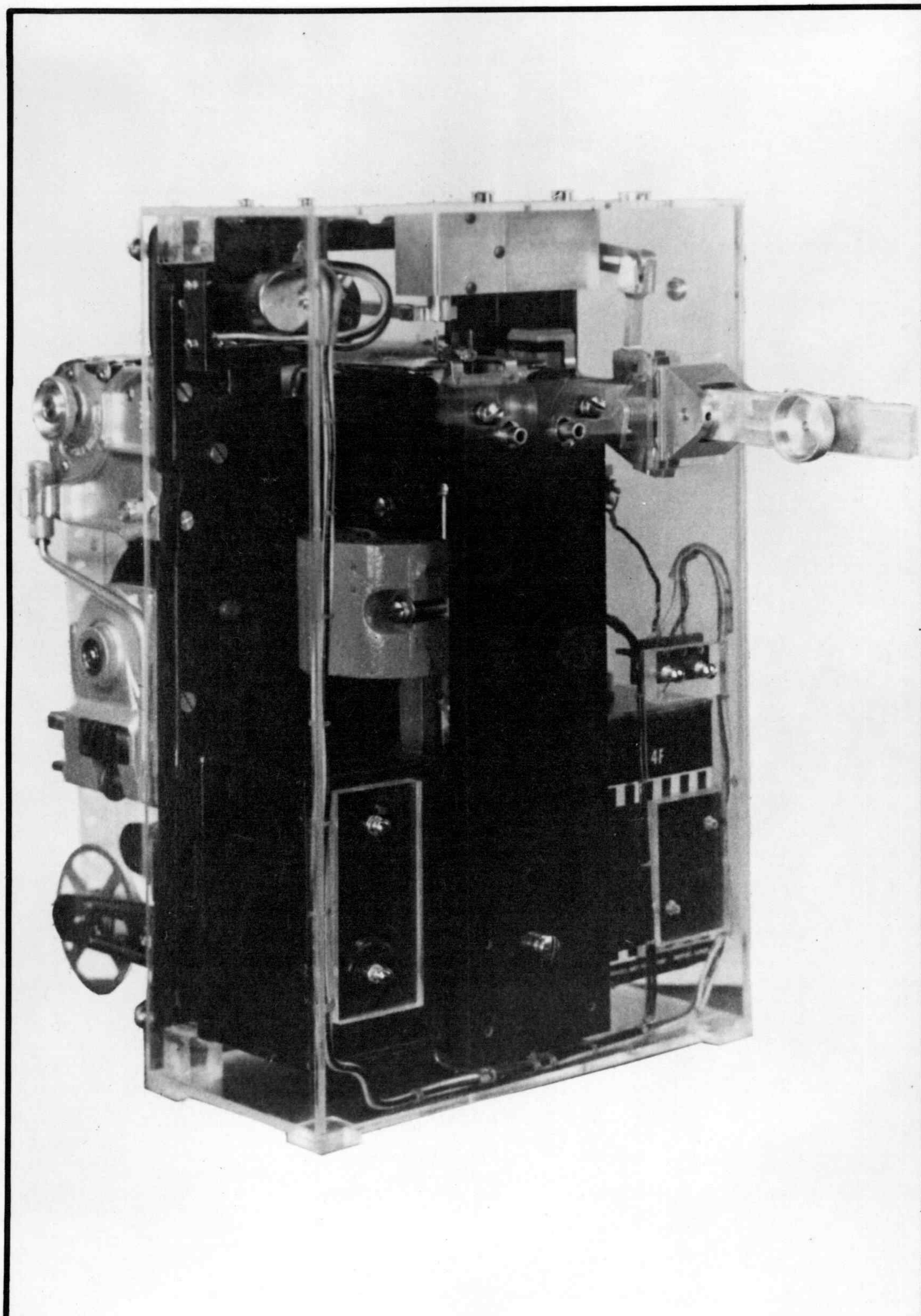


Figure 14. Left rear view of assembled instrument.

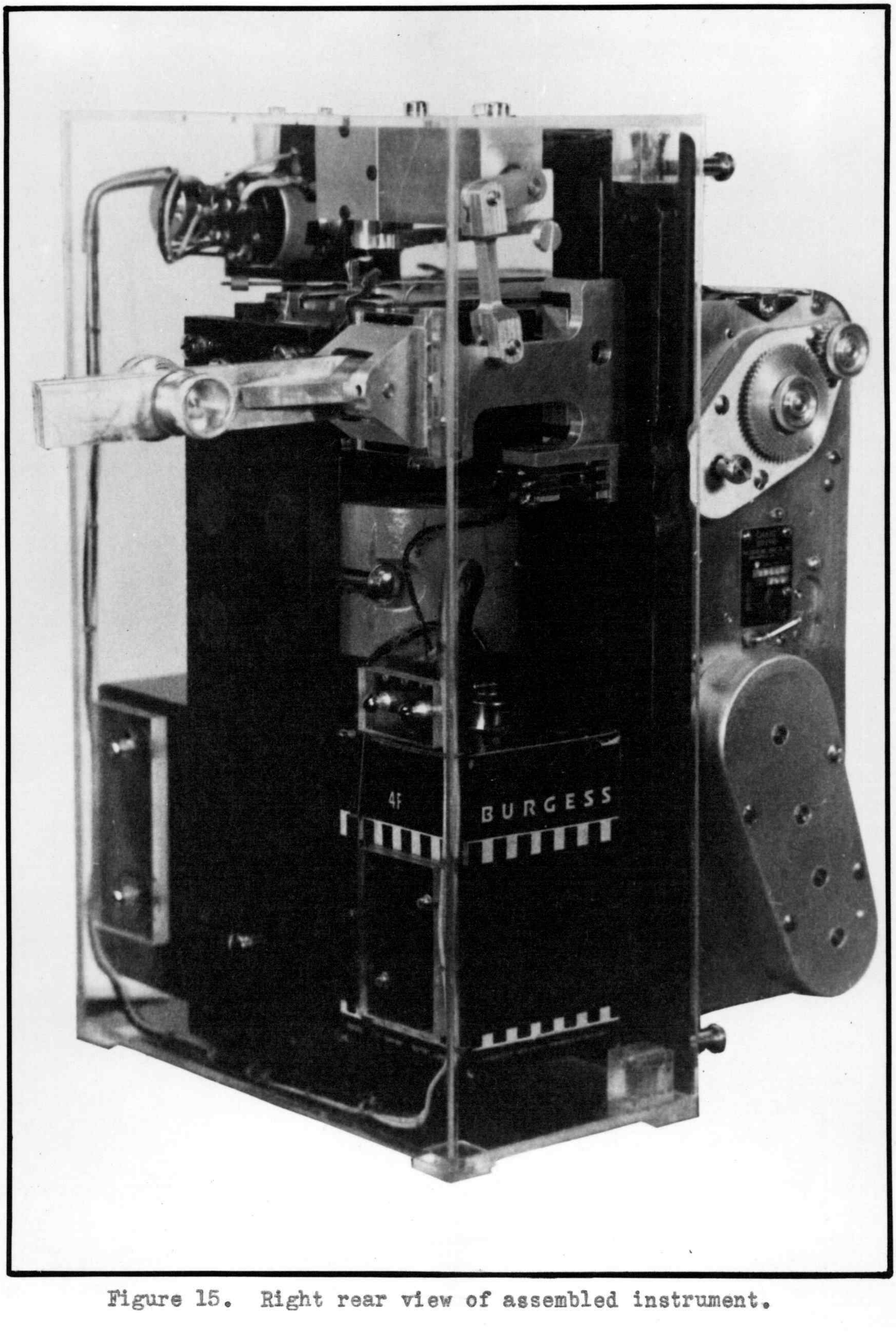


Figure 15. Right rear view of assembled instrument.

the various means available for accomplishing this function that thought to be the least complex was a simple leverage magnification and use of miter gears for change of planes. Space limitation within the case also influenced the making of this decision and possibilities for unit construction were also considered.

The first amplitude transmission mechanism tested was a mock-up of that ultimately used. Elements were essentially the same as shown in Appendix Drawings No. M-2-4a, M-2-4b, and M-2-4c except for lever and connecting rod proportions. These were as required to give the larger amplification ratio initially specified. Parts were improvised from an available supply of war surplus stock. The test was incidental to that of the spring actuated mechanism first developed for driving the commutation switch. This mock-up was of course not unit assembled as shown in the referenced drawings.

Test observations showed the applicability of this simple amplitude transmission mechanism. Some element resonance was noted, but offending members were identified and the condition corrected. Basic information required for design and construction of the mechanism shown as a unit in Figure 16 and as assembled in Figure 17 was thus obtained from the single mock-up. Some further analysis was required however for development of the amplitude recording arm shown in Appendix Drawing No. M-2-4c and in previously referenced figures.

It was recognized that the amplitude recording arm would require some individual treatment. Proper performance of its function

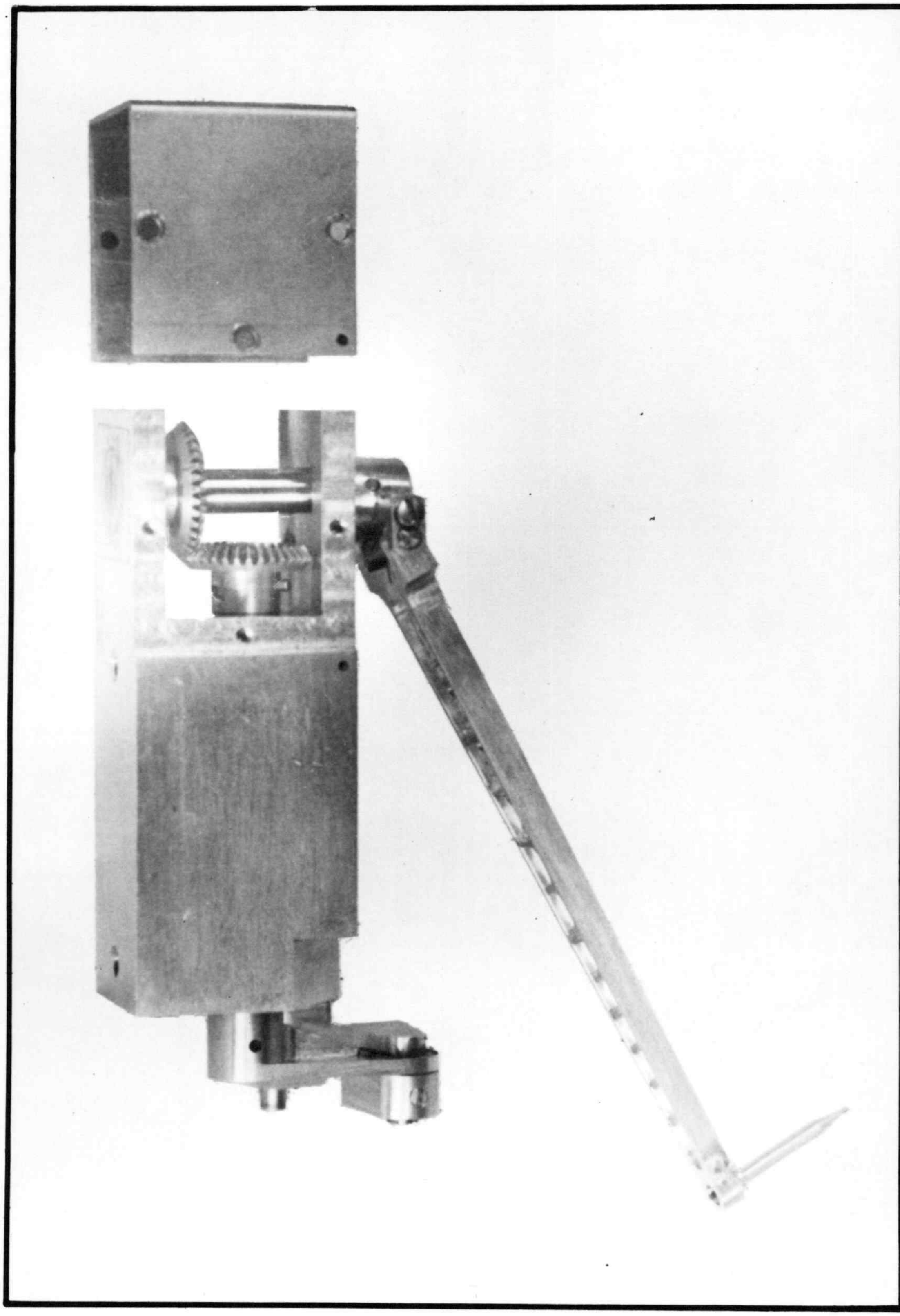


Figure 16. Amplitude team unit with gear box cover removed.

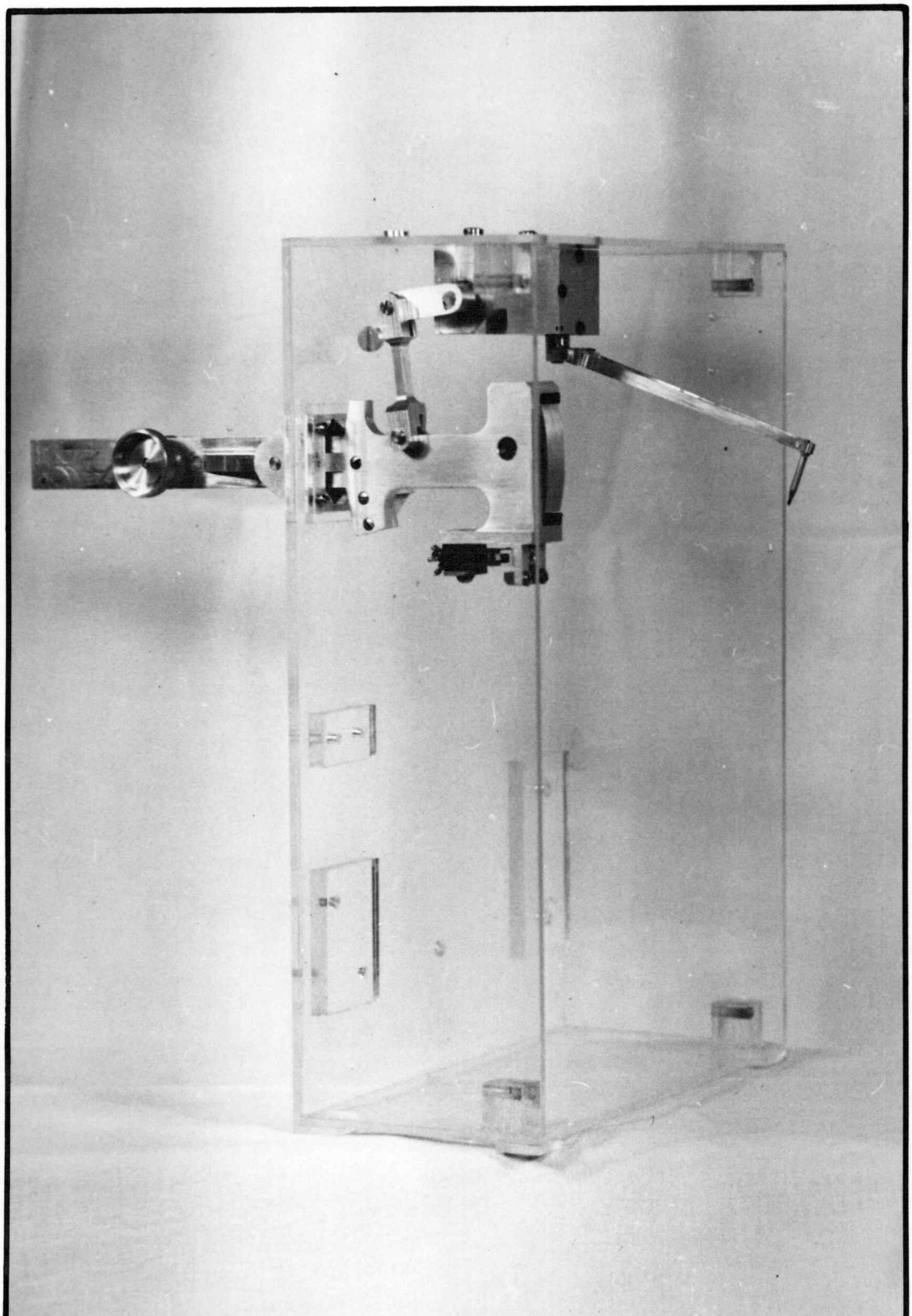


Figure 17. Actuation lever and amplitude train assembly.

demanded an unusual combination of qualities. It must have negligible mass, particularly at its free end, so that inertia forces would be small. In addition it needed to have sufficient stiffness in the horizontal plane to avoid arm resonance. Treated as a part of the amplitude transmission system, arm inertia could not be so large as to overload gears, pins, etc, but stiffness needed to be sufficient to prevent arm resonance which could also overload supporting elements of the train. Since the effect of arm inertia was so dependent upon properties imparted to the train by gears, shafts, and other elements, it was decided to make this the dependent variable in the design determination. Arm stiffness for desirable resonance minimization was thought to be a more easily controlled variable.

Analysis of the arm treated as a beam (6, p25-31) indicated that an arm approximately as shown in Appendix Drawing No. M-2-4c could be expected to have resonance considerably beyond the range of the instrument. These general dimensions were therefore selected and an arm built, but without the lightening holes shown in the drawing. A marking pencil and set screw for clamping the pencil and lead in position per Appendix Drawing No. M-2-4d were also provided. These latter were first constructed of aluminum alloy instead of magnesium as indicated upon the drawing. The arm was then secured in position upon its spindle and the pencil secured in place on the end of the arm. The assembly was thus made ready for testing as a recording element with the instrument.

Testing procedure was to record at a given input amplitude and note the increase of recorded amplitude as frequency was increased. Since resonance of other elements in the amplitude train was known to have been corrected, the increase of recorded amplitude with frequency could only be due to inertia of the recording arm and the spring constant of the system. Substitution of this knowledge into the expression for resonance of such systems (4, p61) permitted calculation of a resonant frequency. That calculated for data taken during this first test showed the resonant frequency of the system to be 55 cycles, a value considered to be much too close to the instrument's operating range for safety.

The low value of resonant frequency noted indicated that considerable recording arm mass needed to be removed. Fabrication of the arm itself from magnesium was considered, but it was decided to first lighten the aluminum arm as much as possible and retest. The pencil and set screw were, however, made of magnesium at this time. Retest showed a vast improvement. The resonant frequency had been increased to 75 cycles per second. This was considered to provide a safe margin above the operating range and so the arm was left in this form. Final dimensions are those shown on the drawings.

<u>Recording medium</u>. Since recorder specifications called for satisfactory operation at temperatures ranging from -30 to +120 F, fluidity of recording ink under all conditions of operation was

of concern. Early investigation showed that comprehensive information of this nature was not available and so Mr. S. E. Graf, Research Engineer of the Oregon State College Engineering Experiment Station, undertook the test of available commercial inks.

Eight specimens of ink in addition to methyl alcohol and water were placed in a rack mounted eight-inch test tubes filled to a height of one inch. The rack was then placed in a freezer and held for twenty-four hours at -20 F, after which the specimens were removed and fluidity of each noted and recorded. These data are given in Figure 18, together with those of subsequent tests.

New samples of the same specimens were then placed in the test tubes as before and the rack was placed in an electric oven the temperature of which was thermostatically maintained at 120 F $\pm$ 2 F. Evaporation rates were measured in terms of linear rate of liquid level decrease in the tubes. Starting level of all specimens with the exception of methyl alcohol was one inch. That for methyl alcohol was two inches because of its rapid evaporation rate. The test was continued for a period of seven days and readings were taken at twenty-four hour intervals.

Results of these tests are tabulated in Figure 18, and curves plotted from these data are shown in Figure 19. Note that only one of the seven inks tested at -20 F remained liquid, and that this ink, Taylor Red Recording Ink, had a very high evaporation rate during the first 50 hours at +120 F. Such performance indicates the presence of a low vapor pressure diluent which provided protection at low

Figure 18

Evaporation at 120 F

Type Ink	Sample No.	Condition at -20 F	24 hr T.T. Lev dec in in.	24 hr Per cent decrease	48 hr T.T. Lev dec in in.	48 hr Per cent decrease	72 hr T.T. Lev dec in in.	72 hr Per cent decrease	96 hr T.T. Lev dec in in.	96 hr Per cent decrease	124 hr T.T. Lev dec in in.	124 hr Per cent decrease	148 hr T.T. Lev dec in in.	148 hr Per cent decrease	168 hr T.T. Lev dec in in.	168 hr Per cent decrease
GE Blue Formula #2	1	Sol	0.10	10	0.22	22	0.35	35	0.44	44	0.54	54	0.64	64	0.67	67
Taylor Red Recording	2	Liq	0.23	23	0.38	38	0.48	48	0.55	55	0.59	59	0.61	61	0.64	64
Tag Red #17910	3	Slushy to Sol	0.10	10	0.23	23	0.33	33	0.44	44	0.52	52	0.56	56	0.59	59
Tag Purple #17910	4	Slushy	0.04	4	0.17	17	0.27	27	0.34	34	0.43	43	0.47	47	0.52	52
GE Red	5	Sol	0.16	16	0.29	29	0.45	45	0.55	55	0.68	68	0.76	76	0.80	80
Palmer Red	6	Sol	0.14	14	0.30	30	0.43	43	0.56	56	0.68	68	0.76	76	0.82	82
Brush Red	7	Sol	0.15	15	0.28	28	0.42	42	0.54	54	0.67	67	0.77	77	0.82	82
Esterline Angus Red	8	Sol	- - - - - - - - - - - - - - Received too late to subject to evaporation test - - - -													
Methyl Alcohol	9	Liq	1.51	75.5	- - - - - - Discontinued test due to too rapid evaporation - - - - -											
Tap Water	10	Sol	0.16	16	0.33	33	0.45	45	0.61	61	0.74	74	0.84	84	-	-

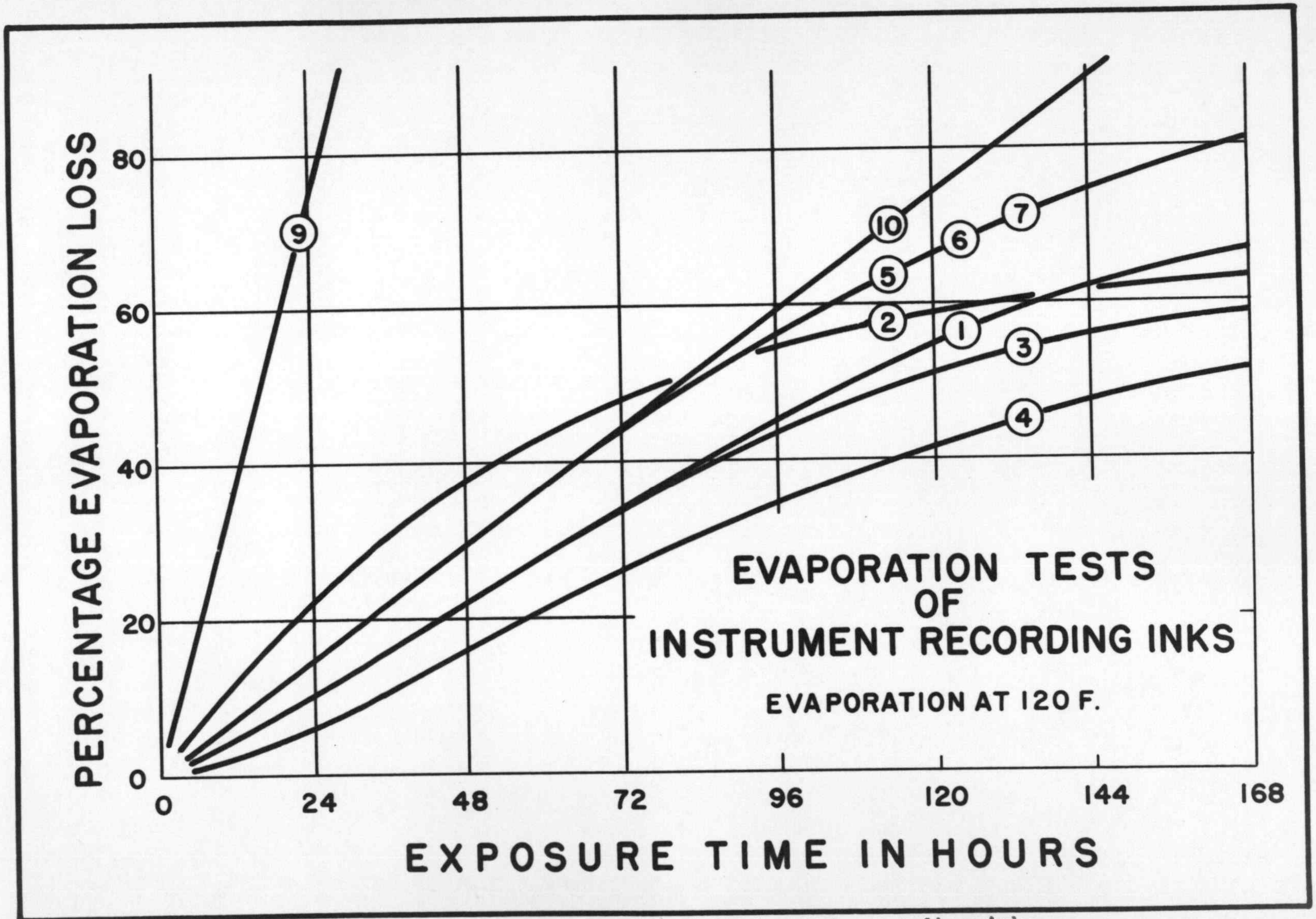


Figure 19. Evaporation tests of instrument recording inks.

temperature but boiled off at elevated temperature. This ink would lack this protection if again subjected to low temperatures, and would freeze as the others did. Lesser degrees of this same characteristic were displayed by other inks tested.

From these tests it appears that none of the samples would perform well throughout the entire temperature range specified. Samples are included in the lot tested which would serve in either the low or the high range, however, with addition of either methyl alcohol or water as a diluent. Methyl alcohol would serve to prevent freezing, and water would prevent too rapid evaporation at elevated temperatures. It is noted that Esterline Angus recommends this practice in use of their ink (7, p70). Adoption of this practice would appear to offer the best available solution to the problem both in consideration of the characteristically gradual temperature variation in the Northwest and the economy and convenience of using only one basic ink diluted to suit the prevailing temperature.

Use of ink as a recording medium was contemplated for the frequency trace, in compliance with Esterline Angus recommendations, but early experimentation showed the unsuitability of this medium for the amplitude trace. Trial of the Brush Fineline recording pen in this service was disappointing due to the high accelerations of the amplitude recording arm. At an input amplitude of 0.05 inch and a frequency of 50 cycles per second, acceleration of the pen was over 50 G. Since recording was required under such conditions and the Brush pen tried was known to have the smallest ink flow of any commercially available, use of ink for amplitude recording was abandoned.

