N83 24904 D23

# HINGE LATCH MECHANISM

John C. Walker*

## ABSTRACT

The device described in this paper is a modular hinge latch mechanism. It is used to latch and preload four motorized hinges after their deployment. The paper describes design requirements, hardware development and initial testing of this hinge latch module. A special effort has been made to cover the design philosophy incorporated into this hardware.

## INTRODUCTION

This latch meets the hinge-lock requirements for a large space shuttle-flown satellite. The satellite has two solar panels and one antenna which are deployed after it disembarks from the shuttle. A schematic of the satellite's stowed and deployed configurations is shown in Figure 1.

Each of three types of hinges is equipped with a latch mechanism designed to reliably clamp the hinges shut under all combinations of dynamically induced loading. Upon command, the latch must also unclamp, allowing the antenna and solar panels to be restowed. The latch mechanism is modular and common to the three hinge types. Installation of the latch module in each hinge type is illustrated in Figure 2. An adjustment allows the preload to be customized for each hinge. Before being installed in flight hardware, each module is calibrated in a fixture such that the actual preload generated by the device is measured by a strain gauge.

The latch mechanism is a double over-center four-bar linkage. Preload is generated by driving the linkage into a latched position where a spring beam is deflected. The crank link is driven into adjustable eccentric stops by a reversible motor/gearbox. The adjustments allow control of a critical design parameter, over-center distance. While over center in the latched position, the preload generated at the latch tip keeps the linkage biased against its stop. While over center in the unlatched position, a compression spring-plunger assembly provides the bias force.

Latched and unlatched linkage positions are sensed by microswitches. The switch modules are adjusted relative to the eccentric stops to yield 2.54 mm (.100 in) trip arm motion.

*Ball Aerospace Systems Division, Boulder, Colorado

The latch mechanism is electronically and mechanically redundant - with the exception of the gearbox. Dry film lubrication is used on the linkage and gearbox, with Bray Oil Company 3L38RP grease used on the motor ball bearings. A lightweight, compact design was achieved through the use of 6Al-4V titanium and Carpenter Custom 455 stainless steel.

## REQUIREMENTS

The general design requirements for this device consist of:

| | | |
|---|---|---|
| • Temperature: | -54°C to +71°C (-65°F to +159.8°F) | |
| • Humidity: | 0-100% RH nonoperating mode for up to five days<br>0-55% RH during storage | |
| • Electrical: | 21-35 Vdc power for motors and switches | |
| • Switches: | Latched and unlatched position indicators | |
| • Stress Factors of Safety: | 1.5 on Ultimate Tensile Strength<br>1.2 on Yield Strength | |
| • Fasteners: | All fasteners employ means of positive locking | |
| • Outgassing: | Materials to meet requirements of NASA JSC SP-R-0022 | |
| • Redundancy: | All electrical and mechanical devices shall be redundant, with the exception of the gearbox | |
| • Drive Torque: | Static drive torque under worst-case conditions shall be twice the maximum load | |
| • Minimum Preload: | Solar Array Hinge (SAH) | 3343 N (752 lb) |
| | Antenna Inboard Hinge (AIH) | 1879 N (422 lb) |
| | Antenna Outboard Hinge (AOH) | 1467 N (330 lb) |

## LINKAGE - SPRING BEAM

The latch mechanism contains a double over-center four-bar linkage. Linkage hardware is illustrated in Figure 3. Preload is generated by driving the linkage into a position where a spring beam is deflected. Being biased over center while in the latched position allows preload to be maintained with the motor power off. The over-center action, while in the unlatched position, eliminates the need for a motor brake. The high mechanical advantage produced when the linkage goes over center minimizes the size of the motor/gearbox required to drive the mechanism.

Linkage design was optimized on a computer. Final linkage geometry maximizes mechanical advantage near the latched over-center position. Constraining the design was the requirement that the same latch module must fit into all three hinge types. A computer plot of mechanical advantage vs. crank angle is shown in Figure 4. Once the linkage is defined, the motor/gearbox torque requirement is a function of the latch beam spring rate and the preload requirement. The beam spring rate is found by setting a beam deflection which is "large" (10X) relative to thermal expansion effects. The worst-case thermal expansion change in latch-beam to latch-tip position is .05 mm (.002 in). Thus, the latch-beam deflection was set to .50 mm (.020 in). This sets a spring rate since the preload associated with this deflection is known for each hinge. Small beam deflections require lower drive torques since the preload is being generated in the high mechanical advantage section of the linkage action.

