Design and Performance of the Telescopic Tubular Mast

Mehran Mobrem* and Chris Spier*

Abstract

The Telescopic Tubular Mast (TTM) has been under development at Astro Aerospace – Northrop Grumman Aerospace Systems for a number of years and has found several applications including deployment of the Large Sunshield on the James Webb Space Telescope. The TTM is composed of a number of large diameter tubes that are deployed and retracted using a Storable Tubular Extendable Mechanism (STEM). In order to study and evaluate the feasibility of utilizing the TTM concept for long boom applications a special design case (34.4-m long) was selected in 2005, which later was built and tested. This paper describes the design and analysis of this design case as well as testing performed. Special attention is given to the deployed stiffness and frequencies, which is a key requirement for space applications. Also, stabilization features required for the deployment are discussed and finally a feedback control system to drive the STEM deployer during the deployment is selected and the difficulties of controlling the system are discussed.

Introduction

The TTM is composed of a number of large-diameter thin-wall composite or metallic tubes as required to achieve a given deployed length, stowed envelope and structural characteristic. When deployed, the tube sections are latched together by multiple tapered pins to achieve a stable extended structure. These pins are withdrawn automatically to enable retraction. The Telescopic tubes are deployed and retracted by a version of Astro Aerospace's STEM deployer. The STEM deployer has a long history of being used in space from the earliest small satellites in the 1960s as antennas and gravity gradient booms to the current GPS series of spacecraft. For the TTM application, the STEM has been evolved into a higher force actuator capable of more than 445 N (100 lb) deployment force.

The TTM has to be deployed within a given time and it is desirable to have a constant velocity during the deployment or retraction. However, due to friction, latch up loads, and external tip loads, the velocity fluctuates. Also the friction and latch up forces fluctuate during the deployment/retraction depending on the individual segments. The STEM deployer inherently has a large dead band due to the winding/unwinding of its STEM element around the spool, therefore, each time the motor changes its direction it can not affect the deployment/retraction for a short period. Hence, the feedback control system should minimize the motor reversal during a given operation, e.g. deployment. Finally, it is desirable to limit the average velocity.

TTM Top Level Information and Background

The TTM test hardware is comprised of 17 telescoping tubes, excluding the fixed base tube. The tubes are optimized for stiffness, strength, and mass and are made of high modulus graphite composite with wall thicknesses ranging from 1.02 mm (0.040 in) to 0.38 mm (0.015 in). The tube closest to the fixed base is 31.8 cm (12.5 in) diameter, and tubes decrease in diameter by 1.3 cm (0.5 in) to the last tube at the tip which is 11.4 cm (4.5 in) diameter.

The tubes are nested inside the fixed base tube and the nested group of tubes is deployed using a STEM deployer that is attached to the fixed base tube. The STEM deployer pushes the group of nested tubes

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* Astro Aerospace – Northrop Grumman Aerospace Systems, Carpinteria, CA

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from inside the fixed base tube. Latching features built into each tube allow the tubes to latch together as the system is deploying and creating a stable structure once fully deployed. The latching features are explained below.

Stowed Design Features

The base tube is designed to support the tubes and STEM deployer for a typical spacecraft application. There are two sets of attachment points one at the base and one in the side close to the tip. Also there are graphite load rings at the tube tips to provide a stiff structure for supporting the payload. The stowed configuration of the TTM is shown in Figure 1.

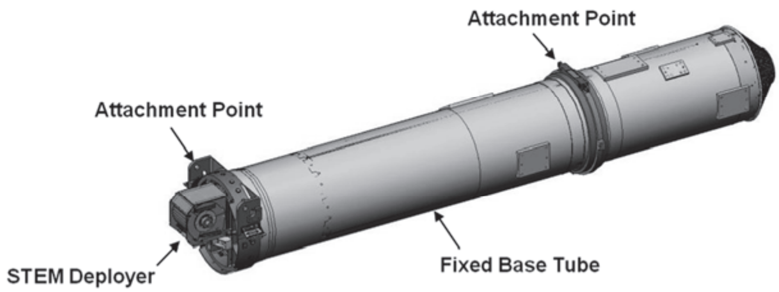


Figure 1. Stowed TTM

Deployment Design Features

There are three major components in the design to stabilize the TTM during its deployment. Each tube has three rails along the tube axis that are spaced 120 degrees apart to stabilize the rotation about the boom axis during the deployment. These rails also have secondary effects to increase the deployed stiffness. The second major component is a secondary stiffening ring at the base about one diameter apart from the latch ring to stabilize the rotations about the two lateral axes. Finally, there is one centralizer in each tube to support the STEM element and stabilize/improve its buckling capability. These main features of each tube that help stabilize the TTM during deployment are highlighted in Figure 2.

