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CANNON LAUNCHED ELECTROMECHANICAL CONTROL ACTUATION SYSTEM DEVELOPMENT

N83 24893

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ABSTRACT

The evolution of an electromechanical control actuation system (EM-CAS) from trade study results through breadboard test and high-g launch demonstration tests is summarized in this report. Primary emphasis is on design, development, integration and test of the gear reduction system.

INTRODUCTION

Future small missiles will likely use infrared (IR) or radar frequency seekers and are expected to operate over increased ranges. Size, weight, and bandwidth limitations of pneumatic control actuation systems will almost certainly compromise the design of the missile system and necessitate a close examination of possible CAS alternatives.

Trade studies conducted at Martin Marietta Aerospace during 1981 for a Direct Fire Projectile Study resulted in identifying the electromechanical control actuation system (EM-CAS), shown in Figure 1, as the most likely candidate for improved performance at lower weight and cost compared to pneumatic and hydraulic systems. Proper selection of gearing and gear ratios to match load requirements to the dc motor characteristics was important in optimizing the system.

During the last half of 1981, Martin Marietta undertook a program to design and fabricate a breadboard electromechanical actuator which successfully proved performance feasibility.

The 1982 CAS development program was the direct result of CAS Trade Studies, breadboard development, and computerized performance simulation completed in 1981. The long range objective is to develop a baseline design for a family of small missile electromechanical control actuation systems. Specific objectives for 1982 were:

1. To establish and optimize the component parts and assembly configuration of a prototype EM-CAS.
2. To demonstrate the performance capability of the prototype CAS in laboratory tests.
3. To demonstrate the suitability of the prototype CAS for small missile applications by exposure to a high-g launch environment.

During 1982, a prototype EM-CAS for a 155 mm cannon launched projectile was designed, fabricated, tested, and canister launched at temperatures from -45°F to +145°F at 9000 to 10,000g's acceleration with excellent results. Selection of gear reducer types and ratios was of primary importance in ensuring hardness to withstand high acceleration and ensuring a high efficiency gear train to minimize energy consumption from the missile battery system. Reducer trade studies comparing spur gear, ball screws, worm gears, spiroid gears and gear combinations were made with a compromise of acceleration hardness, energy consumption, and complexity to optimize production cost. Design of the dc servomotor and brake, as well as packaging the electronics, was also influenced by the high acceleration hardness requirements.

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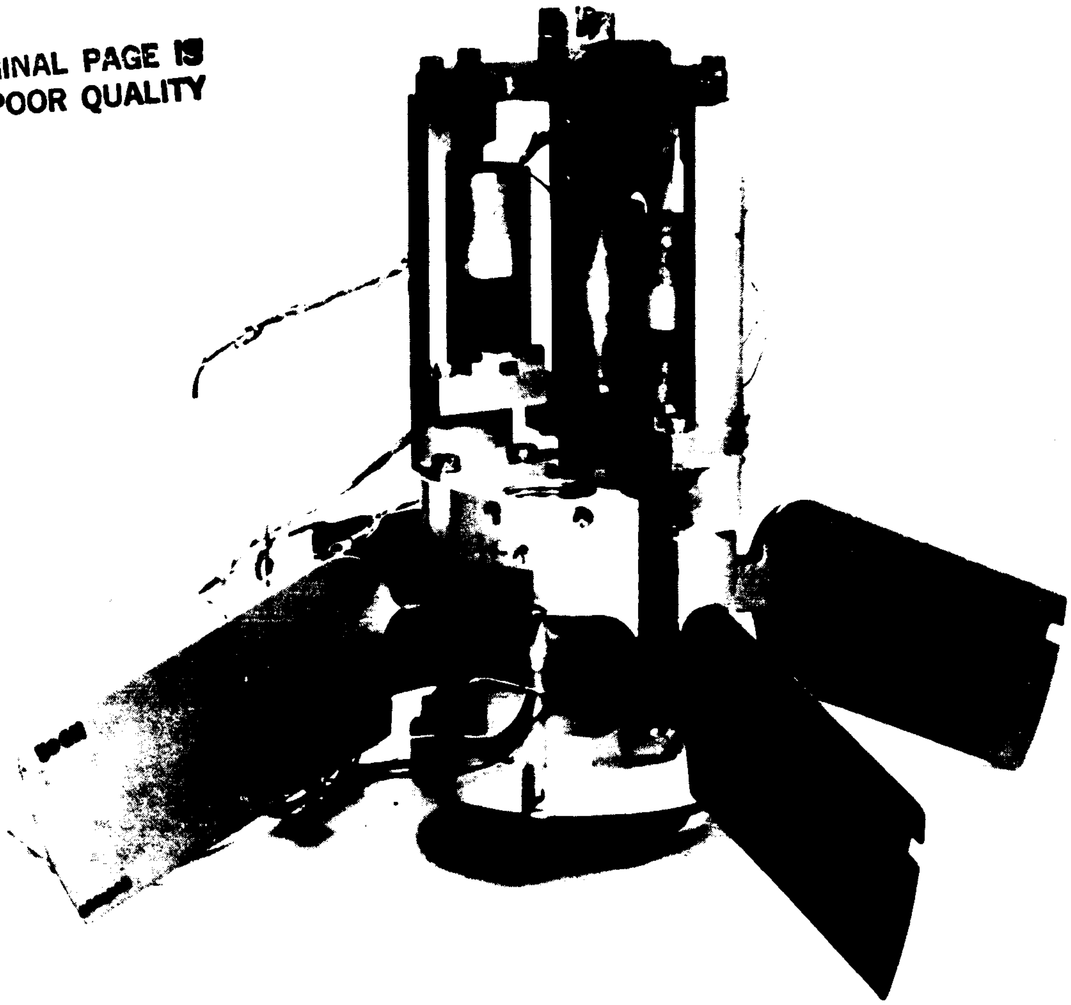

Figure 1. EM-CAS

The ability of an EM-CAS to meet the requirements of small guided projectiles has been demonstrated through detailed analysis, computer simulation, breadboard testing, and five prototype 10,000g canister launches. Breadboard testing using motors from 3 vendors demonstrated that the EM-CAS could meet bandwidth and slew rate requirements for a typical small missile (155 mm). Power analysis and bench tests demonstrated that power requirements are not prohibitive for a direct drive servomotor EM-CAS.

Overall, an EM-CAS using a samarium cobalt servomotor will compete favorably with pneumatic and hydraulic systems because of its inherent simplicity, reduced weight and envelope, and increased bandwidth (over pneumatic). Cost studies indicate that the EM-CAS will also be price competitive.

BREADBOARD DEVELOPMENT

The objectives of breadboard development were to establish EM-CAS technology and expand and verify the EM-CAS data base. To meet these objectives, the breadboard was designed with maximum flexibility for accepting different motors and gear reducer configurations (spur gear, worm gears, ball screws, etc.) while incorporating tailored electronics.

The mechanical assembly shown in Figure 2 consists of existing projectile pitch shaft fin, bearing, and feedback potentiometer; as well as a tension spring loading fixture and adjustable mounting brackets which accept different motors and gear reducers.