The need for very small mass at the end of the amplitude recording arm, as previously described, seriously limited the range of recording medium selection. Ball point pens were tried at one stage of the investigation but were rejected because of a lack of dependability and the mass of the ink and ink container. Best prospect for solution of the problem was thought to be use of sensitized recording paper in conjunction with a very small metal stylus for marking. A canvas of manufacturers was therefore instigated to find suitable recording paper and to learn, if possible, of methods employed in similar applications.

Replies to inquiries were not very encouraging. Firms which produced sensitized paper stated that their process was only applicable to paper of such heavy weight as to prohibit use in this application. Furthermore the sensitization process did not allow treatment of ruled paper without damage to rulings, and sensitized paper could not be ruled without damaging ruling equipment. Since light weight ruled paper was required in this application to meet unattended operation specifications and to allow proper interpretation of recorded data respectively, the above findings were discouraging. It was decided to not abandon hope for use of this recording medium, however, until after experimental trial of various means of sensitizing already ruled paper.

Water-glass silication was the first sensitization method tried. All of the bad results outlined above were found to be inherent in this process. Rulings were blurred and the paper was wrinkled by the wetting required for proper impregnation of paper fibers. The treated paper could be written upon with any soft metal such as silver or brass, however. These results, although unsatisfactory in important particulars, prompted analysis of basic requirements. It was reasoned that the silication process imbedded small particles of abrasive material in the nap of the paper and between the fibers. Paper so treated could be written on with metal style because this abrasive material caused particles of metal to be removed from the stylus and to be left upon the paper as a mark. It followed that any means of getting abrasive into the fiber of the paper which would not involve wetting would produce the desired results.

Experiments were first conducted to determine the grade of abrasive material required. Several meshes of metallographic polishing powder were manually worked into the fibers of the recording paper and tested for marking qualities. It was found that Linde polishing powder having an average particle diameter of one micron produced the best results. Manual application of the powder did not produce desirable uniformity of treatment, however, and so other means of application were considered.

It was recalled that spray applied particles of paint pigmentation were usually found deeply imbedded in the fibers and among the surface cells of unsealed wood to which application had been made. Since the abrasive particles considered for use were of comparative size to those of paint pigment, it was decided to try spraying a mixture of this powder in a vehicle of collodion thinned with acetone. The experiment proved successful. Spraying was found to be easily controlled for uniformity of treatment, and surface burnishing to remove excess abrasive produced a smooth, unwrinkled paper with undamaged rulings and upon which metal style left a clear mark. Sufficient paper was sensitized for both laboratory use and field trials.

Laboratory use of this recording medium proved to be entirely successful, but field trial results were not completely satisfactory. Recordings of variable amplitude were found to lack definition and extremely small amplitudes were not easily resolved. Increase of abrasive proportions in the treatment solution was tried, and although darker lines resulted, the quality of the recording was not improved. In view of the favorable results obtained in the laboratory using this medium, field recordings should have been better. It was decided that the disparity was due to inability to simulate in the laboratory the dynamic conditions of field operation. Although the method was considered usable, it was thought that results achieved did not justify the cost and effort necessary in chart treatment. It was decided to abandon this medium in favor of the much simpler one used in the final instrument; pencil lead upon untreated paper. Quality of recording in this latter medium was found to be just as good, recording time was comparable, and other considerations were also favorable. Appendix Drawing

No. M-2-4d shows the pencil used in the final instrument.

Meter damping. By the fall of 1951 development had outgrown the laboratory stage. Elements of the instrument were essentially as described, and performance, based upon laboratory tests, was considered to be excellent. It was decided that further efforts to simulate field conditions in the laboratory would be a waste of time. The Bonneville Power Administration was therefore so advised and arrangements were made for conducting field tests.

Site of the tests was tower No. 127 on the Kelso-Chehalis 230 KV line of the Bonneville Power System. As can be seen in the photographs of Figures No. 3 and 4, this is a double suspension tower having three power phases and two ground wires. Locale of the tower is an open field removed some five hundred yards from the nearest trees or other break in ground contour. This and other features of the site gave assurance of representative field operating conditions.

The tower installation of both the new instrument and its predecessor was made, as shown in the photographs, upon opposite sides of the same tower. In each case attachment was made to the ground wire, 18 inches out from the suspension clamp, and connection to the instrument was by means of glass twine. Twine connectors were used for securing twine, both at the cable clamp and at the instrument. String dampers were used upon both installations. The completed installation of the new instrument is shown in Figure 20.

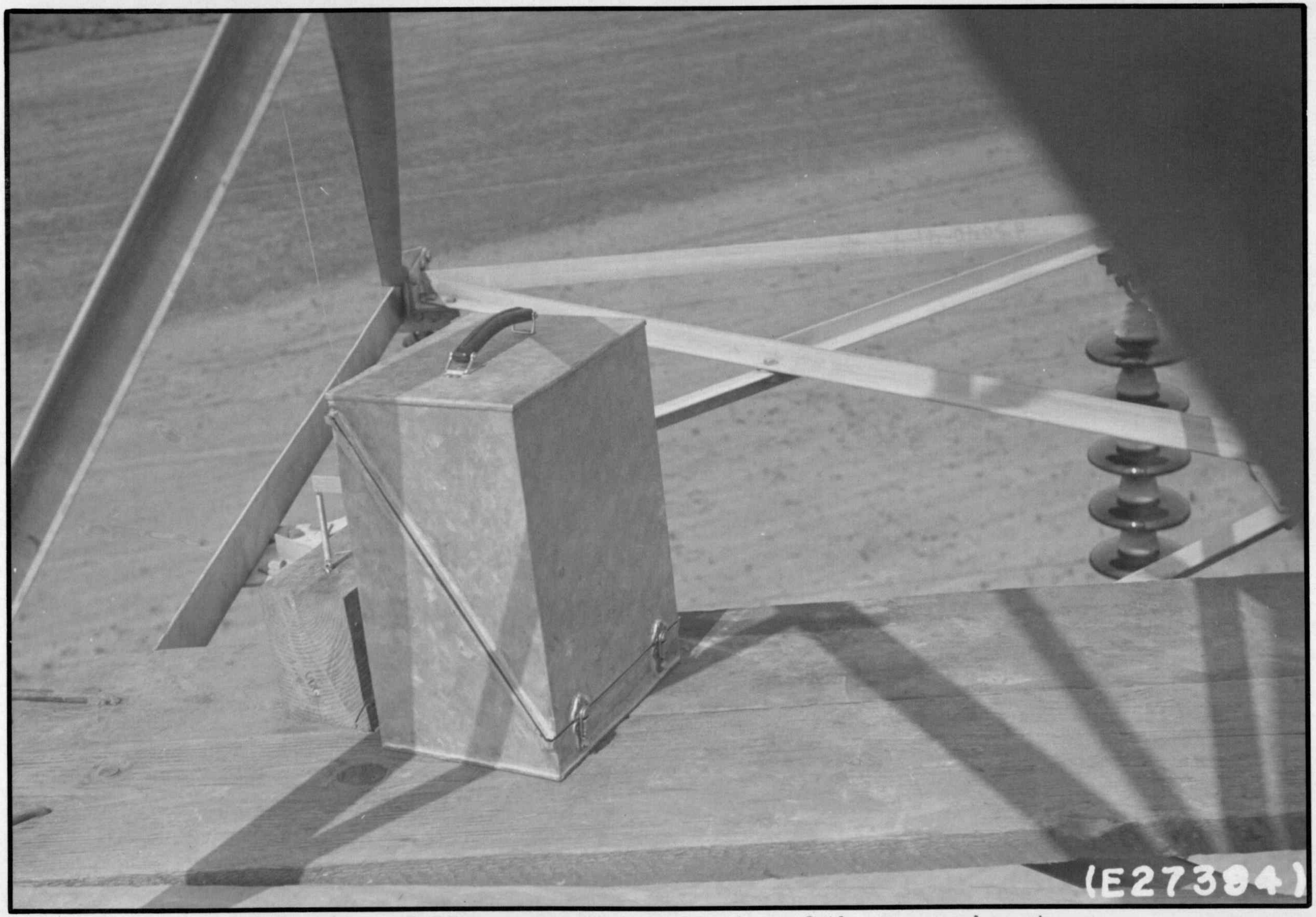


Figure 20. Field test tower installation of the new equipment.

Favorable vibration conditions existed during the three days of the tests and adequate recording for proper evaluation of instrument performance resulted. Most important of the various observations made was the fact that frequency recording was much too responsive. Vibration encountered in this case was of constant frequency but variable amplitude. Furthermore some part of each amplitude cycle was less than that necessary to actuate the frequency sensing mechanism. When this occurred, the instrument felt no vibration, commutation ceased, current in the meter circuit stopped, and the frequency recording pen immediately dropped to or approached zero on the chart. A fraction of a second later, when the amplitude had again built up to a value sufficient to actuate the frequency sensing element, recording proceeded as before cut-out. The result was that a wide frequency trace was recorded and ink was consumed at a much faster rate than anticipated. Since conservation of ink supply was necessary if the specified unattended operation was to be achieved, and also in view of the specified requirement for a line frequency trace, correction of this characteristic was needed.

Upon first consideration of this behavior it appeared that instrument sensitivity needed to be increased to prevent cut-out. Upon close inspection of records, however, it was noted that there were numerous instances of such frequency recording where no corresponding amplitude trace was recorded. In some such cases the frequency was in the range from 20 to 30 cycles per second. Reference to the response curve of Figure 11 shows that the amplitudes of concern in this range of frequencies were less than 0.001 inch. Increase of instrument sensitivity beyond this range was not considered practicable, especially in view of the fact that the curves of Figure 7 indicate cut-out would still be troublesome at lower frequencies. It was therefore decided that increase of sensitivity would not provide desired correction.

It was obvious that use of a less responsive meter would provide desirable limitation of frequency trace, but the damping characteristic of available meters was found to introduce error. In the case where frequency fluctuation was systematic, the ballistically damped meter would provide a roughly average frequency trace. In the case of unsystematic frequency fluctuation, the value of frequency recorded would be indeterminate. Either response would be unacceptable, and adaptation of the usual means of meter damping was therefore rejected.

The significant factor in this rationalization was the emphasis upon need for unilateral damping. Obviously, if the frequency trace was to have meaning, a recording of maximum frequency was required. For this purpose a mechanism was needed which would discriminate between the two directions of meter motion, and apply damping to retard back swing only, while allowing uninhibited motion in the direction of increasing frequency. Additional characteristics desired were small size, damping control, temperature stability, mechanical rigidity, interchangeability, unit construction, etc, as for other elements of the instrument.

Consideration of the various means for providing damping suggested two which might be applicable. The first made use of armature eddy currents established by shorting the meter poles when cut-out occurred. Motion discrimination was to be accomplished by means of a switch, in electrical series--parallel with the meter, which would be mechanically actuated by the meter spindle. This method was not considered entirely practicable because of the vulnerability of the switch and the limitation of damping adjustment. The second utilized a rachet or clutch drive which would engage a dashpot in one direction of motion, and it was this means which appeared to offer the best assurance of success. Experimentation was therefore directed toward development of this type damper.

The first experimental design constructed sought to establish an hydraulic gradient in a viscous medium in much the same manner as the paddle wheels of the old sternwheel river boats. Elements of this design were paddles fastened to a spider or hub upon which was also mounted a rachet with very fine teeth. This assembly was free to rotate upon the meter spindle and was driven unilaterally by means of a rachet pawl secured solidly to the spindle. The paddles were immersed in viscous oil contained in a transparent plastic cup (see Figure 8) mounted upon the meter frame. Action of the paddles was easily observed through the walls of the plastic cup.

Oils of various viscosities from SAE 30 to SAE 140 were tested in this device. It was noted that although the desired hydraulic gradient was established by movement of the paddles through the oil, damping action was not great. Whereas a half cycle delay of ten seconds was desired, SAE 140 oil produced only three seconds' delay for a return swing of the meter from full-scale position. Although it would no doubt have been possible to improve performance of this device by incorporation of design refinements, it was decided to abandon the principle in favor of one having greater promise of success.

The next design adapted the same rachet and pawl device for directional discrimination, and utilized the same type of cup for containing the working fluid, but depended upon the familiar principle of viscous drag for its action. Appendix Drawing No. M-2-5a shows constructional details of working elements, and Figure 21 shows these elements pictorially. The object to the left in Figure 21 will be recognized as the cup of Figure 8, now made of aluminum since transparency is no longer of value. Note both the mounting flange, by means of which the assembly is secured to the meter frame, and the central spindle bearing which serves to position the meter armature in the magnetic field. The next object is the rotor which takes the place of the paddles of the previous design. The third object is the stator and cup cover. Note that the concentric cylinders of the stator fit in the spaces between those of the rotor. As can be seen by reference to the detail drawing, these cylinders mesh with a clearance of 0.005 inch. The fourth object is the rachet which secures to the top of the rotor and which is driven by the pawl shown at the extreme right. Note that the pawl carrier is

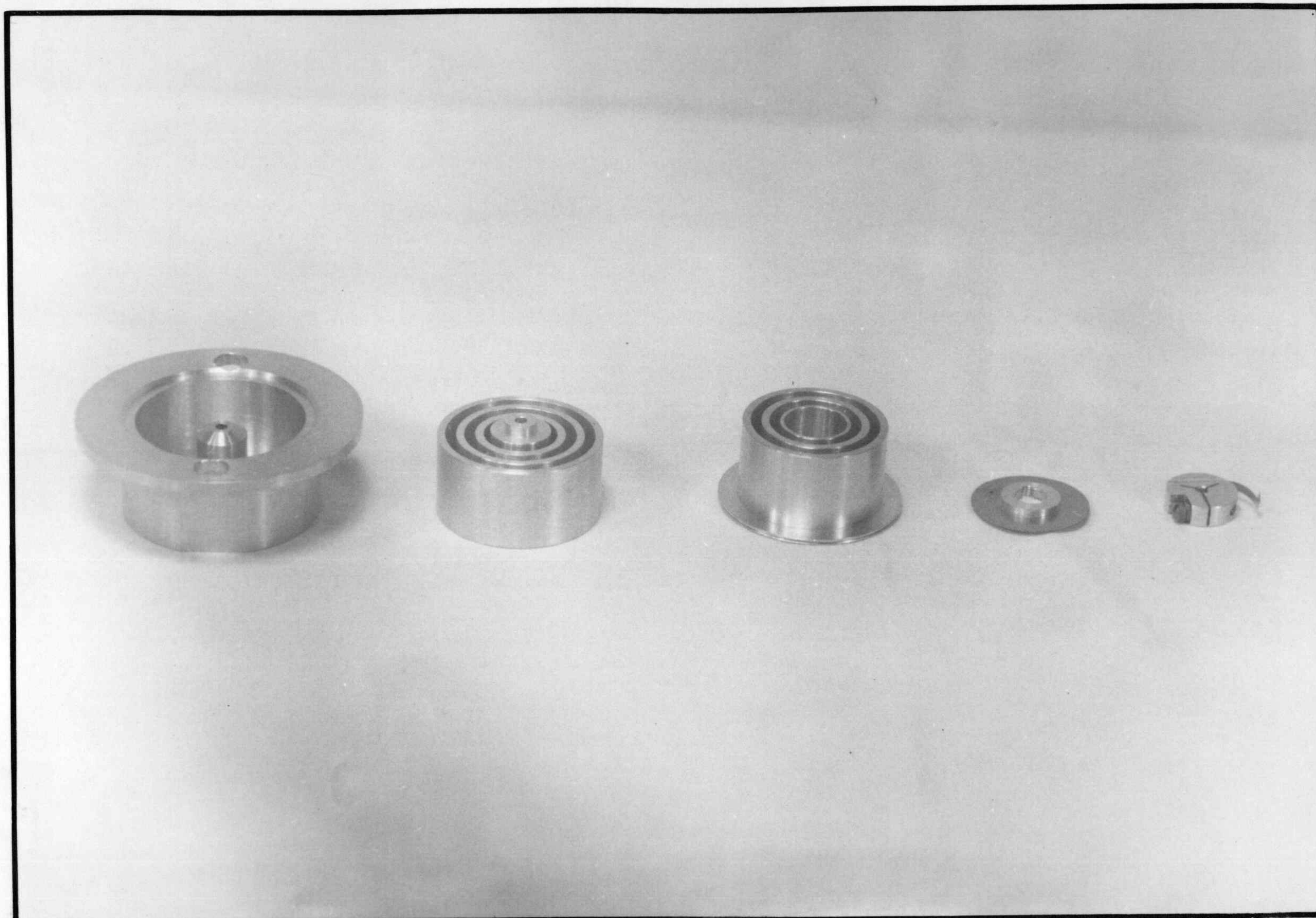


Figure 21. Meter damper parts.

provided with a clamp by means of which it may be fastened to the spindle. The pawl itself is made from a piece of spring steel 0.001 inch thick and is adjusted so as to have negligible drag upon the rachet teeth, pitched approximately 0.01 inch. When assembled, all elements except the pawl carrier mesh into a compact unit within the cup, and the cover, which is integral with the stator, prevents leakage of the working fluid from the cup.

Tests of this device showed that ample damping action was obtained, and that this action could be varied over a wide range by changing viscosity of the working medium. For example, SAE 140 oil produced a return swing delay of approximately three minutes at room temperature. At 20 F damping was not critical and was therefore not comparable, but half swing was noted to take three and a half to four minutes. This latter was cause for concern, since essentially linear damping response was desired throughout the range of temperatures from -30 F to +120 F. The need for using some working medium other than petroleum base oils was apparent.

The solution of this problem was found to be utilization of the Dow Corning silicone base lubricants. In the range of viscosities and temperatures of concern for this use, these remarkable fluids have a change of damping effect in the ratio of but 3 to 1 compared to a similar ratio for petroleum base fluids of 2500 to 1 (8, p14). Retest of the damper, using these fluids, showed that acceptable damping action was obtained over the specified temperature range.

Two silicone base fluids were tried in laboratory tests. The DC 200 fluids having a viscosity at 25 C of 12500 and 500 centistokes respectively were selected as providing the most probable range of utility for this application. It developed that the more viscous of the two permitted summation of transients in the lower frequency ranges, and so completely reduced sensitivity in the range to 20 cycles per second as to make recording meaningless. Damping in the range above 20 cycles per second was satisfactory however. The other fluid was thought to provide somewhat less than the desired damping action, but decision in this matter can be made only after extensive field trials under all conditions of operation. The nature of operating conditions, as well as the difficulty and expense of simulating these conditions in the laboratory, dictate such procedure. It seems assured that this means of damping will provide satisfactory meter performance when the proper viscosity of fluid has been determined.

External case. The need for a protective case external to the instrument case was first considered incident to the evaporative and freezing tests of inks. Behavior of these recording mediums at extremes of temperature suggested the need for some means of minimizing the range. Provision of heat from an external source to raise temperature in the low range, and insulation to lower temperature in the high range offered the obvious solution. Since the instrument case did not lend itself to such treatment, it was decided that a suitable external case was needed which would contain the instrument case and also afford space for the heating unit and insulation. After investigation of auxiliary requirements, the case shown in Figure 20 was designed and built. This will be recognized as the one used during the field trials held in the fall of 1951.

It was found that although the case was satisfactory in many respects, it needed certain modifications to be entirely satisfactory. For instance, it was observed that the effort to make the case weather tight had resulted in insufficient air circulation to dry the inked record. It was also noted that initial adjustment of glass twine on the actuating lever arm was difficult because the movement of the amplitude arm with adjustment of string length could not be readily seen. Means of correcting these difficulties were provision of air vents, and addition of a window, as can be seen in Figure 22 and Appendix Drawing No. M-1-3a.

At the time of the field trials, the range of permissible compensating spring force was not known. It will be noted in Figure 20 that the spring used for compensating pull of the glass twine upon the actuating lever was jury-rigged to a wooden block nailed to the test platform. By adjustment of this spring and comparison of results with those recorded by the older type recorder, it was determined that a spring force ranging from 14 to 18 pounds produced the best results. With this information as a guide, the counter spring support shown in Figure 23 and in Appendix Drawing No. M-1-3b was constructed, and installed upon the back of the external case. A compression spring was used in the final model, as

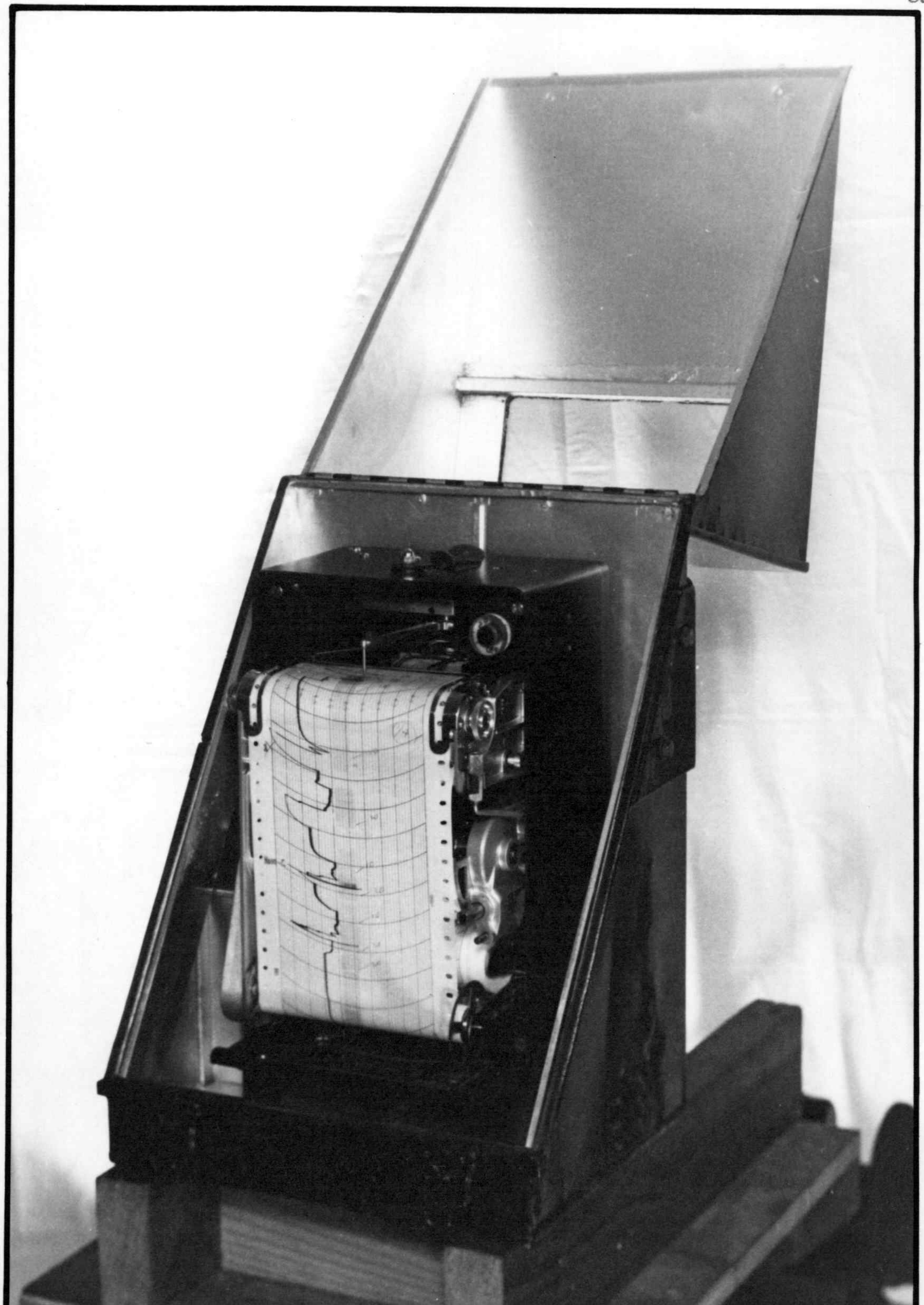


Figure 22. Laboratory test installation of final instrument.

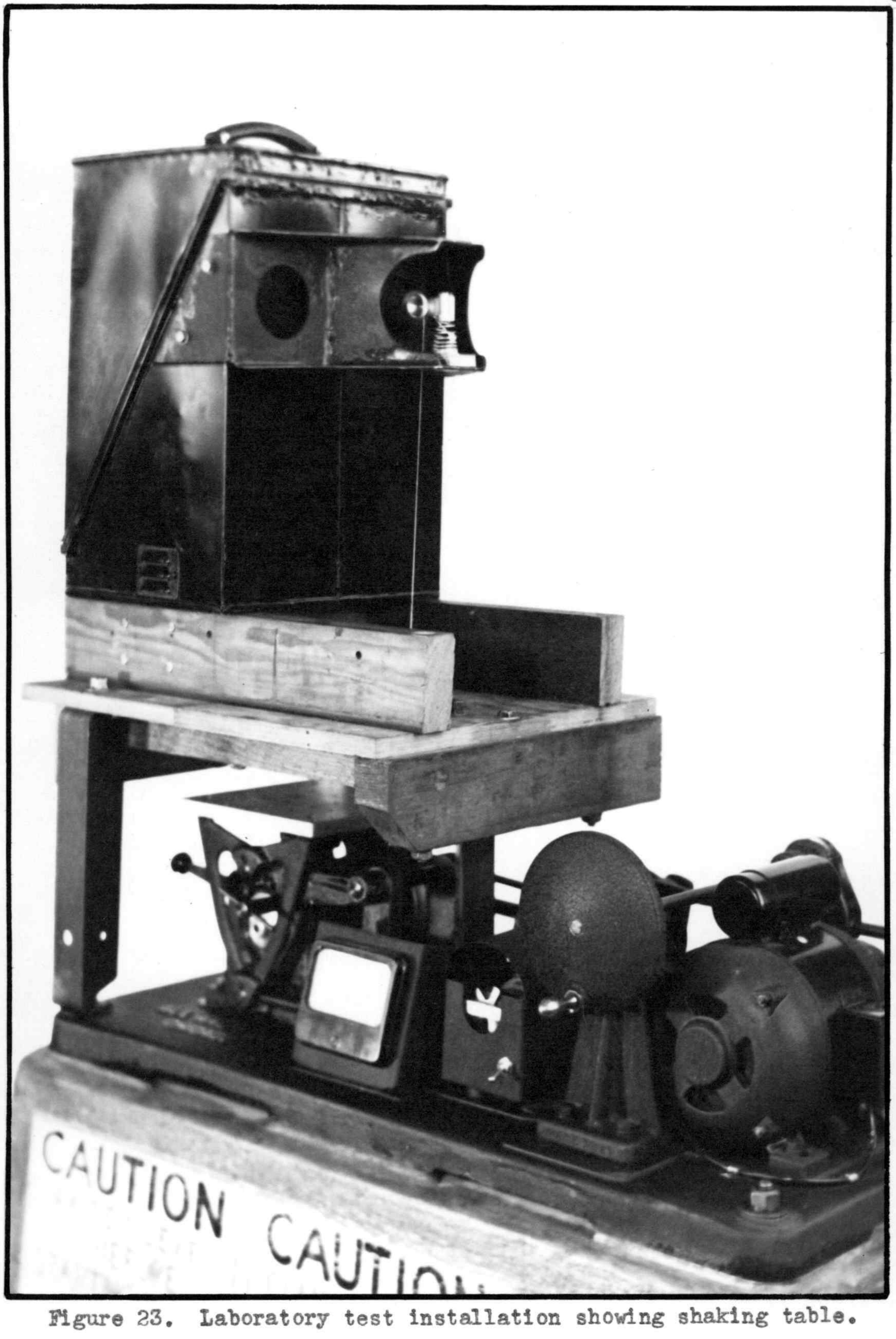


Figure 23. Laboratory test installation showing shaking table.

shown in Figure 23, because of the increased compactness thereby achieved.

