DEPLOYMENT MECHANISMS ON PIONEER VENUS PROBES*

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ABSTRACT

Deployment mechanisms were developed to position scientific instruments during probe descent into the Venus atmosphere. Each mechanism includes a provision for pyrotechnic release of the enclosure door, negator springs for positive deployment torque, and an active damper using a shunted d-c motor. The deployment time requirement is under 2 seconds, and the deployment shock must be less than 100 g's. The mechanism is completely dry lubricated and constructed mainly of titanium for high strength and high temperature stability. The mechanism has been qualified for descent decelerations up to 565 g's and for instrument alignment up to 940°F. The paper describes the mechanism requirements, the hardware design details, the analytical simulations, and the qualification testing.

INTRODUCTION

The Pioneer Venus mission includes a Probe Bus that carries three small probes and one large probe to Venus for release and descent into the Venus atmosphere. Each of the three small probes, as shown on Figure 1, has two deployment mechanisms designed to stow scientific measurement instruments during the 565-g deceleration of descent. When the Probe has reached 65 Km altitude from the surface, the enclosure doors are pyrotechnically released to allow rapid deployment of the Small Probe Net Flux Radiometer (SNFR) and the Small Probe Atmospheric Structure Experiment (SAS) instruments. Atmospheric data are taken for the last 65 Km of descent until the Probe lands. The Venus atmosphere is very dense, reaching 1400 psia at the surface, and aerodynamic heating causes a maximum temperature of 940°F on the mechanism.

The generation of the mechanism design requirements has covered many variables such as entry angles, spin speed and direction, Probe nutation and aerodynamic turbulance torques. For instance, the lumped environmental torques could either aid the SAS deployment as much as 3.0 in-lbs or possibly retard SAS deployment as much as 8.5 in-lbs. This extreme variation in environmentally applied torques meant that a large spring force must be applied to ensure deployment, but damping provisions must also be made to prevent instrument damage should these environmental extreme forces not be present.

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In addition to the deployment requirements, the instruments have to be protected structurally and thermally during the deceleration prior to deployment. The SAS and SNFR instruments are stowed aft of the conical heat shield to avoid temperatures of several thousand degrees during initial entry. A full enclosure, which is covered with silicone rubber ablative material, is still required for mechanism protection.

The six mechanisms are now installed on the small probes and are undergoing system testing. The launch of the Pioneer Venus multiprobe mission is planned for August 1978.

REQUIREMENTS

The mechanism design requirements can be divided into three basic modes of operation as listed below. Table 1 lists the specific parameters required of each mode.

1. Stowed Mode. The instruments are kept folded into a retracted position by the enclosure door. While in the folded condition, the mechanism is subjected to vibration and entry deceleration simulations up to 706 g's for qualification.

2. Deployment Mode. The enclosure door is pyrotechnically released by a bolt cutter. As the door is sprung open, the instruments must be deployed in less than 2.0 seconds. However, if the deployment is too rapid, the stopping shock must not exceed 100 g's at the instrument tip.

3. Descent Mode. The mechanism must withstand the high pressure and temperatures up to 940^{o}F while maintaining alignment of the instruments within $\pm$ 1 degree.

MECHANISM TRADEOFFS

The deployment time requirement of less than 2.0 seconds along with the deployment shock restriction made the possible mechanism approaches very limited. An undamped spring deployment design showed impact shocks above 400 to 500 g's. In addition, the rebound problems with the undamped spring approach could only be solved with a deployed latch which had difficult dynamic and strength requirements.

An active d-c brush motor could be sized to drive the mechanism but requires active rate feedback to fall within the impact requirement. The rate feedback closed-loop electronics for each of six mechanisms makes the cost and weight penalties much higher than the baseline. The more simplified stepper motor driver electronics are more attractive for rate control than the d-c motor system, but the stepper motor would be much heavier. A heavier stepper motor is required because the 2.0 second deployment time means a small gear ratio and high motor torque for control.