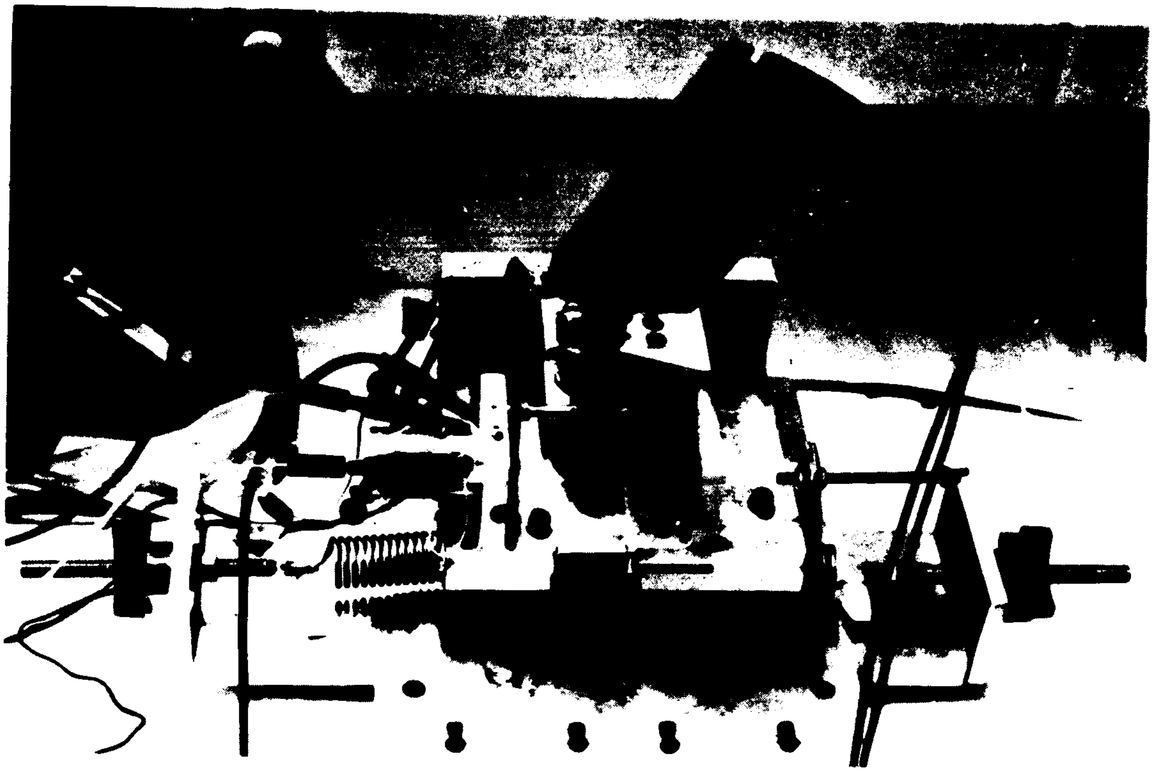

Figure 2. Mechanical Assembly

Tests were performed on two brush motors (Simmonds Precision and Inland Motors) and one brushless motor (MPC). Gear ratios were selected for each motor based on motor speed/torque capability and on a compromise between output load and frequency response. Tests were run for each motor to find frequency response, step response, and stall torque. Frequency response of 22 to 29 Hz, fin rates of 500 degrees per second, and stall torques up to 31.4 Nm (278 in lb) were demonstrated.

EM-CAS PROTOTYPE REQUIREMENTS

EM-CAS prototype requirements, primarily based on 155 mm projectile specifications, are summarized in Table I. Using these prototype requirements, dc motor/brake requirements were established (Table II).

EM-CAS SYSTEM DESCRIPTION

A photograph of the electromechanical actuator for a 155 mm cannon launched projectile is shown in Figure 1. Three independent axis of control (pitch, yaw plus roll, and yaw minus roll) requires three motor and gear train combinations. For expediency, simplicity, and lower cost, it was decided that all three motor and gear reducers are identical even

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Table I. Prototype Requirements

Stall Torque (min per fin)	6.22 Nm (55 in lb)		
Loaded Rate (5.09 Nm per fin min)	120 deg/sec		
Shaft Deflection-Yaw/Roll Axes	± 22 deg		
Pitch Axis	± 17 deg		
Duty Cycle Duration(-25°F to +145°F) (all fins)			
Command	**Offset**	**Torque Rate/Fin**	**Duration**
d_c = 3 Sin 20t	2.33 deg	0.339 Nm/deg (3 in lb/deg)	50 sec
d_c = 5 Sin 20t	2.33 deg	0.339 Nm/deg (3 in lb/deg)	20 sec
		Total	70 sec

Frequency Response (±2.5 deg command)

Axis	Offset	Torque Rate	Frequency Response
Pitch	10 deg	0.283 Nm/deg (2.5 in lb/deg)	12.1 Hz min at -80° lag
Yaw/Roll	15 deg	0.142 Nm/deg (1.25 in lb/deg)	14.32 Hz min at -75° lag

Environment

Temperature (soak)	-25°F to +145°F
Set Back	10,000g
Set Forward	1,900g
Lateral	750g
Angular	75,000 rad/sec^2
Radial	345g
Electrical Power Requirements	Use Existing Battery (≅8600 Joules allowance)

Table II. Torque Motor and Brake Performance Requirements

Parameter	Symbol	Unit
Torque (rated)*	T_r	0.106 Nm (15 in oz)
Current (armature rated)*	I^P	6.5A (6.5 amps)
Voltage (armature rated)	V^P	25V (25 volts)
Speed (rated)*	RPM	6000 min-1 (6000 rpm)
Maximum Applied Armature Voltage	V_{tm}	35V (35 volts)
Maximum Applied Armature Current	I_{tm}	16.7A (16.7 amps)
Torque constant*	K_t	0.0162 Nm/A(2.31 in oz/amp)
Rotor Inertia (max)	J_m	21.2g cm^2(3x10^{-4} in oz-sec^2)
Armature Resistance at 25°C	Ra	2.1Ω (2.1 ohms)
Coulomb Friction (max)	Tc	0.00752 Nm (1.064 in oz)
Envelope (motor and brake)		3.175 cm Ø x 7.62 cm long (1.25 in Ø x 3 in long)
Brake Torque	T_b	0.071 Nm (10 in oz static dynamic)
Brake Voltage (dc)	V_b	30 ± 5V (30 ± 5 volts)
Brake Reaction Time (max)	t_B	0.010 sec (0.010 sec)
Brake Current (max) (25°C)	I_B	0.150A (0.150 amps)
Brake Inertia (max)	J_B	1.13g cm^2(0.16x10^{-4} in oz-sec^2)
Environment - As per Table I		

*Worst case tolerances and temperatures

though the pitch axis requirements were essentially twice the yaw/roll axis. The fins, fin shaft and bearing arrangements, feedback potentiometers, battery location, and all other useable parts that could be retained from a prior 155 mm missile system were used. The spur gear/worm gear combination was selected for the gear reducer because of breadboard demonstration, hardness, availability, low cost, and other lesser trade study criteria.

The electronics was packaged in the aft end of the control housing due to envelope and structural constraints and because a large inherent heat sink is available. Photographs of the major components and the complete assembly with fins folded are shown in Figures 3 and 4.

