A BALL TRUNNION CAPTURE LATCH

David V. Adams and Brendan Alchorn*

ABSTRACT

The Ball Trunnion Capture Latch, developed under a research and development program conducted by Lockheed Missiles and Space Company, was designed to restrain a spacecraft deployable appendage in three translational directions. The latch is capable of supporting an appendage during STS ascent and landing events and is capable of releasing and restowing an appendage distorted in three translational directions by thermal growth. This paper discusses requirements, design, analyses, and tests conducted on a development unit of the latch.

INTRODUCTION

The function of the Ball Trunnion Capture Latch is to restrain a spacecraft deployable appendage through ascent and landing conditions. The latch must also recapture and preload the appendage during on-orbit conditions, when relative thermal growth may occur between the deployable appendage and the spacecraft supporting structure. Since the latch is to be of general utility, it must be insensitive to thermal growth in three orthogonal translational directions. The latch must be tolerant of the spatial distortion of such an appendage, and it must overcome any loads associated with pulling a distorted appendage back into place. The latch consists of dual four-bar linkages which are actuated by a motor-driven ball screw. In the over-center position, one link, called the latch jaw, closes down on a ball (spherical) trunnion, locking it between two conical cups. The ball trunnion provides the interface between the latch and the deployable appendage.

REQUIREMENTS

The latch must be capable of withstanding a limit opening load of 44,500 N (10,000 lb) against the jaw when the latch is closed. This load includes any opening loads induced by side loads reacting against the conic slopes of the two capture cups. Additionally, the latch must supply a ball trunnion preload of 4,450 N (1,000 lb) to prevent rattle during typical acoustic conditions seen at launch.

*Lockheed Missiles and Space Company, Sunnyvale, California

The latch must also be capable of pulling a trunnion displaced 0.762 cm (0.3 in) in the direction of actuation against a load varying from 0 to 4,450 N (1,000 lb). This must be accomplished while the trunnion is also offset 0.254 cm (0.1 in) in both directions [0.359 cm (0.141 in) total] orthogonal to the direction of jaw travel, overcoming an orthogonal load varying from 0 to 9,900 N (2,000 lb, vector sum). The latch must be capable of a minimum on-orbit life of 3 years, and single-point failures (excluding structure) must be minimized. The latch must operate between temperature extremes of -34°C (-30°F) and +88°C (+190°F) under conditions ranging from 0- to 100-percent humidity. The envelope of the latch measures 45.7 cm by 22.9 cm by 11.4 cm (18 in by 9 in by 4.5 in). An acoustic vibration criterion of 16.7 GRMS was selected based on typical responses measured on components of this type during launch of Lockheed-built spacecraft.

DESIGN DESCRIPTION

A ball-end trunnion caught between two conical cups was chosen as the design solution. This design configuration permits capture and restraint of the trunnion in three translational directions, while it allows some relief from rotational loads (depending on friction). Figure 1 shows the latch in the over-center position. The latch jaw is shown closed down on the ball trunnion, locking it between the two conical cups. One conical cup is located on the latch jaw while the other cup is fixed rigidly to the latch structure. A half-cone angle of 45° was selected for both cups, based on a trade-off between lock-down capability and capture capability. A narrower cone would have reduced the vertical reaction on the jaw but would have increased the distance required for jaw travel. The peak load on the ball screw actuator would have increased because loading would have begun when the mechanical advantage was lower. A shallow half-cone angle would have required higher jaw vertical loads to pull the trunnion into place.

Preload is achieved by placing a shim under the fixed cup. Motive power is supplied to the ball screw through a 1:2 speed-increasing gear train from redundant motors driving through a planetary differential into a common output shaft.

The link pivots and the ball screw supports are redundant. Each joint consists of a pin surrounded by a bushing which is free to rotate on either its inner or outer surface. The latch is shown in the fully open position in Figure 2.

ANALYSIS OF LATCH

Analysis of the latch was conducted in two phases: analysis of ascent loads sustained when the latch is closed and analysis of loads occurring during latch closure.