The damped spring approach could use viscous fluid, rubbing friction, or electromagnetic damping. The nearby science payloads along with the wide temperature excursion ruled out the viscous damper because of concern for outgassing. The electromagnetic damper was selected over the friction damper due to the fine tuning capability and consistency of the d-c motor. The d-c motor can be fine tuned using a shunt resistor to allow for changes in requirements or variations from unit to unit.

DESIGN DESCRIPTION

The deployment mechanisms and protective housings are designed to accommodate both the SNFR and SAS instruments interchangeably. There are differences in instrument attachment, interconnections, and instrument aerodynamic shapes. The mechanism has different deployment springs, different stop angles, and damper settings for each type of instrument. The diagram of Figure 2 describes the mechanism with the SAS in the deployed position.

The mechanism includes a machined structural base that supports both the mechanism and the protective housing. The protective cover is deployed 90 degrees by a torsional spring after the tie-down bolt is cut by a pyrotechnically driven device. The squibs and squib drivers are redundant. The instruments are deployed by a negator spring once the door motion has started. The door is deployed in 0.1 to 0.2 seconds which is more rapid than the instrument motion. As a backup, there is a mechanical finger on the instrument platform to safely push the cover ahead in case of interference. The cover support is captured by the wedging action of a C-shaped clamp to absorb excess energy and to prevent rebound.

The instrument deployment is driven by a 301 CRES negator spring selected for its constant torque properties. The damping is provided by a d-c brush motor geared to a higher speed by a 38.3:1 ratio. The motor is mounted on ball bearings to preserve the air gap, but dry-lubricated journal bearings are used elsewhere. The titanium gears have Vitrolube 1220 MoS_2 dry lubricant which was selected for the binder cure temperature of 950°F to minimize further outgassing during descent. There is no latch in the deployed position. A cantilevered beam provides a spring action stop that is rigid enough to maintain alignment after a short settling time.

The photographs of Figures 3 and 4 show the mechanisms in the deployed condition with the housings removed. The instrument deployment negator spring and the cantilevered beam stop are depicted in Figure 3. The base structure is machined 6AL4V titanium because of the high strength requirements during entry and thermal expansion compatibility with the mating probe structure. Beryllium copper is the shaft material selected to act as a journal bearing and for shock absorbing properties, since it has a low modulus of elasticity compared to steel. The mechanisms' physical characteristics are shown in Table 2.

ANALYTICAL MODEL

The flight loading environment at the time of deployment, as depicted in Figure 1, is severe and difficult to predict. Trajectory dependent and hence time varying aerodynamic conditions are primarily within the transonic region characterized by bow shocks and complex local flow fields. Attitude motion of the probe generates time varying inertial loads in addition to axial deceleration. Asymmetric ablation on the aeroshell can induce vehicle spin of up to 100 rpm resulting in centrifugal force fields. In addition, the mechanism performance is dependent on design parameters such as friction, spring tolerances, cable bending effects, and motor damping which are all in varying degrees environmentally dependent.

The time sequencing and complex interaction of both environmental effects and dispersed design parameters necessitates the development of analytical models in order to assess mechanism performance and structural integrity. A computer program was written incorporating both probe and mechanism dynamics for time simulation of the rigid body aspects of the deployment process. A finite element model and standard modal analysis techniques are employed to determine structural response at impact. Parametric studies are utilized to establish the combination of extreme conditions which result in worst case fast and slow performance as summarized for the SAS in Table 3.

The level of torque for the negator deployment spring is selected to ensure positive torque margin throughout deployment without benefit of momentum. In other words, should the instrument momentarily stall at any intermediate angle, the negator must have sufficient torque to restart. The significant torque variables are displayed on Figure 5 for conditions influencing the SAS deployment. The static torque margin shown on Figure 5 never falls below zero. The resulting fast and slow deployment times are graphically shown on Figure 6. The computer results are shown in Table 4 depicting the wide spread in all parameters between fast and slow deployment conditions.