Beam deflection is set with an adjustable latch tip. The tip has a spherical radius such that it may accommodate ±1 deg of misalignment. The maximum pull-in capability of the latch is 12.7 mm (.50 in).

The linkage uses redundant journal bearings throughout. The connecting link and lever link bearing pins are free to rotate in either of their bushings. The crankshaft is mounted in two pairs of concentric bushings.

## MOTOR

The drive motor is a reversible dc torque motor with samarium cobalt magnets. The motor is sized to deliver a minimum of .325 N•m (2.90 in-lb), which is twice the maximum static load. The motor delivers this at minimum voltage (21V) and maximum temperature (+71°C). The motor was selected for its high motor constant (Km). This means it is very efficient at turning electrical input power into mechanical output torque. The actuation time requirement dictated torque to be more important than power in this application. The samarium cobalt magnets contribute to the high motor constant and are insensitive to demagnetization during high current pulses. Motor redundancy is achieved by mounting two motors on one shaft. The primary operating mode uses one motor at a time. A summary of the motor's operating characteristics is given in Table 1.

## GEARBOX

High motor torque allows the use of a single-stage gearbox. A single-stage gearbox is desireable because it minimizes size, weight and complexity. The motor pinion is cantilevered off the motor shaft while the output gear is cantilevered off the linkage crankshaft. The gearbox output gear is integral with the crankshaft. The crankshaft drives the crank link through an involute spline.

The gears were designed per AGMA standards. Spur gearing was chosen for high efficiency and jam-free operation. Gear mesh characteristics are listed in Table 2.

Peak loading on the drive train occurs when the motor drives the crank link into the unlatched eccentric stop. The effects of motor torque and inertial loading are combined in the dynamic simulation of Figure 5. This simulation assumes that the motor is running at maximum no-load speed when it impacts the stop. To keep drive train stresses at an acceptable level, the output gear has been designed to act as a torsion spring. The peak dynamic load was reduced from 75.7 N•m (670 in-lb) to 44.7 N•m (396 in-lb) through this cushioning effect.

The gear tooth stresses due to this load are:

Tooth Strength (Motor Pinion Bending Stresses)

- Actual 849 M•Pa (123,160 psi)
- Allowed 1379 M•Pa (200,000 psi) ($S_{ult}$ = 1.5)
- Margin 0.08

Tooth Durability (Mesh Contact Stresses)

- Actual 2065 M•Pa (299,500 psi)
- Allowed 2637 M•Pa (382,500 psi) ($S_y$ = 1.2)
- Margin 0.06

In order to endure high contact stress without pitting, the gears are being case-hardened to RC 60 minimum with the ionitride process. The case depth will be .08-.12 mm (.003-.005 in) to ensure subsurface shear stresses will not flake the case off.

## SWITCH MODULE

Each latch mechanism contains two switch modules. One indicates when the linkage (crank link) is in the latched position while the other indicates the unlatched crank link position. Two microswitches are mounted in each switch module for redundancy. One trip arm actuates both microswitches when the crank link has depressed the contact button 2.54 mm (.100 in). The microswitch trip points are individually adjusted relative to the trip arm. Redundant compression springs bias the trip arm against an adjustable down stop. Figure 6 illustrates how these components are configured.

## MATERIAL AND LUBRICANTS

The structural components used in the latch mechanism are made from 6Al-4V titanium. This alloy was chosen because of its corrosion resistance and strength-to-weight ratio. Titanium acounts for approximately 39% of total latch module weight. The more highly stressed components are made of passivated Carpenter Custom 455 corrosion-resistant steel. These components include the gears, links, stops and journal bearings. The steel is heat treated to tailor its strength vs. toughness properties for each application. Custom 455 accounts for approximately 25% of the total latch module weight. The close match (within 20%) in thermal expansion coefficients between these two materials keeps bearing clearances and gear-mesh backlash relatively constant. The bulk of the remaining module weight (31%) is made up of the various materials used in the drive motor. The total latch module weight is 62.3 N (14.0 lb).

