MASTER

**CONTROL ROD DRIVE PROTOTYPE
TEST REPORT**

Submitted to
**U. S. ATOMIC ENERGY COMMISSION
NORTHERN STATES POWER COMPANY**
and
CENTRAL UTILITIES ATOMIC POWER ASSOCIATES

by

**ALLIS-CHALMERS MANUFACTURING COMPANY
ATOMIC ENERGY DIVISION
Milwaukee 1, Wisconsin**

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between

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under

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PATHFINDER ATOMIC POWER PLANT

CONTROL ROD DRIVE PROTOTYPE

TEST REPORT

DISTRIBUTION

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FOREWORD

One of a series of reports on research and development in connection with the design of the Pathfinder Atomic Power Plant, this particular report deals with the control rod drive experiments.

The Pathfinder plant will be located at a site near Sioux Falls, South Dakota, and is scheduled for operation in 1963. Owners and operators of the plant will be the Northern States Power Company of Minneapolis, Minnesota. Allis-Chalmers is performing the research, development, and design as well as being responsible for plant construction.

The U.S. Atomic Energy Commission, through Contract No. AT(11-1)-589 with Northern States Power Company, and Central Utilities Atomic Power Associates (CUAPA) are sponsors of the research and development program. The plant's reactor will be of the Controlled Recirculation Boiling Reactor type with Nuclear Superheater.

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1.0 INTRODUCTION

The Control Rod Drives of the Pathfinder Power Plant are designed to perform under rather unique operating conditions and it is desirable that they operate for a minimum of two years before requiring maintenance. Therefore, extensive testing was done to determine the compatibility of all components under expected design conditions.

Figure 1 shows a cutaway drawing of the Control Rod Drive Prototype Unit. Figure 2 shows a drawing of the assembled Control Rod Drive Production Unit.

2.0 OBJECTIVE

To evaluate the design of the control rod drive prototype unit under simulated Pathfinder reactor operating conditions.

3.0 CONCLUSION

The control rod drive prototype unit will perform satisfactorily under all Pathfinder operating conditions.

4.0 DESCRIPTION OF THE PATHFINDER CONTROL ROD DRIVES

The Pathfinder control rod drives are located on the reactor vessel head and operate in a pool of shielding water (Fig. 3). The shielding water temperature will be approximately 105 F during reactor operation, and the reactor vessel will be pressurized to 600 psig at 489 F. The drive rack housing extends 11-1/2 ft above the vessel cover support pipes. Electrical power is supplied to the drive motors through waterproof cables.

