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Argonne National Laboratory

FARET CONTROL ROD DRIVE MECHANISMS

by

E. Hutter, T. E. Sullivan,
and G. D. Giorgis

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ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

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Reactor Engineering Division

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FARET CONTROL ROD DRIVE MECHANISMS

by

E. Hutter, T. E. Sullivan
and G. D. Giorgis

ABSTRACT

FARET was designed to be a 50-MW(t) sodium-cooled fast reactor experimental facility. Twelve control drives, operating above the reactor vessel, but inside of a large shielded cell, were to move 12 poison control rods over a 24-in. vertical distance. The control drives are electro-mechanical devices, that include pneumatic scram accelerators and hydraulic decelerators. A gripper mechanism at the lower end is submerged in 1150°F sodium and is remotely controlled to grip or to release a control rod. The detailed design of the control drive cluster, its operating characteristics, and preliminary tests of some control drive components are described.

I. INTRODUCTION

This report is one of a series designed to record, from inception in 1962 to termination in 1965, research and development activities pertinent to design of a reactor complex for irradiation-performance testing of advanced fuels for large, fast reactors.

The conceptual design of this complex, identified as the Fast Reactor Test Facility (FARET), and the proposed experimental program, were described briefly in ANS-100.¹ A more detailed description of the program objectives and nature of experiments was published as ANL-6708.²

Certain design features of the overall plant description are summarized in this report. The balance of the report is confined to a detailed description of the control rod drive mechanisms and results of out-of-pile component tests conducted up to termination of the FARET Project in December, 1965.

II. REACTOR AND PLANT DESCRIPTION

The FARET facility has been designed as a versatile fast reactor system having a heat removal capacity of 50 MW(t) at a maximum mean

coolant temperature at the core outlet of up to 1200°F. It can accommodate a diversity of reactor configurations, coolant flow rates, and core sizes operating over the full range of power levels (0 to 50 MW) and at a variety of temperature levels. The reactor vessel can accommodate cores having diameters of up to $3\frac{1}{2}$ ft. Smaller cores for high-power-density experiments can also be assembled.

The basic unit of the reactor is a subassembly of the EBR-II type. The subassemblies are hexagonal in cross section and are arrayed in a triangular lattice. Within the subassembly a variety of fuel elements can be arranged. Each subassembly is in itself a test unit. Test units can be arranged singly, in groups, or as whole cores. Burnup rates of up to 1 percent per month are possible. Provision for instrumentation of core components is an essential feature of the facility. In addition to the subassembly test units, closed in-pile, re-entrant-type loops are provided.

The facility comprises a single building of conventional construction and a few miscellaneous structures. The building consists of two principal areas: a reactor building and a support wing. The reactor building houses the containment structure, the reactor vessel and reactor coolant systems, the fuel-handling equipment, and various auxiliary systems as shown in Fig. 1. The support wing contains the offices, personnel facilities, plant support utilities, and the reactor control room.

The containment structure is composed of three integrated, steel-reinforced, steel-lined monolithic concrete enclosures: the cell, the vault, and the reactor cavity. These enclosures provide biological shielding, inert gas environment, and containment in event of a nuclear incident.

The cell is located above the ground-floor level of the reactor building, and has an upper and a lower section. The lower section is 20 ft wide, 68 ft long, and about 37 ft high. The walls are of 5-ft-thick ordinary concrete below the crane haunch and diminish to 3 ft of ordinary concrete above the haunch. The ceiling is 4-ft-thick ordinary concrete. The lower cell contains the upper portion of the reactor vessel, the control rod drive mechanisms, the fuel-handling and disassembly equipment, and the upper portion of the wet fuel-storage tank. The area is serviced by a crane. Provisions are made for the later addition of remote viewing and handling equipment.

The upper section of the cell, 36 ft long by 24 ft wide by 29 ft high, is located over the reactor vessel. The walls are 3-ft-thick ordinary concrete with a 4-ft-thick concrete roof. The upper cell contains equipment for raising and lowering the reactor vessel cover and integral control drive mechanisms. When the cover is fully elevated, the lower cell crane can pass under the cover to service the reactor vessel area. The reactor vessel cover and upper cell floor are connected by an impregnated fabric bellows as an atmospheric seal between the upper and lower cell areas.

1. BUILDING CRANE
2. STACK
3. VESSEL COVER HOIST
4. VESSEL COVER AND BELLOWS ASSEMBLY
5. SECONDARY SODIUM PUMP
6. HEATING AND VENTILATING EXHAUST
7. HEATING AND VENTILATING SUPPLY
8. CELL CRANE
9. CONTROL ROD DRIVES
10. FUEL HANDLING MACHINE
11. PRIMARY SODIUM PUMP
12. SODIUM TO AIR HEAT EXCHANGER
13. FUEL AND EQUIPMENT PORT
14. WET FUEL STORAGE TANK

15. REACTOR VESSEL
16. FUEL TRANSFER CASK
17. PERSONNEL ACCESS LOCK
18. PRIMARY HEAT EXCHANGER

Fig. 1. Reactor Building and Components

III. CONTROL ROD DRIVE MECHANISMS

A. Design Criteria

The following criteria were established for the design of the control rod drive system:

