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SUBJECT: Reliability Aspects of Control Drum Drive Systems

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SUMMARY: The reliability of various parts of a control drum actuation system is discussed. Alternate schemes are compared, possible failure modes are analyzed, and redundancy effects on the overall system are studied.

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RELIABILITY ASPECTS OF CONTROL DRUM DRIVE SYSTEMS

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CONCLUSIONS

A study is presented comparing and discussing the reliability of two basic methods of driving control drum by means of pneumatic actuators. Proposed variations in design of component parts are examined. Recommendations and reservations are expounded.

The conclusion is reached that a mechanically coupled system driven by three actuators is six times as reliable as the presently accepted system of individual actuators for all control drums. A more general conclusion from reliability trade off studies favors minimizing the number of actuators. This approach is recommended for larger systems, subject to the assurance of functional independence through the use of highly reliable decoupling and override features.

BACKGROUND

The means of control for the reactor must be of the highest feasible reliability. Eventual space usage re-emphasizes this requirement.

In order to attain the highest feasible reliability for the drum control system, development work must proceed on mechanical design aspects of the system as well as in the electrical portion.

A beginning for this work lies in determining deficiencies in the present system and/or finding a new approach that demonstrates decided advantages that warrant change.

The NRX system uses control drums as the final link in the control system. These drums are moved, as required, by individual actuators using closed loop control.

The two features of this system which may lead to a reliability deficiency are the use of feedback potentiometers, and overall complexity. Control Systems Engineering has suggested that these features can be eliminated, and that a change is warranted. As a result, a new drum control system has been proposed. Their proposal is discussed in detail in WANL-TME-1312. The basis of this proposal is reliability improvement. For this reason, an extensive reliability evaluation was undertaken.

The results of these evaluations are collectively presented in the following pages. They are a consolidation of information originally issued in letters No. RQA: 663, 692, 703, 732, and 720, and represent the Reliability Engineering supporting effort. Much of the material has been re-arranged, and some new material has been added for WANL-TME-1312. In particular, data provided by the Actuator Development group was used in Appendix C.

INTRODUCTION

The reliability of a component or system is the probability of it successfully completing the intended operation or mission. This reliability evaluation is not concerned with initial defects uncovered during the shakedown or developmental period; nor does it try to predict wearout failures beyond the conclusion of normal operating life. Rather, it assumes that the component will perform in accordance with specifications under specified environmental conditions for as long as required, subject to chance failures.

The accuracy of a reliability evaluation is dependent on the completeness of the design and on the availability of reliability data on similar component parts operating in a similar environment. The present study is lacking in both respects, and any quantitative comparisons can be considered significant only if the net results vary by an order of magnitude.

Definition of failure is an important factor in a reliability study. The present study was performed for an arrangement of twelve control drums, where any failure of any one drum is assumed to result in a system failure. (A more general approach is discussed in Appendix A.)

For purposes of this study, the use of three actuators was stipulated for the coupled drive scheme. The loss of a single actuator is acceptable (since all drums are still operable), but the loss of two actuators constitutes system failure (inadequate torque). It is assumed that an actuator failure will take the form of its removal from the system rather than a freezing in place. This is a reasonable - though nonconservative - assumption.

ACTUATOR

A pneumatic control drum actuator was developed for NERVA by General Electric under Contract No. 7T-AG20-A01. (A later contract, P. O. 59-NYC-49563, is a continuation of this effort on the actuator only.) As described in their report No. R64AT11, it ". . . comprises a piston, rack, and pinion actuator controlled by a single state two-sided pressure control servo valve. Drum position is sensed by a potentiometer whose output is then compared to the electrical position command and the result modified in the amplifier to drive the servo valve torque motor. In this way, the loop is closed. A shaft lock, soft stop, and rebound limiting spring are integral parts of the over-all assembly." Design details as well as a reliability evaluation are shown in the G.E. report.

Features of the PCA and CVA Designs

The G.E. actuator was devised to be connected directly to an individual control drum. Other designs are now being considered by WANL to permit a small number of actuators to drive the total number of control drums by means of a mechanical coupling arrangement. The proposed designs are the controlled velocity actuator (hereafter designated CVA), Figure 1.a and 1.b, and the pressure controlled actuator (PCA), Figure 2.

Controlled Velocity Actuator
Figure 1.a

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TABLE 1 - RELIABILITY COMPARISON OF INDIVIDUAL CLOSED LOOP ACTUATOR vs. CONTROLLED VELOCITY ACTUATOR

Reference: GE Report No. R64ATII, pp. 350 - 358

Not Included: Interconnection, decouplers, load springs and other drum components, velocity or acceleration limiting system.

Functional Block	Part Name	Item Number (or similar)	Qty.	Failure Rate per 10 ⁶ Hrs.
<u>Eliminated:</u>				
Locking	Complete Assembly			64.17
Servo Valve	Complete Assembly			34.55
Potentiometer C	Complete Assembly (See Note 8)			9.73
Shaft, spring & bearings	Spring, Soft Stop (rebound)	51 (partial)	1/2	10.00
Housing	Cover Assembly			
	Servo Valve	61 61		2.65
	Cover, Potentiometer	72 Trade for end cover		
Total for parts being eliminated				121.10
<u>Added:</u>				
Locking	Bellows & Pin Assembly	(63)	1	3.00
	Sleeve	(70)	1	.01
	Bushing			.05
	Insert, Helical	(2-4)	1	.00
	Ring, Retaining	(42)	1	.01
	Wire, Lock	(10)	2	.02
	Seal, Lock Pin	(68)	2	.02
Valving	Valve, Solenoid	(25)	2	20.00
	End Plate	(30-3)	2	.02
	Seal, Static - Large	(64)	4	2.40
	Seal, Static - Small	(68)	4	.04
	Orifice	(59-11)	2	.30
	Check Valve		1	6.00

APPENDIX B - TABLE 1 (continued)

Functional Block	Part Name	Item Number (or similar)	Qty.	Failure Rate per 10 ⁶ Hrs.
Housing	Cover, Accumulator	(72)	1	1.10
	Seal, Static	(52, 71)	2	1.20
	Insert, Helical	(2-3)	8	.05
	Screw, Cap	(8)	8	.00
	Cover, Lock & Load Rack Brg.	(72)	1	1.10
	Seal, Static	(71)	1	.60
	Insert, Helical	(2-3)	6	.05
	Screw, Cap	(8)	6	.00
				<u>35.97</u>
	Total for parts being added			35.97
Net decrease in failure rate for CVA design			85.13	

TABLE 2 - COMPARATIVE RELIABILITY PREDICTION OF TWO ACTUATOR DESIGNS

Design	Failure Rate Per Million Hours	Predicted MTBF, Hours
Closed loop controlled (GE)	244.91	4,080
Controlled Velocity ("flow control")	159.78	6,260

The proposed designs differ from the G.E. design in several respects:

1. The capacity of each design was increased to enable it to drive several drums.
2. Operation was changed from modulating control (by servo valve, with position feedback by potentiometer used in the G.E. and other current designs) to on - off control (by solenoid valves actuated via the power controller).
3. The locking device was greatly simplified by eliminating the provisions for remote operation.
4. The rebound spring was deleted. The changes were motivated mainly by a desire to improve actuator and system reliability, mindful of ultimate flight applications.