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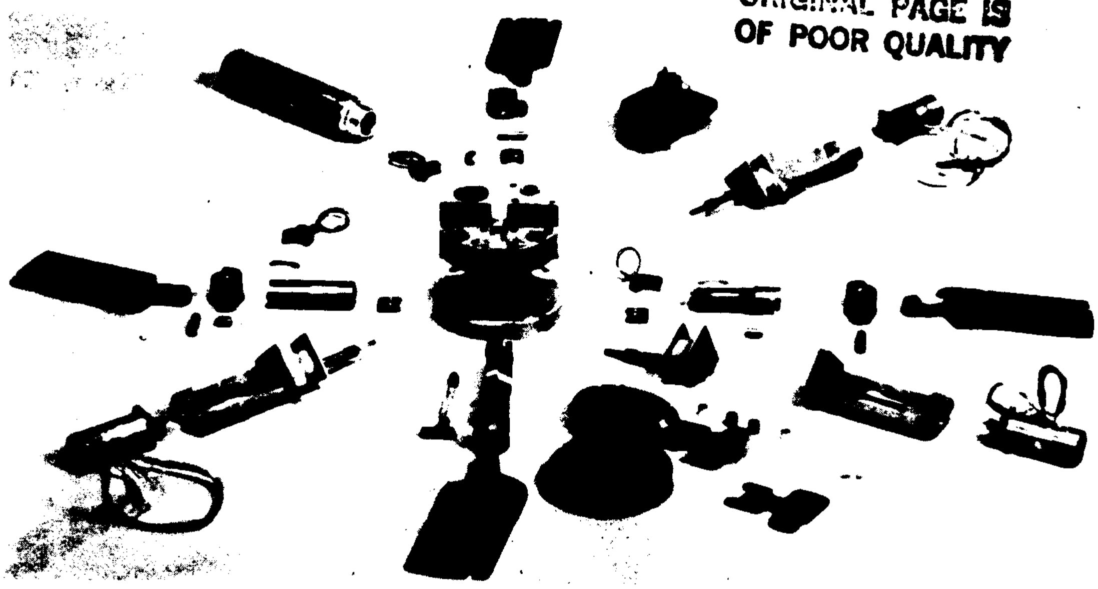

Figure 3. Major Components

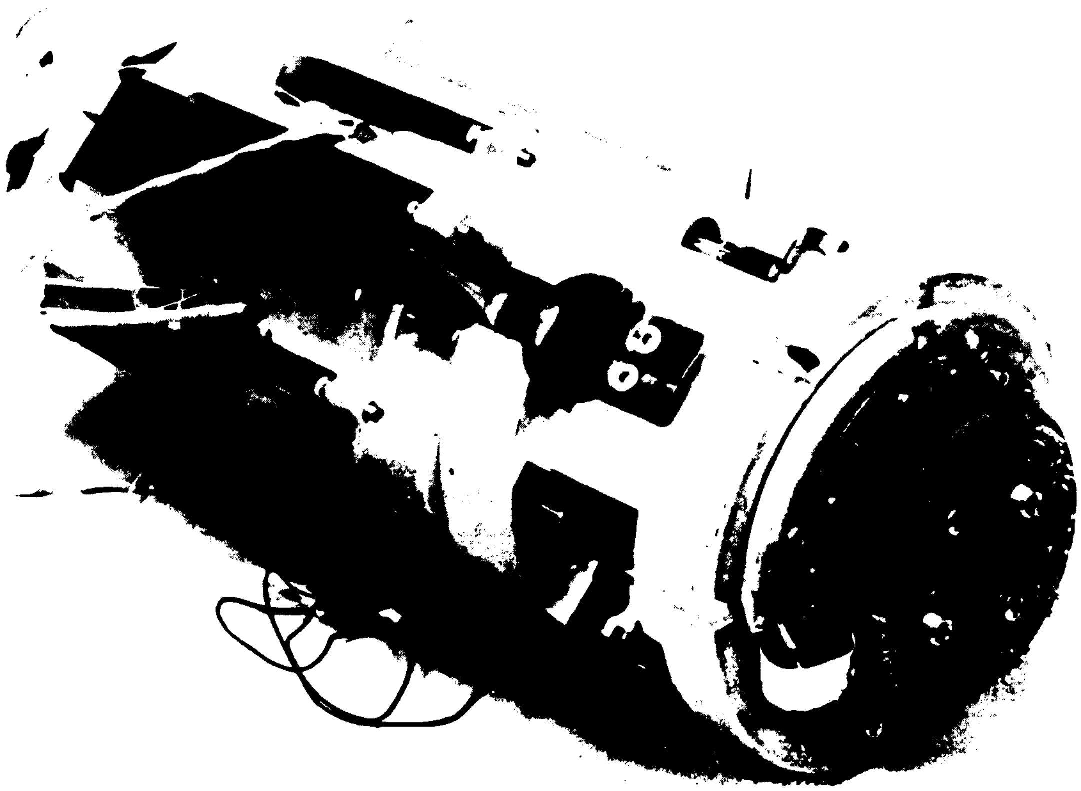

Figure 4. EM-CAS Electronics End with Fins Folded

GEAR REDUCER

The gear reducer consists of a 3.89/1 aluminum spur gearbox (Figure 5) which drives a worm gear segment with a two-threaded hardened steel worm. The overall gear ratio from motor shaft to fin output shaft is 194.44/1.

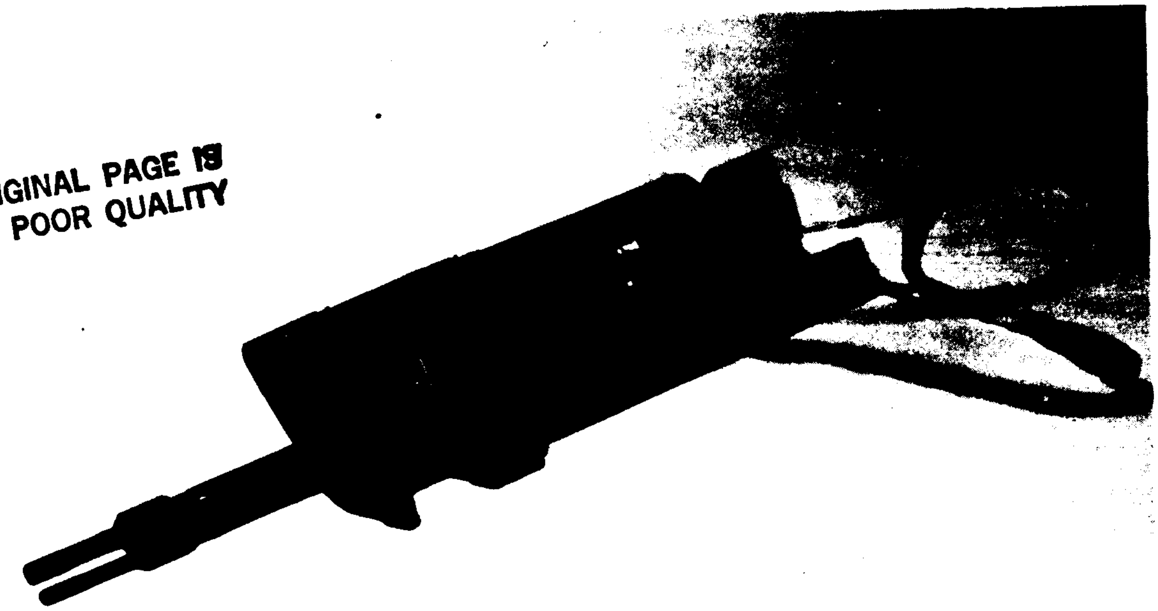

Figure 5. Gear Reducer

All spur gears are 2024-T4 aluminum, AGMA-12 precision from Pic Gear Corp. The worms and worm gears were purchased from Reliance Gear Co., Ltd. Worm gear material is QQ-S-763 class 10 steel (440B), and the worm is heat treated 1117 steel.