Ascent loads were treated as a static loading case, since the latch is closed over the ball trunnion and its internal dynamic reactions are insignificant. An analytical model of the capture cups predicts that the latch jaw takes loads only in the direction of its travel (+Y direction at closure). This is due to the fact that the jaw stiffness in the plane orthogonal to the direction of jaw travel (the X-Z plane) is low compared to that of the fixed capture cup. Further analysis shows that this assumption is conservative, since it results in calculating higher jaw vertical loads. The results of these analyses forced a change in the design of the latch. It was determined that pin diameters at each of the joints would have to be increased to accommodate the 44,500-N (10,000-lb) opening load.

An analytical model simulating operational performance of the latch was developed to verify latch closure in the presence of deployable appendage thermal growth. The model simulates latch kinematics, stiffness of the linkages and support structure, bearing friction, and ball trunnion/latch jaw pull-in forces. Dynamic forces were neglected, since actuation speed is low.

The kinematic relationship of the latch linkages is shown in Figure 3. A motor-driven ball screw, represented as link DE, actuates the four-bar linkage consisting of ground link AB, jaw BC, compression link CD, and tension link AD. At the full extension of DE, the jaw angle A6 is 0° and links AD and CD align with the Y (vertical) axis. This is the over-center, or closed, position of the latch.

Figure 4 shows how the stiffnesses of the various linkages, including the support structure, are modeled. Links AD, CD, and DE are considered simple extensional springs. Structural stiffnesses at joints A and B are considered decoupled for simplicity. Joint E, which is the ball screw support, is treated as an eccentrically loaded cantilever beam. Flexibility at E is represented by the two-dimensional flexibility matrix $[f_E]$. Latch jaw stiffness is computed by assuming it to be a simply supported beam with an overhanging load. For each spring shown in Figure 4, an effective stiffness is computed at the jaw based on the kinematic relationship between a unit spring displacement and the corresponding jaw displacement. By the principle of conservation of energy, unit strain energy in each link is equated to unit strain energy of an effective spring located at the jaw. An incremental change in length is computed for each spring due to the unit strain energy. This change in length causes a corresponding change in the position of the jaw. The jaw displacement is considered to be the displacement of an effective spring located at the jaw. Knowing strain energy and effective jaw spring displacement allows the calculation of an effective stiffness (transferred to the jaw) for each element. The effective stiffnesses of all the elements are then added in series to determine the total effective jaw stiffness. The total effective jaw stiffness changes with the geometry of the mechanism. Figure 5 shows the plot of jaw stiffness vs. jaw open angle (0° is the closed position).

If the effective stiffness of the jaw is known, then the position of the ball trunnion during pull-in can be calculated, based on the equations of static equilibrium. The simulation of ball trunnion/latch jaw pull-in forces assumes a rigidly fixed capture cup, a jaw with a finite stiffness in the horizontal (X-Z) plane, and a contact angle (based on the slopes of the capture cups) of 45°. For simplicity, we also assume that the trajectory of the ball trunnion projected onto the horizontal (X-Z) plane would consist of straight lines.

Forces and reactions in the latch linkages are then calculated based on the latch jaw load. Torque losses are computed for each joint due to an incremental change in position. The torque losses are considered work done by the ball screw during an incremental change in length. Since work done by the ball screw is equal to force time displacement, and the displacement and the work done are known, then an added force on the ball screw due to friction is computed for each joint. The various friction losses are then summed to obtain a total force on the ball screw force due only to friction. Frictional forces are then added to the ball screw for screw force assuming no joint friction to obtain the total ball screw force. Total ball screw force is shown vs. ball screw extension for a 13,350-N (3,000-lb) preload case in Figure 6, along with test data collected for this case.

