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DESIGN AND DEVELOPMENT REPORT ON TREAT CONTROL ROD DRIVE II

by

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DISCLAIMER

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DESIGN AND DEVELOPMENT REPORT ON TREAT CONTROL ROD DRIVE II

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I. INTRODUCTION

The control rod drives initially designed for the TREAT reactor were the product of an early development program. Though the drives lacked some of the design requirements necessary in the TREAT test program, they were temporarily installed to permit preliminary reactor operation while a final drive was being designed and tested. The description and operating characteristics of these initial drives are discussed in the TREAT summary report, ANL-6034.*

This report is a documentation of the development of TREAT control rod drive II. More specifically, it describes the basic design, the problems involved with this design, and the various methods pursued in arriving at the final drive. The development and test program was conducted in the Reactor Engineering Division. Fabrication of the prototype and final drives was, for the most part, performed in the Laboratory shops.

The new control rod drives were installed in the TREAT reactor in April, 1960, and have been performing satisfactorily since that date.

II. DESIGN REQUIREMENTS AND BASIC DRIVE DESCRIPTION

A. Control Rod Drive Requirements

One of the purposes of the TREAT reactor is to evaluate reactor fuels and materials under conditions simulating various types of nuclear excursions. This program requires a great degree of versatility in the control rod drive system, since the nuclear excursions are initiated by the rapid insertion of transient rods or rapid removal of control rods. To be adaptable, the control rod drives were required to meet the following specifications:

1. Drives are to be installed below the reactor.
2. Control rods are to be actuated upward out of the core to provide for a downward insertion or scram.

*G. A. Freund et al., Design Summary Report on the Transient Reactor Test Facility TREAT, ANL-6034 (June 1960).

3. Rods must scram from any raised position and must be provided with an adjustable means of regulating scram speed down to the minimum drop time of 100 msec for the central 122 cm of stroke.
4. Rod travel is to be 153 cm (60 in.).
5. Two rod actuating speeds, up or down, are to be available. These are to be approximately 50 cm/min or 17 cm/min.
6. Remote actuation and positioning are required.
7. Operation is to be fail-safe, i.e., reactivity is to be removed on loss of power.
8. Each drive is to actuate two control rods on either 28.8-cm or 22.7-cm centers.
9. Provision is to be made on all drives for assembly of a transient safety latch which would prevent control rod scram in the event of power failure. The latch would be assembled to those drives which actuate transient rods.
10. The transient safety latches must be remotely operated and there must be visual indication of latch position.
11. The maximum overall height of the drive is to be less than 4.0 meters.
12. Drives are to be adaptable to either selsyn or potentiometer-type position indicators. Visual indication of rod fully up or down is to be provided.
13. Complete interchangeability of drives is required.
14. The maximum cross-sectional dimensions of drive are to be determined by established control rod placement in core.
15. The maximum current requirement for control latches and actuating switches is to be less than five amperes.

B. Design Procedure

The approach to a suitable design was mainly influenced by the condition that a variable scram speed was necessary and, further, that the fastest scram should approach a minimum time of 100 msec for the central 122 cm of travel. To achieve the flexibility of producing readily adjustable forces at arbitrary rod positions, air control appeared to be the most convenient and practical method.

On the basis of using an air cylinder for rapid insertion of the rods, commercial cylinder manufacturers were contacted in an effort to obtain essentially standard equipment for the application. The use of standard equipment provided the advantages of reduced initial equipment costs plus the immediate availability of parts for replacement. A pneumatic cylinder and hydraulic shock absorber combination was proposed by one manufacturer, based on their design and construction. It was around this unit that the remainder of the drive was designed.

C. Description of Basic Drive

The drive, as initially designed, is shown in Fig. 1. It is basically a pneumatic-accelerated fast scram cylinder combined with a motor-driven lead screw mechanism for re-latching and intermediate positioning. A magnetic latch holds and releases the control rods as required. The drive is designed to operate at a maximum cylinder loading of 34 atm.

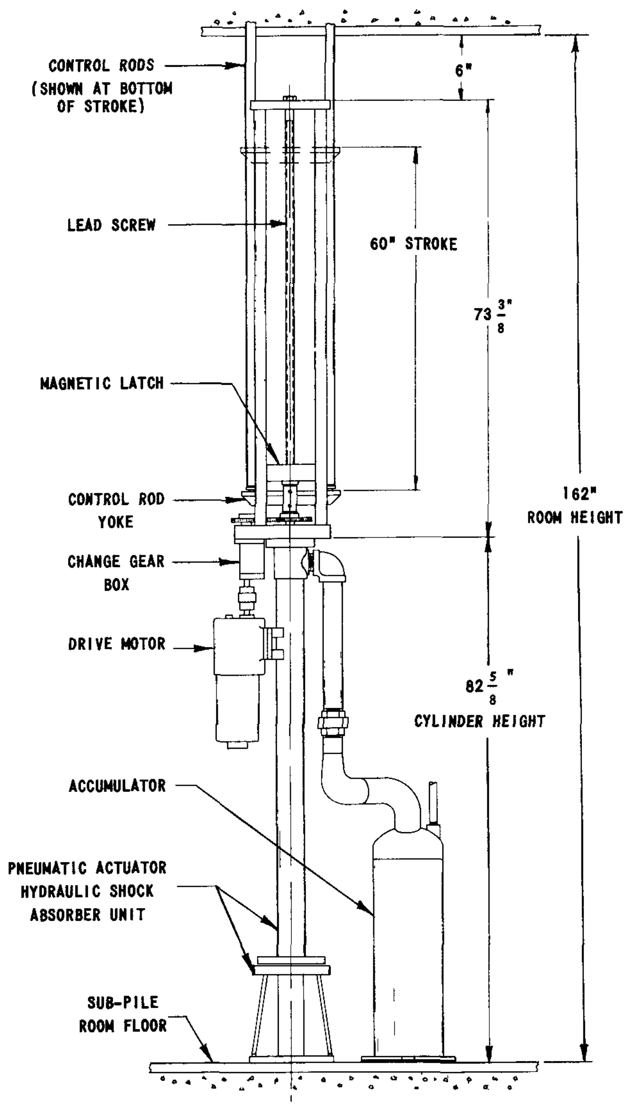

FIG. 1
BASIC CONTROL ROD DRIVE DESIGN

1. Pneumatic Actuator - Hydraulic Shock Absorber

The actuator-shock absorber unit is shown in Fig. 2. The unit consists of an open-bottom air cylinder mechanically fastened to a hydraulic shock absorber. The rod end of the cylinder is connected through manifolds to a system accumulator which maintains the necessary firing pressure in the cylinder regardless of rod position. The piston rod is internally threaded for attachment to the magnet armature and control rod yoke. Dynamic seals prevent loss of cylinder pressure with rod movement.

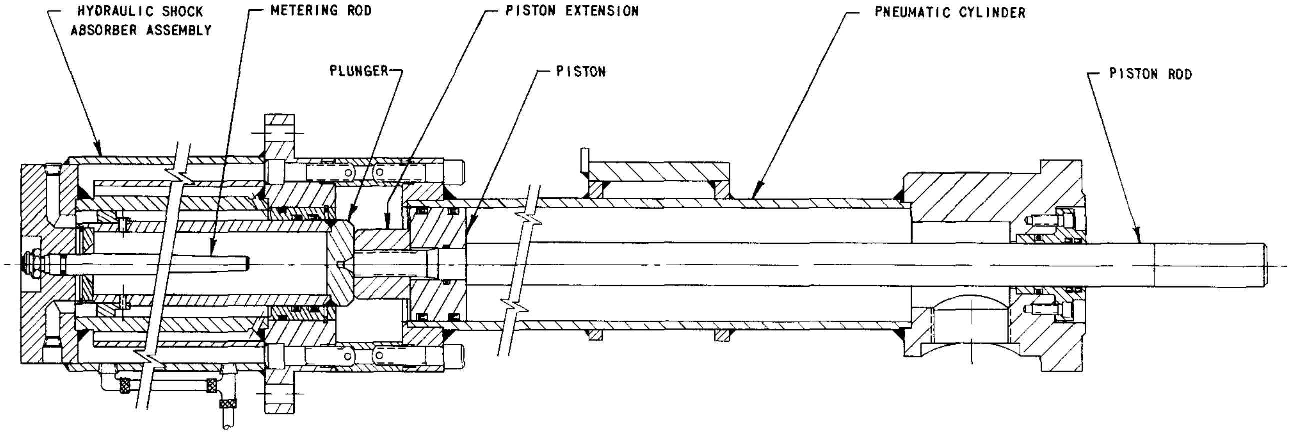

FIG. 2
PNEUMATIC CYLINDER WITH HYDRAULIC SHOCK ABSORBER

The hydraulic shock absorber is a constant-pressure-type absorber which operates on a variable fluid-discharge principle. The plunger, with a fixed orifice at the bottom, is passed over a tapered metering rod such that the flow area for the fluid being displaced by the plunger is continually decreasing. When the plunger is fully depressed, the flow area is essentially zero. Oil flowing through the varying orifice dissipates energy in a linear fashion. The displaced dashpot oil is stored in a reservoir around the pressure chamber. The reservoir is kept pressurized to return the plunger to its extended position when the cylinder piston is raised.

The shock absorber is fastened to a rigid stand which, when bolted to the floor, provides support for the drive assembly.

2. Drive Mechanism

Movement and positioning of the control rods are accomplished by a pair of Acme lead screws which are driven through a set of spur gears by a $\frac{1}{2}$-hp, vertically mounted gear head motor. Rotation of the lead screws imparts motion to the magnet assembly through a pair of bronze nuts assembled into the magnet plate. The normal rate of control rod travel is 50.1 cm/min; however, a set of change gears is provided, which can be readily assembled to provide a positioning speed of 17 cm/min. The lead screws operate through guide bushings in the lower tie plate and are supported by thrust bearings in the upper support plate.

3. Magnetic Latch

A magnetic latch (see Fig. 3) was chosen for control rod release because of its simplicity, reproducibility, and freedom from mechanical malfunction.