Failure Mode Analyses

A comparison of the PCA and CVA designs was based on several methods of reliability evaluation. These methods included Failure Mode Analyses (FMA) and reliability prediction.

A detailed FMA was performed for the CVA, evaluating those features of its design which were different from the G.E. design. Then, reliability predictions were made using the G. E. design as a frame of reference.

Comments resulting from these evaluations are discussed below. The details of the FMA are given in Appendix B. The reliability prediction is shown on Pages 7 and 8.

Results of FMA and Reliability Predictions

The general notes, resulting from these analyses are given below. They relate to both the CVA and the PCA designs. All work was done with the G.E. design as a frame of reference.

1. It may be concluded from Table 2 that the mean time between failures (MTBF) for a single CVA unit is 53% higher than for a single G.E. actuator. More dramatic benefits accrue from the fact that fewer actuators will be used in the proposed CVA system and that redundancy can be achieved. See "Composite System", Page 23.
2. Reliability of the actuator will decrease appreciably from values shown if a velocity or acceleration limiting device has to be incorporated. See "Decoupling Device", Page 11.
3. CVA and PCA designs are both highly susceptible to entrained or residual particles becoming lodged on valve seats or in orifices. Special effort is required to prevent introduction of particulate matter due to:
 - a. Insufficient filtration of actuating gas.

- b. Inadequate cleaning during assembly procedure.
 - c. Surface degradation during operation (separation of coating material, wear of moving parts).
 - d. Formation of ice due to moisture hide-out.
 - e. Disintegration of electrical insulation, seals, or piston rings at low temperature and vacuum.
 - f. Careless application of dry lubricant during assembly.
 - g. Surface corrosion during storage or shipment.
4. Susceptibility to contamination, surface defects, and deformation, as well as tendency toward misalignment can be reduced in both schemes by use of spherical plugs against 45° chamfered seats. This design tends to be self-cleaning and self-aligning and is easier to fabricate and maintain. See sketch on Page 32
 5. Orifices for both schemes should be accessible or removable (for inspection and cleaning) and adjustable (for calibration). Corresponding procedures must be specified and enforced.
 6. It is recommended that solenoid valves and orifices, possibly check valves and accumulators, be removed from CVA actuator housings. In this manner, all the actuators will conform to the basic G.E. design, and the solenoid valves can be of a simple, standard design. Fabrication, cleaning, inspection, and maintenance are simplified, and redundancy can be achieved by installing duplicate valves and orifices.
 7. Evaluation of the pressure controlled actuator is based on a conceptual schematic diagram. A detailed failure mode analysis cannot be made until physical parts have been designed. A few comments of a general nature are presented:
 - a. Pneumatic loading of the various actuators has to be equalized. This can be done by interconnecting the various pilot volumes, eliminating individual pairs of solenoid valves. With the present design, unequal leakage through various solenoid valves would result in unequal loading of the (ganged) actuators, causing excessive loading of the coupling device and weakening of the drive system as a whole.
 - b. A leak across the seat of either solenoid valve will cause continuous power swings, may result in valve chatter and premature wear. This will be minimized by proposed solution (a) above.
 - c. The metal diaphragms will pose an alignment problem during initial fabrication. They also tend to cock after being stroked a number of times.

- d. Control volumes will have to be eliminated completely or sized very small in order to permit small position changes. This is especially important for the condition described under (b), but will be alleviated by proposed solution (a).
 - e. Return springs should be mounted on individual control drums to assure that subcritical condition can be attained following decoupling or failure in any part of the drive system.
 - f. The PCA design is more complex than CVA and also deviates to a greater degree from the firmly established G.E. design. Pneumatic force requirements are doubled, with a corresponding increase in size and weight of components. Its use can be justified only if the CVA response characteristics are not acceptable.
8. A number of potentiometer failures have been experienced during preliminary testing of the G.E. actuator. Additional test runs will be conducted to determine whether the failures represented initial defects associated with shakedown operation or inherent weaknesses of the component design. If the latter is concluded, a re-evaluation of potentiometer failure rate will be in order. A history of potentiometer operation is presented in Appendix C.

DECOUPLING DEVICE

Design parameters for a decoupler are determined mainly by the characteristics of the actuator pneumatic system. If a "freezing" stoppage occurs in a control drum or actuator, the pneumatic pressure, hence the system torque will build up to a certain value to overcome the seizure. At this point it is conceivable that the system will be released, either by removal of the causative blockage (if a spring biased decoupler is used) or by disengagement of the decoupler (if of a detent or clutch design). The built-up pneumatic pressure under worst case conditions will now throw the unrestrained actuators, hence the control drums, into a new position. The magnitude of this movement may or may not be damaging to the core, depending on the torque which the system was permitted to reach prior to release.

Control Systems Engineering has determined that torque build-up, above normally anticipated friction, for the complete drive system must be kept below 200 in.-lbs. In line with the prohibition against a single control drum's abandonment at or movement to an extreme position, it was also decided that a spring biased decoupler design would be mandatory. The following rules were established:

- 1. A decoupler will not be installed on actuators. Reliability of the actuators is very high, as shown by analysis as well as testing to date, obviating the need for a decoupler. On the other hand, the limiting torque for an actuator decoupler would have to be set quite high, resulting in a drum recoil of 14° to 36° on release of a stuck actuator. The maximum allowable swing is approximately 4° .

2. Spring biased decouplers (Dwg. 979D168) will be installed on drums. They must be pre-loaded so that the particular drum position will not deviate from nominal system position (in either direction) until a torque of 75 in.-lbs. has been reached.
3. Beyond the pre-loaded torque point, sticking drums are allowed to build up decoupler spring bias of no more than 200 in.-lbs. (actually for the entire system) as the free drums rotate away from the frozen one(s).

It must be admitted that on previous test runs, individual control drums have developed hang-up torques far in excess of the 200 in.-lbs. stipulated for the complete system. The assumption is that this is the consequence of an initial design flaw which will be corrected in the course of current development work. Should this effort prove unsuccessful, a new and rather complicated sub-system will have to be added which can sense the acceleration resulting from release and instantly apply a brake or unload the pneumatic system to limit the movement of recoiling actuators and drums. Future development work must also include additional testing of actuators at cryogenic conditions to justify omission of the decoupler.

The interaction between decouplers and actuators will depend on a number of factors.

A. Decoupling Effects - engaged condition

1. Effect of individual engaged decoupler on overall system is a function of developed spring bias or release torque in decoupler vs. breakaway friction torque of system. Values as presently specified or estimated are:

Drum decoupler bias, max., for 12 drums	200 in.-lbs.
Drum friction, normal, for 12 drums	100 in.-lbs.
Actuator friction, normal, for 3 actuators	75 in.-lbs.
Friction added by interconnection	120 in.-lbs.

2. When any particular drum position has been reached with one or more components hung up, the system's reaction will depend on the type of actuator used and possibly the decoupling torque developed.
 - a. The PCA actuator will adjust simply by developing a higher pilot gas pressure, hence a higher pressure against the piston to offset the unbalanced torque during movement and at final hold.
 - b. The CVA actuator will move to and then hold its nominal position, provided that net torque due to decoupler(s) is less than the static friction torque capability of the overall system.