The cantilevered beam used as a spring stop is necessary because the tip accelerations during stopping exceed 100 g's even with an active damper. A key contributor to the stopping acceleration is the shock produced by ongoing motor rotor energy after the instrument reaches the end of travel. Therefore, the use of higher damping coefficients cannot directly solve the problem without far exceeding the 2.0 second limit in slow deployment cases. Figure 7 describes the relationship between stop spring stiffness and tip acceleration for the fast deployment case. The figure also shows the overtravel tradeoff which is an important consideration for enclosure clearance.

TEST PROGRAM

A comprehensive test program was conducted on a prototype unit to cover all aspects of environments, including high-g, vibration, deployments under many environmental simulations, and descent temperatures. Two of the six flight units were tested to qualification levels of acceleration, temperature and vibration. The remaining four units were subjected to acceptance levels of acceleration, temperature, and vibration. The acceptance criteria were in terms of deployment time, deployment shock, and deployed alignment measurements. The pyrotechnic bolt cutter release was done once in development and four times in qualification to ensure compatibility with the mechanism. In order to save on the cost of the non-reusable bolt cutters, manual releases were done for most deployment tests.

The high-g testing turned out to be most revealing in discovering design deficiencies. Several cases occurred where structural distortion at 706 g's caused mechanical contacting of delicate parts not designed to carry loads. These areas had to be reinforced with structural stiffeners along with better supports for electrical wiring. During the first high-g test, the free end of the negator spring was driven off its support post so that subsequent deployment could not occur. A special hook was designed to prevent this unacceptable negator motion during high-g forces and to spring away during normal deployment.

The probe descent simulation of the high pressure and $940^{\circ}F$ temperature was conducted on the development model only because of permanent damage to some parts of the system. Since it would be very difficult to measure instrument angle change during exposure, measurements were made on a fixture before and after exposure. Unfortunately, the instrument angle changed about 1.5 degrees as a result of this test. Subsequent testing was conducted to isolate the problem to permanent set of the beryllium copper cantilever stop spring shown on Figure 3. It was discovered that beryllium copper creeps to a new permanent set when exposed to $940^{\circ}F$ under the residual preload of the negator deployment spring (approximately 15 in-lbs). The cantilever stop spring was then changed to a 17-4 PH CRES design with the same spring rate. The mechanism exposure was rerun with a small acceptable change in alignment of only 0.035 degree.

The deployment tests were conducted at high and low temperatures, but most data were collected at room ambient conditions. A loading fixture was designed to apply either aiding or restricting torques to simulate the extremes of the environment. Provision was made to add a shunt resistor to the motor to trim the extent of damping of each unit. As it turns out, a 5 ohm resistor was suitable for all units. The results are summarized in Table 5. It should be noted that the data show results that are less than one-half the allowable specification limits in time and g-loading. The analytical results showed a much wider dispersion predicted at Venus because the aerodynamic induced torques as shown on Figure 5 have a more severe effect than the linear simulation used in test. The computer simulations were rerun using the laboratory induced torques and verified this difference.

CONCLUDING REMARKS

The development of these mechanisms shows a high degree of sophistication and confidence in performance predictions compared to just a few years ago. For instance, the use of dry lubrication on gears, ball bearings, journal bearings, and motor brushes is now well enough defined to predict friction and life instead of just survival in space. The use of the computer simulations of mechanism behavior now allows all variables to be studied so that the testing matrix can be very much reduced. However, it continues to be important to touch on each test environment to check the level of accuracy of the computer simulations and be sure that some critical condition has not been overlooked.