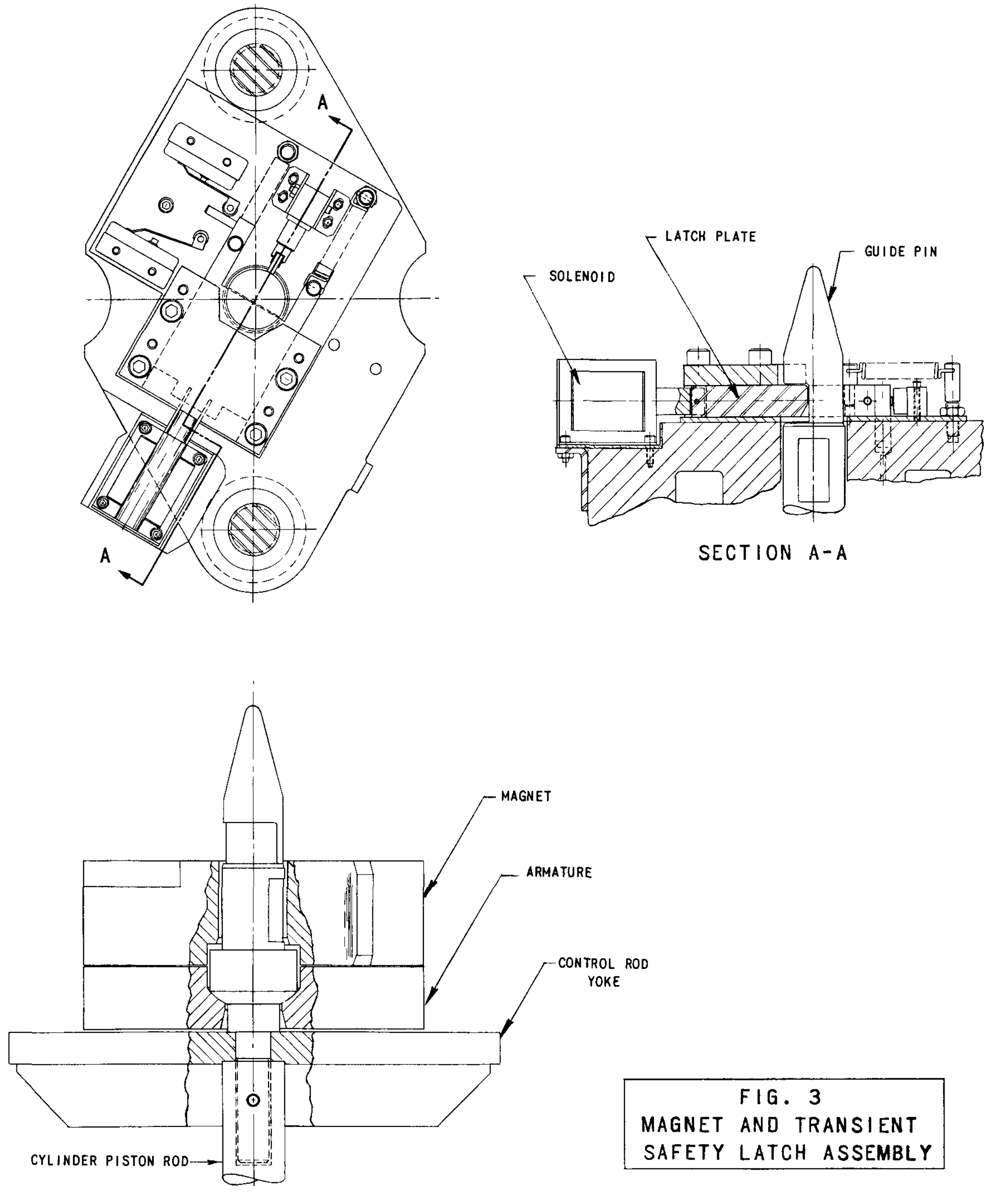

FIG. 3
MAGNET AND TRANSIENT
SAFETY LATCH ASSEMBLY

The function of the latch is to provide fast disassembly of the control rods from the driving mechanism to obtain rapid insertion of poison into the reactor core. The latch consists of an armature plate and a magnet plate assembly which contains the magnet coil. The yoke, which supports the two control rods, is fastened to the armature plate by means of the guide pin. The guide pin is threaded into the piston rod of the air cylinder so that downward motion of the poison rod must be accompanied by downward motion of the control rods. This arrangement provides for accurate positioning, as well as offering an axially applied latching force to eliminate eccentric bending forces on the control rods.

To re-latch the magnet after a scram, the magnet plate is driven down to contact the armature plate by rotation of the lead screws. Actual contact between magnet and armature is prevented by nonmagnetic stainless steel stops which maintain a 0.025-cm gap between the two components. The gap is provided to accelerate the release of the armature once the magnet coil current is interrupted. With the magnet plate in the contact position, energizing of the magnet coil magnetically couples the armature to the magnet plate. When the energized magnet is raised, the armature, control rod yoke, control rods, and piston rod follow accordingly. Interruption of the coil current allows the pressurized piston rod to rapidly retract the rods.

4. Transient Safety Latch

In addition to conventional control rods, the TREAT reactor is provided with transient rods, which differ from the control rods in that the neutron-absorbing poison is positioned to enter the core when the rod is driven upward. When the rods are scrammed, poison is removed from the core. With this condition, insertion of transient rods into the core unexpectedly due to magnet or power failure could be very hazardous. To prevent such uncontrolled insertions of reactivity, a special latch is provided on all transient drives to secure the transient rods until rod insertion is so desired.

The transient safety latch, shown in Fig. 3, is a detachable assembly mounted to the top of the magnet plate. The latch basically consists of a solenoid, latch plate, and spring return. When the magnet plate and armature are in the latched position, the guide pin, which is fastened to the cylinder piston rod, extends through a hole above the top face of the magnet plate. De-energizing the latch solenoid allows the latch plate springs to slide the latch plate into engagement with a machined slot on the guide pin. With the latch plate in position, the guide pin cannot be withdrawn should the magnet circuit open. Since the yoke and transient rods are contained by the guide pin, downward motion is constrained. To de-latch the assembly, the solenoid is energized to withdraw the latch plate from the guide pin slot.

5. Control Rod Knuckle

The control rods are connected to the control rod yoke through self-aligning knuckle joint assemblies, as shown in Fig. 4. Each knuckle joint assembly consists of a monoball bearing, a set of thrust washers, and a series of Belleville springs. The monoball bearing is allowed lateral movement which, in addition to its spherical action, insures axial alignment of the control rod.The Belleville washers are installed to absorb axial shocks in both directions.

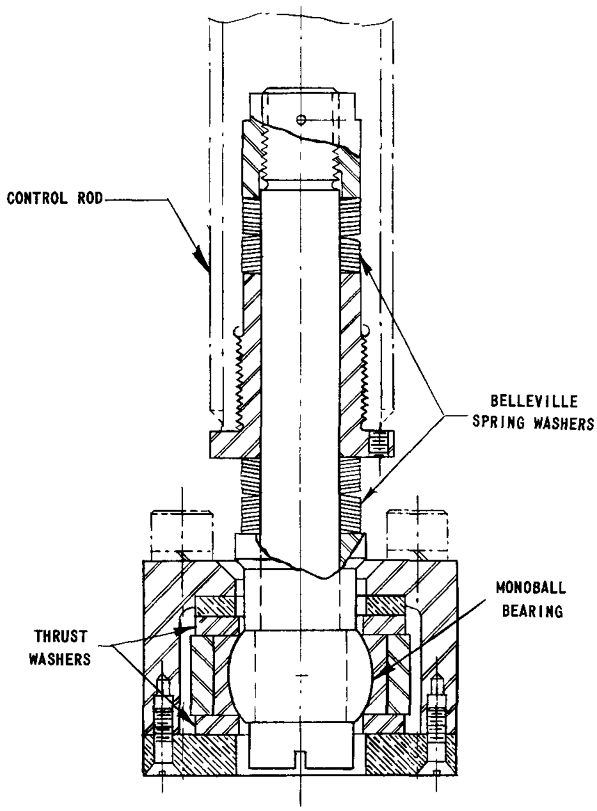

FIG. 4
CONTROL ROD KNUCKLE

The knuckle housings are bolted to each end of the control rod yoke. Since two possible control rod spacings are required, each yoke is provided with two sets of mounting holes.

6. Control Rod Position Indicators

Control rod drives No. 1 and No. 2 are used for fine reactivity control and accordingly are equipped with selsyn position indicators. The remaining drives are equipped with less accurate potentiometer-type indicators.

Each position indicator is mounted in a compact subassembly to the lower tie plate of the drive and is driven directly from the lead screw. Appropriate precision gears in each position-indicator subassembly convert the lead screw rotation into desired indicator rotation.

III. TEST AND EVALUATION OF BASIC DRIVE

A. Test Procedure

A prototype of the basic drive was fabricated and assembled in the test laboratory. Two mild steel bars, each 4.45 cm in diameter and weighing 31.8 kg, were threaded at one end to act as control rods. A steel structure was erected above the drive, and bearing plates containing graphitar guide bushings were positioned at appropriate heights to support the dummy rods. The drive and dummy rods were mounted to simulate actual reactor mounting conditions.

The first phase of the test was concerned primarily with the general mechanical operation of the drive. A nitrogen gas bottle was connected to the rod end of the pneumatic cylinder to provide the actuating force necessary for fast scrams. The cylinder input pressure was controlled by a regulator at the nitrogen bottle, and a bourdon tube pressure gauge gave constant indication of cylinder pressure. A solenoid-operated, four-way valve installed between the gas bottle and the cylinder permitted exhausting of the cylinder pressure as required.

The magnet power supply was assembled and mounted in a control panel near the drive. A variac was placed in the rectifier input circuit to provide a means of varying the magnet current so that the holding characteristics of the magnet could be studied. A toggle switch installed in the rectifier output circuit between the magnet coil and the rectifier served as the rod scram switch.

The first procedure was to cycle the drive through its required travel by means of the drive motor so as to visually check the overall operation and to set all limit and indicating switches properly. When all adjustments were completed, the drive was operated through a series of rod drops, starting with no gas pressure in the cylinder and increasing in 3.4-atm increments to a maximum loading of 13.6 atm. To provide a visual record of rod motion and dashpot reaction, a series of high-speed motion pictures were taken at each pressure interval.