- (1) The core lattice shall contain twelve control rods spaced 6.138 in. apart in a circular configuration approximately 24 in. in diameter.
- (2) Each control rod shall be operated by an identical and interchangeable drive mechanism.
- (3) The twelve control rod drive mechanisms shall be assembled atop the reactor vessel cover within the space restrictions imposed by the cover-lifting structure and bellows.
- (4) Each control rod shall be moved vertically at a constant speed of 6 in./min over a distance of 24 in.; upward movement shall tend to increase reactivity and downward movement decrease it.
- (5) Electrical and mechanical safety devices shall be employed to prevent simultaneous raising of all control rods any significant distance above their fully inserted position.
- (6) Each control rod shall be capable of rapid downward movement (scram), from any operating level in excess of 8 in. above the fully inserted position, with accelerated speeds of 1 to 2 g.
- (7) Maximum elapsed time between electrical initiation and mechanical response of a scram stroke shall be 0.020 sec.
- (8) Interruption or loss of electrical power shall cause immediate scram of all control rods.
- (9) Deceleration of the scrambled control rods shall be accomplished within the last 8 in. of the stroke.
- (10) Each control rod drive mechanism shall be capable of positive engagement or disengagement with its rod by remote control.
- (11) It shall be mechanically impossible to disengage the control rods at other than their fully inserted position.
- (12) There shall be remote indication of positive engagement or disengagement of the control rods.
- (13) Devices for sealing drive-shaft penetrations in the reactor vessel cover shall be gastight without restricting axial motion of the shafts.
- (14) Materials for all components within the reactor vessel shall be compatible with sodium vapor at 1150°F.

B. Mechanical Design

The major features and operating characteristics of the control rod drive design evolved are summarized in Table I.

TABLE I. Summary of Control Rod Drive Design and Operating Characteristics

Number of control rods and drives	12
Radial distance from vertical core centerline (avg)	11.83 in.
Calculated reactivity worth of each rod	0.00667 $\Delta k/k$
Nature of drive	Electromechanical (ball-nut screws) with pneumatic scram assist.
Control rod travel	24 in.
speed	6 in./min
Scram delatching	Electromagnet
delatch time	0.017 sec
Pressure in pneumatic scram assist	12 psig
Scram acceleration	1.5 g
Scram speed (max)	125 in./sec
Elapsed time for initial 16 in. of 24-in. scram stroke	0.232 sec
Scram decelerating device	Hydraulic (oil) dashpot

As shown in Fig. 2, the twelve control rod drive mechanisms and supporting structure are located immediately above and attached to the reactor vessel cover. The compact cluster of drive mechanisms and auxiliaries is surrounded by a metal-reinforced elastomer bellows which isolates the control rod drive region from the cell atmosphere. Sufficient lateral space is provided around the cluster for access by maintenance personnel. Preparatory to fuel handling, the integrated vessel cover-control rod drive assembly is hoisted into the cell annex, as shown in Fig. 3.

With reference to Fig. 2, the major components of each drive mechanism include a pneumatic cylinder (scram accelerator) and shock absorber, a motor-driven ball-nut screw and ball-nut yoke assembly, a yoke-mounted electromagnet, and a motor-driven gripper actuator. The piston in the pneumatic cylinder forms the top end of the vertical drive

Fig. 2. Installation of Control Rod Drive Mechanism

Fig. 3. Handling System for Reactor Vessel Cover

shaft which extends downward through the electro-magnet and a bellows-sealed nozzle in the reactor vessel cover. At the bottom end, the drive shaft is fitted with a gripper which engages the control rod.

During normal operation, the armature on the drive shaft is magnetically latched to the vertically driven electromagnet to impart axial movement of the control rod into or out of the reactor core. Rapid insertion (scram) of the control rod is effected by de-energizing the electromagnet and combined gravitational and pneumatic acceleration of the rod.

The twelve control rod drive mechanisms are mounted and supported in a circular pattern on a horizontal platform which can be raised or lowered several inches by motor-driven jackscrews. For example, the platform is lowered to ensure engagement between the grippers and the control rods preparatory to reactor startup; it is raised to obtain ample clearance between the disengaged grippers and the control rods before the vessel cover is lifted for fuel handling.

The control drive-platform complex is mounted atop a base plate located just above the vessel cover insulation. This intermediate base plate isolates the thermally expanding cover from the fixed-mounted control drive components.

The following descriptions of the mechanical components and auxiliaries apply to each of the twelve control rod drives, starting with the gripper immediately below the reactor vessel cover in Fig. 2.

1. Control Rod Gripper

The primary function of the gripper is to effect engagement between the drive shaft and the control rod by remote actuation of the gripper shaft. The secondary function is to provide remote indication of positive completion of either operation.

This critical mechanism (see Fig. 4) is contained in a tubular body (of $2\frac{1}{2}$ -in. OD and $9\frac{3}{8}$ -in. long) which is attached, at the upper end, to the control rod drive shaft. The lower end is formed into a guide funnel and receiving chamber for the cone-shaped upper adapter of the control rod. A transverse shaft ($3/8$ in. in OD), supported by the body, serves as a common hinge pin for a pair of single-toothed jaws and also as a guide for the sensing device.