Table 1. Requirements summary for qualification

Launch Mode (Stowed)	
Sine vibration - lateral	45 g 45-60 Hz
thrust	30 g 25.5-100 Hz
Random vibration	12 g rms overall
Acceleration	706g
Temperature	-98°F to $+170^{\circ}$F
Deployment Mode	
Deployed angle - SAS	$160^{\circ} \pm 1^{\circ}$
SNFR	$120^{\circ} \pm 1^{\circ}$
Peak tip acceleration	100 g
Deployment time (including door opening)	2.0 seconds
Overtravel allowance	5°
Temperature	-98°F to $+123^{\circ}$F
Acceleration	5.5 g
External torques - SAS	13.0 in-lbs peak
(lumped) SNFR	8.5 in-lbs peak
Deployed Mode	
Deployed angle error	$\pm 1^{\circ}$
Temperature	-98°F to $+940^{\circ}$F

Table 2. Mechanism physical characteristics

Enclosure structure	6AL 4V titanium with silicone rubber ablative material	
Enclosure size (inches)	4.2 W x 9.4 L x 5.8 H	
Weight summary (lbs)	SAS	SNFR
Enclosure	2.01	2.01
Base structure	1.04	1.04
Mechanism parts	1.71	1.74
Instruments	0.29	0.80
Total	5.05	5.59
Instrument connections		
SAS	6 electrical wires 1 bellows	
SNFR	9 electrical wires	
Damper motor		
Weight (lbs)	0.15	
~~Static torque (in-lbs)~~	~~0.94~~	
Damping (in-lb-sec/rad)	1.10	
Motor resistance (ohms)	6.5	
Negator spring torque		
SAS (in-lbs)	19 $\pm$ 1	
SNFR (in-lbs)	16 $\pm$ 1	

Table 3. Definition of worst-case deployment conditions

	SAS	
	FAST	SLOW
Trajectory and aerodynamics	$\gamma_E = -20^o$ (1) Maximum aiding aero.	$\gamma_E = -75^o$ Maximum retarding aero.
Angle of attack	-12^o	$+12^o$
Attitude dynamics	Outboard = +0.3 g Inboard = -1 g	Outboard = -0.3 g Inboard = -1 g
Negator spring	20 in-lbs	18 in-lbs
Motor damping	1.5 in-lb-sec/rad	5.2 in-lb-sec/rad
Temperature	123^oF	-103^oF

(1) γ_E is the probe trajectory entry angle relative to zenith.

(2) Aiding and retarding aerodynamic torques have 30% dispersion margin.

Table 4. Performance results of the computer simulation

CONDITIONS		DEPLOYMENT PARAMETERS			
		Peak Accel. (g's)	Elapsed Time (sec)	Tip Overtravel (deg)	Alignment Error (deg)
SAS	Fast	51	0.24	2.8	-0.005
	Slow	6	1.52	0.3	+0.14
SNFR	Fast	75	0.23	3.0	-0.05
	Slow	8	1.29	0.3	+0.1
Specification maximum		100	2.0	5.0	1.0

TABLE 5. Deployment data

Configuration	Temperature	Average External Torque (in-lbs)*	Deployment Time (sec)	Deployment Shock (g)	Deployed Angle Error (deg)
SAS	Ambient	0	0.34	42	0.45
	Ambient	+ 4.5	0.29	43	0.70
	Ambient	-10.0	0.80	13	-0.05
	Ambient	0	0.34	40	-0.08
	High	+ 4.5	0.29	45	-0.05
	Low	-10.0	0.83	10	-0.73
	Ambient	0	0.35	35	-0.22
SNFR	Ambient	0	0.37	48	0.11
	Ambient	+ 5.2	0.27	60	0.26
	Ambient	- 6.3	0.80	12	-0.02
	Ambient	0	0.35	51	-0.01
	High	5.2	0.22	65	+0.06
	Low	6.7	0.88	11	-0.39
	Ambient	0	0.31	42	-0.06

*Aiding torques are positive and restricting torques are negative.