The holding characteristics of the magnet were determined by an observation of the minimum currents necessary to support the control rod assembly at given pressure loadings. The procedure was to lift an applied load magnetically to a given height and then, by means of the variac, slowly decrease the rectifier input current until the armature fell away from the magnet. The rectifier current at each breakaway was recorded by a circuit ammeter. Current readings were taken at 3.4-atm intervals up to the maximum loading of 13.6 atm.

The transient safety latch was operated through a series of latchings and de-latchings and through a cycle of simulated magnet failures at similar pressure loadings as used in the previous tests. Latch performance and solenoid current requirements were carefully noted at each loading.

The test was discontinued at this point to review and evaluate the initial results.

B. Evaluation of Test Results

The basic drive, though designed to operate at a cylinder pressure of 34 atm, was not tested beyond 13.6 atm because it became apparent early in the rod drop tests that the segregated design of the actuator-shock absorber unit (see Fig. 2) was not acceptable. The effect produced by the piston rod, moving at high speed, striking the stationary steel plunger to initiate cushioning action was not desirable. The noise at impact was intense, approaching the deafening level at the higher scram pressures. In addition to the noise, the dashpot did not function as the supplier had specified. The dashpot was to have dissipated the kinetic energy of the piston rod at a linear rate over a 33-cm effective length. The high-speed movies indicated that the piston rod came to an abrupt stop within 3 cm of travel, after which it slowly settled the remaining 30 cm. The excessive impact forces produced by the sudden stoppage were evidenced by stress cracks in the threaded portion of the piston rod and by the deformation of the control rod yoke. The dashpot plunger, however, at this point showed no evidence of damage from the impact forces.

The current requirements for holding by the magnet were determined to a 13.6-atm loading. Extrapolation of the data obtained gave indication that a current of approximately 5 amp would be necessary to lift the rods with 34 atm of gas applied to the cylinder piston.

Examination of the transient safety latch showed no indication of failure or wear. Throughout the limited test, the latch had performed to satisfaction without hesitation. There was, however, an undesirable assembly problem which demanded critical adjustment of the guide pin. It was necessary to locate the guide pin very accurately to insure proper latch plate engagement and to provide pin engagement with a minimum amount of axial clearance. An excessive clearance would result in large impact forces which could fracture the guide pin should the magnet power ever fail. If, however, too little clearance was provided, the friction of the pin against the latch plate was great enough to cause an excessive starting current through the solenoid for latch plate withdrawal.

The remainder of the drive was disassembled and visually examined for deformation or wear. The only component to show indication of excessive use was the monoball bearing in the knuckle joint assembly. The bearing, when initially assembled, had contained no lateral clearance, movement of the ball being restricted to rotational movement only. After the test, a very slight amount of axial ball movement was possible. Close measurement of the bearing indicated that the lower portion of the ball race had increased slightly in diameter. The ball race was fabricated from an unhardened alloy steel, and flow had taken place under the excessive impact loads caused by the improperly functioning dashpot.

IV. BASIC DRIVE MODIFICATION

A. Actuating and Shock Absorbing Cylinder

The major design change required with the drive was basically concerned with the method of decelerating the rods. Originally, it had been planned to use commercially available components for the actuating and shock-absorbing cylinder, but it was apparent from the test that, to meet our requirements, a special unit would have to be designed.

Before finalizing on a design, manufacturers of pneumatic cylinders were again contacted to find those interested in fabricating a unit of our design. Only one company expressed interest in the job, so the cylinder was designed around their type of construction, using as many standard parts as possible. The redesigned unit is shown in Fig. 5. The hydraulic dashpot on this unit is rigidly fastened to the bottom of the actuating cylinder. The inside of the dashpot housing is lined with a tapered bronze sleeve which decreases in internal diameter from top to bottom. The dashpot plunger, which is an integral part of the piston rod, is a concentric steel member with a bullet-shaped nose. When the control rod is released, acceleration continues until the dashpot plunger enters the oil seal at the entrance to the dashpot chamber. As the plunger moves through the dashpot, the displaced oil from underneath the plunger must pass through a continually decreasing annular area between the plunger and sleeve. The degree of sleeve taper is designed to produce a constant dashpot pressure at maximum loading and, consequently, a constant deceleration force. The radial clearance between the plunger and sleeve varies from 0.08 cm to 0.007 cm through the dashpot length of 15 cm. The dashpot oil displaced by entry of the plunger is stored in a surge reservoir connected to the top of the dashpot housing. Gravity and expansion of reservoir air return the dashpot oil to the chamber when the plunger is withdrawn.

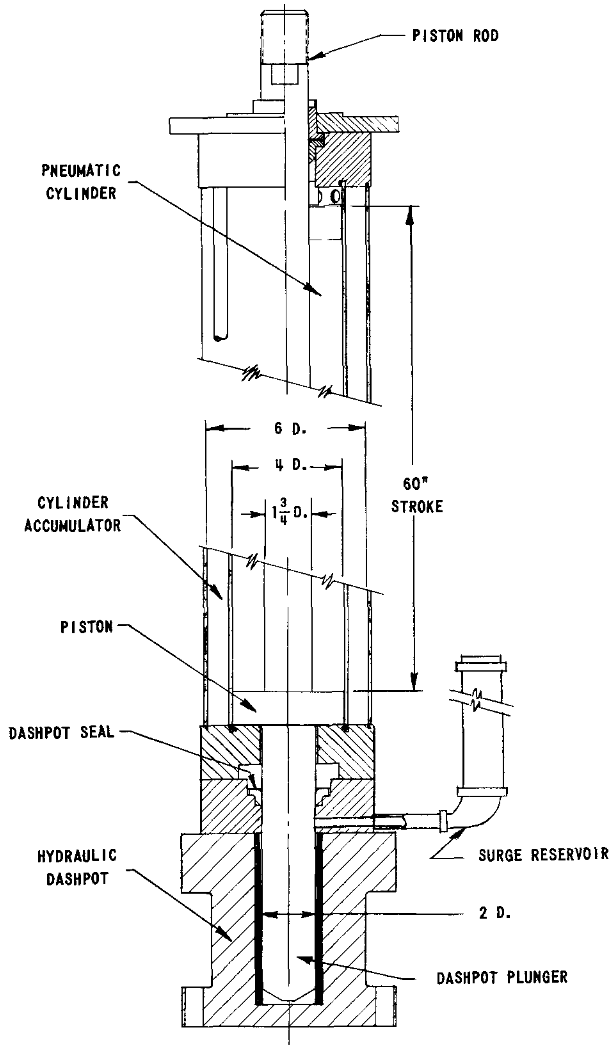

FIG. 5
ACTUATING AND SHOCK ABSORBING CYLINDER

To eliminate the need for external accumulators and large supply manifolds as required on the initial cylinder, a concentric tube was placed around the actuating cylinder to act as an accumulator. The self-contained accumulator stores a volume of air in excess of the volume of air displaced by the working piston. The accumulator is connected to the actuating cylinder by a series of large orifices at the rod end of the cylinder. Air is supplied to the cylinders through a small line from system compressors. Compressors were installed in the pressurization circuit to eliminate the need for the initially used nitrogen bottles. (See TREAT summary report, ANL-6034).

B. Magnet

The holding-current requirements of the initially proposed magnet indicated that a stronger magnet would be more versatile and desirable. To provide a stronger magnet, it was necessary to change to a rectangular coil design, since the largest possible circular coil that could be assembled between the control rods had already been used. The new magnet (see Fig. 6) was designed to lift 3200 kg with a current flow of 3 amp; a 0.025-cm air gap existed between the magnet plate and armature.