- c. Should decoupling torque exceed breakaway torque of system (i.e. exceeding the 200 in.-lbs. specified above), the CVA actuator will cycle within the limits of the control deadband against the force of the decoupler spring(s).

B. Decoupling Effects - following sudden release

1. If CVA actuators are used and decoupling occurred under condition A-2-b, a sudden release will have little if any effect on the overall drive system.
2. If decoupling took place under condition A-2-c or against PCA actuators (condition A-2-a), the control drums will swing in the direction of the most recent movement. The magnitude of the resultant power excursion will presumably exceed an allowable 4% only if the specified torque of 200 in.-lbs. has been exceeded.

It should be noted that an additional torque contribution may be transmitted to the system by the elasticity of the straps in the bands and straps coupling scheme. The direct strap connection scheme, while also composed of elastic straps, would tend to be more self-balancing. The gear drive has no springiness; rather, its high angular momentum would tend to produce a flywheel effect to suppress recoil action of actuators. See Table 4, Line 6.

A reliability evaluation of twelve control drum decouplers is shown on Table 3.

TABLE 3 - RELIABILITY OF DECOUPLERS

<u>Part Name</u>	<u>Reference</u>	<u>Quantity</u>	<u>Failure/10⁶ Hours</u>
Inner Sleeve		12	6.00
Decoupler Spring	1/2 GE No. 51	12	120.00
Decoupler Bearing	GE No. 38	24	120.00
Lower Bearing Housing	GE No. 35	12	0.12
Drum Shaft Nut	GE No. 50-4-4	12	0.12
Locking Cup		12	0.12
Spacer Sleeve	GE No. 50-4-5	24	0.24
Decoupler Cap		9	0.09
Key	GE No. 61-6	12	<u>2.04</u>
Total			248.73

COUPLING DEVICE

A number of arrangements have been proposed for interconnecting the various control drums and actuators. Their relative **merits** are tabulated qualitatively on Table 4. Letters A, B, and C are assigned for each factor, 'A' representing a favorable quality and 'C' signifying a weakness.

This evaluation is preliminary, subject in part to personal opinion and judgment. It can be refined only as design studies of the various schemes are completed. Layouts for these designs are not included in this report.

TABLE 4 - RELIABILITY DESIGN RATING FOR VARIOUS COUPLING SYSTEMS

Evaluated Rating Factors	Straps to common band (s)	Bull gear with pinions	Chain drive with sprockets	Flexible shafts with pinions	Straps between adj. drums	Links between adj. drums	Wire ropes between drums
1. General simplicity of design	C	C	B	C	A	B	A
2. Simplicity of fabrication, assembly	B	B	A	B	B	B	B
3. Special support or bearing requirements	A	C	A	C	A	A	A
4. Special material or lubrication requirements	B	B	C	C	A	C	C
5. Lowest weight	B	C	B	C	A	B	A
6. Resistance to mechanical perturbations	C	A	B	B	C	B	C
7. Subject to thermal distortion, backlash	B	B	A	B	C	B	C
8. Single failure can affect several drums	A	A	A	A	C	B	C
9. Possibility of wear	A	C	C	C	A	C	B
10. Possibility of fatigue	B	A	B	C	B	A	C
11. Possibility of loosening or disengaging	A	B	C	C	A	A	A

A reliability analysis was conducted on three schemes showing the greatest promise. The results of this study are summarized on Tables 5 through 9. Predicted failure rates are of a common order of magnitude and cannot be regarded as a decisive demonstration of the superiority of one of the schemes.

More work is needed on the various devices. Efforts should be made to reduce the number of parts for the Direct Strap and Common Band systems as shown on Tables 7 and 8. On the other hand, the gear drive design might benefit from an increase in knowledge about operation of bearing and "loose" gears under cryogenic conditions.

TABLE 5 - RELIABILITY OF DRUM COUPLING SYSTEM: Parts Common To All Designs

<u>Part Name</u>	<u>Reference</u>	<u>Quantity</u>	<u>Failures/10⁶ Hours</u>
Drive Shaft W/Cplgs.	GE No. 46	2*	10.00
Splined Coupling	GE No. 46	2*	10.00
Pulley or Pinion Housing	GE No. 50-4-3	12	12.00
Cap Screws	GE No. 31	48	0.00
Helical Inserts	GE No. 2-3	48	0.00
Lock Wire	GE No. 10	12	<u>.12</u>
Total			32.12

* 3 installed, 2 required for system operation. Benefits accruing from installation of third (spare) actuator are neglected on these items.

TABLE 6 - RELIABILITY OF THE DRUM COUPLING SYSTEM: Gear Drive

See WANL-TME-1312, Figure 3.4 through 3.6.

<u>Part Name</u>	<u>Reference</u>	<u>Quantity</u>	<u>Failures/10⁶ Hours</u>
Gear Ring (Size and shape, gear ratio, loose fit, no lubrication, temp., vacuum)		1	20.00
Pinion Gear (Loose fit)		12	60.00
Gear Ring Guide Wheel		3	6.0
Bearing	GE No. 38	6	30.0
Guide Wheel Flange (locking)		3	3.0
Bearing Spacer Sleeves	GE No. 50-4-5	<u>6</u>	<u>.06</u>
Total		31	119.06

TABLE 7 - RELIABILITY OF THE DRUM COUPLING SYSTEM: Direct Strap Connections

Assume 1 strap between every 2 adjacent drums, otherwise per Dwg. 937J503.

<u>Part Name</u>	<u>Reference</u>	<u>Quantity</u>	<u>Failure/10⁶ Hours</u>
Strap	GE No. 79	12	24.0
Mounting Block		24	12.0
Machine Screw	GE No. 22	48	0.0
Cap Screw	GE No. 80	24	12.0
Lock Wire or Cup	GE No. 10	<u>48</u>	<u>0.5</u>
Total		156	48.5

TABLE 8 - RELIABILITY OF THE DRUM COUPLING SYSTEM: Common Band With Straps

Assume 1 strap in each direction per drum.

Assume 1 band clockwise and 1 band counter-clockwise.

<u>Part Name</u>	<u>Reference</u>	<u>Quantity</u>	<u>Failures/10⁶ Hours</u>
Band	GE No. 79	2	4.0
Straps	GE No. 79	24	48.0
Mounting Block		48	24.0
Machine Screw	GE No. 22	96	0.0
Cap Screw	GE No. 80	48	24.0
Lock Wire or Cup	GE No. 10	<u>96</u>	<u>0.9</u>
Total		314	100.9

TABLE 9 - CONCLUSIONS ON RELIABILITY OF DRUM COUPLING SYSTEMS

<u>System</u>	<u>Failure Rate per 10⁶ Hours</u>	<u>Mean Time Between Failures - Hours</u>
Gear Drive	151.18	6,620
Direct Strap Connections	80.62	12,400
Common Band With Straps	133.02	7,520

CONTROL DRUM

In the course of various test runs, the control drums have suffered a number of malfunctions in terms of bowing, sticking, and scraping. These were not totally unexpected, having been predicted by prior stress analysis. (See WANL-TME-681, "Stress Analysis: NRX-A1".) The solution to the basic bowing problem becomes more urgent when pneumatic actuators are to be used, with emphasis on minimizing torques. Efforts in this direction are considered to be developmental, and to be completed successfully prior to normal operating life. Reliability of the drums during normal operation is evaluated on Table 10.