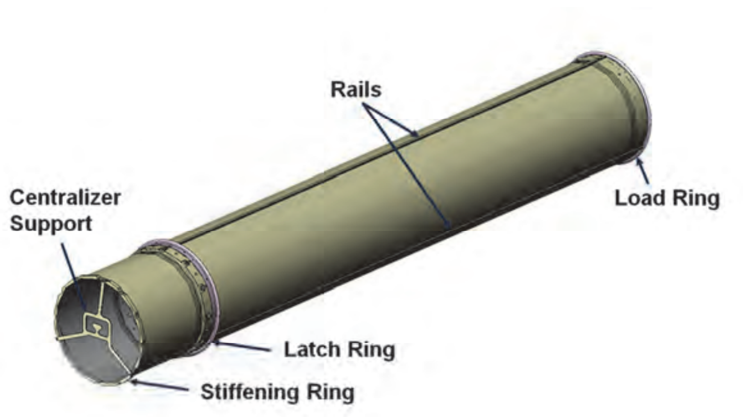


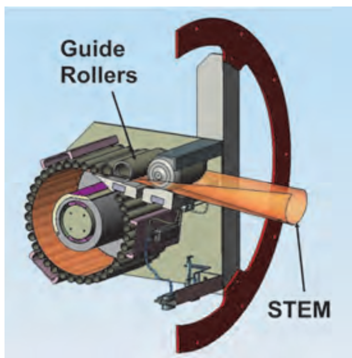
Figure 2. Tube Features

Deployed Design Features

The deployed TTM has nonlinear stiffness mainly due to local flexibility of the latching feature. The stiffness increases as the applied loads to the latch-pin increases. Several design features have been implemented to increase the deployed stiffness. A load ring has been added to each tube tip where the latch-pins are engaged to stabilizes the local deformation of the thin walled tubes. As discussed earlier these load rings are part of launch restraint to support the payload during the launch. Also, the three deployment rails along each tube add to the overall stiffness. Note that by pre-loading the latch-pins the stiffness would be increased and the non-linearity could be reduced. For many applications, there may be a type of tension load which could pre-load the TTM; however, if needed the STEM deployer could be utilized to do this function.

STEM Deployer

The drive system for the TTM is a STEM deployer powered by a brushless dc motor which pushes against the tip plate. The STEM consists of a "C" section of thin formed metal that is flattened so it can be rolled onto a spool for launch, as shown in Figure 3. Deployable booms in the STEM family are simple and extremely lightweight; they have been successfully deployed over 300 times in space without any known failures.



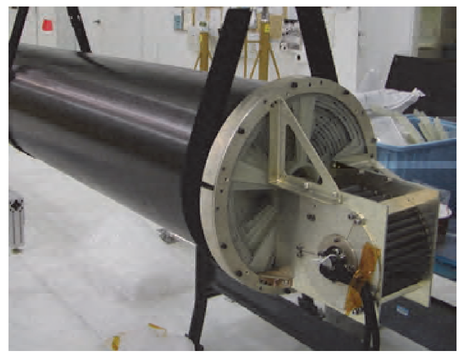


Figure 3. STEM device used for TTM deployment/retraction

The payload and package of stowed tube segments are pushed from the inside of the fixed external base tube by the STEM. When the package reaches the end of the fixed segment, the outer tube in the package latches to it. This tip deployment process repeats sequentially until all tubes are latched into place. The same sequence is reversed to retract. The innermost of the undeployed tubes is fixed to the tip of the STEM in order to stabilize the moving package of tubes.

To minimize the number of tubes, they are all the same length and are stowed coincident with each other. The latches fit in the annular gap between adjacent tubes in a stiffening ring at the lower end of each tube. The adjacent larger tube in turn necks down to a thin stiffening ring at the upper end. The stiffening ring helps to center and align the adjacent smaller tube and to lessen local deformations between the latched segments in bending.

Tube Latching

Small tapered pins are distributed circumferentially in the stiffening ring at the lower end of each tube. The pins are loaded radially outward by short springs to engage with tapered holes at the upper end of each larger adjacent tube, as shown in Figure 4. When stowed, the springs and pins are compressed by the interior surface of the adjacent larger tube. During

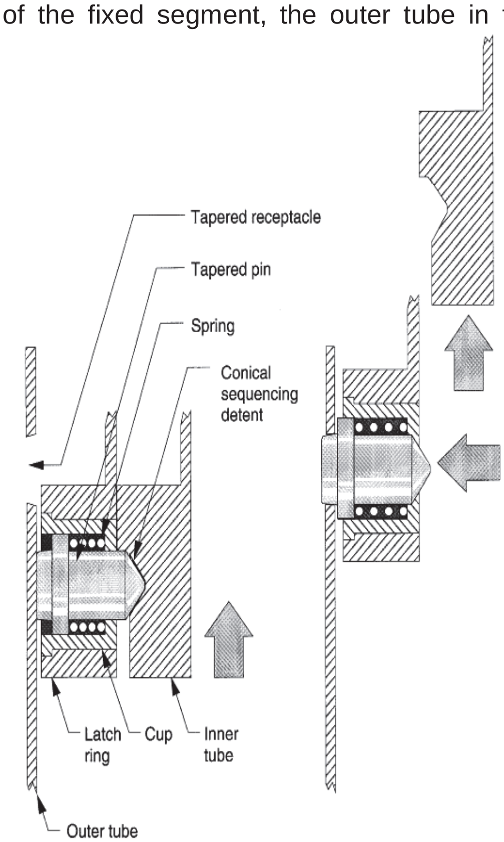


Figure 4. Tapered pins used for latching

deployment, the tips of the pins slide on the surface until they pop into the tapered seats to latch.

Sequencing

Since the tubes are stowed coincident to each other, each ring of compressed latch pins can engage the adjacent smaller ring with simple detents as shown in Figure 4. All the nested tubes are thus locked together so that they can be pushed as a package during deployment. When the latch ring in the outermost tube of the package locks it into deployed structure, the detents retaining that tube to the moving package of tubes are released. The now smaller package of moving tubes continues without interruption.

The male component of the detent on the interior end of the latch pin is conically shaped to make the latching function fail-safe. If one or more springs fail, the affected pin is forced out of the way by the female side of the detent, which acts as a ramp, as shown in Figure 4. Without the spring to preload the pin in the tapered receptacle, that pin cannot contribute to the deployed stiffness of the boom, however, deployment will not be impeded.

Retraction

To retract a given tube, its latch pins are pulled from engagement with the next larger tube by ramps in the next smaller tube. The ramps are hollowed out of the latch rings to engage conical rims at the male detent end of the latch pins, as shown in Figure 5.