Two basic lubricant types are used in the latch module. A dry, bonded lubricant is the preferred type for intermittently operated devices which must survive long nonoperating periods. This is because it has a low evaporation rate and no surface creep. For applications with low total travel and moderate contact stresses, Ball Aerospace Systems uses a bonded molybdenum disulphide-based dry lubricant. This lubricant has been used at all latch module moving interfaces except the motor ball bearings and brushes. The motor ball bearings use a molybdenum disulphide-based grease to endure the high contact stresses encountered here. The motor brushes are made of a silver/molybdenum disulphide material, Stackpole SM 476. This material has been tested to twice the required latch motor life and found to wear only .018 mm (.0007 in) at each brush block.

## DEVELOPMENT PROBLEMS AND SOLUTIONS

### Thermal Effects on Preload

Thermal effects strongly influence drive requirements and worst-case loading. Because of the over-center linkage's mechanical advantage characteristics, the drive torque requirement is minimized by decreasing the preload spring deflection. That deflection must not be made so small that thermal expansion/contraction causes large preload variations. Thermal expansion/contraction effects on preload must be accounted for in the nominal preload setting and in worst-case structural loading, as indicated in Table 3.

### Peak Drive Train Loading

When the crank link is driven into the "hard" eccentric stops, loads occur which are much higher than the motor's stall torque. A single stage, low reduction gearbox increases this peak loading since gearbox stiffness is proportional to the inverse square of gear ratio. To lower stresses to an acceptable level (a 41% reduction) the output gear is designed to act as a cushion between the motor rotor inertia and the eccentric stops.

### Preload Spring Selection

A significant weight reduction was brought about by using the mounting structure itself as the dominant spring. Preload is generated in this device through the deflection of a spring. Initially, the lever link was designed as the dominant spring in this system. Choosing the lever link drove the spring rate requirements of the hinge-to-latch mounting structure up in order to achieve the desired deflection. This increased hinge weight.

## CONCLUSIONS

This mechanism achieves all hinge latch design requirements with a compact modular unit. The hinge latch module has a calculated reliability of .9997 for its design life. Design philosophy stresses redundancy where redundancy does not overcomplicate the design or vastly increase the parts count. The use of easily accessible adjustments on critical performance parameters yields increased machinability and lower cost. Computer automation of the linkage analysis (which would have been done graphically) allowed Design Engineering the time to optimize latch performance. Design response to changes in hinge latch requirements is also greatly enhanced through this automation.

Development testing of the hinge latch module is under way. Initial testing has substantiated the computer predictions of preload generation and drive torque requirements associated with a given preload setting. The development test hinge latch module is shown in Figure 7.

## ACKNOWLEDGEMENT

I would like to thank Hilary Lane and Gloria Bucco for their invaluable assistance in compiling and editing this paper. I also thank George Ahlborn for his work on the lubrication section and Lew Martin for his dynamic simulation of the latch impact.

## REFERENCES

1. John C. Walker; SAAM Hinge Latch Mechanism; BASD TN82-20; 11 November 1982.

2. Dave Paule; Hinge Latch Loads; BASD SER B8650-81-041 27 July 1981.

Table 1
MOTOR OPERATING CHARACTERISTICS

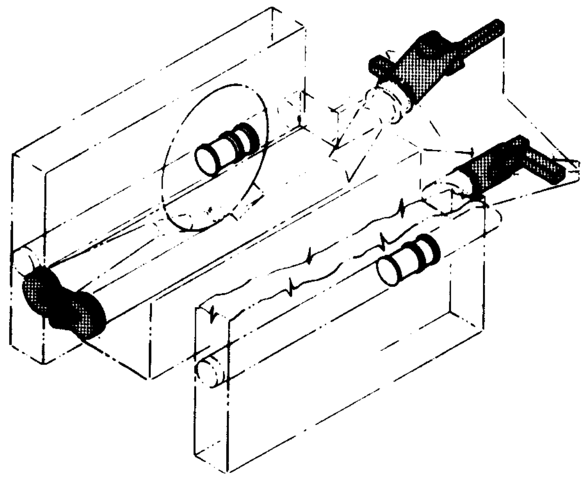
| | |
|---|---|
| MOTOR TYPE | DC TORQUE WITH BRUSHES |
| MAGNETS | SAMARIUM COBALT |
| MOTOR CONSTANT | .209 $N \cdot m/\sqrt{Watt}$ (29.6 $in\text{-}oz/\sqrt{Watt}$ ) |
| RESISTANCE | 54Ω |
| MAXIMUM NO-LOAD SPEED | 25.2 rad/s (241 rpm) |
| TORQUE CONSTANT | 1.54 N•m/A (218 in-oz/A) |
| BACK EMF CONSTANT | 1.54V/(rad/s) |
| ROTOR INERTIA | .125 $mm \cdot Kg \cdot s^2$ (.173 $in\text{-}oz\text{-}s^2$) |
| WEIGHT | 22N (5.0 lb) |

