UNFOLDING THE AIR VANES ON A SUPERSONIC AIR-LAUNCHED MISSILE

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ABSTRACT

ASALM (Advanced Strategic Air-Launched Missile) is a supersonic air vehicle designed to be air-launched from a carrier aircraft. The four aft aerodynamic control vanes are folded to maximize the number of missiles carried on-board. The unfolding (erection) system must be small, energetic, fast, and strong. This paper describes the materials selected and problems that arose during development of the unfolding system. An artist's concept of the ASALM in flight is shown in Figure 1.

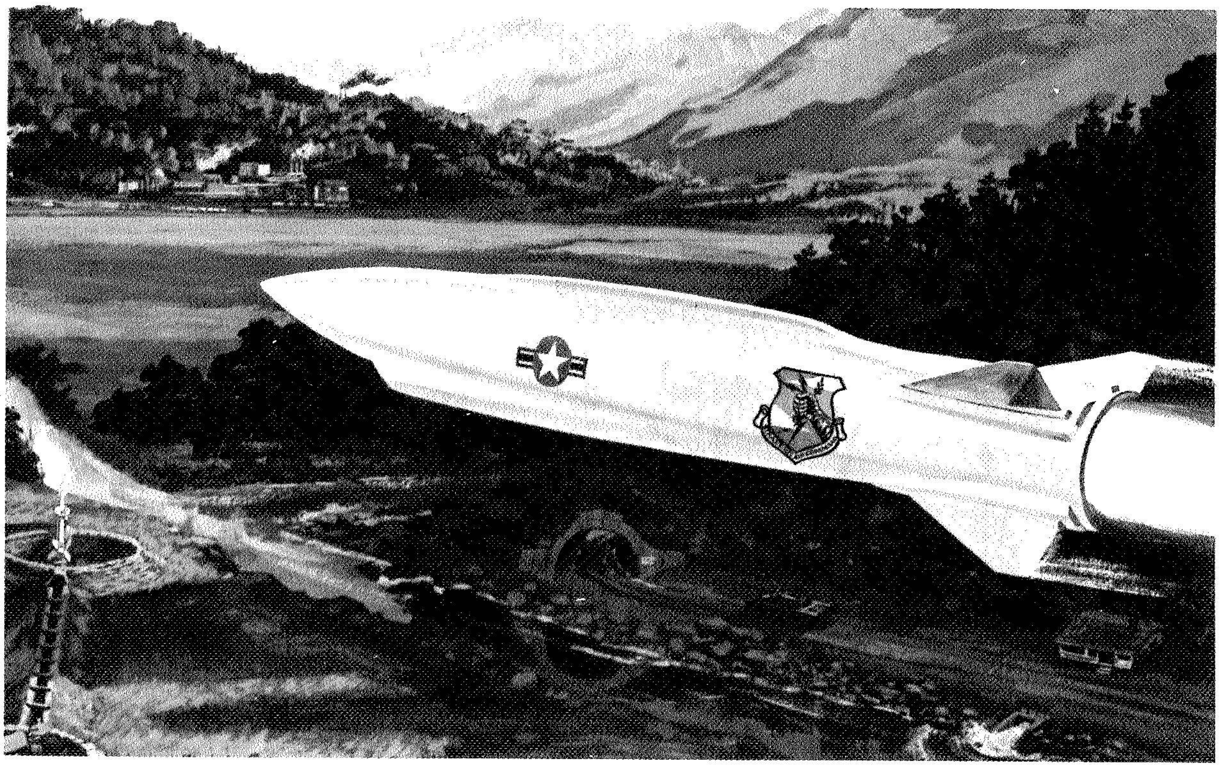

Figure 1. The Advanced Strategic Air-Launched Missile (ASALM)

INTRODUCTION

To maximize internal carry packaging efficiency on one of several carrier aircraft, the four aft aerodynamic control vanes on the ASALM must be folded to fit within a pie-shaped sector around the launcher spindle. Analysis of the launch sequence has shown that 350 ms can be allocated for vane

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deployment. However, 250 ms of this allocation are required for the missile to separate from the launcher and clear the weapons bay door, leaving 100 ms for vane opening. Figure 2 shows the ASALM/weapons bay interface.