TABLE 10 - RELIABILITY OF CONTROL DRUMS

<u>Part Name</u>	<u>Similar Part GE Reference</u>	<u>Quantity per Drum</u>	<u>Failures/10⁶ Hours per Drum</u>
Shaft, dome end bearing	45	1	.30
Shaft, nozzle end bearing	45	1	.30
Housing	50-4-3	1	1.00
Plate Assembly	7	3	.03
Spring	3 - 6	1	.01
Bolt	80	8	4.00
Locking Device, bolt		8	.08
Pin		6	.06
Cylinder		1	2.00
Nut, bearing retaining	49	2	1.00
Lockwasher, bearing retaining		2	.02
Housing, bearing		1	.50
Bearing, ball	38	1	10.00
Total, one control drum			19.30
Total, 12 drums			231.60

Additional parts for individual drives (with GE actuators):

<u>Part Name</u>	<u>Similar Part GE Reference</u>	<u>Quantity per Drum</u>	<u>Failures/10⁶ Hours per Drum</u>
Drive shaft w/couplings	46	1	5.00
Splined coupling	46	1	5.00
Load spring		1	5.00
Bolt, spring retaining	80	1	0.50
Locking device, bolt		1	.01
Sub-total, additional, one control drum			15.51
Sub-total, additional, twelve control drums			186.12
Total, incl. additional, twelve control drums			417.72

An unexpected failure occurred during NRX-A3 tests when the dome end weld was severed on Control Drum No. 11. Reliability effort took the form of the failure mode analysis in an attempt to determine failure causes. The detailed results of this analysis are given in Appendix D. Reliability Engineering will continue to monitor design modifications performed for the prevention of this and previously mentioned failures.

ACTUATOR CONTROLS

The actuator control subsystem currently used is described in Figure 3. The subsystem is comprised of 12 individual local controllers having a common input point. This common point of control obtains a demand signal from the main power controller. Each local controller is in itself a closed loop servo system. A demand is balanced against the output of the feedback pot connected to the output shaft of the actuator. This shaft is then positioned (and thus the drum) as required.

Reliability improvement of the control subsystem would require the use of redundancy at a local level for best results. This would significantly affect system complexity.

The control subsystem proposed for the CVA system is shown in Figure 4. The principle of operation in this subsystem differs from that currently used. The differences are:

1. This subsystem used on-off control and is **amenable to digital design** techniques.
2. The local control is not closed loop, but closure of the **main power loop** provides the feedback.

WESTINGHOUSE ELECTRIC CORPORATION

FIGURE 3
Controls For Individual Actuators

WESTINGHOUSE ELECTRIC CORPORATION

FIGURE 4

3. The design is simpler from both quantity and detail complexity requirements.

The overall CVA control subsystem shown in Figure 4 utilizes a redundant power controller. The design shown in the sketch does not provide functional independence (therefore it is not truly redundant) but it does approximate the total complexity to be expected in a CVA control subsystem.

Reliability estimates have been made for the control subsystems discussed. These estimates are given in Tables 11, 12 and 13 in the section on system reliability.

COMPOSITE SYSTEM

Table II shows a reliability evaluation of the control subsystems and actuators as a group, with due consideration given to the benefits of redundancy. Calculations in support of these figures can be found in Appendix A.

Two subsystems are considered. Subsystem A utilizes the present GE actuator, in the present one to one configuration; i.e., one actuator for each drum. Subsystem B utilizes the proposed controlled velocity actuator, redundantly connected to the drums via a common drive mechanism.

Based on available GE predictions for reliability of their actuator, and utilizing MIL-Hd-BK-17 for the controls portion, the reliability of subsystem A is predicted for several mission times. Modifying the GE prediction for the CVA and again using MIL-Hd-BK-17 for the proposed controls, the reliability of subsystem B is also predicted for several mission times.

TABLE II - OVERALL RELIABILITY EVALUATION - ACTUATORS AND CONTROLS

Reliability	t = Mission Time (hrs.)		
	t = 1	t = 10	t = 20
Actuator Subsystem A	.997060	.970600	.941200
Actuator Subsystem B	.9 ⁷ 234	.9 ⁵ 234	.9 ⁴ 694
Control Subsystem A	.9 ⁴ 3584	.999358	.998717
Control Subsystem B	.9 ⁴ 8616	.999862	.999723
Total System R _A	.9970	.9706	.9400
Total System R _B	.9 ⁴ 861	.99986	.99972

Note: $9^3 = .999$

It is evident from the results in Table II that:

1. Subsystem B offers reliability advantages that cannot be ignored.
2. The limiting factor in subsystem A appears to be total actuator complexity.
3. The limiting factor in subsystem B appears to be the controls reliability.

The limitation of controls reliability in subsystem B may be obviated by design improvement. As noted in the discussion on controls, no strong effort has been made to design a highly reliable controller. When these subsystems are joined with the couplers, drums and decouplers,

a reliability estimate of the system can be made. A final consolidation of all the reliability evaluations is presented in Tables 12 and 13 for coupled and individual drive systems, respectively, in terms of three mission durations. Any conclusions to be drawn from this comparison must be subjected to critical scrutiny in line with the reservations expressed in the Introduction.

The results in the composite table for the CVA design (Table 12) now show that the reliability of the couplers and decouplers bear significantly on the overall reliability. Using the product rule, it is seen that the reliability of these two items plus the drum essentially determine reliability of the system, and although the controls subsystem reliability could be improved, more effort may be required in other areas.

A strong effort to maximize reliability for the couplers, decouplers, and drums would result in conclusive advantages for the ganged drum system.

It should be recalled that single drum failures imply system failure in this evaluation. The picture does not change, however, even when more than one drum failure is permitted. As shown in Appendix A, a tradeoff between small systems with redundancy and large systems with redundancy still decidedly favors the smaller system, the CVA design.

As expected in the present system, the results of Table 13 show the effect of the overall number of actuators on system reliability.

