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CONTROL-DRUM DRIVE
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AEC Research and Development Report

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ABSTRACT

The SNAP 2, 10A, and 8 nuclear systems are designed to provide electrical power on space vehicles. Although each system has different power capabilities, the control-drum-drive components are basically the same.

Development of the various control-drum-drive components and subcomponents from GFY 1962 to mid-GFY 1964 is discussed. The design criteria are outlined for each SNAP system. Comparisons are made between the systems to evaluate their design and operation in the prescribed environment.

The components and subcomponents reviewed are the actuator, controller, bearings, gears, and flexible couplings. Data are presented to describe development of the SNAP 10A reference design, and both the reference and backup designs for the SNAP 2 and SNAP 8 systems.

In addition to the integrated control-drum-drive reference designs, four possible backup design schemes are also presented for SNAP 2 and SNAP 8. Each scheme is discussed in terms of structural and mechanical design, thermal heating, and nuclear shielding considerations. The concept of a direct-drive actuator mounted above the drum is promising, and further study is planned.

A control-drum-drive scheme that provides failsafe scram capabilities for ground testing of the reactor system has been designed and proof tested. The details of this design as it applies to SNAP 8 systems are presented.

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Note: This report, sponsored chiefly by the SNAP 8 program, covers studies performed for and jointly funded by the SNAP 10A, 2, and 8 programs. The SNAP 2 reactor has been superseded by the Interim 10A/2 reactor.

I. INTRODUCTION

The SNAP 2, 10A, and 8 nuclear reactors are designed to provide electrical power on space vehicles. All the reactors are liquid-metal cooled (NaK), and uranium-zirconium hydride fueled, although each employs a different power conversion system. The reactors are controlled by rotating the external beryllium reflectors to increase or decrease the fraction of leakage neutrons. The reflectors are semicylindrical drums that surround the reactor core vessel and rotate toward the core, or away, as required to increase or decrease neutron reflection into the core.

The drum-drive system is composed of an electromagnetic actuator that rotates the drums either directly or through a gear train upon command from a programmed controller. The system is designed to meet the startup and operational requirements of each reactor throughout the operational life of 10,000 hours or more.

This report discusses the drive-system development that has been conducted on the various components and subcomponents in order to evaluate the adequacy and status of the total effort. The development effort covers the period from FY 1962 to mid-FY 1964 and compares present thinking with the ideas originally published at the start of the effort. The development of all components and system concepts has been improved by the knowledge gained during the effort. This has been due, in part, to laboratory results, design improvements, and necessary changes made to obtain compatibility between components.

Figures 1, 2, and 3 illustrate the SNAP 2, 10A, and 8 drum-drive systems, respectively. The SNAP 2 and 10A are each four-drum systems with two "coarse" drums and two actuator-driven "fine" drums. The SNAP 8 system has six drums, three coarse-type and three actuator-driven. In all cases, the coarse drums are spring-loaded and rotate to the full-in position upon telemetry command to their release actuators. At this point, the reactor is still subcritical, and inward rotation of the fine control drums by the actuator is required to achieve criticality and subsequent power.

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Figure 1. SNAP 2 Reactor Control-Drum Drive System

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Figure 2. SNAP 10A Reactor Control-Drum Drive System

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Figure 3. SNAP 8 Reactor Control-Drum Drive System

7568-5539B

Various drive systems have been considered for each SNAP system. Table 1 is a summary comparing the various concepts evaluated and pointing out current development status. For SNAP 10A only the reference design is included since the final design for this system has been fixed and new concepts are not required. Columns 1 through 5 in the table compare major characteristics and their effects upon each other. The comparison for each concept is arbitrarily based on the SNAP 2 reference design as 100. Thus, numbers higher than 100 are comparatively better than SNAP 2, and numbers lower represent a less desirable design.

Columns 6 through 10 are status and information listings comparing each drum-drive concept. The maximum calculated thermal level is shown in column 6. The criterion for grading here is that the lowest temperature is the most attractive feature.

Columns 7 and 8 relate the current development status and the approximate time remaining to complete each program. The program most complete and having the least time remaining to complete scores the highest in overall evaluation. The completion times are based on past performance and estimated times determined from present knowledge of the various problems to be solved.

Column 9 shows relative tolerance of the drum-drive systems to dust, sand, and dirt environments. All design concepts are rated "poor" here since the bearings, brakes, and other moving parts would be equally affected. Column 10 takes into account the effect of distortion due to temperature transients on bearing and gear tolerance. The strength of materials under the respective environmental conditions is also considered in this analysis.

TABLE 1
COMPARISON OF SNAP REACTOR DRUM-DRIVE SYSTEM DESIGN CONCEPTS

Design Concept	Actuator Location	System	(1) Reliability*	(2) Structural Design Simplicity*	(3) Ease of Installation *	(4) Ease of Actuators Shielding Scatter*	(5) Degree of Simulation of Ground Tests to Flight Conditions*	(6) Thermal Levels (Calculated Maximum) (°F)	(7) Development Status (Complete) (%)	(8) Time to Complete Including Tests	(9) Tolerance to Dust, Sand, and Dirt	(10) Tolerance to Thermal Distortion or Other Strain and Material Strength
Indirect-Drive Actuator (Reference Design)	Above Drum	SNAP 2 (Figure 1)	100	100	100	100	100	775	60	1 yr	Poor	Poor
		SNAP 10A (Figure 2)	95	100	100	90	75	675	90 [†]	6 mo	Poor	Poor
		SNAP 8 (Figure 3)	100	100	90	100	100	950	70	2 yr	Poor	Fair
Direct-Drive Actuator	Directly Above Drum (without flexible coupling)	SNAP 2	110	140	130	60	100	500	30	3 yr	Poor	Fair
		SNAP 8	120	140	130	70	100	950	30	3 yr	Poor	Good
	Offset Above Drum (flexible coupling)	SNAP 2	115	120	120	100	100	500	30	3 yr	Poor	Good
		SNAP 8	115	120	120	100	100	950	30	3 yr	Poor	Good
	Below Drum	SNAP 2	105	70	90	100	75	800	30	3 yr	Poor	Fair
		SNAP 8	80	80	90	100	75	1100	30	3 yr	Poor	Good
	Below Shield	SNAP 2	90	40	60	-	75	500 to 800	20	3 yr	Poor	Poor
		SNAP 8	90	20	60	-	75	~600	20	3 yr	Poor	Poor

*Assume SNAP 2 current design as 100

[†]For 90 days only

II. DESIGN CRITERIA

In general, the design criteria are similar for all three drum-drive systems, SNAP 2, 10A, and 8. The orbital startup action for each system is initiated by ground command signal which activates the controller and releases the locking devices on the control drums.

The coarse control drums snap in, reducing the amount of subcriticality, and the fine control drums then start their rotation inward. The stepping rate for each system is different due to differences in drum configuration and reactivity rate required. In each case, however, the actuators step the drums one-half degree per step. For SNAP 2 and 10A the two fine drums are stepped together while SNAP 8, having three fine drums, steps the drums in sequence. In the S2 and S8 systems, the drum rotation rate is decreased as the reactor becomes critical and temperature or heat generation is established. When the reactor power level reaches the desired heat generation, negative reactivity is inherently introduced by the temperature coefficients of the grid plates and fuel, thus reversing the power transient. After the initial power transient, there is a gradual increase in power with corresponding rises in system temperatures and coolant flows. The process continues until the reference outlet coolant temperature is reached, at which time the controller is signaled by the control temperature switches to stop power to the control-drum actuators.

When the outlet temperature of the reactor drops below the reference temperature, the controller is signaled by the control temperature switches and the actuators step the drums inward to again insert reactivity. The SNAP 10A reactor will normally operate in a passive mode (no control actions) throughout its lifetime. The system is designed for short-term (72 hours) automatic control after reactor power has been established. At this time the controller is commanded off and the drum-drive system remains inactive for 90 days. The controller can be reactivated by ground command and the reactor outlet temperature may be increased above the setpoint by a manual override if desired. The temperature-override capability is employed to compensate for converter degradation and unexpected events.

The SNAP 2 and 8 reactors are designed to maintain automatic control for one year after startup. Thus, the controller and control-drum actuator are in constant operation as required to hold the reactor outlet temperature in the setpoint temperature band.

The control-drum drive motors for each system are designed to impart a nominal 0.5° step rotation to the control drums. The allowable step size is determined by the maximum differential worth of a control drum. Table 2 itemizes the basic design criteria for each drive system. Tables 3 and 4 outline the shock and vibration test requirements for the three drum-drive systems.

TABLE 2
DESIGN CRITERIA – SNAP REACTOR DRUM-DRIVE SYSTEMS

Design Criteria	Drum Drive System		
	SNAP 2 (125 kw)	SNAP 8 (600 kw)	SNAP 10A (43 kw)
Maximum temperature range ($^\circ\text{F}$)	750	900	650
Drum rotation per step during startup ($^\circ/\text{step}$)	0.5	0.5	0.5
Drum rotation per step during operation ($^\circ/\text{step}$)	0.5	0.5	0.5
Cumulative error allowable ($^\circ/\text{step}$)	± 0.1	± 0.1	± 0.1
Dwell time between steps during startup (sec)	100	2.5	150
Dwell time between steps during operation (sec)	400	~ 120	150 (1st 72 hr only)
Number of control drums	4	6	4
Number of actuator (fine) controlled drums	2	3	2
Maximum allowable drum bearing friction (in. -lb/drum)	2.5	2.5	2.5
Control required	Automatic constant temperature setpoint	Automatic constant temperature setpoint	Automatic constant temperature setpoint
Radiation	$1.5 \times 10^{19}\dagger$	$1 \times 10^{20}\dagger$	$1.5 \times 10^{19}\dagger$
	$3.9 \times 10^{10}\S$	$3 \times 10^{11}\S$	$1.5 \times 10^{10}\S$
Minimum torque required to control drum bearing (per drum) at temperature (in. -lb) ϵ^4	5.0	12.5	10.0
Storage life before launch (yr)	2	2	2
Design life (hr)	10^4	10^4	10^4
Shutdown	Ground command, end of life, or reentry	Ground command, end of life, or reentry	Ground command, end of life, or reentry

*Average for actuator

$\dagger$ Fast neutrons (nvt)

$\S$ Gamma (rads)

**Includes safety factor

TABLE 3
SNAP 2 AND 10A SHOCK AND VIBRATION
TEST REQUIREMENTS

Test	Description	Evaluation
Vibration	<u>Sinusoidal Sweep</u> A vibration test shall be applied along each of three mutually perpendicular axes, at the vibration amplitudes specified below. The test along each axis shall consist of a single sinusoidal sweep starting at the lower frequency limit and proceeding at a constant octave sweep rate to the upper frequency limit in not less than 45 min	After each axis sweep the drum lockout pin shall be withdrawn and the simulated drum shall be rotated by hand to test free motion
	<u>Longitudinal Axis</u> 5 to 20 cps at 1/4-in. double amplitude 20 to 200 cps at 5 g 200 to 400 cps at 5 g 400 to 3000 cps at 7.5 g	
	<u>Lateral and Normal Directions</u> 5 to 13 cps at 1/4-in. double amplitude 13 to 200 cps at 3 g 200 to 400 cps at 5 g 400 to 3000 cps at 7.5 g	
Shock	Subject the assembled shake test fixture to two (2) shocks in each direction of the three mutually perpendicular axes for a total of twelve shocks. Each shock shall be 20 g and of ~6-millisecond duration in a half-sine wave form.	Upon completion of the shock testing, the drum lockout pin shall be withdrawn and the simulated drum shall be rotated by hand to test free motion

TABLE 4
SNAP 8 SHOCK AND VIBRATION
TEST REQUIREMENTS

Test	Description	Evaluation
Vibration	<u>Sinusoidal Sweep</u> A vibration test shall be applied along each of three mutually perpendicular axes, at the vibration amplitudes specified below. The test along each axis shall consist of a single sinusoidal sweep starting at the lower frequency limit and proceeding at a constant octave sweep rate to the upper frequency limit in not less than 45 min.	After each axis sweep, the drum lockout screw shall be withdrawn and the simulated drum shall be rotated by hand to test free motion
	<u>Longitudinal Axis</u> 5 to 12 cps at 1/2-in. double amplitude 12 to 400 cps at 3.5 g 400 to 3000 cps at 7.5 g	
	<u>Lateral and Normal Axes</u> 5 to 10 cps at 1/2-in. double amplitude 10 to 250 cps at 2.5 g 250 to 400 cps at 5.0 g 400 to 3000 cps at 7.5 g	
Shock	The fixture shall be subjected to two shocks in each direction along each of three mutually perpendicular axes for a total of twelve shocks. Each shock shall be 20 g and of ~6-millisecond duration in a half-sine wave form.	After each axis orientation the drum lockout screw shall be withdrawn and the simulated drum shall be rotated by hand to test free motion.

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III. COMPONENTS

The basic components comprising the drum-drive system are the actuator, controller, bearings, gears, and flexible couplings.

The actuator is the prime mover in the system, supplying the torque required to rotate the control drum reflector, as well as the braking force to hold the drum to accuracies of 0.5° per step. The controller supplies the electrical signals to the actuator thus controlling its direction, rate, and order of stepping action. The bearings play an important part in the actuator, the gears, and the control drum by providing low friction and high accuracy during the lifetime of the drum-drive system. Although only the control drum bearings are discussed here, the material and bearing tolerances are typical of the other drive system components. The gears transmit the actuator torque to the control drum and, in addition, provide a mechanical advantage to the actuator output shaft. The flexible coupling is employed to obtain high torsional transmission through a nonrigid shaft.