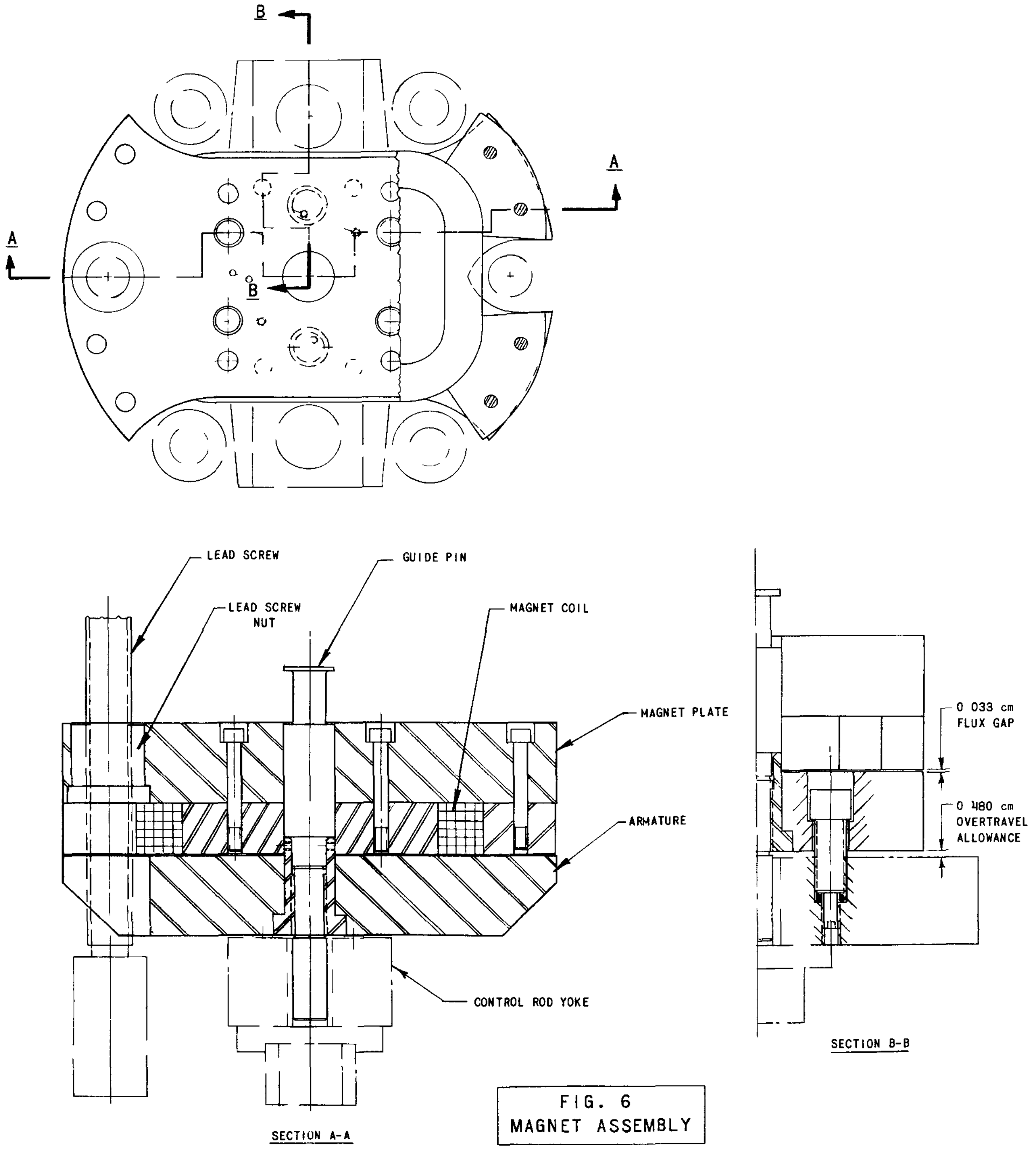

FIG. 6
MAGNET ASSEMBLY

The magnet coil, consisting of 750 turns of No. 20 magnet wire, was bonded to the inner pole piece with epoxy resin. The pole pieces were held to the magnet plate by nonmagnetic stainless steel cap screws, which also served to produce the 0.025-cm air gap. Nonmagnetic screws were used to prevent a direct magnetic path between magnet plate and armature.

A prototype magnet was fabricated and tested to determine the lifting current requirements, the time delay before armature release, and the evenness of armature release. A Sanborn two-pen recorder was used to chart the time interval between the interruption of magnet current and the actual beginning of armature movement. In operation, the actuation of the scram switch closed the Sanborn circuit to produce a pen deflection. A sensitive, precision snapswitch in contact with the control rod yoke interrupted the Sanborn circuit when downward motion began. Delay-time data were recorded at pressure intervals of 6.8 atm up to the maximum loading of 34 atm. At each pressure interval, data were obtained for more than one magnet-holding current. The results are tabulated in Table I. The minimum holding current required at 34 atm was found to be 1.1 amp.

Table I

DELAY TIME OF MODIFIED MAGNET

Pressure, atm	Magnet Current, amp	Delay Time, msec
Free Fall	0.1	300
	1.5	1542
6.8	0.4	57
	1.1	350
	1.5	422
13.6	0.6	36
	1.1	147
	1.5	190
20.4	0.8	27
	1.1	72
	1.5	89
27.2	0.95	18
	1.0	23
	1.5	42
34	1.2	16
	1.5	20

High-speed photography was used to observe the armature breakaway from the magnet. The films indicated that the armature did not leave the magnet face evenly, but had a tendency to hold on to one side. Since this type of release is typical of magnets with two magnetic surfaces in contact, the initially nonmagnetic screws were checked and found to have become magnetic from exposure to the magnet flux. The magnetic screws were removed and a series of tests were made to determine the best insulating material and thickness that could readily be adapted to the magnet to produce a minimum delay time without producing an excessive increase in magnet current. Thin sheets of varying thicknesses of copper, brass, and nonmagnetic stainless steel were cut to the shape of the armature and used as separators between the magnet and armature. Some sheets were completely solid, whereas others were perforated in an effort to compare the effect of each. Results of the tests are shown in Table II.

The results indicated that a 0.033-cm perforated brass shim had yielded the shortest delay times. Since the perforated sheet was preferable to a solid sheet, a third sheet was made in the form of a 1.27-cm-wide strip of brass that fitted the outer periphery of the armature but left the center completely open. Delay times produced with this hollow sheet were comparable with the times produced by the solid sheet, but were not as short as those from the perforated sheet. The test result showing that the perforated sheet gave shorter delay times than the almost complete air gap was not expected. The laminated effect of the gap available when using the perforated sheet obviously dissipated the flux field or residual magnetism at a faster rate than either the solid insulator or essentially solid air gap.

Based on the above findings, a brazing torch was used to deposit 0.63-cm rows of brass on the armature plate in a checker-board pattern. The brass surface was then ground parallel to the armature face, leaving a 0.033-cm-high cross-hatched pattern to provide the flux gap. The magnet plate and pole segments were fastened together with short carbon steel screws instead of the longer nonmagnetic screws initially used.

C. Transient Safety Latch

The operation of the transient safety latch had indicated that, since the available current was limited, the elimination of a solenoid-powered actuating force would provide a more reliable and versatile latch. Since adequate air was available at the drive, an air-cylinder-powered latch was designed. The new latch was designed automatically to take up any axial clearance between the latch and guide pin to eliminate the need for critical pin adjustment.

Table II

MAGNET DELAY TIMES WITH BRASS COPPER AND STAINLESS STEEL SEPARATORS

Pressure atm	Material	Sheet Type	Sheet Thickness cm	Magnet Current amp	Average Delay Time msec
Free Fall	Brass	Solid	0 025	0 2	195
Free Fall	Brass	Holes	0 025	0 2	107
Free Fall	Copper	Solid	0 025	0 2	425
Free Fall	Brass	Solid	0 025	1 5	905
Free Fall	Copper	Solid	0 025	1 5	1240
Free Fall	Brass	Solid	0 033	0 2	73
Free Fall	Brass	Solid	0 036	0 2	120
6 8	Brass	Solid	0 025	0 5	65
6 8	Brass	Holes	0 025	0 5	40
6 8	Copper	Solid	0 025	0 5	107
6 8	Copper	Solid	0 025	1 1	320
6 8	Brass	Solid	0 025	1 1	255
6 8	Brass	Solid	0 036	0 6	34
6 8	Brass	Solid	0 036	1 0	88
6 8	Brass	Holes	0 036	1 0	54
13 6	Brass	Solid	0 025	0 7	37
13 6	Brass	Holes	0 025	0 7	30
13 6	Copper	Solid	0 025	0 7	39
13 6	Copper	Solid	0 025	1 1	119
13 6	Brass	Solid	0 025	1 1	106
13 6	Brass	Holes	0 025	1 1	94
13 6	Brass	Solid	0 036	1 0	29
13 6	Brass	Holes	0 036	1 0	17
20 4	Brass	Solid	0 025	0 8	27
20 4	Copper	Solid	0 025	0 8	30
20 4	Copper	Solid	0 025	1 5	84
20 4	Brass	Solid	0 025	1 5	57
20 4	Brass	Solid	0 033	1 0	18
20 4	Brass	Holes	0 033	1 0	18
20 4	Stainless Steel	Solid	0 033	1 0	29
27 2	Brass	Solid	0 025	1 0	10
27 2	Copper	Solid	0 025	1 0	21
27 2	Copper	Solid	0 025	1 5	52
27 2	Brass	Solid	0 025	1 5	29
27 2	Brass	Solid	0 033	1 5	15
27 2	Brass	Holes	0 033	1 5	15
27 2	Stainless Steel	Solid	0 033	1 5	30
34 0	Brass	Solid	0 025	1 2	10
34 0	Copper	Solid	0 025	1 2	14
34 0	Copper	Solid	0 025	1 5	23
34 0	Brass	Solid	0 025	1 5	15
34 0	Brass	Solid	0 033	1 6	12
34 0	Brass	Holes	0 033	1 6	12
34 0	Stainless Steel	Solid	0 033	1 6	16

The latch assembly schematically illustrated in Fig. 7 is shown in the de-latch position. The transient safety latch is assembled to the top of the magnet plate by means of mounting lugs which locate the latch slide mechanism over the guide pin hole. When the magnet is driven down to engage the armature, the guide pin moves up through the magnet plate and latch slide mechanism to contact the latch cam pin. Further motion of the guide pin causes the cam pin and, subsequently, the slide stop cam, which is integral with the cam pin, to pivot about the cam pin shaft. When the magnet is in the latch position, the stop cam has rotated past its crest, allowing the compression spring to drive the slide mechanism forward until the latch block engages the undercut portion of the guide pin. After the latch block has engaged the undercut portion of the guide pin, the slide mechanism continues its forward motion and results in the raising of the latch block, which is supported on a pair of inclined ways, until it contacts the head of the guide pin. In this position, all clearance between guide pin and latch has been removed. With the guide pin latched, the transient rods cannot be scrammed by loss of power.