Figure 1...Control Rod Drive Prototype

SECTION C-C

NORMAL POSITION
MK 502
MK 503

15° POSITION.
MK 504

SECTION "D-D"
SCALE 1/2" = 12"
LATCH RING ROTATION

QTY REQ	CHK REQ	QTY REQ	QTY REQ	QTY REQ	QTY REQ	DESCRIPTION	MATERIAL	PART NUMBER DRAWING MK.	N°
1	-	-	-	-	1	CONTROL ROD DRIVE ASSEMBLY		43-500-938 501	
-	-	1	1	2	TERMINAL POST ASSEMBLY		43-500-951 501		
-	-	1	1	3	RACK HOUSING		43-500-938 501		
-	-	1	1	4	1/4 SWAGelok MALE CONNECTOR (900-12-3/6)		00101598		
YES	YES	YES	YES	5	ASSEMBLY PROCEDURE		43-101-600 401		
-	-	1	1	6	WATER ACCUMULATOR ASSEMBLY		43-500-939 401		
-	-	1	7	DRIVE MOTOR & LIMIT SWITCH ASSEM.		43-500-936 502			
-	-	1	1	8	ACCUMULATOR MOTOR HOUSING ASSEM.		43-500-919 501		
-	-	1	1	9	1/4 SWAGelok MALE ELBOW ("400-2-2-3/6")		00121348		
-	-	1	1	10	OIL FILL LINE		43-401-367 502		
-	-	1	1	11	NOTE SCREWED AIR VENT (WATERHEAD)	BRASS	00094724		
-	-	1	1	12	GASKET		43-500-936 001		
-	-	16	16	13	1/8 UNC-3A1 LG. HEX. SOC. HD. CAP SCREW		43-101-381 422		
-	-	16	16	14	1/2 MEDIUM SPRING LOCKWASHER		43-101-377 030		
-	-	1	1	15	LEAD CABLE		43-401-267 501		
-	-	1	1	16	PINION HOUSING ASSEMBLY		43-500-965 501		
-	-	1	1	17	WATER ACCUMULATOR PURGE TUBE		43-301-910 502		
-	-	16	16	18	1/2 MEDIUM SPRING LOCKWASHER		43-101-377 030		
-	-	16	16	19	1/8 UNC-2B HHX S.F. HEX. NUT		43-101-376 108		
-	-	16	16	20	1/8 UNC-2A1 3/16 HHX S.F. HEX. HD. BOLT		43-101-379 150		
-	-	1	1	21	1/8 UNC-2A1 3/16 HHX S.F. HEX. HD. BOLT		43-101-379 150		
-	-	1	1	22	DOWEL PIN		43-101-396 385		
-	-	1	1	23	STEAM SUPPLY TUBE		43-401-316 501		
-	-	1	1	24	1/4 SWAGelok MALE ELBOW ("400-2-2-3/6")		00121353		
-	-	1	1	25	SEAL VENT TUBE		43-401-364 502		
-	-	4	4	26 SHIM		43-301-681 514			
-	-	2	2	27 SHIM		43-301-681 516			
-	-	4	4	28 SHIM		43-301-681 518			
-	-	1	1	29	DOWEL PIN		43-101-396 523		
1	1	1	30	CONTROL ROD LATCH ASSEMBLY		43-500-926 501			
-	-	1	1	31	ADAPTER		43-202-288 502		
-	-	2	2	32 SHEAR PIN		43-202-089 502			
YES	YES	YES	YES	33	MARKING SPEC.		43-101-282 401		
YES	YES	YES	YES	34	CLEANING SPEC.		43-201-661 402		
-	-	1	1	35	UNFINISHED OIL - IN WATERPROOF - 1/4 PLATE RD	ACM 403	THIS IS	035	
-	-	16	16	36 1/8 UNC-2A1 3/16 HHX S.F. HEX. HD. BOLT		43-101-379 150			
-	-	1	1	37	DRIVE MOTOR & LIMIT SWITCH ASSEMBLY		43-500-936 503		
-	-	1	1	38	CONTROL ROD DRIVE ASSEMBLY		43-500-938 502		
-	-	1	1	39	NAME PLATE		43-202-571 001		
-	-	2	2	40 1/8 UNC-3A1 LG. HEX. SOC. HD. CAP SCREW		43-101-381 093			
-	-	2	2	41 1/2 MEDIUM SPRING LOCKWASHER		43-101-377 016			
1	1	1	42	CONTROL ROD DRIVE TAG (SEE TABLE 1)		43-202-575 001			
1	1	1	43	RUN & LATCH INDICATOR (SEE TABLE 1)		43-202-576 001			
-	-	45	45	NEOLUBE DRYFILM LUBRICANT (SEE NOTE 4)		THIS	044		
-	-	45	45	TEFLON PIPE THREAD SEALANT TAPE (SEE NOTE 4)		THIS	045		
-	-	64	64	VELOCITE "G" OIL (USE QO13047B)		THIS	046		
-	-	-	-	47 CONTROL ROD DRIVE ASSEMBLY		43-500-936 503			
-	-	-	-	48 CONTROL ROD DRIVE ASSEMBLY		43-500-936 504			
-	-	YES	YES	49 FLUID PENETRANT, INSPE.		43-104-173 401			
-	-	YES	YES	50 WELDING SPEC.		43-101-567 401			
-	-	108	108	51 WELD WIRE ASTM A371-53T CL ER 308L	ACM 1937	THIS IS	051		
-	-	2	2	52 CLAMP (MARINUM-AEROSQUIP #1120-5 1/2)		THIS IS	052		
-	-	45	45	WELD WIRE (INCONEL #82)		THIS IS	053		
-	-	1	1	54 OIL FILL CLAMP		43-101-780 001			
-	-	1	1	55 1/8 UNC-2A1 3/16 HHX S.F. HEX. HD. BOLT		43-101-379 150			
-	-	2	2	56 1/8 UNC-2A1 3/16 HHX S.F. HEX. HD. BOLT		43-101-379 150			
-	-	1	1	57 GASKET		43-202-936 002			
-	-	1	1	58 GASKET		43-202-936 003			

[illegible]

UNLESS OTHERWISE NOTED CONTAINING NO OTHER NOTED EXCEPT FOR THE FOLLOWING		PAGE 2	
REQUIRE	UP TO 2 INCH.	OVER 2 TO 4 INCH.	OVER 4 IN.
FUNCTIONAL	E	E	E
GENERAL	A	A	B

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3043

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**CONTROL ROD
DRIVE ASSEMBLY**

43-500-956

Figure 2A...Control Rod Drive Production Model

Figure 3...Location of Control Rod Drives on Reactor Vessel Head.

The 20 control rod drives are all basically of the same design, and are designated as outer boiler drives, inner boiler drives or superheater drives. The inner boiler drives differ from the rest in that the inner boiler drive motors have an output speed of 1/3 that of the others. The superheater drives differ from the boiler drives in that they will not be scrambled when an emergency shutdown occurs.

The drive mechanism is of the rack and pinion type (Fig. 1). The driving power is supplied by a 3-phase reversible induction gear motor with integral spur gear box and mechanical brake. The output of the motor is transmitted through a fail-safe combination backstop cam clutch, magnetic scram clutch, and drive shaft to miter gears which change the shaft axis 90° . The power is then transmitted by a pinion shaft through two rotary face seals, one high pressure and one low pressure, to a pinion that drives the control rod drive rack. The control rod drive rack, guided by a backup roller and a Stellite bushing, is latched directly to the control rod extension.