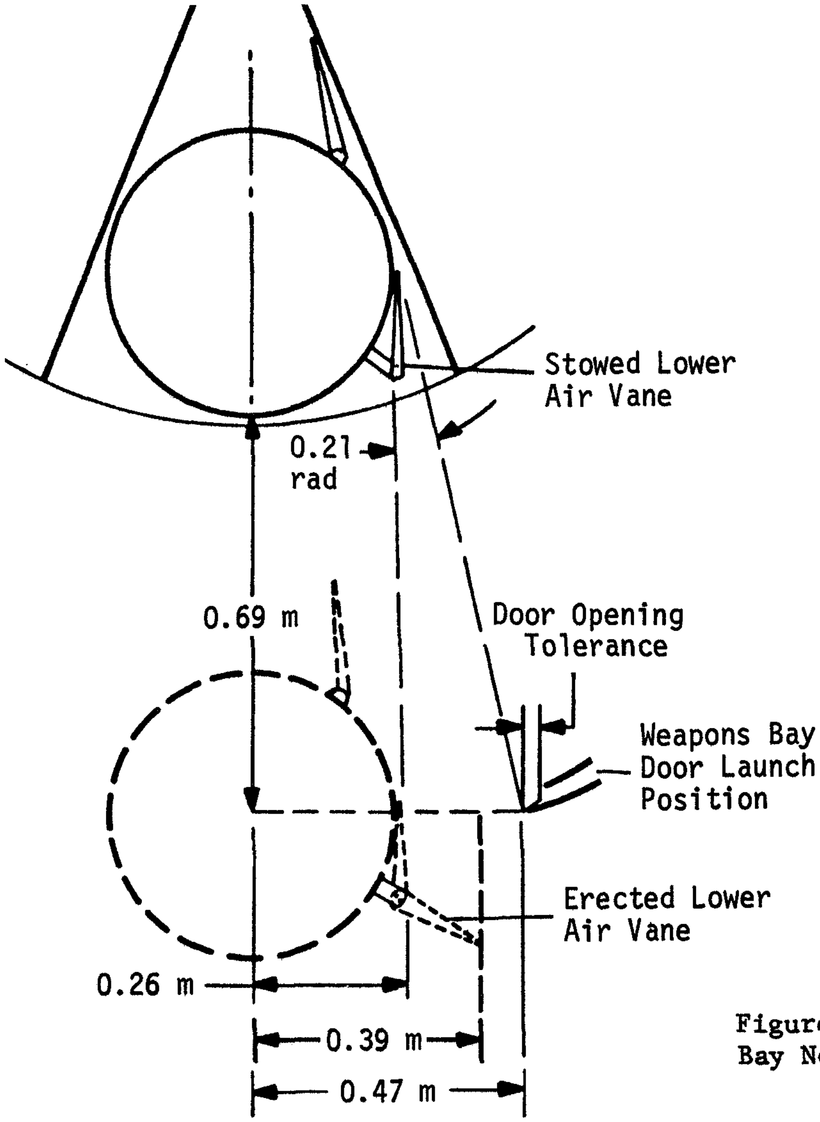

Figure 2. ASALM/Weapons Bay Nominal Clearances

Loads analysis which considered separation velocity, aircraft flow field, missile attitude, and various launch aircraft cruise conditions resulted in two cases of predicted load. The first case, a supersonic launch, shows that the unfolding air vane is opposed by an increasing moment which reaches a maximum of 339 N·m (3000 in-lb) as the air vane approaches its fully erect position. The second case is an aiding air load which peaks at 243 N·m (2150 in-lb) at approximately one-half the fold angle. These requirements necessitate an extremely compact, high-energy air vane erection system (AVES) which must act within the allocated 100 ms.

DESIGN CONCEPT

The concept selected for this mechanism (a type of "yankee" screwdriver) uses a small gas generator (thruster) to pressurize a piston in the fixed root of the air vane. Two helical tangs located diametrically opposite on the piston ride in helical grooves machined in the piston bore. These grooves impart rotary motion to the piston. The forward end of the piston rod is splined and engages female splines in a sleeve which is rigidly connected to the folding part of the air vane, thus erecting the vane as the piston is forced forward. The end of the piston/splined rod opposite of the gas pressurized end forces hydraulic fluid through a sharp-edged orifice, producing damping forces proportional to the square of the piston translational velocity. Damping is essential to reduce the variation in air vane angular velocity between the two aerodynamic load cases. Both up- and downlock provisions have been incorporated, along with manual erection and folding capability. Air vane dimensions are approximately 63.5 cm (25 in) root chord, 22.9 cm (9 in) span, and 3.8 cm (1.5 in) maximum thickness. Figure 3 presents an exploded view of the AVES.