The flexible coupling designs apply only to SNAP 8. In the SNAP 8 design, due to a higher operating temperature, it was desirable to move the actuator away from the core vessel as far as possible without its protruding outside the shield shadow cone and without compromising drive torque. Several design concepts are discussed.

A. ACTUATOR

The purpose of the actuator is to position the drum reflectors as required by the signal from the controller. The actuators under development are electromagnetic devices in the form of stepping motors and brakes. Development objectives include production of a unit of low weight, low power consumption, and high reliability. The actuator must also provide an accurate output position as required by the reactor control circuit.

At the start of the first SNAP program (SNAP 10A), there were no commercially available actuators of the type desired that would function reliably for 10,000 hours in the described space environment. The high neutron and gamma flux, the high steady-state temperature and temperature gradients, and the high vacuum of outer space required new technology in rotary actuator design. It was necessary to develop new insulating materials for coils and lead wires.

TABLE 5
CURRENT STATUS OF PERMANENT MAGNET ACTUATOR TESTING

Design	Status	Total Hours	Vacuum (Torr)	Temperature (°F)	Insulation Resistance to Ground (ohms @ 45 volts)		Torque (oz-in)		Remarks
					Motor	Brake	Clockwise	Counter-Clockwise	
SNAP 10A Reference Design (AI-B30-8P)	Completed	9,000	2.2×10^{-5}	800	35K	10K	14	14	Completed 9 000 hr quite successfully, further testing at 1000°F showed slight Cu migration resulting in loss of torque and of resistance to ground
SNAP 10A with Commercial Insulation (Westinghouse 3002)	Terminated	7,881	2×10^{-5}	1000	5K	-	14 4 [†] 25 6 [§]	16 0 [†] 25 6 [§]	Test terminated due to high-vapor pressure materials lead, boron, and silica plating out and reducing the resistance to ground
SNAP 10A Reference Design (AI-B30-6P)	Terminated	1,200	3×10^{-5}	840	79.9K	5K	4.8	4.8	Test terminated due to low resistance, inspection showed rotor to have been hitting stator because of excessive bearing clearance. Probable initial cause: excessive bearing clearance.
SNAP 10A with Commercial Insulation (Westinghouse 3001)	Terminated	5,500	4×10^{-6}	1000	17K	-	12 8 [†] 19 2 [§]	12 8 [†] 19 2 [§]	Test terminated due to problems with the test fixture rather than with the actuator itself. Also, certain high-vapor pressure materials, lead, boron, and silica were plating out and reducing the resistance

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† @ 17.5 volts

§ @ 28 volts

New bearings to minimize self-welding and seizure were also required. High-temperature materials that had previously been satisfactory were now marginal. In general, the state of the art in actuator design was behind the SNAP reactor requirements.

Several different actuator types have been studied for use on SNAP drum-drive systems. Each new design represented improvements on the concepts of the preceding design. These improvements resulted from development testing, reliability studies, and continuous design review.

Thus far, five actuator types have evolved from study effort on the SNAP 2, 10A, and 8 drum-drive systems. These types are discussed as follows:

1. Indirect Drive

- a. Permanent Magnet in Rotor (S10A Reference Design)

The actuator with a permanent magnet in the rotor (see Figure 4) consists of a stepper motor and brake. The essential parts are: a stator with eight salient poles, its associated winding and encapsulating system; a rotor which consists of a permanent magnet, slotted endcaps, shaft and journals; and two end-bell assemblies with bearings. The brake is a more or less conventional axial-operating spring brake which releases by magnetic pull when electrically energized. Thus, when the actuator is energized, the brake is "off" (no braking force). When the actuator is not energized, the brake is "on" and prevents the rotor from turning.

The actuator is not conventional in that special materials are used for environmental operation, such as high-temperature insulation (glass-covered copper wire and encapsulant), carbon graphite and aluminum oxide bearings, and aluminum oxide braking surfaces. Magnetic parts are chrome-plated. Titanium structural parts are used for weight-saving and low neutron scatter. The motor operates on a synchronous-reluctance principle made possible by the magnetic alignment of uneven numbers of rotor and stator slots. The "bifilar" winding makes it possible to operate the motor in either direction from a single-ended power supply. The rotor steps 1.8° of arc per pulse. The current status of permanent magnet actuator testing is shown in Table 5.

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Figure 4. Indirect-Drive, 1.8° Step Actuator, Including Partial Assemblies—Permanent Magnet in Rotor (SNAP 10A Reference Design)

b. Solid Rotor (S2DRM-1 and S8DRM-1 Reference Design)

The solid rotor actuator (Figure 5) is an improved design of the SNAP 10A actuator. Several major design changes were made in the rotor, stator, and brake. The rotor is a one-piece unit without the permanent magnet. The rotor slots are milled "in-line" in contrast to the SNAP 10A actuator, in which the slots are misaligned by one-half the pitch of the slot. These changes enable the rotor to operate more reliably at the higher temperature and also eliminate many parts such as endcaps, holddown nuts, lock pins, etc., thus making manufacturing easier and less costly. The end result is a more accurate and reliable rotor assembly, lighter in weight.

The stator is changed from a bifilar winding with two coils per pole as in the SNAP 10A actuator to a simple winding with but one coil per pole. The

Figure 5. Indirect-Drive, 1.8° Step,
Solid-Rotor Actuator (S2DRM-1
and S8DRM-1 Reference Design)

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winding consists of glass-covered, chrome-plated copper wire with encapsulant. The stator is simpler, easier to wind, and hence more reliable. More gross copper can be used, thus reducing heating losses as well as power consumption.

The brake is redesigned and incorporates the same electrical constituents as the stator but is "wet-wound."* It also has a new type of flexure and optimized design configuration to meet the applicable requirements. A disassembled view of this actuator is shown in Figure 6.

In general, the solid-rotor actuator possesses more torque, greater accuracy, and higher reliability at lower power consumption than the SNAP 10A actuator.

c. Wet-Wound Solid Stator (S2DRM-1 and S8DRM-1 Backup) – Figure 7

This actuator is the same as the solid-rotor actuator except that the stator, as well as the brake, are "wet-wound." The stator is "solid" (machined), rather than being a stack of punched laminations as in the SNAP 10A and solid-rotor actuators. The winding is inserted from the outside diameter rather than from the inside as in the previously described actuators. Each coil is inserted as a "wet-wound" unit.

*Wet-winding refers to the technique of applying a liquid ceramic insulation to a single wire as the wire is wound into a coil. The coil is subsequently oven-cured to solidify the insulation.

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Figure 6. S8DRM-1 Solid-Rotor Actuator, Disassembled

With this method, further improvement in design performance is possible. Reliability is greatly increased chiefly because a wet winding has a higher insulation resistance capability in the order of magnitude of two decades over the vacuum-encapsulated dry winding.

2. Direct-Drive, $1/2^\circ$ Step

a. Reluctance Stepper (S2 and S8 Backup Design)

The one-half degree stepper is an advanced design actuator (see Figure 8). This device will utilize the materials previously described for the insulation systems, conductors, bearings, etc. However, the principle of operation is different from the aforementioned devices.

The device is essentially a reluctance stepper which takes a $1/2^\circ$ step per pulse. The "angle of advance" is provided by a discrete, asymmetrical

placement of the stator pole pairs rather than the usual mismatch of rotor and stator slots. In this design, the theoretical number of rotor and stator slots are the same. An electromagnetic brake is incorporated into the device as an integral unit. It is inherently more reliable than the other devices as it is directly coupled to the control drum and thereby eliminates the need for a gear train. The proof-of-principle model has already been assembled and bench-tested. Preliminary results are quite promising and the principle is proven to the sound.

b. Linear Rotary-Reversible Stepper (S2 and S8 Backup Design)

Another advanced concept is the linear actuator (Figure 9). Several designs have been analyzed and the best of the various concepts have been incorporated into a single design. This is a rotary-reversible stepping actuator which will produce a discrete angular step upon command signal, and hold the control drum fixed under no-power conditions or when no command signal is applied. This design also eliminates the requirements for a gear and pinion.

The actuator is designed to produce step size in the 0.5° range which can be set by internal adjustments and physical stops. Flexural pivots are used to eliminate friction and backlash. The envelope size required for this design is less than the SNAP 10A or SNAP 8 stepper actuator and, due to its flat shape, it can be mounted either above or below the control drum well within the radiation shield cone. The electrical control is simple, using only three magnet solenoids to produce the stepping motion.

Preliminary data from bench tests of the developmental model have shown that design improvements are necessary to ensure the performance goals of step accuracy and torque output.

3. Materials

Considerable research was conducted on high-temperature magnetic materials that could be used in these actuators. For the SNAP 10A and solid-rotor actuators, silicon-iron electrical grade steels such as No. 5 Relay and M-36 are used. These steels have adequate strength, are low in cost, and machine well. As a finishing operation the parts are coated with chromium to deter oxidation. Their magnetic properties are satisfactory up to 1300°F but suffer somewhat reduced capability at the higher temperatures.

Hiperco 27, a cobalt-iron alloy steel, are used in the $1/2^\circ$ stepper and linear actuator. This material has a higher temperature capability and

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Figure 7

Indirect-Drive, 1.8° Step
Actuator (SNAP 2DRM-1
and SNAP 8DRM-1
Backup)

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Figure 8. Direct-Drive, $1/2^\circ$ Step
Actuator (SNAP 2 and SNAP 8
backup)

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Figure 9.

Direct-Drive, $1/2^\circ$ Step
Linear Actuator (SNAP 2
and SNAP 8 backup)

allows for a higher magnetic flux level than the previously mentioned materials at the high temperatures. It also appears that plating is not required as protection against oxidation.

The electrical insulation used to encapsulate the coil windings, referred to as PBX, is composed of silica, alumina, and traces of chromium oxide as solids in a dilute phosphoric acid mixture. This system, when cured, results in a molecular bonded SiO_2 and Al_2O_3 coating that has proved to be satisfactory for SNAP 10A operating temperatures. However, a superior insulation, designated as SAP II, has been developed which shows better performance at any of the temperature levels and has a higher temperature capability. The SAP II can be used up to 1600°F and will be employed as design requirements dictate.

Both the PBX and SAP II insulations have a tendency to absorb moisture. Coils that have absorbed atmospheric moisture may show a low insulation resistance and cause delay in acceptance testing. However, the actuator still operates reliably and as the unit heats up from continued use the moisture is driven off, bringing the insulation resistance back to normal. The net result is that the controller may see a slightly higher current during initial operation. In any case, under vacuum and/or temperature this type of insulation gives up the absorbed moisture readily.

Tables 6, 7, and 8 summarize the various actuator design concepts in terms of functional, structural, operational, and performance features.

B. CONTROLLER

The function of the controller is to perform reactor startup and control during the lifetime of the system by properly sequenced signals to the control drum actuators. The direction, rate, and order of stepping action is controlled by the state of the input signals to the controller.

Various controller designs can be implemented according to the needs of the control system. Certain designs will be more complex than others depending upon the performance requirements. Basically the three SNAP drum-drive systems will operate in the following manner. Upon receipt of a startup signal, the controller steps the actuators at a preset rate. For SNAP 10A the actuators will be signalled to step once every 150 sec. There will be no change in step rate

TABLE 6
DESIGN COMPARISON OF SNAP ACTUATORS

		Indirect-Drive, 1 8° Step Actuators				Direct-Drive Actuators	
		Permanent Magnet (S10A)	Solid Rotor		Solid Stator	1/2° Step	1/2° Step, Linear
			(S2DRM-1)	(S8DRM-1)	(S2 and S8 Backup)	(S2 and S8 Backup)	(S2 and S8 Backup)
Design Considerations	Principle		Magnetic reluctance stepper				Solenoid mechanical contact stepper
	Permanent Magnet	In rotor			None		
	Electromechanical Brake				Yes		
	Weight (lb)	4 5	5 5	5 5	5 5	7	5
	Maximum diameter (in)	3.5	3 5	3 5	3 5	5	5
	Maximum length (in)	7	8 125	6	6 25	2 25	2
	General configuration		Cylinder			Pancake	
	Step size (°)	1 8	1 8	1 8	1 8	0 5	0 5
	Phase-winding pulse sequence§§	AB BC CD DA	A B C D	A B C D	A B C D	A B C D	B AB AC C
	Cost estimate (1 = lowest)	1	2.0 [†]	2.5 [†]	1	3	3
	Manufacturing ease at present time (1 = easiest)	3	1	1 [§]	2	4	5
Materials	Magnetic	Silicon steel and Almco	Silicon steel			Hiperco	
	Electrical	Glass-covered copper	Glass-covered chromed copper				
	Insulation	PBX [*]	SAP II ^{††} (brake wet-wound)			SAP II (stator and brake wet-wound)	
	Bearings	Carbon graphite and Al ₂ O ₃	Carbon graphite and Al ₂ O ₃		Carbon graphite and Al ₂ O ₃	Carbon graphite and titanium	Carbon graphite and Al ₂ O ₃
	Brake surfaces		Al ₂ O ₃ and chrome		None	Carbon and titanium	Al ₂ O ₃ and Inconel-X
	Structural	Titanium	Inconel-X		Titanium		Inconel-X