Subsequent laboratory trial using this spring showed good instrument response to 47 cycles per second with 18 pounds of spring force. Substitution of a stiffer spring would have extended the response to beyond 50 cycles per second. This was not done since essential compliance with specifications was achieved, and no real purpose was to be served by expending more effort in this endeavor.

Due to delays caused by meter damper development, work contemplated for operation temperature control could not be undertaken. This was not considered a serious deficiency, however, since development of auxiliary means for vibration measurement (4) during the contractual period had modified the original requirements. Among modifications permitted was use of the instrument under very much less extreme temperatures, and need for extensive protective means was not great. It was accordingly decided to provide the absorptive and reflective protection of black and aluminum paint on the outer case for nominal temperature control, and use additives in the ink as necessary to assure successful recording at extremes of temperature. This degree of protection was considered ample pending acquisition of more specific knowledge of field requirements.

Figures 22 and 23 indicate the manner in which instrument installation can be made upon the tower. As may be seen in Appendix Drawing No. M-1-3a, a 5/8" machine nut has been fastened to the external case flush with the bottom. A 5/8" machine screw is passed through a suitably located hole in the tower platform and serves to hold the case in position. Simplicity and compactness of the installation, in comparison with that for the instrument's predecessor, is also shown in these illustrations.

IV. DESIGN AND CONSTRUCTION

General. It is characteristic of developmental work that the fundamental design is established during experimental stages but the ultimate design becomes associated with final construction. In some cases the evolution is not at all obvious, as when analogues are employed in experimental stages. In this case, however, the design treatment is believed to have been direct.

The reader will have noted that very definite design trends were established early in the work. For example, utilization of standard elements, and design for sub-assembly construction, were both given consideration in the very first experimental mock-ups. Other design objectives recognized early in the work were mechanical simplicity and easy reproducibility. As the work progressed, thought was given to operational ruggedness, reliability, and simplicity, access to working parts, ease of maintenance, etc. Many phases of these design and constructional features have been emphasized in previous sections of the thesis. It will be the objective of this section to present additional details of design, and describe the manner in which these features were incorporated into the instrument.

A very important consideration in the development of a machine design is the availability of construction facilities. It is obviously not good general planning to design a machine which cannot be built without access to specialized facilities. Although this is done in some cases to give the owner of such machinery a manufacturing advantage, it is seldom done where the best interests of the ultimate consumer are served. In this case it was understood that the Bonneville Power Administration wished to call for bids on the construction of several instruments of similar design. The design construction facilities required were therefore kept to a minimum.

Machine tools used by the author in construction of the instrument included a jeweler's lathe, an 11-inch machine lathe, an end mill, a band saw, and a drill press. Hand tools used consisted of such gages and small tools as would be found in the tool box of any precision machinist. The design is thus seen to be one which lends itself to construction in a small shop having good equipment and personnel qualified to operate such equipment. In the interest of the instrument purchaser this means that small shops can bid, and although large shops are not excluded from bidding they cannot charge proportionately large overhead to the job and expect to get the contract. With adequate quality control written into the construction specifications, this means a good instrument at lower cost.

Other design considerations which contribute favorably to low first cost and reasonable maintenance are use of standard elements, such as ball bearings, switches, etc, and use of sub-assemblies. The economy in use of both standard elements and sub-assembly design is thought to require no substantiation, in view of the contribution of these two factors in development and economical operation of present-day mechanical devices. Adoption of these factors in the design was a matter of course.

Some degree of reproducibility and mechanical simplicity is thought to have been achieved. The manner in which this simplicity was developed has already been described in some detail and will be further described in the paragraphs to follow. The state of development as regards simplification is still formative, however. As it exists at this writing, the design is thought to be good and the instrument is workable. Recommendations for possible improvement made elsewhere in the thesis are thought to be logical extensions of the basic theory and further applications of these design objectives.

Throughout the text, and upon drawings, use is made of standard designators without naming standards in which such symbolization is set forth. The American Standards Association is understood to be cited in all such cases. Wherever chance for misunderstanding exists, as is the case between certain wire gages, the particular gage is referenced by name. All departures from usual or accepted practice have been similarly completely identified. Instances of this include departures from permitted allowances and tolerances. The drawings are seen to be dimensioned either in fractions or decimals of inches. Wherever a fractional dimension is given, allowance is understood to be 1/64 inch. Wherever a dimension is given as a decimal to thousandths of an inch, the allowance is understood to be one thousandth of an inch, and similarly it is one ten thousandth of an inch where the dimension is given to four decimal places. Hole and pin or shaft tolerances and allowances are in accordance with the American Standards Association and are given on the following page for ease of reference.

HOLE AND SHAFT TOLERANCES IN INCHES

Class of Fit	Hole Tolerance ($x\ d^{-1/3}$)	Shaft Tolerance ($x\ d^{-1/3}$)	Allowance or Interference
Loose	0.0025	0.0025	$+0.0025\ d^{2/3}$
Free	0.0013	0.0013	$+0.0014\ d^{2/3}$
Medium	0.0008	0.0008	$+0.0009\ d^{2/3}$
Snug	0.0006	0.0004	+0.0000
Wringing	0.0006	0.0004	-0.0000
Tight	0.0006	0.0006	-0.00025 d
Medium Force	0.0006	0.0006	-0.0005 d
Heavy Force	0.0006	0.0006	-0.001 d

d = Diameter of hole or shaft in inches.

The nature of the design work precluded the use of any set system of safety factors. The practice was to design for the most severe condition that could be anticipated, and systems or trains of mechanism were designed for uniformally distributed loading where strength was of primary importance. Where other factors such as stiffness or mass were of significance unequal loading was permitted. Consistently conservative design practice was violated in only one case, that of the amplitude transmission miter gears. In this case both the dynamic and wear loading are considered excessive and there is no question but that gears of steel will need to be substituted for the aluminum gears now in use. This was not done during development stages because other work was more pressing and the aluminum gears served the purpose. Replacement will no doubt be required, however, in preparation for a sustained field testing program.

Frequency sensing. In accordance with the principles outlined in the chapter on Theory, proper function of the pendulum involves selection of such values for mass, spring constant, damping, and moment arms as will provide response including the specified frequency of 10 cycles per second and conductor amplitude of 0.0125 inches. Other applicable controls include space limitation within the instrument case, inertia forces of moving parts, resonance of linkages, strength of members, etc. Selection is thus seen to be a balancing process in which the elements having the least variable values serve as controls. Other elements are then designed accordingly.

Of the factors which contribute to proper operation of the pendulum, that known to have the least permissible variation was space limitation within the instrument case. From experimental work it was also known that the pendulum arm needed to be as long as possible, and that the mass should be as large as was consistent with permissible sizes of bearings, linkages, etc. These requirements were obviously contradictory. A compromise involving incorporation of the maximum mass at the maximum pendulum arm consistent with space limitation was finally devised, however, through geometrical consideration. The design presented in Appendix Drawing No. M-2-3a resulted.

The next step in the determination of values was to establish the point of dynamic mass center for the pendulum. It will be recalled that the point "O" of Figure 6 corresponds closely to the dynamic mass center of the pendulum for higher frequencies. The pendulum was therefore mounted upon a lever and supported statically by a spring in much the same manner as indicated in Figure 6, except that the pendulum was turned around with respect to the lever fulcrum as shown in Figure 24. After coupling the free end of the lever to the shaking table, as shown, the table was caused to shake at 60 cps, and various amplitudes, and the center of rotation was noted. This proved to be on the longitudinal symmetrical axis, out 2.50 inches from the support pin, as indicated in Appendix Drawing No. M-2-3a.

With the mass center known, the effective mass of the pendulum was then determined by measuring the balancing reaction required a measured distance from the pin connection, with the pendulum horizontal. This measured reaction referred to the 2.5 inch pendulum arm corresponded to an effective pendulum mass of 66.5 grams.

Determination of the proper size spring to use for supporting the pendulum was the next step in the design. Approach to this problem involved selection of the proper value for s (see Figure 6) and spring constant, k, to result in a pendulum having a natural frequency producing desired sensitivity. Use was made of the final expression derived in the chapter on Theory (page 15). It will be noted that finite values of variables exist where impressed frequency is equal to natural frequency. For this value of impressed frequency the last term is equal to zero and a quadratic expression for s may be

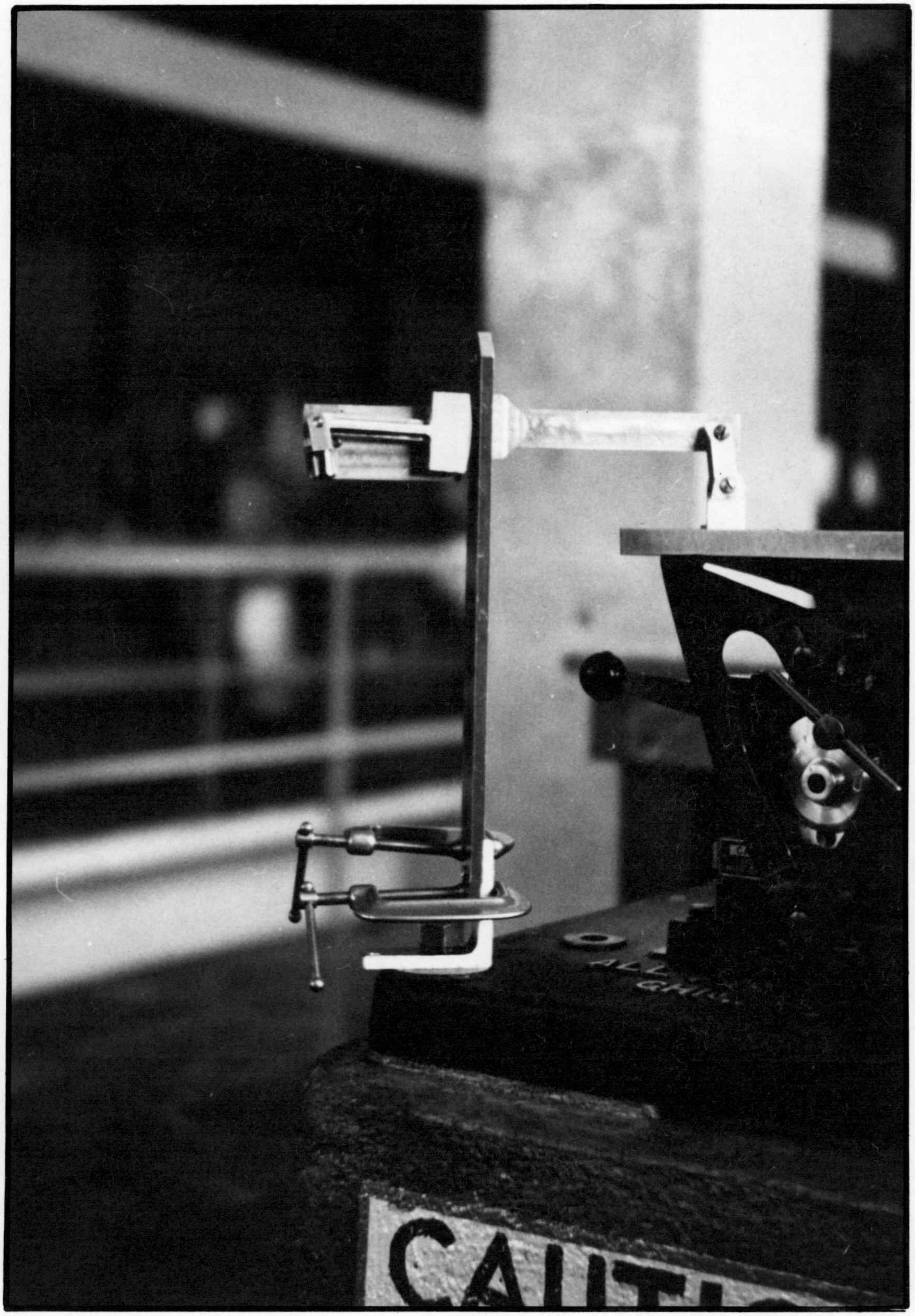


Figure 24. Method of determining dynamic mass center of the pendulum.

obtained in terms of $L = 2.5$ and α . The term $\alpha = a_o/x_o$, and this ratio may be expressed for any value of lever deflection in terms of moment arms about the lever fulcrum. Values of these moment arms were established by geometrical limitations of the lever design and so selection of the point on the free end of the lever at which conductor motion is to be applied permits calculation of the value of s. This selection was made such that the value of S/L would be about 0.18, allowing for solid end connection of a torsion spring. See Appendix Drawing No. M-2-2a for design dimensions.

Selection of a suitable spring was based upon the knowledge, obtained from the generalized curves of Figure 7, that desirable performance required a constant value of pendulum inertia at all frequencies. If this condition was to apply, the inertia level at specified conditions ($f = 10$, $x = 0.0125$) must remain constant. Substitution of these values into the expression for acceleration of the mass, $a = x_1 \omega^2 = x_2 \omega_n^2$, and solving for ω_n corresponding to a value x_2 for which the first derivative becomes a large negative value (as shown in Figure 11) resulted in a decision to use a natural frequency of four cycles per second. Substitution in the expression for natural frequency of the pendulum, derived in the chapter on Theory (page 15), resulted in a value of $k = 4.78$ pounds per inch of deflection. The spring was then designed by conventional methods (9, p36); the result of which is shown in Appendix Drawing No. M-2-3b.

Trial of this spring and pendulum showed good agreement with expected response. As stated above, correction was required for the solid connection of spring ends to the pendulum arm and lever. Effect of this connection was to reduce sensitivity of the pendulum to motion. Increase of the actual value of s from 7/16 to 9/16 inch without changing other calculated values produced the desired response shown in Figure 11. This result is to be expected since the solid end connections add stiffness to the spring, and increasing the length of projecting arms on the spring produces a compensating softening action. The same result could have been obtained by pins connecting the spring ends, but this would also have meant more complex construction and greater difficulty in assembly and disassembly.

Nothing has been said so far of construction of the pendulum pivot pin or fitting of this bearing. Appendix Drawing No. M-2-3a shows the detail of this pin and bearing. It will be noted that an oilite bronze bushing and ground steel pin are called for in this detail. It is possible that a preloaded double row ball or other suitable anti-friction bearing would have served as well, and modification of the design to this extent might well be considered in subsequently built instruments. A ground and lapped bearing was used in this model both to facilitate the construction and for ease of assembly and disassembly. These were both very desirable features in the experimental model but are not of significance otherwise. A preloaded anti-friction bearing for this use in subsequent models would allow further design simplification and permit specification of such bearings for all linkages in the instrument. These are thought to be features worthy of adoption.

Construction of this bearing was as indicated on the referenced drawing. Stock bushings were first force fit to the pendulum arm and carefully reamed. The pin was then made to the dimensions given and the journal surface ground to a zero tolerance, honed, polished, and finally lapped with polishing range on the bushings. The finished bearing was completely free radially but with not the slightest indication of "slop" in the fit. Finished clearance was something under 0.00005" on the diameter.

Appendix Drawings No. M-2-2a and M-2-2b show construction of the lever fulcrum and bearing. No particular design limitations were applicable in this case and construction is seen to be straightforward. The lever was lathe-bored to receive the bearing races with a tight or medium force fit (0.0000" to 0.0003" metal interference) and the pin was ground to allow a clearance of 0.0002" for easy assembly. Other requirements are as shown in the detail drawings.

Various parts of the commutating switch drive, cross-head, and assembly required very painstaking work in fabrication, but it is seen by inspection of Drawings No. M-2-2a, M-2-3a, and M-2-3c that design elements are simple in construction. The design is seen to be an adaptation (10, p2) of the basic Metron tachometer switch involving placement upon the lever so as to permit ease of actuation. Dimensional tolerance can be as much as $\pm$ 0.0020 so long as ease of assembly and satisfactory operation are not thereby impared. Construction methods are believed to be obvious from mechanical configuration indicated. The complete lever and frequency sensing element is shown in Figure 10.

Amplitude Train. The amplitude transmission train is shown pictorially in Figures 15, 16, and 17, and construction details are presented in the M-2-4 series of Appendix drawings. Figure 17 shows the basic mechanism to be a simple four-bar crossed linkage (11, p17-21) with the case providing loop closure. As before stated, the purpose of this train is to transmit conductor motion to the record and to provide the specified 4.5:1 magnification. Miter gears in the train (Figure 16) are of unit ratio both for minimization of tooth loads and to permit gear setting for zero back-lash. This obviously calls for motion magnification in the linkage.

In addition to these requirements, space limitation within the case was an important design consideration. This was accepted as the control design factor and desired configuration of linkages was therefore first established graphically. The next step in the determination was selection of linkage size and mass to avoid resonances in the system. As stated in the chapter on Development, this selection was based upon knowledge gained from an experimental mock up. The first system devised for trial was found to have a bothersome resonance in the long shaft driving the pinion gear of Figure 16. This condition was corrected by increasing the size of shaft and changing from an aluminum alloy to steel. The results of this change were then evaluated. Procedure was to fabricate a solid aluminum disc having considerably greater mass moment of-inertia than the recording arm and install it on the recording arm spindle. The assembly was then driven at constant amplitude through frequencies varying from 10 to 60 cycles per second. Since the resonant frequency of such a system varies as the square root of the reciprocal of mass moment of inertia, this system would resonate at a lower frequency than the actual system. No resonance was noted, however, and so this part of the mechanism train was accepted as satisfactory.

The final step in design of the amplitude train was reduction of recording arm mass. The procedure followed and results obtained in this phase of the design have already been described in connection with the development of this component. As will have been noted, performance of the assembly was consistent with basic design requirements and so no further design changes were made.

Figure 25 presents the calibration for variation of recorded amplitude with frequency. Information for preparation of this figure was obtained by driving the instrument with a solid connecting rod fastening to the shaking table. Variation is of course due to the amplification of input motion as the amplitude train approaches its resonant frequency of 75 cycles per second. The curve is presented in this form to facilitate the interpretation of chart records. Conductor amplitudes are seen to be directly determined from chart amplitudes by means of this curve.

No particular problems were involved in the construction of this mechanism. Design as a unit facilitated construction to the end that the principal effort was directed toward alignment of gears for

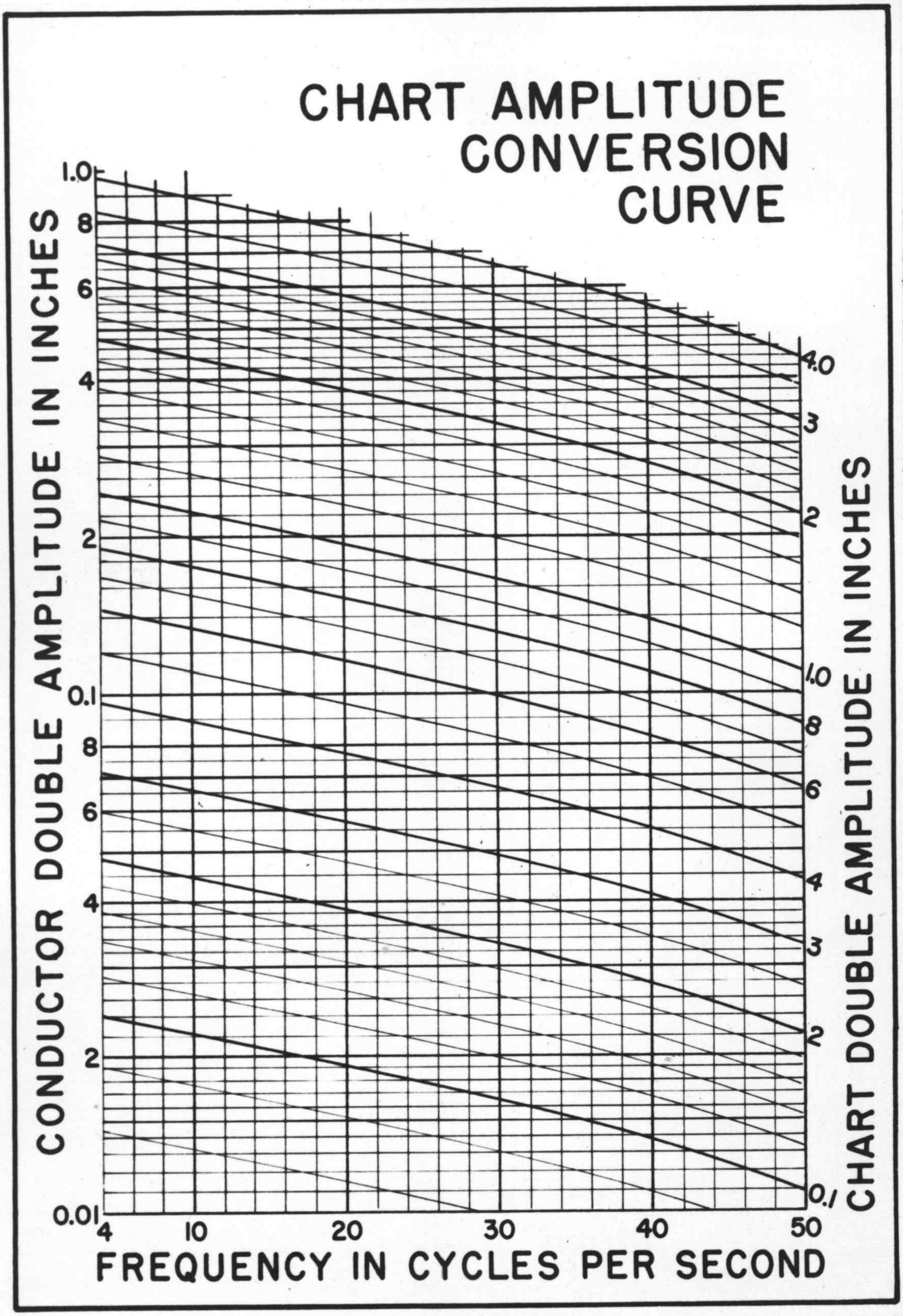


Figure 25. Chart amplitude conversion curve.

negligible back-lash. Use of precision ball bearings throughout the unit and application of ordinary care in fabrication of the mount and bearing seats contributed to success in this regard. As with other components of the instrument, some parts of the amplitude arm required painstaking care in manufacture, but it is thought that need for such care is adequately indicated on applicable drawings. Standard allowances and tolerances were used for all fittings.

Assembly of the unit in the instrument case was by means of two 10-24 machine screws for which provision is made in the top of the gear mount. (See Drawing No. M-2-4b) Matching holes are located, drilled, and head countered in the top of the case such that the amplitude recording pencil makes its trace just one-half inch behind that of the frequency recording pen. When so located, the connecting rod of Appendix Drawing No. M-2-4a should be so positioned that attachment may be made to the actuating lever as shown in Figure 17.

It will be noted in Figure 17 that two large head canister screws are mounted on the top and left side, respectively, of the instrument case and disposed orthogonally with respect to the top pin connection of the above connecting rod. These serve to cover holes provided for gaining access to this pin and its accompanying set screw. (See Appendix Drawing No. M-2-4a) Reference to the drawing will show that this pin is designed in such manner that when its set screw is loosened and the pin turned, the length of the connecting rod is adjusted through 0.04 inch. This means that for a given actuating lever position, the amplitude recording arm position may be adjusted a maximum of 0.17 inch. This adjustment may be used for initial positioning of the recording arm to the center of the chart corresponding to a horizontal or midamplitude position of the actuating lever.

Meter damping. As in the design of other components used in the instrument, space limitation within the meter case required consideration in design of the meter damper. Fortunately, the manufacturers had left a volume approximately one inch in height by one and three-sixteenths inches in diameter within the case immediately above the meter armature. This volume was advantageously located for installation of a damper required to act upon the armature spindle, but was at first considered to be too small a volume in which to place an hydraulic or viscous damping device. Preliminary analysis by means of Newton's familiar expression for viscous drag had indicated that no less than ten square inches of area would be required for desirable function of a viscous damper. Fitting this quite large amount of drag area into such small volume was therefore of concern.

Several basicly similar approaches to a solution of the above problem were suggested by familiar, commonly used machinery. It was obvious that multiple-stacked discs having desired combined surface area could be arranged as in a disc clutch and so spaced as to give desired damping. Another possibility was use of a rotor similar to the usual axial flow turbine, and arrangement of the case so that it could be disassembled in two halves along an axial plane. A third possibility was adaptation of the principle used in some oil centrifuges embodying concentric conical or cylindrical moving surfaces and a cuplike container. Other devices, including positive displacement rotary pumps discharging to an orifice of variable size, were considered but ultimately rejected because of their relatively greater complexity. The simplicity of a device employing the drag between surfaces working in a viscous medium emphasized the desirability of selecting one of the three basic designs first mentioned above.

Consideration was then given to secondary factors applicable in selection of the basic design. It was realized that desired damper performance would be achieved only after trial of several working mediums in the device, and ease of access was therefore significant. Use of a damper case which had to be disassembled along an axial plane was therefore cause for rejection of the axial turbine principle. Similarly, the use of multiple discs would present an assembly problem, and would also call for special provision to assure freeing the working fluid of entrained air. These and other considerations indicated the desirability of using concentric cylindrical moving and stationary surfaces and a cuplike container for the working medium.

The design ultimately adopted has already been presented in the chapter devoted to development. Figure 21 shows the component parts of the damper pictorially and Appendix Drawing No. M-2-5a both gives the construction details and indicates the assembly sequence. The damper was found to fit compactly under the plastic meter cap, and incorporation of the upper armature spindle bearing as part of the damper cup allowed removal of this element from the meter. The flanges left upon the damper cup were drilled to fit the meter frame and replace the carrier for the above-mentioned spindle bearing. Consolidation of parts in this manner allowed the installation to be incorporated into the meter instead of being merely mounted on it.