The jaws are opened or closed over the cone-shaped control rod adapter by sliding two parallel bars against their angular back sides (above and below the hinge pin). These parallel bars are integral with the bottom end of a sleeve which encircles the jaws and which can be moved vertically $1\frac{1}{2}$ in. with respect to the jaws. The inner surface of the sleeve slides on six, equally spaced bearing buttons located circumferentially on the outer surface of the gripper body. At the upper end, the sleeve is attached to four equally spaced keys that protrude through four guide slots in the body and are fastened to the flat sides of a square nut. This nut is threaded onto the bottom end of a tubular gripper shaft (of $7/8$ -in. OD) which is concentric with the outer drive shaft. In this manner, motion is imparted to the sleeve when the gripper shaft is motivated by the gripper drive mechanism above the vessel cover.

As shown in Fig. 4, the teeth of the jaws which actually support the adapter head protrude through the guide funnel. This design feature allows the jaw teeth to recede past the guide funnel upon opening and this eliminates any possibility for the adapter head to hang up on a jaw tooth due to slight misalignment of the control rod and gripper.

The presence or absence of a control rod upper adapter in the gripper guide chamber is detected by a sensing device which is centrally located in the chamber. This device is essentially a flat bar with a longitudinal slot; it is held captive and limited in vertical travel to $3/4$ in. by the

Fig. 4. Control Rod Gripper

hinge pin of the gripper jaws. The upper end of the sensing device is attached to a solid shaft (of $1\frac{1}{2}$ -in. OD); the bottom end is shaped to accept the impinging cone of the control rod adapter. Thus upward or downward movement of the sensing device (and shaft) transmits positive indication to the control console as to whether a control rod adapter has entered or left the gripper guide chamber.

Because it is one of the more difficult components to repair or replace in service, considerable emphasis was placed on durability and reliability of gripper operation in the anticipated high-temperature sodium environment. With respect to materials of construction, Haynes No. 25 alloy (50% Co, 20% Cr, 15% W, 10% Ni, 5% other elements) was selected for all gripper parts except the hinge pin, the two parallel bars, the six sleeve-bearing buttons, and the four keys. These small parts, being subject to greater wear, were to be fabricated of Haynes Stellite No. 6B alloy (59% Co, 30% Cr, 4.5% W, 6.5% other elements).

2. Concentric Shafts and Seals

The concentric drive, gripper, and sensing shafts extend upward from the gripper, through the reactor vessel cover, into the drive mechanism assembly. Since the functions of the gripper and sensing shafts are dependent on their unrestricted axial movement of $1\frac{1}{2}$ -in. and $3\frac{3}{4}$ in., respectively, relative to the drive shaft, effective sealing of the shafts is accomplished by interconnecting bellows assemblies, as shown in Fig. 5.

Two bellows seal assemblies are employed. Each assembly features flexing convolutions formed from Type 321 stainless steel, double-walled tubes whose total wall thickness is 0.010 in. Their extremities are resistance-welded to collars which, in turn, are welded to adapters attached to the respective shafts. The lower bellows, nominally of $1\frac{1}{2}$ -in. OD, is welded at the bottom to the drive shaft and at the top to the gripper shaft. The upper bellows, nominally of $1\frac{1}{8}$ -in. OD, is welded at the bottom to the sensing shaft and at the top to the gripper shaft.

The overall length of the bellows assembly is about $3\frac{1}{2}$ ft. This length was selected to ensure a travel capacity 80 to 100% greater than required and thus reduce the stress imposed on each convolution.

The annuli formed by the three, Type 304 stainless steel shafts are held to a minimum to reduce gamma-ray and neutron streaming. In addition, the diameters of each shaft are stepped so that (as space permits) each normally straight annulus is interrupted by a length of metal sufficient to attenuate the streaming. For example, the diameter of the central portion ($2\frac{1}{2}$ ft long) of the sensing rod is stepped from $1\frac{1}{2}$ in. to $3\frac{3}{4}$ in. The inside diameter of the gripper shaft provides a total operating clearance of $1\frac{1}{8}$ in. for the sensing shaft. The outside diameter is stepped from $1\frac{3}{8}$ in. (lower section) to $1\frac{3}{4}$ in. (upper section). Except for an 18-in.-long central section

which is of $2\frac{7}{8}$ -in. OD the outside diameter of the drive shaft is nominally $2\frac{1}{2}$ in. The wall thickness ranges from a minimum of $1/4$ in. to a maximum of $1/2$ in. where required for shielding.

Each drive shaft operates through a removable, Type 304 stainless steel guide tube installed in the respective vessel cover nozzles. Each tube extends upward from the bottom of the cover to several inches above the drive mechanism lifting platform. The inside diameters of the guide tube are designed to accommodate the drive shaft and its shielding section. In addition, two guide bushings are installed, one at the lower and the other at the upper end of the guide tube. The lower unit, made of Haynes Stellite No. 6B, provides a total operating clearance of 0.075 in. (minimum) for the drive shaft. The upper unit, made of aluminum-bronze Ampco No. 18-13, provides a total clearance of 0.020 in. (minimum); it can be removed independently of the guide tube. The guide tube is sealed to the vessel cover nozzle by a concentric nut which threads onto the nozzle and forces the guide tube flange to compress a soft, copper gasket between the flange and the nozzle gasket seat.

A nesting-type bellows seal is used to effect a gastight connection between the guide tube and the drive shaft throughout its 24-in. travel. This seal, shown in Fig. 6, consists of 400 flat ring disks which are welded together at the inner and outer edges. Each disk is made of Type 347 stainless steel and measures $2\frac{9}{16}$ in. in ID, $4\frac{1}{2}$ in. in OD and is 0.010 in. thick. The lower end fitting is threaded onto the top end of the guide tube, compressing a soft copper gasket to effect a gastight seal.