TABLE 12 - RELIABILITY OF COUPLED CONTROL DRUM DRIVE SYSTEM

<u>Subsystem</u>	<u>Mission</u>	<u>Unreliability x 10⁶</u>	<u>Reliability</u>
3 Actuators (CVA Design) See Table II	1 Hour	.0766	.9999999234
	10 Hours	7.651	.999992349
	20 Hours	40.57	.99996943
Controls See Table II	1 Hour	13.84	.99998616
	10 Hours	138.4	.9998616
	20 Hours	276.8	.9997232
Coupling Device (Direct Straps) See Table 9	1 Hour	80.62	.99991938
	10 Hours	806.20	.9991938
	20 Hours	1,612.40	.998388
12 Decouplers See Table 3	1 Hour	248.73	.9997513
	10 Hours	2,487.30	.997513
	20 Hours	4,974.60	.995025

TABLE 12 (continued)

<u>Subsystem</u>	<u>Mission</u>	<u>Unreliability x 10⁶</u>	<u>Reliability</u>
Control Drums	1 Hour	231.60	.9997684
See Table 10	10 Hours	2,316.00	.997684
	20 Hours	4,632.00	.995368
Composite System	1 Hour	575	.999425
	10 Hours	5,756	.994244
	20 Hours	11,536	.98846

TABLE 13 - RELIABILITY OF INDIVIDUAL CONTROL DRUM ACTUATOR SCHEME

<u>Subsystem</u>	<u>Mission</u>	<u>Unreliability x 10⁶</u>	<u>Reliability</u>
12 Actuators	1 Hour	2,938.9	.99706
(GE Design)	10 Hours	29,389.2	.9706
See Table II	20 Hours	58,778.4	.9412
Controls	1 Hour	64.2	.9999358
See Table II	10 Hours	642	.999358
	20 Hours	1,284	.998716
Control Drums	1 Hour	417.7	.999582
(incl. additional)	10 Hours	4,177.2	.995823
See Table 10	20 Hours	8,354.4	.991646
Composite System	1 Hour	3,421	.99658
	10 Hours	34,208	.9658
	20 Hours	68,417	.9316

APPENDIX A

COMMENTS ON THE USE OF ACTUATOR REDUNDANCY IN DRUM CONTROL SYSTEMS

Some General Considerations

The proposal for providing gang control of drums is based on reductions of controls and actuator complexity and where required, applying redundancy (at lesser complexity) to provide a high level of reliability.

One aspect of this approach is optimization in terms of reliability and complexity. The following considerations apply.

In general, if the system requires a minimum number of actuators, due to a variety of constraints, and if it is known that the system can accommodate as many as N total actuators, then of interest is the difference, which defines the number of redundant actuators used. Assuming independence between actuators, the gain in reliability by using all available space can be determined.

The results in Table A depict reliability gains in various sized systems subject to the actuator reliability, and the minimum number required for operation. It is significant to note from this data that:

1. Gain in reliability is related to a redundancy fraction $\frac{X}{N}$, where X is the number of redundant components in a system of size N . This is evident by the relatively greater reliability improvement in the smaller systems when redundancy is added.
2. As unit reliability, p , increases, the amount of redundancy required to meet a minimum goal decreases at a rapid rate.

These two factors are significant in the optimization process.

A Particular Consideration

The previous discussion dealt generally with some concepts of redundancy, without particular consideration to the actuators, the control drums or the interconnections required.

TABLE A - RELIABILITIES WITH REDUNDANT ACTUATORS

Actuator Reliability	Redundant Actuators	Reliability Determined by X Redundant Actuators out of a Total N		
		N = 24	N = 12	N = 8
0.9	0	.0798	.2824	.4305
	1	.2925	.6590	.8131
	2	.5643	.8891	.9619
	3	.7857	.9743	.9950
	4	.9149	.9957	.9996
	5	.9723		
	6	.9925		
0.99	0	.7857	.8863	.9227
	1	.9761	.9938	.9973
	2	.9983	.9998	.9999
0.999	0	.97630	.98807	.99203
	1	.99970	.99993	.99997
	2	.99999		

If it is desired to apply these general considerations to optimizing actuator designs, one must first answer the question: Do we not already have redundancy in the NRX systems, what with 12 drums and 12 actuators? The answer is maybe, for it has not been demonstrated that the reactor system can stand one or more failures and still meet well defined goals. Thus lack of demonstration and lack of definition of failure may preclude taking advantage of inherent redundancy.

To better understand these implications, consider first of all the assumption that a single drum failure* is catastrophic to the system. Under these conditions there is strong argument for doing everything possible to simplify the design, by reducing the number of actuators, and at the same time, **improve** reliability.

* A drum failure will be defined at this time as any failure of the drum, its linkage or its actuating mechanism which results in inability to control, including conditions of uncontrolled run out or run in.

On the other hand, if more than one drum failure is permissible, the optimization process may be significantly affected. The results of Table A may also be used to demonstrate this situation. For example, in a 12 drum system where three drum failures are permitted, system reliability improves from .8863 (no failure permitted) to .9998 based on a unit reliability of .9900. Thus permitting failures in the present system implies intrinsic redundancy, and in terms of independent failures, it implies a high system reliability.

Pursuing this logic further, perhaps there is too much actuator redundancy in the presently used system, with one actuator per drum.

It was mentioned earlier, that the fraction of redundancy $\frac{X}{N}$ is significant in reliability gains. It follows that a possible tradeoff may exist between the present system with its intrinsic redundancy and a smaller system with actuator redundancy. This possibility was investigated and the results are given below.

A Balance of Complexity and Redundancy

Consider a 12 drum system where it is desired to optimize the number of actuators used. Assume that all actuators have the same reliability and that the minimum number of actuators, L , is chosen by the following rule:

$$L = \frac{12}{n+1}$$

Where n is the number

Where n is the number of permissible drum failures and the brackets symbolize choosing the nearest integer containing the result of the arithmetic indicated.

Then in a system containing L actuators, a single actuator failure would cause system failure. But for n permissible drum failures there exists a balance between a drum system containing L actuators plus some redundancy and a system where there is an actuator for each drum.

Let the original system reliability, R_A , be defined by

$$R_A = 1 - \sum_{k=n+1}^{12} \binom{12}{k} q^k p^{12-k}$$

Where p is the actuator reliability and q is the actuator unreliability.

Now starting with the minimum number of actuator (L) in the simplified system, there is need to add as many as x redundant actuators to obtain a reliability R_B such that $R_B \geq R_A$. The reliability of the simplified system, R_B , is defined by

$$R_B = 1 - \sum_{M=x+1}^{L+x} \binom{L+x}{M} p^M q^{L+x-M}$$

For an assumed actuator reliability of $p = 0.9$, R_A was evaluated, and in turn the number (x) of redundant actuators was determined for $R_B \geq R_A$. The results of this evaluation are given in Table B.

TABLE B - THE NUMBER OF REDUNDANT ACTUATORS, REQUIRED TO OBTAIN $R_B \geq R_A$ WHEN ACTUATOR RELIABILITY, $p = 0.9$

Permissible Drum Failures	System Reliability	Basic Number of Actuators	Redundant Actuators	Total Actuators To Achieve $R_B \geq R_A$
n	$R_A = R_B$	L	x	N
0	.282	12	0	12
1	.659	6	1	7
2	.889	4	1	5
3	.974	3	2	5
4	.996	3	3	6
5	.9995	2	4	6
6	.9999	2	5	7

The data in Table B indicates that, for the conditions imposed and with three or more drum failures permitted, 6 actuators should be used and that connecting these actuators pair-wise would be a good choice. This arrangement would tend to maximize the reliability of the system.

The results given in Table B are based on a reliability of $P = 0.90$. This was done to show the effect of using a relatively poor actuator reliability. If in fact the pneumatic actuator, less the feedback pot is considerably more reliable than one with a feedback pot, (and this has not yet been shown) then the choice of number of actuators is not constrained and becomes limited only by the design of reliable torque override and decoupling features, and the torque capability of the actuators. For example, assume that a design using no feedback pot shows a magnitude improvement in failure probability, i.e., $p = 0.99$, then for system goals as given in Table B the results in Table C are obtained.