Construction of the damper presented some problems in machining as inspection of the detail drawing will indicate. Since operational success called for very close tolerance in working clearance of the cylinders, both machining and fitting-up work needed to be done with good precision. Also, since space was at a premium, very thin sections had to be fitted, and "fit and trim" metal had to be peened without distortion. This called for the utmost care both in machining and assembly of component parts, as well as the use of spacers and jigs to assure correct holding while such work was in progress. Such measures are of course obvious to those familiar with precision machine work and so further detailing of the methods employed is redundant.

It will be noted that the stator also serves as a cover to keep the working medium from spilling out of the cup. An interference of 0.001 inch is provided in the fit of this cover on the cup so that a tight closure will result. Also note that the closure edge on the cup is mitered and that on the cover is grooved. This provision allows the insertion of a knife blade to pry off the cover when access to the cup is desired.

Assembly of the damper follows the sequential arrangement of parts shown on the detail drawing. Starting from the bottom, the cup presumably contains the required amount of working medium. The rotor is then placed within the cup and pressed gently down so that the working medium is forced up through the holes in the bottom of the rotor and into the spaces between cylinders. There should be sufficient liquid to provide complete filling of the concentric spaces of the rotor where holes in the bottom provide access for the liquid. Next place the stator into the rotor and press down gently so as to force liquid into all spaces of the damper. Note particularly that liquid should ooze out around both the inner and outer periphery of the cover before tight closure against the cup lip. This indicates that liquid completely fills the damper, and that no air was entrained. The last step is to properly align the ratchet hub on the protruding rotor shoulder so that the 2-56 keying screw may be inserted and set.

Assembly of the damper in the meter is thought to be obvious to anyone contemplating such procedure. One note of caution will be made, however. It is very important that the pawl carrier not rub on the ratchet in normal operation and so adequate clearance should be provided during damper installation in the meter. The writer finds that 0.002 inch thick shim stock serves admirably for positioning the pawl carrier with respect to the ratchet, and for holding this position while tightening the pawl carrier upon the armature spindle.

Electrical components. Electrical components and wiring have been apparent in instrument assembly Figures No. 12, 13, 14, and 15 previously referenced. Figures No. 26, 27, and 28 present these components and their assembly more in detail, and the series of Appendix drawings prefixed with the letter E give constructional details. Drawing No. E-1-6a shows the components of the system, and No. E-1-6b indicates the manner in which the system is assembled into the instrument case. This latter drawing is a schematic representation of the front view of the electrical assembly, comparable to the pictorial presentation of Figure 27. This drawing also gives the wiring harness color code and shows the elements relegated to the control panel, as well as the wiring employed on this panel. Figure 26 and Appendix drawings of the E-1-4 series show control panel assembly and constructional details respectively.

The control panel was designed as a unit both to serve as a consolidation point for the several components assembled there, and to provide one-point control of the instrument. Normal instrument control involves the following actions: (See E-1-6a)

1. Turning on the electrical circuit by closing the switch S_2.
2. Placing the meter M, in series with the cell B, and resistances R_1 and R_2 by means of switch S_3.
3. Calibrating the meter by adjustment and setting of R_2 and R_1 under certain conditions of laboratory operation.
4. Readjusting R_1 for subsequent settings (after the cell has begun to wear) so that the meter deflects

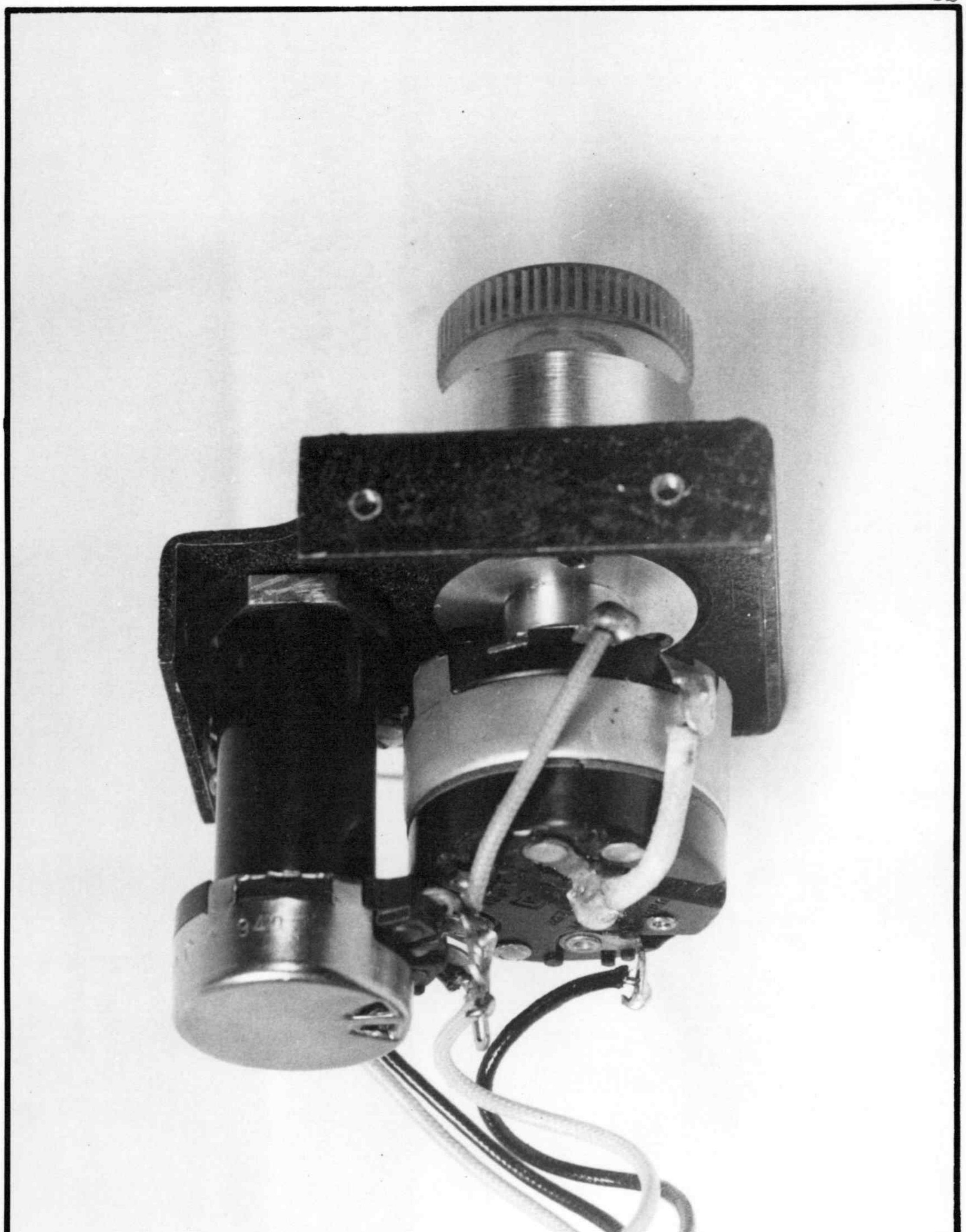


Figure 26. Close-up of control panel.

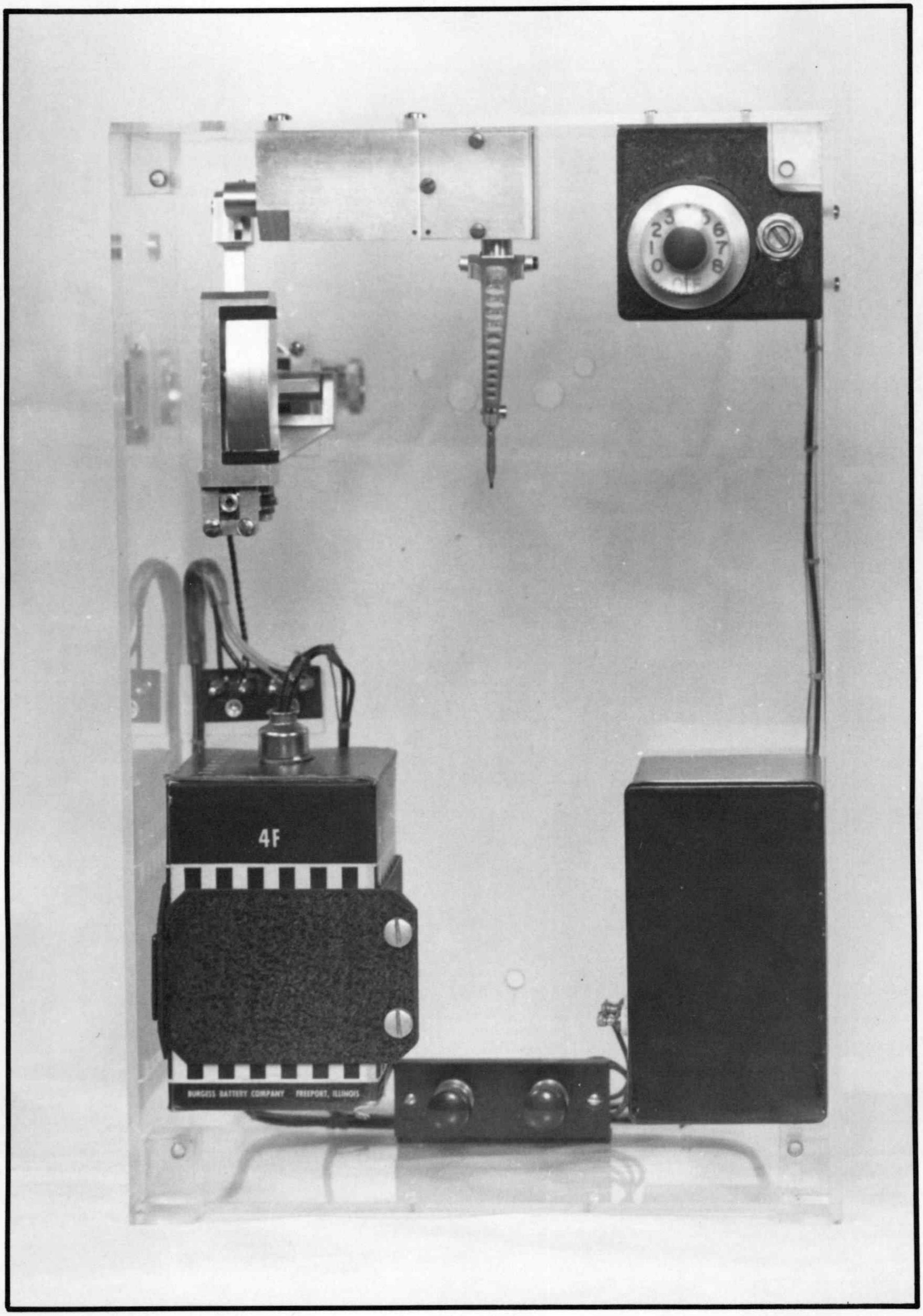


Figure 27. Front view showing electrical component assembly.

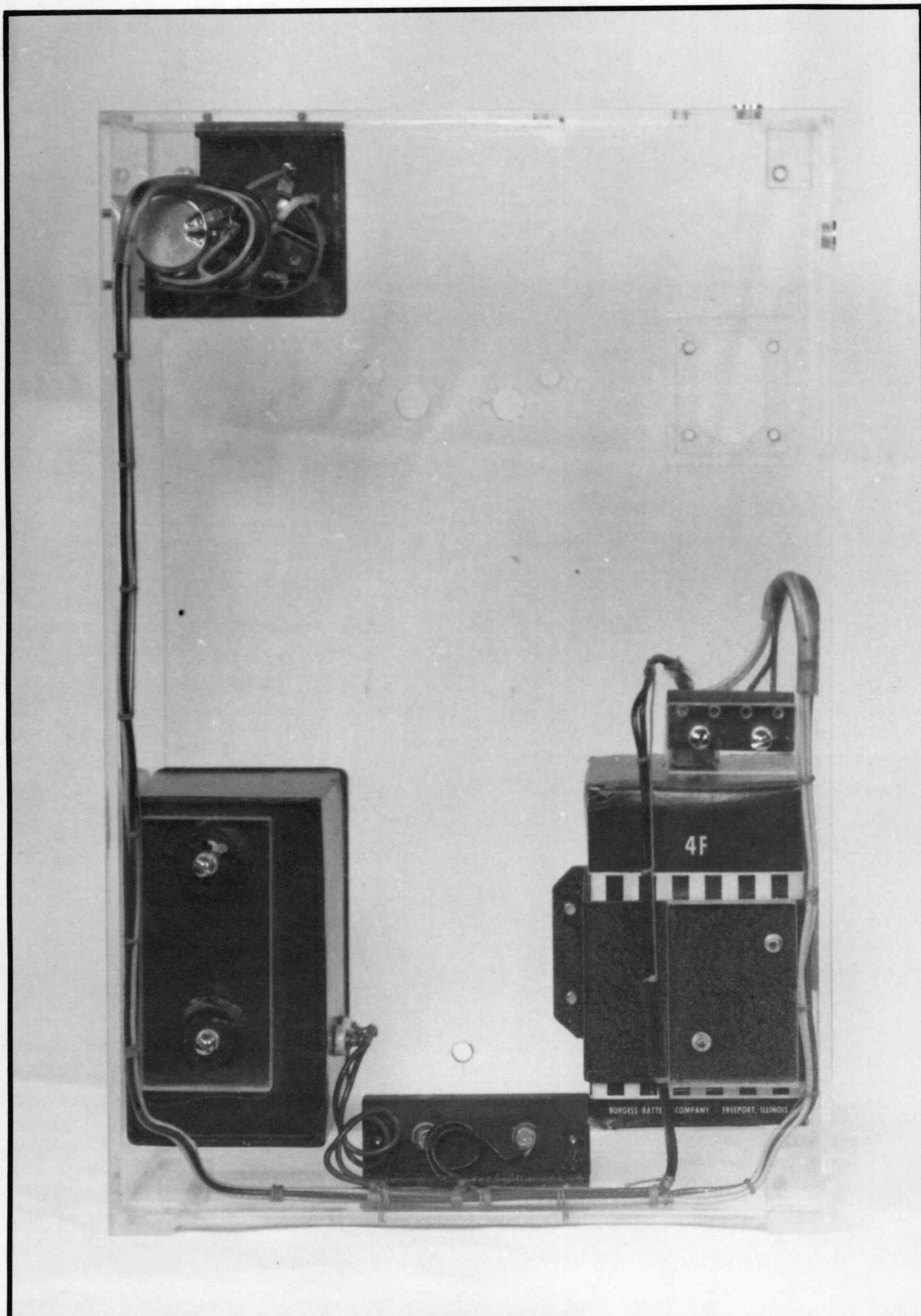


Figure 28. Rear view of electrical component assembly and wiring.

full scale. This readjustment resistance value is calibrated so that knowledge of the battery voltage level may be learned therefrom.

5. Placing the meter in series with the commutating switch S_1 , the cell B, and the potentiometer R_1 by means of the switch S_3 for normal operation of the instrument.

All of the above actions and adjustments are made by manipulation of controls located on the panel. Switch S_2 , potentiometer R_1 and switch S_3 are all operated by means of the control knob seen at the upper right in Figure 27. Turning the knob from zero to one on the scale calibration actuates the switch S_2 to energize the circuit. Further turning of the knob adjusts the potentiometer R_1 to lower values of resistance, and pushing the knob throws the switch S_3 as required in subparagraph 2 on the previous page. Releasing the knob normalizes the circuit as required in subparagraph 5 above. Initial adjustment of the potentiometer R_2 is accomplished by screwdriver manipulation of its slotted shaft (see Figure 27). All adjustments and controls required for normal operation are thus seen to be located upon this panel. Details of design which make this possible are shown in the referenced figures and drawings, and described further in later paragraphs.

Construction of the panel involved fabrication in several mediums. Appendix Drawing No. E-1-4b shows the knob to be made of clear lucite with a black bakelite hub. Lucite was used for the knob because it was desired to curve the face so as to magnify the dial numbers for easy reading. Joinure of lucite to bakelite was by means of dimethylene cement. Construction was as shown.

Another material applied in the fabrication of panel elements was neoprene rubber, used as a spring to position switch S_3 . Drawing E-1-4b shows the installation. Sponge rubber was used in this application because a spring of relatively low spring constant was desired and it was feared that use of a conventional spring in this application would result in undesirable resonance. It was therefore decided to use rubber which provides its own damping. The manner in which this switch actuation mount was placed upon the panel can be determined by referring to the E-1-4 series of Appendix drawings and examination of referenced figures.

Drawing No. E-1-4b also shows construction of the friction clutch designed to prevent inadvertent or vibrational alteration of the potentiometer (R_1) setting. This was designed and incorporated as an alternative to the more complex separate adjustment lock normally used in such applications. Laboratory trial under violent conditions of shock and vibration have proved its suitability for this use.

Assembly of the panel in the case is by means of four 4-40 canister head machine screws. Provision for this fastening to the case is made so as to position the face of the panel immediately behind the mounting board of the Esterline Angus clockwork chart drive. Holes are cut through this mounting board in way of the control knob and screwdriver adjustment of the potentiometer R_2. The manner in which this is done is indicated in Figure 13. Center locations and diameters of these holes may be taken from the E-1-4 series of drawings.

Very little design went into the remaining components of the electrical system. Parts were located within the instrument case in such manner as to provide best accessibility, or where adequate space was found. These will be seen to consist mainly of standardized elements or adaptations of standard designs. Customary electrical practice was followed throughout. Harness wiring was employed (see E-1-6c) and all components were wired and soldered before assembly in the case. This was considered essential from the maintenance point of view because of the space limitation within the case. Similarly, pig-tails were left at all terminals for ease of assembly and service. These and other factors of significance in construction are believed to be adequately illustrated in the figures.

Modifications to basic instrument. Although the Esterline Angus instrument was found to be very adaptable it was to be expected that some minor modifications should be found necessary for best utilization. In each case, when the prospect of making such change presented itself, an effort was made to design so as to avoid the necessity of alteration. This was done in the attempt to retain as much of the original character as possible and thus obtain an application of the instrument and not a reconstruction. The objective was use of the instrument as a standard component, and modification by addition of components without material change in the original essentials. Minor alterations were found necessary, however. Some of these have been described and some were too minor to mention in previous description. It is thought necessary to submit a listing of these latter, and in the interest of completeness, the following also contains mention of those alterations already described.

As regards the meter, the major alterations incidental to installation of the damper have been described. Reference was not made, however, to the change made in the pen carrying yoke which mounts on the top of the armature spindle. This alteration was found necessary incidental to the need for gaining access to the damper. The original yoke was fastened to the spindle by solder, and since the yoke needed to be removed before the bakelite meter cap could be removed, a less permanent fastening needed to be devised. Appendix Drawing No. M-2-5b shows the redesigned yoke, the fastening for which is obviously easily made by screwdriver.

The clockwork chart-drive mounting panel was subject to several alterations in the effort both to gain space within the case for added components and to gain access to others. Also several unused parts which normally mounted upon this panel were removed. These alterations are briefly listed as follows:

(a) Remove two round-head anchor screws from in back of the panel to avoid interference with seismic pendulum and substitute countersunk flat-head machine screws.

(b) Mill flush that part of a stiffening rib and boss which interfered with motion of the seismic pendulum.

(c) Cement 0.050 inch thick shims to faying surfaces at mounting corners to permit amplitude pencil to ride upon drawing board.

(d) Cut 1 1/8 inch diameter hole through the panel to allow control knob to protrude.

(e) Cut 1/8 inch diameter hole through the panel to permit screwdriver adjustment of potentiometer R_2.

(f) Remove meter pen stops.

(g) Remove index arrow from meter pen arm; also the scale for same, and fittings for supporting the scale upon the panel.

The instrument case was of course drilled in several places to permit inside fastening of components. This was required, as previously indicated, in spite of the fact that most of the components secured to the two aluminum mounting strips (see Drawing No. M-2-1c) substituted for the bakelite terminal strips included in the original instrument. Other than these, the only alterations were the installation of pedestal legs (see Drawing No. M-2-1b) and removal of the carrying handle from the top of the case for installation upon the top of the external case as shown in Figure 23.

V. OPERATION

The reader who has followed the text to this point will have achieved a knowledge of the fundamental functions necessary to permit operation of the instrument. The extreme simplicity of controls and functions hardly warrants re-emphasis. This chapter is therefore intended either for those who have no interest in the foregoing subject matter, or for those who wish a quick review of operating principles. The objective is to present such information as will be useful in the successful operation of the instrument.

Proper operation of the instrument begins in the laboratory where calibration of the frequency record is established. Procedure for so doing is to drive the instrument by means of a solid connection to a shaking table and so adjust controls that full-scale recordings result at 50 cycles per second, the meter zeroes properly and the frequency scale is linear. The steps involved are as follows:

(1) Turn on the meter by twisting the lucite control knob to the right until a click is heard, then turn the knob back to place the index line on the knob over the figure 1 on the dial. This places maximum resistance in the series R-C circuit. (See Appendix Drawing No. E-1-6a)

(2) Depress the knob to shunt out the commutation switch and place the cell in series with the meter. This results in deflection of the meter.

(3) So adjust the balancing spring on the meter that the recording pen returns to zero on the chart. This adjustment is made by rotation of the lever which protrudes from under the lower edge of the clockwork mounting board, and is directly coupled to the meter balancing springs.

(4) Drive the meter at 50 cycles per second and adjust the knob actuated potentiometer to a value of resistance such that the meter records full scale on the chart.

(5) Shut down the shaking table and without changing the value of knob actuated resistance adjust the screw-driver actuated potentiometer such that when the knob is depressed the meter records full scale. This provides the reference standard for full-scale deflection and is the means of duplicating standardized conditions in the field.

(6) Drive the instrument at various mid-frequencies to check linearity of recording. No adjustment is provided for altering the frequency function. The values of components are selected for a linear characteristic curve, but it sometimes happens that the damper prevents proper zero setting of the meter and the scale is distorted. If this occurs the zero adjustment and all subsequent steps in the calibration must be repeated.

With the instrument calibrated as outlined on the previous page, it may be removed to the location of desired field test and set up for recording as follows:

(1) Turn on the meter as described in step (1) on the previous page.

(2) So adjust the value of knob controlled resistance that depressing the knob causes the meter to record full scale. As the cell ages and voltage decreases, the value of resistance in series with the cell and meter which will permit full-scale deflection will decrease. This may be used as a measure of the condition of the cell and indicate when replacement should be made. Such replacement, however, will not alter calibration of the instrument since calibration is a function of current. The higher voltage of the new cell will call for a larger value (a lower dial number) of knob controlled resistance, but the calibration resistance (screwdriver adjusted) will be unchanged and so instrument calibration will be the same.

Use of the meter in the field calls for attachment of the actuating lever to the conductor by means of a glass twine of sufficient length to span the height (usually about twelve feet) between platform and conductor. This length of twine is of sufficient length and unit mass is such as to have at least one and sometimes more resonant frequencies in the range of operation of the instrument. Since modes of vibration of this system are not under investigation, it is desirable to prevent resonance insofar as possible. Professor R. F. Steidel has designed vibration absorbers for this purpose; details of which are shown in Appendix Drawing No. M-1-2. In use, these absorbers are fastened to the twine by clamping them at equal intervals along the length. Use of three such absorbers is sufficient for the usual length of twine required.

Since twine used is made by twisting strands of spun glass together and bonding with wax, it is very difficult to tie or otherwise secure this material to another object. Also glass fibers, although very strong in tension, are very brittle, and break when subjected to sharp bending or excessive wear. It was therefore considered necessary to devise a means of securing twine both to the conductor and to the actuating lever of the instrument. This device is shown in detail in Appendix Drawings No. M-1-1a and M-1-1b. It is also shown pictorially in Figures 9, 10, 12, 14, 15, 17, and 23. Figures 9 and 23 show the connector with twine attachment in place. It is seen that the connector incloses a pin about which a loop of twine may be passed and the free end then can be twisted back on the main lead which is given several wraps around the drum and holds tight by friction. This holding action is increased if the twine is first coated with resin in way of the drum. In use an abrasive disc is interposed between the drum and the object to which attachment is made, and the drum is kept from turning on its axial screw by friction against this disc. Loosening of the screw makes it possible to turn the drum, and results in adjustment of twine length upon the drum. Thus the adjustment of connecting twine length necessary for proper positioning of the amplitude arm is made possible at the twine connector on the actuating lever arm.

The procedure for making a typical field test installation, in regard to physical attachment of the instrument to the conductor, is as follows:

(1) Impregnate a suitable length of glass twine with resin in way of both absorbers and connectors.

(2) Clamp absorbers onto the twine, making sure that fastening does not bruise the twine or otherwise cause stress concentration or breakage of strands.

(3) Secure the upper end of the twine to the connector on the cable clamp and then install the clamp upon the conductor to be tested.

(4) Secure the lower end of the twine to the instrument actuating arm connector.

(5) With the clamping screw loose, adjust the twine length and tension against the compensating spring tension (see Figure 23) so as to position the amplitude recording arm at the desired position on the chart. This adjustment is accomplished at the connector located upon the actuating arm. Note particularly that the twine must lead away from the instrument side

of the connector drum. If the twine leads down from the instrument (the conductor is below the instrument platform), the connector drum must be on the side of the actuating lever opposite to that for which the twine leads up. (The conductor is above the instrument platform.) This is necessary because the line of action of the twine must always be at the same point on the actuating lever. This point is located on the instrument side of the connector drum.

(6) Tighten the connector screw by turning the knurled knob provided for the purpose and the connection is securely made.

Prior to start of the test there are certain additional routine preparations to be made to assure a successful run. Such preparations as winding the clockwork, checking the chart supply, filling the inkwell, priming the pen, and sharpening the amplitude recording pencil are illustrations. These are considered to be preparations which would normally occur to any careful operator. Separate itemization of this more general class of directions is therefore omitted. Such instructions as these, which are applicable to the basic Esterline Angus meter, have been prepared by the manufacturers and will be found contained in reference 7. The interested reader and the instrument operator in search of such information are referred to this publication.

On the subject of instrument maintenance, very little can be said pending acquisition of considerably more operational experience. It is known that the Esterline Angus basic instrument is well built, and the writer believes that the same can be said for the modifications made to the basic instrument. At the present writing it is not definitely known what the conditions of operation are to be. This fact makes the outline of a reasonable maintenance program particularly difficult. No specific recommendations can therefore be made, but it is hoped that the following quite general statements will be of assistance in formulating the program ultimately adopted.

(1) Lubrication points and lubricants used are as follows:

(a) Seismic pendulum spindle; SAE 10 or 20 oil. Note that an Oilite bushing was used here, and very infrequent oilings are required.

(b) Commutating switch cross-head; light weight silicone gear grease. (The same as used on the amplitude transmission miter gears)

(c) Commutating switch connecting-rod pin; SAE 10 or 20 oil.

(d) Amplitude transmission miter gears; light weight silicone gear grease.