*Approximate
†Includes Inconel-X parts
§Except for bottom end bell

**Commercially available high-temperature encapsulant
††AI-developed high-temperature encapsulants
§§A, B, C, and D refer to phase-windings for pairs of poles in clockwise sequence

TABLE 7
COMPARISON OF TOLERANCE OF SNAP ACTUATORS TO
ENVIRONMENTAL AND OPERATING CONDITIONS

Component	Low Temperature Operation	Oxidizing Atmospheres	Sand, Dust, and Other Particles	Endurance in Space Environment	Alignment	Expected Thermal Distribution	Shock and Vibration	Humidity
Bearings	Meet design criteria	Marginal	Marginal	Preliminary data satisfactory	Poor	Preliminary data satisfactory	Preliminary data satisfactory	Preliminary data satisfactory
Electrical parts	Meet design criteria	Meet design criteria	Marginal	Indirect-drive, permanent magnet actuator meets design criteria, testing incomplete on other models	Meet design criteria	Preliminary data satisfactory	Indirect-drive permanent magnet actuator meets design criteria, other models not tested	Indirect-drive permanent magnet actuator meets design criteria, other models not tested
Magnetic parts	Meet design criteria	Protective coating required	Marginal	Preliminary data satisfactory on indirect-drive permanent magnet actuator, testing incomplete on other models	Poor on direct-drive linear actuator, all other models meet design criteria	Meet design criteria	Indirect-drive permanent magnet actuator meets design criteria, other models not tested	Meet design criteria
Mechanical and structural parts	Fair on direct-drive linear actuator, all other models meet design criteria	Meet design criteria	Poor on direct-drive linear actuator, all other models meet design criteria	Indirect-drive permanent magnet actuator meets design criteria, testing incomplete on other models	Poor on direct-drive linear actuator, all other models meet design criteria	Preliminary data satisfactory on indirect-drive, permanent magnet and solid stator, and direct-drive 1/2° step actuators, poor on direct-drive linear model, solid rotor results inconclusive	Indirect-drive permanent magnet actuator meets design criteria, results poor on direct-drive linear model, other models not tested	Not tested

Note Remarks apply to all models, except where specifically mentioned.

TABLE 8
PERFORMANCE COMPARISON OF SNAP ACTUATORS

		Indirect Drive, 1 8° Step Actuators				Direct-Drive Actuators	
		Permanent Magnet (S10A)	Solid Rotor		Solid Stator (S2DRM-1 Backup)	1/2° Step (S8 or S2 Backup)	1/2° Step, Linear (S8 and S2 Backup)
(S2DRM-1)	(S8DRM-1)						
Accuracy	Step size accuracy limitation	Dependent on load torque					Independent of load torque (unless starting torque is exceeded)
	Sand and dust, and other contaminants	Unaffected by differential growth					Affected by differential growth
	Alignment during operation due to stress, etc.	Bearing clearance affected					
	Accuracy with respect to unforeseen high-load torque not in excess of starting torque	Moderate accuracy	High accuracy				Affected
	Tolerance to mis-alignment	Moderate	Low				High
Torque and Power	Starting torque at temperature	15 oz-in.	20 oz-in. (calculated)	20 oz-in.	30 oz-in. (calculated)	720.0 oz-in.	480.0 oz-in.
	End of step torque (assuming zero step error)	0	0	0	0	0	Extremely high
	Torque at room temperature (starting)	35 oz-in.	40 to 50 oz-in (calculated)	40 to 50 oz-in.	50 to 60 oz-in.	900.0 oz-in.	600 oz-in.
	% Optimum design (based on starting torque at temperature)	20	25	25	30	75	75
	Electrical power at temperature	24 watts	20 watts (calculated)	20 watts	28 watts	26 watts	24 watts
	Electrical power at room temperature	70 watts	60 watts (calculated)	60 watts	80 watts	75 watts	70 watts
	Power required for 100% design (based on starting torque at temperature)	100 watts	90 watts	90 watts	90 watts	60 watts	60 watts
	Differential growth manifestations	Bearing clearance affected	Not affected	Bearing clearance affected			Bearings and solenoid clearance affected
Stepping Output	Nominal step size	1.8°	1.8°	1.8°	1.8°	1/2°	1/2°
	Step size reduced by combination pulse	No	0.9°	0.9°	0.9°	1/4°	No
	Step size changed by mechanical adjustment only	No					Infinitely variable within a narrow band
	Step size changed by re-design in same envelope	Possible	Limited				Incremental increase, larger step acceptable, smaller step limited
	Power required if step size changed by redesign	Same					Smaller step, less power, larger step, more power
	Recommendation for smaller step by redesign	No	Possible				Yes (with limit 0.4°)
	Recommendation for larger step by redesign	No	Yes				Yes, will require more power and larger package

throughout this system's lifetime. The SNAP 2 system, however, is controlled at one step every 100 sec during startup and then "shifts gears" to one step every 400 sec as required during operation. In the SNAP 8 system the controller steps the actuators at a fast rate, one step every 2.5 sec, during startup. This controller is designed to have several intermediate rates, each of which will be set as required during startup until the reactor outlet temperature enters the dead-band for the first time. The controller then changes to the long-term rate of one step every 120 sec and steps the actuators at this rate in the appropriate direction as signaled by the temperature-sensor switches.

For all three reactor systems, the long-term control drums must maintain their positional relationship regardless of the direction of movement. For SNAP 8, the controller must provide the memory function to assure that the right control drum is actuated in response to the low or high signal from the temperature sensor. The controller is then the "heart" of the drum-drive system and must function accurately and reliably throughout the reactor operational life of 10,000 hours or more.

In the development of the various drum-drive system controllers, several schemes were studied. In preliminary design it was necessary to make assumptions as to basic requirements of the controllers and then evaluate several designs based on complexity. For these purposes the following assumptions were made:

- 1) Environmental stresses will have the same proportionate effect on all designs regardless of complexity.
- 2) Inputs to the controller are step inputs from both temperature-sensor switches and rate-changing switches.
- 3) The actuator used is the indirect-drive, 1.8° step, solid-rotor actuator.
- 4) Increased power requirements due to increased complexity are not considered.

For this analysis the controller can be broken down into three basic blocks as shown in Figure 10. These may be defined as follows.

1. Time Base Generator

The time base generator is a source of pulses which trigger the sequencer. These pulses appear whenever the High or Low switches indicate nondeadband conditions. The delay period between pulses is determined by the input from the rate change switches. The time base generator contains a basic clock plus divider stages to divide the clock frequency down to frequencies of the order desired, e.g., for SNAP 8 operation, 1 cycle every 120 sec. Rate changes are accomplished by switching the divider stages either in or out or by changing the clock frequency.

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Figure 10. Controller Block Diagram

2. Sequencer

The sequencer is triggered periodically by the time base generator and, in turn, drives the power switches. Each time the sequencer is triggered it generates a sequence — according to the state of the High and Low switches — which is then directed to the power switches for the appropriate actuator. The sequencer consists of a clock which sets the stepping rate, and a bidirectional shift register which when triggered generates a step sequence and directs the steps to the appropriate power switches.

3. Power Switches

The power switches receive their drive from the sequencer and, in turn, switch high-current, unregulated, d-c power to the actuators. This unit may utilize either relay or transistor switches.

For the purposes of this discussion, arbitrary "complexity factors" will be assigned to the various blocks shown in Figure 10. Primary attention will be focused on the sequencer as the largest variety of changes could occur there depending upon the number of steps taken per sequence and the number of actuators which must be alternately driven. (See Figures 11 through 17.) Table 9 lists assigned complexity factors for each of the three blocks, time base generator, sequencer, and power switches. It should be emphasized that the "complexity factors" assigned are completely arbitrary, and that the weighting between the three basic blocks may be out of proportion.

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Figure 11. Sequencer — Any Number of Actuators in Parallel,
Four Steps per Sequence

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Figure 12. Sequencer — Two Actuators Driven Alternately,
Four Steps per Sequence

TABLE 9
COMPLEXITY FACTORS OF CONTROLLER BLOCKS

Block	Actuators				Complexity Factor
	Number	Drive Method	Steps per Sequence	Rate Change	
Time Base Generator	-	-	-	0 (120-sec period)	10
	-	-	-	1 } Rate changes	12
	-	-	-	2 } related by	14
	-	-	-	3 } binary factors	16
	-	-	-	1 } No binary	15
	-	-	-	2 } relationship	20
	-	-	-	3 }	25
Sequencer (See Figures 11 through 17)	Any number	Parallel	4	-	10
	2	Alternately	4	-	20
	Any number	Parallel	2	-	12
	Any number	Parallel	1	-	16
	3	Alternately	1	-	48
	3	Alternately	4	-	30
Power Switches	6	Alternately	4	-	60
	1	-	-	-	5
	2	-	-	-	10
	3	-	-	-	15
	6	-	-	-	30

To illustrate how these tabulations might be employed, a comparison is given in Table 10 of a controller which meets the current SNAP 10A requirement vs controllers meeting the S8DRM-1 requirement and possible SNAP 2 and SNAP 8 requirements. Review of Tables 9 and 10 shows that the controller unit becomes more complex with any increase in the number of operations required.

TABLE 10
COMPARISON OF CONTROLLER REQUIREMENT
COMPLEXITY FACTORS

System	Actuators				Complexity Factor			
	Number	Drive Method	Steps per Sequence	Rate Change	Time Base Generator	Sequence	Power Switches	Total
SNAP 10A	2	Parallel	4	0	10	10		30
SNAP 2	2	Alternately	4	1 (Binary related rate change)	12	10	10	42
S8DRM-1	3	Alternately	4	3 (Rates related in pairs; i.e., 125 sec and 40 sec, 30 sec and 120 sec)	19	30	15	64
SNAP 8	6	Alternately	4	3 (Related in binary pairs)	19	60	30	109

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Figure 13. Sequencer — Any Number of Actuators in Parallel,
Two Steps per Sequence

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Figure 14. Sequencer — Any Number of Actuators in Parallel,
One Step per Sequence

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Figure 15. Sequencer — Three Actuators Driven Alternately, One Step per Sequence

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Figure 16. Sequencer — Three Actuators Driven Alternately, Four Steps per Sequence

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Figure 17. Sequencer — Six Actuators Driven Alternately,
Four Steps per Sequence

The SNAP 10A design utilizes the technique of all-transistor logic. The logic and time-base generation is performed with transistorized logic modules and the output switching is done with relays. The internal frequency source or time base is a transformer-coupled multivibrator. This design has been proven in long-term environmental tests, irradiation tests, and bench tests.

The SNAP 2 design is similar to SNAP 10A except for time sequences, output switching, and power requirements. A breadboard has been constructed using S10A design principles and components, and tests are underway to evaluate performance.

The SNAP 8 controller design utilizes the technique of core-transistor logic. This approach allows a reduction in the total number of transistors required as compared to straight transistor logic techniques. The cores and windings which replace the transistors are inherently more reliable and more radiation-tolerant. The core-transistor logic modules perform all the sequencing and timing functions within the controller. Relays perform the power switching to the actuators.

The overall reliability of the controller will be determined by the relays, since they are more sensitive to radiation. Radiation screening tests have been initiated in an attempt to find a power transistor which can replace the relay.

The SNAP 2 and 8 backup controller is patterned after the SNAP 10A unit in that it utilizes the technique of all-transistor logic. Components, circuit designs, and test results from the 10A program have been utilized to improve the reliability of the original design. This approach requires more semiconductors than the SNAP 8 reference design; however, it has proved highly successful in the SNAP 10A application.

Advanced designs are being developed to provide design information for possible utilization in controllers with increased tolerance to the imposed environment and increased reliability over current designs. Two advanced logic concepts are under study. The first is the all-magnetic logic approach and the second utilizes a family of ceramic components known as TIMM's* which are based on the tiny heaterless ceramic vacuum tube concept. Both advanced designs being developed are inherently capable of high temperature and radiation tolerance.

*Thermionic Integrated Micromodules

C. CONTROL-DRUM BEARINGS

The SNAP 2, 8, and 10A bearings under development are similar to the journal type except that the journal bushings are held within spherically shaped cartridges to provide self-alignment. The reflector drum bearings discussed herein are characteristic of all other SNAP component bearings. The typical design configuration is shown in Figure 18.

SNAP 2 and SNAP 10A drum bearings are located outside the shield shadow cone and therefore must be made from a material with a low neutron scattering cross section. The SNAP 10A bearings will operate at temperatures of approximately 700°F, and the SNAP 2 bearings will operate at approximately 900°F. Titanium was chosen as the structural material for SNAP 2 and SNAP 10A application because it has a good strength-to-weight ratio at temperatures up to 900°F and has a relatively low neutron scattering cross section.