Latch pins are alternated with retraction ramps and detents in increments around the circumference of each ring. Each successive tube in the assembly is indexed by one such increment relative to its neighbors so that everything meshes properly. The length of the trough is controlled so that the detents will engage before the deployed tube is unlatched, as shown in the second inset of Figure 5.

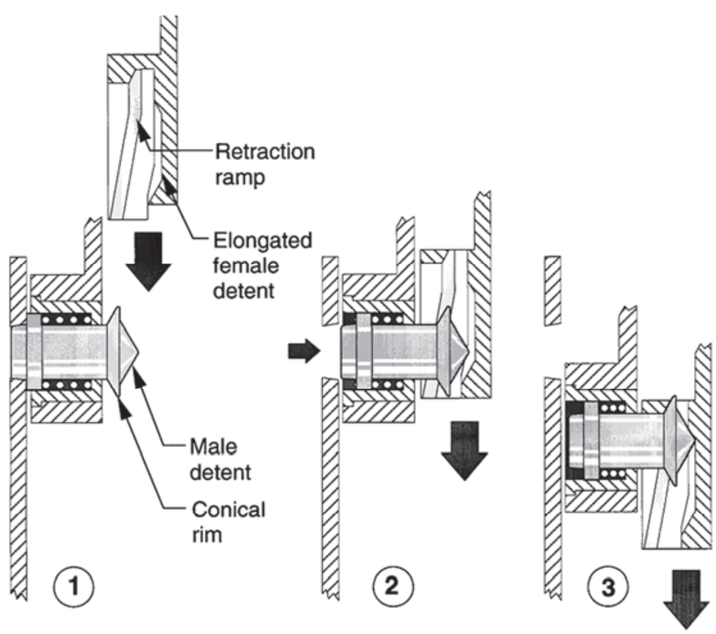


Figure 5. Boom retraction sequence

TTM Special Design Case

In order to study and evaluate the feasibility of utilizing the TTM concept for long boom applications, a special design case was selected in 2005 which was later built and tested. The key parameters and top level requirements for this design are:

- Deployed length of 34.3 m (1350 in) from bottom of base to tip
- Stowed length of 2.16 m (85 in)
- Mass is approximately 58 kg (128 lb)
- Deploy time within 14 minutes
- Stowed frequency should be greater than 35 Hz
- Deployed frequency should be greater than 0.1 Hz
- Mast should be able to retract by reversing motor
- Tip displacement during the deployment/retraction should stay within ±17.8 cm (7 in)

<u>Tube Properties</u>

There are 17 telescoping tubes, excluding the fixed base tube. Table 1 lists properties for the minimum and maximum diameter telescopic tubes, along with Tube 9 which is the tube in the middle between the base and the tip. Tube 1 is the tube closest to the base tube and is 31.8 cm (12.5 in) diameter, while Tube 17 is the last tube at the tip and is 11.4 cm (4.5 in) diameter. Listed in the table are the different stiffness properties for each of the tubes. EI is the bending stiffness of the tube, GJ is the torsional stiffness of the tube, and EA is the axial stiffness of the tube.

Table 1. Tube Properties

Tube	Tube Diameter (cm)	Thickness (mm)	EI (x10^3 N·m^2)	GJ (x10^3 N·m^2)	EA (x10^6 N)	Mass/Length (kg/m)
1 (Max)	31.8	0.51	868.9	346.9	69.0	1.56
9	21.6	0.38	140.3	105.8	24.1	1.47
17 (Min)	11.4	1.02	81.1	41.9	49.6	1.93

Stowed Frequency Analysis

A stowed finite element model of the TTM was created to compute the natural frequencies. The fundamental computed natural frequency is 37.88 Hz which exceeds the 35 Hz requirement. The first ten stowed natural frequencies are listed in Table 2 and the first three mode shapes are shown in Figure 6. Note that these frequency results have not been validated by a test.

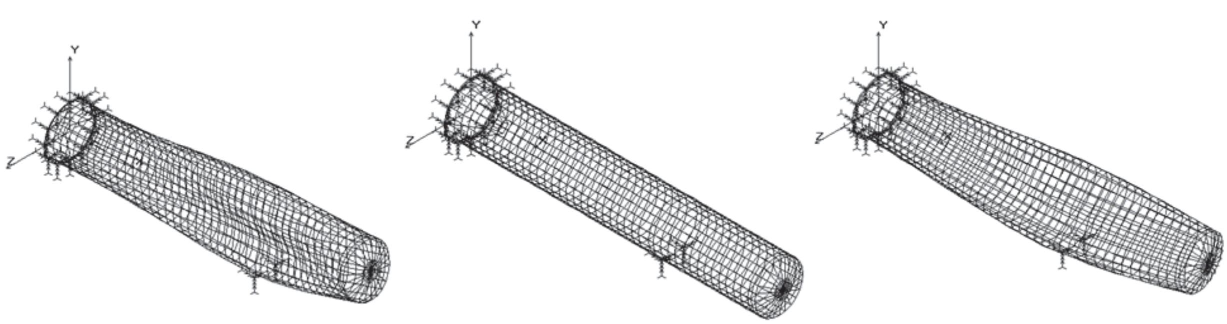


Figure 6. First three modes in the stowed configuration

Table 2. Stowed natural frequencies

Mode	Frequency (Hz)	Mode	Frequency (Hz)
1	37.88	6	62.92
2	48.69	7	69.23
3	51.83	8	76.58
4	52.68	9	82.42
5	57.46	10	85.28

Deployed Stiffness Analysis and Test

A deployed nonlinear FEM was created to determine the deployed stiffness, natural frequencies, and its load capability. The nonlinear FEM was used to determine the deployed stiffness under different load conditions and was validated by test. Measurements were performed on the deployed TTM to characterize its stiffness under different loading conditions. The test setup consisted of the deployed mast offloaded with a total of 17 helium balloons. Each balloon was attached to the tip end of their respective tubes as shown in Figure 7. The lift of the individual balloons was adjusted to the mass of the supported tube sections. The tip load was applied with a motorized slide mechanism that had a load cell and position transducer attached.