TABLE C - THE NUMBER OF REDUNDANT ACTUATORS REQUIRED WHEN ACTUATOR RELIABILITY $P = 0.99$

Permissible Drum Failures	System Reliability	Total Actuators Required For $R_B \geq R_A$
0	.282	12
1	.659	6
2	.889	4
3	.974	3
4	.996	3
5	.9995	3
6	.9999	3

These results indicate a trend to fewer actuators when a more reliable actuator is used. They also tacitly assume the use of the torque override and decoupling features, otherwise a single drum failure or an actuator failure would be tantamount to mission failure under these conditions.

Conclusion

It is evident from this evaluation that even though it is assumed that the present system may contain a considerable amount of intrinsic redundancy, a new actuator system, using as few as three actuators can be at least as reliable as the presently used method by virtue of greater effective redundancy.

Furthermore, the advantage of fewer actuators (with actuator redundancy) is increased when a 24 drum system is considered.

The above noted conclusions make little sense if one downgrades the concept of functional independence. We have postulated override and decoupling features and we have assumed perfect reliability for these devices. It is possible that unreliability of these devices could prevent the implementation of an apparently more reliable system.

APPENDIX B

FAILURE MODE ANALYSIS OF CONTROLLED VELOCITY ACTUATOR

Note: Only those items which differ from the original GE design are covered.

PART NAME Check Valve

PAGE 32

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME F.O. _____

DATE 6-2-65

SKETCH:

MATERIAL:

Possible design.

FUNCTION:

NO. PER ENGINE: 1

To permit gas flow into accumulator, prevent reverse flow.

MODES OF FAILURE:

Valve fails to close completely or fails to open when required.

POSSIBLE CAUSE OF FAILURE:

Sticking of poppet due to friction, distortion or dirt.

RESULTS OF FAILURE:

Lose "fail safe" protection for loss of gas supply. Interfere with "in" movement.

REQUIRED TO PREVENT FAILURE:

Proper design, testing, cleaning, choice of materials.
See General Note No. 6, Page 10.

PART NAME Lock Bellows

PAGE 33

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME F.O. _____

DATE 6-2-65

SKETCH:

MATERIAL:

FUNCTION:

NO. PER ENGINE: 1

1. Hold lock pin in locked position (spring action).
2. Seal internal gases at lock pin penetration.

MODES OF FAILURE:

Rupture of bellows or of joint to adjacent part.

POSSIBLE CAUSE OF FAILURE:

1. Torque applied when pin is raised by turning bushing.
2. Inadequately designed or fabricated for full gas pressure with bellows extended.

RESULTS OF FAILURE:

Continuous loss of gas, loss of coolant flow through actuator resulting in hot spots.

MATERIAL

REQUIRED TO PREVENT FAILURE:

Proper design, quality control. Minimize clearances at bushing to limit gas leakage rate. Minimize friction between pin snap ring and bushing, or use square lock pin.

PART NAME Lock Pin

PAGE 34

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME F.O. _____

DATE 6-2-65

SKETCH:

MATERIAL:

FUNCTION:

NO. PER ENGINE: 1

Lock shaft in sub-critical position

MODES OF FAILURE:

Pin jammed in locked position

POSSIBLE CAUSE OF FAILURE:

1. Galling of pin against mating part.
2. Misalignment of flange

RESULTS OF FAILURE:

Delay in critical start-up, possibly loss of one actuator.

REQUIRED TO PREVENT FAILURE:

Proper choice of contacting materials, satisfactory clearance and finish. Line up flange by pinning or indexing against housing.

PART NAME Stop Soft Spring

PAGE 35

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME F.O. _____

DATE 6-2-65

SKETCH:

MATERIAL:

FUNCTION:

NO. PER ENGINE: 1

Decelerate actuator, drums and ganging assembly at non-destructive rate after scram or other rapid movement.

MODES OF FAILURE:

Excessive overshoot or rebound surge of reactor power after scram. Loosening or yielding of components due to shock.

POSSIBLE CAUSE OF FAILURE:

1. Insufficient or excessive spring stiffness for various conditions, e.g. one actuator inoperative,
2. Improper adjustment of the various actuators to engage springs simultaneously.

RESULTS OF FAILURE:

Undesirable power surge or fluctuation.
Misalignment, backlash develops in drive system.

REQUIRED TO PREVENT FAILURE:

Proper design for all credible conditions. Consider transferring this function to ganging device.

PART NAME 3-way solenoid valve

PAGE 36

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME F.O. _____

DATE 6-2-65

SKETCH:

MATERIAL:

FUNCTION:

NO. PER ENGINE: 2

Connect one cylinder to gas supply or exhaust in order to move or stop piston.

MODES OF FAILURE:

POSSIBLE CAUSE OF FAILURE:

Failure of electrical insulation in coil.

Overheating if valve is energized before coolant gas flow is initiated. Particularly applicable if one valve is reversed to cause actuator to fail in scrambled position on loss of power (i.e. normally energized).

RESULTS OF FAILURE:

Inability to move piston toward burned-out coil except by (external) scram.

REQUIRED TO PREVENT FAILURE:

Design conservatively, provide heat sink next to coil. See General Note No. 6, Page 10.

PART NAME 3-Way Solenoid Valve

PAGE 37

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME F. O. _____

DATE 6/2/65

SKETCH:

MATERIAL:

See Page 36

FUNCTION:

NO. PER ENGINE: 2

See Page 36

MODES OF FAILURE:

Cracking of external armature.

POSSIBLE CAUSE OF FAILURE:

Thermal stress due to gradient between cryogenic coolant and heating by I²R and gamma, with concurrent loss of ductility at low temperature.

RESULTS OF FAILURE:

Loss of supply pressure into cooling passages may cause loss of actuator function. Short circuiting of coolant may cause local hot spots.

REQUIRED TO PREVENT FAILURE:

Choice of material ductile at low temperature. Avoid notches, sharp corners, heavy sections.

PART NAME 3-Way Solenoid Valve

PAGE 38

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME F. O. _____

DATE 6/2/65

SKETCH:

MATERIAL:

See Page 36

FUNCTION:

NO. PER ENGINE: 2

See Page 36

MODES OF FAILURE:

Valve jammed in either position or partly open.

POSSIBLE CAUSE OF FAILURE:

Particle trapped between seat and disc. See general note No. 3, Page 9.

RESULTS OF FAILURE:

Loss of actuator function. Several units could fail during same run if caused by dirt in supply gas.

REQUIRED TO PREVENT FAILURE:

See general notes No. 3, 4, and 6, Pages 9 and 10.

PART NAME 3-Way Solenoid Valve

PAGE 39

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME F. O. _____ DATE 6/2/65

SKETCH:

MATERIAL:

See Pages 36 and 38

FUNCTION:

NO. PER ENGINE: 2

See Page 36

MODES OF FAILURE:

Excessive pressure drop through valve. Or, disc will not move when coil is energized.