The SNAP 8 drum bearings are located within the shield shadow cone and will function at temperatures above 1100°F which is above the efficient service temperature of titanium. The reflector support bracketry in the SNAP 8 design employs attachment supports with wide spacing centering at the interface between the support hardware and the beryllium reflector. Thus, because of the large

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Figure 18. Control Drum Bearing Set

TABLE 11
SNAP DRUM BEARING DESIGN DATA

System	Design	Component		Material	Diametral Clearance (in.)			Total Diametral Clearance (in.)			
					Room Temperature	Operating Temperature		Room Temperature	Operating Temperature		
						Radial	Thrust		Radial	Thrust	
SNAP 10A	Reference Design	Shaft		Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)	0.0019 to 0.0029	0.0018 to 0.0028*	0.0018 to 0.0028*	0.0024 to 0.0044	0.0027 to 0.0047*	0.0028 to 0.0048*	
		Ball		K-162B							
		Socket		Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)							
	Backup	Shaft		Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)	0.0019 to 0.0029	0.0021 to 0.0031*	0.0025 to 0.0035*	0.0024 to 0.0044	0.0030 to 0.0050*	0.0040 to 0.0060*	
		Ball Assembly	Bushing	P-5N Graphite							
			Ball	K-162B							
Socket		Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)									
SNAP 8	Reference Design	Shaft		Al ₂ O ₃ , LA-2 Plated Inconel-X	0.0012 to 0.0020	0.0006 to 0.0014†	0.0008 to 0.0016†	0.0013 to 0.0029	0.0013 to 0.0029†	0.0016 to 0.0033†	
		Ball Assembly	Bushing	K-162B							
			Ball	Al ₂ O ₃ , LA-2 Plated Inconel-X							
		Socket		Al ₂ O ₃ , LA-2 Plated Inconel-X							
	Backup	Shaft		Al ₂ O ₃ , LA-2 Plated Inconel-X	0.0020 to 0.0028	0.0034 to 0.0044†	0.0041 to 0.0049†	0.0021 to 0.0037	0.0037 to 0.0055†	0.0046 to 0.0062†	
		Ball Assembly	Bushing	CDJ-83 Graphite							
			Ball	Al ₂ O ₃ , LA-2 Plated Inconel-X							
		Socket		Al ₂ O ₃ , LA-2 Plated Inconel-X							
	SNAP 2	Reference Design	Shaft		Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)	0.0009 to 0.0019	0.0011 to 0.0021§	0.0015 to 0.0025§	0.0014 to 0.0034	0.0020 to 0.0040§	0.0026 to 0.0046§
			Ball Assembly	Bushing	P-5N Graphite						
Ball				K-162B							
Socket			Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)								
Experimental "X"		Shaft		Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)	0.0005 to 0.0013	0.0002 to 0.0010§	0.0001 to 0.0009§	0.0010 to 0.0028	0.0021 to 0.0039§	0.0021 to 0.0045§	
		Ball		P-5N Graphite							
		Socket		Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)							
Experimental "Y"		Shaft		Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)	0.0005 to 0.0013	0.0005 to 0.0013§	0.0005 to 0.0013§	0.0010 to 0.0030	0.0015 to 0.0036§	0.0018 to 0.0038§	
		Ball Assembly	ID Bushing	P-5N Graphite							
			Ball	Ti(6Al-4V)							
			OD Bushing	P-5N Graphite							
		Socket		Al ₂ O ₃ , LA-2 Plated Ti(6Al-4V)							
Experimental "Z"		Shaft		Tungsten	0.0005 to 0.0015	0.0005 to 0.0015§	0.0005 to 0.0015§	0.0010 to 0.0030	0.0010 to 0.0030§	0.0010 to 0.0030§	
		Ball		P-5N							
		Socket		Tungsten							

*at 700°F

†at 1100°F

§at 850°F

Note: Al₂O₃ is flame-sprayed LA-2, K-162B is a sintered TiC + Ni, P-5N is Purebon carbon graphite.

temperature range of the system, a bracket and bearing material with a thermal expansion rate very close to that of beryllium is required to minimize thermal stresses in the reflector and support hardware. Inconel-X was chosen for the SNAP 8 bearings and support bracketry because it has a high strength-to-weight ratio at 1100°F and a thermal expansion rate very close to that of beryllium over the temperature range from 70 to 1100°F.

Low-friction materials to be used as surface coatings with the bearing structural materials were selected from sliding block samples which had been tested at vacuum below 10^{-8} Torr at more severe loading and wear than will be required in actual application. Material combinations of mechanical grades of carbon graphite (CDJ-83 and P-5N) and Al_2O_3 are used for the primary friction surfaces. Sodium silicate bonded dry-film lubricants are being considered for the secondary all-ceramic friction surfaces. Bearing material combinations and clearances of various bearing surfaces are shown on Table 11.

1. Design Tests of Reflector Bearings

Prototype bearings were fabricated and subjected to a series of tests at temperature and vacuum under design loads. The upper and lower drum bearing assemblies were mounted into a simulated drum and cycled periodically. The drum simulation provided the bearing loads typical for each system. The actuation stroke rotated the shaft $\pm \sim 85^\circ$. The frictional reaction of the bearings was opposed by a buckout rod coupled to a force transducer and associated recording equipment. The assembled test fixture is shown in Figure 19. Note that the lower end of the bearing shaft is eccentric by 0.030 in., thus imposing a stringent misalignment condition for test purposes.

The environmental tests were conducted in an oil diffusion pump vacuum chamber. The test equipment is shown in Figure 20. The environmental temperature was simulated by radiant heating from quartz lamps located inside the vacuum chamber. This permitted the upper and lower bearing couples to operate at different temperatures in close simulation of the reactor environment. The bearing temperatures during the tests were set at the drum-drive operating requirements for SNAP 2 and 10A. Due to equipment limitations, SNAP 8 maximum temperatures were not simulated. Temperatures were mapped and in the various tests to date ranged from 505 to 960°F. Test vacuum pressures ranged from 2×10^{-6} to 2×10^{-7} Torr and test duration times ranged from 100 to 2300 hours.

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Figure 19. Drum Bearing Test Fixture

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Figure 20. Test Fixture Suspended Into Vacuum Chamber

Nine prototype bearing sets tested at operating temperatures revealed drum maximum torques of 0.05 to 1.1 in.-lb. The test results are shown in Table 12. Although some of the bearing sets showed wear primarily on the thrust faces, wear was not of a magnitude to measurably change clearances. The bearing sets using K-162B against Al_2O_3 showed maximum torques from 0.7 to 1.1 in.-lb. Bearing sets using K-162B against Al_2O_3 sockets and Al_2O_3 against mechanical graphite bushings and thrust faces showed maximum torques ranging from 0.05 to 0.5 in.-lb. The bearing set using K-162B against Al_2O_3 with dry-film lubricant showed 0.65 in.-lb maximum torque after 1487 hours or 62 days. The bearing set using K-162B against Al_2O_3 sockets with dry-film lubricant and Al_2O_3 against mechanical graphite bushings and thrust faces showed a maximum torque of 0.35 in.-lb. This test was the longest test, lasting over 90 days.

TABLE 12
SUMMARY OF MATERIAL COMBINATIONS USED
IN PROTOTYPE DRUM BEARING TESTS

Bearing Material Combinations [*]		Number of Tests	Duration of Tests (hr)	Number of Rotations [†]	Maximum Torque (in.-lb)
Shaft to ball and thrust face surfaces	Ball to socket surfaces				
K-162B + Al_2O_3	K-162B + Al_2O_3	3	330	8500	1.1
			290	2500	1.0
			190	120	0.7
CDJ-83 + Al_2O_3	K-162B + Al_2O_3	4	270	1900	0.3
			270	2600	0.05
			300	2650	0.5
			270	2600	0.5
K-162B + Al_2O_3 + X-15	K-162B + Al_2O_3 with X-15	1	1487	570	0.65
CDJ-83 + Al_2O_3	K-162B + Al_2O_3 with X-15	1	2340	770	0.35

*K-162B is a sintered $\text{TiC} + \text{Ni}$
CDJ-83 is a carbon graphite, X-15 is a sodium-silicate bonded MoS_2 plus graphite dry-film coating.

†One rotation is 85° clockwise and 85° counterclockwise

In addition to testing each bearing design under operating environments, thermal cycling tests have been conducted to evaluate temperature transients on the bearing tolerances.

Nine bearing sets were thermally cycled from 200 to 1000°F. These tests were performed with the bearings assembled but without the simulated drum weight. The temperature change rates were approximately 7.5°F/min, increasing, and 4.5°F/min, decreasing. These rates are lower than design requirements due to limitations of the existing equipment. The tests were performed at a pressure of $\sim 1 \times 10^{-6}$ Torr. Six of the tests were thermally cycled five times and three sets thermally cycled ten times. Results of these tests revealed slight discoloration of the bearing sets (Figure 18) but no appreciable weight change, dimensional change, flaking of coatings, or structural failure. It should be noted that the first nine sets tested for torque, temperature, and vacuum could also be considered as single-cycle thermal cycle tests. These tests produced no noted adverse effects.

Analysis of differential temperature effects were made. Table 13 shows the allowable differential temperature between various reference design bearing parts for each SNAP system at 70°F and at operating temperature. Adequate margins are provided at both conditions.

TABLE 13
ALLOWABLE TEMPERATURE DIFFERENTIAL BETWEEN
REFERENCE DESIGN DRUM BEARING PARTS

	Radial Bearing		Thrust Bearing	
	ΔT , Shaft to Ball (°F)	ΔT , Ball to Socket (°F)	ΔT , Shaft to Ball (°F)	ΔT , Ball to Socket (°F)
System at Approximately Maximum Operating Temperature				
SNAP 10A	1100	213	1100	96
SNAP 2	680	213	680	163
SNAP 8	170	32	170	46
System at Approximately 70°F Operating Temperature				
SNAP 10A	1200	120	1200	83
SNAP 2	568	120	568	83
SNAP 8	343	14.5	343	12.7

In a sliding-couple bearing, a thermal gradient across the bearing seems to have little effect upon friction. The prototype drum bearing tests using mechanical graphites against aluminum-oxide-coated surfaces, and similar sliding block tests, all yielded low coefficients of friction even though entirely different methods of heating were employed. The results of heating the prototype bearings (radiant heating from quartz lamps) produced very small temperature differences between mating parts, whereas the heating method of the sliding block tests (conduction through the samples) produced temperature differences of about 100°F for 800°F temperature settings.

Shock and vibration tests to simulate launch and handling conditions were conducted on reflector bearings. The shock and vibration testing, as in the friction testing, was performed with simulated drum loadings. Figure 21 shows the SNAP 10A and 2 vibration test fixture. The SNAP 8 drum bearings were tested on a similar fixture. All tests were conducted at laboratory ambient temperature and pressure.

The first of the SNAP 10A shock and vibration tests showed that the ceramic coating spalled off wherever substrate edges were relatively sharp. Design changes of improving coating techniques and chamferring sharp edge

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Figure 21. SNAP 10A/2 Vibration
Test Fixture

TABLE 14
DRUM BEARINGS – ENVIRONMENTAL TEST RESULTS

Bearing Test No	Bearing Set No	Materials				Thermal Cycle Tests Completed (200 to 1000°F)	Shock and Vibration Tests to Design Specification Completed	High Vacuum, Elevated Temperature Drum Torque Tests								
								Test Duration (hr)	Actuation Cycles	Torque (in.-lb)			Temperature (°F)		Vacuum (Torr)	Remarks
		Balls	Bushings	Shafts	Sockets					Initial	Maximum	Final	Radial Bearing	Thrust Bearing		
1	1	K-162B		T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2	5	Yes	330	8500	1.0	1.1	1.1	950	780	1 x 10 ⁻⁶	A number of operational problems occurred during this test. An additional run was made for 70 hours and 350 more actuation cycles. Results were identical.
2	3	K-162B	CDJ-83	T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2	5	No	270	1900	0.1	0.3	0.25	960	780	7 x 10 ⁻⁷	
3	2	K-162B	CDJ-83	T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2	5	Yes	270	2600	0.05	0.05	0.05	900	730	1 x 10 ⁻⁶	
4	4	K-162B	CDJ-83	T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2	5	Yes	300	2650	0.2	0.5	0.5	910	730	1 x 10 ⁻⁶	
5	5	K-162B	CDJ-83	T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2	5	Yes	270	2600	0.1	0.5	0.5	900	720	7 x 10 ⁻⁷	
6	6	K-162B		T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2	10	Yes	100	680	1.0	2.5	2.5	890	730	1 x 10 ⁻⁶	
6B	6	← Same set as above →						290	2500	0.5	1.0	0.9	890	730	1 x 10 ⁻⁶	SNAP 10A backup design Balls became tight in sockets
7	13	K-162B		T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2	None	No	190	120	0.3	0.70	0.70	920	750	8 x 10 ⁻⁷	
8	13A	← Same set as above plus X-15 on all Al ₂ O ₃ , LA-2 surfaces →				None	No	1487	570	0.2	0.65	0.5	930	750	8 x 10 ⁻⁷	
9	14	K-162B	CDJ-83	T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2 + X-15	None	No	2340	770	0.05	0.35	0.35	665	505	2 x 10 ⁻⁷	
10	15	K-162B	P-5N	T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2 + X-15	None	Yes	← 90-day test scheduled to start mid-April 1964 →								
81	81	Inconel-X + Al ₂ O ₃	K-162B	Inconel-X + Al ₂ O ₃ , LA-2 + X-15	Inconel-X + Al ₂ O ₃ , LA-2 + X-15	None	Yes	← To be tested in ultrahigh vacuum →								
11	7	K-162B		T ₁ + Al ₂ O ₃ , LA-2	T ₁ + Al ₂ O ₃ , LA-2	5	Yes	← To be tested in ultrahigh vacuum →						Tested in SNAP 10A reflector shock and vibration at one and two times design specification.		
12	10	T ₁ + Cr ₃ C ₂	P-5N	T ₁ + Cr ₃ C ₂	T ₁ + Cr ₃ C ₂	10	Yes	← Not tested to date →								
13	11	T ₁ + Cr ₃ C ₂	CDJ-83	T ₁ + Cr ₃ C ₂	T ₁ + Cr ₃ C ₂			← Not tested to date →								

Note K-162B is a sintered TiC + Ni, P-5N is a Purebon carbon graphite, LA-2 is flame-sprayed Al₂O₃, X-15 is a sodium-silicate bonded MoS₂ plus graphite dry-film coating, CDJ-83 is a carbon graphite

configurations remedied the spalling problem for all successive bearing tests. With the improved design, no dimensional or structural changes were noted in succeeding tests. Five of the bearings which completed the shock and vibration tests were later subjected to the torque-vacuum-temperature tests previously described. No deleterious effects were observed.