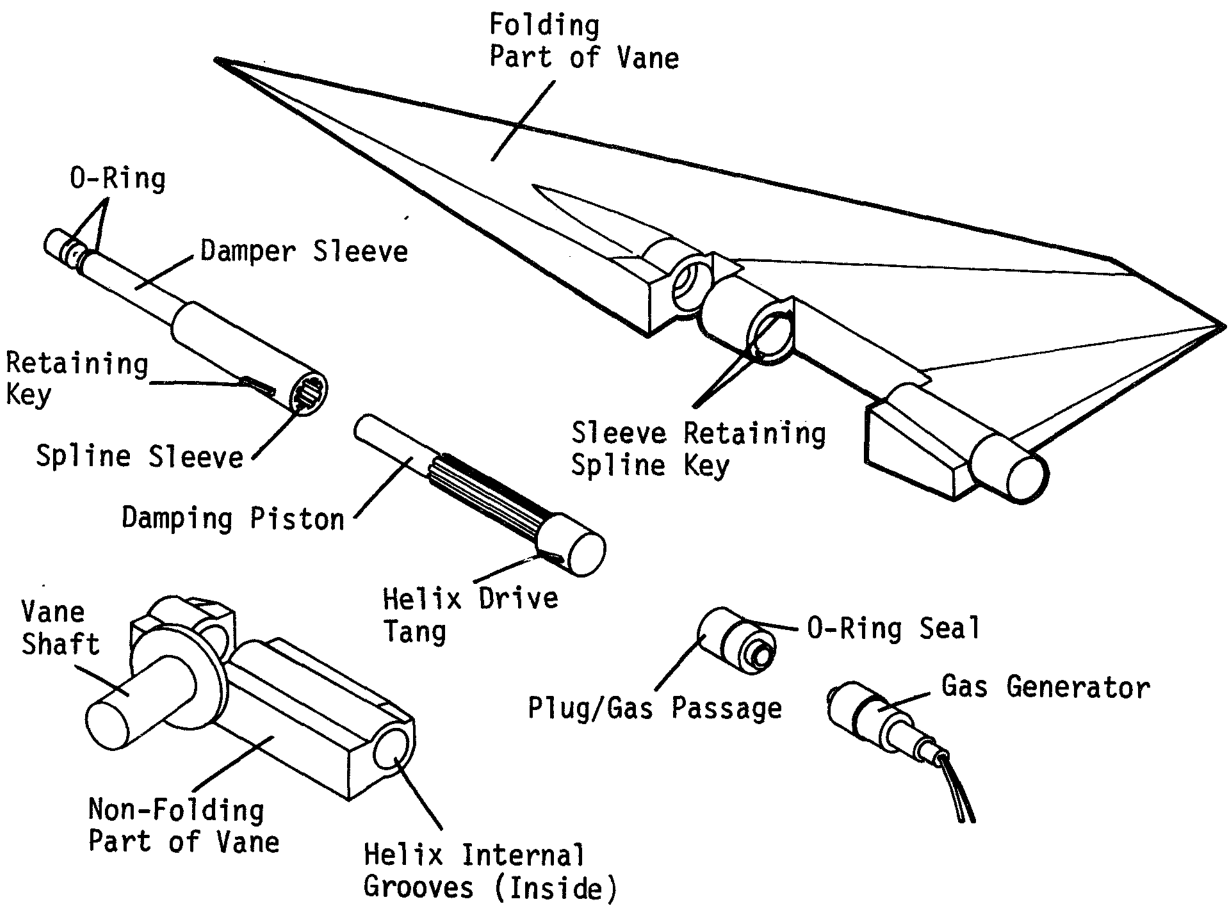

Figure 3. Piston-Helix AVES Concept

MATERIAL SELECTION

Due to high air loads on the vane and the small size requirements for the AVES, stress analysis dictated the use of high strength materials. Candidate steels which could meet the high strength requirements include AISI 4340, 250 and 300 grade maraging, Custom 455, 13-8-PH, 17-4-PH, and 440C CRES. AISI 4340 and 250 grade maraging steels were selected because of their ready availability. The folding component was fabricated from titanium 6-2-4-2.

To alleviate or minimize galling effects, the piston/spline component was made of 250 grade maraging steel. The helix groove component and female spline sleeve were constructed from 4340 steel heat-treated to a strength of 1790 MPa (260,000 psi). Some concern was initially expressed in the use of 4340 at such a high strength level. Opinion was that ductility was very poor and that problems could be expected during the application of dynamic loads during air vane unfolding. As discussed later, however, no problems with the 4340 material were encountered throughout the entire test program. Damping fluid seals were made of ethylene propylene, which is compatible with the Dow Corning 510 silicone oil damping fluid. These seals were effective against high pressure warm gases.

GALLING FRICTION TEST

A test was devised to determine the galling/friction characteristics of maraging steel sliding against 4340 steel. Torque loads, up to and including ultimate, were applied to a drive piston/spline while a hydraulic ram translated the drive piston/spline in mating parts. The test pieces were geometrically similar to AVES parts, however, all motion was linear. Both male and female parts were coated with Everlube 811 dry film molybdenum disulfide lubricant. The test set-up is shown in Figure 4, and Table I presents the measured results. Inspection of the parts revealed no galling or other detrimental wear, thus indicating satisfactory choice of materials and lubricant.