The gearbox was designed to provide flexibility in changing gear ratio and to minimize backlash on the output worm shaft. Gearbox ratios from 2/1 to 7/1 are feasible permitting selection of overall reducer ratios from 100/1 to 350/1. A gearbox ratio of 3.89/1 (194.44/1 overall) has proven optimum for the selected motor.

The bearings (KP3AL and KP3A) used in the gearbox were standard aircraft ball bearings supplied by TRW. The KP3A bearing was installed to carry the forward and aft thrust loads of the worm shaft in addition to radial loads. Any axial movement of the worm shaft translates directly into backlash on the output fin shaft. To minimize axial movement, the shaft was shimmed tightly on each side of the KP3A thrust bearing, and the bearing was installed with a 0.0178 mm (0.0007 in) mean tight fit in the aluminum housing and a 0.0076 mm (0.0003 in) mean tight fit on the steel shafts. This installation reduced internal bearing axial movement from 0.076 mm (0.003 in) to 0.013 mm (0.0005 in) as demonstrated by tests. Additional assurance of minimizing worm shaft axial movement was obtained by installing a disk spring beneath the KP3AL gearbox bearing and pre-loading the worm shaft to 68 Kg (150 lb) of thrust load.

Additional investigation to minimize backlash considered the required gear precision and centerline location tolerance. Results of this analysis are shown in Figure 6 where fin backlash is plotted against axis center tolerance for AGMA-10, AGMA-12, and AGMA-14 gears. Maximum backlash tolerated by the actuator fin shaft was established at 0.25 degree.

This analysis assumes no play in the gear shaft bearing. As a result of this analysis, AGMA-12 gears and 0.051 mm (0.002 in) axis center tolerance were selected.

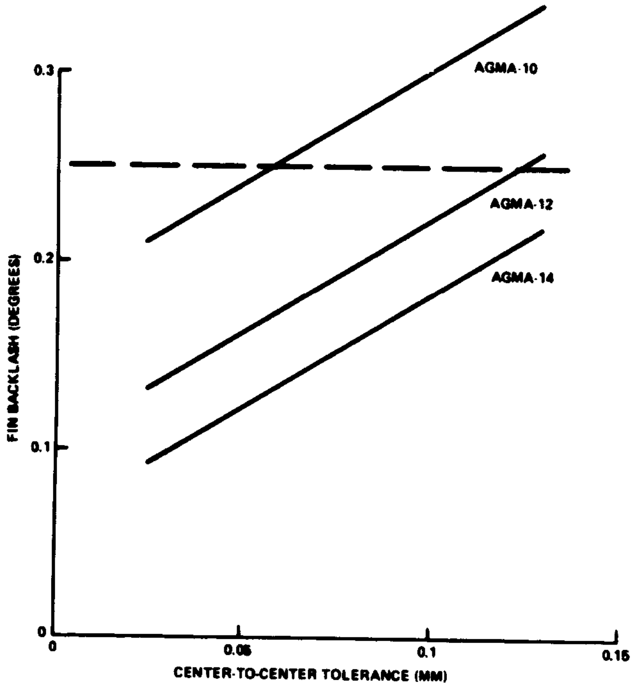

Figure 6. Centerline Tolerance Effects on Backlash

The spur gears and worm gear set are lubricated with FEL-PRO C100 grease which contains stabilized molybedinum disulfide and lead in an organic viscus carrier. The ball bearings were supplied presealed and prepackaged with lubricant conforming to MIL-G-23827.

Gear Reducer Trade

A trade study of various gear reducers was made for the purpose of designing the cheapest actuator which is capable of operating for a minimum of 70 seconds at the specified duty cycles and using an existing system battery. Results of this trade study are shown in Table III.

The single worm, the lowest cost and least efficient reducer, will only give 58 seconds of flight duration. The double-threaded worm will provide 70 to 82 seconds of flight duration and costs slightly more than the single worm. The four-threaded worm concepts offer little increased flight duration to justify the cost increase. The spiroid gear concept gives less duration for a higher price; however, it has the positive feature of almost zero backlash. The spur/bevel gear will achieve the same flight duration as the ball screw (max of 85 to 88 seconds) for a lower cost; however, it probably cannot be packaged in the available envelope. The ball screw is too expensive and will probably not withstand the 10,000g setback launch loads. Cost and high packaging risk eliminates the spur/bevel gear from further consideration. For these reasons, the double-threaded worm was selected for the EM-CAS prototype.

Table III. Reducer Trade

		SPUR/BEVEL	BALL SCREW	SINGLE WORM	2-WORM	4-WORM	SPIROID	4-WORM
DIA PITCH:	OUTPUT GEAR	0.52917M (48 dP)	–	0.52917M (48 dP)	0.52917M (48 dP)	0.52917M (48 dP)	–	1.05833M (24 dP)
	OTHER GEARS	0.26458M (96 dP)	0.26458M (96 dP)	0.26458M (96 dP)	0.26458M (96 dP)	0.26458M (96 dP)	0.26458M (96 dP)	0.26458M (96 dP)
RATIO:	NOMINAL	200/1	200/1	200/1	200/1	200/1	200/1	200/1
	MAXIMUM OBTAINABLE	250/1	248/1	600/1	300/1	250/1	400/1	250/1
STAGES		4	2	2	2	3	2	3
WEIGHT (Kg) (COMPONENTS ONLY) (LB)		0.837 (1.845)	0.313 (0.69)	0.472 (1.04)	0.476 (1.05)	0.658 (1.45)	0.680 (1.50)	0.680 (1.50)
COST $/CAS (PARTS ONLY)		$258	$443	$105	$168	$205	$250	$205
COST $/CAS		+153	+338	0	+63	+100	+145	+100
MAXIMUM MOTOR (NM) TORQUE (IN OZ)		0.088 (12.46)	0.0866 (12.26)	0.165 (23.32)	0.124 (17.54)	0.1074 (15.2)	0.2304 (32.6)	0.1023 (14.48)
MAXIMUM CURRENT LIMIT (AMPS)		5.24	5.16	9.81	7.38	6.39	13.71	6.09
BATTERY ENERGY (JOULES)		7071/ 6839	7038/ 6869	10.436/ 7916	8662/ 7324	7936/ 7100	10522/ 10383	7686/ 7068
DURATION-MIN (SEC)		85	85	58	70	76	57	78
-MAX (SEC)		88	88	76	82	85	58	85
BACKLASH – MAX (DEG)		0.05	0.20	0.20	0.26	0.3	0	0.3
10,000g SENSITIVE		LOW	VERY HIGH	LOW	LOW	LOW	LOW	LOW
PACKAGING RISK		HIGH	MED.	LOW	LOW	MED	HIGH	MED
SELECTION		2	6	3	1	4	7	5

Gear Reducer Efficiency Analysis

Gear efficiency prediction is inexact because surface finish, lubricant, temperature, rubbing speed, accuracy of teeth, and installation all influence gear effectiveness in transferring power from one gear to the other.

According to Mark's Engineering Handbook, the efficiency of worm gearing is approximately (dependent on thread angle (ψ) and coefficient of friction (f)) as follows:

$$\text{Efficiency, } E = \tan\psi \, \frac{(1 - f \tan\psi)}{\tan\psi + f} \qquad \psi = \text{Thread Helix Angle}, \quad f = \text{Coefficient of Friction}$$

Friction data from several sources using different lubricants indicate that friction factors for hard steel rubbing on hard steel with pressures of 2.76×10^6 KPa (400 KSI) are 0.058 for graphite and 0.033 for molydisulfide. A grease consisting of molydisulfide and graphite was selected for this application.