One SNAP 10A and one SNAP 8 bearing set were coated with a sodium-silicate bonded MoS_2 dry-film lubricant prior to shock and vibration. These sets (all four bearings) were found to be tight in the sockets upon disassembly. Investigation of the coating showed that spalling had occurred. From this test it was hypothesized that the sodium-silicate binding agent in the dry-film lubricant was not baked at a high enough cure temperature. This has been resolved by developing an improved technique for applying the sodium-silicate bonded dry-film lubricants. Table 14 summarizes the results of all environmental tests conducted on the reflector bearings.

Several experimental bearings have also been designed and manufactured using P-5N on the spherical surface. One design utilizes balls made completely from solid graphite (Figure 22). Another design utilizes a configuration in which

Figure 22. Solid Graphite Spherical Ball

Figure 23. Graphite Segment Spherical Balls

the ball is formed by spherical segments surrounding a metal cylinder which houses a graphite bushing (Figure 23). A variation of this latter design utilizes spherical concave graphite surfaces assembled in the socket pieces by a shrink-fitting process. The solid carbon-graphite ball design has been subjected to the SNAP 10A shock and vibration tests with no apparent damage.

2. Materials Oxidation Tests

The dry-film lubricants and carbon-graphites that are selected for space application may necessarily be of different material than the materials required for ground test of the drum-drive system. When the environmental temperature exceeds 700°F the oxidation rates of the materials for the space system, especially the dry-film lubricants, are excessive at pressures of 1×10^{-3} Torr and above. Sliding block tests have been conducted on reference design materials to establish a "safe" operating pressure for ground test of the drum-drive system. These tests employ simple blocks in sliding couple in an oil diffusion pump vacuum chamber. The test pressure is held constant by regulating a controlled leak between the test chamber and the diffusion pump. Data from the materials tests at poor vacuums are shown in Table 15.

Results of these tests show that only slight oxidation of the graphite had occurred after slightly more than 1000 hr at 1×10^{-4} Torr and 1100°F, while at 1×10^{-3} and 10^{-2} Torr, excessive oxidation occurred after 500 hr or less. Dimensional stability could be affected since considerable oxide scale is produced at 10^{-2} Torr. Although this study is incomplete, preliminary data indicate that pressure levels should be 1×10^{-5} Torr or below for ground test applications where temperatures exceed 700°F.

A number of samples have also been subjected to 100-hr poor vacuum tests, as shown in Table 16. Analysis of these samples is not complete; however, there has been a large apparent oxidation of all materials except Fluorspar, pyrolytic graphite, and National Carbon WCG graphite cloth. The dry-film lubricants have been analyzed after test and the MoS_2 was shown to be completely converted to MoO_3 .

TABLE 15
CONTROLLED VACUUM FRICTION STUDIES

Material Combination	Friction Coefficient		Temperature (°F)	Vacuum (Torr)	Linear Travel (in)	Hours in Test	Bearing Load (psi)	Remarks
	Start	Finish						
Purebon P-5N Al ₂ O ₃ , LA-2	0 18	0 09	1050	1 x 10 ⁻⁴	2600	624	40	Slight oxidation on exposed surface of P-5N, no oxidation on covered surface
Purebon P-5N Al ₂ O ₃ , LA-2	0 17	0 23	1300	5 x 10 ⁻³	1060	5	40	Al ₂ O ₃ coating failed, requiring test shutdown Slight oxidation of the P-5N occurred on the exposed surfaces
Purebon P-5N Al ₂ O ₃ , LA-2	0 17	0 6	1050	1 x 10 ⁻³	3417	542	40	Gross oxidation occurred on the exposed P-5N Dimensional loss of 0 001 in resulted
Purebon P-5N Al ₂ O ₃ , LA-2	0 19	0 16	1100	1 x 10 ⁻⁴	7616	1032	40	Slight oxidation on exposed surface of P-5N No oxidation at center of exposed surface
Purebon P-5N Al ₂ O ₃ , LA-2	0 26	0 18	1100	1 x 10 ⁻⁴	6571	1080	40	Slight oxidation on exposed surface of P-5N No oxidation at center of exposed surface
Molykote X-15 (MoS ₂) Al ₂ O ₃ , LA-2	0 21	0 86 0 19	1100	1 x 10 ⁻⁴	220	576	40	Oxidation occurred on the exposed area, no noticeable oxidation was evident between the bearing faces "Stiction" observed at start of cycles
Molykote X-15 (MoS ₂) on K-162B Al ₂ O ₃ , LA-2	0 20	0 55 0 19	750	1 x 10 ⁻²	12025	144	40	Approximately 80% of the coating oxidized Stickslip occurred after 4 days continuous run The MoO ₃ oxide continued to prevent seizure
Drilube 867N (MoSe ₂) Al ₂ O ₃ , LA-2	0 15	0 30 0 12	1000	1 x 10 ⁻³	788	24	40	Stickslip occurred at 1100°F after 5-hr dwell Seizure was prevented by the contaminant film Typical results for two tests
Drilube 861N (WSe ₂) Al ₂ O ₃ , LA-2	0 15	0 82 0 70	900	1 x 10 ⁻³	670	24	40	Seizure occurred due to film breakdown Oxidation appears to be rapid at 700°F
C1co 3026A (MoS ₂ + LiF) Al ₂ O ₃ , LA-2	0 32	0 70 0 36	1300	5 x 10 ⁻³	1450	24	40	Lubricated galling occurred, however, seizure was prevented by the film contamination
Molykote X-15 (MoS ₂) Stellite 6-B	0 15	0 23 0 12	750	5 x 10 ⁻³	12505	48	40	Highly burnished surface under bearing contact No oxidation evident on this short-term test
Molykote X-15 (MoS ₂) Stellite 6-B	0 15	0 78 0 10	1300	1 x 10 ⁻³	23375	120	40	Stickslip occurred after 3 days, running friction remained low No seizure occurred Oxidation occurred on the exposed coating
Molykote X-15 (MoS ₂) on Al ₂ O ₃ Purebon P-5N	0 11	0 1	1100	1 x 10 ⁻⁴	5379	936	40	No oxidation, no galling, negligible wear on dry film Slight oxidation on exposed surface of P-5N

Notes Friction coefficients recorded as 'start' are values for running friction recorded at 70°F "Finish" values are the maximum and minimum recorded during operation at the noted temperature
Dry-film coatings are approximately 0 0003 to 0 0008 in thick

TABLE 16

MATERIALS SUBJECTED TO 100-HR CONTROLLED VACUUM TESTS

Material	Pressure (microns)	Temperature (°F)
Purebon P-5N
Molykote X-15 (bonded MoS ₂ plus graphite)		
National Carbon CDJ-83		
National Carbon ZTA		
National Carbon CDJ-97		
National Carbon WCG (graphite cloth)		
Drilube 862 (unbonded WS ₂ powder)		
Molykote Z (unbonded MoS ₂ powder)		
Fluorspar Powder		
Purebon P-03NHT		
Purebon P-303N		
Purebon P-5R		
Pyrolytic Graphite		
Lubeco 6001A (bonded MoS ₂)		
Lubeco 6021 (bonded MoS ₂)		
	0.5 to 100	1100

3. Effect of Contaminants

No provision has been made for sand and dust contamination in the bearing design. A number of approaches have been considered to minimize contamination. Localized sealing of the surfaces in contact causes the frictional load to increase. A seal that will sublime quickly in space is a possibility. Rigid environmental control of the reactor throughout ground handling and use of a dust-tight nose cone may be a more practical solution. The latter is being planned for SNAP 8.

4. Self-Welding Tests

Self-welding tests in ultrahigh vacuum have shown that all materials tested except some mechanical graphites or dry-film-coated materials will self-weld with appreciable bonding strength. The use of dry lubricants and carbon graphites is considered absolutely necessary to minimize this adhesion. A sample in which Molykote X-15 dry film was applied between Inconel-X and stainless steel has been tested for over 3000 hours at 850 to 900°F with no self-welding.

TABLE 17

SNAP REFERENCE DESIGN GEAR CHARACTERISTICS

System	Component	Materials	Gear Ratio	Pressure Angle	No of Teeth	Circular Pitch (in)	Diametral Pitch (in)	Centerline to Centerline Dimension (in)	Pitch Diameter (in)	Major Diameter (in)
SNAP 10A (S10A FSM-1)	Pinion	Stellite 6-B	13 625 1	20°	16	0 09821	32	3 6562 +0 0003 -0 0000	0 500 ±0 001	0 562 +0 000 -0 005
	Gear	Ti (6Al-4V)	13 625 1	20°	218	0 09821	32	3 6562 +0 0003 -0 0000	6 8125 ±0 001	6 875 +0 000 -0 005
SNAP 2	Pinion	Hastelloy-B	13 625 1	20°	16	0 09821	32	3 6562 +0 0003 -0 0000	0 500 ±0 001	0 550 ±0 001
	Gear	Hastelloy-B	13 625 1	20°	218	0 09821	32	3 6562 +0 0003 -0 0000	6 8125 ±0 001	6 8625 ±0 0030
	Pinion (backup)	Purebon P-5N	13 625 1	20°	16	0 09821	32	3 6562 +0 0003 -0 0000	0 500 ±0 001	0 550 ±0 001
SNAP 8 (S8DRM-1)	Pinion	Inconel-X	13 8 1	20°	20	0 04911	64	2 3125 +0 0006 -0 0000	0 3125 +0 0000 -0 0007	0 3435 +0 0000 -0 0020
	Gear	Inconel-X	13 8 1	20°	276	0 04911	64	2 3125 +0 0006 -0 0000	4 3125 +0 0000 -0 0007	4 3436 +0 0000 -0.0020

Frictional characteristics thus far tested indicate that a low-shear contaminant lubricant is necessary to maintain friction coefficients below 0.2. The problem in the space environment is to maintain this low-shear boundary lubrication continuously for 10,000 hours. Tests in ultrahigh vacuum to date show that if the friction coefficient exceeds 0.2, the mechanism becomes unstable and very high friction or wear occurs.

5. Alignment Tests

Special tests have been conducted to determine the dimensional misalignment from the axial centerline that the drum bearings can sustain without binding. The SNAP 10A bearings will function properly up to one degree misalignment; i.e., the shafts misaligned by more than 0.170 in. at their extremes. The SNAP 2 bearings can function properly with up to 2 degrees, 12 minutes misalignment or 0.340 in. misalignment of the shafts. The SNAP 8 bearings can function properly with up to 1 degree, 42 minutes misalignment or 0.190 in. misalignment of the shafts.

D. GEARS

As previously discussed, control of the SNAP reactor criticality is accomplished by accurately positioning the control-drum reflectors. The reference design control-drum drive system is a pinion and gear power train driven by a stepper type of actuator. The minimum step of the actuator is 1.8° at the shaft. The reflector must be positioned in increments of 0.5° ; therefore, a single-step gear reduction is introduced between the actuator output shaft and the reflector. This gear reduction transmits the actuator torque to the reflector drive system and also provides, through the actuator brake, the required braking force to hold the reflector in place during periods of dwell.

1. Materials Combinations

The gear materials and general dimensions for the reference design pinions and gears on the three drum-drive systems are summarized in Table 17. These designs utilize metal against metal with bonded dry-film lubricant coatings. One backup design of carbon graphite is also being considered. The gears on SNAP 10A are partially outside the shadow cone of the shield; therefore titanium is used as it has a low neutron scatter cross section. The SNAP 10A

gear design also has a copper tooth insert which meshes with the Stellite pinion during launch to prevent fretting. During startup, the pinion turns past the copper tooth onto the regular titanium teeth. The SNAP 2 and 8 gears are located inside the shield shadow area and thus do not have a neutron scatter problem.

From the results of sliding-block screening tests for materials compatibility and actual configuration tests in SNAP environments, two types of material couples showed promise for gears. These were graphite vs ceramic and hard metal surfaces lubricated with sodium-silicate bonded dry-film lubricants. Several combinations of these materials have been performance tested to evaluate their design application. The gear materials must be considered not only for low-friction characteristics but also for material strength, and ease of manufacture.