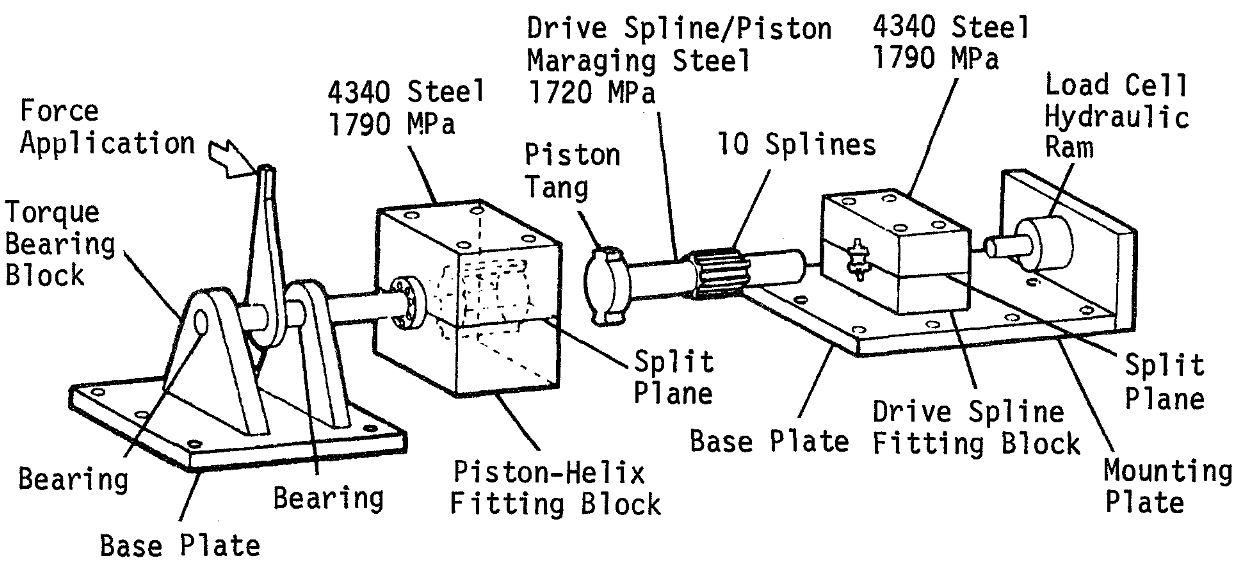

Figure 4. AVES Galling/Friction Test Set-up

TABLE I. FRICTION TEST SUMMARY

Run Number	Applied Torque (N•m)	Axial Load (N)			Friction Coefficient			Velocity (m/s)
		Breakaway	Min	Max	Breakaway	Min	Max	
1	51	~3,114	1,010	-	-	0.23	-	0.831
2	111	3,220	2,440	2,990	0.19	0.14	0.17	0.820
3	226	3,810	3,420	4,480	0.11	0.10	0.13	0.679
4	330*	5,600	5,340	6,940	0.11	0.10	0.14	0.396
5	336*	6,800	5,200	6,610	0.13	0.10	0.13	0.287
6	503**	8,900	7,540	9,270	0.11	0.10	0.12	0.267
7	620***	10,140	9,210	10,880	0.11	0.10	0.10	0.231

* Design limit load (repeat runs)
** Design ultimate load
*** Proof load (2X limit load)

ACTUATOR SIMULATOR

The gas generator which provides the energy for erection of the AVES was being developed concurrently with AVES fabrication. Since this hardware was not available for gas generator development and verification testing, an actuator simulator was designed and fabricated (Figure 5). This device simulated the reaction forces of the damping fluid, mechanical friction, etc., and duplicated the time rate of change of the gas chamber volume. Its construction provided for gas and fluid pressure instrumentation as well as piston position versus time. Prior to using the simulator for gas generator testing, trial runs were made using 34.5 MPa (5000 psi) of gaseous nitrogen. These experiments demonstrated reliable operation of the simulator and provided a check on damper orifice sizing.

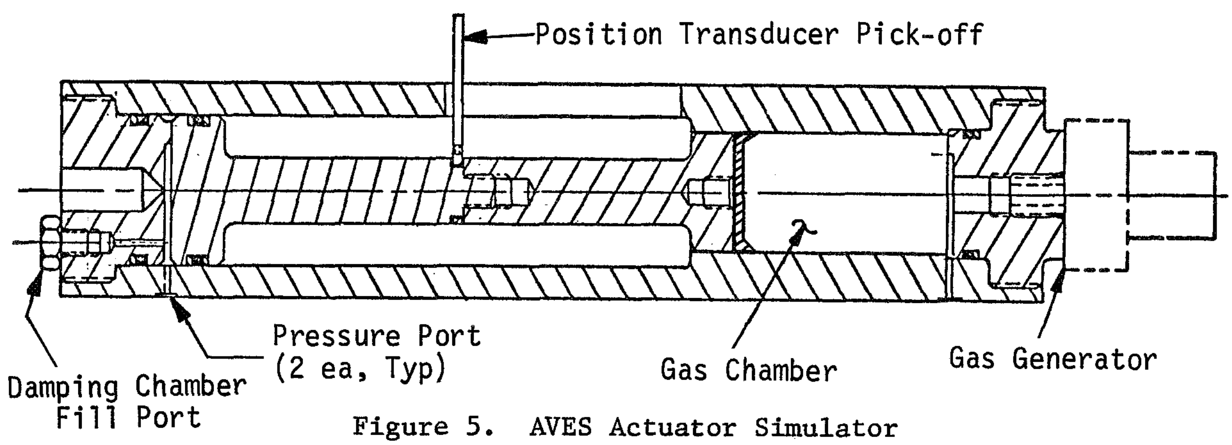

Figure 5. AVES Actuator Simulator

In October 1977, all materials were available to begin testing. The AVES was required to operate against aiding and opposing loads at the minimum and maximum expected temperatures (Table II).

TABLE II. AVES TEST SEQUENCE

Test No.	Gas Source	Pressure (MPa)	Load Condition	Temperature
CG1	Manual Checkout	-	-	Ambient
CG2	Bottled GN_2	3.45	None	Ambient
CG3	Bottled GN_2	13.8	None	Ambient
CG4	Bottled GN_2	34.5	None	Ambient
CG5	Bottled GN_2	34.5	None	Ambient
WG1	Warm Gas	-	None	Ambient
CG6	Bottled GN_2	34.5	Opposing	Ambient
CG7	Bottled GN_2	34.5	Opposing	Ambient
WG2	Warm Gas	-	Opposing	Ambient
CG8	Bottled GN_2	34.5	Aiding	Ambient
WG3	Warm Gas	-	Aiding	344°K
WG4	Warm Gas	-	Opposing	219°K
WG5	Warm Gas	-	Opposing	344°K

All of the cold gas (CG) runs were completed without incident; however, warm gas test (WG1) failed due to gas leakage and erosion of the TFE (Teflon) seal in the "square" corner.