The magnetic scram clutch disconnects the drive shaft from the gear motor and allows the control rod, with rod extension and rack attached, to drop into the reactor for an emergency shutdown. Gravity with the aid of the scram spring drops the 164 lb boiler rod, the full stroke of 73 in. (including the dashpot) in about 1 second. The hydraulic dashpot surrounding the rack deaccelerates the rack and the control rod during the last 6 in.

The backstop cam clutch prevents reactor pressure from raising the control rod, should the pressure be lowered sufficiently in the rack housing. It does not, however, prevent the control rod from scrambling. The drive motor bypasses the backstop clutch. Thus, it can drive the control rod both up and down.

When scrambling, a void is formed in the rack housing due to a large portion of the rack leaving the housing. To fill this void with water quickly, a steam supply line in addition to the clearance between the rack and its guide bushing is provided. The control rod drive housing and steam supply line are not insulated from the cool shield pool water and as a result the steam from the reactor is rapidly condensed as it passes through the supply line to the upper end of the rack housing.

To eliminate a circulating steam path through the control rod drive, which would result in excessive heat losses to the shield pool, a spring loaded piston type accumulator is provided near the top of the rack housing. The bottom of the accumulator is vented to reactor pressure through the drive by the steam supply line. A water passage connects the upper end of the accumulator to the top of the rack housing. The piston of the accumulator forms a seal between the upper and lower connection points. On scrambling, when the rack is withdrawn from the rack housing, the pressure in the housing and upper end of the accumulator drops. Reactor steam on the lower side of the accumulator piston then pushes the piston up and forces the water from the

accumulator into the rack housing. After a scram, a spring forces the piston downward thus reducing the pressure in the rack housing slightly. This causes steam to flow through the rack bushing from the reactor. The steam condenses and allows the accumulator piston to return to its normal position. Sixty seconds are needed for rod and rack withdrawal and only 20 to 30 seconds are needed for the rack housing pressure to build up and provide pressure equalization. Therefore, the drive can be scrammed repeatedly without loss of scram speed.

A servo position transmitter is driven off the main drive shaft by a gear train and indicates the control rod position at all times. Limit switches also actuated by the gear train prevent over-travel of the rod in either direction, and provides, by means of lights on the control panel, an independent way of sensing control rod position.

The motor housing is filled with a transformer grade oil. The low pressure side of the rotary seal on the pinion shaft limits leakage of this oil from the drive housing. Any leakage past the seal goes into an alarm metered seal drain where it is monitored.

Reactor steam condenses in the rack housing which will fill with water at about 105 F and 600 psig. The stellite bushing, which surrounds the rack, acts as a seal to retard water leakage back into the reactor. The high pressure rotary face seal on the pinion shaft prevents leakage of the high pressure water from the rack housing. Any leakage past this seal face goes into the same metered seal drain that carries away oil leakage from the low pressure rotary face seal on the drive housing.

All electrical and other service connections to the control rod drives are made above the water level of the shield pool. The connecting lines are brought from the side of the shield pool to the drives by a hinged service platform.

The control rod is attached to the rack by means of a bayonet latch, which is locked or unlocked by rotating the round rack 45 degrees. Rotation of the round rack is accomplished by means of a special unlatching tool which rotates the turning tube and therefore the rack. While the reactor is operating, the turning tube is locked in place to prevent rotation of the rack. After a latching of the control rod has been attempted, the rack is driven in the Up Direction and the motor current is noted. If the control rod is latched, an increase in motor current over the unlatched condition will be noticed. Thus, the determination of whether or not a control rod is latched can be made.

When power is removed from the motor, the rack motor coast is less than 1/8 in. The motor is rapidly stopped by an integral mechanical brake acting on the low inertia rotor of the motor. Due to the speed limiting characteristics of the a-c motor, a dangerous withdrawal rate of the control rods can never be reached.

The control rod drive assembly is removable as a unit from the vessel cover. The operation can be performed without removing the vessel cover from the reactor provided that the reactor is shut down, depressurized and the vessel is flooded. The operation is performed

by means of a quick-disconnect mechanism. A cutaway sketch of this mechanism is shown in Figure 4. The upper shaft, (Item 1), is rotated using a special tool. Rotation of the upper shaft causes the lower shaft, (Item 2), which is pinned to the upper shaft to rotate. The actuator gear, (Item 3), which is keyed to the lower shaft (Item 2) also rotates, causing the ring gear, (Item 4), to rotate. The threads on the ring gear and control rod drive mounting flange (Item 4 and Item 5) are thus screwed together. This causes the Marman Aeroquip Conoseal Gasket (Item 8) to be compressed and provide an effective seal.

5.0 INDIVIDUAL TESTS

These tests were run on one or more components, which function together in the control rod drive unit, to determine and evaluate performance independently of all other components of the control rod drive unit.

5.1 Control Rod Rack Housing Heat Transfer Test

This test was made to determine the heat loss to the shield pool, when different types of internal water supplies were used. This water supply is necessary for proper scram operation of the control rod drive. The test setup for this test is shown in Figure 5.