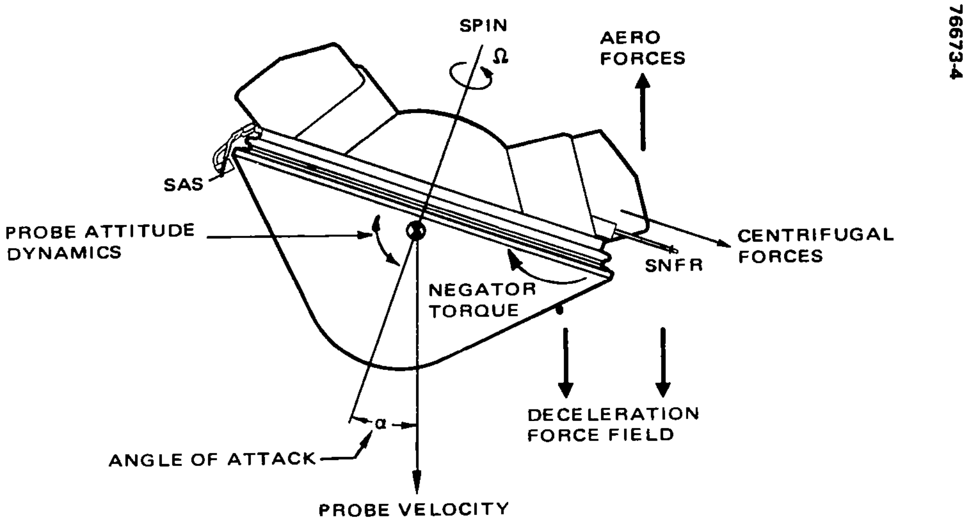

Figure 1. Pioneer Venus Small Probe

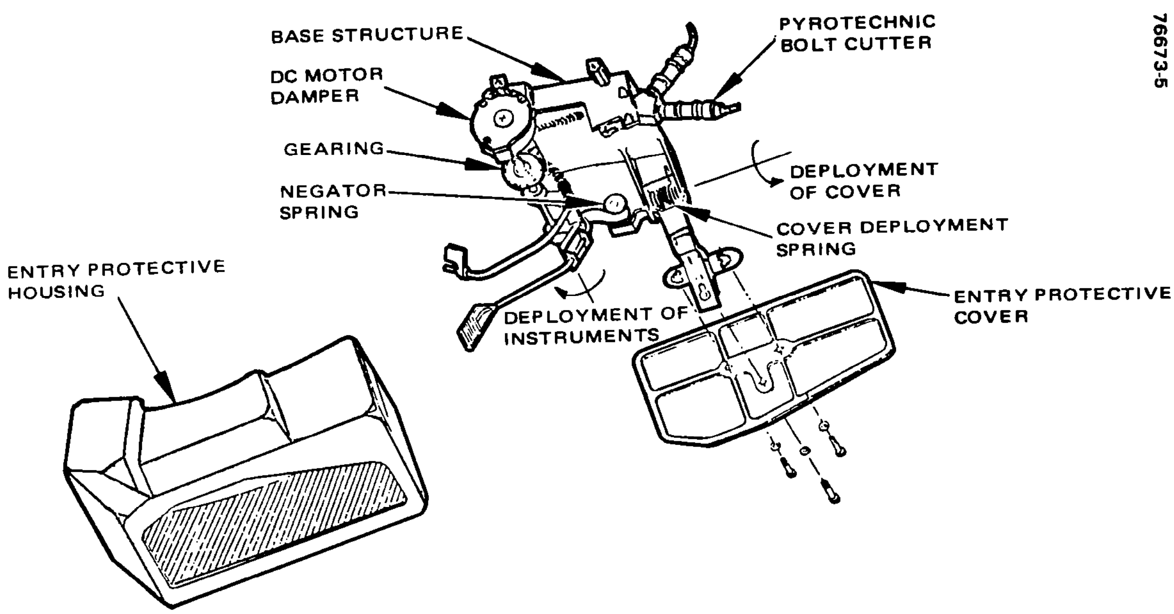

Figure 2. SAS Deployment