Figure 7. Deployed stiffness test configuration

<u>Baseline Tip Load – No Preload</u>
A baseline tip load test was performed with no preload. For this test, the mast was loaded in shear only at the tip to the levels shown in the plot in Figure 8. The blue curve in the figure shows the load-deflection characteristics of the mast with zero preload. The red line represents the predicted performance of the mast. The TTM measured stiffness correlates well with predictions, however it is lower at low amplitude and is higher at high amplitude compared to the predictions.

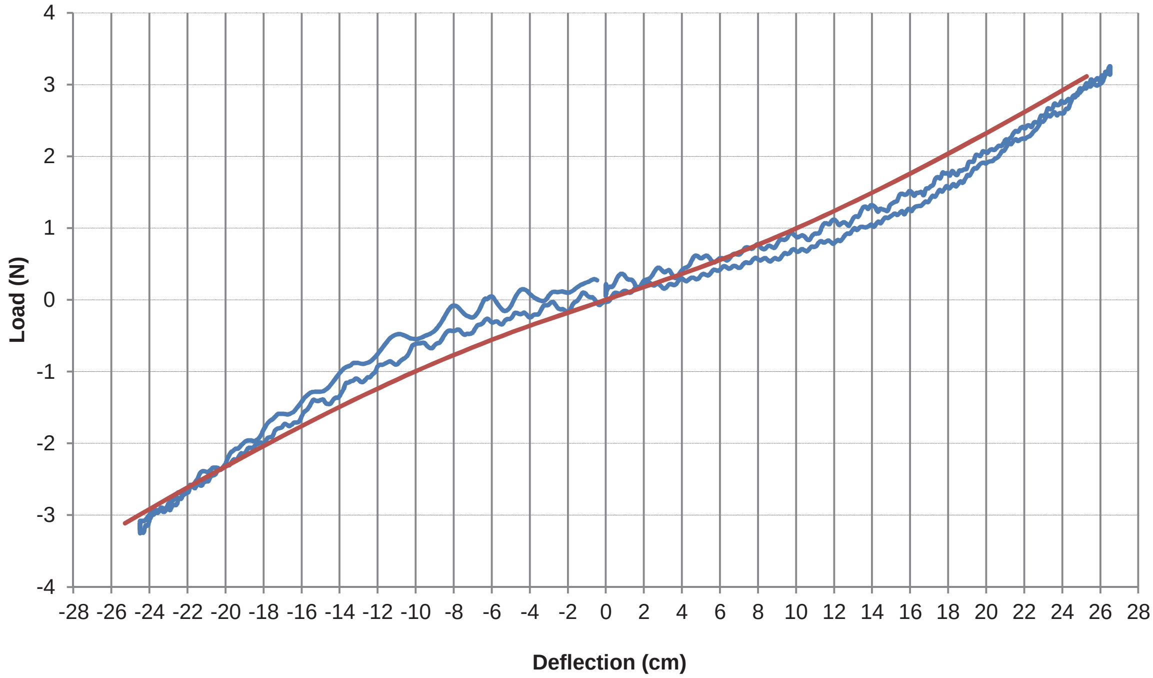


Figure 8. Plot of tip displacement versus load for the no preload load case

Deployed Preload Condition

The deflection of the fully deployed mast due to the application of a tip preload was also characterized. This type of loading generally relates to typical flexible solar panels or sun-shields under tension. The test configuration is shown in Figure 9. A typical 55.6-N (12.5-lb) load was applied in the direction shown at the tip and the resulting deflection was measured at the tip.

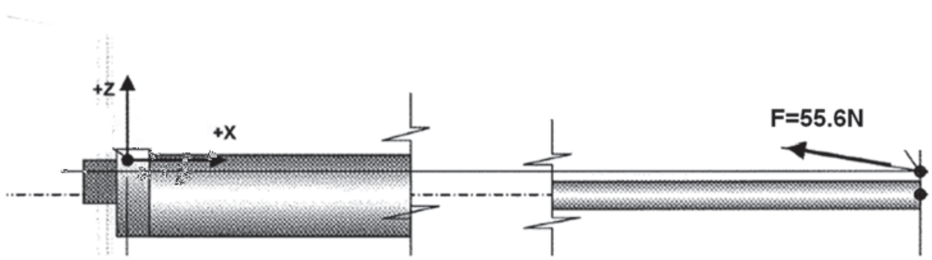


Figure 9. Deployed preload condition

The results of the preloaded test are shown in Figure 10. For this test, the mast was loaded with a 55.6-N (12.5-lb) tethered load, which creates an axial, shear, and moment load, and then an additional shear load was added at the tip to the levels shown in the plot. In this plot, zero deflection corresponds to the deflected shape under the tethered load, which was 13.8 cm (5.45 in).

The red line represents the predicted performance of the mast. This line shows the predicted tip deflection of 12.7 cm (5.0 in) under the tethered load versus the 13.8 cm (5.45 in) observed in the test. It was observed from this test that lower than predicted stiffness occurred at lower loads while higher than predicted stiffness occurred at higher loads.