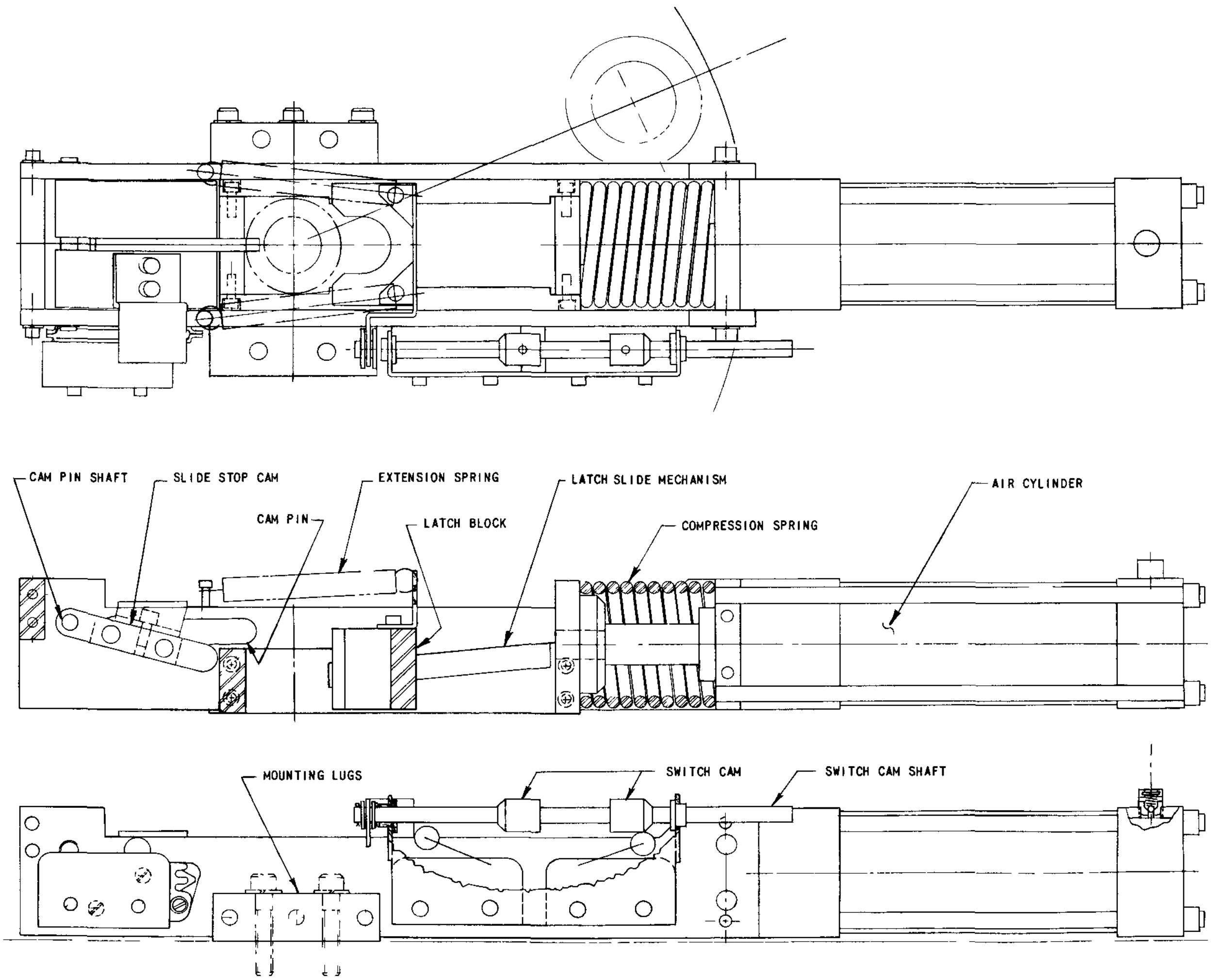

FIG. 7
TRANSIENT SAFETY LATCH ASSEMBLY

To unlatch the assembly, a solenoid which actuates a small air valve is energized to allow air to flow into the head end of the air cylinder. The air pressure overcomes the force of the compression spring and retracts the slide mechanism to its initial position. The latch block extension springs return the latch block to its position at the bottom of the inclined ways. When the control rods are scrammed, the guide pin vacates the slide mechanism, allowing the cam pin and stop cam to fall into the lock position. When the solenoid is de-energized, exhausting the air from the head end of the cylinder, the slide is retained in its "cocked" position by the cam stop. In this position, the latch block is again ready to engage the guide pin.

A cam shaft operated by the latch block actuates two snap switches to give remote indication of position of the latch block. A third snap switch operated by the cam pin gives remote indication that the armature is in position to be magnetically latched to the lead screw drive.

D. Control Rod Knuckle

A hardened alloy steel, cup-type, race retainer was made to enclose the monoball to eliminate the possibility of elongation of the ball race. A pair of hardened retainer monoball assemblies were subjected to impact loads for a 1500-cycle test period at maximum load conditions. No deformation of any type was detectable, and movement of the monoball was as initially installed.

E. Guide Pin

The method of guide pin assembly was changed to insure a more fail-safe design. Initially, the guide pin was threaded into the cylinder piston rod and passed through clearance holes in the magnet plate, armature, and yoke. With this design, the guide pin carried the piston load at all times so that, even with the magnet coupled, pin failure could drop the rods.

In the re-design (see Fig. 6), the armature was fitted with a threaded bushing to contain and locate the guide pin. The yoke was re-bored and fitted with a bronze guide bushing so that the guide pin, though threaded into the armature, would be guided by the yoke assembly. The pin was guided in the yoke to stabilize the armature in event of failure of one of the springs supporting the armature.

Two large, socket-head cap screws were provided to hold the yoke assembly to the armature. These cap screws are threaded into the yoke and retained with lock washers and set screws to prevent movement. The screws allow the armature to float axially on the compression springs when in the unlatched position.

V. TEST AND PERFORMANCE CHARACTERISTICS OF THE FINAL DRIVE

A. Description of Final Drive

The original prototype was modified as discussed in the previous section and completely assembled for final test. The fully assembled drive is shown in Fig. 8.

The operation of the final drive is essentially the same as that of the prototype with the exception of that of the transient safety latch, which has already been discussed. The drive is supported at the base by a series of floor bolts extending through the flange of the dashpot housing. Oil level in the dashpot is indicated by the stand pipe and shutoff valve assembly at the base of the housing. With the dashpot plunger removed from the dashpot chamber, the chamber oil level should be equal in height to the top of the stand pipe.

Air for cylinder and latch operation is supplied at the top of the drive and carried by pressure tubing to the head end of the actuating cylinder. Air for the transient safety latch cylinder is extended from this point by flexible high-pressure hose to a regulator which provides control of cylinder pressure. A spring-operated pulley guides and controls the moving flexible hose between regulator and transient cylinder.

Power and control cables are brought to the drive from overhead and coupled to the connector bracket at the top of the drive. Cables from this point are connected to the main terminal strip on the motor hanger plate. The magnet and transient safety latch circuits are supplied from the main terminal strip through flexible coiled cords capable of expanding 152 cm. Motor switches limit magnet travel in both up and down directions. Compression springs between the armature and yoke allow for motor overtravel in the latching position. Snap switches are positioned to give remote indication of the rods fully down, the magnet fully down, the magnet in latch position, and the transient safety latch in either latched or unlatched position.

B. Test Procedure

The final drive was installed in the test frame and assembled with dummy control rods as previously described. To determine rod drop speeds, a series of photoelectric cells were mounted to the support frame opposite one face of the control rod yoke. The cells were vertically spaced at definite intervals throughout the full yoke travel.

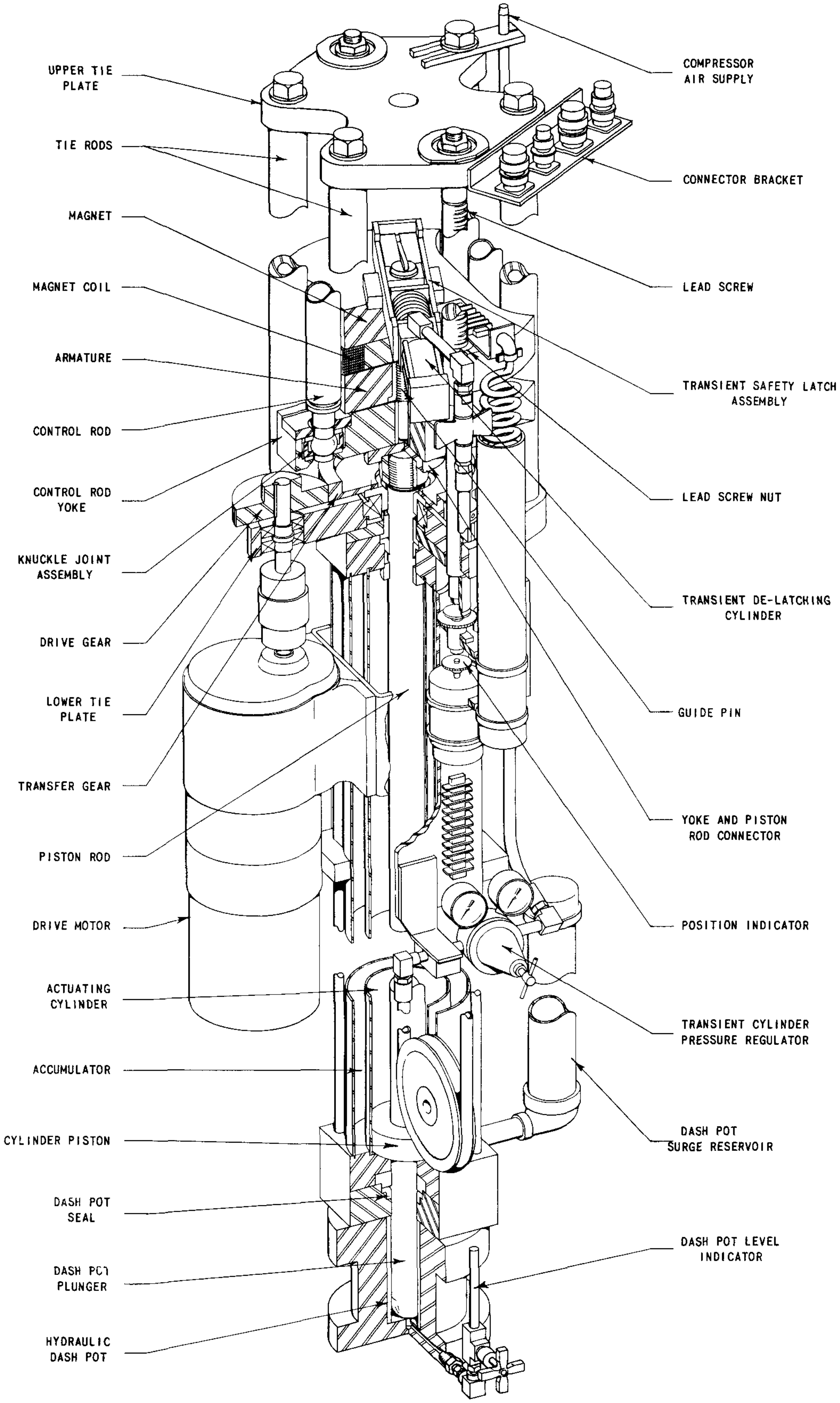

FIG. 8
TREAT CONTROL ROD DRIVE II

A small flashlight, with its beam centered on the vertical center line of the cells, was secured to the yoke. Movement of the yoke allowed the flashlight beam to fall consecutively on each cell. To eliminate excessive light reception, the cells were covered with tape to allow only a small slit at the center of each cell to be exposed. The cells, electrically wired in parallel, were connected with a direct-current power supply to the input of the Sanborn recorder. The passing of light over a cell completed the electrical circuit and produced a pen deflection on the recorder. Time intervals between successive cells were obtained by this method to establish true speed curves.