Eight longitudinal grooves are machined into the outer surface of the drive shaft. These grooves provide ample flow area for the gas that is forced out of the rapidly collapsing bellows during a scram stroke.

Immediately above the bellows, the drive shaft penetrates a housing for the gripper shaft jackscrew mechanism. This mechanism converts rotary input from the gripper drive to linear movement of the gripper shaft required to actuate the gripper jaws.

At this point, the gripper shaft is threaded through a spur gear which is rotated by an idler gear and a driving gear contained in a horizontal extension of the housing. The shaft of the driving gear extends downward from the housing; it is cone-pointed and fitted with a cross driving pin. In operation, this pin is engaged by the output shaft coupling of the gripper drive mechanism.

A horizontal lever is attached to the top end of the gripper shaft. This lever protrudes through the housing wall and actuates the gripper data transmission assembly. A similar lever on the sensing shaft actuates the sensing data transmitter.

Fig. 5. Concentric Shafts and Seals

Fig. 6. Drive-shaft Seal Bellows

Several inches above the jackscrew housing, the control rod drive shaft is attached to and protrudes through a ball joint-magnet armature assembly (see Fig. 8). This assembly permits some misalignment between the magnet-armature interface. Just above the armature, the drive shaft is coupled, by a threaded connection, to the piston rod of the scram accelerator.

3. Scram Accelerator and Shock-absorber Assembly

The functions of this assembly (Fig. 7) are (1) to enhance gravitational acceleration of the drive shaft (and control rod) during a scram stroke, and (2) to effect gradual deceleration of the rod over the last 8 in. of travel.

The scram accelerator is a pneumatic cylinder-piston assembly (4 in. in OD, and 24 in. long) which terminates at the top end in a gas-inlet fitting (of $1\frac{3}{4}$ -in. dia). This fitting is connected through a 2-in. quick-disconnect coupling and a short length of hose to a toroidal gas accumulator which supplies each of the twelve accelerators. Each of the interconnecting hoses features a flow-check valve which closes automatically when the hose is disconnected from the cylinder. A constant force of about 150 lb is maintained on the piston with an accumulator gas pressure of 12 psig. During a scram stroke, this results in 1.5-g acceleration of the control rod.

The lower end of the pneumatic cylinder is attached to the shock-absorber assembly, consisting of a hydraulic plunger and dashpot. The plunger is an open-bottom cylinder which is attached to the piston rod. It is tapered in steps (diminishing diameters) in the direction of the dashpot.

The dashpot is an oil-filled annular well, about 8 in. deep which is formed by an inner standpipe (of 2-in. OD) and an outer housing (of $3\frac{3}{8}$ -in. ID). This housing is encircled by a reservoir which collects the oil displaced by the plunger. A transparent plastic window in the side wall of the reservoir provides visual indication of the oil level. Remote warning of a drop in oil level is transmitted by a float-operated magnetic switch in the reservoir.

During a scram stroke, gradual absorption of the kinetic energy by displacement of the oil results in a maximum deceleration of about 4.5 g.

4. Drive-shaft Actuating Mechanism

This mechanism (see Fig. 8) raises and lowers the control rod drive shaft. It consists basically of an electric motor, a pair of lead-screws and ball-nuts, and a ring electromagnet. The shaft-attached armature is held only by magnetic attraction to the electromagnet (when energized), which is made to traverse a vertical distance of 24 in. This method of

Fig. 7. Scram Accelerator and Shock-absorber Assembly

Fig. 8. Drive-shaft Actuating Mechanism

engagement frees the scrambling shaft of encumbrances and also eliminates scram-induced shock loads on the driving components and data-transmitting synchros.

The gear train associated with the drive mechanism is installed below the shock absorber. This gear train is driven by a $1/4$ -hp, polyphase electric motor, equipped with a magnetic brake, and an integral worm-gear speed reducer. The output shaft (36 rpm) is coupled to a pair of 2:3 ratio bevel gears which, in turn, drive a friction-disk torque limiter at 24 rpm.

A spur gear on the torque limiter drives one of the two spur gears attached to the vertical lead screws on either side of the piston rod. The second spur gear is driven by an idler gear which rotates on a fixed bearing through which the piston rod travels.

The lead screws have a nominal diameter of 1 in. and a pitch of 0.25 in. They are made of SAE 4150-H steel hardened to Rockwell C-56 (min). Each screw rotates in radial ball bearings at the upper end, and in needle bearings at the lower end.

Two ball-bearing nuts, which thread onto the lead screws, are rigidly attached to each end of a yoke which spans the screws. The center of the yoke contains a large bronze bearing. This bearing houses an annular electromagnet (of $1\frac{7}{8}$ -in. ID) through which the accelerator piston rod travels during a scram stroke. Downward movement of the electromagnet is restricted, but it can be forced upward slightly against three spring-loaded plunger assemblies mounted in the yoke. This arrangement eliminates critical adjustment of the bottom limit switch of the drive mechanism. The switch is actuated when the magnet is driven downward to contact the armature after a scram stroke. Contact is indicated by a yoke-mounted switch which is actuated by a rod that passes through a hole in the magnet and touches the armature.

The Armco iron core of the electromagnet is $4\frac{1}{4}$ in. in OD and $2\frac{3}{4}$ in. long; it is chrome-plated. Embedded concentrically in the magnet is a helical coil of No. 20 B&S gauge, enameled copper wire.