For cylindrical worm gears (EM-CAS type) the variation of friction coefficient with rubbing speed for a carburized and ground steel worm and phosphor bronze gear was shown to vary from 0.08 to 0.03 for rubbing speeds from 12 to 254 cm/sec.

Using friction coefficient of 0.03 to 0.08 as boundaries, worm gear efficiency is plotted versus helix angle (ψ) in Figure 7. Breadboard test data for two different helix angles (4°46' and 9°28') are also shown for comparison. Based on this data, EM-CAS worm gear efficiency of 60 percent to 80 percent is predicted.

Schematically, the gear train consists of four spur gears driving a worm gear set as shown in Figure 8. Assuming worm gear efficiencies predicted in Figure 7, overall gear reducer efficiency (from motor input to aerodynamic fin output) is predicted to vary with motor speed from 50 to 60 percent.

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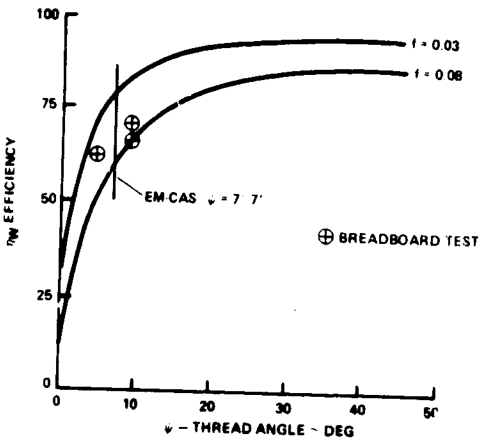

Figure 7. Worm Gear Efficiency

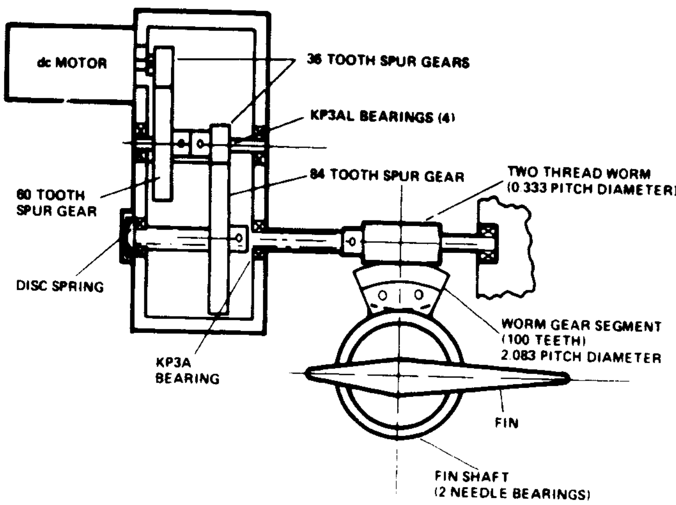

Figure 8. Gear Reducer Schematic

Gear Reducer Efficiency Test Results

Gear reducer efficiency tests were made to determine the effectiveness of the reducer in transmitting power to the aerodynamic fin. Separate tests had to be made on the dc motor, dc motor/gearbox, and EM-CAS to determine reducer losses and separate these losses into gearbox (spur gear) and worm gear components. These tests were made using the dynamometer test setup shown in Figure 9 and the EM-CAS test fixture.

Motor torque-speed performance with and without gearbox are compared in Figure 10. The gearbox efficiency curve, also shown, was derived from this test data and plotted against motor torque. Conclusions derived from this data indicates that at high torque (low motor speed), gearbox efficiency (N_{GB}) approaches 92 percent, as predicted, and reduces as torque decreases.

To estimate worm gear efficiency, it was necessary to operate the complete CAS under different load conditions and subtract the fin shaft bearings and gearbox losses. Results of these tests are plotted in Figure 11 showing that overall reducer efficiency is highly dependent on worm gear efficiency and that worm gear efficiency is 34 percent lower than predicted from analysis and breadboard test data. Detailed inspection of the worm and worm gear hardware revealed worm surface finish rougher than 20 microinch (RMS) and worm gear tooth surface finish rougher than 65 microinch

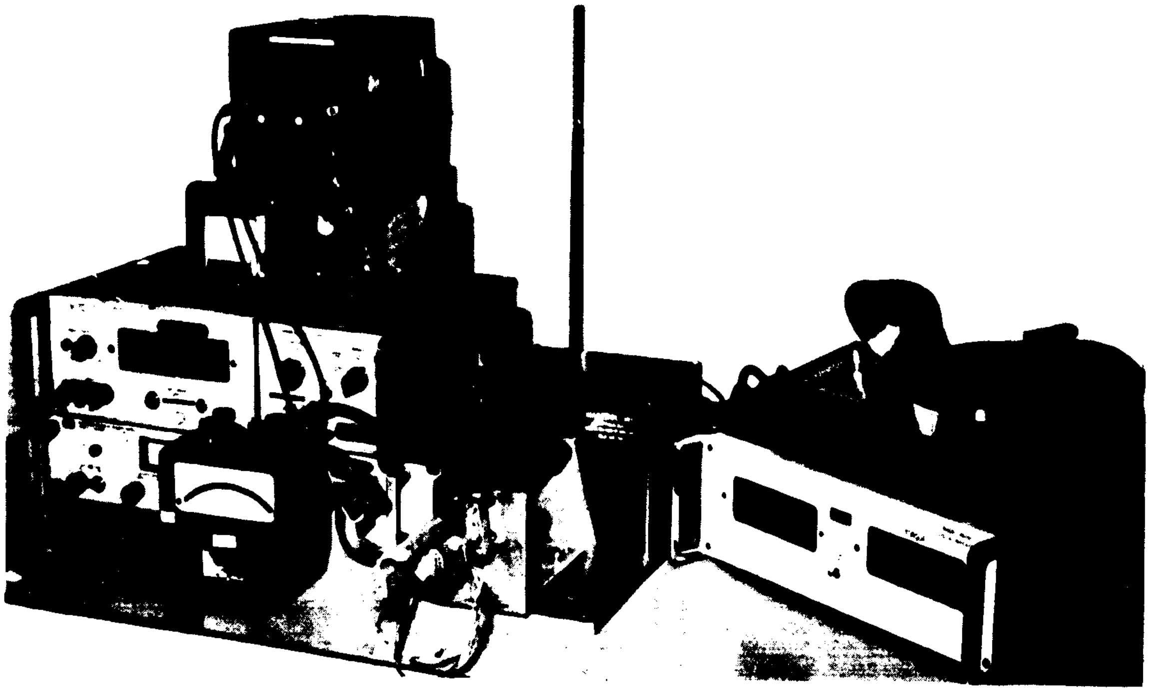

Figure 9. Dynamometer Test Set Up

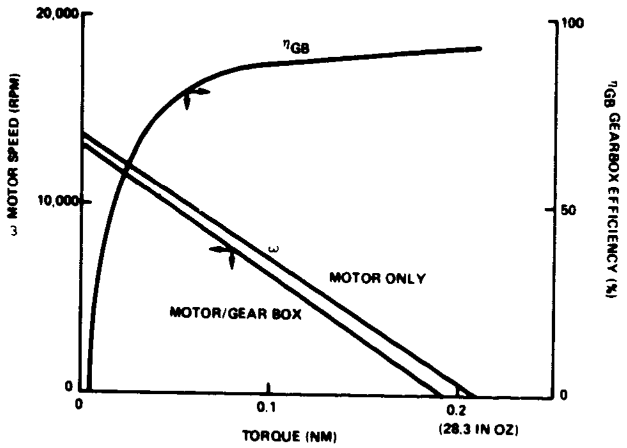

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Figure 10. Gearbox Efficiency