A seal material and configuration test plan was developed in an attempt to curtail gas leakage. The candidate seal materials were required to be effective at 219°K (-65°F) and 344°K (+160°F), and were to provide initial edge preload when installed. They were also required to be resilient so that the gas chamber pressure tending to compress the seal would also create additional edge loads.

Five seal candidates were selected: lead seals 0.32 mm (0.125 in) thick and 0.64 mm (0.25 in) thick; RTV rubber seals 0.32 mm (0.125 in) thick; and copper shim seals composed of stacks of shims 0.008 mm (0.003 in) thick in one concept and 0.013 mm (0.005 in) thick in another. Seventeen layers of shim stock went into the making of one copper seal. Each of the individual shim stocks were made slightly oversize such that the package of shims had to be forced into the test device. With this array of seals, testing began using a modified actuator simulator. Parallel grooves simulating the helical grooves in the AVES were electric-discharge machined in the gas side of the cylinder. The gas piston was modified to include two diametrically opposite tangs. In this manner, the square corner sealing problem would be duplicated. Warm gas was supplied by a handloaded cartridge which used a $BKNO_3$ boosted initiator and IMR 4198 smokeless rifle powder (shotgun shell). Each seal candidate was tested at room temperature; those which did not leak were then tested at 219°K (-65°F) soak temperature. As shown in the summary (Table III), the RTV rubber seal did not leak at either temperature.

The 0.32 mm (0.125 in) RTV seal was then molded to fit the helical grooves in the AVES. A warm gas test was conducted using the AVES hardware and a shotgun shell. Immediate leakage was detected by both thermocouple and pressure instrumentation. The physical characteristics of the actuator simulator and the AVES were carefully checked and compared to determine what effects (other than the helical twist of the AVES grooves) might have contributed to the seal failure. The dimensional check revealed that the helical groove in the AVES was rougher than that in the simulator (4.75 μm [190 μin] versus 3.25 μm [130 μin]), and that the clearance between the tang and the groove was much greater (0.36 mm [0.014 in] versus 0.05 mm [0.002 in]) on a side. Based on this information, the helical tangs on the AVES gas piston were built-up with flame-spray techniques and remachined to reduce the clearance. The AVES was then assembled with the newly machined piston and RTV seal installed. A warm gas test using the same propellant charge as in the simulator tests was run. Again the seal failed. At this point, it was believed that sealing square corners would involve a high-risk development program. Therefore, an alternate concept was configured wherein the tanged piston was sandwiched between two cylindrical pistons, both of which could readily be sealed against high pressure fluids and gases with the use of O-rings. To verify this concept, the AVES test hardware was modified to accept a cylindrical piston extension and O-rings.

TABLE III. SEAL TEST SUMMARY

Seal Configuration	Ambient Temperature	219°K Temperature
0.32 mm RTV	No leak	No leak
0.32 mm lead	Leak	N/A
0.64 mm lead	No leak	Leak
0.008 mm copper shim	Leak	N/A
0.013 mm copper shim	Leak	N/A

Figure 6 illustrates the redesigned configuration. Minimal changes were required. The new part (A) is an extension of the original piston (part C). The piston head of part A is circular and allows the installation of a standard O-ring seal. Part B, the cylinder containing piston A, replaced the titanium extension in the original configuration. Fastening bolt diameters were the same but the lengths were shortened. Both parts A and B were made of AISI 4340 steel heat-treated to a hardness of 1790 MPa (260,000 psi). Original parts C and D required some machining.