The autoclave simulates the reactor, and the 3/8 in. tube (Fig. 5) represents one of the proposed methods of supplying steam to the top of the rack housing. Scramming is simulated by opening the solenoid valve (Fig. 5) which allows some water to drain. This water drainage causes steam to enter the loop.

Figure 4...Quick-Disconnect Mechanism

Figure 5...Control Rod Drive Rack Housing Heat
Transfer Test Setup

The heat loss was evaluated by measuring the flow rate and temperature rise of the water passing through the cooling jacket. It was found that any method of supplying water by an open direct connection between the rack housing and the reactor vessel allows excessive heat loss to the shield pool.

This test setup also allowed various types of bushings to be tested to determine if a suitable design could be achieved whereby steam would pass through the bushing into the rack housing, condense and remain there without significant water flow back into the reactor. (the rack passes through this bushing). All types tested resulted in excessive heat loss to the shield pool.

Further testing revealed that a set of steam baffles between the bushing and the drive pinion housing effectively reduced the heat loss to the shield pool. It was found that an axial baffle spacing of 1.5 in. gives a minimum heat loss of between 1.1 and 1.4 kw per drive. This steam baffle design was incorporated into the final design of the control rod drive. The data from this test has been plotted, and is shown in Figure 6.

5.2 Drive Motor and Clutch Tests

These tests were performed to determine if the motor and clutch would meet the specifications for application in the control rod drive.

The characteristics to be determined for the motor were: starting time, stopping time, reversal time, a current characteristic which could be used to indicate, by ammeter, when a control rod was latched

Figure 6...Effect of Baffle Spacing on Heat Loss
to the Shield Pool

or unlatched from the rack, and construction to permit operation submerged in transformer oil. The characteristics to be determined for the clutch were: fast release time and proper operation in transformer oil. The test setup is shown in Figure 7. Preliminary tests in air on a single-phase induction motor with an integral brake using a torque wheel and weights to simulate load conditions, indicated that this motor will meet the requirements for starting, stopping, reversal time, and current difference between loadings. Tests also proved the motor to be satisfactory for operation in transformer oil.

Testing of a Gearmaster, type P35/14P, three-phase gear motor with a Stearns H42 integral brake showed that its current difference between no-load and load was more suitable, than that of a single -phase motor, for determining whether or not a control rod was latched. The data for the three-phase motor operating in oil, was:

		No-Load	240 in-lb Load
Starting time (milliseconds)	up	140-150	240-280
	down	140	130-140
Stopping time (milliseconds)	up	150	170
	down	150	230
Reversal time (milliseconds)	up to down	145-165	170
	down to up	150	230
Motor rpm	up	1640	1240-1265
	down	1650	1770
Output rpm	up	15.5	12.0
	down	15.6	16.6
Power-total (watts)	up	95	170
	down	90	70
Line Current (aver.) (amperes)	up	1.13	1.3
	down	1.12	1.15

Figure 7...Drive Motor and Clutch Test Setup

A servomotor and double worm gear reducer were also tested. However, this type of drive was rejected due to excessive coast in the Down Direction and teeth failure of the worm gear reducer.

The data showed that the Gearmaster motor was satisfactory for operating in oil and was chosen for the production units.

A 3-3/4 in. diam Fawick magnetic clutch was initially tested in oil. This revealed clutch slippage at 280 in-lb of torque. The release time was 13 to 14 milliseconds. Later testing in oil of a 4-1/2 in. diam Fawick clutch revealed no clutch slippage at 650 in-lb torque. This clutch has a release time at 240 in-lb of torque of 17 to 18 milliseconds, which is satisfactory. Thus, the 4-1/2 in. diam clutch was chosen for the production units.

5.3 Latch Tests

The purpose of these tests was to determine if a control rod could be remotely latched and unlatched many times in both the cold and the hot condition, determine if the materials were compatible, and determine the maximum offset and tilt that the latch could withstand and still operate properly. The test setup is shown in Figures 8 and 9. After a total of 800 trouble free cycles, the latch was considered satisfactory from both the materials and operations standpoint.

The results of the latch test with respect to compatibility of the materials were as follows:

Figure 8...Latch Test Setup

Figure 9...Latch Test Setup

1. No appreciable wear of any components.
2. As expected, slight galling occurred at the contact surfaces between the lower plunger and the latch adapter.

The lower plunger was made of Inconel "X", and the latch adapter was made of Type 304 Stainless Steel. This galling did not increase the torque required to turn the latch assembly while latching or unlatching.

The maximum offset that the latch could withstand was $3/8$ in. That is, the rod could move $3/8$ in. from its centered position and the latch mechanism would still operate properly. The maximum tilt of the rod that the latch could withstand and still operate properly was 0 degrees and 7.7 minutes. Both the maximum offset and tilt that the latch can withstand are greater than those calculated for the rod in the reactor.

5.4 Dashpot Test

The dashpot in the control rod drive is used to decelerate the rack and control rod at a constant deceleration for about the last six inches of travel during a scram. It was decided to test the dashpot separately, due to the difficulty of measuring the internal pressure of stopping action in the full assembly. Also, in the assembled control rod drive, it is impossible to determine if the dashpot returns to the Up Position after a scram. Therefore, a sufficient number of cycles were run to insure that no galling or sticking would occur, since failure of the dashpot to return could cause damage to the control rod drive and the control rod on the next scram.