The thickness of the chrome plate on both the magnet pole and armature surfaces (0.0015 in.) provides an optimum magnetic gap of 0.003 in. The coil is energized by a d.c. power supply of 6 V at 1.3 A.

5. Gripper Drive Mechanism

Rotary input to the gripper-shaft jackscrew mechanism is effected by the gripper drive assembly, which is mounted on the control rod drive lifting platform.

As shown in Fig. 9, the rotary output shaft of the gripper drive terminates in a flexible coupling made of two steel shaft hubs connected by a molded elastomer sleeve. The lower hub is keyed to the output shaft. The upper hub features a bore-and-slot arrangement designed to accommodate the drive pin of the jackscrew shaft that extends downward from the housing attached to the control rod drive shaft. When the drive shaft is scrambled, the cross-pin of the jackscrew shaft lacks 1 in. of being engaged with the upper hub of the coupling. Full engagement is effected only when the coupling is moved upward by the push-type solenoid in the gripper drive.

The gripper drive housing supports an external 1/8-hp, 220-V, polyphase gearmotor (30 rpm) which drives a friction-plate torque-limiting clutch. A clutch-mounted spur gear, acting through an idler gear, drives another spur gear. The square bore of this spur gear is designed to receive, with a sliding fit, the squared portion of the output shaft that passes through it.

The lower end of the output shaft is ball bearing-mounted in a yoke that straddles two vertical guide posts. Assembled to the guide posts are compression springs which hold the yoke against the armature of the push-type solenoid. Two switches, actuated by the yoke at either end of its 1-in. travel, provide indication of shaft movement.

Just below the flexible coupling, the output shaft is geared to a power-takeoff shaft which drives the lifting platform safety interlock system. This system is actuated by two of the twelve gripper drive mechanisms.

6. Transmission of Gripper and Sensing Data

Two sets of equipment are used to transmit gripper and sensing data to the control room console. Each set (see Fig. 10), consisting of two limit switches and a Size 18 synchro, is mounted on opposite sides of a common support frame attached to the lifting platform. Both sets are operational only when the control rod drive shaft is in the fully inserted position. In this position, the horizontal arms attached to the gripper and sensing shafts contact the respective actuating plungers of the equipment. During subsequent control rod gripping operations, each plunger is depressed the distance corresponding to the travel of the gripper shaft ($1\frac{1}{2}$ in.) and the sensing shaft ($\frac{3}{4}$ in.).

Each plunger is a vertical, round rod that slides in sleeve bearings mounted in the synchro housing. A concentric coil spring is installed between the housing and the adjustable, upper end fitting of the rod. This spring returns the depressed plunger to the preoperative position. Attached to the top end fitting is a horizontal bar which actuates a travel limit switch.

Fig. 9. Gripper Drive Mechanism

Fig. 10. Transmitters for Gripper and Sensing Data

Linear motion of the gripper plunger rod is converted to about 238° of synchro rotation. This is accomplished by a gear rack (machined in the lower portion of the rod) which drives the pinion shaft of the synchro.

7. Transmission of Drive-shaft Position Data

During normal operation, an assembly of two Size 15 synchros and a precision gear train provide continuous indication of control rod position. The assembly is inoperative during a scram stroke.

Both synchros are mounted on the lead-screw drive-gear housing and interconnected by spur gears in the ratio 36:1. The faster synchro is also connected through a 6:1 ratio gear train to a precision spur gear which is attached to the upper end of one lead screw. Since the lead screws made 96 revolutions during normal control rod travel, the fast synchro makes 16 revolutions and the slow synchro less than one revolution (160°) throughout the 24-in. stroke. Outputs of the respective synchros are transmitted to corresponding "fine" and "coarse" rod travel indicators in the control room.

Seven switches actuated by the drive shaft also provide information on control rod movement. Six switches are used to indicate upper and lower midtravel transfer ranges. The seventh switch is used with other equipment to determine elapsed time for the upper 16 in. of the 24-in. scram stroke.

The switches are mounted at predetermined intervals on both sides of a vertical bar attached to the lifting platform support structure. They are actuated by a pair of 5-in.-long, ski-shaped cams attached to a sliding member which is guided by a long slot in the support bar. A driving pin, pressed into the sliding member, is held loosely captive in a slotted bracket which is fastened to the drive shaft. In operation, the latter bracket moves with the drive shaft, thus raising or lowering the sliding ski-shaped switch actuators over the full shaft stroke.

8. Lifting Platform and Support Structure

The twelve control rod drive mechanisms and the gas accumulator are mounted and supported on a platform (see Fig. 11) which can be raised or lowered $4\frac{1}{8}$ in. This is done to effect simultaneous disengagement or engagement of all control rod grippers before and after fuel-handling operations.

The platform is a steel plate approximately 50 in. long, 46 in. wide, and $2\frac{1}{2}$ in. thick. Four equally spaced columns extend upward from the platform and are joined at the top by a common circular framework. Upper portions of the drive mechanisms and the gas accumulator are supported by this framework. An intermediate circular framework provides bearings for the lower ends of the drive mechanism lead-screws.

Fig. 11. Lifting Platform and Base-plate Assembly

The toroidal gas accumulator (of 64-in. OD), mounted atop the structure, has a volume of 16.5 ft³. It is fed through a check valve and a constantly open inlet line by a pressurized supply of argon gas. Equally spaced around the accumulator are twelve outlet hoses that supply the scram accelerators. The accumulator is equipped with a pressure gauge, an overpressure-relief valve, and a high-low-pressure electrical switch. The normal operating pressure is 12 psig.