It was thought desirable to check the results of the analytical model internally. To do this, an algorithm was added which compares the work done by the ball screw to work done on the latch in the form of friction, strain energy stored in the compliance of the latch, and work done to pull the trunnion into place. This algorithm helped to point out errors in both the coding and the synthesis of the model. Calculations of work done by the ball screw vs. work done on the latch typically differ by less than 1 percent.

DEVELOPMENT TESTS

A development unit of the latch was subjected to tests in order to verify capture, preload, and release capabilities of the latch and to validate the latch analytical model. A total of 10 tests were conducted on the development unit. The tests conducted fall into three general categories: compliance tests, tests of latch operational performance, and vibration sensitivity tests.

Compliance tests were performed in order to calibrate the jaw vertical load, measure the effective jaw stiffness, and determine the flexibility of the ball screw support structure. Calibration was accomplished by placing strain gages on either side of the tension and compression links and then pulling up with a known force on a ball fixture locked in the latch. Link bending due to friction torque required that the strain gages be wired in a moment-compensating circuit. This was an early indication of the significance of joint friction. This test allowed the determination of vertical jaw load in subsequent tests. Effective stiffness at the jaw was measured by inserting shims of various thicknesses under the fixed cup and recording variations in jaw vertical load. The results showed an effective stiffness of 8,400 N/cm (48,000 lb/in), which contrasts with an expected value of about 350,000 N/cm (200,000 lb/in). The difference between the two values can be attributed to bending and shearing of the linkage pins, phenomena which were not considered in the initial analysis. Later versions of the latch will feature larger diameter pins to accommodate ascent conditions and therefore should have higher effective stiffnesses. Pin stiffnesses were incorporated into the analytical model by merely adding them in series to the stiffness of the existing spring elements in the input of the model.

Two preload tests were conducted by closing the latch over a loose ball fixture resting in the fixed capture cup. The preload tests were conducted to determine the effect of joint friction without cup friction. Cup friction was eliminated, since the ball fixture did not move. Shims were placed under the fixed cup to obtain preloads of 6,230 N and 13,350 N (1,400 lb and 3,000 lb) and the latch was then closed and opened while recordings were taken of ball screw extension and required input torque. These tests indicated that joint friction approximately doubled the peak required drive-unit torque. The tests also verified the simulation of joint friction torque in the analytical model.

The Z direction pull-in test was conducted in order to determine the ability of the latch to capture, pull in, and release a ball fixture offset to the side of the latch. In this test, a ball fixture was attached to the end of a long threaded rod with a known axial stiffness (Figure 7). The threads on the rod allowed for position adjustment of the ball. The surfaces of the capture cups and the ball fixture were coated with solid-film lubrication prior to the test. The ball was positioned to rest on the near side of the fixed cup if not loaded. The latch was then closed, pulling the ball down into the cup. Tensile force developed in the rod was indicated by a load cell. Deflections of the ball, the jaw, and the fixed cup were recorded along with ball screw input torque, ball fixture axial force, and jaw vertical load for various ball screw extensions. Latch capabilities of capture, closure, and release were demonstrated. This test yielded information that verified the simulation of cup friction and side-direction stiffness of the jaw. The test also indicated that side forces on the jaw cause rubbing to occur at the side of the jaw, thus reducing the efficiency of the mechanism. This problem can be alleviated by treating the rubbing surfaces with solid-film lubrication.

The X and Y direction pull-in test was performed in order to determine the capability of the latch to capture, pull in, and release a ball fixture which was offset above (vertically) and forward (horizontally) of the fixed capture cup. The setup was similar to that used in the Z pull-in test, except that the ball fixture was not attached directly to either threaded rod. Instead, the ball was centered on a short rod which was supported at both ends by two U-shaped brackets (Figure 8). The brackets were attached to the threaded rods. As the latch was closed, the ball was pulled in simultaneously in both the X and Y directions. Again, latch capabilities for capturing, pulling in, and releasing a ball trunnion were verified. Data gathered in previous tests allowed the analytical model to simulate this test with fair accuracy.