Table 2
GEAR MESH CHARACTERISTICS

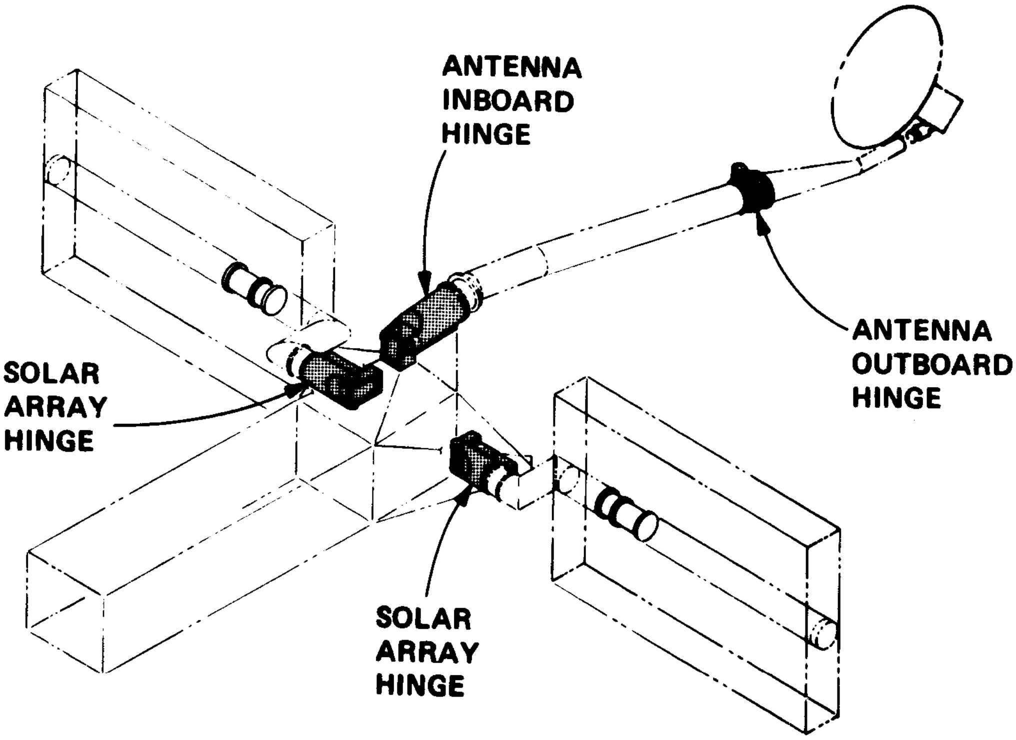
| | |
|---|---|
| TOOTH TYPE | SPUR GEARS |
| TOOTH FORM | FULL DEPTH INVOLUTE |
| AGMA QUALITY NUMBER | Q7A |
| PRESSURE ANGLE | 25 deg |
| DIAMETRAL PITCH | 48 |
| GEAR RATIO | 17.833:1 |
| CENTER DISTANCE | 59.8 mm (2.354 in) |
| MINIMUM FACE WIDTH | 7.04 mm (.277 in) |

Table 3
PRELOAD VARIATION DUE TO THERMAL EXPANSION
AND INSTRUMENTATION UNCERTAINTIES

| | Required Preload | Minimum Preload | Nominal Preload (22°C) | Maximum Preload |
|---|---|---|---|---|
| SAH | 3340N (752 lb) | 4010N (902 lb) | 4420N (993 lb) | 4990N (1120 lb) |
| AIH | 1880N (422 lb) | 2250N (507 lb) | 2390 (537 lb) | 2550N (574 lb) |
| AOH | 1468N (330 lb) | 1760N (396 lb) | 1830N (412 lb) | 1900N (428 lb) |