Mechanism and Cover

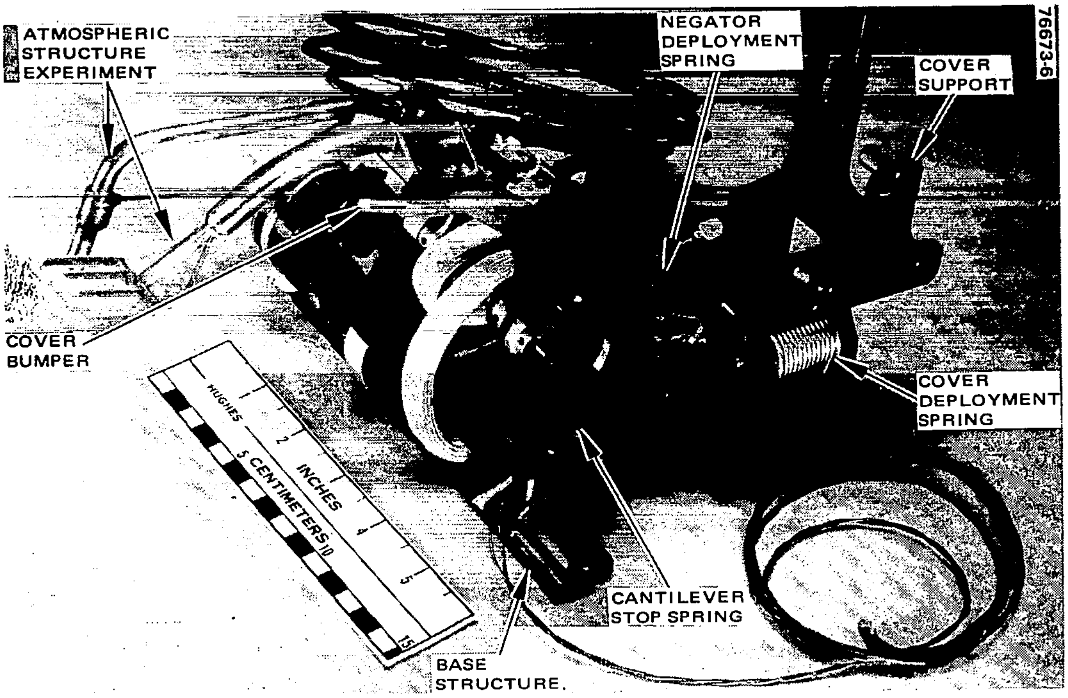

Figure 3. SAS Deployment Mechanism

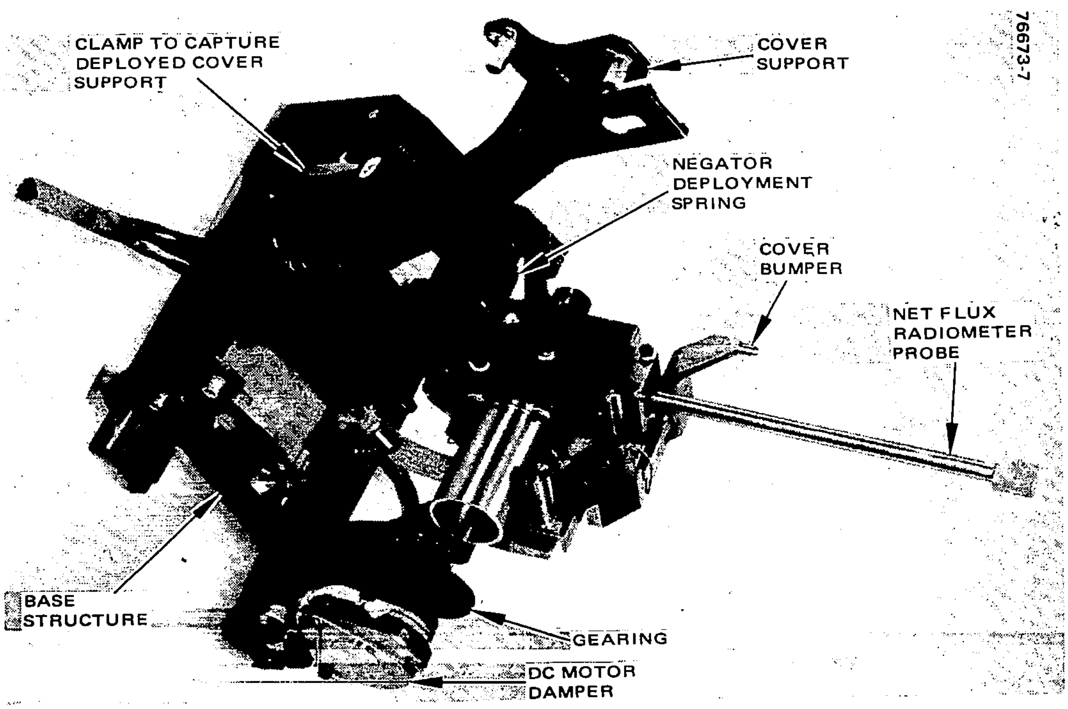