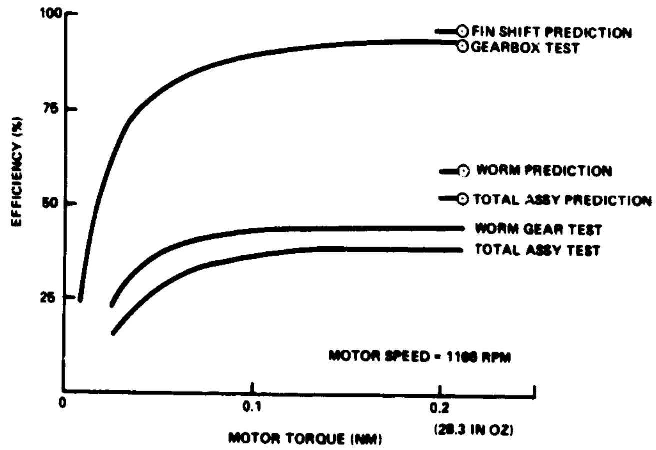

Figure 11. Component Efficiency

(RMS). This compares with the breadboard worm and worm gear finishes which are smoother than 20 microinch (RMS). Test data found in literature indicates that the friction coefficient can increase from 80 percent to 300 percent as surface finish changes from 2 to 65 microinch (RMS). Therefore, rough surface finish is the most likely reason for the observed low efficiency of the worm gear.

DC MOTOR AND BREAK ASSEMBLY

The dc motor and brake assembly was designed, fabricated, tested and supplied by Inland Motors, Division of Kollmorgan Corp. A photograph of the disassembled unit is shown in Figure 12. The complete unit consists of a dc brush motor assembly and a brake assembly. The dc motor assembly consists of three main subassemblies: armature, field, and brush ring.

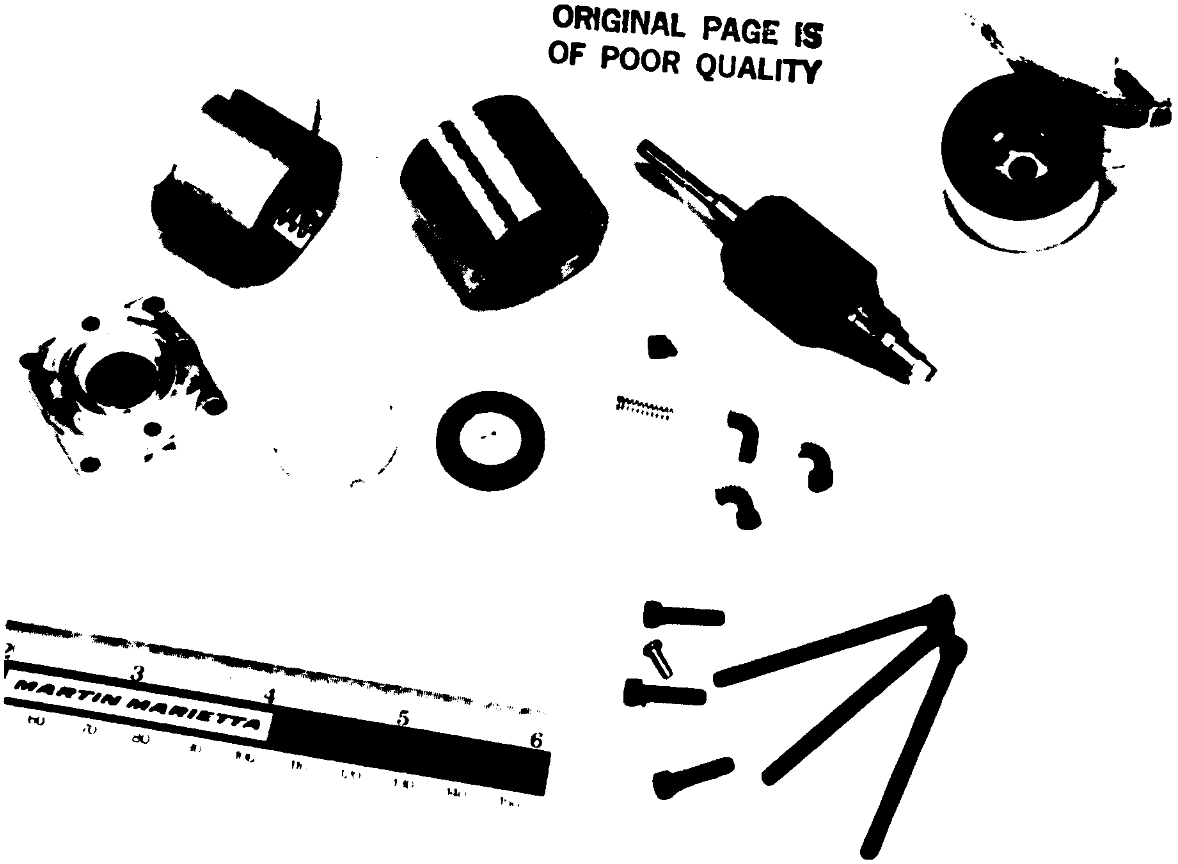

Figure 12. Disassembled Motor/Brake Assembly

The armature (rotor) assembly is made of thin laminations to reduce eddy current effects at high speeds and are bonded together to form the armature core which is insulated and wound with heavy insulation magnet wire.

The field assembly (stator) is the stationary outside case of the motor. Four radially oriented samarium cobalt magnets are bonded to a cold rolled steel yoke section and held physically in place on all four sides by shoulders.

A very small commutator diameter was chosen for this application to provide for better commutation at the higher operating speeds. In addition, four cartridge brushes are used to provide better high speed commutation. The four brush housings, in addition to the two EMI capacitors, and wiring and shielding connections, are molded into a plastic housing which is sandwiched between the stator housing and spacer (aft bearing support). The rotor is mounted on two special load transfer bearings which limit ball load due to "set back" and "set forward" launch accelerations.

The brake consists of an armature, clapper, spring and housing assembly. The brake is failsafe which engages when power is off. The brake disengages the clapper from the armature when greater than 20 volts is applied to the brake coil. The brake will engage when this voltage reduces to approximately 5 volts.

The motor is designed to be pilot centered and flange mounted with four screws. Actual torque/speed test data on Inland Motor S/N 15 is plotted in Figure 13. For this motor, torque constant (K_T) measures 0.01674 Nm/amp (2.37 in oz/amp).