POSSIBLE CAUSE OF FAILURE:

Valve gap too large or too small. Reasons: Differential thermal expansion; fabrication and assembly tolerances specified or executed too wide. Note: Flaw gap at valve seat reportedly will be .002" for flat seat design.

RESULTS OF FAILURE:

Actuator too sluggish or inoperative.

REQUIRED TO PREVENT FAILURE:

Check and enforce close tolerances. Choose materials with similar thermal expansion, considering coefficient as well as local temperature difference. See general notes No. 4 and 6, Page 10.

PART NAME 3-Way Solenoid Valve

PAGE 40

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME _____

F. O. _____

DATE 6/2/65

SKETCH:

MATERIAL:

See Pages 36 and 38

FUNCTION:

NO. PER ENGINE: 2

See Page 36

MODES OF FAILURE:

Gas leakage through closed valve port.

POSSIBLE CAUSE OF FAILURE:

Scratched seat surface. Poor finish on seat.
See also Page 38. Gasket leak between valve
chamber and control volume.

RESULTS OF FAILURE:

May oppose piston movement if defect is of same magnitude as orifice size.

REQUIRED TO PREVENT FAILURE:

Thorough inspection of parts. Hydrostatic or pneumatic pressure tests across seats.
See general notes No. 4 and 6, Page 10.

PART NAME Orifice

PAGE 41

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME _____

F. O. _____

DATE 6/2/65

SKETCH:

MATERIAL:

FUNCTION:

NO. PER ENGINE: 2

Limit gas exhaust flow in order to produce constant piston velocity in each direction.

MODES OF FAILURE:

Excessive or insufficient gas flow rate.

POSSIBLE CAUSE OF FAILURE:

Wrong size of hole. Analytical predictions are generally inaccurate.

RESULTS OF FAILURE:

Reactor control ramp too steep or too shallow. Possibility of loss of one errant actuator.

REQUIRED TO PREVENT FAILURE:

Design the orifices to be adjustable. Calibrate following assembly in actuator. See General Notes No. 5 and 6, Page 10.

PART NAME Orifice

PAGE 42

PART NUMBER _____

ENGINE CVA

PLANNING PART NO. _____

NAME F. O.

DATE 6/2/65

SKETCH:

MATERIAL:

See Page 41

FUNCTION:

NO. PER ENGINE: 2

See Page 41

MODES OF FAILURE:

Partial or total loss of gas flow.

POSSIBLE CAUSE OF FAILURE:

Plugging of hole by entrained particle (dirt, chip, ice, corrosion product).

RESULTS OF FAILURE:

Loss of actuator function. Possibility of several actuators failing in same run if general dirt or moisture contamination of system, causing loss of reactor control.

REQUIRED TO PREVENT FAILURE:

High degree of cleanliness required.

Orifice to be removable or accessible for easy inspection and cleaning. See general notes No. 3, 5, and 6, Pages 9 and 10.

APPENDIX C

RELIABILITY ASPECTS OF ACTUATOR POTENTIOMETERS

A preliminary review of the failure history of Markite potentiometers has been made. The purpose of this review is to establish background and data for estimating the reliability of these potentiometers, in conjunction with the analysis of various actuator configurations.

The G.E. actuator design uses Markite potentiometers whose basic features are similar to those used in the NRX test series. A significant difference, however is the provision for cooling used in these potentiometers.

The review of data from the NRX-A2 and NRX-A3 tests reveals the following: A total of six failures were noted. All of these were found during checkout. No failures occurred during tests. All failures were apparently related to either checkout equipment or operator difficulties.

The Actuator Development Group estimates conservatively that each potentiometer acquires at least 40 hours of operation from the time of receiving inspection until completion of the test series. Assuming that none of the six failures (as noted in Table D below) are valid, then the NRX potentiometers have a minimum MTBF of $40 \times 24 = 960$ hours.

Table D

NRX-A ACTUATOR POT FAILURES

<u>Date</u>	<u>QRDR</u>	<u>No. of Pots</u>	<u>Test Phase When Failure Occurred</u>	<u>Cause</u>
29 Jan. 65	00710	2	Checkout at R-Mad	Operator error - too high voltage applied
	00717	2	Checkout at R-Mad	Operator error - too high voltage applied

TABLE
APPENDIX C - Table D (continued)

<u>Date</u>	<u>QRDR</u>	<u>No. of Pots</u>	<u>Test Phase When Failure Occurred</u>	<u>Cause</u>
22 Feb. 65	00756	2	Checkout on car at R-Mad	Wiring change on test car resulting in high voltage on pot wiper.
5 Feb. 64	-	1	C/O on car	Unknown - Pot checked at WANL revealed burned spot.

Under operating conditions of a similar nature the potentiometers used in the G.E. actuator have not fared as well. Table E below shows the failure history of the "flat track" potentiometer first used in the actuators. This data clearly shows deficiencies, with an apparent MTBF of 45 hours.

As noted in Table E a principal cause of failure is contamination.

Table E

OPERATING HISTORY OF THE FLAT TRACK DESIGNS

<u>Unit</u>	<u>Hours Operated</u>	<u>Comments</u>
Actuator No. 1 pot	42.5	Actuator sent to SNPO. Pot still working.
Actuator No. 2 pot	62.6	Removed because of dirt on track. Sent to Markite for rework.
Actuator No. 3 pot	59.81	Removed for rework at Markite although pot was still functioning O.K.
Actuator No. 4 pot	34.57	Removed for use in actuator no. 5. Pot was still functioning O.K.
Actuator No. 4 pot used in Actuator No. 5	23.3 here 57.87 total	Total running time 57.87 hrs. Evidence of dirt on track after 47.47 hrs. usage and again after 56 hrs. Finally removed and sent to Markite for rework.

APPENDIX C - Table E (continued)

<u>Unit</u>	<u>Hours Operated</u>	<u>Comments</u>
Actuator No. 5 pot	35.51	Removed for rework at Markite although pot was still functioning O.K.
Actuator No. 6 pot used in Actuator No. 2	14.2	Removed during vibration test because of evidence of dirt on track. Sent to Markite for rework.

A change in potentiometer design was implemented, resulting in the "crown track" models. Results of testing on this model are given in Table F below.

Table F

OPERATING HISTORY OF THE CROWN TRACK DESIGN

<u>Unit</u>	<u>Hours Operated</u>	<u>Comments</u>
Actuator No. 2 pot S/N 3U-1634	43.9	Indication of open circuit in 1 deck. Disassembled at Markite and found to have improper assembly.
Actuator No. 3 pot S/N 3U-1532-2	32.4	Pot still in use.
Actuator No. 4 pot S/N 3U-1635	226.1	First used on life test (Actuator No. 4) for 196.1 hrs. then used in actuator No. 3 for 30 hrs. (Hydrogen test). Removed because of noisy output.
Actuator No. 6 pot S/N 3U-1630-4	73.11	Used for cryogenic test for 65.53 hrs. then put in actuator No. 5 for 7.58 hrs. Pot still O.K.
Actuator No. 5 pot S/N 3U-1595-10		Pot not in use at this time. No hours accumulated.