Figure 4. SNFR Deployment Mechanism

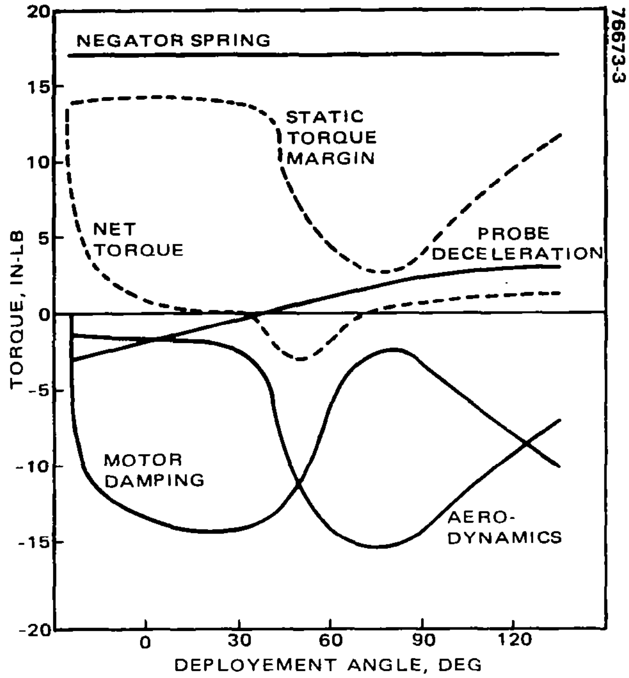

Figure 5. SAS Minimum Deployment Torques

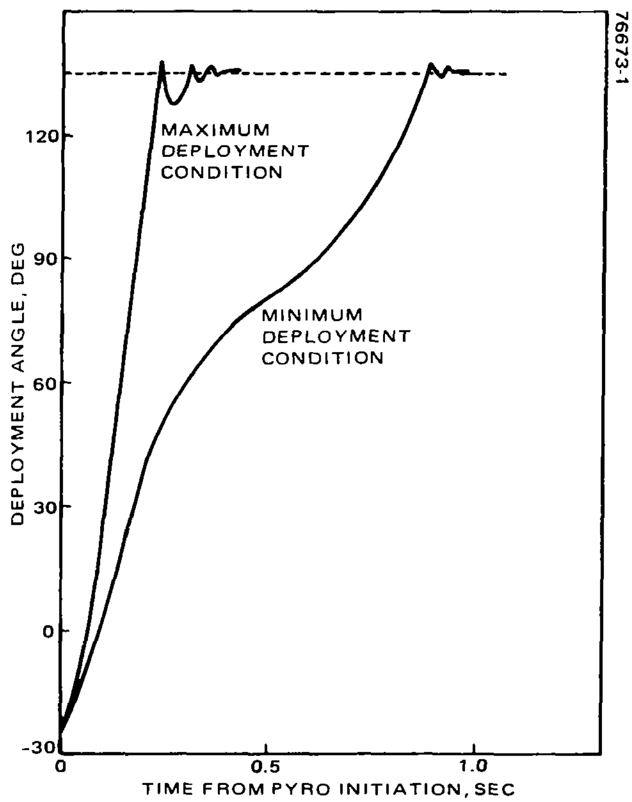