The platform structure is raised or lowered by four jackscrews. The jackshafts are driven by a 3/4-hp polyphase gearmotor, equipped with a magnetic brake, through a friction-disk torque-limiting clutch and an arrangement of miter-gear assemblies and connecting shafts. The jacks are synchronized by placing shims of proper thicknesses under the jackscrew bases.

During reactor operation, the platform is supported on six adjustable posts which rest on pads attached to a ring that encircles the drive shaft nozzles. After reactor shutdown, and preparatory to downward travel of the platform, the pads are rotated out from under the posts by a motorized actuator.

The gripper drive mechanisms of two, diametrically opposed control rod drive assemblies operate respective safety interlock devices attached to each side of the platform. In case of electrical interlock failure, these devices prevent simultaneous raising of all control rod grippers any significant amount above their lowest operating elevation.

Each interlock is connected through a flexible shaft to a power take-off shaft on the gripper drive mechanism. Each device houses a mechanism that extends or retracts a sliding bolt when the gripper jaws are closed or opened, respectively. This bolt engages a slotted restraining bracket fastened to the base plate. This bracket permits the platform to be raised 0.1 in.

During reactor operation, the control rods, at their lowest operating elevation (or after a scram stroke), are suspended 7/8 in. above their support bars in the core lattice. This clearance prevents damage to the control rod when the rod is driven down or scrammed. When it is desired to separate the control rods from the grippers, the platform must be used to lower the twelve grippers to the "Gripper Function" elevation. Before attempting downward movement, the platform is first raised 0.1 in. to the "Pads Released" elevation, so that the pads can be rotated from under the platform support posts, and then lowered 7/8 in. to rest the control rods on the grid bars. Since the control rods and the grippers may vary slightly in elevation, the platform is lowered an additional 1/4 in., depressing the spring-supported control rods nominally the same amount. This procedure assures full insertion of the upper adapter cone of the control rod into the gripper guide chamber. At this point, it is possible to open or close all

jaws simultaneously. With the platform at the "Gripper Function" elevation, the jaws are driven open. Then the platform is raised $4\frac{1}{8}$ in. to its highest or "Gripper Clear" elevation to provide a generous clearance between the grippers and the control rods before the reactor vessel cover is raised for fuel handling.

At two different elevations during platform ascent, checks are performed to ascertain control rod separation. At $1\frac{1}{4}$ in. of upward travel, the platform stops at the "Sensing Check" elevation. Here the sensing indicators are checked to determine that no control rod is stuck in its gripper receiving chamber. The platform is raised $1\frac{1}{2}$ in. more to the "Jaw Check" elevation. Here the gripper jaws are driven closed to ensure that no control rod is stuck to its gripper sensing device. If no interference is detected, the jaws are driven open again before resuming upward travel.

Returning to the operating mode requires that the platform first be lowered to the "Gripper Function" elevation. The gripper jaws are driven closed, and the platform is raised to the "Pads Released" elevation. At this point the pads are rotated under the platform posts, and the platform is driven down on them at the "Operating" elevation.

Platform movement is controlled by an assembly of six switches housed in a box fastened to the base plate on one side of the platform. The switches are contacted by a common actuator connected to a bar which extends from the platform. With one exception, each switch stops the platform precisely at a certain elevation: the "Operating" elevation switch is actuated when the platform has been lowered to a point $1/16$ in. above the "Operating" elevation. However, instead of stopping the motor directly, the switch energizes a timing relay which stops the motor only after the platform has settled on the pads and the torque-limiting clutch in the drive system has slipped several revolutions.

9. Base Plate

In Fig. 11 is shown the steel base plate which supports the control rod drive mechanism-lifting platform complex. This plate (59 in. in diameter and $2\frac{1}{2}$ in. thick) has twelve holes for the drive-shaft nozzles and a large central hole to accommodate the experimental nozzle in the center of the vessel cover. An adjustable ring, fitted with a bronze ring insert, is fastened around the periphery of the central hole and closely fitted to the machined outer diameter of the nozzle flange to permit accurate lateral positioning of the base plate.

The base plate is supported by six equally spaced posts arranged in a circular pattern near its outer edges. The bottom flanges of the posts are rigidly fastened to the top surface of the vessel cover; the upper ends protrude through radially slotted holes in the base plate. Load on the base

plate is transferred to radially aligned linear roller-bearing assemblies mounted on each post. This arrangement minimizes heat transfer from the vessel cover to the platform and permits the cover to flex without affecting the base plate.

10. Routing of Electrical Cables and Gas Lines

The base plate is encircled by a cable raceway of rectangular cross section which is fastened to the floor of the vessel cover-lifting structure. This raceway is equipped with a large opening to accommodate entry of gas-supply hoses for the accumulator and 28 electrical cables (of $1\frac{1}{2}$ -in. dia). These feeder cables and hoses are suspended from large pulleys mounted in the cable take-up mechanism attached to the cell annex wall (see Fig. 3) and are retained by a multiple-clamping bracket just above the raceway entry.

The cables divide at the point of entry and proceed in both directions to twelve outlets spaced around the raceway. At each outlet, conductors from the large feeder cables enter a group of small flexible conduits or cables of appropriate sizes. Each group emerges from the wall of the raceway and connects to receptacles on a control rod drive mechanism.