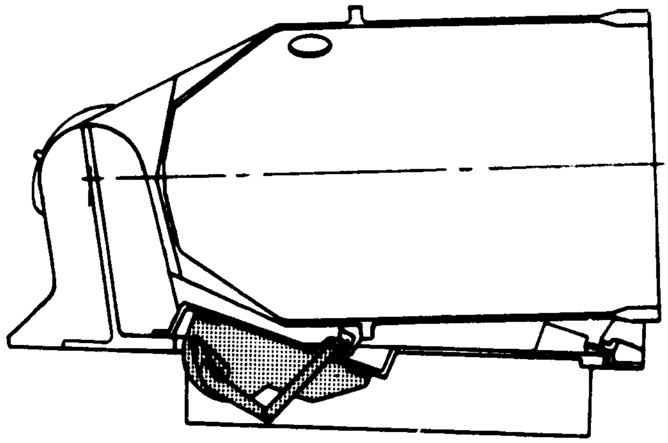
STOWED



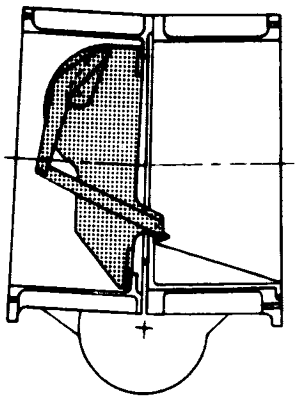

DEPLOYED

A/N 2972

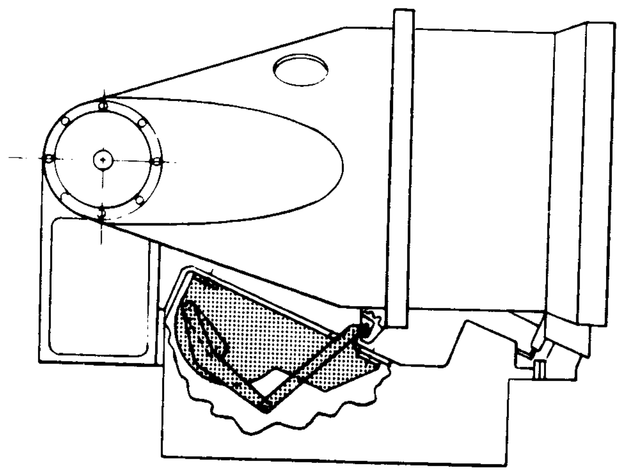
Figure 1 Stowed and Deployed Hinge Configurations