The final test was a measurement of latch sensitivity to vibration. A free-floating trunnion ball was clamped into the latch so that a preload of 4,450 N (1,000 lb) was developed. The latch was then vibrated along three translational axes to determine the capability of the latch to remain closed and maintain its preload in the presence of vibration. The latch was subjected to sine sweep and random vibration levels of 16.7 GRMS, thus simulating conditions that might be seen during a typical launch. The latch remained closed. Typical landing conditions (with thermally induced loads orthogonal to the direction of jaw actuation) were not simulated, due to the complexity of the required test setup. Response of the latch was termed low (Reference 1). Preload capability was verified by subsequently releasing the ball fixture and then reclosing the latch while taking strain gage measurements.

CONCLUSIONS

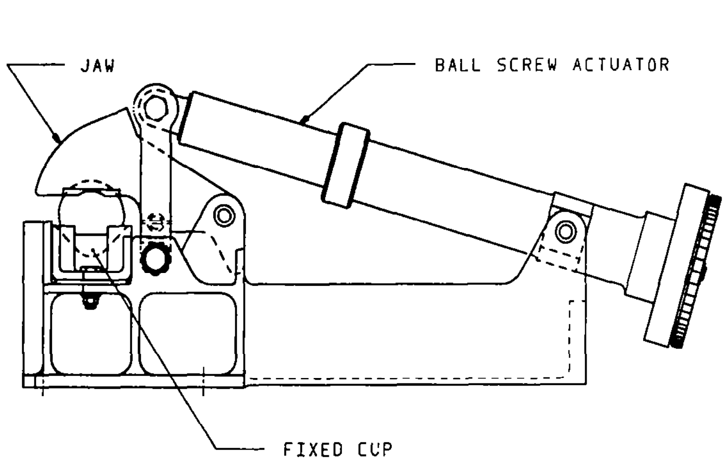
A capture latch capable of restraining, releasing, and recapturing a spacecraft deployable appendage in three translational directions was developed by Lockheed Space Systems Division. Performance of the latch and a computer simulation of latch operation were verified by tests conducted on a development unit. Bending stiffness of the joint-connecting pins was found to have significant effect on overall latch stiffness; however, larger pins necessitated by the static opening load of 44,500 N (10,000 lb) will alleviate this condition. The latch is currently under consideration for use on several LMSC payloads.

ACKNOWLEDGMENTS

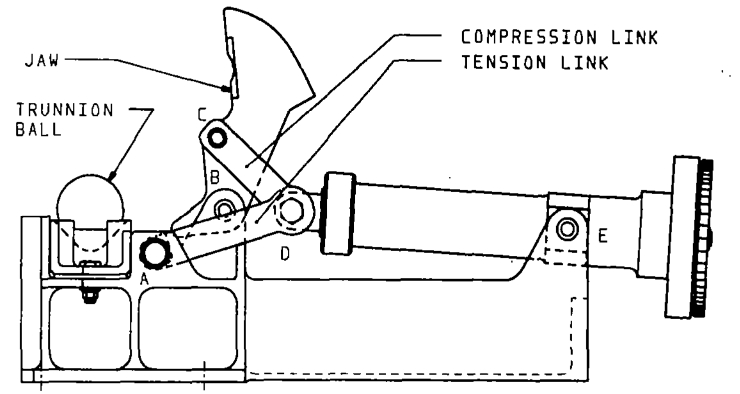
We wish to acknowledge the efforts of Jim Larson for his work on the initial design of the latch, Joe Wilson for his contribution to the stiffness portion of the analytical model, and Ken Buckley for his supervision in the development of the latch design.