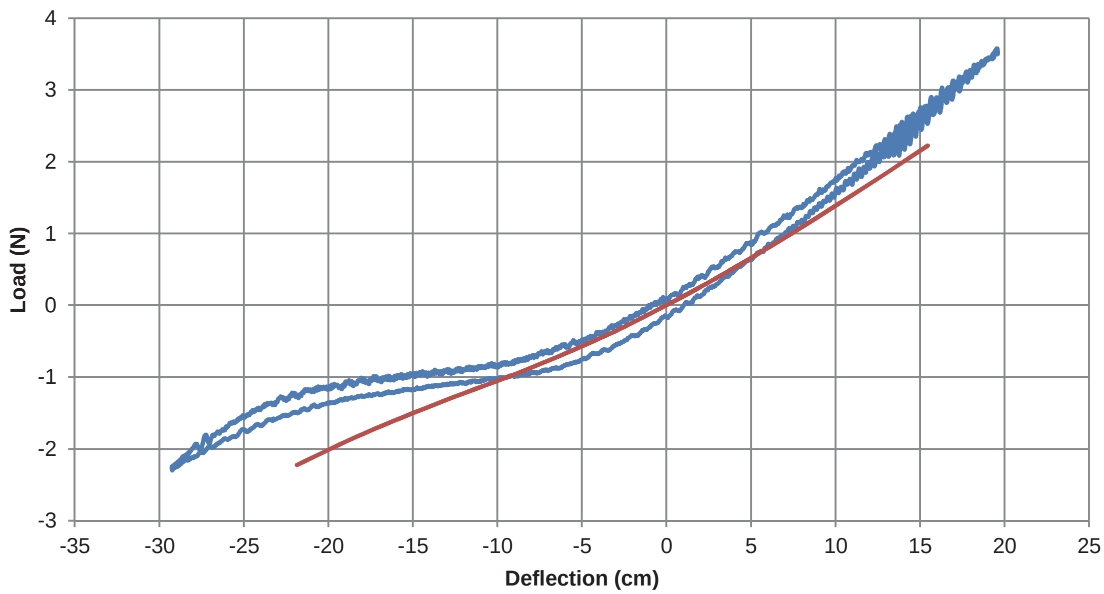


Figure 10. Plot of tip displacement versus load for the preload condition

Deployed Natural Frequency Analysis

A linear FEM was created from the nonlinear model with applied tethered loads to compute the natural frequencies. The fundamental computed natural frequency is 0.1 Hz which meets the 0.1 Hz requirement. Note the mode shape is orthogonal to the plane of load application where the boom stiffness is at minimum stiffness region. The first eight deployed natural frequencies are listed in Table 3 and the mode shapes are shown in Figure 11. Note that these frequency results have not been validated by a modal test. A detailed view of the deployed FEM is shown in Figure 12.

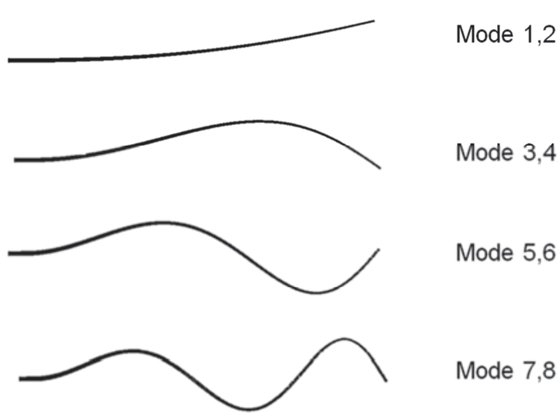


Figure 11. First eight deployed modes

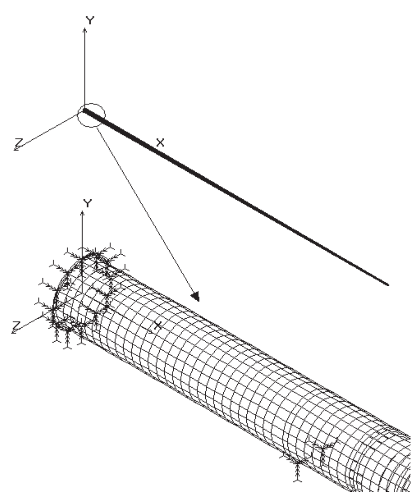


Figure 12. Deployed FEM

Table 3. Deployed natural frequencies

Mode	Frequency (Hz)	Mode	Frequency (Hz)
1	0.10	5	1.56
2	0.15	6	2.09
3	0.56	7	3.09
4	0.77	8	4.09

Deployment Characteristics and Control

The goal was that the TTM should be deployed within 15 minutes. It is desirable to have a constant velocity during the deployment or retraction; however, due to friction, external tip loads, and latch up loads the velocity fluctuates. Also, the friction and latch up forces fluctuate during the deployment/retraction depending on the individual segments. A typical resistive force during a segment latch up is show in Figure 13.