(e) Amplitude arm hinge; SAE 10 or 20 oil.

(f) All ball bearings; very infrequent application of a light weight instrument oil or specially compounded lubricant for ball bearings.

(2) It is recommended that steel gears be procured as replacements for the aluminum amplitude transmission gears used.

Both dynamic and wear loading on the aluminum gears are known to be excessive. Consideration might be given to fabrication of an entire amplitude train unit using the steel gears if instrument shut-down time for making the change is of significance.

(3) If it is decided to standardize on use of this instrument design, consideration might be given to fabrication of spare component assemblies to simplify and expedite maintenance. For example, a spare unit comprising the actuating lever and pendulum-commutating switch assembly could be stocked. A spare amplitude transmission train could also be kept on hand. In fact, the unitization of design makes the keeping of spares by units a distinct possibility for effecting economical maintenance.

(4) The most fragile part of the instrument is believed to be the "throw leaves" of the commutation switch. Although it is claimed that the Metron instruments operate for long periods without failure, the best maintenance policy would seem to be having spares of this item on hand.

VI. COMMENTS AND CONCLUSIONS

Although it might be said that the instrument is a notable success insofar as basic principles of operation are concerned, the author is aware of a number of design modifications and extensions of basic design which would make for a better instrument. This statement in no way implies that less than the maximum effort was applied to make this first instrument a complete success, however. Rather it illustrates the fact that inspiration seldom comes as a flash of light. Development of any idea proceeds slowly. By the time one revelation has been completed, another is being resolved. The second could not come before the first, however, because it depends upon the first for its basic form. The present instrument is therefore already replaced by a better one in the mind of the author. Some design features and performance characteristics thought to be possible in this instrument will be described.

The seismic pendulum could be made very much smaller in mass with use of a differential or toggle type of spring, and also very much simpler in shape. Support for the pendulum could be electrically insulated from the lever and the pendulum itself used as the "throw lever" of the commutation switch. Since the pendulum could be made smaller and the mass of the commutation switch eliminated, the actuation lever could also be less sturdily proportioned and have less mass. This reduction in mass of moving parts would result in a higher upper frequency limit for the same value of external spring counter-balancing force. It is thought that this and other modifications proposed would make possible the construction of an instrument having an extended upper frequency limit of perhaps 100 cycles per second.

A simple system of levers has been devised which could be employed to drive a very much lightened amplitude arm. This train of mechanism could be so reduced in mass as to have a very much higher natural frequency, and therefore very much less mechanical amplification of motion at higher frequencies. Elimination of miter gears and use of precision micro-ball bearings for pin connections would completely eliminate back-lash in the system.

It is believed that both frequency and amplitude traces could be made linear on the transverse chart scale instead of an arc as at present. To accomplish this the chart would be caused to pass over transverse straight-edges placed under the paper in such manner that straight transverse ridges would be presented to blade type pens or pencils on frequency and amplitude arms. Traces on rectangular coordinates would result.

The author would consider use of a very small storage cell recently announced by General Electric Company. This cell reportedly has a remarkably long life, and in this application would serve to reduce both the weight and the space limitation within the instrument.

Further study would be made of meter motion discrimination. It is thought that an over-riding clutch of simple design could be devised which would provide a more positive connection between the armature spindle and the hydraulic damper. Such a device would remove all flexibility and apparent back-lash from this system.

In summation, it appears to the author that instrument improvement along the lines of the basic design offers promise of better and more complete service. In its present form there are certain limitations which restrict its application to specific fields of usefulness. In an improved form it would undoubtedly have an extended field, and might have additional fields. It is the hope of the author that the comments made here will inspire some further development of a basic design which he believes to be worthy.

VII. BIBLIOGRAPHY

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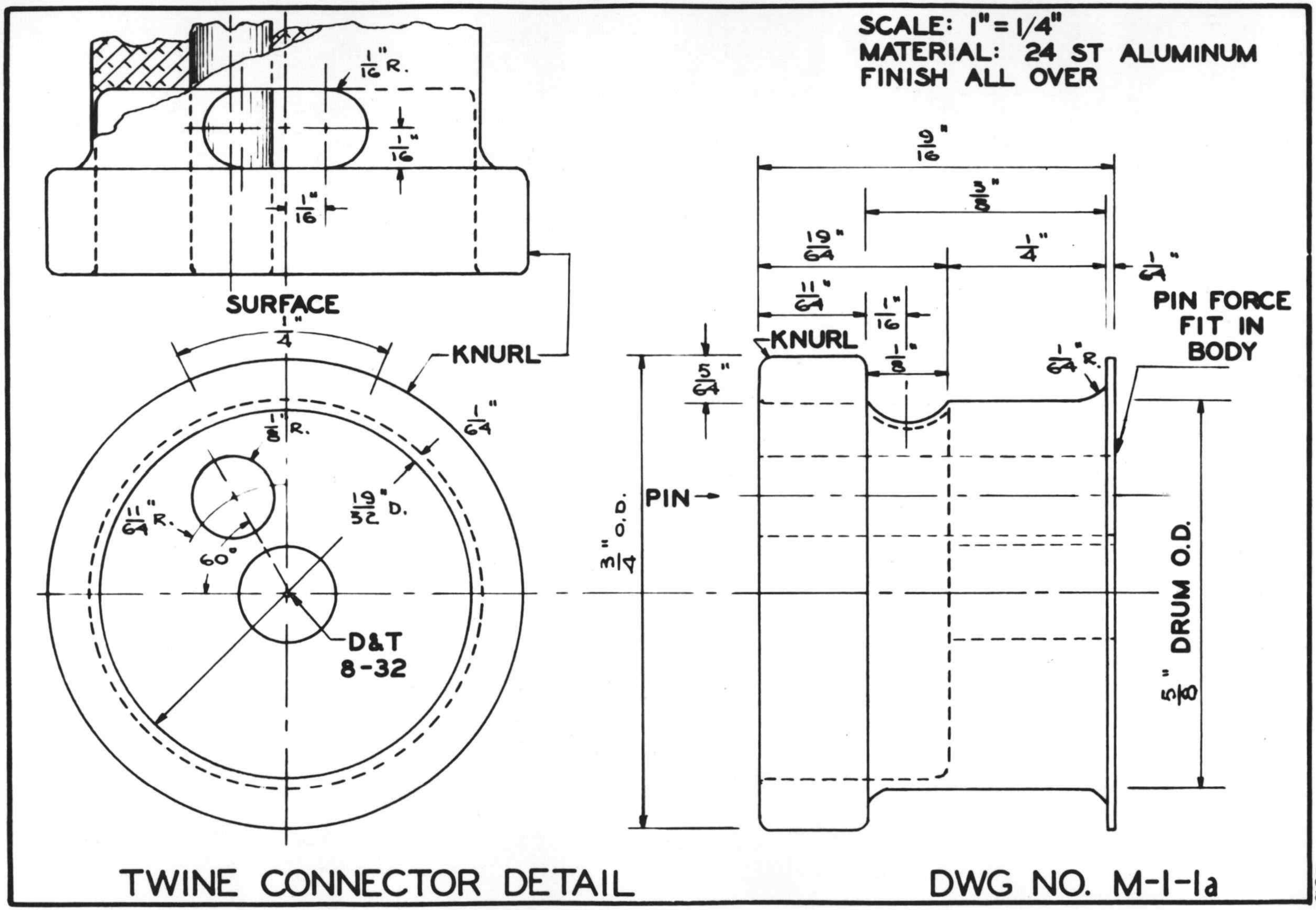
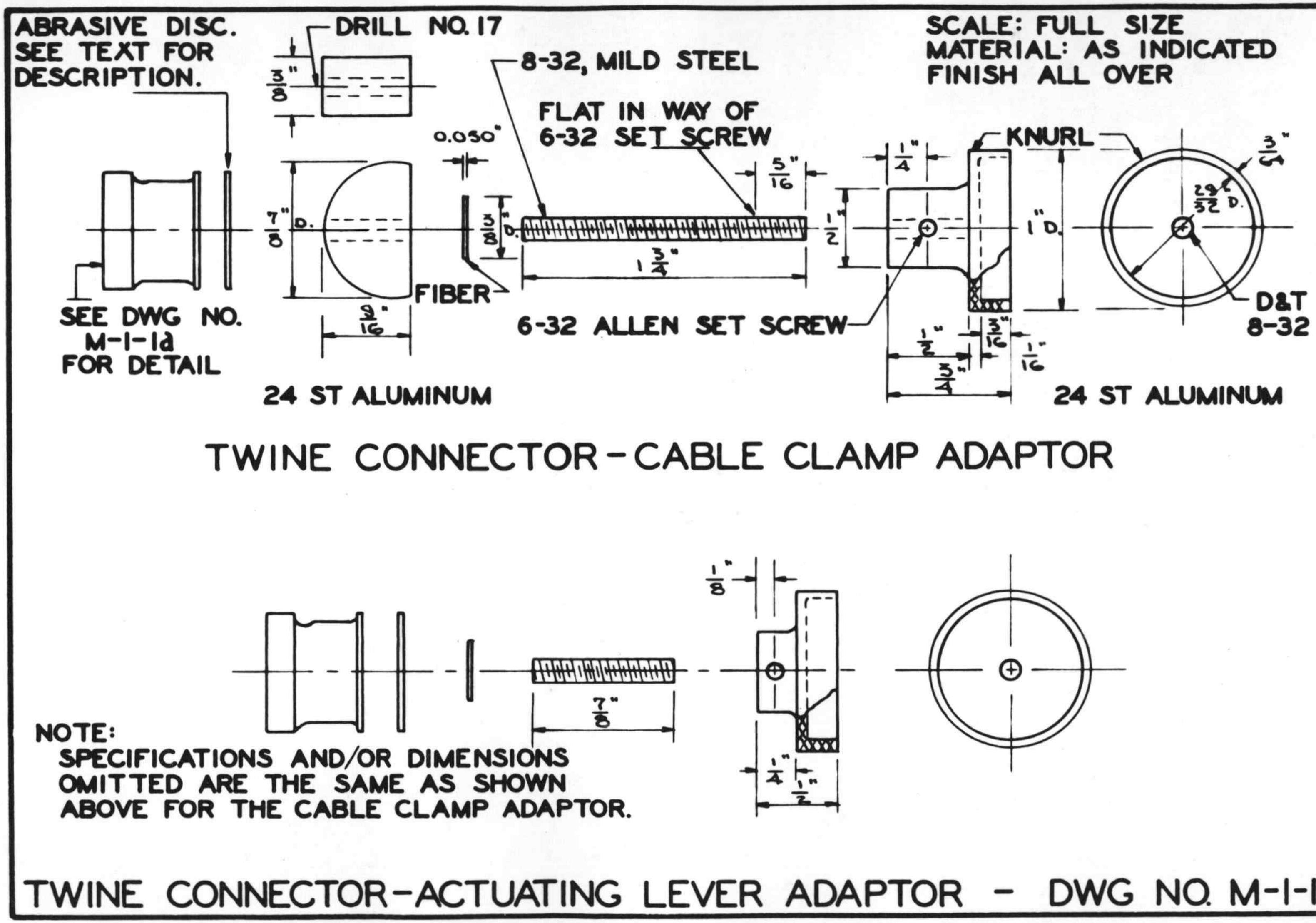
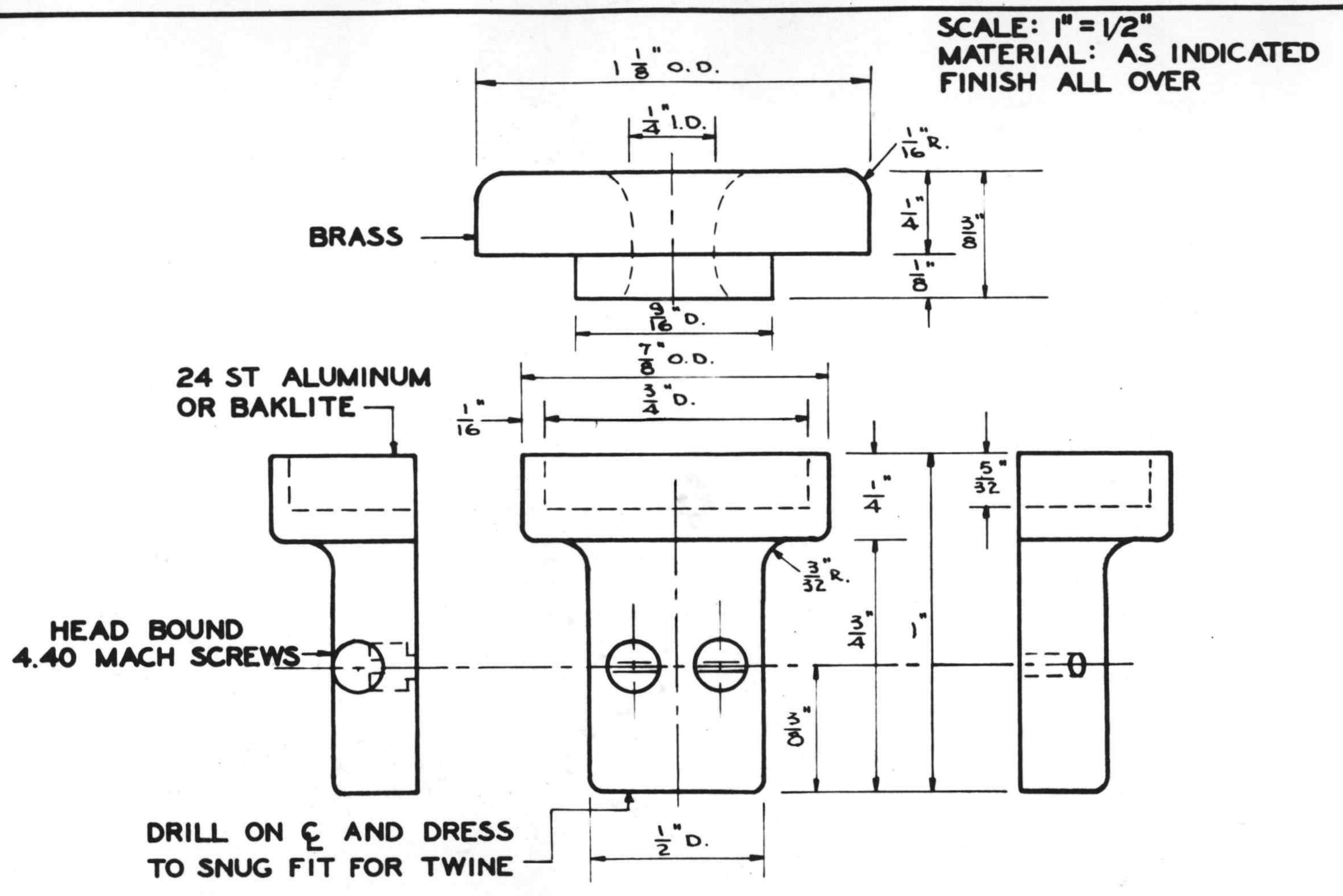
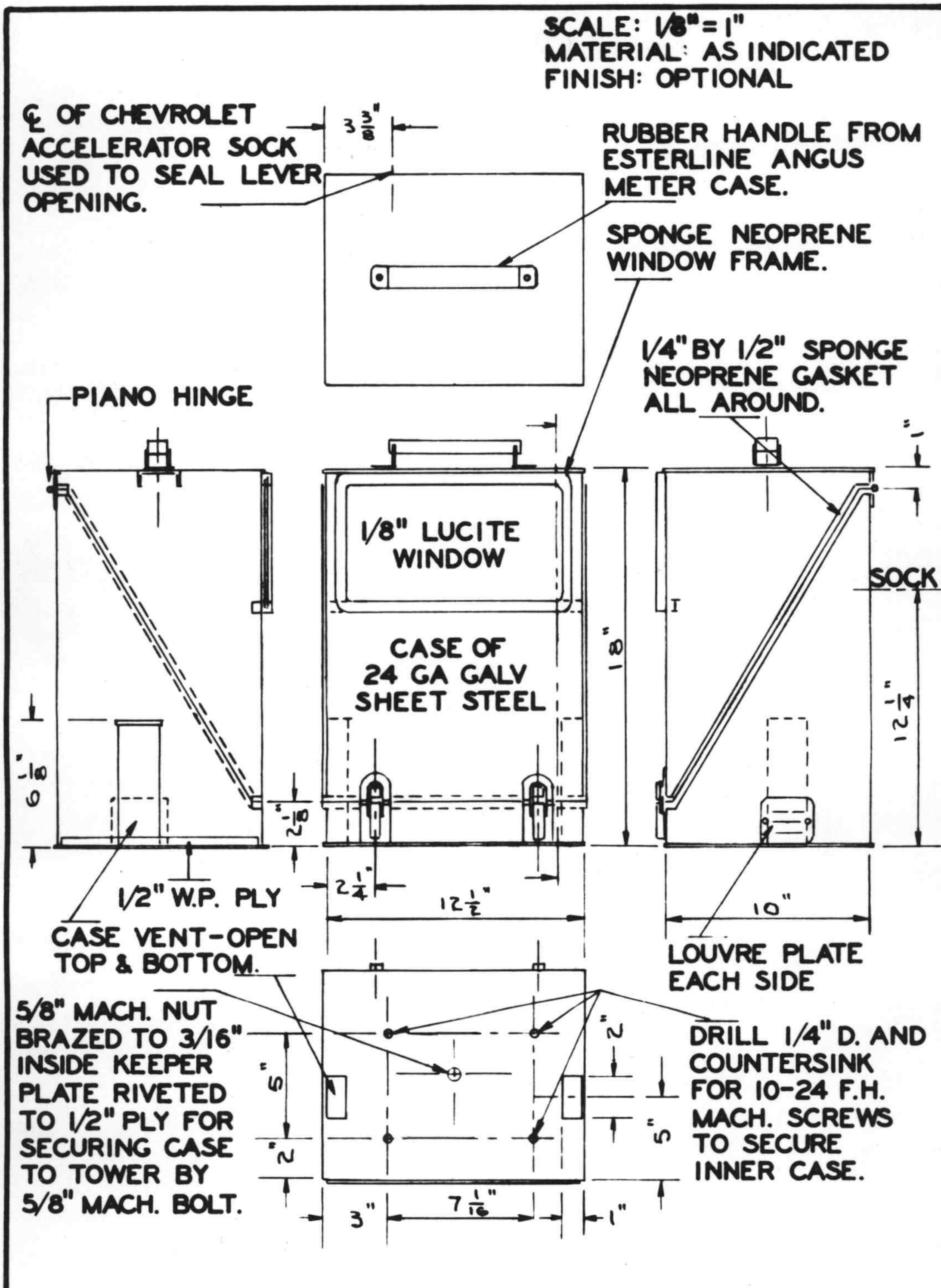
APPENDIX


SCALE: 1" = 1/4"
MATERIAL: 24 ST ALUMINUM
FINISH ALL OVER
1/16" R.
1/16"
1/16"
SURFACE
KNURL
1/4"
1/8" R.
1/64"
19/32" D.
11/64" R.
60°
D&T 8-32
9/16"
3/8"
19/64"
1/4"
1/64"
11/64"
1/16"
KNURL
1/8"
5/64"
1/64" R.
PIN FORCE FIT IN BODY
3/4" O.D.
PIN
5/8" DRUM O.D.
TWINE CONNECTOR DETAIL
DWG NO. M-1-1a


ABRASIVE DISC.
SEE TEXT FOR
DESCRIPTION.
DRILL NO. 17
8-32, MILD STEEL
SCALE: FULL SIZE
MATERIAL: AS INDICATED
FINISH ALL OVER
FLAT IN WAY OF
6-32 SET SCREW
0.050"
KNURL
FIBER
6-32 ALLEN SET SCREW
D&T
8-32
SEE DWG NO.
M-1-1a
FOR DETAIL
24 ST ALUMINUM
24 ST ALUMINUM
TWINE CONNECTOR - CABLE CLAMP ADAPTOR
NOTE:
SPECIFICATIONS AND/OR DIMENSIONS
OMITTED ARE THE SAME AS SHOWN
ABOVE FOR THE CABLE CLAMP ADAPTOR.
TWINE CONNECTOR-ACTUATING LEVER ADAPTOR - DWG NO. M-1-1b



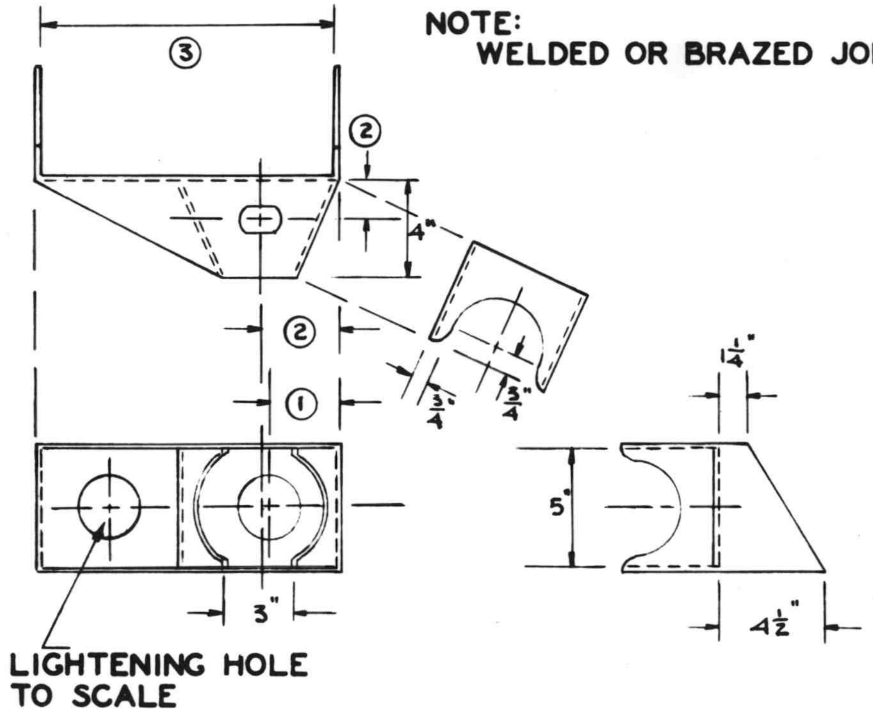
GLASS TWIN VIBRATION DAMPER DETAIL - DWG NO. M-1-2



EXTERNAL CASE DETAIL DWG NO. M-1-3a

SCALE: 1/8" = 1"
MATERIAL: 3/16" 24 ST ALUMINUM
FINISH: BLACK CRACKLE

NOTE:
WELDED OR BRAZED JOINURE.



① DIMENSIONS TO SUIT SOCK LOCATION ON BACK OF EXTERNAL CASE. SEE DWG NO. MI-3a.

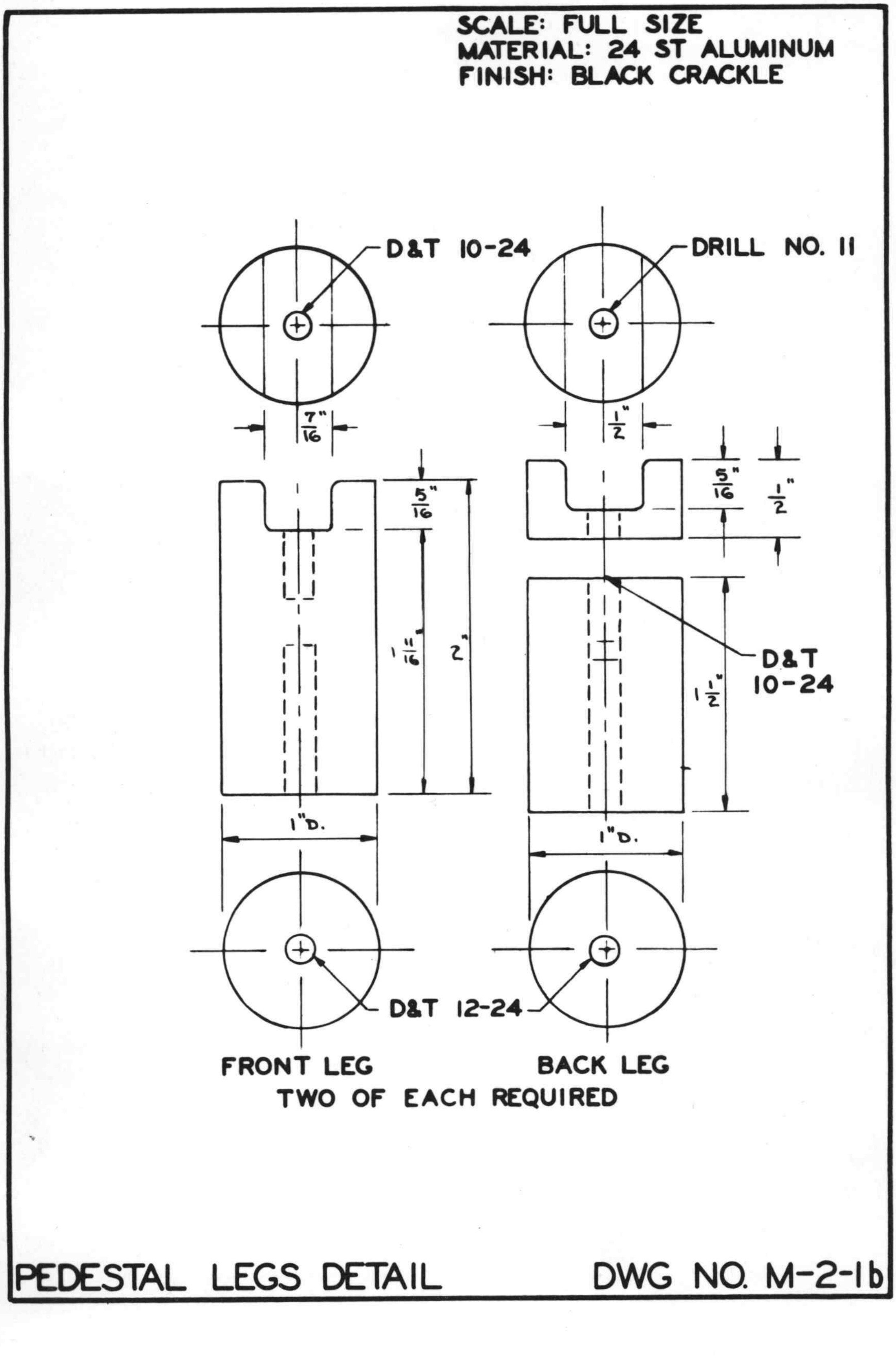
② DIMENSIONS TO SUIT LEVER PLACEMENT. HOLE TO ACCOMMODATE GLASS TWINE.