2. Friction and Self-Welding Tests

Pinions and gears have been designed and tested successfully for the SNAP 10A operating conditions of 700°F in 10^{-9} Torr vacuum. The units were made of Stellite 6-B and titanium using a dry-film lubricant on the gear. The gear tooth form was a 20° pressure angle involute tooth of standard design. These units were further tested at 800°F and 10^{-5} Torr to simulate ground test conditions. The gears were loaded with a constant load simulating maximum allowable design bearing drum friction. A prototype SNAP 10A actuator was used to drive the gears. The test setup is shown in Figure 24. Test duration was 430 hr for the 800°F condition. The gear and pinion showed no measurable wear or self-welding. The prototype gear sector warpage was less than 0.001 in. for the total sector angle.

3. Shock and Vibration Tests

Shock and vibration tests have also been conducted on SNAP 10A gears. The S10DRM-1 has two gear sets on the mockup loading assembly and all tests were performed at laboratory room temperature. The shock and vibration test was performed twice on the same hardware. The gear teeth were slightly abraded at the position of contact. The abrasion appeared to be caused by the fretting phenomenon and was about 0.003-in.-deep. The normal functioning of

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Figure 24. Gear Test Fixture

the gear after drum lockout release was apparently not affected. The materials of the mating gear teeth in contact were Stellite 6-B against the copper gear insert. Later tests with dry-film-lubricant-coated parts reduced the fretting considerably. Apparently the dry film cushions the mating surfaces and thus reduces fretting and provides lubrication.

4. Tests on Other Materials

Limited success has been obtained with other materials. Several gear segments were ceramic-coated by vapor deposition in an attempt to meet the ceramic vs graphite configuration. However, the coating proved to be of poor quality, cracks developed easily, and substrate bonding was inadequate. Because of the gear geometry and size of the gear teeth it is impractical to coat gears or pinions by the same method used to coat bearings. Thus, pending new developments in ceramic coating techniques, consideration of this configuration has been set aside.

Several pinions were made up of sintered iron and graphite, also of nickel and graphite. These pinions were successfully manufactured, but later sliding block tests indicated the material to be poor for space environments.

5. Temperature Considerations

Design parameter studies have been conducted to establish limitations on backlash and accuracy at low temperature and at temperature differentials. The effect of low temperatures upon the SNAP 10A, 2, and 8 designs is less than 0.0001 in. increased backlash per 100°F temperature drop. Analysis shows that low temperatures should not increase contact stresses on the reference gear designs since the coefficients of expansion of the materials are similar.

Gear drives and materials in sliding couple have not been tested for low-temperature (below 70°F) effects. However, no problems are expected with gears that have been dry-film coated.

The maximum differential temperatures allowable between various fine control-drum elements were calculated and are presented in Table 18. The calculations show for comparative evaluation the maximum temperature increase allowable between the gears and their mounting bracketry. Any further temperature increase would result in binding. Each calculation assumes all machining and assembly tolerances for the most adverse conditions. A study

TABLE 18
MAXIMUM ALLOWABLE
TEMPERATURE CHANGE
BETWEEN GEAR AND PINION

	ΔT at Room Temperature (°F)	ΔT at Operating Temperature (°F)
SNAP 10A	65	76
SNAP 2	37	41
SNAP 8	130	103

has indicated that the probability of all components reaching their maximum tolerances is so low that the allowable temperature differential could be increased by a factor of about 50%. Therefore, it is probable that the temperature gradients will not cause gear binding.

A change of gear center-to-center distance relationship has been determined graphically for the fine-tooth gears utilized in SNAP designs. The following equation gives the change in backlash for a given change in distance between gear and pinion center lines:

$$\Delta \text{ backlash} = 0.690 \Delta (\text{centerline to centerline}).$$

6. Effect of Contaminants

Sand and dust within the meshed gear teeth will not affect accuracy as long as binding or excessive wear is not caused. A hard particle could cause increased wear if lodged into the pinion during launch, but the effect of this on the control reflector performance is unknown.

The open design and scuffing kinematics of spur gear teeth tend to minimize clogging of the gears by sand and dust. Table 19 shows the maximum particle size allowable for SNAP 10A, 2, and 8 gears at room temperature and operating temperatures. Particles any larger would cause binding.

To date, no sand and dust tests have been performed. If proper precautions are taken to maintain cleanliness — such as shrouding the drum-drive system with a nose cone — sand and dust should not be a problem on the gears.

TABLE 19
MAXIMUM ALLOWABLE
CONTAMINANT SIZE

	At Room Temperature (microns)	At Operating Temperature (microns)
SNAP 10A	30	34
SNAP 2	18	25
SNAP 8	53	53

E. FLEXIBLE COUPLINGS

One major problem in integrating the actuator, gear train, and control drum reflector into a working drum-drive system is the prevention of system binding. If the actuator pinion and drive gear are not aligned properly, binding will occur between the mating teeth. Such misalignment would also sideload the bearings in both the gear brackets and actuator housing. If the side loads are high enough, or if shaft constraint is excessive, the actuator will stall. On all three systems, the drum reflector bearings are designed to compensate for misalignment by employing a ball and socket with a rotating journal shaft that provides a self-aligning ability. It is difficult to use self-aligning bearings in the actuator due to a close tolerance requirement between the rotor, which is carried by the actuator bearings, and the stator.

On SNAP 2 and 10A, the upper drum bearing is contained in the same housing on which the actuator is mounted. In this case, differential expansion between the actuator pinion and the driven gear are minimized by mounting the gear directly on the control drum shaft (see Figures 2 and 3).

TABLE 20
FLEXIBLE COUPLING COMPARISON

Design Concept	Number	Type	Backlash	Torsional Rigidity	Axial Misalignment Force	Radial Misalignment Force†	Angular Misalignment Force§	Remarks
Double Trunion	1	Single Joint (Figure 25a)	Excessive	Error in-significant	Undesirably high	Requires two stages, doubling error	Negligible	Size impractical unless extremely close fits for pins maintained, Oxidation-sensitive due to trunion bushing material.
	2	Antibacklash	None with-in capacity	Error in-significant	See Remark	Requires two stages, doubling error	~ 0	Will not allow axial misalignment. No spring-loaded antibacklash types tested.
	3	Flexible pivots	None	Error in-significant	See Remark	Requires two stages, doubling error	No data available	Will not allow axial misalignment. No test data available.
Elastomer	4	Elastomer	None	← Function of elastic material →				No suitable elastic material available.
Bellows	5	Welded (Figure 25b)	None	Within allowable limits	Will not over-load bearings. (S8ER 550 lb/in.) (S8DRM-1 no test data)	Will not over-load bearings. (S8DRM-1 1000 lb/in.)	Will not over-load bearings. (S8ER 0 003 lb/°)	Torsional rigidity can be designed very high if diam greater than 1 in. Much better resistance to fatigue failure than convoluted type.
	6	Formed convoluted (Figures 25c and d)	None	Within allowable limits	← No data available →		Approximately same as welded (see above)	Torsional rigidity can be designed very high if diam greater than 1 in.
Flexible Disc	7	Single (Figure 25e)	None		← Function of the design →		No data	No test data on torsional rigidity, however, commercially available units within envelope range have excessively high torsional windup
	8	Double (Figures 25f and g)						
Machined Flexures	9	Multiple Disc (Figure 25h)	None	Dependent upon design	Undesirably high		No data	These comparisons apply to analytical values, no test data available.
Dihedral Geared	10	Dihedral Geared (Figure 25i)	Excessive	Error in-significant	Very small	Requires two stages, doubling error	Very small	Backlash similar to double trunion.

Results in bearing side load

†Results in bearing thrust load

§Results in bearing side and thrust load

In the SNAP 8 design, the actuator is mounted on the top of the stationary reflector structure and away from the core vessel. Again, differential expansion between the actuator pinion and the driven gear is minimized by mounting the gear on the same housing as the actuator. A flexible coupling, used to connect the driven gear to the control drum, prevents shaft and bearing constraint.

The coupling must transmit the torque from the actuator to the drum with little or no torsional deflection, yet be flexible enough to allow axial misalignment between the gear drive and the control drum. The couple must also have little or no backlash, while at the same time it must transmit radial or angular forces first one way and then another as the drum is stepped into or away from the core.

Ten different types of couplings were considered for use on SNAP 8 designs. These are outlined in Table 20. The table shows a comparison of the various types and relates the important design parameters such as backlash, torsional rigidity, axial and radial misalignment, to the SNAP 8 application. Figure 25 illustrates several of the design configurations. The types considered can be classed into two general types: those employing a design of mechanical joints for relative movement, and those employing elastic deformation of some elastic member.

In the SNAP 8 design, 0.1° is the maximum error (total shaft position error) allowable in the drive system for any particular actuator step. The coupling error must therefore be less than this. The mechanical joint concepts have the disadvantage of backlash at the joint. The backlash can be reduced by take-up mechanisms which are usually limited in capacity, add complexity, and increase the forces required for flexing. The other approach is to design with extremely close joint clearances and large radius at the joint, as shown in Figure 26. The relative motion at the mechanical joint immediately introduces backlash due to pin clearance as well as other bearing problems. For these reasons, no further investigation was made of these concepts.

Only those concepts utilizing flexural material suitable for SNAP application were considered. Of these, it appears that both the welded and formed metallic bellows types (Nos. 5 and 6 in Table 20) offer the best potential. These couplings can be designed to meet the particular performance requirements. The bellows

(a)

DOUBLE TRUNION SINGLE JOINTS,
UP TO 40° ANGULAR OPERATION

(b)

WELDED BELLOWS

(c)

FORMED BELLOWS

(d)

FORMED CONVOLUTED SHEET
FLEXIBLE BELLOWS

(e)

SINGLE FLEXIBLE DISC

(f)

DOUBLE FLEXIBLE DISC

(h)

MACHINED MULTIPLE
DISC

(i)

DIHEDRAL GEARED

(g)

DOUBLE FLEXIBLE DISC COUPLINGS COMBINED

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Figure 25. Flexible Coupling Designs

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Figure 26. Input-Output Error of Double Trunion Joint

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selected for the final design must represent the best compromise of all the influencing parameters. Figures 27 and 28 show relative torsional deflection vs torque for the two bellows-flexure concepts.

Both types of bellows were designed, fabricated, and tested for the SNAP 8 Experimental Reactor (S8ER) reflector drive. In addition to meeting the flexibility and torsional rigidity requirements at operating temperatures of 1100°F, the bellows were tested satisfactorily through 1000 design stress loading cycles under maximum load at 1100°F.

The reference S8DRM-1 reflector drive concept utilizes the bellows couplings to isolate the actuator from the control drum (see Figures 29 and 30). A formed bellows coupling was procured for the actuator drive and a welded bellows coupling for the position sensor coupling. Development testing on these parts indicates that either type will meet the SNAP system requirements.

Special bellows-type couplings are currently being designed to meet the requirements for a direct-drive actuator, which is the backup actuator design for use on the SNAP system.

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Figure 27. Torsional Deflection of Shallow Convolute (Formed) Bellows Coupling

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Figure 28. Torsional Deflection of Welded Bellows Coupling

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Figure 29. S8DRM-1 Control Drum Fine Drive Coupling Assembly

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Figure 30. S8DRM-1 Coarse Drive Coupling Assembly

IV. INTEGRATED DRIVE SCHEMES

Several design concepts have been studied for integrating all of the control-drum drive components into an optimum functioning unit. Five design schemes have been compared in terms of structural, mechanical, thermal, and shielding considerations.

The reference designs of SNAP 2 and SNAP 10A are discussed together as the designs are similar. The SNAP 10A design has been finalized and incorporated into the flight reactor. Several backup design concept variations for SNAP 2 and SNAP 8 are evaluated. An analysis of the five concepts was given in Table 1 (Section I).

A. REFERENCE DESIGN, INDIRECT-DRIVE ACTUATOR LOCATED ABOVE DRUM

1. SNAP 2/10A

a. Structural and Mechanical Design

The SNAP 2/10A reactor drive system (see Figures 2 and 3) uses a gear linkage to connect the drum actuator and the fine control drum. Three basic problems are encountered with the gears in this concept. First is the problem of self-welding of the gear teeth in the space environment. Second is the problem of impact loads by the gear teeth during launch; and finally, it is desirable to maintain a minimum backlash under all operating conditions to prevent limit cycling when this system is used with bilateral control.

The control-drum actuator is mounted rigidly to the main reflector block and has a 1/2-in.-diameter pinion gear attached to its shaft. The mating gear is attached to the control drum and serves also as the upper drum bracket which carries both longitudinal and lateral loads.

On SNAP 10A, the pinion gear is Stellite 6-B and the gear segment is titanium. The problem of self-welding is reduced by applying dry-film lubricant to the gear segment. As this dry-film layer will sublime in space, it is questionable whether the gears will not be self-welded after 10,000 hours in orbit. On SNAP 2 the backup design pinion gear is a mechanical grade carbon-graphite and the gear segment is Hastelloy-B. The use of carbon-graphite in the pinion gear provides the best protection available against self-welding, although tooth strength is greatly reduced.

The control drums are supported by two self-aligning bearings, one of which carries all longitudinal loads. During launch, the drums are prevented from rotating by the shanks of the pistons in the drum release actuators. The drum bracket which engages the piston shank fits closely so that drum motion is restricted, limiting the magnitude of impacts imparted to the gear teeth. To allow for the expected impacts on the SNAP 10A system, a soft copper insert is used in the gear segment to protect the Stellite 6-B pinion. On the SNAP 2 reactor the width of the gear was increased to protect the teeth of the graphite gear from high-impact loads.