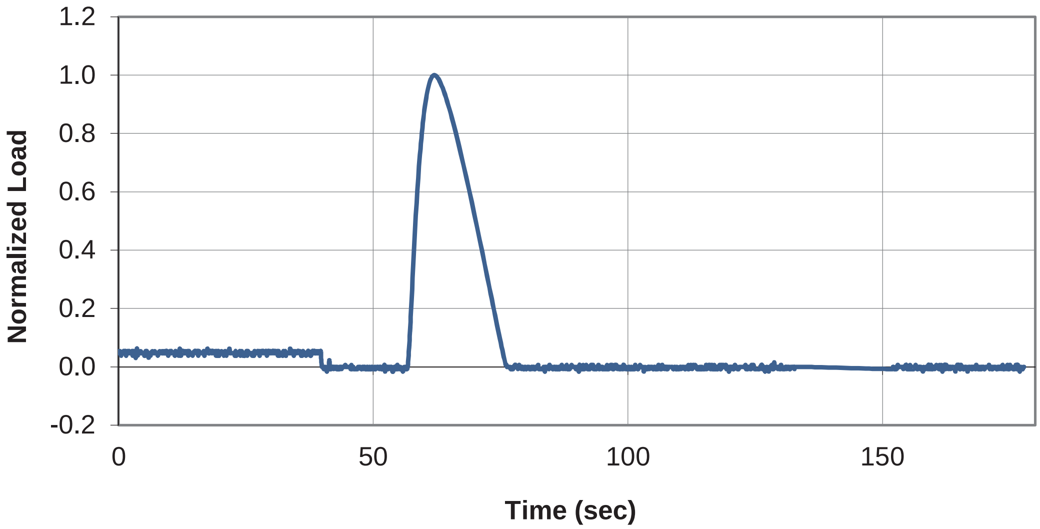


Figure 13. Typical latch loads during deployment

The drive system is a STEM deployer powered by either a brushless dc motor or a stepper motor which pushes against the tip plate. The STEM deployer inherently has large dead band due to the winding/unwinding of its STEM element around the spool, therefore each time the motor changes its direction it can not affect the deployment/retraction for while. Hence the feedback control system (required for brushless dc motor application) should minimize the motor reversal during the deployment operation. Finally, it is desirable to limit the average velocity.

Deployment Drive Feedback Control System

To achieve the deployment criteria, a feedback control system containing the following is considered (see Figure 14):

- Position loop
- Velocity loop inside the position loop
- Feed forward time dependent velocity profile - trapezoidal
- Time dependent reference position – integral of velocity profile
- Both velocity and position loops should be:
  - Stable for minimum inertia due to the backlash (motor inertia only) as well as maximum inertia
  - Open loop frequency bands should be less than 70% of the structural natural frequency
  - Nonlinear gains used in both position and velocity loops
  - Gain and phase margins should be greater than 10 dB and 40 degrees respectively
- Current limit is required to prevent high deployment force

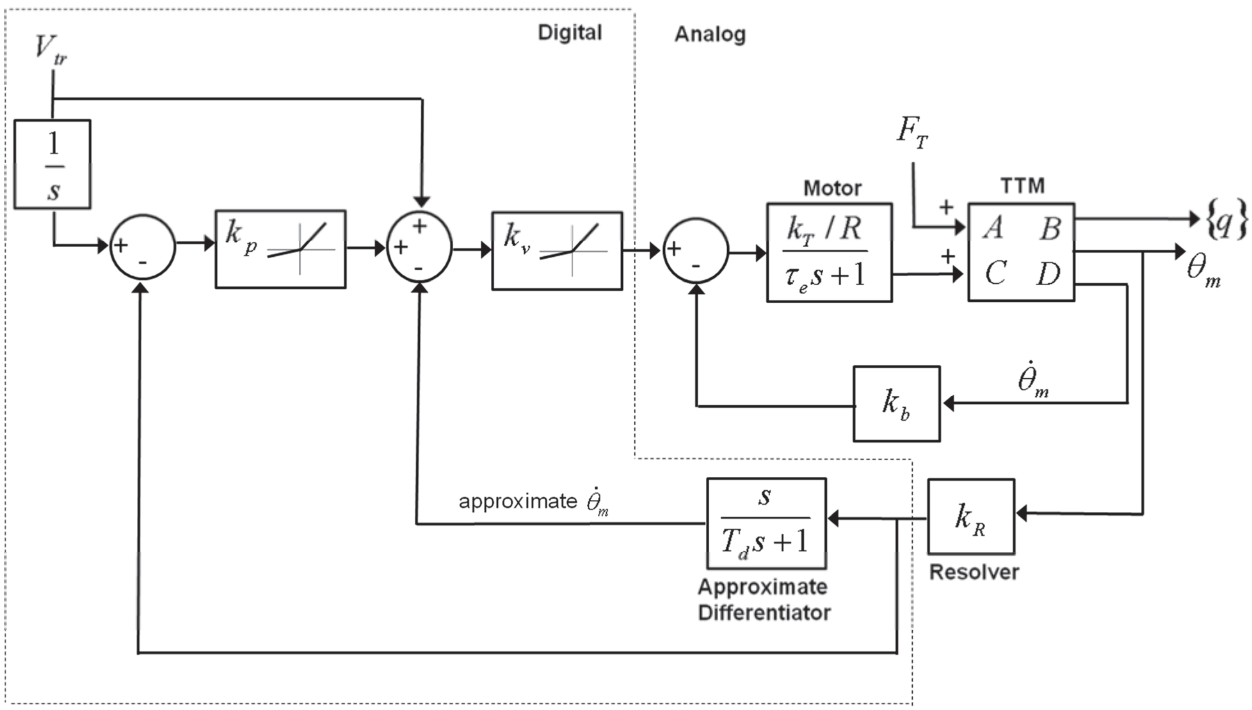


Figure 14. Control system block diagram

In Figure 14, V_{tr} is the translational drive reference velocity, T_d is the approximate differentiator time constant, L is the motor inductance, R is the motor resistance, τ_e is equal to L/R, k_T is the motor torque constant, and k_b is the back emf constant.

The gains for the control system are: k_p is the position loop gain, k_v is the velocity loop gain, and k_R is the resolver gain. It is desirable to have nonlinear gains in both the position and velocity loops in order to prevent the motor reversal caused by back winding in the STEM. A possible nonlinear control gain that can be implemented is a piece-wise linear gain that is high gain for lag and low gain for lead.

The TTM is a time variant system and, because of this, there are non-conservative forces during the deployment. These forces become significant as the deployment velocity and acceleration increases

[2,4]. A similar feedback control system was designed and tested for the Space Station Freedom Mobile Transporter [2,3]. For the Mobile Transporter, an avionics control breadboard simulator was designed to simulate and verify the operation of the Mobile Transporter control system.

The following dynamic models are used to derive the TTM system state space matrix [A,B,C,D]. Note that during deployment the length of the TTM is changing causing a change in natural frequencies.