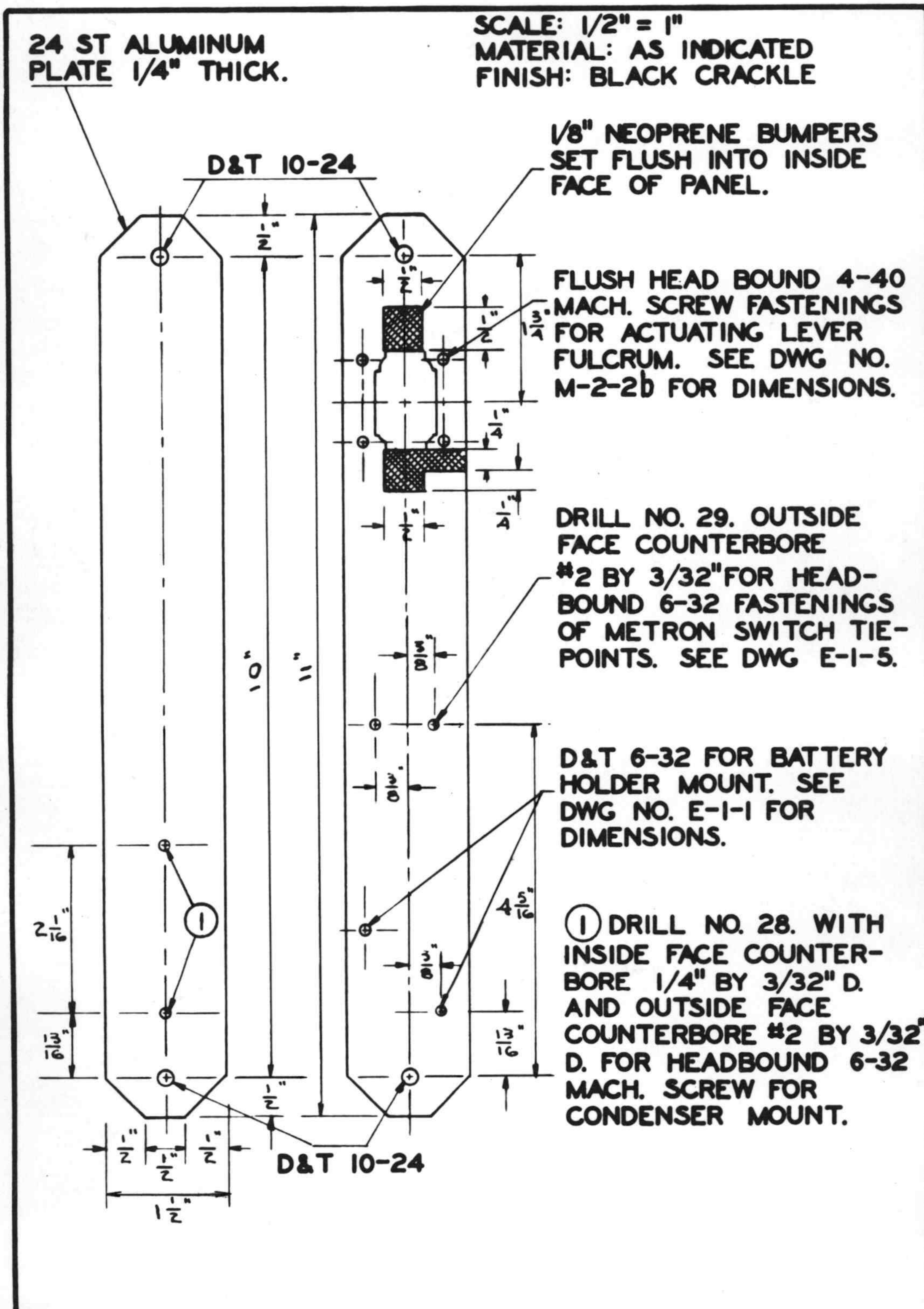
③ DIMENSIONS TO FIT OUTSIDE OF EXTERNAL CASE.

NOTE:
SPRING SUPPORT MOUNTED ON OUTSIDE BACK OF EXTERNAL CASE PER ASSEMBLY PHOTOGRAPHS. MOUNTING STIFFNESS SUPPLEMENTED BY INSIDE ANGLE BRACKETS WHICH CARRY LOADING TO BOLT CONNECTIONS ON BASEBOARD OF CASE.

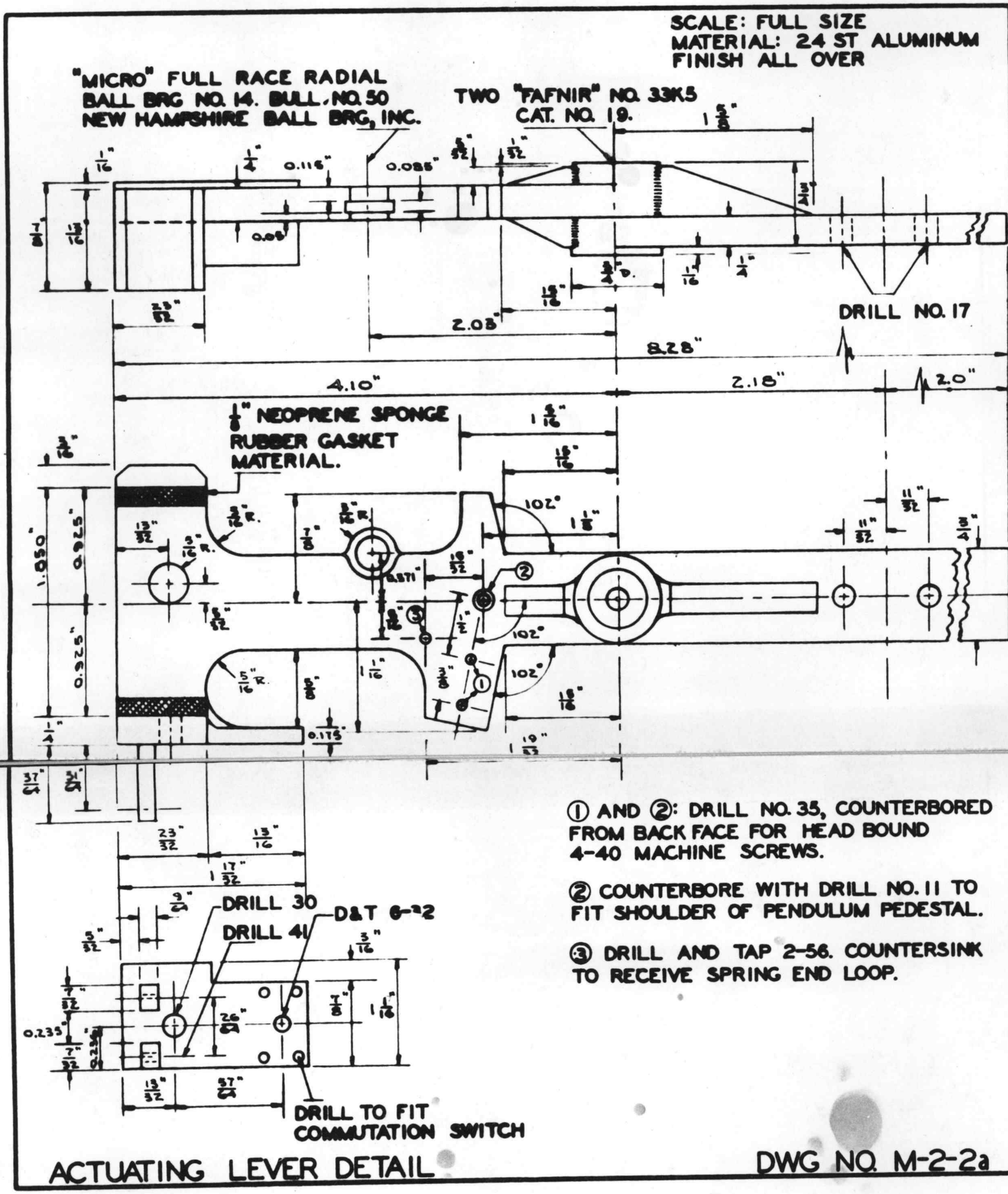
SEE DWG NO. MI-3C FOR COUNTER SPRING DETAIL.

COUNTER SPRING SUPPORT - DWG NO. M-1-3b

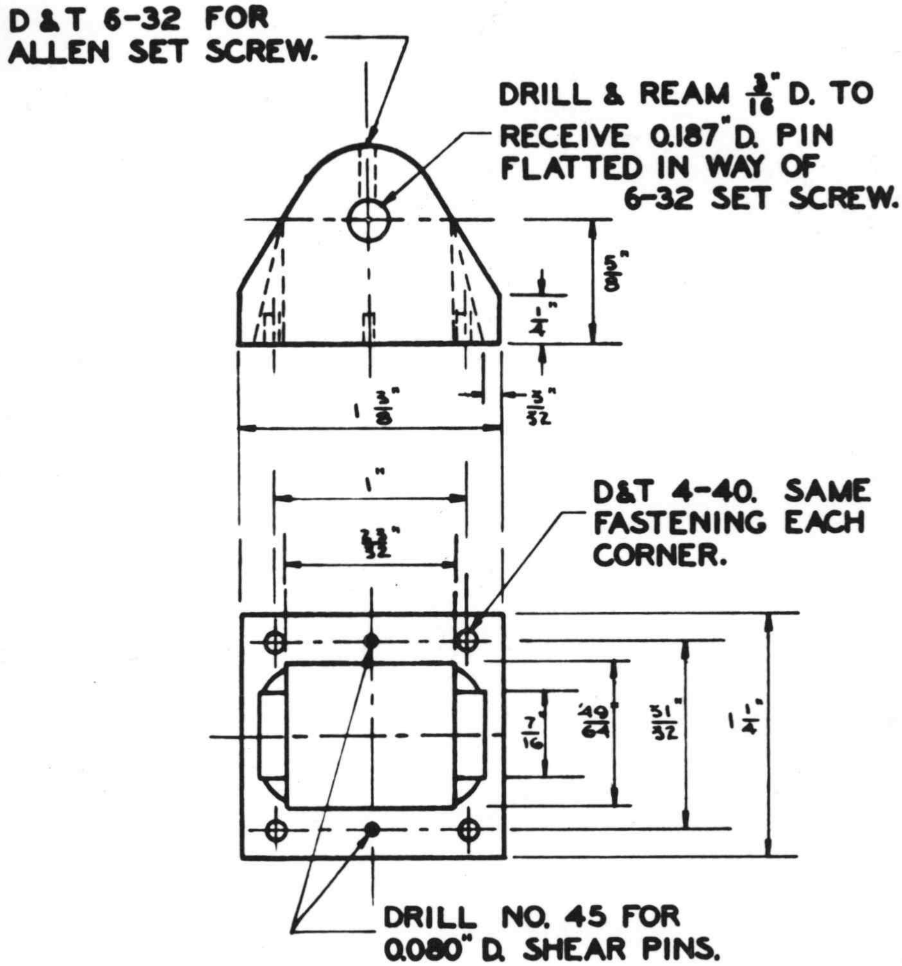

SCALE: FULL SIZE
MATERIAL: 24 ST ALUMINUM
FINISH: BLACK CRACKLE
D&T 10-24
DRILL NO. 11
7"/16
1"/2
5"/16
1"/2
5"/16
1 11/16"
2"
D&T 10-24
1 1/2"
1"D.
1"D.
D&T 12-24
FRONT LEG
BACK LEG
TWO OF EACH REQUIRED
PEDESTAL LEGS DETAIL
DWG NO. M-2-1b



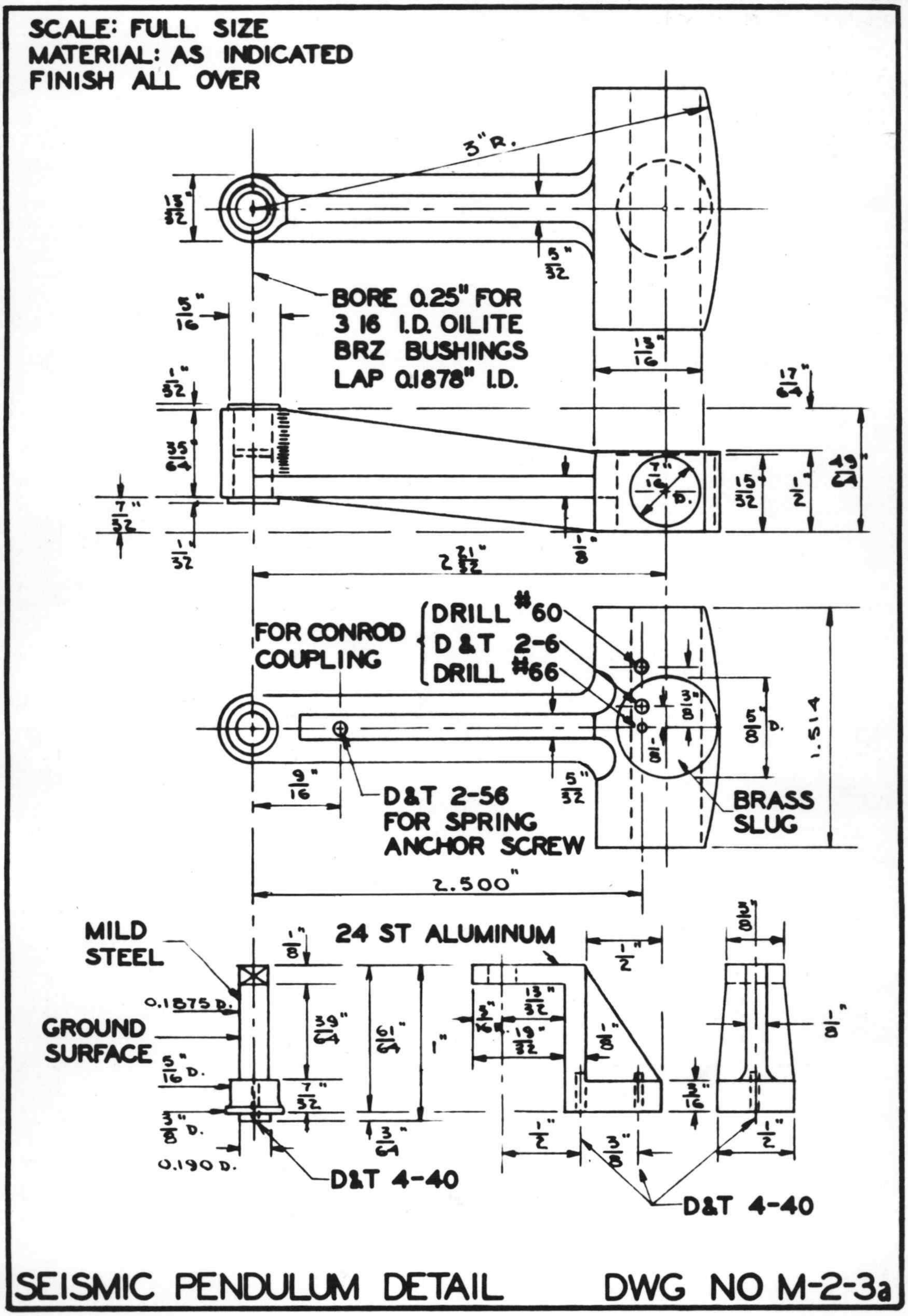
MOUNTING PANELS DETAIL - DWG NO. M-2-1c


SCALE: FULL SIZE
MATERIAL: 24 ST ALUMINUM
FINISH ALL OVER
"MICRO" FULL RACE RADIAL BALL BRG NO. 14. BULL. NO. 50 NEW HAMPSHIRE BALL BRG, INC.
TWO "FAFNIR" NO. 33K5 CAT. NO. 19.
DRILL NO. 17
1/8" NEOPRENE SPONGE RUBBER GASKET MATERIAL.
DRILL 30
DRILL 41
D&T 6-32
DRILL TO FIT COMMUTATION SWITCH
① AND ②: DRILL NO. 35, COUNTERBORED FROM BACK FACE FOR HEAD BOUND 4-40 MACHINE SCREWS.
② COUNTERBORE WITH DRILL NO. 11 TO FIT SHOULDER OF PENDULUM PEDESTAL.
③ DRILL AND TAP 2-56. COUNTERSINK TO RECEIVE SPRING END LOOP.
ACTUATING LEVER DETAIL
DWG NO. M-2-2a

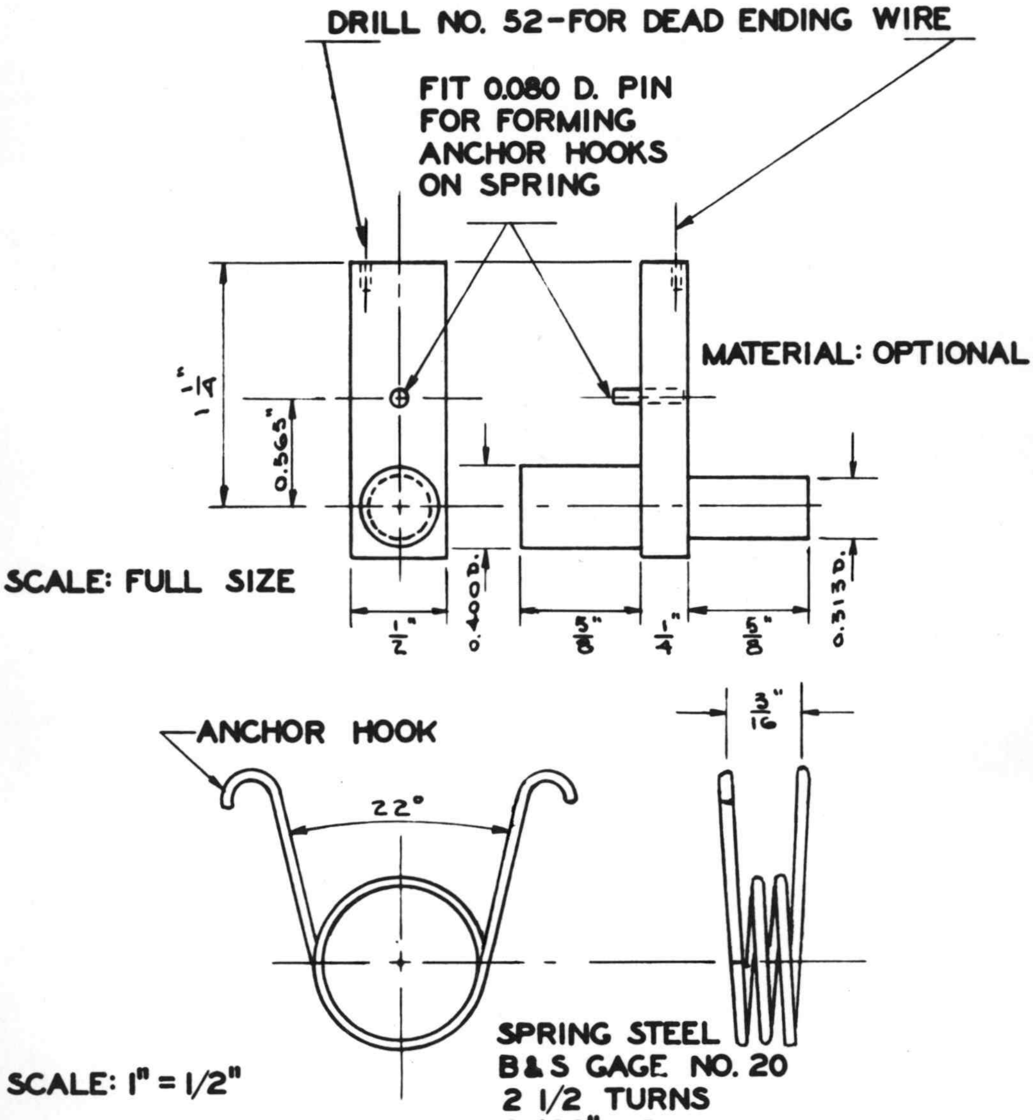
SCALE: FULL SIZE
MATERIAL: 24 ST ALUMINUM
FINISH ALL OVER



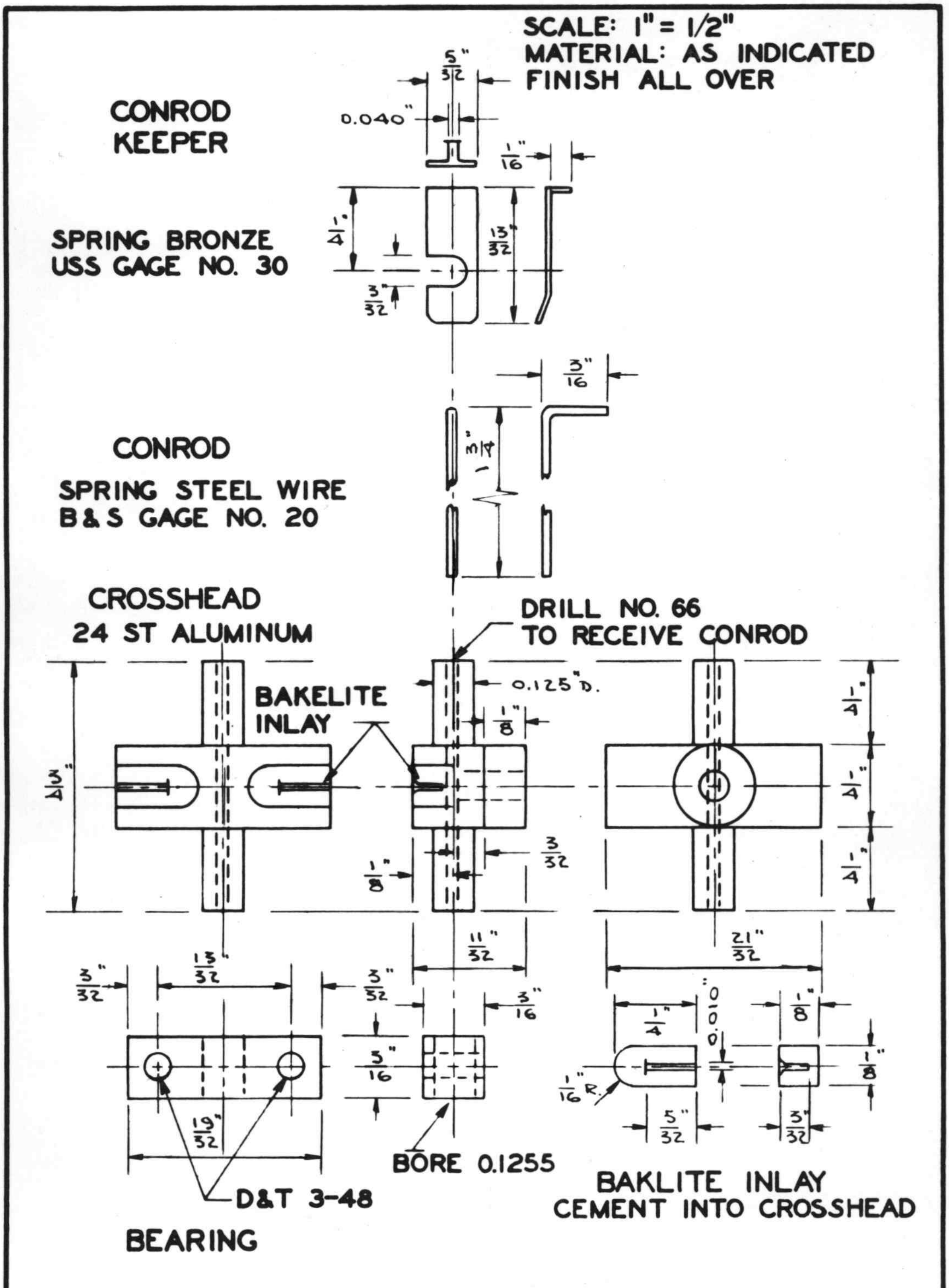
ACTUATING LEVER FULCRUM - DWG NO. M-2-2b


SCALE: FULL SIZE
MATERIAL: AS INDICATED
FINISH ALL OVER
3" R.
13/32
5"/32
BORE 0.25" FOR
3 16 I.D. OILITE
BRZ BUSHINGS
LAP 0.1878" I.D.
5"/16
13"/16
17"/64
1"/32
35"/64
7"/16 D.
15"/32
1"/2
49"/64
7"/32
1"/32
1"/8
2 21"/32
FOR CONROD
COUPLING
DRILL #60
D&T 2-6
DRILL #66
3"/8
5"/8 D.
1.514
1/8
9"/16
D&T 2-56
FOR SPRING
ANCHOR SCREW
5"/32
BRASS
SLUG
2.500"
MILD
STEEL
24 ST ALUMINUM
1"/8
0.1875 D.
GROUND
SURFACE
5"/16 D.
3"/8 D.
0.190 D.
39"/64
61"/64
1"
7"/32
3"/64
D&T 4-40
13"/32
3"/16
19"/32
1"/8
1"/2
3"/16
1"/2
3"/8
3"/8
1"/8
1"/2
D&T 4-40
SEISMIC PENDULUM DETAIL
DWG NO M-2-3a