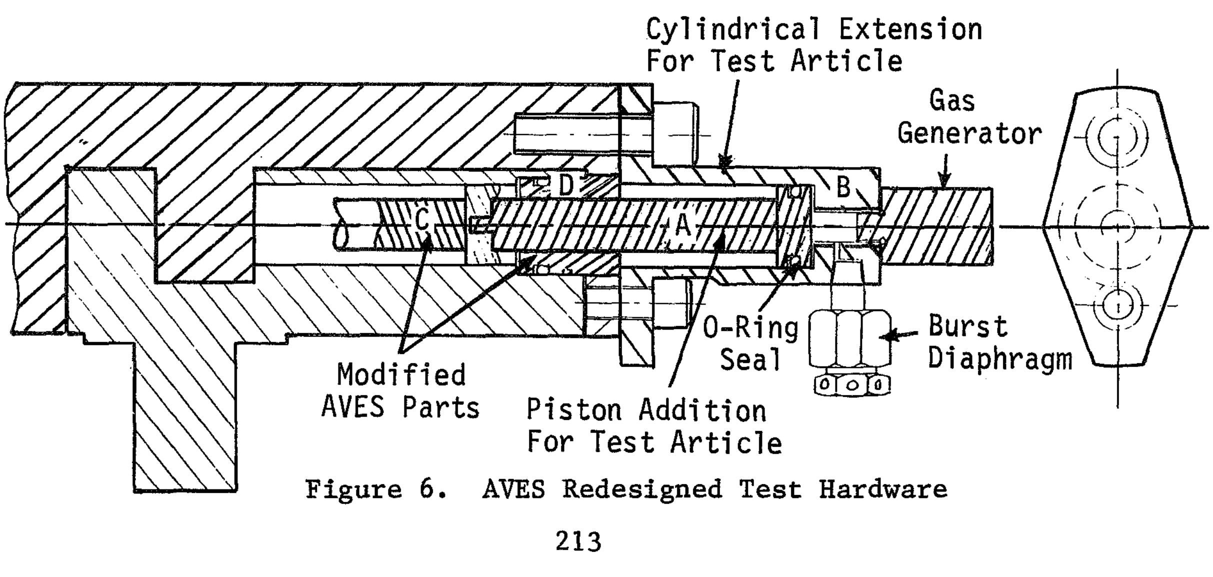

Figure 6. AVES Redesigned Test Hardware

Prior to checking the performance of the redesigned AVES with warm gas generators, two cold gas and three shotgun shell erection tests were conducted. The air vane erected in all cases except the first shotgun shell test. In that test, the burst diaphram ruptured in the first several milliseconds. Reduction of the shotgun shell propellant charge by one-half allowed successful erection of the air vane in the subsequent tests discussed below:

- Test WG1 was conducted at ambient temperatures with no external loads applied to the AVES. The air vane opening time was 78 ms. Maximum warm gas and fluid damping pressures were 40 MPa (5800 psi) and 103 MPa (15,000 psi), respectively. Post-test examination of the AVES revealed no mechanical damage - all seals worked effectively and the locking pin fully engaged.

- Test WG2 was conducted at ambient temperatures with an opposing simulated external aerodynamic torque of 339 N·m (3000 in-lb) applied to the AVES. The air vane erected to within 0.05 rad (3 degrees) of full opening in 96 ms. Partial engagement of the locking pin verified the vane erection angle. Examination of the driving piston following AVES post-test disassembly and inspection revealed galling of the piston surface. This galling action was the result of a skewed alignment between the axes of the cylinder and the driving piston (localized titanium yielding was involved). Failure of the vane to fully erect was believed due to the restraint forces introduced by galling. This effect was eliminated from succeeding tests by machining the drive piston to an approximate spherical shape.

- Test WG3 was conducted at 344°K (160°F) with an aiding simulated aerodynamic torque of 243 N·m (2150 in-lb) applied to the AVES. The vane erected with full locking pin engagement within 66 ms. The angular velocity at full erection was 26.2 rad/s. Peak warm gas and damping pressures of 54.5 MPa (7900 psi) and 142 MPa (20,600 psi), respectively, were recorded. Post-test examination revealed no mechanical damage, other than partial extrusion of the hydraulic damping fluid Teflon sealing backup rings.

- Test WG4 was conducted at 219°K (-65°F) with a simulated opposing load applied to the AVES. The vane erected fully within 114.5 ms and then backed off about 0.05 rad (3 degrees) due to the opposing load. The locking pin did not move - indicating a frozen condition. After warm-up, however, the locking pin engaged. An opening time in excess of 100 ms was partially attributed to opposing torques generated in the test fixture equipment (i.e., bearings). Measurement of this torque ranged from 8.5 to 17 N·m (75 to 150 in-lb). The work involved in erecting the AVES is increased about 6 percent due to this effect.

- Test WG5, conducted at 344°K (160°F) with an opposing load applied to the AVES, was expected to develop the highest pressures of any of the tests. The gas pressure rose to 75.8 MPa (11,000 psi), when the safety burst diaphragm (rated 77.6 to 81.0 MPa [11,250 to 11,750 psi]) ruptured. The erection process terminated in 30 ms, at which point the air vane had erected 1.05 rad (60 degrees). No physical damage was sustained by the AVES other than Teflon backup ring extrusion on the damping fluid seals (pressure $\sim$169.6 MPa [24,600 psi]).

For subsequent testing, the damping orifice area was increased by a factor of 2.0 to reduce the erection time for the opposing load, 219°K (-65°F) condition. This selected value was based on results of dynamic analyses which showed that air vane opening time was reduced 9 ms by increasing the damping orifice area to 1.5 times the nominal value. An additional 3 ms was gained by doubling the area. Only 1 additional ms would be gained if the area were to be increased 2.5 times.

FINAL AVES TESTING

In April 1978, additional gas generators were purchased to repeat AVES tests WG2, WG4, and WG5. A brief description of each of these tests follows:

- Test WG2R was conducted at ambient temperatures with an opposing simulated external aerodynamic torque of 339 N·m (3000 in-lb) applied to the AVES. The air vane erected with full locking pin engagement in 99 ms. Expected opening time was approximately 75 ms. Prior to the test, the wiring on one of the two initiators was accidentally broken. Each of the initiators contained 115 mg of propellant. With only one-half the propellant available, the initial chamber pressure was considerably reduced for this test. Despite this deficiency, all test requirements were met.