The parabolic dashpot surface required for a uniform deceleration was approximated by two straight-line tapers for improved fabrication. The test setup is shown in Figure 10. The instruments that were used for this test were:

1. a high-speed motion picture camera,
2. a Baldwin SR-4 pressure cell.
3. strain gages,
4. a Tektronic oscilloscope with camera.

The pressure cell was used to measure the internal pressure of the dashpot. The strain gages were tangentially adhered to the lower region of the dashpot casing, and one was placed on the upper region of the ram. The high speed camera was used to reveal the stopping action of the dashpot. The outputs of the strain gages and the pressure cell were fed to the oscilloscope, where pictures were taken of them.

In the set up, an extra inch of travel was allowed for the weight over the amount of travel allowed in the drive, but the piston in the dashpot was prevented from bottoming. This extra travel made it possible to analyze the velocity with which the control rod will hit the grid plate with greater accuracy than if motion were stopped at the expected point of impact.

First tests showed a peaking of the pressure and strains in the dashpot shortly after impact, which was about three times the average values. Modifications in the taper minimized this peaking.

Figure 10...Dashpot Test Setup

The pressure peaks, as the ram started its stroke, were caused by the separation of the ram and the falling weight.

Two different weights were used in this test; a 227 pound and a 303 pound weight. Calculations from the pressure curve indicated that the velocity at the grid plate would be about 42 in/sec for the 303 pound weight and about 22 in/sec for the 227 pound weight.

The dashpot stuck after 48 cycles due to a metallic deposit inside the outer casing. Examination showed a number of shallow axial scratches on the upper guide of the ram. The deposit was removed, and the dashpot operated adequately after reassembly.

After six more scrams, the dashpot stuck again, and a metallic deposit and more scratches were observed. The cause of dashpot sticking was a piece of metal which had become lodged between the guide and the outer casing. Machining on the outer casing after it has been chrome plated left the chrome edge rough, so that a piece of chrome got into the dashpot. The taper on the upper guide of the ram has reduced binding and friction. The following modifications were made:

1. The inside of the outer casing was honed to remove the metallic deposit.
2. A 1/32 in. taper was put on the lower end of the cylinder to allow for easier insertion of the ram and piston ring.
3. The outside diam of the upper guide on the ram was turned with a slight taper added which removed all scratches and any indication of binding or friction.

After 30 more cycles, no additional signs of binding appeared.

The conclusions drawn from this test are as follow : The velocity at the bottom of the stroke, that is where the control rod hits the grid plate, is at most 22 in/sec. This is for a velocity of 230 in/sec at which the control rod is traveling, at the time the dashpot starts to function. This is quite satisfactory. However, the velocity 1/4 in. above this is nearly double. Therefore, due to manufacturing inaccuracies, the dashpot taper has been reduced to 5-1/2 in. rather than the full 6 in.

A metallic piston ring was found to be the best seal for the ram. A rubber O-ring did not allow satisfactory return of the ram to the Raised Position after a scram. The ring manufacturers recommend a brass ring instead of the cast iron type used in this test, since cast iron would not be satisfactory for long periods in water.

The maximum stress, that was set up in the dashpot casing, was 18,800 psig using the 227 pound weight and 20,300 psig using the 303 pound weight. These stresses are well below the yield strength of the material. With the 303 pound weight, the velocity at the bottom of the stroke, that is where the control rod hits the grid plate, is about 42 in /sec. This velocity is too high. But, since the combined weight of the control rod, the latch, and the rack is 230 pounds, this dashpot design is satisfactory as is.

5.5 Quick-Disconnect Mechanism Test

This test was performed to determine suitable materials and thread lubricants for the quick-disconnect mechanism which is used to produce

a seal with a Marman Aeroquip Conoseal Gasket #50887-400-S. It was desirable to limit the torque to a value of approximately 100 ft-lb. The test setup is shown in Figure 11.

Hydrostatic testing at 1250 psi showed the coupling to be leak-tight. The coupling was then added to the control rod drive prototype assembly. The threads on Item 4 and Item 5 in Figure 4 were lubricated with a copper-in-oil colloid. The metallic conical gasket was seated with 150 ft-lb torque. The mechanism operated satisfactorily for the entire test.

In a separate test, a quick-disconnect with silver plated threads was tested, and this combination worked well with a torque up to 300 ft-lb. Therefore, it has been chosen as the reference design.

A locking device prevents the quick-disconnect from loosening except when turned through the drive shaft by a special tool. The locking action is obtained by means of a helical spring inserted into a housing with a slight interference fit. The locking device spring can be rotated in the direction which tends to wind it and thus decrease its diameter, but cannot be rotated in the opposite direction which tends to unwind and increase its diameter.

The locking device cannot release due to vibration since there is interference at all times between spring and spring housing. When the gear shaft tries to back-off it is immediately stopped because of this interference. The spring must be wound up slightly in order to tighten the quick-disconnect coupling and is under a precompression

Figure 11...Quick-Disconnect Test Setup

to hold this position. No trouble due to vibration was experienced during testing of the drives.