In the SNAP 10A system, the problem of backlash is not severe, as a unilateral control system is used. The backlash of both SNAP 2 and 10A designs is approximately equal and is held to a minimum by controlling the center distance between the two gears under all operating conditions. On SNAP 10A, the shafts for both gears are located by the same bracket; therefore the center-to-center dimension is held accurately. On SNAP 2, the bracket holding the shaft of the gear segment is adjustable so that the gear center distance can be accurately set. The top and bottom drum shafts in both cases are accurately aligned to prevent shaft misalignment and binding of the gear segment as it rotates. Control of the gear center distance at operating conditions was established by selection of materials with compatible coefficients of expansion. On SNAP 10A, the gear segment is the same material as the supporting bracket and therefore will expand equally. On SNAP 2 the Hastelloy-B material was chosen to match the combined coefficient of expansion of the beryllium block and titanium support bracket.

b. Thermal Considerations

The orbital operating temperatures of the actuator housings for SNAP 2 and SNAP 10A are 750 and 650°F, respectively. The bearings on the SNAP 10A concept will operate at a temperature of 675°F, and on SNAP 2 at 775°F.

c. Shielding

The advantage of the geared system is that the control drum actuator can be moved inward so that it falls within the shield angle. On SNAP 2, the actuator is totally within the shield angle although on SNAP 10A a portion does protrude.

2. SNAP 8

a. Structural and Mechanical Design

The reference design of the SNAP 8 reflector drive system with its plug-in scram kit is shown schematically in Figure 31. The stepper-type actuator is mounted rigidly to the top of the stationary reflector structure. The output pinion of the actuator is meshed with the control drum drive gear.

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Figure 31. SNAP 8 Reflector Drive System
with Attached Scram Kit

To preserve the tolerance of the control drum motion the gearing kinematics must be accurate. Even though gears can be fabricated to close tolerances, it is difficult to maintain a parallel center distance between mating gears during operation. The SNAP 8 design minimizes this variation by mounting both gear centers on a single small housing; thus, close dimensional tolerances can be held. This approach makes it possible to match the gear and the pinion for the optimum fit prior to assembly. The drive gear is attached to the control drum with a nonbacklash, torsionally rigid, bellows coupling. The coupling takes up all the dimensional variation between the control drum and

NOTE:
ALL TEMPERATURES SHOWN
ARE IN DEGREES FAHRENHEIT

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Figure 32. S8DS Component Temperatures

its bearing on one end, and the actuator and the drive gear on the other. This arrangement eliminates the problem of machining and maintaining concentric or parallel center lines between the actuator and the control drum.

b. Thermal Considerations

In the SNAP 8 design, the actuator is mounted above the reflector on legs which decrease thermal conduction from the reflector. As a result, the primary contribution to the temperature of the actuator is from neutron and gamma heating. Cooling of the actuator is accomplished by thermal radiation to the surrounding environment.

A detailed temperature map of the present S8DS configuration is shown in Figure 32. These temperatures are for 600-kwt reactor power. (Reactor operation at 450 kw would reduce all temperatures about 40° F.) Some of the more pertinent operating temperatures on the actuator are:

Rotor operating temperature	855° F
Stator operating temperature	811° F
Maximum bearing operating temperature	950° F
Pinion gear operating temperature	946° F

c. Shielding

The entire drive motor and gear assembly is located within the shadow cone and does not contribute to the radiation levels at the payload. The neutron dose (fast) at the control drum drive motor will be about 1×10^{20} nvt for 10,000 hours of operation at 600 kw, while the gamma dose will be about 1×10^{10} rads.

B. DIRECT-DRIVE ACTUATOR LOCATED DIRECTLY ABOVE DRUM

1. SNAP 2

There are two concepts for a drum drive with the actuator located directly above the drum. In one concept, the actuator is attached directly to the top of the main reflector block (Figure 33). The shaft of the actuator drives the control drum through a flexure which takes up any shaft misalignment. In the other concept, the actuator housing is attached directly to the ball of the upper spherical bearing.

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Figure 33. SNAP 2 Direct-Drive Actuator Located Directly Above Drum.

a. Structural and Mechanical Design

In the concept where the actuator is mounted directly to the top of the reflector, the flexure at the top takes lateral loads while the spherical bearing at the bottom takes both lateral and longitudinal loads. When the actuator housing is attached to the ball of the upper spherical bearing, the upper bearings take the lateral and longitudinal loads and the lower bearing takes lateral loads. As these concepts are identical structurally to the the bottom-mounted concepts, the structural and mechanical considerations will be discussed in Section IV-D.

b. Thermal Considerations

The outside housing temperature of the directly coupled top-mounted actuator will be about the same as that for the flexibly-coupled, top-mounted actuator which was calculated to be 500°F. The temperature of the outside housing of top-mounted actuators is lower than that of the bottom-mounted actuator due to better heat radiation to space.

c. Shielding

In the top-mounted, direct-coupled concept, part of the actuator is outside the cone angle of the shield. The radiation at the bottom edge of the shield is consequently increased from an estimated 3.7×10^{12} nvt to 7.7×10^{12} nvt.

2. SNAP 8

a. Structural and Mechanical Design

The arrangement for this design is similar to the SNAP 8 reference concept except that the gears are eliminated. The actuator output is connected directly to the flexible coupling. The actuator, either the 1/2° stepper or the linear actuator, is rigidly mounted on top of the reflector structure. The flexible coupling still serves the same function of absorbing the misalignment and other dimensional changes between the actuator and the drum. Because the gears have been eliminated, the accuracy required for reflector positioning under maximum torsional loads must be established by the actuator.

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Figure 34. SNAP 2 Direct-Drive Actuator Offset
Within Shield Cone Angle

b. Thermal Considerations

The design of the direct-drive system will accommodate either the 1/2° stepper or the linear actuator. Some of the more important temperatures for these actuators when mounted at the top of the S8DS reactor operating at 600 kwt are:

	<u>1/2° Stepper</u>	<u>Linear Actuator</u>
Maximum rotor temperature	950° F	--
Maximum stator temperature	850° F	--
Maximum solenoid temperature	--	1100° F
Maximum bearing temperature	950° F	1100° F

The higher temperatures shown for the linear actuator are due to isolation of the solenoids and lower efficiency of radiation to space. Reactor operation at lower than 600 kwt would serve to reduce these temperatures in a manner similar to the decrease in the indirect-drive concept.

c. Shielding

The direct-drive actuators are larger in diameter than the reference indirect-drive actuator and when mounted directly over the control drum will extend beyond the present shadow cone about 3 in. The neutrons which scatter from the actuator would result in an excessive dose at the periphery of the payload. Increasing the size of the shield to accommodate this configuration would increase the shield weight by about 250 lb.

C. DIRECT-DRIVE ACTUATOR OFFSET WITHIN SHIELD CONE ANGLE

1. SNAP 2

In this concept, the actuator drives the control drum through a flexible coupling, thus permitting the actuator to be offset within the shield cone angle (Figure 34).

a. Structural and Mechanical Design

The manner in which the control drum is supported and rotated is quite similar to the reference design for the coarse control drums. The coarse control-drum configuration and supports from the reference design could be used directly with the exception that the upper drum shaft would be attached to one end of the flexible coupling instead of the snap-in spring. The longitudinal and lateral loads from the control drum during liftoff would be carried by the

upper spherical bearing and only lateral loads would be taken by the lower spherical bearings. In this concept, the control drum is supported independently of the actuator. Thus, the only loads on the actuator shaft and bearings would be those due to the rotor, and to the bending of the flexible coupling.

The gear train is replaced with a flexible coupling which relies on elastic deformation rather than sliding surfaces. A bellows is an example of the elastic coupling. It has relatively high torsional rigidity, but is quite flexible laterally and axially. A bellows that would perform this task has been investigated. It is a little less than 2 in. long, about 1.5 in. in diameter, and has ten convolutions. It requires about 10 in.-lb to bend the coupling 30°. The bending will cause a constant 7-1/2-lb load on the rotor bearings in the actuator. The drag on the rotor will therefore be about 1 in.-lb, or at the most 3 in.-lb. There will also be a constant load of about one pound on the spherical bearings. The coupling should not present any vibration problems, but this will have to be checked out experimentally.

b. Thermal Considerations

The outside housing temperature of the top-mounted actuator was calculated to be 500°F at a 125-kw power level and an average reactor vessel temperature of 1200°F. Solar heating, nuclear heating, radiation to space, and conduction and radiation from the reflector were considered in these calculations. The actuator was assumed to have an emissivity of 0.85. The high emissivity and unrestricted view of space account for the relatively low outside housing temperature. The other values of emissivity and absorptivity used in the calculations are consistent with those values previously used in determining the reflector temperature.

The temperature gradient across the housing on the top of the actuator is estimated to be 100°F from the outer edge to the center. This section of the housing is relatively thin and therefore a poor conductor. This condition can be alleviated by designing a thicker cover and providing a larger contact area between the outer edge of the cover and the rest of the actuator.

c. Shielding

Since in this concept the actuator has been rotated back within the cone angle of the shield, there should be no increase in radiation at the instrument and control package.

2. SNAP 8

a. Structural and Mechanical Design

A modification of the direct-drive actuator with its plug-in scram kit is shown in Figure 31. By skewing the actuator centerline with respect to the control-drum centers, the actuator can be located within the shadow cone of the payload shield. Either the $1/2^\circ$ stepper or the linear actuator can be adapted to this design. The connection between the actuator output and the control drum is again accomplished by means of a flexible coupling. However, in this case the coupling always operates under a bending load while performing its drive and misalignment absorption function. This imposes an additional requirement upon the coupling which the preliminary investigation indicates can be accommodated by a bellows coupling. The continuously flexed coupling also imposes reaction loads upon the actuator output shaft and the control drum bearings. However, these loads should not increase the total bearing loads appreciably. The probable longer required length of the coupling may necessitate further evaluation of its stability during shock and vibration.

b. Thermal Considerations

The thermal environment of the actuator and the coupling on this design is essentially unchanged from the previous direct coupling design. By designing the support for the linear actuator so that it will reduce the heat conduction path, the temperature can be minimized.

c. Shielding

This design approach locates the actuator toward the core centerline and uses a flexible coupling to connect the drum (see Figure 35). Thus, the irradiation dosage to the actuator would remain unchanged from other top-mounted actuators located inside the shield cone.

D. DIRECT-DRIVE ACTUATOR MOUNTED BELOW DRUM

1. SNAP 2

There are two basic concepts for mounting the direct-drive actuator at the bottom of the reflector between the reflector and the shield. In the first concept (Figure 36), the actuator is attached directly to the main reflector support block. The shaft of the actuator drives the control drum through a flexure

TOP VIEW

SECTIONAL ELEVATION

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Figure 35. SNAP 8 Direct-Drive Actuator Offset Within Shield Cone Angle

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Figure 36. SNAP 2 Direct-Drive Actuator Below Drum

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Figure 37. SNAP 2 Direct-Drive Actuator Mounted Below Drum (Actuator attached to self-aligning bearing)

which takes up any shaft misalignment while transmitting only negligible longitudinal loads. In the second concept (Figure 37), the actuator housing is attached directly to the ball of the lower spherical bearing. The housing is keyed to the reflector and the shaft is attached to the control drum. This enables the actuator to pivot to allow for shaft misalignments.

From the thermal and nuclear standpoint, the concepts are equivalent, but there are significant structural differences.

a. Structural and Mechanical Design

(1) Actuator Mounted Directly to the Reflector

In the direct-mounted concept, the lateral loads transmitted from the control drum during liftoff will be taken by the spherical bearing at the top of the drum and the flexure at the bottom. The longitudinal loads will be carried by the spherical bearing at the top. The longitudinal and lateral loads from the actuator will be transmitted directly into the reflector. The lateral load on the flexure will be transmitted through the actuator shaft and bearings into the actuator housing and then to the reflector.

The reference design incorporates a shallow hub with the internal bearings spaced at about 1-3/8 in. If the reference control drum and reflector configuration is selected, the actuator shaft should be extended about 1-3/8 in. to pick up the lateral load from the flexure. This load will not overstress the shaft, but it will cause ~0.01-in. deflections in the shaft.

If the hub on the actuator were lengthened by about one inch and the bearings spaced accordingly, it would facilitate mounting the actuator, lower the shaft stresses and bearing loads, and decrease the deflection of the shaft.

Note that compared to the previous design the use here of a flexure eliminates two bearing surfaces. A flexure has been designed which will not buckle under the resultant lateral loads. It will take a $\pm 1/16$ -in. longitudinal motion with less than 2-1/2 lb resistance.

(2) Actuator Housing Mounted to Ball of Spherical Bearing

In this concept, the lateral loads imposed by launch on the control drums are taken at the top and bottom of the drum by spherical bearings which also allow for misalignments. The longitudinal loads are taken by the lower

spherical bearing. Thus, both the lateral and longitudinal loads from the control drum are transmitted through the shaft and bearings of the actuator to the reflector.

b. Thermal Considerations

The outside housing temperature of the bottom-mounted actuator was estimated to be 700°F for a 125-kw power level and a 1300°F NaK outlet temperature. This result should apply to either the direct-mounted or the ball-mounted concept. Nuclear heating, solar heating, radiation and conduction from the reflector, radiation from the shield, and radiation to space were considered. The emissivities and absorptivities assumed in the calculations are consistent with the assumptions made in calculating the reflector temperatures for various power levels and outlet temperatures. The relatively low outside housing temperature is due to the assumption that the housing has a high emissivity coating ($\epsilon = 0.85$). This lowers the housing temperature by increasing the radiation heat transfer to space.