Electrical components of the lifting platform are connected similarly by cables at convenient raceway outlets.

IV. SUPPORTING RESEARCH AND DEVELOPMENT

This section describes the design criteria established for certain critical components and the analytical and out-of-pile experimental techniques employed to ensure that these criteria were satisfied.

A. Electromagnet

1. Design Criteria

The close spacing between control rod drives limited the OD of the magnet to $4\frac{1}{4}$ in. Also, in order to permit rapid travel (scram) of the drive shaft (of $1\frac{5}{8}$ -in. OD) through the magnet, the ID of the magnet was held to $1\frac{7}{8}$ in. The lifting capability of the magnet was specified as 1100 lb; this was determined by adding the scram weight of the rod, the scram assist force, and a suitable safety factor.

2. Design Calculations

With reference to Fig. 12, the required effective area of the magnet was calculated from the relation

$$A = 7.213 \times 10^7 F/B^2 = 7.55 \text{ in.}^2,$$

where

$$F = \text{lifting force} = 1100 \text{ lb};$$

$$B = \text{flux density} = 1.03 \times 10^5 \text{ lines/in.}^2 \text{ (based on 16,000 gauss for Armco iron).}$$

NOTE: $A_1 + A_2 = A$. To ensure a uniform flux path, A_1 should be approximately equal to A_2 , with the balance of the cross-sectional area (A_3) utilized for the coil.

Fig. 12. Design Dimensions of Electromagnet

Knowing the cross-sectional area of the coil, the height of the coil was selected proportional to the overall height of the magnet. Based on this height, and using No. 20 B&S gauge, enameled copper wire, the total number of turns was calculated to be 534.

The 0.003-in. gap between the armature and the magnet was obtained by plating both surfaces with chromium to a thickness of 0.0015 in.

The current requirements of the magnet were calculated and checked against the actual requirements ($I = 1.3 \text{ A}$), with the following results:

The calculated value for I was obtained from the familiar relationship

$$NI = 0.313 B l / \mu,$$

where

I = current in amperes;

N = number of wire turns = 534;

B = flux density = $1.03 \times 10^5 \text{ lines/in.}^2$;

μ = permeability of flux path

= ~1200 Armco iron

= unity for air;

ℓ = length of flux path
 = 6.30 in. for the iron
 = 0.006 in. for two air gaps.

Accordingly,

$NI = 169$ for the iron = 189 for the two air gaps.

Therefore,

$$I = (169 + 189) / 534 = 0.67 \text{ A.}$$

Compared to the actual value of 1.3 A, the calculated value indicates an efficiency of ~50%. This low efficiency could be attributed to (1) the permeability for Armco iron, which may be somewhat lower than 1200, or (2) Eq. (1) may be based on a magnet of ideal geometry.

3. Prototype Tests

Figure 13 shows the test assembly used to develop and evaluate the optimum performance characteristics of the prototype magnet shown in Fig. 12. Load was applied to the magnet with a hydraulic jack. Direct current was supplied to the magnet from a rectifier through an ammeter, with a voltmeter connected across the magnet.

Fig. 13. Test Assembly to Determine Electromagnet Lifting Force

Figure 14 shows the switching circuitry employed to measure the load-release time. The magnetic gaps were varied by inserting brass shims between the magnet and armature.

Fig. 14

Armature Load-release Time Circuitry. Limit switch (A) is attached to the magnet support, with the actuator adjusted to barely close the switch when the armature contacts the magnet. When pushbutton switch (B) is depressed, the magnet power circuit is interrupted, and switch (C) closes in series with switch (A), completing the timer circuit. When sufficient load is applied to the armature, it breaks away from the magnet. Switch (A) opens and de-energizes the timer, which indicates the load release time in milliseconds.

The scope of the tests included magnet lifting force and armature release time vs coil current for different air gaps, and magnet temperature rise vs coil current. Load on the armature was maintained at 750 lb (estimated scram weight plus the scram-assist force).

On the basis of the test results (see Figs. 15 and 16), it was concluded that the optimum gap should be 0.003 in. With this gap, the

Fig. 15. Performance Characteristics of Prototype Electromagnet

Fig. 16. Characteristics of Prototype Electromagnet--Coil Current vs Temperature

magnet lifting capability (1100 lb), load-release time (0.017 sec), and temperature rise (23°F) satisfied the criteria specified for the FARET control rod drive mechanisms.

B. Shock Absorber

1. Design Criteria

The shock absorber must completely absorb the energy of the control rod drive shaft and rod, which is accelerated at 1.5 g. Moreover, deceleration of the moving mass should not exceed 5 g.

2. Design Calculations

The shock absorber utilizes the principle of a stepped-diameter piston entering a constant-diameter dashpot filled with oil to absorb the energy of the accelerated drive shaft and control rod. Both the shock-absorber plunger and the dashpot were proportioned by balancing the energy of the accelerated mass (1.5 g) to the energy of the oil flowing past the annulus formed by the stepped piston entering the dashpot.