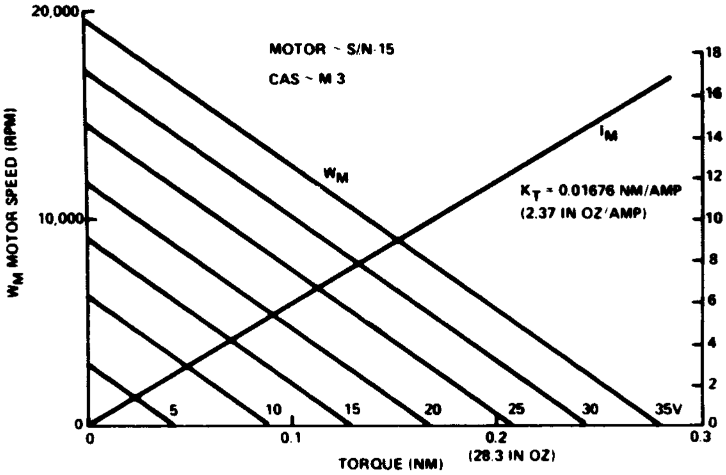

Figure 13. Motor Torque/Speed Characteristics

ELECTRONICS

The electronics package was designed, fabricated, assembled, and tested by Martin Marietta and consists of four shaped printed wiring fiberglass boards, an aluminum heat sink, and the required electronic components.

The first printed wiring board (PWB) is a two-layer board and carries the pitch, yaw plus roll (Y&R) and yaw minus roll (Y-R) command and feedback amplifiers. Error voltage is produced by summing the guidance commands with feedback position voltage.

The second PWB is a two-layer board and carries the dynamic lead/lag compensation amplifiers and the voltage limiting amplifiers.

PWB number 3 is a two-layer board and is attached to the aluminum heat sink which provides mounting for the four sets of complimentary pair of PNP and NPN Darlington power transistors. The board also provides capacitors to minimize crossover distortion and provides the circuitry to the power transistors.

The fourth PWB is a six-layer board and provides the amplifier for the summing junction currents proportional to commands and motor feedback currents proportional to motor speed. Phase lead compensation is provided for the phase lag due to the motor armature inductive lag.

Performance Tests

Performance tests were conducted to demonstrate compliance with the requirements. Low efficiency of the worm gear impacted the pitch axis in stall torque performance and the yaw/roll axes in frequency response performance. Both problems were solved by increasing electrical current limits in all three axes. A performance summary of the pitch and yaw/roll axis is listed in Table IV.

Table IV. Performance Summary

ITEM	REQUIREMENT	PITCH	EM-CAS AXIS YAW+ROLL	EM-CAS AXIS YAW-ROLL	COMMENTS
STALL TORQUE (NM)	12.44	13.68			Pitch
(NM)	6.22		9.89	9.89	Yaw/Roll
LOADED RATE (DEG/SEC)	120	264	357	385	
FREQ. RESPONSE (HZ)					
AR = -3 dB	12.1 Hz	18 Hz	17.5 Hz	17.5 Hz	
ØLag = -75°	14.32Hz		18.0 Hz	18.0 Hz	
-80°	12.1 Hz	17.5Hz			

The required stall torque is 6.22 Nm/fin (55 in lb per fin) or 12.4 Nm pitch (110 in lb pitch). To eliminate the effects of inertia, stall torque was measured by imposing a slow triangle wave command into the EM-CAS electronics. For a pitch current limit increase of 50 percent, pitch axis stall torque increased 50 percent from 9.125 Nm (80.7 in lb) to 13.68 Nm (121 in lb).

Minimum loaded vane rate performance requirements are 120 deg/sec at 5.088 Nm/fin (45 in lb per fin)(10.18 Nm pitch/fin). The EM-CAS demonstrated 264 deg/sec in pitch and 357 deg/sec in the yaw/roll axes.

The ability of each actuator to follow triangle, square, and sine waves is evident in Figure 14. Very little backlash is displayed with the triangle and sine waves as the actuator crosses the zero axis. Note also that closed loop position error is very small. The zero overshoot shown in the square wave indicates damping close to critical which is also indicated in frequency response data.

Higher performance from the EM-CAS is possible by increasing current limits and gains; however, this increased performance requires additional battery power which reduces flight duration.

EM-CAS Battery Test

A series of tests on the EM-CAS were performed using energy supplied by a standard 1 7/8-inch diameter by 5 3/4-inch long thermal battery. Twelve tests were run: three ambient, five cold (-25°F), and four hot (+145°F). The purpose of these tests was to verify that the existing standard thermal battery could provide sufficient energy to operate the EM-CAS with simulated worst case flight duty cycles for a minimum of 70 seconds.

Other required loads were simulated with a 20 ohm resistor, and the three motor brakes released and held during tests using the B+ cell.

Eleven of these tests were performed with the initial electronic package which had the original current limits. One final battery test was conducted with the increased current limits required to meet stall torque and frequency response requirements.

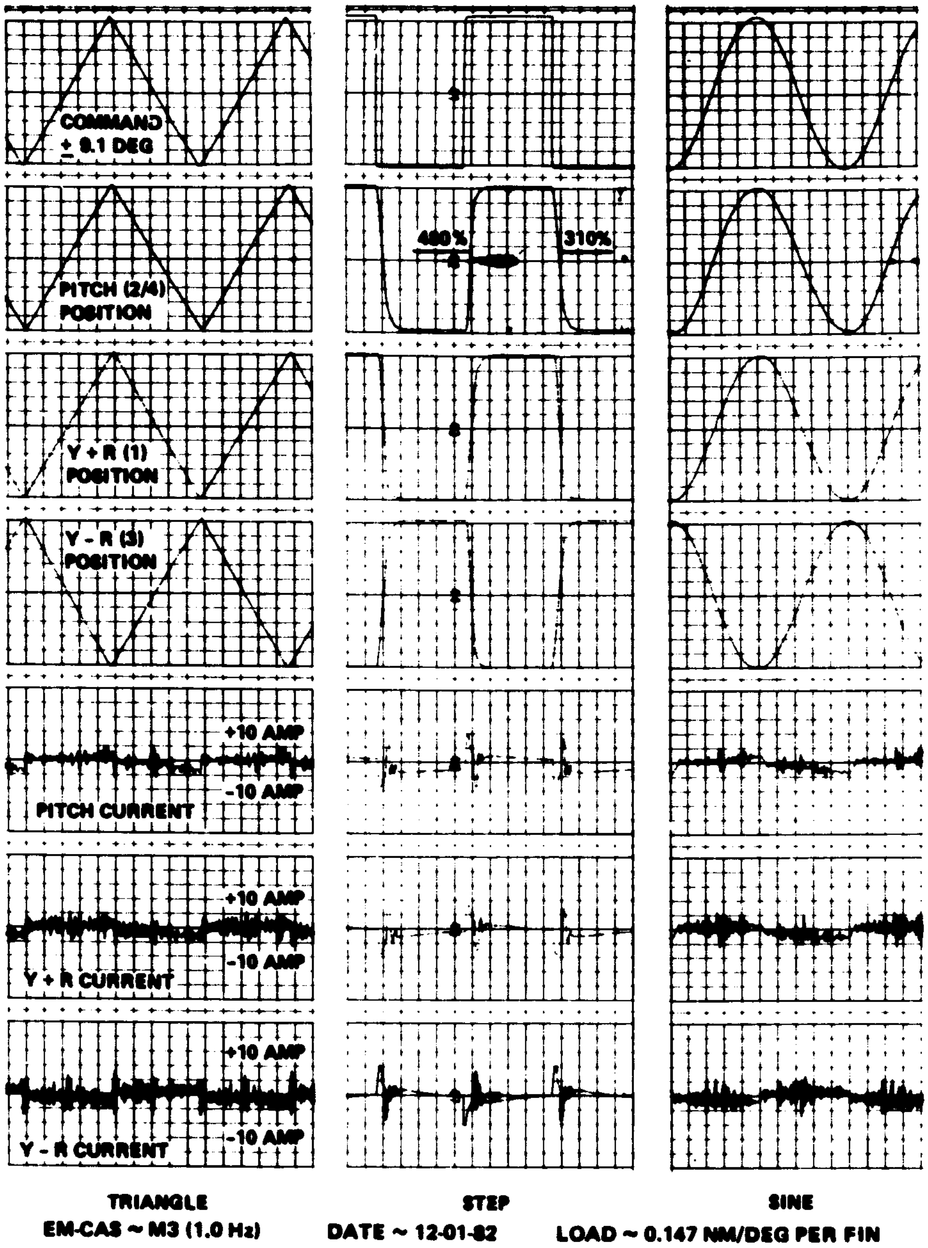

Figure 14. Response after Electronic Mod.