ANTENNA INBOARD HINGE



ANTENNA OUTBOARD HINGE



SOLAR ARRAY HINGE

A/N 2972

Figure 2 Latch Module Installations

ORIGINAL PAGE IS
OF POOR QUALITY

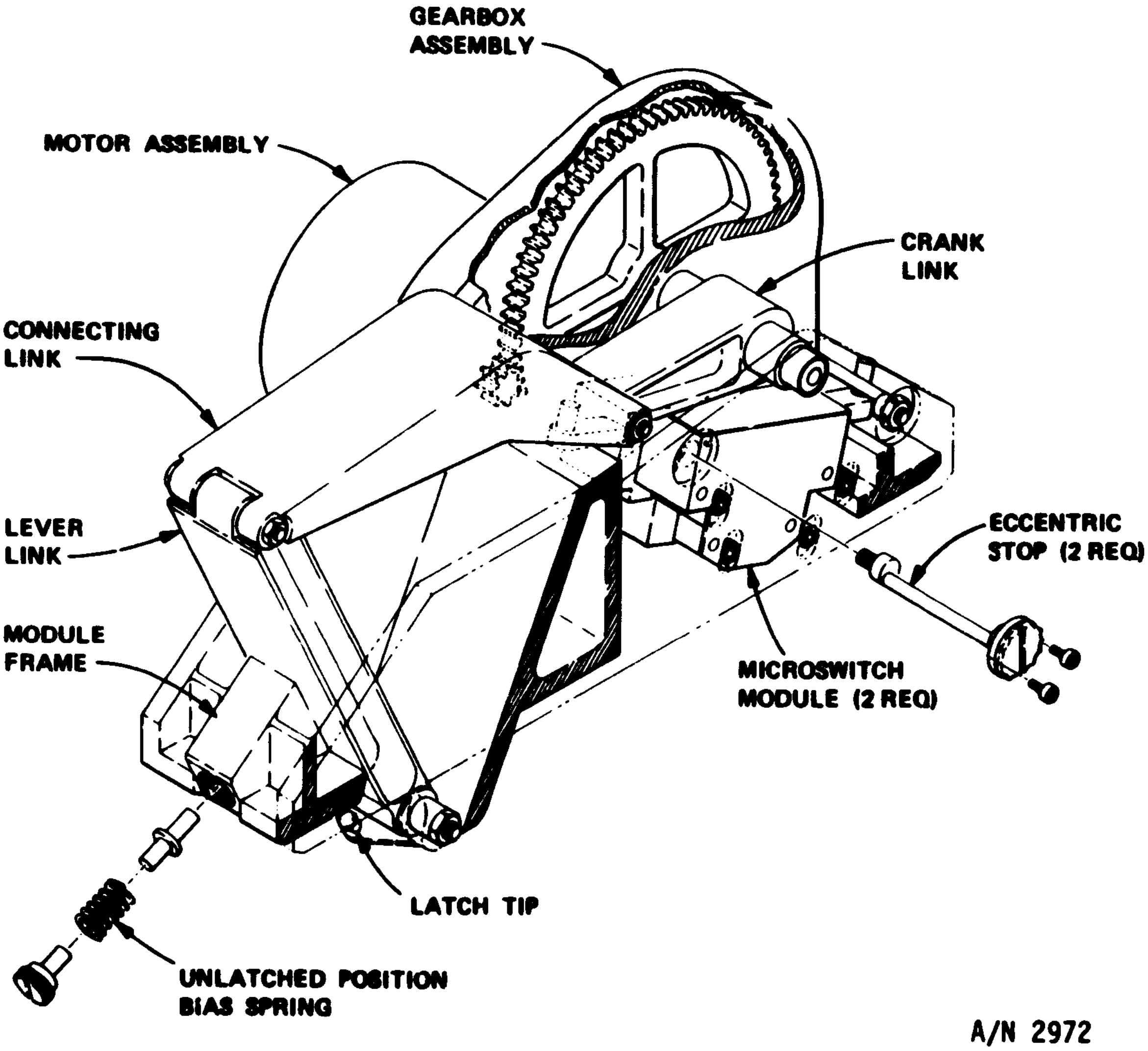
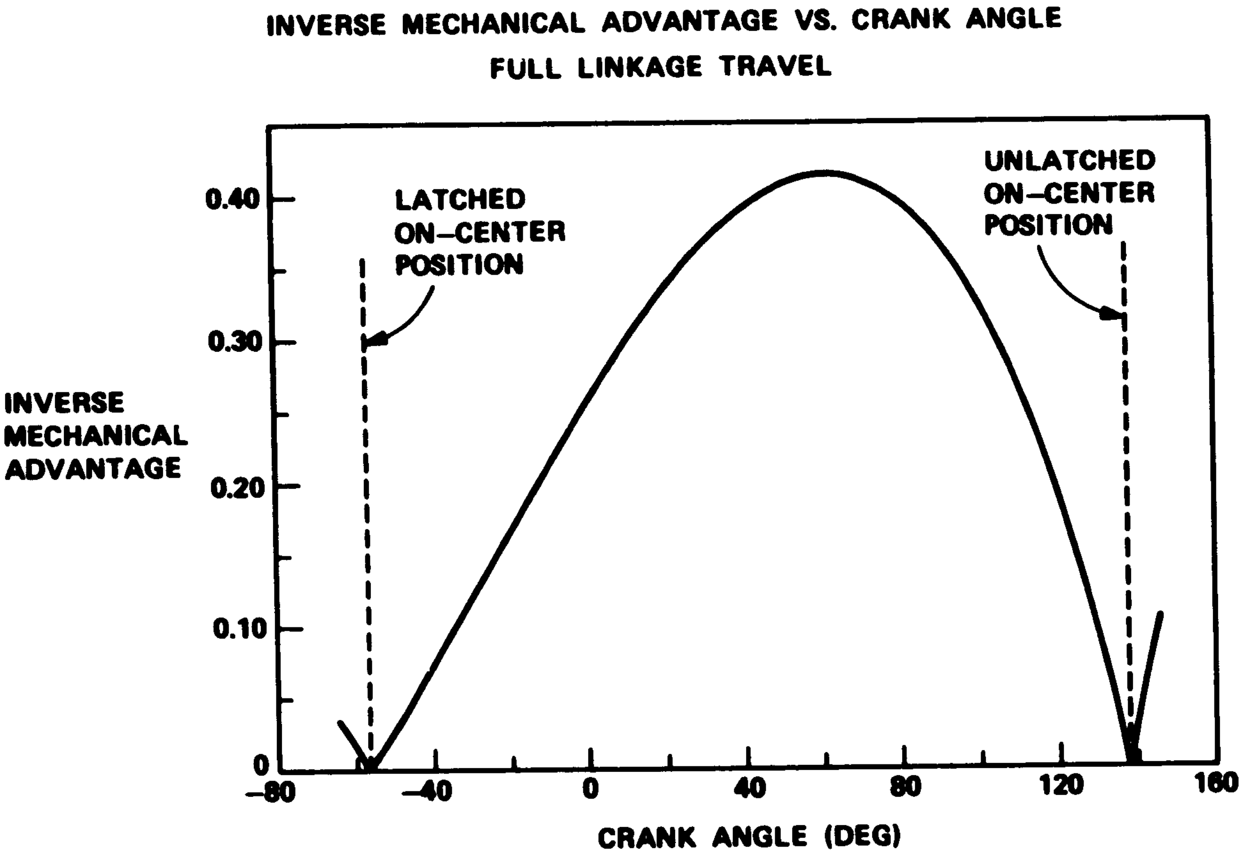