Primary Dynamic Model

The primary dynamic model used for the design of the control system is shown in Figure 15. This model represents the deployment and axial vibration of the TTM. The moving mass M is the mass of the deploying tubes plus the mass of the payload and $f_{resistive}$ are loads due to friction, tube latching, and any external loads such as sun-shields tension and/ or space craft attitude control. For the STEM deployer, k_{STEM} is the axial stiffness of the partially deployed STEM and varies by length, θ_m and θ_s are the rotation of the motor and spooler, respectively, J_m and J_s are the inertias of the motor and spooler, respectively, and n is the gear ratio.

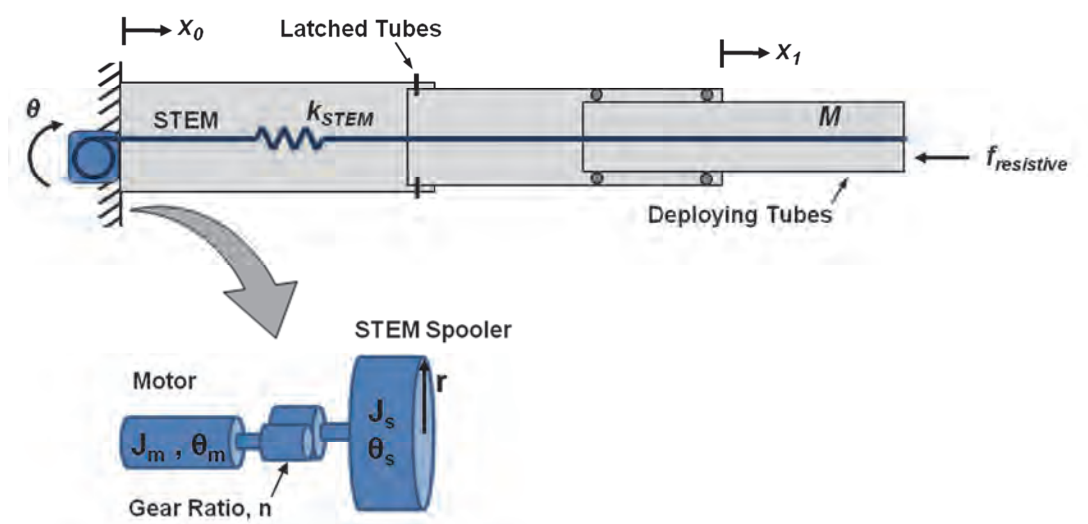


Figure 15. Dynamic model for control system

The equations of motion for the STEM deployer and the system of deploying tubes shown in Figure 15 are

$$M\ddot{x}_1 + k_{STEM}\left(x_1 - x_0\right) = -f_{resistive} \tag{1}$$

The axial natural frequency changes as the STEM deploys and the boom length changes. A plot showing the change in axial natural frequency as the system deploys and the length increases and moving mass of the tubes decreases is shown in Figure 16.

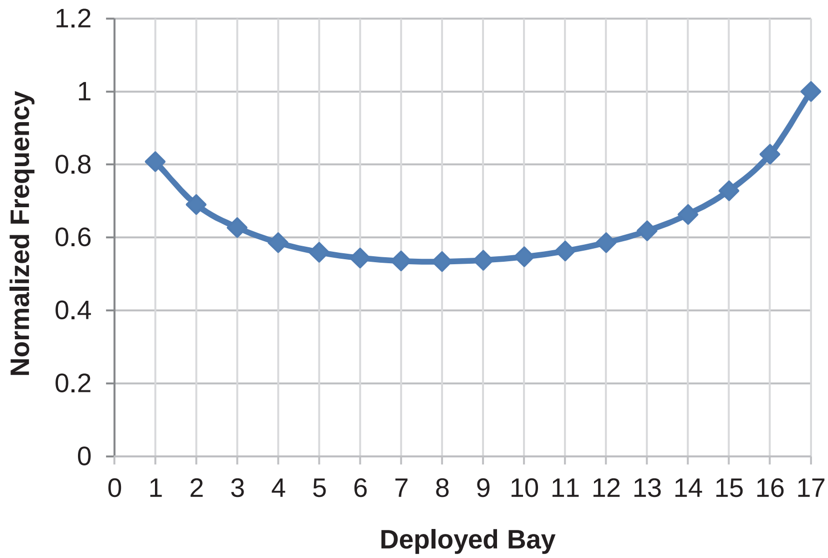


Figure 16. Plot of axial frequency versus deployed tube bays

<u>Secondary Dynamic Model</u>

The secondary dynamic model used for the design of the control system represents the lateral vibrations of the TTM during the deployment. This can be represented by an axially moving cantilever beam with a tip mass as shown in Figure 17.

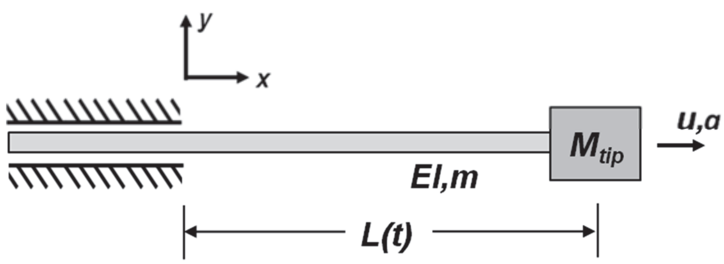


Figure 17. Cantilever beam with tip mass and time dependent length

The equation of motion for the system shown in Figure 17 for a cantilever beam whose length changes with time is given by [2]

$$\left[\frac{33}{35}mL(t)+4M_{tip}\right]\ddot{\eta}+\frac{33}{35}mu\dot{\eta}+\left[\frac{12EI}{L(t)^3}-\left(1.028m+4.8\frac{M_{tip}}{L(t)}\right)a-0.6857\frac{m}{L(t)}u^2\right]\eta=0 \qquad (2)$$

where u is the velocity of the moving beam, a is the acceleration of the moving beam, EI is the bending stiffness of the beam, m is the mass of the beam, $L(t)$ is the instantaneous length of the beam, M_{tip} is the tip mass and η is the normal coordinate representing the lateral displacement y. Note that equation (2) has non-conservative forces which could become important depending on the deployment velocity and acceleration.