SCALE: FULL SIZE & 1" = 1/2"
MATERIAL: AS INDICATED
FINISH: OPTIONAL

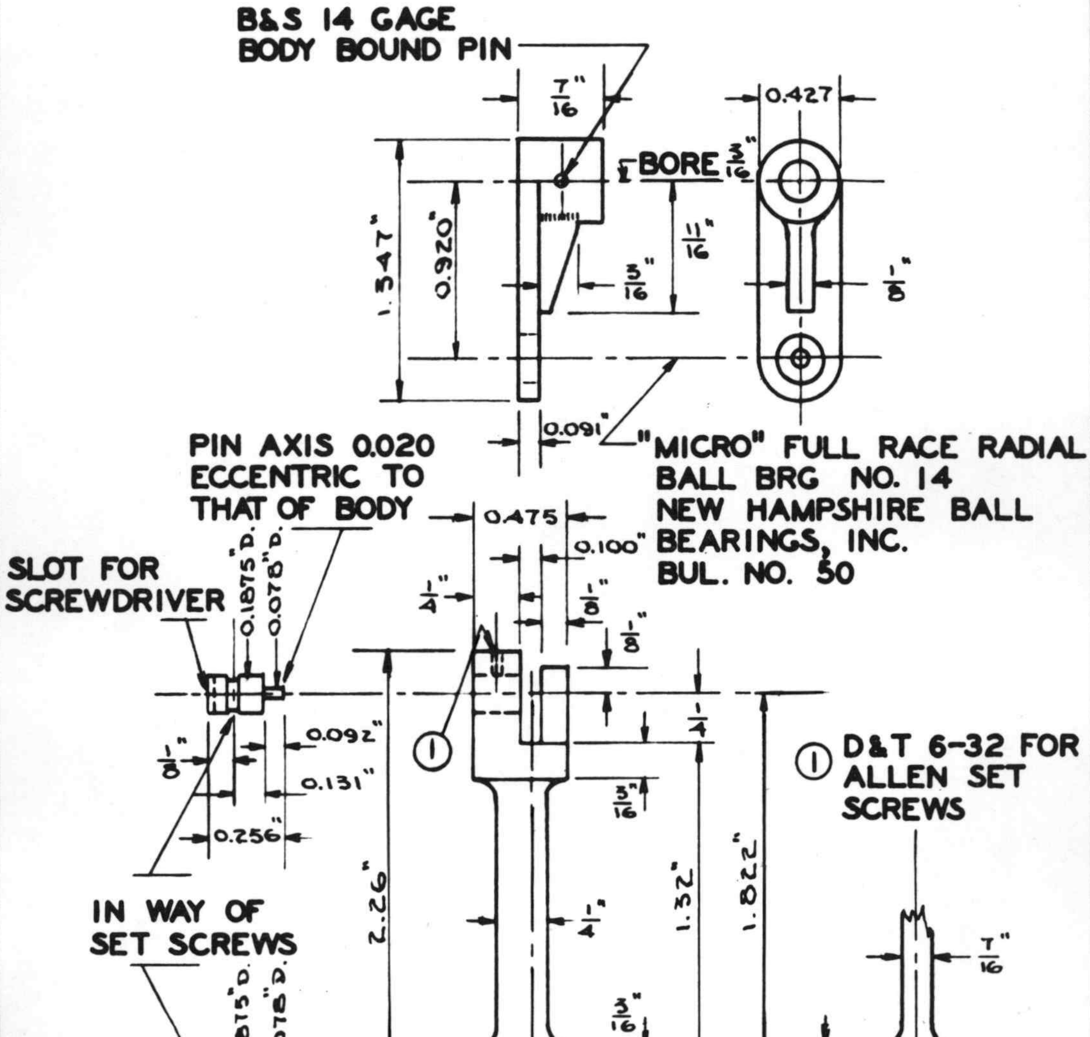


PENDULUM SPRING & MANDREL- DWG NO. M-2-3b



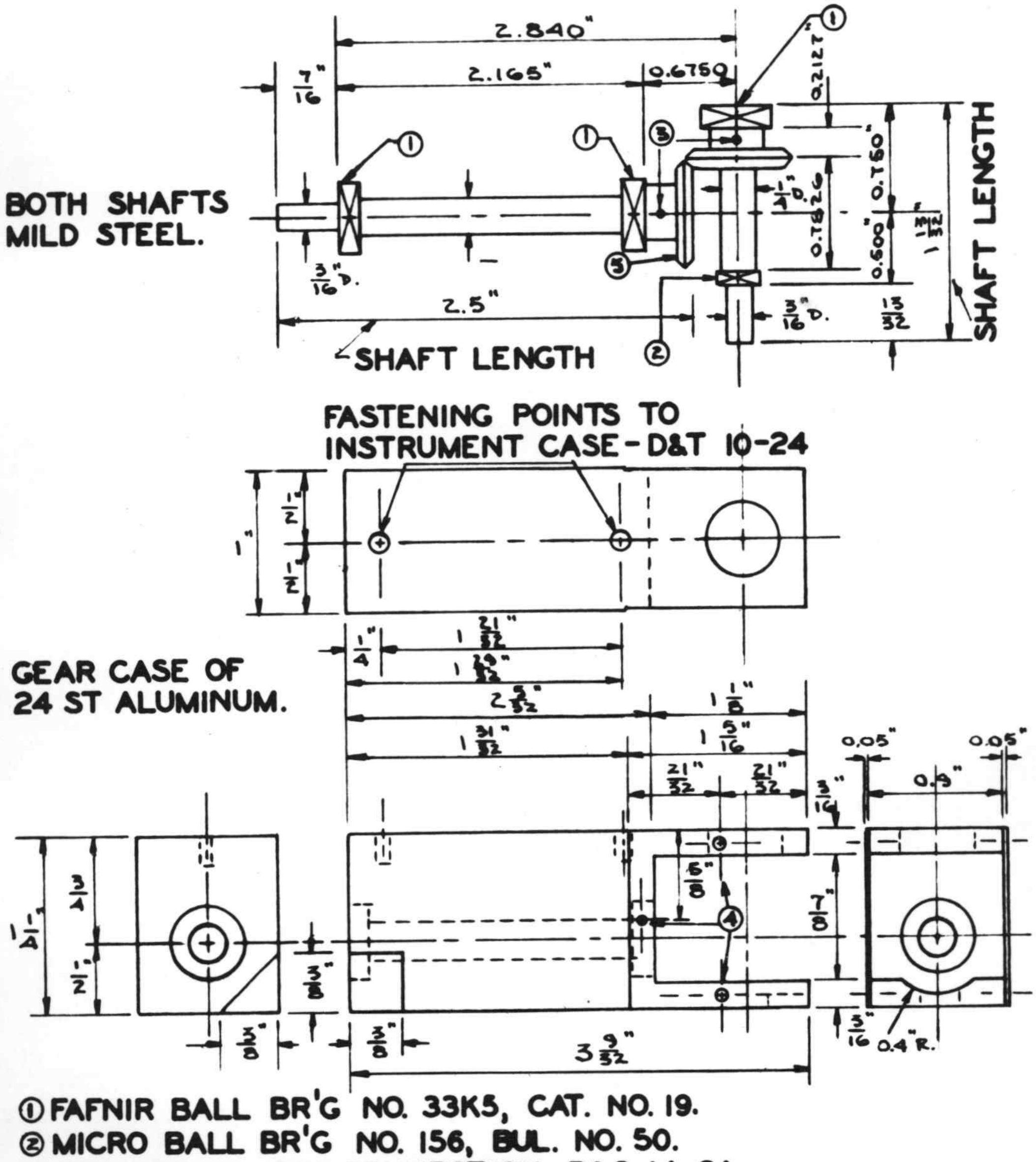
CONROD-CROSSHEAD DETAIL - DWG NO. M-2-3c

SCALE: FULL SIZE
MATERIAL: 24 ST ALUMINUM
FINISH ALL OVER



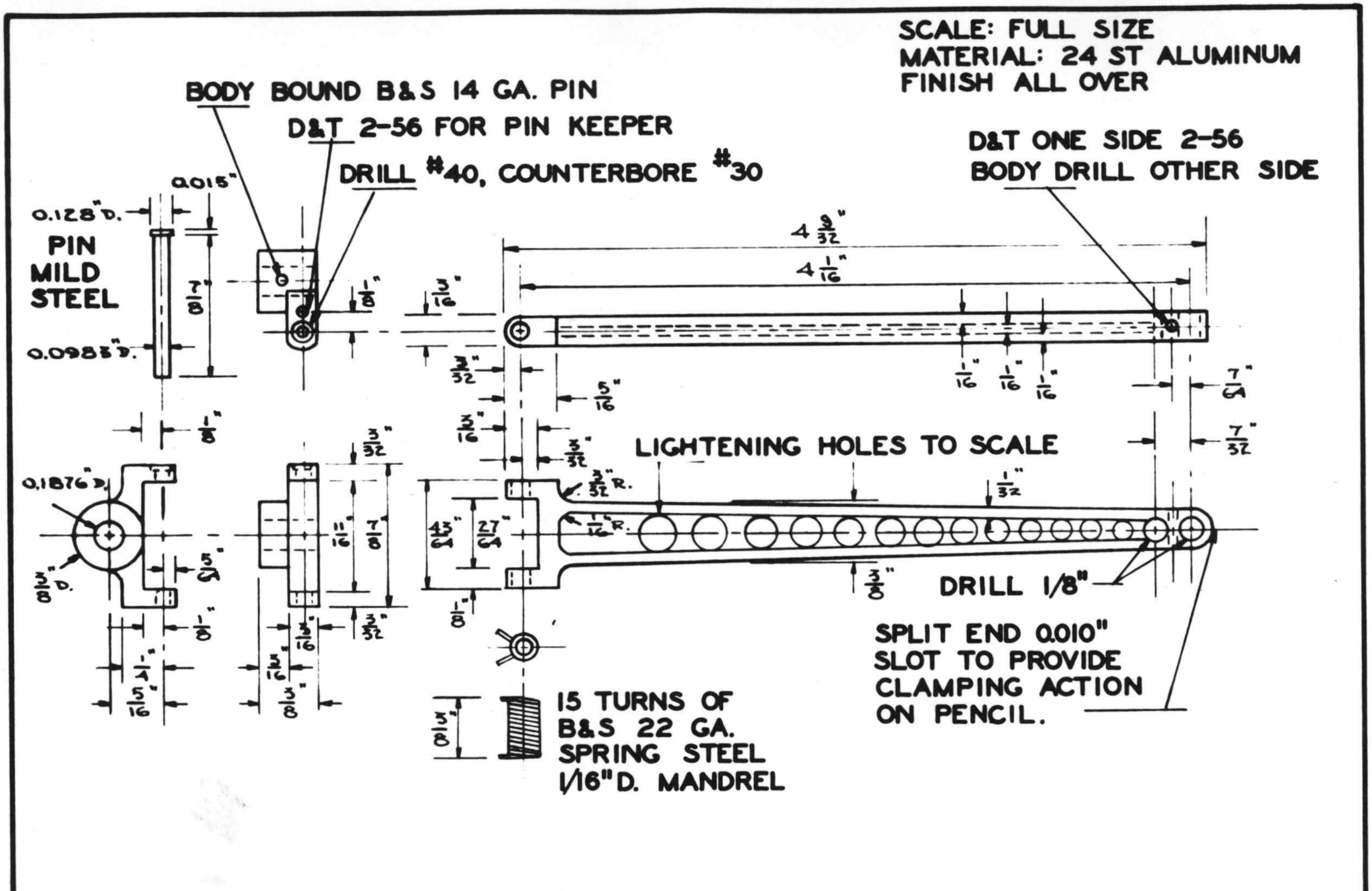
CONROD-LEVER DETAIL DWG NO. M-2-4a

SCALE: 3/4" = 1"
MATERIAL: AS INDICATED
FINISH ALL OVER



① FAFNIR BALL BR'G NO. 33K5, CAT. NO. 19.
② MICRO BALL BR'G NO. 156, BUL. NO. 50.
③ BODY BOUND PIN CONNECTION, B&S 14 GA.
④ FASTENING POINTS FRONT & BACK FOR GEAR CASE COVER. BEND COVER FROM B&S 36 GA. ALUMINUM TO COVER FRONT, END, AND BACK OF GEAR SPACE.
⑤ 36 T. 48 PITCH MITER, P.D. = 0.75".

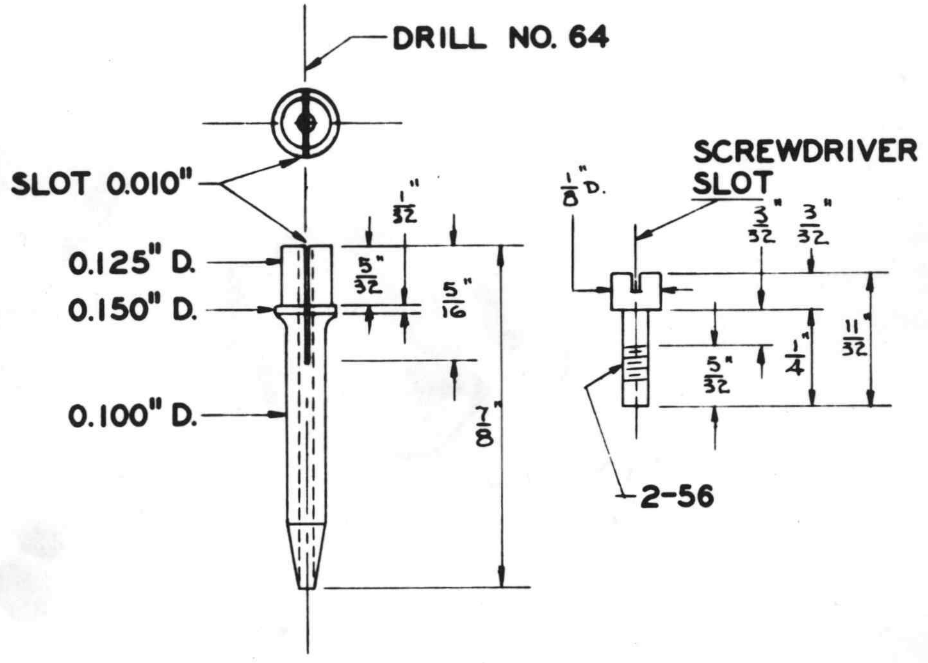
GEAR MOUNT DETAIL DWG NO. M-2-4b



AMPLITUDE RECORDING ARM DETAIL

DWG NO. M-2-4c

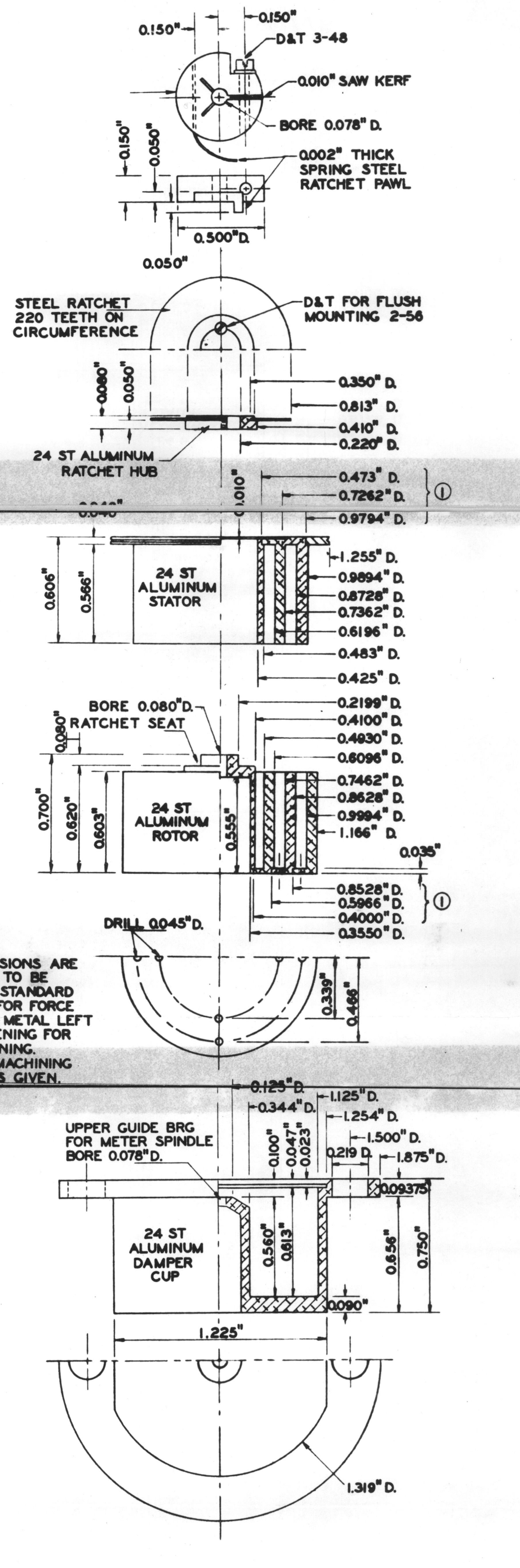
SCALE: 1" = 1/2"
MATERIAL: MAGNESIUM
FINISH ALL OVER



NOTE: THE PENCIL IS DRILLED TO HOLD "FINELINE" LEAD. THE SCREW SERVES BOTH TO HOLD LEAD IN PENCIL AND PENCIL IN ARM. SEE DWG NO. M-2-4C FOR APPLICATION.

AMPLITUDE PENCIL DETAIL - DWG NO. M-2-4d

SCALE: 1" = 1/2"
MATERIAL: AS INDICATED
FINISH ALL OVER

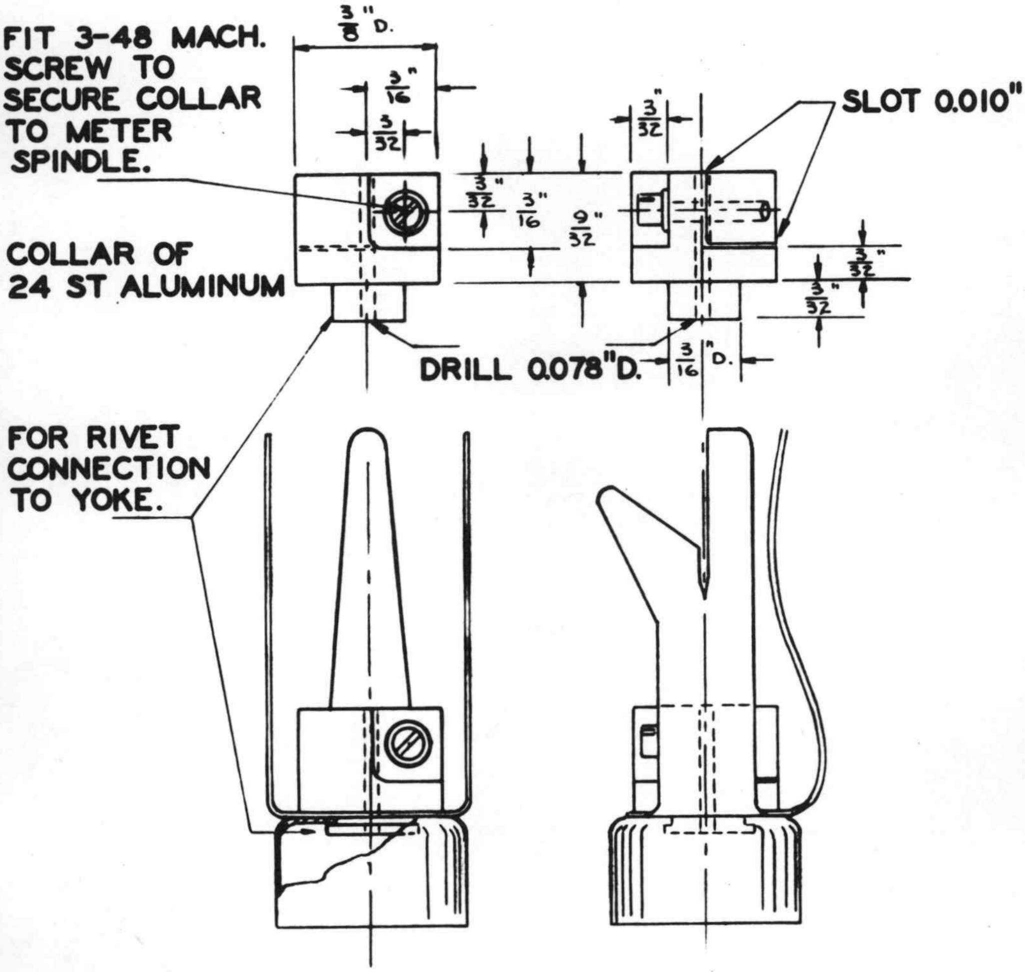


(1) FITTED DIMENSIONS ARE GIVEN. PARTS TO BE MACHINED TO STANDARD TOLERANCES FOR FORCE FIT AND TRIM METAL LEFT TO ALLOW PEENING FOR SECURE FASTENING. SUBSEQUENT MACHINING TO DIMENSIONS GIVEN.

METER DAMPER DETAIL

DWG NO. M-2-5a

SCALE: 1" = 1/2"
MATERIAL: AS INDICATED
FINISH ALL OVER

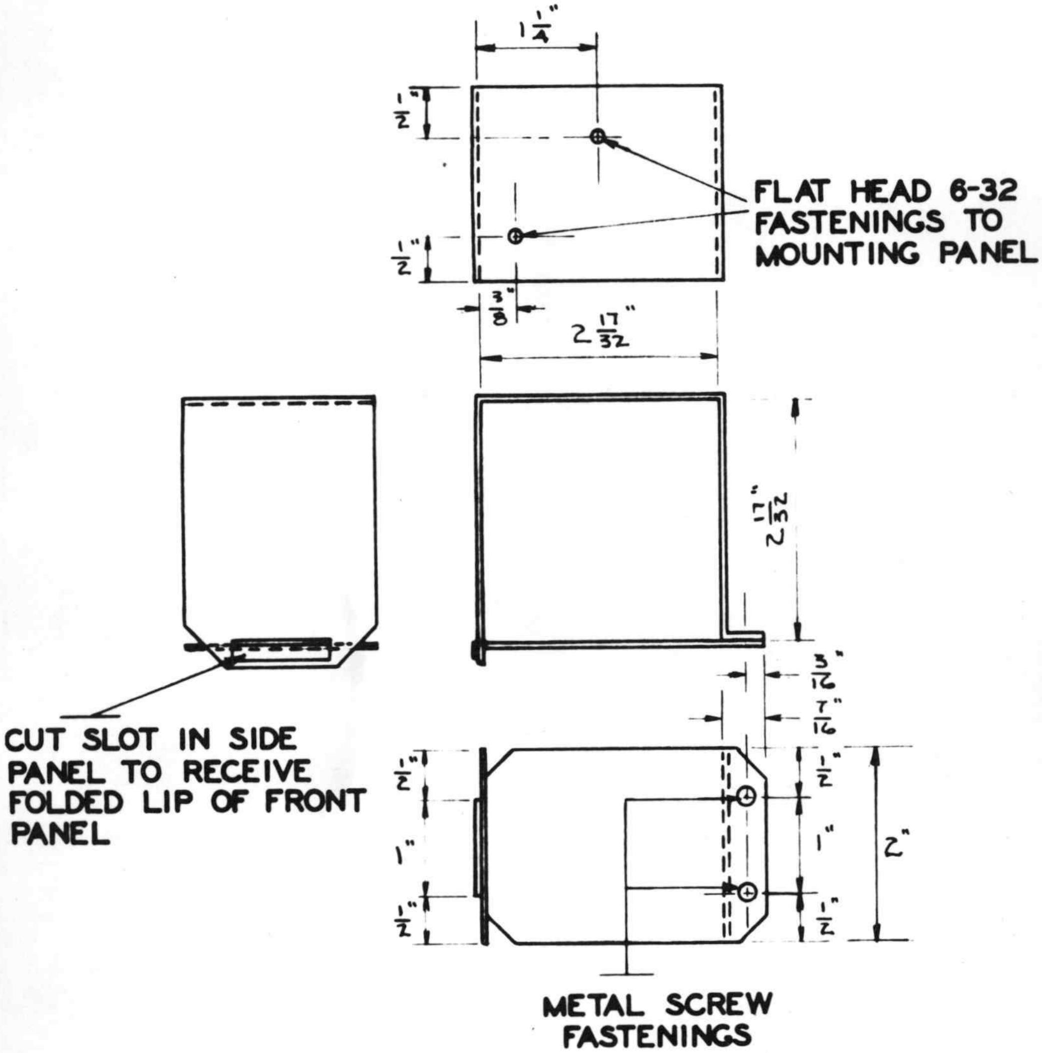


YOKE COLLAR FURNISHED ORIGINALLY WITH METER FOR SOLDER CONNECTION OF YOKE TO SPINDLE TO BE REMOVED, AND THE ABOVE 24 ST ALUMINUM COLLAR SUBSTITUTED FOR EASE IN GAINING ACCESS TO METER DAMPER.