Significant improvement is evident in the longevity of individual crown track potentiometers. These potentiometers exhibit a minimum MTBF of 114 hours, exceeding by at least a factor of two the best of the flat track designs, as given in Table E.

Thus with three potentiometer designs we have:

<u>Potentiometer Type</u>	<u>Apparent MTBF (Hours)</u>
NRX	960 (Minimum)
Flat Track	43
Crown Track	114 (Minimum)

Since even the "Crown Track" design showed evidence of increased noise output, as evidenced in Table F, it was requested that this characteristic be compared with the NRX types. This data was not available. The Actuator development group is proceeding to obtain this and other post test data for both NRX-A2 and NRX-A3 potentiometers.

If it is assumed that the NRX types are within specification then some question of the reliability of the crown track design may remain. Along these lines, it may be advisable to determine more precisely the effects of open cooling in the G.E. design potentiometers and take corrective action to eliminate contamination. On the other hand, a wear characteristic not noticeable until after say 200 hours of operation would not be significant in applications of 40 hours or less.

Based on this preliminary review, the conclusion that potentiometer reliability will not be a problem is reasonable. A reliability goal of greater than 2,000 hours MTBF is feasible.

Reliability growth for the crown track design, especially through removal of the effects of contamination and through operation in the expected environment, should substantiate this conclusion.

APPENDIX D

NRX-A3 CONTROL DRUM WELD FAILURE ANALYSIS

General Notes

1. Failure of weld during NRX-A3 test series may have been caused as described on Page 48. A temperature gradient of only 130° R between end cap and cylinder can initiate thermal stress conditions, and this was undoubtedly exceeded during cooldown. Yield strength of the dome end weld could be exceeded during a transient that produces a thermal profile such as this:

Cylinder mean temperature	450° R
Housing mean temperature	440° R
Nozzle end cap temperature	300° R

2. "Bottoming" of the cylinder against nozzle end cap or against solidly compressed spring may have been responsible for the past failure (Pages 51 and 52), but only if combined with a poor weld. Using A-3 conditions and as-built overall length, the weld could barely be stressed to its yield strength. It should be noted, though, that drawings and inspection procedures did not limit deviation from overall reference dimension (52.882"). The initial gap might therefore have been smaller than intended in some drums, resulting in higher stresses. A tolerance has now been specified for future reactors.
3. There is evidence at this time that the failure under discussion was contributed to by weld defects. It must be pointed out that present inspection methods do not permit a conclusive evaluation of the weld as completed. Should radiographic or ultrasonic inspection be required on future welds, it will be necessary to change weld preparation design as well as assembly procedure.

Even with a sound weld, it must be assumed that the T6 temper is lost in the heated zone, with a corresponding drop in ambient yield strength from 40,000 psi to 8,000 psi. This applies especially to a repair weld (such as the one that failed), where three passes are made and a lower strength filler metal is introduced. Post-weld heat treatment would not be advisable because of the danger of distortion and destructive thermal stress.

4. The forms on the following pages indicate steps that would be required to prevent specific modes of failure. Their adoption may not be justified on presently completed assemblies, especially if the consequences of weld failures are not considered serious.

PART NAME Control Drum HousingPAGE 48

PART NUMBER _____

ENGINE _____

PLANNING PART NO. _____

NAME F. O. _____

DATE 6/29/65SKETCH:MATERIAL: Aluminum 6061-T6

Dome End

Housing

Be Cylinder

Nozzle End

Coolant Flow

FUNCTION:NO. PER ENGINE:

Structural member between bearings. Contains fragments in case of fracture. Channels coolant flow over boron plate.

MODES OF FAILURE:

Tensile failure at dome end weld.

POSSIBLE CAUSE OF FAILURE:

Nozzle end of housing seizes Be cylinder due to radial contraction of (colder) nozzle end cap. Subsequent differential axial contraction between housing and cylinder produces high tensile stress. Failure at weld because thinnest (.030"), weakest (annealed) part.

RESULTS OF FAILURE:

Rotation of severed housing relative to Be cylinder could change coolant flow pattern, aggravate bowing problem. Possibility of particle formation during rupture or subsequent abrasion.

REQUIRED TO PREVENT FAILURE:

Reduce O. D. of Be cylinder at nozzle end (see also Page 50.)

PART NAME Control Drum Housing

PAGE 49

PART NUMBER _____

ENGINE _____

PLANNING PART NO. _____

NAME F. O. _____

DATE 7/2/65

SKETCH:

MATERIAL: Aluminum 6061-T6

FUNCTION:

NO. PER ENGINE:

See Page 48.

MODES OF FAILURE:

Failure at either weld by torsional or tensile loading within design limits.

POSSIBLE CAUSE OF FAILURE:

Defective weld - incomplete penetration, sub-surface crack, or other flaw not evident by present inspection methods.

RESULTS OF FAILURE:

See Page 48.

REQUIRED TO PREVENT FAILURE:

Redesign weld preparation to provide lip or trepan on end caps. Specify 100% radiography of welds.

PART NAME Control Drum Housing

PAGE 50

PART NUMBER _____

ENGINE _____

PLANNING PART NO. _____

NAME _____

F. O. _____

DATE 7/2/65

SKETCH:

MATERIAL: Aluminum 6061-T6

FUNCTION:

NO. PER ENGINE:

See Page 48.

MODES OF FAILURE:

Failure at either weld by bending or shear.

POSSIBLE CAUSE OF FAILURE:

Differential radial contraction of cylinder and end cap during cooldown due to:

1. Different rate of contraction in the two materials.
2. Difference in temperature associated with heat transfer characteristics and relative location.

RESULTS OF FAILURE:

See Page 48.

REQUIRED TO PREVENT FAILURE:

Reduce O. D. of Be cylinder at both ends (See also Page 48.)

PART NAME Control Drum Housing

PAGE 51

PART NUMBER _____

ENGINE _____

PLANNING PART NO. _____

NAME _____ F. O. _____ DATE 7/6/65

SKETCH:

MATERIAL: Aluminum 6061-T6

FUNCTION:

NO. PER ENGINE:

See Page 48.

MODES OF FAILURE:

Tensile failure at either weld.

POSSIBLE CAUSE OF FAILURE:

Differential axial contraction during cooldown causes contact between cylinder and end cap, then builds up tension in housing.

Occurs only if initial gap is too small - possibly as a result of weld repair procedures.

RESULTS OF FAILURE:

See Page 48.

REQUIRED TO PREVENT FAILURE:

Assure adequate gap by proper dimensional inspection during assembly. Specify tolerance on dimension 'X' after welding.

PART NAME Control Drum Housing

PAGE 52

PART NUMBER _____

ENGINE _____

PLANNING PART NO. _____

NAME _____

F. O. _____

DATE 7/6/65

SKETCH:

MATERIAL: Aluminum 6061-T6

See Page 51.

FUNCTION:

NO. PER ENGINE:

See Page 48.

MODES OF FAILURE:

Tensile failure at either weld.

POSSIBLE CAUSE OF FAILURE:

Same as on Page 51 - except that contact is established through solidly compressed spring.

RESULTS OF FAILURE:

See Page 48.

REQUIRED TO PREVENT FAILURE:

Same as on Page 51.