Figure 6. SAS Deployment Time History

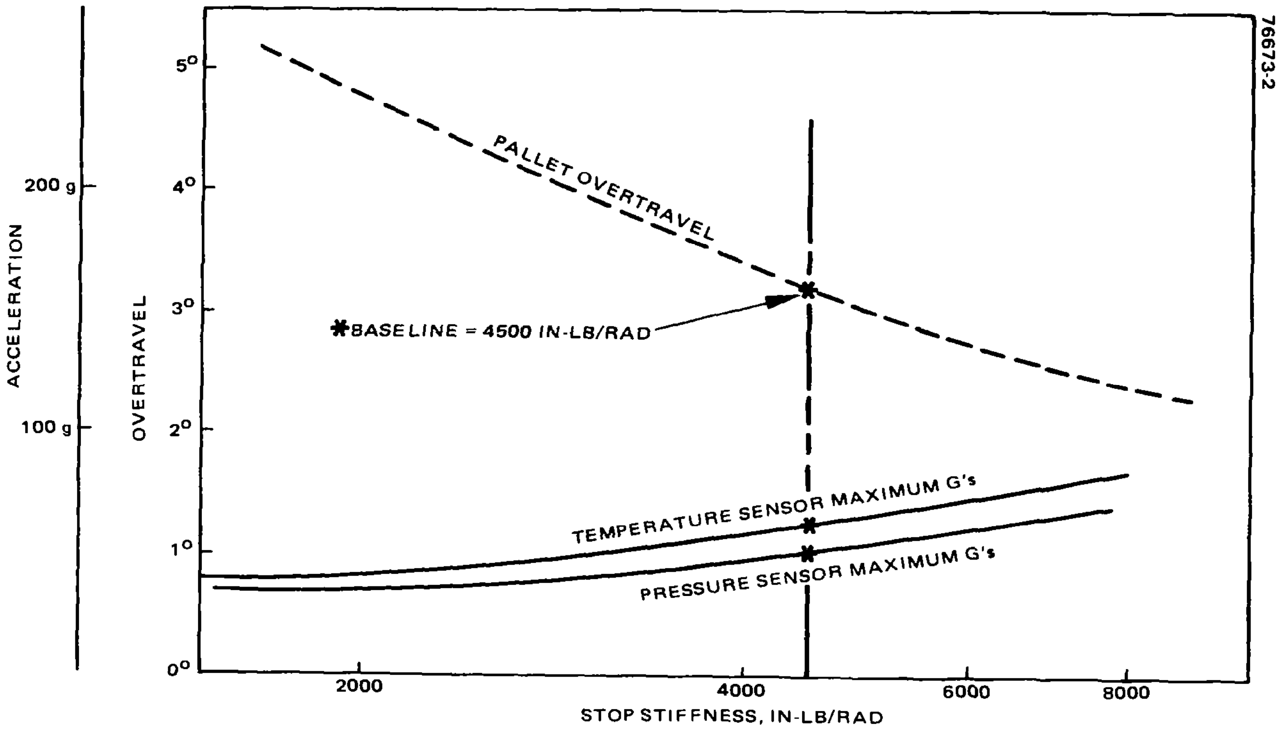

Figure 7. SAS Impact and Overtravel Response