The original design utilized a Graphitar bushing with a Type 431 Stainless Steel shaft, Item 6 and Item 2 respectively in Figure 4. When the assembly was submerged in demineralized water, the stainless steel began to rust and pit, as did the PH 15-7 MO snap ring, which was in contact with the Graphitar. The same was noticed when Type 416 Stainless Steel was placed against Graphitar in demineralized water. Further tests showed that an Inconel "X" spring housing, Item 7 in Figure 4, which acted, also, as a bushing for the Type 431 Stainless Steel shaft, caused galling on the shaft.

In a separate test, an Oilite bushing and spring housing, which replaced the original ones, were used and the results showed that the Oilite bushing was adequate for the required torque and did not induce pitting or rusting when submerged in water with stainless steel. Thus, this design was incorporated into the production model.

5.6 Rotary Seal Test

The purpose of this test was to determine materials for the rotary Stein Seal #1110-P4 for the pinion shaft, which seals the high pressure water in the rack housing and the low pressure oil in the motor housing from atmosphere. These materials must limit the leakage rate to a small value for the design lifetime. The test setup is shown in Figure 12.

Figure 12...Rotary Seal and Bearing Test Setup

A test was set up to simulate the conditions under which the seal is to operate , except for temperature. It was felt that the 120 F operating temperature would have little effect. Therefore, the tests were run at room temperature.

Using an aluminum oxide water-side rotor against a carbon face, the leakage rate was 21 cc/hr after five days of accelerated testing (equivalent to one-third of its design lifetime).

Using a tungsten carbide water-side rotor, the leakage rate was reduced to about 1 cc/hr for the same test period with high-, medium-, and low-pressure biased carbon faces. Therefore, the tungsten carbide water-side rotor was selected as the reference design. Chrome plated 17-4 PH was used for the oil-side rotor.

5.7 Bearing Test

This test was performed to find a suitable radial and thrust bearing for the pinion shaft, which is to be run in a very thin oil with a viscosity of 56 SSU at 100 F. The test setup used was the same as that used for the rotary seal test and is shown in Figure 12.

Deep-groove steel bearings with brass and steel retainers and an angular contact steel bearing with a steel retainer were tested and failed due to high thrust loading and poor lubrication. These bearings had a 40 mm outside diameter and 17 mm bore. A larger angular contact bearing with a 47 mm outside diam, a 17 mm bore, and a steel retainer was tested and found to handle the thrust loading satisfactorily.

5.8 Control Rod Drive Pinion Housing Hydrostatic Test

This test was performed to determine if any elastic deformation would occur in the control rod drive pinion housing since the stresses in many regions could not be determined analytically. The pinion housing is shown in Figure 13.

The pinion housing was hydrostatically pressurized to 1300 psig. There was no indication of plastic deformation. Therefore, the design was deemed satisfactory. Stress coat was used to determine regions of highest stress and the magnitudes were measured using strain gages.

6.0 ASSEMBLED TESTS

In this phase of testing, the object was to test all component parts of the control rod drive as they function as a unit under simulated reactor conditions such as the full power reactor temperature and pressure, and submergence in the shield pool. A control rod was also used in conjunction with these tests to double check the latching and unlatching capability of the drive unit and, also, for scram purposes.

The prototype drive is shown mounted on the hot test pressure vessel in Figure 14.

6.1 Preliminary Tests

Prior to the assembled control rod drive tests the following preliminary tests were made.

Figure 13...Control Rod Drive Pinion Housing

Figure 14...Control Rod Drive Prototype Mounted
on Hot Test Pressure Vessel

6.1.1 Position Indicator System

After the first assembly of the unit, it was placed on a stand and the Vard synchro-driven position indicating system was checked for accuracy. Using one position receiver the accuracy was within $\pm 3/8$ in. (usually within $\pm 1/4$ in.). Using two position receivers, the accuracy was reduced to $\pm 1/2$ in., and the pointer moved erratically. Thus, the one-receiver system was chosen for the hot tests.

6.1.2 Cold Test

The preliminary testing involved running of the drive unit cold while it was bolted to the hot test pressure vessel and hydrostatically pressurized to 700 psig.

Scram times with this solid water system were measured at 3.5 sec without an accumulator and 1.5 sec with an accumulator.

During these tests, the 3-3/4 in. magnetic clutch slipped, and it was replaced with a 4-1/2 in. clutch.

Latching, unlatching, and locking of the rack in the Up Position were performed repeatedly with success. With the control rod latched or unlatched, a motor current difference was observed.

6.1.3 Hot Test

This preliminary testing involved running of the drive unit on the hot test pressure vessel with the vessel hot and pressurized (489 F and 600 psig.).

During this testing, a leak was present in the steam supply tube for the scram accumulator. This caused the control rod to drop for 36 in. and slowly coast the rest of the stroke, when the drive was scrammed.

When the accumulator was removed from the drive, the control rod would drop about 10 in. before slowing down. The scram time without an accumulator was measured to be about 5 sec.