- Test WG5R was conducted at 344°K (160°F) with an opposing load applied to the AVES. The air vane opened with full locking pin engagement (visual observation) in 51 ms.

- Tests WG4R and WG4R2 were conducted at 219°K (-65°F) with simulated opposing loads applied to the AVES. The air vane failed to fully erect in either test due to failure of the O-rings to seal the warm gas pressures; however, full erection was achieved in test WG4, and no gas leakage past the O-ring seal occurred in that test. Post-test examination of the O-ring in both tests indicated erosion of the O-ring over a sector width of about 1.05 rad (60 degrees). Position of this sector was symmetrical about the piston topside. Failure of the O-rings to seal was attributed to excessive yielding of the titanium folding air vane in the vicinity of the gas generator. This yielding caused misalignment between the axes of the gas cylinder and O-ring sealed gas piston. As a result, a greater amount of O-ring on the piston topside was exposed to the hot gases than on the piston sides and bottom. Yielding of the titanium was first noticed on test WG2. Apparently, the amount of yielding was not enough to cause leakage on test WG4; however, continued testing increased the yielding. In an AVES redesigned for tactical use, this type of structural behavior would not occur. Typical dynamic response characteristics of these tests are shown in Figure 7.

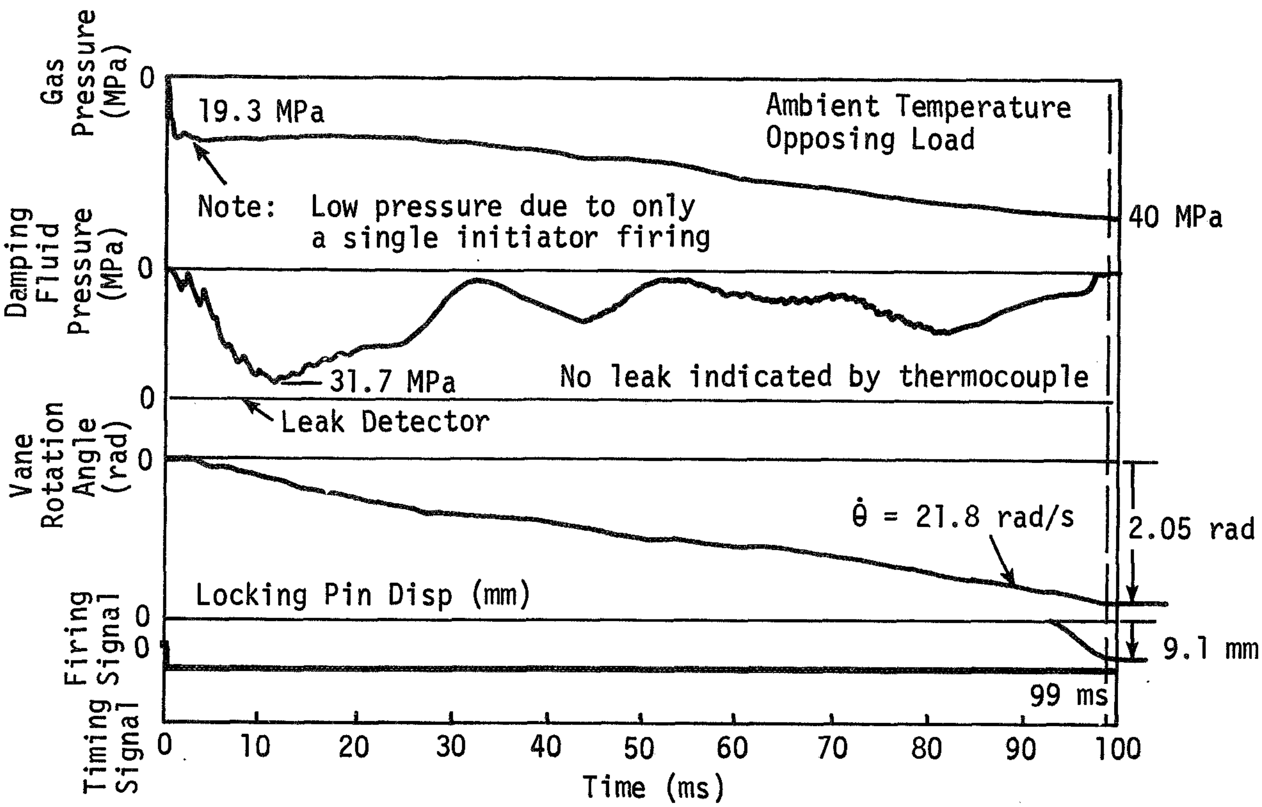

Figure 7. AVES Dynamic Response for Test WG2R (Typical)

Figure 8 summarizes AVES performance with respect to opening time and angular position requirements. For all load and temperature conditions, the AVES erected fully on tests WG1, WG2R, WG3, WG4, and WG5R within 100 ms, and had full engagement of the locking pin except on test WG4. The erection time performance of WG4 (cold temperature opposing load) will meet 100 ms requirement by doubling the damping orifice area. An alternate locking pin design has been implemented which will prevent freeze-up at cold temperatures.