Figure 3 Hinge Latch Module

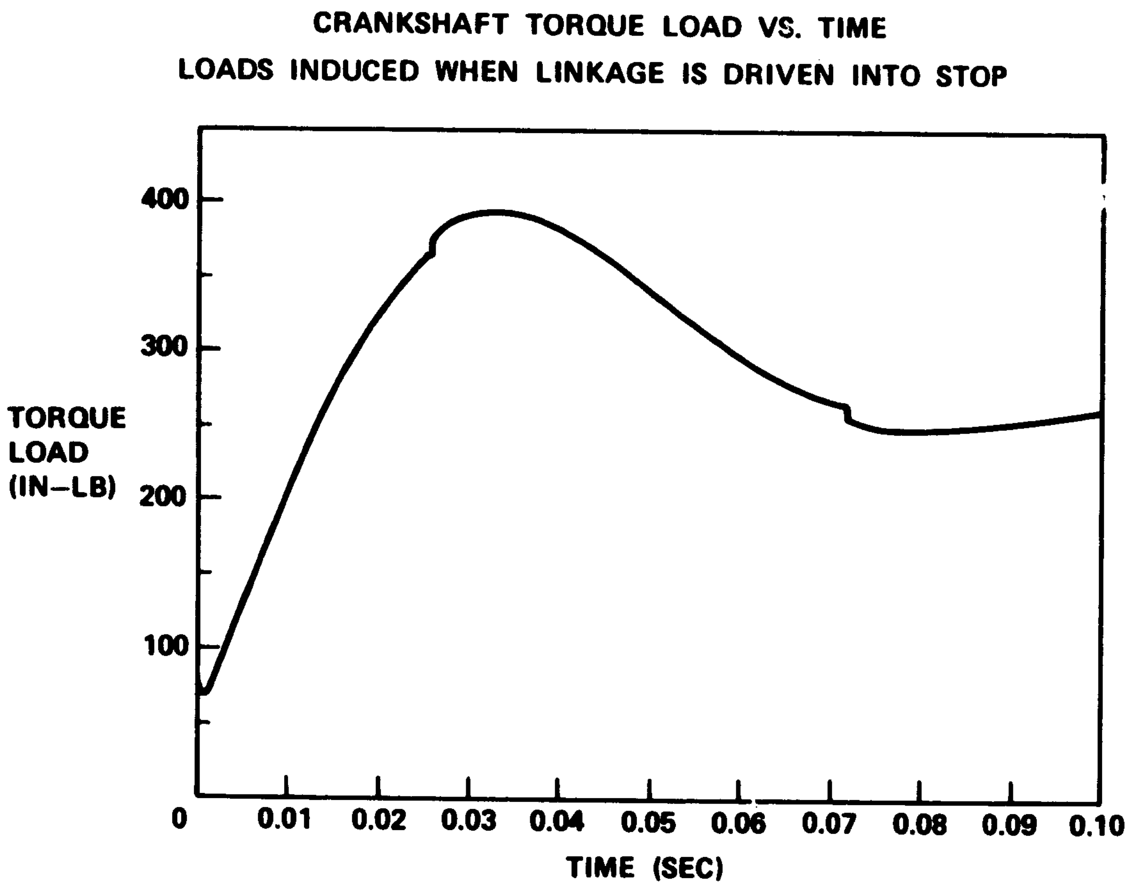
ORIGINAL PAGE IS
OF POOR QUALITY




A/N 2972

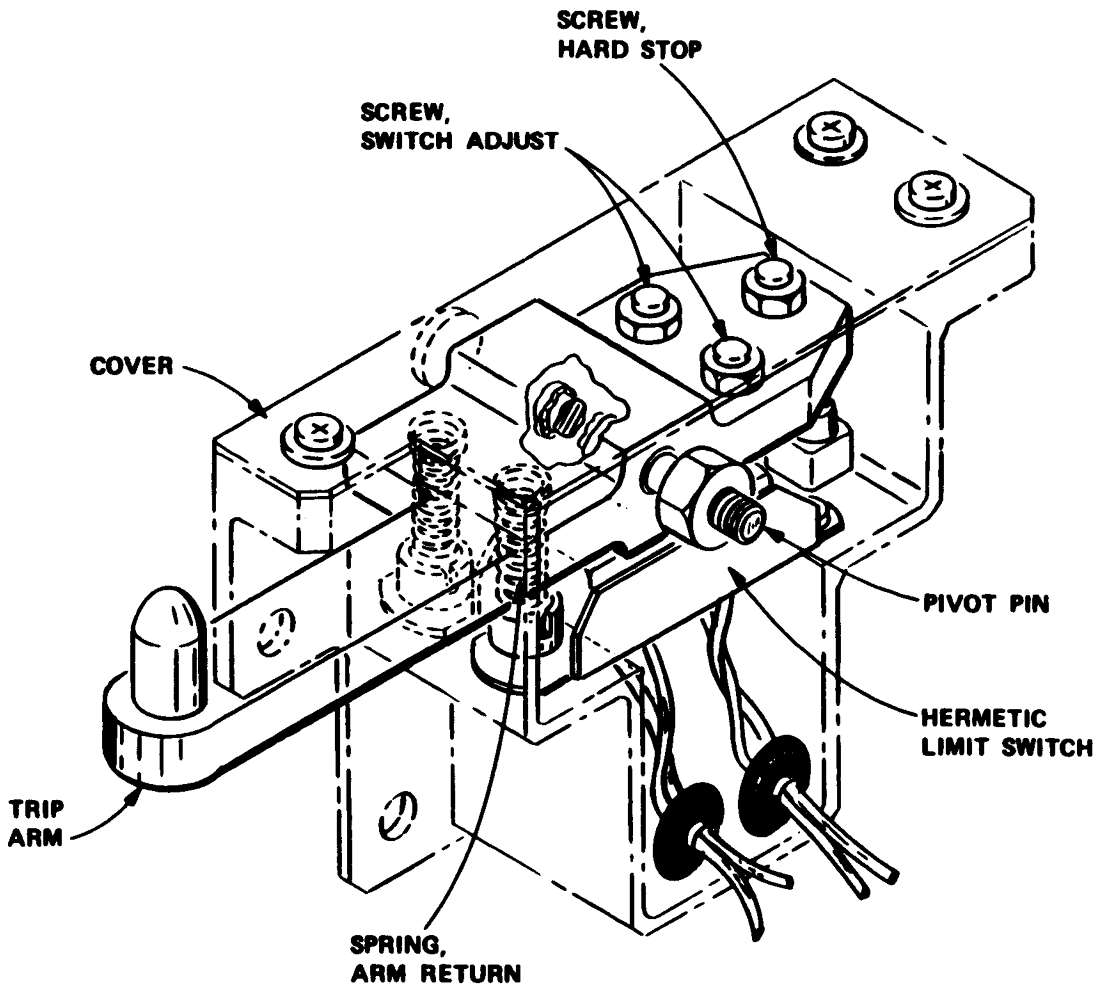
Figure 4 Linkage Mechanical Advantage

ORIGINAL PAGE IS
OF POOR QUALITY




A/N 2972

Figure 5 Dynamic Loading




A/N 2972

Figure 6 Switch Module

Figure 7 Development Test Latch Module