The magnet delay time was recorded by the second pen of the Sanborn recorder. As before, a sensitive snap switch was positioned to contact the yoke at its maximum height and open with any downward movement. The scram switch, when closed, gave a continual pen deflection until yoke motion caused the snap switch to open the pen-supply circuit. This produced a chart indicating the time required between scram switch actuation and actual control rod motion.

The series of tests began with free fall of the rods and were continued in 3.4-atm increments up to 34 atm. At each test pressure, the minimum magnet-holding current was determined by the gradual reduction of magnet current flow. Using a series of selected magnet currents, twenty-five rod drops were made at each test pressure, and the corresponding delay times and rod speeds were recorded.

Throughout the rod drop tests, high-speed cameras were used to provide visual indication of magnet and armature release and of dashpot action. A chart calibrated in inches was attached along the side of the drive to provide a length scale so that the time required per inch of dashpot travel could be determined from the film.

The transient safety latch was operated through a test cycle which included the intermittent interruption of magnet current to facilitate magnet failure. Latch action and response was closely observed throughout the period.

C. Test Conclusions and Drive Characteristics

Throughout the test program, the drive had performed as expected and all components had functioned as required. Examination of the individual parts after disassembly produced no indication of excessive wear or future failure of parts. The transient safety latch had operated smoothly and the function of the slide assembly to remove clearance between latch and guide pin was exemplified by the fact that the transferral of the load to the latch under simulated magnet failure was not visually noticeable.

When the magnet was re-energized, the latch was readily withdrawn by the air cylinder. The high-speed movies verified proper magnet release and good dashpot action.

By use of the high-speed film, the time interval for each consecutive inch of dashpot motion was obtained. The time was determined by dividing the number of frames of film required for the yoke to pass through each inch marking on the calibrated chart by the film speed at that particular time. The data obtained from the film is plotted in Fig. 9.

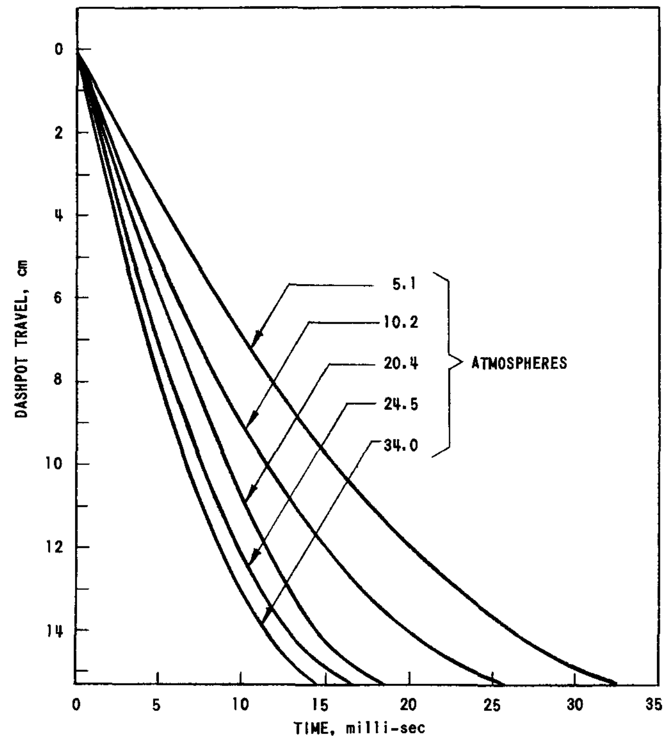

FIG. 9
TIME VS. TRAVEL THROUGH DASHPOT

The minimum magnet-holding currents observed in the tests are plotted in Fig. 10. The graph indicates that for any particular magnet, the holding current is a linear function of the cylinder pressure.

The Sanborn charts provided the data for the curves shown in Fig. 11. The curves represent the distance versus scram time characteristics of the control rods under various actuating pressures. The graph considers only the time involved from the beginning of control rod movement to the entry of the plunger into the dashpot.

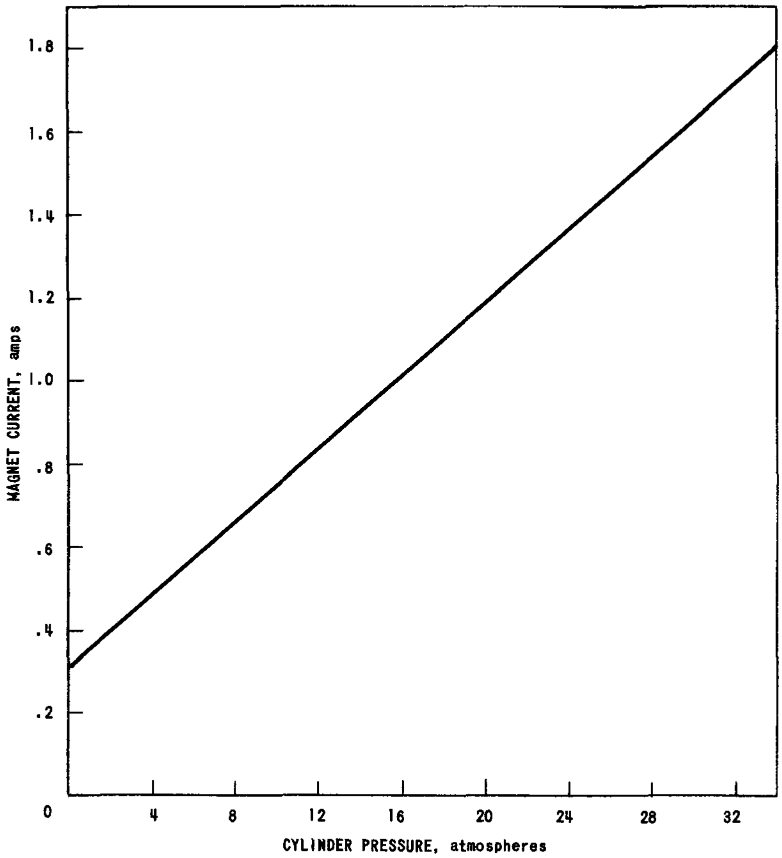

FIG. 10
MINIMUM MAGNET HOLDING CURRENT
VS.
CYLINDER PRESSURE

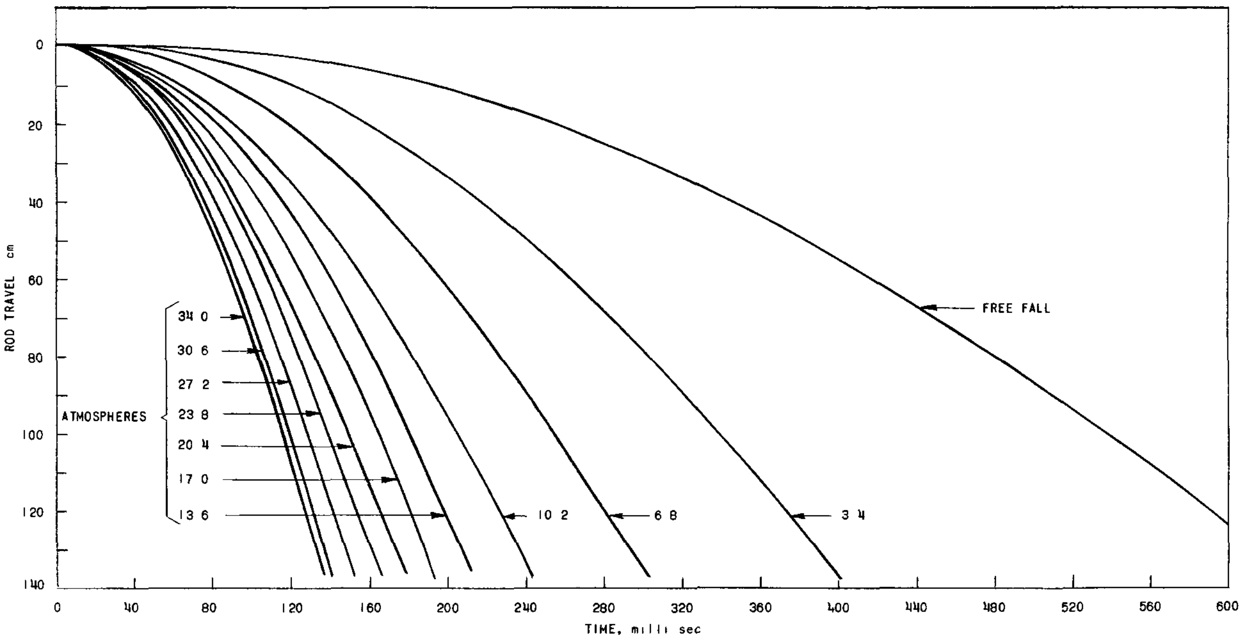

FIG. 11
ROD TRAVEL VS. TIME

A complete scram time curve is shown in Fig. 12. The curve represents the total time required from the actuation of the scram switch to the stopping of the control rod in the dashpot chamber. The plot was made from data recorded at the 20.4-atm test and is characteristic of the curves obtained at other experimental pressures.

The final drive fulfilled all the initial design requirements and performed reliably throughout the test program. The control rods were fully raised and scrammed approximately 500 times, which is estimated to be equivalent to a year of operation, without evidence of deformation or excessive wear. The versatility of the drive was apparent and the characteristic curves verified the selectiveness available in times for rod insertion.