6.2 Assembled Hot Tests

The assembled hot tests provided data to evaluate the following items under full power reactor temperature and pressure:

- Operating life of the drive

- Drive motor and clutch characteristics

- Heat transfer rates to the shield pool

- Seal leakage

- Vibration or instability

- Scram time

- Dashpot operation

- Position indication

This testing was done in five phases, which are described below.

6.2.1 Phase I

In this test phase, 2400 complete 72 in. cycles and 200 scrams from various heights were performed with the drive.

This is equivalent to at least 20 years of normal operation.

These tests showed that the following design requirements were met.

The latching and unlatching operations were performed successfully many times. The motor current difference between the latched and unlatched conditions was satisfactory (0.04 amp, which is about 4 percent of full load current). The synchro-position indicator worked very smoothly and accurately. The dashpot functioned properly even though particles of black iron oxide were later found inside. The rack teeth wear was minor. The clutch performed well, and no wear was observed.

The following features were found to require improvements:

1. An early design high-pressure rotary shaft seal leaked excessively. This type utilized a chrome-plated 17-4 PH stainless steel high pressure rotor. Its leakage rate reached a maximum of 1500 cc/hr. To correct this, another early design, high-pressure rotary seal was incorporated into the drive.
2. The steam supply tube did not provide sufficient steam to the accumulator for a full 72 in. scram at pressures above 200 psi. To correct this, the lower steam supply tube was increased from 3/8 to 1/2 in. in diam.
3. The limit switches went out of adjustment several times. The unit was modified, using more rugged actuators, followers, and cams to make the switches less sensitive to temperature changes.

4. Three Type 420 Stainless Steel snap rings rusted and broke during this testing; one in the high-temperature, high-pressure steam environment, and two in the low-temperature, high-pressure water region. Other snap rings in the steam region were pitted and about to break. These rings were replaced by PH 15-7 MO rings.
5. The pinion wore about 0.040 in. on the driving side of the teeth. This may have been caused by running it with an inaccurately machined rack. All of the teeth on the rack were not parallel, which reduced the contact area and caused a highly loaded area at different points on the pinion teeth.

6.2.2 Phase II

This phase of testing consisted of 3600 complete 72 in. cycles and 250 scrams. The results and observations of this testing follow below.

Prior to this phase of testing, a successful hydrostatic pressure test at 1250 psi showed that there would be no problem with face seals and O-rings, when the reactor vessel is hydrostatically tested. The new, more rugged limit switch actuators and cams, which had been installed, worked well making the limit switches very sensitive to cam movement but not sensitive to temperature changes.

Another early design, high pressure rotary seal failed, giving a leak rate of about 6000 cc/hr.

Several Type 420 Stainless Steel snap rings in the low-temperature region broke during this testing, while the PH 15-7 MO snap rings appeared to be sound. Thus, in the third phase, all 420 snap rings were replaced by PH 15-7 MO rings.

The clutch slipped occasionally when the test temperature was raised. It was concluded that the Graphitar bushing added as a rack roller backup was overloading the clutch, since the clutch was found to be properly adjusted. The function of the rack roller is to guide and backup the rack at the point where the pinion driving force tends to separate the pinion and rack. Normally the roller contacts the smooth surface of the rack. However, at the end of the stroke where the teeth are cut completely around, the rack can move up and down or bounce on the roller as it passes. To eliminate the possibility of rack or roller damage at this point, the Graphitar bushing was added between the pinion and rack housing just above the rack roller. However; since the Stellite number 3 rack roller did not seem affected by the bounce-back at the end of the stroke, the Graphitar bushing was permanently removed.

Modifications of the steam supply tube, which were made to decrease the scram time, consisted of a 1/2 in. diam internal tube and a 1-5/8 in. diam external tube. The external tube had a short

length of 5/8 in. diam tube formed in a loop at the lower end to allow for differential expansion. Using this arrangement, the scram time, including the dashpot time, was 1.1 sec at 600 psig and 489 F with a steam system and 1.4 sec at 600 psig and 80 F with a solid water system. After this phase of testing, the steam supply system was further modified to include dual 1/2 in. diam internal steam supply tubes to decrease scram time and 3/4 in. diam expansion loop.

The rack and pinion used in Phase I were also used in this Phase. The rack teeth were considerably rounded due to wear, but it was deemed satisfactory to enter Phase III of testing. The pinion was used in this test to determine whether or not it can be run without chrome-plating, since all of the chrome had been removed from the worn areas of the teeth. After this phase of testing, the pinion teeth were found to be worn. Approximately 2/3 of the tooth thickness was lost through wear.

6.2.3 Phase III

One thousand full 72 in. cycles were performed during this phase. The results and observations of these tests follow:

With the dual 1/2 in. diam internal steam supply tubes, the scram time was 1.1 sec at 600 psi and 120 F with a solid water system and 1.0 sec at 600 psi and 489 F with a steam system. These scram times included dashpot travel of about 0.1 sec.

The heat loss rate to the shield pool was determined experimentally.

While the drive is operating, 5 kw are lost to the shield pool; and while the drive is static, only 1 kw is lost.