The internal temperature rise due to nuclear heating was estimated to be less than 100°F. Internal temperature differences between components can be minimized by using high-emissivity coatings.

c. Shielding

Since the actuator is within the cone angle of the shield in both concepts, there should be no increase in radiation to the instrument and control package.

2. SNAP 8

a. Structural and Mechanical Design

The direct-drive actuator concept places the actuator, either 1/2° stepper or linear, between the reflectors and the shield. While presenting no structural or mechanical problem itself, it does affect the reactor destruct system and the end-of-life reflector ejection pivots. Due to interference with the actuator, the location of the explosive charge would require revision. The redesign of the reflector ejection pivot presents no major problem. The flexible coupling being developed for the reference design should be suitable for this application. The plug-in ground test scram kits cannot be used directly, however. For this concept it would be necessary to deactivate the actuator and mount the scram kit on the top of the reflector during ground checkout and testing.

b. Thermal Considerations

Since the bottom-mounted actuators cannot "see" as much space, the operating temperatures could be somewhat higher than in the top-mounted concept. Detailed analyses have not been made but the temperatures are estimated as follows:

	<u>1/2° Stepper</u>	<u>Linear Actuator</u>
Maximum rotor temperature	975° F	-
Maximum stator temperature	875° F	-
Maximum solenoid temperature	-	1100° F
Maximum bearing temperature	1000° F	1100° F

c. Shielding

As the actuator would be located well within the shadow cone no shielding problem will exist. The radiation dosage to the actuator should not be significantly different from the top-mounted designs.

E. ACTUATORS LOCATED BELOW SHIELD

1. SNAP 2

a. Structural and Mechanical Design

There are two concepts for mounting the reference design actuator below the shield. In one concept, the control drum would be driven by a gear train located at the base of the drum. The pinion gear would be mounted on the top of a shaft which extends below the shield to the actuator. In the other concept, both the actuator and gears would be located below the shield. In this concept, the flexible coupling is replaced by a universal joint with sliding pins which allow longitudinal motion and disengage completely when the reflector is ejected. A stepped shaft connects the universal joint to the actuator. This shaft is supported at one end by the universal joint and at the other end by the actuator. In the universal joint, which is located just below the drum, the sliding pins also allow for relative axial motion between the shield and the connecting shaft. To permit disengagement of the universal joint without binding, the axis about which the reflector assembly pivots during ejection must coincide with one axis of the joint.

Figure 38. SNAP 8 Indirect-Drive Actuator
Mounted Below Shield

b. Thermal Considerations

The temperature of the actuator depends on the actual mounting location below the shield. Depending upon the exact location, the operating temperature could vary from 500 to 800° F.

c. Shielding

To prevent neutron streaming through the access hole in the shield it is necessary to offset or jog the drive shaft or use a stepped hollow shaft filled with lithium hydride.

2. SNAP 8

a. Structural and Mechanical Design

The drive system which appears most practical for an actuator located below the shield is shown in Figure 38. To accommodate this drive system, the reference design reflector and support system is modified to consist of three sections. Each section has one startup drum and one long-term control drum. The actuator for the long-term control drum is located below the neutron shield. The actuator gear train drives the control drum through a stepped drive shaft which is filled with lithium hydride. To compensate for differential thermal expansion, the drive shaft has nonbacklash torsionally rigid, flexible couplings at the actuator and at the control drum. The lower flexible coupling is axially rigid to support the mass of the drive shaft during launch. The upper flexible coupling allows axial play and is designed to disengage completely upon reflector ejection. The pivot for each reflector and support segment is located on the same centerline as that of the upper flexible coupling. When the reactor is to be shut down, the reflector section rotates about this centerline until it becomes disengaged from the reactor, at which time the breakaway action of the upper coupling allows the reflector to be ejected.

In all other respects, the reflector and drive system is the same as the reference design. However, the reactor core vessel mounting can be modified to increase the distance between the core vessel and the shield if required. The stepped holes in the shield required to accommodate the drive shaft will be lined with the shield casing material.

The major revision required for this design is in the fabrication of the shield. The addition of the access holes for the drive shafts increases the problem of casting a void-free lithium hydride shield.

Further study is required to determine the clearance between the drive shaft and the access hole. The clearance must be adequate to accommodate the relative movement of the shaft and the hole due to distortions from differential thermal expansion at operating temperature. As shown in Figure 39, the increase in the clearance also greatly increases the irradiation dosage in the local payload area.

In order to maintain the required accuracy of the drive system, the two flexible couplings on the drive shaft must prevent backlash but allow the necessary flexibility and strength to withstand launch loads.

Preliminary estimates indicate that this design will increase the weight of the reflector and drive system 30 to 60 lb. The weight increase is due to the increased complexity of the components.

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Figure 39. Effect of Drive-Shaft Tube Clearance on Payload Radiation

b. Thermal Considerations

The operating temperatures for this concept are dependent on the actual mounting location of the actuator. However, the maximum operating temperature is not expected to exceed 600°F, which would considerably increase the operating life of the actuator and gear lubricants.

c. Shielding

The total integrated radiation dosage to the actuators and gears would be reduced to approximately 1×10^{13} nvt and 1×10^7 rads gamma.

F. RECOMMENDATIONS FOR FURTHER STUDY

The concept of locating the actuator below the shield presents shield fabrication problems. However, in any case it is desirable to reduce the environmental temperature of the actuator. Several suggestions which merit further study are:

- 1) Submerge the actuator part way into the shield between the thermal and neutron shield.
- 2) Place the actuator between the two sections of the lithium hydride shield.
- 3) Raise the actuator from its S8DRM-1 position and provide thermal and gamma shielding.
- 4) Provide thermal and/or gamma shielding to the top-mounted direct-drive actuator offset within the shield shadow cone.

G. GROUND TEST CONFIGURATION

Optimum failsafe scram capabilities, required during nuclear ground tests, can be achieved by locating the actuator for the ground test drive system above the reflector. A removable spring-driven scram mechanism has been designed for each of the top-mounted actuator designs (see Figures 36 and 40). Use of this mechanism would permit ground checkout of a drive system which would be almost identical to the top-mounted flight system in performance, step-size errors, and temperature distribution. A prototype scram mechanism has been fabricated and proof tested. Successful operation has been achieved at temperatures to 800°F in vacuums of 10^{-6} Torr. Long-term temperature and vacuum endurance tests are in progress on a developmental model.

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Figure 40. Top-Mounted Actuator Designs With Removable Scram Mechanisms

V. CONCLUSION

A. DESIGN CRITERIA

Various control-drum drive concepts have been evaluated and compared to the reference designs. The SNAP 10A concept has been incorporated into the flight design. A comparison of the SNAP 8 and SNAP 2 concepts is summarized in Table 21.

To date, extensive development studies have been completed on the reference design concepts and design criteria have been met satisfactorily. Life tests will be conducted.

B. COMPONENTS

1. Actuators

The direct-drive actuator concepts appear to offer a less complex and thus more reliable design than the reference indirect-drive concept. The direct-drive design eliminates the gear train and uses a motor which provides more torque and is inherently more reliable.

2. Gears and Couplings

In the reference design, the driving torque is transmitted from the actuator to the control drum by a single set of spur gears. The gears provide a mechanical advantage to the actuator output. They also reduce the holding torque the actuator brake must apply to hold the drum position during space flight maneuvers.

The reference design gears on SNAP 2 are Hastelloy B; and on 10A the pinion is Stellite 6-B and the gear, titanium (6Al-4V). The pinion and gear for SNAP 8 design are both Inconel-X. In all cases both pinion and gear are coated with a sodium-silicate bonded dry-film lubricant using MoS_2 as the pigment. The SNAP 2/10A gears have been successfully tested in ultrahigh vacuums at 700°F for 120 days. These designs have also passed shock and vibration testing.

In the reference design system for SNAP 2 and 10A the actuator and gear train are mounted on the same housing to minimize gear backlash and misalignment. The SNAP 8 design locates the actuator away from the core vessel to reduce the environmental temperature of the drive system. The drive gear is connected to the control drum through a flexible bellows coupling.

TABLE 21
COMPARISON OF CONTROL DRUM ACTUATOR DRIVE CONFIGURATIONS
FOR THE SNAP 2 AND SNAP 8 REACTORS

Drum-Drive Concept	Actuator Location	System	Actuator Weight (lb)	Mounting Weight (lb)	Temperature of Actuator (Calculated) (°F)	Developmental Problems	Remarks
Indirect-drive (Reference design)	Above drum	SNAP 2	6	6	775	1) Gear self-welding 2) Gear fabrication with nonself-weld materials	1) Largest amount of developmental information available on the actuator, with operational tests up to 10,000 hours at 850°F. 2) Step size error in actuator can be minimized by multiple steps and high-ratio gearing.
		SNAP 8	6	12	950	3) Minimizing errors from gear and drive shaft	
Direct-drive (1/2° Stepper)	Above drum	SNAP 2	7	7	~500	1) Angular offset drive coupling	1) Step accuracy is a function of bearing friction and drum inertia reaction. 2) Step size errors are not cumulative. 3) Bottom mounting space very limited on SNAP 2 and 8 and would not permit plug-in scram kit adaptation. 4) Bottom mounting design for SNAP 2 requires additional pivot point for torque reaction of actuator.
		SNAP 8	7	14	~950		
	Below drum	SNAP 2	7	7	800 (Estimated)	1) Special self-aligning bearing for combination drum and actuator shaft	
		SNAP 8	7	14	~1000	2) Special quick release scram pivot assembly	
Direct-drive (1/2° Linear)	Above drum	SNAP 2	5	5	700	1) Angular offset drive coupling	1) Step size accuracy should not be appreciably affected by bearing friction or drum reaction. 2) Estimated step accuracy $\pm 0.02^\circ$. 3) Step size is adjustable after manufacture and assembly. 4) Bottom mounting space very limited and would not permit plug-in scram kit adaptation. 5) Bottom mount design for SNAP 2 requires additional pivot point for torque reaction of actuator.
		SNAP 8	5	10	~1100		
	Below drum	SNAP 2	5	5	800	1) Special self-aligning bearing for combination drum and actuator shaft	
		SNAP 8	5	10	~1100	2) Special quick release scram pivot assembly	

The coupling is designed to absorb any axial, lateral, or angular misalignment between the gear and the drum.

3. Bearings

Mechanical carbon-graphite in sliding couple with alumina (Al_2O_3)-coated metals has proved to be a low-friction, low-wear bearing material for the control drum bearings. The mechanical graphites have proved to be structurally sound when used in a precompressed bushing design. These materials exhibit their lowest friction coefficients at temperatures above 500°F in vacuum. The friction coefficient rises from less than 0.2 at 500°F to about 0.6 at room temperature. This increase is not observed, however, in poor vacuum (100-micron pressure) or at atmospheric pressure.

Sodium-silicate bonded dry-film lubricants on the Al_2O_3 surfaces or between Al_2O_3 surfaces provide friction coefficients of less than 0.2 at temperatures from room ambient to 1100°F in vacuum.

Both the carbon-graphites and dry-film lubricants will oxidize rapidly at temperatures above 700°F in air at one atmosphere pressure. Only slight oxidation occurs, however, in 1000 hours at 1100°F and air at 1×10^{-4} Torr.

C. INTEGRATED DRIVE SCHEMES

Of the various locations studied for the actuator, that in which the actuator is mounted directly above the drum is considered to be optimum. If the actuator were mounted directly below the drum, it would be located in a higher-temperature region. Location of the actuator below the shield would greatly reduce the temperature but would require a long drive shaft and an access hole through the shield which would result in neutron streaming toward the payload. Table 22 shows the estimated temperatures of the actuator in the various drum-drive concepts.

From comparison of all the concepts studied, it is concluded that the optimum design is a direct-drive actuator mounted directly above the drum with flexible coupling. This concept and the same configuration without flexible coupling will be studied further.

TABLE 22

ESTIMATED TEMPERATURES OF SNAP 8 ACTUATORS

Drum-Drive Concept	Actuator Location	Component	Temperature (°F)
Indirect-drive (Reference design)	Above drum	Rotor	855
		Stator	811
		Upper bearing	800
		Lower bearing	950
Direct-drive (1/2° Stepper)	Directly above drum	Rotor*	950
		Stator*	850
		Upper bearing*	900
		Lower bearing*	950
	Above drum, offset within shield cone	Rotor*	950
		Stator*	850
		Upper bearing*	900
		Lower bearing*	950
	Below drum	Rotor*	975
		Stator*	875
		Upper bearing*	1000
		Lower bearing*	950
Direct-drive (1/2° Linear)	Directly above drum	Bearing	1100
		Solenoids	1100
	Above drum offset within shield cone	Bearing	1100
		Solenoids	1100
	Below drum	Bearing	1100
		Solenoids	1100
Direct-drive or indirect drive	Below shield	Total assembly	~600

*Estimated $\pm 50^\circ\text{F}$

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