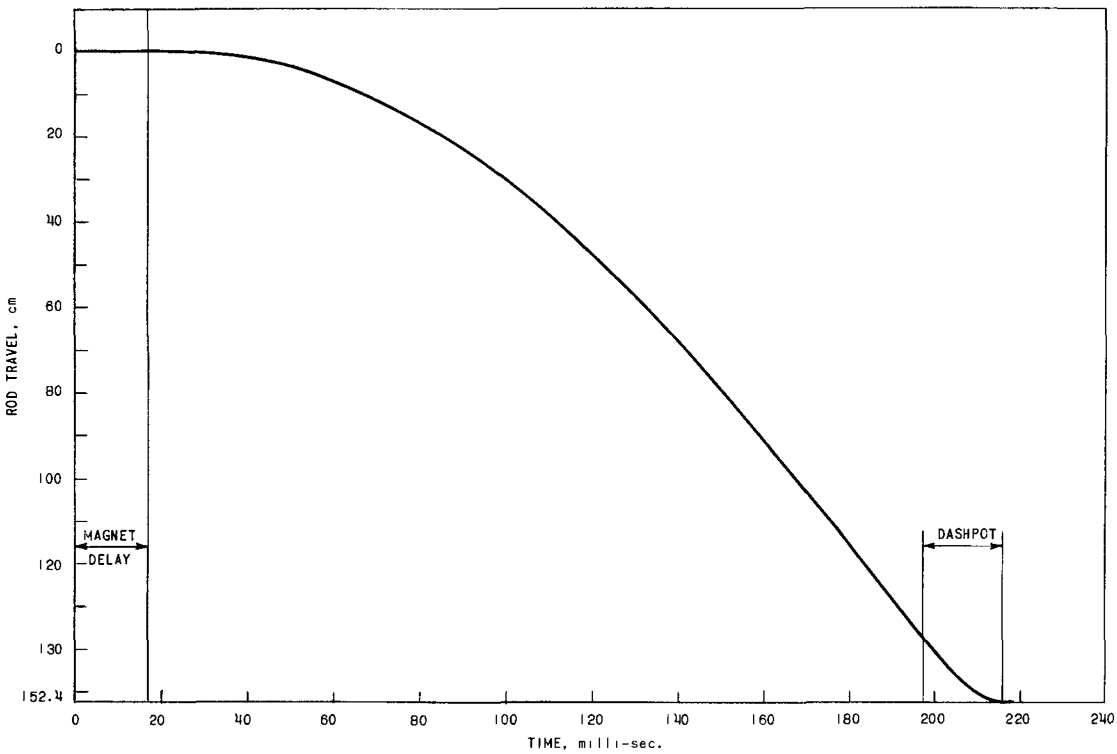

FIG. 12
CONTROL ROD SCRAM CURVE AT 20.4 ATMOSPHERES

Appendix A

AUXILIARY EQUIPMENT

A. Additional Drive Requirements

In addition to the drive requirements specified in Section II, it was desired to have available a means of producing and selecting scram times slower than free fall and extending upward to a 20-sec duration for full downward movement. It was also desirable that these longer scrams occur at a constant velocity throughout the full distance. It was required that any drive could be selected to produce these slow scrams. However, not more than one drive was to be slow scrammed at any one time.

B. Auxiliary Equipment Design

The design of a piece of equipment to be permanently attached to each drive for rod slow down was not thought to be practical for three reasons. For one reason, although it was desired to have slow scram available for each drive, it was a possibility that only the transient rods would ever be used in this fashion. Secondly, the space limitations around each drive were such that the addition of extra permanent equipment was very restrictive. Thirdly, since only one drive was used in slow scram at any one time, other available equipment was essentially surplus.

On this basis, a portable self-contained unit was proposed, which could be readily wheeled to any selected drive and rapidly connected to convert the drive to slow-scram operation. The actuating cylinder piston was chosen as the means of producing slow scrams. The method was to fill the cylinder volume underneath the piston with hydraulic oil and control the rod descent by adjusting the rate of oil removal.

The oil supply for the cylinder was provided by a 11.36-liter/min pump, capable of 27.2-atm operation, mounted on a 38-liter reservoir. Two 2.54-cm diameter high-pressure hoses were used to connect the actuating cylinder to the hydraulic pump assembly. The use of hose provided the line flexibility necessary for attachment to all drives. Fast-disconnect, self-sealing couplings were used for the cylinder connections. This type of coupling seals the flow circuit in the uncoupled position.

Each hose was connected to the reservoir through a two-way, normally closed, solenoid-operated valve. In one of the return lines, a fourteen-turn vernier orifice valve was installed. The valve had a graduated micrometer type dial to permit reproducible valve settings. The outlet of the pump was connected to the other return line through a check valve. The check valve prevented the loss of cylinder oil through the pump circuit when the pump was not in operation. A schematic of the auxiliary hydraulic system is shown in Fig. 13.

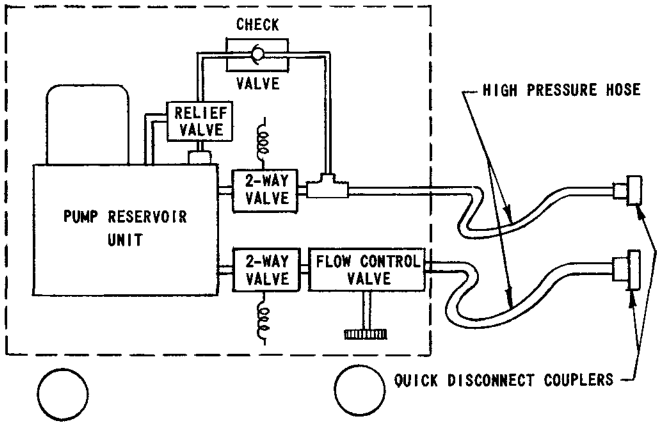

FIG. 13
SCHEMATIC OF AUXILIARY HYDRAULIC UNIT

The auxiliary hydraulic unit was compactly assembled and mounted on a four-wheeled frame. The unit could be easily wheeled to any drive as needed and readily removed again when not in use.

C. Test and Performance Characteristics

The attachment of the auxiliary hydraulic unit to the control rod drive required the addition of mating couplers to the drive to correspond with those on the hoses. One of the couplers was threaded into the cylinder air-exhaust port and the other was threaded into the cap of the dashpot surge reservoir. The hose which carried the auxiliary pump output was connected to the surge reservoir.

The photocell equipment and circuitry used with the control rod drive for obtaining time data were similarly used to determine slow-scram time information. The solenoid-operated two-way valves were electrically connected with the magnet release switch so that either one or both valves, as required, would be opened when the magnet current was interrupted.

In the first series of tests, both return lines were used to exhaust the cylinder oil. The procedure was to raise the rods and fill the cylinder with oil, and then record the time necessary for the rods to descend under different pressure loadings and exhaust orificing conditions. In filling the cylinder with oil, it was important that air was not introduced into the system to produce irregular operation. To prevent any possible air entry, the auxiliary pump was started with the magnet energized in the down position. With the magnet energized, the solenoid valves in the return lines were closed so that the pump output was dumped directly back to the reservoir through the relief valve in the pump circuit. When the drive motor was started, hydraulic oil was supplied to the cylinder volume vacated by the rising piston. The pump output was greater than the cylinder demand, so excessive oil was continually dumped back into the reservoir during control rod lift. When the rods were fully raised and the drive motor stopped, the pump was shut off. The oil under the piston remained in the cylinder, since both return lines and pump circuit were closed.

The first test was made with no air pressure on the cylinder piston and with the vernier valve in the return line full open at the fourteen mark on the graduated dial. When the magnet release switch was actuated to de-energize the magnet, the solenoid valves simultaneously opened both return lines to the reservoir. The Sanborn recorder charted the time intervals between consecutive photocells up to the point of dashpot entry.

The same procedure was repeated, using decreasing vernier dial settings until the valve was fully closed. Then the same series of valve settings were repeated with air loadings on the cylinder. Times were recorded at 3.4-atm intervals up to a pressure of 20.4 atm. Figure 14 shows the curves of time versus valve setting obtained at the test pressures.

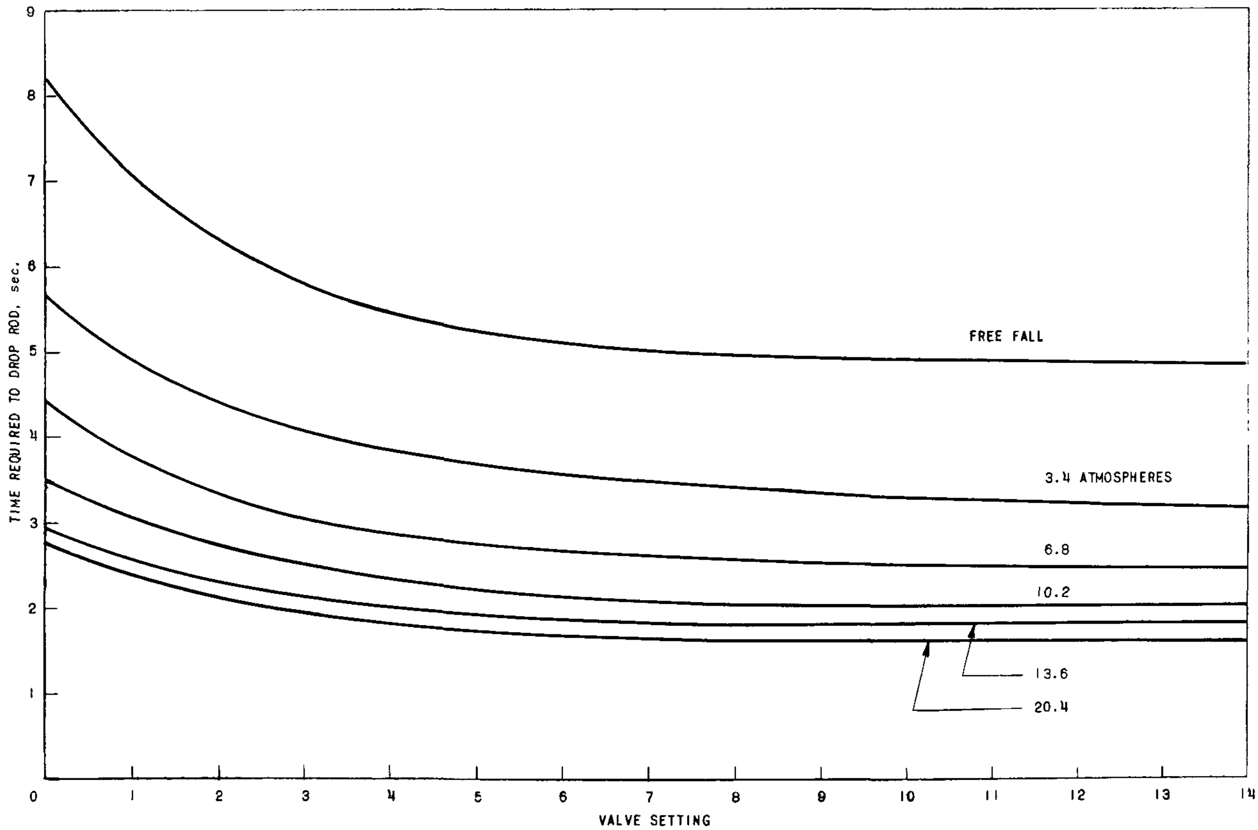

FIG. 14
TIME VS. VALVE SETTING FOR TWO LINE CIRCUIT
TREAT SLOW SCRAM HYDRAULIC CONTROL SYSTEM

In the second series of tests, only the return line with the vernier valve was used to empty the cylinder. The test procedure was the same as in the previous series with the exception that only one return line was opened when the magnet was de-energized. The valve settings and pressure loadings of the initial tests were repeated. Figure 15 shows the curves obtained from this series.