PEN YOKE DETAIL

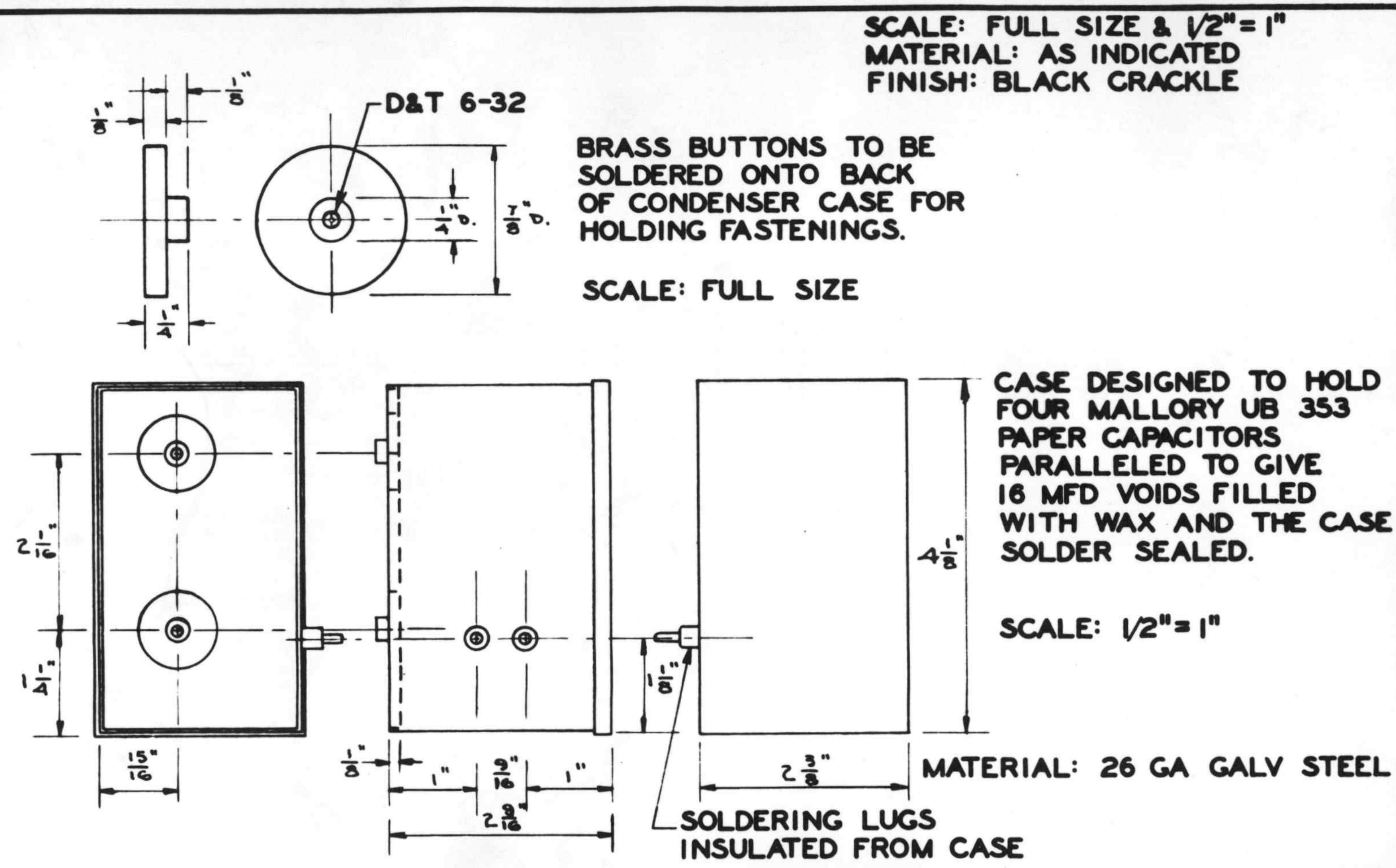
DWG NO. M-2-5b

SCALE: 1/2" = 1"
MATERIAL: 14 GA ALUMINUM
FINISH: BLACK CRACKLE



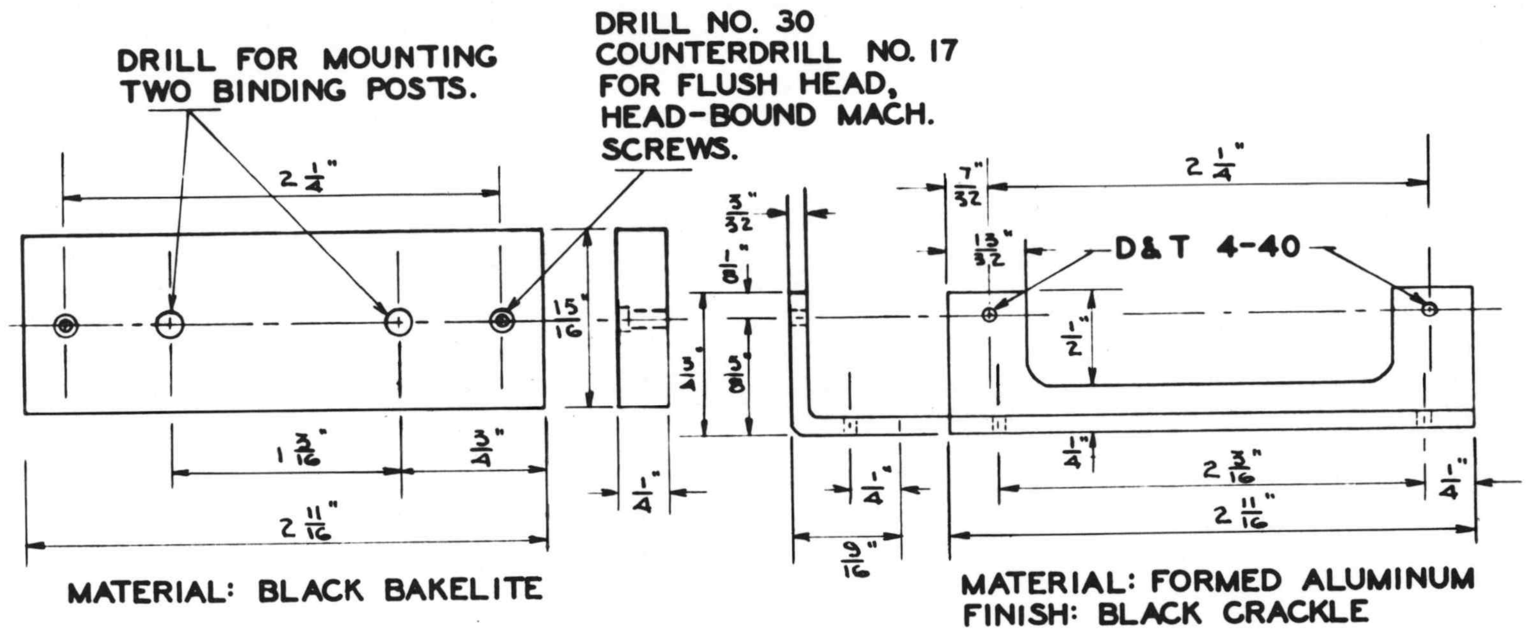
BATTERY HOLDER DETAIL

DWG NO. E-1-1



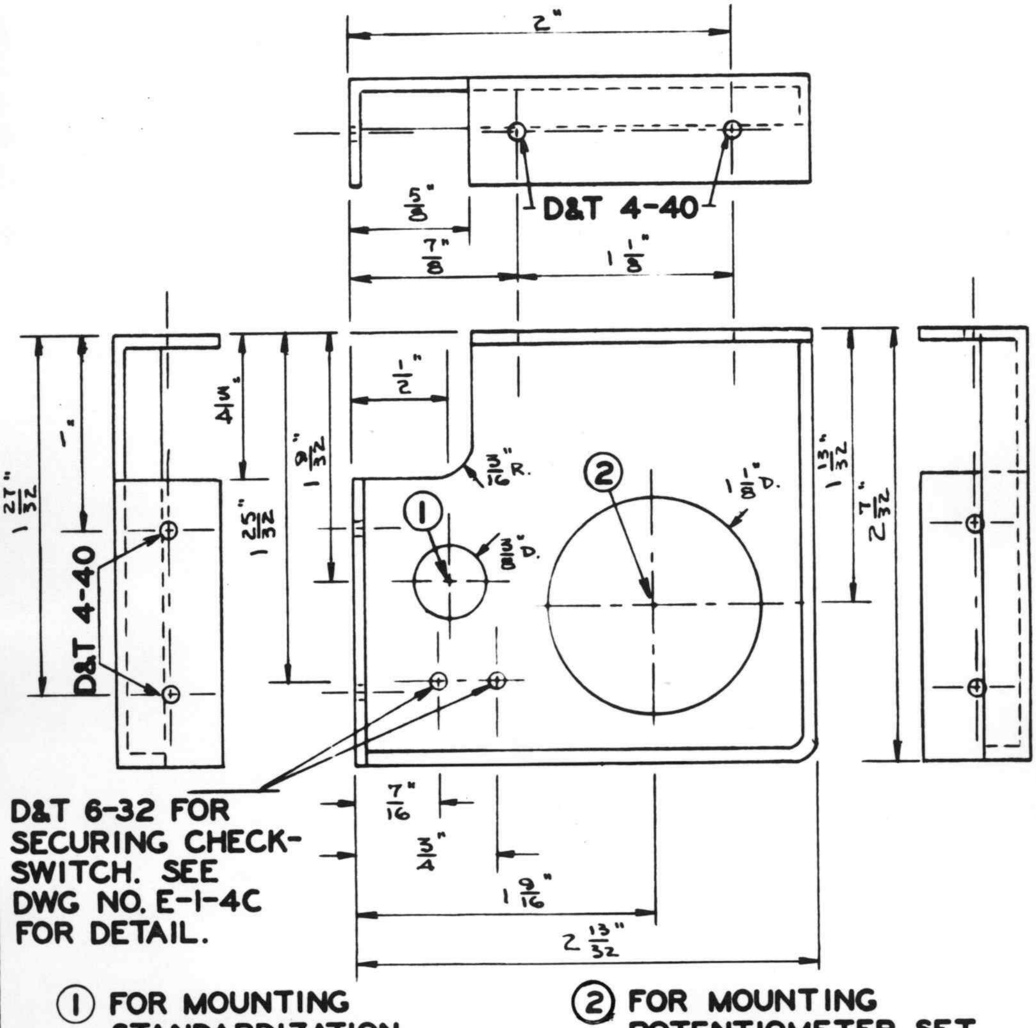
CONDENSER CASE DETAIL DWG NO. E-1-2

SCALE: FULL SIZE
MATERIAL: AS INDICATED
FINISH: AS INDICATED



METER TERMINAL BOARD DETAIL DWG NO. E-1-3

SCALE: FULL SIZE
MATERIAL: 14 B&S GA, 24 ST ALUMINUM
FINISH: BLACK CRACKLE ENAMEL

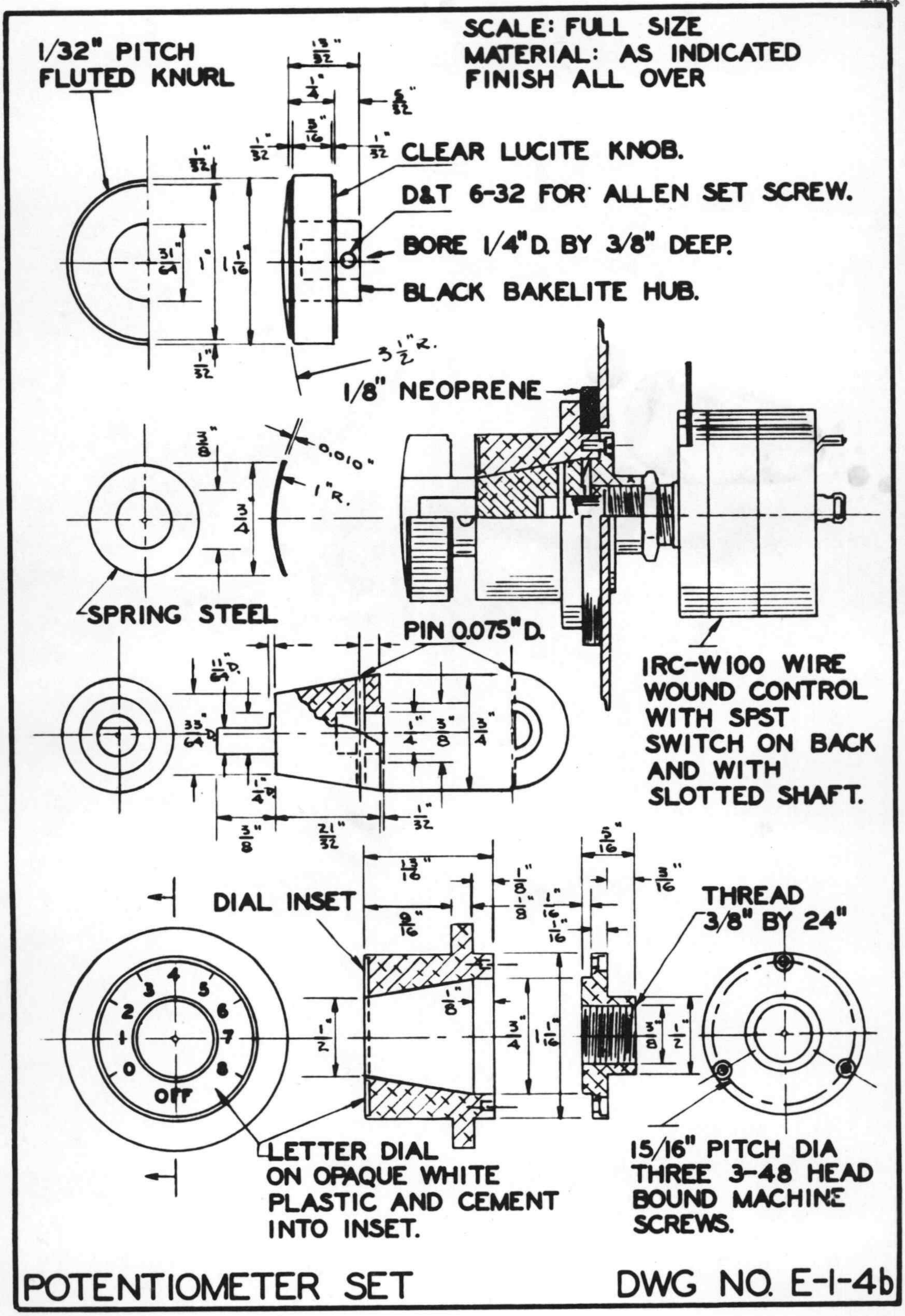


① FOR MOUNTING STANDARDIZATION POTENTIOMETER. SEE DWG NO. E-1-4C FOR DETAIL.

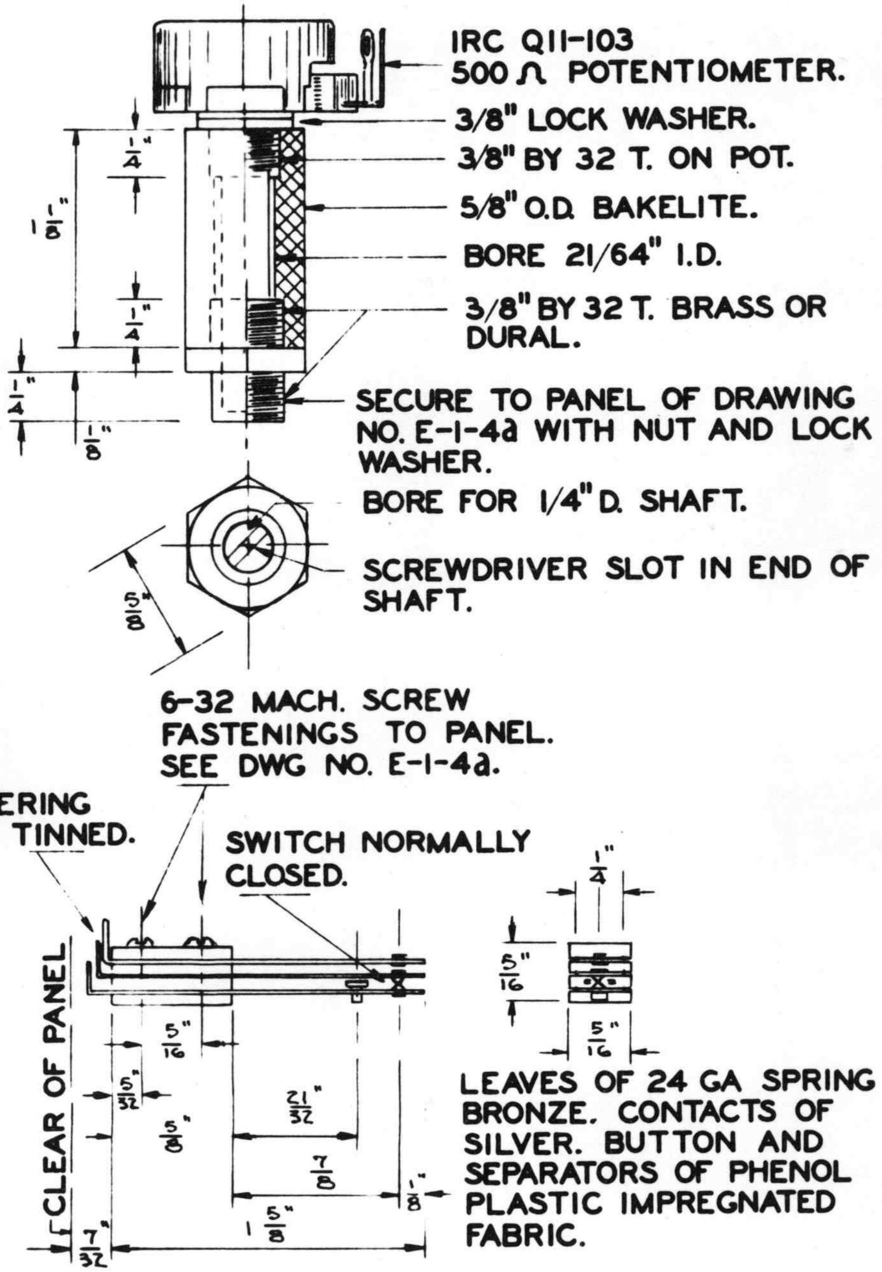
② FOR MOUNTING POTENTIOMETER SET AND ON-OFF SWITCH. SEE DWG NO. E-1-4b FOR DETAIL.

NOTE:
THE ABOVE AS SEEN FROM BACK OF PANEL.

CONTROL PANEL DETAIL DWG NO E-1-4a


SCALE: FULL SIZE
MATERIAL: AS INDICATED
FINISH ALL OVER
1/32" PITCH
FLUTED KNURL
CLEAR LUCITE KNOB.
D&T 6-32 FOR ALLEN SET SCREW.
BORE 1/4" D. BY 3/8" DEEP.
BLACK BAKELITE HUB.
3 1/2" R.
1/8" NEOPRENE
0.010"
1" R.
SPRING STEEL
PIN 0.075" D.
IRC-W100 WIRE
WOUND CONTROL
WITH SPST
SWITCH ON BACK
AND WITH
SLOTTED SHAFT.
DIAL INSET
THREAD
3/8" BY 24"
LETTER DIAL
ON OPAQUE WHITE
PLASTIC AND CEMENT
INTO INSET.
15/16" PITCH DIA
THREE 3-48 HEAD
BOUND MACHINE
SCREWS.
OFF
POTENTIOMETER SET
DWG NO. E-1-4b

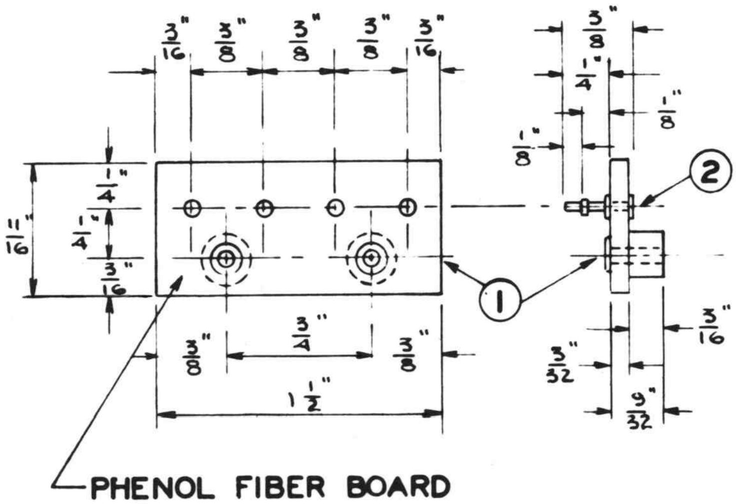
SCALE: FULL SIZE
MATERIAL: AS INDICATED
FINISH ALL OVER



STANDARDIZATION POTENTIOMETER AND CHECK SWITCH DETAIL

DWG NO. E-1-4c

SCALE: FULL SIZE
MATERIAL: AS INDICATED
FINISH: OPTIONAL

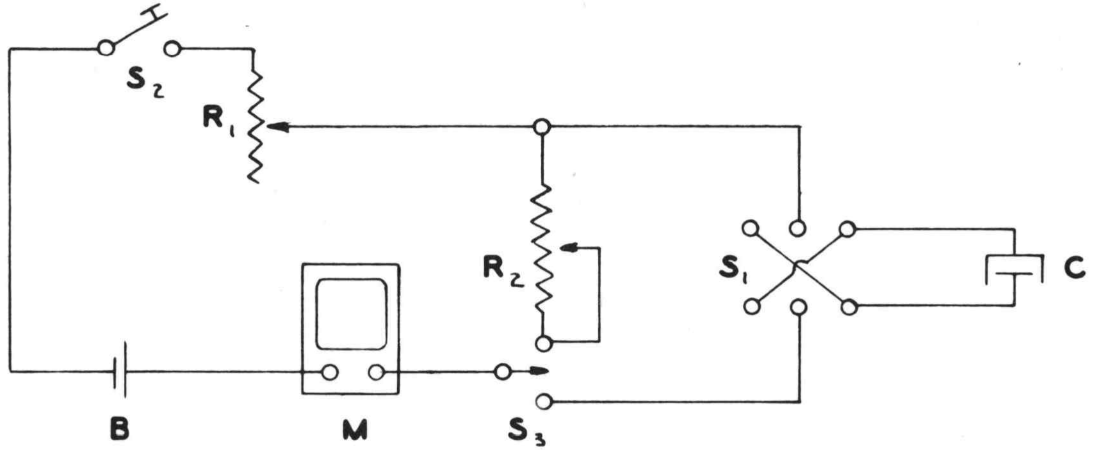


(1) STAND-OFFS OF 24 ST ALUMINUM THROUGH-RIVETED TO BOARD AND D&T 6-32 FOR FASTENING TO MOUNTING PANEL.

(2) BRASS SOLDERING LUGS THROUGH-RIVETED TO BOARD.

COMMUTATION SWITCH TIE-POINT

DWG NO. E-1-5



M ESTERLINE-ANGUS 0-5 DC MA RECORDER.

S_1 METRON COMMUTATING SWITCH.

S_2 SPST SWITCH FOR BACK MOUNTING ON R_1.

S_3 SPDT LEAF SWITCH. SEE DWG NO. E-1-4C.

B BURGESS, 4F, EVEREADY 742, OR RCA, VS004, 1 1/2 VOLT CELL.

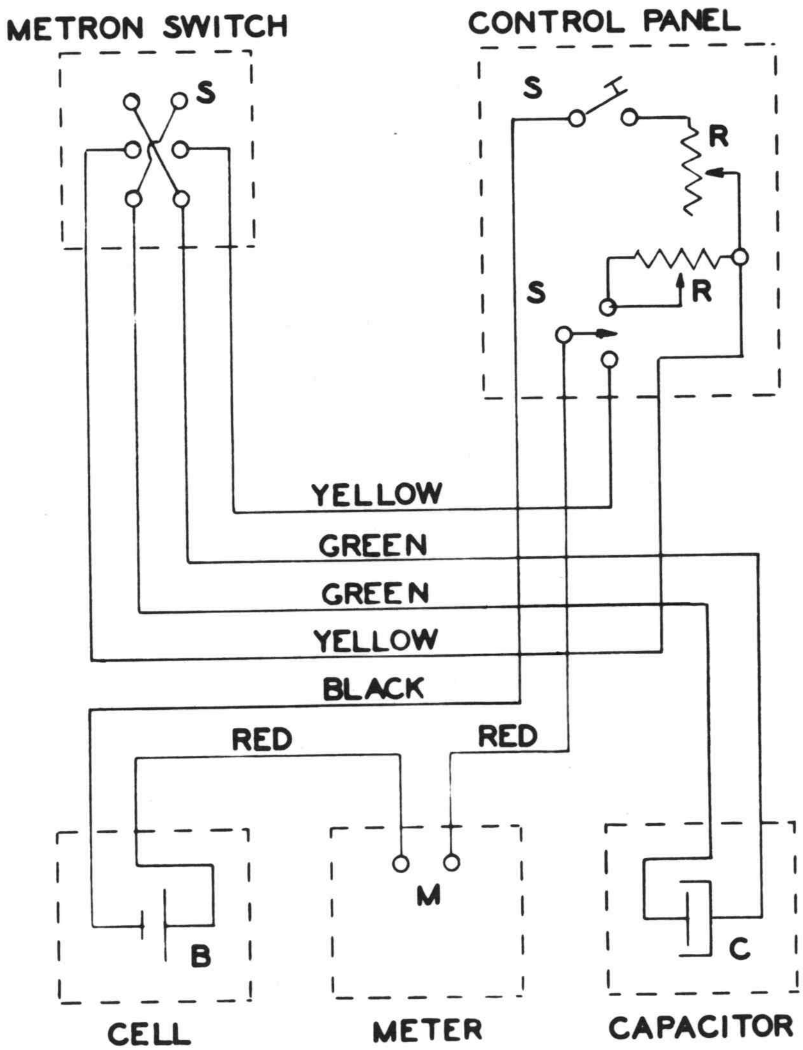
C FOUR MALLORY UB 353 CAPACITORS IN PARALLEL. TOTAL CAPACITANCE 16 MFD.

R_1 IRC W100 CONTROL.

R_2 IRC Q11-103 CONTROL.

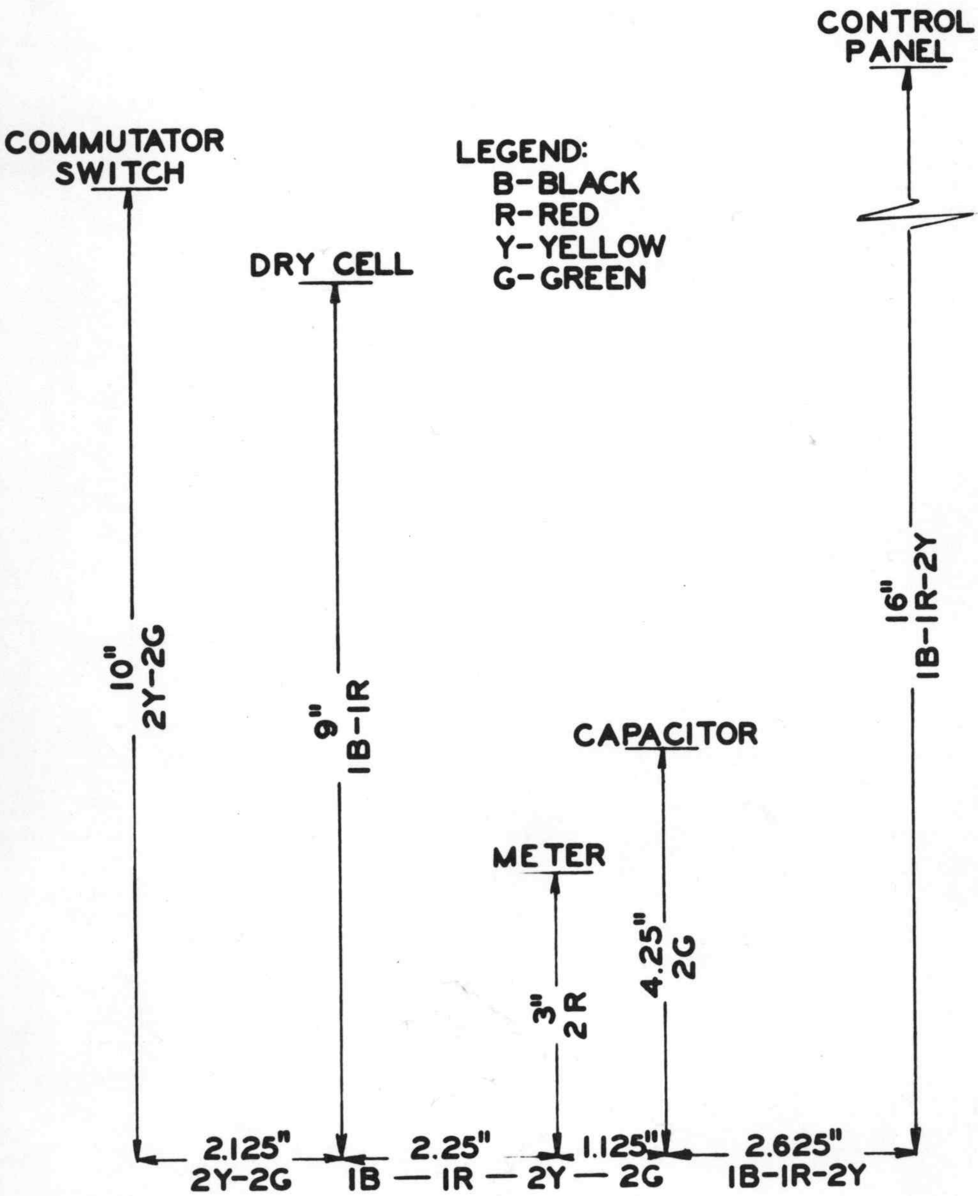
WIRING, SCHEMATIC DIAGRAM

DWG NO. E-1-6a



WIRING, HARNESS DIAGRAM - DWG NO. E-1-6b

SCALE: 1/2"=1"
MATERIAL: 18 GA COPPER WIRE
FINISH: AS INDICATED



NOTE:
HARNESS TO BE MADE UP WITH NO. 10 WAXED HARD LINEN THREADS. FABRIC OR PLASTIC SPAGHETTI COVERS TO BE PLACED AT ALL BENDS. DIMENSIONS GIVEN ALLOW FOR PIG-TAIL ASSEMBLY.

HARNESS BOARD DETAIL DWG NO. E-1-6c