REFERENCES

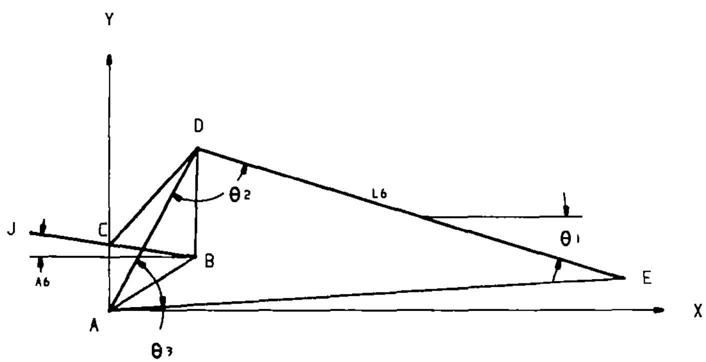
1. LMSC Test Report No. TA B152, "Test Report for the Capture Latch Development Test," September 16, 1981.



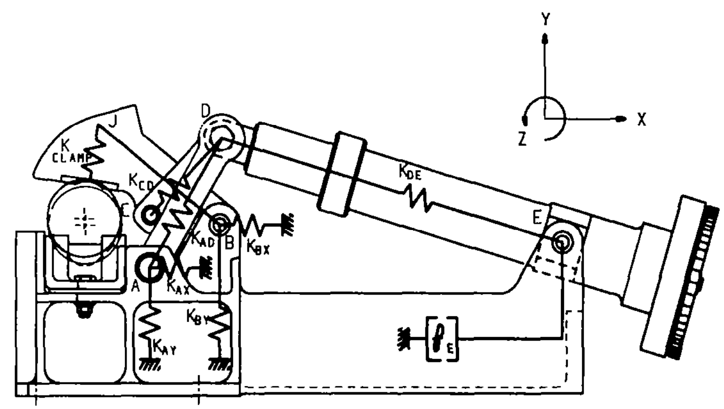
CLAMP CLOSED
FIGURE 1



CLAMP OPEN
FIGURE 2



LATCH KINEMATICS
FIGURE 3



LATCH STIFFNESS MODEL
FIGURE 4

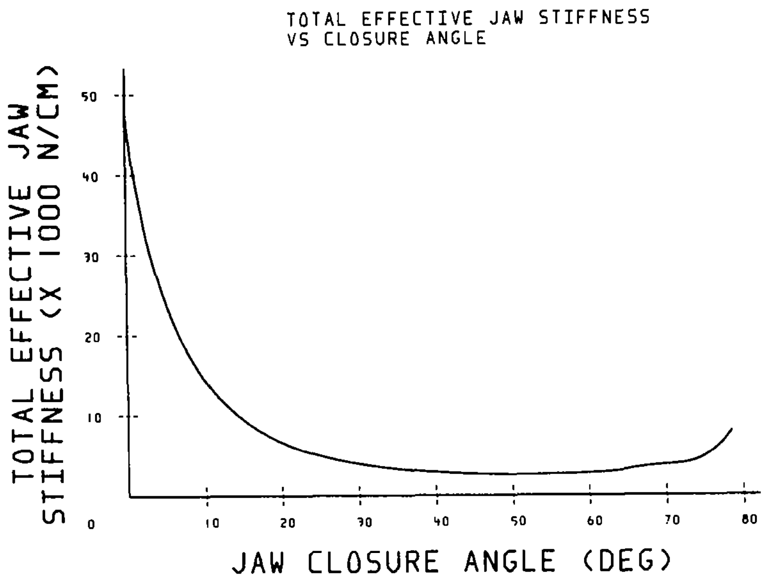


FIGURE 5

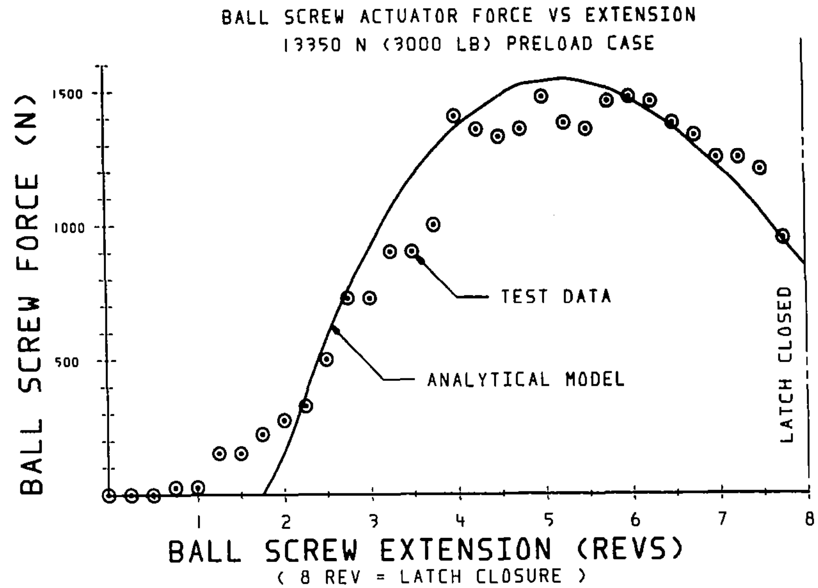
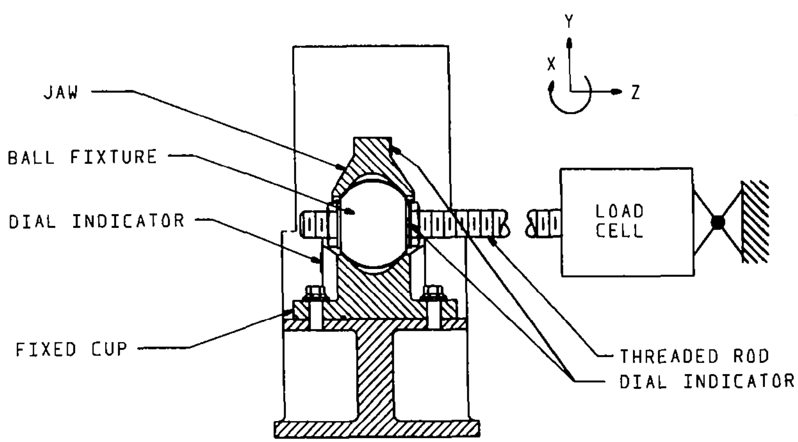
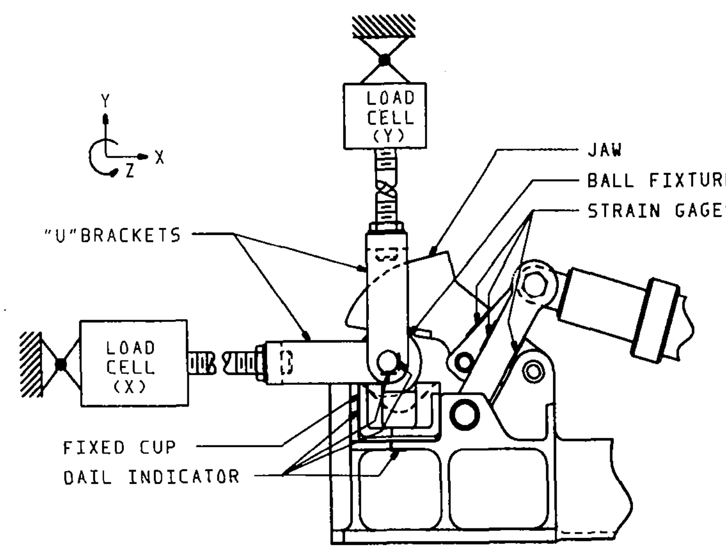


FIGURE 6



Z PULL-IN TEST SET-UP

FIGURE 7



X AND Y PULL-IN TEST SET-UP

FIGURE 8

David V. Adams
Lockheed Missiles and Space Company
P.O. Box 504
Sunnyvale, California 94086

Mr. Adams is an engineering analyst for the Mechanisms and Separation Analysis Group at Lockheed. He received his B.S. degree in General Engineering from the University of Illinois in 1980.

Co-author of this paper is Mr. Brendan Alchorn who is also with Lockheed Missiles and Space Company at Sunnyvale.