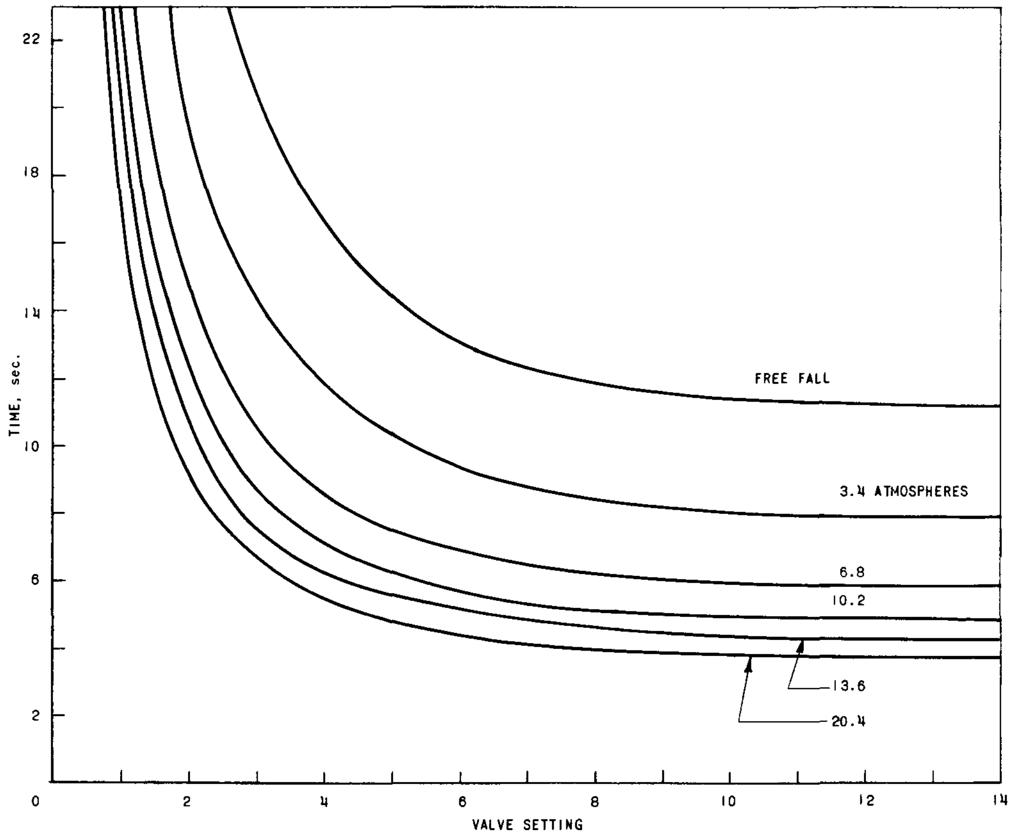

FIG. 15
TIME VS. VALVE SETTING FOR ONE LINE CIRCUIT

The results of the tests indicated that the auxiliary hydraulic unit was capable of producing controlled rod drops from 1.6 sec upward based on a maximum cylinder loading of 20.4 atm. The 1.6-sec minimum time was sufficiently short enough to accomplish the desired reactor test results so that further pressure loadings to obtain shorter times were not attempted.

The rod drop speeds produced by the auxiliary equipment were linear at all pressures and valve settings tested. A typical plot of rod travel against time is shown in Fig. 16.

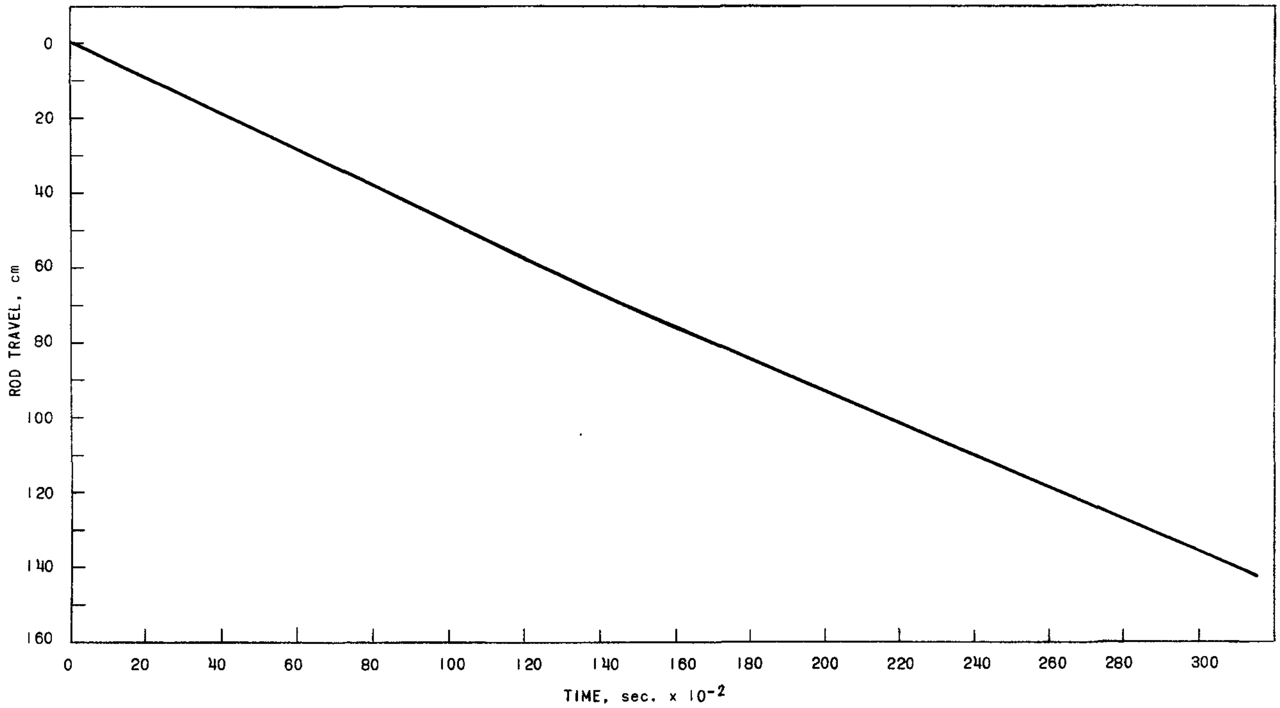

FIG. 16
ROD TRAVEL VS. TIME AT 20.4 ATMOSPHERES
TWO LINE SYSTEM WITH 0 VALVE SETTING

Appendix B

ROD DROP CALCULATIONS

From Newton's Law,

$$F = ma = m\frac{dv}{dt} = mv\frac{dv}{dx} \quad , \tag{1}$$

where

F = force, grams
m = mass of moving components, gm-sec²/cm
a = acceleration, cm/sec²
x = distance traveled, cm
v = velocity of moving components, cm/sec.

The downward force is

$$F = W + PA - P_aA_a \tag{2}$$

where

W = weight of moving components, gm
P = pressure on cylinder piston at any distance of piston travel, gm/cm²
A = rod end piston area, cm²
P_a = atmospheric pressure, gm/cm²
A_a = cylinder area, cm².

If the expansion of air is taken to be an isentropic process, the pressure P at any distance x is found by the expression

$$PV^k = P_0V_0{}^k \quad ,$$

where

P_0 = initial pressure at $x = 0$, gm/cm²
V_0 = initial volume at $x = 0$, cm³
V = volume at any distance x, cm³
k = ratio of molal heat capacities = 1.4, for air.

Thus

$$P = P\left(\frac{V_0}{V}\right)^k$$

and, since

$$V = V_0 + AX \quad ,$$

therefore

$$P = P\left[\frac{V_0}{V_0 + AX}\right]^k \quad . \tag{3}$$

Neglecting friction losses and assuming that the pressure of the exhaust air beneath the piston remains atmospheric, the velocity at any distance X can be found from equations (1), (2), and (3), thus

$$mv\frac{dv}{dx} = W + P_0A\left[\frac{V_0}{V_0 + AX}\right]^k - P_aA_a$$

or

$$v\,dv = \frac{W}{m}dx + \frac{P_0A}{m}\left[\frac{1}{1 + \frac{AX}{V_0}}\right]^k dx - \frac{P_aA_a}{m}dx \quad .$$

Integrating gives

$$\frac{v^2}{2} + C = g\,X + \frac{P_0A}{m}\left[-\frac{1}{\frac{A}{V_0}(k - 1)\left(1 + \frac{AX}{V_0}\right)^{k-1}}\right] - \frac{P_aA_a\,X}{m} \quad .$$

When X = 0, the velocity is also zero; therefore, solving for the constant C,

$$C = \frac{P_0A}{m}\left[\frac{1}{\frac{A}{V_0}(1 - k)}\right]$$

$$= \frac{P_0V_0}{m(1 - k)} \quad .$$

Therefore, the velocity at any distance X is given by

$$v = \sqrt{2g\,X + \frac{2\,P_0V_0}{m(k - 1)}\left[1 - \left(\frac{V_0}{V_0 + AX}\right)^{k-1}\right] - \frac{2\,P_aA_a\,X}{m}} \quad .$$

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W. J. Kann	N. G. Sobol
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