CONCLUSIONS - LESSONS LEARNED

The AVES development program has shown that the piston-helix concept, initially shown feasible through analyses and later demonstrated experimentally, is a sound concept. This design withstood 11 warm gas, 5 shotgun shell gas, and 8 cold gas erection tests with no degradation in performance until the last 2 tests. Future development effort will concentrate on weight reduction, ease of manufacture, good maintainability, and incorporation of lessons learned from the initial development effort (i.e., gas sealing and locking pin mechanism).

Lessons learned from the AVES development test program include:

1 Gaps between the helix drive tang and helix internal grooves could not be effectively sealed against high gas pressures. O-rings proved to be effective seals.

2 The O-ring sealed, spring-loaded locking pin was inoperative at 219°K (-65°F). A concept change to remove O-rings or the use of gas pressure as the locking force is required for a redesigned tactical AVES.

3 Damping bellows should be designed such that repeatable rupture patterns occur during vane erection.

4 A gas generator with a smaller temperature sensitivity coefficient is recommended to reduce variations in AVES performance due to temperature environments.

5 Ethylene propylene O-rings will satisfactorily seal dynamic damping fluid pressures as high as 165 MPa (24,000 psi) and gas pressures as high as 75.8 MPa (11,000 psi).

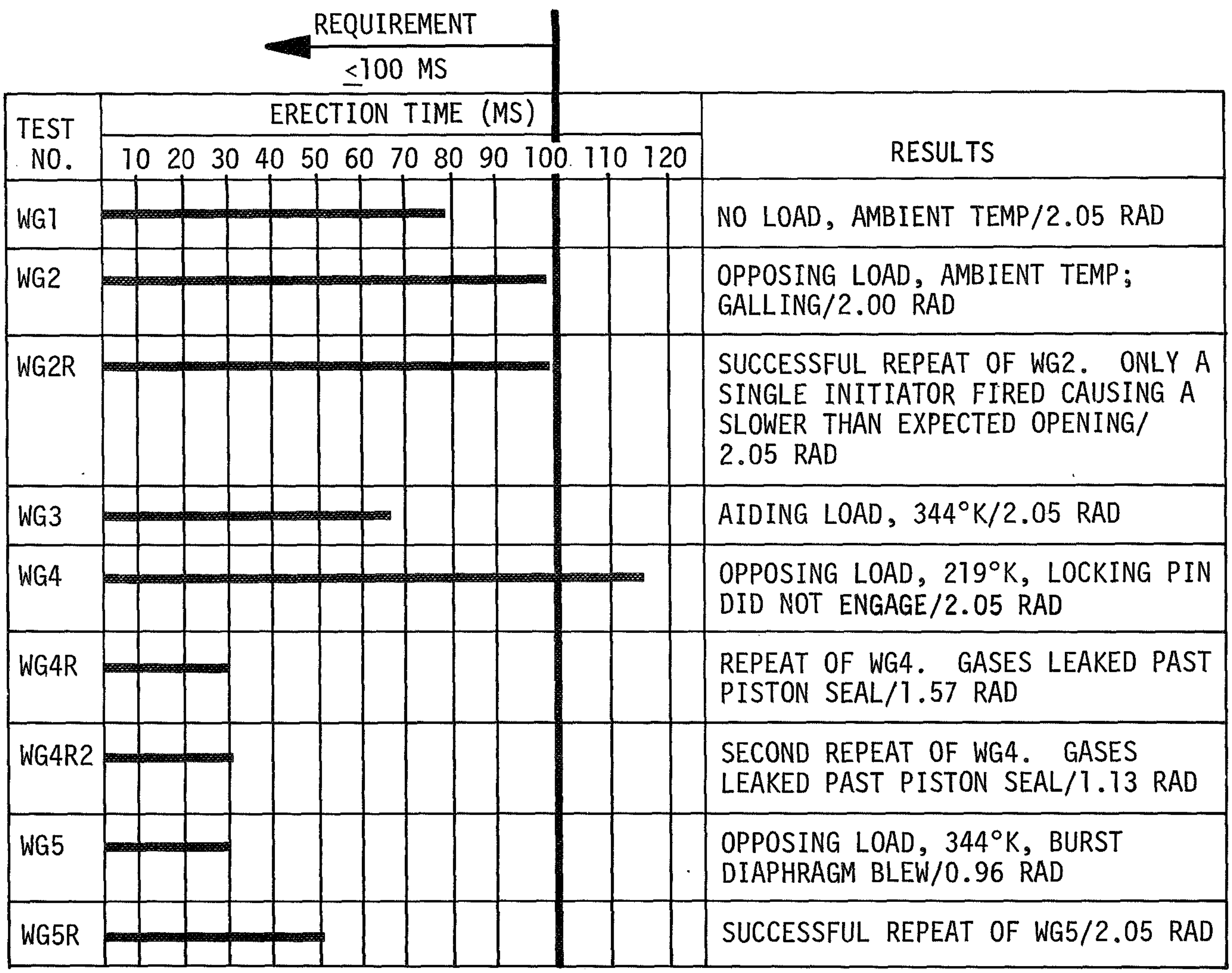

Figure 8. AVES Test Performance