If the velocity and accelerations are small, equation (2) reduces to

$$\left[\frac{33}{35}mL(t) + 4M_{tip}\right]\ddot{\eta} + \frac{12EI}{L(t)^3}\eta = 0 \tag{3}$$

Control System Simulation
A simulation of the control system was performed using Simulink. A resistive force representative of the one shown in Figure 13 is used to simulate latching loads. Non-linear control gains are used for the position and velocity loop gains and a deadband of ±7.6 cm (3 in) is used.

Simulation results for the TTM deployment are shown in Figures 18 and 19. Figure 18 is a plot of velocity versus time and includes deadband in the system. Figure 19 is also a plot of velocity versus time, but does not include deadband in the system. In both plots, time is in seconds and velocity is in meters per second.

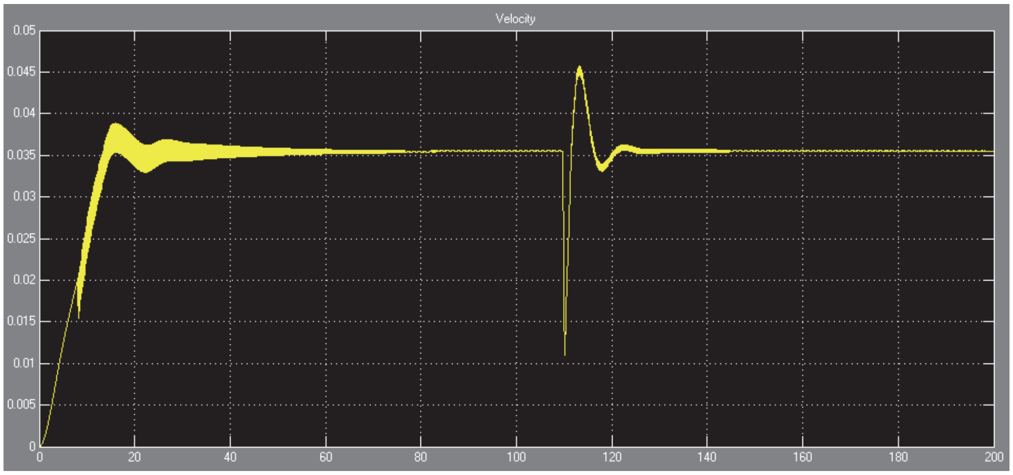


Figure 18. Velocity versus time plot with deadband

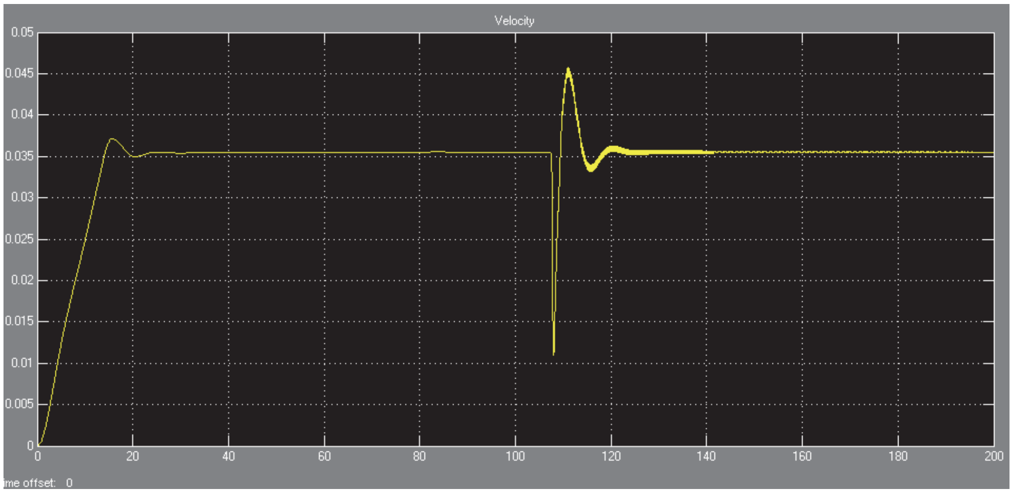


Figure 19. Velocity versus time plot with no deadband

Summary

A deployable and retractable telescoping boom that contains large-diameter, thin-walled composite tubes has been developed that is capable of high deployed stiffness. Tapered pins are used to control the deployment and retraction between the tubular sections and create a stable extended structure when deployed. The latch design and STEM work together to eliminate the need for segments to overlap when deployed giving a lightweight design.

The key parameters and requirements for the TTM study are given, and it is shown by analysis and test that the requirements are met. Frequency analysis on the deployed and stowed configurations was performed using FEA software to show that the frequency requirements are met. Testing was also performed on the deployed TTM to characterize the stiffness under different loading conditions. In all the load conditions, lower than predicted stiffness occurred at lower bending loads and higher than predicted stiffness occurred at higher loads.

Control of the deployment drive for the TTM presents several challenges due to the design of the STEM actuator and the latching loads during deployment. To analyze the system, a dynamic model was developed that includes the deploying tubes and the STEM motor and spooler. The equations of motion for the system were derived and a feedback control system was designed. The work done to date and the progress made on the TTM system are presented here.

References

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