An oil of known viscosity (SAE 30W) was assumed and the discharge energy of the exiting oil was calculated by means of the energy balance relationship

$$\Delta KE = \Delta QH,$$

where

ΔKE = kinetic energy absorbed by first increment of plunger entering the dashpot;

Q = oil displaced by first increment of plunger, lb;

H = total head of the flowing oil

$$= h_{\text{exit}} + h_f + (V^2/2G).$$

By substituting $0.5(V^2/2G)$ for h_{exit} , and the Darcy-Weisbach relationship for h_f , the total head of the flowing oil was given by

$$H = 0.5(V^2/2G) + f(L/D_e)(V^2/2G) + (V^2/2G),$$

where

V = average velocity of oil flowing through the annulus, ft/sec;

L = length of first increment of plunger, ft;

D_e = equivalent hydraulic diameter of annulus, ft;

f = friction factor which varies from 0.7 to 0.9, depending on Reynolds Number for the various velocities of the oil.³

After determining ΔKE from the above relations, it was subtracted from the total kinetic energy of the system to obtain the net kinetic energy remaining for the second increment of plunger entering the dashpot:

$$KE_{Net} = KE_{Total} - \Delta KE,$$

where

$$KE_{Total} = 1/2 MV^2.$$

This procedure was repeated for each succeeding step of the plunger until the total energy absorbed by the dashpot equalled the total energy of the system. In this manner, the number of steps required and the clearances were determined; these, in turn, established the length of the dashpot ($6\frac{1}{2}$ in.)

3. Prototype Tests

Figure 17 shows the test arrangement employed to determine the optimum shock-absorbing characteristics of the prototype absorber assembly for various scram-assist air pressures.

As evidenced by Figs. 18 and 19, a 1.5-g acceleration was attained with a scram-assist pressure of 12 psig; maximum deceleration was 4.5 g.

C. Bellows Seal for Drive Shaft

1. Design Requirements

During normal operation, each of the twelve control rod drive shafts travels up or down over 24 in. at a constant velocity of 6 in./min. It may remain in one position for an extended period of time and, on occasion, may be accelerated downward (scram stroke), from any operating level, to effect rapid shutdown of the reactor. In all instances, the shafts must operate through seals designed to ensure unrestricted movement of the shafts

Fig. 17. Shock Absorber Test Assembly

without compromising the gastight integrity of the reactor vessel into which they penetrate. The operating environment in the vessel is a mixture of argon gas and sodium vapor at 1150°F, with the attendant hazard of sodium or sodium oxide deposition on the shaft surface.

2. Design Specifications

In view of the foregoing requirements, a welded, metallic bellows seal was selected because (1) the bellows were less complicated and more space-saving than other sealing methods that were considered, and (2) advancements in the art of fabricating welded bellows have made available extremely flexible and durable assemblies.

The outside diameter of the drive shaft ($2\frac{1}{2}$ in.) dictated a bellows of $2\frac{9}{16}$ -in. ID, thus providing an internally guided bellows system with a total diametral clearance of $1/16$ in. The outside diameter of the bellows ($4\frac{1}{2}$ in.) was selected to provide a "span" which is less than 30% of the OD recommended by a bellows fabricator. [NOTE: A span is defined as the width of a bellows disk or $1/2$ (OD-ID)].

Although the selected thickness of the disk material (0.010 in.) would provide a recommended minimum pitch of 0.030 in. between nested convolutions (pairs of disks welded at their ID), a pitch of 0.045 in. was chosen to allow greater latitude in adjusting the compressed height of the assembly. From past experience with EBR-II drive-shaft bellows seals and the addition of a 25% safety factor, it was assumed that a 0.125-in. stroke could be expected from each convolution. Thus with a nominal 25-in.

travel (including compression by the lifting platform) each bellows assembly comprised 200 convolutions or 400 disks.

Fig. 18 Scram Characteristics--Displacement vs Time

Fig. 19. Scram Characteristics--Velocity vs Displacement

Each disk was impressed with concentric ripples or waves to enhance the long-stroke capability and to reduce the spring rate of the assembly. Moreover, the free length of the assembly was made equal to 60% of the extended length. This placed the null point of the bellows near the center of the shaft stroke, thus dividing the total force accumulation due to the spring rate of the assembly.

Finally, the disks and end fittings were fabricated of Type 347 stainless steel because of its good welding characteristics and corrosion resistance.

3. Prototype Tests

At the time of termination of the FARET Project, a life-expectancy test of a slightly modified prototype bellows assembly was in progress. This assembly consisted of three bellows sections, each consisting of 82 convolutions as described in the preceding section, aligned with restraining rods to ensure uniform flexing of the coupled sections.

This assembly was being used to effect a dynamic seal between a simulated drive shaft and the nozzle of a resistance-heated vessel containing 1150°F sodium blanketed with low-pressure argon. The upper end of the bellows was fastened to the drive shaft which, in turn, was attached to a yoke. This yoke was supported at each end by ball-nut screws threaded onto respective lead-screws.

In operation, the lead-screws were rotated by a gearmotor, causing the yoke-mounted shaft to move vertically at 11 in./min, thereby expanding or compressing the bellows over a 36-in. stroke.

A reasonable life expectancy for this type of bellows is 10,000 cycles of such testing. Nonetheless, at the time of FARET Project termination, the bellows assembly had accumulated in excess of 17,400 cycles,

with no loss of gas pressure in the sodium container, no discernible deterioration of the bellows proper, and no significant impairment of performance due to sodium or sodium oxide deposition on the bellows and drive-shaft surfaces.

Plans are to continue the test until there is sufficient evidence to warrant disassembly and inspection of the prototype bellows seal.

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