The top of the rack housing was intentionally misaligned in various directions up to 3/8 in. to determine if the rack locking surface at the top of the housing, which locks the rack when the drive is removed from the vessel cover, could possibly stop the rack from scrambling during normal operation. Scrams and normal runs were made without incident.

High motor currents were recorded during the test and were probably due to the worn condition of the pinion. The pinion showed additional wear after this testing phase. Since it was about 3/4 through the teeth it was decided to discard this pinion and rack and not make use of them for further testing.

6.2.4 Phase IV

This phase of testing consisted of 1100 full 72 in. cycles, using a new pinion and a new rack. The new pinion was a spare which originally was heat treated at 900 F for 4 hours. The chrome-plating was not replaced. The hardness of the teeth was 31 Rc after this second heat treatment. The heat treatment specification for 17-4 PH was altered (from a range of 1050-to-1075 F) to (a range of 1090-to-1110 F) for 4 hours on the recommendation of the steel supplier, to improve stress corrosion resistance. The new rack was machined straight within 0.0025 in.

with all teeth parallel within 0.002 in. After a heat treatment at 1100 F for 4 hours, the rack was straight within 0.0065 in. Its hardness was 32 Rc.

At the end of the test, the rack and pinion were both worn badly. The pinion teeth were worn half through, and large burrs surrounded the contact area. The rack teeth edges were found to be rounded from wear.

At the conclusion of this phase, the drive failed to scram due to the jamming of the rack and pinion caused by excessive wear. This was the first and only time the drive would not scram. Also, at the end of this phase, clutch slippage was noted and undoubtedly was due to overloading caused by worn gears. The badly leaking seal was caused by metal chips from the gears.

From the results of Phases I, II and III, which used a 17-4 PH pinion whose teeth surfaces were not entirely covered by the chrome plating and an inaccurately machined rack, it was assumed that, if the chrome-plating had adhered satisfactorily and if the rack had been properly machined, the pinion would have been satisfactory. Thus, the chrome-plated 900-F heat treated 17-4 PH pinion was chosen as the reference design and used for tests in Phase V.

During operation of the drive with transformer oil, rust formed on the bearings. Two oils with rust inhibitors were tested. With Mobil DTE 797 turbine oil, the motor speed dropped off, and the motor current increased. This oil was not satisfactory. Mobil Velocite No. 6 oil, which has the same viscosity as transformer oil; 56 SSU; was tested and the drive motor operated satisfactorily and no rusting was observed. Therefore the Velocite No. 6 oil was selected for use in the motor housing.

6.2.5 Phase V

In this phase, a 900 F heat treated and chrome-plated 17-4 PH pinion shaft was tested with a spare Type 304 Stainless Steel rack, since a 17-4 PH rack was not immediately available. The hardness of the Type 304 Stainless Steel rack was 22 Rc, which was considered sufficiently close to the hardness of the 1100-F 17-4 PH rack to provide a valid tests of the pinion.

The test was run for 13,000 full 72 in. cycles. and 150 scrams. This proved that the reference design for the pinion, 900-F 17-4 PH with chrome-plating on the teeth, was satisfactory, since the wear of the pinion teeth was negligible. The chrome-plating adhered extremely well. The leading edges of the teeth had slight chips removed. These chips consisted of base metal as well as chrome-plating. Otherwise there was no wear on the pinion.

Since, during Phase IV, the rotary shaft seal failed due to chips getting into the seal, a seal baffle was added to the drive unit to keep chips and dirt out of the seal. This baffle functioned as expected during this phase of testing .

During this phase of testing the following observations were made in regard to the motors. The stator of the slow speed motor was changed to a weaker one, because the motor with the stronger stator became saturated. Thus, no current difference was noticed between the latched and unlatched conditions. Also, on both the high- and the slow-speed motors, the voltage applied to the motor had to be the rated voltage or slightly less, otherwise the motor would become saturated and no current difference could be noticed between the latched and unlatched conditions.

6.3 Motor Brake and Back Stop Clutch Test

This test was performed on the drive assembly to determine if the Stearns motor brake and the Morse PB-6 back stop clutch would keep the rack from going up into the rack housing under high loading of the rack. The Morse back stop clutch positively locks to transmit torque in one relative direction, but it will overrun freely in the other direction. Its action is like a self-contained ratchet with an infinite number of teeth. This high loading on the rack may occur in the event that an accident would cause the pressure in the rack housing to drop to near atmospheric pressure. In this case, the high differential pressure on the ends of the rack would create a high loading on the rack

itself, tending to push the rack out of the reactor. This loading could be as high as 600 lb. The motor brake and back stop clutch would have to prevent the rack from moving upward, when the magnetic clutch is disengaged.

The test was set up so that the high axial forces could be applied to the rack for various lengths of rack withdrawal. The forces were applied with a hydraulic jack.

The maximum force applied was 2500 pounds. With this force applied, the back stop clutch jammed so that the rack could not be scrambled but could be run either up or down. The clutch and motor brake did not fail or slip. Therefore, this back stop clutch was considered to be satisfactory and was used on the production drives.