A comparison of average battery terminal voltage minimums for ambient, cold and hot soak conditions is shown in Table V. Battery terminal voltage minimums for the final battery test (high current limits) are shown in the last column of Table V.

Table V. Average Minimum Voltage/Time Conditions

TIME (SEC)	AMBIENT		COLD		HOT		FINAL TEST	
	+B	-B	+B	-B	+B	-B	+B	-B
3	34	32.5	33.5	33.2	33.5	31.5	36	35
50	32	29.0	31.2	32.7	32.2	30.2	35	32
70	29.5	28.5	26.2	30.4	29.7	27.7	34	30
100	29.5	21.0	21.7	25.2	29.2	23.8	34	26

*70 to 100 sec maximum current is supplied by the negative (-B) cell.

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It is concluded from these series of tests that the existing thermal battery has sufficient capacity, even at high current limits, to operate the EM-CAS for the required flight duration.

MISCELLANEOUS TESTS

Miscellaneous tests during development to minimize failure risk included bearing thrust and "slop" investigations, and a hydrogen embrittlement study.

A thrust load test on the KP3A bearing selected to support the worm shaft was conducted to show bearing operability after a 50 percent thrust overload was imposed. As installed in the EM-CAS gearbox, the bearing showed acceptable operation after 1000 percent of recommended thrust load had been imposed, and the unsupported bearing withstood up to 400 percent of recommended thrust load before failure.

Hence, the selected KP3A bearing will safely carry the 1020 pound thrust load during launch without damage and will provide effective CAS operation afterwards.

Measurements were made on eight KP3A bearings to establish relative axial displacement between inner and outer bearing races for estimating bearing effects on backlash. Axial displacement on the free bearings varied from 0.051 to 0.102 mm (0.002 to 0.004 in). The bearings were then pressed into an aluminum housing and axial displacement was again measured. Results show that bearing axial displacement after pressing was less than 0.0127 mm (0.0005 in) in all measurements. These results showed that all bearings supporting the worm shaft must be pressed into their respective housings to minimize backlash.

Standard hydrogen embrittlement tests were performed on M4 and M6 metric screws which had been heat treated after plating.

CANNON LAUNCH TESTS

A total of five EM-CAS units were canistered and exposed to the gun launch environments as listed in Table VI. Three units (M1, M2 and M3) were assembled by Martin Marietta. Two assemblies (D1 and D2) were produced by Diehl GmbH & Co., West Germany. The pitch axis gearbox with a Lucas motor, assembled by Diehl, was present in the M1 unit. The Y+R and Y-R axes had Martin Marietta gearboxes and Inland motors.

Table VI. Canister Launch Test Summary

Unit	Date Launched	Unit Temp. °F	Setback Acceleration gs	Post Launch Results
M1	7-16-82	Ambient	9,356	Circuit Card Spacers Failed
D1	8-18-82	-45°F	10,058	No Structural Anomalies
M2	8-18-82	-45°F	10,084	Circuit Card Screws Failed
D2	9-09-82	+145°F	9,960	No Structural Anomalies
M3	9-09-82	+145°F	9,900	No Structural Anomalies

All five 83.5 Kg (184 lb) projectiles were launched and parachutes recovered at the Redstone Arsenal in Huntsville, Alabama, using the 203 mm (8 in) cannon.

Hardware failures in the first and third tests were attributed to perimeter screws on the heat sink which were too long. The package was loose and free to move on the four perimeter screws during setback acceleration. The phenolic spacers in the first launch absorbed the setback shock and failed. The substitution of aluminum spacers produced a more efficient joint which was capable of transmitting the circuit card inertia to the two M5 screwhead shoulders with subsequent tensile fracture. With the shorter length screws installed in the last tests, no further failure occurred.

PROBLEMS ENCOUNTERED AND SOLVED

Mechanical and electronic problems encountered and solved during development are summarized in Table VII. Excessive shaft axial clearance coupled with bearing clearances made the actuator pitch axis sensitive to limit cycle. This appeared as backlash to the actuator and when the shaft was properly shimmed, the problem disappeared.

Table VII. Problems Encountered and Solved

Problem	Solution
Excessive Shaft Axial Clearance	Redesigned Shaft Retainer to Accept Shims
Collar Flange Interference with Housing	Chamfered Housing
Rubbing Segment Gear Rollpin	Installed Shorter Pin Flush with Bottom
Motor Wiring/Cover Interference	Special Cutouts in Gearbox Covers
Phenolic Bushing Failure	Changed Material to Aluminum
Electronic Package Screw Failure	Reduced Perimeter Screw Lengths to Prevent Blind Hole Bottoming
Low Gear Reducer Efficiency	Increased Motor Current Limit
PWB Numbers 1 and 2 Cross Talk	Added Grounded Copper Shield to M1, Circuit Changes on M2 and up Eliminated Requirement
PWB Numbers 2, 3 and 4 Conductor Errors	Hard Wired Correctly and Epoxied in Place
Heat Sink Grounding	Added Grounding Screw
Darlington Transistor Noise Sensitivity	Added Filter Capacitor and Epoxied to PWB Number 3
Heat Sink Screw Head Position Inspection	Added Cutouts in Heat Sink
Low Bandwidth in Current Amplifier	Added R-C Intergrator in Feedback Circuit (Epoxied in Place)
Spacer Frame Component Interference	Spacer Frame Modification

The phenolic bushing failure on the first cannon launch was solved by using an aluminum material, and the screw failure on the third launch was solved by using shorter perimeter screws and confirming proper seating by visual inspection.

Cross talk between boards, conductor errors and grounding problems were finally solved by circuit changes and hardwiring. Conductor and component modifications were epoxied to the printed wiring board after checkout, and survived the cannon accelerations very well. Circuit changes had to be made to balance outputs and increase current limits for stall torque and bandwidth requirements.

CONCLUSIONS

Specific 1982 objectives were achieved as follows:

1. The assembly configuration and components of a prototye EM-CAS were established and optimized.
2. The performance capability of the prototype EM-CAS was demonstrated in laboratory tests.

3. The prototype EM-CAS for small missile high-g applications was demonstrated by exposure of 5 separate units to 10,000g launch environment using the 203 mm cannon at MICOM in Huntsville, Alabama.

In addition to specific objectives, the following was demonstrated:

1. 12 thermal battery tests, (4 hot, 5 cold and 3 ambient) operating the EM-CAS for greater than 83 seconds under worst-case duty cycle, were performed.
2. The EM-CAS is 14 percent lighter and 20 percent smaller than the present pneumatic CAS.
3. Cost studies show that the EM-CAS will be cost competitive with the pneumatic CAS.